

INDUSTRIAL COGENERATION SYSTEM APPLICATION CONSIDERATIONS

by

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Mr. Kovacik is a 1960 graduate of Pennsylvania State University, with a B.S. degree in Mechanical Engineering, a registered professional engineer in the State of New York, and a member of ASME and the Technical Association of the Pulp and Paper Industry. He has authored many papers on the techno-economic aspects of industrial power generation, and he has been awarded three patents in the field of steam turbine and heat recovery steam generator design. He has received awards from several technical societies in appreciation for his efforts in advancing the technology in energy-related fields.

ABSTRACT

This paper reviews many of the factors that require evaluation in the development of steam turbine and gas turbine cogeneration systems. The impact of federal energy legislation on the gas turbine cogeneration alternatives that merit consideration is also discussed.

INTRODUCTION

Cogeneration is a practice that has been used by many industries during this century as a reliable, economical means of generating power in conjunction with satisfying their process heating needs. Prior to the 1960's, most cogeneration systems were based on use of steam turbine-generators, in both nonextraction and automatic-extraction designs. Most applications were associated with the pulp and paper, textile, chemical and foods industries.

During the 1960's, industries recognized the economic benefits that could be realized through use of gas turbine cogeneration systems. As a result, where suitable fuels were economically available, gas turbine cogeneration applications flourished. This prime mover has maintained a favorable reputation as a critical element contributing to effective cogeneration systems. Furthermore, recent legislation has stimulated interest and provided additional potential for gas turbine cogeneration systems in the years ahead.

This paper will review application considerations for both steam turbine and gas turbine cogeneration systems. Furthermore, the impact of the Public Utilities Regulatory Policies Act

(PURPA) on the development of alternatives that merit consideration will be briefly discussed.

COGENERATION

Cogeneration has often been defined as the sequential production of useful thermal energy and shaft power from a single energy source. The shaft power can be used to drive either mechanical equipment, such as pumps and compressors, or electric generators. Power produced in the manner described is called a "topping" cogeneration cycle. (A "bottoming" cogeneration system produces power resulting from low level energy recovery associated with the process).

The more effective use of energy in topping cogeneration systems is illustrated in Figure 1. The power generation case is typical of what can be expected from a modern coal fired power generating plant. The 35% energy utilization is equivalent to a 9750 BTU/kWh HHV heat rate for this coal fired system.

The "cogeneration" arrow in Figure 1 illustrates the higher energy utilization effectiveness realized when heat and power are delivered from a single system. In a topping cogeneration system, the process becomes the heat sink (or condenser) for the power generation system. Thus, the cogeneration system can often deliver 80% or more of the fuel input energy as useful output (heat and power) for plant use.

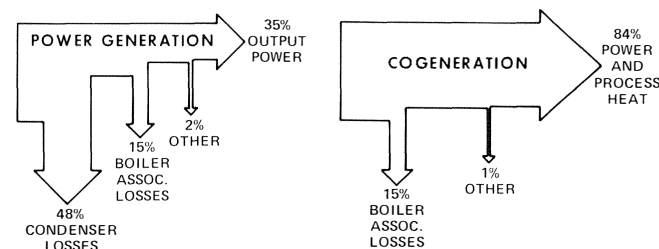


Figure 1. Fuel Utilization Effectiveness.

STEAM TURBINE CYCLES

Steam turbine cycles are frequently applied in those industrial plants having process by-products available as the cogeneration system fuel. These by-product fuels include black liquor and hog fuel in the pulp and paper industry, wastes associated with the food industry, blast furnace and coke oven gas from steel mill operations, and others. Also, in those applications where coal or residual fuel oil are the economical plant fuels, steam turbine cycles are usually preferred relative to other topping cogeneration cycle options.

Development of economical steam turbine cogeneration systems requires careful evaluation of the following aspects of the system design:

- Prime mover size
- Initial steam conditions

- Feedwater heating
- Condensing power

Prime Mover Size

The application of a large, efficient steam turbine-generator rather than a group of smaller, less efficient mechanical drive steam turbines as drivers for small power plant loads can significantly improve energy utilization. The example given in Table 1 shows that 70% more power can be generated by expanding the steam required for process in the larger turbine-generator rather than by expanding the smaller mechanical drive units. Considering the losses associated with the transformation and distribution of energy from the turbine-generator to the motors for the small plant auxiliaries, the net gain is approximately 57%, rather than the 70% implied by the data in Table 1.

Table 1. Influence of Prime Mover Size on Cogenerated Power.

Type Prime Mover	Approximate Efficiency	Cogenerated Power Per 100 Million BTU/Hr Net Heat to Process
500 hp Single Stage Mechanical Drive Units	45%	2800 kW eq.
5000 kW Multi Valve Multi Stage Turbine-Generator	72%	4770 kW

Basis:

- 1) Initial steam conditions, 600 psig, 750°F.
- 2) Process steam required at 50 psig.
- 3) Process returns at 180°F.
- 4) No feedwater heating has been included.

For grassroots facilities, major industrial power plant expansions or modernization programs, economics usually favor expansion of steam in a steam turbine-generator, and use of motors rather than turbines to drive small mechanical loads. However, replacement of existing noncondensing turbines driving miscellaneous loads is more difficult to justify economically, since the capital burden of the displaced capacity must be justified by the annual savings due to the effective increase in cogenerated power. For example, based on Table 1 conditions, the incremental capital cost (installed) of the 5000 kW turbine-generator, the replacement motors, and the associated electrical equipment would have to be justified based on 1600 kW incremental increase in power generated.

Initial Steam Conditions

The increased thermodynamic availability of steam at various initial steam conditions and exhaust pressures relative to 600 psig, 750°F steam is illustrated in Figure 2. The data presented are based on initial steam temperatures that will provide essentially the same expansion line at a 75% turbine efficiency, regardless of the initial pressure selected. The data illustrate that the magnitude of the gain in cogenerated power that can be realized is a function of the initial steam conditions selected, as well as of the pressure level or levels required in process.

The effect of turbine inlet steam conditions on the amount of power that can be cogenerated per 100 million BTU/hr net heat to process (NHP)* at different steam pressures is shown in

600 PSIG 750 F AS "BASE"

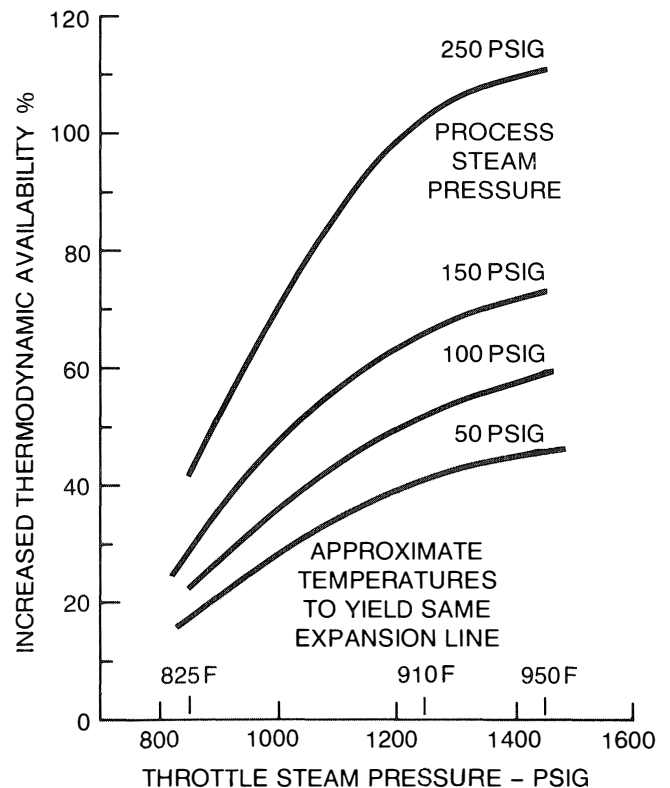


Figure 2. Increased Thermodynamic Availability for Various Throttle Steam Conditions.

Figure 3. The data presented include the benefits of regenerative feedwater heating to the feedwater temperatures noted in the figure. The increase in cogenerated power through use of higher initial steam conditions is readily apparent. Figure 3 also illustrates the output gains that are forfeited if plants are designed with excessive margin in the process steam distribution systems.

Studies have shown that the higher steam conditions can be economically justified more easily in industrial plants having relatively large process steam demands. Data given in Figure 4 provide guidance with regard to the initial steam conditions that are normally considered for industrial cogeneration applications. Higher energy costs experienced since the mid-1970's are favoring the upper portion of the bands shown in Figure 4.

Feedwater Heating

Feedwater heating provides a means of increasing plant steam requirements and thus the amount of power that can be cogenerated in a steam turbine cycle. The simple example given in Figure 5 shows that the addition of the closed heater at 225 psig increases the steam demand by about 10% and the amount of power that can be cogenerated by 8.3%.

Since many industrial plants have several process pressure levels, the individual process pressures may be logical locations for feedwater heaters. Generally speaking, the use of three stages of feedwater heating is economical for most industrial applications using steam turbine cogeneration cycles.

*Net heat to process is the net energy delivered to process without consideration of the boiler inefficiency.

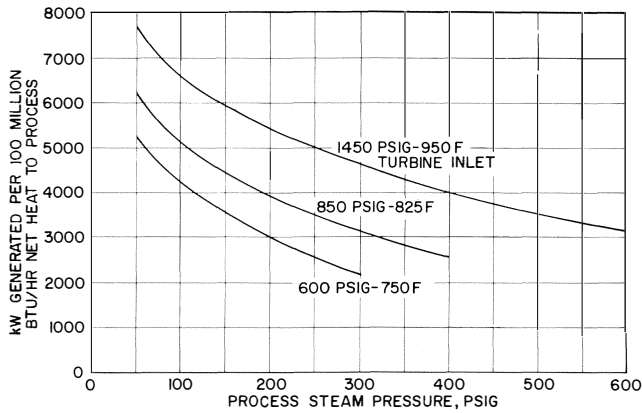


Figure 3. Effect of Inlet Steam Conditions and Process Steam Pressure on Power Cogenerated with Steam Turbines.

Basis:

- 1) 70% of process steam flow returned as 200°F condensate and balance as 80°F makeup.
- 2) Power cycle credited for feedwater heating to 445°F for 1450 psig unit, 400°F for 850 psig, 370°F for 600 psig.
- 3) Turbine efficiency 75%.

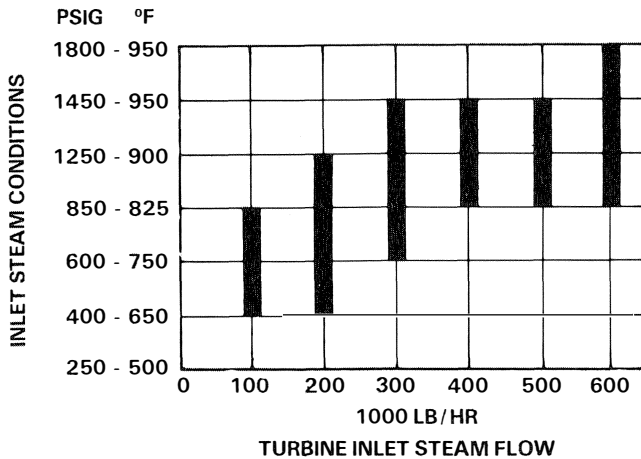


Figure 4. Range of Initial Steam Conditions Normally Selected for Industrial Steam Turbines.

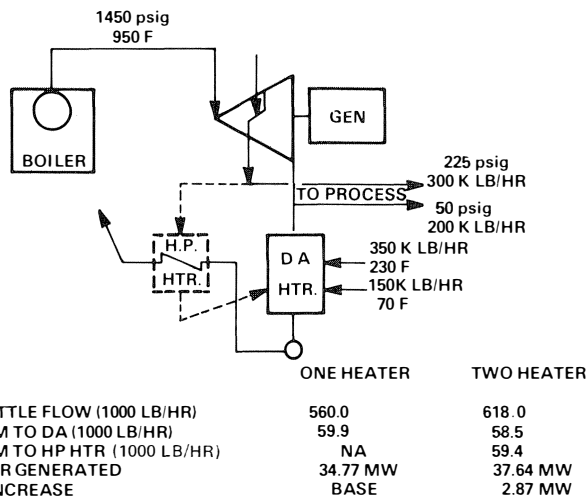


Figure 5. Effect of Added Feedwater Heating on Cogenerated Steam Turbine Power.

Condensing Power Generation

The amount of power that can be cogenerated in steam turbine cycles is a function of the initial steam conditions, the process pressure level(s), and the feedwater heating cycle selected. Also, the data in Figure 3 show that, even for the most effectively developed systems, it is not likely that the amount of power generated per unit of heat to process will exceed 80 kW per million BTU NHP. This is usually less power than that required to satisfy most industrial plant electrical energy needs. Thus, condensing power is frequently considered to augment steam turbine cogenerated power.

The impact of adding condensing power generation to a "pure" steam turbine cogeneration system (noncondensing steam turbine power) is illustrated in Figure 6. Even though condensing power is not necessarily energy efficient, it can occasionally prove economical. Favorable economics can frequently be realized for the following conditions:

- Use of condensing power for utility demand control
- Low cost fuels such as wood, coal, or excess process by-products are available as plant fuels
- Thermal energy from process is available for expansion to a condenser (bottoming cogenerated power)
- Reliability of other power sources is questionable

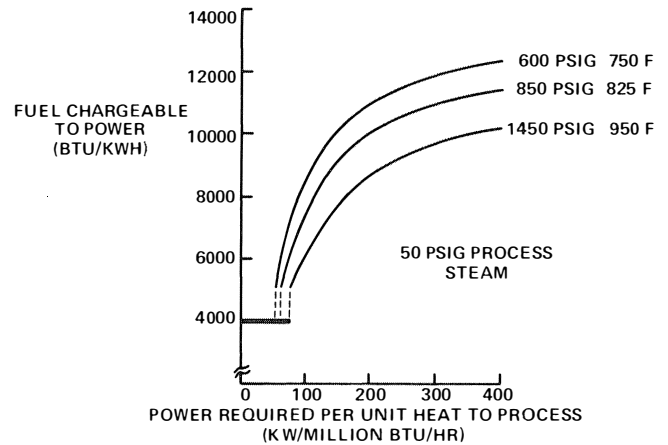


Figure 6. Fuel Chargeable to Power—Various Steam Turbine Cycles.

Basis:

- 1) Process steam at 50 psig.
- 2) Cycle specifics as in Figure 3.
- 3) Fuel chargeable to power (FCP) is the incremental increase in fuel consumption due to the cogeneration system, divided by the net power generation credited to the cycle.
- 4) FCP based on credit for process heat at 84% boiler efficiency.

GAS TURBINES AND COMBINED CYCLES

Gas turbine cycles provide the opportunity to generate a larger power output per unit of heat required in process relative to noncondensing steam turbine cogeneration systems. This characteristic, combined with a favorable fuel chargeable to power (FCP)** and proven reliability, is why this prime mover has had wide acceptance in industrial plants where suitable fuels are economically available.

**Fuel chargeable to power is the incremental increase in fuel consumption due to cogeneration, divided by the net power generation credited to the cycle.

The economics of gas turbines in process plant applications usually depend on effective use of the exhaust energy. The most common use of this energy is for steam generation in heat recovery steam generators (HRSG's), in unfired as well as exhaust fired designs. However, the gas turbine exhaust can also be used as a source of energy for unfired and fired process fluid heaters, or for preheated combustion air for power boilers.

Cycle Configurations

The various cycle options for gas turbines with HRSG's are illustrated in Figure 7. Configuration "A" has the HRSG generating steam at the appropriate steam conditions for process use and is the most simple configuration available.

The HRSG in Figure 7B generates steam at elevated steam conditions, so both steam turbine and gas turbine power can be cogenerated. This configuration yields the highest power to heat ratio for any configuration presented in Figure 7.

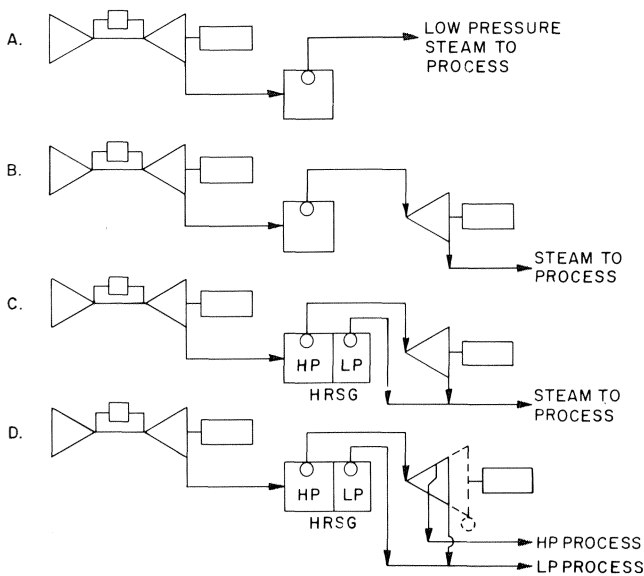


Figure 7. Possible Plant Energy Systems—Gas Turbines with Heat Recovery Steam Generators.

An arrangement commonly applied as a result of the rapid increase in fuel costs experienced since 1973 is shown in Figure 7C. This multiple pressure HRSG is usually applicable when the gas temperature entering the HRSG surface is about 1200°F or lower. Multiple pressure HRSG's provide increased recovery of the gas turbine exhaust energy relative to the Figure 7B configuration, and thus contribute to the favorable FCP associated with these cycles. FCP improvements in the 10 to 15% range are typical for these two-pressure level systems.

Figure 7D, an arrangement that includes a condensing section on the steam turbine-generator, is an extension of the Figure 7C configuration. The condensing section on the steam turbine provides cycle flexibility, permitting utilization of HRSG steam production in excess of process steam demands, but at an increased cycle FCP.

Heat Recovery Steam Generators

Developers of gas turbine cogeneration systems may have several HRSG options. These include the following:

- Unfired HRSG's
- Supplementary fired HRSG's

- Fully fired HRSG's

Unfired HRSG—An unfired HRSG is an extended surface convective heat exchanger designed to recover a portion of the sensible heat in the gas turbine exhaust. These units can produce steam at low steam conditions, such as 150 psig saturated, for direct use in process. Alternatively, steam can be generated at elevated steam conditions for expansion in a steam turbine prior to delivery to the process. Some gas turbine units have exhaust temperatures of 1000°F or somewhat higher, which are temperatures adequate for generation of steam for use at 1500 psig and 925°F.

The unfired HRSG steam production is a function of the exhaust flow and temperature entering the unit. Thus, the unit is a slave of the gas turbine and cannot be controlled.

Supplementary Fired HRSG—In a supplementary fired application, an auxiliary burner is used to increase the turbine exhaust temperature to as high as 1700°F. These units are essentially convective heat exchangers whose construction is similar to those of unfired designs. The primary difference relative to unfired HRSG's is in the heat transfer section immediately downstream of the burner, where bare tubes and/or tubes with reduced fin pitch and height shield the unit from the radiant energy associated with the burner. The auxiliary burner permits modulating the steam production capability of the HRSG essentially independent of the gas turbine operating mode.

Fully Fired HRSG—A fully fired type HRSG is similar in appearance to a power boiler. The design usually admits to its combustion system only the amount of turbine exhaust gas required to generate the desired amount of steam. The balance of the exhaust flow is bypassed and rejoins the gases used for combustion ahead of the heat recovery section.

The maximum amount of steam that can be generated in a fully fired HRSG is usually six to seven times that available from an unfired HRSG. Also, these cycles provide the lowest (best) FCP. Even so, fully fired HRSG's have not been widely used in industrial applications.

Estimating Steam Production—Unfired and Supplementary Fired Units—A simplified diagram illustrating the temperature relationships governing unfired HRSG designs is shown in Figure 8. The temperature difference (T_2-T_3) is frequently referred to as the "pinch point" and is governed by the effectiveness and the degree of subcooling designed into the economizing section. Figure 8 also shows the importance of using a low feedwater heating temperature in gas turbine HRSG systems, in contrast to steam turbine cogeneration systems, where a larger amount of feedwater heating is usually desirable.

The data presented in Figure 8 also illustrate the lower energy recovery if an HRSG is designed to provide the higher steam conditions required to support combined cycles. However, in these instances, opportunities for multiple levels of energy recovery, as shown in Figure 9, can provide the added benefits of steam turbine cogeneration and a reasonable stack temperature.

The amount of steam that can be generated in single pressure unfired or supplementary fired HRSG's can be estimated using the following relationship:

$$W_{stm} = \frac{W_{exh}}{10^6} \times F_1 \times F_2$$

where

W_{stm} = the steam generated

W_{exh} = the gas turbine exhaust flow (lb/hr)

F_1 = the saturated steam production, based on the steam pressure desired and the gas temperature

entering the heat transfer surface (see Figure 10)

F_2 = a factor that adjusts the HRSG production to the desired steam temperature (see Figure 11)

For units fired to average exhaust gas temperatures of 1700°F or less, the HRSG fuel requirements can be estimated using Figure 12.

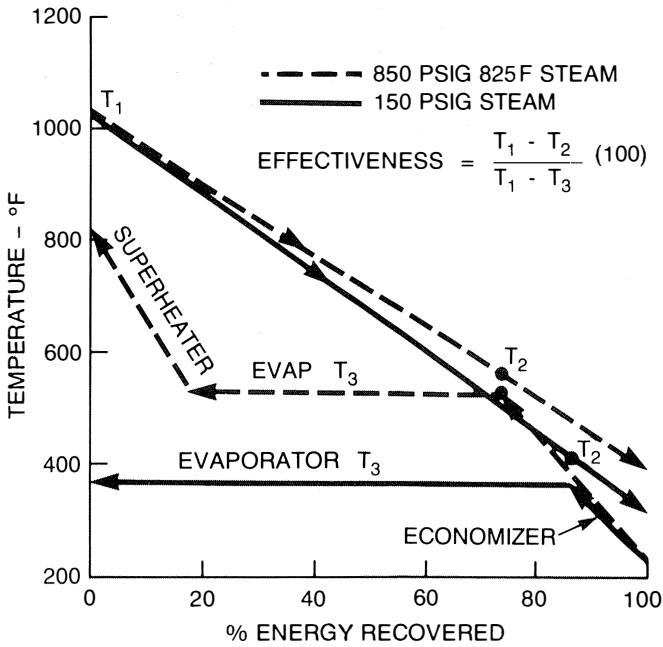


Figure 8. Temperature Relationships—Unfired Heat Recovery Boiler.

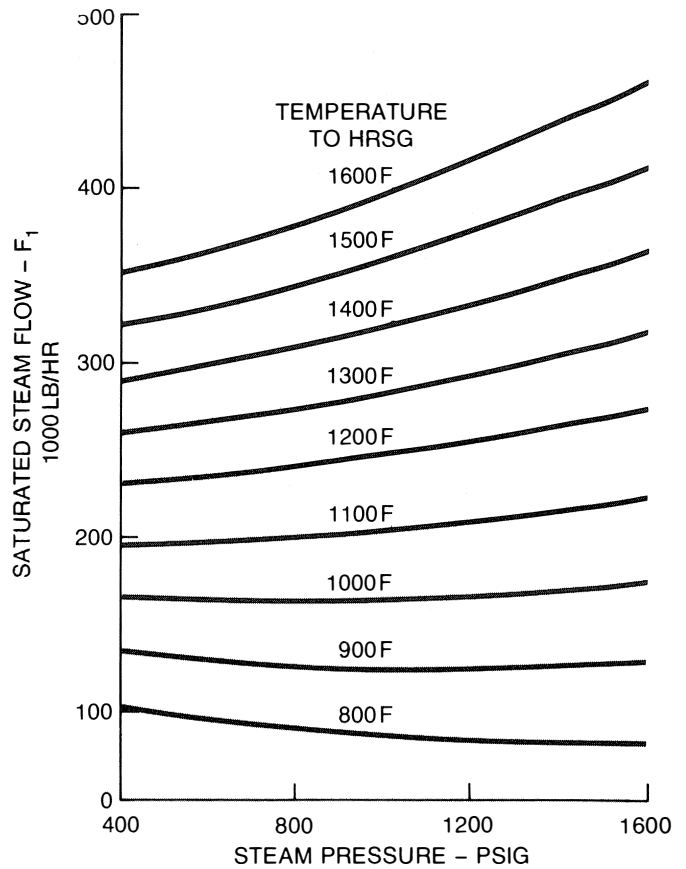


Figure 10. Saturated Steam Production in Heat Recovery Steam Generators.

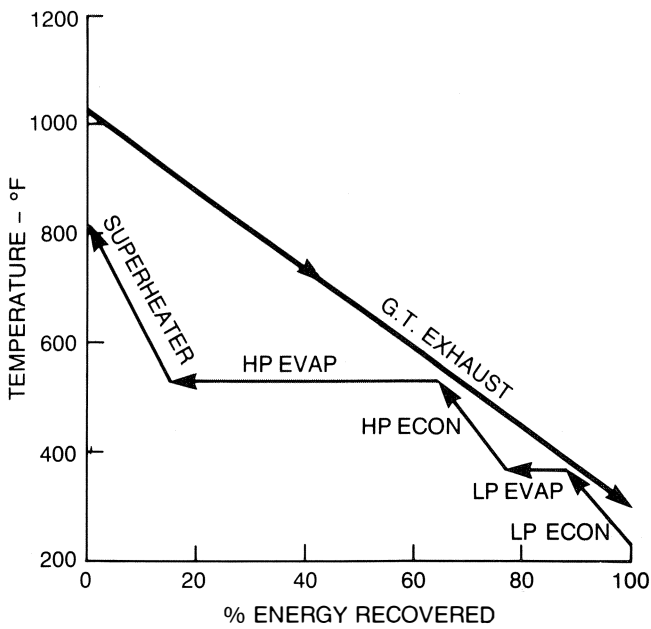


Figure 9. Temperature Relationships—Two-Pressure Level, Unfired Heat Recovery Boiler.

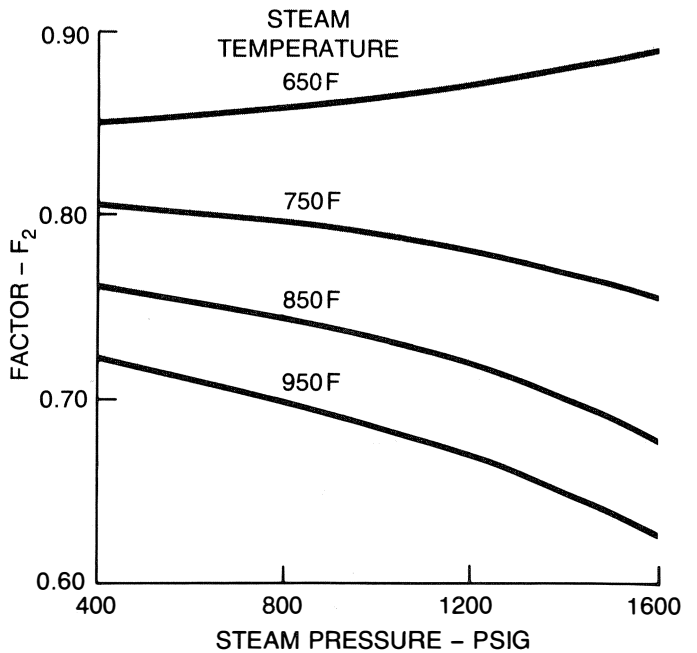


Figure 11. Superheat Adjustment Factor.

Cycle Design Flexibility

One method of displaying the many options available using a gas turbine in a cogeneration application is shown in Figure 13. This diagram has been developed for the General Electric Company MS7001E gas turbine-generator (78,400 kW ISO, natural gas fired). A summary of the performance used to develop the performance envelope given in Figure 13 is presented in Table 2.

Point A represents the MS7001E gas turbine-generator exhausting into an unfired, low pressure HRSG. Point C is a combined cycle configuration, based on the use of a two-pressure level, unfired HRSG. The steam turbine in the C cycle is a noncondensing unit, expanding the HP HRSG steam to the 150 psig process steam header.

Points B and D in Figure 13 represent operation of the HRSG with supplementary firing to a 1400°F average exhaust gas temperature entering the heat transfer surface. The temperature used for the HRSG firing in Figure 13 has been arbitrarily limited to 1400°F, even though considerably higher firing temperatures, and thus steam production rates, are possible in the exhaust of this unit.

The "envelope" defined by points A, B, C, and D in Figure 13 represents the most effective use of a gas turbine in a cogeneration application. Operation along line CE or DF, or at any intermediate point to the left of line CD represents the use of condensing steam turbine power generation, with line EF applicable for combined cycle operation without any heat supplied to process. Thus, the cycles along line EF are combined cycles providing power alone.

The per unit costs of power generation for the cycles A through F illustrated in Figure 13 are presented in Figure 14.

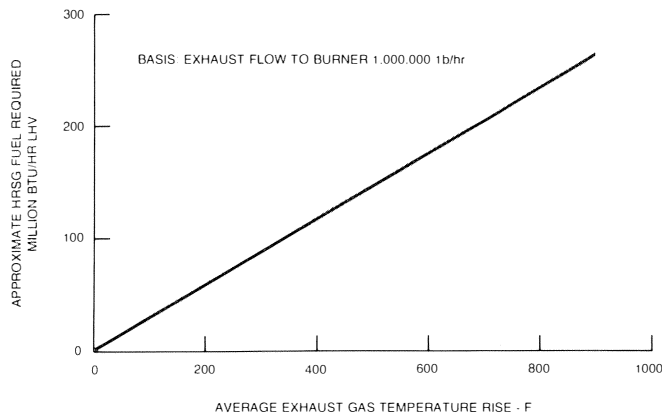


Figure 12. Estimated HRSG Fuel Requirements.

Table 2. Performance of Gas Turbine Cogeneration Cycles—MS7001E.

Cycle	A	B	C	D	E	F
Net Output (MW)	70.4	70.1	84.2	96.4	103.1	124.4
NHP (M BTU/hr)	382	627	346	521	0	0
Net Fuel (M BTU/hr HHV)	396	370	439	496	851	1117
FCP (BTU/kWh HHV)	5620	5280	5210	5150	8250	8980
Power Per Unit Heat to Process (kW/M BTU/hr)	184	112	243	185	NA	NA

Basis:

- 1) Cycle definitions as given in Figure 13.
- 2) Net output is the total power credited to the cogeneration cycle.
- 3) Net fuel includes credit for the net heat to process (NHP) at an 84% process boiler efficiency.

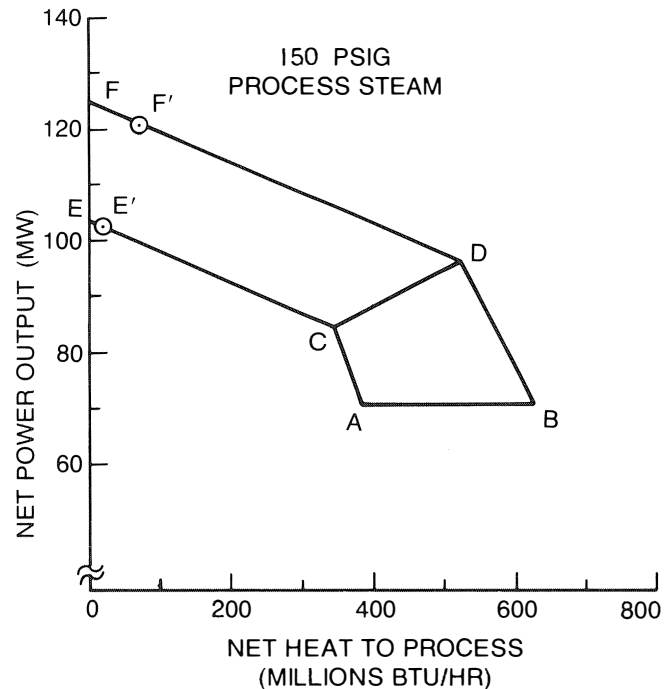


Figure 13. Performance Envelope for Various Gas Turbine Cogeneration Systems—MS7001E Gas Turbine-Generator.

Basis:

- 1) Sea level site, 80°F ambient, natural gas fuel.
- 2) Cycle A—Unfired low pressure HRSG
Cycle B—Supplementary fired (1400°F) HRSG, low pressure process steam
Cycle C—Combined cycle, unfired, 2-pressure level HRSG, HP at 1450 psig, 950°F, LP at 150 psig sat., noncondensing steam turbine-generator
Cycle D—Combined cycle, supplementary fired HRSG, steam at 1450 psig, 950°F, noncondensing steam turbine-generator
Cycle E—Same as Cycle C, but with extraction/admission condensing steam turbine-generator
Cycle F—Same as Cycle D, except with straight condensing steam turbine-generator
- 3) Process returns and makeup enter the 5 psig deaerating heater at a mixed temperature of 180°F.
- 4) Cycles E' and F' represent minimum extraction for process at 150 psig to meet requirement of a qualifying cogeneration facility under PURPA.

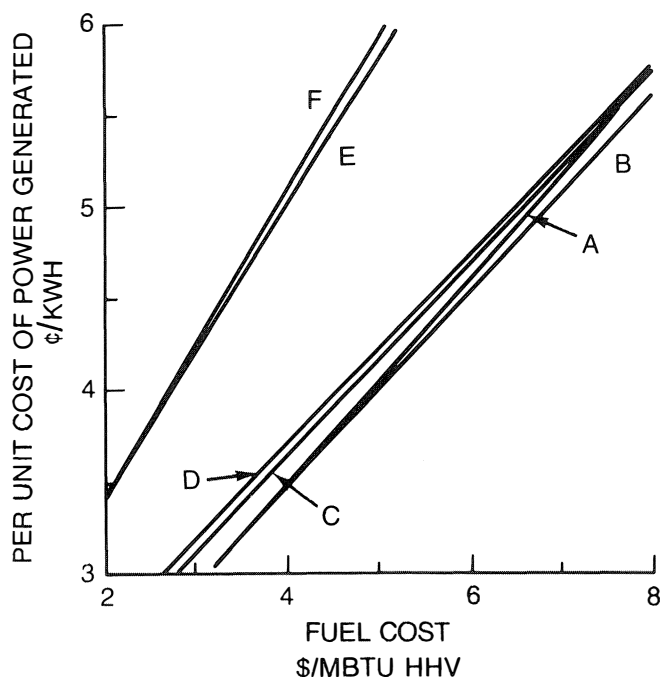


Figure 14. Per Unit Cost of Power Generation MS7001E Cogeneration Systems, 25% Discounted Rate of Return.

Basis:

- 1) Add-on to an existing facility.
- 2) Cycles as defined in Figure 13.
- 3) Operation 8400 hr/yr.

These per unit power generation costs represent the value required to generate a 25% discounted rate of return (DROR) on the investment for the cycle presented. Stated another way, applications where power has a value equal to or greater than that shown in Figure 14 will yield a DROR of 25% or more for the specific cycle configuration being examined.

The "pure" cogeneration cycles (cycles A through D), cycles without any condensing steam turbine power generation, have a much lower per unit cost threshold than the combined cycles designed to produce only power (cycles E and F). Even so, site-specific fuel and power cost considerations may dictate cycles with considerable condensing power as the appropriate economic choice.

THE PUBLIC UTILITY REGULATORY POLICIES ACT

The legislation having the greatest impact on the development of cogeneration cycles is the Public Utility Regulatory Policies Act (PURPA). The broad objective of PURPA is to encourage conservation and the effective use of energy resources. Furthermore, PURPA includes provisions that remove disincentives to cogeneration that have evolved since the beginning of this century. These provisions state that the utility must meet the following requirements:

- Buy power from qualifying cogeneration facilities at a rate related to the utility's avoided cost of power generation
- Provide backup power to the cogenerator at nondiscriminatory rates
- Exempt qualifying cogenerators from state and federal regulations with regard to public utilities

The avoided cost provision of PURPA, which had originally been challenged and later upheld by the Supreme Court, is the primary factor contributing to a number of recent applications. Furthermore, many of these applications have been developed to produce significantly more power than that required by the industrial facility.

In order to derive the economic benefits associated with the sale of cogenerated power, the energy supply system configurations being evaluated must satisfy basic criteria identified in this legislation. These criteria are thermal and efficiency standards summarized in Figure 15.

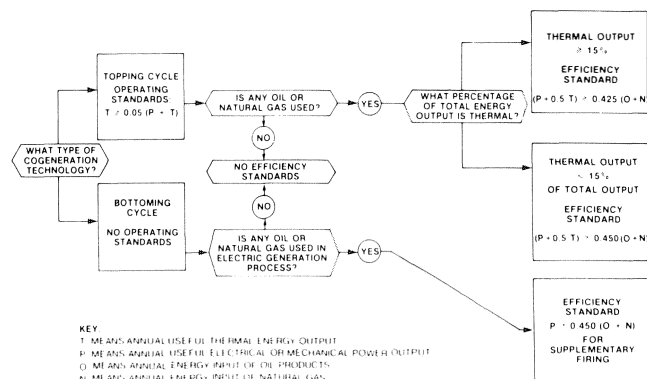


Figure 15. Operating and Efficiency Standards for New Cogeneration Facilities.

PURPA AND COGENERATION CYCLE DEVELOPMENT

The abilities of various cycles to satisfy the PURPA qualification criteria are shown in Figure 16 for steam turbine cycles, and in Figure 17 for gas turbine cycles using unfired HRSG units.

For the steam turbine cogeneration system (Figure 16), operation at 100% steam to process is based on a noncondensing steam turbine cycle expanding 1450 psig, 950°F steam to the process pressure levels noted. As the percent steam to process decreases, the cycle expands a portion of the boiler steam to a condenser, with the limiting condition of no steam to process being a straight condensing steam turbine cycle based on 1450 psig, 950°F initial steam conditions. In order to meet the qualification criteria of PURPA, this steam cycle would have to deliver at least 50% steam for a 50 psig process pressure level, or 57% steam if the plant steam demand was at 250 psig. If initial steam conditions were lower, even greater quantities of steam would have to be delivered to process to attain qualifying facility status.

The data presented in Figure 17 show the ease with which the gas turbine combined cycle configurations noted can meet the PURPA efficiency standard. Operation at 100% steam to process is equivalent to use of the "C" configuration in Figure 13. As the steam to process is decreased, more of the exhaust energy recovery from the gas turbine is used for condensing steam turbine power generation. At zero flow to process, the cycle operates similar to a "STAG" (stagnation) configuration, providing only electric power to the system (point E in Figure 13). Note that all configurations except the MS5001P meet the 45% PURPA efficiency, while supplying only 5% thermal energy, the minimum quantity allowed under PURPA, to the process. The MS5001P cycle meets the 42.5% PURPA efficiency criterion at a 15% thermal energy output to process. These data show that present legislation permits gas turbine cycles to be developed providing a small amount of steam to process, while still receiving qualifying facility status under PURPA.

because of their favorable thermal performance. (The minimum heat to process to comply with PURPA qualification criteria for the MS7001E system envelope given in Figure 13 is noted "E" for the unfired HRSG configuration and "F" for the supplementary fired HRSG system.)

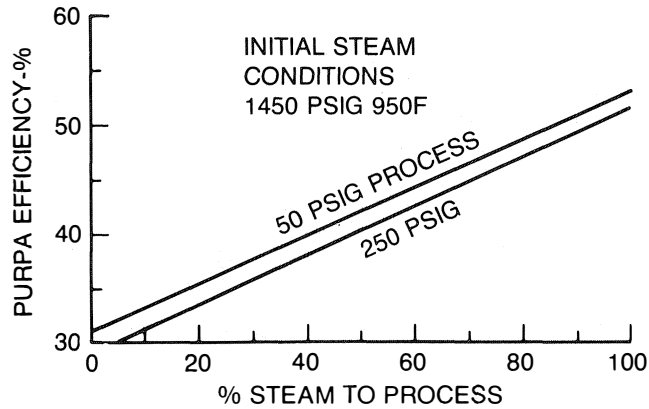


Figure 16. PURPA Efficiency at Various Process Steam Demands—Steam Turbine Cycles.

Basis:

- 1) Use of extraction condensing steam turbine-generator expanding steam to process pressures noted.
- 2) Condensing pressure is 2½ in. HgA.
- 3) Steam turbine-generator efficiency is 75%.
- 4) Feedwater heating to 445°F.
- 5) Process returns 100% of the steam delivered, 180°F.
- 6) $PURPA \eta = \frac{Power + \frac{1}{2} NHP}{Fuel(LHV \text{ Basis})} (100)$

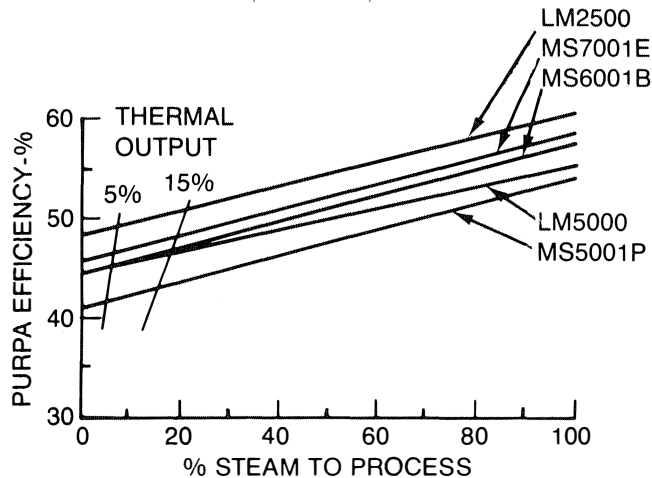


Figure 17. PURPA Efficiency at Various Process Steam Demands—Gas Turbine Combined Cycles.

Basis:

- 1) Basic gas turbine information given in Table 2.
- 2) Sea level site, operation at 80°F ambient capability, natural gas fuel.
- 3) Based on 2-pressure level unfired HRSG. HP HRSG is 850 psig, 825°F for all cases except the LM5000, which is developed at 600 psig, 750°F. LP HRSG feeds 150 psig steam header.
- 4) Process steam demand at 150 psig, 100% returns at 180°F.
- 5) Steam turbine efficiency is 75% for all cases.

6) Condenser pressure is 2½ in. HgA in applicable cases.

$$7) PURPA \eta = \frac{Power + \frac{1}{2} NHP}{Fuel(LHV \text{ Basis})} (100)$$

$$8) Thermal \ output = \frac{NHP}{NHP + Power} (100)$$

Examples

The favorable effect of the PURPA regulations on what utilities must pay for power from qualifying facilities can have a profound influence on the development of cogeneration facilities, even for applications having small process heating demands. For example, assume an industrial facility has a requirement for 45,000 lb/hr of 150 psig saturated steam. The data presented in Table 3 show that the largest cogeneration system, providing about 100 MW of electric power, would yield the highest DROR for the specific conditions given, except at \$8/M BTU HHV fuel with a 7¢/kWh credit for all power generated. The most efficient cycle, the LM2500, is economically preferred only for the high fuel cost case. However the DROR is less than 15%, and the project would probably not meet the minimum criteria for discretionary investments.

Table 3. Cogeneration Example—45,000 lb/hr Process Steam Demand.

System	LM2500	MS6001B	MS7001E
Net Output (MW)	24.1	47.2	100.6
Net Fuel (M BTU/hr HHV)	157.1	372.4	796.9
FCP (BTU/kWh HHV)	6520	7890	7920
Estimated Total Installed Cost (\$ Millions)	15.5	22.3	35.9
DROR (%)			
@\$4/M BTU HHV & 7¢/kWh	37.0	42.5	54.0
@\$6/M BTU HHV & 7¢/kWh	26.5	26.0	34.0
@\$8/M BTU HHV & 7¢/kWh	14.8	3.5	6.8

Basis:

- 1) Process steam demand at 150 psig sat.
- 2) Natural gas fuel, sea level site, 80°F ambient.
- 3) 1983 investment costs for adding the cogeneration system to an existing industrial boiler plant with low pressure boilers. Costs include electrical substation and transformation to 138 kV.
- 4) Maintenance costs are 2.5% of the estimated total installed cost. Operating labor is \$200,000 more than the labor cost without cogeneration.
- 5) Makeup water cost is \$2/1000 gal., based on 50% loss of steam delivered to process for combined cycles. Makeup for cooling tower system is 5¢/1000 gal.
- 6) Operation 8400 hr/yr.
- 7) DROR based on 10% investment tax credit, 5-year depreciation, straight line method, 20-year economic life, no salvage value, 3% total property taxes and insurance, 50% income tax rate. All comparisons are with case without cogeneration.
- 8) Net output includes credit for auxiliary power requirements displaced due to the addition of the cogeneration system.
- 9) Net fuel includes a credit for fuel required in 84% efficient process boilers providing steam to process at 150 psig sat. Credit is approximately 1.27 M BTU HHV/1000 lb process steam.

10) All cycles meet "qualification facility" requirements under PURPA.

11) Cycles	LM2500	MS6001B	MS7001E
HRSG (unfired)			
HP (psig/°F)	850/825	850/825	1450/950
LP (psig/°F)	150/sat	150/sat	150/sat
Steam turbine			
Extraction			
Condensing at 3 in. HgA	~5 MW	~14 MW	~30 MW

An example for a somewhat larger industrial facility is given in Table 4. In this instance, the process heat demand is an ideal match with the thermal energy available from the exhaust of the LM2500. Thus, it provides not only the best FCP, but also the most favorable economics for scenarios where the fuel cost is equal to \$5/M BTU HHV or higher when power has a value of 7¢/kWh.

If the plant power requirement for the Table 4 example was 60 MW, and plant management was not interested in generating "excess" power, the LM2500 cogeneration system would provide the most favorable economics. However, the MS6001B system would also be a potential candidate, and its DROR might be sufficiently attractive to qualify for management consideration if adequate funds were available for this larger cogeneration system.

Table 4. Cogeneration Example—85,000 lb/hr Process Steam Demand.

System	LM2500	MS6001B	MS7001E
Net Output (MW)	19.5	44.1	98.4
Net Fuel (M BTU/hr HHV)	109.5	301.6	749.3
FCP (BTU/kWh HHV)	5615	6840	7615
Estimated Total Installed Cost (\$ Millions)	9.4	21.0	35.4
DROR (%)			
@\$4/M BTU HHV & 7¢/kWh	52.5	46.2	55.0
@\$6/M BTU HHV & 7¢/kWh	42.0	32.8	35.5
@\$8/M BTU HHV & 7¢/kWh	31.0	16.5	11.8

Basis:

1) See items 1 through 10 of Table 3.

2) Cycles	LM2500	MS6001B	MS7001E
HRSG (unfired)			
HP (psig/°F)	NA	850/825	1450/950
LP (psig/°F)	150/sat	150/sat	150/sat
Steam Turbine			
Extraction			
Condensing at 3 in. HgA	NA	~11 MW	~28 MW

The third example, presented in Table 5, is based on an industrial heat demand that is a match for the MS6001B in a combined cycle configuration. Even so, economics favor the larger MS7001E at fuel costs to about \$6.15/M BTU HHV. Once fuel costs exceed this value, the MS6001B cogeneration cycle would be economically preferred.

All examples were based on adding a cogeneration system to an existing process plant. Thus, there was no "offsetting" process boiler plant investment that could be credited to the cogeneration plant investment, as is the case when cogeneration is considered in new or grassroots facilities. If the evaluations were based on new facilities, the investment credit for the process boiler plant would have the most favorable effect on the gas turbine alternatives most consistent with the plant heat demands. For example, for the Table 5 plant situation, the offsetting process boiler plant investment would reduce the incremental cogeneration investment for the LM2500, MS6001B and MS7001E alternatives to 68%, 75% and 86%, respectively, of the values given in Table 5. The magnitude of the reduction in investment would have a more favorable effect on the LM2500 and MS6001B alternatives than on the MS7001E cogeneration system.

Table 5. Cogeneration Example—180,000 lb/hr Process Steam Demand.

System	LM2500	MS6001B	MS7001E
Net Output (MW)	26.6	39.6	92.9
Net Fuel (M BTU/hr HHV)	129.3	208.1	632.6
FCP (BTU/kWh HHV)	4860	5255	6810
Estimated Total Installed Cost (\$ Millions)	15.4	19.3	33.9
DROR (%)			
@\$4/M BTU HHV & 7¢/kWh	43.0	50.0	59.0
@\$6/M BTU HHV & 7¢/kWh	35.5	40.7	41.6
@\$8/M BTU HHV & 7¢/kWh	27.3	30.0	22.3

Basis:

1) See items 1 through 10 of Table 3.

2) Cycles	LM2500	MS6001B	MS7001E
HRSG	Fired	Unfired	Unfired
HP (psig/°F)	850/825	850/825	1450/950
LP (psig/°F)	NA	150/sat	150/sat
Steam Turbine Type	Noncond ~7 MW	Noncond ~6 MW	Extr Cond ~22.5 MW

CONCLUSION

This paper has briefly reviewed technical considerations in the application of steam turbine and/or gas turbine cogeneration systems. These basic principles, combined with sound cost estimating procedures, can provide system designers an effective means of quickly defining which alternatives merit serious consideration.

Years ago, the appropriate cogeneration system for a specific application was significantly influenced by its "fit" into the plant heat and power requirements. However, the promulgation of PURPA introduces an additional "degree of freedom," since cycle power generating capability relative to the heat that can be supplied to process is no longer significantly constrained. Furthermore, economic analyses are illustrating that, in many applications, this is the appropriate method of cogeneration cycle development.

