Optimization of Building CHP Systems

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ABSTRACT

Building combined heat and power (BCHP) feasibility and design depend on the magnitude, duration, and coincidence of electrical and thermal loads, as well as on the selection of the prime mover and waste heat recovery systems. This article assesses the current marketplace for BCHP systems in the light of recent events in California and the FERC standard market design. The article then reviews the basic components of BCHP systems and discusses their integration in the design of a large hospital complex located in Toledo, Ohio. Computer simulation is used to optimize the proposed design of the BCHP system. The findings presented illustrate that the first cost premium is recuperated through the BCHP annual energy and cost reductions.

INTRODUCTION

When planning newly constructed multi-building facilities, often historical operating data, diversity and mixed occupancy factors are not available. Prototypical building studies also may offer little real help. Accordingly, use of reliable (i.e. thoroughly beta tested) computer programs can provide a useful means of estimating their combined annual energy use when configured with various alternative mechanical/electrical/plumbing (MEP) systems being considered at the project preliminary design phase.

Simulation of annual energy use for each of the proposed multi-use building structures is then possible knowing only their respective building dimensions and proportionate use of envelope construction materials, various occupancy requirements and time of use and properly formatted NOAA (National Oceanic and Atmospheric Administration) or equivalent local microclimatic annual weather data.

THE CALIFORNIA SCENE

Current prospects for California deregulation remain uncertain. Clearly the trading practices of some well publicized merchant plant operators, the subject of a report by the California Independent System Operator (ISO) to the Federal Energy Regulatory Commission (FERC) in October 2002, leaves serious and yet unresolved issues. For example, the ISO has alleged that "gaming practices" of some well-known players may have "compromised the interconnected western states marketplace in the period January 2000 and June 2001"(1).

Examples of deceptive trading practices they claim have ranged from over-scheduling load (code name: Fat Boy), exporting power out of state using so called circular schedules (code name: Death Star), shifting back to create congestion and then getting paid to relieve congestion, selling back ancillary services (code name: Get Shorty), scheduling of counterflows on out of service lines (code name: Wheeling Out), megawatt laundering (codename: Ricochet) and selling "non-firm" energy as "firm" energy and scheduling energy production to collect congestion charges. For further details on the above ISO report, see their website at www.caiso.com.

To deal with the above and other foreseeable problems, the three major California energy agencies subsequently drafted an energy action plan intended to "eliminate energy outages and excessive price spikes in electricity or natural gas," which proposes the following:

1. Optimize energy conservation and resource efficiency.
2. Ensure reliable, affordable electricity generation.
3. Upgrade and expand the electricity transmission infrastructure.
4. Promote customer and utility owned generation.
5. Ensure reliable supply of reasonably priced natural gas.

Their final goal was to deliver an "action plan," following receipt of stakeholders' comments, to Governor Davis and to the California legislature (2). While we wish them every success at some future date, at least three of the above five objectives can be met now by those planning building combined heat and power (BCHP) facilities; namely: action items 1, 2 and 5.

Recognizing the flaws of California's AB 1890 deregulation history, one should also, at an early design stage, consider use of an on-site building cooling-heating-power (CHP) or BCHP system (3).

However, converting potential users into sales of BCHP systems in this economy and against the backdrop of the above referenced uncertainties regarding deregulation reform requires one to focus on how best to achieve the maximum utilization of fuel energy at the lowest annual owning and operating cost—no small order.

Recently Primen, a Boulder, Colorado based energy market intelligence company, reported in a distributed generation (DG) survey study entitled "Releasing the Potential for Distributed Energy."

Energy decision makers at these companies were asked about the following:

- Current use of on-site generation
- Willingness to dispatch existing standby generators to help utilize times of peak demand
- Likelihood of adopting grid-alternative on-site generators in two years
- Whether they would be most likely to buy or lease generators
- Benefits of and barriers to adoption of on-site generators
- Awareness and preferences for specific technologies
- Payback, return on investment, and other financial requirement
- Preferred vendors

Primen's survey results revealed fewer industrial and commercial businesses in the North American market (i.e., U.S. and Canada) expressed an interest in "distributed energy" versus one year ago. Its authors estimated that there are 1700 large establishments representing 1-6 gigawatts of load that are strong near-term prospects. The key reason given was that the latter represent a more sophisticated group of energy buyers with a greater knowledge about distributed energy, particularly (on-site) options, and currently more interested in projects that reduce cost versus averting blackouts.

FERC'S STANDARD MARKET DESIGN

FERC in December of 1999 issued Order 2000 in their effort to bring competition to the electric industry. Order 2000 requested that utilities form regional transmission organizations, or RTOs, which are independent agencies without any financial interest in either transmission wires or power plants.

Under Order 2000, utilities would, in effect, turn over control of their transmission wires to an RTO, which would auction use of the wires during every hour of the day. In addition, RTOs would plan new transmission lines and lead regional efforts such as electric energy conservation. Utility response to date has been predictably slow.

FERC has conceded that competition in the nation's electricity market is not yet achievable: "Since one cannot rely on the interaction of supply and demand in all instances to ensure that prices are competitive and thus just and reasonable." In a well functioning market as prices rise, consumers increasingly decline to purchase a commodity, to buy a substitute if available.

In electricity markets however, market conditions do not apply. Electricity consumers in restructured markets can't buy a substitute, nor do they have the ability to recognize when prices spike because they get their bills once a month. Additionally, power plants and transmission wires take years to be sited and built.

Accordingly, FERC is under pressure to stimulate investment in transmission wires due to a national shortage of transmission wires. Utilities remain uncertain if they can recoup their investments.

Also, utility power generation facilities typically release 60 to 70% of initial fuel energy to ambient in addition to transmission line delivery loses approaching an additional 9% less to the customer's electric service meter (4) (5).

Before any power provider can meet the public demand for affordable, reliable power three steps must first take place on a national level. Local government takeovers of utilities aren't the answer to the nation's energy problems. Accordingly;

1. The country needs new rules to lower uncertainty and volatility in its wholesale power markets.
2. The country needs comprehensive federal electricity legislation that enables the nation's electric system to keep pace with demand.
3. The electric power industry needs to restore investor confidence.

BCHP BASICS

Efficient BCHP systems maximize all available opportunities to utilize fuel energy that the prime mover is unable to convert into shaft energy. If waste heat cannot be utilized effectively, the resulting BCHP plant efficiency, in effect, defaults to the limit of the prime mover. Obviously, smaller prime movers cannot match the comparable performance of utility size prime movers. Where building thermal energy requirements can utilize the waste heat available from the prime mover, on-site requirements are reduced and overall plant efficiency increased.

BCHP feasibility and designs depend on the magnitude, duration, and coincidence of electrical and thermal loads, as well as on the selection of the prime mover and waste heat recovery systems. Integrating the design of the project's electrical and thermal requirements with the cogeneration plant becomes a major challenge. The basic components of the BCHP plant are (1) prime mover and its fuel supply system, (2) generator, (3) waste heat recovery system, (4) control system, (5) electrical and thermal transmission and distribution systems, and (6) connections to building mechanical and electrical services.

Reciprocating Engines

Reciprocating engines are the principal prime mover used in smaller (i.e., under 1-5 MW) cogeneration plants. Reciprocating internal combustion engines except for some small air cooled units, provide for reclaimed heat from the jacket cooling system. These engines are available in sizes up to 27,000 brake horsepower (bhp) and use all types of liquid and gaseous fuels, including methane from landfills or sewage treatment plant digesters. Internal combustion engines that use the diesel (compression ignition) cycle can be fueled by a wide range of petroleum products (up to No. 6 oil), although No. 2 diesel oil is the most commonly used. Diesel cycle engines can also be fired with gaseous fuel in combination with liquid fuel, called pilot oil, used as the ignition agent in dual fuel engines.

Those engines, which permit coolant to reach 250°F at above atmospheric pressure, must allow the coolant to flash into low-pressure steam (15 psig) after leaving the engine jacket (ebullient cooling).

Waste heat in the form of hot water or low-pressure steam can be recovered from the engine jacket manifolds and exhaust, and additional heat from the lubrication system can also be recovered. The distribution of input fuel energy for a representative engine operating at rated load can be broken down as follows:

a) Shaft power 33%
b) Convection and radiation 7%
c) Rejected in jacket water 30%
d) Rejected in exhaust 30%

It should be appreciated that these percentages will vary with engine load and design.

High-volume, lower temperature (from approximately 700°F at full load to below 500°F for low loads) exhaust gas offers less heat recovery opportunities. Low-speed engines generally cannot be ebulliently cooled since their cooling systems must operate at approximately 170°F or below, often presenting problems for single stage absorption chillers on bottoming cycle duty.

Combustion Turbines

For gas turbine cycle engines their average fuel to electrical shaft efficiencies generally range from approximately 12% to above 35%. The remainder of the fuel energy is discharged in the exhaust and through radiation or internal coolants in large turbines. A minimum stack exhaust temperature of approximately 300°F however, is required to prevent condensation. Since the heat rate efficiency is lower, the quantity of heat recoverable per unit of power is greater for a gas turbine than for a reciprocating engine. The net result is an overall thermal efficiency of 65% or higher. Because gas turbine exhaust contains a large percentage of excess air, afterburners may be installed in the exhaust to create a supplementary boiler system providing additional steam or to level the steam production during reduced turbine loads. Afterburners can increase cycle efficiency to an estimated maximum exceeding 90% (7).

Steam Turbines

Steam turbine exhaust (or extracted steam) when reduced in both pressure and temperature from throttle conditions can be used to generate prime mover shaft power. Furthermore, they may be fed to heat exchangers, absorption chillers, steam turbine-driven centrifugal chillers, pumps, or other equipment as will be seen (8)

Heat-to-Power Ratios

Ideally, the amount of recoverable heat from the prime mover tracks the power load.

The following methods can be used to produce required on-site power and thermal energy.

- Match the heat/power ratio of the prime mover to that of the user's hourly load profile.
- Store excess power as chilled water or ice when the thermal demand exceeds the coincident power demand.
- Store excess thermal production as heat when the power demand exceeds the heat demand. Either cool or heat storage must be able to productively discharge most of its energy before it is dissipated to the environment.
- Sell excess power or heat through approved paralleling protocols on a mutually acceptable contract basis to a user outside of the host facility. Often the buyer is the local utility, but sometimes it is nearby or "over the fence."

Quality of Heat

The quality of recovered energy is another major determinant in selecting the prime mover. When the quantity of high-temperature (above 260°F) recoverable heat available from an engine's exhaust is significantly below that demanded, a combustion or steam turbine should be considered (6).

Lower heat to power ratios in the order of 1 to 3 lb/hr steam per horsepower output of the prime mover indicate the need for one with a high shaft efficiency of 30 to 45% (shaft energy/fuel input). This would appear to be a good match for engines because their heat output is normally available as 15 psig steam or 250°F water. Higher temperature to pressure ratios are available, but only from a exhaust gas recovery system. For the case where 30% of the fuel energy is in the exhaust, approximately 50% of it is recoverable (at approximately a 300°F final exhaust gas temperature). However, below 50% is recoverable where higher steam pressures may be required.

Higher heat to power ratios of the order of 4 to 11 lb-steam/hp-hr normally available with combustion turbines are often lower in shaft efficiency. Smaller turbines are only 20 to 25% efficient for example, with 75 to 80% of their fuel energy released to ambient.

BCHP Component Integration

BCHP systems often afford better opportunities to maximize on-site prime movers (engine or turbine) waste heat for generating additional power by means of a combined cycle approach and simultaneous heating or cooling where needed.

Recognizing all of the basic variables and equipment parameters needed to optimize a BCHP plant referenced above, it should be apparent that there is no one single path to follow. To begin, one should first demonstrate site-specific factors that determine how best to achieve maximum utilization of input fuel energy is to project BCHP plant operations for a typical year. Yet to do so requires simultaneously comparing alternative solutions and their comparative economics to serve as a benchmark for one's decision process. To illustrate both the path and the process, let us next examine the following real world case study undertaken for a former client.

We will next describe a simulation experiment designed to benchmark the use of separate electric driven chiller plants for a nominal 100,000 square meter medical facility. The facility is composed of two 12-story medical office towers and an adjacent 6-story hospital incorporating gas and steam turbine driven chillers and separate gas turbine driven synchronous generator against each building with conventionally designed, separate plants using head to head computer simulation scenarios.

The methodology employed is a three-stage computer analysis program used to evaluate energy demand and consumption, energy costs and life cycle costs so as to determine the comparative economic benefits of three separate central plants versus a single BCHP serving on-site cooling, heating and power needs. This article presents the results of selected computer simulations employed to arrive at our BCHP design recommended for construction, after confirming an estimated 1.8 year simple payback.

METHODOLOGY

The Energy System Analysis Series (ESAS) is a group of computer programs to model an hour-by-hour, full-year basis the energy performance and system characteristics of commercial, industrial, and institutional buildings and systems under a variety of design, operating, and ambient weather conditions. The entire library of programs can be installed and run on any 20 MB or greater, and a math co-processor. The programs, support files, and samples use about 2.7 MB of disk space, and a "typical" study might need another 5 MB of free disk space for the files generated during the runs. Following is a brief description of the ESAS programs employed by us:

To determine the economic benefits of differing HVAC plant configurations, we employed a three-stage computer analysis program. The program was used to evaluate energy demand and consumption, energy costs and life-cycle costs. Analysis usually is performed for a typical weather year, but actual weather data can be used to compare actual performance to potential performance as follows.

Buildings & Distribution System Program (ERE)

This program module calculates the thermal and electrical loads hourly for the building (or section of the building) and simulates air-distribution-system operation in satisfying these loads for a full year. Additionally, one can observe the effect of changes in various operating parameters.

Building Section Summation & Cooling Storage Program (TCR)

This program module sums up hourly, full-year loads from multiple ERE computer runs of various building sections to find total diversified system loads that must be satisfied by a given mechanical plant configuration. This program also is used to model cooling (or heating) storage systems, with recharge rates, tank sizes and operating strategy as variables. One also can modify selected loads in the hourly data file, permitting the load profile to be "tuned" to specific requirements (9).

Up to nine (9) ERE computer outputs can be summed or modified in each TCR computer run. Those summations can be combined in subsequent TCR computer runs, thereby permitting an infinite number of building sections to be merged before imposed on a mechanical plant configuration.

Mechanical Plant Analysis Program (EEC)

This program module simulates mechanical-equipment operation on an hourly, full-year basis. Equipment responds to loads imposed by the building's airside system (and cooling storage system, if used) to find monthly and annual energy demand and consumption for various systems under evaluation. The program also permits monthly utility demand and consumption data to be grouped into various time-of-day brackets. Up to six plant configurations can be simulated in each EEC computer run with up to four sets of hourly load requirements use in each.

PROJECT DESCRIPTION

The above referenced hospital/office complex consisted of three buildings: a six-story hospital and two identical 12-story office towers served at the Toledo, Ohio, site. The on-site BCHP plant to be simulated would serve all three buildings within the hospital/office from an available location in reasonably close proximity yet visually concealed by attractive landscaping from the complex taking full advantage of the natural contours of the selected site. Concerns regarding environmental issues, e.g., low exhaust emissions[3] and noise levels were also carefully addressed.

Hospital Building

The hospital building is a six-story, rectangular building with a gross floor area of 41,850 m^2 (6,975m^2/floor) of which 1,674 m^2 (279m^2/floor) or approximately 4% is unconditioned floor area, and the remaining 40,176 m^2 (6,696 m^2/floor) or approximately 90% is conditioned floor area. All floors are 4 m floor-to-floor and 2.74 m floor-to-ceiling. The long sides of the building face north to south as illustrated in Figure 1.

The vision glass area is 31% of the gross exterior wall area (45% of the floor-to-ceiling wall area). The glass characteristics include a shading coefficient of 0.55, and overall U-value of 3.35 W/m^2-K, and visible light transmittance of 0.67. The opaque wall is of concrete panel and brick or stone with an overall U-value of 0.3 W/m^2-K. The lighting is provided by recessed fluorescents with varying design lighting (15 W/m^2 to 27 W/m^2) for different types of functions. The design receptacle load is also variable by type of function (5.4 W/m^2 to 10.8 W/m^2). Design occupancy can also be varied from 9.3 m^2/person to 37.2 m^2/person.

The hospital is serviced by its own laundry, which when combined with its food service area, provides a rather large service water-heating load. Hourly profiles of lighting, receptacle, occupancy, thermostat, fan operation, and service water heating are varied extensively between

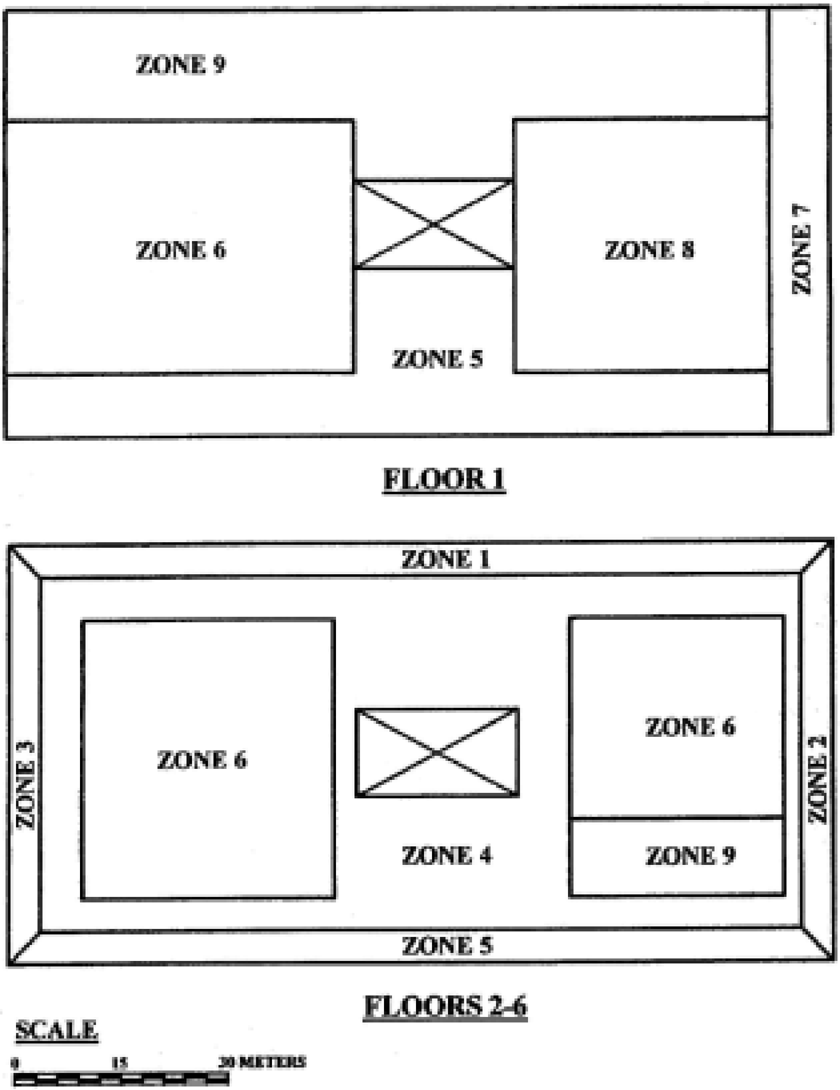

Figure 1. Typical Floor Plans for Floors 1 and 2 through 6 for Hospital Building

various types of functions. Except for the administrative area, all areas have a continuous (8760 hours) fan operation and are heated and cooled by variable-air-volume (VAV) distribution systems with reheat boxes activated at the required minimum primary air damper setting (e.g., 1.27 L/s-m^2 to 2.03 L/s-m^2 for different functional areas). Nine separate VAV distribution systems (zones) distinguished by exposure and space function are identified (as shown in Figure 1) as follows for energy analysis purposes, namely:

Zone 1: North patient perimeter (Floors 2-6)
Zone 2: East patient perimeter (Floors 2-6)
Zone 3: West patient perimeter (Floors 2-6)
Zone 4: Patient interior (Floors 2-6)
Zone 5: Administration
Zone 6: Surgery (Floors 1-6)
Zone 7: Food service perimeter (Floor 1)
Zone 8: Food service interior (Floor 1)
Zone 9: Laundry (Floors 1-6)

The heating thermostat is set at 22.2°C and the cooling thermostat at 22.3°C with dead-band control (float) between these temperatures. The ventilating air requirement is modeled as a constant percentage ranging from 20% to 80% (i.e., depending upon occupancy needs) of the supply airflow. All VAV systems have both outdoor air economizer and fan inlet vane damper control.

Medical Office Towers

The hospital/office complex also contains two identical office towers. Each office tower is a 12-story, rectangular building with a gross floor area of 30,026 m^2 (2,052 m^2/floor) of which 28,688 m^2 (2391 m^2/floor) or approximately 5% is unconditioned floor area. All floors are 4 m floor-to-floor and 3 m floor-to-ceiling. The long sides of the tower face north and south. Figure 2 shows a typical floor plan for one of the two office towers.

The office tower walls are constructed from glass spandrels with an overall U-value of 0.466 W/m^2-K. The façade of a typical floor is 31% vision glass on all exposures with a shading coefficient of 0.55, without interior or exterior shading, and a glass visible light transmittance of

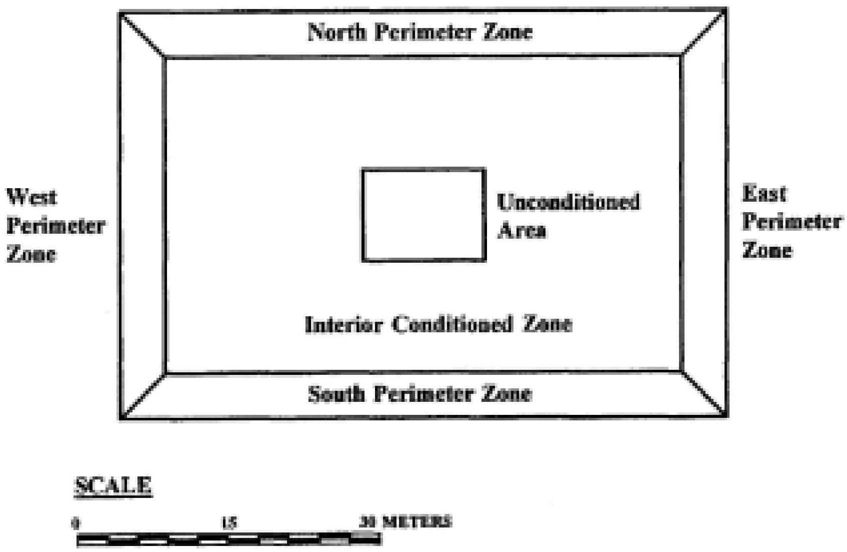

Figure 2. A Typical Floor Plan for Office Tower

0.67. The overall U-value for windows is 3.24 W/m^2-K. The roof construction has an overall U-value of 0.3 W/m^2-K. Lighting is provided by recessed fluorescent fixtures with a design lighting load of 21.5 W/m^2 for all task areas, which include 100% of the perimeter zones and 67% of the interior conditioned area. Non-task areas (33% of the interior conditioned area) have a lighting load of 10.8 W/m^2. The unconditioned areas have a lighting load of 2.2 W/m^2. Design receptacle load is 10.8 W/m^2 of the conditioned floor area.

Each medical office tower is heated and cooled by VAV distribution systems. Separate VAV systems (one per exposure) serve each of the perimeter zones and one VAV system serves the interior zone of the office towers for all floors. Primary air supply temperature is set at 12.8°C at design conditions, but each zone can independently use demand reset up to a maximum of 15.6°C.

During the heating season, thermostats are set at 21.1°C and during the cooling season at 23.9°C. Dead-band thermostat control exists when the space temperature is above 21.1°C and below 23.9°C. During this time, the VAV air supply boxes are at minimum position without reheat. The minimum VAV damper setting is 1.52 L/s-m^2 for all zones. The minimum ventilation air quantity is 0.76 L/s-m^2 of the conditioned floor area during occupied hours. All VAV systems have both economizer and fan inlet vane damper control.

Simulated BCHP Plant

The proposed BCHP plant, illustrated in Figure 3, consists of a gas-turbine engine driven centrifugal chiller, a package gas-turbine cogeneration unit (PGTCU) with a heat-recovery steam generator unit (STGU), a deaerator, a thermal energy storage (TES) system (cold and hot), and cooling towers (10). A control room built in the plant contains the turbine control system panel, a boiler and auxiliary equipment control, and a monitoring system. The plant also has two 1760-kW gas-fired hot water boilers and one 300-kW hot water converter with one hot-water storage tank for space heating service, and one 780-kW hot water heater for domestic hot-water service. The plant will be located above the ground and adjacent to the hospital/office complex. The plant provides the required heating, steam, chilled-water for cooling, and a peak electrical power of 3787 kW for the entire hospital/office complex.

The gas-turbine/centrifugal chiller (GT/CC) is a nominal 7,000-kW centrifugal chiller driven by a gas-turbine capable of delivering 1170 kW at an inlet air temperature of 15℃ and operates with a standard 5.6℃ temperature-differential across the evaporator and condenser. The gas-turbine driver produces less power when the inlet air temperature exceeds 15℃. A cooling coil with a 6.7℃ entering chilled-water temperature is provided in the air inlet to maintain a 15℃ inlet air temperature(4). While this takes some cooling tonnage, there is a net gain in energy cost that varies depending on the inlet air humidity. The available exhaust-heat energy is ducted to the HRSG to generate steam, which then drives a condensing steam turbine with a power output of 2325 kW (10).

The PGTCU comprises a gas turbine, a separate generator, a turbine exhaust system, an heat recovery steam generator (HRSG), and a control system(5). The nominal 1.5-MW cogeneration unit dissipates its waste heat together with the available exhaust-heat energy from the GT/CC transferred into the HRSG. At the International Standardization Organization (ISO) conditions (15℃, sea level), the conventional gas-turbine unit provides 2,129 g/s per hour (pph) unfired (turbine-only) and 3,717 g/s (fully-fired) superheated steam without steam injection to feed the condensing steam-turbine.

The efficiency of the gas-turbine driver varies with the inlet air temperature. With an inlet air temperature of 40℃, the capacity of the

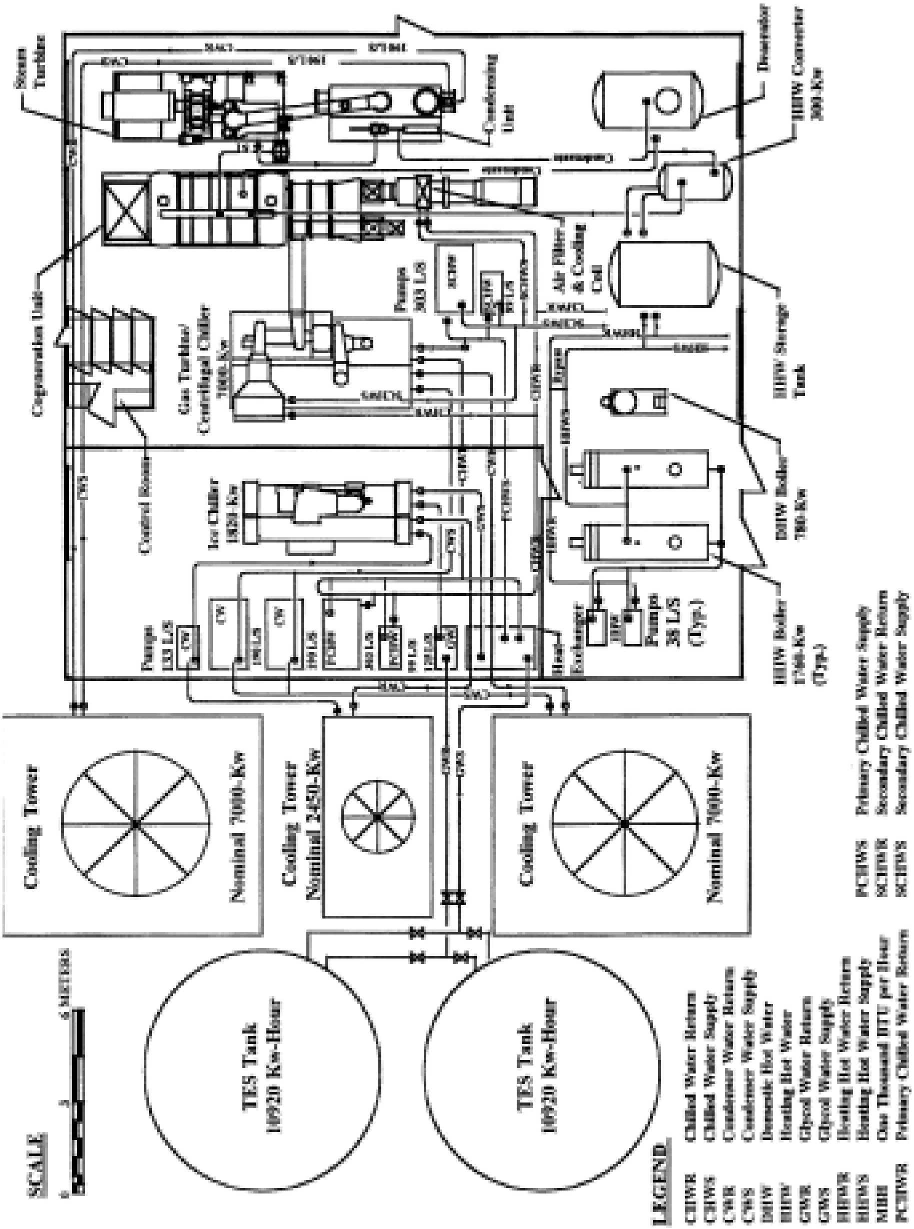

Figure 3. Proposed Plant: Modular Hybrid Combined-Cycle Cogeneration and Thermal Energy Storage Plant

gas turbine is derated to 1119 kW without steam injection. A cooling coil with a 6.7℃ entering chilled-water temperature is provided in the air inlet to maintain a 15℃ inlet air temperature. The power outputs respectively of the gas turbine at the inlet air temperature of 15℃ are 1584 kW with steam injection and 1462 kW without steam injection. The gas-turbine exhaust system includes two flow diverter valves and two duct burners, and is located upstream of the HRSG.

The above referenced STGU is a multistage condensing type and consists of both a steam turbine and a condensing unit. The superheated steam produced a 3,717 g/s in the above referenced HRSG and is piped to the STGU where it generates 2325 kW of additional electricity. Its condensing unit is equipped with two condensate pumps and two 189.3 L/s cooling-water pumps supplying cooling water at 29.4℃ to the condenser based on a cooling water temperature rise of 11.1℃. The TES system of the proposed DHCC plant remained the same as that described above for the conventional DHCC plant.

When the load falls below 1462 kW, the PGTCU commences operation. The steam generated from the HRSG is supplied to the hot-water converter to produce hot water. The hot water then goes into the hot-water storage tank producing up to 17,586 kW for space heating. If the storage tank is full, the steam is dissipated. When the electrical load exceeds 1,462 kW (e.g., 1,700 kW), more than the power output of the PGTCU, the STGU is automatically brought on-line. The STGU now uses the unfired heat-recovery from the PGTCU to try to drive it. If the heat is not enough to produce the steam in order to produce the additional 238 kW, which it needs out of the STGU, the PGTCU duct burner fires to produce the required 238 kW at the STGU.

The STGU is designed to produce 2325 kW at 13,608 kg of heated steam input. In the summertime, the STGU uses the heat-recovery from both the GT/CC and the PGTCU to produce the full power output of 2325 kW. In the wintertime when the cooling load does not exist (and, therefore, the GT/CC in inoperative), the STGU can produce only 50% of its rated output. Therefore, in the wintertime when the load is 3300 kW, part of that power will have to be purchased because the STGU cannot produce the entire 3300 kW requirement without the GT/CC operating.

SIMULATION

This simulation was based on the initial schematics, preliminary plans, and other programmatic information developed by the architectural and engineering (A/E) team. Due to the size of the project and adjacencies of buildings within the hospital/medical office complex we decided to conduct a comparative simulation using state of the art computer programs to decide whether to construct separate conventionally designed HVAC plants within each of the three buildings (11).

When designing a thermal-energy-storage (TES) plant, one must take into consideration three major points: plant operation; operating costs vs. installation costs; and system sizing. The above referenced three operational modes generally must be weighed.

One mode is to serve the base-cooling load with one chiller and use the TES system to satisfy peak loads. A second mode is to serve the base cooling with the TES system and use the chiller to satisfy the peak. A third mode would require the TES system to satisfy 100 percent of the peak load and recharging the TES tank during off-peak hours. Since the hospital has a fairly high continuous cooling load, a full TES system is not warranted.

ANALYSIS

Table 1 summarizes the comparative installed first costs of three individual "base case" plants versus the proposed BCHP plant. Table 2 summarizes the comparative results of the economic analysis conducted for both the above referenced conventional (or base case) and proposed BCHP plants. Electric and gas consumption costs were simulated by a commonly used energy analysis program: Energy System Analysis Series (ESAS), which also was used to evaluate the design and operating characteristics. This included energy demand and consumption, energy costs, and life-cycle cost on an hourly, full-year basis for both the benchmark case (i.e., individual building plants) and the proposed BCHP plant. The Building & Distribution System Program (ERE) module was used to calculate the thermal and electrical loads hourly for each of the representative buildings and simulated the operation of the air-distribution system in satisfying these loads for a full year.

The second program group or Building Section Summation & Cooling Storage Program (TCR) module summed up the hourly, full-year loads from the multiple ERE computer runs described above of various building sections to find total diversified system loads that must be satisfied by a given mechanical plant configuration. This program was also used to model cooling (or heating) storage systems, with recharge rates, tank sizes, and operating strategy as variables.

Finally, the third program group, Mechanical Plant Analysis Program (EEC) module, simulated the operation of the various pieces of mechanical plant equipment on an hourly, full-year basis as they respond to loads imposed by the building's air-side system (and cooling storage system, if used), to establish monthly and annual energy demand and consumption for both conventional and proposed DHCC plants.

SUMMARY OF FINDINGS

The above comparative analysis has shown substantial first cost and annual operating savings favoring use of the proposed BCHP plant, resulting from an overall reduction in utility and maintenance costs. Operating cost savings are mainly due to the fact that approximately 95% of the total required annual electrical power is produced on-site by cogeneration9. Therefore, only 5% of its annual estimated power needs must be purchased from a utility.

Additionally, the reduction in the number of plant operators for the proposed plant has also made sizable contribution to the annual operating cost savings. Although the installed first cost of the proposed plant was higher in comparison, the total operating cost savings of approximately $1,244,500 per year resulted in a cost-effective simple payback period of 1.8 years for the proposed BCHP plant. Additionally, the design of the proposed BCHP plant has resulted in a substantially compact central plant (an approximately 29% decrease in physical size) in comparison to the base plant having three separate central plants. The fully-dedicated proposed BCHP plant(10) as demonstrated by the results of building simulation reported earlier contributed to reduced demand charges associated with the time-of-use electrical rates, better operation and maintenance, efficient energy management, higher overall system efficiency, and reduced levels of emissions and noise offering significant benefits.

In estimating maintenance and repair costs for both the base and proposed plants, manufacturers of respective equipment were consulted.

Table 1. Comparative Installed First Costs

	Base Plant Cost For Individual Buildings ($)		*Total Base Plant Cost ($)*	*Total Proposed Plant Cost ($)*
	Hospital Building	*Office Towers*		
Chillers	280,720	578,920	859,640	
Cogeneration Unit				2,179,150
Gas Turbine/Chiller Unit			1,000,000	
Steam Turbine/Condensing Unit				940,000
Separation Wall				6,400
Dearator				30,700
Cooling Towers	77,400	104,400	181,800	277,230
HHW Boilers	78,890	157,780	236,670	110,900
HHW Converter				1,500
HHW Storage Tank				3,500
TES System	581,790		581,790	590,370
Pumps	81,150	102,400	183,550	129,350
Piping	97,650	87,660	185,310	204,100
Control Room A/C				4,000
Subtotal	1,197,600	1,031,160	2,228,760	5,477,200
Overhead and Profit	239,520	206,230	445,750	477,568
Total	**1,437,120**	**1,237,390**	**3,674,510**	**5,954,768**

Table 2. Comparative Economic Analysis

Cost	*Base Plant*	*Proposed Plant*
Annual Energy Cost		
Electricity	1,916,329	276,305
Gas	124,813	1,049,725
Total	2,041,142	1,326,030
Annual Cooling Tower Water Costs		
Water	2,975	3,856
Sewer	7,328	9,499
Chemicals	4,699	6,091
Total	15,002	19,446
Annual Maintenance Cost		
Chillers	125,706	110,376
Boilers	226,708	58,692
Gas and Steam Turbines		262,800
Plant Operator	919,860	306,600
Total	1,272,274	738,468
Total Operating Cost	3,328,418	2,083,944
Savings vs. Base Plant		1,244,474
Installed First Cost (refer to Table 1)	3,674,510	5,954,768
Cost Premium vs. Base Plant		2,280,258
Payback Period		1.8 years

These costs include major chillers, cogeneration equipment and boiler overhaul over the life span of the equipment. Accordingly, the average maintenance and repair costs for the chillers are assumed to be $43.80/ton-year and $4.38/MBtu-year, respectively. The maintenance and repair costs for the gas and steam turbines for the proposed plant are estimated to be $262,800/year. The plant operator costs is estimated based on one person, 24 hr/day, 365 days/year at $35/hr fully loaded per plant. The lower portion of Table 1 presents the operating cost savings and installed cost premium for the proposed plant. The ratio of these two quantities gives the simple payback period as 1.8 years for the proposed plant (11).

CALIFORNIA'S FORESEEABLE LANDSCAPE

California electricity prices may surge again this summer (2004), reaching levels last seen during the state's power crisis two years ago, as drought in the Pacific Northwest limits hydroelectric generation.

The lack of snow and rain means flows will be at least 25% below normal at Columbia River dams, according to the Bonneville Power Administration, the federal agency that sells power from 31 dams in Washington, Oregon, Idaho and western Montana. Some meteorologists are forecasting even lower flows.

"Hydro is going to be a disaster this year," said Jim Duncan, head of research for energy trading at ConocoPhillips, the third largest U.S. oil company. A lack of hydropower could contribute to a repeat of the "out-of-control" prices experienced in California in 2000 and 2001, he said.

California's failed deregulation plan caused rolling blackouts that left millions of customers without power. Gov. Gray Davis eventually agreed to have the state sign $43 billion in multi-year supply contracts to keep the lights on as California's biggest utilities were pushed to insolvency by high wholesale power prices.

This year, wholesale power prices already reflect some concern about the limited hydroelectric generation. Next-day power at Washington's Mid-Columbia trading point, a benchmark for the Northwest, touched $51.82 a megawatt hour last week, the highest since August 2001, according to Bloomberg data.

Wholesale power for delivery in Northern California has averaged $46.65 a megawatt-hour this year (2003), almost double its average at this time last year.

Higher prices may be coming, according to utilities and industrial customers. Puget Energy, Inc., owner of the largest electric utility in Washington, has said profit will be pinched because of higher power costs. The British Columbia Power & Hydro Authority is limiting power sales to save water behind its dams.

Bonneville Power predicted in December 2002 that water flows through its dam in the Dalles, Ore., would be 25% below normal this year. That suggests water will be low throughout Bonneville's system of dams along the Columbia river and its tributaries, agency spokesman Mike Hansen said.

The forecasts are "abysmal," Hansen said. An unofficial, mid-January update showed flow at the Dalles dam was lower than forecast.

CONCLUSIONS

Demand for energy in the 21st Century, particularly in less developed countries, could rise precipitously as they strive to increase their living standards. Global average temperature has been estimated to rise 1°C by the year 2025 and approximately 3°C by the end of year 2100. This could severely disrupt agriculture, natural ecosystems and human settlements, as they exist today. Introducing on-site BCHP plants capable of incorporating state-of-art energy conservation, heat recovery and waste minimization methods driven by meaningful cost savings, reduced emissions of greenhouse gases and ozone depletion rates, should allow for a more sustainable balance for the long term (12).

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NOMENCLATURE

CHWR	Chilled Water Return
CHWS	Chilled Water Supply
CWR	Condenser Water Return
CWS	Condenser Water Supply

DHW	Domestic Hot Water
HHW	Heating Hot Water
GWR	Glycol Water Return
GWS	Glycol Water Supply
HHWR	Heating Hot Water Return
HHWS	Heating Hot Water Supply
L/S	Liters Per Second
PCHWR	Primary Chilled Water Return
PCHWS	Primary Chilled Water Supply
SCHWR	Secondary Chilled Water Return
SCHWS	Secondary Chilled Water Supply
$	US Dollars

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