2001

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Published in Gee, T. A., Cao, J., & Mathias, J. A., & Christiensen, R. N. (2001). Experimental Testing and modeling of a Dual-Fired LiBr-H_2O Absorption Chiller. ASHRAE Transactions, 107(2), 3-11. ©2001 ASHRAE (http://www.ashrae.org/)

Recommended Citation

Gee, Tom A., Cao, Jiming, Mathias, James A. and Christensen, Richard N. "Experimental Testing and modeling of a Dual-Fired LiBr-H2O Absorption Chiller." (Jan 2001).

Experimental Testing and Modeling of a Dual-Fired LiBr-H_20 Absorption Chiller

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ABSTRACT

An LiBr-H_20 chiller was modified to utilize heat sources from natural gas combustion and/or from hot fluid. This was achieved by replacing the original gas-fired generator with a dual-fired generator. Steam was used as the hot fluid. The generator of the chiller can be powered by each source separately or both sources simultaneously. Experimental investigation was performed to obtain capacity and coefficient of performance (COP) of the original chiller and the modified chiller. During the experiments, the modified chiller was powered solely by steam, natural gas, or both. There was a significant increase in COP and capacity when steam was used as a heat source. The tests using natural gas resulted in performance similar to the original chiller. The experimental conditions were closely modeled by a numerical program.

INTRODUCTION

Interest is growing for cooling systems that can be powered by more than one type of energy source. Absorption cooling systems require heat as the primary energy input and can be powered by more than one heat source. In addition, absorption cooling systems have lower operating costs when they utilize exhaust heat or solar energy. The ability of absorption cooling systems to utilize more than one kind of energy input would be the most advantageous in regions of the world where electricity is expensive or inaccessible. An absorption chiller would also be a viable option wherever heat is readily available. The heat source to an absorption chiller can be a high-temperature liquid or gas. Such a fluid can be obtained from factory waste heat, solar energy, or combustion.

Cooling from an absorption cycle has many advantages over a vapor compression cycle. An absorption cycle is advantageous where the cost of electricity is high or where there is an abundance of energy in the form of heat. The working fluid of an absorption system is environmentally benign, while many of those used in vapor compression cycles are harmful to the environment.

Solar energy could be used to provide the necessary heat input to an absorption cycle. Hammad and Audi (1992) suggested that cooling is needed the most where solar energy is the most available. Some experiments have produced absorption chillers powered by solar energy that are cost-effective compared to other methods of cooling.

The concept of employing more than one heat source to power an absorption cycle is not new. Grossman et al. (1981) mentioned the idea of using solar energy as a primary heat source and using an alternate heat source when the solar power was insufficient. They discussed the ideas of using solar power, a conventional heat source, and using both concurrently. Their suggestions were to either use a solution preheater or an auxiliary generator in order to increase cycle efficiency.

Best and Ortega (1998) suggested that the main technical problem of cooling systems that are powered by solar energy is that the system is so dependent upon the environment. Factors such as the temperature of the cooling water, the temperature of the ambient air, solar radiation, wind speed, etc., would all affect the system's performance. According to their review, solar-powered cooling was also not economical at that time due to the initial cost of the system.

Critoph (1999) noted that the most expensive part of an absorption chiller system that uses solar energy is the collector array. A secondary or supplementary source of heat would reduce the amount of collector area necessary to power the

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system, and this would result in a lower initial cost of the system. Such a solution could take the form of using a backup hot water boiler or a system that would be able to take both the fluid from a solar array and fossil fuel combustion products as the heat input. These dual-fired cycles would be the most economical to build and operate. These systems would also be more reliable in meeting cooling loads than a purely solar-fired system.

Henkel (2000) predicts a considerable decrease in manufacturing costs of solar arrays. He predicts that due to development of the technology and mass manufacturing, the costs of a solar array will be reduced by more than a factor of two. Solar-powered cooling will be made economically feasible by the projected decrease in manufacturing costs. An absorption chiller in California (Duff et al. 1995) that is powered by solar energy has operated for two years. This installation at a 740 m^2 (8000 ft2) commercial office building features a 100 m^2 (1100 ft2) collector array driving a 70 kW (20 RT) double-effect chiller that was modified from gas-fired to hot water-powered. A separate hot water heater that uses natural gas provides backup energy when there is insufficient solar energy available. This backup hot water heater increases the initial cost and maintenance of the absorption chiller. Christensen (2000) presents an economic feasibility study that reports an 18.8% improvement in life-cycle costs of a dual-fired system over a system with a solar-powered chiller with a backup boiler.

Chen and Hihara (1999) suggested a hybrid system that was powered by both solar and electrical power. The system they proposed was a heat driven absorption cycle in parallel with a vapor compressor. The result of their theoretical model was that the system was more efficient than the traditional cooling cycles. This redundant system obtains the benefits of using solar power while providing cooling at a constant rate.

Berry (1988) used the second law of thermodynamics to determine operating parameters and to optimize the cycle performance of a double-effect absorption cycle. Specifically, given the remaining cycle temperatures, the minimum operating temperature of the first stage of a double-effect cycle was determined, which resulted in the maximum cycle COP. He also discussed how solar-powered systems would be attractive in less developed nations and suggested that using solar power, being a renewable energy source, would be a viable option for food refrigeration, cooling, and heating.

Absorption cycles can be modeled with an absorption simulation program (Grossman et al. 1991). Herold et al. (1996) used this type of program to simulate a number of different cycles with a variety of different working fluids. Among the cycles simulated were single-effect and double-effect absorption chillers and heat pumps. These cycles used $LiBr\text{-}H_20$, $H_20\text{-}NH_3$, and $NaOH\text{-}H_20$.

This paper discusses the experimental investigation and theoretical modeling of the performance of an absorption chiller that was modified to accommodate heat input by two heat sources, natural gas and/or solar energy or low-grade heat. The objective was to test a dual-fired system that delivered comparable performance to a commercial chiller that is only fired by natural gas. The performance of the original and dual-fired system were evaluated in terms of the cooling capacity and COP. Theoretical modeling was conducted to simulate the experimental results and would facilitate the design of similar absorption cycles.

EXPERIMENTAL SETUP

The absorption chiller that was modified and tested was a double-effect system that used $LiBr\text{-}H_20$ for the working fluid. The original chiller has a natural gas-fired generator. The generator is a device wherein vapor is generated from a binary mixture. In this case, water vapor is the refrigerant generated from $LiBr\text{-}H_20$ solution.

Prior to the experimental test, the chiller was connected to chilled water, cooling water, and natural gas. The chilled water supply that cools the building was directed through the absorption chiller. The building chiller and the absorption chiller were both used to provide cooling for the building. Cooling water was supplied using city water. Additional city water was constantly added to the cooling water to maintain a constant temperature, near the design condition of 29.4°C (85.0°F), of the cooling water at the inlet of the chiller. Excess cooling water was drained from the cooling water loop to maintain the desired amount of water in the coolant loop. Natural gas flowed through a gas meter and then to the burner of the chiller.

After the original chiller was tested, it was modified into a dual-fired system. The chiller was modified to accommodate a new set of generators. The first change to the chiller was that the original high-temperature generator was removed to make room to install the new generator set. The new generator set was two dual-fired generator units that were installed in parallel. The two dual-fired generators employ a counterflow arrangement between the high temperature fluid and $LiBr\text{-}H_20$ solution. A co-current arrangement is used between the combustion gases and the $LiBr\text{-}H_20$ solution. Figure I is a drawing of a cross-sectional, top view of the dual-fired generator (Cao and Christensen 1999). See Cao and Christensen (1999) for a more detailed description of the dual-fired generator configuration and design.

The dual-fired generator is similar to a shell-and-tube heat exchanger with modifications. The shell is an annular space that is formed from two concentric cylinders. Inside the annular space of the shell are a number of fluted tubes. Fluted tubes are used because of their enhanced heat transfer (Srinivasan and Christensen 1992). The $LiBr\text{-}H_20$ solution flows in the annular region around the tubes, being directed by a series of baffle plates. The high-temperature fluid coming from the low-grade heat source flows vertically through the fluted tubes. The natural gas burner is located at the bottom and the center of the inner cylinder. A ceramic, cylindrical insert is placed inside the inner cylinder to increase the mass flux of the combustion gas and, thus, increase heat transfer. Toward the

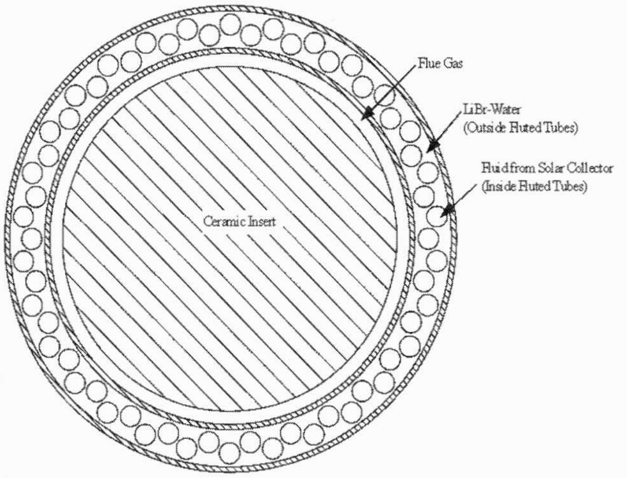

Figure 1 *Cross-sectional, top view of dual-fired generator.*

top of the generator, heat transfer from the combustion gas to the LiBr-H_2O solution is enhanced with a set of straight fins. Straight fins are placed only near the top of the generator in order to prevent structural damage to the fins. Structural damage may occur to the straight fins if the temperature of the combustion gas is too high when it comes in contact with the fins.

When the experiments were performed with the chiller that had been modified, steam was used to simulate a high-temperature fluid from a solar array. This is considered a reasonable substitute for high-temperature fluid from a solar array. The heat transfer coefficient of condensation of steam has been calculated to be approximately equal to the heat transfer coefficient of high-temperature water flowing through the fluted tubes at the designed flow rate. The temperature of the steam was adjusted so that its temperature would be near that of fluid from a solar collector. Steam traps were installed at the exit of each generator to ensure that the steam was condensed at the exit.

Temperatures of the chilled water, cooling water, generator inlet and exit, steam inlet and exit, and natural gas were measured by thermocouples. The thermocouples were placed in the fluid near the center of the pipe. The chilled water temperatures are critical parameters in evaluating cooling capacity and COP. Additional resistance temperature detectors (RTDs) were utilized to measure the temperature of the chilled water. Figure 2 shows a system schematic and the locations where temperature measurements were taken are marked with a "T" in the figure.

Each of the RTDs and thermocouples was calibrated at three different temperatures. They were calibrated by measuring the temperature of boiling water, cold water with a large amount of ice, and water at room temperature. The thermometers used during calibration conformed to ISO 9000 standards. From the calibration, it was determined that the temperature probes were accurate within ±0.3°C (±0.5°F).

Flow rates for the cooling and chilled water were determined by measuring the pressure difference between the inlet and the exit of the chiller of both the cooling and chilled water. The chiller manufacturer's correlation was then used to calculate the flow rate that corresponded to the pressure drop measured. The flow rate of LiBr-H_2O solution was measured by a magnetic flowmeter that measured the velocity of the fluid. This flowmeter was calibrated by flowing water through the flowmeter and recording the reading from the flowmeter, the mass of the water, and the elapsed time. The mass flow rate for LiBr-H_2O solution was obtained by multiplying by the specific gravity of solution. The steam flow rate was calculated by weighing the condensate taken during a given time.

The mass flow rate of LiBr-H_2O solution and the temperatures measured by the thermocouples were processed by a data acquisition system and then recorded by a computer. The data were taken after the chiller had been operating for more than three hours, and it was observed that the temperatures of the various fluids were stable. When the experimental data were taken, temperature measurements were recorded every four seconds.

An uncertainty analysis was performed for the calculations of the cooling capacity and COP for the experiment performed that was dual-fired (powered by steam and natural gas) at full-load conditions. The uncertainty of the cooling capacity depends on the accuracy of the temperatures and the mass flow rate of the chilled water. The total uncertainty in the reported cooling capacity is ±11 kW (3.13 RT). This uncertainty represents 6% of the cooling capacity of the experimental data of the chiller that was dual-fired. The uncertainty of the COP depends on the uncertainties of the cooling capacity and

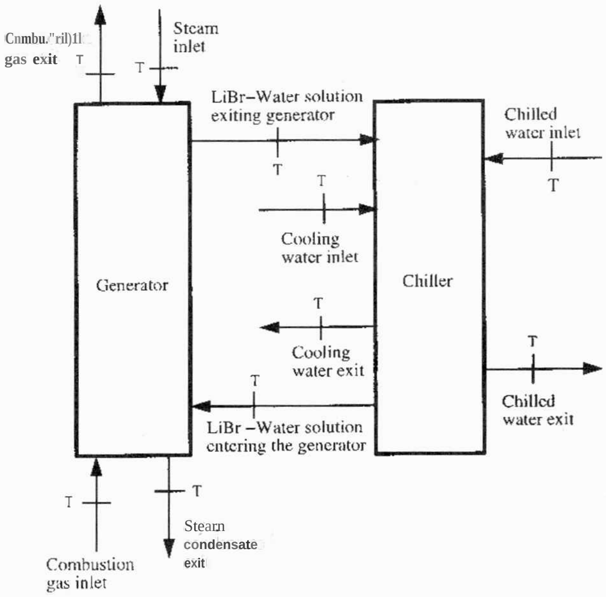

Figure 2 System diagram with temperature probe locations.

the heat input. The uncertainty of the COP was determined to be ±0.06, which is also approximately 6% of the COP value.

EXPERIMENTAL RESULTS AND DISCUSSION

Experimental Results of the Original Chiller

The original chiller was tested at full cooling capacity. The chiller had a rated capacity of 176 kW (50 RT) and COP of 1.0 at design conditions. The test was performed near all the design values. Two of the critical design values are 7.0°C (44.6°F) for the temperature of the chilled water exiting the chiller and 29.4°C (85.0°F) for the temperature of the cooling water entering the chiller. All the design values for temperature, flow rate, and pressure are given in the manufacturer's installation manual (Yazaki 1996).

The heat input to the original chiller was the combustion of natural gas. Electrical power is consumed to run pumps and controls but this power usage is negligible in comparison to the heat input. When calculating the COP of an absorption cooling system, the energy input used is the heat input to the generator. The equation for COP calculation is shown in Equation 1.

$$COP_{cooling} = \frac{\dot{Q}_{evap}}{\dot{Q}_r} = \frac{\dot{m}_{c} c_p (T_{c,i} - T_{c,e})}{\dot{V}_g Q_{HHV}} \qquad (1)$$

The higher heating value of the natural gas used was 38 MJ/m^3 (1020 Btu/ft^3) as approximated from periodic measurements done by the local natural gas utility.

The measured data for the original chiller, shown in Table 1, are used as a baseline condition. The data shown are average values of the measurements taken during the test period. To ensure steady-state operation, the average values are calculated over a 20-minute time period where the total variance of the chilled water temperature at the inlet of the chiller was less than 0.6°C (1.0°F). In general, all the temperatures and flow rates were very constant before and during the period when the experimental data

TABLE 1
Experimental Results of the Original Chiller, of the Modified Chiller at Full and Partial Load with Steam Input, at Full Load Using Natural Gas, and at Full Load with Dual Heat Input

Cycle Performance	Units	Original Chiller Full Load	Modified Chiller Steam Input Full Load	Modified Chiller Steam Input Partial Load	Modified Chiller Natural Gas Full Load	Modified Chiller Dual Heat Input Full Load
Capacity	kW (RT)	159.0 (45.2)	195.9 (55.7)	141.8 (40.3)	168.8 (48.0)	179.5 (51.1)
COP		0.94	1.30	1.26	0.96	1.10
Flow Rates						
Chilled water	m^3/min (gpm)	0.471 (124.4)	0.494 (130.5)	0.484 (128.0)	0.484 (128.0)	0.484 (128.0)
Cooling water	m^3/min (gpm)	0.708 (187.1)	0.684 (180.7)	0.695 (183.5)	0.695 (183.5)	0.700 (184.9)
LiBr-$H_2$0 solution into generator	kg/min (lbm/min)		51.9 (114.4)	47.9 (105.6)	50.4 (111.6)	51.6 (113.2)
Steam	kg/min (lbm/min)		4.3 (9.54)	3.16 (6.97)		2.24 (4.93)
Natural gas	m^3/min (cfm)	0.267 (9.4)			0.280 (9.9)	0.134 (4.75)
Temperatures						
Chilled water inlet	°C (°F)	12.4 (54.4)	13.0 (55.3)	14.4 (57.9)	12.2 (53.9)	13.9 (57.1)
Chilled water exit	°C (°F)	7.5 (45.5)	7.3(45.1)	10.2 (50.3)	7.2 (44.9)	8.6 (47.5)
Cooling water inlet	°C (°F)	28.2 (82.8)	26.7 (80.0)	26.4 (79.6)	28.3 (83.0)	26.6 (79.9)
Cooling water exit	°C (°F)	34.0 (93.1)	34.3 (93.7)	31.9 (89.5)	34.7 (94.4)	33.4 (92.1)
LiBr-$H_2$0 solution into generator	°C (°F)		130.0 (266.0)	116.7 (242.1)	125.7 (258.3)	133.5 (272.3)
LiBr-$H_2$0 solution out of generator	°C (°F)		148.7 (299.7)	133.9 (272.9)	143.7 (290.7)	142.4 (288.4)
Steam inlet	°C (°F)		158.2 (316.7)	142.2 (287.9)		147.9 (298.2)
Natural gas	°C (°F)				22.4 (72.4)	24.4 (75.9)
Combustion Analysis						
Excess air	%				9.85	

were taken. The performance of the original chiller was close to the manufacturer's specifications; the capacity was approximately 10% below the rated value, while the COP was approximately 5% below the rated value.

Experimental Results of the Modified Chiller

The modified chiller was tested at full and partial cooling capacities by using steam and natural gas as heat inputs. Experimental results were obtained by using only steam as the source of heat input, using only natural gas as the source of heat input, and by using steam and natural gas simultaneously as heat inputs. The performance of the modified chiller was compared to the original system.

Steam-Fired Mode. The COP for the experimental tests using steam as the heat input is calculated by Equation 2.

$$COP_{cooling} = \frac{\dot{Q}_{evap}}{\dot{Q}_s} = \frac{\dot{m}_c c_p(T_{c,i} - T_{c,e})}{\dot{m}_s(h_{s,i} - h_{s,e})} \qquad (2)$$

The temperature and pressure ofthe steam were measured to determine the enthalpy at the inlet. The steam was a saturated vapor at the generator inlet. The steam trap at the exit of the generator ensured that the steam exiting the generator was a saturated liquid. In an actual system, the high-temperature fluid from the solar array or the low-grade heat source is circulated in a closed, insulated loop and is reheated after leaving the generator. Therefore, in the calculation of COP, the rate of heat input from the steam is taken as the enthalpy difference of the incoming steam and outgoing condensate.

Table I shows a summary of the average data ofthe modified chiller at full and partial cooling capacity using steam as the heat input. Data for each run were taken over a ten-minute period after steady-state conditions were established.

The performance of the original chiller operating at full-load conditions was compared to the performance of the modified chiller that was powered by steam. It was found that the modified chiller powered by steam had a greater cooling capacity and COP than the original chiller. The modified chiller powered by steam had an increased cooling capacity of 36.9 kW (10.5 RT). This is a significant increase (23%) in capacity compared to the capacity of the original chiller.

There was also a noticeable increase in the COP of the modified chiller that was powered by steam. The COP of the modified chiller increased 38% compared to the COP of the original chiller. The increase of the COP was largely because the energy of the high-temperature fluid leaving the chiller will not be lost but instead will be recirculated while the fluid is reheated. This assumption is valid for a high-temperature fluid (condensate in this experiment) that is returning to a solar array or other energy source. Sample calculations show that if it is assumed that none ofthe energy in the steam that exits the generator is reused and the COP is calculated by the cooling load divided by the energy input, the COP of the modified chiller powered by steam would be comparable to the COP of the original chiller.

To compare two different experimental runs, the operating conditions should be similar. It was difficult to maintain the same chilled and cooling water temperatures as the experimental run of the original chiller. With one exception, the chilled and cooling water temperatures were within 3°C (5.4°F) of the temperatures ofthe experimental run of the original chiller. The chilled water temperature for the experimental run of the modified chiller at full load with steam input is nearly the same as the chilled water temperature of the experimental run of the original chiller. The chilled water temperature at the exit of the chiller for the experimental run of the modified chiller with steam input at partial load is 2.7°C (4.9°F) higher than the same temperature of the experimental run of the original chiller. This warmer chilled water temperature may make it easier for the modified chiller to perform better than the original chiller. The temperature of the cooling water into the chiller was approximately 3.0°C lower for the experimental run of the modified chiller at full and partial load using steam input compared to the experimental run of the original chiller. This colder cooling water temperature may account for part of the increase in capacity and COP of the modified chiller with steam input.

It is noteworthy that the modified chiller was powered with steam at a temperature of only 158.2°C (316.7°F). This temperature can be obtained from a solar array or from waste heat from industries. The smallest temperature difference between the steam and the LiBr-H_2O solution exiting the generator is 9.5°C (17.0°F). This temperature difference indicates that the generator is properly sized for this application.

The temperature of the chilled water was held approximately constant during all the experiments at full-load conditions performed with heat input from steam, natural gas, or both steam and natural gas. The temperature of the chilled water was close to the standards of the Air Conditioning and Refrigeration Institute (ARI). ARI requires chillers to deliver water at a temperature of 7.2°C (45°F) to the indoor air-handler. The temperature of the chilled water exiting the chiller was very close to 7.2°C (45°F); the greatest deviation was for the experiment that was performed with the dual-fired input where the exit temperature ofthe chilled water was 8.6°C (47SF).

Natural Gas-Fired Mode. For the experiment performed with natural gas as the heat input, the COP was calculated the same as the experiments with the original chiller. Table I gives a summary of the data taken from the modified chiller at full load with heat input from natural gas combustion. CO and O_2 concentrations in the combustion products were measured where they exited the generator, and from these measurements, the percent excess air was calculated. The data were taken at steady-state conditions over a ten-minute time period.

From the data shown between the original chiller and the modified chiller that was powered by the combustion of natural gas, it is observed that the cooling capacity and COP are approximately equal. The chilled water and cooling water temperatures were also approximately equal between the two experimental runs. A combustion analysis was taken for the tests performed with the modified chiller using natural gas as the heat source. The percent of excess air for the modified chiller was 9.85%. Using a percent of excess air below this could cause a significant amount of carbon monoxide to increase in the combustion gases leaving the generator. The burner used in the experiment was a metal-fiber, mesh burner that produces a low amount of carbon monoxide. The carbon monoxide reading was 66.5 ppm for the experiments powered with natural gas performed at full cooling capacity. This value of carbon monoxide is acceptable for a device that uses natural gas at this low rate. An experiment was performed at part-load conditions and the burner was turned down to use less than one-half of the rate of natural gas compared to the full-load conditions. The burner operated properly at part-load conditions and can be operated at any firing rate between the firing rates corresponding to full- and part-load conditions and possibly lower than the firing rate at part-load conditions.

Dual-Fired Mode. The experimental tests performed with the modified chiller in the dual-fired mode were performed in the following way. The modified chiller had been operating in natural gas-fired mode at partial load. The steam valve was opened and the temperature of the steam was adjusted until the chiller operated near the rated capacity. The transition between the generator being powered solely by natural gas to a dual-fired mode went very smoothly. It is anticipated that the start-up procedure of the chiller in dual-fired mode occurs without difficulties and can begin with either or both sources of heat providing energy to the generator.

Table I summarizes the data taken from the dual-fired run that used both steam and the combustion of natural gas simultaneously as heat inputs. The COP for the experiments performed with heat input from both steam and natural gas is calculated by Equation 3.

$$COP_{cooling} = \frac{\dot{Q}_{evap}}{\dot{Q}_t} = \frac{\dot{m}_{cc} p C T_{c,i} - T_{c,e})}{V_g Q_{HHV} + I\sim s(h_{s,i} - h_{s,e})} \tag{3}$$

The data were taken when the chiller was operating at steady-state conditions. From the data shown between the original chiller and the modified chiller that was powered by both steam and natural gas, the cooling capacity and COP are greater for the modified chiller. There is a 13% increase in capacity and a 17% increase in COP compared to the baseline results of the original chiller. Sample calculations show that a large portion of the increase in the COP is due to the assumption that the energy in the steam leaving the generator will be reheated; if this assumption is not made, the modified chiller operating in dual-fired mode still has a COP that is comparable to the original chiller. It is recognized that the chilled water temperature was approximately 2.3°C (4.1°F) warmer and the cooling water temperature was approximately 2°C (3.6°F) colder for the experimental run with dual heat input. Both of these differences in temperature may help the modified chiller with dual heat inputs perform better than the original chiller.

The heat input from the steam for the dual-fired run accounted for 48.6% of the total heat input. This heat input was supplied by steam at a temperature of 147.9°C (298.2°F). This temperature can easily be produced by a solar array or from waste heat from industrial processes.

CYCLE MODELING

The absorption cooling cycle was modeled on a numerical solving, absorption simulation program. The program is used to predict the steady-state performance of absorption refrigeration cycles. Given the components and the inputs to the program, the program generates the necessary equations governing species equilibrium and mass, species, and energy conservation. The equations are solved iteratively. For the modeling of the modified chiller operating in dual-fired mode, there were a total of 47 equations (Gee 2000).

The user specifies the convergence criteria of the program. Convergence can be determined by the function residuals of all the equations or by the relative error between two successive iterations of the values of the variables. The function residuals are the error terms of the equations. The iteration procedure terminated when the normalized value of the functions' residuals was less than 1×10^{-5}. This convergence was met after 51 iterations when modeling the modified chiller operating in dual-fired mode.

In order to simplify the modeling process, the program makes the following assumptions.

- The system operates at a steady state.
- The only heat loss from the system is from the exiting combustion gases.
- The pump work is negligible.

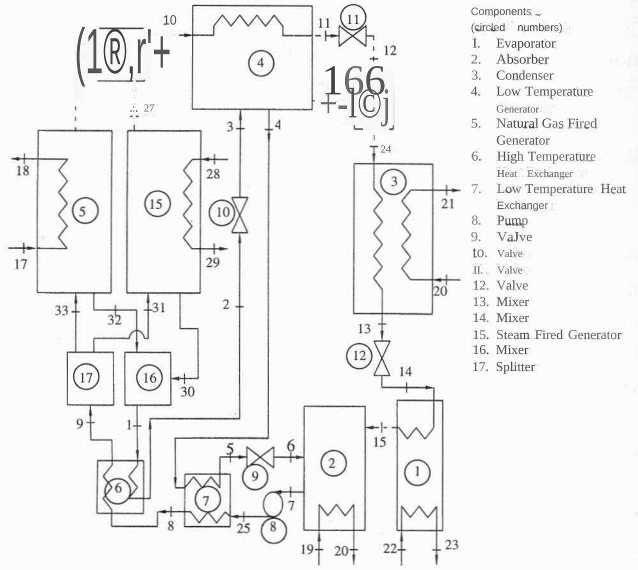

Figure 3 Schematic of absorption simulation program input file for dual-fired cycle.

- Solution in the generators and absorber exists at a saturated state.
- Pressure drop across heat exchangers is neglected.

The dual-fired generator is a unique design, and there is no standard module for modeling this component. In the experiment the dual-fired generator was one integral unit; however, during the modeling process, it was modeled as two generators in parallel. In the model, one generator was powered by steam and the other powered by natural gas. The generator that was powered by steam was a counterflow unit and the generator that was powered by natural gas was a co-current-flow unit. The solution flowing to the generators was split so that solution would flow into both generators. By enforcing conservation of mass, species, and energy, the absorption simulation program determined the mass flow rate ofLiBr-H$_2$0 solution into each generator. The temperature of the LiBr-H$_2$0 solution at the inlet and exit of each generator was set to be equal. Figure 3 shows a schematic of the absorption simulation model for the dual-fired cooling cycle.

The flow rates ofthe chilled water, LiBr-H$_2$0 solution at the generator inlet, and cooling water were held constant at the measured values. The temperatures at the inlet ofthe chiller of the chilled and cooling water were also held constant at the measured values. The inputs of the model, which were not measured in the experiment, were varied to match the measured state points and performance. Other parameters that were varied included the closest approach temperature (CAT) for each of the heat exchangers and the solution concentration. The highest priority was placed on matching the cooling capacity. Given that the flow rates of the chilled and cooling water were held constant in the model, the accuracy of the calculated cooling capacity and COP were linked together.

The pressures of the solution and refrigerant were not measured in the testing of the chiller. In modeling the cycle, state points were varied to yield system pressures that were close to the design values. Vapor fractions were not measured in the chiller testing. Some values were allowed to vary so that the absorption simulation program would determine the values by satisfying its energy, species, and mass balance equations. Key inputs to the model were varied so that the vapor fraction of the steam exiting the generator was assumed to be near zero. Heat transfer occurred while the steam passed through the fluted tubes and a steam trap only allowed condensate to leave the generator. Neglecting heat losses to the ambient, the heat of condensation must have completely entered into the LiBr-H$_2$0 solution.

The concentrations of LiBr were not measured in the system. These values were assumed to be near the design values. Concentration values slightly different from the design points are justified in that the temperature of the cooling water has an effect on the solution concentration. Specifically, cooling water at lower temperatures made for lower LiBr concentration. In testing the chiller, the temperature of the cooling water varied between 26.4°C and 28.3°C (79.6°F and 83°F).

TABLE 2
Comparison of Model Results to Experimental Results for Dual-Fired Mode

		Experimental	Model
Cycle Performance			
Capacity	kW (RT)	179.5 (51.0)	179.5 (51.0)
COP		1.10	1.10
Temperatures			
Cooling water inlet	°C tF)	26.6 (79.9)	26.6 (79.9)
Cooling water exit	°C (OF)	33.2 (92.1)	33.2 (91.8)
Chilled water inlet	°CtF)	13.9 (57./)	*13.9 (57./)*
Chilled water exit	°C (OF)	8.6 (47.5)	8.6 (47.4)
LiBr-H$_2$0 solution into generator	°C (OF)	133.5 (272.3)	130.9 (267.7)
LiBr-H$_2$0 solution out of generator	°C (OF)	142.4 (288.4)	142.4 (288.3)
Flow Rates			
Chilled water	*m^3/min (gpm)*	*0.48 (/28.0)*	*0.48 (/28.0)*
Cooling water	*m^3/min (gpm)*	*0.70 (/84.9)*	*0.70 (/84.9)*
LiBr-H$_2$0 solution into generator	*kg/min (lbllimin.)*	*5/.6 (/13:2)*	*5/.6 (/13:2)*
Steam	*kg/min (lb,/min.).*	2.2 (4.93)	2.2 (4.93)
Natural gas	*m^3/min (cftn)*	*0.13 (4.75)*	*0.13 (4.75)*

The accuracy of the model is considered by comparing values from the model with measured values. Table 2 shows the values from the model and the measured values. Only the state points that correspond to a measured value are shown. The values that were used as inputs in the model are in italics in Table 2.

The model fits the experimental data with good accuracy. The difference between the experimental values and the model values is less than 1% of the experimental data. The largest difference between the experimental and model values is the temperature of the LiBr-H_2O solution at the inlet of the generator. One possible reason for differences between the values is that the absorption simulation program assumes that a two-phase solution is constantly in thermal equilibrium; this assumption may not be completely valid. Heat losses to the ambient and pressure drop in the components, which the absorption simulation program does not account for, are other possible reasons for the differences between the experimental and model values.

When a model was produced that matched the experimental values of the modified chiller test with dual heat input, the overall heat transfer coefficient (VA) of each of the components of the chiller were obtained from the model. The model was then run with the same VAs of the components but with the inputs of the different experimental runs. The outputs of the model matched each of the outputs of the experimental runs very closely. The VA of the generator was changed when the model was used to match the experimental run of the original chiller. This provides confidence that the model could be used to predict performance of the modified chiller at off-design conditions.

Figure 4 shows the COP and cooling capacity as a function of the cooling water inlet temperature. This figure was generated from the experimental results and computer model of the modified chiller that was dual-fired. In an actual absorption system, the temperature of the cooling water is largely determined from the ambient air temperature; therefore, Figure 4 shows how the chiller will perform with changes in ambient temperature. The trend of decreasing COP and cooling capacity with increasing cooling water temperature is expected. The computer model predicts only a 3.7% decrease in COP and cooling capacity as the cooling water temperature rises 5.6°C (10.1°F) from 26.6°C (79.9°F) to 32.2°C (90.0°F).

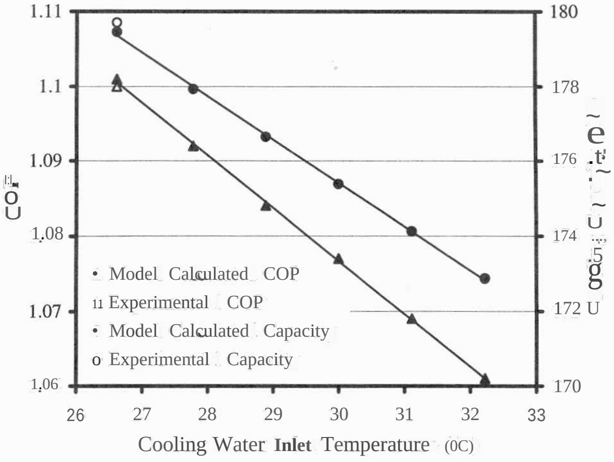

Figure 4 COP and cooling capacity as a function of the cooling water inlet temperature.

CONCLUSION

An LiBr-H_2O chiller was modified such that it would be able to utilize heat inputs from natural gas combustion and/or from a hot fluid. The original high-temperature generator was replaced with an experimental generator. The experimental generator utilized the two heat sources separately and simultaneously. The COP and capacity of the modified chiller in the experimental runs were equal to, or greater than, those of the baseline system. There was a significant increase in COP and capacity when steam was used. The tests using natural gas resulted in performance similar to the original chiller.

The cycle was successfully modeled using the absorption simulation program. The model values are close to the measured data taken from the test. The VAs obtained from the model were held constant, and the model matched all the different experimental runs. The model can be used as a guideline for predicting performance and state point values in case of modifications or operation at off-design conditions.

NOMENCLATURE

c_p = specific heat capacity ($J kg^{-1} K^{-1}$)
COP = coefficient of performance
h = enthalpy of steam ($J kg^{-1}$)
m = flow rate ($kg s^{-1}$)
$\dot{Q}$ = rate of heat transfer (kW)
$\dot{Q}_{evap}$ = rate of heat transfer to the refrigerant in the evaporator (kW)
Q_{HHV} = higher heating value of natural gas ($J m^{-3}$)
RT = refrigeration ton (1 RT = 3.52 kW)
T = temperature (°C)
VA = overall heat transfer coefficient
V = volumetric flow rate ($m^3 s^{-1}$)

Subscripts

c = chilled water
e = exit
g = natural gas
i = inlet
s = steam
t = total

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