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SIMULATION OF AN ABSORPTION CHILLER DRIVEN BY THE HEAT RECOVERY ON AN INTERNAL COMBUSTION ENGINE

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ABSTRACT

In the present work a previous cycle analysis (Lazzarin et al. 2000) of a water-lithium bromide absorption chiller, driven by the heat recovery on an i.c. engine, has been implemented by a specific simulation code of all the different devices of the absorption machine. The whole integrated refrigerating system includes a reciprocating engine, a vapour compression chiller and an absorption unit. The reciprocating engine drives the vapour compression chiller. The thermal energy recovered from the i.c. engine exhaust is used to drive a double effect water-lithium bromide cycle, while the heat recovered from the cooling jacket of the engine drives a single effect water-lithium bromide cycle. The two absorption cycles were integrated into a single unit with a common evaporator and absorber.

The system was analysed in the typical operative range of air conditioning chillers, evaluating the sensibility to the main boundary conditions.

INTRODUCTION

The absorption machines might represent an interesting alternative to traditional vapour compression units as they are free from ozone depleting fluids and they can be driven by thermal energy at relatively low temperature levels, such as the heat recovered from an industrial process or from a thermal engine. It is possible to find several applications of absorption chillers driven by the heat recovery on the exhaust from a gas turbine plant; however, the absorption unit generally was not specifically developed for this application, being only a modified standard machine. In recent literature, some works on the thermodynamic analysis of combined Diesel engine and absorption units (Mustafavi et al. 1997 and 1999) or combined Otto engine and absorption units (Nakamura et al. 1996) were presented. At Purdue 2000, the authors of present paper (Lazzarin et al. 2000) presented a cycle analysis of a water-lithium bromide absorption chiller driven by the heat recovery on an i.c. engine. In the present work the above model has been implemented by a specific simulation code of all the different devices of the absorption machine to evaluate the real performance and size of the system.

The system includes a reciprocating engine, a vapour compression chiller and an absorption unit. The reciprocating engine, based on an Otto cycle, produces around 1000 kW of mechanical power together with 600 kW of thermal power at low temperatures (70-80°C) from the cooling jacket of the cylinders and 600 kW of thermal power at medium temperatures (150-500°C) from the heat recovery on the exhaust. The mechanical power is used to drive the vapour compression chiller which supplies around 3000 kW of refrigerating capacity. The thermal energy recovered from the i.c. engine exhaust is used to drive a double effect water-lithium bromide cycle, while the thermal energy recovered from the cooling jacket of the engine is used to drive a single effect water-lithium bromide cycle. The two absorption cycles were integrated into a single unit with a common evaporator and absorber, which supply a refrigerating capacity of around 1200 kW. The absorption machine consists of three generators, two condensers, an evaporator, an absorber and two regenerative heat exchangers. Each device was simulated by a specific computer code which reproduces the internal heat and mass transfer processes considering the effective exchange area and the thermodynamic, thermophysical and transport properties of the operative solution. Particular attention was devoted to the simulation of the most critical component, the absorber, adopting the Nakoryakov and Grigor'eva 1977 model, complemented with the experimental data available in the open literature. The system was analysed in the typical operative range of air conditioning chillers evaluating the sensibility to the main boundary conditions: cooling water inlet temperature and refrigerated water outlet temperature.

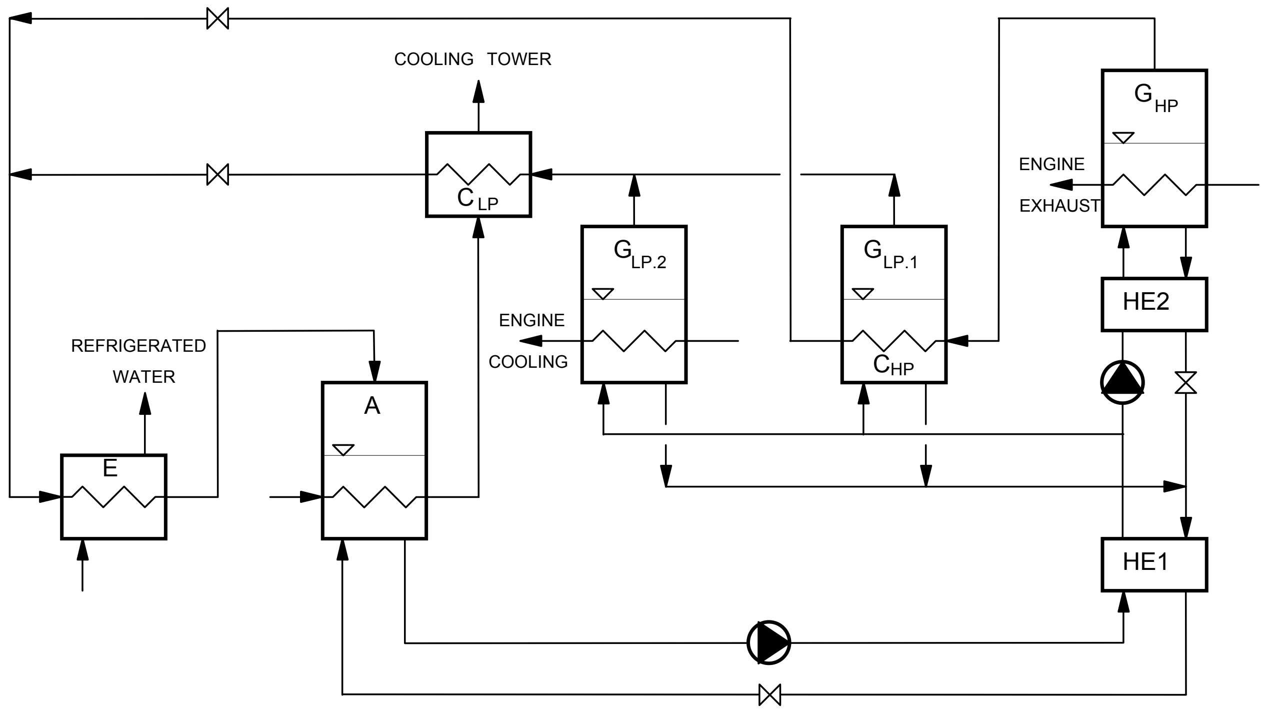

Figure 1. Block diagram of the absorption unit.

SYSTEM DESCRIPTION

The integrated refrigerating system proposed includes a reciprocating engine, a vapour compression chiller and an absorption unit. The reciprocating engine is based on an Otto cycle and is supplied with natural gas: table 1 gives its nominal characteristics and performances. It produces around 1000 kW of mechanical power together with 600 kW of thermal power at low temperatures (70-80°C) from the cooling jacket of the cylinders and 600 kW of thermal power at medium temperatures (150-500°C) from the heat recovery on the exhaust. The mechanical power is used to drive a single stage vapour compression chiller cooled by water which supplies refrigerated water at 7°C with a refrigerating capacity around 3000 kW.

Table 1. Nominal characteristics of the reciprocating engine.

Fuel Chemical Power Input (referred to NCV)	2595 kW
Mechanical Power	1034 kW
Mechanical Efficiency	39.8%
Cooling Water Inlet / Outlet Temperature	70 / 79.6°C
Heat Recovery from cooling water	600 kW
Exhaust engine Outlet / Discharge temperature	500 / 150°C
Heat recovery from exhaust	585 kW

The thermal energy recovered from the i.c. engine exhaust is used to drive a double effect water-lithium bromide cycle, while the thermal energy recovered from the cooling jacket of the engine drives a single effect water-lithium bromide cycle. The two absorption cycles were integrated into a single absorption unit with a common evaporator and absorber: figure 1 illustrates the block diagram of this absorption unit. The steam developed in the high pressure generator G_{HP} by the engine exhaust is used in the high pressure condenser C_{HP} to drive the low pressure generator $G_{LP.1}$. The second generator working at low pressure, $G_{LP.2}$, is driven by the heat recovery on the cooling jacket of the cylinders. The absorber A is linked to the generators in parallel configuration: the rich solution at the outlet of the absorber is subdivided into three separated flows, each of which is sent to one of the generators and the poor solutions at the outlet of each generators come back separately to the absorber. The rich and poor solutions perform two regenerative heat exchanges in the heat exchangers HE1 and HE2. The absorber A and the low pressure condenser C_{LP} are cooled in series by a cooling tower, whereas the evaporator E produces refrigerated water.

THEORETICAL MODEL AND SIMULATION CODE

A computer code was developed to simulate the system. The i.c. engine was simulated considering its nominal performance, while the vapour compression chiller was evaluated assuming an evaporator and a condenser thermal efficiency at 60%, a COP equal to the 40% of the Carnot refrigerating efficiency in the same temperature range and a mechanical efficiency of the transmission of 98%. A specific model was developed to simulate the absorption machine. Each device was simulated by a specific subroutine which reproduces the internal heat and mass transfer processes considering the effective exchange area and the thermodynamic, thermophysical and transport properties of the operative solution.

The absorber A consists of a horizontal smooth tube bundle, with the cooling water flowing inside the tube and the water-lithium bromide solution outside. The heat transfer coefficient on the water side was computed using the Dittus and Boelter 1930 equation, whereas the heat and mass transfer on the solution side was evaluated by the Nakoryakov and Grigor'eva 1977 model. This model refers to the absorption of saturated steam by a liquid film flowing along an isothermal surface under laminar flow which is governed by the following second order differential equations:

$$u\ (\partial T/\partial x) = a\ (\partial T^2/\partial y^2) \qquad (1)$$

$$u\ (\partial C/\partial x) = D\ (\partial C^2/\partial y^2) \qquad (2)$$

where u is the solution velocity along the surface, x and y are the coordinates along and right-angled to the surface, a is the thermal diffusivity of the solution, D is the diffusion coefficient of the water into the solution, T is the temperature, C the water concentration of the solution. The solution of the above differential equations gives the heat transfer coefficient h_E and mass transfer coefficients h_D:

$$h_E = q\ /\ (T_i - T_w) = \lambda\ (\partial T/\partial y)_{y=\delta}\ /\ (T_i - T_w) = \lambda\ /\ \delta \qquad (3)$$

$$h_D = m\ /\ (C_i - C_w) = \rho\ D\ (\partial C/\partial y)_{y=\delta}\ /\ (C_i - C_w) = \rho\ (D\ /\ \delta)\ (Le\ /\ \pi\ \xi)^{0.5} \qquad (4)$$

where q and m are the heat and mass fluxes, ρ and λ are the solution specific heat capacity and thermal conductivity, $\delta = \Gamma\ /\ (2\ \rho\ u)$ is the solution film thickness, Γ is the solution specific flow rate, $Le = a\ /\ D$ is the Lewis number and $\xi = a\ x\ /\ u\ \delta^2$ is a non-dimensional parameter. The subscript i and w refer to the interface and wall conditions respectively. This model, originally developed for isothermal plane surfaces, was applied to a horizontal tube bundle using a row by row iterative approach.

The model was compared against the experimental data relative to steam absorption by water-lithium bromide falling film over a bundle of horizontal smooth tubes. The experimental measurements, available in the open literature, generally refer to the overall absorption performance, whereas data points relative to the local heat and mass transfer are limited. The available experimental data are somewhat conflicting about the influence of the basic parameters on the absorber performance. The experimental measurements considered for the comparison include:
- data by Burdukov et al. 1980 relative to absorption over a column of six tubes,
- data by Nagaoka et al. 1987 relative to absorption over a bundle of horizontal tubes,

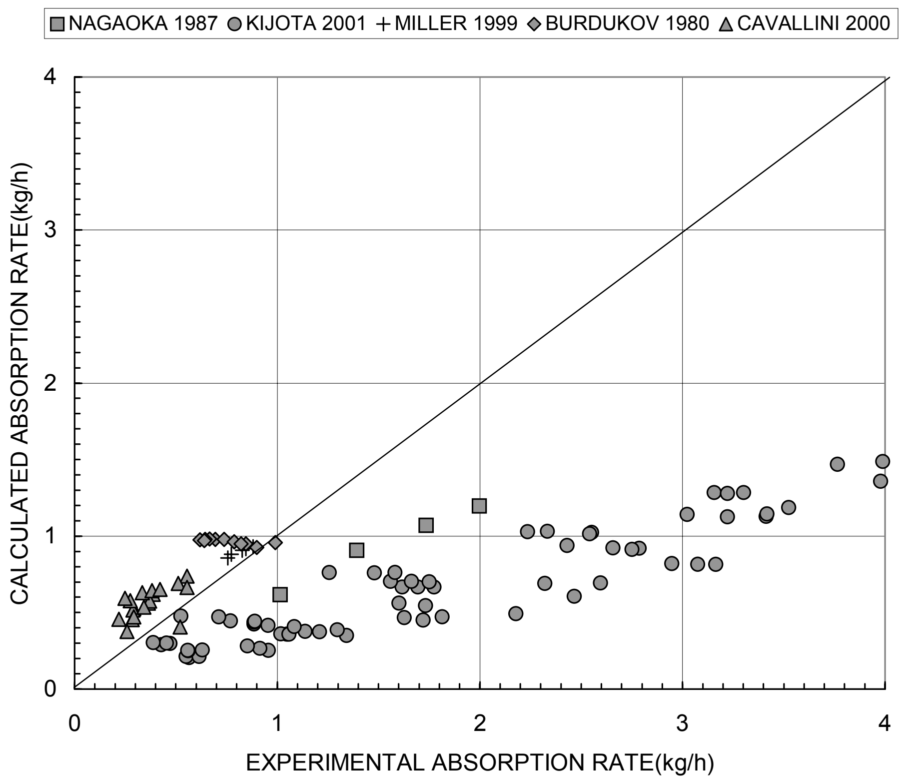

Figure 2. Comparison between experimental data relative to steam absorption by water-lithium bromide falling film over a bundle of horizontal smooth tubes and calculated values.

- data by Miller 1999 relative to absorption over a column of six tubes,
- data by Cavallini et al. 2000 relative to absorption over a column of five tubes,
- data by Kiyota et al. 2001 relative to absorption over columns with different numbers of tubes (from one to ten).

Figure 2 shows the comparison between the experimental absorption rates and the calculated values using the Nakoryakov and Grigor'eva 1977 model. In spite of the great scattering, it is possible to observe that the model generally underestimates the experimental data with a mean deviation around -27%. A similar result was obtained by Grossman 1983.

Moreover, the charge of the water-lithium bromide absorption machine usually includes additives (surfactants) which enhance the transfer rates in the absorbers. Several works concerning additives' effects on absorption process are available in the open literature including measurements of the heat and mass transfer rates. Unfortunately, the data scattering is, if possible, even more widespread than in the case of pure solution. However evidence is generally given to an enhancement in the absorption rate never lower than 40%.

Regarding the previous statements, in the present work the Nakoryakov and Grigor'eva 1977 model was modified multiplying the original heat and mass transfer correlations (eqs. 3 and 4) by a factor of two to account for the tendency of the model to underestimate and for the heat and mass transfer enhancement due to additives.

The high pressure generator G_{HP} is a kettle reboiler heated by the i.c. engine exhaust flowing inside the tube bundle, whereas the solution is regenerated on the shell side. The heat transfer coefficient inside the tubes was computed by the Dittus and Boelter 1930 equation, whereas the heat transfer coefficient on the shell side h_E was evaluated using the Palen 1983 model for nucleate boiling:

$$h_E = h_0 + h_1 F_B F_C \qquad (5)$$

where h_0 is the natural convection heat transfer coefficient outside a tube bundle, h_1 the pool boiling heat transfer coefficient for a single tube, F_B a factor which accounts for bundle effect with respect to single tube and F_C a correction term for solution regeneration with respect to pure substance.

The low pressure generator $G_{LP.1}$ and the high pressure condenser C_{HP} are integrated into a single kettle reboiler with the steam condensing inside the tube and the solution regenerated outside the bundle. The condensation heat transfer coefficient h_C was computed using the Boyko and Kruzhilin 1967 model:

$$h_C = 0.024\ (\lambda_L/d)\ Re^{0.8}\ Pr_L^{0.43}\ [1 + (\rho_L/\rho_V)^{0.5}] / 2 \tag{6}$$

where Pr_L is the liquid phase Prandtl number and Re the Reynolds number referred to the total steam mass flow rate and the liquid phase properties. The solution side heat transfer coefficient was calculated using the Palen 1983 model.

The low pressure generator $G_{LP.2}$ is a kettle reboiler driven by the cooling water of the i.c. engine: the heat transfer coefficients on water and solution side are calculated using the Dittus and Boelter 1930 equation and the Palen 1983 model, respectively.

The low pressure condenser C_{LP} is a tube bundle with the cooling water flowing inside the tube and the steam condensing on the shell side. The condensation heat transfer coefficient h_C was calculated using the Nusselt 1916 equation in the form:

$$h_C = 1.51\ (\lambda_L^3 \rho_L^2 g / \mu_L^2)^{1/3}\ (4\Gamma/\mu_L)^{1/3} \tag{7}$$

where Γ is the condensate specific flow rate. The cooling water heat transfer coefficient was computed using the Dittus and Boelter 1930 equation.

The evaporator E is a tube bundle in which the refrigerated water passes inside the tube while the condensate to be vaporised flows on the bundle as a film. Considering the thermal resistance concentrated in the conduction in the liquid film, the vaporisation heat transfer coefficient was derived from the Nusselt analysis for film condensation in the following form:

$$h_E = \lambda_L / \delta = \lambda_L / [1.385\ (\mu_L \Gamma / g\ \rho_L^2)]^{1/3} \tag{8}$$

The refrigerated water heat transfer coefficient was computed using the Dittus and Boelter 1930 equation.

Table 2 gives the main geometrical characteristics of the different heat exchangers: number of tubes N_T, tube length L_T, inside and outside diameters of the tubes d_I and d_O, tube side number of passages N_P. The thermodynamic and thermophysical properties of the water/lithium bromide solution (with no additives) was evaluated by a specific code in accordance with Lower 1961, McNelly 1979 and Hellmann & Grossman 1996.

The main program links the unit subroutines in order to reproduce the absorption machine in a sequential approach. Once set the specific boundaries conditions (refrigerated water inlet and outlet temperature, cooling water inlet temperature, maximum regeneration temperature) and assumed initial guess for the unknown parameters (evaporation temperature, condensation temperature, intermediate temperature inside the machine), the final solution is obtained by an iterative approach until convergence between the guessed and calculated parameters. The final output includes temperature, pressure, concentration in each component together with the heat and mass flow rates processed.

Table 2. Geometrical characteristics of the different heat exchangers in the absorption unit.

Heat Exchanger	N_T	L_T (m)	d_I (mm)	d_O (mm)	N_P
High Pressure Generator G_{HP}	31	6	39.5	44.5	2
Low Pressure Generator $G_{LP.1}$	96	6	12	16	8
Low Pressure Generator $G_{LP.2}$	282	6	19	22	6
Absorber A	900	6	16	20	2
Condenser C	782	6	16	20	2
Evaporator E	566	6	16	20	2

PERFORMANCE ANALYSIS

The performance analysis of the absorption unit was carried out considering the traditional Coefficient of Performance (COP), whereas the efficiency of the whole system was investigated considering the Primary Energy Ratio (PER), defined as the ratio between the whole refrigerating capacity (sum of the separate contributes of absorption and vapour compression units), and the thermal power of the fuel supplied to the engine. The absorption unit was designed to work with a maximum temperature at the high pressure generator from 130 to 140°C, the optimum range calculated by the cycle analysis (Lazzarin et al. 2000). The parametrical analysis investigates the effect of the main boundary conditions, cooling water inlet temperature and outlet temperature of the refrigerated water, on the PER of the system and the COP of the absorption unit.

Figure 3 shows the COP of the absorption unit and the PER of the whole system as a function of the cooling water inlet temperature for three different refrigerated water outlet temperatures: 4, 7 and 10°C. The PER of the whole system decreases with the cooling water inlet temperature, whereas the COP of the absorption unit remains constant around 1. The increase in the cooling water temperature reduces the concentration difference between rich and poor solutions progressively and so, for each operative conditions, it is possible to determine a maximum value compatible with a correct operation of the absorption unit. For example, under a refrigerated water outlet temperature around 7°C, the maximum cooling water inlet temperature at the absorption unit is around 35-36°C, as illustrated in figure 4. Moreover, the absorption unit cannot work with a refrigerated water outlet temperature less than 3°C, as the evaporation temperature becomes lower than the water ice point.

Under the nominal operative conditions for air conditioning chiller, refrigerated water outlet temperature at 7°C and cooling water inlet temperature around 32°C, the absorption machine shows a COP around 1, very close to the performance of a traditional two stage absorption chiller, whereas the PER of the whole system is around 1.6, a value 25% higher than that of a simple vapour compression unit (PER= 1.2). This performance increase is due to the absorption unit which supplies around 1050 kW out of 4100 kW of the whole refrigerating capacity, as shown on figure 5.

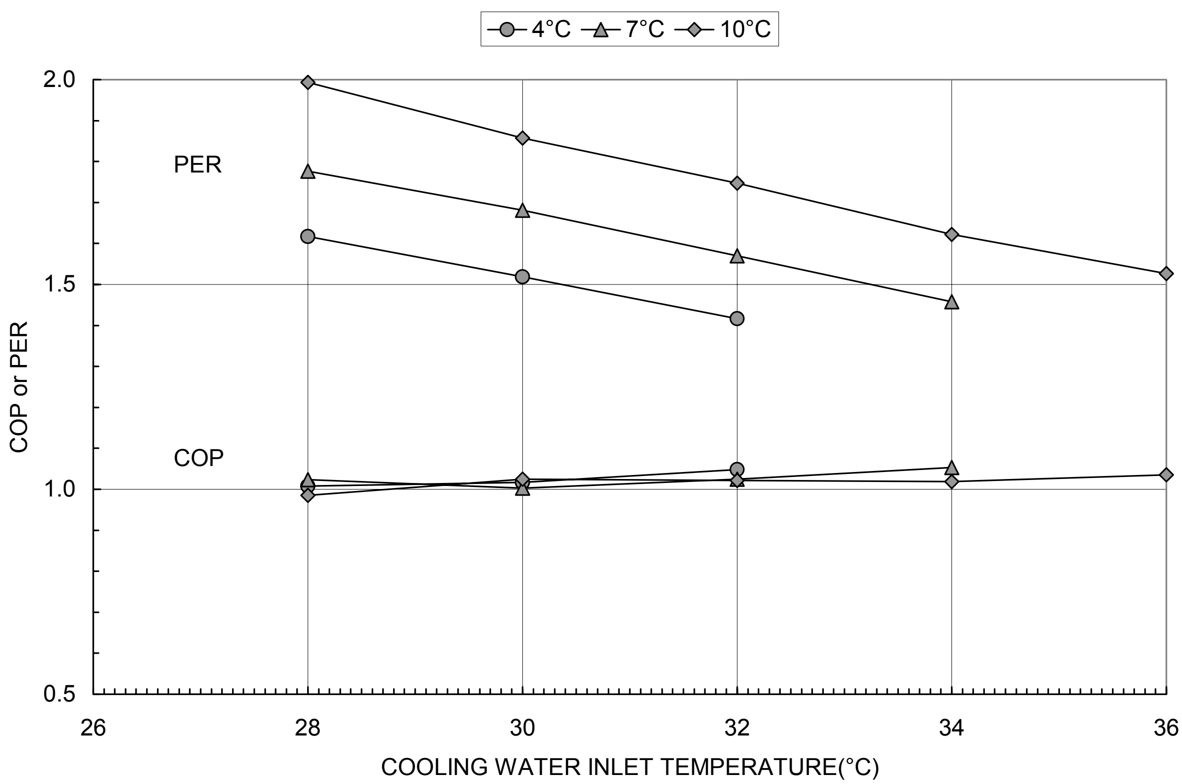

Figure 3. COP and PER vs. cooling water inlet temperature.

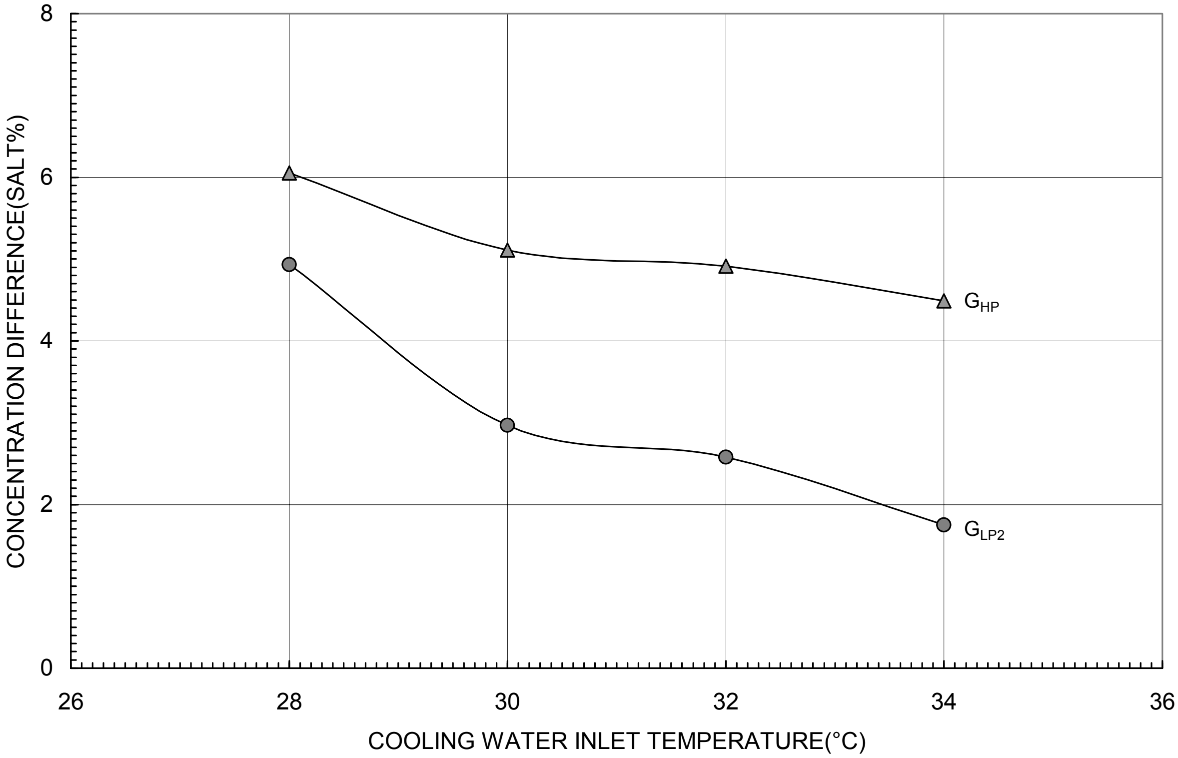

Figure 4. Solution concentration range in the generators vs. cooling water inlet temperature.

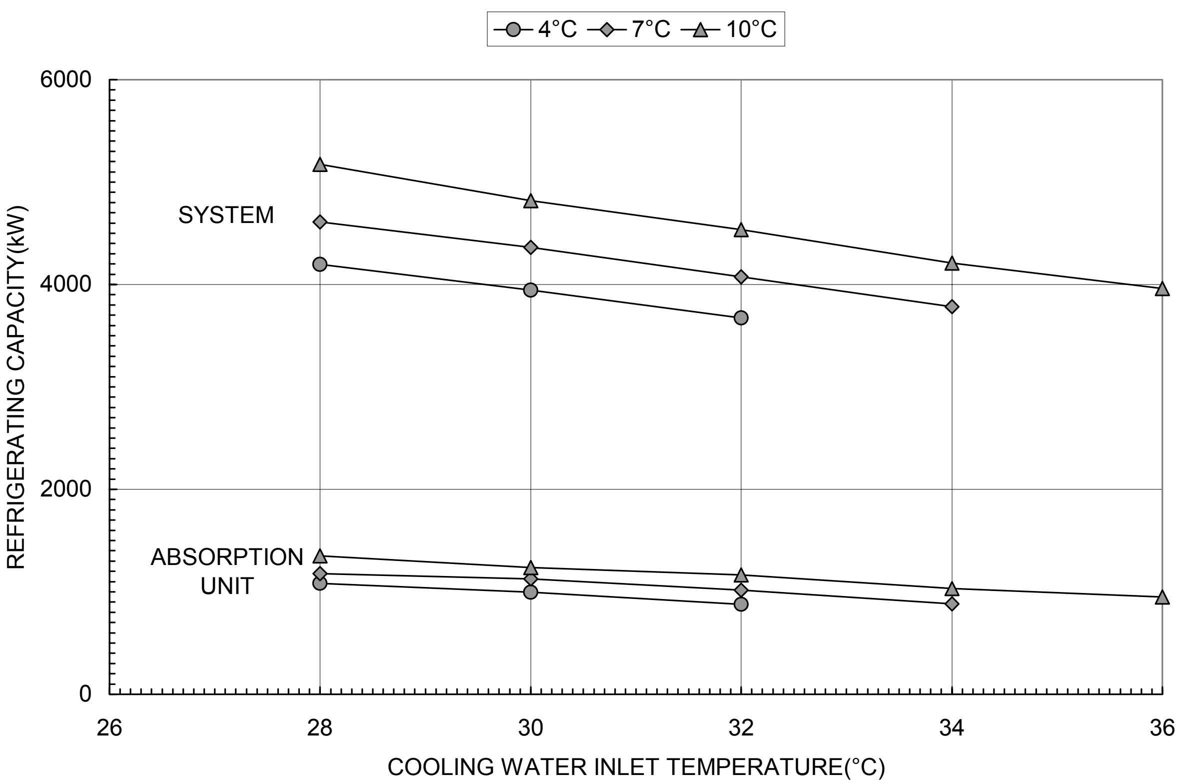

Figure 5. Refrigerating capacity vs. cooling water inlet temperature.

CONCLUSION

An integrated refrigerating system which includes a reciprocating engine, a vapour compression chiller and an absorption unit was analysed in order to evaluate the performance and the sensibility to the main boundary conditions and the operative range. The absorption unit was simulated by a specific computer code which reproduces the internal heat and mass transfer processes considering the effective exchange area and the thermodynamic, thermophysical and transport properties of the operative solution. In the typical operative range of an air conditioning application (refrigerated outlet temperature at 7°C and cooling water inlet temperature at 32°C), the COP of the absorption unit is around 1, while the PER of the whole system is around 1.6, a value 25% higher than that of a traditional vapour compression unit (PER = 1.2). This performance increase is due to the absorption unit which supplies around 1050 kW out of 4100 kW of whole refrigerating capacity. The absorption unit could operate only with cooling water inlet temperature lower than 35-36°C and refrigerated outlet temperature higher than 3°C.

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