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SECOND LAW ANALYSIS OF A CONVENTIONAL STEAM POWER PLANT

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ABSTRACT

A numerical investigation of exergy destroyed by operation of a conventional steam power plant is computed via an exergy cascade. An order of magnitude analysis shows that exergy destruction is dominated by combustion and heat transfer across temperature differences inside the boiler, and conversion of energy entering the turbine/generator sets from thermal to electrical. Combustion and heat transfer inside the boiler accounts for 53.83 percent of the total exergy destruction. Converting thermal energy into electrical energy is responsible for 41.34 percent of the total exergy destruction. Heat transfer across the condenser accounts for 2.89 percent of the total exergy destruction. Fluid flow with friction is responsible for 0.50 percent of the total exergy destruction. The boiler feed pump turbine accounts for 0.25 percent of the total exergy destruction. Fluid flow mixing is responsible for 0.23 percent of the total exergy destruction. Other equipment including gland steam condenser, drain cooler, deaerator and heat exchangers are, in the aggregate, responsible for less than one percent of the total exergy destruction. An energy analysis is also given for comparison of exergy cascade to energy cascade. Efficiencies based on both the first law and second law of thermodynamics are calculated for a number of components and for the plant. The results show that high first law efficiency does not mean high second law efficiency. Therefore, the second law analysis has been proven to be a more powerful tool in pinpointing real losses. The procedure used to determine total exergy destruction and second law efficiency can be used in a conceptual design and parametric study to evaluate the performance of other steam power plants and other thermal systems.

NOMENCLATURE

A = area, m²
C_p = specific heat, KJ/(kg.c)
E_p = exergy, MJ
g = gravitational acceleration, m/s²
h = enthalpy, KJ/kg
h⁰ = methalpy, KJ/kg, h⁰=h+gz+V²/2
HV = heating value of fuel, KJ/kg
m = mass flow rate, kg/s
P = pressure, KPa

Q = heat rate, KW
 R = ideal gas constant, for water vapor $R=0.461$ KJ/(kg.K)
 s = entropy, MJ/K.kg
 S = entropy, MJ/K
 t = time, sec
 T = temperature, K or C
 v = specific volume, m^3/kg
 V = velocity, m/s
 W = gross turbine power, KW
 x = mass flow rate ratio of mixing
 z = elevation, m
 Z_h = enthalpy departure factor
 Z_s = entropy departure factor

GREEK LETTERS

η_I = first law efficiency
 η_{II} = second law efficiency
 Δ = (pressure) difference

SUBSCRIPTS

a = ambient (reference)
 b = boiler
 bfpt = boiler feed pump turbine
 cond = condenser
 cr = critical point
 dc = drain cooler
 des = destroyed
 fri = due to friction
 gen = entropy generation
 gsc = gland steam condenser
 ht = high pressure turbine
 hx = heat exchangers
 i = inlet, inflow
 it = intermediate pressure turbine
 l = liquid
 loss = due to loss
 lt = low pressure turbine
 mak = water makeup
 mix = due to mixing
 o = outlet, outflow
 rh = reheater
 ssr = steam seal regulator
 v = vapor, steam
 w = water

INTRODUCTION

A conventional steam power plant unit located in Valmy, northern Nevada is shown in Fig. 1. In this Plant, steam generated in the coal-fired boiler enters a high-pressure turbine. Most of the steam leaving the high-pressure turbine enters the intermediate-pressure turbine via a reheater. A small fraction of the steam leakage flow from the high-pressure turbine enters the intermediate-pressure turbine directly (without reheating). After passing through the intermediate-pressure turbine, the steam then powers a low-pressure double flow turbine-generator. The main output steam from the turbine-generator is condensed into water through a condenser and is then diverted back to the boiler through a series of heat exchangers and a deaerator. After partial pressurization, the condensed water is heated inside heat exchangers by the steam coming down from three different turbines through bleeds. Two other pieces of auxiliary equipment in the system schematic are the BFPT and SSR. BFPT is the boiler feed pump turbine which supplies direct drive power to the boiler feed pump. The "steam seal regulator" (SSR) can adjust the steam pressure in the turbine gland seals.

Although there have been a considerable number of prior exergy studies on power generation (Salamon et al, 1980; El-Masri, 1985; Ishida et al, 1987; Bejan, 1988; Lozano and Valero, 1988; Stecco and Desideri, 1988; Valero and Torres, 1988; El-Sayed, 1988; Kalina and Tribus, 1989; Dunbar et al, 1991; Kalina, 1991; Horlock, 1991; Bidini and Stecco, 1991; Tsatsaronis et al, 1991), a second law assessment of exergy cascade for this kind of plant is not available.

Properly quantified performance of a steam power plant must not only account for the energy gains and losses as dictated by the First Law of Thermodynamics, it must also account for the quality of the energy. However, energy quality can be only determined from the Second Law. Exergy analysis is a powerful tool for the evaluation of the thermodynamic and economic performance of thermal systems. In this research, the application of exergy analysis in the evaluation of the steam power plant is described in detail. An energy analysis is also performed for purposes of comparison with the exergy analysis.

EXERGY ANALYSIS AND EXERGY EQUATIONS

As described in the above section and Figure 1, the unit is composed of a coal-fired boiler, reheater, high-pressure turbine, intermediate-pressure turbine, low-pressure double flow turbine generator, heat rejector (condenser), gland steam condenser (GSC), drain cooler (DC), deaerator, six heat exchangers, boiler feed pump turbine, steam seal regulator, transportation pipes, pumps, valves and other auxiliary equipment. The working fluid is water/steam. Exergy destruction happens where energy transfer and momentum transportation are involved. In this analysis, only main

components in the system which play important roles in energy transfer and momentum transportation are considered.

Exergy destruction in the boiler manifests from highly irreversible combustion and heat transfer. The fuel oxidation consumes a certain percentage of the usable fuel energy. The huge gap between steam temperature and combustion temperature necessarily causes similarly huge exergy destruction. The value of exergy loss which is obtained by subtracting exergy leaving from exergy arriving at the boiler is the product value of entropy generation and reference temperature. There are several various proposals for the expression of second law efficiency about power plant components (Horlock, 1991). This investigation adopts Fratzscher's definition which is also recommended by Horlock. The second law efficiency is the ratio of exergy leaving over exergy arriving.

$$E_{b, des} = S_{b, gen} T_a = E_{b, i} - E_{b, o} \quad (1)$$

$$\eta_{II, b} = \frac{E_{b, o}}{E_{b, i}} \quad (2)$$

Exergy arriving includes the exergy of fuel input (fuel exergy), the exergy of water returning from heat exchangers and the exergy coming from high-pressure turbine then entering reheater which is built inside the boiler. Exergy gained by the working fluid which is delivered to the high-pressure turbine and the intermediate-pressure turbine respectively is considered as exergy leaving. The fuel exergy is given by

$$E_{fuel} = \int Q_{fuel} \left(1 - \frac{T_a}{T_{flame}}\right) dt \quad (3)$$

where

$$Q_{fuel} = m_{fuel} HV \quad (4)$$

Bejan (1988) gives the flow exergy as,

$$E=m[h^0-h_a^0-T_a(s-s_a)]\Delta t \quad (5)$$

which is defined as the difference between the flow availability of a stream and that of the same stream at its restricted dead (reference) state. In the above equation, h^0 is methalpy representing the summation of thermal, potential and kinetic energy. After neglecting potential and kinetic energy, the exergy of water from the heat exchangers is expressed by adopting equation (5).

$$E_{b,i,hx}=\int m_{b,i,hx}[h_{b,i,hx}-h_a-T_a(s_{b,i,hx}-s_a)] dt \quad (6)$$

The flow exergy entering the reheater from the high-pressure turbine can be described by a same expression.

$$E_{b,i,ht}=\int m_{b,i,ht}[h_{b,i,ht}-h_a-T_a(s_{b,i,ht}-s_a)] dt \quad (7)$$

The flow exergy leaving the boiler for the high-pressure turbine is given by

$$E_{b,o,ht}=\int m_{b,o,ht}[h_{b,o,ht}-h_a-T_a(s_{b,o,ht}-s_a)] dt \quad (8)$$

The flow exergy leaving the reheater for the intermediate-pressure turbine is given by

$$E_{b,o,it}=\int m_{b,o,it}[h_{b,o,it}-h_a-T_a(s_{b,o,it}-s_a)] dt \quad (9)$$

so the exergy destruction and second law efficiency become,

$$E_{b,des}=S_{b,gen}T_a=E_{fuel}+E_{b,i,hx}+E_{b,i,ht}-E_{b,o,ht}-E_{b,o,it} \quad (10)$$

$$\eta_{II,b} = \frac{E_{b,o,ht} + E_{b,o,it}}{E_{fuel} + E_{b,i,hx} + E_{b,i,ht}} \quad (11)$$

Similar analyses of the exergy balance related to exergy destruction and second law efficiency for other components in this steam power plant are based on equation (5) and the following equation for work output or input.

$$E_w = \int W dt \quad (12)$$

Exergy destruction occurs in the flowing fluid throughout the entire system because of energy and momentum loss. Thermodynamic irreversibility in incompressible water flow only depends on temperature drop (Bejan, 1988).

$$S_{f,l,gen} = \int [mC_p \ln \frac{T_o}{T_i}] dt \quad (13)$$

or when

$$\Delta P \ll \frac{C_p}{V} T_i \quad (14)$$

then

$$S_{f,l,gen} = \int [\frac{mV}{T_i} \Delta P] dt \quad (15)$$

In the case of steam, entropy generated by fluid friction is proportional to the pressure gradient and difference of entropy departure factors.

$$S_{f,v,gen} = \int mR [\ln \frac{P_i}{P_o} + (Z_{s,o} - Z_{s,i})] dt \quad (16)$$

The first term in the above expression is based on the ideal gas

assumption (Bejan, 1988). But ideal gas relations should not be used in steam power plant applications where high pressure superheated steam is usually involved. At high pressure, steam deviates considerably from ideal gas behavior. A correction term including enthalpy departure factors and entropy departure factors must be incorporated (Cengel and Boles, 1989). The second term is added to correct for entropy departure.

When two streams of differing temperature and pressure mix with one another, the accompanying energy and momentum exchange necessarily involves exergy destruction. There are many locales where such mixing occurs in the system. Bejan (1988) gives the mixing entropy generation for incompressible fluid as,

$$S_{m,l,gen} = \int m C_{p,w} \left[\frac{x}{2} (1-x) \left(\frac{T_1 - T_2}{T_1} \right)^2 + \frac{xv}{C_{p,w} T_1} \left(\frac{P_1 - P_3}{P_3} \right) + \frac{(1-x)v}{C_{p,w} T_1} \left(\frac{P_2 - P_3}{P_3} \right) \right] dt \quad (17)$$

where m is total mass flow rate after mixing, x is ratio of mass flow rate from the inlet with temperature T_1 and pressure P_1 over the total mass flow rate. Variables with subscripts 1 and 2 represent two inlet flow while the outlet flow of mixing is expressed with the subscript 3. The corresponding expression for steam is

$$S_{m,v,gen} = \int m \left[C_{p,v} x(1-x) \left(\frac{T_1 - T_2}{T_1} \right)^2 + \frac{xR}{C_{p,v}} \left(\frac{P_1 - P_3}{P_3} \right) + \frac{(1-x)R}{C_{p,v}} \left(\frac{P_2 - P_3}{P_3} \right) \right] + xR(Z_{s,3} - Z_{s,1}) + (1-x)R(Z_{s,3} - Z_{s,2}) dt \quad (18)$$

The first three terms in the above equation are based on the ideal gas assumption (Bejan, 1988). Similar to the expression for entropy generated by fluid friction (steam), the last two term is added to correct for entropy departure. Then, exergy destruction can be obtained by multiplying entropy generation and T_a .

Pumps, of course, contribute themselves to the overall destruction of exergy through the system. Like most steam-turbine cycle calculation, their thermodynamic irreversibilities are neglected. Inlet and outlet flows of the steam seal regulator have the same temperature and pressure, so that flow through the steam seal regulator generates no entropy and the net exergy loss is zero.

ENERGY ANALYSIS

Energy balance is based on counting the energy input, energy output and energy losses which is dominated by the first law of thermodynamics. The value of energy loss is obtained by subtracting energy output from energy input.

$$\text{EnergyLoss} = \text{EnergyInput} - \text{EnergyOutput} \quad (19)$$

The first law efficiency is the ratio of energy output over energy input.

$$\eta_I = \frac{\text{EnergyOutput}}{\text{EnergyInput}} \quad (20)$$

For example, considering energy balance for the boiler.

$$\text{EnergyLoss} = \int \{ (mHV)_{fuel} + (mh)_{b,i,hx} + (mh)_{b,i,ht} - (mh)_{b,o,ht} - (mh)_{b,o,it} \} dt \quad (21)$$

therefore

$$\eta_{I,b} = \frac{\int \{ (mh)_{b,o,ht} + (mh)_{b,o,it} \} dt}{\int \{ (mHV)_{fuel} + (mh)_{b,i,hx} + (mh)_{b,i,ht} \} dt} \quad (22)$$

Similar analyses of the energy balance and first law efficiency for other components in the power plant are based on equation (19) and (20).

RESULTS AND DISCUSSION

The preceding integrals are evaluated numerically for a single day using the heating value of coal, temperatures and pressures provided by Sierra Pacific Power Company. Table 1 presents energy and exergy quantities in the energy cascade and exergy cascade from heat source to heat exchanger 7. Comparing the second columns with the third columns for every component in Table 1, it should be noticed that exergy input is always less than energy input. This

means that thermal energy input is not high quality energy and only part of it is available.

Table 2 shows the first law efficiency and the second law efficiency for components and the plant. This table shows that some components, such as the turbine/generator, have high first law efficiencies but low second law efficiencies. The first law efficiency of the turbine/generator set is almost one hundred percent while its second law efficiency is just 54.4 percent. The reason is that a lot of hot steam flows down to the heat exchangers through bleeds. This bleed-off steam can be considered as the hot stream (source) of the heat exchangers and the output of turbine/generator. These thermal energy outputs are much lower quality of energy than the highly refined energy of electrical output. It is also interesting to note that all heat exchangers have a one-hundred percent first law efficiency but not a one-hundred percent second law efficiency, the difference being caused by exergy destruction due to heat transfer. The heat exchanger numbered one has a lower second law efficiency than the other heat exchangers. This datum, crucial to effective energy management, means that heat exchanger one has a larger availability loss; a loss which escapes the methods and techniques of first law analyses.

After the investigation of exergy destruction, it is found that combustion and heat transfer losses in the boiler are responsible for 53.83 percent of the total exergy destruction. Combustion losses can hardly be reduced with present technology because the conventional fuel oxidation via the highly irreversible combustion process consumes about 30 percent of the usable fuel energy (Dunbar et al, 1991). A possible remedy for this waste of exergy would be the application of fuel cells in future, even though fuel-cell technology for large-scale generation of electrical power remains to be determined. The heat transfer loss causing destruction of exergy is due to the difference between high temperature gas and low temperature water/steam. This loss may be reduced by using high temperature and pressure steam.

Conversion of thermal energy into mechanical energy then electrical energy accounts for 41.34 percent of total exergy destruction. Entropy is generated by the expansion of vapor from high temperature and pressure to low temperature and pressure. The magnitude of entropy generated depends on turbine isentropic efficiency.

Heat rejection from steam to the atmosphere is responsible for 2.89 percent of the total exergy destruction. This is due to heat transfer between steam and its ambient surroundings. Lowering the condenser pressure can, of course, reduce temperature differences between working fluid and the atmosphere so that entropy generated in heat rejection can be reduced, but the technical and economical feasibility of doing this should be considered together.

Friction losses account for 0.50 percent of the total exergy destruction. Although it poses a small fraction of the total exergy destruction, it should be noticed that this loss is much bigger than the losses caused by the boiler feed pump turbine, flow

mixing, and any of the heat exchangers in the system. The friction losses may be reduced by installing smooth surface transportation pipes and better insulation on pipes so that pressure and temperature drops are reduced.

The loss caused by the boiler feed pump turbine is responsible for 0.25 percent of the total exergy destruction. Mixing losses account for 0.23 percent of the total exergy destruction. The mixing losses may be reduced by selecting two streams with smaller pressure and temperature difference for mixing purpose. The gland steam condenser, drain cooler, deaerator and other heat exchangers are collectively responsible for 0.96 percent of the total exergy destruction. Theoretically, the irreversibility of heat exchangers depends on two factors, heat transfer across the temperature difference between the hot and cold streams and the pressure drop caused by friction. Large heat exchangers may have a lower exergy destruction rate because they have more heat transfer area and more heat transfer.

CONCLUSION

A methodology is presented to calculate the exergy delivered and the exergy destroyed by operation of a conventional steam power plant. It is shown that combustion and heat transfer inside the boiler and conversion of thermal energy to electricity are responsible for most of the exergy destruction. Heat loss from the condenser makes the next largest contribution. Flow with friction, the boiler feed pump turbine and flow mixing manifest a very small fraction of the total exergy destruction. Heat transfer across temperature differences and frictional pressure drop involved with the gland steam condenser, drain cooler, deaerator and heat exchangers also reduce delivered exergy. The second law analysis is a powerful tool of thermodynamic research for power plants and other thermal systems.

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Table 1. Energy and Exergy Quantities

COMPONENTS	ENERGY INPUT (MJ)	EXERGY INPUT (MJ)
Boiler	1.51270 E08	8.58632 E07
Turbine/generator	1.29420 E08	5.38873 E07
BFPT	2.71932 E06	776627
Condenser	3.30799 E07	1.75705 E06
GSC	2.77810 E06	58312
DC	2.97503 E06	64511
#1 Heater	5.06362 E06	368189
#2 Heater	6.42929 E06	557701
#3 Heater	9.83891 E06	1.34524 E06
#4 Heater	1.29291 E07	2.16142 E06
#5 Heater	1.53978 E07	2.69885 E06
#6 Heater	1.99722 E07	4.13567 E06
#7 Heater	2.31962 E07	5.59951 E06

Table 2. Efficiencies

COMPONENTS	η_I	η_{II}
Boiler	85.6%	62.8%
Turbine/generator	99.7%	54.4%
BFPT	100%	80.6%
Condenser	8.1%	2.4%
GSC	100%	91.6%
DC	100%	93.0%
#1 Heater	100%	73.4%
#2 Heater	99.9%	91.6%
#3 Heater	100%	92.0%
#4 Heater	100%	95.8%
#5 Heater	100%	98.9%
#6 Heater	100%	98.0%
#7 Heater	100%	98.0%
Total Plant	42.8%	37.8%

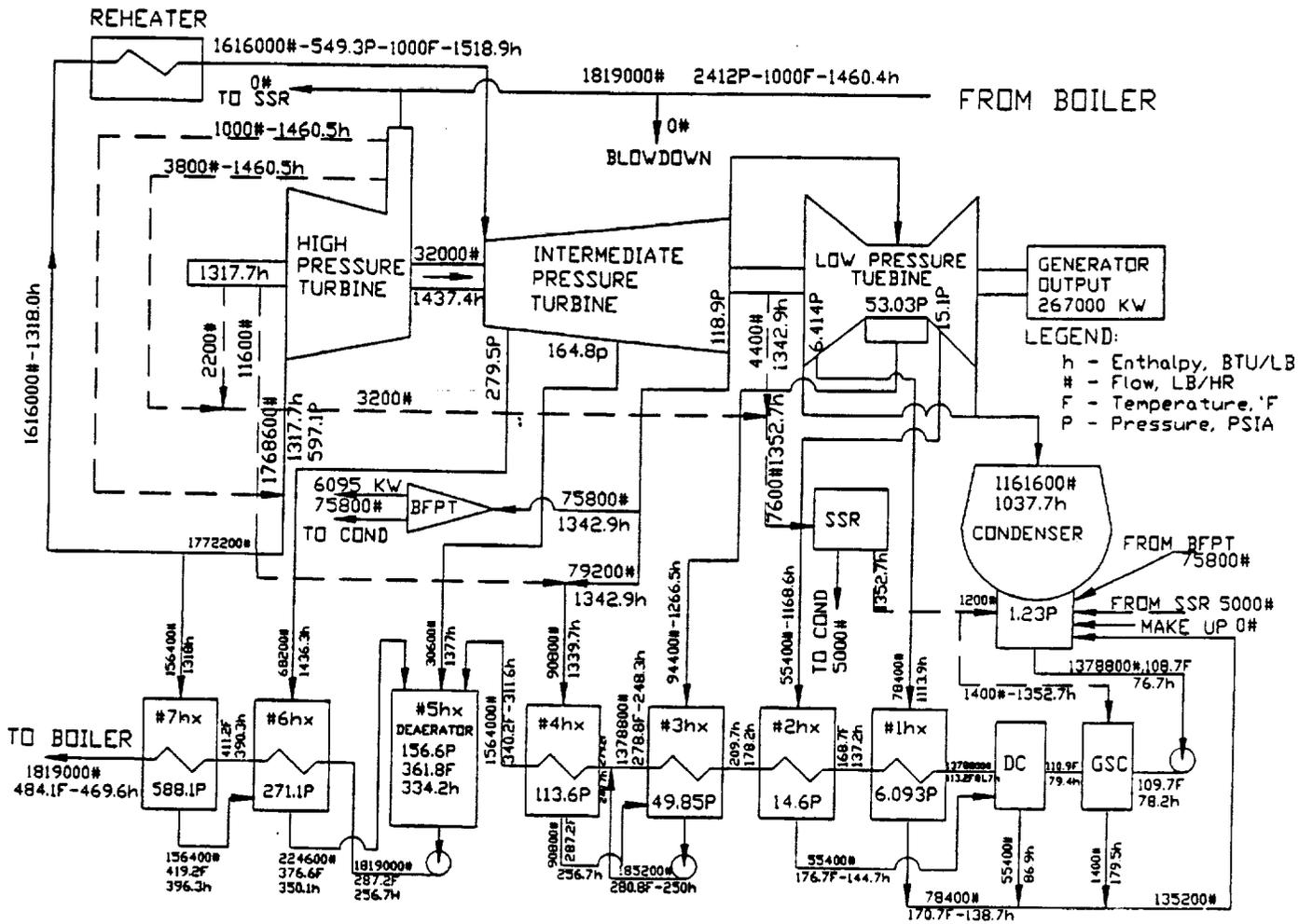


Fig.1 SYSTEM SCHEMATIC

Session Three

Code Innovations

