



Technical and environmental performance of a solar/gas driven absorption chiller using $\text{NH}_3/\text{LiNO}_3$

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ARTICLE INFO

Article history:

Received 22 July 2022

Accepted 06 October 2022

Keywords:

Solar cooling

Absorption chillers

Ammonia

Lithium Nitrate

Natural Gas

CO₂ emissions

ABSTRACT

This study deals with the evaluation of the technical and environmental operation of solar/gas driven $\text{NH}_3/\text{LiNO}_3$ absorption chillers for malls under the environmental conditions of the city of Barranquilla, Colombia. It involves a sensitivity study on the absorption chillers performance at chilled water temperatures of 12 °C and 6 °C and different solar irradiations. It also includes the share of cooling and CO₂ equivalent emissions provided by the absorption chillers considering as a base case the energy consumed by the mechanical vapour compression chillers operating in three selected malls. Results showed that solar energy could provide around 40% and 50% of the total energy required to drive the absorption chillers on a service day. This value could be increased by adding a higher solar panel area, however, the available roof area for solar equipment was a limiting factor. Moreover, the COP and SCOP of the absorption chillers were found up to 0.64 and 0.44, respectively. Additionally, the solar/gas driven absorption chillers could cover up to 83%, 100%, and 35% of the peak cooling requirements in malls 1, 2, and 3, respectively, while providing the highest reductions in CO₂ equivalent emissions.

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ISSN: 1112-2242 / EISSN: 2716-8247



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1. Introduction

Malls located in tropical environments consume high amounts of electrical energy to provide cooling to large volumes of air to satisfy the comfort of customers. The most common refrigeration system used to provide the cooling effect is the mechanical vapour compression system [1]. In order to move to sustainable energy technology systems, it is of interest to identify adequate alternatives to reduce the dependence of cooling units on the electrical grid. Absorption cooling systems have become one of the most promising options to mechanical vapour compression systems on the move toward environmentally friendly sustainable cooling technologies [2]. Absorption systems require small amounts of electrical energy to be powered while the main energy input is thermal energy. This thermal energy can come from different heat sources such as renewable energies, residual heat, or natural gas [3–5]. The main difference between vapour absorption systems and conventional vapour compression units relies on the way how the refrigerant flow coming from the evaporator becomes a high temperature and pressure vapour, which is under the required conditions to be condensed. This task is carried out by the compressor in the conventional compression system while the same objective is achieved by the absorber and generator pack where intensive heat and mass transport processes occur [6,7].

The performance of solar cooling technologies depends on the environmental conditions and system operating conditions [8]. Those places in tropical regions present great potential for solar cooling [9]. Moreover, solar thermal cooling is becoming attractive for air conditions applications in terms of performance and cost [8,10]. One of the reasons is that absorption cooling systems can be feed by various available thermal heat sources [2,5]. It has been demonstrated that investment returns could be obtained below 4 years for absorption chillers driven by waste heat. Also, studies have reported that solar energy can contribute up to 58% of the total energy needed to drive a 10 kW solar/gas driven water/LiBr absorption chiller in Algeria [11]. Additionally, studies have shown that parabolic collectors are preferred to evacuated tubes to drive a water/LiBr absorption system given their higher performance [12], however, it can increase the complexity of the solar collection system. Studies have also demonstrated the feasibility of solar-assisted water/LiBr absorption cooling units in buildings for commercial activities in Saudi Arabia [13] and buildings for residential purposes in The United Arab Emirates [14]. More recently, solar/gas assisted H₂O/LiBr absorption systems showed great potential to reduce the electrical energy requirements in cooling systems applied to commercial buildings in the Caribbean region of Colombia, while the internal rate of return

could range between 40% and 54.6%, which depends on the cooling thermal load provided by the absorption systems [15].

From previous studies, it could be said that solar system dimensions are key in the technical and economic feasibility of the solar cooling absorption system. It strongly affects the payback period which also considers the environmental conditions, the cooling requirements, working fluids used, and operating conditions [8,9]. Regarding the working fluid used in absorption chillers, the most common is the H₂O/LiBr for air-conditioning applications, however, this working pair presents high risks of crystallization in warm environments. As an alternative mixture, the NH₃/LiNO₃ appears as a promising option to the conventional mixture since there is no risk of crystallization at high environmental temperatures as those in tropical weather [16]. In Colombia, solar cooling is not exploited despite the high solar irradiation available in various regions because there is a lack of knowledge about the technologies with the promising use for solar cooling [17]. In Barranquilla, Colombia, which weather is tropical Savanna, there is a high demand for air conditioning and this need is mainly covered by compression chillers. Therefore, the objective of this study is to evaluate the performance and environmental impact of solar and/or gas driven NH₃/LiNO₃ absorption chillers for malls in Barranquilla. It involves a sensitivity study on the performance of the absorption chillers at different chilled water temperatures and solar irradiation, as well as the share of cooling and CO₂ equivalent emissions provided by the absorption chillers considering as a base case the energy consumed by the mechanical vapour compression chillers operating in three selected malls.

2. Methodology

The methodology followed to carry out the present study involves the selection of three malls with different area specifications. In these buildings, the cooling loads were estimated considering the electrical power consumed by conventional compression units used for that purpose. Then, a thermodynamic simulation was conducted to evaluate the performance indicator of absorption chillers under various case studies as a cooling alternative to mechanical compression chillers.

The three malls selected are located in the city of Barranquilla, Colombia. The city of Barranquilla is a well-recognized city in Colombia for its economic growth. Its population is over 1.2 million people. The climate in this city is defined as Tropical Savanna, with environmental temperatures ranging from 24 °C to 35 °C and relative humidity above 68% [18].

Table 1. Air-conditioned area and available roof area of the selected malls.

Mall	Area for air conditioning (A_{CA}) (m^2)	Available roof area (A_{SC}) (m^2)	Ratio (A_{CA}/A_{SC})
M1	28390	4043	7
M2	21000	2391	9
M3	19800	1214	16

Table 1 presents the main details of the malls in terms of air-conditioned areas and available roof areas. The roof areas are of special interest since it marks the area limits for solar panel applications. From Table 1 can be noted that mall 3 (M3) could be the case with the lowest potential for solar cooling given its wider air-conditioned area in relation to the available roof area. Regarding the power measurements of the conventional chillers operating in the malls, a power analyser FLUKE 435 II was employed. Data were collected from 7:00 to 20:30 hours, every one minute. The power data for M2 were collected in March 2019. The power data for M1 and M3 were collected in October 2019.

Fig. 1 shows the hourly mean solar irradiation for Barranquilla. From this figure can be noted that March is the month with the highest mean irradiation and November the one with the lowest mean irradiation. The irradiation per day is on average around 846.4 Wh/m^2 [18]. This value is used for the dimensioning of the solar panels.

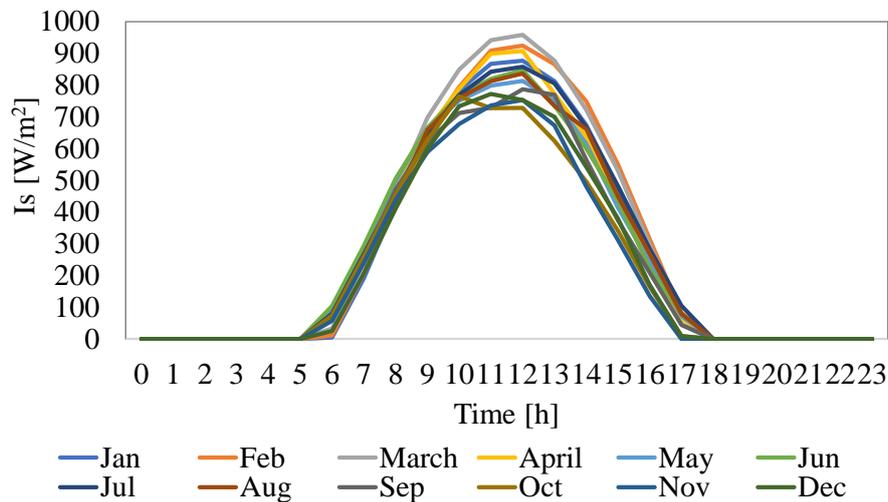


Fig 1. Hourly mean solar irradiation for Barranquilla [18]

2.1 System description

The system evaluated consists of an absorption chiller, which uses $NH_3/LiNO_3$ as a working fluid, and its integration with a solar panel system and natural gas burner as shown in Fig. 2. The operation of the absorption unit is similar to that of a mechanical vapour compression

system. However, in the absorption unit, the refrigerant flow in vapour phase coming from the evaporator is absorbed in the absorber by a mixture of $\text{NH}_3/\text{LiNO}_3$ which is weak in NH_3 . The resulting mixture, high in NH_3 , is then pumped to a generator where the desorption process takes place. In order to make the desorption process happens, the energy input to the chiller is supplied through the generator. This energy input comes from the solar energy captured by the solar panels and the thermal energy released in the natural gas burner. The vapour flow generated in the generator is then sent to the condenser while the mixture of $\text{NH}_3/\text{LiNO}_3$, weak in NH_3 , flows to the absorber one more time. Finally, the condensed flow leaving the condenser is sent to the evaporator where the cooling effect occurs.

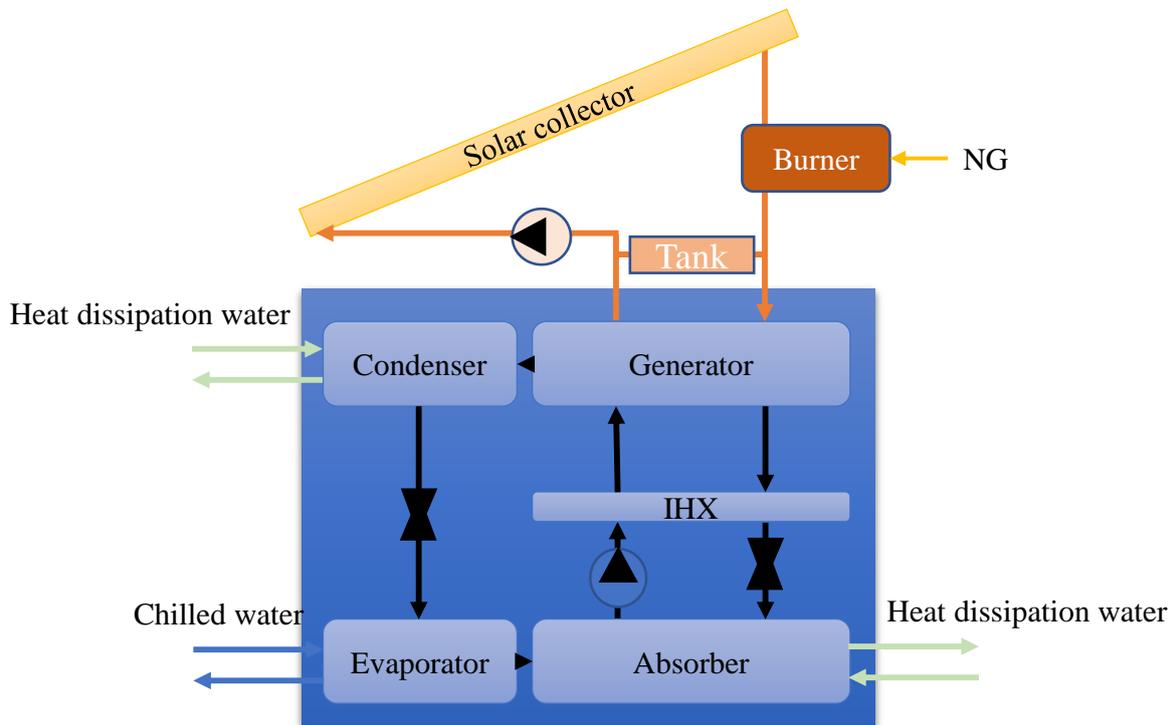


Fig 2. Simplified diagram of the solar/gas driven absorption chiller

This study is focused on the evaluation of the absorption system powered by solar energy and/or gas to cover the cooling needs provided by conventional chillers operating in the selected malls. Therefore, a comparison of the cooling load covered by each configuration is presented.

2.2 Model details

The model of the chiller consists of energy, mass and species balances in each component of the whole chiller by applying Eq. 1, 2 and 3 [19]. The components considered include the absorber, generator, solution pump, internal heat exchanger (IHX), solution expansion valve, evaporator, condenser, and refrigerant expansion valve.

$$\dot{Q} + \sum \dot{m} \cdot h_{in} = \dot{W} + \sum \dot{m} \cdot h_{out} \quad (1)$$

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (2)$$

$$\sum \dot{m} \cdot X_{in} = \sum \dot{m} \cdot X_{out} \quad (3)$$

The main considerations for the model simulation are presented as follows:

- Saturated vapour at the condenser inlet.
- Saturated liquid leaving the condenser.
- Saturated vapour leaving the evaporator.
- Adiabatic flow in pipes.
- Gas burner with efficiency (η_{GN}) of 0.85 [11].
- The solution pump with an efficiency of 0.6.

Regarding the solar panels, the solar energy collected (\dot{Q}_s) is estimated as [20]:

$$\dot{Q}_s = A_c I_s \quad (4)$$

where A_c refers to the solar collector area, and I_s is the solar irradiation. The thermal efficiency of the solar system is presented as:

$$\eta_c = \frac{\dot{Q}_u}{\dot{Q}_s} = \frac{\dot{m}_{water} c_p (T_{out} - T_{in})}{A_c I_s} \quad (5)$$

In Eq. (5), \dot{Q}_u is the heat flow to the pressurized water that flows in the solar collector. In this study, evacuated tube collectors are employed. In this case, the collector efficiency is approached as in Eq. (6) where ΔT_{avg} is the temperature difference between the temperature in the solar panel taken as an average value and the environmental temperature [21,22]:

$$\eta_c = 0.82 - \frac{2.198 \cdot \Delta T_{avg}}{I_s} \quad (6)$$

The coefficient of performance was estimated as in Eq. 7. The solar coefficient of performance was approached as in Eq. 8:

$$COP = \frac{\dot{Q}_e}{\dot{Q}_d + \dot{W}_p} \quad (7)$$

$$SCOP = \frac{\dot{Q}_e}{\dot{Q}_s} \quad (8)$$

In the previous Equations, \dot{Q}_e refers to the evaporator heat flow and \dot{Q}_d is the energy input to the vapour generator. \dot{Q}_d includes the thermal energy coming from the solar panel and gas

burner. \dot{W}_p refers to the pump power input. Eq. 9 indicates the relation between the cooling effect produced by the absorption units and that produced by the mechanical compression chillers:

$$\text{Share of cooling (\%)} = \frac{\dot{Q}_e}{\dot{Q}_{e \text{ comp}}} \cdot 100 \quad (9)$$

The Equation Engineering Solver (EES) software was used in this study. Thermodynamic and transport properties of the $\text{NH}_3/\text{LiNO}_3$ were obtained from [23–25]. Enthalpies were estimated following the methodologies reported in [26,27].

Regarding the environmental analysis, the environmental impact of natural gas is estimated by means of the $\text{CO}_2 \text{ eq}$ emissions as in Eq. 10 [28]:

$$\text{CO}_2 \text{ eq} = (\dot{m}_{\text{GN}} \cdot C_{em \text{ GN}} + \dot{W}_p \cdot C_{em \text{ el}}) \cdot \text{time}_{\text{use}} \quad (10)$$

Where $C_{em \text{ el}}$ is the emission coefficient in kg of CO_2 due to the electrical energy consumed (kWh). Its value was set to 0.181 kg CO_2 / kWh. $C_{em \text{ GN}}$ is the CO_2 emission coefficient in kg for the natural gas consumption (Nm^3). Its value was set to 2.15 kg CO_2 / Nm^3 [28].

The $\text{CO}_2 \text{ eq}$ emissions related to the electrical energy consumed by the conventional chillers were considered as follows [28]:

$$\text{CO}_2 \text{ eq} = (\dot{Q}_{e \text{ comp}} / (\text{COP}_{\text{comp}}) \cdot \text{time}_{\text{use}} \cdot C_{em \text{ el}} \quad (11)$$

The $\text{CO}_2 \text{ eq}$ emissions related to the electrical energy consumed by the absorption chillers were considered as follows [28]. It is important to highlight that based on the cooling flow covered by the absorption chillers, the remaining cooling need is covered by the conventional chillers, and therefore, this electrical energy consumption is also considered:

$$\text{CO}_2 \text{ eq} = (\dot{Q}_{e \text{ abs}} / \text{COP} + \dot{W}_p) \cdot \text{time}_{\text{use}} \cdot C_{em \text{ el}} \quad (12)$$

The $\text{CO}_2 \text{ eq}$ emissions related to the electrical energy and natural gas consumed by absorption chillers were considered as follows [28]:

$$\text{CO}_2 \text{ eq} = ((\dot{Q}_{e \text{ comp}} / \text{COP} + \dot{W}_p) \cdot C_{em \text{ el}} + \dot{m}_{\text{GN}} \cdot C_{em \text{ GN}}) \cdot \text{time}_{\text{use}} \quad (13)$$

3. Results and discussion

3.1 Electrical energy measurements

Fig. 3 presents the electrical energy consumed by the mechanical vapour compressor chillers in the malls selected. These malls are usually opened around 13 h every day which corresponds to the operation time of the chillers. From Fig. 3 can be noted that M3 is the mall with the highest electrical energy consumption, followed by M1, and M2. M3 is also the mall with the smallest available roof area (See Table 1). In general, the chillers electrical energy consumption in M3

varied between 330 kW to 400 kW for most of the day. In the case of M1, it varied between 267 kW and 330 kW, while for M2, it varied between 200 kW and 240 kW. Each mall showed variations in the chiller’s electrical energy consumption due to the normal activities that take place in the malls.

To estimate the cooling effect provided by the conventional chillers, a COP of 3 was considered [9,29]. Therefore, it can be said that the cooling load of the Malls 1, 2 and 3 were up to 1001 kW, 735 kW, and 1187 kW, respectively. The estimated values were used as the thermal loads required to be provided by the solar and/or gas driven absorption units in each corresponding mall.

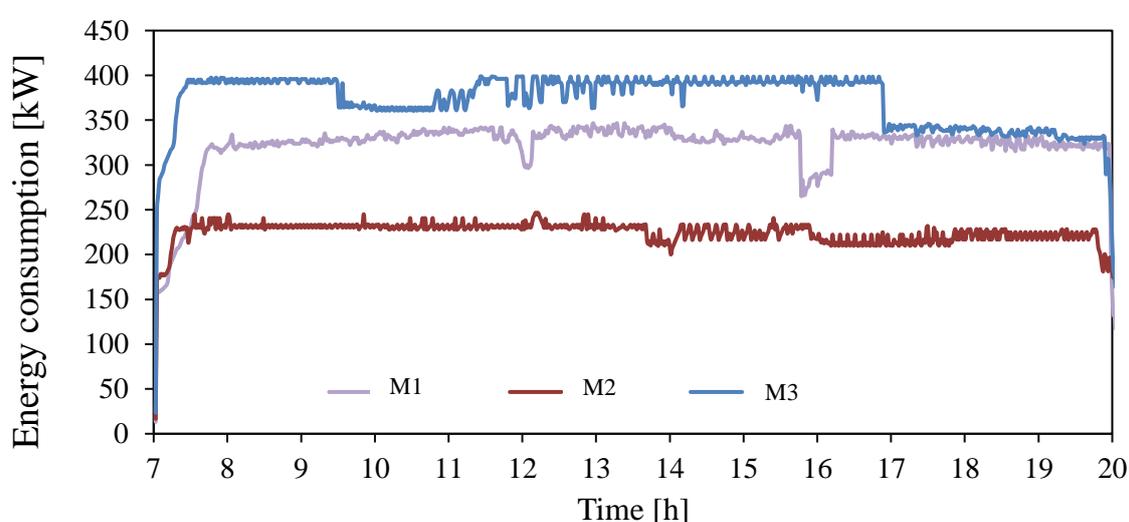


Fig 3. Power measurements of chillers in M1, M2, and M3 [15].

3.2 Energy performance

The development of the model considered as a reference base the external currents operating conditions and nominal cooling capacities of the absorption chillers from the brand Thermax which use water/LiBr as working fluid [30]. This means that the absorption chiller model, which uses NH₃/LiNO₃ as working fluid, was also developed for a nominal cooling capacity of 352 kW, while the temperature of the chilled water was set to 6.7 °C. In the case of the heat dissipation water for the condensation and absorption processes, it was set to 29.4 °C. The pressurized water to drive the chiller was set to 90.6 °C [30]. Regarding the external currents water flows in each component, they were set considering those defined by Thermax [30]. Once defined the external currents conditions, the internal solution mass flow leaving the absorber was estimated as 4.791 kg/s. The mass flow obtained is around 2.8 times higher than that with water/LiBr, which is reasonable considering the differences in the transport properties. For higher cooling loads, the solution mass flow increases proportionally. Also, reasonable effectiveness values were defined for the main components of the chiller as shown in Table 2.

Table 2. Effectiveness values [19,31].

Component	evaporator	absorber	solution IHX	generator	condenser
Effectiveness	0.85	0.46	0.80	0.50	0.68

Considering the estimated thermal loads, it is convenient to define the number of absorption chillers and the area of the solar panels required in each mall. The relation of Eq. 4 to 6 allows for estimating the evacuated tubes collector area. For one absorption chiller of 352 kW, the area of solar panels required would be 1105 m². This area contrast with the area required to drive an absorption chiller using water/LiBr at the same operating conditions (around 840 m²) [15]. This difference is due to the higher COP of the water/LiBr chiller (around 0.78, nominal) in comparison to that of the NH₃/LiNO₃.

Regarding the number of absorption chillers required, the maximum cooling load provided by the conventional chillers was up to 1001 kW in M1, then, three absorption chillers could be used at a nominal capacity. However, if the chilled water temperature is increased to 12 °C, the cooling capacity of the chiller can increase to 406 kW, which indicates that two solar/gas absorption chillers could be used in M1. Following the same consideration, two absorption chillers are also required in M2. In M3, the limiting factor is the available roof area so only one solar/gas driven absorption chiller can be used. If flat panels were to be considered in these case studies, the total area required to drive one absorption chiller could increase by 1.68 times. Considering that each solar/gas driven absorption chiller could operate at steady-state conditions at a chilled water temperature of 12 °C, the COP and SCOP of the absorption chillers were estimated around 0.60-0.64 and 0.41-0.44, respectively. Table 3 presents a summary of the number of chillers and total evacuated solar panel area used in each case.

Table 3. Absorption chillers and solar panel area used in each case.

Mall	Chiller units	Area of solar panels (m ²)
M1	2	2210
M2	2	2210
M3	1	1105

Fig. 4 shows the hourly mean radiation in March and November vs the heat transfer flow in the generator (Q_d), evaporator (Q_e), and the solar heat flow to the solar panel (Q_s) for one chiller. The horizontal lines represent the nominal cooling capacity of the chiller at an evaporation temperature of 12 °C and 6 °C. In the case of the heat dissipation water temperature, this was varied considering the wet bulb temperature of the environmental temperature. It is to be highlighted that the available roof area in each mall limits the possibility of using a larger solar

panel area in order to collect and store a higher amount of thermal energy.

Fig. 4,a,b shows that in March (the month with the highest mean radiation), the solar thermal energy collected is enough to drive the chiller at full load to produce chilled water at 12 °C, between 10:00 h and 13:00 h. This period is one hour wider for an evaporation temperature of 6 °C given the lower nominal cooling load. For the rest of the time, the natural gas burner must be on. In the case of November (the month with the lowest mean radiation), the solar thermal energy is not enough to drive the chiller at full load at any time (see Fig. 4,c,d). Therefore, the natural gas burner operation is required to provide additional energy and make the absorption chiller operate at full load. In general, solar energy can provide around 50% of the total energy required to drive the chiller during a working day in March. This value is reduced to 40% in November. Additionally, it is worth mentioning that at the operating conditions established, no risk of crystallization of the NH₃/LiNO₃ solution was detected. This is one of the advantages of the mixture NH₃/LiNO₃ in comparison to the mixture of water/LiBr.

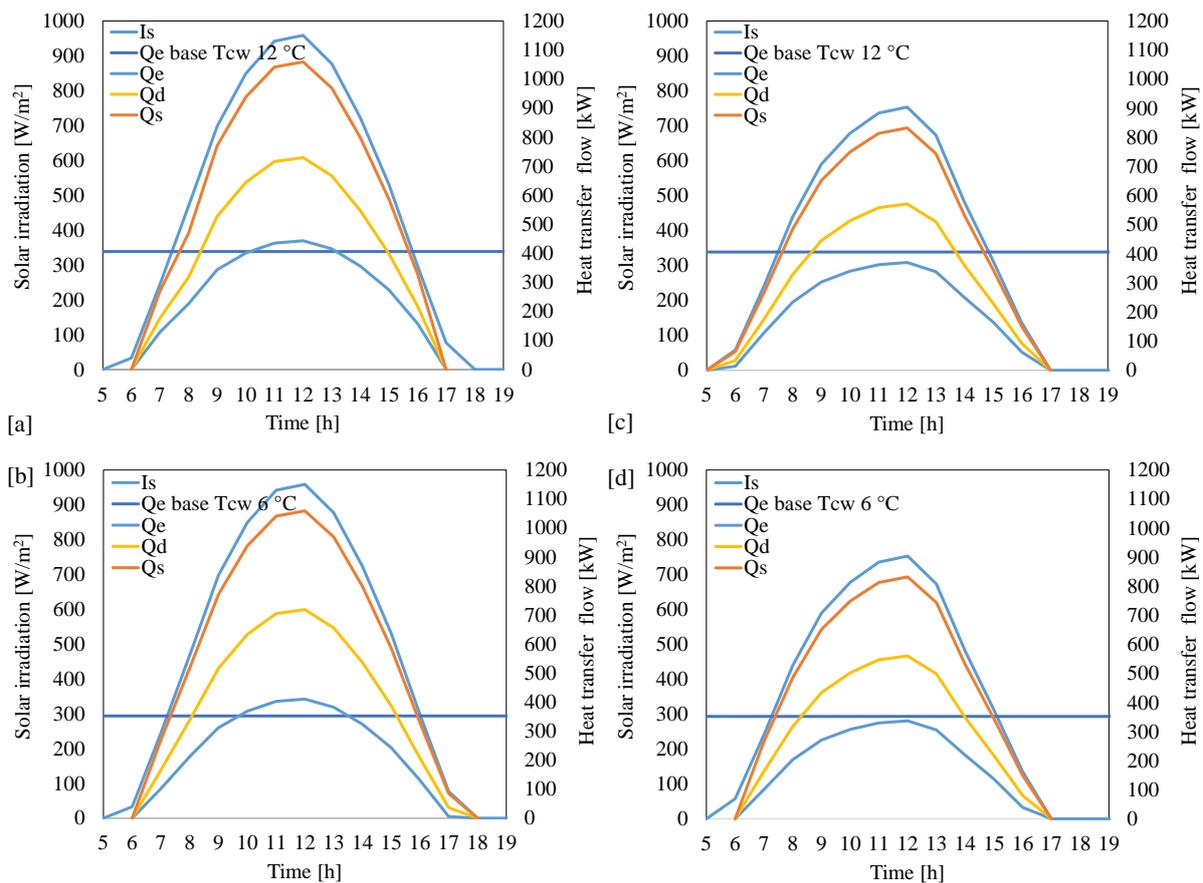


Fig 4. Hourly mean solar irradiation vs heat transfer flow in one chiller for [a,b] March, and [c,d] November.

3.3 Share of cooling demand analysis

This subsection presents the share of cooling that could be provided by the absorption chillers in M1, M2, and M3. The chilled water temperature was established as 12 °C in this study.

Moreover, the low heating value for the natural gas was set to 47.04 MJ/kg [31]. Fig. 5 shows 4 case studies for each mall. 1 (comp) represents the share of cooling provided by using only mechanical compression chillers, 2 (solar) represents the share of cooling provided by using only solar-driven absorption chillers, 3 (comb) represents the share of cooling provided by solar/gas driven absorption units, and 4 (gas) represents the share of cooling provided by using only gas driven absorption chillers. Accordingly, Fig. 5 shows that in most of the cases (except M2, cases 2 and 3), the absorption chillers do not cover all the required cooling capacity, therefore, the mechanical compression chillers are needed to complete the total cooling load. In M1, M2, and M3, the solar-driven absorption chiller is the configuration that requires the major support of the mechanical compression systems. This is expected since solar energy is not available during the whole malls service time. On the other hand, the solar/gas and gas driven absorption chillers could provide a major share of cooling since they could be operating a full load regardless of the solar energy availability, being the natural gas the main energy input to the chillers. In M3,4, since the solar panels are not used in this case, the number of gas driven absorption chillers can be increased in one unit to cover a major share of cooling and then reduce the share of cooling provided by the mechanical compression chillers. In general, the solar/gas driven absorption chillers could provide between 55% and 100% of the cooling load of M1 and M2. In the case of M3, this share is reduced given the limitations in the available roof area of the building.

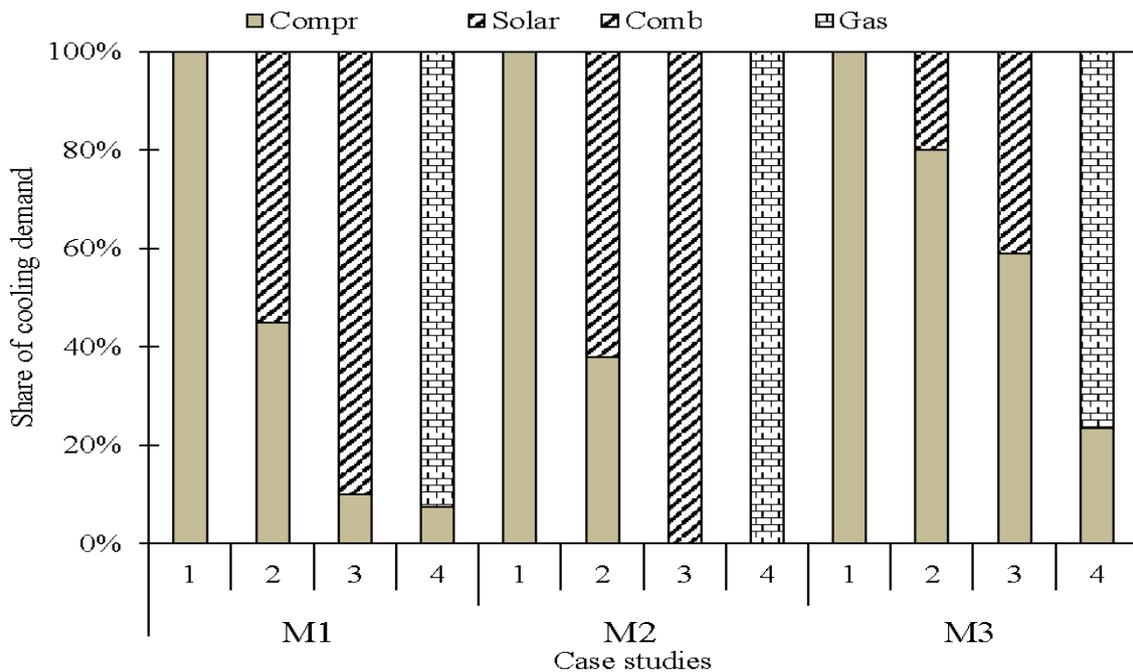


Fig 5. Share of the cooling provided by the NH₃/LiNO₃ absorption chillers and conventional compression units for each case study

3.4 Environmental analysis

Fig. 6 shows the same configuration cases as in Fig. 5, but the results are focused to present the equivalent CO₂ emissions provided by each case study. The gas emissions values presented are related to the use of electrical energy and natural gas in the corresponding configuration. It also considers the share of cooling provided by the mechanical compression chillers. Fig. 5 shows that the gas driven absorption chiller was the configuration with the highest gas emissions because of the higher support required by the mechanical compression systems. On the other hand, the solar/gas driven absorption chillers provided the lowest gas emissions in M1 and M2 given the combination of solar energy and that from natural gas, and the low share of cooling provided by the mechanical compression chillers. In the case of M3, the lowest gas emissions could be provided by the gas driven absorption chiller since in this mall 2 units can be installed. In the case of the solar/gas driven absorption chiller, only one unit can be installed given the limitations in the available roof area, so a major share of cooling by the mechanical compression systems is required to reach the cooling objective, therefore, a higher gas emission is provided in comparison to the gas driven chillers. If the contribution of the mechanical compression chillers is not considered, the gas/driven absorption chiller appears as the best option to reduce the gas emissions impact and provide the highest share of cooling possible for most of the time. Finally, results evidence that the energy and environmental impact of the absorption system relay on the system configuration, available roof area for solar applications, cooling load covered, and building characteristics. Therefore, the feasibility of one case study must be evaluated independently so results reported in the literature from studies with similar approaches cannot be generalized.

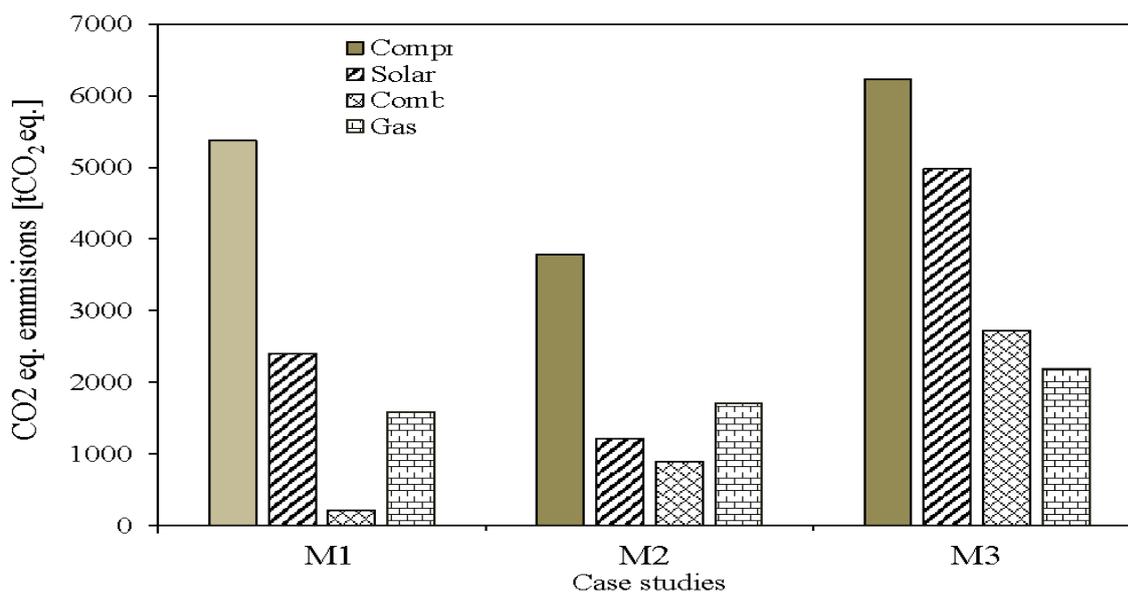


Fig 6. Estimated CO₂ eq. emissions for each case study using the NH₃/LiNO₃ absorption chillers

4. Conclusion

In this research, the technical and environmental feasibility of an $\text{NH}_3/\text{LiNO}_3$ absorption chiller using natural gas and solar energy as energy sources was presented. Thermodynamic simulations were carried out to study the absorption chillers performance for three malls located in Barranquilla, Colombia. The following conclusions can be drawn from the present study:

- The solar energy could drive the absorption chillers at full load conditions only for around 3 hours during the month of the highest irradiation. Therefore, the energy input from natural gas was required to operate the absorption chillers at their full load conditions. Solar energy could provide around 40% and 50% of the total energy required to drive the absorption chillers on a service day. This value could be increased by adding a higher solar panel area, however, the available roof area for solar equipment is a limiting factor. Moreover, the COP and SCOP of the absorption chillers were found up to 0.64 and 0.44, respectively.
- The solar/gas driven absorption chillers could cover up to 83%, 100%, and 35% of the peak cooling requirements in malls 1, 2, and 3, respectively. The area available for solar collectors in mall 3 did not allow using more than one solar/gas assisted absorption chiller to fully cover the cooling requirements. Furthermore, the solar/gas driven absorption chiller appears as the best option in terms of an environmental impact considering the specific characteristics of the mall under study and the cooling load covered.
- The available roof area for solar panels in mall buildings is a limiting factor for solar absorption cooling, especially in those with a vertical architecture.

5. Acknowledgements

The authors acknowledge the Universidad de la Costa for supporting the present research through the INDEX CONV-13-2018.

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Nomenclature

A_c : Area of collectors [m^2]
 C_p : Specific heat of water [$kJ/kg \cdot K$]
 h : Specific enthalpy [kJ/kg]
 I_s : Solar irradiation [W/m^2]
 \dot{m} : Mass flow rate [kg/s]
 P : Pressure [kPa]
 \dot{Q} : Heat transfer flow [kW]
 T : Temperature [$^{\circ}C$]
 v : Specific volume [m^3/kg]
 \dot{W} : Power [kW]
 X : Concentration of $LiNO_3$ in solution
COP: Coefficient of performance
SCOP: Solar Coefficient of performance

Subscripts

a: Absorber
abs: Absorption chiller
Amb: Environment
c: Solar collector
cond: Condenser
comp: Mechanical compression chiller
d: Desorber
e: Evaporator
GN: natural gas
s: Solar
shx: Heat exchanger
p: Pump
u: Useful

Greek letters

ϵ : effectiveness
 η_o : Solar collector Efficiency
 η_{GN} : Gas burner Overall efficiency