Heat Recovery in Food Refrigeration Systems

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Abstract

Heat recovery from commercial food refrigeration systems offers significant potential to improve energy efficiency, reduce operational costs, and support decarbonization in the food sector. Refrigeration systems account for a large share of electricity consumption in supermarkets, cold storage facilities, and food processing plants, while simultaneously rejecting considerable amounts of heat to the environment. Recovering and utilizing this thermal energy can substantially enhance overall system performance.

This chapter reviews the thermodynamic principles of refrigeration and the major sources of recoverable heat within vapor-compression systems. It presents key heat recovery technologies, including desuperheaters, condenser heat reclaim, heat pump integration, and thermal energy storage. Advanced approaches such as Organic Rankine Cycle systems, thermoelectric generators, and ejector-based configurations are also discussed. In addition, the chapter examines heat recovery potential in advanced refrigeration architectures such as transcritical CO_2 booster systems. Finally, the role of intelligent control strategies and digitalization in optimizing heat recovery performance is outlined, along with key research challenges and future directions for sustainable refrigeration systems.

Key Points / Objectives

- Commercial food refrigeration systems reject large amounts of recoverable thermal energy
- Heat recovery can significantly improve system efficiency and reduce operational costs
- Vapor-compression cycles provide the primary source of recoverable heat through condenser rejection
- Direct and indirect heat recovery methods enable utilization for space heating, hot water, and process applications
- Advanced systems (e.g., CO_2 booster and ejector-based cycles) enhance heat recovery potential
- Integration with thermal storage and heat pumps improves flexibility and energy matching
- Digital control strategies enable optimized, demand-responsive heat recovery operation
- Technical, economic, and knowledge barriers remain key challenges for large-scale adoption

1. Introduction

Heat recovery in commercial food refrigeration systems has emerged as a critical strategy for enhancing energy efficiency, reducing operational costs, and mitigating the environmental footprint of the food sector. Across the entire supply chain, from primary production and processing to transportation, storage, and retail, refrigeration systems are indispensable for maintaining product quality, safety, and shelf life. However, this essential function comes with substantial energy demand. Refrigeration systems are among the most energy-intensive subsystems in food-related facilities. In supermarkets, they typically account for 40–60% of total electricity consumption (Evans *et al.*, 2014), with values reaching approximately 60% in the United States and ranging from about 29% in large hypermarkets to over 50% in smaller retail formats in the United Kingdom (Spyrou *et al.*, 2014). These variations reflect differences in store layout, refrigeration display case density, operating temperatures, and product mix. In cold storage warehouses, refrigeration often represents 60–70% of total electricity consumption, while in food processing plants the share generally ranges between 25–35% (Evans *et al.*, 2014).

From a thermodynamic perspective, the high energy consumption of refrigeration systems is accompanied by a correspondingly large rejection of heat at the condenser. In ideal vapor-compression cycles, the total heat rejected equals the sum of the cooling load extracted at the evaporator and the mechanical work input to the compressor. Consequently, a substantial fraction of the electrical energy supplied to refrigeration systems ultimately appears as low- to medium-grade thermal energy discharged to the environment. Traditionally, this heat has been treated as waste. Heat recovery seeks to transform this paradigm by capturing and repurposing rejected heat for productive applications, such as space heating, domestic hot water production, process water preheating, and in some cases integration with district heating networks. By converting a by-product of cooling into a usable energy stream, facilities can significantly improve overall system efficiency. In energy terms, this increases the primary energy utilization rate of the installation. In exergy terms, it reduces the destruction associated with heat rejection to ambient conditions and enhances overall system performance.

The economic and environmental drivers for heat recovery are becoming increasingly compelling. Electricity prices have risen steadily in many regions, with annual increases on the order of 3–5% reported in recent years (Eurostat, 2024; Comstock, 2025). At the same time, decarbonization policies, carbon pricing mechanisms, and stricter building energy regulations are placing additional pressure on commercial and industrial operators to reduce emissions. Because refrigeration systems operate continuously and exhibit relatively stable thermal loads, they offer a highly reliable and predictable source of recoverable heat. As a result, heat recovery is no longer viewed merely as a supplementary efficiency measure but increasingly as a strategic design element in modern commercial refrigeration systems.

This chapter examines heat recovery from commercial food refrigeration systems from both technical and practical perspectives. It first presents the thermodynamic foundations governing refrigeration cycles and the associated heat rejection processes, highlighting typical sources and temperature levels of recoverable heat. It then discusses system configurations and integration strategies for commercial applications, including supermarkets, cold storage warehouses, and food processing plants. Key technologies and design considerations are analyzed with respect to efficiency, controllability, and operational reliability. Real-world case studies are presented to illustrate performance outcomes, economic viability, and implementation challenges. Finally, current trends, technological innovations, and future research directions are discussed, with particular emphasis on system integration, low-global-

warming-potential refrigerants, and sector-coupling approaches that support broader energy transition goals.

2. Thermodynamics of Heat Recovery in Refrigeration Systems

2.1 Basic Concept of Refrigeration and Energy Balance

The fundamental objective of refrigeration is to extract heat from a substance or enclosed space and maintain its temperature below that of the surrounding environment. In food sector applications, this may involve cold storage rooms, refrigerated display cabinets, processing equipment, or transported goods to preserve product quality and ensure safety. Maintaining a region at a temperature lower than ambient conditions requires the continuous removal of heat entering the system through thermal conduction, air infiltration, product loading, lighting, and internal equipment operation. From a thermodynamic standpoint, refrigeration entails transferring heat from a low-temperature reservoir to a higher-temperature reservoir, typically the ambient environment. According to the second law of thermodynamics, heat does not spontaneously flow from a colder body to a warmer one. Therefore, this process requires an external work input (Figure 1). To accomplish this, refrigeration systems employ an intermediate working fluid, known as the refrigerant, which undergoes cyclic thermodynamic transformations while interacting alternately with the low- and high-temperature reservoirs.

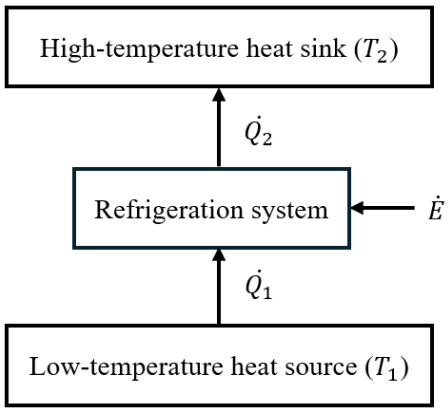


Figure 1. Basic operating principle of a refrigeration system. The system absorbs heat $\dot{Q}_1$ from a low-temperature heat source at temperature T_1 through the input of external power $\dot{E}$. As a result, a total heat flow $\dot{Q}_2$ is rejected to a high-temperature heat sink at temperature T_2, where $\dot{Q}_2 = \dot{Q}_1 + \dot{E}$ during ideal operation.

Based on the sequence of thermodynamic states experienced by the refrigerant, various refrigeration cycles can be implemented to absorb heat from a low-temperature source. Among these, the vapor-compression refrigeration cycle is by far the most widely used in food refrigeration applications, both in residential and commercial contexts. Other established cycles include absorption, adsorption, and Stirling refrigeration systems, which are typically applied in specialized or niche applications.

2.2 Thermodynamic Analysis of the Vapor-Compression Cycle

The vapor-compression refrigeration cycle is the dominant cooling technology in the food sector, applied in systems ranging from small commercial display units to large-scale industrial cold storage facilities. A clear understanding of its thermodynamic operation, in particular, its

heat rejection mechanisms, is essential for identifying and quantifying opportunities for effective heat recovery.

2.2.1 Cycle components and operation

The idealized vapor-compression cycle consists of four principal components (Figure 2):

- Evaporator: A low-pressure heat exchanger in which the refrigerant absorbs heat from the refrigerated space. During this process, the refrigerant undergoes phase change from liquid to vapor. In food applications, evaporator temperatures typically range from −40 °C (frozen storage) to +5 °C (fresh produce and dairy) (Karampour *et al.*, 2016).
- Compressor**:** A mechanical device that raises the pressure and temperature of the refrigerant vapor through work input. Common compressor types include reciprocating, scroll, screw, and centrifugal compressors, selected based on capacity, efficiency, and operational constraints.
- Condenser**:** A high-pressure heat exchanger in which the refrigerant rejects heat to a cooling medium (air or water) and condenses back into liquid form. The condenser represents the primary heat rejection location and therefore the main opportunity for heat recovery.
- Expansion device**:** A throttling or expansion valve that reduces refrigerant pressure prior to re-entering the evaporator. The process is typically modeled as isenthalpic.

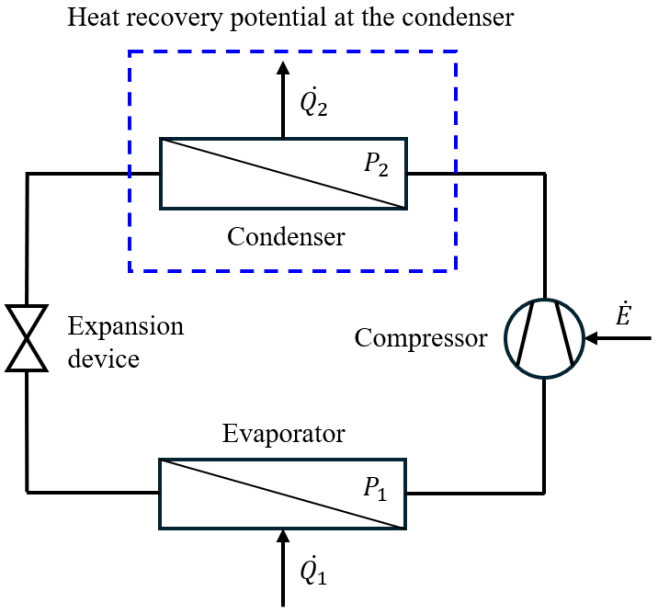


Figure 2. Schematic of a simple vapor-compression refrigeration system. P_2 denotes the high pressure at the condenser, where heat $\dot{Q}_2$ is rejected to the surroundings. P_1 denotes the low pressure at the evaporator, where heat $\dot{Q}_1$ is absorbed from the refrigerated space. Compressor power input $\dot{E}$ drives the thermodynamic cycle. The condenser represents the primary location for potential heat recovery.

2.2.2 Energy performance and COP

The condenser heat rejection constitutes the largest single source of recoverable thermal energy in refrigeration systems. Applying the first law of thermodynamics to the steady-state cycle yields:

$$\dot{Q}_2 = \dot{Q}_1 + \dot{E} \quad (1)$$

where $\dot{Q}_2$ is the heat rejected at the condenser, $\dot{Q}_1$ is the cooling capacity (heat absorbed in the evaporator), and $\dot{E}$ is the compressor power input.

This fundamental relationship shows that the condenser must reject both the extracted cooling load and the mechanical work supplied to the compressor. Consequently, refrigeration systems inherently generate a substantial stream of thermal energy at a temperature level above ambient, an intrinsic opportunity for heat recovery.

The thermodynamic efficiency of refrigeration systems is characterized by the coefficient of performance (COP):

$$COP = \frac{\dot{Q}_1}{\dot{E}} \quad (2)$$

For typical commercial refrigeration systems, COP values range between 2 and 5, depending on operating conditions and system design.

Rearranging Eq. (1) using the definition of COP:

$$\dot{Q}_2 = \dot{Q}_1(1 + \frac{1}{COP}) \quad (3)$$

For a system operating at COP = 3.0, the condenser heat rejection is approximately 1.33 times the cooling capacity. Thus, a 100 kW refrigeration system rejects roughly 133 kW of thermal energy, highlighting the significant magnitude of recoverable heat.

2.2.3 Thermodynamic state analysis and performance parameters

The thermodynamic processes of the simple vapor-compression cycle are commonly illustrated in the pressure–enthalpy (P–h) diagram shown in Figure 3.

The refrigerating capacity, defined as the rate of heat absorption in the evaporator, is expressed as:

$$\dot{Q}_1 = \dot{m}(h_1 - h_4) \quad (4)$$

where $\dot{m}$ is the refrigerant's mass flow rate, h_4 is the refrigerant enthalpy at the evaporator inlet, and h_1 is the enthalpy at the evaporator outlet. The term $(h_1 - h_4)$ therefore, represents the specific enthalpy rise of the refrigerant during evaporation.

Because the majority of refrigeration systems employ volumetric compressors, the volumetric flow rate is a key design parameter in addition to the mass flow rate. The volumetric flow rate at the compressor inlet is given by:

$$\dot{V}_1 = \dot{m}v_1 \quad (5)$$

where v_1 is the specific volume of the refrigerant at the compressor inlet.

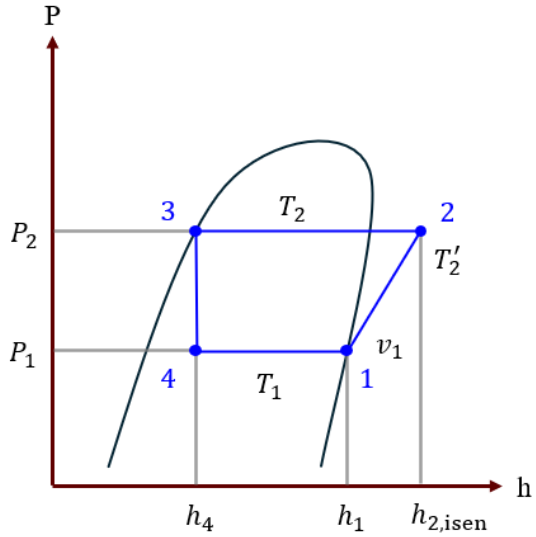


Figure 3. Pressure–enthalpy (P–h) diagram of a simple vapor-compression refrigeration cycle. Process 1–2 represents isentropic compression; 2–3 is isobaric heat rejection in the condenser at pressure P_2; 3–4 is isenthalpic expansion; and 4–1 is isobaric heat absorption in the evaporator at pressure P_1. States h_1, $h_{2,\text{isen}}$ and h_4 denote the refrigerant enthalpies at the compressor inlet, compressor outlet (isentropic), and evaporator inlet, respectively.

The volumetric refrigerating effect q_v (kJ/m^3) represents the cooling capacity per unit volumetric flow rate entering the compressor and is defined as:

$$q_v = \frac{\dot{Q}_1}{\dot{V}_1} = \frac{h_1 - h_4}{v_1} \tag{6}$$

This parameter indicates the amount of cooling achieved per cubic meter of refrigerant circulated and directly influences compressor sizing and system compactness.

Similarly, the volumetric compressor work ε_v is defined as the compressor energy input per unit volumetric flow rate. For ideal (isentropic) compression:

$$\varepsilon_v = \frac{\dot{E}}{\dot{V}_1} = \frac{h_{2,\text{isen}} - h_1}{v_1} \tag{7}$$

where $h_{2,\text{isen}}$ is the enthalpy at the compressor outlet under isentropic conditions.

Based on the definition of the COP in Eq. (2), and recognizing that both the refrigeration capacity and the compressor power are proportional to the volumetric flow rate, the COP may alternatively be expressed in terms of volumetric performance parameters. Accordingly, it can be written as the ratio of the volumetric refrigerating effect to the volumetric compressor work:

$$COP = \frac{q_v}{\varepsilon_v} = \frac{h_1 - h_4}{h_{2,\text{isen}} - h_1} \tag{8}$$

2.3 Cycle Modifications Affecting Performance

To evaluate performance variations in vapor-compression systems, it is instructive to examine the effects of liquid subcooling and vapor superheating on cycle behavior and efficiency.

2.3.1 Effect of subcooling

Subcooling occurs when the liquid refrigerant leaving the condenser is cooled to a temperature below its saturation temperature at the condensing pressure. In the P–h diagram, this process is represented by an extension of the condensation line into the subcooled liquid region. Figure 4 illustrates the effect of subcooling on the refrigeration cycle. The continuous line represents the cycle with subcooling, while the dotted line corresponds to the baseline saturated case.

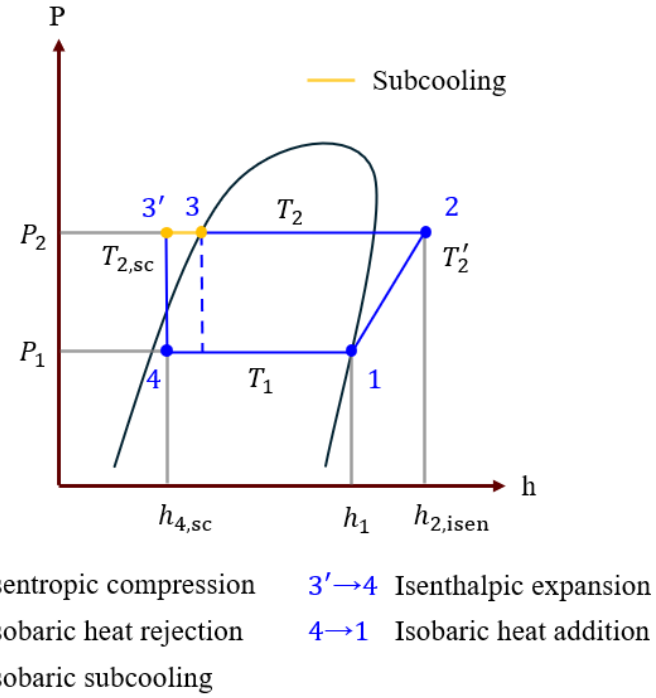


Figure 4. Pressure–enthalpy (P–h) diagram of a vapor-compression refrigeration cycle with liquid subcooling. State 3′ represents the subcooled liquid at the condenser pressure P_2, with corresponding temperature $T_{2,sc}$. The enthalpy at the evaporator inlet is reduced to $h_{4,sc}$ due to subcooling. Other state variables retain the definitions introduced in Figure 3.

Subcooling reduces the enthalpy of the liquid refrigerant entering the expansion device. After isenthalpic expansion, the refrigerant enters the evaporator at a lower enthalpy $h_{4,sc}$, while the compressor inlet state h_1 remains unchanged. Consequently, the refrigeration capacity becomes:

$$\dot{Q}_1 = \dot{m}(h_1 - h_{4,sc}) \tag{9}$$

The increase in enthalpy difference across the evaporator $(h_1 - h_{4,sc})$ leads to an increase in refrigeration capacity. Since the specific volume at the compressor inlet is unaffected, the volumetric refrigerating effect also increases directly with the degree of subcooling. Because the compressor inlet and outlet states remain unchanged, the electrical power input $\dot{E}$ remains approximately constant. Consequently, the coefficient of performance (COP) increases, indicating improved thermodynamic efficiency. The enhancement in performance is therefore

primarily attributable to the increased enthalpy difference in the evaporator without a corresponding rise in compressor power.

2.3.2 Effect of superheating

Superheating occurs when the refrigerant vapor is heated above its saturation temperature at the evaporating pressure before entering the compressor. Starting from the subcooled cycle shown in Figure 4, we now examine the effect of increasing the degree of superheating. As illustrated in Figure 5, the compressor inlet state shifts to the right on the pressure–enthalpy (P–h) diagram. While the suction pressure P_1 remains unchanged, the inlet temperature and specific enthalpy increase.

An increase in suction temperature influences two key parameters in the expression for volumetric refrigerating effect. First, the specific volume at the compressor inlet v_1 increases due to the higher vapor temperature. Second, the enthalpy at the compressor inlet h_1 increases, thereby enlarging the enthalpy difference across the evaporator (h_1–h_4). The net effect on the volumetric refrigerating effect depends on the relative magnitude of these changes. The increase in specific volume tends to reduce the cooling capacity per unit volumetric flow rate, whereas the increase in enthalpy difference tends to enhance it. In practice, these opposing effects often nearly offset one another, and the net change in volumetric refrigerating effect is typically small.

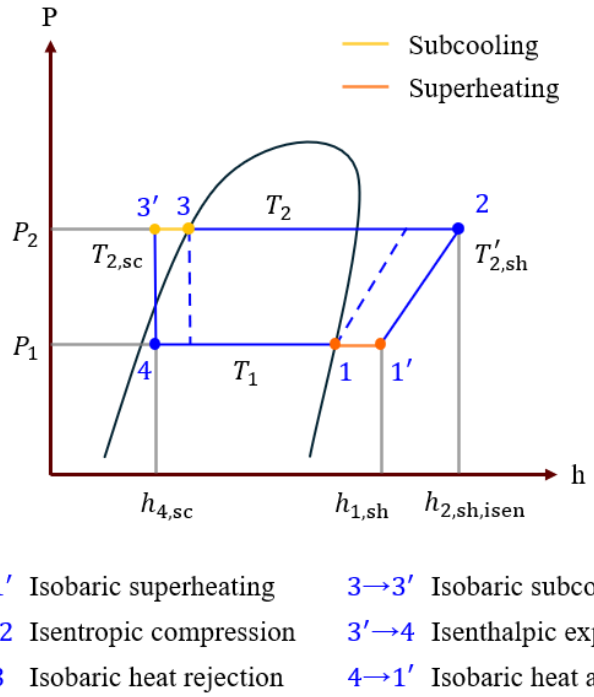


Figure 5. Pressure–enthalpy (P–h) diagram of a vapor-compression refrigeration cycle with liquid subcooling and vapor superheating. $h_{1,\mathrm{sh}}$ denotes the enthalpy of the superheated vapor at compressor inlet, whereas $h_{2,\mathrm{sh,isen}}$ and $T'_{2,\mathrm{sh}}$ represents the enthalpy and temperature after isentropic compression from the superheated state respectively. Subcooling reduces the enthalpy at the evaporator inlet to $h_{4,\mathrm{sc}}$. Other state variables retain the definitions introduced in Figures 3 and 4.

From a compression perspective, the isentropic compression process now begins from a higher enthalpy state. Consequently, the enthalpy rise during compression ($h_{2,\mathrm{sh,isen}} - h_{1,\mathrm{sh}}$) increases compared to the non-superheated case, as the isentropic curves between the compressor pressure levels diverge for higher suction temperatures. This leads to a higher electrical power

input $\dot{E}$ for the same mass flow rate. Because the increase in compressor work generally outweighs any gain in refrigerating capacity, the overall COP tends to decrease with excessive superheating.

Superheating can be classified as internal or external. Internal superheating occurs within the evaporator, where the additional heat absorbed contributes directly to the refrigeration load. In this case, the increase in enthalpy difference across the evaporator corresponds to useful cooling, and the efficiency penalty is limited. In contrast, external superheating occurs outside the refrigerated space, for example in suction lines exposed to ambient conditions. This additional heat does not contribute to useful cooling but still increases compressor work and suction specific volume. As a result, external superheating reduces system efficiency and is generally undesirable.

2.4 Heat Recovery Potential

Understanding the temperature–entropy relationship during the heat rejection process of the vapor-compression refrigeration cycle is essential for the design and optimization of heat recovery systems. The condenser does not reject heat uniformly; rather, the heat rejection process occurs in three distinct thermodynamic regions, as illustrated in Figure 6.

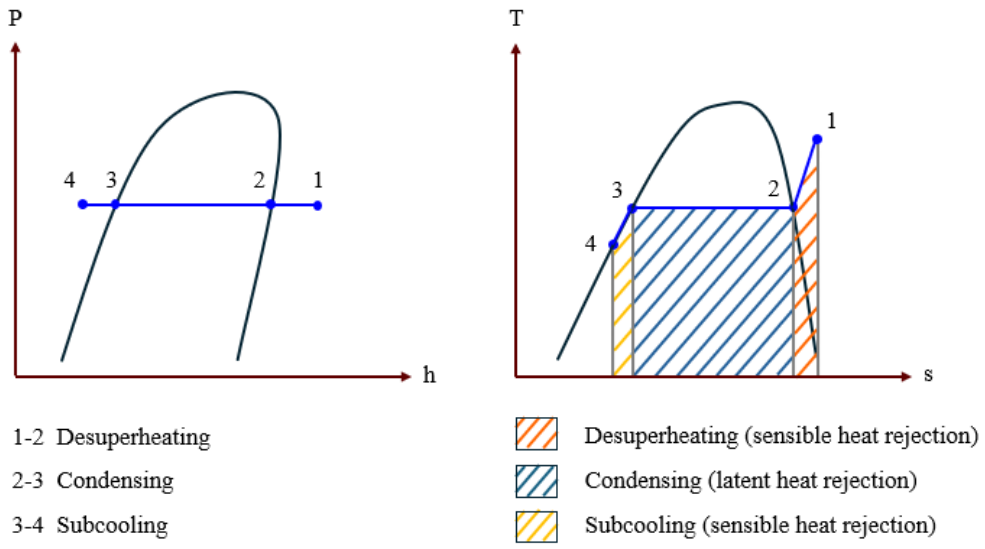


Figure 6. Thermodynamic representation of condenser heat rejection in a vapor-compression refrigeration cycle shown in (a) pressure–enthalpy (P–h) and (b) temperature–entropy (T–s) diagrams. The heat rejection process consists of three regions: 1–2 desuperheating of the superheated vapor, 2–3 condensation at approximately constant temperature and pressure, and 3–4 subcooling of the liquid refrigerant.

The first region is the *desuperheating zone (1–2)*. In this stage, the superheated vapor leaving the compressor is cooled at approximately constant pressure until it reaches the saturation temperature corresponding to the condensing pressure. The desuperheating process typically accounts for 10–20% of the total condenser heat rejection (Granryd *et al.*, 1999). Because the refrigerant temperature is highest in this region, the recovered heat is available at the highest temperature level, making it particularly attractive for high-temperature applications such as domestic hot water production or process heating. The magnitude of desuperheating heat depends strongly on compressor discharge conditions. In particular, it is influenced by compressor isentropic efficiency. Higher isentropic efficiency results in a lower discharge

temperature and therefore reduces the fraction of heat released in the desuperheating zone. Conversely, lower isentropic efficiency increases the discharge temperature, enlarging the share of high-temperature recoverable heat in this region (Granryd *et al.*, 1999).

Following desuperheating, the refrigerant enters the *condensation zone (2–3)*, where phase change occurs at nearly constant pressure and saturation temperature. This region represents the dominant portion of condenser duty, typically accounting for 70–80% of the total rejected heat (Granryd *et al.*, 1999). Although the temperature level is lower than in the desuperheating zone, the large quantity of heat released makes this region the primary source of recoverable thermal energy in most practical systems. The saturation temperature is determined by the condensing pressure and refrigerant properties, which in turn influence the suitability of the heat for various recovery applications.

The final region is the *subcooling zone (3–4),* where the condensed liquid is cooled below its saturation temperature at the same pressure. This zone generally contributes 5–10% of total condenser heat rejection (Granryd *et al.*, 1999). While the temperature level of this heat is lower than in the previous two regions, it can still be utilized effectively for low-temperature applications such as preheating feedwater or ventilation air.

From a heat recovery perspective, the total available thermal energy consists primarily of the heat released during desuperheating of the compressor discharge vapor, and the latent heat released during condensation. In many systems, subcooling is achieved through internal heat exchange (for example, via liquid–suction heat exchangers), which simultaneously induces superheating on the suction side. For this reason, subcooling is sometimes excluded from external heat recovery potential assessments.

Beyond improving refrigeration capacity and COP, subcooling and superheating also influence both the quantity and temperature level of recoverable heat. In particular, superheating raises compressor discharge temperature, increasing the temperature grade of desuperheating heat, whereas subcooling modifies the distribution of heat rejection within the condenser. Therefore, a comprehensive heat recovery analysis must consider not only total heat quantity but also its thermodynamic quality and temperature suitability for downstream applications.

2.5 Matching Recoverable Heat with Demand

In heat recovery applications, both the quantity and quality of available thermal energy must be considered. The quantity refers to the total recoverable heat flow rate, while the quality refers to its temperature level and thermodynamic usefulness. Although superheat recovery provides heat at a higher temperature level than condensation heat, the corresponding quantity is significantly smaller. In contrast, the condensation region offers a substantially larger heat flow rate, albeit at a lower temperature. Effective heat recovery design therefore requires careful matching between the available heat characteristics and the intended application.

Refrigeration systems are typically categorized according to the temperature requirements of the stored products. This classification directly influences evaporation temperature, compressor operating conditions, and ultimately the temperature level of rejected heat.

- Medium-temperature applications are used for refrigerated products such as dairy, fresh produce, and beverages, where product temperatures remain in the range of 1−14 °C. The corresponding evaporation temperature generally ranges between −15 °C and 5 °C (Karampour *et al.*, 2016).
- Low-temperature applications serve frozen product storage, including meat and ice cream, where product temperatures are maintained between −18 and −12 °C. Evaporation temperatures typically range from −40 °C to −30 °C (Karampour *et al.*, 2016).

Lower evaporation temperatures increase compressor pressure ratio and discharge temperature, thereby influencing both system efficiency and the temperature distribution of recoverable heat. The temperature at which heat is rejected in the condenser is primarily governed by refrigerant properties, system pressure, and condenser type. Different condenser technologies operate at different condensing temperatures, which directly affect both refrigeration performance and heat recovery potential:

- Air-cooled condensers typically operate at condensing temperatures 10–15 °C above ambient air temperature. In moderate climates, this corresponds to condensing temperatures of approximately 40–50 °C (Hosoz and Kilicarslan, 2004).
- Water-cooled condensers benefit from higher heat transfer coefficients and more stable cooling temperatures, allowing operation at lower condensing temperatures, generally between 35 °C and 45 °C (Suamir *et al.*, 2015).
- Evaporative condensers, which combine air and water cooling, can achieve the lowest condensing temperatures, typically in the range of 30–40 °C depending on ambient wet-bulb temperature (Harby *et al.*, 2016).

The temperature level of rejected heat determines its suitability for different recovery applications and is commonly described as the "quality" of heat. Higher temperature levels generally allow more versatile reuse options. Within the area of food refrigeration systems, available heat for recovery can be classified based on temperature levels as:

- Low-grade heat (30–40 °C): Often associated with evaporative condensers. Suitable for space heating, ventilation air preheating, and other low-temperature demands.
- Medium-grade heat (40–60 °C): Commonly available from air- or water-cooled condensers. Appropriate for domestic hot water production and moderate industrial process heating.
- High-grade heat (>60 °C): Achievable in certain systems, particularly those using natural refrigerants such as carbon dioxide (CO_2) operating at elevated discharge temperatures. This heat can support industrial processes requiring higher temperature levels.

The suitability of recovered heat for a given application depends primarily on three parameters: (i) temperature level, (ii) available heat flow rate, and (iii) temporal matching between heat supply and demand. Because refrigeration systems operate continuously while heat demand may vary seasonally or diurnally, system integration must consider both thermodynamic compatibility and operational synchronization. Table 1 summarizes typical heat demand categories and their corresponding temperature requirements, providing guidance for effective heat recovery integration.

Table 1. Typical heat demand applications and corresponding minimum temperature requirements (Stoecker, 1998). The values listed in Table 1 illustrate that a substantial portion of building-related heat demands fall within the low- to medium-temperature range (30–50 °C), which aligns well with typical condenser heat rejection temperatures in commercial refrigeration systems.

Potential use	Minimum temperature (°C)
Underfloor heating of subfreezing spaces	10 − 15
Defrost water (water defrost systems)	25 − 30
Space heating	
— docks	30 − 35
— warehouses	30 − 40

— offices and retail spaces	40 − 45
Reheat coils for humidity control	30 − 40
Preheat tap water	45 − 50
Preheating boiler feed water	45 − 50

2.6 Refrigerant-Specific Considerations for Heat Recovery

The thermodynamic and transport properties of the refrigerant play a decisive role in determining heat rejection characteristics and, consequently, heat recovery potential. Discharge temperature, pressure ratio, heat transfer coefficients, and operating regime (subcritical or transcritical) directly influence both the quantity and temperature level of recoverable heat.

Traditional synthetic refrigerants such as R-404A and R-507A exhibit moderate compressor discharge temperatures, typically in the range of 60–80 °C under standard operating conditions (Minor *et al.*, 2014; Fricke, 2011). These temperature levels are generally sufficient for space heating and domestic hot water production but offer limited suitability for high-temperature industrial heat recovery. Their well-established thermophysical properties and predictable system behavior have made them widely adopted in commercial refrigeration systems. However, their high Global Warming Potential (GWP) has led to increasing regulatory restrictions, accelerating their phase-down in many regions.

Natural refrigerants offer broader heat recovery opportunities due to favorable thermodynamic characteristics and low environmental impact. Carbon dioxide (CO_2, R-744), particularly in transcritical operation, can reach discharge temperatures between 80 °C and 140 °C depending on gas cooler pressure and ambient conditions (Sarkar, Bhattacharyya and Ram Gopal, 2007). These elevated temperatures enable recovery of higher-grade heat suitable for industrial water heating and process applications. Moreover, CO_2 systems often operate at high pressure levels, which significantly influence both compressor work and available heat temperature.

Ammonia (R-717), widely used in large-scale industrial refrigeration, also produces relatively high discharge temperatures, typically between 60 °C and 90 °C, which is strongly affected by the compressor type (Stoecker, 1998). Its excellent thermodynamic efficiency and high latent heat make it particularly suitable for integrated heat recovery systems in food processing plants, where waste heat can be reused for cleaning, pasteurization, or facility heating. Hydrocarbons such as propane (R-290) and isobutane (R-600a) exhibit discharge temperatures comparable to conventional HFCs and are therefore well suited for medium-grade heat recovery applications (Colbourne *et al.*, 2010). Their high thermodynamic efficiency and low GWP make them attractive for decentralized and small-to-medium capacity systems, although flammability considerations impose design constraints.

The newer generation of low-GWP synthetic refrigerants, particularly hydrofluoroolefins (HFOs) and HFO blends, demonstrate operating temperature levels similar to conventional HFCs. However, differences in thermodynamic glide (in the case of blends) and heat transfer properties may influence condenser design and heat recovery performance. In particular, temperature glide can affect the effective temperature profile during condensation, which may be advantageous or disadvantageous depending on the target heat application.

3. Impact of System Configurations on Heat Recovery

The configuration of a refrigeration system plays a decisive role in determining both the quantity and the temperature level of recoverable heat. System architecture influences the number of heat rejection points, operating pressure levels, discharge temperatures, and opportunities for thermal integration. Consequently, an understanding of refrigeration system configurations is essential for optimizing heat recovery strategies in food refrigeration applications.

3.1 Heat Recovery from Single-Stage, Multi-Stage, and Cascade Systems

In single-stage vapor-compression systems, all rejected heat is discharged at a single condenser operating at one pressure level. This concentration of heat rejection simplifies heat recovery design and control. Retrofitting such systems with desuperheaters or heat reclaim heat exchangers is generally straightforward, requiring minimal additional control complexity. However, single-stage systems provide a relatively narrow temperature range for heat recovery. The condenser temperature is determined primarily by ambient conditions and system design, typically limiting recovered heat to moderate temperature applications such as space heating, domestic hot water production, or low-temperature process heating in food facilities. While practical and economically attractive for retrofit applications, their flexibility in supplying multiple temperature levels is limited.

Multi-stage refrigeration systems incorporate two or more compression stages operating at different pressure levels. These systems may include intercoolers, flash tanks, or intermediate heat exchangers. As a result, heat rejection occurs at multiple temperature levels within the system. This structural feature enhances heat recovery potential in two ways: (1) Multiple discharge temperature levels allow different heat grades to be recovered simultaneously; (2) Improved thermodynamic efficiency at high compression ratios reduces overall compressor work while maintaining elevated discharge temperatures. Multi-stage configurations are particularly advantageous in low-temperature food storage facilities, where high pressure ratios would otherwise reduce system efficiency. From a heat recovery perspective, they provide greater flexibility in matching heat supply to diverse process demands within food processing plants.

Cascade refrigeration systems combine two or more independent refrigeration cycles, typically using different refrigerants optimized for high- and low-temperature stages. The low-temperature cycle removes heat from the refrigerated space and rejects it to an intermediate heat exchanger, which serves as the evaporator of the high-temperature cycle. This architecture produces distinct heat rejection zones at different temperature levels. The high-temperature stage can supply higher-grade recoverable heat, while the low-temperature stage may offer additional recoverable energy at a lower temperature level. Although cascade systems introduce greater design and control complexity, they can achieve highly efficient and tailored heat recovery solutions. This is particularly beneficial in facilities requiring both deep-freezing and moderate heating loads, where different temperature levels of recovered heat can be strategically utilized.

3.2 System Configurations Based on Component Placement

In commercial food refrigeration, system configuration is strongly influenced by the physical placement of major components such as compressors and condensers. In supermarkets and food retail environments, refrigeration systems are generally categorized into three main

configurations: plug-in (self-contained) cabinets, remote systems, and cold room installations. These architectural differences significantly affect heat recovery feasibility and performance.

Plug-in cabinets are self-contained refrigeration units in which all components, including compressor, condenser, evaporator, and expansion device, are integrated within a single enclosure. These systems require only an electrical connection for operation and are commonly used in small retail settings or for specific display applications. Because the condenser is located within or immediately adjacent to the cabinet, rejected heat is typically released directly into the surrounding retail space. While simple local heat recovery is technically possible, the distributed nature and relatively small individual capacity of these units limit their overall recovery potential. Furthermore, integrating centralized heat recovery from multiple plug-in units can be complex. Historically, open display cabinets employed air curtains to reduce infiltration losses by creating a high-velocity air barrier between the refrigerated space and the ambient environment. However, modern eco-design regulations have led to widespread adoption of glass doors, reducing cooling load and heat rejection by approximately 50% compared to open cabinets. Although this improves energy efficiency, it proportionally reduces the quantity of recoverable heat.

Remote display cabinet systems separate the evaporator section from the compressor and condenser, which are typically located in a central machinery room or in outdoor condensing units. The display cabinet itself usually contains only the evaporator and expansion device. These systems allow multiple cabinets to be connected to centralized compressor racks. When numerous smaller condensing units are distributed throughout the facility, the system is often referred to as a distributed architecture. Remote systems offer significantly greater heat recovery potential compared to plug-in cabinets due to: (1) Higher centralized heat rejection capacity; (2) Easier integration of heat reclaim heat exchangers; (3) More controllable condenser temperature levels; (4) Improved access for retrofitting recovery equipment. Because heat rejection is concentrated in machinery rooms or at centralized condensers, these systems are well suited for reclaiming condenser heat for space heating, domestic hot water, or process applications.

Cold rooms are large enclosed storage spaces used for refrigerated or frozen product preservation. They are typically connected to remote or centralized refrigeration systems, relying on external compressor and condenser units. Due to their higher cooling capacities and continuous operation, cold room systems generate substantial and stable heat rejection loads. This makes them particularly attractive for integrated heat recovery solutions, especially in food processing plants where simultaneous heating demands often exist.

A brief overview of the main system configurations is provided below.

Distributed systems with direct expansion (DX)

In distributed direct expansion (DX) systems, each display cabinet or a small group of cabinets, is connected to its own dedicated condensing unit. This configuration limits the refrigerant charge per circuit, so in the event of leakage or failure, refrigerant loss is confined to a single unit. Such systems are therefore often favored in small retail outlets or convenience stores where simplicity and containment are priorities. However, the presence of multiple condensing units increases capital cost, spatial requirements, and maintenance complexity. Although each unit contains only a modest refrigerant charge, the cumulative refrigerant inventory across the entire facility may still be significant. Moreover, the higher number of mechanical joints increases the total number of potential leakage points. From a heat recovery perspective, distributed DX systems offer limited integration potential. Because heat rejection is dispersed across multiple small condensers, centralized heat recovery becomes technically challenging

and economically less attractive. These systems are therefore generally unsuitable for large-scale structured heat reclaim applications.

Distributed systems with direct expansion and remote cooler

In distributed DX systems with a remote cooler (Figure 7), multiple compressors are arranged in parallel and connected to a shared condenser, typically water-cooled. The condenser heat is transferred via a secondary cooling loop to an external dry cooler or evaporative cooler. The condensed refrigerant is subsequently distributed to multiple evaporators operating in parallel. This hybrid configuration combines the flexibility of distributed evaporators with the efficiency benefits of a centralized condensing stage. By concentrating heat rejection at a shared condenser, the system enables more practical heat recovery integration compared to fully independent condensing units. The use of a secondary fluid loop enhances thermal control and may improve overall heat transfer performance. Furthermore, concentrating the condenser heat into a single or limited number of rejection points improves the feasibility of installing heat reclaim heat exchangers for space heating or domestic hot water applications.

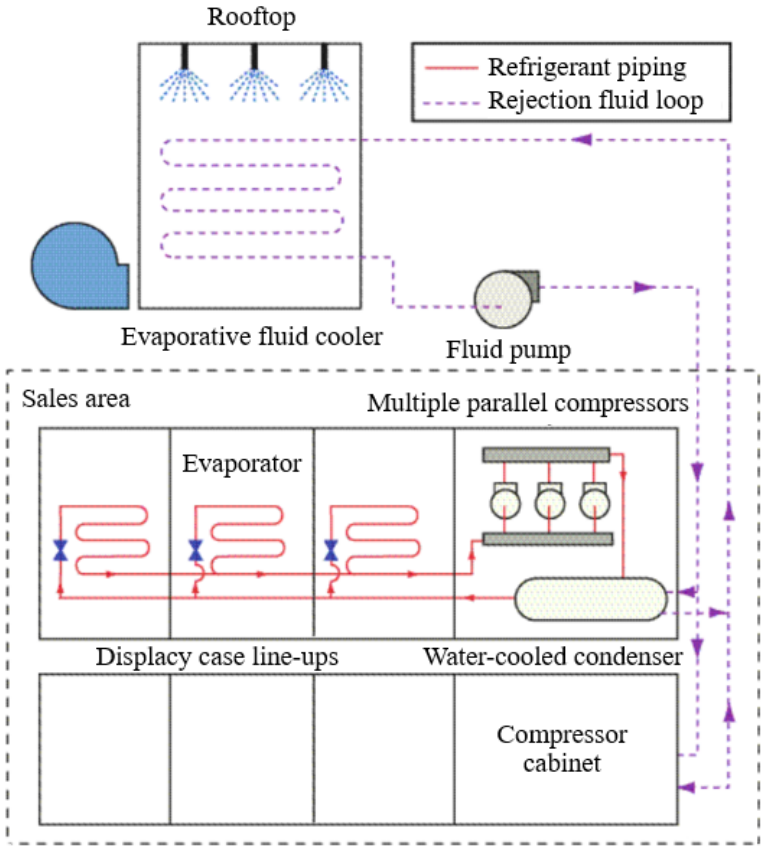


Figure 7. Distributed direct expansion (DX) refrigeration system with a shared water-cooled condenser and remote evaporative fluid cooler (Baxter, 2003). Refrigerant (solid lines) from multiple display case evaporators is routed to parallel compressors located in a central cabinet. The compressed refrigerant is condensed in a water-cooled condenser. A secondary water loop (dashed lines) circulates cooling water between the condenser and an evaporative fluid cooler located on the rooftop, where heat is ultimately rejected to the ambient environment.

Centralized (remote) systems

Centralized refrigeration systems are commonly used in medium to large supermarkets. In these systems, compressors, condensers, and receivers are located in a remote machinery room,

while only evaporators are distributed throughout the sales and storage areas (Figure 8). This architecture provides several operational advantages: (1) Centralized compressor staging allows load modulation and improved part-load efficiency; (2) System expansion is relatively straightforward; (3) Noise and waste heat are removed from customer areas.

However, centralized DX systems typically require large refrigerant charges, often in the range of 1000–2000 kg. Extensive piping networks increase both installation complexity and leakage risk, with annual leakage rates historically around 15–30% (Baxter, 2003). Long refrigerant lines may also result in pressure drops, reducing evaporating pressure and potentially lowering system performance.

Despite these drawbacks, centralized systems are highly favorable for heat recovery. Because all rejected heat is concentrated at one or a few condenser locations, integration of heat reclaim heat exchangers becomes technically straightforward. The large and continuous heat rejection capacity makes these systems particularly suitable for space heating and water heating applications.

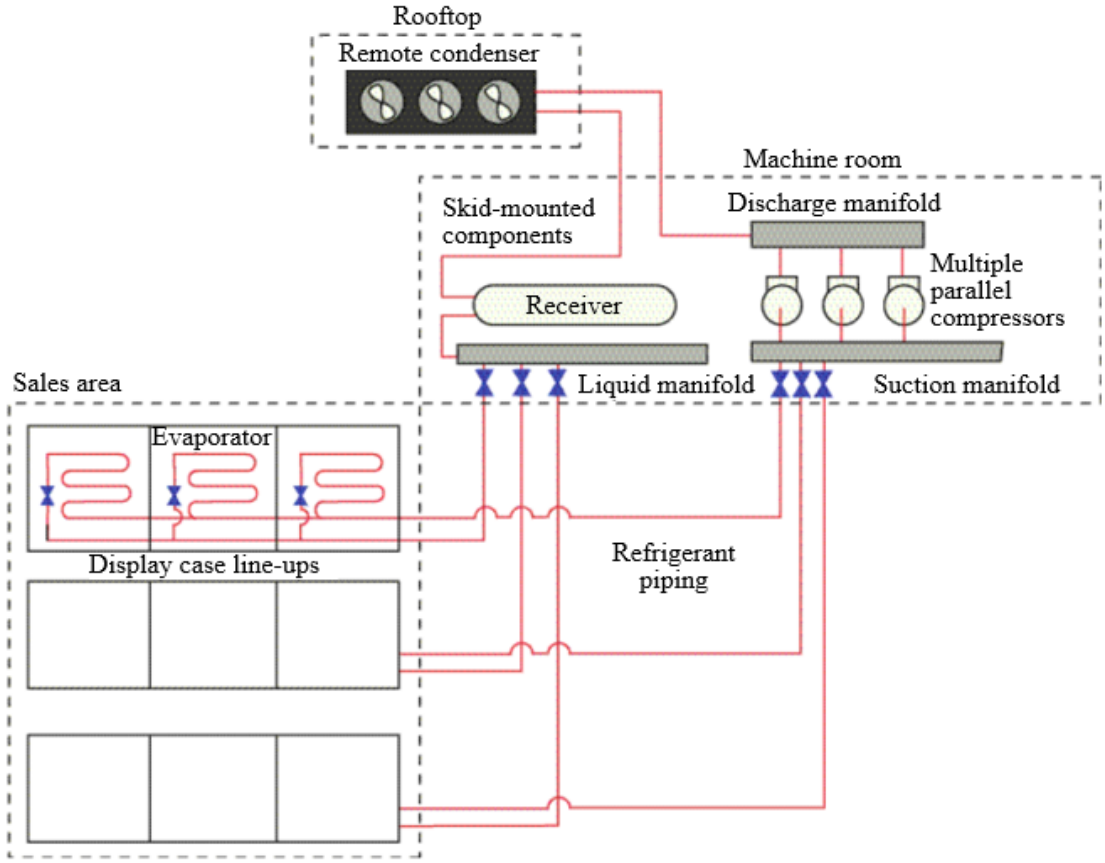


Figure 8. Centralized (remote) direct expansion refrigeration system (Baxter, 2003). Evaporators located in the sales area are connected via refrigerant piping to a remote machinery room housing multiple parallel compressors, receiver, and manifolds. The compressed refrigerant is discharged to a rooftop remote condenser, where heat is rejected to the ambient environment. Only the evaporators are distributed throughout the sales area, while all major refrigeration components are centrally located.

Secondary fluid remote systems

Secondary fluid remote systems were developed to mitigate the high refrigerant charge and leakage risks associated with centralized DX systems (Figure 9). In this configuration, the

primary refrigerant loop is confined to the machinery room. Heat from the refrigerated spaces is absorbed by a secondary fluid, typically brine or glycol, which circulates through the sales area and exchanges heat with the refrigerant via an intermediate heat exchanger. By limiting the refrigerant to a compact central circuit, these systems reduce refrigerant charge to about 10−15 % and curb leakage risk to only around 2−10 % (Baxter, 2003). Long piping runs contain only secondary fluid, eliminating pressure drop issues in refrigerant lines. From a heat recovery perspective, secondary fluid systems offer similar advantages to centralized systems, as condenser heat remains concentrated in the machinery room. However, the introduction of pumps for circulating the secondary fluid increases electrical energy consumption. The selection of the secondary fluid, based on viscosity, freezing point, and thermal properties, affects pumping power and overall system efficiency.

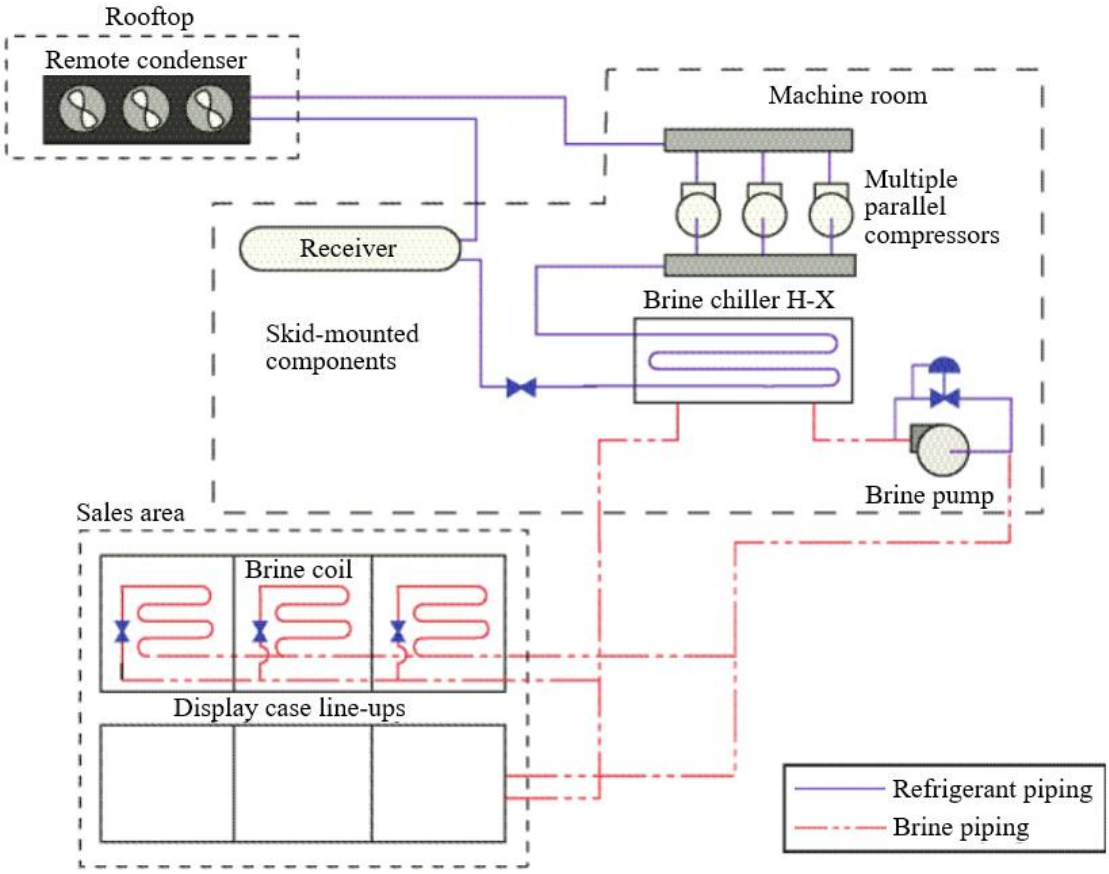


Figure 9. Secondary fluid remote refrigeration system (Baxter, 2003). The primary refrigerant circuit is confined to the machine room, where multiple parallel compressors, receiver, and a brine chiller heat exchanger are located. Heat from the display cases in the sales area is absorbed by a secondary fluid (brine) circulating through brine coils and pumped to the machine room. There, heat is transferred to the refrigerant in the chiller heat exchanger before being rejected at the remote rooftop condenser. By confining the refrigerant circuit to a compact central location, secondary fluid systems significantly reduce refrigerant charge and leakage risk while maintaining favorable conditions for centralized heat recovery.

4. Heat Recovery Technologies and Methods

4.1 Direct Heat Recovery Methods

Direct heat recovery methods extract thermal energy directly from the refrigeration cycle without intermediate storage or conversion systems. These approaches are particularly effective in food refrigeration facilities where heat demand (e.g., domestic hot water, cleaning, or space heating) coincides with refrigeration operation.

Depending on the location within the cycle and the temperature requirements of the end use, different heat exchanger configurations can be implemented. The most common direct heat recovery arrangements are discussed below.

4.1.1 Desuperheaters

Desuperheaters represent one of the simplest and most widely implemented methods for direct heat recovery. Installed between the compressor discharge and the condenser, they are designed to extract sensible heat from the high-temperature, high-pressure refrigerant vapor before it enters the main condenser. Because compressor discharge vapor is at a higher temperature than the condensation temperature, it constitutes a valuable source of high-grade thermal energy. The recovered heat is commonly used for preheating domestic hot water, cleaning water, or boiler feed water in food processing facilities.

A major advantage of employing a dedicated desuperheater lies in its improved heat transfer performance compared to integrated condenser-based recovery systems. In conventional integrated condensers, desuperheating, condensation, and subcooling occur within a single heat exchanger. Since condensation follows immediately after desuperheating, a temperature pinch point may arise (Figure 10). This pinch point restricts the maximum achievable temperature of the heating fluid and limits the recoverable high-grade heat. In contrast, when desuperheating occurs in a separate upstream heat exchanger, the pinch limitation associated with simultaneous phase change is avoided. As a result, higher outlet water temperatures can be achieved, and heat transfer remains effective even when the heating medium approaches elevated temperature levels. Under such conditions, the outlet water temperature may even exceed the refrigerant condensation temperature, since only sensible cooling of the superheated vapor is involved and no phase change constraint is imposed.

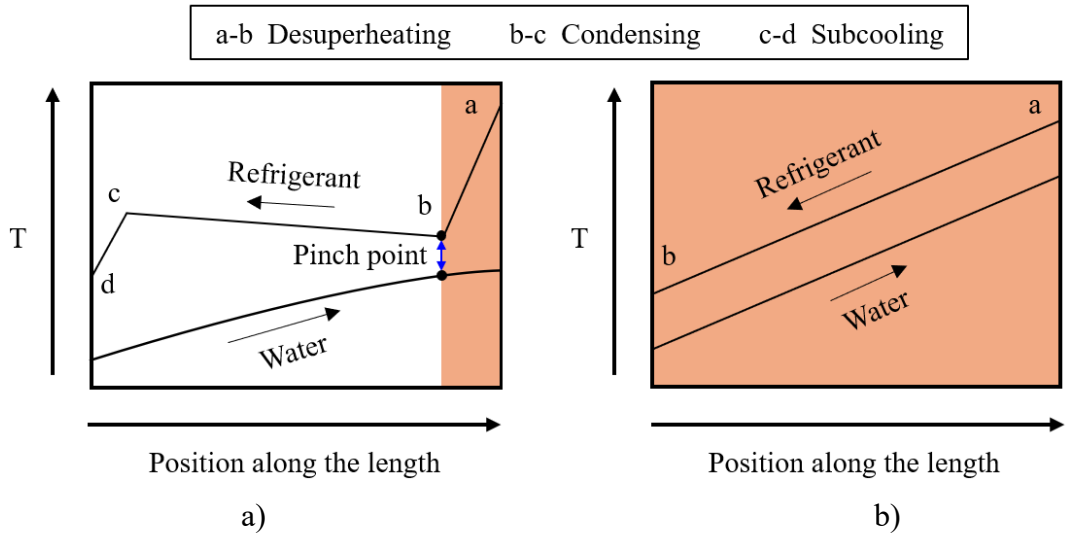


Figure 10. Temperature profiles along the heat exchanger length for (a) an integrated condenser and (b) a dedicated desuperheater. Process a-b, b-c, c-d refers to desuperheating, condensing

and subcooling respectively. In the integrated configuration, desuperheating is immediately followed by condensation, leading to a temperature pinch point that restricts recovery of high-temperature superheat. In contrast, a dedicated desuperheater allows sensible cooling of the superheated refrigerant without phase change, eliminating the pinch constraint and enabling higher outlet water temperatures.

Desuperheaters are designed primarily for sensible heat transfer, meaning the refrigerant is cooled from its superheated state to near saturation without intentional phase change. Although condensation within the desuperheater is generally avoided, small amounts of liquid refrigerant may form under certain operating conditions. Proper system design must therefore ensure that any condensed liquid can drain safely back to the condenser. Ideally, the desuperheater is installed above the condenser to allow gravity-driven liquid return. When vertical placement is not feasible, piping must be carefully designed to prevent liquid accumulation. Recommended gas velocities in the range of 5–10 m/s help ensure adequate vapor transport and minimize liquid holdup, thereby maintaining stable and efficient operation (Granryd *et al.*, 1999).

In multi-stage systems, recovering heat from the discharge of the low-stage compressor provides additional benefits. It satisfies heating demand while simultaneously lowering the thermal load on the high-stage condenser. This can decrease overall compressor power consumption and improve system-level efficiency.

4.1.2 Condenser heat recovery

Heat recovery from condensers is one of the most widely implemented strategies in commercial refrigeration systems, since the majority of rejected heat occurs during the condensation process. As shown previously, condenser heat rejection represents the combined effect of the refrigeration load and compressor work input. Because this heat is available continuously during system operation, it constitutes a substantial and reliable energy source. Condenser heat recovery systems typically redirect part or all of the condenser heat rejection to a secondary heat exchanger, where thermal energy is transferred to a heating medium such as water or air. In water-cooled systems, condenser water can be routed through a plate or shell-and-tube heat exchanger to preheat domestic hot water, cleaning water, or process water. In air-cooled systems, a heat reclaim coil may be integrated into the air handling system to preheat ventilation air or provide space heating. Although the temperature level of condensing heat is generally lower than that of desuperheated discharge vapor, the available heat quantity is significantly larger. As a result, condenser heat recovery is particularly suitable for applications requiring moderate-temperature heat in substantial volumes.

A practical example can be found in milk collection centers, where a shell-and-coil heat exchanger has been implemented to recover waste heat from the condenser of a refrigeration system. The recovered thermal energy is used to heat water for cleaning milk processing equipment. In this application, the system is capable of recovering the entire superheat portion and approximately 35% of the latent heat released during condensation. The integration of the heat recovery exchanger increased the system COP from 3.0 to 4.8 under operating conditions, reflecting improved overall energy utilization. At a representative milk collection center, the installation resulted in monthly energy savings of approximately 600 kWh for water heating. The recovered heat was able to raise water temperature up to 64.5 °C. Depending on system load and operating COP, the heat recovery fraction varied between approximately 37% (at COP ≈ 4) and 85% (at COP ≈ 5.2) (Sapali *et al.*, 2014). These results demonstrate the substantial potential for energy cost reduction and improved system performance when condenser heat is effectively integrated into process heating applications.

4.1.3 Cascade systems with heat reclaim

Cascade refrigeration systems consist of two or more vapor-compression cycles thermally coupled through a cascade heat exchanger. This intermediate heat exchanger functions as the condenser of the low-temperature (LT) cycle and the evaporator of the high-temperature (HT) cycle. By separating temperature levels between cycles, cascade systems enable very low evaporating temperatures while allowing each cycle to operate near its optimal pressure ratio and thermodynamic efficiency. This architecture is particularly suitable for applications requiring simultaneous cooling at different temperature levels, such as supermarket chilled and frozen food storage, as well as industrial food processing facilities. From a heat recovery perspective, cascade systems offer multiple heat rejection points at distinct temperature levels, enhancing flexibility in thermal integration.

Carbon dioxide (CO_2, R-744) is frequently used in the low-temperature stage due to its favorable thermophysical properties at sub-ambient conditions. The high-temperature stage may employ ammonia (NH_3, R-717), hydrocarbons, or optimized HFC/HFO blends designed for higher condensing temperatures. Many modern cascade systems rely on natural refrigerants such as CO_2 and NH_3, offering environmental advantages through low global warming potential and reduced refrigerant charge compared to conventional HFC-based systems.

From a heat recovery perspective, cascade systems provide multiple opportunities for thermal integration. Heat can be recovered from the high-temperature condenser, where moderate- to high-grade heat is available, as well as from the discharge line of the low-temperature stage, where desuperheating heat is present at elevated temperature levels. Because the high-temperature stage typically operates at higher condensing temperatures than single-stage systems, the recovered heat is often suitable for domestic hot water production, space heating, and selected industrial process applications.

Overall COP values exceeding 4.0 have been documented in optimized cascade installations. Studies indicate annual energy cost reductions of approximately 5–6% for CO_2-based cascade systems compared to conventional direct expansion systems (Ge and Tassou, 2014). In certain configurations, such as CO_2/NH_3 cascade systems, annual energy savings of up to 35% have been reported relative to traditional R-404A systems. A laboratory study conducted in Brazil comparing a CO_2/R404A cascade system with conventional R404A and R22 direct expansion systems observed approximately 22% higher efficiency for the cascade configuration (da Silva, Bandarra Filho and Antunes, 2012).

The effectiveness of heat recovery in cascade systems depends strongly on control strategy and climatic conditions. In colder climates, floating head pressure control is often employed to reduce compressor work while maintaining sufficient discharge temperature for useful heat reclaim. A UK supermarket case study of an all-CO_2 cascade system demonstrated that floating head pressure control combined with heat recovery delivered cost savings of approximately 5.6% compared to the base case (Ge and Tassou, 2014). In warmer regions, maintaining adequate heat recovery temperature may require controlled elevation of discharge pressure or deliberate subcooling strategies, which increase available heat temperature at the expense of higher compressor power. Careful selection of refrigerants, cascade heat exchanger sizing, and operating setpoints is therefore essential to balance refrigeration efficiency, heat recovery performance, and overall economic viability. When properly designed and controlled, cascade refrigeration systems offer a highly effective solution for achieving high-efficiency, low-GWP cooling with integrated heat recovery.

4.1.4 Hot water preheating using heat reclaim

A heat reclaim tank, illustrated in Figure 11, is a practical implementation of desuperheating heat recovery for domestic hot water preheating. The system consists of a water storage tank containing a submerged refrigerant coil. The tank is installed in the cold water supply line upstream of a conventional water heater, thereby functioning as a preheating stage that reduces the primary heater's energy consumption. On the refrigeration side, superheated vapor leaving the compressor discharge manifold is diverted through a three-way control valve into one or more reclaim tanks. The refrigerant flows through the submerged coil, transferring sensible heat to the water before returning to the main condenser. The three-way valve is typically controlled by the outlet water temperature from the reclaim tank, ensuring that the preheated water temperature does not exceed the designated hot water set point. When the target temperature is reached, refrigerant flow bypasses the reclaim tank and proceeds directly to the condenser.

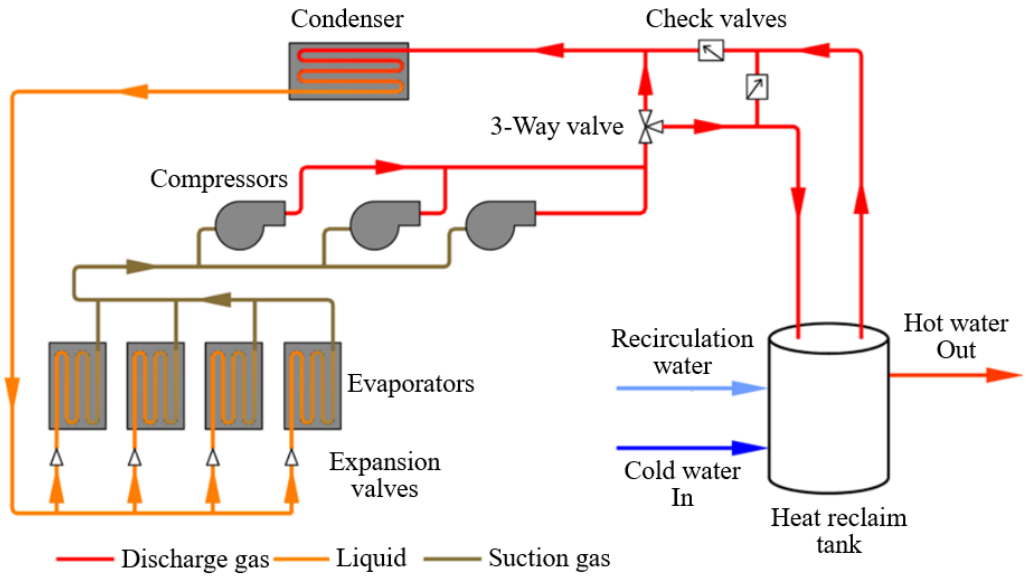


Figure 11. Hot water preheating using a heat reclaim tank for recovery of refrigerant desuperheating heat (Reis *et al.*, 2015). Superheated discharge vapor from multiple parallel compressors is diverted through a three-way control valve to a submerged coil inside the heat reclaim tank. Heat is transferred to incoming cold water before the refrigerant returns to the condenser. Check valves regulate flow direction, and the three-way valve modulates refrigerant flow based on hot water temperature and demand.

The effectiveness of this arrangement depends on hot water demand. During periods of low demand, the tank temperature rises and approaches the hot water set point. Under these conditions, the temperature difference between the refrigerant and the water decreases, limiting further heat transfer and preventing condensation of the refrigerant within the reclaim coil. Conversely, during periods of high hot water demand, incoming cold water lowers the tank temperature, increasing the temperature difference and enhancing heat recovery from the superheated vapor.

The refrigerant coil within the tank is carefully sized to ensure that the vapor remains predominantly in the superheated region during passage through the coil. Avoiding condensation within the reclaim tank simplifies refrigerant and oil return management and prevents operational instability. In properly designed systems, the heat reclaim tank operates as a controlled desuperheater, providing efficient and reliable preheating while maintaining stable refrigeration performance.

4.1.5 Integration of heat reclaim with HVAC systems

In commercial buildings, HVAC systems perform four primary functions: heating, cooling, dehumidification, and humidification. Refrigeration heat recovery can be effectively integrated with HVAC systems, particularly for ventilation air preheating and dehumidification reheat applications. In such arrangements, heat recovered from the refrigeration system offsets heating demand in the air handling unit (AHU), thereby reducing total building energy consumption. In a direct refrigerant reclaim configuration, a refrigerant coil is installed within the air handler. Superheated discharge vapor from the refrigeration system is diverted through a three-way control valve to the reclaim coil before entering the main condenser. As outside air enters the air handling unit, it first passes over the reclaim coil, where it is preheated by the refrigerant. The refrigerant, having transferred sensible heat, exits the reclaim coil as a gas and proceeds to the primary condenser for further heat rejection.

In dehumidification applications, incoming air is first cooled below its dew point to remove moisture. The now-cooled and dehumidified air is subsequently reheated using the recovered refrigerant heat before being supplied to the occupied space. This approach avoids overcooling and excessive reheat energy consumption while maintaining indoor humidity control. A schematic representation of this configuration is shown in Figure 12.

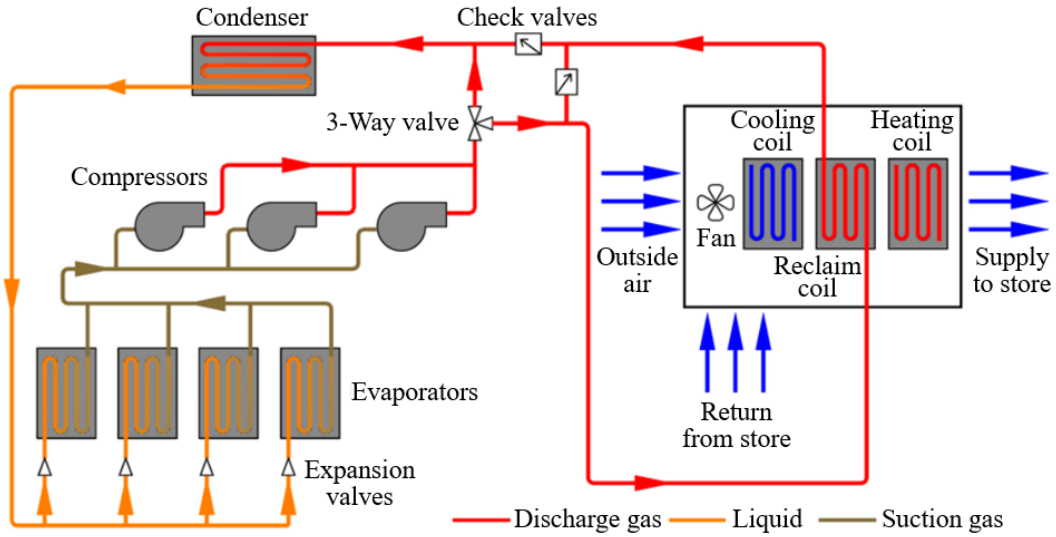


Figure 12. Direct refrigerant heat reclaim integrated with an air handling unit for dehumidification reheat (Reis *et al.*, 2015). Superheated discharge vapor from multiple parallel compressors is diverted through a three-way control valve to a reclaim coil installed in the air handler. After transferring sensible heat to the supply air stream, the refrigerant returns to the main condenser. Check valves ensure proper flow direction, and the reclaim capacity is modulated based on HVAC heating demand.

In applications where higher heating capacity is required, a condensing reclaim configuration may be implemented. In this arrangement, refrigerant from the compressor discharge is redirected through a water-cooled reclaim condenser located upstream of the primary condenser. Within this reclaim heat exchanger, the refrigerant may undergo desuperheating, partial condensation, or complete condensation before flowing to the main condenser. A secondary water loop transfers the recovered heat from the reclaim condenser to a reclaim coil within the air handling unit. This indirect configuration is advantageous when the air handler is located remotely from the refrigeration machinery room. If the air handler is situated directly above or adjacent to the condenser, the intermediate water loop may be omitted, and direct

refrigerant heat exchange can be employed. A schematic of the condensing reclaim system with a water loop is shown in Figure 13.

When heating demand is consistently high, the system can operate in a parallel condensing configuration. In this mode, all refrigerant passing through the reclaim condenser is fully condensed and directed to the receiver. This arrangement allows the reclaim condenser to be installed below the primary condenser, provided it remains above the receiver to ensure proper liquid management. Capacity control of the reclaim heat exchanger is essential to guarantee complete condensation and stable system operation.

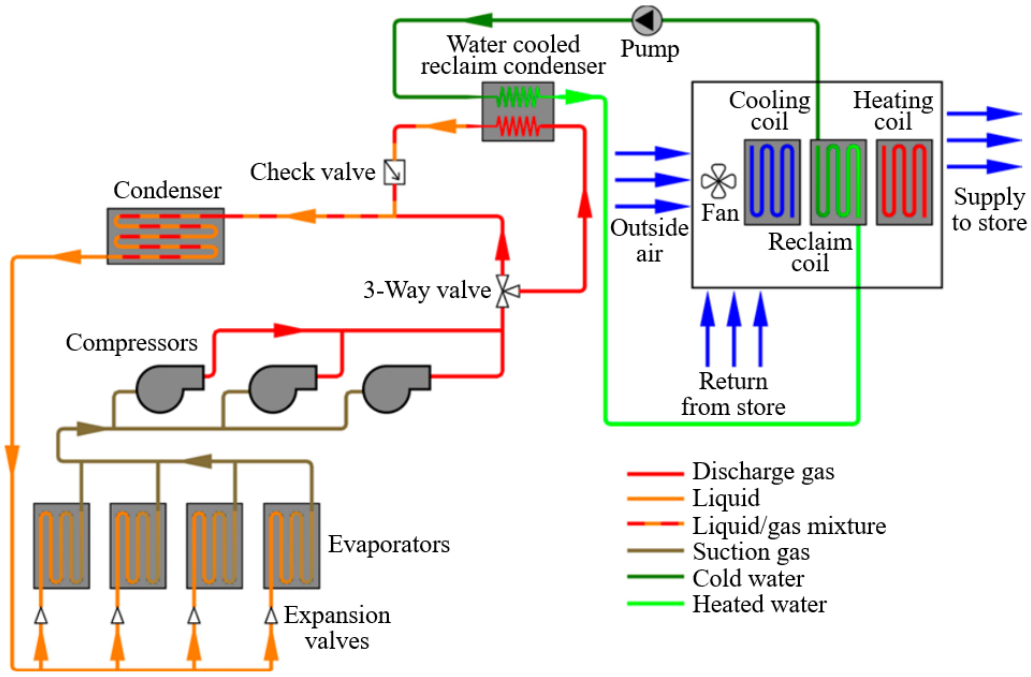


Figure 13. Indirect condensing heat reclaim system with water loop for HVAC air preheating and dehumidification reheat (Reis *et al.*, 2015). Superheated discharge refrigerant is diverted through a three-way valve to a water-cooled reclaim condenser, where desuperheating and partial or complete condensation may occur. The recovered heat is transferred via a pumped water loop to a reclaim coil installed in the air handling unit. After leaving the reclaim condenser, the refrigerant flows to the primary condenser for further heat rejection depending on operating conditions. Flow direction is regulated by check valves, and reclaim capacity is controlled according to HVAC heating demand.

4.2 Indirect Heat Recovery Methods

Indirect heat recovery methods incorporate intermediate systems such as heat pumps or thermal energy storage to decouple the availability of refrigeration waste heat from heating demand. Unlike direct refrigeration heat recovery systems, which serve simultaneous cooling and heating loads, indirect approaches enable temporal and temperature-level flexibility. These systems are particularly suitable for facilities with variable or time-shifted heating needs, including food processing plants operating on intermittent schedules or buildings connected to district heating and cooling networks.

4.2.1 Heat pump integration

Heat pumps provide an effective means of upgrading low- to medium-temperature waste heat from refrigeration systems to temperature levels suitable for space heating, domestic hot water, or industrial process applications. In many commercial refrigeration systems, condenser heat

is available at approximately 30–45 °C, which is often insufficient for direct use. By integrating a heat pump, this thermal energy can be elevated to 60–75 °C or higher, enabling its utilization in hot water production or process heating (Assaf *et al.*, 2010). In a typical configuration, waste heat from the refrigeration condenser loop is supplied to the evaporator of a heat pump. The heat pump then raises the temperature to the desired level through mechanical compression. This approach allows a large fraction of rejected heat to be recovered without significantly increasing the condensing temperature of the primary refrigeration system. Maintaining a moderate refrigeration condensing temperature is essential to avoid excessive compressor power consumption and reduced refrigeration COP.

The thermodynamic performance of the heat pump must be carefully optimized. Increasing the condensing temperature of the heat pump to achieve higher outlet temperatures inevitably reduces its COP and increases electrical demand. Therefore, system design requires balancing the temperature lift against overall energy efficiency. Achieving this balance demands detailed analysis of the heating load profile, the refrigeration system's heat rejection characteristics, and precise control of valves, compressors, and expansion devices. Proper coordination ensures that the heat pump operates in synergy with the refrigeration process without compromising system stability.

An integrated control strategy is particularly important in supermarket applications where heating and cooling loads fluctuate. In one Canadian supermarket case study, compressor discharge gas was successfully utilized to supply heat to rooftop heat pumps, as shown in Figure 14. The integrated system provided space heating while reducing annual specific electrical energy consumption by approximately 6% and energy costs by nearly 18% (Baxter, 2003).

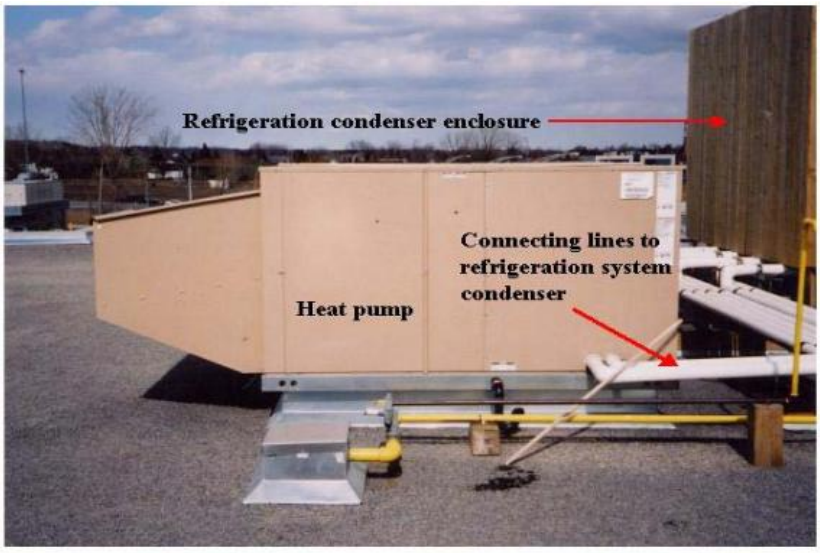


Figure 14. Rooftop heat pump integrated with a supermarket refrigeration system in Canada (Baxter, 2003). Waste heat from the refrigeration condenser is supplied to the heat pump, which upgrades the thermal energy for space heating applications. The enclosure contains the heat pump unit and associated connections to the refrigeration condenser loop.

Heat pump integration is also widely implemented in the food processing industry, including slaughterhouses, dairies, and breweries, where simultaneous heating and cooling demands are substantial. These facilities typically reject significant quantities of low-temperature heat from refrigeration systems, particularly from ammonia-based installations. By upgrading this heat, high-temperature hot water can be supplied for cleaning, sterilization, and space heating (Jouhara *et al.*, 2018). For example, a model-based prediction study showed that the integration

of a transcritical CO_2 heat pump onto an ammonia-based milk refrigeration system for condenser heat recovery can preheat boiler feed water to 70 °C, reducing total energy cost and CO_2 emissions by 33.8% and 45.7% respectively (Singh and Dasgupta, 2017). In a numerical analysis of a Swedish slaughter and meat processing plant, installation of an additional heat pump was shown to be economically attractive, with payback periods below three to four years and annual CO_2 emission reductions of 2.2–4.0 kt (Fritzson and Berntsson, 2006). When properly designed and controlled, indirect heat recovery through heat pump upgrading provides a flexible and highly effective strategy for maximizing the value of refrigeration waste heat.

4.2.2 Thermal energy storage

Thermal Energy Storage (TES) is not, strictly speaking, a heat recovery method; rather, it functions as an enabling technology that enhances the effectiveness of heat recovery systems. By storing recovered heat for later use, TES decouples heat availability from heating demand. This capability is particularly valuable in refrigeration systems, where cooling loads and heating requirements often vary over daily or seasonal cycles. In commercial food refrigeration, excess heat may be generated during periods of low heating demand, such as overnight operation. Without storage, this thermal energy would be rejected to the ambient environment. By integrating TES, surplus heat can be stored and later reused for space heating, domestic hot water, or process applications, thereby reducing reliance on fossil-fuel boilers. TES also facilitates integration with renewable energy sources, such as solar thermal collectors or photovoltaic-powered heat pumps, by absorbing surplus generation and delivering it during periods of insufficient supply. In addition, TES supports demand-side management by shifting electrical consumption to off-peak tariff periods, contributing to reduced operating costs.

In refrigeration-based systems, TES is most commonly implemented as sensible heat storage using buffer tanks. In this configuration, thermal energy is stored by increasing the temperature of a storage medium, typically water, without undergoing phase change. The buffer tank can be charged by transferring heat recovered from desuperheaters or condensers and later discharged to meet heating loads. A typical arrangement is shown in Figure 15, where heat extracted from the refrigerant via a desuperheater heat exchanger is stored in a buffer vessel. Variable-speed pumps regulate both charging and discharging flows to match system demand.

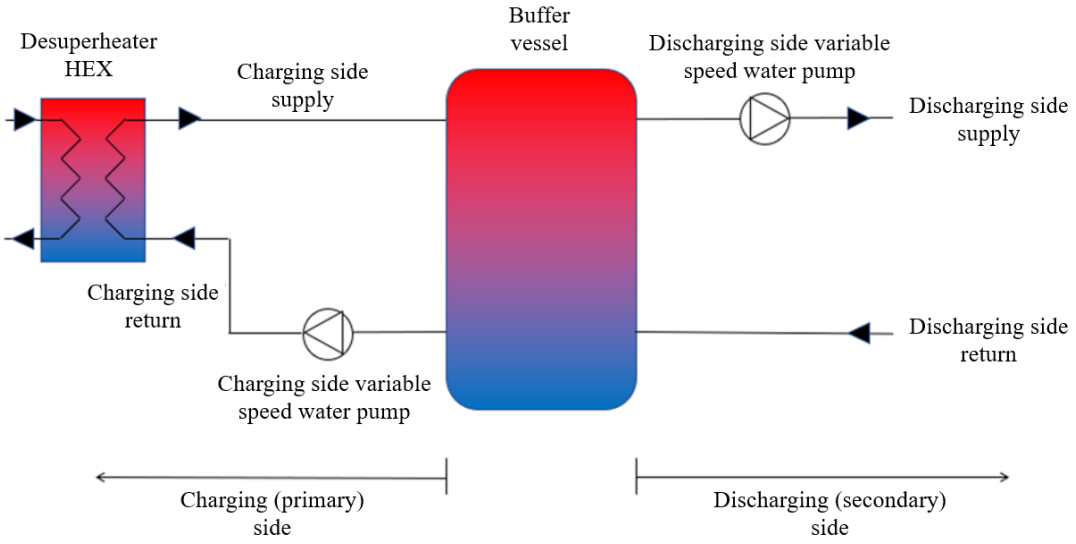


Figure 15. Thermal energy storage configuration using a stratified buffer vessel to store heat recovered from a desuperheater heat exchanger (Maouris *et al.*, 2020). The primary (charging) circuit transfers recovered thermal energy to the storage tank via a variable-speed pump, while the secondary (discharging) circuit supplies heat to the load through an independently

controlled pump. This arrangement enables flexible decoupling between heat recovery and heating demand.

The integration of TES influences the thermodynamic performance and operational strategy of refrigeration systems. By smoothing load fluctuations, TES can improve overall system utilization and reduce peak heating demand. Refrigeration Integrated Heating and Cooling (RIHC) systems, which simultaneously provide refrigeration and building heating, have demonstrated energy savings of approximately 17–18% and greenhouse gas emission reductions of 12–13% compared to conventional configurations using separate gas boilers for space heating (Maouris *et al.*, 2020). The addition of a buffer vessel has been reported to increase recoverable heat utilization by 11–12% and reduce annual operating costs by at least 2.2% relative to systems without storage (Maouris *et al.*, 2020).

More advanced configurations combining RIHC, air-source heat pumps (ASHP), and controlled thermal storage in a 1,856 m^2 supermarket have demonstrated substantial performance improvements, including a 62% reduction in energy demand and a 54% reduction in CO_2 emissions, with a reported 6.3% return on investment and a 16-year simple payback period, when evaluated using a simulation model (Sarabia Escriva *et al.*, 2022). In another case study involving the modelling of a RIHC system in a 1,300 m^2 supermarket in the Netherlands, annual energy savings of approximately 13% was achieved compared to a conventional refrigeration and gas boiler configuration (Shi *et al.*, 2017). However, strategies aimed at increasing recoverable heat, such as raising condenser pressure to elevate heat temperature, may negatively impact refrigeration COP. Therefore, TES integration must be optimized to balance heat recovery, refrigeration efficiency, and electrical consumption.

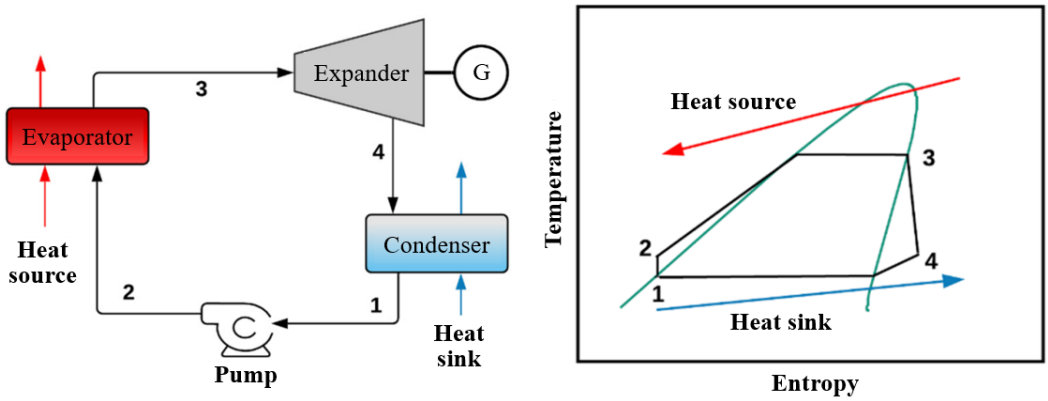
Maintaining stable storage temperatures and minimizing thermal losses from the buffer tank are essential for preserving stored energy. Although TES adds system complexity, proper thermodynamic modeling and advanced control algorithms enable coordinated operation between refrigeration, heating, and storage subsystems. When effectively implemented, TES enhances flexibility, supports renewable integration, and reduces peak electrical demand while maximizing the utilization of recovered refrigeration waste heat (Sarabia Escriva *et al.*, 2022).

4.3 Advanced and Emerging Heat Recovery Technologies

Conventional heat recovery in refrigeration systems primarily targets space heating or hot water production. However, advanced technologies extend the value of waste heat by converting it into mechanical or electrical power. Among these technologies, Organic Rankine Cycle (ORC) systems and thermoelectric generators represent promising approaches for enhancing overall energy utilization in food refrigeration applications.

4.3.1 Organic Rankine cycles

The Organic Rankine Cycle (ORC) is a thermodynamic power generation technology that converts low- to medium-grade thermal energy into mechanical work or electricity. While it operates on the same fundamental principles as the conventional Rankine cycle, it employs organic working fluids with lower boiling points than water, enabling efficient operation at comparatively low heat source temperatures. A basic ORC system, illustrated in Figure 16, consists of an evaporator, an expander (such as a scroll or screw expander), a condenser, and a pump. Thermal energy supplied to the evaporator vaporizes the working fluid, which then expands through the expander to produce mechanical power. This power can either drive a generator to produce electricity or be mechanically coupled to auxiliary equipment. After expansion, the working fluid rejects heat at a lower temperature in the condenser. The condensed working fluid is subsequently pumped back to the evaporator, completing the cycle.



a) b)

Figure 16. Organic Rankine Cycle (ORC) power generation system: (a) schematic layout and (b) temperature–entropy (T–s) diagram (Camilo Jiménez-García *et al.*, 2023). The thermodynamic processes are defined as follows: 1–2 compression of the liquid working fluid; 2–3 heat addition in the evaporator; 3–4 expansion through the expander producing mechanical power; and 4–1 heat rejection in the condenser.

The integration of ORC systems with transcritical CO_2 (R744) booster refrigeration systems in supermarkets has received considerable attention, particularly in warm and hot climates. In such configurations, the ORC recovers waste heat from the CO_2 gas cooler or compressor discharge line. Studies indicate that the ORC can offset between 4% and 24% of total electricity demand, depending on climatic conditions. In warm climates such as Athens and Madrid, the contribution typically ranges from 4–5%, whereas in hotter regions including New Delhi, Bangkok, Phoenix, and Riyadh, electricity coverage increases to approximately 14–24%. ORC systems generally become thermodynamically viable when ambient temperatures exceed approximately 27–28 °C, as higher gas cooler outlet temperatures provide sufficient driving potential. When combined with parallel compression in CO_2 systems, further performance improvements have been reported. Configurations incorporating CO_2 with parallel compression and ORC (CO_2 + PC + ORC) achieved additional energy savings of 6–9% in warm climates and 11–14% in hot climates. Economic analyses demonstrated strong feasibility, with reported payback periods below 0.5 years in hot climates and between 1.3 and 2.5 years in warm climates (Sengupta *et al.*, 2023).

Beyond supermarket applications, ORC systems have been investigated in industrial food processing and cold storage facilities. In these settings, waste heat from vapor compression refrigeration system (VCRS) condensers can be converted into electrical power. One study reported that integrating an ORC increased the combined system COP from 3.10 (stand-alone VCRS) to 3.54. More advanced hybrid systems have also been explored. In a cascaded vapor compression–absorption refrigeration configuration, the ORC system powered by waste heat functioned as a cogeneration unit, supplying electricity to drive the VCRS compressor while providing thermal energy to the desorber of the absorption cooling stage. This system achieved an energetic efficiency of 22.3% in cooling mode and 79% in cogeneration mode, while reducing compressor power demand from 19.2 kWe to 8.6 kWe, corresponding to a 55% reduction compared to a stand-alone VCRS (Patel, Desai and Kachhwaha, 2017).

Computational analyses of combined ORC–vapor compression cooling systems further demonstrated the ability to utilize low-grade heat at temperatures as low as 60 °C while delivering refrigeration at temperatures down to –10 °C. Simulated overall COP values ranged between 0.1 and 0.6, depending on evaporator temperature and operating conditions

(Aphornratana and Sriveerakul, 2010). Although these COP values are lower than those of conventional systems, they represent additional cooling obtained from otherwise rejected heat.

Despite their promising potential, ORC systems face several technical and economic challenges. Working fluid selection must balance thermodynamic performance, environmental impact, and safety considerations such as flammability. Performance sensitivity under part-load conditions can reduce efficiency in systems with fluctuating heat availability. Capital costs remain relatively high, largely driven by heat exchanger requirements. Furthermore, integration complexity increases in multi-stage or hybrid configurations. Ongoing research focuses on compact ORC modules with high-efficiency scroll or screw expanders, adaptive control strategies for variable heat sources, and hybridization with thermal energy storage or phase change materials to enhance operational flexibility and economic feasibility.

4.3.2 Thermoelectric generators

Thermoelectric generators (TEGs) provide a solid-state approach for converting refrigeration waste heat directly into electrical power. Their operation is based on the Seebeck effect, whereby a temperature difference across dissimilar semiconductor materials generates an electromotive force. Because TEGs contain no moving parts, require minimal maintenance, and can operate over a wide range of heat source temperatures, they are particularly attractive for distributed waste heat recovery in commercial and industrial refrigeration systems.

A typical TEG module, illustrated in Figure 17a, consists of multiple pairs of p-type and n-type semiconductor elements electrically connected in series and thermally arranged in parallel. When one side of the module is exposed to a hot surface and the other to a cooler heat sink, charge carriers migrate due to the temperature gradient, producing direct current electricity. The performance of a TEG is governed primarily by the imposed temperature difference, the Seebeck coefficient of the semiconductor materials, and their thermoelectric properties, which are collectively characterized by the dimensionless figure of merit, ZT.

Integration of TEGs onto refrigeration systems can occur at several locations where heat is rejected. One common configuration places a combined heat exchanger–thermoelectric generator (HE-TEG) unit in the desuperheating section downstream of the compressor. In this arrangement, high-temperature discharge vapor supplies heat to the TEG modules before entering the main condenser. Alternative integration points include the condenser outlet or a secondary hot-fluid loop extracting heat from the desuperheater and delivering it to a remote TEG assembly. The cold side of the TEG may be cooled by ambient air, a water loop, or even form part of the refrigeration system cooling load itself. Figure 17b illustrates a simplified configuration in which heat recovered from the desuperheater drives the TEG to partially offset compressor electrical demand.

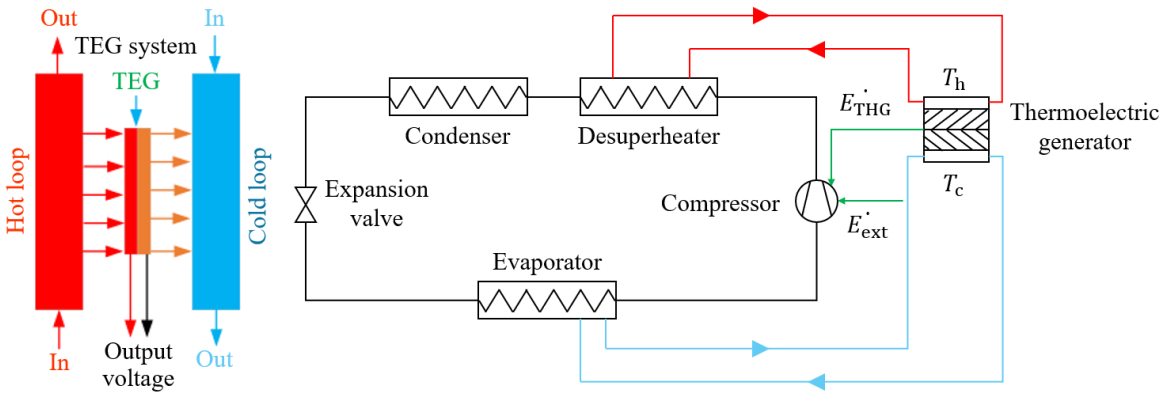


Figure 17. (a) Schematic of power generation using a thermoelectric generator (TEG) based on the Seebeck effect (Wiriyasart, Kaewluan and Naphon, 2022). (b) Configuration of a self-recuperative vapor-compression refrigeration system integrated with a thermoelectric generator. Heat recovered from the desuperheater supplies the hot side temperature T_h of the TEG, while the cold side T_c is cooled by a secondary loop. The electrical power generated by the TEG ($\dot{E}_{THG}$) partially offsets the external power input to the compressor ($\dot{E}_{ext}$), reducing net external electrical demand.

Experimental investigations have demonstrated the feasibility of such integrations. In one study involving an R134a vapor compression refrigeration system, a desuperheater-integrated HE-TEG unit generated between 3 and 20 W under an effective temperature difference of approximately 30 °C. The integration reduced compressor discharge temperature and pressure ratio, resulting in a reported COP improvement of 17–32% under specific test conditions (Attar *et al.*, 2021). In another experimental system, waste heat recovered via a concentric-tube heat exchanger supplied a closed-loop heat sink TEG configuration. A maximum output voltage of 9.8 V was achieved at a temperature difference of 19.8 °C, accompanied by increases in cooling capacity and COP of approximately 40% and a reduction in power consumption of nearly 23% (Wiriyasart, Kaewluan and Naphon, 2022). Industrial-scale prototypes have also been reported, producing approximately 29 W at an 80 °C temperature gradient (Ando Junior, Calderon and de Souza, 2018).

Despite these promising results, thermoelectric generators generally exhibit low heat-to-electricity conversion efficiencies, typically on the order of 3–5% in practical applications (Tohidi, Ghazanfari Holagh and Chitsaz, 2022). Although laboratory efficiencies approaching 9–12% (Champier, 2017; Shi, Zou and Chen, 2020) have been reported for advanced materials, real-world performance is often constrained by limited temperature gradients, contact resistances, and heat exchanger inefficiencies. Consequently, TEG technology remains in an early stage for large-scale refrigeration applications.

Ongoing research aims to improve both material performance and system integration. Material-level advances focus on nanostructured and novel semiconductor compositions that increase the figure of merit (ZT) by reducing thermal conductivity while preserving electrical conductivity. At the system level, optimization efforts include advanced heat exchanger geometries, hybrid configurations combining TEGs with phase change materials (PCM) or vapor compression cycles (VCR), and compact modular designs suitable for cylindrical piping or condenser surfaces. While TEGs are unlikely to replace mechanical power recovery technologies such as ORC systems in high-capacity applications, they offer potential advantages in modular, maintenance-free, and distributed waste heat recovery scenarios.

5. Heat Recovery Potential in Advanced Refrigeration Cycles

5.1 Heat Recovery in Transcritical CO_2 Booster Systems

Carbon dioxide (CO_2, R-744) has become a leading natural refrigerant in food refrigeration due to its favorable environmental profile, with zero Ozone Depletion Potential (ODP) and a Global Warming Potential over 100 years (GWP_{100}) of 1 (Bolaji and Huan, 2013). In addition, CO_2 offers attractive thermophysical characteristics for compact and efficient system design. Its high volumetric refrigerating effect enables lower volume flow rates for a given cooling capacity, supporting smaller compressors and reduced pipe diameters. Its relatively high thermal conductivity and vapor density can further enhance heat transfer performance and mitigate pressure drop penalties in suitably designed systems.

A key distinction from many conventional refrigerants is CO_2's low critical temperature (≈31 °C) combined with a high critical pressure (≈74 bar). Under typical ambient conditions, this often leads to transcritical operation, in which heat rejection occurs above the critical temperature and condensation does not take place. Consequently, transcritical CO_2 systems employ a gas cooler rather than a condenser (Maina and Huan, 2015).

From a heat recovery perspective, transcritical CO_2 systems are particularly attractive because compressor discharge temperatures can be high. Instead of rejecting this high-grade heat directly in the gas cooler, a dedicated heat recovery heat exchanger can be installed upstream of the gas cooler to extract useful thermal energy for domestic hot water production or space heating. In practical implementations, CO_2 may be cooled from approximately 120–140 °C down to 20–40 °C within the heat recovery exchanger, depending on operating mode and heat demand (Evans and Foster, 2015). Recovering heat at this location reduces the thermal load on the gas cooler, may allow smaller gas cooler sizing, and can improve overall system performance by reducing irreversibilities and external energy consumption. These advantages make heat recovery a central value proposition of transcritical CO_2 refrigeration, particularly in mild climates where simultaneous refrigeration and heating demand is more common.

In supermarket applications, the CO_2 booster configuration (Figure 18) is widely adopted because it can serve both medium-temperature (MT) and low-temperature (LT) loads within a single system. The MT compressors handle vapor returning from medium-temperature display cases, whereas LT compressors compress vapor from low-temperature evaporators. The LT discharge is typically routed to the MT suction level (or an intermediate level depending on architecture), and the combined flow is ultimately compressed to the high-pressure side and rejected in the gas cooler. A key component in this architecture is the flash tank (liquid receiver), which separates the two-phase refrigerant into liquid and vapor. The liquid is distributed to MT and LT evaporators after expansion, while the vapor phase ("flash gas") is routed to the MT suction level. In advanced configurations, flash gas can be compressed directly to the high-pressure level using parallel compressors, reducing throttling losses and improving efficiency, particularly at higher ambient temperatures.

Performance assessments in warm climates consistently highlight the importance of reducing gas cooler outlet temperature and improving high-pressure control. Pre-cooling strategies upstream of the gas cooler can mitigate efficiency degradation under high outdoor temperatures by lowering the required discharge pressure and reducing compressor pressure ratio. In turn, this improves compressor efficiency and can increase the quantity and usefulness of recoverable heat, provided that heat recovery is coordinated with pressure optimization rather than achieved simply by elevating high-side pressure.

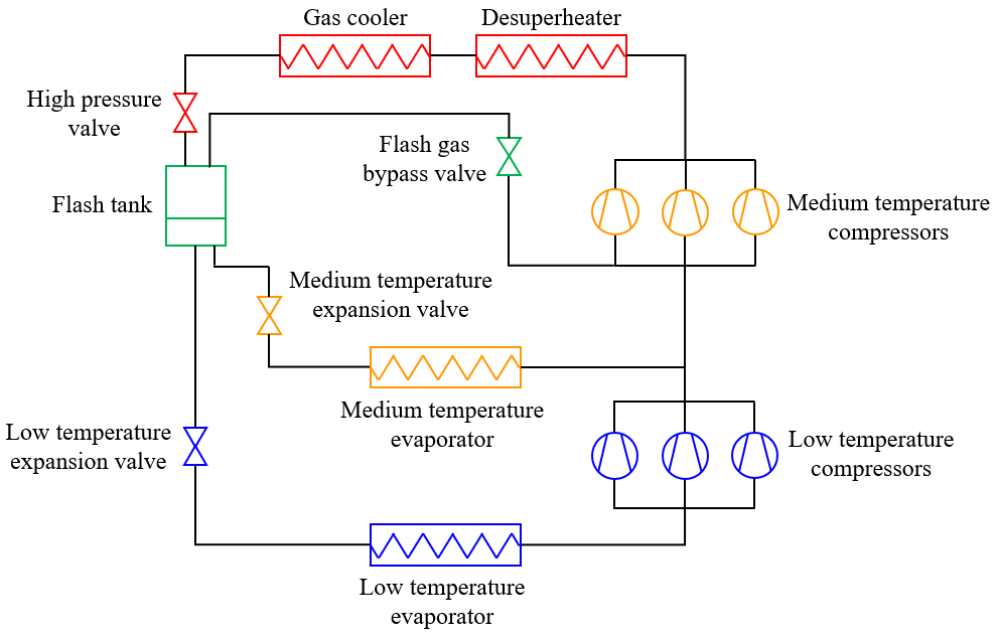


Figure 18. Transcritical CO_2 booster refrigeration system serving medium-temperature (MT) and low-temperature (LT) loads. The flash tank (receiver) separates high-pressure refrigerant into liquid and vapor phases. Liquid refrigerant is distributed to MT and LT evaporators via dedicated expansion valves, while flash gas is routed to the MT suction level (or compressed via parallel compression when implemented). This intermediate pressure separation reduces throttling losses and lowers the pressure ratio of the MT compressors, improving overall system efficiency under varying refrigeration demands.

Additional integration can be realized by incorporating a secondary heat exchanger near the high-pressure side, often upstream of the flash tank, to deliver chilled water for space cooling (Figure 19). In this arrangement, the refrigeration plant can simultaneously supply MT/LT refrigeration and air-conditioning. Effective operation requires coordinated compressor staging and pressure management to satisfy both refrigeration and HVAC cooling demands while maintaining stable receiver conditions. When heat recovery systems are incorporated, these CO_2 booster configurations can independently accommodate the refrigeration, space cooling and heating demands of food retail stores.

This integrated approach can reduce both equipment duplication and installation footprint by consolidating refrigeration, heating, and cooling functions into a single CO_2-based platform. Azzolin *et al*. (Azzolin *et al.*, 2021) reported an integrated transcritical CO_2 booster system installed in a 1500 m² supermarket in Northern Italy that provides refrigeration, air conditioning, and space heating via heat recovery from the high-pressure side. The system incorporates parallel compression and receiver control to enable distinct seasonal operating modes. During winter operation, transcritical conditions may be intentionally maintained to ensure sufficient heat recovery capacity for space heating and domestic hot water, while an external evaporator can be employed to increase heat production. In summer, parallel compression improves air-conditioning capability, although system efficiency can be penalized under very high ambient temperatures if high-pressure control and heat rejection are not optimized.

A supermarket case study in Stockholm, Sweden (Steuer *et al.*, 2023) reported that operational costs were reduced by 29% with an existing heat recovery solution compared to no heat recovery, and that a further 31% reduction was achievable using a control strategy optimized for maximizing heat recovery. In such approaches, recovered heat from the refrigeration system

is prioritized to meet space heating and other thermal demands, which can also reduce municipal hot water consumption where recovered heat replaces conventional water heating.

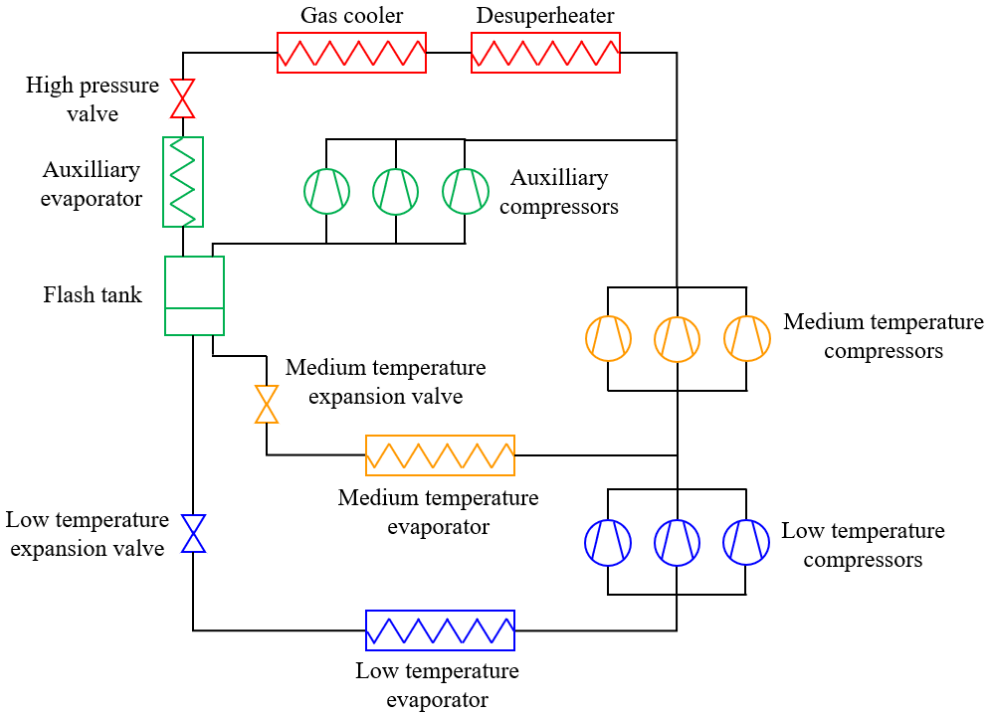


Figure 19. Integrated transcritical CO_2 booster system providing medium-temperature (MT) and low-temperature (LT) refrigeration together with chilled-water production for air-conditioning. An auxiliary evaporator, upstream of the flash tank, serves the air-conditioning load, while auxiliary compressors raise the vapor to the high-pressure level. The system enables simultaneous provision of refrigeration and cooling within a unified CO_2 architecture, requiring coordinated control of flash tank pressure, compressor staging, and high-side pressure to ensure stable and efficient operation.

Karampour and Sawalha (Karampour and Sawalha, 2015) compared conventional and advanced CO_2 booster systems with two-stage heat recovery for integrated space heating, cooling, and refrigeration, highlighting reductions in installation footprint relative to separated systems. Their results indicate that integrated CO_2 systems can outperform conventional HFC-based air-conditioning at moderate ambient temperatures (below approximately 20–25 °C). At higher ambient temperatures, conventional HFC air-conditioning may exhibit higher standalone cooling COP; however, considerations such as environmental impact, regulatory pressure on fluorinated gases, and system compactness can still support CO_2 adoption. Importantly, system-wide efficiency can increase when air-conditioning is integrated appropriately; for instance, at an ambient temperature of 40 °C, the total system COP was reported to increase by 28% relative to the case without air-conditioning, provided that the air-conditioning subsystem COP exceeds unity.

A further techno-economic study of refrigeration-integrated heating and cooling in a UK supermarket (Sarabia Escriva *et al.*, 2022) found that although enhanced heat recovery configurations increase capital cost, operational savings driven by reduced electricity consumption and lower emissions can justify adoption over the system lifetime. Reported annual energy savings for integrated systems relative to conventional HFC-based configurations are often on the order of 6–18%, particularly in regions with high electricity prices or environmental levies. Policy support can play a decisive role in overcoming investment barriers; for example, the UK Renewable Heat Incentive (RHI) has been reported

to offset a substantial share of costs, with subsidies for air-source heat pumps covering up to 89% of generated savings in some assessments. Similarly, internal carbon pricing or a carbon tax on the order of £100/tCO_2e can shorten payback periods to approximately 9–11 years (Sarabia Escriva *et al.*, 2022). These benefits must be evaluated alongside increased system complexity, higher compressor operating loads in integrated modes, and potentially higher maintenance requirements, which should be incorporated into lifecycle cost analysis.

Despite significant progress, transcritical CO_2 booster systems present technical challenges that influence heat recovery performance. High operating pressures require robust component design and careful material selection (Cortella and D'Agaro, 2015). System performance is highly sensitive to gas cooler pressure, receiver pressure, and suction pressure control, particularly under varying ambient conditions and fluctuating refrigeration and HVAC loads (Karampour and Sawalha, 2015). Advanced control strategies are therefore critical, including regulation of gas cooler pressure and outlet temperature to balance refrigeration efficiency against heat recovery requirements. One approach reported in the literature involves stepwise regulation in which the gas cooler is initially operated at high capacity to reduce outlet temperature and enable optimal high-side pressure control, followed by controlled fan speed reduction or bypass operation once an operating optimum is reached, thereby increasing recoverable heat under appropriate demand conditions (Karampour and Sawalha, 2018).

Thermal storage is increasingly deployed to address mismatches between recovered heat availability and demand, particularly because heat recovery potential may peak during summer while space heating demand is minimal. Advanced storage control, such as shifting charging away from peak electricity price periods, can substantially improve operating economics. A UK supermarket analysis reported that storage-informed control could increase cost savings markedly relative to baseline strategies (Sarabia Escriva *et al.*, 2022). In addition, thermal storage can reduce reliance on auxiliary heating during winter operation, for example by mitigating the impact of defrost cycles in air-source heat pumps and smoothing heating delivery under variable conditions (Sarabia Escriva *et al.*, 2022).

5.2 Ejector-Enhanced Refrigeration Systems

5.2.1 Ejector expansion for throttling loss recovery

Ejector-based refrigeration cycles represent a significant advancement over conventional vapor compression systems, particularly in addressing throttling losses associated with expansion valves. In standard vapor compression cycles, expansion through a throttling device is isenthalpic and inherently irreversible, leading to entropy generation and exergy destruction. These throttling losses may account for up to 20% of the compressor input power under certain operating conditions (Santini *et al.*, 2019). By replacing or supplementing the expansion valve with an ejector, part of this lost expansion work can be recovered, thereby improving overall cycle performance.

A schematic layout and pressure–enthalpy (P–h) representation of a typical ejector-based refrigeration cycle are shown in Figure 20. The ejector functions as a static thermally driven device that performs partial compression of the suction vapor without mechanical work input. The ejector consists primarily of a motive nozzle, suction nozzle, mixing chamber, and a diffuser. Its operation involves a sequence of pressure and momentum transformations. High-pressure refrigerant from the gas cooler, referred to as the primary or motive stream, expands through the nozzle. During this expansion, pressure energy is converted into kinetic energy, accelerating the fluid to high velocity while reducing its static pressure. The resulting low-pressure region at the nozzle exit entrains refrigerant vapor from the evaporator through the suction nozzle, which constitutes the secondary stream.

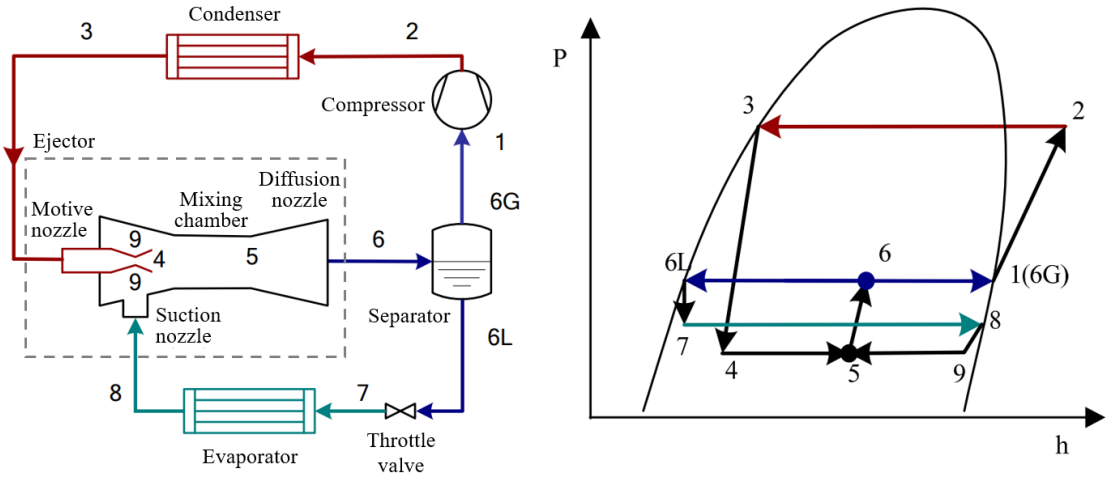


Figure 20. Ejector-based refrigeration cycle: (a) system layout and (b) corresponding pressure–enthalpy (P–h) diagram (Zhang *et al.*, 2015). High-pressure refrigerant leaving the condenser (state 3) expands through the motive nozzle to state 4, converting pressure energy into kinetic energy. The low-pressure vapor from the evaporator (state 8) is entrained through the suction nozzle (state 9) and mixes with the primary stream in the mixing chamber (state 5). The combined flow undergoes pressure recovery in the diffuser (state 6) before entering the separator, where vapor (state 6G) is directed to the compressor suction and liquid (state 6L) is expanded through a secondary throttling valve (state 7) to the evaporator pressure. The compressor raises the vapor pressure from state 1 (6G) to state 2, completing the cycle. The P–h diagram illustrates the reduction in throttling irreversibility and the intermediate pressure lift achieved by the ejector.

Both streams mix within the mixing section, where momentum transfer occurs from the high-velocity primary flow to the lower-momentum secondary flow. This interaction increases the pressure of the combined stream while reducing its velocity. The mixed flow then passes through a diffuser, where kinetic energy is further converted into pressure. The diffuser outlet pressure reaches a level higher than the evaporator pressure but lower than the gas cooler pressure. This partial recompression of the evaporator vapor is the fundamental energy recovery mechanism of the ejector.

Downstream of the ejector, the two-phase mixture enters a separator. Vapor is routed to the compressor suction at an intermediate pressure, while the liquid phase is expanded through a secondary throttling valve to the evaporator pressure. The liquid then absorbs heat in the evaporator, completing the refrigeration cycle.

Two primary thermodynamic effects contribute to COP enhancement in ejector-based cycles. First, the compressor no longer compresses vapor from the lowest evaporation pressure but instead draws vapor from the higher intermediate separator pressure. This reduces the compression ratio and compressor specific work, thereby lowering power consumption (Li and Groll, 2005). Second, the expansion process is effectively split into two stages: part of the pressure reduction occurs in the ejector, where useful pressure recovery is achieved, and the remaining pressure drop across the secondary valve is smaller. This reduces entropy generation and avoids unnecessary enthalpy degradation before the evaporator. Figure 21 illustrates these favorable enthalpy changes, where the increase in refrigeration capacity (Δh_{ref}) and reduction in compressor work (Δh_{comp}) jointly improve cycle COP.

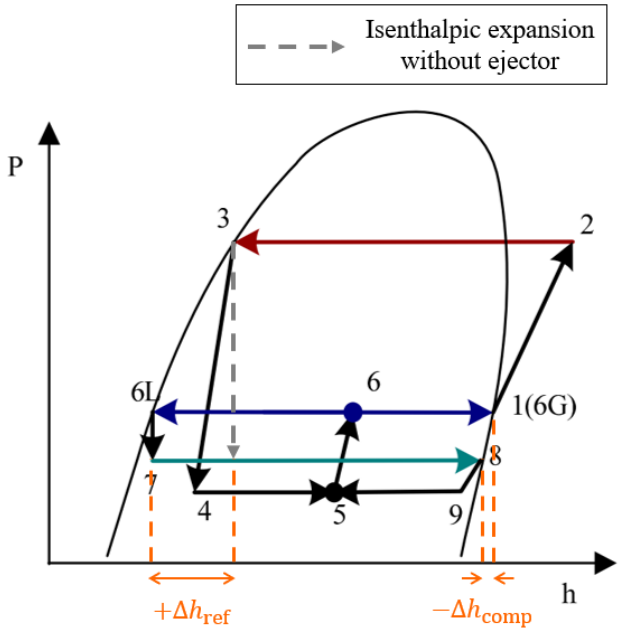


Figure 21. Illustration of favorable enthalpy changes in an ejector-based refrigeration cycle shown on a pressure–enthalpy (P–h) diagram (modified from (Zhang *et al.*, 2015)). The dotted grey arrow indicates the isenthalpic expansion process in a throttling device. The increase in refrigeration effect of ejector-based systems compared to conventional vapor-compression systems, $+\Delta h_{\mathrm{ref}}$, corresponds to the larger enthalpy rise between the evaporator inlet (state 7) and outlet (state 8). The reduction in compressor work, $-\Delta h_{\mathrm{comp}}$, results from compressing vapor from the intermediate separator pressure (state 1 = 6G) rather than directly from the evaporator pressure. These two effects jointly enhance the cycle COP.

Ejector-based cycles are particularly effective in systems exhibiting large pressure differentials between the high-pressure condenser side and the evaporator, where throttling losses are substantial. The technology is especially attractive for refrigerants operating under transcritical conditions, such as CO_2 (R-744), where high expansion ratios amplify throttling irreversibilities (Besagni, Mereu and Inzoli, 2016; Sun, 1999; Cizungu, Mani and Groll, 2001). Although refrigerants such as R152a and R134a have also been investigated, CO_2 has demonstrated particular suitability due to its high operating pressures and associated expansion losses.

The performance of ejector systems depends strongly on geometry and operating conditions. A major challenge lies in matching the ejector's fixed geometry to variable refrigeration loads. Under part-load conditions, reduced evaporator mass flow rates can shift operating points away from the ejector's optimal entrainment ratio, reducing effectiveness. To address this limitation, two principal strategies have been developed, as shown in Figure 22. Adjustable ejectors incorporate a movable needle or variable nozzle geometry that modifies the motive flow area to accommodate varying loads. Alternatively, multiple fixed ejectors of different sizes may be installed in parallel, with combinations activated depending on system demand.

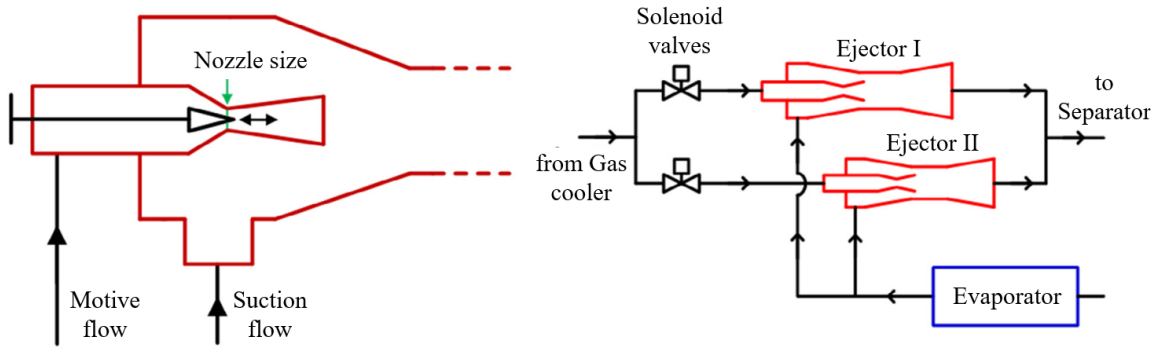


Figure 22. Strategies for maintaining ejector performance under variable load conditions: (a) adjustable ejector with a movable needle regulating the effective motive nozzle area, allowing adaptation of the entrainment ratio to changing operating conditions; (b) multiple fixed-geometry ejectors connected in parallel, where solenoid valves selectively activate individual ejectors to match system load. Both approaches aim to maintain optimal pressure lift and suction flow entrainment across a wide range of refrigeration demands (Elbel and Lawrence, 2016).

Experimental and simulation studies report COP improvements typically ranging from 12% to 30% compared with conventional vapor compression cycles (Besagni, Mereu and Inzoli, 2016). Advanced configurations often combine ejectors with parallel compression and dynamic bypass strategies to maintain stable operation across ambient conditions. For example, ejector bypass may be implemented at low ambient temperatures (e.g., below 15–20 °C), where expansion losses are less significant and ejector benefits diminish.

Beyond basic ejector expansion cycles, numerous hybrid configurations have been explored. These include combined compression–ejector systems designed to reduce compression ratio, multi-stage and multi-evaporator ejector layouts for improved adaptability, and combined ejector–absorption or ejector–adsorption systems that exploit low-temperature waste heat sources. Some designs eliminate mechanical pumps, reducing maintenance requirements while preserving thermodynamic benefits. Reported performance gains in advanced configurations range from 12% to over 50% improvement in COP, depending on system architecture and operating conditions (Besagni, Mereu and Inzoli, 2016).

Despite substantial progress, accurate modeling of the complex two-phase flow phenomena occurring inside ejectors remains challenging. Shock waves, droplet formation, condensation dynamics, and non-equilibrium effects introduce uncertainties in design prediction and scaling. Continued research into computational fluid dynamics (CFD), adaptive geometries, and robust control strategies is essential for optimizing ejector performance and enabling broader commercial adoption.

5.2.2 Heat-driven ejector systems for waste heat recovery

Beyond reducing throttling losses within a single refrigeration cycle, ejector technology can also be devised into heat recovery configurations that utilize waste heat from a primary refrigeration system to drive a secondary cooling process. In such arrangements, recovered thermal energy, typically from the desuperheating region of a vapor compression or transcritical CO_2 system, is used as the driving source for a heat-driven ejector refrigeration cycle. This approach enables additional refrigeration capacity without proportional increases in compressor power, thereby improving overall system efficiency.

A representative recuperative configuration is illustrated in Figure 23. In this coupled system, heat rejected during the desuperheating process of a transcritical CO_2 cycle is transferred to the generator of a secondary ejector refrigeration cycle using refrigerant R152a. The recovered high-temperature heat vaporizes the motive stream of the ejector cycle. This high-pressure vapor expands through the ejector nozzle, entraining the low-pressure secondary stream and enabling partial recompression through momentum transfer.

The secondary stream in the ejector cycle absorbs heat from a subcooler integrated into the primary CO_2 cycle. In this manner, the ejector cycle provides subcooling to the CO_2 refrigerant exiting the gas cooler. After mixing and pressure recovery in the ejector diffuser, the two-phase mixture enters the condenser of the ejector cycle, where heat is rejected to the environment. Downstream of the condenser, the working fluid is separated into a high-pressure liquid stream and a low-pressure stream. The high-pressure stream is pumped back to the generator, while the low-pressure stream expands through a throttling device before re-entering the subcooler.

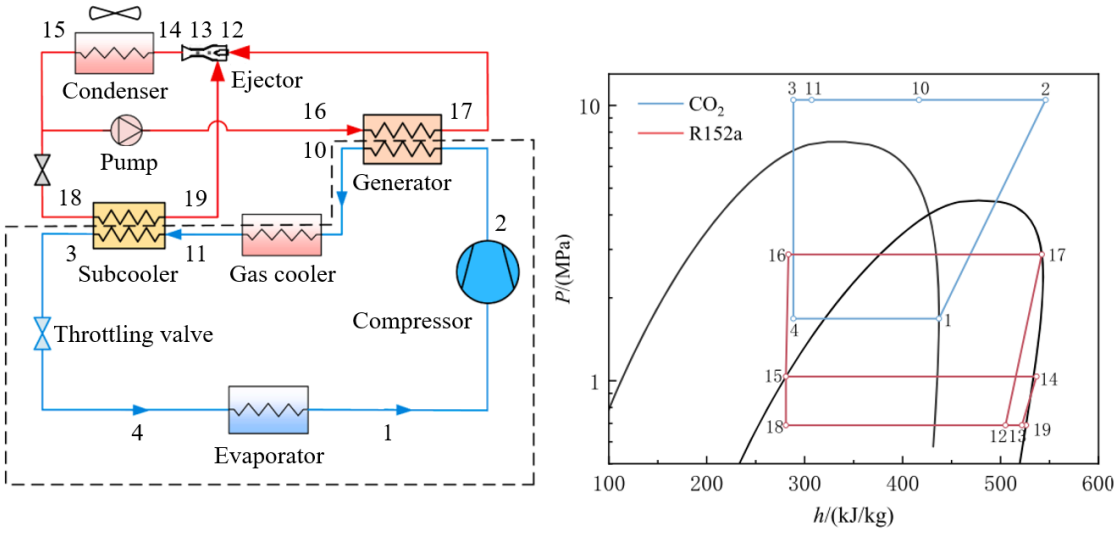


Figure 23. Recuperative heat-driven ejector refrigeration system coupled with a transcritical CO_2 primary cycle: (a) schematic layout and (b) pressure–enthalpy (P–h) diagram (Lv *et al.*, 2025). Waste heat from the desuperheating region of the CO_2 cycle is supplied to the generator of the secondary ejector cycle (R152a). The ejector cycle provides subcooling to the CO_2 refrigerant downstream of the gas cooler, thereby increasing the specific refrigeration effect and improving overall system COP. State numbers correspond to the thermodynamic processes of the CO_2 primary loop and the R152a ejector loop as shown in the P–h diagram.

The thermodynamic benefit of this recuperative configuration arises from the additional subcooling provided to the transcritical CO_2 cycle. Increased subcooling reduces the enthalpy at the expansion valve inlet of the CO_2 system, thereby increasing the specific refrigeration effect without increasing compressor work proportionally. Because the driving energy for the ejector cycle originates from recovered desuperheating heat rather than external mechanical input, the net power penalty is limited to the pump work of the ejector cycle, which is comparatively small.

As a result, the coupled system demonstrates improved overall performance. Reported COP increases range from approximately 12.5% to 19.8%, depending on ambient temperature and operating conditions (Lv *et al.*, 2025). When compared to a configuration in which a dedicated desuperheater operates independently without ejector-driven subcooling, the integrated ejector-based heat recovery system can yield even greater effective COP improvements, particularly

when the additional external power consumption of auxiliary heat rejection devices is considered.

6. Control Systems and Digital Integration

Maximizing the energy, exergy, and economic performance of integrated heat recovery systems requires moving beyond conventional rule-based control strategies. Modern refrigeration plants increasingly operate as multi-energy hubs, simultaneously providing cooling, heating, and potentially grid services. Managing such tightly coupled subsystems demands advanced supervisory control capable of balancing thermodynamic efficiency, economic objectives, and operational constraints. Model Predictive Control (MPC) has emerged as a leading approach for managing the complex dynamics of integrated heating, ventilation, air conditioning, and refrigeration (HVAC&R) systems (Hovgaard *et al.*, 2012). Unlike traditional feedback controllers, MPC predicts future system behavior over a defined horizon and computes optimal control sequences that minimize a specified objective function subject to system constraints. Typical objectives include Coefficient of Performance (COP) maximization, operational cost minimization, or reduction of exergy destruction.

The choice of objective function significantly influences control outcomes. Energy-based MPC strategies commonly aim to maximize COP; however, such approaches may overlook thermodynamic irreversibilities that degrade the quality of energy. In contrast, Exergy Destruction Minimization (EDM)-based controllers use transient exergy destruction as the optimization metric. Comparative studies show that EDM-based control can substantially outperform energy-based MPC. In one case, exergetic efficiency improved by more than 40%, while the energy-based controller exhibited 103% greater exergy destruction in the condenser and 85.4% more in the evaporator (Jain and Alleyne, 2015). These findings highlight the importance of thermodynamically informed control design in advanced refrigeration systems.

From a practical standpoint, Economic MPC has gained particular relevance. Here, the objective explicitly targets operational cost reduction under dynamic electricity pricing. This capability is essential for integration into smart grids and Demand Side Management (DSM) schemes (Coccia *et al.*, 2019). DSM strategies enhance the feasibility of Waste Heat Recovery (WHR) by leveraging the inherent thermal inertia of refrigerated goods or water tanks as temporary storage. Electrical consumption is shifted from peak-price to off-peak periods while maintaining product quality. For example, refrigerated warehouses implementing limited-load strategies achieved energy cost reductions of up to 50% while maintaining storage temperatures below −18 °C, even under controlled temperature fluctuations of up to 5 °C (Altwies and Reindl, 2002).

In supermarket systems, Water Loop Heat Pumps (WLHPs) can serve as both heat recovery devices and flexible thermal storage units. By adjusting temperature setpoints within acceptable comfort limits, additional demand response flexibility can be achieved. A Demand Response (DR) study of an integrated supermarket HVAC, refrigeration, and WLHP system demonstrated annual cost savings of up to 4.67%. The highest savings were obtained by allowing space heating and cooling setpoints to vary by ±2 °C and WLHP auxiliary heater setpoints by ±3 °C, combined with a 200 m^3 water storage tank (Coccia *et al.*, 2019).

Maouris *et al.* evaluated a Refrigeration Integrated Heating and Cooling (RIHC) system in a UK supermarket based on a CO_2 booster cycle (Maouris *et al.*, 2020). Their hybrid recovery controller dynamically adjusted the water tank temperature setpoint (e.g., 45 °C for heating) in response to real-time electricity prices. During high-price periods, the controller reduced the

setpoint to limit heat recovery and avoid costly operation. During low-price periods, it increased the setpoint by elevating condenser pressure (up to approximately 100 bar), reducing gas cooler fan speed to enhance desuperheating heat recovery, and, if necessary, activating a false load evaporator to increase rejected heat. Although these enhanced heat recovery strategies reduce refrigeration COP locally, the combined RIHC and thermal storage configuration achieved overall energy reductions of 17–18% and greenhouse gas emission reductions of 12–13% compared to conventional gas boiler systems (Maouris *et al.*, 2020).

Advanced control also enables participation in external energy markets, such as District Heating Systems (DHS). Huang *et al.* validated a dynamic Heat Recovery Controller for a supermarket in Copenhagen (Huang *et al.*, 2022). The controller continuously optimized water tank setpoints based on the real-time marginal cost gradient between electricity and district heating prices. Field validation demonstrated that prioritizing self-consumption of recovered heat and selling only surplus energy to the DHS achieved cost reductions of up to 93% compared to operation without heat recovery. Similarly, Minko *et al.* simulated a supermarket integrating refrigeration, HVAC, and thermal storage under dynamic economic signals (Minko *et al.*, 2016). Compressor discharge pressure and HVAC damper positions were optimized using forecast inputs such as ambient temperature, occupancy, and real-time electricity prices. Increasing discharge pressure from 50 bar (≈58 °C) to 100 bar (≈130 °C) in transcritical mode significantly increased recoverable heat, while damper control ensured compliance with ventilation requirements. These results demonstrate how predictive algorithms can dynamically balance the trade-off between refrigeration electricity consumption and the thermal value of recovered heat under real-time pricing structures.

Beyond supervisory control, digitalization is transforming diagnostics and optimization in refrigeration systems. Smart sensors combined with machine learning algorithms enable early detection of faults such as refrigerant leakage, preventing efficiency degradation and equipment failure (Wijethunga *et al.*, 2024). Artificial Neural Networks (ANNs) are increasingly used as surrogate models for rapid system evaluation. For example, an ANN model predicted air-cooled condenser backpressure for thousands of operating conditions within seconds, achieving within ±6% accuracy for 93.5% cases (Haffejee and Laubscher, 2021). In another study, ANN-based predictive control of condenser water temperature reduced total cooling energy consumption by 5.6% compared with fixed set-point operation (Kim *et al.*, 2021).

7. Research Opportunities and Future Directions

The continued expansion of the global cold chain, rising energy costs, and increasingly stringent decarbonization policies necessitate a focused research agenda in industrial and commercial heat recovery. Future developments must move beyond simply maximizing the quantity of recovered heat toward enhancing its quality, temperature level, controllability, and economic value. Achieving this objective requires coordinated advances in thermodynamic system design, advanced materials, digital control, and integrated energy management. Priority should be given to research pathways that demonstrate clear routes to commercialization. Technologies capable of retrofitting existing refrigeration infrastructure, minimizing operational disruption, and scaling across diverse food industry segments will likely achieve the greatest near-term impact. At the same time, deeper integration between physical heat recovery systems and digital optimization platforms represents a promising avenue for step-change improvements in performance, flexibility, and profitability.

7.1 Emerging Cooling Technologies

Although vapor compression remains the dominant refrigeration technology, substantial research efforts are directed toward alternative and next-generation cooling concepts with the potential for improved efficiency, lower environmental impact, and enhanced compatibility with waste heat utilization.

Solid-state cooling technologies, including thermoelectric, electrocaloric, elastocaloric, and magnetocaloric systems, offer compelling long-term opportunities (Takeuchi and Sandeman, 2015; Ismail, Yebiyo and Chaer, 2021; Qian *et al.*, 2023; Zhou *et al.*, 2025; Xu *et al.*, 2024; Torelló and Defay, 2022; Klinar *et al.*, 2024; Zhang *et al.*, 2026). These technologies eliminate conventional compressors and refrigerants, reducing leakage risks and environmental impact. Moreover, certain caloric technologies (e.g., elastocaloric systems) may enable novel heat recovery configurations due to their direct thermo-mechanical coupling. However, material limitations, such as fatigue resistance, thermal conductivity, and limited temperature span, currently constrain system-level efficiency and reliability.

In gas-based alternatives, thermoacoustic cooling represents another promising direction (Ismail, Yebiyo and Chaer, 2021; Fazli *et al.*, 2024; Xu and Luo, 2022; Jaworski and Mao, 2013; Zink, Vipperman and Schaefer, 2009). By utilizing acoustic pressure oscillations to induce compression and expansion processes, thermoacoustic systems avoid complex moving components such as mechanical compressors and expansion valves. This design simplicity could enhance durability and reduce maintenance requirements. Nevertheless, efficiency limitations, acoustic driver losses, and heat exchanger optimization remain key technical barriers.

A critical research gap across emerging cooling technologies lies in system integration. When applied to food refrigeration, these technologies require reconfiguration of conventional system layouts, including reconsideration of evaporator placement, heat rejection interfaces, and heat recovery integration. Early-stage research must therefore incorporate heat recovery considerations into component and architecture design rather than treating them as add-on features.

Beyond integration challenges, several priority research gaps must be addressed:

- *Scalability and upscaling*: Most emerging cooling technologies demonstrate promising laboratory-scale performance, yet translating these results to kilowatt- and megawatt-scale applications introduces nonlinear thermodynamic and mechanical challenges. Research is

needed on modular system architectures, reliable scaling laws, and hybrid configurations that maintain efficiency at commercial capacities.

- *Material durability and fatigue*: Caloric materials and other solid-state components undergo repeated thermo-mechanical cycling. Long-term stability under millions of load cycles, fluctuating temperatures, and industrial stress conditions remains insufficiently characterized. Accelerated lifetime testing and predictive degradation models are essential.
- *Cost reduction and manufacturability*: Many advanced systems rely on specialized materials or precision fabrication. Research should focus on design-for-manufacturing strategies, alternative low-cost materials, modularization, and supply chain optimization to achieve competitive capital costs.
- *Standardization and certification frameworks*: Unlike vapor compression systems, emerging technologies lack mature performance standards and regulatory pathways. Development of standardized testing procedures, safety guidelines, and reliability benchmarks is necessary to reduce perceived risk and accelerate market adoption.
- *Lifecycle sustainability and circularity*: Environmental benefits must be assessed across the full lifecycle, including material extraction, manufacturing energy intensity, recyclability, and end-of-life recovery. Integrating lifecycle carbon assessment into early-stage design is critical for ensuring genuine sustainability gains.

7.2 Digitalization and Control

The integration of digital technologies into hybrid refrigeration and heat recovery systems represents a critical enabler for next-generation performance optimization. Advanced sensing, data acquisition, and real-time monitoring platforms allow continuous characterization of cooling demand, heat recovery potential, electricity pricing, and emissions intensity. Such data-driven infrastructure forms the foundation for intelligent supervisory control and system-level optimization. Transitioning from static set-point operation to adaptive, predictive control strategies significantly enhances heat recovery performance. By incorporating model-based or data-driven feedback loops, system operation can be dynamically optimized according to refrigeration load profiles, heat demand fluctuations, and time-varying electricity tariffs. Predictive algorithms based on Model Predictive Control (MPC) enable anticipatory adjustments in compressor staging, condenser pressure, water tank temperature, and heat recovery flow rates (Hovgaard *et al.*, 2012; Jain and Alleyne, 2015). This improves thermodynamic efficiency while reducing operational cost and exergy destruction.

Further benefits arise when refrigeration facilities are integrated into smart grid ecosystems. Bidirectional communication with the power grid allows facilities to exchange real-time information on energy production, consumption, carbon intensity, and demand flexibility. Such integration enables participation in demand response programs, peak shaving strategies, and ancillary grid services. Refrigeration systems, owing to their inherent thermal inertia and storage capacity, can function as flexible energy assets rather than passive electricity consumers.

Digital integration also supports transparency and accountability in sustainability reporting. Real-time tracking of energy efficiency metrics, greenhouse gas emissions, and recovered heat utilization enables data-driven compliance with regulatory frameworks and corporate decarbonization targets. In the longer term, the development of digital twins, virtual replicas of refrigeration systems updated with live operational data, may further enhance lifecycle optimization, predictive maintenance, and investment decision-making.

7.3 Economic Barriers and Policy Imperatives

Despite substantial technical progress, the widespread adoption of advanced heat recovery technologies remains constrained by non-technical barriers, particularly financial risk, capital intensity, and institutional uncertainty. While many configurations, such as Organic Rankine Cycle (ORC) systems, advanced heat pumps, and integrated CO_2 architectures, demonstrate annual energy savings in the range of 6–18% (Baxter, 2003; Sarabia Escriva *et al.*, 2022; Sengupta *et al.*, 2023), their high upfront capital cost often limits deployment in commercial food refrigeration facilities.

Investment decisions in the food sector are typically driven by short payback thresholds and risk-averse capital planning. Even when lifecycle savings are favorable, longer payback periods can deter adoption. Consequently, the economic question is not solely whether heat recovery increases efficiency, but whether the incremental performance gain justifies the additional capital expenditure and operational complexity. This tension is particularly pronounced in retrofit scenarios, where integration into existing infrastructure may require downtime, system redesign, or compatibility adjustments.

Policy mechanisms therefore play a decisive role in accelerating diffusion. Carbon pricing, investment subsidies, low-interest financing, tax incentives, and renewable heat support schemes can substantially improve project economics. For example, analyses of the UK Renewable Heat Incentive (RHI) indicate that subsidy mechanisms can shorten payback periods from approximately 16 years to 9–11 years under certain configurations (Sarabia Escriva *et al.*, 2022). Similarly, the introduction of meaningful carbon pricing signals can shift investment decisions by internalizing environmental externalities and improving the competitiveness of low-carbon technologies. Beyond direct financial incentives, regulatory frameworks must also evolve to support decentralized and integrated energy architectures. Policies that encourage embedded generation, mandate emission reductions in food manufacturing and retailing, and facilitate participation in district heating networks or demand response markets can create additional revenue streams for recovered heat. Clear standards, streamlined permitting processes, and harmonized grid-interconnection rules further reduce uncertainty and transaction costs.

Future research should therefore not only address thermodynamic optimization but also develop robust techno-economic models that incorporate dynamic electricity pricing, carbon taxation scenarios, financing structures, and policy incentives. Integrating economic modeling into early-stage system design will be essential for identifying configurations that are not only technically efficient but also commercially viable.

7.4 Knowledge Gaps and Perceived Risk

In addition to financial constraints, limited technical awareness and perceived technological risk represent significant barriers to the adoption of advanced heat recovery solutions. Even when techno-economic analyses indicate favorable payback periods, decision-makers in the food industry may hesitate to invest in unfamiliar technologies due to uncertainty regarding performance reliability, integration complexity, and long-term maintenance requirements.

Survey evidence from the UK food sector illustrates this challenge: approximately 36% of engineers reported viewing heat pump technology as “risky” or expressed uncertainty about its benefits, despite documented European payback periods of 2–5 years in several applications (Law, Harvey and Reay, 2013). This discrepancy highlights a critical gap between demonstrated technical feasibility and industry confidence. The diffusion of advanced heat recovery technologies is therefore constrained not only by economic metrics but also by

informational asymmetry and limited internal expertise. Many industrial facilities lack in-house specialists capable of evaluating complex thermodynamic systems, advanced control architectures, or integrated energy platforms. As a result, technology selection may default to conventional solutions perceived as lower risk, even when suboptimal from an efficiency or decarbonization perspective.

To address this challenge, the development of structured decision-support tools and knowledge-based systems (expert systems) is essential. Such platforms can decentralize expertise by providing data-driven guidance on system sizing, techno-economic comparison, risk assessment, and lifecycle performance. Coupling these tools with validated field-performance databases and standardized performance metrics would further reduce uncertainty and increase investor confidence. In parallel, legislative and regulatory clarity is necessary to reduce institutional risk. Establishing standardized communication protocols, cybersecurity frameworks, and interoperability guidelines for smart-grid-enabled refrigeration systems can facilitate broader adoption. Enabling new market actors, such as demand response aggregators or decentralized heat suppliers, through clear regulatory pathways will be particularly important for small and medium-sized enterprises, which often lack the resources to navigate complex energy markets independently.

8. Conclusion

Heat recovery from commercial food refrigeration systems represents a highly effective pathway toward improving energy efficiency, reducing operating costs, and supporting decarbonization across the food cold chain. As demonstrated throughout this chapter, refrigeration systems inherently reject substantial quantities of thermal energy, which can be captured and utilized through a range of direct and indirect recovery methods. Established technologies such as desuperheaters, condenser heat reclaim, heat pumps, and thermal energy storage already enable significant performance improvements in practical applications. In parallel, advanced system architectures, including transcritical CO_2 booster systems and ejector-based cycles, offer enhanced heat recovery potential by reducing thermodynamic losses and enabling integrated provision of cooling, heating, and air-conditioning services.

Beyond hardware solutions, the integration of intelligent control strategies and digital technologies plays a crucial role in maximizing the value of recovered heat, particularly under variable load and market conditions. However, despite strong technical and economic potential, widespread adoption remains constrained by system complexity, upfront investment costs, and knowledge gaps within industry practice. Future research should therefore focus on system-level integration, scalable and cost-effective designs, and robust control frameworks that can operate reliably under real-world conditions. In this context, the combination of advanced refrigeration technologies, thermal storage, and digital optimization is expected to transform refrigeration systems into multifunctional energy hubs, contributing significantly to the transition toward more sustainable and resilient energy systems.

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