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**OPTIMIZATION OF THE PERFORMANCE OF AN
ORGANIC RANKINE CYCLE USED AS WASTE
HEAT RECOVERY SYSTEM IN A BIO-LIQUID
DIESEL ENGINE POWER PLANT**

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A 30 ECTS credit units Master's thesis

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ABSTRACT

Energy savings and efficiency improvements of energy systems are an important task on the path towards a more sustainable future. This thesis describes an optimization of a heat recovery system for an electric power plant running on bioliquid. This system is a part of an electric power plant being designed by Utras Engineering in Great Britain. Plant oils are chosen in this investment to be used as fuel, and therefore the availability of plant oils and their properties as a fuel for diesel engines is examined in the thesis. The heat recovery system uses heat contained in the cooling water from the engine block as well as heat in exhaust gases, and converts it into electric energy. It is also a part of a larger more intricate system where a part of the heat is used for heating buildings, preheating fuel, running an absorption cooling unit. The recovered heat is converted to electricity using Rankine Cycle, different working fluids are addressed. The Steam and Organic cycles are examined. The system is optimized using three different objective functions: power output, exergetic efficiency and net present value. The optimization process is solved in the mathematical model implemented into Engineering Equation Solver. The model is able to perform a complex simulation of the waste heat recovery system. The model simulates the energy and exergy balance of the system and is also equipped with economical instrument: net present value. The results show that according to all of the objective functions Organic Rankine Cycle with isopentane as working fluid used as a recovery system is the most optimal solution. Its electrical output is 1741 kW, exergetic efficiency 27,6% and the system pays back in 6 years. Also further heat recovery is examined. And it is found that there are possibilities for recovering more heat out of the Organic Rankine Cycles. It was found that the superheated steam leaving the turbine is at insufficient temperature and does not contain enough enthalpy for it to be a feasible source for the heating applications though it could be used for regeneration improving cycle efficiency. But this issue is not studied in this work.

After the optimization process the case study is performed to examine if such a power plant with waste heat recovery system could be implemented at different conditions. Poland is examined as a location for the power plant. It is concluded that Poland is not a feasible place for this type of power plant. That is because undeveloped plant oil industry and not sufficient political incentives for biofuels.

PREFACE

This thesis was done as a finalization of one year master degree program called “RES: The School of Renewable Energy Science” in Iceland. The program was financed by EEA Grand which was guided by the PL0460 Project. I am really grateful to Professor Tomasz Wiśniewski who was the leader of PL0460 Project. He always was expressing the willingness for helping me.

The purpose of making this thesis was my high interest in improving the efficiency of the existing energy systems which was inspired by Dr. Paul A. Ericson lectures.

It is obvious that when something is done it is logical to do it in the best possible way. That is my life attitude and it is also my point of view as an engineer. So I started to look for an interesting topic in the field of efficiency improvements.

I found that Mr.Thorhallur Bjarnason from Utras Engeneering is doing an interesting project about the optimization of the waste heat recovery system for a bio diesel power plant. I took that topic and Mr.Thorhallur became my first advisor. I would like to thank him for his help and support with the data and project documentation that I needed in the optimization process. He was always open for helping me. I would also like to thank my second advisor Dr.Gudrun Seavarsdottir from University of Reykjavik with whom I exchanged tens of email from which everyone was pushing the thesis forward. She supported me with academic knowledge and helped me to form the thesis for an professional academic work. The third person who helped me was Dr.Pall Valdimarsson. He supported me with the core of the code which I used to optimize the cycle. Professor William Scott Harvey was supporting me with his personal knowledge about the turbines and the cost analysis of the system. These were the people who helped me directly in making of the thesis.

But I would also like to mention the people who were helping me not directly but were encouraging me and giving their support when I had some hard times. Firstly I would like to say thank you to Bjorn Gunnarson and Sigrun Loa Kristjansdottir who as main RES stuff assisted and helped me during my stay in Iceland even when they had their own problems.

I would like to thank my friends here in Iceland, my college students. When I was coming to Iceland I did not know you, now you are my best friends.

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TABLE OF CONTENTS

1	Introduction.....	1
1.1	Biofuels.....	2
1.2	Goals of the study	5
1.3	Background information	5
1.4	Works Contribution	6
1.5	Research methodology.....	6
1.6	Anticipated results	6
2	Modeling and optimization of the waste heat recovery system.....	7
2.1	Waste heat recovery at bio-liquid power plants.....	7
2.1.1	Power Plant description.....	7
2.1.2	Comparison of different fuels	9
2.2	The mathematical model of the ORC	11
2.2.1	Simple ORC	12
2.2.2	ORC with regeneration.....	21
2.2.3	Steam Cycle.....	22
2.2.4	Different cooling conditions.....	24
2.3	The working fluids of the system	27
2.3.1	Isobutene R600a.....	27
2.3.2	Isopentane.....	29
2.3.3	Steam.....	30
2.4	Definition of objective functions of the ORC optimization	31
2.5	Exergy analysis of the ORC system	32
2.5.1	Exergy introduction.....	32
2.5.2	Exergy destruction definition	36
2.6	Economical analysis of the ORC system.....	36
2.6.1	Investment cost of WHR system.....	37
2.6.2	NPV analysis	38
2.7	EES Model Explanation	39
2.8	Optimization Results	42
2.8.1	Simple ORC and Steam Cycle results comparison	44
2.8.2	Further heat recovery from the flue gases.....	49

3	Feasibility Study Poland	52
3.1	Polish conditions.....	52
3.2	Policy Support on Bio-liquid applications.....	53
3.3	Future development of Bio-liquid Power Plants with WHR.....	55
4	Conclusions.....	57
5	Further Studies	58
	References	59

LIST OF FIGURES

Figure 1-1 World marketed energy consumption in three Economic Growth cases, 1990-2035 (quadrillion Btu) (World Energy Demand and Economic Outlook, 2010)	1
Figure 1-2 Anticipated World energy-related carbon dioxide emissions, 2007-2035 (billion metric tons) (Energy-Related Carbon Dioxide Emissions, 2010)	2
Figure 1-3 World Production of bioethanol (The IEA Energy Technology Essentials, 2010)	3
Figure 1-4 World Production of biodiesel (The IEA Energy Technology Essentials, 2010)	4
Figure 2-1 Scheme of the biodiesel engine with TCS and ORC units (MAN Diesel, 2010)	8
Figure 2-2 The Selective Catalytic Reduction System (Blue NG Becton Project Documentation, 2010)	11
Figure 2-3 The Simple Organic Rankine Cycle on the T(s) diagram	13
Figure 2-4 Pinch point set for the vaporizer	17
Figure 2-5 Pinch point set for the condenser	18
Figure 2-6 Scheme of the ORC waste heat recovery system	21
Figure 2-7 Steam Rankine Cycle on T(s) diagram	23
Figure 2-8 Scheme of the steam waste heat recovery system	24
Figure 2-9 Duration of the dry bulb temperature in London (U.S Department of Energy, 2010)	25
Figure 2-10 Changes in recovery system parameters in different cooling temperature conditions (Grabiński, 2011)	26
Figure 2-11 Changes in net present value after 5 years with different cooling temperatures conditions (Grabiński, 2011)	26
Figure 2-12 Isobutene T-s diagram	28
Figure 2-13 Isopentene T-s diagram	30
Figure 2-14 Water T-s diagram	31
Figure 2-15 EES interface	39
Figure 2-16 The block scheme of the pre-heater procedure	40
Figure 2-17 The block scheme of the condenser procedure	42
Figure 2-18 Sankey diagram of the optimized bioliquid engine with waste heat recovery system	43
Figure 2-19 Net work and exergetic efficiency of the turbine as a function of vaporizer pressure (heat exchange areas assumed) (Grabiński, 2011)	47
Figure 2-20 Net present value of the whole investment after 5 years as a function of vaporizer pressure (heat exchange areas assumed) (Grabiński, 2011)	47

Figure 2-21 Net present value changes in years for all working fluids (Grabiński, 2011) .	49
Figure 2-22 Choosing the dew point temperature of the exhaust gases on the Molier diagram (Grabiński, 2011).....	51
Figure 3-1 Changes in the harvest of rapeseed in years [thousands tons] (Krajowe Zrzeszenie Producentów Rzepaku, 2011)	52
Figure 3-2 Rapeseed oil production sites in Poland (Polskie Stowarzyszenie Producentów Oleju, 2011).....	55

LIST OF TABLES

Table 2-1 Comparison of the properties of the traditional diesel and plant oils (A.K. Hossain, 2009) (*at 38°C)	10
Table 2-2 Biofuel specification (Blue NG Becton Project Documentation, 2010).....	10
Table 2-3 The Constraints of the System (Blue NG Becton Project Documentation, 2010)	12
Table 2-4 Abbreviations for the ORC waste heat recovery system	13
Table 2-5 The Constrains of the system (Blue NG Becton Project Documentation, 2010)	16
Table 2-6 The Constrains of the system (continued) (Blue NG Becton Project Documentation, 2010)	16
Table 2-7 Total heat transfer coefficients (Valdimarsson, 2010).....	19
Table 2-8 Basic Properties of Isobutane(Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)	27
Table 2-9 Basic Properties of Isopentane (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)	29
Table 2-10 Basic properties of Steam.....	30
Table 2-11 Prices of the elements of the ORC system (Perry, 1997).....	38
Table 2-12 The optimal values of the objective functions	44
Table 2-13 The optimal values of objective functions (continued).....	44
Table 2-14 The optimal values of the parameters of the waste heat recovery system	45
Table 2-15 The optimal values of the parameters of the waste heat recovery system (continued).....	46
Table 2-16 Heat exchanger adjusted areas	48
Table 3-1 The minimal EU requirements for the % of biofuels in Poland's transportation market (Government, 2007)	53

1 INTRODUCTION

This thesis is done in the field of world energy. This area is highly complex and all of its relationships are hard to explain in one sentence. It is necessary to introduce basic notions connected to the field of world energy for better understanding the issues of this thesis.

Energy use is as old as human race . At the beginning wood was used for heating the caves and dwellings. As human civilization developed, fossil fuels became more and more prominent and started to be used not only for heating purposes but also craftsmanship work as pottery, blacksmithing, manufacturing of weapons, shipbuilding etc. The number of humans was growing. Dwellings changed into cities. Manufactures started to combine to form guilds. The demand for energy grew higher and higher. But the big breakthrough in the field of energy was still to come. The industrial revolution was made possible through inventions like steam engine, electricity and internal combustion engine. These tools derived from thermodynamics, made it possible to convert energy from heat to work and literally changed the world. Fossil fuels started to be used not only in the heating and manufacturing fields but also in transportation, lighting and house appliances. Energy became available to almost everyone in the world that could afford it. This was a positive issue because it increased the well being and enabled modern development.

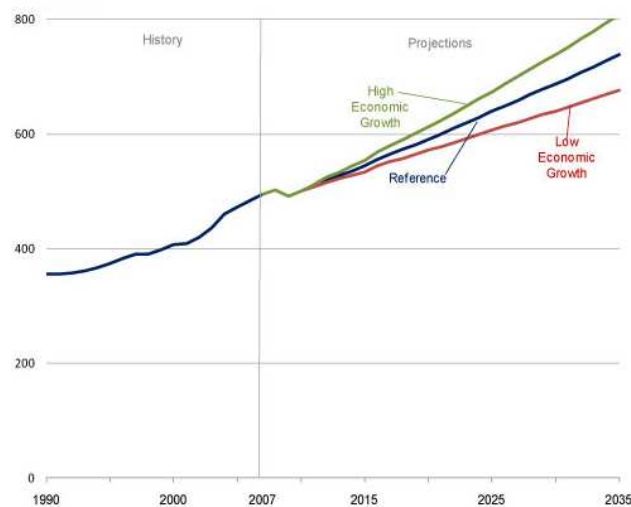


Figure 1-1 World marketed energy consumption in three Economic Growth cases, 1990-2035 (quadrillion Btu) (World Energy Demand and Economic Outlook, 2010)

The present general situation in global energy can be characterized in few points. Firstly the demand for power is still increasing rapidly. The consequence of that is an increasingly negative effect on the environment. One of the most notable effects of fossil fuel combustion is the uncontrolled release of carbon dioxide to the atmosphere. Although carbon dioxide is not toxic most scientists now agree that it has a strong influence on the global climate, so human induced climate change may change living conditions all over the world. Figure 1-2 shows the energy related carbon dioxide emissions. It is visible that the future emissions will be caused by non developed countries. It is predicted that they will increase almost twice in next 25 years.

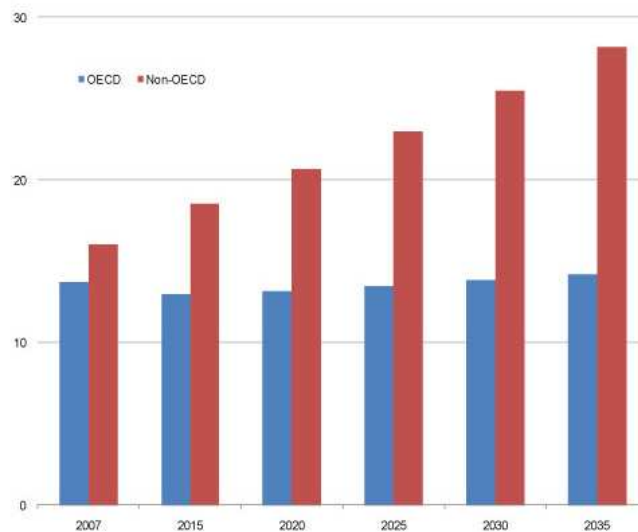


Figure 1-2 Anticipated World energy-related carbon dioxide emissions, 2007-2035 (billion metric tons) (Energy-Related Carbon Dioxide Emissions, 2010)

Secondly at the same time the world is running out of natural energy resources such as: coal, oil and natural gas. This leads to a point where it is necessary to find other possible energy sources which don't pollute the environment on such a scale and can last for much longer period of time. These sources can support the power system with, so called, renewable energy. Examples of it are the: solar, wind, bio-fuel and geothermal energy. They can help in solving the problems mentioned above but usually only locally, on a small scale. Also solar and wind application are intermittent and not possessing high enough potential in many places around the globe. Geothermal sources occur only in few places on earth. As to the bio-fuels there is a big discussion right now about if they really are renewable. This debate is concentrated mostly on the life cycle analysis of the bio-fuel. This issue is described more in details in next chapter.

In all these controversies it is usually forgotten that when approaching sustainability a very important contribution is just saving the energy which is being already used. By increasing the efficiency of energy production or any other industrial process the energy savings can be huge. Of course this leads also to the decrease of the environmental impact (less fuel use) and the increase of profitability of the investment (smaller fuel use and additional energy sold).

The motivation of this study is the large possibilities that lie in improving the energy efficiency of the system.

1.1 Biofuels

Biofuels are the solid, liquid and gaseous fuels made from biomass. This is defined as all organic material that grows by the photosynthetic conversion of solar energy. Because of the topic of the thesis only liquid biofuels are discussed.

There are many brand names of biofuels but in general three main groups can be specified: bioalcohols, biodiesel, vegetable oils.

The main representative of bioalcohols is bioethanol. It is produced in the process of fermentation of sugars or starch with pre-treatment with enzymes. The feedstocks for ethanol are the plants rich in sugar or starch like maize or sugar cane. Bioethanol is used in transportation for internal combustion spark ignition engines. It mixes with gasoline making blends called E5, E10 and E85. It has lower energy content than gasoline.

Another type of liquid biofuel is biodiesel. There can be two types of biodiesel: so called green diesel and FAME (fatty acid methyl esters).

Green diesel is produced in Fischer-Tropsch process. This process is a set of chemical reactions which converts the mixture of carbon monoxide and hydrogen into liquid hydrocarbons. These hydrocarbons are used as traditional diesel substitute. The source of the carbon could be the biomass. This type of biodiesel is not commonly used because of high costs of the producing facility and also high operation and maintenance costs.

FAME is commonly used as biodiesel. It is produced in the process of transesterification. This process is mixing the plant oils of animal fats with methanol in the presence of catalyst. Plant oils are mixed with methanol in the proportion of 5:1. Plant oils used in the production of FAME are from soy, rapeseed, and palm. The plant oil used for making of the biodiesel is the same plant oil use as food. Plant oil can be also used directly as a fuel in the diesel engines.

The main producers of the biofuels are Brazil and United States these counties specialize in the production of bioethanol. That is mainly because large maize plantations. The biodiesel producers are mainly European Union counties: Germany, France and Italy.

Bioethanol

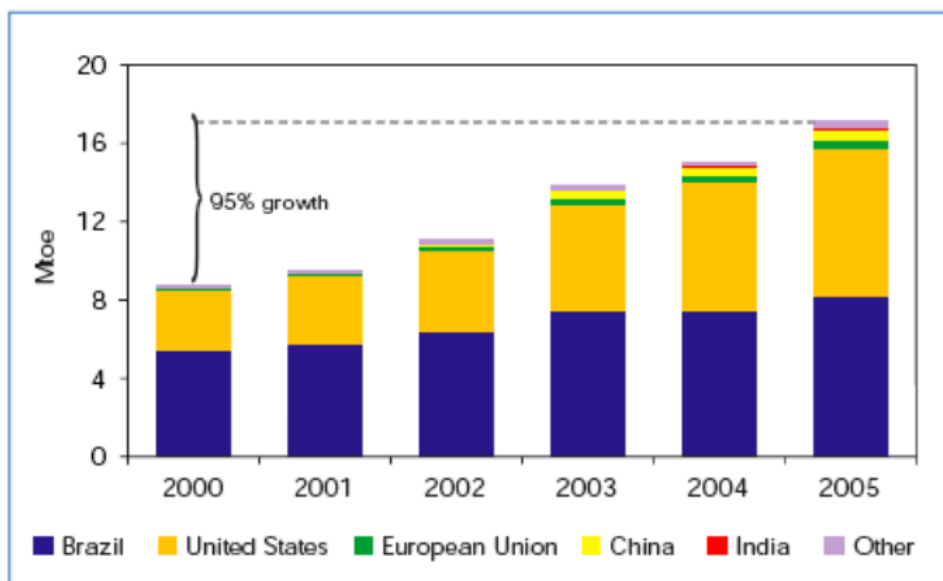


Figure 1-3 World Production of bioethanol (The IEA Energy Technology Essentials, 2010)

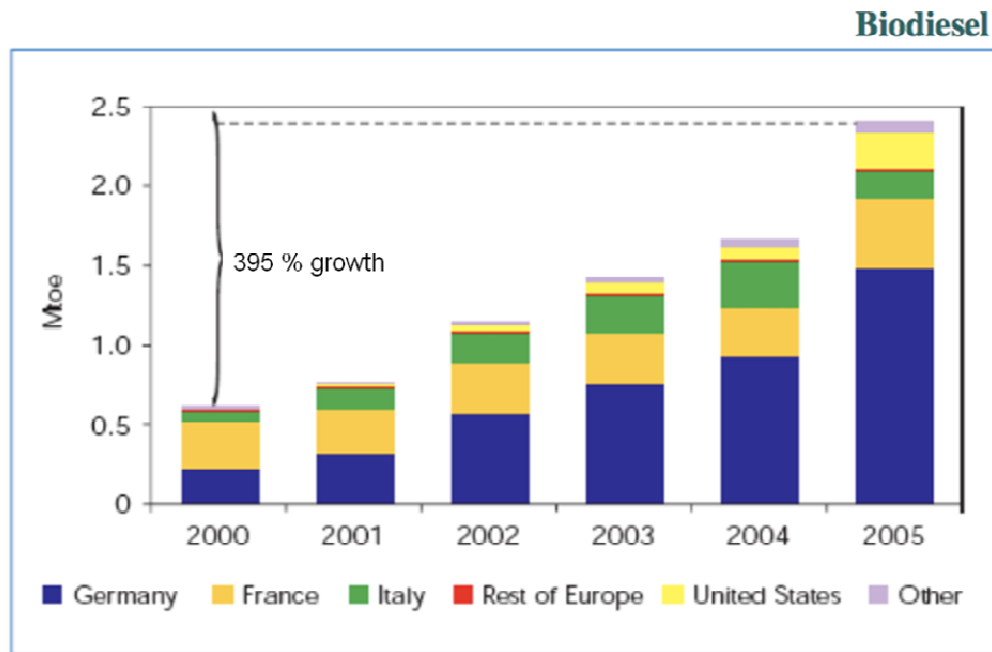


Figure 1-4 World Production of biodiesel (The IEA Energy Technology Essentials, 2010)

Major growth in the usage of biofuel occurred recently. The world production of bioethanol was 95% from year 2000 to 2005. Biodiesel grew even more rapidly, from 0,5 metric tonnes of oil equivalent to almost 2,5 Mtoe. That is almost 4 times.

Even though the biofuels are having their golden era there are many controversies about them. Firstly biofuels are made of the plants that can be used as food. As food is the most basic need of human kind, it is hard to decide that some of the food will be just burned as a fuel. This problem is also connected to the area used for biofuel plants which is competing with area for food crops. Area required for biofuel connected plants is also taken from the forests. The deforestation causes smaller CO₂ absorption. Second issue is the biofuels energy content. It is smaller than in traditional fossil fuels. It is necessary to remember that the whole process of making the biofuels also includes energy usage. Because of the transportation trucks, tractors and other farming equipment used in the process of making the biofuels the energy used in that process could be in some conditions even higher than energy content of biofuel produced. Thirdly it is acknowledged that biofuel are carbon neutral that is they realise the same amount of carbon that was captured by them during their growth. Therefore there are accounted as a renewable energy. Again if the whole life cycle assessment is made it appears that because of carbon realises during the process of production and transportation that is not true that biofuel are carbon neutral.

Because of all these problems the production and usage of biofuels should be managed in the sensible way. The law should support in the reasonable way, to make biofuels a feasible business but also encourage the producers of biofuel to improve the production processes. The law should also encourage users to use more efficient engines to fully use the potential of biofuel.

1.2 Goals of the study

The main goal of this study is to optimize the waste heat recovery system used in the bio-liquid power plant. Two configurations of the recovery system are made: the water steam system and the organic fluid system. These systems are optimized regarding the same objective functions.

Different working fluids of the organic cycle are examined.

After the optimization of the system the case study is performed to examine if the bio-liquid power plants with waste heat recovery system are practically applicable in Poland.

1.3 Background information

This master thesis project is being made as a part of the Útrás Engineering project. This company is designing a waste heat recovery system for a company in UK which is building biodiesel power plants.

Útrás Engineering supports me with the technical specification of the project (Blue NG Becton Project Documentation, 2010). That data is used as the input data in calculating and modelling phase of the thesis.

A literature study has been performed in order to examine work done on ORC optimization. This has been done in order to not redo the studies already made and also learn more about the used methods and software. Several studies have been done on this topic, the most interesting ones are referred to in this work.

Donghong (Donghong, Xuesheng, Zhen, & Jianming, 2008) made an optimization on ORC which ran on HFC-245fa working fluid which was driven by exhaust heat of gas turbine. The system power input was 100 kW. The inlet temperature was in a range of 337-377 °C. As a modelling tool Modelica/Dymola was used. As a working fluid data base, REFPROP 6.01 was used. The study focuses on the exergy analysis of the system.

Vaja (Vaja & Gambarotta, 2010) analyses the performance of ORC as a bottoming system for 12 cylinders 4 stroke super-charged natural gas fired engine. In that work three working fluids are checked: R11, R134a, benzene. Also different configurations of the system are analyzed:

- The simple cycle system thermally powered by engine exhaust gases
- The simple cycle system thermally powered by engine exhaust gases and engine cooling water
- The cycle with regeneration thermally powered by engine exhaust gases

The used software is not specified in this work.

Chacartegui (Chacartegui, Sanchez, Munoz, & Sanchez, 2009) studies the ORC as the bottoming cycle for large and medium recuperated gas turbines. The following organic fluids were considered: R113, R245, isobutene, toluene, cyclohexane and isopentane. The properties of these fluids are taken from EES software (Engineering Equation Solver) but the performance curves of each ORC component are extrapolated from data of similar equipment in real facilities.

1.4 Works Contribution

The following work is contributing to the studies mentioned above by studying ORC as a heat recovery system from 2 stroke bio-liquid stationary engines. That means that system is running in specific performance parameters. The chosen working fluids parameters have to fit to the specific range. The configurations additional to this works are examined. Thermal and economical optimization is combined.

Also there are no previous studies about possibilities of using bio-liquid power plants with waste heat recovery system in Poland. Performing such a study could lead to very interesting conclusions.

1.5 Research methodology

The mathematical model of Organic and Steam Rankin Cycle is made. The model is based on the energy and mass balance equations in all specific points of the cycle. The model also involves the pinch point analysis of the heat exchangers. The cycle is performed in steady state. The Engineering Equation Software (EES) is used for modelling. Specific working fluids are chosen on the basis of the temperature range of available wasted heat. Different configurations of the system are examined:

- The Simple ORC powered by both exhaust and HT cooling water heat
- Steam Rankine Cycle

Different objective functions of the optimization will be performed.

All of the issues mentioned above are defined in chapter 2.

1.6 Anticipated results

The results of the optimization will probably show that there is a trade-off between investment costs and the efficiency of the waste heat recovery system. It will be examined which of the system (steam or organic) is more feasible in this solution. Because of smaller investment cost the expected choice would be the steam system.

2 MODELING AND OPTIMIZATION OF THE WASTE HEAT RECOVERY SYSTEM

The optimization of the ORC is done for a real case study. The company “Blue-NG” is building a bioliquid diesel engine power plant in England. This system is planned to be linked with a gas expansion process and an absorption cooling system. In such a complex arrangement the fuel in the engine must be used very efficiently to turn back the investment in a reasonable time. Using of the bio-liquid as a fuel also requires a high efficiency of the combustion engine. That is mainly because of two issues:

- usually lower heating value than traditional oil (Table 2-1)
- fuel used during bio-diesel production and transportation of bio-liquid on site

These are the causes of implementing a waste heat recovery system in such an application. By reusing the heat from the engine the thermal efficiency of the whole process increases. Also the payback time of the investment is shorter thanks to the incomes from selling more electricity to the grid.

2.1 Waste heat recovery at bio-liquid power plants

In bio-liquid power plants there usually two sources of heat which can be utilized as electricity. First one is a low temperature source which is the heat from the cooling water for the engine. Second one is the high temperature heat recovered from the exhaust gases of the power plant. Depending on the size of the power plant one of them (heat of exhaust gases) or both are used in heat recovery system. The heat from the engine cooling water is used if the power plant is big enough to provide sufficient amount of power maintained in that heat. The cooling water coming out of the engine block also needs to have high enough temperature to be used in a waste heat recovery system.

2.1.1 Power Plant description

The “Blue – Ng” power plants are identical in their structure; they only differ in the power output of the engine. The current study focuses to optimize the waste heat recovery system for one of these applications. The project is located in Blackrod.

The diesel engine used is MAN 14K60MC-S with turbo compound system (TCS) and Organic Rankin Cycle as waste heat recovery system. The engine produces 28 MW of electricity from the combustion of the fuel.

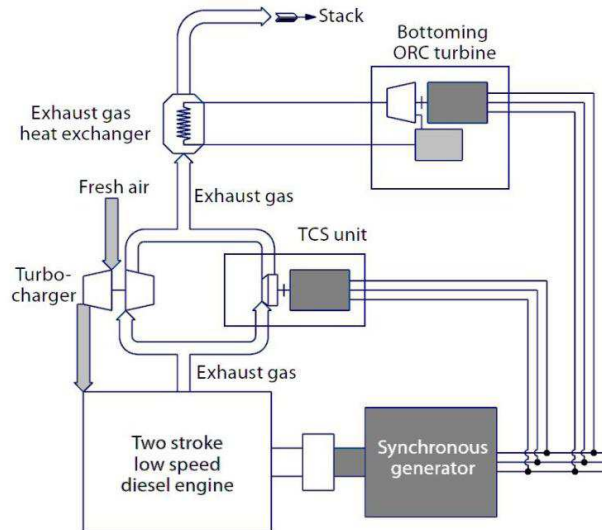


Figure 2-1 Scheme of the biodiesel engine with TCS and ORC units (MAN Diesel, 2010))

The figure above is made by the producer of the engine it includes only exhaust gases as a heat source for the ORC; actually in this case also the heat from the cooling of the engine block is used to run the ORC system.

There are three systems linked to the engine to absorb as much heat as possible from the motor. Two of them are used as a cooling of the engine. The diesel requires cooling from 117 °C to 58 °C. This process is divided into two parts: Low Temperature Cooling and High Temperature Cooling. The LT cooling has too small temperature range to be used directly in the heat recovery system, but it is used to preheat the natural gas entering the gas expander which is also a part of the system but is not the topic of this study. Therefore this issue is not be discussed. Pressurized water is used as a cooling medium in the HT cooling system. The cooling water is in the sub-cooled liquid region under the pressure of 10 bar. It is being heated from 60 to 112°C. The heating power of HT cooling heat exchanger is 9200 kW.

The third waste heat recovery system is collecting the energy from the exhaust gases of the engine. The exhaust gases have the temperature of 287°C. If the simple organic Rankine cycle is used as a heat recovery system than the working fluid of the HT cooling system is also pressurized water but it works on different pressure: 60 bar. The water is also kept in sub-cooled liquid conditions. Water is being heated from 112 to 260°C. The heating power of the exhaust gases heat exchanger is 10500 kW. The exhaust gas is cooled to 120 °C. It is examined if the exhaust gas can be cooled more for further heat extraction. This heat could be used for preheating the fuel of the diesel engine and for heating the offices.

The HT cooling system and the heat from the exhaust gases is used to run the ORC waste heat recovery system. The HT cooling as a pre-heater and exhaust gases as a evaporator. The ORC is optimized in a way to consume as much as possible of the heat from the mentioned heat exchangers. In case of the failure of the waste heat recovery system an auxiliary air cooling is mounted. The working fluid and the principle of the work of the ORC is discussed later in the work.

The important thing to mention is also that the ORC itself has a useful wasted heat that could be used for regeneration and increased efficiency of the cycle. This heat will be instead used for pre-heating the bio-liquid which is used in the diesel engine and heat the

offices. The power needed for those purposes is 3000 kW; needed temperature of the fuel is 85°C. It is examined if the extra heat has sufficient parameters to meet this demand. One of the design options suggests using this heat also to run the absorption cooling. This issue will not be discussed.

The condenser of the ORC is an air cooled heat exchanger.

2.1.2 Comparison of different fuels

As it was mentioned in the introduction the environmental aspects start to play a major role in the designing process of any system. It was chosen to run these power plants on bio-liquids. Because the project is located in the UK it needs to respect the Renewable Obligation which states that no more than 10 % fossil fuels are consumed by the engines i.e. (diesel, gasoline) except during start-up and shut-down. (The Department of Energy and Climate Change, 2010)

The fuel types which can be used in the engine are specified in the project. (Blue NG Becton Project Documentation, 2010) They are mainly straight vegetable oils or recovered vegetable oils.

Fuels that can be used in the “Blue-NG” power plants:

- Refined rapeseed oil
- Refined soya bean oil
- Crude soya bean oil
- Rape acid oil
- Palm oil
- Factory vegetable oil (FVO, off grade material)
- Recovered vegetable oil (RVO)

Table 2-1 shows the comparison of the properties of the different vegetable oils and fossil diesel. It can be observed that all plant oils have the lower heating value lower than fossil diesel. That means that to obtain the same power as from diesel fuel more vegetable oil needs to be combusted. That is why the efficiency of the combustion system plays such an important role.

Secondly it is visible in the table 2-1 that the plant oils are much more viscous than traditional diesel therefore and important issue is to investigate the possibilities of preheating the plant oils to decrease their viscosity and by that the performance during the combustion process. It is done later in this paper.

Rapeseed oil is chosen as a fuel used in the engine. That is because rapeseed is a most popular oil plant in Europe. Another reason is that the properties of plant oils do not vary a lot between each other and one is chosen as a representative.

*Table 2-1 Comparison of the properties of the traditional diesel and plant oils (A.K. Hossain, 2009) (*at 38°C)*

Fuel name	LHV [kJ/kg]	Density [kg/m ³]	Kinematic viscosity in 27 °C [cSt]	Flash point [°C]
Diesel	43350	815	4,3	45-60
Sunflower oil	39525	918	58,5	73
Cottonseed oil	39648	912	50,1	234
Rapeseed oil	37620	914	39,2	246
Peanut oil	39800	903	39,6*	271
Soybean oil	39623	914	65,4	254
Palm oil	36510	918	39,6*	267

The designer gives strict requirements for the properties of the fuel used in the diesel engine.

Table 2-2 Biofuel specification (Blue NG Becton Project Documentation, 2010)

Property	Unit	Value
Density at 15 °C	kg/m ³	1010
Kinematic viscosity at 100°C	cSt	55
Flash point	°C	≥60
Carbon Residue	%(m/m)	22
Ash	%(m/m)	0,15
Water	%(m/m)	1,0
Sulphur	%(m/m)	5,0
Vanadium	ppm(m/m)	600
Aluminium + Silicon	mg/kg	80
Sodium + Potasium	ppm(m/m)	200
Calcium	ppm(m/m)	200
Lead	ppm(m/m)	10
TAN (Total Acid Number)	mgKOH/g	<25
SAN (Strong Acid Number)	mgKOH/g	0

Selective Catalytic Reduction is used as an emission abatement system. This system purpose is to capture the NO_x which is formed during the combustion process of the fuel.

As it can be seen on figure 2-2 the process consists of few steps. Firstly the reducing agent (usually ammonia) is sprayed into the exhaust gases. Afterwards the mixture enters the reactor where in the presence of the catalyst NO_x is reduced to diatomic nitrogen and water and it is absorbed. The catalyst is a porous ceramic material. The process requires temperature at a level of least 290°C . Below that temperature the residence time of the ammonia should increase for the reaction to take place. The reaction between NO_x and ammonia is as follows:

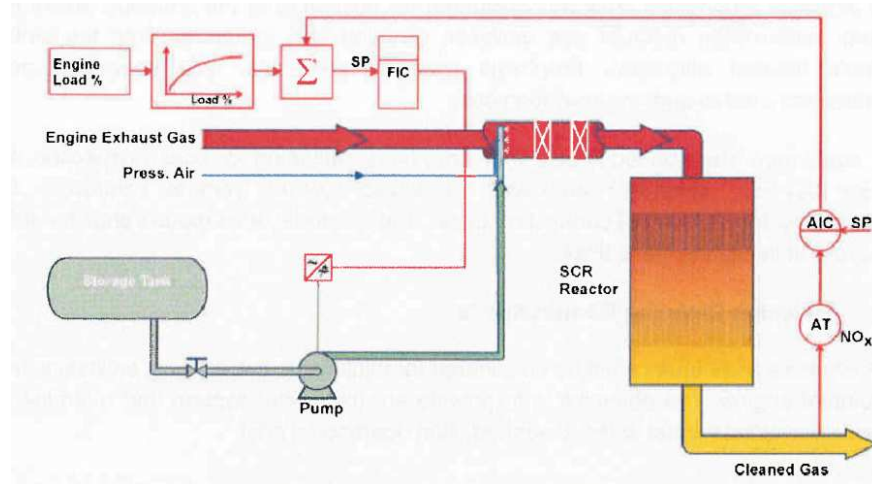
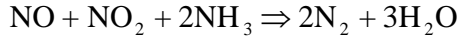


Figure 2-2 The Selective Catalytic Reduction System (Blue NG Becton Project Documentation, 2010)

The problems that could appear on this stage is the reaction between sulfur in the exhaust gases and water which is the product of the selective catalytic reduction. Sulfur acid could be forced causing major corrosion problems.

The literature research was done to examine the sulfur content of the rapeseed oil which is used as a fuel to the engine. Two studies were found. One of the studies measures the composition of the exhaust gases of rapeseed oil combustion. No sulfur is detected. (Pedersen, Ingemarsson, & Olsson, 1999) The other study measured the content of the sulfur in the oil made out of rapeseed cake. The content is 0,11%. (Ozcimen & Karoosmanoglu, 2004) That is a marginal value and would not do any harm to the system.

Another problem is high reactivity of the ammonia with copper. If the exhaust gas temperature is too low part of the ammonia does not react in the selective catalytic reduction process. Ammonia droplets are present in the fumes. If the heat exchangers are made of copper that corrosion problems will appear. It is important to make the heat exchangers from non ammonia reactive material.

After the exhaust treatment system the gases go to the heat exchanger which is the part of the heat recovery system.

2.2 The mathematical model of the ORC

For solving the optimization study the mathematical model of the application is done. The model is based on the energy balance equations and thermodynamic properties defined for each specific point of the Rankine cycle. The key issue is to make the optimized cycle the

function of the constraints of the system. That is done by setting the pinch point temperatures.

These limits can be defined as:

- The temperature and heat of the engine block cooling water
- The temperature of the exhaust gases
- Temperature and heat of the internal heating processes
- The temperature of the environment (surrounding air)

All the details of the system are shown in the table below.

Table 2-3 The Constraints of the System (Blue NG Becton Project Documentation, 2010)

Parameters	The Constrains			
	HT cooling	Exhaust gases	Internal heating	Environment
Temperature	117 °C	287°C	85 °C	9°C
Power	9200 kW	10500 kW	3000 kW	-

2.2.1 Simple ORC

Thermodynamic analysis

The Organic Rankine Cycle is a cycle of the heat engine. It consists of four thermodynamic processes:

- Isobaric heat addition (point a-c)
- Isentropic expansion (point c-d_s; including the efficiency of the turbine c-d)
- Isobaric heat rejection (point d-f)
- Isentropic compression (pumping, point f-a, pump efficiency included temp. a rises)

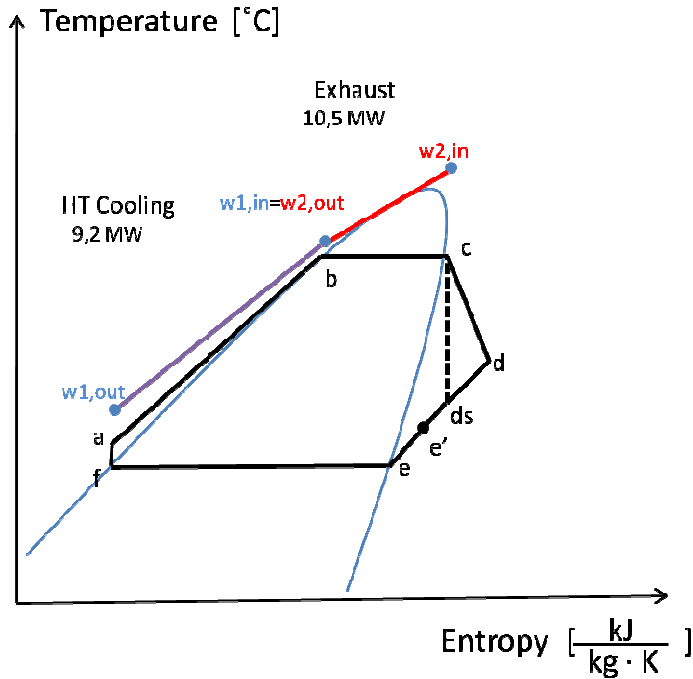


Figure 2-3 The Simple Organic Rankine Cycle on the $T(s)$ diagram

Firstly the abbreviations need to be explained for clearer understanding of the text. The letters of the alphabet are assigned for the points of the ORC. The abbreviations with “w” are assigned to the water side of the cycle.

Table 2-4 Abbreviations for the ORC waste heat recovery system

Abbreviation name	Description
a	After the pump, before preheater
b	After the preheater, before evaporator
c	After the evaporator, before turbine
d	After the turbine actual
ds	After the turbine ideal
e'	Probable beginning of the condenser
e	Organic fluid condensation start
f	After the condenser, before the pump
w1_in	Engine block cooling water inlet
w1_out	Engine block cooling water outlet
w2_in	Water heated with exhaust gases inlet
w2_out	Water heated with exhaust gases outlet
w3_in	Water for internal heating inlet
w3_out	Water for internal heating outlet

Engineering practice enforces to divide the above mentioned heat transfer thermodynamic processes into specific heat exchangers. Isobaric heat addition should in practice be divided into two heat exchangers. First one would be the pre-heater of the ORC. It is shown on Figure 2-3 as a line between points “a” and “b”. Starts just after the pump (f-a) and ends on the liquid saturation line at point “b”. This is the point where the first bubble of vapor appears; the evaporation starts. This process takes place at constant temperature at associated saturation pressure. Point “b” is also the start of second heat exchanger of the heat addition process. This exchanger is called the evaporator. The evaporation ends when all the fluid is vaporized and at that stage the vapor leaves the evaporator and enters the turbine. The expansion within the turbine occurs between points “c” and “d”. In the ideal case the expansion is from point “c” to “d_s” (turbine efficiency 100%) but in the real life the efficiency of the turbine is always smaller than 100% and some entropy is generated therefore the expansion is in practice from “c” to “d”. Due to the expansion the turbine performs useful work which is transformed to electricity in a generator to which it is connected through a shaft. All points a-b-b'-c are at the same pressure level called the pressure of vaporization (boiling pressure). There is a 0,5 bar pressure drop on each heat exchanger on the vaporization side of the cycle.

Points d(d_s)-e-e'-f are on the same pressure line. They are on the low pressure side of the cycle (condensation side). In point “d”, after the turbine, the steam is still is superheated.. As it was mentioned in paragraph 2.1.1 there is a need for 3000 kW of heat for preheating the fuel and heating the offices. This need can be satisfied by the superheated steam after the turbine, before the steam reaches the saturated vapor state at point “e”. The optimization will show what part of the d – e' line is required to sustain the system need. Point d-e mark the heat exchanger used to preheat the fuel and heat the offices. This heat need will be called “process heating”.

At state “e' ” steam is reaching the saturation vapor line. The condensation is started. The condenser is air cooled. Its working pressure is linked to the air parameters. The condensation ends at point “f” which is on the liquid saturation line. From there the working fluid is being pumped to the pre-heater where the whole cycle starts again. During the pumping (points f-a) the condensation pressure is increased to vaporization pressure. On the condensation side there are no pressure drops on the heat exchangers included.

As mentioned in the last chapter the mathematical model consists of energy balance equations on each heat exchanger. It also takes into account the isentropic efficiency of the turbine equation and the work of the pump equation. These equations and balances are shown below:

The Pre-heater:

$$Q_{ab} = m_{water1} \cdot C_p \cdot (T_{w1_in} - T_{w1_out}) = m_{wf} \cdot (h_b - h_a)$$

Where:

Q_{ab} – heating power of the heat exchanger [kW]

m_{water1} – mass flow rate of the water boiler 1 [kg/s]

C_p – specific heat of water at constant pressure [kJ/kgK]

T_{w1_in} – temperature of water entering the pre-heater [°C]

T_{w1_out} – temperature of water exiting the pre-heater [°C]

m_{wf} – mass flow rate of the working fluid of the ORC [kg/s]

h_a - specific enthalpy of the working fluid before the pre-heater [kJ/kg]

h_b - specific enthalpy of the working fluid after the pre-heater (before vaporizer) [kJ/kg]

The Vaporizer:

$$Q_{bc} = m_{water2} \cdot C_p \cdot (T_{w2_in} - T_{w2_out}) = m_{wf} \cdot (h_c - h_b)$$

Where:

Q_{bc} - heating power of the vaporizer [kW]

m_{water2} - mass flow rate of the water in boiler2 [kg/s]

T_{w2_in} – temperature of water entering the vaporizer [°C]

T_{w2_out} – temperature of water exiting the vaporizer [°C]

h_c - specific enthalpy of the working fluid after the vaporizer (before turbine) [kJ/kg]

The Turbine

$$\eta_{turbine} = \frac{h_c - h_d}{h_c - h_{ds}}$$

Where:

$\eta_{turbine}$ - isentropic efficiency of the turbine

h_d - specific enthalpy of the working fluid after the turbine [kJ/kg]

h_{ds} - specific enthalpy of the working fluid after the turbine after ideal expansion (efficiency 100%) [kJ/kg]

The Heat Exchanger for process heating

$$Q_{de} = m_{water3} \cdot C_p \cdot (T_{w3_out} - T_{w3_in}) = m_{wf} \cdot (h_d - h_e)$$

Where:

Q_{de} - heating power of the process heating heat exchanger [kW]

m_{water3} - mass flow rate of the water for process heating [kg/s]

T_{w3_in} - temperature of the water entering the process heating [°C]

T_{w3_out} - temperature of the water exiting the process heating [°C]

h_e - specific enthalpy of the working fluid after the process heating [kJ/kg]

The Condenser

$$Q_{ef} = m_{air} \cdot C_{p_air} \cdot (T_{c2} - T_{c1}) = m_{wf} \cdot (h_e - h_f)$$

Where:

Q_{ef} - heating power of the condenser [kW]

m_{air} - mass flow rate of the water for process heating [kg/s]

T_{c1} - temperature of the air entering the condenser [°C]

T_{c2} - temperature of the air exiting the condenser [°C]

h_f - specific enthalpy of the working fluid after the condenser [kJ/kg]

The Pump

$$W_{pump} = m_{wf} \cdot v_f \cdot (P_a - P_f)$$

Where:

W_{pump} - power of the pump [kW]

v_f - specific volume of the working fluid before the pump [m³/kg]

P_a - pressure after the pump [kPa]

P_f - pressure before the pump [kPa]

The left side of the balance equation is given and it is a constraint of the system. And so the know constraints of the set of the equation are:

Table 2-5 The Constrains of the system (Blue NG Becton Project Documentation, 2010)

Parameter	m_{water1}	T_{w1_in}	T_{w1_out}	Q_{ab}	m_{water2}	T_{w2_in}	T_{w2_out}
Value	43 kg/s	112°C	60°C	9300 kW	17 kg/s	260°C	112°C

Table 2-6 The Constrains of the system (continued) (Blue NG Becton Project Documentation, 2010)

Parameter	Q_{bc}	$\eta_{turbine}$	m_{water3}	T_{w3_in}	T_{w3_out}	Q_{de}
Value	10500 kW	0,8	32 kg/s	40 °C	85 °C	3000 kW

Pinch Point and Logarithmic Mean Temperature Difference Analysis

The pinch analysis in the heat exchanger is carried out in order to set the temperatures in the heat exchanger in a way to not cross the thermodynamic laws. Pinch point analysis requires setting the minimum temperature difference of the heat exchanger. (Bejan, Tsatsaronis, & Moran, 1996) This is shown on figure 2-4.

After setting the limit for the temperatures in the heat exchanger, LMTD analysis can be made. That analysis allows calculating the heat exchange area needed for the heat exchanger.

For the case studied in this thesis the minimum temperature difference needs to be implemented in two elements of the system: the vaporizer and the condenser.

After the examination of the vaporizer it becomes obvious that the minimum temperature difference should be defined between point “b” of the organic Rankine cycle which is actually the bubble point of the cycle and the corresponding to it point $T_{w2,out}$. During the optimization process the minimum temperature difference is changing but the important issue is to have the temperature $T_{w2,out}$ always higher than T_{bubble} .

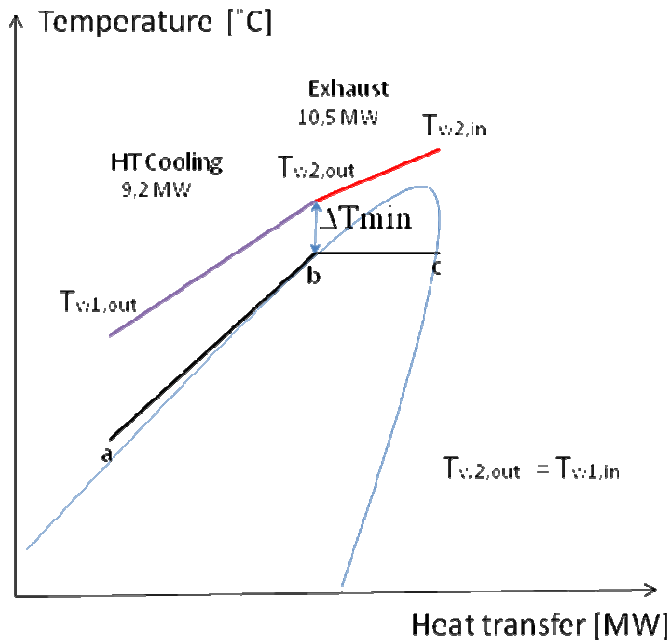


Figure 2-4 Pinch point set for the vaporizer

Optimization for the pinch point prevents the model from running in non thermodynamical way; makes sure it will not transfer heat from lower temperature source to higher temperature source. The ΔT_{min} is set point of highest risk for the violation to appear, if it is kept secure in that point it is secure in every point in the element.

Pinch point is also set in the condenser of the cycle. In this case it is set between the dew point temperature of cooling air and the corresponding to it temperature of the working fluid. The desuperheater is understood as a part of the condenser, but the heat will be used for internal heating purposes. This issue is investigated more in the results part of the thesis. This is shown on the figure 2-5.

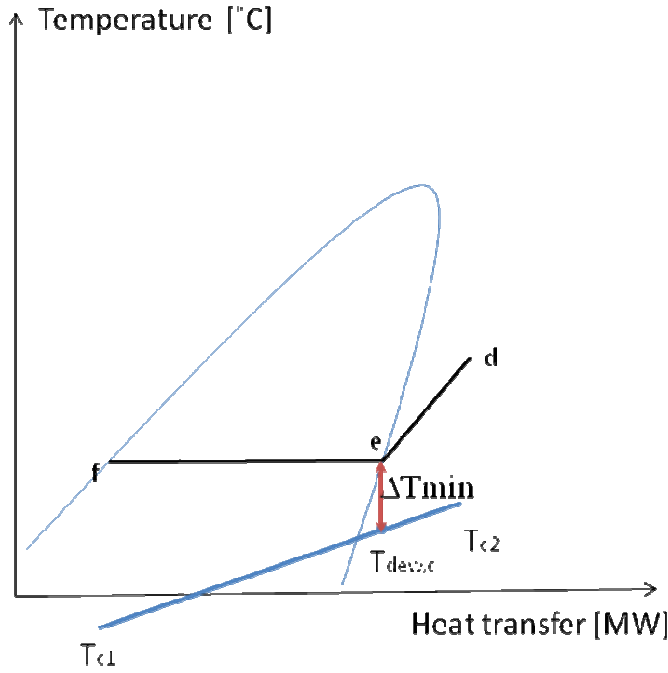


Figure 2-5 Pinch point set for the condenser

In the condenser it is important to keep the temperature of the working fluid always higher than the dew point temperature of the air.

After securing the temperatures of the heat exchangers by the pinch points the LMTD equations are made.

For the practical reasons the vaporizer is divided into two heat exchangers as mentioned above. The condenser is also divided into two heat exchangers: the de-super-heater and the condenser.

The logarithmic mean temperature difference analysis is based on forming the equations for the heat transferred in the heat exchanger. That value is a result of the multiplication of the total heat transfer coefficient U , heat exchange area A and the logarithmic mean temperature difference of the heat exchanger LMTD. Each heat exchanger has its own specific values of U , A and LMTD. (Bejan, Tsatsaronis, & Moran, 1996)

Value of the total heat coefficient is chosen before the optimization and it is kept constant. But choosing its value is not an easy issue. There could be made a whole new work on calculating the U value. In this study the U value is chosen from the literature and the way of its calculation is only mentioned. To calculate the total heat transfer coefficient the convective resistance from both sides of the heat exchanging element is needed. Also the conductance and the thickness of the material through which the heat is conducted needs to be know. For example the equation for the total heat transfer through the flat plate is shown below.

$$U = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta}{k} + \frac{1}{\alpha_2}}$$

Where:

U – total heat transfer coefficient [$\text{W}/\text{m}^2\text{K}$]

α_1 – internal convection coefficient [W/m²K]

δ – thickness of the heat exchanger wall [m]

k – conductance of heat exchanger wall [W/mK]

α_2 – external convection coefficient [W/m²K]

In short words it can be said that the total heat transfer coefficient is the inverse of the sum of all heat resistances of the heat exchanger. These resistances are linked in series.

Internal and external convective resistances vary depending on the heat transfer conditions on both sides of the heat exchanger. These conditions are dependent on: velocity, temperature and density of the working fluids. For calculating these resistances complex methods with using dimensionless numbers (Nusselt number, Reynolds number) are necessary.

On the other hand the conductance is only a function of the properties of the material of the heat exchanger. So it can be taken from the property tables.

The chosen total heat transfer coefficients vary depending on the configuration of the system.

So as mentioned above the U values chosen from the literature are as follow.

Table 2-7 Total heat transfer coefficients (Valdimarsson, 2010)

Configuration	U_{ab}	U_{bc}	U_{de}	U_{ef}
ORC	1000 [W/m ² K]	1200 [W/m ² K]	500 [W/m ² K]	100 [W/m ² K]
Steam	1000 [W/m ² K]	100 [W/m ² K]	100 [W/m ² K]	100 [W/m ² K]

The equations of heat transferred in the heat exchangers of the waste heat recovery system are as follow. (Bejan, Tsatsaronis, & Moran, 1996)

The Pre-heater

$$Q_{ab} = U_{ab} \cdot A_{ab} \cdot LMTD_{ab}$$

$$LMTD_{ab} = \frac{(T_{w2,out} - T_b) - (T_{w1,out} - T_a)}{\ln \left[\frac{(T_{w2,out} - T_b)}{(T_{w1,out} - T_a)} \right]}$$

The Evaporator

$$Q_{bc} = U_{bc} \cdot A_{bc} \cdot LMTD_{bc}$$

$$LMTD_{bc} = \frac{(T_{w2,in} - T_c) - (T_{w2,out} - T_b)}{\ln \left[\frac{(T_{w2,in} - T_c)}{(T_{w2,out} - T_b)} \right]}$$

The de-super-heater

$$Q_{de} = U_{de} \cdot A_{de} \cdot LMTD_{de}$$

$$LMTD_{de} = \frac{(T_d - T_{c2}) - (T_e - T_{dew,c})}{\ln \left[\frac{(T_d - T_{c2})}{(T_e - T_{dew,c})} \right]}$$

The condenser

$$Q_{ef} = U_{ef} \cdot A_{ef} \cdot LMTD_{ef}$$

$$LMTD_{ef} = \frac{(T_e - T_{dew,c}) - (T_f - T_{c1})}{\ln \left[\frac{(T_e - T_{dew,c})}{(T_f - T_{c1})} \right]}$$

During the optimization the values of the area, heat transferred and logarithmic mean temperature difference are obtained.

Practical solution

The equations mentioned in the previous chapter are the mathematical models of the real elements of the system. The scheme of the system is shown on figure 2-6. The important issue is that the preheater (ab) and the evaporator (bc) are not linked to the engine directly. The preheater is linked to the 10 bar water boiler which cools the engine block. The evaporator is linked to the 60 bar water boiler which is heated directly by the exhaust gas. The evaporator boiler is kept under such a high pressure to keep the water in the liquid state. The water is used as a coolant because of the environmental reasons. If heating oil was used for cooling the exhaust gas or if organic fluid was used directly than in case of leakage the potentially toxic and/or flammable substance would get into the stack causing major pollution. In the system aiming in being environmental friendly that is not a good situation. This is the reason for using such a high pressure water boiler. Water will flash into steam in case of the leakage not causing any environmental problems.

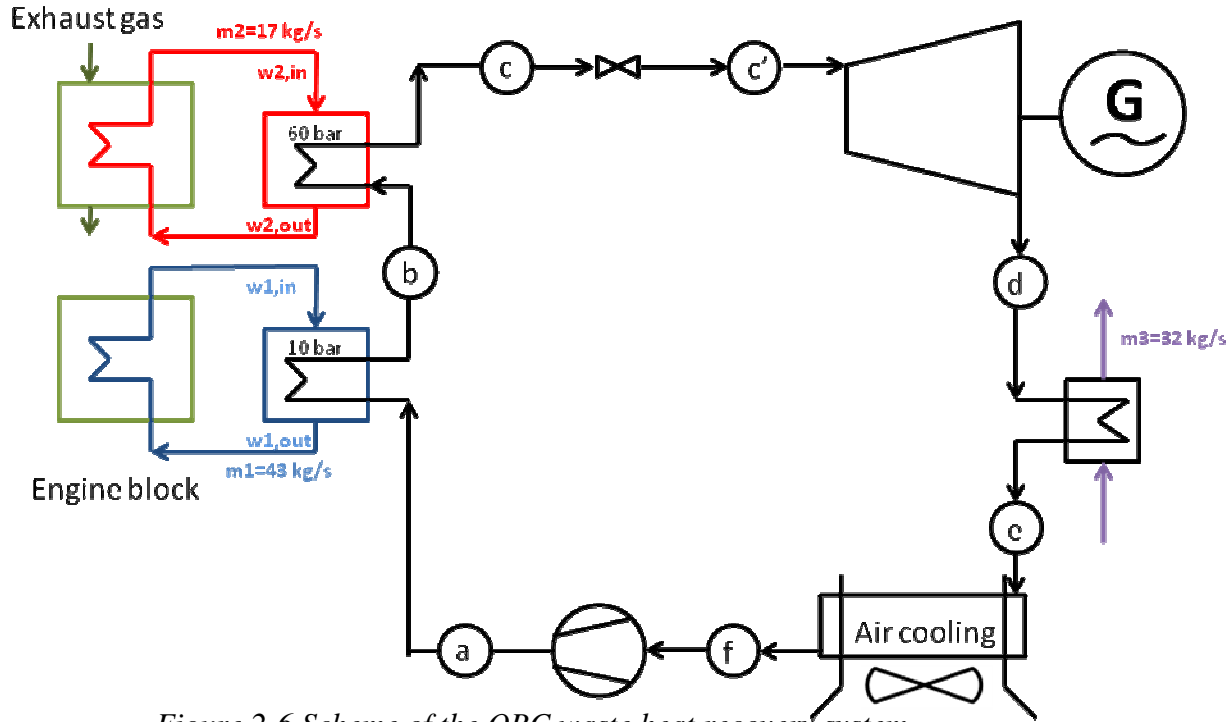


Figure 2-6 Scheme of the ORC waste heat recovery system

Another issue worth discussing is the heat exchanger (de) after the turbine collecting the heat from the superheated steam. During the optimization process it is examined if there is enough heat in the super heated steam to heat the fuel and the offices. If not there is no reason to use (de) heat exchanger than steam after the turbine goes directly to the condenser.

2.2.2 ORC with regeneration

The regeneration in the Organic Rankine Cycle is done by transferring the heat after the turbine to the pre-heater. This is done by the heat exchanger. The maximal value of heat that can be transferred is the difference of the enthalpy after the turbine and the enthalpy on the saturated vapor line of the ORC.

In this specific case maximal heat cannot be transferred because of the heat required for the process heating. So the heat that could be transferred is the difference of the enthalpy of the working after the process heating heat exchanger and the enthalpy on the saturated vapor line.

The energy balance of the regenerator is as follows:

$$m_{wf} \cdot (h_{a_reg} - h_a) = m_{wf} \cdot (h_e - h_{e'})$$

Where:

h_{a_reg} - enthalpy in the pre-heater after regeneration

h_e - enthalpy of the working fluid at the saturation vapor line

Because of using of the regeneration the energy balance equations of the preheater and the vaporizer are changing.

In the studied case was examined in first phases of the research. It was decided not to implement the regenerator to the system because it would not influence the system in a major way. But investigating the possibility for regeneration in this system would be a great study in the future.

2.2.3 Steam Cycle

Thermodynamic analysis

Steam cycle is a Rankine Cycle of the heat engine. From the point of view of thermodynamics it consists of the same processes as an Organic Rankine cycle. These processes are mentioned in the chapter about Simple Organic Rankine cycle. The major difference between these two cycles lies not in the processes themselves but in the way they progress. That is caused by differences in the properties of the water steam from organic fluids. The thermodynamic processes of Steam Rankine Cycle are as follows:

- Isobaric heat addition (point a-c)
- Isentropic expansion (point c-d_s; including the efficiency of the turbine c-d)
- Isobaric heat rejection (point d-f)
- Isentropic compression (pumping, point f-a)

The Steam Rankine cycle is shown on figure 2-7.

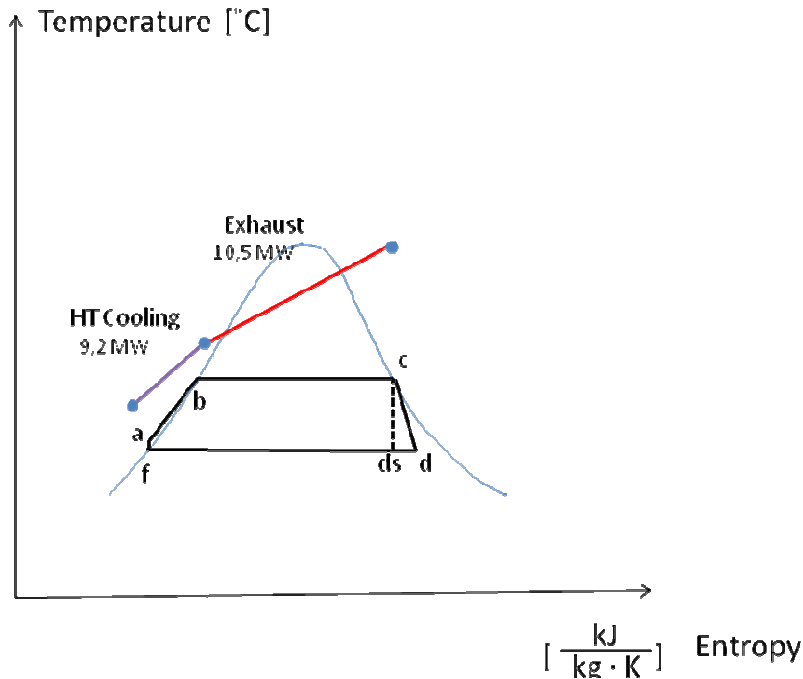


Figure 2-7 Steam Rankine Cycle on T(s) diagram

It can be seen that the only difference between Organic Rankine and Steam Rankine cycle is in the condensation process. The working fluid in the steam cycle lands in the wet steam region. That means that the steam is not superheated. And has no available heat to be used for heating purposes from that source. Therefore while using steam cycle another source of heat for heating the fuel and offices must be found. It is researched if the exhaust gas can be cooled more for extracting more heat out off it. It is examined in chapter 2.8.2.

Another issue is that there cannot be too much liquid in the turbine because that decreases the vitality of the turbine. To control the level of the liquid in the turbine at the sufficient level it was assumed that during the optimization the quality of the wet steam exiting the turbine cannot be lower than 0,85.

Practical solution

From the practical point of view there are two major differences between Simple Organic Rankine Cycle and Steam Cycle.

Firstly the evaporator of the system is mounted directly on the stack. So the heat exchange in the evaporator is between gas and steam. That simplifies the system; one less heat exchanger is used.

Secondly the turbine in the steam cycle has to deal with much less dense fluid so it has to be larger in size than ORC turbine to produce the same amount of electricity. The ORC cycle deals with dense organic fluid. The research in turbine prices showed that even though the steam turbine is larger its price per kW is smaller than the organic turbine.

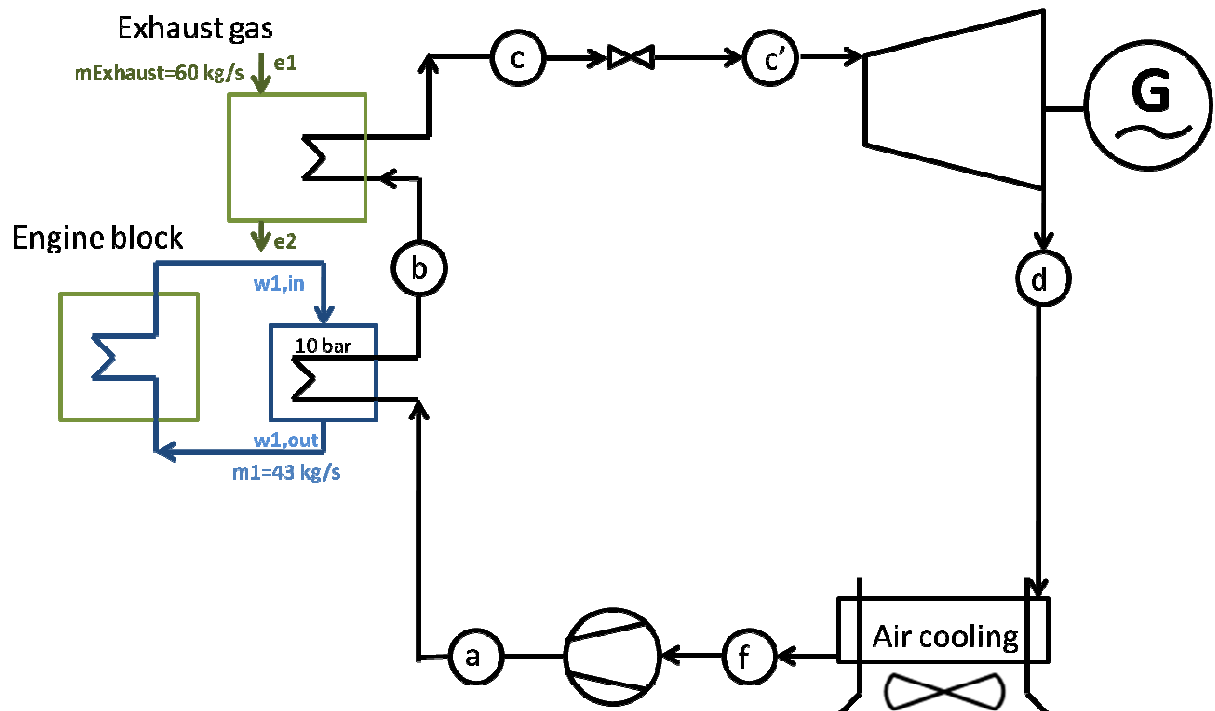


Figure 2-8 Scheme of the steam waste heat recovery system

2.2.4 Different cooling conditions

Air cooling was chosen to be used in this application. That solution is less complex than water cooling. The cooling can also take place at lower temperatures than in water cooling case because water freezes at 0°C . The problem with air cooling is the variations of the air temperature during the year. That has an influence on the condenser parameters and can affect the electricity production of the waste heat recovery system. Therefore it is necessary to examine the work of the optimized system at different temperature conditions during the year. The temperature data is presented on the duration curve. This curve shows for how many hours a year the single temperature appears. The flatter the curve is the more stable the temperature conditions.

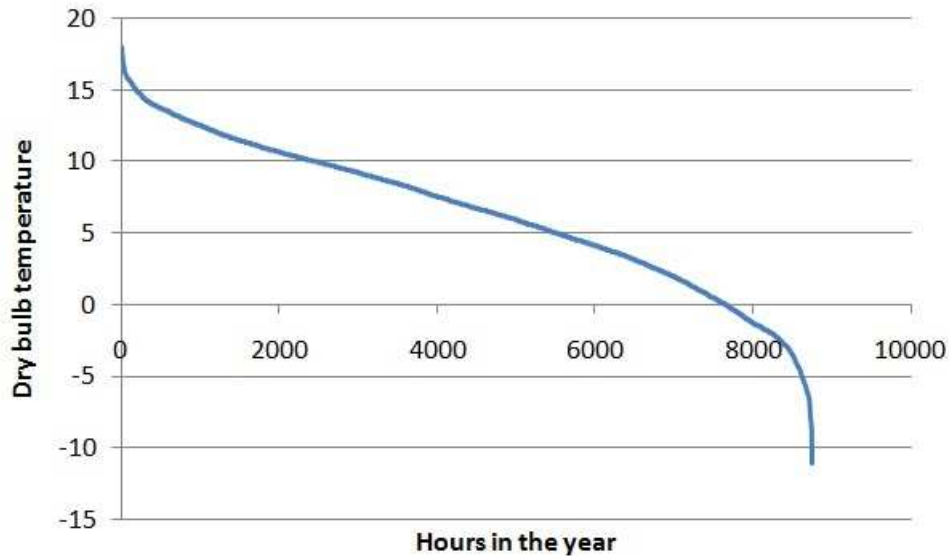


Figure 2-9 Duration of the dry bulb temperature in London (U.S Department of Energy, 2010)

Fifteen different single temperatures from the duration curve are chosen for the evaluation. The results are shown on figures 2-10 and 2-11.

As it can be seen on figure 2-10 while the temperature of cooling air is dropping the system improves: the net work output. Also performance of the cycle improves from the economical point of view which is visible in figure 2-11. The maximal obtained parameters are for the lowest possible cooling air temperature. And comparing them to the objective function values at the highest temperatures; the 40-50% growth in their value is observed.

In studying the waste heat recovery system air temperature was assumed at constants level: 9°C. As it can be seen the cooling air temperature has a major influence on the performance of the system.

This influence is made by changing the condensation pressure with the temperature of cooling air. While the temperature drops the condensation pressure drops too that enables the expansion in the turbine to go further. Therefore more work is produced and because of that net present value grows.

Important issue to notice is that isopentane achieves the highest performance during whole of the year. Economically the steam cycle is very close in performance to the isopentane even though it produces less electricity than isobutane during most of the year.

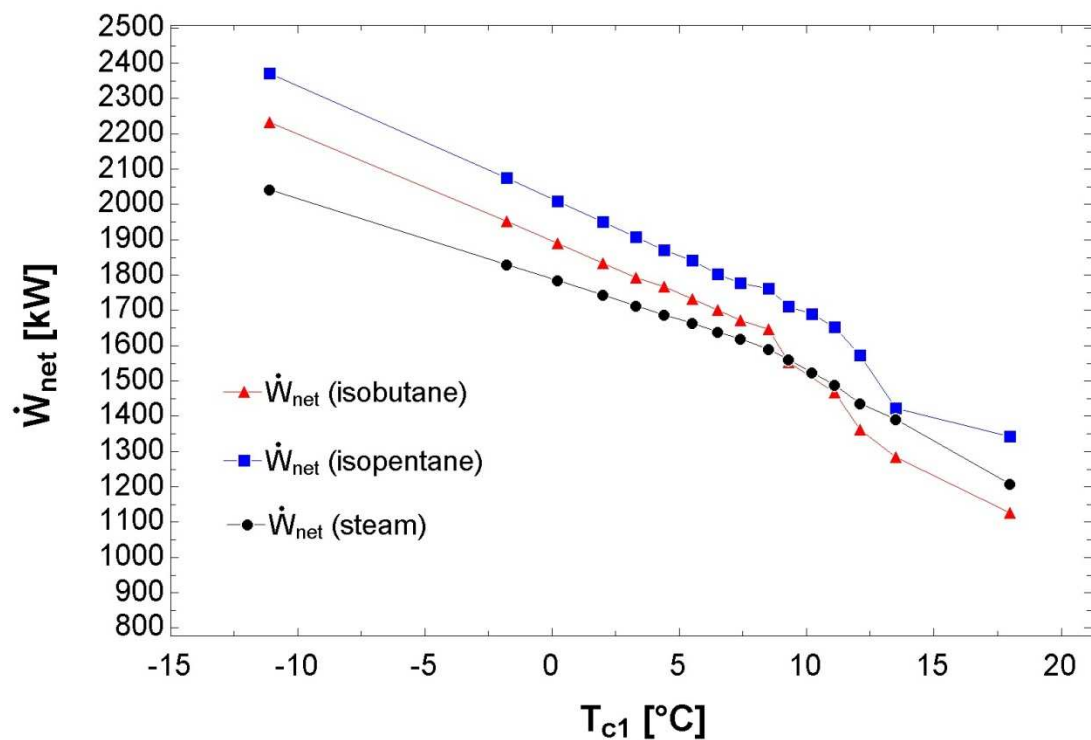


Figure 2-10 Changes in recovery system parameters in different cooling temperature conditions (Grabiński, 2011)

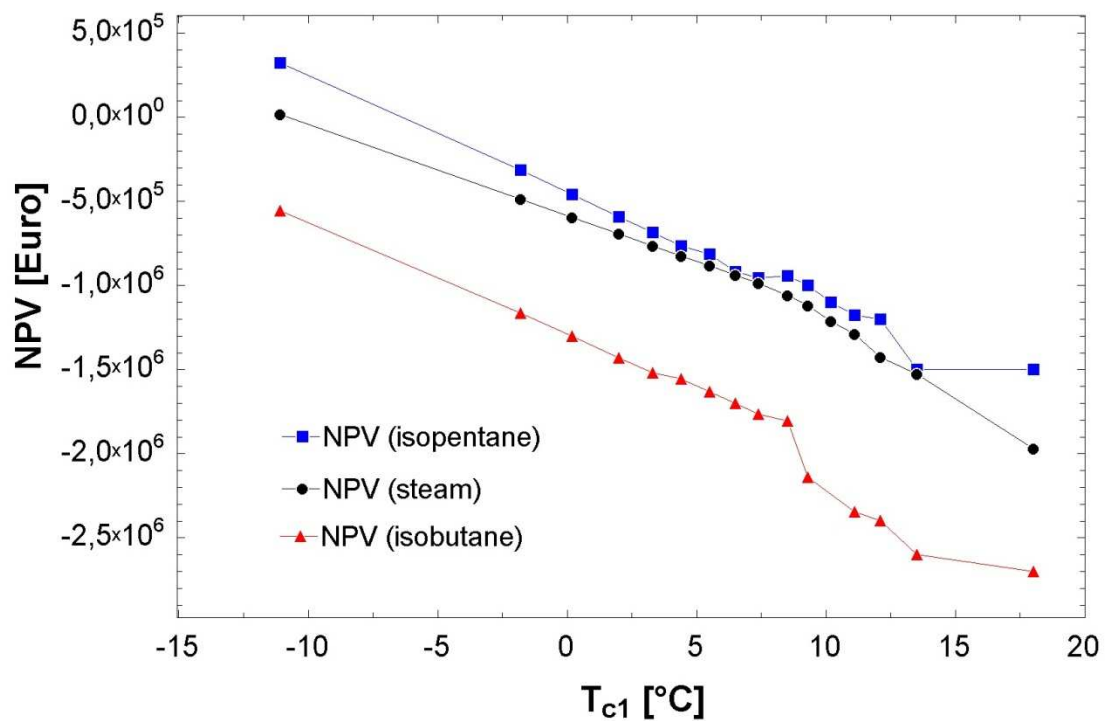


Figure 2-11 Changes in net present value after 5 years with different cooling temperatures conditions (Grabiński, 2011)

2.3 The working fluids of the system

In this paper the following organic working fluids are examined: isobutene, isopentane, steam. Isobutane and isopentane are used as working fluids in Organic Rankine Cycle. Steam is used as a working fluid in the Steam Rankine Cycle.

2.3.1 Isobutene R600a

R600A is a natural substance; it is formed during the biodegradation of mainly plants (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009).

General Characteristics

Table 2-8 Basic Properties of Isobutane(Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)

Parameter	Value	Unit
Chemical equation	$\text{CH}(\text{CH}_3)_3$	-
Molar mass	58,13	kg/kmol
Critical temperature	135,0	°C
Critical pressure	36,45	bar
Specific critical volume	0,004526	m ³ /kg
Ozone depletion potential ODP	0	-
Global warming potential GWP	3	-
Flammability	flammable	-
Colour	colourless	-
Smell	sweetish	-

Solubility with oils

It is recommended to use low viscous additive mineral oils with R600a (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009). Isobutene dissolves easily in these oils and leads to a further decrease in oils viscosity.

Solubility with water

Solubility with water at atmospheric pressure is 54 mg/l (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009).

Reacting with metals

Compatible with all metals used as piping in the industrial solutions (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009). However isobutene is never pure, there is a probability that the corrosion will appear.

It is visible on the isobutene property figure Figure 2-12 that it has specifically tilted the saturation line. That property of this fluid is used in the Simple Organic Rankine Cycle. The steam after the turbine is still superheated therefore its heat can be utilized further. Another positive issue of such a shape of the saturation line is that there is no

liquid in the turbine therefore it could work for a long time. Droplets of liquid are deadly for the turbines.

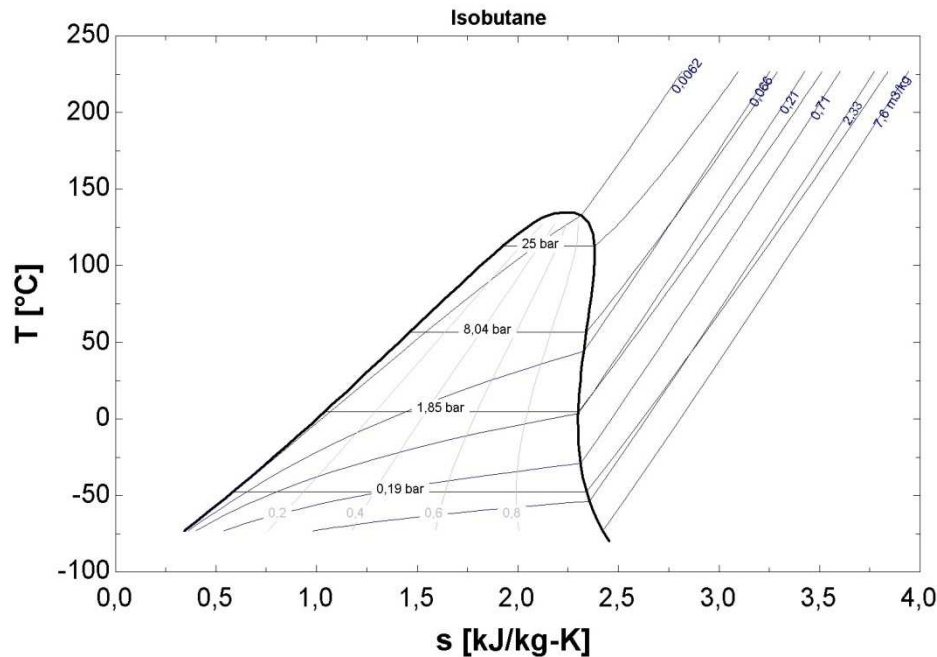


Figure 2-12 Isobutene T-s diagram

Reacting with the plastics

Isobutene can work with most of the plastics despite natural rubber, silicon and EPDM rubber. (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)

Combustibility and explosiveness

Combustibility and the explosiveness are the problems with isobutene. Minimum ignition temperature is 460 °C, lower explosive limit is 1,8% and the upper explosive limit is 8,5%. Isobutene has A3 class as to the ASRAE standards. That means it is inflammable and nontoxic. (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)

Reacting with human organism

Can cause burns if the stream of gas is directed on uncovered skin or eyes. Breathing R600a in high concentrations could lead to the strangulation. At first breathing causes disability of movement, then loss of consciousness and then after further breathing death. The symptoms of inhalation of isobutene are headache, nausea, salivation, vomiting.

2.3.2 Isopentane

Isopentane has very similar properties to isobutane. The main difference is the higher temperature of the critical point. That is an important issue which distinguishes between these two fluids and influences the results of the optimization process.

General Characteristics

Table 2-9 Basic Properties of Isopentane (Ruciński, Pluta, Rusowicz, & Grzebielec, 2009)

Parameter	Value	Unit
Chemical equation	$(\text{CH}_3)_2\text{-CH-CH}_2\text{-CH}_3$	-
Molar mass	72,2	kg/kmol
Critical temperature	187,2	°C
Critical pressure	33,7	bar
Specific critical volume	0,004237	m ³ /kg
Ozone depletion potential ODP	0	-
Global warming potential GWP	11	-
Flammability	Highly flammable	-
Colour	colourless	-
Smell	sweetish	-

Figure 2-13 shows the T(s) diagram of isopentane. Analogical conclusions as for isobutene are made.

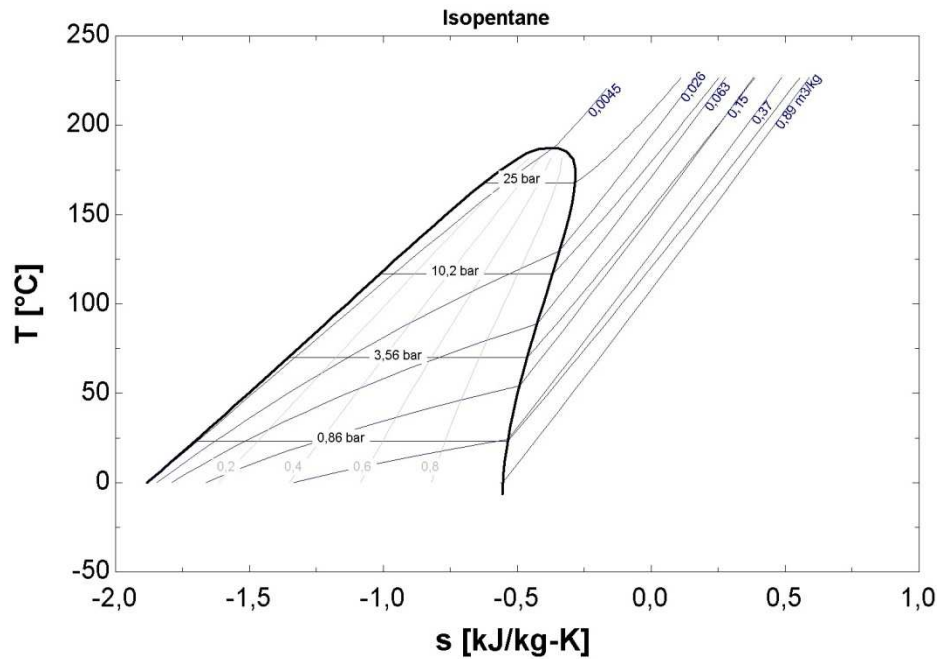


Figure 2-13 Isopentene T-s diagram

2.3.3 Steam

Steam is characterized by much higher critical points than organic fluids. But its specific critical volume even at 10 times bigger pressure is 10 times smaller than in Organic fluids.

Table 2-10 Basic properties of Steam

Parameter	Value	Unit
Chemical equation	H ₂ O	-
Molar mass	18	kg/kmol
Critical temperature	374	°C
Critical pressure	220,6	bar
Specific critical volume	0,003106	m ³ /kg
Ozone depletion potential ODP	0	-
Global warming potential GWP	-	-
Flammability	Not flammable	-
Colour	colourless	-
Smell	No smell	-

There is a substantial difference in the water T(s) diagram and organic fluids figures. The saturation line for water has an opposite tilt as compared to the organic fluids. It is like a symmetrical hill. Therefore after exiting the turbine in the case of Simple Steam cycle (no superheating) the steam is saturated with quality less than one. That means that there is no possibility for regeneration in the steam cycle.

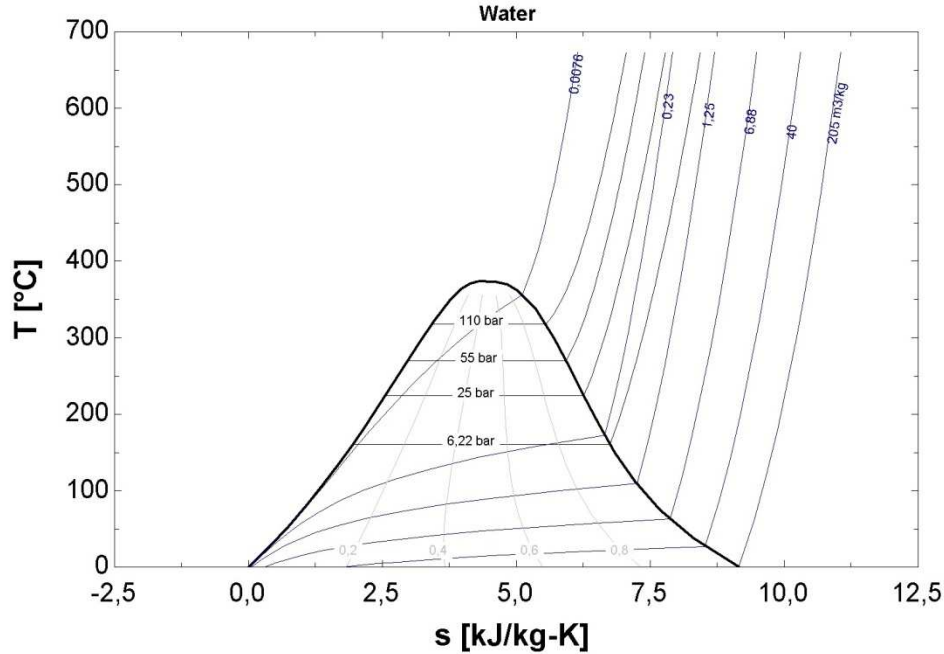


Figure 2-14 Water T-s diagram

2.4 Definition of objective functions of the ORC optimization

The optimization can be realized by using different objective functions. These functions can have engineering, economical or ecological character. Depending on the chosen objective function the optimized cycles can differ. In this work the following objective functions will be used: net work of the ORC system, exergy efficiency and net present value of the investment of the ORC. All the objective functions will be maximized.

- *Net work of the ORC system*

This objective function is defined as follows:

$$W_{net} = W_{turbine} - W_{pump}$$

Where

$$W_{turbine} = m_{wf} \cdot (h_d - h_c)$$

m_{wf} - mss flow rate of the working fluid [kg/s]

h_c - specific enthalpy before the turbine

h_d - specific enthalpy after the turbine

- *Exergy Efficiency*

Exergy efficiency, which is often called Second law efficiency, is an indication of how well the system performs as compared to the theoretical maximum. The Exergy efficiency is used as an objective function to optimize the system in the pure thermo-dynamical way, maximizing the efficiency but not looking on the costs side of the optimization. (Kotas, 1985)

$$\eta_{Exergy} = \frac{W_{net}}{(E_{Exhaust_in} - E_{Exhaust_out}) + (E_{HT_cooling_in} - E_{HT_cooling_out})}$$

Exergy efficiency is introduced in details in chapter 2.5.1

- *Net Present Value*

The net present value is used to show the different view on the optimization. This parameter only sees the economic side of the system implementation. The NPV is a powerful tool also because it combines the investment cost analysis with the benefits analysis. And actually it is a function on the net work of the ORC.

$$NPV = -I_{total} + \frac{B}{d} \left(1 - \frac{1}{(1+d)^n} \right)$$

Net present value is introduced in details in chapter 2.6.2

2.5 Exergy analysis of the ORC system

The analysis of the waste heat recovery system requires measuring the usefulness of the wasted heat. It is also important to know is the waste heat utilized in the efficient way. These are the reasons for defining the notion of exergy and providing the exergy analysis of the system.

2.5.1 Exergy introduction

Exergy is defined as a useful work done by one system to reach the equilibrium with the other system. In thermodynamics that interaction is between the studied system and the system that surrounds that system. The surrounding system is called the environment. The properties of the environment are so called dead state properties and they by definition are uniform. That means that if the system reaches the dead state parameters than there is no useful work to be done. In other words if the system and the environment are in the equilibrium state no work can be done (the system has no exergetic value). (Bejan, Tsatsaronis, & Moran, 1996)

There are four different types of exergy: E_{PH} physical exergy, E_{CH} chemical exergy, E_{KN} kinetic exergy, E_{PT} potential exergy. Their sum gives E_{Total} total exergy:

$$E_{Total} = E_{PH} + E_{CH} + E_{KN} + E_{PT}$$

The kinetic exergy and the potential exergy in the studied case can be neglected. That is because their value influences the total exergy in a marginal way. There are no chemical reactions which could lead to the exergy generation so that the chemical exergy is also not taken into account. Therefore the only exergy component is the physical exergy. Physical exergy can be defined as follows:

$$E_{PH} = (H - H_0) - T_0(S - S_0)$$

Where:

h - enthalpy of the system state [kJ]

h_0 - enthalpy of the dead state [kJ]

T_0 - temperature of the dead state [K]

s - entropy of the system [kJ/K]

s_0 - entropy of the dead state [kJ/K]

For practical purposes the notion of specific exergy is used. All the parameters of the equation above are presented per mass unit (kg).

$$e_{PH} = (h - h_0) - T_0(s - s_0)$$

Where:

h - specific enthalpy of the system state [kJ/kg]

h_0 - specific enthalpy of the dead state [kJ/kg]

T_0 - temperature of the dead state [K]

s - specific entropy of the system [kJ/kgK]

s_0 - specific entropy of the dead state [kJ/kgK]

After introducing the exergy notion the analysis can begin. At first the boundaries of the system will be defined. The system is shown on figure 2-6 for Organic Rankine Cycle and figure 2-8 for Steam Cycle.

In the case where the waste heat recovery system runs in subcritical conditions the elements of the system are as mentioned in chapter 2.2.1.

For each point of the ORC cycle the exergy is defined. Also the parameters of the dead state are introduced:

$$T_0 = 9 \text{ }^{\circ}\text{C}$$

$$p_0 = 1 \text{ bar}$$

$$s_0 = s(T_0, p_0)$$

$$h_0 = h(T_0, p_0)$$

The fluid of the environment is air.

The pre-heater

$$e_a = (h_a - h_0) - T_0(s_a - s_0) - \text{exergy before the pre-heater}$$

$$e_b = (h_b - h_0) - T_0(s_b - s_0) - \text{exergy after the pre-heater}$$

The vaporizer

$$e_c = (h_c - h_0) - T_0(s_c - s_0) - \text{exergy after the vaporizer}$$

Exergy before the vaporizer is the same as exergy after the pre-heater.

The turbine

$$e_d = (h_d - h_0) - T_0(s_d - s_0) - \text{exergy after the turbine}$$

Exergy before the turbine is the same as exergy after the vaporizer.

The internal heating

$$e_e = (h_e - h_0) - T_0(s_e - s_0) - \text{exergy after the internal heating heat exchanger}$$

Exergy before the internal heating heat exchanger is the same as exergy after the turbine.

The condenser

$$e_f = (h_f - h_0) - T_0(s_f - s_0) - \text{exergy after the condenser}$$

Exergy before condenser is the same as exergy after the internal heating.

The pump

The exergy before the pump is e_f the exergy after the pump is e_a

Two types of exergy analysis will be produces in this work. Firstly the exergy efficiency of the ORC will be calculated. It will be one of the objective functions of the optimization. Secondly the exergy destruction will be calculated for each element of the ORC system. For that part of the study also the exergy values of the steam side of the cycle need to be

defined. It will be recognized which element of the system causes most losses. By optimizing the exergy efficiency the losses should be minimized.

Exergy efficiency is defined as following:

$$\eta_{Exergy} = \frac{W_{net}}{(E_{Exhaust_in} - E_{Exhaust_out}) + (E_{HT_cooling_in} - E_{HT_cooling_out})}$$

Where:

W_{net} – net work of the system defined in the chapter 2.4

$E_{Exhaust_in}$ – exergy of the exhaust boiler water in

$E_{HT_cooling_in}$ – exergy of the HT cooling boiler in

$E_{Exhaust_out}$ – exergy of the exhaust boiler water out

$E_{HT_cooling_out}$ – exergy of the HT cooling boiler out

For the analysis of the exergy destruction also the exergy values of the fluids on the opposite to the working fluid side of the heat exchangers needs to be defined.

The exergy of the exhaust boiler water entering the vaporizer:

$$E_{Exhaust_in} = m_{water2} \cdot ((h_{Exhaust_in} - h_0) - T_0(s_{Exhaust_in} - s_0))$$

The exergy of the HT cooling entering the pre-heater:

$$E_{HT_cooling_in} = m_{water1} \cdot ((h_{HT_cooling_in} - h_0) - T_0(s_{HT_cooling_in} - s_0))$$

The exergy of the exhaust boiler water exiting the vaporizer:

$$E_{Exhaust_out} = m_{water2} \cdot ((h_{Exhaust_out} - h_0) - T_0(s_{Exhaust_out} - s_0))$$

The exergy of the HT cooling exiting the pre-heater:

$$E_{HT_cooling_out} = m_{water1} \cdot ((h_{HT_cooling_out} - h_0) - T_0(s_{HT_cooling_out} - s_0))$$

The exergy of the internal heating water entering the heat exchanger:

$$E_{Internal_heating_in} = m_{water3} \cdot ((h_{Internal_heating_in} - h_0) - T_0(s_{Internal_heating_in} - s_0))$$

The exergy of the internal heating water exiting the heat exchanger:

$$E_{Internal_heating_out} = m_{water3} \cdot ((h_{Internal_heating_out} - h_0) - T_0(s_{Internal_heating_out} - s_0))$$

The exergy of the condenser cooling air entering the condenser:

$$E_{air_in} = m_{air} \cdot ((h_{air_in} - h_0) - T_0(s_{air_in} - s_0))$$

The exergy of the condenser cooling air exiting the condenser:

$$E_{air_out} = m_{air} \cdot ((h_{air_out} - h_0) - T_0(s_{air_out} - s_0))$$

2.5.2 Exergy destruction definition

When exergy is defined in every point of the ORC cycle then it is possible how much useful work is lost due to the processes in the cycle. To obtain the exergy losses in the elements of the ORC cycle the exergy balance of each one of the elements should be carried out. (Bejan, Tsatsaronis, & Moran, 1996)

- Exergy balance – Pre-heater

$$E_{loss_preheater} = m_{wf} \cdot (e_a - e_b) + E_{HT_cooling_in} - E_{HT_cooling_out}$$

- Exergy balance – Vaporizer

$$E_{loss_vaporizer} = m_{wf} \cdot (e_b - e_c) + E_{Exhaust_in} - E_{Exhaust_out}$$

- Exergy balance – Turbine

$$E_{loss_turbine} = m_{wf} \cdot (e_c - e_d) - W_{turbine}$$

- Exergy balance – Internal heating

$$E_{loss_internal_heating} = m_{wf} \cdot (e_d - e_e) + E_{Internal_heating_in} - E_{Internal_heating_out}$$

- Exergy balance – Condenser

$$E_{loss_condenser} = m_{wf} \cdot (e_e - e_f) + E_{air_in} - E_{air_out}$$

- Exergy balance – Pump

$$E_{loss_pump} = m_{wf} \cdot (e_f - e_a) + W_{pump}$$

2.6 Economical analysis of the ORC system

To evaluate the waste heat recovery system from the economical point of view, the analysis will be done. The analysis will not only take into account the investment costs of the system but also the income from selling the recovered electricity. To provide such a analysis the net present value of the investment is calculated. Net present value combines the investment cost analysis with the assumed future income from selling the produced

electricity. It takes also into consideration the changing values of the money by using the discount rate.

Net present value will be used as one of the objective functions of the optimization.

2.6.1 Investment cost of WHR system

The investment cost of the waste heat recovery system is a sum of costs of all of its components. These components are as mentioned in previous analysis: pre-heater, vaporizer, turbine, internal heating heat exchanger, condenser and pump. The costs of piping will be also added. The costs of all heat exchangers are based on the heat exchange area and the price of the heat exchanger per m². The costs of the turbine and the pump are based on their power and the price per kW.

$$I_{total} = C_{preheater} + C_{vaporizer} + C_{condenser} + C_{turbine} + C_{pump} + C_{piping}$$

$$C_{preheater} = c_{preheater} \cdot A_{preheater}$$

$$C_{vaporizer} = c_{vaporizer} \cdot A_{vaporizer}$$

$$C_{condenser} = c_{condenser} \cdot A_{condenser}$$

$$C_{turbine} = c_{turbine} \cdot W_{turbine}$$

$$C_{pump} = c_{pump} \cdot W_{pump}$$

Where:

I_{total} – total investment cost

C_i – cost of the element in Euro

c_i – price of the element in Euro/m² for the preheater, evaporator and the condenser and in Euro/kW for the pump and the turbine

A_i – heat exchanger area

$W_{turbine}$ – nominal power of the turbine in kW

W_{pump} – nominal power of the pump in kW

The prices are taken from Chemical Engineering Handbook by Perry. Their values are increased by the inflation of 3% from the base year 1997.

Table 2-11 Prices of the elements of the ORC system (Perry, 1997)

Element of the system	Price/unit
Pre-heater	400 Euro/m ²
Evaporator	400 Euro/m ²
Internal heating heat exchanger	400 Euro/m ²
Condenser	400 Euro/m ²
Turbine	600 Euro/kW
Pump	600 Euro/kW

The areas of the heat exchangers and the powers of the turbine and the pump will be optimized. Therefore the investment cost will also be optimized.

2.6.2 NPV analysis

Net present value analysis which is done in this study is showing how well the investing in the waste heat recovery system does pay.

The formula for the Net Present Value is as follows:

$$NPV = -I_{total} + \sum_{i=0}^n \frac{B}{(1+d)^n}$$

Where

I_{total} – total investment cost

B – yearly benefits of the ORC system [Euro]

d – discount rate

n – number of years

The total investment cost was defined in the previous chapter. The yearly benefits of the ORC system come from selling the produced electricity. The price of electricity is assumed 9 € cents.

In the model the simplified equation for the NPV sum after “n” years will be used.

$$NPV = -I_{total} + \frac{B}{d} \left(1 - \frac{1}{(1+d)^n} \right)$$

The value of the discount rate is taken from analogical case studies. It has assumed value of 5%. (International Energy Agency, 2011)

2.7 EES Model Explanation

In the study Engineering Equation Solver is used to optimize the waste heat recovery system.

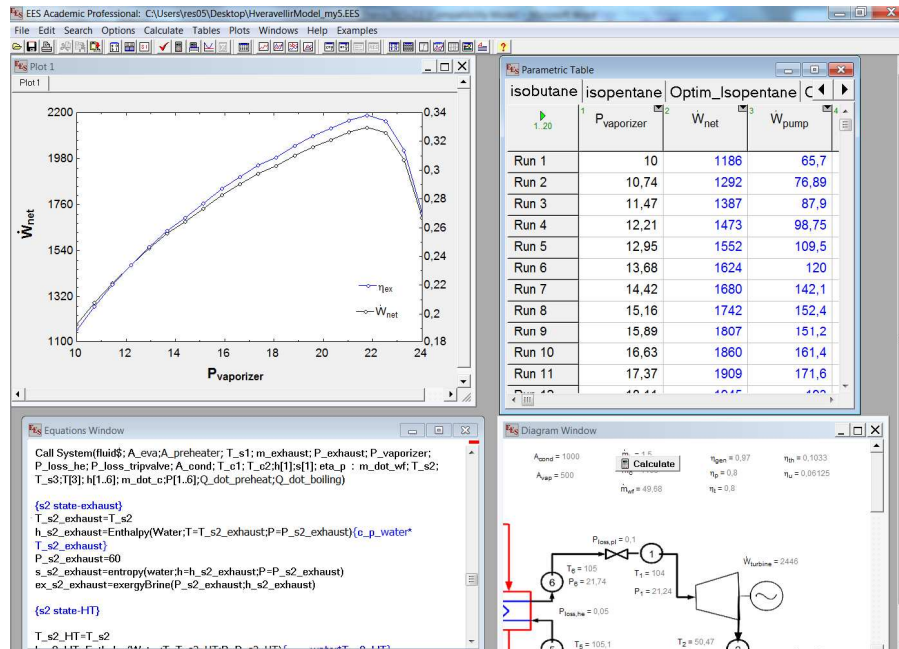


Figure 2-15 EES interface

The EES uses the iteration methods to solve the set of equations. This set of equations was introduced in chapter 2.2. They are energy and exergy balance equations, pump and turbine equations, heat transfer equations with logarithmic mean temperature difference.

But the optimization process is more than only solving the set of these equations. It requires solving the equations with loops checking each time are the restriction of the system not crossed.

For solving the optimization problem every thermo dynamical point of the cycle was defined. Three optimization procedures were created: the preheater, the evaporator and the condenser procedure. Detailed procedures description is presented below.

Pre-heater procedure

This procedure calculates how much of available heat can be utilized in the pre-heater without breaking the barriers set by the pinch point in the heat exchanger. The procedure is a “repeat until” loop. The inputs of the procedure are: vaporizer pressure, pre-heater area, working fluid type, maximal heat utilized by the pre-heater, minimal heat utilized by the pre-heater, and the temperature of the water flowing out of the pre-heater ($T_{w2,out}$). This temperature point corresponds to the bubble point of the organic Rankine cycle.

$P_{vaporizer}$ and $A_{preheater}$ are assumed. After knowing the $P_{vaporizer}$, T_{bubble} is calculated. $Q_{pre-heater}$ is defined as a mean arithmetic value of Q_{max} and Q_{min} . Then the pinch point constrain is checked. If T_{bubble} is higher than $T_{w2,out}$ that means that $Q_{pre-heater}$ is too high because the heated stream has higher temperature than heating stream. That is a validation of thermodynamic law. Q_{max} is assumed equal to $Q_{pre-heater}$. The loop starts from the beginning. This action lowers the $Q_{pre-heater}$ calculated in the next step. That should lower $T_{w2,out}$ but in this particular case $T_{w2,out}$ is fixed. That means that the vaporizer pressure

must be chosen in a way for T_{bubble} to be lower than $T_{w2,\text{out}}$. Otherwise the model does not work properly. If the $P_{\text{vaporizer}}$ is chosen properly and T_{bubble} is smaller than $T_{w2,\text{out}}$ than the loop goes by the “right branch” of the scheme shown below. The logarithmic mean temperature difference is calculated than. Corresponding to that LMTD and to the $Q_{\text{preheater}}$ the area of the heat exchanger (A_{calc}) is calculated. Afterwards the calculated area (A_{calc}) is compared to the assumed area ($A_{\text{preheater}}$). If the calculated area is higher than area assumed that mean that too much heat is being transferred through the heat exchanger. Than the present $Q_{\text{preheater}}$ is set as Q_{max} . This actually decreases the $Q_{\text{preheater}}$ calculated in the next loop. The loop is repeated. If the area calculated is smaller than the assumed area of the pre-heater that means that more heat can be transferred through the pre-heater. Than the present $Q_{\text{preheater}}$ is set as Q_{min} . This actually increases the $Q_{\text{preheater}}$ calculated in the next loop. The loop is repeated. The loop is repeated until the value of Q_{max} is close enough to the Q_{min} . This condition is defined as follows. The difference of Q_{max} and Q_{min} divided by Q_{max} should be smaller than 0,001.

Below the scheme of the procedure is shown.

Evaporator procedure

The procedure for calculating the evaporator is analogical as the pre-heater procedure. The only difference is in setting the initial Q_{max} , Q_{min} and set evaporator area is different than pre-heater area.

The mass flow rate of the system is calculated as by dividing the sum of evaporation and pre-heater heat by the enthalpy difference of enthalpy after the evaporator and the enthalpy before the pre-heater.

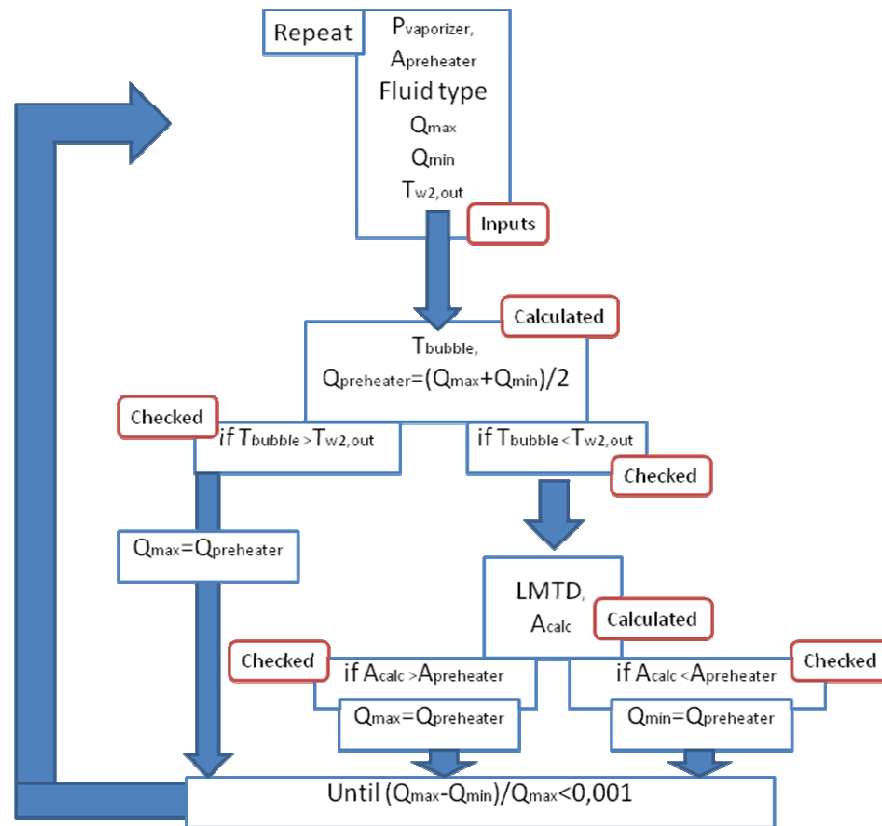


Figure 2-16 The block scheme of the pre-heater procedure

Condenser procedure

Condenser procedure is also analogical as to the method to the evaporator and pre-heater procedures. It used different parameters so it is explained separately. The condenser procedure optimizes the condensation pressure to the value which will be able to condensate all of the working fluid at calculated previously mass flow rate. This optimization has a barrier set by the pinch point. The saturation temperature must be always higher than the corresponding dew point temperature of the air.

The procedure starts by assuming minimal and maximal condensation pressures and the area of the condenser. In the next step the condensation pressure is calculated as an arithmetic mean of the minimal and maximal pressure assumed. When this pressure is known all of the enthalpies of the condenser are calculated. To calculate the enthalpy just before the condenser the turbine equation is used. After knowing the enthalpies in the condenser, the condenser is divided into two heat exchangers: the desuperheater and the condenser. This is done for practical reasons for making possible calculating the LMTD of each heat exchanger separately. The heat of the condensation and superheat are calculated. Knowing the condensation and evaporation heat the mass flow rate of the cooling air is calculated using the energy balance equation for the whole condenser. Then the temperature of the air corresponding to the dew point of the working fluid is calculated using the energy balance equation only on the condensing part of the condenser.

After the calculating part the pinch point is checked. If the dew point temperature of air corresponding to the saturation temperature of the heat exchanger is higher than the saturation temperature that means that the set condensation pressure is too low. The actual condensation pressure (P_{cond}) is set as $P_{\text{cond_min}}$ in the next iteration. This increases the condensation pressure calculated in the next iteration.

If the dew point temperature is higher than the corresponding to it air temperature than the LMTD of both desuperheater and the condenser are calculated. Knowing the LMTD of the heat exchangers and the heat exchanged the area of heat exchange is calculated. The sum of the area of the desuperheater and the condenser is named A_{calc} . That value is compared to the condenser area assumed at the beginning of the iteration.

If the calculated area is higher than assumed at the beginning than the actual condensation pressure is set as minimal condensation pressure in the next step. This move decreases the heat exchange area in the next iteration.

If the calculated area is lower than assumed at the beginning than the actual condensation pressure is set as maximal condensation pressure in the next step. This move increases the heat exchange area in the next iteration.

The iteration takes place until the maximal condensation pressure and the minimal condensation pressure are close enough. This is defined that the difference between maximal and minimal condensation pressure divided by minimal condensation pressure should be smaller than 0,001.

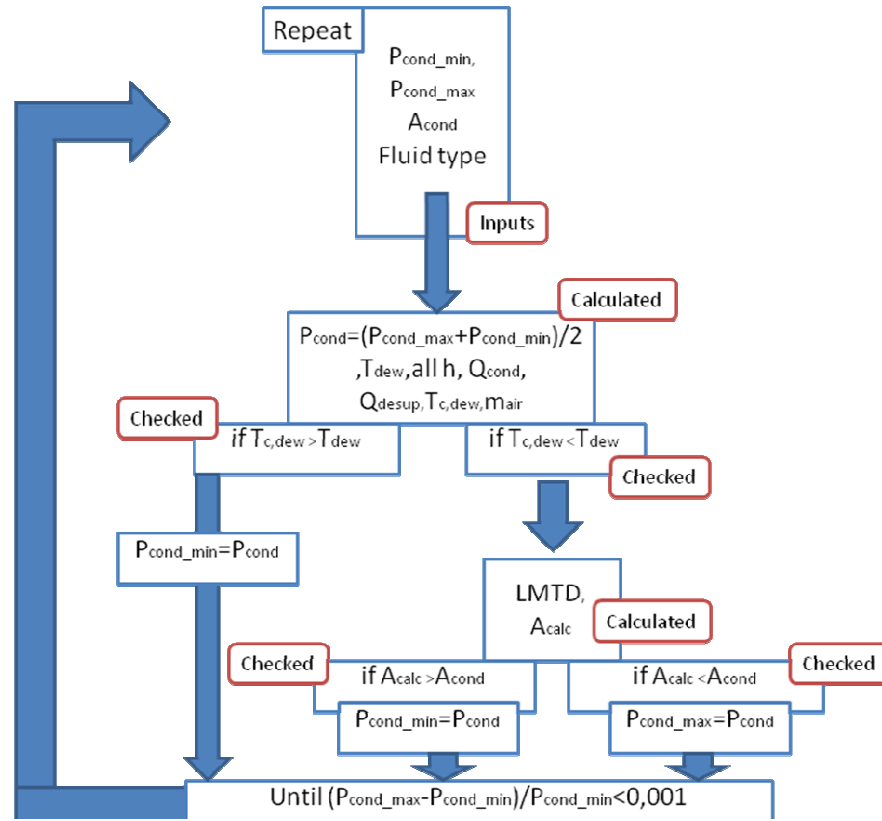


Figure 2-17 The block scheme of the condenser procedure

The outputs of the whole optimization procedure are: condensation pressure, all points of the cycle. The whole optimization work in a way to set the condensation pressure and all the points of the cycle in a way to best fit assumed vaporization pressure and the pre-heater, evaporator and condenser area. Therefore for finding the optimal value of the vaporization pressure the table is made in which vaporization pressure in change in some range. For each vaporization pressure the whole cycle is optimized. For each vaporization pressure the objective function is calculated. The optimal pressure is the pressure for which the objective function reaches its minimum or maximum (depending on the type of the function). The areas assumed are kept constant during the optimization of the vaporization pressure. After choosing the optimal vaporization pressure the areas of the heat exchangers are optimized. It is also done in the table. This time the vaporization pressure is kept constant. The first step is changing the pre-heater area to reach the optimal values of the objective function. The evaporator and the condenser area are kept constant. When the optimal value of the pre-heaters area is chosen, it is being fixed and the evaporator area is being changed. After choosing the optimal value of the evaporator area it is also fixed and the area of the condenser is being changed. After choosing the optimal condenser area the loop starts again. If the areas of the heat exchangers do not change in the next manual iteration steps the actual areas are chosen as optimal.

2.8 Optimization Results

The ORC with isopentane was chosen as a most efficient solution for a waste heat recovery system for bioliquid diesel power plant. The detailed optimization results are shown and discussed in the next subchapter. On the Figure 2-18 the Sankey diagram of the whole optimized system is shown. Sankey diagram represents the energy balance of the system.

As it can be seen 48,4% of the energy input as a fuel to the engine is lost. It is converted into the heat of the engine block low temperature and high temperature and the heat of exhaust gases. The low temperature heat from the engine block is not used in the waste heat recovery system because it is used for other purposes. The heat which is recovered from the engine comes from high temperature engine cooling and the exhaust gases. 19,4% of the fuel energy is converted into the heat of the exhaust gases, 17,4% of total fuel input heats the block of the engine. So the waste heat recovery input is 36,7 % of the initial fuel input is running the recovery system. As it is visible on the figure almost 90 % of recovered heat is dumped in the condenser or the recovery cycle. Because of low heat value that heat cannot be used. But that loss is still necessary for the work of the recovery system. About 7 to 9 % of recovered heat is turned into electricity. That means that the recovery system increases total electrical output of the system by 3,1 %.

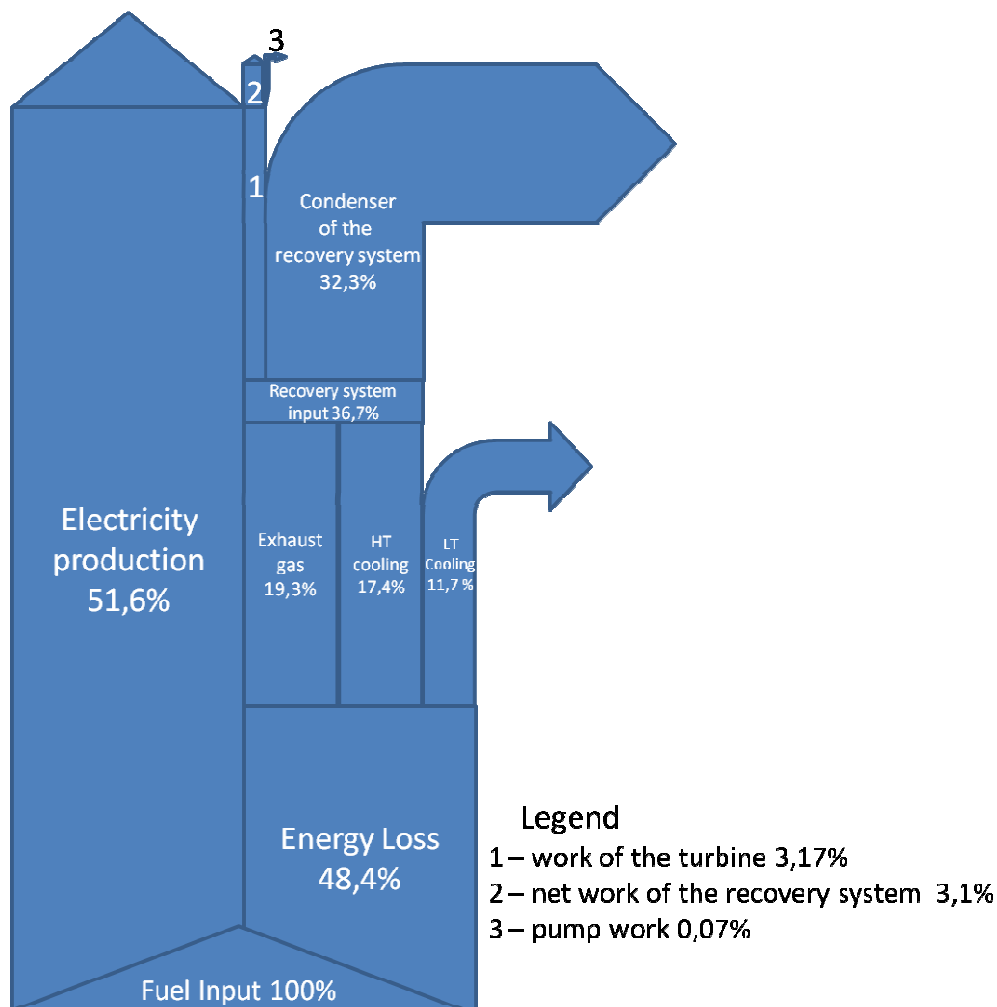


Figure 2-18 Sankey diagram of the optimized bioliquid engine with waste heat recovery system

2.8.1 Simple ORC and Steam Cycle results comparison

The results of the optimization of the Organic Rankine Cycle with isobutane, isopentane as working fluid and Steam Cycle are shown below. Figures 2-19 and 2-20 show the optimal objective function values as a function of different vaporization pressures. As mentioned in chapter 2.7 this optimization was done with the areas of the heat exchangers assumed constant. These areas were as follows: evaporator area 500 m², preheater area 1500 m², condenser area 5000 m². So the results shown below do not have most cost effective areas. The optimal values of the objective functions and the optimized cycle parameters are shown in the tables 2-12, 2-13 and 2-14, 2-15 .

Table 2-12 The optimal values of the objective functions

Objective function	Value			
	Isopentane	Isopentane adjusted area	Isobutane	Isobutane adjusted area
Net work of the system	1756 kW	1741 kW	1648 kW	1630 kW
Exergetic efficiency	27,9 %	27,6 %	26,2%	25,9%
Net present value after 5 years	-1,3mln €	-0,99mln €	-1,8mln €	-1,8 mln €

Table 2-13 The optimal values of objective functions (continued)

Objective function	Value	
	Steam	Steam adjusted area
Net work of the system	1520 kW	1573 kW
Exergetic efficiency	24,2%	25,1%
Net present value after 5 years	-1,4mln €	-1,1 mln €

Table 2-14 The optimal values of the parameters of the waste heat recovery system

Parameter		P_{vap}	P_{cond}	m_{wf}
Value	Isobutane	22,5 bar	7,2 bar	55,4 kg/s
	Isobutane adjusted area	22,5 bar	7,3 bar	56,7 kg/s
	Isopentane	8,4 bar	2,1 bar	48,4kg/s
	Isopentane adjusted area	8,3 bar	2,2 bar	47,9 kg/s
	Steam	1,422 bar	0,063 bar	4,5 kg/s
	Steam adjusted area	1,422 bar	0,066 bar	4,7 kg/s
Parameter		T_{cl}	T_{c2}	m_{air}
Value	Isobutane	9°C	24°C	1178 kg/s
	Isobutane adjusted area	9°C	24°C	1197 kg/s
	Isopentane	9°C	24°C	1191 kg/s
	Isopentane adjusted area	9°C	24°C	1177 kg/s
	Steam	9°C	24°C	645,7 kg/s
	Steam adjusted area	9°C	24°C	676,5 kg/s

Table 2-15 The optimal values of the parameters of the waste heat recovery system (continued)

Parameter		Q_{cond}	Q_{desup}	T_d	T_{dew}
Value	Isobutane	16396 kW	1355 kW	63,9°C	52°C
	Isobutane adjusted area	16683 kW	1366 kW	64,3°C	52,6°C
	Isopentane	15650 kW	2307 kW	76,1°C	51,4°C
	Isopentane adjusted area	15478 kW	2268 kW	76,4°C	51,9°C
	Steam	9734 kW	-	36,92°C	-
	Steam adjusted area	10198 kW	-	37,82°C	-
Parameter		$Q_{\text{evaporator}}$	$Q_{\text{preheater}}$	T_b	I_{total}
Value	Isobutane	10490 kW	9005 kW	106,9°C	7,2 mln €
	Isobutane adjusted area	10249 kW	9291 kW	106,9°C	7,2 mln €
	Isopentane	10490 kW	9291 kW	107,3°C	7,1 mln €
	Isopentane adjusted area	10490 kW	9059 kW	107,3°C	6,7 mln €
	Steam	2011 kW	9291 kW	109,8°C	6,4 mln €
	Steam adjusted area	2638 kW	9182 kW	109,8°C	6,2 mln €

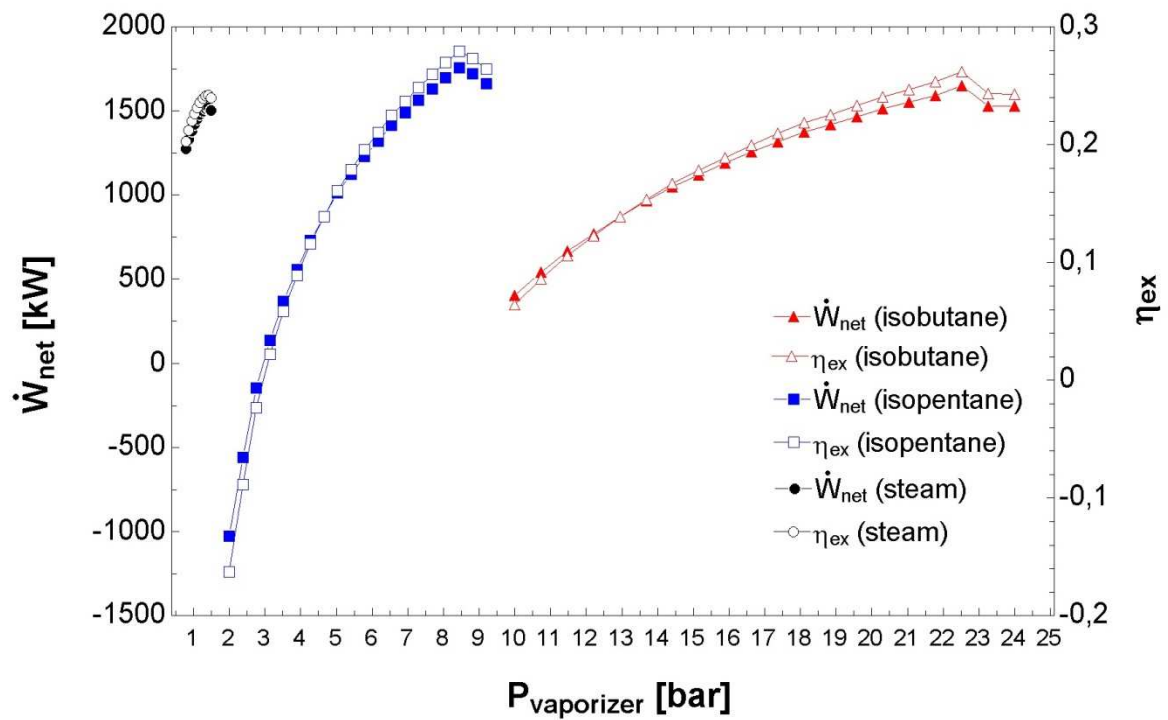


Figure 2-19 Net work and exergetic efficiency of the turbine as a function of vaporizer pressure (heat exchange areas assumed) (Grabiński, 2011)

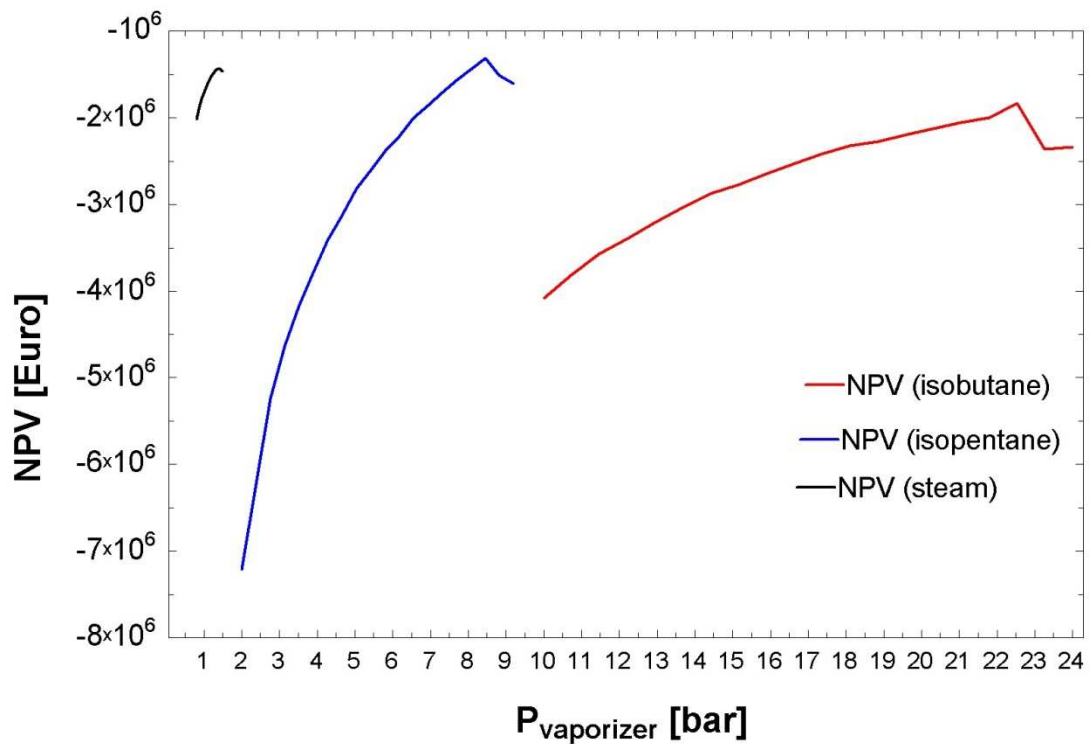


Figure 2-20 Net present value of the whole investment after 5 years as a function of vaporizer pressure (heat exchange areas assumed) (Grabiński, 2011)

All objective functions give the same value of the optimal vaporization pressure. All the objective functions have the same curve shape. That is logical. Net present value is a direct function of the investment cost and the net electricity generation. In this type of optimization the investment cost concerning the heat exchangers is fixed. So the only influence on the NPV is because of the turbine and the pump costs and net work of the system. Basically it can be said that in this type of optimization objective functions are functions of each other.

Further optimization of the system can be done by adjusting the heat exchanger area at constant selected previously optimal vaporization pressure. This final part of optimization is done only according to net present value as an objective function. This is done in a way mentioned at the end of chapter 2.7. The adjusted areas of the heat exchangers are shown in the table 2-16.

Table 2-16 Heat exchanger adjusted areas

	<i>Isobutene</i>	<i>Isopentane</i>	<i>Steam</i>
<i>Preheater</i>	1648 m ²	1479 m ²	1052 m ²
<i>Evaporator</i>	206,9 m ²	213,8 m ²	655,2 m ²
<i>Condenser</i>	5000 m ²	4869 m ²	5000 m ²

It can be concluded from tables 2.12 and 2.13 that adjusting the areas of the heat exchangers decreases the performance of the ORC system: net work of the system and exergetic efficiency is getting lower. But at the same time net present value after 5 years is increased. This increase in net present value is because of the decrease in investment costs of heat exchangers. And that is caused by the adjustment of the heat exchangers area. The main conclusion is that adjusting of the area of the heat exchangers leads to increased economical profitability even when the performance from the point of view of thermodynamics is decreased.

The adjustment of the heat exchanger areas of the Steam Cycle increases the performance but not enough to make it competitive with ORC Cycles.

It can be also concluded that with changing the working fluid in the ORC for the fluid with higher critical point the system is becoming more efficient and more financially feasible. Going from isobutene to isopentane the system achieves higher net work output, exergetic efficiency and net present value after 5 years.

It also needs to be admitted that the ORC does not use fully its potential. The desuperheater heat could be used for the purpose of regeneration. That would improve the efficiency of the cycle and make it even more dominant for the Steam Cycle.

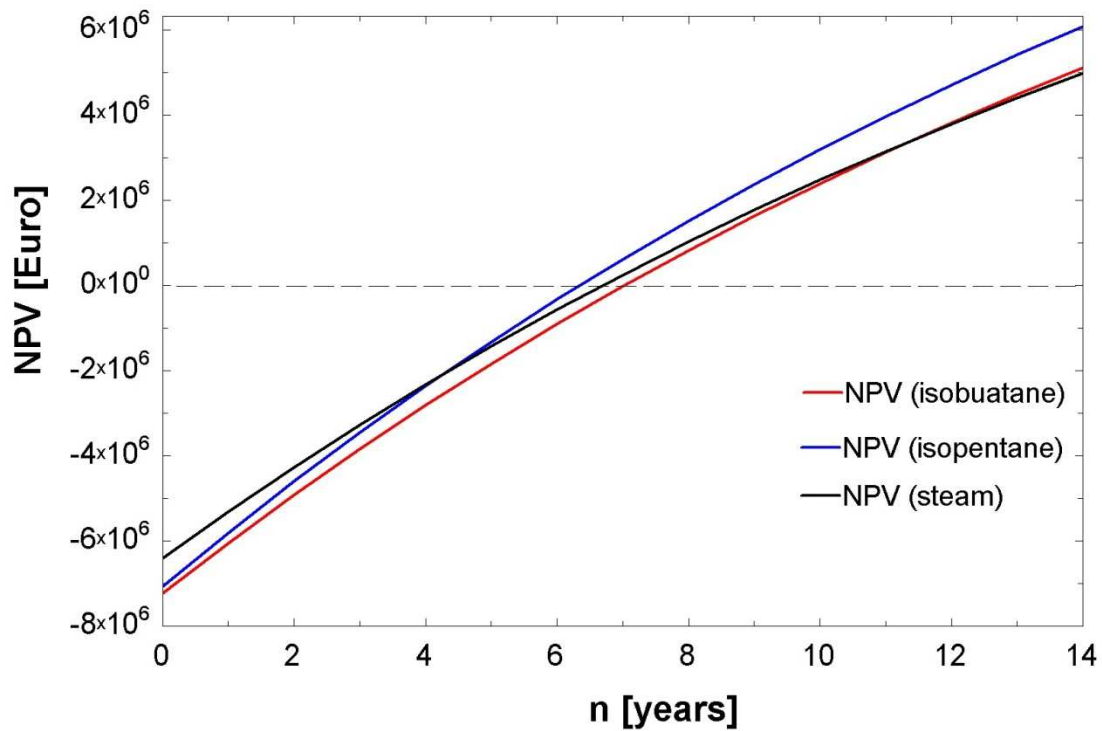


Figure 2-21 Net present value changes in years for all working fluids (Grabiński, 2011)

The net present value changes in years are shown in figure 2-21. As it can be seen the investing in waste heat recovery system pays back in 6-7 years. The isopentane ORC system pays back most rapidly almost in 6 years.

2.8.2 Further heat recovery from the flue gases

As mentioned previously there is a need for 3000 kW of heat. This heat would be used to preheat the fuel entering the diesel engine and heating the offices. The fuel needs to be heated to 85°C. The heat needed for heating the offices should be at the same temperature level. As it is seen in

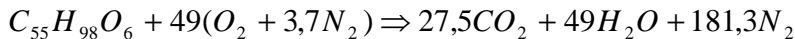
Table 2-15 there is some heat available for that purposes. It is the heat of the superheated steam exiting the turbine in the Organic Rankine Cycles. But it is not at sufficient level to meet the demand of the heating purposes. The heat in the superheated steam in the isobutene ORC is 1544 kW and for isopentane 2557 kW. Also the temperatures of these steams are too low to support the heating system. The temperature of isobutene superheated steam is 49,2°C and for isopentane 63,5°C. It is still valuable heat to be used for regeneration. It would be a great study for the future to examine that option.

There is no superheated steam in the water steam cycle to be used for heating purposes. The condensation is in to low temperature to support heating.

Because of these reason another solution is examined. In the studied system the exhaust gas is cooled to 120°C to heat the evaporator of the recovery system. But the exhaust gases could be exploited further. Heat from them could be used to preheat the fuel and heat the offices. The factor limiting the temperature to which the exhaust gas can be exploited is the dew point of the moist air in the exhaust gases. If the exhaust gases are cooled to too low temperature than moisture in the air condenses. That causes the good conditions for corrosion to start.

At first it is examined how much of moisture is made by combusting the bioliquid. It is assumed the combustion is total and complete. That means that all carbon atoms are oxidized and are oxidized only to CO₂. As fuel a fat triglyceride is assumed. It is a chemical name for plant oil. Its structural chemical equation is as follows: C₅₅H₉₈O₂.

The stoichiometric combustion is written below.



The ratio of water and the air is known.

$$Humidity_ratio = \frac{49H_2O}{49(O_2 + 3,7N_2)} = 0,132 \frac{g - H_2O}{g - air}$$

The humidity of the air which reacts in the combustion process is also taken into account. It has a humidity ratio of 0,0072 at atmospheric temperature. Therefore the total humidity ratio of gases is 0,139. Using the Mollier diagram the dew point temperature of the moist air in the exhaust gas is obtained. The dew point is found in the point where humidity ratio crosses the relative humidity line of 1. The dew point temperature is 59°C.

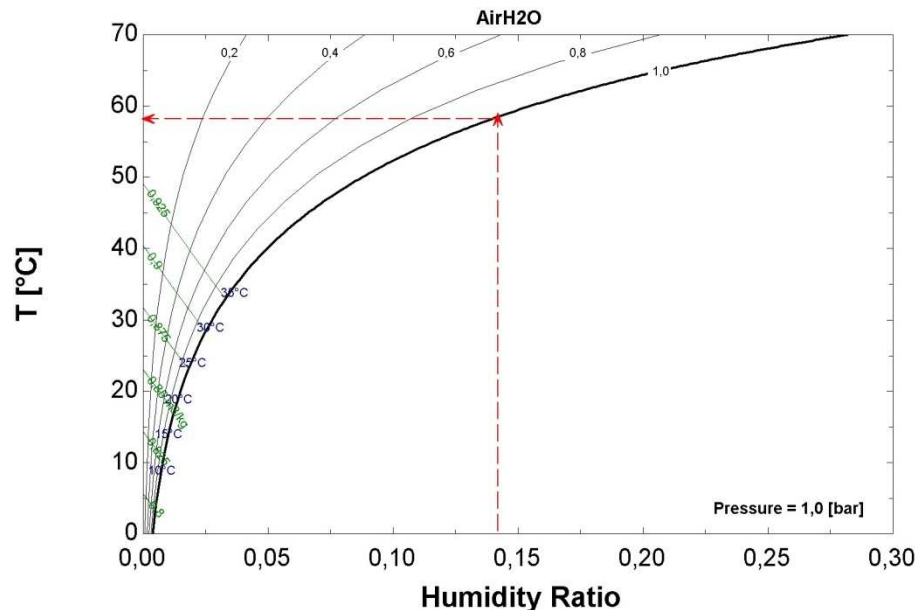


Figure 2-22 Choosing the dew point temperature of the exhaust gases on the Molier diagram (Grabiński, 2011)

Now knowing the dew point, the exhaust gases mass flow rate 60 kg/s. The heat available for further utilization can be calculated.

$$Q_{extra_exhaust} = m_{exhaust} \cdot C_p \cdot (T_{e2} - T_{e3})$$

Where

$$T_{e2} = 120^{\circ}\text{C}$$

$$T_{e3} = 69^{\circ}\text{C} \text{ (10}^{\circ}\text{C added to the dew temperature to be sure dew is not reached)}$$

$$C_p = 1,005 \text{ kJ/kgK}$$

The extra heat from the exhaust is 3,08 MW. That is enough to preheat the fuel and heat the offices. It is also at the sufficient temperature level.

3 FEASIBILITY STUDY POLAND

It was concluded that the optimized bioliquid diesel engine with waste heat recovery system is a feasible solution for electricity production purposes. Such applications are presently being built in England.

The reason of making this chapter is to investigate if such an application could be implemented in different country in different conditions. Because of EEA grand requirements under which this thesis is made Poland was chosen for the investigated country.

The function of the studied bioliquid power plant was to support the whole district with electricity. It could be also used to support few villages with electricity. The other application of this system would be the industrial facilities and factories.

The issue is if Poland has enough potential and conditions in oil plants to support the fuel at a reasonable price for a bioliquid system. Other problem is the support for biofuels in Poland. It is examined if the level of the support is sufficient.

3.1 Polish conditions

Rapeseed is a main polish oil plant. In the last years the acreage of rapeseed varied between 95-97% of oil plants in Poland. The magnitude of the production of rapeseed in Poland characterizes with large variations. They are caused by the changing of the yield because of rapeseed sensitivity for atmospheric conditions.

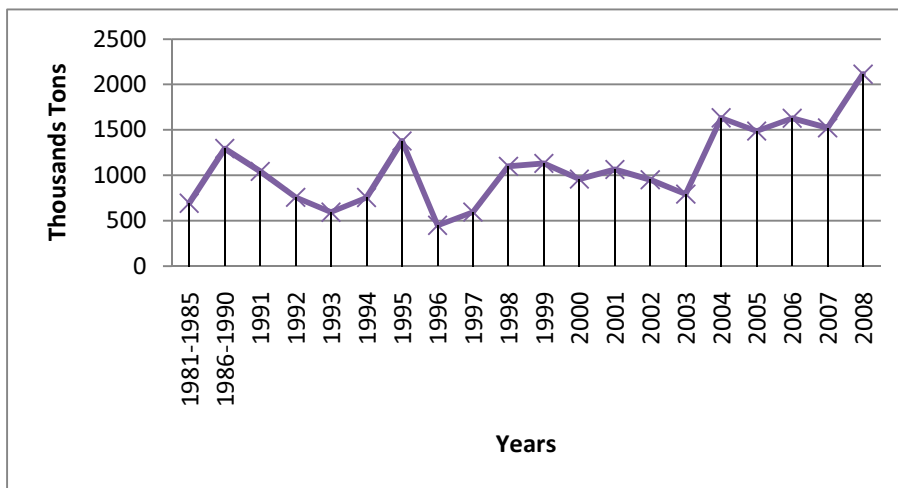


Figure 3-1 Changes in the harvest of rapeseed in years [thousands tons] (Krajowe Zrzeszenie Producentów Rzepak, 2011)

The other issue is that the rapeseed must be competing in price with wheat. The farmers will not plant rapeseed if they will receive more income for planting wheat. The problem in producing rapeseed oil for the purpose of electricity production lies also in the other way of using the plant oil. The main usage of oil is as food oil. So if there is a high demand for

food oils all rapeseed are used for making food oils. The oil companies are buying the rapeseed oil as contracted, and when there is a year with higher yield than the oil site contracted there is some extra rapeseed to be used for energy purposes. But as it was mentioned it is hard to predict the yield of rapeseed.

So as it can be seen the field of oils used directly for making the electricity is complex and unpredictable. These could be changed if a proper support from the government and the EU would be set.

3.2 Policy Support on Bio-liquid applications

Poland has a governmental program for promoting the biofuels. It is called: “Long-term program for the promotion of biofuels and other renewable fuels for years 2008-2014” (Government, 2007). The program main goal is to reach the EU directive (2003/30/WE) requirements about the amount of biofuels in the transportation sector.

Table 3-1 The minimal EU requirements for the % of biofuels in Poland’s transportation market (Government, 2007)

Year	2007	2008	2009	2010	2011	2012	2013	2014
% of biofuel in the transportation market	2,3	3,45	4,6	5,57	6,2	6,65	7,1	7,55

The requirements of this directive were implemented in Poland in the *act about biocomponents and liquid biofuels and about the monitoring and control of the quality of the fuel system* from 25 August 2006. (Government, 2007) This act implemented three important issues concerning biofuels:

1. It creates the possibility for farmers to produce for their own use all kinds of liquid biofuels and plant oils. His not paying the excise and does not have to pay excise security. Biofuel produced for own purposes should preserve minimal environmental requirements. The limit of the production is 100 l/ha of the agricultural land.
2. From 1 January 2008 the obligation was put on the companies which are producing or importing liquid fuels. This obligation sets the minimal requirements for biocomponents and biofuels in the transportation sector as in 2003/30/WE directive
3. Implementing into polish law the notion of “selected fleet” which enabled the usage of the fuel with increased biofuel content

Of course implementing these issues to the law is not solving the biofuels problem. Certain help mechanisms must be implemented. The government financial help for the biofuels according to polish and EU law must not be too high. The level of financial support by law should not be higher than the difference of the production of the biocomponents (bioethanol and biodiesel) and the whole sales prices of gasoline and diesel oil without taxes.

That difference was calculated according to EU requirements and concerning the fact that the lower heating value of the biofuels is lower than in gasoline and traditional diesel.

The maximal amounts of financial support for biofuel in Poland are: 647,48 €/m³ of produced bioethanol and 397,4 €/m³ for the production of biodiesel.

Implementing mechanisms which will support the biofuels at the calculated above level is necessary to make the development of biofuels possible in Poland. These mechanisms will support the biofuels at each stage of their life cycle from the production and to the use.

There are five six mechanisms of support for biofuels mentioned in the program (Government, 2007).

1. The support of biocomponents in the tax system and the fuel charge
2. The support for the plantations of plants used in the production of biofuel
3. The financial support for the production of biocomponents and liquid biofuels from the EU funds and public funding
4. Actions that increase the demand side of liquid biofuels
5. R&D studies on liquid biofuels
6. The information and education activities on liquid bio fuels

Concerning the topic of the thesis only the mechanisms which support directly the oil plants as a fuel will be discussed in detail.

Firstly the changes in tax system will be discussed. The producer of biocomponents which can be used directly as a fuel (plant oils) will have to pay a symbolic excise of 0,026 €/liter of made fuel. Another financial incentive for biocomponent producers will be lowering the income tax by 19% of additional biocomponent production costs above the costs of traditional fuels production. The fuel charge will be canceled for biocomponents that can be used directly as fuel.

Secondly the support for the plantations of plants used as a biofuel is examined. The new funds from the EU became possible. The funding is at a level of 45 €/ha of the cultivation land for plants that are used for making the biocomponents. That EU funding is additionally supported by the government at a level of 45 €/ha and it is specially connected to rapeseed plantation. Also the governmental funding at 50% of costs of creating the long term plantation became a law.

Thirdly there is a governmental program called “The program for the development of rural areas for years 2007-2013”. It supports individual projects concerning inter alia the production of refined plant oils. The minimal single support level is set as 26 thousand € and the maximal value of the single support is 5 Million €.

The support for oil plants cultivation seems high. It is hard to check if it has any real influence on the oil plants market. There was an analysis done by the government itself. It seems that the analysis is objective because it shows not only positive aspects of the act.

The report states that the amount of biofuels used in Poland increased in 2008 and it was 659 thousand tones. Their prices were compatible with the traditional fuel prices. The report sees also major disadvantages of the program. The biofuel sold comes mainly from import. Also there is no major growth in plants for biofuels cultivation. No new biofuels production facilities are open. Even regression appears in that field.

The polish support for biofuels requires further development. The good biofuels support law could also make the implementation of biodiesel engines with recovery system more feasible. Under some conditions and scale they could be implemented even now.

3.3 Future development of Bio-liquid Power Plants with WHR

As it was mentioned above polish conditions are not friendly for the development of biodiesel power plants. But that could change after the law is improved. As for current conditions the bioliquid power plants can be situated close to the current source of the plant oil: the refineries. They could be supported with overproduction oil from the refineries. The power plants could supply the refineries themselves and the nearby villages with electricity. On Figure 3-2 there are shown polish rapeseed oil production sites. Each of these sites could be the place of implementing the biodiesel power plant with waste heat recovery system.



Figure 3-2 Rapeseed oil production sites in Poland (Polskie Stowarzyszenie Producentów Oleju, 2011)

It is examined if one of these facilities could be a place for future bioliquid power plant. Firstly it is evaluated how much of rapeseed oil is needed to support the 28 MW diesel engine for a year.

$$m_{fuel} = \frac{Q_{fuel}}{LHV_{fuel}} = 1,46 \left[\frac{kg}{s} \right]$$

$$Q_{fuel} = \frac{W_{output}}{\eta_{engine}} = 55MW$$

Where:

m_{fuel} [kg/s] – mass flow rate of the fuel supplying the engine

Q_{fuel} [kW]– power input with the fuel

$\eta_{engine} = 55,5\%$ - electrical efficiency of the engine with waste heat recovery system

$W_{output} = 30,5$ MW – electricity generated in the power plant

$LHV_{fuel} = 37620$ kJ/kg – lower heating value of the rapeseed oil

The mass flow rate needed to for the engine is 1,46 kg/s. When assuming that the engine runs 8400 hours a year (which was assumed also in the economical analysis) it is conducted that a yearly need for the rapeseed oil for the 30,5 MW bioliquid power plant is 44,15 thousand tones.

It was examined what are the production of the oil powers of companies located in Poland. The polish biggest plant oil company, Kruszwica reveals in a report for year 2009, that in that year 116,5 thousand tons of oil were sold. (Kruszwica, 2009) That means that in order to support one bioliquid power plant almost 40 % of its oil should be used only for that purpose. That is currently not a realistic scenario. Concluding polish oil market is still not ready for bioliquid power plants.

4 CONCLUSIONS

Recovering waste heat and converting it to electricity from the 28 MW bioliquid diesel power plant is economically and thermodynamically feasible. Waste heat recovery system when optimized properly can increase the electricity output of the engine by 3,17 %. When operated with the optimal parameters of the system the waste heat recovery pays back after 6 to 7 years, given the assumptions made in this study. The optimization was based on meteorological data from London, and would need modification if application to other locations is of interest. Each case should be calculated separately.

Organic Rankine Cycle with isopentane as working fluid was found to be the optimal solution for this specific case. This solution gives the highest electricity output with the lowest associated cost and the investment pays back in the shortest time. It was also concluded that there are no possibilities of further heat recovery from the Steam Rankine Cycle. Therefore in this configuration the fuel of the engine and the surrounding offices should be heated from different source. It was examined that the exhaust gas can be the source of that heat. It can be cooled more than it is planned to be cooled by the waste heat recovery system. The lower temperature limit of absorbing the heat from the exhaust gas is the dew point in the flue gas.

The Simple Organic Rankine Cycles could be even more dominant over the Steam Rankine Cycle than what the models show here. They have the heat that could be recovered from them: the heat of the superheated steam after the turbine. It was evaluated that in cases of both working fluids (isobutene and isopentane) that steam is not valuable to be used for the preheating the fuel and heating the offices. As it was mentioned exhaust gas can be use for those purposes. Even if the heat of the superheated steam does not have the capacity nor a sufficient temperature to be used to achieve this goal, it is valuable enough to be regenerated and used to preheat the working fluid in the boiler. That would improve the recovery system efficiency and could increase its dominance over the Steam Cycle configuration.

In the Poland case study it was concluded that even when possessing the optimised solution (28 MW biodiesel power plants with heat recovery system) it is still not enough to implement it as an actual investment. That is because political and geographical circumstances influence the result more than economical and financial feasibility. Polish rapeseed oil production capacity is not large enough to support such large bioliquid diesel plants. The political support is not sufficient to change that situation.

5 FURTHER STUDIES

The most important future study would be to include regeneration in the Organic Rankine Cycle system, and examine how that will affect the efficiency and economy of the system. Also other working fluids could be examined. Interesting issue would be examining different mixtures of the working fluids. Thirdly it could be examined if adding of the superheater would influence the ORC and the Steam Cycle in a positive way. Supercritical cycle could be the solution which could compete with the isopentane ORC solution in term of electricity production. More detailed exergy analysis could be performed, it could be examined which element of the cycle causes the highest exergy destruction and how can that be improved. Also the possibilities of using different fuels could be investigated. Another country could be studied on the possibilities of using such an application. It would be a good solution to choose the country with different conditions than Poland with higher oil plants potential.

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