

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL

OPTIMIZING INTEGRATED HEAT RECOVERY NETWORKS
WITH VARIOUS POWER CYCLES



ERDEM BAYDIR

A THESIS OF MASTER OF SCIENCE
DEPARTMENT OF MECHANICAL ENGINEERING

ADVISOR: PROF. DR. ALP ER ŞEVKİ KONUKMAN

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T.C.
GEBZE TEKNİK ÜNİVERSİTESİ
LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ

**ÇEŞİTLİ GÜÇ ÇEVİRİMLERİYLE ENTEGRE ISI GERİ
KAZANIM AĞLARININ OPTİMİZASYONU**

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To my Family

ABSTRACT

Recent technological advancements have increased the demand for electricity in industries. Additionally, considering the depletion of fossil fuels and their negative impacts on the environment, energy recovery and energy efficiency studies have been gaining momentum. It is possible to obtain useful energy by recovering waste heat in industries, and various methods are employed for waste heat recovery. One of these methods is power cycles.

In this thesis study, the aim is to design an integrated heat exchanger network and power cycle with the minimum total annual cost. Five different power cycles, including the Organic Rankine Cycle, Rankine Cycle, Supercritical Carbon Dioxide Cycle, Trilateral Flash Cycle, and Kalina Cycle, are compared. In the proposed method, sequential optimization is performed: first, the heat exchanger network is optimized and designed using the SYNHEAT model to achieve minimum total annual cost and maximum energy recovery. Subsequently, the constraints from the optimized heat exchanger network are used to model the power cycle in UNISIM DESIGN, which is then optimized in Python. Based on the optimization performed in Python, the total annual costs of the power cycles integrated into the heat exchanger network are analyzed.

The proposed method offers an alternative perspective to the methods used in the literature. Incorporating a program with a robust thermodynamic data set into the optimization process strengthens the method. The proposed method has been compared with an example studied in the literature, and its strengths and weaknesses have been evaluated. Additionally, the findings obtained from the analyzed examples indicate that, although more power is generated in Rankine Cycle among the compared power cycles, Organic Rankine Cycle is a more economical option due to the thermophysical properties of its working fluid.

Keywords: Optimization, Heat Exchanger Network, Power Cycles, Energy Recovery

ÖZET

Son yıllarda teknolojideki yaşanan gelişmeler sanayinin elektriğe olan talebini artırmaktadır. Ayrıca fosil yakıtların ömrünün sonuna gelmesi ve çevreye olan negatif etkileri göz önüne alındığında enerji geri kazanımı ve enerji verimliliği çalışmaları hız kazanmaktadır. Endüstrilerdeki atık ısı, geri kazanılarak faydalı enerji elde edilmesi mümkündür ve atık ısıyı geri kazanmak için birçok yöntem kullanılmaktadır. Bunlardan biri de güç çevrimleridir.

Bu tez çalışmasında, ısı değiştirici ağı ile güç çevrimi entegre edilerek minimum toplam yıllık maliyetle tasarlanması amaçlanmaktadır. Organik Rankine Çevrimi, Rankine Çevrimi, Süperkritik Karbon Dioksit Çevrimi, Üçlü Flaş Çevrimi ve Kalina Çevrimi gibi beş farklı güç çevrimi karşılaştırılmaktadır. Önerilen yöntemde sıralı optimizasyon yapılarak, önce ısı değiştirici ağı SYNHEAT model ile minimum toplam yıllık maliyet ve maksimum enerji geri kazanımı sağlanarak optimize edilip tasarlanmaktadır. Sonrasında optimize edilen ısı değiştirici ağından gelen kısıtlar ile güç çevrimi UNISIM DESIGN'da modellenip Python'da optimize edilmektedir. Python'da yapılan optimizasyon sonucunda, ısı değiştirici ağına entegre edilen güç çevrimlerinin toplam yıllık maliyetleri analiz edilmektedir.

Önerilen yöntem, literatürde kullanılan yöntemlere alternatif bir bakış açısı sunmaktadır. Termodinamik veri seti güçlü olan bir programın optimizasyona dahil edilmesi, yöntemi güçlendirmektedir. Önerilen yöntem, literatürde çalışılmış örneklerle karşılaştırılmış ve güçlü ve zayıf yönleri değerlendirilmiştir. Ayrıca, çalışılan örneklerden elde edilen bulgular, karşılaştırılan güç çevrimleri arasında, Rankine Çevrimi'nde daha fazla güç elde edilmesine rağmen, Organik Rankine Çevrimi'nin çalışma akışkanının termofiziksel özelliklerinden dolayı daha ekonomik bir seçenek olduğunu göstermektedir.

Anahtar Kelimeler: Optimizasyon, Isı Değiştirici Ağı, Güç Çevrimleri, Enerji Geri Kazanımı

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LIST OF SYMBOLS AND ABBREVIATIONS

RC	: Rankine Cycle
ORC	: Organic Rankine Cycle
KC	: Kalina Cycle
s-CO ₂ C	: Supercritical Carbon Dioxide Cycle
TFC	: Trilateral Flash Cycle
HEN	: Heat exchanger network
HE	: Heat exchanger
MINLP	: Mixed Integer Non-Linear Programming
TAC	: Total Annual Cost
MER	: Maximum Energy Recovery-Minimum Energy Requirement
GCC	: Grand Composite Curve
U	: Overall heat transfer coefficient (kW/m ² K)
A	: Heat exchanger area (m ²)
Q	: Heat transfer (kW)
HU	: Hot utility
CU	: Cold utility
W1	: Cooling water
S1	: Steam
ST	: Stage
NoS	: Number of stage
HP	: Hot process stream
CP	: Cold process stream
<i>i</i>	: Hot process stream
<i>j</i>	: Cold process stream
<i>k</i>	: Stage
β	: Cost coefficient exponent of the heat exchanger
h	: Enthalpy (kJ/kg)
η	: Thermal efficiency
hr	: Hour

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1. INTRODUCTION

In today's world, industrial and technological developments are at the peak levels, both the increasing demand for electricity and the increase in fuel prices, the end of the life of fossil fuels, as well as increasing concerns about global warming are increasing people's interest in renewable energy technologies. In this scope, industries have accelerated their efforts to recover waste heat from processes, reduce environmental pollution and minimize fuel consumption. Waste heat is thermal energy that is released into the environment and wasted. It is possible to recover waste heat and the efficiency of the process increases when waste heat is recovered [Jouhara *et al.*, (2018)].

The second law of thermodynamics states that work cannot be done without waste heat. Waste heat is available in every sector where there is production. The heat flows sent to the environment here are considered waste heat and have significant exergy. With the help of waste heat recovery technologies, it is possible to obtain work from these heat flows.

The temperature of the waste heat is the key criterion for determining whether the heat flows from the process to the environment can be utilized. In order for waste heat recovery technologies to work efficiently, waste heat is categorized into three different categories according to temperature. Waste heat with a temperature higher than 400°C is considered high temperature, those with a temperature between 100-400°C are considered medium temperature, and those with a temperature lower than 100°C are considered low temperature waste heat.

The technologies applied to recover waste heat can be classified as shown in **Figure 1.1** below. There are many technologies available to recover waste heat. Technologies that can generate power from waste heat by heat and power integration are very popular. There are many studies on waste heat utilization using thermodynamic power cycles [Brückner *et al.*, (2015)].

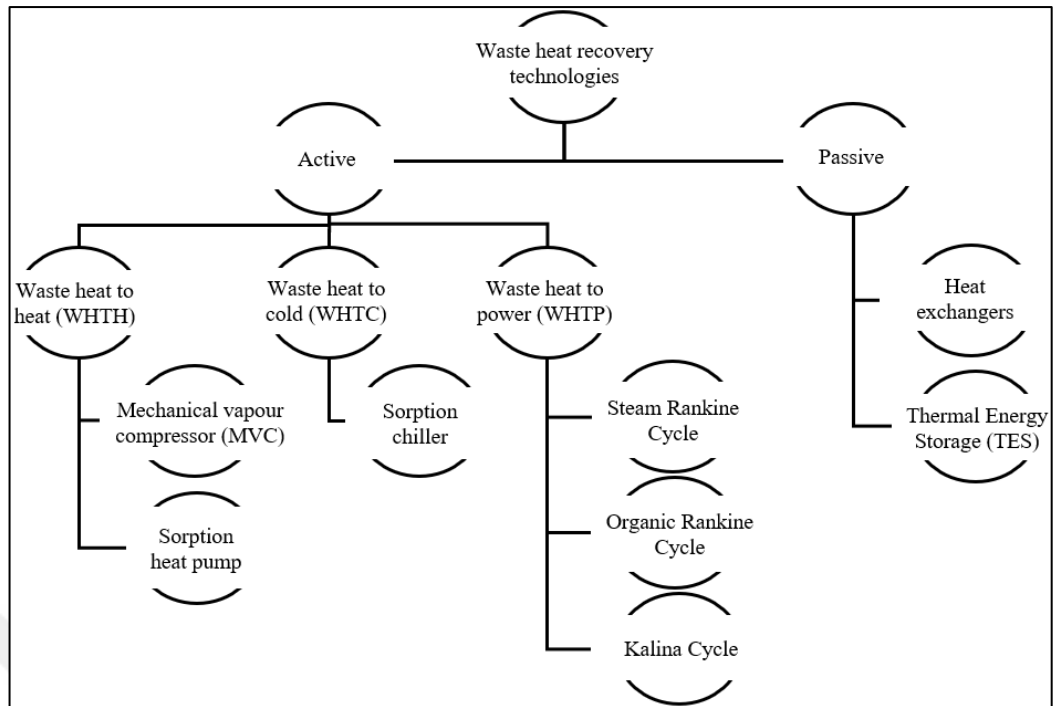


Figure 1.1: Classification of waste heat recovery technologies [Brückner *et al.*, (2015)].

Thermodynamic power cycles are the most well-known method for generating power using waste heat. Power cycles operate by converting the thermal energy, which is required for their operation, into power using waste heat. Different power cycles can be used depending on the temperature of the waste heat. The temperature ranges in which the cycles work appropriately are shown in **Figure 1.2** below.

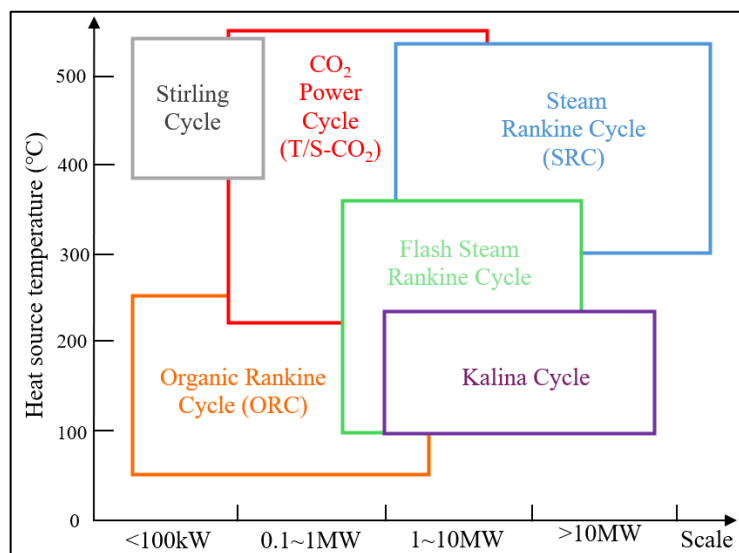


Figure 1.2: Thermodynamic cycles used at different waste heat temperatures [Liu *et al.*, (2020)].

Steam Rankine Cycle (RC) operates more efficiently at waste heat in the medium and high temperature range. Stirling Cycle is used to utilize waste heat at high temperatures. In the Organic Rankine Cycle (ORC), organic fluids with low boiling points are used, while in the Kalina Cycle (KC), a mixture of ammonia and water is utilized to efficiently operate with low-grade waste heat. The Supercritical Carbon Dioxide Cycle (s-CO₂C) is suitable for operation over a wide range of waste heat temperatures. s-CO₂C is overshadowed by ORC because of environmental concerns as the cycle operates under high pressures. ORC, which is used for low-grade waste heat, is the most studied cycle in recent years [Liu *et al.*, (2020)].

The aim of this thesis is to design the heat recovery network integrated with the power cycle with minimum cost by optimization. First, the heat recovery network is optimized and designed with minimum cost. According to the results of the optimization, it is expected to reach the minimum cost by optimizing within the cycle by integrating the waste heat into five different power cycles. Heat Exchanger Network (HEN) optimization is performed by hand calculation with the pinch method or with the help of GAMS with the SYNHEAT model. The power cycle is modeled in UNISIM DESIGN with the identified constraints and integrated with Python for a parametric study. Unlike the literature, the model developed in UNISIM DESIGN can be integrated and optimized in Python. In addition, the comparative evaluation of five different cycles in terms of cost also contributes to the literature. A schematic of HEN integrated with the power cycle is shown in **Figure 1.3** below.

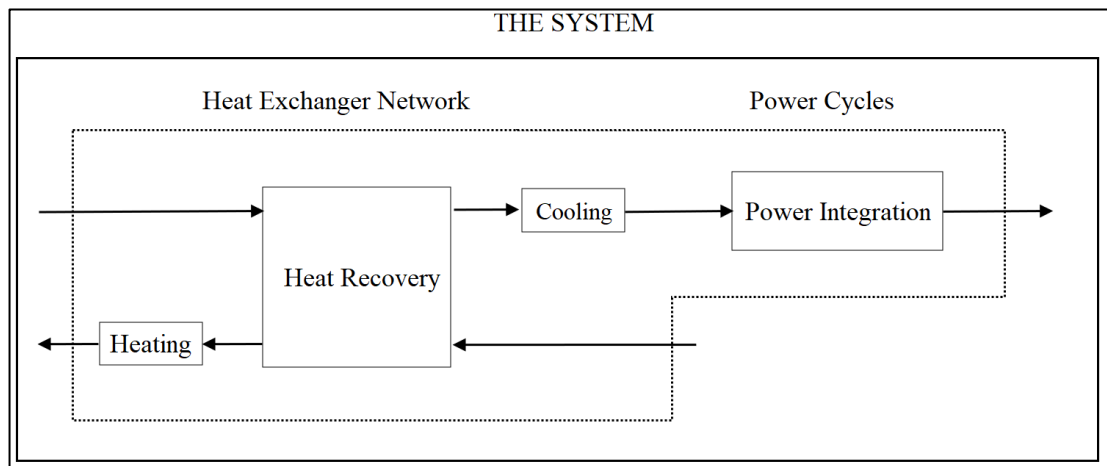


Figure 1.3: Schematic of HEN integrated with the power cycle.

Some studies that contribute to the literature are mentioned in Chapter 2. In Chapter 3, heat recovery technologies are explained in detail. Pinch analysis and MINLP model solution are explained and the characteristics of power cycles and the equipment used are introduced. In the next section, the method proposed for this study is mentioned and in section 5, the method is evaluated through two case studies. In the last section, the findings of the study are presented and discussed.



2. LITERATURE REVIEW

In the last fifty years, a lot of research has been focused on the design of HEN. The development of the concept is still being investigated up to the present day. Hohmann's PhD thesis **Hohmann, (1971)** is an important work that forms the basis for the optimization of HEN. It is considered to be the first work to introduce the ideas of the Pinch Analysis methodology.

The aim is to optimize HEN to ensure economical operation of chemical processes. The assumption of a minimum temperature approximation ($\Delta T_{\min}$), the so-called pinch temperature, is emphasized to impact on optimization. By analyzing how the temperature difference changes the heat exchanger area and equipment cost, the minimum temperature difference is used as an optimization parameter in the design.

By minimizing the heating and cooling requirements, the optimum energy transfer paths are analyzed. It is aimed to reduce unnecessary energy use by maximizing the energy transfer between hot and cold streams. During the synthesis of HEN, a perspective for the evaluation of alternative HEN is put forward. Among the designs created, the regions close to the economic solution are mentioned. The importance of optimizing energy integration in regions close to the pinch point is emphasized.

Linnhoff and Flower, (1978), a new technique for optimization of HEN is introduced by developing the method of temperature ranges, adhering to the laws of thermodynamics. Based on **Hohmann, (1971)** doctoral thesis, the method used to determine energy quantities with enthalpy-temperature diagrams was developed. A HEN is systematically constructed to ensure maximum energy recovery. By creating sub-networks, design possibilities are reduced and the process is simplified. The method proposed in this study is a frequently used method for achieving maximum energy recovery by predicting minimum energy requirements.

Umeda et al., (1979) analyzed and systematically improved the quality of energy (available energy) based on the second law of thermodynamics. By introducing the heat availability diagram and T-Q diagrams, it provides visualization for optimization and provides the basis for the development of the composite curves method.

Linnhoff and Hindmarsh, (1983) introduced the pinch design method. With this method, it is aimed to reduce the total costs of HEN that satisfy the minimum energy requirements. The minimum number and area of heat exchangers and minimum utility consumption are calculated. The economic relationship between energy recovery and capital costs is analyzed. The relationship between the decrease in the number of units and energy consumption is examined. A simplified method is developed to solve complex problems by hand calculation.

Gundersen and Naess, (1988) emphasizes that during the design of HEN, the matching of streams, material selection of equipment, physical properties changing with temperature and pressure, and pressure drops should also be taken into account during design. Among the constraints, the difficulties of the research such as some temperatures being constant and optimizing at some temperatures are discussed. In case of different energy prices, regional factors such as cooling water temperature and cost should be considered in the design. An important work has been done for the integration of thermodynamic concepts with mathematical optimization to make the HEN design more effective. The importance of mathematical programs in solving the problem together with the knowledge and abilities of engineers is expressed.

Papoulias and Grossmann, (1983) makes an important contribution to the literature by developing the transshipment model to optimize the heat exchanger network. In the developed model, heat streams are organized and thermal constraints are taken into account. A structure is introduced that allows for the splitting of streams and cyclic matching of streams. A mixed integer linear programming (MILP) method is developed to optimize the number of units used and the minimum energy requirement.

Yee and Grossmann, (1990), a mathematical model that simultaneously solves the cost factors of energy, number of units and area with mixed integer non-linear programming (MINLP) is created. With the superstructure model that creates all possible matchings in the HEN, optimum networks with lower total cost are designed compared to the old methods. Although a Pinch method based solution is created, unlike the old method, the energy recovery is optimized simultaneously instead of being determined in prior. Better results than the Pinch method are obtained and solutions are achieved in more complex problems. Flexible network structures are designed with heuristic rules.

In the study by **March, (1998)**, $\Delta T_{\min}$ values are experienced according to different industrial sectors. As shown in **Table 2.1** below, the variation of $\Delta T_{\min}$ values according to sectors is given. In oil refineries, the $\Delta T_{\min}$ value is chosen higher because of the high fouling in heat exchangers and relatively lower heat transfer coefficients. In chemical industries, it is seen that re-boiling and condensing result in good heat transfer coefficients and lower fouling. In the chemical and petrochemical industrial sectors, $\Delta T_{\min}$ is considered to be between 10°C and 20°C. The power required for cooling in low temperature processes is expensive. $\Delta T_{\min}$ value reduces in the lower cooling temperature range.

Table 2.1: $\Delta T_{\min}$ values for several types processes [**March, (1998)**].

Industrial Sector	Experience $\Delta T_{\min}$ Values
Oil Refining	20-40°C
Petrochemical	10-20°C
Chemical	10-20°C
Low Temperature Processes	3-5°C

In the study by **Ononogbo et al., (2023)**, the classification, development, advantages and disadvantages of waste heat recovery applications are evaluated. The environmental and economic impacts of different applications are examined and improvements such as reducing CO₂ emissions and increasing energy efficiency are highlighted. The advantages of ORC include its simplicity of design and its ability to recover low temperature waste heat, which makes it more popular than other cycles. Since it operates at lower pressures than KC, it is emphasized that the materials and seals of the equipment used are more economical. The negative aspect of the cycle is that the organic fluids used pose a risk of flammability in some cases. In KC, it is mentioned that it shows performance that can compete with ORC at waste heat temperatures above 200°C and that it is used efficiently in low temperature heat sources during the non-isothermal phase change process in the evaporator thanks to the ammonia-water mixture. The disadvantage of KC is emphasized that the cycle is favorable to corrosion. It is emphasized that the Trilateral Flash Cycle (TFC) can be used as a competitor to ORCs with its high performance in waste heat temperatures between 70 and 200°C and power generation per unit waste heat. However, it is

evaluated that the low isentropic efficiency of the dual-phase expanders used causes TFC to remain in the background. It is stated that s-CO₂C can generate more power and operate at higher efficiency than ORC in waste heat recovery. It is emphasized that the absence of a dual-phase process in the evaporator of the cycle provides a better thermal coupling and low irreversibility. Since CO₂ has a low critical temperature (31.1°C), lower temperatures are needed for its condensation. Cooling the fluid at low temperatures causes the design complexity of the cooling process. Working at high pressures creates safety problems. This is seen among the disadvantages of the cycle.

In the study by **Tchanche *et al.*, (2011)**, the properties of organic fluids used in ORC and water are compared. The critical temperature and pressure values, boiling point and specific heats of organic fluids are lower than water, while their viscosity is relatively high. It is shown that organic fluids have acceptable condensation pressure values against the low condensation pressure of water. Environmental impacts such as flammability and toxicity, which are not the case for water, are fluid-dependent conditions in organic fluids and require attention. Organic fluids are more difficult to access and somewhat more expensive than water.

In the study by **Herath *et al.*, (2020)**, the effects of seven different fluids used in ORC such as Benzene, Propane, Ethanol, R-245fa, R-134a, Methanol and Acetone on the performance of the cycle are compared. Instead of fluids such as chlorofluorocarbon and hydrochlorofluorocarbon, which have high negative environmental impacts, different environmentally beneficial fluids are evaluated as alternatives. It is observed that thermal efficiency increases with increasing evaporator pressure, while efficiency decreases with increasing temperature in the condenser. In the selection of the fluid, different temperature and pressure levels, generated power ratio, specific mass flow rates for each fluid are evaluated and these criteria are taken into consideration in the selection of the fluid. It is stated that Benzene and Methanol, which provide high performance of ORC, can be used as alternative fluids.

In the study by **Chen *et al.*, (2010)**, 35 different fluids to be used in ORC are examined. In addition to thermodynamic and physical properties such as density and enthalpy of vaporization, environmental impacts, safety and costs are also evaluated among the selection criteria. Fluids are analyzed in three groups as dry, wet and isentropic. Dry fluids are preferred in ORC since they do not form liquid droplets during expansion.

It is emphasized that wet fluids reduce turbine efficiency as superheating is required. Isentropic fluids are used in both ORC and Supercritical Rankine Cycle (SRC). Benzene and Toluene are emphasized to be suitable for use in ORC due to their high critical temperature and high stability properties. CO₂ is considered to be suitable for SRC. Although it is preferred due to its environmental impacts and low cost, it is stated that the low critical temperature creates some difficulties.

Chen et al., (2006) in a study, The Transcritical Carbon Dioxide Cycle (T-CO₂C) for power generation from low temperature waste heat is compared with ORC using R123 as the working fluid. The thermodynamic analysis of these two cycles is performed to analyze which cycle is more efficient, as well as the environmental impacts and design advantages. The heat transfer in the heat exchanger between the heat source and the working fluid in the T-CO₂C is analyzed. It is shown that due to the “temperature glide” of T-CO₂C above the critical temperature, a better match is achieved because the evaporator of the cycle does not have the pinch constraint that occurs in the evaporator in the ORC. It is observed that the T-CO₂C provides slightly higher power output than the ORC due to the higher operating pressure and a better heat transfer coupling in the evaporator of the cycle. It is emphasized that the total power output should also be taken into account, as interpretation based on thermodynamic efficiency may give misleading results.

In the study **Liu et al., (2020)**, the place of s-CO₂C in the literature and its thermodynamic advantages in converting waste heat into energy are examined. It is shown that s-CO₂C is used in waste heat sources at low, medium and high temperatures. It is indicated that s-CO₂C performs efficiently over a wide temperature range and has a low carbon footprint. It provides a more compact and flexible design than the conventional RC in generating power from waste heat. The performances of different s-CO₂C such as simple, regenerative, two-stage are analyzed at different temperature levels. It is evaluated that two-stage or cascade combined cycles are advantageous for medium and high temperature waste heat. It is stated that transcritical cycle gives better performance for low temperature waste heat and supercritical cycle gives better performance for medium and high temperature waste heat. The difficulties in the development and maintenance of equipment such as heat exchangers and turbines in commercial applications of s-CO₂C are evaluated. Disadvantages such as pressure drops in heat exchangers are also evaluated.

Li *et al.*, (2018) in their study, an experimental analysis of ORC and T-CO₂C for electricity generation from low-temperature waste heat and comparison of the two cycles is performed. The experimental data of these cycles mentioned in previous theoretical studies are compared and calculated under similar operating conditions. The advantages of the CO₂ cycle are evaluated by thermodynamic and exergy calculations. Important data for the design and control of heat exchangers, turbines and other equipment are shared. ORC using organic fluids such as R245fa as the working fluid has been extensively researched for generating electricity from low temperature waste heat. Although ORC is widely used, its disadvantage compared to CO₂C is that the constant temperature evaporation in the evaporator causes pinch restriction and losses in heat transfer. CO₂C bypasses the two-phase region, resulting in less losses and better thermal coupling. It requires special equipment due to the high critical pressure. In this study, the importance of data such as temperature, pressure and mass flow rate affecting turbine efficiency is emphasized. Recommendations are made for turbine and heat exchanger designs.

In the study by **Iqbal *et al.*, (2020)**, TFC is preferred for low temperature waste heat recovery, highlighting its advantages over the widely used ORC. The study reports that TFC provides high efficiency from heat sources below 80°C. TFC, which is usually investigated theoretically, has been studied experimentally using an organic working fluid and a fixed converging-diverging nozzle. In this study, the applicability of TFC for the recovery of low temperature industrial waste heat is demonstrated. The economic case of the TFC integrated system is analyzed and the payback period of the investment cost is less than ten years. Due to the phase change of the working fluid of the ORC, losses are experienced in the evaporator, while these losses can be minimized in TFC. Since the expansion in TFC takes place in two phases, there is a better match in the heat transfer curves, which increases energy efficiency. The efficiency of TFC depends on the isentropic efficiency of the nozzle and the turbine. It is shown that the performance is improved for nozzles with larger geometry. It is emphasized that when TFC is integrated into an existing ORC, the total energy capacity can increase to around 130%.

In the study by **Lai *et al.*, (2019)**, the performances of TFC and ORC are compared based on the pinch point concept. Since the heat input process in TFC is non-isothermal and there is no phase change, it provides higher exergy efficiency in the pinch point

problem. ORC and TFC are analyzed using different working fluids (R245fa, R134a, R236fa, R1233zd). It is shown that the net power generation increases when R245fa is used in TFC. The minimum temperature difference in both cycles is taken as 5°C and it is shown that TFC provides higher heat transfer values than ORC. According to the results of the study, 30% more power generation is achieved in TFC under the same conditions. However, due to the high pump power of TFC, it is explained that the costs and reduction of the pump power should be investigated.

In the study by **Rohde *et al.*, (2018)**, the performance of RC and TFC is compared. It is aimed to determine the potential of TFC by optimizing power generation from low temperature heat sources. The study analyzes the efficient use of heat sources, especially at low temperatures such as 100°C, 150°C and 200°C, and provides recommendations. The effects of system dimensions such as heat exchanger area on performance are examined. The effects of working fluids with low environmental impacts on the cycle are also examined. The challenges of dual-phase expansion in TFC and the selection of suitable equipment such as screw expanders are highlighted. It is argued that RC has higher expander isentropic efficiency and TFC performs better at low temperatures and under certain conditions.

In **Smith, (1993)** research, the power potential for heat sources between 100°C-200°C is analyzed and its advantages over RC are discussed. It is stated that the dual-phase expansion in TFC improves energy efficiency. It is emphasized that hydrocarbons such as n-pentane are the most suitable working fluid in TFC due to thermal stability and cost reduction. The working principles for designing dual-phase expanders and their effects in TFC compared to RC are analyzed. The design, applicability and efficiency of the expander are evaluated. Based on the net power generated per unit mass and cost per unit power generated between TFC and RC, it is calculated that the expander used in TFC is approximately 20-60% more costly than RC. The fact that the expander has an adiabatic efficiency of more than 75% means that TFC's theoretical superiority can be translated into practice is shown. It is shown that TFC can be constructed at similar costs to RC and its thermodynamic advantages. A modeling approach for optimizing the heat exchangers, expander and costs of TFC is presented.

In the study by **Wang *et al.*, (2024)**, a new system is proposed to improve the performance of KC for power generation from a low temperature heat source. By

integrating the KC with an ammonia-water absorption refrigeration cycle, the thermal efficiency can reach 100% by utilizing the entire waste heat. The area of the heat exchangers, the power of the turbine and other equipment in a combined heating and power generation plant based on KC are analyzed from a thermodynamic and economic point of view. A system is presented in which both power and hot water are obtained from low temperature waste heat with ammonia-water solution. It is stated that increasing the ammonia concentration and turbine inlet temperature and decreasing the turbine inlet pressure enhance the exergy efficiency of the cycle. The proposed system is shown to be economically viable with a payback period of 2.31 years.

In the study by **Zhu *et al.*, (2022)**, integration of ORC and KC with the s-CO₂ Brayton Cycle is analyzed. It is aimed to determine the advantages of ORC and KC under different conditions. Comparisons are made between the working fluids used in ORC and KC, such as R32 and ammonia, in terms of cycle efficiency, environmental impacts and safety. It is emphasized that the heat exchanger in KC operates in accordance with the temperature changes in the heat source due to the temperature glide property, but this feature cannot be used due to the slope close to the critical point of s-CO₂C. In the design of the evaporator, the effects of superheating and ammonia-water mixture on reducing heat transfer losses are analyzed. Although ORC is popular for the recovery of low temperature waste heat due to the high preference of the working fluid used, it is stated that the working fluid causes exergy losses by increasing the temperature differences in the evaporator until it reaches the saturated vapor phase. ORC has better cycle efficiency and compactness while KC is better in terms of better matching with the temperature of the heat source is indicated.

In the study by **Hu *et al.*, (2020)**, a new ammonia water power cycle based on the KC is introduced. A new method for power generation from low and medium temperature geothermal energy sources is presented. The power generation rate is optimized by analyzing the performance of ammonia and water mixture under different operating conditions. According to the theoretical results obtained, the optimum circulation ratio of ammonia water solution in the model is stated as 4 times. It is shown that the concentration difference of ammonia water mixture is 12-13%. It is stated that the amount of power generated increases at higher ammonia water concentration. Different ammonia-water concentrations are examined and it is observed that 35%

concentration is at the lowest levels in the efficiency of the cycle and the efficiency of the power produced. It is stated that the efficiency of the cycle depends on many factors such as circulation rate, ammonia concentration, temperatures of the sources from which the cycle receives and gives heat, temperature differences in heat exchangers.

In the study by **Modi et al., (2015)**, the economic optimization of KC is performed to minimize the investment cost and KC is compared with RC. The efficiency and cost of the cycle are examined by including turbine outlet pressure, separator inlet temperature and ammonia water concentration parameters in the optimization. Ammonia water concentration is evaluated between 0.5 and 0.8 and total investment cost per unit power produced is calculated. According to the results, although KC and RC have almost the same investment costs, KC is slightly more economical. The maximum investment cost is calculated when the ratio of ammonia water mixture is 0.7.

In the study by **Elsayed et al., (2013)**, instead of ammonia water mixtures in KC, zeotropic mixtures are studied. KC is compared with ORC at low temperature heat sources. KC using ammonia water mixture with 0.55 concentration is reported to be 20-40% more efficient than ORC using pure ammonia and pure R134a at the same operating conditions. 19 different zeotropic fluids are tested and it is shown that mixtures containing propane and propylene can be used as an alternative to ammonia water mixture. Despite the environmental impacts of the ammonia water mixture, such as low global warming and zero ozone depletion potential, it is emphasized that care should be taken in safety measures as it is a toxic mixture.

In the work of **Beck and Hofmann, (2018)**, optimization of HEN in three different cases. In the first HEN example, the total area and annual cost of the structure analyzed by pinch solution as a linear model are calculated. The same HEN is then analyzed for total area and cost using the MINLP formulation. In the MINLP model solution, total heat exchanger area and utility increases while total annual cost and number of units decreases. It is presented as a mixed integer linear programming (MILP) method by simplifying the complexity of classical MINLP methods. It is emphasized that the optimization process is accelerated by expressing nonlinear equations together with linear inequalities.

In the study by **Watanapanich *et al.*, (2022)**, a mathematical model that simultaneously optimize HEN and ORC is presented. It is mentioned that the thermodynamic properties of the cycle are taken by REFPROP and modeled with linear regression equations using microsoft excel, which simplifies the solution of the mathematical problem. With GAMS, wide temperature ranges are obtained in the simultaneous optimization of the ORC and HEN. Four different ORC configurations are analyzed as well as the total cost based on the unit price of electricity. It is also mentioned that the total cost decreases as the unit price of electricity increases.

In the study by **Hipólito-Valencia *et al.*, (2014)**, ORC and HEN are optimized simultaneously for electricity generation and electricity revenue by using low-temperature waste heat. In configurations where ORC is considered, operating costs, investment costs and energy integration are optimized together. In addition to the stages with heat exchangers between the hot and cold streams, an additional stage is used, where heat exchangers connected to the ORC are installed. The model aims to generate the maximum amount of electricity while ensuring minimum cost.

3. HEAT RECOVERY TECHNOLOGY

In this section, the steps and details of the pinch analysis and MINLP model optimization used in the design of the heat exchanger network are described. Then, the power cycles integrated into the heat exchanger network are introduced.

3.1. Pinch Analysis

The fundamentals of Pinch analysis date back to the 1970s. In 1978, it was developed by Linnhoff et al. The pinch method is continuously being improved and modified by experts. This is a reason why, in order to arrive at an optimal design, some insights are needed that are not intuitively clear or recognised by common sense. The first step of the pinch analysis is to identify the energy target. The pinch method is used to maximise heat recovery and minimise external heating and cooling, so-called utilities. In the Pinch method, manual calculations are made using $\Delta T_{\min}$. The minimum temperature approach refers to the temperature difference between the hot and cold streams in the heat exchangers [Kemp, (2007)].

In the Pinch method, the first law of thermodynamics is first analyzed from Equation (3.1) below. In other words, hot streams transfer their heat to cold streams and reach the target temperature, while cold streams reach the target temperature values with the heat they receive. In order for heat transfer to occur, there must be a positive temperature driving force between the hot and cold streams. In also, the outlet temperature of the cold stream cannot exceed the inlet temperature of the hot stream. This means in addition to the first law, the second law must be satisfied [Douglas, (1988)].

$$Q = F_{CP} \Delta T \quad (3.1)$$

The Pinch method provides maximum energy recovery (MER) between streams in heat exchangers and minimum energy requirement (MER) to minimize the heating and cooling load used in utilities. It determines the investment and operating costs via MER [Dimian, (2003)].

3.1.1. Temperature Intervals and Cascade Diagram

In the Pinch method, the first law analysis is first checked. In the first law analysis, it is calculated how much heat the hot streams can transfer and how much heat the cold streams can receive. Then $\Delta T_{\min} 10^{\circ}\text{C}$ (minimum driving force) is assumed between the hot and cold streams. Two separate temperature scales are created for hot and cold streams depending on $\Delta T_{\min}$. The data given in the problem description are transferred to this graph. The inlet and outlet temperatures of the streams are shown on the graph by means of an arrow.

Heat transfer takes place from the stream with high temperature to the stream with low temperature. How much heat the streams transfer to each other in each temperature range is read on the graph. According to the first law, the point where the heat flow stops refers to the pinch point. In order to ensure the operation of HEN, it is necessary to use enough hot utilities to pass heat through the pinch point. It is determined that the excess waste heat is supplied by the cold utility and the deficient heat is supplied by the hot utility. The hot and cold utilities used can be easily read from the Cascade Diagram. Representative Temperature Interval and Cascade Diagrams are shown in **Figure 3.1** and **Figure 3.2** below [Douglas, (1988)].

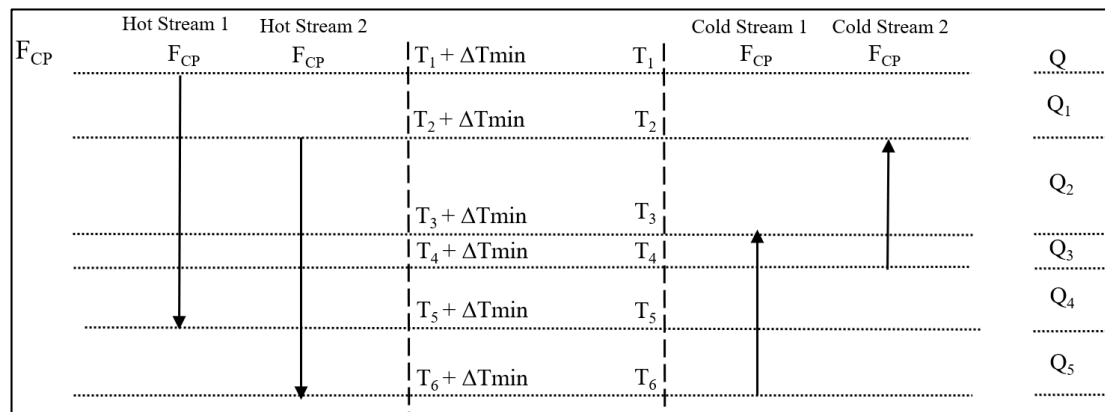


Figure 3.1: Temperature intervals diagram [Douglas, (1988)].

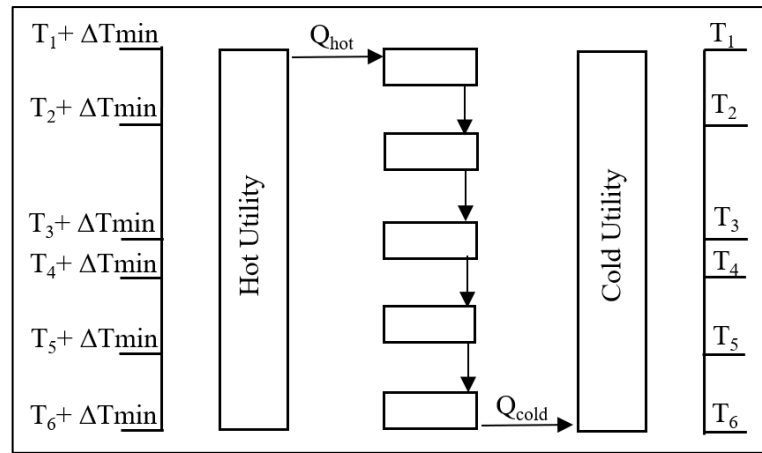


Figure 3.2: Cascade diagram [Douglas, (1988)].

3.1.2. Composite and Grand Composite Curves

The key point in the pinch method is the Grand Composite Curve (GCC). Composite Curve provides minimum energy requirement from stream data. A representative Composite Curve and GCC is shown in **Figure 3.3** below. It is expressed as the minimum hot utility Q_{hot} and minimum cold utility Q_{cold} required for the operation of HEN. The heat recovery between hot and cold streams is expressed as $Q_{recovery}$ and the minimum temperature approach is expressed as ΔT_{min} .

The composite curve is obtained when the hot and cold streams are integrated separately into the Temperature-Enthalpy Diagram. Then, when these two curves are combined, GCC is obtained. GCC shows the amount of hot and cold utilities, heat recovery and pinch point. It plays an important role in determining the location of the power cycle in heat and power integration [Dimian, (2003)].

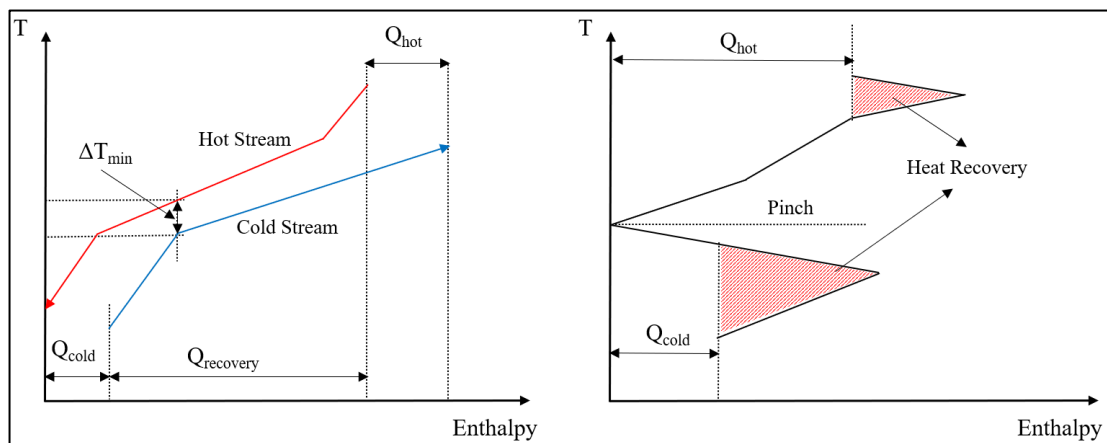


Figure 3.3: Composite and grand composite curve [Dimian, (2003)].

3.1.3. Golden Rules of Pinch Method

To design HEN, the problem is divided into two parts: below pinch and above pinch. This is necessary to simplify the problem. In the pinch method, there are some golden rules to ensure maximum energy recovery and minimum utility consumption. These;

- Heat transfer through the pinch is not allowed.
- Only hot utility is used on above the pinch, no cold utility is used.
- Only cold utility is used below the pinch, no hot utility is used.

When designing HEN, which is divided into two as above pinch and below pinch, the design is made by starting from the pinch point. Hot utilities are only used above the pinch and cold utilities are only used below the pinch. Above the pinch, all hot streams must reach the pinch temperature, giving up their heat to the cold streams. On the contrary, below the pinch all cold streams must reach the pinch temperature by receiving heat from the hot streams [Kemp, (2007)].

Since no cold utilities are used above the pinch, the number of cold streams and their heat capacity must be greater than or equal to the hot streams. Below pinch, the number of hot streams and their heat capacitance must be greater than or equal to the number of cold streams since no hot utilities are used. For possibilities that do not correspond to this situation, splitting the streams means that the design problem has to be started from the beginning. The general algorithm is schematized in **Figure 3.4** shown below [Linnhoff and Hindmarsh, (1983)].

- $N_{\text{Hot Stream}} \leq N_{\text{Cold Stream}}$ and $CP_{\text{Hot}} \leq CP_{\text{Cold}}$ (for above the pinch) all the streams.
- $N_{\text{Cold Stream}} \leq N_{\text{Hot Stream}}$ and $CP_{\text{Cold}} \leq CP_{\text{Hot}}$ (for below the pinch) all the streams.

where $N_{\text{Hot Stream}}$ and $N_{\text{Cold Stream}}$ denote the number of hot and cold streams. Also CP_{Hot} and CP_{Cold} represent the heat capacity of the hot and cold streams [Kemp, (2007)].

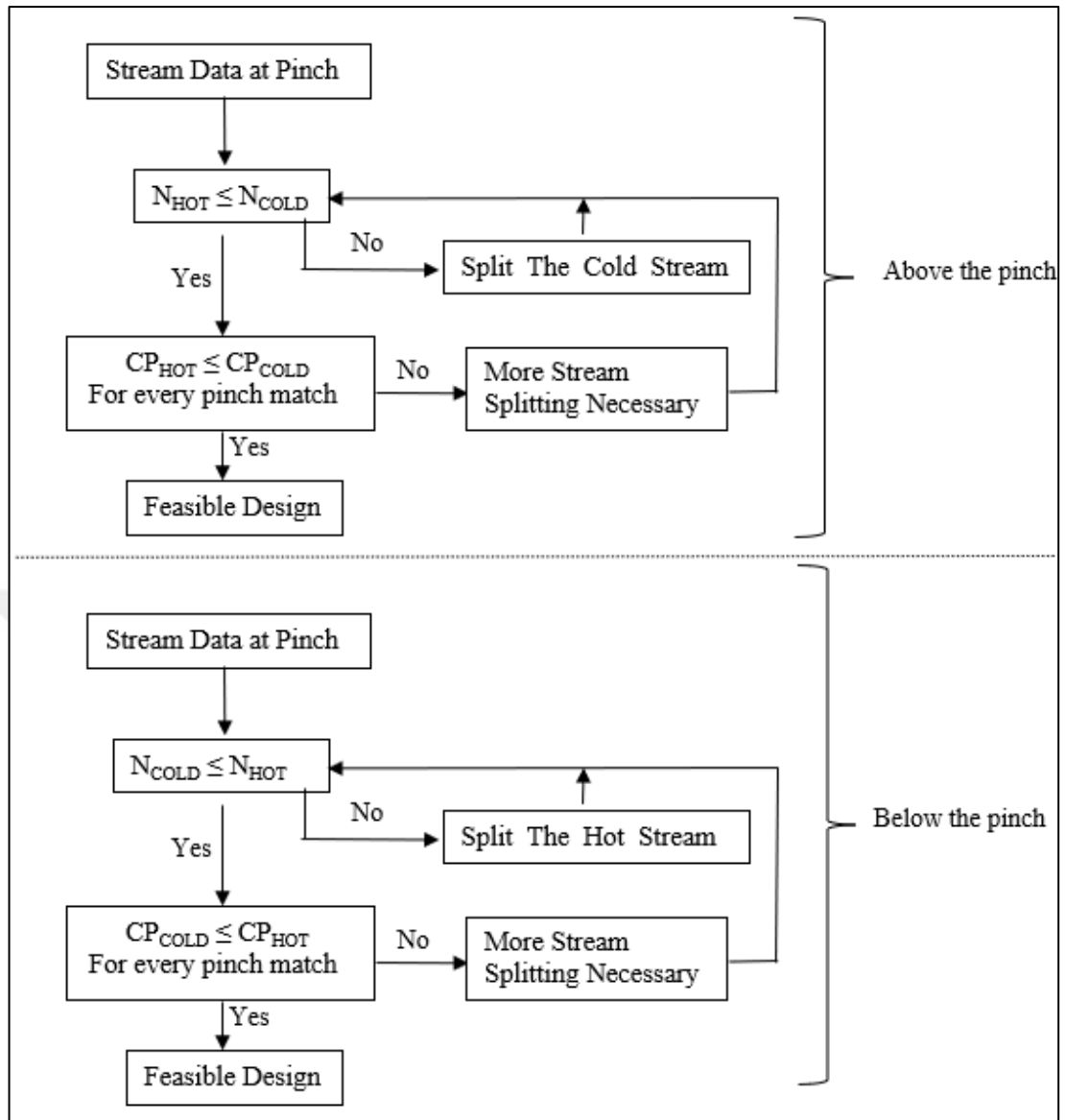


Figure 3.4: Stream splitting algorithms [Linnhoff and Hindmarsh, (1983)].

3.1.4. Minimum Temperature Approximation ($\Delta T_{\min}$)

In order to design HEN, a minimum temperature driving force must be selected. The minimum level of utility usage and at the same time the minimum number of heat exchangers used depends on this temperature approach. As $\Delta T_{\min}$ decreases, the utilities used decrease and the energy cost decreases accordingly. The opposite is the case for the area. As $\Delta T_{\min}$ decreases, the utility decreases and the area increases, thus the capital cost increases [Biegler *et al.*, (1997)]. Figure 3.5 below shows the relationship between $\Delta T_{\min}$, area and utilities consumption [Linnhoff and Ahmad, (1990)].

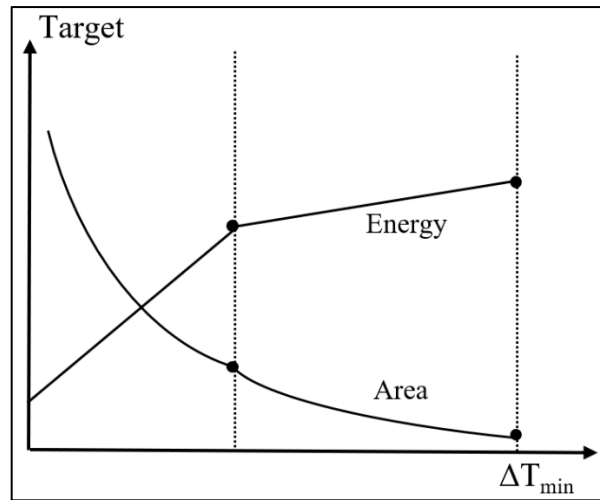


Figure 3.5: The relationship between $\Delta T_{\min}$ and area [Linnhoff and Ahmad, (1990)].

In **Figure 3.6**, the relationship between investment, operating and total cost depending on $\Delta T_{\min}$ is shown on the graph. Increasing the cost of the utilities leads to an increase in the operation cost. An increase in the area leads to an increase in the investment cost. The sum of the Operating and Investment Costs is called the Total Annual Cost (TAC) and as can be seen from **Figure 3.6**, the point where TAC is minimum is the optimum value of $\Delta T_{\min}$ [Ahmad *et al.*, (1990)].

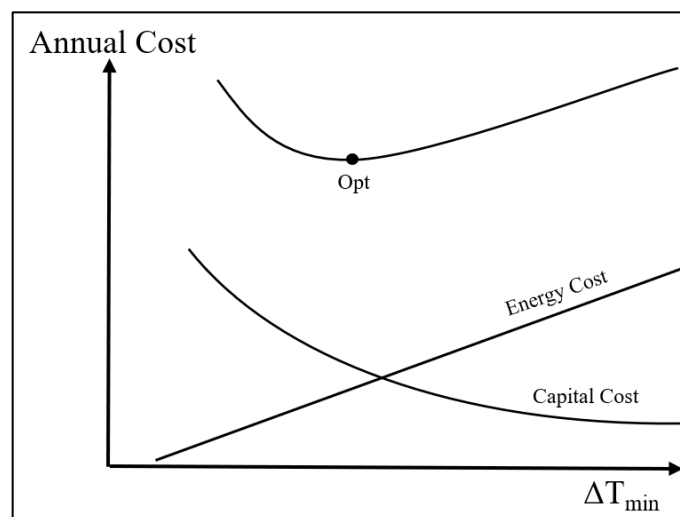


Figure 3.6: The relationship between TAC and $\Delta T_{\min}$ [Ahmad *et al.*, (1990)].

3.1.5. Loop and Paths

In order to minimize the number of heat exchangers, the concept of paths and loops needs to be analyzed. Loops and paths show the possibilities of shifting loads in HEN.

A loop refers to the way starting from a heat exchanger and returning back to the same heat exchanger. The loop can consist of heat exchangers as well as utilities. If there is a loop in the network, it means that there is an extra heat exchanger being used. Hence, breaking the loop implies a reduction in the number of heat exchangers. The representation in **Figure 3.7** below is called a Grid Diagram and shows a loop.

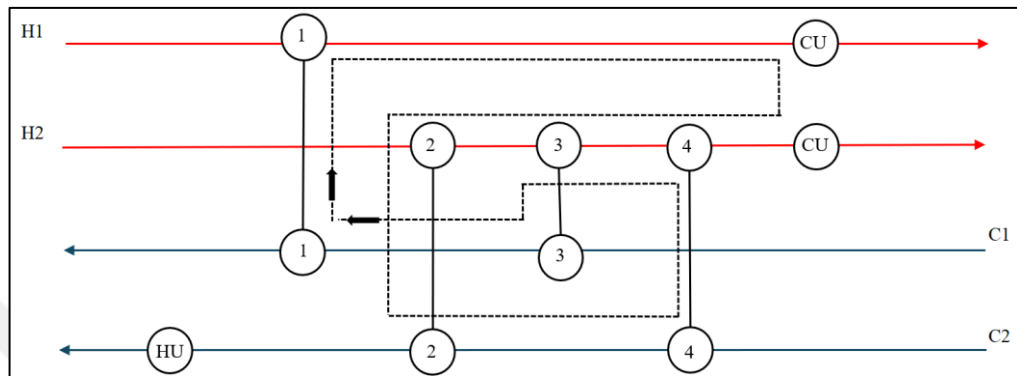


Figure 3.7: A loop representation [Douglas, (1988)].

A path represents the way between the hot utilities and the cold utilities in the heat recovery network. The loads of heat exchangers on one path can be shifted along the same path. The load shift on the utilities can be covered by another heat exchanger on the same path without changing the total heat load. **Figure 3.8** below shows a representative path via a Grid Diagram.

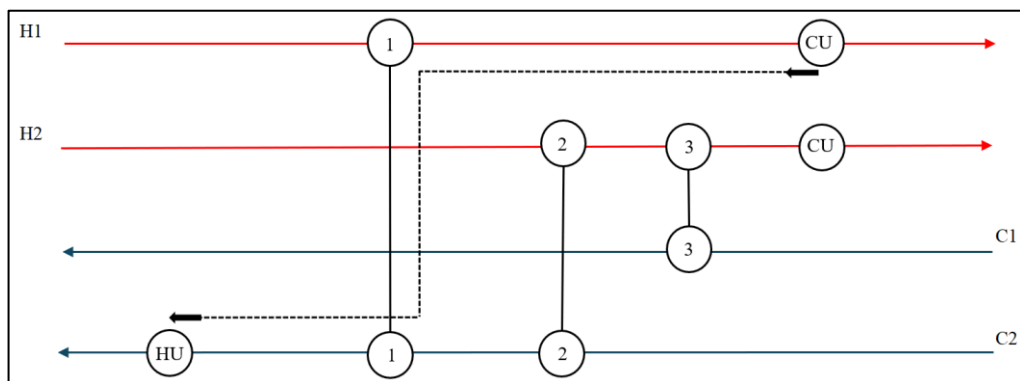


Figure 3.8: A path representation [Douglas, (1988)].

Loop breaking has three steps as a heuristic method. First, the loop with the heat exchanger with the lowest heat load is broken. The heat exchanger with the lowest heat load is always eliminated from the loop. If a loop is broken passing the pinch point,

$\Delta T_{\min}$ is violated. If $\Delta T_{\min}$ is violated, a restoration is necessary. The path concept is used for this [Douglas, (1988)].

3.1.6. Number of Unit and Area

Achieving maximum heat recovery and minimum consumption of external resources, i.e. utilities, will reduce operating costs. However, to design the optimum heat recovery network, it is not enough to reduce only operating costs. Since TAC depends on both operating and investment costs, investment costs should also be considered. Two important factors that make up the investment cost are the number and area of heat exchangers. The more heat exchangers used in the heat recovery network, the higher the investment cost will be. Because each heat exchanger has a certain fixed price for installation in the plant. Also, like the number of heat exchangers, the area also affects the costs. As the area of the heat exchanger increases, the investment cost will naturally increase [Towler and Sinnott, (2008)].

Hohmann, (1971) calculated the minimum number of heat exchangers as follows;

$$E_{\min} = N - 1 \quad (3.2)$$

where $E_{\min}$ is minimum number of heat exchangers (including heaters and coolers), N is number of streams (including utilities).

While the relationship used by **Hohmann, (1971)** is applicable for simple networks, **Linnhoff *et al.*, (1979)** derived the following equation for complicated heat exchanger networks;

$$E_{\min} = N - L - S \quad (3.3)$$

where L is number of loops in the networks, S is number of separate components available in the network.

For optimum design, extra heat exchangers are avoided, so the network is designed with $L = 0$. Usually the separate components are calculated with $S = 1$ and the general Equation (3.2) above is obtained. [Towler and Sinnott, (2008)], [Kemp, (2007)]

The number of heat exchangers is also expressed as follows, as calculated by **Douglas, (1988)**;

$$\left(\begin{array}{c} \text{Number} \\ \text{of} \\ \text{Exchangers} \end{array} \right) = \left(\begin{array}{c} \text{Number} \\ \text{of} \\ \text{Streams} \end{array} \right) + \left(\begin{array}{c} \text{Number} \\ \text{of} \\ \text{Utilities} \end{array} \right) + \left(\begin{array}{c} \text{Number} \\ \text{of} \\ \text{Loops} \end{array} \right) - \left(\begin{array}{c} \text{Number} \\ \text{of} \\ \text{Independent} \\ \text{Problems} \end{array} \right) \quad (3.4)$$

The other factor affecting the design and also the cost is the heat exchanger area. The area calculation should be taken into account to reach the minimum cost in the targeted design. **Nishida *et al.*, (1981)** reviewed studies for the calculation of the minimum heat transfer area. The following Equation (3.5) is presented for equal heat transfer coefficients of all streams and countercurrent heat transfer conditions;

$$A = \frac{1}{U} \int_1^2 \frac{dQ}{\Delta T} \quad (3.5)$$

$$A_{min} = \frac{1}{U} \sum_j \frac{Q_j}{\Delta T_{LMj}} \quad (3.6)$$

If the overall heat transfer coefficient (U) is not given, it can be calculated from Equation (3.7) below using the individual heat transfer coefficients of the hot and cold streams [**Douglas, (1988)**];

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_j} \quad (3.7)$$

Townsend and Linnhoff, (1984) proposed Equation (3.8) by considering the individual heat transfer coefficient for streams. Designing with the individual heat transfer coefficient instead of the overall heat transfer coefficient provides a significant contribution to area and cost estimates.

$$A_{min} = \sum_j \frac{1}{\Delta T_{LMj}} \sum_k \left[\frac{\Delta q_k}{h_k} \right] \quad (3.8)$$

where U is the overall heat transfer coefficient, h_i, h_j is individual transfer coefficients for stream, Q is the heat transfer, ΔT is the difference between two streams, A represents the area, ΔT_{LM} is log-mean temperature difference for streams. [**Gundersen**

and Naess, (1988)] The heat exchanger area is calculated using ΔT_{LM} formula given in Equation (3.9) below.

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \quad (3.9)$$

where $\Delta T_1 = (T_{h,i} - T_{c,o})$, $\Delta T_2 = (T_{h,o} - T_{c,i})$, the sub-index h represents the hot stream and the sub-index c represents the cold stream. The sub-index i represents the inlet location and the sub-index o represents the outlet location. **[Seider *et al.*, (2009)]**

3.1.7. Cost of Heat Exchanger Networks

Part of the investment costs in a heat network are caused by the heat exchangers. Therefore, the calculation of the investment cost of a network depends on the area. In order to calculate the investment cost of a heat network, it is necessary to know the heat exchanger areas. The cost of a heat exchanger with surface area A is evaluated according to Equation (3.10) below;

$$\text{Installed Capital Cost of Heat Exchanger} = a + b(A)^c \quad (3.10)$$

where a , b , c coefficients are cost constants that vary depending on many factors such as heat exchanger type, pressure value, construction material. After calculating the cost of a heat exchanger using Equation (3.10) above, this equation is applied to all heat exchangers in the network **[Smith, (2005)]**.

Investment cost consists of the fixed installation cost of components such as heat exchangers and the costs of heat exchangers due to the area. Operating costs are the costs related to the utilities consumed. In the calculation of TAC, there is a triple relationship between the number of heat exchangers, area and utilities consumption as shown in **Figure 3.9** below. Optimizations in costs should be made simultaneously for the utilities, number of heat exchangers and area **[Yee and Grossmann, (1990)]**.

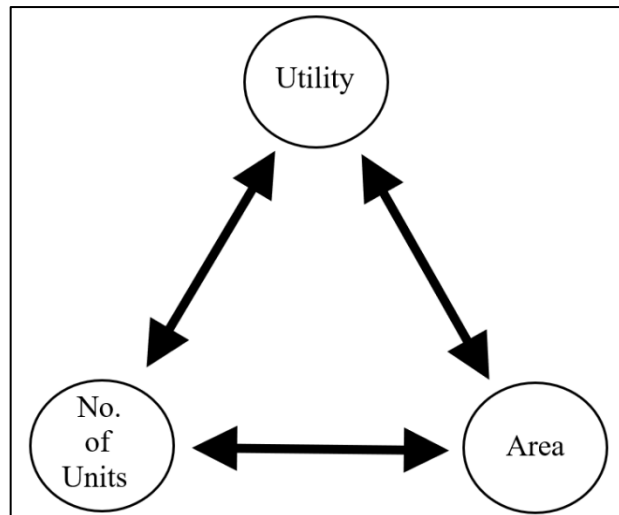


Figure 3.9: Trade-off between costs in design [Yee and Grossmann, (1990)].

3.2. Superstructure MINLP Model

Optimizations involving continuous and integer variables are called Mixed Integer Problems. The presence of integer variables in the problem allows complex decisions to be modeled through graphical representation called superstructures. These superstructures allow simultaneous determination of the optimal structure and parameters of a heat network. MINLP are applied in engineering applications such as HEN synthesis.

In the MINLP model solution, it is determined which streams in HEN should be matched with which streams or which utilities, and the annual cost of HEN is optimized by determining how much load will be on the heat exchangers. An expression that results in (0 and 1) is called a binary variable. And they represent whether a heat exchanger is present or not at any location in the heat network [Floudas *et al.*, (1999)].

HEN optimization aims to determine the structure of the process and design variables such as inlet/outlet temperatures of the streams, heat exchanger areas, utility consumption. In HEN optimization, integer variables are used to select heat exchangers from all possible matches between hot and cold streams. The heat exchanger network (HEN+utility) problem is a MINLP and is difficult to solve. The MINLP model problem is solved by defining the initial parameters and boundary conditions given in the problem definition in GAMS.

Yee and Grossmann, (1990) have developed a MINLP model problem as a SYNHEAT superstructure. HEN superstructure is shown in **Figure 3.10** below. The SYNHEAT solution by **Yee and Grossmann, (1990)** is basically based on the pinch method. The techniques of synthesizing HEN with SYNHEAT simultaneously have some limitations. These are; In the optimization, the utilities are not included in the superstructure or model. Instead, the utilities were positioned at the hot and cold ends of the superstructure. The inclusion of the mass flow rates of the utilities is not included in the optimization since it would complicate the solution of MINLP. Possible matching of utilities can only be in parallel, because an utility is considered as a sub-stream for a possible matching with process streams [**Martelli et al., (2017)**].

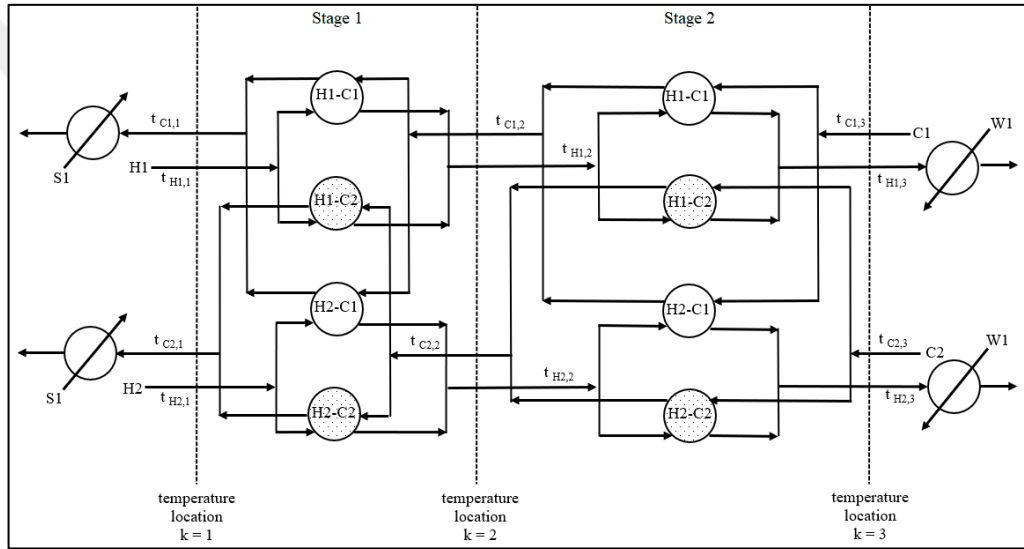


Figure 3.10: Heat exchanger network superstructure [**Yee and Grossmann, (1990)**].

A simple optimization problem without many variables and constraints, for example a problem with two hot and two cold streams, can be solved by hand using the pinch method. However, as the number of variables and constraints increases, i.e. when the problem becomes more complex, such as a problem with four hot and three cold streams, manual solution with the pinch method will make the solution difficult, so MINLP model solution with SYNHEAT is preferred. As shown in **Figure 3.10** above, the superstructure is divided into stages and the possible matches at each stage are identified. Where k indicates the number of steps. The SYNHEAT solution determines the possible matches and temperatures at each of the k stages, as well as the heat exchanger loads and areas, and the utility loads [**Yee and Grossmann, (1990)**].

In the SYNHEAT model designed by **Yee and Grossmann, (1990)**, Chen Approximation developed by **Chen, (1987)** is used for area calculation instead of ΔT_{LM} . ΔT_{LM} and Chen Approximation are quite similar. When using the ΔT_{LM} method, uncertainty occurs in the denominator in case $\Delta T_1 = \Delta T_2$ as seen in Equation (3.9). With the Chen Approximation shown in Equation (3.11) below, this uncertainty is eliminated [**Yerramsetty and Murty, (2008)**].

$$\text{Chen Approximation} = \left\{ \Delta T_1 \Delta T_2 \left(\frac{\Delta T_1 + \Delta T_2}{2} \right) \right\}^{\frac{1}{3}} \quad (3.11)$$

In the SYNHEAT model solution, the following Equation (3.12) represents the mathematical modeling of the objective function, Equation (3.13) the constraints and Equation (3.14) the variable bounds. In the objective function that minimizes TAC of HEN, the first expression includes the used utilities costs, then the fixed installation costs for each heat exchanger. The last expression shows the costs of the heat exchangers due to the area. The constraints shown in Equation (3.13) represent heat loads of process streams, utility amounts and temperature constraints at each stage in the superstructure model. The variable bounds shown in Equation (3.14) represent the limits of heat loads that process streams can supply or absorb and the temperature limits at each stage in the superstructure model [**Floudas et al., (1999)**].

OBJECTIVE FUNCTION (3.12)

$$\begin{aligned} \min_{T, Q, \Delta T, z} \quad & \sum_{i \in HP} C_{CU} Q_{CU,i} + \sum_{j \in CP} C_{HU} Q_{HU,j} + \sum_{i \in HP} \sum_{j \in CP} \sum_{k \in ST} C_{F_{ij}} z_{ijk} + \sum_{i \in HP} C_{F_{i,CU}} z_{CU,i} + \sum_{j \in CP} C_{F_{j,HU}} z_{HU,j} \\ & + \sum_{i \in HP} \sum_{j \in CP} \sum_{k \in ST} \left[\frac{C_{ij} Q_{ijk}}{U_{ij} \left[\frac{\Delta T_{ijk} \Delta T_{ijk+1} (\Delta T_{ijk} + \Delta T_{ijk+1})}{2} \right]^{\frac{1}{3}}} \right]^{\beta} \\ & + \sum_{i \in HP} \left[\frac{C_{i,CU} Q_{CU,i}}{U_{CU,i} \left[\frac{\Delta T_{CU,i} (T_{out,i} - T_{in,CU}) (\Delta T_{CU,i} + T_{out,i} - T_{in,CU})}{2} \right]^{\frac{1}{3}}} \right]^{\beta} \\ & + \sum_{j \in CP} \left[\frac{C_{j,HU} Q_{HU,j}}{U_{HU,j} \left[\frac{\Delta T_{HU,j} (T_{in,HU} - T_{out,j}) (\Delta T_{HU,j} + T_{in,HU} - T_{out,j})}{2} \right]^{\frac{1}{3}}} \right]^{\beta} \end{aligned}$$

where C_{CU} , C_{HU} are the cost coefficients of hot and cold utilities; $Q_{CU,i}$ and $Q_{HU,j}$ are the heat transfer between process streams and utilities and $Q_{ij,k}$ is the heat transfer between the process streams at stage k . CF_{ij} , $CF_{i,CU}$, $CF_{j,HU}$ are the fixed cost of heat exchangers between process streams, hot stream and cold utilities and cold stream and hot utilities, respectively, according to their sub-indices also $z_{ij,k}$, $z_{CU,i}$ and $z_{HU,i}$ are binary variables to determine the presence of a heat exchanger between the hot and cold streams at stage k , between the hot stream and the cold utility and between the cold stream and the hot utility, respectively. C_{ij} , $C_{i,CU}$ and $C_{j,HU}$ are the area-dependent cost coefficients of the heat exchangers between the process streams and between the process streams and utilities. U_{ij} , $U_{CU,i}$, $U_{HU,j}$ are the overall heat transfer coefficient in the heat exchangers between the process streams and between the process streams and utilities. ΔT_{ijk} , $\Delta T_{CU,i}$, $\Delta T_{HU,j}$ express the temperature approach in the heat exchangers between the process streams at stage k , between the hot streams in the process and the cold utility, and between the cold streams and the hot utility, respectively.

CONSTRAINTS (3.13)

$$\sum_{k \in ST} \sum_{j \in CP} Q_{ijk} + Q_{CU,i} = (T_{in,i} - T_{out,i})Fcp_i, \quad \forall i \in HP \quad (3.13a)$$

$$\sum_{k \in ST} \sum_{i \in HP} Q_{ijk} + Q_{HU,j} = (T_{out,j} - T_{in,j})Fcp_j, \quad \forall j \in CP \quad (3.13b)$$

$$\sum_{j \in CP} Q_{ijk} = (T_{i,k} - T_{i,k+1})Fcp_i, \quad \forall k \in ST, \forall i \in HP \quad (3.13c)$$

$$\sum_{i \in HP} Q_{ijk} = (T_{j,k} - T_{j,k+1})Fcp_j, \quad \forall k \in ST, \forall j \in CP \quad (3.13d)$$

$$T_{i,1} = T_{in,i}, \quad \forall i \in HP \quad (3.13e)$$

$$T_{j,NoS+1} = T_{in,j}, \quad \forall j \in CP \quad (3.13f)$$

$$T_{i,k} - T_{i,k+1} \geq 0, \quad \forall k \in ST, \forall i \in HP \quad (3.13g)$$

$$T_{j,k} - T_{j,k+1} \geq 0, \quad \forall k \in ST, \forall j \in CP \quad (3.13h)$$

$$T_{i,NoS+1} = T_{out,i}, \quad \forall i \in HP \quad (3.13i)$$

$$T_{j,1} = T_{out,j}, \quad \forall j \in CP \quad (3.13j)$$

$$Fcp_i T_{i,NoS+1} - Q_{CU,i} = Fcp_i T_{out,i}, \quad \forall i \in HP \quad (3.13k)$$

$$Fcp_j T_{j,1} + Q_{HU,j} = Fcp_j T_{out,j}, \quad \forall j \in CP \quad (3.13l)$$

$$Q_{ijk} - Q_{ij}^{max} z_{ijk} \leq 0, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.13m)$$

$$Q_{CU,i} - Q_{i,CU}^{max} z_{CU,i} \leq 0, \quad \forall i \in HP \quad (3.13n)$$

$$Q_{HU,j} - Q_{HU,j}^{max} z_{HU,j} \leq 0, \quad \forall j \in CP \quad (3.13o)$$

$$\Delta T_{ijk} - T_{i,k} + T_{j,k} + \Delta T_{ij}^{max} z_{ijk} \leq \Delta T_{ij}^{max}, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.13p)$$

$$\Delta T_{ijk+1} - T_{i,k+1} + T_{j,k+1} + \Delta T_{ij}^{max} z_{ijk} \leq \Delta T_{ij}^{max}, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.13q)$$

$$\Delta T_{CU,i} - T_{i,NoS+1} + \Delta T_{i,CU}^{max} z_{CU,i} \leq \Delta T_{i,CU}^{max} - T_{out,CU}, \quad \forall i \in HP \quad (3.13r)$$

$$\Delta T_{HU,j} + T_{j,1} + \Delta T_{HU,j}^{max} z_{HU,j} \leq \Delta T_{HU,j}^{max} + T_{out,HU}, \quad \forall j \in CP \quad (3.13s)$$

$$\left\{ \sum_{j \in CP} z_{i,j,k} \leq 1, \quad \forall i \in HP, \forall k \in ST \right\} \text{Arithmetic Mean Temperature Difference} \quad (3.13t)$$

$$\left\{ \sum_{i \in HP} z_{i,j,k} \leq 1, \quad \forall j \in CP, \forall k \in ST \right\} \text{Arithmetic Mean Temperature Difference} \quad (3.13u)$$

where $\Delta T_{ij}^{max} = \max\{0, T_{j,in} - T_{i,in}, T_{j,in} - T_{i,out}, T_{j,out} - T_{i,in}, T_{j,out} - T_{i,out}\}$,
for all $i \in HP \cup \{HU\}$ and $j \in CP \cup \{CU\}$; and $Q_{ij}^{max} = \min\{(T_{in,i} - T_{out,i})F_{cp,i}, (T_{out,j} - T_{in,j})F_{cp,j}\}$, for all $i \in HP \cup \{HU\}$ and $j \in CP \cup \{CU\}$. (3.13v)

where T_{in} is the inlet temperature of the stream and T_{out} is the outlet temperature of the stream. F_{CP} is the heat capacity of the streams. $T_{i,k}$ and $T_{j,k}$ denote the temperature of the hot and cold streams at stage k . Q^{max} means the upper limit of heat transfer and ΔT^{max} means the upper limit of temperature difference [Floudas *et al.*, (1999)].

(3.14)

VARIABLE BOUNDS

$$0 \leq Q_{ijk} \leq Q_{ij}^{max}, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.14a)$$

$$0 \leq Q_{CU,i} \leq Q_{i,CU}^{max}, \quad \forall i \in HP \quad (3.14b)$$

$$0 \leq Q_{HU,j} \leq Q_{HU,j}^{max}, \quad \forall j \in CP \quad (3.14c)$$

$$T_{i,out} \leq T_{ik} \leq T_{i,in}, \quad \forall k \in ST, \forall i \in HP \quad (3.14d)$$

$$T_{j,in} \leq T_{jk} \leq T_{j,out}, \quad \forall k \in ST, \forall j \in CP \quad (3.14e)$$

$$\Delta T_{min} \leq \Delta T_{ijk} \leq \Delta T_{ij}^{max}, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.14f)$$

$$\Delta T_{min} \leq \Delta T_{CU,i} \leq \Delta T_{i,CU}^{max}, \quad \forall i \in HP \quad (3.14g)$$

$$\Delta T_{min} \leq \Delta T_{HU,j} \leq \Delta T_{HU,j}^{max}, \quad \forall j \in CP \quad (3.14h)$$

$$z_{ijk}, z_{CU,i}, z_{HU,j} \in \{0,1\}, \quad \forall k \in ST, \forall i \in HP, \forall j \in CP \quad (3.14i)$$

3.3. Power Cycles

Waste heat has a certain thermal energy, even at low temperatures, and can be used to generate power to reduce fossil fuel consumption. Thermodynamic cycles are widely used to convert thermal energy into electricity and to recover industrial waste heat. The thermodynamic cycles used include various power cycles such as RC, ORC, KC, etc. [Pethurajan *et al.*, (2018)].

A thermodynamic cycle is the process by which a system undergoes some changes from its initial state and then returns to its initial conditions in the final state. In a cycle with water as the working fluid, an example of thermodynamic change is that when heat is added to the working fluid, it changes phase to vapor and then expands in the turbine. A system that takes heat from a high-temperature source and transfers it to a lower-temperature sink while simultaneously producing work is called a heat engine [Rackley, (2017)].

According to Kelvin-Planck expression in the second law of thermodynamics; it is impossible for a system that receives heat from a source to produce net work, it must necessarily reject some of the heat. In other words, a heat engine has to reject some of the heat it receives. Because of Kelvin-Planck expression, the efficiency of the heat engine can not be one hundred percent [Çengel and Boles, (2015)].

3.3.1. Rankine Cycle

RC is a process that efficiently converts thermal energy from an external source into mechanical work, i.e. electricity. In the traditional RC, water is used as the working fluid due to its low cost and environmental benefits. As shown in **Figure 3.11** below, a simple RC consists of four components: evaporator, condenser, pump and turbine.

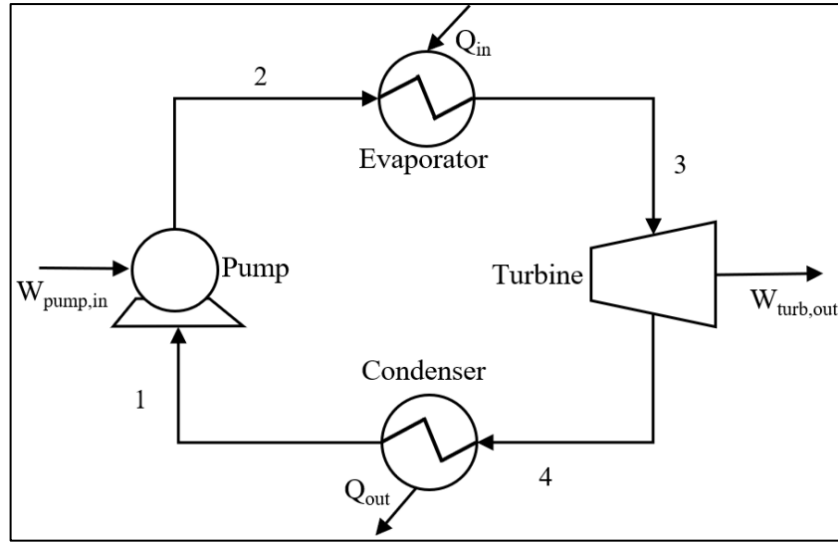


Figure 3.11: Schematic of Rankine Cycle [Wu *et al.*, (2021)].

The working fluid enters the pump at low temperature and pressure. By the pump, the temperature and pressure of the water increases. Then in the evaporator, with heat input at constant pressure, the temperature of the working fluid rises and it changes from liquid phase to vapor phase. As can be seen from the Temperature-Entropy Diagram given in **Figure 3.12** below, the working fluid is in the vapor state at high temperature and pressure at the turbine inlet. Afterwards, the temperature and pressure of the working fluid, which expands in the turbine, decreases. As can be seen from the T-S Diagram shown in **Figure 3.12**, in ideal conditions, the compression and expansion processes take place isentropically. The fluid, whose temperature and pressure drop, enters the condenser, releases some heat and returns to its initial state. Where Q_{in} and Q_{out} are the heat input/output to the cycle. $W_{pump,in}$ indicates the power consumed by the pump and $W_{turb,out}$ indicates the power generated by the turbine [Wu *et al.*, (2021)].

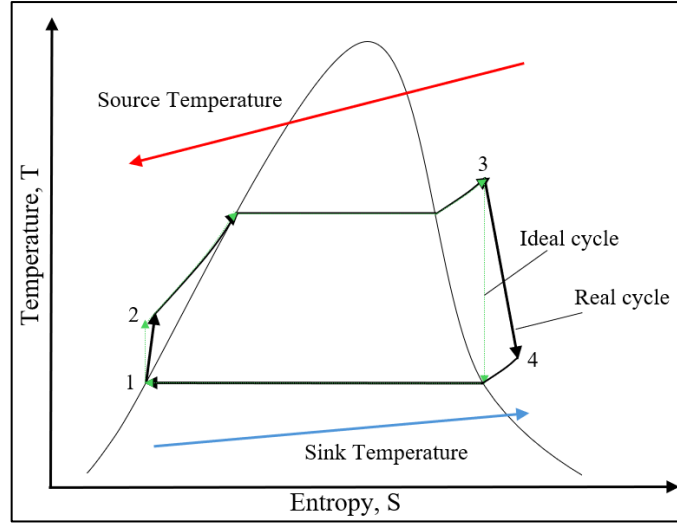


Figure 3.12: T-s diagram of Rankine Cycle [Wu *et al.*, (2021)].

The following Equations (3.15) - (3.18) are used to calculate the heat input/output and the power of the turbine and pump in the cycle. In Equation (3.19), the net power of the cycle is calculated as the generated power minus the consumed power. Equation (3.20) is used to calculate the efficiency of the cycle.

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2) \quad (3.15)$$

$$\dot{Q}_{out} = \dot{m}(h_4 - h_1) \quad (3.16)$$

$$\dot{W}_{turb,out} = \dot{m}(h_3 - h_4)\eta_{turb} \quad (3.17)$$

$$\dot{W}_{pump,in} = \frac{\dot{m}(h_2 - h_1)}{\eta_{pump}} \quad (3.18)$$

$$\dot{W}_{net} = \dot{W}_{turb,out} - \dot{W}_{pump,in} \quad (3.19)$$

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (3.20)$$

where η_{cycle} is the thermal efficiency of the power cycle. $\dot{W}_{net}$ is the net power produced also $\dot{m}$ is the mass flow rate of the working fluid, h is the enthalpy of water at that point and η is the efficiency of the turbine or pump [Victor *et al.*, (2013)].

In the studies, increasing the efficiency of the RC is related to the vaporization temperature, i.e. turbine inlet temperature and pressure. If the turbine inlet temperature increases, the energy efficiency of the cycle also increases. The conventional RC is an

accepted process for generating power from waste heat and the water used as the working fluid is a very logical application in terms of cost, reliability and availability. Condensation temperatures in RC are usually between 298-303K. The condenser pressure is also lower than atmospheric pressure, so operation and maintenance costs increase slightly when air infiltration occurs [Wu *et al.*, (2021)].

3.3.2. Organic Rankine Cycle

ORC is the most advanced of the thermodynamic cycles in recovering low-level waste heat and converting it into power. ORC is considered to be ahead of KC, RC as well as waste heat recovery technologies due to its simplicity of design and ability to recover waste heat at low temperatures. ORC and RC operating principles are the same. As shown in **Figure 3.11** above, a simple ORC consists of an evaporator, condenser, pump and turbine. The only difference from the RC is that an organic fluid is used as the working fluid instead of water. Due to the low boiling point of the organic working fluid, it is preferred to recover heat from a low-grade heat source [Ononogbo *et al.*, (2023)].

The overall thermal efficiency of ORC varies between 2-19%. The efficiency of ORC depends on different factors such as heat source, sink and operating conditions. The most important factor affecting the efficiency is the thermodynamic properties of the working fluid. Therefore, it is necessary to be careful when selecting the working fluid.

The organic fluids used in ORC have low boiling temperatures and pressures and high molecular weights. Their high molecular weight reduces the number of steps in the expansion process, affecting complexity and cost. Since ORC receives heat from a low-grade waste heat source, organic fluids with lower latent heat of vaporization have better thermal performance. The low specific volume of the organic fluid reduces the size of the expander and heat exchanger, which affects the cost. The critical temperature values of the organic fluid close to the maximum temperature of the heat source of the ORC increase the efficiency of the cycle and the freezing point should be less than the minimum temperature in the cycle [Mahmoudi *et al.*, (2018)].

Organic fluids (Hydrocarbons) have explosive and flammable properties, despite their low cost, environmental compatibility and high performance in waste heat recovery.

Therefore, they need to be thoroughly investigated to ensure compliance with environmental control and regulations [Ononogbo *et al.*, (2023)].

3.3.3. Kalina Cycle

KC is a cycle designed by **Kalina, (1982)** in the early 1980s. KC, like the ORC and RC, is a cycle used to generate electricity from waste heat and operates according to the operating principles of the RC. In addition to the components of the RC, a recuperator, separator and mixer are used as shown in **Figure 3.13** below. The working fluid is a water-ammonia mixture instead of steam in RC or organic fluid in ORC. Since KC does not use a single fluid, the temperature does not remain constant during boiling and condensation, which increases the efficiency of the cycle.

In single-fluid cycles, the working fluid is uniformly heated to the vaporization temperature at which superheated steam begins to form. As KC has two different mixtures of working fluids, the temperature of each fluid is raised separately during vaporization because water and ammonia have different boiling points [Jouhara *et al.*, (2018)].

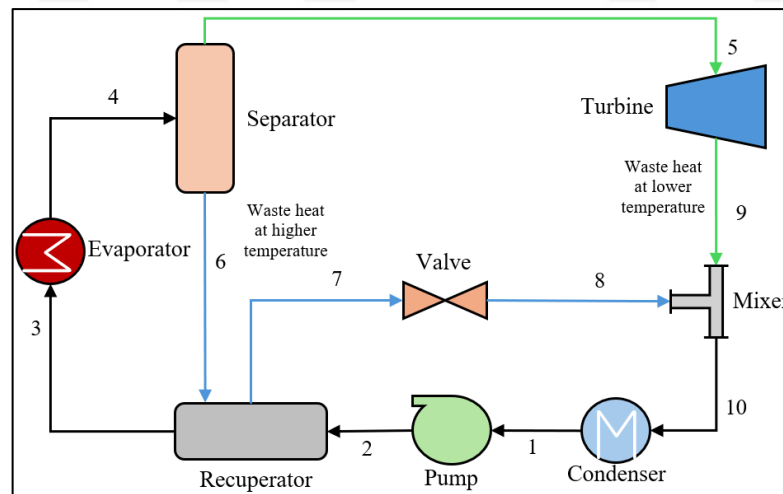


Figure 3.13: Configuration of Kalina Cycle [Wang *et al.*, (2024)].

The temperature and pressure of the ammonia and water mixture is increased with the help of a pump (1-2). Then the temperature of the mixture is increased a little more in the recuperator (2-3). The temperature of the mixture is increased to the maximum amount by taking waste heat from the evaporator (3-4). Then, in the separator, the fluid is divided into ammonia rich and ammonia poor (4-5,4-6). The temperature of the

ammonia-poor mixture is still high as can be seen from the T-s diagram shown in **Figure 3.14** below and it transfers its heat to the mixture with lower temperature at the pump outlet (6-7). The ammonia-rich fluid goes to the turbine where it expands and its temperature and pressure are reduced (5-9). With the help of the mixer, the fluid whose flow rates are combined again enters the condenser (8,9-10). In the condenser, with the help of cooling water, the fluid, which completely passes into the liquid phase, completes the cycle (10-1) [Wang *et al.*, (2024)].

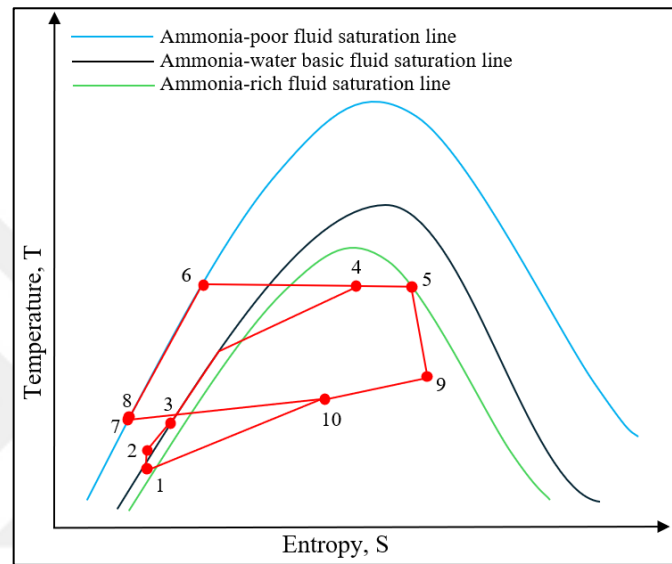


Figure 3.14: T-s diagram of Kalina Cycle [Wang *et al.*, (2024)].

When selecting the working fluid, it is desirable to have high thermal conductivity and enthalpy of vaporization and low condensation pressure. Although the boiling point of water is high, since ammonia can be dissolved homogeneously in water, the boiling point of the mixture decreases if the amount of ammonia in the water increases. The enthalpy of vaporization of ammonia is lower than water, but the heat absorption ability of the mixture is higher than ammonia. The fact that the ammonia-water mixture has varying bubbles and dew point causes it to behave like different fluids in the cycle, and the high condensation pressure of the mixture is one of the disadvantages of the ammonia-water mixture.

The advantages of ammonia water mixture can be summarized as follows;

- Because ammonia is a fluid produced from nature, it is less threatening to the environment than other organic fluids, and it is easily accessible, cheap and reduces acidic pollution in the atmosphere.

- Its enthalpy of vaporization is higher than other organic fluids.
- Since the freezing temperature of water in atmospheric conditions is 0°C and that of pure ammonia is -78°C, the freezing temperature of the ammonia water mixture is very low and its boiling at low temperatures allows the utilization of low level waste heat.
- Since the ammonia water mixture has a non-constant boiling temperature, it ensures that heat transfer losses are low.
- A conventional steam turbine can be used for ammonia because the vapor behavior of ammonia is almost similar to that of water vapor due to the close molar weights of ammonia and water (17.03 g/Mol, 18.015 g/Mol). [Köse *et al.*, (2022)]

3.3.4. Supercritical-CO₂ Cycle

s-CO₂C is a power cycle operating according to RC principles. Carbon dioxide (CO₂), which is used as the working fluid in the s-CO₂C, has the desired characteristics of low cost, medium critical point value, low environmental impact and abundant availability in nature. The disadvantages of the cycle are that the condensation process can take place at low temperatures due to the low critical temperature of CO₂ and the cycle operates at high pressures so safety precautions must be taken.

The heat addition process of the s-CO₂C does not consist of a two-phase region as in a conventional RC or ORC. The CO₂ goes directly from the liquid phase to the vapor phase by exceeding the critical point and is heated to supercritical states before the expansion process. This results in a better thermal coupling in the evaporator [Chen *et al.*, (2010)].

s-CO₂C with regenerator and T-s diagram is shown in **Figure 3.15** below. Since carbon dioxide has an intermediate critical pressure (73.8 bar) and low critical temperature (31.1°C), as seen in the T-s diagram in **Figure 3.15** below part of the cycle occurs in supercritical regions, meaning that the s-CO₂C can be called a transcritical cycle [Chen *et al.*, (2006)].

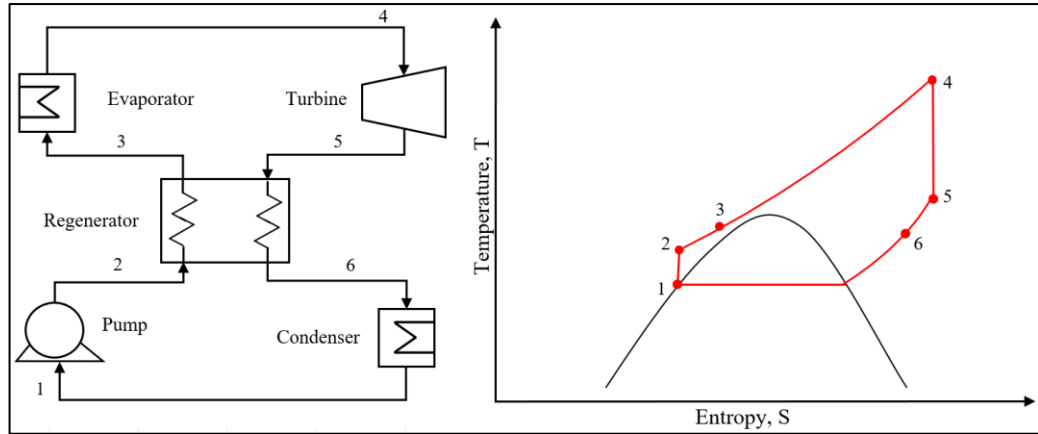


Figure 3.15: Schematic of s-CO₂C and T-s diagram [Chen *et al.*, (2006)].

The “temperature shift” in the evaporator of s-CO₂C allows a better matching of the working fluid than in other cycles where the working fluid evaporates below the critical point. In this case, the pinch problem in a countercurrent heat exchanger can be avoided with s-CO₂C. This is represented in **Figure 3.16** below [Chen *et al.*, (2006)].

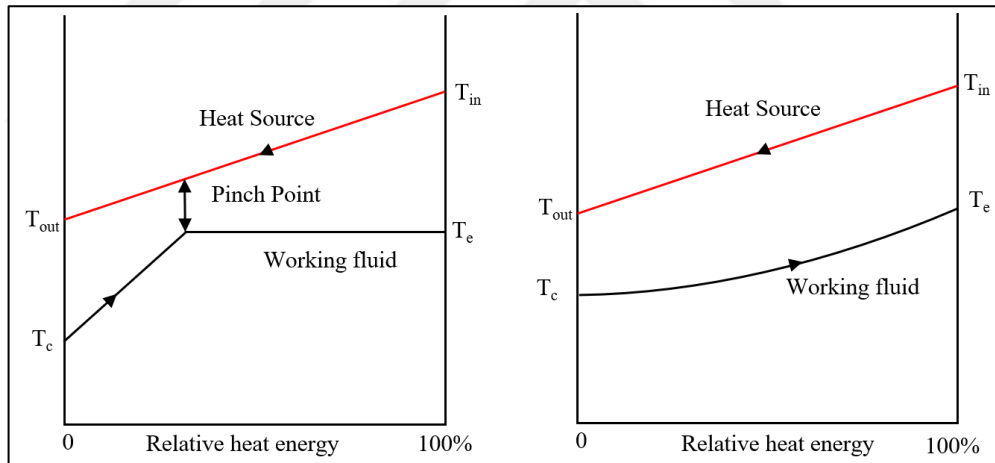


Figure 3.16: Heat transfer in the evaporator ORC and s-CO₂C [Chen *et al.*, (2006)].

3.3.5. Trilateral Flash Cycle

TFC works with the same working principle and the same equipment as RC. The only difference with RC is in the heating process. In TFC, it is sufficient to use only liquid heater. The heating process consists of a single phase. In RC, since the heating process consists of two phases, it is the vaporizer on the liquid heater. As shown in **Figure 3.17** below, the start of expansion in TFC starts from the saturated liquid phase, not

from the saturated vapor phase. 3 is the turbine inlet state in RC, 3* is the turbine inlet state in TFC. Cases 1 and 2 have the same characteristics as RC. 4* represents the condenser inlet state of TFC. In the turbine of TFC, two-phase expansion is performed.

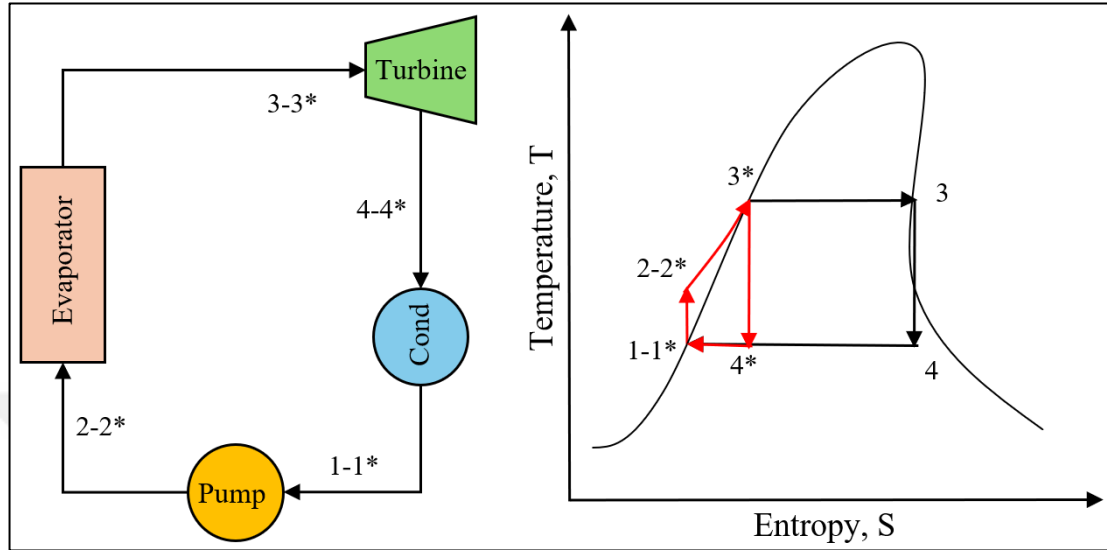


Figure 3.17: Schematic diagram of TFC and T-s diagram [Imre and Ahmed, (2023)].

In TFC, just like ORC, many organic working fluids with high molar weight and low boiling point temperature can be used to replace water. In the selection of the working fluid, the critical temperature values, specific heat, mole weights and other properties of the fluid play an important role in TFC as in ORC. When a dry fluid is used as the working fluid, the expansion in the ORC may stop in the dry vapor state due to overheating and the same happens in the TFC. Therefore, a suitable fluid should be selected and at the same time some waste heat should be removed before the condenser [Imre and Ahmed, (2023)].

The turbine used in TFC is more expensive than the one used in ORC, which makes ORC more preferred. ORC requires the use of a heat exchanger with more area than in TFC due to the large amount of waste heat when using super dry working fluid. However, the heat exchanger costs less than the turbine and ORC is cheaper than TFC for dry working fluids [Ahmed, (2024)].

4. METHOD

It is proposed to optimize a heat recovery network with hot and cold streams with constant inlet and outlet temperatures to achieve maximum energy recovery and minimum energy requirement, and to integrate a power cycle into this heat recovery network to obtain power. The streams that enter the heat recovery network at low temperature and need to be heated are called cold streams and the streams that enter at high temperature and need to be cooled are called hot streams.

In the proposed method, HEN is first analyzed manually by pinch solution. The minimum amount of utility consumption is calculated. Then TAC of HEN is calculated according to the given economic parameters. After the pinch method, HEN is solved as MINLP with SYNHEAT model by including the economic parameters in the optimization. In SYNHEAT solution, more economical results can be obtained compared to pinch solution or the same results can be obtained.

After the cost of HEN is minimized with the MINLP model solution, power integration is performed. In the MINLP model solution, the superstructure is divided into stages. The first two stages include the heat exchangers between the process streams. The amount of heat that needs to be cooled from the hot streams is considered as waste heat. All streams using cold utility are integrated into the power cycle. In the proposed method, another stage is added next to the last stage and waste heat is sent to the power cycle. This is called the power stage. The proposed new superstructure model is shown in **Figure 4.1** below.

According to the temperature constraints, the maximum and minimum temperatures of the power cycle are determined from the optimization of HEN when $\Delta T_{\min}$ is included. In the optimization within the power cycle, temperatures are not decision variables but fixed parameters.

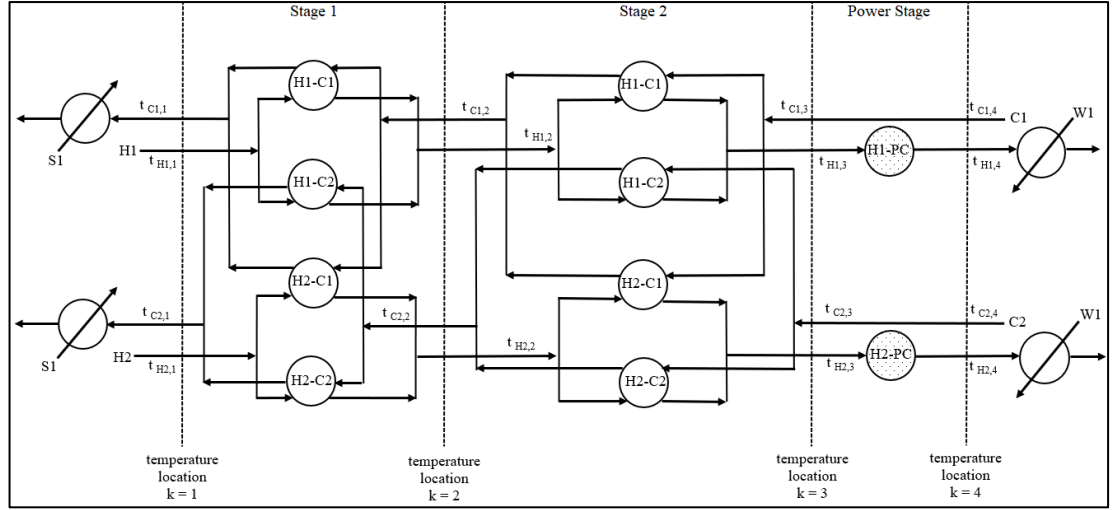


Figure 4.1: Proposed new superstructure model.

The optimization of the HEN without a power cycle is completed once the TAC is minimized while satisfying the constraints in the optimization using Pinch Analysis or the SYNHEAT model with GAMS. After minimizing the cost of the HEN, the potential locations for the power cycles are identified. In the HEN, the heat exchanger used for cold utilities is removed, and the available heat is transferred to the power cycle. The inlet temperatures for the turbine and pump in the cycle are determined by the HEN. The simulation of the power cycles is carried out using UNISIM DESIGN. The results from UNISIM DESIGN simulations can be integrated with Python for optimizing the power cycle. Once the potential locations for the power cycles are determined, the next step involves developing a thermodynamic model and optimizing it iteratively.

A thermodynamic model is developed in UNISIM DESIGN based on the temperature constraints in the HEN. The simulation results are then optimized iteratively in Python by performing a parametric study on various parameters that affect the power output and indirectly the costs, such as pressures, mass flow rates, fluid vapor quality, and heat rejection due to temperature constraints. If the constraints are satisfied during the iterations in Python, the heat recovery network integrated with the power cycle is designed. If the constraints are violated, the iteration process is repeated. The flow diagram of the optimization strategy for the overall system, integrating UNISIM and Python, is presented in **Figure 4.2** below.

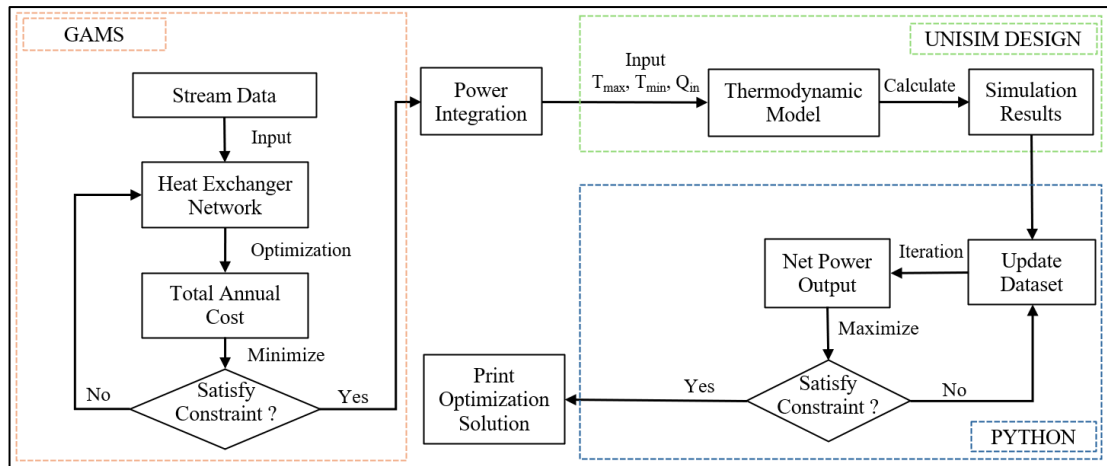


Figure 4.2: Flow diagram of the optimization strategy.

Most of the studies in the literature are optimized simultaneously. Thermodynamic data such as the enthalpy of working fluid at the cycle is sent into GAMS using REFPROP. Thus, the power cycle integrated with HEN is optimized simultaneously. The difference of this thesis from the literature is that the thermodynamic properties are calculated in UNISIM DESIGN. After HEN is optimized using GAMS, UNISIM DESIGN is integrated with Python and HEN integrated with the power cycle is optimized sequentially.

5. RESULT AND DISCUSSION

In this section, two case studies from the literature are examined under subheadings 5.1 and 5.2. The results obtained using the proposed method, along with a comparison with the literature, are presented. The strengths and weaknesses of the method, compared to those in the literature are discussed.

5.1. Case Study 1

The stream data for Case Study 1 **Beck and Hofmann, (2018)** is shown in **Table 5.1** below. In a system with two hot and two cold streams (H1, H2, C1, C2) as well as steam and cooling water (HU, CU) used as hot and cold utilities, the inlet and outlet temperatures, heat capacities and individual film heat transfer coefficients of the streams have been determined. $\Delta T_{\min}$ is set to 10°C as used in the reference.

Table 5.1: Case study 1 stream data [**Beck and Hofmann, (2018)**].

Stream	T_{in} (°C)	T_{out} (°C)	F_{CP} (kW/K)	h (kW/m ² K)
H1	270	160	18	0.5
H2	220	60	22	0.5
C1	50	210	20	0.5
C2	160	210	50	0.5
HU	250	250		1
CU	15	20		1

When the first law analysis is conducted using Equation (3.1), H1 can supply 1980 kW, H2 can supply 3520 kW, C1 can absorb 3200 kW and C2 can absorb 2500 kW. According to the first law, there is a 200 kW heat deficit for the cold streams to reach to the targeted temperature values by absorbing heat from the hot streams. This heat deficit is eliminated by using utility. In **Figure 5.1** shown below, the streams are plotted according to the temperatures in Temperature Interval Diagram and Cascade Diagram.

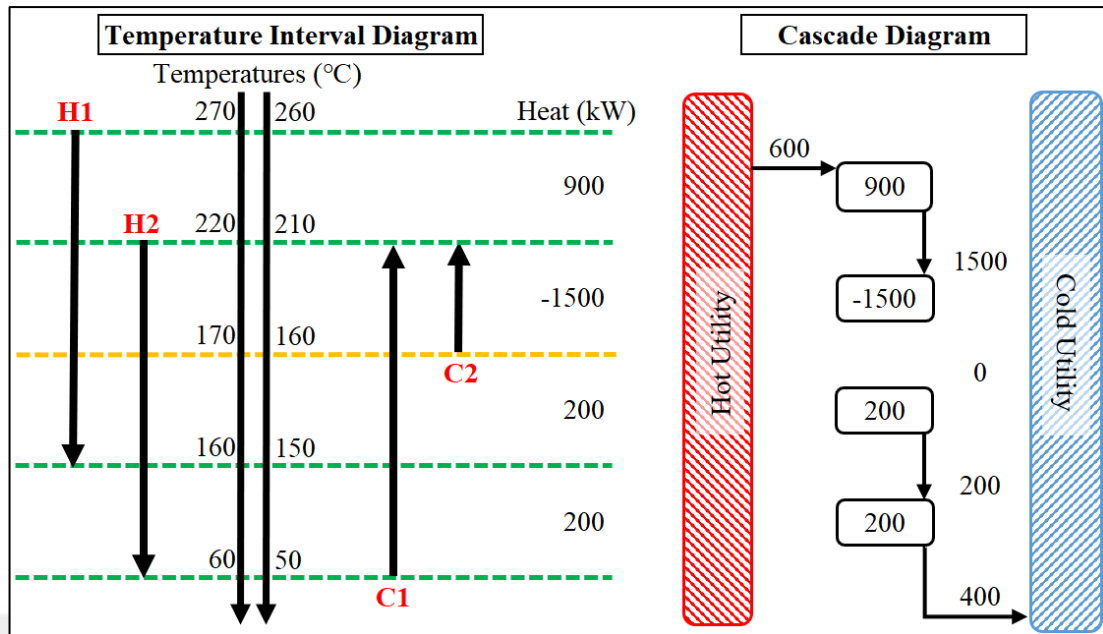


Figure 5.1: Temperature interval and cascade diagram for case study 1.

In the first interval there is a 900 kW heat surplus; in the second interval, there is a 1500 kW heat deficit; and in the third and fourth intervals, there is a 200 kW heat surplus. According to the second law, heat flow naturally occurs from high to low temperatures. Heat flow is also represented in the Cascade Diagram. A minimum of 600 kW hot utility is required to ensure heat flow. The heat flow stops at the end of the second interval. The temperature range between 170°C and 160°C corresponds to the pinch temperature. Below the pinch point, excess heat is transferred to lower temperatures, and 400 kW of cold utility are required.

Firstly, Composite Curves of the hot and cold streams are drawn, as shown in **Figure 5.2** below. These Composite Curves represent the heat loads that the streams will receive or release according to their temperatures. Subsequently, GCC derived from the Composite Curves is illustrated in **Figure 5.3**. Interpreting GCC is crucial for the pinch method, as it reveals the pinch temperature, the amount of utilities required, and the potential locations for a power cycle or heat pump. As can be understood from **Figure 5.3**, the pinch point is the location where the heat flow stops, specifically the point that touches the y-axis. The distance between the starting point of the curve and the y-axis indicates the hot utility, while the distance between the endpoint and the y-axis represents the cold utility. The 400 kW heat surplus below the pinch point, addressed by the cold utility, denotes the possible location for a power cycle.

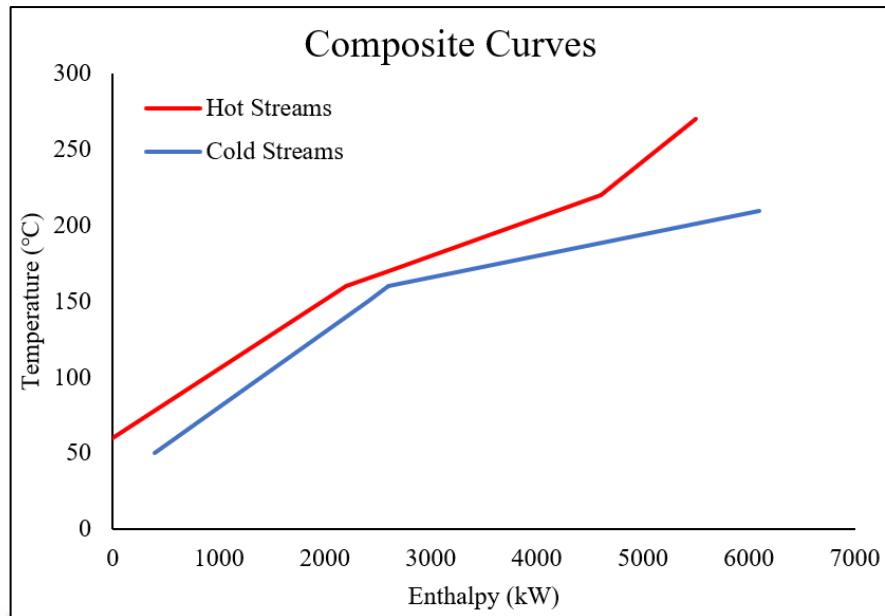


Figure 5.2: Composite curves for case study 1.

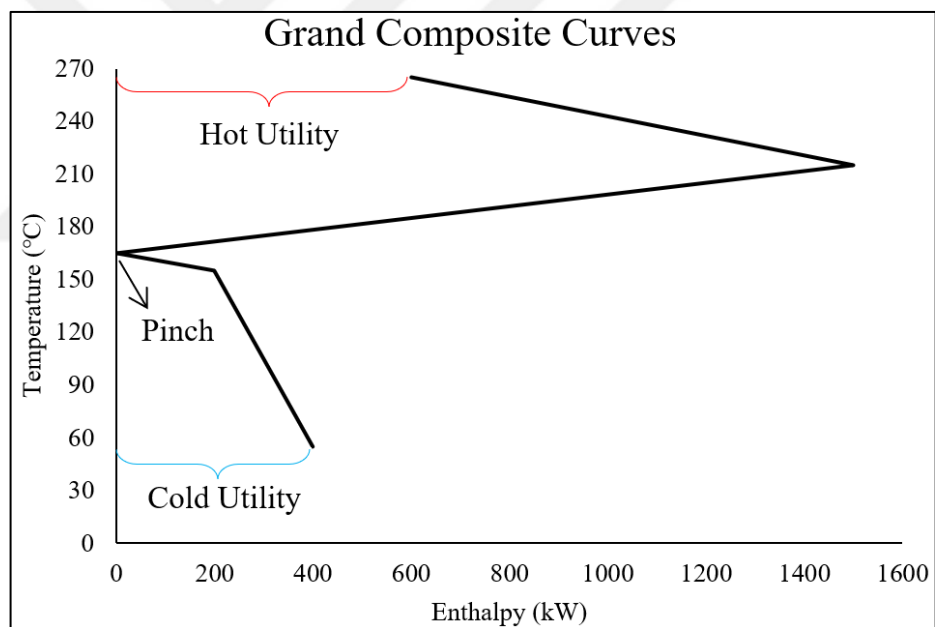


Figure 5.3: Grand composite curves for case study 1.

According to the pinch rules described in Chapter 3, Grid Diagram shown in **Figure 5.4** below is obtained. A total of seven heat exchangers are used: four between the process streams and three for the utility. This design refers to maximum energy recovery (MER) and minimum energy requirement (MER). A heat transfer of 5100 kW occurs between the process streams. A 400 kW cold utility and a 600 kW hot utility are required.

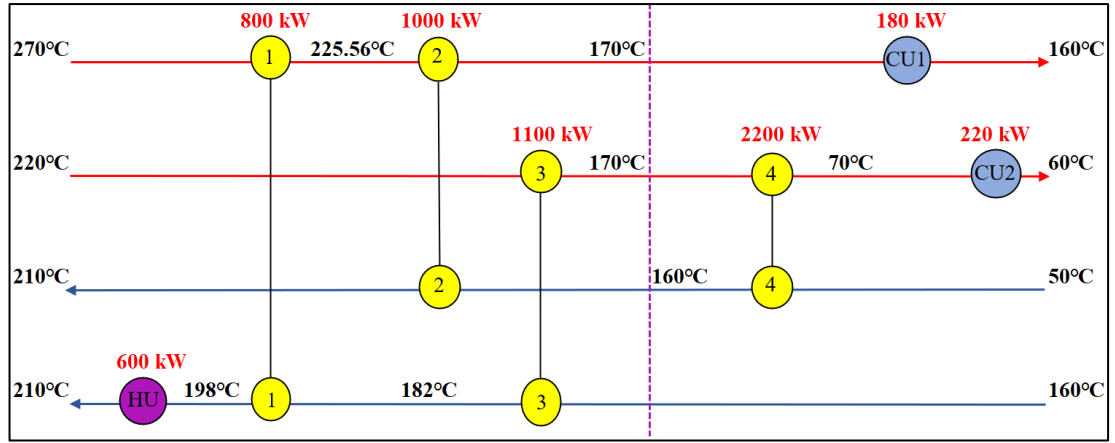


Figure 5.4: Pinch solution for Case Study 1.

Using the individual heat transfer coefficients of the streams, the overall heat transfer coefficient for each heat exchanger is calculated from Equation (3.7). ΔT_{LM} is then calculated from Equation (3.9), and the areas of each heat exchanger are determined from Equation (3.6). The areas of the heat exchangers are shown in **Table 5.2** below.

Table 5.2: Case study 1 heat exchanger areas for pinch solution.

Heat Exchanger	Area (m ²)
1	56.55
2	318.14
3	210.60
4	610.16
HU	39.36
CU1	3.66
CU2	13.91
Total Area = 1252.37 m ²	

The Pinch Solution shows the required minimum utility. However, it does not account for the fixed installation costs, and area-dependent costs of heat exchangers, and the cost of utilities. Investment and operating cost coefficients are shown in **Table 5.3** below [Beck and Hofmann, (2018)] [Dong *et al.*, (2020)] [Wang *et al.*, (2024)] [Watanapanich *et al.*, (2022)].

When the investment costs of heat exchangers and operating costs for utilities are calculated, Annual Investment Cost is € 263360.8, Annual Operating Cost is € 128000, and Total Annual Cost, which represents the sum of operating and investment costs in the pinch solution, is € 391360.8.

Table 5.3: Economic parameters for case study 1 [Beck and Hofmann, (2018)], [Dong *et al.*, (2020)], [Wang *et al.*, (2024)], [Watanapanich *et al.*, (2022)].

Investment Cost Economic Parameters		
Heat Exchanger	$4000 + 500 [\text{Area (m}^2\text{)}]^{0.83}$	€ (y) ⁻¹
Pump	$[\text{MF}/\text{PB}_{\text{year}}] [3540 (W_{\text{pump}})^{0.71}]$	€ (y) ⁻¹
Turbine	$[\text{MF}/\text{PB}_{\text{year}}] [6000 (W_{\text{tur}})^{0.7} + 60 (W_{\text{tur}})^{0.95}]$	€ (y) ⁻¹
Separator (for KC)	$[\text{MF}/\text{PB}_{\text{year}}] [280.3 (\dot{m}_{\text{in}})^{0.67}]$	€ (y) ⁻¹
Valve (for KC)	$[\text{MF}/\text{PB}_{\text{year}}] [114.5 \dot{m}]$	€ (y) ⁻¹
Mixer (for KC)	$[\text{MF}/\text{PB}_{\text{year}}] [100]$	€ (y) ⁻¹
Operating Cost Economic Parameters		
Hot Utility	200	€ (y) ⁻¹ (kW) ⁻¹
Cold Utility	20	€ (y) ⁻¹ (kW) ⁻¹
H _y	7000	hr (y) ⁻¹
C _{el}	0.14	€ (kWh) ⁻¹

The area shown in **Table 5.3** represents the area of each heat exchanger. W_{pump} is the power consumed by the pump, and W_{tur} is the power generated by the turbine. The separator, valve and mixer are used only for the KC, and $\dot{m}$ is the mass flow rate of the working fluid. Multiplication factor (MF) accounts for unexpected losses in investment costs, installation, and maintenance costs, and its value is 1.5. PB_{year} refers to the investment plan of the plant, representing the payback period or the number of years over which the plant is built, and its value is taken as 15. H_y refers to the annual operating hours of the plant. C_{el} represents the electricity sales price per unit kWh.

In the SYNHEAT model solution derived by Grossmann, HEN is solved in GAMS as a MINLP. In addition to the stream data, when economic parameters are determined and included in the calculation, it is possible to obtain a more economical HEN compared to the pinch solution. This is because the SYNHEAT model solution performs triple optimization for the number and area of heat exchangers, as well as the utility used. The MINLP model solution is shown in **Figure 5.5** below.

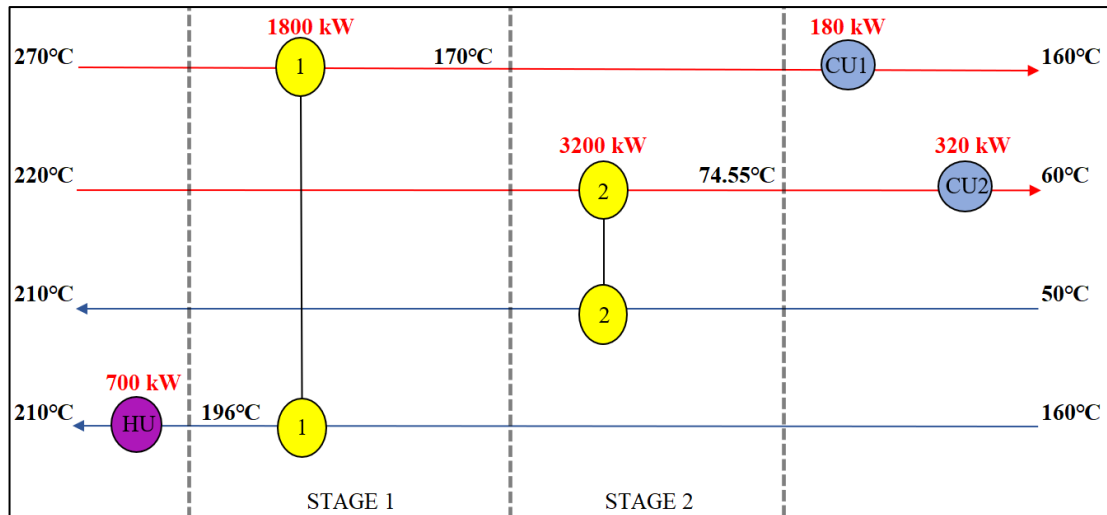


Figure 5.5: SYNHEAT solution for case study 1.

Compared to the Pinch Solution, the number of heat exchangers used decreases from seven to five in the SYNHEAT model, and the loops from the Pinch Solution are broken. The number of heat exchangers is reduced. At the same time, since the area of the heat exchangers is reduced, the costs caused by the area are reduced. The areas of the heat exchangers in the SYNHEAT model solution are shown in **Table 5.4** below. The total area of heat exchangers obtained from the Pinch Solution decreases from 1252.37 m² to 1087.89 m² in the MINLP model solution. However, the amount of utility used increases from 400 kW to 500 kW for cold utility, and from 600 kW to 700 kW for hot utility. Despite the increase in the amount of utility, both the heat exchanger area and the number of heat exchangers decrease.

Table 5.4: Case study 1 heat exchanger areas for SYNHEAT solution.

Heat Exchanger	Area (m ²)
1	229
2	790.86
HU	45.02
CU1	3.66
CU2	19.35
Total Area = 1087.89 m ²	

If cost calculations are evaluated, Annual Investment Cost is € 211731.02, Annual Operating Cost is € 150000, and Total Annual Cost is calculated as € 361731.02. Annual operating costs are obtained by multiplying the quantity of utility consumed by the annual cost per kW. The annual investment cost is calculated according to the

investment cost coefficients of heat exchangers according to Equation (3.10). TAC is expressed as the sum of annual operating and annual investment costs. Compared to the Pinch Solution, the SYNHEAT model solution shows a slight increase in the operating cost due to the increase in utility consumption, while the investment cost decreases. In the total cost calculation, the HEN cost is minimized with a decrease of € 29629.78 compared to the Pinch Solution.

For the purpose of this study, sequential optimization is performed for the integration of heat exchanger network and power cycle to be designed with minimum cost. Firstly, the HEN cost is minimized with SYNHEAT model. In the cost-minimized HEN, some waste heat remains in the hot streams that use cold utility. The waste heat is planned to be sent to the power cycle to generate power. As shown in **Figure 5.5** above, there is 500 kW of waste heat. The performances of five different power cycles, namely ORC, RC, s-CO₂C, TFC and KC, are compared, and the most cost-optimal one is evaluated. One of the points that differentiates this study from the literature is that these five cycles are integrated with a heat exchanger network and compared both in terms of power generated and cost. The integration of the five different power cycles ORC, RC, s-CO₂C, TFC and KC with HEN is shown in **Figures 5.6-10** below, respectively.

There are many different organic working fluids used in ORC; however, some fluids frequently studied in the literature show good economic results. Among these working fluids, methanol shows the best results for this case study. Therefore, methanol is used as the working fluid in both ORC and TFC. Water is used as working fluid in RC, CO₂ in s-CO₂C, and ammonia water mixture in KC at 75-25% concentration. The heat transfer coefficients of the organic fluid used in ORC and TFC are shown in **Table 5.5** below. [Dong *et al.*, (2020)]

Table 5.5: Heat transfer coefficients for ORC and TFC [Dong *et al.*, (2020)].

Heat transfer coefficients for ORC/TFC	h (kW/m ² K)
Liquid phase	1
Evaporating	3.6
Superheated vapor	0.06
Condensing	1.6

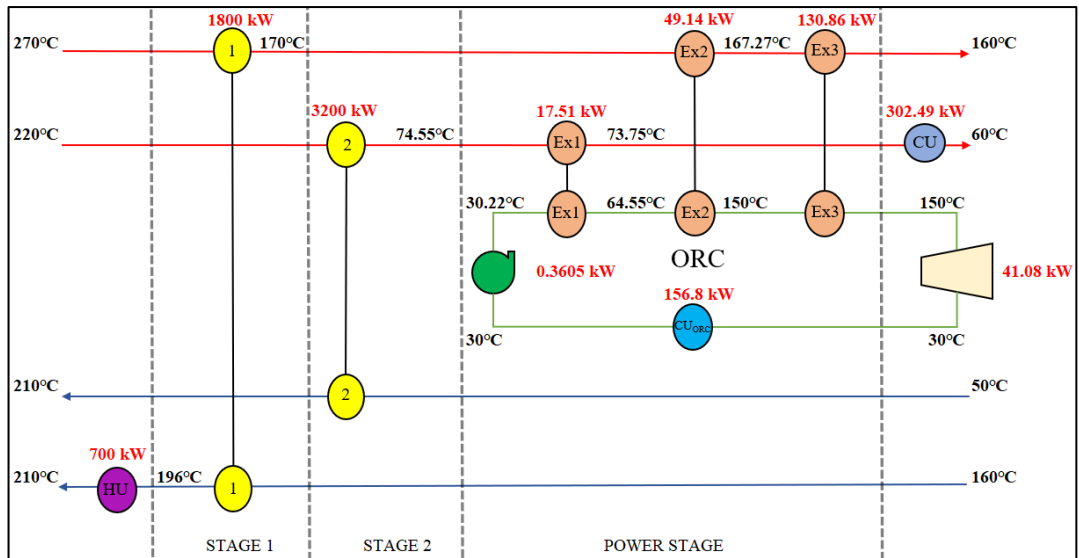


Figure 5.6: ORC and HEN integration for case study 1.

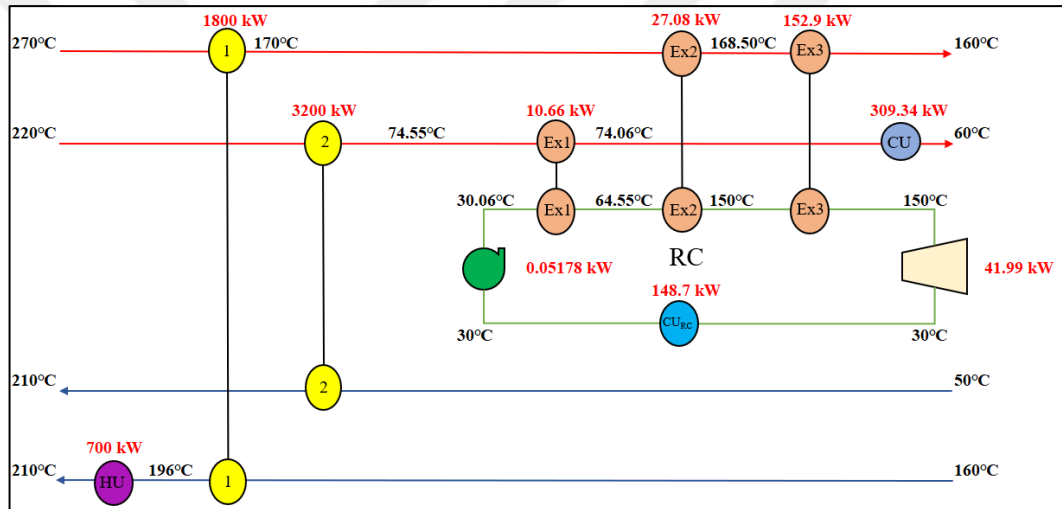


Figure 5.7: RC and HEN integration for case study 1.

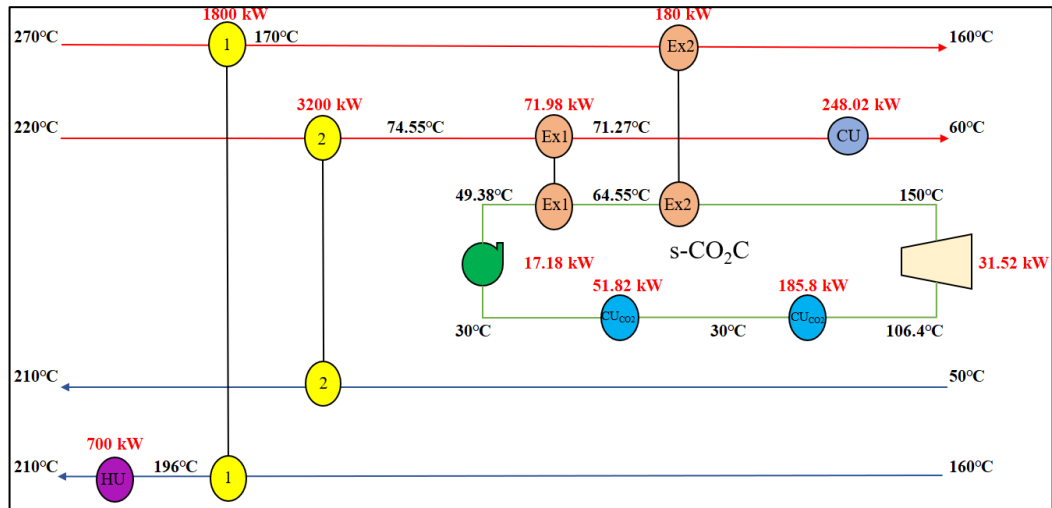


Figure 5.8: s-CO₂C and HEN integration for case study 1.

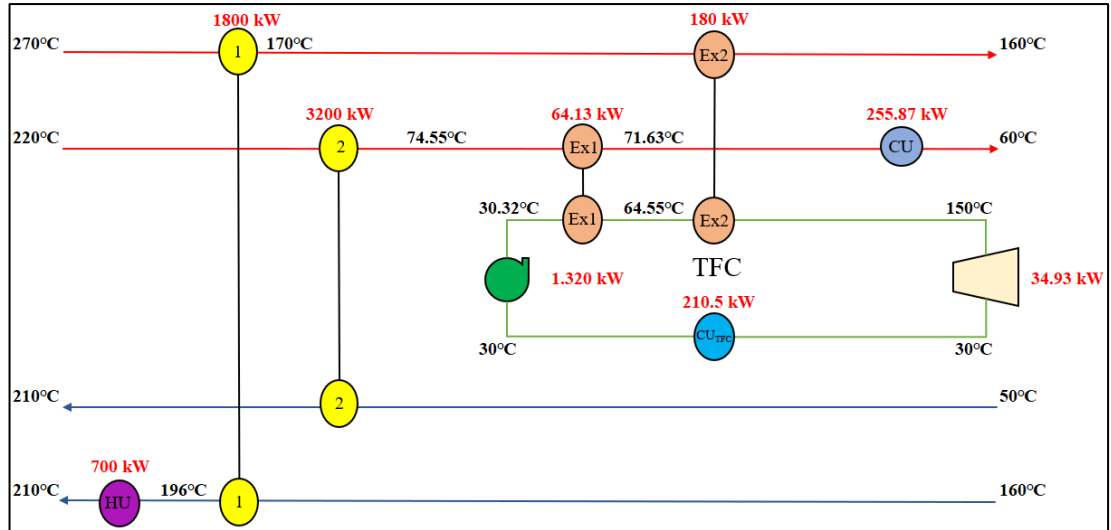


Figure 5.9: TFC and HEN integration for case study 1.

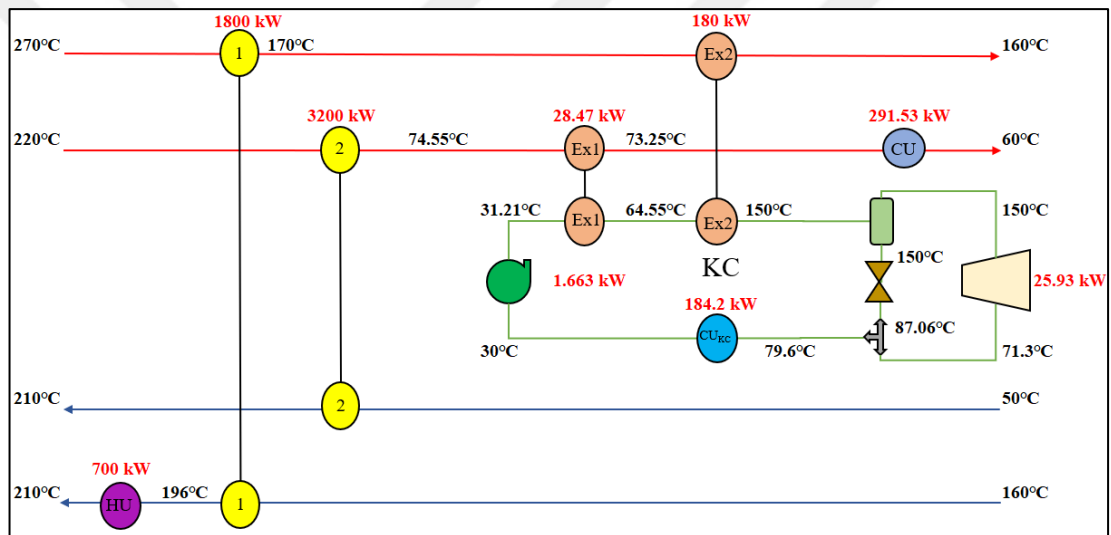


Figure 5.10: KC and HEN integration for case study 1.

The outlet temperatures of the hot streams using cold utility are 160°C and 60°C, as shown in **Figure 5.5** above. The stream with the higher outlet temperature is used in the evaporator of the power cycle, while the other hot stream is used to heat the working fluid in the power cycle. Taking ΔT_{min} into account, the maximum temperature that the cycle can reach is 150°C. Similarly, since the outlet temperature of the cooling water is 20°C, the inlet temperature of the pump in the power cycle becomes 30°C.

All of the 180 kW of heat that the H1 hot stream can supply is transferred to the power cycle. The amount of heat that the H2 stream can transfer is sufficient to raise the temperature in the cycle to 64.55°C, adhering to the temperature constraint. The heat

that can be transferred from HEN to the power cycle varies depending on the power cycle. The amount of heat the working fluid in the cycle can absorb in the liquid or vaporization phase can be calculated by analyzing the mass flow rates in Python. In s-CO₂C, the minimum total cost target is achieved by performing parametric studies in Python based on the pressures, and in KC, it is achieved based on the vapor quality of the working fluid.

The heat used in the cold utility is transferred to the power cycle. Optimization within the power cycle is performed by integrating Python with UNISIM. After defining the power cycle constraints, such as maximum and minimum temperatures and heat input, parametric studies are conducted in Python using other variables, such as pressures, mass flow rates, fluid quality, and intermediate temperatures. The results obtained from UNISIM are then evaluated in Python. As a result of the optimization, the net powers produced, total heat exchanger areas, TAC of all cycles are shown comparatively in **Table 5.6** below.

Table 5.6: TAC results for all cycles for case study 1.

	ORC	RC	s-CO ₂ C	TFC	KC
Total HE Area [m ²]	1131.64	1144.01	1123.47	1128.56	1109.28
Net Power Output [kW]	40.72	41.94	14.34	33.61	24.27
Investment Cost [€ (y) ⁻¹]	245499.57	248161.80	245163.53	242073.49	233441.19
Operating Cost [€ (y) ⁻¹]	149185.80	149160.80	149712.80	149327.40	149514.60
Electricity Revenue [€ (y) ⁻¹]	39905.11	41099.46	14053.20	32937.80	23781.66
Total Annual Cost [€ (y) ⁻¹]	354780.26	356223.14	380823.13	358463.09	359174.13

As seen in **Table 5.6**, ORC shows the most economic result in terms of TAC. Annual electricity revenue is calculated based on the net power generated by the power cycle, annual operating hours, and the unit cost per kWh. In terms of power generation, although RC is the cycle that produces the most power and hence provides electricity revenue, its annual cost is higher than ORC. In RC, heat exchanger areas increase due

to the different thermophysical properties of water used as the working fluid compared to organic fluids, which leads to higher investment costs. TFC achieves good results after ORC in terms of cost. However, since the working fluid enters the turbine in the saturated liquid phase in TFC, the turbine used is different from other cycles. This increases the turbine cost in TFC. Although there is no clear cost coefficient for the turbine used, the coefficients in the turbine cost calculation for TFC are estimated by increasing by 30% compared to other turbines, based on literature values. Since the pump inlet temperature is 30°C in s-CO₂C, the condenser pressure increases according to the critical temperature values of the fluid. This results in a reduction of the power generated. In KC, good results are obtained at low temperatures, but it is not as popular as ORC.

The impact on the total cost of working fluids used in ORC, which are frequently tested in the literature, is analyzed. Methanol, n-pentane, n-hexane, benzene, toluene, and R245fa are tested. The cost impact of different working fluids used in ORC is shown in **Figure 5.11** below.

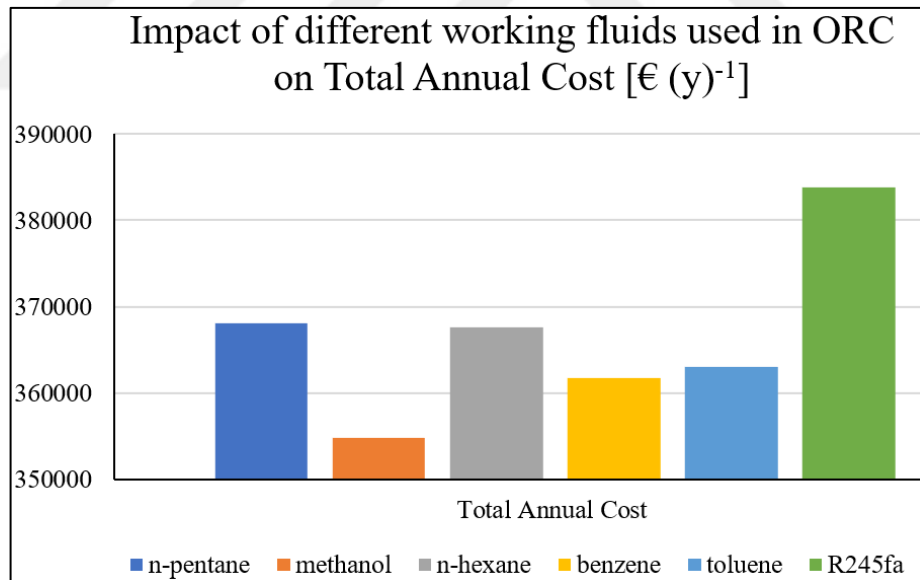


Figure 5.11: Impact of different working fluids used in ORC on TAC.

TAC of HEN without integrating the power cycle is € 361731.02. After the integration of the power cycles, the profits obtained by power cycle are shown in **Figure 5.12** below. The power cycle that provides the most profit is ORC. In s-CO₂C, there is a loss, not a profit.

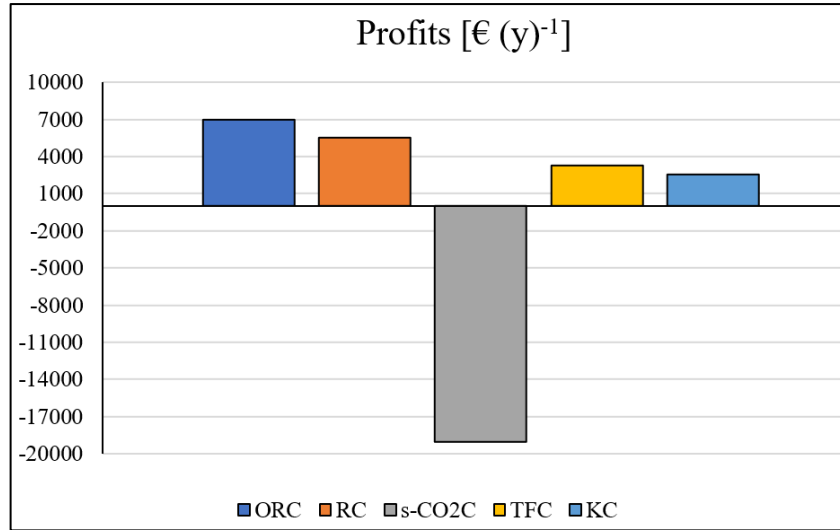


Figure 5.12: Profit rates according to cycles of case study 1.

5.2. Case Study 2

The stream data for Case Study 2 **Watanapanich *et al.*, (2022)** is given in **Table 5.7** below. As in Case Study 1, a plant with two hot and two cold streams is optimized, and $\Delta T_{\min}$ is set to 10°C. The first law analysis shows that hot streams can transfer a total of 83000 kW, while cold streams can absorb 56000 kW of heat. In this case, there is an excess heat of 27000 kW.

Table 5.7: Case study 2 stream data [**Watanapanich *et al.*, (2022)**].

Stream	T_{in} (°C)	T_{out} (°C)	F_{CP} (kW/K)	h (kW/m ² K)
H1	187	77	300	1
H2	127	27	500	1
C1	147	217	600	1
C2	47	117	200	1
HU	300	250		5
CU	15	30		1

The same steps as in Case Study 1 are applied in this case. Temperature Interval and Cascade Diagrams of the streams are shown in **Figure 5.13** below. Pinch temperature is between 147°C and 157°C. Hot utility is 33000 kW, and cold utility is 60000 kW.

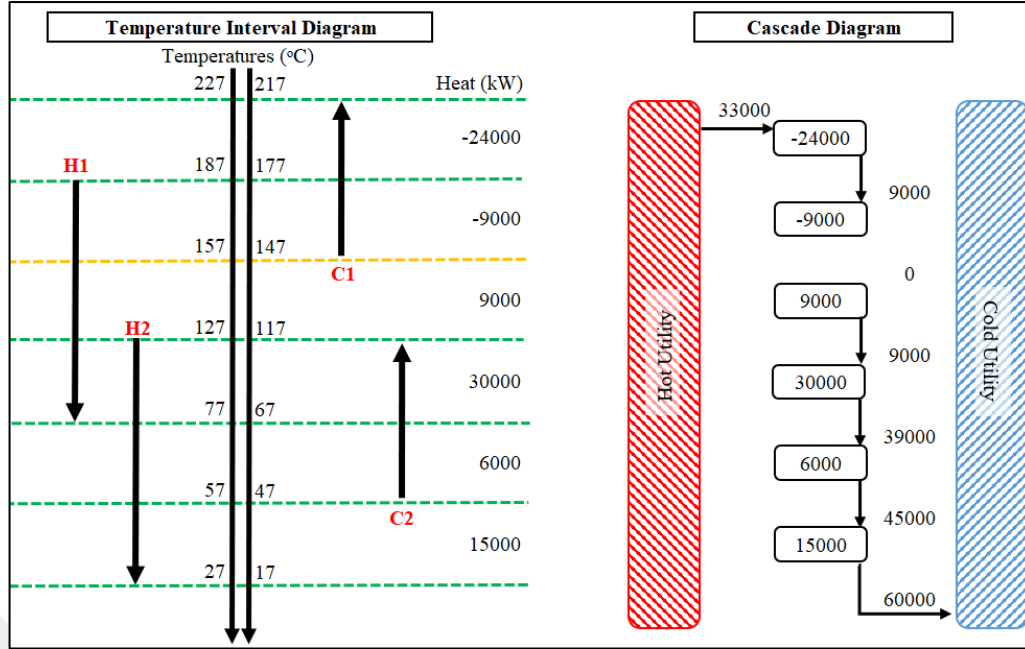


Figure 5.13: Temperature interval and cascade diagram for case study 2.

Investment and operating cost economic parameters for Case Study 2 are shown in Table 5.8 below [Watanapanich *et al.*, (2022)], [Dong *et al.*, (2020)], [Wang *et al.*, (2024)].

Table 5.8: Economic parameters for case study 2 [Watanapanich *et al.*, (2022)], [Dong *et al.*, (2020)], [Wang *et al.*, (2024)].

Investment Cost Economic Parameters		
Heat Exchanger	[MF/PB _{year}] [16620 + 575 [Area (m ²)]]	\$ (y) ⁻¹
Pump	[MF/PB _{year}] [3540 (W _{pump}) ^{0.71}]	\$ (y) ⁻¹
Turbine	[MF/PB _{year}] [6000 (W _{tur}) ^{0.7} + 60 (W _{tur}) ^{0.95}]	\$ (y) ⁻¹
Separator (for KC)	[MF/PB _{year}] [280.3 ($\dot{m}_{in}$) ^{0.67}]	\$ (y) ⁻¹
Valve (for KC)	[MF/PB _{year}] [114.5 $\dot{m}$]	\$ (y) ⁻¹
Mixer (for KC)	[MF/PB _{year}] [3000]	\$ (y) ⁻¹
Operating Cost Economic Parameters		
Hot Utility	100	\$ (y) ⁻¹ (kW) ⁻¹
Cold Utility	20	\$ (y) ⁻¹ (kW) ⁻¹
H _y	7000	hr (y) ⁻¹
C _{el}	0.14	\$ (kWh) ⁻¹

In this case, which is frequently studied in the literature, both the hand calculation with the pinch method and the SYNHEAT model solution give the same result. First, the cost of HEN is minimized. HEN design is shown in **Figure 5.14** below.

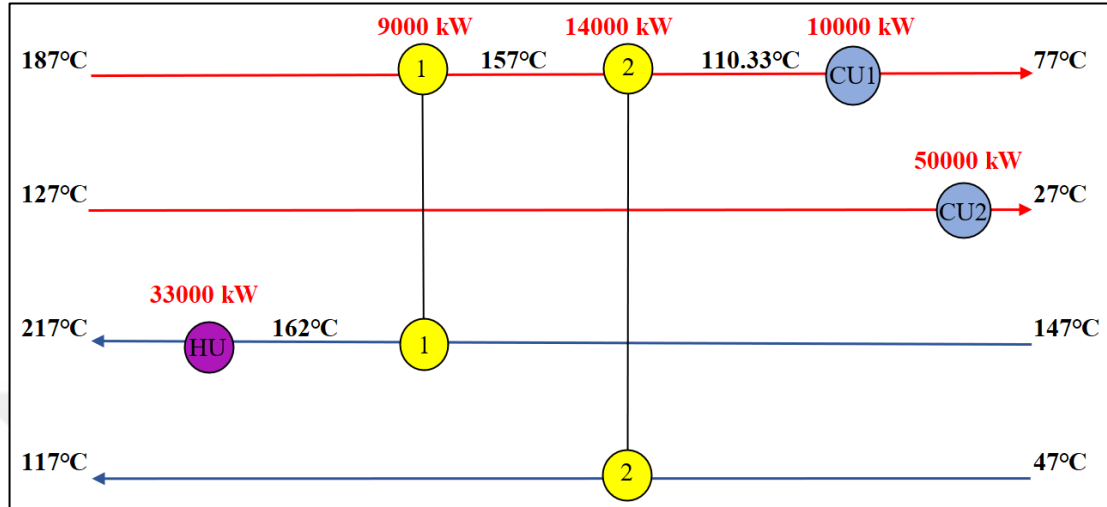


Figure 5.14: HEN for case study 2 without power cycle.

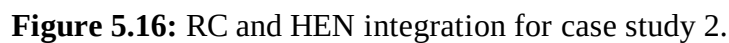
HEN is designed with minimum cost using a total of five heat exchangers: two between the process streams and three in the utility. The 60000 kW of waste heat delivered to the utility from the hot streams is transferred to the power cycle in the maximum amount allowed by the temperature constraints. The areas of heat exchangers in the HEN without power cycle integration are provided in **Table 5.9** below.

Table 5.9: Case study 2 heat exchanger areas.

Heat Exchanger	Area (m ²)
1	1100.56
2	551.47
HU	463.29
CU1	282.60
CU2	2507.37
Total Area = 4905.30 m ²	

As stated in the reference where economic parameters are given, MF is 1.5, PB_{year} is 5 is assumed. When the cost calculations are analyzed in this HEN designed with five heat exchangers and using 93000 kW of utility in total, Annual Investment Cost is \$ 871094.51, Annual Operating Cost is \$ 4500000 and Total Annual Cost is \$

In this study, a more economical HEN is designed according to the compared paper. The first step is completed by optimising only HEN without the power cycle. Then, the power cycle is integrated with the temperature constraints from HEN and the second optimization is performed within the cycle. The integration of the five different power cycles and the compared study is shown in **Figures 5.15-20** below.



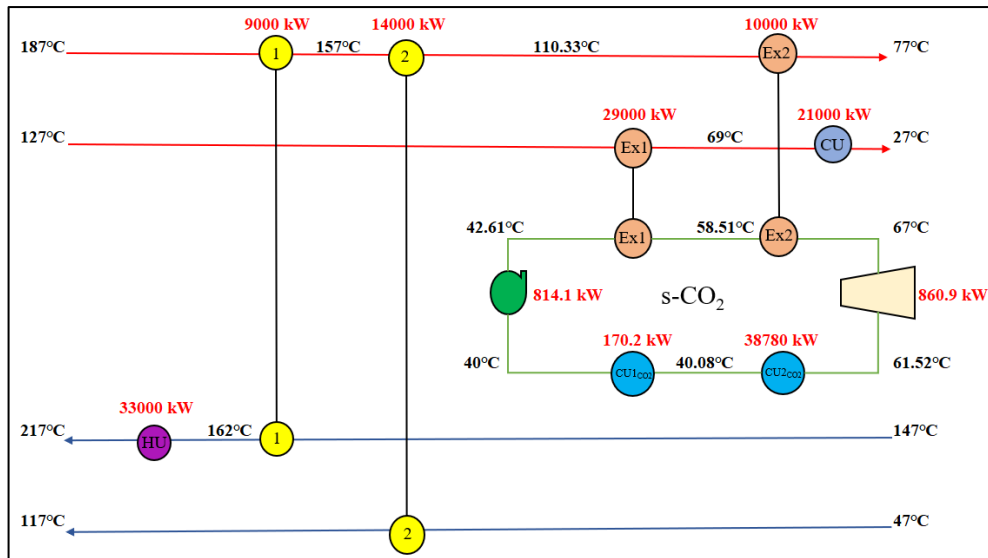


Figure 5.17: s-CO₂C and HEN integration for case study 2.

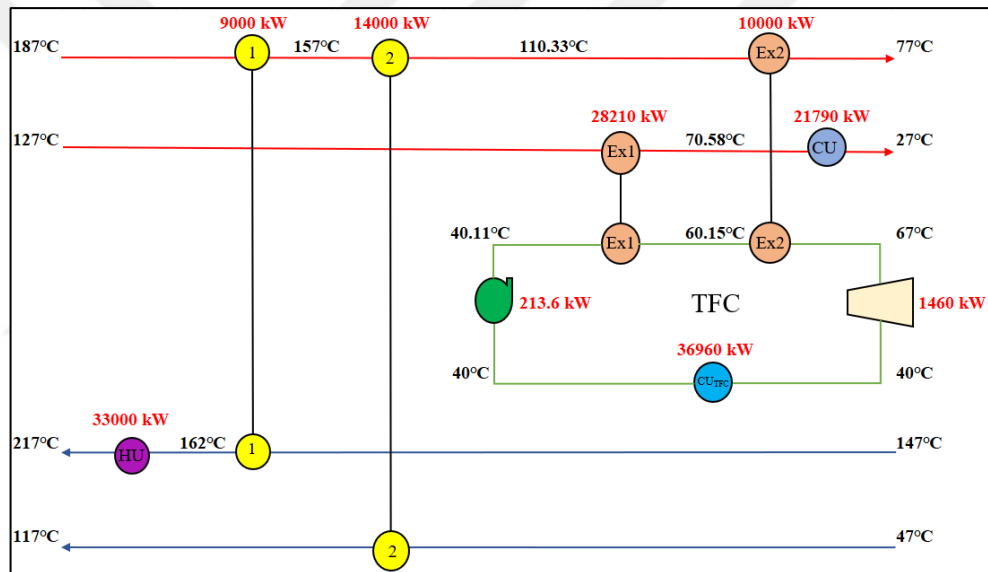


Figure 5.18: TFC and HEN integration for case study 2.

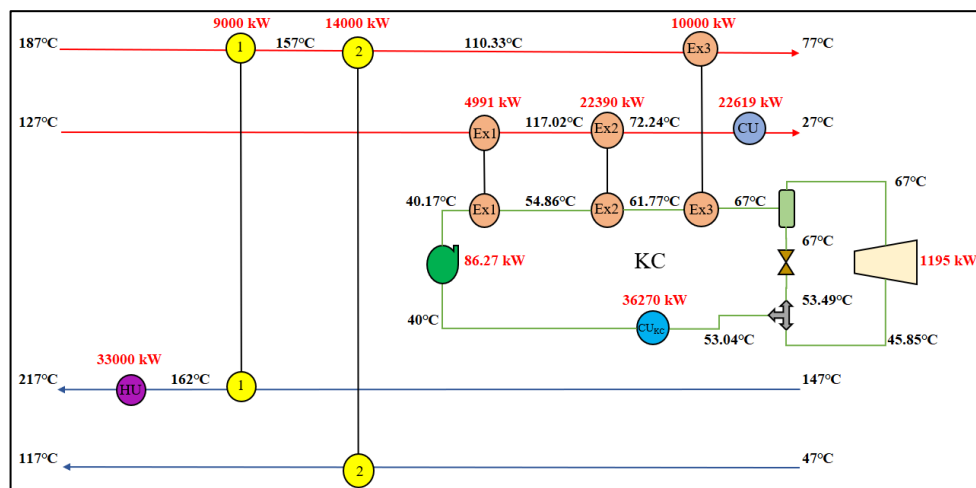


Figure 5.19: KC and HEN integration for case study 2.

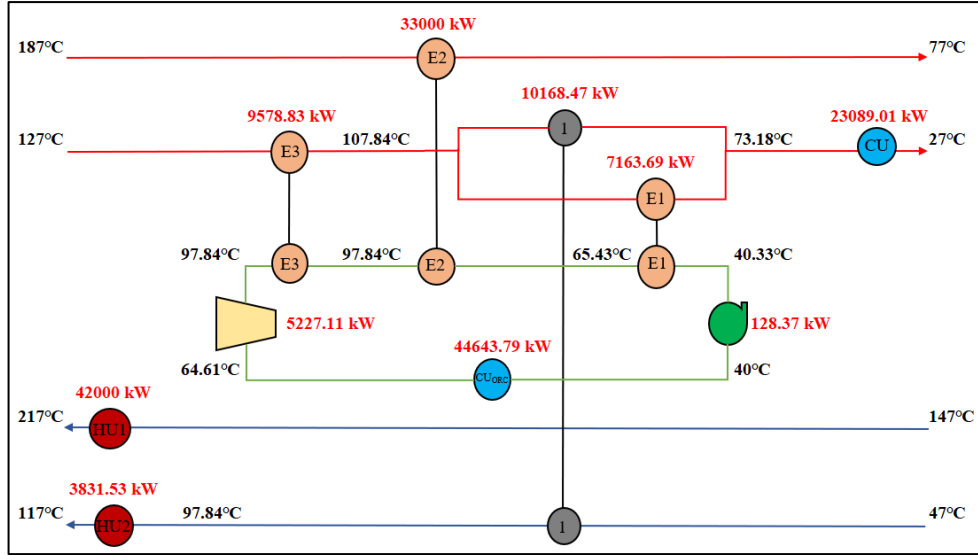


Figure 5.20: Integration of the ORC and HEN [Watanapanich *et al.*, (2022)].

Power cycles are integrated with temperature constraints coming from the optimization of HEN. The output temperature drops to 77°C as the hot stream send waste heat to the utilities. Accordingly, the maximum temperature the power cycle can reach is 67°C with $\Delta T_{\min}$ of 10°C. Since the outlet temperature of the cooling water is also 30°C, the pump inlet temperature are designed as 40°C. Also, unlike Case Study 1, this case study uses n-pentane as the working fluid in ORC and TFC.

HEN is divided into stages as the method of this thesis. As shown in **Figure 5.15**, the first two stages include heat exchangers used for energy recovery between process streams. In addition to the last stage, another stage is added and this is called the power stage. In the power stage, the waste heat sent to the cold utility is recovered and the heat exchanger is eliminated and integrated into the power cycles.

In the power stage, all waste heat in H1 is transferred to the cycle. The waste heat that H2 can transfer is the heat that the H2 stream can send up to the temperature constraint. Therefore, it can send heat until the temperature of H2 is 77°C. The waste heat that the cycle can take after the applied temperature constraints is determined by the mass flow rate of the working fluid. After the thermodynamic model designed in UNISIM DESIGN, appropriate mass flow rates are found by parametric operation in Python. Thus, the maximum amount of waste heat that the cycle can take and the maximum power it can generate are found.

The economic results of the cycles are presented in **Table 5.10** below. Although RC is more prominent in terms of electricity revenue, when annual investment costs are

considered, ORC is the most economical cycle as in Case Study 1. It is becoming popular in the RC in the event of a significant increase in the unit price of kWh of electricity.

When compared with the example from the literature in Case study 2, even if the annual investment and operating costs seem more economical, it falls behind the referenced article in TAC since the electricity income is low in direct proportion to the net power generated. In the referenced study, the heat transferred from the heat exchanger network to the cycle is 49742.52 kW and the turbine inlet temperature is 97.84°C. This is because the study employs simultaneous optimization.

In the method proposed in this thesis, energy transfer in the first two stages occurs between the processes, while the heat in the power stage is directed to the power cycle. In simultaneous optimization, the heat sent to the cycle can also be in the first stages according to the optimization result. Therefore, the power produced increases as the amount of heat received by the cycle increases and also as the maximum temperature that the cycle can reach increases. The economic results of all cycles and the compared reference study are also shown in **Figure 5.21** below.

Table 5.10: TAC results for all cycles for case study 2.

	ORC (Article)	ORC	RC	s-CO ₂ C	TFC	KC
Total HE Area [m ²]	9930.84	8888.91	10050.20	7947.40	8813.46	8521.37
Net Power Output [kW]	5098.74	2145.79	2307.56	46.80	1246.40	1108.73
Investment Cost [\$ (y) ⁻¹]	2568834.21	2003179.47	2209288.87	1749715.02	2010774.42	1811573.61
Operating Cost [\$ (y) ⁻¹]	5937809	4457200	4453800	4499004	4475000	4477780
Electricity Revenue [\$ (y) ⁻¹]	4996765.20	2102874.20	2261413.21	45864	1221472	1086555.40
Total Annual Cost [\$ (y) ⁻¹]	3509878.01	4357505.27	4401675.66	6202855.02	5264302.42	5202798.21

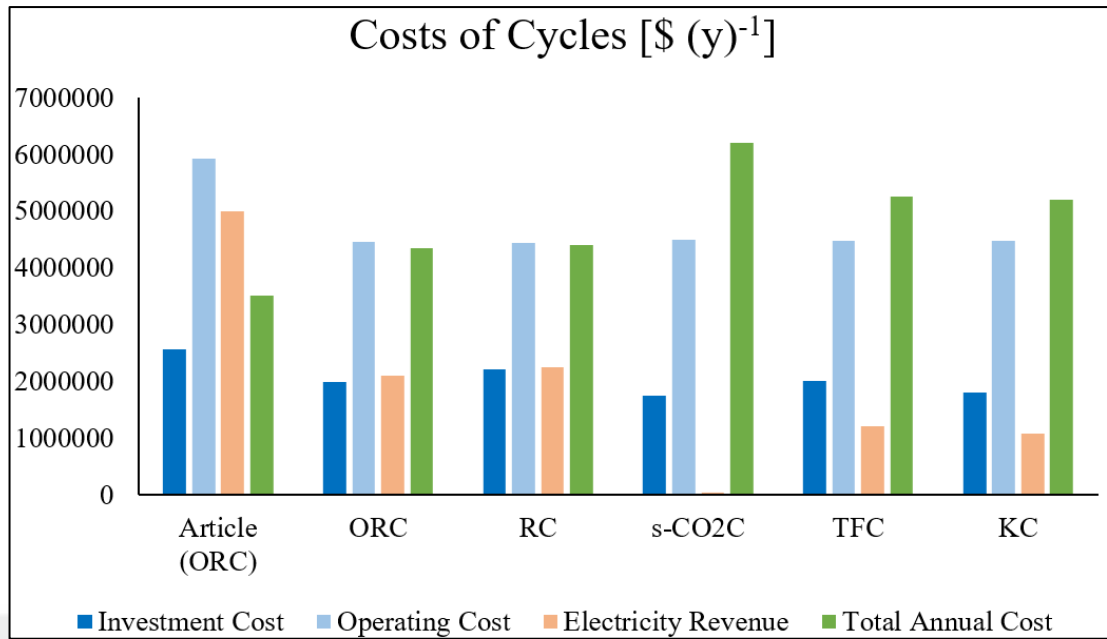


Figure 5.21: Cost of all cycles.

TAC of HEN without power cycle is \$ 5371094.51. The comparative annual profits after the integration of the power cycles is visualized in **Figure 5.22** below. The annual profit of the reference study is the highest as shown in **Figure 5.22** below. Then the profit of the heat exchanger network of this study with ORC integration is the highest. In s-CO₂C, on the other hand, there is not profit but loss.

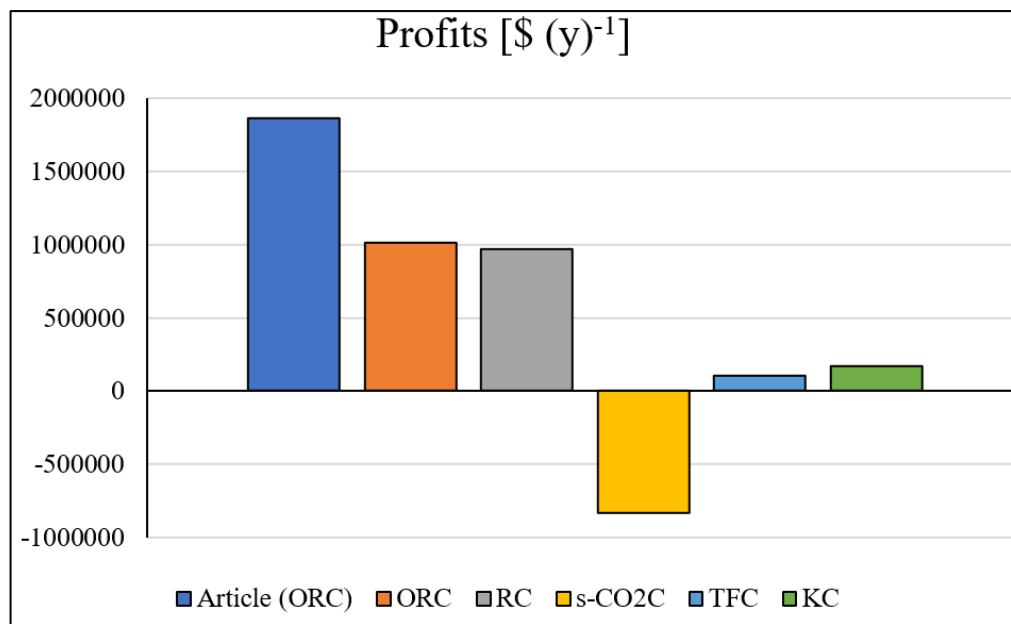


Figure 5.22: Profit rates according to cycles of case study 2.

6. CONCLUSIONS

In this study, total annual cost of power cycles integrated with heat exchanger networks were optimized. Five different power cycles were compared. Heat exchanger network cost was minimized with the SYNHEAT model. Power cycles were modeled with the constraints formed according to the results of heat exchanger network optimization. The modeled power cycles were then optimized in Python.

The best results in terms of total annual cost were in ORC as shown in Case Study 1 and 2. The thermophysical properties of the organic fluids used in ORC provide advantages over the fluids used in other cycles such as water, CO₂ and ammonia-water mixtures. RC seems to be the best cycle in terms of power generated. However, due to the thermophysical properties of the working fluid water used, the total annual cost was higher than ORC as the heat exchangers integrated with heat exchanger network were costly. If there is a significant increase in the unit price of electricity, RC may be more economically viable than ORC.

Even if the same organic fluid is used in TFC, it cannot perform as well as ORC because turbine inlet phase is saturated liquid. In addition, since more costly turbines are used, it is seen that it is not preferred in terms of total annual cost. Although KC is a cycle used like ORC to generate power at low temperatures, it was insufficient in terms of the power produced. This affects total annual cost. s-CO₂C was the most cost inefficient cycle. Also, one of the other reasons why this cycle is not preferred is the safety issues. Because the cycle operates under high pressures and this situation both creates a safety problem and requires more costly equipment selection.

The proposed method was compared with the literature. Since sequential optimization was performed instead of simultaneous optimization, this study was weak when comparing ORC results with the literature. Simultaneous optimization results in a wider range of temperature constraints. In addition, the amount of heat transferred to the power cycle is higher than the proposed method, which is advantageous in terms of power generated. On the other hand, the advantage of this study compared to the literature is the software used, UNISIM DESIGN. Realistic results were achieved using a program with a powerful thermodynamic data base, and integration was

achieved using Python. In addition, in power cycle optimization, multiple parameters can be included in the optimization at the same time to access the enthalpy data of the working fluid in the cycle. Selection of decision variables are more various. In spite of the disadvantages of the method, the use of a program with a strong thermodynamic background and the practicality of the method give this study an advantage.



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BIOGRAPHY

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PUBLICATIONS AND PRESENTATIONS FROM THE THESIS

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