AUGUST 2019

M.Sc. in Mechanical Engineering

NECDET SUAT ATAY

REPUBLIC OF TURKEY

GAZİANTEP UNIVERSITY

GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES

ENERGY RECOVERY USING ORGANIC RANKINE CYCLE (ORC) FROM MUNICIPAL SEWAGE SLUDGE AS SUSTAINABLE ENERGY RESOURCE

M.Sc. THESIS

IN

MECHANICAL ENGINEERING

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MECHANICAL ENGINEERING DEPARTMENT

Name of the Thesis : Energy Recovery Using an Organic Rankine Cycle (ORC) from Municipal Sewage Sludge as Sustainable Energy Resource

Name of the Student : Necdet Suat ATAY

Exam Date : 7 August 2019

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Necdet Suat ATAY

ABSTRACT

ENERGY RECOVERY USING AN ORGANIC RANKINE CYCLE (ORC) FROM MUNICIPAL SEWAGE SLUDGE AS A SUSTAINABLE ENERGY RESOURCE

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M.Sc. in Mechanical Engineering
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August 2019
115 pages

In this study, thermodynamic analysis of the wastewater sludge incineration plant established for incineration of waste sludge from GASKI wastewater treatment plant is performed based on plant actual data. The temperature of the waste gas from the combustion plant is 279.95 ºC and the mass flow rate is 5.258 kg/s. In order to use this waste heat, an organic Rankine cycle (ORC) is designed in which the waste gas was used as a source. In the designed system, seven different working fluids are simulated using four different pressure ratios and two different mass flow rates. The most suitable fluid for the system as a result of simulation is found to be n-pentane. The pressure ratio is 20 and the mass flow rate is 2 kg/s. The net work (electricity) produced in the n-pentane system operating with these values is 198.68 kW. The exergetic efficiency of the system is 44.59%. In addition to this work, hot water production is found in the system at a mass flow rate of 6.576 kg/h at 60ºC.

Key Words: Sewage Sludge Organic Rankine Cycle, n-pentane, Incineration Plant Thermodynamic Analysis

ÖZET

SÜRDÜRÜLEBİLİR BİR ENERJİ KAYANĞI OLARAK EVSEL ATIK SU ÇAMURUNDAN ORGANİK RANKINE ÇEVRİMİ KULLANARAK ENERJİ GERİ KAZANIMI

ATAY Necdet Suat
Yüksek Lisans Tezi, Makine Mühendisliği
Danışman: Doç Dr. Ayşegül ABUŞOĞLU
Ağustos 2019
115 sayfa

Bu çalışmada GASKI atıksu arıtma tesisinden çıkan atık çamurun yakılması için kurulan gerçek bir yakma tesisinin verileri kullanılarak termodinamik analizi yapılmıştır. Yakma tesisinden çıkan atık gazın sıcaklığı 279,95ºC ve kütlesel debisi 5,258 kg/s'dir. Bu atık ısıdan yararlanabilmek için atık gazın kaynak olarak kullanıldığı bir Organik Rankine Çevrimi (ORC) tasarlanmıştır. Tasarlanan sistemde yedi farklı akışkan, dört farklı basınç oranı ve iki farklı kütlesel debi kullanılarak simulasyon yapılmıştır. Simülasyon neticesinde, sistem için en uygun akışkan n-pentan olarak tespit edilirken optimum basınç oranı 20 ve kütlesel debi de 2 kg/s olarak bulunmuştur. Bu değerler ile çalışan n-pentan sisteminde üretilen net iş (elektrik) 198,68 kW, sistemin toplam ekserji verimi de %44,6 olarak bulunmuştur. Üretilen işe (elektriğe) ek olarak sistemde 60ºC de 6,576 kg/s kütlesel debide sıcak su üretimi gözlenmiştir.

Anahtar Kelimeler: Atık Su Çamuru, Organik Rankine Çevrimi, n-pentan, Termodinamik Analiz, Yakma Tesisi.

‘’Dedicated to my family’’

ACKNOWLEDGEMENTS

I would first like to thank my thesis advisor Assoc. Prof. Dr. Ayşegül Abuşoğlu from the Mechanical Engineering Department at the University of Gaziantep. She consistently allowed this study to be my own work but steered me in the right direction whenever she thought I needed it. I must express my very profound gratitude to my mother, to my wife for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this thesis. This accomplishment would not have been possible without them.

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LIST OF SYMBOLS

kWh	Kilowatt-hours
kW	Kilowatt
MW	Megawatt
Ex	Exergy (kJ/kg)
T	Temperature (^oC)
$\dot{m}$	Mass flow rate (kg/s)
h	Enthalpy (kJ/kg)
s	Entropy (kJ/kg*K)
$\dot{W}$	Power
P	Pressure (bar)
ε	Exergy efficiency
Δ	Difference
η	Energy efficiency
Ψ	Specific flow exergy (kJ/kg)
u	Universal
f	Functional
s	Source
is	Isentropic
a	Actual
mech	Mechanical
wf	Working fluid
exh	Exhaust

LIST OF ABBREVIATIONS

BFBC	Bubbling Fluidized Bed Combustion
CAC	Charge Air Cooler
CCHP	Combined Cooling Heating and Power
CFB	Circulating Fluidized Bed
CFBC	Circulating Fluidized Bed Combustion
CPU	Central Processing Unit
DFC	Down-Flow Combustion
EEA	Eroupean Environment Agency
EfW	Energy From Waste
EORC	Organic Rankine Cycle With Ejector
EU	European Union
EVP	Evaporator
FBC	Fluidized Bed Combustion
GASKI	Gaziantep Water and Sewage Plant
GWIT	Geothermal Water Inlet Temperature
GWP	Global Warming Potential
HDPE	High Density Polyethylene
HRGV	Heat Recovering Vapor Generator
I/O	Input/Output
IGWT	Intermediate Geothermal Water Temperature
LHV	Low Heating Value
LNG	Liquified Natural Gas
MA-ES	Multi Approach Evaluation System
MGT	Micro Gas Turbine
MHF	Multiple Heart Furnace
MSW	Municipial Solid Waste
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
ORCC	Organic Rankine Cycle Condenser

ORCP	Organic Rankine Cycle Pump
ORCT	Organic Rankine Cycle Turbine
OW	Organic Waste
PFBC	Pressurized Fluidized Bed Combustion
PR	Pressure Ratio
RDF	Refuse Derived Fuel
RFB	Rotating Fluidized Bed
SOFC	Solid Oxide Fuel Cell
SSIP	Sewage Sludge Incineration Plant
TIT	Turbine InletTemperature
TSORC	Two Stage Organic Rankine Cycle
VFBI	Vortexing Fluidized Bed Incinerator
WTE	Waste to Energy
WWTP	Waste Water Treatment Plant

CHAPTER 1

INTRODUCTION

1.1 Background

Energy is a phenomenon that our world needs and we are becoming more and more dependent on. Many methods have been tried to meet this increasing need for centuries. Mankind's first energy need was to warm up and, thanks to this need, he invented fire. Afterwards, the developing civilizations and societies tried to do bigger jobs and realized that manpower would not be able to do these things. This need brought about mechanization. The first settled civilization started to produce agricultural machinery for agricultural works. Then the need for different machines for different jobs continuously increased, and this dynamic has brought us to today. The most important work of the invented machines was to convert energy from one form to another. The first machine that converts energy into its various forms is the steam engine made by James Watt. One of the questions that came to mind with the first machine produced was the efficiency of the machines and this question has remained on the agenda in every energy cycle until today. It is predicted that fossil fuels, currently used as energy sources, will be exhausted or significantly reduced in a few decades. Therefore, it is very important to look for alternative energy sources as well as how to increase the energy efficiency of energy cycles.

Although renewable energy seems to be the solution to the search for energy resources, it is important to use them efficiently as usual. One of the renewable energy sources is wastewater sludge, which is a human product. The energy potential of wastewater sludge is higher than some coal types, such as swamp coal and lignite. Although the energy potential of wastewater sludge varies from country to country, it is a very important renewable energy source that is increasing day by day in the world. While some countries have realized this source, some countries still consider wastewater sludge only as fertilizer.

With the developing world, the newly invented energy cycles paved the way for more efficient use of energy. Thus, the net work and efficiency of energy systems have increased significantly. Compared to the big systems, the efficiency of the small systems has increased tremendously and the recovery of the waste heat in the system operation has been increased. Especially in an energy cycle with low energy potential, one of the best ways to ensure that waste heat thrown out of the system is converted to reusable energy is to use organic Rankine cycle (ORC). The ORC contains the same components as the standard Rankine cycle, but the only difference is the working fluid used. While water or similar fluids are used in the standard Rankine cycle, organic liquids selected according to the temperature of the heat source are used in the ORC.

In this study, a thermodynamic analysis of an actual sewage sludge incineration plant is carried out. Together with the analysis, the waste heat generated in the system is determined as the heat source of the ORC designed with a computer program. The designed system is simulated with a computer program, and energy and exergy efficiencies of the system and net work production are calculated. Although seven different organic work fluids (n-pentane, benzene, iso-pentane, toluene, R134a, ethanol, cyclohexane) are used in the system, five different pressure ratios (1-5 bars, 1-10 bars, 2-10 bars, 2-20 bars and 1-20 bars) are used for each work fluid and two different mass flow rates (2kg/s and 2.5 kg/s) are used for each pressure ratio. As a result of the simulation performed for each model, the best working fluid, the best pressure range and the best mass flow rate are determined by considering the net work and efficiency produced by the system.

1.2 Scope and Outline of the Study

In this thesis, firstly, thermodynamic analysis of the existing wastewater sludge incineration plant is performed. The procedure and formulations of such a comprehensive analysis are provided, and they are applied to an actual sewage sludge incineration plant (SSIP) located in Gaziantep, Turkey. Secondly, an ORC is designed to recover the waste heat generated in the system. The various working fluids, pressure ratios and mass flow rates of this cycle are tried and the most suitable

results are obtained for the system in terms of energy and exergy efficiencies and net work output. The outline of the study with respect to the chapters is as follows:
In Chapter 2, an exhaustive literature survey is presented. The survey is presented under five titles: sewage sludge combustion, fluidized bed incineration, thermodynamic analysis and evaluation of incineration process, ORC related to sewage sludge incineration and thermodynamic analysis of ORC.

In Chapter 3, sewage sludge statistics and an overview of energy recovery from sewage sludge in European countries and Turkey are presented. This contains the sewage potential of European countries and Turkey and disposal statistics of sewage sludge in the countries. Moreover, in detail, the number of wastewater treatment plants in Turkey, sludge production and energy potential of sludge produced in Turkey are presented.

In Chapter 4, a detailed explanation of the SSIP located in Gaziantep, Turkey is presented. In the chapter, system, operation in the facility, components of the system, investment cost and operating cost are presented with detailed figures and calculations.

In Chapter 5, thermodynamic analysis of the facility is presented. Then ORC is designed that use waste heat of the SSIP as a heat source and system is simulated with seven different working fluids. For each working fluid, five different pressure ratios and two different mass flow rates are selected as parameters. After all simulations of the models, the optimum solution is presented in terms of energy and exergy efficiencies and net work output.

In Chapter 6, conclusions drawn from the study are pointed out and certain recommendations and future works are provided.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

The history of sewage sludge incineration is based on the 1930s in the world that is located the first sewage sludge incineration plant in the United States. The process couldn’t be popular until the 1960s. The first generation constructed sewage sludge incineration plants were inefficient, needed a very high amount of fuel, and couldn’t produce any valuable energy. After time goes, new generation of sewage sludge incinerators were constructed by using developing scientific knowledge and engineering technics. Nowadays, there are many sewage sludge incineration plants to use disposal of sewage sludge.

However many improvement is added sewage sludge incineration there is still needed to be upgraded especially in manner of energy efficiency. In this paper, one of the sewage sludge incineration plants was thermodynamically analyzed and alternative ways of increasing energy efficiency are investigated. Hypothetically design of the Organic Rankine Cycle (ORC) is the main topic of this study.

2.2. Sewage Sludge Combustion

The Sewage sludge combustion process is one of the most important disposal methods for municipal sewage sludge and many researchers have studied this issue. Many incinerator types are used in sewage sludge incineration plants but most important are rotary kiln incinerator, multiple heart incinerator and fluidized bed incinerator. Many studies are given in below that can be found some important information about sewage sludge combustion and related issues with sewage sludge combustion.

Mininni et al. [1] studied a design model of sewage sludge incineration plants. Without drying of sewage sludge, fuel used for incineration amount (149-192 Nm^3/ton sludge cake at 25% concentrations) is much more than fuel used for incineration dried sewage sludge (20Nm^3/ton sludge cake at 44% concentration). Werther & Ogada [2] studied various issues related to sewage sludge combustion. Treatment methods of sewage sludge, thermal processing sewage sludge including mono-combustion, co-combustion and alternative process, co-combustion with pulverized coal and co-incineration with municipal solid waste in various furnaces, alternative thermal process to sludge combustion, the importance of the drying and devolatilization processes for sewage sludge combustion are some of issues in the study. With this study, many kind of incineration was realized by researchers in manner of investigation. Flaga [3] suggest that only long term sludge thermal utilization is incineration of sludge. Also the sludge which is incinerated should be dried because of prevent high fuel consumption. CChem & Hudson [4] studied if incineration is suitable case for sewage sludge or not and the study showed that, incineration is one of the best method for sewage sludge if incineration method is provided by scientific and engineering approach. After incineration is concerned with sewage sludge disposal methods, many scientists have worked on this subject. One group of these scientists was worked on mathematical modelling of fluidized bed sewage sludge incineration and this transient mathematical modelling of fluidized bed incinerator for sewage sludge may effect significantly to reduce effort of experimental study [5]. Searches have increased from day to day, and more details were studied in this area. The mathematical model of a large-scale sewage sludge incineration plant was developed. The model assumes the bed to consist of a fast gas phase, an emulsion phase, and a fuel particle phase with specific consideration for thermally – thick fuel particles. This model predicts fluctuation of sewage sludge bed response against sludge feeding rate variation [6].

New type of sewage sludge incinerator was studied by Murakami et al. [7]. This new type of incinerator proposed combination of turbocharger driven by flue gas and pressurized fluidized bed combustor. Also, in this study, the operation and combustion characteristics of a demonstration plant were clarified, and the design data for a commercial plant were obtained. On the other side, it was claimed that classic combustion of sewage sludge is not a good choice for sewage sludge disposal when it is compared with co-combustion of sewage sludge with natural resources, pyrolysis, and gasification and combined processes [8].

The EU countries such as Poland go further steps in thermal sewage sludge utilization. The restriction of the EU in sludge using at agricultural area without treatment pushed these countries to utilize sewage sludge. The sludge utilization gains more and more popularity in Poland due to decline of environmental and agricultural usage of sewage sludge as well as its storage. The sludge incineration provides full utilization and removal of the sludge from the environmental system. The products of this system are sanitary safe. It is probably the best way to reduce sludge volume and weight [9]. Lin & Ma [10] also studied about co-incineration simulation of sewage sludge with municipal solid waste (MSW). It was studied co-incineration of sewage sludge and MSW with different dryness ratios of both. According to study, co-incineration of 10 wt.% wet sludge with MSW can ensure the furnace temperature, the residence time and other vital items in allowable level, while 20 wt.% of semi-dried sludge can reach the same standards. With lower moisture content and higher low heating value (LHV), semi-dried sludge can be more appropriate in co-incineration with MSW in a grate furnace incinerator.

After incineration of sludge, ash is formed as a byproduct. Donatello & Cheeseman [11] investigated the usage of sewage sludge incineration ash in a better and more efficient way unlike usage landfilling. The environmental effects and burden of thermal utilization of sewage sludge were studied by Xu et al. [12]. The study showed that a sewage sludge-treatment scenario with anaerobic digestion, dewatering, and incineration technologies was the most environmentally and economically suitable method to treat sewage sludge because of energy recovery. All new sewage treatment plants should be constructed to operate according to this method, and existing plants should be retrofitted. After all development, optimization of treatment of sewage sludge was one of the important issues. Vadenbo et al. [13] studied this treatment optimization. Many environmental objectives were blended with different treatment options of sewage sludge such as mono-incineration, co-incineration and etc.

Li et al. [14]. studied on optimal energy efficiency of a sludge drying-incineration combined system. Different dryness ratios of sewage sludge were analyzed and incineration performances were observed. This study can be helpful for energy efficiency in sewage sludge incineration plants. Also other researchers, Li et al. [15] studied more specific incineration system with integrated drying system. It was experimental study which capacity of experimental plant is 100 t/d. The study shows that an integrated incineration system with a drying unit is a single feasible unit of sewage sludge treatment and can be constructed successfully at intended capacity.
Zhu et al. [16] focused on the environmental problem of sewage sludge disposal. When dried sewage sludge is disposed of by two stage systems, gasification and incineration, the emission values are very promising. So in the future, this kind of disposal method can be attractive for governments that take care of environmental problems much.

Scientists who think that only one disposal method is not efficient for sewage sludge disposal try to find a hybrid system for more energy recovery and efficiency. Speidel et al. [17] studied a new process concept for highly efficient conversion of sewage sludge by combined fermentation and gasification and power generation in a hybrid system consisting of a solid oxide fuel cell (SOFC) and gas turbine. With this hybrid system, the overall wide electrical efficiency was calculated as 53%. This efficiency value is higher than one stage sewage sludge disposal methods.
Not all scientists focus just on incineration or combustion of sewage sludge. Also there are many scientists focusing effects of drying systems or drying methods for getting energy of sewage sludge combustion. Bianchini et al. [18] studied integration of WTE power plant with process for thermal drying of sewage sludge. In the study, the described integration offers a solution for the issues related to sludge disposal and generates positive effects on WTE power plant in terms of energy efficiency and water demand.

One of the last studies is sewage sludge disposal methods, with particular emphasis on combustion as a priority disposal method. The study represented the result of experimental studies aimed at determining the mechanisms (defining the fuel combustion region by studying the effects of process parameters, including the size of the fuel sample, temperature in the combustion chamber and air velocity, on combustion) and kinetics (measurement of fuel temperature and mass changes) of fuel combustion in an air stream under different thermal conditions and flow rates [19].

All these studies show that incineration is still an attractive option for disposal of sewage sludge. The topic was investigated past, still is investigated, and will be investigated in future. Understanding of which investigation was done by using which furnace type may be easier if it is presented as table 2.1;

Table 2.1 Combustion types used in studies and publishing date of the studies

No	Type of Combustion	Research Name	Writer of Research	Publishing Date
1	Fludized Bed Combustion (FBC) / Multiple Heart Furnace (MHF) Combustion	A Design Model of Sewage Sludge Incineration Plants with Energy Recovery	Mininni, G., Di Batolo Zuccarello, R., Lotito, V., Spinosa, L., Di Pinto, A.C.	1997
2	MHF Combustion / FBC / Combined MHF-FBC / Cyclone Furnace / Smelting Furnace / Rotary Furnace	Sewage Sludge Combustion	Werther, J., Ogada, T.	1999
3	FBC / Rotary Furnace / Shelf Furnace / Grill Furnace / Pyrolysis & Gasification	The Aspects of Sludge Thermal Utilization	Flaga A.	2003
4	FBC / MHF / Rotary Furnace	Incineration – Is There A Case?	CChem, P., L., Hudson, J., A.	2005
5	FBC	Transient Mathematical Modelling of a Fludized Bed Incinerator for Sewage Sludge	Khiari, B., Marias, F., Zagrouba, F., Vaxelaire J.	2006
6	Bubbling Fluidized Bed Combustion (BFBC)	Dynamic behavior of sewage sludge incineration in a	Yang, Y., B., Sliwinski, L., Sharifi, V., Swithenbank, J.	2007

		large – scale bubbling fluidized bed in relation to feeding – rate variations		
7	Pressurized Fluidized Bed Combustion	Combustion characteristics of sewage sludge in an incineration plant for energy recovery	Murakami, T., Suzuki, Y., Nagasawa, H., Yamamoto, T., Koseki, T., Hirose, H., Okamoto, S.	2009
8	Pyrolysis & Gasification / Bubbling Fluidized Bed Combustion (BFBC) / Circulating Fluidized Bed Combustion (CFBC) / Rotary Furnace / Dust Boiler / SVZ Technology	A review of methods for the thermal utilization of sewage sludge: Polish perspective	Werle, S., Wilk, R., K.	2010
9	Fluidized Bed Combustion	Thermal utilization of municipal sewage sludge – examples of polish solutions	Latosinska, J., Turdakow, A.	2011
10	Grate Furnace Combustion	Simulation of co-incineration of sewage sludge with municipal solid waste in a grate furnace incinerator	Ma, X., Lin, H.	2012
11	Fluidized Bed Combustion (FBC)	Recycling and recovery routes for incinerated sewage sludge ash (ISSA): A review	Donatello, S., Cheeseman, C., R.	2013
12	Fluidized Bed Combustion (FBC)	Multi – objecti optimization of waste and resource management in industrial networks – Part II: Model	Vadenbo, C., Guillén-Gosálbez, G., Saner, D., Hellweg, S.	2014

		application to the treatment of sewage sludge		
13	Circulating Fluidized Bed Combustion (CFBC)	Integrated drying and incineration of wet sewage sludge in combined bubbling and circulating fluidized bed units	Li, S., Li, Y., Lu, Q., Zhu, J., Yao, Y., Bao, S.	2014
14	Down-Flow Combustion (DFC)	Experimental investigation of gasification and incineration characteristics of dried sewage sludge in a circulating fluidized bed.	Zhu, J-G., Yao, Y., Lu, Q-g., Gao, M., Ouyang, Z-q.	2015
15	Solid Oxide Fuel Cell (SOFC) Combustion	A new process concept for highly efficient conversion of sewage sludge by combined fermentation and gasification and power generation in a hybrid system consisting of a SOFC and a gas turbine.	Speidel, M., Kraaij, G., Wörner A.,	2015
16	Fluidized Bed Combustion (FBC)	Mechanism and kinetics of granulated sewage sludge combustion.	Kijo-Kleczkowska, A., Środa, K., Kosowska-Golachowska, M., Musiał, T., Wolski, K.,	2015

2.3. Fluidized Bed Incineration

Sewage sludge incineration plants use different types of bed in incineration system. In this study, fluidized bed incineration is focused as one kind of incineration type of sewage sludge. Also, there are many past study in this issue and it will be more study for the issue.

One of the oldest studies is about an advanced fluidized-bed swirl incinerator for dioxin control during municipal waste disposal. In the study, the principles and performance of an advanced fluidized-bed incinerator installed with an independently controlled secondary swirl combustion chamber. Also, behavior of dioxins and their precursors as well as their homologue distribution along gas flow was studied [20]. Another study including fluidized bed combustor is about influence of combustion conditions on dioxin in an industrial fluidized-bed incinerator. The study was experimental and it aims measuring incineration process parameters and dioxin emissions are measured under different operating conditions. Dioxin level measured at boiler outlet and it is mainly influenced by combustion conditions. Higher flue gas O_2 level, secondary to primary air ratio and total air supply are favorable for lower dioxin formation [21]. Miyamoto et al. [22] studied on an operating fluidized bed incinerator, where a hybrid system consisting of fuzzy systems and neural networks has been realized, which assesses the fuel-feeding state on the basis measured values and combustion image processing, and operates with low CO/NO_x concentrations by means or air-fuel ratio control. One of the most important studies was published by J. V. Caneghem et al [23]. In the study, design, operational and environmental issues were explained and inspected the parameters that govern the design and operation of these incinerators. The design strategy of a fluidized bed incinerator was outlined, which involved considerations of hydrodynamic (velocities, mixing) thermal (heat balances) and kinetic (reaction rate burnout) nature. One of the newest investigations about fluidized bed combustor has been published by Shukrie et al [24]. In the study, high pressure drop in fluidized bed system problem was tried to solve by designing novel air distributor that consists of circular edge segments that contributed to low pressure drop and improvement of heat and mass transfer in fluidized bed combustor.

In 1998, United Kingdom prohibited sea dumping of sewage sludge. With this regulation, researchers from Sheffield University investigated sewage sludge incineration in a new type rotating fluidized bed incinerator (RFB). In the study, large quantities of sewage sludge was burned in the incinerator and as a result sludge throughput for the Rotating Fluidized Bed (RFB) incinerator was significantly higher than the conventional fluidized bed incinerator [25]. Also fluidized bed incinerator was not only studied for sewage sludge but also other materials such as wool scouring sludge. In experimental study, one vertical axis rotating fluidized bed (RFB) designed and one of experiment was investigation of fluidization performance of the RFB. The experimental results obtained have suggested that incineration was successful and ash particles elutriated were fine due to good missing and turbulence in the RFB. This also reflects the RFB as an effective incinerator [26].

There are investigations for fluidized bed in Japan as every developed country. Piao et al. [27] investigated combustion of refuse derived fuel (RDF) in fluidized bed with different two commercial sized RDF. Two type RDF was used and thermogravimetric measurements were done such as some flue gases measured with changing some parameters of fluidized bed during incineration. There is a case study in literature and it shows that changing multiple heart furnace by fluidized bed incinerator is more advantageous both economically and environmentally [28]. In 2006, Hernandez & Atonal et al. [29] investigated combustion of refuse – derive fuel in a fluidized bed. Main objective of the study was to investigate the RDF combustion characteristics and the associated pollutant emissions in a fluidized bed combustor.

Another topic about fluidized bed incinerator is co-combustion. Cliffe, & Patumsawad investigated co-combustion of waste from olive oil production with coal in fluidized bed [30]. Another co-combustion of municipal solid waste with some type of coal was investigated by Suksankraisorn et. al. [31]. In the study, effects of mass fraction of a model MSW in the fuel mixture, as against 100% lignite, on the combustion characteristics that may be inferred from temperature distributions, carbon combustion efficiency and CO and CO_2 concentrations and on the emission of major gaseous pollutants, including CO, SO_2, NO and N_2O. Municipal solid waste burning in a fluidized bed incinerator is also another topic about fluidized bed incineration. Same researchers (Cliffe, & Patumsawad) [32] demonstrated the technical feasibility of a fluidized bed as a clean technology of burning high moisture municipal solid waste. For co-firing high moisture two type MSW (vegetable waste and olive oil waste) at different moisture contents with coal, the combustion efficiency and the flue gas composition were investigated and compared to burning 100% coal.

After all these studies, some researchers published a book to help understanding combustion and incineration processes and systems including fluidized bed incineration. A book as source about fluidized bed combustion was published by Faulkner [33]. In this book, two types of fluidized bed combustion – bubbling and circulating FBC are focused. Also development of fluidized bed combustion boilers, fundamental processes in fluidized bed combustion boiler furnaces and fluidized bed combustion applications are explained. Detailed basic knowledge was given in the book [34]. Suksankraisorn et al. [35] also investigated burning three type high moisture municipal solid waste in fluidized bed incineration. “Fluidized Bed Technologies for Near-Zero Emission and Gasification” edited by Scala [36-43] contains many information about fluidized bed combustion. Many researchers contributed chapters of the book. Over viewing of fluidization science and fluidized bed technologies, pressurized fluidized bed combustion (PFBC) combined cycle systems, conversion of liquid and gaseous fuels in fluidized bed combustion and gasification, fluidized bed reactor design and scale-up, modelling of fluidized bed combustion processes, atmospheric (non-circulating) fluidized bed (FB) combustion, pressurized fluidized bed combustion and circulating fluidized bed combustion are some of chapters of the book. Another source book is “Thermal Power Plant Design and Operation” by Sarkar [44]. Basic and technical details and also some instructive problems are given in chapters of the book.

The use of fluidized bed combustion system brought some problems with it such as low thermal efficiency, high emissions, bed agglomeration. Valuable experiences were gained from the study that belongs to Yan et al. [45]. Fluidized bed combustor is used co-combustor of agricultural residues. Residues are burned with coal or any other fuel. Ghani et al. [46] explained the behavior of biomass-fired fluidized bed incinerator, biomass from agricultural residues (rice husk and palm kernel) were co-fired with coal in a 0.15 m diameter and 2.3 m high fluidized bed combustor. The combustion efficiency and carbon monoxide emissions were studied and compared with those for pure coal combustion. Different materials combustion in fluidized bed combustor was analyzed and studied. F. Burgess et al. [47] studied combustion of high density polyethylene (HDPE) polymer pellet in a bubbling fluidized bed and measured concentrations of CO and CO_2 in the off-gas, enabling burnout-times to be derived with different combustion temperatures in range of 400 and 900^{o}C. Liu et al. [48] studied process simulation of formation and emission of NO and N_2O during decoupling combustion in a circulating fluidized bed combustor using Aspen Plus.

Among all experimental and case study, some different approaches were applied to fluidized bed combustion. Mathematical modelling of sewage sludge incineration in a bubbling fluidized bed (BFB) with special consideration for thermally thick fuel particles is one of these different approaches. In the study, mathematical model was developed for the simulation of a large scale sewage sludge incineration plant [49]. Another detailed approach was experimental analysis of bubble velocity in a RFB. The bubble velocity in a two-dimensional RFB was experimentally analyzed. The motion of bubbles was observed through a high-speed video camera, and the radial and angular components of bubble velocity were experimentally measured [50]. Adamczyk et al. [51] studied numerical simulations of the industrial circulating fluidized bed boiler under air- and oxy-fuel combustion. Measured and numerical results of air-fuel combustion process within large scale industrial circulating fluidized bed (CFB) boiler was presented.

Palm plantation is another biomass source that fluidized bed combustor can be used especially in developing countries such as Malaysia, Thailand and Indonesia. Razuan et al. [52] studied combustion of oil palm stone in a pilot-scale fluidized bed reactor. The main objective was to investigate the thermochemical conversion of oil palm stone in a pilot-scale fluidized bed combustor.

Kitchen waste can be counted as MSW also. So, incineration of this waste became a topic for research by Duan et al [53]. In the study, incineration of kitchen waste with high nitrogen in vortexing fluidized-bed incinerator (VFBI) and its NO emission characteristics was investigated. As result, compared with other types of combustor, VFBI reduces the CO and NO emission concentrations much better when burning MSW with high nitrogen content.

Okasha & Zeidan [54] studied propane combustion in a novel fluidized bed configuration. Experimental study on propane combustion was performed in a novel fluidized bed configuration. This configuration has a jet that issues vertically in the upper part of the bed while allowing two methods of feeding which they are jetting air-propane mixture and staged-air combustion technique respectively.

Researchers studied different types of Energy from Waste (EfW) incinerator plants. Four suitable case study incinerator plants were chosen to compare technologies (moving grate, fluidized bed and rotary kiln), plant economics and operations. It was concluded that one of the major difficulties encountered by waste facilities is the appropriate selection of technology, capacity, site waste suppliers and heat consumers [55]. Another experimental study was about comparisons of polypropylene combustion in porous and nonporous alumina bed materials were conducted in a semi-pilot scale fluidized bed combustor. The result indicated that polypropylene can be effectively used as a fuel in both bed materials. The combustion efficiencies of polypropylene in porous alumina and nonporous alumina are above 99.3% [56].

Co-combustion hazardous waste in existing fluidized bed combustor is an alternative to hazardous waste treatment facilities. Tannery sludge is a kind of hazardous waste. It is fit for co-combustion with coal in fluidized bed boilers. When mono-combustion of bituminous coal and combustion of tannery sludge with coal was compared, the result showed that Cd, Ni, Cu, Pb and Mn amounts are almost same but As and Cr amount is different from mono-combustion and higher than limits [57].

2.4 Thermodynamic Analysis and Evaluation of Incineration Process

In the literature, there are some studies related with thermodynamic analysis of sewage sludge incineration even not as many as fluidized bed incineration.

Miyakazi et al. [58] published one of the oldest studies. In the study, the objective was to develop a combined power generation cycle using incineration and LNG cold energy and to conduct parametric analysis to investigate the effects of the key parameters on the thermal and exergy efficiencies. It was found that the thermal and exergy efficiencies of combined cycle were 1.53 and 1.43 times higher than those conventional cycles, respectively. In another study, thermogravimetry was used for measurement of parameters of combustion modifications of three different sludge during combustion mixed with coal. The combustion of pyrolyzed sewage sludge was studied [59]. Potentialities of using gasification as a part of two-stage process for incineration of industrial and hazardous waste was described by Bébar et al [60]. The application of gasification technology brings about a whole range of benefits inclusive the possibility of realization of thermal decomposition in temperature range below melting point of ash as well as a considerable decrease of the amount of flue gas originating in the process. The energy analysis of the systems was done in experimental manner. With tightening legislation and obligation to reduce environmental impact of sludge disposal, expectations on wastewater sludge treatment increased. One of the studies was published to meet this expectation. In the study, the aim was to analyze the performance of a heat and power generating sludge combustion plant from technical and economical viewpoints and to compare the studied concept to optional sludge treatment technologies. Three plant concepts were selected (co-generation, heat-only and pure electricity generation). The selection of the optimal technology for sludge treatment was a complicated task [61].

As many country, Spain is interested about this issue. Gomez et al. [62] from Spain studied electricity generation from human and animal waste in Spain by using three different wastes such as municipal solid waste, sewage sludge, and livestock manure. Also several energy-recover options were analyzed for the first one, in other words the collection of landfill gas, incineration and anaerobic digestion. The study showed that, most economical option is the incineration of municipal solid waste, with an entry cost of around 4c€/kWh. Another study was about efficiency of energy recovery from waste incineration. In the study, in the light of Waste Framework Directive (Directive 2008/98/EC of the European Parliament and of the Council of 19 November 2008 on waste and repealing certain Directives), which allows high efficiency installations to benefit from a status of “recovery” rather than “disposal”. In the study the systems was critically analyzed scientific- based approach of exergy efficiency [63]. Garbage incineration systems are originally designed solely purpose of disposing urban waste. Nowadays the schemes which are characterized by best using of energy resources are the only ones which are economically justified. Thus, cogeneration systems have been developed for the better usage of energy resources. Meratizaman et al. [64] studied a waste incineration system that was coupled with power cycle. The power cycle used hot gas produced from combustion of waste material. Cooling energy of Liquefied Natural Gas (LNG) is used to cool the operating fluid in the power cycle.

Combined cooling heating and power (CCHP) systems for waste incineration is one of important issue. Gao et al. [65] studied energy matching and optimization for this kind of systems. CCHP system as a poly-generation technology has received an increasing attention in field of small scale power systems for applications ranging from residence to utilities. It will also play an important role in waste to energy application for megacities. In the study, energy level and exergy analysis are implemented on energy conversion processes to reveal the variation of energy amount and quality in the operation of CCHP system. In one of the latest study related with energy and exergy analysis of sewage sludge incineration is a case study for plant located in Bergen, Norway. Both first-law and second-law efficiency of a combined heat and power plant were calculated. Both household and industrial waste was converted into electricity and district heating by incineration [66].

2.5 Organic Rankine Cycle Related to Sewage Sludge Incineration

In the literature, many papers were published about Organic Rankine Cycle (ORC) too. Studies about ORC related to sewage sludge incineration were given in below paragraphs.

Sung et al. [67] analyzed the thermoeconomic of a biogas-fueled micro-gas turbine (MGT) system, which is coupled with a bottoming ORC for a target biogas plant in Busan, Republic of Korea. Maria and Micale [68] studied an integrated anaerobic digestion (AD) and aerobic bioconversion facility, equipped with an ORC, was analyzed for the management and recovery of energy from organic waste (OW). These authors also were studied exergetic and economic analysis of energy recovery from the exhaust air of organic waste aerobic bioconversion by ORC [69].

Tchanche et al. [70] studied about relations between heat resource and ORC. Various Rankine cycle architectures for single fluids and other improved versions operating with ammonia/water mixture were presented in the paper. Untapped heat resources and their potential for driving ORC were outlined. The nature – state and temperature of the heat source significantly influences the choice of the type of ORC machine.

Fraia et al. [71] studied ORC system with different approach. In this work, the use of geothermal energy is proposed for electric and thermal energy generation for wastewater and sludge treatment. An energy, exergy and economic analysis for the developed system are carried out. The study is carried out for a district wastewater treatment plant on the island of Ischia, in southern Italy, which presents diffused low-medium enthalpy geothermal sources, considered in this work to power an ORC system for electric energy production and to heat the desiccant flow for sludge drying.

Quoilin et al. [72] studied overview of different ORC applications including sewage sludge incineration. Also market review and manufacturers of ORC systems was presented. An in-depth analysis of the technical challenges related to the technology, such as working fluid selection and expansion machine issues was then reported.

One of the recent studies about ORC was studied by Mahmoudi et al. [73]. In the study a review of studies both theoretical and experimental on ORC usage for waste heat recovery and investigation on the effect of cycle configuration, working fluid selection and operating condition on the system performance, which have been developed during the last four years were presented. Also the related statistics were reported and compared regarding the configuration and the employed working fluid with type of the heat source.

2.6 Thermodynamic Analysis of ORC

On the literature, there are also many paper studied about thermodynamic analysis, energy accounting and exergy accounting of ORC systems. On below, it can be founded many studies from oldest to newest in manner published date who explained energy, exergy and thermodynamics of ORC in different systems.

One of the oldest studies belongs to Invernizzi et al. [74]. In the study the possibility of enhancing the performances of micro-gas turbines through the addition of a bottoming ORC which recovers the thermal power of the exhaust gases typically available in the range of 250 – 300ºC was investigated. A specific analysis of the characteristics of different classes of working fluids was carried out to define a procedure to select the most appropriate fluid, capable of satisfying both environmental and technical concerns. Moreover, a thermodynamic analysis was performed to ascertain the most favorable cycle thermodynamic conditions, from the point of view of heat recovery. Low temperature ORC were studied as bottoming cycle in medium and large scale combined cycle power plants. The analysis aimed to show the interest of using these alternative cycles with high efficiency heavy duty gas turbines, for example recuperative gas turbines with lower gas turbine exhaust temperatures than in conventional combined cycle gas turbines [75].

Chen et al. [76] reviewed thermodynamic cycles and working fluids for the conversion of low-grade heat into electrical power, as well as selection criteria of potential working fluids, screening of 35 working fluids for the two cycles and analyses of the influence of fluid properties on cycle performance. The paper discussed the types of working fluids, influence of latent heat, density and specific heat, and the effectiveness of super-heating. For high – temperature ORC, working fluids were studied. Alkanes, aromates and linear siloxanes were considered as working fluids. First “isolated” ORC processes with the maximum temperatures of 250ºC and 300ºC were studied at sub or supercritical maximum pressures. With internal heat recovery, the thermal efficiencies n^{th} averaged over all substances amount to about 70 % of the Carnot efficiency and increase with the critical temperature. Second it was included a pinch analysis for the heat transfer from the heat carrier to the ORC working fluid by an external heat exchanger [77].

Some of cycles used in industrial process were compared including inverted Brayton Cycle, Stirling Engine and ORC. According to parametric investigation of thermodynamic performance ORC performed highest electric efficiency more than 20% with reference to the input heat content [78]. Another study is about performance analysis of ORC with super-heating under different hear source temperature conditions. In the study organic fluids; R-12, R-123. R-134a and R-717 were compared in manner second law efficiency, irreversibility of the system, availability ratio, work output, mass flow rate with increase in turbine inlet temperature (TIT) under different heat source temperature conditions. The calculated results revealed that R-123 produces the maximum efficiencies and turbine work output with minimum irreversibility for employed constant as well as variable heat source temperature conditions. Hence, selection of a non-regenerative ORC during super-heating using R – 123 as working fluid appeared to be choice system for converting low – grade heat to power [79].

A novel cogeneration system driven by low-temperature geothermal sources was investigated by T. Guo et al. [80]. The system consisted of a low-temperature geothermally – powered (ORC) subsystem, an intermediate heat exchanger and a commercial R134a – based heat pump subsystem. The main purpose was to identify appropriate fluids which may have yielded high PPR (the ratio of power produced by the power generation subsystem to power consumed by the heat pump subsystem) value and QQR (the ratio of heat supplied to the user to heat produced by geothermal source) value. Optimization of waste heat recovery ORC was another study. In the study, it was focused that both on the thermodynamic and on the economic optimization of a small scale ORC in waste heat recovery application. A sizing model of the ORC was proposed, capable of predicting the cycle performance with different working fluids and different components sizes. The working fluids considered were R245fa, R123, n-butane, n-pentane and R1234yf and Solkatherm [81]. One of the studies was about thermodynamic modeling of a 2 kW biomass-fired CHP system with ORC. Three environmentally friendly refrigerants, namely HFE7000, HFE700 and n-pentane, were selected as the ORC fluids. The results of modelling showed that under the simulated conditions of the study, the ORC thermal efficiency with any selected ORC fluid was well below the Carnot Cycle efficiency; the ORC efficiency depended on not only the modeling conditions but also the ORC fluid. [82].

In the literature, also there is a many studies investigate ORC working in real conditions. In of the study, the system performance of a 50 kW ORC system subject to influence of various working fluids was analyzed [83]. Energetic and exergetic efficiencies were investigated of an ORC with different heat source temperature. The thermal efficiencies of the ORC at different heat source temperatures of about 100, 90, 80, and 70ºC were explored. The thermodynamic irreversibility that taken place in the evaporator, condenser, turbine, pump, and separator was revealed [84].

Gewald et al. [85] studied producing electricity from waste heat recovery from a landfill gas-fired power plant. The aim of the paper was to study the possibilities of using large amount of heat (which power station consisted of 15 ICEs and has an installed capacity of 23.5 MW) to increase the electricity production and efficiency of the power station. The water/steam cycle and the ORC were examined and evaluated by thermodynamic cycle simulation and by calculating their specific costs of power generation. Experimental study about ORC and radial turbine using R245fa working fluid was studied. In this study, an ORC capable of generating electric power using a low-temperature heat source was developed and an experimental study was conducted. A radial turbine directly connected to the high-speed synchronous generator was also designed and developed. Experiments were conducted to analyze the operational characteristics and performance of the developed ORC [86].

Technical and market review of ORC in power generation was studied. The paper presented an overview of the technical and economic aspects, as well as the market evolution of the ORC [87]. Analysis of the use of waste heat obtained from coal – fired units in ORC and for brown coal drying is one of the study in the literature. In the study, it was presented two energy technologies that, if used will increase the efficiency of electricity generation. It was also presented an analysis of the feasibility of and potential for using waste to heat obtained from exhaust gases to feed ORC's [88].

Because of limited using of ORC with low-grade heat source in the industry due to low efficiency, researchers studied power output capacity and efficiency of an Organic Rankine Cycle with Ejector (EORC). In the EORC, an ejector and a second – stage evaporator were added to the ORC. As a result efficiency of the system was increased [89]. Chen et al. [90] studied a new design method for ORCs to fully couple the ORC with the heat source. The heat source is characterized by the mass flow rate, inlet and outlet heat carrier fluid temperatures. The result showed that a higher turbine inlet temperature required a lower turbine inlet pressure, leading to a lower system thermal efficiency. The optimal (maximum) thermal efficiency appeared at the saturated or slightly – super-heated vapor state at the turbine inlet.

In the literature two stage Rankine cycle for electric power plants was studied. In the paper, it was made of a water steam Rankine Cycle and an ORC. By using an organic fluid with higher density than water, it was possible to reduce the installation size and to use an air – cooled condenser [91]. A comprehensive thermodynamic modelling was reported of a trigeneration system for cooling, heating (and/or hot water) and electricity generation. The system consisted of a gas turbine cycle, an ORC, a single – effect absorption chiller and a domestic water heater. Energy and exergy analyses, environmental impact and sustainability were evaluated. The exergy efficiency of the trigeneration system was found to be higher than that of typical combined heat and power systems or gas turbine cycles [92].

Also there is another study about trigeneration system coupled with ORC system. In the study, energy and exergy analyses of a biomass trigeneration system using an ORC were presented. Four cases were considered for analysis; electrical-power, cooling-cogeneration, heating-cogeneration and trigeneration cases. The result obtained revealed that the best performance of trigeneration system considered can be obtained with the lowest ORC evaporator pinch temperature considered, and the lowest ORC minimum temperature. The study revealed that there is a significant improvement when trigeneration is used as compared to only electrical power production [93]. In ORC systems, sometimes it is studied each component for example evaporator, compressor and etc. In the study, calculation result for the pumping work in the ORC systems were presented. Analysis has been carried out for 18 different organic fluids that can be used as working media in subcritical ORC power plants [94]. Vetter et al. [95] studied comparison of sub- and supercritical Organic Rankine Cycles for power generation from low-temperature/low enthalpy geothermal wells, considering specific net power output and efficiency. The paper presented analysis of sub – and supercritical processes using propane, carbon dioxide and ten other refrigerants as working fluids. The impact of crucial indicators for optimization, such as specific net power, thermal efficiency and heat input was discussed in detail.

Wang et al. [96] studied another efficiency investigation for ORC in 2012. The study proposed a thermal efficiency model theoretically based on an ideal ORC to analyze the influence of working fluid properties on the thermal efficiency, the optimal operation condition and exergy destruction for various heat source temperatures were also evaluated utilizing pinch point analysis and exergy analysis. The proposed model exhibited excellent agreements with the theoretical data and showed better performance than existing models. According to the evaluation of optimal operation condition, different working fluids have little impact on the optimal operation condition of ORC and selection of working fluid reasonably based on heat source temperature will help to optimize the ORC performance. Exergy analysis indicated that the evaporator contributes the major exergy destruction while condenser has the smallest except pump. Another thermodynamic analysis and optimization of an ORC was studied in 2012. In the study, it was examined that the effects of key thermodynamic design parameters, including turbine inlet pressure, turbine inlet temperature, pinch temperature difference and approach temperature difference in HRGV (heat recovery vapor generator), on the net power output and surface areas of both the HRGV and the condenser using R123, R245fa and isobutene. The result showed that turbine inlet pressure, turbine inlet temperature, pinch temperature difference and approach temperature difference have significant effects on the net power output and surface areas of both the HRGV and the condenser [97].

Another trigeneration system investigation was studied F.A. Al-Sulaiman et al. [98-99]. The study considered two part. In the first part of study it was presented the thermoeconomic optimization formulations of three new trigeneration systems using ORC: SOFC-trigeneration, biomass-trigeneration, and solar-trigeneration systems. In the part II of the study, three new trigeneration systems were examined. These systems were SOFC-trigeneration, biomass-trigeneration, and solar-trigeneration systems. This study revealed that the maximum trigeneration – exergy efficiencies were about 38 % for the SOFC-trigeneration system, 28% for the biomass-trigeneration system and 18% for the solar-trigeneration system. Moreover, the maximum cost per exergy unit for the SOFC-trigeneration system was approximately 38 $/GJ, for the biomass-trigeneration system was 26 $/GJ, and for the solar-trigeneration system is 24 $/GJ. This study revealed that the solar trigeneration system offered the best thermoeconomic performance among the three systems. This was because the solar-trigeneration system has the lowest cost per exergy unit.

Zhou et al. [100] studied was heat recovery by using ORC from low-temperature flue gas. An experimental system for heat recovery from low-temperature flue gas based on ORC was constructed. In the system R123 was selected as working fluid, a scroll expander was used to produce work, and fin tubes heat exchanger was designed as evaporator. Low-temperature flue as produced by a liquefied petroleum gas (LPG) stove was designed as the heat source to simulate industrial flue gas, and its temperature could be controlled in the range of 90 – 220ºC. The results showed that the cycle efficiency, the output power of the expander and its exergetic efficiency increased whilst the heat recovery efficiency decreased with the increment of the evaporating pressure at a certain temperature of the heat source. Exergy analysis for maximizing power of ORC power plant driven by open type energy source is another study in the literature. In the study, geothermal water and hot air emitted from the clinker brick cooler were used as energy sources. Such energy sources were defined as the open type energy sources. The difference between the energy capacity and the energy supplied to the power plant was highlighted. It was proved that the use of the power plant thermal efficiency was not sufficient for optimization of the power plant operation, and can often guide to wrong conclusions. Analysis of the thermodynamic imperfection of the power plant cycle, based on determination of the exergy losses, appeared then as more useful [101].

In steel industry, ORC is used also for power generation. In the literature, there is a one study about power generation by ORC in steel industry. In the study, actual plant data was used for exergy and energy analysis of ORC. Variations of energy and exergy efficiencies of the system with the evaporator/condenser pressures, super-heating and sub cooling were illustrated [102].

Another case study in the literature is about power generation using waste heat recovery by ORC in oil and gas sector. The study utilized the ORC in an existing gas treatment plant in Egypt, as a case study, to recover the waste heat and convert it into electricity. A simulation Aspen HYSYS v7.1 was built up for the case study. Two different cycles, the basic and the regenerative cycles, were studied. Various working fluid were investigated using different parameters such as net work produced, efficiency, volumetric flow rate and the irreversibility. The simulation showed that regenerative cycle using either benzene or cyclohexane was the most promising choice. The capital cost and profitability study showed that benzene was more suitable as working fluid than cyclohexane [103]. Meinel et al. [104] studied effect and comparison of different working fluids on a two – stage ORC concept. The paper presented Aspen Plus (v7.3) simulations of a two – stage ORC concept with internal heat recovery. The proposed system was compared to state – of – the – art processes with four different working fluids distinguished by the slope of the saturated vapor curve in the corresponding T – s diagram.

Thermodynamic optimization of binary ORC power plants using in low- medium geothermal sources was studied in 2013. In the study, a MATLAB® code was created to define the optimal combination of fluid, cycle configuration and cycle parameters. An extensive thermodynamic analysis was performed considering geothermal sources in the temperature range of 120ºC – 180ºC [105]. Waste heat recovery form gas turbine by ORC was studied. The paper presented a comparison between four different working fluids in order to identify the best choice. The selected fluids were: toluene, benzene, cyclopentane and cyclohexane. The design was performed with a sensitivity analysis of the main process parameters and the organic Rankine cycle was optimized by varying the main pressure of the fluid at different temperatures of the oil circuit; moreover, the possible use of a superheater was investigated for each fluid to increase electrical power [106].

Ayachi et al. [107] studied ORC optimization for medium grade heat recovery. The result of the study indicated that the global exergy efficiency was strongly linked to critical temperature of the working fluid. Ibara et al. [108] studied performance of a 5kWe ORC at part-load operation. The paper analyzed the performance of an ORC system at part load operation. The objective was to understand its behavior from a thermodynamic perspective, identifying which elements were the most critical and which were the best operating points for each level of demanded power.

Zeotropic mixtures are some working fluids used on ORCs. Lecompte et al. [109] studied exergy analysis of zeotropic mixtures as working fluid in Organic Rankine cycles. The thermodynamic performance of non-superheated subcritical ORCs with zeotropic mixtures as working fluids were examined based on a second law analysis. The zeotropic mixtures under study were: R245fa-pentane, R245fa-R365mfc, isopentane-isohexane, isopentane-cyclohexane, isopentane-isohexane, isobutane-isopentane and pentane-hexane. The result showed an increase in second law efficiency in the range of 7.1% and 14.2% was obtained compared to pure working fluids. Geothermal cogeneration plant is one of the issue for ORC system investigation. One of the studies in the literature is about geothermal – energized cogeneration plants based on ORC. The paper presented and investigated several new hybrid integration of CHP plants driven by low-temperature geothermal water. The main reason was optimization of the heat source utilization in vapor of promoting the net output power of ORC as power plant operating in CHP models at variety of heating plant parameters. The simulations demonstrated that the power production has been considerably increased by the new CHP cycles, where optimization ratios could reach values till 130%, compared with conventional plants, at same heating plant conditions [110].

Another study about working fluids of ORC is studied in 2014. In the study, a thermodynamic model which mainly included Jacob number and the ratio of evaporation temperature and condensation temperature was proposed to forecast the thermal efficiency, output work and exergy efficiency of ORC system with zeotropic mixture. For different heat source inlet temperature, using different zeotropic mixture pairs, output work that was objective function was maximized by optimizing the evaporation temperature. The study showed that if the other working conditions are fixed, the heat source inlet temperature has a significant influence on the best composition of zeotropic mixtures at the optimal evaporation temperature [111]. Using ORC with renewable energy sources is another investigation topic. Minea studied [112] power generation with ORC machines using low-grade waste heat or renewable energy. The paper focused on the technical feasibility, efficiency and reliability of a heat-to-electricity conversion, laboratory beta-prototype, 50kW Organic Rankine Cycle (ORC) machine using industrial waste or renewable energy sources at temperatures varying between 85ºC and 116ºC. The thermodynamic cycle along with the selected working fluid, components and control strategy, as well as the main experimental results were presented.

Usitalo et al. [113] studied thermodynamic analysis of waste heat recovery from reciprocating engine power plants by ORCs. The study represented and discussed an idea of directly replacing the charge air cooler (CAC) of a large turbocharged engine with an ORC evaporator to utilize the charge air heat in additional power production. A thermodynamic analysis different ORCs was carried out with working fluids toluene, n-pentane, R245fa and cyclohexane. The effect of different ORC process parameters on the process performance were presented and analyzed to investigate the heat recovery potential from the exhaust gas and charge air. The result showed that power output of the selected engine which 16.6 MW gas-fired diesel engine can be increased by 11.4% by utilizing exhaust gas heat and 2.4% by utilizing the charge air heat. Long et al. [114] studied exergy analysis and working fluid selection of organic Rankine cycle for low grade waste heat recovery. The internal and external exergy efficiencies were adopted to analyze the impact of working fluids on the performance of the ORC, and a simplified internal exergy efficiency model was proposed to indicate this impact. The calculation results showed that the thermophysical properties of the working fluid have little impact on internal exergy efficiency, but they do play an important role in determining external exergy efficiency.

Liu et al. [115] studied sensitivity analysis of system parameters on the performance of the ORC system for binary – cycle geothermal power plants. The main purpose of the study was to analyze the sensitivity of system parameters to the performance of the ORC system quantitatively. A thermodynamic model of the ORC system for binary-cycle geothermal power plants has been developed and verified. The results showed that the geothermal temperature influences the range of the factors to the net power output, SP factor of radial inflow turbine, and the total heat transfer capacity, but it has no effect for the range of the factors for the thermal efficiency and the power decrease factor of the pump. Imran et al. [116] studied thermo-economic optimization of regenerative ORC for waste heat recovery application. The study dealt with the thermos-economic optimization of basic ORC and regenerative ORC for waste heat recovery applications under constant heat source condition. Thermal efficiency and specific investment cost of basic ORC, single stage regenerative and double stage regenerative ORC has been optimized by using Non-dominated Sorting Genetic Algorithm – II (NSGA-II).

Different approaches have applied to ORC systems. A multi-approach evaluation system (MA-ES) is one of the approaches. The study provided comprehensive evaluations on ORC used for waste heat utilization. The MA-ES covered three main aspects of typical ORC performance; basic evaluations of energy distribution and system efficiency based on the 1st law of thermodynamics; evaluations of exergy distribution and exergy efficiency based on the 2nd law of thermodynamics; economic evaluations based on calculations of equipment capacity, investment and cost recovery [117]. Another study about thermodynamics of ORC was applicability of entropy, entransy and exergy analyses to the optimization of the ORC. In the study, based on the theories of entropy, entransy and exergy, the concepts of entropy generation rate, revised entropy generation number, exergy destruction rate, entransy loss rate, entransy dissipation rate and entransy efficiency were applied to the optimization of the ORC. The optimization goal was to produce maximum output power. The results showed that when both the hot and cold stream conditions are fixed, all the entropy principle, the exergy theory, the entransy loss rate and the entransy efficiency are applicable to the optimization of the ORC, while entransy dissipation is not [118]. The one of the latest investigation about thermodynamic of ORC was published in 2014. In the work, it was concerned the performance enhancement of a two-stage serial organic Rankine cycle (TSORC) for geothermal power generation. Results showed that the system performance was coupled with geothermal water inlet temperature (GWIT), intermediate geothermal water temperature (IGWT), and evaporating temperatures. The two-stage evaporation significantly reduced the irreversible loss, thereby enhanced the net power output [119].

2.7 Conclusion

In this section, previous studies have been examined and a perspective has been formed for this study. When the open literature was examined, it was found that no similar study has been performed. In previous studies, combustion systems, ORC systems and the contribution of ORC systems to increasing energy efficiency in some large systems have been studied, but no study has attempted to establish an ORC system with the aim of gaining energy from the waste heat of the fluidized bed sludge incineration plant. This is the main motivation behind this study. Moreover, the most important difference of this study is that the analysis is performed by using actual SSIP data and the waste heat generated in the system is converted to useful energy with the help of a computer program simulated using ORC systems.

CHAPTER 3

SEWAGE SLUDGE POTENTIAL AS BIOFUEL IN EUROPE AND TURKEY

3.1 Introduction

In the last decades, researchers, governments and private sector companies understood that fossil fuel has a capacity in the world and energy demand of the world wouldn't be satisfied by only fossil fuels. So, the sustainability of energy and renewable energy became more popular and important. In this manner, sewage sludge has a very big potential for generating electricity as renewable and sustainable energy source.

With the increase of industry and population, a large amount of sewage sludge is formed due to the treatment of wastewaters. The resulting waste sludge is increasing day by day. There is no comprehensive study on the current situation regarding the management and final disposal in a manner of inventory of sludge in Turkey. However, it is known that most of the sewage sludge formed until recently has been disposed of by primitive storage method. Relevant Municipalities have difficulty in finding a place for storing and continuously increasing the waste sludge. At the same time, the excessive and widespread occurrence of the waste sludge is a threat to the environment and has become a necessity to be disposed of with appropriate methods. Various methods have been developed and started to be applied in sludge management for many years. The legal obligations and penal sanctions brought in recent years lead waste sludge producers to dispose of waste in a manner appropriate to the technique without damaging the environment. This means that the sludge has been disposed of by landfill method and for a while the greater part of this method occurs when removed reveals the results will be considered Turkey's current social and economic conditions.

3.2 Wastewater Treatment Plants and Sewage Sludge Statistics in Europe

Production of sludge starts with the treatment of wastewater. To determine sludge potential as an energy source, wastewater production should be known. In the world, there are many wastewater treatment plants but finding treatment statistics of all plant are extremely difficult. Although we have some difficulties about knowing the potential of sludge, we can make a consistent approach by using population.

European Environment Agency (EEA) release some report about wastewater treatment plant each year. According to the last report, the 5 biggest treatment plants are given below as a list [120].

Table 3.1 Five biggest wastewater treatment plant in Europe [120]

Name	**Country**	**Entering load (equivalent population)**	**Capacity (equivalent population)**	**Flow (m^3/s)**
PSYTTALIA	Greece	5.205.100	5.630.000	27
PARIS Seine Aval	France	4.283.333	7.500.000	24
LONDON Beckton STW	UK	3.380.000	3.578.977	18.20
ECOLO CHIEF	Poland	2.641.847	3.045.000	14.35
PARIS Seine Amont	France	2.628.883	3.600.000	14.22

As it is seen in the table, these plants satisfy wastewater treatment of 18.2 million people approximately. On the other side, there are some tables also about total sewage production, total sewage disposal and sewage disposal by incineration in Europe [121]. These tables help us to understand the potential of sludge as an energy resource.

Table 3.2 Sewage sludge production from urban wastewater [121]

Sewage Sludge Production from Urban Wastewater (in dry substance (d.s.))					
					***Thousand tonnes
Country / Year	2011	2012	2013	2014	2015
Belgium	-	157.2	-	-	-
Bulgaria	51.4	59.3	60.3	54.9	57.4
Czechia	217.9	263.3	260.1	238.59	210.24
Germany	1946.3	1848.9	1808.7	1837.1	1820.7

Ireland	85.7	72.4	64.6	53.5	58.4
Greece	147	118.7	113.1	116.1	-
Spain	-	1082.7	-	-	-
France	-	987.2	886.5	961.6	-
Croatia	31	42.1 (p)	32.1 (p)	16.4 (b)	18
Cyprus	6.7	6.6	6.1	6.2	6.7
Latvia	19.7	20.1	22.8	-	-
Lithuania	51.9	45.1	41.4	40.7	42.9
Luxembourg	-	7.7	-	-	9.2
Hungary	168.4	158.9	170.3	166.6	156.9
Malta	6.1	10.5	9.7	8.6	8.4
Netherlands	350.8	346.4	339.1	344.2	-
Austria	-	266.3	-	239	-
Poland	519.2	533.3	540,3	556	568
Portugal	-	338.8	-	-	-
Romania	114.1	85.4	172.8	192.4	210.5
Slovenia	26.8	26.1	27.2	28.3	29.1
Slovakia	58.7	58.7	57.4	56.9	56.3
Finland	140.9	141.2	-	-	-
Sweden	200.1	207.5	207.9	200.5	197.5
United Kingdom	-	1136.7	-	-	-
Switzerland	-	-	194.5	-	-

- = not available p=provisional b=break in a time series e=estimated

Table 3.3 Sludge production disposal and incineration of some countries between 2011 and 2015

Sludge Production Disposal and Incineration of Some Countries Between 2011-2015				
				*** Thousand Tonnes
	Average Sludge Production	Average Sludge Disposal	Average Sludge Incineration	Incineration Percentage
Belgium	157.2	107.3	88.8	82.76%
Bulgaria	56.66	36.58	0	0.00%
Czechia	238.026	238.026	8.19	3.44%
Germany	1852.297	1836.981	1067.528	58.11%
Ireland	66.92	66.92	0	0,00%
Greece	123.692	125.70875	38.971	31.00%
Spain	1082.69	1082.69	39.71	3.67%
France	945.073	913.039	179.437	19.65%
Croatia	27.888	16.724	0	0.00%
Cyprus	6.465	6.465	0	0.00%
Latvia	20.866	19.9	0	0.00%
Lithuania	44.386	27.451	0	0.00%
Luxembourg	8.428	6.928	0.729	10.52%
Hungary	164.162	127.04	20.216	15.91%
Malta	8.628	8.628	0	0.00%
Netherlands	345.125	322.8	321.85	99.71%
Austria	252.672	252.672	128.532	50.87%
Poland	543.36	543.34	66.92	12.32%
Portugal	338.8	113.1	0.1	0.09%
Romania	155.016	125.268	0.535	0.43%
Slovenia	27.5	27.24	14.5	53.23%
Slovakia	57.596	57.596	8.232	14.29%
Finland	141.05	141.05	16.5	11.70%
Sweden	202.7	189.9	1.85	0.97%
United Kingdom	1136.7	1078.4	228.9	21.23%

Switzerland	194.5	194.5	188.3	96.81%

Between the European countries, there is not enough information for Turkey, Norway, Denmark, Bosnia Herzegovina, Italy, and Estonia. Because of that, results can't be shown for these countries. As seen, Czechia, Ireland, Greece, Spain France, Cyprus, Latvia, Malta, Netherland, Austria, Poland, Slovenia, Slovakia, Finland, Sweden, United Kingdom, and Switzerland have a high percentage for disposal of produced sludge. But although these high disposal percentages, the disposal method of sludge varies between these countries. For example, the Netherlands use incineration method for disposing of all the sludge but the percentage of incineration in disposing methods is zero for Malta. It shows that countries select different way in disposing of sludge [121].

Between 2011 and 2015 given countries at table 3.1 produced approximately 8,200,000 tons sludge on a dry basis. Also, these countries disposed of 2,420,000 tons sludge by incineration. It means that incineration percentage of sludge disposing method is approximately 31.56%. On the other side, the total population of selected countries was approximately 449 million in 2015. Today, the world population is 7.672 billion approximately. If we take selected countries as a reference to make a consistent approach about world sludge production, it means that 140,000,000 tons of sludge can be produced in the world. If we take incineration percentage as a reference also, 44,184,000 tons sludge is incinerated. If the waste heat of incinerated sludge is used, there will be huge energy recovering.

3.3 Turkey's Wastewater and Sludge Statistics

In Turkey, there is very limited information about wastewater treatment plants and there is approximately no information about sewage sludge. Also, there can be made a consistent approach about sewage sludge by using wastewater amount in Turkey. Table 3.4 shows that Turkey's general statistics about wastewater between 2004 and 2016 [122].

Table 3.4 Turkey's wastewater statistics between 2004 and 2016

	2004	2006	2008	2010	2012	2014	2016
Number of Municipalities	3.225	3.225	3.225	2.950	2.950	1.396	1.397
Municipalities with WWTP	319	362	442	438	536	513	581
WWTP Number	172	184	236	326	460	604	881
Amount of Treated Wastewater in WWTPs (Million m^3/Year)	1.901	2.140	2.252	2.720	3.257	3.484	3.842

From 2004 to 2016 WWTP number and amount of treated wastewater in WWTP has increased continuously.

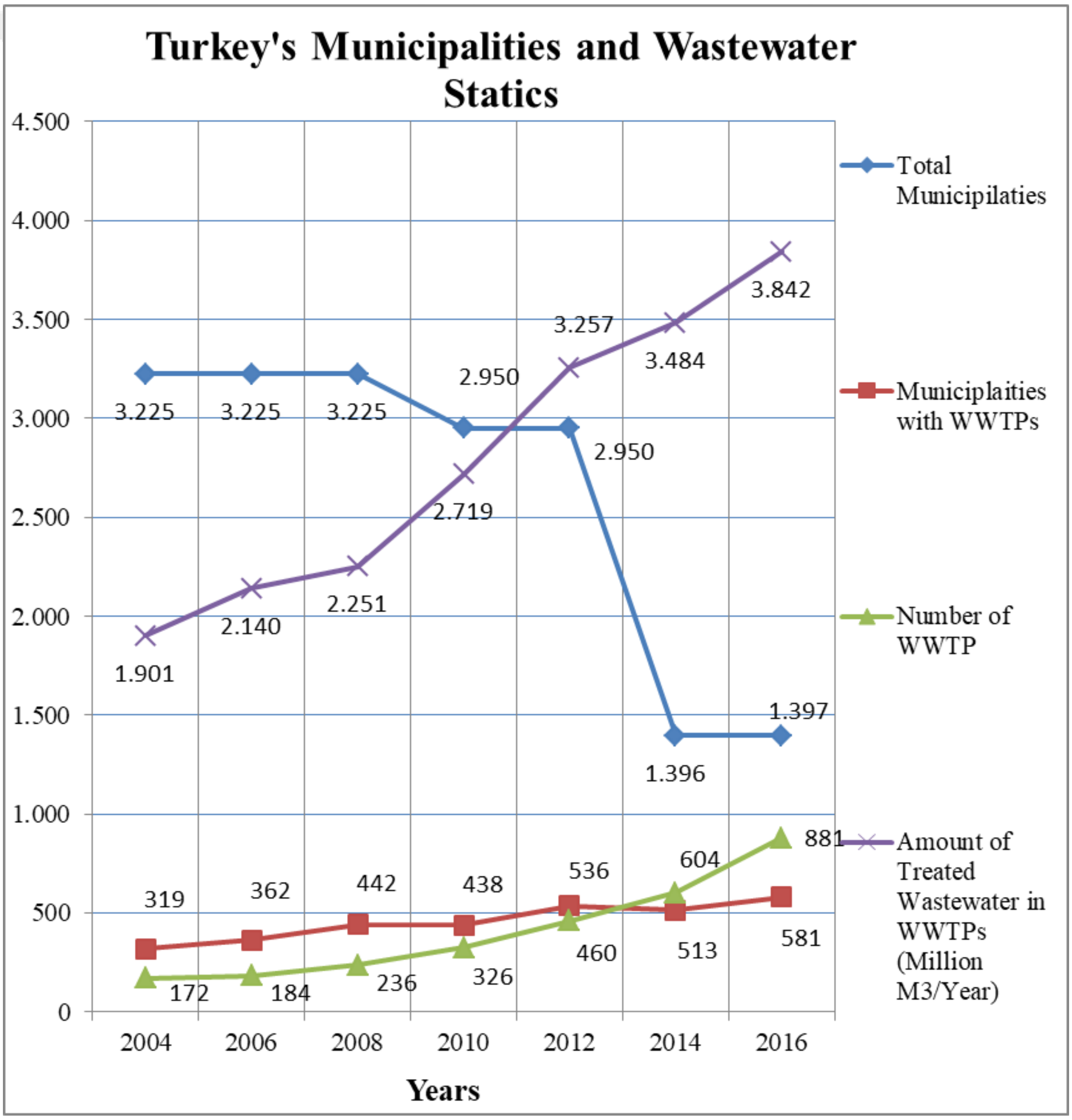


Figure 3.1 Turkey's municipalities and wastewater statistics between 2004 and 2016

It is shown that in the Fig. 3.1, there is a huge amount of an increase in WWTPs number and amount of treated wastewater in WWTPs. Because there is not enough information and available inventory about Turkey's sewage sludge production, European countries statistics can be used as a reference. The fraction of incineration in total sludge disposing method is 31.56% among selected European countries as mentioned below. In many WWTP in Turkey, treated wastewater has 27% of dry matter. So, according to data on Fig. 3.1; 1037.34 thousand tons sludge on a dry basis was produced. Also, if European incineration ratio statistics are accepted as a reference, 327.38 thousand tons sludge was eliminated by incineration.

3.4 Conclusion

In this chapter, the statistics about Europe’s and Turkey’s sewage sludge potential are presented. It is seen that Europe countries act very seriously about sludge dispose. All regulations are strict in this manner. Although there are strict regulations in the region, some of the European countries can’t make progress in this manner. These countries use landfilling entirely. In Turkey, there is not enough information about sewage sludge but researching wastewater treatment plants located in Turkey gives a good approach. About other countries located in Africa, America and Asia, there is no convincing information for sewage sludge potential.

CHAPTER 4

GASKI SEWAGE SLUDGE INCINERATION PLANT

4.1 Introduction

Incineration plants are located integrated with treatment plants because of economic aspects. Each incineration plant is designed according to the capacity of treatment plant. It is also a difficult process for engineers because of amount of wastewater fluctuates depending on weather conditions. Also population increasing should be considered to incineration plant will be enough for treatment plant. In this chapter an overview of incineration plants is provided firstly and then detail of GASKI Incineration plant is presented.

4.2 Sewage Sludge Incineration Plant Overview

Nowadays, there are many waste water treatment plant in the world but by product of this treatment plants is the most important problem for the plant. Therefore, many disposal techniques are developed to overcome this problem. Some of them use high technology and some of them are simple. The Incineration method is one of the disposal methods of sludge. Although there are many types of incineration plants that are running different locations in the world, basically the principle is the same as boiler. The simplest type of incineration plant has one boiler, one sludge tank, a sludge carrying system, ash tanks, and chimney. Figure 4.1 shows a basic schematic diagram of a simple sewage sludge incineration plant.

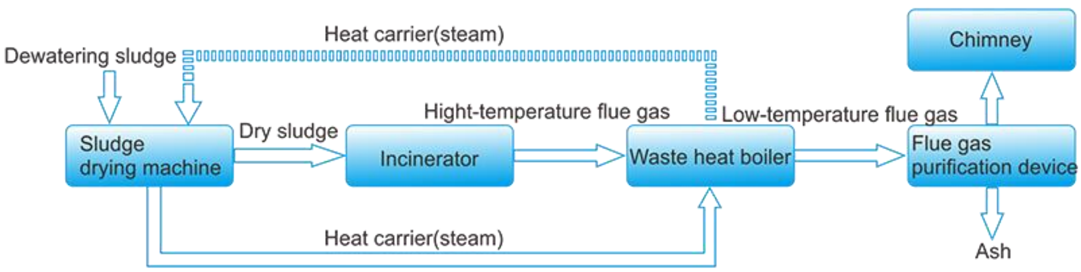


Figure 4.1 Basic schematic of a simple sewage sludge incineration plant

Of course, there are more complex incineration plants that are designed to fulfill requirements, especially in manners of energy recovery. Increasing energy recovery demand means increasing of investment cost of the incineration plant.

4.3 GASKI Sewage Sludge Incineration Plant

In Gaziantep there is only one sewage sludge incineration plant which serves to GASKI Central Wastewater Treatment Plant, Kızılhisar 2nd Wastewater Treatment Plant and other wastewater treatment plant that is located in the same region. The plant is established on 2635 m^2 area which 887.07 m^2 is closed. Incineration capacity of this plant is about 300 tons/day. Coordinates of plant is 37°01’56.8” N and 37°25’54.0 E and satellite view of plant are given below (Fig. 4.2).

Figure 4.2 Satellite view of the GASKI incineration plant

Table 4.1 shows sewage sludge production of the central municipal wastewater treatment plant and the Kızılhisar wastewater treatment plant.

Table 4.1 Sewage sludge production of the central WWTP and Kızılhisar WWTP

Months	Central WWTP	Kızılhisar
January 2018	3.719.200	174.650
February 2018	3.997.580	214.860
March 2018	3.848.260	196.800
April 2018	4.998.940	226.180
May 2018	7.964.130	256.000
June 2018	4.515.620	168.200
July 2018	4.058.280	150.720
August 2018	3.934.960	119.960
September 2018	3.752.200	114.200
October 2018	4.486.235	161.000
November 2018	5.202.040	274.000
December 2018	6.261.750	208.000
January 2019	4.106.080	276.320
February 2019	4.590.240	121.460
March 2019	5.038.360	387.180
April 2019	5.174.500	340.160
Total	75.648.375	3.389.690
Monthly Average	4.728.023	211.856
Daily Average (kg/day)	157.601	7.062

4.3.1 SSIP Description

Central Wastewater Treatment Plant was established in 2010 for the purpose of eliminating the wastewater sludge from the Kızılhisar 2nd Wastewater Treatment Plant and about 300 tons / day from the treatment units in the region and eliminating the harmful effects to the environment.

The treatment sludge contains 73% water when taken into the system. In order for the sludge to be burned, it is necessary to remove the significant amount of water it carries within the structure. For this reason, the sludge is dewatered before being burned. The sludge collected from the treatment system is poured into the sludge conveyor belts. The conveyor belts transfer the waste to the sludge collection pool. The collected sludge is fed into the thermal drying system. The rate of dry sludge, which is 27% in the thermal drying system, is increased to 40% dry matter. Then the sludge is sent to the combustion unit. In the combustion unit, the sludge is burned so that only 1% ash remains. In the treatment plant, decanter yields digested sludge containing 27% solids. In order to eliminate this sludge, the plant was established with an integrated fluidized bed continuous incineration system with a thermal drying system. Process flow diagram is given below (Fig. 4.3). A detailed description of the process flow diagram is given below. The waste sludge taken into operation goes through the following process steps.

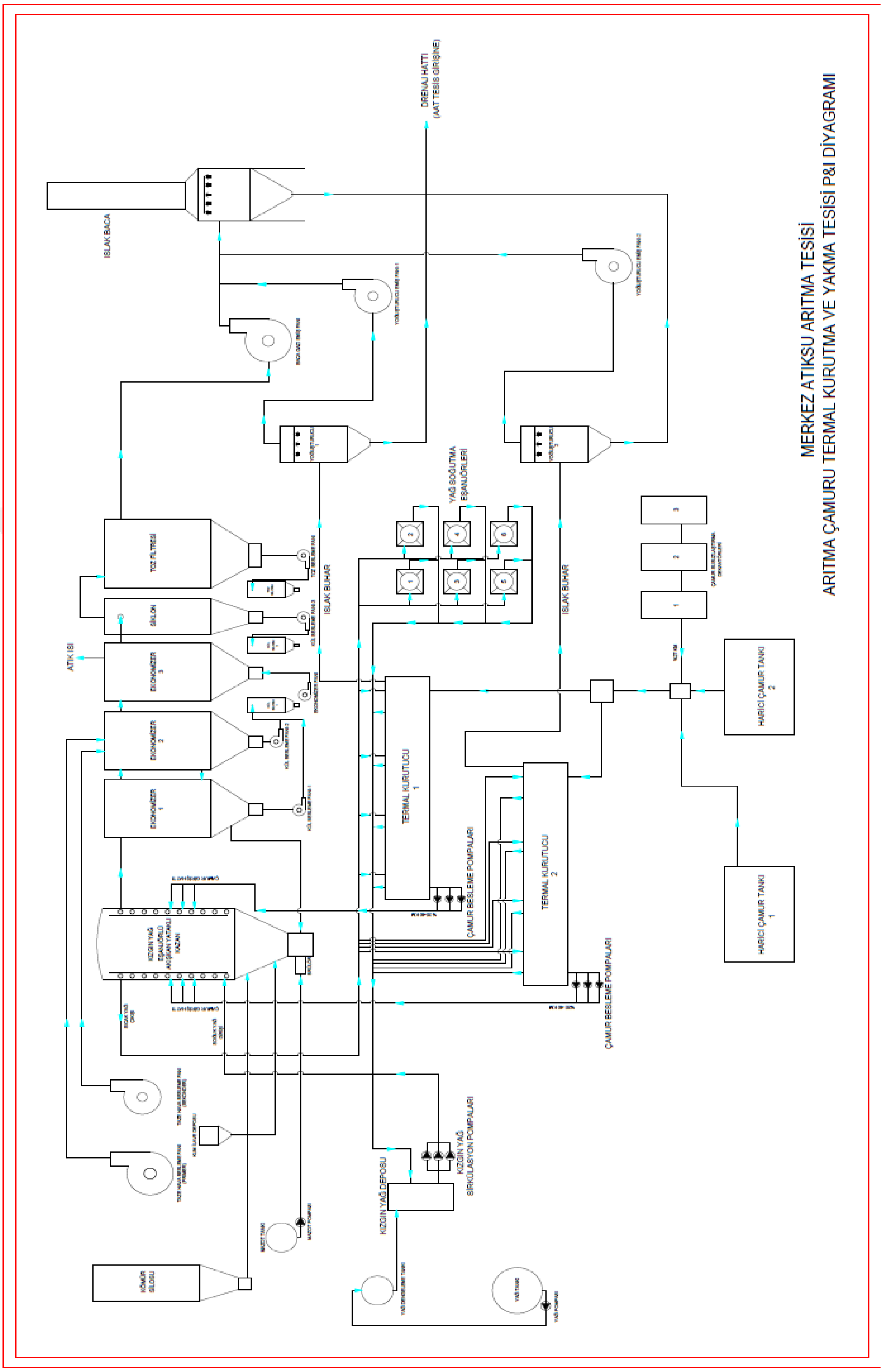


Figure 4.3 Schematic diagrams of the incineration plant and thermal drying system

4.3.1.1 Sludge Tank

The sludge tank is a unit where the sludge from the existing system and the sludge brought to the plant to be disposed of from the outside are stored and can be kept before feeding to the system and has a daily storage capacity. The sludge is sent to the warehouse by conveyor. The feeding of the sludge from the tank to the thermal dryer is also done with the help of the spiral screw. The sludge level in the tank is measured with the help of a laser sensor. The temperature of the sludge can also be measured with the help of the sensor. The sludge intake is made by the sliding mechanism in the tank, which is driven by hydraulic pistons. The control of this mechanism is carried out on the computer with the help of a sensor.

4.3.1.2 Sludge Transfer and Feeding System

Pump and transfer systems are used to transport the accumulated sludge in the tank. The screw transfer units are made of corrosion-resistant materials against harmful and oxidizing molecules in the sludge. The feeding system is specially designed to achieve optimum operating performance. It has an inverter control system and its carrying capacity can be adjusted. Control will be done completely via a computer system.

4.3.1.3 Thermal Drying System

The sludge is sent to the thermal dryer with the transport system (Figure 4.4). The dryer is made of double-walled galvanized steel and the heat of the hot oil passing through the hot oil line pipes placed between the two walls is used to dry the waste sludge. The thermal dryer was made using special materials and was made to prevent heat leakage. Fireproof rock wool industrial boards are used as insulation materials. In order to optimize the process and to make the process more efficient, the parameters such as the rotational speed, oil temperature, sludge feed rate of the thermal dryer drum are continuously monitored by the computer and can be processed immediately. The sludge in the heated thermal dryer with hot oil is dried to a rate of 40% dry matter and sent to the fluidized bed combustion unit. Figure 4.5 shows the thermal drying unit.

Figure 4.4 Transport system of the thermal drying unit

Figure 4.5 Thermal drying unit

4.3.1.4 Hot Oil System

The other system working with the drying system is the hot oil system. This system will take its energy from the heat exchanger located at the combustion furnace outlet. In addition, heat exchangers can be added in electricity production. These heat transfer units, connected in series, consist of hot oil circulating pumps, control valves, heat jackets, conveying pipes and insulation systems. The hot oil circuit is positioned around a central tank. The energy from the diesel burner and the combustion unit is sent to this tank and the heat energy is sent to the required systems. The most important of these systems is the drying system as mentioned previously. The oil heated in the heat exchanger dries the water in the waste sludge by giving energy to the waste sludge. In order to remove the excess energy in the relatively cooled oil passing through the drying system, the oil is sent back to the heat exchanger and cooled further. The oil from the heat exchanger goes to the oil tank and the oil cycle is completed. Figure 4.6 shows the oil heat exchangers.

Figure 4.6 Oil heat exchangers

4.3.1.5 Burner

A fuel support is required for the first incineration of the waste sludge. In this plant, although coal was used as support fuel before, the system was renewed and nowadays, diesel fuel is used as support fuel. In order for the fluid bed furnace to be heated to the temperature at which the sludge from the dryer may burn, the fuel supplied from the reserve tank is sent to the boiler. In addition, this fuel is sent from the reservoir tank to the boiler when the internal temperature of the boiler is reduced, helping to keep the temperature of the boiler constant. This fuel is stored in a reserve tank. The fuel stored in the reserve tank is pumped through the fuel pump into the boiler as a result of the communication of the sensors inside the boiler connected to the computer system. The pumping process and the amount of fuel to be pumped are determined and controlled by the automation system.

4.3.1.6 Condenser

The gas released during the drying of the dried sludge in the thermal dryer is sent to the condenser. Condensation is carried out in the opposite direction with water flow. The amount of condensed water is constant and constant in direct proportion to the evaporated water. This ratio is determined according to the operating parameters of the thermal drying. Condensation water and condensate steam are collected at the bottom of the tower with water pipes. The exhaust gas will be taken from the head of the tower and sent to the combustion unit. In this way, the release of bad odor into the atmosphere is prevented.

4.3.1.7 Gas Heating System

The gas heating system ensures that the air fed to the thermal dryer is heated to the desired level, and works by using a parallel heat exchanger. The oil, which emerges at 200 degrees from the thermal dryer, is sent to the drier at 240 degrees from the hot oil tank. The super-heated oil entering the burning unit which reaches 850 degrees is taken to the oil tank at 260 degrees and sent back to the combustion unit at 210 degrees. In this way, the excess heat energy in the combustion unit is sent to the hot oil tank to provide the heat energy required to dry the sludge in the thermal dryer.

4.3.1.8 Pressurized Air System

For drying system, pressurized air system is settled down. The pressurized air is used in pistons in thermal dryer to do some process such as pushing sludge. The system contains compressor, air dryer, filters and air tank.

4.3.1.9 Electric and Electronic System

The power required by the fans, screw carriers and pumps in the system is provided by using electric motors. The motors are all 3-phase asynchronous motors and they are controlled by changing the frequency applied to the motor with the speed control devices. The energy inputs of the speed controllers are made via the input choke. This not only reduces the impact of the device on sudden fluctuations in the network, but also provides damping of the 3rd and 5th harmonics produced by the electronic structure of the device and prevents other devices connected to the network from being affected. There are various analog and digital sensors on the system. These sensors are connected to the appropriate input and output modules in the panel and distribution boxes placed in the system and which are part of the control system. In addition, these equipment are used to control the motors in the combustion system, the speed control devices of the motors, temperature measurement sensors, flap control devices, air velocity measurement sensors and gas analyzers. In order to achieve the most efficient combustion in the furnace, the supply air temperature, quantity, fuel supply quantity is precisely controlled. In addition to the efficient combustion, the analysis of the chemicals in the flue gas is ensured by measuring the amount of gases that are harmful to the environment and ensuring that the additives are kept in the combustion chamber by dosing the additives. Figure 4.7 shows the control room of the plant.

Figure 4.7 Electric room of the incineration plant

4.3.1.10 Computer and Automation System

Computer and automation system, I/O (input/output) modules that read signals from sensors in the field, central processing units (CPUs) that determine the signals to be sent to the applicators in the field to process these signals and the screens that operate as user interface with these CPUs is an integrated system. The signals from the sensors are disturbed when they are moved analogously. I/O modules are located in IP 65 enclosures near the sensors to avoid these distortions. In these modules, the digitized information is transmitted to the CPU by cabling according to the communication protocol. After this transmission, the CPU, which was previously loaded into the CPU, decides how the system behaves and sends signals back to where it is needed. These signals to the field are sent from digital format to analog format. In order to ensure efficient combustion, fuel, air and heat control is provided to ensure the optimum mixture. The information from the field is processed and the desired values are reached. Data from the system is stored. In case there is a condition other than the conditions that should occur during normal operation in the system, the user is warned by sound, written and illuminated. Fluidized bed combustion system and thermal dryer systems are controlled by common automation system. The central computer system monitors all process parameters. Figure 4.8a and figure 4.8b show the automation control room of the plant.

Figure 4.8a Automation and control room of the incineration plant

Figure 4.8b Automation screens of the incineration and thermal drying system

4.3.1.11 Incineration Unit

In the drying unit, approximately 60% of the water in the sludge is removed. Thus, the sludge contains 40% dry matter. The dried sludge is supplied to the combustion system. The combustion system contains the following units. Fluidized bed combustion system is made of bricks and steel construction with vertical cylindrical geometry and consists of four main parts:

1- Air supply section

2- Air distribution plate

3- Fluid bed material consisting of prime material, generally sand and product ash

4- Overhead combustion chamber, reactor

The fluidized bed combustion unit is operated at a temperature of 850ºC. It works with 25% - 50% excess air supply and gas sweep time is 5 - 8 seconds and this period ensures efficient decomposition of organic materials in waste sludge. Thus, organic content is below 1% in ash. Figure 4.9 shows the incineration unit with economizer.

Figure 4.9 Incineration unit with economizer

Combustion chamber is cylindrical and inside is made of steel sheet structure which is woven with flame wall and it is isolated with 20 cm stone wool. The technical drawing of the combustor is given at figure 4.10. The combustor has 2.8 m diameter in bottom and 4 m diameter at the top. The total height of combustor is 6.55 m. There are air nozzles (tuyeres) located between the bottom of the combustion chamber and the wall. The temperature in the fluidized bed can be fixed at 850 degrees or at the desired value. In this section, the squeeze time varies between 2 and 5 seconds. General dimensions of the fluidized bed combustion unit are presented in the following drawing. A refractory brick was built into this furnace. Approximately 500 kg of fuel must be burned to heat the oven to the desired temperature. As the sludge burns, small ash particles are transported to the upper areas of the furnace, and a small amount of sand is carried with the ash. Therefore, sand is added at a rate of 5% in every 300 hours of operation.

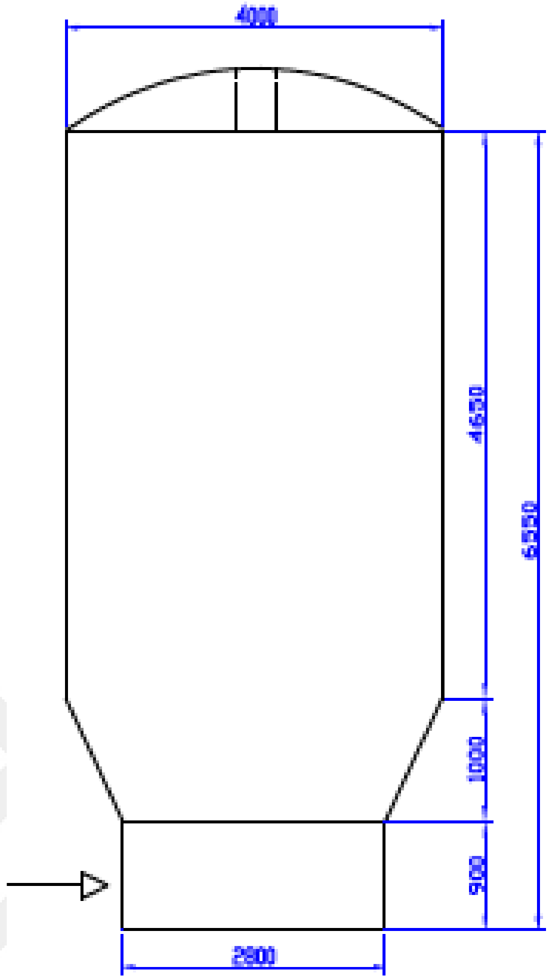


Figure 4.10 Technical drawing of the fluidized bed combustor

Burning of the sludge in the combustion unit occurs in two parts. In the first section, the evaporation of the water and the pyrolysis of organic materials are provided by increasing the temperature of the waste sludge directly in the fluidized bed. In the second part, the remaining free carbon and combustible gases are burned in the freeboard area. A homogeneous mixture is formed in the fluidized bed and 20-50% air is sufficient for complete combustion. In order to obtain low SO_x emissions, limestone is fed and graded air is supplied to obtain low NO_x emissions. The sand bed is 0.8-1 m deep for the fluid bed. In case of fluidization, volume increase is between 80% and 100%. The bulk density of the sand is 1600 kg/m^3. Feeding the sludge to the bed is made from the 1.2 m height into the sand bed. The smoke rate in the combustion chamber is 0.8 - 0.9 m/s and it is designed to have a heat load of 350000-500000 kcal/hour/m^3. Air is supplied in the air supply section at a pressure of 0.2-0.35 atm. Combustion unit has been designed that combustion gas occurs as a result of combustion enters to economizer at 480ºC and re-fed to the combustion chamber at 280ºC. Thus, the amount of heat thrown into the atmosphere is significantly reduced and heat savings are planned.

4.3.1.12 Heat Transferring System and Economizer

At the exit of the furnace, heat is taken from the exhaust gases by means of hot oil exchanger and this heat is sent to the hot oil tank. In addition, some of the heat energy is used to heat the air supplied to the furnace inlet.

When the high temperature waste gas generated by combustion is given to the atmosphere, there is considerable energy loss. Economizer is used in the system to bring the energy of this waste gas back into the system and to increase the energy efficiency. The amount of heat thrown into the atmosphere is reduced significantly and heat is saved. Economizer consists of parallel tube bundles arranged horizontally. The tubes are connected to the inlet and outlet lines. The air flow and gas flow are reversed. In order to maintain the temperature of the oven and to ensure good combustion, the inlet air temperature of the economizer will be increased and supplied to the combustion furnace as supply air. The increase in economizer gas inlet and outlet temperatures indicates a high efficiency of the boiler. However, the economizer gas outlet temperature is not reduced to less than 160 Celsius degrees, which is the dew point of sulfuric acid.

4.3.1.13 Cyclone

Since the exhaust gases and ash particles coming out of the fluidized bed furnace cannot be supplied directly to the atmosphere, so the ash and sand particles in the exhaust gas must be separated. The cyclone is used for this separation process. The cyclone built in the plant is made of cylindrical double-walled galvanized sheet and the insulation material is placed to prevent heat loss between the walls. Due to the specific mass differences, the solids fall down and the gas components are discharged from the top. The ash and the flying sand particles are removed from the bottom outlet of the cyclone and sent to the ash tank. Figure 4.11 shows the technical drawing of cyclone and figure 4.12 shows the cyclone located in the plant.

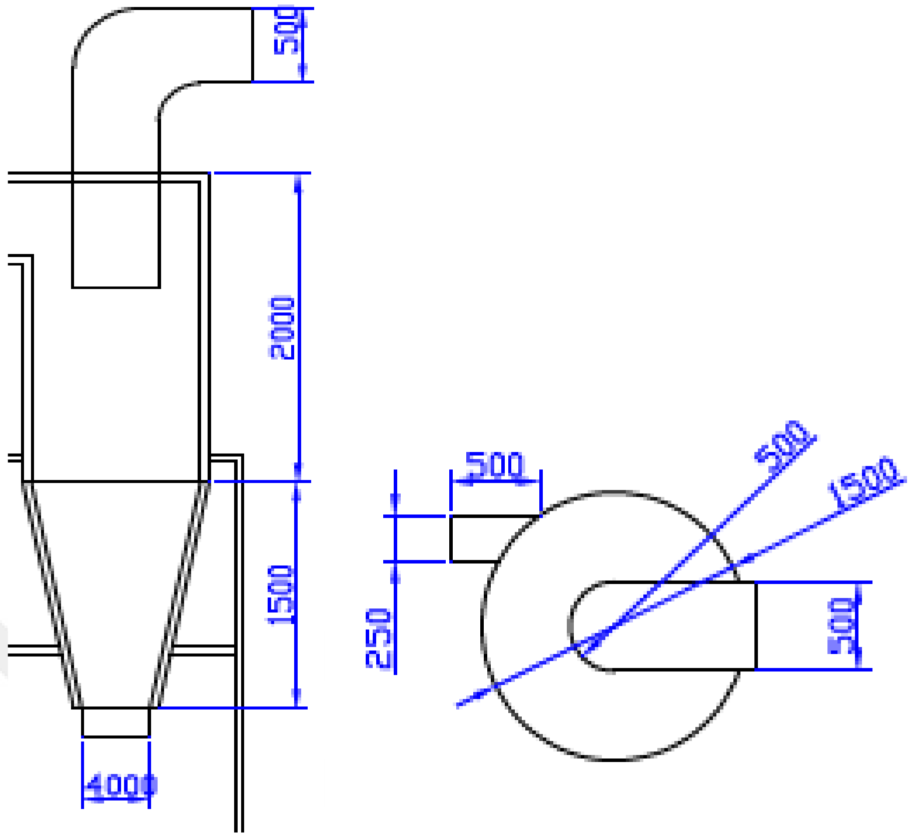


Figure 4.11 Technical drawing of the cyclone

Figure 4.12 Cyclone

4.3.1.14 Dust Filter

Although the process gas from the cyclone is separated from the ash and sand particles, it contains particles in which the cyclone fails to decompose. Therefore, the exhaust gas enters the dust filter after it is released from the cyclone. The dust filter prevents the release of dust and chemical particles which are smaller than the ash particles in the waste gas to the atmosphere. This unit has cleaning systems to create high performance. The removal of harmful gases in the exhaust gases will also take place in this region. Dust filter is a system that catches dust by compressing compressed air. The exhaust gas entering into the filter is exposed to high pressure air and leaves harmful gases and particles on the filter. Exhaust gases free of particles and toxic gases are sent to the chimney with the help of a fan to release into the atmosphere. Dry and storable ash obtained as a result of incineration is discharged from the system with the help of fans after decomposition in the economizer, cyclone and dust filter and stored in ash tanks.

4.3.1.15 Wet Chimney

The exhaust gas, free of particles and toxic molecules, is ready to be released into the atmosphere by passing through the chimney. When exhausting the exhaust gas from the chimney, various measurements are made to control the combustion efficiency and according to these measurements, combustion is regulated by computer control. In order to prevent the formation of liquid in the chimney, a suitable device has been established and its gas content and values are continuously measured by the flue gas measurement system established on the line. The monitoring system prevents the emission of harmful exhaust gas to the environment during the day and night and immediately intervenes when any undesirable value is observed. The chimney is made of 3 mm thick Ni-Cr alloy sheet and has a rain hat at the top of the chimney. There is also a cleaning lid at the bottom of the chimney to clean the ash that accumulates in the chimney. The chimney is designed according to the Regulation on Control of Industrial Air Pollution. A wet chimney is used to reduce emission values. In the flue system, water is sprayed in the opposite direction of the gas flow and the values of harmful gases in the waste gas are kept within the legal limits. Both particles and sulfur molecules in the gas discharged from the wet chimney to the atmosphere are retained. The polluted water that accumulates in the bottom of the chimney is collected in the collection witness here and given to the treatment system. Calcined water circulation is carried out to neutralize the acids in the flue gas. In order to prevent the formation of droplets at the chimney outlet and the appearance of the white vapor layer, the temperature of the clean gas exiting the neutralization column should be kept above 120 degrees. Figure 4.13 shows the wet chimney [123].

Figure 4.13 Wet chimney

4.3.1.16 Ventilation Ducts

There are channel systems in the facility constructed according to the flow requirements of air and exhaust systems. These ventilation ducts are manufactured using stainless materials and are insulated. Thanks to these isolations, the humidity and temperature values of the air in the channel are preserved and the perspiration on the surface of the channel is prevented.

4.3.2 Current Situation of the Plant

Actually, in the schematic layout, it is seen that there are two blowers and two thermal dryer units. But because of some problems, the second fresh air blower and second thermal dryer line are not used. Because of these problems, the incineration plant capacity decrease from 150 tons/day to 76 tons/day. Approximately 76 tons of waste sludge is disposed on a daily basis in the incineration plant. The plant consists of two systems; drying and incineration.

The sludge containing 27% dry matter is sent from the wastewater treatment plant to the sludge tank in the incineration plant and stored there. Then the sludge stored in the tank is sent to the drying units through the sludge pumps. In these units, 27% dry matter rate is increased to 40%. The sludge containing 40% dry matter is again sent to the fluidized bed incinerator with hot oil exchanger by means of sludge pumps. The exhaust gas emitted from the sludge burned in this incinerator is passed through 3 economizers, 1 cyclone and 1 dust filter and sent to the wet chimney and then released to the atmosphere.

The fluidized bed incinerator is 9.5 m high and 2.2 m in diameter. The fluidized bed furnace body is double-walled and thermal insulation is provided by stone wool between the walls with a thickness of 20 cm. Oil pipes used for the drying system are installed on the inner walls of the fluidized bed furnace body. The heat transfer oil circulating inside these pipes takes part of the heat energy generated during incineration and then this heat energy is used in the sludge drying process.

The oil inlet temperature to fluidized bed incinerator and outlet temperature from the incinerator is 140°C and 150°C respectively.

In this whole process, a drying system has been installed to dry the sludge although the main process is incineration. In the drying system, the condenser performs an important task. The water vapor and exhaust gas mixture sent from the drying unit to the condenser are subjected to the separation process. The exhaust gas is separated from the water vapor and is sent to the wet chimney by a fan. Water vapor is sent to the entrance of the wastewater treatment plant for reuse. Table 4.2 shows all components used in the incineration plant.

Table 4.2 Components of the incineration plant

Line	Component Name	Quantity	Unit Power	Flow Rate
1	Fresh Air Supply Fan	1	75	30000
2	Fluidized Bed Combustor with Hot	1	22700	
3	Economizer	3		
4	Cyclone	1		
5	Dust Filter	1		
6	Ash Supply Fan	3	7.5	16000
7	Economizer Fan	1	7.5	16000
8	Dust Supply Fan	1	7.5	16000
9	Hot Oil Circulation Pump **2	3	75	22000
10	Mixer Pump (Drying Process)	1	3	
11	Thermal Dryer	1		
12	Dryer Conveyor Motor	1	90	
13	Sludge Feeding Pump	3	11	
14	Oil Heat Exchanger	6	7.5	
15	Condenser	1		
16	Condenser Suction Fan	1	11	
17	Flue Gas Suction Fan	1	90	50000
18	Wet Chimney	1		

*1. Incinerator capacity is 22700 kW_{th}. **2. The flow rate is 22000 kg/h.

4.3.3 Sewage Sludge Capacity of the Incineration Plant

In the plant, 300 tons of 27% dry matter containing sewage sludge is dried and incinerated. In the drying process, the dry matter ratio of the treatment sludge is removed from 27% to 40% and then incinerated. The capacity calculation of the incineration process is based on the burning (organic) part of the treatment sludge. The following calculations are examined on some 100% dry matters.

Sewage Sludge =

300 tons / day x 27% dry matter = 81 tons/day (100% dry matter)

= 81 tons/day x 1 day/24 hours = 3.57 tons/hour

$= 81$ tons/day x 300 days/1 year = 24300 tons/year

$= 24300$ tons/year x 1 year/12 months = 2025 tons/month

Produced Ash = 300 tons/day x 12% = 36 tons/day

$= 36$ tons/day x 1day/24 hours = 1.5 tons/hour

$= 36$ tons/day x 300 days/year = 10800 tons/year

10800 tons/year x 1 year/ 12 months = 900 tons/month

Plant area is 2635 m^2 that is 887.08 m^2 is covered area which belongs to Gaziantep Municipality. In the plant, 8 people are working that one of them is engineer, one of them is technician and six of them are workers. Also in the plant 100kg fuel is used per day. It means that approximately 30 tons fuel is used annually.

Rated thermal power of plant coming from sludge is calculated as follows;

Sewage Sludge Amount = 81000 kg/day = 3375 kg/h

LHV of Sludge = 3142 kJ/ kg = 751.67 kcal/kg

1 kW = 860.4 kcal / kg

$Q_{(sludge)} = 3375$ kg/h x 751.67 kcal/kg x 1 kW/ 860.4 kcal/h

$Q_{(sludge)} = 2949.87$ kW ≈ 2.95 MW

4.3.4 The Total Initial Cost of SSIP

The plant has two type costs. One of them is investment cost and the second one is operating and maintenance cost. The investment cost of the plant is approximately 7.6 million €. The data has been provided from Gaziantep Municipality. But when it comes operating and maintenance costs, there is no clear information in the facility.

4.4 Conclusion

In this chapter, detailed description and current working situation with of GASKI Incineration Plant has been presented with technical details and cost information. These values are taken from incineration plant's management and Gaziantep Municipality and also from some official reports. The data of the incineration plant and thermodynamic analyses result will be given at Chapter 5.

CHAPTER 5

RESULTS AND DISCUSSION

5.1 Introduction

In this chapter, GASKI SSIP is analyzed by thermodynamically and then one scenario with changing parameters and seven different working fluids of ORC is developed. The performances of the models are evaluated in terms of changes in the pressure and temperature. The results are compared and assessed for the betterment of SSIP.

5.2 Thermodynamic Analysis of the GASKI SSIP

To perform a thermodynamic analysis of the incineration plant, a computer program was used that is called Cycle-Tempo. The program has an educational version, and for this analysis educational version has been used. All values used in the program have been taken from GASKI and the values are actual operating values for the plant. In the thermodynamic analysis, only the incineration process has been analyzed in detail. Only inlet and outlet temperatures and pressures of heat transfer oil were used because of affect effectiveness of the heat exchanger in the system. The fluidized bed combustor has been modeled gasifier. In the schemes, two types exergetic efficiency are shown. One of is the universal exergetic efficiency, and one of is the functional exergetic efficiency. The difference between universal exergetic efficiency and functional exergetic efficiency is explained in detail below.

5.2.1 Explanation of Efficiency Definitions

The difference between universal efficiency and functional efficiency can be illustrated with a simple example: the process of heat transfer in a heat exchanger.

As a given, the purpose of the heat exchanger is to heat a process flow – called primary flow – by withdrawing heat from another process flow – called secondary flow. The exergy of the primary flow will increase as a result of the absorbed heat, and the exergy of the secondary flow will decrease. Figure 5.1 visualizes the exergy change in the process flows.

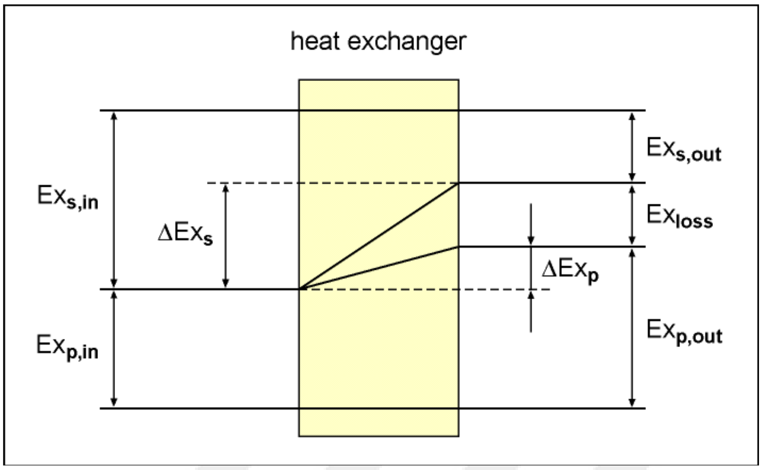


Figure 5.1 Change in exergy quantities at heat transfer

The universal exergy efficiency for this heat exchanger follows from comparison below formula:

$$\eta_{Ex,u} = \frac{\Sigma Ex_{out}}{\Sigma Ex_{in}} \tag{5.1}$$

In which:

ΣEx_{out} is the exergy of the outgoing process and energy flows

ΣEx_{in} is the exergy of the ingoing process and energy flows

$$\eta_{Ex,u\ (heat\ exchanger)} = \frac{\Sigma Ex_{s,out} + \Sigma Ex_{p,out}}{\Sigma Ex_{s,in} + \Sigma Ex_{p,in}} \tag{5.2}$$

If only exergy quantities involved in the process is looked, it is seen the exergy quantities $Ex_{p,in}$ and $Ex_{s,out}$ flowing through the process without any change. These flows can be regarded as ballast flows that are not part of the process and thus need to be considered in the process assessment.

If possible, the specification of a functional efficiency should relate only to changes in exergy quantities. It is assumed that the purpose of the heat exchanger is to heat the primary flow. The exergy change ΔEx_p of the primary flow can be regarded as $Ex_{product}$ from the formula below;

$$\eta_{Ex,f} = \frac{\Sigma Ex_{product}}{\Sigma Ex_{source}} \tag{5.3}$$

The secondary flow provides the supply of exergy. The exergy change ΔEx_p of the secondary flow can thus be regarded as Ex_{source}. For the functional exergy efficiency of the heat transfer process, we can then write:

$$\eta_{Ex,f\ (heatexchanger)} = \frac{\Delta Ex_p}{\Delta Ex_s} = \frac{Ex_{p,out} - Ex_{p,in}}{Ex_{s,in} - Ex_{s,out}} \tag{5.4}$$

In this comparison, the exergy flow seen as ballast is not into consideration. Actually, in comparison (5.2) the exergy loss is related to the total exergy supplied ($Ex_{s,in}$ + $Ex_{p,in}$), while in comparison (5.3) the exergy loss is related to the exergy change in the secondary medium (ΔEx_s). Since ΔEx_s is always smaller than ($Ex_{s,in}$ + $Ex_{p,in}$), the functional efficiency (5.4) is more sensitive to changes in an exergy loss than the universal efficiency (5.2).

In system, both efficiency type (universal efficiency and functional efficiency) are calculated where apparatus is suitable for both calculation.

5.2.2 System Diagram Explanation

In analyses, some data are installed to the program and some data are calculated from the program. But to understand program scheme, each component seen on the diagram should be explained. After all explanations, system and calculation steps can be understood fully. SSIP has divided into two parts and figure 5.2a and figure 5.2b shows technical scheme with results of SSIP thermodynamic analysis. The following tables show which devices used in the program represent the actual system. A description of the devices used is also given in the table 5.1 and 5.2 respectively.

Table 5.1 Explanation of apparatus in the scheme of SSIP thermodynamic analysis scheme part 1

Apparatus No in Scheme	Apparatus Name Given by Program	Apparatus Name for Intended Use	Explanation
1	Gasifier	Fluidized Bed Combustor	In the program fluidized bed combustor is shown as gasifier and all inlets and exit has designed according to program.

2	Heat Exchanger	Economizer 1	Economizer 1 is shown as a heat exchanger that heat transfer is occurred between flue gas and fresh air exit from economizer 2.
3	Sink/Source	Sludge Tank	Sludge tank is shown as a source which mass flow rate, temperature and pressure of sludge can be determined.
4	Heat Exchanger	Economizer 2	Same aim with economizer 1.
5	Sink/Source	Oil Line Inlet	Oil line inlet is shown as a source which supplies oil to fluidized bed combustor for drying process of sludge in drying unit.
6	Sink/Source	Oil Line Exit	Oil line exit is shown as sink which takes oil from fluidized bed combustor with an increased temperature according to oil inlet temperature.
7	Heat Exchanger	Economizer 3	This apparatus is used to show waste heat that is sent to sludge store.
8	Sink/Source	Fresh Air Supply	This apparatus is used to show fresh air supplying from environment. The pressure and temperature of the air is at environment conditions which is determined in the program (T = 15°C P=1.01325 bar)
9	Sink/Source	Fresh Air Supply	This apparatus is used to show supplying fresh air to economizer 3. The temperature and pressure of air is at environment conditions.
10	Sink/Source	Sludge Storage	Sink is used to determine sludge store which waste heat is transferred to.
11	Sink/Source	Sink	This apparatus is used to get values of flue gas as a result of combustion. The values of this sink are used to calculate results in the second part of the scheme.

Table 5.2 Explanation of apparatus in the scheme of SSIP thermodynamic analysis scheme part 2

Apparatus No in Scheme	Apparatus Name Given by Program	Apparatus Name for Intended Use	Explanation
1	Sink/Source	Source from Part 1	This apparatus is used to get results from the first part of the scheme.
2	Separator	Cyclone	The apparatus is used to show cyclone in the system that separates ashes from the flue gas.
3	Separator	Dust Filter	The apparatus is used to show dust filter in the system that separates dust from the flue gas.
4	Compressor	Induced Draft Fan	This apparatus is used to show fan which pushes the flue gas from dust filter to wet chimney cooler.

5	Scrubber	Wet Chimney Cooler	This apparatus is used to show cooler part of wet chimney.
6	Stack	Chimney	This apparatus is used to show stack which the releasing point of flue gas to the atmosphere.
7	Sink/Source	Water Inlet	This apparatus is used to show cooling water supply of wet chimney.
8	Sink/Source	Water Exit	This apparatus is used to show cooling water exit of wet chimney.
9	Compressor	Dust Filter Fan	This apparatus is used to show dust filter fan in the system that transfer dust from dust filter to dust tank.
10	Sink/Source	Dust Tank	This apparatus is used to show dust tank that dust is stored after separation process in dust filter.
11	Compressor	Ash Filter Fan	This apparatus is used to show ash filter fan in the system that transfer ash from cyclone to ash tank.
12	Sink/Source	Ash Tank	This apparatus is used to show ash tank that ash is stored after separation process in the cyclone.

In the analysis, standard flue gas and standard air is used that components are available in the database of the program. Environment condition is determined in order to calculate exergetic efficiency of each apparatus and whole system. Standard flue gas is composed of Ar, CO_2, H_2O and N_2 and the percentages of the components are 0.82%, 9.50%, 19.19%, 70.49% respectively. Standard air is composed of Ar, CO_2, H_2O, N_2, O_2 and percentages of the components are 0.92%, 0.03%, 1.01%, 77.29% and 20.75% respectively. To calculate exergetic efficiency of the system, environment condition should be specified in the program. Specified environment temperature and pressure are 15°C and 1.01325 bar respectively. The numerical enthalpy, entropy and exergy values of states available in the database of the program are different from widely used literature but in thermodynamic analyses, the difference of enthalpy values between states is meaningful for thermodynamic analysis. So, the calculated enthalpy value differences between the two states give very close results when compared with literature. It shows that, there is no problem about using the enthalpy, entropy and exergy numerical values which even are different from values that available in literature. It is assumed that there is no energy loss in pipes. So, temperature and pressure values are the same at the beginning of pipe and at the end of pipe. Although the fans are not shown in the scheme part 1, their energy consumption is taken into account for all calculations. Thermodynamic properties of the states in part 1 and part 2 presented in Figs. 5.2a and 5.2b of the existing SSIP are given in Table 5.3a and Table 5.3b.

Table 5.3a Thermodynamic properties of the states of the 1st part of the SSIP

State	Fluid	P (bar)	T (˚C)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kj/kg)
1	Sludge	1.013	200.00	2.344	-12803.06	8.0141	6286.29
2	Air	1.100	24.00	3.549	-89.76	6.8726	7.09
3	Oil	1.013	200.00	5.555	253.07	0.6587	162.27
4	Oil	1.013	240.00	5.555	325.85	0.8062	192.52
5	Flue gas	1.013	994.00	5.893	-1820.24	8.9532	781.85
6	Air	1.095	538.00	3.549	450.28	7.9208	245.10
7	Air	1.013	669.00	3.549	597.37	8.1112	337.30
8	Flue gas	1.013	876.00	5.893	-1980.69	8.8203	659.69
9	Flue gas	1.012	628.78	5.893	-2305.95	8.5022	426.09
10	Flue gas	1.010	442.00	5.893	-2539.90	8.2124	275.64
11	Air	1.013	24.00	5.258	-89.76	6.8963	0.25
12	Air	1.013	279.95	5.258	172.46	7.5315	79.39

Table 5.3b Thermodynamic properties of the states of the 2nd part of the SSIP

State	Fluid	P (bar)	T (˚C)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kj/kg)
1	Flue gas	1.003	442.00	5.893	-2539.90	8.2145	257.92
2	Flue gas	1.003	222.00	5.186	-2801.26	7.7776	126.82
3	Flue gas	1.003	186.00	4.460	-2842.66	7.6918	111.00
4	Flue gas	1.077	196.29	4.460	-2830.87	7.6956	121.66
5	Flue gas	1.077	65.88	4.461	-3383.60	7.4850	91.25
6	Water	1.077	65,88	1.388	-15695.67	4.4163	10.75
7	Water	1.013	30.00	1.569	-15845.88	3.9487	0.17
8	Ash	1.006	154.00	0.726	-2879.17	7.6084	99.36
9	Ash	1.041	158.56	0.726	-2874.00	7.6102	104.00
10	Ash	1.003	154.00	0.707	-2879.17	7.6084	99.36
11	Ash	1.041	158.56	0.707	-2874.00	7.6102	104.00

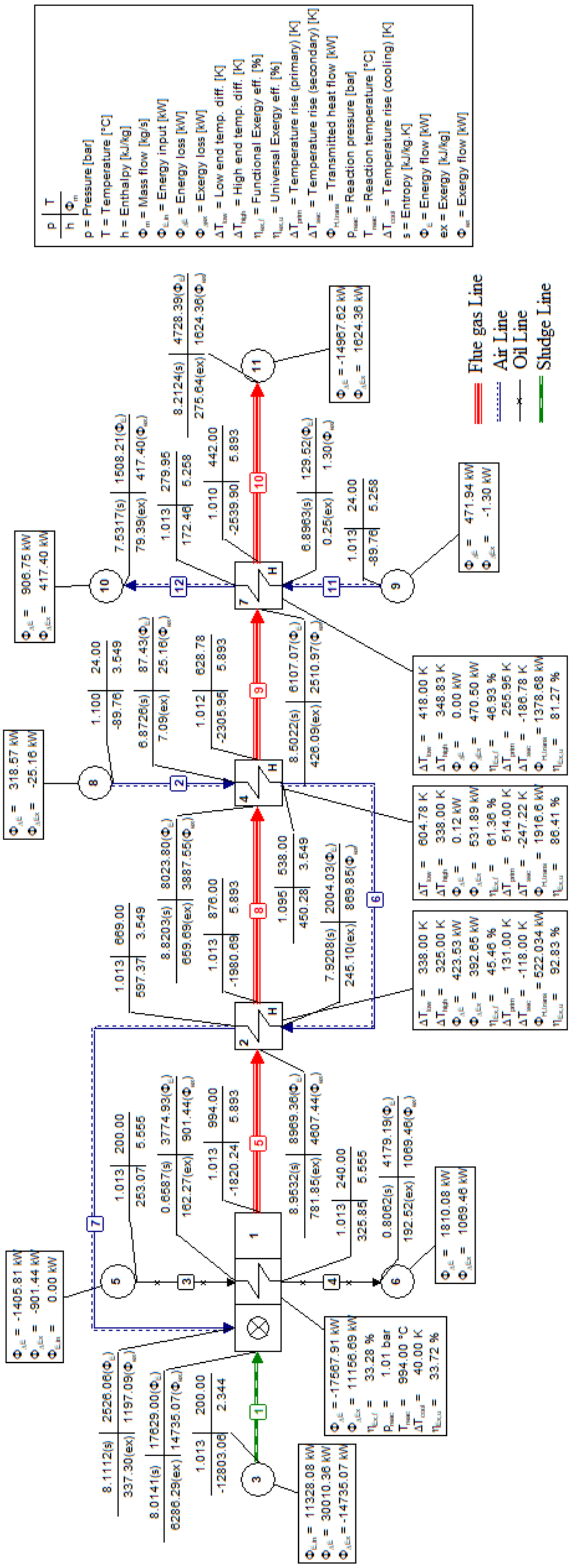


Figure 5.2a Thermodynamic analysis scheme of SSIP - part 1

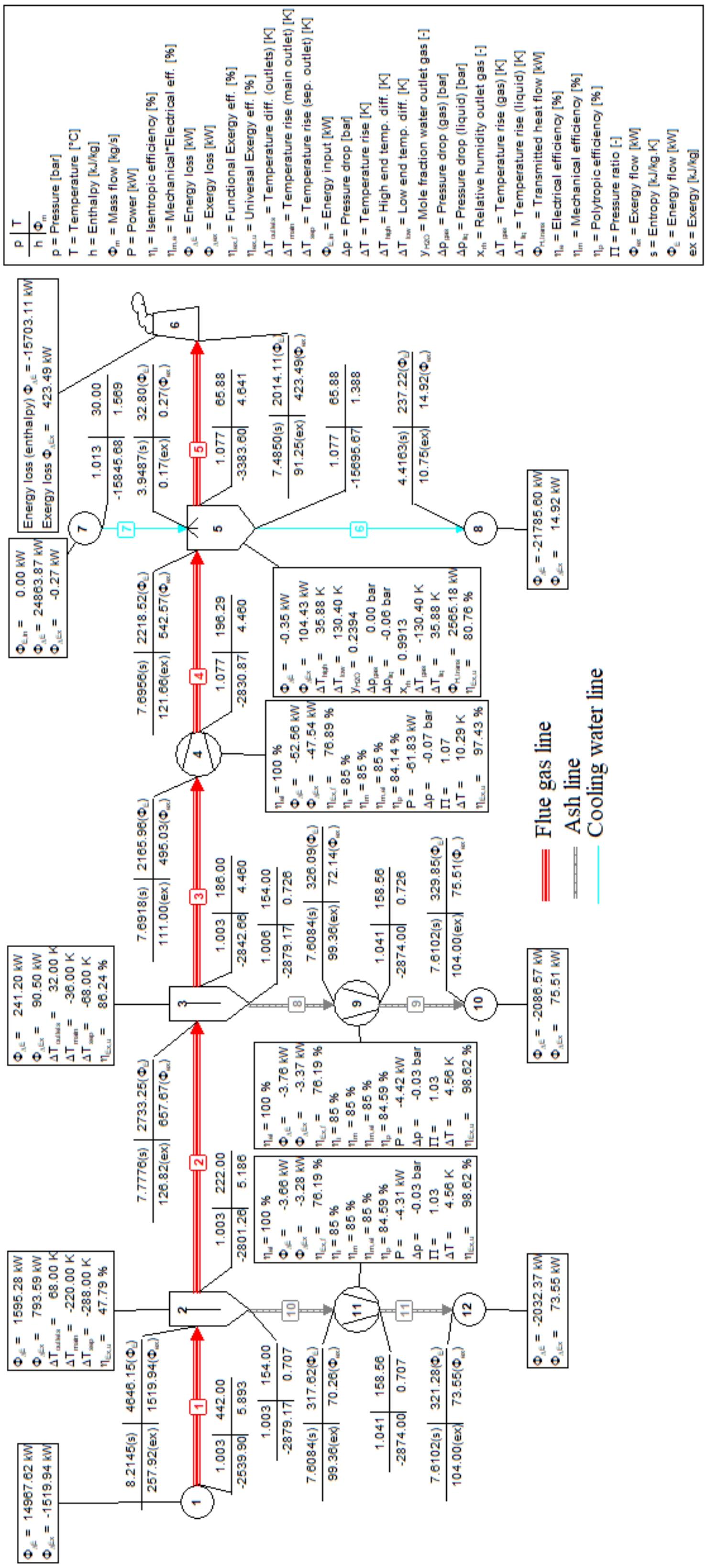


Figure 5.2b Thermodynamic analysis scheme of SSIP - part 2

The heat transfer rates, the work, the exergy destructions and the exergy efficiencies of all sub-components were calculated with the help of the governing equations [124-125] which are given in Table 5.4 and Table 5.5.

Table 5.4 Energy and exergy equations of the subcomponents of the 1st part of the incineration plant

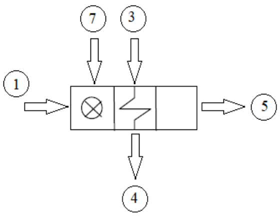
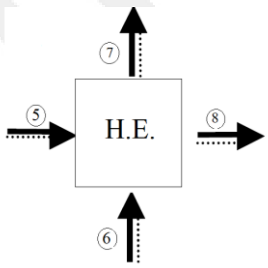
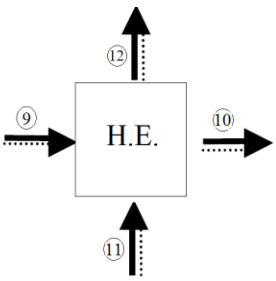
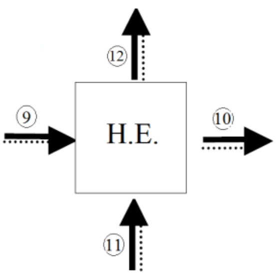
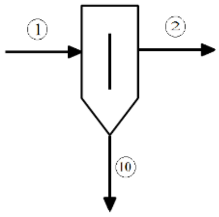
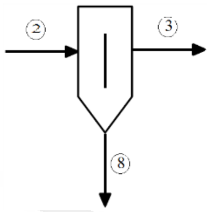
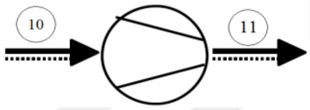
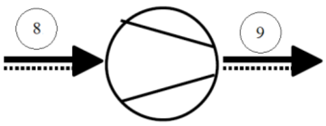
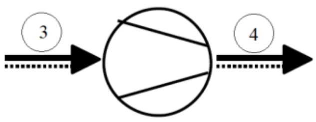
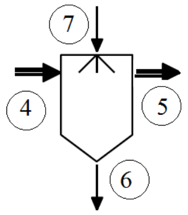
 GASIFIER	$\dot{m}_1 + \dot{m}_7 + \dot{m}_3 = \dot{m}_4 + \dot{m}_5$ $\dot{m}_1 * h_1 + \dot{m}_7 * h_7 + \dot{m}_3 * h_3 = \dot{m}_4 * h_4 + \dot{m}_5 * h_5$ $Ex_D = \dot{m}_3 * \Psi_3 + \dot{m}_1 * \Psi_1 + \dot{m}_7 * \Psi_7 - \dot{m}_4 * \Psi_4 - \dot{m}_5 * \Psi_5$ $\varepsilon = \frac{\dot{m}_4 * \Psi_4 + \dot{m}_5 * \Psi_5}{\dot{m}_3 * \Psi_3 + \dot{m}_1 * \Psi_1 + \dot{m}_7 * \Psi_7}$
 ECONOMIZER 1	$\dot{m}_7 = \dot{m}_6, \dot{m}_8 = \dot{m}_5$ $\dot{m}_5(h_5 - h_8) = \dot{m}_7(h_7 - h_6)$ $Ex_D = \dot{m}_5(\Psi_5 - \Psi_8) - \dot{m}_6(\Psi_7 - \Psi_6)$ $\varepsilon = \frac{\dot{m}_5(\Psi_5 - \Psi_8)}{\dot{m}_6(\Psi_7 - \Psi_6)}$
 ECONOMIZER 2	$\dot{m}_8 = \dot{m}_9, \dot{m}_2 = \dot{m}_6$ $\dot{m}_6(h_6 - h_2) = \dot{m}_9(h_8 - h_9)$ $Ex_D = \dot{m}_8(\Psi_8 - \Psi_9) - \dot{m}_6(\Psi_6 - \Psi_2)$ $\varepsilon = \frac{\dot{m}_6(\Psi_6 - \Psi_2)}{\dot{m}_8(\Psi_8 - \Psi_9)}$
 ECONOMIZER 3	$\dot{m}_{10} = \dot{m}_9, \dot{m}_{12} = \dot{m}_{11}$ $\dot{m}_{10}(h_9 - h_{10}) = \dot{m}_{11}(h_{12} - h_{11})$ $Ex_D = \dot{m}_9(\Psi_9 - \Psi_{10}) - \dot{m}_{11}(\Psi_{12} - \Psi_{11})$ $\varepsilon = \frac{\dot{m}_{11}(\Psi_{12} - \Psi_{11})}{\dot{m}_9(\Psi_9 - \Psi_{10})}$

Table 5.5 Energy and exergy equations of the subcomponent of the 2nd part of the incineration plant

 CYCLONE	$\dot{m}_2 + \dot{m}_{10} = \dot{m}_1$ $\dot{m}_1 * h_1 = \dot{m}_2 * h_2 + \dot{m}_{10} * h_{10}$ $Ex_D = \dot{m}_1 * \Psi_1 - \dot{m}_2 * \Psi_2 - \dot{m}_{10} * \Psi_{10}$ $\varepsilon = \frac{\dot{m}_2 * \Psi_2 + \dot{m}_{10} * \Psi_{10}}{\dot{m}_1 * \Psi_1}$
 DUST FILTER	$\dot{m}_3 + \dot{m}_8 = \dot{m}_2$ $\dot{m}_2 * h_2 = \dot{m}_3 * h_3 + \dot{m}_8 * h_8$ $Ex_D = \dot{m}_2 * \Psi_2 - \dot{m}_3 * \Psi_3 - \dot{m}_8 * \Psi_8$ $\varepsilon = \frac{\dot{m}_3 * \Psi_3 + \dot{m}_8 * \Psi_8}{\dot{m}_2 * \Psi_2}$
 ASH FAN	$\dot{m}_{11} = \dot{m}_{10}$ $\dot{W}_{fan,actual} = \dot{m}_{11}(h_{11} - h_{10})$ $\dot{W}_{fan,isentropic} = \dot{m}_{11}(\Psi_{11} - \Psi_{10})$ $Ex_D = \dot{W}_{fan,a} - \dot{W}_{fan,is}$ $\varepsilon = \frac{\dot{W}_{fan,is}}{\dot{W}_{fan,a}}$
 DUST FILTER FAN	$\dot{m}_8 = \dot{m}_9$ $\dot{W}_{fan,actual} = \dot{m}_9(h_9 - h_8)$ $\dot{W}_{fan,isentropic} = \dot{m}_9(\Psi_9 - \Psi_8)$ $Ex_D = \dot{W}_{fan,a} - \dot{W}_{fan,is}$ $\varepsilon = \frac{\dot{W}_{fan,is}}{\dot{W}_{fan,a}}$
 INDUCED DRAFT FAN	$\dot{m}_4 = \dot{m}_3$ $\dot{W}_{fan,a} = \dot{m}_4(h_4 - h_3)$ $\dot{W}_{fan,is} = \dot{m}_4(\Psi_4 - \Psi_3)$ $Ex_D = \dot{W}_{fan,a} - \dot{W}_{fan,is}$ $\varepsilon = \frac{\dot{W}_{fan,is}}{\dot{W}_{fan,a}}$

 WET CHIMNEY COOLER	$\dot{m}_7 = \dot{m}_6, \dot{m}_4 = \dot{m}_5$ $\dot{m}_4(h_4 - h_5) = \dot{m}_7(h_6 - h_7)$ $Ex_D = \dot{m}_4(\Psi_4 - \Psi_5) - \dot{m}_7(\Psi_6 - \Psi_7)$ $\varepsilon = \frac{\dot{m}_7(\Psi_6 - \Psi_7)}{\dot{m}_4(\Psi_4 - \Psi_5)}$

In the analysis, universal exergetic efficiency is calculated by program instead of functional exergetic efficiency of cyclone and dust filter. Also there is not any exergetic efficiency calculation for chimney. The exergetic efficiencies of all sub-components are given in table 5.6.

Table 5.6 Exergetic efficiencies of subcomponents of the SSIP

Component name	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Eff. (%)
Gasifier	-17567.91	11156.69	33.72(1)
Economizer 1	423.53	392.65	45.46(1)
Economizer 2	0.12	531.98	61.36(1)
Economizer 3	0.00	470.50	46.93(1)
Cyclone	1595.28	793.59	47.79(2)
Dust Filter	241.20	90.50	86.24(2)
Ash Fan	-3.66	-3.28	76.19(1)
Dust Filter Fan	-3.76	-3.37	76.19(1)
Induced Draft Fan	-52.56	-47.54	76.89(1)
Wet Chimney Cooler	-0.35	104.43	80.76(2)

(1) Functional exergetic efficiency is calculated

(2) Universal exergetic efficiency is calculated

In the system the most exergy destruction is calculated in combustor that is 11156.59 kW which leads exergetic efficiency 33.72%. The reason of this big amount of exergy destruction is because of exothermic reaction in fluidized bed combustor. The exergy of combustion spreads surrounding as a heat. To increase exergetic efficiency of the combustor, the combustor should be insulated well. The second most exergy destruction is calculated at cyclone. The reason of this exergy destruction is separation process. Ash is separated from flue gas and ash carries the exergy as heat to ash tank.

Exergetic efficiency of the actual system is calculated by using the formula given 5.1. Exergetic efficiency of the system is 13.24%.

5.3 Model: Organic Rankine Cycle

5.3.1 System Description

In the model, there are four subcomponents which are called ORC pump (ORCP), ORC turbine (ORCT), ORC condenser (ORCC) and evaporator (EVP). The schematic layout of the ORC system is given in figure 5.3. ın the schematic ORCP is shown as apparatus no 6, ORCT, ORCC and EVP are shown as apparatus no 7, apparatus no 1 and apparatus no 4 respectively. As can be seen, working fluid starts circulation between condenser and pump which is show in schematic diagram as line 3. Working fluid is sent from the condenser to pump and after exiting the pump, the fluid is pressurized (line 4). Then the heat of working fluid is increased by the waste heat coming from the incineration plant by using evaporator. Working fluid in the gas phase is sent to the turbine (line 5) and the heat energy stored in working fluid is converted to mechanical energy. After passing through the turbine working fluid finishes the cycle (line 2). For cooling the working fluid water is used and it is shown in the schematic figure as line 6 and line 7 which are inlet to condenser and outlet from condenser respectively. Gained mechanical energy is transformed into electricity by using a generator. The isentropic efficiency of the turbine and isentropic efficiency of the pump are taken constant and values are 85% and 85% respectively according to the common usage in the literature [126-130]. In addition, three separate scenarios were created as pressure ratios 5, 10 and 20, and in these scenarios the energy and exergy analysis of the system were performed. For the scenario with a pressure ratio of 5 and 10, two sub-scenarios were created. For the pressure ratio 5, in one of these sub-scenarios, the fluid pressure at the inlet of the pump was chosen as 1 bar and the outlet pressure as 5 bars. In the second scenario, the pressure of the fluid at the pump inlet was 2 bars and the pressure at the pump outlet was 10 bars. For the pressure ratio 10, in one of these sub-scenarios, the fluid pressure at the inlet of the pump was chosen as 1 bar and the outlet pressure as 10 bars. In the second scenario, the pressure of the fluid at the pump inlet was 2 bars and the pressure at the pump outlet was 20 bars. These values are selected because there are many researches about pressure effect on organic Rankine cycles [131-

137].While analyzing the models, some general assumptions are accepted such that, the system is a steady state, changing of kinetic energy and potential energy is neglected and there is no loss in pipes and heat exchangers.

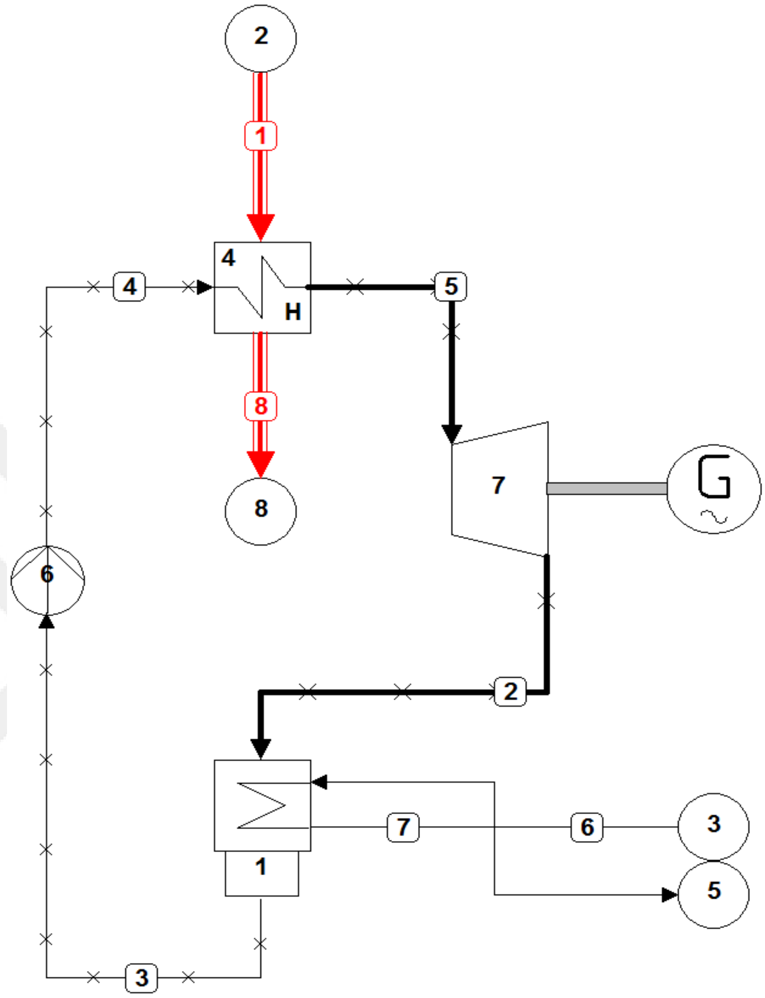


Figure 5.3 Schematic layout of the ORC

Design parameters are determined for seven different working fluids which are cyclohexane, n-pentane, isopentane, ethanol, toluene, R134a and benzene. Table 5.7 shows all design parameters about the system for each working fluid.

Table 5.7 Design parameters for ORC

Inlet Temperature of Heat Source (°C)	279.95
Outlet Temperature of Heat Source (°C)	60
The mass flow rate of Heat Source (kg/s)	5.258
Pressure Ratio of ORC	5/10/20
Isentropic Efficiency of Pump	0.85

Isentropic Efficiency of Turbine	0.85
Inlet Temperature of Cooling Water (°C)	20
Outlet Temperature of Cooling Water (°C)	60
Mass Flow Rate of Working Fluid (kg/s)	2/2.5
Environment Temperature (°C)	15
Environment Pressure (bar)	1.01325

ORC systems are designed for low heat grade sources it works between 80°C and 300°C. The temperature of heat resource of the ORC is approximately 280 °C so, ORC works at its high limits as possible as the working fluid's critical point let. In the literature many working fluids are tried for ORC and these fluids which are given for this analysis are found optimum fluid except fluid mixtures. Selection of the working fluid depends on many criteria. These criterias are as enthalpy, critical temperature, critical pressure, maximum stability temperature and latent heat. Also chemical stability, Ozone Depletion Potential (ODP) and Global Warming Potential (GWP), safety such as flammability, toxicity, and explosivity availability of working fluid in the market and price of the working fluid are another criterias considered when choosing the working fluids. In the literature many researchers studied about working fluid selection on ORC systems. Among all these researches, it is seen that working fluids that selected for this study shows the best performance in most criterias given above [138-149]. Table 5.8 shows the properties of selected working fluids.

Table 5.8 Properties of working fluids used in the ORC system

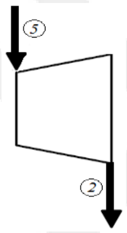
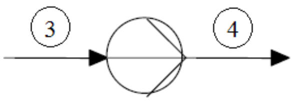
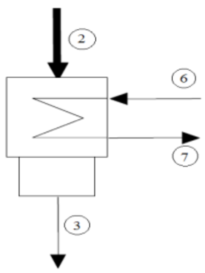
Working Fluid	**Molecular Formula**	**Classification**	**Molecular weight (kg/kmol)**	**P_{cr} (bar)**	**T_{cr} (°C)**	**Fluid Type**
Benzene	C_6H_6	Aromatic hydrocarbon	78.1	9.1	88.9	Dry
Cyclohexane	C_6H_{12}	Cycloalkane	84.2	0.8	80.5	Dry
Ethanol	C_2H_5OH	Alcohol	46.06	1.5	40.8	Wet
Isopentane	C_5H_{12}	Branched-chain alkane	72.1	2.1	87.2	Dry
n-pentane	C_5H_{12}	Linear alkane	72.1	2.1	96.6	Dry
R134a	CF_5CH_2F	Hydrofluorocarbon	102.03	40.6	101.1	Isentropic

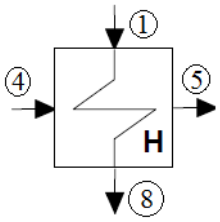
Toluene	C_7H_8	Aromatic hydrocarbon	92.14	41.1	318.7	Dry

5.3.2 Thermodynamic Analysis

The most important results of the thermodynamic analysis are thermal efficiency of system, exergetic efficiency of the system and exergetic efficiency of each component of the system. This result is calculated by using basic governing equations. Table 5.9 shows the used equations for calculation of components efficiencies.

Table 5.9 Thermodynamic equations of subcomponents of the ORC

	$\dot{m}_5 = \dot{m}_2 = \dot{m}_{wf}$ $\dot{W}_{T,a} = \dot{m}_{wf}\,(h_5\text{-}h_2)$ $\dot{W}_{T,s} = \dot{m}_{wf}(\Psi_5\text{-}\ \Psi_2)$ $\dot{E}x_{D,T} = \dot{W}_{T,s} - \dot{W}_{T,a}$ $\varepsilon_T = \frac{\dot{W}_{T,a} * \eta_{mech}}{\dot{W}_{T,is}}$
	$\dot{m}_3 = \dot{m}_4 = \dot{m}_{wf}$ $\dot{W}_{P,a} = \dot{m}_{wf}\,(h_4\text{-}h_3)$ $\dot{W}_{P,s} = \dot{m}_{wf}(\Psi_4\text{-}\ \Psi_3)$ $\dot{E}x_{D,P} = \dot{W}_{P,a} - \dot{W}_{P,is}$ $\varepsilon_P = \frac{\dot{W}_{P,is} * \eta_{mech}}{\dot{W}_{P,a}}$
	$\dot{m}_2 = \dot{m}_3 = \dot{m}_{wf}$ $\dot{m}_6 = \dot{m}_7 = \dot{m}_{water}$ $\dot{m}_{wf}\,(h_2\text{-}h_3) = \dot{m}_{water}\,(h_7\text{-}h_6)$ $\dot{E}x_{D,C} = \dot{m}_{wf}(\Psi_2\text{-}\ \Psi_3) - \dot{m}_{water}(\Psi_7\text{-}\ \Psi_6)$ $\varepsilon_C = \frac{\Psi_7 - \Psi_6}{\Psi_3 - \Psi_2}$

	$\dot{m}_1 = \dot{m}_8 = \dot{m}_{exh}$ $\dot{m}_4 = \dot{m}_5 = \dot{m}_{wf}$ $\dot{m}_{wf}\,(h_5\text{-}h_4) = \dot{m}_{exh}\,(h_1\text{-}h_8)$ $\dot{E}x_{D,EVAP} = \dot{m}_{exh}\,(\Psi_1 - \Psi_8) - \dot{m}_{wf}\,(\Psi_5 - \Psi_4)$ $\varepsilon_{EVP} = \frac{\dot{m}_{wf}(\Psi_5-\Psi_4)}{\dot{m}_{exh}(\Psi_1-\Psi_8)}$

All calculations about ORC are done according to equations which are given in Table 5.3. Moreover, the below equations are used to determine the energy and exergy efficiencies of the ORCs.

$$\eta_{ORC} = (\dot{W}_T - \dot{W}_P) / \dot{m}_{exh}\,(h_1 - h_8) \tag{5.5}$$

$$\varepsilon_{ORC} = (\dot{W}_T - \dot{W}_P) / \dot{m}_{exh}\,(\Psi_1 - \Psi_8) \tag{5.6}$$

For each working fluid, the designed system has many states according to cycle working. For all parametric changing and different working fluid, a state table is prepared with concerning data. In some part of state tables, there is “N/A” in each cell of the table that it means the working fluid can’t be worked with those parameters. So, for the working fluid, there are no state values.

Table 5.10 Properties of states of each working fluid for PR=5 (1-5 bar) and ṁ=2 kg/s

	State	Fluid	T (°C)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	205.37	1.000	2.000	248.02	0.6411	141.46
	3	Working Fluid	76.00	1.000	2.000	-343.25	-0.9542	9.89
	4	Working Fluid	76.16	5.000	2.000	-342.69	-0.9540	10.37
	5	Working Fluid	249.15	5.000	2.000	318.30	0.6150	219.27
	6	Water	20.00	3.000	7.072	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.072	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	204.22	1.000	2.000	306.19	0.7900	150.14
	3	Working Fluid	77.00	1.000	2.000	-291.34	0.8022	11.43
	4	Working Fluid	77.17	5.000	2.000	-290.72	-0.8020	11.97
	5	Working Fluid	236.25	5.000	2.000	370.28	0.7662	221.09
	6	Water	20.00	3.000	7.147	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.147	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	151.87	1.000	2.000	245.84	0.6838	75.40
	3	Working Fluid	35.00	1.000	2.000	-349.15	-1.1247	1.55
	4	Working Fluid	35.19	5.000	2.000	-348.41	-1.1244	2.19
	5	Working Fluid	184.40	5.000	2.000	312.55	0.6560	150.14
	6	Water	20.00	3.000	7.117	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.117	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	167.61	1.000	2.000	197.73	0.5387	141.50
	3	Working Fluid	25.00	1.000	2.000	-410.46	-1.0819	0.27
	4	Working Fluid	25.12	5.000	2.000	-409.91	-1.0816	0.74
	5	Working Fluid	204.25	5.000	2.000	251.14	0.5172	201.11
	6	Water	20.00	3.000	7.274	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.274	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

Isopentane couldn't be used in this scenario because of limit of boiling temperature of isopentane. In the system, isopentane mass flow rate should be more than designed parameters. The same problem occurs for PR=5 (1-5 bar) and mass flow rate is 2.5 kg/s.

The thermodynamic analysis of the system with selected working fluid was done by using the equations mentioned in Table 5.9. The thermodynamic results are given below tables.

Table 5.11 Thermodynamic results of the components in the system with PR=5 (1-5 bar) and ṁ=2 kg/s

		Transmitted Heat Flow (kW)	**Mechanical Power (kW)**	**Energy Loss (kW)**	**Exergy Destruction (kW)**	**Exergetic Efficiency (%)**	η_{ORC} (%)	ε_{ORC}(%)	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-1.24	-1.11	-0.97	78.86	9.48	28.11	125.25
	ORCC	1182.37	-	0	170.2	35.31			
	EVP	1321.81	-		27.86	93.75			
	ORCT	-	126.49	140.55	155.6	81.29			
Cyclo hexan	ORCP	-	-1.37	-1.23	-1.08	78.89	8.62	25.58	113.97
	ORCC	1194.88	-	0	183.49	33.85			

	EVP	1321.81	-	0	27.39	93.85			
	ORCT	-	115.34	128.16	141.89	81.29			
n-pentane	ORCP	-	-1.66	-1.49	-1.28	77.38	8.96	26.57	118.4
	ORCC	1189.89	-	0	54.21	63.3			
	EVP	1321.81	-	0	149.7	66.4			
	ORCT	-	120.06	133.4	149.47	80.33			
Toluene	ORCP	-	-1.22	-1.09	-0.94	76.96	7.18	21.30	94.89
	ORCC	1216.11	-	0	186.85	33.84			
	EVP	1321.81	-	0	44.93	89.92			
	ORCT	-	96.11	106.79	119.19	80.64			

After modelling the system with PR=5 (1-5 bar) and with ṁ=2 kg/s, the system was designed with same pressure interval and values but different mass flow rate which is ṁ=2.5 kg/s. Table 5.12 and table 5.13 shows the state properties of the ORC system and thermodynamic results of the system, respectively.

Table 5.12 Properties of states of each working fluid for PR=5 (1-5 bar) and ṁ=2.5 kg/s

	State	Fluid	T (ºC)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	128.81	1.000	2.500	127.66	0.3676	99.90
	3	Working Fluid	76.00	1.000	2.500	-343.25	-0.9542	9.89
	4	Working Fluid	76.16	5.000	2.500	-342.69	-0.9540	10.37
	5	Working Fluid	172.90	5.000	2.500	186.07	0.3418	165.77
	6	Water	20.00	3.000	7.041	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.041	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	111.08	1.000	2.500	125.14	0.3699	90.17
	3	Working Fluid	45.70	1.000	2.500	-353.60	-0.9884	2.80
	4	Working Fluid	45.84	5.000	2.500	-353.01	-0.9881	3.31
	5	Working Fluid	145.00	5.000	2.500	175.67	0.3465	147.42
	6	Water	20.00	3.000	7.159	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.159	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	94.09	1.000	2.500	123.40	0.3746	42.06
	3	Working Fluid	35.00	1.000	2.500	-349.15	-1.1247	1.55
	4	Working Fluid	35.19	5.000	2.500	-348.41	-1.1244	2.19
	5	Working Fluid	126.75	5.000	2.500	180.21	0.3471	106.79
	6	Water	20.00	3.000	7.068	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.068	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

Toluene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	143.07	1.000	2.500	157.99	0.4459	128.49
	3	Working Fluid	76.60	1.000	2.500	-321.24	-0.8063	10.09
	4	Working Fluid	76.75	5.000	2.500	-320.67	-0.8061	10.60
	5	Working Fluid	180.00	5.000	2.500	208.05	0.4246	184.71
	6	Water	20.00	3.000	7.166	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.166	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

Table 5.13 Thermodynamic results of the components in the system with PR=5 (1-5 bar) and ṁ = 2.5 kg/s

		Transmitted Heat Flow (kW)	**Mechanical Power (kW)**	**Energy Loss (kW)**	**Exergy Destruction (kW)**	**Exergetic Efficiency (%)**	**η_{ORC} (%)**	**ε_{ORC}(%)**	**$\dot{W}_{net}$ (kW)**
Benzene	ORCP	-	-1.54	-1.39	-1.22	78.86	9.83	29.15	129.88
	ORCC	1177.18	-	0	132.52	41.11			
	EVP	1321.81	-	0	57.13	87.18			
	ORCT	-	131.42	146.02	164.66	79.81			
Cyclohexane	ORCP	-	-1.65	-1.48	-1.28	77.8	8.48	25.15	112.04
	ORCC	1196.97	-	0	124.39	43.06			
	EVP	1321.81	-	0	85.28	80.86			
	ORCT	-	113.69	126.32	143.15	79.42			
n-pentane	ORCP	-	-2.07	-1.87	-1.6	77.38	9.51	28.23	125.77
	ORCC	1181.62	-	0	8.46	91.65			
	EVP	1321.81	-	0	184.02	58.7			
	ORCT	-	127.84	142.05	161.85	78.99			
Toluene	ORCP	-	-1.6	-1.44	-1.26	78.88	8.40	24.92	111.05
	ORCC	1198.08	-	0	201.85	31.81			
	EVP	1321.81	-	0	103.25	97.69			
	ORCT	-	112.65	125.17	140.56	80.14			

Another version of designed pressure ratio is 5 but inlet and outlet temperature values are different as mentioned above. The second version of the pressure ratio in the system is 2 and 10 bars. Inlet pressure of pump is 2 bars and outlet pressure of the pump is 10 bars. According to these pressure values and mass flow rates, table 5.14 and 5.15 shows the state values of the designed system with 2 kg/s and 2.5 kg/s mass flow rate, respectively. In this scenario, isopentane wasn't able to be used for 2 kg/s mass flow rate because of the exergetic efficiency value of condenser. In the system, exergetic efficiency is found over 1 which is impossible. For ṁ = 2.5 kg/s toluene isn't used for the system because of temperature values of toluene. Condensation temperature of toluene at 10 bars is 216.19 ºC and for 2.5 kg/s mass flow rate turbine inlet temperature should be lower than condensation temperature of toluene at 10 bars which is impossible because toluene is in liquid form at temperatures below

216.19 ⁰C. So, it is impossible working toluene with this pressure ratio and temperature values.

Table 5.14 Properties of states of each working fluid for PR=5 (2-10 bar) and ṁ=2 kg/s

	State	Fluid	T (⁰C)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	182.38	2.000	2.000	207.95	0.4829	146.96
	3	Working Fluid	50.00	2.000	2.000	-389.45	-1.0921	3.39
	4	Working Fluid	50.29	10.000	2.000	-388.38	-1.0916	4.32
	5	Working Fluid	229.20	10.000	2.000	272.61	0.4577	218.90
	6	Water	20.00	3.000	7.145	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.145	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	161.21	2.000	2.000	215.16	0.5235	135.91
	3	Working Fluid	25.00	2.000	2.000	-391.46	-1.1115	0.42
	4	Working Fluid	25.27	10.000	2.000	-390.31	-1.1109	1.41
	5	Working Fluid	197.25	10.000	2.000	270.44	0.5009	197.69
	6	Water	20.00	3000	7.258	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.258	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	173.76	2.000	2.000	293.74	0.7155	114.18
	3	Working Fluid	55.00	2.000	2.000	-300.33	-0.9718	6.29
	4	Working Fluid	55.44	10.000	2.000	-298.78	-0.9711	7.64
	5	Working Fluid	208.85	10.000	2.000	362.10	0.6883	190.36
	6	Water	20.00	3.000	7.107	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.107	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	176.05	2.000	2.000	209.34	0.5042	163.05
	3	Working Fluid	30.80	2.000	2.000	-400.89	-1.0505	0.79
	4	Working Fluid	31.05	10.000	2.000	-399.79	-1.0499	1.74
	5	Working Fluid	217.00	10.000	2.000	261.15	0.4837	220.75
	6	Water	20.00	3.000	7.300	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.300	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

Table 5.15 Properties of states of each working fluid for PR=5 (2-10 bar) and ṁ=2.5 kg/s

	State	Fluid	T (°C)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene		Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
		Working Fluid	136.76	2.000	2.500	136.90	0.3187	123.23
	3	Working Fluid	80.00	2.000	2.500	-335.79	-0.9333	11.32
	4	Working Fluid	80.33	10.000	2.500	-334.67	-0.9329	12.30
	5	Working Fluid	184.90	10.000	2.500	194.00	0.2940	187.47
	6	Water	20.00	3.000	7.069	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.069	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	149.13	2.000	2.500	191.68	0.4687	128.22
	3	Working Fluid	80.00	2.000	2.500	-284.99	-0.7846	12.68
	4	Working Fluid	80.34	10.000	2.500	-283.75	-0.7840	13.77
	5	Working Fluid	185.75	10.000	2.500	245.04	0.4463	188.04
	6	Water	20.00	3.000	7.127	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.127	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Isopentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	122.40	2.000	2.500	178.27	0.4404	65.64
	3	Working Fluid	44.00	2.000	2.500	-292.63	-0.9764	2.98
	4	Working Fluid	44.47	10.000	2.500	-291.13	-0.9757	4.27
	5	Working Fluid	158.10	10.000	2.500	237.63	0.4137	132.68
	6	Water	20.00	3.000	7.041	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.041	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	118.85	2.000	2.500	171.52	0.4240	75.95
	3	Working Fluid	55.00	2.000	2.500	-300.33	-0.9718	6.29
	4	Working Fluid	55.44	10.000	2.500	-298.78	-0.9711	7.64
	5	Working Fluid	154.90	10.000	2.500	229.87	0.3976	141.92
	6	Water	20.00	3.000	7.057	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.057	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A

After determining thermodynamic properties of each states for PR=5 which inlet pressure of pump is 2 bars and outlet pressure of pump is 10 bars and mass flow rate is 2 kg/s and 2.5 kg/s, the thermodynamic results of each sub-components of the system are given in table 5.16 and table 5.17 respectively.

Table 5.16 Thermodynamic results of the components in the system with PR=5 (2-10 bar) and $\dot{m}$ = 2 kg/s

		Transmitted Heat Flow (kW)	Mechanical Power (kW)	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Efficiency (%)	$\eta_{ORC\,(\%)}$	$\varepsilon_{ORC(\%)}$	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-2.39	-2.15	-4.05	77.08	8.41	24.94	111.13
	ORCC	1194.64	-	0.00	164.33	35.71			
	EVP	1321.81	-	0.00	8.99	97.08			
	ORCT	-	116.38	164.88	185.03	80.20			
Cyclohexane	ORCP	-	-2.57	-2.32	-1.98	76.96	7.33	21.67	96.95
	ORCC	1213.55	-	0.00	175.69	35.18			
	EVP	1321.81	-	0.00	52.92	88.12			
	ORCT	-	99.52	110.58	123.6	80.52			
n-pentane	ORCP	-	-3.44	-3.10	-2.69	78.16	9.57	28.39	126.49
	ORCC	1188.18	-	0.00	122.43	43.27			
	EVP	1321.81	-	0.00	80.12	82.02			
	ORCT	-	123.05	136.72	52.36	80.76			
Toluene	ORCP	-	-2.44	-2.2	-1.89	77.21	7.24	21.47	95.68
	ORCC	1220.40	-	0.00	228.6	29.55			
	EVP	1321.81	-	0.00	7.57	98.3			
	ORCT	-	93.24	103.60	115.40	80.80			

Table 5.17 Thermodynamic results of the components in the system with PR=5 (2-10 bar) and $\dot{m}$ = 2.5 kg/s

		Transmitted Heat Flow (kW)	Mechanical Power (kW)	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Efficiency (%)	$\eta_{ORC\,(\%)}$	$\varepsilon_{ORC(\%)}$	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-3.10	-2.79	-2.45	78.99	9.49	28.14	125.39
	ORCC	1181.83	-	0.00	139.60	50.11			
	EVP	1321.81	-	0.00	7.62	98.29			
	ORCT	-	128.49	142.77	160.63	79.99			
Cyclohexane	ORCP	-	-3.43	-3.09	-2.71	78.99	8.82	26.17	116.61
	ORCC	1191.52	-	0.00	195.20	32.42			
	EVP	1321.81	-	0.00	9.94	97.77			
	ORCT	-	120.04	133.37	149.53	80.28			
Isopentane	ORCP	-	-4.17	-3.75	-3.24	77.75	9.79	29.04	129.37
	ORCC	1177.18	-	0.00	64.16	59.05			
	EVP	1321.81	-	0.00	124.59	72.04			
	ORCT	-	133.54	148.38	167.58	79.69			
n-pentane	ORCP	-	-4.30	-3.87	-3.36	78.16	9.61	28.50	127.00
	ORCC	1179.79	-	0.00	81.47	53.22			
	EVP	1321,.81	-	0.00	109.84	75.35			
	ORCT	-	131.30	145.89	164.93	79.61			

After modeling the system with 5 pressure ratio, the parameter was changed to 10 and 20. For the pressure ratio 10, there are two pressure values used at inlet and exit of the pump which is 1 bar-10 bar and 2 bar-20 bar, respectively. For the pressure ratio 20, working fluid pressure was increased from 1 bar to 20 bars. In modeling, condensate temperature is fixed as close as possible to the boiling temperature of working fluid. On the other hand, as the critical temperature of some working fluids is relatively high, when the heat transfer through the heat exchanger is increased, the second law efficiency of the heat exchanger increases and the amount of energy produced by the working fluid increases in the turbine. On the other side, all T-s diagram are given below to understand working interval of working fluids including R134a and ethanol although the system couldn't be worked with these fluids none of PR values.

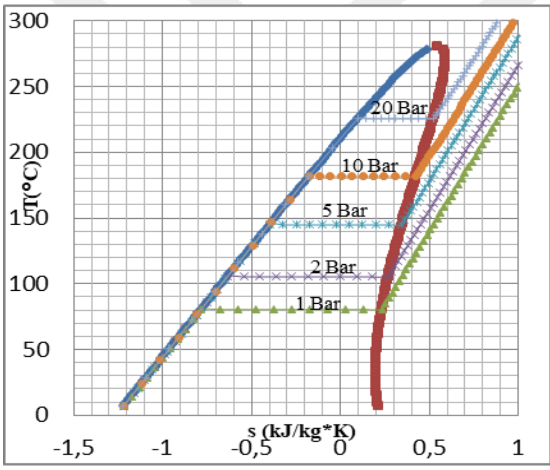


Figure 5.4 T-s diagram of benzene

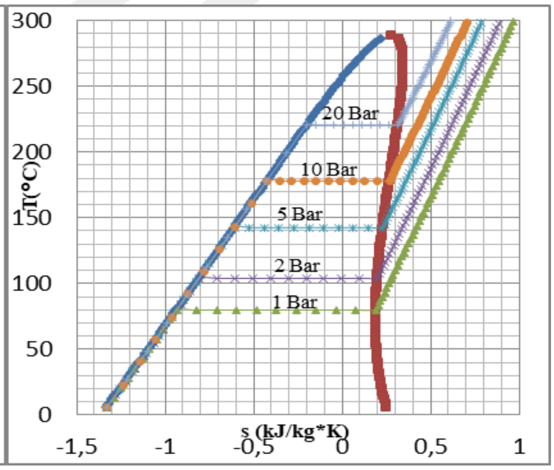


Figure 5.5 T-s diagram of cyclohexane

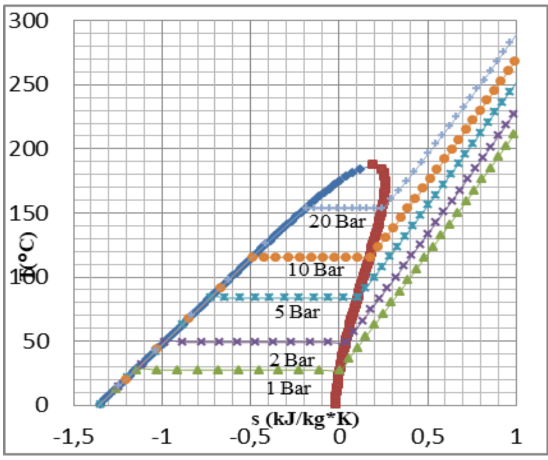


Figure 5.6 T-s diagram of isopentane

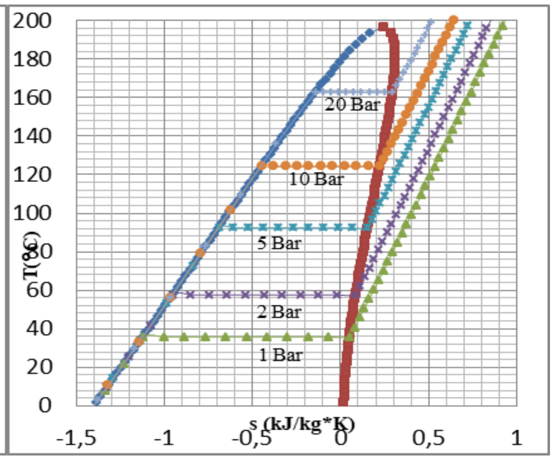


Figure 5.7 T-s diagram of n-pentane

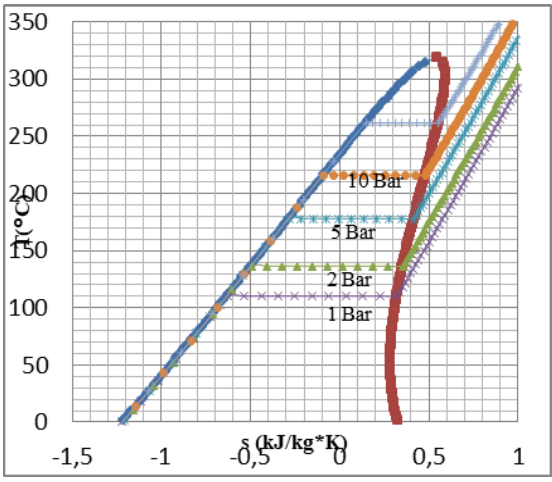


Figure 5.8 T-s diagram of toluene

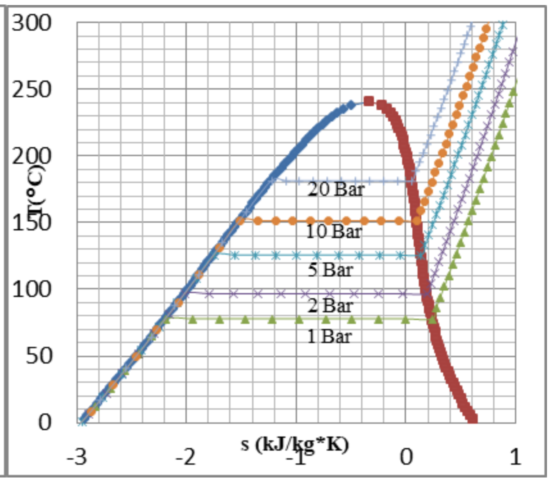


Figure 5.9 T-s diagram of ethanol

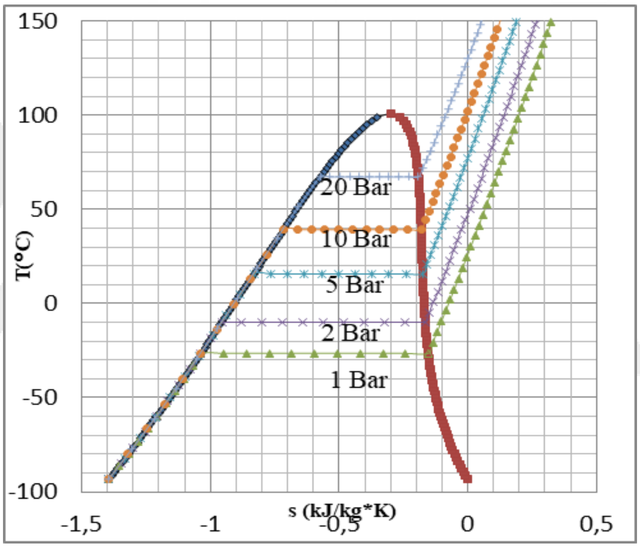


Figure 5.10 T-s diagram of R134a

According to thermodynamic rules, it is expected that if pressure ratio of the system is increased, net work output increases. Also it is the same for first law efficiency and second law efficiency of the system. But there are some differences between working fluids when the system is designed according to the working fluid. To find the best working fluid it is tried to use most suitable working fluids which result of previous studies in this area supports this approach and selected working fluid. The states tables and results tables are given below for PR=10 and PR=20 with ṁ= 2 kg/s and 2.5 kg/s respectively.

Table 5.18 Properties of states of each working fluid for PR=10 (1-10 bar) and ṁ=2 kg/s

	State	Fluid	T (⁰C)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	147.83	1.000	2.000	191.67	0.5351	109.08
	3	Working Fluid	25.00	1.000	2.000	-391.55	-1.1114	0.30
	4	Working Fluid	25.30	10.000	2.000	-390.25	-1.1107	1.41
	5	Working Fluid	197.40	10.000	2.000	270.77	0.5016	197.82
	6	Water	20.00	3.000	6.976	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.976	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	140.77	1.000	2.000	221.24	0.6252	67.70
	3	Working Fluid	35.00	1.000	2.000	-349.15	-1.1247	1.55
	4	Working Fluid	35.42	10.000	2.000	-347.48	-1.1239	2.99
	5	Working Fluid	189.40	10.000	2.000	313.52	0.5855	171.43
	6	Water	20.00	3.000	6.822	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.822	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	161.10	1.000	2.000	187.00	0.5141	137.84
	3	Working Fluid	30.75	1.000	2.000	-401.06	-1.0506	0.67
	4	Working Fluid	31.03	10.000	2.000	-399.82	-1.0500	1.73
	5	Working Fluid	217.00	10.000	2.000	261.15	0.4837	220.75
	6	Water	20.00	3.000	7.034	84.20	0.2964	0.38
	7	Water	60.00	3.000	7.034	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

At PR=10 (1-10 bar) with ṁ=2 kg/s benzene is not working because of exergetic efficiency of evaporator. In the simulated system, exergetic efficiency of evaporator is calculated as 102.48% which is impossible in real to fix mass flow rate at 2kg/s with other designed parameters. For isopentane at 1 bar, boiling temperature of isopentane is 27.57 ⁰C. Cooling water inlet temperature in the condenser is designed as 20 ºC and outlet temperature is designed as 60 ºC. Isopentane is has to be cooled 26.75 ºC which is the maximum temperature in liquid form of isopentane but to reach this temperature, exergetic efficiency is calculated 122.47% in the condenser which is impossible too. The same problem occurs for isopentane when mass flow rate of isopentane is fixed to 2.5 kg/s.

Table 5.19 Properties of states of each working fluid for PR=10 (1-10 bar) and ṁ=2.5 kg/s

	State	Fluid	T (ºC)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	114.22	1.000	2.500	106.89	0.3150	94.30
	3	Working Fluid	76.00	1.000	2.500	-343.25	-0.9542	9.89
	4	Working Fluid	76.37	10.000	2.500	-342.00	-0.9537	10.98
	5	Working Fluid	180.70	10.000	2.500	186.78	0.2781	184.81
	6	Water	20.00	3.000	6.730	84.20	0.2964	0.38
	7	Water	6000	3.000	6.730	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	133.44	1.000	2.500	164.83	0.4702	100.93
	3	Working Fluid	78.00	1.000	2.500	-289.25	-0.7963	11.80
	4	Working Fluid	78.37	10.000	2.500	-287.86	-0.7957	13.02
	5	Working Fluid	183.80	10.000	2.500	240.78	0.4370	186.46
	6	Water	20.00	3000	6.791	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.791	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	84.06	1.000	2.500	103.60	03200	38.01
	3	Working Fluid	35.00	1.000	2.500	-349.15	-1.1247	1.55
	4	Working Fluid	35.42	10.000	2.500	-347.48	-1.1239	2.99
	5	Working Fluid	134.20	10.000	2.500	181.27	0.2812	126.85
	6	Water	20.00	3.000	6.770	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.770	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A

In simulated system at PR=10 (1-10 bar) and ṁ=2.5 kg/s, condensed temperature of toluene is calculated as close as possible of boiling temperature of toluene which is approximately 110 ºC. But when this temperature is used in the program, the exergetic efficiency of evaporator is calculated as 111.56%. So, this value is impossible thermodynamically.

Table 5.20 Properties of states of each working fluid for PR=10 (2-20 bar) and ṁ=2 kg/s

	State	Fluid	T (ºC)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	147.87	2.000	2.000	153.60	0.3589	128.34
	3	Working Fluid	27.35	2.000	2.000	-427.30	-1.2134	0.52
	4	Working Fluid	27.94	20.000	2.000	-424.94	-1.2123	2.55
	5	Working Fluid	222.70	20.000	2.000	222.70	0.3240	220.87
	6	Water	20.00	3.000	6.948	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.948	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Cyclohexane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
Isopentane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	163.70	2.000	2.000	270.45	0.6628	106.08
	3	Working Fluid	55.00	2.000	2.000	-300.33	-0.9718	6.29
	4	Working Fluid	55.98	20.000	2.000	-296.85	-0.9702	9.31
	5	Working Fluid	218.35	20.000	2.000	363.91	0.6247	210.51
	6	Water	20.00	3.000	6.829	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.829	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A

	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A

At PR=10 (2-20 bar) with ṁ=2 kg/s, cyclohexane, isopentane and toluene is not working. For cyclohexane, condensed working fluid temperature is fixed as close as possible to cooling water inlet temperature but with these values when the system is solved, exergetic efficiency of evaporator is calculated 100.04%. As known, this value is impossible thermodynamically. Also this problem occurs for cyclohexane when mass flow rate is fixed to 2.5 kg/s because condensation temperature of cyclohexane at 20 bars is 225.53ºC and heat source temperature is 279.95ºC. It means that, difference between heat source temperature and the condensing temperature of cyclohexane is very close and heat transfer between these two fluids are impossible for 2 kg/s and 2.5 kg/s mass flow rate. For isopentane, when mass flow rate is fixed to 2 kg/s, isopentane is has to be heated more than critical temperature of isopentane to reach 2kg/s mass flow rate even condensed isopentane temperature is fixed to 25ºC. Minimum saturated vapor temperature of toluene at 20 bars is 261.77ºC. It means that, if toluene want to be used in the system, minimum turbine inlet temperature of toluene is 261.77ºC. But in the program, when turbine inlet temperature is fixed that value, the evaporator exergetic efficiency is calculated 124.00% even pump inlet temperature is fixed as high as possible. So, thermodynamically this efficiency value is impossible. The problem is the same for same pressure values with the mass flow rate 2.5 kg/s. So, toluene isn't used in the system with this pressure values and both mass flow rate.

Table 5.21 Properties of states of each working fluid for PR=10 (2-20 bar) and ṁ=2.5 kg/s

	State	Fluid	T (ºC)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
Cyclohexane	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A

	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
Isopentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	113.43	2.000	2.500	159.41	0.3921	60.68
	3	Working Fluid	44.00	2.000	2.500	-292.63	-0.9764	2.98
	4	Working Fluid	45.04	20.000	2.500	-289.26	-0.9748	5.89
	5	Working Fluid	171.30	20.000	2.500	239.39	0.3553	151.28
	6	Water	20.00	3.000	6.760	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.760	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	110.36	2.000	2.500	153.72	0.3781	71.38
	3	Working Fluid	55.00	2.000	2.500	-300.33	-0.9718	6.29
	4	Working Fluid	55.98	20.000	2.500	-296.85	-0.9702	9.31
	5	Working Fluid	169.70	20.000	2.500	231.90	0.3418	160.02
	6	Water	20.00	3.000	6.789	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.789	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
Toluene	1	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A
	2	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	3	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	4	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	5	Working Fluid	N/A	N/A	N/A	N/A	N/A	N/A
	6	Water	N/A	N/A	N/A	N/A	N/A	N/A
	7	Water	N/A	N/A	N/A	N/A	N/A	N/A
	8	Flue gas	N/A	N/A	N/A	N/A	N/A	N/A

For PR=20 (1-20 bar) only limited working fluids can be used in designed system. For mass flow rate 2kg/s only n-pentane and benzene are suitable for the system. For mass flow rate 2.5 kg/s none of working fluid works in the system.

Table 5.22 Properties of states of benzene and n-pentane for PR=20 (1-20 bar) and ṁ=2 kg/s

	State	Fluid	T (ºC)	P (bar)	ṁ (kg/s)	h (kJ/kg)	s (kJ/kg*K)	Ex (kJ/kg)
Benzene	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	129.89	1.000	2.000	129.89	0.3715	100.35
	3	Working Fluid	27.35	1.000	2.000	27.35	-1.2133	0.41
	4	Working Fluid	27.97	20.000	2.000	27.97	-1.2121	2.55
	5	Working Fluid	222.70	20.000	2.000	222.70	0.3240	220.87
	6	Water	20.00	3.000	6.658	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.658	251.39	0.8311	13.52
	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74
n-pentane	1	Flue gas	279.95	1.013	5.258	-2733.85	7.9042	170.48
	2	Working Fluid	131.28	1.000	2.000	200.61	0.5748	61.60
	3	Working Fluid	35.00	1.000	2.000	-349.15	-1.1247	1.55
	4	Working Fluid	35.89	20.000	2.000	-345.62	-1.1230	4.59
	5	Working Fluid	200.40	20.000	2.000	315.37	0.5241	190.96
	6	Water	20.00	3.000	6.576	84.20	0.2964	0.38
	7	Water	60.00	3.000	6.576	251.39	0.8311	13.52

	8	Flue gas	60.00	1.013	5.258	-2985.24	7.3259	85.74

For all these working fluids which can be applied to the system, a system was simulated with a program and for each scenario thermodynamic results of sub-components was calculated. Tables below give the results of sub-components of the system for each scenario.

Table 5.23 Thermodynamic results of the components in the system with PR=10 (1-10 bar) and $\dot{m}$=2kg/s

		Transmitted Heat Flow (kW)	**Mechanical Power (kW)**	**Energy Loss (kW)**	**Exergy Destruction (kW)**	**Exergetic Efficiency (%)**	η_{ORC} (%)	ε_{ORC} (%)	$\dot{W}_{net}$ (kW)
Cyclohexane	ORCP	-	-2.89	-2.6	-2.23	76.96	10.55	31.30	139.47
	ORCC	1166.24	-	0	125.9	42.13			
	EVP	1321.81	-	0	52.83	88.14			
	ORCT	-	142.36	158.17	177.45	80.22			
n-pentane	ORCP	-	-3.73	-3.36	-2.89	77.39	12.28	36.44	162.36
	ORCC	1140.62	-	0	42.67	67.75			
	EVP	1321.81	-	0	108.76	75.59			
	ORCT	-	166.09	184.54	207.42	80.07			
Toluene	ORCP	-	-2.75	-2.47	-2.12	77.21	9.89	29.34	130.71
	ORCC	1175.99	-	0	181.9	33.69			
	EVP	1321.81	-	0	7.59	98.3			
	ORCT	-	133.46	148.29	165.81	80.49			

Table 5.24 Thermodynamic results of the components in the system with PR=10 (1-10 bar) and $\dot{m}$=2.5kg/s

		Transmitted Heat Flow (kW)	**Mechanical Power (kW)**	**Energy Loss (kW)**	**Exergy Destruction (kW)**	**Exergetic Efficiency (%)**	η_{ORC} (%)	ε_{ORC} (%)	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-3.47	-3.13	-2.74	78.87	13.34	39.56	176.27
	ORCC	1125.22	-	0	77.52	63.26			
	EVP	1321.81	-	0	11.05	97.52			
	ORCT	-	179.74	199.71	226.27	79.73			
Cyclohexane	ORCP	-	-3.85	-3.47	-3.04	78.93	12.64	37.49	167.06
	ORCC	1135.38	-	0	133.65	40.03			
	EVP	1321.81	-	0	11.89	97.33			
	ORCT	-	170.91	189.9	213.86	79.91			
n-pentane	ORCP	-	-4.66	-4.19	-3.61	77.39	12.87	38.17	170.09
	ORCC	1131.84	-	0	2.23	97.56			
	EVP	1321.81	-	0	135.95	69.49			
	ORCT	-	174.75	194.16	222.08	78.69			

Table 5.25 Thermodynamic results of the components in the system with PR=10 (2-20 bar) and ṁ=2 kg/s

		Transmitted Heat Flow (kW)	Mechanical Power (kW)	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Efficiency (%)	η_{ORC} (%)	ε_{ORC} (%)	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-5.25	-4.73	-4.05	77.08	10.83	32.13	143.15
	ORCC	1161.65	-	0	164.33	35.71			
	EVP	1321.81	-	0	8.99	97.98			
	ORCT	-	148.40	164.88	185.03	80.2			
n-pentane	ORCP	-	-7.73	-6.95	-6.04	78.18	12.15	36.03	160.54
	ORCC	1141.8	-	0	109.9	44.95			
	EVP	1321.81	-	0	43.10	90.33			
	ORCT	-	168.27	186.96	208.91	80.55			

Table 5.26 Thermodynamic results of the components in the system with PR=10 (2-20 bar) and ṁ=2.5kg/s

		Transmitted Heat Flow (kW)	Mechanical Power (kW)	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Efficiency (%)	η_{ORC} (%)	ε_{ORC} (%)	$\dot{W}_{net}$ (kW)
Isopentane	ORCP	-	-9.37	-8.43	-7.29	77.77	12.91	38.29	170.61
	ORCC	1130.26	-	0	55.47	61.56			
	EVP	1321.81	-	0	82.07	81.58			
	ORCT	-	179.98	199.98	226.52	79.46			
n-pentane	ORCP	-	-9.66	-8.69	-7.55	78.18	12.58	37.31	166.24
	ORCC	1135.05	-	0	73.52	54.82			
	EVP	1321.81	-	0	68.84	84.55			
	ORCT	-	175.9	195.45	221.59	79.38			

Table 5.27 Thermodynamic results of the components in the system with PR=20 (1-20 bar) and ṁ=2kg/s

		Transmitted Heat Flow (kW)	Mechanical Power (kW)	Energy Loss (kW)	Exergy Destruction (kW)	Exergetic Efficiency (%)	η_{ORC} (%)	ε_{ORC} (%)	$\dot{W}_{net}$ (kW)
Benzene	ORCP	-	-5.55	-4.99	-4.27	77.08	14.13	41.91	186.72
	ORCC	1113.16	-	0	112.39	43.76			
	EVP	1321.81	-	0	8.96	97.99			
	ORCT	-	192.27	213.64	241.04	79.77			
n-pentane	ORCP	-	-7.86	-7.07	-6.08	77.41	15.03	44.59	198.68
	ORCC	1099.39	-	0	33.71	71.93			
	EVP	1321.81	-	0	72.88	83.64			
	ORCT	-	206.54	229.49	258.69	79.84			

In all scenarios, system efficiencies are as important as component exergetic efficiency. Thermal efficiency and exergetic efficiency of the systems are calculated according to the formula given 5.5 and 5.6.

5.3.3 Effect of Pressure Ratio on Net Work and System Efficiencies

In the study, seven organic fluids with three different pressure ratios were tried to work in the designed system. For this kind of low-heat source systems, working fluid selecting becomes important. In the study two of seven fluids don't work any of pressure ratios with any designed mass flow rate. These two fluids are R134a and Ethanol. R134a is a fluid using for cooling systems and critical temperature are very low compared to other used working fluids in the study. Because of working temperature intervals of R134a which boiling temperature is -26.45ºC at 1 bar and condensing temperature is 67.22ºC, the fluid is not suitable for the application. On the other side, although ethanol has quite good temperature values at 20 bars and 1 bar, it is not used for the cycle as well. The reason is that ethanol has the similar thermodynamic behavior with water and to get desired pressure values of ethanol, much pump energy is required which is more than energy produced in the turbine. So using ethanol in the system is not suitable.

Other working fluids used in the system, shows the different performance according to pressure ratio, inlet and outlet pressure values of the pump and mass flow rate. Only n-pentane can be used for all parametric scenarios with both mass flow rate values. Benzene is second most used working fluid in systems with many pressure ratios and mass flow rate. Toluene can be used only small PR values. The most restricted used working fluid is isopentane. Isopentane can be used only in two scenarios. The system efficiencies and the net work of the ORC systems are given as a figure below.

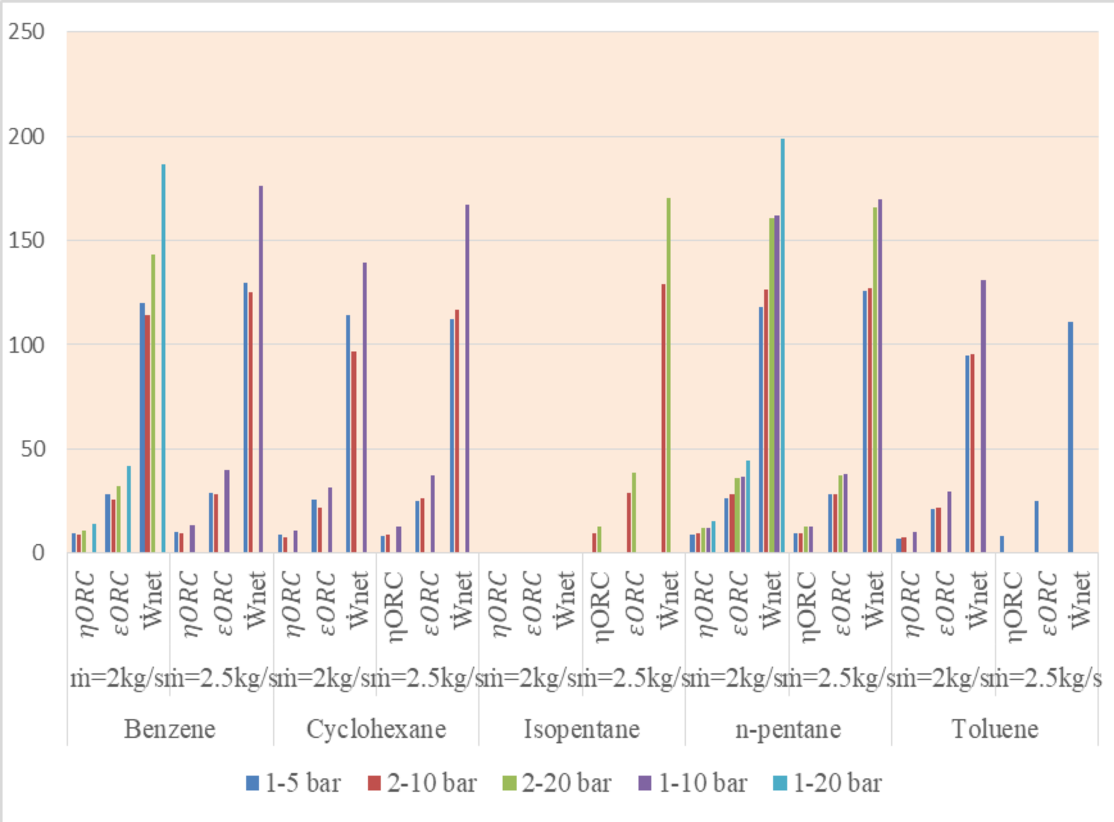


Figure 5.11 System efficiencies and net work output of ORC systems

In designed system the most promising working fluid is n-pentane. Net work output of n-pentane system at PR=20 and ṁ=2 kg/s is 198.68 kW. It is shown that when PR values are increased, thermal efficiency, exergetic efficiency and net work output of the system increase as well. For low grade PR values such as PR=5 (1-5 bar) benzene is the most promising working fluid. Moreover, for PR=5 (2-10 bar) benzene and n-pentane show the best performance but benzene net work output is less than the system works between 1 bar and 5 bars. Normally it is expected increasing net work output when the pressure difference between inlet and outlet of the pump increases. The similar behavior is seen for cyclohexane at 2kg/s mass flow rate with PR=5 (1-5 bar) and PR=5 (2-10 bar). For PR=10 (1-10 bar) benzene shows the best performance for 2.5kg/s mass flow rate. The thermal efficiency, exergetic efficiency and net work output of the system are 13.34%, 39.56% and 176.27 kW respectively. In this condition, isopentane doesn't work in the system for both mass flow rates. Also for mass flow rate 2kg/s, benzene and for mass flow rate 2.5 kg/s, toluene doesn't work in the system. When it comes same PR value but different inlet and outlet pressure values for PR=10, isopentane shows the best performance at 2.5 kg/s mass flow rate. The thermal efficiency, exergetic efficiency and a net work of isopentane are

12.91%, 38.29%, and 170.61 kW respectively. Toluene is the working fluid that shows the worst performance in the study. Toluene has restricted using and when the efficiencies and net work output is considering, the values are much lower than systems used other working fluids. Another question for this study is pump selecting. For all parametric scenarios volumetric flow rate changes between 8.22 m^3/h and 14.61 m^3/h that belongs to benzene and isopentane, respectively. The minimum pressure difference among all scenarios is 4 bars and maximum pressure difference is 19 bars. In the literature there are similar studies that calculate electrical consumption in an experimental way [140]. When results are compared between literature and program, it shows that program has consistent results. Also in the industry, multistage centrifugal pump are used for ORC systems. In the market, the pump which is suitable for the designed system can be found easily. Figure 5.11 shows the pump energy consumption for each working fluid in each designed system.

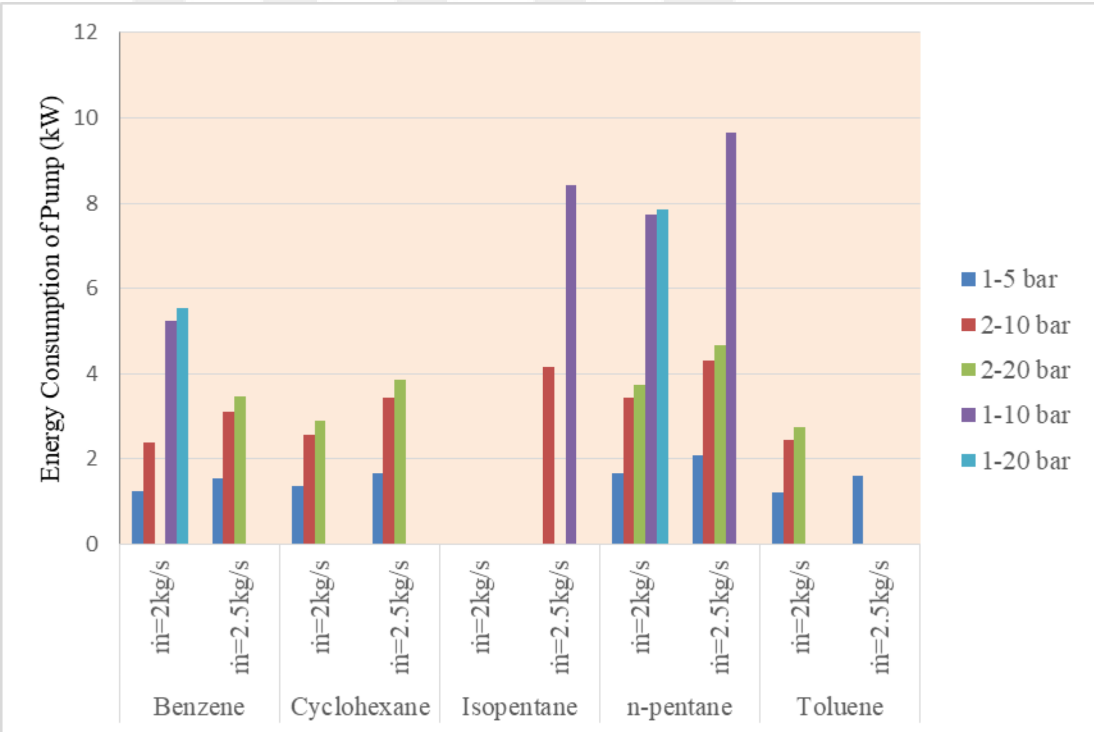


Figure 5.12 Energy consumption of pump regarding PR values

As it is expected, greater pressure difference greater energy consumption of the pump in the systems. However, when net work production is compared with the energy consumed by the pump, for each unit of energy the pump consumes, the energy generated by the turbine is more than the energy consumed by the pump. The best power generation is the same as in n-pentane based systems, and the highest pump energy consumption occurs in systems performed with n-pentane. In parallel with the production in turbines, the least pump energy consumption was observed in systems using toluene. It is seen that the pressure difference, pressure ratio and change of mass flow rate have limited effect on exergy efficiency of pumps used in systems.

5.3.4 Cooling Water Usage in the System

The cooling water used in the systems becomes available in the facility for office heating, worker showers and other hot water demanding works. The energy carried by sixty Celsius degrees of water, together with the flow rate, is a great energy, and the re-use of this energy is very important in terms of energy efficiency. The following figure shows the flow rates of the cooling water used in the systems.

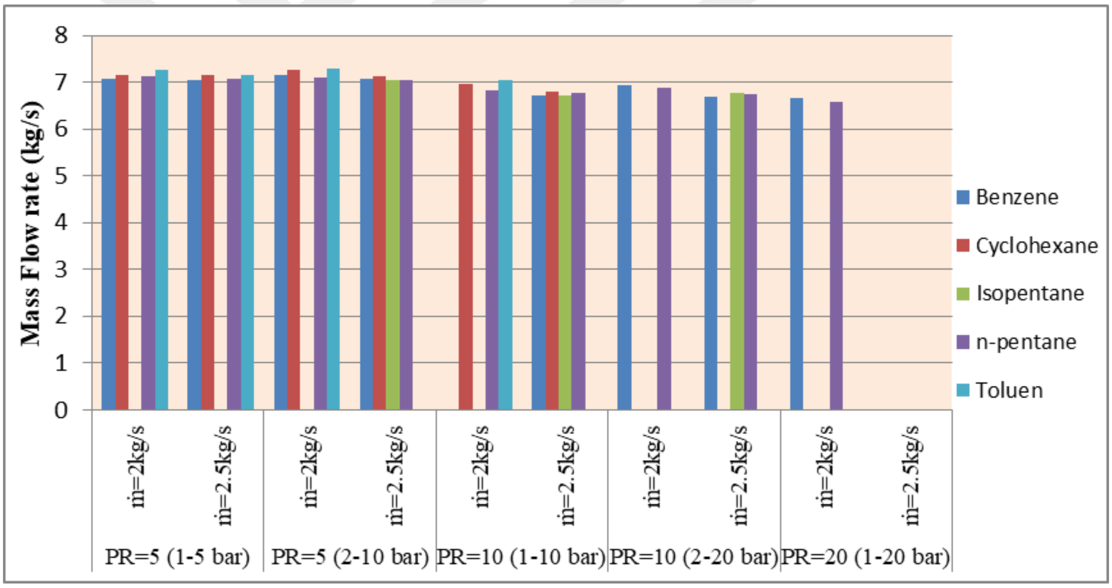


Figure 5.13 Cooling water amount used in the ORC systems

The system using toluene has the highest flow rate and the pump inlet and outlet pressure values of this system are 2 bar and 10 bar, respectively. In this system, the mass flow rate of toluene is 2 kg/s. In this system, the total energy carried by 60ºC water is 1835.02 kW. In the studies in the literature, the amount of hot water usage at a temperature of 60ºC for an apartment is given as 272.5 l/h [150]. The hot water flow produced in the system is 26680 l/h. In this case, the cooling water used in the system can meet the hot water requirement of approximately 98 apartments.

5.5 Conclusion

Energy and exergy analysis of GASKI SSIP are performed. An ORC system is designed to utilize the waste heat generated by the system. A total of 7 working fluids are selected for this system, and since 2 of these 7 are not suitable, only 5 working fluids are simulated by means of computer software. Design parameters are selected for the system. The pressure ratio and the mass flow rate, which are important among the parameters, are changed and how the efficiency of the systems changed against the changing parameters is calculated. At the same time, the usage areas of the energy stored in the cooling water used in the system are mentioned.

CHAPTER 6

CONCLUSION

In this study, thermodynamic analysis of sewage sludge incineration plant and thermodynamic analysis of ORC system modeled by utilizing waste heat generated in the system were performed. The capacity of the plant where the thermodynamic analysis is performed is 300 tons / day, but the incineration system operates with a capacity of 200 tons / day when the thermodynamic analysis data is collected. All thermodynamic analyzes were performed using real data taken from the facility. All the required formulas for analysis are given in table 5.4 and 5.5 in Chapter 5. A computer program was used for the thermodynamic analysis of the facility. The schematic of the analyzed plant was drawn in the program and all the figures in the drawing were explained. All data required for analysis were entered into computer program one by one for each component and thermodynamic analysis of incineration plant was performed.

This analysis is only the first part of this study. The main aim is to increase the energy efficiency of the plant by converting the waste heat produced by this plant to reusable energy. Inefficient delivery of waste heat from the incineration plant to 279.95 Celsius degrees reduces the overall energy and exergy efficiency of the system. The mass flow rate of this waste heat is 5.258 kg / sec. In other words, the total exergy of waste heat not used in the plant is 417.40 kW.

Using this system instead of giving this amount of heat loss to the sludge drying tank inefficiently will also increase the efficiency of the system. Since the temperature value of the flue gas is not considered to be very efficient for a gas turbine, ORC is preferred which can operate at lower temperature sources. The design parameters for the system are also given in table 5.7 in chapter 5. Considering the flexibility of ORC to work with various fluids at various temperatures and pressures, it is necessary to find out which fluid performs best for which pressure and mass flow rate. Therefore, seven different fluids were selected during system analysis. The properties of these fluids are given in table 5.8. Of these fluids, ethanol and R134a could not be used for the system under any given conditions. The reason ethanol cannot be used is because ethanol is chemically similar to water. Since the operation of ethanol in the pressure parameters designed for the system requires very high pump forces, it was found that ethanol is not suitable for the designed system. Although R134 is suitable for the system designed as a chemical structure, R134 could not be operated under any conditions like ethanol since the critical temperature values were not suitable for the designed system.

For the other five working fluids, the system was operated under some conditions and under some conditions wasn't operated for various reasons. Tables 5.10, 5.12, 5.14, 5.15, 5.18, 5.19, 5.20, 5.21 and 5.22 show the state properties of the system for the working fluids that is operated. For each working fluid, the systems are run with predetermined parameters and the results are obtained. In Table 5.11, 5.13, 5.16, 5.17, 5.23, 5.24, 5.25, 5.26 and 5.27, the system components and the energy and exergy results of the systems are given. According to analysis the fluid n-pentane was obtained best results. While the pressure ratio is 20 and the mass flow rate is 2 kg / s, the system running n-pentane produced 198.68 kW net work. Another feature of the fluid is that it is a fluid suitable for 9 of the 10 simulated systems, in which no other fluid can work in the model where n-pentane does not work.

Apart from n-pentane, benzene was another fluid that gave the best results. In the system where the pressure ratio is 20 and the mass flow rate is 2kg/s, benzene produced 186.72 kW net work. This work is 6.02% lower than the work produced by n-pentane.

According to the results, toluene was the worst performing fluid in terms of net work production. There are only 4 models in which toluene can be operated; of which toluene is the fluid produced lowest net work in all systems. As example, in the system where pressure ratio is 5 and mass flow rate is 2 kg/s, benzene produces 36.87% more net work than toluene.

Isopentane is the fluid attracting attention in the study. The narrowest operating range among the design parameters belongs to isopentane. Isopentane could only be operated in two systems. However, in both systems, the maximum net work production belongs to isopentane. Isopentane produced a net work of 129.37 kW in a system with a pressure ratio of 5 (2-10 bar) and a mass flow rate of 2.5 kg/s. This amount means that 1.86% more net work is produced than the n-pentane system which produced 127 kW of work output. In the other isopentane system, the pressure ratio is 10 (2-20 bar) and the mass flow rate is 2.5 kg/s. Only two fluids were able to operate in this system, and the other working fluid was n-pentane. The net production of isopentane and n-pentane in the system is 170.61 kW and 166.24 kW respectively. The difference between net work produced in this system is 2.62% in favor of isopentane.

The energy consumption of the each system which is another important issue for energy systems is shown in Figure 5.11. Also, n-pentane is the most energy consuming working fluid for pumping in the systems that work. For each unit of power spent in the pump, more than one unit of power is gained in the turbine. Therefore, increasing the pressure ratio gives better results in terms of net work produced from the system. In the system with a pressure ratio of 5 (2-10 bar) and a mass flow rate of 2.5 kg/s, the system uses n-pentane and the power consumed by the pump is 4.3 kW. The power generated by the turbine in the system is 130.79 kW. In the n-pentane system with a pressure ratio of 10 (2-20 bar) and a mass flow of 2.5 kg/s, the power consumed by the pump is 4.66 kW. On the other hand, the power generated by the turbine in the system is 166.24 kW. The power spent on the pump between the two systems increased by 0.36 kW. On the other hand, the power gained from the turbine increased by 35.45 kW.

In addition to the net work produced in all simulated models, the temperature of the water used to cool the working fluid increases. As the design parameter, the cooling water inlet temperature is 20ºC and the outlet temperature is 60ºC. With the energy carried by the 60ºC hot water obtained as a by-product from the systems, the hot water requirement of a maximum of 98 and a minimum of 86 apartments can be met.

In this thesis, only thermodynamic analysis is performed for the related systems and system comparisons are made only according to the results of thermodynamic analysis. When a more detailed study is required for all of these modeled systems, economic and environmental impact analyzes can be performed as a continuation of this study.

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