

The design and analysis of an integrated heat supply system for large industrial potato fryers

Utilizing a steam heat pump cycle with frying vapor as primary heat source

D. Berghout



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Acknowledgement

My journey in Delft began with an HBO Bachelor in Mechanical Engineering at The Hague University of Applied Sciences. After completing this programme in four years, I decided to continue my academic journey. At that point, I hesitated between pursuing a master's degree in management or a technical master's programme, but eventually followed the technical path. This decision led me to start the bridging programme for the Master's in Mechanical Engineering. After completing the bridging programme, I continued with the Master's programme, where I further deepened my knowledge of energy-related concepts. This academic path eventually resulted in the topic of my master thesis: The design and analysis of an integrated heat supply system for large industrial potato fryers. Those who know me well will recognise that my enthusiasm for potato products made this topic a particularly fitting and enjoyable choice. As this thesis marks the final chapter of my academic journey spanning more than four years at TU Delft I would like to take this opportunity to express my sincere gratitude to everyone who supported me throughout this process.

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Abstract

Industrial potato frying processes require substantial amounts of thermal energy, of which a dominant share is associated with moisture evaporation from the product. This energy leaves the system in the form of hot frying vapor and is typically rejected to the environment, while process heat is commonly supplied by fossil-fired steam boilers. Recovering and upgrading this latent heat therefore represents a significant opportunity to reduce external energy demand and improve the sustainability of industrial frying operations. This thesis investigates the feasibility of recovering frying vapor heat and returning it to the fryer as high-pressure steam using a closed-loop mechanical vapor recompression (MVR) system with water as the working fluid.

A structured methodology was applied, combining literature study, steady-state thermodynamic modelling, equipment sizing, and dynamic simulation. The energy distribution within the frying process was first analysed, demonstrating that evaporation-related heat dominates the total thermal demand and that the recoverable heat potential depends strongly on the vapor dew point. Based on these findings, a complete system configuration was developed consisting of a vertical shell-and-tube condenser, a flash tank, and a three-stage water-injected screw compressor. The heat exchangers were sized to ensure sufficient heat-transfer capacity, reliable condensate drainage, and robustness against fouling, while maintaining realistic pressure drops and flow velocities. The flash tank was designed to accommodate the large specific volume of low-pressure vapor and to provide adequate thermal and mass buffering capacity. Water injection in the compressor limits discharge temperatures and enables high pressure ratios while maintaining near-saturated compression.

For a representative industrial fryer with a heat load of 8161 kW and a vapor dew point of 87.5 °C, the reference configuration recovers approximately 4045 kW of latent heat from the frying vapor, which is upgraded through compression to deliver 6572 kW of useful heat back to the fryer, with the remaining 1589 kW supplied externally. Under these conditions, the system achieves an MVR coefficient of performance of 2.63, demonstrating that a substantial fraction of the process heat demand can be covered through internal heat recovery under realistic operating conditions. A sensitivity analysis shows that the vapor dew point, flash tank temperature, and condenser temperature approach strongly influence the recoverable heat and compression requirements, with the vapor dew point proving particularly critical: a reduction from 90 to 85 °C lowers the recoverable heat by nearly 1 MW. An improved configuration incorporating an additional vapor compression heat pump enables further cooling of the recovered condensate, allowing additional low-grade heat to be absorbed and upgraded, such that the fryer heat demand can be fully met without external steam. The resulting temperature levels are well suited for ammonia (R717) as a natural refrigerant, offering favourable thermodynamic performance while requiring appropriate safety measures.

To assess operability under realistic production disturbances, a dynamic model of the heat recovery system was developed using time-dependent mass and energy balances. Components with significant thermal inertia, including the fryer oil circuit and the flash tank, were modelled dynamically, while fast-responding components such as the compressor and heat exchangers were represented by steady-state relationships. Dynamic simulations were performed to evaluate system behaviour during changes in production rate and to identify suitable control strategies. Feedback control restores steady operation following load changes but results in relatively slow settling and noticeable transient deviations. Introducing feed-forward control based on production throughput allows disturbances to be anticipated, significantly reducing overshoot and accelerating convergence to the new operating point. The results show that combined feed-forward and feedback control provides superior transient performance and enables stable maintenance of fryer oil temperatures and system pressures, which is essential for preserving product quality and ensuring reliable industrial operation.

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1

Introduction

The Netherlands ranks as the world's tenth-largest producer of potatoes and holds the third-largest processor position [1]. By 2024, Dutch processors had converted around 4 million tonnes of potatoes into 2.2 million tonnes of finished products [2], of which approximately 85% consists of pre-fried goods preserved by freezing or cooling. The potato processing industry consumed 10,296 TJ of primary energy in 2017 [3], with the frying step representing the highest energy use per tonne of product, followed by steam generation for blanching and peeling. This highlights the industry's significant heat demand and the importance of effective heat management.

Frying vapor heat recovery systems have been implemented in various forms throughout the industry; the primary focus has often been on utilizing the recovered heat for other process steps, such as water heating for blanching or air heating for HVAC [4]. However, the direct reuse of this heat in the fryer itself has not been extensively explored, despite advancements in heat pump technology that now enable higher temperature lifts and supply temperatures [5]. As such, recovering and recycling heat from the fryer exhaust back into the frying process presents a promising opportunity for energy savings and CO₂ emissions reduction.

This report investigates a concept for capturing the latent heat from frying vapor, a mixture of oil fumes, steam, and air, and using it to generate low-pressure steam via a heat recovery unit. The generated steam is subsequently compressed using a steam compressor to reach a pressure and temperature suitable for reintegration into the fryer.

This configuration in essence represents a basic compression heat pump, depicted in Figure 1.1, using water as the refrigerant. The waste heat source is the frying vapor, which is utilized to generate low-pressure steam. This steam is subsequently compressed to a higher pressure and temperature to supply process heat, in this case, back to the fryer. After heat delivery, the steam is expanded through the expansion valve to complete the cycle. The heat delivered corresponds to the sum of the recovered waste heat and the mechanical work supplied by the compressor, as shown by the arrow in Figure 1.1. Closing this thermal loop can significantly improve the process because the energy is currently supplied by a gas-fired heater or by electricity used to heat a thermal fluid.

With rising energy costs and the increasing urgency to reduce CO₂ emissions, the potato processing industry faces the challenge of lowering its dependence on gas-fired heaters. Recovering the latent heat contained in frying vapors and returning it to the process offers a promising pathway, but its practical implementation remains unclear. The gap between the availability of recoverable heat and its effective reuse within the core frying step forms the basis of this study. Consequently, this thesis is driven by the following research question.

How can a heat pump system be implemented in a large industrial potato fryer to supply the required energy demand?

This report consists of seven chapters that together provide the foundation necessary to answer the research question.

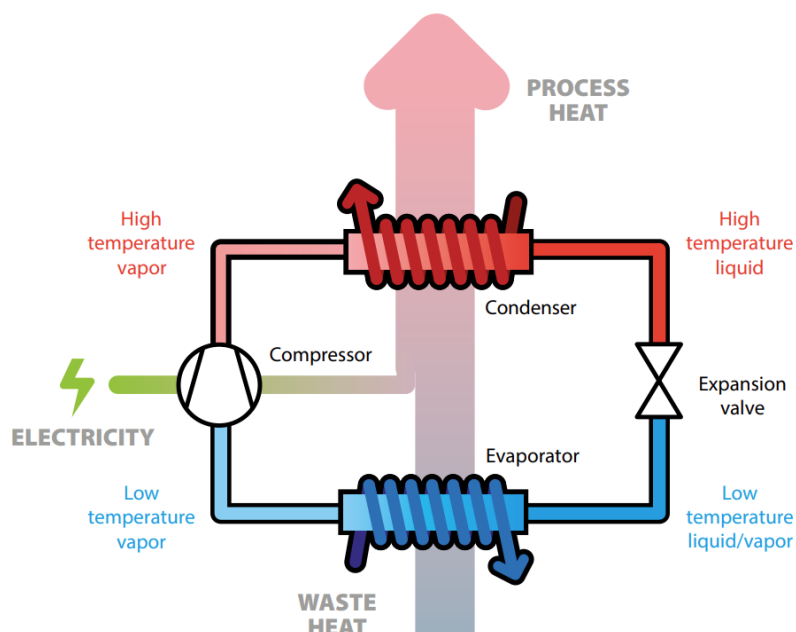


Figure 1.1: Basic (working principle of a) vapor compression heat pump. [6]

Chapter 2 presents an overview of the industrial potato frying process. It describes the process characteristics and the generation of frying vapors that determine the potential for heat recovery. Furthermore, available methods for recovering this heat are discussed. The basic working principle of a heat pump cycle is introduced, followed by an assessment of suitable working fluids, including steam and alternative refrigerants, together with their applicable operating temperature ranges. Finally, the types of steam compressors used to raise low-pressure steam to higher pressure levels are reviewed.

Chapter 3 presents the methodology used for the design, modelling, and analysis of the proposed system. It first introduces the methodology for the steady-state design and analysis, including the overall modelling approach, the formulation of component models, and the definition of control and performance metrics. The chapter then describes the equipment sizing methodology, detailing the design of the vertical flash tank and the general sizing considerations for the heat exchangers. Finally, the development of the dynamic simulation model is presented, including the general modelling approach and solution strategy, the formulation and verification of mass and energy balances, the definition of thermodynamic properties, and the component models used for the dynamic simulations.

Chapter 4 presents the conceptual and general design of the proposed heat recovery and heat delivery system, in which the components introduced in Chapter 2 are integrated into a single installation. Rather than focusing on model derivation, this chapter primarily discusses and evaluates steady-state results obtained for the proposed system configuration. The recoverable heat from the fryer and the amount of heat that can be supplied back to the process are quantified, and several design iterations are assessed to improve overall system performance. A sensitivity analysis is included to evaluate the influence of key parameters on the system behaviour.

Chapter 5 focuses on the results related to component sizing based on the steady-state simulations. This includes the sizing and performance evaluation of the two condensers, the flash tank, and their role within the overall system. In addition, the thermal inventory of the system is analysed to provide insight into the distribution of thermal inertia and the energy required during start-up and transient operation.

Chapter 6 presents the dynamic simulation results of the integrated system, building upon the component sizes and system configuration defined earlier. The dynamic response of the system to variations in production rate is analysed, and the relevant process variables and actuators are identified. Feedback and combined feedback–feedforward control strategies are then evaluated to assess the system performance under transient operating conditions.

Chapter 7 concludes the thesis by summarizing the main findings and drawing conclusions from the results presented in Chapters 2 through 6, and provides recommendations for future work.

2

Background Information

This chapter provides the technical background required to evaluate the potential for heat recovery in the industrial production of French fries. The chapter first outlines the main stages in frozen French-fries production. Building on the frying step of this process description, it describes the associated mass and energy balances and analyzes the characteristics of the frying vapors that determine the potential for heat recovery. Furthermore, the available methods for recovering this heat are discussed. The basic working principle of a heat pump cycle is then introduced. In addition to steam as a working medium, alternative refrigerants are examined together with their operating temperature ranges. Finally, the types of steam compressors used to raise low-pressure steam to high-pressure steam are reviewed.

Together, these sections provide the thermodynamic and technological foundation for the general design presented in Chapter 4.

2.1. Frying Process

A brief introduction to the potato production process is presented first [7], followed by an exploration of the potential for heat recovery, utilizing a steam system.

2.1.1. French fries production process

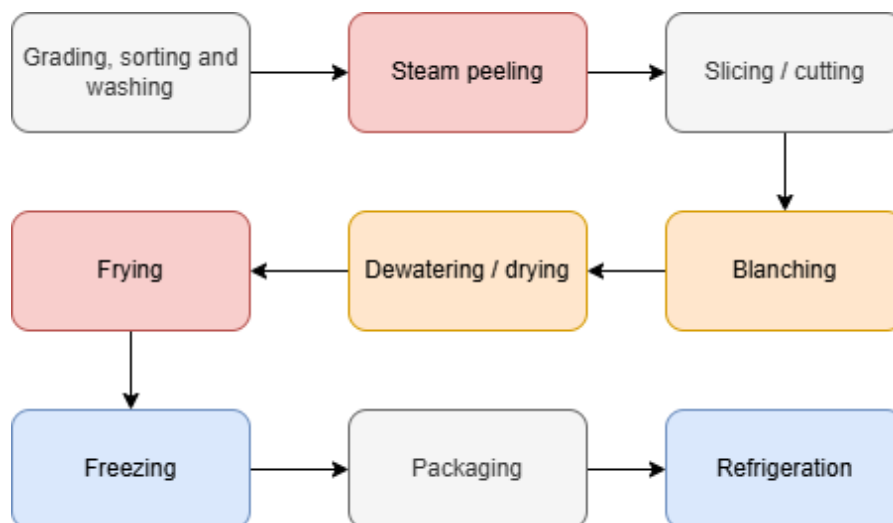


Figure 2.1: Process steps involved in the industrial production of frozen French fries

Grading and Sorting: This step removes damaged or unsuitable potatoes and sorts the rest by size, color, or quality. Larger potatoes are used for French fries or crisps; smaller ones may be used for flakes. In the Netherlands, sorting is mostly automated using technologies such as screen graders, vibrating screens, and optical sorters. Optical sorters enable detailed inspection and are integrated at multiple points in advanced facilities.

Washing and Cleaning: Dirt, stones, insects, and plant residues are removed before further processing. This is often done in multiple stages, potentially using water flumes to combine cleaning and transport. In some cases, antimicrobial treatments are applied.

Peeling: The potato skin is removed to enhance product quality. Steam peeling is the most common method, using high-pressure steam (180–200 °C), followed by rapid depressurization. Other methods include mechanical (knife), abrasion, and chemical (caustic) peeling. Steam peeling is efficient but requires 150–430 kJ of steam/kg, whereas mechanical peeling requires 14 kJ of electricity/kg. Peels and trimmings are typically sold as animal feed.

Cutting/Slicing: Potatoes are cut to final shapes using high-speed water knives or blade systems. The method depends on the product (e.g., thin slices for crisps, ridges for ridged crisps).

Post-Slicing Sorting: After cutting, products are sorted by size and quality. Optical sorters remove visual defects, while length and sliver sorters separate by size. This is typically a mechanical, motor-driven process.

Blanching: Potatoes are briefly immersed in hot water (60–95 °C) to deactivate enzymes and bacteria, preserving quality and reducing internal pressure.

Drying: During the drying process, the moisture content is reduced, and the strips are prepared for frying, where final texture and color are developed. Before entering the fryer, the French fries are transported on an equilibration belt to ensure uniform moisture distribution.

Frying: This is the primary cooking step. Products are dried, then passed through hot oil (160–190 °C). Frying time varies by product (30 seconds to 3 minutes). Energy use ranges from 0.8 to 2.8 MJ/kg and may account for up to 80% of the total process energy.

Cooling: Cooked products are cooled to prepare for storage or packaging. This step has low energy intensity, contributing only about 3% of the total electricity use.

Freezing: Products are quickly frozen (–15 to –40 °C) to preserve quality. This is done using mechanical blast freezers in freezing tunnels.

Packaging: Products are packed in bags, boxes, and pallets using automated equipment. This step accounts for about 3% of the total electricity use and involves electric motors and pneumatic actuators.

2.1.2. Characteristics of raw and fried products

The chemical composition of raw potatoes, potato chips, and French fries is presented in Table 2.1. As shown, there are noticeable differences in water content, primarily due to the production process. Frying is one of the key steps during which a significant amount of water is evaporated.

In food materials, specific heat is influenced by their composition. As the moisture content decreases, the specific heat generally decreases as well. [8]

Table 2.1: Typical Composition of Raw Potatoes, French Fries, and Potato Chips

Composition	Raw Potato	French Fries	Potato Chips	Source
Water (%)	79.8	44.7	1.8	[9]
Protein (%)	2.1	4.3	5.3	
Oil (%)	0.1	13.2	39.8	
Carbohydrate (%)	17.1	36.0	50.0	
Ash (%)	0.9	1.8	3.1	
Specific heat [kJ/kg K]	3.65	2.67	1.54	[10]

In addition to the composition values presented in Table 2.1, it is important to note that the water content of fries and chips at the time they enter the fryer may differ from that of raw potatoes. This variation results from pre-frying processes such as blanching and hot-air drying, which reduce the initial moisture content. Furthermore, during the production of frozen French fries, the moisture content remains considerably higher than that of the final fried product, as indicated in table 2.1. [11]

2.1.3. Frying oil and vapor characteristics

In industrial frying processes, a significant amount of energy is transferred into the vapor stream emitted from the fryer, referred to as frying vapor. This stream is primarily composed of water vapor generated through evaporation of moisture from the product, accompanied by air drawn in as part of the fryer's draft system.

Table 2.2 presents the characteristics of the frying vapor. The dew point is directly related to the air–steam ratio in the vapor mixture, which reflects the extent of water evaporation and air dilution. This ratio, and consequently the dew point, is influenced by the specific fryer design.

Table 2.2: Frying Vapor Characteristics from Various Sources

Vapor Temp. (°C)	Dew Point (°C)	References
130	92.5	[12]
120	-	Kuipers Food Technology
130	85.0	SenterNovem Case Study

Energy delivered to fryer

The thermal energy required to maintain the frying process is supplied by the hot oil entering the fryer. As oil flows through the system, it transfers heat to the product and exits at a lower temperature. This cooler oil is typically reheated via a heat exchanger before recirculation. While minor oil losses occur due to product absorption and fines, these are generally small and often not explicitly accounted for. Typical industrial oil inlet and outlet temperatures are shown in Table 2.3.

Table 2.3: Typical Temperature Ranges for Frying Oil in Industrial Systems. [13] [14]

Stream	Temperature Range (°C)	Notes
Hot oil in	177 – 190	Oil enters fryer
Cold oil out	160 – 174	Oil leaves fryer and is reheated

Energy flows largely depend on the difference in water content and temperature of the potato input and output. The difference in water content between raw potatoes entering the fryer and the fried product exiting the fryer indicates the amount of water that evaporated during frying. Since the latent heat of vaporization for water is significantly higher than its specific heat capacity, this evaporation process constitutes a major component of the total energy input.

Frying oil characteristics

The thermophysical properties of the frying oil used in this study are summarised in Table 2.4. The density, specific heat capacity and viscosity correspond to typical values at a temperature of 180°C. These properties are used as input for the thermal modelling of the fryer and oil circulation system throughout the simulations.

Table 2.4: Frying oil thermophysical properties at 180°C.

Property	Value	Units	References
Density	811	kg/m ³	[15] [16]
Specific heat	2.36	kJ/(kg·K)	[15] [17]
Viscosity	0.0020	Pa·s	[18]

2.1.4. Product Quality and Frying Conditions

The quality of fried food products is determined by several key properties, including moisture content, color, oil content, flavor, texture, yield, nutritional value, and shelf-life stability [19]. Together, these properties define the final product quality and consumer acceptance.

During frying, product quality is mainly influenced by the oil temperature, the product residence time, and the type of fryer used. Oil temperature plays a central role, as it controls the rate of water evaporation, crust formation, browning reactions, and oil uptake [20]. Residence time determines how long the product is exposed to these conditions and therefore affects both internal structure and surface properties.

The type of fryer affects the uniformity of the thermal conditions during frying. Batch fryers typically operate under nearly constant oil temperatures, because of the large oil/product ratio, whereas continuous industrial fryers often show temperature gradients along the fryer length [21]. In large-scale industrial operations, such temperature variations are common and can lead to product quality variability if not properly controlled.

These findings highlight the importance of stable and well-controlled oil temperatures in industrial frying. When introducing alternative or sustainable heat delivery systems, it is essential that the temperature levels and stability in the fryer remain unchanged to ensure that product quality and consistency are maintained.

2.2. Heat recovery

In the case of an air-steam mixture, the dew point depends on the amount of water vapor present in the air, which is quantified by the *humidity ratio*. The humidity ratio is defined as the mass of water vapor per kilogram of dry air. A higher humidity ratio indicates a greater water content, which in turn corresponds to a higher dew point. Thus, the dew point provides insight into the thermal recovery potential of the exhaust stream.

In the context of frying vapor, the condensation process becomes more complex due to the presence of air, which is a non-condensable gas. Several important characteristics must be considered when steam condenses in the presence of non-condensable gases. First, condensation is non-isothermal. It does not occur at a single temperature; as steam gradually condenses, the vapor composition changes and the dew point shifts continuously. Second, there is diffusion resistance. The steam must diffuse through a layer of non-condensable gas before it can reach the cold surface and condense [22]. This additional resistance slows down the overall heat and mass transfer. As a result, the QT diagram for the air-steam mixture differs from that of pure steam because of continuous changes in vapor composition and dew point during condensation. Figure 2.2 shows the condensation curve of humid air with an initial temperature of 130 °C and varying dewpoints of 90, 85, and 80 °C. As observed, the amount of sensible heat removed from the superheated humid air is relatively small compared to the heat released during condensation. This figure also demonstrates how changes in dew point affect the energy profile of the vapor mixture.

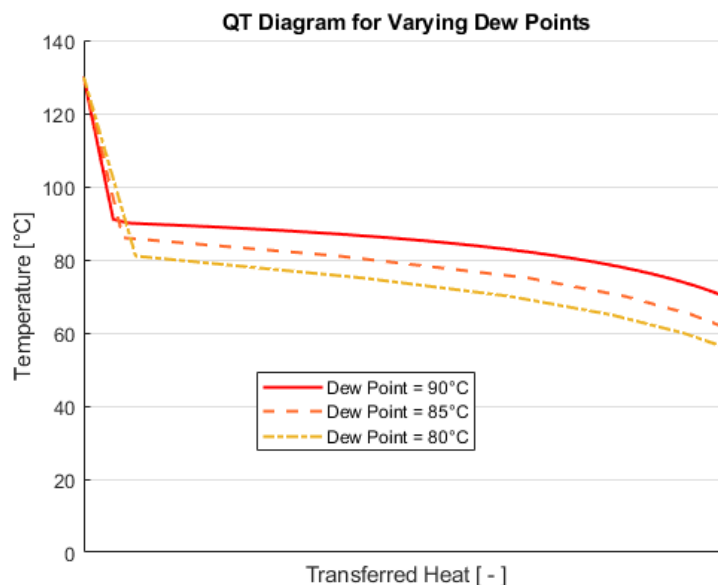


Figure 2.2: QT diagram for humid air with varying dewpoints. The data for this plot was obtained using CoolProp at atmospheric pressure.

Heat recovery from frying vapor can be achieved using two main types of condensers: direct-contact condensers and indirect condensers, both of which are shown in Figures 2.3 and 2.4. In direct-contact systems, the vapor comes into physical contact with the cooling medium, enabling simultaneous heat and mass transfer. This configuration is often used when both condensation and cleaning of the vapor stream are required. Indirect condensers, on the other hand, separate the vapor from the cooling medium via a solid heat-transfer surface. This prevents the two fluids from mixing while still enabling efficient heat transfer. Both approaches are used in industrial processes to recover thermal energy from frying vapors. Frying vapor and oil streams are highly susceptible to fouling due to the presence of aerosolized fats, fine food particles, and thermally degraded compounds. As a result, fouling and cleanability become relevant factors in selecting an appropriate heat exchanger for heat recovery and delivery for the frying processes. When moderate-to-severe fouling is expected, two exchanger types are typically considered: shell-and-tube heat exchangers and gasketed plate heat exchangers.[23]

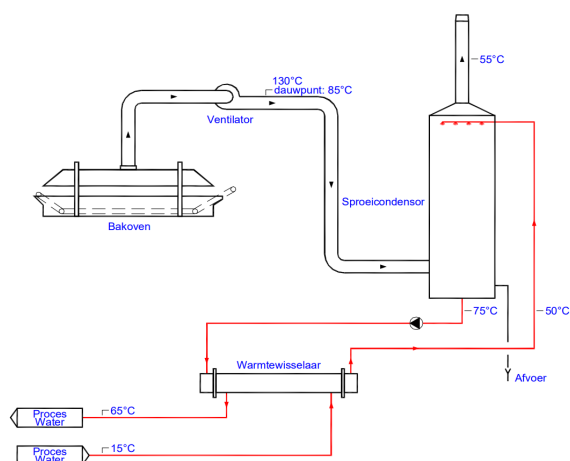


Figure 2.3: Heat recovery setup for frying vapors using a spray condenser.

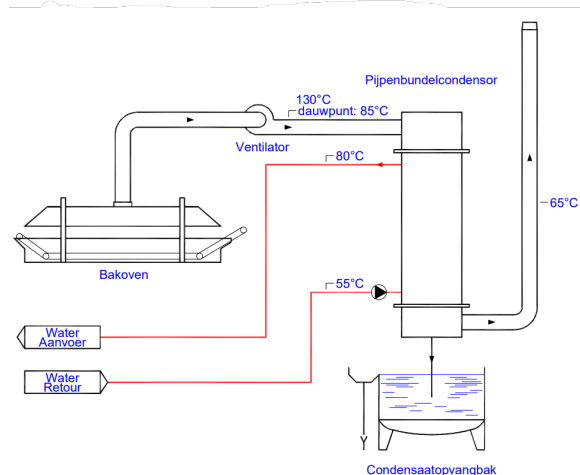


Figure 2.4: Heat recovery setup for frying vapors using a vertical shell-and-tube condenser.

Because frying vapor contains not only water vapor but also particulate matter, volatile organic compounds (VOCs), and aerosolized fats, the spray condenser–plate heat exchanger configuration presents notable disadvantages in terms of system complexity, maintenance, and reliability. Direct-contact condensation results in mixing of condensate and cooling water, which may require additional treatment or the inclusion of an oil–water separator to maintain water quality. The presence of contaminants further increases the risk of fouling in spray nozzles and the plate heat exchanger, necessitating frequent maintenance and increasing downtime. In addition, the spray condenser configuration requires an extra circulation pump and consists of multiple components, increasing system complexity, all of which contribute to higher maintenance demands and reduced operational reliability. The need for several interconnected units also results in a larger installation footprint, increasing required floor space and complicating integration into existing industrial layouts. In contrast, a shell-and-tube condenser provides a simpler and more robust solution, as it comprises a single component with well-established and straightforward tube-cleaning procedures, making it more attractive from both a maintenance, reliability and layout perspective.

A key advantage of a plate heat exchanger is its generally high heat transfer coefficient, compared to shell-and-tube heat exchangers, which allows for a much smaller required heat transfer area. In industrial plants, this is beneficial when the floor area is small. Because plates can be stacked easily, it is easier to achieve smaller approach temperature differences, down to 1 °C. This is reported to be difficult for shell-and-tube heat exchangers, where typical temperature approaches of 5–7 °C are used [24]. However, the advantage of reduced equipment size for a plate heat exchanger is partly offset by the large volume required for the spray condenser, which has typical volumetric heat transfer rates of approximately 10,000 W/m³ K [22] [25]. While the temperature difference between the exiting water and the condensing steam could theoretically approach zero in spray condensers, in practice it is typically 5–6 °C [26]. The optimal temperature approach for heat recovery depends on the trade-off between capital and energy costs, and is generally reported to lie between 10 and 30 °C for shell and tube heat exchangers [24].

Table 2.5: Typical overall heat transfer coefficients U for various heat exchanger configurations, grouped by type and application [22]

Type	Hot fluid	Cold fluid	U [$\text{W/m}^2\text{°C}$]
Heat Exchanger			
Shell and tube	Water	Water	800–1500
Gasket plate	Water	Water	5000–7000
Heater			
Gasket plate	Steam	Water	3500–4500
Shell and tube	Steam	Water	1500–4000
Shell and tube	Steam	Organic solvents	500–1000
Shell and tube	Steam	Light oils	300–900
Condenser			
Shell and tube	Aqueous vapors	Water	1000–1500

2.3. Heat pump

The heat recovery and heat delivery in the proposed system are coupled through the principle of a vapor-compression heat pump. The basic operation of such a heat pump is illustrated in Figure 2.5. It consists of four main components: an evaporator, a compressor, a condenser, and an expansion device. Heat is absorbed in the evaporator, compressed to a higher pressure and temperature, and subsequently released in the condenser. In the present application, the heat pump extracts heat from the frying vapor at a low temperature and upgrades it to a higher temperature for delivery back to the frying oil.

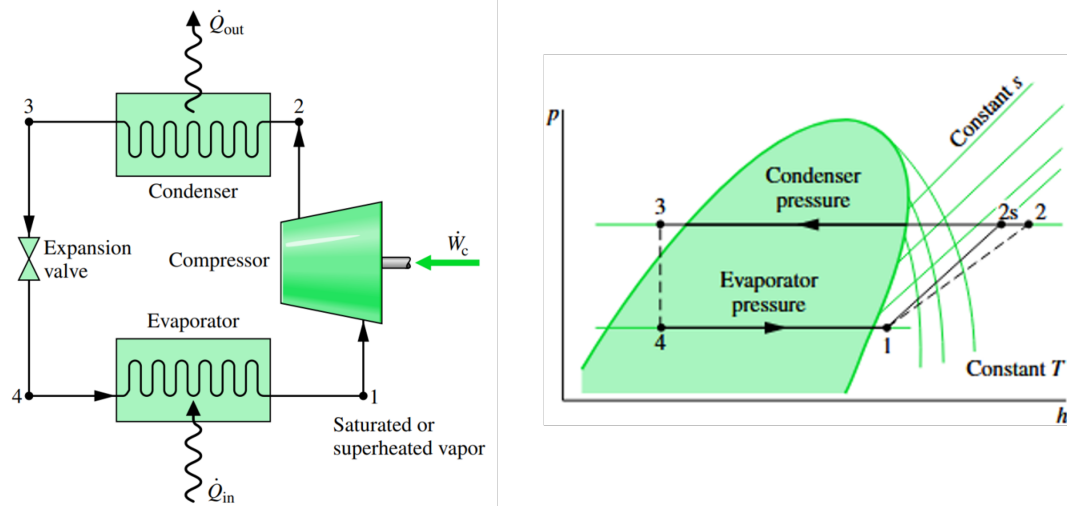


Figure 2.5: Schematic representation of a simple vapor-compression heat pump cycle.

The refrigerant of particular interest for the frying process, also recognized as a strategic focus area by Atlas Copco, is steam. Steam is widely used as a heat carrier in industrial applications, particularly in the chemical and food sectors, due to several important advantages [24]:

- Steam delivers a high amount of energy per unit mass via its latent heat of condensation. Unlike hot oil or flue gas, which transfer heat sensibly across a temperature range, steam releases energy at a constant temperature, making it particularly effective for thermal applications.
- The condensation temperature of steam can be precisely controlled by adjusting the pressure, enabling accurate and stable process control, which is critical for product quality and operational consistency.
- Condensing steam exhibits very high heat transfer coefficients, enabling the use of compact and cost-effective heat exchanger designs.
- Steam is nontoxic, nonflammable, and easily visible in the case of leaks. It is also chemically inert with many process fluids, making it a safe and versatile heat transfer medium.

Given these characteristics, steam is an ideal medium for integration in a heat recovery system designed to support the frying process.

Water vapor has a substantial specific volume in the temperature range of 80–90 °C, which would require oversized heat exchangers to achieve phase change under conventional designs. To avoid the need for such large evaporation surfaces, Chamoun et al. (2014) [27] propose a system that couples a liquid–liquid plate heat exchanger with a flash evaporator, allowing evaporation to occur efficiently without excessively large equipment.

In addition to the large-scale heat pump that drives the main thermal cycle, a smaller vapor-compression heat pump using a conventional refrigerant can be integrated to increase operational flexibility, particularly when moderate temperature lifts or intermediate temperature levels are required.

Figure 2.6 shows the operating temperature ranges and the maximum temperature lift (ΔT_{lift}) for nine refrigerants commonly applied in large-scale ($> 50 \text{ kW}_{\text{th}}$) electric-driven compression heat pumps. For most refrigerants, the maximum ΔT_{lift} lies between 60 and 78 K.

Among these, hydrofluorocarbons (HFCs) represent the dominant refrigerant class. Hydrofluoroolefins (HFOs) and hydrochlorofluoroolefins (HCFOs) have emerged as lower-GWP alternatives while maintaining comparable safety classifications. R717 (ammonia) is the only natural refrigerant included in the comparison. Owing to its high evaporation enthalpy, it is highly suitable for compact, high-capacity systems, although its toxicity and flammability require strict safety controls.

Other natural options such as R718 (water) and R744 (CO_2) were not yet widely implemented in large-scale heat pumps as of 2019. Hydrocarbons like R600 and R601 also offer low GWP but are classified as highly flammable, and according to EN 378-1 their maximum charge limit of 2.5 kg makes them impractical for industrial-scale applications.

Table 2.6 summarises the refrigerant classes, global warming potentials, and PFAS classification for reference.

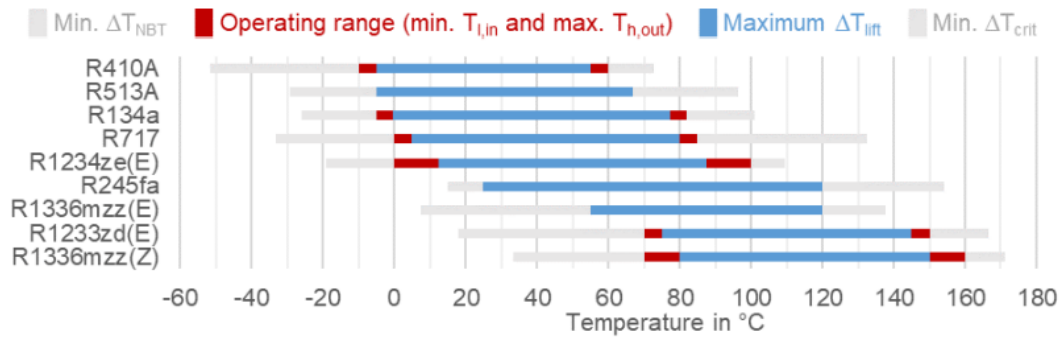


Figure 2.6: Temperature limits for the nine refrigerants, including the normal boiling temperature at 1.013 bar, minimum evaporator inlet temperature ($T_{i,\text{in}}$), maximum temperature lift (ΔT_{lift}), maximum condenser outlet temperature ($T_{h,\text{out}}$), and critical temperature (T_{crit}) [28].

Table 2.6: Overview of refrigerant classes, global warming potential (GWP), and PFAS classification.

Refrigerant	Class	GWP	PFAS
R1336mzz(Z)	HFO	2	Yes
R1233zd(E)	HCFO	1	Yes
R1336mzz(E)	HFO	18	Yes
R245fa	HFC	858	Yes
R1234ze(E)	HFO	<1	Yes
R717 (Ammonia)	Natural	0	No
R134a	HFC	1300	Yes
R513A	HFO/HFC blend	631	Yes
R410A	HFC	2088	Yes

According to the literature, a study indicates that for heat pumps available in 2021, the temperature glide of the heat source lies in almost all cases between 1 and 6 K for HFC, HFO, HCFO, and ammonia. The temperature glide at the heat sink ranges between 5 and 10 K for HFC, HFO, and HCFO, whereas for ammonia a value of around 10 K is common, with reported cases exceeding 20 K. [28]

2.4. Steam compressors

Compressors applied in steam mechanical vapor recompression (MVR) systems typically include piston, screw, and centrifugal (turbo) compressors [5]. Centrifugal compressors are particularly suited for high volumetric flow rates and generally achieve isentropic efficiencies in the range of 0.8–0.9, with intercooling or aftercooling applied downstream of the compression stage. However, the achievable temperature lift of a single-stage centrifugal compressor is limited to approximately 30 °C, which is indicated by the yellow rectangle in Fig. 2.7. Screw compressors, in contrast, typically operate with lower isentropic efficiencies of about 0.6–0.7 and frequently employ water injection to control discharge temperatures [29]. Their maximum single-stage compression ratio is generally limited to approximately 7. Owing to their positive-displacement operating principle, single-stage screw compressors can achieve substantially higher temperature lifts, up to approximately 80 °C. The maximum allowable discharge temperature for screw compressors is typically around 230 °C [30]. Compressing water vapor is inherently challenging due to its physical properties. Steam has a low molecular weight, a large specific volume, and a high adiabatic index. As a result, steam compressors must accommodate high pressure ratios, large volumetric flow rates, and elevated discharge temperatures [31].

Each compressor type offers different advantages and limitations for steam compression. Centrifugal compressors can process large volumetric flow rates but provide only modest single-stage compression ratios, produce high discharge temperatures, and are sensitive to droplets. They also require high-grade, expensive materials for the blades and casing. Screw compressors, in turn, are well suited for wet compression, provide higher compression ratios and stable operation, but their volumetric capacity is lower than that of centrifugal compressors.

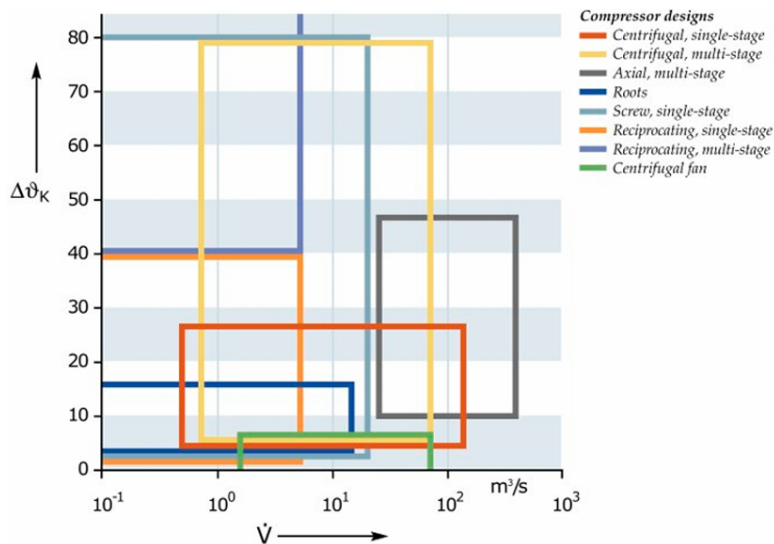


Figure 2.7: Functional ranges of compressors for vapor recompression in accordance with manufacturers' data (increase in condensation temperature of the water vapour $\Delta\vartheta_K$ at an initial state of 1 bar and 100 °C). Adapted from [32].

Table 2.7 presents the specific volume of saturated steam at selected pressures. As the pressure approaches a vacuum, the specific volume increases sharply. This means that at low inlet pressures, the compressor must handle large volumetric flow rates to achieve the required energy transfer, which significantly influences compressor sizing and design.

Table 2.7: Specific volume and energy content of saturated steam at selected pressures. Values obtained using CoolProp.

Pressure (bar)	Temperature (°C)	Specific Volume (m³/kg)	Energy per volume (kJ/m³)
0.1	46	14.67	176.13
0.3	69	5.23	501.98
0.7	90	2.36	1124.59
1.0	100	1.69	1579.14
5.0	152	0.3740	7423.50
10.0	180	0.1944	14288.34

3

Methodology

Chapter 2 provided an overview of the industrial potato frying process, including the associated mass and energy balances, and analyzed the characteristics of the frying vapors that determine the potential for heat recovery. Furthermore, the available methods for recovering this heat were discussed, and the basic working principle of a heat pump cycle was introduced. In addition to steam as a working medium, alternative refrigerants and their operating temperature ranges were examined, and the types of steam compressors used to raise low-pressure steam to high-pressure steam were reviewed.

Building upon the theoretical foundation established in Chapter 2, this chapter presents the overall methodological framework and workflow of the thesis following the literature study. Three main steps are required to achieve a complete design and analysis of the integrated heat supply system. These steps are:

1. **Steady-state calculations**, which are used to evaluate different system concepts and to analyze the sensitivity of the system.
2. **Equipment sizing**, based on the steady-state calculations and relevant for the subsequent dynamic modeling, as well as for the determination of the thermal inventory and pressure buffer.
3. **Dynamic simulation**, which is required to understand the interactions between the different components, to determine the appropriate control strategy for the system, and to assess the effects of each control approach.

3.1. Steady-State design and analysis

3.1.1. Modelling approach

The objective of the steady-state calculations is to quantify the amount of recoverable heat from a large industrial fryer and the usable heat that can be delivered back to the process via a steam heat pump cycle.

For each component k at steady state, the following mass and energy balances are used:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (3.1)$$

$$0 = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} + \dot{W}_k \quad (3.2)$$

For all heat exchangers, counter-current flow is assumed. The temperature approach is defined as the minimum temperature difference between the inlet and outlet streams on the same side of the heat exchanger. The constraints are expressed as:

$$(T_{hot,in} - T_{cold,out}, T_{hot,out} - T_{cold,in}) = \begin{cases} \Delta T_{\text{spray}} & \text{(spray condenser)} \\ \Delta T_{\text{PHX}} & \text{(plate heat exchanger)} \\ \Delta T_{\text{S\&T}} & \text{(shell \& tube condenser)} \end{cases} \quad (3.3)$$

For the other equipment, including the flash tank, expansion valve, steam compressor, and heat pump, models are obtained from the literature; these models will be discussed in detail in section 3.1.2.

In Excel, all models are coupled. The operation and evaluation of the system are based on two sets of parameters: control metrics and performance metrics. The control metrics define the operating conditions, thermodynamic state points, and boundary constraints of each component. The performance metrics, on the other hand, quantify the thermodynamic effectiveness and overall energy efficiency of the system. Together, these parameters provide a comprehensive framework for assessing the performance of the integrated fryer heat supply system. The values used for these parameters are either commonly reported in the literature or derived from Atlas Copco experience. The sensitivity analysis is performed using the Data Table feature, which calculates all desired process variables for varying input parameters.

The thermodynamic properties of the frying vapor and of water/steam are determined using the CoolProp library. For humid air, the HAPropsSI function in CoolProp allows the calculation of all relevant properties given three state variables. In this work, the state of the frying vapor is defined using temperature, humidity ratio, and atmospheric pressure. For water and steam, the thermodynamic properties are generally determined from temperature, pressure, and vapor fraction, providing a complete set of properties for each state point.

The steady-state solution is verified by performing component-level mass and energy balance checks. For each component, the fundamental conservation laws are evaluated and must be satisfied. Under steady-state conditions, no mass accumulation occurs and the internal energy of each control volume remains constant.

3.1.2. Component models

Fryer

Figure 3.1 and Table 3.1 provide a clear overview of the mass balance within a fryer, listing the various input and output streams and their respective compositions. Each stream is categorized by its flow direction (inlet or outlet) and includes components such as water, oil, and other substances. According to the literature, additional flows are sometimes incorporated into similar analyses, such as an exit flow of fines [15]. The oil input listed in the table represents the make-up oil required to compensate for oil absorbed in the potato product during frying. The fines outflow, while present in practice, is considered very small and has therefore been omitted from the table. This table focuses on the most significant mass flows relevant to the system. Table 3.1 presents the mass flow values corresponding to the production capacity of the largest potato processing plant in the Netherlands, with an annual output of approximately 210 kt/year, equivalent to roughly 7 kg/s of final product. [3] The energy balance for the fryer focuses on quantifying the incoming and outgoing energy streams, which are linked to the mass balance and the corresponding enthalpies of each stream. These enthalpies depend primarily on the stream temperatures and their respective specific heat capacities.

The enthalpy of the gas streams, such as the incoming air and outgoing frying vapor, can be determined using thermodynamic property libraries like CoolProp, which require inputs such as pressure, temperature, and humidity ratio. Although the incoming air may contain a small amount of water, this is assumed to be negligible compared to the significant amount of water evaporated from the potato during frying. Given that the frying process is open to the environment, the system is assumed to operate at atmospheric pressure. The conditions of the frying vapor, given in Table 2.2, allow for the calculation of enthalpy for the frying vapor stream. The specific enthalpy of the liquid and solid stream is calculated as:

$$h - h_{ref} = c_p \cdot (T - T_{ref}) \quad (3.4)$$

where h is the specific enthalpy (kJ/kg), c_p is the specific heat capacity (kJ/kg·K), T is the stream temperature (K or °C), and T_{ref} is the reference temperature. The total energy transfer associated with a stream is then given by:

$$\dot{Q} = \dot{m} \cdot (h - h_{ref}) = \dot{m} \cdot c_p \cdot (T - T_{ref}) \quad (3.5)$$

where $\dot{m}$ is the mass flow rate (kg/s). For food products such as potatoes, which are made up of multiple components, a composite c_p is used based on the mass-weighted average:

$$c_{p,mix} = \sum_i x_i \cdot c_{p,i} \quad (3.6)$$

where x_i and $c_{p,i}$ represent the mass fraction and specific heat capacity of component i , respectively.

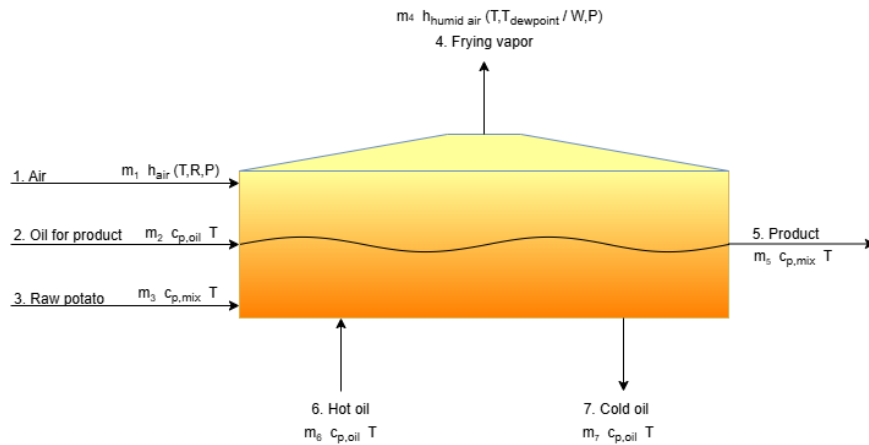


Figure 3.1: Schematic of the energy balance around a continuous deep-fat fryer. Each stream in the system is defined by a mass flow rate ($\dot{m}$), temperature (T), and specific heat capacity (c_p), except for the incoming air stream, which requires additional thermodynamic properties: pressure (P) and relative humidity (ϕ). The outgoing frying vapor stream is a humid air mixture characterized by either its dew point temperature ($T_{dewpoint}$) or its humidity ratio (W), defined as the mass ratio of water vapor to dry air. The energy content of the streams is evaluated using these properties to perform a thermal balance over the fryer system.

Table 3.1: Stream data with specific parameter values for determining the energy balance. The frying vapor has a dew point temperature of 90 °C.

General					Proposed case				
#	Description	In/Out	Composition	$\dot{m}$	X	$\dot{m}$ [kg/s]	C_p [kJ/kg·K]	T [°C]	H [kJ/kg]
1	Air	In	Air	$\dot{m}_3$	1	2.41	1.01	25	100.78
2	Oil for product	In	Oil	$\dot{m}_4$	1	0.35	2.36	100	-
3	Raw potato	In	Water	$\dot{m}_{1w}$	0.78	9.22	4.18	40	-
			Oil	$\dot{m}_{1o}$	0		2.36		
			Others	$\dot{m}_{1s}$	0.22		1.68		
4	Frying vapor	Out	Air	$\dot{m}_{5a}$	0.41	4.98	-	130	1661.65
			Steam	$\dot{m}_{5w}$	0.59				
			Oil	$\dot{m}_{5o}$	0				
5	Product	Out	Water	$\dot{m}_{8w}$	0.66	7.00	4.18	100	-
			Oil	$\dot{m}_{8o}$	0.05		2.36		
			Others	$\dot{m}_{8s}$	0.29		1.68		

Spray Condenser Model

The spray condenser recovers heat from the moist fryer exhaust by direct contact between the vapor and a circulating cooling-water stream. Part of the incoming vapor condenses and mixes with the cooling water, increasing the liquid mass flow rate; downstream, an equal amount of water is removed to maintain a constant inventory.

The total energy balance for the condenser, based on inlet and outlet enthalpy flows, is expressed as:

$$\dot{m}_{v,in}h_{v,in} + \dot{m}_{w,in}h_{w,in} = \dot{m}_{v,out}h_{v,out} + \dot{m}_{w,out}h_{w,out} \quad (3.7)$$

with the corresponding mass balance:

$$\dot{m}_{w,out} = \dot{m}_{w,in} + \dot{m}_{cond}, \quad \dot{m}_{v,out} = \dot{m}_{v,in} - \dot{m}_{cond} \quad (3.8)$$

where $\dot{m}_{cond}$ is the mass of vapor that condenses within the condenser.

At steady state, the condensate leaving with the outlet water is assumed to be in thermal equilibrium with it, such that $h_{cond} = h_{w,out} = h_\ell(T_{w,out})$.

The temperature approach between the vapor outlet and water inlet must satisfy:

$$T_{v,out} - T_{w,in} = \Delta T_{spray} \quad (3.9)$$

The temperature approach between the vapor inlet and water outlet must satisfy:

$$T_{dp} - T_{w,out} = \Delta T_{spray} \quad (3.10)$$

Plate Heat Exchanger

For the plate heat exchanger with a minimum approach of $\Delta T_{PHX} = 1$ K:

$$T_{h,out} - T_{c,in} = \Delta T, \quad (3.11)$$

$$T_{h,in} - T_{c,out} = \Delta T. \quad (3.12)$$

$$\dot{m}_h (h_{h,in} - h_{h,out}) = \dot{m}_c (h_{c,out} - h_{c,in}) = \dot{Q}_{PHX}. \quad (3.13)$$

Flash Tank

The flash tank generates saturated low-pressure steam at $T_{flash} = 70^\circ\text{C}$:

$$\dot{m}_{steam} = \frac{\dot{m}_{w,in} [h_{w,in} - h_l(T_{flash})]}{h_v(T_{flash}) - h_l(T_{flash})} \quad (3.14)$$

Mechanical Vapor Recompression (MVR)

The twin-screw water vapor compressor with water injection is the key component of the MVR system, as its performance strongly influences the overall system efficiency. To evaluate the compressor behavior, a multi-stage compression and water injection model is employed to ensure that the compressed steam remains at saturation at the compressor outlet [33]. In this model, the compression and injection processes are divided into n stages, each maintaining an identical compression ratio. At each stage, a controlled quantity of water is injected into the working chamber to absorb the heat of compression and to ensure that the vapor at the start of each stage remains saturated [34], [35].

The compression process is modeled as a multi-stage process, with each stage represented by steady-state mass and energy balances. For each stage k , the mass flow rate of vapor is obtained from the conservation of mass:

$$\dot{m}_{v,k} = (\dot{m}_{v,k-1} + \dot{m}_{inj,k-1}) \eta_v \quad (3.15)$$

where $\dot{m}_{v,k-1}$ is the vapor mass flow entering stage k , $\dot{m}_{inj,k-1}$ is the injected water mass flow from the previous stage, and η_v is the volumetric efficiency of the compressor stage.

The power required in stage k , $\dot{W}_{comp,k}$, is determined from the first law of thermodynamics using the isentropic efficiency η_{is} :

$$\dot{W}_{comp,k} = \frac{\dot{m}_{v,k} (h_{is,k} - h_{sat,k})}{\eta_{is}} \quad (3.16)$$

where $h_{out,k}$ and $h_{s,k}$ are the actual and isentropic outlet enthalpies of stage k , respectively. Typical values of the isentropic efficiency of twin-screw water vapor compressors reported in the literature range from 0.60 to 0.75 [33] [36].

The actual outlet enthalpy of stage k is calculated as:

$$h_{out,k} = h_{sat,k} + \frac{h_{is,k} - h_{sat,k}}{\eta_{is}} \quad (3.17)$$

where $h_{in,k}$ is the inlet enthalpy and $h_{sat,k}$ is the enthalpy of saturated vapor at the same pressure.

The injected water mass flow in stage k is evaluated from the stage energy balance:

$$\dot{m}_{inj,k} = \frac{\dot{m}_{v,k} (h_{out,k} - h_{sat,k+1})}{h_{sat,k+1} - h_{inj}} \quad (3.18)$$

where h_{inj} is the specific enthalpy of the injected water.

The total mass flow rate of vapor entering the compressor is determined from the suction conditions:

$$\dot{m}_{v,in} = \frac{\rho_s n_{rot} V_c}{60} \quad (3.19)$$

where ρ_s is the vapor density at suction, n_{rot} is the compressor rotational speed, and V_c is the theoretical displacement volume.

The total mass flow rate of vapor leaving the final compression stage is:

$$\dot{m}_{v,out} = \dot{m}_{v,in} + \sum_{k=1}^N \dot{m}_{inj,k} \quad (3.20)$$

Compression Heat Pump Model

In the study by Klute et al. [37], a representative model for closed-loop compression heat pumps (CLCHPs) is presented to describe the thermodynamic performance of vapor-compression systems used for industrial steam generation.

$$\dot{Q}_{src} = \dot{m}_{src} (h_{src,in} - h_{src,out}), \quad (3.21)$$

$$\dot{Q}_{snk} = \dot{m}_{snk} (h_{snk,out} - h_{snk,in}), \quad (3.22)$$

where $\dot{Q}_{src}$ is the heat absorbed from the low-temperature source, and $\dot{Q}_{snk}$ is the heat released to the sink. Both depend on the mass flow rate $\dot{m}$ and the enthalpy difference across the respective process sections.

$$\dot{Q}_{snk} = \dot{Q}_{src} + P_{el,CLCHP}, \quad (3.23)$$

Equation 3.23 represents the energy balance for the adiabatic CLCHP, where the electrical power input $P_{el,CLCHP}$ drives the compressor, linking the heat exchanged at both sides.

$$\Delta T_{Source} = T_{src,in} - T_{src,out}, \quad (3.24)$$

$$\Delta T_{lift,CLCHP} = T_{snk,out} - T_{src,out}, \quad (3.25)$$

where $T_{src,in}$ and $T_{src,out}$ refer to the inlet and outlet temperatures of the heat source, and $T_{snk,out}$ represents the outlet temperature of the sink. The temperature lift, $\Delta T_{lift,CLCHP}$, describes the difference between the sink outlet and the source outlet temperature.

$$COP_{Carnot} = \frac{T_{snk,out}}{\Delta T_{lift,CLCHP}}, \quad (3.26)$$

The Carnot coefficient of performance (COP_{Carnot}) defines the theoretical maximum efficiency achievable by a reversible heat pump cycle operating between the two temperature levels. This value is used as a reference to evaluate real system performance [37].

$$\eta_{ex,CLCHP,H} = \frac{COP_{CLCHP,H}}{COP_{Carnot}}, \quad (3.27)$$

where $\eta_{ex,CLCHP,H}$ represents the exergetic efficiency, quantifying the deviation of the real cycle from the ideal Carnot limit. It serves as an indicator for system losses and thermodynamic irreversibilities.

$$COP_{CLCHP,C} = COP_{CLCHP,H} - 1, \quad (3.28)$$

which expresses the inherent relation between the heating and cooling modes of a compression heat pump.

$$COP_{CLCHP,H} = \frac{\dot{Q}_{snk}}{P_{el,CLCHP}}, \quad (3.29)$$

$$COP_{CLCHP,C} = \frac{\dot{Q}_{src}}{P_{el,CLCHP}}, \quad (3.30)$$

Equations 3.29 and 3.30 define the real coefficients of performance for the heating and cooling modes, respectively, based on the heat transfer rate and the required electrical input power.

In practice, exergetic efficiencies between 0.45 and 0.5 are typical for industrial compression heat pumps, while values up to 0.7 are achievable for large-scale installations [37] [38]. Maintaining small temperature differences on the source side ($\Delta T_{Source} < 20$ K) is recommended to minimize losses and improve system efficiency. Nonetheless, data reported by IEA Annex 58 show that some systems operate with larger temperature glides of around 30 K [39], [40], [41].

Return flow from condenser in the flash tank

The total mass of water in the system remains constant because the cooling water used by the compressor is taken from the flash tank. The model also includes the return of condensate to the flash tank. The magnitude of the mass flow throughout the system depends on the steam generated in the flash tank, the cooling water added during compressor operation, and the mass flow returning from the condenser back into the flash tank. By considering these interactions, the overall mass and energy balances can be established, allowing the calculation of the actual mass flow rates in the system, including the effect of the return flow to the flash tank.

Oil Heat Exchanger

The heat that can be delivered is governed by the steam mass flow rate, the compressor outlet superheated (sh) temperature, steam condensation, and the degree of subcooling (sc). The condensation temperature of the steam is determined by the pressure prevailing in the condenser.

$$\dot{Q}_{\text{delivered}} = \dot{m} (h_{\text{in}} - h_{\text{out}}) = \dot{m} [h(T_{\text{sh,in}}, P_c) - h(T_{\text{sc,out}}, P_c)] \quad (3.31)$$

Condensing steam releases its latent and sensible heat to the fryer oil circuit, which serves as the thermal sink. The energy balance governs the heat transfer on the oil side:

$$\dot{Q}_{\text{oil}} = \dot{m}_{\text{oil}} c_{p,\text{oil}} (T_{\text{oil,out}} - T_{\text{oil,in}}), \quad (3.32)$$

Under steady-state conditions, the heat released by the condensing steam equals the heat absorbed by the oil:

$$\dot{Q}_{\text{delivered}} = \dot{Q}_{\text{oil}}. \quad (3.33)$$

Exergy destruction

Exergy analysis is employed to evaluate the thermodynamic effectiveness of the individual components in the system. Exergetic efficiencies are useful for distinguishing energy utilization concepts that are thermodynamically effective from those that are less so, as they explicitly account for irreversibilities. They also enable the assessment of engineering measures intended to improve system performance by comparing efficiency values before and after modifications.

Following the formulation of Moran and Shapiro [42], the specific flow exergy of a stream is defined as:

$$e_f = h - h_0 - T_0 (s - s_0) + \frac{V^2}{2} + gz, \quad (3.34)$$

where h and s denote the specific enthalpy and entropy of the stream, respectively, h_0 and s_0 are the corresponding properties at the environmental (dead-state) conditions, T_0 is the ambient temperature, and the final two terms represent the kinetic and potential exergy contributions.

For a control volume operating under steady-state conditions, the corresponding exergy balance is expressed as

$$0 = \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j - \dot{W}_{cv} + \sum_i \dot{m}_i e_{f,i} - \sum_e \dot{m}_e e_{f,e} - \dot{E}_d, \quad (3.35)$$

where $\dot{Q}_j$ denotes the heat transfer rate at boundary temperature T_j , $\dot{W}_{cv}$ is the work rate, $\dot{m}$ represents mass flow rates, and $\dot{E}_d$ is the rate of exergy destruction.

3.1.3. Control and performance metrics

The operation and evaluation of the system are based on two sets of parameters: control metrics and performance metrics. The control metrics define the operating conditions, thermodynamic state points, and boundary constraints of each component. The performance metrics, on the other hand, quantify the thermodynamic effectiveness and overall energy efficiency of the system. Together, these parameters provide a comprehensive framework for assessing the operational behavior and performance of the integrated fryer heat recovery system.

Table 3.2: Summary of key process parameters and performance metrics.

Type	Description
Process parameters	
Mass flow of product	Specifies the rate of frying production and determines the corresponding heat demand of the fryer.
Temperature of frying vapor and dew point	Define the thermal characteristics of the frying vapor entering the recovery system.
Flash tank temperature	Controls the amount of heat recovered from the frying vapor condenser and determines the properties of the generated steam.
S&T and plate heat exchanger temperature approach	Sets the temperature difference between the inlet and outlet streams in the heat exchanger.
Spray condenser temperature approach	Regulates condensation conditions and heat rejection in the spray condenser.
MVR outlet temperature and isentropic efficiency	Determine the pressure ratio, compression work, and overall compressor performance.
Subcooled water outlet temperature	Defines the final delivered heat to the fryer oil loop.
Performance metrics	
Recovered heat from the fryer	Thermal energy extracted from the fryer exhaust vapor.
Electric power input	Electrical power consumed by the compressor.
Heat delivered	Useful thermal energy transferred to the fryer oil loop.
Supplementary heating demand	Additional heating required when the heat pump cannot meet the fryer's total demand.
Exergy destruction	Total thermodynamic irreversibilities in the system.
Coefficient Of Performance	Expression of the effectiveness of a heat pump system.

Coefficient of performance represents the ratio of useful heat output to the required electrical input power. In this case, the specific heat delivered is associated with the frying-oil condenser, where hot steam is cooled, condensed, and subsequently subcooled. The heat released during this process, divided by the work performed by the compressor, defines the COP_{MVR} .

$$COP_{MVR} = \frac{\dot{Q}_{del}}{P_{el}}, \quad (3.36)$$

3.2. Equipment Sizing Methodology

Based on the steady-state results for the general design, the equipment will be sized accordingly. Table 3.3 summarizes all equipment being sized and specifies the method used for each piece of equipment. Subsequently, the necessary input parameters for each piece of equipment are provided, along with the objective of the sizing calculations.

Equipment to be sized

Table 3.3: Equipment and corresponding sizing or design methodology.

Equipment	Method
Frying vapor condenser	Sized in Aspen Exchanger Design & Rating (EDR, V12).
Oil condenser	Sized in Aspen Exchanger Design & Rating (EDR, V12).
Flash tank	Sized following the design methodology for gas–liquid separators described by Towler and Sinnott in <i>Chemical Engineering Design</i> (3rd ed.) [43].
Fryer	Sized using characteristic design data and performance specifications obtained from industrial fryer manufacturers, serving as a representative reference for heat-duty estimation and volume assessment.

Inputs for sizing

- Fryer heat demand $\dot{Q}_{\text{demand}}$ and oil-loop inlet/outlet temperatures
- Flash-tank pressure and mass flows
- Oil-condenser operating pressure, temperatures and mass flows
- Frying-vapor dew-point temperature, frying vapor temperature and mass flows
- Nominal stream mass flow rates (vapor, condensate, oil, cooling/spray media)
- Target temperature approaches/pinch values

Design objectives

For each unit, the sizing procedure provides several key thermal design parameters. These include the required heat-transfer area (A) and the overall heat-transfer coefficient (U), which together characterize the unit's heat-transfer performance. The thermal inventory, defined as the liquid hold-up volume on each side, is also determined and is essential for understanding start-up behavior and system dynamics. In addition to the liquid hold-up, the metal mass of the tube bundle contributes significantly to the system's thermal inertia, influencing how quickly the unit responds to temperature changes during start-up and shut-down.

3.2.1. Design of a vertical flash tank

The flash tank in the system also functions as a buffer vessel, providing pressure stability during transient disturbances in the thermal energy balance. A vertical flash tank, or gas–liquid separator, is used to separate entrained liquid droplets from a vapor stream by reducing the vapor velocity sufficiently for the droplets to settle under gravity. Such vessels are commonly installed downstream of condensers, throttling valves, or other process equipment to ensure single-phase vapor flow and to protect downstream machinery such as compressors. A schematic representation of a typical vertical flash tank is shown in Figure 3.2.

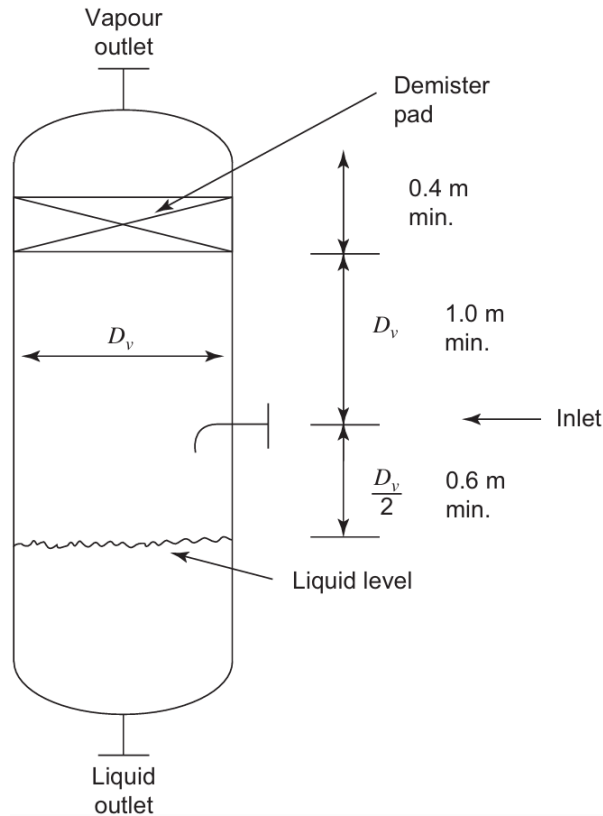


Figure 3.2: Schematic representation of a vertical flash tank (adapted from typical design practice).

Settling Velocity

The fundamental design principle of a flash tank is based on the ability of the liquid droplets to settle out of the vapor stream. The *settling velocity* represents the terminal velocity of a droplet moving through the gas phase and can be estimated using equation 3.37.

$$u_t = 0.07 \left(\frac{\rho_L - \rho_V}{\rho_V} \right)^{1/2} \quad (3.37)$$

where:

- u_t = settling (terminal) velocity of the liquid droplet [m/s],
- ρ_L = liquid density [kg/m³],
- ρ_V = vapor density [kg/m³].

The design vapor velocity, u_s , is then defined as a conservative fraction of the settling velocity to allow for surges and non-ideal flow conditions:

$$u_s = \begin{cases} u_t & \text{with a demister pad} \\ 0.15 u_t & \text{without a demister pad} \end{cases} \quad (3.38)$$

A demister pad, typically a 100 mm thick stainless-steel knitted mesh with a density of around 150 kg/m³, enhances droplet coalescence and allows the separator to operate efficiently at higher vapour velocities, thereby enabling smaller vessel diameters.

Vessel Diameter

The vessel diameter should be large enough to slow the gas flow so that the liquid droplets can settle under gravity. The required diameter is expressed as:

$$D_V = \sqrt{\frac{4V_V}{\pi u_s}} \quad (3.39)$$

where:

- D_V = minimum vessel diameter [m],
- V_V = volumetric vapor flow rate [m³/s].

Vessel Height and Liquid Hold-up

The total height of the separator must provide adequate disengagement space above the liquid level. As a rule of thumb, the height should be at least equal to the vessel diameter or not less than 1 m, whichever is greater. This disengagement zone prevents liquid carryover during transient flow conditions.

Below the vapor disengagement zone, a liquid hold-up section is included to allow smooth level control and temporary accumulation of the separated liquid. The required hold-up volume, V_L , is typically based on a desired residence time, t_h , of 5–10 minutes:

$$V_L = Q_L t_h \quad (3.40)$$

where Q_L is the liquid volumetric flow rate [m³/s]. The resulting volume is used to define the distance between the normal liquid level and the outlet nozzle.

3.2.2. General design considerations for heat exchanger

This section outlines the key design criteria that govern the allocation of fluids to the tube and shell sides in heat exchanger design.

- **Corrosion:** The fluid with the highest corrosive potential is typically assigned to the tube side, as the cost of corrosion protection or surface treatment is lower for the tubes than for the shell.
- **Fouling:** Fluids prone to fouling are preferably placed on the tube side, where higher velocities can be maintained to minimise deposition and the surfaces are easier to clean.
- **Fluid temperature:** When high-temperature fluids require special alloys, placing them on the tube side helps limit the amount of expensive material. At moderate temperatures, locating the hotter fluid in the tubes also reduces the external shell temperature and minimizes heat loss.
- **Operating pressure:** The stream operating at the highest pressure should be assigned to the tube side, as the smaller diameter tubes can withstand internal pressure more effectively and at lower cost than a large shell.
- **Pressure drop:** For the same allowable pressure drop, the tube side generally provides a higher heat transfer coefficient. Fluids with lower permissible pressure drop are therefore often allocated to this side.
- **Viscosity:** The more viscous fluid is usually directed to the shell side, where turbulence can be promoted with baffles and flow distribution can be controlled more effectively.
- **Flow rate:** Fluids with lower flow rates are typically placed on the shell side to obtain a more compact and economical exchanger design.

For the design, the primary consideration is that the frying vapor and frying oil streams are significantly more contaminated than the clean water stream. Although this configuration places the high-pressure stream on the shell side and the higher-flow-rate stream on the tube side, it offers a practical advantage. In both heat exchangers, condensation occurs, and arranging the equipment vertically allows the condensate to drain naturally downward, simplifying operation and improving reliability.

Shell and tube fluid velocities

For shell-and-tube heat exchangers, selecting appropriate fluid velocities is crucial to balance heat transfer performance and pressure drop. For liquid flow, typical design velocities range from 1 to 2 m/s on the tube side, with a maximum of 4 m/s when higher velocities are required to reduce fouling. Water is often designed to flow at 1.5–2.5 m/s on the tube side, while the shell-side velocity is generally maintained between 0.3 and 1 m/s. In contrast, for vapour or gas flows operating under high pressure, the typical velocity range is between 5 and 10 m/s.

Table 3.4: Fluid allocation for the frying vapor condenser and oil condenser

Condenser	Tube side	Shell side
Frying vapor condenser	Frying vapor	Water from flash tank
Oil condenser	Frying oil	Steam

Table 3.5: Viscosity of the working fluids

Fluid	Viscosity [kg/(m·s)]
Water (flash tank outlet)	0.00015
Frying oil (tube side of oil condenser)	0.001486

Pressure drop

The pressure drop on the condensing side is difficult to predict because two phases are present and the vapour mass velocity decreases along the condenser length. A common approach is to estimate the pressure drop using single-phase flow correlations and apply a correction factor to account for the variation in vapor velocity during condensation. For cases of total condensation, Frank [44] suggests taking the pressure drop to be approximately 40% of the value calculated from the inlet vapour conditions.

Table 3.6: Allowable pressure drop for liquids, gases, and vapors

Category / Condition	Allowable pressure drop [kN/m ²]
Liquids	
Viscosity < 0.001 kg/(m·s)	35
Viscosity 0.001–0.010 kg/(m·s)	50–70
Gas and vapors	
High vacuum	0.4–0.8
Medium vacuum	0.1 × absolute pressure
1–2 bar	0.5 × system gauge pressure
Above 10 bar	0.1 × system gauge pressure

The tube side of the frying vapor condenser operates approximately at atmospheric pressure, while the steam side of the oil condenser functions at around 10 bar. Based on typical design recommendations, the allowable pressure drop for the vapor condenser is taken as 0.5 times the system gauge pressure, and for the oil condenser as 0.1 times the system gauge pressure. The resulting allowable pressure drops, also including the adjustment suggested by [44], are summarised in Table 3.7.

Table 3.7: Allowable pressure drops for the frying vapor condenser and oil condenser

Condenser	Tube side	Shell side
Frying vapor condenser	20 kN/m ² (frying vapor)	35 kN/m ² (water)
Oil condenser	50 kN/m ² (frying oil)	40 kN/m ² (steam)

3.3. Dynamic simulation

3.3.1. General approach

Dynamic simulations are performed to investigate the transient behaviour of the heat recovery system and to support the design of an effective control strategy. The simulations provide insight into how the system responds to variations in production rate, allowing for the prediction and optimization of control actions under changing operating conditions. The models are implemented in Matlab using mass and energy balance equations for the relevant components.

The dynamic behaviour of the heat recovery system is simulated in Matlab as a set of interconnected control volumes and algebraic process units. The simulation is executed with a fixed time step over a specified horizon, in which all state variables are updated sequentially based on the current inlet conditions and control actions. Thermodynamic properties of water and steam are obtained using the `XSteam` library. For the flash tank model, selected saturation properties are additionally retrieved from an auxiliary property file. The accuracy of the empirical saturation property correlations employed in this auxiliary model has been assessed and discussed in detail in Section 3.1.2 of the original flash tank model development report of Atlas Copco. In the present work, these property formulations are adopted as given, and no additional validation of the thermodynamic property correlations is performed.

Components with significant thermal inertia, such as the flash tank and fryer, are modelled dynamically. These components contain significant fluid volumes that store thermal energy, as shown in Section 5.3, and influence system transients. In contrast, units that respond much faster (compressor and both condensers) are assumed to have negligible thermal inertia and are therefore represented by steady-state relationships. Heat transfer in the heat exchangers is modelled using the effectiveness–NTU (ε -NTU) method. This combined approach allows the transient system response to be captured, yet provides a computationally efficient representation of the system dynamics, and forms the basis for controller development.

Process control is embedded directly into the dynamic model. Two control methodologies are investigated: (i) conventional feedback control, where controlled variables are compared to their setpoints and actuators respond based on error, and (ii) combined feedforward–feedback control, where disturbances (variations in product throughput) are anticipated through feedforward action while residual deviations are corrected by feedback. The dynamic simulation results obtained under feedback control are used to identify the new post-disturbance steady-state operating point. This operating point subsequently serves as the input to the feedforward control design. This dual strategy is designed to suppress transient deviations during production changes while maintaining a stable fryer operation.

Process control is embedded directly into the dynamic simulation via PID controllers. The flash-tank pressure is controlled by adjusting the compressor rotational speed; the high-pressure side is regulated via the expansion valve, based on the available subcooled area in the oil condenser and the pressure in the vessel; additional steam injection is used to maintain the desired condenser outlet and fryer inlet temperatures; and the fryer oil-flow rate is manipulated to keep the fryer temperature at its setpoint. All controllers are updated at each time step using the current measured variables and their internal integral states.

To analyse system performance under varying production rates, a disturbance scenario is imposed in which the dry-air flow in the frying vapour and the mass flow of fries are suddenly reduced by 25 % after an initial period at nominal conditions. The resulting temperature, pressure, mass flow, and compressor power transients are recorded. Throughout the simulation, overall and component-level mass and energy balances are continuously evaluated. For the global system, all incoming and outgoing mass and energy flows are summed and compared with the change in internal energy of the components possessing thermal inertia. This comparison provides a measure of the total energy balance residual and verifies that energy is conserved at system level. In addition, the mass and energy balances of each individual component are monitored separately, enabling the detection of numerical inconsistencies and ensuring accurate conservation behaviour throughout the model.

3.3.2. Solution strategy

The overall system is solved using a sequential, component-wise solution strategy within each global time step. Rather than assembling and solving a fully coupled system of equations, each process unit is evaluated individually using the numerical or analytical method best suited to its governing equations. The state variables of upstream components provide the boundary conditions for downstream components within the same time step.

For the flash tank, the governing equations are solved using MATLAB's `solve` function, yielding a consistent solution for pressure, temperature, and phase distribution that satisfies thermodynamic equilibrium constraints between the liquid and vapor phase. The frying vapor condenser is solved iteratively using a Picard fixed-point iteration to resolve the interdependence between heat transfer, condensation, and outlet conditions. For the oil condenser, the outlet conditions are obtained by minimizing the residual of the energy balance using the `fminsearch` function, which is suitable for the smooth, low-dimensional nonlinear problem encountered.

For the fryer and associated piping, the governing energy balance reduces, within each time step, to a linear first-order ordinary differential equation with constant coefficients. This equation is therefore solved using its analytical (closed-form) solution rather than a numerical ODE solver. The oil temperature at the end of the time step is obtained from the exact exponential solution, while a special case is handled explicitly when the mass flow rate is zero. As a result, the temperature update is unconditionally stable and free of numerical integration error for the assumed model structure.

In addition to the outlet temperature at the end of the time step, an average outlet temperature over the time step is computed from the analytical solution. This mean temperature represents the time-averaged temperature of the oil leaving the fryer during the current step. The average outlet temperature serves two purposes: first, it provides a physically consistent interface variable when coupling the fryer model to downstream components, ensuring that inter-component energy exchange is based on time-averaged rather than end-point conditions; second, it enables a direct verification of energy conservation by relating the change in stored thermal energy of the oil to the net convective energy flow and the heat extracted by the fries over the same interval.

The sequential coupling of the individual component models is physically justified by the presence of pipe elements with finite mass holdup between process units. Within each time step, the mass contained in these pipe elements exceeds the mass that can leave the element during that interval, for the liquid water pipes. As a result, the downstream boundary conditions are always based on physically available fluid, preventing nonphysical propagation of information or mass deficits. This ensures that the sequential solution approach remains mass-consistent and physically valid for the chosen time-step size.

3.3.3. Energy and mass balance check

To ensure thermodynamic consistency of the dynamic simulation, a system-wide mass and energy balance verification was implemented as a post-processing audit. This procedure does not influence the state evolution of the model, but evaluates whether the simulated temperatures, pressures, densities, mass flows, and energy flows satisfy the fundamental conservation laws at each time step. The balance is evaluated on a discrete time grid with a fixed time step Δt , and all accumulation terms are computed using finite differences between consecutive time steps.

A system-wide control volume is defined that compares the total change of mass and energy inside the system with the in- and outgoing streams, namely injection water and supplementary steam entering the system, and the splitter discharge leaving the system. All mass flows circulating within the system boundary do not contribute to net mass or energy transfer across the boundary and are therefore evaluated separately using component-level mass and energy consistency checks. The system-wide mass balance is formulated as:

$$\frac{dM_{\text{sys}}}{dt} = \sum \dot{m}_{\text{in}} - \sum \dot{m}_{\text{out}},$$

where M_{sys} denotes the total mass stored within the system boundary. Mass accumulation is primarily determined by changes in the mass stored in the flash tank and pressure vessel. For the flash tank, accumulation is calculated as the product of density variations and the known tank volume. At each time step, a mass balance residual is computed as the difference between the calculated accumulation rate and the net mass inflow.

The system-wide energy balance is formulated in rate form as

$$\frac{dU_{\text{sys}}}{dt} = \dot{W}_{\text{comp}} + \sum \dot{m}_{\text{in}} h_{\text{in}} - \sum \dot{m}_{\text{out}} h_{\text{out}} + \dot{Q}_{\text{net}},$$

where U_{sys} is the total internal energy stored within the system, $\dot{W}_{\text{comp}}$ is the compressor shaft work, and $\dot{Q}_{\text{net}}$ represents the net energy exchange associated with the fryer path. The rate of change of internal energy is evaluated for each component with a significant inventory. The fryer oil is treated as a lumped mass, and its energy accumulation is calculated from the oil mass, specific heat capacity, and temporal temperature variation. To approximate the change of internal energy in the flash tank, the backward difference method is used for the derivatives of density, specific enthalpy and pressure. The pipes are treated as lumped control volumes with sensible energy storage, while the pressure vessel contribution is approximated using the backwards difference method using the coolprop internal energy function at the specified temperature and pressure. The total energy accumulation rate is obtained by summing the contributions of all inventories.

Energy inputs to the system consist of compressor shaft work and enthalpy flows associated with mass streams entering and exiting the control volume. In addition, the fryer path is treated explicitly by accounting for the heat extracted from the oil by the product and the heat recovered in the frying vapor condenser. These contributions are combined into a net energy export term to avoid double counting of internally recovered heat while ensuring that only net energy crossing the system boundary is included. At each time step, an energy balance residual is calculated as the difference between the total energy accumulation rate and the net energy input.

In addition to the system-wide balances, local mass and energy balances are evaluated for individual components, including the flash tank, compressor, fryer, heat exchangers, and connecting pipes. These component-level checks compare local accumulation terms with local enthalpy flows and work contributions and are used diagnostically to identify the origin of any imbalance observed at system level. Overall, this balance verification approach provides a robust method for validating the physical consistency of the dynamic model and supports reliable interpretation of the simulation results.

3.3.4. Thermodynamic properties

In this work, the thermodynamic properties of water and steam are obtained using the XSteam library, a MATLAB implementation of the IAPWS-IF97 formulation for water and steam [45]. XSteam allows the user to determine key properties such as enthalpy, entropy, density, and specific volume by specifying two independent thermodynamic variables, ensuring accurate and consistent property data over a wide range of conditions. For example, when pressure and temperature are known, XSteam internally applies the appropriate IAPWS-IF97 region equations to calculate the corresponding state point. The specific enthalpy can then be expressed as a function of pressure P and temperature T , as shown in Equation 3.41.

$$h = f_{\text{XSteam}}(P, T) \quad (3.41)$$

3.3.5. Component models

Flash tank

The dynamic simulation of the flash tank is implemented in Matlab to analyse the transient thermodynamic behaviour of the heat recovery system. The model solves the governing mass and energy balance equations over time using a fixed time step. The property evaluations are updated at each time step to capture variations in pressure and temperature within the tank.

The implementation of the flash tank model relies on repeated evaluations of steam properties. These computations are handled in a separate file. This file contains the formulations for steam properties and their associated derivatives with respect to the key thermodynamic variables. In addition, the definitions and derivatives of the specific enthalpy in the flash tank are provided.

The derivation of the model is based on Equations 3.42, 3.43, and 3.44. By further rearranging these equations and taking into account the below-stated assumptions, the formulation ultimately leads to the energy balance expressed in Equation 3.46. The following assumptions are made in the flash tank model:

- the flash tank is perfectly insulated (no heat gain and losses) $\dot{Q} = 0$,
- no mechanical work is performed within the control volume $\dot{W} = 0$,
- the pressure and temperature are uniform throughout the tank,
- liquid and vapor phases remain in thermodynamic equilibrium.

$$\frac{dE_{\text{system}}}{dt} = \dot{Q} - \dot{W} + \dot{m}_{\text{in}}e_{\text{in}} - \dot{m}_{\text{out}}e_{\text{out}} \quad (3.42)$$

$$e = u + \frac{c^2}{2} + gz \quad (3.43)$$

$$h = u + pv \quad (3.44)$$

Mass balance:

$$V_{FT} \frac{d\bar{\rho}_{FT}}{dt} = \dot{m}_{\text{snk},\text{in}} - \dot{m}_{\text{snk},\text{out}} + \dot{m}_{\text{src},\text{in}} - \dot{m}_{\text{src},\text{out}} \quad (3.45)$$

Energy balance:

$$\begin{aligned} V_{FT} \left(\bar{\rho}_{FT} \frac{d\bar{h}_{FT}}{dt} - \frac{dp_{FT}}{dt} \right) &= \dot{m}_{\text{snk},\text{in}} (h_{\text{snk},\text{in}} - \bar{h}_{FT}) - \dot{m}_{\text{snk},\text{out}} (h_{\text{snk},\text{out}} - \bar{h}_{FT}) \\ &+ \dot{m}_{\text{src},\text{in}} (h_{\text{src},\text{in}} - \bar{h}_{FT}) - \dot{m}_{\text{src},\text{out}} (h_{\text{src},\text{out}} - \bar{h}_{FT}) \end{aligned} \quad (3.46)$$

The average density of the flash tank, which depends on the vapor fraction, is given by:

$$\bar{\rho}_{FT} = (1 - \gamma)\rho_f + \gamma\rho_v, \quad (3.47)$$

where γ is the volumetric vapor fraction, and ρ_f and ρ_v represent the liquid and vapor phase densities, respectively.

During each simulation step, the model updates the state variables (p_{FT} , $\bar{h}_{FT}$, γ) using the instantaneous mass and energy flows. This enables the prediction of the tank's dynamic response, including variations in vapor generation and pressure, which are essential for the design of the system's control strategy.

Oil condenser

The ε -NTU method is used to model the condensing heat exchanger because it allows the heat transfer rate to be determined directly from inlet conditions without requiring prior knowledge of the outlet temperatures. Its low computational complexity and numerical robustness make it particularly suitable for system-level simulations and parametric studies, where repeated evaluations are required. Moreover, the method provides a direct link between heat exchanger size and thermal performance, which is essential for performance assessment and optimization.

The steam-side flow is divided into three regions: superheated (region 1), condensation (region 2), and subcooled (region 3). Each region is modeled using the effectiveness-NTU (ε -NTU) method, where the actual heat transfer rate $\dot{Q}_i$ is expressed as:

$$\dot{Q}_i = \varepsilon_i C_{\min,i} (T_{h,i,\text{in}} - T_{c,i,\text{out}}), \quad i = 1, 2, 3 \quad (3.48)$$

The heat exchanger effectiveness ε_i is determined by the flow configuration and the heat capacity ratio. For the superheated and subcooled regions (1 and 3), the following correlation for a counterflow heat exchanger is used:

$$\varepsilon_i = \frac{1 - \exp[-\text{NTU}_i(1 - C_{r,i})]}{1 - C_{r,i} \exp[-\text{NTU}_i(1 - C_{r,i})]}, \quad \text{NTU}_i = \frac{U_i A_i}{C_{\min,i}}, \quad C_{r,i} = \frac{C_{\min,i}}{C_{\max,i}}, \quad i = 1, 3 \quad (3.49)$$

For the condensation region (2), the heat transfer is driven by the latent heat of phase change, and the effectiveness is defined as:

$$\varepsilon_2 = 1 - \exp(-\text{NTU}_2), \quad \text{NTU}_2 = \frac{U_2 A_2}{C_{\min,2}}. \quad (3.50)$$

The total heat transferred in the condenser is given by:

$$\dot{Q}_{\text{HEX}} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3. \quad (3.51)$$

The total heat exchanger area is constant and distributed among the three regions as:

$$A_{\text{tot}} = A_1 + A_2 + A_3. \quad (3.52)$$

Each region is characterized by its own overall heat transfer coefficient U_i , reflecting differences in dominant heat transfer mechanisms for the superheated, condensation, and subcooled zones.

The total heat transfer area A_{tot} and the regional heat transfer coefficients U_i are determined from the detailed heat exchanger sizing performed in Aspen EDR, ensuring that the model parameters are consistent with the physical design of the condenser.

Two operational modes:

To ensure physically realistic results, the model operates in two distinct modes depending on the condensation behavior:

- **Full condensing mode:** Includes all three regions (superheated, condensation, and subcooling). This mode is used when the entire vapor flow is condensed and subsequently cooled below the saturation temperature.
- **Partial condensing mode:** Activated when the condensation is incomplete. In this case, only regions 1 and 2 are solved. The degree of condensation is then calculated from the fraction of latent heat released:

$$x_{\text{frac}} = \frac{\dot{Q}_2}{\dot{m}_h(h_V - h_L)}, \quad (3.53)$$

where x_{frac} is the vapor mass fraction condensed, and h_V and h_L are the saturated vapor and liquid enthalpies at the operating pressure.

Frying vapor condenser

The frying vapor condenser represents the thermal interaction between frying vapor and the flash tank water stream. The model assumes steady-state operation of a counterflow heat exchanger with a constant overall heat transfer coefficient (UA) and negligible pressure losses. The thermal performance is described using the effectiveness–NTU (ε –NTU) method, where the effectiveness for counterflow configuration is expressed as:

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \exp[-NTU(1 - C_r)]}, \quad NTU = \frac{UA}{C_{\min}}, \quad C_r = \frac{C_{\min}}{C_{\max}}, \quad (3.54)$$

The heat transfer rate is then given by:

$$\dot{Q} = \varepsilon C_{\min} (T_{h,\text{in}} - T_{c,\text{in}}) \quad (3.55)$$

On the hot side, the moist air is assumed to be saturated and undergoes condensation during cooling. To capture this effect, a fictitious specific heat capacity c_h is introduced to represent both the sensible and latent heat transfer. This value is iteratively updated until convergence based on the change in moist air enthalpy:

$$c_h = \frac{h_{h,\text{in}} - h_{h,\text{out}}}{T_{h,\text{in}} - T_{h,\text{out}}}. \quad (3.56)$$

The enthalpy of the humid air per kilogram of dry air is determined from a polynomial correlation as a function of temperature, fitted from psychrometric data at atmospheric pressure:

$$h_{\text{da}}(T) = a_4 T^4 + a_3 T^3 + a_2 T^2 + a_1 T + a_0. \quad (3.57)$$

This formulation ensures that the condensation behavior of the moist air is represented through an equivalent heat capacity, allowing the model to predict a physically realistic outlet temperature and heat transfer rate.

To apply the correlations for the humid-air enthalpy and the appropriate specific heat capacity, given by equations 3.56 and 3.57, certain simplifications are required. The model considers only the latent heat released by the condensing frying vapor. In contrast, the sensible heat of the superheated moist air is neglected because of its relatively small contribution to the total recovered energy. The other simplification is stated below:

The energy balance for the humid air stream can be expressed as:

$$\dot{m}_{\text{ha},\text{in}} h_{\text{ha}}(T_{\text{dp},\text{in}}) = \dot{m}_{\text{ha},\text{out}} h_{\text{ha}}(T_{\text{dp},\text{out}}) + \dot{m}_{w,\text{out}} h_{w,\text{out}}, \quad (3.58)$$

where the enthalpy $h_{\text{ha}}(T_{\text{dp}})$ is evaluated as a function of the dew-point temperature of the humid air.

For practical implementation in the MATLAB model, the calculation is performed on a dry-air basis, as the dry-air mass flow rate remains constant throughout the condenser. Accordingly, the transferred heat from the humid air to the cooling water is evaluated by:

$$\dot{Q}_{\text{from ha}} = \dot{m}_{\text{da}} [h_{\text{da}}(T_{\text{dp},\text{in}}) - h_{\text{da}}(T_{\text{dp},\text{out}})], \quad (3.59)$$

where h_{da} denotes the enthalpy of humid air per kilogram of dry air.

Because this formulation neglects the enthalpy carried away by the condensate term ($\dot{m}_{w,\text{out}} h_{w,\text{out}}$), the energy leaving the system through the condensed water is not explicitly included in the balance. This simplification can lead to a small overestimation of the total heat transferred to the flash tank water. Nevertheless, the approach provides a robust and computationally efficient representation of the heat transfer process for system-level dynamic simulations.

Expansion valve

The specific enthalpy of water vapor remains constant through the expansion valve due to isenthalpic throttling. Therefore, the enthalpy at the inlet and outlet of the valve are equal:

$$h_{\text{exp,in}} = h_{\text{exp,out}} \quad (3.60)$$

The mass flow rate through the expansion valve is controlled by adjusting the effective flow area. The mass flow rate of water through the valve can be calculated using [33]:

$$\dot{m}_{\text{exp}} = C_D A_D [2\rho_{\text{exp,in}} (p_{\text{exp,in}} - p_{\text{exp,out}})]^{0.5} \quad (3.61)$$

where C_D is the discharge coefficient, A_D is the effective flow area, $\rho_{\text{exp,in}}$ is the refrigerant density at the valve inlet, and $p_{\text{exp,in}}$ and $p_{\text{exp,out}}$ are the pressures upstream and downstream of the valve, respectively.

At steady state, the mass flow rate through the expansion valve equals the compressor mass flow rate.

Fryer

The oil temperature in industrial fryers typically decreases along the fryer length: hot oil is supplied at the inlet and cooler oil leaves at the outlet. Detailed heat and mass transfer inside an individual french fry are complex [8] [15], so for system-level energy modelling a simplified approach is required.

Instead of modelling the fryer as a plug-flow reactor (PFR) with spatially varying temperatures, a single continuously stirred tank reactor (CSTR) representation is proposed. In this simplified model, the oil entering the fryer is fixed at 177 °C, and the oil leaving the fryer is maintained at 160 °C. The CSTR approximation treats the outlet temperature as the volume-averaged oil temperature inside the fryer. This simplification has two main consequences:

- Advantage: The CSTR introduces thermal inertia (lumped capacitance), so the dynamic response of the fryer e.g., how quickly the oil temperature changes after a disturbance or variation in product flow can be represented without spatial discretization. This makes transient simulations of control strategies feasible and computationally inexpensive.
- Drawback: The outlet temperature from the CSTR is the average oil temperature in the vessel, not the temperature at the physical end of a real fryer. Therefore, the model cannot capture spatial temperature gradients along the fryer length or the detailed effects of residence-time distribution on product quality.

In the simulation, the required heating power is calculated based on the given product mass flow. Controllers will be used to maintain the desired inlet and outlet temperatures at 177 °C and 160 °C, respectively.

Energy balance:

$$\underbrace{\rho V c_p \frac{dT}{dt}}_{(1)} = \underbrace{\dot{m} c_p (T_{\text{in}} - T)}_{(2)} - \underbrace{k_f \dot{m}_f(t)}_{(3)} \quad (3.62)$$

1. Represents the rate of change of internal thermal energy stored in the oil (thermal inertia of the control volume).
2. Represents the net heat flow associated with oil entering and leaving the fryer. It accounts for energy entering at T_{in} and exiting at T , and therefore reflects the influence of mass flow on temperature dynamics.
3. Represents the heat removed by the product being fried, where $\dot{m}_f(t)$ is the mass flow of product and k_f the adequate energy consumption per unit mass.

Simple steam pipe model

To enable control of the expansion valve and the subcooled region of the condenser, a dynamic calculation of the pressure downstream of the compressor is introduced through an additional control-volume element. This control volume represents the combined internal volume of the steam piping after the compressor and the condenser steam volume. The control volume is assumed to have a constant volume and to contain only steam, with two-phase effects neglected.

The pressure within this control volume is determined from a transient mass balance and is governed by the ideal gas law. Pressure dynamics arise from the imbalance between mass inflow and mass outflow, allowing mass accumulation within the control volume. Consequently, the mass inflow and outflow are not required to be equal at each time step. The mass inflow is prescribed by the compressor, which is modelled as a volumetric machine and therefore imposes a mass flow rate based on its displacement and operating conditions. The mass outflow is determined by the expansion valve and is modelled as a function of the valve opening and the pressure difference between the valve inlet and outlet.

Due to water injection in the compressor, the steam temperature within the control volume is assumed to remain close to the saturation temperature corresponding to the local pressure. Furthermore, pressure variations within the control volume are assumed to be relatively small, allowing the use of a simplified thermodynamic description.

This pressure-holdup model enables dynamic coupling between compressor operation, valve position, and downstream pressure. As a result, valve control actions can be explicitly represented in the simulation while maintaining a physically consistent and computationally efficient description of the system's transient behaviour.

4

Steady state analysis results

Chapter 4 presents the conceptual and general design of the proposed heat recovery and delivery system, in which the various heat-exchange and process components introduced in Chapter 2 are integrated into a single installation. Chapter three discusses the models used for the steady-state calculations. Subsequently, the amount of heat that can be recovered from the fryer is determined, and the amount of heat that can be supplied back to the fryer is evaluated based on the proposed design. Several design iterations are then proposed to improve system performance. Finally, a sensitivity analysis is carried out to assess the influence of key parameters on the overall installation.

4.1. Basic cycle design

The topology of the reference concept is illustrated in Figure 4.1, while the corresponding control parameters associated with the equipment shown in the figure and governing the system performance are summarized in Table 4.1. The justification references provided in the table refer to the specific sections of this thesis in which the selected parameter values are motivated and derived. The cycle begins at the fryer, where raw potato is converted into the final product. The energy requirement of the fryer is determined by the mass flow of incoming raw potato, its moisture and dry-matter content, and the temperature difference between inlet and outlet product streams, as shown in table 3.1. The frying oil loop supplies the thermal energy required to meet the fryer's heat load.

A fan transports the frying vapor from the fryer to the heat recovery unit. Two condenser configurations are considered for recovering heat from this vapor:

1. **Spray condenser in combination with a plate heat exchanger**
2. **Shell and tube condenser**

From the perspective of this steady-state calculation, the only meaningful difference between the configurations lies in how the temperature approach is defined for the spray condenser, the plate heat exchanger, and the shell-and-tube condenser. Based on section 2.2, the shell-and-tube condenser is identified as the more robust condenser option.

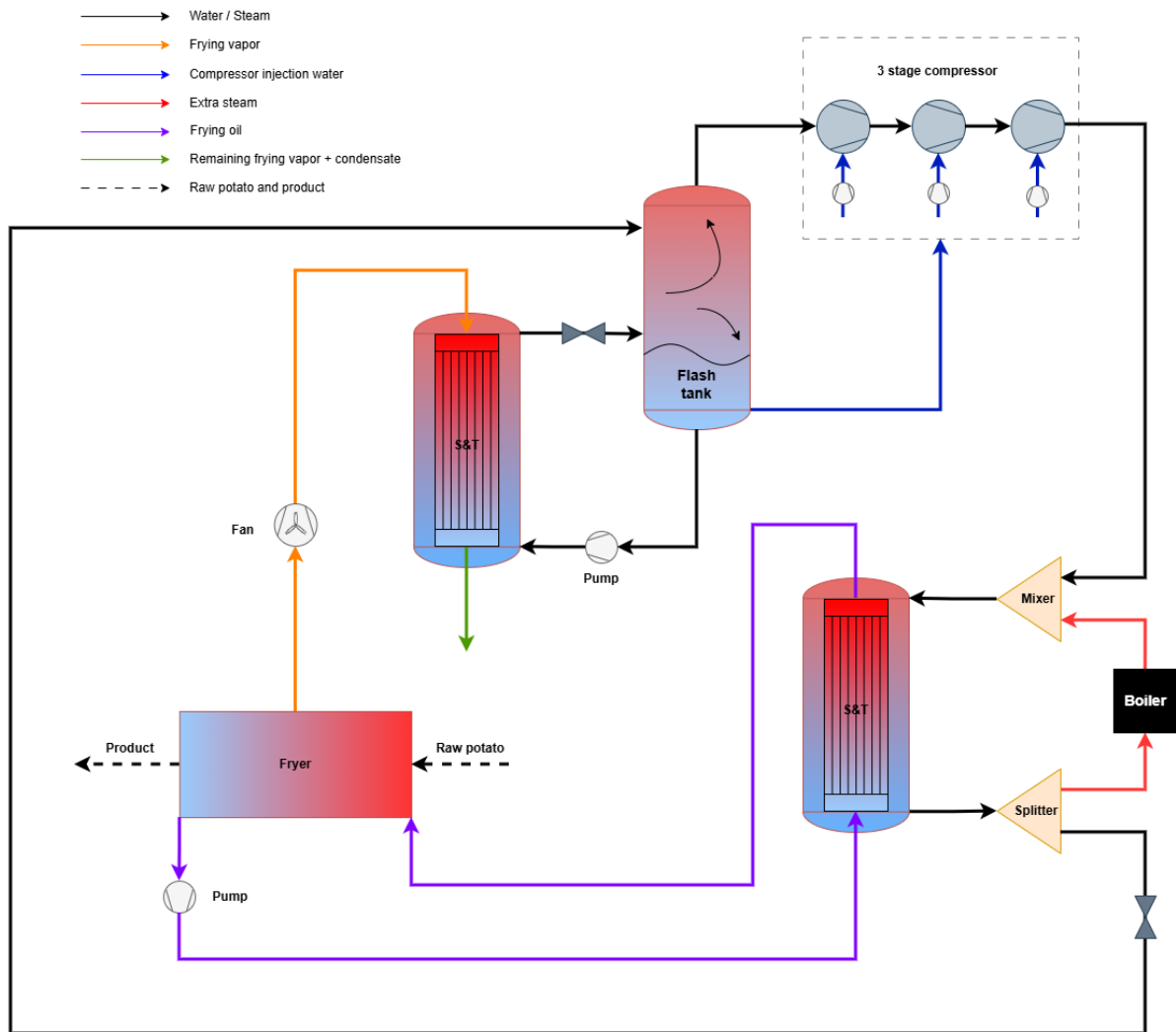


Figure 4.1: Overview of the basic cycle design with material streams and equipment

In the shell-and-tube condenser, the frying vapor is cooled and partially condensed. A combination of remaining frying vapor and condensate leaves the heat recovery unit. The vapor from the fryer heats the water drawn from the flash tank, after which it is returned to the flash tank, where it partially flashes into steam.

The steam compressor then withdraws the generated steam. In consultation with Atlas Copco, it was determined that a three-stage screw compressor configuration is the most realistic for this application. This corresponds to a single stage pressure ratio of approximately 3.17, based on the temperature levels presented in Table 4.1. According to Atlas Copco, the minimum allowable inlet pressure for the compressor is 0.312 bar, which directly determines the operating pressure in the flash tank. Water injection is used to control the discharge temperature and maintain steam saturation throughout compression, except at the exit, where the maximum superheat is limited to 30 °C, according to Atlas Copco.

The high-pressure steam is then routed to the heat exchanger, where it heats the frying oil from 160 °C to 177 °C. As the steam condenses in the heat exchanger with a selected subcooling of 10 °C, the condensate is routed back to the flash tank, where it partially flashes.

The values used for the steady state calculation for the reference concept are shown in Table 4.1.

Table 4.1: Control parameters used for the thermal system configuration.

Control Parameters				
Component	Parameter	Value	Unit	Justifying sec.
Fryer	Product mass flow	7	kg/s	4.1
	Frying vapor temperature	110	°C	2.1.3
	Dew point temperature	87.5	°C	2.1.3
	Oil inlet temperature	177	°C	2.1.3
	Oil exit temperature	160	°C	2.1.3
	Heat load	8161	kW	3.1.2
Flash tank	Pressure	31200	Pa	4.1
	Temperature	70	°C	4.1
Frying vapor HEX	Approach temperature	5	°C	2.2
MVR	Exit pressure	1002810	Pa	4.1
	Saturated temp. out MVR	180	°C	4.1
	Isentropic efficiency	0.7	–	2.4
	Number of stages	3	–	4.1
	Amount of superheated	30	°C	4.1
Frying oil HEX	Subcooled temperature	170	°C	4.1

The performance metrics are summarized in Table 4.2. Approximately 4 MW of heat is recovered from the frying vapor. The steam generated from this recovered heat, together with the steam produced by the returning hot condensate in the flash tank, provides a total thermal input of 6572 kW. This value is based on the thermal energy associated with the mass flow of generated steam and the heat released during cooling from superheated steam at 210 °C to subcooled water at 170 °C.

However, this configuration cannot fully meet the fryer's total heat demand. Consequently, an additional heat input of 1589 kW is required. As illustrated in Figure 4.1, this deficit is supplied by steam from an auxiliary boiler. The required boiler steam flow is determined from the additional thermal energy demand of the fryer and the enthalpy difference between the superheated steam leaving the final compressor stage and the subcooled condensate at the outlet of the frying oil condenser. This supplementary steam is subsequently extracted from the system in the liquid phase after condensation, as indicated by the splitter.

The total exergy destruction of the system amounts to 908 kW. The exergy destruction of the individual components is evaluated using the specific flow exergy definition and the steady-state exergy balance given in Eqs. 3.34 and 3.35, respectively. The kinetic and potential exergy contributions are neglected in this analysis, as the associated terms are small and their influence on the overall exergy balance is negligible compared to the enthalpy and entropy terms. The oil condenser is not included in this calculation because no entropy data were available for the frying oil. The contributions of the individual components to the total exergy destruction are shown in Fig. 4.2. The compressor stages account for the largest share of the exergy destruction. A significant amount of exergy is also destroyed in the expansion valve connecting the oil condenser and the flash tank due to the substantial pressure drop, which results in a considerable loss of energy quality.

Performance metrics	
Parameter	Ref
Heat load fryer	8161 kW
Pelek	2497 kW
Heat recovered from fryer	4075 kW
Heat delivered	6572 kW
Extra heat needed	1589 kW
COP _{MVR}	2.63
Exergy destruction	908 kW

Table 4.2: Performance metrics of the system

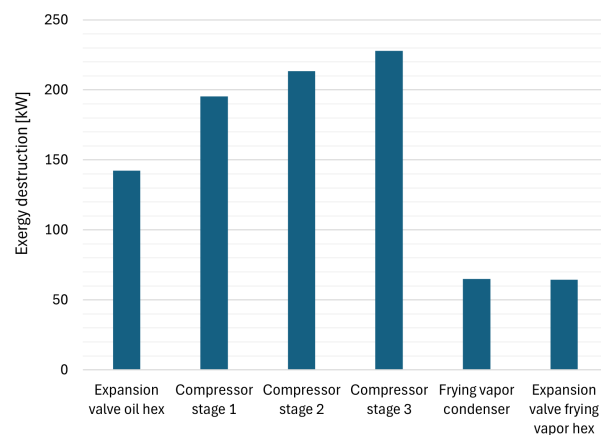


Figure 4.2: Overview of the basic cycle design with material streams and equipment

Based on the reference concept shown in Figure 4.1 and the performance metrics summarized in Table 4.2, several parameters are identified as having a significant impact on the system performance. These include the dew-point temperature of the frying vapor, the temperature in the flash tank, the approach temperatures in the shell-and-tube frying vapor condenser, and the compressor isentropic efficiency.

4.2. Sensitivity analysis

4.2.1. Effect of temperature flash tank

The flash tank temperature strongly affects the COP_{MVR} by governing both the compressor operating conditions and the extent of heat recovery from the frying vapor. Lower flash tank temperatures result in lower saturation pressures and larger temperature lifts, which increase the compressor power demand. In addition, enhanced heat recovery in the frying vapor condenser increases the thermal load on the compressor. Over the investigated temperature range, these combined effects lead to a gradual decrease in COP_{MVR} as the flash tank temperature decreases. Simultaneously, the increased heat recovery reduces the difference between the heat delivered by the MVR system and the required process heat, thereby lowering the demand for supplementary heating, as illustrated in figure 4.3. At a flash tank temperature of 65 °C, the MVR system is able to supply nearly the entire heat demand of the fryer, while achieving a COP_{MVR} of approximately 2.5.

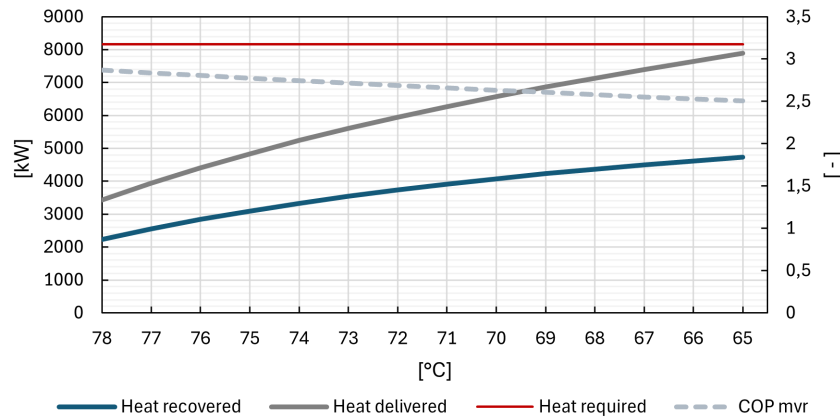


Figure 4.3: Impact of flash temperature on vapor-liquid separation and energy recovery.

4.2.2. Effect of isentropic efficiency of steam compressor

The isentropic efficiency of the compressor determines the compressor power demand and, consequently, the heat delivered by the system. A lower isentropic efficiency increases irreversible losses during compression, resulting in higher electrical power input and increased superheating at the compressor outlet. In the present system configuration, this excess superheat is absorbed by injected water, which lowers the discharge temperature and generates steam that can be condensed at a controlled pressure, thereby increasing the heat delivered by the installation, as shown in figure 4.4. Centrifugal compressors generally exhibit higher isentropic efficiencies than screw compressors (Section 2.4). While this corresponds to a more efficient compression process in terms of electrical power input, it also results in a lower compressor power level and therefore a reduced amount of heat delivered. When recoverable heat is not in excess of the process heat demand, and when the comparison is limited to isentropic efficiency and delivered heat, this trade-off favors the use of a screw compressor, as a larger fraction of electrical input power is converted into useful process heat.

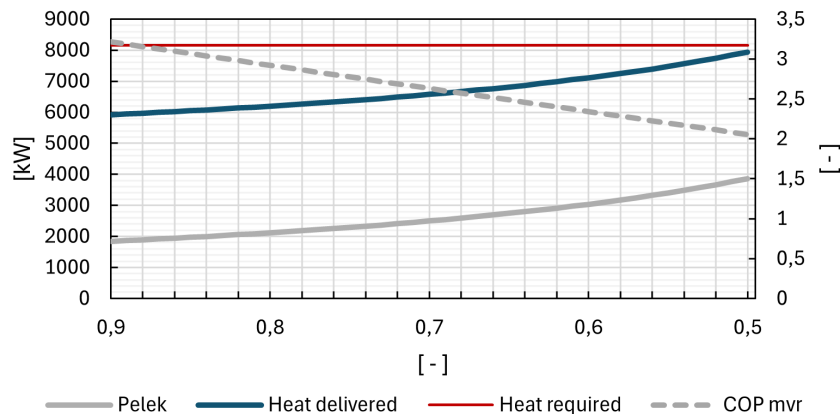


Figure 4.4: Effect of isentropic efficiency on mechanical vapor recompression performance.

4.2.3. Effect of temperature approach in the heat exchanger

The temperature approach in the shell-and-tube condenser influences the amount of heat that can be recovered from the frying vapor, as defined in Section 3.1. A smaller approach temperature increases the recoverable heat but requires a larger heat exchanger surface and results in higher pumping power. While the approach temperature does not affect the COP_{MVR} , it significantly impacts the level of heat recovery. For the investigated operating conditions, a reduction of the approach temperature by $5^{\circ}C$ leads to an increase in recovered heat of approximately 1000 kW. This highlights the need for an economic trade-off analysis between the additional investment and operating costs of a larger heat exchanger and the benefit of increased heat recovery.

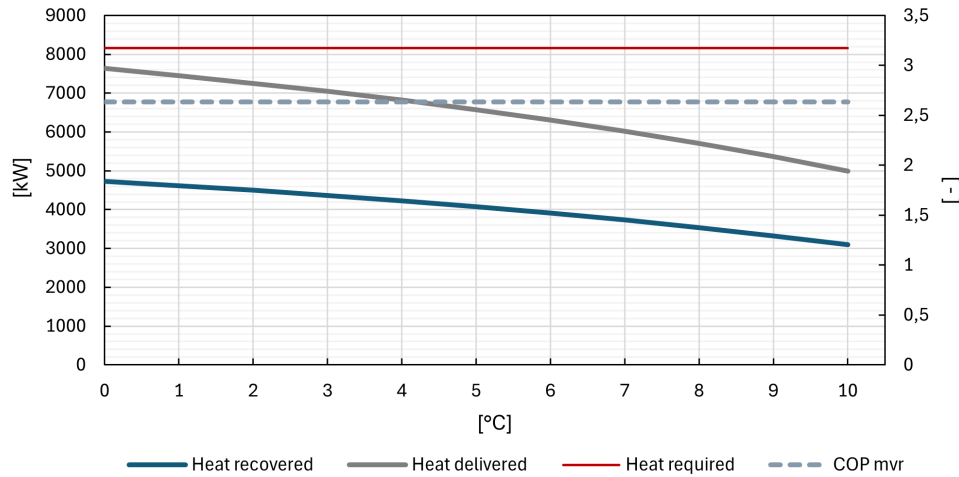


Figure 4.5: Effect of plate heat exchanger approach temperature on system performance.

4.2.4. Effect of dewpoint temperature frying vapor

The dew point temperature strongly influences the amount of heat that can be recovered from the frying vapor. In the present design, the minimum condenser outlet temperature of the frying vapor is constrained by the flash tank temperature, as this represents the lowest temperature at which the vapor can theoretically leave the condenser in the absence of a heat pump. The COP_{MVR} remains constant, while the level of heat recovery varies with the dew point temperature. Increasing the dew point temperature enhances the amount of recoverable heat from the frying vapor and reduces the requirement for supplementary energy input. As shown in Figure 4.6, a dew point temperature of approximately $93^{\circ}C$ allows sufficient heat to be recovered and upgraded by the MVR system to meet the total heat demand of the fryer.

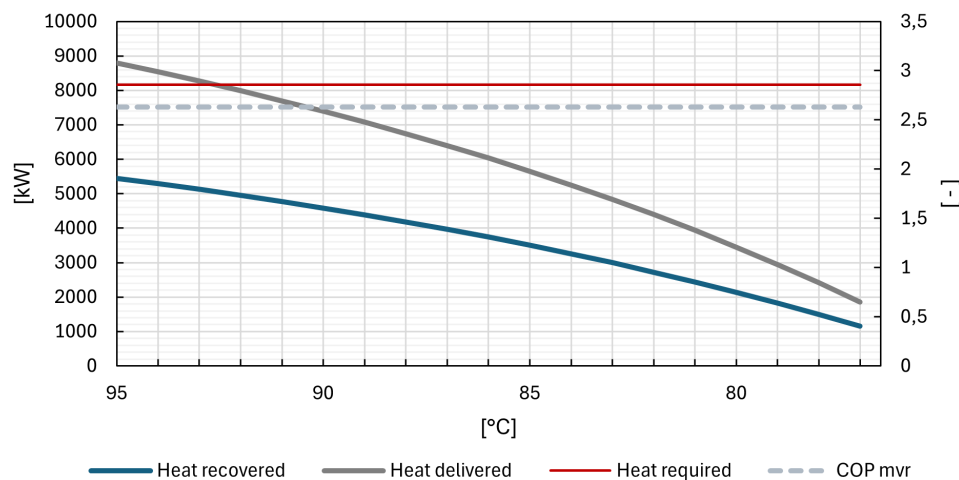


Figure 4.6: Influence of frying vapor dew point temperature on heat recovery characteristics.

4.3. Improved concepts

Based on the sensitivity analysis, several opportunities to improve system performance and increase the amount of heat recovered from the frying vapor have been identified. Since the recovered heat directly determines the amount of heat that can be delivered back to the fryer, these improvements are of key importance.

First, the system should be designed to maximize the dew-point temperature of the frying vapor entering the installation. The analysis shows that a decrease in the dew-point temperature from 90 °C to 85 °C results in an approximate reduction of 1 MW in recovered heat, indicating a strong sensitivity to this parameter.

Second, reducing the approach temperature in the frying vapor condenser increases heat recovery, as the exiting frying vapor and condensate approach the flash tank temperature. Since the flash tank represents the lowest temperature level in the current configuration, lowering its temperature further enables additional heat recovery. However, this requires an increase in the heat exchanger surface area, leading to a higher capital investment. As discussed in Section 2.2, an optimum exists between the capital cost and the resulting energy savings.

A promising option to achieve this is the integration of a closed-compression heat pump, as discussed in Section 2.3. By cooling the flash tank water to a lower temperature level, additional heat can be absorbed from the frying vapor. At the same time, the mechanical work supplied to the heat pump generates additional steam, thereby further increasing the total recoverable heat for the fryer.

Based on the findings in Sections 4.1 and 4.2, three alternative concepts have been generated:

1. A concept in which the utilization of a heat pump supplies the full heat load of the fryer.
2. A concept that focuses on increasing the COP_{MVR} and reducing the exergy destruction over the expansion valve connecting the oil condenser with the flash tank through the use of intermediate flash tanks.
3. A concept proposed by a senior engineer at Atlas Copco, which aims to decrease the size of the first compressor due to the large specific volume of steam at the flash tank pressure.

The concept improvements are implemented with the spray condenser and plate heat exchanger setup, based on the temperature approaches listed in Table 4.3. In comparison with the shell-and-tube version, the frying vapor leaves the condenser approximately 1 °C higher, resulting in a slight difference in heat recovery, as summarized in the results table presented after all improved concepts. The justifying sections refer to the sections where the values used are explained and justified.

It should be noted that all improvements are also feasible for the shell-and-tube configuration. Additionally, the amount of extra heat required is based on the same methodology as presented in the basic design in Section 4.1.

Table 4.3: Control parameters used for the thermal system configuration.

Control Parameters				
Section	Parameter	Value	Unit	Justifying sec.
Plate heat exchanger	Plate HEX pinch	1	°C	2.2
Spray condenser	Spray pinch	5	°C	2.2
Heat pump	Temperature glide sink	6.5	°C	4.3.1
	Temperature glide source	7.0	°C	4.3.1
	Second law efficiency	0.4	–	4.3.1

4.3.1. Concept A - Focus on total heat supply utilizing a heat pump

The design of this concept focuses on supplying the fryer's total heat demand by utilizing energy recovered from the frying vapor. The cycle incorporates a heat pump that extracts heat from the water exiting the plate heat exchanger (I) $71\text{ }^{\circ}\text{C} \rightarrow$ (II) $64\text{ }^{\circ}\text{C}$. This recovered heat is upgraded and used to heat water from the flash tank (III) $70\text{ }^{\circ}\text{C} \rightarrow$ (IIII) $77\text{ }^{\circ}\text{C}$. The lower-temperature water leaving the heat pump is then circulated back to absorb additional heat from the frying vapor, completing the cycle. For the heat pump, temperature glides of $7\text{ }^{\circ}\text{C}$ are assumed for both the heat source and the heat sink. These values are chosen because they result in a total delivered energy that exactly matches the heat demand of the fryer and are commonly reported for heat pump source temperature glides, as discussed in Section 2.3. The corresponding temperature lift yields a second-law efficiency of 0.4 [28]. This effectively means that, compared to the reference concept, the frying vapor can be cooled an additional $7\text{ }^{\circ}\text{C}$, and the corresponding recovered heat, together with the mechanical power supplied by the compressor of the compression heat pump, is used to generate the extra steam. Compared with the reference model (of the spray condenser setup), the combined heat delivered through the upper heat exchanger of the heat pump and the plate heat exchanger is 5060 kW, whereas in the reference case it is 3945 kW.

Larger temperature glides at the source and sink are also possible, which would further cool the water and increase heat recovery from the frying vapor. According to the literature, the maximum allowable temperature rise in recirculated cooling water is limited to approximately 30 °C [22]. However, in such cases, it is more advantageous to use two heat pumps in series. This configuration improves the COP of each heat pump by reducing the temperature difference between the evaporator and condenser, minimizing exergy losses. Additionally, the operating pressures within the heat pumps can be better matched. It also allows the use of refrigerants better suited to the temperature profiles of the source and sink. It should be noted that, compared to the reference concept, this approach offers greater flexibility. The additional heat pump cycle, using a refrigerant from Table 2.6, enables precise control of the amount of heat recovered from the frying vapor. Comparing the temperatures stated above with the possible refrigerants in Table 2.6, five refrigerants are suitable for this temperature profile, including R717, a natural refrigerant.

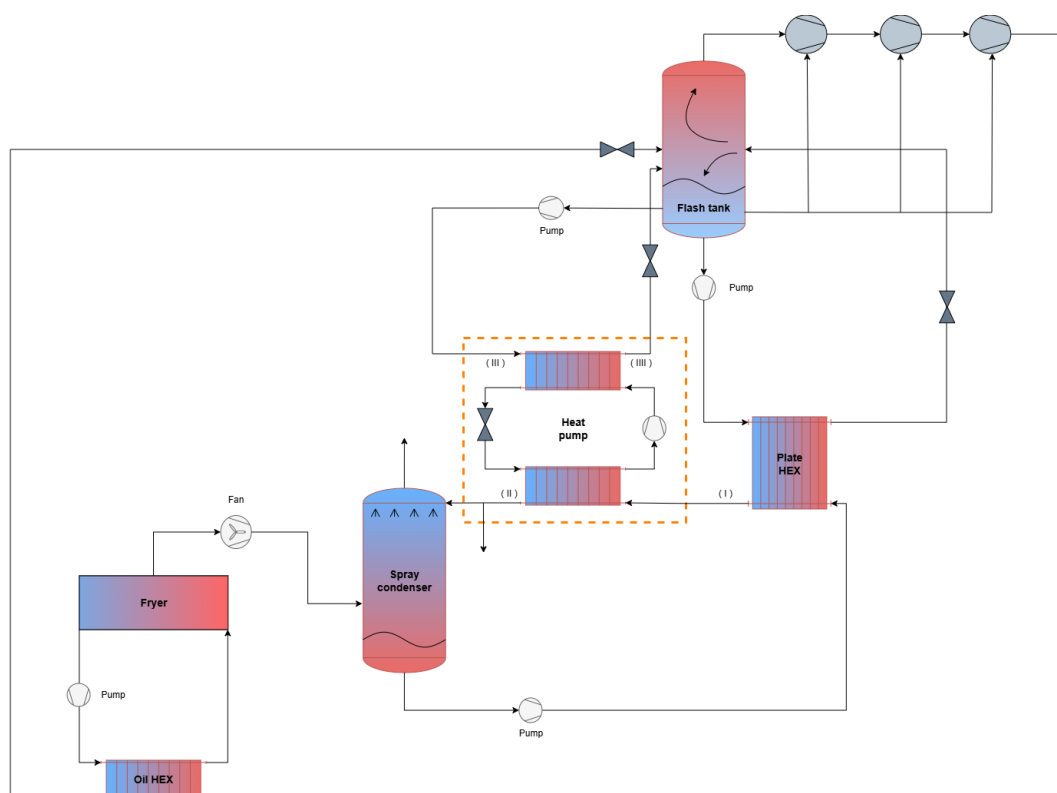


Figure 4.7: Visualization of concept A: Emphasis on maximizing total heat supply using a heat pump.

4.3.2. Concept B - Focus on increasing the COP

This concept focuses on increasing the COP and reducing exergy destruction. In the reference case, a significant amount of exergy is destroyed at the valve located after the frying oil condenser, where the condensate flows back into the flash tank. The pressure drop across this valve is approximately 10 bar. The steam generated during this flashing process must then be compressed again to 10 bar, which requires substantial work.

Since the reference case essentially operates as a standard heat pump, it is helpful to consider alternative designs that are already applied in practice. One commonly discussed concept in literature introduces an intermediate flash tank [29] [42] [46]. Using this intermediate flash tank, the high-pressure condensate is first flashed to the pressure prior to the final compression stage. This prevents the steam from passing through the first two compression stages and reduces the required compression work. The remaining condensate can then be flashed again to the pressure of the previous stage, and so on. The COP of this concept has increased from 2.63 to 2.82 compared with the reference concept. When comparing the exergy destruction of the expansion valve in the reference case with that of the combined valves in the proposed concept, the reference system exhibits an exergy destruction of 138 kW, whereas the combined valves in the new concept result in only 40 kW.

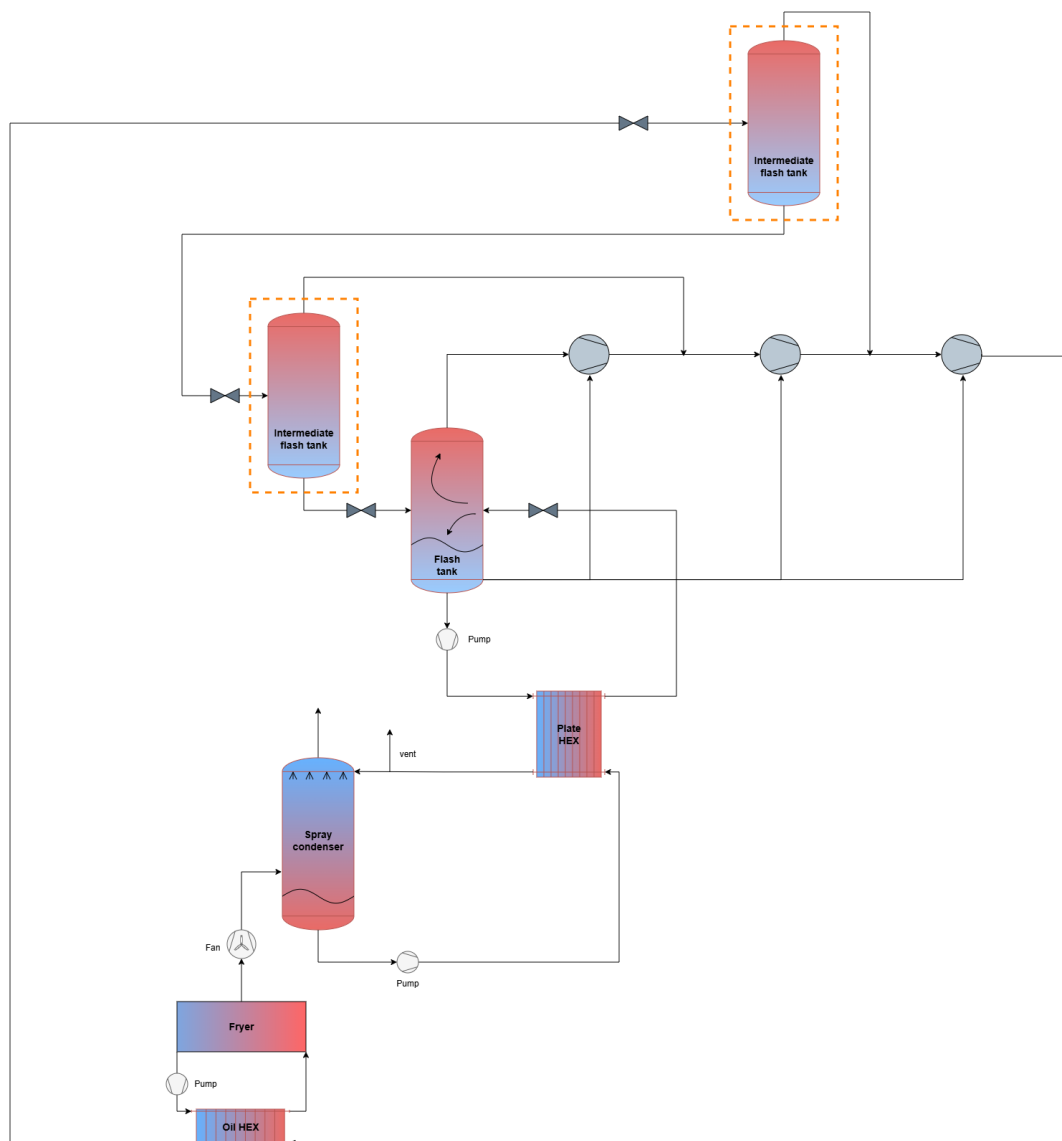


Figure 4.8: Visualization of concept B: Emphasis on increasing the COP.

4.3.3. Concept C - Focus on reducing the size of the first compressor stage

The last concept focuses on reducing the load on the first compressor stage. As presented previously in Table 2.7, steam exhibits a very high specific volume at sub-atmospheric pressures. Consequently, for a given heating capacity, large volumetric flow rates must be compressed, requiring a large first-stage compressor.

In collaboration with Atlas Copco, a potential solution was developed: introducing an additional flash tank enables flashing to occur at a higher pressure than the minimum reference pressure of 31.200 Pa. This reduces the required mass flow through the first compressor from 2.23 kg/s (reference case) to 1.51 kg/s for the same delivered heat, thereby substantially decreasing the workload of the first compression stage. However, this modification also results in a relatively small pressure ratio for the first compressor compared to the second and third stages.

The distribution of compressor work and compression ratios for both the reference system and the proposed concept are presented in Table 4.4.

Table 4.4: Compressor work and pressure ratios for the reference case and proposed flash tank concept.

Compressor Stage	Reference Case		Concept Case	
	Work [kW]	Compression ratio [-]	Work [kW]	Compression ratio [-]
Stage 1	670	3.18	96	1.32
Stage 2	796	3.18	1017	4.94
Stage 3	950	3.18	1297	4.94

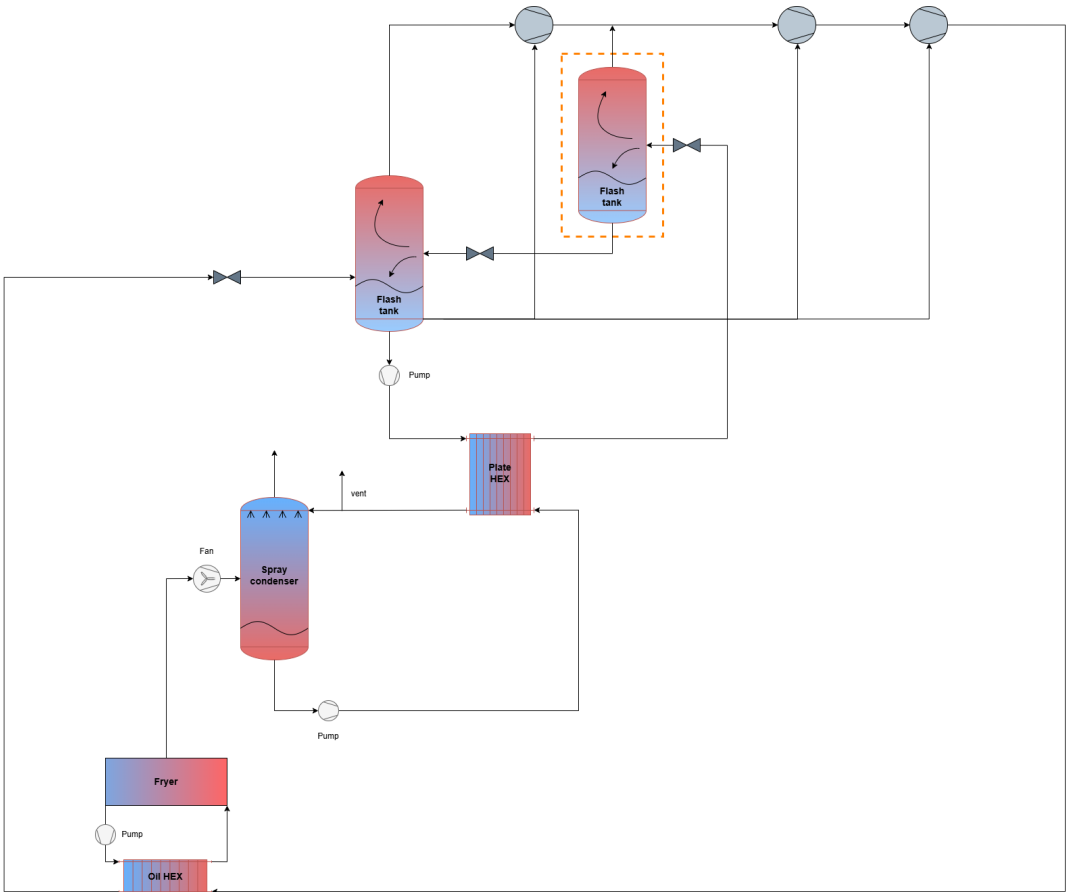


Figure 4.9: Visualization of concept C: Emphasis on reducing the size of the first compressor stage.

4.3.4. Summary

A summary of the different system concepts is presented in Table 4.6. In addition to the reference concept utilizing the shell-and-tube heat recovery configuration, all alternative concepts are evaluated using the spray condenser and plate heat exchanger configuration. The case denoted as Ref* corresponds to the reference concept implemented with the spray condenser and plate heat exchanger instead of the shell-and-tube design. Due to the slightly different temperature approaches of these two configurations, a small difference in the amount of recovered heat is observed.

The concept of incorporating a compression heat pump can fully meet the fryer's heat demand. This is achieved through enhanced heat recovery from the frying vapor, combined with additional work input from the heat pump compressor.

The concepts with intermediate flash tanks reduce exergy destruction and increase the overall system COP. However, these improvements come at the expense of an increased demand for external steam and additional energy input.

Table 4.5 summarizes the meaning and design focus of each evaluated system configuration. The technical comparison between the reference configuration (Ref) and the alternative condenser configuration (Ref*) is discussed in Section 2.2.

Table 4.5: Overview of evaluated system configurations and their primary design focus.

Case	Description
Ref	Reference concept utilizing a shell-and-tube condenser for heat recovery.
Ref*	Alternative reference concept utilizing a spray condenser in combination with a plate hex.
A	Concept focusing on maximizing total heat supply by incorporating a heat pump.
B	Concept focusing on increasing the overall coefficient of performance (COP).
C	Concept focusing on reducing the size and load of the first compressor stage.

Table 4.6: Performance Metrics for reference and alternative system configurations. All energy and exergy values are expressed as rates in kW.

Performance Metrics						
Parameter	Unit	Case				
		Ref	Ref*	A	B	C
Energy sum fryer	kW	8161	8161	8161	8161	8161
Heat recovered from fryer	kW	4075	3945	4880	3945	3945
Pelex	kW	2497	2417	3281	2166	2412
Heat delivered	kW	6572	6361	8161	6111	6356
Energy added to fulfill demand	kW	1589	1799	0	2050	1804
COP_{MVR}	–	2.63	2.63	2.49	2.82	2.64
Exergy destruction	kW	908.25	881.46	–	716.82	882.75

5

Equipment Sizing

Based on the steady-state results for the general design, the equipment will be sized accordingly. Table 3.3 summarizes all equipment being sized and specifies the method used for each piece of equipment. Subsequently, the necessary input parameters for each piece of equipment are provided, along with the objective of the sizing calculations.

Design objectives

For each unit, the sizing procedure provides several key thermal design parameters. These include the required heat-transfer area (A) and the overall heat-transfer coefficient (U), which together characterize the unit's heat-transfer performance. The thermal inventory, defined as the liquid hold-up volume on each side, is also determined and is necessary for understanding start-up behavior and system dynamics. In addition to the liquid hold-up, the metal mass of the tube bundle contributes to the system's thermal inertia, influencing how quickly the unit responds to temperature changes during start-up and shut-down.

5.1. Flash tank

5.1.1. Role of the flash tank as a buffer

The flash tank in the system also functions as a buffer vessel, providing pressure stability during transient disturbances in the thermal energy balance. In this configuration, two main energy inflows enter the flash tank: the heated water leaving the frying vapor condenser and the returning hot water from the oil condenser. At the same time, the compressor extracts energy from the tank by withdrawing vapor (steam) from the system. Disturbances in these flows could cause pressure fluctuations.

One drawback of pressure fluctuations is the occurrence of the shrink-and-swell phenomenon, which is commonly observed in steam drums and flash tanks. Variations in pressure affect the size and volume of steam bubbles present within the liquid phase. As a result, the measured liquid level exhibits an inverse and non-intuitive response to pressure changes [47] [48]. The presence of steam below the liquid level in the drum causes the shrink-and-swell phenomenon, which complicates accurate level control. In addition to this apparent level change, a second physical effect occurs concurrently. An increase in drum pressure leads to enhanced condensation of steam within the drum, thereby increasing the actual liquid water inventory inside the drum. Consequently, the observed level response is the combined result of vapor bubble compression or expansion and changes in liquid volume due to pressure-dependent condensation effects.

The shrink-and-swell behavior can be distinguished as follows:

- **Swell:** When the flash tank pressure decreases, vapor bubbles within the liquid expand, temporarily increasing the effective liquid level. This behavior is typically observed during sudden increases in compressor extraction or reduced inflow.

- **Shrink:** When the flash tank pressure increases, vapor bubbles are compressed and partially condensed, temporarily lowering the effective liquid level. This effect is commonly observed during sudden decreases in compressor extraction or increased inflow.

To describe the system's resistance to rapid pressure changes, the inertia of the flash tank is expressed by means of two time constants. These time constants are defined based on the operating pressure and a predefined lower pressure limit.

The first time constant assumes that the mass in the flash tank remains constant, corresponding to the condition in which no flow leaves the tank via the steam compressor and no flow enters from the oil condenser. In this case, the maximum thermal power transferred through the frying vapor condenser, corrected for the additional heat contribution from the oil condenser's hot-water return, is applied. The resulting time constant represents the time required for the tank pressure to fall from the operating pressure to the lower-pressure limit under this maximum heat-input condition.

The second time constant considers the opposite scenario: no additional heat is supplied to the flash tank, while the steam compressor operates at maximum speed. Here, the pressure decreases due solely to steam removal. The corresponding time constant is the time required to reach the predefined lower pressure limit starting from the operating pressure.

Based on the parameters listed in Table 5.1 and the methodology for gas–liquid separators described by Towler and Sinnott in *Chemical Engineering Design* (3rd ed.) [43], the resulting time constants are calculated to be 12.75 s and 13.18 s. These values indicate that the flash tank provides sufficient buffering capacity to ensure stable operation, even during load fluctuations. Moreover, in the event of a sudden disturbance, the available response time is adequate to detect and counteract a pressure decrease before critical limits are reached.

Table 5.1: Flash tank design conditions, calculated dimensions, and time constant parameters

Parameter	Symbol	Value
Design conditions		
Steam mass flow rate	$\dot{m}_{\text{steam}}$	2.68 kg/s
Vapor volumetric flow rate	V_V	10.50 m ³ /s
Vapor density at 31.200 Pa	ρ_V	0.198 kg/m ³
Liquid water density at 31.200 Pa	ρ_L	977.73 kg/m ³
Liquid holdup time	t_h	60.0 s
Demister pad used	$u_s = u_t$	–
Calculated dimensions		
Vessel diameter	D_V	1.65 m
Liquid volume in vessel	V_L	4.65 m ³
Total vessel height	H_V	5.0 m
Liquid fill percentage		34 %
Time constant calculation		
Maximum system power	P_{max}	6,000,000 W
Operating pressure	p_{op}	31.200 Pa
Lower pressure limit	p_{low}	26.200 Pa
Time constant	τ_1	12.75 s
Time constant	τ_2	13.18 s

Detailed drum boiler dynamics with supporting experimental data are presented in [47].

5.2. Condensers

5.2.1. Oil condenser

Table 5.2 presents the design data for the heat exchanger, based on the steady-state calculations described in Chapter 4. Table 5.3 provides a summary of the key design and performance parameters for the heat exchanger. Considering the reference ranges for typical velocities on both the shell and tube sides, as well as the allowable pressure drops summarized in Table 3.7, the design values for the shell-and-tube condenser are consistent with standard design guidelines. Furthermore, the overall heat transfer coefficient for the fouled surface is in good agreement with literature values for shell-and-tube exchangers. As reported in Table 2.5, for units using steam to heat organic or light oils, they generally fall between 500 and 900 W/m²·K [22].

Table 5.2: Process data for the heat exchanger

Process data	Tube side	Shell side	Unit
Flow rate	200	3.78	kg/s
Inlet temperature	160	210	°C
Outlet temperature	177	170	°C
Allowable pressure drop	50	40	kN/m ²
Fouling resistance	0.0002	0	m ² ·K/W

Table 5.3: Summary of key design and performance parameters for the heat exchanger

Parameter	Value	Unit
Heat transfer parameters		
Heat exchanged	8103	kW
Heat transfer area	348	m ²
Mean temperature difference	8	K
Overall heat transfer coefficient (fouled)	588	W/(m ² ·K)
Overall heat transfer coefficient (clean)	693	W/(m ² ·K)
Film coefficient (shell side)	8406	W/(m ² ·K)
Film coefficient (tube side)	784	W/(m ² ·K)
Exchanger geometry		
Exchanger type	BEM	–
Position	Vertical	–
Number of tubes	969	–
Tube length	6000	mm
Tube inner diameter	14.83	mm
Tube outer diameter	19.05	mm
Shell outer diameter	849	mm
Process data		
Inlet velocity (shell side)	7.19	m/s
Inlet velocity (tube side)	1.46	m/s
Calculated pressure drop (shell side)	0.37	bar
Calculated pressure drop (tube side)	0.16	bar
Thermal inertia parameters		
Mass of tube bundle	5855	kg
Mass of frying oil in tubes	800	kg
Shell side volume	2.27	m ³

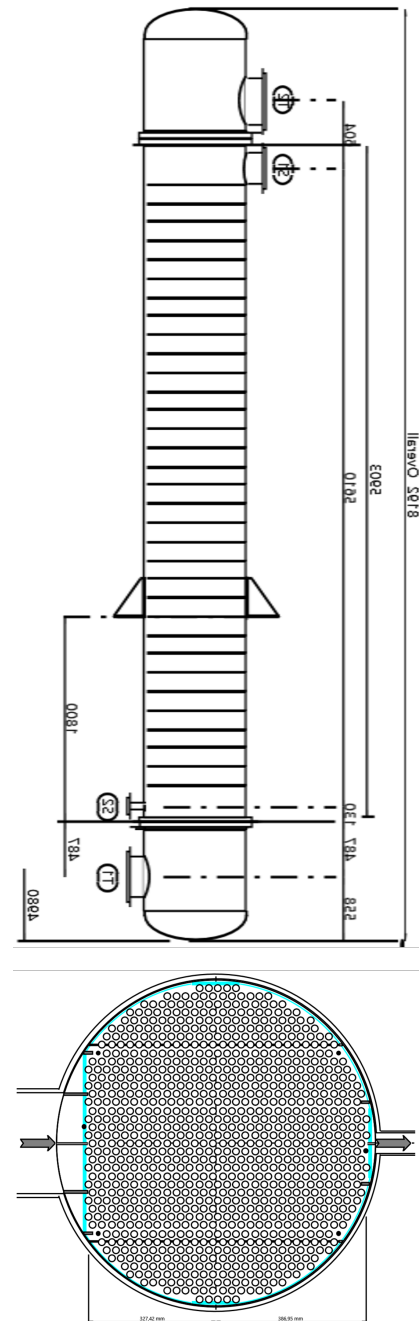


Figure 5.1: Sized oil HEX

5.2.2. Frying vapor condenser

Table 5.4 presents the design data for the heat exchanger, based on the steady-state calculations described in Chapter 4. Table 5.5 provides a summary of the key design and performance parameters for the heat exchanger. Considering the reference ranges for typical velocities on both the shell and tube sides, as well as the allowable pressure drops summarized in Table 3.7, the design values for the shell-and-tube condenser are consistent with standard design guidelines. Furthermore, the overall heat transfer coefficient for the fouled surface is in good agreement with literature values for shell-and-tube exchangers. As reported in Table 2.5, for units condensing aqueous vapors to heat water, typical overall heat transfer coefficients range between 1000 and 1500 W/m²·K [22].

Table 5.4: Process data for the heat exchanger

Process data	Unit	Tube side	Shell side
Flow rate	kg/s	4.39	80
Inlet temperature	°C	130	70
Outlet temperature	°C	76	84
Allowable pressure drop	bar	0.2	0.5
Fouling resistance	m ² ·K/W	0.0002	0

Table 5.5: Summary of key design and performance parameters for the heat exchanger

Parameter	Value	Unit
Heat transfer parameters		
Heat exchanged	5070	kW
Heat transfer area	470	m ²
Mean temperature difference	9.29	K
Overall heat transfer coeff. (fouled)	1208	W/(m ² ·K)
Overall heat transfer coeff. (clean)	1751.7	W/(m ² ·K)
Film coefficient (shell side)	8030	W/(m ² ·K)
Film coefficient (tube side)	2506	W/(m ² ·K)
Exchanger geometry		
Exchanger type	BEM	–
Position	Vertical	–
Number of tubes	1192	–
Tube length	6600	mm
Tube inner diameter	14.83	mm
Tube outer diameter	19.05	mm
Shell outer diameter	949	mm
Process data		
Inlet velocity (shell side)	0.55	m/s
Inlet velocity (tube side)	32.89	m/s
Calculated pressure drop (shell)	0.20	bar
Calculated pressure drop (tube)	0.06	bar
Thermal inertia parameters		
Mass of water in shell	2865	kg
Mass of tube bundle	7574	kg

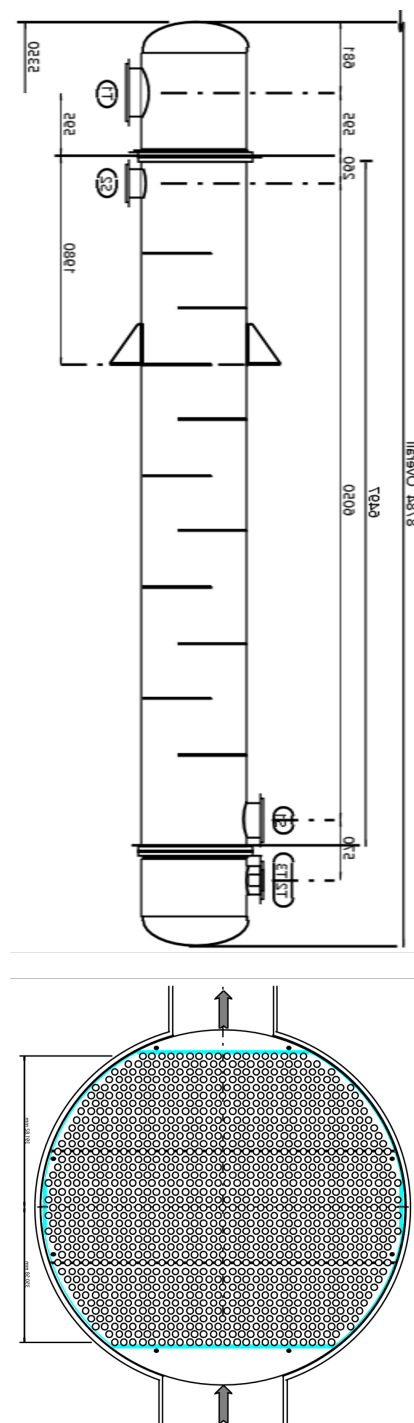


Figure 5.2: Sized frying vapor HEX

5.3. Thermal inventory of system components

In Table 5.6, the thermal inventory of the system is presented. For each component, the relevant mass is multiplied by its specific heat capacity to obtain the energy required to increase the temperature of that component by 1 K. For the flash tank loop, the water in the flash tank and the frying vapor condenser account for the largest portion of the thermal inventory. In contrast, the contribution of the equipment's steel structure is comparatively small. This demonstrates that most of the energy stored in this section of the system resides in the water rather than in the surrounding metal. In the oil loop, the thermal inventory is dominated by the mass of oil in the fryer.

These calculated values indicate the system's heat content during normal operation and the energy required to start the system from cold or to shut it down. They also provide a basis for comparing the relative contributions of different components to the installation's total heat content.

In addition to the stored thermal energy, the maximum mass flow rate of water and oil through the system determines the maximum rate at which heat can be transported in the condensers. The product of the mass flow rate and the specific heat capacity indicates the maximum thermal energy that can be transferred per unit temperature difference. When considered together with the thermal inventory values, this allows comparison of the heat stored in the components with the rate at which heat can be exchanged through fluid circulation, based solely on physical properties and measured flow capacities.

Table 5.6: Thermal Capacitance and Flow Characteristics

Equipment	Type	$C = m \cdot c_p$ [MJ/K]	Frac. [-]	$\dot{m}_{max}$ [kg/s]	$\dot{m}_{max} c_p$ [MW/K]
Flash Tank Loop					
Flash Tank	Water	19.40	0.55	—	—
Frying Vapor Condenser	Steel	3.79	0.11	—	—
Frying Vapor Condenser	Water	12.00	0.34	80	0.334
Oil Loop					
Fryer	Oil	9.73	0.68	—	—
Oil Condenser	Oil	1.60	0.11	200	0.400
Oil Condenser	Steel	2.93	0.21	—	—

6

Dynamic simulation results

Dynamic simulations are performed to investigate the transient behaviour of the heat recovery system and to support the design of an effective control strategy. The simulations provide insight into how the system responds to variations in production rate, allowing for the prediction and optimization of control actions under changing operating conditions. The models are implemented in Matlab using time-dependent mass and energy balance equations for the relevant components.

The dynamic model considers thermal inertia in the flash tank, the fryer, and the connecting pipe sections, as these components contain significant fluid volumes that store thermal energy and influence system transients. In contrast, the compressor and the two heat exchangers are assumed to have negligible thermal inertia and are therefore represented by steady-state relationships. Heat transfer in these steady-state components is modelled using the effectiveness–NTU (ϵ -NTU) method. The compressor is modeled using the same approach as described in Section 3.1.2. This combined approach enables an accurate yet computationally efficient representation of the system dynamics and forms the basis for controller development.

The structure of this chapter is organised as follows. The development of the dynamic component models is presented in the methodology chapter. The control logic is then introduced in Section 6.1, where the controlled process variables and their corresponding actuators are described. In this section, the feedback control strategy is first explained, followed by the combined feedback–feedforward control approach. Next, the operational scenarios under consideration are outlined. Although multiple scenarios are defined, only the results of the most relevant cases are shown, as the steady-state results are demonstrated in the initial feedback-controlled simulation. Finally, Section 6.2 presents and discusses the dynamic simulation results.

6.1.1. Feedback control

The dynamic simulation requires a set of feedback controllers to regulate key process variables throughout the system. Figure 6.1 illustrates the controlled variables (orange dots) and their corresponding actuators (green lines). The following subsections describe the control objectives and the associated control actions.

① Flash Tank Pressure Control

The first process variable is the pressure inside the flash tank. The control objective is to maintain this pressure at the compressor's minimum allowable operating pressure, 0.312 bar. Maintaining a low flash tank pressure maximizes heat recovery from the frying vapor. The actuator is the steam screw compressor, which removes vapor from the flash tank, thereby reducing the pressure inside the flash tank.

② Flash Tank Water Level Control

The second controlled variable is the liquid water level in the flash tank. Since additional water is continuously supplied to the system via compressor water injection and an auxiliary steam line, the level must be maintained constant. This is achieved via a discharge valve that removes excess water. In the simulation, an ideal controller is assumed: all additional water entering the system is removed at the next timestep, maintaining a nearly constant level.

③ Fryer Inlet Oil Temperature Control

The third process variable is the temperature of the oil leaving the oil condenser and flowing back to the fryer. If this temperature becomes too low, additional steam is introduced to the condenser via the auxiliary steam line by adjusting the valve, increasing the amount of heat that can be delivered. This ensures that the fryer receives oil at the required temperature.

④ Condenser Pressure Control

The fourth controlled variable is the downstream compressor pressure and the condenser pressure. This pressure is regulated by the expansion valve, which controls the mass flow rate returning from the high-pressure region to the flash tank. Increasing the mass flow rate decreases the condenser pressure, whereas decreasing the flow rate increases it. The control objective during dynamic operation is to maintain the condenser pressure slightly above 10 bar.

⑤ Subcooled Condensation Area Control

The fifth controlled variable is the area of subcooled condensation in the condenser. A decreasing subcooled region indicates incomplete condensation. When the area falls below a defined threshold, the expansion valve is throttled to reduce the mass flow rate, slightly increasing pressure and ensuring full condensation is maintained.

⑥ Fryer Return Oil Temperature Control

The sixth process variable is the temperature of the oil exiting the fryer and entering the oil condenser. This temperature is regulated through the pump speed. Increasing the pump speed increases the mass flow rate, enabling more heat to be absorbed in the condenser. Under steady-state conditions, the required mass flow rate is determined from

$$\dot{Q} = \dot{m} c_p \Delta T,$$

where $\Delta T = 15^\circ\text{C}$ is the temperature difference between the required inlet and outlet temperature, c_p is assumed constant, and Q depends on the fryer production rate.

⑦ Frying Vapor Dewpoint Temperature Control

The seventh process variable is the dewpoint temperature of the frying vapor entering the frying vapor condenser. Although this aspect is not the primary focus of the thesis, it is assumed that dewpoint temperature is influenced by the fan flowrate, where increasing fan speed is expected to lower the dewpoint. The dynamic simulation uses an ideal controller that maintains the dewpoint at a fixed setpoint.

⑧ Condensate Outlet Temperature Control

The eighth controlled variable is the temperature of the water exiting the frying vapor condenser and returning to the flash tank. To maintain a desired setpoint temperature, the mass flow rate through the condenser is regulated.

⑨ Steam Temperature Control

The ninth process variable is the temperature of the steam exiting the compressor. This temperature must remain at saturated conditions. To achieve this, water is injected into the compressor discharge stream to desuperheat the compressed steam. In the dynamic model, an ideal controller is assumed that ensures steam saturation after compression.

6.1.2. Combined feedback and feedforward control

Within a process control feedback system, significant and rapid disturbances affecting either the process variable (PV) or the system's dynamic response can reduce the effectiveness of feedback control. When disturbances propagate before the controller can react, the process may move far outside its desired operating range, making timely correction to the setpoint difficult. Under such conditions, overall process performance and control accuracy may become unacceptable. [49] [50]

Feedforward control is used to detect and compensate for disturbances before they influence the process, thereby reducing their effect on feedback loop stability. Unlike feedback control, feedforward action does not consider deviations of the process variable from its setpoint. Instead, it responds to measured or estimated disturbances and applies corrective adjustments in advance to minimize their impact.

The principal distinction between feedforward and feedback control can be summarized as follows:

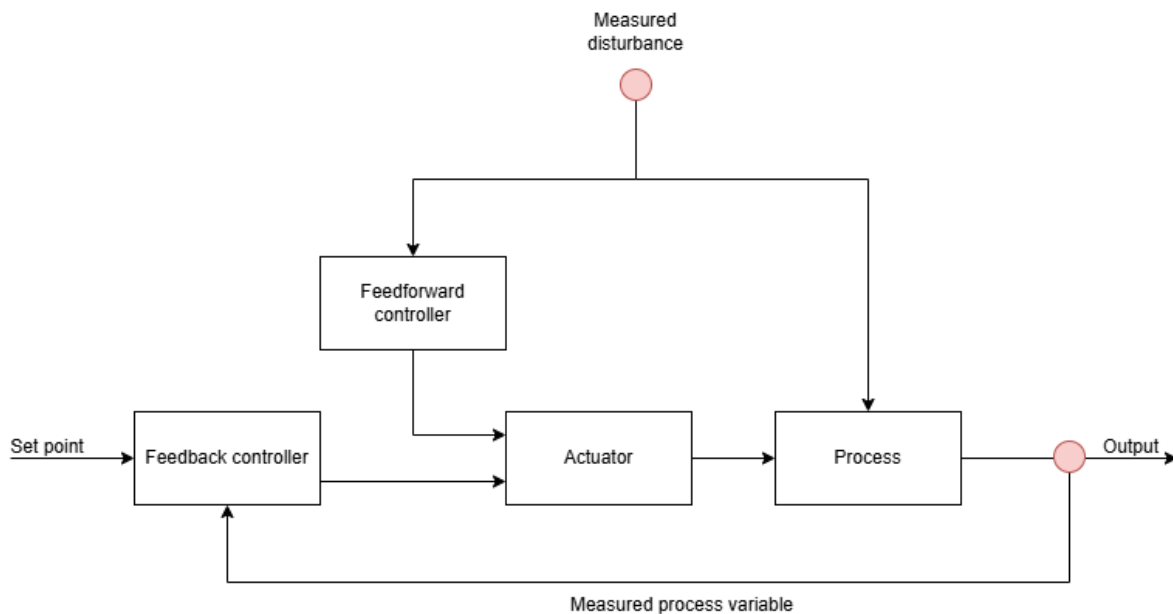


Figure 6.2: Block diagram illustrating a control system combining both feedback and feedforward control strategies.

The frying system employs a combined control strategy to ensure robust regulation. Feedback controllers maintain critical process variables, such as flash tank and condenser pressures, at their setpoints by correcting measured deviations. Simultaneously, a feedforward controller preemptively compensates for the primary disturbance: product throughput. By monitoring the product load, the system adjusts actuators to counteract changes before they impact the frying oil temperature or system pressures. This approach minimizes transient errors and enhances stability under variable operating conditions.

6.2. Dynamic results

6.2.1. Scenario overview

1. **Steady-state at full production**

Objective: Establish the baseline operating point at full production. This concerns the mass flow of all pumps, the compressor speed, and the additional steam required to meet the fryer's energy requirements and temperature setpoints. This establishes reference values and initial conditions for subsequent dynamic scenarios.

2. **Step change in product flow (pressure behind compressor kept constant)**

Objective: apply a step increase or decrease in product mass flow and observe the transient response while maintaining constant pressure in the flash tank.

Key detail: the mass flow out of the compressor plus any externally added steam must equal the mass flow exiting the condenser and re-entering the flash tank through the expansion valve (mass continuity). This is also relevant for the first scenario.

Metrics: transient oil temperatures, compressor power, settling time, and overshoot.

3. **Add steam pipe control-volume (dynamic condenser mass/pressure)**

Objective: include a lumped control volume representing the condenser and downstream piping to account for steam mass accumulation and dynamic pressure behavior. Study how valve position and compressor operation interact to control condenser pressure and the valve mass flow.

Why: reveals challenges in pressure regulation and the risk of insufficient condenser subcooling.

Metrics: condenser pressure transients, valve flow, accumulated mass in the control volume, effects on condenser outlet quality and subcooling margin.

4. **Add feed-forward control (heater/compressor/valve)**

Objective: implement a feed-forward element that uses measured product flow or setpoint changes to preemptively adjust heater power or valve position, in addition to feedback control.

Study: improvement in disturbance rejection performance (reduced overshoot and faster settling), and robustness to modelling uncertainty and measurement noise.

6.2.2. Feed-back control

In the simulation, the system initially operates at full production capacity. At a certain moment, the production load is suddenly reduced to 75% of its original value. As a result of this step change, both the heat load of the fryer and the amount of frying vapor produced decreases. As a result, the frying vapor condenser absorbs less heat, and consequently, less heat is transferred to the flash tank. Since feedback loops control the system, the compressor initially continues to rotate at the speed corresponding to full-load operation, resulting in a temporary decrease in flash tank pressure immediately after the step change. It can be observed that the flash tank pressure (green line) eventually stabilizes again at the original pressure level, showing that the pressure control loop successfully restores the system. The compressor also finds a new steady-state rotational speed.

Approximately 200 seconds are required for the flash tank pressure to be fully restored, and approximately 600 seconds are required after the step change for the target fryer temperatures to be reached again. Initially, the frying oil pump still delivers the higher flow rate associated with full capacity. This results in increased heat transfer within the oil condenser, delivering more heat than necessary: the fryer outlet temperature increases, while the inlet temperature decreases. This effect can be seen in the upper plot of Figure 6.3. As described in Section 6.1, the extra steam supply is regulated based on the temperature of the oil leaving the condenser. Because the inlet temperature temporarily decreases after the step change, the controller adds additional steam, producing a visible spike in the steam supply. The reason for the long recovery time of the fryer temperatures is illustrated in Figure 8.2. Ideally, the green line should follow the pink line exactly. However, during the first 100 seconds after the step change, too much heat is delivered to the fryer, leading to temperature overshoot and delaying the system's reach of its final steady-state condition.

For the pump that regulates the mass flow of the frying oil, a PI controller was implemented. During the simulations, it became clear that a design compromise was required. A low proportional (P) gain resulted in a slow adjustment of the mass flow, which avoided overshoot but produced a relatively slow overall system response. On the other hand, a high proportional gain caused significant overshoot, temporarily reducing the mass flow of frying oil to undesirable levels. This reduction negatively affected the heat transfer capability in the oil condenser, since insufficient oil flow limited heat transfer and hindered proper condensation of the supplied steam. As a consequence, the condenser performance deteriorated, and the overall system stability was reduced.

Through several tuning iterations, suitable proportional and integral gains were determined. With these selected values, the pump response was relatively fast while maintaining minimal overshoot, ensuring sufficient heat-transfer capacity and stable system behavior. These observations indicate that the use of a feed-forward control strategy, in addition to the feedback PI controller, could further improve the transient response by anticipating the effects of production load changes on the oil mass flow.

To control the flash tank pressure, a proportional–integral (PI) controller was initially implemented. However, this configuration resulted in prolonged pressure oscillations. To stabilize the system, a differential action was added, forming a PID controller that effectively damped the oscillations.

An interesting observation followed from the fact that the water flow from the flash tank to the frying vapor condenser was kept constant during the step change. Because the flow of frying vapor decreases while the water flow remains unchanged, the approach temperature changes. This increases the heat transferred, reducing the outgoing water temperature and bringing the exiting frying vapor temperature closer to the ingoing water temperature. This effect is illustrated in Figure 6.4, where the total heat load in the fryer decreases from 8 MW to 6 MW, while the heat transferred in the frying vapor condenser decreases only from 5 MW to 4 MW. This means that a proportionally greater amount of heat is recovered, reducing the additional steam required to meet the fryer heat demand.

The control valve behavior is shown in Figure 6.5. Initially, the valve opening corresponds to full production capacity, after which it is reduced. Two sawtooth oscillations are visible in the subcooled region of the oil condenser plot, caused by the heat exchange area temporarily falling below the threshold value. The controller then reduces the mass flow and increases pressure. The combined dynamics of the flash tank pressure controller, the frying oil pump, the extra steam supply, and the valve control determine the system's transient response.

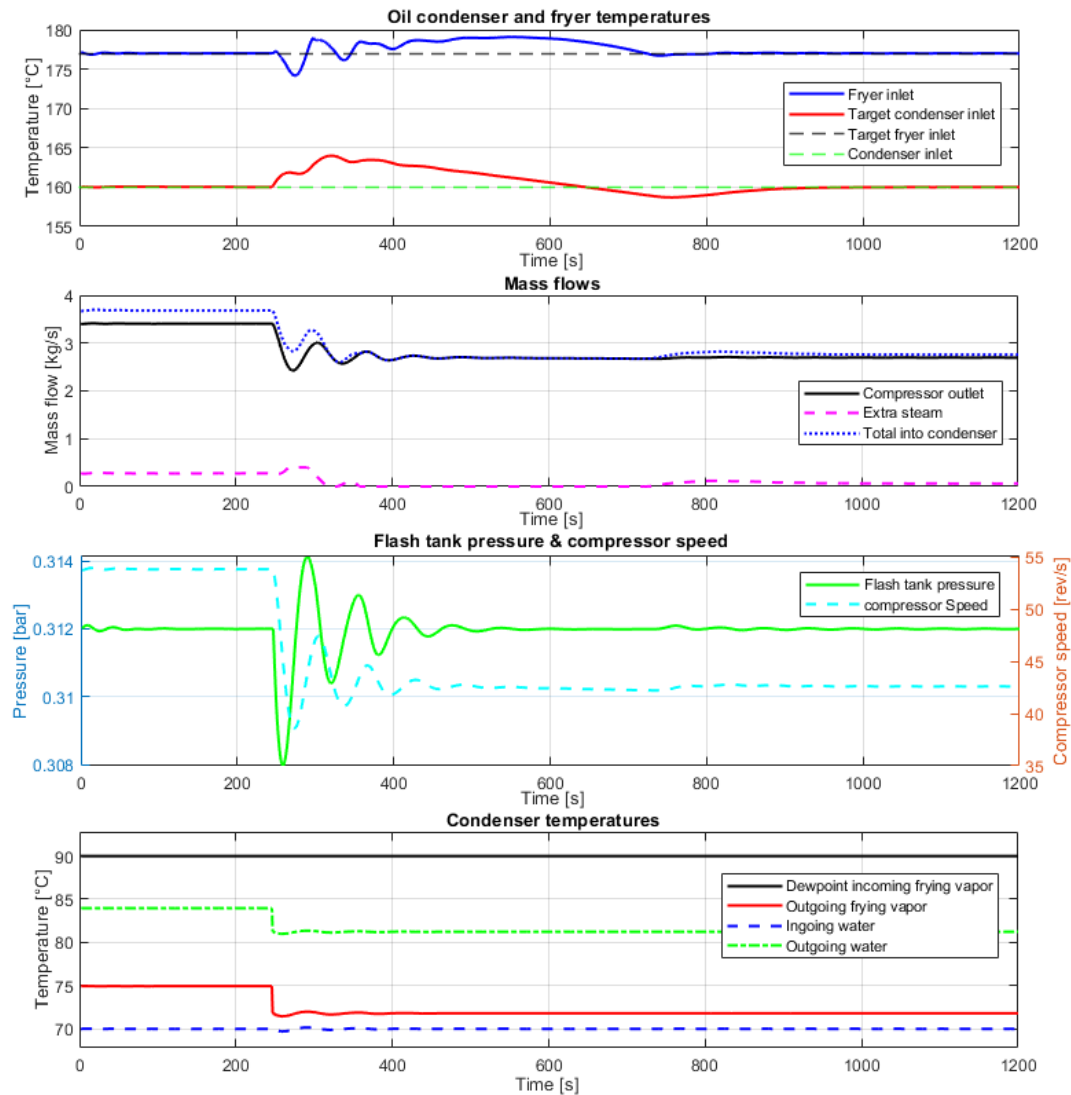


Figure 6.3: Dynamic response of oil temperatures, mass flows, flash tank pressure, and condenser temperatures following a 25% production step-down.

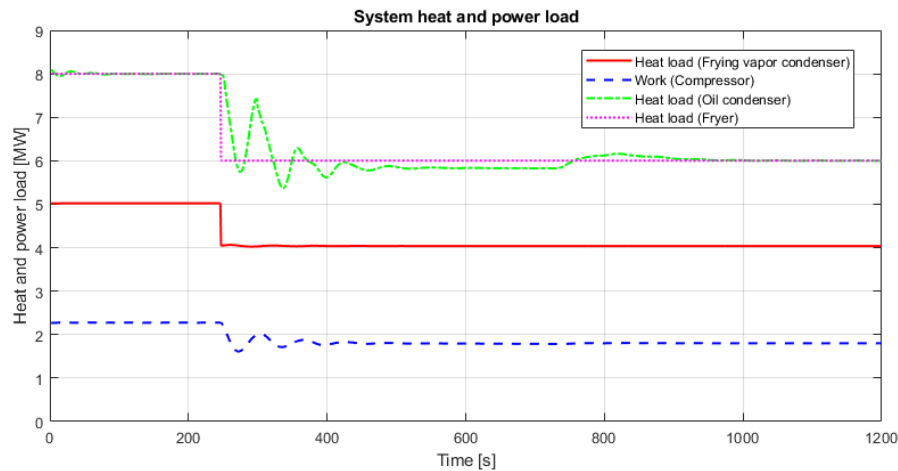


Figure 6.4: System heat load and compressor power during the step change, showing reduced demand and gradual stabilisation of heat transfer.

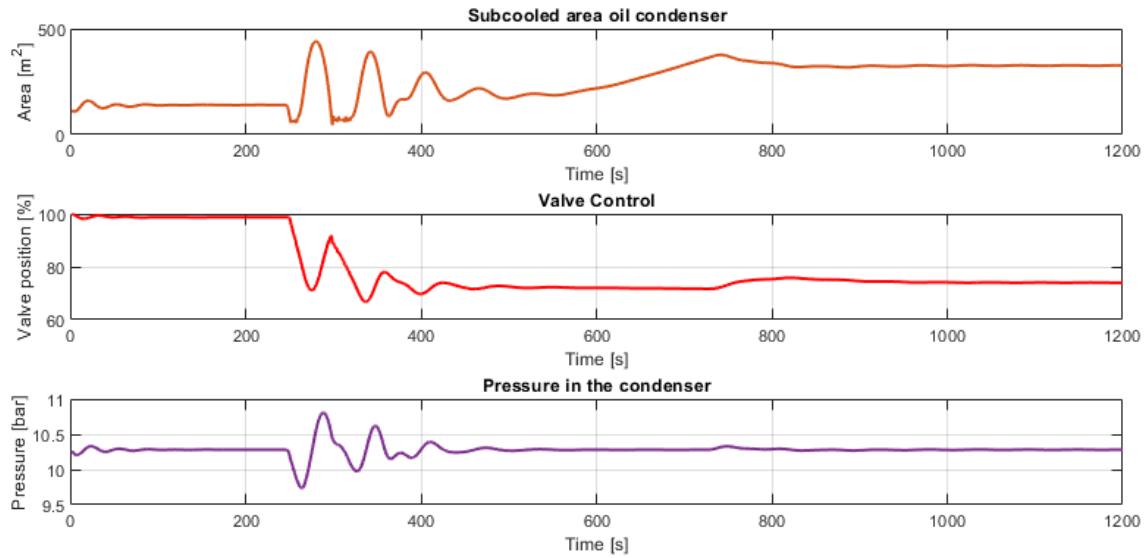


Figure 6.5: Condenser subcooled area, valve response, and high-pressure behaviour after the production step change.

In the presented feedback-control results, the observed crossing of the oil temperature setpoint during load changes indicates that the current control strategy does not yet provide the level of thermal stability required for industrial frying processes. As discussed in Section 2.1.4, oil temperature is a key determinant of product quality, as it directly influences water evaporation rates, crust formation, browning reactions, and oil uptake. Transient temperature deviations may therefore lead to variations in moisture content, texture, color, and oil absorption, ultimately affecting product consistency and consumer acceptance.

For industrial potato processing applications, it is essential that alternative and sustainable heat delivery systems maintain oil temperature levels and stability comparable to those achieved with conventional fossil-fired heating systems. The setpoint crossings observed with feedback-only control demonstrate that further improvements in controllability are required to meet these quality-related constraints.

The results indicate that enhanced thermal stability can be achieved by enhancing the existing PID feedback structure with a feedforward control action. By anticipating load disturbances, such as variations in production throughput, and proactively adjusting pump flow rates and compressor speed, feedforward control can reduce transient deviations and prevent excessive temperature overshoot or undershoot. The combined use of feedback and feedforward control is therefore a prerequisite for maintaining stable frying conditions, preserving product quality, and enabling industrial acceptance of the proposed sustainable heating concept.

6.2.3. Feed-forward control

Based on the results obtained from the feedback-controlled simulation, a feed-forward control strategy was also evaluated. In this simulation, the system was subjected to a gradual load change rather than a step disturbance. The production flow rate was linearly reduced from 100% to 75% over 30 seconds, which seems a realistic change duration under practical production conditions. During this transition, the mass flow rates through the oil and water circulation pumps and the compressor speed were adjusted according to the feed-forward control action. After this coordinated adjustment, the standard PID controller resumed normal operation.

The results show that the fryer temperatures closely follow their setpoints throughout the transition, and the flash tank pressure remains stable. This behaviour is illustrated in Figure 6.7, where the heat load of the fryer and the heat load of the oil condenser closely match, with only small perturbations. The temperature profile of the frying vapour condenser, shown in Figure 6.6, remains essentially unchanged during the transition, indicating that the thermal inertia associated with the steel tube bundle is small compared to the overall thermal inventory.

A comparison with the feedback-only simulation shows that, with feed-forward control included, the new steady-state value is reached almost immediately. This suggests that incorporating feed-forward action in the control strategy can reduce the deviation from nominal operating conditions during load changes and help to avoid large disturbances throughout the system.

For practical implementation, it is essential to note that the feed-forward approach requires a reliably measurable process variable to drive the regulating actuators, including the circulation pumps, the compressor, and the auxiliary steam supply. In the present study, the production flow rate was used as the primary measurable variable for feed-forward control.

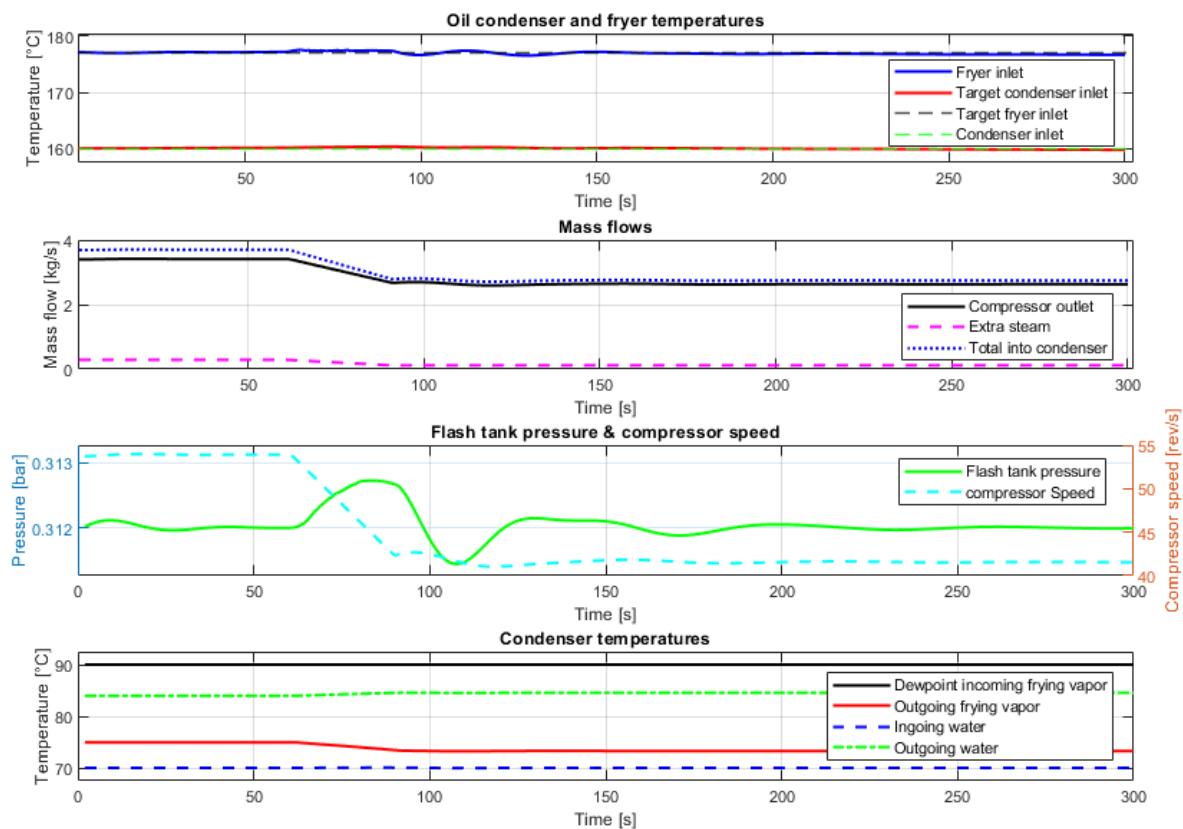


Figure 6.6: Dynamic response of oil temperatures, mass flows, flash tank pressure, and condenser temperatures following a 25% controlled production step-down.

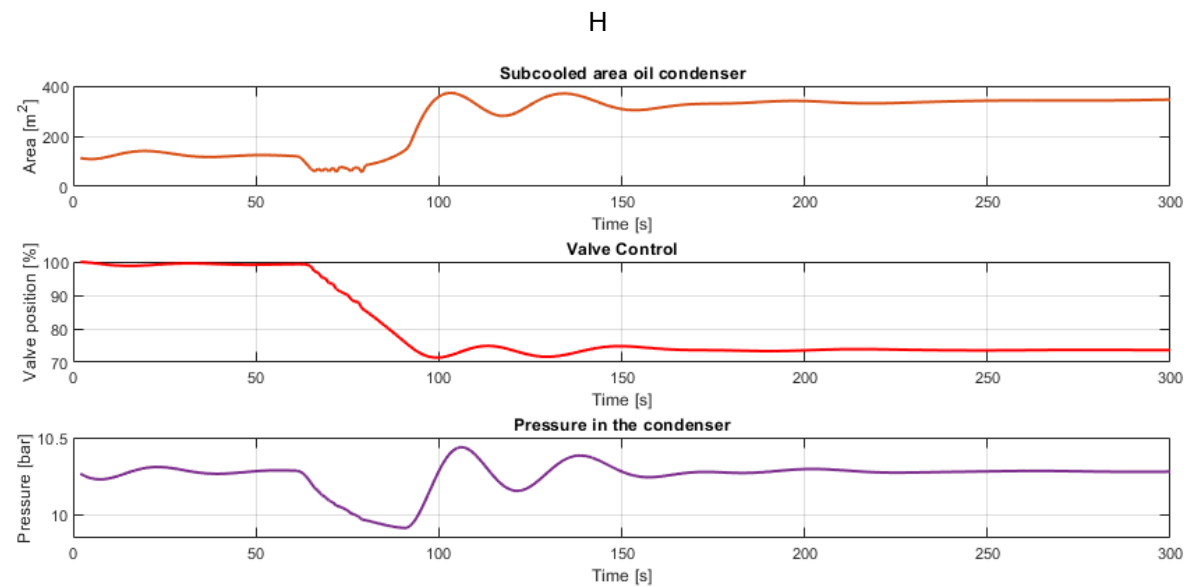


Figure 6.8: Condenser subcooled area, valve response, and high-pressure behaviour after the controlled production step change.

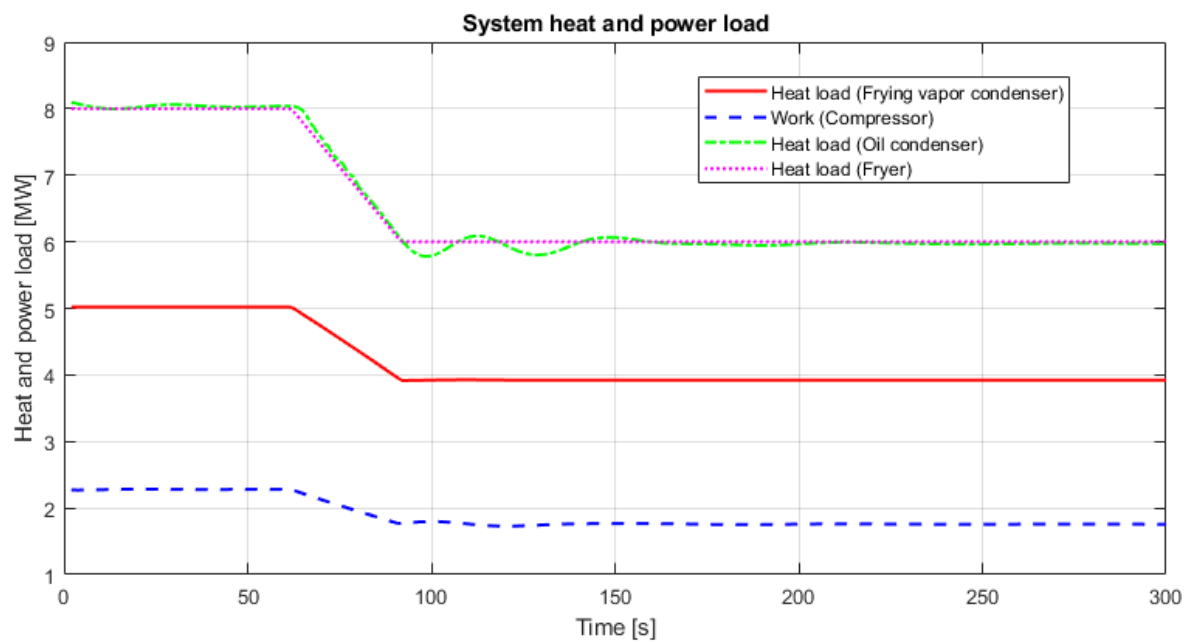


Figure 6.7: System heat load and compressor power during the controlled step change, showing reduced demand and gradual stabilisation of heat transfer.

7

Conclusion

Through a structured methodology comprising literature study, steady-state modelling, equipment sizing, and dynamic simulation, a complete system design was developed to recover latent heat from frying vapor and return it to the fryer as high-pressure steam. The proposed configuration is a closed-loop heat pump system using water as the working fluid, in which low-grade waste heat is upgraded via mechanical vapor recompression and utilized directly to satisfy the fryer's thermal load.

Chapter 2 demonstrated that most of the fryer's energy demand is associated with moisture evaporation in the potato, which exits the system as frying vapor. The recoverable heat and its temperature level depend strongly on the vapor dewpoint, as illustrated in the Q - T diagram. A vertical shell-and-tube condenser is selected as the preferred configuration, as it offers a more robust, practically operable design with fewer components, making it less susceptible to fouling than a spray condenser combined with a plate heat exchanger. The condenser is connected to a flash tank to generate steam and accommodate the large specific volume of vapor at 0.312 bar, making a flash vessel more suitable than a conventional evaporator. To deliver the recovered energy back into the process, the vapor must be compressed from 0.312 bar to 10.2 bar; this can be achieved in three stages using a screw compressor with water injection, which maintains the discharge temperature near saturation while achieving the required pressure ratio.

The basic cycle design demonstrates that significant recovery of frying vapor heat is achievable under realistic industrial operating conditions. For a fryer heat load of 8161 kW and a vapor dew point of 87.5 °C, the reference configuration recovers approximately 4045 kW of latent heat, which, after compression, yields 6572 kW of useful heat delivered back to the fryer. The remaining 1589 kW must be supplied externally. Under these conditions, the installation yields a MVR COP of 2.63. These results confirm that a shell-and-tube condenser with a flash tank and a three-stage screw compressor can recover a substantial share of the frying energy demand.

The sensitivity analysis highlights which parameters most strongly influence performance. Increasing the frying vapor dew point, lowering the flash-tank temperature, and reducing the temperature approach in the condenser each improves recovery by enabling greater heat transfer at appropriate temperatures. The dew point proved especially important: a reduction from 90 to 85 °C lowered the recoverable heat by nearly 1 MW, underscoring the relevance of controlling vapor composition.

The improved concept incorporating a heat pump showed the most promising results. By further cooling the recovered condensate, additional heat can be absorbed from the frying vapor, allowing the system to meet the fryer heat load without relying on external steam. This is achieved by extracting additional heat from the recovered condensate stream and lifting it thermodynamically through the additional heat pump. The resulting temperature range is well-suited for the natural refrigerant ammonia (R717), making this configuration both technically feasible and environmentally advantageous. However, despite its favorable thermodynamic properties and negligible global warming potential, ammonia is classified as toxic and flammable, which necessitates strict safety measures, appropriate system design, and compliance with industrial safety regulations when applied in food-processing environments.

The flash tank provides essential buffering capacity within the system, stabilising pressure during thermal disturbances. The resulting response times of approximately 12 s indicate that the tank can absorb fluctuations before compressor limits are reached, ensuring safe operation during load changes.

The heat exchangers were designed with fouling resistance, cleanability, and reliable condensate drainage in mind. Placing frying vapor and oil streams on the tube side and using a vertical shell-and-tube configuration supports robust operation. At the same time, the selected velocities, pressure drops, and overall heat-transfer coefficients fall within recommended design ranges.

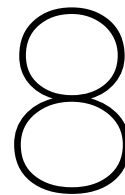
Thermal inventory analysis shows that the dominant heat capacity resides in the flash-tank loop and fryer oil, whereas steel mass contributes minimally. When combined with flow-rate capacity, these values indicate that the installation can store and transport heat effectively, supporting dynamic operation and stable thermal performance.

The feedback-controlled simulation shows that the system can recover from a sudden reduction in production load from 100% to 75%, but the transient response is relatively slow. Approximately 200 seconds are required for the flash tank pressure to return to its setpoint, and approximately 600 seconds are required for the fryer oil temperatures to stabilize. The delay is caused primarily by an initial mismatch between available heat and demand, combined with conservative pump controller tuning to avoid excessive overshoot. Although stable operation is achieved, the results indicate that feedback alone leads to noticeable temperature deviations and a prolonged settling time following a production change.

Introducing combined feedback-feed-forward control substantially improves the system behaviour. When the load change is applied gradually and actuator setpoints are adjusted pre-emptively based on the production flow rate, both fryer temperatures and flash tank pressure track their setpoints closely. The transition to the new operating point occurs almost immediately, with minimal disturbance in the condenser and oil loop response. Comparison with the feedback-only simulation shows that feed-forward control significantly reduces overshoot, limits thermal drift, and shortens the settling time.

In addition, it is essential that stable frying oil temperatures are maintained throughout operation. Fluctuations in oil temperature directly affect the rate of water evaporation from the product and strongly influence oil uptake during frying, both of which are key determinants of final product quality. For potential users in the potato processing industry, it is therefore critical to demonstrate that the proposed sustainable heat delivery system can maintain process conditions comparable to those achieved with conventional fossil-fired heating systems. Ensuring equivalent thermal stability is a prerequisite for guaranteeing unchanged product quality and for facilitating industrial acceptance of the technology.

This study therefore demonstrates that feed-forward action, when combined with the existing PID feedback structure, enhances controllability during load changes and prevents large transient deviations. Successful implementation requires a reliable disturbance indicator, such as production throughput, to drive anticipatory adjustments of pump flow and compressor speed.



Recommendation

All developed models should be validated against experimental or published reference data. In the present work, the dynamic models are primarily verified through conservation of mass and energy at component and system level, ensuring thermodynamic consistency. However, no direct validation against experimental measurements or industrial transient data has been performed. Future work should therefore focus on validating the model predictions against measured pressure, temperature, and liquid level trajectories during charging and discharging of the flash tank, as well as during system transients.

Benchmark cases

A possible validation approach for the flash tank model is to reproduce benchmark cases reported in the literature on steam accumulators. Several studies report complete transient simulations with clearly defined vessel dimensions, initial conditions, inlet/outlet mass flow rates, temperatures, and pressures, as well as resulting time histories of pressure, water level, vapour fraction, and temperatures [51] [52]. These datasets can be used to directly compare the predicted pressure evolution, temperature levels, and liquid inventory of the flash tank model used in this thesis.

Non-equilibrium flash tank model

Published steam accumulator models in literature [51] [52] [53] compare equilibrium and non-equilibrium models. The first one is based on the assumption of thermal equilibrium between the liquid and vapour phases. This equilibrium assumption implies that the liquid water and steam in contact have the same saturation temperature at a common pressure, and that any thermodynamic change is accompanied by evaporation or condensation occurring at infinite rates. While this assumption simplifies the modelling effort, it neglects the possibility of deviations from thermal equilibrium during rapid charging or discharging of the flash tank.

In contrast, thermal non-equilibrium models calculate the thermodynamic states of the liquid and vapour phases separately. In these models, the two phases may have different temperatures despite being in direct contact, and evaporation and condensation are described by finite-rate mass transfer processes. The phase change rates are typically governed by relaxation terms, characterised by a relaxation time or relaxation constant that determines how quickly the system approaches equilibrium between the liquid and vapor [53]. The choice of this relaxation parameter significantly influences the predicted transient behaviour of the accumulator. As demonstrated in the literature, non-equilibrium evaporation during discharging results in a lower accumulator pressure than predicted by equilibrium models, while non-equilibrium condensation during charging leads to a higher pressure than predicted under equilibrium assumptions [52]. Consequently, equilibrium-based flash tank models may overestimate the available pressure during discharge or underestimate the pressure rise during charging. In the context of the present work, this implies that the equilibrium flash tank model may overpredict the pressure level available to the downstream screw compressor during rapid transients. An experiment could show if this is relevant or not for a flash tank connected to a screw compressor.

Experiment

Future work should focus on conducting an experimental study using the flash tank–steam compressor setup at Atlas Copco's site in Wilrijk, Belgium. A simplified schematic of a proposed test configuration, including the required instrumentation, is provided in the appendix; many of the necessary sensors are already available in the existing setup. Such an experiment would provide valuable insight into flash tank transients and the dynamic behavior of the steam compressor under realistic operating conditions. In addition, the results could be used to assess the relevance of non-equilibrium models for describing the behavior of a flash tank connected to a steam compressor and could be used to validate the flash tank and steam compressor models.

Economic assessment and cost--performance trade-offs

To assess the industrial viability of the proposed heat recovery and mechanical vapour recompression system, a comprehensive economic evaluation is recommended. While the present work focuses on thermodynamic performance and system dynamics, the feasibility of implementation in an industrial setting ultimately depends on operating and capital costs. An economic assessment should therefore compare the cost of electricity required to drive the compressor with the cost of fossil fuels that would otherwise be used in a conventional steam boiler.

In addition to direct energy costs, the analysis should consider the impact of carbon pricing or avoided CO₂ emissions, as these factors can significantly influence the relative attractiveness of electrified heat supply systems. By relating the achieved coefficient of performance of the MVR system to electricity prices, fuel prices, and potential CO₂ costs, it can be evaluated whether the system provides sufficient economic benefit under current or projected market conditions.

A further recommended analysis is a cost–performance trade-off between heat exchanger capital expenditure and annual energy savings. Larger heat exchangers with smaller approach temperatures enable increased heat recovery and a higher effective system efficiency, but at the expense of increased investment cost. Quantifying the additional recoverable heat over a year and comparing it to the incremental capital cost would allow an objective evaluation of optimal heat exchanger sizing using metrics such as payback time or net present value.

Integration of External Steam Supply in Heat Recovery Systems

For heat recovery concepts that generate low-pressure steam and upgrade it to high-pressure steam using a steam compressor, it is likely that the recoverable heat alone is insufficient to meet the total process heat demand. While a closed-cycle compression heat pump can be added to increase the amount of usable energy, it is also relevant to investigate alternative ways of supplying the remaining heat demand, such as a hybrid system in which a fossil-fuel boiler provides additional steam. Such a configuration may be necessary to achieve an economically attractive solution for the customer. In this context, the main objective of heat recovery should be to maximize the overall COP by delivering as much heat as possible while keeping the electrical energy consumption of the steam compressor as low as possible. At a more detailed level, future work should focus on the integration of an external steam circuit within the heat recovery and steam delivery system, including its coupling, with particular attention to where and how additional steam is added to or removed from the system, and the associated control strategy.

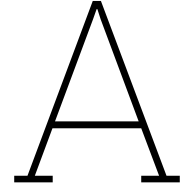
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Time constants flash tank

Model equations

$$E = \dot{M}_{\text{avg,out}} H_{\text{avg,out}} t + M_{\text{water}} H_{\text{water}} + M_{\text{steam}} H_{\text{steam}} \quad (\text{A.1})$$

$$M_0 = \dot{M}_{\text{avg,out}} t + M_{\text{water}} + M_{\text{steam}} \quad (\text{A.2})$$

$$V = \frac{M_{\text{water}}}{\rho_{\text{water}}} + \frac{M_{\text{steam}}}{\rho_{\text{steam}}} \quad (\text{A.3})$$

Eliminate t using the mass equation:

$$t = \frac{M_0 - M_{\text{water}} - M_{\text{steam}}}{\dot{M}_{\text{avg,out}}} \quad (\text{A.4})$$

Substitute the expression for t into the energy equation and rearrange:

$$E = H_{\text{avg,out}} (M_0 - M_{\text{water}} - M_{\text{steam}}) + M_{\text{water}} H_{\text{water}} + M_{\text{steam}} H_{\text{steam}} \quad (\text{A.5})$$

$$E - H_{\text{avg,out}} M_0 = (H_{\text{water}} - H_{\text{avg,out}}) M_{\text{water}} + (H_{\text{steam}} - H_{\text{avg,out}}) M_{\text{steam}} \quad (\text{A.6})$$

Write the two linear equations for M_{water} and M_{steam} :

$$a_1 M_{\text{water}} + b_1 M_{\text{steam}} = c_1 \quad (\text{A.7})$$

$$a_2 M_{\text{water}} + b_2 M_{\text{steam}} = c_2 \quad (\text{A.8})$$

where the coefficients are

$$a_1 := H_{\text{water}} - H_{\text{avg,out}} \quad (\text{A.9})$$

$$b_1 := H_{\text{steam}} - H_{\text{avg,out}} \quad (\text{A.10})$$

$$c_1 := E - H_{\text{avg,out}} M_0 \quad (\text{A.11})$$

$$a_2 := \frac{1}{\rho_{\text{water}}} \quad (\text{A.12})$$

$$b_2 := \frac{1}{\rho_{\text{steam}}} \quad (\text{A.13})$$

$$c_2 := V \quad (\text{A.14})$$

Solution

$$\Delta = a_1 b_2 - a_2 b_1 \quad (\text{A.15})$$

$$M_{\text{water}} = \frac{c_1 b_2 - c_2 b_1}{\Delta} \quad (\text{A.16})$$

$$M_{\text{steam}} = \frac{a_1 c_2 - a_2 c_1}{\Delta} \quad (\text{A.17})$$

Time:

$$t = \frac{M_0 - M_{\text{water}} - M_{\text{steam}}}{\dot{M}_{\text{avg,out}}} \quad (\text{A.18})$$

Explicit forms (substituting coefficients):

$$\Delta = (H_{\text{water}} - H_{\text{avg,out}}) \frac{1}{\rho_{\text{steam}}} - \frac{1}{\rho_{\text{water}}} (H_{\text{steam}} - H_{\text{avg,out}}) \quad (\text{A.19})$$

$$M_{\text{water}} = \frac{(E - H_{\text{avg,out}} M_0) \frac{1}{\rho_{\text{steam}}} - V (H_{\text{steam}} - H_{\text{avg,out}})}{\Delta} \quad (\text{A.20})$$

$$M_{\text{steam}} = \frac{(H_{\text{water}} - H_{\text{avg,out}}) V - \frac{1}{\rho_{\text{water}}} (E - H_{\text{avg,out}} M_0)}{\Delta} \quad (\text{A.21})$$

Assumptions (used in the derivation):

- We are considering a *screw compressor*, which is a volumetric compressor. Therefore the *volumetric flow rate* is treated as constant (the compressor displaces a fixed volume per revolution).
- Because the steam density changes with operating pressure (and with the lower pressure limit), the *mass flow* associated with that fixed volumetric flow changes. In practice the inlet and outlet steam densities are known for the two pressures of interest (begin and end densities).
- To account for the density variation while using a single mass flow in the energy balance we choose an *average mass flow rate* (the given $\dot{M}_{\text{avg,out}}$) together with a corresponding *average enthalpy* $H_{\text{avg,out}}$. These assumptions linearize the energy balance and make the system solvable with the available measurements.
- All properties (H and ρ) are treated as known constants for the operating point; transient effects, heat losses and non-ideal mixing are neglected.

B

Dynamic model fryer

The fryer is modelled as a well-mixed control volume containing a constant volume of oil. Heat losses to the environment and internal heating elements are neglected. The oil temperature is assumed spatially uniform and varies only in time due to convective mixing with the inlet oil stream and heat extraction by the fries.

The total oil mass is

$$m_o = \rho V,$$

where ρ is the oil density and V the oil volume. Applying an unsteady energy balance to the oil yields

$$\frac{d}{dt} (m_o c_p T) = \dot{m} c_p T_{in} - \dot{m} c_p T - \dot{Q}_{fries},$$

with $\dot{m}$ the oil mass flow rate, T_{in} the inlet oil temperature, c_p the specific heat capacity of the oil, and $\dot{Q}_{fries}$ the heat extracted by the fries. The fries heat draw is modelled as

$$\dot{Q}_{fries} = k_f \dot{m}_f,$$

where k_f is a constant heat extraction coefficient and $\dot{m}_f$ is the mass flow of fries in the fryer.

Substituting $m_o = \rho V$ and dividing by $\rho V c_p$ leads to the governing first-order differential equation for the oil temperature:

$$\frac{dT}{dt} = \frac{\dot{m}}{\rho V} (T_{in} - T) - \frac{k_f \dot{m}_f}{\rho V c_p}.$$

This equation can be written in linear form as

$$\frac{dT}{dt} = -aT + b,$$

with

$$a = \frac{\dot{m}}{\rho V}, \quad b = aT_{in} - \frac{k_f \dot{m}_f}{\rho V c_p}.$$

Assuming a and b remain constant over a timestep Δt , the exact discrete-time solution is

$$T_{k+1} = T_k e^{-a\Delta t} + \frac{b}{a} (1 - e^{-a\Delta t}).$$

In the limiting case of zero mass flow ($a = 0$), the temperature evolution reduces to

$$T_{k+1} = T_k + b\Delta t.$$

For applications requiring a representative temperature over the timestep, the time-averaged oil temperature is defined as

$$\bar{T}_k = \frac{1}{\Delta t} \int_{t_k}^{t_k + \Delta t} T(t) dt.$$

Substituting the analytical solution of $T(t)$ yields

$$\bar{T}_k = \frac{b\Delta t - (T_{k+1} - T_k)}{a\Delta t},$$

This is used to evaluate heat transfer terms over the timestep.

The oil temperature T constitutes the state variable of the dynamic fryer model, while $\dot{m}$, T_{in} , and $\dot{m}_f$ act as time-varying inputs. The formulation captures the thermal inertia of the fryer oil and enables transient simulation of fryer temperature under changing operating conditions.

C

Schematic lay-out proposed test setup

