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Optimal Design of a Cogeneration System in a Kraft Process using Genetic Algorithm

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Cogeneration is one of the best methods for energy saving which makes a better use of fuels by using recovered heat and producing heat and power simultaneously. In this study, the implementation of a cogeneration system (CHP) integrated with an industrial pulp and paper mill is studied. The cogeneration system consists of an air compressor, a combustion chamber, a gas turbine and a single pressure heat recovery steam generator (HRSG) with supplementary firing which are integrated with a back-pressure steam turbine available in the mill. This system is designed for producing 35 MW of power required in the mill and 192 t/h of superheated steam at 61 bar required for driving the steam turbine. Aspen plus software is used to simulate the proposed CHP system.
Afterwards, genetic algorithm (GA) is used for optimization of this cycle using both thermodynamic and thermo-economic models. Two objective functions are considered which are Total Annual Cost (TAC) of the cogeneration system to be minimized and exergy efficiency of this system to be maximized and five decision variables are considered in the optimization procedure. By applying the proposed CHP system both power and heating requirements of the pulp and paper mill can be supplied with a payback of 1.1 year. Also, it is shown that by applying GA method, TAC of the CHP system is improved by 18.5 % compared to one resulted from general simulation.

1. Case study– Kraft process

The pulp and paper industry is a very large energy consumer industry, in the form of electricity and heat for pulp drying, liquor evaporation and other operations which represent a high potential for cogeneration (Mostajeran-Goortani et al., 2010). The Kraft process is a chemical process in which the paper pulp is produced by wood chips. Paper pulp is an important source for producing many kinds of paper products. In this work we consider an Iranian pulp and paper mill as case study. The mill requires 155 t/h steam at 12 bar for process applications and 35 MW power for whole of the plant.

1.1 Cogeneration system– Scheme proposal

In existing process, there is a back pressure steam turbine (22.5 MW), which requires superheated steam at 443 °C and 61 bar to work. However, there is no boiler to produce steam with this condition. In the other hand, the power generated by steam turbine is not enough for the plant and the rest must be supplied by the grid. Therefore, it is reasonable to implement a gas turbine cogeneration system to supply the rest of power demand and produce required steam for the steam turbine. The gas turbine is driven by natural gas. The hot exhaust gases from the turbine enters to a HRSG system and the output high pressure steam derives the steam turbine, which produces power and delivers steam at a lower pressure that could be used for process applications. The proposed CHP system is shown in Figure 1.

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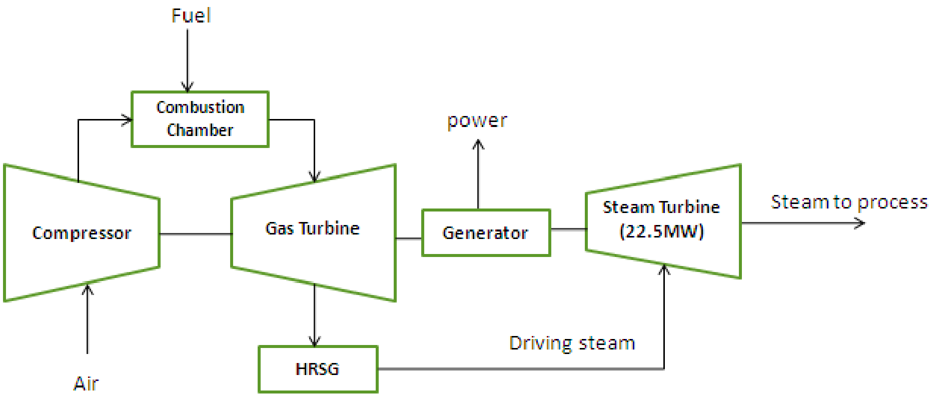

Figure 1: Proposed CHP system

1.2 Simulation of proposed cogeneration system

In this work, a CHP cycle is proposed to integrate with an existing back pressure steam turbine. The back pressure steam turbine requires 192 t/h of superheated steam at 443 °C and 61 bar to work. Hence, by applying the cogeneration system the required steam could be produced and the rest of power demand could be supplied.

To simulate the CHP cycle the Aspen Plus software is used. 262.8 t/h of air after compression in the compressor with 4.45 t/h natural gas ignites in the combustion chamber of the gas turbine. The flue gases at 1071.3 °C are expanded to 582.6 °C and 1.1 bar and 16.5 MW of electricity is produced. The HRSG system which is considered to be single pressure consists of an economizer, an evaporator and a super-heater equipped with a duct burner for supplementary firing. Flue gas from gas turbine plus 9.86 t/h natural gas, in the HRSG system produce 192 t/h steam at 443 °C and 61 bar, which is sent to the back pressure steam turbine. In this turbine the high pressure steam is expanded to 12 bar and 254.9 °C and 18.5 MW of electricity is produced. The 192 t/h produced steam with lower pressure supplies the steam demand of the mill (155 t/h) and the remainder can be sent to another condensate steam turbine to produce power for selling to the grid, which is not considered in this study. Simulation of the cycle is shown in Figure 2.

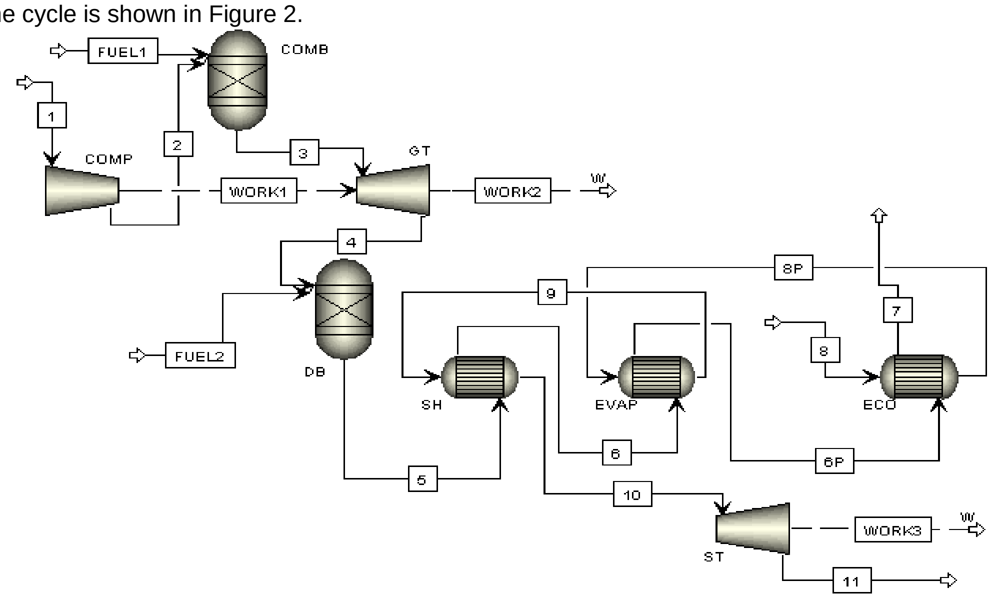

Figure 2: Simulation of the CHP cycle

2. Optimization of proposed cogeneration system

The aim of energy system optimization is modifying the system design according to one or more defined objectives. Toffolo et al. (2002) considered a multi-objective optimization with energetic and economic objectives in a benchmark cogeneration system using evolutionary algorithms. Here, the optimization of proposed CHP system is carried out based on their work. As it described before, the proposed cogeneration cycle is designed to supply the whole of power demand (35 MW) and the required steam for steam turbine. The aim of optimization step is getting a cogeneration cycle for supplying energy demands with minimum TAC and maximum exergy efficiency, which is carried out by varying some effective thermodynamic variables of the cycle using genetic algorithm. Dependent parameters like temperatures, pressures and mass flow rates are obtained again from optimization.

2.1 Thermodynamic model

According to the simulation of proposed cogeneration system, following thermodynamic equations are used to model the cycle. The power produced by steam turbine resulted from simulation is 18.5 MW and considered to be fixed during optimization. So the net power of gas turbine cycle has to be 16.5 MW, for supplying the mill power demand.

Some assumptions are used during the optimization, which are described below:

- Air is considered as ideal gas with constant specific heat and pure methane is assumed as fuel.
- Environmental conditions of the inlet air and also basic conditions for enthalpy and exergy calculations are: P_0 = 1.013 bar, T0 = 298 K.
- Pressure drops in combustion chamber and HRSG system are considered 0.03 and 0.05 of combustion gases, respectively.

Thermodynamic parameters and variables and also fixed parameters of the cycle are presented in Tables 1 and 2.

Table 1: Thermodynamic variables

State	1	2	3	4	5	6	6p	7	8	8p	9	10
P (bar)	1.01	P2	P3	1.1	1.06	-	-	1.01	61	61	61	61
T (K)	298	T2	T3	T4	T5	T6	T6p	T7	298	539.8	549.8	716

Table 2: Fixed variables

Material	Fixed variables
Air	Cp_{air} = 1.004 kJ/kgK, γ_{air} = 1.4
Fuel	LHV= 50,000 kJ/kg, Ex_{fuel} = 51,850 kJ/kg
Gas	γ_{gas} = 1.33
Steam	$\dot{m}_{steam}$= 48.38 kg/s, $h_{10} - h_8$=3.2 MJ/kg, $h_{10} - h_9$=0.33 MJ/kg, $h_{10} - h_{8p}$= 1.96 MJ/kg, $Ex_9 - Ex_7$= 909.8 kJ/kg

For compressor we have:

$$\frac{T_2}{T_1} = 1 + \frac{r_c^{\frac{\gamma_{air}-1}{\gamma_{air}}} - 1}{\eta_c}; \; r_c = \frac{P_2}{P_1}; \; W_c = \dot{m}_{air} \times Cp_{air} \times (T_2 - T_1) \quad (1)$$

For combustion chamber:

$$P_3 = (1 - \Delta P_{cb})\, P_2; \; \dot{m}_{gas1} = \dot{m}_{fuel1} + \dot{m}_{air} \quad (2)$$

$$\dot{m}_{air} \times Cp_{air} \times (T_2 - T_0) + \eta_{cb} \times \dot{m}_{fuel1} \times LHV = \dot{m}_{gas1} \times Cp_{gas1} \times (T_3 - T_0) \quad (3)$$

For gas turbine:

$$\frac{T_4}{T_3} = 1 - \eta_t \left(1 - \left(\frac{P_3}{P_4}\right)^{\frac{1-\gamma_{gas1}}{\gamma_{gas1}}}\right); \; W_{gt} = \dot{m}_{gas1} \times Cp_{gas1} \times (T_3 - T_4); \; W_{net} = W_{gt} - W_c \quad (4)$$

For HRSG system with supplementary firing:

$$\dot{m}_{gas1} + \dot{m}_{fuel2} = \dot{m}_{gas2} \quad (5)$$

$$\dot{m}_{gas} \times Cp_{gas1} \times (T_4 - T_0) + \eta_{cb} \times \dot{m}_{fuel2} \times LHV = \dot{m}_{gas2} \times Cp_{gas2} \times (T_5 - T_0) \quad (6)$$

$$\dot{m}_{gas2} \times Cp_{gas2} \times (T_5 - T_7) = \dot{m}_{steam} \times (h_{10} - h_8) \quad (7)$$

$$\dot{m}_{gas2} \times Cp_{gas2} \times (T_5 - T_6) = \dot{m}_{steam} \times (h_{10} - h_9) \quad (8)$$

$$\dot{m}_{gas2} \times Cp_{gas2} \times \left(T_5 - T_{6p}\right) = \dot{m}_{steam} \times (h_{10} - h_{8p}) \quad (9)$$

The Cp of combustion gases is considered as a function of temperature variable and is calculated according to the following equation (Kurt et al., 2009):

$$\mathrm{Cp(T)} = 0.991615 + \left(\frac{6.99703\mathrm{T}}{10^5}\right) + \left(\frac{2.7129\mathrm{T}^2}{10^7}\right) - \left(\frac{1.22442\mathrm{T}^3}{10^{10}}\right) \quad (10)$$

2.2 Thermo-economic model

The economic model is consists of the cost of all individual components, maintenance cost and the cost of fuel consumption. These cost equations have to be expressed as functions of thermodynamic variables which are the optimization parameters. The capital costs of compressor (C_{comp}), combustion chamber (C_{cb}), gas turbine (C_{GT}) and HRSG system (C_{HRSG}) are determined from (Baghernejad and Yaghoubi, 2011):

$$C_{comp} = \left(\frac{75 \times \dot{m}_{air}}{0.9 - \eta_c}\right) \times r_c \times \ln r_c \quad (11)$$

$$C_{cb} = \left(\frac{48.64 \times \dot{m}_{air}}{0.995 - \frac{P_3}{P_2}}\right)(1 + \exp(0.018T_3 - 26.4)) \quad (12)$$

$$C_{GT} = \left(\frac{1536 \times \dot{m}_{gas1}}{0.92 - \eta_t}\right) \times \log\left(\frac{P_3}{P_4}\right) \times (1 + \exp(0.036T_3 - 54.4)) \quad (13)$$

$$C_{HRSG} = 4745\left\{\left(\frac{Q_{ec}}{(\Delta T_{lm})_{ec}}\right)^{0.8} + \left(\frac{Q_{ev}}{(\Delta T_{lm})_{ev}}\right)^{0.8} + \left(\frac{Q_{sh}}{(\Delta T_{lm})_{sh}}\right)^{0.8}\right\} + 11820\,\dot{m}_{steam} + 658\dot{m}_{gas2}^{\;1.2} \quad (14)$$

To determine the duct burner investment cost (C_{db}), the following relation can be used (Charles, 2003):

$$C_{db} = \eta_{combustion} \times 7000 \times LHV \times \dot{m}_{fuel2} \quad (15)$$

And fuel cost can be estimated by (Dincer and Ahmadi, 2011):

$$C_f = 0.000003 \times LHV \times (\dot{m}_{fuel1} + \dot{m}_{fuel2}) \quad (16)$$

2.3 Objective functions

A multi-objective optimization is carried out in this work including two objective functions of TAC and exergetic efficiency. The minimization of TAC of the cogeneration cycle and maximization of the exergetic efficiency of this cycle is considered.

$$TAC = \sum C_i \times AF \times \emptyset_r + C_f \times 3600 \times N; \quad AF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (17)$$

$$\eta_{ex} = \frac{W_{net} + \dot{m}_{steam}(Ex_9 - Ex_7)}{\dot{m}_{fuel} \times Ex_{fuel}} \quad (18)$$

Five decision variables are considered through the optimization procedure including compressor pressure ratio (r_c), isentropic efficiency of compressor (η_c), isentropic efficiency of gas turbine (η_t), temperature of combustion gases entering the gas turbine (T_3) and the fuel mass flow rate to the HRSG supplementary firing system ($\dot{m}_{fuel2}$).

$$7 < r_c < 20;\ 0.7 < \eta_c < 0.9;\ 0.7 < \eta_t < 0.9;\ 1000\ K \le T_3 \le 1500\ K;\ 1 \le \dot{m}_{fuel2} \le 5 \quad (19)$$

For heat exchange in HRSG system, there are some constraints, which must be satisfied during the optimization:

$$\Delta T_p = T_{6p} - T_9 > 0;\ T_6 \ge T_9 + \Delta T_p;\ T_7 \ge T_8 + \Delta T_p;\ T_7 > 375 \quad (20)$$

3. Results of optimal design

The number of adjustable variables is 5. An initial population of Chromosomes is randomly generated. The population in each generation is taken as 100. The results of optimal design found by the GA are presented in Tables 3 to 5.

Table 3: Optimal values for decision variables and objective functions

Parameter	Value
r_c	19
η_c	0.87
η_t	0.89
T_3(K)	1398
$m°_{fuel2}$(kg/s)	2.01
TAC($/year)	13.8×10^6
η_{ex}	0.51

Table 4: Optimal values for dependent parameters

Parameter	Value	Parameter	Value
$m°_{air}$(kg/s)	54.9	P_3(bar)	18.5
$m°_{gas1}$(kg/s)	55.9	T_2 (°C)	749.5
$m°_{fuel1}$(kg/s)	0.96	T_4 (°C)	765
$m°_{gas2}$(kg/s))	57.9	T_5 (°C)	1891
W_c(MW)	24.9	T_6 (°C)	1706
W_{gt}(MW)	41.4	T_{6p} (°C)	512
P_2 (bar)	19.5	T_7 (°C)	261.8

Table 5: Comparison between cogeneration system before and after optimization

Cost of designed CHP ($/y)	Cost of optimized CHP ($/y)	Improvement%
16.95×10^6	13.8×10^6	18.5

The electricity and steam costs for this plant are about 33 $/MWh and 8.1 $/t, respectively, and the cost of cooling water is taken as 1.2 $/m^3. By considering 8000 hours for annual operating time of the plant, the simple payback time describing by SPB= Investment/Yearly savings, can be calculated. Economic evaluation of the proposed cogeneration system implementation is shown in Table 6.

Table 6: Economic evaluation of the cogeneration system implementation

Net saving (M$/y)= 6.9			TAC of CHP system (M$/y)= 13.8	
Electricity (MW)	Steam (t/h)	Fuel and water cost (M$/y)	Installation cost of CHP system (M$)	Payback (y)
35	192	14.7	7.2	1.1

4. Conclusions

In this work, we have performed an optimum design of a cogeneration system integrated with a pulp and paper mill. For the cycle simulation, Aspen Plus software was used. We have considered 5 variables related to the thermodynamic model to get the best design of the proposed cogeneration system with minimum cost and maximum exergy efficiency using GA method for optimization.

By comparison between the cost resulted by GA (13.8×10^6 $/y) and the one resulted from general simulation by Aspen Plus software before optimization (16.95×10^6 $/y), it was found that by this optimum design; TAC is reduced by 18.5 %. Also, the economic evaluation showed that by implementation of the cogeneration system both power and heating requirements of the pulp and paper mill can be supplied with a payback of 1.1 y, which shows that the cogeneration system application is a cost attractive option.

Nomenclature

AF: Annual factor
C: Compressor
cb: Combustion chamber
C_i: Component investment cost ($)
CHP: Combined Heating and Power
C_p: Heat capacity (kJ/kg°C)
C_f: Fuel cost
Ex: Exergy
db: Duct burner
i: Annual interest rate
h: Specific enthalpy (kJ/kg)
LHV: Lower heating value (kJ/kg)
$\dot{m}$: Mass flow rate (kg/s)
N: Operating hour per year
n: CHP plant lifetime
P: Pressure
ΔP: pressure drop
r_c: Compressor pressure ratio
TAC: Total annual cost
T: Temperature
ΔT_{lm}: Log mean temperature difference
ΔT_P: Pinch point temperature difference
W: Power
γ: Specific heat ratio
η_c: Compressor isentropic efficiency
η_{gt}: Gas turbine isentropic efficiency
η_{ex}: Exergetic efficiency
Ø: Maintenance factor

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