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Investigation of the Performance of a Solar Driven Refrigerator for Post-Harvest Crops in Hot Arid Remote Areas

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Abstract

There are many agriculture crops that are being cultivated in Egypt reclaimed deserts at hot arid remote areas with lack or no electrical grid connection. Consequently, to deliver such crops at high product quality, after harvest to the consumers, the transport storage environment need to be at appropriate temperature and humidity levels to maintain highest product quality to the end user. Therefore for multiple crops carriage, a transportable multi-store solar driven refrigeration system is suggested to be used. In this study, design, modeling and performance investigation of a 15 ft^3 transportable solar driven DC refrigeration system to be utilized for postharvest handling of crops has been carried out at Alexandria, Egypt. There are two compartments with different working temperatures of 5 and 0°C to store different types of fruits and vegetables. The determined cooling loads for the two cold storages are 5.44 and 6.21 kW, respectively. The required PV panel's area for 5 and 0°C compartment are 23.6 m^2 and 32 m^2 Monocrystalline type. In addition, the refrigerator EER values are 9.04 and 7.41 for the 5°C and 0°C compartments, respectively.

Key-words

Solar cooling – Refrigeration – PV – Vapor compression – Cold storage – Remote areas

1. Introduction

Due to increasing rate of using of the conventional energy sources that leads to draining the fossil fuels resources, renewable energy sources are the good alternative to be utilized to produce energy needs in many applications such as cooling, refrigeration and heating. Cooling and refrigeration of post-harvest crops especially at remote areas in hot arid climates are significant, as these crops have to be delivered fresh to the consumers. However, due to the lack of electric grid network at these remote areas, solar driven vapor compression refrigeration system is one of suggested alternative systems. Such system need to be designed and its performance need to be evaluated. Solar cooling system saves electrical energy by about 25-40% when compared to an equivalent conventional water cooled refrigeration system (Afonso, 2006). Also solar-driven cooling systems had a higher initial cost than convention vapor compression systems and that's limit the headway of these technologies. Solar vapor compression cycles had high values of coefficient of performance (COP) compared to solar thermal systems (Abu-Zour and Riffat, 2007). The collector area of the solar thermal system had to be more than six times larger than the corresponding solar electrical collector field to achieve the same amount of saved primary energy (Hartmann et al., 2011). Also solar electrical driven cooling system had much higher COP and a lower Photovoltaic (PV) footprint, however it had highest cost depending on required COP (Otanicar et al., 2012). PV driven refrigeration systems would be the suitable option for remote areas and the problem of varying of the rate of producing electricity with time can be solved by using a variable speed capacity compressor (Mekhilef et al., 2013). For vapor compression refrigeration system with input power of 1000 W, the COP is in the range from 1.1 to 3.3 at evaporator temperature between -5 and 15°C and condenser temperature between 45 and 61°C (Kim and Ferreira,

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2008).

There are many agriculture crops that are being cultivated in Egypt reclaimed deserts at hot arid remote areas with lack or no electrical grid connection at these remote areas. Consequently, to deliver such crops at high product quality, after harvest to the consumers, the transport storage environment need to be at appropriate temperature and humidity levels to maintain highest product quality to the end user. Therefore for multiple crops carriage, a transportable multi-store solar driven refrigeration system is suggested to be used.

In this study, design and performance investigation of a 15 ft^3 transportable solar driven (Direct Current) DC refrigeration system to be utilized for postharvest handling of crops has been carried out through modeling and simulation. There are two compartments with different working temperatures of 5 and 0°C to store different types of fruits and vegetables. The suggested PV refrigeration crops carriage will be designed through modeling based on the hour by hour yearly weather data of reclaimed desert areas in Alexandria, Egypt. The refrigeration crops carriage is driven by the DC power output from PV modules. Phase Change Materials (PCM) is utilized as thermal storage to cover the cooling demand during sunset time. The model is used to investigate the performance of designed solar driven refrigerator crops carriage when operated at different environmental conditions in other locations within Egypt climate with thermal storage.

2. PV Driven Refrigeration System Physical Model

In this study, transportable PV driven vapor compassion refrigeration system consists of cold storage rooms and PV system. The gross cold storage room is a commercial 15ft3 shipping container (the main outer frame of the cold room). It is selected to be the main crops reservation and cargo and it is divided into two equal compartments. Each compartment has a separate vapor compression cycle which consists of DC compressor, condenser, expansion device and evaporator. Refrigerant R134a is used as the working fluid. Moreover, its external walls are thermally insulated by foam and a gap between the internal walls and the compartment storage space is created and are filled by thermal energy storage material (PCM) to store the cold energy required to cover the cooling load demand at the time were there is no or lack of solar energy. This refrigerator is driven by PV system which consists of PV panels connected to controller which is used to regulate output DC electrical energy of PV to be proportion to compressor input power. A schematic diagram of the proposed system is shown in Fig. 1.

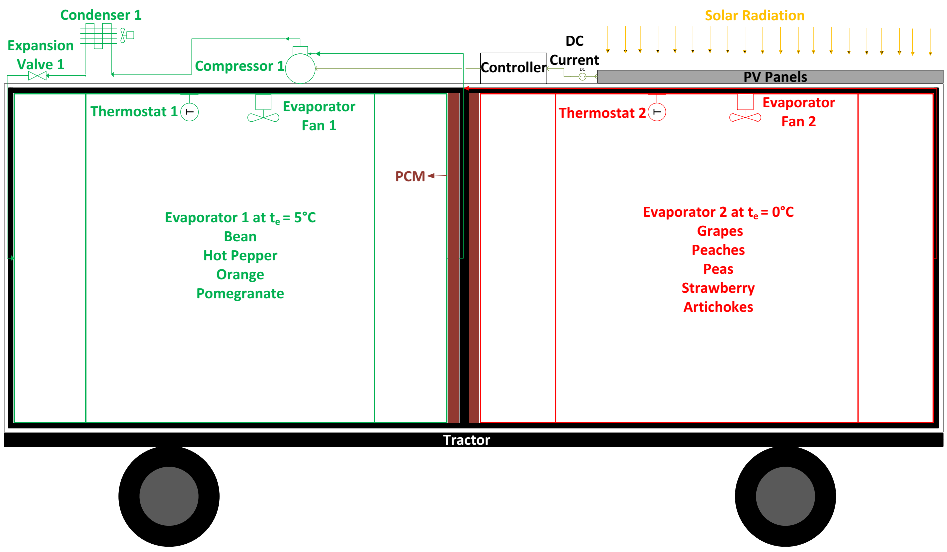

Fig. 1: Schematic diagram of multi-store multiple crops carriage solar driven refrigeration system

Each compartment has its specific storage temperature which is 5 and 0°C, respectively. These temperatures were identified correspondence to the real cultivated crops around Alexandria, Egypt reclaimed deserts in hot arid remote areas. The temperature value of 5°C refrigeration compartment is designed targeting to store the following crops product: Beas, Pepper, Orange and Pomegranate. While the compartment with temperature of 0°C refrigeration is designed targeting to store Grapes, Peaches, Peas, Strawberry and Artichokes. Each compartment has its own thermostat which is set to off and on the power on compressor when each compartment reaches its required storage temperature. Each compartment is designed to store about 2.1 ton of fruits and vegetables. The design point input data are based on the actual working months where targeted crops are harvested. Those crops harvest months are from March to October for the 5°C compartment and from April to June for the 0°C compartment.

3. Mathematical Model

Being driven by solar energy, the performance of the multi-store multiple crops carriage PV solar driven refrigeration system is affected not only by geometrical parameters of the cold storage rooms, but also by local climatic conditions. The system is designed and modeled based on the hour by hour yearly weather data of reclaimed desert areas in Alexandria, Egypt considering on the following design conditions: maximum ambient dry bulb temperature of 40°C, total cold storage room area of 4.55 m * 2.44 m * 2.59 m (L * W * H) with tractor trailer maximum speed of 80 km/h. This weather data include hourly air ambient dry bulb temperature, relative humidity, sun angles, horizontal insolation, wind velocity and atmospheric pressure. The whole system model is consists of integrated subcomponent systems models of the evaporator, compressor, and PV with details of each subsystem are as follows.

3.1. Evaporator Model

As this system designed targeting storing fruits and vegetables, total refrigeration load consists of (1) transmission load, which is heat transferred into the refrigerated space through its surface; (2) product load, which is heat removed from and produced by products brought into and kept in the refrigerated space; (3) internal load, which is heat produced by internal sources (e.g., lights, electric motors, and people working in the space); (4) infiltration air load, which is heat gain associated with air entering the refrigerated space; then safety factor is added to allow for possible discrepancies between design criteria and actual operation. This system is modeled based on ASHRAE (2010). The total refrigeration load Q_T [kW] is calculated as follows:

$Q_T = Q_{TL} + Q_{PL} + Q_{IL} + Q_{IAL}$ (eq. 1)

where, Q_{Tr}: is the transmission heat rate gain [kW]

Q_{PL}: average product cooling load [kW]

Q_{IL}: internal load of energy rate dissipated in storage room such as lighting, fans and people etc. [kW]

Q_{IAL}: average heat gain from infiltration [kW]

3.1.1. Transmission Load

The energy balance on the evaporator walls is shown in Fig. 2. To determine the transmission load, the outside air convection heat transfer coefficient had to be determined for moving air at ambient temperature t_{amb} as follow:

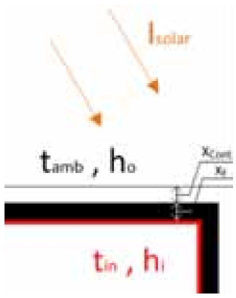

Fig. 2: Detailed section of the evaporator

Sensible heat gain through walls, floor, and ceiling is determined from,

$Q = UA\Delta t$ (eq. 2)

where, Q: heat gain [kW]

U: overall coefficient of heat transfer [$kW/m^2.K$]

A: outer surface area [m^2]

Δt: difference between outside air temperature and air temperature of the refrigerated space [°C]

$U = \frac{1}{1/h_0 + \Sigma x/k + 1/h_i}$ (eq. 3)

where, x: wall and insulation thickness [m]

k: thermal conductivity of wall and insulation material [kW/m.K]

$\Delta t = (t_{amb} - t_c) + \alpha_{st} I / h_o$ (eq. 4)

where, t_{amb}: ambient temperature [°C]

t_c: compartment temperature [°C]

α_{st}: absorbed solar radiation for outside wall material

I: solar radiation [kW/m^2]

In which the heat transfer coefficient h_o is determined as follows:

Reynolds number is determined at tractor trailer speed by using following equation,

$Re = u_T L / \nu$ (eq. 5)

where, Re: Reynolds number

u_T: tractor trailer speed [m/s]

ν: ambient air kinematic viscosity [m^2/s]

L: storage room length [m]

And Prandtl number is determined as following,

$Pr = \mu c_p / k$ (eq. 6)

where, Pr: Prandtl number

μ: ambient air dynamic viscosity [kg/m.s]

c_p: ambient air specific heat capacity [kJ/kg.K]

k: ambient air thermal conductivity [kW/m.K]

and for turbulent flow over a flat plate, Nusselt number (Nu) could be determined from following equation,

$Nu = 0.0296\, Re^{0.8}\, Pr^{1/3}$ (eq. 7)

also Nusselt number equal,

$Nu = h_o L / k$ (eq. 8)

where, h_o: outside air convection heat transfer coefficient

Then h_o could be determined.

3.1.2. Product Load

Heat that is removed from stored vegetables and fruits to reach required storing temperature for each crop type is analyzed to obtain the total product load which determined as follows:

$Q_{PL} = q_1/3600n$ (eq. 9)

where, Q_{PL}: average product cooling load [kW]

n: allotted time [h], where allotted time is time required for heat removal from products.

where, q_1: heat removed above freezing [kJ] is determined as follows:

$q_1 = mC_1(t_i - t_c)$ (eq. 10)

where, m: mass of product [kg]

C_1: specific heat of product above freezing [kJ/kg.K]

t_i: initial product temperature [°C]

t_c: final product temperature above freezing [°C]

3.1.3. Internal Load

Internal load Q_{IL} is all energy dissipated in storage room such as lighting, fans and people. For lighting, led lambs is chosen as it consumes little amount of energy. Each compartment has single fan of total load of 15 W to ensure that air is circulated inside storage room.

3.1.4. Infiltration Air Load

Heat gain from infiltration air which flow directly through the door can amount to more than half the total refrigeration load for the storage room. Heat gain through doorways from air exchange is as follows,

$Q_{IAL} = qD_tD_f(1 - E)$ (eq. 11)

where, Q_{IAL}: average heat gain from infiltration [kW]

q: sensible and latent refrigeration load for fully established flow [kW]

D_t: doorway open-time factor

D_f: doorway flow factor

E: effectiveness of doorway protective device

After the load for each compartment is determined, then the total refrigeration load Q_T is known. The design calculation based on maximum ambient dry bulb temperature of 40 °C, cold storage room area of 2.28 m * 2.44 m * 2.59 m (L * W * H) for each compartment with tractor trailer maximum speed of 80 km/h shows that the maximum required cooling load for the hottest day of the year that is on 11 June the cooling load values are 7.4 and 7.88 kW for the two cold compartments of temperatures 5°C and 0°C, respectively. These loads are for the conditions that each cold storage room is fully loaded instantaneously. However, in practice this doesn't occur to fully load each storage room with 2.1 ton of required crops and harvest it takes about two days. Therefore, each day will be crop load with about 1050 kg (150 boxes, 7 kg each) as working hours per day is 8 hrs. Therefore the real cooling load will be 5.44 and 6.21 kW at 5°C and 0°C compartment respectively. Consequently, depending on the required cooling load the DC compressor is selected and modeled.

3.2. Compressor Model

The compressor that fit the required cooling load has to be variable speed motor driven to match the variation of solar radiation during the daytime. For the 5 and 0°C compartment, a selection of a commercially available DC driven compressor A MASTERFLUX DC, model: ALPINE08-2750Y3. From its datasheet, it can operate at five different speed rpm's which are 2800, 3300, 3800, 4300 and 4800 rpm correspondence to different evaporator temperatures and condenser temperature of 55.4°C. This compressor as a subsystem lead to determine electrical efficiency ratio [EER] of the system that determined by,

$EER = 3.412\ Q_T/W_C$ (eq. 12)

where, W_C: compressor input power [kW]

from the compressor datasheet at evaporator temperaure of 5°C, the Power – Cooling capacity equation is given as follows,

$W_C = \ln(Q_e/0.7897)\ /\ 0.1755$ (eq. 13)

where, Q_e: compressor cooling capacity [kW]

and at evaporator temperature of of 0°C, the Power – Cooling capacity equation is given as follows,

$W_C = 0.0396Q_e^{\ 2} + 0.0791Q_e + 0.8407$ (eq. 14)

Thus, based on cooling load demands, the required DC compressor that would fit 5 and 0°C compartments cooling load require input power of 2.05 and 2.86 kW respectively. Consequantaly, the required input power to the comperssoer leads to the selection of PV modules type and the reqiured total PV area.

3.3. Photovoltaic Model

To determine the total size of PV module required to fit the compressor input power, an approximate expression for calculating the cell temperature t_{Cell} is given by (Ross, 1980),

$t_{Cell} = t_{Air} + \frac{NOCT\text{-}20}{80} I$ (eq. 15)

where, t_{Cell}: cell temperature [°C]

t_{Air}: ambient temperature [°C]

NOCT: Nominal Operating Cell Temperature provided by the PV manufacturer [°C]

I: solar radiation [mW/cm^2]

Therefore PV cell efficiency η_{PV} could be calculated from (Tiwari and Dubey, 2009),

$\eta_{PV} = \eta_{t_{Ref}}[1 - \beta_{Ref}(t_{Cell} - t_{Ref})]$ (eq. 16)

where, η_{tRef}: module electrical efficiency at the reference temperature t_{ref} and at solar radiation of 1000 W/m^2 [%]

β_{Ref}: temperature coefficient provided by the PV manufacturer [%/°C]

then the required total PV area is determined (Nafeh, 2009),

PV area = $W_C/(G_{av} * \eta_{pv} * TCF)$ (eq. 17)

where, PV area: total photovoltaic area [m^2]

W_C: average daily load of compressor [W]

G_{av}: average tilted solar energy input per day [W/m^2]

TCF: temperature correction factor

Then determine the peak power output from the total PV modules,

PV Peak power = PV area * PSI * η_{PV} (eq. 18)

where, PV Peak Power: maximum output power from PV [W]

PSI: Peak solar insolation [W/m^2]

Finally number of PV panels could be determined by using following equation,

No. of Panels = PV Peak Power / PV average output Power (eq. 19)

where, PV average output Power: PV average output peak power for each panel [W]

Then output power of the PV is,

PV Output Power = G_{av} * A_{PV} * N * η_{PV} (eq. 20)

where, PV output Power: output power of PV [W]

A_{PV}: area of PV panel [m^2]

N: number of PV panels

The PV modeling is carried out for three different types of PV panels Monocrystalline, Polycrystalline and Thin Film with the same output peak power of 500 W for each panel. However, this peak power occur few minutes around noon time, therfore the real average monthely output power from each PV panel is determined from average monthely solar radiation for refrigerator working monthes for each compartment. Tab. 1shows the mechanical and electrical specification of each PV panel type. Also total required area and number of PV panels required.

Tab. 1: Sizing of different PV Panels

Solar Cell Type	**Monocrystalline**	**Polycrystalline**	**Thin Film**	
Model	Clearline PV [PV30/500]	Qsolar [QS-500QLX]	ENN Solar [EST-500]	Masdar PV GmbH [MPV500-MXL]
Max-Power [W]	500	500	500	500
Efficiency [%]	16.7	14.9	8.76	8.69
Length [m]	2.898	2.506	2.6	2.6
Width [m]	1.173	1.34	2.2	2.2
Thickness [m]	0.082	0.02	0.045	0.034
Open Circuit Voltage [V]	79.5	75.7	286	286.1
Short Circuit Current [A]	8.5	8.68	2.86	2.74
5°C				
Average PV Output Power [W]	318.24	282.34	278.46	276.21
PV Area [m²]	26.6	25.87	44.01	44.37
PV Peak Power [kW]	3.94	3.85	3.85	3.85
No. of Panels	12.39 ≈ 13 Panel	13.65 ≈ 14 Panel	13.84 ≈ 14 Panel	13.96 ≈ 14 Panel
0°C				
Average PV Output Power [W]	302.6	268.49	265.17	263.06
PV Area [m²]	32.08	35.17	59.82	60.3
PV Peak Power [kW]	5.35	5.24	5.24	5.24
No. of Panels	17.7 ≈ 18 Panel	19.52 ≈ 20 Panel	19.76 ≈ 20 Panel	19.92 ≈ 20 Panel

From Tab. 1, it could be seen that Monocrystalline PV panel is the appropriate one as it has less required total area of 26.6 and 32.08 m^2 for each. This will lead to determine number of required PV panels which is 13 and 18 panels for each compartment. Consecuantly, selection of the conditioner that provide the DC compressor the required input power at the maker defined voltage.

4. Results of Investigation of the System Performance

The performance of the solar driven refrigerator is carried out when the two cold rooms are at 5 and 0°C, respectivelly. The performance is presented by COP that is carried at full load on the design date of 11 June (the hottest day in the year) at Alexanderia. The weather parameters for this date are shown in Fig. 3.a and Fig. 3.b. The results are presented during that day. As it can seen from Fig. 3, maximum ambient temperature is 39.6°C and tiltled solar radiation corrosponding to it is 774.82 W/m^2.

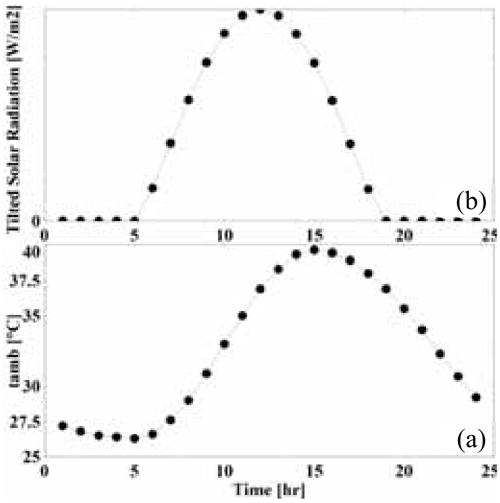

Fig. 3: Alexandria daily variation of (a) ambient dry bulb temperature, (b) tilted solar radiation

4.1. Performance of 5°C Compartment

The results of the performance of the refrigerator based on the input weather data are shown in Fig. 4 where Fig. 4.a shows variation of compressor input power, Fig. 4.b shows variation of resulted evaporator load (summation of cooling load and PCM load), Fig. 4.c show variation of required EER and Fig. 4.d show variation of condenser temperature. EER value obtained at ambient temperature of 39.6°C is 9.04 at cooling load of 5.44 kW, compressor input power of 3.58 kW and higher condenser temperature obtained during the day is 53.6°C while keeping the evaporator temperature at the design point value of 5°C.

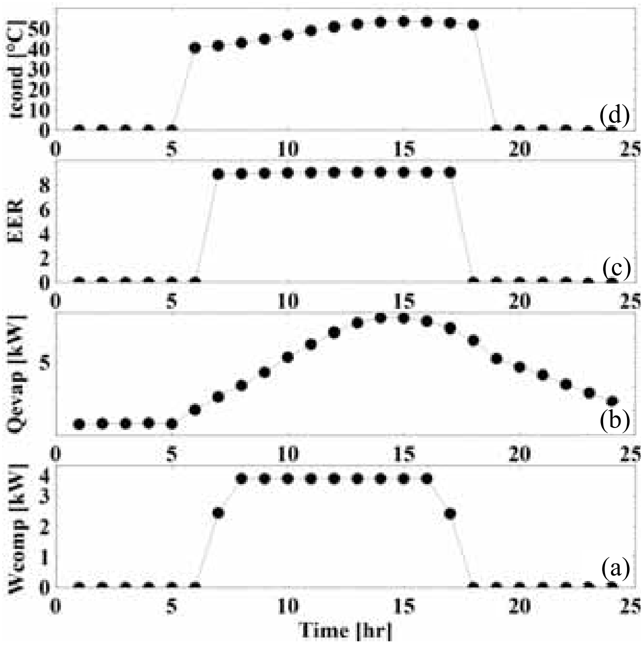

Fig. 4: Alexandria daily variation of (a) compressor input power, (b) evaporator load, (c) COP and (d) condenser temperature at 5°C compartment

4.2. Performance of 0°C Compartment

The results of the performance of the system at 0°C compartment is shown in Fig. 5. EER value obtained at ambient temperature of 39.6°C is 7.41 at cooling load of 6.21 kW, compressor input power of 3.58 kW and higher condenser temperature obtained during the day is 50.6°C while keeping the evaporator temperature at the design point value of 0°C.

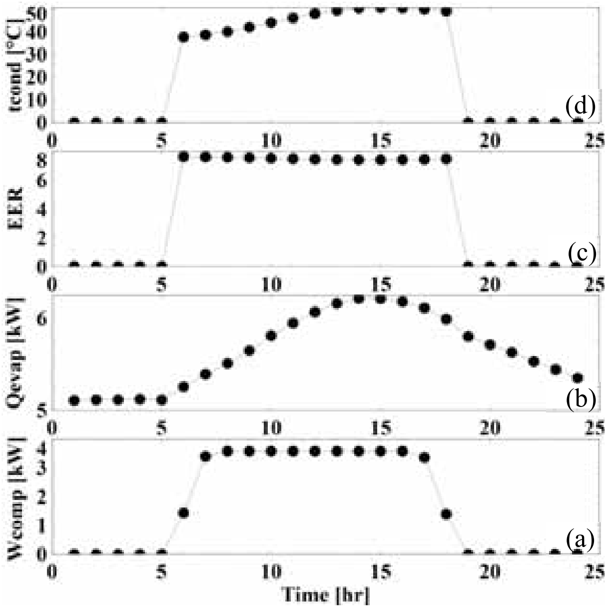

Fig. 5: Alexandria daily variation of (a) compressor input power, (b) evaporator load, (c) COP and (d) condenser temperature at 0°C compartment

4.3. Performance at Different Climates

The performance of this transportable solar driven refrigerator when operated at different environmental conditions is then investigated in other locations within Egypt Climate in Cairo, Asyut and Aswan for the 5 and 0°C compartment. Tab. 2 shows the performance of refrigerator at highest temperature during 2013 year for Cairo, Asyut and Aswan cities and number of required panels for each one.

Tab. 2: Performance of refrigerator at Alexandria, Cairo, Asyut and Aswan at 5 and 0°C

Parameter\Location	Alexandria	Cairo	Asyut	Aswan
Ambient Dry Bulb Temperature [°C]	39.6	42.4	44.7	46.7
Corresponding Tilted Solar Radiation [W/m^2]	774.81	657.92	660.28	633
5°C				
Compressor Input Power [kW]	3.58	3.58	3.58	3.58
Cooling Load [kW]	5.44	5.67	4.92	4.89
EER	9.04	9.05	8.96	8.95
Condenser Temperature [°C]	53.6	56.4	58.7	60.7
PV [Monocrystalline] Area Required [m^2]	23.6	25.23	21.05	20.57
Number of Panel Required	12.38 ≈ 13 Panel	13.88 ≈ 14 Panel	11.89 ≈ 12 Panel	11.66 ≈ 12 Panel

0°C				
Compressor Input Power [kW]	3.58	3.58	3.58	3.58
Cooling Load [kW]	6.21	6.48	5.67	5.63
EER	7.41	7.32	7.54	7.55
Condenser Temperature [°C]	50.6	53.4	55.7	57.7
PV [Monocrystalline] Area Required [m^2]	32.08	35.65	28.85	28.1
Number of Panel Required	17.7 ≈ 18 Panel	21.16 ≈ 22 Panel	17.25 ≈ 18 Panel	16.77 ≈ 17 Panel

From Tab. 2 it could be seen that Cairo have the highest cooling loads among other cities, it would require 5.67 and 6.48 kW cooling load, 14 and 22 panel for each compartment respectively. While for Asyut and Aswan, despite their higher ambient temperature, the required cooling load is lower that that's of Alexandria and Cairo. This is because relative humidity at Aswan and Asyut are much lower than that of Cairo and Alexandria, this lead to decrease of cooling load which lead to decrease in number of panels required. They both require 12 panels for 5°C compartment and Aswan require 17 panel for 0°C compartment. Also from Tab. 2 it can be seen that the compressor input power seems to be constant but this is because at this time of the day compressor is working at its maximum input power to achieve required load. From above Solar driven refrigerator would be applicable and economically valuable at Asyut, Aswan and Alexandria but at Cairo it would cost too much as it requires 14 and 22 panels and installation would be so difficult.

5. Acknowledgement

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