SAWYER'S GAS TURBINE ENGINEERING HANDBOOK

Third Edition
Three Volumes

VOLUME II
SELECTION & APPLICATION

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SAWYER'S GAS TURBINE ENGINEERING HANDBOOK

Third Edition

Volume II of Three Volumes

I Theory & Design

II Selection & Application

III Accessories & Support

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CHAPTER 1

GAS TURBINES IN UTILITY POWER GENERATION

by Roy C. Chetty, Richard W. Foster-Pegg, Thomas J. Radkevich and Ronald J. Thoman

The world is populated with a diversification of people and their cultures; but no matter how large or how small a nation, there is one common denominator--electricity.

Throughout the world, electric generators are powered by a variety of prime movers. Although the gas turbine is a relative newcomer, it has carved out an important niche for itself in utility generation: emergency, reserve, peaking, intermediate and baseload duties.

The gas turbine's versatility has made it a utility workhorse. Requiring no water, it is a popular form of generation in the Middle East, where that commodity is so precious. Fuel flexible, it can burn natural gas, propane, distillate oil, naphtha, residual oil, and crude oil, to name but a few. As a simple cycle, it can be installed quickly at a low capital cost. In a combined cycle, it produces the highest efficiency of any conventional power plant.

The gas turbine is blessed with an open pathway to growth. This includes both technological advances and new applications. For example, gas turbine manufacturers are developing nonpolluting methods to burn coal either directly or indirectly. In this regard, investigations are in progress to adapt gas turbine combined cycles to coal gasification systems, to pressurized fluidized bed combustors (PFBC) and to atmospheric fluidized bed combustors (AFBC). Without a doubt the gas turbine has a bright future in utility application.

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CHAPTER I

1.0 HISTORICAL PERSPECTIVE

The first gas turbine for utility service was installed in 1938. This 4000 kw Brown Boveri unit provided standby power for the city of Neuchatel, Switzerland. The gas turbine incorporated an axial-flow compressor and an expander that had been successfully used together in catalytic cracking plants for oil refineries. A combustor system was added to provide a complete working gas turbine.

Jet engine advances, receiving a stimulus from World War II, provided spinoff to industrial gas turbines. With military superiority in the balance, large budgets were earmarked for jet engine research. This included the development of high-temperature alloys and the improvement of component efficiencies and reliabilities. The high level of military funding has continued to the present time. The industrial gas turbine has been a benefactor of this new know-how. Furthermore, a component supplier infrastructure has developed. That, associated with the manufacture of turbine blades and vanes, has been an especially important element in the transfer of technology to the industrial gas turbine.

Although the market steadily grew as the industrial gas turbine improved, it was not until the 1965 Northeast Blackout in the United States that the utility application of the gas turbine got its big boost. The power interruption awakened the utility industry to the small reserve margins throughout the United States. With electric consumption doubling every ten years, this was a critical deficiency. Gas turbines, both industrial and aviation-derived, became the means to quickly correct this problem.

No other power-producing plant had such short time from order placement to electric generation. The low capital cost of simple cycle gas turbine plants, furthermore, made sense for peak shaving. It was particularly fortunate that during this period quality fluid fuels, natural gas and distillate oil, were readily available and inexpensive.

Figure 1 shows the installed gas turbine electric power generating capacity in gigawatts for 1964 to 1980. This is divided into the total worldwide and the United States installed capacity. As a direct consequence of the Northeast Blackout, the United States had more than half of the total worldwide installed capacity between 1970 and 1975.

Although there had been some earlier experience with gas turbines in combined cycle plants, the higher turbine inlet temperatures of the early 1970s contributed to combined cycle efficiencies that significantly exceeded those for conventional steam plants. This efficiency advantage resulted in a large number of sales of combined cycle plants to the electric utility industry.

The 1973-1974 Arab oil embargo and the associated large price increases for oil contributed to the greatest marketplace dislocation since World War II. The United States utility market for gas turbines nearly disappeared overnight. Energy conservation measures resulted in a sudden generation overcapacity and led to the cancellation or delay of many new power plants on order.

As a result of these severe dislocations in the United States and a redistribution of wealth in the world, the market for gas turbines radically changed character. While the United States domestic market largely vanished, the worldwide market--particularly in the Middle East and other oil-producing countries--increased. With their new oil prosperity, these countries formulated development plans that required new electric generation capacity. With the unavailability of water in many of these countries, the gas turbine was a natural fit.

2.0 UTILITY PLANNING AND ECONOMICS

Power generation planning for utilities throughout the international community is a complex process. In addition to the difficulty in forecasting long range load demands, the planner must evaluate capital cost, fuel cost, efficiency, reliability, availability, siting, and construction lead times. There are situations, furthermore, where overriding constraints dictate equipment selection.

2.1 Load Duration Curve

Although the precise nature and mix of power generation equipment is peculiar to a given utility, the method used to select equipment is essentially the same. An important tool for effective utility planning is the load duration curve.

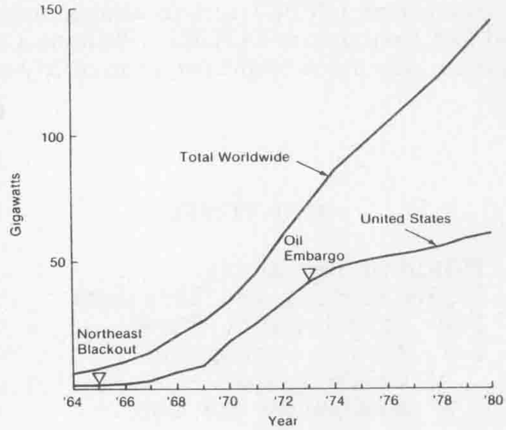

Figure 1. Installed Gas Turbine Electric Power Generating Capacity

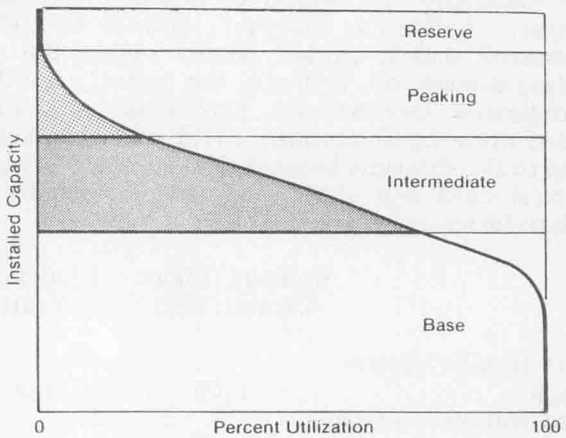

Figure 2. Typical Annual Load Duration Curve

Figure 2 shows a typical load duration curve. This curve indicates the percentage of operating time for each type of capacity: base, intermediate, or peak. Base capacity supplies the major portion of energy throughout the year, while peaking capacity is the source of little of the annual energy produced.

The greatly differing usage of the categories dictates the specific requirements for each generation facility. The fact that baseload units operate for such a high percentage of the time (typically 6,000 hours or more a year) makes low operating cost a major consideration. Since peaking units operate relatively little (generally less than 1,500 hours each year) their operating cost is less important in comparison to their capital cost.

The requirements for intermediate capacity are a compromise. The usage rate, typically 1,500 to 6,000 hours per year, is high enough so that economy of operation is a significant consideration, and yet high capital costs cannot be justified.

Historically, as overall power generation requirements increased, utility systems added only base capacity, moving the older units into intermediate duty and then into peaking duty. As long as new generations of baseload plants had substantially higher efficiencies than those replaced, this procedure was logical.

In recent years, however, the heat rate improvement of baseload equipment has slowed. In fact, requirements for flue gas desulfurization has often resulted in poorer heat rates. Now, new equipment often offers no or little more in operating economy than its predecessors.

A utility can seldom justify replacing existing plants to solely reduce operating costs. Also, many of the current large baseload units have long startup times that prohibit their use in applications requiring frequent cycling. This is particularly true of nuclear units. Today it frequently makes more sense to add new equipment specifically suited to intermediate and peaking duty.

Selection of unit size is a major consideration in utility planning. The optimum unit size depends on many factors. The principle of economy of scale seems to imply that units should be as large as possible. Since costs of construction and installation do not increase proportionally with size, such an approach theoretically will keep total cost per kilowatt to a minimum. The need to maintain a reserve capacity to accommodate maintenance requirements, however, changes this picture considerably.

Reserve facilities can be looked at as a financial drain; they deplete capital while they generate no kilowatts and produce no revenue. Decreasing reserve requirements increases available funds. The required reserve capacity varies directly with the size of the individual units that comprise the total system capacity. The larger a unit, the greater the percentage of total system capacity it represents and the greater the impact its outage has. The capital cost of reserve capacity needed to support a relatively small number of large units is larger than it might otherwise be. A system constituting relatively small units permits a substantial savings by reducing the reserve requirement.

Furthermore, building system capacity with smaller units results in reduced excess capacity by closely matching the demand for incremental power. Gas turbines and combined cycle plant installations are an excellent way to add this smaller capacity. An alternate solution is joint ownership, where several utilities purchase shares of large, economical units.

In many countries, baseload needs are served by nuclear and fossil fueled steam plants; intermediate load by fossil fueled steam and combined cycle plants; and peak load by simple cycle gas turbines. The exact nature of the mix is influenced by the objective of minimizing the utilities annual owning and operating cost.

2.2 Annual Owning and Operating Cost

Annual owning and operating cost is comprised of three components:

- capital - annual revenue required to pay for interest, depreciation, insurance and taxes
- fuel - annual revenue required to pay for the fuel consumed
- operation and maintenance - annual revenue required to pay for labor force, maintenance parts and miscellaneous supplies

These components are related as follows:

Annual Owning And Operating Cost
$$(\$/KW/YR) = (CC)\ (FCR) + (FC)\ (HR)\ (8760)\ (CF)/10^6 + (O\&M)\ (8760)\ (CF)/10^3 \quad (1)$$

where,

CC	-	capital cost, \$/kw
FCR	-	fixed charge rate, %
FC	-	fuel cost, \$/10^6 Btu (\$/10^6 kJ)
HR	-	heat rate, Btu/kwh (kJ/kwh)
8760	-	hours per year
CF	-	capacity factor, %
O&M	-	operation and maintenance cost, mill/kwh

Substituting appropriate values into Eq. (1) yields a screening curve such as shown on Fig. 3. The intersections of the lines define the limits for the most economical form of generation for a given capacity range. For the hypothetical case shown, the most economical type generation equipment is as follows:

Cap. Factor %	Type Generation Equipment
0 to 20	Simple cycle gas turbine (gas fueled)
20 to 38	Combined cycles (gas fueled)
38 to 58	Fossil fueled steam (coal fueled)
58 and above	Nuclear

As shown in this example, coal gasification combined cycle would be a viable alternative for both intermediate and baseload application when it becomes commercially available.

Annual owning and operating cost is just one factor in the decision making process. Consideration must be given to many other variables, such as:

- availability and cost of capital
- availability and cost of fuel
- required installation time
- existing system mix
- existing system size
- system reliability
- system interconnections
- regulatory constraints
- siting constraints
- size (rating) of addition
- water requirements

2.3 Availability and Cost of Capital

Quite often the utility planner is confronted with diametrically opposed requirements for generation additions. If, for example, it is determined that there is a need for an increase in baseload generation, a pure economic evaluation may dictate the installation of nuclear or fossil fueled steam plants. The utility, however, may be so highly leveraged that it cannot readily obtain the required capital, or, if it can, the cost of capital is prohibitive. Gas turbines, particularly in a combined cycle configuration, could provide a solution to this dilemma because of their relatively low capital cost and short lead time compared to other forms of generation:

	Relative \$/kw Capital Cost	Lead Time Years
Gas turbine simple cycle	1.00	1-2
Gas turbine combined cycle	2.00	3-4
Coal gas combined cycle	4.00	4-5
Coal fueled steam	4.00	5-7
Oil fueled steam	3.25	4-6
Nuclear	5.50	8-12

In addition to the lower initial capital cost, a combined cycle requires less overall capital. Because of the shorter lead time, less monies are expended for escalation and interest during construction.

2.4 Availability and Cost of Fuel

When evaluating power generation alternatives for baseload application, fuel becomes a critical parameter since it usually represents the largest expense incurred by a utility. Coal or nuclear fueled power plants are often the most economical at high capacity factors as a result of their relatively low fuel costs. There are many areas or countries, however, where coal or nuclear fuel is either unavailable or not practical. For example, countries with large available reserves of natural gas may find that the use of that fuel is preferred.

For those utilities that are committed to a given fuel, analysis of annual owning and operating cost may dictate alternatives not otherwise considered. The screening curve shown on Fig. 4, for instance, indicates that for natural gas applications, gas turbine combined cycles are the superior choice for baseload application.

2.5 Other Considerations

As discussed, the decision to buy a particular type of power generation equipment is based on many considerations. For example, there are instances where utilities require power in a relatively short time frame. Because steam and nuclear plants cannot meet this criterion, an alternative is a simple cycle gas turbine--with possible conversion to a combined cycle at a later date.

Siting is another constraint. For example, if the required transmission and distribution system is

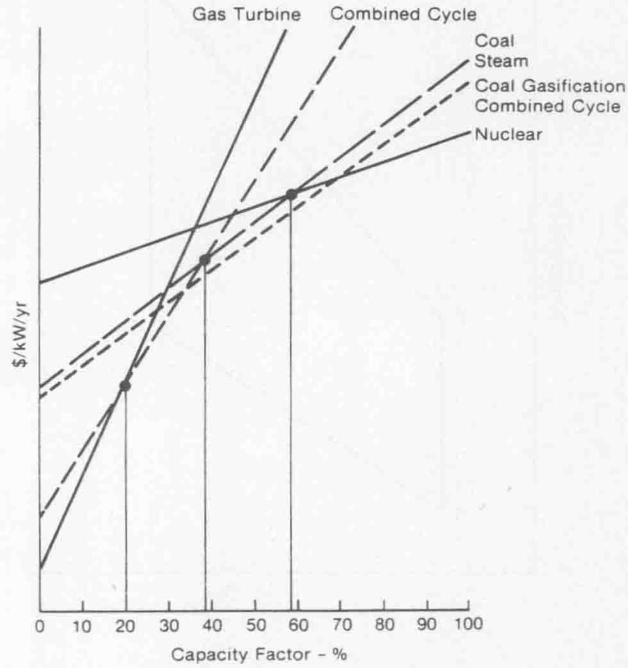

Figure 3. Annual Owning and Operating Cost

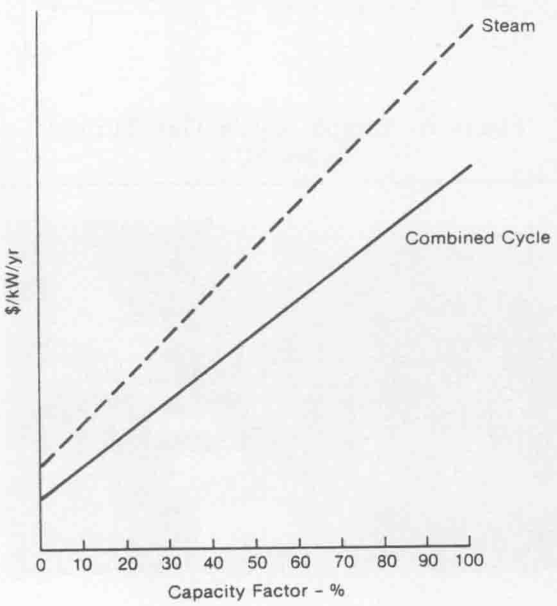

Figure 4. Annual Owning and Operating Cost -- Gas Fuel

not available, the viability of a large centralized power plant is negated. As another example, the substantial water supply needed for a large steam plant may not be available. Both of these constraints can be met with the use of gas turbines since they can be placed at the source of power demand and have low water requirements.

3.0 TODAY'S ELECTRIC GENERATING PLANTS

Around the world the gas turbine has gained acceptance for use in electric generating plants. In either simple cycle or combined cycle, it is used for peaking, intermediate, or baseload duty. Repowering is a special application where gas turbines are added to upgrade an existing steam plant thereby converting it into a combined cycle.

3.1 Simple Cycle Plants

The simple cycle gas turbine system is a complete power plant containing all the functional elements of a conventional steam power plant:

- The compressor is similar to the boiler feed pump.
- The combustor is similar to the boiler.
- The turbine is similar to the steam turbine.
- The atmosphere is an energy sink similar to the cooling water.

3.1.1 Thermal Cycle Description

The ideal prototype of the gas turbine thermodynamic process is the Joules-Brayton cycle shown in Fig. 5. Air is taken from the atmosphere at point 1 and is compressed isentropically to point 2. Along 2-3 heat is added by burning fuel in a combustor. Then the hot gases expand through the turbine and exhaust out the stack at point 4. The total heat added to the cycle is proportional to the area under line 2-3 on the T-S diagram. The total heat rejected is proportional to the area under the line 1-4. The net work from the cycle is the difference.

Figure 6 schematically represents a simple cycle gas turbine. This is an open cycle with the compressor, turbine and load on one shaft. The air entering the compressor is typically compressed to 12-14 atmospheres. In the combustor, the released fuel energy increases the gas temperature typically to 2000°F (1093°C) or above. The hot gases expand in the turbine to produce useful work.

3.1.2 Equipment Description

The simple cycle power plants, designed to be suitable for electric utility application, have the advantage of low initial cost, rapid installation, fuel flexibility, and no water consumption (unless wet NO_x control is required).

The gas turbine engine is divided into two principal types: aircraft-derived and industrial. In the first type, aircraft jet engines are modified to drive ground-based electric generators. In the second type, the engine is specifically designed to be rugged and to lend itself to utility maintenance and operation requirements. For example, casings are generally horizontally split to allow field access and repair. Normally, the industrial engine capability is larger, with a rating near 100 Mw common.

All auxiliary equipment is supplied to make a complete power plant. This includes the electric generator, exciter, power electrics, starting equipment, inlet and exhaust ducts, lubrication system, fuel system, and controls.

3.1.3 Application Example - Saudi Arabia

Generally since no external source of cooling or make-up water is required, the simple cycle power is desirable for arid regions such as Saudi Arabia.

For example, Fig. 7 shows the SCECO Shedgum site with nine Westinghouse W501D gas turbines rated near 90 Mw each. These machines are located outdoors in a remote area of the desert. Large self-cleaning air filters allow the gas turbines to operate unaffected by sand storms.

These units were initially designed to burn low sulfur natural gas; however, due to a delay in the gas gathering and stripping facilities, they operated on unstripped gas for several years. H_2S content of up to 15% was common. The addition of gas heating to above 200°F (93°C) and knock-out drums eliminated most liquids. Special gas nozzles reduced the detrimental effects of any liquid carryover to the combustion system.

This is only one of many gas turbine power plant sites in Saudi Arabia. The total gas turbine installed capacity is about 15,000 Mw. The primary fuel for newer installations is crude oil. The once plentiful waste gas is being liquefied for export. While new installations operate on unprocessed crude oil, they are fully capable of gas fuel operation if economics dictate.

3.2 Combined Cycle Plants

The combined cycle operates with high efficiency because exhaust heat from the gas turbines is recovered to make steam for a steam turbine bottoming cycle. The design requirements of this steam cycle are much different than those for a conventional steam power plant. Awareness of these differences leads to a plant that meets utility requirements.

3.2.1 Thermal Cycle Description

Figure 8 shows a stylized T-S diagram of combined gas turbine and steam turbine cycle. As can be seen, the energy sink for the simple cycle gas turbine process (Joules-Brayton cycle) is the

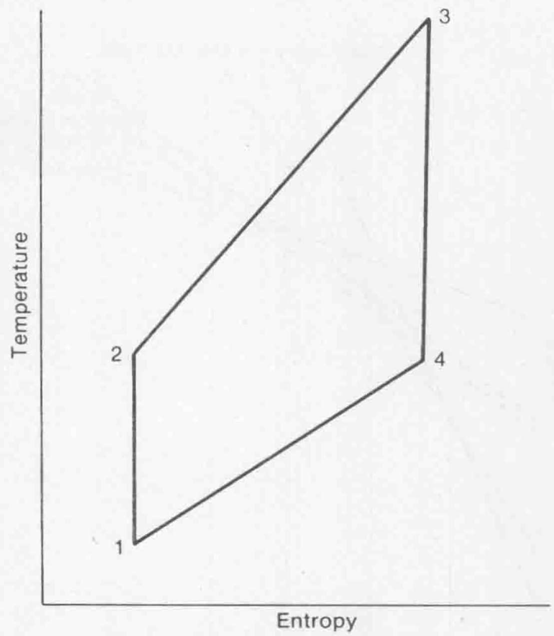

Figure 5. Ideal Joules-Brayton Cycle for Gas Turbines

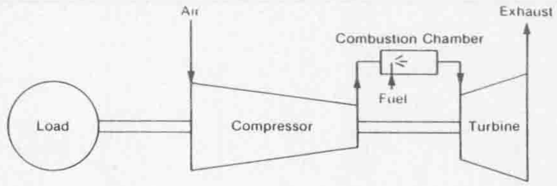

Figure 6. Simple Cycle Gas Turbine Schematic

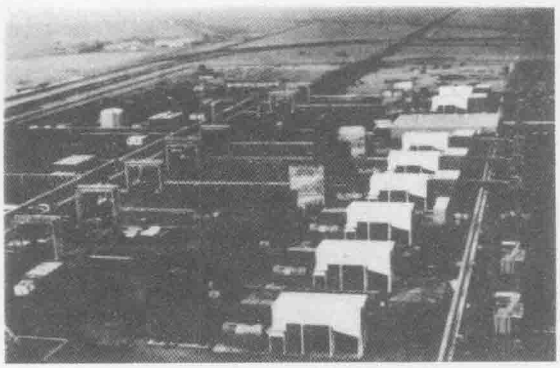

Figure 7. SCECO Shedum Installation in Saudi Arabia

heat source for the steam plant process (Rankine cycle). In this configuration, the steam cycle is often called the bottoming cycle since its maximum operating temperature is lower than the gas turbine cycle. Combined cycles of this type have higher efficiencies than either a separate gas turbine or a steam turbine power plant.

3.2.2 Plant Design

The Westinghouse PACE (Power At Combined Efficiencies) combined cycle plant is typical of those provided to the electric utility industry. Figure 9 shows the cycle schematic. Two gas turbines exhaust into heat recovery steam generators (HRSG). Approximately 80% of the fuel is used in the gas turbines, while 20% is used in the supplementary burners to raise the gas turbine exhaust typically from 950 to 1200°F (510 to 649°C).

The plant design allows for operating flexibility. Each gas turbine and HRSG combination can operate independently of the other. A full capacity steam bypass is included around the steam turbine. If the steam turbine cannot be operated, either gas turbine can operate with the steam going directly to the condenser.

3.2.3 Steam Turbine

Special attention must be given to the steam turbine design requirements:

- Floating pressure operation
- Low pressure end moisture
- Startup
- Load changes

Floating Pressure Operation - The throttle pressure for a conventional steam plant is usually held constant at all loads. To accomplish this, pressure is regulated by sequencing multiple control valves. With a conventional boiler, steam temperature is then maintained by adjusting the firing rate or gas flow through the boiler. In a combined cycle, however, floating pressure operation increases the part-load operating efficiency. It also alleviates low pressure end moisture.

Low Pressure End Moisture - To minimize low pressure end moisture, the steam turbine throttle pressure is allowed to decrease as steam flow is lowered. This coincides with superheat steam temperature reduction. If the combined cycle uses duct burners, steam temperature can also be controlled by modulating their firing rate; however, part-load efficiency may suffer.

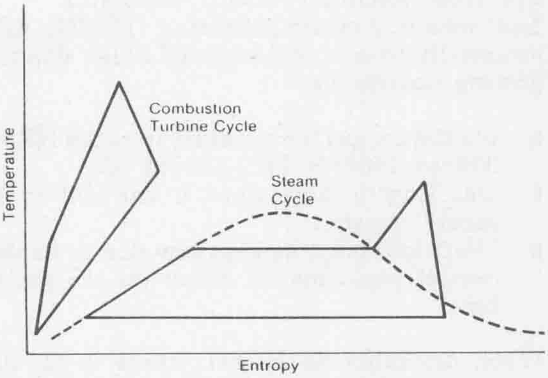

Figure 8. Stylized T-S Diagram for Combined Gas Turbine and Steam Turbine Cycle

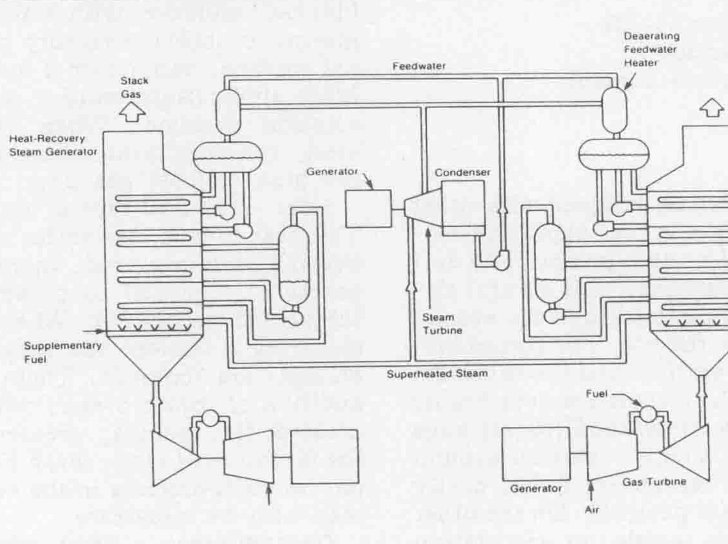

Figure 9. Typical Combined Cycle Schematic

Startup - Startup usually imposes the most severe stress conditions on the steam turbine, with the rotor bore being the limiting factor. Nevertheless, rapid startup is desirable to provide for quick response to load demand and to minimize fuel use. A judicious selection of materials is required to satisfy the high temperature creep limitations and the brittle fracture considerations at low temperatures. If rapid startup is necessary after extended shutdown periods, it is desirable to keep the steam turbine heated by external means.

Load Changes - For intermediate load duty, the plant must respond to frequent load changes. If the conditions for startup are satisfied, the load change ability is usually inherent in the unit. Load change is often accompanied by throttle steam temperature change, especially in an unfired cycle.

3.2.4 Heat Recovery Steam Generator

A heat recovery steam generator (HRSG) differs substantially from a conventional boiler due to the following conditions:

- Maximum gas temperature into the HRSG is 900 to 1400°F (482 to 760°C).
- Gas flow is high since it has 200 to 300% excess oxygen.
- Draft loss must be kept low due to its detrimental performance effect on the gas turbine.

When designing an HRSG, there is no single operating condition that is the most severe. All operating conditions need to be considered to properly size each of the components. Design areas of most concern include the following:

- Circulation
- Drum sizing and cyclic life
- Heat exchanger surface
- Superheat temperature control
- Insulation
- Fuel
- Duct burners

Circulation - HRSGs may be designed with either natural or forced circulation in the evaporator sections. The choice of circulation type normally dictates the boiler tube arrangement. For natural circulation, the gas flow is horizontal and the evaporator tubes are oriented vertically. For forced circulation, the gas flow is vertical and the tubes are oriented horizontally. Each circulation type has its merits. Forced circulation allows the HRSG to have a lower first cost and to occupy a smaller ground area. The field erection, moreover, is less costly since more shop assembly is possible. On the other hand, natural circulation needs no circulation pumps. This contributes to higher reliability and less auxiliary power consumption.

Drum Sizing and Cyclic Life - The primary consideration for sizing a conventional steam plant boiler drum is usually steam purity at full pressure and the maximum steaming rate. In contrast, the main criteria for sizing the HRSG steam drum are adequate capacity for swell during startup, steam purity at a high steaming rate and reduced pressure, and cyclic life. The thermal stresses due to transient induced temperature gradients are most severe in the drum. They must be analyzed to ensure sufficient fatigue life for the expected cyclic duty. It is important to impose no restrictions on the gas turbine operation.

Heat Exchanger Surface - Due to the large volume flow of low temperature gas and the close approach temperatures required, extended surface tubes are used throughout the HRSG. Both serrated- and solid-fin tubes have been used. Close approach temperatures for high efficiency require large surface areas. Figure 10 shows how the required superheater and evaporator surfaces vary with the superheater approach temperature for the same duty.

Superheat Temperature Control - Superheat temperature control is required to provide a wide range of operation, quick plant startup, and reduced moisture in the low pressure end of the steam turbine. Two common types of control are often employed: steam bypass of saturated steam around the superheater and water spray attemperation in the middle or at the outlet of the superheater. Figure 11 shows a superheater bypass arrangement. This system avoids possible water carryover from water sprays. For sufficient control latitude, the superheater must be oversized by about 20%.

Insulation - Three types of insulation systems have been used for the HRSG: internal block or blanket insulation with a thin sheet metal lining, internal castable refractory with an erosion resistant surface, and external insulation with lagging. Most applications employ one of the internal insulation systems. When external insulation is used, the structural casing must be designed for the high exhaust gas path temperatures.

Fuel - The fuel type is important to the design. The best fuel is low sulfur natural gas. If sulfur bearing fuels are used, increased feedwater temperature is needed to prevent acid corrosion at the exit of the HRSG. When heavy oils are used, requiring treatment for vanadium, HRSG design changes are required. These changes include the addition of soot blowers, reduced fin heights, increased fin spacing, greater tube spacing, and the inclusion of water wash provisions. Depending on the contaminants in the fuel, material changes may also be necessary.

Duct Burners - Duct burners provide operational flexibility, better part-load efficiency, improved steam conditions, and additional steam

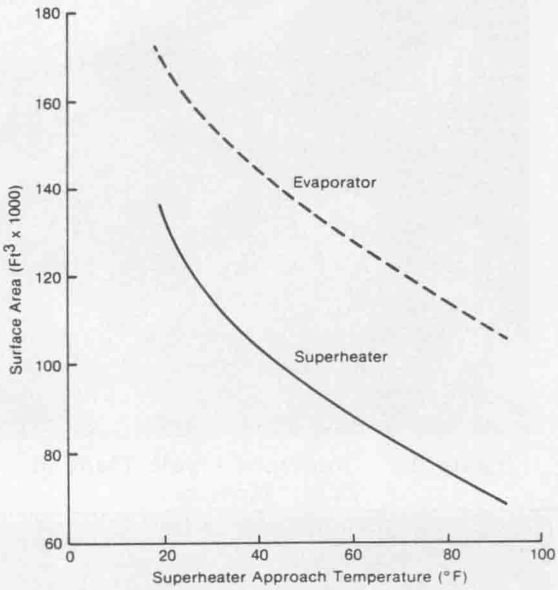

Figure 10. Heat Transfer Area Versus Superheater Approach Temperature

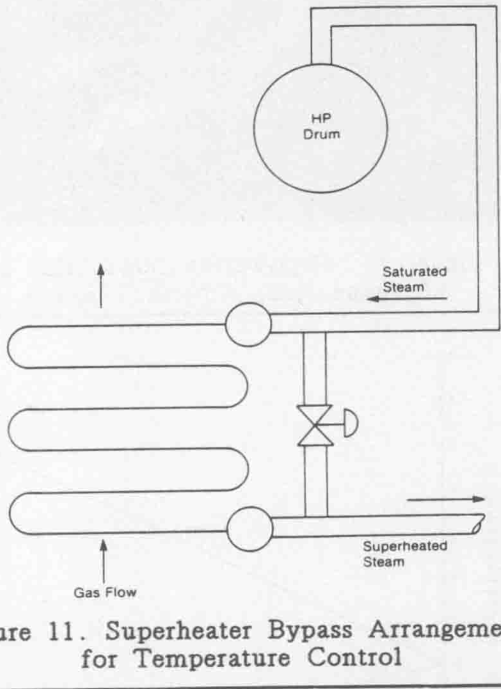

Figure 11. Superheater Bypass Arrangement for Temperature Control

turbine power. A demanding consideration for incorporating a duct burner is supplying it with a uniform exhaust gas flow. The flow leaving the gas turbine is highly turbulent. It is advisable, therefore, to incorporate a flow straightener or a long length of ducting between the gas turbine and duct burner.

3.2.5 Application Example - Mexico

The combined cycle power plant has earned acceptance not only in the United States but also in many other countries. Experience in Mexico is a typical example of the electric utility use of the new high efficiency combined cycles.

In the early seventies, Comision Federal De Electricidad purchased three large combined cycle plants designed by Westinghouse for location at two sites. Both sites chose to put the turbomachinery in large buildings. These sites, Dos Bocas and Gomez Palacio, have operated successfully for many years burning natural gas or high sulfur No. 2 fuel oil.

Due to the success of these early combined cycle plants, additional plants were purchased in 1981. Two Westinghouse plants are now being installed at the Tula site near Mexico City. The plants are designed for sequential installation; that is, the gas turbines are constructed and operated as simple cycles while the steam plant is erected. The plants are located at an elevation of 6900 ft (2100 m). This altitude results in gas turbines producing only 75% of the power they would produce at sea level. The duct burners are designed with sufficient firing so that the steam turbine power is not reduced from normal sea level conditions.

Combined cycle plant efficiency has improved substantially over the decade. Tabulated below is baseload performance comparison for typical plants consisting of two gas turbines and one steam turbine. Performance is given for ISO conditions, No. 2 fuel oil, and 2-1/2 in. HgA (8.4kPaA) steam exhaust pressure.

	1971	1981
Plant net power, kw	248,000	302,000
Plant net thermal efficiency, % (HHV)	39.1	42.9

Figure 12 shows essentially how the Tula site will look after completion. The gas turbines are designed to operate on both natural gas and No. 2 fuel oil. The duct burners will fire only natural gas.

All equipment is controlled from a central control room. Control hardware is located in the electrical skids next to gas turbines, HRSGs, and steam turbines. All necessary signals are multiplexed and transmitted to the central control room. Each gas turbine can also be controlled from the local panel in the skid. A data acquisition system with first-out indication, sequence of events, and 30 minute retention of all measured values aids disturbance diagnosis. CRT displays provide the operator with information through the display of mimic diagrams. The touch sensitive displays speed data retrieval.

3.3 Repowering

Repowering is defined as the integration of gas turbine capacity into an existing steam plant. The advantages of repowering are:

- Fuel savings due to improved cycle efficiency
- Added capacity at low capital cost
- Short installation period

3.3.1 Application Example - Medicine Hat

The repowering project for the Medicine Hat Municipal Electric in Alberta, Canada is a typical example of repowering. The city of Medicine Hat had a power generating station consisting of four steam turbines rated at 32 Mw, 15 Mw, 6 Mw, and 3 Mw and of four conventional gas fired boilers. The boilers were aging and additional capacity was required. After an economic evaluation of various types of generating equipment, the decision was made to repower the existing steam units by installing two gas turbines fitted with supplementally fired HRSGs. The gas turbines added a total of 67 Mw generating capacity. Since this plant operates essentially isolated, the following system requirements were imposed:

- No single failure could reduce the total power output by more than 33 Mw.
- Bypass stacks were required so that all units could operate independently.
- Since the load varies substantially every day, high efficiency had to be maintained over the load range of 20 to 100%.

Figure 13 is a photograph of the plant. One gas turbine produces about 33 Mw; the steam produced from its waste heat produces about another 16 Mw. Duct burners permit full steam power with only one gas turbine operating.

Maximum power was required to meet summer peak. Gas turbine power, however, decreases as compressor inlet temperature increases. To compensate for this, the gas turbine combustor shell is fitted with ports for steam injection. When high power is needed, excess steam produced by firing afterburners, or by a standby boiler, is injected into the gas turbine. Figure 14 shows the effect of steam injection on simple cycle power and heat rate.

Modulating exhaust gas bypass dampers are installed to allow simple cycle operation if an HRSG requires maintenance. Pressurized air between the double-louvered dampers is applied to minimize leakage and maintain maximum cycle efficiency.

To improve the part load efficiency, variable inlet guide vanes (IGVs) are installed. The use of the modulating IGV improves the part-load heat rate up to 10%.

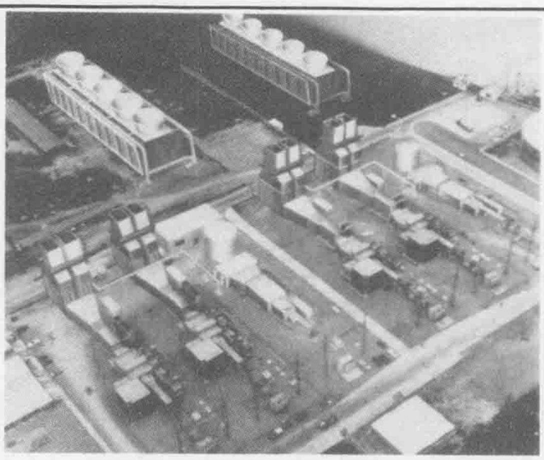

Figure 12. Combined Cycle Plant at Tula, Mexico

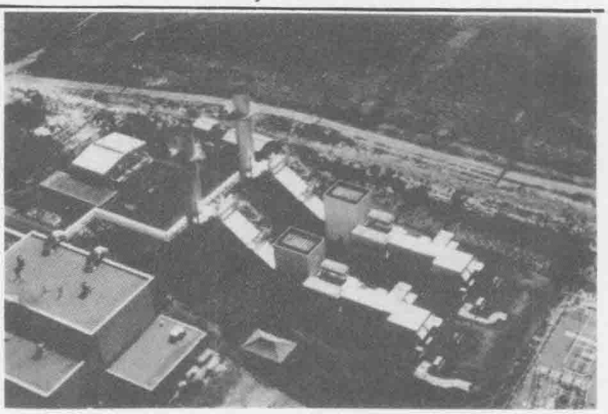

Figure 13. Repowering Installation at Medicine Hat, Alberta, Canada

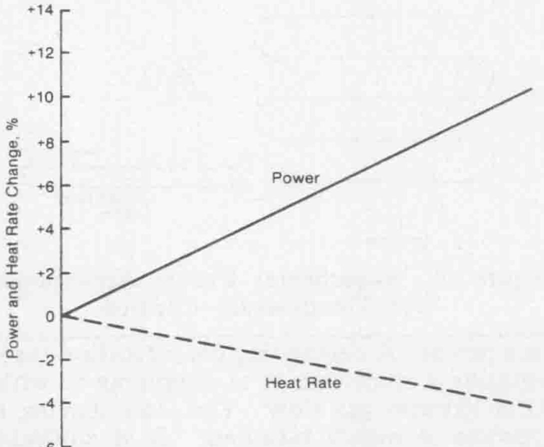

Figure 14. Effect of Steam Injection on Power and Heat Rate

4.0 ADVANCED COMBINED CYCLE PLANTS

As the worldwide supplies of fossil fuels are depleted, their value is expected to escalate at a rate exceeding that of general inflation. This increases the fuel component of power cost relative to the capital component, and elevates the importance of power plant efficiency. A practical means of achieving a significant improvement in the efficiency of power generation from fossil fuels is the adoption of advanced combined cycle plants.

Such plants can be divided into two main categories: those in which the combustion of clean fuels takes place within the gas turbine combustion system and those in which there is a combustion of raw coal, or another dirty fuel, at some point within the cycle--not necessarily the gas turbine combustion system. This latter category includes cycles where the combustion products of the dirty fuel are expanded directly through the gas turbine and indirect heater cycles in which clean, hot air is the actual working fluid.

Advanced combined cycle plants depend on gas turbine progress and on the steam cycle configuration choice. No basic technology advances, however, are foreseen as necessary for the steam cycle.

4.1 Gas Turbine Progress

Since the introduction of industrial gas turbines, their maximum firing temperature has increased at an average rate of 30°F (17°C) per year. Firing temperature is expected to increase at a somewhat lesser rate as the performance benefits of higher temperatures decrease because of the increased cooling losses at the higher temperatures.

To support future higher firing temperatures, advances are required in turbine blade and vane cooling. Water cooling has been investigated [1], but heat removed in a water cooling circuit has little value to the steam bottoming cycle and constitutes a loss. On the other hand, heat removed by air cooling remains in the gas turbine flow path at a useful temperature and is valuable to the steam bottoming cycle. Thus, air is expected to remain the cooling medium. Practical advanced air cooled systems, involving such techniques as transpirational cooling, are being developed to allow significant increases in firing temperatures while maintaining metal temperatures within limits.

There has been some interest in compound gas turbines with intercooling and reheat [2]. Reheat is beneficial for combined cycle efficiency, but intercooling is a heat loss from the system and offers little efficiency benefit in combined cycles.

In cycles having raw coal combustion, gas turbine operating temperatures are expected to be limited to levels significantly below current temperatures, as will be discussed later. For the indirect heated system, development will concentrate on aerodynamic improvement. For gas turbines exposed to combustion products of dirty fuels, the main thrust will be to improve tolerance to deposition, erosion, and corrosion. For instance, an expander is more tolerant to deposition at a lower blade tip speed. Lower speed designs require more stages but include larger passages less affected by deposition and less subject to erosion. Erosion can be reduced further by choice of blade alloys and by armoring areas of high ash bombardment. Corrosion and deposition may be reduced by greater cooling, although airfoils with passages for cooling are more vulnerable to erosion.

4.2 Steam Bottoming Plant Technology

The current technology of steam power plants is sufficient for requirements of combined cycles in the foreseeable future. Boilers for steam bottoming plants are likely to differ from those for conventional steam power plants in the use of multiple pressures, close temperature approach, and extended heat transfer surfaces in the boilers.

As efficiency becomes more important, combined cycles will likely adopt steam reheat in conjunction with higher steam throttle pressure. Today's combined cycles use steam up to about 1200 psig and 950°F (8.27 MPag and 510°C). For conventional power plants, 2400 psig (16.54 MPag) steam pressure and 1000°F (538°C) throttle and reheat steam temperatures are commonplace. It is ultimately expected that combined cycles will use similar steam temperatures with throttle pressure ranging from 1400 to 2400 psig (9.65 to 16.54 MPag) depending on the size of the steam turbine.

Combined cycles incorporating raw coal combustion, including pressurized fluidized bed combustors (PFBCs) and indirect heated atmospheric fluidized bed combustor (AFBC) systems with their lower gas turbine temperatures, optimize thermodynamically with higher proportions of steam power (66-75%). Single pressure reheat systems much like conventional steam power plants are appropriate for these cycles.

4.3 Steam Cycle Configurations

As briefly discussed, steam cycle configurations required for the future can be divided into two categories: those for combined cycles with gas turbines burning clean fuels and those for raw coal combustion cycles.

4.3.1 Combined Cycles with Gas Turbines Burning Clean Fuels

In combined cycles with unfired heat recovery steam generators (HRSGs), the heat input is established by the gas turbine. The best efficiency will occur when the power output of the steam bottoming cycle is at a maximum for a particular gas turbine.

Overall bottoming cycle efficiency is the product of heat recovery effectiveness and steam cycle efficiency. Good heat recovery effectiveness requires low steam pressure. On the other hand, good steam cycle efficiency requires high steam pressure. For best overall efficiency with a single steam pressure, the requirement for heat recovery effectiveness predominates, and the thermodynamic optimum steam pressure is about 200 psig (1.38 MPag) as shown in Fig. 15 [3].

Good heat recovery effectiveness can be combined with good steam cycle efficiency through the combination of high and low pressures in a multiple steam pressure system. The efficiencies of single, double, and triple pressure cycles are displayed in Fig. 16 as a function of the throttle steam pressure. The performance values shown are based on a gas turbine with an efficiency of 31% (LHV) on oil fuel with an exhaust temperature of 1000°F. Figure 17 shows an example of a triple pressure system with steam reheat.

A multipressure reheat steam system has not yet operated as part of a combined cycle but several such systems are now operating in gas cooled nuclear power plants with gas temperatures leaving the reactor comparable to that of the exhaust of a gas turbine.

Additional power generated by supplementary firing generally degrades the basic combined cycle efficiency--particularly at higher levels of supplementary firing. This can be understood from the fact that the efficiency of a high pressure 1000°F (538°C) reheat steam cycle is about 40.5%. Fuel fired in a boiler is converted to power at a lower efficiency than the fuel fired in a high temperature gas turbine, which generates power in the gas turbine and again in the bottoming steam plant at a combined net efficiency that can be near 49%.

There is an important exception to the above principle. Supplementary firing reduces the severity of the boiler pinch, improves heat recovery effectiveness, and allows steam temperature to be raised to 1000°F (538°C) for firing up to about 1100°F (538°C). These combined improvements can exceed the degradation due to additional heat input to the less efficient bottoming cycle; therefore, supplementary firing below 1100°F can actually improve combined cycle efficiency.

The performance of combined cycles with subcritical pressure steam bottoming and various degrees of supplemental firing is approximated in Fig. 18 [4]. Profiles are shown for several levels of gas turbine firing temperatures. Clean fueled combined cycles in the near future are represented by the 2200°F (1200°C) profile; those further in the future are represented by the 2500°F (1370°C) profile. The raw coal combustion cycles are represented by the 1500°F (820°C) profile.

At zero heat input to steam, line A (ordinate) of Fig. 18, the efficiencies are for simple cycle gas turbines. At line B the efficiencies are for combined cycles with unfired HRSGs. The broken lines connecting lines A and B represent only partial recovery of the gas turbine exhaust heat.

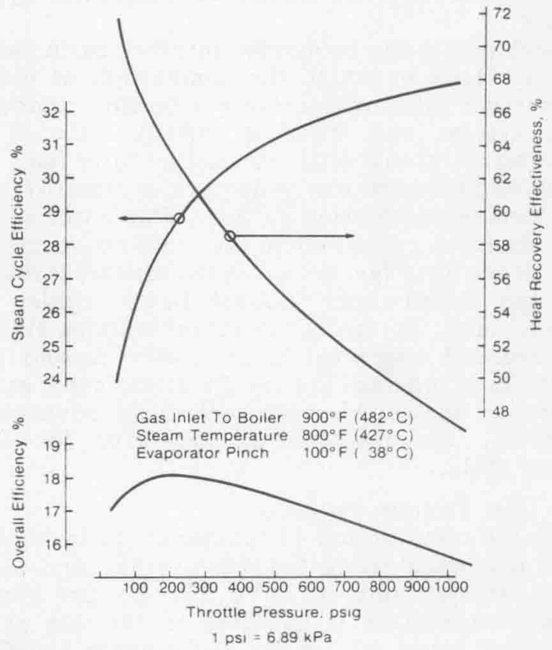

Figure 15. Throttle Steam Pressure Influence on Steam Cycle Efficiency, Heat Recovery Effectiveness, and Overall Bottoming Plant Efficiency

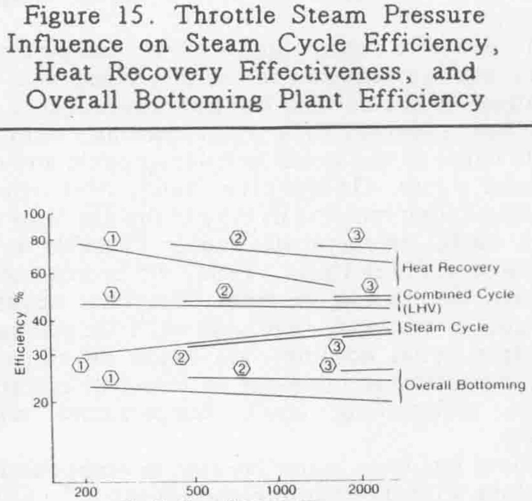

Figure 16. Efficiencies of Single, Double, and Triple Pressure Cycles as a Function of Steam Throttle Pressure

Line C gives efficiencies when the steam temperature is raised to 1000°F (538°C) by supplementary firing. An improvement occurs at all levels of gas turbine firing temperatures. It is greatest for the lower firing temperatures because of the lower steam temperatures without supplementary firing. For all but the lowest gas turbine firing temperatures, firing past line C is detrimental.

Where heat inputs to steam are less than line D, the steam cycles optimize with multipressures. For heat inputs to steam greater than at line D, the high pressure steam production and the economizer water flow relative to gas flow absorbs all available heat leaving the boiler. Consequently, there is no benefit in multiple pressures for greater heat inputs. The best cycle is one with a single high steam pressure where boiler exit gas heats part of the incoming water. The balance of the water is heated by extraction steam as in Fig. 19.

At line E, the oxygen in the airflow is fully used by combustion. Beyond E the airflow is augmented by a forced draft fan. An all steam power plant with no gas turbine is represented by point F.

4.3.2 Raw Coal Combustion Cycles

The PFBC and AFBC combined cycle systems must operate at lower turbine inlet temperatures to ensure effective sulfur capture by the limestone included in the bed. This places a limit on the operating bed temperature at approximately 1600°F (871°C). This is reflected in a turbine inlet temperature of about 1500°F (816°C).

At this lower temperature, the combined cycle works best at a higher proportion of steam power. The net efficiency of a combined cycle with unfired heat recovery boiler and a 1500°F (816°C) gas turbine is about 40%. This is below the efficiency of the steam cycle. As a result, overall efficiency can be improved by additional heat input to the steam cycle. Combined cycle efficiency with a 1500°F (816°C) gas turbine improves as heat input to steam is raised to about 70%. This point is the intersection of the 1500°F (816°C) gas turbine profile and line D on Fig. 18. This corresponds to supplementary firing at about 1400°F (760°C) in an exhaust fired boiler cycle.

5.0 OUTLOOK FOR BURNING COAL

As oil and natural gas supplies become increasingly scarce and therefore more valuable, electric utilities worldwide are looking for better ways to burn lower cost coal. Some countries have large natural reserves of coal; others aim to diversify their imported fuel mix. Studies show that coal-burning power plants incorporating gas turbines promise higher thermal efficiencies, better economics, and less environmental impact than conventional pulverized coal power plants.

Coal technologies are being developed to provide clean fuels for gas turbines. Coal derived liquids provide a direct substitute for today's petroleum fuels. The use of coal derived gases, however, is probably more near term.

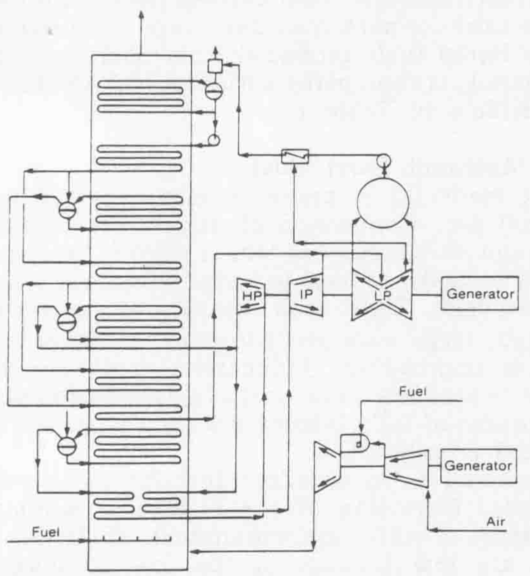

Figure 17. Combined Cycle with Multiple Steam Pressures

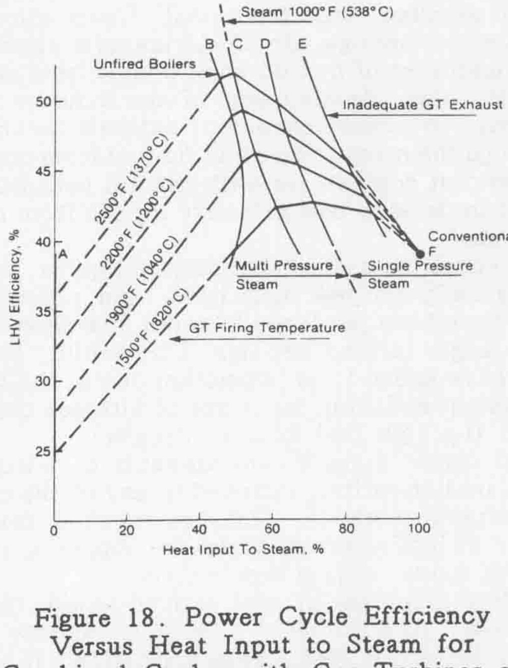

Figure 18. Power Cycle Efficiency Versus Heat Input to Steam for Combined Cycles with Gas Turbines of Various Specific Outputs

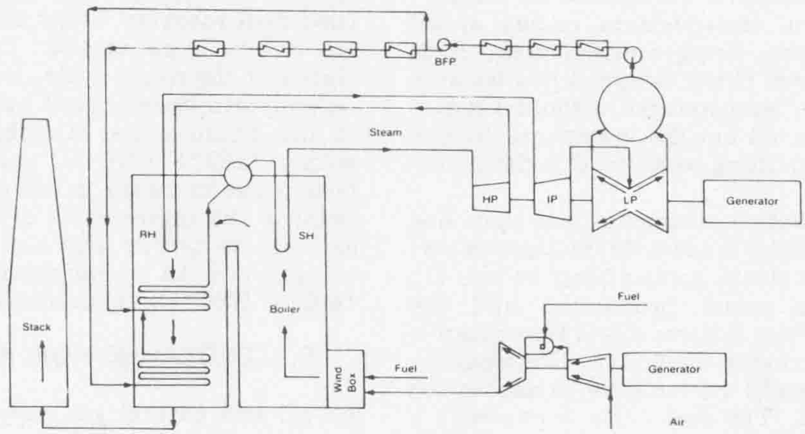

Figure 19. Combined Cycle with Parallel 3 and 7 Heater Water Flow Paths

Raw coal combustion avoids the loss associated with conversion of coal to a clean liquid or gas. In one instance, pressurized products of coal combustion are expanded through the gas turbine. This includes the products from the combustion of coal/water slurries or those from a pressurized fluidized bed combustor (PFBC). Alternately, raw coal combustion takes place in an atmospheric fluidized bed combustor (AFBC). Here, an indirect air heater provides hot pressurized air for expansion through the gas turbine.

5.1 Liquefaction of Coal

During World War II the German military largely ran on gasoline made from coal. Using some of the same technology, South Africa now produces large quantities of hydrocarbon liquids from coal. In both cases, development of the industry was prompted by considerations of national security. Although the production of liquid fuel from coal is not yet cost competitive with natural petroleum, efforts to develop cost effective liquids from coal continues.

To reduce the cost of coal derived liquids, various partially refined fuels have been proposed. Some have been produced in small quantities and tested in gas turbine test rigs. Difficulties, however, have arisen from deposition due to the high ash content and from emissions of nitrogen oxides due to the high fuel bound nitrogen.

Coal contains significant amounts of nitrogen which are not entirely removed in any of the current refining methods. This has required development of innovative measures for suppression of nitrogen oxides during combustion.

Hydrogen content of coal derived liquids tends to be low. This results in a greater tendency for carbon deposition and smoke formation. It also results in higher flame emissivity and higher combustor wall temperatures.

Liquid fuel for gas turbines is a small percentage of the total market. A distribution system for relatively small quantities of a special fuel would be prohibitively costly. Emphasis has been placed, therefore, on the manufacture of coal derived liquids to meet current specifications which could use the established marketing distribution network. Ultimately, coal derived liquids will probably be indistinguishable from other ASTM specification oils. In general, a nitrogen content less than 0.2% and a hydrogen content over 10% should result in a fuel which can be substituted directly for the petroleum derived No. 2 distillate fuel.

All processes for the production of coal derived liquids are projected to result in relatively high cost fuel. As such, coal derived liquids are likely to be used for peak load duty only. Properties of some liquid fuels produced from coal, including methanol, are compared with a typical ASTM No. 2 distillate in Table 1.

5.2 Methanol From Coal

Most methanol is presently manufactured from natural gas. Conversion of natural gas to methanol and its transport in tanker ships is being proposed to make natural gas available from remote gas deposits. Until such deposits of gas are exhausted, large scale production of methanol from coal is improbable. Ultimately, methanol will likely be produced from coal at a gasification plant as a means of fully loading the plant when the gas demand is reduced.

Methanol is an excellent fuel for combustion turbines. Corrosion of the turbine is minimal. Emissions of sulfur are nonexistent. Emissions of NO_x are low because of the low combustion temperature when the methanol is burned as a liquid. Methanol freezes at -140°F (-96°C), so tank heating and heat tracing are unnecessary.