

# On the Teaching of Performance Evaluation and Assessment of a Combined Cycle Cogeneration System

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*Performance evaluation and assessment of combined cycle cogeneration systems are not taught well in academia. One reason is these parameters are scattered in the literature with each publication starting and ending at different stages. In many institutions professors do not discuss or even mention these topics, particularly from a second law perspective. When teaching combined cycle cogeneration systems to undergraduates, the professor should introduce pertinent parameters in a systematic fashion and discuss the usefulness and limitations of each parameter. Ultimately for a given situation, the student should be able to determine which parameters form the most appropriate basis for comparison when considering alternative designs. This paper provides two approaches, one based on energy (the first law of thermodynamics) and the other based on exergy (the second law of thermodynamics). These approaches are discussed with emphasis on the “precise” teaching of the subject matter to undergraduates. The intent is to make coverage of the combined cycle cogeneration systems manageable so that professors can appropriately incorporate the topic into the curricula with relative ease.*  
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## 1 Introduction

The United States Department of Energy [1] lists 5267 electrical power plants operating in the United States, providing a nameplate electrical capacity of 1067 GW. Only 19.5% of this capacity is credited to the 466 power plants operating as a combined cycle, which produced  $6.21 \times 10^8$  MW h, or 15.3% of the net generated total in 2006. The majority of existing power plants are not based on a combined cycle, but the majority of recently commissioned power plants are. The low capital costs and high thermal efficiency of combined cycle plants, which are fueled in 99.0% of cases by natural gas, make these power plants attractive in comparison to others listed in Figs. 1 and 2.

Advances in combined cycles can raise their thermal efficiency to 51% without increasing capital or operating costs [1]. This helps combined cycles remain competitive even in light of recent high natural gas prices. Coal has remained relatively inexpensive and continues to fuel lower efficiency power plants.

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The previous efficiencies do not include useful thermal output, as obtained from combined heat and power (CHP) plants. In 2006, CHP plants produced  $3.22 \times 10^8$  MW h, or 7.9% of the United States’ total electrical output, while also producing  $5.22 \times 10^8$  MW h ( $1.78 \times 10^{15}$  Btu) of useful thermal output, mostly for industrial processes. For comparison, the total United States energy consumption in 2006 was  $2.93 \times 10^{10}$  MW h ( $1.00 \times 10^{17}$  Btu) [2]. Hence, a combined cycle power plant that also produces useful thermal output makes more complete use of the fuel source.

Since the  $\$2.77 \times 10^{11}$  United States electric utility industry [1] relies more extensively on the combined cycle cogeneration system, undergraduate thermodynamics education should include introduction to such systems and the associated performance parameters. Performance parameters may be based on either the first law of thermodynamics or the second law of thermodynamics, thus many parameters may be defined to analyze a combined cycle cogeneration system. Although this type of system is comprehensive, teaching and understanding the processes are indeed very valuable. The overall intent of a combined cycle cogeneration system is to produce electrical power; process energy obtained is an added benefit.

Various thermodynamics textbooks are available to teach undergraduate thermodynamics [3–8]. These textbooks provide only a simple introduction to combined cycle cogeneration systems, whereas Ref. [9] emphasizes exergo-economics in the analysis of combined cycle cogeneration systems. Most, if not all, undergraduate courses in thermodynamics introduce first law analysis; however, most programs do not introduce exergetic analysis based on the second law. This paper presents performance criteria for a combined cycle cogeneration system using both first law and second law analyses. These analyses will allow better assessment of the characteristics of a combined cycle cogeneration power plant. The intent is to provide an example that thermodynamics educators can integrate into their courses.

The sign convention, heat in positive and work in negative, is applied in this presentation consistently [10]. Sign convention is not a trivial issue as consistency of its application in analysis is paramount. This is one of the points that cause confusion to the student in the classroom, as well as during tests due to the inconsistencies that exist in the textbooks mentioned above. Precision and consistency are very important and cannot be trivialized.

Technical data for the gas turbine (GT) are from Ref. [11] for the generator drive PG9351(FA) and are given in Table 1. Use of data from an actual gas turbine shows the student that real industrial gas turbines found in literature can and should be analyzed to bring reality to the classroom. This engine is a modern one on which a combined cycle cogeneration structure can be built. This is the intent of this paper so that complete first and second law analyses—energy versus exergy—can be performed to better understand the details and the meaning of the results. Should time permit, thermo-economics can also be integrated. The appropriate choices are, of course, at the discretion of the individual instructor.

## 2 Combined Cycle Cogeneration System

A typical combined cycle cogeneration system is shown in Fig. 3. The topping cycle is an open Brayton cycle consisting of a compressor, combustion chamber, and gas turbine. A generator converts output shaft power to electricity. Hot exhaust from the topping cycle provides energy to superheat steam in the Rankine cycle-based bottoming cycle. Instead of a conventional boiler that burns fuel, a heat recovery steam generator (HRSG) that includes an economizer, evaporator, and superheater converts compressed liquid water exiting the pump into superheated steam that drives the steam turbine. Within the HRSG a minimum pinch of  $20^\circ\text{C}$  is required to ensure proper thermal exchange. Pinch technology, its use and value are covered in the course that is presented. If this is not the case, it is suggested that it be taught by the instructor who

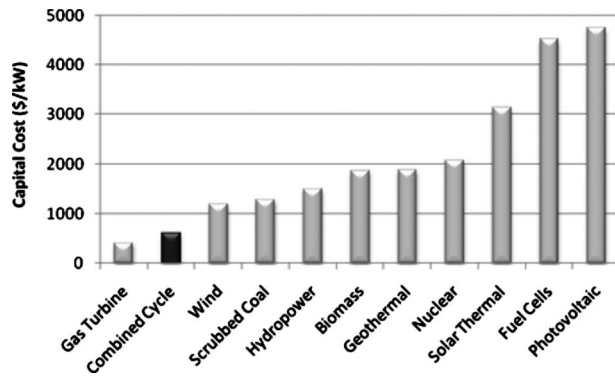


Fig. 1 Capital cost of power plants [1]

wants to use these results. The pinch diagram for the HRSG is shown in Fig. 4. A generator connected to the steam turbine converts shaft power output into electricity. In lieu of a conventional condenser that rejects heat to the environment, heat rejection from the steam exiting the turbine in the condenser provides thermal energy for a useful purpose (process heat).

The combined cycle produces power from the gas turbine and the steam turbine, while power is required to operate the compressor and the pump. As shown in Eq. (1), the net power produced by the system is the sum of all power associated with the system: gas turbine power output (positive value), compressor power input (negative value), steam turbine power output (positive value), and pump power input (negative value)

$$\dot{W}_{net} = \dot{W}_{gas\ turbine} + \dot{W}_{compressor} + \dot{W}_{steam\ turbine} + \dot{W}_{pump} \quad (1)$$

Energy associated with fuel is added to the system in the combustion chamber of the topping cycle. The energy rate associated with the fuel is the product of the lower heating value (LHV) of the fuel and the mass flow rate of the fuel, as shown in Eq. (2). The LHV is used instead of the higher heating value (HHV) since its use would give a more conservative result; however, to make the discussion even better, both definitions can be incorporated and the students can see which is a better definition

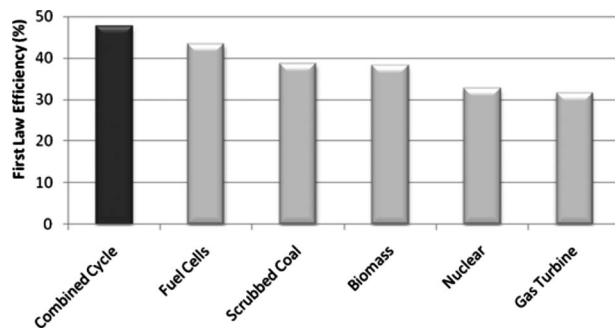


Fig. 2 First law efficiency of power plants operating on purchased fuels [1]

Table 1 Characteristics of the gas turbine used: PG9351(FA) [11]

International Standards Organization (ISO) base rating (kW)	255,600
Heat rate of gas turbine (kJ/(kW h))	9757
Exhaust flow (kg/h)	$2.318 \times 10^6$
Exhaust temperature (°C)	608
Pressure ratio	15.3

$$\dot{E}_{fuel} = \dot{m}_{fuel} LHV \quad (2)$$

Assuming a combustor efficiency to account for losses during conversion of chemical energy of the fuel to thermal energy, the heat transfer rate into the combustion chamber (positive value) is determined by multiplying the combustor efficiency and the fuel energy rate, as shown in Eq. (3)

$$\dot{Q}_{combustor} = \eta_{combustor} \dot{E}_{fuel} \quad (3)$$

Heat rejected from the steam in the condenser (negative value) is the input thermal energy for process heating (positive value). This relationship is shown in Eq. (4)

$$\dot{Q}_{process} = -\dot{Q}_{condenser} \quad (4)$$

To fully describe the total system, performance parameters for the system and its subcomponents must be included. Therefore, below are given those that make most sense with respect to the gas turbine system, the HRSG, the steam turbine system using the first and second laws of thermodynamics and the energy and exergy for the process heat, the work terms, and other quantities of interest. It is important to remember that the intent of this paper is to present concise and precise performance parameters for the combined cycle cogeneration system based on the first and second laws of thermodynamics so that those who wish can use them in the classroom as they see fit.

### 3 First Law-Based Performance Criteria

Based on the first law of thermodynamics, the following performance criteria may be defined. The choice is up to the instructor. Here suggestions are made as to the value or lack thereof for each so that appropriate choices can be made by the instructor for the course. If possible, all should be discussed so that the student can appreciate the importance for each and comparative value in each.

**3.1 Thermal Efficiency ( $\eta_{thermal}$ ).** Thermal efficiency compares the net power produced by the system to the fuel energy rate input to the system, as shown in Eq. (5). Although it is an important quantity, thermal efficiency does not identify losses in utilization of fuel energy

$$\eta_{thermal} = \frac{\dot{W}_{net}}{\dot{Q}_{combustor}} \quad (5)$$

**3.2 Power to Heat Ratio ( $R_{power\ to\ heat}$ ).** The power to heat ratio compares the electrical power output from the system to the heat output used as process heat, as shown in Eq. (6). The primary purpose of a combined cycle cogeneration system is to produce as much electrical power as possible from the perspective of an industrial user, who can produce electricity at a lower cost than purchasing it from a utility, and to use all the electricity produced or to sell the excess back to the utility. However, in situations where the electricity produced will sometime exceed the electric load on site and nothing is paid by the utility for excess electricity fed back into the grid, then the optimal power to heat ratio may not necessarily be the largest value possible. The case in which excess electricity cannot be fed back to the electric power grid might be an interesting exercise for the students. Process heat is a cycle by-product that is useful, but is not the primary desired output. Process heat is the final effort to obtain useful energy from the cycle input fuel energy. Since the ultimate desired output is electrical power, the power to heat ratio should be as large a value as possible since the cycle should not produce a large amount of process heat

$$R_{power\ to\ heat} = \frac{\dot{W}_{net}}{\dot{Q}_{process}} \quad (6)$$

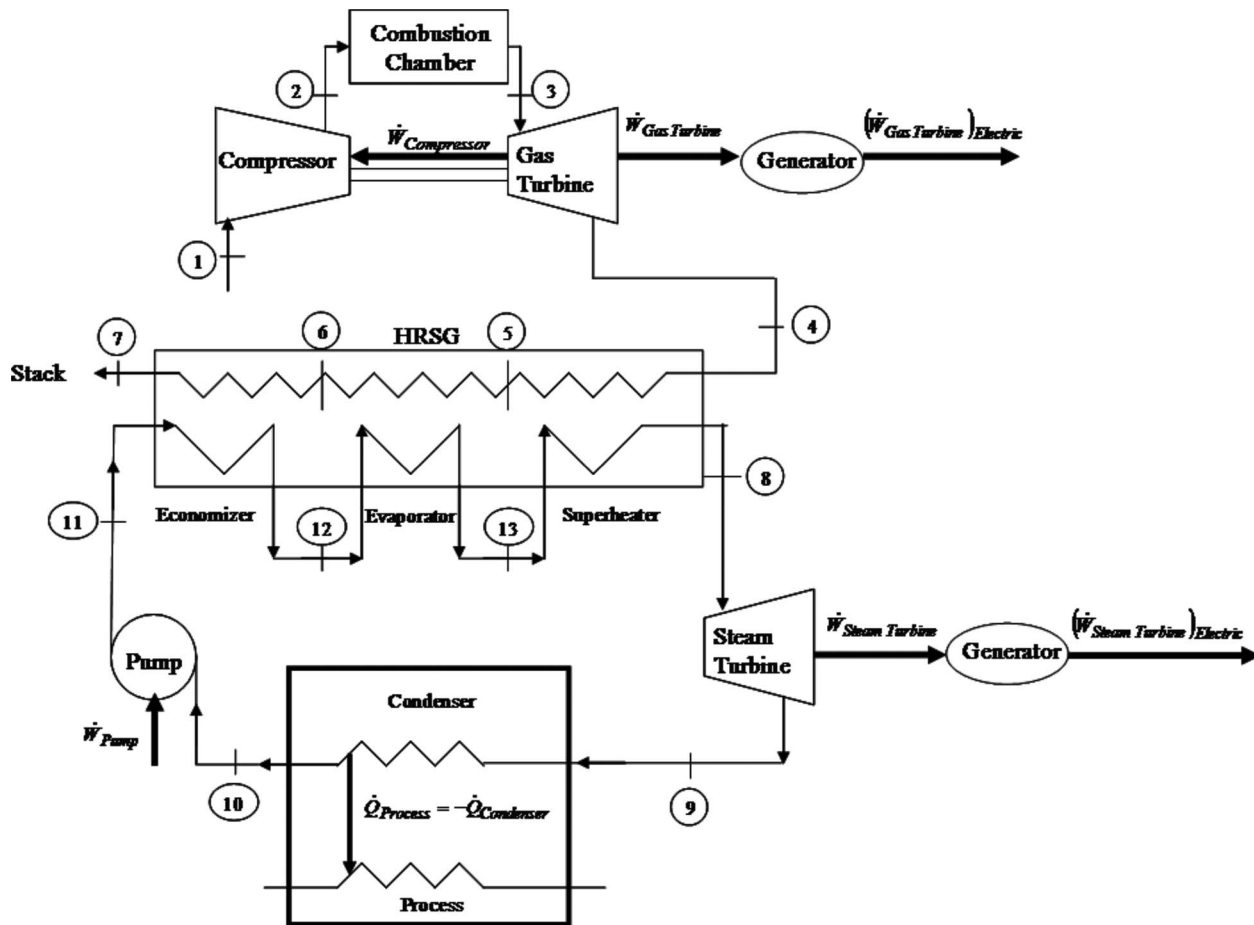


Fig. 3 Combined cycle cogeneration system

**3.3 Utilization Factor ( $F_{utilization}$ ).** The utilization factor, as shown in Eq. (7), compares useful output from the system to combustor heat transfer rate input to the system. Since power output is the primary desired useful output from the system, a high

utilization factor is significant *only if* net power produced is significantly greater than process heat provided. There could be cases, however, where a large process heat is desired in the application. For this case, a lower value may be obtained as long as the

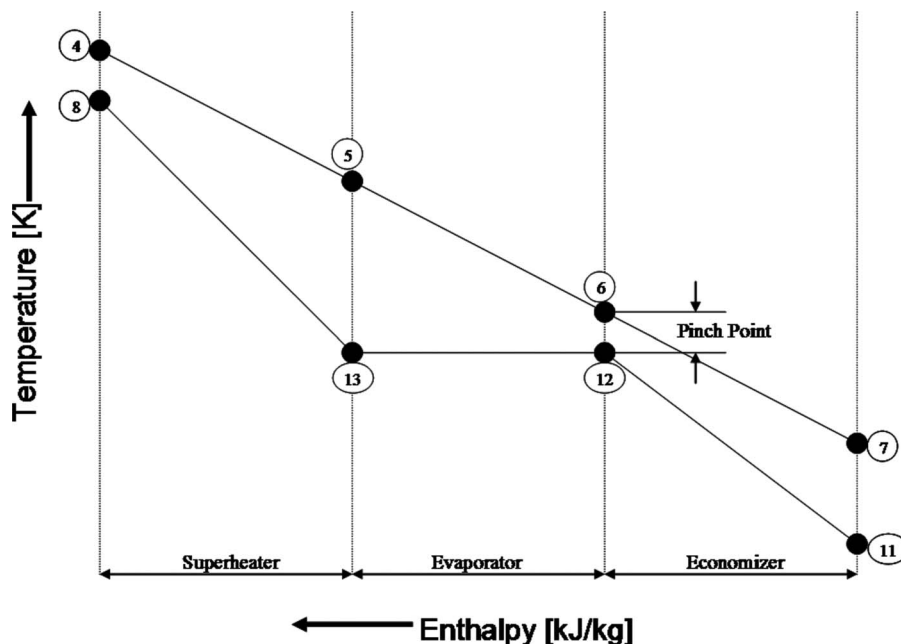


Fig. 4 Heat recovery steam generator pinch diagram

situation that is being studied is kept in perspective

$$F_{\text{utilization}} = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{combustor}}} \quad (7)$$

**3.4 Backwork Ratio ( $R_{\text{backwork}}$ ) for the Gas Turbine.** The backwork ratio indicates the percent of power produced by the gas turbine used to power the compressor, as shown in Eq. (8). A low value is desired since the compressor should not use a large amount of the produced turbine work. However, thermodynamically speaking, since the specific volume is large in gases, the compressor work will also be large compared with a water pump in a steam cycle. High compressor isentropic efficiency will reduce the backwork ratio

$$R_{\text{backwork}} = \frac{-\dot{W}_{\text{compressor}}}{\dot{W}_{\text{gas turbine}}} \quad (8)$$

**3.5 Specific Fuel Consumption (SFC) (Gas Turbine).** The specific fuel consumption indicates the fuel flow rate needed to produce the net power from only the topping (gas turbine) cycle, as shown in Eq. (9). Since the fuel used in the gas turbine supports the total system, specific fuel consumption provides important information about the fuel used per net power output of the gas turbine. A reduction in this quantity is necessary since this will reduce the cost of electricity generated and supplied to the customer, which should be the ultimate goal of power generation

$$SFC = \frac{\dot{m}_{\text{fuel}}}{\dot{W}_{\text{gas turbine}} + \dot{W}_{\text{compressor}}} \quad (9)$$

**3.6 The Heat Recovery Steam Generator Effectiveness ( $\epsilon_{\text{energy}}$ ).** Energy based heat recovery steam generator effectiveness compares the rate of energy transfer provided to the steam cycle to the energy rate that was available in the HRSG, as shown in Eq. (10). This is an important quantity, though based on the first law, in that it shows how well the energy exchange can take place in the HRSG between the exhaust of the gas turbine cycle and the water that is superheated for the steam cycle

$$\epsilon_{\text{HRSG}} = \frac{\dot{Q}_{\text{supplied steam cycle}}}{\dot{Q}_{\text{combustor}} - (\dot{W}_{\text{gas turbine}} + \dot{W}_{\text{compressor}})} \quad (10)$$

**3.7 Power Generation Efficiency ( $\eta_{\text{power generation}}$ ).** Power generation efficiency compares the power produced by the system to the net heat transfer into the system on a fuel energy basis, as shown in Eq. (11). This quantity may be of limited value since it refers to *fuel utilization efficiency* and thus the requirement for a large value of *power to heat ratio*. It should be analyzed in this context

$$\eta_{\text{power generation}} = \frac{\dot{W}_{\text{net}}}{\left( \dot{Q}_{\text{combustor}} - \frac{\dot{Q}_{\text{process}}}{\epsilon_{\text{HRSG}}} \right)} \quad (11)$$

The denominator is sometimes referred to as the *process fuel credit* [12].

**3.8 Fuel Chargeable to Power (FCP).** Fuel chargeable to power identifies how much of the fuel actually contributes to net power generation. It compares the net heat transfer into the system on a fuel energy basis to the net power produced by the system, also known as *net heat rate* [13]. As shown in Eq. (12), fuel chargeable to power is the reciprocal of power generation efficiency. Fuel chargeable to power is important since power generation is the primary desired output from the system

$$FCP = \frac{\left( \dot{Q}_{\text{combustor}} - \frac{\dot{Q}_{\text{process}}}{\epsilon_{\text{HRSG}}} \right)}{\dot{W}_{\text{net}}} \quad (12)$$

$$FCP = \frac{1}{\eta_{\text{Power Generation}}} \quad (12)$$

**3.9 Economic Efficiency ( $\eta_{\text{economic}}$ ).** Economic efficiency examines efficiency from the perspective of selling electrical energy equivalents. It compares the rate of energy available for sale (actual electric power and process heat expressed in terms of equivalent electric power) to the rate of energy bought for the system, as shown in Eq. (13)

$$\eta_{\text{economic}} = \frac{(\dot{W}_{\text{net}})_{\text{electric}} + (\dot{W}_{\text{process equivalent}})_{\text{electric}}}{\dot{Q}_{\text{combustor}}} \quad (13)$$

The net electric power from the system is the product of the net power from the system and the generator efficiency, as shown in Eq. (14)

$$(\dot{W}_{\text{net}})_{\text{electric}} = \eta_{\text{generator}} \dot{W}_{\text{net}} \quad (14)$$

The process heat expressed in terms of electric power  $(\dot{W}_{\text{process-equivalent}})_{\text{electric}}$  is the product of the actual process heat  $(\dot{Q}_{\text{process}})$  and an economic ratio value ( $R_{\text{value}}$ ), as shown in Eq. (15)

$$(\dot{W}_{\text{process equivalent}})_{\text{electric}} = R_{\text{value}} \dot{Q}_{\text{process}} \quad (15)$$

where ( $R_{\text{value}}$ ) is the ratio that compares the value associated with selling process heat to the value associated with selling electric power, as indicated by Eq. (16)

$$R_{\text{value}} = \frac{\text{unit price of process heat}[\$/\text{kW of process heat}]}{\text{unit price of electrical energy}[\$/\text{kW of electricity}]} \quad (16)$$

Economic efficiency compares rate of energy available for sale (in terms of equivalent electric power) to the rate of energy bought for the system.

## 4 Second Law-Based Performance Criteria

Based on the second law of thermodynamics, the following performance criteria may be defined using *exergy*, which is the energy that has been *taxed* by nature prior to use,  $Ex = H - T_0S$ . In these analyses, it is assumed that exergy calculations are covered in the courses taught. If this is not the case, either this section will be omitted or additional discussion will be incorporated to clarify these. Although, in general, these parameters are more meaningful, they should, nevertheless, be used judiciously to emphasize the pedagogical points that are intended.

**4.1 Exergetic Turbine Efficiency ( $\eta_{\text{exergetic turbine}}$ ).** Exergetic turbine efficiency applies to a device, the *turbine*, rather than the overall system. As shown in Eq. (17), exergetic turbine efficiency compares the power produced by the turbine to the decrease in flow exergy from the turbine inlet to its exit. Although it is an important quantity, exergetic turbine efficiency does not identify exergy losses associated with the turbine. To determine these losses, individual calculations must be made for the operational characteristics of the turbine based on exergy to better identify the losses

$$\eta_{\text{exergetic turbine}} = \frac{(\dot{W}_{\text{Net}})_{\text{turbine}}}{(\dot{E}x_{\text{FlowIn}} - \dot{E}x_{\text{FlowOut}})_{\text{turbine}}} \quad (17)$$

**Table 2** Property values obtained from EES for combined cycle cogeneration state points

State	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg K)	Exergy (kW)
1	25	101.3	298.6	5.695	0
2	426.2	1550	712.9	5.788	281,124
3	1267	1550	1685	6.693	791,815
4	608	101.3	912.1	6.824	201,377
5	497.2	101.3	789.8	6.676	144,593
6	232.6	101.3	509.3	6.231	37,141
7	162	101.3	437	6.077	17,892
8	570	2000	3623	7.624	296,659
9	196.3	101.3	2867	7.812	209,092
10	99.97	101.3	419	1.307	154,179
11	100.2	2000	421.4	1.308	154,403
12	212.4	2000	908.6	2.447	170,326
13	212.4	2000	2799	6.34	249,039

**4.2 Second Law Efficiency ( $\eta_{II}$ ).** Second law efficiency applies to the overall system. As shown in Eq. (18), second law efficiency compares the useful exergy output from the system to the rate of exergy provided to the system by the combustor, a parameter that can be calculated using the appropriate exergy definition, thus making it an important parameter to discuss

$$\eta_{II} = \frac{(\dot{W}_{net})_{electric} + \dot{E}x_{process}}{\dot{E}x_{combustor}} \quad (18)$$

**4.3 Exergetic Combustor Efficiency ( $\eta_{exergetic\ combustor}$ ).** Exergetic combustor efficiency applies to the *combustion chamber*. It describes the percent of exergy used from the exergy available in the fuel provided to the combustion chamber. As shown in Eq. (19), exergetic combustor efficiency is the ratio of the total heat transfer rate in the combustion chamber to the fuel exergy rate provided and should be as large as possible so that the exergy in the fuel can better be used in the combustion chamber to raise the pressure and the temperature

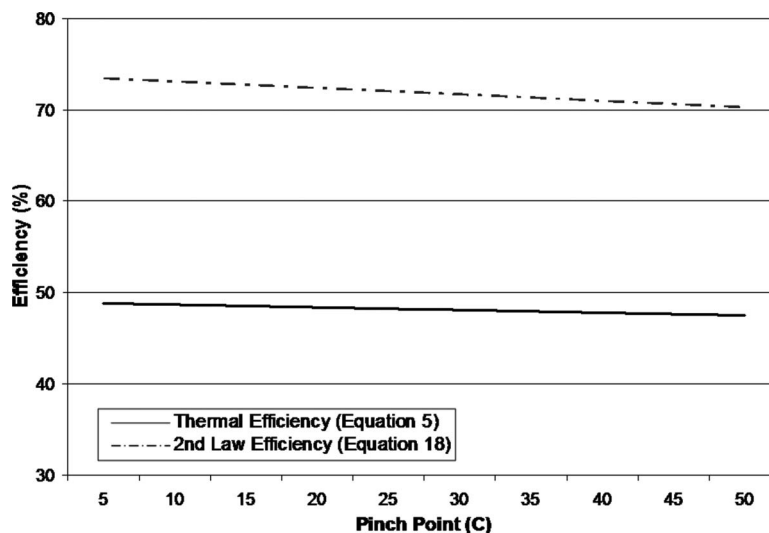
$$\eta_{exergetic\ combustor} = \frac{\dot{Q}_{combustor}}{\dot{E}x_{fuel}} \quad (19)$$

**4.4 Heat Recovery Steam Generator Effectiveness ( $\epsilon_{exergy}$ ).** Exergy based heat recovery steam generator effectiveness compares the exergy rate provided to the steam cycle to the exergy rate that was available in the HRSG, as shown in Eq. (20). This is an important quantity in that it shows how effectively the exergy in the exhaust stream of the gas turbine cycle is transferred to the steam generation in the HRSG thus making the overall effectiveness as high as possible

$$\epsilon_{exergy} = \frac{\dot{E}x_{supplied\ steam\ cycle}}{\dot{E}x_{combustor} - (\dot{W}_{gas\ turbine})_{electric}} \quad (20)$$

## 5 Analysis

This paper presents a number of performance parameters relevant to industrial power generation systems. Thermodynamics educators teaching a two-semester course in thermodynamics should include these parameters in the course material. If only one semester is available, then the first law parameters should be covered in detail, particularly the thermal efficiency, power to heat ratio, utilization factor (keeping in mind that the electric power generated is the important term), and the backwork ratio for the gas turbine cycle, since a small value is desired. For the HRSG, the heat recovery steam generator effectiveness, Eq. (8), should be



**Fig. 5** Pinch point effect on thermal and second law efficiencies

**Table 3 Numerical values for Fig. 5**

Pinch point (K)	5	10	15	20	25	30	35	40	45	50
$\eta_{\text{thermal}}\%$	48.84	48.69	48.55	48.40	48.25	48.11	47.96	47.81	47.66	47.52
$\eta_{\text{II}}\%$	73.47	73.12	72.76	72.41	72.06	71.71	71.35	71.00	70.65	70.29

**Table 4 The results for each term given in equations for the gas turbine selected [11] and other parameters**

Equation No.	Variable	Description	Numerical value
Given or assumed parameters			
	$\eta_C$	Isentropic compressor efficiency	85 (%)
	$\eta_G$	Generator efficiency	98 (%)
	$\eta_{\text{thermal, gas turbine}}$	Gas turbine cycle thermal efficiency	36.897 (%)
	$r_P$	Pressure ratio for GT cycle compressor	15.3
	$(\dot{W}_{\text{GT}})_{\text{electric}}$	Net electric power for gas turbine	255,600 (kW)
	$\eta_P$	Isentropic pump efficiency	80 (%)
	$\eta_{\text{steam turbine}}$	Steam turbine isentropic efficiency	90 (%)
	$\eta_{\text{combustor}}$	Combustion efficiency	98 (%)
(1)	$\dot{W}_{\text{net}}$	Net power produced by system	342,128 (kW)
(2)	$\dot{E}_{\text{fuel}}$	Energy rate associated with fuel	721,303(kW)
(3)	$\dot{Q}_{\text{combustor}}$	Rate of heat transfer into combustion chamber	706,877(kW)
(4)	$\dot{Q}_{\text{process}}$	Rate of heat transfer for process heat	264,322(kW)
(5)	$\eta_{\text{thermal}}$	Thermal efficiency	48.4 (%)
(6)	$R_{\text{power to heat}}$	Power to heat ratio	1.294
(7)	$F_{\text{utilization}}$	Utilization factor	85.79 (%)
(8)	$R_{\text{backwork}}$	Backwork ratio	53.6 (%)
(9)	$SFC$	Specific fuel consumption(gas turbine)	0.00005529 (kg/kW h)
(10)	$\epsilon_{\text{HRSG}}$	Heat recovery steam generator effectiveness	77.49 (%)
(11)	$\eta_{\text{power generation}}$	Power generation efficiency	93.54 (%)
(12)	$FCP$	Fuel chargeable to power	1.069
(13)	$\eta_{\text{economic}}$	Economic efficiency	66.17 (%)
(14)	$(\dot{W}_{\text{net}})_{\text{electric}}$	Net electric power	335,285 (kW)
(15)	$(\dot{W}_{\text{process equivalent}})_{\text{electric}}$	Process heat expressed in terms of electric power	132,161(kW)
(16)	$R_{\text{value}}$	Economic ratio value	0.5
(17)	$\eta_{\text{exergetic turbine}}$	Exergetic turbine efficiency	44.17 (%)
(18)	$\eta_{\text{II}}$	Second law efficiency	72.41 (%)
(19)	$\eta_{\text{exergetic combustor}}$	Exergetic combustor efficiency	94.54 (%)
(20)	$\epsilon_{\text{exergy}}$	Heat recovery steam generator effectiveness	72.72 (%)

taught. If economic considerations are desired, then economic efficiency, Eq. (13) should be discussed. If the second law is well covered in the thermodynamics course, then exergetic turbine efficiency and the second law efficiency are the two most important terms. Additionally, heat recovery steam generator effectiveness based on the Second Law, Eq. (20), should be discussed thoroughly. The control of the pinch point and its effect on the HRSG operation and on the efficiency are also topics to be discussed. If the above is done precisely, then the student will have a better appreciation of the usefulness of the combined cycle cogeneration system. An engineering equation solver (EES) [14] is used to obtain property results given in Table 2 for analysis of the combined cycle cogeneration system in Fig. 3. The effect of the pinch point on thermal efficiency, Eq. (5), and the second law efficiency, Eq. (18), is shown in Fig. 5, and the numerical values are given in Table 3, as calculated by EES [14]. For the data in Ref. [11] and other given information, as shown in Table 4, all of the parameters listed in the text are calculated using EES [14]. Each term is identified by its equation number to facilitate reading. Thus the instructor and student alike can analyze a real system either using the first law or second law of thermodynamics analysis or both. It has to be decided by the individual instructor.

## 6 Conclusions

With the prevalence of combined cycle cogeneration systems in industry, introduction of these systems into undergraduate thermodynamics courses is important. Increasing concern with resource utilization and sustainability presents additional need for further discussion in the context of the course. This paper outlines important parameters for the complete cycle. The individual instructor should exercise discretion as to which parameters are introduced based on students' knowledge bases, the focus of the course, and whether or not a single- or two-semester course is offered.

## Nomenclature

- $E$  = energy (kJ)
- $\dot{E}$  = energy rate (kW)
- $Ex$  = exergy (kJ)
- $\dot{Ex}$  = exergy rate (kW)
- $F$  = factor
- $FCP$  = fuel chargeable to power
- $h$  = enthalpy specific (kJ/kg)
- $LHV$  = lower heating value (kJ/kg)

$\dot{m}$  = mass flow (kg/h)  
 $Q$  = heat transfer (kJ)  
 $\dot{Q}$  = heat transfer rate (kW)  
 $R$  = ratio  
 $s$  = specific entropy (kJ/(kg K))  
 $SFC$  = specific fuel consumption (kg/kJ)  
 $T$  = temperature (K)  
 $W$  = work (kJ)  
 $\dot{W}$  = power (kW)

#### Subscripts

$0$  = ambient  
 $II$  = second law

#### Greek Symbol

$\varepsilon$  = effectiveness  
 $\eta$  = efficiency

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