SUDAN UNIVERSITY
OF SCIENCE AND TECHNOLOGY
COLLOGE OF GRADUATE STUDIES

Aplication Of Combined Exergy And Pinch Analysis For Khartoum North Power Station Phase (2)

تطبيق التحليل المشترك للايكسيرجي والبينش علي محطة كهرباء الخرطوم بحري المرحلة الثانية

A Thesis Submitted in Partial Fulfillment of the Degree of M.Sc. in Mechanical Engineering (POWER)

By:
AHMED MOHAMED ALI KARAR

Supervisor:
Dr. Ali Mohammed Hamdan Adam

April 2014

Dedication

To my father who gives me direction to the sky

To my mother who gives me lovely life

To my brothers and sisters who give me support

To my wife who gives me wormed life

Acknowledgement

First of all I would like to submit my best greeting to my supervisor Dr. Ali Hamdan for his acceptance to supervise my research and for his directions during preparing this research. Alsomy full respect and appreciation to all friends, colleagues and co-workerenhanced me in this research and helped me in data collection stage. Finally, a lot of thanks to Khartoum North Power Station Directorate which gives me permeations to apply this research and analysis in the station and has been supplying me by all data I needed throughout the research.

Abstract

In this research,Khartoum North Power Station Phase (2)has been discussed in accordance with Exergy and Pinch technology analysis.Exergy analysis is introduced and defined as energy quality analysis. Also exergy is work potential of energy. The main parameters of exergy analysis has been considered in this research is exergy, exergydestruction(irreversibilities) and second-law efficiency.As outcomes of exergy analysis discussion, the full power plant performance definition is to specify actual work, irreversibility and second-law efficiency which is founded in KNPSP(2) $60\ MW, 131.3\ MW\ and\ 31.4\%$ respectively. Also, the worst components in the plant needs to redesign and re-evaluation are first and second feed water heat exchanger according to exergyanalysis. For efficiency of regenerative feed water heat exchanger, the net workincludes heat exchangers from first to fourth is inefficient and needs to redesign but the other net work includes fifth and sixth heat exchangers is efficient heat exchange network.

تجريدة

فيهذاالبحث،تحليلالطاقةوالايكسيرجيلمحطة كهرباء بحري الحراري (2) تمت مناقشتها بالاشتراك مع تقنية البنش. المعالمالرئيسيةالتيتمالنظرفيهاهيالايكسيرجي،وتدميرالايكسيرجي، اللاعكوسيةوكفاءةالقانونالثاني. نتيجة مناقشةتحليلالايكسيرجي،أعتبر التعريف الكامل لأداء المحطة الحرارية هو أن تحدد كمية الشغل الحقيقي الناتج من المحطة، اللاعكوسية وكفاءة القانون الثاني؛ و عليه عرف أداء محطة الخرطوم بحري المرحلة الثانية بالنتيجة الاتية 60 ميقاوات، 131.3 ميقاوات و 31.4% على التوالي. كذلك، المكوناتالأسوءفيالمحطة هي مسخن مياه التغذية الأول و الثاني و التي تحتاج الى إعادة تصميم و تقييم من حيث الطاقة المتاحة المستفاد منها. أما بالنسبة لكفاءة شبكة مبادلات ماء التغذية الحرارية فإن الشبكة التي تتكون من المبادل الحراري الأول حتى الرابع تعتبر ذات كفاءة متدنية وتحتاج لإعادة تصميم، أما التي تحتوي على المبادلين الحراريين الخامس و السادس تعتبر شبكة كفؤة .

Contents

Abriviations

$\boldsymbol{CPEA}$	Combined pinch and exergy analysis
$\boldsymbol{fg}$	Flue gases
$\boldsymbol{FLT}$	First Law of Thermodynamics
$\boldsymbol{fw}$	feadwater
$\boldsymbol{FWHEX}(\#)$	Feadwater heat exchanger number (#)
$\boldsymbol{GCC}$	Grand Composite Curve
$\boldsymbol{HEN}$	Heat exchanger networks
$\boldsymbol{HXG}$	Heat Exchanger
$\boldsymbol{KNPSP}(\boldsymbol{2})$	Khartoum North Power Station Phase (2)
$\boldsymbol{Ke}$	Kinetic energy
$\boldsymbol{p}$	Combustion products
$\boldsymbol{Pe}$	Potential energy
$\boldsymbol{r}$	Combustion reactants
$\boldsymbol{SLT}$	Second Law of Thermodynamics
$\boldsymbol{T/H}$	Temperature / Heat load
$\boldsymbol{T-H}$	Temperature-Enthalpy
$\boldsymbol{CV}$	Control Volume

Symbols

$\boldsymbol{X}$	exergy	kJ/kg
$\boldsymbol{I}$	irreversibility	kJ/kg
$\boldsymbol{W_{rev}}$	Reversible work	kJ/kg
$\boldsymbol{W_u}$	Useful work	kJ/kg
$\boldsymbol{\eta_{II}}$	Second-law efficiency	%
$\boldsymbol{\eta_{rev}}$	Reversible cycle efficiency	%
$\boldsymbol{T_L}$	Lowest temperature	K
$\boldsymbol{T_H}$	Highest temperature	K
$\boldsymbol{\eta_{th}}$		%
$\boldsymbol{P}$	pressure	bar
$\boldsymbol{T}$	Temperature	K
$\boldsymbol{h}$	enthalpy	kJ/kg
$\boldsymbol{s}$	entropy	$kJ/kg.K$
$\boldsymbol{P_0}$	Pressure at date state	bar
$\boldsymbol{T_0}$	Temperature at date state	K
$\boldsymbol{h_0}$	enthalpy at date state	kJ/kg
$\boldsymbol{s_0}$	entropy at date state	$kJ/kg.K$
$\boldsymbol{u}$	Internal energy	kJ/kg
$\boldsymbol{V}$	volume	m^3
$\emptyset$	Closed system exergy	kJ/kg
$\boldsymbol{\psi}$	Flow exergy	kJ/kg
$\boldsymbol{Q}$	Heat transfer	kJ
$\boldsymbol{m}$	mass	Kg
$\boldsymbol{\overline{h^\circ}_f}$	enthalpy of formation	$kJ/Kmol$
$\boldsymbol{\overline{h^\circ}}$	the sensible enthalpy at the standard reference state of 25°C and 1 atm	$kJ/Kmol$
$\boldsymbol{\overline{h}}$	the sensible enthalpy at the specified state	$kJ/Kmol$
$\boldsymbol{\overline{s^\circ}(T,P_o)}$	Absolute entropy at pressures 1 atm	$kJ/Kmol.K$
$\boldsymbol{\overline{s}(T,P)}$	Absolute entropy at any pressure	$kJ/Kmol.K$
$\boldsymbol{Cp}$	Specific heat capacity	$KJ/Kg.K$
$\boldsymbol{CP}$	heat capacity flow rate	KW/K
$\boldsymbol{H}$	Heat load	Kw
$\boldsymbol{T_s}$	Initial (supply) temperature	°C
$\boldsymbol{T_t}$	Final (target) temperature	°C

$\boldsymbol{S_s}$	Shifted initial (supply) temperature	℃
$\boldsymbol{S_t}$	Shifted final (target) temperature	℃
$\boldsymbol{A}$	area	m^2
$\boldsymbol{U}$	overall heat transfer coefficient	kW/m^2K
$\Delta\boldsymbol{T_{LM}}$	log mean temperature difference	K
$\boldsymbol{\Delta Tmin}$	Minimum temperature deference	℃

List of Tables

List of Figures

Chapter one: Literature Review

1.1 Introduction

Energy consumption is one of the most important indicator showing the development stages of countries and living standards of communities. Population increment, urbanization, industrializing, and technologic development result directly in increasing energy consumption. This rapid growing trend brings about the crucial environmental problems such as contamination and greenhouse effect. Currently, 80% of electricity in the world is approximately produced from fossilfuels (coal, petroleum, fuel-oil, natural gas) fired thermal power plants, whereas 20% of the electricity is compensated from different sources such as hydraulic, nuclear, wind, solar, geothermal and biogas. Generally, the performance of thermal power plants is evaluated through energetic performance criteria based on first law of thermodynamics, including electrical power and thermal efficiency. In recent decades, the exergetic performance based on the second law of thermodynamics has found as useful method in the design, evaluation, optimization and improvement of thermal power plants. The exergetic performance analysis not only determines magnitudes, location and causes of irreversibilities in the plants, but also provides more meaningful assessment of plant individual components efficiency. These points of the exergetic performance analysis are the basic differences from energeticperformance analysis. Therefore, it can be said that performing exergetic and energetic analyses together can give a complete depiction of system characteristics. Such a comprehensive analysis will be a more convenient approach for the performance evaluation and determination of the steps towards improvement [9].

In addition to energetic performance which based on first law of thermodynamics and exergetic performance which based on second law of thermodynamics, there is the Pinch Technology based on both first and second laws of thermodynamics. The pinch technology is a new set of thermodynamically based methods that guarantee minimum energy level in design heat exchanger networks (HEN). Over the last two decades, it has emerged as an unconventional development design and energy conservation.

In the literature, there exist a number of papers/studies concerning energy, exergy and pinch analysis of thermal power plants will be introduced.

1.1 Previous studies

Alvors,et al.[1] in this paper a comparison between exergy analysis and pinch technology has been applied by studying systems where threshold problems occur and heat pumps are available. Also, the differences between exergy(second law) analysis and pinch technology explained. thus Pinch technology was defined as a method to improve heat exchanger networks(HEN) by matching excess of heat and cold streams in a systemthrough composite curves in a T–H diagram (Temperature – Enthalpy). Unfortunately, many people today uses the pinch technologyas a method of optimizing systems other than HEN.However, pinch technology is only a method to improve structure of a HEN under specific conditions. It should be noticed that there is a fundamental difference between optimization and improvement. Optimization in a general sense involvesthe determination of a highest or lowest value over somerange. Thus, pinchtechnology is just a method to improve the design of HEN underspecified restrictions.

Ataei and Yoo [2] presented simulation of a 325 MW steam power plant was performed in a Cycle-Tempo 5.0 simulator and operational parameters of the Rankine cycle were optimized using the Exergy concept combined with a Pinch based approach. The Combined Pinch and Exergy Analysis (CPEA) first considers the representation of the hot and cold Composite Curves of the Rankine cycle and defines the energy and Exergy requirements. The basic assumption of the minimum approach temperature difference (DTmin) required for the Pinch Analysis is represented as a distinct Exergy loss that increases the fuel requirement for power generation. The exergy composite curves put the focus on the opportunities for fuel conservation in the cycle.Ataei [3] the modification of an olefin plant was performed and its refrigeration cycles were optimized using the exergy concept combined with a pinch-based approach. By the combined pinch and exergy analysis (CPEA), the present research corrected the temperature levels of the refrigeration cycles of an olefin plant to minimize exergy loss in the network and to economically and efficiently reduce shaft work demand and energy consumption in these cycles.The application results of CPEA in the olefin plant showed that its power consumption could be reduced by 2553 kW with a reasonable investment and payback period time.

Al-Doori [4] This study performed an exergetic analysis for a Baiji plant with a gas-turbine of capacity 159MW. Each component of the system was tested in accordance with the laws of mass and energy conversion. The aspects under consideration were the quantitative exergy balance for the entire system and for each component, respectively. At different temperatures, rate of irreversibility of system components, efficiency of exergy and the efficiency flaws were highlighted for each component and for the whole plant. The exergy flow of a material is classified into the groupings of thermal, mechanical and chemical exergy in this study and a stream of entropy-production. Fuel oil of low heating value of 42.9 MJ/kg was used as the fuel. The evaluation addressed the question of how the fluctuations in cycle temperatures influence the exergetic efficiency and exergy destruction in the plant. The rate of exergy destruction in the turbine was around 5.4% whereas that in the combustion chamber was about 36.4%. When a 14°C rise was done in the temperature, exergy efficiency for the combustion chamber and the turbine was calculated to be 45.43% and 68.4%, respectively. According to the results of the study, the combustion chamber and turbine are found to be chief means of irreversibilities in the plant. Also, it was identified that the exergetic efficiency and the exergy destruction are considerably dependent on the alterations in the turbine inlet temperature. On the basis of these results, recommendations are presented for advancement of the plant.

Amir [5] In this paper, the useful concept of energy and exergy utilization is analyzed, and applied to the boiler system. Energy and exergy flows in a boiler have been shown in this paper. The energy and exergy efficiencies have been determined as well.

Ganpathy, et el.[6] this paper deals with an exergy analysis performed on anoperating 50MWe unit of lignite fired steam power plant at Thermal Power Station-I, Neyveli Lignite Corporation Limited, Neyveli, Tamil Nadu, India. The exergy losses occurred in the various subsystems of the plant and their components havebeen calculated using the mass, energy and exergy balance equations. The distribution of the exergy losses in several plantcomponentsduring the real time plant running conditions has been assessed to locate the process irreversibility. The Firstlaw efficiency (energy efficiency) and the second law efficiency (exergy efficiency) of the plant have also been calculated.

Jafari, et al.[7]In this study, The exergy analysis of Shazand power plant's boiler is presented. Theaim of this study is to use energy and exergy analysis to identify the locations andmagnitudes of losses in order to maximize the performance of a boiler. The resultsindicate that about 48 % of heat energy generated in the boiler is destroyed. Alsoresults, show that the major irreversibilities are due to losses Furnace system. Thatensue, combustion process is the most effective factor in irreversibilty of boiler.

Kwambai [8] The processes of electricity production from geothermal resources at *Olkaria I* Power Plant in Kenya were analysed using the exergy analysis method. The objectives of the analysis were to determine the overall second law (exergy) of efficiency of the power plant pinpoint the locations and quantities of exergy losses and wastes and suggest ways to address these losses and wastes. In the analysis, the power plant was simplified into sub-systems, each with distinct exergy inflows and outflows and approximated into steady-state flow. A few assumptions and simplifications were made. The results show that Olkaria I Power Plant has an overall second law of efficiency of 34.6% and an overall first law of efficiency of 15%. The analysis reveals that 6 MW of exergy are wasted in the separated brine while 11 MW exergy are lost in the steam transmission system. Significant losses are found to occur in the turbines, condensers and the GES system. Although the exergy in the wasted brine is relatively small compared to that in the steam, it could still be put to useful work at some investment cost. It is concluded that exergy analysis is an important tool for analysing the performance of geothermal plants and should be incorporated into their designs. It is suggested that the steam transmission system should be investigated further to determine the causes of exergy losses and that ways of utilizing the exergy in the brine be investigated.

Kaushik, et al.[9] The present study deals with the comparison of energy and exergy analyses of thermal power plants stimulated by coal and gas. This article provides a detailed review of different studies on thermal power plants over the years. This review also throws light on the scope for further research and recommendations for improvement in the existing thermal power plants.

Mborah and Gbadam [10] this study aimed to use energy and exergy analysis to identify the locations and magnitudes oflosses in order to maximise the performance of a 500 KW open system steam power plant atBenso Oil Palm Plantation(BOPP). Therequired outputs (work, heat and irreversibility) of the various components are assessed and calculated usingmass, energy and exergy balance equations.

Rashad and El Maihy [11] In this study, the energy and exergy analysis of Shobra El-Khima power plant in Cairo, Egypt is presented. The primary objectives of this paper are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load.

Sciubba [12] this paper presents a brief critical and analytical account of the development of exergy concept and its applications. It is based on a careful and extended (in time) consultation of a very large body of published references taken from archival journals, textbooks and other monographic works, conference proceedings, technical reports and lecture series. it has been tried to identify the common thread that runs through all of the references, to put different issues into perspective, to clarify dubious points, to suggest logical and scientific connections and priorities.

Siahaya [13] This paper was written to evaluate an energy, exergy and thermoeconomic analysis of a binary geothermal power plant. As a case of this study, data of geothermal field at Lahendong, North Sulawesi

Indonesia was used. The results of this analysis shows that the cost formation process throughout a plant from the total capital investment, revenue requirement, main product unit cost ($/kWh) and cost rate associated with the product of the geothermal power plant ($/h) can be determined.

Sanjay and Dr.Mehta [14] In this paper presents energy and exergy analysis method for thermal power plant and analysis carried out on 125 MW coal base thermal power plant. This analysis shows exergy efficiency is less at each and every point of unit equipments. Also presents major losses of available energy at combustor, superheater, economiser and air-pre heater section. In this article also shown energy exergy efficiency, exergy disctructon and energy losses comparison charts.

Vundela, et al.[15] In this paper, a thermodynamic analysis of a coal based thermal power plant and gas based cogeneration power plant has been carried out. The energy and exergy analysis has been studied for the different components of both power plants. The paper analyses the information available in the open literature regarding energy and exergy analysis on high temperature power plant has been included. A comprehensive literature review on thermal power plants, especially boiler in coal base thermal power plants and combustion chamber in gas-steam cogeneration has been included. Finally, explaining the procedure of analysis of thermal power plant systems by exegetical approach.

Vosough, et al.[16] this paper about the concept of exergy, application of exergy in various fields and its characteristic has been discussed and different forms of exergy have been derived. Also a brief comparison between energy and exergy analysis has been done.

Keeping in view the facts stated above, it can be expected that performing an analysis based on energy, exergy and pinch analysis will be meaningful for performance comparisons, assessments and improvement for thermal power plants. Also, it can guide the ways of efficient and effective usage of fuel resources by taking into account the quality and quantity of the energy used in the generation of electric power in thermal power plants.

1.2 Objective

The purpose of this study is to carry out combined exergy and pinch analysis, at the design conditions, for the existing thermal power plant *Khartom North Power Station Phase (2)* [hereafter KNPSP(2)] in order to identify the needed improvement.

1.3 Scope

This research will perform analytical study by using exergy and pinch technology approach on the main components of KNPSP(2)which are boiler, steam turbine, condenser, cooling tower and the set of feed water heaters.

The magnitude and location of exergy losses will be indentified, and the second law efficiency will be calculated. Also the pinch analysis will be applied in the set of feed water heaters and the heat exchangers network weaknesses will be showed.

1.4 Methodology

As exergy is not well known as well as quality of energy is not usually taken into account. Thus exergy concept, exergy analysis approach, second law efficiency (which reflects quality of energy) for systems will be explained in second chapter. The pinch technology and its analytical tools will be introduced in the third chapter. The design conditions of the KNPSP(2) will be illustrated and combined exergy and pinch analysis will be applied and the results of this analytical study will be discussed and the improvements or alternatives will be provided in the fourth chapter of this research.

Chapter Two: Exergy

2.1 Exergy: Work potential of energy

When a new energy source, such as a geothermal well, is discovered, thefirst thing the explorers do is estimate the amount of energy contained in thesource. This information alone, however, is of little value in decidingwhether to build a power plant on that site. What really need to be known isthe work potential of the source. As is well known, the work done during a process depends on the initialstate, the final state, and the process path. That is,

$$Work = f(\,Initial\ sate, Process\ path, Final\ state)$$

A system is said to be in the **dead state** when it is in thermodynamic equilibriumwith the environment. At the dead state, a system isat the temperature and pressure of its environment (in thermal and mechanicalequilibrium); it has no kinetic or potential energy relative to the environment(zero velocity and zero elevation above a reference level); and it does notreact with the environment (chemically inert). Also, there are no unbalancedmagnetic, electrical, and surface tension effects between the system and its surroundings.

The factors that cause a process to be irreversible are called **irreversibilities.** They include friction, unrestrained expansion, mixing of two fluids, heattransfer across a finite temperature difference, electric resistance, inelasticdeformation of solids, and chemical reactions. The presence of any of theseeffects renders a process irreversible. A reversible process involves none ofthese.

2.1.1 Exergy versus Energy

To identify exergy deeply, it is better to compare with energy as in the following table 2.1

Table 2.1 Comparison of energy and exergy.

ENERGY	EXERGY
is dependent on the parameters of matter or energy flow only, and independent of the environment parameters.	is dependent both on the parameters of matter or energy flow and on the environment parameters.
has the values different from zero (equal to mc2 upon Einstein's equation).	is equal to zero (in dead state by equilibrium with the environment).
is governed by the FLT for all the processes.	is governed by the FLT for reversible processes only (in irreversible processes it is destroyed partly or completely).
is limited by the SLT for all processes (including reversible ones).	is not limited for reversible processes due to the SLT.
is motion or ability to produce motion.	is work or ability to produce work.
is always conserved in a process, so can neither be destroyed or created.	is always conserved in a reversible process, but is always consumed in an irreversible process.
is a measure of quantity only	is a measure of quantity and quality due to entropy

2.2 Exergy of Kinetic and Potential Energy

Kinetic and potential energy is a form of mechanical energy, and thus it can be converted to work entirely.

Exergy of Kinetic energy: $X_{ke} = Ke = \frac{V^2}{2}(kJ/kg)$ (2.1)

Where V is the velocity of the system relative to the environment

Exergy of potential energy: $X_{pe} = Pe = gz(kJ/kg)$ (2.2)

Where $g$is the gravitational acceleration and z is the elevation of the system relative to a reference level in the environment.

2.3 Reversible Work and Irreversibility

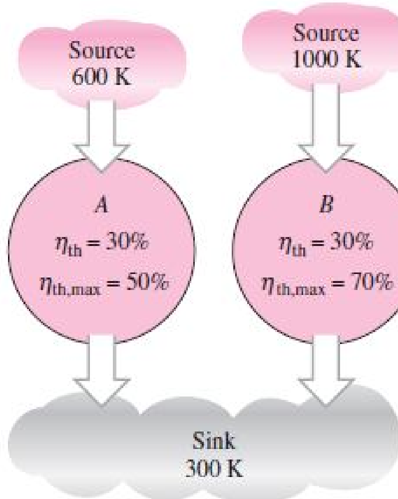
Reversible work Wrev is defined as the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states. Any difference between the reversible work Wrev and the useful work Wu is due to the irreversibilities present during the process, and this difference is called irreversibility I.

$$I = W_{rev,out} - W_{u,out} \qquad or \qquad I = W_{u,in} - W_{rev,in} \quad (2.3)$$

2.4 Second-Law efficiency, η_{II}

The fraction of the heat input that is converted to network output is a measure of the performance of a heat engine and is called the thermal efficiencyη_{th}.The *thermal efficiency* for devices is defined as a measure of their performance. It is defined onthe basis of the first law only, and it is sometimes referred to as the*first-law efficiencies.* The first law efficiency, however, makes no referenceto the best possible performance, and thus it may be misleading.

An illustrative example will be provided to clarify that thermal first-law efficiencies do not lead to the best possible performance. Consider two heat engines, both having a thermal efficiency of 30 percent,as shown in Figure 2.1. One of the engines (engine A) is supplied with heat from a source at 600K, and the other one (engine B) from a source at 1000K. Both engines reject heat to a medium at 300 K. At first glance, bothengines seem to convert to work the same fraction of heat that they receive;thus they are performing equally well. When a second look is taken at theseengines in light of the second law of thermodynamics, however, it will be seen as atotally different picture. These engines, at best, can perform as reversibleengines, in which case their efficiencies would be



\\

Figure 2.1 Two heat engines that have the same thermal efficiency, but different maximum thermal efficiencies

$$\eta_{rev,A} = \left(1 - \frac{T_L}{T_H}\right)_A = 1 - \frac{300\ K}{600\ K} = 50\%$$

$$\eta_{rev,B} = \left(1 - \frac{T_L}{T_H}\right)_B = 1 - \frac{300\ K}{1000\ K} = 70\%$$

Now it is becoming apparent that engine *B* has a greater work potentialavailable to it (70 percent of the heat supplied as compared to 50 percent forengine *A*), and thus should do a lot better than engine *A*. Therefore, it can be said that engine *B* is performing poorly relative to engine *A* even thoughboth have the same thermal efficiency.
It is obvious from this example that the first-law efficiency alone is not a realistic measure of performance of engineering devices. To overcome thisdeficiency, a**second-law efficiency η_{II}** is defined as the ratio of the actualthermal efficiency to the maximum possible (reversible) thermal efficiencyunder the same conditions:

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \qquad (Heat\ engines) \qquad (2.4)$$

Based on this definition, the second-law efficiencies of the two heat enginesdiscussed above are

$$\eta_{II,A} = \frac{0.30}{0.50} = 0.60 \quad and \quad \eta_{II,B} = \frac{0.30}{0.70} = 0.43$$

That is, engine *A* is converting 60 percent of the available work potential touseful work. This ratio is only 43 percent for engine *B*.

The second-law efficiency can also be expressed as the ratio of the usefulwork output and the maximum possible (reversible) work output:

$$\eta_{II} = \frac{W_u}{W_{rev}} (work - producing\ devices) \qquad (2.5)$$

This definition is more general since it can be applied to processes (in turbines,piston–cylinder devices, etc.) as well as to cycles. Note that the second-lawefficiency cannot exceed 100 percent (Figure 2.2).

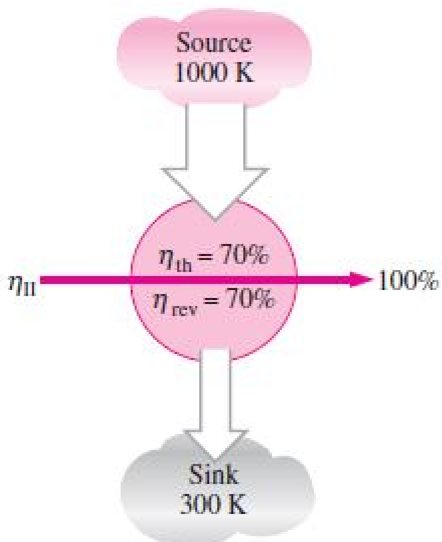


Figure 2.2 Second-law efficiency of all reversible devices is 100 precent

the second-law efficiency of a system during a process is defined as

$$\eta_{II} = \frac{Exergy\ recovered}{Exergy\ supplied} = 1 - \frac{Exergy\ destoried}{Exergy\ supplied} \qquad (2.7)$$

2.5 Exergy Change of System

Relations for the exergies and exergy changes are developed below for afixed mass and a flow stream.

2.5.1 Exergy of a Fixed Mass (a closed system)

In general, internal energy consists of sensible, latent, chemical, and nuclear energies. However, in the absence of any chemical or nuclear reactions, the chemical and nuclear energies can be disregarded and the internal energy can be considered to consist of only sensible and latent energies that can be transferred to or from a system as heat whenever there is a temperature difference across the system boundary. The second law of thermodynamics states that heat cannot be converted to work entirely, and thus the work potential of internal energy must be less than the internal energy itself. But how is much less?

To answer that question, it is needed to consider a stationary closed system at a specified state that undergoes a reversible process to the state of the environment (that is, the final temperature and pressure of the system should be T_0 and P_0, respectively). The useful work delivered during this process is the exergy of the system at its initial state (Figure 2.3).

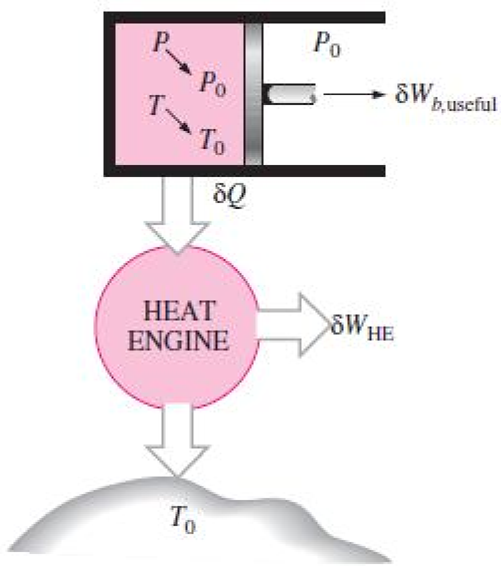


Figure 2.3 The *exergy* of specified mass at a specified state is the useful work that can be produced as the mass undergoes a reversible process to the state of the environment

Consider a piston–cylinder device that contains a fluid of mass m at temperature T and pressure P. The system (the mass inside the cylinder) has a volume V, internal energy U, and entropy S. The system is now allowed to undergo a differential change of state during which the volume changes by a differential amount dV and heat is transferred in the differential amount of δQ. Taking the direction of heat and work transfers to be from the system (heat and work outputs), the energy balance for the system during this differential process can be expressed as

$$\underbrace{\delta E_{in} - \delta E_{out}}_{\text{Net energy transfer by heat, work and mass}} \quad = \quad \underbrace{dE_{system}}_{\text{Change in internal, kinetic, potential, etc., energies}}$$

$$-\delta Q - \delta W \ = \ dU \quad (2.8)$$

Since the only form of energy the system contains is internal energy, and the only forms of energy transfer a fixed mass can involve are heat and work. Also, the only form of work a simple compressible system can involve during a reversible process is the boundary work, which is given to be δW = P dV when the direction of work is taken to be from the system (otherwise it would be −P dV). The pressure P in the P dV expression is the absolute pressure, which is measured from absolute zero. Any useful work delivered by a piston–cylinder device is due to the pressure above the atmospheric level. Therefore,

$$\delta W = PdV = (P - P_0)dV + P_0 dV = \delta W_{b,useful} + \ P_0 dV (2.9)$$

A reversible process cannot involve any heat transfer through a finite temperature difference, and thus any heat transfer between the system at temperature T and its surroundings at T_0 must occur through a reversible heat engine. Noting that dS = δQ /T for a reversible process, and the thermal efficiency of a reversible heat engine operating between the temperatures of T and T_0 is $\eta_{th} = 1 - T_0/T$, the differential work produced by the engine as a result of this heat transfer is

$$\delta W_{HE} = \left(1 - \frac{T_0}{T}\right)\delta Q = \delta Q - \frac{T_0}{T}\delta Q = \delta Q - (-T_0 dS) \rightarrow$$
$$\delta Q = \delta W_{HE} - T_0 dS$$

Then

$$\delta W_{total\,useful} = \delta W_{HE} + \delta W_{b,useful} = -dU - P_0 dV + T_0 dS$$

Integrating from the given state (no subscript) to the dead state (0 subscript)

$$W_{total\,useful} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) \qquad (2.10)$$

Where $W_{\text{total useful}}$ is the total useful work delivered as the system undergoes a reversible process from the given state to the dead state, which is exergy by definition.
A closed system, in general, may possess kinetic and potential energies, and the total energy of a closed system is equal to the sum of its internal, kinetic, and potential energies. Noting that kinetic and potential energies themselves are forms of exergy, the exergy of a closed system of mass m is

$$X = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) + m\frac{V^2}{2} + mgz \qquad (2.11)$$

On a unit mass basis, the closed system (or nonflow) exergy ϕ is expressed as

$$\emptyset = (u - u_0) + P_0(V - V_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$
$$= (e - e_0) + P_0(V - V_0) - T_0(S - S_0)(2.12)$$

Where u_0, v_0, and s_0 are the properties of the system evaluated at the dead state. Note that the exergy of a system is zero at the dead state since $e = e_0$, $v = v_0$, and $s = s_0$ at that state.
The exergy change of a closed system during a process is simply the difference between the final and initial exergies of the system,

$$\Delta X = X_2 - X_1 = m(\emptyset_2 - \emptyset_1) = (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)$$
$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) + m\frac{{V_2}^2 - {V_1}^2}{2} + mg(Z_2 - Z_1)(2.13)$$

or, on a unit mass basis,

$$\Delta\emptyset = \emptyset_2 - \emptyset_1$$
$$= (u - u_0) + P_0(V_2 - V_1) - T_0(S_2 - S_1) + \frac{{V_2}^2 - {V_1}^2}{2} + g(Z_2 - Z_1)$$
$$= (e_2 - e_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)(2.14)$$

Forstationary closed systems, the kinetic and potential energy terms drop out.
When the properties of a system are not uniform, the exergy of the system

can be determined by integration from

$$X_{system} = \int \emptyset \, \delta m = \int_V \emptyset \rho dV \qquad (2.15)$$

Where V is the volume of the system and ρ is density.
Note that exergy is a property, and the value of a property does not change unless the *state* changes. Therefore, the *exergy change* of a system is zero if the state of the system or the environment does not change during the process. For example, the exergy change of steady flow devices such as nozzles, compressors, turbines, pumps, and heat exchangers in a given environment is zero during steady operation.
The exergy of a closed system is either positive or zero. It is never negative. Even a medium at low temperature (T=T0) and/or low pressure (P=P0) contains exergy since a cold medium can serve as the heat sink to a heat engine that absorbs heat from the environment at T0, and an evacuated space makes it possible for the atmospheric pressure to move a piston and do useful work (Figure 2.4).

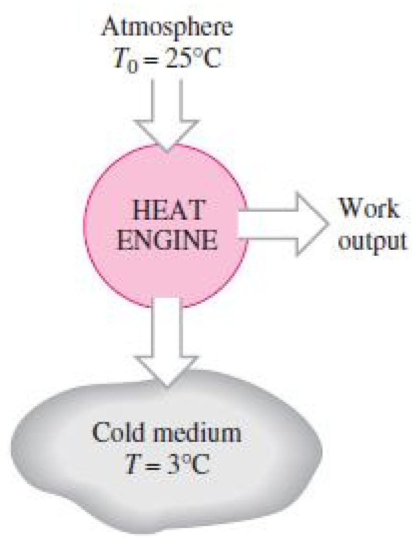


Figure 2.4 The exergy of a cold medium is also a positive quantity since work can be produced by transferring heat to it.

2.5.2 Exergy of a Flow Stream (Open system)

a flowing fluid has an additional form of energy, called the flow energy, which is the energy needed to maintain flow in a pipe or duct, and was expressed as w_{flow}= Pv where v is the specific volume of the fluid, which is equivalent to the volume change of a unit mass of the fluid as it is displaced during flow. The flow work is essentially the boundary work done by a fluid on the fluid downstream, and thus the exergy associated with flow work is equivalent to the exergy associated with the boundary work, which is the boundary work in excess of the work done against the atmospheric air at P0 to displace it by a volume v (Figure 2.5).

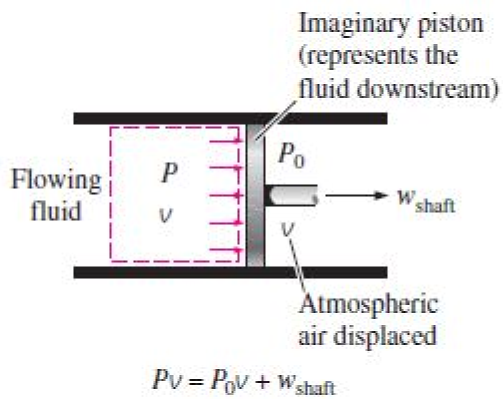


Figure 2.5 The exergy associated with flow energy is the useful work that would be delivered by an imaginary piston in the flow section.

Noting that the flow work is Pv and the work done against the atmosphere is P0v, the exergy associated with flow energy can be expressed as

$$x_{flow} = PV - P_0V = (P - P_0)V \quad (2.16)$$

Therefore, the exergy associated with flow energy is obtained by replacing the pressure P in the flow work relation by the pressure in excess of the atmospheric pressure, P - P0. Then the exergy of a flow stream is determined by simply adding the flow exergy relation above to the exergy relation in Eq. 2.12 for a nonflowing fluid,

$$x_{flowing\ fluid} = x_{nonflowing\ fluid} + x_{flow}$$

$$= (u - u_0) + P_0(V - V_0) - T_0(s - s_0) + \frac{V^2}{2} + gz + (P - P_0)V$$

$$= (u + PV) - (u_0 + P_0V_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

$$= (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \quad (2.17)$$

The final expression is called **flow** (or **stream**) **exergy**, and is denoted by ψ

Flow exergy: $$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \quad (2.18)$$

Then the exergy change of a fluid stream as it undergoes a process from state 1 to state 2 becomes

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1) + \frac{{V_2}^2 - {V_1}^2}{2} + g(z_2 - z_1)(2.19)$$

For fluid streams with negligible kinetic and potential energies, the kineticand potential energy terms drop out.

Note that the exergy change of a closed system or a fluid stream represents the maximum amount of useful work that can be done (or the minimum amount of useful work that needs to be supplied if it is negative) as the system changes from state 1 to state 2 in a specified environment, and represents the reversible work Wrev. It is independent of the type of process executed, the kind of system used, and the nature of energy interactions with the surroundings. Also note that the exergy of a closed system cannot be negative, but the exergy of a flow stream can at pressures below the environment pressure P0.

2.6 Exergy Transfer by HEAT, WORK, and MASS

Exergy, like energy, can be transferred to or from a system in three forms: heat, work, and mass flow. Exergy transfer is recognized at the system boundary as exergy crosses it, and it represents the exergy gained or lost by a system during a process. The only two forms of exergy interactions associated with a fixed mass or closed system are heat transfer and work.

2.6.1 Exergy by Heat Transfer, Q

The work potential of the energy transferred from a heat source at temperature T is the maximum work that can be obtained from that energy in an environment at temperature T0 and is equivalent to the work produced by a Carnot heat engine operating between the source and the environment. Therefore, the Carnot efficiency η_{II} =1 -T0 /T represents the fraction of energy of a heat source at temperature T that can be converted to work.

Heat is a form of disorganized energy, and thus only a portion of it can be converted to work, which is a form of organized energy (the second law). Work can always be produced from heat at a temperature above the environment temperature by transferring it to a heat engine that rejects the waste heat to the environment. Therefore, heat transfer is always accompanied by exergy transfer. Heat transfer Q at a location at thermodynamic temperature T is always accompanied by exergy transfer X_{heat} in the amount of

$$\textit{Exergy transfer by heat: } X_{heat} = \left(1 - \frac{T_0}{T}\right) Q \qquad (kJ) \quad (2.20)$$

Note that heat transfer through a finite temperature difference is irreversible, and some entropy is generated as a result. The entropy generation is alwaysaccompanied by exergy destruction, as illustrated in Figure 2.6. Also note that heat transfer Q at a location at temperature T is always accompanied by entropy transfer in the amount of Q/T and exergy transfer in the amount of (1 - T0 /T)Q.

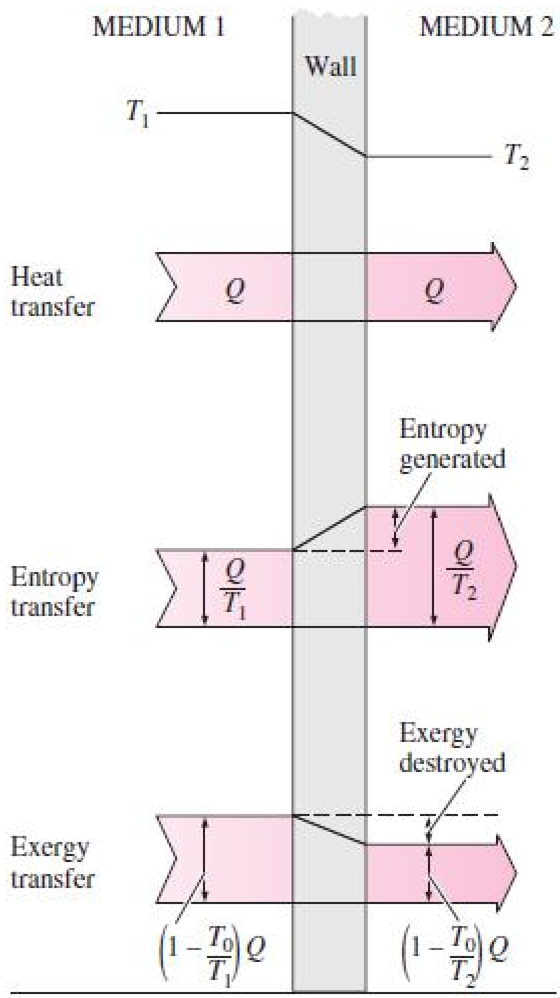


Figure 2.6 The transfer ad destruction of exergy during a heat transfer process through a finite temperature difference.

2.6.2 Exergy Transfer by Work, W

Exergy isthe useful work potential, and the exergy transfer by work can simply be expressed as

$$\textit{Exergy transfer by work:} X_{work} = \begin{cases} W - W_{surr} & (for\ boundary\ work) \\ W & (for\ other\ forms\ of\ work) \end{cases}$$

Where Wsurr =P0(V2 -V1), P0 is atmospheric pressure, and V1 and V2 are the initial and final volumes of the system. Therefore, the exergy transfer with work such as shaft work and electrical work is equal to the work W itself. In the case of a system that involves boundary work, such as a piston–cylinder device, the work done to push the atmospheric air out of the way during expansion cannot be transferred, and thus it must be subtracted. Also, during a compression process, part of the work is done by the atmospheric air, and thus less useful work is needed to be supplied from an external source.

2.6.3 Exergy Transfer by Mass, m

Mass contains exergy as well as energy and entropy, and the exergy, energy, and entropy contents of a system are proportional to mass. Also, the rates of exergy, entropy, and energy transport into or out of a system are proportionalto the mass flow rate. Mass flow is a mechanism to transport exergy, entropy,and energy into or out of a system. When mass in the amount of m entersor leaves a system, exergy in the amount of mc, where ψ =(h -h0)- T_0(s - s0) - V2/2 -gz, accompanies it. That is,

$$\textit{Exergy transfer by mass:} X_{mass} = m\psi (2.21)$$

Therefore, the exergy of a system increases by mψ when mass in the amount of m enters, and decreases by the same amount when the same amount of mass at the same state leaves the system.

2.7 Exergy of Reacting System

Any material that can be burned to release thermal energy is called a **fuel.**Most familiar fuels consist primarily of hydrogen and carbon. They arecalled **hydrocarbon fuels** and are denoted by the general formula C*n*H*m*.Hydrocarbon fuels exist in all phases, some examples being coal, gasoline,and natural gas.

To specify exergy of fuel, it is important to know enthalpy and entropy of fuel respect to standard state of environment.

$$Enthalpy = \overline{h^{\circ}}_f + \left(\overline{h} - \overline{h^{\circ}}\right)(KJ/Kmol) \quad (2.22)$$

Where thefirst term in Enthalpy equation $\overline{h^{\circ}}_f$ is enthalpy of formation, which is the enthalpy of a substance at a specified state due to its chemical composition. The enthalpy of formation of all stable elements (such as O2, N2, H2, and C) has a value of zero at the standard reference state of 25° C and 1 atm. Also, the term in the parentheses represents the sensible enthalpy relative to the standard reference state, which is the difference between $\overline{h}$(the sensible enthalpy at the specified state), and $\overline{h^{\circ}}$ (the sensible enthalpy at the standard reference state of 25°C and 1 atm). This definition enables us to use enthalpyvalues from tables regardless of the reference state used in their construction.

When evaluating the entropy of a component of an ideal-gas mixture, the temperature and the partial pressure of the componentshould be used. Note that the temperature of a component is the same as the temperature of the mixture, and the partial pressure of a component is equal to the mixture pressure multiplied by the mole fraction of the component.
Absolute entropy values at pressures other than $P_o = 1\, atm$ for any temperature T can be obtained from the ideal-gas entropy change relation written for an imaginary isothermal process between states (T,P_o) and (T,P).

$$\overline{s}(T,P) = \overline{s^{\circ}}(T,P_o) - R_u ln\frac{P}{P_o}\left(\frac{KJ}{Kmol\,.K}\right) \quad (2.23)$$

where $P_o = 1\ atm$, $P$is the partial pressure.

Both enthalpy and entropy of fuel discussed above is per moles. To have absolute enthalpy and entropy should be specify moles of reactants and moles of products of combustion. Thus enthalpy of reactants would be reactants' moles times reactants enthalpy and the same for products of combustion. Also, for entropy of reactants would be reactants' moles times reactants enthalpy and the same for products of combustion.

Eventually, exergy for reacting system can be calculated as equation below

$$\left.\begin{aligned}\psi_r &= \sum N_r \left(\overline{h^\circ}_f + \bar{h} - \overline{h^\circ} - T_o\bar{s}\right)_r (KJ) \\ \psi_P &= \sum N_r \left(\overline{h^\circ}_f + \bar{h} - \overline{h^\circ} - T_o\bar{s}\right)_P \ \ (KJ)\end{aligned}\right\} \quad (2.24)$$

2.8 Decrease of ExergyPrinciple

The conservation of energy principle is that energy cannot be created or destroyed during a process. Also the increase of entropy principle can be regarded as one of the statements of the second law, and the entropy can be created but cannot be destroyed.

Consider an isolated system. By definition, no heat, work, or mass can cross the boundaries of an isolated system, and thus there is no energy and entropy transfer. Then the energy and entropy balances for an isolated system can be expressed as

00

Energy balance:$E_{in} - E_{out} = \Delta E_{system} \rightarrow 0 = E_2 - E_1$

Entropy balance:$S_{in} - \overset{0}{S_{out}} + \overset{0}{S_{gen}} = \Delta S_{system} \rightarrow S_{gen} = S_2 - S_1$

Multiplying the second relation by T_0 and subtracting it from the first onegives

$$-T_0 S_{gen} = E_2 - E_1 - T_0(S_2 - S_1)$$

From Eq.2.13

Since $V_2 = V_1$ for an isolated system (it cannot involve any moving boundaryand thus any boundary work).

$$-T_0 S_{gen} = X_2 - X_1 \leq 0 (2.25)$$

Since T0 is the thermodynamic temperature of the environment and thus apositive quantity, $S_{gen} \geq 0$, and thus $T_0 S_{gen} \geq 0$. Then it is concluded that

$$\Delta X_{isolated} = (X_2 - X_1)_{isolated} \leq 0 (2.26)$$

This equation can be expressed as the exergy of an isolated system during a process always decreases or, in the limiting case of a reversible process, remains constant.

2.9 Exergy Destruction

Irreversibilities such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, nonquasi-equilibrium compression or expansion always generate entropy, and anything that generates entropy always destroys exergy. The exergy destroyed is proportional to the entropy generated, as can be seen from Eq.2.26, and is expressed as

$$X_{destroyed} = T_0 S_{gen} \geq 0 \qquad (2.27)$$

Note that exergy destroyed is a positive quantity for any actual process and becomes zero for a reversible process. Exergy destroyed represents the lost work potential and is also called the irreversibility or lost work.

Equations 8–32 and 8–33 for the decrease of exergy and the exergy destruction are applicable to any kind of system undergoing any kind of process since any system and its surroundings can be enclosed by a sufficiently large arbitrary boundary across which there is no heat, work, and mass transfer, and thus any system and its surroundings constitute an isolated system.

The decrease of exergy principle does not imply that the exergy of a system cannot increase. The exergy change of a system can be positive or negative during a process, but exergy destroyed cannot be negative. The decrease of exergy principle can be summarized as follows:

$$X_{destroyed} \begin{cases} = 0 & Reversible\ process \\ > 0 & Irreversible\ process \\ < 0 & Impossible\ process \end{cases}$$

This relation serves as an alternative criterion to determine whether aprocess is reversible, irreversible, or impossible.

2.10 Exergy Balance

2.10.1 Exergy Balance: CLOSED SYSTEMS

The nature of exergy is opposite to that of entropy in that exergy can be destroyed, but it cannot be created. Therefore, the exergy change of a system during a process is less than the exergy transfer by an amount equal to the exergy destroyed during the process within the system boundaries. Then the decrease of exergy principle can be expressed as (Figure 2.7)

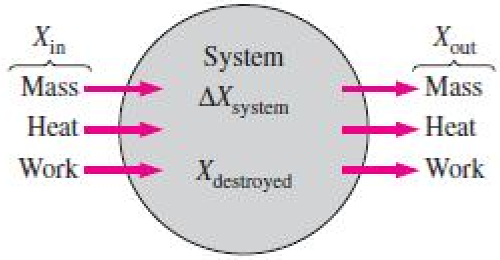


Figure 2.7 Mechanisms of exergy transfer.

$$\begin{pmatrix} Total \\ exergy \\ entering \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ leaving \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ leaving \end{pmatrix} = \begin{pmatrix} Change\ in\ the \\ total\ exergy \\ of\ the\ system \end{pmatrix}$$

Or

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{system} \qquad (2.28)$$

It is mentioned earlier that exergy can be transferred to or from a system by heat, work, and mass transfer. Then the exergy balance for any system undergoing any process can be expressed more explicitly as

$$General{:}\ \underbrace{X_{in} - X_{out}}_{\text{Net exergy transfer by heat, work, andmass}} \underbrace{- X_{destroyed}}_{\text{Exergy destruction}} = \underbrace{\Delta X_{system}}_{\text{Change in exergy}} (kJ) \qquad (2.29)$$

Or, in the rate form, as

General, rate form:

$$\underbrace{\dot{X}_{in} - \dot{X}_{out}}_{\text{Rate of net exergy transfer by heat, work, andmass}} \underbrace{- \dot{X}_{destroyed}}_{\text{Rate of Exergy destruction}} = \underbrace{dX_{system}/dt}_{\text{Rate of Change in exergy}} (kW) \qquad (2.30)$$

where the rates of exergy transfer by heat, work, and mass are expressed as $\dot{X}_{heat} = (1 - T_0/T)\dot{Q}, \dot{X}_{work} = \dot{W}_{useful}, and\ \dot{X}_{mass} = \dot{m}\psi$, respectively.

The exergy balance can also be expressed per unit mass as

General, unit-mass basis:

$$(x_{in} - x_{out}) - x_{destroyed} = \Delta x_{system} (kJ/kg) \qquad (2.31)$$

A closed system does not involve any mass flow and thus any exergy transfer associated with mass flow. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the exergy balance for a closed system can be expressed more explicitly as (Figure 2.8)

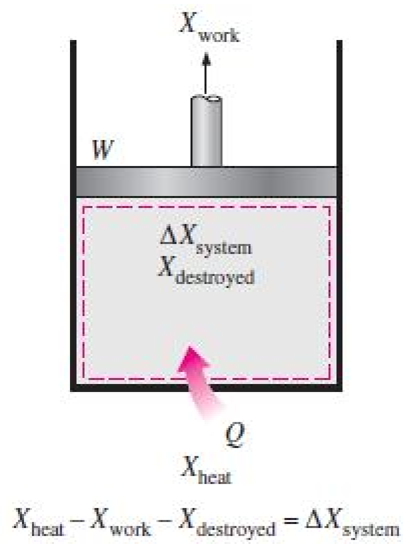


Figure 2.8 Exergy balance for a closed system when the direction of heat transfer is taken to be to the system and the direction of work from the system.

$$\textit{Closed system: } X_{heat} \quad - X_{work} - X_{destroyed} = \Delta X_{system} \quad (2.32)$$

The exergy balance relations presented above can be used to determine the reversible work Wrev by setting the exergy destruction term equal to zero. The work W in that case becomes the reversible work. That is, $W = W_{rev}$ when $X_{destroyed} = T_0 S_{gen} = 0$.

2.10.2 Exergy Balance: CONTROL VOLUMES

The exergy balance relations for control volumes differ from those for closed systems in that they involve one more mechanism of exergy transfer: mass flow across the boundaries. As mentioned earlier, mass possesses exergy as well as energy and entropy, and the amounts of these three extensive properties areproportional to the amount of mass (Fig. 8–42). Again taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general exergy balance relations (Eqs. 8–36 and 8–37) can be expressed for a control volume more explicitly as

$$X_{heat} - X_{work} + X_{mass,in} - X_{mass,out} - X_{destroyed} = (X_2 - X_1)_{CV} \quad (2.33)$$

Or

$$\sum\left(1 - \frac{T_o}{T_k}\right) Q_k - [W - P_0(V_2 - V_1)] + \sum_{in} m\psi - \sum_{out} m\psi - X_{destoryed} = (X_2 - X_1)_{CV} \quad (2.34)$$

It can also be expressed in the rate form as

$$\sum\left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \left(\dot{W} - P_0 \frac{dV_{CV}}{dt}\right) + \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi - \dot{X}_{destoryed} = \frac{dX_{CV}}{dt} \quad (2.35)$$

2.10.2.1 Exergy Balance for Steady-Flow Systems

Most control volumes encountered in practice such as turbines, compressors, nozzles, diffusers, heat exchangers, pipes, and ducts operate steadily, and thus they experience no changes in their mass, energy, entropy, and exergy contents as well as their volumes. Therefore, $dV_{CV}/dt = 0$ and $dX_{CV}/dt = 0$ for such systems, and the amount of exergy entering a steadyflow system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving plus the exergy destroyed. Then the rate form of the general exergy balance (Eq. 8–46) reduces for a **steady-flow process** to (Figure 2.9)

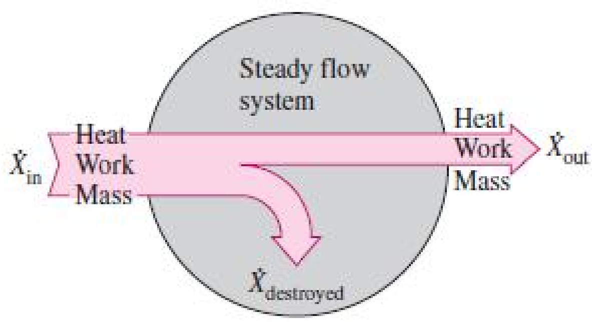


Figure 2.9 The exergy transfer to steady-flow system is equal to the exergy transfer from it plus the exergy destruction within the system.

Steady-flow:

$$\sum\left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \dot{W} + \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi - \dot{X}_{destoryed} = 0 \quad (2.36)$$

For a sigle-stream (one-inlet, one-exit) steady-flow device, the relation above further reduces to

Single-stream:

$$\sum\left(1-\frac{T_0}{T_k}\right)\dot{Q}_k-\dot{W}+\dot{m}(\psi_1-\psi_2)-\dot{X}_{destoryed}=0 \quad (2.37)$$

Where the subscripts 1 and 2 represent inlet and exit states, $\dot{m}$ is the mass flow rate, and the change in the flow exergy is given by Eq.2.19 as

$$\psi_1-\psi_2=(h_1-h_2)-T_0(s_1-s_2)+\frac{{V_2}^2-{V_1}^2}{2}+g(z_1-z_2)$$

Dividing Eq.2.34 by $\dot{m}$ gives the exergy balance on a unit-mass basis as

Per-unit mass:

$$\sum\left(1-\frac{T_0}{T_k}\right)q_k-w+(\psi_1-\psi_2)-x_{destroyed}=0 \quad (kJ/kg) \quad (2.38)$$

Where $q=\dot{Q}/\dot{m}$ and $w=\dot{W}/\dot{m}$ are the heat transfer and work done per unit mass of the working fluid, respectively.
For the case of an adiabatic single-stream device with no work interactions, the exergy balance relation further simplifies to $X_{destroyed}=m(\psi_1-\psi_2)$, which indicates that the specific exergy of the fluid must decrease as it flows through a work-free adiabatic device or remain the same ($\psi_2=\psi_1$) in the limiting case of a reversible process regardless of the change in other properties of the fluid.

2.10.2.2 Reversible Work, Wrev

The exergy balance relations presented above can be used to determine the reversible work *W*rev by setting the exergy destroyed equal to zero. The work *W* in that case becomes the reversible work. That is,

General: $\quad W=W_{rev} \quad when\ X_{destroyed}=0 \quad (2.39)$

For example, the reversible power for a single-stream steady-flow device is, from Eq.2.34,

Single stream: $\quad \dot{W}_{rev}=\dot{m}(\psi_1-\psi_2)+\sum\left(1-\frac{T_0}{T_k}\right)\dot{Q}_k \quad (2.40)$

which reduces for an adiabatic device to

Adiabatic, single stream: $\quad \dot{W}_{rev}=\dot{m}(\psi_1-\psi_2) \quad (2.41)$

Note that the exergy destroyed is zero only for a reversible process, and reversible work represents the maximum work output for work-producing devices such as turbines and the minimum work input for work-consuming devices such as compressors.

2.10.2.3 Second-Law Efficiency of Steady-Flow Devices, η_{II}

The second-law efficiency of various steady-flow devices can be determined from its general definition, η_{II} = (Exergy recovered)/ (Exergy supplied). When the changes in kinetic and potential energies are negligible, the second-law efficiency of an adiabatic turbine can be determined from

$$\eta_{II,turb} = \frac{w}{w_{rev}} = \frac{h_1-h_2}{\psi_1-\psi_2} \quad or \quad \eta_{II,turb} = 1 - \frac{T_0 s_{gen}}{\psi_1-\psi_2} \tag{2.42}$$

where $s_{gen} = s_2 - s_1$

For an *adiabatic compressor* with negligible kinetic and potential energies, the second-law efficiency becomes

$$\eta_{II,comp} = \frac{w_{rev,in}}{w_{in}} = \frac{\psi_2-\psi_1}{h_2-h_1} \quad or \quad \eta_{II,comp} = 1 - \frac{T_0 s_{gen}}{h_2-h_1} \tag{2.43}$$

Where again $s_{gen} = s_2 - s_1$

For an adiabatic *heat exchanger* with two unmixed fluid streams (Figure 2.10), the exergy supplied is the decrease in the exergy of the hot stream, and the exergy recovered is the increase in the exergy of the cold stream, provided that the cold stream is not at a lower temperature than the surroundings.

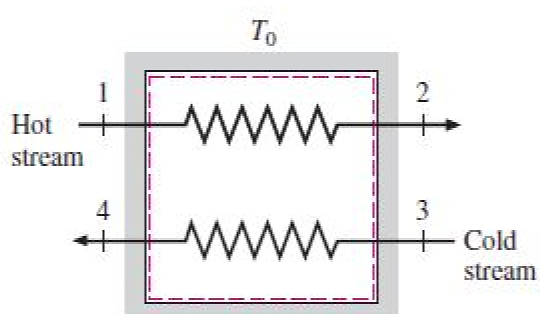


Figure 2.10 A heat exchanger with two unmixed fluid streams.

Then the second-law efficiency of the heat exchanger becomes

$$\eta_{II,HX} = \frac{\dot{m}_{cold}(\psi_4 - \psi_3)}{\dot{m}_{hot}(\psi_1 - \psi_2)} \quad or \quad \eta_{II,HX} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_{hot}(\psi_1 - \psi_2)} \quad (2.44)$$

Where $\quad \dot{S}_{gen} = \dot{m}_{hot}(s_2 - s_1) + \dot{m}_{cold}(s_4 - s_3)$

For an adiabatic *mixing chamber* where a hot stream 1 is mixed with a cold stream 2, forming a mixture 3, the exergy supplied is the sum of the exergies of the hot and cold streams, and the exergy recovered is the exergy of the mixture. Then the second-law efficiency of the mixing chamber becomes

$$\eta_{II,mix} = \frac{\dot{m}_3 \psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad or \quad \eta_{II,mix} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad (2.45)$$

Where $\quad \dot{m}_3 = \dot{m}_1 + \dot{m}_2 \;\; and \; \dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1$.

Chapter Three: Pinch Technology

3.1 Introduction

Pinch technology is a complete methodology derived from simple scientific principles by which it ispossible to design new plants with reduced energy and capital costs as well as where the existing processes require modification to improve performance

Pinch analysis is a method that aims at identifying the heat recovery opportunities by heat exchange in complex thermal processes. The pinch analysis has been mainly developed in the early 70's by Linnhoff and co-workers who developed a graphical method to calculate the minimum energy requirement of a process and design the heat recovery exchanger network and by Grossmann and co-workers who developed a mathematical programming framework for the design of heat exchanger networks. The graphical tools and the mathematical methods have converged to propose nowadays tools and methods that help in the identification of energy recovery by heat exchange and energy savings in site wide complex systems. The pinch analysis has been first developed for studying energy savings in the chemical process industry. Since then, the pinch analysis has been applied in the other industrial sectors where thermal operations occur like food, cement, pulp and paper, metallurgy, power plants, urban systems, etc.

The power of the pinch analysis stands mainly in its ability of offer a holistic analysis of the possible heat exchanges in a large and integrated system.

3.1.1Pinch Analysis

The first key concept of pinch analysis is **setting energy targets**. "Targets" for energy reduction have been a key part of energy monitoring schemes for many years. Typically, a reduction in plant energy consumption of 10% per year is demanded. A 10% reduction may be very easy on a badly designed and operated plant where there are many opportunities for energy saving, and a much higher target would be appropriate. However, on a "good" plant, where continuous improvement has taken place over the years, a further 10% may be impossible to achieve.
As an illustrative example, consider Figure 3.1(a) shows an outline flowsheet representing a traditional design for the front end of a specialty chemicals process. Six heat transfer "units" (i.e. heaters, coolers and exchangers) are used and the energy requirements are 1,722 kW for heating and 654 kW for cooling. Figure 3.1(b) shows an alternative design which

was generated by Linnhoff *et al.* (1979) using pinch analysis techniques (then newly developed) for energy targeting and network integration. The alternative flowsheet uses only four heat transfer “units” and the utility heating load is reduced by about 40%with cooling no longer required. The design is as safe and as operable as the traditional one. It is simply better.Results like this made pinch analysis a “hot topic” soon after it was introduced. Benefits were found from improving the integration of processes, often developing simpler, more elegant heat recovery networks, without requiring advanced unit operation technology.Targets obtained by pinch analysis are different. They are absolute thermodynamic targets, showing what the process is inherently capable of achieving if the heat recovery, heating and cooling systems are correctly designed. In the case of the flowsheet in Figure 3.1 the targeting process shows that only 1,068 kW of external heating should be needed, and no external cooling at all. This gives the incentive to find a heat exchanger network which achieves these targets.

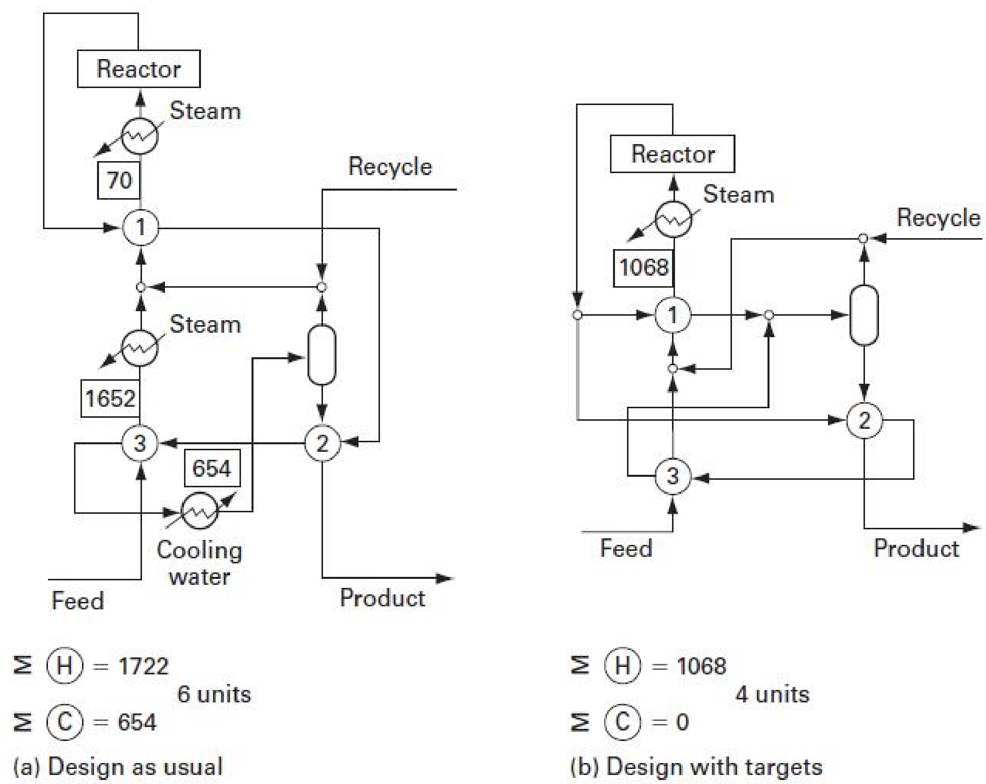


Figure 3.1 outline flowsheets for the front end of a specialty chemicals process

3.2 Key concepts of pinch analysis

In this section, the key concepts of pinch analysis are presented, showing how it is possible to set energy targets and achieve them with a network of heat exchangers.

3.2.1 Heat recovery and heat exchange

3.2.1.1 Heat exchange

Heat exchange is a heat transfer from hot stream to another cold stream. The stream is defined as any flow which requires to be heated or cooled, but does not change in composition. For more explanation, consider the simple process shown in Figure 3.2. Liquid is supplied to the reactor and needs to be heated from near-ambient temperature to the operating temperature of the reactor. Conversely, a hot liquid product from the separation system needs to be cooled down to a lower temperature. There is also an additional unheated make-up stream to the reactor. The feed which starts cold and needs to be heated up is known as a cold stream. Conversely, the hot product which must be cooled down is called a hot stream. Conversely, the reaction process is not a stream, because it involves a change in chemical composition; and the make-up flow is not a stream, because it is not heated or cooled.

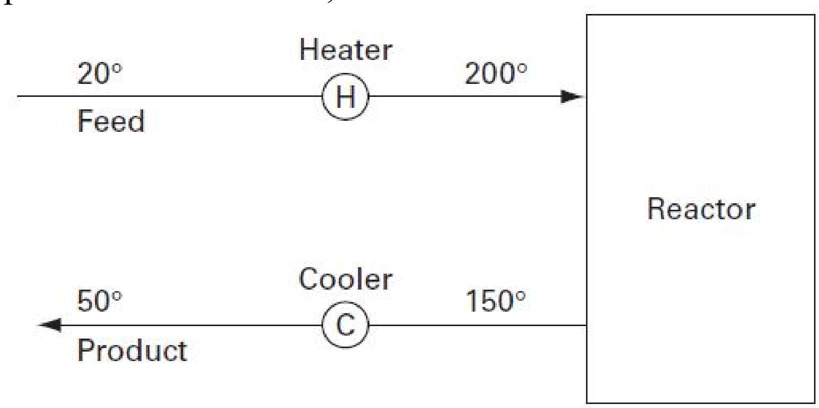


Figure 3.2 Simple process flowsheet

To perform the heating and cooling, a steam heater could be placed on the cold stream, and a water cooler on the hot stream. The flows are as given in Table 3.1.Clearly, it is needed to supply 180 kW of steam heating and 180 kW of water cooling to operate the process.

Table 3.1 Data for simple two-stream example

	Mass flowrate W *(kg/s)*	*Specific heat capacity* C_P *(kJ/kgK)*	*Heat capacity flowrate* CP *(kW/K)*	*Initial (supply) temperature* T_S *(°C)*	*Final (target) temperature* T_T *(°C)*	*Heat load* H *(kW)*
Cold stream	0.25	4	1.0	20	200	−180
Hot stream	0.4	4.5	1.8	150	50	+180

if some heat can be recovered from the hot stream and use it to heat the cold stream in a heat exchanger, less steam will be needed and water to satisfy the remaining duties. The flowsheet will then be as in Figure 3.3. Ideally, of course, all 180 kW would be liked to be recovered in the hot stream to heat the cold stream. However, this is not possible because of temperature limitations. By theSecond Law of Thermodynamics, a hot stream at 150°C can't be used to heat a cold stream at 200°C.

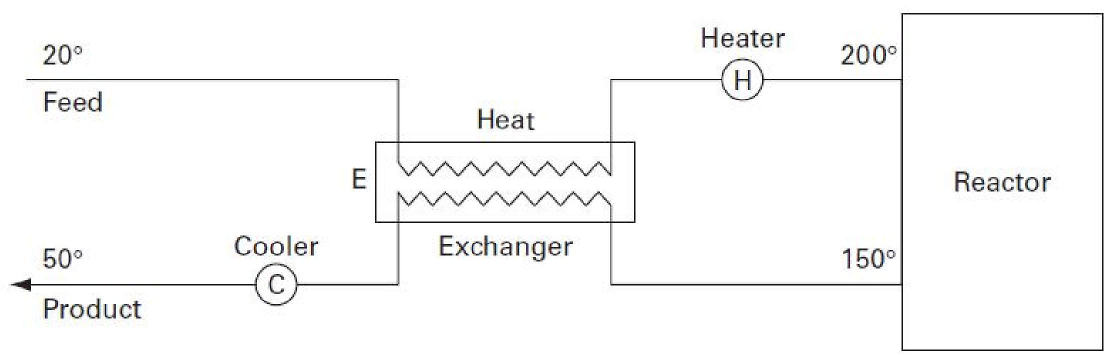


Figure 3.3 Simple process flowsheet with heat exchange

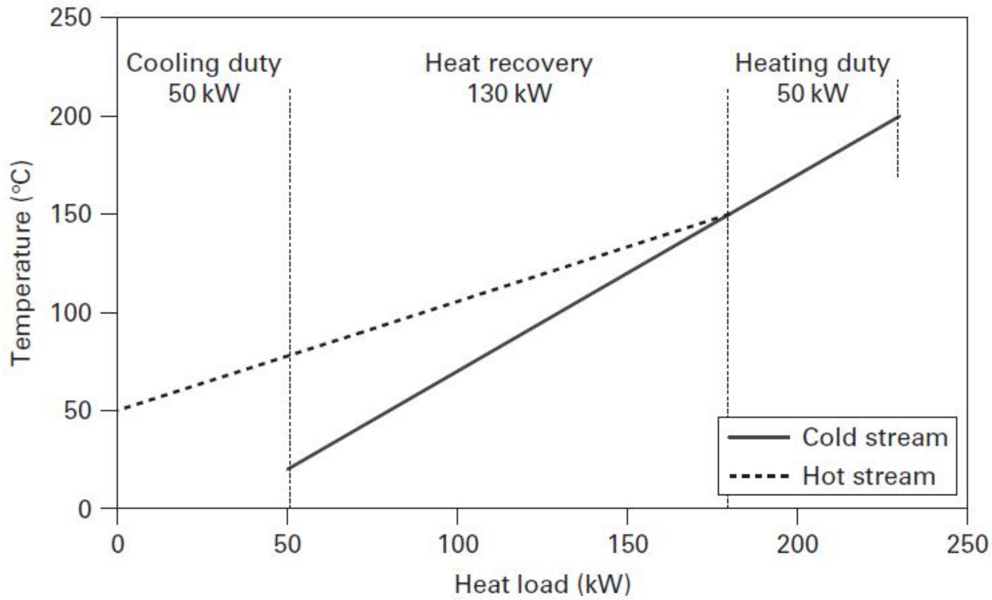


Figure 3.4 Streams plotted on temperature/enthalpy (T/H) diagram with $\Delta T_{min} = 0$

3.2.1.2 Temperature –Enthalpy Diagram

A helpful method of visualization is the temperature–heat content diagram, as illustrated in Figure 3.4. The heat content *H* of a stream (kW) is frequently called its enthalpy; this should not be confused with the thermodynamic term, specificenthalpy (kJ/kg). Differential heat flow dQ, when added to a process stream, will increase its enthalpy (H) by $CPdT$, where:

CP = "heat capacity flowrate" (KW/K) = mass flow $\dot{m}$ (Kg/s) $\times$ specific heat CP(KJ/KgK).
dT = differential temperature change

Hence, with CP assumed constant, for a stream requiring heating ("cold" stream) from a "supply temperature" (T_s) to a " target temperature" (T_T), the total heat added will be equal to the stream enthalpy change, i.e.

$$Q = \int_{T_s}^{T_T} CPdT = CP(T_T - T_s) = \Delta H \qquad (3.1)$$

and the slope of the line representing the stream is:

$$\frac{dT}{dQ} = \frac{1}{CP} \qquad (3.2)$$

The *T*/*H* diagram can be used to represent heat exchange, because of a very useful feature. Namely, since enthalpy *changes* of streams is only interesting, a given stream can be plotted anywhere on the enthalpy axis.

Provided it has the same slope and runs between the same supply and target temperatures, then wherever it is drawn on the H-axis, it represents the same stream.

Figure 3.4 shows the hot and cold streams for this example plotted on the *T*/*H* diagram. Note that the hot stream is represented by the line with the arrowheadpointing to the left, and the cold stream *vice versa*. For feasible heat exchangebetween the two, the hot stream must at all points be hotter than the cold stream, so it should be plotted above the cold stream. Figure 3.4 represents a limiting case; the hot stream cannot be moved further to the right, to give greater heat recovery, because the temperature difference between hot and cold streams at the cold end of the exchanger is already zero. This means that, in this example, the balance of heat required by the cold stream above 150°C (i.e. 50 kW) has to be made up from steam heating. Conversely, although 130 kW can be used for heat exchange, 50kW of heat available in the hot stream has to be rejected to cooling water. However, this is not a practically achievable situation, as a zero temperature difference would require an infinitely large heat exchanger.

In Figure 3.5 the cold stream is shown shifted on the H-axis relative to the hot stream so that the minimum temperature difference, ΔTmin is no longer zero, but positive and finite (in this case 20°C). The effect of this shift is to increase the utility heating and cooling by equal amounts and reduce the load on the exchanger by the same amount – here 20 kW – so that 70 kW of external heating and cooling is required.

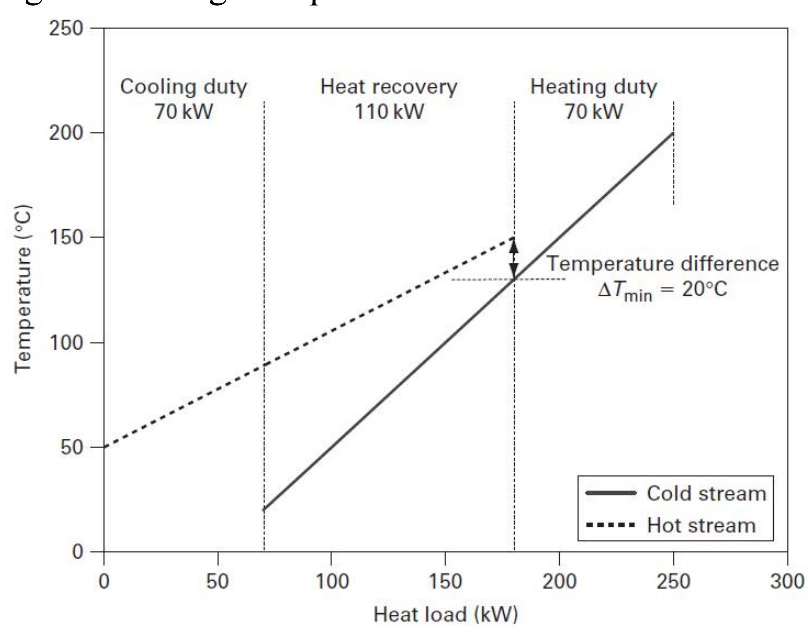


Figure 3.5 T/H diagram with $\Delta T_{min} = 20$℃

This arrangement is now practical because the ΔTmin is non-zero. Clearly, further shifting implies larger ΔTmin values and larger utility consumptions.

From this analysis, two basic facts emerge. Firstly, there is a correlation between the value of ΔTmin in the exchanger and the total utility load on the system. This means that if a value of ΔTmin is chosen, an **energy target** is taken place for how much heating and cooling should be used if heat exchanger designed correctly.
Secondly, if the hot utility load is increased by any value α, the cold utility is increased by α as well. As the stream heat loads are constant, this also means that the heat exchanged falls by α.
It will be rightly pointed out that a method confined to a single hot and cold stream is of little practical use. What is needed is a methodology to apply this to real multi-stream processes. The composite curves give a way of doing so.

3.2.1.3 Composite curves

To handle multiple streams, the heat loads or heat capacity flowrates of all streams existing added together over any given temperature range. Thus, a single composite of all hot streams and a single composite of all coldstreams can be produced in the *T*/*H* diagram, and handled in just the same way as the two-stream problem.

In Figure 3.6(a) three hot streams are plotted separately, with their supply and target temperatures defining a series of "interval" temperatures T_1–T_5. Between T_1 and T_2, only stream B exists, and so the heat available in this interval is given by$CP_B(T1 - T2)$. However between $T2$ and $T3$ all three streams exist and so the heat available in this interval is $(CP_A+CP_B+CP_C)(T2 - T3)$. A series of values of ΔH for each interval can be obtained in this way, and the result re-plotted against the interval temperatures as shown in Figure 3.6 (b). The resulting *T*/*H* plot is a single curve representing all thehot streams, known as the **hot composite curve**. A similar procedure gives a **cold composite curve** of all the cold streams in a problem.The overlap between the composite curves represents the maximum amount of heat recovery possible within the process. The "overshoot" at the bottom of the hot composite represents the minimum amount of external cooling required and the "overshoot" at the top of the cold composite represents the minimum amount of external heating.

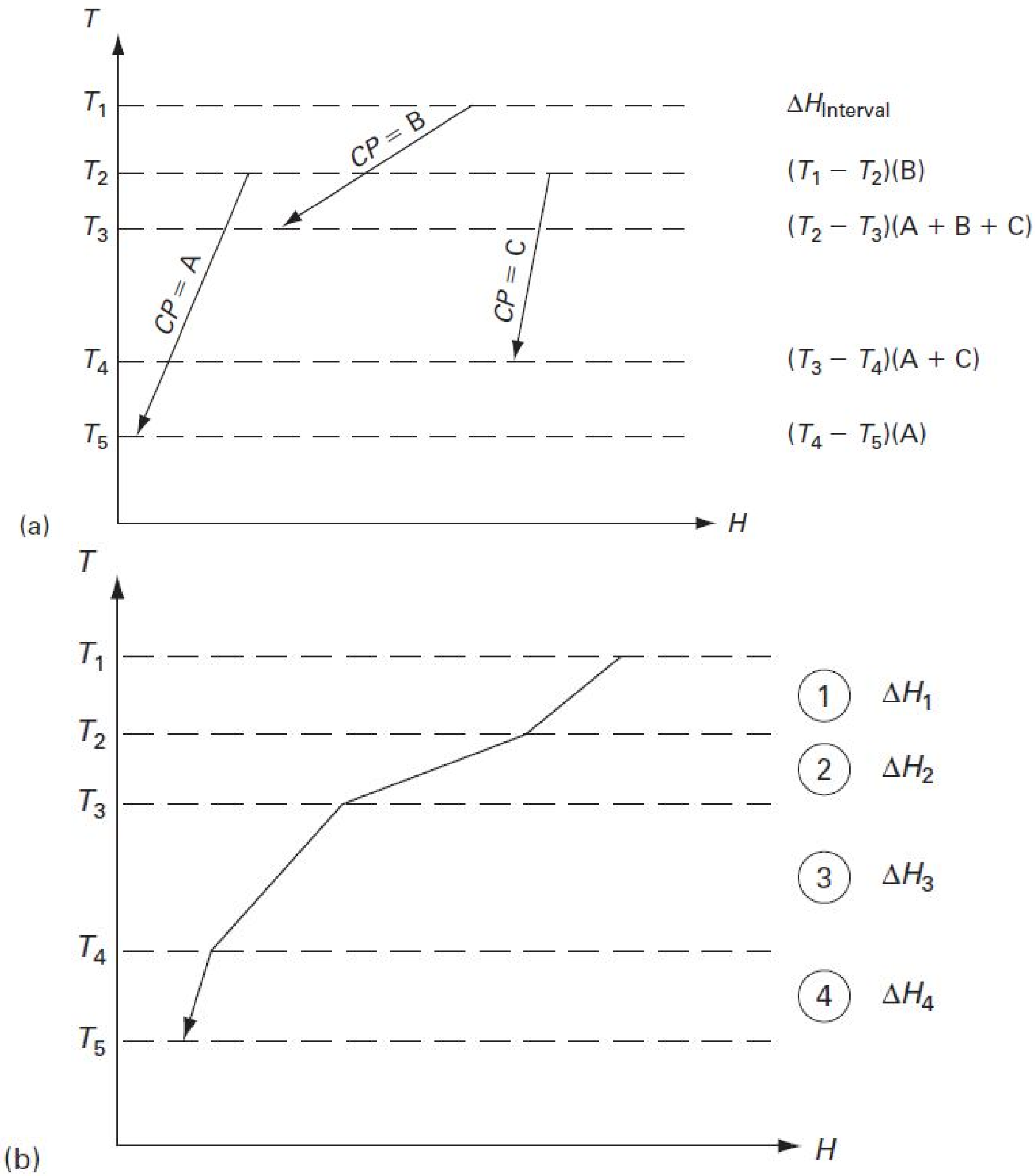


Figure 3.6 Formation of the hot composite curve

Table 3.2 Data for four-stream example

Stream number and type	CP (kW/K)	Actual temperatures		Shifted temperatures	
		T_S (°C)	T_T (°C)	S_S (°C)	S_T (°C)
1. Cold	2	20°	135°	25°	140°
2. Hot	3	170°	60°	165°	55°
3. Cold	4	80°	140°	85°	145°
4. Hot	1.5	150°	30°	145°	25°

ΔT_{min} = 10°C.

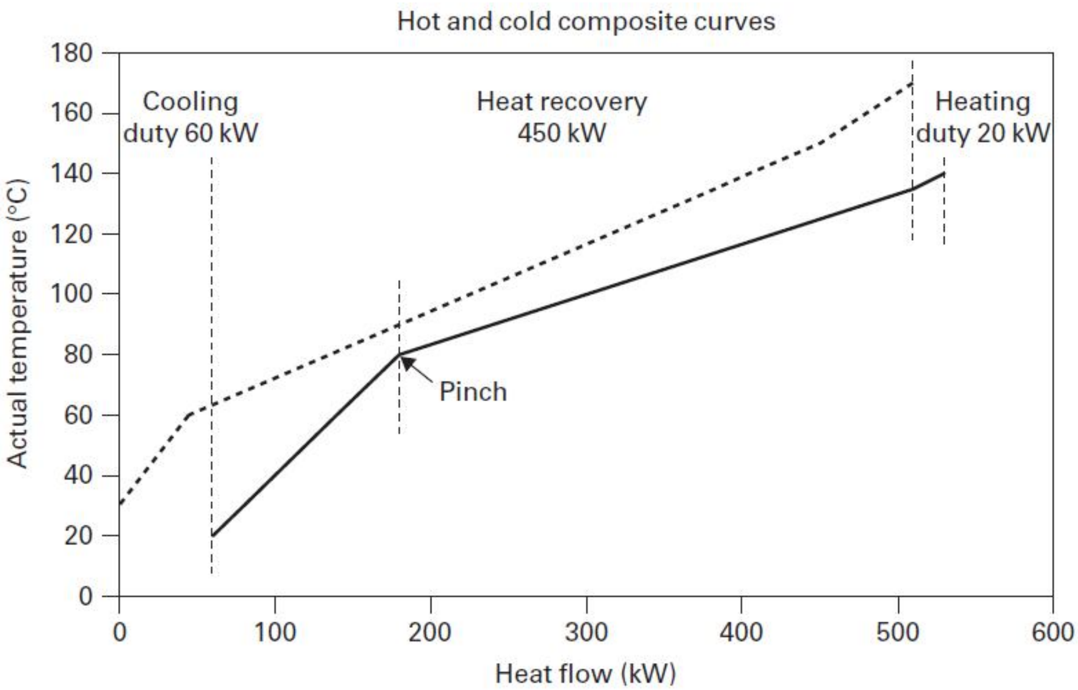


Figure 3.7 Composite curves for four-stream problem

Figure 3.7 shows a typical pair of composite curves, for the four-stream problem given in Table 3.2. Shifting of the curves leads tobehaviour similar to that shown by the two-stream problem. Now, though, the "kinked" nature of the composites means that Δ*T*min can occur anywhere in the interchange region and not just at one end. *For a given value of* Δ*T*min, *the utility quantitiespredicted are the minima required to solve the heat recovery problem.* Note that although there are many streams in the problem, in general Δ*T*min occurs at only one point of closest approach, which is called the **pinch**. This means that it is possible to design a network which uses the minimum utility requirements, where *only the heat exchangers at the pinch* need to operate at Δ*T* values down to Δ*T*min.

3.2.1.4 A targeting procedure (Problem Table)

In principle, the "composite curves" described in the previous sub section could beused for obtaining energy targets at given values of Δ*T*min. However, it wouldrequire a "graph paper and scissors" approach (for sliding the graphs relative toone another) which would be messy and imprecise. Instead, an algorithm is used for setting the targets algebraically, which called the "Problem Table" method.

In the description of the construction of composite curves (Figure 3.7), it wasshown how enthalpy balance intervals were set up based on stream supply andtarget temperatures. The same can be done for hot and cold streams together, toallow for the maximum possible amount of heat exchange within each temperature interval. The only modification needed is to ensure that within any interval, hotstreams and cold streams are at least

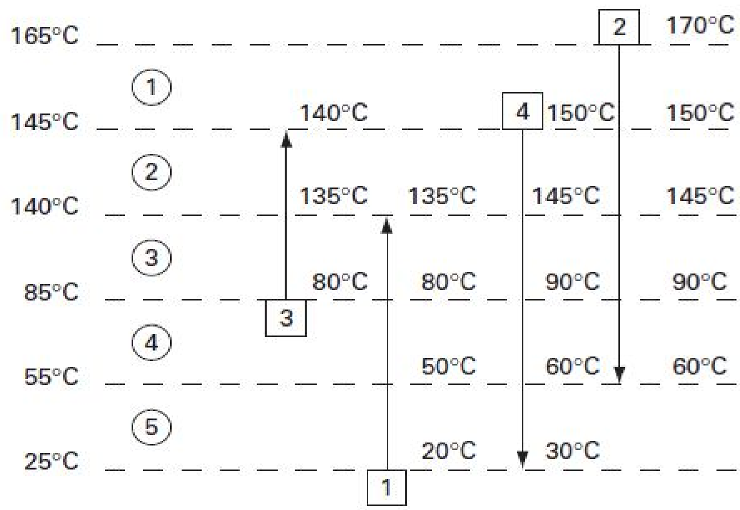


Figure 3.8 Streams and temperature intervals

Δ*T*min apart. This is done by using **shiftedtemperatures**, which are set at 1⁄2Δ*T*min (5°C in this example) *below* hot streamtemperatures and 1⁄2Δ*T*min *above* cold stream temperatures. Table 3.2 shows thedata for the four-stream problem including shifted temperatures. Figure 3.8 showsthe streams in a schematic representation with a vertical temperature scale, withinterval boundaries superimposed (as shifted temperatures). So for example ininterval number 2, between shifted temperatures 145°C and 140°C, streams 2 and4 (the hot streams) run from 150°C to 145°C, and stream 3 (the cold stream) from135°C to 140°C. Setting up the intervals in this way *guarantees* that full heat interchangewithin any interval is possible. Hence, each interval will have either a netsurplus or net deficit of heat as dictated by enthalpy balance, *but never both*.Knowing the stream population in each interval (from Figure 3.8), enthalpy balancescan easily be calculated for each according to:

$$\Delta H_i = (S_i - S_{i+1}) (\sum CP_H - \sum CP_C)_i \quad (3.3)$$

for any interval *i*.

Table 3.3 Temperature intervals and heat loads for four-stream problem

	Interval number i	$S_i - S_{i+1}$ (°C)	$\sum CP_{HOT} - \sum CP_{COLD}$ (kW/°C)	ΔH_i (kW)	*Surplus or deficit*
S_1 = 165°C					
	1	20	+3.0	+60	Surplus
S_2 = 145°C					
	2	5	+0.5	+2.5	Surplus
S_3 = 140°C					
	3	55	−1.5	−82.5	Deficit
S_4 = 85°C					
	4	30	+2.5	+75	Surplus
S_5 = 55°C					
	5	30	−0.5	−15	Deficit
S_6 = 25°C					

The results are shown in Table 3.3, and the last column indicateswhether an interval is in heat surplus or heat deficit. It would therefore be possibleto produce a feasible network design based on the assumption that all "surplus" intervals rejected heat to cold utility, and all "deficit" intervals took heat from hot utility. However, this would not be very sensible, because it would involve rejectingand accepting heat at inappropriate temperatures.

However,a key feature of the temperature intervals is exploited. Namely,*any heat available in interval* i *is hot enough to supply any duty in interval*(i + 1).This is shown in Figure 3.9, where intervals 1 and 2 are used as an illustration.Instead of sending the 60 kW of surplus heat from interval 1 into cold utility, it canbe sent down into interval 2. It is therefore possible to set up a heat "cascade" asshown in Figure 3.10(a).Assuming that no heat is supplied to the hottest interval 1from hot utility, then the surplus of 60 kW from interval 1 is cascaded into interval 2.There it joins the 2.5 kW surplus from interval 2, making 62.5 kW to cascade intointerval 3.Interval 3 has a 82.5 kW deficit, hence after accepting the 62.5 kW it canbe regarded as passing on a 20 kW deficit to interval 4. Interval 4 has a 75 kW surplusand so passes on a 55 kW surplus to interval 5. Finally, the 15 kW deficit ininterval 5 means that 40 kW is the final cascaded energy to cold utility. This in fact isthe net enthalpy balance on the whole problem (i.e. cold utility will always exceedhot utility by 40kW, whatever their individual values).

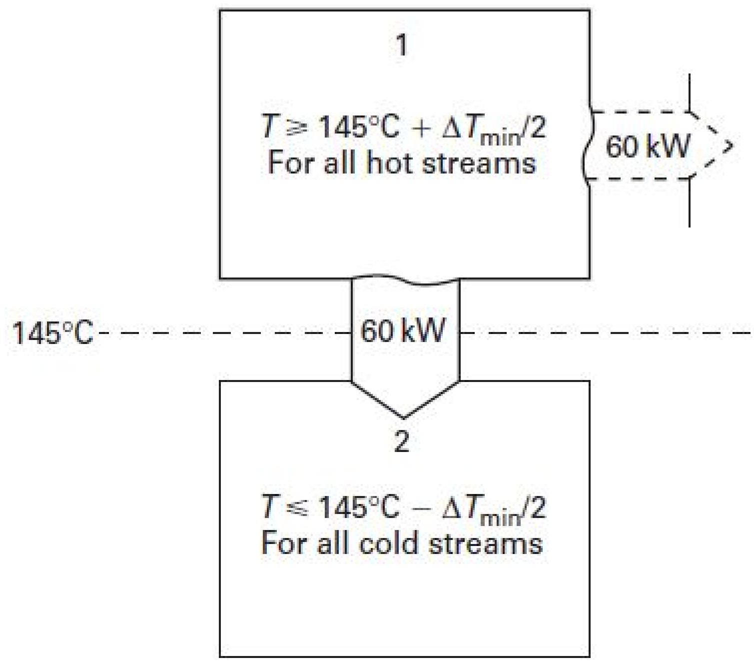


Figure 3.9 Use of a heat surplus from an interval

Looking back at the heatflows between intervals in Figure 3.10(a), clearly the negative flow of 20 kW betweenintervals 3 and 4 is thermodynamically infeasible. To make it just feasible (i.e. equalto zero), 20 kW of heat must be added from hot utility as shown in Figure 3.10(b),and cascaded right through the system. By enthalpy balance this means that allflows are increased by 20kW. The net result of this operation is that the minimumutilities requirements have been predicted (i.e. 20 kW hot and 60 kW cold).

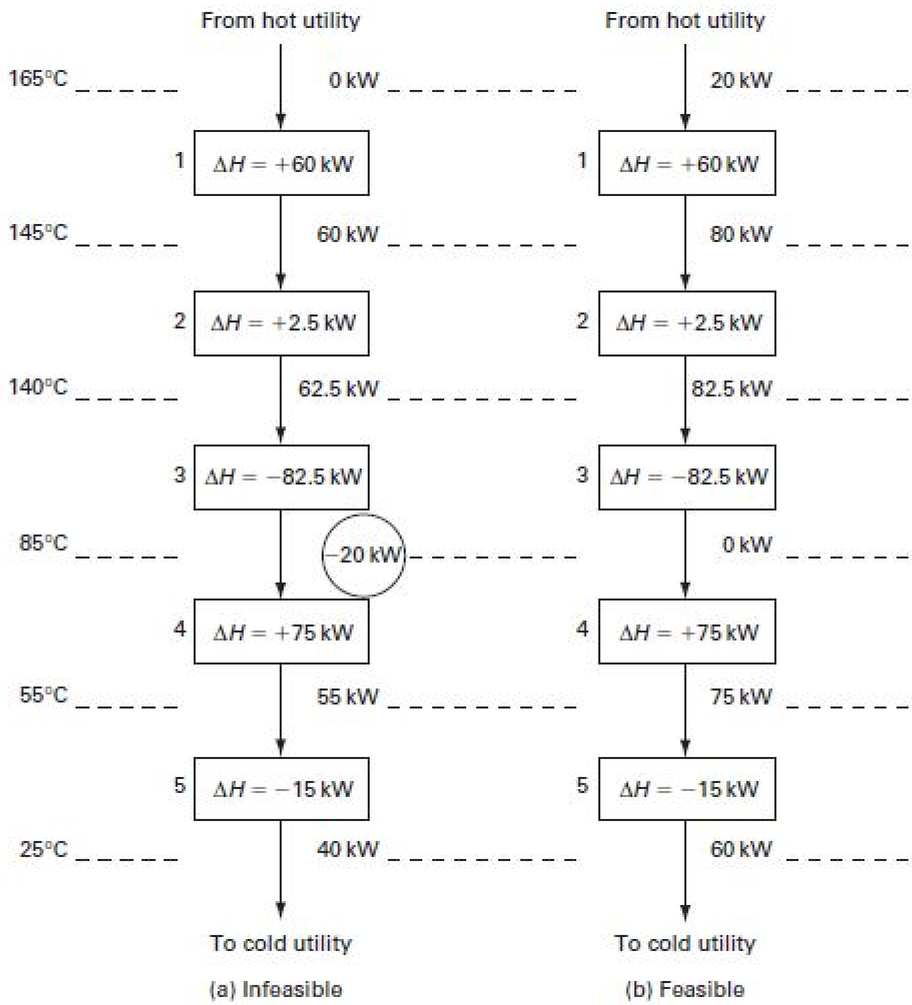


Figure 3.10 Infeasible and feasible heat cascades

Furthermore,the position of the pinch has been located. This is at the interval boundarywith a shifted temperature of 85°C (i.e. hot streams at 90°C and cold at 80°C) wherethe heat flow is zero.
Comparing the results obtained by this approach to the results from the compositecurves, as shown in Figure 3.7. The same information is obtained, but the ProblemTable provides a simple framework for numerical analysis. For simple problems itcan be quickly evaluated by hand. For larger problems, it is easily implemented as aspreadsheet or other computer software. It can also be adapted for the case wherethe value of ΔTmin allowed depends on the streams matched, and is not simply a"global" value. Finally it can be adapted to cover othercaseswhere simplifying assumptions (e.g. CP = constapnt) are invalid.
The total heat recovered by heat exchange is found by adding the heat loads for all the hot streams and all the cold streams – 510 and 470 kWh,

respectively. Subtracting the cold and hot utility targets (60 and 20 kWh) from these values gives the total heat recovery, 450 kWh, by two separate routes. The cold utility target minus the hot utility target should equal the bottom line of the infeasible heat cascade, which is 40 kWh. These calculations provide useful cross-checks that the stream data and heat cascades have been evaluated correctly.

There are in fact three possible ways of moving the hot and cold composite curves closer together by ΔTmin, so that they touch at the pinch. This may be achieved in three ways:

1. Express all temperatures in terms of hot stream temperatures and increase allcold stream temperatures by ΔTmin.
2. Express all temperatures in terms of cold stream temperatures and reduce all hotstream temperatures by ΔTmin.
3. Use the shifted temperatures, which are a mean value; all hot stream temperatures are reduced by Δ Tmin/2 and all cold stream temperatures are increased by ΔTmin/2.

Approach 3 has been the most commonly adopted, although approach 1 has alsobeen used significantly. In this research, shifted temperatures 3 will be used.

3.2.1.5 Grand Composite Curve and Shifted Composite Curves

If the composite curves are re-plotted on axes of shifted temperature, the **shifted composite curves**are obtaine, Figure 3.11. The shifted curves just touch at the pinch temperature, and show even more clearly than the composite curves that the pinch divides the process into two.

Now consider what happens at any shifted temperature *S*. The heat flow of allthe hot streams Q_H, relative to that at the pinch Q_{HP} (fixed), is ΔQ_H. Likewise theheat flow of all cold streams relative to that at the pinch is ΔQ_C. There is an imbalancewhich must be supplied by utilities – external heating and cooling. Above thepinch, $\Delta Q_C > \Delta Q_H$and the difference must be supplied by hot utility. Likewise,below the pinch $\Delta Q_H > \Delta Q_C$ and the excess heat is removed by cold utility.

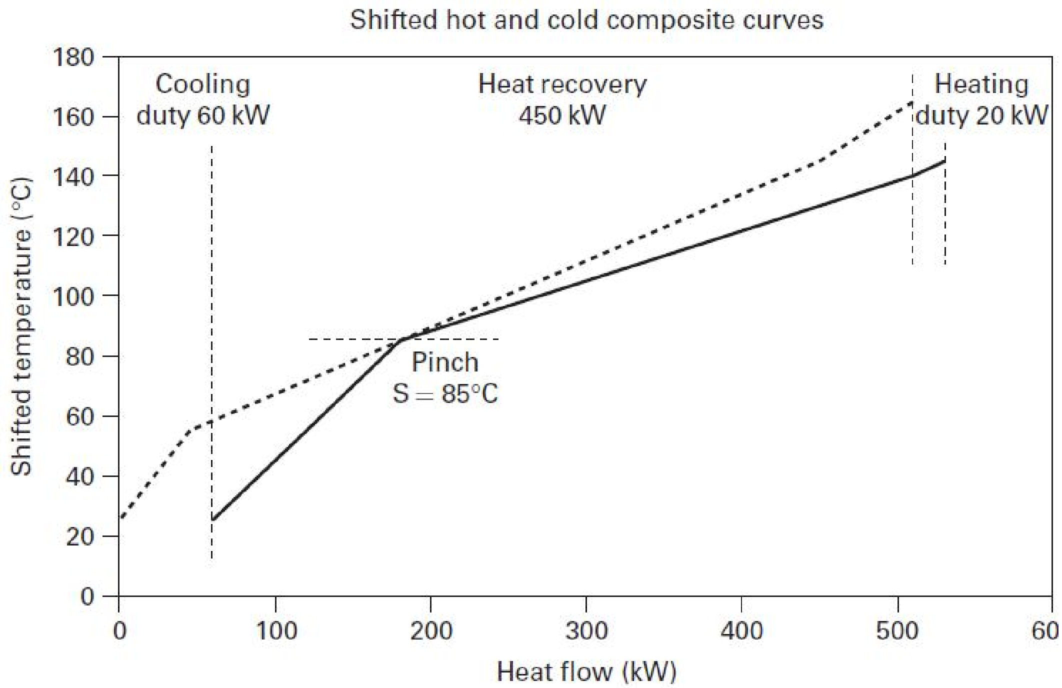


Figure 3.11 Shifted composite curves

Hence, knowing the shifted composite curves, the minimum amountof heating or cooling that needs to be supplied can be found at any given temperature. A graph ofnet heat flow (utility requirement) against shifted temperature can then easily be plotted. This is known as the **grand composite curve** (hereafter abbreviated to**GCC**). Itrepresents the difference between the heat available from the hot streams and theheat required by the cold streams, relative to the pinch, at a given shifted temperature.Thus, the GCC is a plot of the net heat flow against the shifted (interval) temperature,which is simply a graphical plot of the Problem Table (heat cascade).

The GCC for the four-stream inFigure 3.12 The values of net heat flow at the top and bottom end are the heat supplied to and removed from the cascade, and thus the hot and cold utility targets are told. But not only does the GCC show how much net heating and cooling is required, it also specify what temperatures it is needed at. There is no need to supply all the utility heating at the highest temperature interval; much of it can, if desired, be supplied at lower temperatures. The pinch is also easily visualized, being the point where net heat flow is zero and the GCC touches the axis. Moreover, it can be seen whether the pinch occurs in the middle of the temperature range or at one end (a “threshold” problem), and identify other regions of low net heatflow, or even double or multiple pinches.

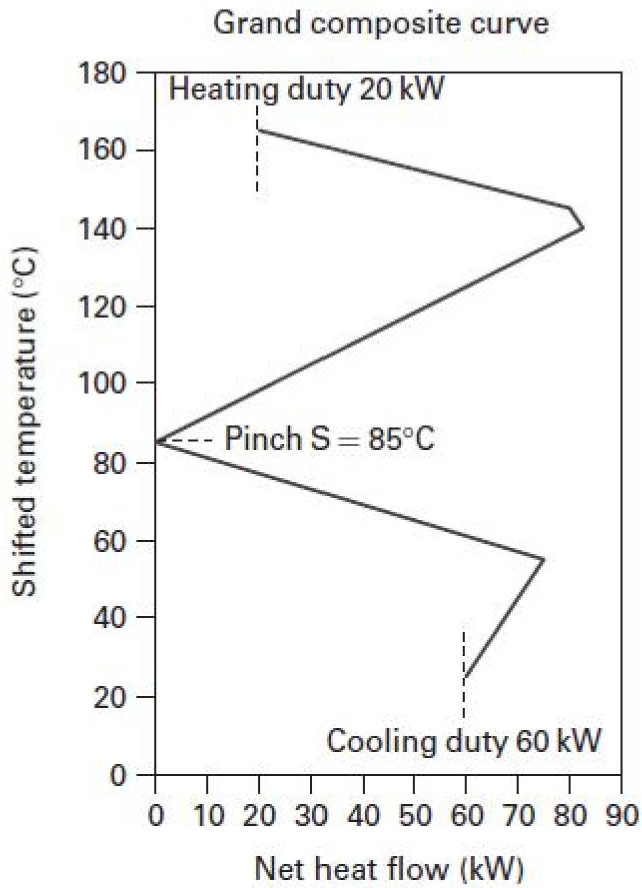


Figure 3.12 GCC for core example

Energy targeting can also be used to settle quickly disputes along the lines of "to integrate, or not to integrate?" Processes often fall into distinct sections ("A" and "B") by reason of layout or operability considerations. The question is often "can significant savings be made by cross integration?" The Problem Table algorithm can be applied to areas A and B separately, and then to all the streams in A and B together. The results of the analysis will quickly settle the question. For example, if the answer is:

- ✓ A alone: 10% savings in total fuel bill possible
- ✓ B alone: 5% savings in total fuel bill possible
- ✓ A and B together: 30% savings possible

Then there is a 15% energy incentive for cross-integrating areas A and B.

To summarize this section on energy targeting, it could be concluded that the composite curves give conceptual understanding of how energy targets can be obtained. And the Problem Table and its graphical representation, the

GCC, give the same results (including the pinch location) more easily. Also, Energy targeting is a powerful design and "process integration" aid.

3.2.2 The pinch and its significance

Figure 3.13(a) shows the composite curves for a multi-stream problem dissected at the pinch. "Above" the pinch (i.e. in the region to the right) the hot composite transfers all its heat into the cold composite, leaving utility *heating only* required. The region above the pinch is therefore a *net heat sink*, with heat flowing into it but no heat flowing out. It involves heat exchange and hot utility, but *no coldutility*. Conversely below the pinch *cooling only* is required and the region is therefore a *net heat source*, requiring heat exchange and cold utility but *no hot utility*. The problem therefore falls into two thermodynamically distinct regions, as indicated by the enthalpy balance envelopes in Figure 3.13 (b). Heat Q_{Hmin} flows into the problem above the pinch and Q_{Cmin} out of the problem below, but the heat flow across the pinch is zero. This result was observed in the description of the Problem Table algorithm. The way in which the pinch divides the process into two is even more clearly seen from the shifted composite curves (Figure 3.11).

It follows that any network design that transfers heat α across the pinch must, by overall enthalpy balance, require α more than minimum from hot and cold utilities, as shown in Figure 3.13 (c). As a corollary, any utility cooling α above the pinch must incur extra hot utility α, and *vice versa* below the pinch. This gives three **golden rules** for the designer wishing to produce a design achieving minimum utility targets:

- ✓ Don't transfer heat across the pinch.
- ✓ Don't use cold utilities above the pinch.
- ✓ Don't use hot utilities below the pinch.

Conversely, if a process is using more energy than its thermodynamic targets, it must be due to one or more of the golden rules being broken.

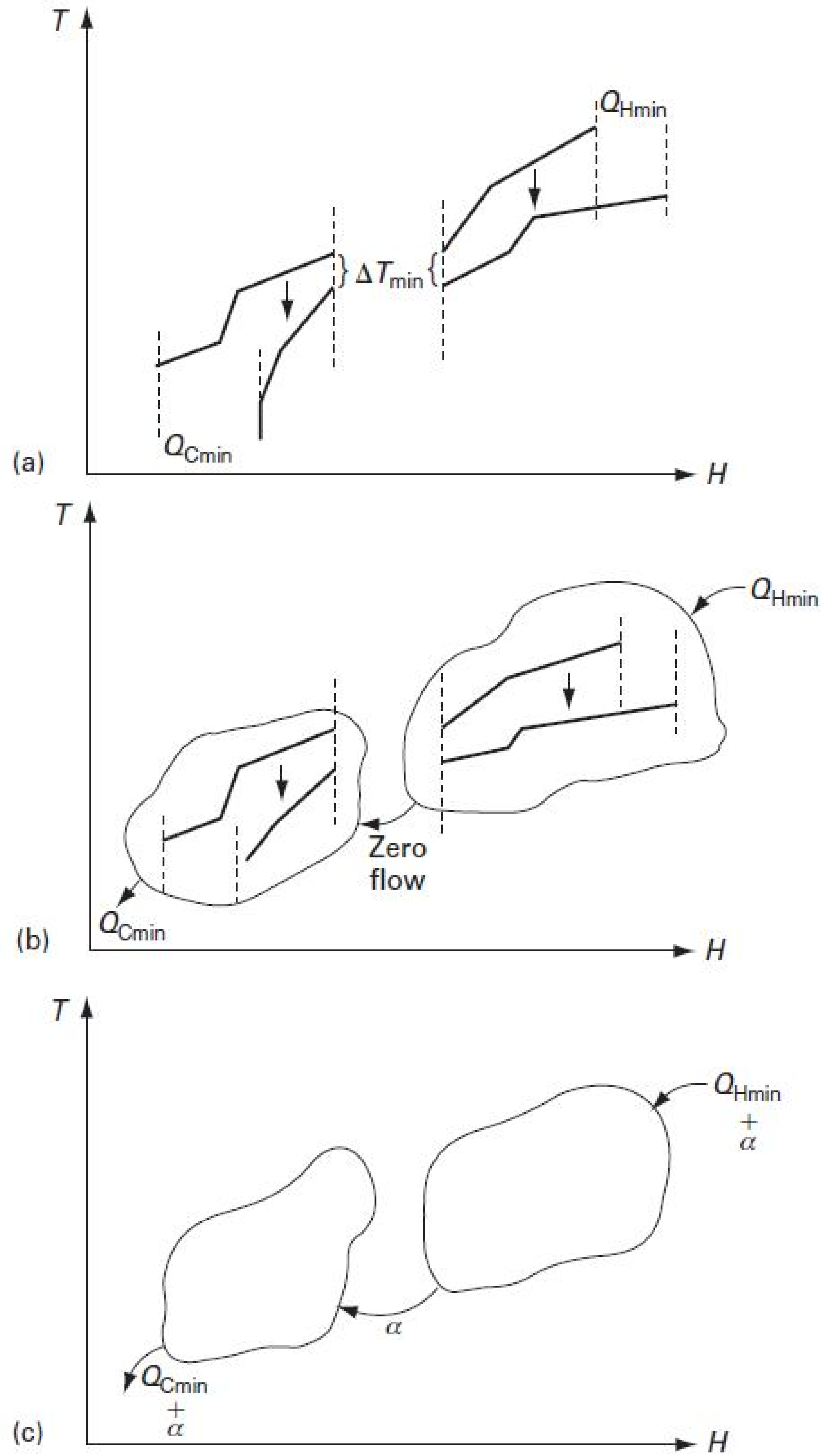


Figure 3.13 Dissecting a problem at the pinch

3.2.3 Heat exchanger network design

3.2.3.1 Network grid representation

For designing a heat exchanger network, the most helpful representation is the "grid diagram" introduced by Linnhoff and Flower (1978) (Figure 3.14). The streams are drawn as horizontal lines, with high temperatures on the left and hot streams at the top; heat exchange matches are represented by two circles joined by a vertical line. The grid is much easier to draw than a flowsheet, especially as heat exchangers can be placed in any order without redrawing the stream system. Also,the grid represents the countercurrent nature of the heat exchange, makingit easier to check exchanger temperature feasibility. Finally, the pinch is easilyrepresented in the grid (as will be shown in the next sub-section), whereas it cannotbe represented on the flowsheet.

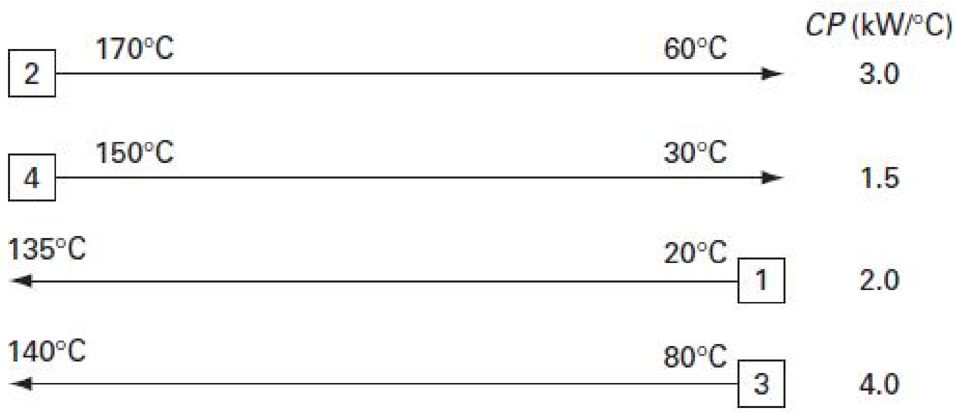


Figure 3.14 Initial grid diagram for four-stream problem

3.2.3.2 A "commonsense" network design

A simple heat exchanger network now will be produced for the four-stream problem and represent it on the grid diagram. Figure 3.14 shows the initial situation. Heat wanted to be exchanged between the hot and cold streams, and logically it should be started at one end of the temperature range. Matching the hottest hot stream 2 against the hottest cold stream 3 should give the best temperature driving forces and ensure feasibility. If the whole of the heat load on stream 3 (240 kW) is matched, it can be calculated that stream 2 has been brought down to 90°C, which is just acceptable for the given ΔTmin of 10°C. Then stream 4 is matched against stream 1, and it found that the whole of the 180 kW in stream 4 can be used while again achieving the ΔTmin =10°C criterion at the bottom end. This raises stream 1 to 110°C, so it needs an additional 50 kW of hot utility to raise it to the required final temperature of 135°C. Finally, a cooler is added to stream 2 to account for the remaining 90 kW required to bring it down to 60°C. The resulting design is shown as a network grid in Figure 3.15. 420 kW of heat exchange have been achieved, so that only 50 kW external heating and 90 kW of cooling are required, with two exchangers.

However, when the energy targets are seen, just 20 kW of heating and 60 kW of cooling should be used, but the design needs 50 and 90kW, respectively. The wrong which have been done is to attempt to produce a better network by trial-and-error. However even with the knowledge of the targets, it is no easy task to design a network to achieve them by conventional means.

Instead, it is better to use the insights given by the pinch concept to find out why the target has been missed. The pinch is at a shifted temperature of 85°C, corresponding to 90°C for hot streams and 80°C for cold streams. It is known that there are three, and only three, reasons for a process to be off-target. Looking at the grid diagram (and assisted by having the pinch drawn in) there are no heaters below the pinch, nor coolers above it. Therefore, heat transfer across the pinch must have taken place, and the problem must be in heat exchanger 2. Although it spans the pinch on both streams, stream 2 can be calculated is releasing 90 kW above 90°C and 90 kW below it, while stream 3 is receiving only 60 kW above 80°C and 120 kW below it. Therefore, 30 kW is being transferred across the pinch, corresponding precisely to the 30 kW excess of the hot and cold utility over the targets.

The targets can be guaranteed without violating the three golden rules by starting the design at the most constrained point – the pinch itself – and working outwards. This in itself explains why "traditional" heat exchanger network designs are almost invariably off-target.

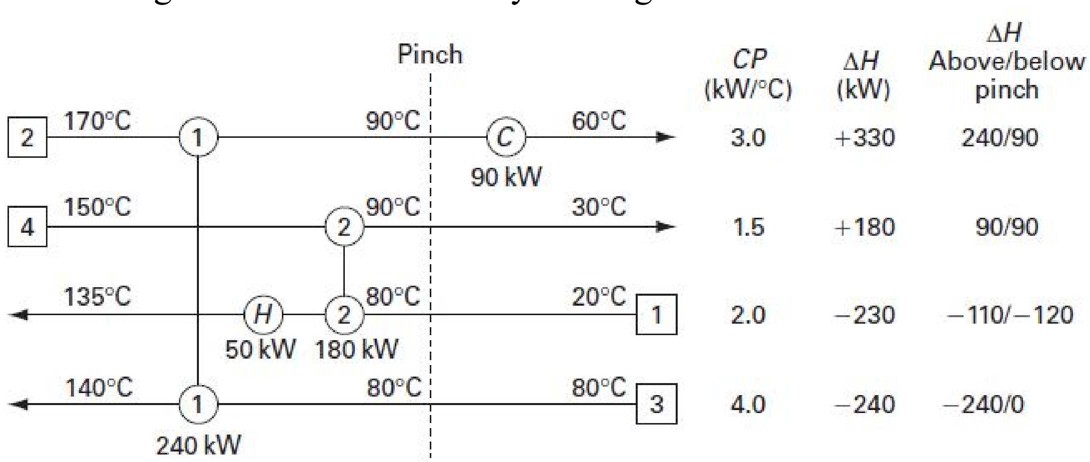


Figure 3.15 "Commonsense" network design for four-stream problem

3.2.3.3 Design for maximum energy recovery

For maximum energy recovery discussion, consider the grid diagram and the new network. Notice that stream number 3 starts at the pinch. In fact in problems where the streams all have constant *CP*s, the pinch is always caused by the entry of a stream, either hot or cold.

It is known that above the pinch, no utility cooling should be used. This means that above the pinch, *all hot streams must be brought to pinch temperature by interchangeagainst cold streams*. the design must therefore be started *at the pinch*, finding matches that fulfill this condition. In this example, above the pinch there are two hot streams at pinch temperature, therefore requiring two "pinch matches". In Figure 3.16(a) a match between streams 2 and 1 is shown, with a *T*/*H* plot of the match shown in inset. (Note that the stream directions have been reversed so as to mirror the directions in the grid representation.) Because the *CP* of stream 2 is greater than that of stream 1, as soon as any load is placed on the match, the ΔT in the exchanger becomes less than ΔTmin at its hot end. The exchanger is clearly infeasible and therefore it must look for another match. In Figure 3.16(b), streams 2 and 3 are matched, and now the relative gradients of the *T*/*H* plots mean that putting load on the exchanger opens up the ΔT.

This match is therefore acceptable. If it is put in as a firm design decision, then stream 4 must be brought to pinch temperature by matching against stream 1 (i.e. this is the only option remaining for stream 4). Looking at the relative sizes of the *CP*s for streams 4 and 1, the match is feasible ($CP_4 < CP_1$).

There are no more streams requiring cooling to pinch temperature and so a feasible design has been found at the pinch. It is the only feasible pinch design because only two pinch matches are required.

Summarising, in design immediately above the pinch, it must meet the criterion:

$$CP_{\text{HOT}} \leq CP_{\text{COLD}}$$

Having found a feasible pinch design it is necessary to decide on the match heat loads. The recommendation is "maximise the heat load so as to completely satisfy one of the streams".

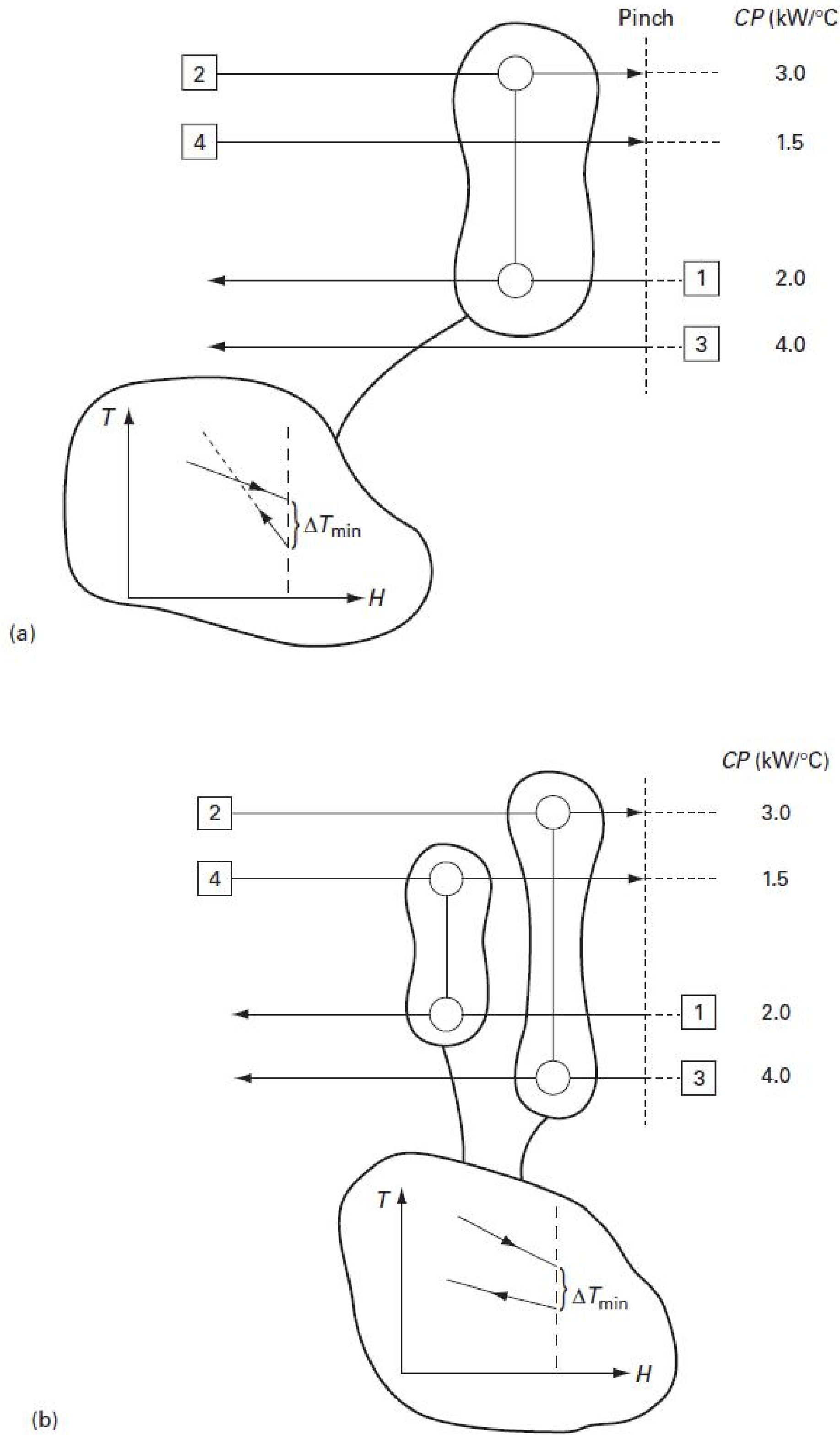


Figure 3.16 Consequences of matching streams of different CPs

This ensures the minimum number of heat exchange units is employed. So, since stream 2 above the pinch requires 240 kW of cooling and stream 3 above the pinch requires 240 kW of heating, co-incidentally the 2/3 match is capable of satisfying both streams. However, the 4/1 match can only satisfy stream 4, having a load of 90 kW and therefore heating up stream 1 only as far as 125°C. Since both hot streams have now been completely exhausted by these two design steps, stream 1 must be heated from 125°C to its target temperature of 135°C by external hot utility as shown in Figure 3.18.This amounts to 20kW, as predicted by the Problem Table analysis. This is no coincidence! The design has been put together obeying the constraint of not transferring heat across the pinch (the "above the pinch" section has been designed completely independently of the "below the pinch" section) and not using utility cooling above the pinch.

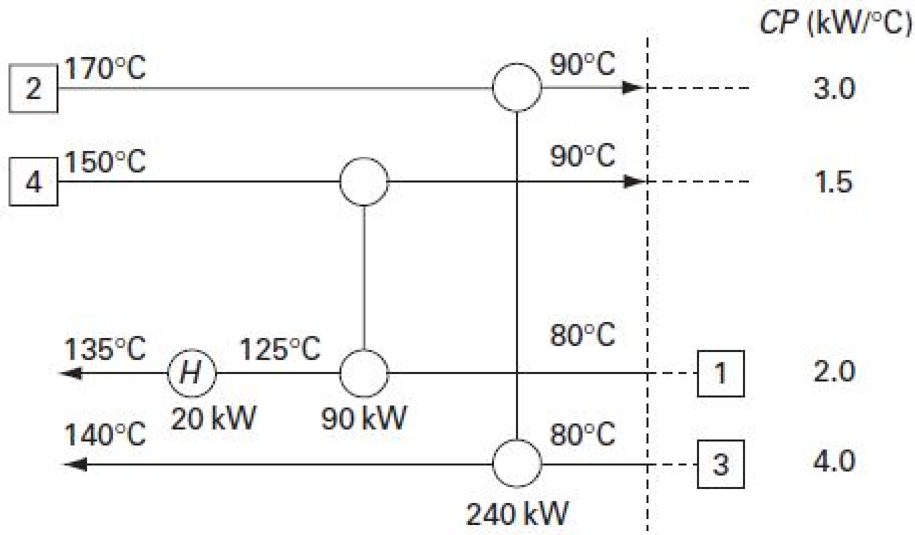


Figure 3.17 Above-pinch network design for four-stream problem

Below the pinch, the design steps follow the same philosophy, only with design criteria that mirror those for the "above the pinch" design. Now, it is required to bring cold streams to pinch temperature by interchange with hot streams, since utility heating is not wanted to be used below the pinch. In this example, only one cold stream 1 exists below the pinch, which must be matched against one of the two available hot streams 2 and 4. The match between streams 1 and 2 is feasible because the *CP* of the hot stream is greater than that of the cold, and the temperature difference increases as moving away from the pinch to lower temperatures. The other possible match (stream 1 with stream 4) is not feasible. Immediately below the pinch, the necessary criterion is:

$$CP_{\text{HOT}} \geq CP_{\text{COLD}}$$

which is the reverse of the criterion for design immediately above the pinch. Maximising the load on this match satisfies stream 2, the load being 90kW. The heating required by stream 1 is 120 kW and therefore 30 kW of residual heating, to take stream 1 from its supply temperature of 20–35°C, is required. Again this must come from interchange with a hot stream (not hot utility), the only one now available being stream 4. Although the *CP* inequality does not hold for this match, the match is feasible because *it is away from the pinch*. That is to say, it is not a match that has to bring the cold stream up to pinch temperature. So the match does not become infeasible (though a temperature check should be done to ensure this). Putting a load of 30 kW on this match leaves residual cooling of 60 kW on stream 4 which must be taken up by cold utility. Again, this is as predicted by the Problem Table analysis. The below-pinch design including the *CP* criteria is shown in Figure 3.18.

Putting the "hot end" and "cold end" designs together gives the completed design shown in Figure 3.19. It achieves best possible energy performance for a ΔTmin of 10°C incorporating four exchangers, one heater and one cooler. In other words, six units of heat transfer equipment in all. It is known as an **MER** network (because it achieves the **minimum energy requirement**, and **maximum energy recovery**).

Summarising, this design was produced by:

- ✓ Dividing the problem at the pinch, and designing each part separately.
- ✓ Starting the design at the pinch and moving away.
- ✓ Immediately adjacent to the pinch, obeying the constraints:

  CPHOT $\leq CP$COLD (above) for all hot streams

  CPHOT $\geq CP$COLD (below) for all cold streams
- ✓ Maximising exchanger loads.
- ✓ Supplying external heating only above the pinch, and external cooling only below the pinch.

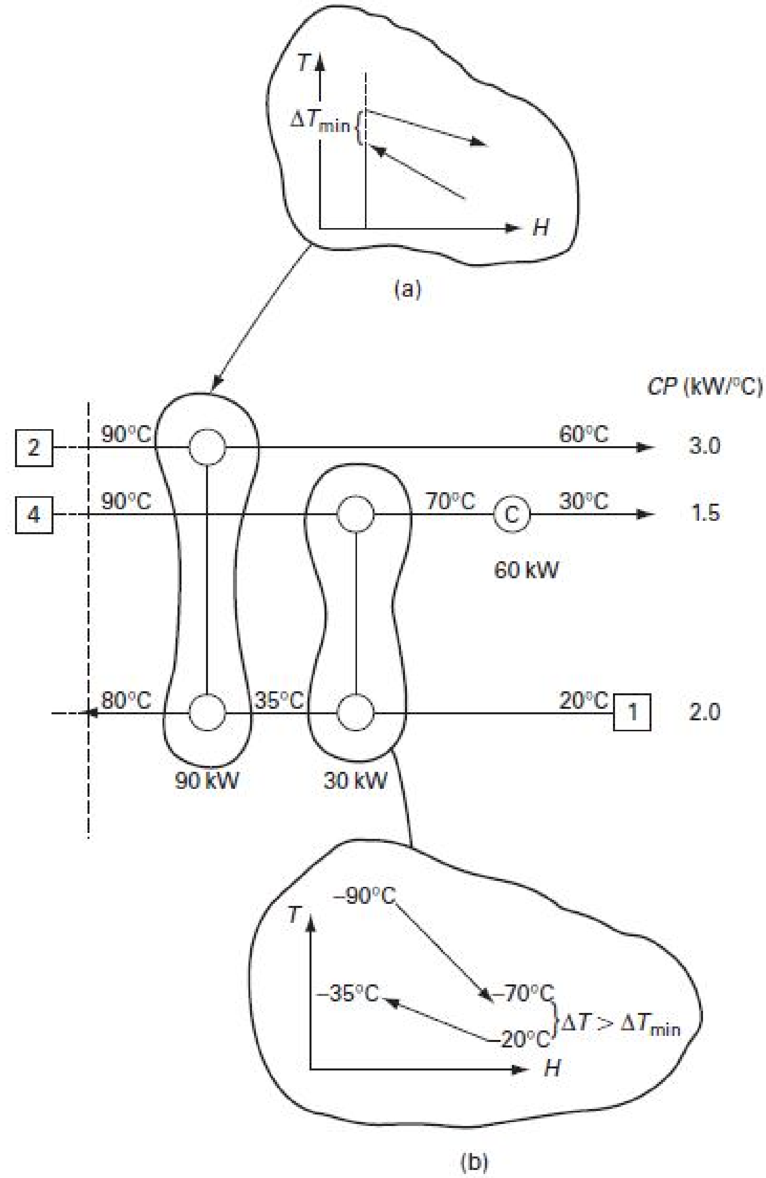


Figure 3.18 Below-pinch design for four-stream problem

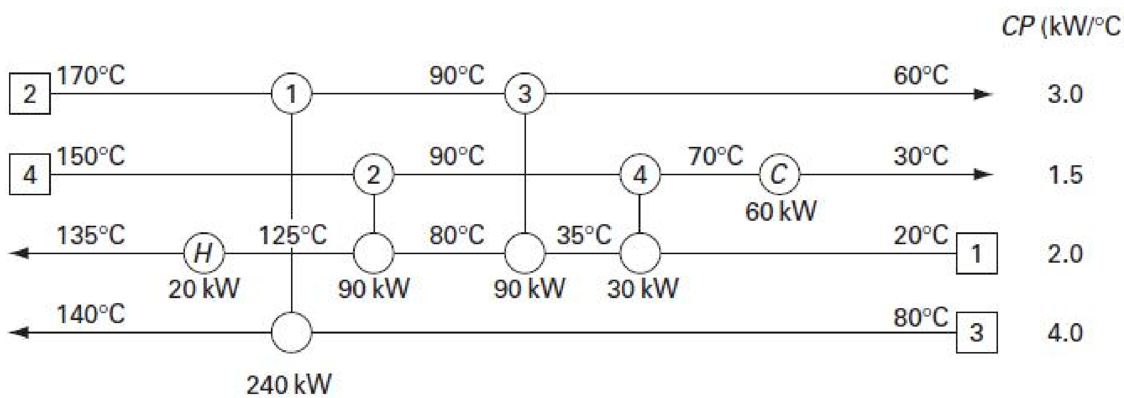


Figure 3.19 Network design achieving energy targets

These are the basic elements of the "pinch design method" of Linnhoff and Hindmarsh (1983). Network design is not always so easy – for example, streams may have to be split to meet the *CP* criteria at the pinch. Usually, trade-offs are made, known as "relaxing" the network to reduce the number of exchangers, at the expense of some increase in utility loads.

3.2.3.4 Choosing ΔTmin: supertargeting

3.2.3.4.1 Further implications of the choice of ΔTmin

So far, higher values of ΔTmin givehigher hot and cold utility requirements, and it therefore seems that a ΔTminwanted to be as low as possible, to give maximum energy efficiency. However, there is a drawback; lower ΔTmin values give larger and more costly heat exchangers. In a heat transfer device, the surface area A required for heat exchange is given by:

$$A = \frac{Q}{U\Delta T_{LM}} \qquad (3.4)$$

A is in m^2, Q is the heat transferred in the exchanger (kW), U is the overall heat transfer coefficient (kW/m^{2}K) and ΔT_{LM} is the log mean temperature difference (*K)*. If a pure countercurrent heat exchanger have been considered, where the hot stream enters at T_{h1} and leaves at T_{h2}, and the cold stream enters at T_{c1} and exits at T_{c2}, so that T_{c1} and T_{h2} are at the "cold end" C and T_{h1} and T_{c2} are at the "hot end" H of the exchanger, then ΔT_{LM} is given by:

$$\Delta T_{\mathrm{LM}} = \frac{\Delta T_{\mathrm{H}} - \Delta T_{\mathrm{C}}}{\ln \frac{\Delta T_{\mathrm{H}}}{\Delta T_{\mathrm{C}}}} = \frac{T_{\mathrm{h1}} - T_{\mathrm{c2}} - T_{\mathrm{h2}} + T_{\mathrm{c1}}}{\ln\left(\frac{T_{\mathrm{h1}} - T_{\mathrm{c2}}}{T_{\mathrm{h2}} - T_{\mathrm{c1}}}\right)} \qquad (3.5)$$

If ΔT_{H}=ΔT_{C}, ΔT_{LM} is undefined and ΔT is used in Equation (3.4). In essence, the heat exchanger area is roughly inversely proportional to the temperature difference. Hence, low values of ΔTmin can lead to very large and costly exchangers, as capital cost is closely related to area. Even if one end of an exchanger has a hightemperature difference, the form of the expression for ΔT_{LM} means that it is dominated by the smaller temperature approach. Obviously a low ΔT_{min} value gives a low ΔT_{LM}.

For the two-stream example, Figure 3.20 plots the utility use and heat exchanger surface area (assuming a value of 0.1kW/m^{2}K for the heat transfer coefficient), and also shows the effect of including heaters and coolers (assuming sensible temperature levels). Here, utility use rises linearly with ΔT_{min}, whereas exchanger area rises very sharply (asymptotically) for low ΔTmin values.

Now, if it is assumed that energy cost is proportional to energy usage, and that heat exchanger cost (as a first approximation) is proportional to surface area, the operating and capital cost can be summed. Since energy cost is per hour and capital cost is a one-off expenditure, either the energy cost is needed to be calculated over a period (say 1–2 years) or the capital cost to be annualised over a similar period. The chosen timescale is known as the **payback** time.This then gives the combined total cost graph at the top of Figure 3.20, including utility, exchanger, heater and cooler cost. It can be seen that there is an optimum for $\Delta T_{\min}$ – in this case, about 15–20°C.

Clearly, it will be important to choose the right value of $\Delta T_{\min}$ for targetingand network design. This can be done by area and cost targeting, or **supertargeting**,based on the concept above. Supertargeting ismuch less exact than energy targeting, because there are many uncertainties – heattransfer coefficients, total area of a exchanger network and costs are all subject to variation. However, it is noted that the total cost curve has a relatively flat optimum, sothere is a fair amount of leeway. As long as the chosen $\Delta T_{\min}$ is not excessively smallor large, a reasonable design should be obtained by using a sensible "experiencevalue" for $\Delta T_{\min}$, at least in the initial stages. Often a value of 10°C or 20°C is best,but in some industries, a very much lower or higher value is appropriate.

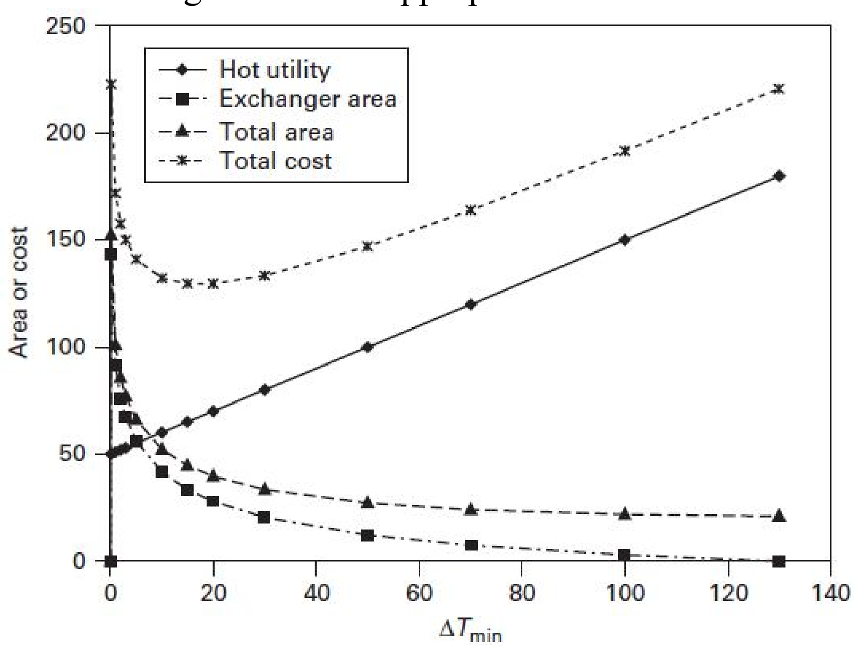


Figure 3.20 Utility use, exchanger area and cost variation with ΔT_{min}

The Summarizing of the key points of area and cost targeting are, usually an optimal value for $\Delta T_{\min}$ part way through the feasible range, Also the optimum is not exact and a significant error can usually be tolerated initially.

Chapter Four: Exergy and Pinch Technology analysis

4.1 Introduction

In this chapter the Exergy and Pinch technology are applied on KNPSP(2). The analysis procedure is as applying set of equations express Exergy on the whole plant's main components as per mentioned in section 1.4. As already has been explained in chapter 2, that the identifying of the Exergy means indentifying of potential work. Also Exergy destruction calculation results point out how that components effect on the efficiency of the whole plant.

On the other hand, the pinch technology analysis is considered by applying the set of equations and techniques on the Feed Water Heaters system as mentioned in section 1.4. As it known the pinch analysis may be applied on the plant design stage to be sure that Heat Exchanger Networks have efficient design and minimum Heat and Cold utilities. However in this research the other purpose of pinch analysis is considered which is to re-evaluation of existing Heat Exchanger Networks which are the feed water heaters system.

The description of plant is introduced and all assumptions considered for any component are figured out.

4.2 Plant Description

The Khartoum North Power Station Phase(2) (KNPSP(2)) has a total installed power capacity of (120) MW. It is located in Industrial Area district, in Khartoum, Sudan. It started to produce power in the end nineties. The power house consists of two steam turbines units (2x60) MW at 100% load. The KNPSP(2) uses Heavy cocker gas oil. The schematic diagram of one (60) MW unit at 100% load is shown in Figure 4.1.

This unit employs regenerative feed water heating system. Feed water heating is carried out in six closed heat exchangers and one open heat exchanger (Daerator). The first closed heat exchanger receives hot stream from outlet of ejector and the second closed heat exchanger receives hot stream from outlet of gland steam system. The extracted steam streams from the turbine are distributed along the other four closed heat exchangers and Daerator as hot streams. Steam is superheated to 512.5 ℃ and 89.50 bar in the Boiler and fed to the turbine. At full load the mass flow rate of the steam entering the high pressure turbine is 64.216kg/s, the turbine exhaust stream at pressure 0.075 bar and it exhausts to a water cooled condenser operates . Then, the cycle starts over again.

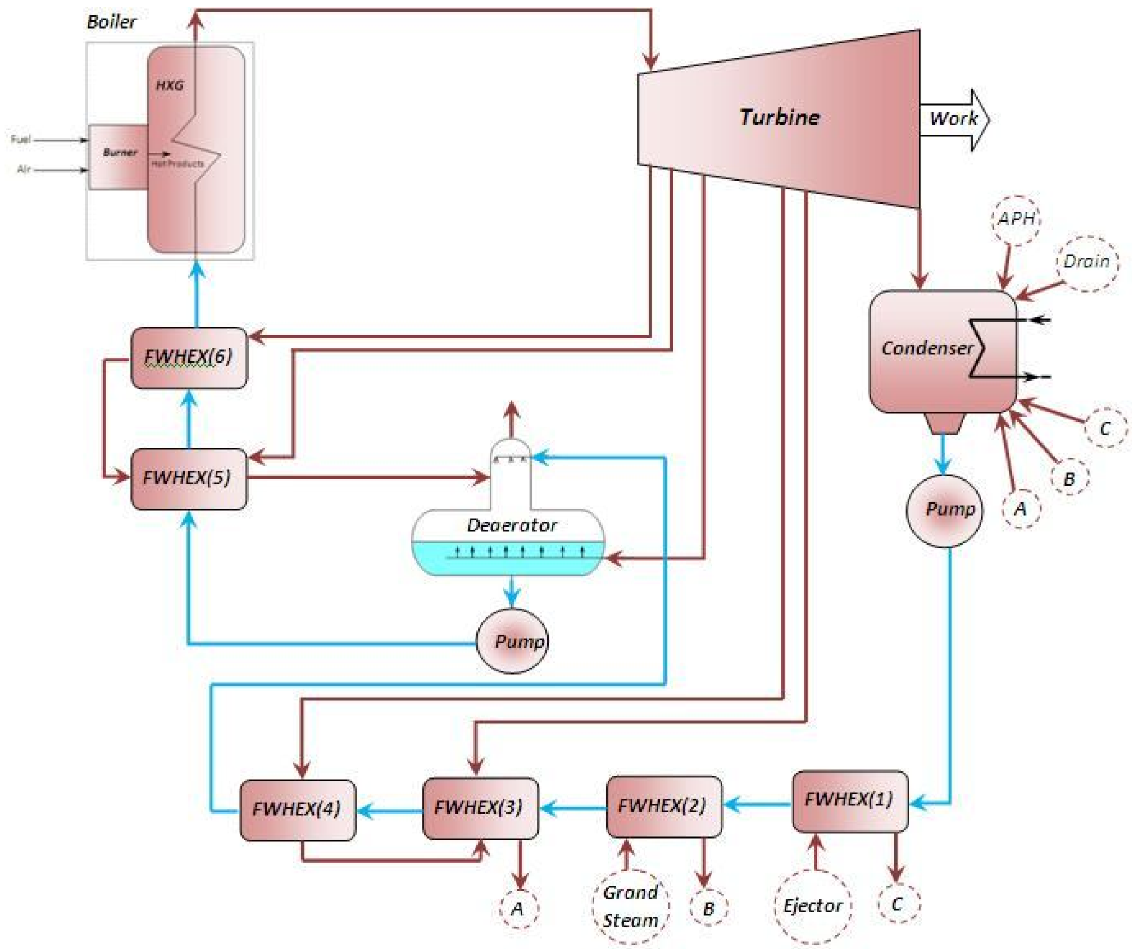


Figure 4.1 The schematic diagram of one (60) MW unit {KNPSP(2)}

4.3 Assumptions

In this research the common assumptions are

- ✓ The kinetic and potential energy changes are negligible.
- ✓ Any flow process in this analysis is considered as steady flow process.

4.4 Exergy analysis

In this section the Exergy analysis is applied on components each alone. The dead state properties of water have been taken at 31℃ and 1 atm to be in equilibrium with environment which is in the same state in the plant.

Table 4.1 working fluid (water) properties at dead state

$T_0(K)$	$P_0(bar)$	$h_0(\frac{Kj}{kg})$	$S_0(\frac{Kj}{kg\ K})$
304	1.01325	130.01360	0.45053

Any component in exergy analysis is undergoing the set of meaningful equations which are Mass, Energy, Exergy and Entropy balance. Applying mass balance equation verifies mass conservation and energy gives the total heat loss out the component. The essential equation in those equations is Exergy balance equation which reflects the potential work destruction (Exergy destruction / Irreversibility) in the component or work reversible if component is work-produce device. Furthermore, Entropy balance is used as cross check to exergy destruction is worked out Exergy balance equation in each component. It is very important to know that applying Entropy balance equation reflects high compatibility of data extracted, calculated results and ensure that there is no cooking takes place in results.

The given data of each point in any process in this analysis are mass flow rate, pressure and temperature. Specific enthalpy and specific are taken from the water-steam table and the specific exergy are worked out the equation (2.18)with negligible kinetic and potential energies.

4.4.1Exergy analysis calculations

4.4.1.1 Steam Turbine

The steam turbine used in KNPSP(2) has one inlet and five outlet as extracted steam and sixth outlet to condenser. The main assumption is the flow through this steam turbine is steady.

Table 4.2 Steam Properties at Steam Turbine

$Stream$	$Notes$	P (bar)	T (°C)	$\dot{m}$ $(\frac{kg}{s})$	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
MS	Main stream	87	510	62.608	3415.1	6.7068	1383.18
6	Extracted steam to FWHX(6)	25.09	338.5	4.421	3101.5	6.7976	1041.977
5	Extracted steam to FWHX(5)	12.57	251.7	3.562	2959.7	6.8529	883.3659
D	Extracted steam to Deaerator	5.05	166.2	3.889	2780	6.8928	691.5363
4	Extracted steam to FWHX(4)	1.94	119.2	2.554	2638.5	6.968	527.1755
3	Extracted steam to FWHX(3)	0.829	94.5	4.183	2666.83	7.4221	417.4591
C	Exhausted steam to Condenser	0.075	40.2	43.999	2258.6	7.2435	63.52352

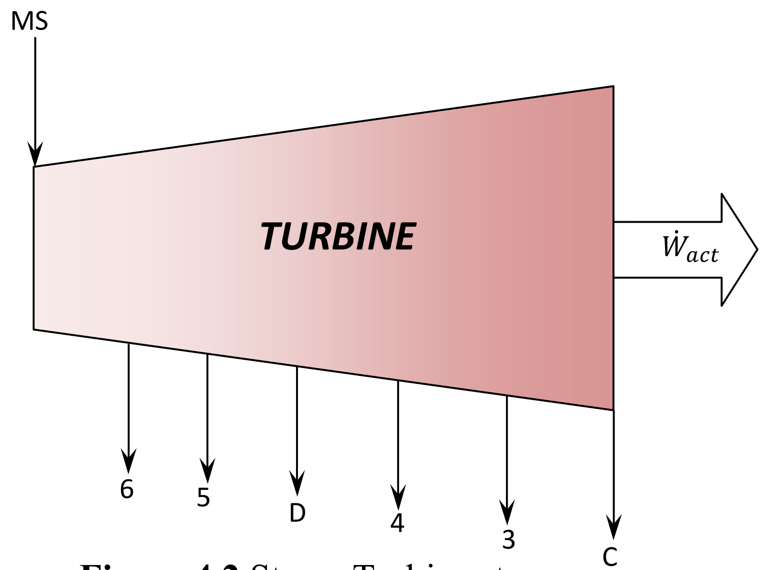


Figure 4.2 Steam Turbine streams

<u>Mass balance</u>

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

$$\sum_{in} \dot{m} = \dot{m}_{ms} = \ 62.608 \, {}^{\mathrm{Kg}}/_{\mathrm{s}}$$

$$\sum_{in} \dot{m} = \dot{m}_6 + \ \dot{m}_5 + \dot{m}_D + \dot{m}_4 + \dot{m}_3 + \dot{m}_C$$

$$= 4.421 + 3.562 + 3.889 + 2.554 + 4.183 + 43.999$$

$$= 62.608 \ {}^{\mathrm{Kg}}/_{\mathrm{s}}$$

Then mass conservation achieved.

<u>Energy balance</u>

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

$$\dot{m}_{ms} h_{ms} = \dot{m}_6 h_6 + \dot{m}_5 h_5 + \dot{m}_D h_D + \dot{m}_4 h_4 + \dot{m}_3 h_3 + \dot{m}_C h_C + \dot{Q}_{loss} + \dot{W}_{act}$$

$$\dot{Q}_{loss} = \dot{m}_{ms}h_{ms} - (\dot{m}_6h_6 + \dot{m}_5h_5 + \dot{m}_Dh_D + \dot{m}_4h_4 + \dot{m}_3h_3 + \dot{m}_Ch_C + \dot{W}_{act})$$

$$\dot{Q}_{loss} = 62.608 * 3415.1 - (4.421 * 3101.5 + 3.562 * 2959.7 + 3.889 * 2780 + 2.554 * 2638.5 + 4.183 * 2666.83 + 43.999 * 2258.6 + 60000)$$

$$\dot{Q}_{loss} = 1476.8\ KW$$

Exergy balance

$$\dot{W}_{rev} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m\psi}$$

$$\dot{W}_{rev} = \dot{m}_{ms}\psi_{ms} - (\dot{m}_6\psi_6 + \dot{m}_5\psi_5 + \dot{m}_D\psi_D + \dot{m}_4\psi_4 + \dot{m}_3\psi_3 + \dot{m}_C\psi_C)$$

$$\dot{W}_{rev} = 62.608 * 1383.1 - (4.421 * 1041.977 + 3.562 * 883.3659 + 3.889 * 691.5363 + 2.554 * 527.1755 + 4.183 * 417.4591 + 43.999 * 63.52352) = 70263\ KW$$

$$\dot{W}_{rev} = 70.3\ MW$$

$$\dot{I} = \dot{W}_{rev} - \dot{W}_{act}$$

$$\dot{I} = 70.3 - 60 = 10.3\ MW$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_6S_6 + \dot{m}_5S_5 + \dot{m}_DS_D + \dot{m}_4S_4 + \dot{m}_3S_3 + \dot{m}_CS_C + \frac{\dot{Q}_{loss}}{T_0} - \dot{m}_{ms}S_{ms}$$

$$\dot{S}_{gen} = 4.421 * 6.7976 + 3.562 * 6.8529 + 3.889 * 6.8928 + 2.554 * 6.968 + 4.183 * 7.4221 + 43.999 * 7.2435 + \frac{1476.8}{304} - 62.608 * 6.7068$$

$$\dot{S}_{gen} = 33.78\ \frac{\text{Kj}}{\text{Kg. K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 33.78 = 10268.07\ Kw = 10.3\ Mw$$

$$\eta_{II} = \frac{\dot{W}_{act}}{\dot{W}_{rev}}$$

$$\eta_{II} = \frac{60}{70.3} = 0.85349 = 85.3\%$$

$$\eta_{II} = 85.3\%$$

4.4.1.2 Condenser

The condenser used in KNPSP(2) has one main hot stream inlet which is steam turbine discharge, also there are five hot streams inlet for consider which are FWHX(1,2,3) outlet, Air pre heater outlet and Plant-drain outlet. the outlet hot stream for consider is consider only one hot stream outlet. The cold stream inlet comes from Cooling tower and cold stream outlet goes back to Cooling tower. The main assumption is the flow through this steam turbine is steady.

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\begin{aligned}\sum_{H,in} \dot{m} &= \dot{m}_{H,in} + \dot{m}_{H,in,1} + \dot{m}_{H,in,2} + \dot{m}_{H,in,3} + \dot{m}_{H,in,APH} + \dot{m}_{H,in,Drain} \\ &= 43.999 + 0.063 + 0.035 + 6.737 + 1.21 + 1.101 \\ &= 53.145\ \text{Kg/s}\end{aligned}$$

$$\sum_{H,out} \dot{m} = \dot{m}_{H,out} = 53.145\ \text{Kg/s}$$

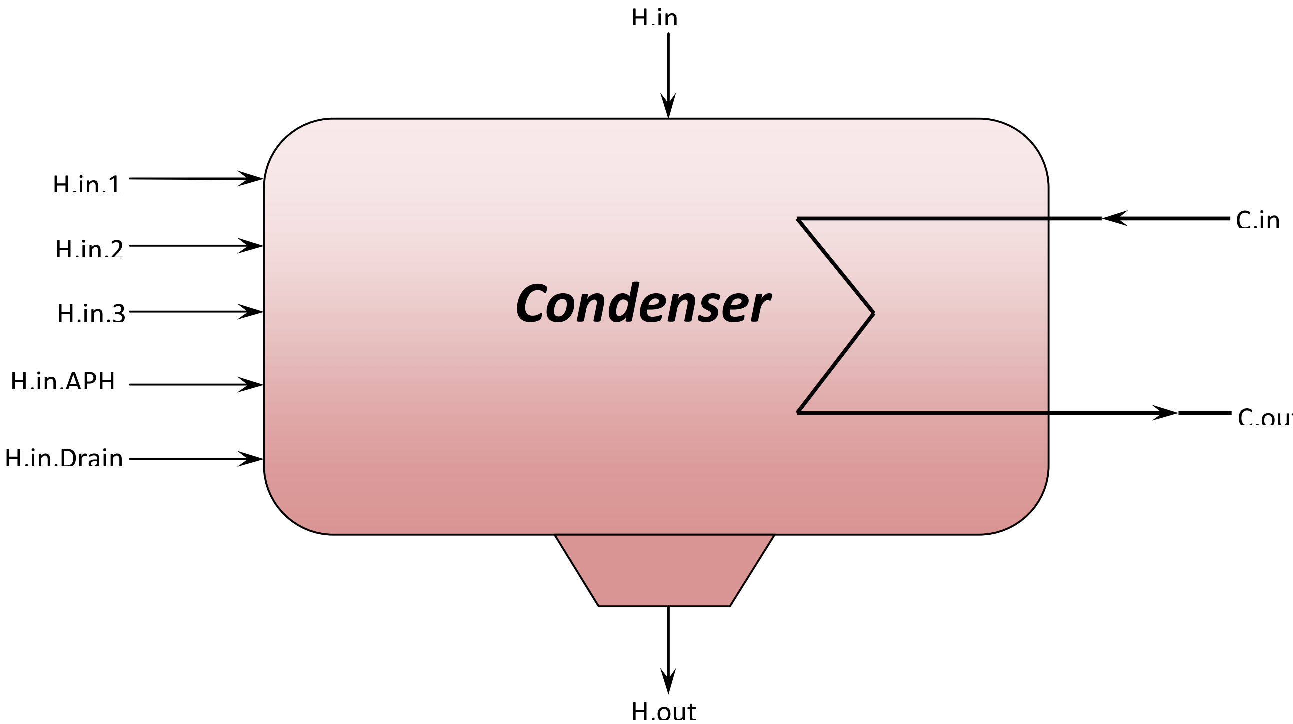


Figure 4.3 Condenser Streams

Table 4.3 Steam/Water properties at Condenser

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (℃)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	43.999	0.075	40.2	2258.6	7.2435	63.52352
H,in,1	Hot stream outlet of HXG(1)	0.063	1	43	180	0.6122	0.83872
H,in,2	Hot stream outlet of HXG(2)	0.035	1	99.6	420	1.3102	28.64672
H,in,3	Hot stream outlet of HXG(3)	6.737	0.729	47.6	199.3	0.6726	1.77712
H,in,APH	Hot stream outlet of Air pre heater	1.21	17.5	100	420.3	1.30573	30.3056

H,in,Drain	Hot stream, outlet of Drain	1.101	1	30	125.8	0.43676	-0.02752
H,out	Hot stream, outlet	53.145	0.075	40.2	168.4	0.575097	0.518032
C,in	Cold stream, inlet	2386	2.6	29	121.7987	0.4229	0.18462
C,out	Cold stream, outlet	2386	0.9	38	159.257	0.54562	0.33604

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 2386 \text{ Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 2386 \text{ Kg/s}$$

Then mass conservation achieved.

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,1}h_{H,in,1} + \dot{m}_{H,in,2}h_{H,in,2} + \dot{m}_{H,in,3}h_{H,in,3} + \dot{m}_{H,in,APH}h_{H,in,APH} + \dot{m}_{H,in,Drain}h_{H,in,Drain} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = \dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,1}h_{H,in,1} + \dot{m}_{H,in,2}h_{H,in,2} + \dot{m}_{H,in,3}h_{H,in,3} + \dot{m}_{H,in,APH}h_{H,in,APH} + \dot{m}_{H,in,Drain}h_{H,in,Drain} + \dot{m}_{C,in}h_{C,in} - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = 43.999 * 2258.6 + 0.063 * 180 + 0.035 * 420 + 6.737 * 199.3 + 1.21 * 420.3 + 1.101 * 125.8 + 2386 * 121.7987 - (53.145 * 168.4 + 2386 * 159.257) = 3066.812\ Kw$$

$$\dot{Q}_{loss} = 3066.812\ Kw$$

Exergy balance

$$\dot{X}_{destoryed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destoryed} = (\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,1}\psi_{H,in,1} + \dot{m}_{H,in,2}\psi_{H,in,2} + \dot{m}_{H,in,3}\psi_{H,in,3} + \dot{m}_{H,in,APH}\psi_{H,in,APH} + \dot{m}_{H,in,Drain}\psi_{H,in,Drain} + \dot{m}_{C,in}\psi_{C,in}) - (\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out})$$

$$\dot{X}_{destoryed} = 43.999 * 63.52352 + 0.063 * 0.83872 + 0.035 * 28.64672 + 6.737 * 1.77712 + 1.21 * 30.3056 + 1.101 * (-0.02752) + 2386 * 0.18462 - (53.145 * 0.518032 + 2386 * 0.33604) = 2455.8\ Kw$$

$$\dot{I} = \dot{X}_{destroyed} = 2.4558\ Mw$$

$$Exergy\ supply = \dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,1}\psi_{H,in,1} + \dot{m}_{H,in,2}\psi_{H,in,2} + \dot{m}_{H,in,3}\psi_{H,in,3} + \dot{m}_{H,in,APH}\psi_{H,in,APH} + \dot{m}_{H,in,Drain}\psi_{H,in,Drain} - \dot{m}_{H,out}\psi_{H,out}$$

$$Exergy\ supply = 43.999 * 63.52352 + 0.063 * 0.83872 + 0.035 * 28.64672 + 6.737 * 1.77712 + 1.21 * 30.3056 + 1.101 * (-0.02752) - 53.145 * 0.518032$$

$$Exergy\ supply = 2817.1\ Kw$$

$$Exergy\ recovery = \dot{m}_C(\psi_{C,out} - \psi_{C,in}) = 2386 * (0.33604 - 0.18462)$$

$$Exergy\ recovery = 361.3 Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{361.3}{2817.1} = 0.128 = 12.8\%$$

$$\eta_{II} = 12.8\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{H,in,1}S_{H,in,1} + \dot{m}_{H,in,2}S_{H,in,2} + \dot{m}_{H,in,3}S_{H,in,3} + \dot{m}_{H,in,APH}S_{H,in,APH} + \dot{m}_{H,in,Drain}S_{H,in,Drain} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = 53.145 * 0.575097 + 2386 * 0.54562 + \frac{3066.812}{304} - (43.999 * 7.2435 + 0.063 * 0.6122 + 0.035 * 1.3102 + 6.737 * 0.6726 + 1.21 * 1.30573 + 1.101 * 0.43676 + 2386 * 0.4229)$$

$$\dot{S}_{gen} = 8.078353 \frac{\text{Kj}}{\text{Kg. K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 8.078353 = 2455.8Kw = 2.4558\ Mw$$

Then equalization of $\dot{X}_{destroyed}$ and $T_0\dot{S}_{gen}$ achieved as cross-check for compatibility of calculated results.

4.4.1.3 Feed water heat exchangers system

The function of the whole feed water heat exchangers system is to warm up the feed water comes from condenser outlet to boiler inlet so as to reduce the heat added to boiler and fuel consumption.

4.4.1.3.1 Feed water heat exchanger (1)

The feed water heat exchanger (1) (hereafter FWHX1) receives hot stream from outlet of ejector (Figure 4.4).

Table 4.4 Steam/Water Properties at FWHEX(1)

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	s $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	0.061	1.05	163.8	2803.3	7.6552	483.06672
H,out	Hot stream, outlet	0.061	1	43	180	0.6122	0.83872
C,in	Cold stream, inlet	53.145	12	40.1	170	0.5733	2.66432
C,out	Cold stream, outlet	53.145	11.70	41.1	173	0.5866	1.62112

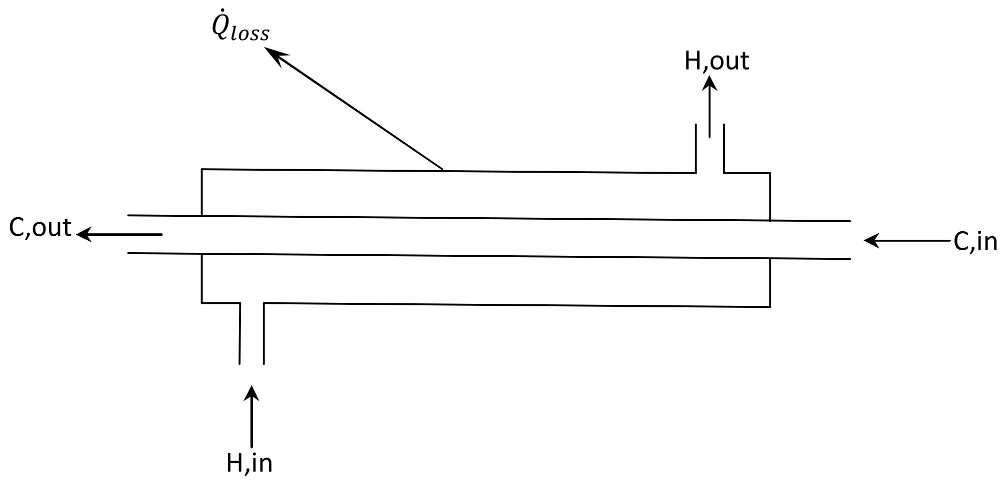


Figure 4.4 FWHEX(1,2,4&6) Streams

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} = 0.061\ \mathrm{Kg/s} \quad , \sum_{H,out} \dot{m} = \dot{m}_{H,out} = 0.061\ \mathrm{Kg/s}$$

Hot stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 53.145 \text{ Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 53.145 \text{ Kg/s}$$

Then mass conservation achieved.

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = \dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = 0.061 * 2803.3 + 53.145 * 170 - (0.061 * 180 + 53.145 * 173) = 5.83\ Kw$$

$$\dot{Q}_{loss} = 0.5863\ Kw$$

Exergy balance

$$\dot{X}_{destoryed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destoryed} = (\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{C,in}\psi_{C,in}) - (\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out})$$

$$\dot{X}_{destoryed} = 0.061 * 483.06672 + 53.145 * 2.66432 - (0.061 * 0.83872 + 53.145 * 1.62112) = 84.86\ Kw$$

$$\dot{I} = \dot{X}_{destroyed} = 84.86\ Kw$$

$$Exergy\ supply = \dot{m}_H(\psi_{H,in} - \psi_{H,out}) = 0.061 * (483.06672 - 0.83872)$$
$$Exergy\ supply = 29.4\ kw$$

$$Exergy\ recovery = \dot{m}_C(\psi_{C,out} - \psi_{C,in}) = 53.145 * (1.62112 - 2.66432)$$
$$Exergy\ recovery = -55.46\ Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{-55.46}{29.4} = -1.886 = -188.6\%$$

$$\eta_{II} = -188.6\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = 0.061 * 0.6122 + 53.145 * 0.5866 + \frac{0.5863}{304} - (0.061 * 7.6552 + 53.145 * 0.5733)$$

$$\dot{S}_{gen} = 0.27913 \frac{\mathrm{Kj}}{\mathrm{Kg.K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 0.27913 = 84.86\ Kw$$

Then equalization of $\dot{X}_{destroyed}$ and $T_0\dot{S}_{gen}$ achieved as cross-check for compatibility of calculated results.

4.4.1.3.2 Feed water heat exchanger (2)

The feed water heat exchanger (2) (hereafter FWHX2) receives the hot stream from outlet of Gland steam system (Figure 4.4).

Table 4.5 Steam/Water Properties at FWHEX(2)

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	0.035	1.05	336.1	3147.4	8.3179	625.70592
H,out	Hot stream, outlet	0.035	1	99.6	420	1.3102	28.64672
C,in	Cold stream, inlet	53.145	11.70	41.1	173	0.5866	1.62112
C,out	Cold stream, outlet	53.145	11.40	41.5	174.7	0.5919	1.70992

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} = 0.035\ \mathrm{Kg/s} \quad , \sum_{H,out} \dot{m} = \dot{m}_{H,out} = 0.035\ \mathrm{Kg/s}$$

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 53.145\ \mathrm{Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 53.145\ \mathrm{Kg/s}$$

Then mass conservation achieved.

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = 0.035 * 3147.4 + 53.145 * 17 - (0.035 * 420 + 53.145 * 174.7)$$
$$= 5.113\ Kw$$

$$\dot{Q}_{loss} = 5.113\ Kw$$

Exergy balance

$$\dot{X}_{destoryed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destoryed} = (\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{C,in}\psi_{C,in}) - (\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out})$$

$$\dot{X}_{destoryed} = (0.035 * 625.70592 + 53.145 * 1.62112)$$
$$- (0.035 * 28.64672 + 53.145 * 1.70992)$$

$$I = \dot{X}_{destoryed} = 16.2\ Kw$$

$$Exergy\ supply = \dot{m}_H(\psi_{H,in} - \psi_{H,out}) = 0.035 * (625.70592 - 28.64672)$$
$$Exergy\ supply = 20.9\ kw$$

$$Exergy\ recovery = \dot{m}_C(\psi_{C,out} - \psi_{C,in}) = 53.145 * (1.70992 - 1.62112)$$

$$Exergy\ recovery = 4.7\ Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{4.7}{20.9} = 0.225 = 22.5\%$$

$$\eta_{II} = 22.5\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{C,in}S_{C,in})$$

$\dot{S}_{gen} = 0.035 * 1.3102 + 53.145 * 0.5919 + \frac{5.113}{304} - \ (0.035 * 8.3179 + 53.145 * 0.5866)$

$$\dot{S}_{gen} = 0.053218 \ \frac{\text{Kj}}{\text{Kg. K}}$$

$\dot{I} = T_0\dot{S}_{gen} = 304 * 0.053218 = 16.2 \ Kw$

Then equalization of $\dot{X}_{destroyed}$ and $T_0\dot{S}_{gen}$ achieved as cross-check for compatibility of calculated results.

4.4.1.3.3 Feed water heat exchanger (3)

The feed water heat exchanger (3) (hereafter FWHX3) receives the hot stream from extracted steam (Figure 4.5).

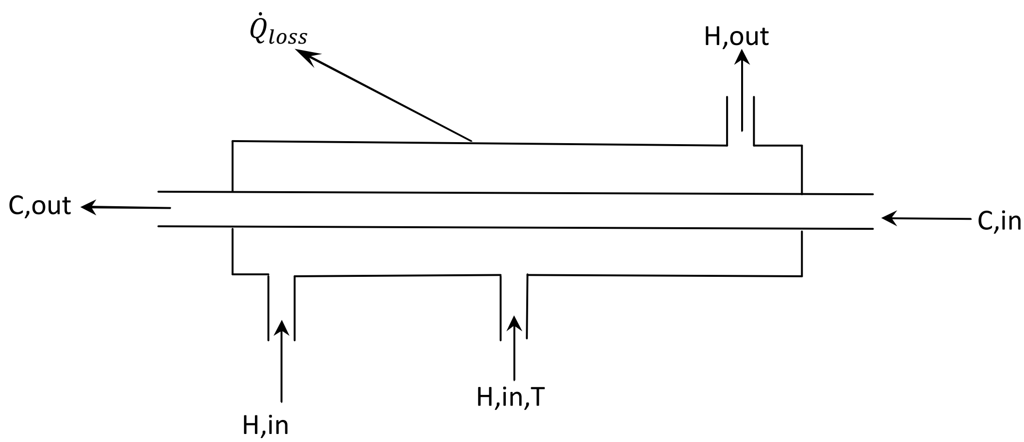


Figure 4.5 FWHEX(3&5) Streams

Table 4.6 Steam/Water Properties at FWHEX(3)

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$
H,in	Hot stream, inlet	4.183	94.5	2666.83	7.4221	417.459
H,in,T	Hot stream, Trap	2.554	92.1	386	1.2168	23.0403
H,out	Hot stream, outlet	6.737	47.6	199.3	0.6726	1.77712
C,in	Cold stream, inlet	53.145	41.5	174.7	0.5919	1.70992
C,out	Cold stream, outlet	53.145	87.2	366.1	1.1594	20.5899

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} + \dot{m}_{H,in,T} = 4.183 + 2.554 = 6.737 \text{ Kg/s} \quad ,$$

$$\sum_{H,out} \dot{m} = \dot{m}_{H,out} = 6.737 \text{ Kg/s}$$

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 53.145 \text{ Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 53.145 \text{ Kg/s}$$

Then the mass conservation achieved.

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,T}h_{H,in,T} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = (4.183 * 2666.83 + 2.554 * 386 + 53.145 * 174.7) - (6.737 * 199.3 + 53.145 * 366.1)$$

$$\dot{Q}_{loss} = 626.56\ KW$$

Exergy balance

$$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destroyed} = \left(\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} + \dot{m}_{C,in}\psi_{C,in}\right) - \left(\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out}\right)$$

$$\dot{X}_{destroyed} = (4.183 * 417.459 + 2.554 * 23.0403 + 53.145 * 1.70992) - (6.737 * 1.77712 + 53.145 * 20.5899)$$

$$I = \dot{X}_{destroyed} = 789.7\ KW$$

$$Exergy\ supply = \dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} - \dot{m}_{H,out}\psi_{H,out} = 4.183 * 417.459 + 2.554 * 23.0403 - 6.737 * 1.77712$$

$$Exergy\ supply = 1793.1\ kw$$

$$Exergy\ recovery = \dot{m}_C\left(\psi_{C,out} - \psi_{C,in}\right) = 53.145 * (20.5899 - 1.70992)$$

$$Exergy\ recovery = 1003.4\ Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{1003.4}{1793.1} = 0.560 = 56.0\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{H,in,T}S_{H,in,T} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = (6.737 * 0.6726 + 53.145 * 1.1594) + \frac{626.56}{304} - (4.183 * 7.4221 + 2.554 * 1.2168 + 53.145 * 0.5919)$$

$$\dot{S}_{gen} = 2.597795\,\frac{\mathrm{Kj}}{\mathrm{Kg.K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 2.597795 = 789.7\ Kw$$

4.4.1.3.4 Feed water heat exchanger (4)

The feed water heat exchanger (4) (hereafter FWHX4) receives the hot stream from extracted steam (Figure 4.4).

Table 4.7 Steam/Water Properties at FWHEX(4)

Stream	*Notes*	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T $(°C)$	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream,inlet	2.554	1.94	119.2	2638.5	6.9680	527.176
H,out	Hot stream,outlet	2.554	1.79	92.1	386	1.2168	23.0403
C,in	Cold stream, inlet	53.145	10.5	87.2	366.1	1.1594	20.5899
C,out	Cold stream, outlet	53.145	10.10	112.9	474.2	1.4498	40.4083

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} = 2.554\ \mathrm{Kg/s} \quad , \sum_{H,out} \dot{m} = \dot{m}_{H,out} = 2.554\ \mathrm{Kg/s}$$

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 53.145\ \mathrm{Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 53.145\ \mathrm{Kg/s}$$

Then the mass conservation achieved.

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = 2.554 * 2638.5 + 53.145 * 366.1 - (2.554 * 386 + 53.145 * 474.2) = 7.91Kw$$

$$\dot{Q}_{loss} = 7.91Kw$$

Exergy balance

$$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destroyed} = \left(\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{C,in}\psi_{C,in}\right) - \left(\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out}\right)$$

$$\dot{X}_{destroyed} = (2.554 * 527.176 + 53.145 * 20.5899) - (2.554 * 23.0403 + 53.145 * 40.4083)$$

$$I = \dot{X}_{destroyed} = 234.3KW$$

$$Exergy\ supply = \dot{m}_{H,in}(\psi_{H,in} - \psi_{H,out}) = 2.554 * (527.176 - 23.0403)$$

$$Exergy\ supply = 1287.5\ kw$$

$$Exergy\ recovery = \dot{m}_{C}\left(\psi_{C,out} - \psi_{C,in}\right) = 53.145 * (40.4083 - 20.5899)$$

$$Exergy\ recovery = 1053.2\ Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{1053.2}{1287.5} = 0.818 = 81.8\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = (2.554 * 1.2168 + 53.145 * 1.4498) + \frac{7.91}{304} - (2.554 * 6.9680 + 53.145 * 1.1594)$$

$$\dot{S}_{gen} = 0.770763 \frac{\text{Kj}}{\text{Kg. K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 0.770763 = 234.3\ Kw$$

4.4.1.3.5 Feed water heat exchanger (5)

The feed water heat exchanger (5) (hereafter FWHX(5)) receives the hot stream from extracted steam (Figure 4.5).

Table 4.8 Steam/Water Properties at FWHEX(5)

Stream	*Notes*	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	s $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	3.562	12.67	261.7	2959.7	6.8529	883.366
H,in,T	Hot stream, Trap	4.421	24.14	189.1	804.8	2.2254	135.226
H,out	Hot stream, outlet	7.983	12.11	157.7	665.6	1.919	89.1715
C,in	Cold stream, inlet	64.538	113	152.7	650.3	1.8573	92.6283
C,out	Cold stream, outlet	64.538	112.50	184.1	786.3	2.1651	135.057

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} + \dot{m}_{H,in,T} = 3.562 + 4.421 = 7.983 \text{ Kg/s} \ ,$$

$$\sum_{H,out} \dot{m} = \dot{m}_{H,out} = 7.983 \text{ Kg/s}$$

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 64.538 \text{ Kg/s} \ , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 64.538 \text{ Kg/s}$$

Then the mass conservation achieved

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,T}h_{H,in,T} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,T}h_{H,in,T} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = (3.562 * 2959.7 + 4.421 * 804.8 + 64.538 * 650.3) - (7.983 * 665.6 + 64.538 * \quad 786.3)$$

$$\dot{Q}_{loss} = 9.82 \ KW$$

Exergy balance

$$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destroyed} = \left(\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} + \dot{m}_{C,in}\psi_{C,in}\right) - \left(\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out}\right)$$

$$\dot{X}_{destroyed} = (3.562 * 883.366 + 4.421 * 135.226 + 64.538 * 92.6283) - (7.983 * 89.1715 + 64.538 * 135.057)$$

$$I = \dot{X}_{destroyed} = 294.3\ KW$$

$$Exergy\ supply = \dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} - \dot{m}_{H,out}\psi_{H,out}$$

$$Exergy\ supply = 3.562 * 883.366 + 4.421 * 135.226 - 7.983 * 89.1715$$

$$Exergy\ supply = 3032.5\ kw$$

$$Exergy\ recovery = \dot{m}_C(\psi_{C,out} - \psi_{C,in}) = 64.538 * (135.057 - 92.6283)$$

$$Exergy\ recovery = 2738.2\ Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{2738.2}{3032.5} = 0.903 = 90.3\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{H,in,T}S_{H,in,T} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = (7.983 * 1.919 + 64.538 * 2.1651) + \frac{9.82}{304} - (3.562 * 6.8529 + 4.421 * 2.2254 + 64.538 * 1.8573)$$

$$\dot{S}_{gen} = 0.967953\frac{\mathrm{Kj}}{\mathrm{Kg.\ K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 0.967953 = 294.3\ Kw$$

4.4.1.3.6 Feed water heat exchanger (6)

The feed water heat exchanger (6) (hereafter FWHX6) receives the hot stream from extracted steam (Figure 4.4).

Hot stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{H,in} = 4.421\ \mathrm{Kg/s} \quad , \sum_{H,out} \dot{m} = \dot{m}_{H,out} = 4.421\ \mathrm{Kg/s}$$

Table 4.9 Steam/Water Properties at FWHEX(6)

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	4.421	25.09	338.5	3101.5	6.7976	1041.98
H,out	Hot stream, outlet	4.421	24.14	189.1	804.8	2.2254	135.226
C,in	Cold stream, inlet	64.538	112.50	184.1	786.3	2.1651	135.057
C,out	Cold stream, outlet	64.538	112.00	219.4	943.6	2.4962	191.703

Cold stream mass balance

$$\sum_{C,in} \dot{m} = \sum_{C,out} \dot{m}$$

$$\sum_{C,in} \dot{m} = \dot{m}_{C,in} = 64.538\ \mathrm{Kg/s} \quad , \sum_{C,out} \dot{m} = \dot{m}_{C,out} = 64.538\ \mathrm{Kg/s}$$

Then the mass conservation achieved

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = 4.421 * 3101.5 + 64.538 * 786.3 - (4.421 * 804.8 + 64.538 * 943.6) = 1.88Kw$$

$$\dot{Q}_{loss} = 1.88Kw$$

Exergy balance

$$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destroyed} = \left(\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{C,in}\psi_{C,in}\right) - \left(\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out}\right)$$

$$\dot{X}_{destroyed} = (4.421 * 1041.98 + 64.538 * 135.057) - (4.421 * 135.226 + 64.538 * 191.703)$$

$$I = \dot{X}_{destroyed} = 352.9KW$$

$$Exergy\ supply = \dot{m}_{H,in}(\psi_{H,in} - \psi_{H,out}) = 4.421 * (1041.98 - 135.226)$$

$$Exergy\ supply = 4008.7\ kw$$

$$Exergy\ recovery = \dot{m}_C\left(\psi_{C,out} - \psi_{C,in}\right) = 64.538 * (191.703 - 135.057)$$

$$Exergy\ recovery = 3655.8Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{3655.8}{4008.7} = 0.912 = 91.2\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = (4.421 * 2.2254 + 64.538 * 2.4962) + \frac{1.88}{304} - (4.421 * 6.7976 + 64.538 * 2.1651)$$

$$\dot{S}_{gen} = 1.1610198 \frac{\text{Kj}}{\text{Kg. K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 1.1610198 = 352.9\ Kw$$

4.4.1.4 Deaerator (an open heat exchanger)

Deaerator is basically a mixing chamber, where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure (Figure 4.3).

Table 4.10 Steam/Water Properties Deaerator

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
H,in	Hot stream, inlet	3.889	5.05	166.2	2780	6.8928	691.5363
H,in,T	Hot stream, Trap	7.983	12.11	157.7	665.6	1.919	89.17152
H,out	Hot stream, outlet	0.01	4.85	150.7	2746.7	6.8308	677.0843
C,in	Cold stream, inlet	53.145	10.10	112.9	474.2	1.4498	40.40832
C,out	Cold stream, outlet	65.007	4.85	150.7	635.2	1.8489	80.08192

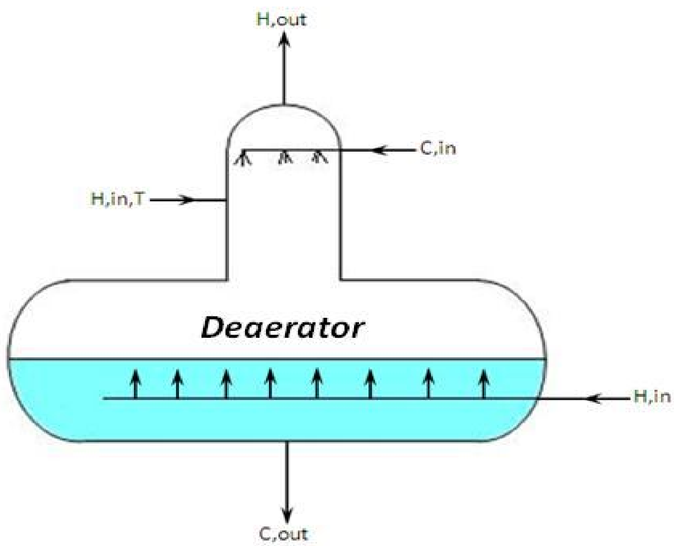


Figure 4.6 Deaerator Streams

Mass balance

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

$$\sum_{in} \dot{m} = \dot{m}_{H,in} + \dot{m}_{H,in,T} + \dot{m}_{C,in} = 3.889 + 7.983 + 53.145 = 65.017 \text{ Kg}/\text{s} \ ,$$

$$\sum_{out} \dot{m} = \dot{m}_{H,out} + \dot{m}_{C,out} = 0.01 + 65.007 = 65.017 \text{ Kg}/\text{s}$$

Then the mass conservation achieved.

Energy balance

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

$$\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,T}h_{H,in,T} + \dot{m}_{C,in}h_{C,in} = \dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out} + \dot{Q}_{loss}$$

$$\dot{Q}_{loss} = (\dot{m}_{H,in}h_{H,in} + \dot{m}_{H,in,T}h_{H,in,T} + \dot{m}_{C,in}h_{C,in}) - (\dot{m}_{H,out}h_{H,out} + \dot{m}_{C,out}h_{C,out})$$

$$\dot{Q}_{loss} = (3.889 * 2780 + 7.983 * 665.6 + 53.145 * 474.2) - (0.01 * 2746.2 + 65.007 * 635.2)$$

$$\dot{Q}_{loss} = 6.4\ KW$$

Exergy balance

$$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$$

$$\dot{X}_{destroyed} = \left(\dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} + \dot{m}_{C,in}\psi_{C,in}\right) - \left(\dot{m}_{H,out}\psi_{H,out} + \dot{m}_{C,out}\psi_{C,out}\right)$$

$$\dot{X}_{destroyed} = (3.889 * 691.5363 + 7.983 * 89.17152 + 53.145 * 40.40832) - (0.01 * 677.0843 + 65.007 * 80.08192)$$

$$I = \dot{X}_{destroyed} = 336\ KW$$

$$Exergy\ supply = \dot{m}_{H,in}\psi_{H,in} + \dot{m}_{H,in,T}\psi_{H,in,T} - \dot{m}_{H,out}\psi_{H,out}$$

$$Exergy\ supply = 3.889 * 691.5363 + 7.983 * 89.17152 - 0.01 * 677.0843$$

$$Exergy\ supply = 3394.4\ kw$$

$$Exergy\ recovery = \dot{m}_{C,out}\psi_{C,out} - \dot{m}_{C,in}\psi_{C,in}$$

$$Exergy\ recovery = 65.007 * 80.0819 - 53.145 * 40.40832$$

$$Exergy\ recovery = 3058\ .4Kw$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{3058\ .4}{3394.4} = 0.901 = 90.1\%$$

Entropy balance

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = \dot{m}_{H,out}S_{H,out} + \dot{m}_{C,out}S_{C,out} + \frac{\dot{Q}_{loss}}{T_0} - (\dot{m}_{H,in}S_{H,in} + \dot{m}_{H,in,T}S_{H,in,T} + \dot{m}_{C,in}S_{C,in})$$

$$\dot{S}_{gen} = (0.01 * 6.8308 + 65.007 * 1.8489) + \frac{6.4}{313} - (3.889 * 6.8928 + 7.983 * 1.919 + 53.145 * 1.4498)$$

$$\dot{S}_{gen} = 1.1051 \frac{\mathrm{Kj}}{\mathrm{Kg.\ K}}$$

$$\dot{I} = T_0\dot{S}_{gen} = 304 * 1.1051 = 336\ Kw$$

4.4.1.5 Boiler

The boiler system of the plant will be discussed in this calculation as two subsystems. The first subsystem is Burner where fuel combusted. The burner has two inlet streams which are fuel stream and air stream, also has one outlet stream which conveys hot products of combustion to a Heat exchanger which is considered as second subsystem. The Heat exchanger subsystem is closed type heat exchanger. It contains two inlet streams, one of them is the hot products stream and the other is feed water stream. Also heat exchanger has two outlet streams which are steam outlet stream and Flue gases outlet stream.***As boiler system is considered reacting system, its analysis is based on section 2.8 and equations (2.22, 2.23 &2.24) are taken into account.***

Table 4.11 Burner inlet streams properties

	T_{in} (K)	P_{in} (bar)
Fuel	320	16
Air	598	0.077

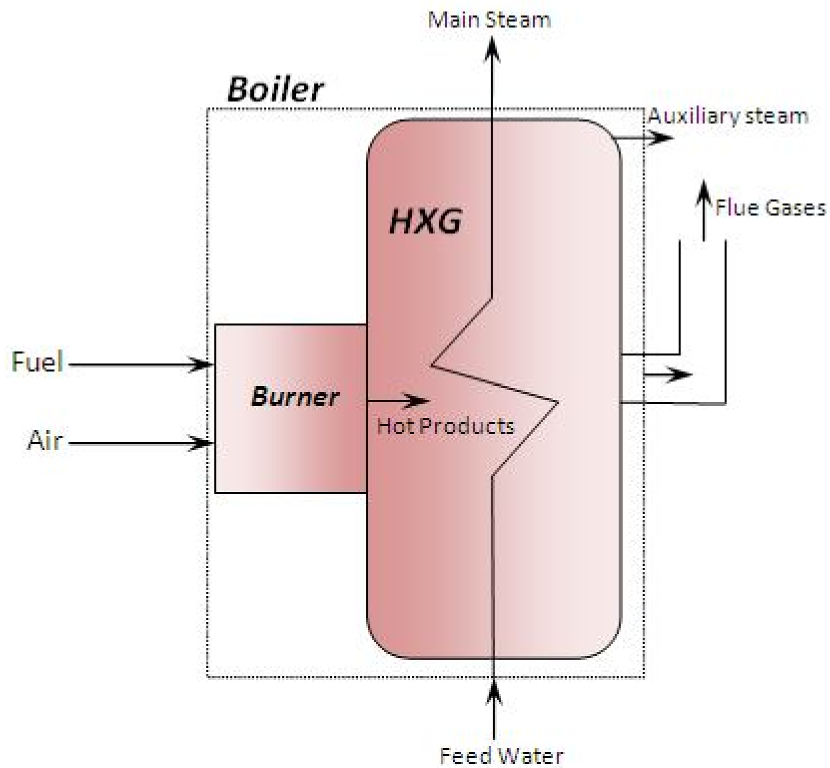


Figure 4.7 Boiler streams and its two subsystems Burner & Heat Exchanger (HXG)

Table 4.12 Fuel Components Properties

Fuel Components	%	$\dot{m}\ (\frac{Kg}{s})$	$M(\frac{Kg}{Kmole})$	$N(Kmole)$
C	88.41	3.18276	12	0.26523
H	11.37	0.40932	1	0.40932
N	0.22	0.00792	14	0.0005657

Fuel combustion equation

$$(0.26523\,C + 0.40932\,H + 0.0005657\,N) + A_{th}(O_2 + 3.76\,N2) = 0.26523\,CO_2 + 0.20466\,H_2O(g) + \left(\frac{0.0005657}{2} + 3.76A_{th}\right)N_2$$

Moles balance for Oxygen (O_2)

$$A_{th} = 0.26523 + \frac{0.20466}{2} = 0.36756$$

$$(0.26523\,C + 0.40932\,H + 0.0005657\,N) + 0.36756\,(O_2 + 3.76\,N2) = 0.26523\,CO_2 + 0.20466\,H_2O(g) + 1.3823\,N_2$$

Table 4.13 Fuel Properties at T=320K and P=16bar

	Fuel properties at T=320 K and P=16 bar							
Fuel Comp.	$\dot{N}$ $\left(\frac{Kmole}{s}\right)$	h_{f298} $\left(\frac{Kj}{Kmole}\right)$	$h_{320} - h_{f298}$ $\left(\frac{Kj}{Kmole}\right)$	S $\left(\frac{Kj}{Kmole.K}\right)$	$R_u \ln\left(\frac{P}{P_o}\right)$ $\left(\frac{Kj}{Kmole.K}\right)$	H_{Total} (Kw)	S_{Total} (Kw/K)	$\dot{\psi}$ $\left(\frac{Kmole}{s}\right)$
C	0.26523	716780	132	158.1	22.9418656	190146.57	35.847992	179248.7802
H	0.40932	217346	453.8	115.929	22.9418656	89149.8141	38.0614939	77579.12
N	0.00056571	470783	453.8	154.513	22.9418656	266.58539	0.07443167	243.9581619
						279562.969	73.9839175	257071.8584

Total enthalpy of fuel at inlet of burner

$$h_{fuel} = 190146.57 + 89149.8141 + 266.58539 = 279562.969\ Kw$$

$$h_{fuel} = 279.6\ Mw$$

Total exergy of fuel at inlet of burner

$$\psi_{fuel} = 179248.7802 + 77579.12 + 243.9581619 = 257071.8584 \text{ Kw}$$

$$\psi_{fuel} = 257.1 \text{ Mw}$$

Table 4.14 Air properties at T=598 K and P=77 mbar

	Air properties at T=598 K and P=77 mbar							
Air Comp.	$\dot{N}$ $\left(\frac{Kmole}{s}\right)$	h_{f298} $\left(\frac{Kj}{Kmole}\right)$	$h_{598} - h_{f298}$ $\left(\frac{Kj}{Kmole}\right)$	S $\left(\frac{Kj}{Kmole\,.K}\right)$	$R_u \ln\left(\frac{P}{P_o}\right)$ $\left(\frac{Kj}{Kmole\,.K}\right)$	H_{Total} (Kw)	S_{Total} (Kw/K)	$\dot{\psi}$ $\left(\frac{Kmole}{s}\right)$
O_2	0.36756	0	9190	226.259	-21.4261162	3377.88	91.039141	-24298
N_2	1.382026	0	8845.3	211.994	-21.4261162	12224.4	322.59258	-85844
						15602.3	413.63172	-110142

Total enthalpy of fuel at inlet of burner

$$h_{Air} = 3377.88 + 12224.4 = 15602.3\text{Kw}$$

$$h_{Air} = 15.6 \text{ Mw}$$

Total exergy of fuel at inlet of burner

$$\psi_{Air} = (-24298) + (-85844) = -110142 \text{ Kw}$$

$$\psi_{Air} = -110.1\text{Mw}$$

Enthalpy of reactants at inlet of burner

$H_r = h_{fuel} + h_{Air}$

$H_r = 279.6 \ + 15.6$

$$H_r = 295.2 \ Mw$$

Exergy of reactants at inlet of burner

$\psi_r = \psi_{fuel} + \psi_{Air}$

$\psi_r = 257.1 + (-110.1) = 147$ Mw

Energy balance

Assumption is considered that the burner is adiabatic system

$H_p = H_r$

$H_p = 295.2 \ Mw$

Table 4.15 Product Components properties at unknown flame temperature

Fuel Comp.	$\dot{N}$ $\left(\frac{Kmole}{s}\right)$	h_{f298} $\left(\frac{Kj}{Kmole}\right)$	$h_T - h_{f298}$ $\left(\frac{Kj}{Kmole}\right)$
CO_2	0.26523	-393546	Δh_{CO2}
H_2O	0.20466	44010	Δh_{H_2O}
N_2	1.3823	0	Δh_{N2}

$$H_p = 0.26523 * (-393546 + \Delta h_{CO2}) + 0.20466 * \left(44010 + \Delta h_{H_2O}\right) + 1.3823 * \Delta h_{N2} = 295200$$

$0.26523 * \Delta h_{CO2} + 0.20466 * \Delta h_{H_2O} + 1.3823 * \Delta h_{N2} = 390573.1$

$T_p \cong 5000 \ K$

$$\Delta h_{CO2} = 279313 \frac{Kj}{Kmol}$$

$$\Delta h_{H_2O} = 242343 \frac{Kj}{Kmol}$$

$$\Delta h_{N2} = 167778 \frac{Kj}{Kmol}$$

Table 4.16 Product Components at T=5000 K and P=16 bar

Products	$\dot{N}$ $\left(\frac{Kmole}{s}\right)$	h_{f298} $\left(\frac{Kj}{Kmole}\right)$	$h_{5000} - h_{f298}$ $\left(\frac{Kj}{Kmole}\right)$	S $\left(\frac{Kj}{Kmole.K}\right)$	$R_u \ln\left(\frac{P}{P_o}\right)$ $\left(\frac{Kj}{Kmole.K}\right)$	H_{Total} (Kw)	S_{Total} (Kw/K)	$\dot{\psi}$ $\left(\frac{Kmole}{s}\right)$
CO_2	0.26523	-393546	279313	366.372	22.9418656	-30298.0186	91.0879746	-57988.7629
H_2O	0.20466	44010	242343	315.915	22.9418656	58605.00498	59.9598817	40377.20095
N_2	1.3823	0	167778	285.958	22.9418656	231919.5294	363.567203	121395.0998
						260226.5158	514.615059	103783.5379

Total enthalpy of Hot products at outlet of burner

$$H_p = (-30298.0186) + 58605.00498 + 231919.5294 = 260226.5158 Kw$$

$$H_p = 260.2\ Mw$$

Total exergy of fuel at inlet of burner

$$\psi_p = (-57988.7629) + 40377.20095 + 121395.0998 = 103783.5379 \text{Kw}$$

$$\psi_p = 103.8 \text{ Mw}$$

$$\dot{X}_{destroyed} = \psi_r - \psi_p = 147 - 103.8 = 43.2 \text{ Mw}$$

Heat exchanger (2nd sub-system)

Table 4.17 Steam/Water at Boiler Heat Exchanger

$Stream$	$Notes$	$\dot{m}$ $(\frac{kg}{s})$	P (bar)	T (°C)	h $(\frac{Kj}{kg\ K})$	S $(\frac{Kj}{kg\ K})$	ψ $(\frac{Kj}{kg})$
MS	main steam	64.289	89.50	512.5	3418.4	6.70361	1387.45
AS	auxiliary steam	0.249	18	210	2803.3	6.3958	865.9243
FW	Feed water	64.538	112	219.4	943.6	2.4962	191.7027

$$\dot{X}_{destroyed} = \dot{\psi}_p + 64.538 * 191.7027 - (64.289 * 1387.45 + 0.249 * 865.9243 + \dot{\psi}_{fg})$$
$$H_{fg} = 64.538 * 943.6 + 260200 - (64.289 * 3418.4 + 0.249 * 2803.3)$$

Cold stream mass balance

$$\sum_{H,in} \dot{m} = \sum_{H,out} \dot{m}$$

$$\sum_{H,in} \dot{m} = \dot{m}_{fw} = 64.538 \text{ Kg/s} \quad , \sum_{H,out} \dot{m} = \dot{m}_{ms} + \dot{m}_{as}$$
$$= 64.289 + 0.249 = 64.538 \text{ Kg/s}$$

Then mass conservation achieved

Energy balance

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

$$\dot{m}_{fw}h_{fw} + H_p = \dot{m}_{ms}h_{ms} + \dot{m}_{as}h_{as} + H_{fg}$$

$$H_{fg} = \dot{m}_{fw}h_{fw} + H_p - (\dot{m}_{ms}h_{ms} + \dot{m}_{as}h_{as})$$

$$H_{fg} = 64.538 * 943.6 + 260200 - (64.289 * 3418.4 + 0.249 * 2803.3)$$

$$H_{fg} = 100.6\ Mw$$

$$H_{fg} = H_p = 0.26523 * (-393546 + \Delta h_{CO2}) + 0.20466 * (44010 + \Delta h_{H_2O}) + 1.3823 * \Delta h_{N2} = 100600$$

$$0.26523 * \Delta h_{CO2} + 0.20466 * \Delta h_{H_2O} + 1.3823 * \Delta h_{N2} = 195973.119$$

$$T_{fg} \cong 3000\ K$$

$$\Delta h_{CO2} = 152891 \frac{Kj}{Kmole}$$

$$\Delta h_{H_2O} = 126563 \frac{Kj}{Kmole}$$

$$\Delta h_{N2} = 92730 \frac{Kj}{Kmole}$$

Table 4.18 Flue gases at T=3000 K and P=16 bar

Flue gases	$\dot{N}$ $\left(\frac{Kmole}{s}\right)$	h_{f298} $\left(\frac{Kj}{Kmole}\right)$	$h_{3000}-h_{f298}$ $\left(\frac{Kj}{Kmole}\right)$	S $\left(\frac{Kj}{Kmole.K}\right)$	$R_u \ln\left(\frac{P}{P_o}\right)$ $\left(\frac{Kj}{Kmole.K}\right)$	H_{Total} (Kw)	S_{Total} (Kw/K)	$\dot{\psi}$ $\left(\frac{Kmole}{s}\right)$
CO_2	0.26523	-400140	152891	334.124	22.9418656	-65577.8523	82.5348375	-90668.4429
H_2O	0.20466	44010	126563	286.42	22.9418656	34909.47018	53.923435	18516.74594
N_2	1.3823	0	92730	266.81	22.9418656	128180.679	337.098922	25702.60664
						97512.29691	473.557195	-46449.0903

$$\dot{X}_{destroyed} = 103783.5379 + 64.538 * 191.7027 - (64.289 * 1387.45 + 0.249 * 865.9243 - 46449.0903)$$

Total enthalpy of flue gases at outlet of heat exchanger

$H_{fg} = (-65577.8523) + 34909.47018 + 128180.679 = 97512.29691 Kw$

$$H_{fg} = 97.5\ Mw$$

Total exergy of flue gases at outlet of heat exchanger

$\psi_{fg} = (-90668.4429) + 18516.74594 + 25702.60664 = -46449.0903\text{Kw}$

$\psi_{fg} = -46.4\ \text{Mw}$

$\dot{X}_{destroyed} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi$

$\dot{X}_{destroyed} = \dot{\psi}_p + \dot{m}_{fw}\psi_{fw} - (\dot{m}_{ms}\psi_{ms} + \dot{m}_{as}\psi_{as} + \dot{\psi}_{fg})$

$\dot{X}_{destroyed} = 103783.5379 + 64.538 * 191.7027 - (64.289 * 1387.45 + 0.249 * 865.9243 - 46449.0903)$

$$\dot{X}_{destroyed} = 73.2\ Mw$$

$Exergy\ supply = \dot{\psi}_p - \dot{\psi}_{fg} = 103783.5379 - (-46449.0903)$
$= 150232.6282\ \text{Kw} = 150.2\ \text{Mw}$

$Exergy\ recovery = \dot{m}_{ms}\psi_{ms} + \dot{m}_{as}\psi_{as_p} - \dot{m}_{fw}\psi_{fw}$
$= 64.289 * 1387.45 + 0.249 * 865.9243 - 64.538 * 191.7027$
$= 77041.28\ \text{Kw} = 77\ \text{Mw}$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{77}{150.2} = 0.513 = 51.3\%$$

The whole Boiler system

$\dot{X}_{destroyed} = \dot{\psi}_r + \dot{m}_{fw}\psi_{fw} - (\dot{m}_{ms}\psi_{ms} + \dot{m}_{as}\psi_{as} + \dot{\psi}_{fg})$

$$\dot{X}_{destroyed} = 147000 + 64.538 * 191.7027 - (64.289 * 1387.45 + 0.249 * 865.9243 - 46449.0903)$$

$$\dot{X}_{destroyed} = 116407.8kw = 116.4\ Mw$$

$$Exergy\ supply = \dot{\psi}_r - \dot{\psi}_{fg} = 147000 - (-46449.0903) =$$

$$193449.0903 \text{ Kw} = 193.4 \text{ Mw}$$

$$Exergy\ recovery = \dot{m}_{ms}\psi_{ms} + \dot{m}_{as}\psi_{as} - \dot{m}_{fw}\psi_{fw}$$
$$= 64.289 * 1387.45 + 0.249 * 865.9243 - 64.538 * 191.7027$$
$$= 77041.28 \text{ Kw} = 77 \text{ Mw}$$

$$\eta_{II} = \frac{Exergy\ recovery}{Exergy\ supply} = \frac{77}{193.4} = 0.398 = 39.8\%$$

4.4.2Exergy Analysis Results and Discussion

After calculating of exergy, irreversibility and second law efficiency for all main systems in KNPSP(2), it is time to analyze results are calculated through equations such as mass, exergy, energy balance.

Methodology of this analysis is considered discussing of any variable such irreversibility in each system according to magnitude of variable and its influence to using energy quality and the total plant efficiency.

Calculated results table below contains outcomes of variables under consideration in discussion for the ten main components in KNPSP(2) from Boiler to FWHEX(6).

As it is well know that Boiler is considered the main energy source in power plant cycles. Also, turbine is considered as work-producing device where energy which is high pressurized and superheated steam (working fluid) received from boiler is converted to mechanical work. The condenser comes after turbine to condensate the steam and recycling the water to boiler. But there are some heat exchangers are put in way of recycled water from condenser to boiler so as to warm up feed water to minimize heat energy is needed in boiler to achieve higher possible thermal efficiency of power plants.

Now what variables are needed to evaluate performance of power plant in accordance with Quantity and Quality of energy. In this research, the selected variables are Heat loss as Energy Quantity analysis; also Exergy, Irreversibility and Second law efficiency as Energy Quality analysis.

Table 4.19 Exergy analysis Results

#	*Component*	$\dot{Q}_{loss}$ *(Kw)*	*Exergy Supply (Kw)*	*Exergy recovery (Kw)*	*I (Kw)*	*Exergy – out (Kw)*	$\eta_{II}(\%)$
1	Boiler	0	193449.0903	77041.28	116407.8	89197.77305	39.8%
2	Turbine	1476.8	70300	60000	10300	2794.971356	85.3%
3	Condenser	3066.812	2817.1	361.3	2455.8	27.53081064	12.8%
4	FWHEX(1)	0.5863	29.4	-55.46	84.86	86.1544224	-188.6%
5	FWHEX(2)	5.113	20.9	4.7	16.2	90.8736984	22.5%
6	FWHEX(3)	626.56	1793.1	1003.4	789.7	1094.250236	56.0%
7	FWHEX(4)	7.91	1287.5	1053.2	234.3	2147.499104	81.8%
8	Daerator	6.4	3394.4	3058.4	336	5205.885373	90.1%
9	FWHEX(5)	9.82	3032.5	2738.2	294.3	8716.308666	90.3%
10	FWHEX(6)	1.88	4008.7	3655.3	352.9	12372.12821	91.2%

4.4.2.1 Energy /Quantity Analysis (Heat Losses)

Results table figures out that the heat loss maximum is 3066.812 KW in condenser and minimum is 0.5863 Kw in FWHEX(1), regardless boiler due to adiabatic assumption takes place.

The result of maximum amount of heat loss is at condenser is acceptable in accordance with Kelvin–Planck statement of the second law of thermodynamics (It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work).however, heat loss should be as minimum as possible to increase the first and second low efficiency of power plant, so such as Turbine heat loss (1476.8 Kw) and FWHEX(3) (626.56 Kw) should be decreased to enhance increase of power plant efficiency. Thus, Turbine and FWHEX(3) where Heat loss analysis must be taken in to account where redesign or revaluation to improve performance of the whole power plant.

4.4.2.2 Exergy /Quality Analysis

Quality analysis is one of important power plant analysis where identifies the maximum useful work will be gotten from a system. The tool of this Quality analysis for power plant is Exergy. In this research, Exergy analysis is discussed as Exergy supply, Exergy recovery, Exergy out, Exergy destruction (Irreversibility) and second law efficiency.

Exergy supply is the potential maximum useful work supplied to a system and the amount of potential maximum useful work recovered by the system expresses Exergy recovery. But work lost in system is defined as Exergy destruction or Irreversibility. Also, second law efficiency is considered as maximum thermal efficiency (Ranking cycle efficiency).

Where Exergy Supply, Exergy recovery and Exergy destruction are considered as a power plant component or system performance index, however, the criterion for working fluid along power plant cycle is consider Exergy-out. Exergy-out identifies the maximum potential useful work out of a component or system.

Referring to results table, the maximum exergy supply is 193.4 Mw at boiler and the minimum is 20.9 Kw at FWHEX(2). Both FWHEX(1) and FWHEX(2) has negligible amount of exergy supply comparing with other all components.

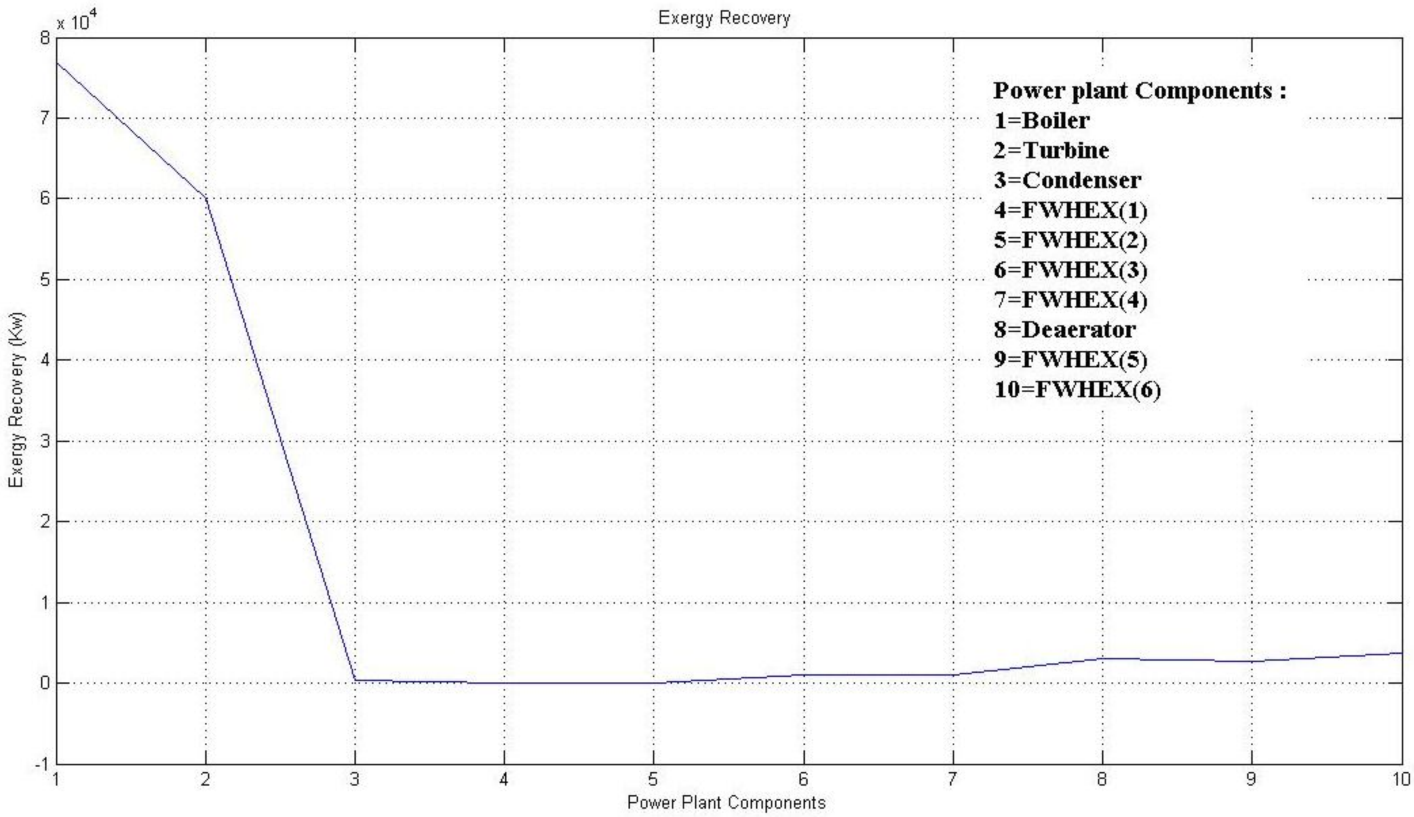


Figure 4.8 Exergy Recovery at each KNPSP(2) components

On the other hands, the maximum exergy recovery is 77 Mw at boiler(Figure 4.8) and the minimum is -55.46 Kw at FWHEX(1) where minus sign is refer to exergy decreasing which reverse to FWHEX(1) function where exergy-out must be higher than exergy-in to has positive exergy recovery sign. Also, FWHEX(2) has a positive exergy recovery but is negligible amount comparing with other all components except FWHEX(1).

Referring to exergy destruction, the maximum amount is 116.4 Mw at boiler (Figure 4.9) and the minimum amount is 16.2 Kw at FWHEX(2).In addition, Turbine and condenser consider have high exergy destruction are 10.2 Mw and 2.5 Mw respectively. The exergy destruction (Irreversibility) is deviation from reversible process which means the smaller exergy destruction the better performance of component.

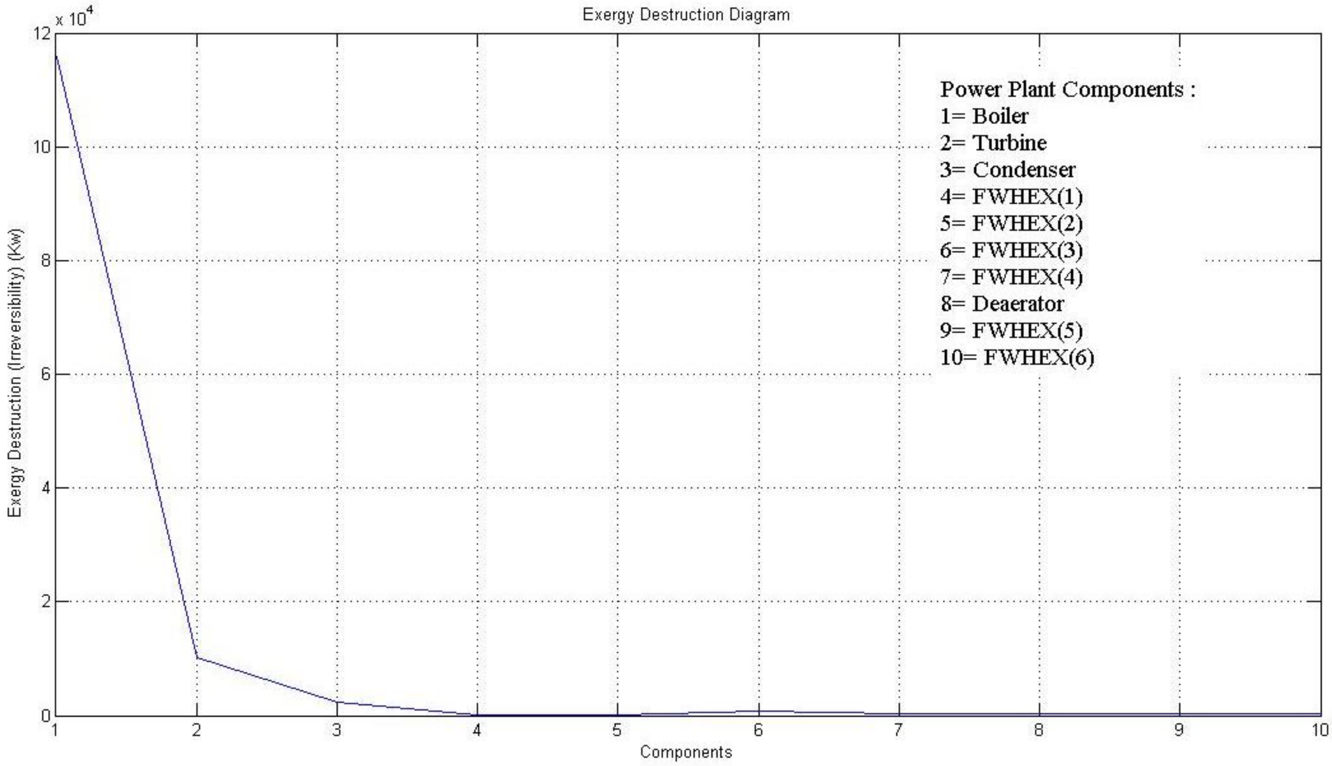
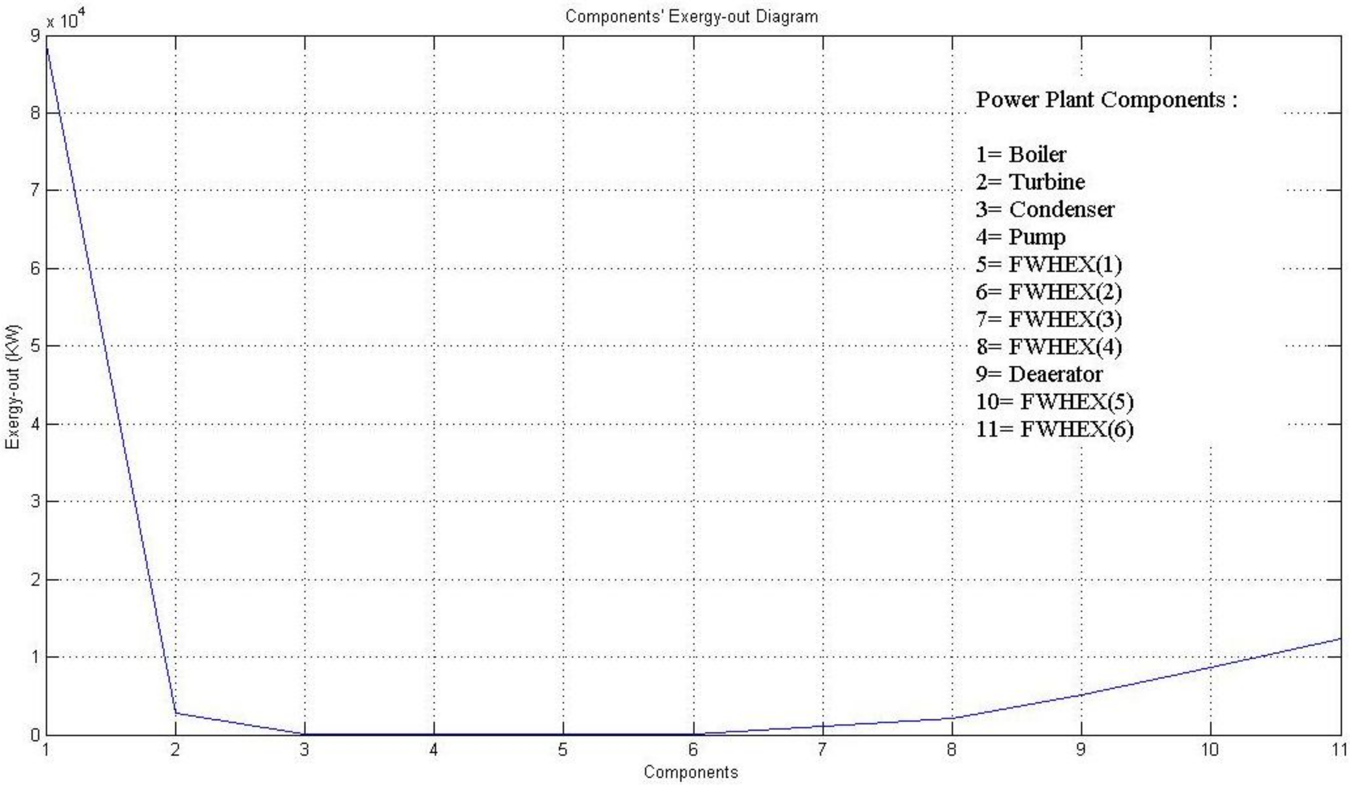


Figure 4.9 Exergy Destruction at KNPSP(2) components

According to second law efficiency, maximum percentage is 91.2% at FWHEX(6) and minimum percentage is 12.8% at condenser. These percentage means the thermal efficiency and performance must be maximum at FWHEX(6) and minimum at condenser. However, FWHEX(1) second law efficiency is -188.6% which means exergy destruction of this component higher than exergy supply by 188.6% so that exergy recovery gets minus sign. Then, based on this point of view FWHEX(1) is consider the worst component in this power plant where thermal performance.

Now after evaluation of components, there is need to evaluate their affect on the whole cycle of power plant. The maximum exergy-out is 89.1 Mw at outlet of Boiler and the minimum exergy-out is 27.5 Kw at outlet of condenser{Table 4.3($\dot{m}\psi_{H,out}$)}. For optimum thermal performance exergy-out must has a maximum at boiler outlet and minimum at condenser outlet after that must be increased along other component until FWHEX(6).but the exergy-out circulating pump (which is considered out of this research scope) is 141.6 Kw which is equal to exergy-in FWHEX(1), then according to optimum status exergy-out FWHEX(1) must be higher than 141.6 Kw, however, it is 86.2 Kw which means there is exergy-out drop takes place at FWHEX(1) after condenser outlet referring to break optimum thermal performance condition of power plant (Figure 4.10).

Figure 4.10 Exergy-out at each KNPSP(2) components



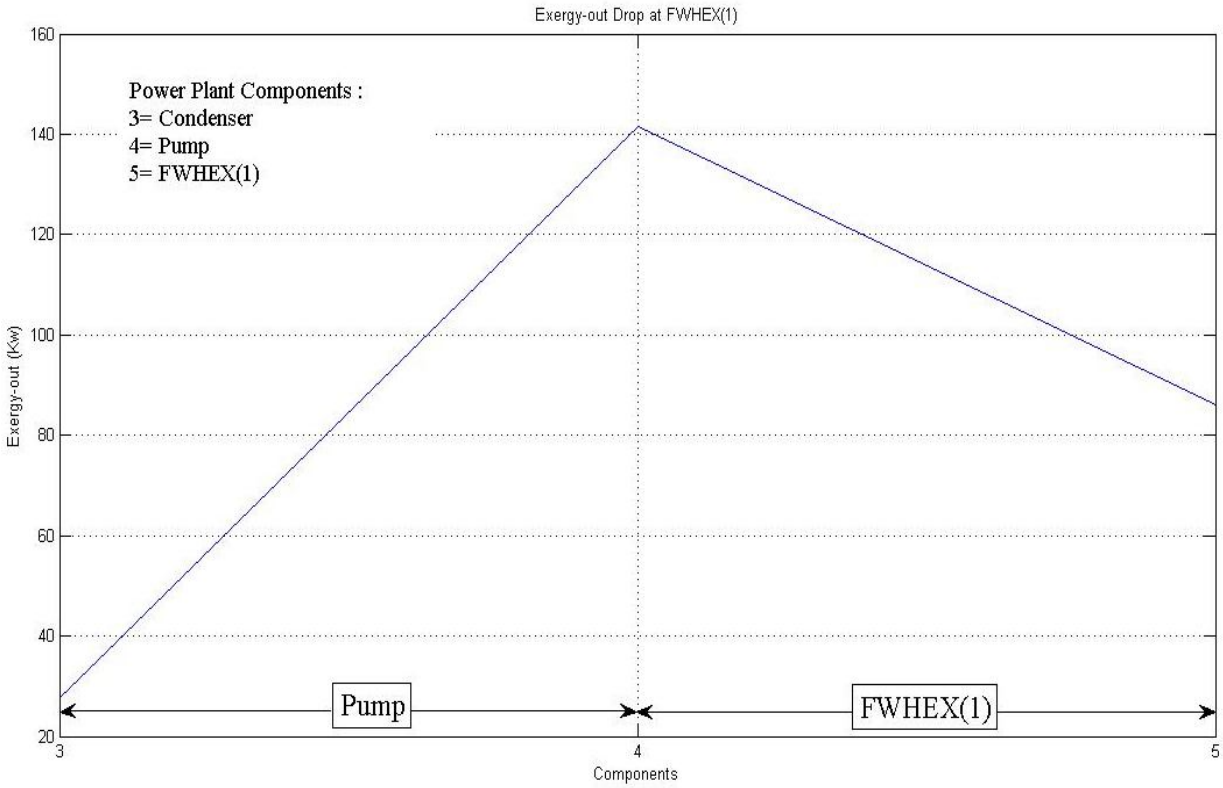


Figure 4.10a Zooming-in at Exergy-out diagram shows Exergy-out drop at FWHEX(1)

It could be concluded as FWHEX(1) and FWHEX(2) must be redesigned and re-evaluated according to Energy Quality analysis (Exergy) to enhance better thermal performance for the power plant. Also FWHEX(3) must be taken in to account in redesign according to Energy Quantity analysis (Heat loss).

Eventually ,this power plant , KNPSP(2), its total irreversibility according to this research scope is 131.3 Mw; which means that is, 131.3 Mw of work potential is wasted during these power plant processes. In other words, an additional 131.3 Mw of energy could have been converted to work during these processes, but was not.

4.5 Pinch analysis

Refer to chapter three which discussed the pinch technology and analysis. This suction is applying the Pinch analysis tool on part of the pant which is regenerative feed water heat exchanger. This application is to evaluate this part of plant in accordance with maximum utilization.

4.5.1 Data collection

To apply pinch analysis, the regenerative feed water heat exchangers should be separated as two groups. The first group includes four closed exchangers before deaerator. Second group includes two closed exchangers after deaerator. This separation due to the pinch technology based on heat exchanging between hot and cold streams, however, deaerator is open heat exchanger and the hot and cold streams mixing takes place.

In those two groups of regenerative feed water heat exchanger there are mixing processes take place. These mixing processes happen between extracted steam comes from turbine as hot stream for a closed heat exchanger and trap stream comes out a heat exchanger as additional hot stream for the same closed heat exchanger. In this case the properties of the hot stream will be taken for the mix of those two hot streams. This special case will be applied at feed water heat exchanger 3 and 5 in this plant. also there is no contradiction with pinch technology bases where mixing processes take place between Hot-Hot streams not Hot-Cold streams as contradicted with pinch technology bases.

Before start analysis the specific heat will be calculated in all cases as formula below:

$$Cp = \frac{\Delta h}{\Delta T}$$

However, it is important to know in pinch analysis for each stream the phase of working fluid (Water) at inlet and outlet so as to know if heat exchanging process includes latent heat which happens as phase changing in stream takes place. This point is considered as crucial essential matter when pinch analysis calculation applied in heat exchangers net work.

4.5.2 First Group Pinch analysis

Table 4.20 First group data collection

$Streams$	$\dot{m}\ (\frac{Kg}{s})$	T_s (°C)	h_s (Kj/Kg)	T_t (°C)	h_t (Kj/Kg)
cold stream	53.145	40.1	170	112.9	474.2
Hot stream (1)	0.063	163.8	2803.3	43	180
Hot stream (2)	0.035	336.1	3147.4	99.8	420
Hot stream (3)	4.183	94.8	2666.83	47.6	199.3
Hot stream (4,trap)	2.554	92.1	386	47.6	199.3
Hot stream (4)	2.554	119.2	2638.5	92.1	386

Now mixed hot stream property which is mixing of Hot stream (3) and Hot stream (4,trap) is being calculated by using mass and Energy balance.

Mass balance

$$\dot{m}_{H,3} + \dot{m}_{H,4,\text{trap}} = \dot{m}_{H,3,mix}$$

$$4.183 + 2.554 = \dot{m}_{H,3,mix}$$

$$\dot{m}_{H,3,mix} = 6.737\ kg/s$$

Energy balance

$$\dot{m}_{H,3}h_{H,3} + \dot{m}_{H,4,\text{trap}}h_{H,4,\text{trap}} = \dot{m}_{H,3,mix}h_{H,3,mix}$$

$$h_{H,3,mix} = \frac{\dot{m}_{H,3}h_{H,3} + \dot{m}_{H,4,\text{trap}}h_{H,4,\text{trap}}}{\dot{m}_{H,3,mix}}$$

$h_{H,3}\ and\ h_{H,4,\text{trap}}\ values\ should\ be\ substituted\ from\ table\ 4.20$

$$h_{H,3,mix} = \frac{4.183 * 2666.83 + 2.554 * 386}{6.737}$$

$$h_{H,3,mix} = 1802.166\ kj/kg$$

$For\ mixed\ hot\ stream\ temperature\ is\ selected\ as\ average\ of\ two\ hot\ streams$

$$T_{H,3,mix} = \frac{\mathrm{T_{H,3}} + \mathrm{T_{H,4,Trap}}}{2} = \frac{94.5 + 92.1}{2} = 93.3\ ℃$$

$then\ from\ tables\ at\ T_{H,3,mix} = 93.3\ ℃\ and\ h_{H,3,mix} = 1802.166\ kj/kg$

$$P_{H,3,mix} = 0.795\ \text{bar}$$

$\underline{Dryness\ factor}$

$$x_{H,3,mix} = \frac{h_{H,3,mix} - h_f}{h_{fg}} = \frac{1802.166 - 390.86}{2664.89 - 390.86} = 0.62$$

Table 4.21 First group data collection after mixing calculation

$Streams$	$\dot{m}\ (\frac{Kg}{s})$	T_s (°C)	h_s (Kj/Kg)	T_t (°C)	h_t (Kj/Kg)
Cold stream	53.145	40.1	170	112.9	474.2
Hot stream (1)	0.063	163.8	2803.3	43	180
Hot stream (2)	0.035	336.1	3147.4	99.8	420
Hot stream (3,mix)	6.737	93.3	1802.166	47.6	199.3
Hot stream (4)	2.554	119.2	2638.5	92.1	386

Now, it is essentially to study each hot and cold stream where it includes latent heat or sensible heat only. Thus Table 4.22 is a detailed data table for first group illustrates and elaborates all essential parameters needed for first group analysis.

Table 4.22 First Group Detailed Data

			In					Out					
		$\dot{m}$	$P\ (bar)$	$T_{sat.}$ (°C)	T (°C)	$h \left(\frac{Kj}{Kg}\right)$	state	$P\ (bar)$	$T_{sat.}$ (°C)	T (°C)	$h \left(\frac{Kj}{Kg}\right)$	state	Δh_{latent}
FWHEX (1)	H/S	0.063	1.05	100.97	163.8	2803.625	steam	1	99.6	43	180	water	A
	C/S	53.145	12	187.96	40.1	170	water	11.7	186.82	41.1	173	water	N/A
FWHEX (2)	H/S	0.035	1.05	100.97	336.1	3147.4	steam	1	99.6	99.6	420	mixture	A
	C/S	53.145	11.7	186.82	41.1	173	water	11.4	185.66	41.5	174.7	water	N/A
FWHEX (3)	H/S	6.737	0.795	93.3	93.3	1802.166	mixture	0.729	91	47.6	199	water	A
	C/S	53.145	11.4	185.66	41.5	174.7	water	10.5	182.02	87.2	366.1	water	N/A
FWHEX (4)	H/S	2.554	1.940	119.2	119.2	2638.5	mixture	1.790	116.74	92.1	386	water	A
	C/S	53.145	10.5	182.02	87.2	366.1	water	10.1	181.33	112.9	474.2	water	N/A

As outcome of Table 4.22 aboveall hot streams in first group has latent heat and vice versa.

As it well known regarding what already discussed in chapter 3, there is two ways to identify pinch point; composite curve method and mathematical technique method. In this research, those methods are applied to insure accuracy of outcomes.

4.5.2.1 Composite Curve Method

Temperature – Heat load diagram (T/H) for each feed water heat exchanger must be drawn to have the composite Curve diagram for first group. In drawing of T/H diagram each stream may separate to processes according to phase changing in the stream. Then, as example, may there is a stream start with sensible heat process after that latent heat process then sensible heat process if there is, and so on. Thus each process in the stream has its own load calculation and temperature interval. Drawing of T/H diagram for each process in one diagram gives the T/H diagram for whole stream.

__FWHEX(1)Processes__

Hot Stream - Process(1-2)							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
0.063	163.8	99.6	2803.625	2674.95	2.0043	0.12627	-8.107

Hot Stream - Process(2-3) (latent)			
$\dot{m}\ (Kg/s)$	T (°C)	Δh (Kj/Kg)	$\Delta\dot{H}$ (KW)
0.063	99.6	2257.514	-142.223

Hot Stream - Process(3-4)							
$\dot{m}\ (Kg/s)$	T_3 (°C)	T_4 °C)	h_3 (Kj/Kg)	h_4 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
0.063	99.6	43	417.436	180.159	4.1922	0.26411	-14.948

Total load of heat stream is -165.278 Kw

Cold stream							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
53.15	40.1	41.1	170	173	3.0000	159.43500	159.435

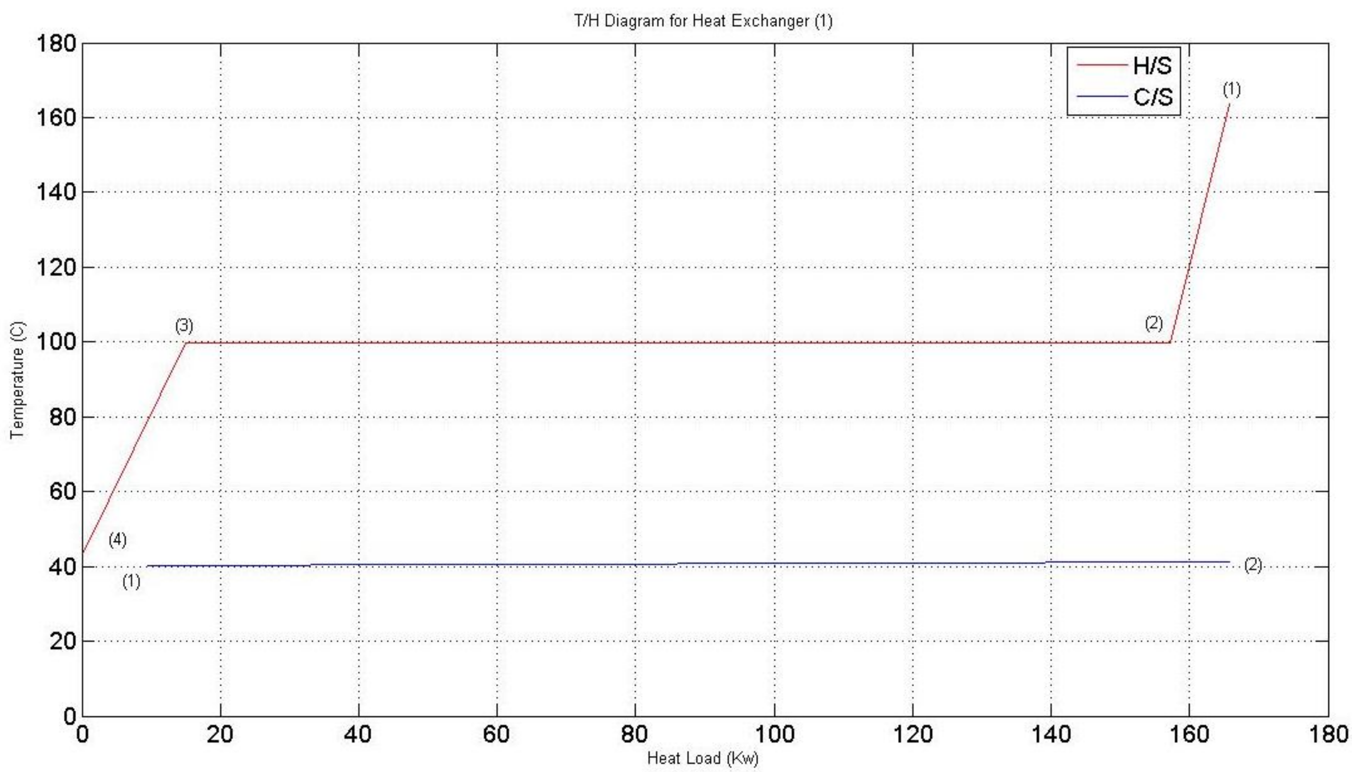


Figure 4.11 Temperature –Heat load diagram for FWHEX(1)

FWHEX(2)Processes

Hot Stream - Process(1-2)							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
0.035	336.1	99.6	3147.4	2674.95	1.9977	0.06992	-16.536

Hot Stream - Process(2-3)			
$\dot{m}\ (Kg/s)$	T (°C)	Δh (Kj/Kg)	$\Delta\dot{H}$ (KW)
0.035	99.6	2254.95	-78.923

Total load of heat stream is -95.459 Kw

Cold stream							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
53.145	41.1	41.5	173	174.7	4.2500	225.86625	90.346

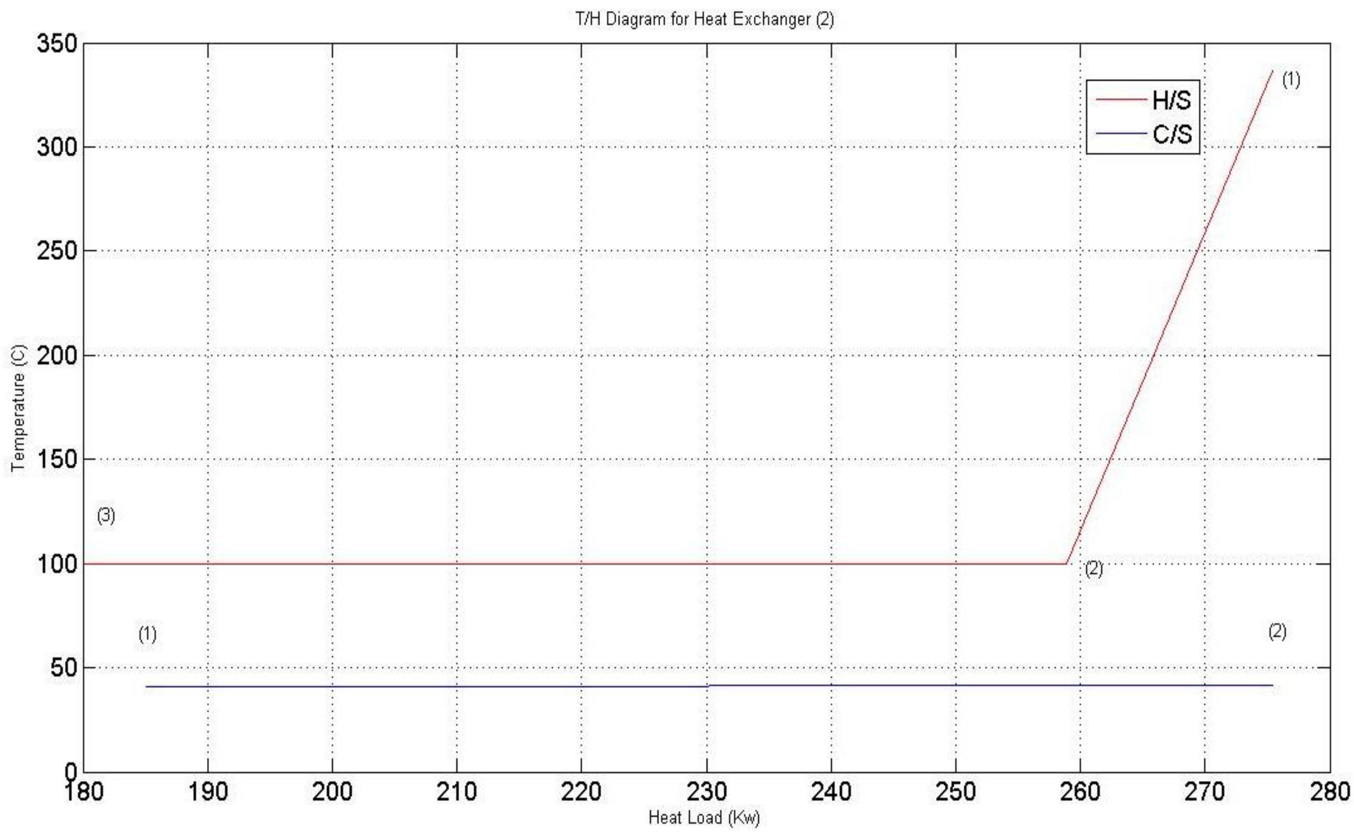


Figure 4.12 Temperature-Heat load diagram for FWHEX(2)

FWHEX(3)Processes

Hot Stream - Process(1-2)			
$\dot{m}\ (Kg/s)$	T (°C)	Δh (Kj/Kg)	$\Delta\dot{H}$ (KW)
6.737	93.3	1411.306	-9507.969

Hot Stream - Process(2-3)							
$\dot{m}\ (Kg/s)$	T_2 (°C)	T_3 °C)	h_2 (Kj/Kg)	h_3 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
6.737	93.3	47.6	390.86	199.3	4.1917	28.23938	-1290.540

Total load of hot stream is -10798.508 Kw

Cold stream							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
53.145	41.5	87.2	174.7	366.1	4.188	222.58103	10171.953

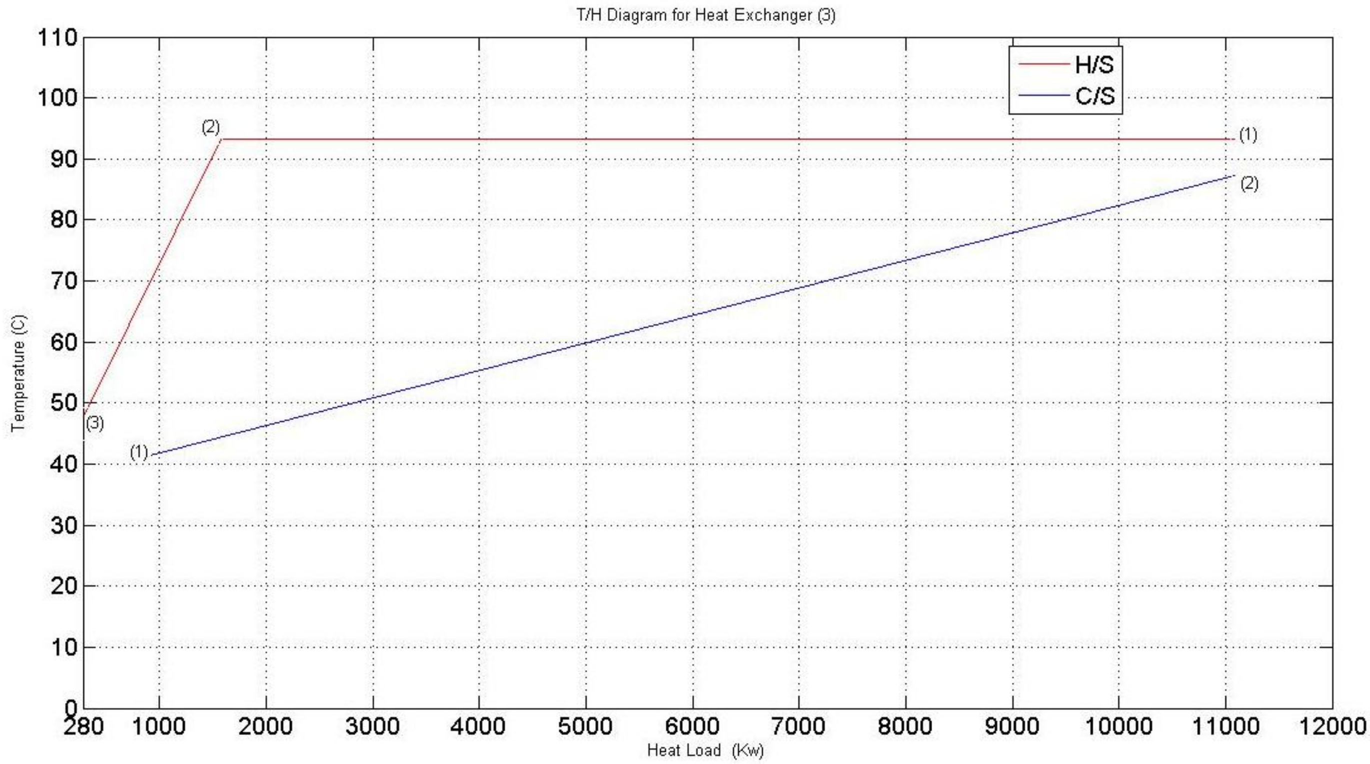


Figure 4.13 Temperature-Heat load diagram for FWHEX(3)

FWHEX(4)Processes

Hot Stream - Process(1-2)			
$\dot{m}\ (Kg/s)$	T (°C)	Δh (Kj/Kg)	$\Delta\dot{H}$ (KW)
2.554	119.2	2137.9	-5460.197

Hot Stream - Process(2-3)							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
2.554	119.2	92.1	500.6	386	4.2288	10.80031	-292.688

The total load of heat stream is -5752.885 Kw

Cold stream							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
53.145	87.2	113	366.1	474.2	4.2062	223.53986	5744.975

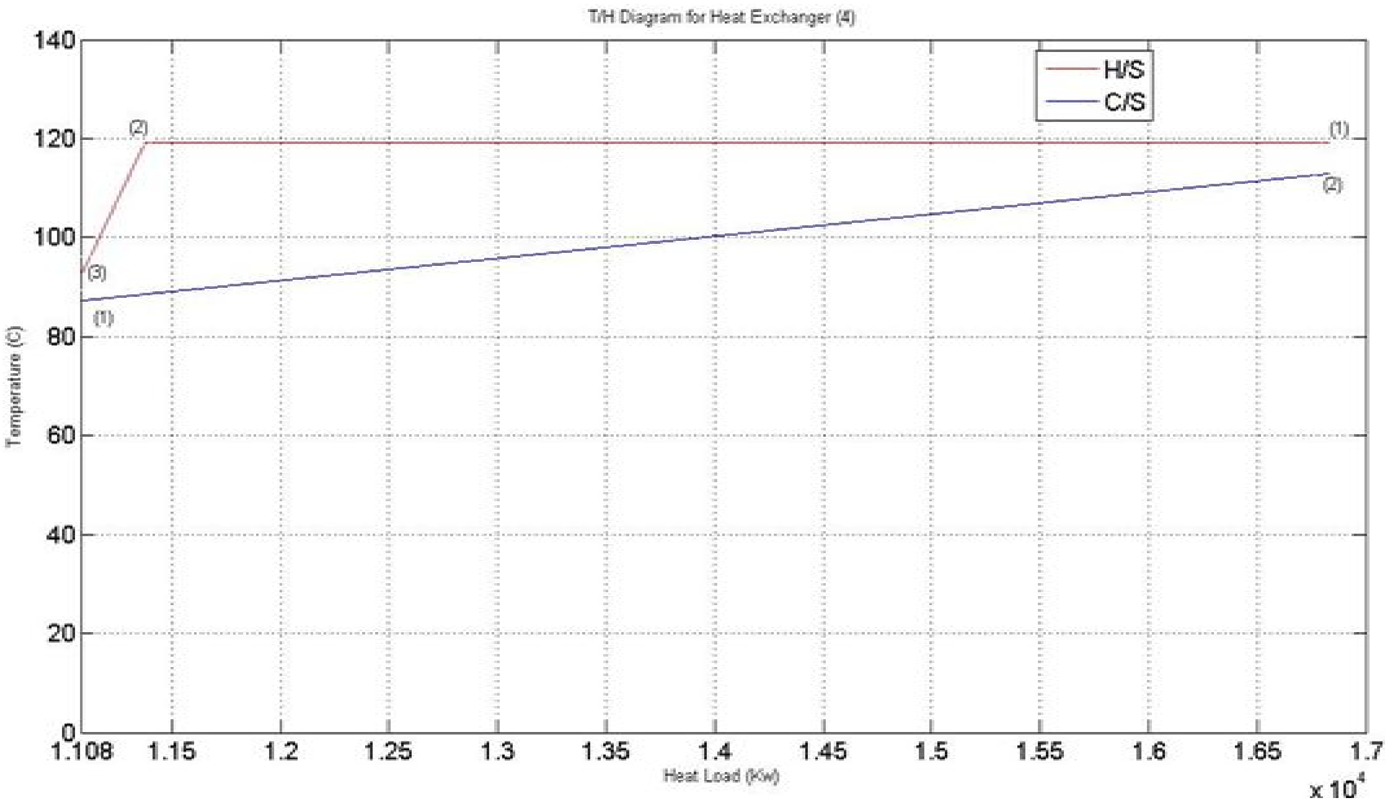


Figure 4.14 Temperature – Heat load diagram for FWHEX(4)

Composite Curve

Composite curve construction needs to identify all common interval temperatures in stream whether hot or cold and each interval start with temperature supply (T_s) and end with temperature target (T_T). Next step is calculation of heat capacity ($\sum \dot{m}\, C_p$) of each interval by sum of heat capacity for each stream in specific interval alone. Then heat load is calculated according to equation (3.1)

$$\dot{Q} = \int_{T_s}^{T_T} CP dT = CP(T_T - T_s) = \Delta\dot{H}$$

$$Where : \; CP = \sum \dot{m}\, C_p$$

The above explanation of sensible heat processes, but in case of latent heat processes

$$\Delta\dot{H} = \dot{m}h_{fg} = \dot{m}(h_g - h_f)$$

$Where:$

h_{fg} $is\ transformation\ enthalpy$

h_g $is\ enthalpy\ at\ gaseous\ phase$

h_f $is\ enthalpy\ at\ lequefied\ phase$

According to four first group heat exchangers T/H diagrams, it is possible to obtain *Hot and Cold Stream Composite Curve Parameters*as follows:

Table 4.23 First Group Hot Stream Composite Curve Parameters

	Hot Stream Composite Curve Parameters				
	T_s	T_T	$\sum \dot{m}\, C_p$	ΔH	Latent Heat
1	336.8	163.8	0.0699	-12.098	N/A
2	163.8	119.2	0.1962	-8.750	N/A
3	119.2	119.2	None	-5460.197	A
4	119.2	99.6	10.9972	-215.545	N/A
5	99.6	99.6	None	-221.146	A
6	99.6	93.3	11.0651	-69.710	N/A
7	93.3	93.3	None	-9507.969	A
8	93.3	92.1	39.3061	-47.167	N/A
9	92.1	47.6	28.5051	-1268.477	N/A
10	47.6	43	0.2641	-1.215	N/A

Total load of composite hot stream is -16812.274 Kw

Table 4.24 First Group Cold Stream Composite Curve Parameters

	Cold Stream Composite Curve Parameters			
	T_s	T_T	$\sum \dot{m}\, C_p$	ΔH
1	40.1	41.1	159.4350	159.435
2	41.1	41.5	225.8662	90.346
3	41.5	87.2	222.5810	10171.953
4	87.2	112.9	223.5399	5744.975

Total load of composite cold stream is 16166.709 Kw

Then according to *Hot and Cold Stream Composite Curve Parameters tables* above, it can be drawn Composite curve of first group Heat exchangers simply.

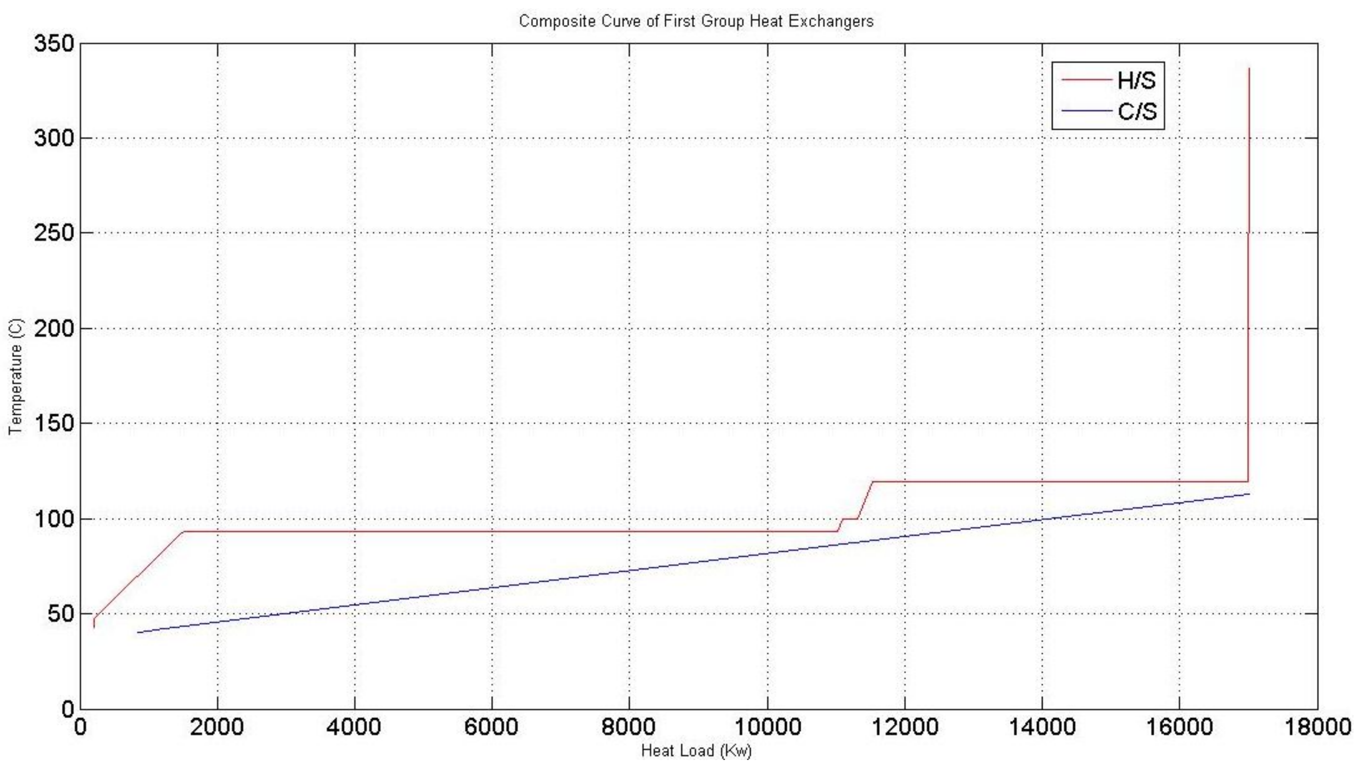


Figure 4.15 Temperature-Heat load Composite Curve of First Group Heat Exchangers

T/H composite curve diagram for first group heat exchanger above illustrates loss of biggest temperature interval in composite hot stream due to extremely low heat capacity corresponding to this temperature interval.

Thus, there is lowest heat load exchanging though extremely big temperature interval. This matter considered as inefficient use of heat exchanging process.

Now, it is important to identify minimum temperature deference ΔT_{min} between Hot and Cold composite streams in T/H composite curve diagram. Visually, there are two points may have ΔT_{min}, first point at right hand side of diagram and another point at the middle of diagram. Then after zoom-in those two points may it is possible to identify where ΔT_{min}takes place.

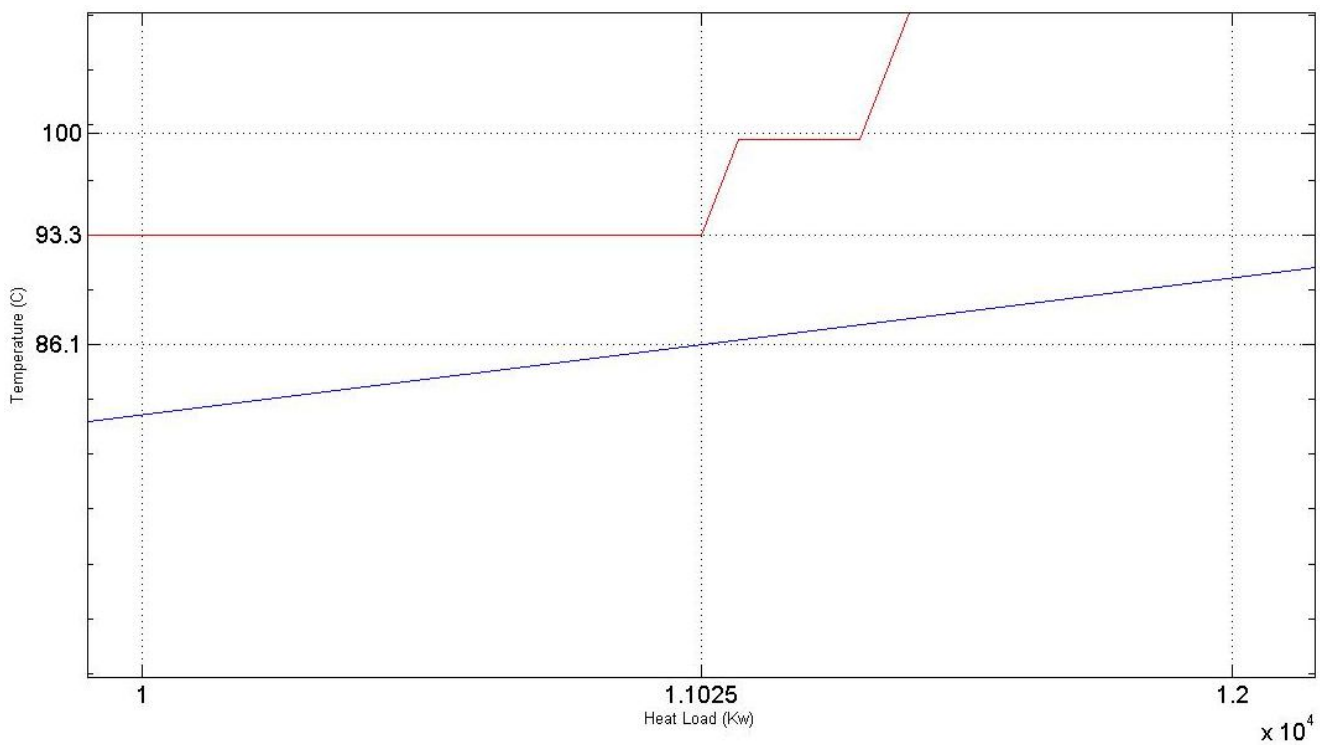


Figure 4.15a Zoomed-in Composite Curve of First Group at middle zone

Zoomed-in T/H composite curve diagram at middle zone (Figure 4.15a) shows that

$$\Delta \mathbf{T_{min}} = \mathbf{93.3} - \mathbf{86.1} = \mathbf{7.2}℃$$

Zoomed-in T/H composite curve diagram at right hand side zone (Figure 4.15b) shows that

$$\Delta T_{min} = 119.2 - 112.8 = 6.4 ℃$$

Then selected ΔT_{min} is 6.4℃

As discussed at chapter (3) hot and cold shifted temperature ($T_{H(shifted)}$ and $T_{C(shifted)}$ *respctively*) *are calculated by using below formula*

$$T_{H(shifted)} = T_H - \frac{\Delta T_{min}}{2}$$

$$T_{C(shifted)} = T_C + \frac{\Delta T_{min}}{2}$$

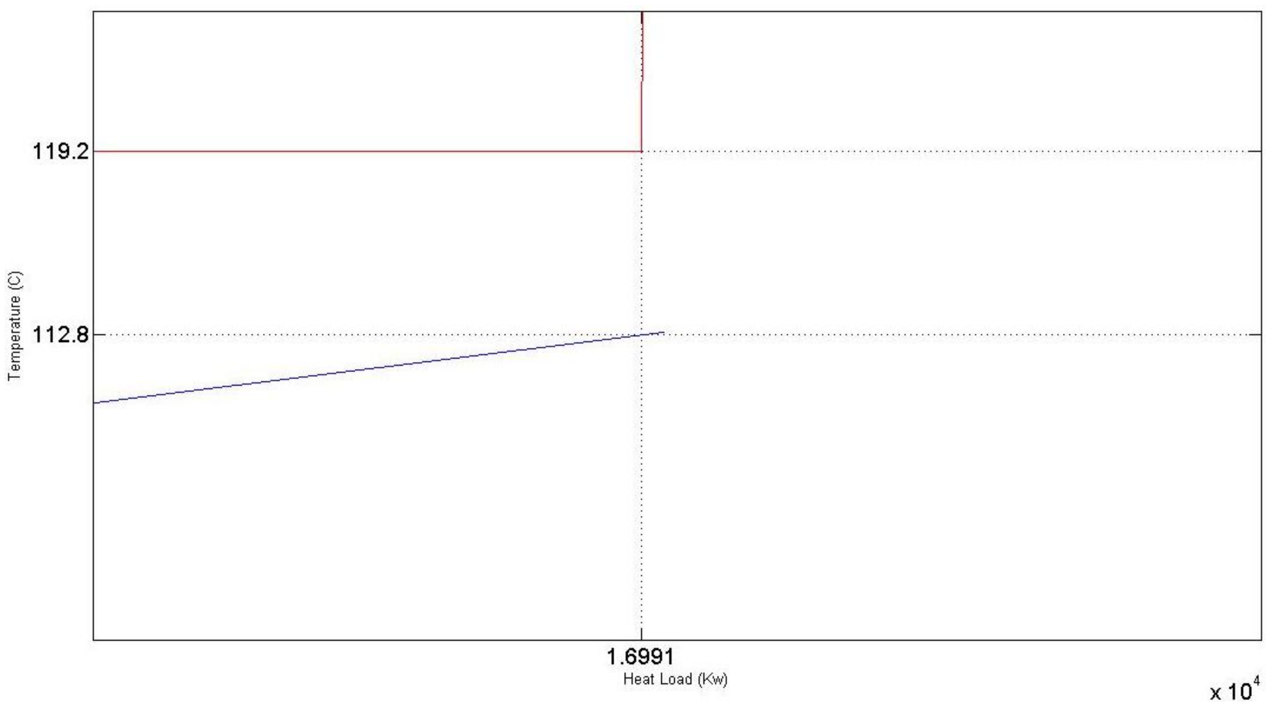


Figure 4.15b Zoomed-in Composite Curve diagram of First Group right hand side zone

Then shifted composite curve diagram are constructed by applying hot and cold shifted temperatures as follows:

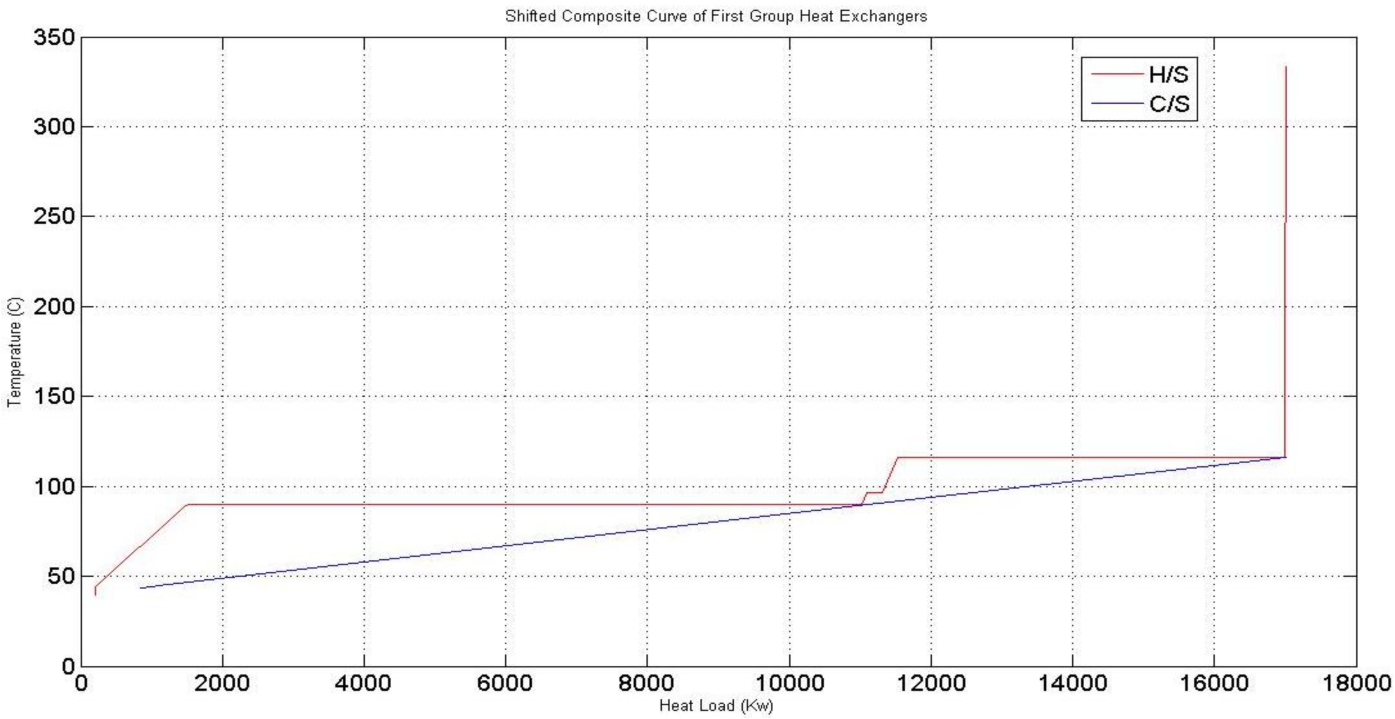


Figure 4.16 Shifted Temperature-Heat load Composite Curve of First Group

In Figure 4.16, it seems composite hot stream touches composite cold stream at two points but by zooming-in at middle point suspected to be tangential point (Pinch Point), in the figure below, it is clear that there is no tangency in this point.

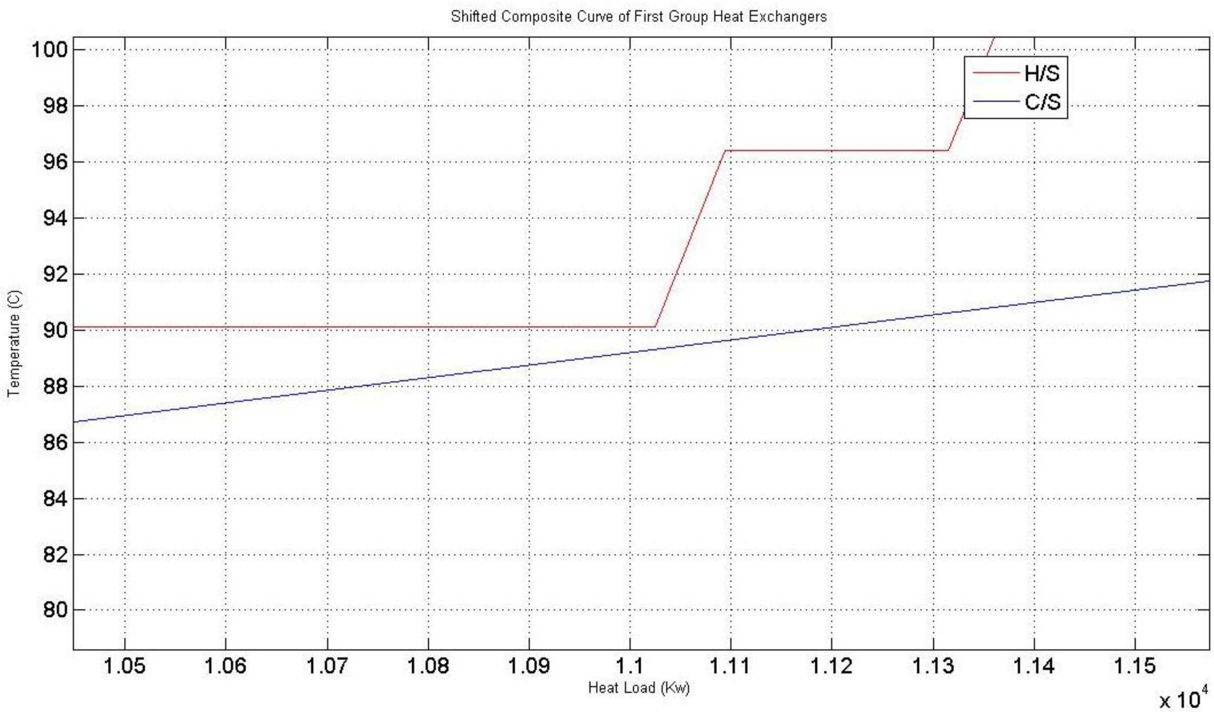


Figure 4.16a Zoomed-in Shifted Composite Curve of First Group at middle zone

According to what already discussed there is one pinch point in this case and it is located at the right hand side of shifted composite curve diagram.

Now, it is desired to know at what shifted temperature this pinch point is. Thus, finally zooming-in at pinch point to simplify identifying of sifted pinch temperature is needed (Figure 4.16b).

Then after zooming-in

$$\boldsymbol{T_{pinch(Shifted)}=116}℃$$

The pinch temperature at hot and cold streams are calculated as follows

$$\boldsymbol{T_{H,pinch}} = \boldsymbol{T_{pinch(Shifted)}} + \frac{\boldsymbol{\Delta T_{min}}}{\boldsymbol{2}} = \boldsymbol{116+3.2=119.2}℃$$

$$\boldsymbol{T_{C,pinch}} = \boldsymbol{T_{pinch(Shifted)}} - \frac{\boldsymbol{\Delta T_{min}}}{\boldsymbol{2}} = \boldsymbol{116-3.2=112.8}℃$$

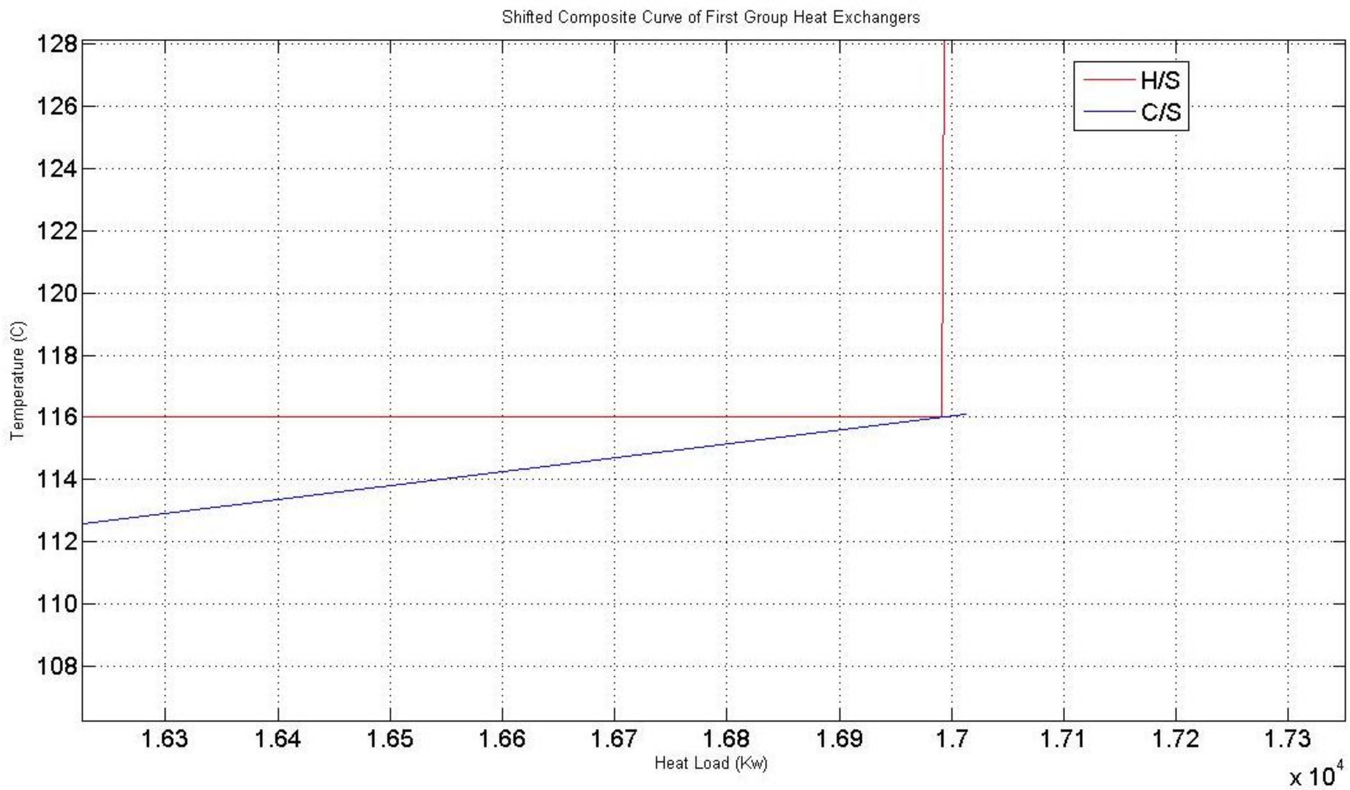


Figure 4.16b Zoomed-in Shifted Composite Curve of First Group at middle zone

Eventually, the composite curve method identifies hot stream pinch temperature is $119.2°\text{C}$ and cold stream pinch temperature is $112.8°\text{C}$ and the minimum defference between Hot and cold stream is$6.4°\text{C}$.

4.5.2.2 Mathematical Technique Method

In this research, applying of this method to check to what extend the composite curves method results are accurate and consistency of two methods outcomes.

First step in this method is identifying shifted temperature intervals after selection of suitable minimum deference temperature ΔT_{min} . Then calculate the heat load by abstract cold stream enthalpy from hot stream enthalpy if sensible heat and specify latent heat if there is, then sum both at lower temperature in each shifted temperature intervals. Now, there is known deference of enthalpies at each lower shifted temperatures. For utilization purposes suppose Hot utility load is zero Kw and calculate the cumulative summation at each lower shifted temperatures. The result of the last step has to possibility whether there is zero value of cumulative summation at one of shifted temperature then this shifted temperature is shifted pinch temperature or there is less zero value for cumulative summation at one of shifted temperature then the calculation will be repeated by positive Hot utility load equal to less zero value. Then where cumulative summation equal zero, the shifted pinch temperature is.

Then from Table 4.26

$\boldsymbol{T_{pinch(Shifted)}}$=**116.00335**℃ ≈**116**℃

$$\boldsymbol{T_{H,pinch} = T_{pinch(Shifted)} + \frac{\Delta T_{min}}{2} = 116.00335 + 3.19665}$$
$$\boldsymbol{= 119.2}℃$$

$$\boldsymbol{T_{C,pinch} = T_{pinch(Shifted)} - \frac{\Delta T_{min}}{2} = 116.00335 - 3.19665}$$
$$\boldsymbol{= 112.8}℃$$

In this case $\Delta \boldsymbol{T_{min} = 6.3933}$℃ ≈ $\boldsymbol{6.4}$℃

Table 4.25 First Group Shifted Temperature intervals

				$\Delta \boldsymbol{T}_{\boldsymbol{min}} = \mathbf{6.3933}$℃	
		Actual		Shifted	
stream	CP	T_s	T_t	T_{ss}	T_{st}
H/S	0.0699	336.8	163.8	333.60335	160.60335
H/S	0.1962	163.8	119.2	160.60335	116.00335
H/S	None	119.2	119.2	116.00335	116.00335
H/S	10.9972	119.2	99.6	116.00335	96.40335
H/S	None	99.6	99.6	96.40335	96.40335
H/S	11.0651	99.6	93.3	96.40335	90.10335
H/S	None	93.3	93.3	90.10335	90.10335
H/S	39.3061	93.3	92.1	90.10335	88.90335
H/S	28.5051	92.1	47.6	88.90335	44.40335
H/S	0.2641	47.6	43	44.40335	39.80335
C/S	159.4350	40.1	41.1	43.29665	44.29665
C/S	225.8662	41.1	41.5	44.29665	44.69665
C/S	222.5810	41.5	87.2	44.69665	90.39665
C/S	223.5399	87.2	112.9	90.39665	116.09665

$Where:$

$$CP = \sum \dot{m}\, C_p$$

$T_s = Actual\ temperature\ supply$

$T_t = Actual\ temperature\ target$

$T_{ss} = Shifted\ temperature\ supply$

$T_{st} = Shifted\ temperature\ target$

Table 4.26 Firs Group Cumulative Summation of Heat loads

$\Delta T_{min} = 6.3933°C$							
Shifted Temp.	**Interval number (i)**	$Ts_i - Ts_{i+1}$ (K)	$CP_h - CP_c$ $(\frac{Kw}{K})$	$\Delta H_{sensible}$ (Kw)	ΔH_{latent} (Kw)	$\Delta H_{total} = \Delta H_{sensible} + \Delta H_{latent}$ (Kw)	***Heat Flow*** (Kw)
Ts1=333.60335							0.0
Ts2=160.60335	1	173.00000	0.06993	12.09789		12.09789	12.1
Ts3=116.09665	2	44.50670	0.19618	8.73141		8.731413	20.8
Ts4=116.00335	3	0.09330	-223.34372	-20.83797		-20.838	0.0
Ts5=96.40335	4	19.60000	-212.54270	-4165.83692	5460.197	1294.36	1294.4
Ts6=90.39665	5	6.00670	-212.47480	-1276.27238		-1276.27	18.1
Ts7=90.10335	6	0.29330	-211.51590	-62.03761	221.146	159.1084	177.2
Ts8=88.90335	7	1.20000	-183.27490	-219.92988	9507.969	9288.039	9465.2
Ts9=44.69665	8	44.20670	-194.07590	-8579.45509		-8579.46	885.8
Ts10=44.40335	9	0.29330	-197.36090	-57.88595		-57.886	827.9
Ts11=44.29665	10	0.10670	-225.60210	-24.07174		-24.0717	803.8
Ts12=43.29665	11	1.00000	-159.17090	-159.17090		-159.171	644.6
Ts13=39.80335	12	4.49330	0.26410	1.18666		1.186663	645.8

Now, by drawing heat flow against shifted temperature, Grand Composite curve can be constructed as in table 4.26.

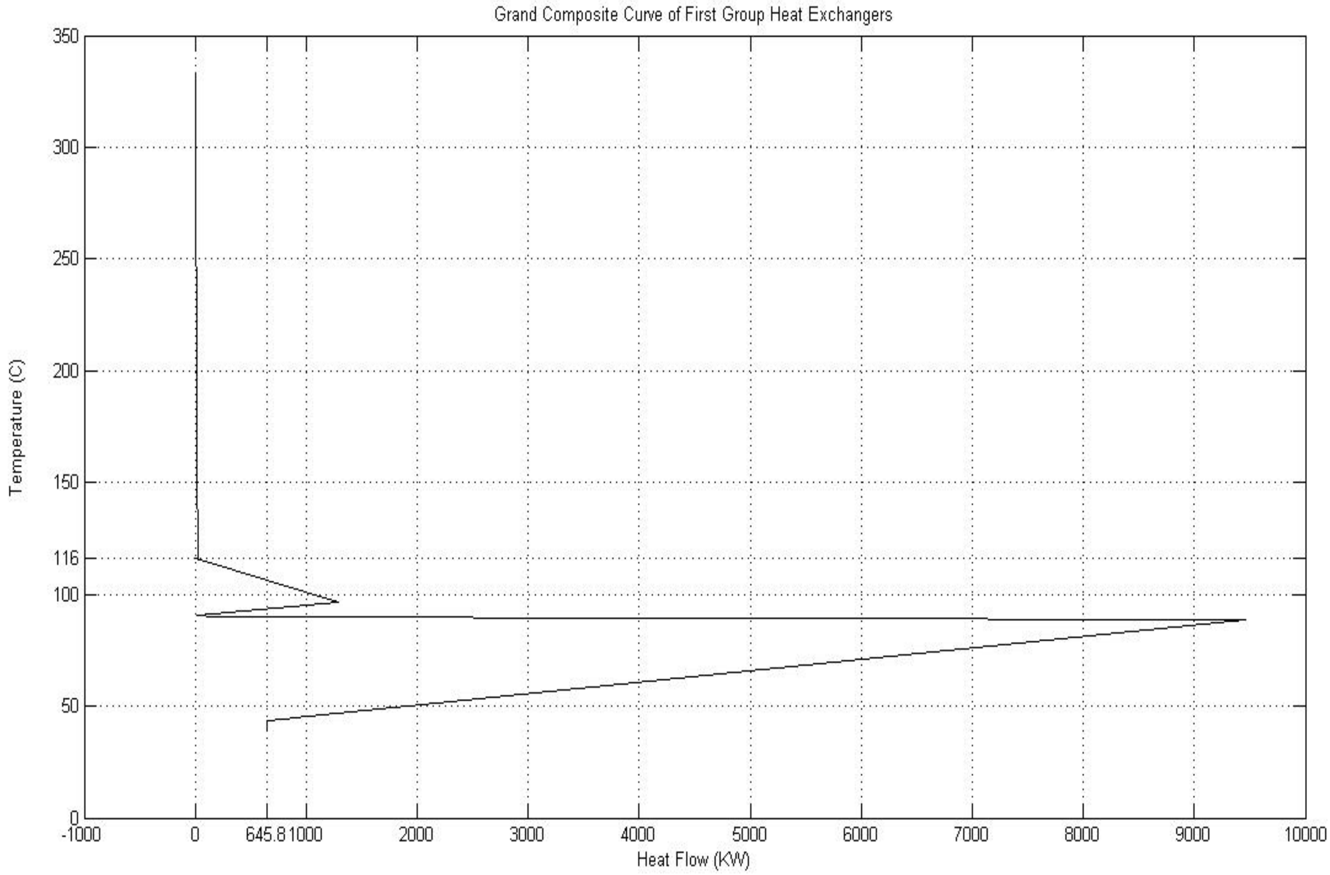


Figure 4.17 Grand Composite Curve of First Group Heat Exchangers

By zooming-in at shifted pinch temperature $116°C$, it is found that heat flow equal to zero exactly what approve that $116°C$ is exact shifted temperature(Figure 4.17a).

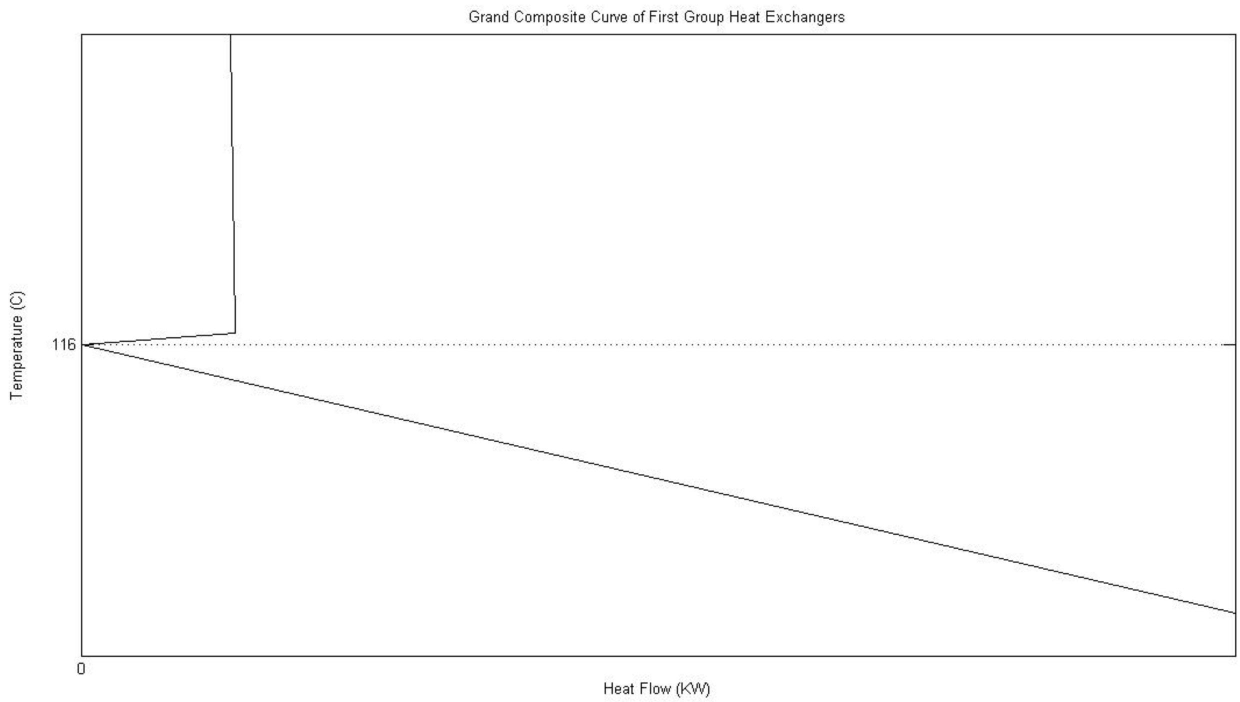


Figure 4.17a Zoomed-in Grand Composite Curve of First Group at 116℃

Then hot stream pinch temperature is 119.2℃ and cold stream pinch temperature is 112.8℃ is final sure result for first group heat exchanger.

4.5.2.3 First Group Analysis Results Discussion

As mentioned in chapter (3), efficient utilization of heat exchanger net works depend on the three golden rules as follows:

- ✓ Don't transfer heat across the pinch.
- ✓ Don't use cold utilities above the pinch.
- ✓ Don't use hot utilities below the pinch.

In commencement, the two last rules is already applied where there is no cold utilities above the pinch and there is no hot utilities below the pinch. But what about the first rule, the four heat exchanger T/H diagram shown before should be revised to approve this rule.

After revision of four diagrams, it is found that FWHEX(1) breaks this rule where heat exchanging between temperature interval at hot stream above 119.2℃with temperature interval at cold stream lower 112.8℃ which means that there is heat exchanging between above pinch part and below pinch part which can be concluded as there is heat transferred across the pinch.

For the same reasons shown above FWHEX(2) and FWHEX(4) break the first golden rule.

Eventually, according to pinch technology analysis the first group heat exchanger network needs redesign to achieve highest heat exchanging utilization and the weak points in this heat exchanger network are FWHEX(1,2&4).

4.5.3 Second Group Pinch analysis

In this section, second group pinch analysis, it will be avoided to repeat same explanations discussed in the first Group Pinch analysis (prior section) unless there is necessity for that.

Table 4.27 Second group data collection

Streams	$\dot{m}\ (\frac{Kg}{s})$	T_s (°C)	$h_s\ (Kj/Kg)$	T_t (°C)	$h_t\ (Kj/Kg)$
Cold stream	64.538	152.7	650.3	219.4	943.6
Hot stream (5)	3.562	261.7	2959.7	157.7	665.6
Hot stream (6,trap)	4.421	189.1	804.8	157.7	665.6
Hot stream (6)	4.421	338.5	3101.5	189.1	804.8

Mixed hot stream property which is mixing of Hot stream (3) and Hot stream (4,trap) is being calculated by using mass and heat balance.

Mass balance

$$\dot{m}_{H,5} + \dot{m}_{H,6,\mathrm{trap}} = \dot{m}_{H,5,mix}$$

$$3.562 + 4.421 = \dot{m}_{H,5,mix}$$

$$\dot{m}_{H,3,mix} = 7.983\ kg/s$$

Energy balance

$$\dot{m}_{H,5}h_{H,5} + \dot{m}_{H,6,\mathrm{trap}}h_{H,6,\mathrm{trap}} = \dot{m}_{H,5,mix}h_{H,5,mix}$$

$$h_{H,5,mix} = \frac{\dot{m}_{H,5}h_{H,5} + \dot{m}_{H,6,\mathrm{trap}}h_{H,6,\mathrm{trap}}}{\dot{m}_{H,5,mix}}$$

$h_{H,5}$ *and* $h_{H,6,\mathrm{trap}}$ *values should be substituted from Table* 4.27

$$h_{H,5,mix} = \frac{3.562 * 2959.7 + 4.421 * 804.8}{7.983}$$

$$h_{H,5,mix} = 1766.312\ kj/kg$$

$$T_{H,5,mix} = \frac{\mathrm{T_{H,5}} + \mathrm{T_{H,6,Trap}}}{2} = \frac{261.7 + 189.1}{2} = 225.4\ ℃$$

From Steam/Water properties tables

$$P_{H,5,mix} = 25.69\ \text{bar}$$

Dryness factor

$$x_{H,5,mix} = \frac{h_{H,5,mix} - h_f}{h_{fg}} = \frac{1766.312 - 968.701}{2802.335 - 968.701} = 0.43$$

Then mixture is in wet zone

Table 4.28 Second group data collection after mixing calculation

$Streams$	$\dot{m}\ (\frac{Kg}{s})$	T_s (°C)	h_s (Kj/Kg)	T_t (°C)	h_t (Kj/Kg)
Cold stream	64.538	152.7	650.3	219.4	943.6
Hot stream (5,mix)	7.983	225.4	1766.312	157.7	665.6
Hot stream (6)	4.421	338.5	3101.5	189.1	804.8

Table 4.29 Second Group Detailed Data

			In					Out					
		$\dot{m}$	P (bar)	T_{sat} (°C)	T (°C)	$h\ (\frac{Kj}{Kg})$	state	P (bar)	T_{sat} (°C)	T (°C)	$h\ (\frac{Kj}{Kg})$	state	Δh_{latent}
HXG(5)	H/S	7.983	25.650	225.4	225.4	1766.312	mixture	12.110	188.38	157.7	565.6	water	A
	C/S	64.538	113	320.11	152.7	650.3	water	112.5	319.77	184.1	786.3	water	N/A
HXG(6)	H/S	4.42	25.050	224.1	338.5	3101.5	steam	24.140	222.1	189.1	804.8	water	A
	C/S	64.538	112.5	319.77	184.1	786.3	water	112	319.44	219.4	943.6	water	N/A

As outcome of Table 4.29 aboveall hot streams in second group has latent heat same as first group.

4.5.3.1 Composite Curve Method

FWHEX(5) Processes

Hot Stream - Process(1-2)			
$\dot{m}\,(Kg/s)$	T (°C)	Δh (Kj/Kg)	$\Delta\dot{H}$ (KW)
7.983	225.4	797.611	-6367.329

Hot Stream - Process(2-3)							
$\dot{m}\,(Kg/s)$	T_2 (°C)	T_3 °C)	h_2 (Kj/Kg)	h_3 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
7.983	225.4	158	968.701	665.6	4.4771	35.74085	-2419.655

The total load of heat stream is -8786.984 Kw

Cold stream							
$\dot{m}\,(Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
64.538	152.7	184	650.3	786.3	4.3312	279.52764	8777.168

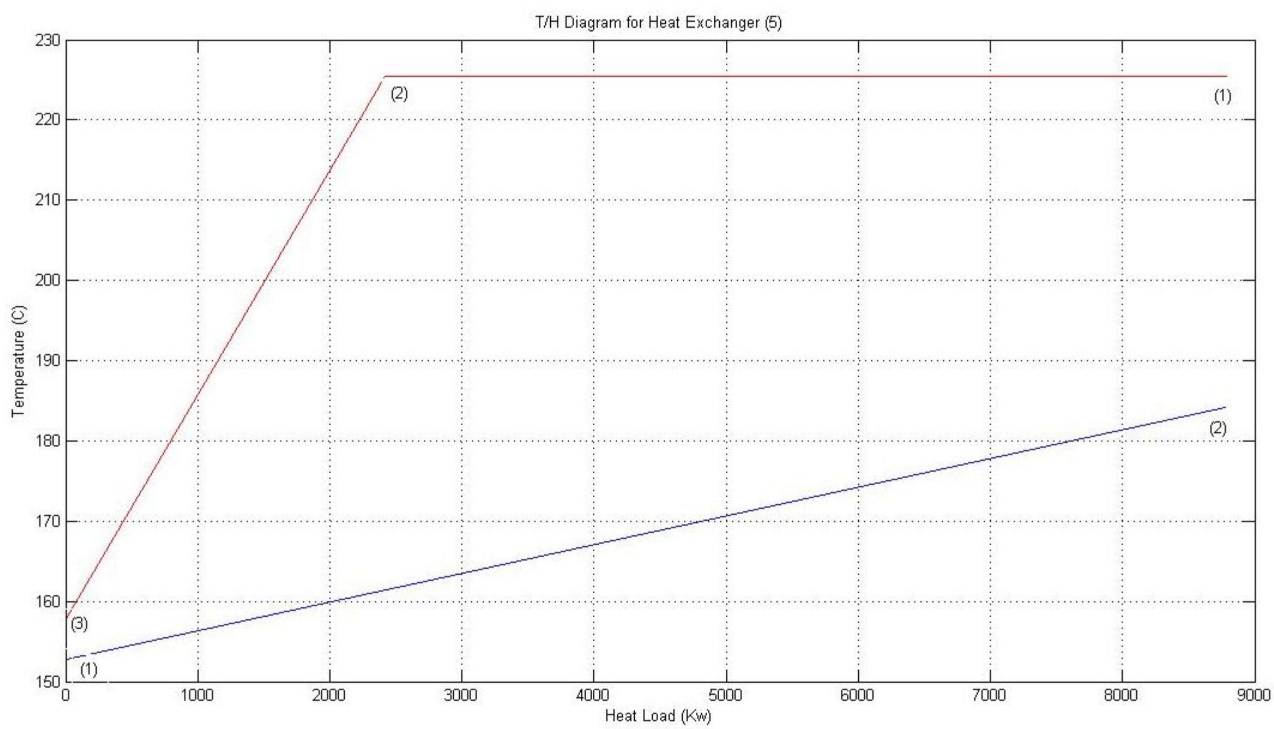


Figure 4.18 Temperature- Heat Load Diagram of FWHEX(5)

FWHEX(6) Processes

Hot Stream - Process(1-2)							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
4.421	338.5	224	3101.5	2802.08	2.6173	11.57096	-1323.718

Hot Stream - Process(2-3)			
m	T	dh	DH
4.421	224.1	1839.212	-8131.156

Hot Stream - Process(3-4)							
$\dot{m}\ (Kg/s)$	T_3 (°C)	T_4 °C)	h_3 (Kj/Kg)	h_4 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
4.421	224.1	189	962.872	804.8	4.5163	19.96675	-698.836

The total load of heat stream is -10153.711 KW

Cold stream							
$\dot{m}\ (Kg/s)$	T_1 (°C)	T_2 °C)	h_1 (Kj/Kg)	h_2 (Kj/Kg)	Cp $(Kj/Kg.K)$	$\dot{m}Cp$ (KW/K)	$\Delta\dot{H}$ (KW)
64.538	184.1	219	786.3	943.6	4.4561	287.58718	10151.827

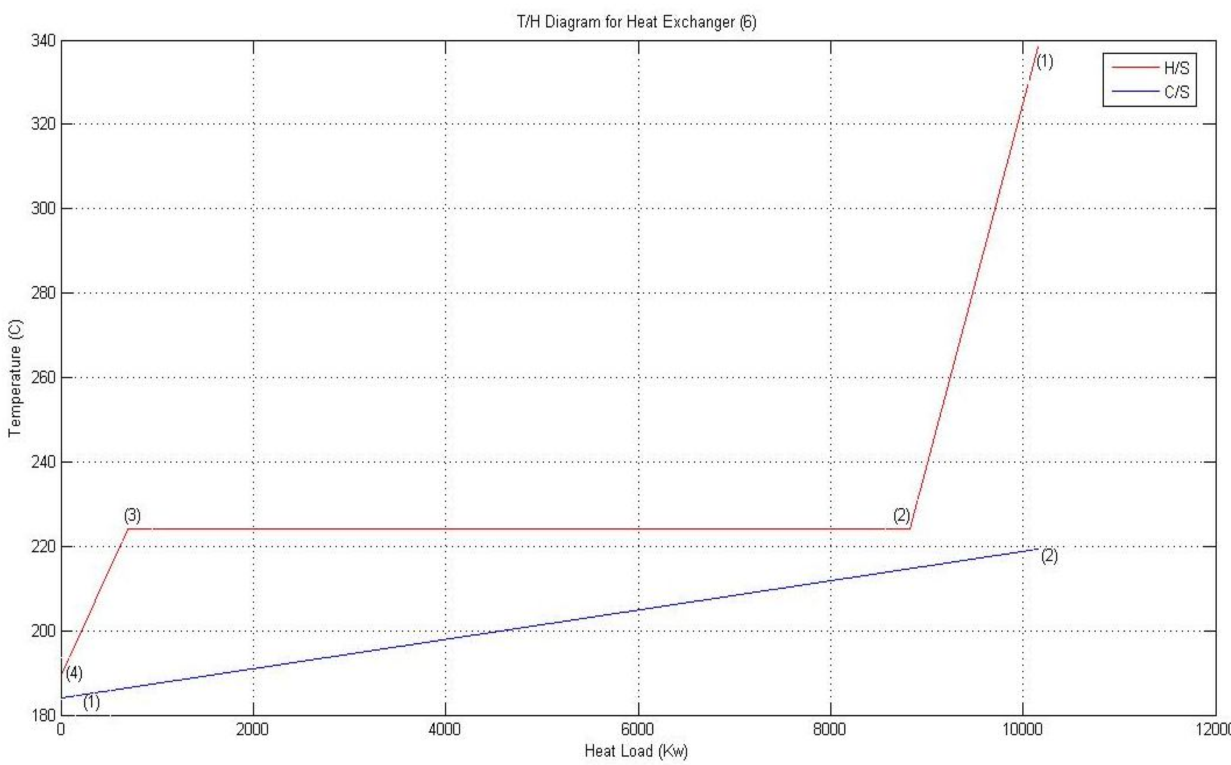


Figure 4.19 Temperature-Heat Load Diagram of FWHEX(6)

Composite Curve

Table 4.30 Second Group Hot Stream Composite Curve Parameters

	Hot Stream Composite Curve Parameters				
	T_s	T_T	$\sum \dot{m}\, C_p$	ΔH	Latent Heat
1	338.5	225.4	11.57096	-1308.676	N/A
2	225.4	225.4	None	-6367.329	A
3	225.4	224.1	47.31181	-61.50535	N/A
4	224.1	224.1	None	-8131.156	A
5	224.1	189.1	55.7076	-1949.766	N/A
6	189.1	157.7	35.74085	-1122.263	N/A

The total load of composite hot stream is -18940.6946Kw

Table 4.31 Second Group Cold Stream Composite Curve Parameters

	Hot Stream Composite Curve Parameters			
	T_s	T_T	$\sum \dot{m}\, C_p$	ΔH
1	152.7	184.1	279.5276	8777.168
2	184.1	219.4	287.5872	10151.8274

The total load of composite cold stream is 18928.9954Kw

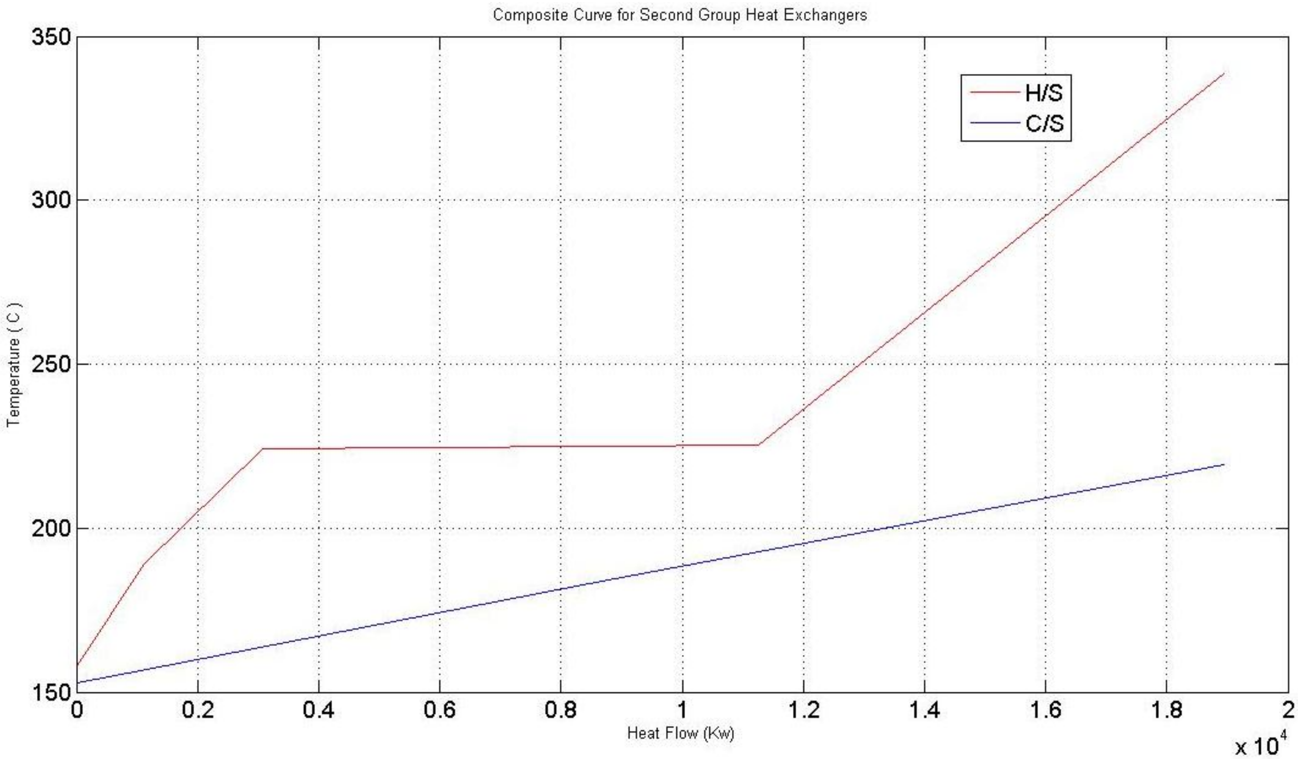


Figure 4.20 Temperature-Heat Load Composite Curve of Second Group Heat Exchangers

In Figure 4.20, only left hand side zone is expected to have pinch. By zooming-in at this side, it is possible to knowΔT_{min}.

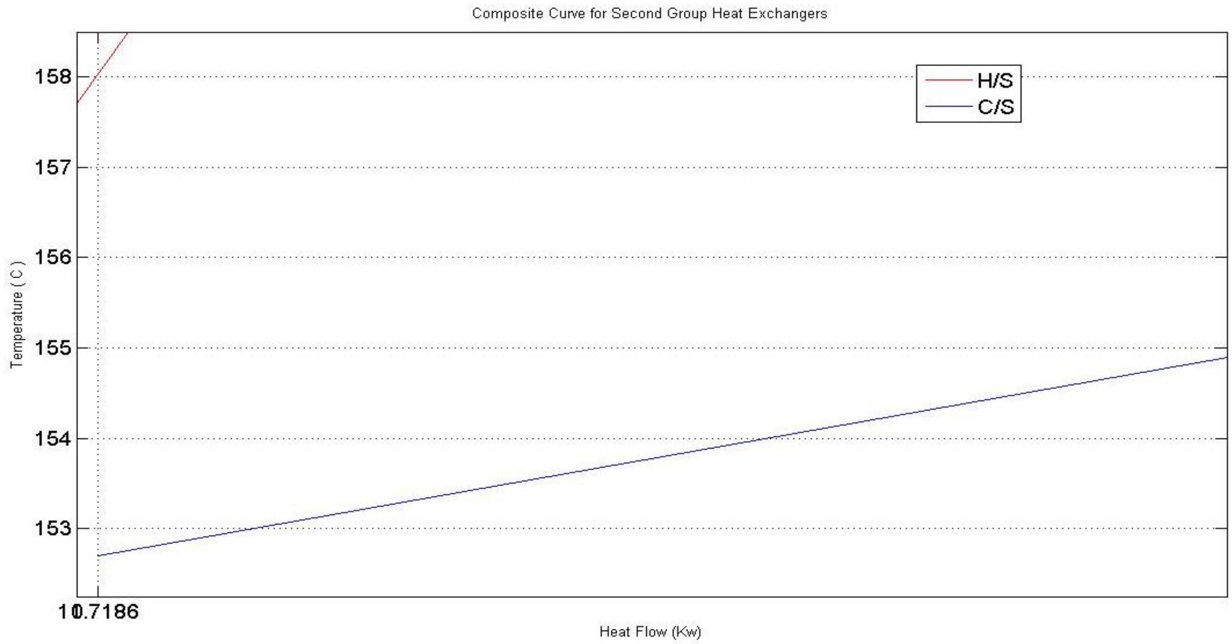


Figure 4.20a Zoomed-in Composite Curve of Second group at left hand side zone

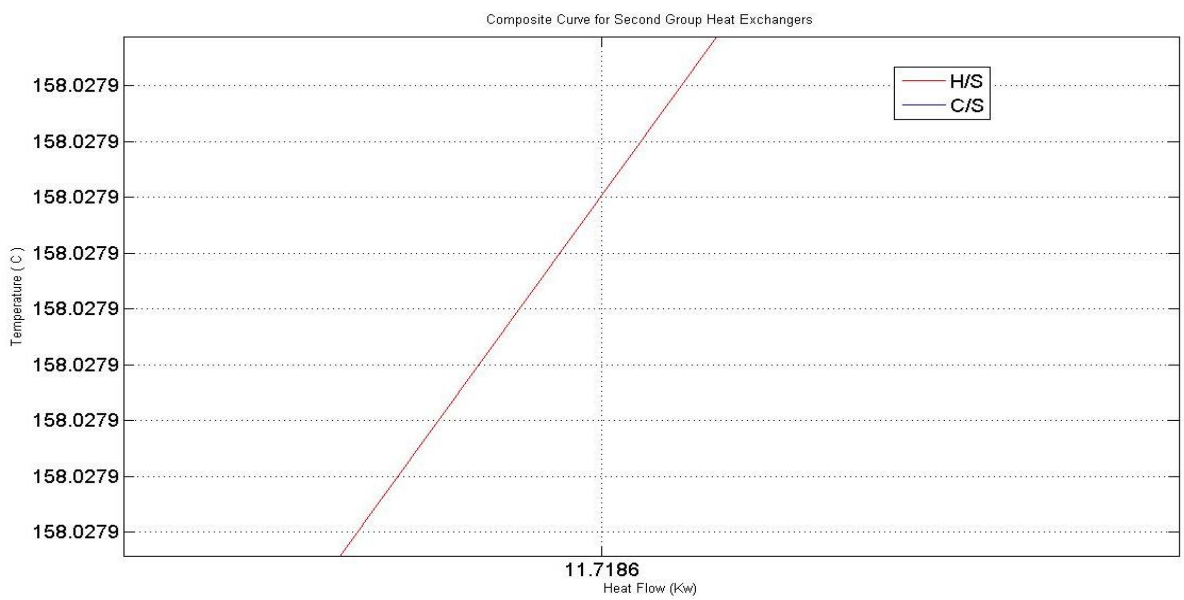


Figure 4.20b super zoomed-in Composite Curve of Second Group to specify

Accurate upper temperature limit of minimum temperature deference

$$\Delta T_{min} = 158.0279 - 152.7 = 5.3279℃$$

$$T_{H(shifted)} = T_H - \frac{\Delta T_{min}}{2}$$

$$T_{C(shifted)} = T_C + \frac{\Delta T_{min}}{2}$$

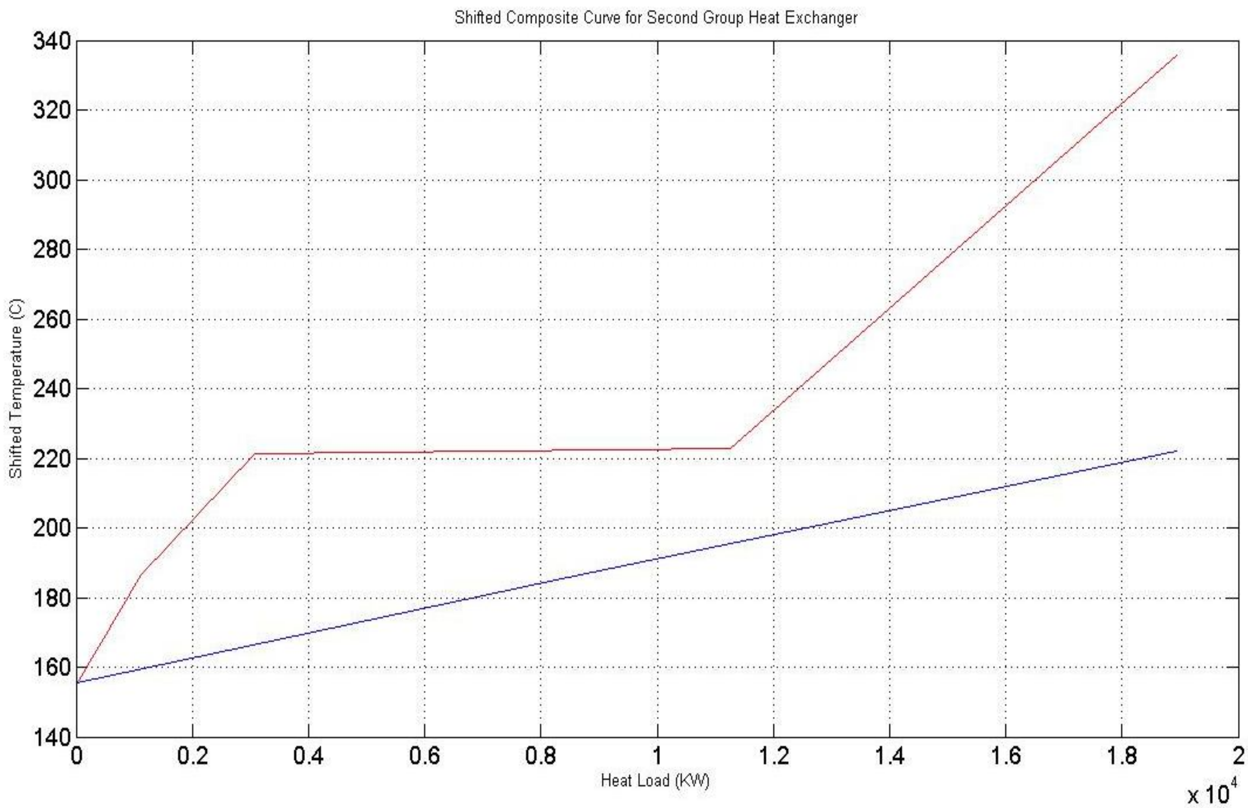


Figure 4.21 Sifted Temperature-Heat Load Composite Curve of Second Group

Now, it is needed to zoom-in at pinch to specify exact shifted pinch temperature.

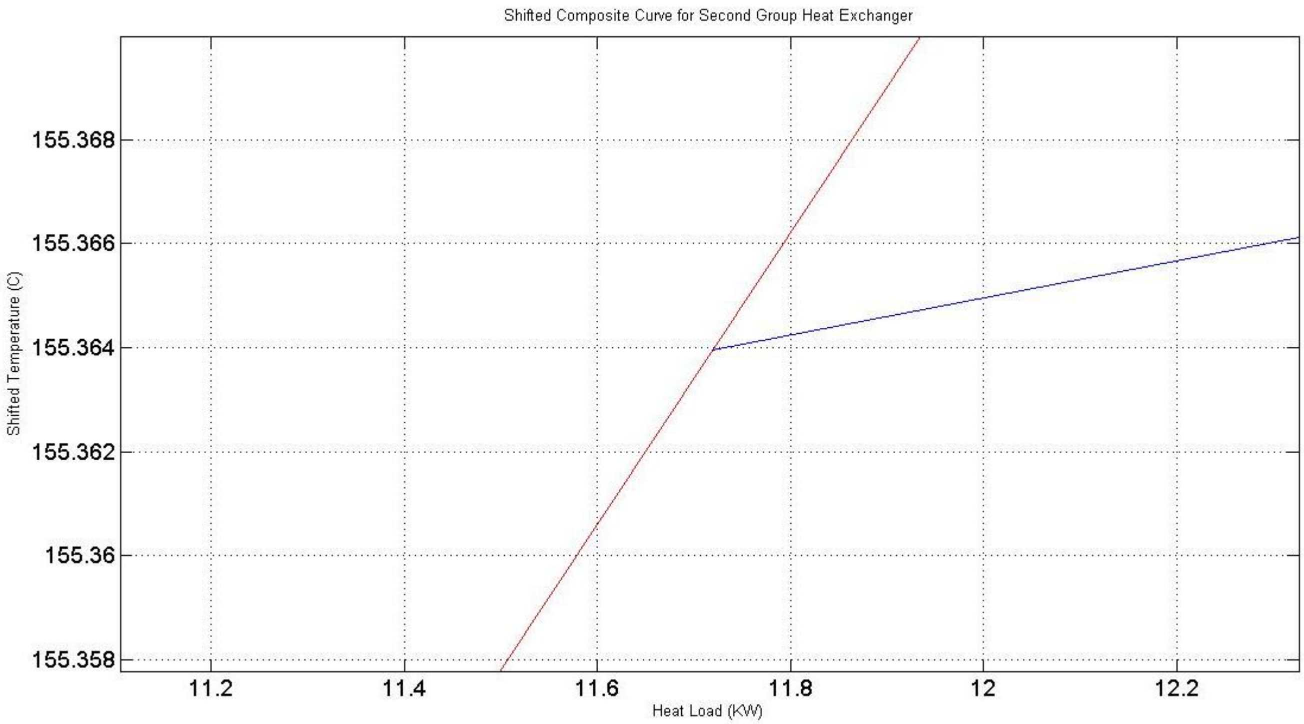


Figure 4.21a Zoomed-in Shifted Composite Curve of Second Group

Regarding to Figure 4.21a

$\boldsymbol{T_{pinch(Shifted)}}$=**155.364**℃

$$\boldsymbol{T_{H,pinch} = T_{pinch(Shifted)} + \frac{\Delta T_{min}}{2} = 155.364 + 2.66395}$$
$$\boldsymbol{= 158.02795}℃$$

$$\boldsymbol{T_{C,pinch} = T_{pinch(Shifted)} - \frac{\Delta T_{min}}{2} = 155.364 - 2.66395 = 152.7}℃$$

Then hot stream pinch temperature is almost159℃ and Cold stream pinch temperature is 152.7℃ and minimum deference temperature is5.3279℃.

4.5.3.2 Mathematical Technique Method

In second group heat exchangers selected $\Delta T_{min} = 5.3279℃$

Table 4.32 Second Group Shifted Temperature intervals

		Actual		$\Delta T_{min} = 5.3279℃$ Shifted	
stream	CP	T_s	T_t	T_{ss}	T_{st}
H/S	11.57	338.5	225.4	335.8361	222.7361
H/S	None	225.4	225.4	222.7361	222.7361
H/S	47.31	225.4	224.1	222.7361	221.4361
H/S	None	224.1	224.1	221.4361	221.4361
H/S	55.71	224.1	189.1	221.4361	186.4361
H/S	35.74	189.1	157.7	186.4361	155.0361
C/S	279.5	152.7	184.1	155.364	186.764
C/S	287.6	184.1	219.4	186.764	222.064

Then from Table 4.33

$T_{pinch(Shifted)}$=155.364℃

$$T_{H,pinch} = T_{pinch(Shifted)} + \frac{\Delta T_{min}}{2} = 155.364 + 2.66395$$
$$= 158.02795℃$$

$$T_{C,pinch} = T_{pinch(Shifted)} - \frac{\Delta T_{min}}{2} = 155.364 - 2.66395 = 152.7℃$$

Table 4.33 Second Group Cumulative Summation of Heat Loads

$\Delta T_{min} = 5.3279$°C							
Shifted Temp.	Interval number (i)	$Ts_i - Ts_{i+1}$ (K)	$CP_h - CP_c$ $(\frac{Kw}{K})$	$\Delta H_{sensible}$ (Kw)	ΔH_{latent} (Kw)	$\Delta H_{total} = \Delta H_{sensible} + \Delta H_{latent}$ (Kw)	**Heat Flow** (Kw)
Ts1=335.83605							0
Ts2=222.73605	1	113.1	11.57096	1308.675576		1308.675576	1308.7
Ts3=222.06395	2	0.6721	47.31181	31.7982675		31.7982675	1340.5
Ts4=221.43605	3	0.6279	-240.2753685	-150.8689039	6367.329	6216.460096	7556.9
Ts5=186.76395	4	34.6721	-231.8795785	-8039.751933	8131.156	91.40406732	7648.3
Ts6=186.43605	5	0.3279	-223.8200433	-73.3905922		-73.3905922	7574.9
Ts7=155.36395	6	31.0721	-243.7867933	-7574.96762		-7574.96762	0.0
Ts8=155.03605	7	0.3279	35.74085	11.71942472		11.71942472	11.7

Now, by drawing heat flow against shifted temperature Grand Composite curve can be constructed as in above Table 4.33.

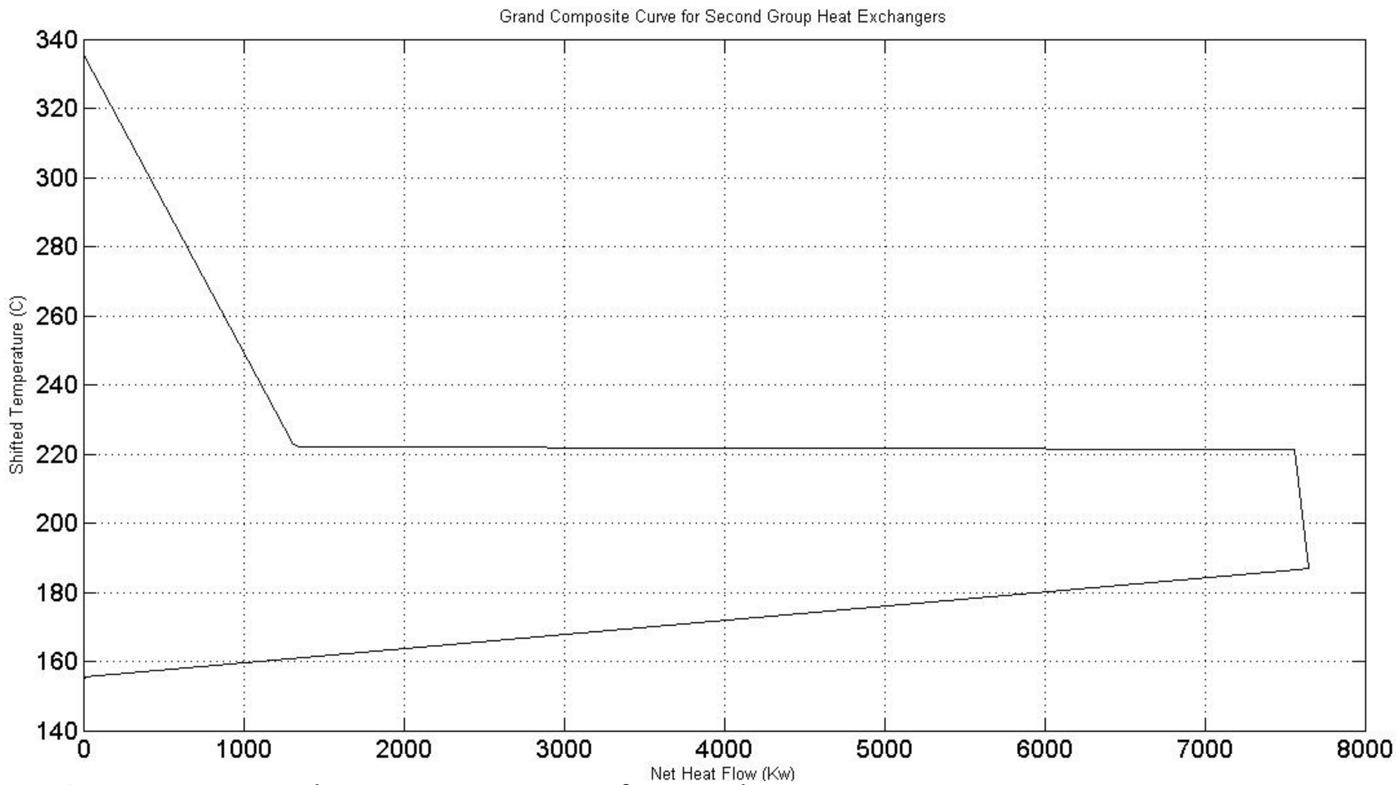


Figure 4.22 Grand Composite Curve of Second Group

By zooming-in at Figure 4.22 below $160°C$ to specify at what shifted temperature the heat flow equal zero.

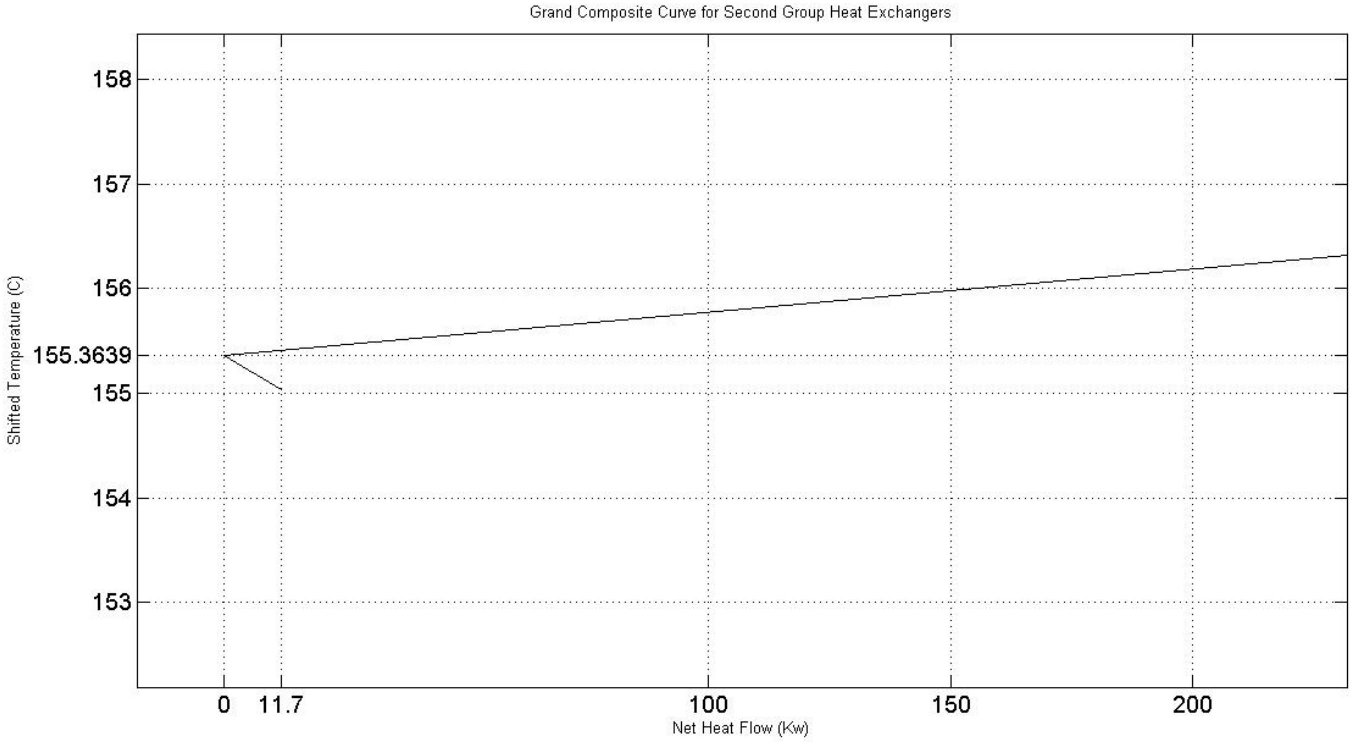


Figure 4.22a Zoomed-in Grand Composite Curve of Second Group

Shifted pinch temperature is approved 155.3639°C according Figure 4.22a.

Then hot stream pinch temperature is almost158°Cand cold stream pinch temperature is 152.7°Cis final sure result for second group heat exchanger.

4.5.3.3 Second Group Analysis Results Discussion

As mentioned in chapter(3), efficient utilization of heat exchanger net works depend on the three golden rules as follows:

- ✓ Don't transfer heat across the pinch.
- ✓ Don't use cold utilities above the pinch.
- ✓ Don't use hot utilities below the pinch.

In commencement, the three golden rules are already applied as shown in composite curve diagram the whole exchanging area is above pinch which means there is heat transferred across the pinch. Also, there are no cold utilities above the pinch and there are no hot utilities below the pinch.

Eventually, according to pinch technology analysis the second group heat exchanger network may consider achieved heat exchanging utilization.

4.6 Conclusion

In this research, the discussion was concentrated on thermal power plant performance. KNPSP(2) has been taken as case study in this research. As exergy analysis specifies low performance plant component in accordance with energy quality conversion matter; the FWHEX(1) and FWHEX(2) have been found as lowest performance components in the plant. Then there is necessity to redesign and re-evaluation for these two components. Regarding to energy quantity, FWHEX(3) is considered as worst component in the plant where high heat loss. Also, wasted work potential in KNPSP(2) is 131.3 MW which is almost equal to twice actual power of this plant, this wasted amount of work could have been converted to work and if so; the total useful work could be 191.3 MW (where actual work is 60 MW). Thus the second-law efficiency of this plant is $(60\ MW)/(191.3\ MW) = 31.4\%$.Furthermore, the full performance definition for any power plant is to specify actual work by using energy balance equation, Irreversibility and second-law efficiency by using exergy balance equations.

In the other hand, regenerative feed water heat exchangers have been discussed in this research in accordance with efficient heat exchangers network by using pinch technology analysis. The pinch analysis is point to minimize hot and cold utility in heat exchangers network and there is golden roles conclude this technique (refer to 3.2.2). if those golden roles have been applied, it means heat exchangers network is efficient. The pinch analysis is applied on closed heat exchanger type. So, KNPSP(2) feed water heat exchangers has been grouped as two groups. First group is from FWHEX(1) until FWHEX(4) and second group is FWHEX(5) and FWHEX(6).the result is that first group of heat exchanger is inefficient heat exchange network where golden roles have beenviolated at FWHEX(1), however, the second group is considered efficient heat exchange network where golden roles have been applied without violation.

4.7 Recommendations

- To judge how good a power plant performance is , should know its actual work, irreversibility and second-law efficiency. Then selection of a power plant would be built in specified area should be after collection of data about avarege temperature and pressure along year and then semulate the power plant performance in different period along year to get average actual work, irrevesibility and second-law efficiency. Thus, it is sintifcly descision will be taken about how good potential to build this power plant in this specified area.
- For power plant complex operation , it is normal to decide what thermal power plant between manyare available in the complex should be shared into grid . like this decision sometimes is taken based on availability of the thermal plant where maintenance records and so on. But for econimical operation , the lower the power plant irreversibility is the better the best selection.thus it is important to know as operating engineer or operation manager the irreversibility of each thermal power plants in the complex to enssure efficient operation decisions are taken.
- Closed monitering for power plant performance is not easy job; but by using exergy analysis in power plant periodecly , it is possible to know at what component of the plant irrevesibility increases. Thus there is opertunity to enhence thermal efficiency of plant by replacing this component or applying deferent maintenance.
- Any heat exchange network should be subjected to pinch technology to optemize the hot and cold utilities. Optemizing utilities is considered as reducing of additional operating energy for the plant to enhance thermal efficiency of the plant.
- As extention for this research, there is need to conduct comperhancive study such what had been done in this research in all power plant in the Sudan to moniter all where efficient performance. The essensial matter behind these efforts is to enssure econimical operation and lost energy means unnecessary additional cost.

References

[1] Alovrs, et al.(1996), " Exergy analysis versus pinch technology", ECOS'96

[2] Ataei and Yoo (2010), " combined pinch and exergy analysis for energy efficiency optimization in a steam power plant" , Full length research paper.

[3] Ataei (2011), " Application of combined pinch and exergy analysis in retrofit of an olefin plant for energy conservation", Scientific research and essays vol.6(12), PP. 2437-2446.

[4] Al-Doori (2012), " Exergy analysis of a gas turbine performance with effect cycle temperatures", IJRRAS 13(2).

[5] Amir (2012), "Improving steam power plant efficiency through Exergy analysis: A review ", ICMPAE'2012.

[6] Ganapathy, et al.(2009), "Exergy analysis of operating Lignite Fired thermal power plant", Journal of engineering science and technology review 2(1) 123-130.

[7] Jafari ,et al., "Exergy analysis of Shazand power plant's Boiler", Department of Mechanical Engineering, K.N. Toosi university of technology.

[8] Kwambi (2005), "Exergy Analysis of OLKARIA I power plant, Kenya", the united nations university, Geothermal training programme.

[9] Kaushik, et el (2011), " Energy and Exergy analyses of thermal power plants: A review", Renewable and sustainable reviews 15 (1857-1872).

[10] Mborah and Gbadam (2010), "On the energy and exergy analysis of 500MW steam power plant at Benso oil palm Plantation (BOPP)", Research Journal of environmental earth sciences 2(4):239-244.

[11] Rashad and ElMaihy (2009), " Energy analysis of steam power plant in Egypt", 13th International conference on AEROSPACE SCIENCES & AVIATION TECHNOLOGY.

[12] Sciubba (2007), " A brief commented History of exergy from the beginnings to 2004", Int.J.thermodynamics Vol.10(No.1) PP.1-26.

[13] Siahaya (2011), " Energy, Exergy and thermoeconomic analysis of LAHEDONG BINARY – Cycle Geothermal Power Plant at NORTH AULAWESI INDONESIA", the 7th international conference on fluid and thermal energy conversion.

[14] Sanjay and Dr. Mehta," Easy method of exergy analysis for thermal power plant", International Journal of advanced engineering research and studies.

[15] Vundela, et al. (2010), " An approach to analyse energy and exergy analysis of thermal power plants : A review ", Smart Grid and renewable energy.

[16] Vosough, et al.(2011), " Exergy concept and its characteristic", international journal of Multidisplinary sciences and engineering vol.2,No.4.

[17] Yunus A.Cengel & Michael A.Boles, " Thermodynamics – An engineering approach", Fifth edition

[18] Ian C.Kemp, "Pinch analysis and Process integration", Second edition