# DEVELOPMENT OF A "HYBRID" ABSORPTION CHILLER FOR COMBINED HEAT AND POWER SYSTEMS

*Ed Bludnicki*
*Product Manager*
*UTC Power*
*South Windsor, CT USA*

*Timothy C. Wagner, Ph.D.*
*Principal Engineer*
*UTRC*
*East Hartford, CT USA*

*Sung-Han Jung*
*Research Engineer*
*UTRC*
*East Hartford, CT USA*

## ABSTRACT

The objective of this effort was the development and testing of an absorption chiller optimized for integration with waste heat from a reciprocating engine in a Combined Heat and Power (CHP) system. The design, which can provide 30% more cooling capacity than a typical single-effect absorption chiller, has been named a "hybrid chiller" due to the fact that it couples a double-effect cycle using exhaust energy and a single-effect cycle using jacket water energy. In this paper, the design of the chiller will be described along with presentation of results from component and system testing of a 387 kW full-scale prototype.

**Key Words:** ***absorption chiller, combined heat and power (CHP), reciprocating engine***

## 1 INTRODUCTION

The Department of Energy has identified Combined Heat and Power (CHP) systems as a critical part of the U.S. energy strategy for achieving current and future energy and environmental goals (LeMar 2002). Effective application of CHP systems is critically dependent on the effective utilization of the thermal energy from the prime mover. A variety of prime movers for CHP systems have been studied, including microturbines, fuel cells, and reciprocating engines (Lazzarin et al. 2000, Maidment et al. 1999, Nayak et al. 2005, Patnaik 2004, Wagner 2004). Reciprocating engines offer an attractive customer value proposition for some applications due to their low cost, high efficiency and wide market acceptance. A disadvantage of reciprocating engine based CHP systems is that the waste heat is split between the exhaust flow at a relatively high temperature and the jacket water at a relatively low temperature. The presence of two waste energy streams results in a lower overall energy recovery capability, especially with regard to driving an absorption chiller. However, the absorber capacity can be enhanced by utilizing the two heat streams separately in a two stage absorber (ASHRAE 2004). The technical challenge in this type of chiller is to enable robust operation for a wide range of engine types and operational scenarios. The objective of the current effort was to develop an absorption chiller optimized for integration with a reciprocating engine and demonstrate stable and efficient operation for multiple engine types and operating scenarios.

There are three types of natural gas reciprocating engine technologies that can be considered for CHP applications: lean combustion, rich combustion, and stoichiometric combustion with cooled exhaust gas recirculation (EGR). Waste heat from all three engine types is available from the exhaust gas (390°C – 600° C) and the engine jacket coolant (85 - 95°C). As engine efficiency increases, the proportion of waste heat available from the jacket water increases and the total waste heat available decreases; that is, the overall quality of the waste heat available decreases as the engine becomes more efficient. One of the key requirements for the cooling technology developed in this study is the flexibility to be seamlessly coupled with any of the aforementioned engine technologies.

In this paper, the modular approach to system design will be presented, the chiller design will be described and results will be presented from both component and system tests.

## 2 MODULAR CHP CONCEPT

The modular CHP system concept shown in Figure 1 was selected following a rigorous concept generation and evaluation process because it provided the best combination of modularity, system flexibility and engine independence.

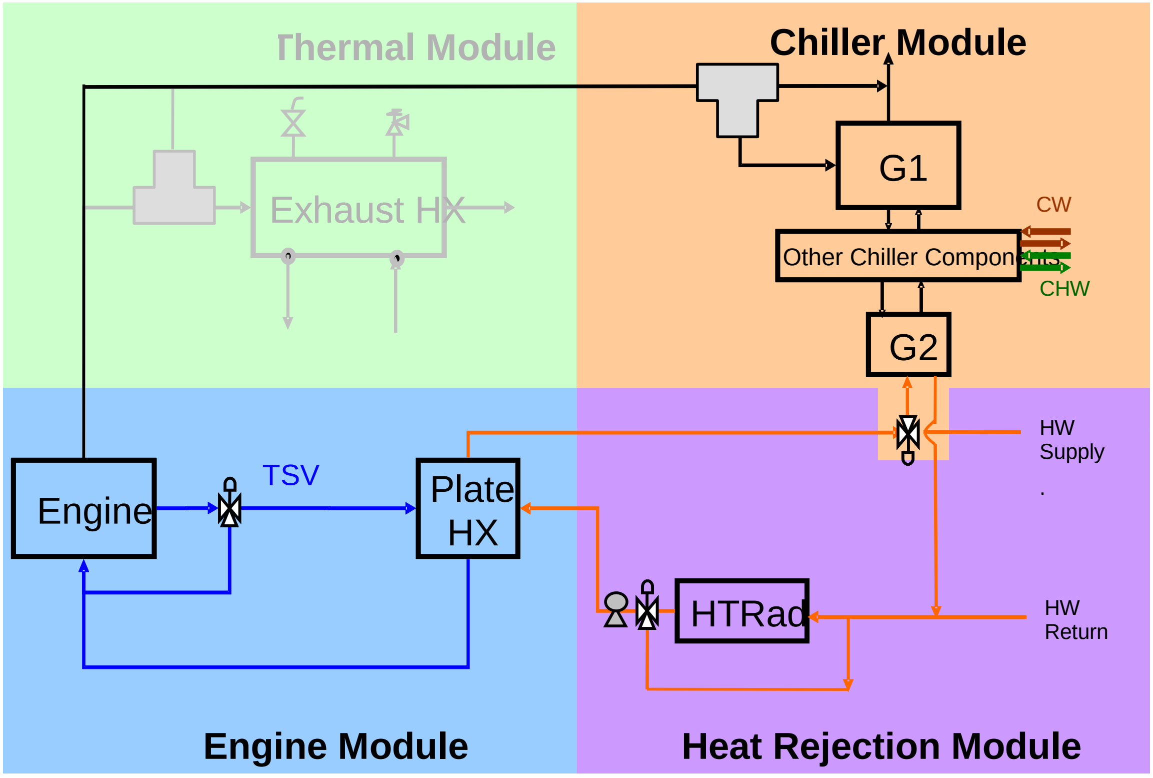


**Figure 1: Modular System Concept**

This concept can be considered as four separate modules. The Engine Module consists of the engine/genset and its controller, as well as its thermal management equipment and a heat exchanger to reject the cooling jacket heat into a secondary fluid for further use in the CHP system. The Chiller Module includes the diverter valve, the hybrid absorption chiller, and the three-way valve to control the flow of hot fluid into the low-stage generator (G2). The chiller controller modulates the diverter valve and three-way valve to achieve the chilled water setpoint in cooling mode, and the hot water setpoint in heating mode. The Heat Rejection Module contains the radiator, radiator bypass and control valve, and the hot water supply and return from the customer supplied hot water heat exchanger. This module rejects excess heat from the intermediary fluid and enables any hot fluid not used by the chiller to be used to generate hot water. Using this module, the customer may obtain simultaneous heating and cooling when the chiller is in cooling mode. The Thermal Module is an optional module which may replace the chiller if the customer does not require cooling. Alternately, the thermal module may be included in the system with the chiller if the customer requires full simultaneous trigeneration; however, this comes at the cost of a second diverter valve and a local controller to coordinate the two diverter valves with the hot water and chilled water setpoints. The thermal module was not demonstrated in the current effort.

## 3 HYBRID CHILLER DESIGN

The design philosophy adopted for the hybrid absorber was that the chiller should be operable with the waste heat available from a wide selection of natural gas engine technologies. Furthermore, consideration was given to the future potential to run the chiller using waste heat from other prime movers and waste heat sources, including diesel/liquid fueled engines, renewable energy sources, and solar hot water. With this in mind, the chiller performance was modeled for a variety of engine conditions. Steady-state and dynamic models of the hybrid chiller and its components were constructed and used to support the machine design and select the cycle configuration. These models were later used in system level models to support system configuration studies. High fidelity simulations of the high- and low-stage generators were used to size and configure the tube banks. The dynamic models were used to predict the control performance of the chiller under various start-up, shut-down, and transient conditions, including operation with a single waste heat stream.

The hybrid chiller is a modified double-effect lithium bromide (LiBr)/water absorption cycle which utilizes waste heat inputs to drive the high- and low-stage generators. The chiller uses a parallel flow cycle and employs a fixed speed pump and bucket trap arrangement to control solution flow and level in the generators as shown in Figure 2. A conceptual rendering is shown in Figure 3.

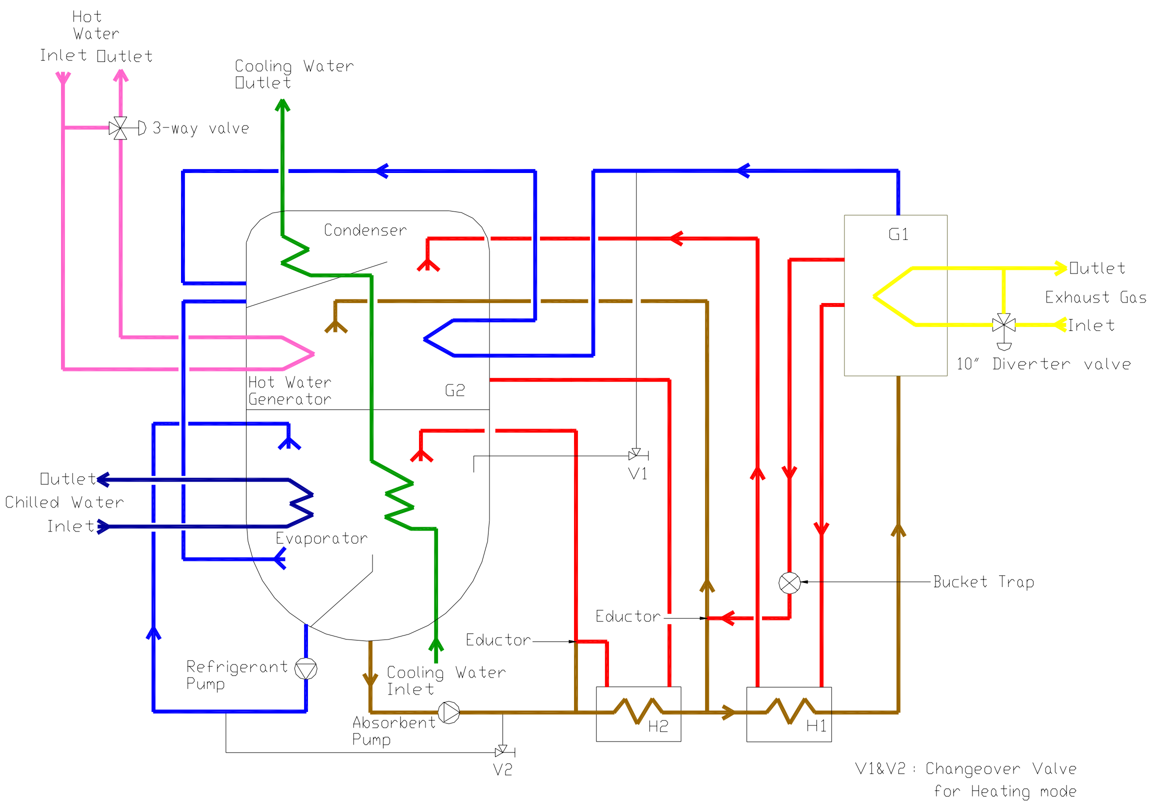


**Figure 2: Schematic of Hybrid Absorption Chiller**

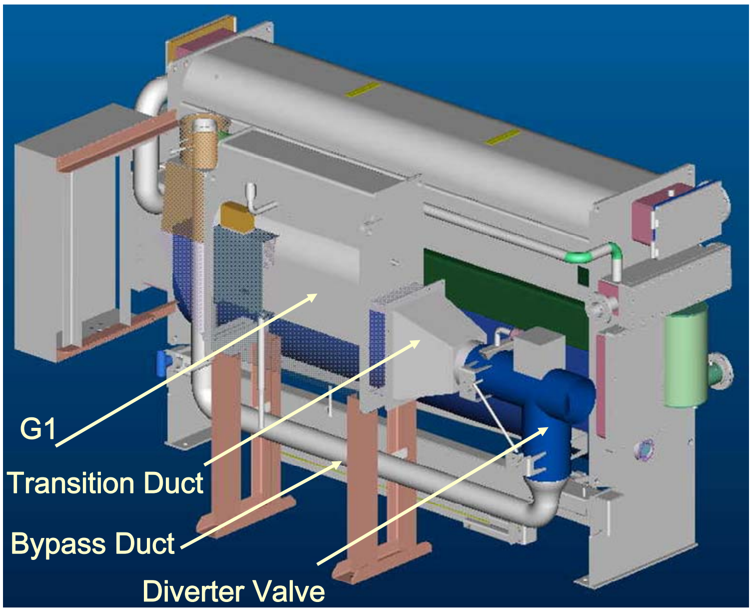


**Figure 3: Conceptual rendering of the hybrid chiller**

## 4 CHILLER TESTING

The hybrid chiller component testing was conducted using hot air from a hot gas generator and hot water from a steam-driven hot water source as shown in Figure 4, to simulate reciprocating engine exhaust and jacket water.

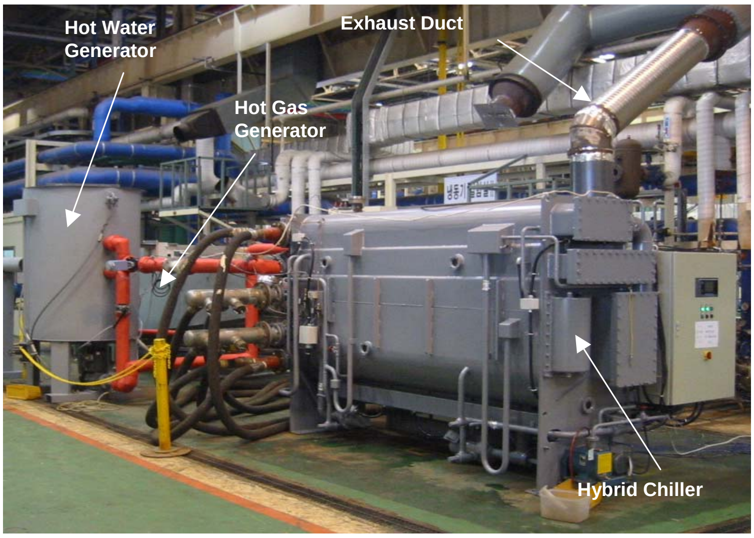


**Figure 4: Photograph of the hybrid chiller and waste heat simulation system**

Heated air and water flows simulating two types of reciprocating engines were selected to benchmark the performance of the hybrid chiller. The first condition simulated a 365 kWe, 33-34% efficient (LHV), rich burn engine; the second condition simulated a 360 kWe, 38% efficient (LHV), rich burn, turbocharged and intercooled engine with exhaust gas recirculation. The hot gas generator used natural gas and excess air to control exhaust flow and temperature. As a result, the exhaust stream from the hot gas generator had significantly lower water content than the exhaust from a rich burn reciprocating engine. Property simulations indicated that the heat capacity of hot gas generator exhaust was 14%

lower than the heat capacity of the exhaust from the rich burn reciprocating engine under consideration. Therefore, the rich burn engine was simulated by increasing the air flow from the hot gas generator to 115% of the published flow rate for this type of engine. The use of a higher flow rate also results in higher velocities in the high-stage generator, potentially reducing heat exchanger effectiveness. A correction factor determined by heat transfer modeling was applied, although the effect on the high-stage generator overall heat transfer coefficient was found to be negligible (< 1%).

For the rich burn engine with EGR, analysis showed that the hot gas generator exhaust would have about 12% higher heat capacity than the engine exhaust. Therefore, the hybrid chiller was tested using a hot air flowrate 12% higher than the published flow rate for this type of engine. The impact of the increased gas velocity on the overall heat transfer coefficient of the high-stage generator was accounted for, although the impact on heat exchanger effectiveness was less than 1%. A summary of the test results for both simulated conditions is presented in Table 1.

**Table 1: Performance with Simulated Waste Heat from Rich Burn and Lean Burn Engines**

| | Rich<br>365 kW<br>33.5% | Lean<br>360 kW<br>38% |
|---|---|---|
| Cooling Capacity (kW/RT) | 494 / 140.4 | 378 / 107.5 |
| COP | 0.871 | 0.803 |
| Hot Gas Mass Flowrate (kg/s) | 0.479 | 0.391 |
| Hot Gas Entering Temperature (°C) | 546 | 378 |
| Hot Gas Leaving Temperature (°C) | 153 | 138 |
| Hot Water Flow Rate ($m^3/h$) | 41.5 | 33.7 |
| Hot Water Entering Temperature (°C) | 91 | 90 |
| Hot Water Leaving Temperature (°C) | 83 | 80 |

## 4.1 Start up tests

Start-up testing was conducted to demonstrate successful start-up and appropriate chiller response to failure modes in the CHP system such that only one form of waste heat is available (i.e., exhaust or jacket coolant). Furthermore, the testing was used to detect any problems with the start-up control logic. The normal start-up of the hybrid chiller with full waste heat availability was successfully demonstrated. The test was conducted with 540 °C exhaust and 80°C hot water flow to the low stage generator. The diverter valve was opened 50% until the feedback signal reached 50%. Then, the jacket water was opened concurrently with the diverter valve until full heat input was achieved. The start-up sequence was found to be stable and appropriate. In the next test, the hybrid chiller start-up was tested using waste heat input to the low-stage generator alone. Start-up was successfully demonstrated, although solution pump cycling was observed due to solution level fluctuations in G1 and the bucket trap. Given that the chiller will only operate in this mode in the unlikely event of exhaust delivery failure, the solution pump cycling is not a significant concern. In the next test, the hybrid chiller start-up was tested with waste heat input to the high-stage generator alone. The test was conducted with the hot water supply system turned off. Start-up was successfully demonstrated with no observed problems.

### 4.2 Envelope testing

Envelope testing was conducted to demonstrate the robustness of the chiller design and chiller controls at the limits of operation. The most difficult operating conditions occur at maximum load and maximum cooling water temperature, minimum load and minimum cooling water temperature, and maximum load and minimum chilled water leaving temperature.

When the chiller is operating at maximum load and maximum cooling water temperature, the LiBr/water solution concentration in the cycle is increased, and there is a risk of crystallization. This test is designed to demonstrate that the chiller can operate in full load and maximum cooling water temperature without developing crystallization problems. The test results indicated that the chiller can successfully operate at maximum load and maximum cooling water temperature. Refrigerant overflow was observed in the evaporator, and this overflow serves to dilute the solution in the absorber and thus protect the chiller against crystallization. The observed capacity was 250 kW, or 51% of the design capacity at a cooling water temperature of 35.8°C.

When the chiller is operating at maximum load and minimum chilled water leaving temperature, the log-mean temperature difference in the evaporator is at a minimum, and the refrigerant evaporation rate is low. Consequently, the solution concentration in the absorber may rise, and crystallization is a risk. To counter the risk, the evaporator is designed with an overflow to dilute the solution and prevent crystallization. The test results indicated that the chiller is operable at maximum load and minimum chilled water conditions. Some refrigerant overflow was observed as expected, and the chiller capacity was reduced to 436 kW, or 89% of design, at a chilled water temperature of 5.1°C. The reduction in capacity is acceptable, and the test was deemed successful.

When the chiller is operating at minimum load and minimum cooling water temperature, there is a risk of refrigerant pump cavitation due to low refrigerant level in the evaporator. The purpose of the test is to determine if the minimum refrigerant level in the evaporator design is sufficient to avoid cavitation. The test successfully demonstrated operation at minimum load and minimum cooling water temperature, and no problems with pump cavitation were observed. The chiller capacity was 91 kW, or 19% of design, at a cooling water temperature of 13.8°C.

### 4.3 Transient testing

Transient tests were conducted to check the robustness of the chiller controls. The cooling water inlet temperature to the chiller was varied in order to simulate the impact of fast and slow changes in ambient temperature and humidity on the hybrid chiller performance. The chilled water inlet temperature to the chiller was varied at a fast rate to simulate the impact of abrupt changes in building loads on hybrid chiller performance. The heat input change test simulated the effect of a sudden cycling of heat input on the chiller stability. The chilled water temperature entering the hybrid chiller evaporator was cycled from 17.5°C to 11°C at a rate of 17°C/h. This rate of change is 1.65 times the standard expected rate of change of 10.3°C. The cooling and chilled water flow rates were kept at 166 $m^3/h$ and 78.4 $m^3/h$, respectively. The cooling water temperature was 29 ± 1 °C. The hybrid chiller operation during the chilled water variability testing was stable, and no problems were observed. The chilled water leaving temperature tracked the inlet temperature well and the test was successfully completed.

The cooling water inlet temperature at full chiller capacity was varied at a fast and a slow rate to simulate the impact of changing ambient conditions on chiller performance. An additional test with cooling water temperature varied at a fast rate at part load was also completed. The cooling water temperature was changed between 30.0°C and 25.0°C in 23 minutes (13°C per hour) in the fast rate tests, and between 28.0°C and 23.0°C in 120 minutes in the slow rate test (2.5°C per hour). The cooling and chilled water flow rates were kept at 166 $m^3$/h and 78.4 $m^3$/h respectively. The chiller response to all three test scenarios was stable, and no problems with cycle stability or controls were observed.

## 5 SYSTEM TESTING

A prototype reciprocating engine CHP system utilizing the hybrid absorption chiller was constructed to support the technology development (see Figure 5).

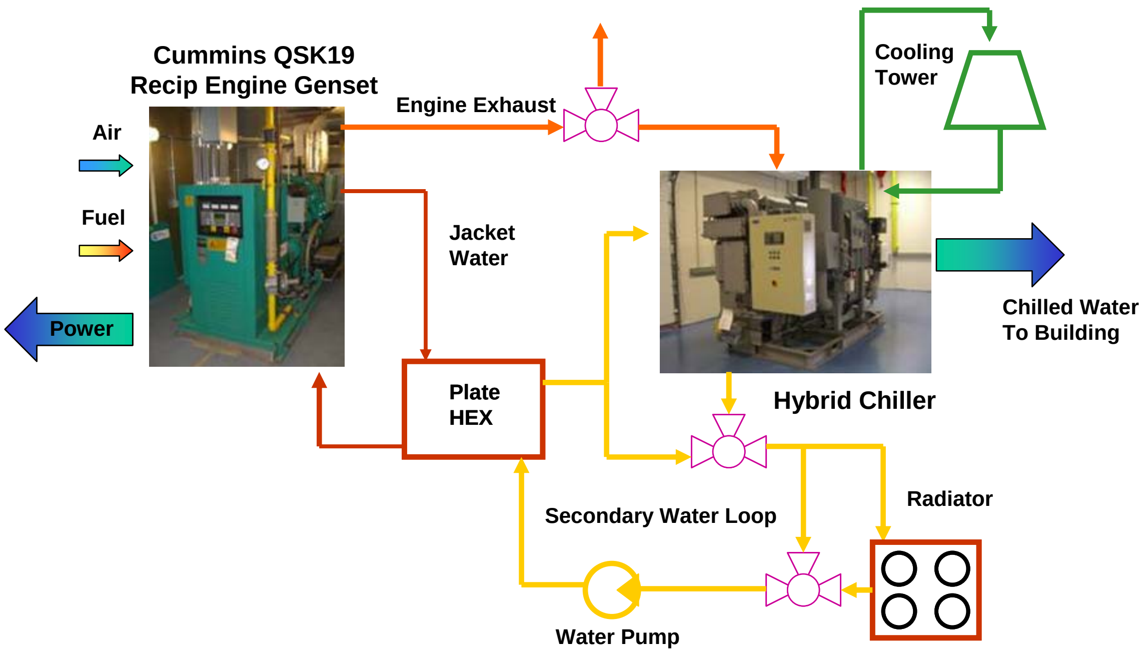


**Figure 5: Reciprocating Engine and Hybrid Chiller Test System Configuration**

The key requirements for the test system were:

- Representative performance, controls, and operability
- Accessibility for troubleshooting and maintenance
- Full instrumentation to meet test requirements
- Safety for operation and maintenance
- Flexibility for future re-use of the test stand with alternate engines and system configurations
- Usability for feasibility demonstration, technology readiness demonstration, and qualification

Test requirements were also established to ensure that the necessary instrumentation and hardware was available on the test stand. The high-level testing requirements were:

- Verification of the manufacturer specifications for engine performance
- Verification of chiller operability
- Verification of robust system operability and performance

- Data collection to support availability and reliability targets
- Demonstration of CHP system performance and efficiency targets.

The engine selection for the system test was driven by the desire to test the hybrid chiller with an engine representative of those currently available on the market. Both rich and lean burn engines were considered, and the lean burn technology was selected because of its wide availability and low emissions without after-treatment. The 334 kW Cummins QSK19 engine was selected based on cost and delivery schedule. The engine was installed with appropriate electrical switchgear and grid paralleling equipment to enable interconnection with the electricity grid.

A detailed test plan based on the test requirements was developed. The objective of the test plan was to verify the operability, controls, and performance of the engine, chiller, system, and balance of plant. In addition, it was desired to collect as much information pertaining to system reliability and availability as possible during the four month test period.

The measured cooling capacity of the hybrid chiller with the 334 kW lean burn engine was 390 kW as opposed to the predicted performance of 404~422 kW. Investigation into the performance shortfall indicated that:

1) The exhaust temperature drop in the duct between the engine exhaust and the chiller inlet was 39$^{o}$C rather than the specification of 8 to 11$^{o}$C. The temperature drop was the result of thermal insulation material with a lower thermal resistance value than specified, leading to degradation in G1 performance.

2) The thermostatic valve for the radiator allowed some leakage of a hot water flow to the radiator at a lower temperature than designed, which resulted in lower hot water inlet temperature to G2, leading to degradation in G2 performance.

When corrected for a short duct length with improved insulation material and a higher G2 entering hot water temperature with a correct thermostatic valve, the predicted chiller capacity is 404 kW. The chiller could also be used with larger engines up to 500 kWe, in which case it would produce correspondingly higher cooling capacity.

Figure 6 shows the results of envelope testing and IPLV (Integrated Part Load Value) testing. The IPLV is defined as:

$$IPLV = 0.01A+0.42B+0.45C +0.12D$$

where A, B, C and D represent the COP at 100%, 75%, 50% and 25%, respectively. The IPLV of the hybrid chiller was 1.20, which is superior to the typical IPLV of 0.7 for single-effect absorption chillers. As indicated in the chiller component envelope testing section, envelope testing was conducted to demonstrate the robustness of the chiller design and chiller controls at the limits of operation. The mechanical design and controls of the chiller properly protected the chiller from crystallization or refrigerant pump cavitation.

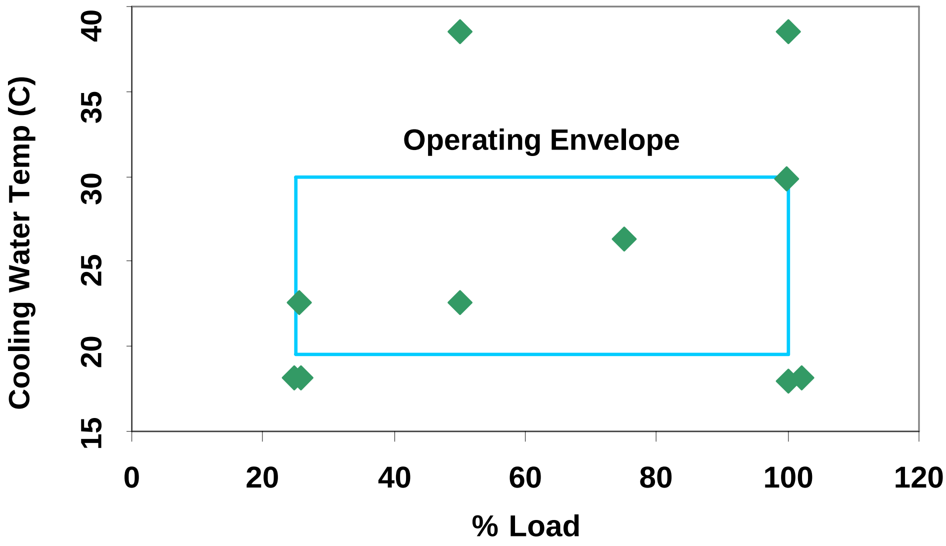


**Figure 6: Envelope and IPLV Testing**

Figures 7 and 8 show the results of two types of chiller transient operation tests. In Figure 7, chiller performance is illustrated when the chilled water return temperature was varied from 21.7 °C to 12.2 °C in 70 minutes, representing a large change in building thermal load. In the second case, the heat input to the chiller was changed by changing the engine power output in the range of 50% (167 kWe) to 100% (334 kWe). The test confirmed that chiller operation is stable and leaving chilled water temperature can be maintained within a desirable range under these conditions. The test results are shown in Figure 8.

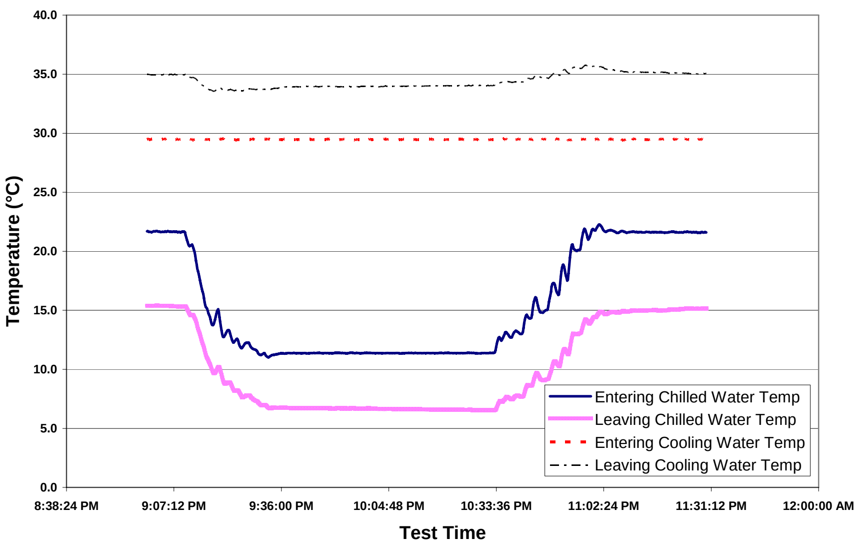


**Figure 7: Results of a Chilled Water Transient Test**

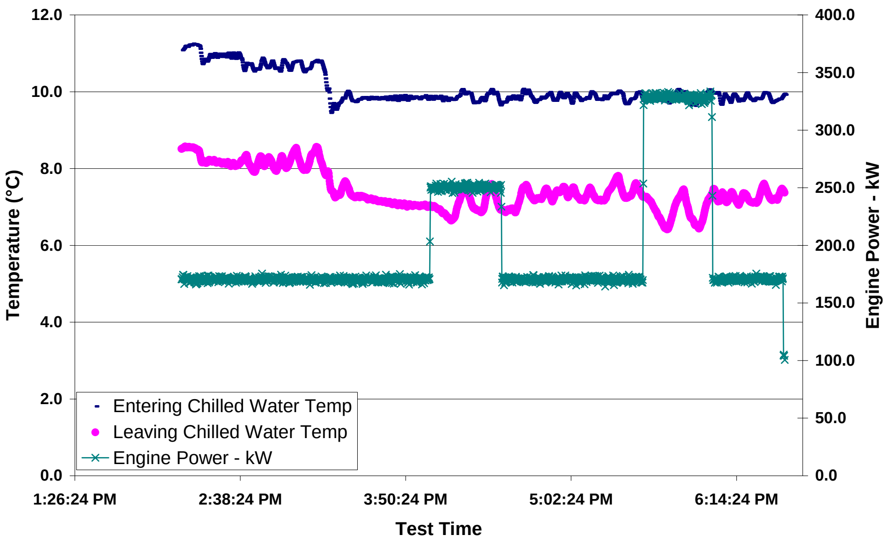


**Figure 8: Results of an Engine Power Transient Test**

The hybrid chiller was tested in heating mode; the capacity using the exhaust energy alone provided 250 kW. Although not tested, analysis showed that a heat rejection module using the jacket water energy alone would provide 200 kW, for a total heating capacity of 450 kW.

Table 2 summarizes system performance for CHP systems with the hybrid chiller versus a single effect chiller. The system with the hybrid chiller provides higher cooling capacities, while the single effect chiller provides higher heating capacities.

**Table 2 Summary of System Performance**

| | Maximum Capacity | | | | System Efficiency [(E + T)/LHV] | |
|---|---|---|---|---|---|---|
| | Electrical Efficiency | Power | Cooling | Heating | Cooling | Heating |
| Recip CHP Hybrid | 31 ± 2 % (<46°C) | 334 kWe | 390 kW | 450 kW | 67% | 73% |
| Recip CHP Single Effect | 31 ± 2 % (<46°C) | 334 kWe | 317 kW | 470 kW | 60% | 80% |

## 6 CONCLUSIONS

A “hybrid” chiller that provides about 30% more cooling capacity than typical single-effect absorption chillers was developed and demonstrated. This enhancement in cooling capacity was achieved by designing the chiller to separately use exhaust heat input into the high stage generator (G1) and hot liquid heat input into the low stage generator (G2) in order to maximize chiller capacity and heat recovery efficiency. The resulting reciprocating engine CHP system is capable of providing trigeneration (simultaneous electricity, cooling and heating). The hybrid chiller can be seasonally changed to heating mode to achieve maximum heating. The hybrid chiller is designed to be flexible and is independent of engine manufacturer in the engine size range of 300-500 kWe.

Chiller testing demonstrated a steady state full-load performance cooling capacity of 390 kW when integrated with a 334 kWe lean burn reciprocating engine. With proper integration, analysis showed a capacity of 404 kW could be attained. Start-up testing under the three different waste heat conditions was successfully completed: 1) full-waste heat availability 2) heat to low-stage generator only, and 3) heat to high-stage generator only. Envelope testing was performed to demonstrate that the mechanical design and control algorithm of the hybrid chiller would function properly to protect the chiller from crystallization or refrigerant pump cavitation. Transient testing was conducted under five different transient conditions, including variations in entering chilled water temperature, entering cooling water temperature, and heat input. The hybrid chiller demonstrated good chilled water temperature control under all of the transient test conditions.

During the course of this program, the hybrid chiller technology was successfully developed and demonstrated, and predicted performance was achieved in both a component and a system level.

## 7 ACKNOWLEDGEMENTS

The authors would like to gratefully acknowledge the support of the U.S. Department of Energy and the significant interaction of Robert DeVault at Oak Ridge National Laboratory.

## 8 REFERENCES

ASHRAE 2004. “ASHRAE Handbook-HVAC Systems and Equipment,” p. 7.28, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta.

LeMar, P. 2002 “Integrated Energy Systems (IES) for Buildings: A Market Assessment: (Final Report),” Resource Dynamics Corporation for Oak Ridge National Laboratory, U.S. Departement of Energy Contract No. DE-AC05-00OR22725.

Lazzarin, R.M., Longo, G.A., and Gasparella, A. 2000, “Analysis of an Absorption Chiller Driven by the Heat Recovery on an I.C. Engine,” Purdue University, pp. 537-545.

Maidment, G.G., Zhao, X., Riffat, S.B., and Prosser, G. 1999, “Application of Combined Heat-and-Power anad Absorption Cooling in a Supermarket,” Applied Energy, Vol. 63, pp. 169-190.

Nayak, S., Radermacher, R., Moran, D., and Garland, P. 2005, “Performance of a Reciprocating Engine Based CHP System for Commercial Office Building,” ISHPC-080-2005, International Sorption Heat Pump Conference, Denver, CO, USA.

Patnaik, P. 2004, “Experimental Verification of an Absorption Chiller for BCHP Applications,” *ASHRAE Transactions*, Vol. 110, Part 1.

Wagner, T.C., Marler, M.E., and Jung, S.H. 2005, “Performance Characteristics of a Microturbine-Double Effect Absorption Chiller CHP System,” Proceedings IMECE 2004: 2004 International Mechanical Engineering Congress and RD&D Expo, Anaheim, California, USA.