

Article

Characterization and Monitoring of a Self-Constructible Photovoltaic-Based Refrigerator

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Abstract: At present, in the developing countries, the power grids do not reach every small town and, even when they do, they are not reliable. For this reason, preservation of food and medicines is a sensitive issue for a large number of people. This work introduces the characterization of a new, self-constructible refrigerator, powered by photovoltaic (PV) energy, suitable for the preservation of food in rural villages, with a specific focus on Africa's tropical-equatorial belt. A thermally-insulated envelope, equipped with thermal energy storage, has been designed to be built on-site, by using local materials, and assembled with technical components (direct current (DC) compressor, heat exchangers, photovoltaic module). The monitoring activity presented in this work was carried out on a first prototype of the system and enabled the validation of numerical simulations, aimed at assessing its appropriate operability under laboratory conditions. In particular, the work demonstrates that the basic module of the refrigerator, with a net capacity of 250 L and an ice storage of 12.5 kg, is able to maintain an internal temperature below 10 °C also in critical boundary conditions.

Keywords: solar refrigeration; photovoltaic (PV) refrigerator; food preservation; self-construction; low-cost refrigerators

1. Introduction

In the rural areas of developing countries, one of the most critical issues is the preservation of food [1]. Settlements located in such areas are typically not served by the grid or—if they are—electricity is available only for a few hours per day [2,3]. Several stand-alone solutions for food preservation have been developed with the aim to solve this problem; some of them use solar thermal collectors as heat sources [4–7] working on sorption refrigeration (adsorption or absorption), others are powered by fossil fuels, such as kerosene or liquid petroleum gas (LPG) [8]. Since, in recent years, a strong reduction occurred in the manufacturing cost of photovoltaic (PV) modules [9,10] and small-size direct current (DC) compressors are commercially available at reasonable prices, the development of PV refrigerators can be considered affordable compared to other solutions [11].

To this extent, some solutions can be found on the market [12–14], although the available products are usually costly [15] for different reasons. For example, the use of batteries for energy storage in the case of electric refrigeration, and the use of complex solutions involving the absorption cycle in the case of solar thermal application—combined with the fact that the components are usually manufactured in developed countries and shipped as ready-to-use systems—substantially increases the final market cost [16,17].

In general, a systematic comparison of the main technologies used for food refrigeration in developing countries and documented in literature is presented in the Table 1 [11,15–22]. The information refers to small/medium size refrigerators for food/vaccine preservation and the reference costs include the accessories required to operate the system (e.g., PV modules, burner, etc.).

Table 1. Comparison of the main technologies used for food refrigeration in developing countries. PV: photovoltaic.

Type of Refrigerator	Initial Cost (€)	Operative Cost	Environmental Impact	Reliability	Need for Specialized Personnel
Vapour compression connected to the electricity grid	500–1000	Cost of electricity, maintenance costs	Linked to the production of electricity	High	Only for maintenance
Vapour compression powered by fossil fuels	700–1500	Fuel cost, cost of fuel transport, maintenance costs	CO ₂ , NO _x , sulphur oxides, and particulate emissions	Medium	Only for maintenance
Vapour compression powered by PV modules with electric storage	2000–4000	Batteries cost, maintenance costs	Battery disposal	Medium	Only for maintenance
Vapour compression powered by PV modules with thermal storage	2000–6000	Maintenance costs	None	High	Only for maintenance
Absorption powered by fossil fuels	1000–2500	Fuel cost, cost of fuel transport, maintenance costs	CO ₂ , NO _x , sulphur oxides, and particulate emissions	Medium	For maintenance and during installation
Absorption powered by solar collectors	1500–3500	Maintenance costs	None	High	For maintenance and during installation
Absorption powered by biomass	2000–3500	Fuel cost, cost of fuel transport, maintenance costs	CO ₂ , NO _x , sulphur oxides, and particulate emissions	High	For maintenance and during installation
Adsorption powered by solar energy	1500–4500	Maintenance costs	None	High	For maintenance and during installation

For such reasons, this work presents the characterization of an innovative solution, which was preliminarily developed within a wider research activity [23] aimed at the application of PV refrigeration in Africa's tropical-equatorial belt. In detail, the system proposed is modular, battery-free, and can be self-constructed on site using local materials and manpower. Moreover, the use of local materials, combined with technological parts manufactured in developed countries, could produce the following advantages:

- Low market price compared with products currently available in Africa, since shipping and manufacturing costs are strongly reduced; the objective is to maintain the final cost of the system below 1000 € for a standard module with 250 L of net capacity;
- High availability; the system can be easily repaired by local people; in addition, full operability without solar power is ensured by the thermal storage; the aim of the research is to ensure the air temperature inside the fridge is always below 10 °C even after 48 h without solar radiation;
- Reduction in the environmental impact during and at the end of the working life; in the proposed system all of the components can be re-used or recycled;
- Active participation of the population during the manufacturing phase, in order to achieve economic benefits and improve technical knowledge. The entire system can be assembled locally without specific expertise or technical equipment. Importing ready-to-use products from developed countries has, in fact, a negative impact on the local technical knowledge.

In addition, the careful selection of the materials for the structure of the refrigerator and the use of specific guidelines on food preservation will guarantee high hygienic performance.

The general aim of the research is, thus, to realize a modular kit, in order to economically and effectively meet the needs of different contexts, taking into specific consideration Africa's tropical

regions. In fact, such areas are the most investigated for the distribution of the technology proposed, and—in particular—a village located in Northwest Cameroon was selected as a case study for the final application. In that village, in the last phase of the research project, the refrigerator will be installed and monitored in its operational environment.

Within such a context, this work describes the first experimental validation of the technology, carried out by means of the construction of a prototype of the system, realized at the Politecnico di Milano (Milan, Italy), and the subsequent testing and monitoring activity, which started in September 2015, as described below.

2. System Description

The PV refrigerator is based on a compression cycle and is composed of the following elements: A PV module, a DC compressor with a forced-air-cooled condenser, a roll-bond evaporator, a thermally-insulated envelope containing ice storage, an expansion valve, and the control system (Figure 1). The main technical specifications of the components are reported in Table 2.

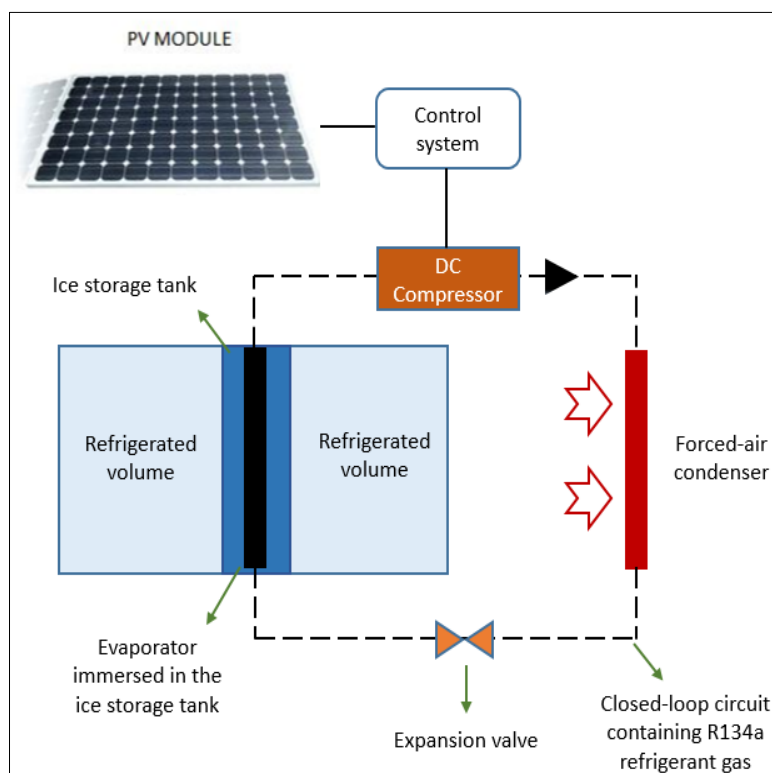


Figure 1. General structure of the system.

Table 2. Technical specification.

Component	Manufacturer	Specification
DC-powered compressor unit	Secop	BD35F–80 W–R134a
Condenser	Vitrifrigio	ND350R2-V
PV module	Sharp	ND-R240A5–240 W _p
Evaporator	Vitrifrigio	PT13
Refrigerated volume	self-constructible	Wood panels and thermal insulation
Ice storage	self-constructible	Aluminium box

The evaporator is placed inside a mass of water (ice thermal storage). Such water, once frozen, ensures an optimal storage temperature of the refrigerated volume even without solar radiation.

The prototype version of the PV-based refrigerator realized at the Politecnico di Milano is a wooden box with a net capacity of approximately 295 L and an internal dimension of the refrigerated volume equal to 65 cm $\times$ 70 cm $\times$ 65 cm ($L \times H \times W$).

For the realization of the envelope, materials available in Italy were selected, ensuring the same thermo-physical properties of the ones that can be found in Cameroon. The volume of the refrigerator and the ice storage are the ones designed.

In detail, the envelope of the prototype refrigerator is composed of:

- Internal layer: oriented strand board (OSB) panel (1 cm thick);
- Thermal insulation: wood-fiber insulation (6 cm thick) and compressed straw (21 cm thick);
- External layer: OSB panel (1 cm thick).

In addition, the box has an internal waterproof layer made of a plastic protecting coating, which is applied like a varnish; the bottom of the refrigerated volume is, instead, covered by an aluminum foil (1 mm). The box is lifted off the ground in order for it to stay dry and to ensure a slight gradient that allows internal condensation water to leave through the specially-designed drain channel.

Inside the refrigerated volume, an aluminum tank is placed, to be used as an ice storage with dimension 45 cm $\times$ 62 cm $\times$ 5 cm ($L \times H \times W$); the capacity of the tank is about 15 L and the volume of the contained water was set equal to 12.5 L, in order to provide roughly 72 h of proper operability without solar power.

The box is equipped with a hinged cover on the top, made with the same materials. Figure 2 shows a vertical and a horizontal section of the prototype while in Figure 3 the external and internal views of the system used for testing purposes are shown.

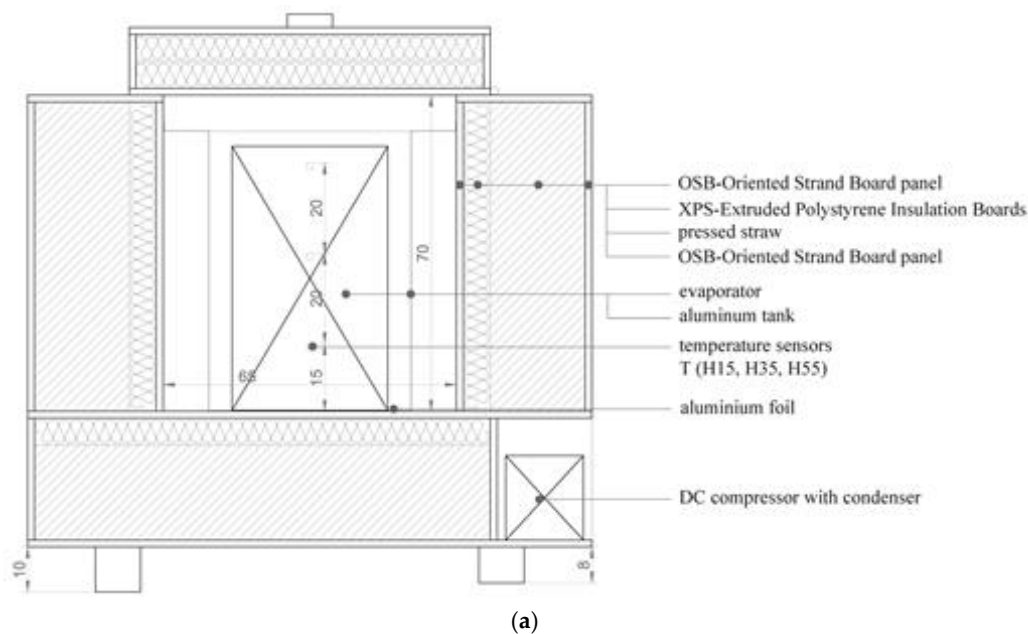


Figure 2. Cont.

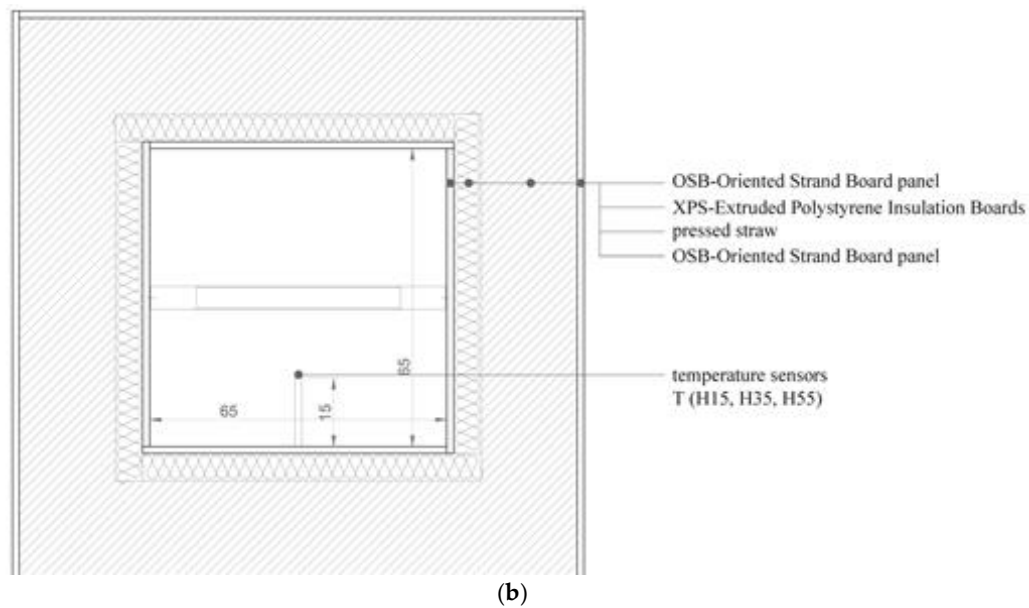


Figure 2. (a) Vertical and (b) horizontal sections of the refrigerator (dimensions in cm).

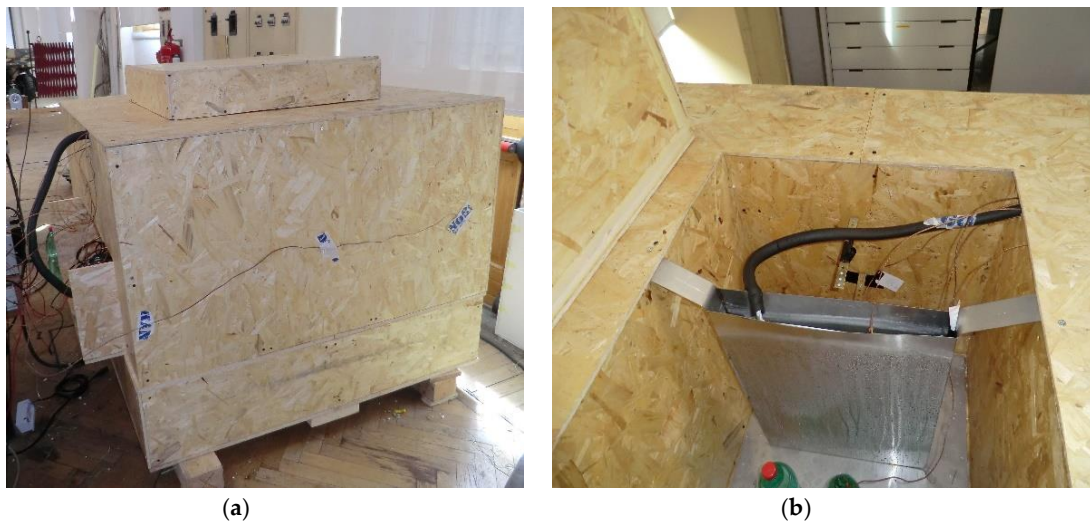


Figure 3. (a) External and (b) internal views of the first prototype of the system used for testing purposes.

3. Energy Model

To assess the energy performance of the system, a simplified energy model was developed. The model estimates the mean air temperature inside the refrigerator and the temperature of the water in the storage container, on the basis of the boundary conditions, assuming the initial conditions are known. The model is based on the energy balances of the internal air and the whole refrigeration system, solved at any desired time step. The thermal inertia of the envelope is considered negligible, since it is realized with lightweight insulation materials.

In addition, in order to define the energy balance of the refrigerator, the items introduced in the fridge to be cooled—such as food and beverages—are considered to be, at each time step, at the same temperature of the surrounding air. It is, thus, assumed that an instantaneous heat transfer between the internal air and the food/beverage occurs. Otherwise, the specific heat transfer coefficient of the food/beverage placed inside the fridge should be known for every kind and size of the same.

This simplification is reasonable insofar that the thermal conductivity and the heat transfer area of the food/beverages inside the fridge are high (e.g., aluminum cans, small fruits rich of water, etc.) and the transient condition during which the load is cooled at the set-point temperature is not considered relevant for the specific scope of this work. The aim of the model is, in fact, to assess the proper functioning of the system (i.e., to maintain the internal set-point temperature in steady state conditions), overlooking the time needed to cool the load.

In detail, the initial/boundary conditions which have to be known are the following:

- The external air temperature at each time step, $T_{ext,n}$;
- The average irradiance on the module surface at a certain time step I_n , in W/m^2 ;
- The internal air temperature and the water temperature in the storage ($T_{int,0}$ and $T_{w,0}$) at the initial time t_0 ;
- The specific heat, cp_{load} , and the mass, m_{load} , of the items placed inside the refrigerator; and
- If there are openings of the refrigerator during a certain time interval.

We assumed that all the quantities of energy removed from the insulated volume of the refrigerator and transferred to the external environment are positive values.

3.1. Thermal Balance of Internal Air

The internal air temperature $T_{int,1}$ and the water temperature inside the energy storage $T_{w,1}$ are calculated on the basis of the thermal balance of the energy storage, defined in the following equation:

$$Q_{sto,0-1} = Q_{env,0-1} + Q_{load,0-1} + Q_{air,0-1} + Q_{o,0-1} \quad (1)$$

which means that, during a certain time interval $\Delta t = t_1 - t_0$, the amount of energy released by the thermal storage $Q_{sto,0-1}$ in the air surrounding the refrigerator is equal to the sum of the heat loss through the envelope $Q_{env,0-1}$, the thermal energy adsorbed by the thermal load inside the refrigerator $Q_{load,0-1}$, the thermal energy absorbed by the air inside the refrigerator $Q_{air,0-1}$, and the thermal energy eventually dissipated during the openings of the cover $Q_{o,0-1}$.

By making explicit all the terms, the equation becomes:

$$\begin{aligned} & U_{sto} \times S_{sto} \times \left(\frac{T_{int,0} + T_{int,1}}{2} - \frac{T_{w,0} + T_{w,1}}{2} \right) \times \Delta t \\ &= U_{env} \times S_{env} \times \left(\frac{T_{ext,0} + T_{ext,1}}{2} - \frac{T_{int,0} + T_{int,1}}{2} \right) \times \Delta t + m_{load} \\ &\times cp_{load} \times (T_{int,0} - T_{int,1}) + m_{air} \\ &\times cp_{air} \times (T_{int,0} - T_{int,1}) + m_{air} \times cp_{air} \times \left(\frac{T_{ext,0} + T_{ext,1}}{2} - \frac{T_{int,0} + T_{int,1}}{2} \right) \times k \end{aligned} \quad (2)$$

where k is conservatively set equal to 0.5 in case of openings at a certain time step or 0 if there are no openings. The two unknown quantities are $T_{int,1}$ and $T_{w,1}$, which can be determined by also considering the thermal balance of the whole system, as described below.

3.2. Thermal Balance of the System

The global thermal balance of the refrigerator can be described with the following relation:

$$Q_{eva,0-1} = Q_{env,0-1} + Q_{load,0-1} + Q_{air,0-1} + Q_{o,0-1} + Q_{w,0-1} \quad (3)$$

which means that, during a certain time interval $\Delta t = t_1 - t_0$, the amount of energy removed by the evaporator in the thermal storage $Q_{eva,0-1}$ is equal to the sum of the thermal energy dispersed by the envelope of the fridge $Q_{env,0-1}$, the thermal energy adsorbed by the thermal load placed inside the fridge $Q_{load,0-1}$, the thermal energy absorbed by the internal air $Q_{air,0-1}$, the thermal energy dissipated during the openings $Q_{o,0-1}$, and the thermal energy absorbed by the water in the thermal storage $Q_{w,0-1}$.

By making explicit all the terms, the equation becomes:

$$\begin{aligned}
 Q_{eva,0-1} = & U_{env} \times S_{env} \times \left(\frac{T_{ext,0} + T_{ext,1}}{2} - \frac{T_{int,0} + T_{int,1}}{2} \right) \times \Delta t + m_{load} \times cp_{load} \\
 & \times (T_{int,0} - T_{int,1}) + m_{air} \times cp_{air} \times (T_{int,0} - T_{int,1}) + m_{air} \times cp_{air} \\
 & \times \left(\frac{T_{ext,0} + T_{ext,1}}{2} - \frac{T_{int,0} + T_{int,1}}{2} \right) \times k + m_{w,tot} \times c_w \\
 & \times (T_{w,0} - T_{w,1})
 \end{aligned} \quad (4)$$

In Equation (4), $Q_{eva,0-1}$, which is the thermal energy removed by the evaporator, is calculated according to the following relation:

$$Q_{eva,0-1} = COP_{0-1} \times E_{com,0-1} \quad (5)$$

where:

$$COP_{0-1} = COP_{rif,0-1} \times [(T_{ext,rif} + 20)/(T_{ext,0} + 20)] \times [(T_{ext,0} + 80)/(T_{ext,rif} + 80)] \quad (6)$$

and:

$$COP_{rif,0-1} = 0.0003 \times T_{eva,0-1}^2 + 0.0441 \times T_{eva,0-1} + 1.7993 \quad (7)$$

$T_{ext,rif}$ is equal to 32 °C, according to the temperature defined by the UNI EN 12900 [24], used by the manufacturer to determine the reference coefficient of performance (COP) of the compressor.

Equations (6) and (7) result from the interpolation of experimental data provided by the manufacturer of the compressor.

It must be noted that the mean temperature of the evaporator within the time step can be calculated as:

$$T_{eva,0-1} = \frac{(T_{w,0} + T_{w,1})}{2} - 5^\circ\text{C} \quad (8)$$

The condition in Equation (8) was determined experimentally, and it is different from the usual case of a roll-bond evaporator operating in air, when T_{eva} is usually 10–15 °C lower than the air temperature [25,26] due to a lower heat exchange coefficient of the plate.

The minimum power required to start the compressor is, in fact, equal to approximately 45 W and the average electric power required during the operation is 55 W, thus assuming a 250 W_p PV module, when I_n is higher than 250 W/m² the generated power is assumed sufficient to run the compressor [27]. Hence:

$$E_{com,0-1} = 55 \times \Delta t \text{ when } I_n \text{ is } > 250 \text{ W/m}^2, \text{ otherwise is } 0 \quad (9)$$

In detail, when the compressor runs, the mean value of 55 W is referred to 3000 revolutions per minute (RPM), a mean evaporating temperature of −5 °C, and a mean environmental temperature of 25 °C. Such conditions can be considered as a reference for the system. In addition to the 55 W needed by the compressor, a 5 W supplementary power for the condenser fan has to be considered.

It must be noted that, during the laboratory experimentation, the compressor was powered by a DC power supply according to a working profile which simulates the average hours of useful solar radiation in the selected application context (Cameroon). Thus, E_{com} is a measured quantity.

The only unknown variables in Equations (2) and (4) are $T_{w,1}$ and $T_{int,1}$, which can consequently be calculated combining the two equations. The process can be repeated to calculate the two temperatures at the following time steps, until $T_{w,n} = 0^\circ\text{C}$ is reached.

As the water in the storage starts to freeze, the unknown quantity becomes the water mass that solidifies during a certain time step, while the water temperature is considered constant during the

freezing process ($T_{w,n} = T_{w,n+1} = 0$ °C). Equation (4) must also include the latent load absorbed by the water during the freezing process, and becomes:

$$\begin{aligned} Q_{eva,n-n+1} = & U_{env} \times S_{env} \times \left(\frac{T_{ext,n} + T_{ext,n+1}}{2} - \frac{T_{int,n} + T_{int,n+1}}{2} \right) \times \Delta t + m_{load} \\ & \times c p_{load} \times (T_{int,n} - T_{int,n+1}) + m_{air} \times c p_{air} \times (T_{int,n} - T_{int,n+1}) \\ & + m_{air} \times c p_{air} \times \left(\frac{T_{ext,n} + T_{ext,n+1}}{2} - \frac{T_{int,n} + T_{int,n+1}}{2} \right) \times k \\ & + m_{w,ice,n} \times \lambda \end{aligned} \quad (10)$$

As in the previous situation, there are two unknown quantities ($T_{int,n+1}$ and $m_{w,ice,n}$), and two available Equations (2) and (10). Thus, the system can be solved and $T_{int,n+1}$ and $m_{w,ice,n}$ can be calculated.

This process has to be repeated until all of the water mass inside the tank is completely frozen ($m_{w,ice} = m_{w,tot}$). Subsequently, Equations (2) and (4) must be used to calculate the ice temperature, changing the value of the specific heat of the water and using that of the ice. When the compressor stops, the calculation process remains the same, assuming $Q_{eva,n} = 0$.

4. Instrumentation and Data Logging

The prototype developed was tested to validate the simulated thermal analysis. For this purpose, an ad hoc testbed was developed. The system allows testing the prototype in actual working conditions. In fact, it permits enforcing a desired power profile over time, thus simulating the behavior of the PV panel in the application context taken into account. The system allows to monitor electrical quantities, voltage, and current, as well as six temperatures.

The system was developed according to a virtual instrument (VI) approach by means of Labview Software (National Instruments, Austin, TX, USA) (Figure 4).

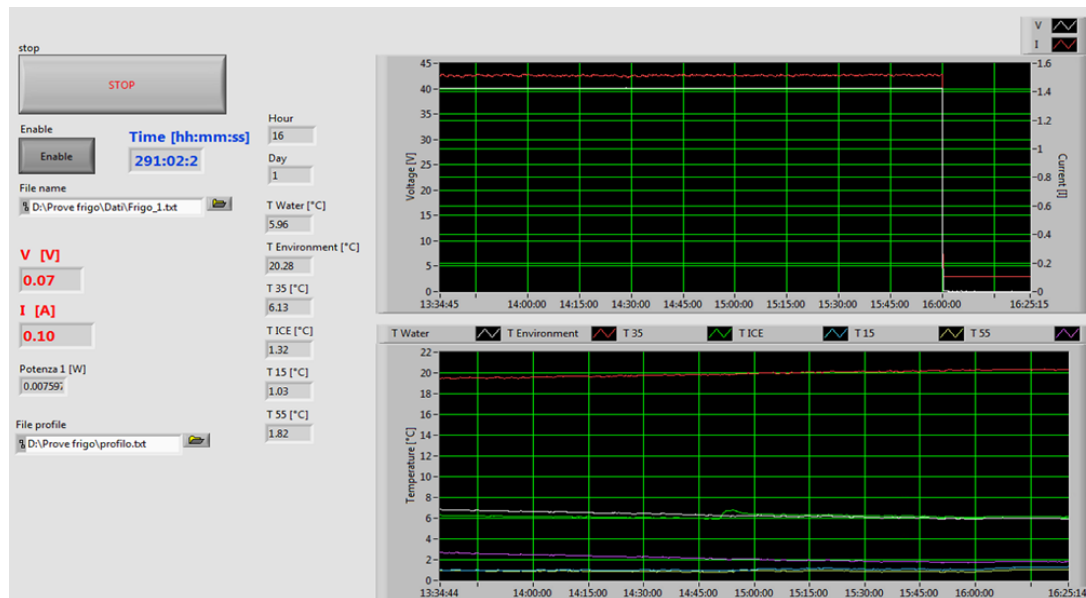


Figure 4. Front panel of the developed virtual instrument (VI).

The PV panel was simulated by means of a DC generator (Sorensen SGI 400-38, AMETEK Programmable Power, San Diego, CA, USA) remotely controlled through a General Purpose Interface Bus (GPIB) connection (Figure 5). The software provides the set point over the time to the generator. The voltage generated and the flowing current were transduced by means of a voltage divider and a Hall effect transducer. The signals were then acquired by means of 16-bit resolution ADCs (National

Instruments, Austin, TX, USA). A sampling frequency of 1280 Hz was used and the samples were averaged over 10 s.

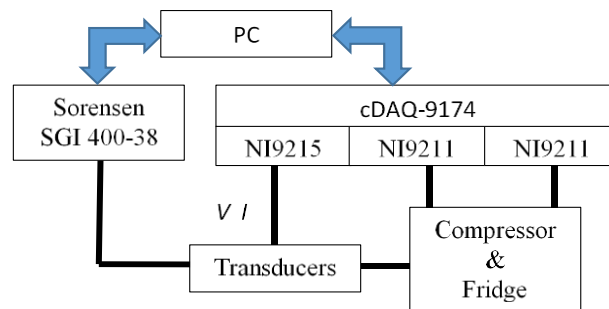


Figure 5. Experimental setup.

During the test, different temperatures were measured by means of thermocouples (k-type) and dedicated acquisition boards (NI9211). Temperatures were sampled at 1 Hz and the average of ten samples was considered. In particular, the following temperatures were monitored:

- Mean water temperature inside the storage (T_w);
- Temperature of the air in the environment in which the fridge is located (T_{ext});
- Temperature of the air inside the fridge 15 cm from the bottom ($T_{int,15}$);
- Temperature of the air inside the fridge 35 cm from the bottom ($T_{int,35}$);
- Temperature of the air inside the fridge 55 cm from the bottom ($T_{int,55}$);
- Mean temperature of the load inside the fridge (T_{load}).

The uncertainty of the temperature measurements is $0.4\text{ }^{\circ}\text{C}$ with a coverage factor $k = 2$.

5. Experimental Results and Model Validation

The simulation model was firstly run considering the reference parameters reported in Table 3, which correspond to the testing conditions used to monitor the prototype.

Table 3. Main reference parameters of the testing conditions.

	Parameter	Symbol	Value	Unit
General features	Air volume	V_{air}	0.295	m^3
	Air density	ρ_{air}	1.22	kg/m^3
	Air thermal capacity	cp_{air}	1006	$\text{J}/(\text{kg K})$
	Mass of the load	m_{load}	9	kg
	Load thermal capacity	cp_{load}	4186	$\text{J}/(\text{kg K})$
	Opening coefficient	k	0	-
	Time step	Δt	10	min
Thermal insulation	Internal surface of the envelope	S_{env}	2.665	m^2
	Thickness	-	0.3	m
Storage system	Average conductivity	-	0.114	W/mK
	Mass of water in tank	m_w	12.5	kg
	Water thermal capacity	c_w	4186	$\text{J}/(\text{kg K})$
	External area of the storage tank	S_{sto}	0.54	m^2
	Water latent heat of fusion	λ	333500	J/kg
	Heat loss coefficient of the storage tank	U_{sto}	6	$\text{W}/\text{m}^2\text{K}$

It must be specified that the heat loss coefficient of the storage tank and the thermal transmittance of the envelope were experimentally verified according to the following procedure: the water inside

the thermal storage was completely frozen and then the compressor was stopped until the mass of ice returned completely to a liquid state. Thus, the time needed for the melting process Δt is known. During the test, the temperature inside the thermal storage T_w , internal air temperature T_{int} , and external air temperature were monitored. Knowing the mass of the water inside the thermal storage $m_{w,tot}$, the heat loss area of the thermal storage tank S_{sto} , and the heat loss area of the external envelope S_{env} , it was possible to calculate the two heat transfer coefficients, U_{sto} and U_{env} .

The working profile of the compressor during a typical week was defined through the study of the solar radiation measured in Bangang, Cameroon, where the final prototype will be installed. In particular, it was observed that during sunny days there are, on average, 7–8 h of irradiation, sufficient to run the compressor (I_n is $>250 \text{ W/m}^2$), while during partially cloudy days there are 4–5 h of viable irradiation. In addition, very few cloudy days with irradiance below 250 W/m^2 , during which no operation of the compressor is possible, were observed.

According to the elaboration of weather information, the working profile reported in Table 4 was selected as a reference to test the refrigerator in laboratory conditions.

Table 4. Reference working profile.

Day	Condition	Start	Stop
Day 1	Sunny	09:00 a.m.	04:00 p.m.
Day 2	Partially cloudy	09:00 a.m.	01:00 p.m.
Day 3	Partially cloudy	09:00 a.m.	02:00 p.m.
Day 4	Sunny	09:00 a.m.	05:00 p.m.
Day 5	Cloudy	-	-
Day 6	Cloudy	-	-
Day 7	Sunny	09:00 a.m.	04:00 p.m.

The first experimental test to validate the theoretical model was started by inserting a thermal load, represented by 9 kg of water ($6 \times 1.5 \text{ L}$ plastic bottles), and running the system according to the pre-defined working profile.

The internal air temperature, calculated, was compared with the temperatures monitored, as shown in Figure 6.

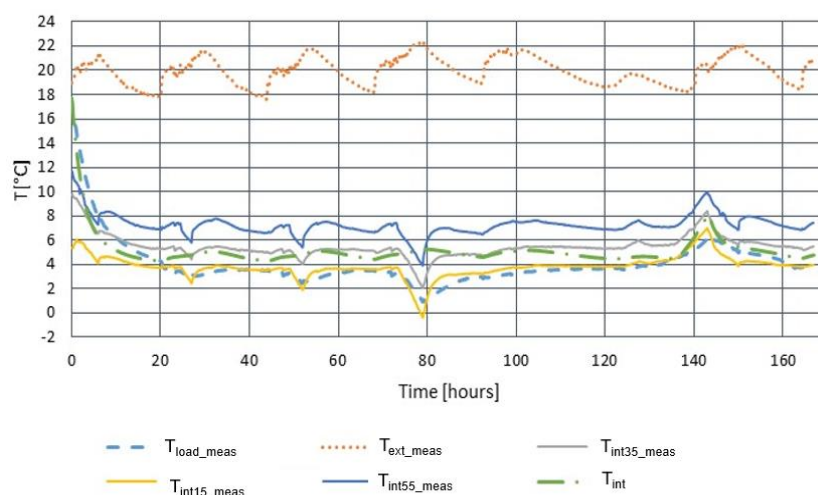


Figure 6. Comparison of the simulated average internal air temperature and the various temperatures measured during a typical working week.

As it can be observed, the simulated internal air temperature suits well with the temperature measured at the average height of the case (35 cm from bottom), which can be considered representative

of the mean air temperature inside the fridge. In detail, the average error between the calculated (T_{int}) and the measured temperatures (T_{int35}) is equal to 5.8%. The measured temperature of the load is, instead, on average equal to 4 °C and is very close to the temperature of the air measured 15 cm from the bottom, which is approximately 3.9 °C. The bottles of water were, in fact, placed directly on the bottom of the fridge, therefore they exchanged heat with the coldest part of the air. In general, in any working condition, the mean internal temperature is always around 5.5 °C and the maximum internal temperature, measured at the top of the fridge is, on average, below 8 °C. If the refrigerator remains closed and no additional load is introduced in the course of the two cloudy days, during which the compressor never works, the amount of ice is sufficient to maintain acceptable temperatures of the air and the internal load.

Figure 7 reports the comparison of the average water temperatures simulated and measured in the storage container during a typical working week, along with the amount of frozen water at every time step.

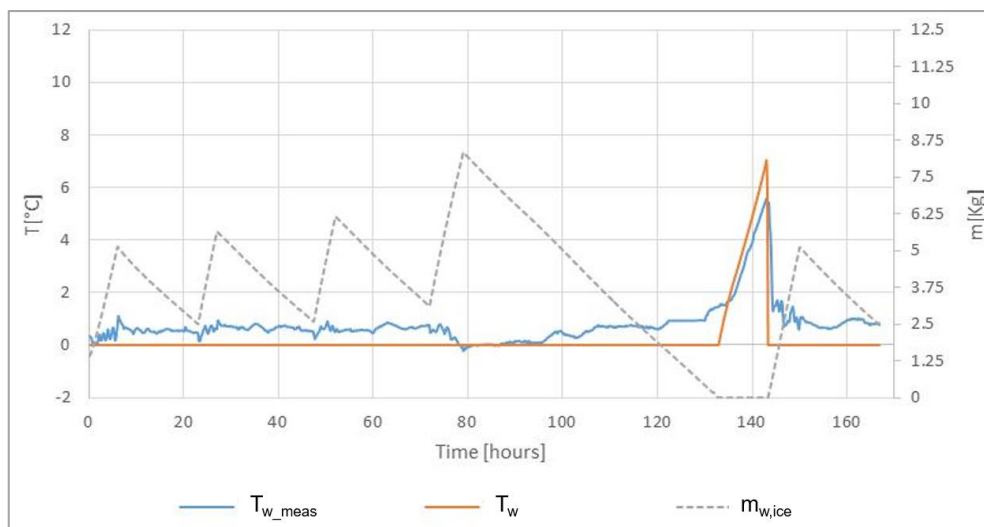


Figure 7. Comparison of the average water temperatures simulated and measured in the storage container during a typical working week, along with the amount of frozen water at every time step.

Subsequently, in order to further validate the simulation model, the test was carried out with different kinds of thermal loads placed daily inside the fridge, thus considering the fact that the system is actually to be used as a domestic refrigerator. The amount of new load inserted daily was estimated after the survey carried out in the Bangang village in Cameroon, supposing that the system is used by a group of 3–4 families.

The results obtained are reported in the following Table 5.

Table 5. Comparison of the measured calculated and the measured average air temperature after 12 h from the introduction of the load.

Day	Working Period of the Compressor (h)	New Load (kg)	Number of Openings	$T_{int,35_meas}$ after 12 h from the Introduction of the Load	T_{int} (°C) after 12 h from the Introduction of the Load
Day 1	7	9	8	5.21	4.96
Day 2	4	4.5	4	4.68	4.38
Day 3	5	4.5	4	4.87	4.64
Day 4	8	9	8	4.69	4.98
Day 5	0	9	8	6.24	5.61
Day 6	0	9	8	9.26	7.85
Day 7	7	0	0	5.49	5.52

By analyzing the data reported in Table 5, it is possible to state that, at the end of the transient condition during which the load is cooled at the set-point temperature:

- The average air temperature measured inside the refrigerator is always below 10 °C, even if a new thermal load is introduced in the course of the completely cloudy days, when the compressor is off. This result preliminarily confirms the adequate design of the system;
- The error between the temperature calculated and that measured is, on average, lower than $\pm 10\%$ and can be mainly attributed to the impossibility to estimate precisely the heat transfer coefficient of the thermal load inside the refrigerator. In any case, as stated before, since the objective of the model is to assess the proper functioning of the system (i.e., to maintain acceptable set-point temperatures in steady state conditions), the approximation introduced can be considered non-relevant.

Finally, it must be noted that, according to the present state of the research, an overall cost of the materials and components needed to realize the refrigerator described in the work is assumed to be around 500 € (excluding value-added tax, shipment costs, and labor). Such results, together with the energy performances above described, can ensure a large penetration of the technology since it can be considered immediately affordable for food retailers and small communities.

6. Conclusions

In this paper, a refrigerator kit—composed of a refrigerated volume, a storage unit, and a PV panel with its control unit—was experimentally tested. For the refrigerator designed, a simplified thermal model is proposed, in order to estimate its operability under the climate conditions of the targeted application context. In addition, a prototype was realized and a test bench was set up in the laboratory. The results of the experimental tests, performed in the lab, confirm the goodness of the thermal model in predicting the steady-state temperature of the food placed inside the refrigerated area as a function of the weather conditions. Experimental results also show that the low-cost and self-constructible refrigerator developed was adequately dimensioned in order to ensure a proper preservation of the food under the typical climate conditions of the Cameroon village that will be the final site of use. In detail, the work demonstrated that the basic module of the refrigerator, with a net capacity of 250 L and an ice storage of 12.5 kg is able to maintain internal temperature below 10 °C even after 48 h of no operation of the compressor and with a new thermal load introduced in the envelope. Finally, the validated model can be successfully used to predict the behavior of the system in different climatic and operational conditions.

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Nomenclature

I_n	Average irradiance on the PV module surface at a certain time step n (W/m^2)
$E_{com,n}$	Electric energy required by the compressor at a certain time step n (Wh)
$P_{com,n}$	Electric power required by the compressor at a certain time step n (W)
$Q_{eva,n}$	Thermal energy removed from the evaporator at a certain time step n (Wh)
$Q_{env,n}$	Thermal energy dissipated on the envelope at a certain time step n (Wh)
$Q_{sto,n}$	Thermal energy dissipated by the thermal storage at a certain time step n (Wh)
$Q_{load,n}$	Thermal energy needed to cool the load (food/beverages) inside the fridge at a certain time step n (Wh)
$Q_{w,n}$	Thermal energy absorbed by the water inside the thermal storage at a certain time step n (Wh)
$Q_{o,n}$	Thermal energy dissipated by the repeated opening of the fridge at a certain time step n (Wh)
COP_n	Coefficient of performance of the compressor at a certain time step n (dimensionless)
$COP_{n,rif}$	Reference coefficient of performance of the compressor at a certain time step n (dimensionless)
U_{env}	Thermal transmittance of the envelope of the fridge ($\text{W}/\text{m}^2 \cdot \text{K}$)
S_{env}	Internal surface of the envelope of the fridge (m^2)
U_{sto}	Heat transfer coefficient of the thermal storage tank ($\text{W}/\text{m}^2 \cdot \text{K}$)
S_{sto}	Heat loss surface of the thermal storage tank ($\text{W}/\text{m}^2 \cdot \text{K}$)
$T_{eva,n}$	Mean temperature of the evaporator at a certain time step n ($^{\circ}\text{C}$)
$T_{ext,n}$	Temperature of the air in the environment in which the fridge is located at a certain time step n ($^{\circ}\text{C}$)
$T_{int,n}$	Mean temperature of the air inside the fridge at a certain time step n ($^{\circ}\text{C}$)
$T_{w,n}$	Temperature of the water inside the thermal storage at a certain time step n ($^{\circ}\text{C}$)
$T_{load,n}$	Temperature of the load (food/beverage) at a certain time step n ($^{\circ}\text{C}$)
cp_{load}	Specific heat of the load (food/beverages) inside the fridge ($\text{Wh}/\text{kg} \cdot \text{K}$)
m_{load}	Mass of the load (food/beverages) inside the fridge (kg)
cp_{air}	Specific heat of the air inside the fridge ($\text{Wh}/\text{kg} \cdot \text{K}$)
V_{air}	Net volume of the internal air (m^3)
m_{air}	Mass of the air inside the fridge (kg)
$m_{w,tot}$	Total mass of the water inside the thermal storage (kg)
$m_{w,ice,n}$	Mass of the water inside the thermal storage which is solidified at a certain time step n (kg)
c_w	Specific heat of the water in the thermal storage ($\text{Wh}/\text{kg} \cdot \text{K}$)
λ	Latent heat of the water (Wh/kg)
k	Factor that considers the percentage of the net volume of air changed every time step, due to infiltration and door openings (dimensionless)
Δt	Length of the time step (h)
Note	The quantities measured have the same nomenclature of the ones calculated, but are identified by the subscript _{meas}

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