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Investigation of the Part-Load Performance of an Absorption Chiller

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ABSTRACT

An experimental investigation designed to determine the part-load performance of an ammonia-water absorption water chiller is described. The steady-state and cyclic performance of the chiller were measured under controlled conditions in an environmental chamber. Two valves were installed in the chiller to separate high- and low-pressure regions during off times, and insulation was applied to the chiller components. By these measures, losses due to cyclic operation were reduced by over 50%, resulting in a 6% to 7% increase in the calculated seasonal performance factor for typical northern and southern climates in the United States. The use of the valves eliminated the need of the "spindown" period, thereby reducing the consumption of parasitic electrical energy.

INTRODUCTION

An absorption water-chiller, like other building heating/cooling equipment, is generally selected to meet the design (i.e., maximum) load. Under design conditions, the water-chiller operates continuously at full capacity. During much of the cooling season, however, the load is less than the design value, and the water-chiller must be cycled on and off to provide the desired indoor conditions. The seasonal performance of the water-chiller is thus dependent on its performance during part-load operation.

Some information is already available in the literature on the performance of absorption water-chillers under part-load conditions.[1,6] The data available, however, demonstrate that the performance of an absorption air conditioner can be significantly degraded by cyclic operation. Reduced performance occurs for several reasons. When an absorption unit is operated, regions of high pressure (in the condenser and generator), and low pressure (in the absorber and evaporator) are established. When the unit is turned off, the pressure difference causes a migration of the working fluids from the high- to the low-pressure regions. This mass transfer is aided by temperature differences that exist between the components. In addition, radiative and convective heat transfer occurs between the individual components and between the components and the environment. When the unit is turned back on, energy must be expended to reestablish the high- and low-pressure regions, as well as to make up for the heat losses that occurred during the off period.

Manufacturers of absorption water-chillers recognize the problem with part-load operation and employ two methods to maintain acceptable performance during part-load conditions. The

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first method involves the selection of an appropriate thermostat. From a comfort standpoint, it would be best to cycle the air conditioner as often as necessary to keep the indoor temperature within narrow limits. Controlling the unit in this manner would, however, result in high on-off cycle rates and significantly degraded performance, unless a quasi-steady-state behavior is established at a very high cycle rate as indicated in Ref. 1. As a result, manufacturers of absorption systems recommend using a thermostat designed to cycle the unit at a maximum of 1.5 cycles per hour. The maximum cycle rate for vapor compression cooling equipment is generally three cycles per hour.[2] Although the part-load performance is improved by controlling the cycle rate, there is an associated penalty in the degree of comfort experienced by the building occupants, since meeting the building cooling load at a lower cycle rate results in greater temperature fluctuations in the air-conditioned space.

Another way in which the part-load performance is improved is by the use of a time-delay control switch, which maintains the pumps and blowers in operation for several minutes after the thermostat cuts off the generator heat input. This period, referred to as the "spindown" period, allows the additional cooling capacity available within the unit and connecting piping to be supplied to the load; this additional capacity would be partially or totally lost without the spindown period, further degrading the part-load performance.

There are two disadvantages associated with the spindown period. First, the additional cooling capacity (which was not called for by the thermostat) causes wider indoor temperature fluctuations and, thus, decreases indoor comfort. Second, the pumps and blowers must operate for a longer period, thus consuming additional electrical energy.

This paper reports the results of an experimental investigation designed to determine the part-load performance of absorption machines, assuming the major factors contributing.to reduced performance during part-load operation are the migration of the working fluids during the off-period and thermal energy losses. Off-period fluid migration was reduced by installing two valves between the high- and low-pressure regions, and thermal energy losses were reduced by insulating the components of the unit. The cyclic performance of the absorption chiller was measured at several cycle rates and part-load operating conditions. The steady-state performance was also measured over a range of operating conditions.

The effect of the valves and insulation on the seasonal performance of the unit was estimated using a modified bin-temperature calculation procedure, which accounts for the reduced performance during part-load operation.

EXPERIMENTAL SETUP

An air-cooled ammonia-water absorption chiller was used in the experimental investigation. The chiller was rated by the manufacturer as capable of cooling 0.45 l/s (7.2 gpm) of water from 12.8°C (55°F) to 7.2°C (45°F) at 35°C (95°F) dry-bulb temperature; this corresponds to a nominal capacity of 10,550 W (36,000 Btu/hr). The unit is fired by natural gas and requires an energy input of 23,150 W (79,000 Btu/hr) to the combustor. It was designed for residential service, with all of the absorption system components included in one modular package. Also included is a pump for circulating chilled water through the evaporator. The electrical energy required to run the blower and the chilled water and solution pumps is 1050 W.

The chiller was installed in an environmental chamber at the National Bureau of Standards in which the operating conditions (i.e., the air and inlet water temperatures) could be controlled. Air was circulated within the chamber to ensure a uniform air temperature. During cyclic tests, the air temperature was maintained within $\pm$1°C ($\pm$ 1.8°F) of the set point, while fluctuations during steady-state tests were less than 0.3°C (0.5°F). The test facility was designed to allow a once-through passage of chilled water through the evaporator at a constant inlet temperature of 12.8°C (55°F).

A schematic diagram showing the major components of the absorption chiller appears in Fig. 1. Thermocouples were installed at the inlet and outlet of every component, which provided meaningful temperature measurements at these locations when the solution pump was in operation. The thermocouples were connected to a data acquisition system, which recorded the temperatures at these locations at one-minute intervals.

The capacity of the chiller was obtained by measuring the flow rate and temperature change of the water flowing through the evaporator. A turbine meter was installed in the inlet stream and calibrated in situ at the design flow rate. An electronic counter was connected to the turbine meter to allow recording of the instantaneous and average flow during a test. The chilled water flow rate and inlet temperature were maintained constant at 0.45 l/s and 12.8°C (55°F) in all tests. Thermocouples were installed in the chilled water inlet and outlet streams close to the unit and were connected to the data acquisition system. In addition, two thermopiles were installed between the two streams and connected to a strip-chart recorder to provide a continuous record of the temperature difference. The estimated precision of the capacity measurement is within 1%.

The volumetric flow rate, temperature, and pressure of the natural gas were measured in each test. The higher heating value of the natural gas was determined in a calorimeter in operation at the National Bureau of Standards. The energy consumption of the combustor heating the generator was determined with an estimated precision of 2%.

Pressure gauges were mounted on the generator and absorber of the unit to measure the high- and low-side pressures. These pressures were recorded at the beginning and end of the burner on-time and spindown periods. In addition, for some cyclic tests, the pressures were recorded at short time intervals during the entire cycle. Both pressure gauges were calibrated on a deadweight gauge-tester prior to the tests.

Before most of the tests were conducted, two remote-activated ball valves were installed. Their positions within the unit are indicated in Fig. 1. In order to install the valves, it was necessary to remove ammonia and to replace it again after the work was completed. The ammonia charge was adjusted to maximize the steady-state capacity at the standard rating point (35°C [95°F] outdoor air temperature, 12.7°C [55°F] chilled water return temperature). This capacity was within 1% of that obtained before the valves were installed.

After testing the influence of the valves, the following parts of the absorption chiller were insulated with glass wool: the solution-cooled absorber, the receiver located before the solution pump, and the rectifier and that part of the generator that is not in direct contact with the burner. In addition, the panel between the components listed above and the condensing unit was insulated. To prevent further heat losses from the generator by convection, the exhaust for the flue gases was closed during off-times. The effect of the insulation on the steady-state performance was checked, and no change in the capacity was detectable.

TESTING PROCEDURE AND EVALUATION

All tests were conducted with a chilled water flow rate of 0.45 l/s (7.2 gpm) and a chilled water inlet temperature of 12.8°C (55°F). Prior to each steady-state test, steady conditions were first established in the environmental chamber and then data were taken and averaged over a 30-minute period. The steady-state capacity, $\dot{Q}_{ss}$, was determined by the relationship

$$\dot{Q}_{ss} = \dot{m} C_p \Delta T_{ss} \tag{1}$$

where

$\dot{m}$ is the mass flow rate of chilled water, C_p is the specific heat of water, and ΔT_{ss} is the steady-state temperature difference between the inlet and outlet chilled water streams. $\dot{Q}_{ss}$ is the instantaneous capacity which is equal to the average capacity in steady-state operation

The steady-state coefficient of performance (COP_{ss}) is defined here as the ratio of the capacity to the sum of the rate of gas energy input to the combustors and the steady-state electrical power input to the blower and pumps, $\dot{E}_{ss}$.

$$COP_{ss} = \frac{\dot{Q}_{ss}}{\dot{E}_{ss}} \tag{2}$$

This definition of COP weights the rate of gas energy input and power consumption equally. Since other equally appropriate definitions of COP exist, the experimental results in Tab. 1 contain the ratio of the electrical power to the rate of gas energy inputs.

For the cyclic tests, steady conditions were first established in the environmental chamber. After a warm-up period, the unit was cycled on and off for the amount of time appropriate for each test for three cycles. The averaged data from the last two cycles were used in the following calculation.

The total cooling during a cycle, Q_{cyc}, was determined by

$$Q_{cyc} = \dot{m}C_p \int_{t_1}^{t_2} \Delta T \, dt \tag{3}$$

where

ΔT is the instantaneous temperature difference between the inlet and outlet chilled water streams and t_1 and t_2 are the times that the chilled water pump was turned on and off, respectively.

The pump operating time coincided with the burner on-time when the spindown period was disabled. Otherwise, the pump operating time was about 4.5 minutes longer than the burner on-time. (In the following discussion, the term "on-time" always means burner on-time, although the capacity was evaluated for the entire period in which the chilled water circulation pump was in operation.) The total gas and electrical energy input to the unit over the interval t_1 to t_2, E_{cyc}, was measured and used to calculate the coefficient of performance for cyclic operation.

$$COP_{cyc} = \frac{Q_{cyc}}{E_{cyc}} \tag{4}$$

The cyclic test data are presented in Tab. 1 in terms of a cooling-load factor and a part-load factor, similar to the factors used to describe the cyclic performance of vapor compression machines.[5] The cooling load factor, CLF, is defined

$$CLF = \frac{Q_{cyc}}{\dot{Q}_{ss} \; T_{cyc}} \tag{5}$$

where

Q_{cyc} is the integral cyclic capacity over one cycle, $\dot{Q}_{ss}$ is the steady-state capacity rate at the same air temperature as for the cyclic test, and T_{cyc} is the cycle period which is the sume of the furner on- and off-times.

Defined in another (equivalent) way, CLF is the ratio of the cooling that is supplied at a particular outdoor air temperature to the steady-state capacity of the machine at that temperature. It is a dimensionless measure of the degree of part-load operation. Values near unity indicate that the machine must operate nearly continuously to meet the load; values near zero occur when the machine is off for most of the time.

The part-load factor, PLF, is defined

$$PLF = \frac{COP_{cyc}}{COP_{ss}} \tag{6}$$

The part-load factor is less than or equal to unity; it is a dimensionless measure of the performance penalty for cyclic operation.

RESULTS AND DISCUSSION

Steady-State Performance

Steady-state tests were conducted at air temperatures of 21.7°C (71°F), 26.7°C (80°F), 35.0°C (95°F), and 38.0°C (100.4°F). The test results appear in lines 1, 2, 3, and 4 of Tab. 1. A plot of the measured capacity versus air temperature is shown in Fig. 2. The capacity is strongly dependent on the air temperature, especially for temperatures higher than about 30°C (86°F). Increasing temperatures decreased the capacity significantly.

Part-Load Performance in the Original Operating Mode

Cycle rates were chosen for most of the tests according to the thermostat characteristics supplied by the manufacturer. (Thermostat cycle rates are not constant but rather a parabolic function of burner on-time.[3] The maximum recommended cycle rate for this absorption chiller is about 1.5 cycles per hour (CPH), which occurs at 50% on-time. At 20% and 80% on-time, the cycle rate with the recommended thermostat is 1.0 CPH.)

The tests listed in lines 5, 6, 7, 8, and 14 of Tab. 1 were conducted to measure the part-load performance of the absorption chiller in its original operating mode (i.e., without additional insulation, without the valves in operation, and with the spindown period enabled). The tests shown in lines 5 and 8 were conducted under the same conditions but at the beginning and in between the other tests, respectively, to check the reproducibility of the experimental data, which was found to be satisfactory. During a cooling season, an absorption chiller is operated at a variety of cycle rates and outdoor air temperatures. Tests were conducted to determine how changes in the outdoor temperature affect the part-load factor (PLF) at a given cycle rate. The results in lines 5, 6, and 7 of Tab. 1 show that both the part-load factor and the cooling load factor (CLF) decrease with increasing outdoor air temperature. The deviation in PLF between 21.6°C (71°F) and 35°C (95°F) is about 7%, while the change in CLF is about 9% In an installation in which the chiller is appropriately sized, part-load operation is more likely to occur at temperatures below the design condition. Therefore, all of the remaining cyclic tests were conducted at 26.6°C (80°F) as a representative condition.

The circles in Fig. 3 show the experimental values of PLF plotted versus CLF for the absorption chiller in its original operating mode over a range of part-load operating conditions. The size of the symbols in Fig. 3 is indicative of the uncertainty in the measured values. For comparison, the part-load performance of vapor compression systems (at a maximum cycle rate of 3 CPH), as assumed in Ref 4, is indicated by line a in Fig. 3. The performance of the absorption system is not considerably lower than that of vapor compression systems when it is operated at the recommended cycle rate, which is about one-half the cycle rate for vapor compression systems.

Part-Load Performance with the Values in Operation and without Insulation Applied

In this section, the part-load performance of the chiller in its original operating mode (lines 5, 6, 7, 8, and 14 of Tab. 1) is compared with its performance when the valves are closed during the burner off-time and open during the burner on-time (lines 10 and 16). The spindown period was disabled when the valves were operated. Tests were also conducted in which the valves were closed after the spindown period was completed. However, these tests resulted in slightly lower part-load factors than those with disabled spindown, and they required significantly more electrical energy. Apparently, the operation of the valves eliminates the need of the spindown period.

Comparing the temperature changes within the unit during the on-time, it was obvious that the average temperatures during the cycle are closer to their steady-state values when the valves are operated than when they are not. Further, when the valves were operated, there was no time delay for the temperature rise of the fluid leaving the solution-cooled absorber. This time-delay, which was 1.5 minutes in the original operating mode, indicates that the liquid absorbent solution traveled during the off-time from the generator to the solution-cooled absorber and needed to be pumped back to the generator. A similar time-delay was observed for the capacity in the original operating mode. Again, it took approximately 1.5 minutes after turning on the unit until it started to cool down the incoming water. This time-delay was not present when the valves were operated.

The valves also affect the behavior of the pressures as illustrated in Fig. 4. During the off-time, the high- and low-side pressures converge to the same pressure, and then they both drop with the same rate in the original operating mode. The low-side pressure remains unchanged during the off-time when the valves are operated, while the high-side pressure drops significantly in the beginning of the off-time but then stabilizes at a relatively high value. During the on-time, the high-side pressure achieves higher values (closer to the steady-state values) when the valves are operated, while the low-side pressure remains stable at its steady-state value. The peaks shown by the high- and low-side pressures after turning off the unit were observed in all cases in which spindown was disabled including those cases in which the valves were not operated. They are due to time-delays in heat and mass fluxes in generator and absorber, respectively. During cyclic operation at high outdoor air temperatures, these peaks could cause the pressure relief valve to open. The peaks could be significantly reduced by a short spindown period, e.g., thirty seconds.

This temperature and pressure behavior was qualitatively observed for all cycle rates. It indicates that, during the on-time, these variables return toward their steady-state values from higher initial values, resulting in an increased COP and PLF. The increase in performance is significant. For example, at 1.0 CPH and 20% on-time, the PLF increases by more than 13% (lines 5 and 10 of Tab. 1). In this case, the valve operation reduces the degradation in performance as a result of cyclic operation to almost half its original value.

Less electrical energy is used when the spindown period is disabled. The difference in electrical use increases as the on-time decreases. For 1.0 CPH and 20% on-time, the power consumption is reduced by 30%; this can be evaluated by comparison of the electricity to gas input ratio in Tab. 1 lines 8 and 10.

Part-Load Performance with Insulation and the Valves in Operation

The chiller was insulated as described earlier in an effort to further improve its part-load performance. The insulation produced no detectable change in the steady-state performance. With the insulation in place, the chiller was operated with the spindown period disabled and the valves in operation (Tests 11, 13, 17, 18, Tab. 1). The effect of the insulation could be seen by examining the temperatures at various points within the unit. With the insulation in place, the temperatures were generally higher when the unit was turned on. For example, the temperature of the weak solution leaving the generator was 75°C (167°F) at the beginning of the on-time in the case of the insulated unit and 47°C (117°F) when the unit was not insulated; the valves were operated in both cases. (For comparison, the steady-state value is 115°C [239°F]). The behavior of the pressures was, within experimental error, not affected by the insulation. Qualitatively, these effects were observed for all cycle rates.

Insulating the chiller components improved the part-load performance of the chiller. The triangles in Fig. 3 show values of PLF plotted versus CLF for the insulated unit with the valves in operation and the spindown period disabled (Tests 11, 13, Tab. 1). The square symbol in Fig. 3 represents a test that was conducted without insulation (line 16, Tab. 1). However, since the PLF for this test was 0.994, a further increase in PLF due to insulation is not expected; the test result indicated by the square in Fig. 3 thus represents both the uninsulated and insulated results with the valves in operation. The PLF at 20% on-time and 1.0 CPH is 15% higher than that for the unit in its original operating mode. The insulation itself achieves an increase in PLF of 3% compared to operation of the valves alone, which is a much smaller contribution than that attributed to the operation of the valves alone (Tests 10, 11, Tab. 1).

Part-Load Performance at High Cycle Rates

The maximum cycle rate recommended for the absorption chiller (1.5 CPH at 50% on-time) is about half of that commonly employed for vapor compression systems. Lower cycle rates result in larger temperature fluctuations in the air-conditioned space and, presumably, lower occupant comfort. To see if this disadvantage of the absorption chiller could be relaxed, the unit was tested with the insulation in place and with the valves in operation at 2.0 CPH, 20% on-time and 3.0 CPH, 50% on-time. The results are shown in lines 17 and 18 of Tab. 1. The performance is almost as good as for the lower cycle rates, which are displayed in lines 11 and 16. The maximum deviation in PLF is 2%. These results demonstrate that, with the insulation and valves, the absorption unit may be operated at cycle rates typical for vapor compression systems while still showing a considerably higher performance than vapor compression systems, as indicated by line a in Fig. 3.

In order to show the influence the cycle rate has on the absorption chiller performance in its original operating mode, the chiller was tested at 3.0 CPH, 50% on-time. The test results appear in line 19 of Tab. 1. The PLF is 7% lower at the higher cycle rate than at the recommended cycle rate (line 14).

Higher cycle rates with spindown enabled result in a larger percentage of electrical energy input. A 37% reduction in electrical energy consumption is achieved at 3.0 CPH, 50% on-time, by eliminating the spindown period (Tab. 1 lines 18 and 19).

Seasonal Performance

The performance data obtained in this investigation were used to calculate the seasonal performance factor, SPF, of the absorption chiller. The seasonal performance factor is defined as the ratio of the total cooling load supplied to the total fuel and electrical energy consumed by the chiller during the cooling season. In the results given below, the fuel and electrical energy were equally weighted. To fairly compare absorption chillers with vapor compression machines, however, the electrical energy consumption should be divided by the efficiency of its generation for both systems.

The calculations were done for residential applications according to the modified bin-temperature method given in Ref 5. The modified bin-temperature calculation procedure uses the information in Fig. 3 to estimate the part-load performance of the chiller in each temperature bin and thereby provides an estimate of the seasonal performance including the effects of cyclic operation. The calculations were done for a generalized northern and southern climate;[5] the results are shown in Tab. 2.

Column 1 in Tab. 2 shows the values of SPF that would be achieved if the unit were to operate under all circumstances with the steady-state COP for any given air temperature. This is an upper limit for the SPF. Column 2 in Tab. 2 displays the SPF obtained by the unit in its original operating mode, while column 3 shows the SPF achieved by the unit with insulation in place, the valves in operation, and the spindown period disabled. Compared with the original operating mode, the insulation and valves increase the SPF by 7.0% in the southern climate and by 6.4% in the northern climate. Expressed in another way, these figures indicate that the losses due to part-load operation can be reduced by approximately 50% by the insulation and valves.

CONCLUSIONS

The results show that migration of the working fluids during the off-time is a major factor contributing to the degradation of the absorption chiller performance during cyclic operation. This migration can be reduced by the installation of automatic valves that separate the low- and high-pressure regions during the off-periods. The use of the valves eliminates the need for the spindown period and results in a significant reduction of electrical energy consumption during cyclic operation. Insulating the chiller components in this way also increased the part-load performance but not to the same extent as the valves. A 6% to 7% increase in the calculated seasonal performance factor results from the use of the valves and insulation.

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TABLE 1
Cyclic Test Results

No.	Air Temp. °C(°F)	Cycle Rate CPH	Burner On/off Time [min]	Burner On Time percent	Elect/ Gas Input *10^2	Valves Applied	Insula-tion Applied	Spindown Enabled	PLF	CLF
1	21.7 (71)	Steady State Test			4.69	N.A.	no	N.A.	1.0	1.0
2	26.7 (80)	"	"	"	4.64	N.A.	no	N.A.	1.0	1.0
3	35.0 (95)	"	"	"	4.73	N.A.	no	N.A.	1.0	1.0
4	37.8 (100)	"	"	"	4.80	N.A.	no	N.A.	1.0	1.0
5	26.7 (80)	1	12/48	20	6.27	no	no	yes	0.771	0.153
6	21.7 (71)	1	12/48	20	6.10	no	no	yes	0.788	0.164
7	35.0 (95)	1	12/48	20	6.56	no	no	yes	0.735	0.149
8	26.7 (80)	1	12/48	20	6.26	no	no	yes	0.769	0.152
9	26.7 (80)	1	12/48	20	6.21	yes	no	yes	0.870	0.171
10	26.7 (80)	1	12/48	20	4.78	yes	no	no	0.882	0.172
11	26.7 (80)	1	12/48	20	4.86	yes	yes	no	0.918	0.180
12	26.7 (80)	0.6	10/90	10	6.52	no	yes	yes	0.679	0.068
13	26.7 (80)	0.6	10/90	10	4.75	yes	yes	no	0.783	0.076
14	26.7 (80)	1.5	20/20	50	5.68	no	no	yes	0.912	0.451
15	26.7 (80)	1.5	20/20	50	5.67	yes	no	yes	0.979	0.489
16	26.7 (80)	1.5	20/20	50	4.72	yes	no	no	0.994	0.491
17	26.7 (80)	2	6/24	20	4.87	yes	yes	no	0.898	0.175
18	26.7 (80)	3	10/10	50	4.86	yes	yes	no	0.996	0.483
19	26.7 (80)	3	10/10	50	6.65	no	no	yes	0.849	0.429

TABLE 2
Calculated Seasonal Performance Results

	Maximum Achievable SPF	Absorption Chiller in Original Operation Mode	Modified Absorption Chiller
Southern Climate	0.434	0.387	0.413
Northern Climate	0.436	0.386	0.405

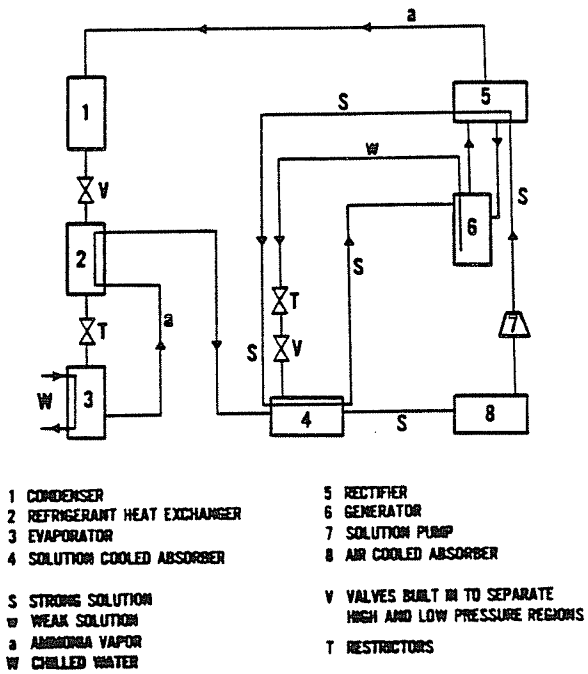

Figure 1. Scheme of the absorption water chiller under investigation

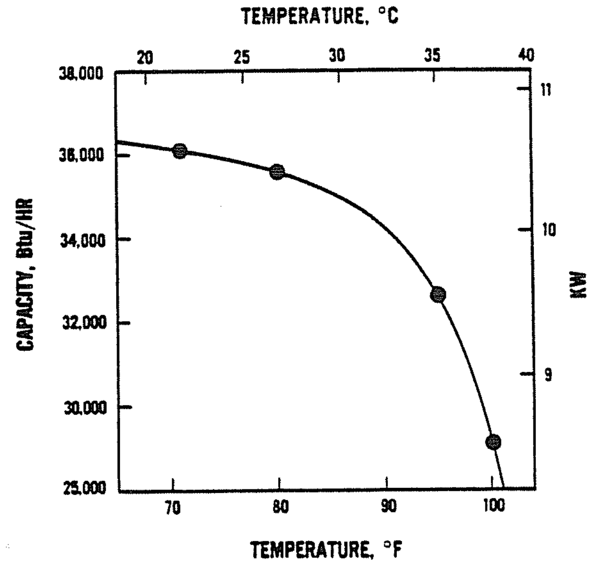

Figure 2. Steady-state capacity of the absorption water chiller versus outdoor air temperature

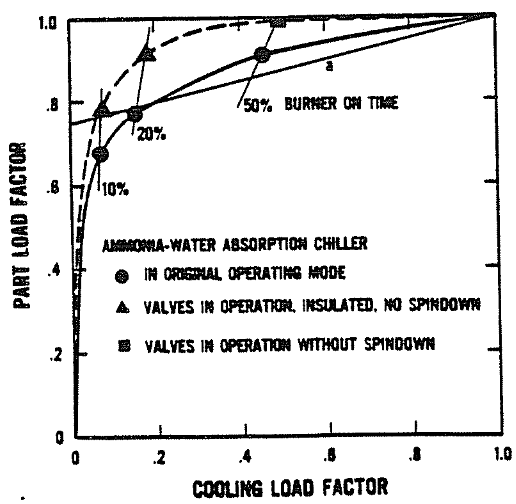

Figure 3. Part load factor versus cooling-load factor for different cycle rates and operating modes of the absorption chiller

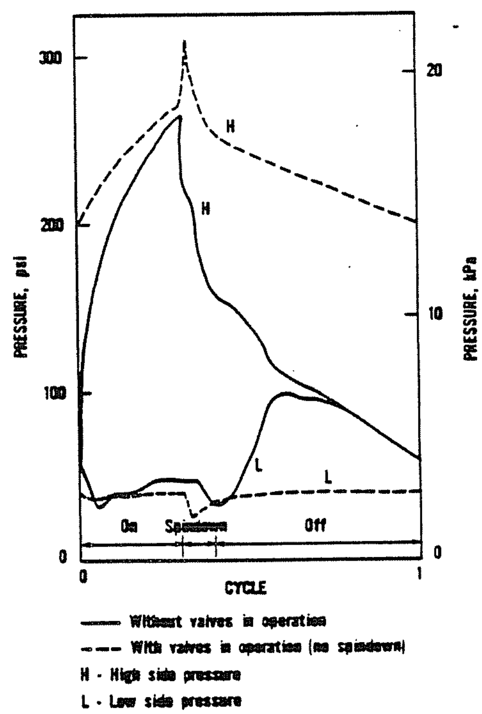

Figure 4. Comparison of typical behavior of high- and low-side pressures during a cycle with and without valves in operation; spindown was enabled in the latter case