REPUBLIC OF TURKEY

YILDIZ TECHNICAL UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

THERMODYNAMIC AND ECONOMICAL ANALYSIS OF ORGANIC RANKINE CYCLE USAGE WITH NATURAL GAS FIRED INTERNAL COMBUSTION ENGINE WASTE HEAT

Yunus Emre TALU

MASTER OF SCIENCE THESIS

Department of Mechanical Engineering

Heat and Processing Program

Advisor

Prof. Dr. Ali PINARBAŞI

April, 2019

REPUBLIC OF TURKEY

YILDIZ TECHNICAL UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

THERMODYNAMIC AND ECONOMICAL ANALYSIS OF ORGANIC RANKINE CYCLE USAGE WITH NATURAL GAS FIRED INTERNAL COMBUSTION ENGINE WASTE HEAT

Yunus Emre TALU

MASTER OF SCIENCE THESIS

Department of Mechanical Engineering

Heat and Processing Program

Advisor

Prof. Dr. Ali PINARBAŞI

April, 2019

REPUBLIC OF TURKEY

YILDIZ TECHNICAL UNIVERSITY

GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

THERMODYNAMIC AND ECONOMICAL ANALYSIS OF ORGANIC RANKINE CYCE USAGE WITH NATURAL GAS FIRED INTERNAL COMBUSTION ENGINE WASTE HEAT

A thesis submitted by Yunus Emre TALU in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE THESIS is approved by the committee on 08.05.2019 in Department of Mechanical Engineering, Heat and Processing Program.

Prof. Dr. Ali PINARBAŞI

Yıldız Technical University

Advisor

Approved By the Examining Committee

Prof. Dr. Ali PINARBAŞI, Advisor
Yıldız Technical University

Asst. Prof. Nader JAVANI, Member
Yıldız Technical University

Asst. Prof. Levent KAVURMACIOĞLU, Member
Istanbul Technical University

I hereby declare that I have obtained the required legal permissions during data collection and exploitation procedures, that I have made the in-text citations and cited the references properly, that I haven't falsified and/or fabricated research data and results of the study and that I have abided by the principles of the scientific research and ethics during my Thesis Study under the title of Thermodynamic and Economical Analysis of Organic Rankine Cycle Usage with Natural Gas Fired Internal Combustion Engine Waste Heat supervised by my supervisor, Prof. Dr. Ali PINARBAŞI. In the case of a discovery of false statement, I am to acknowledge any legal consequence.

Yunus Emre TALU

Signature

Dedicated to my family

and friends

ACKNOWLEDGEMENTS

I would like to thank to my advisor dear Prof. Dr. Ali PINARBAŞI for his guidance. Also, I would like to thank to Research Assistant Ahmet Doğan who patiently followed and controlled all steps of research.

Besides that, I have to thank to my friends and family for their support in this hard period.

Yunus Emre TALU

TABLE OF CONTENT

LIST OF SYMBOLS

b	Bulk index
C	Cost
c_p	Specific heat
crit	critical condition index
cogen	Overall cogeneration system index
con	Condenser
cs	Condenser surface condition index
d	Diameter
des	Destruction
E	Energy
En	Entropy
es	Evaporator surface condition index
Ex	Exergy
f	Fluid phase
gen	Generation condition index
h	Enthalpy of mentioned state point
hrv	Heat recovery index
i	Inlet condition index
inv	Investment index
lm	Logarithmic mean index
M	Mass
$\dot{m}$	Mass flow rate
o	Outlet condition index
op	Operating condition index
P	Pressure
Pr	Prandtl number
Q	Total heat transfer rate for mentioned component
$\dot{q}$	Heat transfer rate for mentioned index
Re	Reynold number
rv	Reversibility index
S	Total entropy of mentioned component
sc	Scrap
T	Temperature
V	Velocity of mentioned index
W	Power output at mentioned index
η	Efficiency of mentioned index
μ	Dynamic viscosity
ρ	Density
ΔT	Temperature difference of mentioned states

LIST OF ABBREVIATIONS

A34	Ashrae 34 Classification
ALT	Atmospheric Life Time
BORC	Basic Organic Rankine Cycle
CEPCI	Chemical Engineering Cost Index
CFC	Chloro Fluoro Carbons
CSOE	Cogeneration System Overall Efficiency of Electricity Production
CSFC	Cogeneration System Fuel Consumption
EEC	Exergy Efficiency of Cycle
EFEC	Engine Fuel Energy Consumption
EOP	Evaporator Outlet Pressure
EOT	Evaporator Outlet Temperature
FES	Fuel Energy Save
GWP	Global Warming Potential
HCFC	Hydro Chloro Fluoro Carbons
HFC	Hydro Fluoro Carbons
HRE	Heat Recovery Efficiency
ICE	Internal Combustion Engine
IHE	Internal Heat Exchanger
LCV	Lower Calorific Value
MFR	Mass Flow Rate
MW	Molecule Weight
NOPT	Net Output Power of Turbine
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PBP	Payback Period
RORC	Regenerative Organic Rankine Cycle
RFEC	Reduction of Fuel Energy Consumption
TEC	Thermodynamic Efficiency of Cycle
TP	Total Profit
TPC	Total Profit of Cogeneration System
WF	Working Fluid

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

Thermodynamic and Economical Analysis of Organic Rankine Cycle Usage with Natural Gas Fired Internal Combustion Engine Waste Heat

Yunus Emre TALU

Department of Mechanical Engineering

Master of Science Thesis

Advisor: Prof. Dr. Ali PINARBAŞI

Nowadays, heat recovery systems are coming into substantially prominence in conjunction with augmentation of pollution allied with fossil fuel dependency and population growth. Internal combustions engines are one of primarily responsible about fossil fuel consumption with widely utilization area beside all benefits to humanity such as transportation, power unit and agricultural machinery. In this study, Organic Rankine Cycle (ORC) usage analyzed as a heat recovery system for exhaust gas heat loss of Internal Combustion Engine (ICE) which is TCG2032 model manufactured by MWM brand of Caterpillar Energy Solutions GmbH. Mathematical models were developed for two different chosen cycles which are basic ORC and regenerative ORC in conscious the simplicity and lower investment cost necessity of a heat recovery system. Regenerative cycle includes an additional internal heat exchanger distinctly from basic cycle for the purpose of maximize cycle efficiency via provide pre-heat with cycle waste heat. Seven different working fluids that R113, R123, R134a, R141b, R245fa, n-pentane and n-butane were taking in consideration for both basic and regenerative cycle to represent different fundamental classes of fluids which are isentropic, dry and wet type. The maximum net power output was selected as the foremost evaluation criterion to analyze heat recovery system effectiveness. Optimization study importance is ensuing with the different working condition necessities for maximize net output power of working fluids relative to different thermal and physical characteristics. Maximum net output power of cycle was defined depending on different heat source conditions by help of optimization study which was made with the results of the first law and second law efficiency of each fluid and cycle combination. The thermal analysis results reveal that the R113 is the most efficient working fluid for regenerative cycle and R141b is the most efficient working fluid for basic cycle. R134a is one of the wet type fluids and it has the lowest net output power as comparatively from the other selected fluids. Generally, regenerative cycle produce more power than basic cycle with the rate of between 30-50%. Also analyzes show that the 6-17% increase at the overall efficiency appear possible with ICE-ORC combined cycle in according to single engine, and as a result of that 400-1000 kW reduce can be achieved for the fuel energy consumption. Besides that, Regenerative ORC was found more profitable with the 8,46 years Payback Period (PBP) against Basic ORC which has 9,82 years PBP.

Keywords: Organic Rankine Cycle, heat recovery system, engine waste heat recovery, economic analysis, optimization study

ÖZET

Doğalgaz Yakıtlı İçten Yanmalı Motorun Atık Isısı ile Organik Rankine Çevrimi Kullanımının Termodinamik ve Ekonomik Analizi

Yunus Emre TALU

Makine Mühendisliği Bölümü

Yüksek Lisans Tezi

Danışman: Prof. Dr. Ali PINARBAŞI

Günümüzde ısı geri kazanım sistemleri, nüfus artışı ve fosil yakıt bağımlılığının artmasına bağlı olarak kirlenmenin artması ile birlikte büyük önem kazanmaktadır. İçten yanmalı motorlar, ulaşım, enerji ünitesi ve tarım makineleri gibi geniş kullanım alanları ile insanlığa sağladığı tüm faydaların yanı sıra fosil yakıt tüketiminde en önemli sorumlulardan biridir. Bu çalışmada, Caterpillar Energy Solution GmbH tarafından üretilen MWM marka TCG2032 model içten yanmalı motorun egzoz gazı ısı kaybından ısı geri kazanım sağlamak amaçlı Organik Rankine Çevrimi (ORC) kullanımı analiz edilmiştir. Bir ısı geri kazanım sisteminin basitlik ve düşük yatırım maliyeti gerekliliğinin bilincinde olarak Temel ORC ve Rejeneratif ORC olmak üzere iki farklı çevrim seçilmiş ve matematik modeller bu iki çevrim için ayrı olarak oluşturulmuştur. Rejeneratif çevrim, temel çevrimden ayrı olarak, çevrimin atık ısısı ile ön ısıtma sağlayıp çevrimin verimliliğini en üst düzeye çıkarmak amacıyla kullanılan ek bir iç ısı değiştirici içermektedir. R113, R123, R134a, R141b, R245fa, n-pentane ve n-butane olmak üzere yedi farklı çalışma akışkanı, izentropik, kuru ve ıslak tip gibi akışkanın farklı temel sınıflarını temsil etmeleri için hem temel hem de rejeneratif çevrim için incelenmiştir. Isı geri kazanım sisteminin etkinliğini analiz etmek için en önemli değerlendirme kriteri olarak maksimum net güç çıkışı seçilmiştir. Optimizasyon çalışmasının önemi, çalışma akışkanlarının farklı termal ve fiziksel özelliklerine göre net çıkış gücünü maksimize etmek amacıyla farklı çalışma koşulu gerekliliklerine bağlı olarak ortaya çıkmaktadır. Her bir akışkan ve çevrim kombinasyonunun birinci yasa ve ikinci yasa verimi ile yapılan optimizasyon çalışmasıyla, farklı ısı kaynağı koşullarına bağlı olarak çevrimin maksimum net çıkış gücü belirlenmiştir. Termal analiz sonuçları, R113'ün rejeneratif çevrim için en verimli çalışma akışkanı olduğunu ve R141b'nin temel çevrim için en verimli çalışma akışkanı olduğunu ortaya koymaktadır. R134a, ıslak tip akışkanlardan biridir ve diğer seçilmiş çalışma akışkanlarına göre en düşük net çıkış gücüne sahiptir. Genel olarak, rejeneratif çevrim 30-50% arasında değişen bir oranla, temel çevrimden daha fazla güç üretmektedir. Ayrıca, yapılan analiz, tek başına motor kullanımına göre 6-17% oranında genel verimlilik artışının ICE-ORC entegre sistemi ile mümkün olduğunu göstermektedir ve bunun sonucu olarak yakıt tüketiminde 400-1000 kW düşüşünün sağlanabileceği gözükmektedir. Bunun yanı sıra, Rejeneratif ORC 8,46 yıllık geri ödeme süresi ile 9,82 yıl geri ödeme süresi olan temel çevrime göre daha karlı olarak bulunmuştur.

Anahtar Kelimeler: Organik Rankine Çevrimi, ısı geri kazanım sistemi, motor atık ısı geri kazanımı, ekonomik analiz, optimizasyon çalışması

1
INTRODUCTION

Energy need and energy supply method have become one of the major factors for designate strong countries in recent years [1]. Fossil fuels are the most common energy production resources for both developed and developing countries [2]. Overall fossil fuel usage is increased 51% as approximately in the period of 1995-2015, and it is estimated that the consumption will increase approximately 18% for the period of 2015-2035 [3]. Population growth is one of the most important reasons for fossil fuel usage increment. Fossil fuel consumption causes environmentally negative effects such as air pollution and global warming. Also fossil fuel usage is indirectly affected to social life as a result of negative effects on human health and life quality of populations [4]. In addition to this, fossil fuels are non-renewable resources and increment of usage is causing to ponder about its availability and sustainability. It would be a logical inference that increases in fossil fuels consumption will be bigger problem for next generations with creating wars and energy crisis.

Fossil fuel resources have widely utilization area such as application of industrial production, climatization, electrical energy production, agricultural machinery and transportation. Fossil fuels are using generally for mechanical energy production from chemical energy with combustion at most of the mentioned utilization area. Internal combustion engines (ICEs) are most known mechanical energy generator and as a reason of that, it is major consumer of fossil fuels [5]. In addition to this, ICEs can be used for electrical energy production from mechanical energy with using integrated generator.

Waste heat recovery systems are become important with increasing pollution depending on increment of fossil fuel usage as a response for growth population needs. [6]. It will be proper choice for a heat recovery system that secondary cycle usage such as Organic Rankine Cycle (ORC) which is using waste heat of ICE as a heat source. ORC integration to ICE improves overall system efficiency and reduces emissions by generating additional power without consume fossil fuel resources [7].

Organic Rankine Cycle can be preferred for small scale power plants by reason of lower level maximum working pressure against Rankine Cycle which is using water as working fluid. Organic working fluids can be efficient for lower temperatures because of organic fluids have lower boiling points and higher vapor pressures than water for same temperature. As a result of that, temperature difference between heat source and evaporator could be bigger for ORC [8]. Also increasing temperature difference between heat source and evaporator has positive effects to heat exchanger size by higher heat flux.

There is a fact that, it is not possible to achieve 100% efficiency for an energy conversion systems, in other saying, it is not possible that entirely convert an amount of energy into another energy form based on second law of thermodynamic. Significant amount energy is lost with this way separately from friction loss for Internal Combustion Engines. This irreversible energy which is not used for produce work causes to heat loss from system. Heat loss of system could be happen with different temperature levels and it determines quality of heat source. Classification of waste heat quality was showed at Table 1.1 which is defined by Auld [9]. Nowadays, industry is more interested in heat recovery systems than ever before because of raising needs cannot be covered with the traditional systems as economically.

Table 1.1 Quality classification of waste heat [9]

Quality	Heat Soruce	Heat Sink
High	>500 °C	>250 °C
Medium	250-500 °C	150-250 °C
Low	<250 °C	<150 °C

Exergy analysis is just as important as energy analysis to designate quality of heat source with calculation of usable part for find out best heat recovery process. It is economically and technically hard that produce work from low quality waste heat due to low exergy level.

Typical ICE can produce averagely 25% useful work, and it loses approximately 40% energy with exhaust gas waste heat, friction and engine block cooling [10]. In this study, only waste heat of exhaust gas taking into consideration for heat recovery system because engine block cooling waste heat is recycling already with producing heating water for investigated facility.

Traditional Rankine Cycle which is using water as working fluids was discovered by William John Macquorn Rankine (1820 - 1872) who is a Scottish engineer, mathematician and physicist [11]. Discovery of Organic Rankine Cycle is based on the ground of the searching about alternative fluid usage as a response of the higher pressure need for evaporator with low temperature sources. ORCs are become more popular with the desire of electricity power production from low and medium quality heat sources by the development of suitable working fluids to replace water in traditional Rankine Cycles. Organic fluids which are replaced with water contain carbon atom such as chloro-fluoro-carbons (CFC), hydro-chloro-fluoro-carbons (HCFC) and hydro-fluoro-carbons (HFC).

Basically ORC has four component which are evaporator (boiler), condenser, pump and turbine (expander) as similarly to traditional Rankine Cycle. Cycle efficiency and investment cost are increasing with relative to cycle complexation. Different application types which are reheating cycle usage, vapor extraction cycle usage, multiple loop usage etc. and also, additional components which are pre-heater, regenerator, open loop feed water heater etc. are investigating with the purpose of increasing cycle efficiency for especially low quality heat source as seen at literature review. In this study, Basic ORC and Regenerative ORC as two different cycle types were investigated by considering system simplicity and lower investment cost for medium quality heat source.

The larger and heaviest components of an ORC are heat exchangers. Also heat exchangers have important impacts to investment cost of system. Different types of heat exchangers are commercially available for ORC systems such as plate type, plate fin type, shell & tube type. Also, flow direction of fluid just as important as heat exchanger type for system design. Optimization study must be done for all heat exchangers with taking into consideration as comparatively for different opposite effects. For example, higher heat transfer surface area of heat exchanger provides higher heat transfer rates with lower exergy destruction, while it is causing to more expensive system. Also, heat exchanger material is important for sustainability and cost. Carbon steel is the most widespread material for these components due to its lower corrosion rates. Shell & tube is the most prevalent heat exchanger type for the ORC and it can be used with different flow directions to serve at different purposes. It is comprised of a bundle of tubes enclosed in a shell with two different stream flow one or more times within parallel or opposite direction. Plate type and plate-fin type heat exchangers could be preferred against shell & tube for small scale applications as cheaper solution [12]

These components are used as evaporator, condenser and IHE for ORC. While evaporator is providing the heat absorption from heat source, condenser provides heat release from cycle to environment. IHE which is used for Regenerative ORC provides internal recovery of the thermal energy from the hot vapor discharged of turbine by the regenerator. In this study, double tube pass and single shell pass type heat exchangers were chosen for both evaporator and condenser with the purpose of provide higher heat transfer area with compact components. Besides that, single shell pass and single tube pass opposite directional type heat exchanger were chosen as IHE for provide higher temperature difference between input and output.

Pump uses to pressurize the liquid phase working fluid from condensation pressure to evaporation pressure. ORC pumps are usually preferred as variable speed multistage centrifugal pumps depending on its proper design for usage of chemistry field. Pump consumes some part of turbine net output power depending on design parameters as the only moving part of cycle. Pump efficiency is significantly effective to overall efficiency especially for small scale plants rather than bigger scale applications [12].

Turbine comprise of generator integration to expander. Expander produces mechanical power as rotational motion with depending on pressure difference between input and output states. The generator converts mechanical power into electrical power and it is directly connected to the expander shaft. Expanders as mainly divided in two categories which are turbomachines and volumetric expanders.

Turbomachines contains single stage or multi stage rotor and stator configurations. High pressurized working fluids gain acceleration with help of static channels of stator. After that, accelerated stream enters the rotor and it transfer its momentum to expander shaft with deflection and expansion over expander blades where are placed in rotor. Turbomachines consists of two different types which are axial and radial depending relative motion of working fluid stream direction with respect to expander shaft. Turbomachines are commonly using for medium and large scale power plants which are power outputs in between 100 kW and 15 MW. Axial type has higher efficiency for especially small scale power plants distinctly from radial type.

Volumetric expanders are more efficient options when power output below 100 kW. Volumetric expanders have same working principal with compressors. While compressors are pressurizing the vapor with power input, volumetric expanders oppositely produce mechanical work with expanding vapor. These components are cheaper than turbomachines because of derived from refrigerant compressors. Also, volumetric expanders show more resistance with lower erosion rates to undesirable liquid stream than turbomachines. Volumetric expanders have low efficiency rates for high pressure difference. This situation constitutes a limitation for evaporator outlet pressure and it effects negatively to approaching maximum outlet power [12].

1.1 Literature Review

There are many parametric and experimental studies about heat recovery cycle application for internal combustion engine in literature. Most of studies in literature were focused on Organic Rankine Cycle usage for diesel engines. In this study, natural gas fired engine was investigated for combined system. Organic Rankine Cycle analysis for integration to ICE has different design parameters such as working fluid selection, optimization of working parameters, thermophysical properties and examination of additional components for higher efficiencies. Beginning system decisions was made depending on below literature research.

Amicabile et al. [13] have a research about organic Rankine Cycle optimization for waste heat recovery from heavy-duty diesel engine with comparison of thermodynamic cycle model, working fluid selection and working condition. They used four different types thermodynamic cycles which are subcritical cycle without recuperator, supercritical cycle without recuperator, subcritical cycle with recuperator and supercritical cycle with recuperator. Recuperator was used as closed loop internal heat exchanger for this study. They concluded that the highest power output value is achieved with using pentane based recuperative supercritical cycle. Also, they found that while pentane based recuperative cycle has 5,5 year payback time, non-recuperative pentane based cycle has 5,9 year payback time.

Wang et al. [14] investigated the performances of different working fluids which are R113, butane, R141b, R123, R11 and R245fa with using standard single loop ORC and regenerative ORC that including additional internal heat exchanger for internal combustion engine waste heat recovery. They found that the internal heat exchanger included ORC thermal efficiency is in range between 9,5-10,5% with different working fluids, while single loop cycle thermal efficiency is in range between 8,4-9,1%. Additionally, results of this research shows that the R11, R141b, R123 and R113 slightly more efficient than other working fluids.

Katsanos et al. [15] focused on a comparative research for Rankine Cycle usage with steam and Organic Rankine Cycle usage with R245ca with applied on diesel engine waste heat. They analyzed Organic Rankine Cycle with additional recuperator as a preheater differently from Rankine Cycle which is consists of only four basic components. They used Shell and Tube type heat exchangers for both evaporator and condenser. As a result of this research, Organic Rankine Cycle usage improve overall fuel save in range between 10,2-8,4% depending on different engine loads from 25% to 100%. Also, Rankine Cycle usage improvement is found in range between 6,1-7,5% for same engine load variations. They concluded that Organic Rankine Cycle operates 15 times higher mass flow rate of working fluid than Rankine Cycle as a reason of efficiency difference.

Mastrullo et al. [16] analyzed heat exchanger design for Organic Rankine Cycle with using R245fa working fluid that powered by a heavy duty diesel engine which produce 362 kW maximum power with six cylinders. Exhaust gas temperature range is indicated between 363-503°C with different engine load for them research. They proposed the shell and louvered mini-tube type heat exchanger for optimal design depending on overall efficiency increment results which is approaching to 8%.

Uusitalo et al. [17] made an experimental study on Organic Rankine Cycle application for heat recovery from charge air waste heat of internal combustion engine which has 1,6 MW maximum power output. They performed Regenerative Organic Rankine Cycle with R245fa which is classified as a hydrofluorocarbon and pentane which is classified as a hydrocarbon. Also, they revealed that the 2% raise of power output achieved at 30 °C condensation temperature and 1,7% raise at 40°C condensation temperature as congruently for both two working fluids.

A parametric analysis was carried out by Wang et al. [18] for a four cylinder light-duty diesel engine waste heat recovery system with taking advantage of dual loop Organic Rankine Cycle with two different working fluids such as R134a and R245fa. In his study, first loop was utilized for exhaust gas waste heat as high temperature loop and second loop was utilized for jacket cooling water waste heat as low temperature loop. Significance of optimization study for working parameter definition was shown depending on effective thermal efficiency increment with rate of 8% increment at maximum engine load. Also, they reported that the R245fa is proper for high temperature loop and R134a is proper for low temperature loop with the results of averagely 19-22% maximum power outlet increment of engine for different engine loads.

Yang et al. [19] evaluated the effects of main parameters which are evaporation pressure, superheat level, and condensation temperature on the efficiency of

Organic Rankine Cycle system installed to turbocharged diesel engine that consist of six cylinders and 247 kW of maximum power output at speed range between 900–1900 rpm. R245fa was used as working fluid for the Organic Rankine Cycle system. The operating conditions of engine affected at first the evaporation pressures. Operating conditions of engine showed a slight effect on superheat temperature and condensation temperature. They presented 0,74 kW/m^2 power output for per unit heat transfer area and 13,84 kW additional net power output at the maximum power of engine. They found that the optimal evaporating pressure is in range between 1000-2970 kPa for different engine speed and also they found optimal superheat temperature is in range between 0 - 3,64°K. Optimal condensation temperature was kept nearly constant at 298,15 °K depending on design temperature of outdoor air.

Song et al. [20] examined the two separated cycle with heat recovery purpose of diesel engine which is produce 996 kW power with six cylinders. Also, it is specified that the engine is producing exhaust gas at temperature of 573°K and jacket cooling water at temperature of 363°K for maximum load. They used high temperature cycle for exhaust gas and low temperature loop for jacket cooling water. While R236ea, R236fa, R600, R600a, R123, R134 and R245fa were chosen for low temperature loop cycle, cyclohexane, benzene, toluene, nonane and decane were chosen for high temperature loop cycle in the study. The analyze results brings out that the 10,2% efficiency increment was achieved with 101 kW additional net output power. Besides that, R245fa was found best working fluid for low temperature loop and benzene was found best working fluid for high temperature loop.

In study of Wang et al. [21] natural gas fired internal combustion engine was used for Regenerative Organic Rankine Cycle implementation at supercritical and subcritical conditions for recover waste heat. They proposed that the Regenerative ORC usage at supercritical conditions can significantly improve overall performance. Also, Regenerative ORC can improve system overall efficiency averagely with rate of 8% for each engine load level with using R245fa and R123zd as best working fluids depending on thermodynamic results.

Larsen et al. [22] comparatively analyzed the marine diesel engine heat recovery with Regenerative ORC and single loop Basic ORC with 24 different working fluids. Dry type working fluids (toluene, pentanes, hexanes and heptanes) was showed higher efficiencies with recuperated process while isentropic and wet type working fluids was found more effective with non-recuperated process when heat source temperature is above 300 °C.

Shengjun et al. [23] were showed that the all different working fluids has to be operated with different optimal working state points based on their research which is about thermal and parametric optimization of single stage ORC with using low-temperature geothermal power at subcritical conditions.

Optimal cycle research was made by Rech et al. [24] with using single stage and two stage Organic Rankine Cycle for liquefied natural gas fired Internal Combustion Engine. They concluded that, optimal application was changing depending on different engine loads which are 4 different load levels. They used R245fa for both cycles and as a result of research 1665,8 MWh annual additional power was achieved for single stage ORC with 6,5% peak cycle thermal efficiency at second speed of engine. Also 2306,6 MWh annual additional power output was achieved as maximum for two stage ORC system 12,6% thermal efficiency was achieved as maximum at first speed of engine.

Srinivasan et al. [25] have a parametric and small scale experimental study for analyze single stage Organic Rankine Cycle usage with dual fuel Internal Combustion Engine which is producing 568°K temperature exhaust gas. They used R113 as working fluid and heat exchanger effectiveness of ORC was identified as an important parameter. They concluded that, highest exergy efficiencies and lower pinch point temperature difference could be achieved at 0,8 and higher effectiveness values. Besides that, lower investment costs could be achieved with lower effectiveness values with lower exergy efficiencies. Also, they found that fuel conversation efficiency could be improve in range between 20-35% with using R113 and 0,7 effectiveness evaporator.

Xu and Yu [26] comparatively investigate 57 different working fluids for thermodynamic results with using Basic ORC at subcritical pressure conditions at wide waste heat source temperature range that as 100-400°C. They proposed R245fa and R141b as proper working fluids for wider temperature range than other working fluids depending on critical temperature. Also, thermal efficiency of cycle was found approximately 10% for 125°C heat resource temperature, 13,5% for 175°C, 16% for 225°C and 19% for 300°C. Cyclopentane is the most efficient working fluid with thermal efficiency of 20,4% at high temperature that 300°C depending on research.

He et al. [27] studied the application of Organic Rankine Cycle to Internal Combustion Engine with different working fluids which are cyclopentane, cyclopentene, 2,2-dimethyl butane, R11, R113, n-pentane and isopentane for evaporation temperature range that 145 °C to 220 °C. As a result of study, they found relative thermal efficiency ranges of ORC to inlet temperature of turbine accordingly, 18-21% for n-pentane, 19-23% for R113, 20-23% for R11 as highest thermal efficiency results.

Saleh et al. [28] have a research about 31 different pure organic fluids which including alkanes, fluorinated alkanes, ethers and fluorinated ethers usage at low temperatures that 100°C evaporation temperature and 30°C condensation temperature for compare the regenerative cycle to single loop cycle. They found regenerative cycle thermal efficiency is averagely 7% higher than basic cycle for dry and isentropic organic fluids. They could not analyze Regenerative ORC for wet type working fluids such as R143a, R227ea and R134a etc. because of turbine outlet temperature which is very close to condensation temperature. They observed a dramatic jump in thermal efficiency of cycle for Basic ORC from 6,11% for propane to 12,91% for n-pentane. Also, n-pentane, n-hexane, RE245 and R245fa were found as most efficient working fluids for both cycle.

Wang et al. [29] examined boiling temperature effect for 13 different working fluids with Basic Organic Rankine Cycle thermal efficiency at different evaporation temperatures. Multi-objective optimization model was generated for study based on heat source temperature which is range of 100–220°C and the pinch point temperature difference range 5–30°C. The results showed that the working fluid boiling temperature effects to optimum evaporating pressure is the main criterion for specify maximum thermal efficiency. While R123 was found as the best fluid for the 100–180°C evaporation temperature range, R141b was found as the optimal fluid with the evaporation temperature higher than 180 °C.

Kajurek et al. [30] presented working fluid selection with using ten different working fluids which are R134a, R152a, R227ea, R236fa, R245fa, R290, R600a, R717, R1234yf, R1234ze(E) for Basic ORC at 80°C evaporation temperature and 30°C condensation temperature. Thermal efficiency based calculation showed that the worst working fluid is R1234yf with 8,38% and best working fluid is R717 with 9,93%. Besides that, they choose R245fa which has 9,5% thermal efficiency for experimental part of study with considering that R245fa has lower flammable risk and GWP value than R717. They revealed that 800W power output was achieved as a result of experimental study while aiming 1000 W power output.

He et al. [31] used 21 different working fluids for determine optimal evaporation temperature according to maximum net output power of single stage ORC. They concluded optimal evaporating pressures which are 1714 kPa, 1835 kPa, 1206 kPa, 1307 kPa, 1048 kPa for working fluids respectively R600a, R142b, R114, R600 and R245fa depending on optimal evaporating temperatures. Also, working pressures were specified as in range between 300-700 kPa for R123, R601a, n-pentane, R11, R141b and R113. And as a result of his research, R114, R245fa, R123, R601a, n-pentane, R141b and R113 were identified as best working fluids with depending on cycle net output power which is in range between 9,10 kW to 9,61 kW.

Organic Rankine Cycle integration to Internal Combustion Engine as a heat recovery system was evaluated by Glover et al. [32] with using 18 different working fluids for optimal evaporating temperature independently from exhaust gas temperature which is approximately 350°C and coolant water temperature which is approximately 110°C. They were explored that the working fluids which had higher critical temperature are shown greater potential to ORC performance increment. They have reported that, n-pentane, isobutane, R245fa, butene and isopentane would be logical selection as working fluids for higher evaporating temperatures depending on cycle thermal efficiency results that in range between 12-13,2%.

Wang et al. [33] generated a maximum output power based selection chart for single stage Organic Rankine Cycle working fluids depending on different heat source temperatures with comparison of 25 different working fluids. R123, R365mfc, R601a, R601 and R141b are proper for 465-500°K, R600, R245fa, n-pentane and R245ca are ideal for 445-465°K, R600a, R142b, R236a, isobutane and butene are suitable for 395-420°K, R22, R190, R134a and R227ea are proper for 365-395°K, R143a and R32 are good for 320-365°K temperature range of heat source depending on generated chart. And also they found the highest exergy destruction rate belongs to evaporator as 50% of total destruction rate.

Mago et al. [34] have a research about performance analyses of Organic Rankine Cycle usage with different working fluids as comparison based different heat source temperatures. They found thermal efficiencies of ORC as 6% with R134a, 11% with isobutene, 13% with R245fa, 16% with R123 and 18% with R113 for 430°K and higher heat source temperatures. Besides, it is shown that, while R123, R245ca and R245fa having the best efficiencies for temperatures between 380°K and 430°K, isobutane has highest efficiency for evaporator temperatures lower than 380°K.

Sung and Kim [35] investigated Organic Rankine Cycle application with 13 different working fluids to 18 cylinder 17550 kW maximum output power internal combustion engine for both exhaust gas and jacket cooling waste heat. They used separated dual loop cycles which are high temperature cycle for exhaust gas and low temperature cycle for jacket cooling. Exhaust gas evaporator inlet and outlet temperatures was accepted respectively 373°C and 120°C for study while jacket cooling water inlet and outlet temperatures was accepted respectively 91°C and 74°C. Condensation temperature accepted 35°C for both cycle. Also, they compared that basic cycle and recuperative cycle net output work based. According to simulation results n-pentane was found optimum working fluid for high temperature loop basic cycle with 589,1 kW additional power generation. 755,5 kW additional power was achieved with R123 based recuperative high temperature cycle which is 105,4 kW higher than basic cycle.

He et al. [36] analyzed combined Organic Rankine Cycle for liquid natural gas fired internal combustion engine waste heat based on thermal efficiency results with using C4F10, CF31, R236ea, R236fa and RC318 as working fluids. Evaporator outlet temperature of working fluid was set in range between 450°K and 600°K. The highest thermal efficiencies were found as similarly 21,6% for both R236fa and R236ea and other three working fluids thermal efficiencies were found very close to highest efficiency as around 21%. Also it is showed that, averagely 14,7% improvement of fuel economy can be achieved with ORC implementation.

Tian et al. [37] compared 20 different working fluids for Organic Rankine Cycle integration to diesel Internal Combustion Engine waste heat. Evaporation pressure was defined as primary criteria for gaining maximum power output from cycle. The top three best working fluids were presented as R141b, R123 and R245fa depending on its thermal efficiency range from 13,3% to 16,6% and net output power per unit mass range from 49 kj/kg to 60 kj/kg. Also, optimum evaporation pressures were found in range between 2800 kPa and 3600 kPa for top three working fluids as a result of his study.

Miao et al.[38] compared a large number of pure fluid zeotropic mixtures with using single stage ORC and dry and isentropic working fluids are found more efficient compared to wet type working fluids. Shu et al.[39] examined parametrically the dual loop ORC thermal and exergy efficiency, and R245fa exergy efficiency range depending on evaporator temperature was found 25-35% and its thermal efficiency also was found 12-19%.

Imran et al. [40] examined thermo-economic analyze of Basic ORC single stage Regenerative ORC and double stage Regenerative ORC for five different working fluids such as R245fa, R11, R123, R134a, R141b. They showed that single stage ORC implementation is improve 1,01% of combined system overall thermal efficiency an additional 187 $/kW cost while Regenerative ORC is improving overall thermal efficiency 1.45% with 297 $/kW additional cost. Specific investment cost of system was specified depending on optimal working pressures and temperatures for both Basic and Regenerative ORC. The lowest specific investment cost was calculated as belong to R245fa, while highest specific investment cost belong to R141b for both cycle based on them research.

Le et al. [41] studied thermodynamic and economic optimization as comparison of different working fluids for industrial waste heat recovery with ORC. Thermodynamic analysis were showed that n-pentane and R245fa has highest exergy efficiency respectively 53,2% and 53%. The economic analysis result was showed that the n-pentane based ORC has the lowest investment cost and shortest payback period.

Asim et al. [42] analyzed the integration of ORC to refrigerant cycle for take advantage of heat loss with six different working fluids, such as butane, R123, R141b, R227ea, R245fa, R1233zd(e). ORC system exergy efficiencies range was found 32% to 38%. It was discovered in his study that the R123 and R141b have highest exergy efficiencies apart from that the butane and R227ea has shortest payback period.

Yang [43] focused on Payback Period reduction of Basic Organic Rankine Cycle with using different type working fluids and its mixtures for waste heat of large marine diesel engine which is produce 308°C exhaust gas with 80.080 kW power output. Firstly he specified the optimal evaporating pressure, expander inlet temperature and condenser outlet temperature for calculation of PBP. R236a, R245fa, R600 and R1234ze were used as working fluids for constitute mixtures. He found that, PBP is changing in range between 5,7-6,4 years depending on working fluid type. R600/R1234ze was showed PBP based best performance with having shorter time than respectively rate of 7,55%, 6,47%, 9%, 9,17%, 0,9%, and 2,88% for R236fa, R245fa, R600, R1234ze, R236fa/R245fa, and R236fa/R1234ze.

In paper of Li et al. [44], single stage and two-stage ORC were compared for thermodynamic and economic points of view which are net output power, efficiency, thermal conductance, volumetric flow ratio, and the economic indicators the electricity production cost (EPC), payback period (PBP) and rate of return on investment (ROROI). Decane, nonane, cyclohex, octane, heptane and i-octane were used for calculation as comparatively. As a thermodynamic result of his study single stage ORC has average thermal efficiency in range between 12,02-16,25% for evaporation temperature that in range between 200-300°C. Two-stage ORC thermal efficiency was found in range between 13,29-17,57% for same conditions. One of the most significant economic result of his study is that while two-stage ORC has 7,2 year PBP, single stage ORC has 14,79 year PBP for maximum thermal efficiency point.

Wang et al. [45] evaluated the double-pressure and single pressure ORC usage at 100°C to 160°C heat source temperature range with using isobutane. They concluded that both single and double-pressure ORC showed better thermo-economic performance at higher heat source temperature. Electricity Production Cost (EPC) based analyze was showed that, single pressure ORC has 0,24-0,13 $/kWh EPC with respectively 100-160°C heat source temperatures and double-pressure ORC has 0,24-0,14 $/kWh EPC.

Economical feasibility research of Organic Rankine Cycle usage for heat recovery from diesel engine waste heat was made by Yang et al. [46]. They aimed that determine most feasible heat recovery cycle in comparison with Basic ORC and Regenerative ORC for 4 different working fluids which are R245fa, R600, R1234ze and R600a by optimization study. Regenerative ORC is designed to have an additional Internal Heat Exchanger, which is used as a pre-heater with the cycle waste heat. They found that Payback Period (PBP) of R1234ze is 5,87 year with pre-heater and 6,52 year for without pre-heater. Also, R245fa has 5,68 year PBP with pre-heater and 6,19 year PBP for without pre-heater, R600 has 5,72 year PBP with pre-heater and 6,31 year PBP for without pre-heater, R600a has 5,79 year PBP with preheater and 6,42 year PBP for without pre-heater.

1.2 Objective of Thesis

In this study, two different Rankine Cycles which are Basic ORC and Regenerative ORC were investigated as a heat recovery system for natural gas powered internal combustion engine which is TCG2032 model manufactured by MWM brand of Caterpillar Energy Solutions GmbH. Both basic and regenerative ORC were comparatively analyzed for each working fluid's energy efficiency, exergy efficiency and overall cogeneration efficiency to provide heat recovery from engine exhaust gas heat loss. Seven different working fluids were studied in this paper as represent of fluids different classes, to maximize efficiency by determine the optimal statement points. Basic and Regenerative ORC were examined economically for most efficient working fluids as a final step of this study.

1.3 Hypothesis

Significant reduction of fuel consumption is expected with both Regenerative and Basic ORC integration to Internal Combustion Engine which is using for electricity power generation in an industrial facility in İstanbul. Integrated ORC is working as heat recovery system for waste heat of ICE and it will supply additional power generation without consuming any fuel source. Regenerative ORC will be operating more efficiently than Basic ORC with help of IHE usage as preheater. While IHE consisting additional investment cost for cogeneration system, PBP expected shorter than Basic ORC. PBP will be shorter than 10 years for both Basic and Regenerative cycles depending on literature research.

2
SYSTEM DESCRIPTION

2.1 Heat Source Parameters

Internal combustion engine which is heat recovery system will be applied, has three different working levels such as 50%, 75% and 100%. Exhaust gas conditions and engine parametric terms which are shared by the manufacturer were presented as Table 2.1 for each working level. Exhaust gas composition and specific heat for average temperature were presented at Table 2.2.

Table 2.1 ICE working parameters for different engine load

Property	Unit	Value	Value	Value
Engine Load	%	50	75	100
Exhaust Temperature	°C	510	480	453
Exhaust Mass Flow	kg/h	12268	17397	22832
Fuel Energy Consumption ($Q_{e,con}$ **)**	kW	5486	7630	9824
Thermal Efficiency	%	46,9	44,6	43,8
Electricity Production Efficiency($\eta_{e,epe}$ **)**	%	39,2	42,3	43,3

Table 2.2 Composition of exhaust gas

Composition	Volumetric Ratio (%)	Molecular Weight (kg/kmol)	Spesific Heat (kj/kg°C)
Argon (Ar)	0,1	40	0,5206
Carbondioxide (CO_2)	4,9	44	1,061
Water (H_2O)	11,2	18	2,013
Oxygen (O_2)	9,9	32	0,993
Nitrogen (N_2)	73,9	28	1,07

2.2 Heat Recovery Cycle

Basic Organic Rankine Cycle (BORC) consists of four components which are pump, evaporator, condenser and turbine. The only difference between Regenerative Organic Rankine Cycle (RORC) and BORC is RORC has Internal Heat Exchanger as additional component. BORC and RORC schematic flow diagrams were respectively presented as Figure 2.1(a) and Figure 2.1(b). Cycle loop begin with pump which is the only moving part of cycle. Pump pressurized the liquid and provides liquid supply to evaporator. Evaporator vaporized the liquid until working fluid reaches saturated vapor or super-heated vapor depending on design. After that, turbine produces power by expanding the vapor to condenser pressure. Expanded vapor goes into condenser and phase change occurs at constant pressure with heat rejection. Liquid outlet of condenser goes into pump and by this way; closed loop cycle was obtained for BORC. Regenerative ORC include an additional closed loop internal heat exchanger (IHE) as mentioned before. IHE was used as preheater for low energy level condenser outlet by using high energy level turbine outlet as heat source. It is expected that the IHE will improve efficiency level but also negative effect to investment cost.

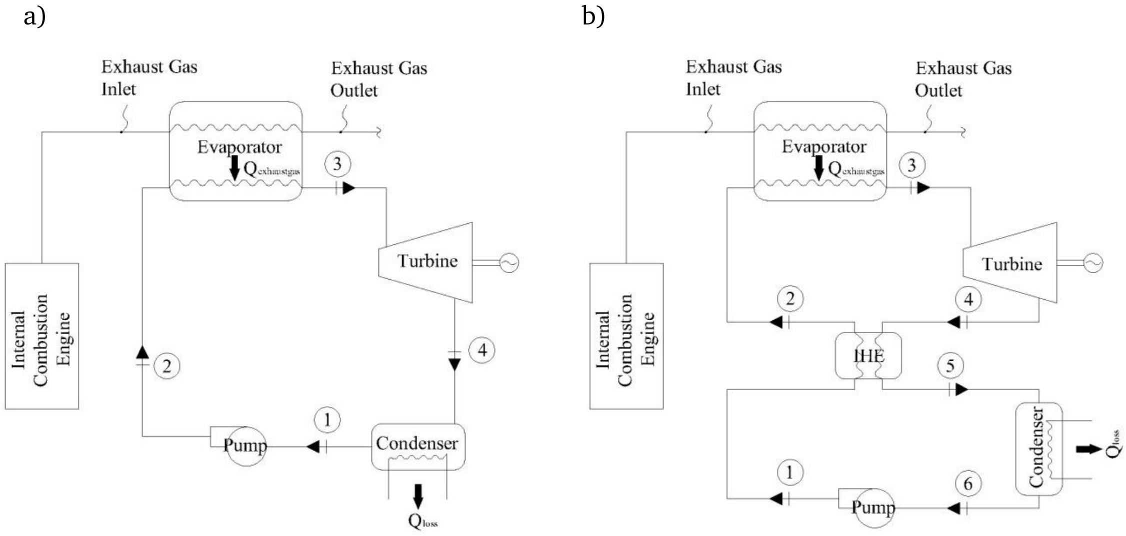

Figure 2.1 Schematic flow diagram of cycles

2.3 Design Parameters

Heat source capacity is the most significant design parameter for a power plant. In this study, exhaust gas of an IHE was used as heat source and exhaust gas contents water vapor and CO_2. Funnel wall or evaporator outside surface can be damaged with acidic water generation in case of exhaust gas condensation. Evaporator was designed based on 120 °C exhaust gas outlet temperature and by this way, it is assumed that there is no acidic condensation at funnel or evaporator. Exhaust gas capacity was found depending on temperature difference between inlet and outlet with using specific heat and mass flow rate. Specific heat of exhaust gas was calculated with the contents mass ratio which is presented at Table.2 for average temperature. Average temperature of exhaust gas assumed 300 °C and specific heat was calculated 1.116 kj/kg°C. Pump has to be pressurized liquid to under subcritical pressure of working fluid. Risks of reaching critical pressure were expressed in detail at next steps of study. Heat sink of ORC is coolant water which is produce by cooling tower. Coolant water temperature regime accepted as 20-30 °C by considering summer conditions. Because of that, condenser outlet temperature assumed 35 °C. Also it is assumed that the condenser outlet is saturated liquid which constitutes another limitation parameter such as turbine expansion pressure. There are various isentropic efficiencies which are in range between 65%-75% for turbine and in range between 75%-85% for pumps depending on literature [8] [14]. Isentropic efficiency was assumed 65% at turbine and 75% at pump for more realistic approach.

All assumptions:

- Exhaust gas evaporator outlet temperature: 120 °C
- Specific heat of the exhaust gas (c_p): 1,116 kj/kg°C
- Condenser outlet temperature: 35 °C
- Condenser outlet is saturated liquid.
- Isentropic efficiency of pump (η_{pump}): 0,75
- Combined Isentropic and generator efficiency of turbine ($\eta_{turbine}$): 0,65
- Pressure drop in pipe, evaporator, condenser and IHE was neglected.
- Components were accepted adiabatic.
- The system operates steady state.

2.4 Working Fluid Parameters

Working fluid selection is directly effective to ORC system design and feasibility researches. Every working fluid has different physical and chemical characteristics which are highly important for environmental based terms and beside that, also thermodynamic characteristics are highly important for cycle type, components design, system maximum efficiency and optimization. First of all, there are basically three different organic fluids types which are dry, isentropic and wet type depending on saturation curve slope. Working fluids were observed for chemically conditions (stability under operating conditions, non-flammable, non-explosive, non-toxic), environmentally conditions (low GWP, low ODP) and thermodynamically conditions (efficient, suitable to conditions) as presented at Table 2.3.

Table 2.3 Working fluids properties

WF	Formula	MW	[1]T_{crit}	[2]P_{crit}	[3]ALT	[4]GWP	[5]ODP	[6]A 34
n-pentane	C_5H_{12}	72,15	196,56	3370	12	4	0	A3
n-butane	C_4H_{10}	58,12	152,01	3796	12	3	0	A3
R113	$C_2F_3Cl_3$	187,38	214,06	3392	85	6130	0,9	A1
R123	$C_2HF_3Cl_2$	152,93	183,83	3672	1,3	120	0,012	B1
R141b	$C_2H_3FCl_2$	116,95	206,96	4212	9,3	700	0,086	A2
R245fa	$C_3H_3F_5$	134,05	154,20	3651	7,2	950	0	B1
R134a	$C_2H_2F_4$	102,03	101,06	4059	14	1300	0	A1

Listed working fluids at Table 3 were chosen for next steps of study as the most commercial and most popular fluids depending on literature research. Combined T-S diagram of selected working fluid shown at Figure 2.2.

[1]T_{crit}: critical temperature (kPA)

[2]P_{crit}: critical pressure (°C)

[3]ALT: atmospheric life-time (years)

[4]GWP: global warming potential, relatively CO2 (100 years)

[5]ODP: ozone depletion potential, relatively R11

[6]A 34: Ashrae 34 Classification: 1:non-flammable, 2:low-flammable, 3:high-flammable;A:lower toxicity, B:higher toxicity

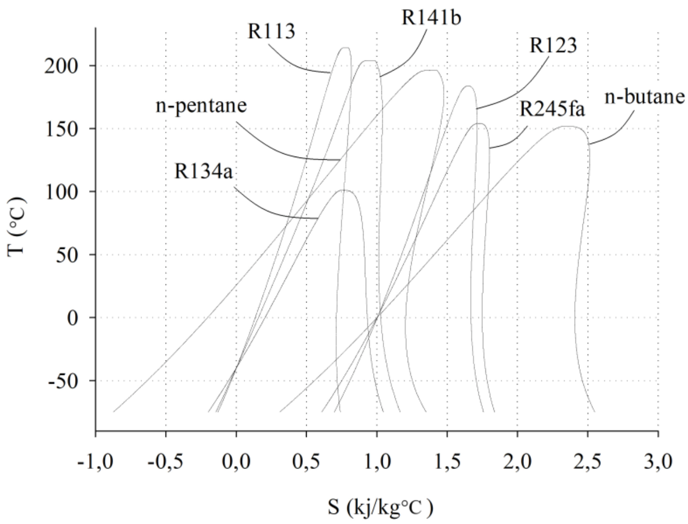

Figure 2.2 T-S Diagrams of selected working fluids

3
MATHEMATICAL MODEL

3.1 Thermodynamic Equations

Thermodynamic equations constitute the base of all mathematical models, balance equations and efficiency terms calculations. Enthalpy and exergy values are also base for efficiency terms. Enthalpy and exergy values were found for all state points with using temperature and pressure. Temperatures and pressures of all statement points are known depending on assumptions which are mentioned at design parameters and independent variables which will be specified at optimization step.

Waste heat of engine can be expressed as follows:

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} c_p (T_{ei} - T_{eo}) \tag{3.1}$$

Working fluid mass flow rate can be expressed as follows:

$$\dot{m}_{wf} = \frac{\dot{Q}_{exhaust}}{q_{evaporator}} \tag{3.2}$$

Evaporator heat intake per unit mass can be expressed as follows:

$$\dot{Q}_{evaporator} = \dot{m}_{exhaust} \left(h_{oe} - h_{ie} \right) \tag{3.3}$$

General Exergy terms for each state point can be expressed as follows:

$$ex_i = (h_i - h_0) - T_0 (s_i - s_0) \tag{3.4}$$

General Exergy destruction terms for each state point can be expressed as follows:

$$\dot{E}x_d = T_0 \dot{S}_{gen} \tag{3.5}$$

General thermal exergy terms for each state point can be expressed as follows:

$$\dot{E}x^{Q_i} = \dot{Q}_i(1 - \frac{T_0}{T_s}) \tag{3.6}$$

Net output power of turbine can be expressed as follows:

$$\dot{W}_{net} = \dot{W}_{tubine} - \dot{W}_{pump} \tag{3.7}$$

Net electricity production of ICE can be expressed as follows:

$$\dot{W}_{need} = \dot{Q}_{e,cons}\eta_{e,epe} \tag{3.8}$$

3.2 Balance Equations

Balance equations were developed as separately for both basic and regenerative ORC as respectively presented at Table 3.1 and Table 3.2 depending on assumptions. System friction and heat loss of pipes, kinetic and potential energy changes were accepted negligible. Balance calculations are performed for each component with consideration of enthalpies, exergies, mass flow rates, pressures and temperatures

Table 3.1 Balance equations for Regenerative ORC

Equation Type	Component	Balance Equation
Mass Balance Equations(M)	Overall System	$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_{wf,r}$
Energy Balance Equation (E)	Turbine	$\dot{W}_{turbine} = \dot{m}_4 h_4 - \dot{m}_3 h_3$
	Condenser	$\dot{Q}_{loss} = \dot{m}_6 h_6 - \dot{m}_5 h_5$
	Pump	$\dot{m}_1 h_1 - \dot{m}_6 h_6 = \dot{W}_{pump}$

Table 3.1 Balance equations for Regenerative ORC (continued)

Equation Type	**Component**	**Balance Equation**
Energy Balance Equation (E)	Evaporator	$\dot{m}_3 h_3 - \dot{m}_2 h_2 = \dot{m}_{wf,r} q_{evaporator}$ $\dot{m}_{wf,r} q_{evaporator} = \dot{Q}_{exhaust}$
	IHE	$\dot{m}_4 h_4 + \dot{m}_1 h_1 = \dot{m}_5 h_5 + \dot{m}_2 h_2$
	Overall System	$\dot{Q}_{exhaust} + \dot{W}_{pump} = \dot{Q}_{loss} + \dot{W}_{turbine}$
Exergy Balance Equation (Ex)	Turbine	$\dot{m}_3 ex_3 = \dot{m}_4 ex_4 + \dot{W}_{turbine} + \dot{E}x_{d,turbine}$
	Condenser	$\dot{m}_5 ex_5 = \dot{m}_6 ex_6 + \dot{E}x^{Q_{loss}} + \dot{E}x_{d,condenser}$
	Pump	$\dot{m}_6 ex_6 + \dot{W}_{pump} = \dot{m}_1 ex_1 + \dot{E}x_{d,pump}$
	Evaporator	$\dot{m}_2 ex_2 + \dot{E}x^{Q_{exhaust}} = \dot{m}_3 ex_3 + \dot{E}x_{d,evaporator}$
	IHE	$\dot{m}_4 ex_4 + \dot{m}_1 ex_1 = \dot{m}_5 ex_5 + \dot{m}_2 ex_2 + \dot{E}x_{d,IHE}$
	Overall System	$\dot{E}x^{Q_{exhaust}} + \dot{W}_{pump} = \dot{E}x^{Q_{loss}} + \dot{W}_{turbine} + \dot{E}x_{d,overall}$
Entropy Balance Equation (En)	Turbine	$\dot{m}_3 s_3 + \dot{S}_{gen,turbine} = \dot{m}_4 s_4$
	Condenser	$\dot{m}_5 s_5 + \dot{S}_{gen,condenser} = \dot{m}_6 s_6 + \frac{\dot{Q}_{loss}}{T_b}$
	Pump	$\dot{m}_6 s_6 + \dot{S}_{gen,pump} = \dot{m}_1 s_1$

Table 3.1 Balance equations for Regenerative ORC (continued)

Equation Type	Component	Balance Equation
Entropy Balance Equation (En)	Evaporator	$\dot{m}_2 s_2 + \frac{\dot{Q}_{exhaust}}{T_{es,wf}} + \dot{S}_{gen,evaporator} = \dot{m}_3 s_3$
	IHE	$\dot{m}_4 s_4 + \dot{m}_1 s_1 + \dot{S}_{gen,IHE} = \dot{m}_5 s_5 + \dot{m}_2 s_2$
	Overall System	$\frac{\dot{Q}_{exhaust}}{T_{es,wf}} + \dot{S}_{gen,overall} = \frac{\dot{Q}_{loss}}{T_b}$

Table 3.2 Balance equations for Basic ORC

Balance Equation Type	Component	Balance Equation
Mass Balance Equations (M)	Overall System	$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_{wf,b}$
Energy Balance Equation (E)	Turbine	$\dot{W}_{turbine} = \dot{m}_4 h_4 - \dot{m}_3 h_3$
	Condenser	$\dot{Q}_{loss} = \dot{m}_1 h_1 - \dot{m}_4 h_4$
	Pump	$\dot{m}_2 h_2 - \dot{m}_1 h_1 = \dot{W}_{pump}$
	Evaporator	$\dot{m}_3 h_3 - \dot{m}_2 h_2 = \dot{m}_{wf,b} q_{evaporator}$ $\dot{m}_{wf,b} q_{evaporator} = \dot{Q}_{exhaust}$
	OverallSystem	$\dot{Q}_{exhaust} + \dot{W}_{pump} = \dot{Q}_{loss} + \dot{W}_{turbine}$

Table 3.2 Balance equations for Basic ORC (continued)

Balance Equation Type	Component	Balance Equation
Exergy Balance Equation (Ex)	Turbine	$\dot{m}_3 ex_3 = \dot{m}_4 ex_4 + \dot{W}_{turbine} + \dot{E}x_{d,turbine}$
	Condenser	$\dot{m}_4 ex_4 = \dot{m}_1 ex_1 + \dot{E}x^{Q_{loss}} + \dot{E}x_{d,condenser}$
	Pump	$\dot{m}_1 ex_1 + \dot{W}_{pump} = \dot{m}_2 ex_2 + \dot{E}x_{d,pump}$
	Evaporator	$\dot{m}_2 ex_2 + \dot{E}x^{Q_{exhaust}} = \dot{m}_3 ex_3 + \dot{E}x_{d,evaporator}$
	Overall System	$\dot{E}x^{Q_{exhaust}} + \dot{W}_{pump} = \dot{E}x^{Q_{loss}} + \dot{W}_{turbine} + \dot{E}x_{d,overall}$
Entropy Balance Equation (En)	Turbine	$\dot{m}_3 s_3 + \dot{S}_{gen,turbine} = \dot{m}_4 s_4$
	Condenser	$\dot{m}_4 s_4 + \dot{S}_{gen,condenser} = \dot{m}_1 s_1 + \frac{\dot{Q}_{loss}}{T_b}$
	Pump	$\dot{m}_1 s_1 + \dot{S}_{gen,pump} = \dot{m}_2 s_2$
	Evaporator	$\dot{m}_2 s_2 + \frac{\dot{Q}_{exhaust}}{T_{es,wf}} + \dot{S}_{gen,evaporator} = \dot{m}_3 s_3$
	Overall System	$\frac{\dot{Q}_{exhaust}}{T_{es,wf}} + \dot{S}_{gen,overall} = \frac{\dot{Q}_{loss}}{T_b}$

3.3 Efficiency Terms

Thermal efficiency and second law efficiencies are important deterministic criteria for optimization of cycle. Efficiency terms of Regenerative ORC were calculated with the equations that presented at Table 3.3. In addition to this, efficiency terms of Basic ORC were calculated with the equations that presented at Table 3.4. On the other hand, ICE and ORC combined system overall efficiency terms is most significant output parameter of the system. Feasibility research was comprised depending on cogeneration system overall recovery terms

Overall efficiency of cogeneration system can be expressed as follows:

$$\eta_{cogen} = \frac{\dot{W}_{need} + \dot{W}_{net}}{\dot{Q}_{e,con}} \tag{3.9}$$

Reduction of ICE fuel energy consumption can be expressed as follows:

$$\dot{Q}_{e,red} = \dot{Q}_{e,con} - \frac{\dot{W}_{need}}{\eta_{cogen}} \tag{3.10}$$

Reversible work is represents the maximum power output which could be provided at working conditions. It can be expressed as follows:

$$\dot{W}_{rv} = \dot{Q}_{loss}(1 - \frac{T_b}{T_{s,ex}}) \tag{3.11}$$

Cogeneration heat recovery efficiency is an indication that shows system heat recovery amount how close to the maximum possible heat recovery amount. Cogeneration system heat recovery efficiency can be expressed as follows:

$$\eta_{hrv} = \frac{\dot{W}_{net} + \dot{W}_{need}}{\dot{W}_{rv} + \dot{W}_{need}} \tag{3.12}$$

Table 3.3 Efficiency terms of Regenerative ORC

Component	First Law Efficiency	Second law efficiency
Pump	$\eta_{I,pump} = \frac{h_1 - h_6}{w_{pump}}$	$\eta_{II,pump} = \frac{ex_1 - ex_6}{w_{pump}}$

Table 3.3 Efficiency terms of Regenerative ORC (continued)

Component	First Law Efficiency	Second law efficiency
Evaporator	$\eta_{I,evaporator} = \frac{h_3 - h_2}{q_{evaporator}}$	$\eta_{II,evaporator} = \frac{ex_3 - ex_2}{q_{evaporator}(1 - \frac{T_{es,wf}}{T_{es,ex}})}$
Condenser	$\eta_{I,condenser} = \frac{q_{loss}}{h_5 - h_6}$	$\eta_{II,condenser} = \frac{q_{loss}(1 - \frac{T_b}{T_{cs,wf}})}{ex_5 - ex_6}$
IHE	$\eta_{I,IHE} = \frac{h_2 - h_1}{h_4 - h_5}$	$\eta_{II,IHE} = \frac{ex_2 - ex_1}{ex_4 - ex_5}$
Overall System	$\eta_{I,overall} = \frac{w_{net}}{q_{evaporator}}$	$\eta_{II,overall} = \frac{w_{net}}{w_{rv}}$

Table 3.4 Efficiency terms of Basic ORC

Component	First Law Efficiency	Second law efficiency
Pump	$\eta_{I,pump} = \frac{h_2 - h_1}{w_{pump}}$	$\eta_{II,pump} = \frac{ex_2 - ex_1}{w_{pump}}$
Evaporator	$\eta_{I,evaporator} = \frac{h_3 - h_2}{q_{evaporator}}$	$\eta_{II,evaporator} = \frac{ex_3 - ex_2}{q_{evaporator}(1 - \frac{T_{es,wf}}{T_{es,ex}})}$
Condenser	$\eta_{I,condenser} = \frac{q_{loss}}{h_4 - h_1}$	$\eta_{II,condenser} = \frac{q_{loss}(1 - \frac{T_b}{T_{cs,wf}})}{ex_4 - ex_1}$
Overall System	$\eta_{I,overall} = \frac{w_{net}}{q_{evaporator}}$	$\eta_{II,overall} = \frac{w_{net}}{w_{rv}}$

3.4 Component Design

Component design contains different effective physical parameters and assumptions. Pump and turbine design physical parameters are inlet pressure, outlet pressure and total mass flow rate of working fluid. Turbine and pump physical parameters were calculated depending on thermodynamic and balance equations which are mentioned at stage 4.1. and 4.2. Turbine type and pump type assumptions were made as proper to commercially logical options. Axial turbine accepted as expander type and close coupled pump accepted as centrifugal pump type by considering suitability to low grade power plant. On the other hand, evaporator, condenser and IHE as basically are heat exchangers. Evaporator is a heat exchanger that was used for heat recovery from ICE exhaust gas to ORC. Condenser is a heat exchanger which was used for condensation of working fluids via heat transfer from ORC to heat sink. IHE is a heat exchanger which was used for heat recovery from turbine outlet to evaporator inlet of cycle. Heat exchanger design contains independent variables separately from system such as co-current condition, flow direction, fluid velocity. All mentioned independent variables affect the heat exchanger size and heat transfer efficiency. Shell & Tube type heat exchangers were selected as most commercial and simple implemented heat exchangers.

Shell & Tube heat exchanger total heat transfer area can be express as follows:

$$A_{ht,x} = \frac{Q_x}{U_x F_{t,x} \Delta T_{lm,x}} \tag{3.13}$$

Where Q_x is the heat transfer rate of heat exchanger, U_x is the overall heat transfer coefficient, ΔT_{lm} is the logarithm mean temperature difference between separated flows, A_{ht} is the total heat transfer surface area, F_t is the correction factor for temperature depending on co-current and flow direction. Different components are represented with x subscript.

Logarithm mean temperature difference can be express as follows:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \tag{3.14}$$

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{3.15}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{3.16}$$

Where $T_{h,i}$ and $T_{h,o}$ represent respectively hot fluid inlet and outlet temperatures, $T_{c,i}$ and $T_{c,o}$ represent respectively cold fluid inlet and outlet temperatures. When the hot fluid outlet temperature equal to cold fluid inlet temperature, ΔT_1 has to be calculated depending on temperature difference between $T_{h,i}$ and $T_{c,i}$ then, ΔT_2 has to be calculated depending on temperature difference between $T_{c,i}$ and $T_{c,o}$.

Overall heat transfer coefficient of heat exchanger can be express as follows:

$$\frac{1}{U_x} = \frac{1}{h_{i,x}} \frac{d_{o,x}}{d_{i,x}} + F_{fi,x} \frac{d_{o,x}}{d_{i,x}} + \frac{d_{o,x} \ln(d_{o,x}/d_{i,x})}{2\lambda_{m,x}} + F_{fo,x} + \frac{1}{h_{o,x}} \tag{3.17}$$

Where h_i and h_o are the respectively inside surface and outside surface convection heat transfer coefficient, d_i and d_o are respectively inside and outside diameters of tube, F_{fi} and F_{fo} are respectively inside and outside surface fouling factors of tube and λ_m is the thermal conductivity of tube depending on material type. Correction factors of given equations were represented at Table 3.5 for all components.

Table 3.5 Correction factor of heat exchangers

Component	F_t	F_{fi}	F_{fo}
Basic ORC Evaporator	0,83	0,0003	0,0004
Basic ORC Condenser	0,97	0,0003	0,0002
Regenerative ORC Evaporator	0,83	0,0003	0,0004

Table 3.5 Correction factor of heat exchangers (continued)

Component	F_t	F_{fi}	F_{fo}
Regenerative ORC Condenser	1	0,0003	0,0002
Internal Heat Exchanger	1	0,0002	0,0002

Inside and outside surface heat transfer coefficients are relative to fluid velocity, thermal conductivity and fluid density. Fluid velocity affects to flow characteristics such as turbulent flow and laminar flow. Turbulent flow is desired to maximize heat transfer coefficient by increasing of fluid velocity. Also, pressure drop in heat exchanger is increasing with the increment of the fluid velocity. Because of that, fluid velocity was accepted as slightly above critical point for optimal working condition. Critical point is representing a limitation for transition point to turbulent flow. Below equations were indicated considering turbulent flow conditions. Exhaust gas thermal conductivity and density were assumed as equal to air parameters at 1 bar and at average temperature in the evaporator [47].

Heat transfer coefficient of tube outside wall (shell side) can be expressed as follows:

$$h_{o,x} = \frac{j_{h,K}\lambda_{f,x}\,\mathrm{Re}_{o,x}\,\mathrm{Pr}^{1/3}\left(\mu_x / \mu_{s,x}\right)^{0,14}}{d_{e,x}} \tag{3.18}$$

$$\mathrm{Re}_{o,x} = \frac{V_{f,x} d_{e,x} \rho_{f,x}}{\mu_x} \tag{3.19}$$

$$d_{e,x} = \frac{1,27}{d_{o,x}}(t_{l,x}^2 - 0,785 d_{o,x}^2) \tag{3.20}$$

$$V_{f,x} = \frac{\dot{m}_{f,x} t_{l,x}}{\rho_{f,x}(t_{l,x} + d_{0,x}) e_x D_{s,x}} \tag{3.21}$$

Where d_e is the equivalent diameter of shell, μ_x and $\mu_{s,x}$ are dynamic viscosity respectively at average temperature and surface temperature, λ_f is the thermal conductivity of fluid, $\Pr$ is the prandtl number, $j_{h,K}$ is the non-dimensional thermal factor depending on Kern method, V_f is the velocity of fluid, ρ_f is the density of fluid, d_o is the outside diameter of tube, t_t is the distance between center of pipes in tube bundle, e is the distance between mixing plates, $D_{s,x}$ is the inside diameter of shell.

Heat transfer coefficient changes depending on boiling or condensing situations. Liquid current in two different phases affects the heat transfer coefficient. This situation is not important for IHE which has single phase liquid current but evaporator and condenser contains two different phase fluid current.

Heat transfer coefficient at inside of tube for liquid current can be expressed as follows:

$$h_{i,l} = \frac{0,023\lambda_{i,f} \operatorname{Re}_{i,f}^{0,8} \operatorname{Pr}_{i,f}^{0,4}}{d_i} \tag{3.22}$$

$$\operatorname{Re}_{i,f} = \frac{V_{i,f} d_i \rho_{i,f}}{\mu} \tag{3.23}$$

$$V_{i,f} = \frac{4\dot{m}_{i,f}}{\rho_{i,f} n_l \pi d_i^2} \tag{3.24}$$

$$n_l = \frac{4\dot{m}_{i,f}}{V_i \rho_{i,f} \pi d_i^2} \tag{3.25}$$

Where V_i is the desired maximum velocity of fluid, $V_{i,f}$ is the actual velocity of fluid, d_i is the inside diameter of tube, n_l is the tube number in one direction tube bundle. Inner side of tube was represented with i subscript.

Heat transfer coefficient at outside wall of tube for condensation at vertical tubes can be expressed as follows:

$$h_{o,condenser} = 0,926\lambda_l \left[\frac{\rho_l(\rho_l - \rho_g)g}{\mu_l \Gamma_l} \right]^{1/3} \tag{3.26}$$

$$\Gamma_l = \frac{\dot{m}_{o,f}}{n_l \pi d_o} \tag{3.27}$$

$$\mathrm{Re}_{o,f} = \frac{4\Gamma_l}{\mu_l} \tag{3.28}$$

Where λ_l is the thermal conductivity of condensed liquid, ρ_l and ρ_g are density respectively of condensed liquid and vapor, g is the gravitational acceleration, μ_l is the dynamic viscosity of condensed liquid, Γ_l is the condensed liquid mass flow rate for per length unit of vertical tube. Equation (3.26) could be used when the $\mathrm{Re}_{o,f}$ value is under 2000.

Heat transfer coefficient calculation of evaporator inside tube was made by iteration method depending on first overall heat transfer coefficient guess. First of all, total heat transfer area was calculated depending on first guess with Equation 13. After that, inside tube heat transfer coefficient is calculated with following equations:

$$q_{i,evaporator} = \frac{Q_{evaporator}}{A_{guess}} \tag{3.29}$$

$$L_{total} = \frac{A_{guess}}{\pi d_d} \tag{3.30}$$

$$n_{evaporator} = \frac{L_{total}}{l} \tag{3.31}$$

$$h_{i,evaporator} = 7,96\left(T_{i,t} - T_{i,s}\right)^3 p^{0,4} \tag{3.32}$$

Where $T_{i,t}$ is the saturation temperature, $T_{i,s}$ is the tube wall temperature, p is the working pressure in bar, L_{total} is the total tube length, l is the desired tube length for one U type pipe, $q_{i,evaporator}$ is the heat flux.

Total heat transfer coefficient was calculated with using inside heat transfer coefficient value which was calculated depending on guess. If calculated total heat transfer value equal to guessed value, inside heat transfer coefficient can be accepted as true. If it is not equal, another guess is made depending on difference between calculated and estimated value until result become equal to guess. Equation (3.26) could be used when the $q_{i,evaporator}$ value is between 3000 and 63000.

While evaporator and condenser were accepted as double tube-pass and single shell-pass type heat exchanger as schematically presented at Figure 3.1 (a) and (b), IHE was accepted as single pass counter-flow type heat exchanger as schematically presented at Figure 3.1 (c).

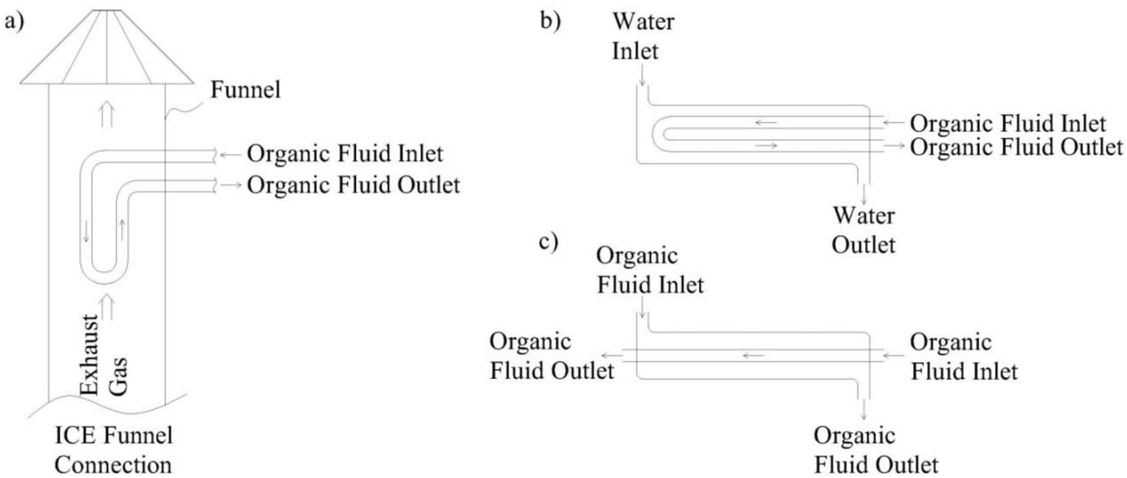

Figure 3.1 Schematic figures of heat exchangers

3.5 Economic Model

Economic model was developed with using Net Present Value (NPV) method for operating cost, maintenance cost, fuel energy save income and scrap income.

Total investment cost of components can be expressed as follows:

$$C_{inv} = C_{inv,evaporator} + C_{inv,pump} + C_{inv,turbine} + C_{inv,condenser} + C_{inv,IHE} \tag{3.33}$$

Where C_{inv} is the total investment cost and all other components are specified as subscript of it. NPV method was applied for yearly periods in the cycle life time. Cogeneration system total profit was calculated with total income of fuel energy save.

Total fuel energy consumption reduction of ICE can be expressed as follows:

$$Q_{red,tot} = Q_{e,red,\%100} t_{op,\%100} + Q_{e,red,\%75} t_{op,\%75} + Q_{e,red,\%50} t_{op,\%50} \tag{3.34}$$

Where $Q_{red,tot}$ is the total fuel energy consumption reduction per year (kWh/year), $t_{op,\%100}, t_{op,\%75}, t_{op,\%50}$ are ORC total operating hour at respectively %100, %75,%50 engine loads per year (hour/year), $Q_{e,red,\%100}$, $Q_{e,red,\%75}$, $Q_{e,red,\%50}$ are fuel energy consumption reduction per hour at respectively %100, %75,%50 engine loads.

Total profit of cogeneration system (TPC) was calculated with NPV method for Cycle Life Time (LT) as follows:

$$TPC = \sum_{t=0}^{LT} \frac{Q_{red,tot} C_f (1+r)^t}{(1+i)^t} \tag{3.35}$$

Where C_f is the cost of fuel energy (\$/kWh), r is the inflation rate and i is the annual loan interest rate. Yearly operating and maintenance cost of system was calculated with the operating cost rate depending on cycle total investment cost.

Total operating and maintenance costs can be expressed as follows:

$$C_{op} = \sum_{t=0}^{LT} \frac{C_{inv} R_{op} (1+r)^t}{(1+i)^t} \tag{3.36}$$

Where C_{op} net present value of total operating and maintenance cost, R_{op} is the operating and maintenance cost rate which is accepted as 2,5% [48]. Cycle Life Time (LT) accepted 15 year [44]. Scrap Cost (SC) of system was calculated depending on scrap cost rate relative to investment cost with using NPV method as follows:

$$C_{sc} = \frac{C_{inv} R_{sc} (1+r)^t}{(1+i)^t} \tag{3.37}$$

Where C_{sc} is the scrap cost net present value, R_{sc} is the scrap cost rate which is accepted 10%.

Overall Total Profit of cogeneration system (TP) within Cycle Life Time can be expressed as follows:

$$TP = TPC + C_{sc} - C_{op} - C_{inv} \tag{3.38}$$

Payback Period (PBP) of system is in the year which TP is zero.

3.6 Estimation of Investment Cost

Investment cost of system was analyzed based on bare module investment cost. Cost of each component for 2001 year was calculated with using Chemical Engineering Plant Cost Index (CEPCI) [49].

Updated cost can be expressed as follows:

$$C_{inv,x} = C_{M,x} \left(\frac{CEPCI_{2018}}{CEPCI_{2001}} \right) \tag{3.39}$$

Where $C_{inv,x}$ is the updated investment cost of component which represented with x subscript, $C_{M,x}$ is the cost of components for 2001 year. $CEPCI_{2001}$ represent the cost index for 2001 year which is 397 and $CEPCI_{2018}$ represent the cost index for 2018 year which is 648,7 [44].

Carbon steel based component cost for 2001 year can be expressed as follows:

$$\log(C_{k,x}) = K_{1,x} + K_{2,x} \log(Z) + K_{3,x} \left[\log(Z)\right]^2 \tag{3.40}$$

$$\log(F_{k,x}) = C_{1,x} + C_{2,x} \log(P) + C_{3,x} \left[\log(P)\right]^2 \tag{3.41}$$

$$C_{M,x} = C_{k,x} \left(\beta_{1,x} + \beta_{2,x} F_{r,x} F_{k,x} \right) \tag{3.42}$$

Where Z represents total heat transfer area for IHE, condenser and evaporator while representing total capacity for pump and turbine. K, C, β and F are constant coefficients which are presented as Table 3.6. P is working pressure of component (bar).

Turbine cost calculation can be expressed as follows:

$$C_{M,turbine} = C_{k,turbine}(F_{k,turbine}F_{r,turbine}) \tag{3.43}$$

Table 3.6 Constant values of investment cost calculation

x	Z	K_1	K_2	K_3	C_1	C_2	C_3	β_1	β_2	F_r
Turb	kW	3,514	0,598	0	0	0	0	0	0	3,4
Pump	kW	3,389	0,0536	0,153	-0,393	0,395	-0,0022	1,89	1,35	1,6
Evap	m^2	4,324	-0,303	0,163	0,0388	-0,112	0,0818	1,63	1,66	1,4
Cond	m^2	4,324	-0,303	0,163	0,0388	-0,112	0,0818	1,63	1,66	1,4
IHE	m^2	4,324	-0,303	0,163	0,0388	-0,112	0,0818	1,63	1,66	1,4

4
THERMODYNAMIC OPTIMIZATION

In this section, thermal output parameters were comparatively examined to identify optimal independent state point conditions such as evaporating pressure, superheat temperature and IHE effectiveness for both Basic and Regenerative ORC. While basic ORC system was investigated for seven different working fluids, Regenerative ORC was investigated for six different working fluids except R134a. There is not IHE usage need for R134a because of turbine outlet temperature was found very close to condensation temperature. As mentioned before, IHE was used as preheater depending on temperature difference between turbine outlet and condenser outlet. Figure 4.1 and Figure 4.2 were generated as a comparison of R134a (wet type fluid) and R113 (dry type fluid) for 35°C condensation temperature with using Basic ORC.

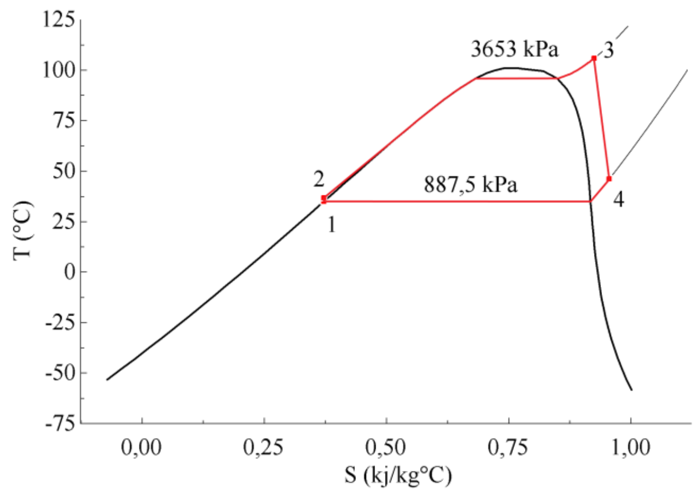

Figure 4.1 T-S Diagram of Basic ORC with R134a

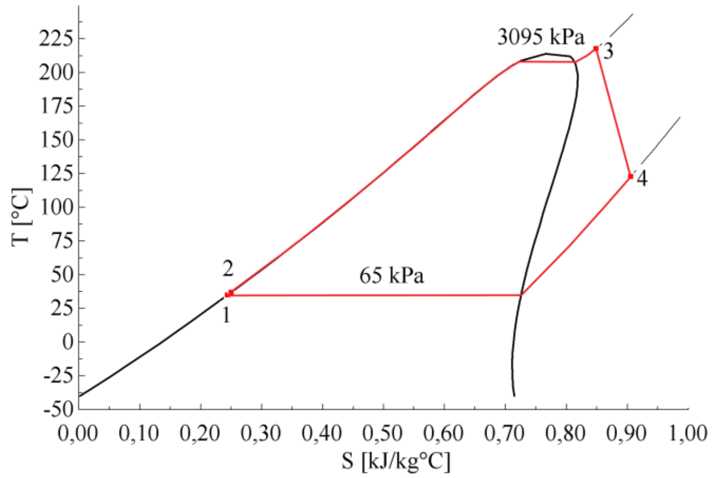

Figure 4.2 T-S Diagram of Regenerative ORC with R113

Mathematical model was developed with all assumptions and limitations by using Energy Equation Solver (EES) and also it is used for all calculations

4.1 Evaporating Pressure Optimization

Figure 4.3 and 4.4 were generated to identify optimum evaporation pressure of both Basic and Regenerative ORC. Superheat temperature was accepted 10 °C for now. Superheat temperature will be investigated at the next step in optimization section.

Basic ORC results were represented at Figure 4.3. Figure 4.3(a) shows that the net output power of turbine increasing with the increment of the evaporating pressure for all different working fluids. Figure 4.3(b) and 4.3(c) respectively represents the first law and second law efficiency change with evaporation pressure. It can be seen that, the both first law and second law efficiencies are increasing with increment of evaporator pressure for all different working fluids. Figure 4.3(d) demonstrates the reduction of fuel consumption of ICE for different evaporator pressures. Reduction of fuel consumption is increasing with increment of evaporator pressure for all different working fluids as shown in Figure 4.3(d).

Figure 4.4 demonstrates various thermodynamic results relative to the evaporation pressure for Regenerative ORC. IHE effectiveness (ΔIHE) was accepted presently 5 °C. It can be seen in Figure 4.4(a) that, the net output power of turbine increases with increment of the evaporating pressures for all different working fluids, which is similar to Figure 4.3 (a). Evaporation pressure increment is positively effective to thermal and exergy efficiency of system for all working fluids as shown in Figure 4.4(a) and (b). Figure 4.4(d) represents that ICE fuel consumption is reducing with increment of evaporator pressure.

As seen at Figure 4.3 and 4.4, increasing of evaporating pressure has positive effect to cycle's thermal output parameters. But there is a critical point limitation for all working fluids. It is hazardous that pass over critical pressure especially for delicate and expensive equipment such as turbine. There is not vapor-liquid balance above critical pressure point. The other fact remains that, the thermal output parameters are increasing very slightly with approaching critical pressure. If error margin of real systems are taking consideration, it is dangerous to approach critical pressure. It will be logical to choose different optimum evaporator pressures for all working fluids depending on critical pressure. The result of that, it is proper to choose 90 percent of critical pressures' as evaporating pressures for all working fluids.

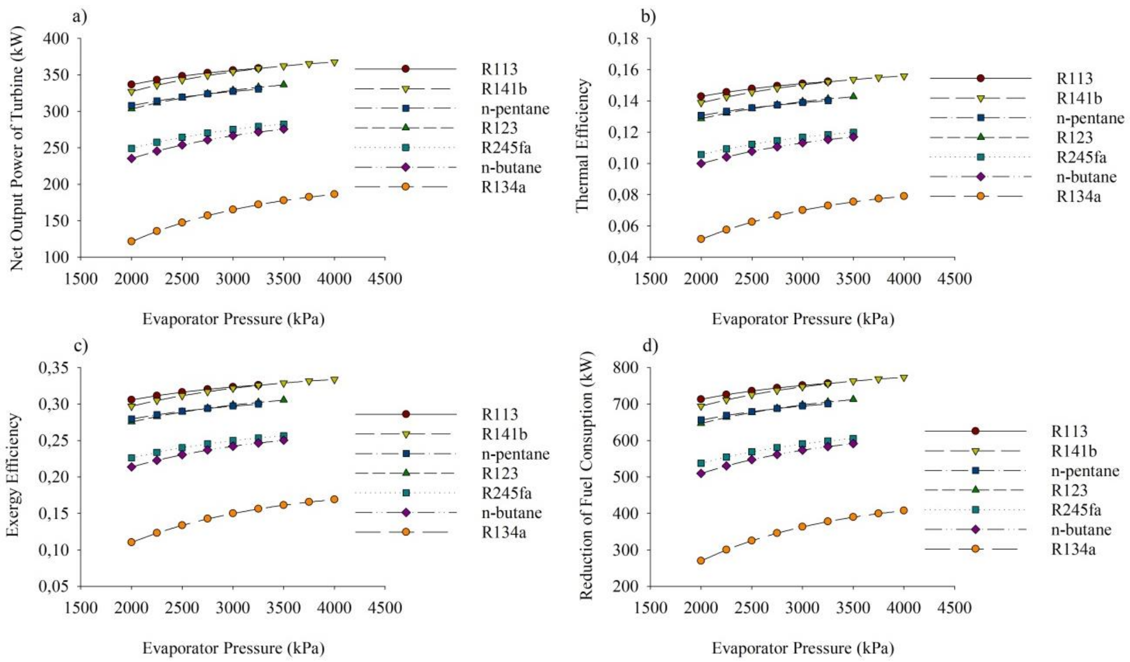

Figure 4.3 Variations of the thermal parameters with the different evaporator pressures for the Basic ORC

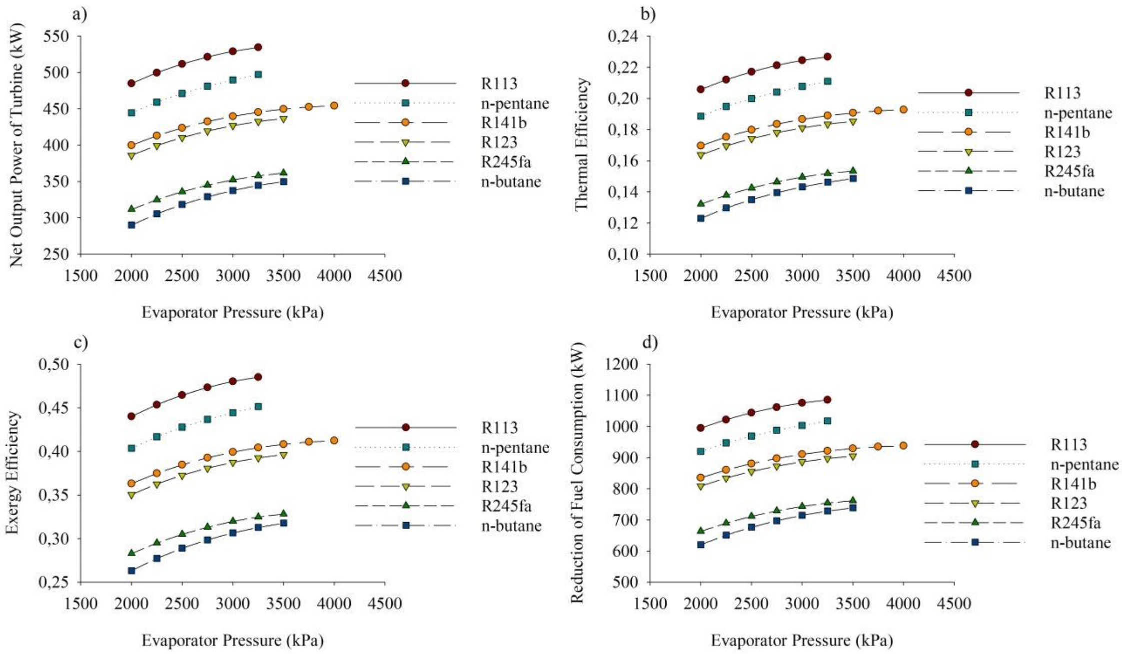

Figure 4.4 Variations of the thermal parameters with the different evaporator pressures for the regenerative ORC

4.2 Super Heat Temperature Optimization

The other independent parameter is superheat temperature and it has to be specified for optimal working condition. Superheat temperature and its effect to thermal parameters were investigated at Figure 4.5 and 4.6. The evaporator pressure accepted 90 percent of critical pressures for all working fluid.

Figure 4.5 was generated for Basic ORC. Figure 4.5(a) shows that, the net output power of turbine slightly increasing with increment of superheat temperature. Figure 4.5(b) and 4.5(c) shows that, the change of first law and second law efficiencies depending on superheat temperature. Figure 4.5(d) represents evaporator outlet temperature changes depending on superheat temperature. Figure 4.6 was generated for Regenerative ORC and ΔIHE accepted presently 5 °C. Figure 4.6(a) shows that the net output power of turbine increasing with increment of superheat temperature. Figure 4.6(b) and 4.6(c) shows that increasing of first law and second law efficiencies depending on increment of superheat temperature. Figure 4.6(d) shows that the evaporator outlet temperature changes depending on superheat temperature.

As seen at Figure 4.5(a) and Figure 4.6(a) super heat temperature increment is more effective for Regenerative ORC. Increasing of super heat temperature has no appreciable positive effect for the Basic ORC after the 10 °C point. The reason of that, there is no heat recovery unit such as preheater IHE and then waste heat of ORC is increasing with increment of superheat temperature. On the other hand, Regenerative ORC needs larger-scale IHE with the increment of super heat temperature. Also, evaporator has to be larger-scale for both regenerative and basic cycles. Caused by this, specific investment cost will be increase also. Decision of optimal superheat temperature has decomposition risk separately from thermal outputs. Fluids become useless by time because of decomposition rate which is the change rate of fluid chemical structure. Decomposition rate increases with the increment of temperature [43]. Andersen and Bruno [50] tested decomposition rates most of organic fluids (R245fa, R143a , R11 etc.) in the range of 2000 kPa and 5000 kPa pressure and found averagely above 350-400°C organic fluid become useless for cycle. The result of that, it is proper to choose 10 °C as superheat temperature for both cycles.

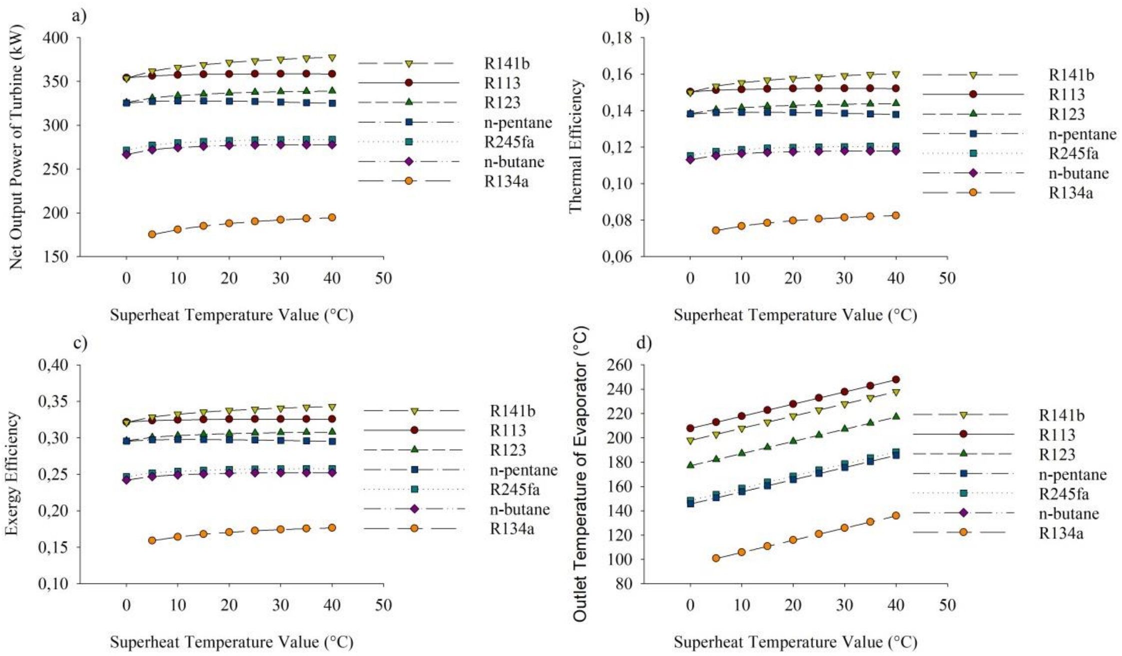

Figure 4.5 Variations of the thermal parameters with the different superheat temperatures for the Basic ORC

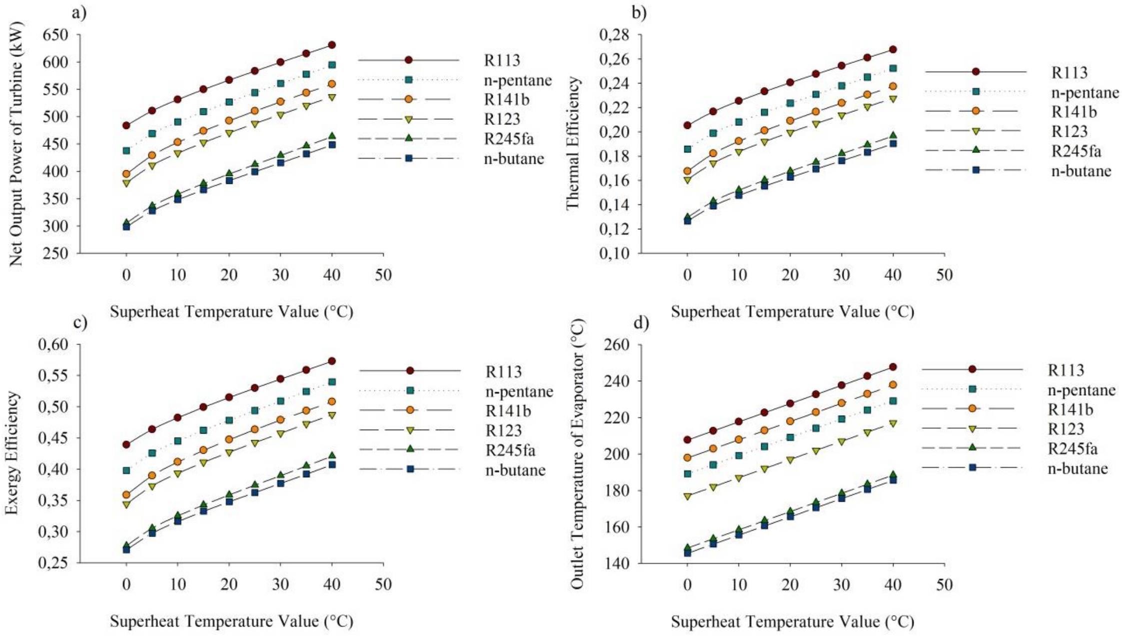

Figure 4.6 Variations of the thermal parameters with the different superheat temperatures for the Regenerative ORC

Fig 4.7 generated for n-pentane based cycles. Figure 4.7(a) represents that the variation of second law efficiency of components for Basic ORC with different evaporator pressure. Figure 4.7(b) shows that the second law efficiency of components for Regenerative ORC depending on different evaporator pressure. Figure 4.7(c) demonstrates the second law efficiency of components for Basic ORC depending on different superheat temperatures. Figure 4.7(d) represents the second law efficiency variation of components for Regenerative ORC depending on different superheat temperatures.

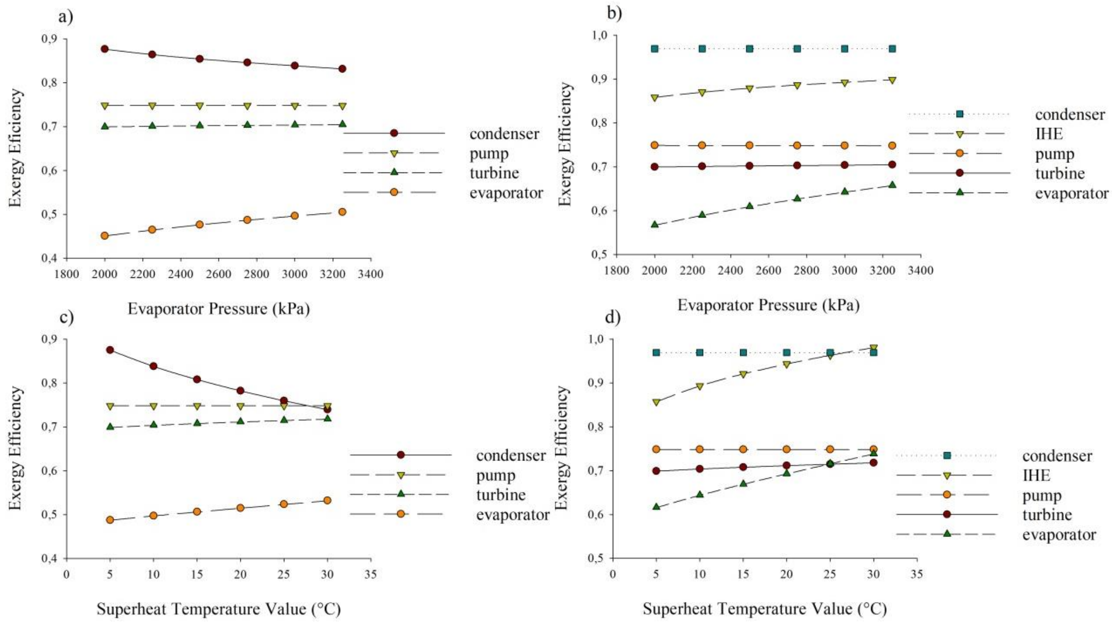

Figure 4.7 Components second law efficiencies depending on superheat temperature and evaporator pressure

4.3 Internal Heat Exchanger Effectiveness Optimization

Condenser temperature is a dependent value to coolant temperature and mass flow that is coming from cooling tower. Condenser temperature must be higher than coolant temperature for condensation with heat rejection to heat sink. On the other hand, the working fluid temperature at condenser outlet desired at least saturated liquid temperature provide stable working condition to cycle. Desired saturated liquid condition constitutes a limitation for pressure level of condenser outlet. The result of that, especially dry and isentropic type fluids have higher temperature of turbine outlet according to condenser outlet temperature. This situation causes to increase heat lost and reduce overall efficiency. Figure 4.8(a) represents the T-S diagram of Basic ORC dry type fluid of n-pentane. Figure 4.8(b) represents the T-S diagram of n-pentane with IHE. Effects of IHE can be clearly seen at Figure 4.8.

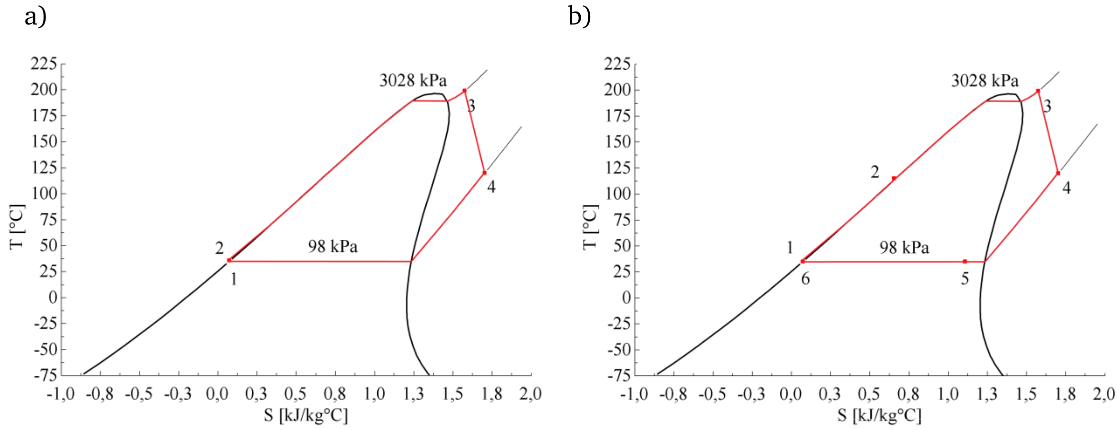

Figure 4.8 T-S diagram of n-pentane based Basic and Regenerative ORC

Figure 4.9(a) shows that the net output power of turbine decreasing with the increment of internal heat exchanger inlet and outlet temperature difference (ΔIHE). Figure 4.9(b) and Figure 4.9(c) shows that decreasing of first law and second law efficiencies depending on increment of Δ_{IHE}. Figure 4.9(d) represents that the working fluid flow rate decreasing with depending on increment of ΔIHE. It can be seen in Figure 4.9 that, working fluid mass flow rates decreasing correspondingly with efficiency terms and net output power of turbine. The result of that, Δ_{IHE} chose 5 °C to provide higher level heat recovery for maximum efficiency. Table 4.1 was generated for represent the numerical results of Figure 4.9 with additionally fuel energy consumption reduction values

Table 4.1 R113 state points, working fluid mass flow rate and thermal outputs depending on different Δ_{IHE}

WF	Δ_{IHE} (°C)	WF MFR (kg/s)	NOPT (kW)	TEC	EEC	RFEC (kW)
R113	5	14,05	531,3	22,54%	48,23%	1079,7
R113	7,5	13,83	523,1	22,20%	47,49%	1064,8
R113	10	13,63	515,3	21,86%	46,78%	1050,7
R113	12,5	13,42	507,7	21,54%	46,09%	1036,8
R113	15	13,23	500,3	21,23%	45,42%	1023,3

Table 4.1 R113 state points, working fluid mass flow rate and thermal outputs depending on different Δ_{IHE} (continued)

WF	Δ_{IHE} (°C)	WF MFR (kg/s)	NOPT (kW)	TEC	EEC	RFEC (kW)
R113	17,5	13,04	493,2	20,93%	44,77%	1010,2
R113	20	12,86	486,3	20,63%	44,14%	997,5

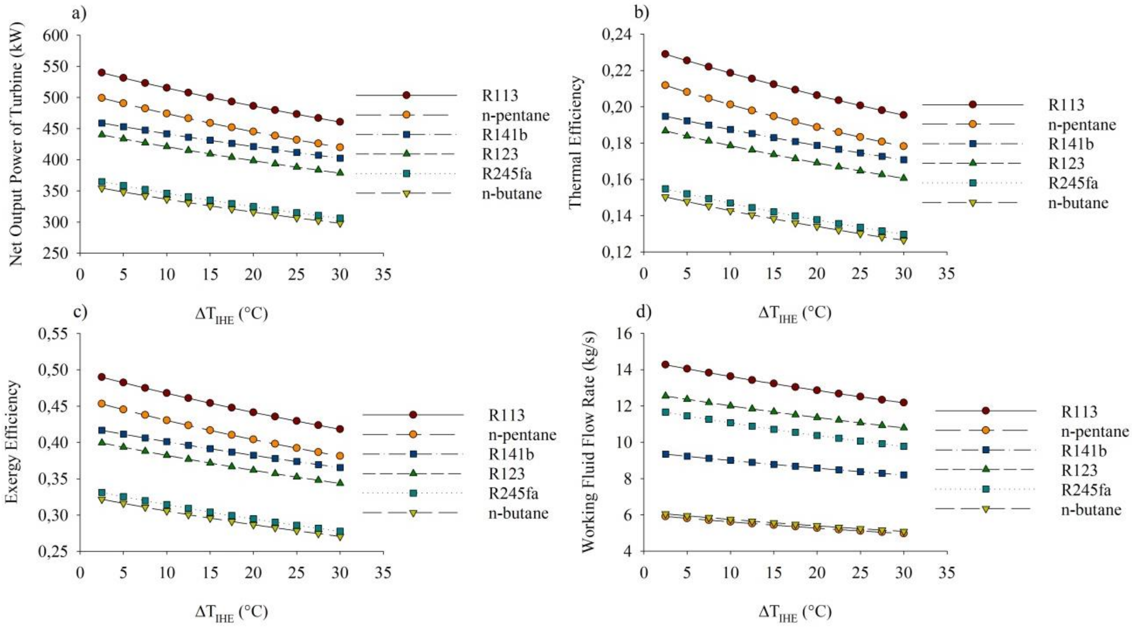

Figure 4.9 Variations of the thermal parameters with the different Δ_{IHE} for Regenerative ORC

5
RESULT AND DISCUSSION

5.1 Thermodynamic Results

In this section, working fluids were comparatively examined for thermodynamic results and overall system results. Also different engine loads examined for fluids most efficient working conditions which are specified at optimization phase. Figure 5.1(a) represents net turbine output power of all selected working fluids for both Basic and Regenerative ORC with different engine loads. R141b with Basic ORC is produces more power than production of R245fa and n-butane with regenerative ORC as seen at Figure 15(a). The exergy efficiencies of working fluids for both basic and regenerative ORC with different engine loads represents as Figure 5.1(b). Exergy efficiency rates of R141b with basic ORC higher than the R245fa and n-butane with regenerative ORC. Exergy efficiencies of systems are slightly increasing with increment of engine load. The reason of that, exhaust gas maximum temperature are reducing with increment of exhaust gas mass flow rate for higher engine load.

Figure 5.2(a) represents the cogeneration efficiency change of all working fluids with both basic and regenerative ORC for different engine loads. As seen at Table 5.2(a) cogeneration efficiency is increasing averagely 8 percent with increment of engine load from 50% to 75% engine load besides that, it is increasing averagely 4 percent with increment of engine load from 75% to 100%. The reason of that, engine is more efficient at higher load levels depending on data which are published by manufacturer as seen at Table 2.1. Cogeneration system overall efficiency is 6-14% higher than the single engine efficiency at 100% engine load depending on cycle type and working fluid type. In addition to this, the overall efficiency of cogeneration system is 4-14% higher at 75% engine load and 6-17% higher at 50% engine load.

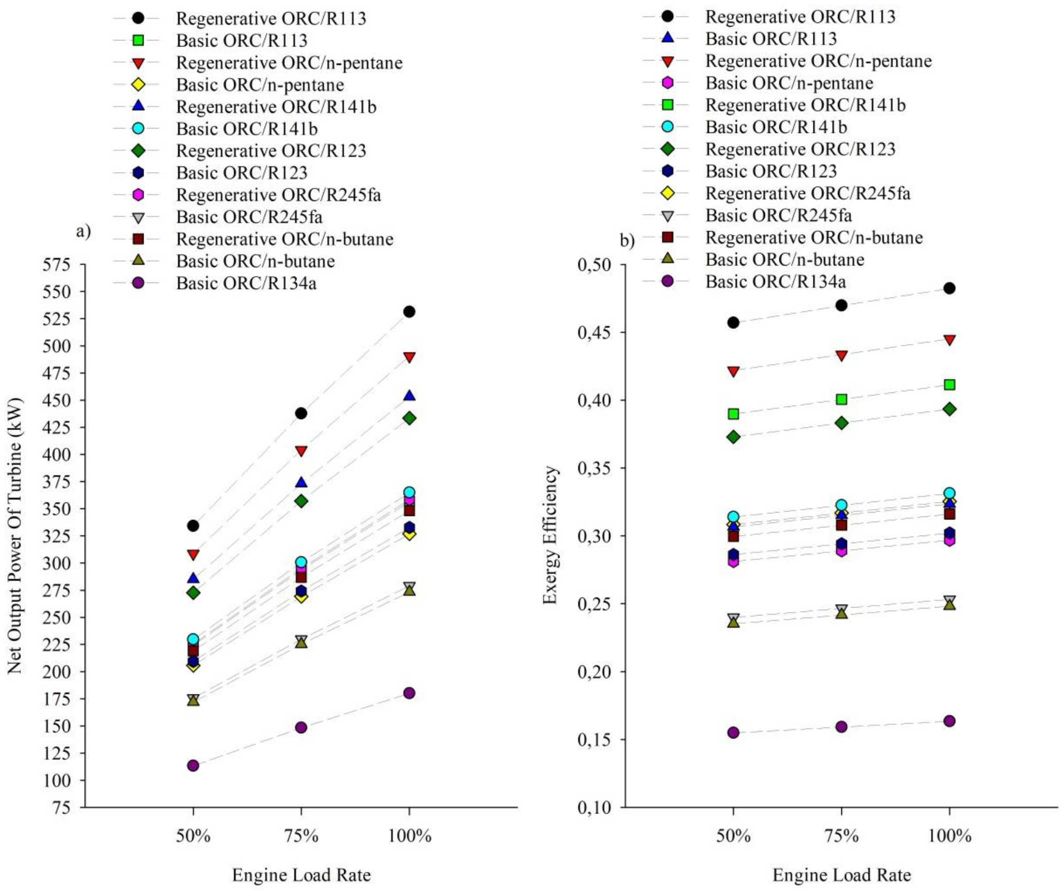

Figure 5.1 Turbine net output power and system exergy efficiency for different engine load

Figure 5.2(b) shows the fuel usage reduction of engine for different working fluid usage with both basic and regenerative ORC depending on different engine loads. Also, results which are shown at Figure 5.2(b) will be basis of economic analyses of cogeneration system. Reduction of fuel consumption is increasing with different increment rates for different working fluids depending on different engine loads. For example, while R113 reduction of fuel consumption is increasing with rate of 44%, R134a was increased with rate of 40% between 50% and 100% engine load. Simultaneously, cogeneration efficiencies increment has same increment slope for different working fluids. The reason of that, reduction of fuel usage is a directly dependent result to ICE total electricity production capacity and net output power of cycle apart from cycle efficiency. Cycle thermal efficiency is indirectly effective to reducing fuel energy consumption. Figure 5.2(c) represents cogeneration system heat recovery efficiency depending on engine load. Cogeneration system heat recovery efficiency is an indication for recovered heat rate depending on reversible work amount. In other words, it shows that the heat recovery system how close to maximum possible efficiency. System heat recovery efficiency changing with the range between 78,5% which is for R134a at 50% engine load as minimum point and 89,5% which is for R113 at 100% engine load as maximum point.

Numerically representation of Figure 5.1 presented as Table 5.1 for maximum engine load level. Also, additionally thermal efficiency, working fluid mass flow rate and dependent variables which are specified at optimization phase such as maximum temperature and pressure of cycle are presented at Table 5.1. Table 5.2 was presented as numerical results of Figure 5.2 for 100% engine load.

Table 5.1 Thermal outputs for different working fluids with both Basic and Regenerative ORC

WF	EOT (°C)	EOP (kPa)	Cycle Type	WF MFR (kg/s)	NOPT (kW)	TEC	EEC
R113	217,7	3095	Regen	14,05	531,3	22,54%	48,23%
			Basic	9,426	356,5	15,12%	32,36%
n-pentane	199,2	3028	Regen	5,813	490,5	20,81%	44,52%
			Basic	3,875	326,9	13,87%	29,68%
R141b	207,9	3824	Regen	9,231	453,1	19,23%	41,14%
			Basic	7,433	364,9	15,48%	33,13%
R245fa	158,5	3286	Regen	11,46	358,5	15,21%	32,54%
			Basic	8,923	279	11,84%	25,33%
R123	187,1	3301	Regen	12,37	433,5	18,39%	39,35%
			Basic	9,494	332,8	14,12%	30,21%

Table 5.1 Thermal outputs for different working fluids with both Basic and Regenerative ORC (continued)

WF	EOT (°C)	EOP (kPa)	Cycle Type	WF MFR (kg/s)	NOPT (kW)	TEC	EEC
n-butane	155,6	3416	Regen	5,955	348,3	14,78%	31,62%
			Basic	4,675	273,5	11,60%	24,83%
R134a	105,8	3653	Basic	12,03	180,1	7,64%	16,35%

5.2 Mathematical Evaluation of System

Mathematical evaluation was performed for most significant result that is net output power of cycle which is obtained with pump consumption extraction from turbine net output power. All final reports which are economic result, fuel save amount, cogeneration system efficiency could be calculated with net output power of cycle. Also, all assumptions and dependent variables of study such as, heat source inlet and outlet temperature, condensation temperature, subcritical pressure conditions, turbine and pump isentropic efficiencies, component design and effectiveness are effective parameters for ORC system. Mathematical evaluation equations were generated by taking into consideration below variables and it is important that equations are valid for that circumstances. In addition to this, parametric study based error margin of evaluated equations for both Basic ORC (R_b) and Regenerative ORC (R_r) were calculated for different working fluids and results were showed at Table 5.3.

All validation of mathematical evaluations:

- Super heat temperature is 10 °C.
- Internal Heat Exchanger effectiveness is 5 °C
- All assumptions which are mentioned at Section 2.3 are valid
- System operates under subcritical pressure conditions

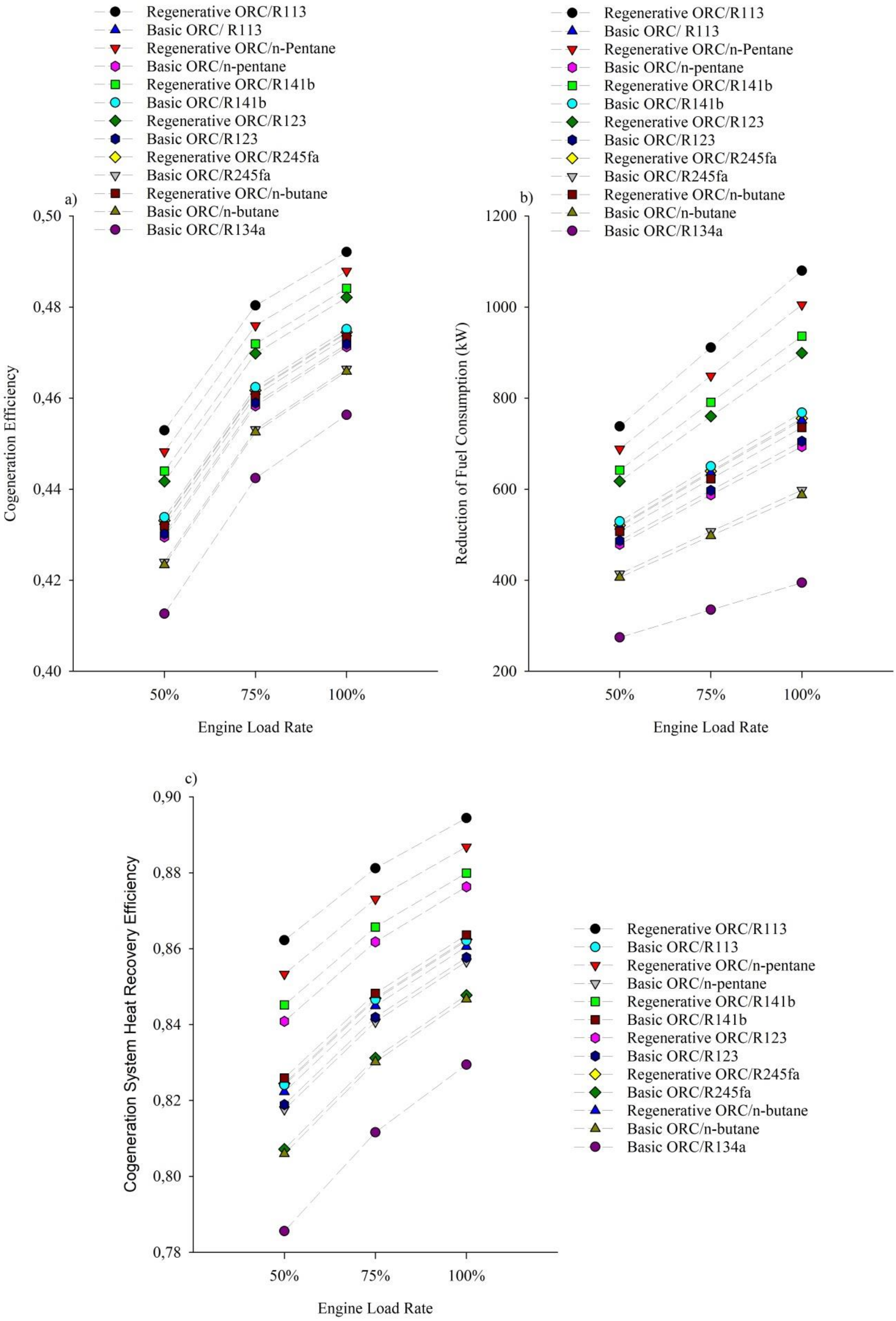

Figure 5.2 Cogeneration efficiency, reduction of fuel consumption and heat recovery efficiency for different load

Table 5.2 Cogeneration system thermal outputs for different working fluids with both Basic and Regenerative ORC

Working Fluid	Cycle Type	EFEC (kW)	CSFC (kW)	CSOE	FES (kW)	HRE
R113	Regen	9824	8744,30	49,21%	1079,70	89,44%
	Basic	9824	9072,35	47,43%	751,65	86,21%
n-pentane	Regen	9824	8818,73	48,79%	1005,27	88,69%
	Basic	9824	9130,35	47,13%	693,65	85,66%
R141b	Regen	9824	8888,08	48,41%	935,92	87,99%
	Basic	9824	9056,02	47,51%	767,98	86,36%
R245fa	Regen	9824	9068,46	47,45%	755,54	86,24%
	Basic	9824	9225,80	46,64%	598,20	84,77%
R123	Regen	9824	8924,86	48,21%	899,14	87,63%
	Basic	9824	9118,73	47,19%	705,27	85,77%
n-butane	Regen	9824	9088,34	47,35%	735,66	86,06%
	Basic	9824	9236,89	46,58%	587,11	84,67%
R134a	Basic	9824	9429,33	45,63%	394,67	82,94%

Net output power of cycle for Regenerative ORC can be expressed as follows:

$$W_{r,net} = K_{r,wf}(-12{,}696P_e^2 + 102{,}91P_e + 245{,}29) \tag{5.1}$$

Where $K_{r,wf}$ is the constant value which is dependent to working fluids of Regenerative ORC, P_e is the evaporation pressure in MPa.

Net output power of cycle for Basic ORC can be expressed as follows:

$$W_{b,net} = K_{b,wf}(-6{,}8416P_e^2 + 60{,}896P_e + 233{,}27) \tag{5.2}$$

Where $K_{b,wf}$ is the constant value which is dependent to working fluids of Basic ORC, $K_{r,wf}$ and $K_{b,wf}$ values are presented at Table 5.3.

Table 5.3 Constant value and error margin of mathematical evaluation

Working Fluid	$K_{r,wf}$	R_b (%)	$K_{b,wf}$	R_r (%)
R113	1,2062	0,73	1,0135	0,82
R141b	1	0,89	1	0,07
R123	0,9692	0,68	0,9280	0,08
R245fa	0,7950	0,69	0,7727	0,68
n-pentane	1,1130	0,56	0,9301	0,64
n-butane	0,7566	1,85	0,7442	1,84

Basic ORC operate with R134a could not be express with common equations with other selected working fluids. Net output power change rate based on evaporation pressure has considerable different increment rate for R134a according to other working fluids. Error margin was found above of acceptable limits, when a common single equation generation which is including R134a with other working fluids. Because of that, R134a was represented with different equation which is presented below.

Basic ORC net output power with R134a can be expressed as follows:

$$W_{net} = -11,361 P_e^2 + 99,893 P_e - 32,096 \qquad (5.3)$$

Error margin of Equation (5.3) is averagely 2,32% for subcritical pressures.

Calculated values depending on balance equation and evaluated values depending on mathematical model were shown as comparatively at Figure 5.3 (a) and (b). Figure 5.3 (a) was generated for Basic ORC and Figure 5.3 (b) was generated for Regenerative ORC.

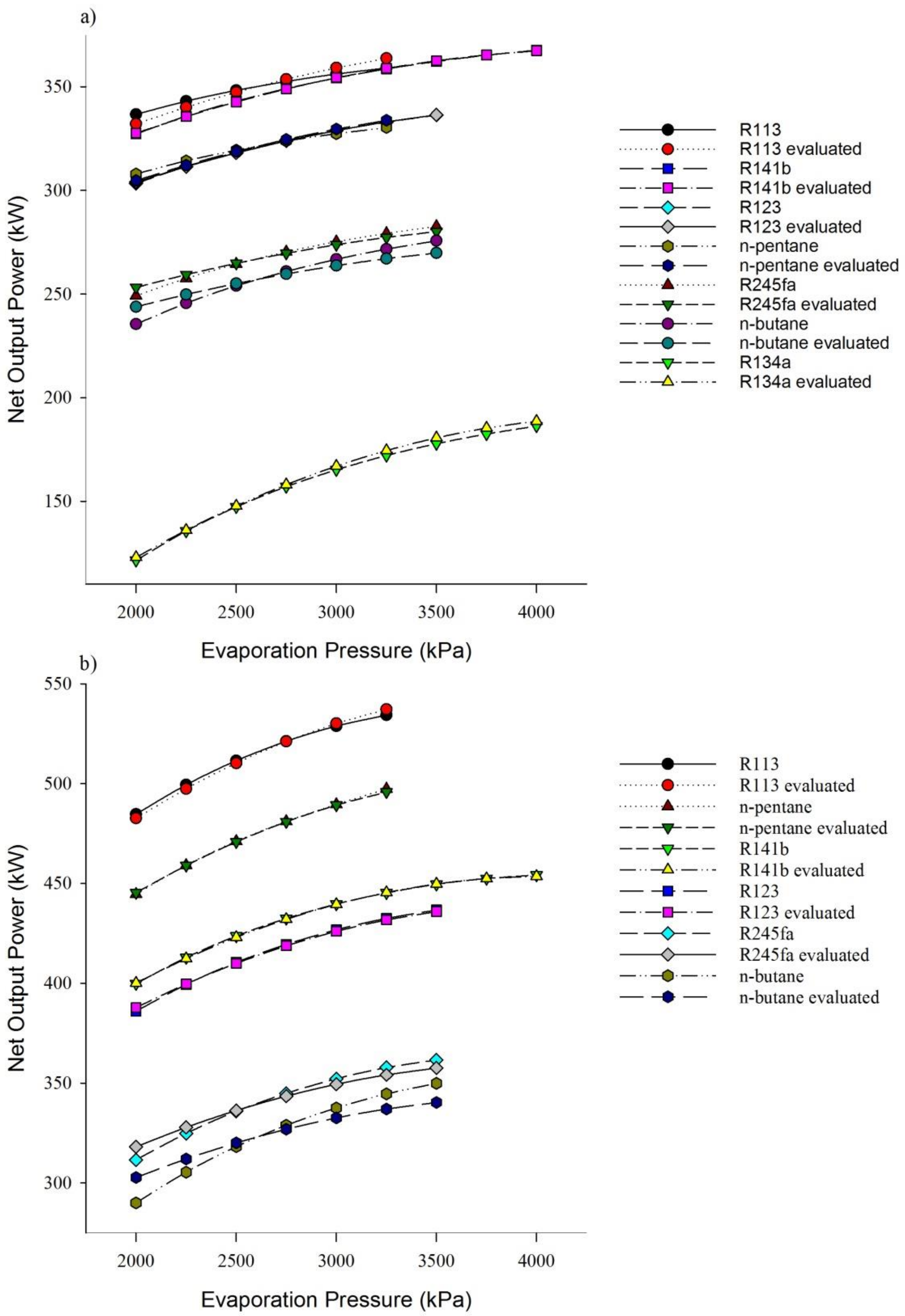

Figure 3.3 Differences between calculated and evaluated values for Basic and Regenerative ORC

5.3 Economic Results

Economic results are most important indicators for feasibility research of cogeneration system. Economic analyze was developed for both Basic and Regenerative ORC with considering most efficient working fluids of each cycle.

R113 is most efficient working fluid for RORC and R141b is the most efficient working fluid for BORC as mentioned at Thermodynamic Results section. Thermophysical results and state point conditions of Regenerative ORC and Basic ORC respectively presented at Table 5.4 and 5.5

Table 5.4 Thermophysical properties of R113 with Regenerative ORC

Property	Unit	Value	Value	Value
Engine Load	%	50	75	100
t_{op}	hour/year	960	2240	4480
m_{wf}	kg/s	8,84	11,57	14,05
$m_{exhaust}$	kg/s	3,41	4,83	6,34
m_{cw}	kg/s	27,32	35,77	43,42
W_{pump}	kW	23,22	30,41	36,91
$W_{turbine}$	kW	357,50	468,10	568,20
$Q_{e,red}$	kW	737,88	911,18	1079,70
Q_{out}	kW	1143,00	1496,00	1816,00
Q_{in}	kW	1483,00	1942,00	2357,00
T_1	°C		37,11	
T_2	°C		118,00	
T_3	°C		217,70	
T_4	°C		123,00	
T_5	°C		35,00	

Table 5.4 Thermophysical properties of R113 with Regenerative ORC (continued)

Property	Unit	Value	Value	Value
T_6	°C		35,00	
P_1	kPa		3095,00	
P_2	kPa		3095,00	
P_3	kPa		3095,00	
P_4	kPa		65,26	
P_5	kPa		65,26	
P_6	kPa		65,26	

Table 5.5 Thermophysical properties of R141b with Basic ORC

Property	Unit	Value	Value	Value
Engine Load	%	50	75	100
t_{op}	hour/year	960	2240	4480
m_{wf}	kg/s	4,68	6,12	7,43
$m_{exhaust}$	kg/s	3,41	4,83	6,34
m_{cw}	kg/s	29,85	39,08	47,44
W_{pump}	kW	19,05	24,94	30,28
$W_{turbine}$	kW	248,60	325,60	395,20
$Q_{e,red}$	kW	529,21	650,09	767,98

Table 5.5 Thermophysical properties of R141b with Basic ORC (continued)

Property	Unit	Value	Value	Value
Q_{out}	kW	1249,00	1635,00	1984,00
Q_{in}	kW	1483,00	1942,00	2357,00
T_1	°C		35,00	
T_2	°C		36,03	
T_3	°C		199,20	
T_4	°C		120,10	
P_1	kPa		112,20	
P_2	kPa		3824,00	
P_3	kPa		3824,00	
P_4	kPa		112,20	

Equipment design was made according to supply maximum load of system. Investment cost of components was calculated depending on 100% engine load with using presented parameters at Table 5.4 and 5.5. Component investment cost and total investment cost result presented as Table 5.6 for both Basic and Regenerative ORC. Investment costs which are presented at Table 5.6 also include labor price and piping price. Regenerative ORC total investment cost higher than Basic ORC as expected. Besides that, Basic ORC evaporator price and condenser price are higher than Regenerative ORC with compatible to expectation. The reason of that, Basic ORC condenser heat loss is higher as shown at Table 5.4 and 5.5. Additionally, Regenerative ORC evaporator is smaller than Basic ORC with help of higher energy level fluid inlet to evaporator. Turbine and pump investment costs are higher for Regenerative ORC as a result of respectively higher output and higher input power. Total investment cost difference mainly relative to Internal Heat Exchanger investment cost and turbine investment cost difference.

Table 5.6 Basic and Regenerative ORC Investment Cost

Component	Symbol	Unit	Regenerative ORC	Basic ORC
Turbine	$C_{inv,turbine}$	$ x 10³	805,068	648,864
Pump	$C_{inv,pump}$	$ x 10³	60,341	54,412
Evaporator	$C_{inv,evaporator}$	$ x 10³	336,556	340,577
Condenser	$C_{inv,condenser}$	$ x 10³	207,609	210,607
IHE	$C_{inv,IHE}$	$ x 10³	184,411	-
Total	C_{inv}	$ x 10³	1593,985	1254,46

Payback Period (PBP) and Total Profit (TP) are relative to reduction of fuel energy consumption. Investigated ICE is using natural gas as fuel source. Natural gas unit price is 1,37 TL/m³ for industrial usage in Turkey. Natural gas unit price was converted to USD for analyze with using 5 TL/USD exchange rate. Converted unit kW based fuel price can be express as follows:

$$C_f = \frac{C_{v,f}}{LCV} \tag{5.4}$$

Where *Cf* is the unit price of fuel as unit kW based, $C_{v,f}$ is the unit price of fuel volumetric base, *LCV* is the Lower Calorific Value of natural gas which is 9,59 kW. Net Present Value was found for all incomes and costs with using inflation rate index (r) which is accepted 2% and annual loan index (i) which is accepted 5%. Final results of economic analyze presented at Table 5.7. The operating and maintenance cost rising according to increasing of investment cost. Also scrap income is higher for Regenerative ORC depending on its investment cost. Besides that, TP is higher for Regenerative ORC with help of higher output power. While

TP is increasing, the investment cost also increasing as shown at Table 5.7. The PBP is major indicator to decide most feasible system in such circumstances. Regenerative ORC has shorter PBP than the Basic ORC. As a result of that Regenerative ORC usage is more profitable than Basic ORC usage.

Table 5.7 Comparative economic results of Basic and Regenerative ORC

Properties	Unit	Regenerative ORC	Basic ORC
TPC	$\$ \times 10^3$ / 15 year	2.815,419	2.005,776
C_{sc}	$\$ \times 10^3$ / 15 year	103,192	81,212
C_{op}	$\$ \times 10^3$ / 15 year	542,801	407,351
C_{inv}	$\$ \times 10^3$	1.593,986	1.254,462
TP	$\$ \times 10^3$ / 15 year	781,822	425,174
PBP	Years	8,46	9,82

6
CONCLUSIONS AND RECOMMENDATIONS

Purpose of this study is research two different ORC with different working fluids as a heat recovery system for exhaust gas of natural gas powered internal combustion engine which is TCG2032 model manufactured by MWM brand of Caterpillar Energy Solutions GmbH. The determinative criterion for heat recovery system effectiveness is reduction of fuel energy usage of engine which is relative to maximum net output power of turbine. Optimization study was made for each working fluid and cycle to maximize net output power of turbine with using thermal parameters. R113, R123, R134a, R141b, R245fa, n-pentane and n-butane was investigated as working fluids. The major conclusions of research are express as follows:

- The most efficient working fluid is R113 for regenerative ORC. System electricity production efficiency could increase from 43,3% to 49,21% for regenerative system (provide 1079kWh fuel energy save) and also increase to 47,23% for basic system (provide 751 kWh fuel energy save). But R113 has the highest GWP value.
- R141b is the most efficient working fluid for basic ORC and its net turbine output power higher then regenerative cycle usage with R245fa and n-butane. R141b could provide 767 kWh fuel energy save with basic ORC and 935kWh with regenerative ORC. Beside that R141b has the highest evaporator outlet temperature.
- R134a is the worst working fluid for heat recovery system depending on thermodynamic results. Fuel energy usage could be reduced 394kWh with using R134a based basic ORC.
- Environmental parameters of working fluids at least as important as thermal parameters. R123 has the lowest ALT value and it could save 899

kWh fuel energy with regenerative ORC, also save 705 kWh fuel energy with basic ORC.

- The lowest GWP value belongs to n-butane which is save 735 kWh fuel energy with regenerative ORC and save 587 kWh fuel energy with basic ORC.
- Regenerative ORC is more profitable than Basic ORC depending on PBP. Regenerative ORC has 8,46 years PBP and Basic ORC has 9,82 years PBP.
- Regenerative ORC has higher net output power than basic ORC for same working fluid. The reason of that, IHE usage is increasing net output power with reducing heat loss of cycle. Net output power increment rate with IHE usage is directly relative the temperature difference between turbine outlet and condenser outlet. The most proper working fluid to IHE usage is n-pentane, its net turbine output power increasing with rate of 50,4% between regenerative and basic ORC. Fuel energy save value of n-pentane is 1005 kWh for regenerative and 693 kWh for basic ORC. IHE usage is negatively affecting to system simplicity and investment cost beside all thermal benefits.
- Different optimal thermal working condition must be defined for each working fluid depending on its thermal and chemical properties to maximize turbine net output power.
- ORC exergy efficiencies are increasing for both basic and regenerative cycle depending on engine load level increment. The reason of that, exhaust gas temperature is reducing with exhaust mass flow rate increment at higher load level.
- Regenerative ORC superheat temperature increment is efficiently more effective than basic ORC. The reason of that, both outlet temperature of turbine and cycle heat loss are increasing with the increment of evaporator outlet temperature for the basic ORC.

REFERENCES

[1] Kuyumcu, M.E., Tutumlu, H. and Yumrutaş, R., (2016). "Performance of a swimming pool heating system by utilizing waste energy rejected from an ice rink with an energy storage tank", Energy Conversion and Management, 121: 349-357.

[2] Martins, F., Felgueiras, C. and Smitková, M., (2016). "Fossil fuel energy consumption in European countries", Energy Procedia, 153: 107–111.

[3] Yıldız, İ., (2018). "1.12 Fossil Fuels", Comprehensive Energy Systems, 1: 521-567.

[4] Lott, M.C. Pye, S. and Dodds, P.E., (2017). "Quantifying the co-impacts of energy sector decarbonisation on outdoor air pollution in the United Kingdom", Energy Policy, 101: 42–51.

[5] Shi, L., Shu, G., Tian, H. and Deng, S., (2018). "A review of modified Organic Rankine cycles (ORCs) for internal combustion engine waste heat recovery (ICE-WHR)", Renewable and Sustainable Energy Reviews, 92: 95-110.

[6] Mohsen, S., Arash, N., Alireza, G. and Mortaza, Y., (2016). "Thermodynamic analysis and multi-objective optimization of various ORC (organic Rankine cycle) configurations using zeotropic mixtures", Energy, 109: 791-802.

[7] Yu, X., Li, Z., Lu, Y., Huang, R. and Roskilly, A.P., (2019). "Investigation of organic Rankine cycle integrated with double latent thermal energy storage for engine waste heat recovery", Energy, 170: 1098-1112.

[8] Vaja, I. and Gambarotta, A., (2008). "Internal Combustion Engine (ICE) bottoming with Organic Rankine Cycles (ORCs)", Energy, 35: 1084-1093.

[9] Auld, A., Berson, A. and Hogg, S.I., (2013). "Organic Rankine cycles in waste heat recovery: a comparative study", International journal of low-carbon technologies, 8: 9-18.

[10] Chuang, Y. and Chau, K.T., (2009). "Thermoelectric automotive waste heat energy recovery using maximum power point tracking", Energy Conversion and Management, 50: 1506-1512.

[11] Bamgbopa, M.O., (2012). "Modeling and Performance evaluation of an Organic Rankine Cycle (ORC) with R245fa as Working Fluid", Master's of Science Thesis, Middle East Technical University, Ankara.

[12] Macchi, Ennio., Astolfi, M., (2017). Organic Rankine Cycle (ORC) Power Systems Technologies and Applications, Elsevier LTD., Cambridge.

[13] Amicabile, S., Lee, J. and Kum., (2015). "A comprehensive design methodology of organic Rankine cycles for the waste heat recovery of automotive heavy-duty diesel engines", Applied Thermal Engineering, 87: 574-585.

[14] Wang, E.H., Zhang, H.G., Fan, B.Y., Ouyang, M.G., Zhao, Y. and Muc, Q.H., (2011). "Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery", Energy, 36: 3406-3418.

[15] Katsanos, C.O., Hountalas, D.T. and Pariotis, E.G., (2012). "Thermodynamic analysis of a Rankine cycle applied on a diesel truck engine using steam and organic medium", Energy Conversion and Management, 60: 68–76.

[16] Mastrullo, R., Mauro, A.W., Revellin, R. and Viscito, L., (2015). "Modeling and optimization of a shell and louvered fin mini-tubes heat exchanger in an ORC powered by an internal combustion engine", Energy Conversion and Management, 101: 697–712.

[17] Uusitalo, A., Honkatukia, J., Backman, J. and Nyyssonen, S., (2015). "Experimental study on charge air heat utilization of large-scale reciprocating engines by means of Organic Rankine Cycle", Applied Thermal Engineering, 89: 209-219.

[18] Wang, E.H., Zhang, H.G., Fan, B.Y., Ouyang, M.G., Yang, F.Y., Yang, K., Wang, Z., Zhang, J. and Yang, F.B., (2014). "Parametric analysis of a dual-loop ORC system for waste heat recovery of a diesel engine", Applied Thermal Engineering, 67: 168-178.

[19] Yang, F., Zhang, H., Bei, C., Song, S. and Wang, E., (2015). "Parametric optimization and performance analysis of ORC (organic Rankine cycle) for diesel engine waste heat recovery with a fin-and tube evaporator", Energy, 91: 128-141.

[20] Song, J., Song, Y. and Gu, C., (2015). "Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines", Energy, 82: 976-985.

[21] Wang, E., Yu, Z., Zhang, H. and Yang, Fubin., (2017). "A regenerative supercritical-subcritical dual-loop organic Rankine cycle system for energy recovery from the waste heat of internal combustion engines", Applied Energy, 190: 574–590.

[22] Larsen, U., Pierobon, L., Haglind, F. and Gabrielii, C., (2013). "Design and optimisation of organic Rankine cycles for waste heat recovery in marine applications using the principles of natural selection", Energy, 55: 803-812.

[23] Shengjun, Z., Huaixin, W. and Tao, G., (2011). "Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation", Applied Energy, 88: 2740-54.

[24] Rech, S., Zandarin, S., Lazzaretto, A. and Frangopoulos, C.A., (2017). "Design and off-design models of single and two-stage ORC systems on board a LNG carrier for the search of the optimal performance and control strategy", Applied Energy, 204: 221–241.

[25] Srinivasan, K.K., Pedro, J., Sundar, M. and Krishnan, R., (2010) "Analysis of exhaust waste heat recovery from a dual fuel low temperature

combustion engine using an Organic Rankine Cycle”, Energy, 35: 2387-2399.

[26] Xu, J. and Yu, C., (2014). “Critical temperature criterion for selection of working fluids for subcritical pressure Organic Rankine cycles”, Energy, 74: 719-33.

[27] He, M., Zhang, X., Zeng, K. and Gao, K., (2011). “A combined thermodynamic cycle used for waste heat recovery of internal combustion engine”, Energy, 36: 6821-29.

[28] Saleh, B., Koglbauer, G., Wendland, M. and Fischer, J., (2007). “Working fluids for low-temperature organic Rankine cycles”, Energy, 32: 1210-1221.

[29] Wang, Z.Q., Zhou, N.J., Guo, J. and Wang, X.Y., (2012). “Fluid selection and parametric optimization of organic Rankine cycle using low temperature waste heat”, Energy, 40: 107-115.

[30] Kajurek, J., Rusowicz, A., Grzebielec, A., Bujalski, W., Futyma, K. and Rudowicz, Z., (2019). “Selection of refrigerants for a modified organic Rankine cycle”, Energy, 168: 1-8.

[31] He, C., Liu, C., Gao, H., Xie, H., Li, Y., Wu, S. and Xu, J., (2012). “The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle”, Energy, 38: 136-43.

[32] Glover, S., Douglas, R., Rosa, M.D., Zhang, X. and Glover, L., (2015). “Simulation of a multiple heat source supercritical ORC (Organic Rankine Cycle) for vehicle waste heat recovery”, Energy, 93: 1568-1580.

[33] Wang, D., Ling, X., Peng, H., Liu, L. and Tao, L.L., (2013). “Efficiency and optimal performance evaluation of organic Rankine cycle for low grade waste heat power generation”, Energy, 50: 343-352.

[34] Mago, P.J., Chamra, L.M. and Somayaji, C., (2007). “Performance analysis of different working fluids for use in organic Rankine cycles”, Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 221: 255-264.

[35] Sung, T. and Kim, K.C., (2016). “Thermodynamic analysis of a novel dual-loop organic Rankine cycle for engine waste heat and LNG cold”, Applied Thermal Engineering, 100: 1031–1041.

[36] He, S., Chang, H., Zhang, X., Shu, S. and Duan, C., (2015). “Working fluid selection for an Organic Rankine Cycle utilizing high and low temperature energy of an LNG engine”, Applied Thermal Engineering, 90: 579-589.

[37] Tian, H., Shu, G., Wei, H., Liang, X. and Liu, L., (2012). “Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE)”, Energy, 47: 125-136.

[38] Miao, Z., Zhang, K., Wang, M. and Xu, J., (2019). "Thermodynamic selection criteria of zeotropic mixtures for subcritical organic Rankine cycle", Energy, 167: 484-497.

[39] Shu, G., Liu, L., Tian, H., Wei, H. and Yu, G., (2014). "Parametric and working fluid analysis of a dual-loop organic Rankine cycle (DORC) used in engine waste heat recovery", Applied Energy, 113: 1188-1198.

[40] Imran, M., Park, B.S., Kim, H.J., Lee, D.H., Usman, M. and Heo M., (2014). "Thermo-economic optimization of Regenerative Organic Rankine Cycle for waste heat recovery applications", Energy Conversion and Management 87: 107-118.

[41] Le, V.L., Kheiri, A., Feidt, M. and Pelloux-Prayer, S., (2014). "Thermodynamic and economic optimizations of a waste heat to power plant driven by a subcritical ORC (Organic Rankine Cycle) using pure or zeotropic working fluid", Energy, 78: 622-638.

[42] Asim, M., Leung, M.K.H., Shan, Z., Li, Y., Leung, D.Y.C. and Ni, M., (2017). "Thermodynamic and Thermo-economic Analysis of Integrated Organic Rankine Cycle for Waste Heat Recovery from Vapor Compression Refrigerant Cycle", Energy Procedia, 143: 192-198.

[43] Yang, M.H., (2018). "Payback period investigation of the organic Rankine cycle with mixed working fluids to recover waste heat from the exhaust gas of a large marine diesel engine", Energy Conversion and Management, 162: 189–202.

[44] Li, T., Meng, N., Liu, J., Zhu, J. and Kong, X., (2019). "Thermodynamic and economic evaluation of the organic Rankine cycle (ORC) and two-stage series organic Rankine cycle (TSORC) for flue gas heat recovery", Energy Conversion and Management, 183: 816-829.

[45] Wang, M., Chen, Y., Liu, Q. and Yuanyuan, Zhao., (2018). "Thermodynamic and thermo-economic analysis of dual-pressure and single pressure evaporation organic Rankine cycles", Energy Conversion and Management, 177: 718–736.

[46] Yang, M.H. and Yeh, R.H., (2015). "Thermodynamic and economic performances optimization of an organic Rankine cycle system utilizing exhaust gas of a large marine diesel engine", Applied Energy, 149: 1–12.

[47] Baldassoa, E., Andreasen, J.G., Mondejar, M.E., Larsen, U. and Haglind, F., (2019). "Technical and economic feasibility of organic Rankine cycle-based waste heat recovery systems on feeder ships: Impact of nitrogen oxides emission abatement technologies", Energy Conversion and Management, 183: 577-589.

[48] Walraven, D., Laenen, B. and D'haeseleer, W., (2015). "Minimizing the levelized cost of electricity production from lowtemperature geothermal heat sources with ORCs: Water or air cooled?", Applied Energy, 142: 144-153.

[49] Turton, R., Bailie, R.C., Whiting, W.B. and Shaeiwitz, J.A., (1998). Analysis, synthesis and design of chemical processes, Prentice Hall Inc., New Jersey.

[50] Andersen, P.C. and Bruno, T.J., (2005). “Thermal Decomposition Kinetics of RP-1 Rocket Propellant”, Ind. Eng. Chem. Res., 44: 1670-1676.

CURRICULUM VITAE

PERSONAL INFORMATION

Name Surname	**:** Yunus Emre TALU
Date of birth and place	**:** 22.05.1993 / Tokat
Foreign Languages	**:** English
E-mail	**:** yunusemretalu@gmail.com

EDUCATION

Degree	**Department**	**University**	**Date of Graduation**
Undergraduate	Mechanical Engineering	Yıldız Technical University	2015
High School	-	Tokat Science High School	2011

WORK EXPERIENCE

Year	**Corporation/Institute**	**Enrollment**
2015-present	Totalteknik	Project Responsible

PUBLISHMENTS

Conference Papers

1. Talu, Y.E., Pınarbaşı, A., Doğan, A. and Ulusoy, I., (2019). "Thermodynamic Analyses and Optimization Study Of Organic Rankine Cycle Usage with Internal Combustion Engine Waste Heat", Ereğli International Science and Academic Congress (INSAC), 9-10 March 2019, Konya.