COGENERATION
and
COMBINED-CYCLE
PRINCIPLES
WORKSHOP ©

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ABSTRACT

This Cogeneration and Combined-Cycle Principles Workshop® document provides an overview introduction to cogeneration and combined-cycle powerplants, including primers on gas turbines; heat recovery steam generators; steam turbines and condensers; methodologies for executing a plant from conception through to synchronization and operation; and operations and maintenance concepts. Typical examples of simple-cycle, cogeneration, combined-cycle and combined-cycle cogeneration plants are provided.
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1.0 THE CASE FOR COGENERATION

Cogeneration is normally defined as the simultaneous production of two or more forms of useful energy, usually electricity and heat, from a single fuel source. In Europe and other countries, this concept is often referred to as Combined Heat and Power (CHP).

In the early 1900's, because of a lack of viable alternatives, many small and large industries employed cogeneration for the simultaneous production of both process heat and electricity. With the subsequent development of large central electrical power generating stations and the creation of reliable electrical transmission and distribution systems, this initial interest in Cogeneration and CHP waned, as industries found it more economical to simply purchase electricity, while producing only their own process heat.

Today in a world of global competition, increasing power costs, economic expansion, transmission constraints, environmental concerns and the uncertainties of deregulation, many industries are once again turning to Cogeneration.

Some industries are increasing their current fuel usage or using a waste product as a fuel, for the purpose of generating electricity, either for self-generation (i.e. load-displacement) or for sale to a utility or other customer. The waste heat associated with this operation is then harnessed to provide process heat and/or more electricity. And in some cases, the waste heat associated with a process is being harnessed to generate electricity.

The net effect of either approach has been a new source of corporate revenue from the lowering of operating costs (i.e. savings) and/or from the sale of electricity. The economic incentive in some instances has been so great as to promote large electrical generation plants, sometimes in excess of the user's own requirements.

In recent years, there is a growing concern for the environment and production of greenhouse gases. Cogeneration produces electricity at much higher efficiencies than can be achieved by conventional central generating stations. Thus, to meet the Kyoto Protocol objectives, electricity production using cogeneration systems must be encouraged.

2.0 BASIC COGENERATION CYCLES

The major equipment associated with a cogeneration cycle traditionally includes one or more of the following:

a) Steam generator or boiler.
b) Steam turbine generator (STG).
c) Gas turbine generator (GTG).
d) Heat recovery steam generator (HRSG).
e) Reciprocating spark-ignited Gas, Dual-Fuel or Diesel engine generator.

The manner in which the equipment is combined and used defines the type of cogeneration cycle.

2.1 Topping Cycle

A Topping Cycle is one in which the primary purpose of the input fuel is to generate electricity, and the waste heat exhausted by this process is captured, and used in industrial processes or for district heating or cooling. Thus, the hierarchy of energy flow in a Topping Cycle is Fuel to Electricity to Process.

Exhibit #1 illustrates the two most common topping cycles currently being employed in cogeneration installations.

The First Case shown on Exhibit #1 is a Steam Generator & Steam Turbine installation. Here the fuel is fired in a boiler and is used to produce steam. The steam in turn is expanded in a steam turbine which is used to rotate a generator and generate electricity.

In this process, of the 100% energy in the input fuel*:

a) Some 15% of the fuel energy is converted to electrical Power.
b) Some 65% can be diverted to process uses (Heat).
c) Some 15% is exhausted to atmosphere (i.e. lost) through the stack of the steam generator.
d) A further 5% is lost in the form of auxiliary power uses in the cycle, and friction and radiated losses.

* In the following examples and Exhibits, the energy content or heat value of the input fuel
will always be based upon the fuel's Higher Heating Value (HHV).

The Thermal Efficiency of this particular Topping Cycle, indicating the amount of useful energy forms provided by the input energy, is:

\[
\text{Thermal Efficiency} = \frac{65\% + 15\%}{100\%} = 80\%
\]

The amount of Process Heat compared to the amount of electrical Power produced can be expressed as a Heat-to-Power ratio. In this case the ratio is:

\[
\text{Heat-to-Power Ratio} = \frac{65\%}{15\%} = 4.3
\]

The Second Case shown on Exhibit #1 is that of a Gas Turbine & HRSG installation. Here, the fuel is fired in the combustor of the gas turbine and used to generate electricity directly. The exhaust gas from the gas turbine is then used to supply the fundamental heat to a heat recovery steam generator. The HRSG generates steam which in turn is used to provide process heat.

In this particular process, of the 100% energy in the input fuel:

a) About 28% is converted to electrical Power.
b) Nearly 50% provides process Heat.
c) Some 17% is ultimately lost to atmosphere.
d) A further 5% is utilised in auxiliary loads and is lost in friction and radiated losses.

The net Thermal Efficiency of the second case, of \([28\% + 50\%] / 100\% = 78\%\) is similar to the first steam-generator & steam turbine case. However, the Heat-to-Power Ratio of this second case is significantly reduced, to only \(50\% / 28\% = 1.8\).

It should be pointed out that in these Topping Cycle examples, the input fuel could have been used to operate other prime movers, such as gas or diesel reciprocating engines. Unlike the gas turbine, the waste heat in the engine jackets is captured through ebullient coolers which provides low grade steam or hot water to process. For this equipment, the overall Thermal Efficiency is in the order of 68% and the Heat-to-Power ratio is usually less than 1.0.

2.2 Bottoming Cycle

A Bottoming Cycle is one wherein the fuel energy has primarily been used to provide process heat first. Waste heat from the process is then captured and used to generate electricity.

Exhibit #2 illustrates a typical HRSG & Steam Turbine bottoming cycle, wherein the hierarchy is Fuel to Process Heat to Electricity. The waste heat left over from an industrial Heat process (e.g. furnace, dryers, thermal oxidiser, etc.) is directed to an HRSG, which uses the waste heat energy to generate steam. The steam is generally expanded through a steam turbine generator, to produce electricity.

In this Bottoming Cycle process, of the 100% energy in the input fuel, typically:

a) Up to 50% of the initial fuel energy could end up as available waste heat to the HRSG/Steam Turbine.
b) In the HRSG some 68% of the waste heat provided could be converted to steam, i.e. 34% of the 100% input fuel energy.
c) Some 16% of the 100% input fuel energy is lost to the atmosphere from the HRSG's stack.
d) Some 11% is ultimately converted to electrical Power via the steam turbine generator.
e) Traditionally, the steam turbine is a condensing type, wherein 21% of the original fuel energy is lost to the atmosphere through a cooling medium, usually, water.
f) Auxiliary power, friction and radiant losses account for a further 2% loss of the original fuel energy.

The resultant net Thermal Efficiency of this particular Bottoming Cycle is typically \([50\% + 11\%] / 100\% = 61\%\). The Heat-to-Power Ratio of this Bottoming Cycle is 50% / 11% = 4.5.

A variation of the above Exhibit #2 Bottoming Cycle would be to use part of the steam from the HRSG for further process use or for heating or cooling. This would increase the Thermal Efficiency of the cycle and increase the Heat-to-Power ratio. The actual quantity of electricity produced, however, would be reduced.
2.3 Combined-Cycle Cogeneration

A Combined-Cycle Cogeneration plant is outlined in Exhibit #3. This cycle combines in tandem, the gas turbine topping cycle with the steam turbine bottoming cycle, with an energy flow hierarchy of Fuel to Electricity to Process to Electricity.

In this typical example, of the 100% energy in the input fuel:

a) The gas turbine part of the cycle converts some 28% of the original fuel energy into electrical Power.

b) GTG auxiliary power uses, and friction and radiant losses account for 5%.

c) The HRSG converts some 50% of the input fuel energy to steam.

d) The HRSG exhausts some 17% of the input fuel energy to the atmosphere.

The steam turbine part of the cycle is used both to generate electricity, and to provide steam to process via an internal "tap-off" (extraction). The resultant distribution of the 50% HRSG steam is:

a) Some 14% to electrical Power via the steam turbine generator.

b) Some 15% to process Heat use via the steam turbine's extraction.

c) A further 20% loss to the atmosphere via the condenser cooling medium.

d) A further 1% or so in auxiliaries and friction and radiant losses.

The net Thermal Efficiency of the illustrated cycle is about \([28\% + 14\% + 15\%] / 100\% = 57\%\), and the Heat-to-Power Ratio is about \([28\% + 14\%] / 0.36\).

Both the efficiency and Heat-to-Power ratio can be varied, by altering the amount of steam to process. The more steam to process, the less to the condenser and more efficient the cycle.

More process steam (Heat) and electrical Power could also be produced by burning additional fuel via duct (supplemental) firing in the gas turbine exhaust on its way to the HRSG. This additional fuel energy boosts all downstream outputs including steam production and the steam turbine.

3.0 PROCESS HEAT-TO-POWER RATIO

To assist in determining the most efficient cogeneration cycle for a given application, the Process Heat requirements and the Electrical Power requirements can be determined, and matched to a set of guidelines.

Thus, the Process Heat-to-Power Ratio can provide an indication of which cycle to employ. General optimisation and equipment selection can then be undertaken.

Exhibit #4 summarises the Heat-to-Power ratios for various types of Topping Cycles*, for various process steam pressures (* bottoming cycles are not shown, since their design will generally be dictated by the type and amount of available waste heat which can captured from the process).

With the Steam-Generator & Steam Turbine topping cycle, a Heat-to-Power Ratio as low as 4, or as high as 25 or above is possible. This large variation is primarily affected by the steam pressure of the steam-generator, i.e. the higher the operating pressure, the lower the ratio, since more electricity is produced, per pound of steam made.

The Gas Turbine & HRSG topping cycle is applicable in Heat-to-Power ratios of about 1.5 (unfired HRSG) to 10 (full supplementary-fired HRSG). The large fluctuation in this range is caused by the firing of additional fuel in the exhaust duct prior to the turbine's exhaust gas entry into the HRSG. The additional heat added in this manner can boost the process heat output, to match swings in the process load.

It is apparent that there is an overlap between these two types of topping cycles, and other factors are usually involved in the selection of one cycle over the other.

The primary objective of a Combined-Cycle Cogeneration scheme is electricity production. As such, much lower Heat-to-Power Ratios are encountered, often in the range of 0.2 to 2.0. In the extreme, the Heat-to-Power Ratio could become zero, when only electrical Power is produced in the pure combined-cycle case. In practice, however, a lower regulatory limit is usually encountered first.
4.0 COGENERATION CYCLE EFFICIENCY

4.1 General

The Industrial User usually sets out to displace purchased electricity, avoiding a portion or all of his energy and demand charges, sometimes with the added benefit of an additional or replacement process heat source.

By comparison, the primary goal of a Developer may be to generate electricity for sale to the Industrial User and/or to a Utility, from which their most significant revenues will be obtained. From their perspective, the process Heat use provides the legitimacy for their powerplant. In many cases, this has presented Industrial Users with conflicting considerations. His plant's process heat use and electrical consumption may dictate a small scale cogeneration facility, that may be economically justifiable for them. On the other hand, for the same process heat and electrical load, a Developer may propose a much larger-scale cogeneration plant, with the sale of significant amounts of electrical Power being his primary objective, and with the Industrial User being the steam host and providing other considerations. Frequently, this arrangement can prove beneficial to both parties.

However, some Industrial Users may instead attempt to achieve the gains of the large-scale Developer, with the possible result that their primary business becomes the generation and sale of electricity, instead of their original product and what they know best.

Hence, one must always review the cycle being proposed and remember its driving function. For the reader to evaluate these concerns, a review of one of the important efficiency criteria set by governing and regulatory bodies is helpful.

4.2 Class 43

In Canada, the applicable tax classification for cogeneration facilities and equipment is Class 43, which defines a minimum plant efficiency target which is required to allow accelerated depreciation of portions of the cogeneration equipment.

The required Efficiency or Heat Rate hurdle is 6,000 btu/kW.hr, and is based on the fuel higher heating value HHV. For non-renewable fuels the Class 43 Heat Rate has been defined as:

\[
\text{Class 43 Heat Rate (btu/kW.hr)} = \frac{\text{Fuel Heat Input (btu/hr HHV)}}{\text{Gross Electrical Output (kW)} + \frac{\text{Net Process Heat (btu/hr)}}{3413 \text{ (btu./kW.hr)}}
\]

For steam being sent for process use, the Net Process Heat would be the total heat content (in btu/lb) of the steam being sent to the process, minus the total heat in the condensate return (in btu/lb).

Cogeneration plants which qualify for Class 43, whether constructed by an approved Industrial User or a Developer, may depreciate a portion of the plant based on a 30% declining balance, thus enhancing the plants economic return.

Additional details concerning Class 43 are available from the federal government.
5.0 TYPICAL COGENERATION APPLICATIONS

The use of cogeneration has become common place in the United States and is quickly gaining favour within Canada. Following are some typical heat balances of cogeneration facilities.

5.1 Steam-Generator and Back-Pressure Steam Turbine Application

Exhibit #5 illustrates a Steam-Generator & Back-Pressure Steam Turbine cogeneration scheme that has been employed in alumina plants, pulp and paper applications and small industrial installations.

This system has an extremely high thermal efficiency, since all of the steam that makes its way through the steam turbine is subsequently usefully employed as Heat in the process.

Some production of electrical Power has also occurred, providing the system with an overall thermal power efficiency of 80% and a Heat-to-Power Ratio of 10.

This particular scheme is particularly attractive when existing steam-generators / boilers exist, which produce steam at a high enough pressure.

5.2 Steam-Generator and Condensing Steam Turbine Application

Exhibit #6 illustrates a Steam Generator & Condensing Steam Turbine cogeneration scheme that has the same process Heat requirements (180,000 lb/hr at 150 psig) as the previous case, only the electrical Power output has been increased from 6,000 kW to 13,000 kW.

To increase the electricity production, the amount of fuel energy input to the system was increased, thus raising the steam output of the steam generator. A portion of the steam is now expanded in the steam turbine to sub-atmospheric pressure and condensed.

The maximum overall thermal efficiency of this condensing cycle is reduced to 66%, since the latent heat contained in the steam passing through to the condenser is essentially wasted. The Heat-to-Power Ratio is correspondingly lower, at 4.6.

This particular cogeneration scheme has found application in the petrochemical and the pulp and paper industries, again particularly when existing high pressure boilers exist.
5.3 Combined-Cycle Cogeneration

Exhibit #7 illustrates a typical Combined-Cycle Cogeneration heat balance for the cogeneration application generally promoted by developers for use at industrial sites.

This scheme is intended to maximise the electrical output for the developer, with a relatively small process steam requirement.

The limitations on this size of installation, in the United States, was governed by PURPA and as such these applications were viewed as PURPA configuration.

In Canada, the Class 43 Heat Rate requirements also limit the size of the cogeneration facilities for a given steam host size.

The thermal efficiency of this particular cogeneration scheme is in the order of 48% and the Heat-to-Power Ratio is about 0.4.

5.4 Gas Turbine and HRSG Application

Exhibit #8 illustrates a typical Gas Turbine & HRSG cogeneration application suitable for a small to medium process plant applications such as a food or material processing plants, or a brewery.

These plants have a very high overall thermal efficiency, and by the addition of supplementary firing in the exhaust gas duct between the gas turbine and the HRSG, can also have a high Heat-to-Power Ratio.

The thermal efficiency of this particular cogeneration application is in the order of 80% and the Heat-to-Power Ratio is about 4.4.

The above exhibits and examples are only typical of the many possible opportunities.

Every cogeneration application is different, due to differing technical, infrastructure, financial, economic and/or operational philosophy reasons.
6.0 SUMMARY

Based on the preceding illustrations, a potential Industrial User of cogeneration can define the cycle that is most applicable to their use.

Once decisions are taken on the hierarchy of the cycle and process heat production, cogeneration facilities can be sized to either meet average or peak process heat requirements or peak or average electrical requirements.

An Industrial User must determine if the primary objective of his cogeneration facility will be the production of process Heat or of electrical Power. A second consideration is the responsibility for process heat production, i.e. does he wish to be the recipient of process heat from a third-party, or does he wish to control it himself?

A very significant variable in any industrial user's analysis will be whether or not a third-party may be involved in the development of the cogeneration facility. Once this path is followed, electricity production for sale to a utility becomes paramount to the Developer. The production of process steam for the industrial user becomes a mandated necessity to justify that the maximum size facility that can be developed.

The net result of either of the above approaches from an Industrial User's point of view is that he can direct a project that meets his process steam and/or electricity requirements more cost effectively. For him, this translates into lower operating and production costs, and a more competitive product.

ACKNOWLEDGEMENTS

A. STEAM GENERATOR/STEAM TURBINE

THERMAL EFFICIENCY = 65 + 15 = 80%
HEAT TO POWER RATIO = 65/15 = 4.3

B. GAS TURBINE/HRSG

THERMAL EFFICIENCY = 50 + 28 = 78%
HEAT TO POWER RATIO = 50/28 = 1.8

ALL EFFICIENCIES ARE HHV
HRSG/STEAM TURBINE

THERMAL EFFICIENCY = 50 + 11 = 61%
HEAT TO POWER RATIO = 50/11 = 4.5

HEAT TO PROCESS 50%

WASTE HEAT 50%

FUEL 100%

16%
HEAT RECOVERY STEAM GENERATOR

STACK

STEAM 34%

STEAM TURBINE

2%
AUX. POWER & LOSSES

GENERATOR

11%
ELECTRICITY

COOLING WATER MEDIUM

21%

ALL EFFICIENCIES ARE HHV

Gryphon
International Engineering Services Inc.

Bottoming Cycle

Jan. 2000
Cogeneration Principles
Exhibit 2
GAS TURBINE/HRSG/STEAM TURBINE

THERMAL EFFICIENCY = 28 + 15 +14 = 57%
HEAT TO POWER RATIO = 15/(28 + 14) = 0.36
Steam Generator/Condensing Steam Turbine

Fuel: $375 \times 10^6$ BTU/hr
HHV: (HHV) 110,000 kW Thermal

Steam Generator:
- 600 PSIG: 280,000 LB/HR
- 150 PSIG: 180,000 LB/HR

Condensing Steam Turbine Generator:
- 650°F
- 16,000 KW Thermal

Stack:
- 255°F

Condensate:
- 50,000 LB/HR

Deaerator:
- 15 PSIG

Condenser:
- 70,000 LB/HR
- 2" HG
- Heat Rejected = 16,000 KW Thermal

Steam to Process (60,000 KW Thermal)

13,000 KW Net Electrical Output

Losses and Auxiliary Power = 5000 KW

HHV Thermal = 13,000 + 60,000 = 66.4%
Efficiency = 110,000

Heat/Power Ratio = 60,000 = 4.6
13,000

Gryphon International Engineering Services Inc.

Typical Heat Balance
Pulp & Paper Application

Jan. 2000
Cogeneration Principles
Exhibit 6
COMBINED CYCLE COGENERATION

STEAM HEAT TO PROCESS
= 12,600 + 7,800
= 20,400 kW THERMAL

NET ELECTRICAL OUTPUT
= 39,800 + 15,000
= 54,800 kW

LOSSES = 6,500 kW

HHV
THERMAL = 54,800 + 20,400 = 47.5%
EFFICIENCY = 158,200

HEAT/POWER = 20,400 = 0.4
RATIO = 54,800

GARYPHON
INTERNATIONAL ENGINEERING SERVICES INC.

TYPICAL HEAT BALANCE
COMBINED CYCLE APPLICATION

JAN. 2000
COGENERATION PRINCIPLES
EXHIBIT 7
GAS TURBINE GENERATOR/HEAT RECOVERY
STEAM GENERATOR

DEAERATOR

CONDENSATE

52,800 LB/HR

STEAM TO PROCESS
(15,660 KW THERMAL)

125 PSIG
(SAT.)

STACK
(4640 KW THERMAL)

3520 KW ELECTRICAL OUTPUT

GAS TURBINE GENERATOR

FUEL
(HHV) (24,000 KW THERMAL)

3.52 x 10^6 BTU/HR

(81.9 x 10^6 BTU/HR

AIR

120,000 LB/HR

1055° F

1830° F

3.24 x 10^6 BTU/HR

HEAT RECOVERY
STEAM GENERATOR

LOSSES AND AUXILIARY
POWER = 600 KW

HHV THERMAL = \frac{3520 + 15,660}{24,000} = 80\%

EFFICIENCY

HEAT/POWER = \frac{15,660}{3520} = 4.4

RATIO

GLYPHON
INTERNATIONAL ENGINEERING SERVICES INC.

TYPICAL HEAT BALANCE
SMALL PROCESS PLANT APPLICATION

JAN. 2000 COGENERATION PRINCIPLES
EXHIBIT 8
1.0 GAS TURBINE CONCEPTS

1.1 The Basic Gas Turbine Cycle

The basic gas turbine thermodynamic cycle is an open Brayton Cycle, using air as the working fluid. Air at ambient state 1 is compressed to a high pressure at state 2; fuel is added and continuously burnt to the firing temperature (state 3); and the resultant high pressure-temperature air is expanded through a turbine to atmosphere (4).

The expander turbine drives the compressor, and the excess power available is used to drive a load.

The higher the turbine inlet temperature T3, for a given pressure ratio, the higher the power output and efficiency of the unit. As gas turbine materials have developed, turbine “firing” temperatures have steadily climbed from the 1400 deg F region, through 2000~2200 deg F, and now up to 2600 deg F in the newest units being offered.

The higher the pressure ratio P2/P1, the higher the unit’s efficiency (for higher firing temperatures), and the higher the specific power output of the engine (hp/lb/sec). Accordingly, most aircraft gas turbines (jet engines, turboprops and turbofans) use high pressure ratio, high firing temperature designs to minimize weight and frontal area. Thus, the resultant aero-derivative GT shares the high-pressure ratio design. Pressure ratios vary from 7.5:1 for the smaller and older technology GT’s, to 35:1 for the most advanced GT’s.

1.2 Turbine Cycle Variations

There are several major variations of the basic Brayton gas turbine cycle, including:

a) **Reheat or sequential combustion** – typically employed in high-pressure ratio GT’s. The hot gases leaving the HP turbine section, are reheated...
by the combustion of additional fuel prior to entry into the LP turbine section. Reheat increases the output of the LP turbine, and raises the turbine’s exhaust temperature, resulting in increases in simple-cycle and combined-cycle power output.

b) **Recuperated or regenerated GT’s**, typically employed in low-pressure ratio units with high firing temperatures. An external regenerative heat exchanger is used to transfer heat from the exhaust to the air leaving the compressor, before it enters the combustor. This saves fuel and increases thermal efficiency, with a slight decrease in power output. Because of the high-flow, high-temperature nature of the process, these heat exchangers are very large and expensive.

c) **Inter-cooled GT’s** – typically employed in high-pressure ratio multi-shaft units. Compressed air from the LP compressor is directed through an external heat exchanger, and a cooling medium (e.g. cooling water, or a process fluid) is used to decrease the air temperature (hopefully close to ambient conditions) prior to entry into the HP compressor. This process decreases the required HP compressor power, thus improving the efficiency and specific output of the unit. Again, because of the external heat exchanger and modifications to the “standard” unit, this process tends to be large and expensive; plus an external cooling system of some form is required.

An interesting recent, and less-expensive intercooling twist is spraywater cooling of the compressor air between the LP and HP compressor of an multi-shaft aero-derivative unit that is controlled to a fixed compressor discharge temperatures T2. The cooling of the HP compressor inlet air allows increased HP compressor mass flow (for the same T2 setpoint), resulting in an increased pressure ratio, increased unit power output and improved efficiency, especially at higher ambient temperatures. Further enhancements can be achieved by spray cooling or fogging the inlet to the LP compressor, during high ambient temperatures.
1.3 Basic Components of the Gas Turbine

The basic components of the gas turbine include a compressor, a combustor and a turbine:

a) **Compressor section** – most frequently a multi-stage axial design. In each stage, a row of stationary blades (stators) acts on the air to impart the correct angle for the succeeding rotating blades. A final set of outlet guide vanes and a diffuser straighten and slow the air stream prior to entry into the combustor section.

In most GT’s, small portions of the compressed air are bled out along the blade path and used for cooling purposes in the hottest portions of the stationary and rotating turbine sections. In some gas turbines, small portions of the compressed air are taken out and used for pressure balance elsewhere in the GT to minimize axial loading.

Compressor air bleed systems are employed to discharge excess air during starting (when the blading design works less efficiently at lower rpm’s), and during part-load operation. Pivoted variable guide vanes are frequently employed on both industrial and aero-derivative units, to manage the bulk inlet air flow into the GT (i.e. inlet guide vanes), and to manage the mass flow further along the blade path for control reasons.

b) **Combustor section** – which can be a multi-can or basket design, or an annular ring design.

For standard diffusion combustion systems (i.e. non dry low-NOx), gaseous or liquid fuels are introduced via nozzles located at the head of each combustor can, or the front of the combustion annulus chamber. A portion of the air from the compressor is introduced directly into the combustion reaction zone (flame), with the remainder introduced afterwards to shape the flame, to quench the flame to an acceptable firing temperature (T3), and to cool the walls of the combustor and downstream liners.

In some cases – for environmental or power enhancement reasons – water or steam is injected into the primary combustion zone, to reduce flame temperatures and the production of thermal NOx.

Current generation dry low-NOx (DLN or DLE) combustion systems operate differently than above, using the lean premix principle, as will discussed further below.

Between the baskets and the turbine blading, transition ducts or liners are used to carefully shape the gases for the turbine section, with velocity & temperature profiles.

Fuel and steam and/or water injection manifolds and hoses are located around the circumference of the combustor section.

c) **Turbine section** – usually a multi-stage axial design. In each stage, a row of stationary nozzles acts on the hot gases to impart the correct angle for the succeeding rotating blades.

The most critical section of the turbine is the first few stages. Both the nozzle and the rotating blade are exposed to “red-hot” gases at the design firing
temperature, which is far in excess of acceptable creep-fatigue limits for the engineered alloys employed. In addition, the rotating blade has to survive while being subjected to high centrifugal and mechanical stresses.

As a result, massive research and development efforts are conducted to design materials and systems for the cooling of these stages.

Internal cooling passages are cast and machined into the nozzles and blade, and cooled compressor bleed air is passed through them to maintain material temperatures at acceptable limits.

Creep-resistant directionally-solidified and single-crystal blade production technology has moved from the aircraft GT world into the industrial heavy-duty GT design world.

Thermal barrier coatings are employed to protect the aerodynamic surfaces and materials from corrosion, oxidization and erosion.


2.0 THE GAS TURBINE ASSEMBLY

2.1 The Basic Gas Turbine Machine

The individual compressor, combustor and turbine sections and casings are bolted together, and supported via struts and baseplates, to make a complete machine.

The rotating compressor and turbine sections are mechanically interconnected, thus the compression power needed is provided by the turbine power made. The excess power available is taken off an output shaft and used to drive a pump, compressor or generator. The graphic below shows a “cold-end” drive unit – “hot-end” drives are also employed on other machines.

Typically, 60% to 70% of the turbine’s power output is needed to drive the compressor section, with the remaining 30% to 40% available as true shaft output power. For example, a typical nominal 50 MW single-shaft industrial gas turbine will produce 150 MW in the turbine section, and provide about 100 MW of it to the compressor section.

Inlet manifolds are installed to accept ducted filtered air. Exhaust gases are directed through a diffuser, to an exhaust stack or an HRSG.

Fuel, steam and/or water manifolds and hoses, cooling air systems, controls and instrumentation are installed, to make a complete unit.

2.2 Gas Turbine Variations

Other configurations of GT’s exist than the single-shafts, including:

a) Single-shaft with PT – industrial and aero-derivative units. A single-shaft GT operates at the speed and firing temperature that keeps itself self-sustained. It’s exhaust gases are passed to an aerodynamic-coupled free power turbine (PT) that drives a load, at either fixed (generator) or variable speed (mechanical drive).

The prime unit is sometimes simply called a “jet”, or “gas-generator”, for convenience.
b) **Multi-shaft, with and without PT** – industrial units designed for variable-speed mechanical drive, and derivatives of aircraft engines.

The basic compressor and turbine sections are divided into HP and LP units, and each usually operates at a different speed depending upon load and ambient conditions.

The LP compressor (LPC) is coupled to and is driven by the LP turbine (LPT). The HP compressor (HPC) is coupled to and is driven by the HP turbine (HPT).

In some three-shaft machines, an intermediate compressor (IPC) and turbine (IPT) are also used, in between the LP and HP sections (configuration not shown).

Fixed or variable-speed loads are driven off the LP shaft portion of the unit. Some units offer the capability to drive off either the cold-end or hot-end of the LP shaft.

In some cases, these multi-shaft units, act as another “gas generator”, and a PT is required to drive the load.

2.3 **Aero-Derivative & Heavy-Duty Industrials**

Aero-derivative and heavy-duty industrial GT’s are basically cousins, sharing the same basic thermodynamic cycle, but each executing the cycle differently depending upon the original design intent.

a) **Aero-derivative GT units** are based upon aircraft engines, and are usually characterized by low weight and low frontal area, both of which are generally inconsequential in industrial service.

The original jet engines developed for military and commercial applications produced forward thrust from the high temperature exhaust gases by the use of nozzles. Most of these jet units receive a PT for industrial power application. The “jets” utilize anti-friction roller bearings, while their PT’s use journal bearings.

Later derivations of the jet engine, especially for commercial application, add more turbine stages which are used to drive propellers (turbo-prop) or large fans for increased thrust (e.g. low and high-bypass ratio turbo-fans). The majority of these use anti-friction bearings, and are industrialized by redesign of the prop or fan takeoff drives.

Because of their high firing temperatures and pressure-ratios, most aero-derivatives are very efficient, compared to their same size industrial cousins. However, their inlet and exhaust flows are significantly less than same-size industrials, because of their high specific output (hp/lb/sec)
Major maintenance of the aero units is accomplished by the complete removal of the gas turbine from its package by special lifting frames. This is followed by the disassembly of its modules into smaller components such as the LPC, HPC, combustion module, HPT and LPT, etc. Most minor maintenance activities can be conducted at the plant site, but major disassembly and overhauls require that the unit be returned to certified shops.

Lease engines are generally available to replace the original engine that is under repair, during the overhaul period.

b) Heavy-duty industrial GT units for power generation are heavy and rugged units optimized to operate within a fairly narrow speed range and at base load duty. Typically, scheduled maintenance intervals are longer than aero units.

The industrial units are characterized by the use of heavy multi-cylinder castings and fabrications, large bolted horizontal and vertical split joints, and heavy built-up rotors, journal bearings and large solid couplings, large baseplates and frames.

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Major maintenance of the larger industrial GT units is usually accomplished by the removal of the top half cylinder, the removal of diaphragms and blade rings, the lifting and removal of the rotor, and subsequent blade removal, all of which can usually be accomplished at the plant site.

<table>
<thead>
<tr>
<th>Performance</th>
<th>Aero-Derivative</th>
<th>Heavy-Duty Industrial</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Up to 50 MW.</td>
<td>Up to 240 MW+</td>
</tr>
<tr>
<td></td>
<td>Up to 41–42% efficiency (LHV).</td>
<td>Up to 35–40% efficiency (LHV).</td>
</tr>
<tr>
<td></td>
<td>Less waste heat opportunity from the exhaust gases.</td>
<td>Good waste heat opportunity.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Large units with high exhaust temperatures allow reheat combined-cycle</td>
</tr>
</tbody>
</table>

| Fuel Aspects | Natural gas to light distillates and jet fuels. Most require relatively high gas pressures. | Natural gas through to distillates and cheaper heavy or residual fuels. Generally require lower gas pressures. Expensive treatment of heavy / residual fuels is required. |

| Start-Up | Quick startup – 5–20 minutes. Relatively low horsepower starters usually electro-hydraulic | 20 to 60 minutes depending on size. High horsepower diesel or motor starters, also some are started by the motoring the generator itself |

| Loading | Quick loading, sometimes 10–25%/min | Slower loading, 1–10%/min depending on size |

| Shutdown | Many larger units require a short time of motoring to cool internals after a trip, but can then be shutdown | Many units require 1–2 days on turning gear after shutdown, but most can be motored to assist quicker cool down |
3.0 THE GAS TURBINE PACKAGE

To prepare the gas turbine machine for service at a site, it needs to be packaged with driven equipment, controls and auxiliaries, in a form which is straightforward to install & commission, and that is easy to maintain.

3.1 Driven Equipment

Industrial gas turbines typically drive generators and process or pipeline compressors, with occasionally use as large pumping sets for oil.

Synchronous generators for gas turbines are rated in accordance with ANSI C50.14, and are usually either 2-pole (3600 rpm for 60 Hz service) or 4-pole (1800 rpm for 60 Hz). Generator output voltages range from 600V for the very smallest GT’s, to 2.4 and 4.16 kV for the 3~8 MW class units, to 13.8 kV for the 10 MW+ units, and 27.6 kV for the 100 MW+ units.

Excitation systems are used to control the generator’s voltage and power factor/var, and can be either brushless (which usually derives it’s excitation power from a shaft-mounted permanent magnet generator) or a thyristor-fired static system (which derives it’s excitation power from an external transformer). Brushless systems are relatively maintenance-free, however their speed of response under transient fault conditions are not as fast as static systems. Accordingly, most large gas turbine generator (GTG) sets employ static excitation, with their attendant high-maintenance brush and slip ring systems.

Generators can be air cooled, water-cooled (TEWAC) or hydrogen-cooled (the largest units).

When the GT machine’s output speed does not match the required generator speed, gearboxes are required. In the larger frame sizes, double-helical gear units are generally employed. Epicyclic gears are sometimes employed in the smaller, high-speed GT classes.

3.2 Air Inlet Systems

Proper air inlet systems for gas turbines are critical to the health of the GT and for noise mitigation.

High-volume multi-stage high-efficiency filtration systems are employed to capture atmospheric particles, to prevent their deposition on the compressor bladepath.

For some GT applications, heating of the inlet air is employed at low ambients, to prevent icing in the compressor or air inlet manifold, and/or to fool a pressure ratio limited aero-derivative GT into operating at a more optimum ambient temperature.

In some cases, inlet air cooling is employed at high ambients, to increase GT power output.

Tuned inlet air silencers are employed to absorb the sound emanations from the gas turbine intake.

3.3 Lubricating Oil Systems

Main, auxiliary and emergency lubricating and control oil (as required) systems are provided for the gas turbine and driven equipment.
Most aero-derivatives require fire-resistant synthetic lube oils for the GT, while their power turbines, gearboxes and generators employ mineral based lube oils. Most heavy-duty industrial GT’s have a common lube oil system for the complete drivetrain.

Lube oil can be cooled either by aerial fin-fan coolers, or oil-to-water heat exchangers.

### 3.4 Fuel Systems

Fuel control systems for gaseous and liquid fuels include filters, strainers and separators; block & bleed valves; flow control/throttle and sequencing valves, manifolds and hoses.

For the complex dry low-NOx units, several throttle valves may be employed, staged and sequenced to fire the pilot, primary, secondary, tertiary and quaternary nozzle and basket sections (as applicable) of the DLE combustion system for startup/shutdown, speed ramps, load changes.

### 3.5 Enclosures

For most of the smaller industrial and almost all of the aero-derivative GTG packages, the complete drivetrain is enclosed in an acoustic enclosure(s). The turbine and generator are compartmentalized and separately ventilated, for hazardous area classification reasons. As a result, these units tend to be pre-packaged at the manufacturer’s facility, and can be easier and quicker to install.

The 40 MW+ industrial GT machines are too large to pre-package in an enclosure, and the components are shipped in major blocks for assembly at site. Enclosures or buildings (if required) are built around the complete drivetrain as site construction progresses.

### 3.6 Controls and Monitoring

Current GTG control systems are complex combinations of digital PLC and/or processor systems, either selected from vendors such as Woodward, or based on vendor-proprietary systems, or occasionally DCS-based.

The systems manage the GT fuel control and speed/load control, and the generator’s voltage, power factor or var control, and breaker synchronization.

The control systems manage the sequencing of all the auxiliaries, and usually include vibration, temperature and pressure monitors.

Generator protective relaying can be provided by the GTG vendor, or can be designed separately for more complex connection requirements.

Sequence of events recorders are employed to determine causes of trips and shutdowns. Metering systems are installed to monitor and catalog billings. Most current GTG control systems will read/write to a plant DCS.

### 3.7 Miscellaneous Auxiliaries

Starting and turning gear systems; inlet manifolds; exhaust diffusers or plenums; water wash systems; water and steam injection (if required); gas detection systems; fire detection and CO2 suppression systems; battery and charger systems; ventilation and heating; exhaust expansion joint, silencer and stack systems (simple-cycle).
4.0 NEWEST GAS TURBINE TECHNOLOGIES

The two primary drivers of new gas turbine technology today are increased gas turbine and combined-cycle efficiency and decreased machine and plant emissions.

Increased gas turbine and cycle efficiency is being achieved by the raising of firing temperatures and pressure ratios, and the design of gas turbines for highly efficient combined cycle power output.

Decreased GT emissions were initially achieved by water or steam injection, however, for new installations today, dry low-NOx combustion systems are preferred, since they are becoming more reliable and are achieving lower emission rates. Other unique emission control technologies are in testing stage.

4.1 Firing Temperatures – Nozzle and Blade

The first stage of the turbine section, composed of a stationary blade row and a rotating blade row, is directly exposed to the hot gases that result from the combustor and the flame cooling system. The nozzles see the highest temperature, but have lower mechanical stresses compared to the rotating blades.

To survive in this environment, expensive alloys that can operate at metal temperature of about ~1400 deg F are utilized. However, since the current firing temperature are well in excess of this, cooling of the nozzles and blades is required.

The earliest gas turbines that were developed from steam turbine technology employed low-grade alloys and low firing temperatures (1400 ~ 1500 deg F) and only required simple cooling methods. In general, cooling air was simply taken from the compressor section of the unit and routed into the rotating section of the machine. The nozzles see the highest temperature, but have lower mechanical stresses compared to the rotating blades.

As firing temperatures increased, nozzle designs were significantly changed. The airfoil section was cast hollow, and small holes were machined inward from the airfoil’s leading and trailing edges, and along the concave and convex surfaces. Cooling air from inside the hollow would discharge through the holes and act to “film-cool” the outer surfaces of the airfoil profile.

To allow increased firing temperatures in the next generation of turbine blades, several small radial cooling passages were cast or electrically machined through the length of the blades. The cooling air entering the blade root would pass up to the tip of the blade, and enter into the gas path. The cooling air supply system was also redesigned to pass through an external cooler, decreasing cooling air supply temperatures, thus improving heat transfer. New alloys were developed to allow higher metal temperatures, and oxidization and corrosion resistant coatings were developed for nozzles and blades.

Past 1600 deg F, protective coatings were required for nozzles and buckets to minimize hot corrosion.

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The blade’s design was also changed from a solid design to a hollow one with an internal serpentine air path. Air discharges radially; through film cooling discharge holes at the leading edge and from holes at the trailing edge. Directionally solidified blades increased creep strength and blade life.
These technologies are also being employed for the turbine stages past the first. Second, and sometimes third stages, are frequently cooled by the same method as the first stage. The later stages are usually cooled by multiple radial passages.

This increasing growth in firing temperatures over time, and the impact on design, has been labeled by various manufacturers as B technology, D, D5, E, EA, F, FA, G and H, to represent the different technology step changes that have been made.

The first G and H machines are being installed today, and employ single crystal blade designs, and more efficient steam cooling (in lieu of air), and are projected to operate at 2600 deg F firing temperature. Steam cooling also improves power output by eliminating the parasitic loss of cooling air to blade cooling passages.

The ultimate goal of turbine blade designers is ceramic blading, with no cooling at all. Research programs are underway to develop suitable materials that can achieve the required strength, durability and impact resistance.

4.2 Firing Temperatures – Combustion Systems

To increase firing temperatures, more combustion air is introduced at the head of a basket, and less quenching air is introduced downstream of the combustion flame. Accordingly, less air is available for cooling the combustor walls, at the same time that bulk temperatures have increased. The affect is similar for DLN systems, where increasing amounts of fuel and air are premixed prior to entering the flame reaction zone.

To improve the creep strength and wear characteristics of the basket / liner and transitions, new materials have been developed that have acceptable strength and life at ~1400 deg F, combined with new thermal barrier coatings that insulate the underlying materials. Maintenance of these coatings is critical to the survival of the combustion system components.

4.3 Dry Low NOx Combustion Systems

Thermal NOx production can be decreased by decreasing the flame temperature, and by decreasing the time that the hot gas mixture is at flame temperature. Coincidentally the highest rate of NOx formation occurs at stoichiometric temperature, where combustion occurs at base load in most conventional combustors.

Conventional diffusion combustion systems and fuel nozzles have been adapted to accept steam or water injection to reduce the local flame temperature and thus NOx. Typical NOx and CO production values with injection are 42 ppm and 25 ppm respectively, on gas.

Current generation dry low NOx combustion systems operate on a completely different principle. To reduce flame temperature, the flame process is conducted in the lean (with excess air) zone, below stoichiometric. The air and fuel are completely pre-mixed prior to entering the flame.
zone to achieve stable combustion.

While the lean zone yields improved emissions, the operating temperature range is much more limited than a diffusion flame, and flame stability degrades. In order to cater for GT ignition, acceleration, and operation over a wide load range, many nozzles must be installed, and each turned on and off as required. As a result, the number of nozzles, manifolds and hoses, external control valves, and the computing power requirements have dramatically multiplied.

The example below is a first-generation staged multi-nozzle arrangement, as applicable to only one of several combustor baskets on a single machine.

GT’s with new and reliable DLN combustion systems are now being offered with near single-digit NOx and CO values.

On the potential horizon for gas turbines are catalytic combustion systems. These systems mix the fuel and air upstream of a proprietary metal catalyst substrate. Approximately 50% of the fuel is oxidized via flameless combustion in the catalyst, with the maximum temperature limited to about 1700 deg F to protect the metal. The remaining 50% of fuel is oxidized downstream of the catalyst, again limited to about 1700 deg F. NOx and CO production in the order of 3 ppm and 5 ppm respectively, has been reported for some of the systems undergoing test.

Another ultra-low NOx development being researched now is a Rich-Quench-Lean combustor, which operate as it sounds. The initial combustion takes place in the rich zone, at low NOx levels, and just below the point of significant smoke production. The flame is then rapidly cooled or quenched by air jets to a low gas temperature, and the remaining combustion takes place in a low NOx lean environment. The primary design problems are stability, firing temperature control and a suitable mechanical quenching and mixing design.

DLN-2
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1.0 INTRODUCTION
A Heat Recovery Steam Generator (HRSG) or waste heat boiler, is the standard term used for a steam generator producing steam by cooling hot gases. Waste heat is obviously a very desirable energy source, since the product is available almost operating cost-free, and increases the efficiency of the cycle in which it is placed, either for process steam generation or for incremental power generation.

HRSGs can regain energy from any waste-gas stream, such as incinerator gases, furnace effluents, or most commonly, the exhaust of a gas turbine set (GT).

2.0 FUNDAMENTAL PARTS of the HRSG
HRSGs can be made up from a number of components, including evaporators, economizers, superheaters, reheaters, integral deaerators and preheaters.

Each of these heat-transfer sections performs a specific task, and the ones that are selected are generally dictated by the required steam conditions for process use or power generation, the type of power generation cycle and/or the efficiency requirements, weighed against HRSG costs.

2.1 Evaporators
HRSG evaporator or boiler sections act to vaporize water and produce steam in one component, like the kettle in the kitchen.

A bank of finned tubes is extended through the gas turbine’s exhaust gas path from a steam drum (top) to a lower (mud) drum. Boiler feedwater is carefully supplied at the appropriate pressure to the upper drum below the water level, and circulates from the upper to lower drum by external downcomers, and from the lower drum back to the upper drum by convection within the finned tubes.

In the steam drum, a “water level” is carefully maintained in the middle of the cylinder – virtually dry steam rises from the water surface, and exits the steam drum through moisture separators and cyclones – delivering essentially 100% dry, albeit saturated steam.

The amount of heat absorbed by the water, and the amount of heat released from the GT exhaust gas to generate steam is the product of the gas mass flow rate, average gas specific heat capacity (C_p), the temperature difference (dT) across the evaporator and the amount of heat transfer surface area installed.
If an infinite amount of surface is installed, the gas temperature leaving the evaporator will equal the saturation temperature of the steam – and maximum heat recovery will be achieved in this component. However, infinite surface is obviously neither practical, economical or good for the gas turbine (resulting in very high exhaust pressure loss and thus reduced GTG power output).

Accordingly, a “reasonable” amount of surface is installed, as dictated by economic considerations – as a result the gas temperature leaving the evaporator section is higher than saturation. This difference between the saturation temperature and the gas exit temperature is called the “pinch point”. The smaller the pinch point, the more surface that has been installed, the more steam that is produced, and the lower the power output of the gas turbine, and vice versa.

To reduce the pinch point requires increasingly dramatic amounts of HRSG surface area, as indicated by the typical figure below. For unfired HRSGs, the optimum pinch point can range from 10 F to 30 deg F.

Instead of pinch point, the amount of surface area in an economizer is quantified by the “approach temperature”, i.e. the difference between the feedwater temperature leaving the economizer, and the saturation temperature in the drum to which it is delivered. For most HRSGs, it is desirable to maintain a discreet approach temperature under all operating conditions (i.e. considering fired, unfired and part-load GT situations), in order to prevent the generation of steam directly in the economizer – where it doesn’t belong. Typical approach temperatures are approximately 25 ~ 40 F.

In a single pressure HRSG, the economizer will be located directly downstream (with respect to gas flow) of the evaporator section. In a multi-pressure unit the various economizer sections may be split, and be located in several locations both upstream and downstream of the various evaporators.

### 2.2 Economizers

The gas temperature leaving an evaporator varies from 300 ~ 600 deg F, depending upon the steam pressure being produced. If no other heat transfer component is installed downstream, this remaining energy is wasted.

Accordingly, economizers are frequently installed downstream (with respect to gas flow) of the associated evaporator, and lower gas temperatures further, thus increasing heat recovery. Economizers are serpentine finned-tube gas-to-water heat exchangers, and add sensible heat (preheat) to the feedwater, prior to its entry into the steam drum of the evaporator.
2.3 Superheaters

While the evaporator produces dry-saturated steam, this is rarely acceptable for large steam turbines, and is frequently not the appropriate condition for process applications.

In these cases, the saturated steam produced in the evaporator is sent to a separate serpentine tubed heat exchanger referred to as a superheater, which is located upstream (with respect to gas flow) of the associated evaporator. This component adds sensible heat to the dry steam, superheating it beyond the saturation temperature.

The superheater can consist of either a single heat exchanger module or multiple heat exchanger modules. The final steam outlet temperature will vary depending upon the gas turbine exhaust and/or duct burner conditions, unless controlled.

For single modules, a temperature controlling desuperheater can be located outside the HRSG, to adjust the final outlet temperature.

If two superheater modules are installed, the temperature controlling desuperheater is generally mounted between the two discreet modules, i.e. interstage attemperation, allowing more precise steam temperature control.

Superheaters can be either bare tubes or finned tubes, depending upon material and inlet gas temperature considerations.

The three primary heat transfer components discussed above, i.e. the economizer, evaporator and superheater, are included in all power generation HRSGs, and most process steam HRSGs.

2.4 Reheaters

Reheaters are a heat transfer component similar to superheaters, and are employed in advanced multi-pressure power generation cycles. They accept superheated or semi-saturated steam at a low pressure from a steam turbine after it’s first section of expansion, and re-superheat or “reheat” the steam back towards the original superheater’s outlet temperature. Accordingly, reheaters are generally interspersed among the superheater sections in the HRSG, so that the same outlet temperatures can be achieved.

In general, reheat systems are very expensive due to the high temperature, large-diameter HRSG tubing and external piping systems required, and complicate startup, operation and load adjustment. Reheaters appear to be justified only in the largest combined-cycle plants where the highest efficiency is prime.

In general, there will only be one reheat section in an HRSG, irrespective of how many steam pressure levels are employed in the unit.
2.5 Integral Deaerators

All power plant cycles employ a deaerator to control oxygen levels in the feedwater. Heating steam is provided to strip oxygen from the condensate falling through the pressurized deaerator’s tray systems.

Normally, this deaerating steam is a parasitic loss from such a cycle. In some HRSG’s, the deaerator can instead be mounted on the HRSG in the final portions of its gas path, with a finned or bare tube bank extended into the gas stream. The pegging steam is produced from the tube bank and provided directly to the deaerator mounted above.

The result is decreased stack temperature, i.e. improved heat recovery, and decreased piping and equipment cost.

2.6 Preheaters

Typically, preheaters are located at the coolest end of the HRSG gas path, and absorb energy from the gas stream to preheat liquids such as condensate, makeup water, water/glycol mixtures or proprietary heat exchange fluids, e.g. Dowtherm.

The most common application is to preheat condensate prior to entry into the deaerator, which reduces the amount of deaeration steam required.

Due to the low gas and fluid temperatures associated with integral deaerators and preheaters, there can be concerns related to water dewpoint corrosion and acid dewpoint corrosion.

a) **Water dewpoint corrosion** can occur when the preheater metal temperatures are below the water dewpoint, leading to accelerated corrosion of carbon steel and some stainless steel heat transfer surfaces, tubes and/or fins. This form of corrosion is especially a concern when the GT is steam or water-injected.

b) **Acid dewpoint corrosion** occurs when trace quantities of sulfur in the fuels form sulfur trioxide $SO_3$, and can lead to localized sulfuric acid corrosion in colder sections of the preheater. Materials must be carefully selected to operate in this environment.

To avoid these forms of corrosion in the HRSG, it is sometimes economically justifiable to incorporate external water-to-water preheaters. The boiler feedwater leaving the deaerators can be used to preheat the incoming makeup water.

3.0 TYPES OF HRSGs

HRSGs types can be classified on whether they are a natural-circulation (NC), forced-circulation (FC) or a once-through (OT) design.

3.1 Natural Circulation HRSG

In natural-circulation HRSG units, the turbine gases flow horizontally past vertical tubes.

In the vertical evaporator tubes (as per the diagram in section 2.1), the density difference between water in the external downcomers and the water-steam mixture in the evaporator tubes is responsible for the circulation through the evaporator system.

Proper selection and sizing of evaporator tubes, downcomer, feeders & risers is required to ensure good circulation rates for the full range of GT, duct burner and HRSG operation that is expected.

In most natural circulation HRSG units, the economizer, superheater, reheater, preheater and integral deaerator sections (as applicable) are all supported and hung from the top of the HRSG structure. As the tube bundles grow thermally, they are allowed to expand vertically down.
Because of this construction and support method, the various sections of the HRSG can frequently be prefabricated in the shop, and shipped to site as a single component, minimizing field erection time and complexity.

Often, NC HRSGs for GT’s up to about 40–50 MW incorporate integral steam drums and internal downcomers, again minimizing field erection.

For larger NC HRSGs, the length of the steam drum dictates that the drum be shipped separately, and be erected at site. In this case, the evaporator tube banks will be located between top and bottom headers, and the drum is connected to the headers with risers to release steam, and with downcomers to be fed with water. Instead of a bottom drum, the bottom headers are interconnected with jumpers.

In a forced circulation HRSG, the gas turbine exhaust flows vertically past horizontal tubes. Steam-water mixture circulation through the evaporator tubes, and to and from the drum is maintained with a “forced circulation” pump.

Traditionally, most HRSGs in Europe have been specified as forced circulation. The claimed advantages of these units include decreased space requirements and faster start-up capabilities. However, their main disadvantage is the complex circulating pumps and their impact on operating costs and reliability.

3.3 Once-Through Steam Generator – OTSG

In a once-through steam generator (OTSG) the gas turbine exhaust flows past vertical and/or horizontal tubes. The unit is basically a single continuous serpentine tube in which all the functions of economizer, evaporation and superheating are carried out, without discrete drums.

In the tube bundle, the phase change zone from liquid to gas is free to move up or down throughout the bundle, depending on gas conditions (flow and temperature) and the operational load.

OTSGs eliminate the need for the steam drums, level controls, blowdown and recirculation systems. Startup times can be greatly due to the absence of thick walled pressure vessels and the steam drum water inventory, which would otherwise require heating.
The OTSG has all the benefits of the forced circulation HRSG, but without circulation pumps, and with decreased start up times. Because there is no blowdown from this type of HRSG, improved feedwater treatment systems, and condensate polishers may be required.

### 4.0 SINGLE vs. MULTI-PRESSURE HRSGs

#### 4.1 Single Pressure HRSGs

Up to this point only single-pressure level HRSGs have been discussed.

Viable steam outlet pressures can range from a low of 60 ~ 100 psig to a high end of 1500 psig. Outlet steam temperatures can range from saturated, up to within ~50 deg F of the GT exhaust temperature (which ranges from 850 to 1200 deg F depending on the unit), although temperature must be limited to 1050 deg F for material considerations.

A typical temperature profile for a superheated, single-pressure level, natural or forced-circulation HRSG shows the final outlet temperature, pinch point, saturation and approach temperatures, and superheater, evaporator and economizer concepts that have been previously discussed.

In this case, a high pressure HRSG has been shown, to illustrate that there is a great deal of additional energy available after the economizer – which is potentially wasted.

#### 4.2 Multi-Pressure HRSG

To recover this additional energy, the unit can be designed with multiple pressure levels – the assumption is made that the additional steam has a customer such as a process user, deaeration, feedwater preheating, gas turbine steam injection and/or a multi-inlet pressure (admission) steam turbine.
The following configurations of HRSGs are possible, arranged in order of increasing cost and complexity:

- Single Pressure
- Dual Pressure
- Dual Pressure with Reheat
- Triple Pressure
- Triple Pressure with Reheat

For each pressure level used, the relative location of the economizer, evaporator and superheater in the gas path are maintained. However, sections of each different pressure levels may be located in between some of the common pressure level sections so that a nearly parallel relationship between the temperature gradients for the gas side and steam/water side is achieved. This is best illustrated in the above temperature profile.

### 5.0 UNFIRED HRSGs vs. FIRED HRSGs

#### 5.1 Unfired HRSGs

When the available GT exhaust energy, the consequential HRSG steam production, and the steam requirements are well balanced, an unfired HRSG can be selected.

The performance (steam output) of the unfired HRSGs will be driven by the GT’s operating conditions, and if for some reason additional steam is required, it would have to be provided by an external source, e.g. auxiliary boilers, existing boilers, etc., or by duct firing.

#### 5.2 Fired HRSGs

The exhaust gases from a typical GT include from 14% ~ 16% oxygen by volume, which makes it possible to locate a supplemental burner downstream of the gas turbine exhaust – frequently called a duct burner.

These duct burners act to raise the gas temperature approaching the superheater or evaporator (as applicable) of the HRSG. Since steam production is a function of temperature differential, steam production will increase. The incremental steam is produced very efficiently (compared to a conventional boiler) since the gas turbine effectively preheats the combustion air, saving the additional fuel required to heat that air.

A gas-fired duct burner consists of several burner rows mounted inside a steel frame. Each row comprises a gas distribution pipe with pre-mounted flame stabilizing shields of refractory steel. The duct burner is designed to minimize gas side pressure drop, usually around 0.5” H2O. A uniform temperature and flow profile in the duct upstream of the duct burner is crucial to ensure a uniform temperature downstream and emissions within predicted limits. HRSG manufacturers flow model all steam generators incorporating duct burners to try and prevent any problems downstream of the duct burner.

Although duct firing can easily double the steam production of an unfired HRSG, there are also design implications such as:

- Higher cost superheater tube and fin material.
- Increased superheater, evaporator and economizer surface area requirements.
- Potential for economizer steaming, i.e. too much economizer surface area for unfired operation.
- Longer HRSG inlet duct to allow for complete combustion of the supplemental fuel.
- Increased heat insulation requirements on HRSG ducting walls.
- Burner management control system for duct burner.

The following single-pressure level HRSG temperature profile illustrates the affect of duct
burner firing from the original 900 deg F GT exhaust temperature, to 1200 deg F HRSG inlet.

In addition to supplementary fired HRSGs, there are two other possible variations:

a) A fully fired HRSG is a unit having the same amount of oxygen in its stack gases as an ambient, air fired power boiler. The HRSG is essentially a power boiler with the GT exhaust as its air supply. Steam production can range up to six or seven times the unfired HRSG steam production rate.

Fuel requirements for the fully fired HRSG will usually be between 7.5% and 8% less than those of an ambient fired boiler providing the same incremental steam capacity. Although fully fired HRSG's provide large amounts of steam, few applications are found in industry.

b) In some critical process steam applications, fresh air fired HRSGs may be applied. Forced draft or induced draft fans, dampers and ducting are installed to allow the introduction of fresh ambient air in case the gas turbine stops.

6.0 POST-COMBUSTION EMISSION CONTROLS

The drive for low NOx and CO production in the prime combustion device, the gas turbine, was covered in the previous chapter.

In certain jurisdictions and non-attainment areas, additional post-combustion emission mitigation measures are mandated beyond the 8–9 ppm\text{VD} to 42 ppm\text{VD} NOx being achieved by gas-fired DLN, steam or water-injected gas turbines.

6.1 Selective Catalytic Reduction Systems

The most common post-combustion process applied to HRSGs is Selective Catalytic Reduction or SCR. Most new combined-cycle plants where stringent emissions limits exist (i.e. 5 ppm or less), are equipped with both DLN combustors on the gas turbine, and an HRSG incorporating an SCR system.

A conventional SCR module is located in the HRSG at a zone that will consistently operate between 550 and 750 deg F gas temperature. Controlled amounts of ammonia are injected into the exhaust gas to react with NOx molecules in the presence of a catalyst, to produce N2, O2 and H2O. Typical catalyst materials are vanadium, platinum and titanium.

Some low-temperature SCR systems that operate between 300 and 400 deg F have been developed and have been in operation for 3~5 years. They can be particularly useful for retrofits, where they could be located downstream of an existing HRSG (assuming real estate is available and existing stacks can be relocated, etc.).
High temperature SCR technology has also been developed that operates in the 800 to 1100 deg F range. These are possible retrofits to existing simple-cycle installations, mounted directly at the GT exhaust.

With the modern DLN gas turbine machines and SCR technology, a percentage of the injected ammonia can pass through the SCR catalyst bed unreacted (ammonia slip), since there are less and less NOx molecules to find. Ammonia slip causes as much environmental concern as NOx and CO emissions.

The use of sulfur-bearing gaseous and/or oil fuels in the gas turbine or duct-burner, can also create environmental and HRSG durability problems. Inhalable secondary particulates of environmental concern (PM$_{2.5}$ and PM$_{10}$) can be formed in the HRSG. And corrosion salts can be formed that attack the low temperature sections of the HRSG, leading to premature failure, if not regularly washed.

SCR catalysts lose activity over time, and must be replaced periodically, typically every 3 to 5 years.

6.2 SCONOx Catalytic Absorption Systems

A relatively new post-combustion alternative to SCR is a technology known as SCONOx. This technology is reputed to reduce stack NOx emissions to less than 1 ppm, removing ~100% of CO, and producing no PM$_{2.5}$.

The system utilizes a single catalyst located in a 300 to 700 deg F zone of the HRSG, and works by simultaneously oxidizing CO to CO$_2$ and NO to NO$_2$, and then absorbing NO$_2$ on to the catalyst surface through a catalyst coating of potassium carbonate. The CO$_2$ produced by the reaction is exhausted up the HRSG stack.

Catalyst must be regenerated periodically by passing a dilute hydrogen reducing gas across the surface of the catalyst in the absence of oxygen. Absorbed nitrogen oxides are broken down into elemental nitrogen and water, and exhausted up the stack instead of NOx. During regeneration, louvres are closed to isolate the catalyst from the exhaust gases.

Acknowledgements: Heat Recovery Steam Generators - Jim McArthur, IST
1.0 STEAM TURBINE CONCEPTS

1.1 The Basics of Steam

Steam is used in more of today’s power generation plants than any other working fluid. Compared to water (whose behaviour is predictable due to nearly constant volume), the physical properties of steam are much more complex. When any one steam property is changed, e.g. pressure, temperature, volume, energy or moisture, all the other properties will also change.

The Mollier Diagram has been developed to show this interrelationship of steam properties, and how they all fit together.

The Mollier Diagram’s vertical axis is Enthalpy (given the symbol H), which is usually measured in btu/lb. Enthalpy is a quantification of the usable internal energy which is contained in steam.

The diagram’s horizontal axis is Entropy (S), which is the energy in the fluid that is irrecoverable, at a molecular level, and is usually quantified in units of btu/lb.F.

Inside the diagram are shown lines of constant pressure, constant temperature, constant moisture, and the steam saturation line (below which the steam is wet, and above which the steam is dry and superheated).

1.2 Steam Energy Processes and the Mollier Diagram

All real steam energy processes, or “engines” can be plotted on a Mollier Diagram.

For example, to extract energy (btu) from a given pound of high pressure and temperature superheated steam at P1, T1 and H1, it is necessary to expand it to a lower pressure P2, which results in a new T2 and H2. The amount of energy released (i.e. the enthalpy drop) from the expansion process is H1 – H2, in btu/lb. In an engine, when 2540 btu are released in one hour, one horsepower (hp) is made for that hour. Similarly, if 3413 btu were released in one hour, one kilowatt (kW) is made for the hour.

If this expansion process could be conducted perfectly and without losses, the steam would expand along a true vertical line, i.e. isentropically. In practical fact, there are always losses associated with the expansion process, and all expansion lines will curve toward the right on a
Mollier Diagram. The more efficient the process, the more vertical the line.

It is obvious that the longer the expansion line and thus the greater the difference between H1 and H2, the more energy which will be extracted from the pound of steam. There are several ways to increase the length of the line, including:

a) increasing the initial temperature
b) increasing the initial pressure
c) decreasing the final pressure
d) increasing the expansion process’s efficiency.

Thus the push for higher pressure and temperature boilers and HRSGs, for decreased turbine exhaust backpressure and condensing pressures, and for higher efficiency units.

From the Mollier Diagram (and from more extensive steam tables), we have a good understanding of the thermodynamics of steam, how it can hold energy, how the addition or removal of energy causes changes to it, how the volume, pressure and temperature, density and energy content of it all relate to each other, how to put energy in and take it out with maximum efficiency, and what materials are suited to working with steam.

1.3 Steam Engines and Steam Turbines

Until the 1880’s, steam was expanded in piston engines, e.g. water pumps, stationary engines and locomotives. Where needed, vertical motion was converted to rotary motion via rods, linkages and beams. The practical temperature limit on these steam engines was about 500 deg F, after which point the lube oil used in the cylinders and valve gear cooked and would cause failures.

In 1884, Charles Parsons developed the first practical, modern high-speed steam turbine, which overcame many restrictions of steam engines.

In it’s simplest terms, the steam turbine consists of a container or casing with steam inlet and outlet connections, enclosing a shaft which holds disk(s), which hold blades or buckets at the periphery. Stationary nozzles, attached to the casing, manage the angle of direction of the steam approaching each rotating blade stage. Because the shaft is supported by bearings outside the casing, and the lubricating oil is no longer in contact with the process, temperatures in excess of 500 deg F are possible.

In the steam turbine, steam is expanded in one of two basic ways:

a) Nozzles are used to expand and direct the steam onto the blades, creating a high speed jet at a suitable angle to push them, i.e. Impulse design.

In an impulse design, all the pressure drop occurs across the nozzle, and no pressure drop occurs along the axial length of the rotating blade.
b) In the **Reaction** design, pressure drop and expansion occurs across both the rotating and stationary blades, and the blades behave like a high-pressure fire hose, rotating from the reaction that occurs due to the sudden change from high pressure to high velocity steam. In comparison to the impulse design, there is a small efficiency gain, however, maintaining rotating blade tip clearances is critical, or else high pressure steam will bypass the blade length, doing no work.

In practice, modern steam turbines combine these two basic blading designs (impulse and reaction) in varying degrees along their blade path, for cost and efficiency reasons.

The steam turbine’s inlet steam flow is controlled by a series of poppet valves, raised and lowered as required. Poppet, spool and/or grid valves are also fitted to manage the quantity of steam entering or leaving the various parts of the unit. All valves are managed by digital control systems utilizing speed, pressure and/or load feedback signals to position them.

In the remainder of this chapter, we’ll concentrate on steam turbine generator drive applications, although steam turbine mechanical drives are also frequently employed, e.g. pumps and compressors.
2.0 STEAM TURBINE EXHAUST CONFIGURATIONS

One of the ways to classifying steam turbines is by the pressure to which they exhaust to, which affects the turbine’s basic configuration, size, cost and power production.

2.1 Backpressure Steam Turbines

Backpressure steam turbines generally exhaust to a steam system that is reasonably above atmospheric pressure. At these pressures, the volume of steam is reasonably small, and consequently the blade path consists of relatively small nozzles and blades, relatively small cylinders, and the exhaust piping is relatively small.

Virtually any inlet-to-exhaust pressure ratio above 2:1 is acceptable (depending upon steam inlet temperature) for a backpressure turbine, although exhaust pressures above 200 psig are rarely used, because of the lack of heat (enthalpy) drop across the machine.

Since all of the exhaust steam flow from a backpressure unit generally goes to a process application, the ultimate capacity of the turbine is only limited by the amount of process steam needed. Having said that, backpressure steam turbines greater than 50 MW are relatively rare.

Typical overall isentropic efficiency levels for backpressure units are as follows:

- **Gear-drive** (i.e. 5000 to 7500 rpm): 73 to 78%, for flows 150 to 300 klb/hr.
- **Direct-drive** (i.e. 3600 rpm): 77 to 82%, for flows of 250 to 500 klb/hr.

2.2 Condensing Steam Turbines

Condensing steam turbines generally exhaust to a system or location that is significantly below atmospheric pressure, where pressure is measured in inches of mercury absolute, i.e. inch HgA.

At these pressures, the specific volume of steam is very high, and to pass the steam mass flow, much larger blading and nozzle areas are required.

Compared to backpressure units, the condensing turbine’s cylinders become very large near the exhaust of the unit, and will include an exhaust hood or diffuser to slow the steam in an orderly manner, to minimize exhaust losses.

The term condensing refers to the condensing system that is attached to the unit, which acts to condense (turn back to water, or condensate) the
condensing STG www.gepower.com

exhaust steam. In the act of condensing, the original volume of the exhaust steam decreases by several orders of magnitude, and since the process takes place in a closed vessel, a steady-state vacuum is created and maintained, which keeps the process going. Condensers are discussed in further detail later.

Conventional condensing turbines can be sized from as small as ~5 MW, up to the 1500 MW+ units employed in utility service. For combined-cycle plants, the maximum size tends to be 200~250 MW.

Any inlet pressure up to about 2400 psig is acceptable, while the maximum inlet temperatures are 900~1050 deg F.

Typical condensing pressures are 0.75 inch HgA for the coldest condensing systems, up to a maximum of ~10 inch HgA for air-cooled condensing systems at high ambient temperatures.

Typical overall isentropic efficiency levels for condensing units are as follows:

- **Gear-drive** (i.e. 5500 to 6000 rpm): 76 to 82%, for units up to about 25 MW.
- **Direct-drive** (i.e. 3600 rpm): 78 to 84%, for units 25 MW and greater.

The expansion diagram provides a simple comparison of the heat drop and power output for a backpressure and a condensing steam turbine unit.

Each turbine receives the same 250,000 lb/hr of 900 psig, 900 deg F steam, and is expanded either to 60 psig in the backpressure unit, or expanded to 1.5 inch HgA in the condensing steam unit.

The backpressure steam turbine produces about 16 MW, and provides process steam at about 405 deg F.

By comparison, the condensing steam turbine produces about 33 MW, but no process steam. The exhaust of the unit results in condensate from the condenser hotwell at about 92 deg F.
Westinghouse Single Auto-Extraction Backpressure Steam Turbine www.siemens.ca

Westinghouse Double Auto-Extraction Condensing Steam Turbine
3.0 EXTRACTION, ADMISSION and REHEAT CONSIDERATIONS

3.1 Extraction Steam Turbines

Extraction steam turbines provide a tapping point somewhere in the blade path downstream of the inlet valves & inlet control stages, where steam at an intermediate pressure can be obtained. Extractions can be either automatic (controlled), or uncontrolled.

In the controlled extraction machines, a series of poppet, spool or grid valves are installed at an intermediate steam chest. The valve position is manipulated by the control system so that some steam is forced out of the steam chest into the extraction piping at the required pressure, while the remainder passes to the lower pressure section of the turbine. This control can be delivered over a wide range of inlet flows, but is expensive because of the additional steam chest.

In uncontrolled extraction machines, there are no controlling valves at the tap-off point, and the delivery pressure will be a function of the amount of steam flow downstream of the tap-off point, i.e. higher flows will deliver high tap-off pressures, and vice versa. Uncontrolled extractions can be applied where the downstream process can tolerate pressure variations.

Some specific steam turbine units are fitted with a series of uncontrolled extraction ports, each with a external control valve. As the unit’s through-flow changes, the external valves are sequentially opened to deliver steam from the appropriate pressure port, and the unit behaves like a controlled extraction turbine, without the cost of a large steam chest.

Controlled and uncontrolled extraction systems can be applied to condensing and to backpressure steam turbines. More than one controlled and/or uncontrolled extraction can be provided on a single unit – as seen on the previous image of a GE condensing steam turbine generator.

3.2 Admission Turbines

Admission steam turbines are similar to extraction units, where a tap-in point is provided in the blade path downstream of the inlet stages.

Admissions can also be either controlled or uncontrolled as discussed for extraction units. The only difference is that admission piping systems require a trip & throttle valve, to prevent the feeding steam to the unit after a turbine trip, which might cause overspeed of the unit.

Controlled and uncontrolled admission systems can be applied to condensing and backpressure steam turbines. Some units can be configured as admission/extraction, and can thus either accept or deliver steam depending upon extenuating circumstances outside the turbine.

3.3 Reheat Turbines

Reheat units are a special condensing turbine configuration applied to large, high inlet pressure power plants. These have been common for large non-nuclear utility units for decades.
Reheat is becoming common for the largest combined-cycle plants, where inlet pressures and temperatures are rising to those commonly found in traditional fossil fuel powerplants.

In the reheat turbine, all the steam expanded through the high-pressure section of the turbine is removed from the unit, and taken as cold-reheat to the reheat section of an HRSG (or boiler in the case of utility generation).

There, it is reheated back to the original inlet steam temperature, and admitted back into the intermediate-pressure section of the steam turbine as hot-reheat, at the same temperature as the inlet although at a lower pressure.

Reheat maintains acceptable levels of wetness in the LP section of the turbine, while the increased length of the expansion line results in increased power and efficiency albeit at a significant increase in equipment cost.

Multi-cylinder configurations are best suited for reheat applications, because of the large diameter of the piping takeoffs and inlets, and for thermal considerations.

Additional reheat and intercept valves are required to control the unit during startup and transients.

Inlet / reheat temperatures tend to be 1000~1050 deg F, with initial pressures varying from 1250 ~ 2400 psig, and reheat pressures 400 ~ 600 psig.
3.4 Combined Gas – Steam Turbine Drivetrain

In an effort to further reduce plant costs, to slightly improve efficiency, and to simplify electrical connections, major manufacturers of large gas turbines and steam turbines offer a combined, common-shaft drivetrain.

The gas turbine is connected to one end of a generator, and a steam turbine to the other end.

A clutch is mounted between the steam turbine and the generator, to allow the gas turbine to commence operation, and the HRSG to commence delivering steam, prior to starting the steam turbine.

The smaller gas turbine manufacturers also offer combined gas & steam turbine drivetrains, employing similar principles.
4.0 TYPES OF CONDENSING SYSTEMS

The actual type of condensing system that can be installed will vary, depending on the environmental conditions. Condensing systems can be broken down into the following categories:

- Water-cooled surface condensers and wet condensing systems
- Air-cooled condensers
- Alternative condensing systems

4.1 Water-Cooled Surface Condensers

The most efficient, and thus most popular condensing systems are the water-cooled surface condenser systems – popular in areas where a large amount of cooling water is readily available, and where governing environmental agencies permit their use.

Water-cooled surface condensers can be further categorised by the means in which the heat rejection function is accomplished:

a) Surface Condensers with Once-Through Cooling Water Systems, which are the simplest type and typically offer the best performance for the least amount of auxiliary power consumption. They are typically applied when the powerplant is located close to an adequately sized river, lake or to the sea, where a pumphouse can be constructed.

Cooling water is pumped directly from the cooling water source into the condenser inlet water box. The tubesheet connects the waterbox to the tubes, allowing cooling water to pass through the inside of the condenser without contacting the steam-condensate circuit. The outside surface of the tubes makes up the condensing surface and it is this surface area and the cooling water temperature that dictates the condenser performance.

Exhaust steam from the turbine enters the condenser chamber and condenses upon contact with the outside surface of the cold tubes. The latent heat discharged by the condensed steam is transferred through the tube walls into the cooling water. Because the condenser chamber is a fixed volume and sealed from the ambient air, the state-change from saturated-steam to saturated-liquid (i.e. condensate) occurs at a constant pressure and a constant temperature. The large reduction in the fluid specific volume causes the pressure in the chamber to drop to a vacuum condition, to achieve a steady-state equilibrium.
Condensate is collected in the “hotwell” at the bottom of the surface condenser and is then pumped out for reuse in the boiler or HRSG feedwater cycle. The cooling water exits the tubes at an elevated temperature and discharges into the outlet waterbox and eventually returns to the cooling source.

To maintain a vacuum in the condenser, a vacuum pump or steam ejector system is used to remove non-condensibles such as air and other gases that infiltrate the system. The vacuum pump or a large “hogging” ejector is also used to evacuate the condenser at start-up.

Single-pass and multi-pass surface condensers are both available to suit site requirements. In some cases, multiple inlet/outlet configurations, with divided waterboxes are desired, to allow for flexibility in operation.

While a once-through cooling water cycle is simple and cost effective, it is not always environmentally acceptable to discharge the warm water directly back into the cooling water source. It is quite common for the environmental agencies to restrict the return water temperature discharging back into the original source to 18 deg F rise or less.

b) **Surface Condensers with Evaporative Cooling Towers** eliminate the discharge of hot water back into the original cooling water source, by providing essentially a closed-circuit system for the condenser cooling water. The circulating water is obtained from the basin of an evaporative cooling tower, and after being heated in the condenser circuit, is returned to the cooling tower for cooling.

In general, air is forced or induced through the bottom of the cooling tower, and extracts heat by evaporation from the cooling water stream as it rises and eventually discharges through the top of the tower.

Some circulating water is constantly lost to evaporation, drift and blowdown from a cooling tower, thus an amount of makeup water is still required. In addition, chemical treatment of the circulating water, and the blowdown water is usually required. All cooling towers will generally exhibit a visible plume during operation, unless specific plume abatement measures are taken.

There are two basic types of evaporative cooling towers available:

i) **Natural-draft cooling towers**, with distinctive hyperbolic shape. The cooling air is induced naturally by the hotter (less dense) air in the tower drawing the colder (denser) air from the outside at the bottom.

ii) **Mechanical-draft cooling towers**, that create an airflow through the tower by using either induced-draft fans (fan at top as below) or forced-draft fans (located at the air inlet to the tower, and susceptible to icing).
4.2 Direct Air-Cooled Condensers

In areas where water is extremely scarce, and powerplants cannot afford even small amounts of cooling water evaporation, sufficient cooling/condensing capacity must be provided without direct air contact with the cooling water circuit. This can be accomplished by using a dry or direct air-cooled condenser.

The turbine exhaust steam enters a central plenum/pipe located above a series of finned tubes, sloped down towards a condensate collection piping system, generally in an A-frame configuration. Cooling fans push ambient air across these finned tubes, causing the steam to condense as it progresses toward the hotwell.

4.3 Alternative Condenser Systems

In addition to the direct-wet and direct-dry condensing systems above, there are several alternate condensing configurations which can be employed, including:

- Parallel or hybrid condensing systems – a combination of a direct air-cooled condenser and a surface condenser and mechanical-draft cooling tower.
- Wet-surface air-cooled condensers.
- Cooling ponds – one to two acres of surface area required, per MW of condensing.

Please refer to the references for additional documents that describe these alternative systems in more detail.
4.4 Condenser Performance Comparison

The figure illustrates the relative performance of the various types of condensing systems available vs. dry bulb temperature, from the once-through surface condensers with river or lake cooling, through surface condensers with evaporative cooling towers, wet-surface air cooled condensers and air-cooled condensers.

ACKNOWLEDGEMENTS & REFERENCES

1. Modern Steam Turbines – Lindsay Scott, Siemens-Westinghouse Power Corporation.


1.0 PROJECT EXECUTION PROCESS

Every cogeneration and combined-cycle power project is different, for any number of reasons. As a way of illustrating the possible process and methodology of executing a project, we are going to follow a hypothetical 125 MW Combined-Cycle project from its conception to the project’s first synchronization, and review the options that may be available during the course of the work.

In general, all power projects will go through the following general phases:

a) Identifying the opportunity
b) Developing the project
c) Planning and financing
d) Design and construction
e) Startup and commissioning

2.0 IDENTIFYING the OPPORTUNITY

All projects start in the imagination of someone, who sees a need to do something better, to decrease costs, to improve efficiency, to minimize risk, and/or to improve the competitiveness of their own, or someone else’s industry.

a) The initial proponent could be someone from within a plant organization, for instance, a technically astute boilerhouse superintendent or chief engineer, an electrical superintendent, a middle level manager, or a financially astute executive who is responsible for utilities, costs and/or operations.

For example, a plant may realize that their electrical costs are increasing or are becoming uncertain, that outages are causing expensive disturbances in their process, and/or that their fuel supply situation is changing.

In addition, some of their existing steam production facilities require expensive upgrade due to age. Instead of simply planning the upgrade or replacement of their boilers and auxiliary equipment, the plant may instead decide to study the viability of replacing portions or all of their electrical purchases with self-generation, and all or part of their steam generation via a cogeneration plant.

b) Or instead, the initial proponent of the project may be an experienced third-party developer who recognizes the energy use pattern of an industry or plant, realizes that some opportunity exists, and approaches the organization with a general offer to examine the opportunity on their behalf, and to develop and execute a project if it proves economically viable – both to the developer and to the “host”.

These third-party approaches are usually made at a corporate or executive level, with the plant’s technical staff brought in to assist management with evaluation of the offer. Letters of intent, exclusivity and/or confidentiality may be signed, with 3–12 month expiry dates, that give the developer the comfort that their efforts and costs may not be wasted.

In our hypothetical case, a large mill consistently experiences several electrical outages each year due to local lightning strikes or a weak distribution system. Each plant electrical outage results in loss of one-half day of production, and increased costs due to cleanup and process startup.

In addition, the mill has been operating for several decades, and although the plant’s boilers work fine, the mill is facing pressure to improve emissions. The local newspaper publishes an article about a new natural gas pipeline that is in the planning stages, that will run within 40 miles of the plant.

The mill requires about 80 MW of electrical power and has a continuous load of about 1,000,000 lb/hr of steam at varying pressure levels. The existing steam boilers operate at 600 psig, 750 deg F.

The boilerhouse superintendent, who once attended a cogeneration and combined-cycle principles workshop, realizes that his plant may soon be a good candidate for an efficient, low-emissions natural gas-fired cogeneration plant. He approaches his boss about the idea, who says “Sure, look into it”.

The superintendent now faces several choices – either look into it himself, or ask for outside help. He could get free advice from the gas turbine, steam turbine or other salespeople that they met.
at the workshop, or he could commission a study by an experienced consulting engineer, or he could approach a developer for initial assistance.

Each approach obviously has its merits and drawbacks, which might affect the course of the subsequent project:

a) The equipment salesperson will usually have an interest in ensuring that their product is the basis of the subsequent project. In addition, the salesperson may not be sufficiently knowledgeable to address all the technical and financial aspects of the potential project. The salesperson sometimes recommends or employs a consulting engineer to assist in the first-cut evaluation.

b) In general, a thermal power consulting engineering firm will treat any study without a bias for or against any particular cogeneration technology, and will act conscientiously to examine all positive and negative aspects of the opportunity.

c) A developer will frequently employ a consulting or in-house engineer to assist in evaluating the opportunity. However, the primary interest of the developer will be their own profit. While they will strive to make the project financially attractive to the host, they may create a project that is larger or smaller than what is actually best for the host. Due to economic, financing or risk management considerations, this may be perfectly acceptable to the host.

In any case, in order to frame the opportunity in terms of reference that are familiar to the power generation industry, any assisting party is going to require fairly accurate top-level technical information about the plant, including power requirements, duration and capacity factors; electrical connection details; process steam loads; fuel usage; site and plant layout details, etc.

In addition, some confidential sharing of financial information may be required, depending upon if the mill wishes to have the economic analysis done by themselves, or by the assisting party.

In any case, the task of the assisting party would potentially include: the basic identification of the type of cycle and project, i.e. bottoming or topping cycle; steam-turbine based; gas turbine / HRSG based; or combined-cycle based; capital and O&M costs for viable alternatives; approximate schedule for viable alternatives; impact on existing operations; potential project execution methods; and identification of potential hurdles or show-stoppers.

In our hypothetical case, the superintendent has commissioned a study by a consulting engineering firm. They have examined the performance and costs of steam turbine; gas turbine and HRSG, and combined cycle plants; and identified that no significant technical hurdles or show-stoppers existed mechanically or electrically.

Realizing that the existing boilers were initially a high-pressure design (900 psig), but were operated at 600 psig only for convenience, the engineers recommended a relatively low-cost backpressure steam turbine project, with the boiler operating pressure raised to 900 psig (with boiler temperatures raised), and with extraction and exhaust steam provided at the required pressures to the process steam system.

The backpressure STG provides a good portion (but not all) of the plant’s required electrical load, at high efficiency and the design would allow the critical services portion of the mill to operate islanded if the main utility tie is lost due to local lightning strikes or system instabilities.

The report includes preliminary mass & heat balances that show the configuration would qualify for Class 43 accelerated depreciation. A spreadsheet is provided, demonstrating that the project would save the mill several million dollars per year, and based upon typical costs for backpressure steam turbine plants, would meet the mill’s normal hurdle rate for investments.

The study did include that larger gas-turbine based cogeneration plants would offset more of the plant’s electrical load, might provide more electricity security, while replacing all or most of the existing boiler capacity, and while providing the same amount of steam. However, due to the greater price associated with these projects, the superintendent, the plant accountant and the engineer agree that this is not the way to proceed.
3.0 DEVELOPING the PROJECT

The hypothetical conceptual report is finalized and submitted by the boilerhouse superintendent to his boss, and is presented to plant management and a committee from head office. If management agrees with the preliminary performance and financial figures, they may choose to proceed with the internal approval and financing of the project, through their traditional corporate path.

If instead, they evaluate that the project, while saving millions, may be too expensive to finance by themselves, and/or that they want to keep the mill operations continuing normal during electrical supply outages, an alternate path may be taken.

A third-party developer could be sought by a formal RFP process, where the technical terms of reference of the opportunity are stated, and proponents are invited to submit technical and financial proposals, in any way, shape and form. Typical developers could include local or regional gas or electrical utilities, or their unregulated arms; domestic or international independent power developers (IPP’s), and/or various equipment manufacturers or their development arms.

A dozen replies are received and evaluated by mill and corporate management and the consulting engineer.

One particular developer offers a project that would include a large combined-cycle power plant located on property adjacent to the mill, consisting of two 55 MW gas turbines, fired HRSGs, plus one condensing steam turbine. The total plant output would be rated at about 125 MW, and would qualify for Class 43.

The developer offers two gas turbines and fired HRSGs so that if one GT experiences a forced outage, or is shutdown for maintenance, the other GT and the steam turbine can supply the mill’s power and steam requirements.

All the process steam would be sold to the host at prices below their current steam production costs. All the electrical power required by the host would be sold at a fixed rate. All sales prices would escalate based upon consumer indices or gas pass-through costs, whichever was worst.

The attractiveness of the offer depends upon the success of the developer selling the excess power from the plant into a merchant power environment.

The developer would develop and finance the project, obtain all approvals, take all risks associated with sale of merchant power, and obtain the necessary gas supply and transportation arrangements.

The developer’s proposal includes a pro-forma economic analysis of the project, from the host’s perspective. The pro-forma demonstrates on a month-by-month basis, for a project life of 20 years, the host’s “as-is” future operating cost if an agreement was not entered into, and the host’s costs if an agreement was signed and the project was built. This pro-forma demonstrates that the agreement would save the mill millions of dollars per year, although not as much as the original backpressure steam turbine project. The developer rarely shows the host his own pro-forma.

This developer’s offer looks attractive to the mill, and they sign a letter of intent with the developer and commence detail discussions and preliminary financial negotiations. Potential issues such as new vs. refurbished equipment; battery limits; site access and location; existing plant and boilerhouse staff; backup power supply; existing boiler capacity and retirements; use of existing condensate and water treatment systems; required mill improvements; risk management; etc. are straightened out. The mill can also discuss the potential for equity sharing in the plant, risk sharing and the potential for profits from the developer’s merchant power income.

At this point, the developer (and the mill, if they are equity sharing) will commence an exercise to firm up capital and operating costs, and to improve the accuracy of their preliminary pro-formas.

In general, an engineer and potentially a contractor will be engaged to prepare a conceptual cycle and plant design, to obtain specific estimates for the major equipment, to prepare preliminary designs and costs for the remainder of the plant and for the interconnections. Schedules and plans for the work will be prepared.
What is clear from experience is that executive champions or sponsors for the project must now exist, both within the mill’s management and/or the corporate head office, plus within the developer’s organization, who firmly believe in the project and who will support the project through its remaining torturous steps. Without the appropriate sponsors to guide and to push the project through its upcoming difficult phases, the project and team may stall or stop at key decision points.

4.0 PLANNING and FINANCING

As the project develops further, the developer will solicit the services of engineering, legal and project development firms (if they don’t exist in-house), to assist with the further planning, development and formation of the project.

Initial contacts, and initial technical and financial submissions will be made to fuel, electrical and services (water, sewer, etc.) authorities and municipalities that will be impacted by the project. Initial studies may be required to assess the cost of infrastructure changes, which will have to eventually be paid for by the project’s developer.

Where necessary, land rights discussions and/or easements will be discussed and negotiations commenced. Initial environmental impact submittals will be made.

Protocols and schedules will be established for the submission of further information and studies as the project design firms up and is finalized.

Business plans will be drawn up for the purposes of soliciting construction and future operational financing. The plans outline the host’s needs; site considerations; the developer’s view of the merchant power environment and his plan to sell into it; the conceptual design of the plant; pro-formas; the range of expected performance, power and steam production, fuel consumption; anticipated range of O&M costs; plant staffing; design and construction methodology; project risks and corresponding risk management methods; the proposed terms and approximate sale prices for energy (electricity and steam); and suggested / requested financing terms, etc.

The business plans will be submitted as RFP’s to financial institutions such as banks, insurance companies and venture capital firms or funds, etc. As necessary, these organizations will form partnerships to secure adequate capital, and may engage the services of a mutually satisfactory lender’s or bank’s engineer, to assess the technical adequacy of the information provided, and to evaluate the project risks and technologies.

The terms offered by the financial institutions are negotiated, and may occasionally result in changes in the debt and equity positions proposed by the developer, changes in the proposed project execution method, and/or in changes in plant design due to financial constraints, etc.

In our hypothetical case, terms for construction financing are agreed to with one financial institution, and the project can now proceed into the design and construction phase.

Some organizations may choose to finance the project on balance sheet, without seeking external financing – the original backpressure steam turbine project could possibly have been handled this way if the corporation had the financial ability to handle the associated cash flow requirements.

5.0 DESIGN and CONSTRUCTION

The project proponents will already have planned their choice of how to get the project designed and constructed, and will have considered:

a) Traditional engineering and contractor approach – in this form, the project proponent would solicit and hire an experienced detail design engineer to prepare the design, and solicit a general contractor to construct the works.

The engineer would finalize the cycle design; electrical, utility, interconnect and environmental aspects; prepare specifications for the project major equipment (e.g. gas turbines generators, HRSGs and duct-burners, steam turbine generators, condenser, switchgear, step-up transformers, switchyard, gas compressors, DCS, etc.); integrate all purchased components into a complete plant design; design the subsystems and interconnects; and prepare construction drawings and documents for bid.
This detail design engineer would most likely be the same engineer that has been working with the proponent to date, who understands the project and the players the best.

The awarded general contractor would construct the project in accordance with the engineer’s documents and drawings, hiring trades and specialty subs as required for the work.

In general, under this traditional approach the proponent (owner or developer) or would accept the majority of the risks for delivery, schedule and/or performance of the major equipment and/or plant.

The above traditional approach is quite acceptable for many small to moderate cogeneration projects, and could easily have been utilized for the backpressure steam turbine generator project that was first proposed to the mill’s management.

Variations of this traditional method exist, including where the proponent breaks out the engineering and contracting into several packages, each to be handled by a different organization; engineer-contractor joint ventures; or where a single firm has both the necessary design and construction contracting experience to carry out most or all of the project for the proponent.

In addition, the following approaches can be taken, which are similar to the above, but with an operating twist:

e) BOO – Build-Own-Operate – in which the turnkey contractor will own (all, or a portion of the plant) and operate the plant on behalf of the owner after completion and startup.

f) BOOT – Build-Own-Operate-Transfer – in which the turnkey contractor owns and operates the plant (and obtains the revenues) until it proves to operate to the satisfaction of the final purchaser. At this point, the purchaser exercises his option to purchase the plant at a pre-agreed price.

For these “turnkey” approaches, the proponent generally engages the services of an Owner’s Engineer, to define the work properly, to review and oversee the turnkey contractor’s work and performance, and to correspond with the lenders and the banks engineers as required.

The Owner’s Engineer accurately defines the basis of the project; the project’s general and specific system and performance requirements (system design basis; power, heat rate and emissions requirement; delivery, schedule and milestone dates and achievements); the codes and standards to be used; establishes liquidated damages rates and remedies for any lack of performance, failure to deliver, not-fit-for-purpose, etc., and defines the performance tests and protocols to be used to verify the suitability and completion of the plant.

In large projects, such as our hypothetical 125 MW project, the terms of financing may require quite a different approach due to risk management, including:

b) Turnkey approach. This approach generally requires a contractor to take the fundamental lead in designing and constructing the project. The approach will lead to less owner control, slightly higher project costs, but decreased owner risk.

For the proponent, the following approaches share many similarities, with varying levels of Owner participation and Owner risk absorption:

a) Turnkey

b) EPC – Engineer-Procure-Construct

c) EPCM – Engineer-Procure-Construct-Manage

d) Design-Build, either led by a contractor or led by an engineer.

In our hypothetical 125 MW combined-cycle plant, a turnkey contractor has been hired by the developer to complete the design and construction of the project. The contractor has signed a negotiated contract that includes accepting the financial risk for the performance, and for the scheduled completion of the plant.

During this phase, the owner / developer will act to complete all electrical, steam and fuel sales and purchase contracts, and operating agreements; and arranges for operational financing of the plant, to take over from construction financing at the appropriate time.
7.0 COMMISSIONING and STARTUP

As a project nears completion of construction, the systems and major pieces of equipment must be energized, checked out, commissioned and started up in an orderly manner.

A commissioning manager and commissioning team should be appointed, either from within the developer’s organization, or from a specialist firm experienced in the startup of thermal power generation plants. Where the plant operators have been hired in advance, or if they come from the host’s boilerhouse staff, these operators should be included in the team, in order to begin learning about the plant.

There is always a hazy and contentious area on a project, in attempting to distinguish when the construction of a particular system or machine finishes, and when the commissioning of it starts, especially when these tasks fall to two or more separate contractors. From experience, it is best to provide a separate section in the original turnkey contract, laying out the rules, and providing definitions, responsibilities and protocols for:

a) Mechanical Completion.
b) Checkout.
c) Training
d) Startup.
e) Commissioning.
f) Initial Operation.
g) Functional Testing
h) Commercial Operation.
i) Correction of Deficiencies.
j) Performance Testing
k) Final Acceptance.

During construction, the commissioning team was responsible for review of all project instruction book materials; review of all contracts placed with suppliers and contractors; and from that, to prepare detail commissioning, startup and testing schedules, protocols and procedures.

Equipment and plant testing procedures are frequently pre-defined in the contractual documents and in this case are generally reviewed by the bank’s engineers.

The commissioning team should be responsible for scheduling all necessary inspections of the plant, and for notifying the relevant authorities in advance of testing dates.

The future operating staff will require a training and education process to prepare them for the operation of the new plant, equipment and systems. As a minimum, this consists of specific equipment vendor training courses; general and system-specific training by the design engineers; and familiarization with the electrical, steam & fuel supply contracts, and operating agreements that will govern plant operation.

In the case of our hypothetical 125 MW combined-cycle power plant, the turnkey contractor was responsible the commissioning, startup and testing of the plant. The gas turbines, HRSGs and steam turbine were purchased with site assistance services provided.

The existing boilerhouse staff became part of the final operations staff, and accordingly, became part of the commissioning team, falling under the management lead of a commissioning specialist that has been hired by the turnkey contractor.

The commissioning team arranged for the necessary specialty subs to conduct system testing that the team can not handle, e.g. relay and switchyard testing, steam blows, etc.

The commissioning team worked in concert with the construction crew, commissioning and starting up each system sequentially, backfeeding and starting the plant, achieving first synchronization of each generator, then operating systems and equipment to maximum capacity, conducting testing; arranging for the remedies in design and equipment as necessary.

Final performance tests were performed for the major equipment, and for the plant as a whole, with the bank’s engineers as witness, and the plant was accepted.

After acceptance, the owner / developer arranged for the transfer from construction to operational financing, and proceeded with the plant operation.
APPLICATION EXAMPLES
SIMPLE-CYCLE POWER PLANTS (Non-cogeneration)

Abu Kamash Emergency Power Plant
Six 16 MW Westinghouse W191G Econopacs, installed to supply 60 MW of emergency power to a petrochemical plant on the coast of Libya, in the event of failure of the main electrical transmission system.

Executed as a turnkey contract by Westinghouse Canada – Turbine and Generator Division (now Siemens-Westinghouse) for the Libyan Secretariat of Electricity (SOE).

Songo-Songo Emergency Power Plant
2 x Stewart & Stevenson LM6000PA gas turbine generators fired on jet fuel, located in Tanzania. Air inlet systems include chilling coils for high-ambient temperature operation.

Units provide peaking power to the national grid during the dry season.

Project executed fast-track by a joint-venture of Stewart & Stevenson (now GE S&S Energy Products), Stone & Webster Canada and South African contractor.

Owned and negotiated by joint-venture of Ocelot and TransCanada Pipelines (OTC), with World Bank financing.
COGENERATION PLANTS

Whitby Cogeneration Plant

A joint-venture between Atlantic Packaging and Westcoast Power Inc.

Located on a site adjacent to the AP’s Whitby paper recycling plant. 50 MW of electricity sold to Ontario Hydro, and process steam to Atlantic Packaging. The existing boilers were retired.

The plant consists of one Rolls-Royce Trent aero-derivative gas turbine generator, with an unfired, single-pressure level once-through steam generator (OTSG) provided by Innovative Steam Technologies (IST), plus gas compressors, water treatment systems, auxiliary boilers, main step-up transformer and switchyard.

The OTSG is designed for future conversion to triple-pressure configuration, to provide steam to a future steam turbine generator. The top transition piece and the stack would be temporarily removed, and the new tube sections inserted.

www.otsg.com

H.J. Heinz Cogeneration Plant

Two natural gas fired Allison 501-KB5 gas turbine packaged by U.S. Turbines (now Rolls-Royce).

GTG’s operated at base load, providing approximately 2 x 3.8 MW of electricity at 4.16 kV to the food processing plant in Leamington, Ontario.

Two fired Deltak D-type water-wall HRSGs with economizers, originally operating at 165 psig and now operating at 300 psig to provide steam for steam injection to the KB5’s and for a future 500 kW backpressure steam turbine generator.

Unfired steam production from each HRSG of about 22 klb/hr, and up to about 62 klb/hr when duct-fired to ~2050 deg F with 40 x 10⁶ btu/hr of gas.
Union Gas Power
Ford Windsor Steam Turbine Generator Project

Cogeneration project developed and executed by Union Gas Power at existing Ford Windsor boilerhouse, to provide steam and electricity to new engine plant.

Ford’s existing gas-fired boilers supply up to 350 klb/hr of inlet steam to a new GE auto-extraction condensing steam turbine generator, with LP uncontrolled extraction. 28.5 MW of electricity and up to 225 klb/hr of 110 psig extraction steam provided to the engine plant.

COMBINED-CYCLE PLANT

Trans-Canada North Bay Enhanced Combined-Cycle Power Plant

OTSG fitted to the exhaust of an existing Rolls-Royce RB211 gas compressor drive unit (in the foreground) in the existing gas compression station, delivering HP and LP steam to adjacent combined cycle plant via long steam lines.

Pratt & Whitney FT8 gas turbine generator, with dual-pressure OTSG, located in new building (background). ABB admission steam turbine generator accepts steam from both HRSGs.

The STG exhaust is condensed via GEA Power Cooling Systems dry air-cooled condenser.
COMBINED-CYCLE COGENERATION PLANTS

Lake Superior Power Plant – Joint-Venture: Westcoast Power and Great Lakes Power

Two 40 MW GE LM6000PA gas turbine generators fired on natural gas with fuel oil as reserve. Air inlet heating for winter operation.

Two fired Deltak dual-pressure HRSGs with bypass stacks, delivering HP and LP steam to 25 MW GE auto-extraction/admission condensing steam turbine generator. Surface condenser with river water circulating cooling water system.

105 MW of electricity sold to Ontario Hydro, and process steam sold to St. Marys Paper via 1 mile steam line.

Plant output via 115 kV switchyard and dual-circuit 115 kV submarine cabling through power canal to nearby Great Lakes Power Clergue hydraulic generating station.

Iroquois Falls Power Plant – Northland Power

Two LM6000PA fired on natural gas. Two Babcock and Wilcox HRSG, plus GE steam turbine generator and Volcano standby boilers. Condensing by river water.

Power output sold to Ontario Hydro, and process steam to Abitibi Paper pulp and paper mill.
**Lockport Power Plant – Lockport, New York**

184 MW combined-cycle cogeneration plant partially owned by Calpine Inc.

Process steam and power sold under the terms of an energy service agreement to the adjacent General Motors Harrison radiator plant. Remaining power is sold to NYSEG.

Three GE Frame 6B gas turbine generators, with fired HRSGs and a GE 70 MW axial exhaust steam turbine generator. Condensing by surface condenser and cooling towers.

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**Independence Power Station**

1040 MW combined-cycle cogeneration plant owned by Sithe Energy in Oswego, New York.

At the time, this was the largest IPP plant in the USA, and is now scheduled for doubling in size.

Four GE 7FA gas turbine generators, HRSGs and two steam turbine generators. Condensing by cooling towers.

Process steam and electricity sold to local Alcan Rolling Products. Remaining electricity sold to local utility.
1.0 O&M CONCEPTS

The fundamentals of good O&M for any type of cogeneration or combined-cycle power plant are well understood, and they include:

a) Operations – qualified, skilled, experienced and trained personnel are required to operate the equipment within the intent of the designers and the limits of the equipment; to the requirements of the owners; and within the mandates of various energy services and operating agreements.

b) Maintenance – regular, preventative and condition-based maintenance of all the major, minor and related equipment and systems in the power generation facility.

c) Spare Parts and Consumables management – to ensure that the right type and correct amount of spare parts are available, and to ensure that normal consumables are on-hand.

d) Plant Management – an organizational structure to manage the plant operations and support staff appropriately; to report to owners and authorities; to correspond with suppliers, energy service clients, lending institutions and regional authorities and utilities; as required on behalf of the owner.

e) Specialists – develop relationships with companies capable of quickly performing specific tasks, providing expensive equipment, or providing services that would otherwise be uneconomical to support within the plant organization itself.

These could include anyone from relay testing firms; crane and tool rental firms; local millwright, instrumentation and high voltage shops; laser alignment specialists; bolt tightening specialists; blade and part rehabilitation shops; and engineering firms.

2.0 O&M OPTIONS

Depending upon the size and configuration of the plant, the ownership structure and the energy supply and operating agreements, the owner of a cogeneration plant has several O&M strategies and combinations thereof available, including:

a) Operate & maintain the plant themselves – in which case, they would hire and train the operations staff themselves, and perform all plant management functions. This is typical for the smaller to medium size plants.

b) Contract out for operations and/or maintenance, or for organizing operations – specialist firms exist that will work with the plant designers, the major equipment suppliers, the owner, and the commissioning team to hire and train the operations staff, and to prepare the plant’s procedures.

These procedures could include operating procedures; preventative maintenance and condition based maintenance procedures and programs; the plant’s emissions monitoring and reporting procedures; environmental, health and safety programs; along with day-to-day plant operation, minor and major repair and overhaul requirements.

The firms could either undertake this work as a 3~6 month contract for the initial commissioning and operational phases of the plant, and/or continue on to manage the plant operations under the terms of a long-term management contract.

c) Contract out for equipment maintenance – if the owner chooses to operate and manage the plant themselves, they still have the option of entering into long-term maintenance agreements with suppliers, and maintenance & overhaul firms.

These agreements could be limited to major equipment or could be extended all the way to include all the equipment within the plant.

When large, expensive equipment such as gas turbines are involved, the major equipment suppliers may offer long-term service agreements (LTSA), e.g. for six years, for a fixed annual fee or £/kW.hr rate, with agreed to escalation clauses. These may or may not include provisions for guaranteed target annual capacity and availability, with or without bonus/penalty clauses and/or creative revenue sharing arrangements.
3.0 O&M CONSIDERATIONS

In no particular order, the owners and operators of cogeneration and combined-cycle power plants should consider the following O&M issues:

a) **Share and exchange non-proprietary information.** Various gas turbine, steam turbine and HRSG users groups exist, that are either sponsored by engineering associations, by magazines or by the major equipment manufacturers. These allow the discussion of problem issues and new technologies in an open and constructive manner, with all benefiting from each other’s experience.

b) **Spare Engines and Lease Engine Programs.** Where any particular owner operates several identical aero-derivative or light industrial gas turbines, at one or more sites, they may consider the purchase of a spare engine which is used to take the place of a unit out of service. Having a spare engine improves the owner’s flexibility in scheduling major maintenance, and provides a safety margin in the event of a forced outage.

As a less-expensive alternative, the owner may consider entering into a lease pool engine arrangement with the equipment supplier, where spare engines are rotated around a large regional fleet of units. For fixed or variable annual fees, the delivery of a spare engine to a site is guaranteed, to cover for forced outages or scheduled maintenance periods.

c) **Computerized parts management systems** – many software programs are now available that will integrate, list and link all the major and/or minor equipment and subassemblies in a plant, their spare parts lists; scheduled maintenance intervals; maintenance and repair procedures; supplier contacts; ordering forms, etc. Theoretically, these programs will ease the tasks of managing parts and supply, once properly organized.

d) **Understand the OEM’s requirements** – it is vitally important to fully understand OEM’s maintenance procedures and time intervals. They must be strictly followed, to avoid long-term warranty issues.

e) **Warranty issues** – if warranty problems arise, and if individual components within a large assembly are replaced under the warranty provisions, keep track not only of the hours of operation of the machine, but of the operational hours of the replaced part. For instance, in some circumstances an individual blade row or combustor basket within a turbine may have 11 months of warranty left, even though the whole machine has been out of warranty for 6 months.

f) **Water washes** – on-line and off-line compressor water washes are vital to maintaining optimum performance and health of a gas turbine. Ensure that they are safely completed as scheduled and/or on-condition. This may also apply to steam turbines receiving poor chemistry steam.

g) **Chemistry** – rigorously maintain water and steam chemistry within defined limits, to achieve best life and performance of the plant.

h) **Plant performance monitoring and analysis.** Various software packages of varying complexity, cost and success are available that allow the plant owner to compare the current vs. new operation of a piece of equipment or the plant; to conduct what-if efficiency or power output analysis; to simplify performance reporting; and to anticipate control problems due instrumentation error or drift, or equipment problems due to wear and degradation.

i) **Relieve the operator’s boredom** – a perfectly operating plant can be a boring place.

Continuous training and educational upgrade is an obvious necessity, but consider empowering the operators by making them responsible for specific equipment and systems, and responsible for the planning, organizing and carrying out all it’s normal, preventative and scheduled maintenance, either hands-on (e.g. water treatment), or by interfacing with the major equipment suppliers (e.g. gas turbine, steam turbine, HRSG, DCS).

This approach can work well in a non-union operations environment.
4.0 STAFFING
A typical staff level and makeup for a stand-alone, owner-operated and maintained 80 ~ 150 MW combined-cycle power plant, is 15+ which could consist of the following:

a) **Plant Manager** – responsible for all scheduling, management and reporting of the facility.

b) **Plant Accountant** – responsible for the typical financial accounting, invoices and payment functions. Where non-fixed fuel contracts or merchant power considerations are a part of the plant strategy, the accountant may undertake the necessary negotiations and bidding tasks, if sufficiently knowledgeable technically.

c) **Plant Secretary** – correspondence and filings.

d) **Chief Engineer** – experienced, qualified and ticketed operator, with overall responsibility for the operating and maintenance staff.

e) **Shift Engineers** (3–4) – leading each shift.

f) **Shift Operators** (5~7).

g) **Electrical Technician** – usually an experienced specialist in high voltage equipment operation and testing, relaying, metering, and communications equipment.

h) **Instrument Technician** – usually an experienced specialist in setup, testing and troubleshooting of instrumentation and control wiring, PLC’s and distributed control systems.

i) **Maintenance / Stores Specialist** – who may also be responsible for the scheduling of major overhauls and shutdowns, and coordinating heavy work.

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**TYPICAL PLANT ORGANISATION CHART**