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Efficient TEC Module Selection and Sizing Strategies for Thermoelectric Cool Chamber Design

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ABSTRACT
This research paper explores the principles and design considerations involved in the sizing of thermoelectric coolers (TECs) for the development of an efficient thermoelectric cool chamber. The selection of an appropriate TEC module is crucial considering factors such as cool chamber dimensions, cooling load, temperature differential and power supply. The research employs both theoretical calculations of thermo-physical and thermo-electrical properties of TEC module and graphical calculations to determine TEC size for development of thermoelectric cool chamber.

I. INTRODUCTION

Ever since the industrial revolution, worldwide energy sources have been dominated by fossil fuels. However, there are two main issues with these conventional energy sources: they are finite, which means they can run out, and they emit carbon dioxide, which is negatively impacting the planet. Largely for these reasons, there has been a push to move from conventional to alternative energy sourcesand in recent years this has been gaining momentum. Due to climatic challenges, there is need to reduce primary energy consumption through an increase of efficiency in production, distribution and end-use, limit carbon dioxide emissions and increase the utilization of renewable energy sources.

Thermoelectric coolers, operating on the peltier effect have gained prominence in various applications due to their compact size, reliability and versatility. Besides these they completely eliminate the need or CFCs or HCFs and heavy compressors in small scale refrigeration. Most refrigeration failures are due to gas leaks or to the failure of moving parts in the compressor.

II. REVIEW OF LITERATURE

Palacios et al., 2009 described an analytical procedure to estimate internal parameters of commercial thermoelectric modules from the performance curves provided by the manufacturers. This can be used to predict the response of the module under any working condition. It was reported that though experimental measurements do notperfectly match the results estimated analytically, they can be considered rather precise considering the difficulties for attaining accurate experimental measurements in a standard laboratory. Ajiwigunaet al., 2018 developed a procedure to use TEC module for specific requirement based on manufacturer's technical data. The cooling system with TEC was designed and tested to maintain 6.6 litre of water at 24 °C while surrounding temperature is 26 °C. It was observed that one TEC module is enough for cooling load of 17.5 W while the cooling capacity is 18.87 W. The analytically predicted set point temperature was achieved with one TEC module. Prada Botiaet al., 2021evaluated thermoelectric devices usingexperimental methods and analytical models. The numerical model was developed based on thermoelectric phenomena. The results obtained are validated throughexperimental comparisons and the technical data of the thermoelectric devices. The comparisonbetween the different results shown a maximum error of 5%. It was reported that the developedmethodology would be robust tool for the realistic analysis of the performance ofthermoelectric generators and thermoelectric coolers.

Operating Principle of the Thermo-electric Cooler

The peltier effect, converting electric current into a temperature gradient, forms the basis

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of TEC operation. The heat transfer occurs from heat source to heat sink by electrons. The TEC consists of many p-n junctions (n-type has excess electrons and p-type has electron deficit) connected in series and thermally in parallel, interconnected by copper strips, integrated thermally by conducting ceramic plates on each side. When a DC flows across the junction of the semiconductors, one side will be cooled, whereas the other is heated depending on direction of current. If electrons flow from p-type to n-type, heat is absorbed in n-type side due to electrons jumping towards a higher energy state. The heat absorbed at the cooling side is transferred to the other side of heat sink and becomes hot. Though not as efficient as vapor-cycle devices, TEC have no moving parts or working fluid and can be very small in size.

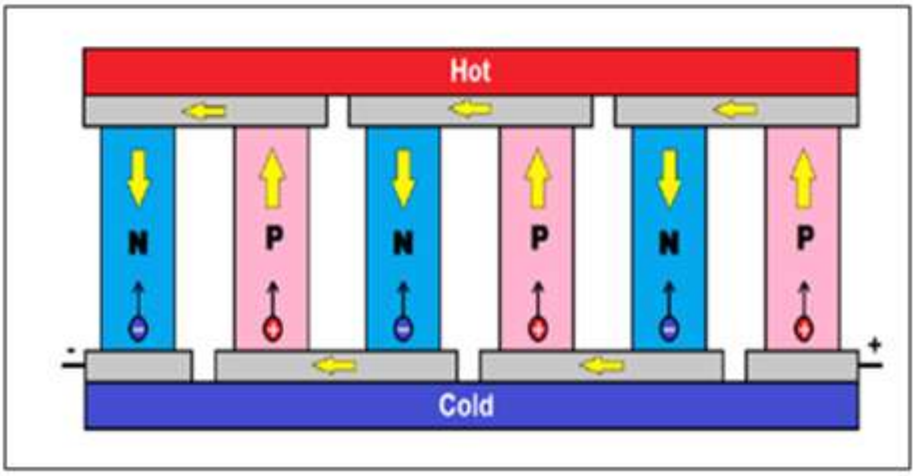

Fig. 1 Peltier Thermopile

Basic Design of Thermoelectric Cool Chamber

The developed thermoelectric cool chamber comprised of PV Panel, solar charger, battery, controller, thermoelectric module with heat sink and insulated box. The schematic design of thermoelectric cool chamber is shown in Fig. 2.

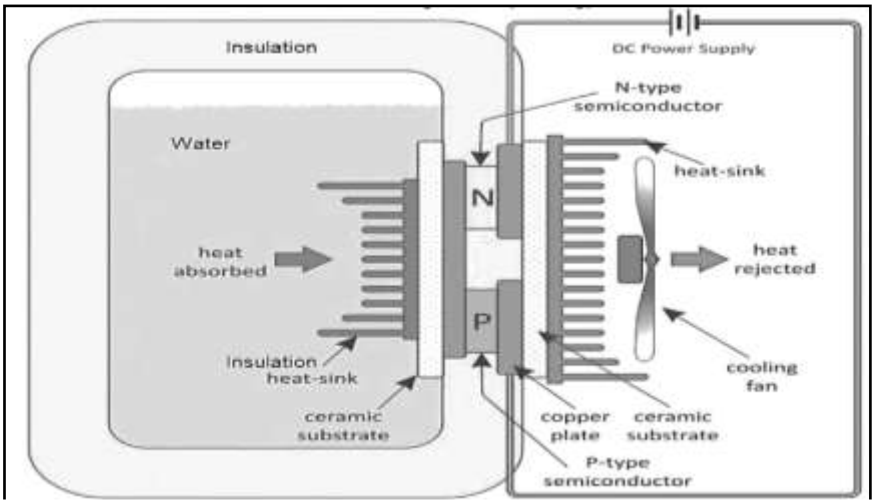

Fig. 2 Schematic diagram of Thermoelectric Cool Chamber

Methodology

The maximum ambient temperature and relative humidity at site was recorded. The refrigerated air temperature and relative humidity based on the safe storage temperature for most of tropical/ sub-tropical fruits and vegetables were taken as the design parameter. The air properties like enthalpy, specific volume, density, humidity ratio at ambient temperature and at safe storage temperature were determined from psychrometric chart.Thedesign parameters of insulated chamber and product to be cooled aredetermined for theoretical calculations (Table 1).

Table 1.Design Parameters of insulated chamber and product to be cooled

S.N.	Parameters
1.	Thermal conductivity of insulated cool chamber, W/m°K
2.	Thickness of cool chamber wall, m
3.	Convection coefficient of air, W/m^2°K
4.	Product to be cooled
5.	Rate of cooling,°C /hr
6.	Heat of respiration of material, W/kg
7.	Specific heat of product to be cooled, kJ/kg °K
8.	Size of thermoelectric cool chamber, m^3 (Internal)
9.	Volume of cool chamber, L

Design of thermoelectric cool chamber

The selected TEC module should have sufficient cooling capacity to maintain the proper temperature, dimensional requirements of the housing and appropriate performance characteristics. The TEC module was selected by considering cool chamber dimensions, cooling load Q_C, module temperature differential ΔT, power supply etc. The size of the TEC was calculated after the heat load of the cool chamber was determined theoretically and by calculating the V_{opt}, I_{opt}, Q_c and COP of TEC module. Similarly, TEC size could be calculated graphically resulting in a less complicated calculation than the theoretical calculation. These results were then compared with the measured results of test conducted.

In designing a thermoelectric cool chamber, heat was pumped from the cool chamber to the atmosphere. Also, the TEC elements itself generate heat which was added to heat loads. Hence, a poor design may fail to cool.

The procedure for designing cool chamber was as follows:

1. The total heat load was estimated.
2. After calculation of the total heat load, a proper TEC was selected followed by study of heat exchange mechanism.
3. Cooling process would fail if the pumped heat was not rejected to the atmosphere. Therefore, a heat sink was designed for the TEC unit.
4. All the system components were mounted and assembled in a unit base.

Estimation of heat loads in cool chamber

The cooler box was a polyurethane box with inner dimensions 0.64 x 0.35 x 0.39 m (length x width x height) with volume of 0.087 m^3 with perfect fit closing lid on top. The air inside the cold room would absorb the heat from different sources and this heat must be removed in order to maintain the cold chamber at 11 °C and 90 % relative humidity. Conductive heat load on the system occurred through lead wires, connected to the TEC which usually had a very small contact area with the TEC assembly. Thus, conductive heat transfer through lead wires was negligible.

The sources of heat load were the transmission through cold room enclosures, infiltrated air from the surrounding, stored product and running equipments like fans, lighting. Based on design specification of the cold chamber and product to be cooled, the equations from the literature could be adapted to appropriately determine the heat load from all sources. TEC size (Q_c) was determined in turn based on heat load estimation (Acharya et al., 2019).

Transmission heat

The transmission heat was thermal energy that flows through the warmer sides of walls into the cold chamber. Heat was transferred from outside to inside of cool chamber. The following equations were used to calculate transmission heat.

$$Q_{transmission} = U \times A \times (T_{amb.} - T_{in}) \quad (1)$$

Where,
$Q_{transmission}$ = Transmission heat load, W
U = Total heat transmission coefficient, W/m^2°K
A = Area of the heat transmission, m^2
$T_{amb.}$ = Ambient Temperature assumed as 31°C
T_{in} = Cool chamber inside temperature 11°C

$$U = \frac{1}{\frac{1}{\alpha_{in}} + \frac{1}{\alpha_{out}} + \sum_{i=1}^{n} \frac{X_i}{\lambda_i}} \quad \ldots (2)$$

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Where,
α_{in} = Coefficient of heat transmission of inside surface (W/m^2 °K)
α_{out} = Coefficient of heat transmission of outside surface (W/m^2 °K)
λ_i = Thermal conductivity (W/m °K)
X_i = Material thickness (m)

Infiltration heat

The infiltration heat load occurred due to air density differences between rooms. The entry of warm air to the cold chamber through doors, windows, cracks in chamber led to minor leakage. It was calculated as

$$Q_{infiltration} = V \times A \times (h_i - h_r) \times \rho_r \times D_t \quad (3)$$

Where,
$Q_{infiltration}$ = Heat produced by air changing and leakage air (KJ/h)
V = Average air velocity, m/s
A = Door Opening area, m^2 = 0.50 x 0.35 = 0.175 m^2
h_i = Enthalpy of infiltration air, kJ/kg
h_r = Enthalpy of refrigerated air, kJ/kg
ρ_r = Density of refrigerated air, kg/m^3
D_t = Decimal portion of time doorway is open
It was assumed that the typical air velocity through door was 0.3 to 1.5 m/s.

$$D_t = \frac{(P\times \theta_P)+(60\times\theta_o)}{3600\ \times\theta_d} \quad \ldots (4)$$

Where,
Dt = Decimal portion of time doorway is open
P = Number of doorway passages
θ_P = Door open-close time, seconds per passage
θ_o = Time door simply stands open, min.
θ_d = Daily time period, h

Product heat

Product heat was the heat brought into the cold chamber from product that was initially at warmer temperature. When new product was placed, heat enters inside the chamber. Therefore, energy was required to cool it. The heat produced by cooling above freezing point could be calculated by following equation.

$$Q_{material} = Wt.of\ material \times Specific\ heat\ of\ material \times (T_{amb.} - T_{Prodouct}) \quad \ldots (5)$$

If fruit or vegetables were product to be cooled then as it continued respiration even after harvest, they release water in atmosphere. Thus, product load involved respiration heat and heat of product (fruit/vegetable) above freezing temperature in cool chamber.

$$Q_{respiration} = \text{Wt. of material} \times \text{Heat of respiration of material} \quad \ldots (6)$$

Other heat sources like fans, lighting

The operation of lights/ fans contributed to internal heat sources. The fans were installed at cold side of TEC to effectively exchange heat within the cool chamber. In the absence of fans as in natural convection, the coldness would not transfer and water in the chamber get condensed on cool side of TEC and formed ice. The ice further caused resistance to heat transfer. As there were three fans installed at inner side of cool chamber with 0.3 A and 12 V.

The total heat released by operation of fan was given as:

$$Q_{fan} = 3\ (Current\ x\ Volt) \quad \ldots (7)$$

The total heat load was the sum of heat generated inside enclosure due to respiration and cooling of material, radiation, convection and conduction.

$$Q_{total} = Q_{transmission} + Q_{infiltra\ tion} + Q_{material} + Q_{fan} \quad \ldots (8)$$

The cooling power of a TEC system depended on various parameters, including the working current, temperature difference and geometrical characteristics. TEC modules could be stacked with different number of stages to obtain more cooling power.

Selecting appropriate TEC

An appropriate TEC was a module which was capable of providing the required cooling power. For this purpose, the performance graphs are used. Palacios et al., (2009); Weera, Sean Lwe Leslie (2014) gave analytical procedure to compute internal parameters from performance curves of thermoelectric modules. These graphs were normalized to provide a universal curve for use with any single or two stage TEC for which the maximum values were known. By using ratios of actual to maximum performance values, performance may be estimated over a wide range of operating conditions.

The left vertical axis of the first performance graph (Fig.3) shows the ratio of required temperature difference to the maximum temperature difference that can be provided by TEC module.

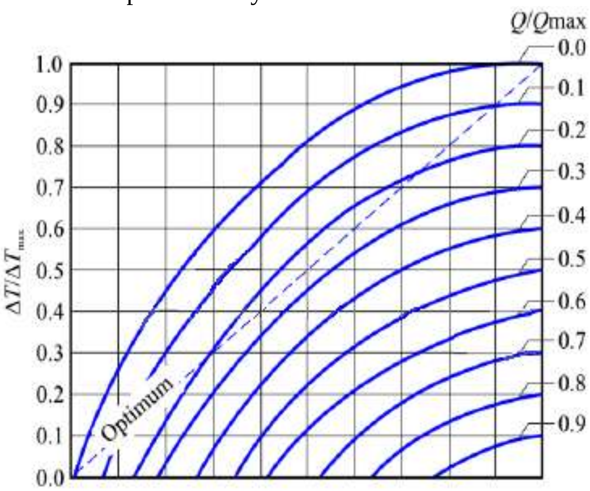

Fig.3 First performance graph for selection of TEC

The concept of the present study was to provide a cooling that was capable to prolong shelf life of vegetables and fruits to about 11 °C. The ambient temperature was considered to be 31 °C and the temperature of heatsink should be a little more than that. In the present study it was assumed the hot side temperature to be T_h = 31 °C. By this assumption,

$$\Delta T = (T_h - T_C) \qquad \dots (9)$$

This equation shows that the TEC system should be capable to work in a $(T_h - T_C)$°C temperature difference. Thus, for the present study ΔT = $(T_h - T_C)$°C and ΔT_{max} = 70°C according to the typical maximum obtainable ΔT for available single stage TEC (Table 1).

Therefore:

$$\frac{\Delta T}{\Delta T_{max}} \qquad (10)$$

The curves on the graph show the constant Q/Q_{max} graphs. The optimum design was shown by the diagonal line on the figure. This curve was used to select an appropriate TEC module for the system. To begin, start with left vertical axis.

By drawing a horizontal line till diagonal dotted line and then finding curved line on Fig. 3, theOptimum $\frac{Q}{Q_{max}}$.

At the intersection of the horizontal line just drawn and the diagonal optimum Q/Q_{max}line, Maximum $\frac{Q}{Q_{max}}$ As the total load of the TEC was Q=121 W. Then, maximum and optimum values of the load could be easily be found,

$$\text{Optimum}Q_{max} = \frac{Q}{\text{Optimum } (\frac{Q}{Q_{max}})} \qquad \dots (11)$$

$$\text{TEC Maximum } Q_{max} = \frac{Q}{\text{Maximum } (\frac{Q}{Q_{max}})} \qquad \dots (12)$$

The selected should be capable of providing a cooling load greater than Maximum Q_{max}, but less than Optimum Q_{max}. A TEC with a Q_{max} close to the optimum value provided maximum efficiency, and close to the maximum value yields a smaller and possible a less expensive module.

Number of TEC modules required

The number of TEC modules required were determined by dividing the maximum Q_{max} by Q_{max} of TEC. This was determined for TEC modules under consideration i.e. SKHC1-07106C. The number of selected module were calculated as,

$$\text{No. of TEC required} = \frac{\text{Maximum } Q_{max}}{Q_{max} \text{ of TEC}} \qquad \dots (13)$$

Hence, the cool chamber under study could be operated with calculated no. of TECs.

Thus these nos. of TEC modules were considered, could handle daily load of $\frac{\text{TEC Maximum } Q_{max}}{No.\ of\ TEC\ required}$W for selected SKHC1-07106C module.

After initial selection of TEC, the theoretical properties of TEC module and

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performance of selected TEC were determined based on manufacturer data sheet. The thermoelectric cooler module material chosen was Bismuth Telluride. The specification of thermoelectric module (SKHC-7106C) according to datasheet is given in Table 2.

Table 2. Specification of Thermoelectric module (SKHC1-7106C) according to datasheet

Sr. No.	TEC Module type	T_h (°C)	ΔT_{max} (°C)	V_{max} (Voltage)	I_{max} (Current)	Q_{max} (Watts)	AC resistance (Ohms)	Tolerance (%)
1.	SKHC1-07106C	27	72	8.9	6.4	35.6	1.06	±10

Where,
T_h (°C) - Hot side temperature at environment: dry air, N_2
ΔT_{max} (°C) -Temperature difference between cold and hot side of the module when cooling capacity is zero at cold side
V_{max} (Voltage) -Voltage applied to the module at ΔT_{max}
I_{max} (Current) -DC current through the modules at ΔT_{max}
Q_{max} (Watts) -Cooling capacity at cold side of the module under $\Delta T = 0$ °C
AC resistance (Ohms) -The module resistance is tested under AC
Tolerance (%) -For thermal and electricity parameters

Determination of theoretical properties of TEC module

A TEC was constructed by using a number of thermocouples thermally in parallel but electrically in series. Thermo-physical properties of a thermoelectric device concerned about temperature dependent physical properties of a module such as the seebeck coefficient, resistivity, thermal conductance and thermal resistance. While thermo-electric properties of a thermoelectric module concerned about temperature dependent electrical properties of the module such as the maximum operation voltage (V_{max}), the maximum operating current (I_{max}), the maximum temperature difference (ΔT_{max}) and maximum cooling capacity (Q_{max}). Basavraj (2020) reported that, figure of merit, cooling capacity of peltier module and coefficient of performance were the performance parameters of TEC. A good thermoelectric material should have a high Seebeck coefficient, high electrical conductivity and low thermal conductivity. The efficiency of Thermo-electric Coolers was limited to 10-15%. The materials with a ZT value above 0.5 could be practically used. The TEC module properties could be computed theoretically using following expressions (Basavraj, 2020).

where,
Q_{max}= Maximum cooling capacity of the TEC at 0 °C temperature difference.
Q_T = Total heat required to be absorbed = Cooling load calculated say 111 W
Q_c = Cooling capacity of module
V_{max} = Maximum voltage the TEC could withstand for given temperature difference specified in the datasheet, Volts
$I_{max} = DC\ current\ through\ the\ modules\ at\ \Delta T_{max}$
$T_h = Hot\ side\ temperature\ of\ selected\ TEC\ module$ as specified in datasheet, °K
T_c= Cold side temperatures of the Peltier module, °K
$\Delta T_{max} = Maximum\ temperature\ difference\ specified\ in\ datasheet\ °C$

R_o = Electrical resistivity, (Ω)
Rq = Thermal resistance °K/W
$\frac{R_q}{Ro}$ = Ratio of thermal to electrical resistance for ΔT_{max}
Z = Quality factor, per °K
α_M = Seebeck coefficient V/ °K
α = Seebeck coefficient, (V/ °K)
ρ = Specific resistivity of the module
ρ_M = Module electrical resistance
Km = Module thermal conductance
K= Coefficient of thermal conductivity
G = Factor of geometry
N = Number of thermocouples in module

Temperature dependent Seebeck coefficient ($\propto_M$), V/ °K

$$\propto_M = \frac{V_{max}}{T_h} \quad \ldots (14)$$

Electrical resistance (R_o), Ω

$$R_o = \frac{{V_{max}}^2}{2 \times Q_{max}} \quad \ldots (15)$$

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Ratio of thermal to electrical resistance for ΔT_{max}

$$\frac{R_q}{Ro} = \frac{2 \times \Delta T_{max}}{{V_{max}}^2 \times \left(1 - \frac{\Delta T_{max}}{T_h}\right)^2} \quad \ldots (16)$$

Thermal resistance, Rq (°K/W)

$$R_q = R_o \times \frac{R_q}{Ro} \quad \ldots (17)$$

Quality factor, Z (/°K)

$$Z = \frac{R_q}{Ro} \times {\alpha_M}^2 \quad \ldots (18)$$

Average temperature of the hot and cold side, T

$$T = \frac{T_h + T_c}{2}$$

Figure of merit, ZT

$$ZT = \frac{R_q}{Ro} \times {\alpha_M}^2 \times (T_h + T_c)/2 \quad \ldots (19)$$

Module Electrical Resistance ρ_M, (Ω)

$$\rho_M = \frac{(T_h - \Delta T_{max})V_{max}}{T_h \times I_{max}} \quad \ldots (20)$$

Module Thermal Conductance (K_M), W/°K

$$K_M = \left[\frac{(T_h - \Delta T_{max})}{2 \times \Delta T_{max}}\right] \times \left[\frac{V_{max} \times I_{max}}{T_h}\right] \quad \ldots (21)$$

Seebeck coefficient is also given by the expression,

$$\propto_M = 2 \propto N \quad \ldots (22)$$

Therefore,

$$\propto = \propto_M / 2N$$

Thermal conductivity of the module is given by the expression,

$$K_M = 2\,N\,K\,G \quad \ldots (23)$$

Therefore, $K = \frac{K_M}{2\,N\,G}$

Specific resistivity of the module (ρ) could be given by following expression

$$\rho_M = \frac{2 \times \rho \times N}{G} \quad \ldots (24)$$

Therefore,

$$\rho = \frac{\rho_M \times G}{2\,N}$$

Cooling Capacity per Module (Q_c), W

The cooling capacity was the capacity of heat that could be absorbed by the TEC module, at a given operating current, voltage, hot side and cold side temperatures. The quantity of heat pumped by the module is obtained from:

$$Q_C = 2\,N\left(\alpha\,I\,T_c - \frac{I^2 \times \rho}{4\,G} - K\,G\,\Delta T\right) \quad \ldots (25)$$

Number of TEC modules required for cooling (n)

$$n = Q_T / Q_c \quad \ldots (26)$$

Electrical Power Input (P_{in}), W

The power required was obtained from the expression

$$P_{in} = 2\,N\left[\alpha\,I\,(T_h - T_c) + \frac{I^2 \rho}{G}\right] \quad \ldots (27)$$

Voltage Input to the Module, V_{in}

The input voltage to the module was obtained from

$$V_{in} = \frac{P_{in}}{I} \quad \ldots (28)$$

Heat Rejected by the Module (Q_R), W

The quantity of heat to be rejected by the module was obtained from

$$Q_R = 2\,N\left(\alpha\,I\,T_c - \frac{I^2 \times \rho}{4\,G} - K\,G\,\Delta T\right) + 2\,N\left[\alpha\,I\,(T_h - T_c) + \frac{I^2 \rho}{G}\right] \ldots (29)$$
$$= Q_c + P_{in}$$

Table 3. Calculated Properties of selected TEC modules

S.N.	Particulars	SKHC1-07106C
1.	Module Seebeck Voltage ($\propto_M$),V/°K	0.0296
2.	Module Thermal Conductance (K_M), W/ °K	0.3006
3.	Module Electrical Resistance (ρ_M), Ω	1.0568
4.	Figure of Merit of module (ZT)	0.72
5.	Cooling Capacity per Module (Q_c), W	15.81
6.	Voltage Input to the Module (V_{in}), V	5.15
7.	Current of Module (I), A	2.87
8.	Electrical Power Input, W	14.78
9.	Coefficient of performance of module ($COP_{electrical}$)	1.07
10.	Heat Rejected by the Module (Q_R), W	30.59

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11.	Number of TEC modules required for cooling (N)	7
12.	Heat load required to be dissipated from hot side, W	214.13

Coefficient of Performance of the System ($COP_{electrical}$)

The electrical COP of the TEC as installed could be calculated from:

$$COP_{electrical} = Q_c / (I \times V_{in}) \quad \dots (30)$$

The properties of selected SKHC1-07106C were determinedand appended in Table 3. While designing a cool chamber, it was important to determine if the peltier modules would operate adequately under the designing conditions. The type and number of peltier (TEC) modules installed on insulated chamber indicated TEC sizing. If TEC sizing was undersized, performance of cool chamber under adverse conditions would suffer. If TEC sizing was oversized, it resulted in high power consumption which was not ideally suitable for small PV systems. Hence, performance of single TEC was important to validate experimentally to avoid future disappointment in handling cooling load. This could be determined in laboratory test by studying its heat exchange mechanism.

TEC Performance- limitations

The TEC was inferior to the compressor cooling for higher amount of heat pumping due to size limitations and internal joule heating. The TEC loses its competitiveness due to three problems

1. Internal joule heating
2. Small temperature difference between the hot side and cold side
3. Thermal conductance in the p-n blocks of the TEC between hot side and the cold side

Internal joule heating effect

$$Q_c = (\alpha_m \times I \times T_c) - \left[(I^2 \times \rho)/2G\right] - (K_m \times \Delta T \times G) \quad \dots (31)$$

The first part of the expression, calculate the total amount of heat that was pumped. It is linear function.The second part of the expression, calculate the total internal joule heating generated in the TEC due to the resistance component in the TEC.The last part of the equation is the heat flow in the p-n block between the hot side and the cold side of the TEC. The bigger ΔT became, the more heat was conducted from cold side to the hot side.

Temperature difference

The average temperature difference of a single stage TEC ranged between 64 - 70 °C. The semiconductor did not retain its zero resistance properties at high temperatures. If superior semiconductors were used there would be zero resistance resulting in zero internal joule heating. Thus, I^2R joule heating effect would disappear and the heat pumping ability would be linear, making TEC more competitive in space cooling applications due to its greater heat pumping capabilities.

Thermal conductance in p-n blocks of the TEC between hot and cold side

Martinez et al., (2013) reported that when the inner temperature reached the lower set point and the thermoelectric modules were switched off, a great amount of the heat stored in the heat exchanger at the hot end of the modules went back into the refrigerator, by heat conduction through the modules and the heat extender. This effect significantly increased the electric power consumption of the refrigerator. Therefore, supply minimum voltage to the modules once the inner temperature reached the lower set point, instead of switching them off, prevented heat from going back. In addition, during installation of TEC assembly consisting more than one TEC module, proper matching of heat sink and TEC module dimensions and use of insulation were important to reduce heat conduction (leakage) from hot side into refrigerator.

Optimum operating conditions of selected TEC

The optimum operating conditions of selected TEC were considered for reliable operation of peltier modules for cooling application. The second performance graph was used (Fig.4) to determine optimum current and voltage setting of chosen TEC module. The maximum temperature difference and the maximum cooling capacity varied for different modules. Therefore, recalculate the temperature ratio and cooling load ratio for selected module i.e. SKHC1-01706C. The design considerations were chosen based on technical limitation quoted by Basavraj (2020) in order to extend the life of

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selected peltier modules and so also operation of cool chamber.

Design Considerations

1. Choose the Peltier element with greater heat pump capacity than required ($Q_{max} > Q_c$). This could be achieved using multistage peltier modules or by assembling multiple peltier modules by suitable series/parallel connections. The multistage peltier modules were costlier than single stage peltier modules.
2. Operating current should be well below I_{max} of the Peltier element in use ($I_{max} > I_{op}$). The most commonly recommended input current for a TEC is 60 % to 80 % of the I_{max} for that TEC. Input current of greater than 80 % of I_{max} usually resulted in minimal increases in both heat pumping and ΔT while significantly increasing both power consumption and waste heat generated. Input current of lower than 60 % (of I_{max}) was also common to create a more efficient system (input power versus heat pumping created).
3. Provision of thermal isolation to the Peltier device to prevent heat transfer from hot side to the cold side should be done with thermal insulation sheet.
4. Either increase the size of the heat sink or add fan to keep the hot side temperature of the Peltier device as low as possible so that ΔT i.e. (T_h -T_c) was favourable for high COPs or else the Peltier element would end up drawing more current which was undesirable.
5. The rise of temperature of heat sink should not be more than about 15 °C above ambient.
6. It was assumed that daily 15 °C temperature should be reduced. Hence, $\Delta T = 15\,°C$.

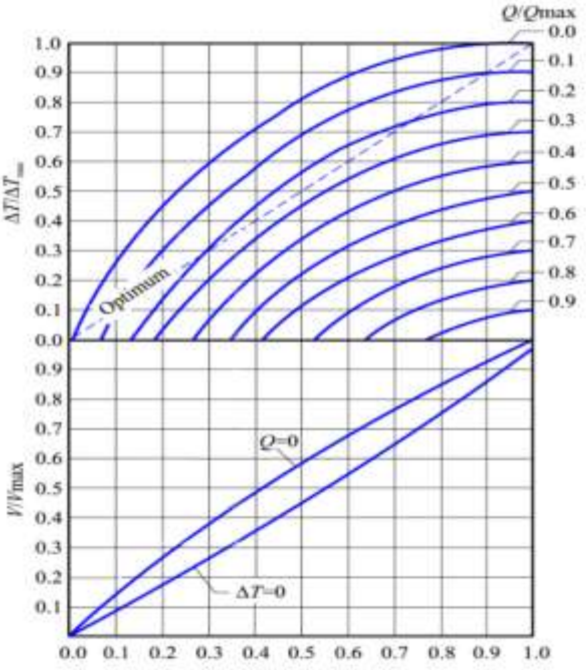

Fig.4 Second Performance graph to compute TEC's performance parameters

$$\frac{\Delta T}{\Delta T_{max}} = \frac{15}{72} = 0.2083 \quad \text{… (32)}$$

$$\frac{Q}{Q_{max}} = \frac{18.5}{35.6} = 0.5196 \quad \text{… (33)}$$

If from the intersection of horizontal line of Fig 4, $\Delta T/\Delta T_{max} = 0.21$and $Q/Q_{max} = 0.52$curve, plot a vertical line on the performance graph, it intersected the I / I_{max} axis at 0.45, the required optimum current was calculated by multiplying I / I_{max} value with maximum current of the TEC,

$$I = I_{max} \times \left(\frac{I}{I_{max}}\right) \quad \text{… (34)}$$

$$I = 6.4 \times 0.45 = 2.88 \text{ amp}$$

The intersection of the vertical line plotted above and the two curves shown at the bottom of the second performance graph (Fig. 4) gives two different values for the ratio of V/V_{max}. These values determine the range of of V/V_{max} for the selected TEC. For large ΔT's (small values of Q/Q_{max}), the voltage would correspond to the upper end of the range, and for small ΔTs (large values of Q/Q_{max}), the voltage would correspond to the lower end of the range. Multiply V_{max} by each of these values to obtain operational range of TEC voltages as follows:

$$V_{up} = V_{max} \times \left(\frac{V}{V_{max}}\right) \qquad \ldots (35)$$

$= 8.9 \times 0.55 = 4.90$ V

$$V_{low} = V_{max} \times \left(\frac{V}{V_{max}}\right) \qquad \ldots (36)$$

$= 8.9 \times 0.40 = 3.56$ V

Thus, the maximum power required by the TEC (Q_{active}) can be found as,

$$Q_{active} = I \times V_{up} \qquad \ldots (37)$$

$= 2.88 \times 4.90 = 14.11$ W

Where,
T_h (°C) - Hot side temperature at environment: dry air, N_2
Tc(°C) - Cold side temperature at environment: dry air, N_2
ΔT_{max} (°C) - Temperature difference between cold and hot side of the module when cooling capacity is zero at cold side
V_{max} -Voltage applied to the module at ΔT_{max}, V
I_{max} -DC current through the modules at ΔT_{max}, A
Q_{max} -Cooling capacity at cold side of the module under $\Delta T = 0$ °C, W
ΔT (°C) - Daily temperature reduction assumed as 15 °C
V - Operating voltage of module, V
I - Operating current of module, A
Q - Cooling load to be handled by each TEC, W (calculated as 18.5°C)

Thus, a single TEC module could generate heat of 14.11 W due to joule loss (i.e. active load) and absorb a heat load of 18.5 W. Hence, the heat sink should handle the total heat load of 32.61 W≅ 33 W.

Experimental validation

The calculated TEC size is compared with measured results from conducted tests, validating the accuracy and reliability of the sizing methodology. Any disparities between theoretical calculations and experimental outcomes are analyzed to refine the sizing process.

Power consumption

The power consumption for operating thermoelectric cool chamber was calculated as per the assembly used for the experimentation.

a) Power consumed by TEC device
An active load was the heat dissipated by the device being cooled due to the input power to the device. The most important active heat load in a TEC system was Joule heat. As three TEC assemblies comprising six TEC modules are used.

$$Q_{active} = (I^2 \times R) \times 3 = 3\,(I \times V) \qquad \ldots(38)$$

b) Power consumed by fans running at cold heatsink and hot side sink (Q_f)
The fans were installed at both cold and hot side of TEC to effectively exchange heat within the cool chamber and heat dissipation at hot side.

$$Q_f = \text{No. of fans x (Current supplied to fans x Voltage to fans)} \qquad \ldots (39)$$

Design of Heatsink

Selection of a heat sink is crucial as heatsink temperature directly affected the cooler hot side temperature, which in turn affected the cold side temperature that could be achieved with a TEC. The heat sink was used to dissipate the heat pumped by the TEC, in addition to the heat dissipated internally by TEC to surroundings. The temperature at the hot side was equal to the sum of the ambient temperature, T_A and rise in temperature across the heat sink from rejecting the heat load, Q and TE module power. The rise of temperature of heat sink should not be more than about 15 °C above ambient as per design consideration.

The hot side temperature of TEC could be reduced using heatsink with natural convection, forced convection by fan or circulating liquid. The choice depended on the requirements and constraints of each application. Typical performance values of 0.5 - 2 °C/W could be expected for natural convection, 0.02 - 0.5 °C/ W for forced convection, and 0.005 - 0.02 °C/W for liquid cooling. Limiting the rise of TEC hot side temperature to 5-15 °C above the surroundings was usually practical due to ΔT (64 °C) between the hot side and the cold side.

The thermoelectric and thermophysical properties of SKHC-07106C peltier module were calculated. It was found thatto handle cooling load of 111 W,7no. of SKHC1-07106CTEC modules were required.Each TEC should dissipate 31 W heat to surrounding.Hence, selected heatsink should be capable to transfer31 W of heat from TEC hotside to surrounding. It was suggested to validate this result for selected TEC experimentally to avoid future disappointment in amount of cooling.

III. CONCLUSION

The modules operational parameters, including ∆T, Qc, I and V are crucial in determining its suitability for a given application. The sizing methodology involves estimating the total heat load of the cool chamber, determination of thermo-physical and thermo-electrical properties of TEC module. The research employs both theoretical and graphical methods for TEC sizing. The theoretical approach involves calculating heat loads and determining optimal values for voltage, current, cooling capacity and coefficient of performance (COP). While the graphical method provides base for determination of operating current and voltage of TEC module.The importance of rejecting pumped heat to the atmosphere is mentioned.

This research paper concludes by summarizing the key findings related to TEC sizing for the development of a thermoelectric cool chamber. The importance of accurate parameter selection and sizing methodologies is emphasized, providing valuable insights for the design of thermoelectric cooling systems.

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