AN OVERVIEW OF COGENERATION TECHNOLOGY— DESIGN, OPERATION AND MAINTENANCE

by

Meherwan P. Boyce

President

Cyrus B. Meher-Homji

Manager of Advanced Technology

and

Alfred B. Focke

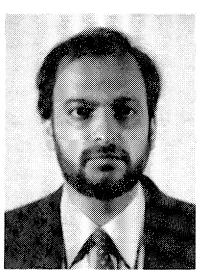
Chief Engineer and Manager of Research and Development

Boyce Engineering International, Incorporated

Houston, Texas

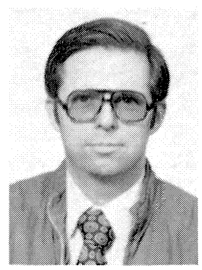
Meherwan P. Boyce is President of Boyce Engineering International, Incorporated, in Houston, Texas. His past experience incorporates many academic and professional positions, including Director of the Gas Turbine Laboratories and Professor of Mechanical Engineering at Texas A&M University, where he organized and was chairman of the Turbomachinery Symposium. Dr. Boyce has authored more than eighty significant papers and technical reports on fluid mechanics and turbomachinery, and is the author of an engineering textbook, Gas Turbine Engineering Handbook. *He is a member of SAE, ASME and several other professional and honorary societies. He is also a registered professional engineer in the State of Texas.*

Dr. Boyce received B.S. and M.S. degrees in Mechanical Engineering from the South Dakota School of Mines and Technology and the State University of New York, respectively, and his Ph.D. in Mechanical Engineering from the University of Oklahoma.



Cyrus B. Meher-Homji is Manager, Advanced Technology with Boyce Engineering International in Houston, where he is involved with a variety of consulting work in the turbomachinery and engineering management areas. His consulting activities include design, vibration and performance analysis, economic analysis and reliability engineering. In the past, he was a development engineer involved in the design and development of a 365 hp prototype, externally fired, steam injected gas turbine. His areas of interest are rotordynamics, turbomachinery prognosis and diagnosis, engineering management and reliability and maintainability.

Mr. Meher-Homji has a B.S. degree in Mechanial Engineering, an M.E. in Industrial Engineering from Texas A&M University and an M.B.A. from the University of Houston. He is a member of ASME, and has several publications in the area of turbomachinery engineering.



Alfred B. Focke is Chief Engineer for Boyce Engineering International, Incorporated, in Houston, Texas. His duties there include the management of engineering projects, as well as overseeing research and development activities of the company. He is heavily involved in computer-based turbomachinery monitoring systems. Mr. Focke has been personally involved in education, plant commissioning, vibration analysis and the development of an externally fired gas turbine.

Prior to joining Boyce Engineering, Mr. Focke was Staff Machinery Consultant with Lagoven in Venezuela and Staff Consultant on the reconstruction of a 100,000 hp platform, after a major fire, and to a 500,000 hp gas turbine and gas compressor shop. He was involved in the engineering and troubleshooting of bearings and seals, vibration, balancing and special repair techniques. He is also former Head of Mechanical Maintenance Engineering with Oil Service Company of Iran (OSHO), where he directed a staff of 21 engineers for 462 gas turbines totalling 3,500,000 hp. He was Chief Designer on fuel systems, air filtration and special testing for the Turbo-Power and Marine Division of Pratt and Whitney.

ABSTRACT

The growth of cogeneration technology started in the 1960s, but has recently accelerated with several users interested in cogeneration applications. An overview of cogeneration technology is presented, with current experience and prospects, including problem areas, in cogeneration plants detailed. An emphasis is placed on gas turbine based cogeneration systems, as these will account for over 50 percent of the market in the future.

As the literature covering cogeneration is so varied, several concepts with references are brought together. The area of reliability, availability and maintainability is addressed in depth, as this is an area where the literature is somewhat lacking. Some future concepts that will have an impact on cogeneration are also discussed, including the use of closed gas turbine cycles, fluidized beds and dual fluid cycles.

INTRODUCTION

Energy costs in the past decade have risen dramatically. With this large increase in energy costs, the acceptability of inefficient engine systems is very limited. Turbine and diesel engine efficiencies are in the low 20 percent range (with some even lower), which implies that between 70 percent and 80 percent of the energy supplied is wasted.

The recovery of waste heat has become economically feasible and is being addressed today on a very large scale. Industrial plants throughout the world are considering cogeneration and combined cycles. Bottoming cycles for gas turbine drives are being investigated with many working fluids, such as steam, freon, ammonia, butane, ethylene, etc. The use of heavy crudes and other dirty fuels, such as pulverized coal and waste products, such as sawdust, support the use of externally fired gas turbines, to overcome the reduced hot gas path life caused by contaminants in the fuel. The use of waste heat in plants from various processes, whether they be in the petrochemical, iron/steel, or the paper industry, require the design of modified power generation units which can directly convert waste energy into useful energy.

Cogeneration, or the combined generation of both power and heat, has been utilized for over a hundred years and has been given a number of names (total energy or combined heating and power). As a result of the energy awareness that started around 1973, United States industry has shown an increasing interest in the cogeneration concept. Early cogeneration systems did not tie into the electric utilities. This created problems in steadily maintaining the demand for electricity and heat. Because of this, some of the early cogeneration systems were not totally successful.

The underlying concept involving cogeneration is that current day prime movers have low efficiencies, which implies that a greater part of the fuel energy is being converted to heat rather than shaft horsepower. Cogeneration would then involve the sequential utilization of the waste heat for some process related need—drying, steam production, etc.

As of 1983, about five percent of the power in the United States was cogenerated. In Europe, where energy costs have been historically higher, cogeneration has been well established. For example, in Germany, about 25 percent of the power consumed is cogenerated. In the fall of 1978, the United States Congress enacted the Public Utility Regulatory Policies Act, commonly known as PURPA. Part of PURPA required that the Federal Energy Regulatory Commission (FERC) develop rules to encourage cogeneration. This resulted in the February 1980 ruling that qualified cogeneration facilities may parallel utility grids and should be paid rates for electricity that are equal to the cost that the utility avoids by not having to generate or obtain power from another source, i.e., the utilities *avoided cost*. Utilities also had to provide standby power at non-discriminatory rates and were obligated to interconnect. State regulatory commissions were then required to implement FERC's rules. By March 1981, most states (and unregulated utilities) had some rules in place. Several states simply rubber stamped the ideas and adopted FERC rules, while other states developed very detailed specific documents, including methods for measurement of avoided costs, setting rate levels and definining contract terms. The rubber stamp approach left it to the utilities to determine how avoided costs were to be measured and how rates were to be structured.

There are several unresolved legal, environmental, interconnection and regulatory problems that make implementation of cogeneration difficult. The best advice for any potential cogenerator is to get *expert* help in dealing and negotiating with the state agencies and utilities, and to keep informed of changes occurring in the laws [1,2]. There are several firms that specialize in the legal, regulatory and procedural steps of initiating a cogeneration project and these consultants should be utilized. There are several cases in which obstacles will seem insurmountable and professional help is the best way to solve this problem.

The term avoided cost is deceptively simple to define, but it is very difficult to find much agreement on the term. Several legal battles have occurred, and probably millions of dollars have been spent in legal fees on this issue. As utilities utilize economic dispatch programs, they use their most efficient units first and their least efficient plants last. Therefore, as a cogenerator comes on-line, the utility would shut down the least efficient unit. The avoided cost would thus relate to this unit, which explains why avoided costs are high (typically $0.05 to $0.07/kWh).

Gas turbine cogeneration is far more efficient than a typical utility central plant. About 75 percent heat utilization can be realized for power and heat, with about 25 percent leaving in the exhaust gases. In a fossil plant, only 35 percent of the fuel energy is obtained as power with condenser losses and boiler losses accounting for 48 percent and 15 percent, respectively [3].

In a modern coal fired plant, a 13 to 19 billion dollar industrial cogeneration market is projected [4]. A 15,000 MW increase in cogeneration capacity is estimated by 1995. Gas turbine based cogeneration and combined cycles will dominate this market. Some interesting projections regarding gas turbine cogneration predict that about 70 percent of cogeneration utilization will occur in Texas and Louisiana, with about half of the sales occurring in the petrochemical industry. Most of the systems will be for units with steam capabilities exceeding 100×10^6 Btu/hr.

Cogeneration systems are so varied that classification is not an easy task. One way of classifying the systems is as follows:

- *Utility Cogeneration*—These are funded and operated by a municipality usually involving large units with district heating and cooling. This concept has been widely used in Europe.
- *Industrial Cogeneration*—This is operated for a private sector industry, i.e., a petrochemical plant, paper mill, etc.

An overview of cogeneration may be found in [4, 5].

Cogeneration Qualifications

Cogeneration qualifications are designed to promote *meaningful* energy conservation and to ensure that the cogeneration concept is not abused by the creation of facilities that produce power and a trivial amount of heat. To qualify as a cogenerator, the system designed (utilizing fuel oil or natural gas) must have a "PURPA Efficiency" of at least 42.5 percent on an annual basis. If the thermal energy output is less than 15 percent of the total energy output, the PURPA efficiency to be met increases to 45 percent.

$$\text{PURPA Efficiency} = \frac{\text{Electric O/P} + \frac{1}{2}\text{ Thermal O/P}}{\text{Energy Input (LHV)}} \qquad (1)$$

where: O/P = Output

For example, assume the cogeneration system as shown in Figure 1. The system involves a gas turbine that produces 3825 kW of electricity at a 95 percent utilization (i.e., 8322 hr/yr). The gas turbine heat rate is 12,186 Btu/kWh. The system also utilizes duct burners that are used for 6146 hr/yr, and are rated at 13.6×10^6 Btu/h. The system generates 31,540 lb/hr of 125 psig saturated steam. The electric power output is therefore:

$$E = 3825 \text{ kW} \times 8322 \text{ hr} \times 3412.2 \text{ Btu/kWh}$$
$$= 108{,}616 \times 10^6 \text{ Btu}$$

The system also generates 31,540 lb/hr steam. Thermal output is given by:

$$T = 31{,}540\ \text{lb/hr} \times 8322\ \text{hr} \times 1011.7\ \text{Btu/lb} = 265{,}547 \times 10^6\ \text{Btu}$$

The Energy Input (EI), based on both gas turbine and duct burner fuel flows, is given by:

$$\begin{aligned} EI &= (3890\ \text{kW} \times 8322\ \text{hr} \times 12{,}186\ \text{Btu/kWh}) + (13.6 \times 10^6\ \text{Btu/hr} \times 6146\ \text{hr}) \\ &= 394{,}492 \times 10^6\ \text{Btu} + 83{,}585 \times 10^6\ \text{Btu} \\ &= 478{,}078 \times 10^6\ \text{Btu} \end{aligned}$$

PURPA Efficiency, therefore, is given by:

$$\text{PURPA}\eta = \frac{E + \frac{1}{2}T}{EI}$$

$$\text{PURPA}\eta = \frac{108{,}616 + \frac{1}{2}(265{,}547)}{478{,}078} = 50.5\ \text{percent}$$

As this exceeds 42.5 percent, the system is "qualified" in this aspect. It must be noted that there are other qualifiers. The operating standard qualifier requires that a topping cycle produce a minimum of 5 percent of the total energy output as useful thermal energy. The ownership qualification states that a utility may not own more than 51 percent of a cogeneration plant.

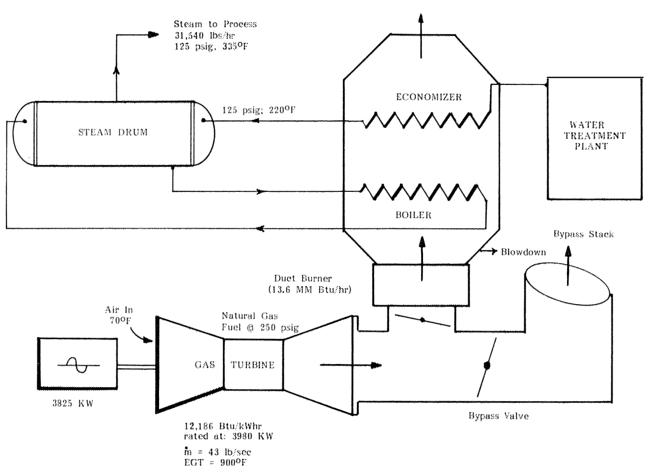


Figure 1. Typical Cogeneration System.

GAS TURBINE CYCLE CONSIDERATIONS

The utilization of gas turbine exhaust gases in regenerators, for steam generation or the heating of other heat transfer mediums is not a new concept and is currently being exploited to its full potential. In the United States there are about 45 combined cycle power plants, producing approximately 6900 MW in electrical utility use. There are also installations of various types in the petrochemical industry totalling in excess of one million horsepower.

Regenerators, combined with gas turbines, capture waste heat and increase the air inlet temperature to the combustor. This reduces the amount of heat necessary to be added to the gas in the combustor, thus increasing the overall turbine efficiency. Turbine efficiency for simple cycle units is dependent on the pressure ratio developed in the compressor and the cycle peak temperature of the combustion gases. The relationship between horsepower and efficiency for simple cycle gas turbines, and of regenerative cycles, is described in [6]. It is important to note that *not all turbines* would benefit from regenerators, especially those operating at low pressure ratios and firing temperatures. The effects of pressure ratios and firing temperatures on performance are shown in Figure 2. With the right pressure ratio and temperature, however, gas turbine efficiencies can be increased by ten to twenty percent. This increase in turbine efficiency could provide a fuel savings of about three million dollars for every ten percent increase in efficiency (assuming a 20,000 hp unit operating base loaded for a year and a fuel cost of $4.00 per million Btu). In most cases, the additional cost of a regenerator installation can be recovered in the first 12 to 18 months of operation. It is this large return on investment which makes regeneration a very interesting proposal. It is important to note that while regeneration involves waste heat recovery, it does *not* constitute cogeneration. Several cogeneration facilities, however, use regenerated gas turbines. Waste heat recovery has also been used successfully by gas distribution companies, where as much as 0.2 to 0.3 percent of the gas transported is utilized in running the pumping stations [7].

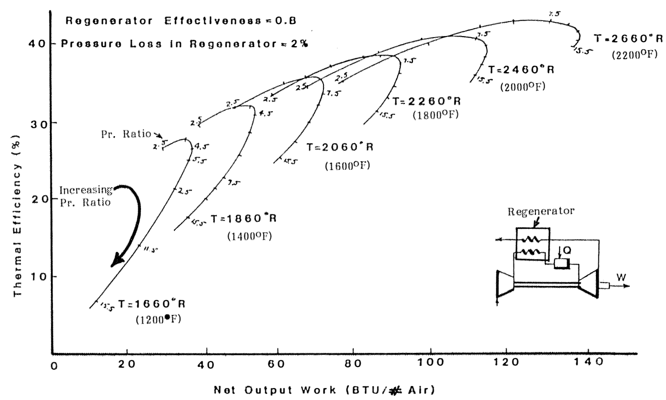


Figure 2. Performance Map Showing the Effect of Pressure Ratio and Turbine Inlet Temperature on a Regenerative Cycle [6].

The combined cycles used in power plants have generally been limited to the use of steam as the bottoming cycle. A typical combined cycle power plant uses the exhaust gases from a gas turbine to produce steam in a boiler (which may have the capability of supplementary firing), for further utilization in a condensing steam turbine. Again, a pure combined cycle plant is not a cogeneration system.

In some petrochemical applications the steam is used for various other processes. This type of plant is called a cogeneration plant. In most of these applications, the gas turbine is a single shaft unit, exhausting gas at a temperature between 900°F and 1050°F, depending on the turbine efficiency and the turbine inlet temperature. The hot gases (approximately 720,000 lb/hr for a 15 MW turbine) are piped into a boiler where steam is generated for use as process heat or for use in an extraction or backpressure steam turbine. The steam turbines usually drive separate generators; however, the system can be designed in a way that both the gas turbine and the steam turbine drive the same generator. A typical combined cycle installation is shown in Figure 3. Dampers may be used between the gas turbine exhaust and the heat recovery steam generator. In this type of configuration, the turbines can be used easily in a simple cycle mode, thus allowing great flexibility in operation. The effect of the gas turbine pressure ratio and inlet temperatures on a combined cycle power plant is depicted in Figure 4. Note the high efficiencies which are possible. There are a number of combined cycle plants with efficiencies well above 40 percent. A comparison of heat rates between different cycles is shown in Figure 5. Capital costs for these plants vary anywhere between $400 and $800 per kilowatt.

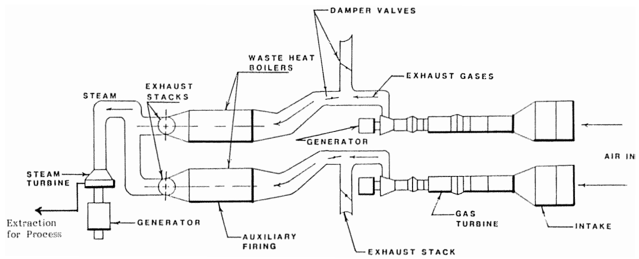


Figure 3. Combined Cycle-Gas Turbine/Steam (Brayton-Rankine).

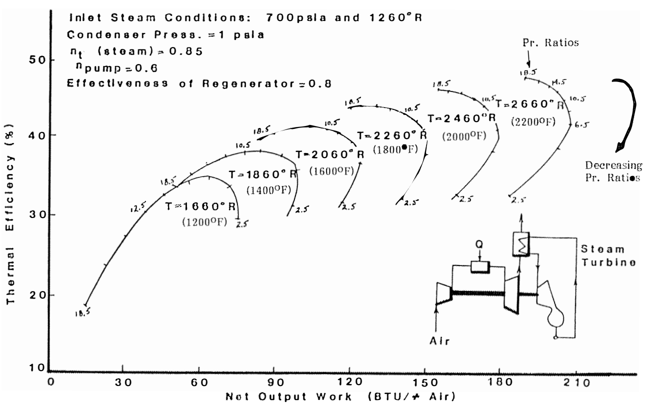


Figure 4. Performance Map Showing the Effect of Pressure Ratio and Turbine Inlet Temperature on a Brayton-Rankine Cycle [6].

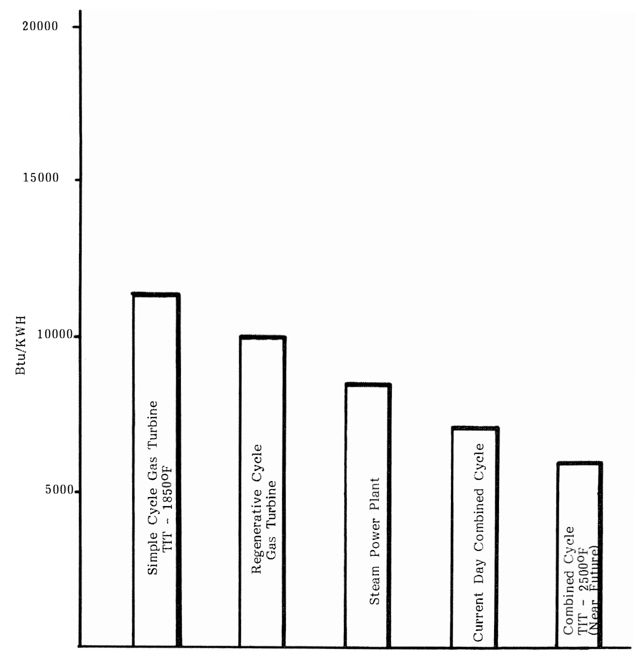


Figure 5. Comparison of Heat Rates of Various Cycles [6].

The use of bottoming cycles other than steam are being investigated, since they offer high energy transfer in the temperature ranges lower than those in which gas or steam turbines operate efficiently. The cost of installing these organic bottoming cycles can sometimes be very high, between $350 and $700 per kilowatt, due to the high cost of the heat exchangers, condensers, pumps and piping required in addition to the turbine. The turbine or expander cost would be approximately $30 to $50 per kilowatt. There are many organic fluids which can be considered. The following are some of the major thermodynamic criteria which an ideal fluid must have:

- It must be stable over all operating conditions.
- The specific heat must be low and the critical temperature must be high, probably over 500°F.
- The inlet pressure should be below 2000 psi in order to keep costs within reason and above 500 psia to produce a reasonable amount of work per pound of fluid.
- The saturation pressure must not be below atmospheric conditions at − 30°F in order to avoid a vacuum situation during shutdown. The saturation pressure, however, must not be too high at operating conditions, thus avoiding excessive back pressure on the turbine and in the condenser.

Typical bottoming cycle fluids and their corresponding molecular weights are shown in Table 1. The major advantage for the use of organic bottoming cycles rather than steam is the low temperature of the gas. Closed cycles, using fluids such as helium, can also be coupled to an organic bottoming cycle as shown in Figure 6. Analysis conducted by General Electric Company in the National Aeronautics and Space Administration (NASA) sponsored Energy Conversion Alternative Study (ECAS) indicates overall combined cycle efficiencies as high as 50 percent [8]. Closed cycle gas turbines are discussed in more detail in later sections.

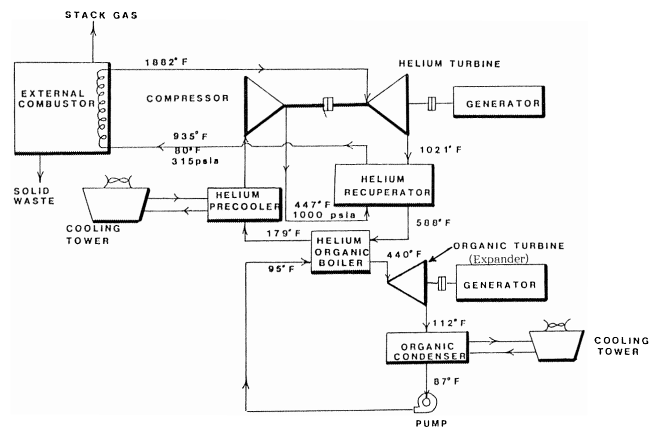


Figure 6. Schematic of a Closed-Cycle Gas Turbine and Organic Rankine Power System.

COMBINED CYCLE PLANTS

Combined cycle plants are those units in which both the Brayton (gas turbine) and Rankine (steam turbine) cycles are "combined" together. This is shown thermodynamically on a temperature-entropy (T-S) diagram in Figure 7. Combined cycle plants have several advantages. These include: 1) high thermal efficiencies (37 to 47 percent), 2) rapid startup (two hour cold start), and 3) low first installed costs ($400 to $800 per kilowatt). Maintenance costs for combined cycles range from $0.003 to $0.007 per kilowatt hour (similar to cogeneration plants).

As several gas turbine cogeneration plants utilize steam turbines (both back pressure and condensing), cogeneration and combined cycles are similar in several aspects—*design and operation lessons learned in one can be applied to the other*.

Table 1. Bottoming Cycle Fluids Characteristics.

Fluids	Formula	M.W.	Working Temp and Pressures
Steam	H_20	18	Press. 1000-1500 psia Temp. 800-1000°F
FL-85	$0.85\ CF_3CH_2OH$ $+0.15\ H_2O$	87.7	Press. 500-700 psia Temp. 450-600
Ammonia	NH_3	17	Press. 1000-1500 Temp. 450-800
n-Butane	$C_4\ H_{10}$	58.1	Press. 500-650 Temp. 450-600
iso-Butane	$C_4\ H_{10}$	58.1	Press. 500-650 Temp. 450-600

An excellent review of worldwide experience in combined cycles is provided in [9] and shows that overall good results have been obtained for availability and efficiency. In Europe there are several combined heating and power plants that involve gas and steam turbines and provide district heating. Combined cycles are becoming increasingly popular, and as a movement toward coal based fuels and higher turbine inlet temperatures continues, combined cycles will become more important. There is also a significant interest in converting existing gas turbines in simple cycle operation to combined cycle operation [10]. Results indicate that for a base loaded plant running 5000 full load equivalent hours per year, the production costs are 26 percent less for a combined cycle plant at a fuel cost of $34/bbl. A commonly used rule of thumb is that 50 percent to 60 percent of the gas turbine output can be added by using the Rankine cycle without any further fuel usage.

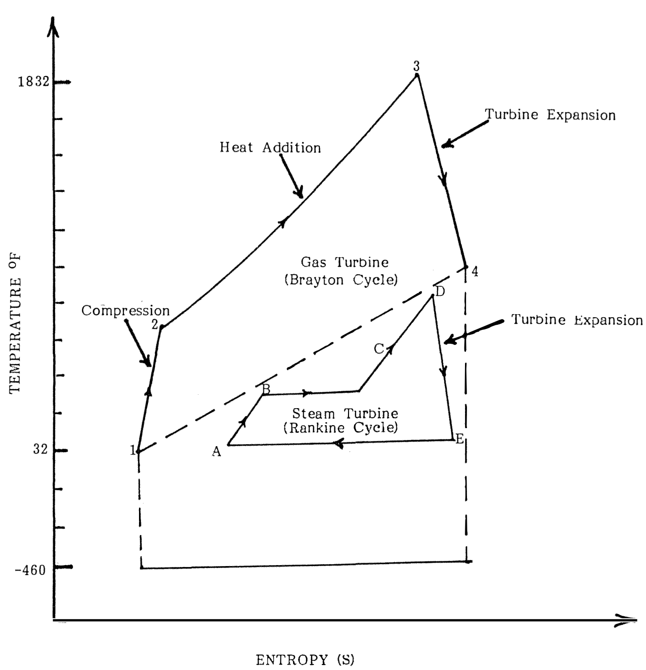


Figure 7. T-S Diagram Showing Combined Cycle.

Several studies indicate that coal fired combined cycles will be of importance in the near future. Several demonstration plants have been developed, including: 1) coal gasification plants deriving fuel for use in a gas turbine combined cycle, 2) cycles utilizing atmospheric and pressurized fluidized beds (AFB and PFB), and 3) gas turbine reheat cycles that will operate at efficiencies around 55 percent.

An interesting development in combined-cycle technology is the ambitious "Moonlight Project" in Japan. The advanced gas turbine currently being tested is a 124 MW (ISO), 484 lb/sec, 55 percent combined cycle thermal efficiency unit. The machine operates at a 55:1 pressure ratio (intercooled by an evaporative intercooler) and at a turbine inlet temperature of 2373°F [11, 12]. The concept or reheat applied here involves the use of two combustors. Several extensive studies on the reheat gas turbine have been conducted by Rice [13, 14]. The second combustor is used to reheat the air between the intermediate pressure and low pressure turbines.

COGENERATION FEASIBILITY

Cogeneration projects are very capital intensive with installed first costs ranging from $400 to $1000 per kilowatt. The choice of plants depends on a host of factors including:

- Location of the Plant
- Power and Heat Demand Ratios
- Fuel Cost
- Plant First Cost ($/kW)
- Operations and Maintenance Costs
- Annual Utilization
- Utility Electric Rates
- Net Fuel Rate
- Rate of Return Criteria

Two types of cycles are possible—topping and bottoming. If power is generated first and rejected energy used as process heat, the system is known as a topping cycle. Gas turbine based cogeneration systems fall into this category. If the rejected heat process is used to generate electricity, then the system is known as a bottoming cycle. This discussion will focus on gas turbine based topping cycles. As steam is a very commonly used and convenient heat transfer medium, it is widely used in cogeneration applications. However, any heat transfer fluids may be utilized for heat transfer.

Some typical power/heat ratios for prime movers are [15]:

Diesel engines	1.5
Brayton-Rankine cycle	0.7 to 1.0
Gas turbine (Brayton Cycle)	0.5 to 0.7
Steam turbine (Rankine cycle)	0.018 to 0.13

A representation made by Gorges [15] indicating the typical performance of cogeneration systems with different types of prime movers is featured in Table 2. It may be seen that the combined cycle is by far the best performer.

Table 2. Types of Cogeneration Systems [15].

	Steam Turbine	*Gas Turbine*	*Combined Cycle*	*Diesel*
Characteristics				
Ratio of power kWh/steam MBtu	50	200	250	400
Ratio of power MWe/steam MWt	0.171	0.683	0.853	1.365
Incremental heat rate (per kWh)	4755 Btu	7265 Btu	6708 Btu	8560 Btu
Cogeneration				
Fuel consumption (Btu fuel/Btu steam)	1.377	2.453	2.677	4.852
Ratio of power Btu/fuel Btu	0.124	0.278	0.319	0.281
Ratio of steam Btu/fuel Btu	0.726	0.408	0.374	0.206
Fuel utilization	84.0%	68.6%	69.3%	48.7%
Separate Generation				
Fuel consumption (Btu fuel/Btu steam)	1.676	3.176	3.676	5.876
Savings by Cogeneration				
Fuel Consumption	17.9%	22.8%	27.2%	17.4%

An ideal cogeneration situation exists when there is a coincidence between power and thermal demands. In several cases, the heat demand may vary with seasonal changes (greater heat loads during the winter and less during the summer). The choice of equipment (type, size, etc.) is strongly influenced by the load demand pattern. An excellent overview of application considerations is given by Kovacik [16].

The following data is required to evaluate cogeneration feasibility:

- Power demand with variations (seasonal and daily)
- Required heat loads (heat content and mass flow)
- Peak requirements for heat and power.
- Special plant requirements—heaters, chillers, plant air requirements, etc.
- Fuel availability
- Environmental impact
- Reliability, availability and maintainability considerations.

A method for performance evaluation and equipment selection for combined cycles used in cogeneration, and to determine the most fuel efficient cycle and combination of equipment to meet the load profile, is provided in [17].

STEAM GENERATION CALCULATIONS

Most gas turbine manufacturers have a set of curves for their different gas turbine models showing steam generating capabilities under different conditions. A curve for a 12 MW gas turbine is shown in Figure 8. If curves are not available, a relatively simple calculation with regard to the waste heat boiler can be done to get a ballpark estimate. A schematic representation of a simple waste heat recovery boiler (WHRB) is shown in Figure 9. In order to compute the amount of steam generated, the following items are required:

- Gas turbine mass flow rate (lb/hr)
- Exhaust gas temperature (°F)
- Steam conditions (pressure (psig) and temperature(°F)) required
- Feedwater temperature (°F).

Some general rules of thumb may be stated, as they are implicit in the computation. First, the steam outlet temperature will be 50°F or more below the exhaust gas temperature. Also, the exhaust gas leaving the boiler evaporator must be 40°F greater than the saturation temperature. The pinch point is taken to be not less than 40°F and the water entering the evaporator must be 15°F below the saturation temperature. Based on a simple heat balance the steam flowrate is:

$$\dot{m}_{steam} = \frac{\dot{m}_{air} \times C_p \times (T_2 - T_1) \times K}{(h_2 - h_1)}$$

where:
$\dot{m}_{steam}$ = Steam flowrate (lb/hr)
$\dot{m}_{air}$ = Gas turbine mass flow rate (lb/hr)
T_2 = Exhaust gas temperature
T_1 = Stack exhaust temperature (saturation + 40°F)
C_p = Specific heat at constant pressure (average between T_1 and T_2)
h_2 = Steam final enthalpy (leaving HRSG)
h_1 = Enthalpy of water entering evaporator
K = Radiation loss factor (approximately 0.985)

More details on heat recovery unit design are presented in a later section. It must be noted that any process fluid may be utilized in the heat recovery unit.

GAS TURBINE HEAT RECOVERY

The waste heat recovery system is a critically important subsystem of a cogeneration system. In the past it was viewed as a separate "add-on" item. This view is being changed with the realization that good performance, both thermodynamically and in terms of reliability, grows out of designing the heat recovery system as an integral part of the overall system. Excellent design guidelines are provided in [18, 19].

Some impotant points and observations relating to gas turbine waste heat recovery are:

Multipressure Steam Generators—These are becoming increasingly popular. With a single pressure boiler there is a limit to the heat recovery because the exhaust gas temperature cannot be reduced below the steam saturation temperature. This problem is avoided by the use of multipressure levels.

Pinch Point—This is defined as the difference between the exhaust gas temperature leaving the evaporator section and the saturation temperature of the steam. Ideally, the lower the pinch point, the more heat recovered, but this calls for more surface area and, consequently, increases the back pressure and cost. Also, excessively low pinch points can mean inadequate steam production if the exhaust gas is low in energy (low mass flow or low exhaust gas temperature). General guidelines call for a pinch point of 40°F to 60°F. The final choice is obviously based on economic considerations.

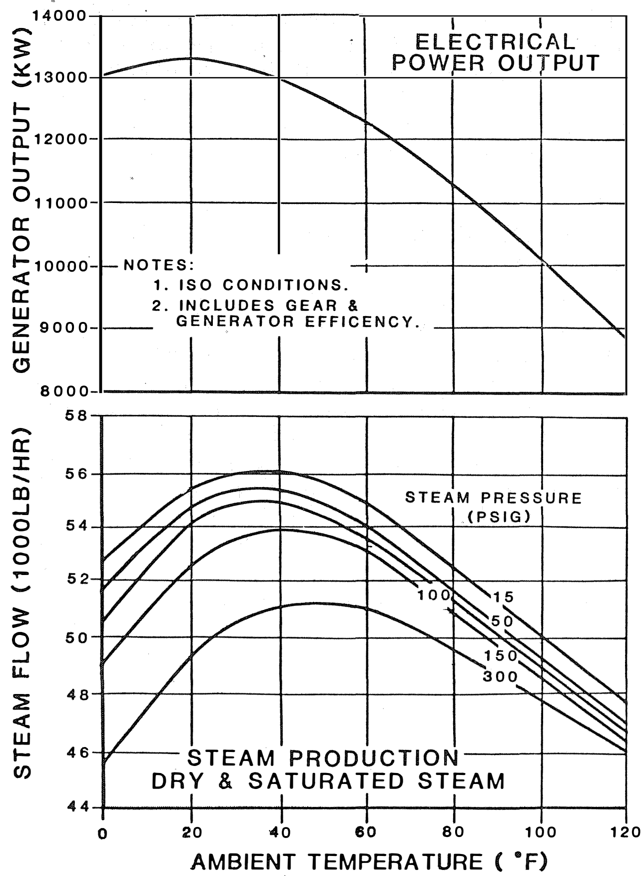


Figure 8 . Steam Production and Electrical Output Vs. Ambient Temperature for a 12 MW (ISO) Gas Turbine.

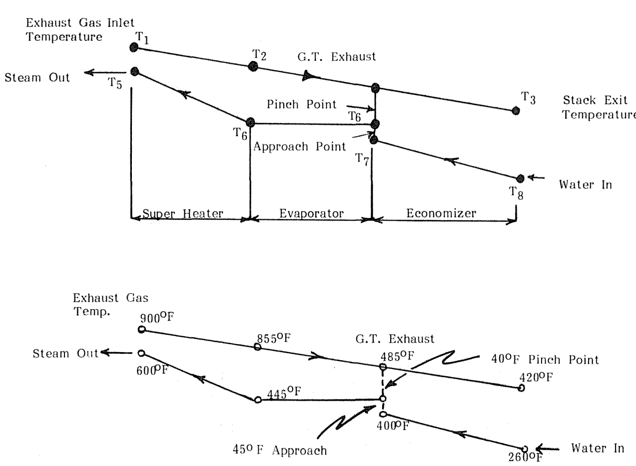


Figure 9. Gas Temperature Profiles in Turbine Waste-Heat Boiler Showing Pinch and Approach Temperatures [19].

Approach Temperature—This is defined as the difference between the saturation temperatures of the steam and the inlet water. Lowering the approach temperature can result in increased steam production, but at increased cost. Conservatively high approach temperatures ensure that no steam generation takes place in the economizer. Typically, approach temperatures are in the 20°F to 50°F range. A typical temperature profile showing the pinch point and approach temperature is also shown in Figure 9.

Off-Design Performance—This is an important consideration for waste heat recovery boilers. Gas turbine performance is affected by load, ambient conditions and gas turbine health (fouling, etc.). This can affect the exhaust gas temperature and the air flowrate. Adequate considerations must be given to how steam flows (low pressure and high pressure) and superheat temperatures vary with changes in the gas turbine operation. This is discussed in [20].

Evaporators—These usually utilize a fin-tube design. Spirally finned tubes of 1.25 in to 2 in outer diameter (OD) with three to six fins per inch are common [19]. In the case of unfired designs, carbon steel construction can be used and boilers can run dry. As heavier fuels are used, a smaller number of fins per inch should be utilized to avoid fouling problems.

Forced Circulation System—Using forced circulation in a waste heat recovery system allows the use of smaller tube sizes with inherent increased heat transfer coefficients. Flow stability considerations must be addressed. The recirculating pump is a critical component from a realiability standpoint and standby (redundant) pumps must be considered. In any event, great care must go into preparing specifications for this pump.

Back Pressure Considerations (Gas Side)—These are important, as excessively high back pressures create performance drops in gas turbines. Very low pressure drops would require a very large heat exchanger and more expense. Typical pressure drops are eight to ten inches of water.

SUPPLEMENTARY FIRING OF HEAT RECOVERY SYSTEMS

There are several reasons for supplementary firing a waste heat recovery unit. Probably the most common is to enable the system to track demand (i.e., produce more steam when the load swings upwards, than the unfired unit can produce). This may enable the gas turbine to be sized to meet the base load demand with supplemental firing taking care of higher load swings.

Raising the inlet temperature at the waste heat boiler allows a significant reduction in the heat transfer area and, consequently, the cost. Typically, as the gas turbine exhaust has ample oxygen, duct burners can be conveniently used.

An advantage of supplemental firing is the increase in heat recovery capability (recovery ratio). According to Pasha [21], a 50 percent increase in heat input to the system increases the output 94 percent, with the recovery ratio increasing by 59 percent. This is shown in Figure 10. The performance of a 200×10^6 Btu/hr furnace when supplementally fired to 300×10^6 Btu/hr is shown in Figure 11. Pasha also provides some important design guidelines to ensure success. These include:

• Special alloys may be needed in the superheater and evaporator to withstand the elevated temperatures.

• The inlet duct must be of sufficient length to ensure complete combustion and avoid direct flame contact on the heat transfer surfaces.

• If natural circulation is utilized, an adequate number of risers and feeders must be provided as the heat flux at entry is increased.

• Insulation thickness on the duct section must be increased.

PROPERTIES OF GAS TURBINE EXHAUST

The exhaust from gas turbines in the simple cycle or regenerative mode represents a large amount of waste heat. Mass flow rates can vary depending on the size of the gas turbine from 4 lb/sec for a 180 kW unit, to a tremendous 1100

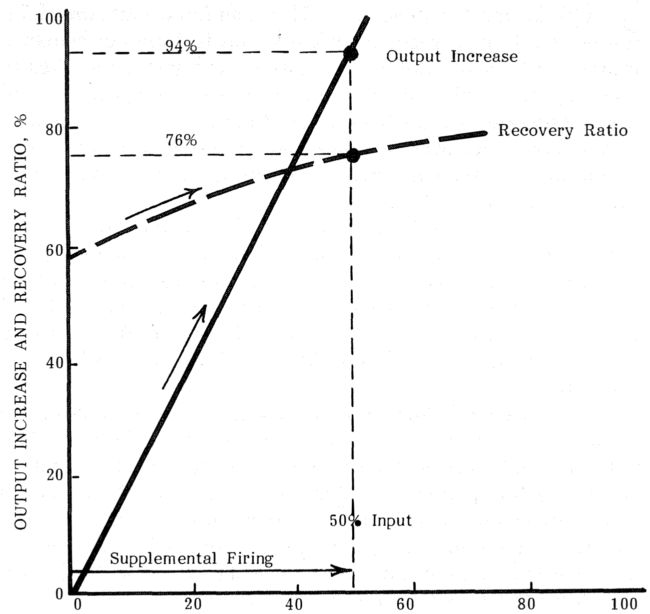


Figure 10. Heat Recovery Ratio and Output with Supplemental Firing [21].

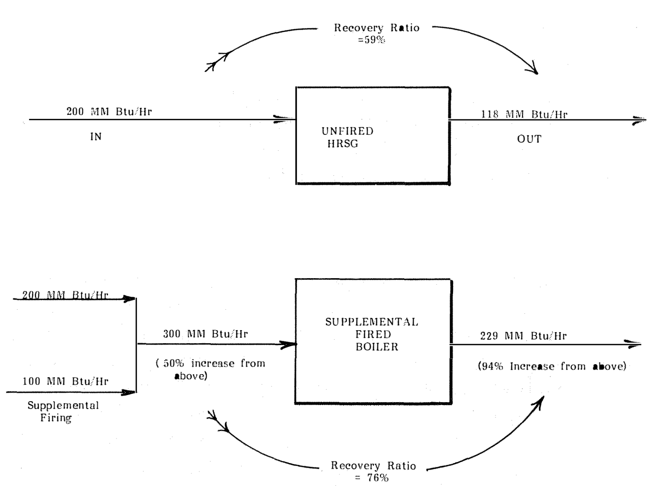


Figure 11. Effects of Supplemental Firing.

lb/sec for a large 143 MW gas turbine. In most cases, exhaust tempeatures will be between 800°F and 1000°F. Gas turbines utilizing recuperators may have exhaust gas temperatures of around 950°F. As gas turbines operate with large amounts of excess air, about 18 percent oxygen is available in the exhaust and this allows supplemental firing.

Temperatures after supplemental firing can be as high as 1600°F and it is often possible to double the steam production by supplemental firing. Variation in the exhaust gas specific heat for combustion is linear with temperature, varying from 0.259 at 800°F to about 0.265 at 900°F (C_p in Btu/lb/°F).

When computing exhaust heat usable for a cogeneration project, capability variations in the gas turbine performance must be taken into account. These include ambient temperature variations, altitude effects, etc. These are discussed in detail in [22]. The effect of back pressure on the gas turbine capability is also important. Back pressure from heat recovery equipment should not exceed ten inches of water (gage). The effect of hot ambient temperatures on gas turbine performance can be significant from a cogeneration perspective, as both electrical output and steam generation may drop off (as was shown in Figure 8). Typically, there is about an 0.5 percent drop in power output for every 1°F rise in temperature. This drop has led to a number of schemes, such as evaporative cooling and chilling of the inlet air, to overcome the problem [23].

COMPUTER PROGRAMS FOR COGENERATION SYSTEM DESIGN

A wide variety of computer codes exist for evaluation of cogeneration projects. Most original equipment manufacturer (OEM) and engineering and construction (E&C) corporations have computer programs available. Programs can vary from disk mounted programs for personal computers for quick feasibility analysis to more complex codes which enable optimization and the consideration of several options. The computer codes could involve detailed financial computations on payback and discounted cash flow. Sample data entry and outputs for one such program are presented in Figures 12, 13 and 14.

```
*******************************************************************
*                                                                 *
*                                                                 *
*              STEAM = 27,000 #/HR                                *
*        STEAM @ $ 4.10/MLBS                                      *
*                                                                 *
*        Date :  3 AUG. 1984  4:26 PM                             *
*                                                                 *
*******************************************************************
```

STEAM FLOW REQUIRED 27000.0 LBS/HR
STEAM PRESSURE REQUIRED 170 PSIA
TEMERATURE REQUIRED SATURATION

GAS TURBINE MASS FLOW REQUIRED 43.51 LBS/SEC

MANUFACTURER AND MODEL NUMBER	EXHAUST GAS TEMPERATURE	MASS FLOW	OUTPUT POWER KW	HEAT RATE	COST $/KW	P2/P1
	925	45.0	4060	11300	300	12.00

SITE AMBIENT PRESSURE IS 14.70 PSIA

MONTH	TEMP	GTURB KW	GT HT RATE	STM FLO	SUP FLO	STURB KW	GT BTU EXHST.	PW/HT RATIO
1	53	3843	11130	24650	2350	0	16410	.41
2	52	3854	11106	24698	2302	0	16442	.41
3	65	3702	11421	24086	2914	0	16035	.39
4	68	3660	11494	23950	3050	0	15944	.39
5	75	3563	11664	23636	3364	0	15735	.38
6	83	3454	11859	23288	3712	0	15503	.37
7	85	3427	11907	23203	3797	0	15446	.36
8	84	3441	11883	23245	3755	0	15475	.36
9	79	3508	11762	23461	3539	0	15618	.37
10	70	3632	11543	23859	3141	0	15884	.38
11	61	3759	11324	24271	2729	0	16158	.40
12	55	3823	11179	24554	2446	0	16346	.40

PURPA EFFICIENCY 52.01 PERCENT

Figure 12. Computer Printout for Cogeneration—Turbine Selection and Performance.

Prior to the use of computer codes, it was possible to make computations that were only relatively accurate. Typically, the following information is needed: 1) plant electricity cost, 2) plant fuel cost, 3) utilization, and 4) gas turbine performance for output and steam generation (obtained from the gas turbine OEM). Installed costs can be estimated at $600 to $1000 per kilowatt. Turbine maintenance costs may be taken at 0.35 to 0.4 cents/kWh. This has to be increased for intermittent operation or heavy fuel operation.

It is important to note that even with highly sophisticated programs, common sense has to be applied. A variety of scenarios should be tested and non-quantitative criteria must be taken into account. *There is no substitute for a brainstorming session with experienced engineers to evaluate the computer results.* All assumptions *must* be critically analyzed. Moreover, it is most important to note that "successful operation" on a computer printout may turn out to be a fiasco

```
FUEL COST                          3.500  DOLLARS/MMBTU
FUEL ESCALATION RATE               6.000  PERCENT

MAINTENANCE COST                   .0030  DOLLARS/KW-HR.
MAINTENANCE ESCALATION RATE        3.000  PERCENT

ELECTRICITY COST                   .0500  DOLLARS/KW-HR.
ELECTRICITY ESCALATION RATE        6.000  PERCENT

STEAM COST                         4.100  DOLLARS/1000 LBS/HR.
AVAILABILITY FACTOR                95.00  PERCENT

AMORTISATION PERIOD                 5.00  YEARS
INTEREST RATE                      14.50  PERCENT
DEPRECIATION PERIOD                10.00  YEARS

GAS TURBINE COST                 1218000  DOLLARS
STEAM TURBINE COST                     0  DOLLARS
HEAT EXCHANGER COST               291600  DOLLARS
EQUIPMENT COST                   1509600  DOLLARS


ENGINEERING COST                  301920  DOLLARS
INSTALLATION COST                1471860  DOLLARS

CONSTRUCTION INTEREST             238045  DOLLARS

TOTAL COST                       3521425  DOLLARS


COST PER KILOWATT                    867  DOLLARS
COST PER MILLION BTU              188933  DOLLARS


  ********************************************
  *      NOTE : TOTAL COST OVERWRITE         *
  *   ENTERING SENSITIVITY ANALYSIS MODE     *
  ********************************************
  INPUT VALUE USED FOR $/KW
  $/KW                              814.0
  INSTALLED COST USED             3304840
```

Figure 13. Computer Printout for Cogeneration—Financial Input.

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5
ELEC. REVENUE	1514076	1604920	1701216	1803288	1911486
STEAM REVENUE	815751	864696	916578	971572	1029867
SYSTM REVENUE	105482	111811	118519	125630	133168
SALES REVENUE	2435309	2581427	2736313	2900491	3074520
FUEL COSTS	1219972	1293170	1370760	1453006	1540187
SUP FUEL COSTS	97990	103870	110102	116708	123710
MAINT. COSTS	136340	140430	144643	148982	153452
INSURANCE COST	4806	4806	4806	4806	4806
G AND A COSTS	29182	30846	32606	34470	36443
OPERATING COST	1488290	1573121	1662917	1757972	1858597
GROSS INCOME	947019	1008305	1073396	1142519	1215923
LOCAL TAXES	15096	15096	15096	15096	15096
CAPITAL COST	3304840				
DEPRECIATION	495726	727065	694016	694016	694016
PRINCIPAL OWED	2643872	2247842	1794387	1275182	680692
REPAYMENT	779392	779392	779392	779392	779392
INTEREST PAID	383361	325937	260186	184901	98700
PRINCIPAL PAID	396030	453454	519205	594490	680691
BEFORE TAX	52836	-59792	104097	248505	408110
TAXES	24304	0	47885	114313	187731
NET INCOME	52836	-59792	56212	134193	220380
AFTER TAX CASH	56068	241322	231023	233719	233705
BEFORE TAX CASH	56068	241322	278908	348032	421435
DISCOUNTED CASH	50971	199440	173571	159633	145112

Figure 14. Computer Printout for Cogeneration—Financial Output.

in practice, unless great care is taken in the system design and installation.

RELIABILITY AND AVAILABILITY CONSIDERATIONS

The success of a cogeneration system depends upon the system being availble, i.e., operating and meeting its demands under expected environmental conditions. A high availability, in turn, depends on reliability (indicating how often the system fails), and maintainability (indicating how fast it can be returned to a satisfactory operating state). A low availability will adversely affect important economic criteria for the project such as discounted cash flow and payback. This section provides a structure by which these important parameters can be addressed at the *design evaluation stage*. Material provided herein on reliability, availability and maintainability and maintenance is taken from [24].

Historically, most of the studies and literature relating to cogeneration systems have dealt with thermodynamic performance or financial aspects. The areas of reliability, availability and maintainability have not been addressed significantly. Economic risks are directly related to technical risks. Experience has shown that numerous cogeneration projects have failed because of an *inadequate* evaluation of the technical risk. Technical risk evaluation must include a variety of factors affecting system operation. The basic risks inherent in a cogeneration project are:

- Failing to complete the project. The risks here would be construction phase risks, such as construction cost and schedule problems, technical problems in design and construction and regulatory problems.
- Failure of the cogeneration system to operate in a manner to generate the expected cash flow. This implies that the expected rate of return is adversely affected as a result of uncontrollable fluctuations in revenues or expenses. Reliability, availability and and maintainability considerations come into play here. Logistic support factors, such as spare part planning, training, etc., are also important.

Since the cogeneration system operating profit depends on the revenue generated through its sales of thermal and electric power, evaluating and reducing risks in the design phase is especially important. A cogeneration facility which utilizes an excellent design concept and is financially and thermodynamically very attractive, but has a poor system design and installation (to lower the installed cost), may cause the availability for several months to be below 40 percent. A design upgrade phase must then be undertaken in which the earlier mistakes are rectified.

While reliability, availability and maintainability (RAM) studies have been used extensively in the power generation field for several years with the Billington and Krasnodebski landmark paper [25] published in 1973, its application to small cogeneration projects has not been widespread. Component and overall system reliability and maintainability must be built in during the *design phase* of any cogeneration project. Reliability and maintainability considerations must touch every aspect of the project, from the choice of the number of gas turbines and waste heat recovery units to be utilized, to specifying small components, such as pumps, dampers, heat exchangers and auxiliary systems. Reliability and maintainability studies must be initiated in the *conceptual phases* of the project and be carried out through the preliminary engineering, project design (procurement and construction phases) and the project testing and commissioning phases. This procedure is shown in Figure 15. Details of the design process and how reliability and maintainability considerations should be addressed may be found in [26].

RELIABILITY, AVAILABILITY AND MAINTAINABILITY DEFINITIONS

The availability of a system is the probability that the system will operate satisfactorily when operated under its design environment. There are several definitions of availability and two commonly used ones are given below.

$$\text{Opertional Availability} = \frac{\text{Uptime}}{\text{Uptime} + \text{Downtime}} \qquad (3)$$

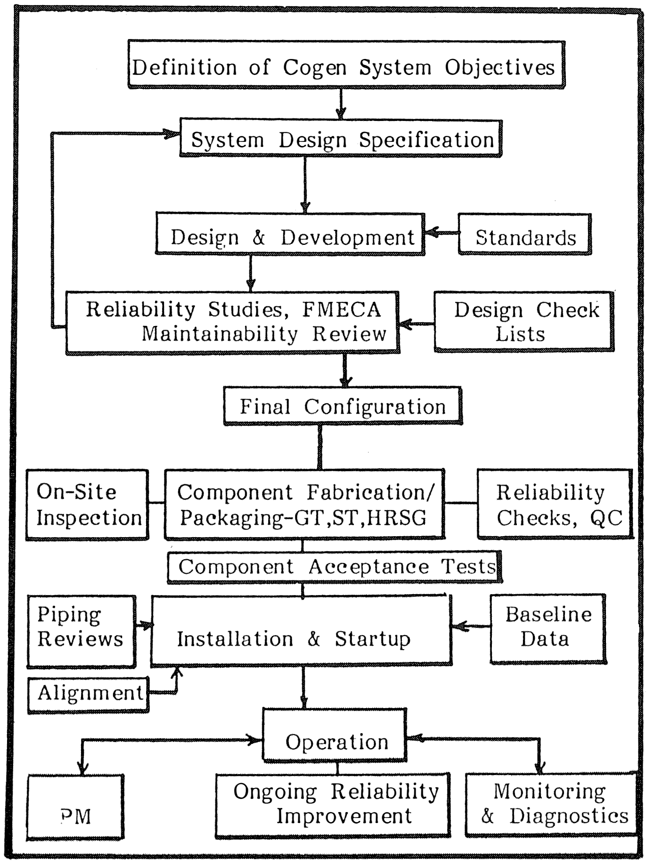


Figure 15. Reliability and Maintainability through Design Operation Phases [24].

The downtime term does not take into account what the specific reason for the downtime was, i.e., it could include maintenance problems, administrative delays, etc.

Another commonly used measure is the inherent availability, which is given by:

$$\text{Inherent Availability} = A_i = \frac{\text{MTBF}}{\text{MTBF} + \text{MTTR}} \qquad (4)$$

where MTBF = mean time between failures
MTTR = mean time to repair

The inherent availability excludes preventive maintenance actions, logistic supply and administrative downtime and is a good measure of the reliability (indicated by MTBF) and maintainability (indicated by MTTR) of the system. In industry, availability is expressed as:

$$A = \frac{\text{Period Hours} - \text{Forced Outage Hours} - \text{Schedule Hours}}{\text{Period Hours}} \qquad (5)$$

The reliability of a system is an inherent characteristic of system design and is the probability that the system will operate in a satisfactory manner for a specified period when used under the stated conditions. The reciprocal of the MTBF is the failure rate (λ).

$$\frac{1}{\text{MTBF}} = \text{Failure Rate} = \lambda \qquad (6)$$

In industry, reliability is often expressed as:

$$R = \frac{\text{Period Hours} - \text{Forced Outage Hours}}{\text{Period Hours}} \qquad (7)$$

The maintainability of a system is also an inherent characteristic of equipment design and installation which is a measure of the ease, economy, safety and accuracy with which maintenance actions can be performed. Common measures for maintainability are Mean Time Between Maintenance (MTBM), or Mean Time to Repair (MTTR). The reciprocal of the MTTR is the repair rate (μ).

$$\frac{1}{\text{MTTR}} = \text{Repair Rate} = \mu \qquad (8)$$

When comparing gas turbine engines in a cogeneration system, aero-derivative units are, in general, easier to maintain (than heavy duty industrial gas turbines), because of their modular design. For example, it is possible to tear down a 34,000 exhaust gas horsepower (EGHP) gas generator module in about 12 hours.

As availability is a function of both reliability and maintainability, it is important that an attempt be made to improve both reliability and maintainability during the design phase. It is important to note that reliability can be significantly reduced by improper installation and startup procedures.

FAILURE MODE, EFFECTS AND CRITICALITY ANALYSIS

Failure Mode, Effects and Criticality Analysis (FMECA) is a technique that allows for a systematic review of all components and subcomponents within the cogeneration system. FMECA lists potential failure modes and the effect of the failure modes on the overall system with an estimate of the criticality of the failure. This enables the user to focus on those areas that represent weaknesses in design. For example, in gas turbine based combined cycle operation, a major source of downtime is caused by gas turbine controls and auxiliaries, as opposed to the hot section parts. Thus, an FMECA focuses attention on particular problem areas.

The steps required to perform FMECA are:

- Define the system under consideration. A convenient way to do this is through the use of block diagrams. A detailed list of the system hardware should also be put on paper.
- Find the failure modes of the system. This can best be accomplished by bringing together individuals with experience in design, construction, operations and maintenance of cogeneration systems and conducting a brainstorming session. This can be performed by analyzing the system block diagram, defining undesirable events and then applying a fault tree analysis technique to the event.

Consider, for example, a simple subsystem of a cogeneration system—the bypass damper. This component is located within the exhaust duct of the gas turbine prior to the heat recovery steam generator (HRSG). Usually two valves will be provided—one in the bypass stack (which will allow the gas turbine to exhaust to the atmosphere prior to the HRSG in the event the HRSG is unusable), and another in the HRSG duct (to isolate the HRSG and the gas turbine). An FMECA focusing on this subsystem would highlight the following:

- If both valves were closed inadvertently, the back pressure would cripple the gas turbine performance. This would require a design solution which could involve a mechanical linkage between the two valves, to ensure they could *never* both be closed.
- In the case of two gas turbines feeding a HRSG, the FMECA would also point out that in the event of one turbine being down, it must be ensured that hot gases do not enter the non-operating gas turbine.

When dealing with cogeneration systems, it is instructive to analyze experience obtained with combined cycles. An excellent study has been conducted by Dolbec [27]. Based on a survey of 14 plants, the contribution of the different components to the outage hours and outage events is shown in Table 3. A good representation of the outage hours and outage events

Table 3. Combined Cycle Outage Contribution [27].

	% of Forced Outage Hours	% of Forced Outage Events
Combustion Turbine System	65.3	58.4
HRSG System	16.5	20.3
Steam Turbine System	12.7	10.3
Plant Level Equipment	5.4	11.0
TOTAL PLANTS	100	100

for different components related to the gas turbine is featured in Figure 16. Mechanical auxiliaries and controls are in the right upper corner of this map indicating that they are responsible for a large number of both outage hours and outage events. The results on combined cycle plants are shown in Figure 17, as reported by Strong, et. al. [28]. It is interesting to note that a report from a 143 MW combined cycle plant in Japan [29], 52 percent of the outages were caused by gas turbine auxiliaries, 28 percent by the HRSG auxiliaries and 20 percent by gas turbine controls. Thus, in this plant, controls and auxiliaries accounted for *all* of the outages. Some outage causes in a large combined cycle power plant [30] are shown in Figure 18.

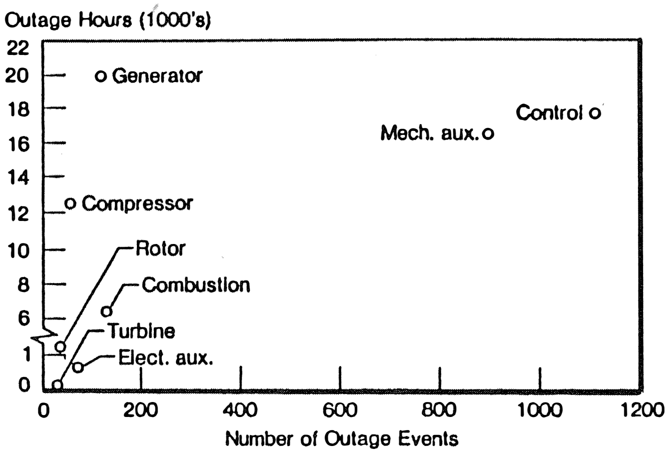


Figure 16. Outage Hours and Events for Combined Cycles [27].

DESIGN REVIEW CHECKLISTS FOR RELIABILITY AND MAINTAINABILITY

Design reviews must be conducted keeping reliability and maintainability in mind. Some important questions that may be used as a checklist for cogeneration systems are presented in the Appendix. The checklist is not an exhaustive one, but represents a framework for the questions that should be asked from a reliability and maintainability standpoint. The checklist has been derived and modified from a number of sources including [31].

REALIABILITY CONSIDERATIONS

There is a close relation between the theory of reliability and probability. Some very fundamental concepts are provided

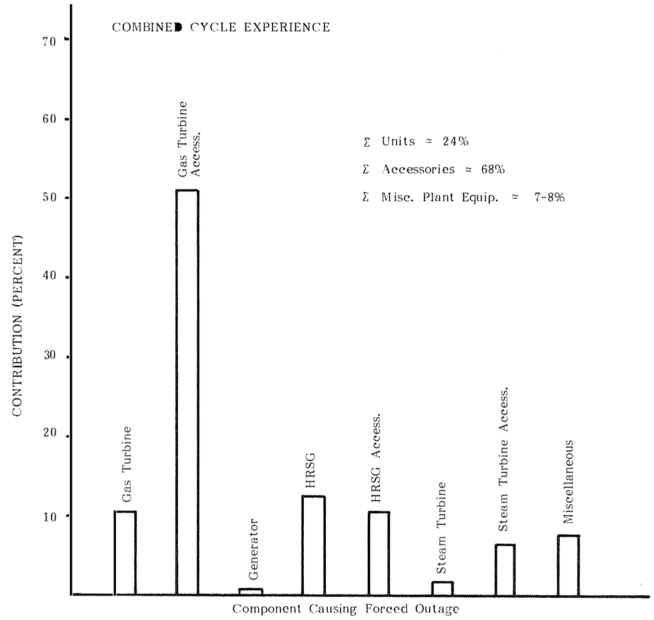


Figure 17. Combined Cycle Outages [28].

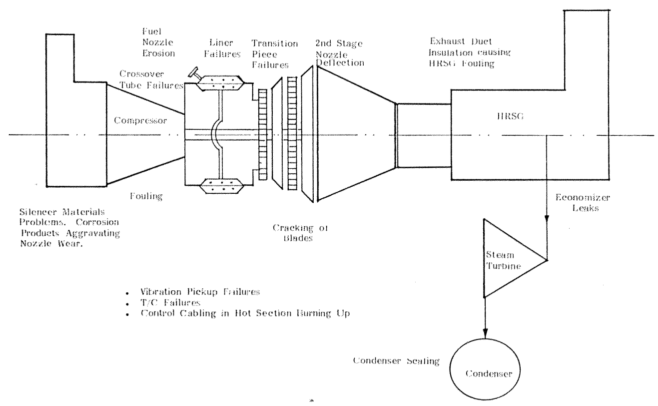


Figure 18. Problem Areas of Combined Cycle [30].

here. An excellent treatment of reliability is provided in [32, 33, 34].

If a system has "n" number of components in series, and if the failure of one component is independent of all other failures, then the system reliability is given by:

$$R_s = R_1 \times R_2 \times \ldots . R_n = \prod_{1}^{n} R_i \tag{9}$$

where R_i = reliability of the components within the system.

If a parallel configuration exists, i.e., a case with redundancy then the system reliability is given by:

$$R_s = 1 - \prod_{i=1}^{n} (1 - R_i) \tag{10}$$

Complex systems can be reduced to a simple system of series and parallel components.

As would be expected, a higher component reliability is required in a series system to keep system reliability high. As the number of components increases, component reliabilities need to be even higher. The opposite holds true for a system in parallel. A graphical representation of the difference between series and parallel systems appears in Figure 19. The simple concepts of series and parallel reliability can be useful tools in evaluating cogeneration system design. For a specified component reliability, adding more components in a series system *decreases* system reliability, while the addition of components in a parallel system *increases* system reliability.

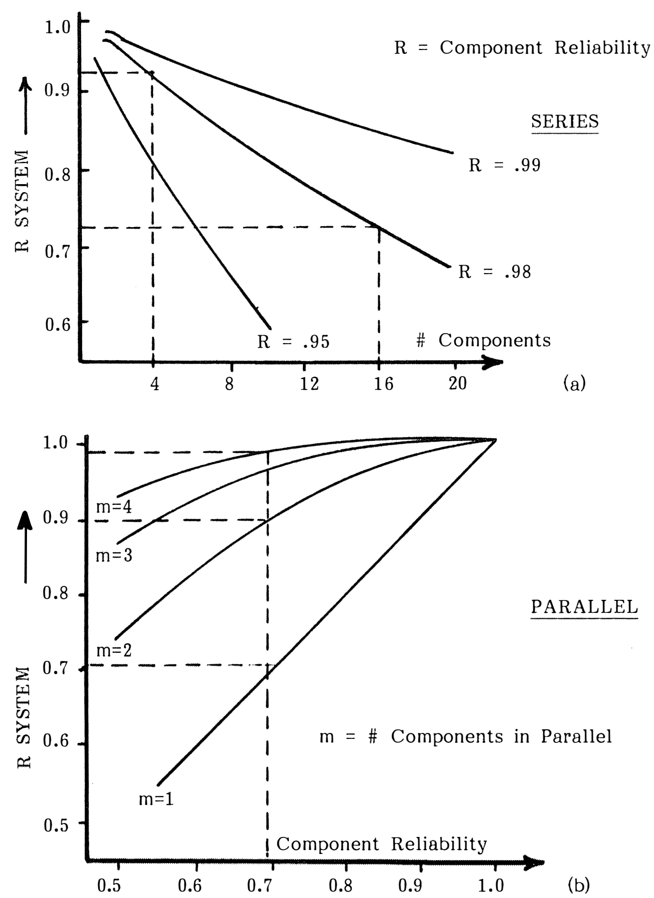


Figure 19. System Reliability Series and Parallel.

For a very simplified reliability calculation for a cogeneration system, assume an exponential distribution to be valid. The example is provided for conceptual purposes only.

Assume that the cogeneration system can be split up as a reliability tree shown in Figure 20. The failure rates for each component are represented by λ. The component reliability is given by:

$$R = e^{-\lambda t} = e^{-t/\theta} \tag{11}$$

where $t = 1$ year (8,760 hrs) and
$\theta = \text{MTBF}$

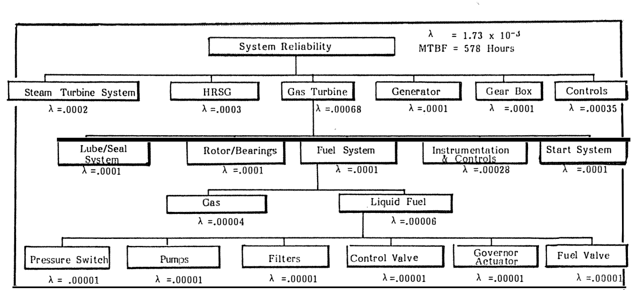


Figure 20. Simplified Reliability Logic Tree for Cogeneration System.

The availability of liquid fuel requires the operation of pressure switches, pumps, filters, control valves, governor activators and fuel valves. These then become a "series" system resulting for the availability of liquid fuel. Other components shown in the tree would also have similar "branches" which have been omitted for simplicity. For example, the HRSG system would have several subsystems included. The overall failure rate of the system is 1.73×10^{-3}, which yields a MTBF of 578 hours. If the Mean Time to Repair (MTTR) of the system is 25 hours, then availability is given by:

$$A = \frac{\text{MTBF}}{\text{MTBF} + \text{MTTR}} = \frac{578}{578 + 25} = 0.96 \tag{12}$$

Redundancy in System Design

Concepts of redundancy play an important role when choosing the basic system configuration. For example, how many gas turbines should be used with one heat recovery unit, and how many independent subsystems should be required? An evaluation of reliability, economics and performance is necessary. Another important decision relates to components such as pumps, in which the number and size of pumps for each application needs to be ascertained (i.e., four, one-third capacity pumps or three, one-half capacity machines).

Statistical theory provides the following relationships. If p is the probability of success, then the probability of at least K successes out of m trials is given by:

$$p(x \geqslant K, m) = \sum_{x=k}^{m} \frac{m!}{x!(m-x)!} p^x (1-p)^{m-x} \tag{13}$$

For three parallel elements where at least two are required for system success:

$$R = p(x \geqslant 2; 3) = \sum_{x=2}^{3} \frac{3!}{x!(3-x)!} p^x (1-p)^{3-x} \tag{14}$$

$$R = \frac{3!}{2!\ 1!} p^2 (1-p) + \frac{3!}{3!\ 0!} p^3 = 3p^2 - 2p^3 \tag{15}$$

Similarly application of equation (13) to a three out of four case yields:

$$R = 4p^3 - 3p^4 \tag{16}$$

Based on equations such as these, it is possible to relate the calculated reliability to the availability, and thus obtain an economic basis for a decision. Obviously, a balance must be attained between the costs associated with the additional redundancy and the economic benefits from the increased availability.

A hypothetical case of a system consisting of a gas turbine and heat recovery steam generator (HRSG) is now considered. Three 50 percent capacity units (i.e., one gas turbine and one HRSG available for backup) are utilized. Two arrangements are possible. In the first arrangement, the gas turbine exhausts are tied into a common duct which feeds the three HRSGs. This arrangement is one of *component* redundancy. The second arrangement is one of *system* redundancy in which each gas turbine and HRSG are linked together forming three parallel paths. The reliability of the two different types of redundancy are examined in Figure 21, assuming a constant HRSG reliability of 97 percent, with the gas turbine reliability varying from 70 percent to unity. The component redundancy case performs more reliably than the case of system redundancy. This is intuitively acceptable, because in the case of *system redundancy*, the failure of a gas turbine makes its associated HRSG useless. In the case of *component redundancy*, all that is required is for any two gas turbines and any two HRSGs to be operating. A similar case of four, one-third capacity units is shown in Figure 22. It can be noted that the reliability of the system consisting of three, one-half capacity units is better than the system utilizing four, one-third capacity units.

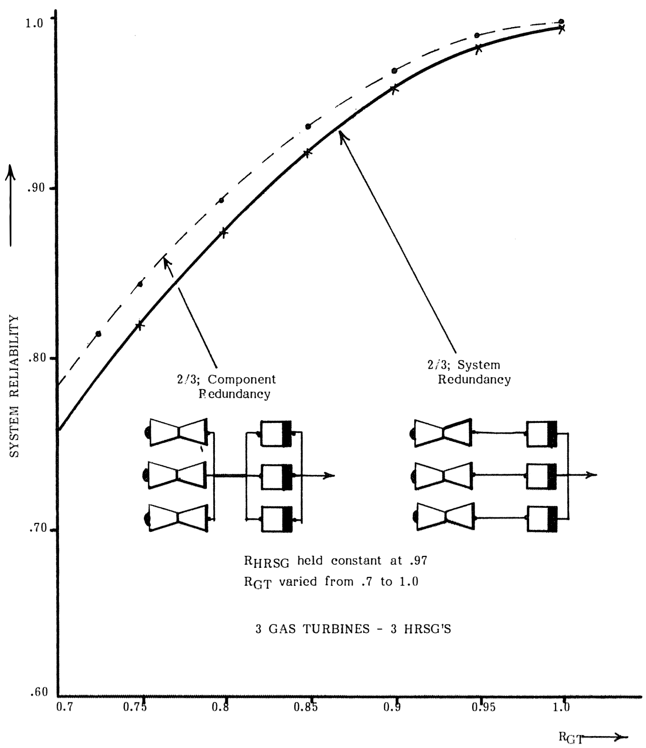


Figure 21. System and Component Redundancy (3 units, 50 percent capacity each).

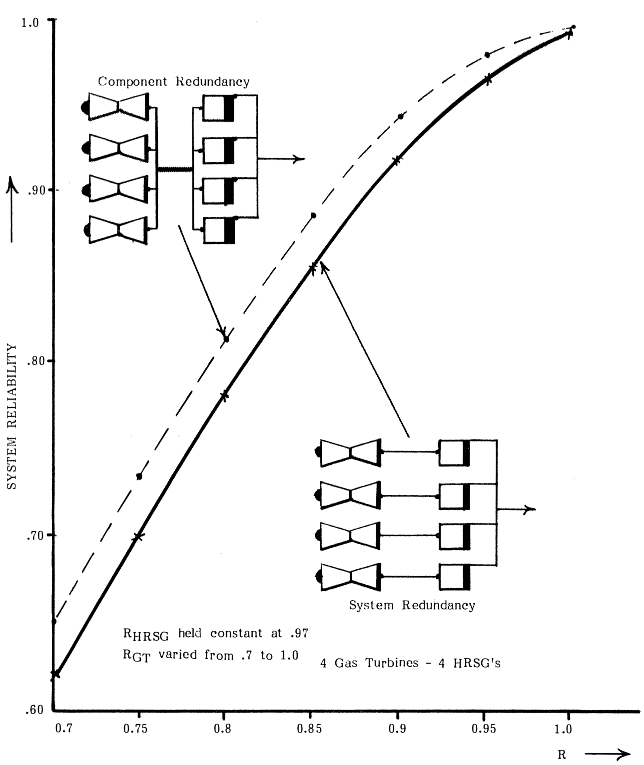


Figure 22. System and Component Redundancy (4 units, 33⅓ percent capacity each).

With cogeneration plants (or combined cycle plants), it is possible for the system to operate under a partial or derated condition. A concept known as equivalent availability may be used, which reflects the amount of time the unit is available and operating at less than full load.

$$\text{Equivalent Availability} = \frac{\text{Available Hrs} - \text{Equiv. Derated Hrs} + \text{Equiv. Seasonal Derated Hrs}}{\text{Period Hours}} \quad (17)$$

A detailed discussion of systems involving multiple gas turbines and HRSGs is provided in [35].

Sources for Reliability Data

There are several sources of historically compiled reliability data. The National Electrical Reliability Council (NERC) is one such source. Original equipment manufacturers (OEMs) also have developed data bases over the years. Combined cycles are in a sense quite similar to gas turbine based cogeneration systems, and data from combined cycles can be used for cogeneration system design. Data cannot be directly applied as one has to take into account things such as unit size, service duty, fuel types and environmental conditions.

With respect to rotating machinery, a very useful reliability assessment framework has been provided by Sohre [36]. Sohre evaluates reliability based on several factors, including the type of equipment, equipment size, number of bodies in the unit, operating parameters, casing support type, coupling type, pipe strain, operator and maintenance personnel qualifications and some other miscellaneous factors. The framework provided points to areas where improvement efforts may be focused when evaluating a system. The concepts used here are not those of classical reliability, but provide an excellent framework for comparative assessment of reliability. A good reference for procedures to assure reliability in process plants has been provided by Bloch [37].

Another useful source of data is the *Handbook of Loss Prevention* [38]. This book includes a historical survey of problems associated in a wide variety of equipment, including electric motors, generators, boilers and steam system components. Knowledge of failure causes and downtime statistics for different components allows system designers to focus on certain areas during the design and specification phases. According to [38], failures on gas turbines can be split into three groups: 1) product faults (including design problems), erection problems and manufacturing and repair defects, accounting for about 69 percent of the failures; 2) user induced defects included material defects and maintenance defects; and 3) operating errors accounting for about 15 percent of the failure, with extraneous influences accounting for about 16 percent.

Gas Turbine Maintainability

For gas turbine engines, some important features for maintainability include: the availability of borescope ports, vibration and condition monitoring systems, diagnostics for electrical and control circuits, availability of special tools and facilities for maintenance alignment, balancing, etc., accessibility of auxiliary pumps, compressor washing system, quick filter changes, heat recovery boiler, etc. Decisions taken during the design phase can have a significant impact over the cogeneration project life cycle. Key items to be reviewed are:

Ease of Access of High Maintenance Items—combustion liners, fuel nozzles, first stage nozzles, etc. For example, some gas turbine units have combustion liners and caps with hinges, enabling them to be unbolted and swung to one side without the use of special lifting tackle.

Ease of Access for Inspection—borescopes and inspection doors should be provided to gain access to the compressor intake and to the hot section. In the case of large gas turbines and heat recovery units, adequate safety handrails, walkways and ladders should be provided. Provisions should also be made for the availability of lighting, power points for tools, etc.

Auxiliary Systems and Controls for the Gas Turbine or Steam Turbine (if applicable)—must be accorded special attention. Arrangements should be made for duplex filters so that one filter can be cleaned while the other is in service. Lube oil systems should have provisions for connections with centrifuges so that oil cleaning can be accomplished as fast as possible.

Site Plans and Piping Plans—must be reviewed by maintenance engineers to ensure adequate space is available for overhead cranes or swing cranes.

In the case of industrial gas turbines, sufficient adjacent space must be provided for removing and temporarily storing both the upper casing and the rotor. A review must also be made of all piping to ensure that isolation valving and spool pieces exist to facilitate disassembly.

Experience has shown that in the case of gas turbines, a major area of deficiency has been the accessories—including clutches, pumps, couplings, air compressors, dampers, instrumentation and controls. The user must take extra care when specifying parts in these areas. Instrumentation and dampers become more unreliable with higher environmental temperature. The following areas should be noted in particular:

- Temperature control thermocouple junction boxes must be of the right size and properly mounted.
- Instrumentation must *not* be mounted on vibrating structures.
- All wiring must be supported appropriately.
- The unit should have provisions for instrumentation, performance monitoring and vibration monitoring.

An important part of maintainability is ensuring that adequate spare parts are on hand to avoid excessive downtime in waiting for replacement parts. Another important aspect is training of both the operators and maintenance personnel. By having maintenance personnel trained to deal with forced outages, the repair times can be significantly reduced.

INSTALLATION, COMMISSIONING AND STARTUP

Adequate care taken during installation, commissioning and startup can go a long way in improving system reliability. Specific individuals with adequate experience must be assigned to supervise the overall machinery assembly at the cogeneration site. Special care should be taken during component lifting, cleaning and inspection of bearings, seals and couplings, and other critical components. Proper alignment is most important and must include a hot alignment technique. All lube and seal oil systems must be cleaned and flushed. Gear components must be checked to ensure good face-to-face contact. Trained individuals should review the completeness of various subsystems in the cogeneration system. Special emphasis should be placed on piping, ensuring that it is as per design, and that no piping strain exists.

Performance tests data must be fully obtained and documented. It is also a good idea to record vibration data from the rotating machinery components so that this data could form a useable baseline when problems develop down the line. Adequate documentation is most important in reducing downtime. All installation and maintenance instructions, specifications and performance maps, spare parts lists and *accurate* updated dimensional drawings must be kept on file.

COGENERATION SYSTEM OPERATIONS AND MAINTENANCE CONSIDERATIONS

Gas turbine cogeneration systems are made up of gas turbines, generators, waste heat boilers, boiler feed pumps, steam turbines (possibly), auxiliary systems and controls. In several cases, gas turbines represent the "new machine" in the plant, as steam systems are well established. The emphasis here will be, therefore, on some key aspects of gas turbine maintenance.

Some important factors that impact gas turbines' operating reliability are fuel considerations (including fuel quality, filtration and treatment), compressor air filtration, compressor washing and health monitoring.

It is important to note that maintaining the gas turbine at its peak operating efficiency is most important from a fuel savings standpoint. A pie chart showing a comparison of first cost, maintenance cost and life cycle fuel costs (for present values) for a gas turbine cogeneration system (assumptions: 30 MW unit, 750 $/kW, gas turbine efficiency = 26 percent, fuel cost $4.50/million Btu escalated at seven percent per year, two year life cycle, with maintenance costs taken at five percent of fuel costs per fired hour) is presented in Figure 23. This chart shows the importance of *choosing a high efficiency* gas turbine and most importantly *maintaining* the gas turbine at its peak thermal efficiency, over time. In cogeneration applications, good overall efficiency can be obtained even with the use of turbines with poor heat rates, because rejected heat is utilied for steam generation or for the process, but the use of low efficiency gas turbines should be carefully evaluated.

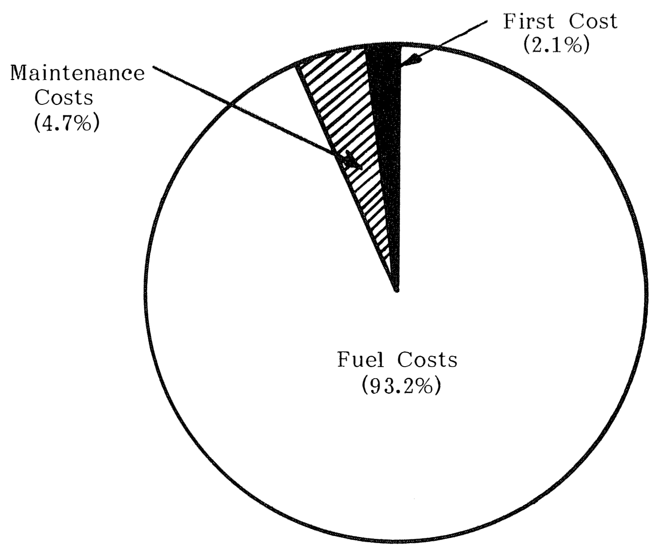


Figure 23. Life Cycle Cost Breakdown (Installed, Fuel and Maintenance Costs) for a Cogeneration System [24].

Compressor fouling is typically responsible for a drop in load output caused by a drop in pressure ratio, air flowrate, compressor efficiency, and overall thermal efficiency. Of these, the most evident is the drop in load—10 percent to 12 percent not being uncommon with a dirty compressor. Effective and well scheduled compressor washing procedures are of paramount importance. Compressor degradation (fouling) can be responsible for 70 percent to 80 percent of the output loss. On a simple cycle gas turbine, a five percent drop in air flow and compressor efficiency can cause an output drop of approximately 13 percent and a heat rate increase of about five percent [31].

The following items should be considered for inspection and maintenance:

• All gages and indicators should be monitored with specific action levels defined, i.e., what the operators should do when a problem is detected. Excessive combustor temperature spreads must be noted and acted upon. All pumps must be visually inspected daily with a special emphasis on seal problems.

• All critical instrumentation and controls must be inspected weekly and during a shutdown. This is a major source of problems. Included here are steam drum controls, pressure control valves and flame scanners.

• Visual inspection must be made of the filters and compartment ventilation systems. Filter pressure drop (ΔP) must be also checked. If evaporative coolers are used, these must be checked every two to three months.

• Several items should be checked on a yearly basis. Air filters should be replaced, the gas turbine combustion section should be checked, fuel nozzles should be inspected, the hot section should be inspected and repaired, the boiler feed pump may be repacked and duct burners should be inspected. The steam drum and deaerator should be opened for visual inspection. A full check for turbine monitoring inspection should be made. Breakers and excitation systems should be inspected.

• Recommended inspections and maintenance must be performed on the machines. These may have to be modified based on environmental conditions, experiences, etc.

COMPRESSOR AIR FILTRATION AND COMPRESSOR CLEANING

Past filter reviews have found that an air filter which controls corrodents also does a good job on foulants. Based on this experience, this discussion will be weighted toward corrosion control, since the foulant control is a healthy byproduct.

Modern day gas turbines operate at a turbine inlet temperature of between 1700°F and 2100°F, which makes them highly sensitive to corrosion problems. This calls for care to be taken with respect to air filtration and compressor washing. Even with good filtration, salt can collect in the compressor section. During the collection process of both salt and other foulants, an equilibrium condition is quickly reached, after which reingestion oflarge particles occurs. This reingestion has to be prevented by the removal of salt from the compressor prior to saturation. The rate at which saturation occurs is highly dependent on filter quality. In general, salts can safely pass through the turbine when gas and metal temperatures are less than 1000°F. Aggressive attack will occur if the temperatures are much higher. During cleaning, the actual instantaneous rates of salt passage are very high, together with greatly increased particle size.

A good air filtration system should include:

First Stage Filter—should be an inertial filter which should remove all particles greater than ten microns.

Second Stage Filter—should be a roughing filter capable of removing particles to the order of five microns.

Third Stage Filter—should be a high efficient filter with filter material that is hydrophobic (unwettable) in nature. In this recommended scheme, the first and second filtration stages are to ensure a long life and efficient operation of the third stage high efficiency filter, as well as to keep operating costs low.

Even with good air filtration, salt deposits will occur in the compressor. As the air moves through the compressor toward the combustion section, it is heated and compressed, causing removal of the remaining moisture from the airborne salt particles. These particles are deposited heavily in the first few stages, sometimes going back as far as half way through the compressor. In general, the condensation nuclei that pass through the compressor without being entrapped, can also pass through the turbine without depositing themselves, or adhering to the hot parts. A difficulty arises, however, when the salt that has been collected within the compressor stages becomes so thick that large flakes are reingested into the engine. When this occurs, the local concentration of salt in the air immediately surrounding these large flakes is extremely high. These salt flakes actually have sufficient mass to stick firmly to the turbine hot parts, and are responsible for many gas turbines suffering from hot corrosion damage. Saturation of the compressor with salts thus still remains a problem and must be dealth with. Compressor cleaning will continue to be necessary even with a machine utilizing an efficient air filter, but on a greatly reduced schedule. Compressor cleaning is very important for the efficient and trouble-free running of a machine and must be carried out on a regular basis.

With an adequate high efficiency air filtration system, cleaning of the compressor should not be needed more than once in four months. Careful performance calculations can indicate when the compressor is beginning to foul, and cleaning can then become a part of a scheduled maintenance program, rather than something done on an emergency or haphazard basis. Salt is soluble in water, and one method of cleaning is to wash out the salt using a soap and water solution. The soap is added to break down or remove any oil which may have condensed within the compressor. While steam cleaning of the compressor is very efficient, water washing, using a water-soap mixture, is the most efficient method of cleaning. This cleaning is most effective when carried out in several steps which involve the application of a soap and water solution, followed by several rinse cycles. Each rinse cycle involves the acceleration of the machine to approximately 50 percent of the starting speed, after which the machine is allowed to coast to a stop. A soaking period follows during which the soapy water solution may work on dissolving the salt. If the machine is cleaned under full operating conditions using rice, wheat, walnut shells or some other solid material injected into the inlet, the foulants and corrosive elements deposited within the compressor are removed rapidly and flushed through the turbine.

If the machine is running at substantial load and the gas temperatures are in the region where the salt can be fused, it will have the opportunity ofcombining with the sulphur (which is present in the fuel), and form sodium sulphate. The sodium sulphate may then fuse, become sticky and deposit itself on the hot parts of the turbine causing severe and rapid attack, which is characteristic of hot corrosion (sulphidation). If the metal temperatures and gas temperatures are less than 1000°F, then the salt will pass safely through the turbine, as it never has the opportunity to become molten.

The method recommended for determining whether or not the foulants have a substantial salt base is to soap wash the turbine and collect the water from all drainage ports available. Dissolved salts in the water can then be analyzed. The amount of water passed through the machine and the time at which it was collected allows the establishment of a proper washing procedure. A program of simple performance tests should be conducted to detect compressor fouling. Pressure drops over filter elements should be recorded daily. Water washing a machine under power must be extremely closely controlled to prevent the possibility of liquid water impinging upon hot turbine parts. A carefully controlled water flow, using turbine operating parameters as control functions, can produce efficient cleaning with minimal or no damage to the machine, and with a very short period of reduced power output. A graph showing the effect of cleaning on power output [40] is shown in Figure 24.

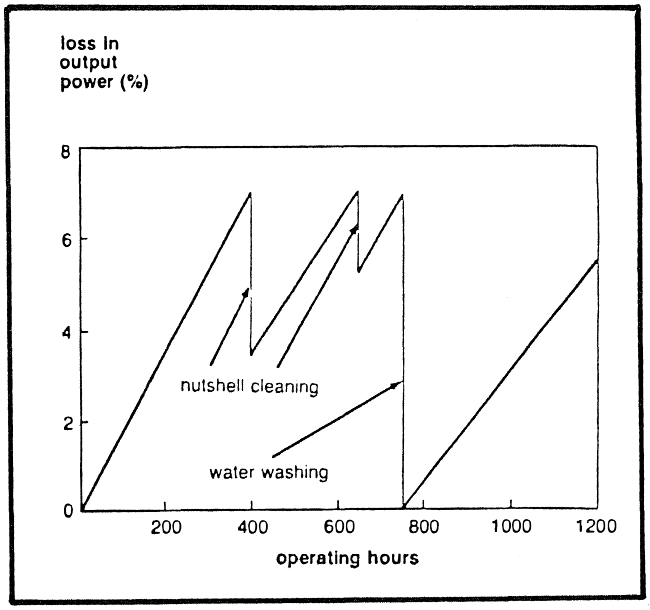


Figure 24. Effects of Compressor Washing [40].

Fuel System Considerations

The construction of the fuel system and of the actual oil or fuel used by a gas turbine must be investigated and considered from a reliability operating standpoint during the design phase of the project.

It is commonly known that the movement of combustible fuels by barge is always accompanied by the ingress of some water into the fuel. One gallon of sea water in one million gallons of oil is sufficient to render that fuel unfit for combustion in gas turbines without the use of a fuel cleanup system. The presence of sodium (both airborne and within the fuel) and vanadium (within the fuel) has a very corrosive effect on hot section life. Most turbine manufacturers insist on levels at or less than one part per million (1 ppm).

Water will generally settle and carry salt out of the oil during storage periods. If pure water is added to the fuel, it will generally be more salty when recovered. This means that the water concentration near the bottom of the storage tank will always be higher than the water concentration near the upper surface of the liquid. It is important that a floating suction system be used for delivering gas turbine fuel. Floating suctions should be installed in these tanks which would preclude the pickup of moisture settled at the bottom of the tank.

Depending on the water binding characteristics of the particular fuel involved, several methods of filtration are available. Paper filters will under no condition render sufficient fuel cleanliness to preclude the ingress of water into the turbine. These filters have been found to remove water, provided the water is not bound in emulsion within the fuel. If the water is bound within emulsion, it is necessary to inject a demulsifier and break the emulsion, either by centrifuge or by an electrostatic precipitation.

Electrostatic precipitation involves the addition of between five to ten percent of pure water within a mixing system. The water and salt dissolved by this added water are then removed either by centrifuge or by electrostatic precipitation. Heating for viscosity reduction may be required.

Adequate fuel filtration and salt reduction can be achieved by these methods. Edge type filters such as the Cuno type and paper filters do not generally provide adequate protection against entrained water in fuel. The presence of vanadium in the liquid fuel is usually inhibited by the addition of magnesium sulphate (epsom salts). An excessive amount of this material causes turbine fouling and, consequently, more frequent turbine washing.

Fuel nozzles, which draw their fuel from a common manifold, are always more sensitive to plugging and poor distribution of their fuel spray patterns. Careful monitoring and control of the temperature spread in the exhaust gas must be maintained. Exhaust gas temperature spreads of more than 60°F, or sudden changes of 30°F above normal variations in a carefully monitored machine, should be grounds for concern. If high exhaust gas temperature spreads are detected, the machine should be shut down and the burner nozzles cleaned or inspected, to ensure that the uneven distribution of fuel does not occur. If this spread is not corrected very rapidly, combustor distress will occur.

In the case of a gas turbine fired by natural gas, the problem of sodium and vanadium is not present. However, care should be taken that no liquid slugs be present in the fuel. This can be achieved by adequate fuel preheating, the use of a knock-out drum and scrubbing. Excessive heating applied during the preheating can, however, cause problems with coking of condensibles.

Coatings

The superalloys currently used by gas turbine engine manufacturers for fabrication of blades and vanes for turbine hot sections are predominantly either nickel or cobalt-based, with compositions emphasizing high temperature mechanical properties. Their chemical formulations, therefore, preclude sufficient amounts of those elements, primarily chromium and aluminum, that could make these alloys inherently resistant to the chemical reactions encountered during engine service. Although the operating environment of turbine blades and vanes varies widely depending on engine design and application, the requirements for protective coatings are basically the same, namely, resistance to hot corrosion and oxidation for long intervals of engine operation.

A wide variety of high temperature coatings for new and engine run turbine components are available which differ substantially in complexity and cost, thereby enabling the user to obtain the desired protection for parts at reasonable expense. Currently, several types of compressor blade coatings are available which may (1) improve erosion resistance, (2) improve corrosion resistance, and (3) provide aerodynamically smooth blades, and, consequently, better compressor efficiency.

Health and Efficiency Monitoring

Health monitoring in a gas turbine based cogeneration system can go a long way in improving efficiency and availability. Health monitoring should include both aero-thermal performance monitoring and mechanical monitoring.

Aero-thermal performance monitoring would include the collection, correction and analysis of pressures, temperatures, flows and speeds throughout the cogeneration system. This enables a knowledge of key performance factors such as gas turbine component efficiencies (compressor, combustor and turbine) and peformance of the HRSG and steam turbine. Important health parameters such as the turbine inlet temperature and the air flowrate can be obtained by computational methods. Knowledge of component efficiencies allows the pinpointing of problem areas for fast maintenance and reduced downtime.

Mechanical analysis would include the surveying of vibration data, bearing metal temperatures, axial positions, etc. Real time spectrum analysis techniques can help in troubleshooting problems, once again reducing downtime. Automated diagnostics are also available. Health monitoring, if effectively applied and acted upon, can reduce maintenance downtime and help in reducing inefficient operation [41, 42].

INTERCONNECTION CONSIDERATIONS

The issue of interconnection with the utility is one which has several technical and institutional facets to it. It has traditionally been a problem area for cogeneration projects and it will take time before utilities and cogenerators become comfortable and understand each other. The underlying problem is that the cogenerator's main objective is to have an uninterrupted power supply while the utility, understandably, wants to isolate (drop) the cogenerator if there is a problem. Because of these problems, *good business and technical relationships should be established by the cogenerator with the utility at the earliest opportunity in the cogeneration project*. Important considerations are: the choice of synchronous or induction generators, circuit breaker design for minimal problems and safety, transformer connections, and the negotiations of the buy/sell contract with the utility. The topic of interconnection is described in detail in [43, 44].

A *synchronous generator* consists of a stator carrying a three-phase winding and a salient pole rotor which carries a direct current (DC) field. The rotor is driven at synchronous speed (3600 rpm for 60 Hz) by the turbine. A three-phase voltage is generated as the magnetic flux caused by the DC field excitation is cut by the stator windings. The frequency is controlled by the prime mover speed and the alternating current (AC) voltage is controlled by the variation of current in the DC field windings. Thus, synchronous generators require excellent driver speed control to ensure operation at the correct frequency. When interconnected, a magnetic coupling will hold the machine at synchronous speed. When interconnected to a utility, even a fraction of a cycle from rated frequency is unacceptable. Synchronous generators operate at a constant power factor with load. Typical large synchronous generators can have efficiency of about 98 percent.

An *induction generator* is identical to an induction motor except that it is driven by a prime mover above synchronous speed. The kilowatt output of an induction motor is about 0.746 times its design horsepower. The voltage and frequency is set by the interconnected utility. Induction generators are being increasingly used for cogeneration applications because of their low cost and simplicity. Induction generator efficiencies are size dependant and are at most 96 percent efficient. Interconnection costs for induction generators are lower than for synchronous generators because less protective relaying is required.

The desired *generation voltage* is a function of the size of the generator and the cost of existing, or installed, switchgear and transformers. Typically, 480 V is most economical up to 500 kVA, 4160 V up to 750 kVA and larger, and 13.8 kV for 2500 kVA systems upwards.

Circuit breakers are automatic switches designed to open (by protective relays) when a heavy current is flowing. Under the correct conditions, even a small cogenerator can affect a large utility operation and disrupt operation. Because a fault may be fed by either the utility, the cogenerator or both, it is important that proper coordination between circuit breaker relays exists. Careful synchronization is a must to prevent catastrophic situations such as sheared generator shafts. Several times a utility breaker may reclose after a transient problem in the utility network is resolved. This automatic closing can cause problems to the connected cogenerator. *Careful design in the setting and selection control relays for circuit breakers, with the utilities review and approval is the best way to ensure safe, trouble free operation*. Some interconnection problems that must be addressed are:

- Isolation of cogenerator's generator
- Reclosing of the cogenerator's breakers
- Hazards to linesmen (cogenerator feeding into a shut down line)
- Relay settings (where relay trip limits should be set)
- Grounding considerations

EMISSIONS CONSIDERATIONS

With the growing movement for environmental protection, the area of gas turbine emissions has become critical for cogeneration projects. Environmental Protection Agency (EPA) regulations and negotiations, relating to emissions and offsets by states in an area, are best handled by professionals with experience in this area. EPA standards are a function of turbine energy consumption and intended use. Rules and decisions regarding emissions are in a state of flux, which is why expert help in addressing this area is necessary.

The trend towards higher firing temperatures and the use of heavy fuels creates problems with respect to nitrogen oxide (NO_x) formation. There are two sources of NO_x—fuel bound NO_x and thermal NO_x. Fuel bound NO_x is created by the reaction of fuel bound nitrogen with combustion air (i.e., it is related to the fuel). Thermal NO_x is produced by the oxidation of atmospheric nitrogen at high firing temperatures.

NO_x Suppression Methods

Steam Injection—involves the injection of steam into the combustor to lower combustion temperatures and, hence, curtail NO_x formation. The output of the turbine is also increased by virtue of the increased mass flowrate. There is also a small increase in thermal efficiency. Steam injection is still being used extensively.

Water Injection—involves a similar principle, but can result in increased maintenance problems in the combustor. Also NO_x and carbon monoxide (CO) exist in an inverse manner. An increase in water injection would reduce NO_x, but *increase* CO.

Catalytic Converters—are capable of reducing 60 to 80 percent of the NO_x. This reduction is brought about by the reaction of NO_x with ammonium hydroxide to produce nitrogen and water in the presence of a catalyst.

Extensive research and development work is underway for dry combustor design, in which NO_x control is accomplished by attempting to inhibit the creation of NO_x. The control scheme includes partially burning the fuel and cooling air quenching the hot mixture within a reconfigured combustor.

COGENERATION UTILIZING GASIFICATION AND FLUIDIZED BEDS

Several programs are underway for the gasification of coal and heavy oils. The gasification process would result in low or medium Btu gas, which would then be utilized by a gas turbine based cogeneration system.

Fluidized beds have two basic advantages. First, sodium dioxide (SO_2) and NO_x emissions are controlled within the bed and, second, great fuel flexibility does exist. Fluidized bed combustion offers a method for the utilization of coal. In this case, the gas turbines would utilize external combustion. Compressed air from the compressor discharge of the gas turbine would be routed to a fluidized bed heat exchanger, after which it is returned to the turbine nozzle. Several atmospheric fluidized bed (AFB) systems are in commercial operation [45, 46, 47, 48, 49] and considerable work has been accomplished in the area of pressurized fluidized beds (PFB) [50, 51].

CLOSED CYCLE COGENERATION

Closed cycle gas turbines are essentially externally fired gas turbines which allow the use of any type of fuel including water, coal and coal derived fuels. Closed cycle gas turbines have been used for several years in Europe. A closed cycle cogeneration concept as described in [52] is shown in Figure 25. Cycles such as these provide the following advantages:

- Multi-feed capability
- Efficient fuel utilization
- High part load efficiency
- Suitability for small scale design

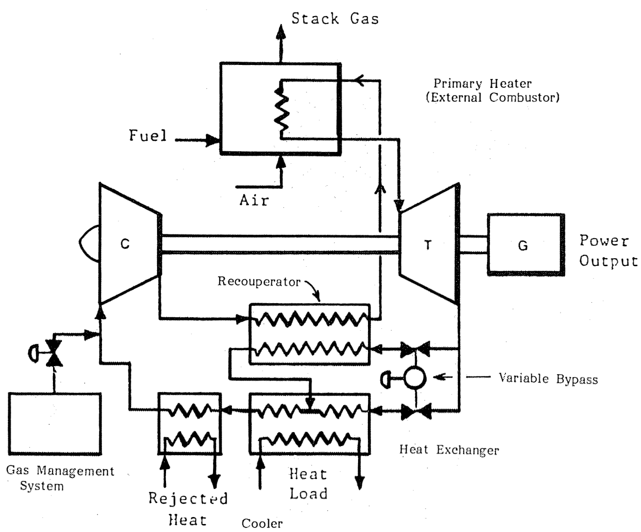


Figure 25. Variable Recuperator Bypass Cogeneration Configuration for the Closed Cycle Gas Turbine [52].

A discussion of closed cycle gas turbines is also made by Sawyer [53]. In the closed cycle concept, exhaust gas *has* to be cooled (so it can be returned to the compressor inlet), and this is done by steam generation. Thus, the closed cycle and cogeneration concepts intermesh perfectly.

Several of the larger closed cycle plants are located in Germany. These include:

Haus Aden Plant—6.4 MW_e and 8 MW_t (since 1983). (subscripts e and t stand for electrical and thermal).

Courburg Plant—6.6 MW_e and 16 MW_t (25 years operation).

Oberhausen 1 Plant—13.7 MW_e and 28 MW_t. This plant was originally designed for pulverized coal but is now operated on natural gas.

Oberhausen 2 Plant—50 MW_e and 53 MW_t thermal output. This plant operates on oil and medium Btu coke (started up in 1975) and uses helium as its working fluid.

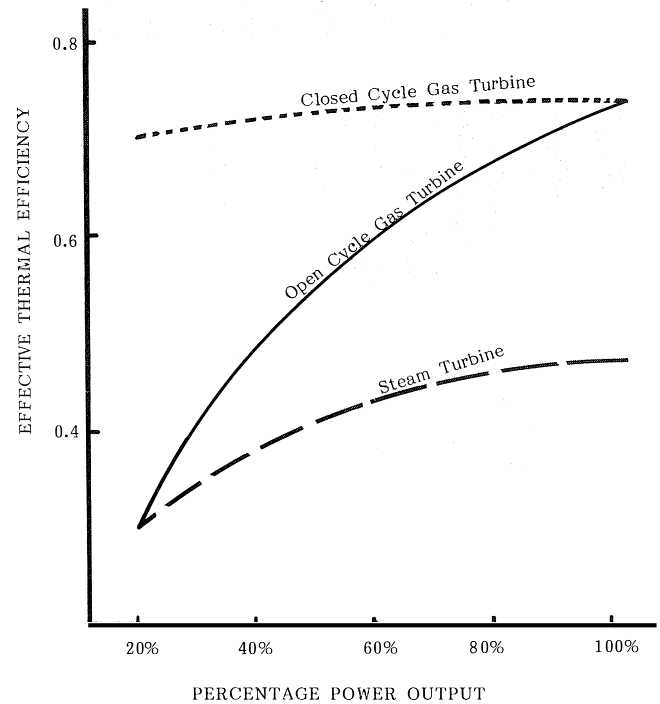
Performance characteristics of some candidate cogeneration cycles showing that the closed cycle has the best part load efficiency are shown in Figure 26 [54]. Further sources on closed cycle concepts are available in [55, 56, 57, 58].

With an important emphasis on coal derived fuels, prospects for using closed cycle gas turbines in combined cycle modes with atmospheric fluidized beds (AFB) and pressurized fluidized beds (PFB) looks very positive [59].

According to several industry experts, closed cycle gas turbines will play a very important role in future cogeneration technology.

Dual Fluid Cycle (DFC) Cogeneration

When steam demands from a cogeneration system drop, the overall efficiency can drop from 70 percent to 80 percent down to 20 percent to 30 percent. This is because the turbine has to be operated at reduced load, or exhaust gas has to be vented. The dual fluid cycle has the ability to accommodate fluctuations in thermal load. The concept utilizes superheated steam injection into the gas turbine combustor [60]. The concept was designed utilizing an Allison 501-KC turbine, but is also applicable to other turbines [61, 62, 63].



All Systems at 0.66 Power/Heat Ratio

Gas Turbine : 80°F Inlet; TIT = 1550°F

Pr. Ratio = 7.8:1; η_t = 0.9; η_c = 0.86

Steam Turbine: 600 psia, 800°F; 2-inch Hg condensing; Single extraction; Process steam @ 190 psia.

Figure 26. Comparison of Part Load Performance for Three Cogeneration Cycles [54].

There are several advanced cogeneration cycles utilizing a wide variety of technologies. These include technologies utilizing solid waste fuel (refuse derived fuel (RDF)). Approximately 20 plants are in operation in the United States today [64]. Japan is the leader in the application of this technology. Cogeneration technologies also exist utilizing solar energy. In addition, there is work being done in the utilization of fuel cell technology [65].

CONCLUSION

The concept of cogeneration has taken a firm hold in the U.S.A. The inhereht flexibility and efficiency of cogeneration systems will create a great demand for cogeneration systems. There are several regulatory and environmental hurdles that will have to be overcome. These will, however, be overcome as the demand and technology of cogeneration grows.

APPENDIX

Design Review Checklist [24]

Design/Manufacturing Related

What are the desired functions of the system under review? (Thermodynamic performance, reliance and maintainability).

- Does the design meet all the desired functions?
- In what areas will it perform better than required?
- Will complexity or cost be reduced if these areas are eliminated? Will this effect reliability?
- What alternative methods of achieving the desired function were considered before adopting the present design? This includes system and component redundancy – Gas Turbines & HRSG, Pumps, etc.
- What does this device have in common with existing successful designs? Are concepts prototype in nature – e.g. unusual fuel type being used?
- Has the operational environment been defined in terms of temperature extremes, humidity, shock and vibration, water quality?
- What standards have been used (API, NEMA, ASME, etc.)?
- Have adequate anti-icing measures been taken for the Gas Turbine?
- Is air filteration adequate?
- What fuel treatment equipment is provided?
- Prepare a list of major or vital components and identify the materials used for each (Turbine, HRSG, Steam Turbines).
- What three *material* properties were most important in the selection of this material for this application? Consider fuel type, temperature, service, coatings, etc.
  a. What fuel capabilities does the GT have? (Dual Fuel)
  b. If liquid fuel, is there a floating suction in bank?
  c. Is fuel inlet pipe at least 18 in from bottom?
- What novel or unique manufacturing methods are required to achieve this design?
- What testing techniques are used?
  a. What acceptance tests have been decided upon?
  b. What overspeed protection is provided?
- If welding is used, what welding and weld repair procedures are used? (Blade and Nozzle repairs)
- Have arrangements been made for project supervision at OEM facilities during package construction?
- Is foundation design sound? Any possibility of foundation resonances?
- At what percentage of its critical speed does each shaft rotate? Is there a sufficient margin from operating speed?
- What provisions are made for balancing?
- What thermal transients have been considered that might cause shaft bending?
- Has adequate turning gear equipment been provided?
- Identify any bearings in the device. Analyze bearing type loading, effects of misalignment, lubrication.
- What is there to prevent debris, dust, grit, etc., from entering the bearing or lube oil system?
- What is the effect of such dust, grit, etc?
- What self-aligning features are present in radial bearings?
- What provisions are made to prevent brinelling of anti-friction bearing during shipping?
- Have adequate rotor dynamic studies been conducted to ensure no instabilities and problems? (stability analysis, unbalance response)
- Are lube and seal oil systems adquately redundant?
- Are thrust bearings adequate?
- Can lube filters be changed during operation?
- Identify any seals in the turbines, compressors, pumps, etc. – dampers, isolation valves, etc.
  a. What type of seal is this?
  b. What testing has been performed on this actual seal?
  c. What is the maximum permitted leak rate?
  d. What happens if this leak rate is exceeded?
  e. How is excessive leakage detected?
  f. What backup is provided for this seal?
- Identify any static seals in the system
  a. What type of seal is this?
  b. What happens if this seal leaks?
  c. What testing has been performed on this seal?
- How much does the application of the above seals represent an extrapolation of proven performance?
- Can exhaust stacks withstand high wind velocities?
- Are prime movers in systems adequate (lube oil, pump motors, auxiliary turbines, etc.)?
- Are couplings adequate?
- Are water chemical treatments for steam system adequate?
- Identify any components carrying large pressure differentials (steam drums, etc.).
  a. What code requirements apply to this component?
- What documents identify compliance with these code requirements?
- For what emergency conditions have the pressure vessels been designed?
- To what hydro-test conditions will the pressure be subject?
- For what areas has a tolerance stackup study been performed?
- What essential clearances must be maintained in all machines?
  a. How are these clearances maintained?
- Identify all critical screwed or bolted joints and locking methods.
- In what ways is axial thrust from rotating parts resisted?
- What interchangeability provisions are incorporated in the design?
- What provisions are there to drain all fluids from the device?
- What standard parts does the device employ?
- Where can "non-standard" parts be replaced by standard parts?
- Identify any areas subject to wear (sliding, rubbing, rolling, impact surface, corrosion, erosion, fouling).
  a. How might this wear be avoided?
  b. How might this wear be reduced?
  c. What is the effect of the wear?
  d. Can this be tolerated?
  e. How is compressor and train washing accomplished?
- What indications are there in the operation of the device to alert the user to wear taking place? e.g., what will be used as a fouling indicator?

Reliability/Maintainability Aspects

- Has mean life been determined?
- Have adequate derating factors been established and adhered to where appropriate?
- Has equipment design complexity been minimized?
- Is protection against secondary failures (resulting from primary failures) incorporated where possible?
- Has the use of adjustable components been minimized?
- Do the connectors incorporate provisions for moisture prevention?
- Identify any moving surfaces exposed to contact by persons close to the device.
- Identify any large moving masses. What provisions are made to retain pieces if these masses should fracture?
- What provisions are made for grounding and shielding electrical circuits?
- Are noise levels acceptable?
- For how long will the device be out of service for each of the above oprations?
- What special tools are required for maintenance?
- Are the types and quantity of spare/repair parts recommended by an OEM appropriate for the estimated demand? Too many or too few spares can be costly.
- Have spare rotors been considered?
- Are cables fabricated in removable sections?
- Are cables routed to avoid sharp bends?
- What are PM schedules?
- What are the critical components?
- How does maintenance affect plant availability?
- What other components require periodic attention and what are the intervals?
- How might the design be changed to avoid upsetting critical clearances?
- Has a lubrication management program been defined?
- Has an inspection schedule been developed for auxiliaries-pumps, exciter brushes, water levels, oil levels, hose connections?
- Has an FMECA on the overall cogeneration system been performed?
- Has the system/equipment wearout period been defined?
- Have failure modes and effects been identified?
- Are item failure rates known?
- Have parts with excessive failure rates been identified?

Accessibility

- Access available for liners, nozzles, first-stage nozzles?
- Are access doors provided where appropriate? Are hinged doors utilized?
- Are access openings adequate in size and located for the access required?
- Are access doors and openings labeled in terms of items that are accessible from within?
- Are adequate walkways, safety handrails, ladders provided for gas turbine and HRSG?

- Are access door fasteners minimized?
- Are access door fasteners of the quick-release variety?
- Can access be obtained without the use of tools?
- If tools are required to gain access, are the number of tools held to a minimum? Are the tools of the standard variety?
- For heavy items, are hoist lugs (lifting eyes) incorporated? Hoist lugs should be provided on all items weighing more than 150 pounds.
- Are hoist and base lifting points identified relative to lifting capacity? Are weight labels provided?
- How are components supported?
- What levelness or verticality requirements are there?
- What external connections are required?
- What is the smallest opening through which the device must pass for installation?
- What services are required; electrical, air, fluids?
- What are the weights of major components?
- What pull-space access is required?

Instrumentation and Controls

- What is measured? To what accuracy?
- What is not measured that could be?
- What reliability and quality control features does instrumentation and controls have?
- Boroscope parts for gas turbine provided?
- Adequate vibration sensors (proximity probes and accelerometers)?
- Are vibration sensors of appropriate temperature and frequency range?
- Are performance monitoring parts/system provided?
- Are automated diagnostics provided to enable quick troubleshooting?
- Thermocouple junction boxes sturdy?
- Are cables routed to avoid pinching?
- Is cable labeling and clamping adequately?
- Are connectors that are mounted on surfaces far enough apart so that they can be firmly grasped for connecting and disconnecting?
- Are connectors and receptables labeled?
- Are connectors standardized?

Documentation

- What has been provided by OEMs regarding maintenance support documentation?
- Have operating and maintenance procedure requirements been defined? Have the necessary procedures been prepared?
- Are operating and maintenance procedures adequate from the standpoint of presenting simple step-by-step instructions; including appropriate use of illustrations; and including tables for presenting data?
- Do the operating and maintenance procedures specify the correct test and support equipment spare/repair parts, transportations and handling equipment and facilities?
- Do the procedures include special warning notices in areas where safety is a concern?
- Are special training programs required?

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