DESIGNING SUSTAINABLE ON-SITE CHP SYSTEMS

By Milton Meckler, P.E., Lucas Hyman, P.E., and Kyle Landis, P.E. -- January 28, 2007 ASHRAE Meeting

ABSTRACT

Sustainable on-site cooling-heating-power (CHP) systems for large multi-building projects require a simplified design and implementation approach from conventionally designed mini-utility type CHP systems employing large volume/footprint, costly, high thermal mass heat-recovery-steam-generators (HRSGs) and 24/7 stationary engineers.

This paper will demonstrate the use of prefabricated, skid-mounted hybrid steam generators with internal headers, fully integrated with low pressure drop heat extraction coils located in the gas turbine exhaust, and employing environmentally benign heat transfer fluids. The proposed thermal tracking Integrated CHP Gas Cooling System (ICHP/GCS) includes close coupled plate and frame heat exchangers, pumps, and self-regulating controls, interconnected via a closed, low-pressure, non-volatile recirculation loop capable of efficient, year-round transfer to on-demand HVAC&R building heat sinks including absorption chillers.

Available waste heat is transferred directly to a gas turbine exhaust extraction heat exchanger, interconnected to a recirculating, closed circuit, non-volatile, low-pressure heat transfer fluid loop. Available waste heat is cascaded to serve multi-building space cooling, heating, and domestic hot water loads, which permits maintaining high log-mean-temperature-differentials (LMTDs) at the subject extraction coil, significantly lowering gas turbine back-pressure, and permitting significant life-cycle-cost savings. These benefits were demonstrated during a recent, comparative CHP study of a 3.5 MW gas turbine installation at a central California university campus.

INTRODUCTION

In today's expanding energy hungry world, sustainability is no longer an option, it has become the design standard for design professionals. What does one mean by the term "sustainability" and is it different from "building sustainability" or "cooling-heating-power (CHP) sustainability"? Ray Anderson, Chairman of Interface Inc., was quoted as stating "sustainability implies allowing a generation to meet its needs without depriving future generations of a way to meet theirs". The ASHRAE Board of Directors approved the position document "Building Sustainability" on 6/23/02, which stated "ASHRAE supports building sustainability as a means to provide a safe, healthy, comfortable indoor environment while simultaneously limiting the impact on the Earth's natural resources". A subtle additional component for "CHP sustainability" is implied in Mr. Anderson's use of the words "allowing a generation to meet its needs". The latter recognizes the MEP (mechanical, electrical, plumbing) consultant's "real world" need to justify (or sustain) "value added" CHP benefits for its clients. What better way to attract funding for CHP than to utilize life cycle cost (LCC) methods to select among traditional versus more attractive CHP alternatives to secure client commitment and thereby advance overall "green" project sustainability.

Other factors in addition to LCC analysis include: waste heat versus prime energy utilization, building operator skill-sets, reliability, local utilities "real time" billable costs, related environmental concerns, and "green" marketing benefits to refocus initial client goals when setting long-term budgetary, building design, and operational parameters. This is particularly true when considering whether to employ on-site cooling CHP systems that rely in part or exclusively on available local gas and electric utilities to serve their new or renovated, large-scale, tenant occupied or leased building facilities. And when doing so, one must realistically ask: how foreseeable are future energy costs likely to be, given present world conditions being what they are?

In looking at traditionally designed mini-utility type CHP plants, is it practical to continue to employ large, bulky and costly heat-recovery-steam-generators (HRSGs) for 3.5-MW and above building CHP applications? Furthermore, are HRSGs still suitable as the most efficient means of waste heat extraction and if not, what are the available alternatives? Studies suggest that HRSG costs are disproportionately high, require 24/7 operators for code safety compliance and also require costly high-pressure steam and condensate distribution systems, along with architect-engineer (A/E) concerns regarding available local contractors with CHP system familiarity. Additional concerns include questionable shopping of specified "or equal" HVAC&R (heating, ventilating, air-conditioning and refrigeration) components, and owner concerns regarding the availability of operator skill-sets. These concerns and challenges often lead to overly conservative CHP construction budgets and associated risk factors for owners and their financing that can make for "deal breakers".

HRSG OPERATION UNDER TURBINE CYCLIC CONDITIONS

To better understand the nature of HRSGs operating in conjunction with gas-fired turbines, refer to **Figures 1a and 1b**. Figures 1a and 1b illustrate the time delays associated with each of three (3) representative start-up procedures for the listed combustion gas turbine (CGT) operating with flow rates ranging from 500,000 to 4,000,000 lb/h (63 to 504 kg/s) exhaust gas flow. The least downtime required for a complete shutdown, depending upon the required exhaust gas flow rate, results with a hot start. The greatest downtime similarly results with a cold start. A complete shutdown of the CGT will also affect the operating temperatures within the downstream HRSG. To reduce the turbine downtime, "soak period" apparatus can be used to warm the interior turbine surfaces during downtime.

HRSGs, suitable for the generally steady operations of large-scale electric utility plants, may prove unsuitable for cyclic building loads due to cyclic thermal stress fatigue. This effect was attributable to their inherent large thermal mass, particularly when tracking highly cyclic, transient and variable diurnal thermal demands, such as building space and domestic hot water heating and air conditioning loads.

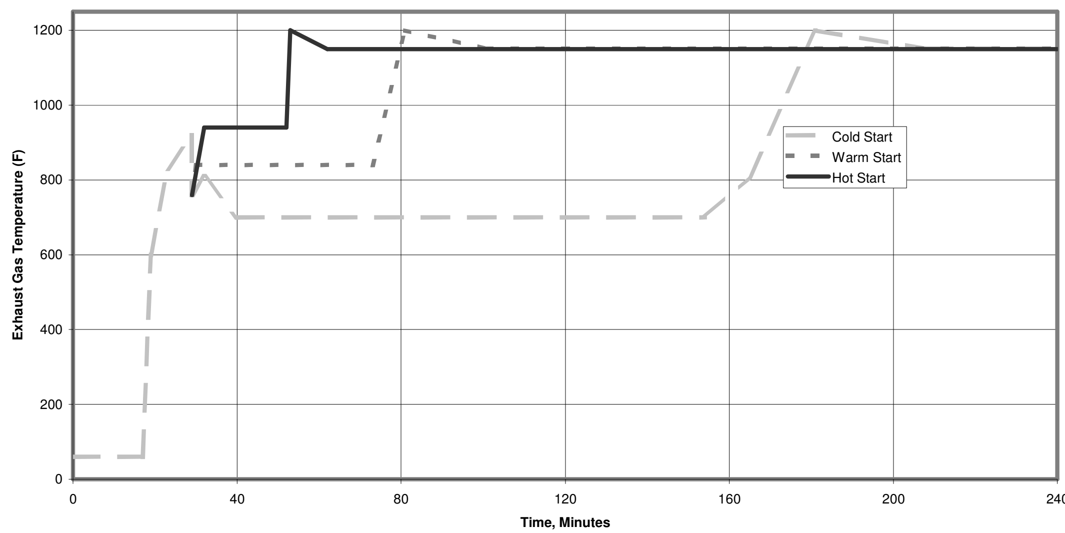


Figure 1a - Typical Frame 7 Gas Turbine Exhaust Temperatures

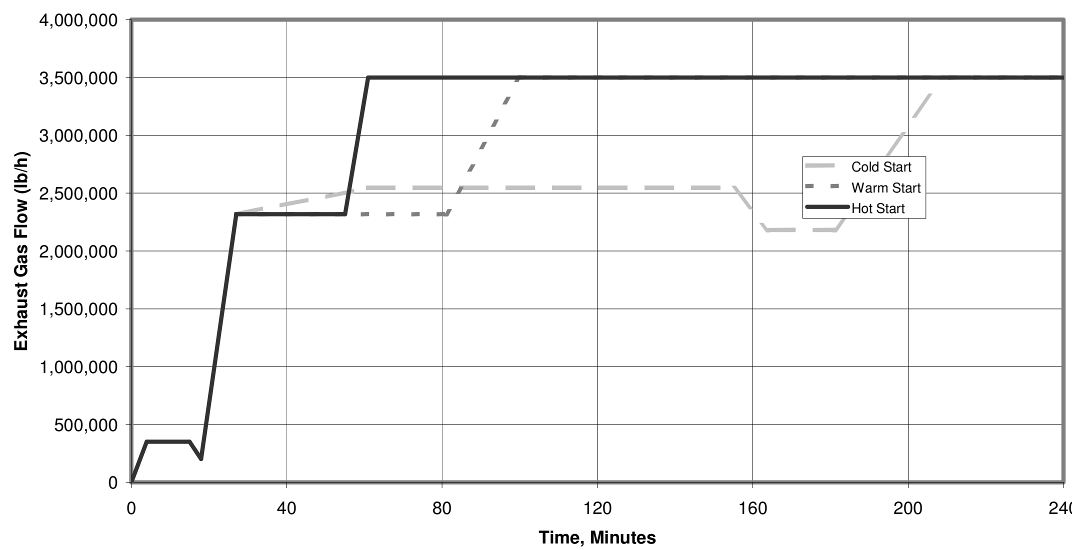


Figure 1b - Typical Frame 7 Gas Turbine Exhaust Flows

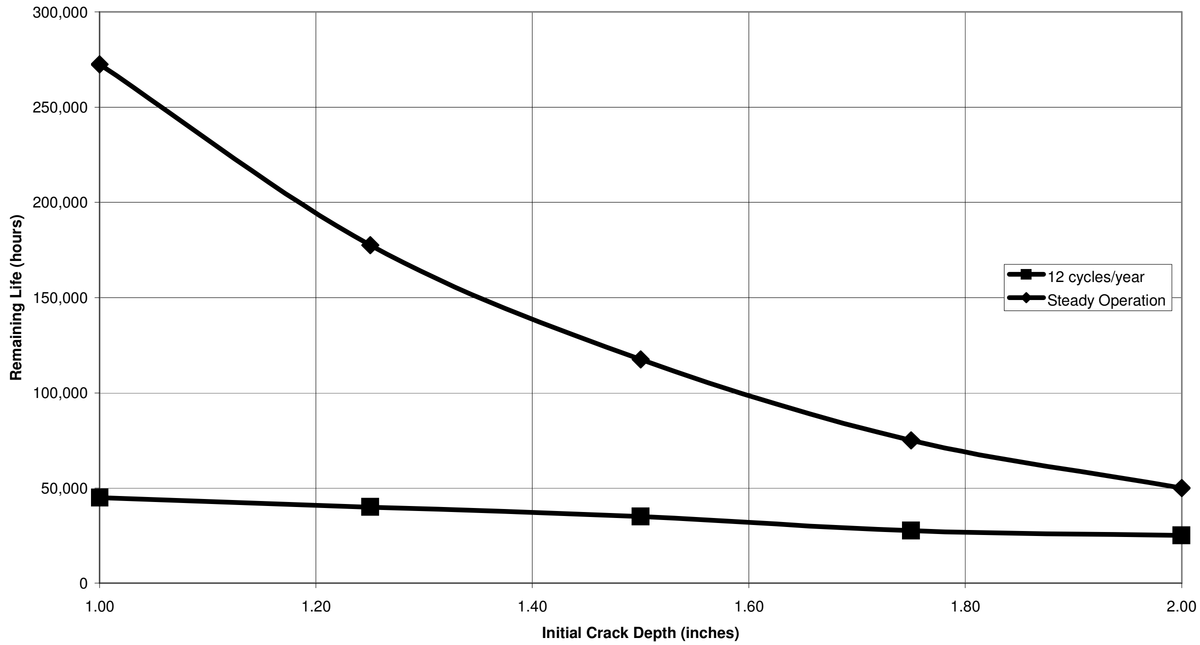


Figure 2 - Impact of Cycling on Useful Life of Header

Referring to **Figure 2**, notice that the impact of turbine cycling on the useful life of a HRSG header can be expressed as initial crack depth in inches versus the remaining life in hours for indicated HRSG operating temperature and header piping material (Piper 2002). For example, for a 1.00 inch (25.4 mm) crack depth to appear in a newly installed HRSG header of the material type and operating temperature illustrated, only 12 cycles per year will accelerate the HRSG header failure rate by approximately 560% over that expected with the same HRSG operated under utility type ”steady operations”.

Under highly cyclic building load operations, for reasons given above, the above referenced HRSG header failure rate could be further accelerated, requiring more frequent monitoring which could also prove costly in additional gas-turbine downtime required for HRSG interior inspection. Furthermore, although 9% chrom-moly steels (9Cr-1Mo) had been used successfully in U.S. fossil boilers from the 1980’s, in recent years the alloy (referred to as P91 in piping and T91 in tubing applications) has also been applied in large HRSGs in order to reduce thermal fatigue and creep damage in main steam piping and desuperheaters, however with limited success. Combined-cycle plants have experienced major trouble with this same alloy in the fabrication, production and repair of P91/T91 components. For example, HRSG users have had to contend with failures in dissimilar metal welds and transition areas in less than 1,000 operating hours, and failures caused by poor weld geometry or inappropriate heat treatment in less than 5,000 operating hours (Swankamp 2002). Although combined cycle Integrated CHP Gas Cooling System ICHP/GCS applications have also been explored, they are beyond the scope of this study, which is intended to deal only with gas turbine simple-cycle ICHP/GCS power applications.

ABSORPTION CHILLER CYCLE INTERACTION

Among the many chiller technologies available in the market today, single and two stage Lithium Bromide (LiBr) absorption chillers have proven to be the most cost efficient topping cycle options for converting available high temperature waste heat, e.g. 350 - 400 °F (177 - 204 °C), into chilled water cooling. On the bottoming cycle end of available cascading lower temperature waste heat, e.g. 200 - 250 °F (93 - 121 °C), ammonia–water and diethylene methanol tri-ethylene glycol (DEMTEG) absorption chillers also offer cost efficient production of ice for a variety of thermal energy storage (TES) options that can significantly lower design day cooling demand by impacting both the size and operating cost of above referenced topping cycle absorption chillers.

Achieving the above described synergies within on-site CHP systems requires thinking “out of the proverbial box” to identify similar converging opportunities by enhancing gas turbine engine performance at lower prime energy and overall capital cost. Close coupled turbine inlet cooling supplied from two (2) stage and/or single stage steam (or hot water) absorption chillers benefit enhanced turbine power performance.

Although the above referenced indirect fired 2-stage and single stage LiBr absorption chillers utilize steam for activation they can also employ waste heat directly to generate chilled water. In fact, efforts to supply turbine exhaust directly to a modified 2-stage direct gas fired LiBr absorption chiller configuration have already been demonstrated (Berry et al 2004, Berry et al 2005, Meckler 2005, Pathakji et al 2005).

SYSTEM DESCRIPTION

Having identified HRSG operating problems and capital cost issues described earlier, the authors decided to seek an alternative, lower cost means of extracting turbine exhaust waste heat without sacrificing overall CHP cycle efficiency. CGT back pressure performance effects were investigated when attempting to select HRSGs for the pre-selected 3.5 MW power requirement. The author's found that the associated HRSG pressure drops ranged from 4.5 to 6.5 inches w.g. (1121 to 1619 Pa) depending upon manufacturer. This corresponded to a 0.75 to 1.5 % loss at the rated CGT turbine power output. In selecting the CGT exhaust coil, it was determined that the turbine backpressure could be reduced by a factor of 4:1 and its substitution could also improve CGT power performance.

Accordingly, it was determined that CGT turbine power losses could be significantly reduced through substitution of the above referenced HRSG with a low thermal mass and low pressure drop extraction coil. This extraction coil would be placed directly in the exhaust gas stream in order to achieve a high LMTD. In this way the CGT back pressure could be reduced to approximately 1.0 to 1.5 inches w.g. (249 to 374 Pa), by requiring less extraction coil surface area at greater finned spacing to match or exceed equivalent HRSG design waste heat extraction rates.

Thermal tracking CHP utilization can be maximized by maintaining year-round high log-mean temperature differences (LMTDs) at the ICHP/GCS CGT extraction coil contributing to lower CGT lower discharge gas temperature to ambient. In selecting the turbine exhaust extraction coil (TGEC) for ICHP/GCS applications, the inlet and outlet exhaust gas and HTHTF temperatures must be specified for either a parallel or counter-flow configuration. The temperature change that takes place across the TGEC from entrance to exit is not linear. A precise temperature change between the above turbine exhaust gas and HTHTF fluid streams is best represented by LMTD.

The LMTD is defined in terms of the higher (HTD) and lower temperature difference (LTD) as follows:

$$\text{LMTD} = (\text{HTD} - \text{LTD}) / \text{Ln}\,(\text{HTD}/\text{LTD})$$

The following standard restrictions apply:

a) Ln is computed using the natural logarithmic base
b) Constant TGEC and HTHTF flowrates
c) Constant overall heat transfer coefficient
d) HTHTF fluid temperature is uniform across the tube cross section,and
e) Heat losses are negligible

The use of several commercially available high-temperature-heat-transfer-fluids (HTHTFs) was explored, and several were found that met the criteria of being environmentally benign, low-pressure, non-volatile, low viscosity and stable at operating temperatures ranging from 150 to 600 °F (66 to 316 °C).

DESCRIPTION OF COMPARED SYSTEMS

Two comparative cogeneration systems were developed to partially meet the electric, cooling and heating requirements of a central California university campus. The systems are identical in terms of turbine configuration, and differ in the manner in which exhaust heat is extracted and utilized. One alternative uses a conventional cogeneration arrangement with a HRSG, while the other alternative uses the Integrated CHP/Gas Cooling System (ICHP/GCS) approach. Refer to **Figure 3** for a schematic of the conventional plant, and **Figure 4** for a schematic of the ICHP/GCS plant. The plants were sized to meet the average base electric load of the campus (approximately 3.5 MW). However the CGT will turndown on weekends and other periods of relatively low campus occupancy to match the electric demand. Exporting energy to the serving utility was found to be uneconomical since the cost to produce the electricity is typically greater than the amount that the utility pays for exported electricity.

Electric, cooling, and heating loads used in the analysis are based on actual campus data, and averaged into four seasonal 24-hour profiles. The CGT utilized in both alternatives has a fuel consumption (at 3.5 MW electric output) of 42.7 X 10^6 Btu/h (12.5 X 10^6 W). The boilers utilized in both alternatives are assumed to have an efficiency of 80%, and the electric chillers utilized in each alternative are assumed to have an efficiency of 0.6 kW/ton (COP = 5.9).

Conventional Cogeneration Plant

The conventional plant uses a HRSG to produce high-pressure-steam (HPS), which is used to drive a 2-stage absorption chiller with an assumed steam consumption of 9 lb/ton (1.2 kg/kW) before being reduced to low-pressure-steam (LPS). The LPS is then used to make heating-hot-water (HHW) for distribution to the campus. Any energy not utilized by the plant is rejected to a dump condenser. A radiator then cools the condenser cooling water. The balance of heating and cooling loads that are not served by the cogeneration plant are served with gas-fired boilers and electric driven centrifugal chillers.

ICHP/CGS Plant

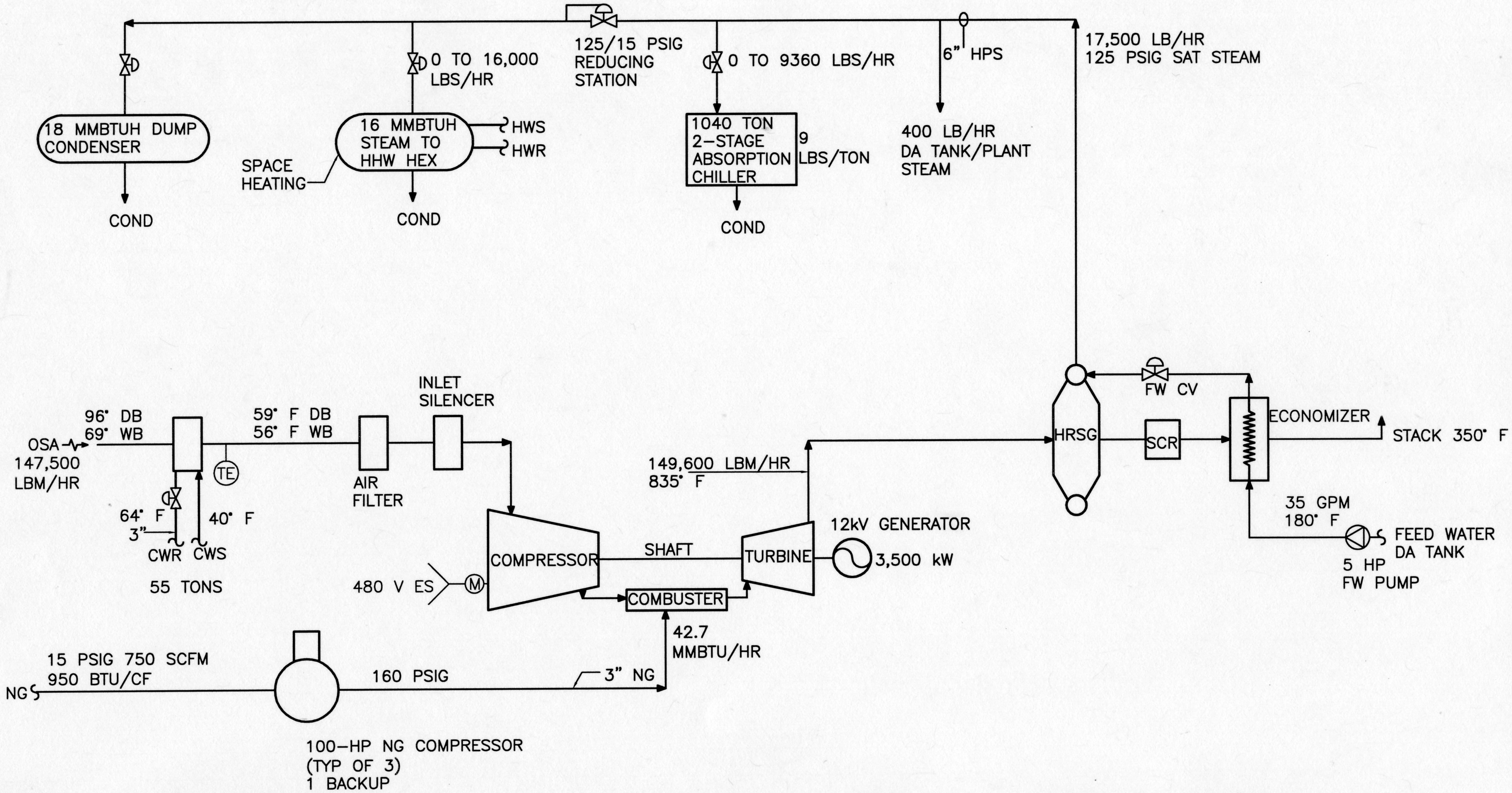
The inherently self- regulating ICHP/GCS met the nominal 1040-ton (3,658-kW) cooling requirement of our 3.5 MW campus project by employing more efficient, commercially available low-mass hybrid steam generators and utilizing commercially available, smaller footprint nominal 240-ton (844-kW) 2-stage HTHTF adapted heated absorption chiller with an assumed heat rate of 10,600 Btuh/ton (COP=1.13) of the type illustrated in **Figure 5**.

The ICHP/GCS plant can be functionally integrated with controls, plate and frame heat exchangers, turbine inlet cooling coil, pumps, interconnecting piping, CGT waste heat extraction coil, and prefabricated (for minimal on-site erection) on modular skids. Notice that HTHTF is initially supplied to a nominal 240-ton (844-kW) 2-stage hot water type absorption chiller operated in parallel with the nominal 800-ton (2,814-kW) 15-psig (103 kPa) steam heated single stage absorption chiller with an assumed heat rate of 17.5 lb/ton (2.3 kg/kW).

The ICHP/GCS plant uses an exhaust-to-HTHTF heat exchanger (HEX) to recover the exhaust heat by heating the HTHTF from approximately 250°F to 600°F (121 °C to 316 °C). The HTHTF first supplies a hybrid HEX that produces LPS. The LPS is used to drive a single-stage absorption chiller. The HTHTF is then used to drive a two-stage absorption chiller, and then goes to a plate and frame HEX to produce HHW. Note that domestic-hot-water (DHW) can also be produced to further utilize the recovered heat. However, in the specific case analyzed herein, the majority of recovered heat was utilized for campus heating and cooling demands, and dumping of recovered heat was minimal.

The thermal utilization is arranged in this order due to the heat temperature and quality requirements of the various system components. For example, the 2-stage absorption chiller has a maximum HTHTF inlet temperature of 425°F (218°C). Therefore, some of the recovered heat must be utilized prior to the 2-stage absorption chiller. Though the most efficient way to use heat would be to produce HHW prior to the 2-stage absorption chiller, the coincident campus cooling and heating loads are not such that the HHW HEX would always reduce the HTHTF below 425°F (218°C). Therefore, the single-stage absorption chiller is the first in the sequence. Since the HHW HEX requires lower temperature HTHTF than the 2-stage absorption chiller, the HEX was placed downstream of the chiller. Like the conventional plant, the balance of heating and cooling loads that are not served by the cogeneration plant are served with gas-fired boilers and electric driven centrifugal chillers.

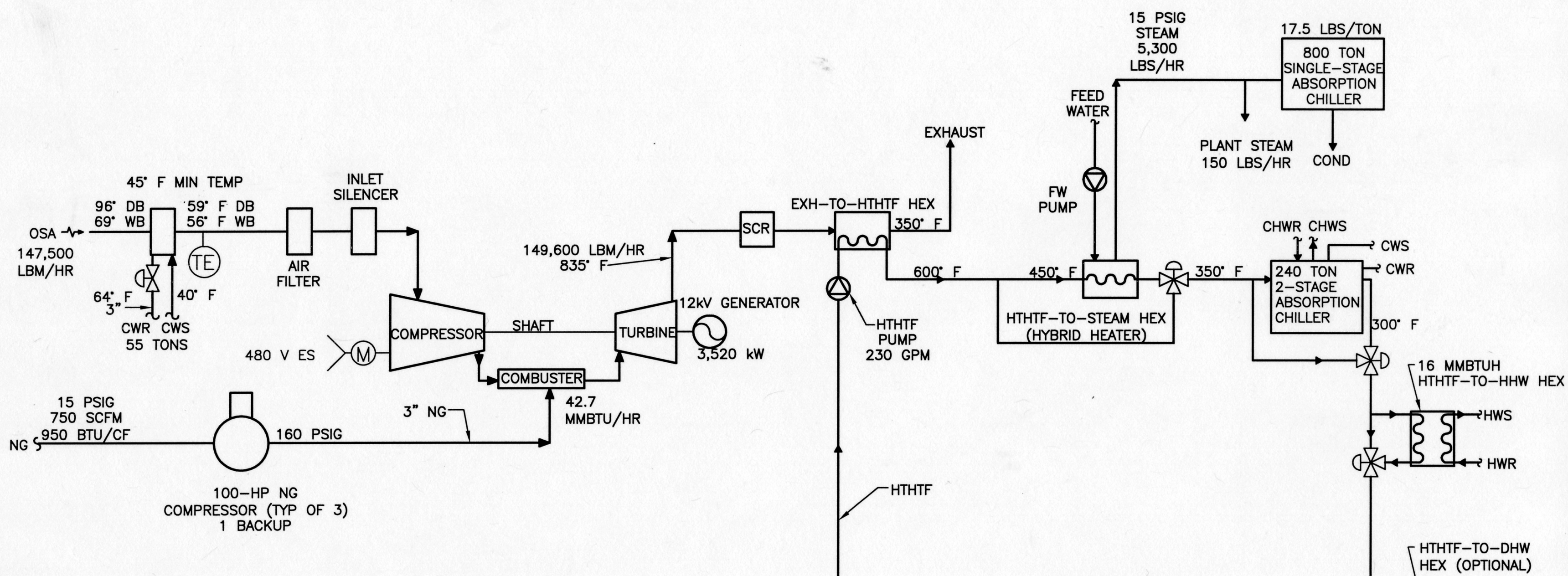
FIGURE 3 CONVENTIONAL PLANT SCHEMATIC DIAGRAM



LEGEND:

COND	CONDENSATE	HWR	HOT WATER RETURN
DA	DEAERATOR	HWS	HOT WATER SUPPLY
EXH	EXHAUST	NG	NATURAL GAS
HHW	HEATING HOT WATER	OSA	OUTSIDE AIR
HPS	HIGH PRESSSURE STEAM	SCR	SELECTIVE CATALYTIC REDUCTION

FIGURE 4 ICHP/GCS PLANT SCHEMATIC DIAGRAM



LEGEND:

COND	CONSENSATE	HHW	HEATIING HOT WATER
DHW	DOMESTIC HOT WATER	HTHTF	HIGH-TEMPERATURE RESISTANT HEAT TRANSFER FLUID
DHWR	DOMESTIC HOT WATER RETURN	HWR	HOT WATER RETURN
DHWS	DOMESTIC HOT WATER SUPPLY	HWS	HOT WATER SUPPLY
DA	DEAERATOR	NG	NATURAL GAS
EXH	EXHAUST	OSA	OUTSIDE AIR
HEX	HEAT EXCHANGER	SCR	SELECTIVE CATALYTIC REDUCTION

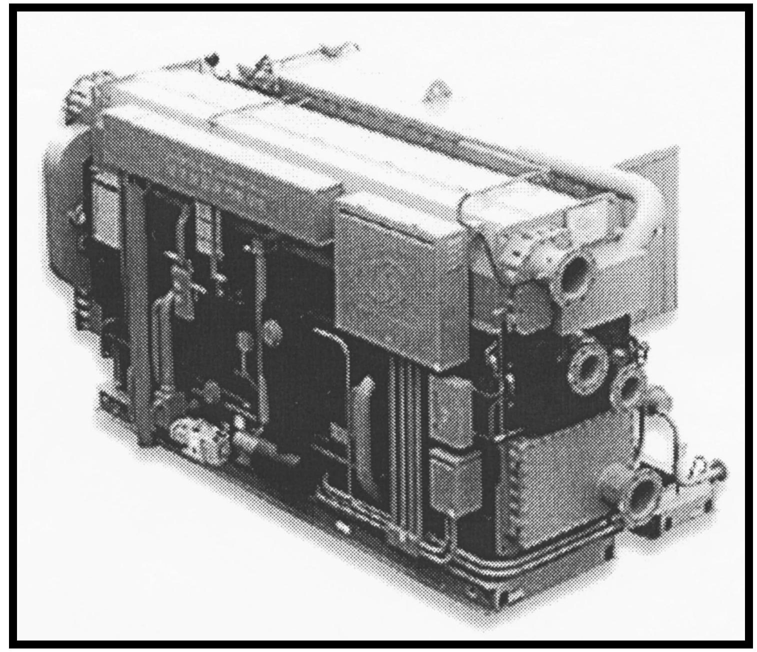


Figure 5 - HTHTF Adapted Heated Absorption Chiller

SUMMARY OF COMPARED LIFE CYCLE RESULTS

Capital Cost Comparison

Table 1 below shows the approximate differential material cost for major equipment. Equipment that is the same for either plant is not included in the estimate.

Table 1: Capital Cost Comparison

Conventional Plant

HRSG	$360,000
1040-Ton 2-Stage Absorption Chiller	$416,000
16 MMBtuh Steam to HW HEX	$70,000
18 MMBtuh Dump Condenser	$80,000
500-Ton Electric Chiller	$100,000
Misc.	$100,000
TOTAL	**$1,126,000**

ICHP/GCS Plant

CGT Exhaust HEX	$90,000
Hybrid HEX	$30,000
800-ton Single-Stage Absorption Chiller	$240,000
240-ton 2-Stage Absorption Chiller	$210,000
16 MMBtuh HTHTF-to-HHW HEX	$80,000
850-Ton Electric Chiller	$170,000
Misc.	$50,000
TOTAL	**$870,000**

As shown in the above table, the cost for major equipment for the conventional plant is approximately $250,000 higher than the HTHTF plant.

Energy Cost Comparison

An energy model was prepared to calculate the energy usage and cost differences between the two plants. **Table 2** below summarizes the energy costs for the two plants. **Table 3** shows the electric and natural gas rates that were assumed. **Table 4** shows the Summary of the Conventional Plant CHP Annual Energy Costs, and **Table 5** shows the Summary of the ICHP/CGS Annual Energy Costs.

Table 2: Annual Energy Costs

	Conventional Plant	ICHP/GCS Plant
Natural Gas Cost	$918,000	$1,053,000
Electricity Cost	$2,817,000	$2,799,000
Total Energy Cost	$3,735,000	$3,852,000

Table 2 shows that the annual energy costs for the conventional plant are approximately $120,000 less than the HTHTF plant. This is primarily due to the fact that the conventional plant produces more cooling with a 2-stage absorption chiller, leaving more recovered energy for campus heating loads. This results in less fired boiler use and lower natural gas cost for the conventional plant. This slight energy cost penalty avoids the need to operate the ICHP/GCS at above a 15 psi (103 kPa) steam pressure. Although annual energy costs of the ICHP/GCS system listed in Table 2 are higher from an energy economics perspective, it is not necessarily a negative from an environmental perspective. That is because the relative annual average cost of natural gas on a $/unit volume or electricity on a $/kilowatt- hour delivered basis to any U.S. location is inherently site specific and will vary depending upon applicable rate structures. When considering sustainability from an environmental standpoint, one must first estimate the energy content of fuel delivered to the serving electric utility for each purchased kilowatt-hour delivered versus the energy content for each 1000 ft^3 (28.3 m^3) of natural gas delivered on a comparable source energy basis adjusted for transmission losses.

Personnel and Maintenance Cost Comparison

Personnel and Maintenance costs were calculated for the two plants. **Table 6** below summarizes the differences in the costs. As with the equipment costs above, costs that are the same are not included.

Table 6: Operation and Maintenance Costs

Conventional

Absorption Chiller	$8,000
17500 lb/hr HP/LP Steam System	$5,000
Operator Cost (FT Operator)	$480,000
Total	**$493,000**

ICHP/GCS Plant

Absorption Chiller	$16,000
5300 lb/hr LP Steam System	$2,000
Operator Cost (FT Operator)	$80,000
Total	**$98,000**

As shown above, the significant difference is the cost of 24/7 stationary engineers for the conventional case, due to the use of HPS. The conventional case assumed six full time operators at $80,000 each per year. The ICHP/GCS case assumed one full time operator (40 hours/week) at $80,000 per year. The $80,000 per year assumed cost is fully burdened, and includes salary, payroll taxes, Social Security, Medicare, healthcare, retirement, etc. The annual operation and maintenance costs of the conventional plant are $400,000 more than the ICHP/GCS plant.

20-Year Life Cycle Cost

Based on the above capital, energy, and maintenance costs, 20-year life-cycle-cost (LCC) comparisons were prepared. Life cycle cost analysis is a process by which system costs are calculated not just for a particular period, but for the life of the system. In addition, life cycle cost analysis is a process by which the time value of money is taken into consideration. The LCC analysis prepared assumes a discount rate of 6%, an operation and maintenance escalation rate of 3%, and an energy escalation rate of 2%. The discount rate equates future values with present values. That is, the discount rate is the number used to determine the equivalent present dollar value given some future dollar value. In general, the discount factor should equal the long-term cost of money.

Table 7 below summarizes the LCC comparison:

Table 7: Life Cycle Cost Comparison

Case	Life Cycle Cost	Life Cycle Cost Savings
Conventional Cogeneration Plant	$58,431,918	
ICHP/GCS Plant Plant	$53,979,824	$4,452,094

As shown, the estimated LCC savings of the HTHTF plant over the conventional are more that $4.4 million dollars.

SUMMARY AND CONCLUSIONS

ICHP/GCS systems are easier to operate, are inherently more user friendly and responsive to the highly variable occupancy cooling and heating thermal loads than traditional mini-utility CHP plants employing downsized HRSGs. One major benefit was the elimination of the code requirement for 24/7 stationary engineers necessary in the conventional CHP base case (note that one full time 40-hour per week operator was still assumed in the ICHP/GCS case). The ICHP/GCS schematically illustrated in Figure 4 lends itself to the use of smaller-footprint, prefabricated vertical hybrid steam generators. These can be mounted on modular skids complete with piping and controls for rapid on-site interconnection with similar functionally integrated equipment, e.g. heat exchangers and pumps that are pre-piped on modular skids with points of connection identified for ease of on-site interconnection prior to charging with HTHTF.

Avoidance of HRSGs also provides the following CHP balance of plant (BOP) advantages:

1) Elimination of cold start time delays,
2) Improved ability to track cyclic rapid building HVAC&R load variations,
3) Elimination of costly "soak period" equipment to reduce HRSG start-up times,
4) Elimination of costly HRSG exotic materials to withstand thermal cycling, and
5) Need for more frequent inspection of HRSGs to monitor potential header failure.

Claimed advantages of the ICHP/GCS include:

1) Smaller thermal mass of hybrid steam generator permits quick response to varying building HVAC&R loads,
2) Low pressure operation of HTHTF recirculation loop eliminates need for 24/7 stationary engineer code requirement,
3) Reduced CTG exhaust extraction coil pressure drop improves CTG power performance,
4) Lower overall life cycle cost,
5) Reduced installation time and operation complexity, and
6) Reduced CHP system downtime and overall footprint.

ASHRAE's policy statement on Global Warming in effect acknowledges that greenhouse gases are linked to global warming and must now be taken seriously by its members. ASHRAE's MEP members responsible for engineered building facilities lasting 20 to 30 years on average, and can minimize such global warming impacts by advocating sustainability through cost effective CHP today. ASHRAE Building Sustainability goals are likely to be significantly advanced through efficient and value-based on-site CHP systems differentiated using LCC methods.

REFERENCES

Berry, J.B., Mardiat, E., Schwass, R., Braddock, C., Clark, E., Innovative on-site integrated energy system tested, Proceedings of the World Renewable Energy Congress VIII, Denver, CO, 2004.

Berry, J.B., Schwass, R., Teigen, J., and Rhodes, K., Advanced absorption chiller converts turbine exhaust to air conditioning, Proceedings of the International Sorption Heat Pump Conference, Denver, CO, 2005. Paper No. ISHPC-095-2005.

Butler, C.H., 1984, Cogeneration engineering, design, financing and regulatory compliance, McGraw – Hill , Inc.

Kehlhofer, R., 1991, Combined-cycle gas and steam turbine power plants, The Fairmont Press, Inc.

Mardiat, E.R., 2006, Everything is big in Texas: Including CHP, Seminar 36 Video , Real Energy and Economic Outcomes from CHP Plants , ASHRAE 2006 Winter Meeting.

Meckler, M., 1997, Cool prescription: hybrid cogen/ice-storage plant offers an energy efficient remedy for a Toledo, Ohio hospital/office complex, Consulting-Specifying Engineer, April.

Meckler, M., 2002, BCHP design for dual phase medical complex. Applied Thermal Engineering, Permagon Press, November, pp. 535-543.

Meckler, M., 2003, Planning in uncertain times, IE Engineer, June.

Meckler, M., 2004, Achieving building sustainability through innovation, Engineered Systems, January.

Meckler, M., Hyman, L.B., 2005, Thermal tracking CHP and gas cooling, Engineered Systems, May.

Piper, J., 2002, HRSG's must be designed for cycling, Power Engineering, May, pp. 63-70.

Orlando, J.A., 1996, Cogeneration design guide, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.

Pathakji, N., Dyer, J., Berry, J.B., Gabel, S., 2005, Exhaust-driven absorption chiller-heater and reference designs advance the Use of IES Technology, Proceedings of the International Sorption Heat Pump Conference, Denver, CO, 2005. Paper No. ISHPC-096-2005.

Payne, F.W., 1997, Cogeneration management reference guide. The Fairmont Press, Inc.

Punwali, D.V., Hulbert, C.M., 2006, To cool or not to cool, 2006, Power Engineering, February, pp.18-23.

Swankamp, R, 2006, Handling nine-chrome steel in HRSG's: steam-plant industry wrestles with increased use of P91/T91 and other advanced alloys, Power Engineering, February, pp. 38-50.