

ENERGY AND EXERGY EVALUATION OF ICE PRODUCTION BY ABSORPTION REFRIGERATION

L. A. B. Cortez, D. L. Larson, A. da Silva

ABSTRACT. *This article presents an energetic and exergetic analysis of absorption refrigeration and the results from tests of ice production by absorption refrigeration using steam as the principal energy input at the UNICAMP hospital. The analysis found the UNICAMP absorption refrigeration system COP to be 0.36 and the exergetic efficiency to be about 10.4%, both similar to reported values, but less than desired. Exergy analysis further identified the generator, strong solution pre-heater and absorber subsystems for additional performance improvement evaluation due to the high irreversibilities of these components.* **Keywords.** *Absorption refrigeration, Refrigeration, Exergy, Energy analysis.*

Several technologies are available to generate low temperatures for the different cooling applications existing in the various climatic conditions around the world. Vapor compression refrigeration has become the dominant cooling technology wherever electricity is conveniently available. However, with increasing electricity costs and awareness of the need to conserve available energy resources, other technologies for cold generation may become more viable. Absorption refrigeration, powered by low temperature thermal energy, may be an attractive cooling alternative where low cost heat is available.

Absorption refrigeration often is described as inefficient and costly compared to mechanical vapor compression refrigeration. This description usually is based on relative coefficients of performance (COP), defined as the output refrigeration effect divided by the input electrical energy, and the availability of inexpensive electricity. However, COP is not an appropriate parameter for comparison of technologies powered by different energy inputs and electrical energy is unavailable or costly in many places. Exergy analysis is the only consistent method for comparing energy use effectiveness and value since it effectively accounts for energy capability.

A commercial absorption refrigeration unit was installed and operated at the Sao Paulo State University at Campinas (UNICAMP) Hospital to test and demonstrate the technology. The aquammonia system, donated by MADEP S.A., has a nominal refrigeration capacity of 23.25 kW for ice production. The heat source is low pressure steam supplied by the hospital boilers. The system was

instrumented and connected to a computer data acquisition system to record temperature, pressure, and other operating parameters. This article reports results of UNICAMP absorption refrigeration system performance tests and energetic and exergetic analysis of absorption refrigeration.

ABSORPTION REFRIGERATION BACKGROUND

Absorption refrigeration dates at least to developments by Carré in the 1850s. From 1859 to 1862, fourteen patents were registered for absorption refrigeration using aquammonia as the fluid pair (Stephan, 1983). Beginning in the 1880s, vapor compression refrigeration development gradually reduced absorption refrigeration use and only high electrical energy costs or limited energy availability, such as during World War I, elicited much interest in absorption refrigeration.

Absorption refrigeration developments between 1920 and 1940 included two-stage systems and the use of different solid and liquid refrigerant pairs (Stephan, 1983). Large absorption systems were installed in a number of European applications. Widespread household use resulted from introduction of the Electrolux absorption refrigerator, based on Platen and Munters patents, in Sweden around 1930. These refrigerators were commercialized until the 1950s, when advances in compression refrigeration outdated the Electrolux systems. In the USA in the 1940s, water-lithium chloride, and later, water-lithium bromide, absorption units were developed for air-conditioning by such companies as Carrier and Servel (Bjurström and Raldow, 1981).

Recently, interest in use of waste heat, from cogeneration and heat recovery projects, has revitalized large scale absorption refrigeration use while remote and portable refrigeration demands have stimulated small unit use. Today, companies manufacturing absorption refrigeration equipment include Trane, Carrier, and York (Guimarães, 1993).

Thermodynamic analysis is important to evaluation of such refrigeration system parameters as alternative refrigerants, cycle differences, and the effect of operating conditions. Exergetic evaluation was applied to analysis of absorption refrigeration by Altenkirch and Nierbergall at the beginning of the century (Stephan, 1983). It was shown

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that the coefficient of performance does not reveal the thermodynamic essence of the process, does not reveal all exergy losses and their causes. Exergetic analysis was shown to consider not only the amounts of energy transferred, but also the thermal potential, the capacity to produce work, of each energy stream.

A general method for calculating exergy flowrates and exergy variations and irreversibilities for energy processes was presented by Szargut et al. (1988). Kotas (1985) reported in detail the irreversibilities associated with such processes as separation, mixture, heat exchange, and expansion. Jordan (1985) described application of exergy evaluation to an absorption refrigeration system. However, second law analysis is not always included in thermodynamic analysis. For example, the ASHRAE Handbook of Fundamentals (1993) presents an exergetic evaluation for the compression refrigeration cycle, but not for absorption refrigeration.

Representative aquammonia properties were essential to the analysis of the UNICAMP absorption refrigeration system. The thermodynamic properties of mixtures can be calculated from pure substance data and mixture laws or state equations. Pure water and ammonia properties have been presented by Keenan and Keys (1936) and Haar and Gallagher (1978), respectively. Alternatively, mixture properties can be approximated via polynomial equations determined from test data. In 1938, Jennings and Shannon presented a review of published data for saturated aquammonia properties, specifically for absorption cycles, data later compiled in tabular form by Scatchard et al. (1947). In 1964 a new review was prepared for IGT which

combined published data with IGT measurements (Macriss et al.). More recently, El-Sayed and Tribus (1985) compiled a review of information on thermodynamic properties of aquammonia mixtures.

Jordan (1992) developed a computer program ("AQUAM") for calculating aquammonia properties, including entropies and exergies from values of pressure and concentration, or temperature and concentration, for saturated states. For superheated steam and sub-cooled liquid, temperature, pressure and concentration values are needed. The AQUAM program, based on the Ziegler and Trepp (1984) and Reynolds (1980) reports, was employed in the UNICAMP evaluation because it is easy to use and compatible with either SI or English units.

UNICAMP ABSORPTION SYSTEM

The absorption refrigeration system installed at the UNICAMP hospital uses aquammonia as the working fluid pair and a small portion of the process steam generated by the hospital oil-fueled boilers as the heat source. The equipment working at high pressure (generator pressure) are at the top, the equipment operating at the evaporating pressure at the bottom of the absorption refrigeration cycle diagram shown in figure 1. In the diagram, three operating temperature ranges can be identified: at the right, the generator high temperature (T_G); in the center, the water cooling temperature (T_O), and at the left, the evaporation temperature (T_E).

In the **generator**, steam heating causes the ammonia to be separated at high pressure from the strong solution

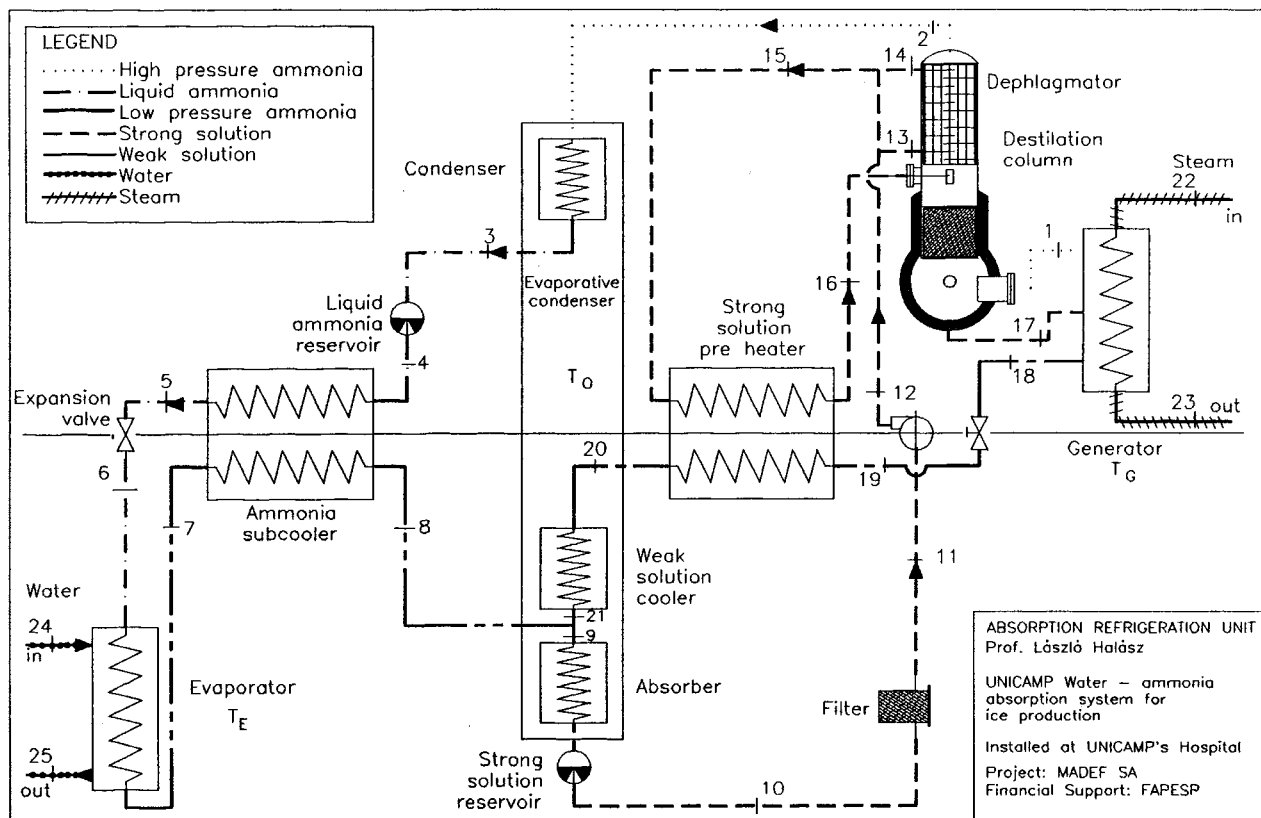


Figure 1—Absorption refrigeration system flow schematic diagram.

which enters the generator at (17). The high pressure ammonia vapor passes from the top of the generator (1) to the **distillation column**, where it mixes with the strong solution coming down in counter-current flow yielding an increase in ammonia concentration. The vapor rises through the column to the top section where it is cooled by the strong solution. As the vapor cools, there is partial vapor condensation (largely water) which increases the ammonia concentration leaving at (2).

The ammonia vapor then flows to the **evaporative condenser** where it is cooled and condensed and sent to the **liquid ammonia reservoir** (3). The liquid ammonia passes through the **ammonia subcooler** where it transfers heat to the ammonia vapor coming from the evaporator. The ammonia continues (5) to the **expansion valve** where its pressure is reduced to evaporation level (6) with a consequent temperature drop. As ammonia evaporates in the **evaporator**, the water in the outer tubes cools, producing ice.

The ammonia vapor leaving the evaporator (7) flows to the subcooler where it becomes superheated at low pressure. The vapor then passes to the **absorber** entrance (8), where it joins the weak solution coming from (9) the **generator**. The solution from the generator (18), passes through an expansion valve, decreasing its pressure to the absorption pressure, and flows (19) through the **strong solution pre-heater** where it transfers heat to the strong solution. The solution then passes (20) to the **weak solution cooler** where it is cooled to the ideal absorption temperature at the exit (21).

In the **absorber**, the combined weak solution is cooled and concentrated, becoming the strong solution stored in the **strong solution reservoir**. This solution passes through a set of filters (10) and then through the solution pump (11) which increases solute pressure to generator level (12). The solution is pre-heated when it passes through the distillation column (13) and the strong solution preheater (15). The solution from the preheater (16) finally enters the distillation column and flows against the vapor coming from the generator (17), where it is separated again, thus completing its cycle.

The freeze cycle was controlled by a timer with the freeze duration set at approximately 9 min. When the timer “opens” the circuit, the solenoid valve opens and ammonia vapor passes from the high pressure portion of the circuit into the evaporator, displacing liquid ammonia to the top of the evaporator coils. The thaw cycle duration was about 1 min. Hot gas condensation inside the tubes frees the formed ice from the inner and outer sections of the condenser. Then the valve positions are inverted again, initiating a new freezing cycle. A pressure control valve

located at the top of the distillation column maintains constant pressure during the defrosting cycle (A pressure drop jeopardizes separation and rectification process stability). The principal characteristics of the system are summarized in table 1.

THERMODYNAMIC ANALYSIS

Absorption refrigeration was considered to be a steady state process. Other principal assumptions of the thermodynamic analysis of the UNICAMP system (da Silva, 1994) were:

- Process fluids considered were: strong and weak solutions, steam, condensate, water, ice, and ammonia.
- Heat exchange to the environment, except at the evaporative condenser, was negligible.
- Pressure drop in tubes, valves and other fittings was negligible.
- The cooling water temperature at the condenser, the absorber and weak solution cooler was T_o and the variation in exergy due to heat transfer was negligible.
- There was isoenthalpic expansion through the ammonia valve.

From these assumptions and boundary conditions, the general mass, energy, exergy and irreversibility equations can be written as follows:

Continuity equation

$$\sum \dot{m}_i = \sum \dot{m}_e \quad (1)$$

Energy conservation

$$\dot{Q}_{c.v.} + \sum (\dot{m}_i \times h_i) = \sum (\dot{m}_e \times h_e) + \dot{W}_{c.v.} \quad (2)$$

Exergy variation

$$\Delta \dot{Ex}_{c.v.} = \sum (\dot{m}_i \times ex_i) - \sum (\dot{m}_e \times ex_e) + \dot{W}_{c.v.} \quad (3)$$

Irreversibility

$$\dot{I} = \dot{W}_{c.v.} - \Delta \dot{Ex}_{c.v.} - \Delta \dot{Ex}_{c.v.}^Q \quad (4)$$

Specific exergy

Table 1. Characteristics of the UNICAMP absorption refrigeration test unit

<i>Refrigeration capacity:</i> 23.25 kW	<i>Evaporating temperature:</i> -10°C
<i>Thermal energy needed:</i> 46.5 kW at 110°C	<i>Electrical power:</i> 220 V — 3 phase
<i>Absorber, condenser and weak solution cooler:</i> coil type cooled by evaporative condenser	
<i>Generator:</i> shell and tube type with ammonia solution between the tubes	
<i>Distillation column:</i> rectification with strong solution in a shell and tube heat exchanger	
<i>Strong solution pre-heater:</i> concentric tube type	
<i>Ammonia subcooler:</i> shell and tube type	
<i>Ice generator:</i> flood type evaporator with 5 annular tubes	
<i>Solution pump:</i> 0.75 kW	<i>Condenser water pump:</i> 0.37 kW
<i>Condenser ventilator:</i> 0.75 kW	<i>Ice generator water pump:</i> 0.37 kW
<i>Ice production capacity:</i> 20 kg of ice/cycle	<i>Freezing cycle duration:</i> 10 min
<i>Ice thickness:</i> 5 to 6 mm; Ice breaker: 0.75 kW	<i>Hot gas defrosting period:</i> 1 min

$$ex_1 = ex_o + (h_1 - T_o \times s_1) - (h_o - T_o \times s_o) \quad (5)$$

These equations are applied to system components as follows:

EVAPORATOR

During tests, the solenoid valve was closed, interrupting refrigerant to flow to the subcooler, so refrigerant entering equaled that leaving the evaporator. The ammonia concentration was considered to be 100% pure. With inlets at points 6 and 24 and outlets at points 7 and 25, the evaporator relationships are:

Mass balances

$$\dot{m}_7 = \dot{m}_6 \quad (6)$$

$$\dot{m}_{24} = \dot{m}_{25} \quad (7)$$

Energy balance

$$\dot{Q}_E = \dot{H}_{24} - \dot{H}_{25} = \dot{H}_7 - \dot{H}_6 \quad (8)$$

Exergy variation

$$\Delta \dot{E}x_{ice} = \dot{E}x_{25} - \dot{E}x_{24} \quad (9)$$

Irreversibility

$$\dot{I} = \dot{E}x_{25} + \dot{E}x_7 - \dot{E}x_6 - \dot{E}x_{24} \quad (10)$$

AMMONIA EXPANSION VALVE

Ammonia expansion valve inlet and outlet points are 5 and 6; valve relationships are:

Mass balance

$$\dot{m}_5 = \dot{m}_6 \quad (11)$$

Energy balance

$$\dot{H}_6 - \dot{H}_5 = 0 \quad (12)$$

Irreversibility

$$\dot{I} = -\Delta \dot{E}x_{valv.exp.} = \dot{E}x_5 - \dot{E}x_6 \quad (13)$$

SUBCOOLER

Subcooler inlet points are 4 and 7; the outlet points 5 and 8. Considering only irreversibility due to heat transfer, the subcooler equations are:

Mass balances

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_7 = \dot{m}_8 \quad (14)$$

Energy balance

$$\dot{Q}_{SUB} = \dot{H}_4 - \dot{H}_5 = \dot{H}_8 - \dot{H}_7 \quad (15)$$

Irreversibility

$$\dot{I} = -\Delta \dot{E}x_{SUB} = \dot{E}x_5 + \dot{E}x_8 - \dot{E}x_4 - \dot{E}x_7 \quad (16)$$

ABSORBER

The absorber inlet points are 8 and 21; the outlet point is 10. Absorber relationships are:

Mass balances

$$\dot{m}_{10} = \dot{m}_8 + \dot{m}_{21} \quad (17)$$

$$\dot{m}_{10} \xi_{10} = \dot{m}_8 \xi_8 + \dot{m}_{21} \xi_{21} \quad (18)$$

Energy balance

$$\dot{Q}_{ABS} = \dot{H}_8 + \dot{H}_{21} - \dot{H}_{10} \quad (19)$$

Irreversibility

$$\dot{I} = -\Delta \dot{E}x_{ABS} = \dot{E}x_{10} - \dot{E}x_8 - \dot{E}x_{21} \quad (20)$$

SOLUTION PUMP

The solution pump inlet and outlet points are 11 and 12. The relationships follow, where the enthalpy difference was calculated considering work to be minimum and the pump and electric motor efficiencies equal to 0.7 and 0.6, respectively.

Mass balance

$$\dot{m}_{11} = \dot{m}_{12} \quad (21)$$

Energy balance

$$\Delta h = \frac{v_{11} \times [\Delta P_{(11-12)}]}{\eta_{pump}} \quad (22)$$

$$\dot{W} = \frac{\dot{m} \times \Delta h}{\eta_{motor}} \quad (23)$$

Exergy variation

$$\Delta \dot{E}x_{SP} = \dot{E}x_{12} - \dot{E}x_{11} \quad (24)$$

Irreversibility

$$\dot{I} = \dot{W} - \dot{E}x_{11} + \dot{E}x_{12} \quad (25)$$

CONDENSER

The condenser equations are as follows, where the inlet and outlet points are 2 and 3 :

Mass balance

$$\dot{m}_3 = \dot{m}_2 \quad (26)$$

Energy balance

$$\dot{Q}_c = \dot{H}_3 - \dot{H}_2 \quad (27)$$

Irreversibility

$$\dot{I} = -\Delta\dot{E}x_c = \dot{E}x_2 - \dot{E}x_3 \quad (28)$$

WEAK SOLUTION COOLER

The weak solution cooler equations follow, where the inlet and outlet points are 20 and 21, respectively:

Mass balance

$$\dot{m}_{21} = \dot{m}_{20} \quad (29)$$

Energy balance

$$\dot{Q}_{WSC} = \dot{H}_{20} - \dot{H}_{21} \quad (30)$$

Irreversibility

$$\dot{I} = -\Delta\dot{E}x_{WSC} = \dot{E}x_{20} - \dot{E}x_{21} \quad (31)$$

STRONG SOLUTION PRE-HEATER AND SOLUTION EXPANSION VALVE

The equations for the subsystem containing the strong solution pre-heater and subsequent expansion valve are as follows, where the inlet points are 15 and 18 and the outlet points are 16 and 20:

Mass balances

$$\dot{m}_{15} = \dot{m}_{16} \quad (32)$$

$$\dot{m}_{18} = \dot{m}_{20} \quad (33)$$

Energy balances

$$\dot{Q}_{SSR} = \dot{H}_{16} - \dot{H}_{15} = \dot{H}_{18} - \dot{H}_{20} \quad (34)$$

Irreversibility

$$\dot{I} = -\Delta\dot{E}x_{SSR} = \dot{E}x_{15} + \dot{E}x_{18} - \dot{E}x_{16} - \dot{E}x_{20} \quad (35)$$

GENERATOR + DISTILLATION COLUMN

The generator and distillation column were considered to be one subsystem since measurements were not made at points (1), (14) or (17). Subsystem inlet points are 12, 16, and 22; outlet points are 2, 15, 18, and 23.

Mass balances

$$\dot{m}_{22} = \dot{m}_{23} \quad (36)$$

$$\dot{m}_{12} = \dot{m}_{15} \quad (37)$$

$$\dot{m}_{16} = \dot{m}_{18} + \dot{m}_2 \quad (38)$$

$$\dot{m}_{12}\xi_{12} = \dot{m}_2\xi_2 + \dot{m}_{18}\xi_{18} \quad (39)$$

Energy balance

$$\dot{Q}_G = \dot{H}_{22} - \dot{H}_{23} \quad (40)$$

Irreversibility

$$\dot{I} = -\Delta\dot{E}x_{G+DC} = \dot{E}x_{12} + \dot{E}x_{22} + \dot{E}x_{16} - \dot{E}x_{15} - \dot{E}x_2 - \dot{E}x_{23} - \dot{E}x_{18} \quad (41)$$

ABSORPTION CYCLE EFFICIENCY

The theoretical absorption refrigeration cycle can be defined from a Carnot cycle with specified water cooling, evaporation, and vapor generation temperatures. The restrictions presented by Martins (1993) for the Carnot cycle are: (1) adiabatic expansion of refrigerant between the condenser and the evaporator, and also of weak solution between the generator and the absorber; (2) pure ammonia vapor leaves the generator; and (3) refrigerant and absorbent fluids form an ideal mixture.

From an energy balance and application of the second law, the Carnot coefficient of performance (COP) may be calculated as:

$$COP_{ideal} = \frac{T_E \times (T_G - T_O)}{(T_O - T_E) \times T_G} \quad (42)$$

where

T_E = evaporator temperature

T_G = generator temperature

T_O = environment temperature to which heat is rejected

For a real cycle, the COP may be given by:

$$\text{COP}_{\text{real}} = \frac{\dot{Q}_E}{\dot{Q}_G} \leq \text{COP}_{\text{ideal}} \quad (43)$$

where

$$\begin{aligned} \dot{Q}_E &= \text{evaporator heat flow (kJ/h)} \\ \dot{Q}_G &= \text{generator heat flow} \end{aligned}$$

The equality applies to a reversible process, where the COP is maximum. The relationship between the effective COP of a system and the maximum COP can also be a measurement of cycle efficiency.

The second law thermodynamic efficiency can be computed using the equation presented by Szargut et al. (1988), where heat transferred to the working fluid at the generator is positive, as:

$$\psi_Q = - \frac{Q_E}{Q_G} \times \frac{(T_E - T_O)}{T_E} \times \frac{T_G}{(T_G - T_O)} \quad (44)$$

The above equation presumes exergy transfer in the generator and evaporator occurs under constant T_G , T_E , and T_O . For a real process, the heat transfer is computed from temperature difference, head loss, etc. Thus, it is possible to calculate the second law efficiency directly from the exergy flowrates of the refrigerated medium (e.g., ice) and the heating vector (e.g., steam). Kotas (1985) defines exergetic efficiency (ψ) using the following relationship between output or useful exergy and the input exergy:

$$\psi = \frac{\Delta \dot{Ex}_{\text{ice}}}{\Delta \dot{Ex}_{\text{vapor}}} \quad (45)$$

Exergy efficiency also can be calculated relative to the work done by the solution pump and other auxiliaries (e.g., ventilator and evaporative condenser water pump (CE) and ice generating water pump). Rewriting the previous equation to include these terms yields:

$$\psi = \frac{\Delta \dot{Ex}_{\text{ice}}}{\Delta \dot{Ex}_{\text{vapor}} + \dot{W}_{\text{pump}} + \dot{W}_{\text{auxiliaries}}} \quad (46)$$

Szargut et al. (1988) also presented an analysis of the influence of the heat transfer irreversibility on overall exergetic efficiency. For an absorption system which operates under an ideal cycle, the irreversibility is shown to be a consequence of only the finite temperature difference between the heat source and the working fluid. Exergetic efficiency then is computed as:

$$\psi_{\text{ideal}} = \frac{(T_G - T_O - 2 \times \Delta T) \times (T_E - \Delta T) \times (T_O - T_E) \times T_G}{(T_O - T_E + 2 \times \Delta T) \times (T_G - \Delta T) \times (T_G - T_O) \times T_E} \quad (47)$$

This equation yields the maximum exergetic efficiency for an absorption refrigeration cycle in comparison with the other presented equations.

The achievable COP for a 10 kW water-lithium bromide absorption chiller may be 0.47 and for a direct fired system may be 0.89, according to Carvalho (1990). For aquammonia single stage absorption refrigeration systems, the COPs tend to be lower (typically 0.4 to 0.5) due to greater irreversibilities in the separation and rectifying stages.

COP values of 1.7 for single stage compression refrigeration systems to 3.4 for two-stage systems were presented by ASHRAE (1981). Auracher (1979) reported a calculated COP value of 3 and exergetic efficiency of 0.44 for a single mechanical compression refrigeration system having an evaporating temperature of -25°C (temperature of the cooling medium of -2°C) and temperature of surrounding air of 17°C . The largest irreversibilities were computed for the compressor (23%), expansion valve (14%), and condenser (11%).

Szargut et al. (1988) presented an exergetic evaluation of an absorption refrigeration system using aqua-ammonia with a capacity of 140 kW and evaporating temperature of -33°C . The heating vector was 158°C steam. The COP was 39.5% and the exergetic efficiency, including the mechanical work of auxiliary equipment, 8%. The greater irreversibilities were found at the generator (19.5%), absorber (16%), evaporator (13.9%) and condenser (8.4%).

Jordan (1985), in an evaluation of a theoretical absorption refrigeration system with evaporating temperature of -4°C (brine) and generation temperature of 55°C (vapor) computed an exergetic efficiency of 17%. The larger irreversibilities were in the generator (21.6%), solution heat exchanger (22.9%) and the absorber (13.3%).

EXPERIMENTAL PROCEDURE

The UNICAMP absorption refrigeration system was tested to determine its energetic performance, particularly the relative energy and exergy losses of its components. A test began with 2 h of operation to obtain steady-state conditions, signified by a constant rate of ice production (da Silva, 1994). Operational data were collected during the following 90 min of continued steady-state operation. The pressure transducer then was moved to read the condensation pressure and operational data were collected for another 30 min with unchanged system settings. The results obtained in the evaluation tests were averaged and tabulated for each data acquisition point; data then were used in computer analysis of system thermodynamic performance. Additional testing is being conducted to evaluate performance under different environmental conditions.

DATA ACQUISITION

Data sensor input ports were installed in the absorption refrigeration system to facilitate installation of measurement devices for temperature, pressure and concentration. An orifice flow meter was installed in the strong solution circuit and two liquid reservoirs permitted measurement of the ammonia flowrate by closing the valve at (4). Measurement data were digitized by an A/D data converter and input to a 386 microcomputer for data processing.

Temperature data were obtained using T-type thermocouples with mineral isolation and steel protection to avoid ammonia corrosion. The thermocouples were calibrated using an ice bath and by comparing the results with a standard calibrated thermometer having a precision of 0.1°C. The recorded pressure measurements were high pressure, measured at the distillation column exit (2), and low pressure, sensed at the subcooler exit (8). The pressure transducers, manufactured by SODMEX, supplied a linear signal of 4 to 20 mA for the 0 to 1700 kPa pressure range. Bourdon type manometers also were installed to provide reference pressure measurements. The steam pressure at regulating valve inlet and outlet ports was measured with Bourdon manometers having a precision of 19.6 kPa.

Solute sample was removed at the evaporator inlet (6) to determine the liquid ammonia concentration. The ammonia was allowed to evaporate from the sample, leaving mostly water, and the difference was used to estimate ammonia concentration. To determine ammonia concentrations in the strong and weak solutions, samples also were taken from points (12) and (21). The samples were preserved in an ice bath, density and temperature were measured, and concentration was estimated using an ammonia density – concentration ($\rho - \xi$) diagram.

The ammonia flowrate at condenser outlet (3) was determined by timing the filling of the liquid reservoir located in that pipeline. The strong solution flowrate was measured with an orifice flow meter installed at the solution pump outlet (12). The quantity of produced ice was determined by weighing the ice produced in each cycle and converting production rate to kilograms per hour (kg/h).

RESULTS

Due to equipment problems and measurement difficulties, test results reported here are considered to be preliminary (da Silva, 1994). Solution pump operation caused the greatest difficulty; the pump presented cavitation problems when suction pressure was less than 176.5 kPa and pump gaskets required early replacement. Ammonia flow rate measurement also was less than satisfactory, a limitation met by repeated measurements.

Ice production averaged 28.8 kg/cycle while system COP varied from 0.31 to 0.40 in the test of the UNICAMP absorption refrigeration system. The energy and exergy balances computed from test data are summarized in tables 2 and 3 and shown diagrammatically in figures 2 and 3, respectively. Thermal energy inputs to the generator

Table 2. Refrigeration system energy balance based on 2 h of steady-state operation

Energy Balance			
Equipment	Q(+) (kJ/h)	Q(-) (kJ/h)	W(+) (kJ/h)
Generator	208 581		
Condenser		87 215	
Valve VI		0	
Evaporator	75 249		
Absorber		123 232	
Solution pump			1240
Auxiliaries			2647
Weak solution cooler		38 229	
Total	283 830	248 675	3887

Table 3. Exergy and irreversibility of the refrigeration elements based on 2 h of steady-state operation

Equipment	Ex (+)		I		Ex (useful)	
	(kJ/h)	(%)	(kJ/h)	(%)	(kJ/h)	(%)
Generator	54 706	93.37	16 882	28.81		
Solution pump	1240	2.12	294	0.50		
Auxiliaries	2647	4.52	529	0.90		
Condenser			2437	4.16		
Evaporator			4259	7.27	6078	10.37
Absorber			7145	12.19		
Exp. valve			369	0.63		
Strong sol. pre-heater			13 181	22.50		
Subcooler			1181	2.02		
Weak sol. cooler			4120	7.03		
Losses			2118	3.61		
Total	58 594	100.00	52 516	89.63	6078	10.37

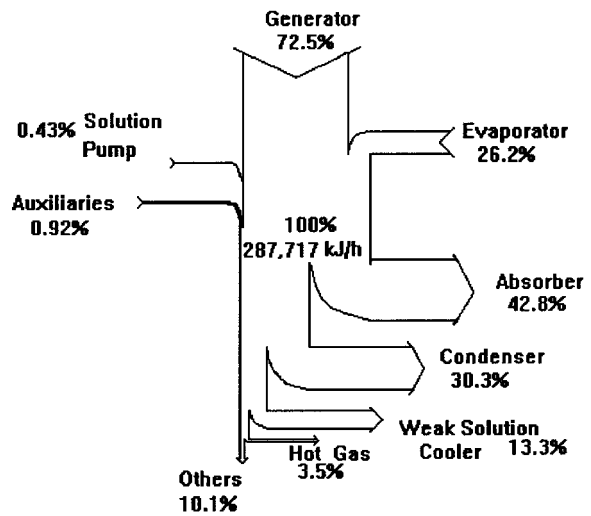


Figure 2–Absorption refrigeration system energy flow diagram.

(72.5%) and evaporator (26.2%) accounted for over 98% of the total energy input to the refrigeration process while most of the energy losses were from the absorber (42.8%), condenser (30.3%), and weak solution cooler (13.3%).

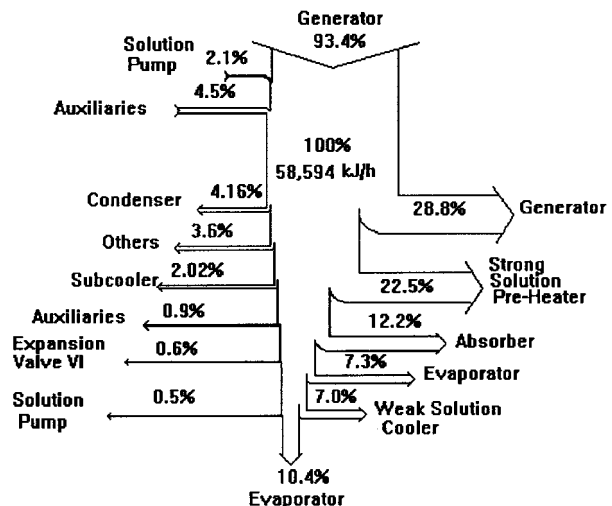


Figure 3–Absorption refrigeration system exergy flow diagram.

Table 4. Refrigeration system energetic and exergetic efficiencies

Absorption Refrigeration System Efficiency		
Energetic	Equation	Value
COP _{ideal} (Carnot)	$\frac{T_E \times (T_G - T_O)}{(T_O - T_E) \times T_G}$	2.014
COP _(gross)	$\frac{\dot{Q}_{ice}}{\dot{Q}_{vapor}}$	0.361
COP	$\frac{\dot{Q}_{ice}}{\dot{Q}_{vapor} + \dot{W}_{total}}$	0.354
Exergetic	Equation	Value
ψ_{ideal} (Szargut)	$\frac{(T_G - T_O - 2 \times \Delta T) \times (T_E - \Delta T) \times (T_O - T_E) \times T_G}{(T_O - T_E + 2 \times \Delta T) \times (T_G - \Delta T) \times (T_G - T_O) \times T_E}$	0.304
ψ_Q	$-\frac{Q_E}{Q_G} \times \frac{(T_E - T_O)}{T_E} \times \frac{T_G}{(T_G - T_O)}$	0.179
ψ_{gross} (Kotas)	$\frac{\Delta \dot{Ex}_{ice}}{\Delta \dot{Ex}_{vapor}}$	0.111
ψ (Kotas)	$\frac{\Delta \dot{Ex}_{ice}}{\Delta \dot{Ex}_{vapor} + \dot{W}_{pump} + \dot{W}_{auxiliaries}}$	0.104

The generator subsystem accounted for an even greater portion of the exergy input (93.4%), while the evaporator was the source of a lesser portion of the exergy loss (7.3% of the total), when compared to the energy analysis results summarized above. Exergy losses also were distributed among the various subsystems quite differently than energy losses. The generator (28.8%) and strong solution pre-heater (22.5%) accounted for the largest exergy losses, while their energy losses were minimal, and the absorber exergy loss was only 12.2% while its energy loss was a much larger 42.8%.

Direct electrical energy accounted for a very small portion of the total energy (1.35%) or exergy (6.6%) inputs. Including indirect energy associated with the electrical input would change the energy balance little, but would make the electrical exergy input a more substantial element.

Thermodynamic analysis results, summarized in table 4, show the system operated with a lower energy efficiency, represented by the system COP of 0.361, than expected for units operating with aquammonia (from 0.4 to 0.5). Two system deficiencies identified to be responsible for excessive energy “losses” were insufficient evaporator insulation and loss of water at the ice generator reservoir.

An exergetic efficiency computed for the UNICAMP system was 10.4% using the relationship between input and output exergy values (table 4). This value does not differ substantially from the values of 8%, reported by Szargut et al. (1988), and 16%, presented by Jordan (1985) (see comparison in table 5). Differences in steam and evaporation temperatures may account for most of the efficiency differences among the three systems. However, all

Table 5. Comparison of the exergetic efficiencies from three absorption refrigeration studies

Features	HC-Unicamp (1994)	Szargut (1988)	Jordan (1985)
Refrigerant pair	aquammonia	aquammonia	aquammonia
Cooling capacity (kW)	23	140	421
Heat source	steam (278 kPa)	steam (588 kPa)	steam (544 kPa)
Evaporation temperature (°C)	-10	-33	-22
COP/Carnot COP (%)	36	40	42
Exergetic efficiency (%)	10	8	16
Main sources of irreversibility	Generator (29%) Strong solution pre-heater (22%) Absorber (12%)	Generator (19%) Absorber (16%) Evaporator (14%)	Generator (25%) Strong solution pre-heater (27%) Absorber (16%)

three of these efficiencies are considerably lower than the ideal efficiency of 30% for a reversible system (table 4).

ENERGY COSTS

Economic benefit, rather than energy use effectiveness, probably is the principal factor motivating refrigeration system selection or use. Absorption refrigeration therefore is of greatest interest where electrical energy is unavailable or more costly than thermal energy. Sugar cane bagasse is available in large quantities at a very low cost in some Brazilian locations. For the ice production application described above (3 Mg ice/day @ 80% machine capacity), it is estimated that the annual energy costs using absorption refrigeration (\$4770/yr if COP = 0.4) using bagasse as fuel (bagasse energy content of 2.6 kWh/kg, 80% boiler efficiency, bagasse cost of \$8.50/Mg) could be one third the energy costs with compression refrigeration using electrical energy (\$1630/yr if COP = 2.0 and electricity costs \$0.06/kWh).

CONCLUSIONS AND RECOMMENDATIONS

UNICAMP absorption refrigeration system energetic and exergetic performance was comparable to that reported in the literature for similar systems.

Energy analysis provided a valuable performance insight not provided by energy analysis alone.

System performance was much lower than ideal. To improve performance, the generator, strong solution pre-heater and absorber were identified by high irreversibility rates as subsystems requiring more detailed evaluation.

Absorption refrigeration research will be continued, largely due to the abundant availability of low cost biomass energy resources.

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SYMBOLS

LATIN

CE	evaporative condenser
COP	coefficient of performance
\dot{E}_x	exergy flow (kJ/h)
H	enthalpy (kJ/h)
I	irreversibility rate (kJ/h)
P	pressure (MPa)
Q	heat transfer (kJ)
\dot{Q}	heat flow (kJ/h)
\dot{S}	entropy (kJ/h.K)
SS	strong solution
T	absolute temperature (K)
WS	weak solution
\dot{W}	power rate (kJ/h)
X	quality (%)
c_p	specific heat (kJ/kg K)
e_x	specific exergy (kJ/kg)
h	specific enthalpy (kJ/kg)
\dot{m}	mass flowrate (kg/h)
s	specific entropy (kJ/kg.K)
t	temperature (°C)
v	volumetric flowrate (m ³ /h)

GREEK

Δ	variation (final – initial)
ψ	exergetic efficiency
η	first law efficiency
ξ	concentration (%)
ρ	density (kg/m ³)
v	specific volume (m ³ /kg)
θ	diameter (mm)

SUBSCRIPTS

A	absorber
C	condenser
E	evaporator
G	generator
SP	solution pump
o	reference state (ambient)
e	exit
i	inlet
c.v.	control volume

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