BCHP COMPUTER SIMULATION FOR MEDICAL COMPLEX DESIGN

M. Meckler[1]
[1]P.E., CPC, Director, Los Angeles Power Association;
Fmr. Adjunct Professor Engineering, CSUN; President/CEO,
Design Build Systems (DBS), 17525 Ventura Blvd., Encino, CA., 91316, U.S.A.

ABSTRACT

This paper will describe a simulation experiment designed to benchmark the use of separate electric driven chiller plants for a nominal 100,000 sq. meter medical facility comprising two (2) twelve (12) story medical office towers and an adjacent six (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 paper will describe a methodology employing 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 (3) separate central plants versus a single BCHP serving on site cooling, heating and power needs. Paper will present the results of selected computer simulations employed to arrive at our BCHP design recommended for construction, after confirming an estimated 2.6 year simple payback.

INTRODUCTION

Increasing worldwide demand for more electricity to satisfy individual building, heating, ventilating and air conditioning (HVAC) system requirements has generated greater interest in gas cooling alternatives. Fortunately gas cooling technologies have improved and have recently gained popularity due to limited available power generation capacity and well known associated negative environmental consequences if increased substantially above current operating levels. Widespread computer use and higher outdoor ventilation rates[8] needed to maintain acceptable indoor air quality (IAQ) and ozone depletion issues, etc. demand new pressing global warming solutions[1].

Based on a population increase from 5.3 billion to 8.1 billion, world demand for electrical power is expected to double by the year 2020. According to current data, North America with about 949 MW of installed capacity serves 377 million people at an annual rate of 2.52 kW per capita. On the other hand, 1.2 billion people in China[2] have 181 MW installed capacity at a rate of 0.15 kW per capita, while South Asia and sub-Saharan Africa are the lowest consumers at 0.09 kW per person. Additionally gas turbines are fast coming on line as production units as well as for peak handling, offering lower emission[7] levels, utilization rates exceeding 90% uptimes, and improved efficiencies.

Gas-turbine packages and deregulation of power production already underway in 40 U.S. states are pointing toward widely distributed generation grids with current utilities eventually providing transmission, maintenance and billing services. In implementing the wider use of these systems, however, manufacturers and HVAC design professionals require appropriate simulation methods to properly analyze, select and specify more optimal gas/electric (hybrid) systems for site-specific requirements e.g. operating characteristics, space requirements, system integration, installed first and operating costs (including continuous maintenance), and life-cycle cost.

BCHP plants combined with thermal energy storage (TES) systems[6] also help serving electric utilities save energy and reduce harmful emissions. TES shifts energy consumption from peak afternoon electricity rates to low nighttime periods when utilities generally operate only their most efficient plants, which results in saving significant amount of source energy and rate reduction to building owners. In this way, the building also avoids high peak-demand charges associated with the energy used during summer daytime periods. Time-of-day rates during the evening hours can be much less than those of daytime rates. In Chicago, for example, large office buildings may have an effective summertime energy cost of $0.154 per kWh in comparison to $0.028 per kWh at night.

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 of 20MB 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.

Up to nine 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)

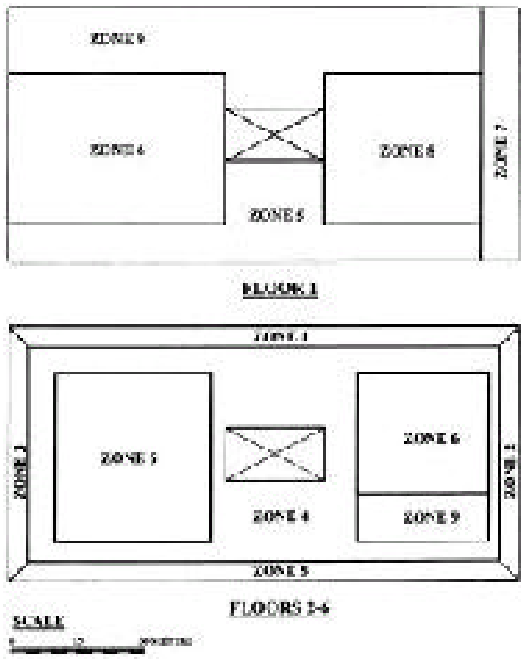
This program module simulates mechanical-equipment operation on an hourly, full-year basis. Equipment responds to loads imposed by the building's airside system (and cooling storage system, if used) to find monthly and annual energy demand and consumption for various systems under evaluation. The program 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.

EXPERIMENT

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 (6,975 m^2/floor) of which 1,674 m^2 (279 m^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.



TYPICAL FLOOR PLANS FOR FLOORS 1 AND 2 THROUGH 6 FOR HOSPITAL BUILDING
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.

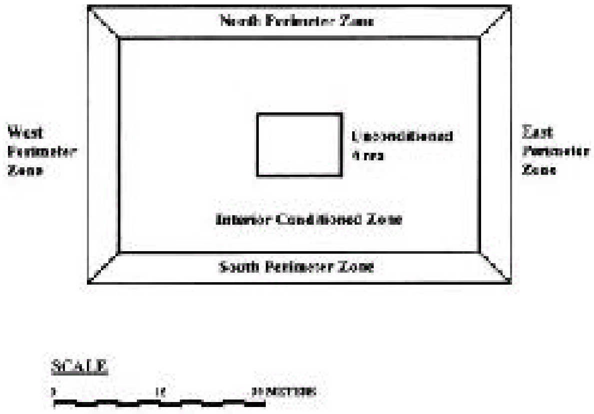
The hospital is serviced by its own laundry which when combined with its food service area provides a rather large service water-heating load. Hourly profiles of lighting, receptacle, occupancy, thermostat, fan operation, and service water heating are varied extensively between various types of functions. 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 Variable Air Volume (VAV) distribution systems (zones) distinguished by exposure and space function are identified (as shown in Fig. 1) as follows for energy analysis purposes, namely:

- Zone 1: North patient perimeter (Fls 2-6)
- Zone 2: East patient perimeter (Fls 2-6)
- Zone 3: West patient perimeter (Fls 2-6)
- Zone 4: Patient interior (Fls 2-6)
- Zone 5: Administration
- Zone 6: Surgery (Fls 1-6)
- Zone 7: Food service perimeter (Fl 1)
- Zone 8: Food service interior (Fl 1)
- Zone 9: Laundry (Fls 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. Fig. 2 shows a typical floor plan for one of the two office towers.



A TYPICAL FLOOR PLAN FOR OFFICE TOWER

Figure 2

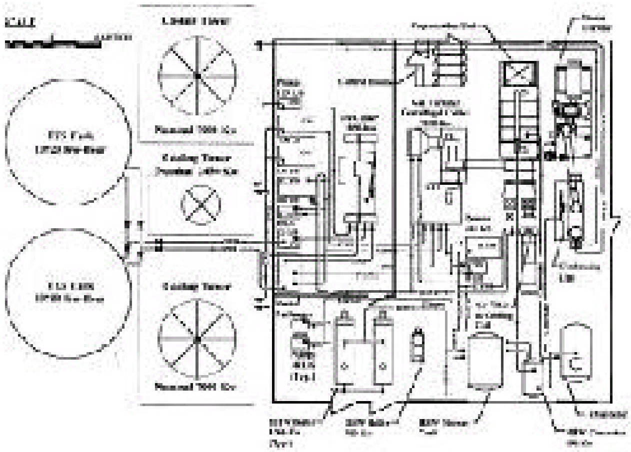
The office tower walls are constructed from glass spandrel with an overall U-value of 0.466 W/m^2-K. The façade of a typical floor is 31% vision glass on all exposures with a shading coefficient of 0.55, without interior or exterior shading, and a glass visible light transmittance of 0.67. The overall 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 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 TES system (cold and hot), and cooling towers. 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.



PROPOSED PLANT : MODULAR HYBRID COMBINED-CYCLE COGENERATION/TES PLANT
Figure 3

The Gas-Turbine/Centrifugal Chiller (GT/CC) is a nominal 7000-kW centrifugal chiller driven by a gas-turbine capable of delivering 1170 kW at an inlet air temperature of 15°C and operates with a standard 5.6°C temperature-differential across the evaporator and condenser. The gas-turbine driver produces less power when the inlet air temperature exceeds 15°C. A cooling coil with a 6.7°C entering chilled-water temperature is provided in the air inlet to maintain a 15°C inlet air temperature[4]. While this takes some cooling tonnage, there is a net gain in energy cost hat 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.

The PGTCU comprises a gas turbine, a separate generator, a turbine exhaust system, an 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 Organization for Standardization (ISO) conditions (15°C, 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°C, the capacity of the gas turbine is derated to 1119 kW without steam injection. A cooling coil with a 6.7°C entering chilled-water temperature is provided in the air inlet to maintain a 15°C inlet air temperature. The power outputs respectively of the gas turbine at the inlet air temperature of 15°C 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 is piped to the STGU where it generates of 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°C to the condenser based on a cooling water temperature rise of 11.1°C. 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 its HRSG supplied to the hot-water converter to produce hot water. The hot water then goes into the hot-water storage tank up to 17586 kW for space heating. If the storage tank is full, the steam is dissipated. When the electrical load exceeds 1462 kW (e.g., 1700 kW), more than the power output of the PGTCU, the STGU is automatically brought on-line. The STGU now uses the unfired heat-recovery form 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 is 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 our client's 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 (3) buildings.

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 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.

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[1]	$2,975	$3,856
SEWER[2]	$7,328	$9,499
CHEMICALS	$4,699	$6,091
TOTAL	$15,002	$19,446
ANNUAL MAINTENANCE COST		
CHILLERS[3]	$125,706	$110,376
BOILERS[4]	$226,708	$58,692
GAS & STEAM TURBINES		$262,800
PLANT OPERATOR[5]	$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 2)	$2,674,510	$5,954,768
COST PREMIUM VS. BASE PLANT		$3,280,258
PAYBACK PERIOD		2.6 YEARS

Comparative Economic Analysis
Table 1

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 s 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.

CONCLUSIONS

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 cogeneration[9]. Therefore only 5% of its annual estimated power needs are needed to 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 has resulted in a cost-effective simple payback period of 2.6 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. 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 and 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 cost 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 2.6 years for the proposed plant.

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.

ACKNOWLEDGEMENTS

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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