MARMARA UNIVERSITY

ENGINEERING FACULTY

# RECOVERY OF HEAT LOSS DUE TO FLASH STEAM BY STEAM CONDENSING HEAT EXCHANGER

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GRADUATION PROJECT REPORT

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# Contents

## Symbols

| | |
|---|---|
| T | Temperature (K & C) |
| S | Entropy (kJ/kg.K) |
| Q | Heat Content (J) |
| V | Flow Rate ($m^3/s$) |
| ρ | Density ($kg/m^3$) |
| $C_p$ | Specific Heat (J/kg.K) |
| q | Heat (J) |
| h | Specific Enthalpy (J/kg) |
| W | Work (J) |
| $T_{ci}$ | Inlet temperature of coil |
| $T_{co}$ | Outlet temperature coil |
| $T_{hi}$ | Inlet temperature of annulus |
| $T_{ho}$ | Outlet temperature |
| $d_0$ | Outer diameter of coil |
| D | Inner diameter of coil |
| $C_p$ | Specific Heat |
| k | Thermal conductivity |
| Pr | Prandtl no |
| Re | Reynolds no |
| C | Outside diameter of cylinder |
| B | Inside Diameter of cylinder |
| p | Pitch of the coil |
| $D_h$ | Average radius of helix |
| r | Average radius of helix |
| $D_{h1}$ | Inside diameter of helix |
| $D_{h2}$ | Outside diameter of helix |
| L | Length of coil for one turn |
| $A_f$ | Area of cross section of coil |
| $A_a$ | Area for fluid flow in annulus |
| $V_c$ | Volume by one turn of coil |
| $V_a$ | Volume of annulus |
| $V_f$ | Volume for flow of fluid |
| $D_e$ | Shell side equivalent diameter |
| $A_a$ | Area for fluid flow in annulus |
| $m_a$ | Mass flow rate in the annulus |
| $m_c$ | Mass flow rate in the coil |
| Ra | Fouling factor of shell side |
| Rt | Fouling factor (For tap water) |
| $h_i$ | Heat transfer c. inside the coil |
| $h_0$ | Heat transfer c. in the annulus |
| U | Overall heat transfer c. |
| A | Area for heat transfer |
| N | Number of turns |
| H | Height of coil |
| $L_t$ | Total length of the coil |
| $\mu$ | Dynamic viscosity |
| $\rho$ | Density |
| ci | Cold inlet |
| co | Cold outlet |
| hi | Hot inlet |
| ho | Hot outlet |
| o | Outer diameter |
| h | Helix |
| h1 | Inside helix |
| h2 | Outside helix |
| a | Annulus |
| c | Coil |
| f | Flow |
| e | Equivalent diameter |
| o | Outside the coil (Annulus) |

# Abbreviations

| | |
|---|---|
| STHE | Shell and Tube Heat Exchanger |
| WHR | Waste Heat Recovery |
| ORC | Organic Rankine Cycle |
| LHC | Longitudinal Heat Conduction |
| HE | Heat Exchanger |
| TEG | Thermoelectric Generators |
| TEM | Thermoelectric Module |
| RC | Rankine Cycle |
| ICE | Internal Combustion Engines |
| CNG | Compressed Natural Gas |
| HTC | Heat Transfer Coefficient |
| HVAC | Heating, Ventilation, and Air Conditioning |
| HCHE | Helical Coil Heat Exchanger |
| LMTD | Log Mean Temperature Difference |
| NPS | Nominal pipe size |

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# 1. INTRODUCTION

In this report, it is aimed to recover heat loss due to flash steam. Steam condensing heat exchanger will be used for this purpose. Required information such as history, components, working principle etc. is obtained from literature research about heat loss types, flash steam and heat exchangers. Steam condensing heat exchanger will be considered for the investigation and most appropriate one for factories will be designed with the help of these theoretical information.

## 1.1. WASTE HEAT

Waste heat is the heat that is created as a byproduct of performing work by a machine or another process that uses energy. The laws of thermodynamics dictate that all such processes emit some waste heat. Compared to the original energy source, waste heat has less utility (or, in thermodynamics jargon, less exergy or higher entropy). All human activities, all natural systems, and all living things can produce waste heat. For instance, incandescent light bulbs get warm, refrigerators warm the air in the room, buildings heat up during peak hours, internal combustion engines produce high-temperature exhaust gases, and electronic components warm up when they're working [1].

Capturing and transporting waste heat from a process using a gas or liquid back to the system as an extra energy source is one form of waste heat recovery [2]. The energy source can be utilized to generate more heat or electrical and mechanical power.

Waste heat can be rejected at any degree; however, the higher the temperature, the greater the quality of the waste heat, and the easier the waste heat recovery process can be optimized. It is also critical to determine the largest quantity of recovered heat with the greatest potential from a process and to guarantee that a waste heat recovery system achieves optimal efficiency.

The following equation can be used to compute the quantity or amount of accessible waste heat.

$$Q = V\ x\ \rho\ x\ C_p\ x\ \Delta T \qquad (1)$$

where, Q (J) is the heat content, V is the flowrate of the substance ($m^3/s$), ρ is density of the flue gas ($kg/m^3$), $C_p$ is the specific heat of the substance (J/kg.K) and ΔT is the difference in substance temperature (K) between the final highest temperature in the outlet ($T_{out}$) and the initial temperature in the inlet ($T_{in}$) of system. [3]

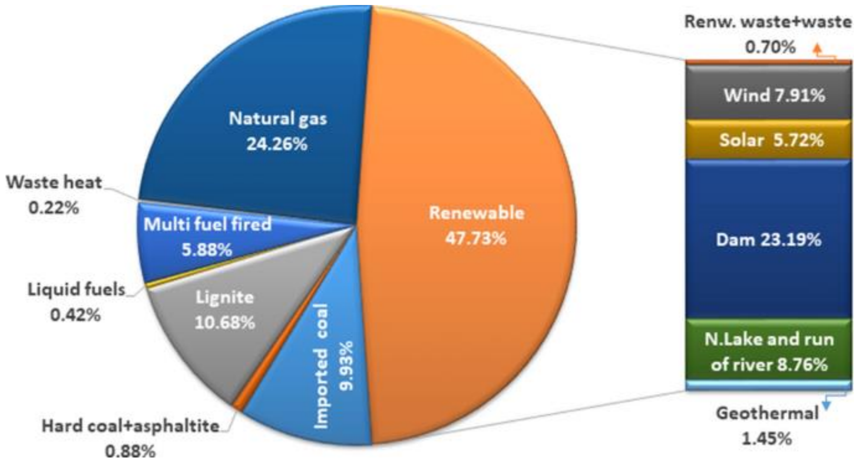


*Figure 1 Turkey's installed capacity distribution by primary energy resources in 2018 [4]*

To justify which waste heat recovery system may be employed, it is necessary to study the amount and grade of heat recovered from the process, depending on the kind and source of waste heat.

There are several heat recovery methods available for capturing and recovering waste heat, the majority of which consist of energy recovery heat exchangers in the form of a waste heat recovery unit. Air preheaters, including recuperators, regenerators, including furnace regenerators and rotary regenerators or heat wheels and run-around coil, regenerative and recuperative burners, heat pipe heat exchangers, plate heat exchangers, economizers, waste heat boilers, and direct electrical conversion devices are the most common waste heat recovery systems. All of these devices operate on the same concept of capturing, recovering, and exchanging heat with a potential energy content in a process.

Severe energy problems and environmental degradation have had a detrimental influence on all countries throughout the world. These difficulties are mostly related to industrial production and operation. Nonetheless, the introduction of improved waste heat recovery technologies has created new potential and problems in the present dire energy and environmental circumstances. The recovery and use of waste-heat in various industries is an efficient way of increasing economic advantages, preserving energy, and lowering emissions [5].

### 1.1.1. Industrial Waste-Heat Classification

WHR is an important technology for strategic energy deployment since it can efficiently ease energy shortages and minimize pollution emissions. According to temperature range attributes, waste-heat sources may be roughly categorized into three grades: low-grade waste heat (less than 230 C), medium-grade waste heat (between 230 and 650 C), and high-grade waste-heat (more than 650 C) [6]. This section provides a basic classification of common businesses that are rich in waste heat from an energy-grade standpoint, as seen in Figure 2.

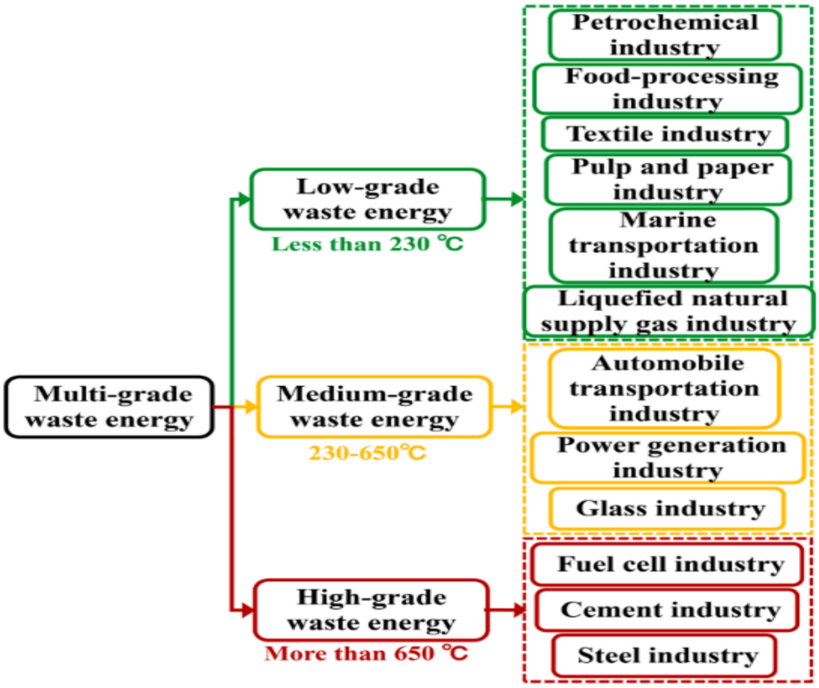


*Figure 2 Classification of industrial waste-heat [5]*

#### 1.1.1.1. Low-Grade Waste Heat

The drawbacks of low-grade waste heat in numerous sectors, such as low temperature, difficult use, and limited potential, are frequently overlooked by businesses and researchers. The overall amount of low-grade waste heat in various sectors is rather significant, accounting for 15-23% of total waste heat in the industry [7]. Recovery and utilization of low-grade waste heat may greatly reduce energy consumption and provide economic advantages; consequently, they are essential study subjects for energy departments across the world. This section offers advances and breakthroughs in the recovery and utilization of low-grade waste heat in typical industries, as well as useful suggestions for researchers.

##### 1.1.1.1.1. Food Processing Industry

The food-processing sector uses a lot of energy and entails a lot of laborious operations including preservation, transformation, sorting, and extraction. It is one of the most promising areas for substantial contributions to green and sustainable energy paradigms. Recovery and use of waste heat generated in food processing, in particular, are efficient techniques for enhancing energy utilization efficiency and offering a feasible alternative to costly and limited fossil-fuel energies. However, the waste heat generated in the food-processing sector is often low-grade energy, such as cooling water, drying heat, and hot gas; hence, efficient recovery and use of this type of waste heat is a difficult task under the current circumstances.

Cooling water accounts for a major amount of waste heat in the food processing sector. As a result, recovering and using low-grade waste heat from cooling water may not only reduce energy consumption but also preserve water resources, which is a significant energy-saving aim in the food-processing business. created a dynamic prediction framework for WHR based on the thermodynamic principle, and evaluated the recovery potential of the latent heat carried by cooling water according to the operating data of food-processing companies. According to the findings, this method might save 730000 MW and $16,6000 while also providing good economic advantages. Furthermore, drying is an essential stage in food preparation. [8]

*Table 1 Technical details of WHR technologies in the food-processing industry [5]*

| Measures | Waste-heat sources | Conditions | Performance |
|---|---|---|---|
| Heat-exchanger | Condensate water (below 90°C) | Mass flow:73.5 kg/s | Output-power:7.3 × $10^8$ kW Economic benefits: $166000 |
| Heat-exchanger | Flue-gas (approximately 96°C) | Mass flow:0.0342 kg/s | Improved thermal-efficiency:8.64% Pay back period:1 years |
| An innovative adsorption cycle | Warm and humid air (below 160°C) | Condensation temperature:95°C | Saved energy:40% |
| Thermal energy storage + ORC | Flue-gas (180–230°C) | Mass flow:1 kg/s | Pay back period:5 years Economic benefits: $224 |
| Absorption heat pump + ORC | Flue-gas (100–233°C) | Mass flow:0.81 kg/s | Output-power:932 kW Thermal-efficiency:15.43% Pay back period:6.3 years |
| Dual heat source ORC | Flue-gas (approximately 120°C) | Mass flow:3.172 kg/s Evaporation pressure: 2.1 MPa and 0.8 MPa | Output-power:212.95 kW Thermal-efficiency:15% |
| Thermoacoustic | Flue-gas (approximately 150°C) | Mass flow:0.358 kg/s | Output-power:1029 kW Improved thermal-efficiency:5.42% |

It has the potential to dramatically decrease food damage during storage and transportation, and it accounts for 20-25% of the total energy in the food processing chain. built heat-exchange equipment, including an air drier, and used flue gas to heat and transfer air into the drying equipment in the heat exchanger, thereby replacing the original drying process and decreasing energy usage directly. [9]

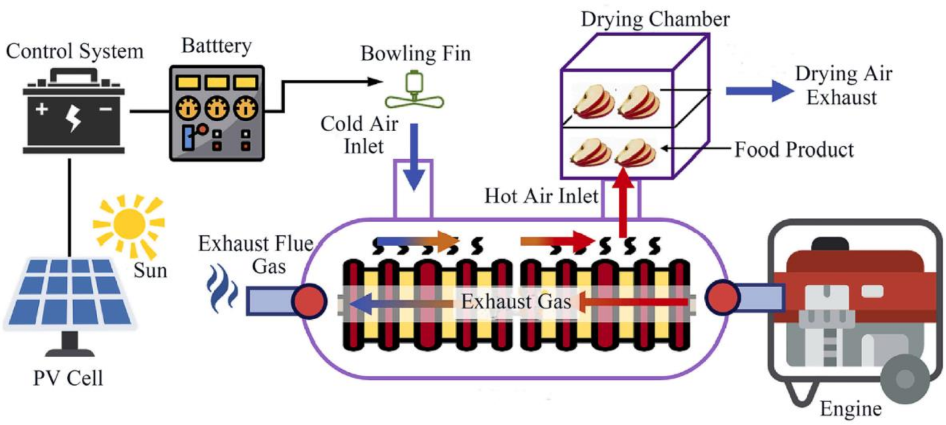


*Figure 3 Drying system experimental flow. [5]*

### 1.1.1.2. Medium-Grade Waste Heat

In terms of temperature, medium-grade waste heat has a major advantage over low-grade waste heat; this greatly improves the capacity for WHR and broadens the implementation scope of the WHR strategy. This part examines the recovery and utilization of medium-grade waste heat in typical companies and displays the benefits and drawbacks of various approaches, offering practitioners useful advice policies [5].

#### 1.1.1.2.1. Power Generation Industry

Thermal power generation is the primary source of power in modern society, accounting for 77% of total power generation [10]. However, throughout the power-generating process, a considerable quantity of unneeded energy is discharged into the environment, accounting for more than 20% of the total. Furthermore, substantial pollution (such as greenhouse gases and nitrogen sulfide) is created during the power production process, which has a negative impact on the surrounding environment and health [11]. As a result, an enhanced WHR technique for thermal power generation must be explored to boost firms' economic benefits while reducing the environmental pollution.

The economizer is now the most often used WHR device in thermal power generation. Wang investigated the thermodynamic performance of a thermal power generation system with a low-pressure economizer and examined the link between the low-pressure economizer and power production performance in various locations. The results showed that adding the low-pressure economizer boosted the power production system's output power and thermal efficiency by 3830 kW and 0.36%, respectively. Furthermore, the system saved 3.85 g/kWh of standard coal equivalent, proving the economizer's significant energy-saving potential [5].

Due to temperature constraints, an economizer is often fitted after an electrostatic precipitator to conserve energy by heating the condensed water.

Xu developed a revolutionary design of integrated high- and low-temperature air preheaters in which a low-temperature economiser was placed between the electrostatic precipitator and the low-temperature air preheater. This approach might conserve high-quality extraction steam while also providing larger economic benefits. In comparison to the traditional design, the scheme's output power reached 9 MW, and a profit of $2600000 was generated in 5500 hours. The recovery and use of flue gas waste heat from a boiler is an efficient way for a power production facility to save energy and minimize emissions.

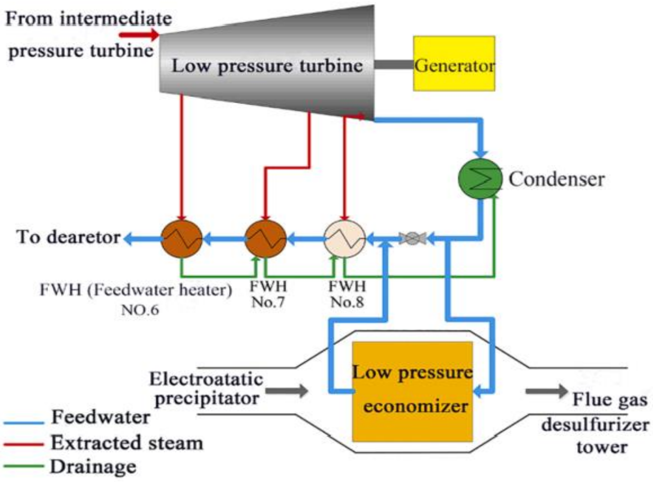


*Figure 4 Layout of low-pressure economiser [5]*

Liu updated the old WHR method. Who suggested a sophisticated proposal that combined a supercritical carbon dioxide cycle with an economizer, as seen in the image below. A thorough simulation and optimization were carried out. The thermodynamic study revealed that the cycle efficiency was 17.39% and the exergy destruction was reduced from 17.78 MW to 3.86 MW, indicating outstanding thermodynamic performance. An economic study revealed that the system could lower the standard coal equivalent by 5.19 g/kWh and that the investment cost could be repaid in 3.067 years, demonstrating that this technology was cost-effective and could be deployed on a wide scale [12].

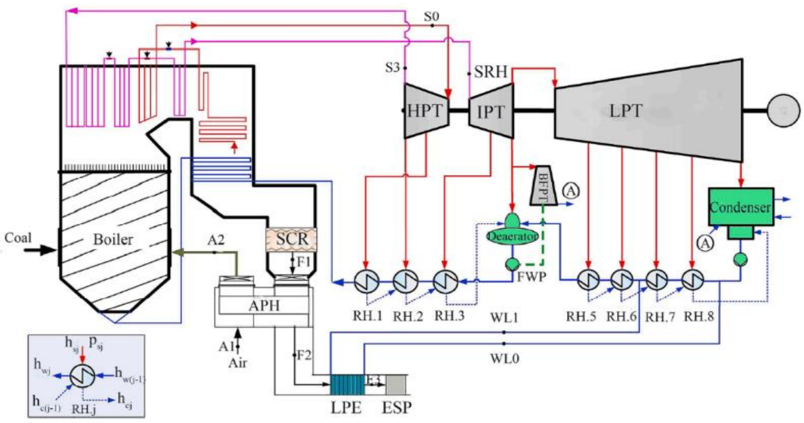


*Figure 5 Power Generation Industry, Waste heat application*

#### 1.1.1.3. High-Grade Waste Heat

During the manufacturing and operating operations of typical industries, a big volume of high-grade waste heat with significant potential is created. The recovery and utilization of high-grade waste heat have been identified as a fundamental and critical topic for boosting efficiency and lowering emissions globally, attracting academics to conduct a relevant study. This section examines practical ways for recovering and using high-grade waste heat in typical businesses to present practicable suggestions to concerned practitioners [5].

##### 1.1.1.3.1. Steel Industry

The steel industry is considered an energy-intensive industry, accounting for around 4-5% of total global energy consumption. Furthermore, the manufacture of one ton of steel produces around 1.9 tons of carbon dioxide, aggravating global warming. Thus, energy conservation and emission reduction measures in the steel sector are critical, and WHR is regarded as the most promising strategy for successfully collecting and using the vast quantity of unneeded waste heat [13].

Most slag is now handled by classic water-quenching granulation, resulting in a low waste-heat conversion capability. As a result, numerous important studies on slag waste heat have been undertaken. Li used an enthalpy-exergy compass approach to analyze and investigate the thermodynamic performances in the recovery and use of slag waste heat in different technologies based on a thermodynamics law. Kaska also presented an ORC for high-temperature flue gas in the steel sector, and based on real operating data, evaluated the thermodynamics and emission reduction potential of the WHR system. The results showed that evaporation pressure had a substantial influence on thermodynamic performance, with the evaporator causing the most exergy degradation.

The high-temperature waste-heat in the steel industry is mainly distributed in products, slag, and flue gas, as depicted in Figure 6.

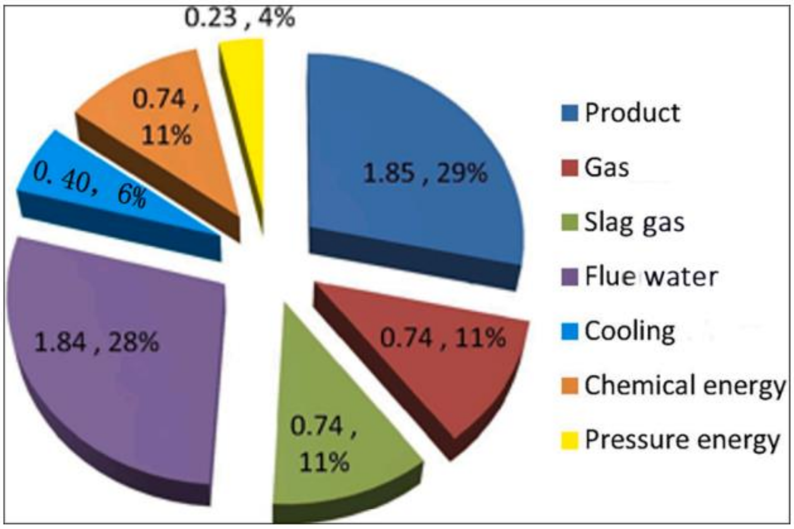


*Figure 6 Distribution of waste-heat of steel industry [5]*

As shown in Figure 6, high-temperature waste heat in the steel industry is mostly dispersed in products, slag, and flue gas. Slag is a metal substance with an exceptionally high temperature that is one of them. It is produced by complicated chemical and physical interactions during high-temperature activities in the steel industry, and it accounts for 11% of total waste heat. The table presents the most important energy data for the steel sector [14].

*Table 2 Main energy information of steel industry [5]*

| Process | Source | Type | Temperature (°C) |
|---|---|---|---|
| Coking | Coke oven | Coke | 1000 |
| | | Coke oven gas | 800 |
| | | Flue-gas | 200 |
| Sintering | Ring cold machine | Sintering | 800 |
| | Sintering machine | Flue-gas | 300 |
| Steelmaking | Converter | Basic oxygen furnace gas | 1500 |
| | | Chemical heat | 2000 kcal/m$^3$ |
| | | Slag | 1550 |
| | Electric-arc furnaces | Flue-gas | 1200 |
| | | Slag | 1500 |
| Rolling | Casting-rolling workshop | Billet | 900 |
| | Heating furnace | Flue-gas | 900 |
| | Heating-furnace hearth | Cooling water | 200 |

The steel industry WHR technical details are shown in the table below. In comparison to traditional WHR technologies, advanced processes such as tubular heat exchangers, phase-change materials, and liquid-phase ion-stripping have emerged to demonstrate superior heat-exchange process performances (such as reduced exergy loss), demonstrating their potential application value.

*Table 3 Technical details of WHR technologies in the steel industry [5]*

| Measures | Waste-heat sources | Conditions |
|---|---|---|
| RC | Slag | Temperature:1500°C |
| Air Brayton cycle | Slag | Temperature:1500°C |
| Externally fired gas + turbocharging | Flue-gas | Evaporation temperature:750°C |
| ORC | Steam (approximately 122°C) | Mass flow:16.23 kg/s<br>Pressure:1.08 MPa |
| ORC + thermochemical + hydrogen compression | Flue-gas (approximately 810°C) | Mass flow:9.2 kg/s |
| Cogeneration | Flue-gas (approximately 413°C) | Mass flow:73.5 kg/s |
| Flat heat-pipe heat-exchanger | Flue-gas (approximately 500°C) | Mass flow:0.42 kg/s |
| Tubular heat-exchanger | Sewage (approximately 85°C) | |
| Phase-change material | Steam | Mass flow:11 kg/s<br>Steam generator size:28800 kWh |
| Liquid-phase ion-stripping | Low temperature waste-heat (below 100°C) | Water yield rate:0.55 |

## 1.2. Waste Heat Recovery Methods

### 1.2.1. Heat Exchangers

Heat exchangers have been present since the 1880s, with its primary uses being in the food and beverage sectors. Albrecht Dracke of Germany was given the first reported patent for a plate heat exchanger in 1878. The first modern and commercial examples of heat exchangers, on the other hand, would be seen in the early 1900s. During the 1920s, the first designs of shell and tube heat exchangers (STHE) were created for growing usage in the oil industry. They were used in crude oil facilities as oil heaters/coolers, reboilers, and condensers. They also discovered uses that worked in harsh conditions or at high pressures and tensions. However, nothing was understood about the design of tubular heat exchangers. The majority of designs were based on material strength, whereas heat transmission was estimated. Heat exchangers are used in practically every sector, including chemical, petrochemical, pharmaceutical, fertilizer, agrochemical, paint, metal, paper, edible oil, and so on. While the fundamental principles of these heat exchangers have not changed, advancements in fluid circulation/flow control, digital designing and simulation tools, material technology, fabrication methods, and the ability to easily integrate all of these factors have resulted in versatile, highly efficient heat exchangers at significantly lower costs. [15]

#### 1.2.1.1. Working Principles

A heat exchanger transfers heat from higher to lower temperatures. If a hot fluid and a cold fluid are separated by a heat-conducting surface, heat can be transmitted from the hot fluid to the cold fluid. Heat exchanger operas within the laws of thermodynamics.

Heat can be transmitted by conduction, convection, or radiation. Conduction is the movement of a fluid, such as hot air or water, that transfers thermal energy from one medium to another. Thermal radiation is a heat energy transmission method characterized by the emission of electromagnetic waves from a heated surface or item, whereas convection is the movement of thermal energy from one surface to another by the motion of a fluid such as hot air or water. The underlying ideas that underpin heat exchangers are the laws of thermodynamics.

The Zeroth Law of Thermodynamics states that thermodynamic systems have the same temperature when they are in thermal equilibrium. If two systems are in thermal equilibrium with a third, the two previous systems must likewise be in thermal equilibrium with one another; hence, all three systems are at the same temperature. According to the First Law of Thermodynamics, Energy cannot be generated or destroyed, although it may be transferred from one medium to another, such as heat.

The Second Law of Thermodynamics defines entropy (S) as an additional attribute of thermodynamic systems, describing the inherent invariable tendency of a closed thermodynamic system to increase in entropy over time.

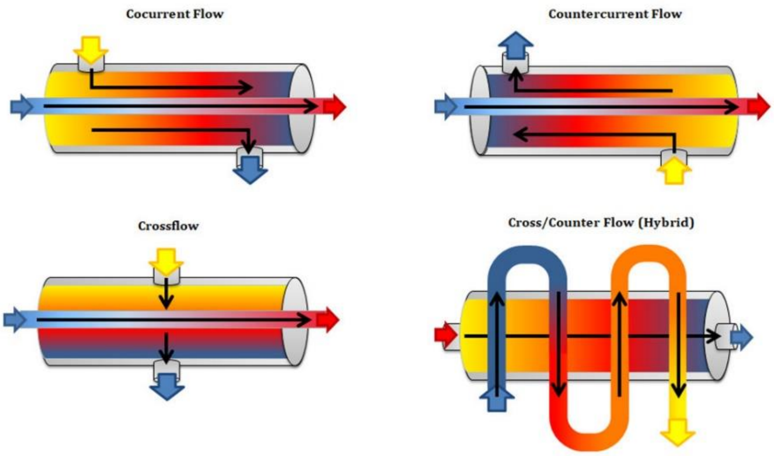


*Figure 7 Flows in Heat Exchangers [16]*

The Zeroth Law establishes temperature as a measurable property of thermodynamic systems, the First Law describes the inverse relationship between a system's internal energy and its surrounding environment, and the Second Law expresses the tendency for two interacting systems to move towards thermal equilibrium. So, in the heat exchanger, a higher-temperature fluid ($T_1$) directly or indirectly interacts with a lower-temperature fluid ($T_2$), allowing heat to pass from $T_1$ to $T_2$ and progress towards equilibrium. $T_1$ temperature decreases and $T_2$ temperature increases following heat transfer. As a result, heat exchangers may be employed to either heat or cool a fluid. [16]

In order to transmit heat between two or more process fluids, a heat exchanger is used. Heat exchangers are used extensively in both household and industrial settings. Every heat exchanger is subject to particular influences and has a unique geometric form that determines its efficiency.

Based on the flow path arrangement, contact, construction features, and compactness factor, Zohuri categorized the different types of heat exchangers. Parallel flow, counter flow, single-pass cross flow, and multi-pass counter flow heat exchanger were the four types of flow route design that were identified [17]. In contrast, a direct contact type heat exchanger is one that uses two immiscible fluids, whereas an indirect contact type heat exchanger uses surface heat exchangers. However, the varieties of heat exchangers categorized based on construction features include tabular (shell and tube), plate, plate-fin, tube-fin, and regenerative heat exchangers. It has been observed that as the compactness factor rises, heat transmission rises as well, increasing efficiency. The Plate-fin heat exchanger's compactness factor has the best value (6000 $m^2/m^3$). The best sort of heat exchanger in this categorization is seen in cross-section in Figure 8.

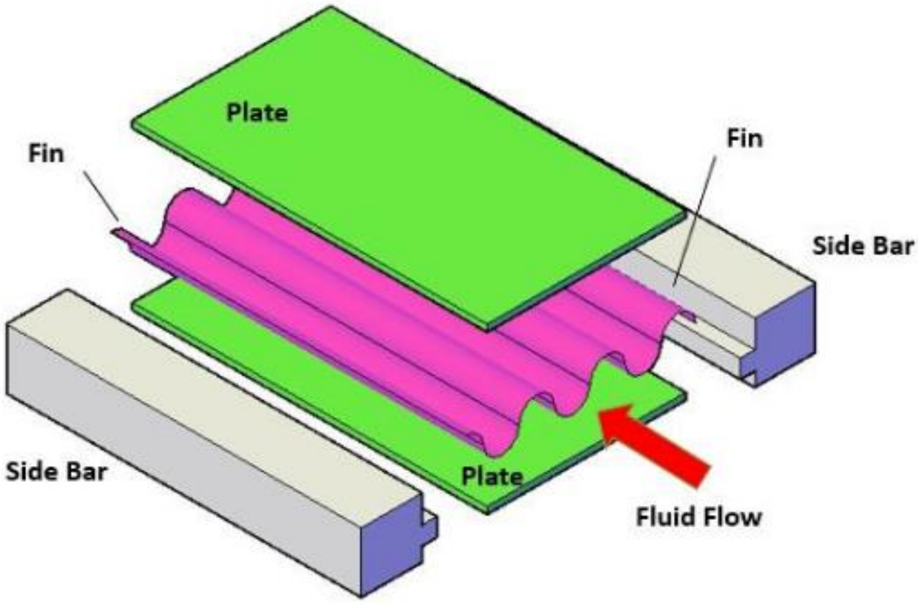


*Figure 8 Cross-section of Plate-fin heat exchanger [21].*

On the other hand, each form of heat recovery displayed a range for efficiency when the research on heat recovery applications for the housing sectors were based on the classification of types and flow arrangement. That example, the Fixed Plate's efficiency ranged from 50 to 80%, the Heat Pipe's from 45 to 55 percent, the Run between 45 and 65%, and the Rotary Wheel's from 45 to 80%, respectively. Based on the aforementioned findings, rotary wheel HE emerged as the most effective technique [18].

Heat pumps appear to be good for low-temperature waste heat recovery as they give the capability to upgrade waste heat to a higher temperature and quality. However, to prevent cross contamination, plate heat exchanger and heat pipe systems are used to transfer heat from one source with different temperature ranges. In a power-house construction, Liu et al. looked into the impacts of longitudinal heat conduction (LHC) on the heat recovery efficiency of the heat wheel, which is presently the best heat exchanger [19]. Due to the LHC effect, the heat wheel's unexpectedly low temperature efficiency and the difficulty in achieving high temperature efficiency (85%) [20]. The best kind, the heat wheel HE, according to this study, is depicted schematically in Figure 9.

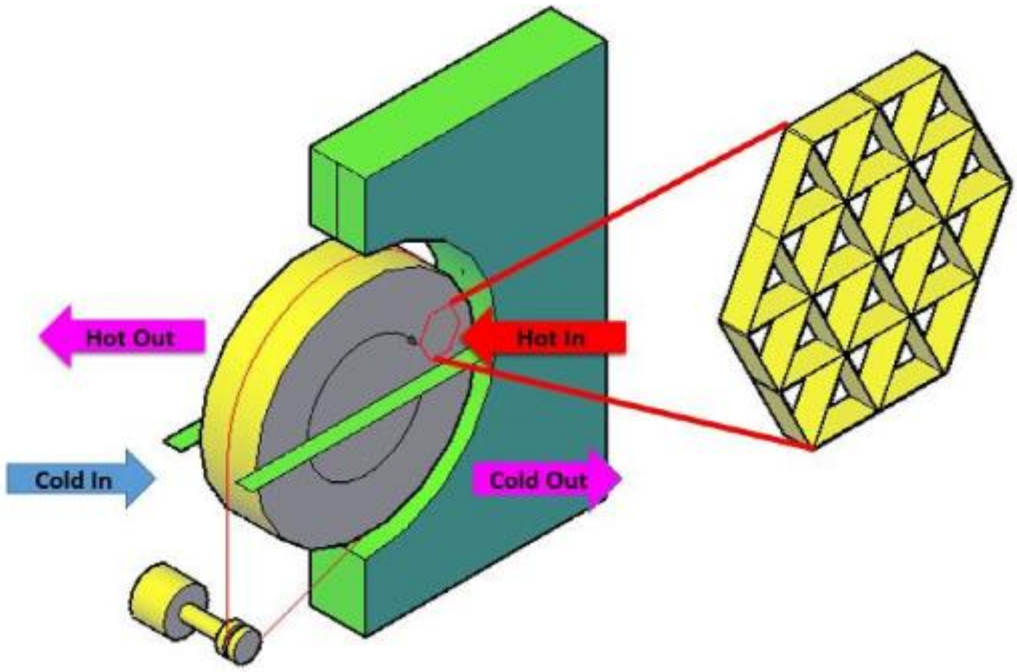


*Figure 9 Schematic of heat wheel HE [21].*

It is advised to employ plate type heat exchangers for thermoelectric generators (TEGs). The TEG receives primary heat from this heat exchanger. Fin structure, heat exchanger construction, and operational settings are additions that boost the heat exchanger's thermal consistency and heat transfer rate. Additionally, heat is transferred from the engine exhaust flowing into the heat exchanger (HE), which lowers the pressure in various structures under a variety of operating situations. However, the choice of the internal construction of the HE is controlled by engine operating conditions, permissible pressure drop, and temperature limit of thermoelectric module (TEM). Table 4 provides an overview of several studies on heat exchangers, along with a list of key goals and subsequent findings.

The most crucial component of heat recovery systems is the heat exchanger. Thus, increasing the efficiency of the entire system by evaluating several heat exchangers and choosing the optimal one. According to studies, plate-fin and rotary wheel heat exchangers outperform all other designs. As a result, it is impossible to draw a firm conclusion on the ideal heat exchanger for research purposes. Then, to increase the efficiency of the HE and the entire WHR system, researchers should conduct in-depth studies on heat exchangers, their kinds, and the improvements that may be made [21].

Table 4 Summary for some heat exchangers studies [21]

| Ref. | Main Objective(s) | Method | Results | Best HE |
|---|---|---|---|---|
| Zohuri and McDaniel (2018) [17] | Discussing heat exchangers, their selection guides and further main characteristics | HE | • Heat transmission increases and efficiency rises as the compactness factor rises.<br>• The best compactness value is achieved by the plate-fin heat exchanger (6000 m2/m3). | Plate-Fin HE |
| Mardiana-Idayu and Riffat (2012) [18] | Review on heat recovery technologies for building applications | HE | • The optimal solution to employ was determined to be the rotary wheel heat exchanger. | Rotary wheel HE |

## 1.2.2. Rankine Cycle

The Rankine cycle (RC), a dependable method for effectively converting low- and medium-temperature heat sources into electricity, has long been hailed as a potential way to recover waste heat. The condenser, turbine, boiler, and pump are its four essential components. This cycle is a crucial part of many waste heat recovery technologies. Rankine cycle components are shown in Figure 10. The fundamental technology to achieve the complete and effective use of energy, nevertheless, are waste heat recovery techniques. Due to its ease of construction and usage, the ORC system—which can convert heat at low and medium temperatures into mechanical energy or electricity—is studied and utilized extensively [22]

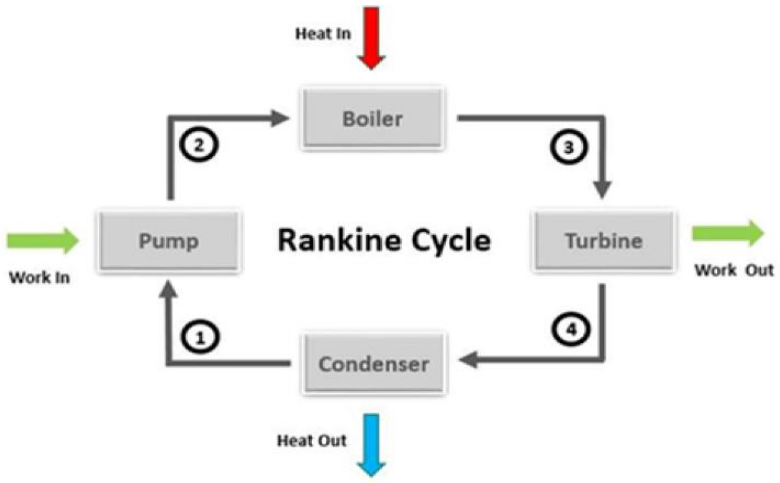


Figure 10 Rankine cycle components [21]

A potential technique for energy recovery from the waste heat rejected by IC engines is the ORC system. Diesel engines (Regenerative dual loop), ICEs, and gas turbines are the three main sources of WHR by ORC. However, although the condensation temperature has negative feedback on ORC system performance, the increase in heat source has a favorable one. Additionally, critical state, sensible heat, and ratio of vaporization latent heat are the most significant thermo-physical properties of working fluids determining system performance. Consequently, fluid mixing improves this performance. In other words, ORC plays a significant role in the recovery and conversion of low-grade heat energy due to the small-sized, simple, low-cost system components and the properties of organic fluids that can utilize low and variable temperature heat sources. As a result, this technology is exceptional since it is inexpensive and has high thermal efficiency [23].

Typically, only one ORC system is able to effectively recover energy from each of the aforementioned waste heat sources. Systems that use hydrocarbons as the working fluids perform well thermally. However, due to safety considerations, the flammability of hydrocarbons restricts their usage. This problem is proposed to be solved by a dual loop ORC system that is cascaded to collect energy from the coolant and exhaust gases of the engine independently. The results demonstrate that two environmentally friendly refrigerants, R1233zd and R1234yf, are used as working fluids to recover energy from waste heat of a compressed natural gas (CNG) engine, and the proposed dual loop ORC system could achieve better performance than other ORC systems for similar applications [24].

### 1.2.3. Thermoelectric Generators

Thermoelectric generators (TEGs) are described as direct electrical conversion devices that produce electricity directly from waste heat, eliminating the need to convert heat from mechanical to electrical energy. A schematic describing the operation of TEGs is shown in Figure 11.

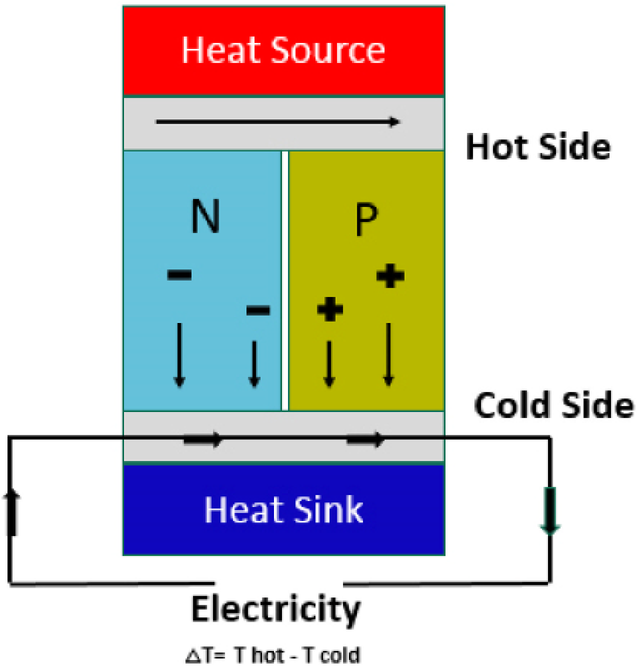


*Figure 11 Schematic explaining the working principle of TEGs [21]*

Numerous research that summarizes heat transfer rate, thermal consistency, and pressure descent in thermoelectric materials and heat exchangers can be found in the literature. The following were the key conclusions: In a TEG, power is produced by a semiconductor with low electric resistance and high thermal resistance; the power of a TEG increases as the temperature differential between the heat source and heat sink increases. Additionally, the power generated by a TEG is impacted by the physical characteristics of fluids.

Additionally, boosting the material's electric conductivity, thermal resistivity, and seebeck coefficient can all help to increase the efficiency of thermoelectric conversion. The study of TEG installed on a chimney wall. As a result, it is demonstrated that the TEG output power is not significantly affected by raising the entrance velocity of the hot gases without also increasing the velocity of the cooling air. In addition, for the same total flow velocity entering the chimney, the heat transfer of the TEG's cold side is more effective on the TEG output power than that of the hot side. In their research, Zabeka and Morinib show that thermoelectric waste heat recovery methods require a heat source temperature of 100 °C, which demonstrates suitable generator power densities of up to 80 °W/mm$^2$ [25]. The most crucial elements that must be improved for a TEG system's construction are TEM and HE. However, the heat exchanger's job in this energy transfer unit is to transport the waste heat from the hot fluid to the TEMs by absorbing it. In order to optimize the performance of the HE, it is required to improve the heat transfer between the HE and hot fluid. As a result, the pressure drop should be as low as feasible without disrupting the HE's regular functioning. As an alternative, Mahajan et al. offered several improvement methods to optimize the energy transferred from the hot exhaust gas to the hot side of the TEM, leading to an increase in TEG productivity [26]. Due to the existence of metal foams, the surface heat transfer coefficient (HTC) at the heat tube walls is magnified, which enhances the heat transfer from the majority of the exhaust gas to the hot side of the TEM, according to the study of an arithmetic model of metal foam-enhanced TEG. As a result, significant progress is made in terms of power density and thermoelectric efficiency. Nevertheless, setting up the generation of electricity with the presence of metal foam produced a net power that was seven times greater than what was obtained with a procedure without metal foam.

In many heat recovery systems today, the key to converting recovered heat into energy is the use of thermoelectric generators. The efficiency of TEG is, however, affected by a variety of parameters, including thermoelectric numbers, filling factor, length of P-N legs, couple numbers of TEC, and total thermal conductance. Thus, strengthening TEGs and improving its functioning can contribute to the enhancement in the total recovery system [21].

## 1.3. STEAM

Steam is the gas created when water changes state from liquid to gaseous. At the molecular level, this happens when $H_2O$ molecules manage to break away from the bonds that hold them together (i.e. hydrogen bonds).

$H_2O$ molecules are continually joining and separating in liquid water. However, when the water molecules heat up, the connections that bind them to begin to disintegrate faster than they can form. When enough heat is applied, some molecules will eventually break away. These 'free' molecules combine to generate the clear gas known as steam or more particularly dry steam.

Steam was critical to the industrial revolution. The early 18th-century modernization of the steam engine resulted in key advances such as the creation of the steam locomotive and the steamboat, not to mention the steam furnace and the steam hammer. The latter refers to a steam-powered hammer used to form forgings rather than a water hammer used in steam pipework.

Internal combustion engines and electricity, on the other hand, have frequently supplanted steam as a power source. Nonetheless, steam is still frequently employed in power plants and several large-scale industrial uses [27].

### 1.3.1.Principal Application for Steam

Steam is employed in a variety of sectors. Common applications for steam include steam-heated operations in plants and factories, as well as steam-driven turbines in power plants, however, the uses of steam in the industry go well beyond this.

Here are some common industrial applications for steam:

- Heating/Sterilization
- Propulsion/Drive
- Cleaning
- Moisturization
- Humidification

In the following sections, we will examine several sorts of steam uses and present some instances of steam-using equipment to explain them.

### 1.3.1.1. Steam for Heating

Positive pressure is often used to create and distribute steam. Most of the time, this implies it is fed to equipment at pressures more than 0 MPaG (0 psig) and temperatures greater than 100°C (212°F).

Positive pressure steam heating applications may be found in food processing industries, refineries, and chemical plants, to mention a few. Saturated steam is used to heat process fluid heat exchangers, reboilers, reactors, combustion air preheaters, and other heat transfer equipment. The steam raises the temperature of the product in a heat exchanger by heat transfer, then condenses and exits through a steam trap.

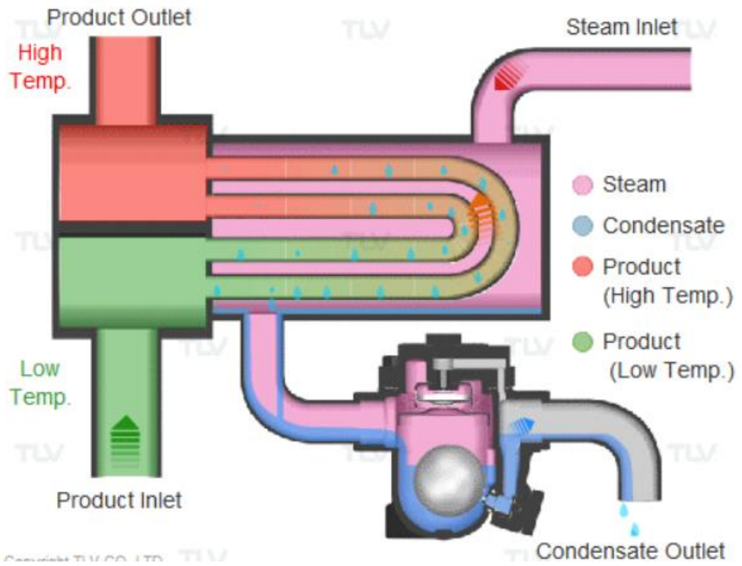


*Figure 12 Shell and Tube Heat Exchanger [27]*

### 1.3.1.2. Steam for Propulsion / Drive

In applications such as steam turbines, steam is frequently utilized for propulsion (as a driving force). The steam turbine is a piece of equipment used in thermal electric power plants to generate energy. Progress is being made toward the utilization of steam at ever-higher pressures and temperatures in an effort to increase efficiency. Superheated steam is frequently utilized in steam turbines to minimize equipment damage caused by condensate ingress. However, in particular types of nuclear power plants, high-temperature steam must be avoided since it will cause difficulties with the material used in the turbine machinery.

High-pressure saturated steam is often utilized instead. Separators are frequently placed in supply pipework to remove entrained condensate from the steam flow when saturated steam is required.

Aside from power production, other common propulsion/drive uses include turbine-driven compressors or pumps, such as gas compressors and cooling tower pumps.

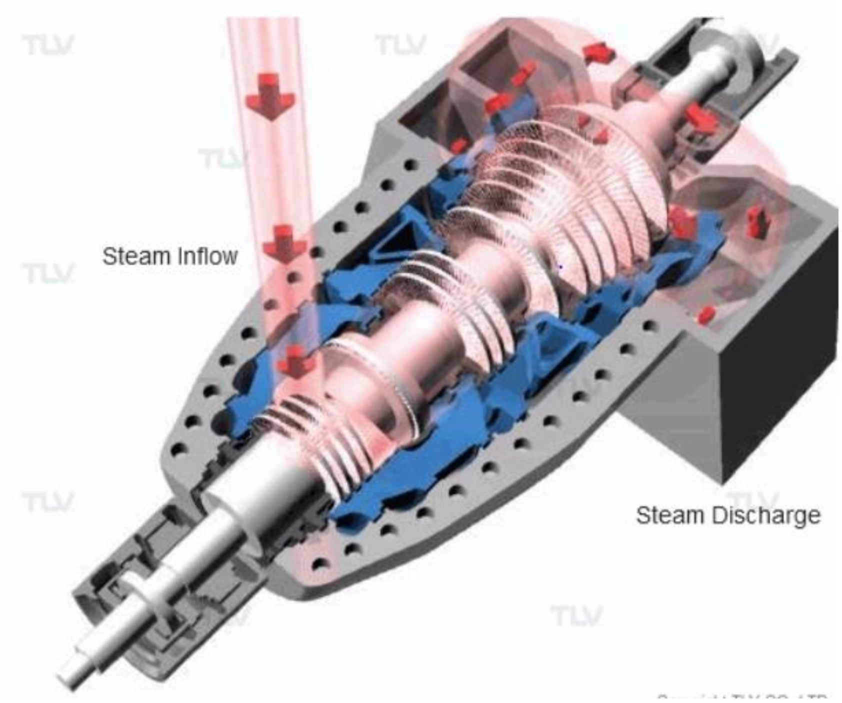


*Figure 13 Generator Turbine [27]*

### 1.3.1.3. Steam as Motive Turbine

Steam may also be utilized as a direct "motive" force in the pipework to move liquid and gas streams. To separate and purify process vapor streams, steam jet ejectors are used to pull a vacuum on process equipment such as distillation towers. They are also used to remove air continuously from surface condensers to keep the proper vacuum pressure on condensing (vacuum) turbines. The entrance nozzle allows high-pressure motive steam to enter the jet ejector and be dispersed. This results in the formation of a low-pressure zone, which draws air from the surface condenser.

In a similar use, steam is the principal motor fluid for secondary pressure drainers, which are used to pump condensate from vented receiver tanks, flash vessels, or stalling steam equipment.

### 1.3.1.4. Steam for Moisturization

Steam is occasionally utilized to give moisture to a process while also providing heat. For example, in the paper industry, steam is used to moisten the paper so that it does not suffer microscopic splits or rips when it moves through rolls at high speeds. Pellet mills are another example. Direct-injected steam is frequently used in the conditioner section of mills that manufacture animal feed in pellet form to both heat and increase water content in the feed material.

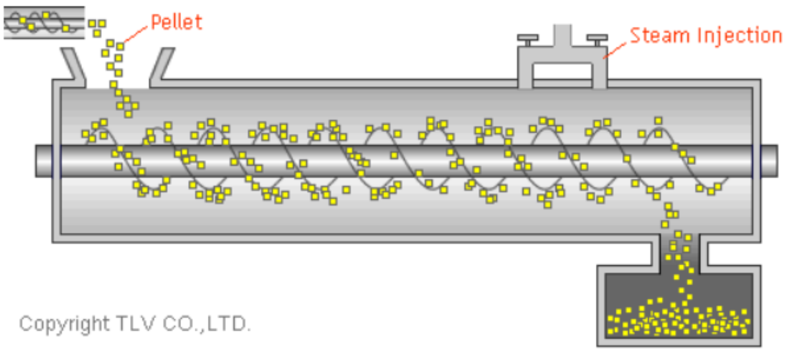


*Figure 14 Pellet mill conditioner [27]*

### 1.3.1.5. Steam for Humidification

For interior seasonal heating, many major commercial and industrial establishments, particularly in colder locations, use low-pressure saturated steam as the primary heat source. HVAC coils, which are frequently paired with steam humidifiers, are the equipment used for conditioning the air for indoor comfort, book preservation, and infection prevention. The relative humidity of the air reduces when the cold air is heated by the steam coils, and it must then be corrected to normal levels with the addition of a regulated injection of dry saturated steam into the downstream air flow.

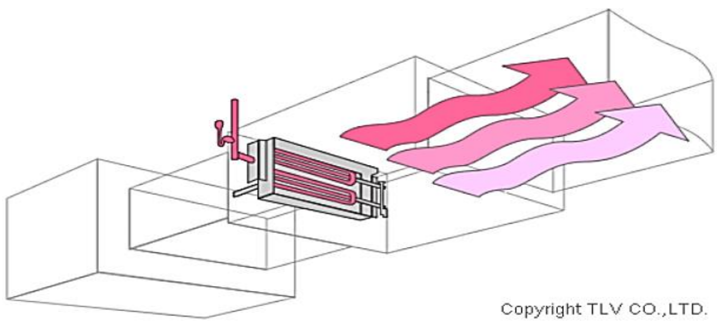


*Figure 15 Steam Humidifier in air duct [27]*

## 1.3.2. Flash Steam

When the pressure is low, the steam created from hot condensate is referred to as flash steam. Flash steam is identical to regular steam; it is only a convenient phrase used to describe how the steam is created. Normal or "live" steam is generated by a boiler, steam generator, or waste heat recovery generator, whereas flash steam is created when high-pressure/high-temperature condensate is exposed to a large pressure drop, such as when escaping a steam trap.

High-temperature condensate includes a surplus of energy, preventing it from remaining liquid at lower pressures. As a result of the extra energy, a portion of the condensate flashes. Because of the pressure differential, condensate emitted from a trap orifice partially evaporates.

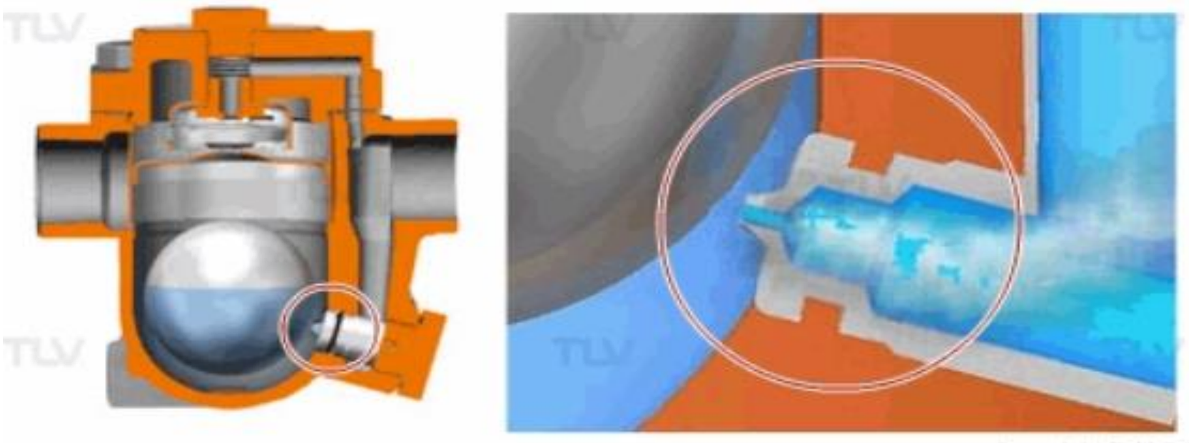

*Figure 16 Illustration of flash evaporation [27]*

Because the saturation point of water fluctuates with pressure, flash steam arises. At atmospheric pressure, the saturation point of water is 100 °C (212 °F). So, what happens when condensate at 184 °C (363 °F) is discharged into the atmosphere? The condensate has too much energy (enthalpy) to be completely liquid, so some of it evaporates, lowering the temperature of the remaining condensate to the saturation point. This is referred to as flash evaporation. To put it another way, when hot condensate is released into a lower-pressure environment, its enthalpy (total energy) remains constant, but its saturation point decreases (the temperature at which condensate can exist in both the liquid and gaseous state).

To compensate for the surplus energy, some of the water molecules absorb it as latent heat and evaporate to generate steam. The steam clouds that can emerge outside a non-subcooling trap releasing into the atmosphere are one of the first things that spring to mind when picturing flash steam.

These steam clouds are sometimes misunderstood as a live steam leak when, in reality, they are merely flashed condensate with tiny water droplets suspended, created by hot condensate flashing and being discharged to the atmosphere.

The % of flash steam generated (flash steam ratio) can be calculated from:

$$Flash\ Steam\ Percentage(\%) = \frac{h_{f1} - h_{f2}}{h_{fg2}} \qquad (2)$$

where:

$h_{f1}$ = Specific Enthalpy of Saturated Water at Inlet[1]
$h_{f2}$ = Specific Enthalpy of Saturated Water at Outlet
$h_{fg2}$ = Latent Heat of Saturated Steam at Outlet

As seen in the following cases (Figure 17), when condensate is released into atmosphere (Example 1), a larger percentage of flash steam is created than when it is discharged into a closed return system (Example 2):

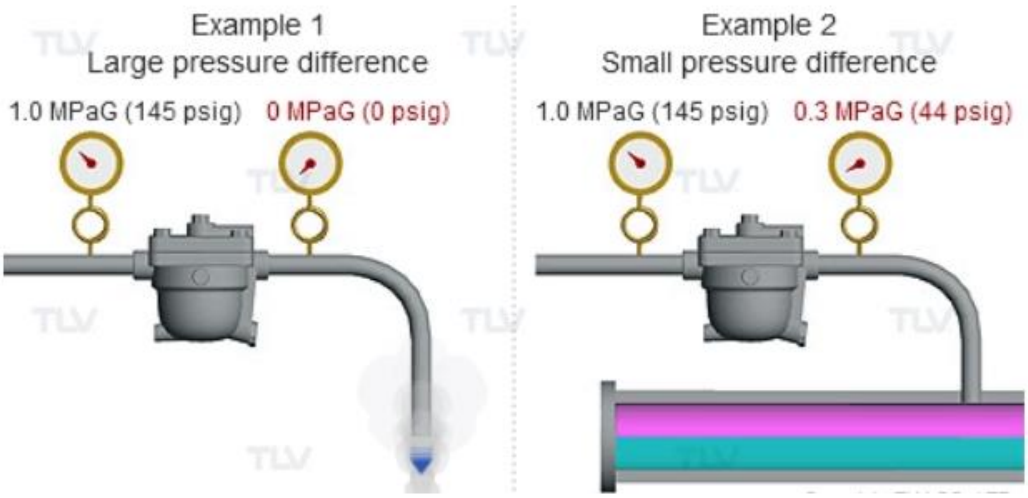


*Figure 17 Example cases for flash steam releases [27]*

[1] The sensible heat of condensate at the trap intake can be much lower than when anticipated using inlet pressure saturated steam values in traps intended to have a large degree of sub-cooling of the condensate before discharge.

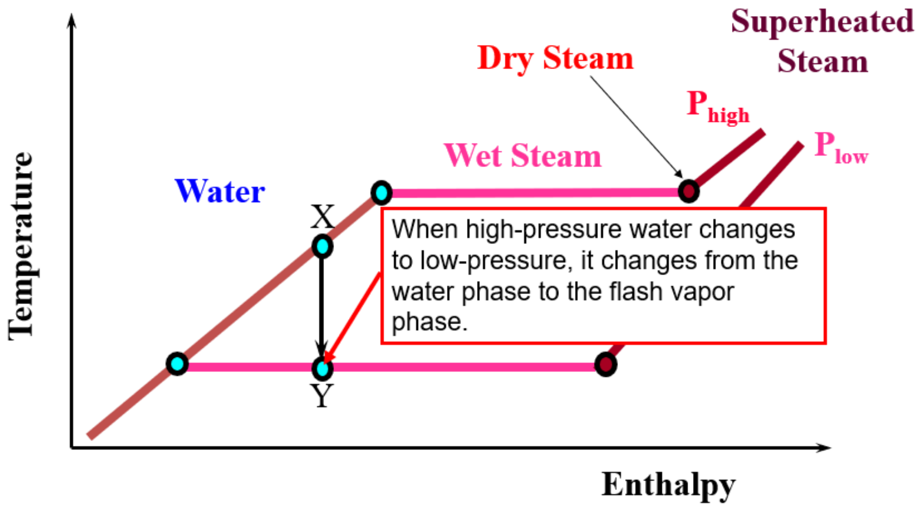


*Figure 18 Flash release explanation on T-S graph [27]*

The flash steam vapor cloud is a natural byproduct of condensate discharge. Because flash steam has the same quality as live steam, contemporary facilities frequently reuse large volumes of flash steam whenever practical.

Reusing flash steam generated by a higher-pressure system for use in a lower-pressure system can result in significant energy savings while also enhancing the working environment of a facility by eliminating vapor clouds. Condensate recovery systems and flash steam recovery systems are frequently considered as a pair when attempting to establish a waste heat management system.

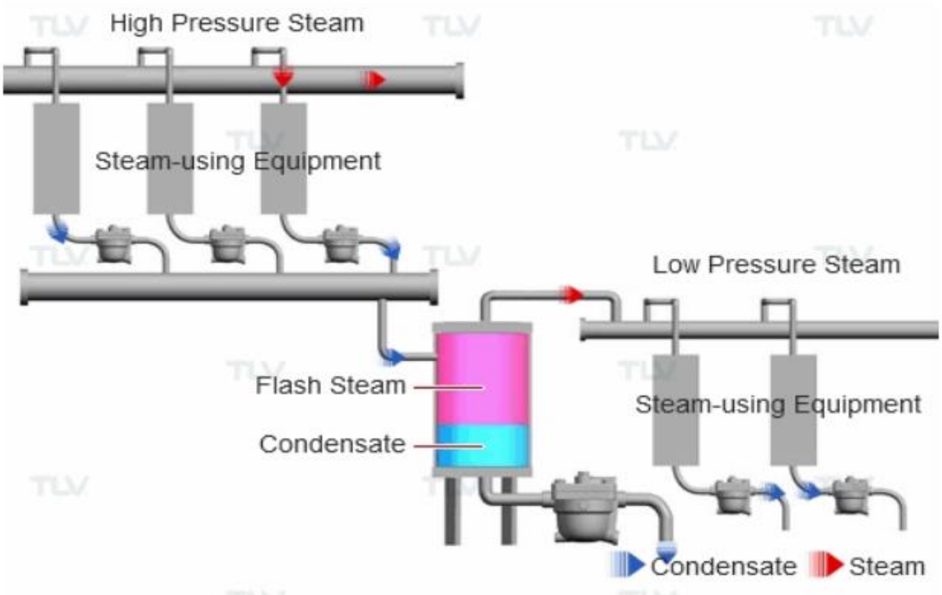


*Figure 19 Flash steam from a high-pressure system is collected and utilized as steam in a low-pressure system through a flash tank. [27]*

## 1.4. HEAT RECOVERY FROM CONDENSING HEAT EXCHANGER

During the combustion processes in a boiler, fuel and oxygen mix to generate heat, which is then used for a variety of reasons. Water vapor is generated during this process, and the loss of this vapor with the dry flue gases constitutes a significant heat loss in the form of latent and specific heat. This heat recovery is achievable by chilling the flue gases from the boiler below the dew point temperature and transferring it to a lower temperature fluid, often boiler make-up water. This sort of heat recovery has the potential to increase total efficiency (by up to 10-15%).

In addition to heat recovery, the condensing heat exchanger improves air pollution by lowering particle and sulfur dioxide emissions.

Flue gas passes over the tubes and cold feed water flows within the tube in this form of heat exchanger. To prevent corrosion from acidic condensate, corrosion-resistant materials such as stainless steel or fiberglass are utilized, or exposed surfaces are coated with a corrosion-resistant substance such as Teflon. A traditional boiler only uses sensible heat from the flue gas, while latent heat is wasted in the boiler stack. A traditional boiler's total efficiency can be increased by including advanced features and controls such as air preheaters, economizers, combustion controls, blow down, and so on. Figure 20 depicts heat losses from a standard boiler with a thermal efficiency of 82% [28]. A condensing boiler employs efficient heat exchangers that are designed to recover all available sensible heat from the fuel as well as some latent heat from flue gases by cooling the flue gas below the dew point temperature of the water vapour present in the flue gas. This form of "heat exchanger" may boost the total efficiency of a boiler plant by up to 10%; the overall efficiency of a boiler can rise by up to 90%, as illustrated in Figure 21 [28]. Low return water temperature is essential to attain high condensing efficiency. Figure 22 depicts the improvement in boiler efficiency with increasing return water temperature [28].

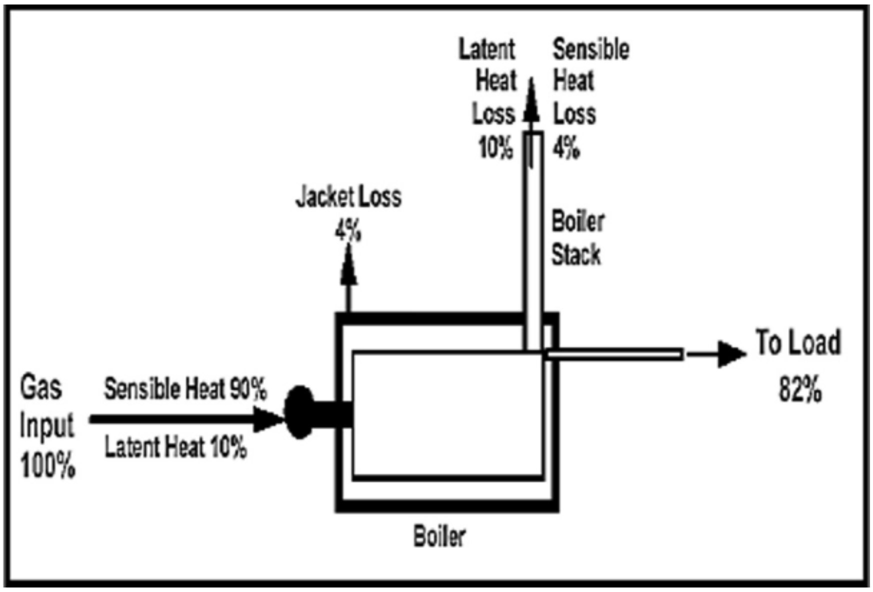


*Figure 20 Efficiency of Conventional Boiler [29]*

A condensate film forms on the tube when flue gas (a combination of a vapor and a non-condensable gas) is fed to a heat exchanger over a tube surface at a temperature below the dew point. The partial pressure of water vapor falls during condensation.

To recover the greatest amount of latent heat accessible with flue gas, a coolant at a temperature much below the dew point temperature of water vapor in flue gas is necessary. Water is created as a result of the combustion reaction between oxygen in the supply air and hydrogen in the fuel. As a result, the amount of water vapor generated and the degree of latent heat recovery vary depending on the fuel [30].

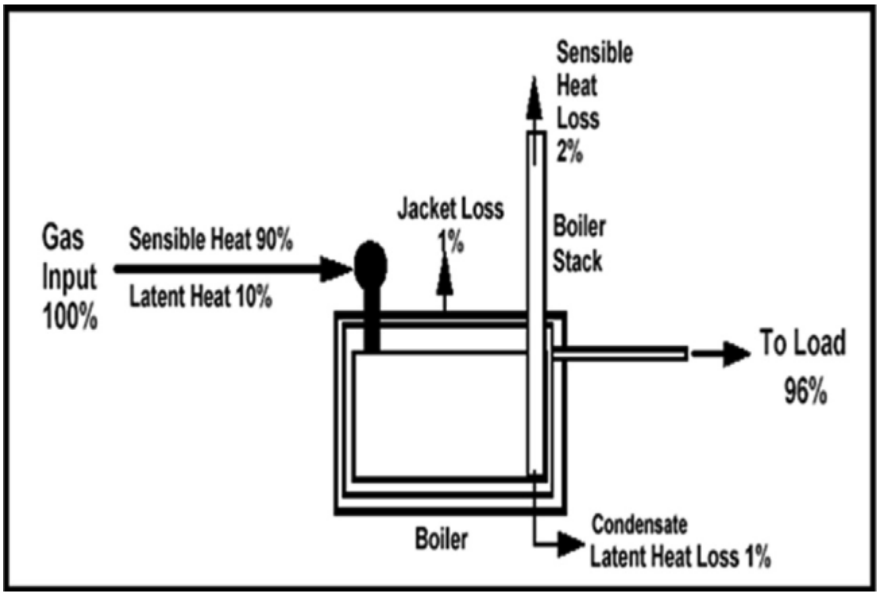


*Figure 21 Efficiency of Condensing Boiler [29]*

Several methods for estimating heat recovery from condensing heat exchangers have been presented. Jeong has built an experimental model based on his experimental findings [31].

An approximation approach was also established on the assumption that the fluctuation of heat flux with heat transferred per unit time, q, along the heat exchanger is parabolic in nature [32].

Condensing heat exchangers capture both sensible and latent heat from boiler exhaust gas, resulting in higher energy savings. Waste heat is recovered in boiler exhaust gas heat exchangers by feeding flue gas from one side and cold fluid from the other side of the heat exchanger.

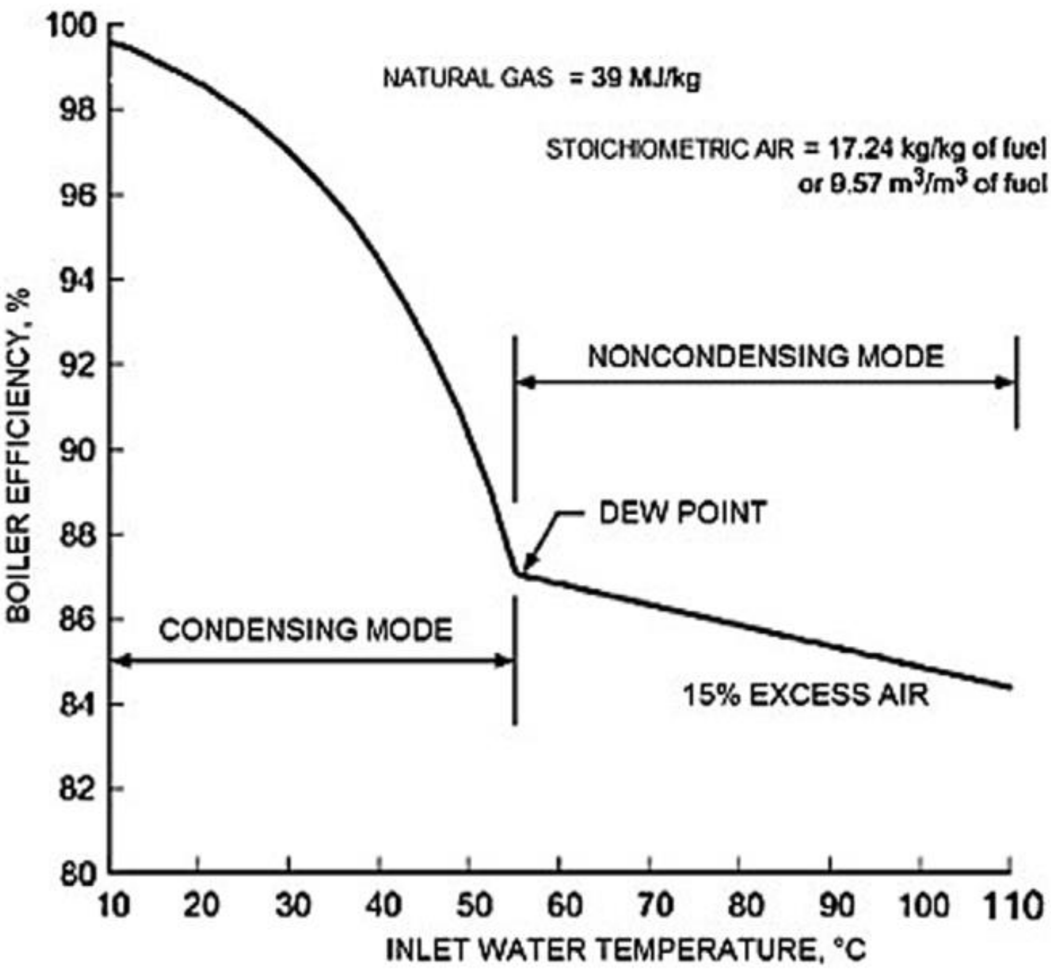


*Figure 22 Effect of Return Water Temperature on Boiler Efficiency [29]*

# 2. METHODOLOGY

Heat exchangers are widely utilized in various industries such as fertilizer plants and nuclear power plants to perform functions like heating, cooling, vaporizing, or condensing different fluid streams. There are multiple types of heat exchangers available, each with its own advantages and drawbacks. Techniques to enhance the heat transfer coefficient include both active and passive methods. Active techniques require external forces like fluid vibration, electric fields, or surface vibration, while passive techniques involve specific surface geometries or fluid additives like tube inserts. Extensive research has led to the development of new heat exchanger types, aiming to improve heat transfer efficiency using these techniques.

In most industrial applications, shell and tube heat exchangers are commonly employed. However, there is a desire to introduce alternative heat exchanger types, such as the Helical Coil Heat Exchanger (HCHE), which demonstrates superior heat transfer performance compared to traditional shell and tube designs. The HCHE is currently being designed and its thermal characteristics are being studied for potential implementation in next-generation nuclear power plants. It is anticipated that a majority of upcoming nuclear power plants will adopt helical coil heat exchangers/steam generators instead of shell and tube configurations. U-tube steam generators are preferred over shell and tube heat exchangers due to their various advantages. Helical coil heat exchangers (HCHE) offer a larger surface area for efficient heat transfer and can handle high temperatures and pressures. Additionally, they are cost-effective in terms of installation and maintenance. The main objectives of this project are to determine the necessary geometric parameters through empirical methods to achieve the desired heat transfer, and then compare the thermal performance of HCHE both empirically and numerically. The design calculations will be performed using the Log Mean Temperature Difference (LMTD) method, while the numerical analysis will utilize commercial software. In heat exchanger design, two methods can be employed: the effectiveness-NTU method and the LMTD method. Due to the project requirements, the LMTD method will be used. The procedure will involve the process design of the HCHE, including determining the average values of temperature, density, viscosity, and mass flow rate on the coil side and annulus side. The overall heat transfer coefficient will be calculated using convection correlations, and the overall surface area will be determined. Next, the coil length and the number of turns required will be determined. Finally, a comprehensive mechanical design based on ASME codes will be developed. In summary, the project aims to design and compare the thermal performance of HCHE using empirical and numerical approaches. The procedure involves process design, determination of average values, calculation of heat transfer coefficients, surface area determination, coil length and turns calculation, and a mechanical design based on ASME codes [33]

In shell and tube heat exchangers, heat is transferred between fluids without significant interaction with the separating walls. However, helical coil heat exchangers introduce turbulence due to the coil's helix, which can cause vibrations. These vibrations are generally not significant at low velocities or mass flow rates. The mechanical design of a helical coil heat exchanger involves calculations that consider factors such as material strength, stability, and robustness in relation to the system's operating conditions. The strength requirements for the components in the system are crucial considerations in determining the geometry and dimensions of the elements. Modifications may be necessary in the mechanical design of the helical coil to mitigate the effects of these limitations and design conditions. When an inquiry for a heat exchanger is received, the first step is to analyze its intended application. The design engineer must identify the appropriate type of heat exchanger that can meet the specific requirements of the application. Design temperature, pressure, and maximum allowable pressure drop need to be defined for both the product and service fluids. Design guidelines or standards provide formulas to calculate the minimum required thickness of the cylindrical shell under internal pressure and the maximum operating pressure. These calculations ensure that the helical coil heat exchanger is structurally sound and can operate safely within specified limits. [34]

## 2.1. NUMERICAL DESIGN OF HCHE

The subsequent step involves analyzing the physical properties of the product and service fluids involved in the application. A comprehensive understanding of the fluid properties is crucial for designing an effective heat exchanger. Mistakes in determining the physical properties can lead to erroneous heat exchanger designs. Once the physical properties are defined, the next step is to apply an energy balance. Typically, the flow rate of the product fluid and the desired entry and exit temperatures are specified. The type of service fluid also needs to be defined, and at least two of the following parameters should be known: flow rate, service entry temperature, or service exit temperature. By applying the energy balance equation, the third parameter can be determined. This process results in fixed flow rates and entry/exit temperatures for both the product and service fluids. Using an empirical approach, the design engineer proceeds to define the geometry of the heat exchanger. This includes determining the tube diameter, the diameter of the shell where the tubes are placed, the number of turns, the wall thickness, and the length of the inner tube. Additionally, the choice of materials is made during this step. [35]

*Table 5 Properties of Material*

| **MATERIAL** | **DENSITY** (kg/m$^3$) | **SPECIFIC HEAT** (J/kg-K) | **THERMAL CONDUCTIVITY** (W/m-K) |
|---|---|---|---|
| Copper | 8978 | 381 | 387.6 |
| Aluminum | 2719 | 871 | 202.4 |

Copper has been chosen for the coil material because the thermal conductivity of copper is higher than the aluminums.

The following figure gives us the optimized design parameters for the highest efficiency from the literature. Also, Figure 24 shows the matched selections with known values from factory.

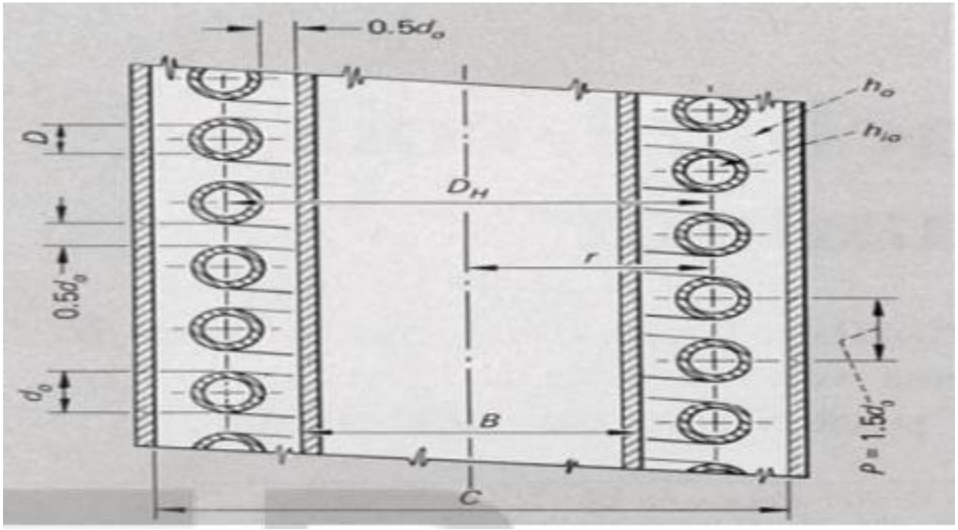


*Figure 23 Design parameters for HCHE [36]*

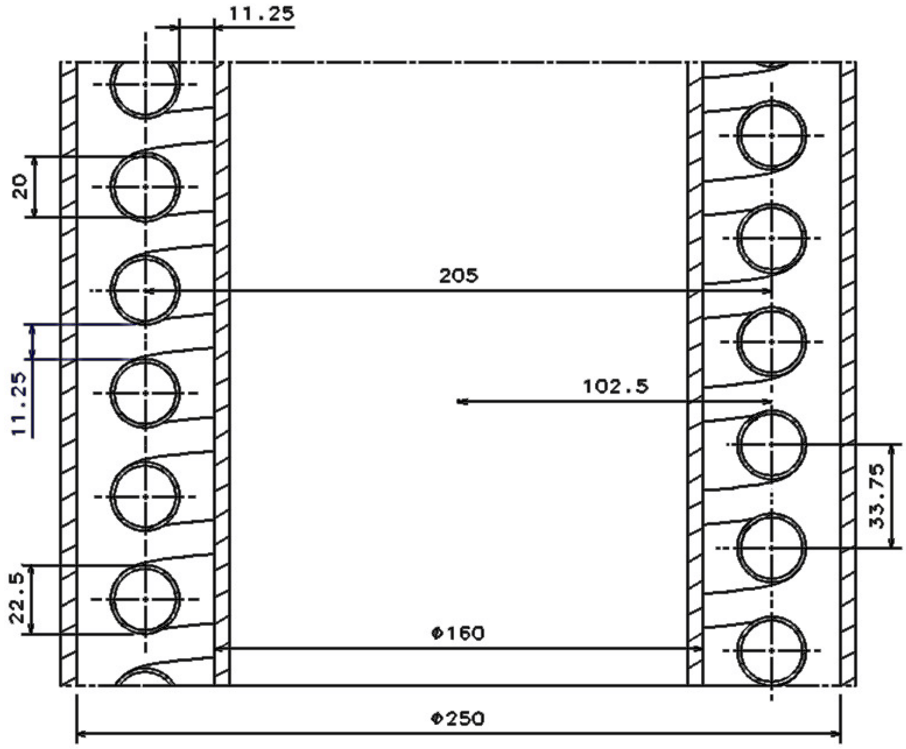


*Figure 24 Desired design parameters for HCHE*

From the geometry as shown in figure above,

- Average diameter of helix = $D_h$ = (B+C) / 2 = 0.205 m
- Average radius of helix = r = $D_h$ / 2 = 0.1025 m
- Inside Diameter of helix: $D_{h1} = (D_h - \mathrm{d}_0) = 0{,}1825\ m$
- Outside Diameter of helix: $D_{h2} = (D_h + \mathrm{d}_0) = 0{,}2275\ m$
- Pitch of the coil = p = 1.5*$d_0$ = 0,03375

*Table 6 Properties of Shell Side*

| S. No | DIMENSIONAL PARAMETERS | DIMENSIONS (mm) |
|---|---|---|
| 1. | Inlet diameter ($D_o$) | 80 |
| 2. | Shell length(L) | 1102,5 |
| 3. | Core diameter ($D_i$) | 150 |
| 4. | Shell thickness | 5 |

*Table 7 Properties of Coil Side*

| S. No | DIMENSIONAL PARAMETERS | DIMENSIONS (mm) |
|---|---|---|
| 1. | Tube outer diameter($d_o$) | 22,5 |
| 2. | Tube inner diameter($d_i$) | 20 |
| 3. | Tube thickness(t) | 1.25 |
| 6. | Pitch(p) | 33,75 |

Length of coil for one turn can be easily calculated by the formula given by:

$$L = p + \sqrt{(2\pi\, r)^2}$$

$$L = 0{,}03375 + \sqrt{(2\pi * 0{,}1025)^2} = 0.67745 \text{ m}$$

Volume occupied by one turn of coil;

$$V_c = \frac{\pi}{4} * {d_o}^2 * L = 0.000269\ m^3$$

Volume of annulus (1 turn coil);

$$V_a = \frac{\pi}{4}\ (\ C^2 - B^2\ ) * p$$

$$V_a = \frac{\pi}{4}\ (\ 0.25^2 - 0.16^2\ ) * 0.03375 = 0.000978\ m^3$$

Volume in annulus;

$$V_f = V_a - V_c = 0.000709\ m^3$$

$$D_E = 4*V_F\ /\ \pi * D_o * \mathrm{L} = \frac{4*0.000709}{(\pi * 0.0025 * 0.67745)} = 0.533$$

$$\text{Clearance} = \left(\frac{C-B}{2} - d_o\right)/\ 2 = 0.01125 \text{ m}$$

*Table 8 Essential Parameters*

| Essential Parameters | | |
|---|---|---|
| Parameter | Coil Side (Water) | Annulus (Shell Side) |
| $T_{ci}$ | 40 C | 110 C |
| $T_{co}$ | 50 C | 50 C |
| Mass flow rate | 1.11 kg/s | 0.0196 kg/s |

- The inlet and outlet temperatures for the coil above are filled with data from the factory.
- Annulus side has been assumption by taking necessary research and information from the factory. In order to prove the correctness of the assumption, the following calculations will be made. If the required efficiency cannot be achieved for the factory, the calculations will be continued as assumption 2.

At the beginning of the design process, the desired outlet temperature of the secondary or product fluid is defined, given the inlet temperature.

Inlet temperature ($Tc_i$) = 40 °C

Outlet temperature ($Tc_o$) = 50 °C

When evaluating properties related to convective heat transfer, such as density, viscosity, thermal conductivity, and specific heat, the bulk temperature or average bulk fluid temperature is considered a convenient reference point. Since these properties vary with temperature, it is important to determine the appropriate temperature for evaluating these properties. In natural convection, the fluid properties are evaluated at the film temperature, which is the average wall temperature and the bulk temperature. However, in forced convection scenarios, such as in a helical coil heat exchanger, the fluid properties are evaluated at the mean bulk temperature of the helical coil. This mean bulk temperature is calculated as the average of the bulk inlet temperature and the bulk outlet temperature.

$$\Delta T_C = (T_{ci} + T_{co}) / 2 = 40 + 50 = 45°C$$

As mentioned earlier, the thermal properties of water, such as thermal conductivity, density, and specific heat, exhibit variations with temperature. It is essential to consider these variations in the design process to ensure accuracy. For the secondary fluid (water) flowing through the helical coil, the density, specific heat, thermal conductivity, and viscosity are evaluated at 45 °C.

*Table 9 Properties of Water at 45 °C*

| | |
|---|---|
| $\rho$ | 990.1 |
| $C_p$ | 4180 |
| $\mu$ | $0.596 * 10^{-3}$ |
| k | 0.637 |
| Pr | 3.91 |

Similarly, the same procedure is followed for the primary fluid which is steam, which exchanges heat with the secondary fluid. For the primary fluid, the properties are evaluated at 80 °C.

Inlet temperature = $T_{hi}$= 110 °C

Outlet temperature = $T_{ho}$= 50 °C

$$\Delta T_h = ( T_{hi} + T_{ho} ) / 2 = 110 + 50 = 80 \text{ °C}$$

*Table 10 Properties of H2O at 80 °C*

| | |
|---|---|
| $\rho$ | 971.8 |
| $C_p$ | 4197 |
| $\mu$ | $0.355 * 10^{-3}$ |
| k | 0.670 |
| Pr | 2.22 |

In our helical coil setup, the tubes are constructed using copper. Copper is chosen for its enhanced thermal conductivity, which promotes efficient heat transfer. Additionally, it provides stability and structural integrity to the system. According to the literature, the thermal conductivity of copper is reported as follows.

Thermal conductivity (K) = 398 W/m.K

For the coil side;

$$m_a = \rho\, \vartheta_c A_f$$

$$A_f = \pi\, D^2/4$$

$$A_f = 3.1415 * (0.02)^2/4 = 31.41 * 10^{-5}\, m^2$$

Substituting the values of Area $A_f$, density p, and mass flow rate $m_a$ , the velocity is given as

$$\vartheta_c = \frac{m_c}{A_f * \rho} = 1.11 / 31.41 * 10^{-5} * 990.1 = 3{,}57 \text{ m/s}$$

$$\vartheta_c = 3{,}57\, m/s$$

Reynolds number for the fluid flowing through the tube is given as:

$$Re = \frac{(\theta * V_c * D)}{\mu}$$

$$= 990.1 * 3{,}57 * 0.02 * / 0.596 * 10^{-3} = 118568.462$$

The mass flow rate of hot fluid in the annulus:

$$m_a = 0.0196$$

Area of flow in annulus = $A_a = \frac{\pi}{4} * ((C^2 - B^2) - ({D_{h2}}^2 - {D_{h1}}^2))$

$$= \frac{\pi}{4} * (0.25^2 - 0.16^2) - (0.2275^2 - 0.1825^2)) = 0.01449059\, m^2$$

Velocity at the annulus side ($v_a$) can be found from eq. as:

$$v_a = \frac{m_a}{Q * A_a} = \frac{0.0196}{4935.632 * 0.01449059} = 0.00027404 \text{ m/s}$$

The Reynolds number at the shell side now is given by

$$\text{Re} = \frac{(Q * v_a * D_e)}{\mu} = (4935.67 * 0.0027404 * 0.533 / 0.355 * 10^{-3}) = 2030.746$$

Nusselt number = Nu = $\frac{hd}{K}$

Heat transfer coefficient on Coil Side:

As Re > 10000 so, the following co-relation is used for turbulent flow:

$$\frac{h_i d_0}{K} = 0.023Re^{0.8}Pr^{0.4}$$

$$\frac{h_i d_0}{K} = 0.023 * 118568.462^{0.8} * 3.91^{0.4}$$

$$h_i = 454.75 * \left(\frac{K}{d_0}\right) = 454.75 * \left(\frac{0.637}{0.0225}\right)$$

$$h_i = 12874.5$$

Heat transfer coefficient on Shell Side:

$$\frac{h_o d_0}{K} = 0.6Re^{0.5}Pr^{0.31}$$

$$\frac{h_o * 0.533}{0.67} = 0.6 * 2030.746^{\ 0.5} * 2.22^{0.31}$$

$$h_o = 43.52$$

Fouling factor of shell side (For tap water) Ra =0.0005 $h.m^2.K_o\ /\ kcal$ = 4.3×$10^{-4}$ $m^2.K_o/W$

Fouling factor of shell side (For tap water) Rt = 0.002 $h.m^2.K_o$ / kcal = 1.72×$10^{-3}$ $m^2.K_o$/ W

Overall Heat Transfer Coefficient:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{x}{k_c} + \mathrm{R}_a + \mathrm{R}_t$$

$$\frac{1}{U} = \frac{1}{12874.5} + \frac{1}{43.52} + \frac{0.00125}{387.6} + 4.3 * 10^{-4} + 1.72 * 10^{-3}$$

$$U = 39.67\ \mathrm{W/\ m^2K}$$

LMTD depends on the hot and cold fluid temperature differences at the inlet and exit of the heat exchanger. As it is known, the steam condensed by cooling. So. LMTD also depends on the saturation temperature of steam.

$$\Delta_{Tm} = \frac{(T_{sat} - T_{wi}) - (T_{sat} - T_{wo})}{\ln\left(\frac{T_{sat} - T_{wi}}{T_{sat} - T_{wo}}\right)}$$

$$= \quad = \frac{(110-40)-(110-50)}{\ln\left(\frac{110-40}{110-50}\right)} = 64.87$$

Area of Heat Transfer:

$$A = \frac{Q}{U * \Delta T_m}$$

$$A = \frac{4935.67}{51.191 * 64.84} = 1.919\ m^2$$

Turns:

$$N = \frac{A}{\pi * d_0 * L}$$

$$N = \frac{1.486}{\pi * 0.0225 * 0.67745} = 40.07 \approx 40$$

Height:

$$\mathrm{H} = (\mathrm{N} \times \mathrm{p}) + \mathrm{d}_0$$

$$\mathrm{H} = (40 * 0.03375) + 0.0225 = 1.3725\ \mathrm{m}$$

Length:

$$\mathrm{L}_t = \mathrm{N} * \mathrm{L}$$

$$\mathrm{L}_t = 40 * 0.67745 = 27\ \mathrm{m}$$

## 2.1.1.Efficiency & Cost Analysis

The main purpose of the project is to recover the energy of the flash steam and to provide preheating to the return water in the heat exchanger circuit. As it is known, 70.6 kg/h of flash steam comes out from 1 ton of condensate and 94% of recovery can be achieved. The heating system has a water flow of 4 $m^3$/h.

With the HCHE system, temperature of water is increased from 40 °C to 50 °C. It will provide 545 tons/year steam recovery with our 10 °C gain. In addition to energy gain, water savings will be achieved by transferring the water formed after the energy of the flash steam is taken to the condensate tank.

This project, in which we save energy and water, can be easily expanded due to its environmental friendliness and short return time

**Steam Side:**

$$Q = Q_{fg} + Q_{ht}$$

$$Q_{fg} = m*h_{fg} = 0.0196*2229700 = 43702.44 \text{ W}$$

$$Q_2 = m*cp+(110-50) = 0.0196*4197*60 = 4935.672 \text{ W}$$

$$Q = 48638.112 \text{ W}$$

**Water Side:**

$$Q = m_a * C_p * (T_{hi} - T_{ho})$$

$$1.11 * 4180 * (50 - 40) = 46398 \text{ W}$$

**Efficiency:**

$$\varepsilon = \frac{Q_{water}}{Q_{Steam}} = \frac{46398}{48438.112} = 0.94 = \%94$$

70.6 Kg/h flash steam = 46.398 kW = 39895.1 Kcal/h energy

(1 Ton Steam Cost = 42.7 Euros)

Investment Cost = 10,250 Euros

Annual Earnings = 26,289 Euros

ROI (Project turnaround time) = 142 Days

## 2.2. SIMULATION

### 2.2.1.Computer Aided Design

This study aims to conduct a thorough performance analysis of a heat exchanger employing copper as the material for the inner tubes, operating with steam as the shell fluid and water as the tube fluid in a counterflow configuration. The primary objective of this analysis is to assess the efficacy and efficiency of the heat transfer process within the system. Through meticulous calculations based on the provided dimensions of the heat exchanger, crucial insights regarding its overall performance will be derived.

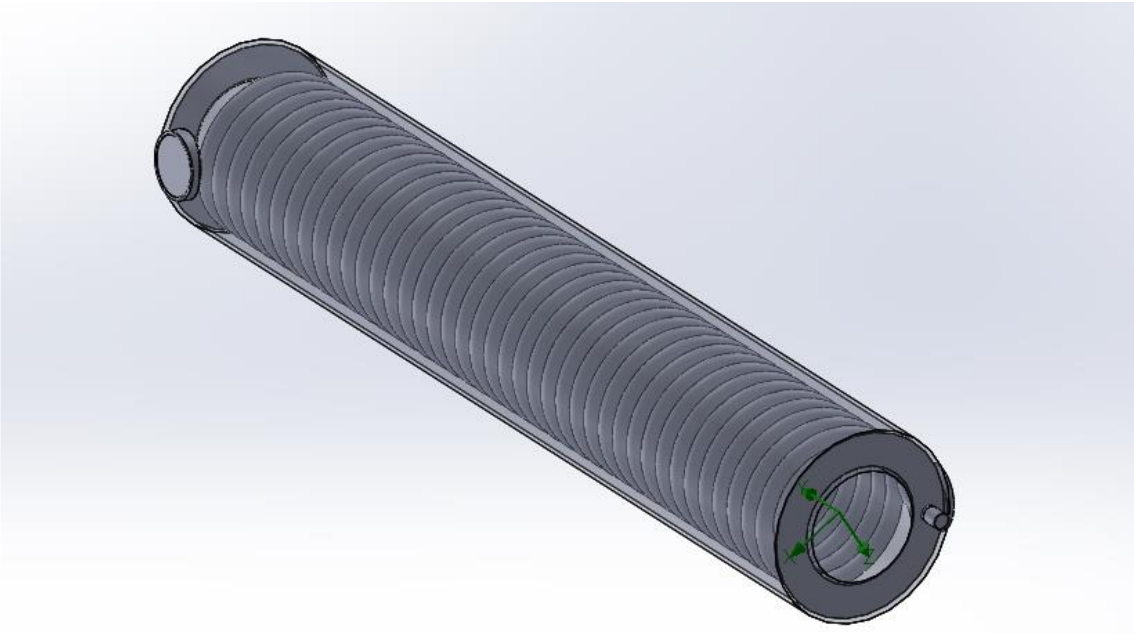

*Figure 25 SolidWorks 3D Model of Heat Exchanger*

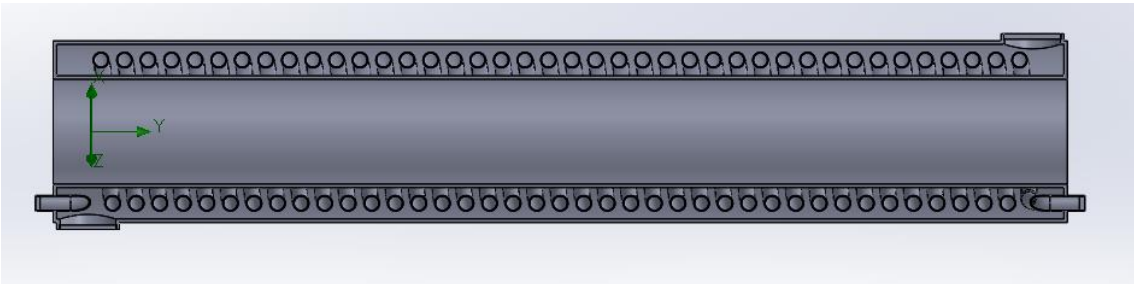


*Figure 26 Section view of the Heat Exchanger*

## 2.2.2.Mesh Details

In the SolidWorks flow analysis software, the automatic mesh settings are employed, which offer a range of scale from 1 to 7. Here, a value of 1 denotes a finer mesh, whereas 7 represents a coarser mesh. The distinction between the options of 3 and 5 is elucidated below.

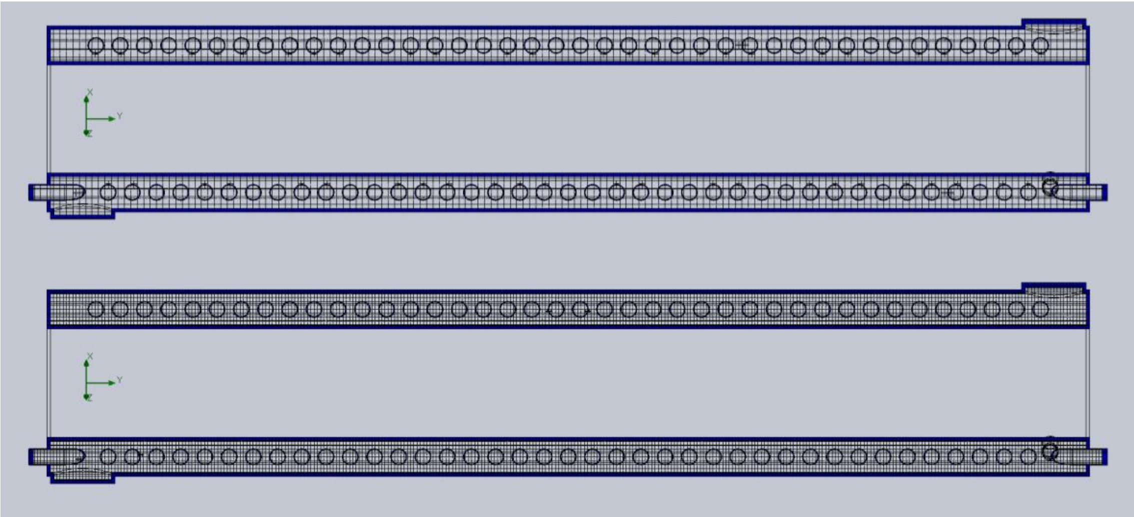

*Figure 27 Mesh views for 3 and 5 selections*

Upon comparing the results obtained from mesh options 3 and 5, a marginal disparity was observed, amounting to a temperature difference of less than 1°C. Consequently, the mesh configuration with a lower frequency was employed to expedite the computation process. It should be noted that all subsequent results presented in the subsequent sections are based on the initial mesh structure depicted in Figure 27.

## 2.2.3.Initial Conditions

The software automatically determines the domains of the steam and water flows, as illustrated in the following depiction.

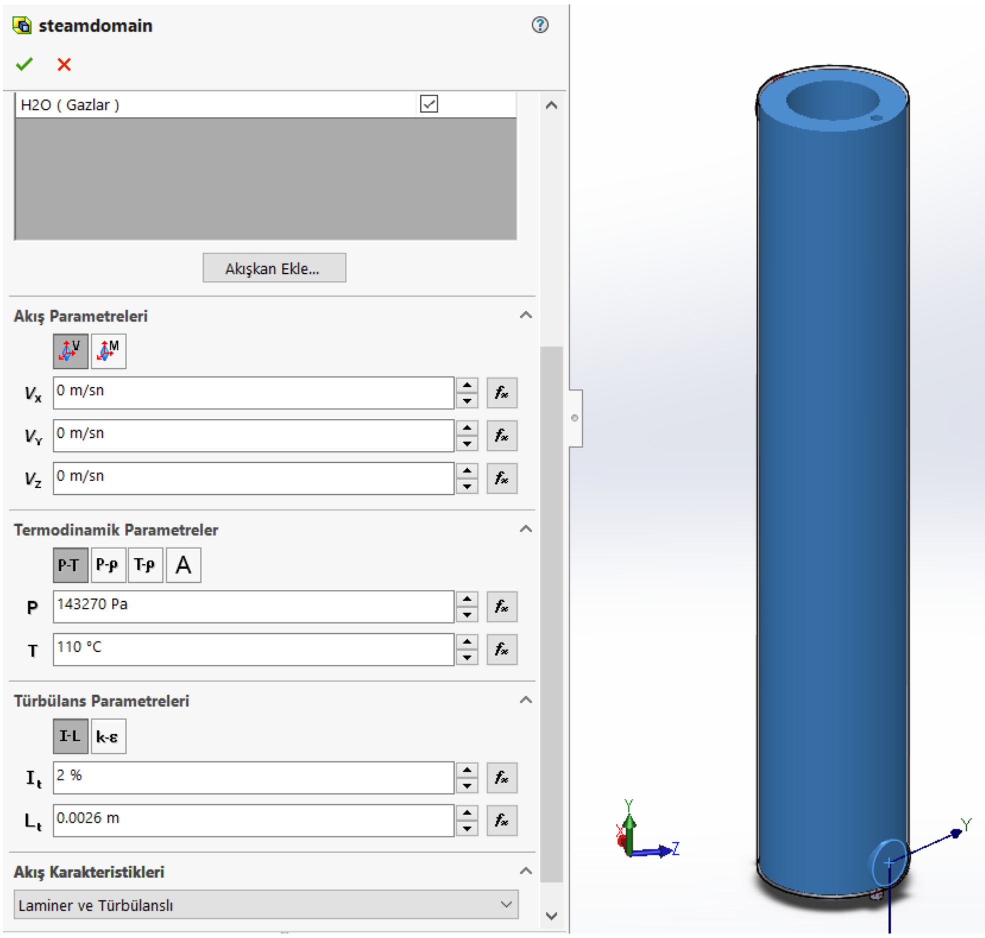


*Figure 28 Parameters for steam domain*

At the outset, considering the absence of any initial flow, the flow parameters are initialized to zero. The steam's initial temperature is defined as 110 °C, a previously established value. Additionally, the pressure of the saturated steam is set to 143270 pascals. The default turbulence parameters provided by the software are employed in the analysis.

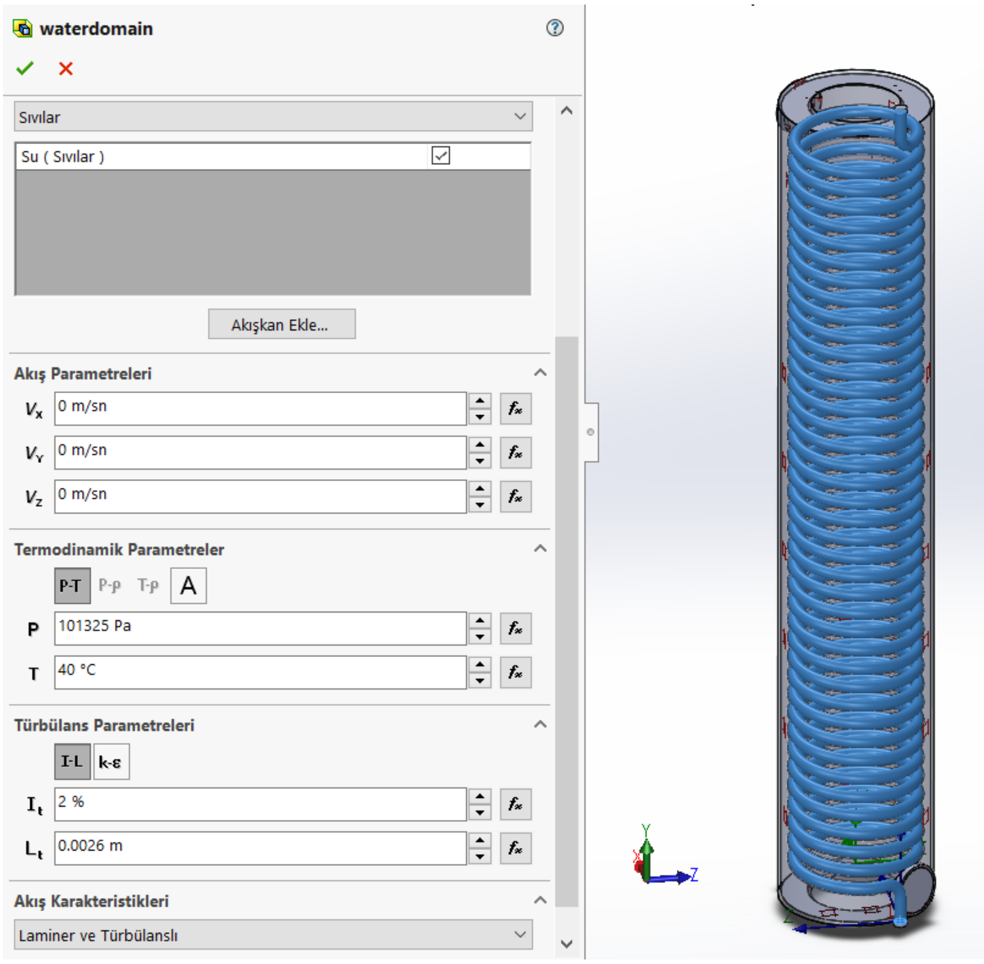


*Figure 29 Parameters for steam domain*

At the initial state, there is an absence of flow, thus necessitating the setting of flow parameters to zero. The water's initial temperature is determined to be 40 °C, based on prior knowledge. Additionally, the pressure of the steam is set to 101325 pascals, which corresponds to atmospheric pressure. The default turbulence parameters provided by the software are selected for the analysis.

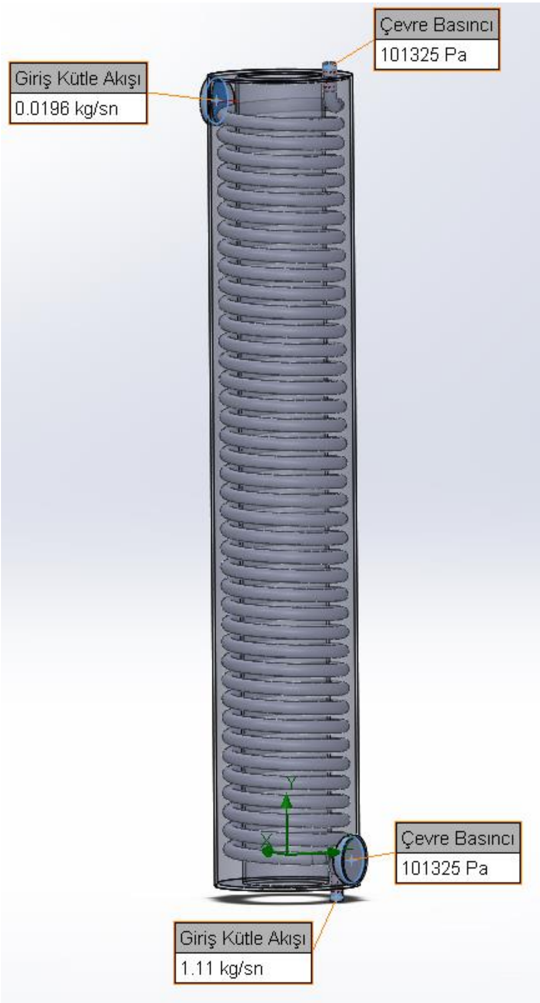


*Figure 30 Input and output boundary conditions*

As evident from the Figure 30, the inlet boundary conditions are established based on previous knowledge. Specifically, the steam mass flow rate is prescribed as 0.0196 kg/s at the inlet, while the outlet boundary is defined by atmospheric pressure. Furthermore, the water inlet is set at a mass flow rate of 1.11 kg/s, with the outlet also maintaining atmospheric pressure.

## 2.2.4.Simulation Results

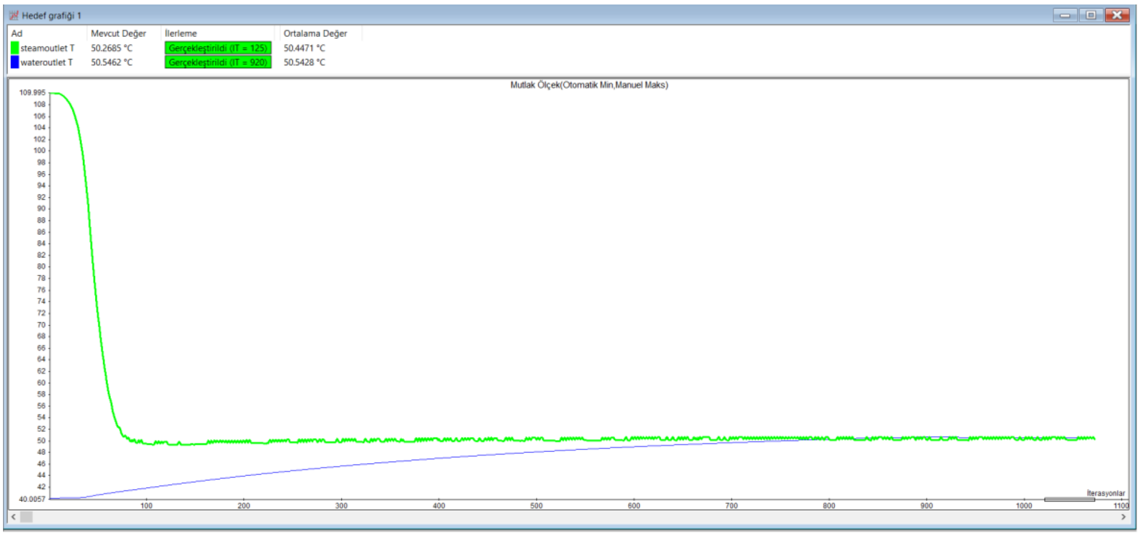


*Figure 31 Calculated value vs iteration number graph*

The iteration process and convergence behavior are documented in the preceding analysis. The outlet temperature of the steam reaches convergence at the 125th iteration, stabilizing at 52.27 °C. Conversely, due to the comparatively complex coil structure, the water outlet temperature requires a more extensive iteration process. It eventually converges at the 920th iteration, yielding a temperature of 50.55 °C.

The converged outlet temperatures for both water and steam align with the previously calculated results. Detailed graphical representations of the final results are presented in the subsequent figures.

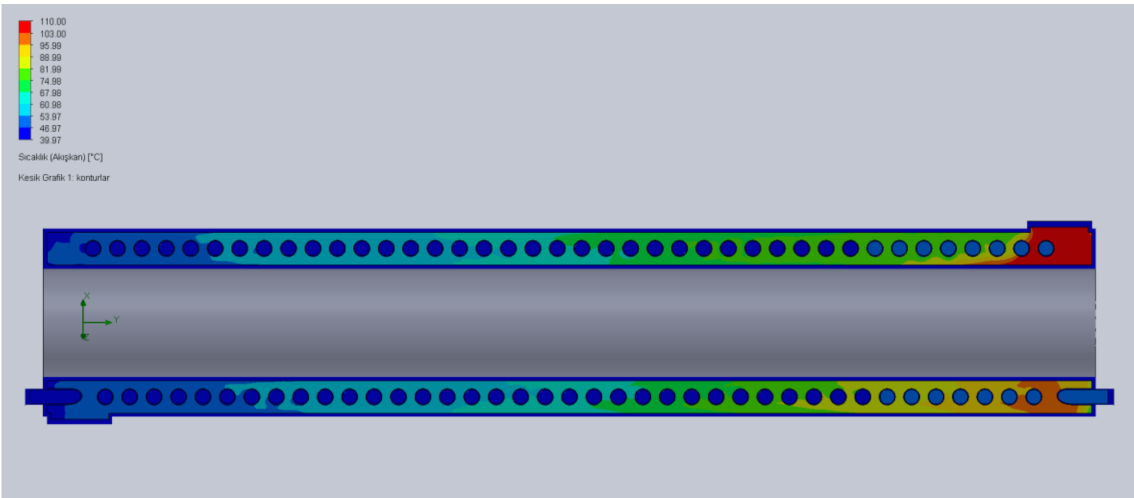


*Figure 32 Temperature contour plot at clip plane*

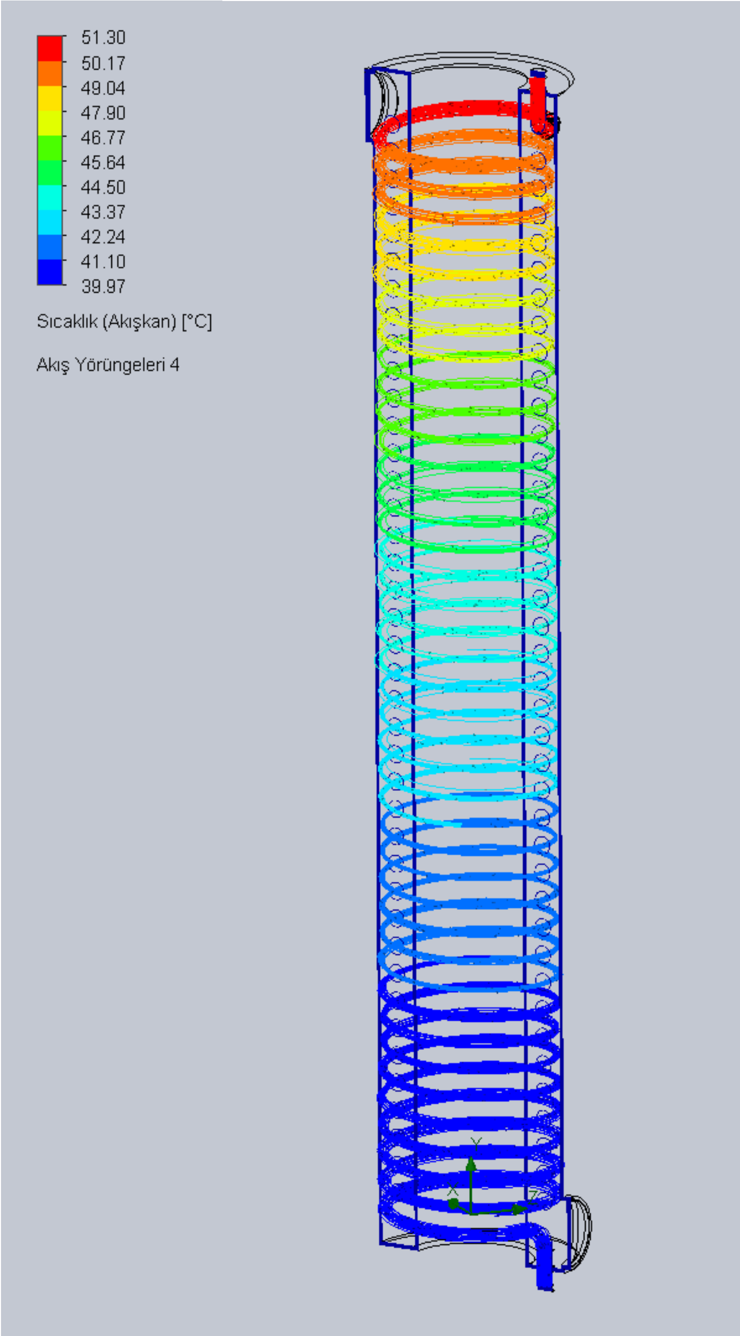


*Figure 33 Fluid flow through the coil with streamlines*

The water enters the coil at the lower end, characterized by an initial temperature of approximately 40 °C. It traverses through the coil's internal structure, interacting with the heat transfer surfaces, and undergoes a thermal exchange process. Finally, the water emerges from the top outlet of the coil, having attained an elevated temperature of around 50 °C. This temperature increase is indicative of the heat absorbed from the steam within the heat exchanger, signifying an effective heat transfer mechanism.

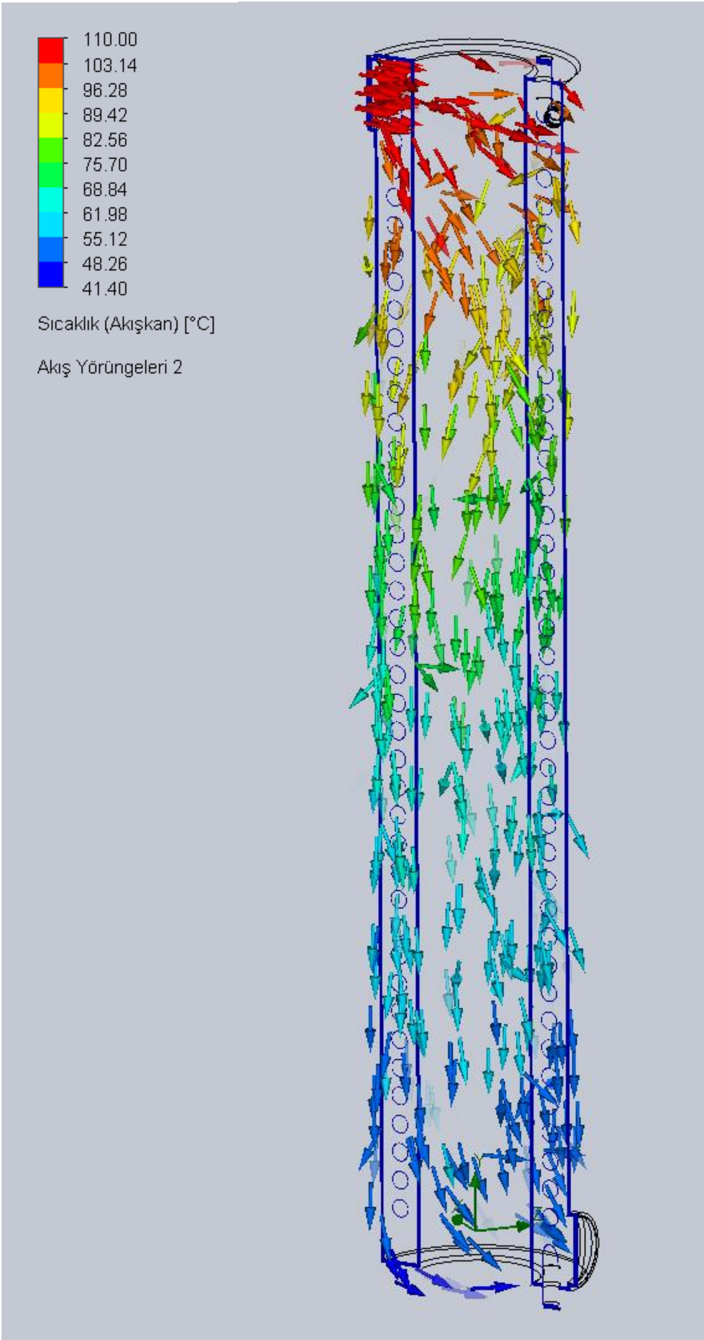


*Figure 34 Fluid flow through the shell with arrows*

The steam enters the coil from the upper inlet, exhibiting an initial temperature of approximately 110 °C. It flows through the intricate coil structure, engaging in a heat transfer process with the surrounding surfaces. As a result of this thermal interaction, the steam undergoes a significant temperature reduction. Consequently, it exits the heat exchanger through the lower outlet with a temperature of approximately 50 °C. This substantial temperature decrease signifies the successful transfer of thermal energy from the steam to the water within the heat exchanger.

# 3. CONCLUSION

Based on our calculations, we have determined that the overall heat transfer coefficient (U) depends on various factors such as the heat transfer coefficient inside the coil ($h_i$) and in the annulus side ($h_o$). It is observed that the influence of $h_i$ on U is relatively less significant compared to $h_o$. There exists a linear relationship between ho and U, with $h_o$ increasing as U increases. This relationship closely approximates an ideal trend, resembling a straight line. Consequently, the primary factor affecting overall heat transfer is the heat transfer coefficient in the annulus ($h_o$).

The average bulk temperature on the annulus side should be higher, approximately double, compared to the average bulk temperature inside the coil. Specifically, maintaining a mean bulk temperature of 80°C in the annulus and 45°C inside the coil is recommended for efficient heat transfer.

Altering the tube diameter significantly affects the temperature drop. Optimal performance can be achieved by using a coil with a small diameter and a tube with a large diameter. This configuration facilitates desirable pressure and temperature drops since temperature drop is inversely proportional to the mass flow rate.

The cross-sectional area of the coil ($A_f$), which is determined by the coil's inner diameter (D), plays a crucial role in modifying the velocity inside the coil ($V_c$). Any adjustments to $A_f$ consequently affect the velocity within the coil.

Our results clearly indicate that the variation of U with $h_i$ is less significant compared to its relationship with $h_o$. A linear correlation is observed between U and $h_o$.

The number of turns (N) in the coil is inversely proportional to both the velocity of the annulus ($V_a$) and the velocity inside the coil ($V_c$). Notably, changes in va have a more pronounced effect on U compared to changes in vc.

The efficiency of the helical coil is notably higher at low Reynolds numbers.

Through our investigation, we have discovered that increasing the mass flow rate of either the cold water in the coil or the hot water in the annulus leads to an increase in both the overall heat transfer coefficient (U) and heat transfer rate. Hence, as the mass flow rate increases, the heat transfer rate also rises.

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