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INTEGRATION OF AN INDIRECT-FIRED ABSORPTION CHILLER IN THE MICROTURBINE-BASED CHP SYSTEM

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ABSTRACT

Combined Cooling, Heating and Power (CHP) Systems offer the potential for a significant increase in the United States' (U.S.) fuel use efficiency by generating electricity onsite near the load and recycling the exhaust gas for heating, drying, cooling, and/or dehumidifying. A key challenge for CHP is the efficient and cost-effective integration of distributed generation (DG) equipment with thermally-activated technologies (TAT) and the compact/cost-effective packaging of such a system. The research and development performed at the Oak Ridge National Laboratory's (ORNL) CHP Test Laboratory focuses on assessing the operational and emissions performance of current DG and TAT systems operated both individually and in combination as a CHP system; developing and verifying mathematical models of the individual components and overall CHP systems; and providing data and model results for supporting the development of test protocols and standards for assessing CHP technologies.

Results of the performance study of an indirect-fired single-effect absorption chiller (AC) incorporated in the above-mentioned CHP system are presented. The exhaust gas from the microturbine generator (MTG) was used as input to the air-to-water heat recovery unit (HRU) to heat water; the hot water, in turn, was used to produce chilled water in the AC. The AC performance was studied with two different generations of the HRU (same manufacturer). The first generation HRU had an effectiveness of 76%, and the second generation HRU has an effectiveness of 92%. Test results indicate an improvement of AC and CHP performance with the use of the HRU with higher effectiveness and high speed of cooling tower fan operation.

Keywords: Cooling, Heating, and Power (CHP); absorption chiller; distributed generation (DG); waste heat recovery.

NOMENCLATURE

Abbreviations:

AC	– absorption chiller
AHU	– air handling unit
CHP	– cooling, heating, and power
COP	– coefficient of performance
CT	– cooling tower
DG	– distributed generation
DOE	– U.S. Department of Energy
E	– efficiency
HHV	– higher heating value (*i.e.*, of natural gas)
HRU	– heat recovery unit (air-to-water heat exchanger)
HVAC	– heating, ventilation, and air conditioning
MTG	– microturbine generator or micro-turbogenerator or microturbine for short
ORNL	– Oak Ridge National Laboratory
TAT	– thermally-activated technologies

Variables:

C_P	– heat capacity, kJ/kg.^{o}C
G	– volumetric flowrate, m^3/min
Q	– heat input, thermal input, cooling capacity, kW
t	– temperature, ^{o}C
W	– electric power, kW
ρ	– density, kg/m^3

Subscripts:

chw	– chilled water
in	– input, inlet
out	– output, outlet

INTRODUCTION

The centralized generation model that has been used by the electric power industry for several decades is confronting a number of economic, technical and environmental problems including long lead times, high capital cost, and the need for modernization of the transmission and distribution systems, electric generation's recent price pressure on natural gas and significant environmental impact. At the same time, digital electric loads (i.e., data centers) are demanding better power quality and higher reliability (more than the four 9s, 99.99%, offered by central generation). All of these issues are reasons to pursue other forms of electric generation such as distributed generation (DG) that is located near the end-use load. Further, DG technology is becoming more reliable, efficient, prevalent, and less expensive. In addition, DG can reduce power delivery losses on the transmission and distribution lines by placing the generation next to the load. In a report prepared in 2001 by the National Energy Policy Development Group, the concept of combined Cooling, Heating and Power (CHP), is identified as a strategy for addressing increased energy demands and peak power issues [1]. Recent developments in DG technologies have opened new opportunities for relatively small-scale CHP that can be used in buildings. DG in combination with thermally-activated technologies (TAT) use waste heat for heating, desiccant dehumidification and/or absorption cooling and provide important opportunities for CHP to be a viable technology for buildings [2, 3].

Microturbine generator (MTG) technology, as a prime mover, currently represents 400 kW or smaller sized units that have efficiencies of 25%[1] or lower (including parasitic losses such as the natural gas compressor). However, most MTGs would have a much lower efficiency without the use of a recuperator. The efficiency of the current technology is limited by the fact that more than two-thirds of the energy is in the form of heat which is exhausted out the stack. In order to increase the overall efficiency of current MTGs above 50%, the MTG must be combined with waste heat recovery technology like TAT desiccant systems and/or absorption technology [2].

The CHP Test Laboratory, which is located on the North-end of the ORNL Campus, was commissioned in 2001. The goal of the CHP program is to increase the overall energy efficiency of DG systems by integrating them with waste heat recovery and TAT. The TAT systems use the DG's hot exhaust gas (by-product of power generation) to produce heating, cooling and/or dehumidification. It provides drying to regenerate desiccant material used by the dehumidification systems. The scope of the facility is to test DG in combination with TAT for optimum waste heat recovery and overall energy efficiency. The objectives of the laboratory include [4]:

- Collection of performance data on current DG and TAT both individually and operated as an integral part of a CHP System,
- Identification of component and system improvements for "Next Generation" CHP products and applications,
- Development of models of the individual devices and verification of a CHP System model based on integrated operation, and
- Providing support for the development of testing protocols and standards for assessing CHP technologies.

The CHP Test Laboratory has a flexible test-bed configuration for testing various heat recovery systems (Figure 1) in conjunction with the DG. The exhaust gas from the DG can either be used directly and/or routed to an air-to-water heat exchanger (also referred to as a heat recovery unit or HRU). The exhaust gas and water flows from the HRU can be varied and directed via automated damper controls to test various CHP configurations and operating modes. The exhaust gas can be conditioned with outside air in an air-mixing chamber. The MTG has been operated individually as well as integrated with various waste heat recovery configurations. Although, the current configuration at the CHP Laboratory only includes MTG-based CHP; it could be extended to encompass many other DG systems such as reciprocating engines and fuel cells. The plans for 2005 include installation of a reciprocating engine genset to test with the TAT equipment at the CHP Test Laboratory.

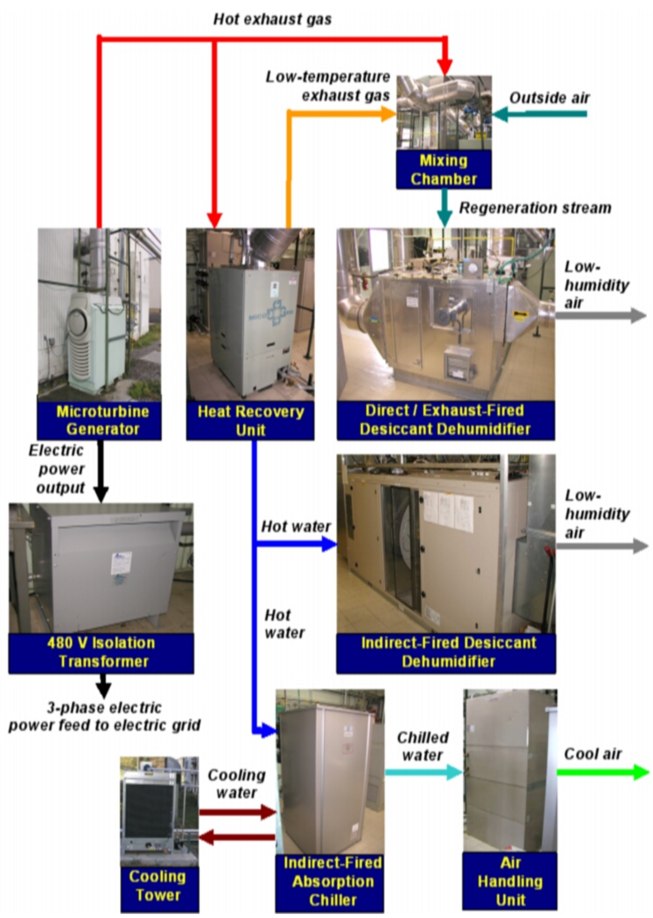

Figure 1. CHP Laboratory at Oak Ridge National Laboratory.

[1] Hereinafter all the efficiencies and coefficients of performance given in this paper are calculated at higher heating value (HHV) of natural gas

EXPERIMENTAL SET-UP

The heat recovery components used with the MTG in this study consisted of an air-to-water HRU, and a hot water-fired (indirect-fired) single-effect lithium bromide (LiBr) ─ water 10-ton (35 kW) absorption chiller (AC) equipped with a cooling tower (CT) and air handling unit (AHU). Two generations of HRUs were investigated: the first generation HRU had lower effectiveness and heat recovery (76% and 43 kW, respectively) and the second generation HRU has higher effectiveness and heat recovery (91% and 50 kW, respectively). The second generation HRU was developed, in part, based on results for the earlier tests of the first generation unit at the CHP Test Facilty. The MTG, which is located on the outside of the CHP Laboratory building, is a three-phase 480 VAC/30 kW rated grid-connect unit that can operate at 50 or 60 Hz, although it is operated at 60 Hz, and at unity power factor. The natural gas-fired microturbine and the electric generator of the MTG are on the same shaft. The MTG, which is designed to operate at a maximum speed of 96,000 rpm, produces high-frequency alternating current power that is rectified to direct current and then converted to 50 or 60-Hz alternating current power by the power conditioning electronics of the digital power controller [5].

In the CHP mode, the exhaust gas comes off of the MTG recuperator at a temperature of ~275^{o}C. It then passes through the HRU and leaves it at a temperature of ~120^{o}C (for the 1st generation HRU) or ~96^{o}C (for the 2nd generation HRU). The hot water produced in the HRU has a temperature of ~175^{o}C (for the 1st generation HRU) or ~182^{o}C (for the 2nd generation HRU) and is directed to the generator of the AC. It should be noted that the 2nd generation HRU has double the heat transfer area and improved effectiveness that resulted in part from the backpressure testing at the CHP Test Laboratory, which showed minimal effect on the performance of the MTG up to 0.02 atm (8 ”wc).

TEST PROCEDURES

Previous tests at the CHP Test Laboratory on the CHP-based performance of the AC [6] showed that the heat output produced with the 1st generation HRU was not enough to drive the AC at full or close to full rated load of 35 kW of cooling. This results in the generation of higher chilled water temperatures (11-12^{o}C) as compared to the design point operation of the AC with 7^{o}C chilled water temperature (full-load operation). The aim of these tests was to study the CHP-based performance of the AC using thermal input from the 2nd generation HRU with higher effectiveness and heat recovery close to 50 kW.

The current series of tests studied the behavior of cooling capacity (Q_{AC}) and coefficient of performance (COP) of the AC, and overall CHP efficiency over the ambient temperature range typical for the AC operation (26-30^{o}C). These tests were performed at a high CT fan speed of 1,740 rpm and an MTG power output setting of 30 kW. Net power output of the MTG depended on ambient temperature and was around 22 kW at the time of the tests. The test flow parameters included hot water flow rate from the HRU, chilled water flow rate from the AC, and cooling water flow rate from the CT. These values are listed in Table 1.

Table 1. Flow Parameters of the Tests with Two Different HRUs.

Parameter	The 1st generation HRU tests	The 2nd generation HRU tests
Hot water flowtare, m^3/min	0.14	0.14
Chilled water flowtare, m^3/min	0.11	0.11
Cooling water flowrate, m^3/min	0.26	0.24

Also the effect of the CT fan speed on the CHP-based AC performance with the 2nd generation HRU was investigated for two different fan speeds: 355 rpm (low fan speed) and 1,740 rpm (high fan speed). The water flowrates in both cases were the same (0.14 m^3/min of hot water, 0.11 m^3/min of chilled water, and 0.24 m^3/min of cooling water).

The AC cooling capacity (Q_{AC}) is defined as:

$$Q_{AC} = C_{p\,chw} \cdot \rho_{chw} \cdot G_{chw} \cdot (t_{chw\,in} - t_{chw\,out}) \qquad (1)$$

where $C_{P\,chw}$ is the water heat capacity at the average temperature; ρ_{chw} is the density of water at the average temperature; G_{chw} is the volumetric flow rate of chilled water; and $t_{chw\,in}$ and $t_{chw\,out}$ are the chilled water temperatures entering and leaving the AC unit, respectively.

Efficiency of the overall CHP system consisting of the MTG, HRU, and AC is defined as:

$$E = (W_e + Q_{AC})/(Q_{in\,HHV} + W_{total}) \cdot 100\ \% \qquad (2)$$

where W_e is the net electric power generated by the MTG, Q_{AC} is the AC cooling capacity, $Q_{in\ HHV}$ is the natural gas input (based on the HHV of natural gas), and W_{total} is the total electric power consumed by the HRU, AC, CT, and pumps.

COP of the AC is defined as:

$$COP = (Q_{AC} / Q_{HRU\,(AC)}) \qquad (3)$$

where Q_{AC} is the AC cooling capacity and $Q_{HRU\,(AC)}$ is the heat supplied by the HRU to the AC. It should be noted that this value may be 5-10% less than the heat recovered by the HRU due to heat losses in the hot water loop from the HRU to the AC.

The test instrumentation and the measurement accuracies are given in Table 2.

Table 2. Instrumentation Used in the CHP Tests.

Measurement	Sensor	Range	Precision
Temperature	Resistive temperature detector (RTD)	-200 to 850°C	±0.1°C
Water flow	Flow meter	0 to 0.38 m^3/min	±1%
Natural gas flow	Test meter	0 to 11.8 m^3/h	±0.2%
Natural gas pressure	Pressure transducer	0 to 50 kPa	±0.5% of full scale
Power	Watt transducer	0 to 40 kW	±0.5% of full scale

DISCUSSION

CHP-Based Performance of Absorption Chiller with Two Different Heat Recovery Units

Replacement of the 1st generation HRU with the 2nd generation HRU, which has an improved effectiveness and heat recovery, had positive effect on the AC performance. Figures 2-4 show the change in AC cooling capacity, AC outlet temperature, and CHP efficiency. Cooling capacity increased from 28 to 32 kW (14% improvement), although maximum rated cooling capacity (35 kW) still was not achieved. There was significant reduction in chilled water temperature at the AC outlet, from 11.5 to 8.5°C (by 3°C). Overall CHP efficiency also increased from 40 to 43%, but the COP of the AC stayed the same – approximately 0.67.

One of the reasons the rated AC performance was not achieved is due to thermal losses in the hot water line between the HRU and AC which can be up to 10% of the heat recovered by the HRU. Assuming zero losses in the hot water line, and the AC COP value of 0.67, then the AC cooling capacity may reach 34 kW, which is close to the rated cooling capacity. Therefore, minimization of thermal losses between separate components of the CHP system is one of the most important tasks needed to optimize efficiency: this could be achieved in integrated packaged systems where all components are packaged as close together as possible.

It should be noted that all AC performance parameters over the temperature range studied did not change significantly. This is due to the fact that an increase in the MTG exhaust gas temperature entering the HRU with increased ambient temperature was compensated by a decrease in MTG electric power output and amount of the exhaust gas resulting in almost the same HRU heat output coming to the AC over the temperature range studied.

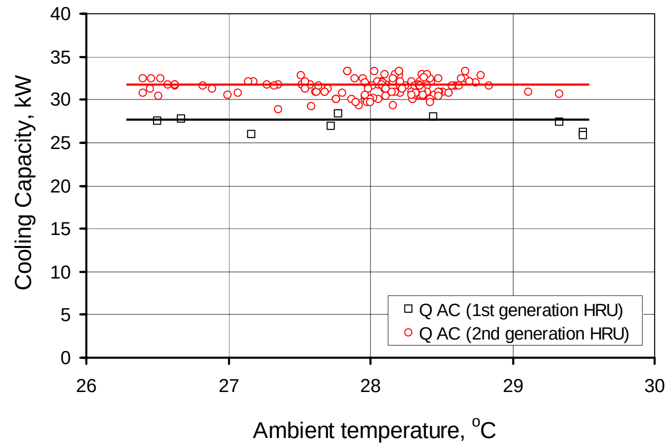

Figure 2. Cooling Capacity of Absorption Chiller (AC) Operated with the 1st and the 2nd Generation Heat Recovery Unit (HRU).

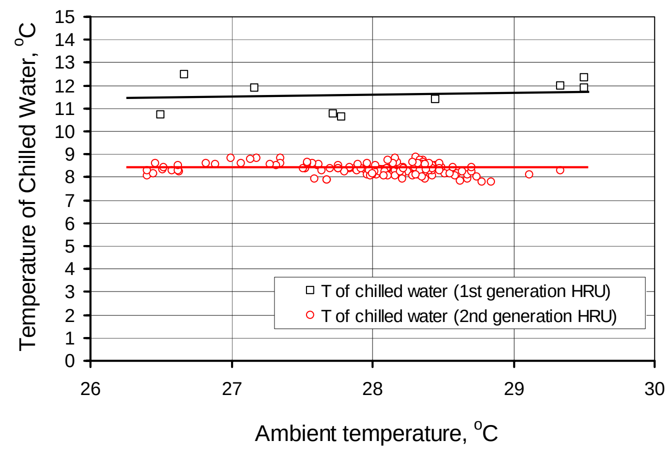

Figure 3. Chilled Water Temperature at the Outlet of Absorption Chiller (AC) with the 1st and the 2nd Generation Heat Recovery Unit (HRU).

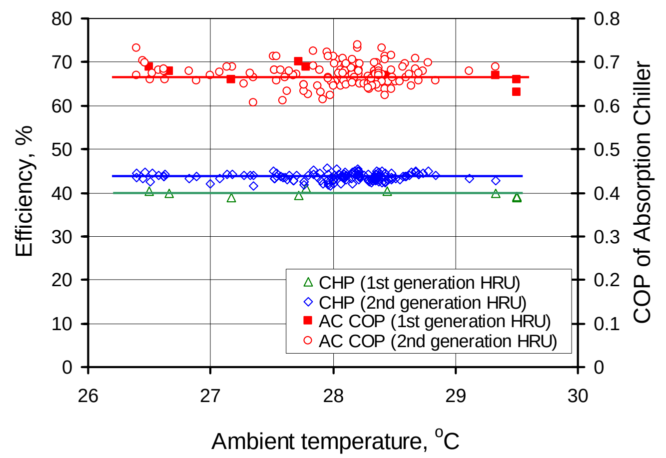

Figure 4. Overall CHP Efficiency and COP of Absorption Chiller (AC) with the 1st and the 2nd Generation Heat Recovery Unit (HRU).

Effect of Cooling Tower Fan Speed on the Absorption Chiller Performance

Increasing the fan speed of the CT that supplies cooling water to the AC, i.e. intensification of air cooling in the CT, results in some increase in absolute and relative performance parameters of the AC. For example, as shown in Figure 5, the AC cooling capacity increases by 3 kW from 29 to 32 kW (10% improvement). Overall CHP efficiency and COP of AC also increased by 2% and 4% respectively (Figure 6). This indicates that, in spite of the obvious increase in total power consumption during high speed operation of the CT fan, absolute and relative performance parameters of the AC and CHP as a whole improved.

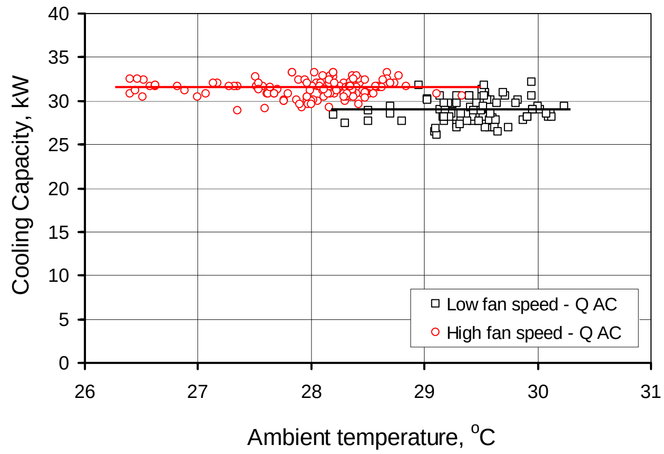

Figure 5. Cooling Capacity of Absorption Chiller (AC) Operated with the 2nd Generation Heat Recovery Unit (HRU) at two Different Cooling Tower Fan Speeds.

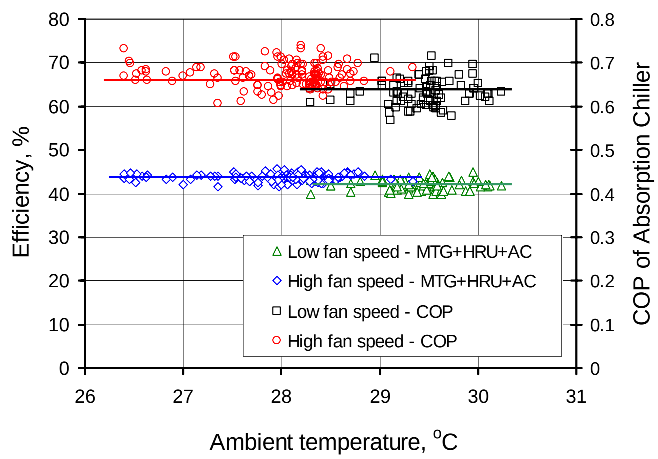

Figure 6. Overall CHP Efficiency and COP of Absorption Chiller (AC) Operated with the 2nd Generation Heat Recovery Unit (HRU) at two Different Cooling Tower Fan Speeds

CONCLUSIONS

Performance tests of a CHP configuration consisting of a microturbine generator, two different generations of heat recovery unit (HRU), and absorption chiller (AC) were performed at the CHP Test Laboratory. The tests were performed to evaluate the benefit of the improved (2nd generation) HRU with higher heat exchanger effectiveness. The effect of different modes of cooling tower operation on AC and CHP performance was also assessed. The tests revealed that the 2nd generation HRU with higher effectiveness provided better heat recovery and as a result, improvements in cooling capacity, chilled water temperature, COP of the absorption chiller, as well as overall CHP efficiency. Cooling capacity increased from 28 to 32 kW (a 14% improvement), although maximum rated cooling capacity of 35 kW was still not achieved. One of the reasons for this lost capacity is the thermal losses in the hot water line between the HRU and the AC. Therefore, for a packaged CHP system, emphasis should be placed on the minimization of thermal losses between components of the CHP system. There was significant reduction in chilled water temperature at the AC outlet, from 11.5 to 8.5°C, but due to the above-mentioned reasons, the rated chilled water temperature (7°C) was not achieved. Overall CHP efficiency also increased from 40 to 43%, but the COP of the AC remained nearly constant (~0.67). Application of high speed cooling fan operation also resulted in an improvement of AC and CHP performance parameters: AC cooling capacity increased from 29 to 32 kW (a 10% improvement), AC COP increased from 0.65 to 0.67, and overall CHP efficiency increased from 41 to 43%. Ambient temperature over the range studied did not have a significant impact on AC performance due to the counteracting actions of the MTG power output and MTG exhaust gas with increasing ambient temperature. The MTG electric power output decreased with the increase in ambient temperature while the amount of exhaust gas supplied to the HRU by the MTG increased.

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