Pergamon

Energy 25 (2000) 1215–1229

ENERGY

www.elsevier.com/locate/energy

# Exergetic and engineering analyses of gas turbine based cogeneration systems

E. Bilgen *

*Ecole Polytechnique University of Montreal, C.P. 6079, centre ville, Montreal, Qc, Canada H3C 3A7*

**Abstract**

This paper presents exergetic and engineering analyses as well as a simulation of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine. The exergy analysis is based on the first and second laws of thermodynamics. The engineering analysis is based on both the methodology of levelized cost and the pay back period. To simulate these systems, an algorithm has been developed. Two cogeneration cycles, one consisting of a gas turbine and the other of a gas turbine and steam turbine and process to produce electricity and process heat have been analyzed. The results showed good agreement with the reported data. © 2000 Elsevier Science Ltd. All rights reserved.

## 1. Introduction

Cogeneration involves the production of both thermal energy, generally in the form of steam or hot water, and electricity. The ratio of electric power to thermal energy varies depending on the plant type. A cogeneration plant may be conceived to supply thermal energy or electric power. In the first case, electric power is considered to be a by product and is relatively small. In the latter case, which is often encountered by public utility companies, thermal energy is considered as a by product. There are conceptually different cogeneration plants: the steam turbine based, gas turbine based, and diesel engine based plant (see, for example, Ref. [1]). In the present study, gas turbine based cogeneration plants are considered. The gas turbines differ in power output, cycle efficiency, cycle pressure ratio, firing temperature, exhaust temperature and exhaust flow rate. The thermodynamic and engineering performance of combustion gas turbine cogeneration systems can be found in the literature [2–10]. Description of various cases and engineering data are given in Refs. [2,3]. The thermodynamic analyses are given in Refs. [4,5]. Various other aspects relating to the present study are discussed in Refs. [6–10]. Rice [4] based his study on

* Tel.: +1 (514) 340-4711, ext. 4579; fax: +1 (514) 340-5917.
*E-mail address:* bilgen@meca.polymtl.ca (E. Bilgen).


PII: S0360-5442(00)00041-4

## Nomenclature

| | |
|---|---|
| $a_{ik}$ | number of atoms of the *k*th element present in chemical species *i* |
| $A_k$ | total number of atomic weight of the *k*th element |
| $B_f$ | exergy of fuel (kJ) |
| $B_p$ | exergy of process heat (kJ) |
| $c$ | levelized cost ($/kWh) |
| $c_f$ | cost of fuel ($/GJ) |
| $c_s$ | cost of steam ($/kg) |
| $C_a$ | present value of additional investment ($) |
| $C_o$ | credit for cogeneration products ($) |
| $C_{su}$ | start-up cost ($) |
| $C_t$ | total equipment cost ($) |
| $E_f$ | fuel energy (kJ) |
| $f_{dc}$ | interest rate on the borrowed capital during construction (%) |
| $F$ | capital charge rate (%) |
| $F_{om}$ | operation and maintenance charges (%) |
| $h$ | enthalpy of produced steam (kJ/kg) |
| $h_f$ | enthalpy of saturated water (kJ/kg) |
| $h_t$ | enthalpy of combustion gases at turbine exit (kJ/kg) |
| $h_{pp}$ | enthalpy of gas mixture at pinch point temperature (kJ/kg) |
| $i$ | interest rate during amortization period (%) |
| $i_e$ | effective interest (%) |
| $m$ | plant operating lifetime (year) |
| $m_a$ | mass of air (kg) |
| $m_s$ | mass of steam (kg) or mass flow rate of steam in Eq. (18) (kg/year) |
| $n$ | pay back period of additional investment (year) |
| $n_i$ | number of moles for component *i* |
| $n_{dd}$ | number of capital draw down per year |
| $n_t$ | total number of capital draw down during construction |
| $P$ | pressure (Pa) |
| $Q_f$ | fuel consumption (GJ/year) |
| $Q_p$ | thermal energy of process heat (kJ) |
| $r$ | inflation rate in next m year (%) |
| $r_{dc}$ | interest rate during construction (%) |
| $r_{fa}$ | fuel–air ratio |
| $r_{ph}$ | power-to-heat ratio |
| $R$ | gas constant (kJ/kmol·K) |
| $s$ | entropy of produced steam (kJ/kg·K) |
| $s_f$ | entropy of condensate return (kJ/kg·K) |
| $T$ | thermodynamic temperature (K) |
| $T_0$ | temperature of the environment (K) |
| $W_e$ | electrical energy (kJ) or (kWh/year) in Eq. (18) |
| $W_t$ | electric energy produced in a year (kWh/year) |
| $y_i$ | molar fraction of component *i* |

*Greek symbols*

| | |
|---|---|
| $\eta$ | fuel utilization efficiency |
| $\varepsilon$ | exergy efficiency |
| $\gamma$ | construction time (year) |
| $\lambda$ | Lagrangian multipliers |
| $\varepsilon_f$ | fuel exergy factor |
| $\varepsilon_p$ | process heat exergy factor |

the first law of thermodynamics and developed a graphic solution showing the interrelationship of the relevant parameters. He also presented a simple algorithm how to use his graphic solution. Huang [5] on the other hand, developed a methodology for performance evaluation based on first and second laws of thermodynamics, where he showed that performance evaluation based on the first law alone was not adequate. Sarabchi [6] carried out a parametric study based on the methodology of [5]. He showed once more the importance of the inclusion of the second law of thermodynamics in the analysis of cogeneration systems. Rosen and Le [7] considered combined heat, power and district cooling system and examined, based on energy and exergy analyses, the efficiencies of integrated systems. They derived general efficiency measures suitable for combined heat, power and cooling systems. Korobitsyn and Hirs [8] studied various cogeneration systems using five different steam and power generating alternatives on the basis of constant process heat demand. They compared them based on fuel saving and energy analysis, and established a definition of a critical electric efficiency indicative of zero fuel saving. Lior and Arai [9,10] carried out thermodynamic and engineering analyses of a system which consisted of a novel chemical gas turbine and a Rankine bottoming cycle. They determined first law efficiency, energy requirement and heat transfer equipment size of these systems for conditions imposed. They showed that the system efficiency of the base case could be 61%, which could be increased to, as high as 74%, although the heat transfer equipment size for the latter were estimated to be larger compared to the base case.

The review shows that in the analysis of cogeneration systems, it is essential to consider both the first and second laws of thermodynamics and also engineering aspect if any variation or improvement is desired. Although this is the case in recent studies on cogeneration systems, the literature review showed that a clear and simple algorithm for a methodology combining the first and second laws of thermodynamics (the exergy analysis) and engineering economics does not exist.

The aim of the present study is to complement previous studies using the exergy concept, to present a modular technique for engineering economics and to develop an algorithm useful for modeling cogeneration systems. The thermodynamic models are based on the methodologies using the first and second laws of thermodynamics and the exergy concept [11]. The engineering methodology is based on standard engineering methodologies for design, cost evaluation and economics of the electrical energy produced and the pay back period of the additional investment for process heat production [12].

## 2. Gas turbine based cogeneration plants

A basic gas turbine based cogeneration system consists of a gas turbine cycle (compressor, combustion chamber and expander), a heat recovery system for steam production and a steam turbine. Fuel is introduced into the combustion chamber of the gas turbine where combustion takes place with compressed air coming out from the compressor. Hot exhaust gases from the turbine are the waste heat source for process heat production. The quantity and quality of process heat produced depend on the temperature of hot exhaust gases entering the heat recovery system and the resulting temperature of steam produced. Steam produced can be used either for process heat or electric power that is generated by a steam turbine or both. The cases considered are then (i) gas turbine electric power production–process heat production, (ii) gas turbine electric power production–electric power production by steam turbine, and (iii) gas turbine electric power production–electric power production by steam turbine–process heat production. The schematics of these cycles are shown in Figs. 1–3, respectively.

In the following three sections, methodologies for thermodynamic and engineering evaluation are discussed and then an algorithm is presented.

## 3. Thermodynamic evaluation

### 3.1. Combustion analysis

The composition of the combustion gas mixture is calculated using the direct minimization of the Gibbs function [13]. The problem is to find the equilibrium composition for a given temperature, $T$, and pressure, $P$, and for a given feed to the combustion chamber.

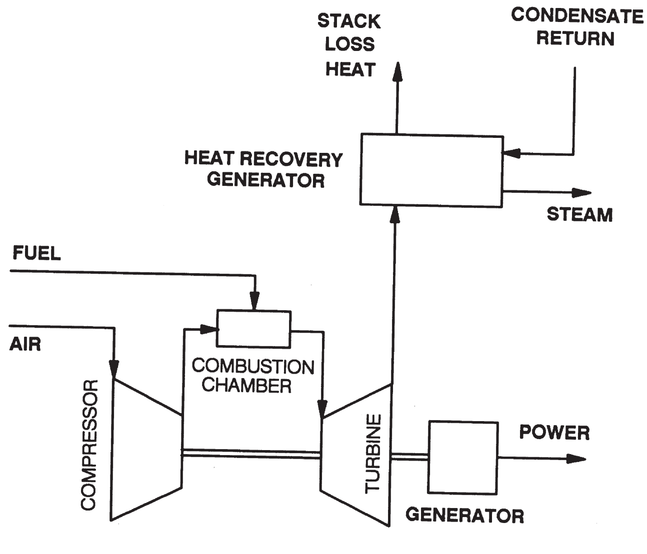


Fig. 1. Schematic of the cycle for gas turbine electric power production–process heat production.

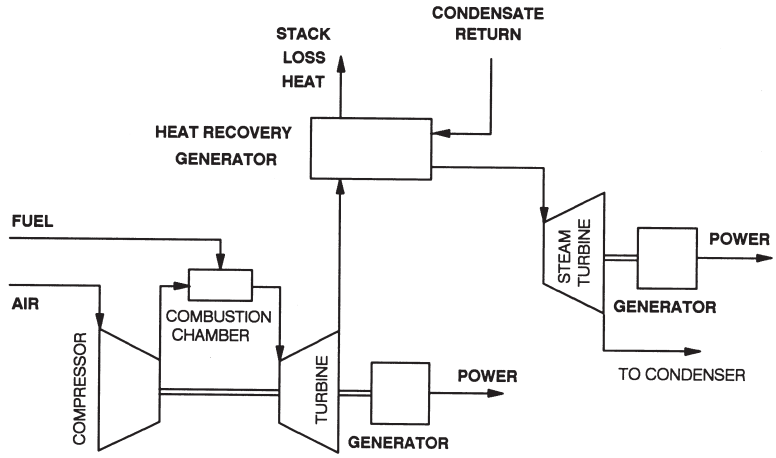


Fig. 2. Schematic of the cycle for gas turbine electric power production–electric power production by steam turbine.

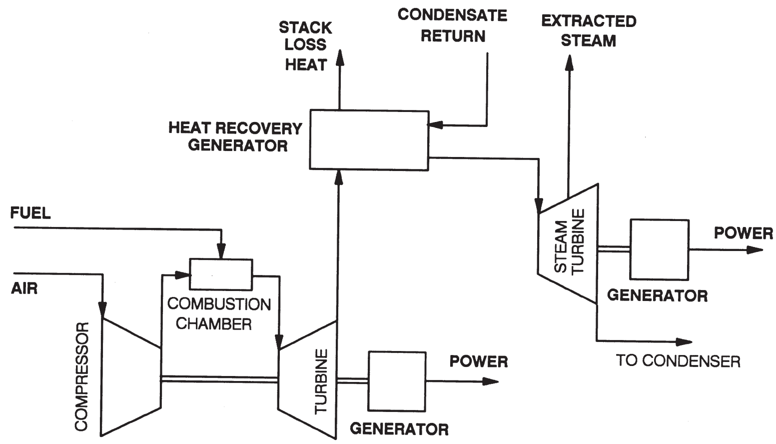


Fig. 3. Schematic of the cycle for gas turbine electric power production–electric power production by steam turbine–process heat production.

Minimization of the Gibbs function, using Lagrangian multipliers $\lambda_i$, leads to a set of $i$ equations in the form of

$$\Delta G^0_{fi}+RT\ln P+RT\ln y_i+\sum_k(\lambda_k\alpha_{ik})=0 \tag{1}$$

where $\Delta G^0_{fi}$ is the standard Gibbs function of formation of each compound $i$ from its constituent elements at temperature, $T$, $y_i$ is the molar fraction of component $i$ in the mixture and $a_{ik}$ is the number of atoms of the $k$th element present in each molecule of chemical species $i$.

In addition, there are $k$ material balance equations

$$\sum_i(y_ia_{ik})=\frac{A_k}{\Sigma_in_i} \tag{2}$$

where $n_i$ is the number of moles for component $i$ and $A_k$ the total number of atomic weights of the $k$th element present in the system.

In addition

$$\sum_i y_i=1 \tag{3}$$

resulting in $i+k+1$ non linear equations, which must be solved with $i$ number $y_i$, $k$ number $\lambda_k$ and $\Sigma_i\ n_i$ unknowns.

Energy and exergy of a hydrocarbon fuel are calculated from (see, for example, Ref. [11])

$$E_{\mathrm{f}}=\sum_p n_{\mathrm{e}}h_{\mathrm{e}}-\sum_r n_i h_i \tag{4}$$

$$B_{\mathrm{f}}=\left(\sum_r n_i g_i-\sum_p n_{\mathrm{e}}g_{\mathrm{e}}\right)+RT_0\ \ln\left(\frac{y_{\mathrm{O_2}}^{y1}}{y_{\mathrm{CO_2}}^{x1}y_{\mathrm{H_2O}}^{x2}}\right) \tag{5}$$

where $h_i$, $h_{\mathrm{e}}$ are enthalpies and $g_i$, $g_{\mathrm{e}}$ are the Gibbs functions of reactants (shown with $r$) and products (shown with $p$) for stoichiometric reaction of fuel evaluated at 1 bar and at 298 K, $y_i^{\alpha}$ is the mole fraction of component $i$ in the environment.

A fuel exergy factor is defined as

$$\varepsilon_{\mathrm{f}}=\frac{B_{\mathrm{f}}}{E_{\mathrm{f}}}. \tag{6}$$

## 4. Cycle analysis

The useful products of a cogeneration system are electrical energy, $W_{\mathrm{e}}$ and thermal energy or process heat, $Q_{\mathrm{p}}$, usually in the form of steam at saturated state. The thermodynamic performance is based on the first law efficiency, defined as fuel utilization efficiency

$$\eta=C\frac{W_{\mathrm{e}}+Q_{\mathrm{p}}}{E_{\mathrm{f}}} \tag{7}$$

where $E_{\mathrm{f}}$ is the energy of fuel, determined from Eq. (4), and following [4], a parasitic system loss of 2% is assumed for simple turbine cycles, hence $C$=0.98.

In Eq. (7), it is seen that electrical and thermal energies are treated as equal without qualification. Therefore, to have a better insight into the thermodynamic performance, an exergy efficiency is defined as

$$\varepsilon=\frac{W_{\mathrm{e}}+B_{\mathrm{p}}}{B_{\mathrm{f}}} \tag{8}$$

where $W_e$ is work, hence considered all exergy, $B_p$ is the exergy content of process heat produced and $B_f$ is the exergy content of fuel input evaluated using Eq. (5).

The exergy content of the process heat produced is evaluated as

$$B_p = m_s[(h-h_c)-T_0(s-s_c)] \tag{9}$$

where $m_s$ is the mass of steam, $s$ is the entropy of the produced steam, $s_c$ is the entropy of condensate return, both at the process heat pressure and $T_0$ is the temperature of the environment.

The first part of Eq. (9) represents the energy of the process heat, which is

$$Q_p = m_s(h-h_c). \tag{10}$$

The process heat generator consists usually of three main parts, the first an economizer, the second evaporator, and the last a superheater, all working at a process heat pressure. The water at $T_c$ enters the economizer and exits at $T_f$, at saturated liquid state. Then, the saturated liquid enters the evaporator and exits at the same temperature at saturated vapor state. Finally, the saturated steam enters the super heater and exits at superheat temperature $T$. The combustion gases, in counter flow, enters the superheater at the turbine exit condition, at $T_t$, goes through the evaporator and exit it at a pinch point temperature, $T_{pp}$, which is determined from $\Delta T=(T_{pp}-T_f)$ of the heat exchanger. It is shown that this parameter is an important parameter affecting the thermal performance of the system [5]. The combustion gases enter the economizer at $T_{pp}$ and exit at $T_e$. In many simpler systems, the process heat may be in saturated state, hence no need for a superheater.

For simplicity it may be assumed that the pressure drop in the heat recovery steam generator is negligible and the heat recovery steam generator is well insulated. Energy balance at the system consisting of superheater and evaporator is

$$m_s(h-h_f) = m_a(1+r_{fa})(h_t-h_{pp}) \tag{11}$$

where $m_a$ is the mass of air in gas turbine engine, $r_{fa}$ is the fuel–air ratio used in combustion process, $h_t$ is the enthalpy of gas mixture at turbine exit, $h_{pp}$ is the enthalpy of gas mixture at pinch point temperature.

Solving Eq. (10) for $m_s$, inserting it in Eq. (11), and rearranging, the process heat produced per unit mass of air flow is found as

$$\frac{Q_p}{m_a} = (1+r_{fa})(h_t-h_{pp})\frac{(h-h_c)}{(h-h_f)}. \tag{12}$$

To have a better assessment, some useful ratios, such as process heat exergy factor and power-to-heat ratio are defined as

$$\varepsilon_p = \frac{B_p}{Q_p} \tag{13}$$

$$r_{ph} = \frac{W_e}{Q_p}. \tag{14}$$

## 5. Engineering evaluation

A modular cost estimating model is used, which is based on standard chemical engineering costing techniques [12,14]. A module consists of one major equipment, which performs a specified operation and which includes all necessary elements and materials such as piping and valves, instruments and control, civil works, structures, painting and insulation, electrical installation, material erection, indirect costs (freight, insurance, taxes, etc.), material types, internals, drives, etc. The model produces the total installed equipment cost $C_t$. The evaluated costs were verified from the reported price levels for basic gas turbine power plant packages [15].

The economical parameters, the levelized cost of product as well as pay back period of additional investment for cogeneration are calculated as follows.

The start-up cost $C_{su}$ is calculated as

$$C_{su}=(1+f_{dc})(1+r_{dc})^{\gamma}C_t \tag{15}$$

where $\gamma$ is the construction time and $f_{dc}$ is the capital draw-down during construction, which is calculated as

$$f_{dc}=0.5\frac{1+n_t}{n_{dd}}. \tag{16}$$

Factor of the levelized operation and maintenance cost, $F_{om}$ is calculated as

$$F_{om}=F_{om,i}\frac{\sum_{\alpha=1}^{m}(1+r)^{\alpha}(1+i)^{-\alpha}}{\sum_{\alpha=1}^{m}(1+r)^{-\alpha}} \tag{17}$$

where $F_{om,i}$ is the initial operation and maintenance charges calculated as a percent of the start-up cost. This is usually 1% of the start-up cost.

In Eq. (17), $m$ is the plant amortization period, normally 20 years, $r$ is the inflation rate during amortization period, often assumed to be zero, for a constant dollar assumption, $i$ is the interest rate during amortization period, $m$.

The levelized cost per kWh electric energy produced by the gas turbine over m years of operation is calculated as

$$c=\frac{(F+F_{om})C_{su}+(Q_f)(c_f)-(m_s)(c_s)-(W_e)(c_e)}{W_t} \tag{18}$$

where $Q_f$ is the total fuel consumed in GJ/year, $c_f$ the cost of fuel in \$/GJ, $m_s$ the total steam produced in kg/year, $c_s$ the credit for steam in \$/kg, $W_e$ the electric energy produced by the steam turbine in kWh/year, $c_e$ the credit for it in \$/kWh, and $W_t$ the total electric energy produced by the gas turbine in kWh/year.

The cost of energy is calculated as levelized over the amortization period in \$/kWh. In this calculation, the electric energy produced by the gas turbine is the main product, and process steam and electric energy produced by the steam turbine are considered as by products, which represent credits. The credit for each product is calculated based on the current purchasing price, $c_s$ and $c_e$.

The pay back period of additional investment for heat recovery steam generator and steam turbine plant, if any, is calculated by amortizing it with the credit of by products (process steam and electric energy from the steam turbine).

$$n=\frac{\ln\left(\frac{C_a\times i_e}{C_o}+1\right)}{\ln(1+i_e)} \tag{19}$$

where $C_a$ is the present value of additional investment for heat recovery steam generator and steam turbine plant in \$, $C_o=m_s c_s+W_e c_e$ is the total credit for cogeneration products (steam and electric energy from the steam turbine) in \$ per year and $i_e$ the effective interest, which is the difference between the interest and the inflation rate during the amortization period.

## 6. Algorithm

An algorithm was developed based on the analyses presented in the previous two sections.

- Compute combustion gas composition at the inlet rotor turbine temperature, Eqs. (1)–(3).
- Using an iterative method compute compressor and turbine outlet gas temperatures.
- Determine specific work output and the mass of air flow.
- Compute enthalpy and exergy of the product gas, water, and steam at various states.
- Compute fuel exergy, Eq. (5).
- compute gas turbine cycle efficiency, Eq. (7) with $Q_p$=0.
- Compute process heat per unit mass of air flow, Eq. (12).
- Compute $\varepsilon_p$ [Eq. (13)], $\varepsilon_f$ [Eq. (6)], $\eta$ [Eq. (7)], $r_{ph}$ [Eq. (14)], $\varepsilon$ [Eq. (8)], steam flow rate [Eq. (10)], exergy of process heat [Eq. (9)] and total work.
- Compute costs of each module, and the total cost, $C_t$.
- Compute levelized cost of power [Eq. (18)], the pay back period for additional investment [Eq. (19)].

## 7. Case studies

Two cases of cogeneration systems are carried out. With reference to three cases discussed in Section 2, case (i) is shown in Fig. 1, which is the gas turbine electric power production–process heat production. Case (ii) shown in Fig. 3, is the gas turbine–steam turbine–process heat production.

In both case studies natural gas is used as fuel and the capacity factor of the plant is assumed to be 80%. Cost data are from [12] updated to 1996 US\$. Assumed cost data for various items are $c_f$=3 \$/GJ, $c_s$=0.01 \$/kg steam and $c_e$=0.05 \$/kWh. Economical assumptions are amortization period, $m$=20 years, inflation rate during amortization, $r$=0 and interest rate during amortization

Table 1
Base-load gas turbine data, ISO conditions from [4][a]

| | |
|---|---|
| Turbine shaft work (kW) | 22007.0 |
| Cycle efficiency (%) | 37.0 |
| Cycle pressure ratio | 18.7 |
| Air mass flow rate (kg/s) | 66.9 |
| Specific work output (kJ/kg air) | 328.9 |
| Turbine rotor inlet temperature (K) | 1485.0 |
| Exhaust temperature (K) | 786.0 |
| Exhaust excess air (%) | 226.0 |
| Compressor isentropic efficiency (%) | 70.4 |
| Turbine isentropic efficiency (%) | 92.6 |

[a] 288 K, 101.325 kPa, 60% relative humidity.

period, $i$=0.05. 1% initial operation and maintenance charge, 1/2 year cogeneration plant construction time are also assumed. Other parameters, if any, are presented later with the case studies.

### *7.1. Gas turbine–process heat system*

Base-load gas turbine data at ISO conditions (288 K, 101.325 kPa, 60% relative humidity) are from a case study given in [4] for an industrial gas turbine presently on the market. These parameters are shown in Table 1.

The base parameters for compressor, turbine and system are: isentropic efficiencies of compressor and turbine, 0.704 and 0.926 respectively, intake air temperature, 288 K, pressure of the process steam (saturated), 2026 kPa, temperature of condensate return, 373 K and the pinch point temperature difference, 50 K.

The composition of combustion products in mole is calculated for the combustion of natural gas with 226% air: 1 $CO_2$, 0 CO, 2 $HO_2$, 0.001 OH, 0 $NO_2$, 0 NO, 24.515 $N_2$, 4.52 $O_2$. Other parameters are also evaluated and compared to available data. The analysis of this cycle with GE, LM2500PE turbine has been reported in [4,5]. The available parameters from these studies, namely cycle efficiency, air flow, specific work output, exhaust temperature from [4] and fuel utilization efficiency, exergy efficiency and power to heat ratio from [5] are compared with the results from this study and presented in Table 2. It is seen that the agreement is excellent.

Table 2
Comparison with the reported data for GE, LM2500PE turbine cited in [4,5] and this study

| | Ref. [4] | Ref. [5] | This study |
|---|---|---|---|
| Cycle efficiency (%) | 37.00 | | 37.62 |
| Air mass flow rate (kg/s) | 66.90 | | 66.33 |
| Specific work output (kJ/kg air) | 328.90 | | 331.66 |
| Exhaust temperature (K) | 786.15 | | 786.15 |
| Fuel-utilization efficiency (%) | | 77.00 | 77.02 |
| Exergy efficiency (%) | | 52.00 | 50.06 |
| Power-to-heat ratio | | 0.92 | 0.93 |

Table 3
Process heat results

| | Case (i) | Case (ii) |
|---|---|---|
| Process steam temperature (K) | 486.19 | 686.15 |
| Saturation temperature of steam (K) | 486.19 | 486.15 |
| Pinch point temperature (K) | 536.19 | 536.19 |
| Process heat produced (kJ/kg air) | 358.28 | 179.16 |
| Specific steam flow rate (kg/kg air) | 0.151 | 0.120 |
| Steam mass flow rate (kg/s) | 10.02 | 3.98 |
| Exergy of process heat (kJ/kg air) | 132.37 | 57.71 |
| Exergy factor of process heat | 0.37 | 0.32 |
| Temperature of extracted steam (K) | | 599.59 |
| Pressure at steam turbine outlet (kPa) | | 4.00 |
| Specific work of steam turbine (kJ/kg air) | | 66.92 |
| Steam turbine shaft work | | 4440.00 |

Process heat results are shown in Table 3. Cost and economics results are shown in Table 4.

The cycle efficiency for the compressor–gas turbine system without cogeneration was calculated as 37.62%. The fuel utilization efficiency 77.02% in Table 2 is that for the cogeneration system. Using cogeneration, the efficiency is therefore improved by 105%. The exergy efficiency of the cogeneration system is 50.06%, which may be compared with to 35.78% for the system without cogeneration. The improvement is about 40%.

The cost and economics results in Table 4 show that the typical cost for 15% capital charge rate is improved by 43% and the pay back period is just about two months. The product cost for case (i) as a function of the capital charge rate has been computed using Eq. (18), with $F$ from 0 to 20% and shown in Fig. 4 together with that for the system without cogeneration. It is noted that for case (i), $W_e$=0 and for the base case both $m_s$ and $W_e$ are zero in Eq. (18). It is seen that the product cost is an increasing function of the capital charge rate for both systems with a quasi-linear relationship. Therefore, the cost improvement is almost the same at any capital charge rate.

Table 4
Cost and economics results

| | Case (i) | Case (ii) |
|---|---|---|
| Gas turbine package cost (M$) | 7.310 | 7.310 |
| Heat recovery system cost (M$) | 0.431 | 0.483 |
| Steam turbine system cost (M$) | | 1.830 |
| Total cost (M$) | 7.741 | 9.623 |
| Typical product cost without cogeneration ($/kWh) | 0.037 | 0.037 |
| Typical product cost with cogeneration ($/kWh) | 0.021 | 0.023 |
| Pay back period (years) | 0.175 | 0.906 |

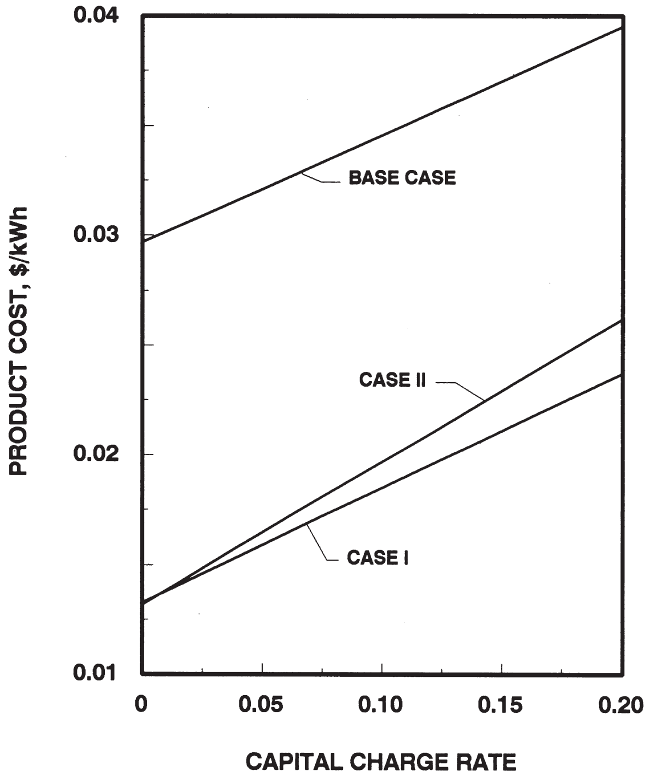


Fig. 4. Production cost from the gas turbine system for the base case (no cogeneration), case (i) (process steam production) and case (ii) (power production by a steam turbine and process heat production by steam extraction from the steam turbine).

### 7.2. Gas turbine–steam turbine–process heat system

For this case, the same data as in the previous case is used, however, the cogeneration process is simulated to produce steam in the heat recovery steam generator and electrical energy by a steam turbine and process steam by extracting steam from the turbine. The schematic of this example is shown in Fig. 3.

The base parameters are identical to the previous case with the exception of process heat and steam turbine data which are assumed to be 50% steam extraction at 1013 kPa and 303 K condenser sink temperature.

The simulation of this case gives identical results to the previous one for combustion results, turbine results, compressor results and the gas–turbine system results. The results for process heat, cogeneration, costs and economics are different and presented in Tables 3 and 4.

The comments made for case (i) apply also to this case. The results show that the fuel utilization efficiency is 64.49% and lower than that in case (i). The reason for this is obviously due to waste heat in the condenser. The exergy efficiency was calculated as 49.22% which is also slightly lower for the same reason. It should be noted that power-to-heat ratio defined by Eq. (14) is 2.22 for this case compared to 0.93 for case (i).

The cost and economics results in Table 4 show that the typical cost for 15% capital charge rate is improved by 37% with respect to the base case and the pay back period is almost a year. The product cost for case (ii) as a function of the capital charge rate, *F*, has also been computed and shown in Fig. 4. It is seen that due to non-zero $m_s$ and $W_e$ in Eq. (18) for this case, the slope is different than the others. At low capital charge rates, the cost difference between cases (i) and (ii) is negligible, but it increases as the charge rate increases. It is also seen that the cost improvement with respect to the base case slightly decreases as the charge rate increases.

Parametric studies were carried out for case (ii) by considering various parameters as a function of power-to-heat ratio. In Fig. 5, first and second law efficiencies and percent steam extraction as a function of power-to-heat ratio while in Fig. 6, power from steam turbine, total power, process heat production and pay back period as a function of power-to-heat ratio are presented. Fig. 5 shows that the steam extracted is a decreasing function of the power-to-heat ratio. The reason for this is for a given gas turbine system, as the steam turbine power is increased, the amount of steam extracted from the turbine decreases. It can also be seen in Fig. 5 that the first law efficiency (or fuel utilization efficiency) is a strong function of the power-to-heat ratio. It decreases with increasing power-to-heat ratio, i.e., with increasing power production, a result which is expected since as the energy is converted to power, the waste heat is increased, and the first law efficiency drops. The second law or exergy efficiency, on the other hand, varies little, indicating that the exergy content of the steam plus power generated from the steam turbine is not degraded much.

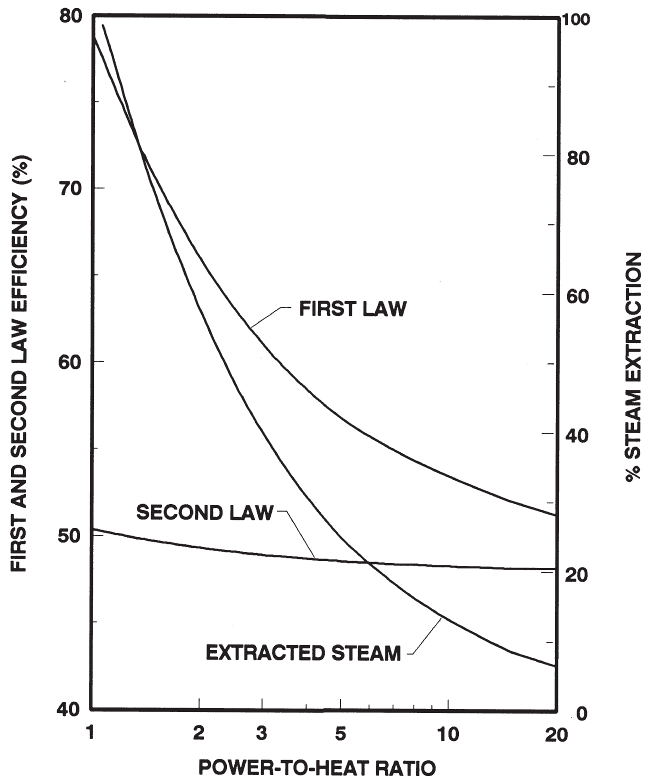


Fig. 5. First law and exergy efficiency and % steam extraction as a function of power to heat ratio.

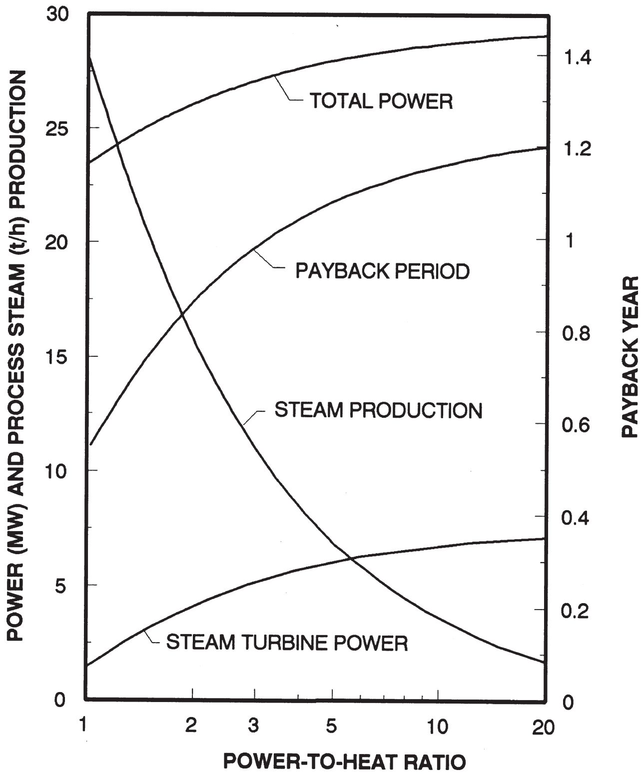


Fig. 6. Power production by steam turbine, total power production, process steam extraction and pay back period of the investment as a function of power to heat ratio.

Following [5] and using Eqs. (6), (7), (13) and (14), the relationship between exergy, $\varepsilon$, and fuel utilization, $\eta$, efficiencies can be expressed in terms of $r_{ph}$, $\varepsilon_f$ and $\varepsilon_p$.

$$\varepsilon=\frac{n_f}{\varepsilon_f}\left[\frac{r_{ph}+\varepsilon_p}{1+r_{ph}}\right]. \tag{20}$$

It is noted that $\varepsilon_f \approx 1$ and for a constant process heat exergy factor, $\varepsilon_p < 1$, the relationship between $\varepsilon$ and $\eta$ depends on $r_{ph}$. Hence, the term in the brackets of Eq. (20) will always be smaller than one. As a result, $\varepsilon$ will always be smaller than $\eta_f$, and the difference will be larger for smaller $r_{ph}$ and becomes smaller for increasing $r_{ph}$, as shown in Fig. 5.

Process steam production in (t/h) in Fig. 6 follows the same relationship as that of % steam extraction in Fig. 5. The power from steam turbine, and its contribution to total power, as a function of power-to-heat ratio visualizes how the power by steam turbine increases while the steam production decreases with increasing power-to-heat ratio. It is seen that the pay back period increases with increasing power-to-heat ratio. This result is expected since the investment for the heat recovery steam generator system stays constant for a given gas turbine system while the investment for the steam turbine increases with increasing power, thus increasing the pay back period. A parametric study was carried in the practical price range of 0.05 \$/kWh, 0.01 \$/kg steam–0.15 \$/kWh, 0.03 \$/kg steam. It was seen that when the electricity price was higher, the same trend was observed (not presented in figures), however the pay back period followed flatter curves with increasing electricity price.

## 8. Conclusions

Thermodynamic analyses based on first law of thermodynamics and exergy, and engineering evaluation based on the levelized cost methodology and pay back period of the investment have been carried out. Based on these analyses, an algorithm has been developed for thermodynamic performance and engineering evaluation of combustion gas turbine cogeneration systems. Simulation results of gas turbine systems with cogeneration show good agreement with the reported data. It is shown that by changing various relevant parameters, parametric studies can be carried out to determine thermodynamic and engineering parameters as well as suitable operational conditions.

## References

[1] Wilkinson BW, Barnes RW, editors. Cogeneration of electricity and useful heat. Boca Raton, FL: CRC Press, 1980.
[2] Hamm JR. Cogeneration systems using combustion (gas) turbines. In: Wilkinson BW, Barnes RW, editors. Cogeneration of electricity and useful heat. Boca Raton, FL: CRC Press, 1980. p.85–119.
[3] Nakhamkin M. Cogeneration. In: Seidman AH, Mahroua H, editors. Handbook of electric power calculations. New York: McGraw-Hill, 1984. p.17-1–17-16.
[4] Rice IG. Thermodynamic evaluation of gas turbine cogeneration cycles: part I — heat balance method analysis. Journal of Engineering for Gas Turbines and Power 1987;109:1–7.
[5] Huang FF. Performance evaluation of selected combustion gas turbine cogeneration systems based on first and second-law analysis. Journal of Engineering for Gas Turbines and Power 1990;112:117–21.
[6] Sarabchi K. Parametric analysis of gas turbine cogeneration plant from first and second law viewpoints. IGTI ASME COGEN-TURBO 1992;7:485–91.
[7] Rosen MA, Le M. Efficiency measures for processes integrating heat and power and district cooling. AES Thermodynamics and the Design Analysis and Improvement of Energy Systems 1995;35:423–34.
[8] Korobitsyn MA, Hirs GG. Analysis of cogeneration alternatives. In: Cogen-Turbo Power Conference, Vienna, Austria, CTP-11, ASME, 1995.
[9] Lior N, Arai N. Analysis of the chemical gas turbine system, a novel high-efficiency low emissions power cycle. In: EC-Vol. 5, ASME 1997 International Joint Power Generation Conference, vol. 1, ASME, 1997. p. 431–7.
[10] Lior N, Arai N. Performance and heat transfer considerations in the novel chemical gas turbine system. In: Proc. AIAA/ASME Thermophysics and Heat Transfer Conference, vol. 4, 1998. p. 33–9.
[11] Szargut J, Morris DR, Steward FR. Exergy analysis of thermal, chemical, and metallurgical processes. New York: Hemisphere, 1988.
[12] Peters MS, Timmerhaus KD. Plant design and economics for chemical engineers. New York: McGraw-Hill, 1991.
[13] Perry RH, Green DW, Maloney JO. Chemical Engineers' Handbook. 6th ed. New York: McGraw-Hill, 1984.
[14] Guthrie KM, Grace WR. Data and techniques for preliminary capital cost estimating. Chem Engineering 1969;76(6):114–42.
[15] De Brasi V, Farmer R. 1988–1989 Handbook gas turbine world, vol. 11. Fairfield, CT: Pequot, 1989.