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EXERGETIC ANALYSIS OF A 5 TR EXPERIMENTAL ABSORPTION REFRIGERATION UNIT POWERED BY LIQUEFIED PETROLEUM GAS (GLP) AND EXHAUST GASES

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Abstract. This study investigated a 5 tons of refrigeration, TR, experimental absorption refrigeration unit assembled in the laboratory to produce cold and heat. The methodology was based on the use of the first and second law of thermodynamics to evaluate either the energetic (or first law) or the exergetic (second law) efficiencies of the system. The work consisted of the design and assembly of a prototype in the laboratory, its characterization and instrumentation. In a final stage, using the experimental measurements, a thermal and exergetic analysis of the system was performed, aiming the optimization of the operating and project parameters for maximum thermodynamic performance of the produced technological innovation. The results show the dependence of the thermal and exergetic efficiencies on the water mass flow rates of the cold and hot sides of the absorption refrigerator.

Keywords. *absorption refrigerator, cold and hot heat exchangers, cogeneration, exergetic analysis*

1. Introduction

Nowadays the reduction of energy sources (water power, fossil fuel) and the increase of pollution, the research centers are maximize the available energy, using cogeneration systems.

Lazzarin *et al*. (1996) showed an experimental study with an ammonia-water absorption chiller. The system in refrigerating version able to operate down to -10^{o}C, using a mixture of water/glycol at 35% of glycol. The study predicts the increase of cooling capacity and operate water chiller down to -25°C with a high concentration in ethylene glycol.

De Francisco *et al*. (2002) developed prototype of an ammonia-water absorption system for low capacity (2kW), designed for solar-powered refrigeration. The system showed a low efficiency due to the solar-powered is not enough for low temperatures in refrigeration systems.

Kodal *et al*. (2003) showed a thermoeconomic optimization study to determine the optimal operation and design parameters of irreversible absorption refrigerator and heat pump systems. The effects of the internal irreversibility, the economical parameters and the external temperatures on the global and optimal performances were discussed.

Bulgan (1997) showed a study that use low temperature (85-110°C) energy sources in ammonia-water absorption refrigeration system. Using the mathematical model, a thermodynamic analysis is obtained. The optimum working are determined.

Misra *et al*. (2003) developed a thermoeconomic optimization. The results showed a significant improvement in the system performance without additional investments.

The aim of this study is to find the optimum operating conditions for the ammonia water absorption chiller. The analysis is performance by doing energy and exergy balances for the system. The efficiency of the system is calculated and the results are discussed.

2. Description of equipment and performance

The equipment is basically a 5 ton ammonia-water absorption chiller manufacture by Robur SpA. The equipment produces simultaneously hot water up to 65°C and chilled water down to –5°C. The mixture charge of 10.0 kg of water and 7.0 kg of ammonia. The generator is heated up by a 28 kW gas burner (LPG). The nominal cold and hot water flow rate are 2.5 m^3/h and 3.0 m^3/h. The only electric power required by the solution pump is 540W. The unit makes a wide variety of convenient applications available: installations where simultaneous heating and cooling output is required, (process applications, conditioning installations) where cooling and pre-heating is required (hotels, centres fitness, swimming pools, etc.); low temperature heating systems, where there are available renewable energy sources (lake water, river water or ground water).

The scheme of the chiller is showed on Fig. 1.

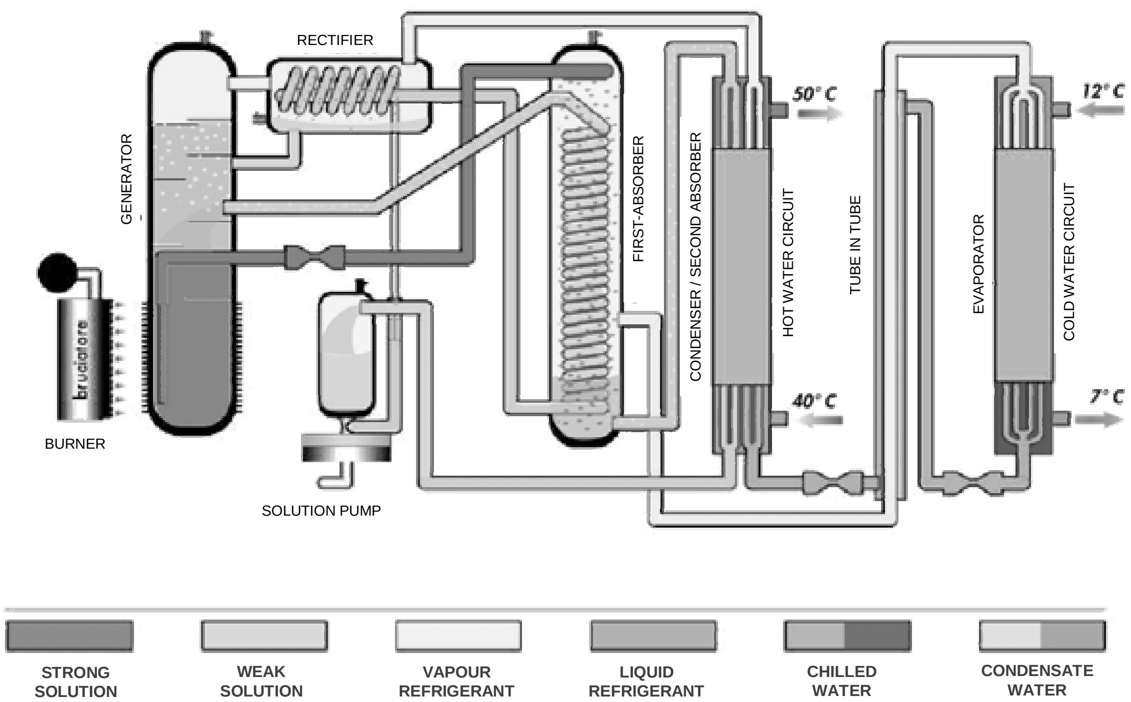

Figure 1. Ammonia-water absorption chiller (Robur, 2005).

The system showed above is composed by a generator, where the ammonia-water mixture is heated by a LPG (liquefied petroleum gas) burner. Of this step the ammonia vapour is separated, but it is mixed up with fractions of water vapour. The ammonia vapour is purified from water in the rectifier which is cooled by a solution rich in ammonia that flows after the pump. The ammonia vapour refrigerant (pure) proceed to the condenser, where it happen the heat exchange with the hot water circuit. After this step, ammonia liquid from the condenser passes through the tube in tube heat exchanger. Ammonia vapour is again produced in the evaporator, where it happen the heat exchange with the cold water circuit. The ammonia vapour is absorbed by the strong solution from generator in a first absorber cooled by the strong solution that has just cooled the rectifier. The absorption is completed in a second absorber (condenser). Then the weak solution at the lower pressure is sending to the higher-pressure generator, using a diaphragm pump, driven by an oil rotary pump.

3. Mathematic model

3.1. Energy analysis

The system to be analyzed is the cold and the hot water sink, that represent thermal loaded for cooling and heating, respectively. The energy balance of the system is shown below.

The capacities of cooling and heating are defined in function of project and operating parameters. The Eq.(1) shows the cooling capacity ($\dot{Q}_L$) in function of the variation of the cold and hot flow rates. The heating capacity ($\dot{Q}_H$) follows the same procedure, as shown in Eq.(2).

$$\dot{Q}_L = f_1\left(\vec{X}, \dot{m}_H, \dot{m}_L\right) \quad (1)$$

$$\dot{Q}_H = f_2\left(\vec{X}, \dot{m}_H, \dot{m}_L\right) \quad (2)$$

The cooling and heating capacity of the system can be expressed as:

$$\dot{Q}_L = \dot{m}_L \overline{cp} \Delta T_L \quad (3)$$

$$\dot{Q}_H = \dot{m}_H cp_W \Delta T_H \quad (4)$$

Where ($\dot{m}_L$) and ($\dot{m}_H$) are the cold and hot water flow rate, and (ΔT_L) and (ΔT_H) are the differences of the outlet and inlet water temperatures for cold and hot sink.

As cold cycle use the mixture of 25% ethylene glycol in water, the specific thermal capacity of the mixture ($\overline{cp}$) is determine by Eq.(5).

$$\overline{cp} = 0{,}25\, cp_{eg} + 0{,}75\, cp_w \quad (5)$$

For the range of temperatures use in these tests, the specific thermal capacity of ethylene glycol (cp_{eg}) and water (cp_w) are considered as constant.

The system was studied using a burn of the LPG as heat source. The heat transfer of the fuel LPG ($\dot{Q}_{LPG}$) is shown in Eq.(6), where ($\dot{m}_{LPG}$) is the fuel flow rate and (LHV_{LPG}) is the lower heat value of the fuel. The LPG is a mixture of the 50% propane (C_3H_8) and 50% butane (C_4H_{10}).

$$\dot{Q}_{LPG} = \dot{m}_{LPG} LHV_{LPG} \quad (6)$$

The efficiencies of the system based on the first law of thermodynamics are shown in the Eqs. (7), (8) and (9). Where ($\eta_{I,L}$) is the efficiency of the cold cycle, ($\eta_{I,H}$) is the efficiency of the hot cycle and ($\eta_{I,comb}$) is the combined efficiency.

$$\eta_{I,L} = \frac{\dot{Q}_L}{\dot{Q}_{LPG}} \quad (7)$$

$$\eta_{I,H} = \frac{\dot{Q}_H}{\dot{Q}_{LPG}} \quad (8)$$

$$\eta_{I,comb} = \frac{\dot{Q}_L + \dot{Q}_H}{\dot{Q}_{LPG}} \quad (9)$$

3.2. Exergy analysis

Exergy is defined as the maximum possible reversible work obtainable in bringing the state of a system to equilibrium with that of environment (Bejan *et al*, 1996). In the absence of nuclear, magnetic, electrical, and surface tension effects, the total exergy of a system can be divided in to two components: physical exergy (E^{PH}), and chemical exergy (E^{CH}).

$$\dot{E} = \dot{E}^{PH} + \dot{E}^{CH} \tag{10}$$

The system to be analyzed is the cold and the hot water sink, that represent thermal loaded for cooling and heating, respectively. The exergy transfer associated with a stream of matter is shown below:

$$\dot{E}^{PH} = \dot{m}\left(h - h_0\right) - T_0\left(s - s_0\right) \tag{11}$$

The exergy of cold cycle ($\dot{E}_L$) was determined by Eq.(12), where ($h_{i,L}$) and ($s_{i,L}$) are enthalpies and entropies of the inlet and outlet of the mixture of cold fluid, respectively. Consider the reference temperature (T_0) as 298,15 K.

$$\dot{E}_L = \dot{m}_L\left[\left(h_{out,L} - h_{in,L}\right) - T_0\left(s_{out,L} - s_{in,L}\right)\right] \tag{12}$$

The exergy of hot cycle is shown in Eq.(13). Where ($h_{i,H}$) and ($s_{i,H}$) are enthalpies and entropies of the hot water, respectively.

$$\dot{E}_H = \dot{m}_H\left[\left(h_{out,H} - h_{in,H}\right) - T_0\left(s_{out,H} - s_{in,H}\right)\right] \tag{13}$$

The exergy of fuel ($\dot{E}_{LPG}$) was determined by Eq.(14), where ($\dot{m}_{LPG}$) is the fuel flow rate and ($\xi_{CH,LPG}$) is the chemical exergy of the fuel.

$$\dot{E}_{LPG} = \dot{m}_{LPG}\, \xi_{CH,LPG} \tag{14}$$

The efficiencies based on the second law of thermodynamics for the cold, hot and combined cycles are shown below:

$$\eta_{II,L} = \frac{\dot{E}_L}{\dot{E}_{LPG}} \tag{15}$$

$$\eta_{II,H} = \frac{\dot{E}_H}{\dot{E}_{LPG}} \tag{16}$$

$$\eta_{II,comb} = \frac{\dot{E}_L + \dot{E}_H}{\dot{E}_{LPG}} \tag{17}$$

4. Results and Discussion

For evaluate the optimal parameters (cooling and heating capacity) several tests were done. The mainly aim of this study was verify the maximum cooling and heating capacity through of the variation of the cold and hot flow rate. The optimum point of any system is characterized by the high point in the middle of the plot and the low points in the extremes.

The range of operation of the machine consists in 0.3 kg/s in minimum flow rate and 0.90 kg/s in maximum flow rate. The system turns off for flow rates below 0.3 kg/s. The range of measure of the flow meter from 0.17 kg/s to 0.93 kg/s.

The Fig. (2), (3) and (4) show the variation of cold fluid flow rate from 0.33 to 0.83 kg/s in function of differences of temperatures, cooling and heating capacities and efficiencies. The hot fluid is fixed in 0.90 kg/s.

The Fig. (2) shows the differences of the water temperatures for the cooling (ΔT_L) and heating (ΔT_H) cycle in function of the variation of cold fluid flow rate.

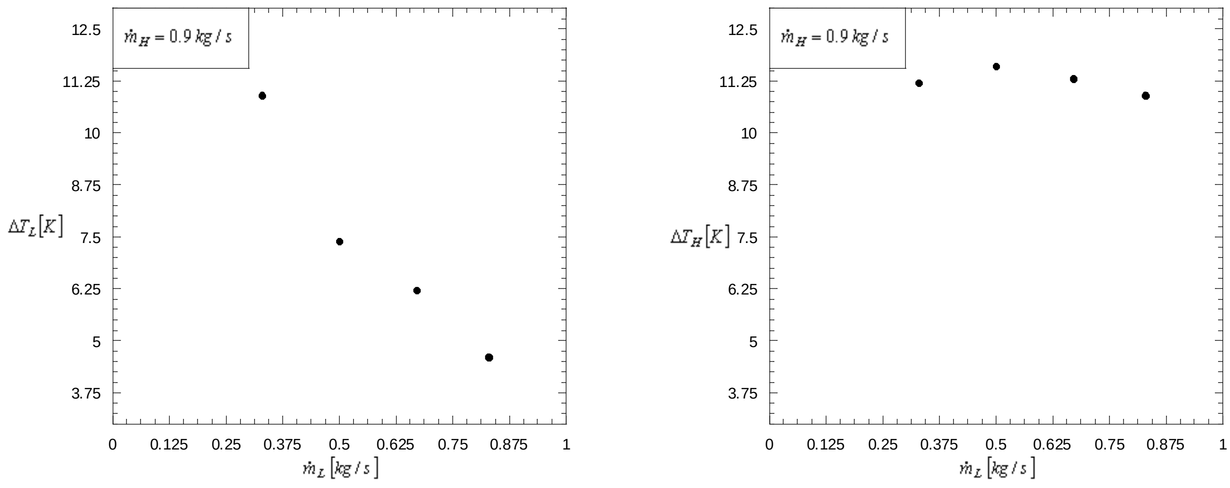

Figure 2. Variations of the cold water flow rate in function of the differences of water temperature.

It can observe in the Fig. (2) the difference of cold water decrease with the increase of cold water flow rate. On the other hand the hot water remains constant values.

The Fig. (3) shows the cooling capacity in function of the variation of cold fluid flow rate (0.33 to 0.83 kg/s).

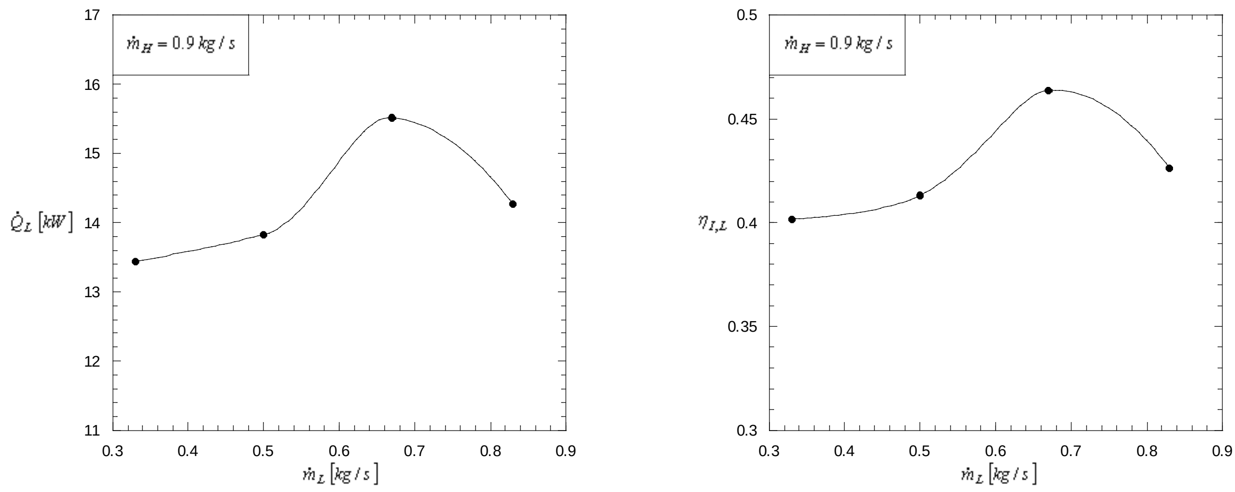

Figure 3. Variations of the cold water flow rate in function of the cooling capacity and efficiency.

This result suggest that the optimal point of the cold water flow rate for maximum cooling capacity is 0.67 kg/s. The maximum cooling capacity is 15.52 kW. This value represents 8.1 to 13.4% of gain the refrigerant effect.

The maximum First law efficiency of cold water for this flow rate variation is 46.4%.

The heating capacity did not show the same behavior of cooling capacity and efficiency for the same variation of flow rate, that mean the optimum point did not verify.

The Fig. (4) shows the exergy of cooling water in function of the variation of cold fluid flow rate.

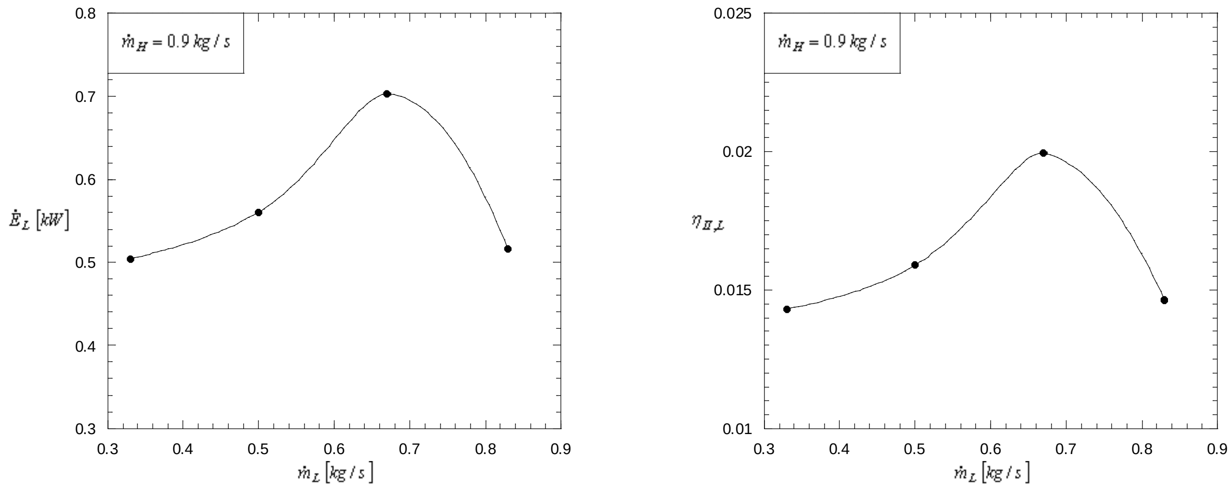

Figure 4. Variations of the cold water flow rate in function of the exergy of cooling water and efficiency.

This results confirm the optimum point as 0.67 kg/s. The maximum available work (exergy) is 0,70 kW and the maximum second law efficiency of cold water is 2.0%.

The Fig. (5), (6) and (7) show the variation of hot water flow rate from 0.50 to 0.90 kg/s in function of differences of temperatures, cooling and heating capacities and efficiencies. The cold fluid is fixed in 0.67 kg/s.

The Fig. (5) shows the differences of the water temperatures for the cooling (ΔT_L) and heating (ΔT_H) cycle in function of the variation of hot fluid flow rate.

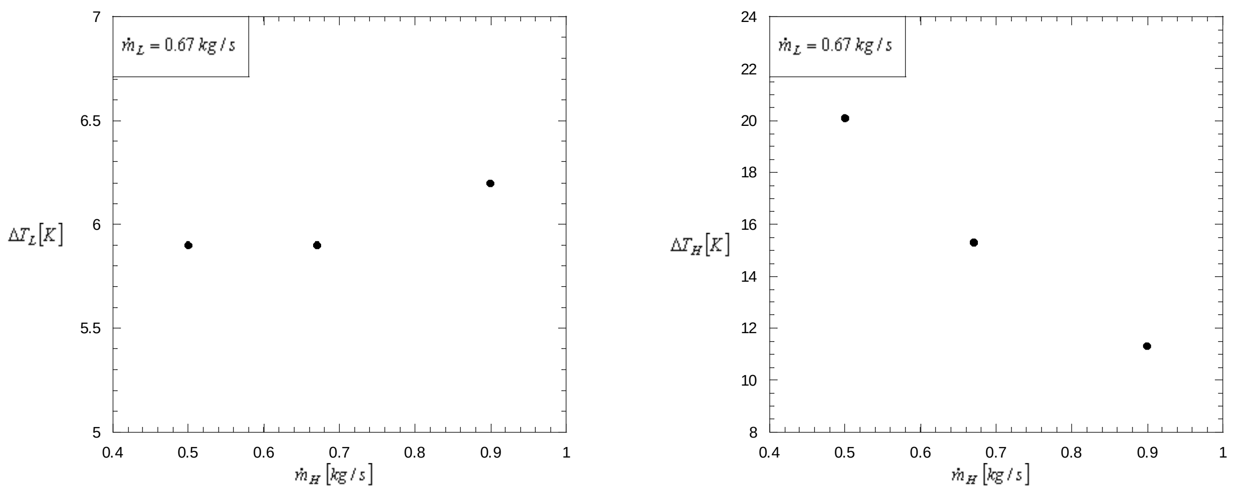

Figure 5. Variations of the hot water flow rate in function of the differences of water temperature.

The differences of hot water (ΔT_H) decrease while the hot water flow rate increase. The difference cold water remains constant values.

The Fig. (6) shows the heating capacity in function of the variation of hot water flow rate (0.5 to 0.9 kg/s).

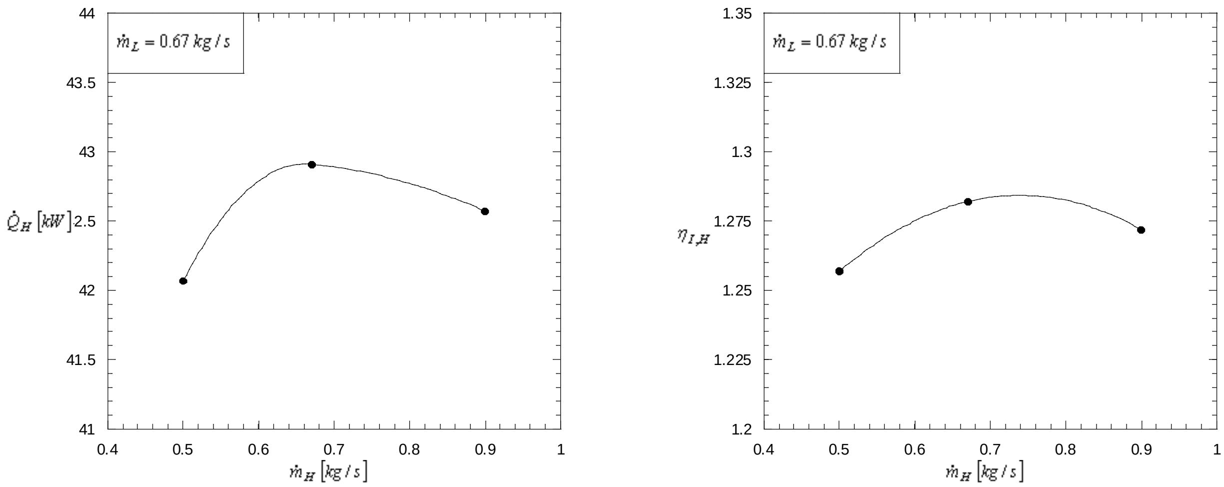

Figure 6. Variations of the hot water flow rate in function of the heating capacity and efficiency.

This result suggests that the optimal point of the hot water flow rate for maximum heating capacity is 0.67 kg/s. The maximum heating capacity is 42.9 kW. The maximum First law efficiency of cold water for this flow rate variation is 128.2%.

The Fig. (7) shows the exergy of heating water in function of the variation of hot fluid flow rate.

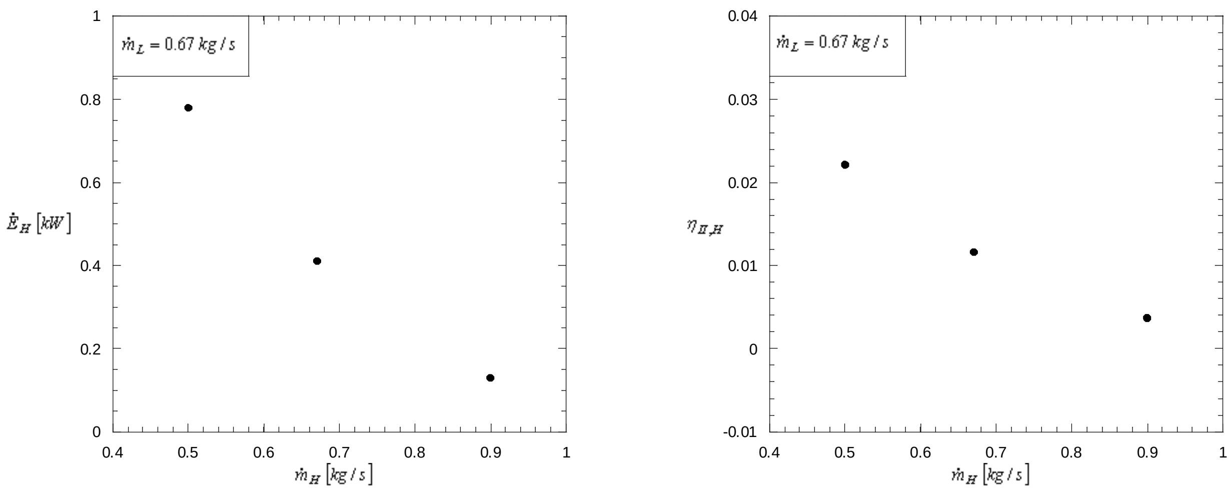

Figure 7. Variations of the hot water flow rate in function of the heating capacity and efficiency.

This result suggests a different optimum point to heating capacity. The maximum available work (exergy) must show the same optimum point.

The optimization study is performed to determine the optimal operation and design parameters for ammonia-water absorption chillers. The optimum value for cooling capacity and exergy was determined for some operation parameters: 0.67 kg/s cold fluid flow rate and 0.90 k/s hot water flow rate.

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