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TECHNICAL AND ECONOMIC ANALYSIS OF STEAM-INJECTED GAS-TURBINE COGENERATION

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ABSTRACT

Industrial cogeneration is gaining popularity as an energy and money saving alternative to separate steam and electricity generation. Among cogeneration technologies, gas-turbine systems are attractive largely because of their lower capital cost and high thermodynamic efficiency. However, at industrial plants where steam and electricity loads vary daily, seasonally, or unpredictably, the economics of conventional gas turbines are often unfavorable due to low capacity utilization.

Steam-injected gas-turbine cogeneration overcomes the part-load problem by providing for excess steam to be injected back into the turbine to raise electrical output and generating efficiency. Under provisions of the Public Utilities Regulatory Policies Act, any excess electricity can be sold to the local grid at the prevailing avoided cost of electricity. Steam-injected gas-turbine cogeneration can result in a consistently high rate of return on investment over a wide range of variation in process steam loads. Moreover, this technology can also give rise to greater annual electricity production and fuel savings per unit of process steam generated, compared to simple-cycle cogeneration, making the technology attractive from the perspective of society, as well as that of the user.

Steam-injected gas-turbines may soon find applications in electric utility base-load generation, as well, since it appears that electrical generating efficiencies in excess of 50% can be obtained from turbines producing of the order of 100 MW of electricity at a fully-installed capital cost as low as $500/kW.

INTRODUCTION

Industrial cogeneration was made economically more attractive by the 1978 passage of the Public Utilities Regulatory Policies Act (PURPA). The Act, and the rules implementing it, which have been upheld in Supreme Court decisions of 1982 and 1983, require electric utilities to (i) purchase power from qualifying cogenerators at a price that reflects the costs the utilities would avoid by not having to provide the electricity themselves and (ii) provide back-up power at rates that do not discriminate against cogenerators.

Commercially available cogeneration technologies include gas turbines, steam turbines, gas turbine/steam turbine combined cycles, and diesel engines. One technology is chosen over another primarily based on economics, as influenced by factors such as relative fuel prices, security of fuel supply, the size and variability of particular steam and electricity loads, the reliability of the system, and environmental constraints.

Gas-turbine cogeneration is attractive largely because of its low installed capital costs compared to competing technologies (see Fig. 1), and its high thermodynamic efficiency in baseload applications (e.g., for meeting process heat loads in capital-intensive, energy-intensive industries characterized by relatively constant steam loads).

The recently introduced steam-injected gas-turbine extends the range of applicability of gas turbines to cogeneration applications characterized by variable heating loads -- applications for which conventional gas turbines are generally not well suited.

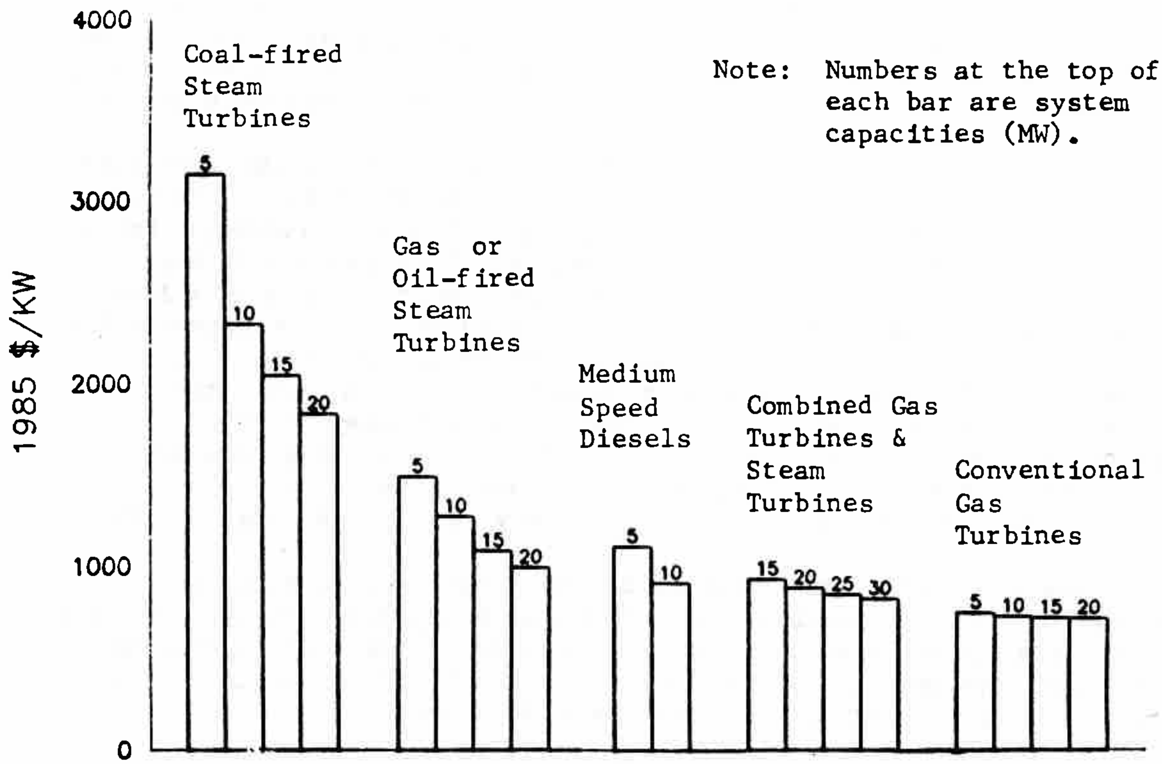


Fig. 1. Installed capital costs of small cogeneration systems.[1]

TECHNICAL PERFORMANCE MEASURES OF COGENERATION

Several technical performance indicators provide a basis for analyzing all cogeneration technologies.

First-Law Efficiency: The most universally understood measure is probably the overall first-law efficiency: the ratio of the energy in the useful heat plus the electricity to the heating value of the fuel consumed. For the separate production of electricity and steam (e.g., in a central station power plant and a stand-alone boiler), first-law efficiencies range from 60 to 70%. Steam-turbine cogeneration efficiencies are typically 80 to 90%; those for gas turbines and combined cycles, 70-80%; and for diesel cycles, 60-70% (see Table 1).

TABLE 1. TYPICAL ENERGY PERFORMANCE INDICATORS FOR ALTERNATIVE COGENERATION SYSTEMS.[2]

	FUEL FRACTION CONVERTED TO		OVERALL 1st LAW EFF.	FUEL CHARGEABLE TO POWER IN kJ/kJ AND (BTU/kWh)(b)	ELECTRICITY-STEAM RATIO kJ/kJ AND (kWh/MBTU)	SECOND LAW EFF.(c)	FUEL SAVINGS RATE (kJ FUEL PER kJ TO PROCESS STEAM)
	ELEC	STEAM(a)					
STEAM TURBINE(d)	0.12	0.73	0.85	1.38 (4700)	0.17 (50)	0.40 [0.34]	0.27
GAS TURBINE(e)	0.30	0.46	0.76	1.60 (5450)	0.64 (188)	0.48 [0.35]	0.86
COMBINED CYCLE(f)	0.35	0.42	0.77	1.49 (5080)	0.84 (245)	0.52 [0.35]	1.21
DIESEL(g)	0.35	0.25	0.60	2.02 (6900)	1.38 (405)	0.45 [0.35]	1.01

(a) SATURATED STEAM AT 1 MPa (150 PSIG).
(b) ASSUMING A STAND-ALONE BOILER EFFICIENCY OF 88%.
(c) ASSUMING FEEDWATER AT 100°C (212°F) AND AN AMBIENT TEMPERATURE OF 15°C (59°F). BRACKETED NUMBERS ARE SECOND-LAW EFFICIENCIES FOR SEPARATE STEAM PRODUCTION AND ELECTRICITY GENERATION AT A CENTRAL STATION PLANT.
(d) FOR SYSTEMS PRODUCING OF THE ORDER OF 50-100 MW OF ELECTRICITY.
(e) FOR A GAS TURBINE/STEAM TURBINE SYSTEM PRODUCING 73.8 MW OF ELECTRICITY.
(f) FOR A SYSTEM PRODUCING 87 MW OF ELECTRICITY.
(g) FOR UNJACKETED DIESELS PRODUCING OF THE ORDER OF 20 MW OF ELECTRICITY.

Fuel Chargeable To Power: Another measure is the amount of fuel chargeable to power (FCP), defined as the fuel required in a cogeneration facility in excess of that required to produce steam in a separate (stand-alone) boiler. The FCP is expressed dimensionlessly as the fuel energy charged per unit of electrical energy produced:

$$FCP = (1 - S/\eta_b)/E, \qquad (1)$$

where S and E are the fractions of the fuel converted to steam and electricity, respectively, and η_b is an assumed efficiency for the stand-alone boiler. The FCP is commonly expressed in units of BTU/kWh, in which case Eqn. 1 becomes

$$FCP = 3413 * (1 - S/\eta_b)/E. \qquad (1a)$$

At a central station generating plant, all fuel is charged to electricity production, typically yielding a FCP of about 2.93 kJ/kJ (10,000 BTU/kWh). For steam-turbine cogeneration, the FCP is typically less than half this value -- about 1.4 kJ/kJ (4700 BTU/kWh). For gas-turbine systems, it is of the order of 1.6 (5500), for combined cycles, 1.5 (5000), and for diesel engines, 2.0 (7000) (see Table 1).

The first-law efficiency and the FCP can be somewhat misleading indicators of overall thermodynamic performance because they account primarily for quantity rather than quality of energy. Thus, they tend to favor cogeneration schemes that produce a relatively small amount of electricity per unit of process steam, such as steam-turbine systems, which typically convert only 10-15% of the fuel into electricity, but 65-75% into process steam (see Table 1). The higher electricity-to-steam ratios of gas-turbines, combined cycles, and diesels mean they typically produce 4-8 times more electricity per unit of process steam than steam-turbines (see Table 1), and thus have the largest potential for generating electricity in excess of onsite needs for sale back to the grid.

Second-Law Efficiency: A better measure of overall thermodynamic performance, which takes into account the high thermodynamic quality of electricity, is the second law efficiency, defined as the ratio of the minimum available work required to produce steam plus electricity, B_{min}, to the actual available work used up in the process, B_{act}.[3] For a cogeneration system producing saturated process steam at a temperature, T, and pressure, P,[4]

$$B_{min} = mv(p-p_o) + mc_p[(T-T_1)-T_o\ln(T/T_1)] + mh_v[1-(T_o/T)] + E, \quad (2)$$

where m is the mass of saturated process steam at p, p_o and T_1 are the inlet pressure and temperature of the feedwater, v is the specific volume of the feedwater, c_p is the specific heat (at constant pressure) of the feedwater, and h_v is the enthalpy of vaporization of water at pressure p. The available work in fossil fuels roughly equals the lower heating value (LHV)[3] — the heat released by complete stoichiometric combustion of the fuel, with water in the products remaining in vapor form (The higher heating value (HHV) is equal to the lower heating value plus the heat released by condensing the product water.)

As shown in Table 1, second law efficiencies for separate steam and central station electricity generation are all about 0.35, while those for cogeneration range from 0.40-0.52, with the highest of these values corresponding to systems characterized by the highest electricity-to-steam ratios -- gas-turbines, combined cycles, and diesels.

The Fuel Savings Rate: One additional measure, the fuel savings rate (FSR), also takes account of energy quality, but is somewhat less abstract than the second law efficiency. Defined as the amount of fuel saved in producing electricity (by cogeneration rather than central station generation) for each unit of process steam generated,[4] the FSR is commonly expressed in dimensionless terms (e.g., kJ of fuel per kJ to steam) and is calculated from

$$FSR = (2.93 - FCP) * ESR, \quad (3)$$

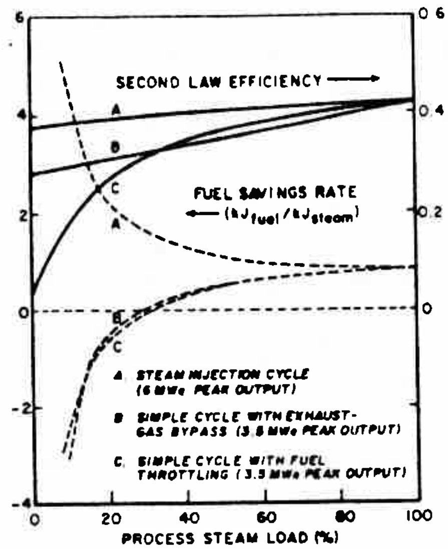
in which ESR is the electricity-to-steam ratio.

As Table 1 indicates, the greatest fuel savings come from the cogeneration technologies that produce the largest amount of electricity per unit of process steam -- gas-turbines, combined cycles, and diesels.

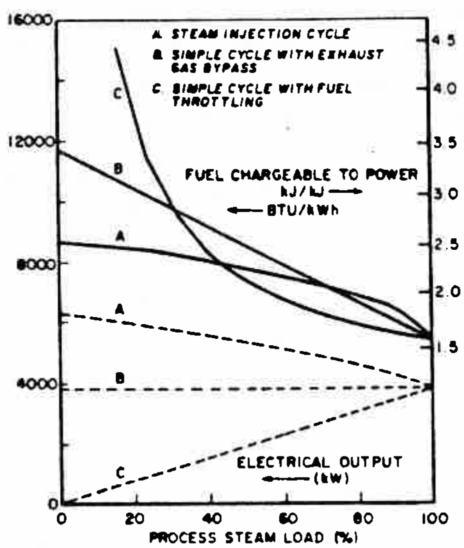
PART-LOAD PERFORMANCE OF GAS-TURBINE COGENERATION SYSTEMS

The attractive thermodynamics of gas-turbine cogeneration systems operating at full-load, as discussed in the preceding section, tends to degrade when they are operated at part-load for a sizeable fraction of the time, as is the case in many smaller scale industrial and commercial applications.

The second law efficiency and the FSR drop precipitously in conventional gas-turbine systems when the process steam demand drops



(a)



(b)

Fig. 2. Part-load gas-turbine cogeneration performance, based on the Detroit Diesel Allison 501-KB turbine.[5] Case A is with steam injection (6 MW peak output); Case B is a conventional system with exhaust-gas bypass (3.5 MW peak); Case C is a conventional system with the fuel flow throttled (3.5 MW peak).

and the fuel input to the turbine is throttled, as shown in Fig. 2a (case C)[6]. In addition, the FCP rises sharply and the electrical output drops off (see Fig. 2b). Because of these poor part-load performance characteristics, the more commmon mode of operation is to bypass the steam generating system with the hot exhaust gases, allowing full electrical production to continue at the expense of an increasing FCP (see Fig. 2b, case B). The second law efficiency falls off somewhat more slowly (see Fig. 2a, case B), but the FSR drop is equally sharp. The unfavorable economics resulting from poor part-load performance has restricted the use of conventional gas turbines largely to applications where steam loads are relatively constant.

The recently-commercialized steam-injected gas-turbine, however, overcomes problems associated with part-load operation by providing (through only minor hardware modifications) for excess steam to be injected into the combustor of the turbine system. As a result of the greater mass and energy flows through the turbine, obtained without additional compressor work, net electricity production increases significantly, but the FCP increases only moderately (see Fig. 2b, case A). The second law efficiency remains essentially constant for all process steam loads, while the FSR actually rises as the process load drops (Fig. 2a).

One of the keys to the viability of steam-injected gas turbines in cogeneration applications is the availability of markets for excess electricity, which now exist as a result of the PURPA legislation. In addition, general apprehensions several years ago to investing in natural gas-fired equipment, arising from fears of significant increases in natural gas prices, have proved to be unfounded. The average price of natural gas paid by industry in the U.S. (in inflation-corrected dollars) is currently projected to fall by about 12%, 1983-1995.[7]

There is also interest in the use of steam injection by utilities for the generation of power only, due to the fact that the FCP when no process steam is required is significantly lower than that for most large central station power plants, even for the very small gas turbines focussed on in this article (see Fig. 2b)! For moderately-sized systems (~100MW), it appears that FCPs of the order of 1.8 kJ/kJ (6200 BTU/kWh) are attainable.[8]

A REVIEW OF CONVENTIONAL GAS-TURBINE COGENERATION THERMODYNAMICS

Conventional gas-turbine cogeneration systems embody an open Brayton cycle, with heat in the turbine exhaust used to produce steam in a heat-recovery steam generator (HRSG).

Brayton Cycle Calculations:[9] As indicated in Fig. 3, the standard open-cycle gas turbine system incorporates: a compression process (state 1 to 2), requiring an input of work; a heat addition process (2 to 3); and an expansion process (3 - 4), by which work is produced. Hardware constraints will dictate: compressor pressure ratio ($P_{rC} = P_2/P_1$); turbine inlet temperature (T_3); the mass flow rate through the compressor (M_C); compressor adiabatic efficiency (η_C), defined as the work input required to isentropically compress a unit mass (w_{Cs}) divided by the actual work (w_C); turbine adiabatic efficiency (η_T), defined as the actual work output per unit mass (w_T) divided by the work output for isentropic expansion (w_{Ts}); and gearbox-generator efficiency (η_g), defined as the ratio of the electrical work output of the system divided by the net shaft work output.

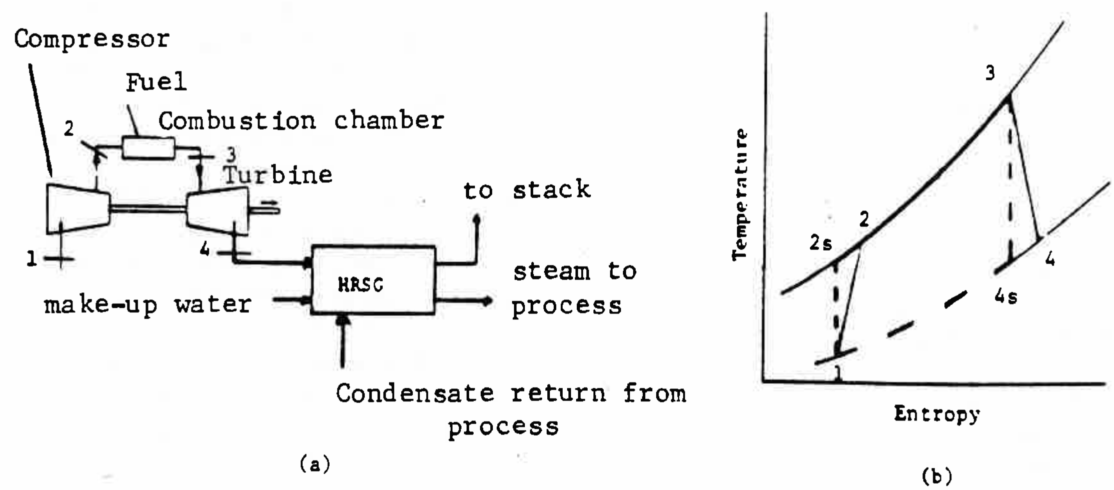


Fig. 3. Schematic (a) and temperature-entropy diagram (b) for a gas-turbine simple-cycle cogeneration system.

To calculate the net power output and generating efficiency of this cycle, the temperature of the compressor discharge is first determined. Assuming that the working fluid (usually air) behaves as an ideal gas (an assumption used throughout this article), so that its enthalpy is a function of temperature only [$h = h(T)$], and that its specific heat at constant pressure, c_{pC} is constant (so that $\Delta h = c_{pC} * \Delta T$), the compressor outlet temperature is obtained from the isentropic compressor efficiency:

$$\eta_C = w_{Cs}/w_C = c_{pC} * (T_{2s} - T_1)/c_{pC} * (T_2 - T_1) \tag{4}$$

or

$$T_2 = T_1 + (T_{2s} - T_1)/\eta_C. \tag{5}$$

In Eqn. 4, the specific heats are evaluated at the mean of the temperature range with which they are associated. In this case, since the difference between T_2 and T_{2s} is small, the specific heats cancel, as indicated by Eqn. 5. For isentropic compression,

$$T_{2s} = T_1 * (P_2/P_1)^{[(k-1)/k]} = T_1 * P_{rC}^{[(k-1)/k]}, \tag{6}$$

where k is the ratio of the specific heat at constant pressure to that at constant volume.

The heat input required to the combustor per unit mass of working fluid (q_{in}), assuming constant pressure combustion and a constant specific heat, c_{pcb}, is

$$q_{in} = c_{pcb} * (T_3 - T_2). \tag{7}$$

In practice, heat addition adds mass -- the products of combustion -- to the working fluid and changes its specific heat. However, since typical turbine air-to-fuel ratios are in the neighborhood of 40:1, for a first-order analysis, the changes in mass flow and specific heat can be neglected.

The turbine exhaust temperature is calculated from the turbine isentropic efficiency,

$$\eta_T = w_T/w_{Ts} = c_{pT} * (T_4 - T_3)/c_{pT} * (T_{4s} - T_3) \tag{8}$$

or

$$T_4 = T_3 - \eta_T * (T_3 - T_{4s}). \tag{9}$$

Defining the turbine pressure ratio as

$$P_{rT} = P_4/P_3, \tag{10}$$

then

$$T_{4s} = T_3 * (P_{rT})^{[(k-1)/k]}. \tag{11}$$

For a first order analysis, P_{rT} can be assumed to equal the inverse of P_{rC}.

The net work output per unit mass is simply the sum of the actual compressor and turbine work, expressions for which are given in Eqns. 4 and 8:

$$w_{net} = w_T + w_C \tag{12}$$

(Note that work and heat input to the system are defined to be positive.) The net shaft and electrical power outputs are

$$P_s = M_T * w_T + M_C * w_C \tag{13}$$

$$P_e = P_s * \eta_g \tag{14}$$

and the cycle shaft and electrical power production efficiencies are

$$\eta_s = |w_{net}/q_{in}| \tag{15}$$

$$\eta_e = \eta_s * \eta_g \tag{16}$$

Equations 4-16 constitute a useful set of equations for doing gas-turbine analysis, as illustrated in Fig. 4, which indicates that at a fixed TIT, one value of P_{rC} maximizes work output, and another maximizes efficiency (see Fig. 4).

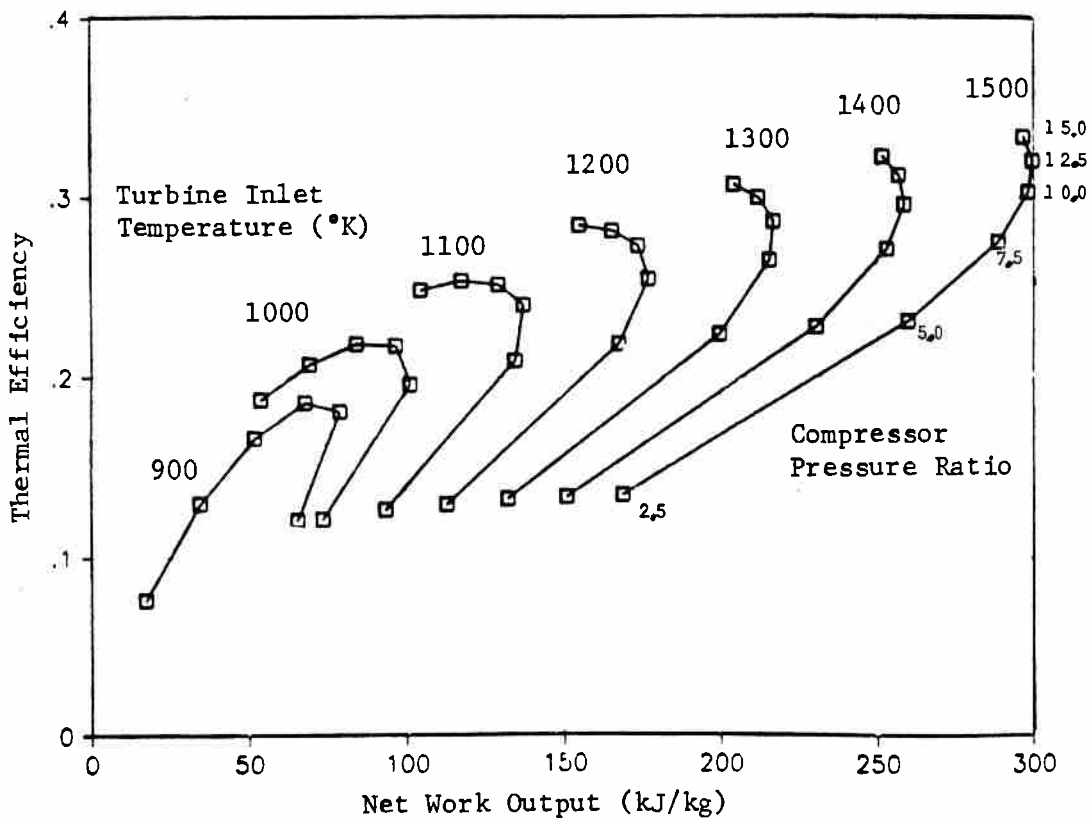


Fig. 4. Performance map for a gas-turbine simple-cycle.

<u>The Heat-Recovery Steam Generator:</u>[10] The heat-recovery steam generator (HRSG) usually consists of an economizer, in which the feedwater (part cold make-up and part condensate return) is heated to near its saturation temperature, and a boiler in which the water is converted to saturated steam. Most industrial processes use

saturated steam, but if superheated steam is required, a superheater follows the boiler.

In some applications, a supplemental firing system -- a duct burner -- is inserted between the turbine exit and the HRSG to increase the steam-generating capacity of the HRSG. Ample oxygen exists in the turbine exhaust to permit supplementary fuel to be completely combusted in the duct burner. The high temperature of the turbine exhaust gases results in very high combustion efficiencies and correspondingly high steam-generating efficiencies[11,12].

Energy Balances: For pedagogical purposes, the three sections of an HRSG can be considered as a single counterflow shell-and-tube heat exchanger, with the hot exhaust gases (from the turbine or the duct burner) passing in one direction on the shell side and water and/or steam traveling through tubes in the opposite direction. The temperature profiles in the HRSG as functions of the percent of heat transferred are shown in Fig. 5.

The gas-side temperature profile is essentially linear in this figure, since the heat tranferred per unit time from the gas, Δq, along any section from one point to another can be expressed as

$$\Delta q = M_g * c_{pg} * \Delta T \tag{17}$$

where M_g is the gas mass flow rate, c_{pg} is the specific heat of the gas per unit mass (which is assumed constant at its average value in the HRSG), and ΔT is the change in gas temperature between the two points. The water-side profiles will also be essentially linear, but with a different slope for each section of the HRSG.

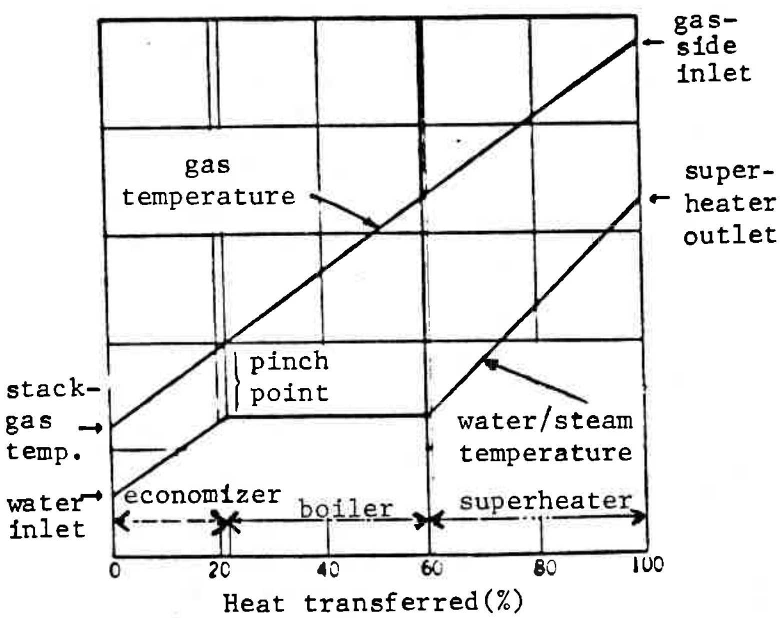


Fig. 5. Typical temperature profiles in a heat recovery steam generator.

In the HRSG of most cogeneration systems using air and water as the working fluids, the closest point of approach between the gas and liquid temperatures at any point will be at the inlet to the boiler, the so-called pinch point (see Fig. 5), where the temperature difference between the gas side and the water side is defined as ΔT_{pp}. From the gas-side inlet, where $T=T_{gi}$, to the pinch point, the total heat released per unit time by the gas is

$$Q_g = M_g * c_{pg} * [T_{gi} - (T_{sat} + \Delta T_{pp})] \qquad (18)$$

Typically 1-2% of this energy is lost by radiation or other means,[13] so that the heat absorbed by the water is

$$Q_{ws} = 0.98 * Q_g \qquad (19)$$

The enthalpy change per unit mass of water between the pinch point and the point it leaves the HRSG is

$$\Delta h = h_s - h_f \qquad (20)$$

where h_s is the enthalpy of the superheated steam exiting the HRSG and h_f is the enthalpy of saturated liquid at the HRSG pressure. Note that for any given system, the upper limit on the steam temperature at the superheater outlet (and hence on h_s) is determined by the temperature of the turbine exhaust gas at that point. Based on practical considerations,[2] the minimum difference between these two temperatures, $(\Delta T_{so})_{min}$, is commonly in the range 15-30°C (30-50°F).

The mass flow rate of steam is calculated from Eqns. 19 and 20:

$$M_{st} = Q_{ws}/\Delta h \qquad (21)$$

Decreasing $(\Delta T_{so})_{min}$ (i.e., increasing the superheat of the steam) has only a relativelyminor effect on the mass of steam that is generated.[2] The stack-gas temperature can be obtained from an energy balance on the economizer:

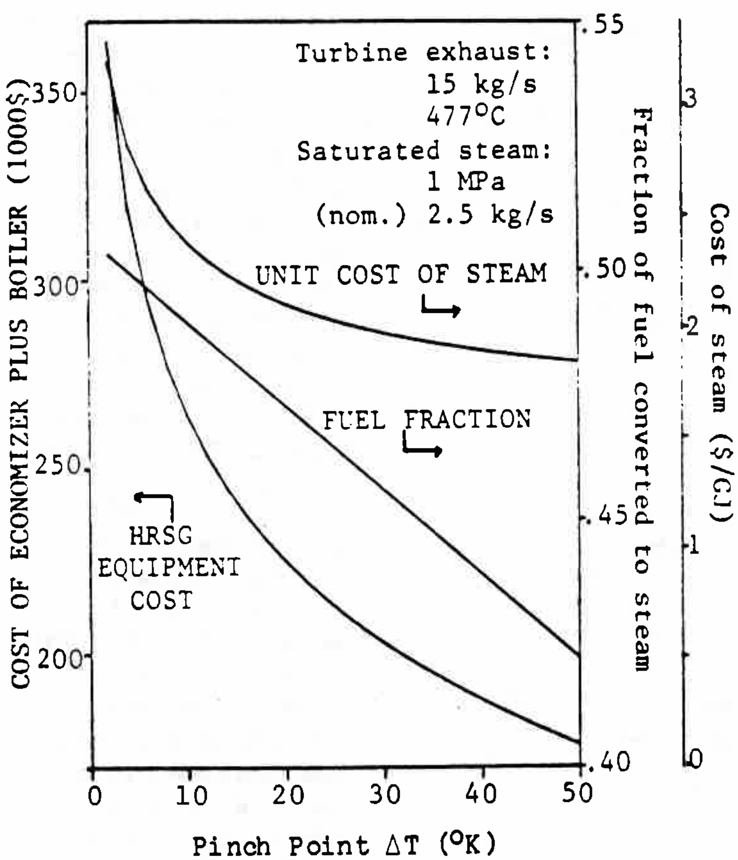


Fig. 6. The influence of the pinch point on a gas-turbine simple cycle cogeneration system.[2] The unit cost of steam represents the capital equipment fraction of the total cost of steam.

$$T_{stack} = T_{sat} + \Delta T_{pp} - \{M_{st}*(h_f - h_{fw})*(1 + B)/[(M_g*c_{pg})*0.98]\}. \quad (22)$$

Here h_{fw} is the enthalpy of saturated water at T_{fw} (a weighted average of the known make-up and condensate return temperatures) and B is the continuous blow-down fraction -- an amount of saturated liquid at T_{sat} (expressed as a fraction of the steam flow) that is continuously added to and subsequently flushed from the system to prevent mineral buildup in the boiler. Typically, B is about 0.05.[13] To ensure that condensation does not occur, T_{stack} must be above the dew point of the exhaust gas, which is typically in the neighborhood of 60-70°C (140-160°F). Stack gas temperatures are generally maintained at 150°C (300°F). (If there is a use for low temperature heat beyond that needed in the economizer, additional energy equivalent to up to 20% of that already extracted in the HRSG can be extracted from the turbine exhaust before condensation occurs.)

The Influence of the Pinch Point: As is evident from Eqns. 18-22, for a fixed process steam quality and HRSG gas-side inlet temperature, ΔT_{pp} determines the steam flow that can be generated, or alternatively, the fraction of the original fuel that is converted to steam, which is a linearly decreasing function of ΔT_{pp} in a typical system (see Fig. 6). The pinch point is also important in determining the HRSG heat-transfer surface area requirements, and consequently capital cost, which varies roughly as the inverse of the logarithm of ΔT_{pp} (see Fig. 6).

The net influence of decreasing ΔT_{pp} is to increase the capital investment required per unit of steam generated. As seen in Fig. 6, this cost rises sharply in the neighborhood of 15-20°C (27-36°F) pinch points, a range typical of current HRSG design practice.[14]

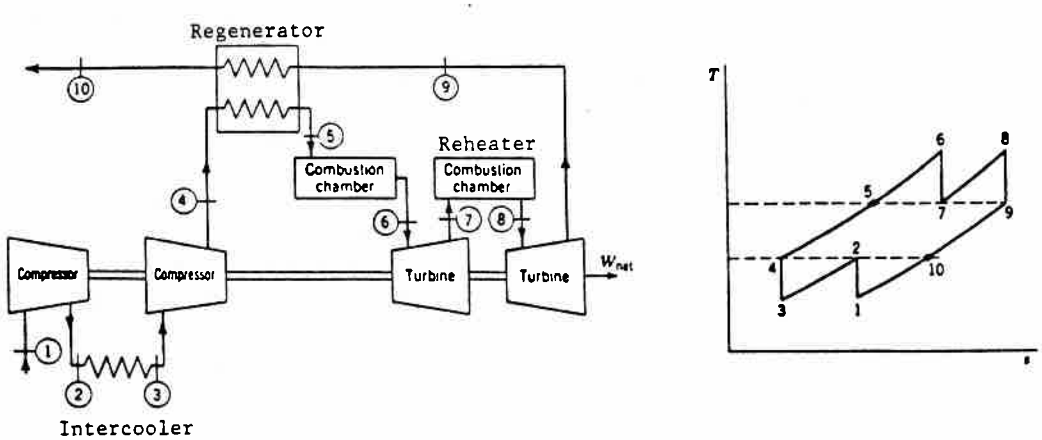


Fig. 7. Schematic and temperature-entropy diagram for alternative gas-turbine configurations for generating power.

Textbook Variations of Simple-Cycle Cogeneration: Simple-cycle gas turbine systems are by far the most commonly used in practice, but the basic analyses of the Brayton cycle and the HRSG presented above can be applied to alternative gas-turbine cycle configurations involving combinations of: heat exchange between the turbine exhaust gases and the cooler compressor outlet gases through use of a regenerator to reduce overall fuel requirements; multiple-stage compression with intercooling between each stage to decrease overall compressor work; and multiple-stage expansion with reheating between turbine stages to increase overall turbine work. These alternatives are shown schematically in Fig. 7.

Results of calculations based on these alternatives (Fig. 8) indicate that gas-turbine systems can operate over a wide range of conditions. Such more-elaborate systems could gain greater popularity in the future, but it appears that the higher cost and greater complexity (and hence more difficult maintenance problems) associated with regeneration and multiple-stage compression and expansion cycles have acted thus far to limit the actual implementation of these often higher efficiency and/or output systems[15].

THE THERMODYNAMICS OF STEAM-INJECTED GAS-TURBINE COGENERATION

Injecting water or steam into a gas turbine is not a new idea, but only recently have steam-injection cogeneration systems been introduced. The steam-injected gas-turbine (STIG) cycle is described in textbooks[16,17] and in a number of other publications (see, e.g., references 18-21). The use of water (not steam) injection for short periods was commmon in the past for thrust augmentation in jet-aircraft engines, although this is now usually done with afterburners.[15] Water is injected into the combustors of many stationary gas turbines used today to keep combustion temperatures low, thereby suppressing the formation of NO_x pollutants,[22,23] to comply with pollutant emissions regulations in many states.

The passage of PURPA has focussed attention on the steam-injection technology for cogeneration applications, primarily because of its flexible and highly

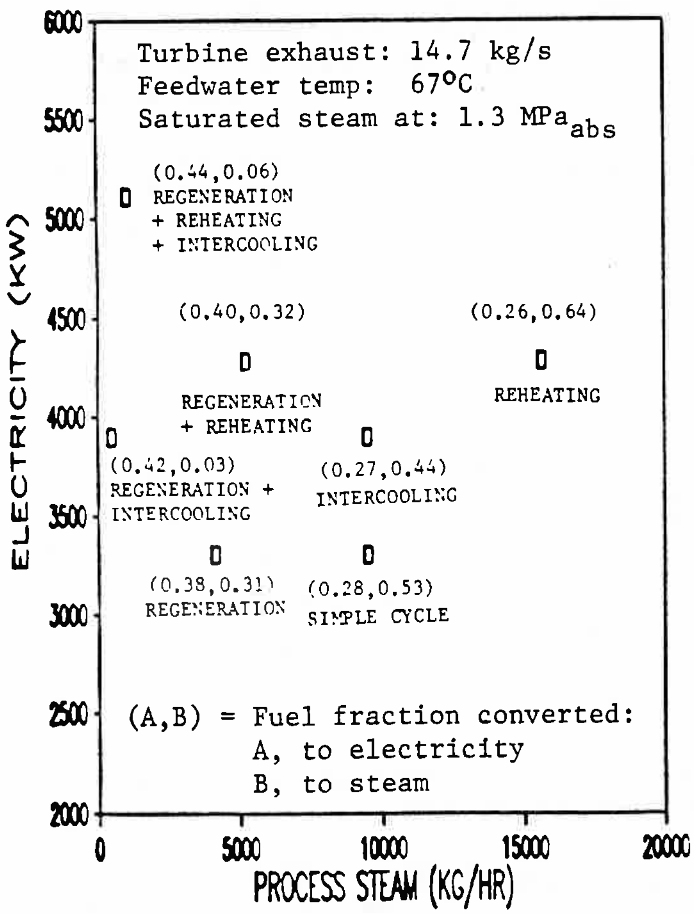


Fig. 8. Performance characteristics for alternative simple-cycle cogeneration configurations.[2]

efficient operation.[24,25] Two US companies -- International Power Technology (IPT), Palo Alto, CA[26] and Mechanical Technology, Inc. (MTI), Latham, NY[27] -- now offer packaged STIG cogeneration systems based on the Allison 501-KH turbine (the 501-KB modified for steam injection). IPT has installed a single unit at San Jose State University, San Jose, California and dual units at a Sunkist Growers, Inc. processing plant in Ontario, CA.[26] Dah Yu Cheng of IPT holds a patent[27] which claims rights to the operation of any STIG cycle in the region of its peak electrical efficiency, which as subsequent calculations will illustrate, is defined uniquely for each STIG cycle.

A "Back-of-the-Envelope" Approach to the STIG Cycle: The basic operation of the STIG cycle involves generating steam with the turbine exhaust heat and injecting some or all of it back into the combustor (see Fig. 9). Thus, unlike the simple-cycle cogeneration calculations, the HRSG and turbine analyses are coupled. As subsequent calculations will show, the peak electrical efficiency for a DDA-501 based system occurs when a mass flow of steam somewhat greater than 15% of the air flow is injected into the combustor. Based on this percentage, a 4-step "back-of-the-envelope" calculation can be made to illuminate the thermodynamics of steam-injected gas-turbine cycles.[30]

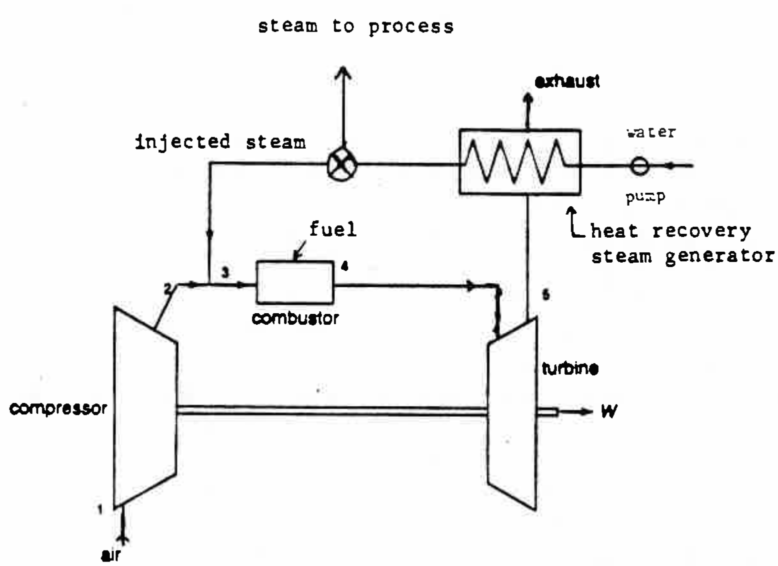


Fig. 9. Schematic of a steam-injected gas-turbine cogeneration cycle.

Step 1: Reference Case: Based on a simplified calculation using air as the working fluid, an Allison 501-KB turbine in a simple cycle will produce about 3350 kW at an efficiency of 27%.

Step 2: "Free" Extra Mass: One result of injecting steam is to increase the mass flow through the turbine. To help understand the mass effect, suppose that 15% additional air (not steam) is supplied (in an unspecified manner) to the turbine inlet at the required temperature and pressure.

In most gas turbines, the working fluid flows from the combustor to the turbine through a nozzle in which the flow is sonic, and thus choked, so that additional mass cannot pass without an increase in the pressure upstream of the nozzle.[15] Introducing additional mass there induces a pressure rise to the level which will permit the injected mass to flow through the nozzle. This pressure rise is felt at the compressor outlet and results in a rise in the compressor pressure ratio, since compressors are designed to operate with a constant mass flow.

As a result, the pressure ratio of the compressor for the Allison 501 turbine is increased according to:[31]

$$\Delta P_{rC} = (13.47 / M_C)*\Delta M_T, \tag{23}$$

where ΔM_T is the increase in the turbine mass flow with steam injection (in kg/s).

Using Eqn. 23 together with differentiated expressions for power and efficiency developed from Eqns. 4-16, changes in power output and efficiency that accompany changes in the turbine mass flow can be determined:

$$\Delta(P) = 574*\Delta M_T \tag{24}$$

$$\Delta(\eta) = 0.056*\Delta M_T \tag{25}$$

where ΔP is in kW when ΔM_T is expressed in kg/s. Adding the increments predicted by these equations to the power and efficiency calculated in Step 1 yields an output of 4625 kW and an efficiency of 38%, both of which are significantly higher than the reference case.

<u>Step 3: Paying for the Extra Mass</u>: The efficiency increase is large in step 2 because no account was taken of the work and heat needed to raise the extra mass to the turbine inlet conditions.

Since we are ultimately interested in steam, in this step we will continue to neglect the work required to raise the mass to the turbine inlet pressure (pumping a liquid requires negligible work compared to compressing air). However, despite the fact that we can recover enough energy from the turbine exhaust to create steam for injection, additional heat must be supplied in the combustor to heat the steam from its injection temperature, T_{inj}, to the turbine inlet temperature, TIT. The total heat added, is, therefore,

$$Q_{in} = M_C*c_{pcb}*(TIT - T_2) + (\Delta M_T)*c_{pinj}*(TIT - T_{inj}), \tag{26}$$

where T_2 is the compressor outlet temperature, and c_{pcb} and c_{pinj} are the specific heats of air and the injected fluid at their respective average temperatures in the combustor.

Equation 24 for the change in power output is applicable for this step, but Eqn. 25 is replaced by

$$\Delta(\eta) = 0.028*\Delta M_T \tag{27}$$

where ΔM_T is expressed in kg/s. Because of the additional heating that is required for the same output, the efficiency drops from 38 to 34%.

<u>Step 4: The Specific Heat Effect</u>: In this last step, we account for the increase in specific heat of the mass flowing through the

turbine that occurs when steam is injected. The specific heat per unit mass of air flow in the turbine will be

$$c_{pT} = S/A * c_{ps} + c_{pair} \tag{28}$$

where S/A is the mass ratio of steam to air. Values of c_p for steam[32] and air[33] (in units of kJ/kg-K) are given by

$$c_{ps} = 4.6 - 103.6*T^{-0.5} + 967.2*T^{-1} \tag{29}$$

$$c_{pair} = 1.003 + 1.816x10^{-4}*T \tag{30}$$

where T (in Kelvin) is the average temperature at which the process occurs. For a S/A = 0.15, c_{pT} is about 25% higher than c_{pair}.

Since 2 variables are changing in this case, the changes in power output and efficiency are

$$\Delta(P) = 768*\Delta M_T + 2027*\Delta c_{pT} \tag{31}$$

$$\Delta(\eta) = 0.028*\Delta M_T + 0.176*\Delta c_{pT}, \tag{32}$$

where ΔP is in kW when ΔM_T is in kg/s and Δc_{pT} is in kJ/kg-K. Because the specific heat of steam is roughly double that of air, both the turbine work output and the heat input requirements increase, with a net result that 5420 kW of electricity are produced at an efficiency of 39%

Summary: Steps 2 and 3 of this "back-of-the-envelope" calculation demonstrate that since the compressor typically consumes 1/2-2/3 of the total turbine work output in a simple-cycle gas-turbine, if extra mass can be provided to the turbine without requiring additional compression work (as is the case with steam), both efficiency and net cycle output will increase. Step 4 demonstrates that the increase in specific heat of the fluid passing through the turbine has by far the most important effect on improving cycle performance. (For accurate calculations, therefore, accurate values for the specific heat should be used in the calculation.)

More Careful STIG Cycle Calculations: Accurately calculating actual STIG-cycle performance requires closer attention to detail than the simplified approach just described, but conceptually there are no differences, and indeed, the results of the "back-of-the-envelope" calculation turn out to be surprisingly accurate.

To calculate cycle efficiency and net power output as functions of S/A, the compressor discharge temperature and work requirements are first determined, as in the case of the simple Brayton cycle. Then (passing over the combustor for the moment), with the turbine inlet temperature (TIT) and S/A specified, the turbine calculation can be performed to give the turbine outlet temperature (TOT) and work output, also using the simple-cycle procedure, but with adjusted values for the specific heat parameters.

With the turbine exhaust temperature, the HRSG pressure, and the steam flow rate specified, the enthalpy of the superheated steam is determined by the pinch point temperature difference. The steam injection temperature (T_{inj}) at this enthalpy and pressure is obtained from the steam tables. If the difference between TOT and the calculated T_{inj} is less than the specified $(\Delta T_{so})_{min}$, then T_{inj} is set equal to TOT - $(\Delta T_{so})_{min}$ (which implies an increase in the pinch point temperature difference).

The total heat addition to the cycle can now be calculated using Eqn. 26, based on which cycle efficiency can be determined.

For an Allison 501-KH system, with TIT = 982°C (1800°F), ΔT_{pp} = 10°C (18°F), ΔT_{so} = 30°C (54°F), and T_{stack} > 150°C (300°F), the calculated peak cycle efficiency occurs at a S/A of about 0.17, as shown in Fig. 10. For S/A up to this value, the ($T_{so})_{min}$ constraint determines the superheat temperature of the steam, as there is enough energy and "temperature" in the turbine exhaust to heat all of the steam to the maximum specified value [=TOT-$(\Delta T_{so})_{min}$]. As S/A continues to increase beyond this critical point, there is insufficient energy in the turbine exhaust to raise all of the steam to this temperature, so additional energy input to the combustor is required. Beyond this critical point, the additional work derived from the larger mass flow through the turbine is more than offset by the increased fuel requirement, so that cycle efficiency begins to fall.

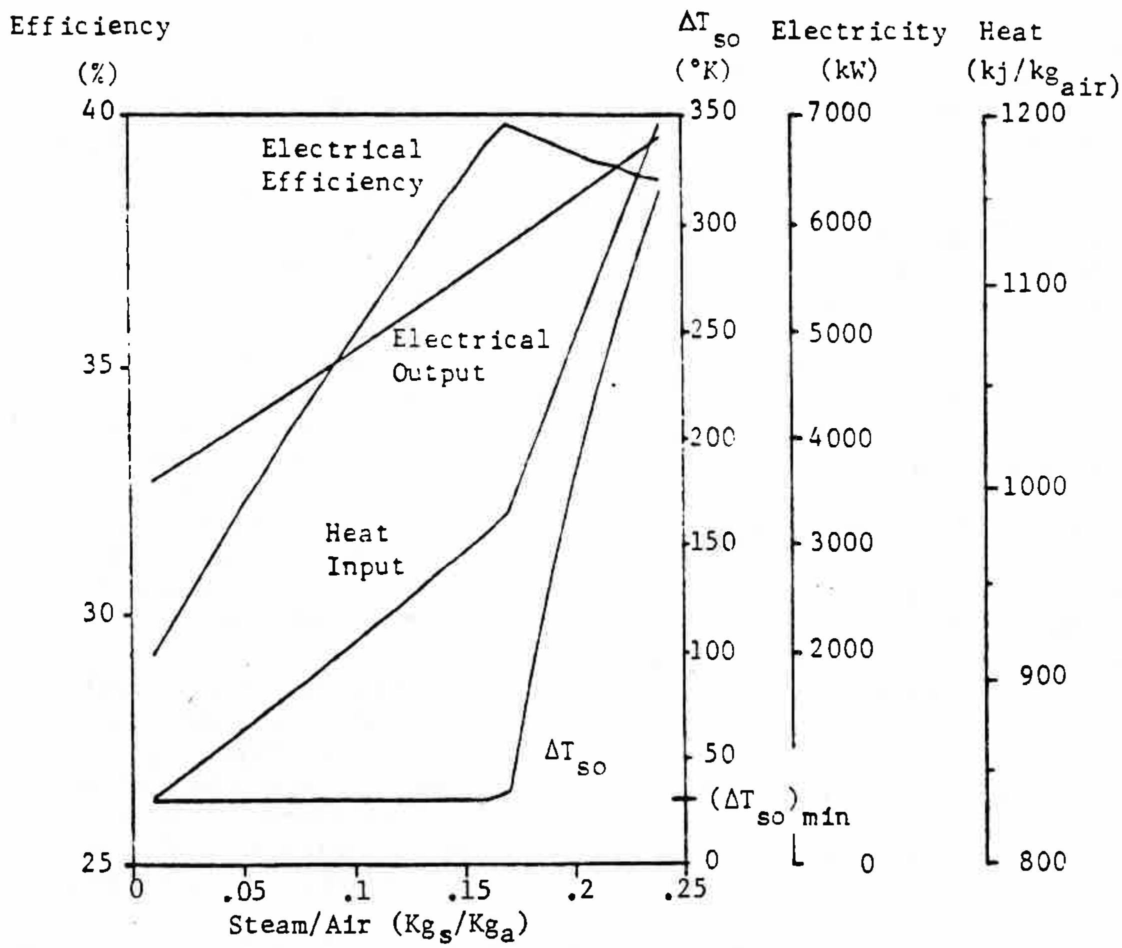


Fig. 10. Operating characteristics of a steam-injected gas-turbine cogeneration system based on the Allison 501-KH turbine.[5]

In a cogeneration application, as the process steam demand decreases, S/A increases. The corresponding increases in efficiency and output account for the attractive part-load performance seen earlier in Fig. 2.

STIG Hardware-Related Considerations: The operating regime of a STIG cogeneration system, defined in terms of electrical output and steam generation, can be considerably expanded by the use of supplemental firing in a duct burner, which is standard equipment on the systems offered by IPT and MTI. IPT claims their system will operate anywhere within "Region A" in Fig. 11, producing from 3.5 to 6 MW of electricity and from 0 to 13 MW (45 MBTU/hr) of process steam.[26]

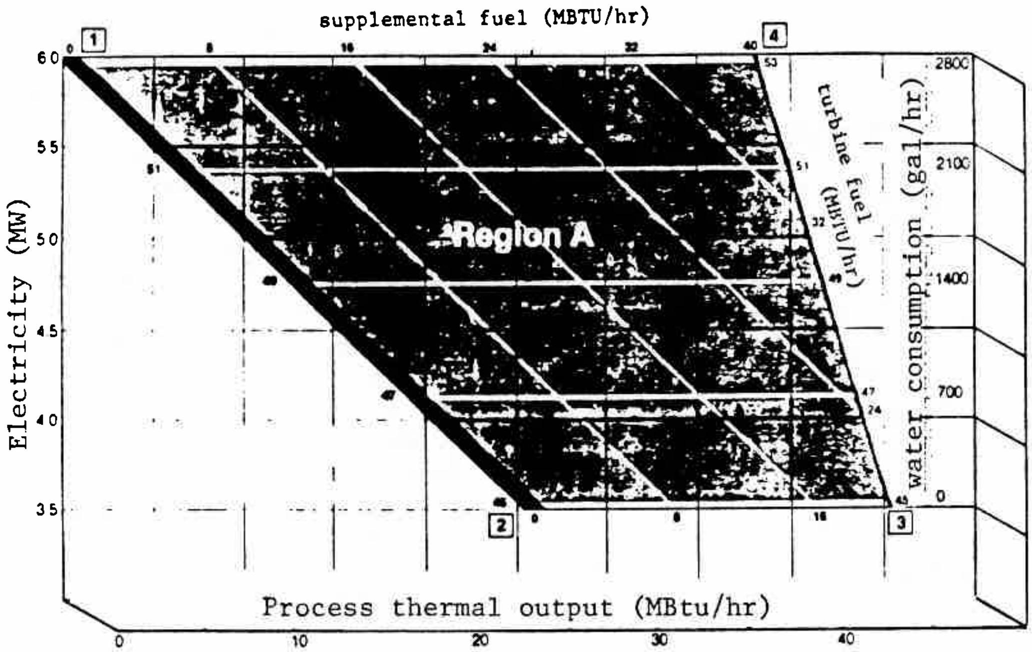


Fig. 11. Performance map for a commercial steam-injected gas-turbine cogeneration system with supplementary firing capability.[26]

STIG cycle thermodynamics are quite attractive, but a number of practical benefits also accompany steam-injection,[2] including generally lower peak operating temperatures, better cooling of turbine blades, NO_x pollution reduction, and better heat recovery in the HRSG (due to the higher specific heat of the turbine exhaust).

Not all turbines are as easily adapted for STIG operation as the Allison 501-KB, which is an aircraft-derivative turbine, and thus has a torque limit (6190 kW) considerably in excess of its design peak value (3500 kW). However, other aircraft-derivative turbines, such as the General Electric LM series, appear to be well suited for steam-injected operation.[8]

THE ECONOMICS OF STEAM-INJECTED GAS-TURBINE COGENERATION

Businesses often make investments based on an expected internal rate of return, determined by solving for the discount rate in an equation which equates the total initial investment required for the new equipment to the total discounted operating-cost savings that are expected to result from replacement of the existing facility.[34] To illustrate this method of assessing the economics of STIG cogeneration, we consider a hypothetical plant in California (where 3 STIG systems have already been installed) currently purchasing electricity from the grid and operating a stand-alone natural-gas fired boiler (83% efficient) to provide its process steam. To simplify the analysis, we assume the plant's full steam demand is 9090 kg/hr (20000 lb/hr) of 1.4 MPa (205 psig) saturated steam (at the HRSG) and full electrical demand is 3500 kW (corresponding to the maximum outputs of the IPT STIG-cycle cogeneration system with no steam injection and no supplemental firing -- point 2 in Fig. 11). We also assume an industrial plant load profile: full steam and electricity demand Monday-Friday from 7a.m. - 6p.m., and half demand at all other times. Finally, we neglect all subsidies and taxes in our analysis (investment tax credit, property tax, etc.)

TABLE 2. ESTIMATED CAPITAL AND INCREMENTAL OPERATING COSTS FOR AN ALLISON 501-KH-BASED STEAM-INJECTED GAS-TURBINE COGENERATION SYSTEM (6000 kW PEAK OUTPUT).[2]

FULLY INSTALLED CAPITAL COST	\$5 MILLION
INCLUDES: GAS-TURBINE GENERATOR HRSG WITH DUCT BURNER CONTROL SYSTEM BUILDING AND MISCELLANEOUS INSTALLATION	
INCREMENTAL OPERATING COSTS OVER A STEAM BOILER PLANT	
TURBINE OVERHAUL ONCE EVERY 3 YEARS @	\$ 220,000/OVERHAUL
WATER CONSUMED DURING STEAM-INJECTION OPERATION	\$ 2/1000 GALLONS
NON-TURBINE MAINTENANCE	\$ 60,000/YEAR
ADDITIONAL TECHNICAL SUPERVISION	\$ 40,000/YEAR
INSURANCE	\$ 37,500/YEAR
PLANT LIFE IS ESTIMATED TO BE 20 YEARS.	

The initial cost incurred to replace the existing stand-alone boiler is the estimated installed capital cost of \$5 million for the STIG system. (Here it is assumed that the construction period is sufficiently short that interest charges accumulated during construction can be neglected.) Operating the STIG system requires some additional expenditures -- maintenance, fuel, water, technical supervision, and insurance (see Table 2). But savings accrue from no longer having to purchase electricity and from being able to sell electricity to the grid at the utility's avoided cost.

The calculation is complicated by the fact that purchased electricity prices and avoided costs vary with the time of day and season of the year. For example, in the Pacific Gas & Electric (PG&E) territory in California, the purchased-electricity price structure includes peak, mid-peak, and off-peak rates, as does the avoided cost structure. In addition, there is a capacity charge applied to the peak power level reached each month by a purchaser of electricity and a capacity payment to a cogenerator if it can guarantee continuous delivery of firm power to the utility.[35]

At PG&E's prices for gas -- \$4.70/GJ (\$4.96/MBTU) (LHV) -- and electricity (\$.106/KWh peak) and avoided cost paid for cogenerated

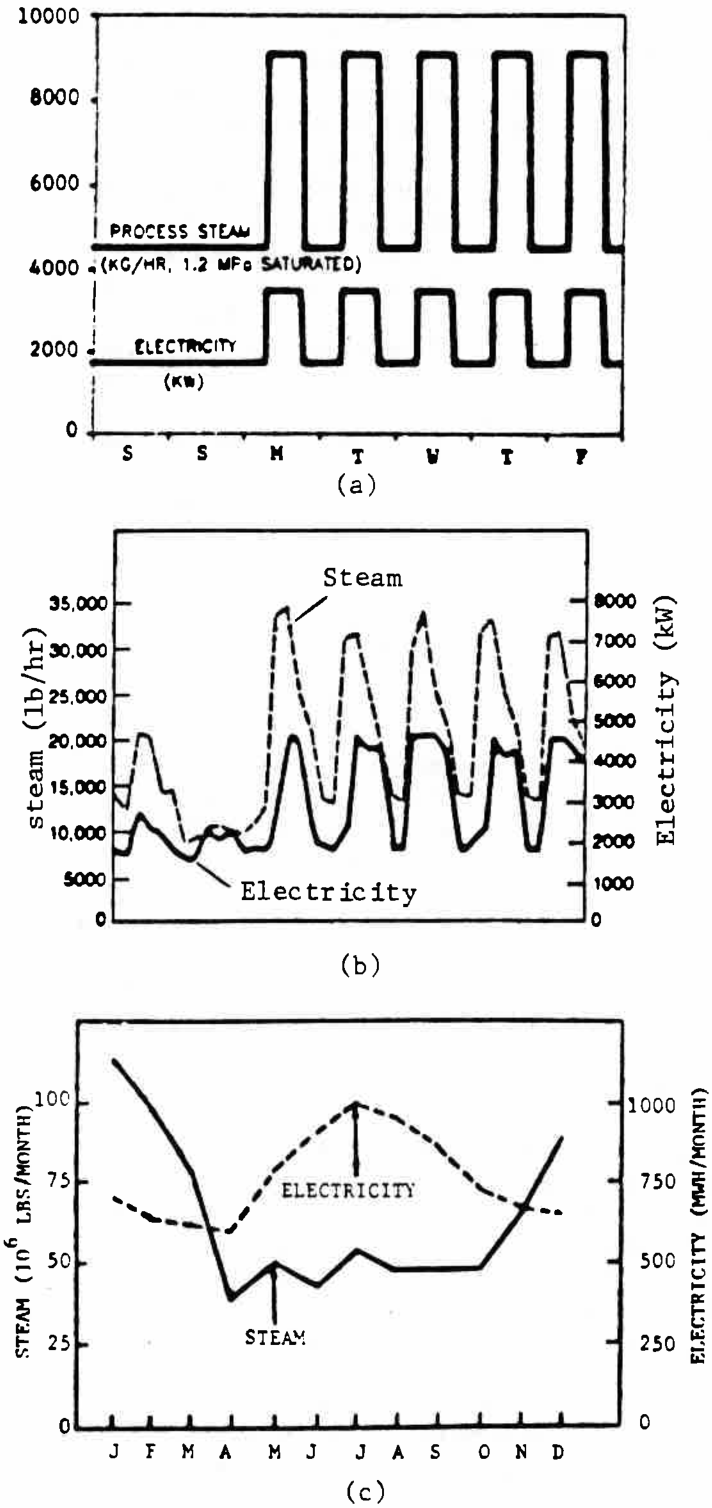


Fig. 12. Plant steam and electricity (a) idealized weekly load used in the analysis, (b) typical weekly load,[12] (c) example seasonal load profile.[36]

electricity ($0.086 peak),[35] the inflation-corrected internal rate of return on an investment in a STIG cogeneration system is about 21%/year, a respectable real rate of return. However, the rate of return for a conventional gas-turbine system with 3500 kW of electrical capacity (at a capital cost of $3.6 million) is about the same. Thus, for the rate structure and load profile we assumed, there appears to be no great incentive for a plant owner to invest in a STIG system.

However, the steam and electricity loads we assumed were very predictable (Fig. 12a), and the size of the cogeneration systems were such that they were operating near their maximum capacity all the time, a rather idealized situation. More typical load profiles are shown in Fig. 12b and 12c, but even these do not account for unexpected shut-downs or other load changes that may occur in the future, e.g., due to the installation of energy conserving equipment.

With unexpected load drops or plant shut-downs, the internal rate of return on an investment in a conventional system can drop substantially, but in a STIG system, since process steam can be redirected to produce additional electricty for sale to the grid, the rate of return is unaffected (Fig. 13). Thus, investing in a STIG system instead of a conventional one eliminates the financial risk associated with unpredicted load changes or plant shut-downs.

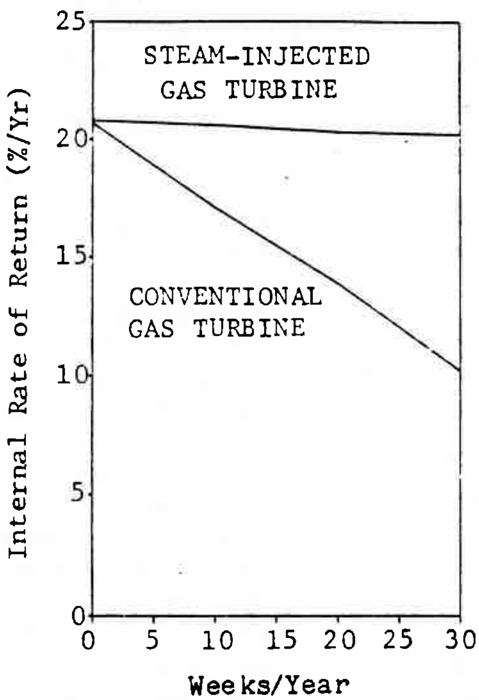


Fig. 13. Internal rates of return on investments in STIG or in conventional gas turbine cogeneration as a function of plant idle time.

Even under a "no surprise" scenario, the STIG system will annually produce about 25% more electricity and save nearly twice as much fuel per unit of process steam generated as the conventional system. As down-time rises, both the electricity produced and the fuel savings rate increase still further. For a 20-week shut-down, the STIG system will produce over 40% more electricity and save over 8 times as much fuel per unit of process steam.

Thus the user's benefit from STIG -- reduced financial risk -- is complemented by the societal benefits of a greater power generating and greater fuel savings potential than what is offered by the simple-cycle gas-turbine system.

FUTURE PROSPECTS FOR STEAM-INJECTED GAS TURBINES

Larger steam injected gas turbine systems, operating at higher pressure ratios and turbine inlet temperatures, are under active development. Based on modelling and some preliminary tests, the General Electric Company (GE) indicates that with only minor modifications, the electrical output of their LM-5000 turbine could be raised from 33 MW in the conventional mode to 50 MW in the steam-injection mode, while the efficiency in generating electricity alone would rise from 37% (LHV) to 44% (LHV).[8] With another 2-3 years of developmental work, GE indicates that 108 MW could be produced at an efficiency of 55% by a system using intercooling with steam injection.[37] A back-of-the-envelope calculation indicates that this level of performance is thermodynamically attainable.

The Simpson Paper Company in Anderson, California currently operates an LM-5000 turbine in a conventional cogeneration configuration. Before the end of the year, they will be making the minor modifications necessary to inject steam, and thus produce electricity at up to 44% efficiency (LHV).[38]

Steam-injected gas-turbines operating on natural gas may soon come to compete successfully with large coal-fired power plants for utility base-load generating capacity in many regions of the country. The favorable economics for central station STIG plants arise from very low capital costs [installed capital costs (in 1984$) as low as $500/kW,[5] compared to $1200-1500/kW for large coal-fired facilities[39]], and relatively low fuel costs [because of efficiencies as high as 55% and expectations that natural gas prices will remain essentially constant for the next 10-15 years in many regions of the US[7]]. Finally, much shorter construction lead times and the capability to make incremental additions (~100 MW) to the generating base with STIG systems would help utilities avoid the inherent risk-taking involved in embarking on expensive 7-10 year construction projects, which may eventually not be needed due to slower-than-expected growth in demand for electricity during the construction period.

Utilities might also utilize STIG cogeneration systems as "electrical peaking systems." In the non-supplementary-fired mode, these systems could provide steam for process needs and base-load electricity, while supplementary firing could be used to provide extra steam for injection to meet peak electrical demands.

PG&E is pursuing still another STIG application. According to a PG&E plan under active discussion,[38] aging boilers which produce steam for heating about 100 commercial buildings in San Francisco will be replaced by a single STIG system based on the LM-5000 turbine, allowing PG&E to increase its base-load generating capacity during the warmer months of the year when steam demands drop and electricity demands rise.

PROBLEMS

1. For a simple-cycle gas turbine cogeneration system with a fuel chargeable to power of 1.6 kJ/kJ and an electricity to steam ratio of 0.70, state any assumptions made, and determine: (a) the fraction of fuel converted to steam; (b) the fraction of fuel converted to electricity; (c) the overall first-law efficiency; (d) the second-law efficiency and; (e) the fuel-savings rate.

2. For a simple-cycle gas turbine system with the following characteristics: compressor inlet temperature, pressure ratio, air flow, and efficiency of 15°C, 9.3, 15 kg/s, 0.84 respectively; turbine efficiency and inlet temperature of 0.90 and 982°C, respectively, calculate the net shaft-power output and generating efficiency. Do not assume that specfic heats are constant, but state any other assumptions made.

3. Assuming specific heats are constant and the pressure drop between the compressor exit and the turbine inlet is negligible, determine for a fixed turbine inlet temperature, an expression for the compressor pressure ratio that will: (a) maximize work output per unit mass and; (b) maximize shaft-power generating efficiency.

4. For a gas-turbine cogeneration system with an air flow of 15 kg/s and a steam generation rate of 2 kg/s, show that the pinch point in the HRSG must occur at the entrance to the boiler.

5. Assuming that a HRSG (operating with a 10°C pinch point) is attached to the gas turbine in problem 2, calculate the fraction of the fuel used in the turbine system that is converted to saturated steam at 1 MPa (gauge). (Assume reasonable, constant values for specific heats).

6. Using the simple-cycle cogeneration system described in problems 2 and 6 as a reference case, and referring to the cycle states numbered in Fig. 7, develop a figure analogous to Fig. 8 using: (a) a regenerator effectiveness [defined as $(T_5 - T_4)/(T_9 - T_4)$] of 0.8; (b) 2-stage compression with intercooling, with pressure drops across each stage that will minimize total compressor work, assuming that specific heats are constant and that $T_3=T_1$ and $T_4=T_2$ and; (c) 2-stage expansion with reheating, with expansions across each stage that will maximize total turbine work, assuming that specific heats are constant and that $T_7=T_5$ and $T_8=T_6$.

7. Develop "back-of-the-envelope" equations similar to Eqns. 24, 25, 27, 31, and 33 for a steam-injected gas turbine, assuming a ratio of injected-steam-to-air of 0.05. For basic turbine characteristics, use the cycle described in problem 2.

8. Based on the costs given in Table 2 for a steam-injected gas-turbine cogeneration system, determine the expected internal rate of return to a fruit processing facility which invests in a steam-injected gas-turbine cogeneration system as a replacement for its 83% efficient, natural-gas [at $5.00/MBTU (LHV)] boiler and the purchase of electricity. Assume that for three months of the year, the facility continuously requires 9000 kg/hr of steam and 3500 kW of electricity, but requires no steam or electricity for the other 9 months of the year. Assume the plant does not use supplemental firing, but operates at full steam injection during the 9-month period to produce electricity for sale to the grid. Also assume the plant pays $0.10, $0.08, and $0.06/kWh during peak, mid-peak, and off-peak periods (all weekend, noon-6p.m. M-F, and all other times, respectively). And assume corresponding avoided costs are $0.08, $0.07, and $0.06/kWh.

What would the rate of return be for an investment in the conventional ($3.6 million) gas-turbine cogeneration system, in which the exhaust gases bypass the HRSG when the need for process steam drops?

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6. To reduce process-steam production in conventional gas-turbine cogeneration systems, the fuel flow to the turbine can be throttled, resulting in lower turbine inlet and outlet temperatures, or the hot turbine exhaust can be bypassed around the heat recovery system. These are the only two alternatives in nearly all gas-turbine systems used to generate electricity, since most incorporate only a single shaft for the compressor, turbine and generator. Since the generator speed must be maintained constant (and turbo-machinery is designed to move a constant volume of air at a fixed speed), the mass flow through the system cannot be throttled.
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