ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

MSc THESIS

Çağlar KARAOĞLU

FLUE GAS WASTE HEAT RECOVERY IN A STEAM POWER PLANT

DEPARTMENT OF MECHANICHAL ENGINEERING

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ÇUKUROVA UNIVERSITY
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DEPARTMENT OF MECHANICAL ENGINEERING

We certify that the thesis titled above was reviewed and approved for the award of degree of the Master of Science by the board of jury on 04/09/2018

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ABSTRACT

MSc THESIS

FLUE GAS WASTE HEAT RECOVERY IN A STEAM POWER PLANT

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UNIVERSITY OF CUKUROVA
INSTITUTE OF NATURAL AND APPLIED SCIENCES
DEPARTMENT OF MECHANICHAL ENGINEERING

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The aim of the study is to determine applicability of waste heat recovery system in a steam power plant. ORC (Organic Rankine Cycle) was integrated into existing steam Rankine cycle. Energy and exergy analyze were applied for each component of ORC. Different working fluid options was taken into account and compared with each other in graphs. On the basis of results steam powered circulating water pump has been adapted to system via engaged with ORC turbine. Economic analyzes of waste heat recovery system was carried out and the results were commentated. Full load (660 MW) parameters were considered in the calculation. Regarding design parameters, acquired results were discussed. Thermodynamic, economic and environmental facts, are the main restriction of the study.

According to experienced results, the main criteria is to select a proper fluid for ORC. In order to shorten payback period either covered flue gas duct against acid dew point on the downstream of ORC evaporator thus possible to reduce more flue gas temperature or to supply loyal customer utilizing flue gas waste via any ORC component.

Keywords: Energy, Exergy, Organic Fluids, ORC

ÖZ

YÜKSEK LİSANS TEZİ

Çağlar KARAOĞLU

BUHARLI BİR GÜÇ SANTRALİNDE BACA GAZI ATIK ISI GERİ KAZANIMI

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Bu çalışma, mevcut bir buharlı güç santralinde atık ısı geri kazanımı sisteminin uygulanılabilirliğine karar vermek için yapılmıştır. Organik Rankine Çevrimi, buhar çevrimine entegre edilmiştir. Enerji ve ekserji analizleri her bir Organik Rankine Çevrim elemanı için yapılmıştır. Farklı organik akışkan seçenekleri hesaba katılmış ve birbirleriyle grafikler üzerinden kıyaslama yapılmıştır. Çıkan sonuçlar neticesinde, buhar tahrikli ana soğutma suyu pompası ORC türbinle akuple edilerek sisteme entegre edilmiştir. Atık ısı geri kazanım sisteminin ekonomik analizi yapılmış olunup, sonuçları yorumlanmıştır. Hesaplama esnasında tam yük (660 MW) parametreleri dikkate alınmıştır. Dizayn parametleri göz önüne alınmış ve çıkan sonuçlar müzakere edilmiştir. Bu çalışmadaki ana kısıtlar sırasıyla termodinamik, ekonomik ve çevresel gerçeklerdir.

Edinilen tecrübelere göre, ORC ana kriteri uygun akışkan seçimidir. Amortisman sürecinin kısaltılması için tercih edilebilecek yöntemler ya ORC buharlaştırıcısı çıkış sıcaklığını daha fazla düşürebilmek için baca gazı kanalını asit çiğlenmesine karşı kaplamak ya da baca gazı atık ısı geri kazanımından ORC elemanları vasıtasıyla sürekli faydalanacak tüketicilerin sağlanmasıyla mümkündür.

Anahtar Kelimeler: Enerji, Ekserji, Organik Akışkan, ORC

EXTENDED ABSTRACT

Energy is the power and indispensable part of life in terms of several aspects. We must have kind of energy features such as sustainable, affordable and environmentally friendly supplies. In spite of the fact that running out of fossil fuel reserves and having many questionable environmental issues such as global warming threat and climate change risks, oil demand increment is to be 47.5% between 2003 and 2030, 91.6% for natural gas and 94.7% for coal.

Greenhouse gas (GHG) emissions are doubtful for future life so one of the way to decrease role of conventional plant products (harmful emissions) or more relying on sharing overall portfolio with renewable energy. Instead of fully replacing renewable by conventional systems, it's better to implement partial integration of renewable methods (e.g. waste heat recovery technique) on such conventional power plants already in operation. The process after burning in fossil fuel use yields hazardous and inevitable emission values. On the other hand, it is not needed any fossil fuel during operating in renewable fields. Renewable energy based on natural resources, e.g. wind, solar power, biomass and waste heat furthermore it requires lower operating costs and maintenance as well. It’s not that all satisfied operating in stable conditions so fluctuating output is a weak point of renewable aspect. By looking into Turkey’s energy supplying policy is actually satisfying consumer total demand without any adverse effect on its economic development. As Turkey rapidly develops, Turkey’s economic growth acceleration is represented by positive values. Parallel to economy, Turkey’s energy industry, including thermal and renewable energy based power generation, has expanded rapidly.

According to TEIAS values Turkey’s installed capacity has been reached around 87.600 MW. Turkey electricity daily generation is 45.000 MW in peak hours. Seasonally the generation value instantly fluctuates depending on demand and supply curve. For instance during spring and autumn season generation decreases to

37.500-38.000 MW and approximately 40.000 MW for winter season. Thermal power plants plays important roles in terms of Turkey's safe and sustainable energy supply policy. Therefore, it's needed to develop such techniques on thermal power plants in order to decrease its weak point just as inevitable environmental effect and to be able to increase its relatively lower thermal efficiency. In this case, we apply methods for one of renewable techniques which is actually waste heat recovery used to reduce thermal waste in steam power plant. By looking at thermal power plants it is basis on pulverized coal firing system, integrated with its components, in the steam generator (boiler) is to provide through the combustion of certain bituminous coals the necessary thermal energy for the generation of steam which in turn is required for driving the downstream turbine with downstream generator for power generation. Boiler is such a massive heat exchanger that approximately 1500 MW thermal power has been transferred by the means of heat transfer methods as radiation, convection and conduction onto the steam circulating inside evaporator walls and other bundles. The coal system for the steam generator to be operated with a steaming capacity of 1887 t/h (%100 boiler load) is an opposed firing system vertically staggered burner arrangement. The required fuel heat output of the firing system within 100% boiler load is 1454 MW.

Flue gases leaving the boiler combustion chamber pass over the convection banks of the super-heaters, re-heaters and then enter regenerative air heater. There the gases exchange their thermal energy with the primary and secondary air. The temperature of the gas entering the air heater is reduced from about 400 ºC to 150 ºC. The cooled gases then pass to electrostatic precipitator where most of the dust is removed. Via induced draft fan, the cleaned gases then pass to the FGD (flue gas desulphurization) plant for desulfurization and then to atmosphere via the stack. A method apply on flue gas path that is already in use economizer as primary recovery and regenerative air preheater as secondary recovery. In addition to that operation it is briefly ORC takes place after regenerative air preheater section and the target is to utilize waste heat of flue gas either via power generation up to few MW values.

Depending upon the fluid selections recovered value converted to electric power can be altered.

Design and modeling of steam power plant on Ebsilon Professional program based on real heat and flow values so the results taken by Ebsilon Professional obeys all thermodynamic laws and engineering formulas as well. Both cycle (Air/Flue gas cycle and Organic Rankine Cycle) in cooperation each other is plotted and designed according to mass, energy and exergy balance principle. Designed power plant model is then compiled and then the data given is run thanks to simulation capability of Ebsilon Programme assistance.

Two proper organic fluids (R236FA, R245FA) were used as working fluid. On the basis of results steam powered circulating water pump has been adapted to system via engaged with ORC turbine. Working fluid strongly depends on the conditions of boiler exhaust temperature and flue gas contents. Thus, it may work on different working fluids in spite of having less recovery values respect to protect system and its auxiliary equipment(flue gas ducting, absorber, stack) based on environmental limitations(acid rainfall). Regarding the other facts working fluid options could be duplicable like R365MFC, R123 and R245CA. Economic analyses of waste heat recovery system has been computed and commentated on the results. Full load (660 MW) parameters were considered during calculations. Regarding design parameters, acquired results has been discussed. Thermodynamic, economic and environmental facts in series, are the main restriction of the study. Environmental facts to be considered as enough importance as it could possible. Thus, the flue gas temperature after utilizing itself does not go down below critical and harmful effect level.

Based on thermodynamic results the recovered amount for the usage of R236FA 4.67 MW from the flue gas waste and 4.78 MW recovered via using R245FA. Those values are equal to 8.49% of all internal consumption of unit for R236FA and 8.65% for R245FA. This reflects positively to system as 0.314% efficiency increase by R236FA and 0.322% by R245FA. Amount of exergy destruction is greater in boiler section for both used organic fluid and condenser section as secondary exergy lost component. It seems exergy is destructed in less amount while computing pump parameters. ORC efficiency is 11.85% for R236FA 12.75% for R245FA as well.

Also, economic analyze will deal with several inter related parts. Those are basically as investment costs, operating and maintenance costs and return of investment part (ROI). Investment costs consist of required modification on existing system, main parts and components of waste heat recovery system (WHRS) and subsequently its piping, auxiliary systems as well. Operating hour parameters of components is taken from power plant archives.

The return of investment by using R245FA for different scenario options both yielded design project of ORC is in feasible range in terms of whole operation life of steam power plant respectively. According to the Scenario A (8000 hours steam powered pump running) ROI becomes 5.02 years and to the Scenario B (5000 hours steam powered pump running)

ROI becomes 6.3 years. In this case, regarding to whole life of steam power plant 11.1% is going to be spent to equalize payback period for high demand operating hours scenario and %14 for low demand operating hour scenario respectively. To be able to shorten the payback period it is necessary to take into account different options while maximizing recovered values because of that reason superior maintain ability of flue gas ducting and stack coating against acid formation on path. Hence, it could go below 110 ^{O}C limit downstream ORC evaporator, reducing waste as well.

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LIST OF ABBREVIATIONS

BWR	: Back work ratio
CO	: Carbon monoxide
CO_2	: Carbon dioxide
E_{in}	: Energy inlet
E_{out}	: Energy outlet
ESP	: Electrostatic Precipitator
FGD	: Flue gas desulphurization
GHG	: Greenhouse gas
GWP	: Global Warming Potential
h	: Enthalpy
h_{in}	: Enthalpy at inlet
h_{out}	: Enthalpy at outlet
h_0	: Enthalpy at dead state
HC	: Hydrocarbons
HFCs	: Hydrofluorocarbons
HFEs	: Hydrofluoroethers
HP	: High Pressure
I	: Irreversibility
ICE	: Internal Combustion Engine
IP	: Intermediate pressure
KE	: Kinetic Energy
kW	: Kilowatt
LP	: Low Pressure
LTE	: Low Temperature Economizer
$\dot{m}$	: Mass flow rate
$\dot{m}_{in}$	: Inlet mass flow rate

Mtoe	: Million tons of oil equivalent
$\dot{m}_{out}$	: Outlet mass flow rate
MW	: Megawatt
NO_X	: Nitrogen oxides
ORC	: Organic Rankine Cycle
ODP	: Ozone Depleting Potential
PE	: Potential Energy
Q	: Heat Transfer
Q_{in}	: Input heat
Q_{loss}	: Heat loss
Q_{out}	: Output heat
SAPG	: Solar Aided Power Generation
S_{gen}	: Entropy generation
s_0	: Entropy at dead state
SO_X	: Sulfur oxides
T_0	: Ambient temperature
V_{in}	: Inlet velocity
V_{out}	: Outlet velocity
W	: Work
W_{in}	: Input work to pump
W_{act}	: Actual work from turbine
W_{net}	: Net Work output
W_{rev}	: Reversible Work
X	: Wetness Factor
X_{des}	: Rate of exergy destruction
z_{in}	: Level at inlet
z_{out}	: Level at outlet
ΔE	: Change of Total Energy
ΔU	: Change in internal energy

ΔE_k : Change in kinetic energy

ΔE_p : Change in potential energy

θ : Energy of flowing fluid per unit mass

η_{cyc} : First Law Efficiency

η_{II} : Second Law Efficiency

ΔX_{sys} : Rate of change of exergy

ψ : Exergy per unit mass

ψ_{in} : Specific exergy at inlet

ψ_{out} : Specific exergy at outlet

1. INTRODUCTION

Energy is the power and indispensable part of life in terms of several aspects. In the near future, we must have kind of energy features such as sustainable, affordable and environmentally friendly supplies. Conventional fossil energy sources will be replaced by renewable sources, gradually. Renewable technologies are considered as clean sources of energy and optimal use of these resources minimize environmental impacts, produce minimum secondary wastes and are sustainable based on current and future economic and social needs (Nrel, 2013).

It is generally expected that coal will continue to play a key role in the future energy mix as it is the most abundant and cheapest fossil fuel source. Such solid fossil fuels are combusted in steam power plants, where the power cycle is based on the steam-Rankine thermodynamic cycle, using a steam turbine.

Fossil fuels has massive influence on world electricity production. The majority of the world's power generation is met by particularly coal and natural gas. However the growth of renewable energy installations like wind, solar and geothermal power, the tendency on fossil fuels is expected to continue until 2050.Inspite of the depletion threat of fossil fuel reserves and environmental concerns such as climate change, the growth in oil demand is expected to be 47.5% between 2003 and 2030, 91.6% for natural gas and 94.7% for coal (Regulagadda et al, 2010).

According to International Energy Agency (IEA) estimations this amount will reach the values, between 14.8 billion toe and 18.7 billion toe by the year 2035, with a raise of 16–47%, approximately. The world's energy consumption expected to increase by more than 40% by the year 2035 (Ministry of Energy and Natural Resources (MENR)). Table 1.1 shows world primary energy demand in years. Total demand increased by 40% in 20 years.

Table 1.1. World Primary Energy Demand Scenario 2008-2030 (International Energy Agency 2009) (Mtoe)

	2008	2015	2030
Coal	3284	4023	4908
Oil	4193	4525	5109
Gas	2612	2903	3670
Nuclear	720	817	901
Hydropower	275	321	414
Biomass and waste	1176	1375	1662
Other renewables	94	158	350
Total	12,354	14,121	17,014

Mtoe: Million tons of oil equivalent.

A major challenge for the future electric grid is to integrate renewable power sources such as solar, wind and biomass also types of waste heat recovery applications on conventional power plants (United States Senate, One Hundred Eleventh Congress, 2009). Figure 1.1 indicates that coal is leading world electricity distribution whereas renewable leads for Europe. Energy policies of global and Europe are differing from each other.

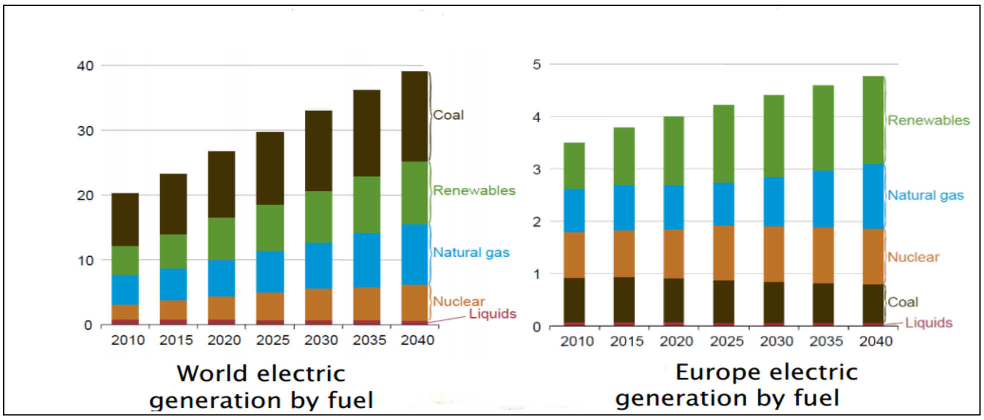

Figure 1.1. Global& European Energy Mix towards 2010-2040 (U.S Energy Information Administration 2015).

Such sources own both many advantages and some disadvantages. Some of those present as follows. Renewable are variable and clean, intermittent unlike traditional sources that provide a controllable, steady stream of power. Instead of fully replacing renewable by conventional systems, it's better to implement partial integration of renewable methods on such conventional power plants already in operation.

Greenhouse gas (GHG) emissions are doubtful for future life so one of the way to decrease role of conventional plant products (harmful emissions) or more relying on sharing overall portfolio with renewable energy. Whether for electricity, heating, cooling, gaseous fuels or liquid fuels, including integration directly into end-use sectors, the renewable energy integration challenges are site specific and include the adjustment of existing energy supply systems. There are multiple pathways for increasing the shares of renewable energy, some relates to directly energy source and some relates to appropriate integration methods such as waste heat recovery applications. The type of integration varies depending on region, characteristics specific to the sector and the technological issues (Hondo H, 2005).

Renewable energy systems are increasingly being used for electricity generation, either at small-scale decentralized systems with capacity in the kW scale or even medium-scale systems (often called utility-scale) with capacity of a few MW. The more energy consumption needed due to global technological and industrial development the more installed capacity to satisfy the energy needs thus it is vital utilizing renewable and conventional energies either. Figure 1.2 shows that energy source types. It is basically classified into two parts. Major part of generation belongs to fossil fuels in the world.

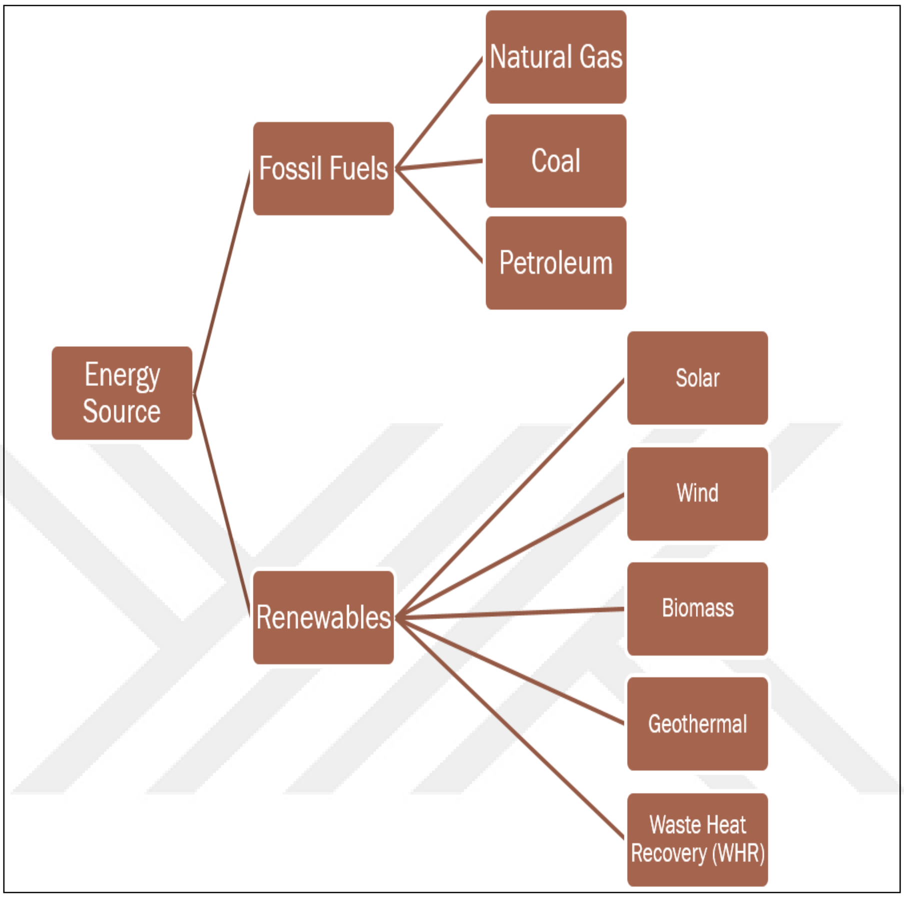

Figure 1.2. Energy source types

One of the most important disadvantages of conventional technologies is the environmental impact. The combustion of fossil fuels such as coal leads to the unavoidable production of carbon dioxide (CO_2), during operation rest of harmful gases are to be considered such as carbon monoxide (CO), nitrogen oxides (NO_X), sulfur oxides (SO_X), unburned hydrocarbons (HC), and solid particles. In order to protect environment against pollutants, auxiliary units being installed on steam power plant such as De-NOx and FGD (Flue gas desulphurization). Another remarkable disadvantage of conventional technologies is that they need perpetual fuel supply to operate, which contributes to the operating costs.

On the other hand, renewable energy technologies do not require any fossil fuel during normal operation. It is replaced by clean and more futuristic sources. Their operation idea is based on the benefit of natural resources, such as the sun and wind, lowering input resource operating costs, although they still need to account maintenance cost. Lastly, it should be aware of renewable energy technologies is still questionable of the fluctuation of their power output (Kosmadakis G et al, 2011). There are many advantages and disadvantages of fossil and renewable sources either. Figure 1.3 states that advantages of fossil fuels are generation stability, easy to control and cheap. On the other hand emission trouble is weak point of fossil fuel sources. Renewable sources are clean, sustainable and environmentally friendly whereas fluctuating power output is questionable for renewable sources.

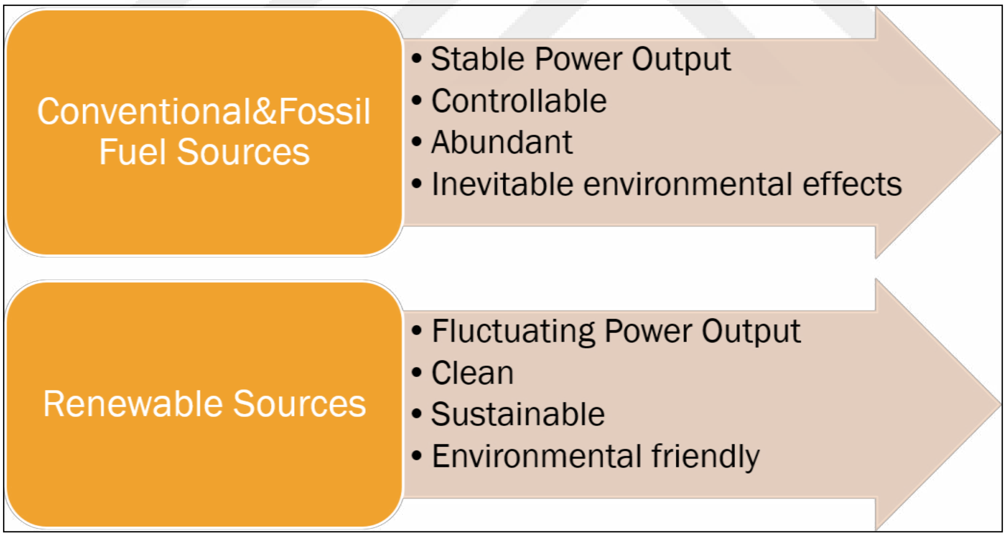

Figure 1.3. Conventional & Renewable source features

1.1. Overall Energy View in Turkey

Energy supply is backbone of the development of Turkey. Turkey's energy supplying policy is satisfying consumer total demand without any adverse effect on its economic development. As Turkey rapidly develops, Turkey's economic growth acceleration is represented by positive values. Parallel to economy, Turkey's energy industry, including thermal and renewable energy based power generation, has expanded rapidly (WECTNC, 2013).

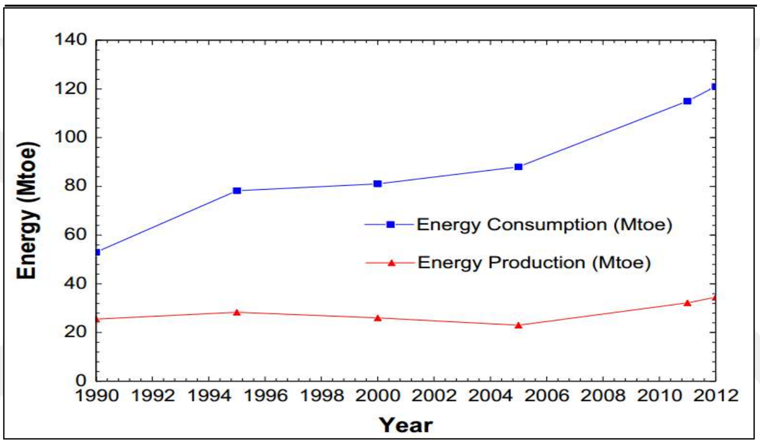

Figure 1.4. Turkey's energy production and consumption

As shown in Figure 1.4 wide gap between energy production and consumption rate is observed in Turkey. The major reason for this deficiency should be connected with the high increase in population and economic growth in spite of limitations in the domestic primary energy resources of the country (EUAS, 2013).

Renewable energy generation except for hydropower constituted only 2.6% of the total generation, wind power constituting the 2% while geothermal and renewable waste constituted less than 1%. The total installed capacity reached 87.600 MW, of which 24% is the renewable capacity hydropower constituting 32% of the total capacity.

The Turkish government has declared the importance of energy security in its strategy for the sector. Domestic coal, renewable energy, and to some extent nuclear power, are going to lead the national goal that reduce imports of natural gas and oil. It is shown in Figure 1.5 between 2014-2016 years, installed primary sources are altering. For instance in 2014 and 2016 fossil fuels are leading the contribution of installed capacity. On the other hand, in 2015 hydro types are leading energy sources.

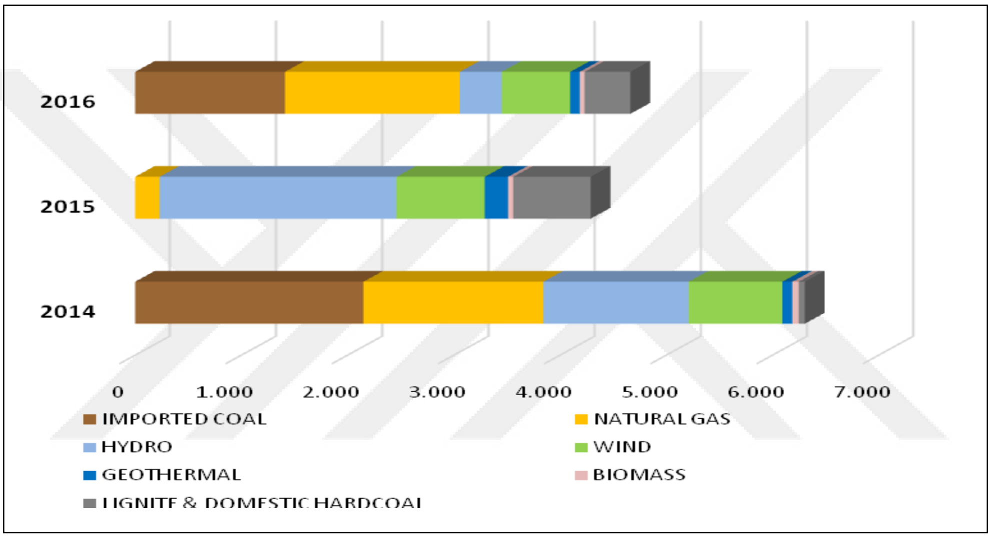

Figure 1.5. Additional installed capacity by years

Renewable targets set out in the Energy Strategy Paper are;

- 30% of total electricity production from renewable energy until 2023
- The whole economically usable hydro power potential of Turkey will be provided for generating electrical energy until 2023
- 600 MW geothermal energy will be implemented by 2023
- 20,000 MW wind energy will be in operation in 2023

(Saygin and Cetin, 2011).

1.1.1. Energy Sources in Turkey

Turkey has rich coal reserves. Total fossil energy sources of Turkey are given as 2454 Mtoe, and 48% of this amount belongs to lignite and major part obtained in Elbistan (15%) and approximately quarter of full reserve belongs to hard coal (28%). The most significant fossil energy source of the country is brown coal, and it could be noted that nearly 30% of Turkish domestic energy supply is lignite and imported coal combination. Figure 1.6 compares domestic coals rate and imported coals rate in years. It seems to be Turkey was imported coal more in recent years.

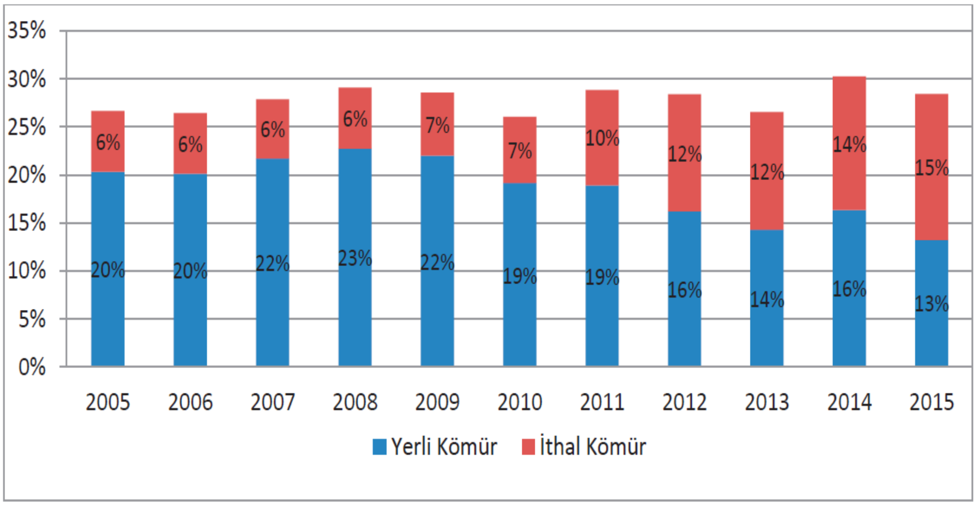

Figure 1.6. Share of coal types contribution on electricity production (TEIAS)

Table 1.2 also shows us coal fired steam power plant types, its efficiencies, years started in operation and combustion technology as well.

Table 1.2. Coal Fired Steam Plant in operation after year 2000 (M.Ersoy, 2015)

Power Plant Name	Capacity (MW)	Coal Type	Operation Year	Combustion Technology	Efficiency (%)
Sugozu	1320	Imported bituminous	2004	PC	39
Afsin-Elbistan B	4*360	Lignite	2009	PC	39
Çolakoglu	190	Imported	2012	PC	34
18 Mart-Çan	2*160	Lignite	2003	CFB	41
Silopi	135	Asphaltite	2009	CFB	38
Biga	3*135	Imported	2005	CFB	37
ZETES	160	Imported	2014	CFB	36
ZETES -2	2*615	Imported		Super Critic	41
Bekirli	2*600	Imported	2013	Super Critic	42
Atlas	2*600	Imported	2014	Super Critic	44
İzdemir	350	Imported	2014	Super Critic	44

Nearly 80 % of the natural gas production in the country is provided in Hamitabat field in the North West region of Turkey. 98.5% of natural gas demand of Turkey is met by imports from Russia (58%), Iran (19%), Algeria (9%), Azerbaijan (9%), Nigeria (3%) and spot import (2%). (BOTAS, Petroleum Pipeline Company). Electricity generation in Turkey by natural gas are dominated in recent years. Despite the futuristic policy rely on domestic sources, it seems natural gas still play an important role in both low demand and high demand scenarios.

In Figure 1.7 60% of total generation belongs to fossil fuels in Turkey's electricity generation distribution. Natural gas is individually satisfied 30% of whole generation of Turkey's generation amount.

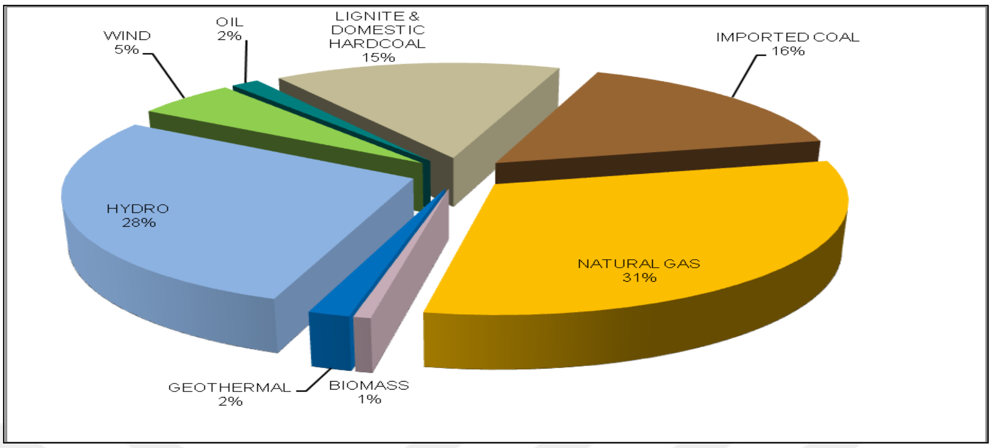

Figure 1.7. Electricity Generation Distribution Depend on Energy Sources (2016)

It should be noted that renewable energy generation capacity is not necessarily a direct replacement for fossil-fuelled generation capacity. There are several reasons for this discrepancy. First, as fossil generation capacity is not needed of on the availability of natural resources, such as wind or solar radiation, coal can be transferred as required. As a result, availability and contribution to load for thermal generators tend to be considerably higher than renewable generators. Second, the geographical distribution of renewable energy resources means that wind and solar power may be available but transmission may be limited by the capacity of the electricity grid. This may limit the ability of renewable resources to meet local demand. Due to these reasons a greater renewable energy generation capacity may be needed to replace an equivalent quantity of fossil-fuelled capacity (The International Institute for Sustainable Development, 2015).

1.2. Steam Rankine Cycle

The Rankine vapor power cycle shown in Figure 1.8 is most widely used thermodynamic cycle for power generation. William John Macquorn Rankine is the Scottish engineer who along with Rudolph Clausius and William Thomson developed the scientific background of this cycle. Rankine cycle has mainly four components (pump, boiler, turbine and condenser). To be able to make advanced and more efficient cycle has been modified (Moran MJ et al, 2006).

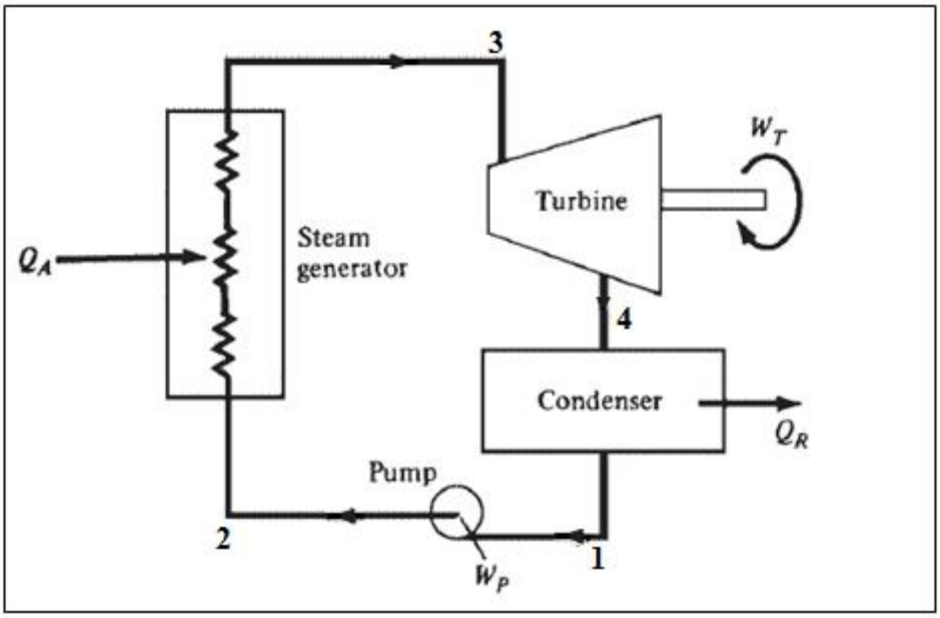

Figure 1.8. Steam Rankine Cycle (Mc Graw Hill, 2002)

The ideal theoretical Rankine thermodynamic cycle consists of four processes as shown in Figure 1.8;

Process 1-2: isentropic compression by a pump
Process 2-3: isobaric heat addition in a boiler
Process 3-4: isentropic expansion in a turbine or expander
Process 4-1: isobaric heat rejection in the condenser

Figure 1.9 shows actually modified Rankine Cycle that steam extraction has been added to system. In this way, preheating the condensate and feed water, system efficiency has been increased.

In modified Rankine Cycle, water enters the pump as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The water temperature increases somewhat during this isentropic compression process due to a slight decrease in the specific volume of water.

Water enters the boiler then as a compressed liquid and leaves as a superheated vapor. The boiler is basically a large heat exchanger where the heat composing from chemical energy of coal after combustion process the product of flue gas, nuclear reactors, or other sources is transferred to the water essentially at constant pressure. The boiler, together with the section where the steam is superheated (the super-heater and re-heater as well), is often called the steam generator. The superheated vapor enters the turbine, where it expands isentropically and produces work by rotating the shaft connected to an electric generator. On the way shaft it could be a coupled turbine series which are generally sequentially HP (High Pressure) turbine, IP (Intermediate Pressure) turbine and LP (Low Pressure) turbine series as can be seen in Figure 1.10. The pressure and the temperature of steam drop during this process to the values, where steam enters the condenser. At this state, steam is usually a saturated liquid–vapor mixture with a high quality. The quality of steam is assessed by wetness factor x. Wetness factor at the end of turbine that's to say condenser inlet area is approximately between 0.85 -0.9. Steam is condensed at constant pressure in the condenser, which is basically a large heat exchanger, by rejecting heat to a cooling medium such as a lake, a river, or the atmosphere. Steam leaves the condenser as saturated liquid and enters the pump, completing the cycle (Cengel and Boles, 2006).

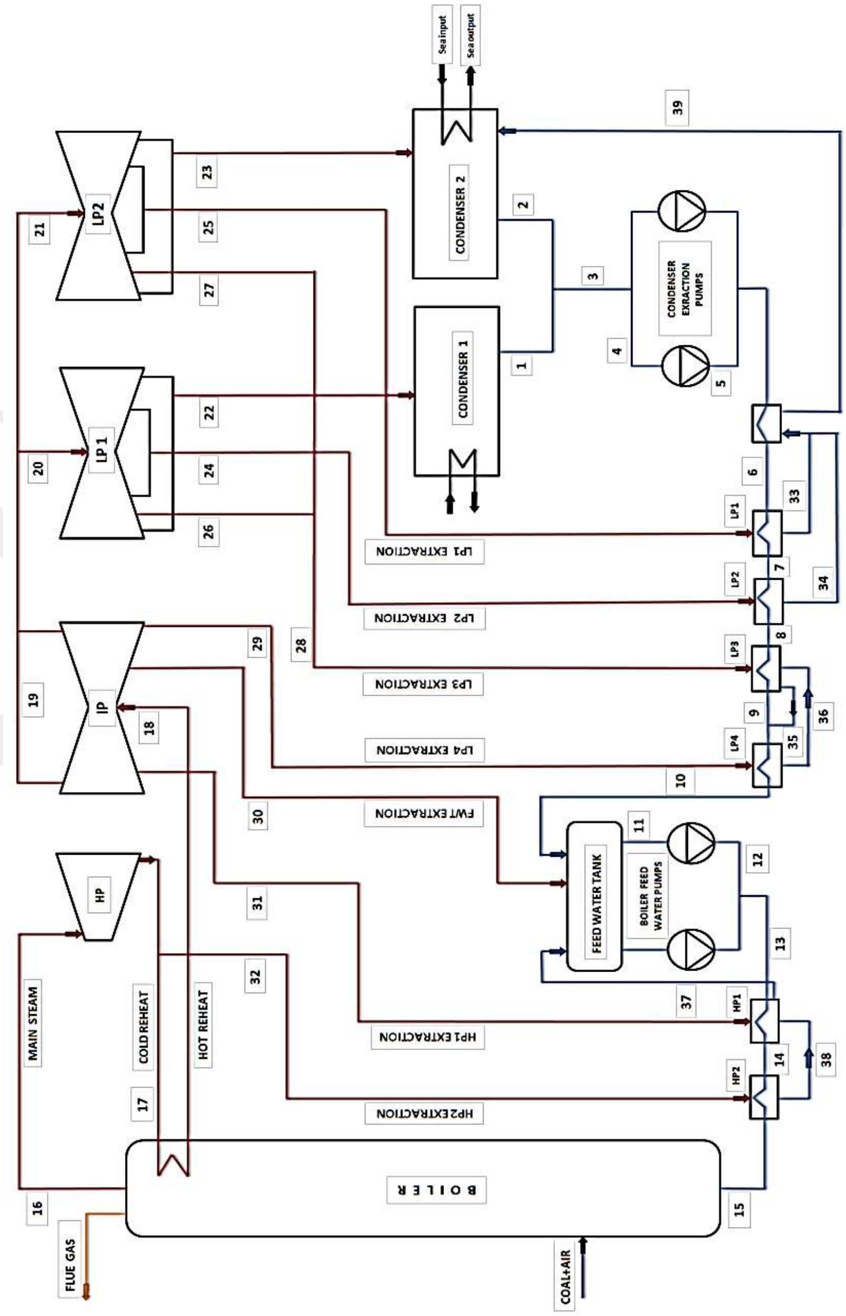

Figure 1.9. Modified Water/Steam Cycle in Steam Power Plant

Typical representation of T-s diagram of a modified steam Rankine cycle is shown in Figure 1.10.

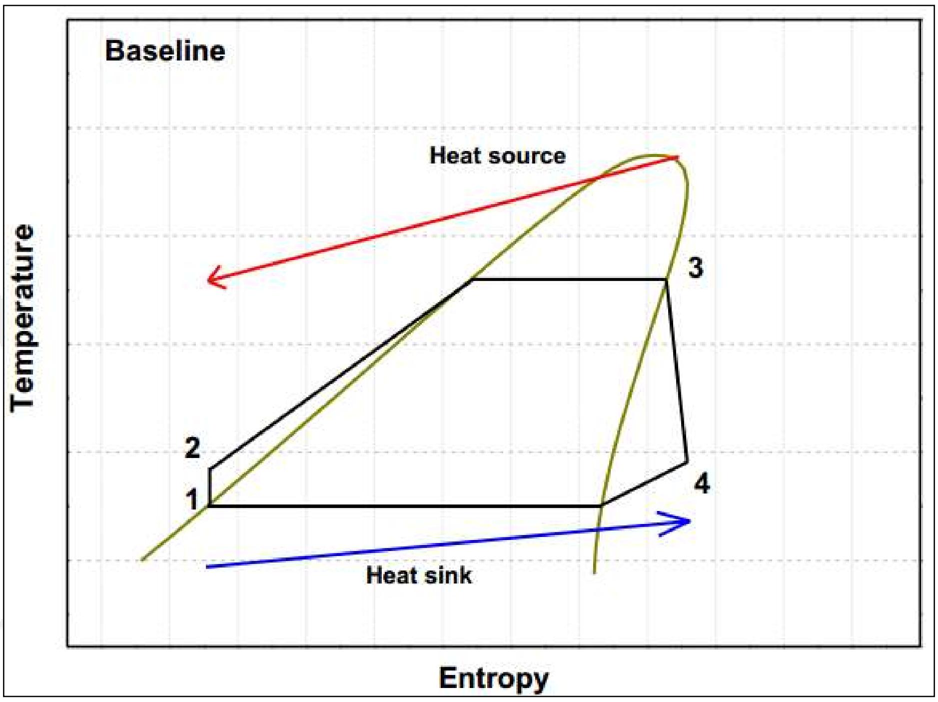

Figure 1.10. T-s diagram of a modified steam Rankine cycle

Typically, water is used as the working fluid. As shown in Table 1.3 water properties are basically;

Table 1.3. Water Characteristics (Wali E, 1980)

ADVANTAGES
VERY GOOD THERMAL/CHEMICAL STABILITY (no risk of decomposition)
VERY LOW VISCOSITY (less pumping work required)"
GOOD ENERGY CARRIER (high latent and specific heat)
NON-TOXIC, NON-FLAMMABLE AND ENVIRONMENTAL FRIENDLY (zero ODP, zero GWP)
CHEAP AND ABUNDANT
DISADVANTAGES
NEED OF SUPERHEATING TO PREVENT CONDENSATION DURING EXPANSION
RISK OF EROSION OF TURBINE BLADES (wetness factor)
EXCESS PRESSURE IN THE EVAPORATOR
COMPLEX AND EXPENSIVE TURBINES

1.3. Organic Rankine Cycle

Organic Rankine Cycle is similar to a Steam Rankine Cycle. Indeed working principle is based on vaporization of a high pressure liquid in steam generator state which is in turn expanded to a lower pressure in turbine, therefore yielding mechanical work. The cycle is completing by condensing the low pressure vapor and pumping it back to the high pressure. Therefore, the ORC involves the same components as a conventional steam power plant (a boiler, a work producing expansion device, a condenser and a pump). However, the working fluid is an organic compound characterized by a lower evaporating temperature than water and allowing power generation from low heat source temperatures.

The high temperature heat is absorbed in the evaporator at the high pressure from the heat source and the working fluid (called as organic fluid) then shift into a saturated gas state. Then the high enthalpy saturated gas is then send to the turbine in order to produce the power output in the connected generator. After that the low pressure superheated gas is cooled down to the saturated liquid in the condenser (Quoilin S, 2011).

Figures 1.11 and 12 express, ORC may depend on location, type of selected organic fluid, environmental cases, finance& budgets, source and cooling fluid that is actually heat sink and so many reasons. It's possible to say ORC offer varying large generating scale from a few kW to some MW electricity power. To be able to reach more effective and efficient cogeneration techniques, ORC can be modified as many types (exchanger and/or reheat).

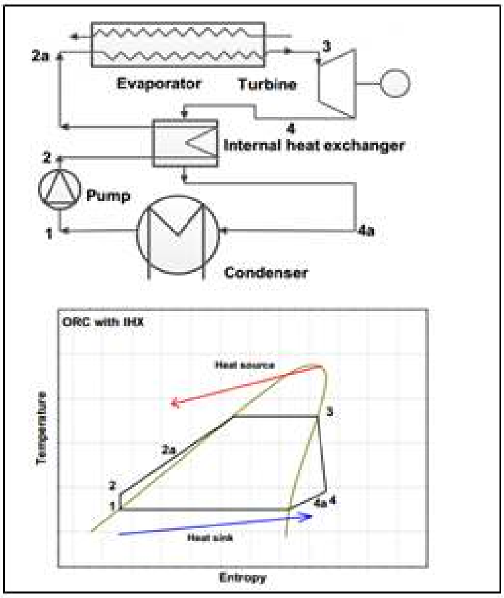

Figure 1.11. ORC with heat exchanger

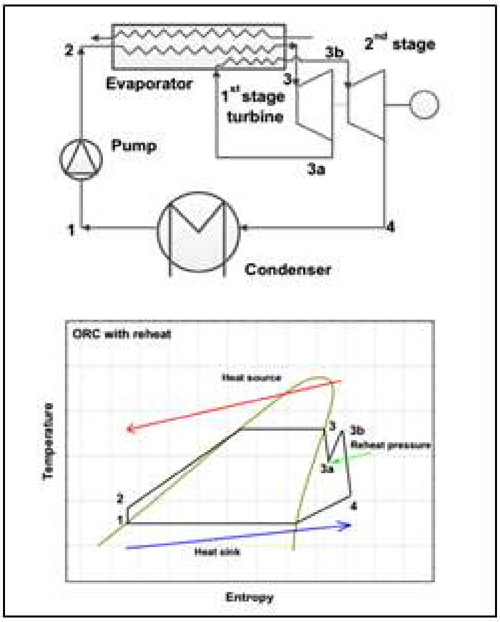

Figure 1.12. ORC with reheat

As mentioned above, working medium can be classified basically,

Low temperature medium → < 150 C^o

Mid temperature medium→ 150-300 C^o

High temperature medium → > 300 C^o

Categorization of systems follows as integrating to cogeneration applications in buildings and, in cars whereas small systems could be applied to recovered exhaust gas from small internal combustion engines (ICE).

Medium and large system sound suitable for industrial onsite additional power generation, distributed power generation in isolated areas or for grid connection. (Cengel, 2001).

ORC mainly differs from Steam Rankine Cycle in the case of fluid selections. Water is favorable in steam power plant and steam rankine cycle applications whereas water is not suitable working with ORC. Due to the lower evaporating temperature needs, different fluids just like hydrocarbons (HCs), hydrofluorocarbons (HFCs), hydrofluoroethers (HFEs), ammonia, etc takes place in ORC applications. A working fluid in ORC machine plays a key role. It determines the performance and the economics of the plant (Papadopoulos A., 2010). However, the aim of an ORC machine is the power generation and as secondary purpose cogeneration or tri-generation with hot water production for heating/cooling. Figure 1.13 shows various applications in agreement with heat source temperature and machine type.

ORC can be compared with the Steam Rankine cycle in terms of several aspects (Table 1.4).

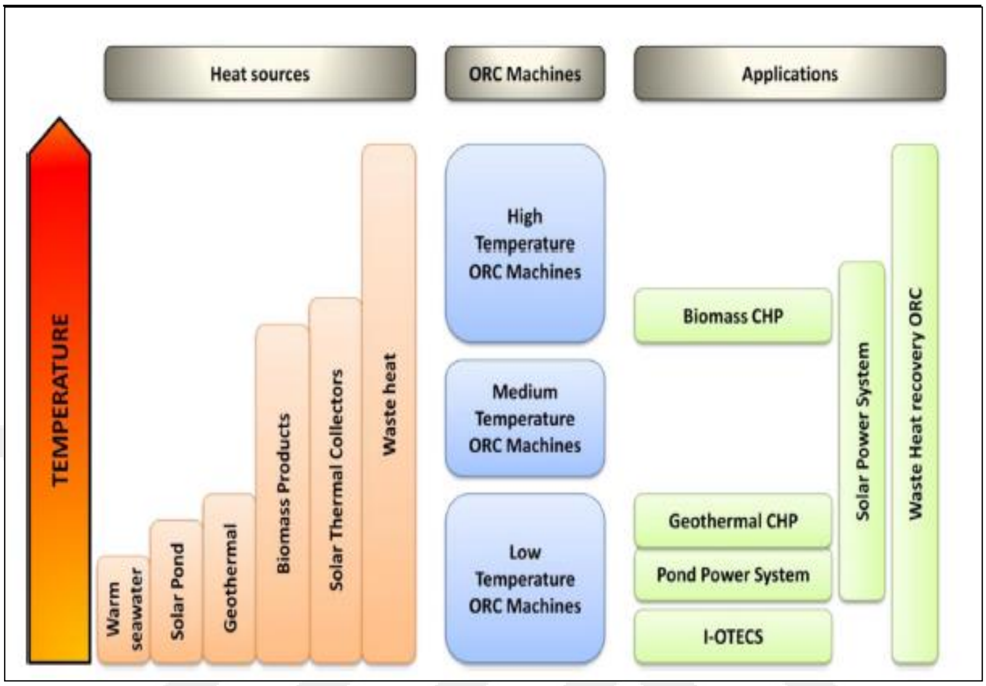

Figure 1.13. ORC applications

Table 1.4. Comparison of Steam Rankine Cycle and Organic Rankine Cycle (Vanslambrouck B. 2011)

Features	**Steam Rankine Cycle**	**Organic Rankine Cycle**
Superheating	Saturated mixture at the end of turbine blades, erosion risk	Organic fluid remains superheated at the end of turbine blades, no erosion risk threat.
Heat Recovery	Restricted heat recovery	Possible lower ambient heat recovery
Components size	Needs higher velocity thus bigger piping, heat exchangers, turbines	Relatively lower component necessity
Boiler design	Larger boiler sections, evaporator and super-heater &re-heater zones	No need steam drum and circulation pump, only heat exchanger(evaporator)
Turbine inlet temperature	Due to super-heating necessity higher inlet temperature	Lower inlet temperature
Pump consumption	Lower BWR	Higher BWR
Condensing pressure	Water, sub-atmospheric condensing pressure	Organic fluids has both higher and lower atmospheric condensing pressure
Fluid characteristics	Treatment needs, leakage possible, natural, easy to work	Large scaled , different operating opportunities
Efficiency	Higher efficiency	Relatively lower efficiency

1.4. Fluid Selection

The selection of the working fluids plays key role on cycle achievement level. Indeed it depends on quality and quantity of heat source. Working fluids include organic compounds and inorganic compounds (Bao J. et al. 2013).

Most favorite selected fluid features to meet expectations is to be non-toxic, non- flammable and non- corrosive too. Some special components like accurate diaphragm pump or hermetic expander are developed to minimize the leakage and corresponding risk of some flammable or toxic fluids. Mago PJ, (2008) says that R11 and R113 fluids are best offer to study however having high GWP (Global Warming Potential) and high ODP(Ozone Depleting Point). To be able to get best result on studies, it is be aware of fluid environment influence and thermodynamic properties together. The priority of working fluids is having good thermodynamic performance. Nevertheless, it should be optimized with environmental facts. The critical temperature of fluid is one of the most important parameters influencing cycle performance. It is listed in Table 1.5 that some of working fluid properties.

Table 1.5. Working fluid properties

Working fluids	Critical temperature (°C)	Critical pressure (MPa)	Molecule weight (g/mol)	Autoignition temperature (°C)	ODP	GWP
Organics						
R134a	101.1	4.06	102.3	-	0	1300
R227ea	101.75	2.93	170.03	-	-	3500
Butane	151.98	3.80	58.12	287	0	20
R245fa	154.05	3.65	134.05	-	0	820
R123	183.68	3.66	152.93	Non-flammable	0.02	77
Isopentane	187.2	3.38	72.15	420	-	-
Pentane	196.55	3.37	72.15	260	-	-
R11	197.96	4.41	137.4	Non-flammable	1	4600
R141b	204.35	4.21	116.95	532	0.086	700
R113	214.06	3.39	187.37	680	0.9	6000
Acetone	234.95	4.70	58.097	465	-	-
Heptane	266.98	2.74	100.2	204	-	-
Benzene	288.87	4.89	78.11	560	-	-
Octane	296.17	2.49	114.23	206	-	-
Inorganics						
CO_2	31	7.38	44	Non-flammable	0	1
Water	374	22.06	18	Non-flammable	0	0

2. LITERATURE SURVEY

It is possible to see several studies that deal with renewable energy integration to conventional systems related to thermodynamic analysis. Some studies more focused on waste heat recovery application

Xu G. et al. (2014) studied integration of waste heat recovery system on flue gas path. To achieve extra work, installed low temperature economizer after ESP takes place. Exergy analysis and economic analysis have been done and discussed on results.

Li C. et al. (2016) investigated on generated waste heat quantities during industrial production. In order to maximize net power output, an optimal combination of cycle configuration, fluid and cycle parameters under different heat source condition, the following researches have been performed. To sum up, results indicate that the regenerative organic trans-critical cycle produces the maximum power output at source temperatures up to about 500 C, and different optimum working fluids are obtained under different heat source temperature.

Li C. et al. (2016) has been studied for waste heat recovery from flue gas in a wide range temperature. The study also dealing with optimum relationship between selected fluid and heat source in order to achieve better thermal efficiency. Evaluation, optimization comparison of many cycles has been evaluated.

Han X. et al. (2016) studied on flue gas waste heat recovery method by extracting water from high moisture lignite coal thus yielding increase of boiler efficiency. Energy saving potential has also been evaluated. To achieve this, different options take into consideration like economizer or spray tower. Thermodynamic and economic performance of system are investigated for each scheme.

Xia J. et al. (2016) performed thermo-economic analysis of combined system which is named combined cooling and power system for engine waste heat recovery. Vital part of the system thermodynamically based were studied. Exergo-economic evaluation is another branch of this study. Via the single-objective optimization, the lowest average cost per unit of exergy product for the overall system is obtained.

Nazaria N. et al. (2016) proposed multi objective optimization of combined steam& Organic Rankine Cycle based on exergy and exergo economic analysis of waste heat recovery application. Combined cycle has been performed in terms of exergo economic and exergy efficiency. Total product cost rate and exergy efficiency were cornerstone of this study. Three different type fluid selection has been monitored to perform system characteristic as well.

Wang Z. et al. (2016) has been presented evaluation of flue gas waste heat and water recovery from a fossil fuel boilers. Several parameters has been studied like efficient heat and water recovery for various kinds of fossil fuel boilers, heat recovery dependency parameters like moisture content.

Zhao X. et al. (2016) presented another waste heat recovery method, known peak shaving heat pump. This study consist of improving transmission network and distribution capacity, Also study proposed by heating capacity, and reduced heating energy consumption .As such, the proposed system is advantageous in terms of energy saving, emission reduction, and economic benefits.

Frattini D. et al. (2016) performed on hydrogen production techniques by integrating renewable sources those are biomass and biogas already. This study capture impact of renewable integration with ammonia synthesis plant and also comparing energy efficiency, investigating sustainability issues and evaluating greenhouse gas emission effects.

Zhai R. et al. (2016) completed study about solar power cooperation and conventional system. The study case here is by analyzing solar aided coal fired power plant in terms of thermodynamically and economical aspects. Design study based on integrating solar power into the regenerative Rankine cycle section of plant that is to utilize renewable source by preheating feed-water and therefore saving extracted steam and fuel as well. Rankine cycle components also considered as energy and exergy analyses.

Wang D. et al. (2011) studied based on waste heat recovery of flue gas after combustion. The aim of study briefly extracting water vapor and its latent heat from flue gas. Obtaining water vapor that is demineralized and well quality can be supplied internally to power plant application some of those are contributing directly to water steam cycle process as boiler make up water or using other basic units which Flue gas desulfurization can be accounted as one of those.

Peris B. et al. (2015) dealt about experimental study of Organic Rankine Cycle application for low grade heat recovery on ceramic industry. Performance of Organic Rankine equipment has been verified, according to data obtained, available system has been modeled, energy and environmental impacts has discussed on experimental results.

Bai Z. et al. (2016) designed new solar biomass power generation. The target is to produce solar fuel named gasified syngas. By using the fuel directly in combined cycle for power generation overall efficiency has been investigated numerically. Some of simulation also performed to evaluate thermodynamic performances.

Hou H. et al. (2015) studied integrated solar aided power plant application. The target of study is highlighted as a coupling analysis about solar aided power generation (SAPG) system is made. The off-design performances of turbine and boiler are considered in SAPG system. Matrix Thermal Balance Equation is used in SAPG system models.

Xu G. et al. (2014) searched for increasing efficiency and one of the most applied methods to use is benefiting extraction steam thereby obtained extra power work. Instead preheating condensate water by extraction steam, they studied on waste heat recovery deal with segmented air preheater thus possesses High temperature and low temperature air preheaters. Besides, LTE is settled between LT air heater and ESP zones. In this way it is obtained power plant efficiency increment and more benefits on economics of power plant.

3. MATERIAL AND METHOD

Energy has many type of forms that exists as mechanical, thermal, chemical and electrical too. Transfer of energy forms is separated into two groups called heat and work. For control volumes, energy can also be transferred by mass flow. Temperature difference is main reason to energy transfer to or from a closed system as in heat form.

Energy values of heat and work flows are absolute, while the energy values of material flows are relative. Enthalpy examination depends on a reference level. For the determination of some energy efficiencies, however, the enthalpies must be evaluated relative to specific reference levels (Dinçer and Rosen, 2011).

The energy that a system possesses as a motion caused is called kinetic energy (KE). The energy caused because of elevation is called potential energy (PE). Total energy of system consist of multiple energy forms including internal energies, kinetic energy and potential energy of whole system. Despite electric, surface tension effects and the magnetic issues are another additional forms, they are generally neglected. (Cengel and Boles, 2006).

$$KE = m * \frac{v^2}{2} \tag{3.1}$$

$$PE = m * g * z \tag{3.2}$$

$$E = U + KE + PE = U + m\frac{v2}{2} + mgz \tag{3.3}$$

3.1. First Law of Thermodynamics

The first law of thermodynamics, also known as the conservation of energy principle. It deals with the relationships between energy forms and its interactions. The first law of thermodynamics states that energy can be neither created nor destroyed during a process; it can only change forms. The first law of thermodynamics is the law of the conservation of energy, the principle emphasizes forms of energy interchangeable but it can be neither created nor destroyed. For all adiabatic processes among closed systems, the net work done is the same due to the no heat transfer (Q=0) between system and its surroundings (Çengel, 2006).

Besides the conservation of energy principle can be expressed as follows: The net change (increase or decrease) in the total energy of the system during a process is equal to the difference between the total energy entering and the total energy leaving the system during that process.

Total Energy Entering the System – Total Energy Leaving the System = Change of Total Energy

$$E_{in} - E_{out} = \Delta \mathrm{E} \tag{3.4}$$

Energy change in the system is determined by the difference between the initial and final states.

$$\Delta \mathrm{E}_{\mathrm{sys}} = E_{final} - E_{initial} = E_2 - E_1 \tag{3.5}$$

While the system in motion, changes in the kinetic energy ΔE_k and potential energy ΔE_p must be added to the change in internal energy ΔU. These three terms together constitute then the change in the energy of the system, ΔE. Thus the more general form of the energy balance is:

$$Q - W = \Delta \mathrm{E} \tag{3.6}$$

$$\Delta \mathrm{E} = \Delta U + \Delta E_k + \Delta E_p \tag{3.7}$$

Where,

ΔU = change in internal energy

ΔE_k = change in kinetic energy

ΔE_p = change in potential energy

Energy balance equation for steady flow processes as:

$$\dot{E}_{in} = \dot{E}_{out} \tag{3.8}$$

$$\dot{Q}_{in} + \dot{W}_{in} + \Sigma_{\text{in}}\, \dot{m} * \theta = \dot{Q}_{out} + \dot{W}_{out} + \Sigma_{\text{out}}\, \dot{m} * \theta \tag{3.9}$$

$$\theta = h + ke + pe \tag{3.10}$$

Thus,

$$\dot{Q} - \dot{W} = m\left[h_{out} - h_{in} + \frac{{v_{out}}^2}{2} - \frac{{v_{in}}^2}{2} + g[z_2 - z_1]\right] \tag{3.11}$$

Mass balance equation for processes is expressed as:

$$\dot{m}_{in} = \dot{m}_{out} \tag{3.12}$$

Despite the first law of thermodynamics is universally accepted law of thermodynamics, it only states the quantity, on the other hand restricted expression about the quality term. Therefore, the second law of thermodynamics is required to establish the difference in quality between mechanical and thermal energy and to indicate the directions of spontaneous processes.

3.2. Second Law of Thermodynamics

The second law deals with quality term that is to say evaluation of limits that takes a place on process from heat to mechanical work. Interchanging of work to heat as always actualized, on the other hand heat cannot be fully converted to work. The slice that dissipated during conversion of heat to mechanical is called unavailable energy. Here we come up with a term called entropy is always generated by the nature of process conversion. The second law states that the thermal efficiency of converting heat to work, in a power plant, must be less than 100 percent (Mc Graw Hill, 2002).

Entropy generation is an expression that indicates the determination of process possibility in three ways

$$S_{gen} = \Delta S_{total} = \Delta S_{sys} + \Delta S_{sur} \tag{3.13}$$

In this way,

$$S_{gen} \begin{cases} > 0 \ Real\ Process \\ = 0 \ Ideal\ Process \\ < 0 \ Impossible\ Process \end{cases}$$

Exergy is a definition that measure work potential of energy. Exergy is only conserved or in balance for a reversible process, but partly consumed in an irreversible process. Thus, exergy is never in balance for real processes (Dinçer and Rosen, 2011).

Exergy Balance Equation for Steady Flow Processes as:

$$X_{in} - X_{out} - X_{dest} = \Delta X_{sys} = 0 \tag{3.14}$$

Where,

X_{in} - X_{out} = Rate of net exergy transfer by heat, work and mass

X_{des} = Rate of exergy destruction

ΔX_{sys} = Rate of change of exergy

Irreversibility (I),

$$I = T_0 S_{gen} \tag{3.15}$$

Or;

$$I = W_{rev} - W_{in} \tag{3.16}$$

Availability, maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state.

$$\psi = (h - h_0) - T_0(s - s_0) \tag{3.17}$$

Second law efficiency for turbine, pump and fan express as;

$$\eta_{\text{second law eff.}} = \frac{W_{act}}{W_{rev}} \tag{3.18}$$

For heat exchanger,

$$\eta_{\text{second law eff.}} = \frac{m_{cold}(\psi_4 - \psi_3)}{m_{hot}(\psi_1 - \psi_2)} \tag{3.19}$$

3.3. Coal Fired Steam Power Plant

Coal is an important energy resource for meeting the further demand for electricity, as coal reserves are much more abundant than those of other fossil fuels. Coal fired power plants are a type of power plant that make use of the combustion of coal in order to generate electricity. Their use provides around 40% of the world's electricity and they are primarily used in developing countries (World Coal Association, 2015). Power plants are actually operated in series of energy conversion that is being finalized electricity generation. Due to the first chain of primary energy supply is coal, those kind of power plants are known as coal fired steam power plant. In order to convert fossil fuel based chemical energy to electromagnetic energy, steam power plants are established. Basically, the following proceedings happen in steam power plant as processes;

The chemically bound energy of fuel is converted into thermal energy of hot flue gases in the furnace by means of combustion with air. These flue gases are cooled down at the heating surfaces of the steam generator and transfer their thermal energy to water and steam along with it. The heating surfaces cause the water to evaporate. It becomes superheated. At the same time it absorbs thermal energy. The turbine converts this thermal energy of steam into mechanical energy of rotor speed of turbine and generator. The generator converts the mechanical energy into electromagnetic energy that is available at the generator terminals (Babcock, 2002). Working principle of steam power plant is shown in Fig 3.1.

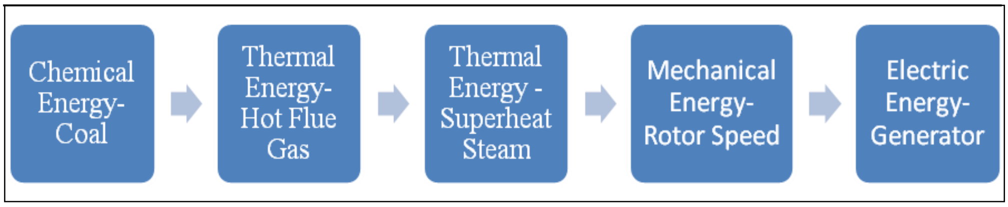

Figure 3.1. Energy Conversion in Coal Fired Steam Power Plant

Coal is supplied to the steam generator after pulverizing in the fuel drying plant combined with the combustion air. It generates slag and flue gas at the combustion in the furnace. The flue gas is purified from the containing nitrogen oxides in a Denox plant by using ammonia, in a following precipitator (ESP) it is particularly removed under generation of fly ash, afterwards it is desulfurized in a flue gas desulfurization plant (FGD) by using limestone resulting the generation of gypsum. The purified flue gas flows through stack into the atmosphere. Besides, limestone the flue gas desulfurization plant also needs air and additional water to absorb harmful substances that is partially emitted again as waste water.

The feed water supplied to steam generator is heated by flame and flue gas, is evaporated and superheated. The generated superheated steam of high pressure carries out work in the turbine that is converted in the attached generator into electromagnetic energy. Figure 3.2 shows overall view of steam power plant.

Figure 3.2. Coal fired steam power plant

3.4. Combustion Technology in Steam Power Plant

Nowadays, there are many methods adopted for burning fossil fuels such as coal, named fixed bed combustion method, fluidized bed combustion method and pulverized fuel combustion method too. Because of the coal fired steam power plant is to be analyzed operated by pulverized coal combustion method, brief description will be about this method.

In this technique, finely ground coal is sprayed into an air stream from the burners. This allows adjustment to within narrow limits of the fuel/air ratio, which is vital to efficient combustion and to the minimization of flue gas heat losses. Carbon burn out efficiency is higher, gas velocities are deliberately kept high in order to entrain all the fuel and ash. Combustion temperatures are high (typically over 1500 K) giving fast combustion, but also causing the undesirable vaporization of some of the components in the fuel (for instance alkalis and heavy metals). Melting of some of the ash also occurs. Although pulverized fuel firing offers many advantages, it requires the provision of coal mills to produce the finely powdered coal, and these mills are expensive, making the technique most suited to larger output boiler units generally more than 100 MW (National Power PLC, 1997). In Figure 3.3, Benson type boiler arrangement and its instrument was shown.

Here are design parameter of boiler and primary fuel coal are as follows:

- Dry ash boiler
- Burner arrangement 24 low NO_x DS burners at four different levels
- Six burners being assigned to one mill distributed to the front and rear wall
- Tower type construction
- Evaporation system : Benson type boiler

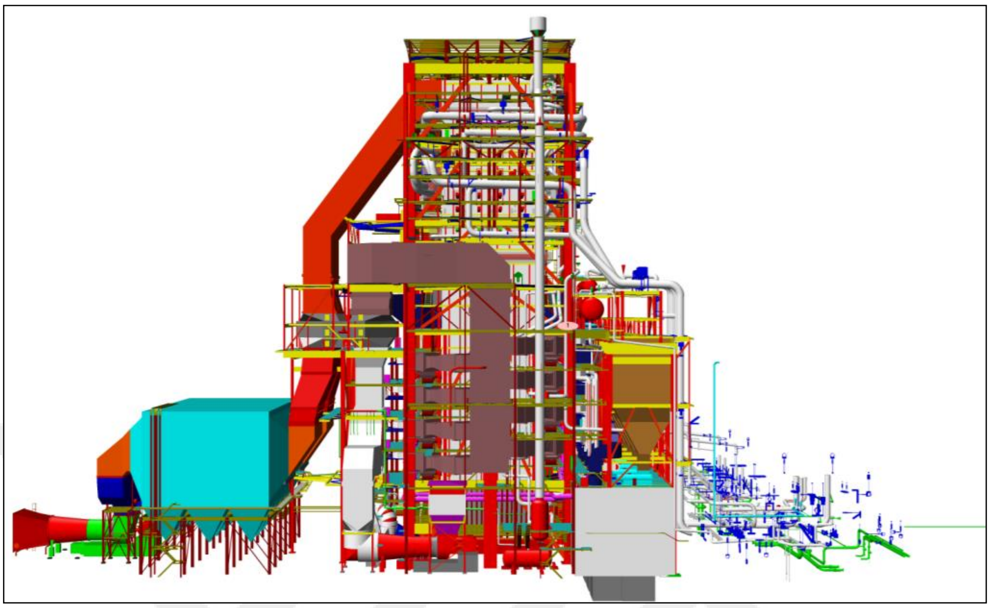

Figure 3.3. Boiler arrangement in coal fired steam power plant

The coal system for the steam generator to be operated with a steaming capacity of 1887 t/h (100% boiler load) is an opposed firing system vertically staggered burner arrangement. The required fuel heat output of the firing system within 100% boiler load is 1454 MW.

Table 3.1 shows fuel analyze report. According to data carbon as known main source of obtained thermal power after combusting coal in the boiler. Also, coal contains sulphur and nitrogen. After combustion takes place, sulphur and nitrogen involved in harmful gases such as SO_2 and NOx.

Table 3.1. Fuel analyze data

Parameters	Unit	Design Coal	Fuel Range
Carbon	%	66	65 -
Hydrogen	%	3.66	3.5 - 4.6
Oxygen	%	4.16	2.9 - 7.0
Nitrogen	%	1.5	1 - 1.7
Sulphur	%	0.9	0.6 - 1.5
Ash	%	13.72	8.0 - 17.5
Water	%	10	7 - 12.0
Volatile	%	30	23 -
Low Heating Value	kJ/kg	25750	23820 - 27110
Deformation Temp.(Slag)	^{o}C	1150	>1150
Softening Temp.	^{o}C	1290	>1290
Fluid Temp.	^{o}C	1360	>1360

Burner structure and its subcomponents are shown in Figure 3.4 and Figure 3.5. Burner is basically consist of segmented air ducts, flame monitors oil lance, igniter and swirls as well.

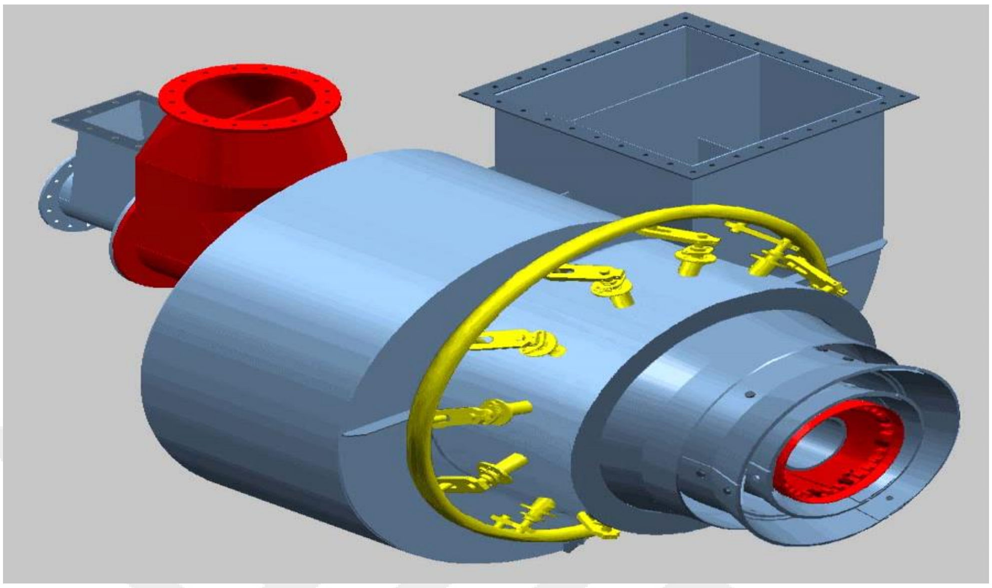

Figure 3.4. Burner structure

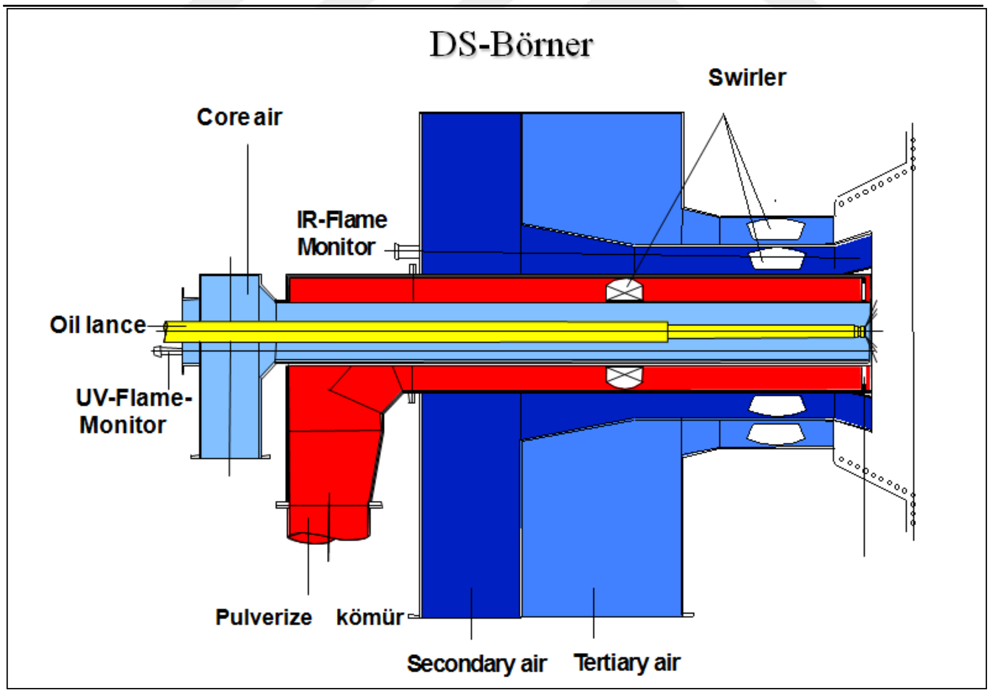

Figure 3.5. Low NO_x DS burner

The burners are arranged at 4 half levels on the front wall and 4 half levels on the rear wall of the boiler. In each case 2 vertically staggered and opposed half levels with 3 burners each belong together and form one whole burner level with 6 burners. Each burner level with 6 burners is supplied by one coal mill.
Raw coal is transported from the coal bunker through a feeder into the mill where it is uniformly ground and dried. Drying as well as subsequent transport of the pulverized coal through the various pulverized coal pipes to the burners take place by means of carrier air.

The coal mill (Figure 3.6) consist of spring loaded, free running grinding rollers, which are rotated by the table as it revolves under power. Coal is carried between the rollers and the table, and the process is continued until the desired fineness required for pulverized fuel is achieved.

Hot primary air is fed into the body of mill at high velocity, via number of angled cones which control the direction of air flow. This high velocity air carries the ground coal upwards into the classifier, which ensures that only particles of the requisite fineness are passed to the burners. Larger particles are returned to the table to be ground. The roller mill is a medium speed device, with a table rotating speed. The rollers, although spring loaded in order to apply crushing force to the coal, are fitted with adjustable stops to prevent metal to metal contact between rollers and the table (National Power PLC,1997) .

The coal mass flow is controlled, as a function of the boiler load demand, through the velocity of the feeders, taking into account the respective coal weight measured, normally the coal is delivered uniformly through all operating mills (Babcock, 2002).

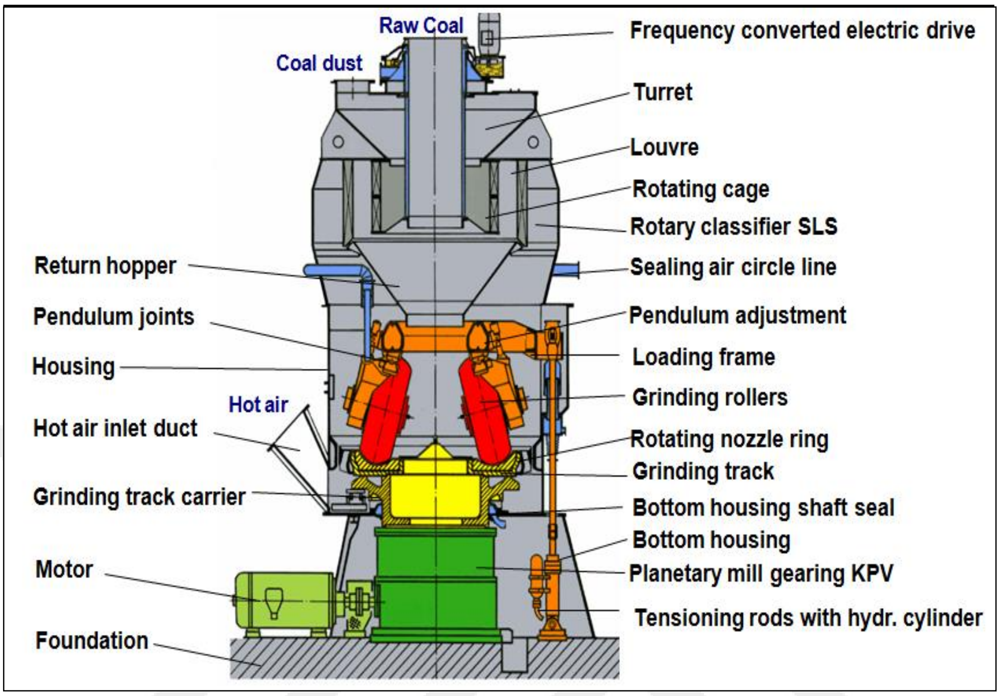

Figure 3.6. MPS mill with rotary classifier and hydropneumatic system

3.5. Air / Flue gas System

The purpose of the system is to supply sufficient air demand in order to complete full combustion of fuel which is actually coal for normal operations and oil for start-up /shut down operations. All combustion air passes initially through forced draught fan (FDF). Downstream of the fresh air is branched off via primary air fan to the mills in order to dry and transport the coal into the combustion chamber. Major part of the primary air (PA) is heated up by the regenerative air preheater to app. 300 ^{o}C. The pulverized coal leaves the mill owing to primary air through coal dust pipes to the burners. To be able to equalize fuel distribution for each burners, there is manually actuated orifice on the pipelines. The remaining combustion air supplied by FD fan also passes through air preheater section via waste heat carrier flue gas before entering furnace and combustion chamber thus yielding cooling down of the flue gas significantly. Nevertheless, As shown in Figure 3.7, the flue gas which leaving boiler combustion chamber pass over convection banks of the super-heaters, re-heaters and economizer and then to air preheater, has a residual potential of waste heat. After flue gas exchanging its thermal energy with primary and secondary air, the temperature remains around 140-150 ºC on itself. The cooled flue gas is passes in turn electrostatic precipitator and then induced draught fan (IDF). Via ID fan the cleaned gases is sent to Flue gas desulphurization (FGD) plant and last to the atmosphere via the stack.

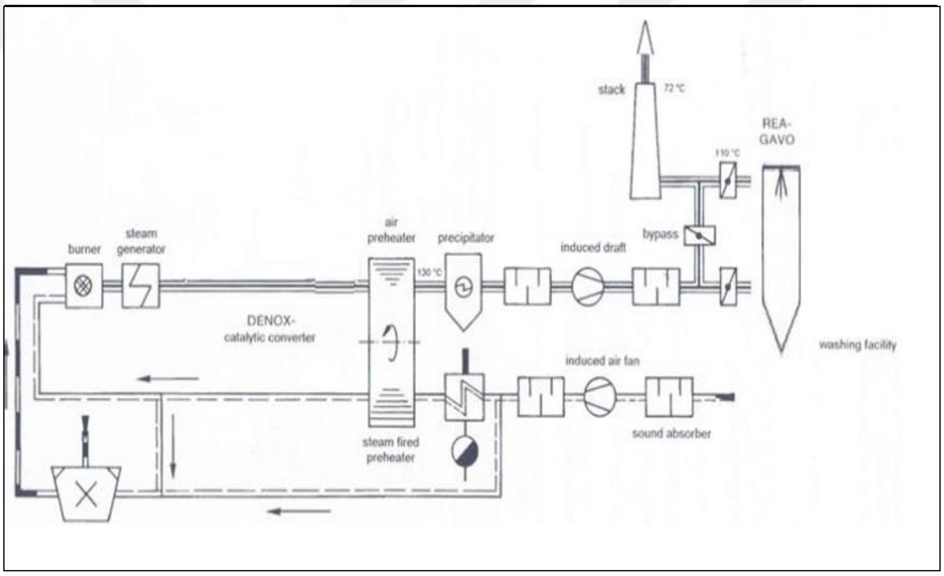

Figure 3.7. A path of flue gas

The waste heat potential of the flue gas based on chemical reaction of boiler dynamics that is actually combustion of main fuel. Secondly on the way to the atmosphere of its travel there is inevitably entropy generation which actually takes place of ESP and ID fan.

On the other hand, economizer and regenerative air preheater is already exist in order to recover waste heat loss on the way to flue gas path. The air flue gas components mainly as follows:

- ✓ Forced Draught Fan (Combustion and transport air supplier)
- ✓ Regenerative Air Preheater (Thermal Energy recovery between flue gas and fresh air)
- ✓ Induced Draught Fan (Evacuating the furnace and transfer flue gas to stack)
- ✓ Steam Air Heater (Combustion air preheating to ensure air-preheater heat transfer plate against acidic corrosion)

Table 3.2 expresses design parameters of fans which is a part of air flue gas system.

Table 3.2. Fan design data

		IDF		FDF		PAF	
						a)	b)
Pressure fan inlet	mbar	-29,2		-2,9		27,9	22,3
Pressure fan outlet	mbar	22,4		40,1		168,5	142,6
Pressure increase	mbar	51,6		42,9		140,6	120,2
Mass flow	kg/s	722,7		654,9		132,0	149,5
Volume flow	m^3/s	868,9		587,9		115,0	130,9
- Temperature	°C	150		40		40	40
- Specific gravity	kg/m^3	0,832		1,114		1,148	1,142
Basis							
Boiler load		100 %		100 %		100 %	100 %
No. of operating mills		3 Mills		3 Mills		3 Mills	4 Mills
Fuel		Design Coal		Design Coal		Check Coal	Check Coal
Air ratio		$\lambda = 1,15$		$\lambda = 1,15$		$\lambda = 1,15$	$\lambda = 1,15$

Boiler is such a massive heat exchanger that approximately 1500 MW thermal power has been transferred by the means of heat transfer methods as radiation, convection and conduction onto the steam circulating inside evaporator walls and other bundles. To be able to satisfy the thermal power, it has to be reached around 1500 ^{C}O combustion chamber.

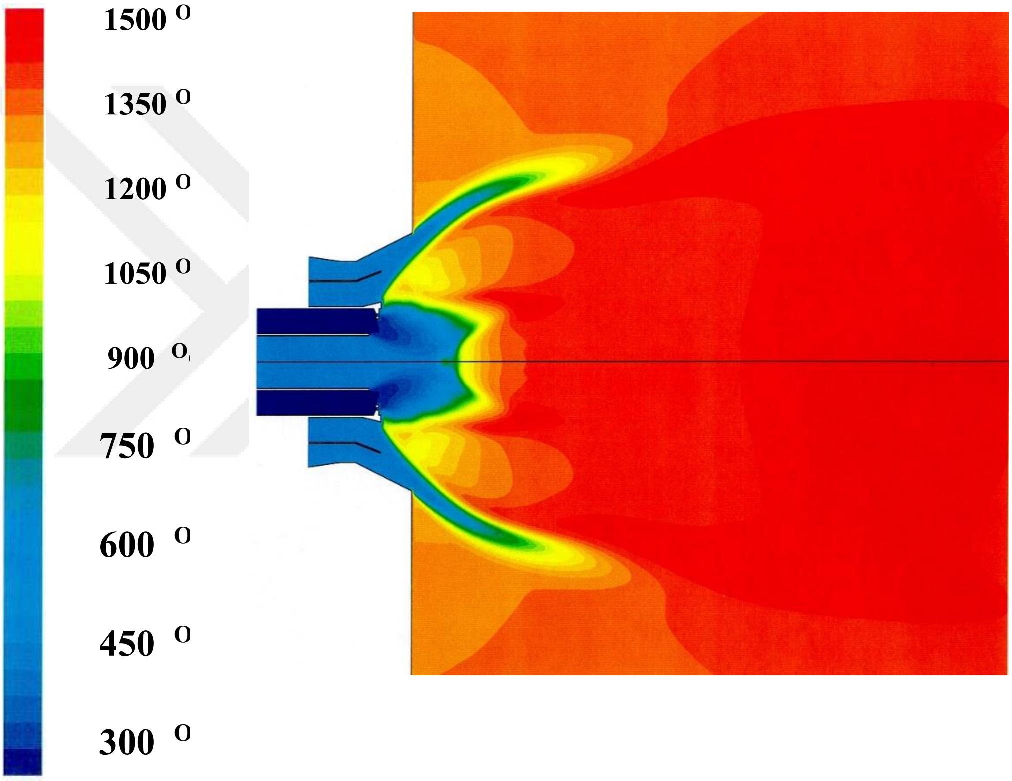

Figure 3.8. DS burner& combustion chamber temperature

The higher effect due to the radiation is around evaporator and super-heater 1 heating surfaces. The rest of bundles are being mostly under affected by convective heat transfer between flue-gas. The graph below shows flue gas temperature profile change during heat transfer zones and the amount of heat transfer on each heating surfaces passes through shell side of boiler and heat recovery zones as well (Figure 3.9).

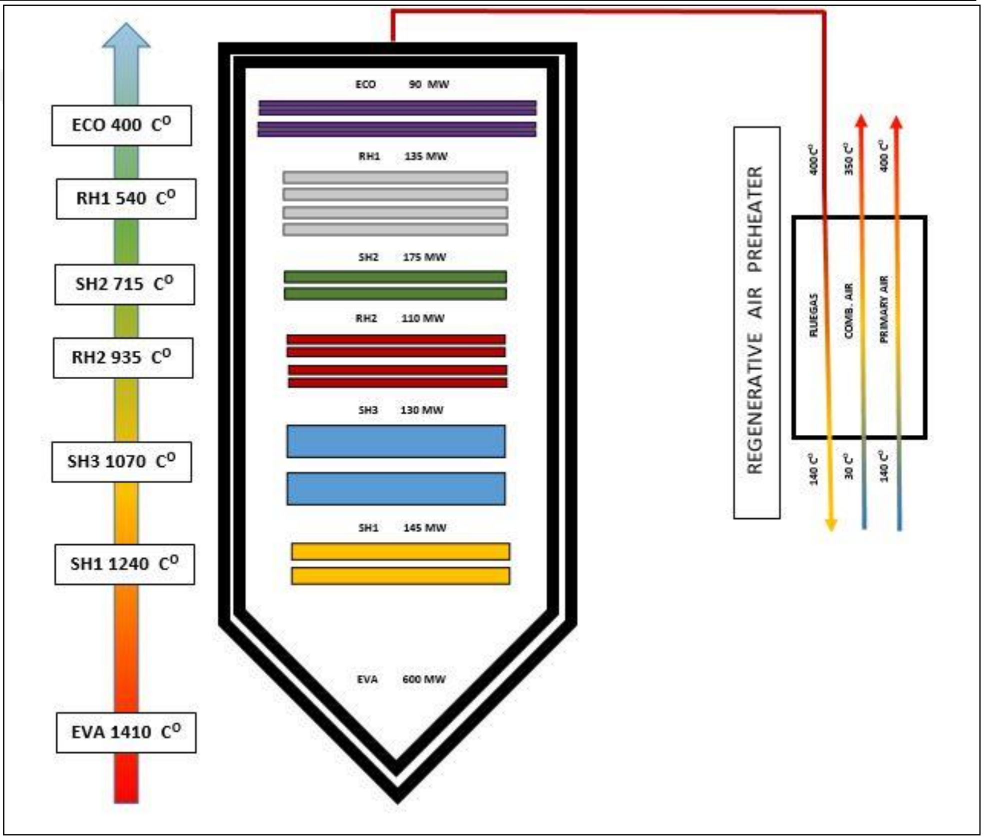

Figure 3.9. Flue gas temperature profile

3.6. Outline of the Study

In this study, it is going to be dealt with one of the most applicable recovery method in use for thermal power plant named as waste heat recovery based on flue gas stack losses. The purpose of the study is basically evaluating system efficiency improvement options by using thermodynamic laws and analysis as well as optimizing by economical facts. It is mainly build up ORC on flue gas path, hence reducing stack losses and contribution to efficiency increment. The organic fluid selection will be according to boiler dynamic parameters. Due to the steam power plant is operated by coal as main fuel, it should be considered the dew point treat on emissive gases during numeric calculations.

The schedule is as follows:

By Ebsilon program assistance, modelling organic rankine cycle demonstration thus it will be obtained optimum numeric data for thermodynamic calculations. The organic rankine cycle was performed under boiler full load conditions. The thermodynamic analyze performance will be repeated by different selected organic fluids. To sum up, it is obtained multiple results to compare each condition in order to optimize the project feasibility and to determine the applicable technique probably that converting one of the internal consumer from electrical driven pump to steam powered pump via flue gas heat recovery thanks to ORC. Here below is step by step to follow during next chapter.

The temperature of the flue gas upstream of the absorbers is around 140-150°C and the volumetric flow is ~760m^3/s. Therefore, there is considerable heat potential, if it can be utilized in an appropriate and efficient way. It is assumed that the temperature at the outlet of the absorber (inlet of the stack) doesn't change or change marginally because of water / steam saturation effect. Indeed if the flue gas inlet temperature of the absorber is lower, then less heat energy is provided into the FGD system and less amount of water will be vaporized. Figure 3.10 shows the exact point of flue gas waste heat recovery system on the flue gas path. In this case it has arranged between IDF and FGD unit.

This region is convenient using ORC application that includes ORC pump, evaporator, turbine and condenser. Design, modeling ,simulation and optimization was performed by Ebsilon Proffesional v12.05 Software programme.

EBSILON Professional is software programme that reliably calculates the potential performance behavior and efficiency of a plant under the widest possible range of operating conditions. Algorithms are based on physical equations, polynomials and characteristic curves. And there are no limits to the modeling options. EBSILON Professional works with conventional power plants, nuclear and solar power plants, desalination plants, fuel-cell applications, and user-specific processes.

Regarding the criteria and limitations of cycle many iterations were performed by Ebsilon Proffesional. Thus it makes the way easier to find optimum working point while system is in operation. It is possible to draw a real water steam cycle within a real parameters. Also due to embedded wide fluid options inside Ebsilon programme, it is possible to perform and check many organic fluids to each other. In this case, it's very important to use that kind of programme while trying to increase the efficiency of conventional systems and also detecting the weaknesses.

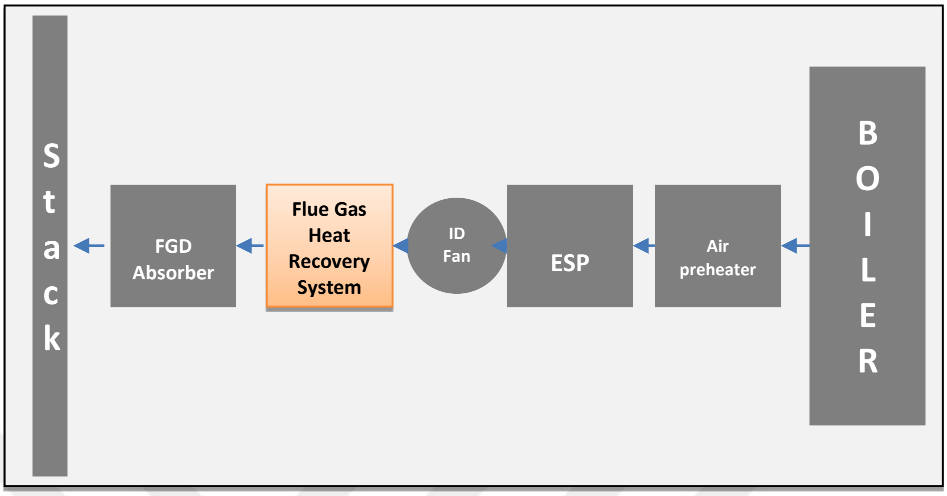

Figure 3.10. Flue gas waste heat recovery process

4. RESULTS AND DISCUSSION

In this chapter, water steam cycle and air flue gas parameters were dealth with together in order to take the best results during full load operations. As shown in Figure 4.1. First step to do, steam power plant is designed based on heat /flow diagram of the steam power plant. The obtained result is 95 % reasonable. The heat flow diagram belongs to steam power plant that is already in operation without waste heat recovery application.

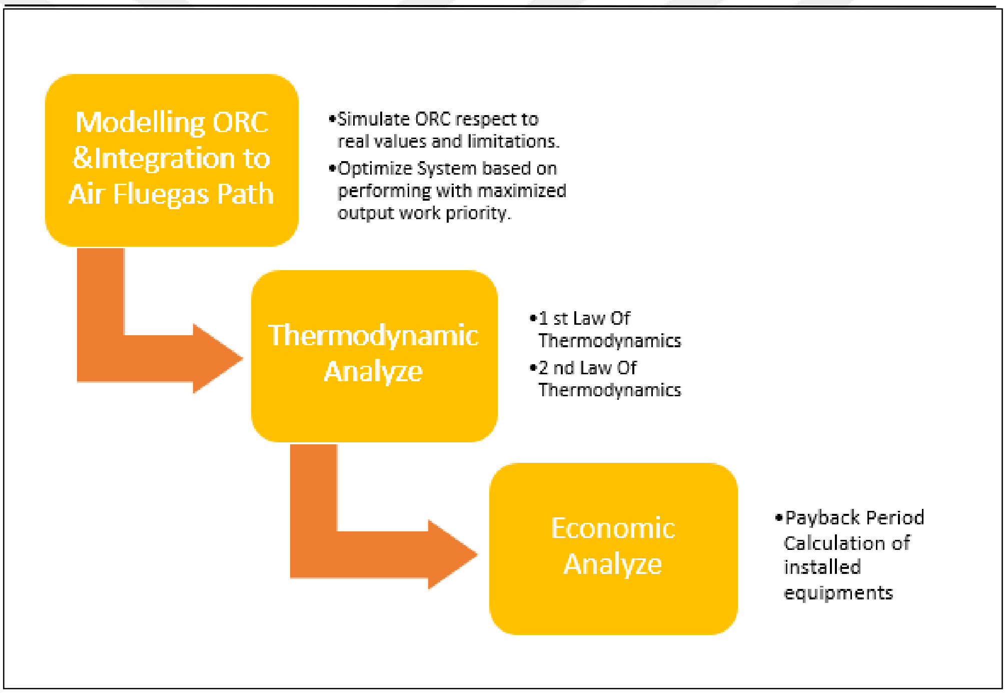

Figure 4.1. ORC Integration & Analyze Path

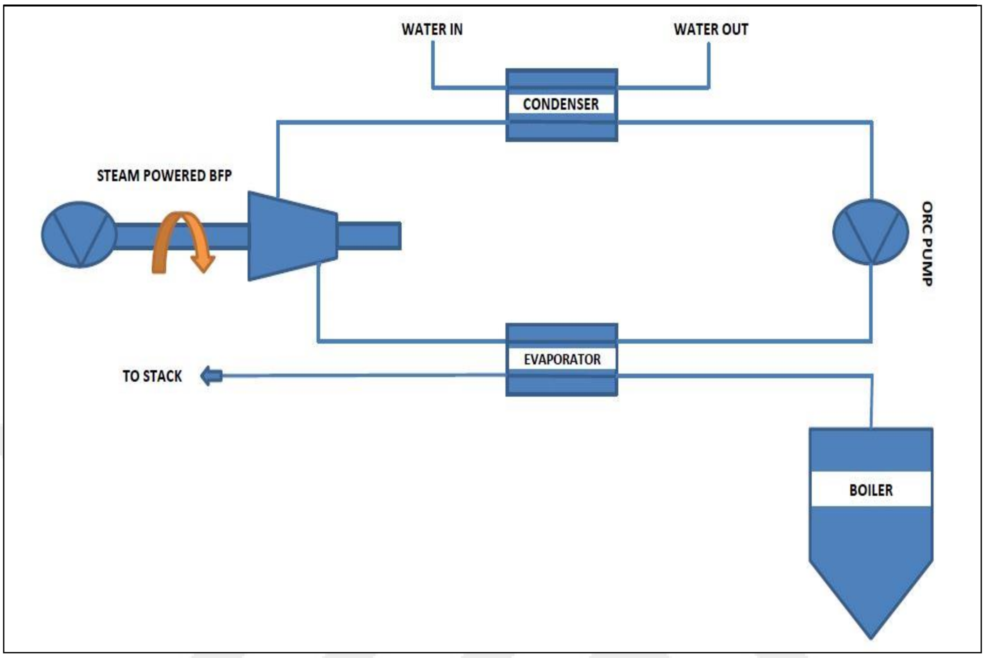

Figure 4.2. ORC & Subcomponent

Design and modeling of steam power plant on Ebsilon Professional program based on real heat and flow values so the results taken by Ebsilon Professional obeys all thermodynamic laws and engineering formulas as well. Both cycle in cooperation each other is plotted and designed according to mass, energy and exergy balance principle. Designed power plant model is then compiled and then the data given is run thanks to simulation capability of Ebsilon Programme assistance.

The study basically aims integration of ORC (Figure 4.3) onto the flue gas ducting thus yielding more energy saving and reducing waste heat additionally economizer and regenerative air preheater sections. Pay attention, the study prior based on thermodynamic and economic facts on the other hand environmental facts to be considered as enough importance as it could possible. Thus, the flue gas temperature after utilizing itself does not go down below critical and harmful effect level.

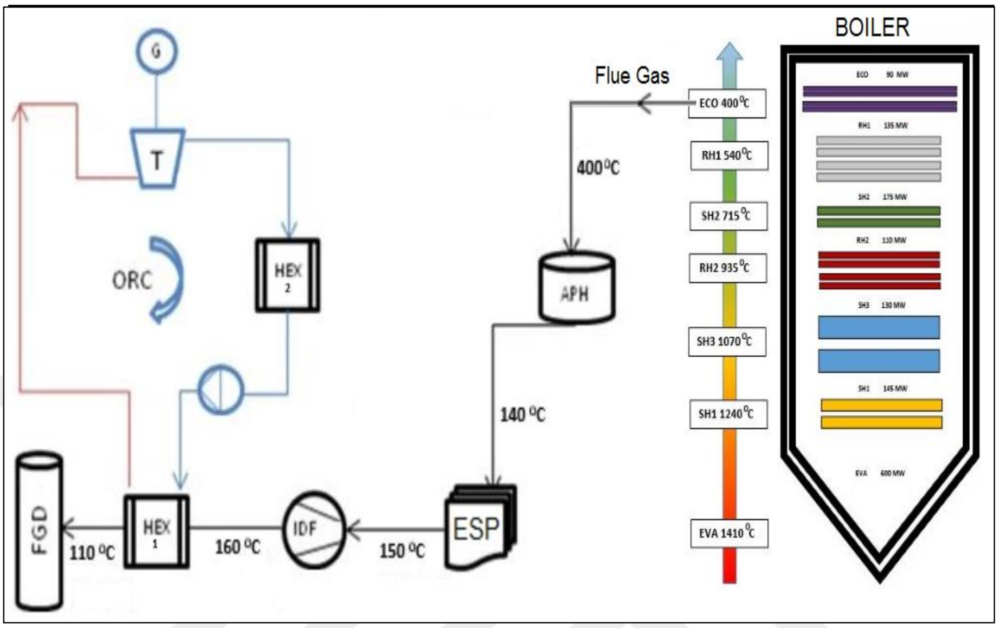

Figure 4.3. Steam power plant with ORC (Steam Rankine Cycle&ORC)

Table 4.1 shows physical fluid properties that are needed to calculate ORC subcomponent energy and exergy analysis. Some of the measured values are directly taken as steam power plant online data. The rest part is coming up by Ebsilon Proffesional calculations.

Table 4.1. Physical properties of the fluids

No	Zone	Stream	Mass flow rate(kg/s)	Temperature (ºC)	Pressure (Bar)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)
0	Dead State	Air		25	1	25,425	6,949
1	FD fan inlet	Air	586.266	25	1	25.425	6.949
2	FD fan outlet	Air	586.266	27.96	1.03	28.439	6.951
3	ID fan inlet	Fluegas	636.373	158.079	0.978	164.944	7.198
4	ID fan outlet	Fluegas	636.373	161.055	1	168.112	7.2
4	Evap. inlet	Fluegas	636.373	161.055	1	168.112	7.2
5	Pump inlet	R236FA	193.358	30.926	3.3	237.794	1.130
6	Pump outlet	R236FA	193.358	32.060	21.420	239.485	1.131
7	Evap. Outlet	Fluegas	636.373	110	0.999	114.201	7.068
8	Turbine Inlet	R236FA	193.358	104.978	21.420	417.540	1.630
9	Turbine Outlet	R236FA	193.358	46.11	3.3	393.387	1.641
10	Seawater inlet	Seawater	1927.416	22	1.5	87.829	0.309
11	Seawater out	Seawater	1927.416	25.926	1	103.532	0.362
5	Pump inlet	R245FA	148.312	32.178	1.920	242.016	1.144
6	Pump outlet	R245FA	148.312	32.790	14	243.160	1.145
8	Turbine inlet	R245FA	148.312	104.638	14	476.887	1.793
9	Turbine outlet	R245FA	148.312	49.218	1.920	444.655	1.807
10	Seawater inlet	Seawater	1455.294	22	1.5	87.829	0.309
11	Seawater out	Seawater	1455.294	27.178	1	108.543	0.378

- There is steady-state operation condition for all components of the system
- Negligible potential and kinetic energy differences
- Atmospheric air can be taken as ideal gas
- Seawater can be taken as constant temperature
- Heat transfer and pressure drops in the connecting tubing can be neglected
- Thermodynamic properties of the refrigerants are taken using the Ebsilon Professional (V12.05 (EPV-12.05) Software Program (EPSP).

4.1. Thermodynamic Analyze

4.1.1. FD Fan (Force Draught Fan)

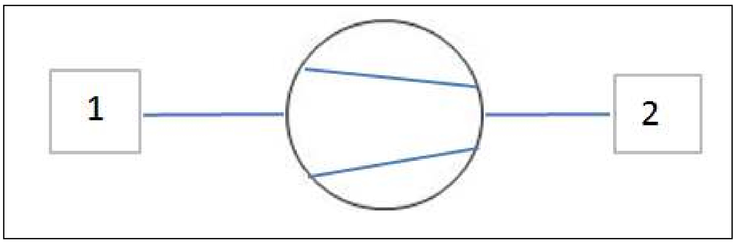

Figure 4.4. FD fan

FD fan inlet and outlet mass flow are same. Using equation (3.12) mass balance is calculated, and using equation (3.11) energy balance is calculated;

Mass Balance: $\dot{m}_1 = \dot{m}_2 = 586.266\ kg/s$

Energy Balance: $E_1 = E_2$

Assuming no heat transfer and kinetic, potential energy change is zero.

$$\dot{W}_{act} = \dot{m}\,(h_2 - h_1)$$

Fan actual work;

$$\dot{W}_{act} = -1767.005\ kW$$

Entropy generation of fan is calculated using Eq. (3.13)

$$\dot{S}_{gen} = \dot{m}\,(s_2 - s_1)$$

$$\dot{S}_{gen} = 586.266 * (6{,}951 - 6{,}949) = 1.172\ kW/K$$

Irreversibility of fan calculated as;

$$I = S_{gen} * T_O$$

$$I = 1.172 * 298 = 349.41\ kW$$

$$\dot{W}_{rev} = \dot{W}_{act} - I = 1767.005 + 349.41 = 1417.595\ kW$$

Second law efficiency of FD fan is;

$$\eta_{\Pi} = \frac{\dot{W}_{rev}}{\dot{W}_{act}} = \%\ 80.22$$

4.1.2. ID Fan (Induced Draught Fan)

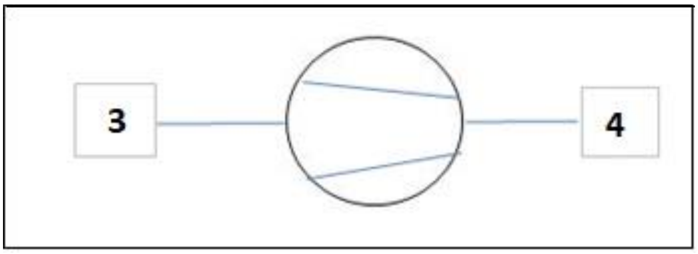

Figure 4. 5. ID fan

ID fan inlet and outlet mass flow rate are the same. Using equation (3.12) mass balance is calculated, and using equation (3.11) energy balance is calculated;

Mass Balance: $\dot{m}_3 = \dot{m}_4 = 636.373\ kg/s$

Energy Balance: $E_3 = E_4$

Assuming no heat transfer and kinetic, potential energy change is zero.

$$\dot{W}_{act,in} = \dot{m} * (h_4 - h_3)$$
$$\dot{W}_{act,in} = -2016.029\ kW$$

Entropy generation of fan is to calculate using Eq. (3.13)

$$\dot{S}_{gen} = \dot{m}(s_4 - s_3)$$
$$\dot{S}_{gen} = 636.373 * (7.200 - 7.198) = 1.272\ kW/K$$

Irreversibility of fan is calculated as;

$$I = S_{gen} * T_O$$
$$I = 1.272 * (25 + 273) = 379.056 kW$$

$$\dot{W}_{rev,in} = \dot{W}_{act,in} - I = 2016.029 + 548.232 = 1636.973\ kW$$

Second law efficiency of ID fan is;

$$\eta_{\Pi} = \frac{\dot{W}_{rev,in}}{\dot{W}_{act,in}} = \%\ 81.19$$

Figure 4.6 expresses the basic ORC and its subcomponents. Vapor at the turbine exhaust is condensed by seawater.

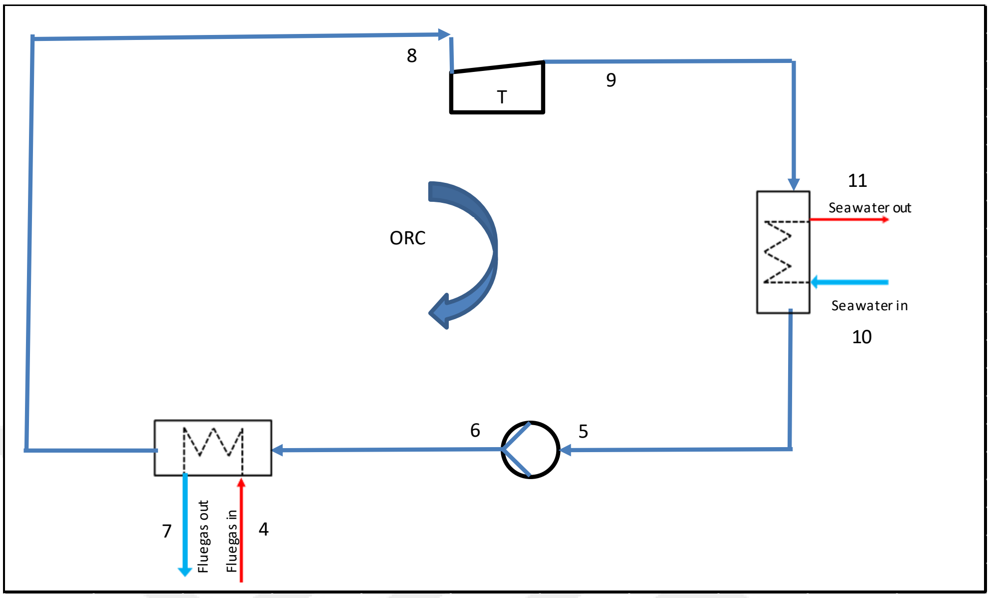

Figure 4.6. Basic ORC and its subcomponents

4.1.3. ORC Pump (R236FA)

To calculate ORC following components and T_O is taken as 25 ^{O}C;

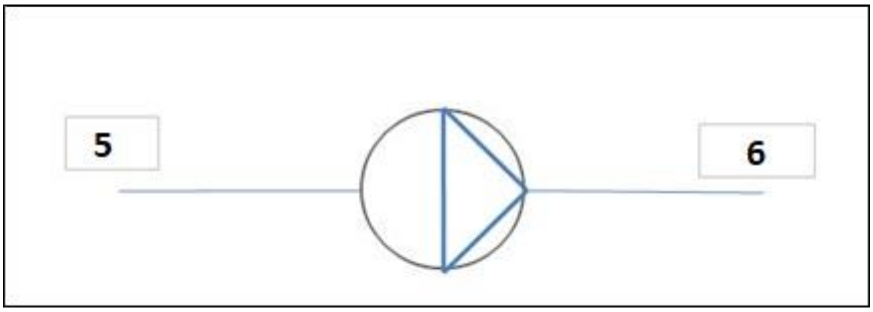

Figure 4.7. ORC pump for R236FA

ORC pump inlet and outlet mass flow rate keeps constant. Using Equation 3.12 mass balance is calculated, and using Equation 3.11 energy balance is calculated;

Mass Balance: $\dot{m}_5 = \dot{m}_6 = 193.358 kg/s$

Energy Balance: $E_5 = E_6$

Assuming no heat transfer and kinetic, potential energy change is zero.

$$\dot{W}_{act,in} = \dot{m} * (h_6 - h_5)$$

$$\dot{W}_{act,in} = -326.968\ kW$$

Entropy generation of ORC pump is to calculate using Eq. (3.13)

$$\dot{S}_{gen} = \dot{m}\ (s_6 - s_5)$$

$$\dot{S}_{gen} = 193.358 * (1.131 - 1.130) = 0.193\ kW/K$$

Irreversibility of ORC pump is calculated as;

$$I = \dot{S}_{gen} * T_O$$

$$I = 0.193 * (25 + 273) = 57.514 kW$$

$$\dot{W}_{rev,in} = \dot{W}_{act,in} - I = 326.968 - 57.514 = 269.454 kW$$

Exergy balance is calculated as shown;

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{des} = \Delta\dot{X}_{sys}$$

$$\dot{W}_{rev,in} + \dot{m} * \dot{\psi}_8 = \dot{m} * \dot{\psi}_9$$

$$\dot{W}_{rev,in} = \dot{m} * [(h_6 - h_5) - T_0(s_6 - s_5)]$$

$$\dot{W}_{rev,in} = 269.454\ kW$$

$$\dot{X_{des}} = \dot{W}_{act,in} - \dot{W}_{rev,in} = 57.514\ kW$$

As seen above $\dot{X}_{des} = I = 57.514\ kW$

Second law efficiency of ORC pump is;

$$\eta_{\Pi} = \frac{\dot{W}_{rev,in}}{\dot{W}_{act,in}} = \%\ 82.40$$

4.1.4. ORC Evaporator (R236FA)

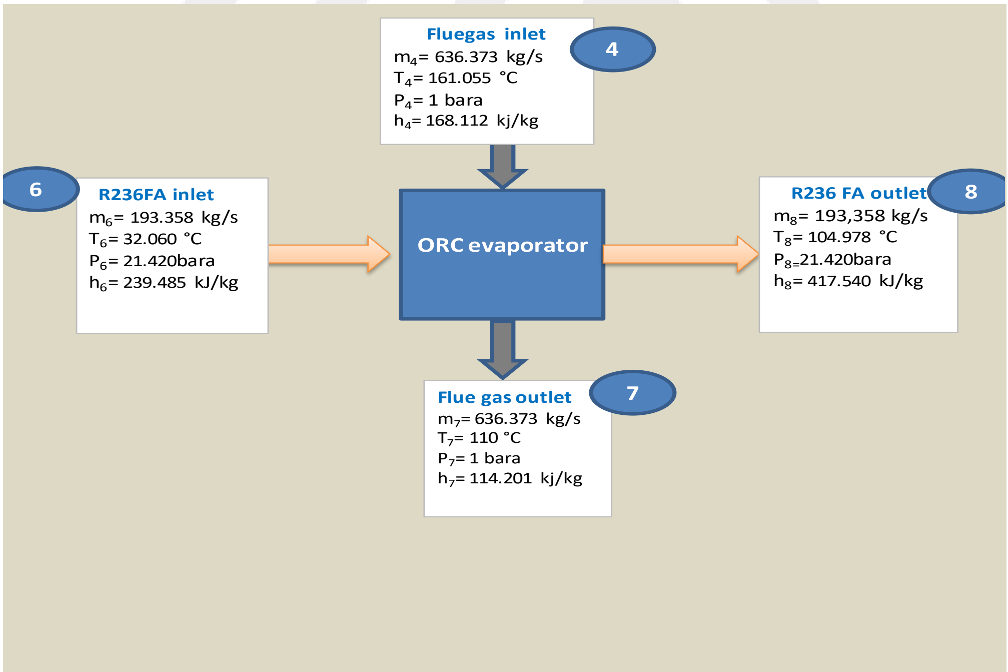

Figure 4.8. ORC evaporator for R236FA

Using Equation 3.12 mass balance is calculated, and using Equation 3.11 energy balance is calculated;

Inlet and outlet mass flow rate for each stream is same.

Mass Balance: $\dot{m}_4 = \dot{m}_7 = 636.373\ kg/s$ and

$\dot{m}_6 = \dot{m}_8 = 193.358 kg/s$

Energy Balance: $E_{in} - E_{out} = Q_{loss}$

$$Q_{loss} = \sum \dot{m}_{in} * h_{in} - \sum \dot{m}_{out} * h_{out}$$

$$Q_{in} = \dot{m}_4 * h_4 + \dot{m}_6 * h_6$$

$$Q_{out} = \dot{m}_7 * h_7 + \dot{m}_8 * h_8$$

$$Qloss = Q_{in} - Q_{out}$$

$$Qloss = 123\ kW$$

Exergy balance is calculated by equation below;

$$\dot{X}_{des} = \dot{X}_{in} - \dot{X}_{out}$$

Exergy inlet is calculated by Eq. (3.17);

$$\psi_4 = (h_4 - h_0) - T_o(s_4 - s_0)$$

$$\psi_4 = 16.851\ kj/kg$$

$$\psi_6 = (h_6 - h_0) - T_o(s_6 - s_0)$$

$$\psi_6 = 67.889 kj/kg$$

Exergy outlet is calculated by Eq. (3.17);

$$\psi_7 = (h_7 - h_0) - T_o(s_7 - s_0)$$
$$\psi_7 = 53.251\ kj/kg$$
$$\psi_8 = (h_8 - h_0) - T_o(s_8 - s_0)$$
$$\psi_8 = 46.216\ kj/kg$$

Exergy destruction is calculated;

$$\dot{X}_{des} = (\dot{m}_4 * \psi_4 + \dot{m}_6 * \psi_6) - (\dot{m}_7 * \psi_7 + \dot{m}_8 * \psi_8)$$
$$\dot{X}_{des} = 3597\ kW$$

Second law efficiency is calculated by Eq. (3.19);

$$\eta_{eva} = \frac{\dot{m}_6 * \psi_6 + \dot{m}_7 * \psi_7}{\dot{m}_4 * \psi_4 + \dot{m}_8 * \psi_8}$$
$$\eta_{eva} = \%71.43$$

4.1.5. ORC Turbine (R236FA)

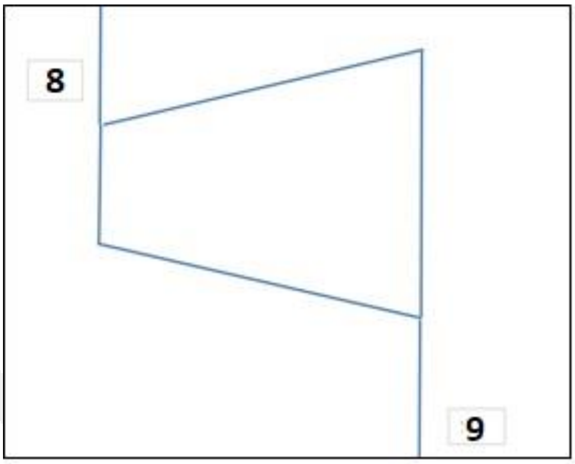

Figure 4. 9. ORC Turbine for R236FA

ORC turbine inlet and outlet mass flow rate are the same. Using equation (3.12) mass balance is calculated, and using equation (3.11) energy balance is calculated;

Mass Balance: $\dot{m}_8 = \dot{m}_9 = 193.358 kg/s$

Energy Balance: $E_8 = E_9$

Assuming no heat transfer and kinetic, potential energy change is zero.

$$\dot{W}_{act,out} = \dot{m} * (h_9 - h_8)$$

$$\dot{W}_{act,out} = 4670.175\ kW$$

Contrary to pump and fan, work is generated by ORC turbine because of that reason positive sign is used here.

Entropy generation of ORC pump is to calculate using Eq. (3.13)

$$\dot{S}_{gen} = \dot{m}\,(s_9 - s_8\,)$$

$$\dot{S}_{gen} = 193.358 * (1.641 - 1.630) = 2.126\;kW/K$$

Irreversibility of ORC turbine is calculated as;

$$I = \;\dot{S}_{gen} * T_O$$

$$I = 2.126 * (25 + 273) = 633.827kW$$

$$\dot{W}_{rev,out} = \dot{W}_{act,out} + \;I = 4672.58 + 633.827 = 5304.002\;kW$$

Exergy balance is calculated as shown;

$$\dot{X_{in}} - \dot{X_{out}} - \;\dot{X_{des}} = \;\Delta\dot{X_{sys}}$$

$$\dot{m} * \dot{\psi}_8 = \;\dot{m} * \psi_9 \dot{+} \dot{W}_{rev,out}$$

$$\dot{W}_{rev,out} = m * [(h_8 - h_9) - T_0(s_8 - s_9)]$$

$$\dot{W}_{rev,out} = 5304.002\;kW$$

$$\dot{X}_{des} = \;\dot{W}_{rev,out} - \dot{W}_{act,out} = 633.827\;kW$$

$$\dot{X}_{des} = I = 633.827\;kW$$

Second law efficiency of ORC turbine is;

$$\eta_{\Pi} = \frac{\dot{W}_{act,out}}{\dot{W}_{rev,out}} = \%\;88.05$$

4.1.6. ORC Condenser (R236FA)

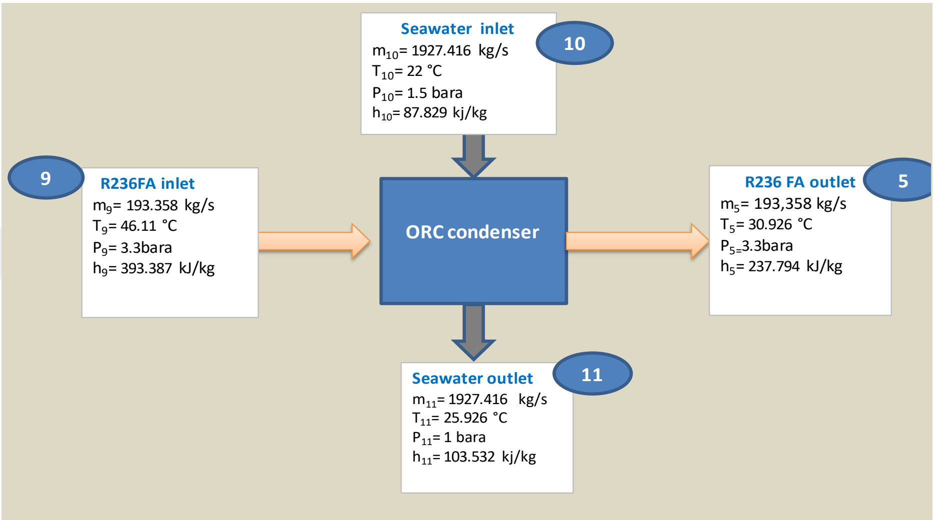

Figure 4.10. ORC condenser for R236FA

Using equation (3.12) mass balance is calculated, and using equation (3.11) energy balance is calculated;

Inlet and outlet mass flow rate for each stream is same.

Mass Balance: $\dot{m}_{10} = \dot{m}_{11} = 1927.416 kg/s$ and

$\dot{m}_9 = \dot{m}_5 = 193.358 kg/s$

Energy Balance: $E_{in} - E_{out} = Q_{loss}$

$$Q_{loss} = \sum \dot{m}_{in} * h_{in} - \sum \dot{m}_{out} * h_{out}$$

$$Q_{in} = \dot{m}_{10} * h_{10} + \dot{m}_9 * h_9$$

$Q_{out} = \dot{m}_{11} * h_{11} + \dot{m}_5 * h_5$

$Q_{loss} = Q_{in} - Q_{out}$

$Q_{loss} = 29939.244\ kW$

Exergy balance is calculated by equation below;

$\dot{X}_{des} = \dot{X}_{in} - \dot{X}_{out}$

Exergy inlet is calculated by Eq. (3.17);

$\psi_9 = (h_9 * h_0) - T_o(s_9 - s_0)$

$\psi_9 = 18{,}792\ kj/kg$

$\psi_{10} = (h_{10} * h_0) - T_o(s_{10} - s_0)$

$\psi_{10} = 0{,}078 kj/kg$

Exergy outlet is calculated by Eq. (3.17);

$\psi_{11} = (h_{11} * h_0) - T_o(s_{11} - s_0)$

$\psi_{11} = 15{,}47\ kj/kg$

$\psi_5 = (h_5 * h_0) - T_o(s_5 - s_0)$

$\psi_5 = 0{,}239\ kj/kg$

Exergy destruction is calculated;

$\dot{X}_{des} = (\dot{m}_9 * \psi_9 + \dot{m}_{10} * \psi_{10}) - (\dot{m}_{11} * \psi_{11} + \dot{m}_5 * \psi_5)$

$\dot{X}_{des} = 822.581\ kW$

$$\eta_{con} = \frac{\dot{m}_{10} * \psi_{10} + \dot{m}_5 * \psi_5}{\dot{m}_9 * \psi_9 + \dot{m}_{11} * \psi_{11}}$$

$\eta_{con=}$ 69.41 %

Using Eq. (3.11)

ORC efficiency for R236FA is calculated as:

$$\eta_{cyc} = \frac{W_{net}}{Q_{in}}$$

$$\eta_{cycforR236FA} = \frac{4076.269}{34370.248} = \%11.85$$

Also Figure 4.11 states T-s diagram of ORC in which R245FA is used. According to the figure; fluid enters pump in 1.92 Bar and 32 ^{O}C as saturated liquid. Then pumped to evaporator by 14 Bar. After heat is transferred by flue gas, refrigerant first changed phase into gas and then slightly superheated. R245FA enters turbine and still remains in vapor phase at turbine exhaust. To be able to complete the cycle it condenses by seawater in the condenser. Critic temperature of R245FA is 154 ^{O}C. It seems to be there is no blade erosion risk at the last stages of ORC turbine.

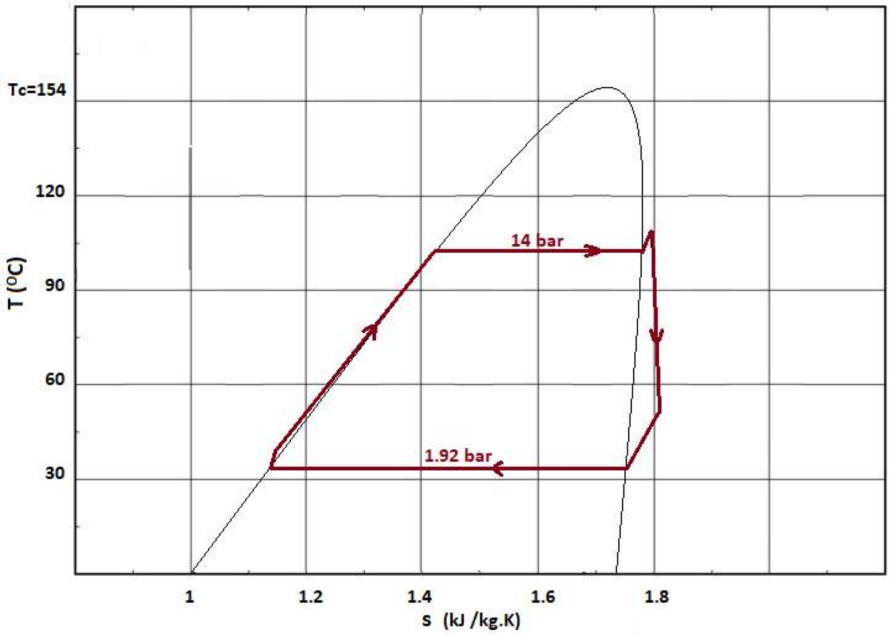

Figure 4.11. ORC T-s diagram for R245FA

4.1.7. ORC Pump (R245FA)

Refrigerant 245FA is selected as a reference refrigerant to calculate ORC following components and T_O is taken as 25 ^OC;

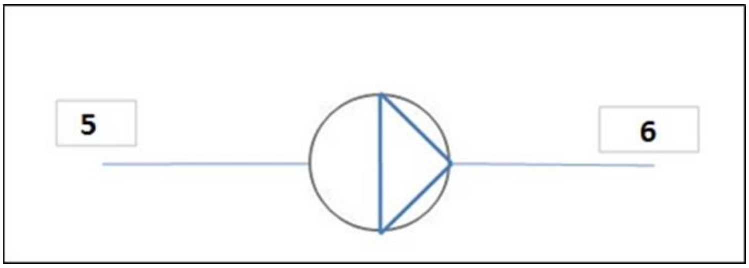

Figure 4.11. ORC pump for R245FA

ORC pump inlet and outlet mass flow rate are the same. Using equation (3.12) mass balance is calculated, and using equation (3.11) energy balance is calculated;

Mass Balance: $\dot{m}_5 = \dot{m}_6 = 148.312 kg/s$

Energy Balance: $E_5 = E_6$

Assuming no heat transfer and kinetic, potential energy change is zero.

$$\dot{W}_{act,in} = \dot{m} * (h_6 - h_5)$$

$$\dot{W}_{act,in} = -169.668\ kW$$

Entropy generation of ORC pump is to calculate using Eq. (3.13)

$$\dot{S}_{gen} = \dot{m}\ (s_6 - s_{15})$$

$$\dot{S}_{gen} = 148.312 * (1.145 - 1.144) = 0.148 kW/K$$

Irreversibility of ORC pump is calculated as;

$$I = \dot{S}_{gen} * T_O$$

$$I = 0.148 * (25 + 273) = 44.104 kW$$

$$\dot{W}_{rev,in} = \dot{W}_{act,in} - I = 169.668 - 44.104 = 125.564 kW$$

Exergy balance is calculated as shown;

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{des} = \Delta\dot{X}_{sys}$$

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

$\dot{W}_{rev,in} + \dot{m} * \dot{\psi}_{15} = \dot{m} * \dot{\psi}_{16}$

$\dot{W}_{rev,in} = \dot{m} * [(h_6 - h_5) - T_0(s_6 - s_5)]$

$\dot{W}_{rev,in} = 125.564\ kW$

$\dot{X}_{des} = \dot{W}_{act,in} - \dot{W}_{rev,in} = 44.104\ kW$

As seen above $\dot{X}_{des} = I = 44.104\ kW$

Second law efficiency of ORC pump is;

$$\eta_{\Pi} = \frac{\dot{W}_{rev,in}}{\dot{W}_{act,in}} = \%\ 74.05$$

4.1.8. ORC Evaporator (R245FA)

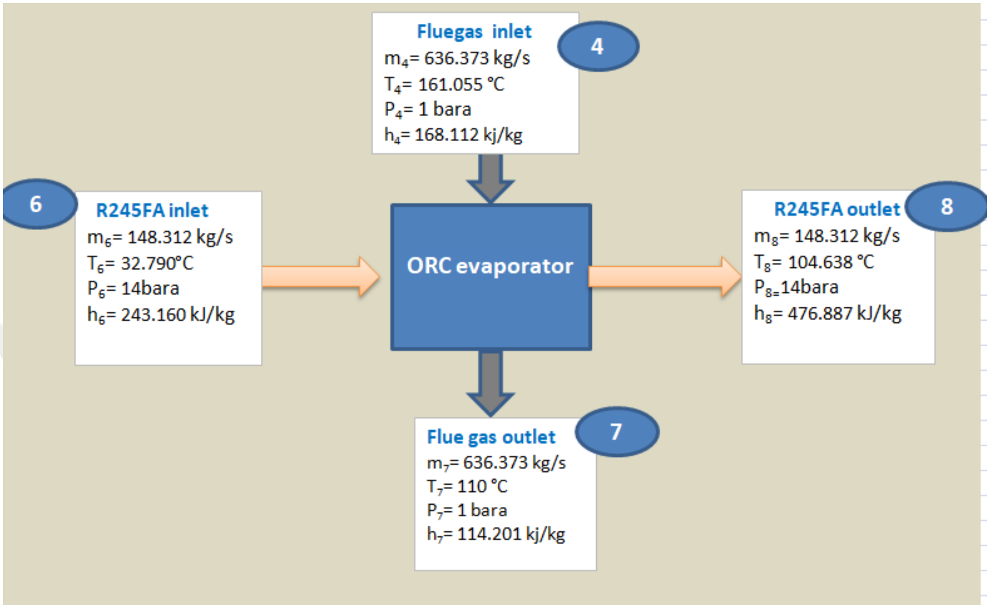

Figure 4.12. ORC evaporator for R245FA

Using Equation 3.12 mass balance is calculated, and using Equation 3.11 energy balance is calculated;

Inlet and outlet mass flow rate for each stream is same.

Mass Balance: $\dot{m}_4 = \dot{m}_7 = 636.373\ kg/s$ and

$\dot{m}_6 = \dot{m}_8 = 148.312 kg/s$

Energy Balance: $E_{in} - E_{out} = Q_{loss}$

$$Q_{loss} = \sum \dot{m}_{in} * h_{in} - \sum \dot{m}_{out} * h_{out}$$

$$Q_{in} = \dot{m}_4 * h_4 + \dot{m}_6 * h_6$$

$Q_{out} = \dot{m}_7 * h_7 + \dot{m}_8 * h_8$

$Q_{loss} = Q_{in} - Q_{out}$

$Q_{loss} = 86.308\ kW$

Exergy balance is calculated by equation below;

$\dot{X}_{des} = \dot{X}_{in} - \dot{X}_{out}$

Exergy inlet is calculated by Eq. (3.17);

$\psi_4 = (h_4 - h_0) - T_o(s_4 - s_0)$

$\psi_4 = 67.889\ kj/kg$

$\psi_6 = (h_6 - h_0) - T_o(s_6 - s_0)$

$\psi_6 = 16{,}858\ kj/kg$

Exergy outlet is calculated by Eq. (3.17);

$\psi_7 = (h_7 - h_0) - T_o(s_7 - s_0)$

$\psi_7 = 53.251\ kj/kg$

$\psi_8 = (h_8 - h_0) - T_o(s_8 - s_0)$

$\psi_8 = 150{,}733 kj/kg$

Exergy destruction is calculated;

$\dot{X}_{des} = (\dot{m}_4 * \psi_4 + \dot{m}_6 * \psi_6) - (\dot{m}_7 * \psi_7 + \dot{m}_8 * \psi_8)$

$\dot{X}_{des} = 3250.257\ kW$

Second law efficiency is calculated by Eq. (3.19);

$$\eta_{eva} = \frac{\dot{m}_6 * \psi_6 + \dot{m}_7 * \psi_7}{\dot{m}_4 * \psi_4 + \dot{m}_8 * \psi_8}$$

$$\eta_{eva} = \%53.77$$

4.1.9. ORC Turbine (R245FA)

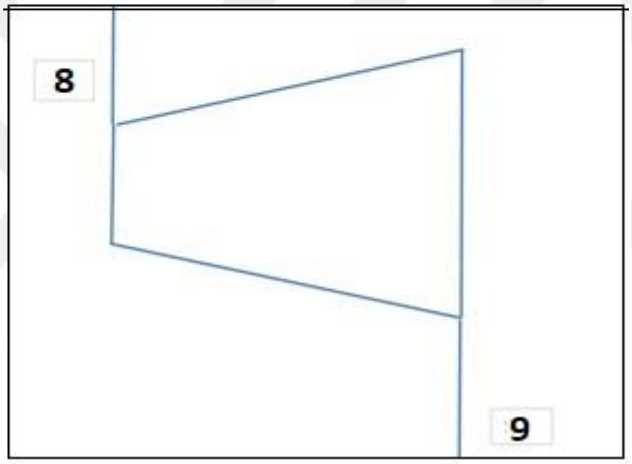

Figure 4.13. ORC Turbine for 245FA

To calculate ORC following components and T_O is taken as 25 ^{O}C;

Mass Balance: $\dot{m}_8 = \dot{m}_9 = 148.312 kg/s$

Energy Balance: $E_8 = E_9$

Assuming no heat transfer and kinetic, potential energy change is zero.

$\dot{W}_{act,out} = \dot{m} * (h_8 - h_9)$

$\dot{W}_{act,out} = 4780.392\ kW$

Entropy generation of ORC pump is to calculate using Eq. (3.13)

$\dot{S}_{gen} = \dot{m}\ (s_9 - s_8)$

$\dot{S}_{gen} = 148.312 * (1.807 - 1.793) = 2.076\ kW/K$

Irreversibility of ORC turbine is calculated as;

$I = \dot{S}_{gen} * T_O$

$I = 2.076 * (25 + 273) = 618.757 kW$

$\dot{W}_{rev,out} = \dot{W}_{act,out} + I = 4780.392 + 618.757 = 5399.149\ kW$

Exergy balance is calculated as shown;

$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{des} = \Delta\dot{X}_{sys}$

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

$\dot{m} * \dot{\psi}_8 = \dot{m} * \psi_9 \dot{+} \dot{W}_{rev,out}$

$\dot{W}_{rev,out} = \dot{m} * [(h_8 - h_9) - T_0(s_8 - s_9)]$

$\dot{W}_{rev,out} = 5399.149\ kW$

$\dot{X}_{des} = \dot{W}_{act,out} - \dot{W}_{rev,out} = 618.757\ kW$

$\dot{X}_{des} = I = 618.757\ kW$

Second law efficiency of ORC turbine is;

$$\eta_{\Pi} = \frac{\dot{W}_{act,out}}{\dot{W}_{rev,out}} = \% \ 88.53$$

4.1.10. ORC Condenser (R245FA)

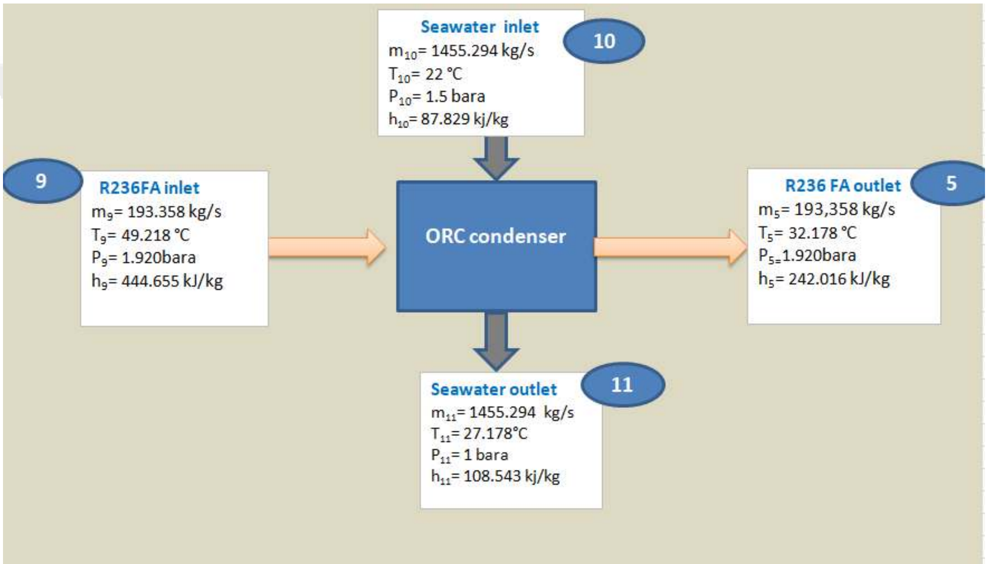

Figure 4.14. ORC condenser for 245FA

Using Equation 3.12 mass balance is calculated, and using Equation 3.11 energy balance is calculated;

Inlet and outlet mass flow rate for each stream is same.

Mass Balance: $\dot{m}_{10} = \dot{m}_{11} = 1455.294 \ kg/s$ and

$\dot{m}_9 = \dot{m}_5 = 148.312 \ kg/s$

Energy Balance: $E_{in} - E_{out} = Q_{loss}$

$$Q_{loss} = \sum \dot{m}_{in} * h_{in} - \sum \dot{m}_{out} * h_{out}$$

$$Q_{in} = \dot{m}_9 * h_9 + \dot{m}_{10} * h_{10}$$

$$Q_{out} = \dot{m}_5 * h_5 + \dot{m}_{11} * h_{11}$$

$$Q_{loss} = Q_{in} - Q_{out}$$

$$Q_{loss} = 193856.372 - 193855.852 = 0.52\ kW$$

Exergy balance is calculated by equation below;

$$\dot{X}_{des} = \dot{X}_{in} - \dot{X}_{out}$$

Exergy inlet is calculated by Eq. (3.17);

$$\psi_9 = (h_9 * h_0) - T_o(s_9 - s_0)$$

$$\psi_9 = 12.305 kj/kg$$

$$\psi_{10} = (h_{10} * h_0) - T_o(s_{10} - s_0)$$

$$\psi_{10} = 2.922 kj/kg$$

Exergy outlet is calculated by Eq. (3.17);

$$\psi_5 = (h_5 * h_0) - T_o(s_5 - s_0)$$

$$\psi_5 = 0.404\ kj/kg$$

$$\psi_{11} = (h_{11} * h_0) - T_o(s_{11} - s_0)$$

$$\psi_{11} = 2.472\ kj/kg$$

Exergy destruction is calculated;

$$\dot{X}_{des} = (\dot{m}_9 * \psi_9 + \dot{m}_{10} * \psi_{10}) - (\dot{m}_{11} * \psi_{11} + \dot{m}_5 * \psi_5)$$

$$\dot{X}_{des} = 621.680\ kW$$

$$\eta_{con} = \frac{\dot{m}_{10} * \psi_{10} + \dot{m}_5 * \psi_5}{\dot{m}_9 * \psi_9 + \dot{m}_{11} * \psi_{11}}$$

$$\eta_{con=}\ \%79.534$$

Using Eq. (3.11)

ORC efficiency for R245FA is calculated as:

$$\eta_{cyc} = \frac{W_{net}}{Q_{in}}$$

$$\eta_{cyc\ forR245FA} = \frac{4380.510}{34355.734} = \%12.75$$

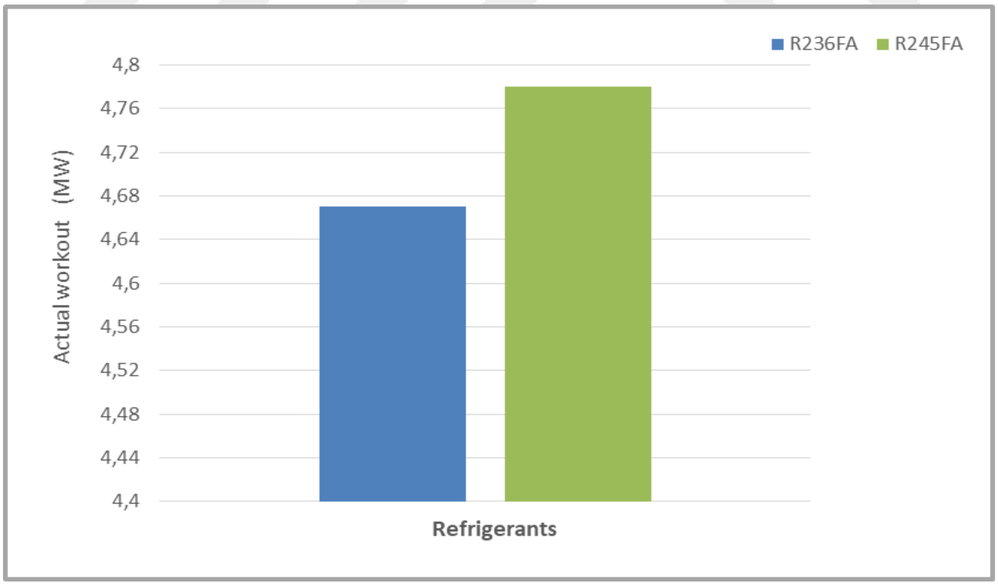

Figure 4.15. ORC Turbine Actual works for full load

Figures 4.16, 4.17, 4.18 and 4.19 show comparison of turbine actual and ideal work. Also comparing pump actual and ideal consumption parameters for two selected working fluids. The result taken after energy and exergy balance equation indicates working by R245FA is better than R236FA in terms of several thermodynamic aspects such as turbine actual works out and pump consumption. Thus, it's possible to get a higher recovery values by using R245 FA. To be able to maximize electrical output on recovery system it could be added recuperator onto the turbine exhaust line. Figure 4. 20 compares organic fluids in terms of ORC efficiencies.

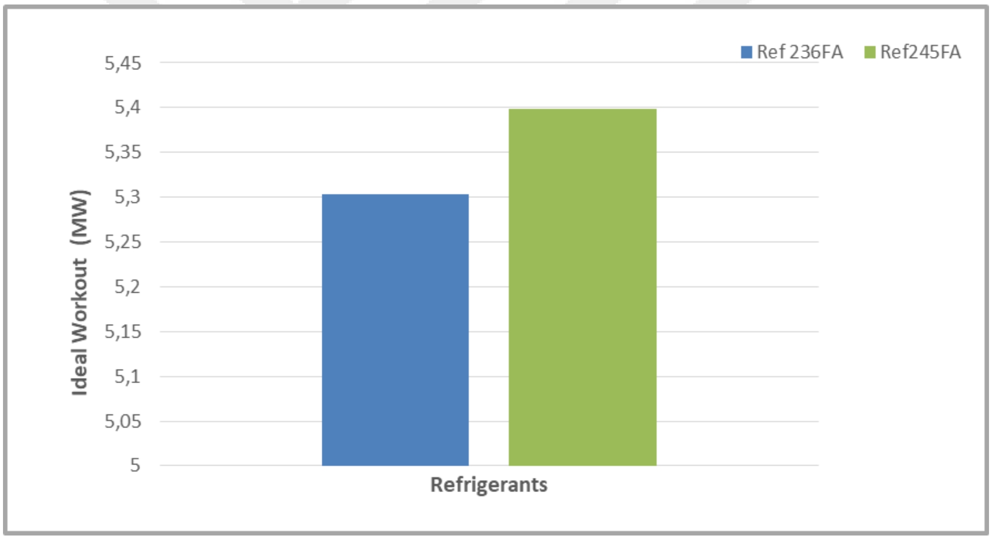

Figure 4.16. ORC Turbine Ideal works for full load

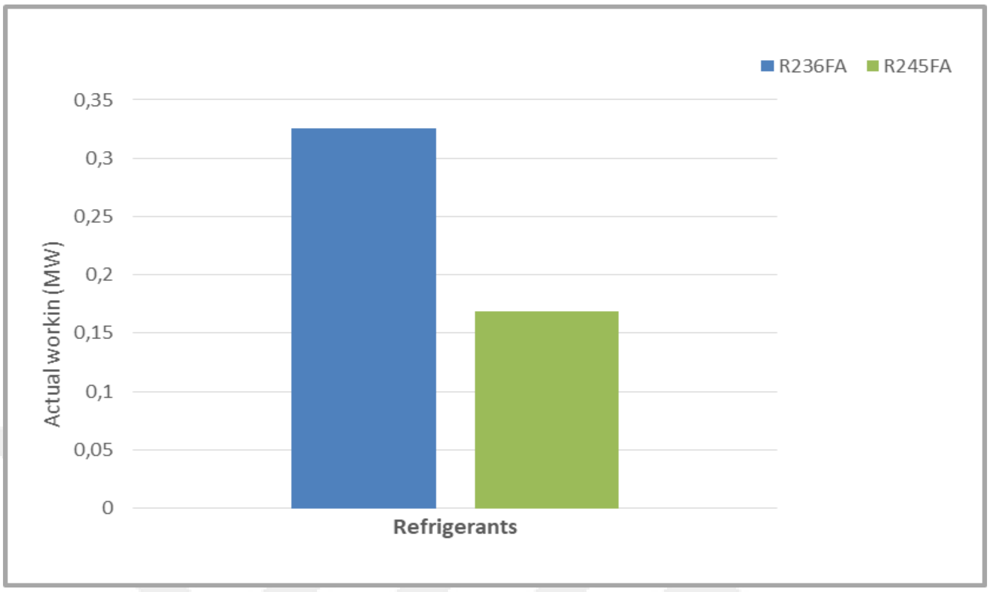

Figure 4.17. ORC Actual Pump Consumptions for full load

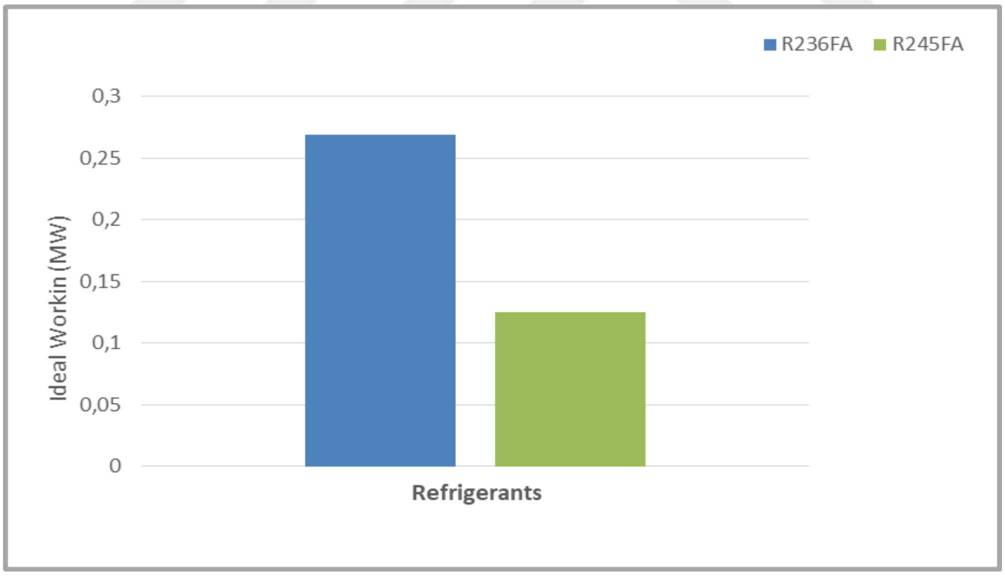

Figure 4.18. ORC Ideal Pump Consumptions for full load

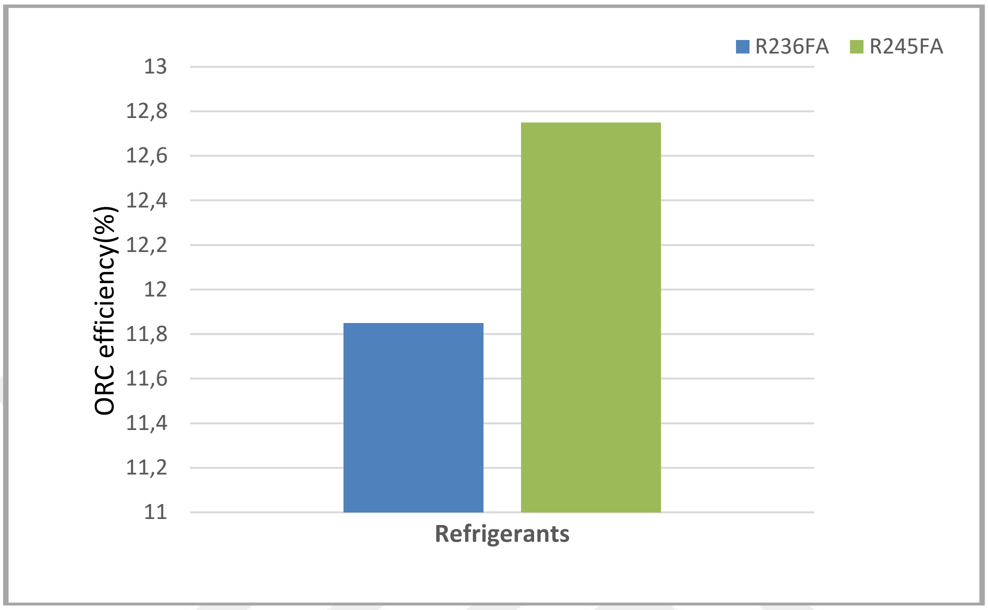

Figure 4.19. ORC efficiency for full load

Working fluid strongly depends on the conditions of boiler exhaust temperature and flue gas contents. Thus, it may work on different working fluids in spite of having less recovery values respect to protect system and its auxiliary equipment (flue gas ducting, absorber, stack) based on environmental limitations (acid rainfall). Regarding the other facts working fluid options could be duplicable like R365MFC, R123, R245CA shown in Figures 4.21 and 4.22.

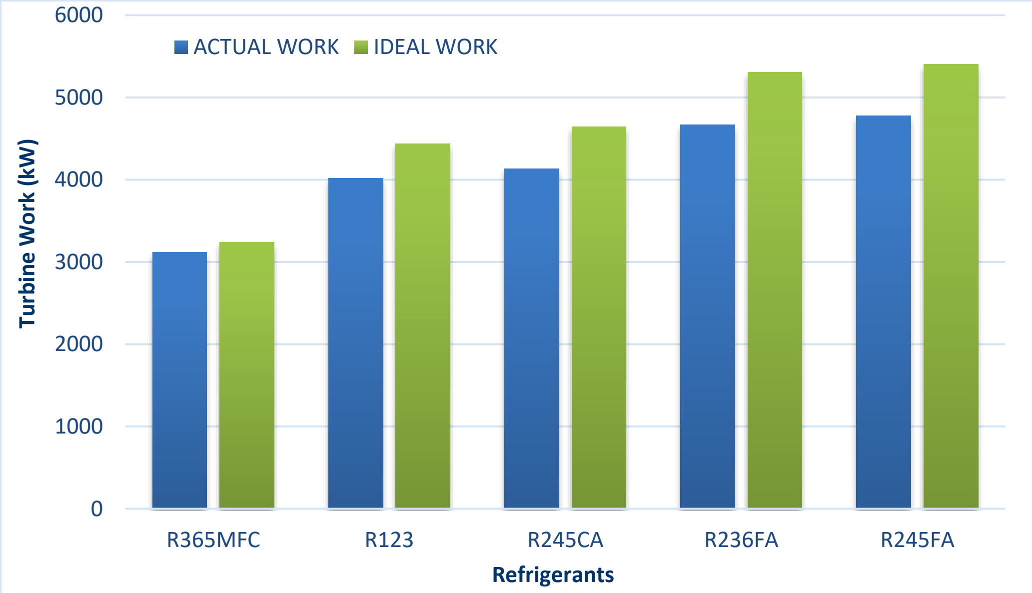

Figure 4.20. Comparison of 5 different working fluid types

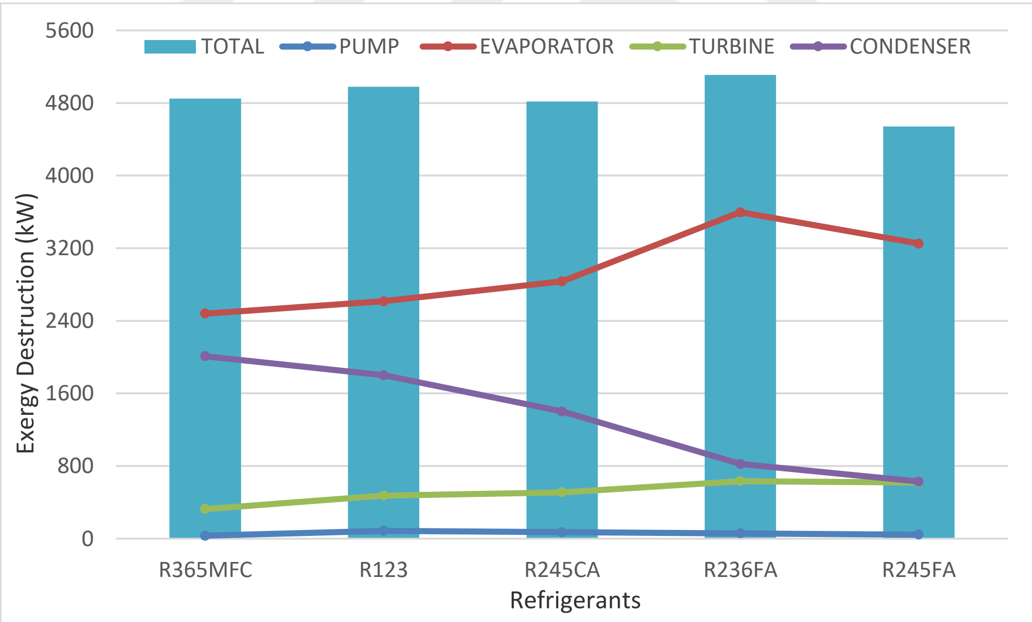

Figure 4.21. Variation of Exergy Destructions of ORC components for different refrigerants

4.2. Economic Analyze

In this project, it was considered several inter related parts for calculation of economic analyze. Those are basically as investment costs, operating and maintenance costs and return of investment part (ROI). Investment costs consist of required modification on existing system, main parts and components of waste heat recovery system (WHRS) and subsequently its piping, auxiliary systems as well. Operating hour parameters of components is taken from power plant archives.

On the other hand, ORC components and additional equipment prices are accounted by extrapolation. Therefore, the result respect to data, is obtained in applicable and approximate values. In this step of thesis similar studies taken into consideration in terms of some economical calculation approaches.

Table 4.2. Investment cost specifications

		Assumptions	**Cost (USD)**
Investment	Modification of the ESP	ESP modification (0,7% improvement)	500.000
	Flue gas heat recovery system (FGHRS)	The budget figure from Babcock	3.500.000
	Circulating Water Pump (Steam Powered Pump)	Extrapolated from similar project	350.000
	Piping and auxiliary systems	Estimation (10% of FGHR)	350.000
	Others	Estimation (10% of total investment)	480.000
	TOTAL		**5.180.000**

Investment cost section includes ESP modification because of residual dust content of flue gas upstream of ESP on existing plant is adversely affected heat transfer of WHRS. Thus, WHRS provider recommends to upgrade ESP in order to make a suitable dust parameters with respect to WHRS (Table 4.3)

Table 4.4. Operating and Maintenance Cost

	Full Load Op. Hours		**8000h**	**5000h**
O&M Costs	O&M costs, consumables, etc.	Water for continuous cleaning the FGHRS	16.000($)	10000($)
	O&M auxiliary power + or -	The power consumption of the pumps and increase in the power consumption of the IDF to overcome the pressure loss	90.520($)	60.325($)
	O&M maintenance	Piping, insulation, pumps, soot blower, Cleaning, (jet cleaning)	100.000($)	100.000
	O&M cost others		10.000($)	10.000($)
	O&M Sum:Cost		**216.520($)**	**180.325($)**

O&M expenditures is fulfilled according to 8000 hours (scenario A) and 5000 hours (scenario B). Due to the archives of power plant circulating water pump is in operation more than 8000 hours annually. Nevertheless, low demand scenario take into account as 5000 hours in operation option (Table 4.5).

Table 4.6. Return of Investment (Years)

	Equivalent Full Load Op. Hours		**8000h**	**5000h**
ROI	Annual Saving	Fuel and energy saving	**1.377.266($)**	**527.090($)**
	Interest Rate	2% of the total investment	**105.864($)**	**105.864($)**
	Annual Cost	O&M Sum	**216.520($)**	**180.325($)**
	Net Annual Saving		**1.054.882($)**	**840.000($)**
	ROI (years)		**5,02**	**6,30**

As a result, it seems to be return of investment of this project is around 5-6 years period. In fact, payback period strongly depends on power plant yearly generation values, availability percentage. It is supposed to be steam power plants looked after well O&M period can be withstand until 40 years. That is to say, it is necessary to operate 15% of whole life of power plant to get payback of this project (Table 4.7).

5. CONCLUSION

In this study, ORC design studies have been applied on steam power plant in use and thermodynamic calculations have been performed for each ORC component. The results obtained based on first law of thermodynamic and second law of thermodynamic have also cross checked by entropy balance equations. Due to the most of the life of steam power plant in operation actualized around full load, the calculation parameters are taken with respect to 100 load%, 660MW. Consequently, the design project success depends on the power plant full load operating hours. Therefore, it's considered while performing economic analyze and payback calculations as scenario A (higher full load operating hours) and scenario B (lower full load operating hours) respect to operating hour stats from power plant archives.

Secondly, we have both internal and external further options for recovering steam power plant waste. Those are mainly as follows:

- Feed water & Condensate preheating: Target is to decrease coal consumption thus while minimizing adverse effect of coal in boiler section and along flue gas ducting and stack also increasing the power plant system efficiency by combusting less coal for the same amount electricity generation.
- Fresh air heating: Assisting to regenerative air preheater combustion air is also preheated by flue gas wastes. Therefore, it can be obtained better combustion efficiency in boiler and thus reducing boiler losses such as unburnt coal amount.
- Clean gas heating: This offer is more related in environmental facts. By increasing clean gas temperature via establishing heat exchanger

between raw flue gas path and clean gas to stack path, it could be diverged from acid rainfall threat zone. Nevertheless, it's improbable offer due to the only deal with environmental utility factor for the power plant operation aspects.

- Heating greenhouses: While looking at location issues, power plant has been installed closer to the agricultural area. In this way, it's possible to establish a steam transfer line to preheat fruits and vegetables during 5-6 months in area.

But this offer have been limited with agricultural processes and only offer seasonal recovery benefits.

Lastly, the recovered amount for the usage of R236FA 4.67 MW from the flue gas waste and 4.78 MW recovered via using R245FA. Those values are equal to 8.49% of all internal consumption of unit for R236FA and % 8.65 for R245FA. This reflects positively to system as 0.314% efficiency increase by R236FA and 0.322% by R245FA.Thus, net system efficiency increases from 39.8% to 40.1 %.

The return of investment for different scenario options both yielded design project of ORC is in feasible range in terms of whole operation life of steam power plant respectively. According to the Scenario A (8000 hours steam powered pump running) ROI becomes 5.02 years and to the Scenario B (5000 hours steam powered pump running)

ROI becomes 6.3 years. In this case, regarding to whole life of steam power plant 11.1% is going to be spent to equalize payback period for high demand operating hours scenario and 14% for low demand operating hour scenario respectively. To be able to shorten the payback period it is necessary to take into account different options while maximizing recovered values because of that reason superior maintain ability of flue gas ducting and stack coating against acid formation on path. Hence, it could go below 110 ^{O}C limit downstream ORC evaporator, reducing waste as well.

5.1. Future Works

It is possible to integrate further applications on conventional steam power plant to be able to eliminate weaken part of conventional system by renewable techniques. In this respect, condensate & feedwater preheating system could be replaced with solar assisted preheating system thus either more kept turbine extraction and obtained more power output or power boosted availability of steam turboset machinery during frequency influence period. As much as solar assisted system in use is possible due to the availability of daily, monthly and seasonal weather conditions, the payback period of investment cost will be lowered.

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CURRICULUM VITAE

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APPENDIX

Thermodynamic properties for each stream at full(%100) load.

No	Zone	Stream	Mass flow rate(kg/s)	Temperature (ºC)	Pressure (Bar)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)
0	Dead State	Air		25	1	25,425	6,949
1	FD fan inlet	Air	586.266	25	1	25.425	6.949
2	FD fan outlet	Air	586.266	27.96	1.03	28.439	6.951
3	ID fan inlet	Fluegas	636.373	158.079	0.978	164.944	7.198
4	ID fan outlet	Fluegas	636.373	161.055	1	168.112	7.2
4	Evap. inlet	Fluegas	636.373	161.055	1	168.112	7.2
5	Pump inlet	R236FA	193.358	30.926	3.3	237.794	1.130
6	Pump outlet	R236FA	193.358	32.060	21.420	239.485	1.131
7	Evap. Outlet	Fluegas	636.373	110	0.999	114.201	7.068
8	Turbine Inlet	R236FA	193.358	104.978	21.420	417.540	1.630
9	Turbine Outlet	R236FA	193.358	46.11	3.3	393.387	1.641
10	Seawater inlet	Seawater	1927.416	22	1.5	87.829	0.309
11	Seawater out	Seawater	1927.416	25.926	1	103.532	0.362
5	Pump inlet	R245FA	148.312	32.178	1.920	242.016	1.144
6	Pump outlet	R245FA	148.312	32.790	14	243.160	1.145
8	Turbine inlet	R245FA	148.312	104.638	14	476.887	1.793
9	Turbine outlet	R245FA	148.312	49.218	1.920	444.655	1.807
10	Seawater inlet	Seawater	1455.294	22	1.5	87.829	0.309
11	Seawater out	Seawater	1455.294	27.178	1	108.543	0.378