Improvement Power Plant Efficiency with Condenser Pressure

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Abstract– **In this study, the energy and exergy analysis of an ideal Rankin cycle with reheat is presented. The primary objectives of this paper are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses at cycle. In addition, the effect of varying the condenser pressure on this analysis will also be presented. 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 has been presented. Energy losses mainly occurred in the condenser where 2126KW is lost to the environment while nothing was lost from the boiler system because it assumed adiabatic. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system 86.27% and then condenser and stack gas 13.73%. In addition, the calculated thermal efficiency was 38.39 % while the exergy efficiency of the power cycle was 45.85%. The boiler is the major source of irreversibilities. For improvement the power plant efficiency the effect condenser pressure has been studied.**

Key words– Ideal Rankine Cycle, Energy Analysis, Exergy Analyses and Condenser Pressure

I. INTRODUCTION

The Carnot Cycle is the most efficient cycle for any given source and sink temperatures of a heat engine. It is useful to refer to the Carnot Cycle Efficiency when referring to changes in plant operating characteristics.

$$\text{Carnot efficiency} = 1 - \frac{T_L}{T_U} \quad (1)$$

T_L is the Lower Temperature in the Cycle and T_U is the Upper or Higher Temperature in the Cycle. The efficiency of the Carnot Cycle will be increased if the value of T_U is increased or the value of T_L is reduced. In the case of a steam power plant T_U is increased if the mean temperature at which heat is supplied in the boiler is increased. T_L Is reduced if the temperature of condensation of the turbine exhaust steam is reduced.

In broad terms the upper temperature is limited by constructional material considerations, and the lower temperature by the temperature of the cooling water supplied to the condenser. Condensation occurs when the temperature of the condenser cooling water is below the saturation temperature of the low pressure steam entering the condenser. In practice this means a temperature difference of about 10K to 15K. If the condenser cooling water is at 20℃, then the condensing temperature in the condenser would be in the region of 35℃. The condensing pressure would be 0.055 bar. If the condenser cooling water was at 30℃, then the condensing temperature would be about 45℃, and the corresponding condensing pressure would be 0.1 bar. The lower condensing pressure will result in a greater expansion ratio in the turbine meaning that more work will be done in the turbine, leading also to a more efficient power plant. Reference to the Carnot Cycle Efficiency confirms that the lower value of T_L will result in higher plant efficiency. It is therefore essential to keep the condenser pressure as low as possible. It is the temperature of the condensing cooling water supply that controls the condensing pressure. Sea water is used for cooling the condensers in our local power stations, it is therefore to be expected that lower condenser pressures will occur in winter than in summer.

Bejan [1] draw outlines the fundamentals of the methods of exergy analysis and entropy generation minimization (or thermodynamic optimization-the minimization of exergy destruction). The paper begins with a review of the concept of irreversibility, entropy generation, or exergy destruction. Examples illustrate the accounting for exergy flows and accumulation in closed systems, open systems, heat transfer processes, and power and refrigeration plants. George and Park [2] discusses how to estimate the avoidable and unavoidable exergy destruction and investment costs associated with compressors, turbines, heat exchangers and combustion chambers.

This general procedure, although based on many subjective decisions, facilitates and improves applications of exergoeconomics. Kotas [3] explained in this work the concept of exergy used to define criteria of performance of thermal plant. Ganapathy *et al.* [4] studied with an exergy analysis performed on an operating 50 MWe unit of lignite fired steam power plant at Thermal Power Station-I, Neyveli Lignite Corporation Limited, Neyveli, Tamil Nadu, India. The distribution of the exergy losses in several plant components during the real time plant running conditions has been assessed to locate the process irreversibility.

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. Kamate and Gangavati [5] studied exergy analysis of a heat-matched bagasse -based cogeneration plant of a typical 2500 tcd sugar factory, using backpressure and extraction condensing steam turbine is presented. In the analysis, exergy methods in addition to the more conventional energy analyses are employed to evaluate overall and component efficiencies and to identify and assess the thermodynamic losses. Boiler is the least efficient component and turbine is the most efficient component of the plant. The results show that, at optimal steam inlet conditions of 61 bar and 475◦C, the backpressure steam turbine cogeneration plant perform with energy and exergy efficiency of 0.863 and 0.307 and condensing steam turbine plant perform with energy and exergy efficiency of 0.682 and 0.260. Datta *et al.* [6] was presented work on exergy analysis of a coal-based thermal power plant is done using the design data from a 210 MW thermal power plant under operation in India.

The exergy efficiency is calculated using the operating data from the plant at different conditions, viz. at different loads, different condenser pressures, with and without regenerative heaters and with different settings of the turbine governing. The load variation is studied with the data at 100, 75, 60 and 40% of full load. Effects of two different condenser pressures, *i.e.* 76 and 89 mmHg (abs.), are studied. It is observed that the major source of irreversibility in the power cycle is the boiler, which contributes to exergy destruction of the order of 60%. Part load operation increases the irreversibilities in the cycle and the effect is more pronounced with the reduction of the load. Increase in the condenser back pressure decreases the exergy efficiency. Successive withdrawal of the high pressure heaters shows a gradual increment in the exergy efficiency for the control volume excluding the boiler.

Aljundi [7] was presented in this study, the energy and exergy analysis of Al-Hussein power plant in Jordan is presented. The primary objectives of this paper are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses. In addition, the effect of varying the reference environment state on this analysis will also be presented. Energy losses mainly occurred in the condenser where 134 MW is lost to the environment while only 13 MW was lost from the boiler system. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (77%) followed by the turbine (13%), and then the forced draft fan condenser (9%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 26% while the exergy efficiency of the power cycle was 25%.

For a moderate change in the reference environment state temperature, no drastic change was noticed in the performance of major components. Dai *et al.* [8] was done exergy analysis for each cogeneration system is examined, and a parameter optimization for each cogeneration system is achieved by means of genetic algorithm to reach the maximum exergy efficiency. The cement production is an energy intensive industry with energy typically accounting for 50-60% of the production costs. In order to recover waste heat from the preheated exhaust and clinker cooler exhaust gases in cement plant, single flash steam cycle, dual-pressure steam cycle, organic Rankine cycle (ORC) and the Kalina cycle are used for cogeneration in cement plant. The optimum performances for different cogeneration systems are compared under the same condition. The results show that the exergy losses in turbine, condenser, and heat recovery vapor generator are relatively large. Rosen [9] reported results were of energy- and exergy-based comparisons of coal-fired and nuclear electrical generating stations. A version of a process- simulation computer code, previously enhanced by the author for exergy analysis, is used. Overall energy and exergy efficiencies, respectively, are 37% and 36% for the coal-fired process, and 30% and 30% for the nuclear process.

The losses in both plants exhibit many common characteristics. Energy losses associated with emissions (mainly with spent cooling water) account for all of the energy losses, while emission-related exergy losses account for approximately 10% of the exergy losses. The remaining exergy losses are associated with internal consumptions. Dincer and Rosen [10] present effects on the results of energy and exergy analyses of variations in dead-state properties, and involves two main tasks: 1) examination of the sensitivities of energyand exergy values to the choice of the dead-state properties and 2) analysis of the sensitivities of the results of energy and exergy analyses of complex systems to the choice of dead-state properties. A case study of a coalfired electrical generating station is considered to illustrate the actual influences. The results indicate that the sensitivities of energy and exergy values and the results of energy and exergy analyses to reasonable variations in dead-state properties are sufficiently small.

II. CONDENCE TYPES

There are two primary types of condensers that can be used in a power plant: 1. Direct Contact, 2. Surface.

Direct contact condensers condense the turbine exhaust steam by mixing it directly with cooling water. The older type Barometric and Jet-Type condensers operate on similar principles. Steam surface condensers are the most commonly used condensers in modern power plants. The exhaust steam from the turbine flows on the shell side (under vacuum) of the condenser, while the plant's circulating water flows in the Tube side. The source of the circulating water can be either a closed-loop (i.e. cooling tower, spray pond, etc.) or once through (i.e. from a lake, ocean, or river). The condensed steam from the turbine, called condensate, is collected in the bottom of the condenser, which is called a hot well. The condensate is then pumped back to the steam generator to repeat the cycle.

A. Condenser Types Steam Surface Condenser Operation

The main heat transfer mechanisms in a surface condenser are the condensing of saturated steam on the outside of the tubes and the heating of the circulating water inside the tubes. Thus for a given circulating water flow rate, the water inlet temperature to the condenser determines the operating pressure of the condenser. As this temperature is decreased, the condenser pressure will also decrease. As described above, this decrease in the pressure will increase the plant output and efficiency. Due to the fact that a surface condenser operates under vacuum, noncondensable gases will migrate towards the condenser. The noncondensable gases consist of mostly air that has leaked into the cycle from components that are operating below atmospheric pressure (like the condenser). These gases can also result from caused by the decomposition of water into oxygen and hydrogen by thermal or chemical reactions. These gases must be vented from the condenser for the following reasons:

The gases will increase the operating pressure of the condenser. Since the total pressure of the condenser will be the sum of partial pressures of the steam and the gases, as more gas is leaked into the system, the condenser pressure will rise. This rise in pressure will decrease the turbine output and efficiency. The gases will blanket the outer surface of the tubes. This will severely decrease the heat transfer of the steam to the circulating water. Again, the pressure in the condenser will increase. The corrosiveness of the condensate in the condenser increases as the oxygen content increases. Oxygen causes corrosion, mostly in the steam generator. Thus, these gases must be removed in order to extend the life of cycle components.

B. Stem Surface Condenser Air Removal

The two main devices that are used to vent the noncondensable gases are Steam Jet Air Ejectors and Liquid Ring Vacuum Pumps. Steam Jet Air Ejectors (SJAE) use high-pressure motive steam to evacuate the noncondensables from the condenser (Jet Pump). Liquid Ring Vacuum Pumps use a liquid compressant to compress the evacuated noncondensables and then discharges them to the atmosphere. (See the HEI Primer on Vacuum on the HEI Website, www.heatexchange.org, for further information about Steam Jet Ejectors and Liquid Ring Vacuum Pumps.) To aid in the removal of the noncondensable gases, condensers are equipped with an Air-Cooler section.

The Air- Cooler section of the condenser consists of a quantity of tubes that are baffled to collect the noncondensables. Cooling of the noncondensables reduces their volume and the required size of the air removal equipment. Air removal equipment must operate in two modes: hogging and holding. Prior to admitting exhaust steam to a condenser, all the noncondensables must be vented from the condenser. In hogging mode, large volumes of air are quickly removed from the condenser in order to reduce the condenser pressure from atmospheric to a predetermined level. Once the desired pressure is achieved, the air removal system can be operated in holding mode to remove all noncondensable gases.

C. Steam Surface Condenser Configuration

Steam surface condensers can be broadly categorized by the orientation of the steam turbine exhaust to the condenser. Most common are side and down exhaust. In a side exhaust condenser, the condenser and turbine are installed adjacent to each other, and the steam from the turbine enters from the side of the condenser. In a down exhaust condenser, the steam from the turbine enters from the top of the condenser and the turbine is mounted on a foundation above the condenser. Condensers can be further delineated by the configuration of the shell and tube sides. Tube side The tube side of a steam surface condenser can be classified by the following: Number of tube side passes Configuration of the tube bundle and water boxes Most steam surface condensers have either one or multiple tube side passes. The number of passes is defined as how many times circulating water travels the length of the condenser inside the tubes. Condensers with a once-through circulating water system are often one pass. Multiple pass condensers are typically used with closed-loop systems.

The tube side may also be classified as divided or non-divided. In a divided condenser, the tube bundle and water boxes are divided into sections. One or more sections of the tube bundle may be in operation while others are not. This allows maintenance of sections of the tube side while the condenser is operating. In a non-divided tube side, all the tubes are in operation at all times. The shell side of a steam surface condenser can be classified by its geometry. Examples of types are:

- Cylindrical
- Rectangular

The choice of the above configuration is determined by the size of the condenser, plant layout, and manufacturer preference. Steam surface condensers can be multiple shell and multiple pressure configurations, as well. See HEI *Standards for Steam Surface Condensers* for examples of shell and tube configurations and for additional information about steam surface condensers

III. EXERGY ANALYSES

The process flow diagram for the power plant is shown in Fig. 1. The process parameters for the power plant are shown in Table 1. The following thermodynamic analysis of the power plant will consider the balances of mass, energy, entropy and exergy. Unless otherwise specified, the changes in kinetic and potential energies will be neglected and steady state flow will be assumed. For a steady state process, the mass balance for a control volume system in Fig. 1 can be written as Solar collectors or waste heat are suggested as heat sources to operate the cycle.

$$\sum_i \dot{m}_\imath = \sum_i \dot{m}_e \tag{2}$$

The energy balance for a control volume system is written as:

$$\sum_i \dot{E}_\imath + \dot{Q} \quad = \sum_{out} \dot{E_{out}} + \dot{W} \tag{3}$$

The entropy balance for a control volume system is:

$$\sum_i \dot{S} + \sum_i \frac{\dot{Q}}{T} + \dot{S_{gen}} = \sum_{out} \dot{S} + \sum_{out} \frac{\dot{Q}}{T} \tag{4}$$

The exergy balance for a control volume system is written as:

$$\sum_i \dot{E_{x,\imath}} + \sum_k \left(1 - \frac{T}{T_k}\right) \dot{Q}_k + \dot{Q}$$
$$= \sum_{out} \dot{E_{x,out}} + \dot{W} + \dot{E_{x,d}} \tag{5}$$

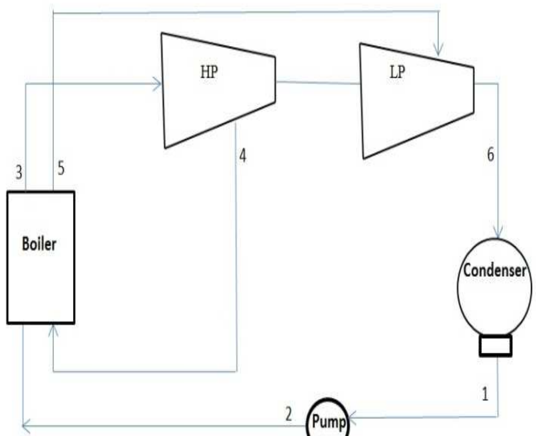

Fig. 1. Schematic diagram of a power plant with reheat

Where the exergy rate of a stream is

$$E_{x,i} = \dot{m}\, e_x \tag{6}$$

$$\dot{m}\, e_x = \dot{m}(e_x^{tm} + e_x^{ch}) \tag{7}$$

The above exergy balance is written in a general form. For the combustion process, the heat input will be included when calculating the chemical exergy of gas. The heat exergy term in Eq. (5) will be used to calculate the exergy loss associated with heat loss to the surroundings. The specific exergy is given by

$$e_x^{tm} = (h - h_0) - T_0(S - S_0) \tag{8}$$

Thermal efficiency and exergy efficiency of power plant calculated from equation 8, 9.

$$\eta_{th,pp} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{9}$$

$$\eta_{ex,pp} = 1 - \frac{\dot{E}_{x,d}}{\dot{E}_{x,in}} \tag{10}$$

$$\dot{Q}_{in} = h[5] + h[3] - h[4] + h[2] \tag{11}$$

$$\dot{E}_{x,in} = \dot{Q}_{in}\left(1 - \frac{T_0}{T_{af}}\right) \tag{12}$$

Where T_{af} is the adiabatic flame temperature? The energy balance equation for calculating adiabatic flame temperature is:

$$\sum N_r \left(\bar{h}_f^0 + \bar{h} - \bar{h}^0\right)_r = \sum N_p \left(\bar{h}_f^0 + \bar{h} - \bar{h}^0\right)_p \tag{13}$$

Table 1: Thermodynamic properties of working fluid of power plant at T_0 (ºC)= 25, P=101 Kpa

Node	P (KPa)	T (ºC)	h (kJ/kg)	s (kJ/kg.K)	e_x (kJ/kg)
1	20	60.06	251.4	0.832	-3.634
2	8000	60.4	259.5	0.8321	4.47
3	8000	500	3400	6.727	1240
4	3000	345.2	3105	6.727	945.7
5	3000	500	3457	7.236	1134
6	20	60.06	2385	7.236	61.74

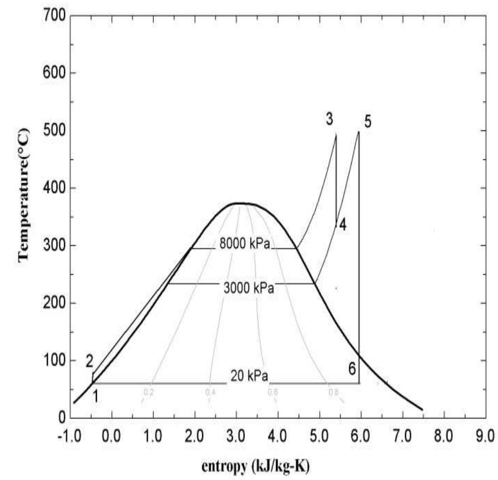

Fig. 2. T-S diagram of cycle power plant

Fig. 2 shows T-S diagram of power-plant cycle. It is a sub critical power-plant; critical power-plant.

V. RESULT AND DISCUSION

The power plant that used to analyze in this research is a power plant with a subcritical boiler, reheat, Hp and Lp turbines. The boiler, turbines and pump assumed to be adiabatic in the paper. The power plant was analyzed using the above relations noting that the environment reference temperature and pressure are 298.15 K and 101.3 kPa, respectively. The energy balance of the power plant is presented in Table 2. It shows that the thermal efficiency (39.8%). The energy balance also reveals that two thirds of the fuel energy is lost in the condenser and carried out into the environment, In addition, losses of energy can be large quantity while it is thermodynamically insignificant due to its low quality. Exergy- based efficiencies and losses, however, provide measures of approach to ideality or deviation from ideality.

Exergy and percent of exergy destruction along with the exergy efficiencies are summarized in Table3 for all components present in the power plant. It was found that the exergy destruction rate of the boiler is dominant over all other irreversibilities in the cycle. It counts alone for 86.27% of losses in the plant, while the exergy destruction rate of the condenser and stack gas is only 13.73%.

According to the first law analysis, energy losses associated with the condenser are significant because they represent about 60.86% of the energy input to the plant. An exergy analysis, however, showed that only 13.73% of the exergy was lost in the condenser and stack gas. The real loss is primarily back in the boiler where entropy was produced. Contrary to the first law analysis, this demonstrates that significant improvements exist in the boiler system rather than in the condenser. The calculated exergy efficiency of the power cycle is 45.85%. This indicates that tremendous opportunities are available for improvement. However, part of this irreversibility cannot be avoided due to physical, technological, and economic constraints

Table 2: Energy balance of the power plant components and percent ratio to fuel energy input

	Energy balance	Percent ratio(%)
boiler	0	0
Hp turbin	0	0
Lp turbin	0	0
Q_in	3493	100
condenser	2126	60.86
W_net	1359	39.9
W_pump	8	0.22

Table 3:Exergy destruction and exergy efficiency of the power plant components when T_0 = 25 ºC, P=101 Kpa

	Exergy destruction (KW)	exergy efficiency (%)	Percent exergy Destruction (%)
boiler	1339	46.69	86.27
Hp turbin	0	100	0
Lp turbin	0	100	0
E_{x_in}	1552	-	100
Condenser	212	58.8	13.22
W_pump	8	100	0.51

A. Parametric Study

Fig. 3 shows the effects of condenser pressure on the cycle performance. It is evident that the efficiency decrease with an increase in the condenser pressure parameters. Decreasing the cycle Condenser pressure and temperature will result to higher power output for the same mass flow rate of steam and fuel input into the boiler, resulting in higher work output of the turbine.

Fig. 4 shows the condenser pressure Vs. Net power and specific volume. The turbine outlet enthalpy is a function of condenser pressure. Decreasing the turbine outlet enthalpy causes the turbine output work to increase. Therefore in order to increase the turbine work, condenser pressure should be

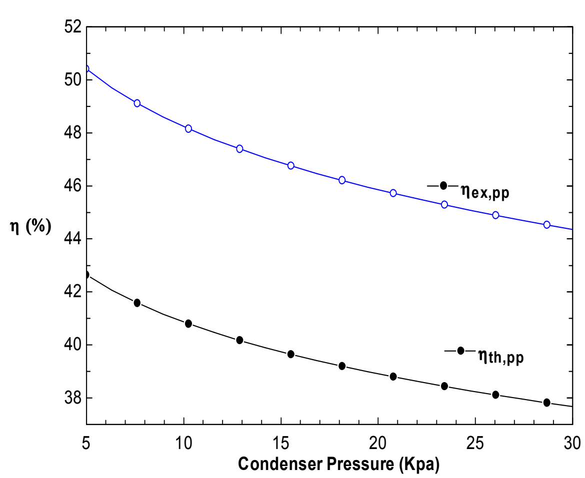

Fig. 3. Condenser pressure vs. thermal efficiency and exergy efficiency

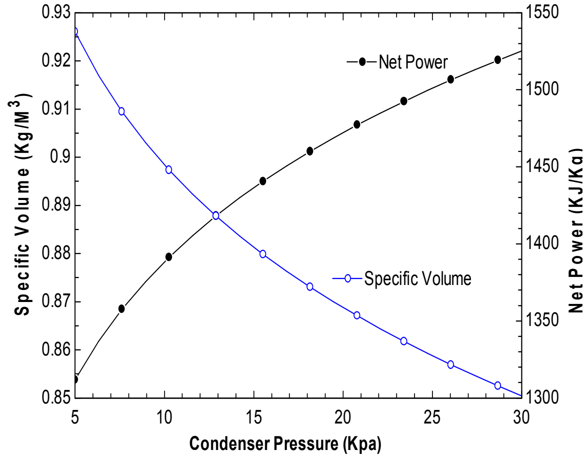

Fig. 4. Condenser pressure Vs. Net power and specific volume

reduced. However, there are the following limitations in the reduction of condenser pressure:

1). The turbine outlet loss increases by reduction of the condenser pressure. In fact reduction of condenser pressure decreases the steam quality or increases the water droplets at the turbine exhaust. These droplets will create a drag force which tends to reduce the turbine output power.

2). The droplets will be also responsible for erosion of the turbine last stage blades.

3). Further reduction of the condenser pressure; will cause the water outlet from condenser to freeze.

Therefore based on the above limitations the condenser optimal pressure range is selected from 6.9 kPa to 13.8 kPa [13].

The condenser is a large shell-and-tube type heat exchanger. This is positioned next to the turbine in order to receive a large flow rate of low pressure steam. This steam in the condenser goes under a phase change from vapor to liquid water. External cooling water is pumped through thousands of tubes in the condenser to transport the heat of condensation of the steam away from the plant. Upon leaving the condenser, the condensate is at a low temperature and pressure. Removal of this condensate may be considered as maintaining the low pressure in the condenser continuously.

The phase change in turn depends on the transfer of heat to the external cooling water. The rejection of heat to the surroundings by the cooling water is essential to maintain the low pressure in the condenser. This heat is absorbed by the cooling water passing through the condenser tubes. The rise in cooling water temperature and mass-flow rate is related to the rejected heat by the following equation. Figure5 illustrated the condenser pressure Vs. heat transfer to boiler and heat rejection of condenser. It shows that rising in condenser pressure is a reason for rising in heat rejection and it due to rising for heat loss. Heat loss means consumption energy without any use. Fig. 5 shows that to have better energy efficiency the condenser pressure must be as low as possible.

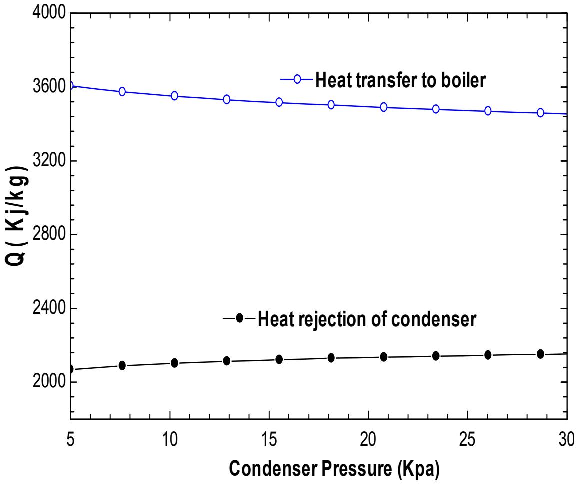

Fig. 5. Condenser pressure Vs. heat transfer to boiler and heat rejection of condenser

VI. CONCULATION

In this study, an energy and exergy analysis as well as the effect of varying the condenser pressure on the subcritical model power plant has been presented. The analyses show that the condenser pressure is an important parameter that affects the output power, power potential and thermal and exergy efficiency of the cycle. Considering the inherent limitation of this parameter as well as the turbine limitation, the minimum allowable condenser pressure should be chosen to produce maximum efficiency and output power. This pressure should be always controlled during the power plant operation. The maximum energy loss was found in the condenser where 60.86% of the input energy was lost to the environment. The exergy analysis of the plant showed that lost energy in the condenser is thermodynamically insignificant due to its low quality. In terms of exergy destruction, the major loss was found in the boiler where 86.271%, of the fuel exergy input to the cycle was destroyed at it. The percent exergy destruction in the condenser and other components was 13.22%. The calculated thermal and exergy efficiency of the power cycle was found to be 38.39%, 45.85.

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REFERENCES

[1] A. Bejan, “Fundamentals of Exergy Analysis, Entropy Generation Minimization, and the Generation of Flow Architecture,” International Journal of Energy Research, Vol. 26, No. 7, 2002, pp. 545-565.

[2] G. Tsatsaronis and M. Park, “On Avoidable and Unavoidable Exergy Destructions and Investment Costs in Thermal Systems,” Energy Conversion & Management, Vol. 43, No. 9-12, 2002, pp. 1259-1270.

[3] T. J. Kotas, “Exergy Criteria of Performance for Thermal Plant: Second of Two Papers on Exergy Techniques in Thermal Plant Analysis,” International Journal of Heat and Fluid Flow, Vol. 2, No. 4, 1980, pp. 147-163.

[4] T. Ganapathy, N. Alagumurthi, R. P. Gakkhar and K. Murugesan, “Exergy Analysis of Operating Lignite Fired Thermal Power Plant,” Journal of Engineering Science and Technology Review, Vol. 2, No. 1, 2009, pp.123-130.

[5] S. C. Kamate and P. B. Gangavati, “Exergy Analysis of Cogeneration Power Plants in Sugar Industries,” Applied Thermal Engineering, Vol. 29, No. 5-6, 2009, pp. 1187-1194.

[6] A. Datta, S. Sengupta and S. Duttagupta, “Exergy Analysis of a Coal-Based 210 mw Thermal Power Plant,” International Journal of Energy Research, Vol. 31, No. 1, 2007, pp. 14-28.

[7] I. H. Aljundi, “Energy and Exergy Analysis of a Steam Power Plant in Jordan,” Applied Thrmal Engineering, Vol. 29, No. 2-3, 2009, pp. 324-328.

[8] Y. Dai, J. Wang and L. Gao, “Exergy Analyses and Parametric Optimizations for Different Cogeneration Power Plants in Cement Industry,” Applied Energy, Vol. 86, No. 6, 2009, pp. 941-948.

[9] M. A. Rosen, “Energy- and Exergy-Based Comparison of Coal-Fired and Nuclear Steam Power Plants,” Exergy, Vol. 1, No. 3, 2001, pp. 180-192.

[10] I. Dincer and M. A. Rosen, “Effect of Varying Dead-State Properties on Energy and Exergy Analyses of Thermal Systems,” International Journal of Thermal Sciences, Vol. 43, No. 3, 2004, pp. 121-133.

[11] H. Erdem, A. V. Akkaya., A. Dagdas, S. H. Sevilgen, B. Sahin, I. Tek, C. Gungor and S. Atas, “Comparative Energetic and Exergetic Performance Analyses for Coal-Fired Thermal Power Plants in Turkey,” International Journal of Thermal Sciences, Vol. 48, No. 11, 2009, pp. 2179-2186.

[12] A. Vidal, R. Best, R. Rivero and J. Cervantes, “Analysis of a Combined Power and Refrigeration Cycle by the ExergyMethod,” Energy, Vol. 31, No. 15, 2006, pp. 3401-3414.