

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

MSc THESIS

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ENERGY AND EXERGY ANALYSIS OF A STEAM POWER PLANT

DEPARTMENT OF MECHANICAL ENGINEERING

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ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

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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 01/08/2013

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ABSTRACT

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ENERGY AND EXERGY ANALYSIS OF A STEAM POWER PLANT

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INSTITUTE OF NATURAL AND APPLIED SCIENCES
DEPARTMENT OF MECHANICAL ENGINEERING

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In this study, the energy and exergy analyses of the steam power plant have been performed with three different operating loads (100%, 70% and 40%). This steam power plant is designed to operate at a subcritical steam conditions and has a power capacity of 660 MW at full load. The primary objectives of this study are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses. Influences of three different loads on the useful power, reversible power and irreversibility have been investigated for each component. In addition, the second law efficiency of system components and the overall efficiency of the plant have been computed.

Energy losses mainly occur in the condenser and that exergy losses mainly take place in the boiler. According to these results, the exergy analysis is more significant compared to the energy analysis. It is found that if the exergy losses are reduced, the power plant efficiencies are positively affected.

Key Words: Energy Analysis, Exergy Analysis, Operating Load, Second Law Efficiency

ÖZ

YÜKSEK LİSANS TEZİ

BUHARLI GÜÇ SANTRALİNİN ENERJİ VE EKSERJİ ANALİZİ

Mehmet TONTU

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Bu çalışmada, buharlı güç santraline enerji ve ekserji analizi üç farklı çalışma yükü (%100, %70 ve %40) için uygulanmıştır. Bu buhar santrali kritik altı durumlarda işletilecek şekilde tasarlanmıştır ve tam yükte 660 MW güç kapasitesine sahiptir. Bu çalışmanın amacı, sistemi oluşturan her bir ekipmanı ayrı ayrı inceleyerek en fazla enerji ve ekserji kaybı olan ekipmanları belirlemek olup ayrıca üç farklı çalışma yükünün sistemi oluşturan ekipmanların faydalı güç, tersinir güç ve tersinmezlik miktarları üzerindeki etkisi incelenmiştir. Ek olarak her bir ekipmanın ve santralin ikinci yasa verimi hesaplanmıştır.

Üç farklı yükte de en fazla enerji kaybı kondenserde olmakta olup, ekserji kaybı kazanda olmaktadır. Sonuçlara göre ekserji analizinin enerji analizine göre önemli olduğu kanısına varılmıştır. Ekserji kayıpları azaltılması durumunda santral verimleri olumlu yönde artacaktır.

AnahtarKelimeler: Enerji Analizi, Ekserji Analizi, Çalışma Yükü, İkinci Yasa verimi.

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NOMENCLATURE

CV	: Control volume
E	: Energy
fg	: Flue gas
FLT	: First law of thermodynamics
h	: Enthalpy
h_0	: Enthalpy at dead state
h_{in}	: Enthalpy at inlet
h_{out}	: Enthalpy at outlet
HP	: High pressure
Hz	: Hertz
I	: Irreversibility
IP	: Intermediate pressure
kW	: Kilowatt
LP	: Low pressure
m	: Mass flow rate
M	: Molar mass
m_f	: Fuel mass flow rate
m_{in}	: Inlet mass flow rate
m_{out}	: Outlet mass flow rate
MRA	: Minimum required amount of air
n	: Molar ratio
P	: Pressure
P_0	: Ambient pressure
Q	: Heat
q	: heat per unit mass
Q_f	: Heat due to fuel
Q_{in}	: Input heat
q_{in}	: Input heat per unit mass
Q_l	: Heat loss

q_{out}	: Output heat per unit mass
Q_{out}	: Output heat
RAA	: Required amount of air
rpm	: Revolution per minute
s	: Entropy
s_0	: Entropy at dead state
S_{gen}	: Entropy generation
SLT	: Second law of thermodynamics
T	: Temperature
T_0	: Ambient temperature
T_k	: Surrounding temperature
V_{in}	: Inlet velocity
V_{out}	: Outlet velocity
W	: Work
w	: Work per unit mass
W_{in}	: Input work to pump
W_{out}	: Output work from turbine
W_{rev}	: Reversible work
X	: Exergy
x	: Molar ratio
X_{in}	: Inlet exergy
X_l	: Exergy loss
X_{out}	: Outlet exergy
z_{in}	: Level at inlet
z_{out}	: Level at outlet
Δh	: Enthalpy differences between inlet and outlet
Δke	: Kinetic energy differences between inlet and outlet
Δpe	: Potential energy differences between inlet and outlet
η_l	: First law efficiency
η_{II}	: Second law efficiency
θ	: Energy of flowing fluid per unit mass

λ	: Excess air coefficient
ψ	: Exergy per unit mass
ψ^{ch}	: Chemical exergy per unit mass
ψ_{in}	: Exergy at inlet
ψ_{out}	: Exergy at outlet
ψ^{ph}	: Physical exergy per unit mass

1. INTRODUCTION

Energy is one of the basic necessities in modern life. There aren't any fields which are not being used energy in everyday life. Nowadays, energy consumption is a measurement for development of communities. Especially electrical energy and heat energy play on important role in our lives. These energies which are produced from scarce natural resources are becoming more valuable by the day due to an increase in demand (Yazıcı and Selbaş, 2011).

With a young and growing population, low per person electricity consumption, rapid urbanization and strong economic growth, Turkey is one of the fastest growing power markets in the world, for nearly two decades. The growth in electricity generation in recent years was below growth in electricity demand. Energy is considered to be a key player in the generation of wealth and also a significant component in economic development (Tekel, 2006).

Energy consumption is one of the most important indicators of economic and social development. Population growth, industrialization, expansion of technology and rising of welfare direct proportion to the increase in energy consumption is inevitable. In order to reach the level of developed countries, as if Turkey, developing countries such as much more must do more production (IEA, 2004).

The world energy needs heavily on fossil fuels for electricity production. The majority of the world's power generation is met by fossil fuels, particularly coal and natural gas. Despite the growth of renewable energy installations like wind and solar power, the heavy dependence on fossil fuels is expected to continue for decades. Despite the depletion 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. Even though cleaner renewable sources of energy are being rapidly developed, their relative cost and current state of technology have not advanced to a stage where they can significantly reduce our dependence on fossil fuels. Therefore, given the continued reliance on fossil fuels for some time, it is important that fossil fuel plants reduce their environmental impact by operating more efficiently (Regulagadda et al, 2010).

1.1. Electricity Generation in Turkey

Turkey is placed between Europe and Asia. It is situated in Anatolia and southeastern Europe bordering the Black Sea, the Aegean Sea and the Mediterranean Sea. Turkey extends more than 1,600 kilometers from west to east but generally less than 800 kilometers from north to south. Its total land area is about 779,452 square kilometers, of which 755,688 square kilometers are in Asia and 23,764 square kilometers in Europe. From the point of view of this geographical location, Turkey is behaved as bridge between southeastern Europe and Asia. Moreover, the economy and population of Turkey grow rapidly. As a developing country, Turkey's population is estimated to be over 100 million by the year of 2020. For this reason, demands for energy and particularly for electricity have been growing fast. Electricity energy is a vital input for the technological, social and economic development of Turkey (Bilgili, 2007).

Electricity in Turkey is generated from the thermal, hydro, wind and the geothermal power plants. The development of Turkey's installed capacity in electricity generation between 1923 and 2005 is illustrated in Figure 1.1.

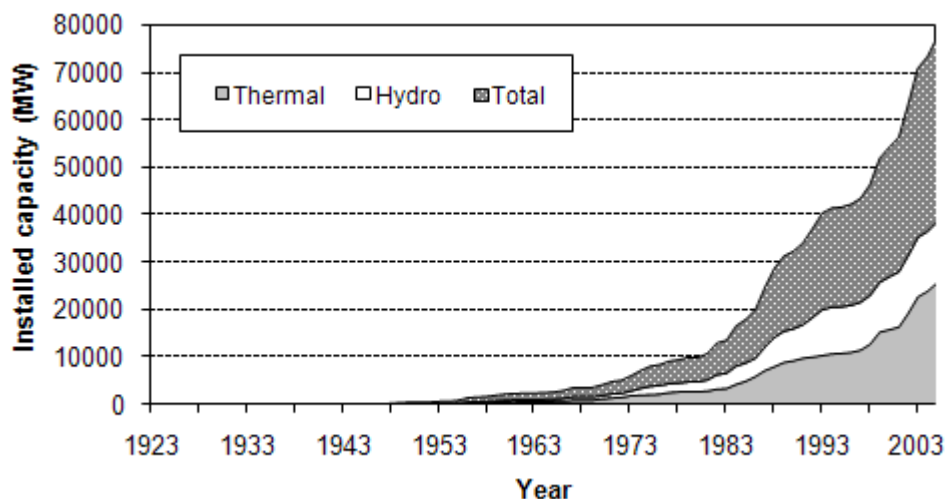


Figure 1.1. Development of Turkey's installed capacity in electricity generation (TEİAŞ, 2011)

The first power generation in Turkey had been from a 2 kW dynamo connected to water-mill in Tarsus-Mersin, in 1902. Although this power plant was the first experience of Turkey in electricity generation, After that, coal-fired Silahtarağa Power Plant was put into service in Istanbul, in 1913. When the Republic of Turkey was founded in 1923, its installed capacity of electricity production and total electricity production were 32.8 MW and 45 GWh, respectively (IEA, 2001; Yılmaz and Uslu, 2007). In 1935, the Electrical Power Resources Survey and Development Administration (Elektrik Isleri Etut Idaresi, EIE) was established for the purpose of carrying out surveys and preparatory work to assess hydro potential and prepare a hydro power plant projects. Construction of power plants began on a larger scale, by both private and publicly-owned entities in the 1950s. At the beginning of the decade, installed capacity was about 408 MW and electricity generations reach to 500 million kW/h (IEA, 2001). From beginning 1950 to the end of 1960, five hydroelectric power plants, and one thermal power plant are founded.

By 1970, installed capacity of electricity was increased to about 2,235 MW, and both growing power consumption and the government's electrification plans required more coherent organization of the power industry. At the time, only 7% of all villages were electrified. As a consequence, the government established the Turkish Electricity Authority (Türkiye Elektrik Kurumu, TEK) as a fully state-owned and state run entity that year. In 1993, TEK was split into 2 separate state-owned electric companies called as the Ministry of Energy and Natural Sources (MENR): The Turkish Electricity Generation Transmission Corporation (Türkiye Elektrik Üretim-İletim A.S., TEAS) and Turkish Electricity Distribution Corporation (Türkiye Elektrik Dağıtım A.S., TEDAS). In 1999, Turkey's installed power generation capacity reached 26,117 MW, and 99.9% of its population was connected to the electricity grid (IEA, 2001; Bilgili, 2007).

At the end of 2012, Turkey has reached to currently total power amount 57,039.8 MW of 35,027.2 MW thermal energy, 162.2 MW geothermal energy, 19,589.9 MW hydro energy and 2,260.6 MW wind energy. At the end of 2012, demand of gross electricity was 242.4 billion kWh and power demand was 39,044.9 MW. Total 239.5 billion kWh of electricity were produced and 5.8 billion kWh were

imported and 2.9 billion kWh electricity of total supply were exported.

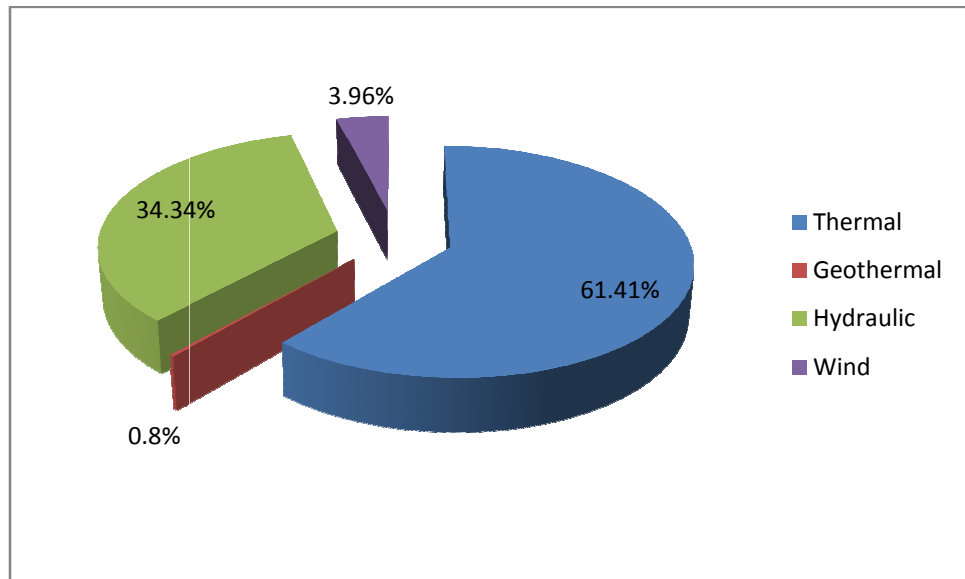


Figure 1.2. Installed capacities in 2012 according to plant type (EUAŞ, 2012)

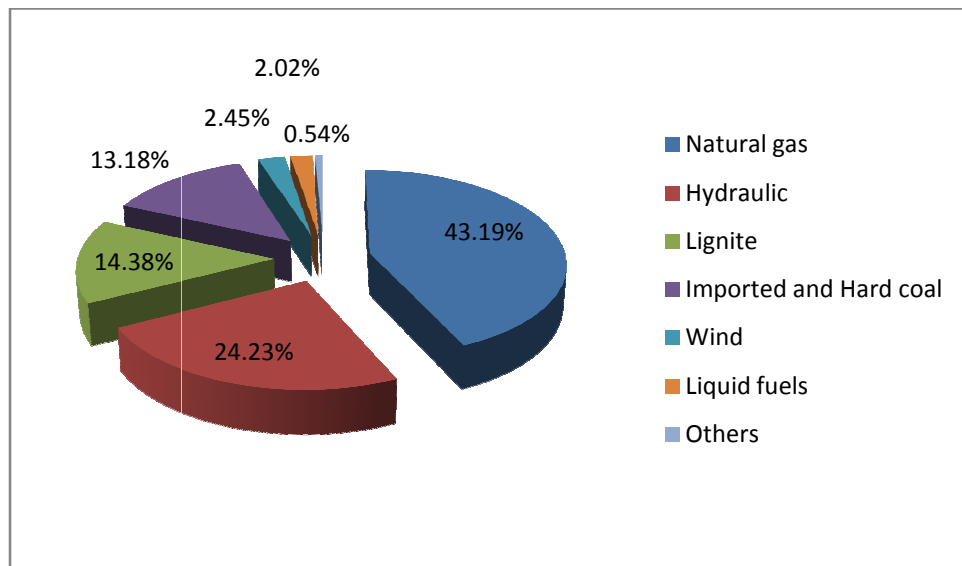


Figure 1.3. Electrical generations in 2012 according to energy sources (EUAŞ, 2012)

In 2013, electrical energy demand will be expected with an increase to 255.0 billion kWh by 5.5% to compare the previous year. The current system has ability (capacity) to produce total amount 289.4 billion kWh belongs to 216.2 billion kWh of thermal power plants, 64.6 billion kWh of hydroelectric power plant, 7.5 billion kWh of wind power plants, 1.0 kWh of geothermal power plants. Electric power

producing and transmission development planning provides to meet electricity demand sustainability, reliability, high-quality and cost-effectiveness. While looking over value of consumption of recent years, consumption has mostly increased in spite of decreasing in consumption growth rate in consisting economic crisis and earthquake years. Demand will expected to increase by 7.5% on average in next-ten-years period according the results of demand forecasting studies which considering basic parameters of population, industry and economic developments. Thus, expected energy demand to be 255.0 billion kWh in 2013 is estimated to reach 303.1 billion kWh in 2015. In addition, expected plant powers demand to be 41,500 MW in 2013 is estimated to reach 46,800 MW in 2015.

Turkey has been one of the fast growing power markets of the world for the last two decades due to its young population and growing energy demand per person, its fast growing urbanization and its economic development. It is expected that the demand for electric energy in Turkey will be 580 billion kWh by the year 2020. (TEİAŞ, 2012).

1.2. Thermal Power Plants

Thermal power plants are widely utilized throughout the world for electricity generation, and coal is often used to fuel these plants. Although the world's existing coal reserves are sufficient for about two centuries, the technology largely used today to produce electricity from coal causes significant negative environmental impacts. To utilize coal more effectively, efficiently and cleanly in electricity generation processes, efforts are often expended to improve the efficiency and performance of existing plants through modifications and retrofits, and to develop advanced coal utilization technologies.

Steam is an important medium of producing mechanical energy. Steam has the advantage that, it can be raised from water which is available in abundance it does not react much with the materials of the equipment of power plant and is stable at the temperature required in the plant. Steam is used to drive steam engines, steam turbines etc. Steam power plant is most suitable where coal is available in

abundance. In most countries, numerous steam power plants driven by fossil fuels so there are different types of coal combustion system. These are stoker furnaces, pulverized coal furnaces and fluidized bed combustion system.

A stoker furnace is one of the oldest types of furnaces and is still in use today. In this type, the coal is introduced on large grate and it is finally burned in a stationary bed. The stoker furnace burns coarsely crushed coal. Part of combustion air is primary air which is introduced below burning bed. The primary air initiates the combustion process and also cools the grate. Secondary air is introduced from above the burning bed to complete combustion process.

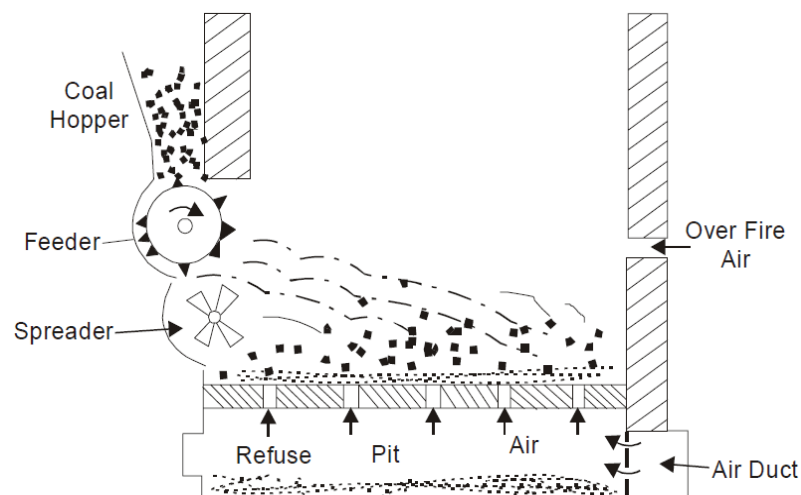


Figure 1.4. Stoker furnaces combustion system

Stoker furnace is shown in Figure 1.4. In this stoker the coal from the hopper is fed on to a feeder which measures the coal in accordance to the requirements. Feeder is a rotating drum fitted with blades. Feeders can be reciprocating rams, endless belts, spiral worms etc. From the feeder the coal drops on to spreader distributor which spread the coal over the furnace. The spreader system should distribute the coal evenly over the entire grate area. The spreader speed depends on the size of coal.

Pulverized coal furnace burns finely powdered coal. This combustion system produces much higher capacities than stoker furnaces. It gives fast response because

of increase coal surface exposure thus permitting rapid combustion therefore there is little unburned fuel in the boiler also it reduces the amount of excess air required for complete combustion and this reduces nitrogen oxides emission. This furnaces is suitable for variety coal types.

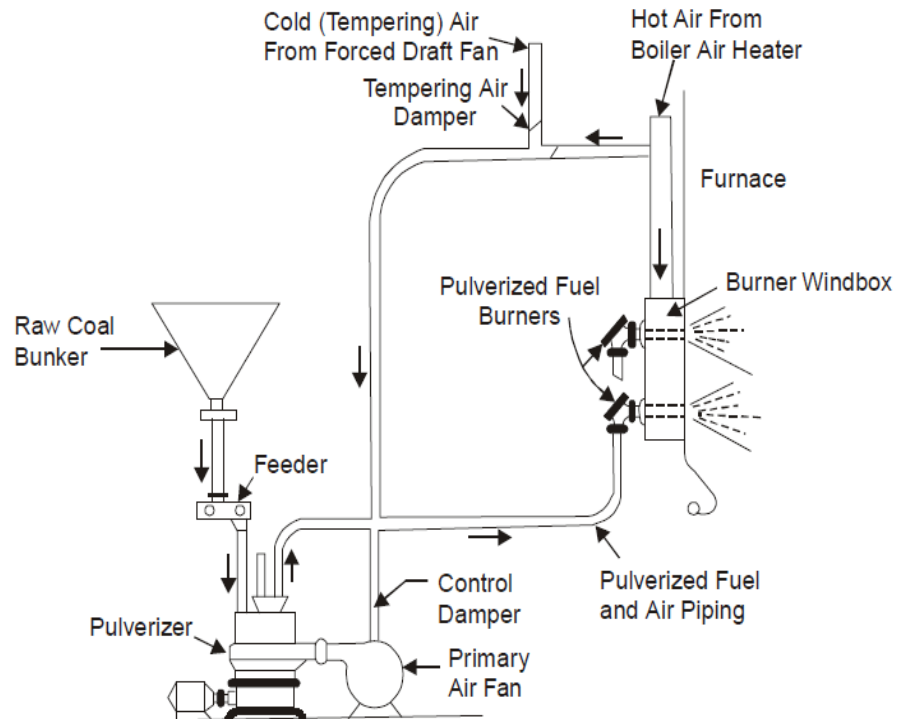


Figure 1.5. Pulverized coal combustion system

Pulverized coal furnace is shown in Figure 1.5. In this type coal is fed to mill by means of feeder. Coal is crushed finely by mill, size of pulverized coal is approximately less than 0.3 mm. After pulverizing process, pulverized coal is conveyed pneumatically to burners using primary air, this air is also used for coal drying. Mill exit temperature is arranged by control dampers for fire protection.

Fluidized bed combustion system is shown in figure 1.6. In this type of firing, crushed particles of coal and limestone are injected into the fluidized bed so that they spread across an air distribution grid. The combustion air, blown through the grid, has an upward velocity sufficient to cause the coal particles to become fluidized. Unburned carbon leaving the bed is collected in a cyclone separator and returned

back to the bed for another go at combustion. The boiler evaporator tubes are immersed directly in the fluidized bed and the water tube produces very high transfer rates, reducing the size of unit. This arrangement also produces very low combustion temperatures compared to pulverized coal combustion. Low combustion temperatures inhibit formation of nitrogen oxides also provides reducing fouling around a water tubes. The main advantage of fluidized-bed combustion is the ability to desulfurization the fuel during combustion in order to meet air quality standards for sulfur dioxide emissions. Desulfurization is accomplished by the addition of limestone directly to the bed. Fluidized-bed combustion is still undergoing development and has other attractive features.

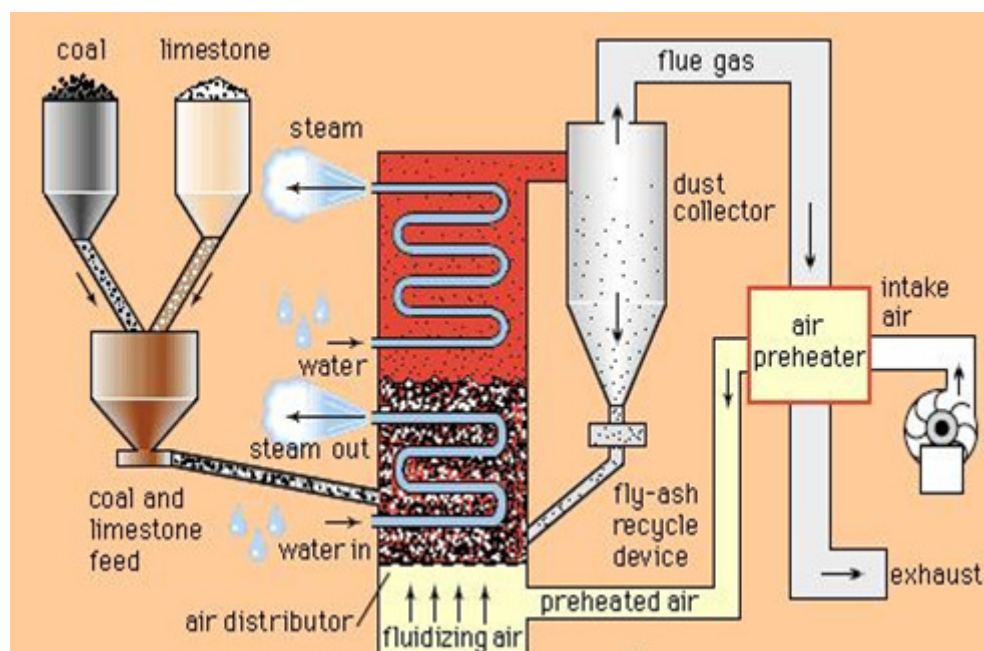


Figure 1.6. Fluidized bed combustion system

The simple reheat rankine cycle is shown in Figure 1.7. Water enters the pump at state 1 as a saturated liquid and is compressed to the operating pressure of the boiler through LP and HP heaters. LP and HP heaters increase the temperature of saturated liquid and increase overall plant efficiency. Water enters the boiler as a compressed liquid at state 4 and leaves as a superheated vapor at state 5. The boiler is basically a large heat exchanger where the heat originating from combustion gases,

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 is often called the steam generator. The superheated vapor at state 5 enters the turbine, where it expands and produces work by rotating the shaft connected to an electric generator. The pressure and the temperature of steam drop during this process to the values at state 6 so that dryness of steam is increased to eliminate erosion on turbine blades and increase efficiency steam is back to boiler to reheating at state 6, after reheating of steam thermal grade of steam increased and fed to LP turbine and produce shaft work then steam enters the condenser. At this state, steam is usually a saturated liquid–vapor mixture with a high quality. 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 (Çengel and Boles, 2006).

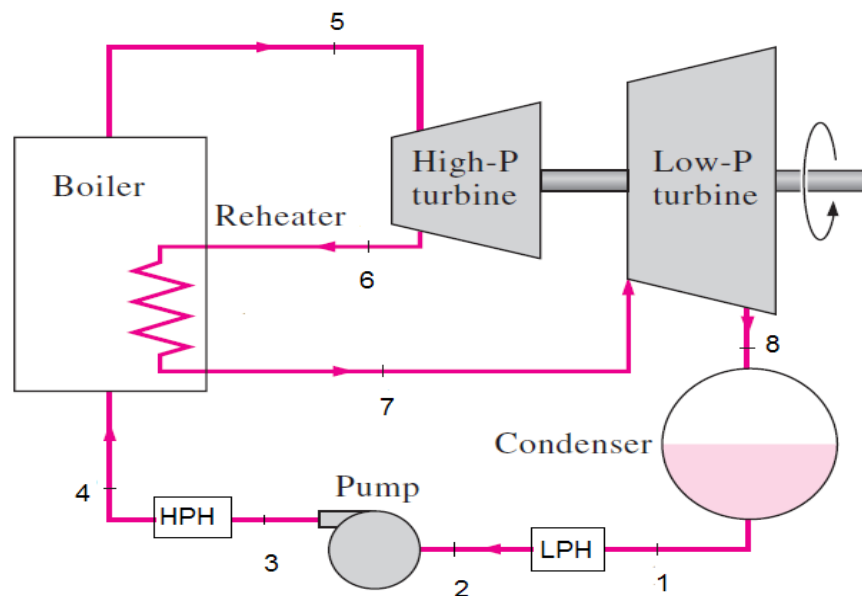


Figure 1.7. The simple reheat rankine cycle (Çengel and Boles 2006).

There are many methods to increase of efficiency of rankine cycle. If condenser pressure is decreased, enthalpy differences between point 7 and 8 is increased so shaft work is increased. Second way is, in case of increasing temperature of point 5, enthalpy of point 5 is increased also shaft work is increased.

Third way is, in case of increasing boiler pressure, not only evaporation heat of steam is decreased but also input energy is decreased therefore rankine cycle efficiency is increased. T-s diagram of reheat cycle is shown in Figure 1.8.

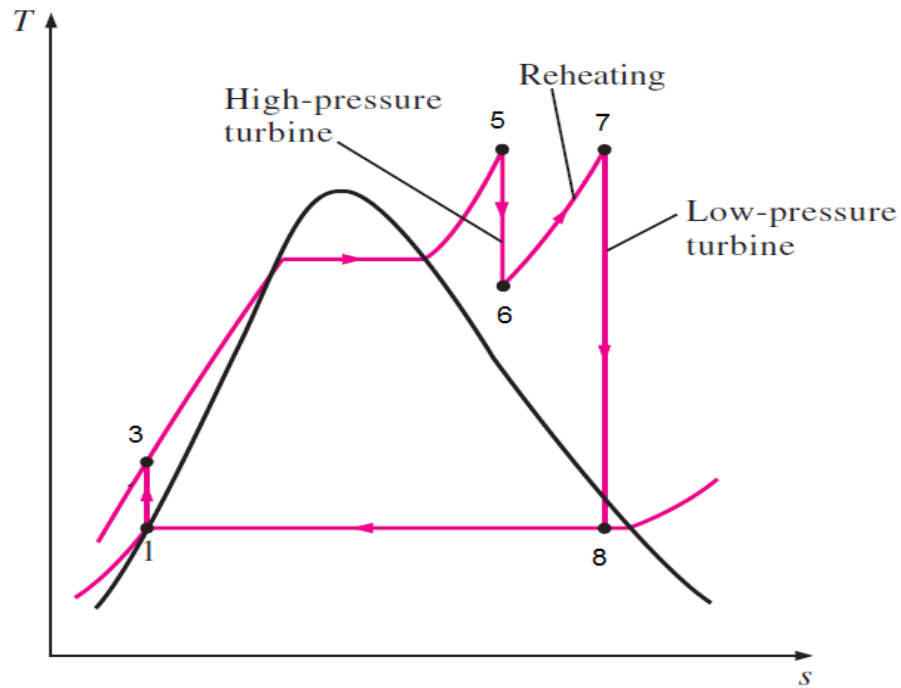


Figure 1.8. T-s diagram of reheat rankine cycle

As of the end of 2012, thermal power plants in Turkey supply approximately 62% of Turkey's total installed capacity for electricity power generation. 173,847.2 GWh of total generation capacity (239,081 GWh) of Turkey has been met by thermal power plant.

Electricity generated by thermal power plants is as follows: 56.85% lignite, 39.49% natural gas, 2.81% hard coal, 0.84% liquid fuels. Fuel used in thermal power plant is shown in Figure 1.9. (EUAS, 2012).

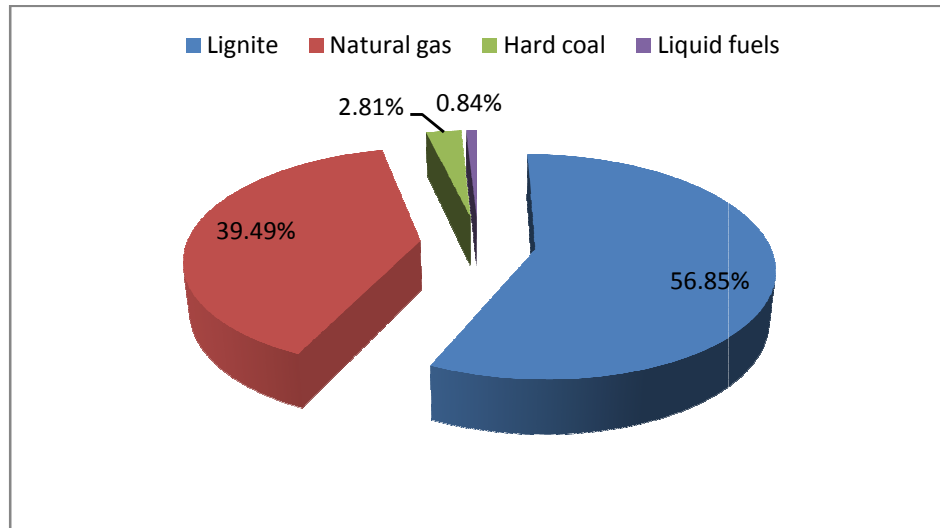


Figure 1.9. Fuels used in thermal power plants

Thermal power plant installed capacity and total power plant capacity are shown in Figure 1.10. This figure shows variation of thermal power plant installed capacity according to total capacity.

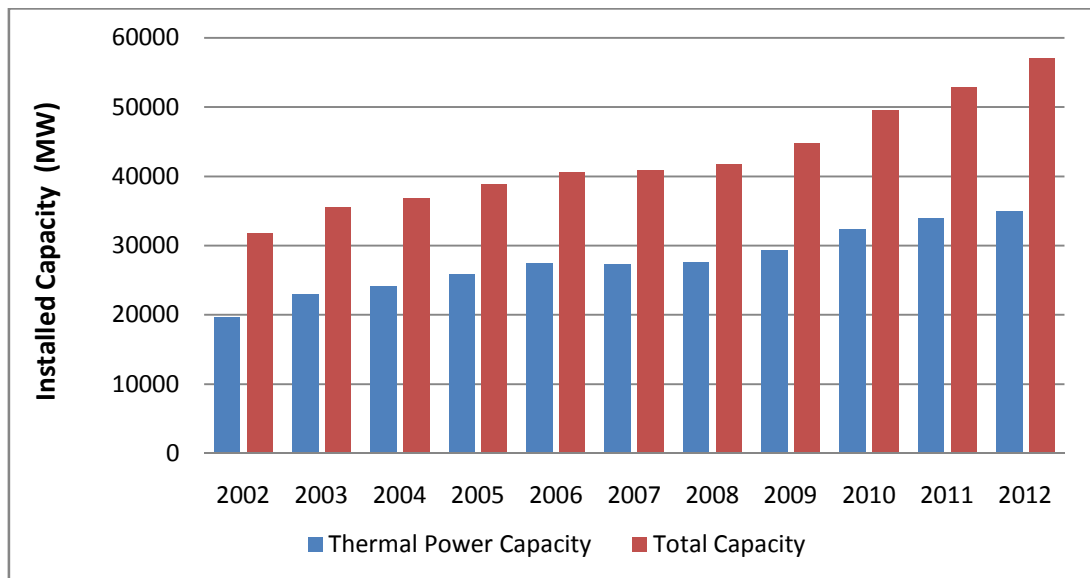


Figure 1.10. Cumulative installed capacity of thermal and total power (ETKB, 2012)

Thermal power plant electricity generation and total power plant electricity generation are shown in Figure 1.11. This figure shows variation of thermal power plant electricity generation with respect to total power plant electricity generation.

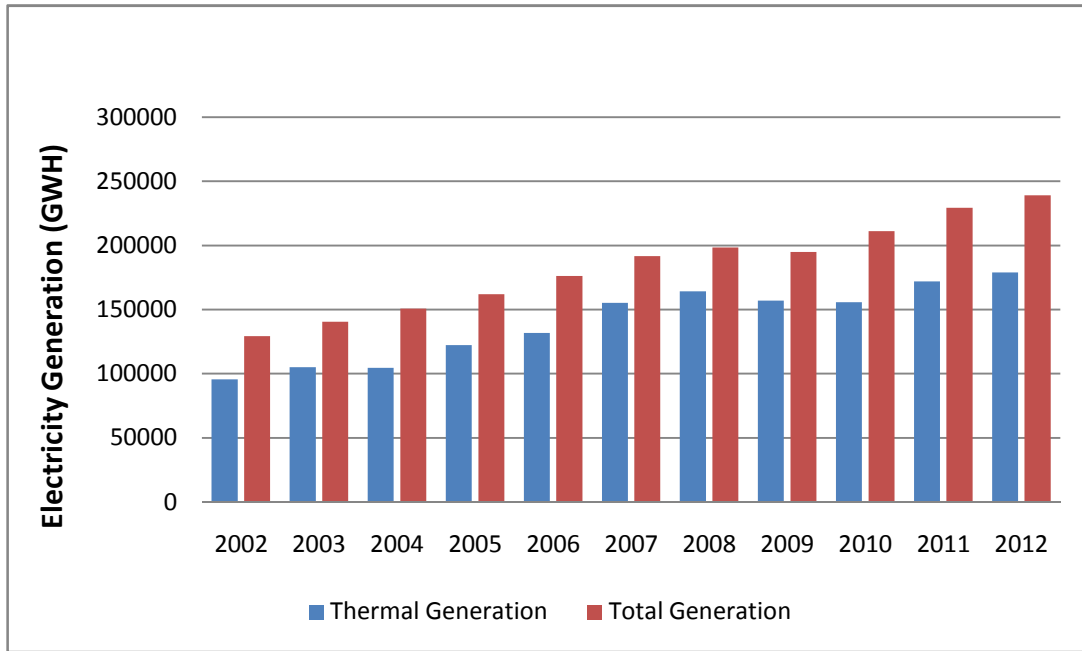


Figure 1.11. Cumulative electricity generation of thermal and total power (ETKB, 2012).

Turkey includes many different coal fired plant and they are shown in Figure 1.12.



Figure 1.12. Coal fired power plants in Turkey (EPDK, 2013)

Properties of coal fired power plant are shown in Table 1.1.

Table 1.1. Coal fired power plants in Turkey (EPDK, 2013)

Power Plant	Fuel Type	Total Power (MW)	Burning Type	Operation Year
Sugözü	Imp. Coal	1320	Pulverized	2003
Eren	Imp. Coal	1360	Fluidized Bed	2010
			Pulverized	2010
İçdaş	Imp. Coal	405	Fluidized Bed	2005 - 2009
Afşin-Elbistan A	Lignite	1355	Pulverized	1984
Afşin-Elbistan B	Lignite	1440	Pulverized	2004
Çan	Lignite	320	Fluidized Bed	2004
Çayırhan Park	Lignite	620	Pulverized	2000
				1987
Kangal	Lignite	457	Pulverized	2000
				1989
Kemerköy	Lignite	630	Pulverized	1993-1995
Orhaneli	Lignite	210	Pulverized	1992
Seyitömer	Lignite	600	Pulverized	1973 - 1989
Soma A	Lignite	44	Pulverized	1957
Soma B	Lignite	990	Pulverized	1981 - 1992
Tunçbilek A	Lignite	65	Pulverized	1956
Tunçbilek B	Lignite	300	Pulverized	1978
Yatağan	Lignite	630	Pulverized	1982 - 1986
Yeniköy	Lignite	420	Pulverized	1986 - 1987
Çatalağzı	Hard Coal	300	Pulverized	1991
Çolakoğlu 2	Hard Coal	180	Pulverized	N/A

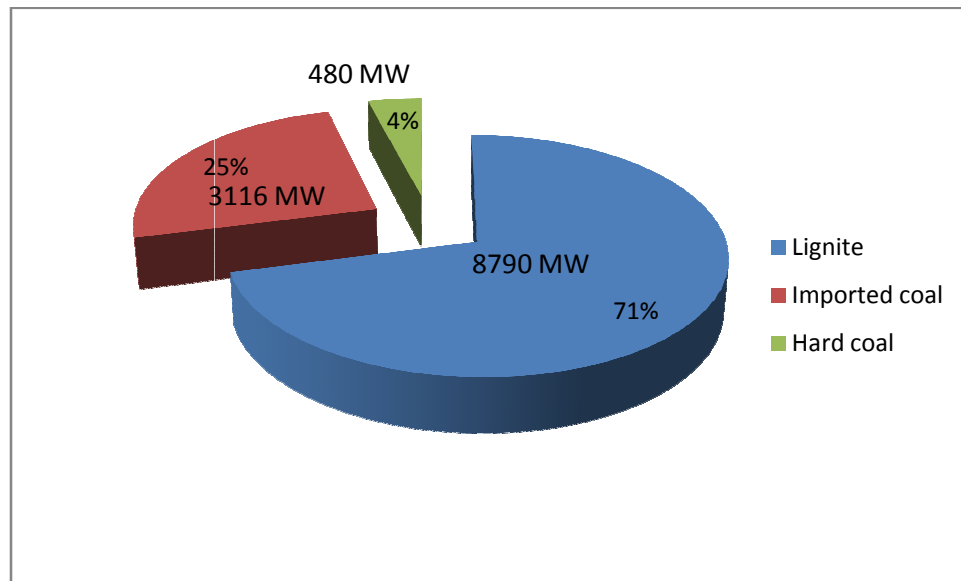


Figure 1.13 Distribution of installed capacity according to coal types

1.3. Energy and Exergy

Energy is the capacity for doing work, generating heat, and emitting light briefly (Raja et al, 2006). A fundamental concept of thermodynamics also one of the most significant aspects of engineering analysis. Energy can be stored within systems in various macroscopic forms: kinetic energy, gravitational potential energy, and internal energy (Moran, 1999). Energy is always conserved, it can neither be produced nor consumed and energy is a measure of quantity (Mborah and Gbadam, 2010). Energy can be transferred to or from a closed system in two distinct forms: heat and work. For control volumes, energy can also be transferred by mass flow. An energy transfer to or from a closed system is heat if it is caused by a temperature difference. Otherwise it is work, and it is caused by force acting through a distance (Çengel and Boles, 2006).

Exergy is the maximum possible work that can be produced by a system as it is brought into equilibrium with a specified reference environment. At specified reference environment useful work potential is zero therefore it is called unavailable energy. Exergy can be reckoned as a measure of the quality or usefulness of energy. Unlike energy, exergy is not conserved, but it is destroyed in any practical process.

The exergy destruction during a process is proportional to the entropy generation in it, which accounts for the inefficiencies due to irreversibility (Ray et al, 2010). Energy, exergy and unavailable energy are shown in Figure 1.14.

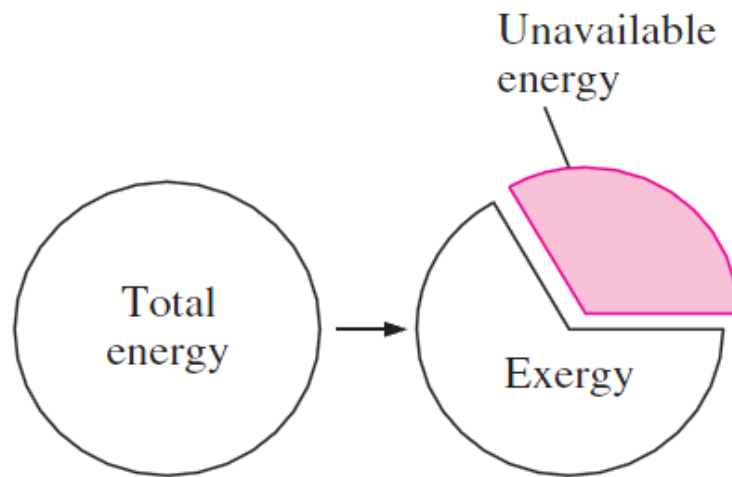


Figure 1.14. The relationship between energy and exergy (Çengel and Boles 2006).

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. Like energy, exergy can be transferred or transported across the boundary of a system. For each type of energy transfer or transport there is a corresponding exergy transfer or transport. Exergy analysis takes into account the different thermodynamic values of different energy forms and quantities that are work and heat. The exergy transfer associated with shaft work is equal to the shaft work. The exergy transfer associated with heat transfer, however, depends on the temperature at which it occurs in relation to the temperature of the environment (Dinçer and Rosen, 2011). Comparison of energy and exergy is shown in Table 1.2.

As a result, energy analysis is traditionally used in industries to carry out performance comparisons and optimizations. Exergy analysis is to identify the magnitudes and the locations of exergy losses, in order to improve the existing systems, processes or components (Ray et al. 2010)

Table 1.2. Comparison of energy and exergy (Dinçer and Rosen, 2011)

Energy	Exergy
Dependent on properties of only a matter or energy flow, and independent of environment properties.	Dependent on properties of both a matter or energy flow and the environment
Has values different from zero when in equilibrium with the environment	Equal to zero when in the dead state by virtue of being in complete equilibrium with the environment
Conserved for all processes, based on the FLT	Conserved for reversible processes and not conserved for real processes based on the SLT
Can be neither destroyed nor produced	Can be neither destroyed nor produced in a reversible process, but is always destroyed in an irreversible process
A measure of quantity only	A measure of quantity and quality

1.4. Aim and Outline of Study

The amount of energy consumption increases continuously. The growing demand of energy which is the most important need for humanity has put forward the efficient energy utilization and environmental issues especially. Hence, the studies to more clean and efficient use of limited energy resources have become popular. In this regard, energy and exergy analysis are used for evaluating the performances of power production and consumption processes.

Suresh et al. (2006) reported that the rate of depletion of fossil fuel reserves has necessitated the operation of power plants in the most efficient manner. The performance of thermal power plants is evaluated through energy performance criteria based on first law of thermodynamics, including electrical power and thermal efficiency. They stated that in recent decades, the exergy performance based on the

second law of thermodynamics has been found to be a useful method in design, evaluation, optimization and finally improvement of thermal power plants.

In the work of Kaushik et al. (2011), it was stated that the second law performance analysis can not only determine magnitudes, location and causes of irreversibility's in the plants, but also provides more meaningful assessment of plant individual components efficiency. These points of the second law performance analysis are the basic differences from first law analysis. Therefore, it can be said that performing second law and first law analyses together can give a complete depiction of system characteristics.

The present study was divided into two groups. Firstly, required data was obtained as such as main steam temperature, and condenser pressure etc. at three different loads. Afterwards the data were prepared for calculation of energy and exergy analysis.

Secondly, the energy and exergy analysis were performed using actual data taken from the existing power plant. After obtaining data, first of all, energy loss of equipments was calculated based on first law of thermodynamics. Then, physical exergy and chemical exergy and second law efficiency and corresponding irreversibility for each equipment were determined.

The necessary thermal analysis were performed for each equipments and the irreversibility's or exergy loss, energy loss and the second law efficiencies of them were also computed.

In conclusion, the objective of this work is to analyze efficiency and loss of steam power plant in terms of energy and exergy analysis. Sites of energy loss and exergy loss or irreversibility and second law efficiency were determined. The effects of varying the plant load on the energy and exergy analysis were also investigated besides the maximum exergy and energy loss were defined.

2. LITERATURE SURVEY

Filiz (2012) investigated the performance of a gas-fired boiler existing in Kardemir A.Ş. using coke gas, converter gas and furnace gas from the waste gases released during the production of iron and steel as a fuel was evaluated by performing energy and exergy analysis. Also, the boiler was economically analyzed from the point of exergy cost calculation and exergoeconomics. The boiler was approached as the open system and single unit and the energy and exergy analysis were performed considering this state. As a result of energy and exergy analysis of boiler, energy and exergy efficiencies were found to be 91% and 46%, respectively. It was determined that the energy loss and exergy loss (destruction) are 6445.5 kW and 21185.81 kW, respectively. It was also determined that the boiler has improvement potential of 11440.33 kW and the rate of exergy destruction of boiler to fuel using in the boiler is 0.54.

Şahin and Aydın (2012) studied energy and exergy analysis of Atlas Iskenderun Power Plant. This plant designed to operate at supercritical steam conditions with 600 MW output. The energy and exergy flow rates, irreversibilities and efficiencies of each unit were determined. Overall energy and exergy efficiencies of the plant are found to be 44.17% and 40.83%, respectively. The influence of reference ambient temperature on the efficiencies and irreversibilities of power plant have been investigated. Total reversible power in the cycle is equal to the input exergy of fuel and found as 1478.02 MWth. Also, the total irreversibility in the power cycle is calculated as 873.76 MWth which is the 59.11% of total exergy input to cycle. According to the energy analysis, energy losses associated with the condenser are significant. However, exergy analysis shows that only 1.46% of exergy was lost in the condenser.

Yazıcı and Selbaş (2011) in this study, has been made energy and exergy analysis of a steam power plant, some cases have been neglected (friction lost, pressure lost, the kinetic and potential energy etc). The first law of thermodynamics has been applied to write the energy equation of 500 MW steam power plant. Heat given to the boiler, the flow of power fluid, pump power, and transferred heat values

from condenser to the cooling water have been calculated separately in order to achieve to the necessary power in the steam turbine on different boiler temperatures. The second law of thermodynamics applied for steam boiler and condenser which are the main elements of steam power plant, irreversibility values of these elements have been calculated by exergy analysis. Total irreversibility of the system has been calculated and system component which produce the most irreversibility have been determined.

Karagöz (2011) In this study, the energy and exergy analysis were carried out using the actual operating parameters of a combined cycle power plant of 82 MW using natural gas. The power plant had two gas turbine units of 30 MW and steam turbine units of 22 MW. Based on the energy and exergy fluxes at inlet and exit of the devices in the power plant energy and exergy losses were determined. The result of energy analysis shows that the highest energy losses occurred in the chimney and the air-cooled condenser were 50.17% and 39.67%, respectively. The result of exergy analysis shows that the highest exergy losses occurred in the combustion chamber with 48.43% and the gas turbine with 37.33%. Thermal and exergy efficiencies of the combined cycle power plant were determined as 49% and 47%, respectively. Results suggest that exergy losses result from the flow, the combustion and the flue gases. Therefore, improvements must be done on the combustion chamber, steam turbine and condenser units.

Geredelioğlu (2011) studied exergy analysis of Çayırhan thermal power plant which are located in Ankara. In this study, energy, exergy and thermo economic analysis were performed in 2nd unit of a thermoelectric power plant which is still active in Turkey. Thermodynamic features of 46 nodes that selected in the thermoelectric power plant unit were estimated by EES programmer. According to these determinations, energy and exergy of each node were measured. The levels of beneficial power, reversible power and irreversible power of each equipment of the system were detected, general efficiency of the system was measured, and exergoeconomic factors were found out by determining the disappearing exergy ratios.

Regulagadda et al. (2010) performed thermodynamic analysis of a subcritical boiler–turbine generator a 32 MW coal-fired power plant. A parametric study is conducted for the plant under various operating conditions, including different operating pressures, temperatures and flow rates, in order to determine the parameters that maximize plant performance. The exergy loss distribution indicates that boiler and turbine irreversibility's yield the highest exergy losses in the power plant. In addition, an environmental impact and sustainability analysis are performed and presented, with respect to exergy losses within the system.

Mborah et al. (2010) examined the aim of this study is to use energy and exergy analysis to identify the locations and magnitudes of losses in order to maximize the performance of a 500 KW open system steam power plant at BOPP. The required outputs (work, heat and irreversibility) of the various components are assessed and calculated using mass, energy and exergy balance equations. The results indicate that about 50 % of heat energy generated in the combustor is destroyed. In conclusion, further improvement in the combustor will maximize the plant performance hence the results show how energy and exergy have been used to identify inefficient locations in the plant.

Ray et al. (2010) applied exergy analysis of a 500 MWe steam turbine cycle of an operating power plant is conducted under the design and off-design conditions with different degrees of superheat and reheat sprays. The analysis indicates how the first law-based analysis shows an apparent improvement in a feed water heater under an off-design condition, while the actual performance degradation is reflected through exergy analysis. The analysis also helps identifying the contribution of individual equipment in the overall increase of exergy destruction under off-design condition. Exergy analysis is also performed using off-line performance guarantee tests conducted before and after a unit overhauling. Pre-overhauling exergy efficiency figures of the major cycle equipment are compared with their respective design values to assess the need and extent of maintenance work, whereas post-overhaul exergy data is used to quantify the compliance with the guaranteed performance.

Kocaekiz (2010) studied the efficiency analysis of combined cycle Yatağan Thermal Power Plant situated in Yatağan/Muğla for industrial zone using first and second laws of thermodynamics is realized. While analyzing of efficiency of thermal power plant different environmental conditions and loads are observed and considered. Variations of the performance parameters and their magnitudes are studied. The useful power, reversible power and irreversibility are obtained for each component which constitutes the plant, and overall efficiencies of the plant are calculated. The results obtained from the exergy analyses for a steam cycle system predict the plant efficiency more precisely.

Ganapathy et al. (2009) performed exergy analysis on an operating 50MWe unit of lignite fired steam power plant at Thermal Power Station-I, Neyveli Lignite Corporation Limited, Neyveli, Tamil Nadu, India. The exergy losses occurred in the various subsystems of the plant and their components have been calculated using the mass, energy and exergy balance equations. The distribution of the exergy losses in several plant components during the real time plant running conditions has been assessed to locate the process of irreversibility. The First law efficiency and the Second law efficiency of the plant have also been calculated. The comparison between the energy losses and the exergy losses of the individual components of the plant shows that the maximum energy losses of 39% occur in the condenser, whereas the maximum exergy losses of 42.73% occur in the combustor. The real losses of energy which has a scope for the improvement are given as a maximum exergy losses that occurred in the combustor.

Rashad et al. (2009) investigated the energy and exergy analysis of Shobra El-Khima power plant in Cairo, Egypt. The primary objectives of this study are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load. The performance of the plant was estimated by a component-wise modeling and a detailed break-up of energy and exergy losses for the considered plant which has been presented at different loads (Maximum load, 75% load and, 50 % load). Energy losses mainly occurred in the condenser where (404.653 MW at Max load, 306.747 MW at 75% load and 278.849 MW at 50% load) is lost to the environment. The percentage ratio

of the exergy destruction to the total exergy destruction was found to be maximum in the turbine system (42% at Max load, 59% at 75% load and 46.1 at 50% load) followed by the condenser (28% at Max load, 20.3% at 75% load) while at 50% load the feed water heaters cause more exergy destruction (27.7%) than condenser (23.8%) and then the feed water heaters cause exergy destruction 20.8% at max load and 12.1 at 75% load. In addition, the calculated thermal efficiency based on the specific heat input to the steam was 143% while the exergy efficiency of the power cycle was (44% - 48%).

Şen (2006) studied exergy analysis of 320 MW of Çan thermal power plant which is located in Çanakkale. In his study, exergy analysis based on the second law of thermodynamics one of the fluidized bed boiler technology thermal power plant where the coal with low calorie value cleanly and efficiently. The useful power, reversible power and irreversibility component are obtained for each constitutes the plant, and overall efficiencies of the plant are also calculated. The results and evaluations are organized important data of design and economical for thermal power plants construction and operation.

By Erduranlı (1997) energy and exergy analysis have been applied to an existing power plant using real operating parameters. Total input and output energy and exergy and exergy losses have been determined for each component by applying energy and exergy analysis. The exergy losses are considered here due to flow, combustion, heat transfer, and stack gases. The energy and exergy losses were compared with each other for each unit. It was concluded that greatest energy loss occurs at condenser based on energy analysis and that greatest exergy loss occurs at boiler based on exergy analysis.

By Al-Bagawi (1994) the design and actual performance of Ghazlan power plant have been studied based on first and second law analysis. First and second law analysis applied on power plant equipments using computer programmer. A parametric study is conducted for the plant under various operating conditions, including different throttle steam and pressure and temperature, flame temperatures and number of heaters is examined. The results are compared with each other.

3. MATERIAL AND METHOD

3.1. Energy Analysis

Energy analysis is the traditional method of assessing the way energy is used in an operation involving the physical or chemical processing of materials and transfer and/or conversion of energy. This usually entails performing energy balances, which are based on the FLT, and evaluating energy efficiencies. This balance is employed to determine and reduce waste exergy emissions like heat losses and sometimes to enhance waste and heat recovery. However, an energy balance provides no information on the degradation of energy or resources during a process and does not quantify the usefulness or quality of the various energy and material streams flowing through a system and exiting as products and wastes.

Energy values of heat and work flows are absolute, while the energy values of material flows are relative. Enthalpies are evaluated relative to a reference level. Since energy analysis is typically concerned only with energy differences, the reference level used for enthalpy calculations can be arbitrary. For the determination of some energy efficiencies, however, the enthalpies must be evaluated relative to specific reference levels (Dinçer and Rosen, 2011).

3.1.1. First law of Thermodynamics

The first law of thermodynamics is the law of the conservation of energy, which states that, although energy can change form from one to another but it can be neither created nor destroyed. Internal energy is defined as a state function and provides a formal statement of the conservation of energy. Therefore, every bit of energy must be accounted during the process (Çengel and Boles, 1996). The total amount of energy is conserved in all transformations and transfers (Moran and Shapiro, 1995).

3.1.2. Steady Flow for Control Volumes

Components of the thermal power plant which are subject of the present work are examined with the adoption of control volumes. Control volume schematic is shown in Figure 3.1. The following assumptions can be made about control volumes.

- The boundary work is zero for steady-flow system
- During a steady-flow process, no intensive or extensive properties within the control volume change with time. Thus, the volume V , the mass m , and the total energy E of the control volume remain constant.
- The fluid properties at inlet or exit remain constant during the steady flow process.
- The heat and work interactions between a steady-flow system and its surroundings do not change with time (Şen, 2006).

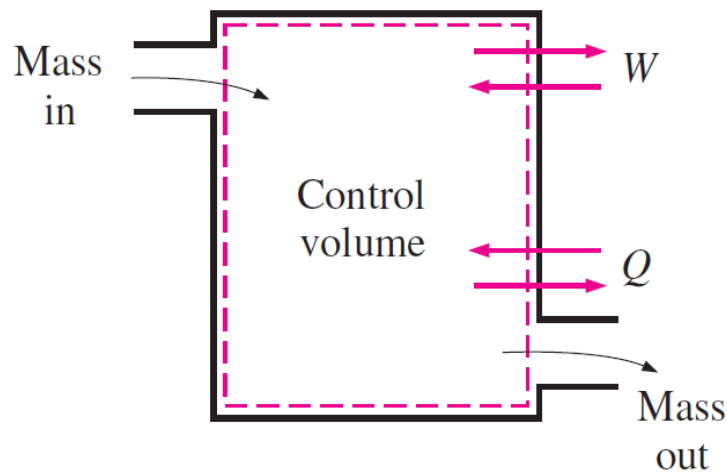


Figure 3.1. Control volume (Çengel and Boles, 2006)

3.1.3. Mass Balance for Steady Flow Process

During a steady-flow process, the total amount of mass contained within control volume does not change with time. Then the conservation of mass principle

requires that the total amount of mass entering the control volume equal to the total mass leaving the control volume.

(Total mass entering the CV during specified Δt) = (Total mass leaving the CV during specified Δt)

Or,

$$\sum m_{in} = \sum m_{out} \quad (3.1)$$

Where m_{in} and m_{out} are the total rates of mass flow into and out of control volume which has a unit of kg/s (Unal, 2009).

3.1.4. Energy Balance for Steady Flow Process

During a steady-flow process, the total energy content of control volume remains constant and thus the change in the total energy of control volume is zero. Therefore, the amount of energy entering a control volume in all forms such as heat, work, and mass must be equal to the amount of energy leaving the control volume.

Energy balance:

(Rate of net energy transfer in) = (Rate of net energy transfer out)

Or

$$E_{in} = E_{out} \quad (\text{kW}) \quad (3.2)$$

$$Q_{in} + W_{in} + \sum_{in} m \theta = Q_{out} + W_{out} + \sum_{out} m \theta \quad (3.3)$$

Since the total energy of flowing fluid per unit mass is

$$\theta = h + ke + pe \quad (3.4)$$

Then,

$$Q - W = m \left[h_{out} - h_{in} + \frac{V_{out}^2 - V_{in}^2}{2} + g(z_{out} - z_{in}) \right] \quad (3.5)$$

$$Q - W = m(\Delta h + \Delta ke + \Delta pe) \quad (3.6)$$

Q = rate of heat transfer between the control volume and its surroundings.

W = power

$$\Delta h = h_{out} - h_{in}$$

$$\Delta ke = \frac{V_{out}^2 - V_{in}^2}{2}$$

$$\Delta pe = g(z_{out} - z_{in}) \quad (\text{Çengel and Boles, 1996})$$

3.2. Exergy Analysis

Regulagadda et al. (2010) indicated that exergy analysis will characterize the work potential of a system. Exergy is the maximum work that can be obtained from the system, when the system state is brought to the dead state. Exergy description is shown in Figure 3.2.

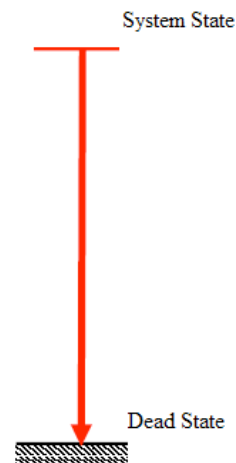


Figure 3.2. Exergy description

In the exergy analysis of this study, the properties at the dead state were denoted by subscript zero. For instance P_0 and T_0 refer to the dead-state pressure and

temperature, respectively. T_0 was assumed to be 25°C (298 K) and P_0 was assumed to be 1bar.

The method of exergy analysis overcomes the limitations of the FLT. The concept of exergy is based on both the FLT and the SLT. Exergy analysis clearly indicates the locations of energy waste in a process and can therefore lead to improved operation or technology. Exergy analysis can also quantify the quality of heat in a waste stream. A main aim of exergy analysis is to identify exergy efficiencies and true magnitudes of exergy losses (Dinçer and Rosen, 2011). Exergy analysis provides those tools and it helps in locating weak spots in the process. This analysis provides a quantitative measure of the quality of the energy in terms of its ability to perform work and leads to a more rational use of energy.

The exergy analysis will be carried out for each component in the subsystems, to evaluate the exergy losses in the individual component and then the analysis is performed on the overall individual subsystems, to find out the exergy losses in each subsystem. Finally the exergy analysis for the overall plant will be carried out and the second law efficiency will be computed.

3.2.1. Second law of Thermodynamics

The second law puts a limitation on the conversion of heat to work. Work can always be converted to heat; however, heat can not always be converted to work. The portion of heat that can not be converted to work is called unavailable energy. It must be rejected as a low-grade heat after work is generated. The second law states that the thermal efficiency of converting heat to work, in a power plant, must be less than 100 percent (Kiameh, 2002).

The second-law analysis is useful to identify the components having maximum irreversibility thus enables proper selection of the process for maintaining high quality of energy. (Eskin et al, 2009)

Quantitative measure of disorder is called entropy of the system at the microscopic level. Entropy generation of the system is as follows:

$$S_{gen} = \sum_{out} ms - \sum_{in} ms - \frac{Q_l}{T_0} \quad (3.7)$$

S_{gen} : Entropy generation of system

S_{in}, S_{out} : Inlet and outlet entropies.

Q_l : Heat transfer with surroundings

T_0 : Ambient temperature (Şen, 2006)

3.2.2. Exergy Balance for Steady flow Process

The exergy balance relation can be stated as the rate of exergy change within the control volume during the process is equal to the rate of net exergy transfer through the control volume boundary by heat, work, and mass flow minus the rate of exergy loss within the boundaries of the control volume.

$$X_{heat} + X_{work} + X_{mass,in} - X_{mass,out} - X_{loss} = X_{out} - X_{in} \quad (3.8)$$

Or,

$$\sum \left(1 - \frac{T_0}{T_k} \right) Q_l - W + \sum_{in} m \psi - \sum_{out} m \psi - X_{loss} = 0 \quad (3.9)$$

Where the subscripts 1 and 2 represent inlet and exit states, m is the mass flow rate, T_0 is the ambient temperature, T_k is the surrounding temperature and the change in the flow exergy is

$$\psi_{out} - \psi_{in} = h_{out} - h_{in} - T_0(s_{out} - s_{in}) + \left(\frac{V_{out}^2 - V_{in}^2}{2} \right) + g(z_{out} - z_{in}) \quad (3.10)$$

$$X = m(\psi_{out} - \psi_{in}) \quad (3.11)$$

3.2.3. Reversible Work, Irreversibility and Availability

Reversible work is the maximum work that can be achieved with the given system interacting with environment at T_0 .

$$\psi_{out} - \psi_{in} = h_{out} - h_{in} - T_0 (s_{out} - s_{in}) + \left(\frac{V_{out}^2 - V_{in}^2}{2} \right) + g(z_{out} - z_{in}) \quad (3.12)$$

$$W_{rev} = m(\psi_{out} - \psi_{in}) \quad (3.13)$$

As the real process produces an amount of work that is less than the ideal reversible work, the difference between these terms is defined as the irreversibility for the control volume. There are many sources of irreversibility in nature. The most important ones are friction, heat transfer, throttling, and mixing (Kiameh, 2002). Irreversibility is directly proportional to the entropy generation inside the control volume. The irreversibility would be zero for a completely reversible process and otherwise is always greater than zero (Çengel and Boles, 2006).

Irreversibility,

$$I = T_0 S_{gen} \quad (3.14)$$

Or,

$$I = W_{rev} - W_{in} \quad (3.15)$$

A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment. At the dead state, a system is at the temperature T_0 and pressure P_0 of its environment: it has no kinetic or potential energy relative to

the environment and it does not react with the environment. At the dead state, the useful work potential of a system is zero.

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.

$$\psi = (h - h_0) - T_0(s - s_0) \quad (3.16)$$

Where h, s are enthalpy and entropy values at given state, h_0 , s_0 are enthalpy and entropy values at dead state. T_0 is ambient temperature (Şen, 2006).

3.3. First Law and Second Law Efficiencies

Efficiency is one of the most frequently used terms in thermodynamics and it indicates how well energy conversion or transfer process is accomplished. Performance or efficiency, in general can be expressed in terms of the desired output and the required input as

$$\text{Efficiency} = \frac{\text{desired output}}{\text{required input}} \quad (3.17)$$

$$\eta_i = \frac{W_{net}}{Q_{in}} \quad (3.18)$$

The first law efficiency makes no reference to best possible performance and thus the first law efficiency alone is not a realistic measure of performance. To overcome this deficiency, second law efficiency should be calculated, which is measure of actual performance relative to the best possible performance under the same conditions. The second law efficiency is defined as ratio of the actual thermal efficiency to the maximum possible thermal efficiency under same conditions.

For turbines,

$$\eta_u = \frac{T_0 S_{gen}}{(\psi_1 - \psi_2)} \quad (3.19) \quad \eta_u = \frac{W_{out}}{W_{rev}} \quad (3.20)$$

For pumps,

$$\eta_u = 1 - \frac{T_0 S_{gen}}{W_{in}} \quad (3.21) \quad \eta_u = \frac{W_{rev}}{W_{in}} \quad (3.22)$$

Where $S_{gen} = m (s_2 - s_1)$

For mixing chamber,

$$\eta_u = 1 - \frac{T_0 S_{gen}}{m_1 \psi_1 + m_2 \psi_2} \quad (3.23)$$

$$\eta_u = \frac{m_3 \psi_3}{m_1 \psi_1 + m_2 \psi_2} \quad (3.24)$$

Where $m_3 = m_1 + m_2$ and $S_{gen} = m_3 s_3 - m_2 s_2 - m_1 s_1$

For heat exchanger,

$$\eta_u = 1 - \frac{T_0 S_{gen}}{m_{hot} (\psi_1 - \psi_2)} \quad (3.25)$$

For overall power plant,

$$\eta_u = 1 - \frac{W_{net}}{X_{in}} \quad (3.26)$$

3.4. Fuel and Combustion Analysis

Ünal (2009) state that boiler needs a heat source which has an enough temperature to generates high pressure and high temperature steam. For this reason, generally energy is obtained with burning fossil fuel in the boiler. The main constituent of coal is carbon. Coal also contains varying amounts of oxygen, hydrogen, nitrogen, sulfur, moisture, and ash. Coal components are shown in Table 3.1. In this study, high calorific imported coal usage is considered and also coal components are demonstrated below. Lower heating value of coal is defined as 25,800 kJ/kg.

Table 3.1. Coal components

Coal Components (% ratio)						
C	H	N	S	O	Ash	Water
66	3.66	1.5	0.9	4.16	13.78	10

A chemical reaction during which a fuel is oxidized and a large quantity of energy is released is called combustion. A fuel is said to have burned completely if all carbon present in the fuel is burned to carbon dioxide, all the hydrogen is burned to water and all the sulfur is burned to sulfur dioxide. During combustion, nitrogen behaves as an inert gas and does not react with other elements, other than forming a very small amount of nitric oxides.

Moran and Shapiro (1995) reported that oxygen is required in every combustion process. Pure oxygen is used only in special application but in this study air is used for combustion. Air is considered to be % 21 oxygen and % 79 nitrogen on a molar basis. With this idealization the molar ratio of the nitrogen to the oxygen is $0.79/0.21 = 3.76$. When air supplies is accompanied by 3.76 moles of nitrogen. Combustion process is shown in Figure 3.3.

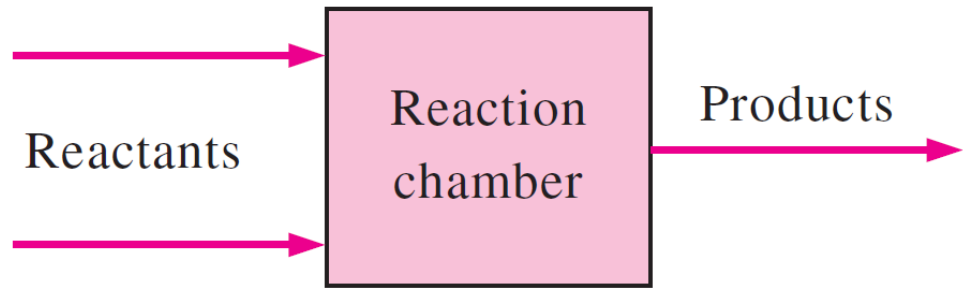
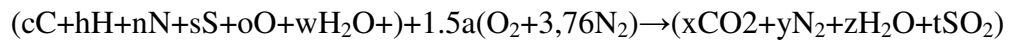


Figure 3.3. Combustion process (Çengel and Boles, 2006)

Combustion process in this study,



Balancing has been applied for this study. Coefficients reactants and products are shown in Table 3.2.

Table 3.2. Coefficient of reactants and products for 100 kg of coal

Coefficient	Value
c	5.5
h	3.66
n	0.106
s	0.028
o	0.26
w	0.55
a	4.208
x	5.5
y	23.79
z	2.38
t	0.028

Molar mass of coal,

$$M_f = \frac{cC + hH + nN + oO + wH_2O}{c + h + n + o + w}$$

$$M_f = 8.52$$

Molar mass of burning air,

$$M_{air} = \frac{1.5a(O_2 + 3.76 N_2)}{1.5(1 + 3.76)}$$

$$M_{air} = 28.84$$

Molar mass of flue gas,

$$M_{fg} = \frac{xCO_2 + yN_2 + zH_2O + tSO_2}{x + y + z + t}$$

$$M_{fg} = 29.90$$

Required minimum amount of oxygen for complete combustion is calculated using this equation;

$$\text{Amount of oxygen} = \%C.2.664 + \%H.7.936 + \%S.0.998 - \%O_2$$

The amount of air required for the combustion of fuel in the minimum amount of air depending on the minimum amount of oxygen must be present. For this, the oxygen mole weight divided by the weight of air and oxygen in the air mass flow rate mole multiplied.

Air is to be considered 21%,

$$0,21 \frac{M_{O_2}}{M_{air}} = 0.233$$

$$\text{Amount of air per kg of coal} = \frac{\text{Amount of oxygen}}{0.233}$$

Excess air coefficient is assumed $\lambda = 1.2$

$$RAA = MAA. \lambda \quad (3.27)$$

Flue gas flow rate can be calculated after finding required amount of combustion air.

As a result of combustion of 1 kg of coal, flue gas flow rate is $1 + RAA$ kg.

Coal energy supplied to boiler,

$$Q_f = m_f h_f \quad (3.28)$$

Flue gas enthalpy,

Molar ratio components of flue gases are,

$$x = \frac{n_i}{n_{total}} \quad (3.29)$$

$$x_{CO_2} = 0.183 \quad x_{N_2} = 0.795 \quad x_{SO_2} = 0.183 \quad x_{H_2O} = 0.0795$$

Enthalpy of flue gases can be calculated at desired temperature using these equations,

$$\bar{h}_{fg} = x_{CO_2} \bar{h}_{CO_2} + x_{N_2} \bar{h}_{N_2} + x_{SO_2} \bar{h}_{SO_2} + x_{H_2O} \bar{h}_{H_2O} \quad (3.30)$$

$$h_{fg} = \frac{\bar{h}_{fg}}{M_{fg}} \quad (3.31)$$

Entropy of components of flue gases,

$$\bar{s}_{CO_2} = \bar{s}_{K,CO_2} - R \ln x_{CO_2} \quad (3.32)$$

$$\bar{s}_{N_2} = \bar{s}_{K,N_2} - R \ln x_{N_2} \quad (3.33)$$

$$\bar{s}_{SO_2} = \bar{s}_{K,SO_2} - R \ln x_{SO_2} \quad (3.34)$$

$$\bar{s}_{H_2O} = \bar{s}_{K,H_2O} - R \ln x_{H_2O} \quad (3.35)$$

$$\bar{s}_{fg} = x_{CO_2} \cdot \bar{s}_{CO_2} + x_{N_2} \cdot \bar{s}_{N_2} + x_{SO_2} \cdot \bar{s}_{SO_2} + x_{H_2O} \cdot \bar{s}_{H_2O} \quad (3.36)$$

$$s_{fg} = \frac{\bar{s}_{fg}}{M_{fg}} \quad (3.37)$$

Chemical exergy of flue gas and combustion air formula represented below respectively.

$$\begin{aligned} \bar{\psi}_{fg}^{ch} &= x_{CO_2} \cdot \bar{\psi}_{CO_2} + x_{N_2} \cdot \bar{\psi}_{N_2} + x_{SO_2} \cdot \bar{\psi}_{SO_2} + x_{H_2O} \cdot \bar{\psi}_{H_2O} \\ &+ RT_0 (x_{CO_2} \ln x_{CO_2} + x_{N_2} \ln x_{N_2} + x_{SO_2} \ln x_{SO_2} + x_{H_2O} \ln x_{H_2O}) \end{aligned} \quad (3.38)$$

$$\psi_{fg}^{ch} = \frac{\bar{\psi}_{fg}^{ch}}{M_{fg}} \quad (3.39)$$

$$X_{fg}^{ch} = m_{fg} \psi_{fg}^{ch} \quad (3.40)$$

$$\bar{\psi}_{air}^{ch} = x_{O_2} \cdot \bar{\psi}_{O_2} + x_{N_2} \cdot \bar{\psi}_{N_2} + RT_0 (x_{O_2} \ln x_{O_2} + x_{N_2} \ln x_{N_2}) \quad (3.41)$$

$$\psi_{air}^{ch} = \frac{\bar{\psi}_{air}^{ch}}{M_{air}} \quad (3.42)$$

$$X_{air}^{ch} = m_{air} \psi_{air}^{ch} \quad (3.43)$$

Coal chemical exergy,

$$\Phi = 1.0437 + 0.1882 \frac{h}{c} + 0.0610 \frac{o}{c} + 0.0404 \frac{n}{c} \quad (3.44)$$

$$\psi_f = \Phi \left(h_f + \frac{w \cdot 2467.3}{4.18} \right) \quad (3.45)$$

$$X_f = m_f \psi_f \quad (3.46)$$

Table 3.3. Standard chemical exergy of components (Geredelioğlu, 2011)

Components	ψ^{ch} kJ/kmol
O ₂	2970
CO ₂	19870
N ₂	720
SO ₂	313400
H ₂ O	9500

3.5. Plant Description



Figure 3.4. The steam power plant

Steam power plant is located in Adana and the steam power plant is shown in Figure 3.4. The power plant has two units and it operates on rankine cycle. It consists of pumps, turbines, boiler, condensers and heaters also includes flue gas desulfurization unit and electrostatic filters. In addition to supplying reliable, efficient and competitive power to the national grid, environmental protection.

Most important equipments for power plants are turbines. The power plant has one HP turbine, one IP turbine and two LP turbines. HP turbine is a single flow and has 13 stages, IP is double flow and has three uncontrolled extraction points and 14 stages at each flow side, LP turbines are double flow and each LP turbines have three uncontrolled extraction points and 14 stages at each flow side. LP turbines moving blades is shown in Figure 3.5. All turbines directly couple the generator which rotates at 3000 rpm (50Hz) and gross outlet power is 660 MW and net power is 605 at full load for each unit.



Figure 3.5. LP turbine moving blades

Boiler is defined as one pass tower and subcritical. Boiler includes heating packages, they are evaporators, superheaters, economizer and reheaters. Solid model of boiler is shown in Figure 3.6. Economizer is parallel flow and placed top of the boiler. It has been estimated that an 6 °C and 7 °C in temperature of the feedwater produced from the heat recovery in the economizer. Feed water enters to evaporator after economizer. The feedwater in the evaporator tubes receives heat from combustion gases and boils further. In exit of evaporator tubes evaporating completed. There are three superheater packages. Superheaters have heat transfer surfaces in which heat is transferred to the saturated steam to increase its temperature and available energy. Superheaters are placed above banks of water tubes to protect them from combustion flames and high temperature. Radiation and convection heat transfer are occurred. After superheater steam becomes superheated steam and goes to the HP turbine. There are two reheater packages. Exhaust steam of HP turbine returns to reheaters for additional superheat before it is enters to the IP turbine. When the exit steam temperature become excessive, some unit employ attemporator in which compressed feedwater is sprayed into superheated steam. Imported coal is used as a main fuel. It is pulverized by mill and conveyed pneumatically to the pulverized coal burners. Pulverized coal burner is shown in Figure 3.7. But oil is

used as a fuel in start-up and shut-down conditions. Also there is soot blowing system to reduce fouling around pipe.

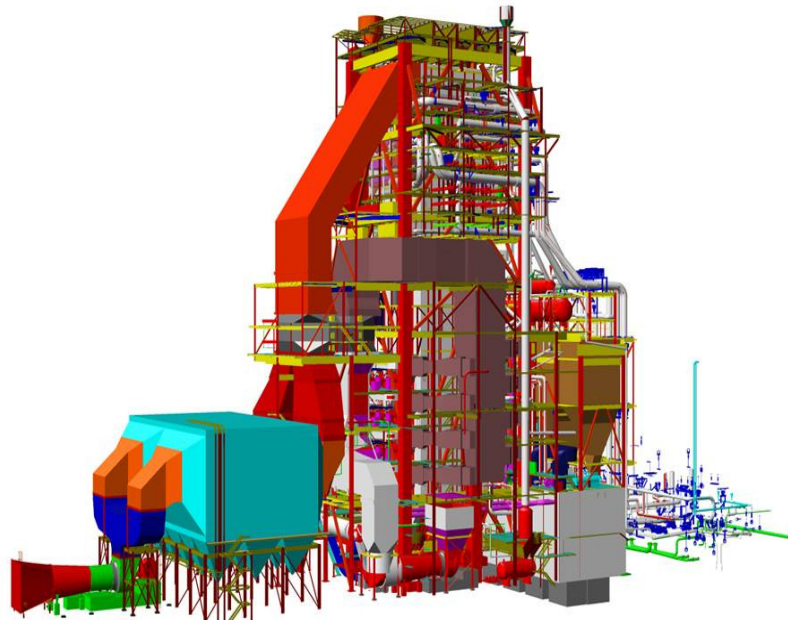


Figure 3.6. Solid model of boiler

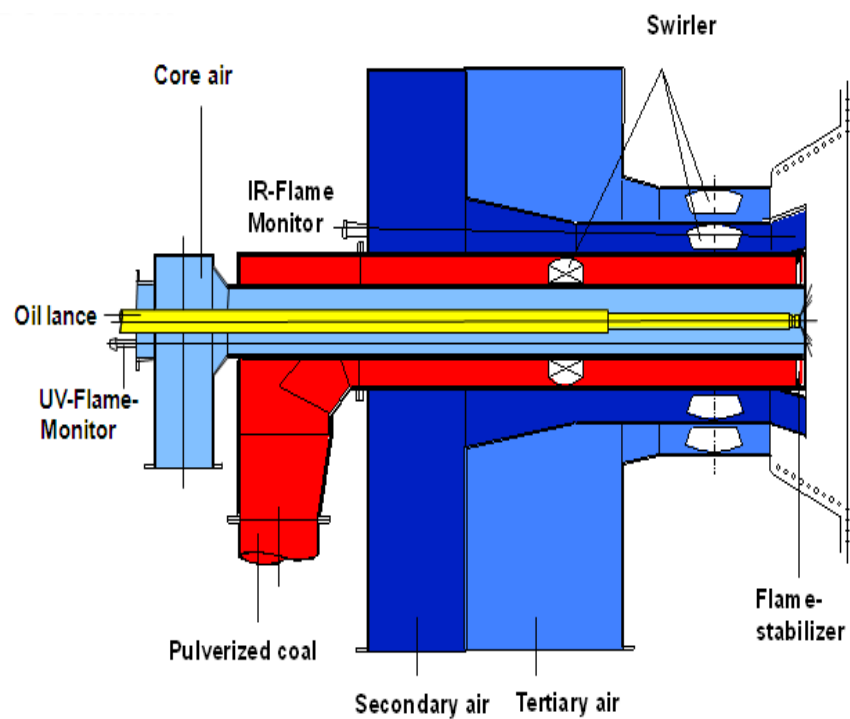


Figure 3.7. Pulverized coal burner

Air preheater is a regenerative heat exchanger installed exit of boiler. Air preheater transfers thermal energy from flue gas to cold combustion air. Preheated air accelerates combustion by producing more rapid ignition and increase plant efficiency reducing fuel consumption.

Feed water tank is mixing chamber installed between condenser and boiler, It is used to remove the air from condensed water. This is necessary because the air in the water produces corrosion of the pipes. Besides condensate water is heated by steam which is coming from IP extraction.

LP and HP heaters are shell and tube exchangers, plant have four LP heaters and two HP heaters. The temperature of condensate and feed water increases steam gradually across the heaters using extraction steam so that temperature of water which goes to boiler increases to higher value. In addition, Heaters recover the latent heat of steam. HP heater is shown in Figure 3.8.

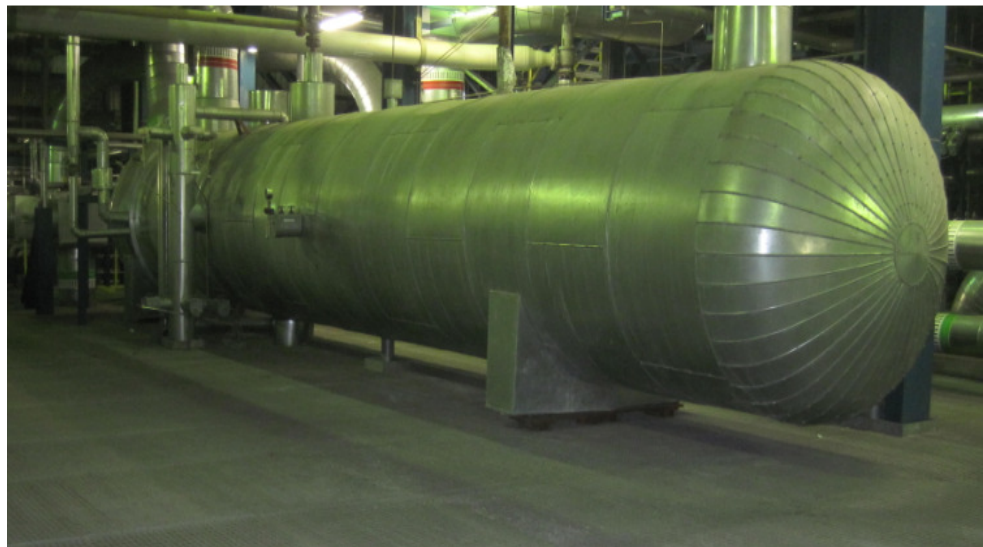


Figure 3.8. HP heater

Condenser is a shell and tube heat exchanger where exhaust steam is condensed in tubes which are cooled by sea water therefore latent heat of steam is transferred to sea water. Condenser pressure is close to perfect vacuum, it should be arranged well because it does directly affect the power plant efficiency.

Condenser extraction pumps are used to pump condensate water from condenser to feed water tank through LP heaters. Boiler feed pumps are used to pump feed water from feed water tank to boiler through HP heaters. Both of them have six stages. Condenser extraction pump is shown in Figure 3.9.



Figure 3.9. Condenser extraction pump

A brief description of the plant operations is as follows: The process of steam power comprises the vaporization of the feed water and the superheat of the operating steam with certain pressure and certain temperature by heat supply unit in the boiler. Afterwards the expansion of the steam happens in the steam turbine. The kinetic energy of the steam is converted in the turbine blade working area into mechanical energy and hence shaft work is obtained. The exhaust steam is then condensed in the condenser by sea water. On this occasion the heat of evaporation of the exhaust steam is removed by the cooling medium. This condensate water is fed to the feed water tank aid with condensate extraction pumps through LP heaters and then fed to the boiler by boiler feed pumps through HP heaters. The water-steam cycle of power plant is shown in Figure 3.10.

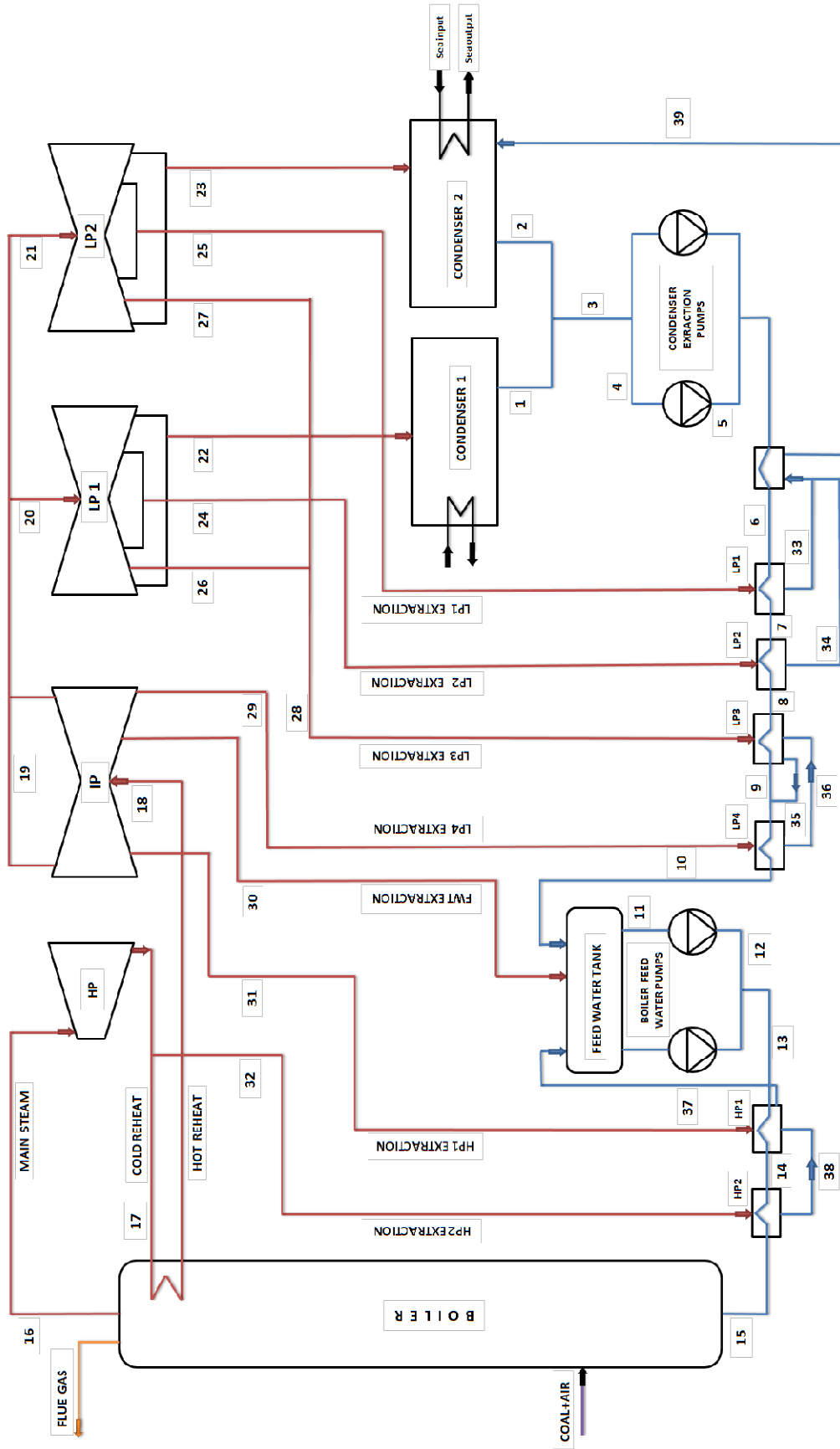


Figure 3.10. The water-steam cycle of power plant

Ideal T-s diagram of power plant is shown in Figure 3.11.

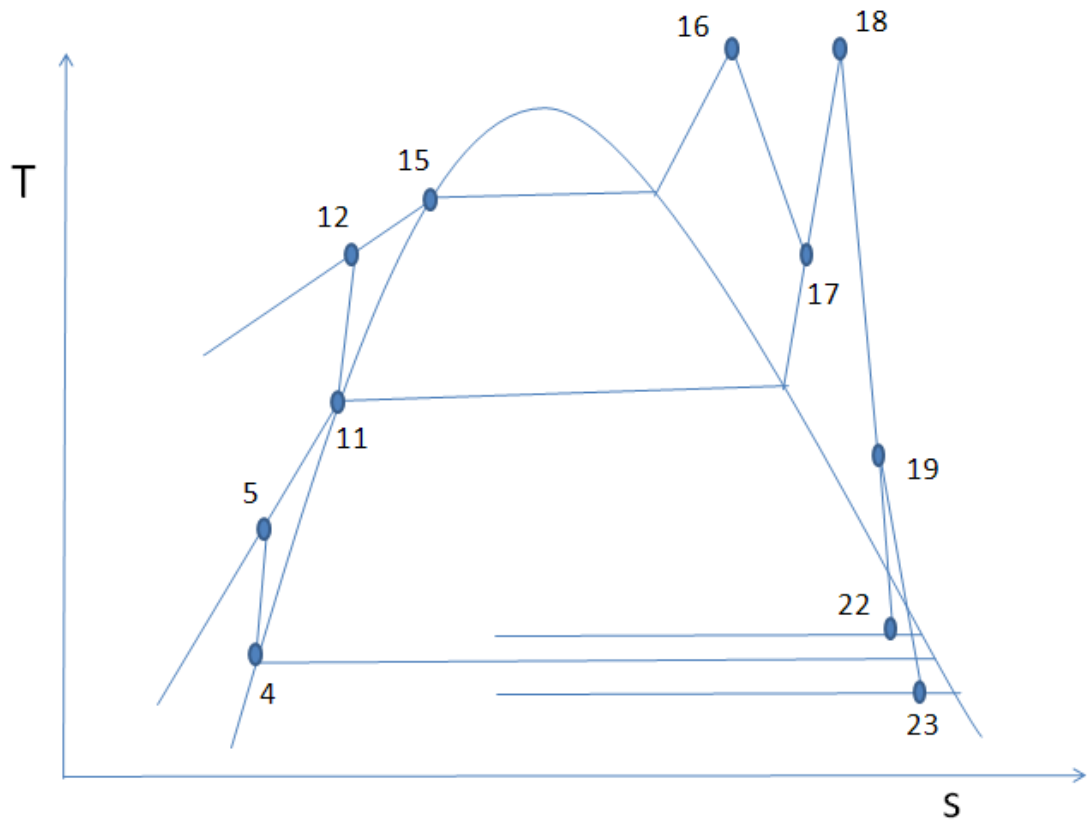


Figure 3.11. T-s diagram of the power plant

4. RESULTS AND DISCUSSION

4.1. Calculations for 100 % load

Table 4.1. Thermodynamic properties for each stream at 100 % load

%100 Load						
No	Statement	Temperature	Pressure	Flow Rate	Enthalpy	Entropy
		T(°C)	P(bar)	m(kg/s)	h(kj/kg)	s(kj/kg.K)
1	water	28,6	0,0391	161,7	119,8	0,417
2	water	31,8	0,047	210,2	133,2	0,461
3	water	30,4	0,042	371,9	127,4	0,441
4	water	30,4	0,042	185,95	127,4	0,441
5	water	30,7	25	185,95	130,94	0,445
6	water	36	14,59	371,9	152,12	0,518
7	water	55,9	14,1	371,9	235,18	0,778
8	water	89,6	13,61	371,9	376,28	1,187
9	water	123,2	13,12	418,4	518,15	1,561
10	water	153	12,63	418,4	650	1,88
11	water	182,7	10,7	262,1	775,13	2,165
12	water	187,2	229	262,1	805,8	2,179
13	water	187,2	229	524,2	805,8	2,179
14	water	219,4	228	524,2	947,7	2,476
15	water	268	227	524,2	1169,5	2,911
16	steam	541	185	524,2	3387,2	6,359
17	steam	351,2	53,26	466,5	3064,5	6,417
18	steam	538	49	466,5	3523,9	7,099
19	steam	243,6	5,95	395,4	2944,3	7,161
20	steam	243,6	5,95	197,7	2944,3	7,161
21	steam	243,6	5,95	197,7	2944,3	7,161
22	mix	26,6	0,039	161,7	2304,4	7,5
23	mix	31,8	0,047	169,2	2312,6	7,47
24	mix	92,8	0,78	24	2614,4	7,317

25	mix	59	0,19	17	2433,7	7,427
26	steam	157,5	2,45	12	2781,31	7,217
27	steam	157,5	2,45	11,5	2781,31	7,217
28	steam	157,5	2,45	23,5	2781,31	7,217
29	steam	243,6	5,95	23	2944,3	7,161
30	steam	324	11,39	21,5	3100	7,148
31	steam	423,2	23,2	26,6	3294,4	7,129
32	steam	351,2	53,26	56	3064,5	6,417
33	mix	59	0,19	17	248	0,818
34	mix	92,8	0,78	24	388,1	1,222
35	mix			46,5		
36	water	126,2	4	23	530,3	1,594
37	water	190,2	22	84,3	808,9	2,236
38	water	22,4	50	57,7	955,7	2,535
39	water	34,2	0,19	41	143,1	0,494

4.1.1. Condenser Extraction Pump



Figure 4.1. Condenser extraction pump schematic

At pump, inlet flow rate have to be equal to outlet flow rate.

$$m_4 = m_5 = 185.95 \text{ kg/s}$$

Heat transfer is neglected because the pump is assumed to be adiabatic, potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are same.

If the first law is applied for condenser extraction pump,

$$W_{in} = m(h_5 - h_4)$$

Pump work,

$$W_{in} = -658.263 \text{ kW}$$

Reversible pump work,

$$W_{rev} = m_4(h_4 - T_0 s_4) - m_5(h_5 - T_0 s_5)$$

$$W_{rev} = -436.611 \text{ kW}$$

To calculate irreversibility's of pumps, entropy generations have to be calculated.

Entropy generation is calculated using equation (3.7),

$$S_{gen} = m(s_5 - s_4)$$

$$S_{gen} = 0.744 \text{ kW/K}$$

Irreversibility of pumps is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 221.652 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16),

$$\psi_4 = (h_4 - h_0) - T_0(s_4 - s_0)$$

$$\psi_4 = 0.448 \text{ kJ/kg}$$

Exergy at outlet is calculated using equation (3.16),

$$\psi_5 = (h_5 - h_0) - T_0(s_5 - s_0)$$

$$\psi_5 = 2.796 \text{ kJ/kg}$$

Exergy differences inlet and outlet gives reversible power of pump,

$$W_{rev} = m(\psi_4 - \psi_5)$$

$$W_{rev} = 436.611 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{in} - W_{rev}$$

$$I = 221.652 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between inlet work and reversible work at the same time.

The second law efficiency is calculated using equation (3.21),

$$\eta_{II} = \frac{W_{rev}}{W_{in}}$$

$$\eta_{II} = 66.3\%$$

4.1.2. LP 1 Heater

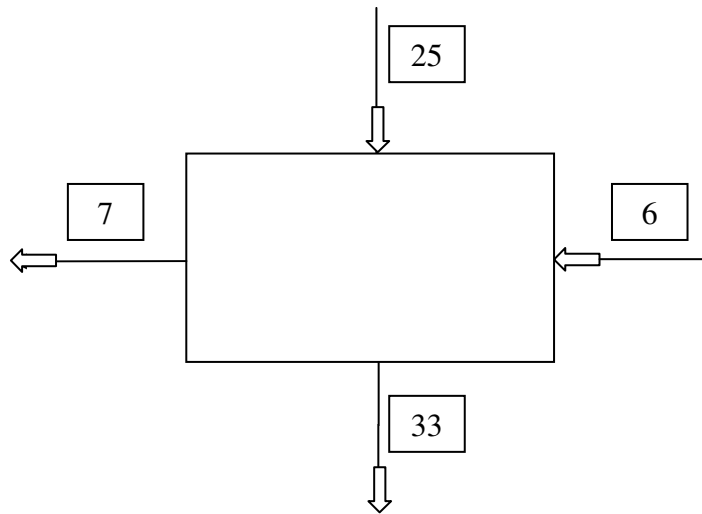


Figure 4.2. LP 1 heater schematic

Inlet and outlet flow rates are the same.

$$m_6 = m_7 = 371.9 \text{ kg/s} \quad m_{25} = m_{33} = 17 \text{ kg/s}$$

If the first law is applied for LP 1 heater, Q_1 is found.

$$Q_1 = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_6 h_6 + m_{25} h_{25}$$

$$Q_{out} = m_7 h_7 + m_{33} h_{33}$$

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

$$Q_l = 6266.8 \text{ kW}$$

Entropy generation of LP 1 heater is calculated using equation (3.7)

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_7 s_7 + m_{33} s_{33}) - (m_6 s_6 + m_{25} s_{25}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 5.37 \text{ kW/K}$$

Irreversibility of LP 1 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 1600 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16),

$$\psi_6 = (h_6 - h_0) - T_0 (s_6 - s_0)$$

$$\psi_6 = 2.222 \text{ kJ/kg}$$

$$\psi_{25} = (h_{25} - h_0) - T_0 (s_{25} - s_0)$$

$$\psi_{25} = 224.92 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16),

$$\psi_7 = (h_7 - h_0) - T_0 (s_7 - s_0)$$

$$\psi_7 = 7.802 \text{ kJ/kg}$$

$$\psi_{33} = (h_{33} - h_0) - T_0 (s_{33} - s_0)$$

$$\psi_{33} = 8.702 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_6 \psi_6 + m_{25} \psi_{25}) - (m_7 \psi_7 + m_{33} \psi_{33})$$

$$X_l = 1600 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.25),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{hot} (\psi_{25} - \psi_{33})} \right)$$

$$\eta_{II} = 56.4\%$$

4.1.3. LP 2 Heater

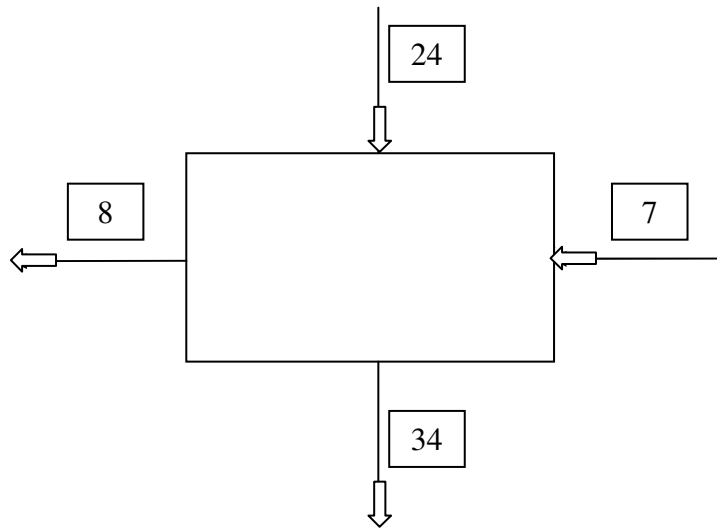


Figure 4.3. LP2 heater schematic

Inlet and outlet flow rates are the same.

$$m_7 = m_8 = 371.9 \text{ kg/s} \quad m_{24} = m_{34} = 24 \text{ kg/s}$$

If the first law is applied for LP 2 heater, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_7 h_7 + m_{24} h_{24}$$

$$Q_{out} = m_8 h_8 + m_{34} h_{34}$$

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

$$Q_l = 956.11 \text{ kW}$$

Entropy generation of LP 2 heater is calculated using equation (3.7)

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_8 s_8 + m_{34} s_{34}) - (m_7 s_7 + m_{24} s_{24}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 9.03 \text{ kW/K}$$

Irreversibility of LP 2 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 2692.58 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16),

$$\psi_7 = (h_7 - h_0) - T_0 (s_7 - s_0)$$

$$\psi_7 = 7.802 \text{ kJ/kg}$$

$$\psi_{24} = (h_{24} - h_0) - T_0 (s_{24} - s_0)$$

$$\psi_{24} = 438.4 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16),

$$\psi_8 = (h_8 - h_0) - T_0 (s_8 - s_0)$$

$$\psi_8 = 27.02 \text{ kJ/kg}$$

$$\psi_{34} = (h_{34} - h_0) - T_0 (s_{34} - s_0)$$

$$\psi_{34} = 28.41 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_7 \psi_7 + m_{24} \psi_{24}) - (m_8 \psi_8 + m_{34} \psi_{34})$$

$$X_l = 2692.58 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.25),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{hot} (\psi_{24} - \psi_{34})} \right)$$

$$\eta_{II} = 72.6\%$$

4.1.4. LP 3 Heater

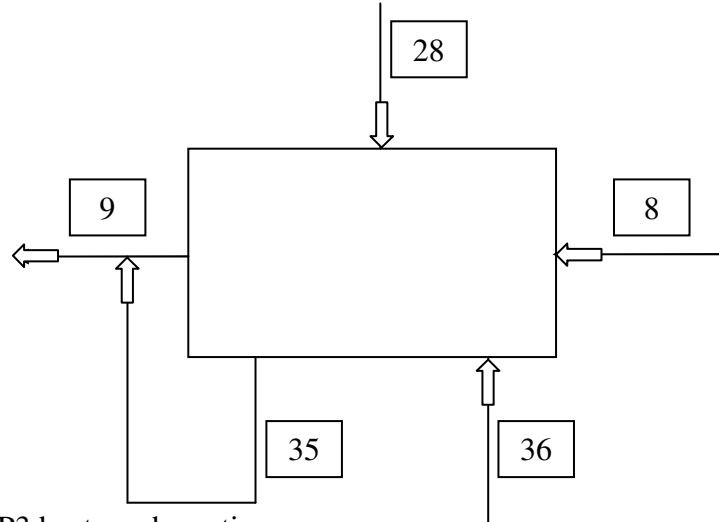


Figure 4.4. LP3 heater schematic

Inlet and outlet flow rates are the same.

$$m_9 = m_8 + m_{35} = 418.4 \text{ kg/s} \quad m_{35} = m_{28} + m_{36} = 46.5 \text{ kg/s}$$

If the first law is applied for LP 3 heater, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_8 h_8 + m_{28} h_{28} + m_{36} h_{36}$$

$$Q_{out} = m_9 h_9$$

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

$$Q_l = 702.25 \text{ kW}$$

Entropy generation of LP 3 heater is calculated using equation (3.7)

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_9 s_9) - (m_8 s_8 + m_{28} s_{28} + m_{36} s_{36}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 7.77 \text{ kW/K}$$

Irreversibility of LP 3 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 2316 .1 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16),

$$\psi_8 = (h_8 - h_0) - T_0 (s_8 - s_0)$$

$$\psi_8 = 27 .02 \text{ kJ/kg}$$

$$\psi_{28} = (h_{28} - h_0) - T_0 (s_{28} - s_0)$$

$$\psi_{28} = 635 .2 \text{ kJ/kg}$$

$$\psi_{36} = (h_{36} - h_0) - T_0 (s_{36} - s_0)$$

$$\psi_{36} = 59 .75 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16),

$$\psi_9 = (h_9 - h_0) - T_0 (s_9 - s_0)$$

$$\psi_9 = 57 .43 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_8 \psi_8 + m_{28} \psi_{28} + m_{36} \psi_{36}) - (m_9 \psi_9)$$

$$X_l = 2316 .1 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_8 \psi_8 + m_{28} \psi_{28} + m_{36} \psi_{36}} \right)$$

$$\eta_{II} = 91 .2\%$$

4.1.5. LP 4 Heater

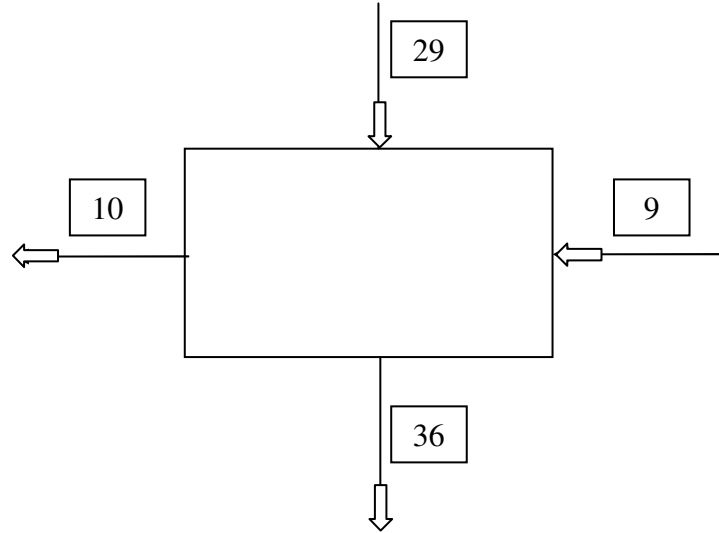


Figure 4.5. LP 4 heater schematic

Inlet and outlet flow rates are the same.

$$m_9 = m_{10} = 418.4 \text{ kg/s} \quad m_{29} = m_{36} = 24 \text{ kg/s}$$

If the first law is applied for LP4 heater, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_9 h_9 + m_{29} h_{29}$$

$$Q_{out} = m_{10} h_{10} + m_{36} h_{36}$$

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

$$Q_l = 355.96 \text{ kW}$$

Entropy generation of LP 4 heater is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_{10} s_{10} + m_{36} s_{36}) - (m_9 s_9 + m_{29} s_{29}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 6.62 \text{ kW/K}$$

Irreversibility of LP 4 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 1973.68 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16),

$$\psi_9 = (h_9 - h_0) - T_0 (s_9 - s_0)$$

$$\psi_9 = 57.43 \text{ kJ/kg}$$

$$\psi_{29} = (h_{29} - h_0) - T_0 (s_{29} - s_0)$$

$$\psi_{29} = 814.7 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16),

$$\psi_{10} = (h_{10} - h_0) - T_0 (s_{10} - s_0)$$

$$\psi_{10} = 94.22 \text{ kJ/kg}$$

$$\psi_{36} = (h_{36} - h_0) - T_0 (s_{36} - s_0)$$

$$\psi_{36} = 59.7 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_9 \psi_9 + m_{29} \psi_{29}) - (m_{10} \psi_{10} + m_{36} \psi_{36})$$

$$X_l = 1973.68 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.25),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{hot} (\psi_{29} - \psi_{36})} \right)$$

$$\eta_{II} = 88.6\%$$

4.1.6. Feedwater Tank

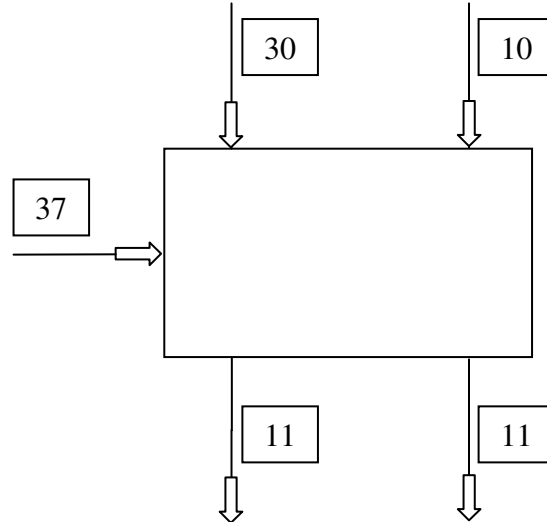


Figure 4. 6. Feedwater tank schematic

Inlet and outlet flow rates are the same.

$$2m_{11} = m_{10} + m_{30} + m_{37} = 524.2 \text{ kg/s}$$

If the first law is applied for feed water tank, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{10} h_{10} + m_{30} h_{30} + m_{37} h_{37}$$

$$Q_{out} = 2m_{11} h_{11}$$

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

$$Q_l = 477.12 \text{ kW}$$

Entropy generation of feed water tank is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = 2(m_{11} s_{11}) - (m_{10} s_{10} + m_{30} s_{30} + m_{37} s_{37}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 7.72 \text{ kW/K}$$

Irreversibility of feed water tank is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 2302.1 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16),

$$\psi_{10} = (h_{10} - h_0) - T_0 (s_{10} - s_0)$$

$$\psi_{10} = 94.22 \text{ kJ/kg}$$

$$\psi_{30} = (h_{30} - h_0) - T_0 (s_{30} - s_0)$$

$$\psi_{30} = 974.362 \text{ kJ/kg}$$

$$\psi_{37} = (h_{37} - h_0) - T_0 (s_{37} - s_0)$$

$$\psi_{37} = 147.038 \text{ kJ/kg}$$

Exergy at outlet is calculated using equation (3.16),

$$\psi_{11} = 2[(h_{11} - h_0) - T_0 (s_{11} - s_0)]$$

$$\psi_{11} = 134.42 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_{10}\psi_{10} + m_{30}\psi_{30} + m_{37}\psi_{37}) - 2(m_{11}\psi_{11})$$

$$X_l = 2302.1 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{10}\psi_{10} + m_{30}\psi_{30} + m_{37}\psi_{37}} \right)$$

$$\eta_{II} = 96.8\%$$

4.1.7. Boiler Feed Pump



Figure 4.7. Boiler feed pump schematic

At pump, inlet flow rate have to be equal to outlet flow rate,

$$m_4 = m_5 = 267.1 \text{ kg/s}$$

Heat transfer is neglected because the pump is assumed to be adiabatic , potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are the same.

If the first law is arranged for condenser extraction pump,

$$W_{in} = m(h_{12} - h_{11})$$

Pump work,

$$W_{in} = -8038.60 \text{ kW}$$

Reversible pump work,

$$W_{rev} = m_{11}(h_{11} - T_0 s_{11}) - m_{12}(h_{12} - T_0 s_{12})$$

$$W_{rev} = -6945.12 \text{ kW}$$

To calculate irreversibility's of pumps, entropy generations have to be calculated.

Entropy generation is calculated using equation (3.7),

$$S_{gen} = m(s_{12} - s_{11})$$

$$S_{gen} = 3.669 \text{ kW/K}$$

Irreversibility of boiler feed pump is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 1093.48 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16),

$$\psi_{11} = (h_{11} - h_0) - T_0 (s_{11} - s_0)$$

$$\psi_{11} = 134.426 \text{ kJ/kg}$$

Exergy at outlet is calculated using equation (3.16),

$$\psi_{12} = (h_{12} - h_0) - T_0 (s_{12} - s_0)$$

$$\psi_{12} = 160.924 \text{ kJ/kg}$$

Exergy differences inlet and outlet gives reversible power of pump,

$$W_{rev} = m(\psi_{11} - \psi_{12})$$

$$W_{rev} = 6848.67 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{in} - W_{rev}$$

$$I = 1093.48 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between inlet work and reversible work at the same time.

The second law efficiency is calculated using equation (3.21),

$$\eta_{II} = \frac{W_{rev}}{W_{in}}$$

$$\eta_{II} = 86.4\%$$

4.1.8. HP 1 Heater

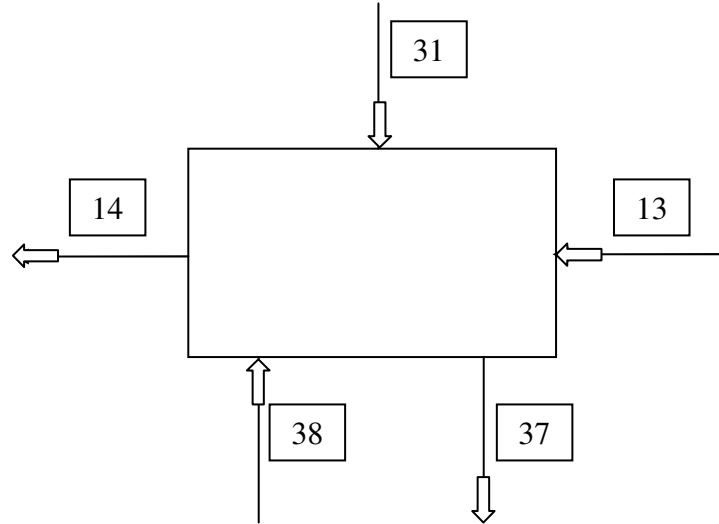


Figure 4.8. HP 1 heater schematic

Inlet and outlet flow rates are the same.

$$m_{13} = m_{14} = 524.2 \text{ kg/s} \quad m_{37} = m_{31} + m_{38} = 84.3 \text{ kg/s}$$

If first law is applied for HP 1 heater, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{13} h_{13} + m_{31} h_{31} + m_{38} h_{38}$$

$$Q_{out} = m_{14} h_{14} + m_{37} h_{37}$$

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

$$Q_l = 200.68 \text{ kW}$$

Entropy generation is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_{13} s_{13} + m_{31} s_{31} + m_{38} s_{38}) - (m_{14} s_{14} + m_{37} s_{37}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 8.95 \text{ kW/K}$$

Irreversibility of HP 1 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 2668.50 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16)

$$\psi_{13} = (h_{13} - h_0) - T_0 (s_{13} - s_0)$$

$$\psi_{13} = 160.924 \text{ kJ/kg}$$

$$\psi_{31} = (h_{31} - h_0) - T_0 (s_{31} - s_0)$$

$$\psi_{31} = 1174.424 \text{ kJ/kg}$$

$$\psi_{38} = (h_{38} - h_0) - T_0 (s_{38} - s_0)$$

$$\psi_{38} = 204.736 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16)

$$\psi_{14} = (h_{14} - h_0) - T_0 (s_{14} - s_0)$$

$$\psi_{14} = 214.318 \text{ kJ/kg}$$

$$\psi_{37} = (h_{37} - h_0) - T_0 (s_{37} - s_0)$$

$$\psi_{37} = 147.038 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_{13}\psi_{13} + m_{31}\psi_{31} + m_{38}\psi_{38}) - (m_{14}\psi_{14} + m_{37}\psi_{37})$$

$$X_l = 2668.50 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{13}\psi_{13} + m_{31}\psi_{31} + m_{38}\psi_{38}} \right)$$

$$\eta_{II} = 97.9\%$$

4.1.9. HP 2 Heater

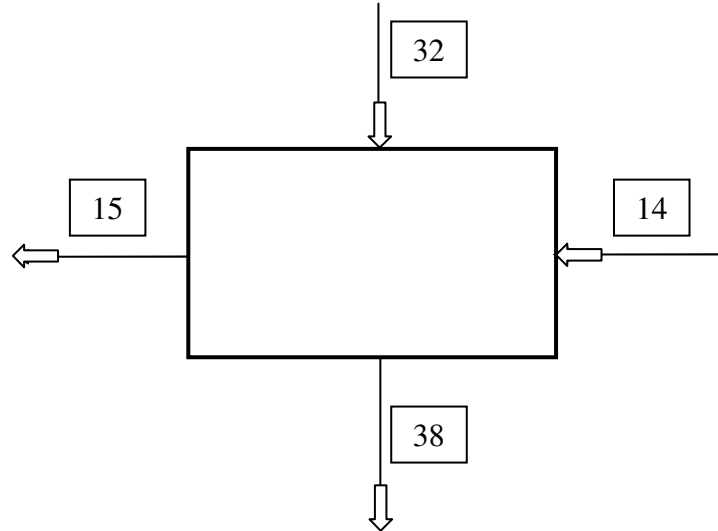


Figure 4.9. HP 2 heater schematic

Inlet and outlet flow rates are the same.

$$m_{14} = m_{15} = 524.2 \text{ kg/s} \quad m_{32} = m_{38} = 56 \text{ kg/s}$$

If the first law is applied for HP 2 heater, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{14} h_{14} + m_{32} h_{32}$$

$$Q_{out} = m_{15} h_{15} + m_{38} h_{38}$$

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

$$Q_l = 200.55 \text{ kW}$$

Entropy generation is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = (m_{15} s_{15} + m_{38} s_{38}) - (m_{14} s_{14} + m_{32} s_{32}) + \frac{Q_l}{T_0}$$

$$S_{gen} = 15.617 \text{ kW/K}$$

Irreversibility of HP 2 heater is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 4654 .011 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16)

$$\psi_{14} = (h_{14} - h_0) - T_0 (s_{14} - s_0)$$

$$\psi_{14} = 214 .318 \text{ kJ/kg}$$

$$\psi_{32} = (h_{32} - h_0) - T_0 (s_{32} - s_0)$$

$$\psi_{32} = 1156 .7 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16)

$$\psi_{15} = (h_{15} - h_0) - T_0 (s_{15} - s_0)$$

$$\psi_{15} = 306 .488 \text{ kJ/kg}$$

$$\psi_{38} = (h_{38} - h_0) - T_0 (s_{38} - s_0)$$

$$\psi_{38} = 204 .736 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_{14} \psi_{14} + m_{32} \psi_{32}) - (m_{15} \psi_{15} + m_{38} \psi_{38})$$

$$X_l = 4666 .38 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.25),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{hot} (\psi_{32} - \psi_{38})} \right)$$

$$\eta_{II} = 97 .4\%$$

4.1.10. HP Turbine

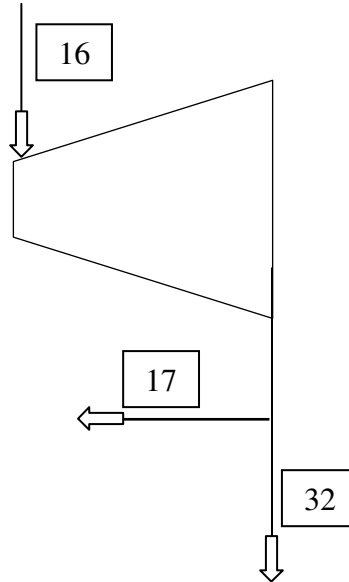


Figure 4.10. HP turbine schematic

At turbine, inlet flow rate have to be equal to outlet flow rate,

$$m_{16} = m_{17} + m_{32} = 524.2 \text{ kg/s}$$

Heat transfer is neglected because the turbine is assumed to be adiabatic, potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are same.

If the first law is arranged for HP turbine,

$$W_{out} = (m_{16} h_{16}) - (m_{17} h_{17} + m_{32} h_{32})$$

Turbine work,

$$W_{out} = 174368.99 \text{ kW}$$

Reversible turbine work,

$$W_{rev} = m_{16} (h_{16} - T_0 s_{16}) - m_{17} (h_{17} - T_0 s_{17}) - m_{32} (h_{32} - T_0 s_{32})$$

$$W_{rev} = 180186 \text{ kW}$$

To calculate irreversibility's of turbine, entropy generations have to be calculated.

Entropy generation is calculated using equation (3.7),

$$S_{gen} = m_{16} s_{16} - m_{17} s_{17} - m_{32} s_{32}$$

$$S_{gen} = 19.49 \text{ kW/K}$$

Irreversibility of HP turbine is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 5809.42 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16)

$$\psi_{16} = (h_{16} - h_0) - T_0 (s_{16} - s_0)$$

$$\psi_{16} = 1496.684 \text{ kJ/kg}$$

Exergy at outlets are calculated using equation (3.16)

$$\psi_{17} = (h_{17} - h_0) - T_0 (s_{17} - s_0)$$

$$\psi_{14} = 1156.7 \text{ kJ/kg}$$

$$\psi_{32} = (h_{32} - h_0) - T_0 (s_{32} - s_0)$$

$$\psi_{32} = 1156.7 \text{ kJ/kg}$$

Exergy differences inlet and outlet gives reversible power of turbine,

$$W_{rev} = (m_{16} \psi_{16}) - (m_{17} \psi_{17} + m_{32} \psi_{32})$$

$$W_{rev} = 180186 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{rev} - W_{out}$$

$$I = 5809.42 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between reversible power and output power at the same time.

The second law efficiency is calculated using equation (3.20),

$$\eta_{II} = \frac{W_{out}}{W_{rev}}$$

$$\eta_{II} = 96.78 \%$$

4.1.11. IP Turbine

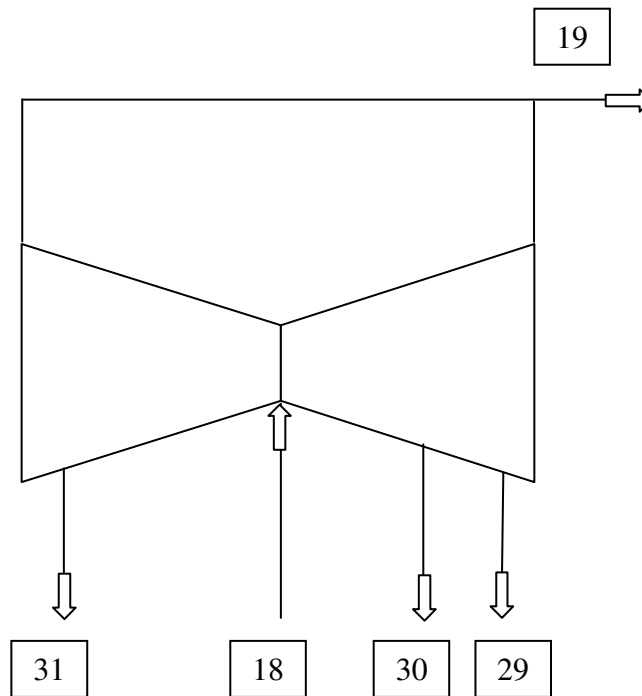


Figure 4.11. IP turbine schematic

At turbine, inlet flow rate have to be equal to outlet flow rate,

$$m_{18} = m_{19} + m_{29} + m_{30} + m_{31} = 466.5 \text{ kg/s}$$

Heat transfer is neglected because the turbine is assumed to be adiabatic, potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are same.

If the first law is arranged for IP turbine,

$$W_{out} = (m_{18} h_{18}) - (m_{19} h_{19} + m_{29} h_{29} + m_{30} h_{30} + m_{31} h_{31})$$

Turbine work,

$$W_{out} = 257723.19 \text{ kW}$$

Reversible Turbine work,

$$W_{rev} = m_{18} (h_{18} - T_0 s_{18}) - m_{19} (h_{19} - T_0 s_{19}) - m_{29} (h_{29} - T_0 s_{29}) - m_{30} (h_{30} - T_0 s_{30}) - m_{31} (h_{31} - T_0 s_{31})$$

$$W_{rev} = 266005.29 \text{ kW}$$

To calculate irreversibility's of pumps, entropy generations have to be calculated. Entropy generation is calculated using equation (3.7),

$$S_{gen} = m_{18} s_{18} - m_{19} s_{19} - m_{29} s_{29} - m_{30} s_{30} - m_{31} s_{31}$$

$$S_{gen} = 27.792 \text{ kW/K}$$

Irreversibility of IP turbine is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 8282.105 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16)

$$\psi_{18} = (h_{18} - h_0) - T_0 (s_{18} - s_0)$$

$$\psi_{18} = 1412.864 \text{ kJ/kg}$$

Exergy at outlets is calculated using equation (3.16)

$$\psi_{19} = (h_{19} - h_0) - T_0 (s_{19} - s_0)$$

$$\psi_{19} = 814.788 \text{ kJ/kg}$$

$$\psi_{29} = (h_{29} - h_0) - T_0 (s_{29} - s_0)$$

$$\psi_{29} = 814.788 \text{ kJ/kg}$$

$$\psi_{30} = (h_{30} - h_0) - T_0 (s_{30} - s_0)$$

$$\psi_{30} = 974.362 \text{ kJ/kg}$$

$$\psi_{31} = (h_{31} - h_0) - T_0 (s_{31} - s_0)$$

$$\psi_{19} = 1174.424 \text{ kJ/kg}$$

Exergy differences inlet and outlet gives reversible power of turbine,

$$W_{rev} = (m_{18} \psi_{18}) - (m_{19} \psi_{19} + m_{29} \psi_{29} + m_{30} \psi_{30} + m_{31} \psi_{31})$$

$$W_{rev} = 266005.29 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{rev} - W_{out}$$

$$I = 8282.1 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between reversible power and output power at the same time.

Second law efficiency is calculated using equation (3.20),

$$\eta_{II} = \frac{W_{out}}{W_{rev}}$$

$$\eta_{II} = 96.9\%$$

4.1.12. LP 1 Turbine

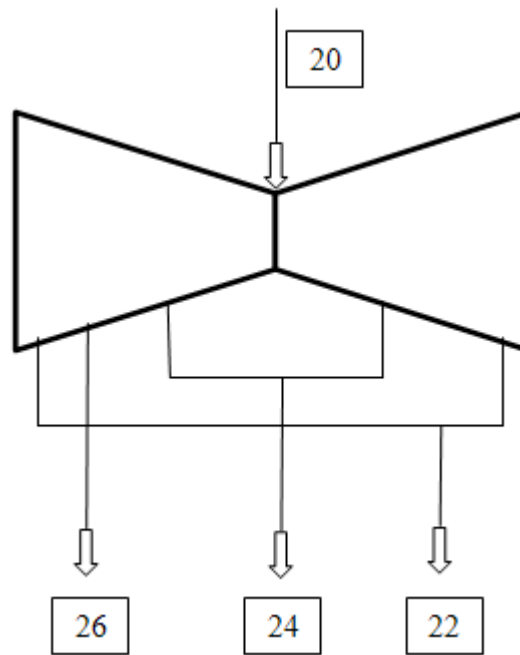


Figure 4.12. LP 1 turbine schematic

At turbine, inlet flow rate have to be equal to outlet flow rate,

$$m_{20} = m_{22} + m_{24} + m_{26} = 197.7 \text{ kg/s}$$

Heat transfer is neglected because the turbine is assumed to be adiabatic, potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are same.

If the first law is arranged for IP turbine,

$$W_{out} = (m_{20} h_{20}) - (m_{22} h_{22} + m_{24} h_{24} + m_{26} h_{26})$$

Turbine work,

$$W_{out} = 113345.3 \text{ kW}$$

Reversible turbine work,

$$W_{rev} = m_{20}(h_{20} - T_0 s_{20}) - m_{22}(h_{22} - T_0 s_{22}) - m_{24}(h_{24} - T_0 s_{24}) - m_{26}(h_{26} - T_0 s_{26})$$

$$W_{rev} = 130996.5 \text{ kW}$$

To calculate irreversibility's of pumps, entropy generations have to be calculated.

Entropy generation is calculated using equation (3.7),

$$S_{gen} = m_{20} s_{20} - m_{22} s_{22} - m_{24} s_{24} - m_{26} s_{26}$$

$$S_{gen} = 59.23 \text{ kW/K}$$

Irreversibility of LP 1 turbine is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 17651.2 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16)

$$\psi_{20} = (h_{20} - h_0) - T_0 (s_{20} - s_0)$$

$$\psi_{20} = 814.788 \text{ kJ/kg}$$

Exergy at outlet are calculated using equation (3.16)

$$\psi_{22} = (h_{22} - h_0) - T_0 (s_{22} - s_0)$$

$$\psi_{22} = 73.86 \text{ kJ/kg}$$

$$\psi_{24} = (h_{24} - h_0) - T_0 (s_{24} - s_0)$$

$$\psi_{24} = 438.4 \text{ kJ/kg}$$

$$\psi_{26} = (h_{26} - h_0) - T_0 (s_{26} - s_0)$$

$$\psi_{26} = 635.2 \text{ kJ/kg}$$

Exergy differences inlet and outlet gives reversible power of turbine,

$$W_{rev} = (m_{20} \psi_{20}) - (m_{22} \psi_{22} + m_{24} \psi_{24} + m_{26} \psi_{26})$$

$$W_{rev} = 130996.5 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{rev} - W_{out}$$

$$I = 17651.2 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between reversible power and output power at the same time.

The second law efficiency is calculated using equation (3.20),

$$\eta_{II} = \frac{W_{out}}{W_{rev}}$$

$$\eta_{II} = 86.5\%$$

4.1.13. LP 2 Turbine

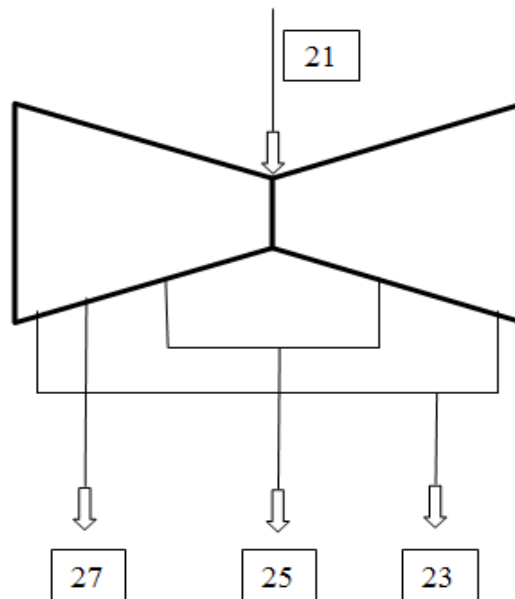


Figure 4.13. LP 2 turbine schematic

At turbine, inlet flow rate have to be equal to outlet flow rate,

$$m_{21} = m_{23} + m_{25} + m_{27} = 197.7 \text{ kg/s}$$

Heat transfer is neglected because the turbine is assumed to be adiabatic, potential energy is neglected because level differences is assumed to be zero and kinetic energy is neglected because of inlet and outlet pipes diameter are same.

If the first law is arranged for IP turbine,

$$W_{out} = (m_{21} h_{21}) - (m_{23} h_{23} + m_{25} h_{25} + m_{27} h_{27})$$

Turbine work,

$$W_{out} = 117438.19 \text{ kW}$$

Reversible pump work,

$$W_{rev} = m_{21} (h_{21} - T_0 s_{21}) - m_{23} (h_{23} - T_0 s_{23}) - m_{25} (h_{25} - T_0 s_{25}) - m_{27} (h_{27} - T_0 s_{27})$$

$$W_{rev} = 134557.96 \text{ kW}$$

To calculate irreversibility's of pumps, entropy generations have to be calculated.

Entropy generation is calculated using equation (3.7),

$$S_{gen} = m_{21} s_{21} - m_{23} s_{23} - m_{25} s_{25} - m_{27} s_{27}$$

$$S_{gen} = 57.44 \text{ kW/K}$$

Irreversibility of LP 2 turbine is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 17119.7 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16)

$$\psi_{21} = (h_{21} - h_0) - T_0 (s_{21} - s_0)$$

$$\psi_{21} = 814.788 \text{ kW}$$

Exergy at outlets are calculated using equation (3.16)

$$\psi_{23} = (h_{23} - h_0) - T_0 (s_{23} - s_0)$$

$$\psi_{23} = 91 \text{ kJ/kg}$$

$$\psi_{25} = (h_{25} - h_0) - T_0 (s_{25} - s_0)$$

$$\psi_{25} = 224.92 \text{ kJ/kg}$$

$$\psi_{27} = (h_{27} - h_0) - T_0 (s_{27} - s_0)$$

$$\psi_{27} = 635.2 \text{ kJ/kg}$$

Exergy loss,

$$W_{rev} = (m_{21} \psi_{21}) - (m_{23} \psi_{23} + m_{25} \psi_{25} + m_{27} \psi_{27})$$

$$W_{rev} = 134557.96 \text{ kW}$$

Irreversibilities can be calculated using equation (3.15) alternatively,

$$I = W_{rev} - W_{out}$$

$$I = 17119.7 \text{ kW}$$

As it can be seen irreversibilities is equal to differences between reversible power and output power at the same time.

The second law efficiency is calculated using equation (3.20),

$$\eta_{II} = \frac{W_{out}}{W_{rev}}$$

$$\eta_{II} = 87.2\%$$

4.1.14. Condenser 1

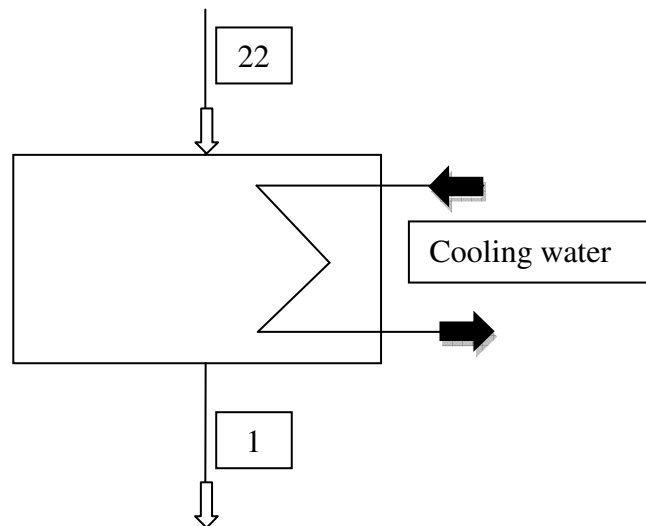


Figure 4.14. Condenser 1 schematic

Inlet and outlet flow rates are the same.

$$m_{22} = m_1 = 161.7 \text{ kg/s}$$

$$m_{CW} = 29500 \text{ kg/s} \quad T_{in} = 23^\circ \quad T_{out} = 25.9^\circ$$

If the first law is applied for condenser 1, Q_1 is found.

$$Q_1 = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{22} h_{22} + m_{CW,in} h_{CW,in}$$

$$Q_{out} = m_1 h_1 + m_{CW, out} h_{CW, out}$$

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

$$Q_l = 2199.82 \text{ kW}$$

$$\text{Total energy loss to outside from condenser} = 353249.8 \text{ kW}$$

Entropy generation is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = m_1 s_1 + m_{CW, out} h_{CW, out} - m_{22} s_{22} - m_{CW, in} h_{CW, in} + \frac{Q_l}{T_0}$$

$$S_{gen} = 12.79 \text{ kW/K}$$

Irreversibility of condenser 1 is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 3812.35 \text{ kW}$$

Exergy at inlet is calculated using equation (3.16)

$$\psi_{22} = (h_{22} - h_0) - T_0 (s_{22} - s_0)$$

$$\psi_{22} = 73.86 \text{ kJ/kg}$$

$$\psi_{CW, in} = (h_{CW, in} - h_0) - T_0 (s_{CW, in} - s_0)$$

$$\psi_{CW, in} = 4248 \text{ kJ/kg}$$

Exergy at outlet is calculated using equation (3.16)

$$\psi_1 = (h_1 - h_0) - T_0 (s_1 - s_0)$$

$$\psi_1 = 0 \text{ kJ/kg}$$

$$\psi_{CW, out} = (h_{CW, out} - h_0) - T_0 (s_{CW, out} - s_0)$$

$$\psi_{CW, out} = 3658 \text{ kJ/kg}$$

Exergy loss,

$$X_l = m_{22} \psi_{22} + m_{CW, in} \psi_{CW, in} - m_1 \psi_1 - m_{CW, out} \psi_{CW, out}$$

$$X_l = 3812.35 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{22} \psi_{22} + m_{CW,in} \psi_{CW,in}} \right)$$

$$\eta_{II} = 51.4\%$$

4.1.15. Condenser 2

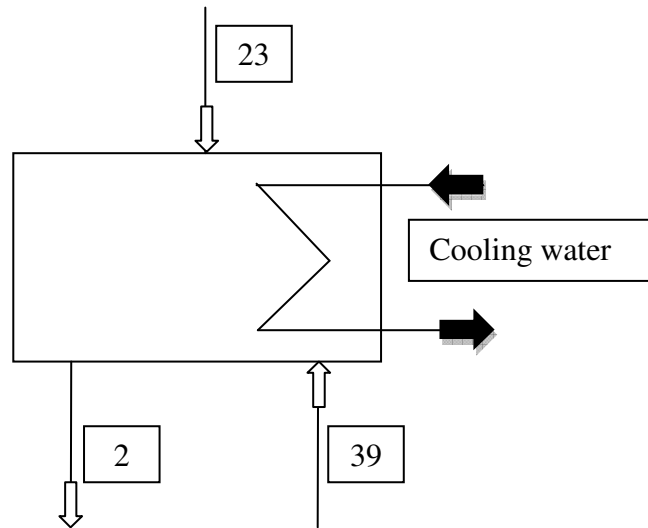


Figure 4.15. Condenser 2 schematic

Inlet and outlet flow rates are the same.

$$m_{23} = m_7 + m_{39} = 169.2 \text{ kg/s}$$

$$m_{CW} = 29500 \text{ kg/s} \quad T_{in} = 25.9^\circ \quad T_{out} = 28.8^\circ$$

If the first law is applied for Condenser 2, Q_1 is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{23} h_{23} + m_{39} h_{39} + m_{CW,in} h_{CW,in}$$

$$Q_{out} = m_2 h_2 + m_{CW,out} h_{CW,out}$$

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

$$Q_l = 3360.8 \text{ kW}$$

$$\text{Total energy loss to outside from condenser} = 349778.9 \text{ kW}$$

Entropy generation is calculated using equation (3.7),

$$S_{gen} = \sum m_{out} s_{out} - \sum m_{in} s_{in} + \frac{Q_l}{T_0}$$

$$S_{gen} = m_2 s_2 + m_{CW,out} h_{CW,out} - m_{23} s_{23} + m_{39} s_{39} - m_{CW,in} h_{CW,in} + \frac{Q_l}{T_0}$$

$$S_{gen} = 19.48 \text{ kW/K}$$

Irreversibility of condenser 2 is calculated using equation (3.14),

$$I = T_0 S_{gen}$$

$$I = 5807.1 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16)

$$\psi_{23} = (h_{23} - h_0) - T_0 (s_{23} - s_0)$$

$$\psi_{23} = 91 \text{ kJ/kg}$$

$$\psi_{39} = (h_{39} - h_0) - T_0 (s_{39} - s_0)$$

$$\psi_{39} = 0.354 \text{ kJ/kg}$$

$$\psi_{CW,in} = (h_{CW,in} - h_0) - T_0 (s_{CW,in} - s_0)$$

$$\psi_{CW,in} = 3658 \text{ kJ/kg}$$

Exergy at outlet is calculated using equation (3.16)

$$\psi_2 = (h_2 - h_0) - T_0 (s_2 - s_0)$$

$$\psi_2 = 0.288$$

$$\psi_{CW,out} = (h_{CW,out} - h_0) - T_0 (s_{CW,out} - s_0)$$

$$\psi_{CW,out} = 4631.5 \text{ kJ/kg}$$

Exergy loss,

$$X_l = (m_{23} \psi_{23} + m_{39} \psi_{39} + m_{CW,in} \psi_{CW,in}) - (m_2 \psi_2 + m_{CW,out} \psi_{CW,out})$$

$$X_l = 5807.1 \text{ kW}$$

As it can be seen here irreversibilities is equal to exergy losses.

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \left(\frac{T_0 S_{gen}}{m_{23} \psi_{23} + m_{39} \psi_{39} + m_{CW.in} \psi_{CW.in}} \right)$$

$$\eta_{II} = 44.7\%$$

4.1.16. Boiler

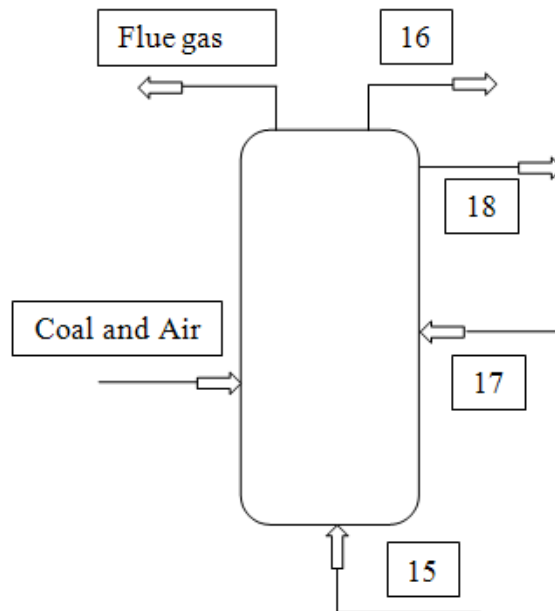


Figure 4. 16. Boiler schematic

Inlet and outlet flow rates are the same.

$$m_{15} = m_{16} = 169.2 \text{ kg/s}$$

$$m_{17} = m_{18} = 466.5 \text{ kg/s}$$

$$m_f = 56.5 \text{ kg/s}$$

$$m_{air} = 586.63 \text{ kg/s}$$

$$m_{fg} = m_f + m_{air}$$

$$m_{fg} = 643.13 \text{ kg/s}$$

$$Q_f = m_f h_f$$

$$Q_f = 1457700 \text{ kW}$$

$$Q_{air} = m_{air} h_{air}$$

$$Q_{fg} = 260038 \text{ kW}$$

$$T_{air} = 610 \text{ K}$$

$$Q_{air} = 187330.3 \text{ kW}$$

$$h_f = 25800 \text{ kJ/kg}$$

$$T_{fg} = 650 \text{ K}$$

Regenerative air preheater is placed exit of boiler therefore burning air is heated then flue gas temperature reduces 420 K after regenerative air preheater, this losses is named as stack gas losses and affect overall efficiency.

If the first law is applied for boiler, Q_l is found.

$$Q_l = \sum m_{in} h_{in} - \sum m_{out} h_{out}$$

$$Q_{in} = m_{15} h_{15} + m_{17} h_{17} + m_{air} h_{air} + m_f h_f$$

$$Q_{out} = m_{16} h_{16} + m_{18} h_{18} + Q_{fg}$$

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

$$Q_l = 8163.4 \text{ kW}$$

Considering stack gas losses,

$$\text{Overall boiler energy loss} = 80871.4 \text{ kW}$$

Exergy at inlets are calculated using equation (3.16)

$$\psi_{15} = (h_{15} - h_0) - T_0 (s_{15} - s_0)$$

$$\psi_{15} = 306.48 \text{ kJ/kg}$$

$$\psi_{17} = (h_{17} - h_0) - T_0 (s_{17} - s_0)$$

$$\psi_{17} = 1156.7 \text{ kJ/kg}$$

Exergy of air is equal to sum of chemical and physical exergy. Physical exergy is calculated using equation (3.16). Chemical exergy of air is calculated using equations (3.41), (3.42), (3.43).

$$X_{air} = X^{ch} + X^{ph}$$

$$X^{ph} = (h_{610 \text{ K}} - h_{298 \text{ K}}) - T_0 (s_{610 \text{ K}} - s_{298 \text{ K}})$$

$$X^{ph} = 102.98 \text{ kW}$$

$$X^{ch} = 2613.67 \text{ kW}$$

$$X_{air} = 2716.65 \text{ kW}$$

Exergy of coal is calculated using equation (3.44), (3.45) and (3.46),

$$X_f = 1547089.3 \text{ kW}$$

Exergy at outlets are calculated using equation (3.16),

$$\psi_{16} = (h_{16} - h_0) - T_0 (s_{16} - s_0)$$

$$\psi_{16} = 1496.68 \text{ kJ/kg}$$

$$\psi_{18} = (h_{18} - h_0) - T_0 (s_{18} - s_0)$$

$$\psi_{18} = 1412.86 \text{ kJ/kg}$$

The exergy of flue gas is equal to the sum of chemical and physical exergy. Firstly, enthalpy is calculated using equations (3.30) and (3.31) secondly, the entropy of flue gas is calculated using equations (3.36) and (3.37). The physical exergy is calculated using equation (3.16). The chemical exergy of flue gas is calculated using equations (3.38), (3.39), (3.40).

$$X_{fg} = X^{ch} + X^{ph}$$

$$X^{ph} = (h_{650 K} - h_{298 K}) - T_0 (s_{650 K} - s_{298 K})$$

$$X^{ph} = 74122.56 \text{ kW}$$

$$X^{ch} = 3538.21 \text{ kW}$$

$$X_{fg} = 77660.77 \text{ kW}$$

Exergy loss,

$$X_l = (m_{15}\psi_{15} + m_{17}\psi_{17} + X_{air} + X_f) - (m_{16}\psi_{16} + m_{18}h_{18} + X_{fg})$$

$$X_l = 728743.9 \text{ kW}$$

Considering stack exergy loss,

$$\text{Overall boiler exergy loss} = 803608.1 \text{ kW}$$

The second law efficiency is calculated using equation (3.23),

$$\eta_{II} = 1 - \frac{T_0 S_{gen}}{m_{15}\psi_{15} + m_{17}\psi_{17} + X_{air} + X_f}$$

$$\eta_{II} = 68 \%$$

4.2. Analysis of Plant

Energy losses of power plant equipments at different load are shown in Figure 4.17. As it can be seen easily from the figure energy loss of condenser is always higher than other equipments. 48.1%, 48.8% and 49.6% of total energy are rejected to outside via condenser at 100%, 70% and 40% loads respectively. Because enthalpy of exhaust steam which comes from LP turbines is higher compared to the condensate water. Reason of condenser losses is the latent heat of exhaust steam is transferred to the cooling water. Also 5.5 %, 6.3% and 6.6% of total energy rejected to ambient via boiler surfaces and flue gases at 100%, 70% and 40% loads respectively.

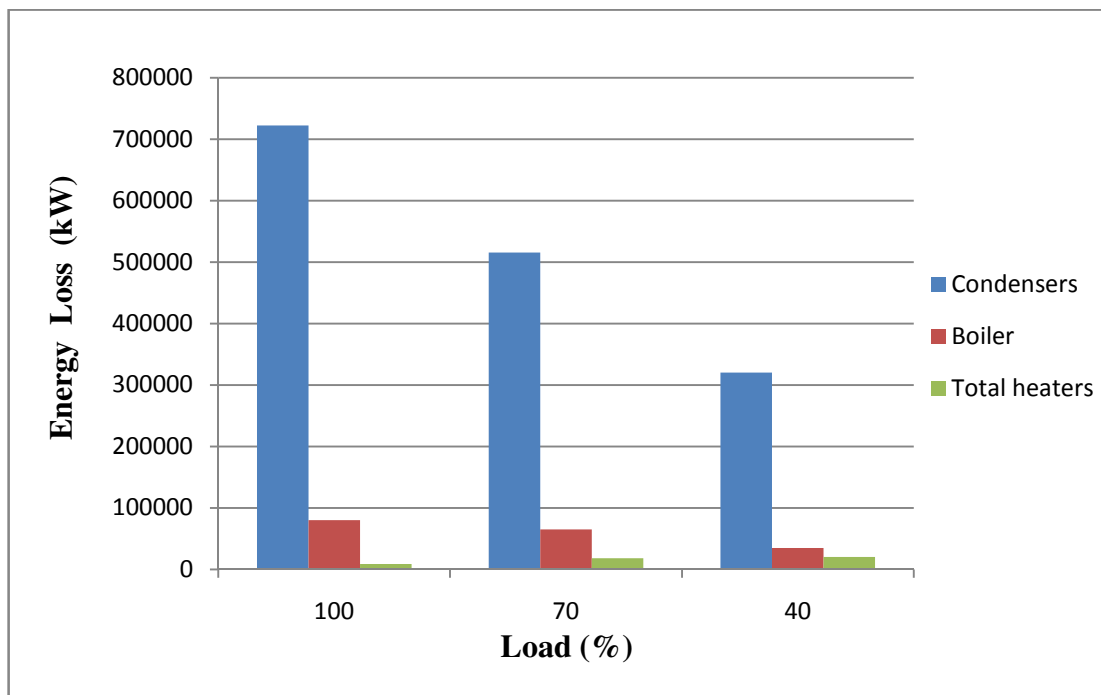


Figure 4.17. Energy loss

Figure 4.18. shows exergy loss of all power plant equipments. As it observed from the figure, exergy loss of the boiler is much higher than other equipments. Despite the fact that major energy loss in the condenser, major exergy loss is placed in the boiler and 51.9 %, 53.6% and 56.1% of input exergy are destroyed gases at 100%, 70% and 40% loads respectively. The high level of exergy

loss in the boiler is due to finite difference between flue gas and working fluid (water and steam). A combustion process usually occurs simultaneously with heat transfer. Both chemical reaction and heat transfer are irreversible processes. The losses in the boiler are due to increase in the entropy generation of flue gases. Additionally substantial amount of heat loss is conveyed by the flue gas to the ambient with high temperature. The exergy losses take place in pumps due to compression ratio and flow control method. Pump losses are usually very small compared to other equipments. Exergy loss in the condenser are due to heat transfer between exhaust steam and cooling water but it's thermodynamically insignificant because temperature differences between them is about 5K so that the exergy losses is small.

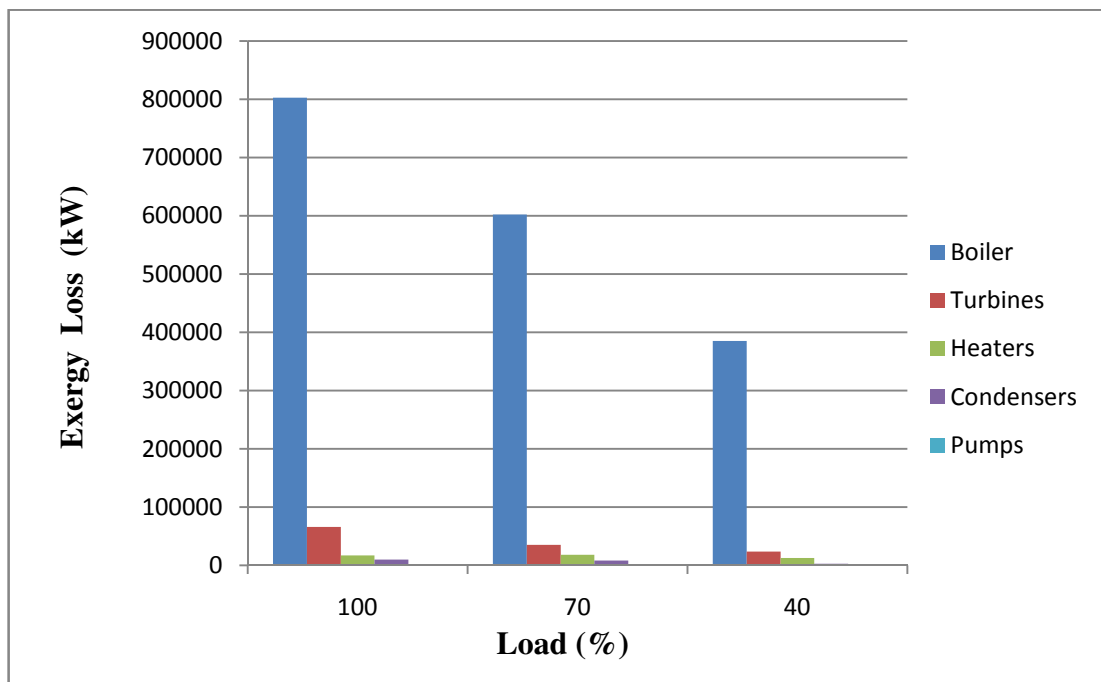


Figure 4.18. Exergy loss

Figure 4.19. shows irreversibility of turbines at three different loads. 5.34%, 5.43% and 5.55% of total exergy are destroyed in turbines at 100%, 70% and 40% loads respectively. The exergy losses in turbines due to pressure drop and expansion process. Fluid friction happens when the fluid expands through the turbine, undergoing internal friction. This friction results in the dissipation of part of its energy into heating itself at the expense of useful work. The fluid then does less

work and leaves through the exhausts with a higher temperature. The more irreversible the process, the more heating effect and the less the work. As the load increases, amount of irreversibility increases in turbines but percent of irreversibility is decreases. The exergy loss in LP turbines is higher than IP and HP turbines because LP turbines work at vacuum pressure so that condensation process begins and steam leaves from LP turbines as a water-steam mixture therefore the rate of entropy and irreversibility increases substantially with condensation process. But the exergy loss in HP turbines is lower because pressure ratio between inlet and outlet is small compared to IP and LP turbines. The value of exergy losses in LP 1 and LP 2 turbine are close to each other, because the operating conditions are almost same.

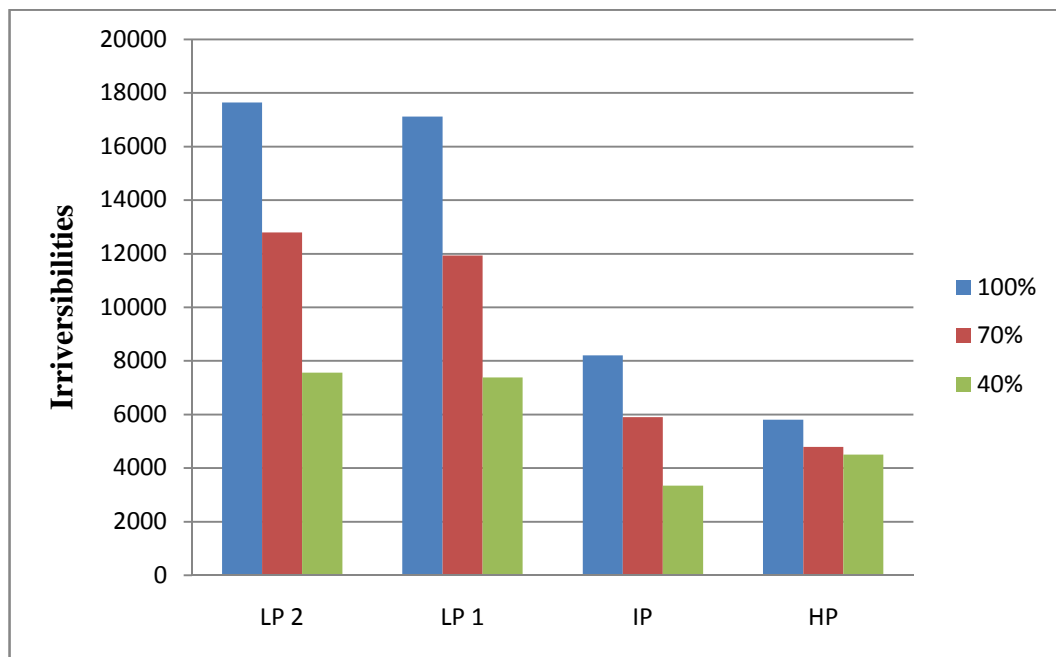


Figure 4.19. Irreversibilities of turbines

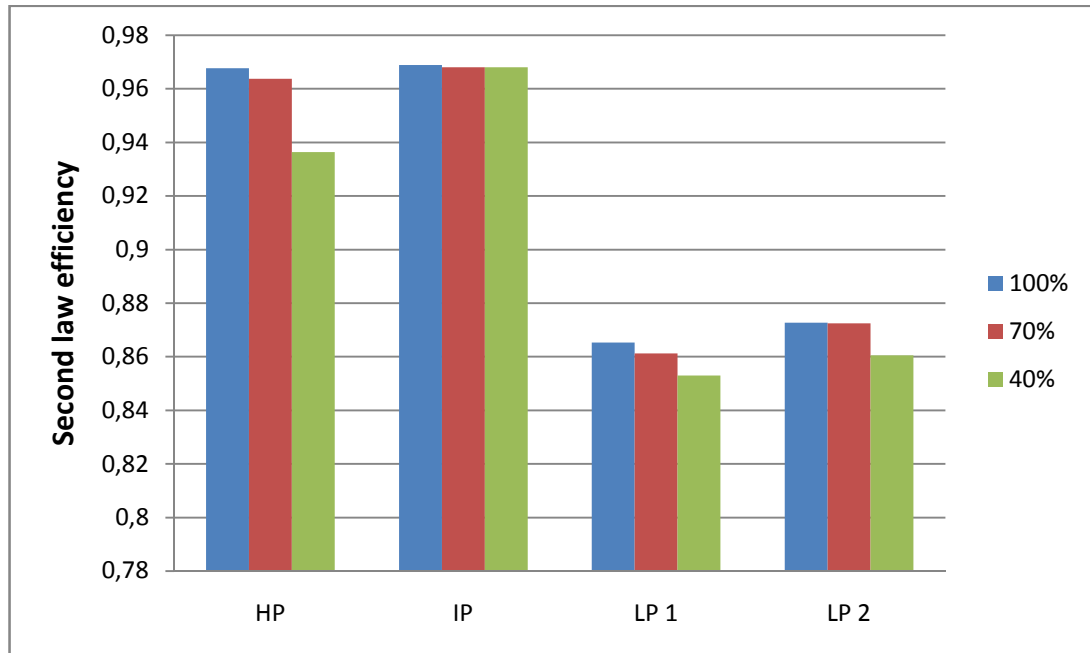


Figure 4.20. The second law efficiency of turbines

Irreversibility in the HP and LP heaters at different load are shown in Figure 4.20. As it can be noted from this figure that irreversibility in the HP heaters is always higher than LP heaters because extraction steam of HP heaters at high pressure and high temperature besides temperature differences between extraction steam and working fluid is higher and also as the load increases, although irreversibility in the HP heaters raises, irreversibility in the LP heaters drops down because while the load increases, temperature differences between extraction steam and working fluid diminishes in LP heaters but in HP heaters temperature differences between them are increases. In addition aim of using HP and LP heaters is recovering latent heat of exhaust steam just before sent to condenser. Also they provide decreasing temperature differences heating condensate and feed water.

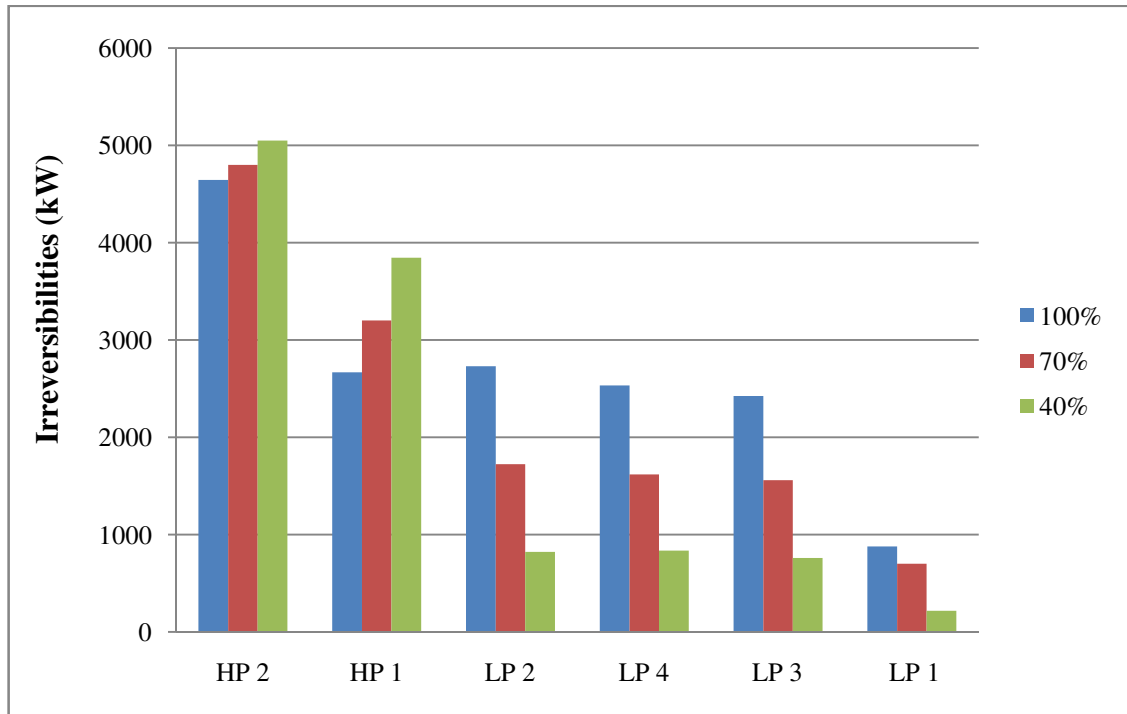


Figure 4.21. Irreversibilities of heaters

The second law efficiency which is belonging to power plant equipments are shown in Figure 4.21. At three different load boiler feed pump are very efficient but boiler is least efficient. And also according to figure second law efficiency of equipments is enhance with increasing load. Besides boiler feed pump efficiency is higher compared to the condenser extraction pump because flow rate of boiler feed pump is controlled by frequency converter but flow rate of condenser extraction pump is controlled by valve (throttling). Pump exergy loss is related with the compression ratios which are 21.4, 21.6 and 22.2 in boiler feed pump and 595.2, 707.8, 913.4 in condenser extraction pump at 100%, 70% and 40% loads respectively.

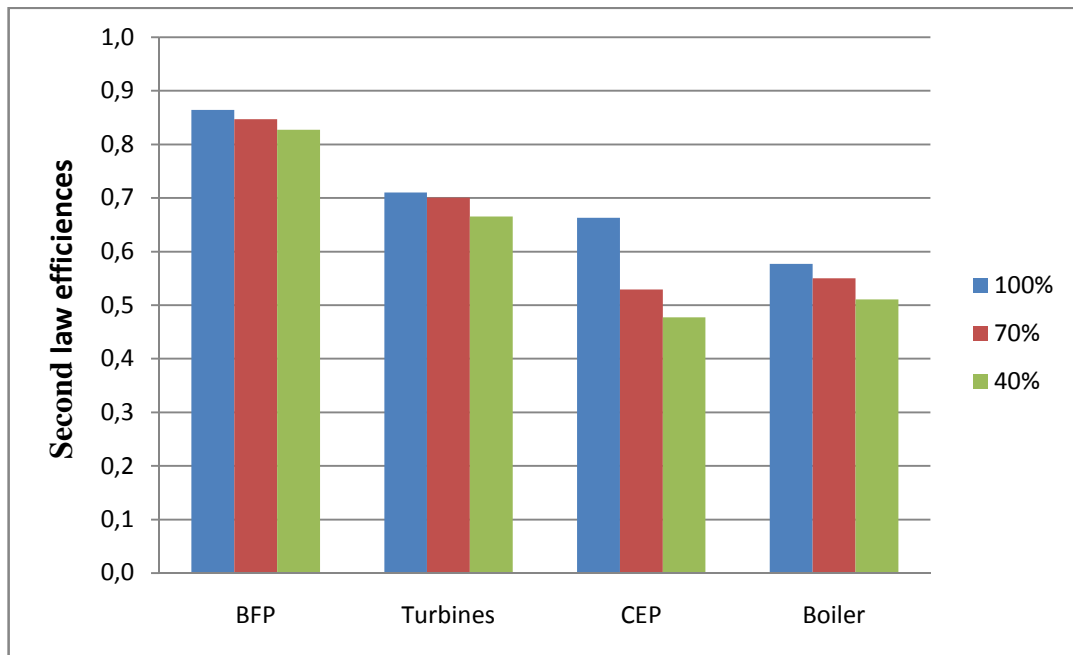


Figure 4.22. The second law efficiency of equipments

First and second law efficiency of boiler versus load are shown in Figure 4.23. As it can be seen from this figure, the first law efficiency slightly increases while load is increased because the boiler pressure reaches to a critical point. Therefore evaporating heat decreases, as a result, input energy to boiler reduced then first law efficiency is increases. The second law efficiency is raises as the load increased because percent of irreversibility in boiler is reduced. To operate boiler efficiently, air-fuel ratio should be arranged well to prevent CO formation considering amount of O_2 at the exit of boiler, boiler exit temperature should not be exceed from normal value. This temperature is controlled reduce fouling around a pipe using soot blowing system efficiently therefore heat transfer between flue gas and working fluid is increased then flue gas temperature is drops.

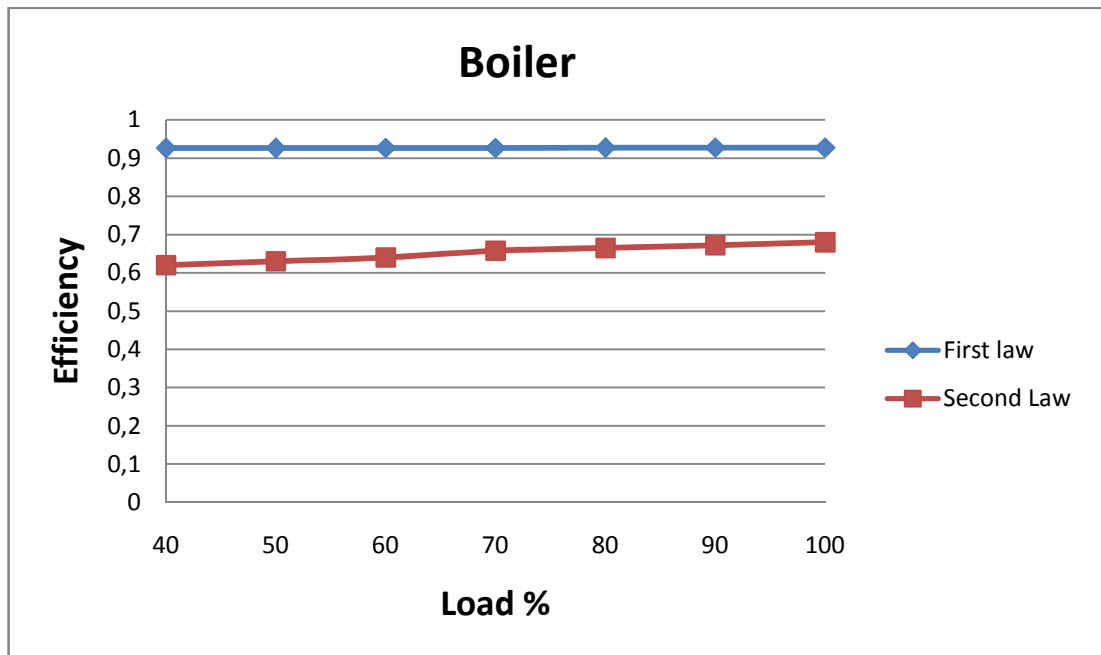


Figure 4.23. The first and second law efficiency of boiler versus load

Overall efficiency of power plant is shown in Figure 4.24. According to energy analysis based on the first law analysis the major energy losses are due to heat rejection in the condenser and heat losses with flue gases but according to exergy analysis based on the second law analysis the major losses are due to steam generation where the fuel exergy is destroyed. It is seen that first and second law efficiencies enhance with increasing load. Overall power plant efficiencies are affected by its component efficiencies and its raises with increasing load. Although the amount of energy loss and exergy loss of power plant is increased, percent of these losses is decreased while load is enhanced so that overall efficiencies are increased. As a consequence, the power plant is not suitable for partial load, it is suitable full load. The first law and second law efficiency of overall power plant were determined using equations (3.18) and (3.26) respectively also net output power of power plant are measured 605, 420, 235 MW at 100%, 70% and 40% loads respectively.

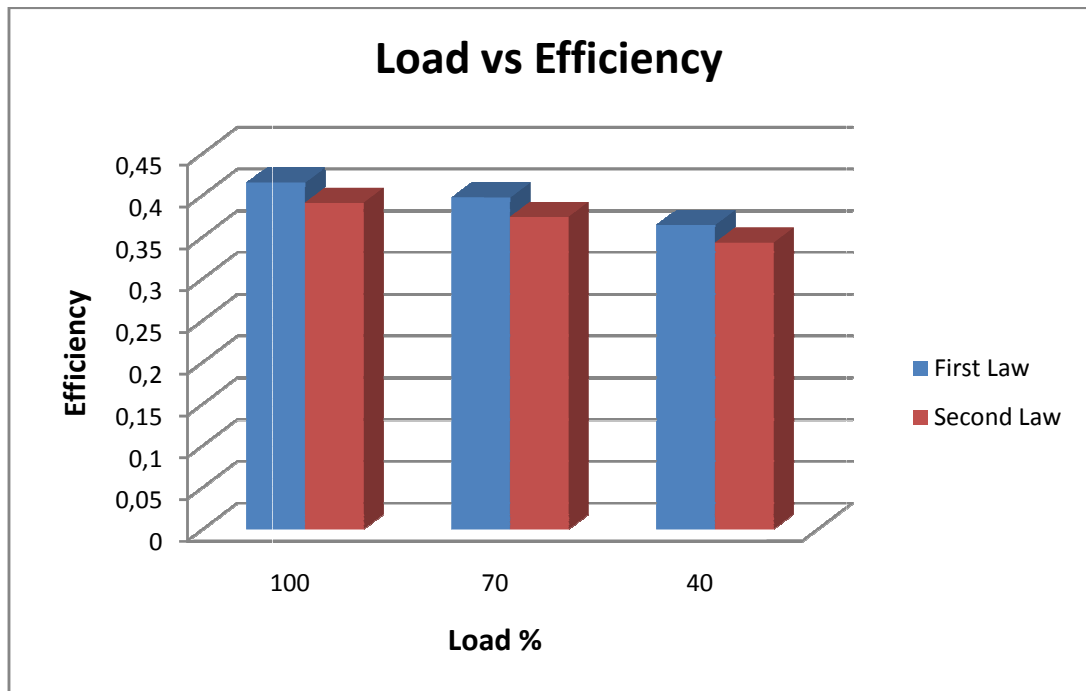


Figure 4.24. Overall efficiencies of power plant

5. CONCLUSION

In this study, energy and exergy analysis of the active power plant were performed using thermodynamic principles for three different loads. Energy and exergy losses and also second law efficiencies of each equipments of the plant and overall first and second law efficiencies were calculated. From the analysis, we can read that where the vulnerable spots are in power production process clearly. In actual thermal power plant operation, abnormal exergy parameters of the corresponding equipment's can be detected and used for the fault location. So the exergy efficiency and loss analysis of the power plant are helpful to malfunctions identification and diagnosis of power plant.

From the results, the following conclusions have been drawn:

1. In considered power plant, the maximum energy loss was found in the condenser at three different loads but the exergy losses are lower because these losses thermodynamically are insignificant which are at low quality. The losses can be used heating greenhouses if it is possible to set near the power plant.
2. In terms of the exergy loss or irreversibility, maximum losses were found in the boiler at three different loads. Because flue gas temperature is very high according to the ambient temperature. These losses may be used in absorptional cooling system for cooling offices and rooms or it may be used for coal drying purposes. Also the losses may be used for heating sites if there is a sites close to power plant. But it is considered that in case of flue gas temperature is below 120° C, acid formation can occur with condensation of flue gas, this situation cause defect the channel and any equipment.
3. As the load increases, irreversibility or the exergy loss increases at each equipment but percentage of irreversibility decreases.

4. Irreversibility in the pumps directly proportional to compression ratio. And the type of flow control method can also affect the irreversibility. As the load increases, compression ratio is decreased but efficiency is increased.
5. Irreversibility in the HP heaters is always higher than LP heaters at three different loads. Additionally, although irreversibility in the HP heaters increases, irreversibility in the LP heaters decreases when the load is increased. Irreversibility in heaters directly proportional to temperature differences between heating medium and working fluid. If the differences between them are increased, heat transfer rate is enhanced as a result, entropy and irreversibility rise.
6. The causes of exergy losses in turbines are mainly related to their design, frictional losses and pressure ratio between inlet and outlet. However, the influence of inter-stage leakages should be investigated and rectified as necessary. It is possible that some inter stage seals are worn thereby allowing some steam leakages which pass through the inter-stages without doing work. Also condensation process should be considered in the last blades of LP turbines. It may cause erodes in the last blades of LP turbines because of water droplets and the entropy and irreversibility increase rapidly.
7. The calculated of the first law efficiency of power plant was found to be 41.5%, 39.7% and 36.4% % at 100%, 70% and 40% load respectively.
8. The calculated of the second law efficiency of power plant was found to be 39.1%, 37.4% and 34.3% at 100%, 70% and 40% load respectively. The power plant is best efficient at full load.

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CURRICULUM VITAE

Mehmet TONTU was born in 15.08.1986 in Ceyhan. He has been lived in Ceyhan since 1986. He is a single. He graduated from Ceyhan High School in 2003 in Ceyhan. In 2005, he enrolled Mechanical Engineering Department of University of ukurova. He graduated from the Mechanical Engineering Department in 2010 and he started to his MSc education in 2011.

Mehmet TONTU is working in Isken sugözü power plant as an operation engineer. He is working here about two years. He has advanced knowledge about power plant operation and its components.

APPENDICES

APPENDIX A

Thermodynamic properties for each stream at 70 % load.

%70 Load						
No	Statement	Temperature	Pressure	Flow Rate	Entalpy	Entropy
		T(°C)	P(bar)	m(kg/s)	h(kj/kg)	s(kj/kg.K)
1	water	27	0,0357	114	113,24	0,395
2	water	29,3	0,0409	145,3	123	0,427
3	water	28,3	0,038	259,3	119	0,412
4	water	28,3	0,038	129,65	119	0,412
5	water	28,7	26,9	64,825	122,8	0,418
6	water	33,3	10,45	259,3	140,6	0,481
7	water	51	10,21	259,3	214,5	0,716
8	water	82,4	9,96	259,3	345,7	1,103
9	water	113,2	9,72	289,3	475,5	1,453
10	water	142,4	9,46	289,3	599,8	1,763
11	water	167,8	7,49	179,7	709,6	2,02
12	water	170,8	162	179,7	731	2,031
13	water	170,8	162	359,4	731	2,031
14	water	202,9	158,8	359,4	871,4	2,335
15	water	248	158,3	359,4	1076,6	2,748
16	steam	541	129,3	359,4	3447	6,58
17	steam	352,8	36,87	322,2	3107,5	6,66
18	steam	538	33,68	322,2	3538,8	7,283
19	steam	246,3	4,17	273,1	2956,4	7,346
20	steam	246,3	4,17	137	2956,4	7,346
21	steam	246,3	4,17	136,1	2956,4	7,346
22	mix	27	0,0357	114	2315,8	7,702
23	mix	29,3	0,0409	120	2323,7	7,661
24	mix	84,3	0,5647	15,4	2623,6	7,473
25	mix	52,9	0,1426	8,9	2444,2	7,557
26	steam	159,7	1,72	7,6	2791	7,4
27	steam	159,7	1,72	7,2	2791	7,4

28	steam	159,7	1,72	15	2791	7,4
29	steam	246,3	4,17	15	2956,4	7,345
30	steam	326,7	7,58	15,1	3114	7,356
31	steam	424,5	16,13	19	3308	7,312
32	steam	352,8	36,87	35,4	3107,5	6,641
33	water	52,9	0,1426	8,9	221,5	0,741
34	water	84,3	0,5647	15,4	353	1,126
35	water			30		
36	water	4	115,1	15	483,2	1,474
37	water	172,8	16	54,4	731,8	2,068
38	water	204,7	36	35,4	874,2	2,371
39	water	31,2	1,6	25,3	130,8	0,453

APPENDIX B

Thermodynamic properties for each stream at 40 % load.

%40 Load						
No	Statement	Temperature	Pressure	Flow Rate	Enthalpy	Entropy
		T(°C)	P(bar)	m(kg/s)	h(kj/kg)	s(kj/kg.K)
1	water	25,5	0,0326	69,8	107	0,373
2	water	26,9	0,0356	86,5	113	0,394
3	water	26,3	0,0342	156,3	110,3	0,385
4	water	26,3	0,0342	78,15	110,3	0,385
5	water	27	31,24	78,15	116	0,395
6	water	30,8	6,74	156,3	129,7	0,447
7	water	43,5	6,65	156,3	182,6	0,618
8	water	71,6	6,56	156,3	300,2	0,974
9	water	99	6,47	172,1	415,3	1,295
10	water	125,4	6,38	172,1	527	1,585
11	water	147,2	4,42	104,85	620,2	1,813
12	water	149	98,5	104,85	634	1,821
13	water	149	98,5	209,7	634	1,821
14	water	179,9	98,27	209,7	761,3	2,126
15	water	219,8	98,11	209,7	945	2,502
16	steam	541	80	209,7	3496	6,851
17	steam	352,6	21,43	188,5	3140,5	6,932
18	steam	514	19,65	188,5	3499,5	7,481
19	steam	232,5	2,46	163	2934,3	7,543
20	steam	232,5	2,46	81,5	2934,3	7,543
21	steam	232,5	2,46	81,5	2934,3	7,543
22	mix	25,5	0,0326	69,2	2348	7,892
23	mix	26,9	0,0356	73,9	2351,1	7,866
24	mix	72,5	0,3475	8,2	2612,6	7,663
25	steam	44,2	0,092	3,8	2439,2	7,732
26	steam	148,7	1,021	4,1	2773,8	7,598

27	steam	148,7	1,021	3,8	2773,8	7,598
28	steam	148,7	1,021	7,9	2773,8	7,598
29	steam	232,5	2,46	7,9	2934,3	7,543
30	steam	310,5	4,7	6,6	3087	7,528
31	steam	404,8	9,4	11	3275,5	7,511
32	steam	352,6	21,43	21,2	3140,5	6,932
33	water	44,2	0,0918	4,4	182,9	0,621
34	water	72,5	0,3455	8,2	303	0,985
35	water			15,8		
36	water	100	2,4	7,9	419,1	1,306
37	water	150,1	9,15	32,2	632,8	1,842
38	water	180,6	21	21,2	766,4	2,143
39	water	28,8	0,092	12,6	121	0,421