Solar Refrigeration using the Peltier Effect

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Thesis submitted in fulfilment of the requirements for a Masters Diploma in Technology in the School of Electrical Engineering at the Cape Technikon.

Date of submission: ....

Acknowledgements

I would like to thank my parents for their continuous moral and financial support and to Dr. Uken and Mr. Kleinhans for their support.

Special thanks to Aneen and Dr. Tapson for their share in preparing this thesis and lastly to Liezel for her encouraging role.

Declaration

I declare that the contents of this thesis represent my own work, and the opinions contained herein are my own and not necessarily those of the Cape Technikon.

J.C. Swart. .... 14 Mei 1996 ....

Date

Synopsis

In order to design a coolerbox, utilising Thermo-electric technology as a heat pump, it was necessary to determine if this type of technology would be suitable for a coolerbox heat pump application. A detailed TEC (Thermo Electric Cooler) sizing estimation was done, using formulas supplied by the manufacturer, and using thermo-electric theory formulas to calculate the absolute theoretical performance parameters for a suitable TEC. The correct heat pumping capability is important since it is possible to obtain TEC's with different input currents, voltages and heat pumping capabilities.

Using the TEC as a heat pump, it was decided to use water cooling due to the extremely high cost of a suitable air cooled heatsink. It should be noted that to cool the hot side of the TEC, a very efficient heat exchanger should be used.

A simulator was constructed to simulate, under variable operating temperatures and input powers, the cooling capacity of the TEC heat exchanger. The cooling characteristics were then used to determine if the theory and manufacturers claims correspond with the cooling characteristics of the coolerbox. It might be possible that the performance of the TEC would be drastically influenced, since installation conditions may not be ideal when installed; and, that the manufacturers performance claims are done when the TEC is operated under ideal conditions. This would ensure optimum results because, should an under sized TEC be used, the result would be poor cooling characteristics; or, if the TEC is over sized, the coolerbox would consume too much power, resulting in an inefficient system.

Using the results obtained from the simulator, it was possible to determine if the TEC performance claims made by the manufacturers could be applied directly to determine the performance of the heat exchanger. The claimed maximum temperature difference

(ΔT) between the cold side and the hot side was compared with the theory; and the maximum heat pumping capability was measured, and whether this value is achieved at the maximum ΔT, or at $\Delta T = 0°C$.

A coolerbox was constructed. The dimensions of the coolerbox were practical so that it would fit into a dwelling with ease, but would at the same time have a large enough cooling space, keeping in mind that the power consumption should be as low as possible. There was a battery storage space underneath the coolerbox for the batteries and the regulator of the PV system.

The prototype coolerbox was tested, and its cooling capabilities were compared with production TEC coolerboxes as well as a compressor type coolerbox.

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CHAPTER 1

Thesis Contents and Structure

1.1 Introduction

The current tendency of the first world is to look at renewable energy resources as a source of energy. This is done for the following two reasons; firstly, the lower quality of life due to air pollution; and, secondly, due to the pressure the ever increasing world population puts on our natural energy resources. From these two facts comes the realisation that the natural energy resources available will not last indefinitely.

In South Africa , as part of the RDP (Reconstruction and Development Programme), the policy is to provide electricity to every household. At the present time, more than half of the population do not have electricity in their homes. A study done by Escom[1] in 1992 showed that of the 7,2 million homes in South Africa, only 3 million at that stage were provided with electricity. Of the remaining 4,2 million, only 2 million could obtain electricity cost effectively. The remaining 2,2 million would be too uneconomical to electrify by cables due to topographical- and structural reasons and their isolation. Therefore, the ideal solution would be to use some type of renewable energy resource to provide these houses with energy without an expensive electrical grid connection.

One solution is a RAPS (Remote Area Power Supply) using an alternative form of energy. A popular renewable energy resource is a PV (Photo Voltaic) panel due to its 12 VDC compatibility with automobile appliances. The high cost of PV panels[2] (R1112 for a 40W PV panel), necessitates a small installation (40W - 70W) which usually provides only enough energy for lighting, a radio and a black and white T.V. No cooking, heating, air-conditioning or electrical geyser (all high power consumption equipment) can be accommodated due to the low power output of the PV panels (although such a high power output is possible, it would be very expensive).

For PV systems to be accepted, it should provide a fair degree of compatibility with household appliances. Therefore either the PV system should be adapted to provide 220VAC (Alternating Current) using an inverter, or appliances should be designed to operate at 12VDC (Direct Current).

A study[3] done by the University of Cape Town's Energy Development Research Centre came up with interesting facts that can be used to support the application of PV systems to Third World housing. The study found that once households received electricity, they still relied on multiple fuel use for cooking, heating etc. The electrical energy is usually used for lighting, because of its low energy consumption and the substantial increase it makes to living standards. In these houses, the most sought after appliances are a television, radio and (despite the cost) refrigerators.

Due to the high cost of locally available 12VDC refrigerators, a need to develop a low cost refrigerator was identified. Different types of heat exchangers were examined and it was found that a compressor system[4] would cost around R800 whereas an evaporation system[5] would cost about R400. A complete TEC (thermo-electric cooler) heat exchanger would cost roughly R200 (see Annexure E, Table E.1) thus proving to be the least expensive type of heat exchanger. Further, the advantage of using a TEC as a heat pump in a coolerbox is threefold. Firstly, a TEC heatpump and installation compared to a compressor system is cheaper; secondly, a TEC has the ability to operate under the continually changing input power which is associated with the output supplied from a PV panel (due to clouds or irradiation changes); and thirdly, the power consumption of a TEC heat pump can be controlled. These qualities make the TEC suitable for use with a PV power source.

The TEC cooler will utilise the power from the PV panels when the battery is fully charged, and at night, will use a small amount of power to maintain the temperature in the coolerbox. In other words, if the battery of the system is fully charged, and there is no appliance to absorb the power generated from the PV panel, it would be wasted, resulting in a poor efficiency factor for the whole PV system. The coolerbox integrated in a RAPS would allow for a very efficient system utilising all the excess generated power from the sun.

Sun energy is used for cooling; therefore, the more the sun shines, the warmer it gets, the more power there will be for cooling.

1.2 Formulating the Problem

Disadvantages of 12V coolerboxes are the following :

- Availability (suitable for Third World application and utilising less than 40W)
- TEC coolerboxes are designed for leisure rather than household purposes.
- Suitability to adapt to a RAPS system particularly in the application of power sharing regulator.
- High cost[6] (more than R800 in 1995).

From the above points we can see that for the average dwelling owner, it is very difficult to obtain or afford a 12V coolerbox. The criteria for development of a coolerbox are:

- Must be readily available
- Must be as inexpensive as possible
- Must be suitable for use with a RAPS system (using a power sharing regulator)
- Must be robust and maintenance free
- Must be environment friendly.

Two companies that sell 12V coolerboxes were approached. The first company, MINUS 40's coolerbox[7] uses a 12V compressor system as the heat exchanger. Although it is the most effective method of cooling, it is very expensive. A 40 litre coolerbox is priced at R2 700. The 12V compressor costs about R800,00. The introduction of the new (more ozone friendly) B134A gas will further elevate the price in the future. The power consumption of the MINUS 40 coolerbox is also too high at 52W (4.3A at 12V). It is also not suitable to be used with a power sharing regulator because the compressor requires a high start up current, and it may be the case that the PV panel can not supply that current, which can result in a damaged compressor. Due to the variable output power from a power sharing regulator, the compressor motor may stall, and be unable to restart again, resulting in overheating of the windings and possible damage.

The second manufacturer of 12V fridges is EDESA[8]. They sell a 40 litre coolerbox priced at R1 200. This coolerbox operates on 12V and uses an absorption cycle. A 75W heater is used as the heat source to boil the Ammonia Hydrogen. This would relate to 6.25A input current at 12V which is too high. The solution of Ammonia Hydrogen has to boil before the heat exchanger will operate effectively. It would therefore not be suitable for a power sharing regulator because when less that 75W is supplied, the ammonia solution will not boil and no effective cooling will take place. To replace or obtain an absorption cycle heat exhanger, would cost about R400,00.

The local thermo-electric coolers (TEC) like the 30 litre Coleman[9] coolerbox are available at about R650 - R700. Although the Coleman provides ample cooling power, its power consumption is too high at 43W (3.6A at 12V) and its cooling power at high temperatures (≥30°C) is unreliable. The TEC can operate at variable input powers and is thus suitable for power sharing regulators. The price of the TEC used in the prototype coolerbox is about R80 - R120 (1995) and it is environmentally friendly needing no gasses (CFC's) and is maintenance free.

1.3 Placing the Problem in Context

To design and build a coolerbox, there are three heat exchangers to choose from :

- Compressor type
- Absorption type
- Thermo-electric type.

Three factors were used to choose the type of heat exchanger :

- Cost
- Ability to operate with a PV system that utilises a power sharing regulator
- Low power consumption.

The only one of these three types of heat exchanger that is able to satisfy the above requirements is a thermo-electric heat exchanger. It is the cheapest, can run on little current and will operate with a power sharing regulator. To see how efficient the TEC is in a coolerbox environment, it has to be tested with this environment simulated by a simulator, where heat loads can be applied to the cold side of the heat exchanger. The ambient temperature influence can then be tested to determine the influence on the efficiency of a TEC and its heat pumping capabilities. This information should provide for a realistic indication of the sizing and behaviour of the TEC in this specific application. The data can also help to predict the behaviour of the coolerbox in power sharing conditions. Theoretically, as will be proved later, the co-efficient of performance (COP) and even the heat pumped, Q_c, could prove to be optimistic, as was discovered by Peterson[10] in his research project to cool down microwave repeaters. The following questions will be answered in relation to the TEC installation :

- What is the maximum temperature difference (ΔT = maximum) between the cold side and the hot side of the TEC?

- The amount of heat that is pumped when the maximum temperature difference between the hot side and the cold side of the heat exchanger is reached.
- The heat pumped when the hot side temperature of the heat exchanger is at 25°C, 30°C, 40°C, 50°C starting at 0,5A input current to the TEC, using 0.5A increments up to the maximum input current for each simulated hot side temperature.
- What is the efficiency of the heat exchanger installed in the coolerbox?

Tests have been done in a simulator which has been constructed. Its operation will be explained in Chapter 3. The results have been tabulated (see Annexure F) and discussed and a coolerbox has been constructed.

1.4 Aims and Objectives

The overall short term aim was to develop a small, inexpensive and compact 12VDC coolerbox using a TEC heat exchanger. As has already been explained in section 1.3, an important design parameter should be the ability to function under variable input power conditions. By using a coolerbox, all the power provided by the PV system could be utilised during the day, achieving very high overall efficiency for the PV system. The main aim of this dissertation is thus :

- To research and understand the Peltier effect.
- To look at commercially available 12VDC coolerboxes.
- To construct a simulator and test the behaviour and specifications of a TEC heat exchanger operating in a coolerbox environment.
- To compare the results with manufacturers results.
- To compare results with theoretical results.
- To design and build an inexpensive coolerbox that is compatible with a 12V RAPS system and can be manufactured easily in a Third World environment.

- To compare the coolerbox with at least two different types of commercially available TEC coolerboxes and a commercially available compressor type.

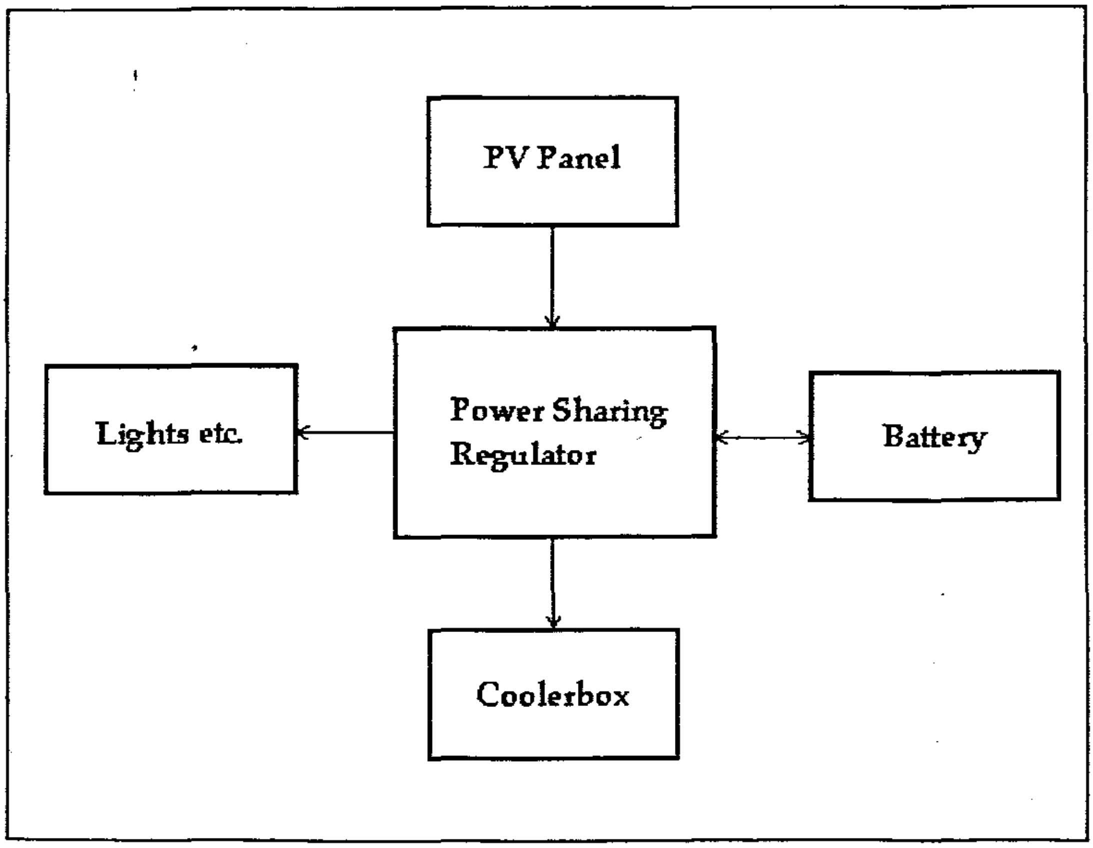

Figure 1.1 Graphical representation of the coolerbox connected to a PV system with a power sharing regulator.

Fig 1.1 illustrates the aim of the whole project. A TEC coolerbox should be integrated to a RAPS, consisting of a PV panel, battery and a power sharing regulator. The power sharing regulator will be used to divert excess power to the coolerbox.

1.5 Preliminary Investigation

The coolerbox was designed to comply with the design aims set out in Section 1.4. Preliminary research showed that the Peltier effect has been researched in detail (which will be discussed in full detail in Chapter 2) although not many research papers could be obtained where the TEC is investigated in its installed environment. This is very important because in the installed environment, conditions are not as ideal as in the mathematical analysis of the performance of the TEC.

1.5.1 Thermo-electric Cooler (TEC)

It was apparent that TEC has proved successful in specialised industrial applications in many parts of the world. The following applications for TEC's were discovered :

- They are used for fibre optic and other electro-optic cooling applications.
- Food service refrigeration and other commercial/institutional applications.
- Appliances such as portable 12V refrigerators, water and beverage coolers, etc.
- Instruments for physical, chemical, optical and electronic analysis (using the resistance change in the TEC for measurement as a reference cooling source).
- Laboratory and scientific instruments, computers and video cameras (cooling components).
- Medical and pharmaceutical equipment (keeping medicine cool at constant temperatures, and freezing specimens).
- Military/aerospace applications (noise free and compact cooling).

1.5.2 Advantages of the TEC

- Reduced size and weight compared to the absorption cycle system and compressor system.
- Reliable solid state cooling with no sound, vibrations or electromagnetic radiation.
- Predicted life span in a good installation is more than 200 000 hours[11].
- Precision temperature control could be achieved by just varying the input power to the TEC.
- It can operate on DC up to 14,8V (specified for a Marlow designed TEC model no. DT1049).

- TEC's can be cascaded to produce temperature differences between the hot side and the cold side of more than 64°C.

1.5.3 Disadvantages of the TEC

- Thermo-electric coolers suitable for PV systems are not manufactured in South Africa at the time of writing (May 1995).
- The co-efficient of performance[12] (COP) of the TEC is generally below unity.
- Low temperature difference between the hot side and the cold side of a single stage TEC is limited to 64°C (This value can be seen as the mathematical ideal and, as shown in Chapter 3, proved to be exaggerated under installation conditions).

1.5.4 Coolerbox Design

The design of the coolerbox must be such that it meets the following design parameters:

- The coolerbox should be neat and compact.
- It should be able to accommodate 4 x 1 litre standard size bottles (eg. Coca Cola bottles), 2 x 500g butter packs and 6 eggs in a standard egg box.
- The space utilization in the coolerbox should be maximised.
- The fridge should be the box type with the lid opening horizontally so that it will contain cold air better than a vertical lid when opened.
- The batteries and the regulator for the RAPS should be housed underneath the cooler, so as to be out of sight and touch of children and to make the RAPS and equipment as neat and compact as possible.

1.5.5 Hot Side Cooling

There are two main methods to cool the hot side:

- Cooling by using a finned heatsink.
- Cooling by using a liquid to air heat exchanger.

The hot side cooling has produced some cooling design problems. In South Africa the climate is quite harsh for any cooling system. Certain parts of the country have hot summers which put a very high demand on a cooling system. The heat exchanger has to work very hard to ensure that the heat exchanger operates at its optimum point (this had to be taken into account since a TEC has a 64°C temperature difference, as mentioned in Section 1.5.3, could be seen as an exaggerated value). This is important, since the hot side heat exchanger should be able to cope easily under hot harsh conditions as can be experienced in RSA.

Cooling by a suitable aluminium finned heatsink is expensive. These heatsinks are bought by the metre and cut to size. A suitable type could cost[13] between R457,35/m and R1 087,00/m. Another problem is that an extension block has to be inserted to move the coldsink and the heatsink further apart (this is done to increase the insulation area between the hot side and the cold side, while keeping in mind that the TEC is less than 4,5 mm thick). The undesired situation is created when there are three surfaces of contact through which heat flows. Each one introduces a thermal resistance relating to a temperature difference[14] of about 3°C. In a TEC heat exchanger with a small ΔT between the hot side and the cold side, this proved to be in-efficient. Another problem is that the suitable heatsink has to be very big and have a thick base because the heat must be evenly spread over the heatsink. From experience it was decided that the hot side of the heat exchanger should be kept as close as possible to ambient temperature. Finned aluminium heatsinks would prove to be far too expensive, and a cheaper finned heatsink will find it difficult to maintain the TEC hot side close to the ambient temperatures which is important for maximum performance.

Liquid cooling was chosen. This eliminates the extension block and there are only two surfaces through which the heat has to flow. Thus, more effective heat conduction results, enhancing the efficiency of the TEC heat exchanger. The hot side and cold side are also then perfectly isolated from each other (no overlapping of the heatsinks). The hot side could then be more easily kept at ambient temperature, and higher system efficiency with greater heat pumping capability is ensured.

1.5.6 Power Sharing Photo Voltaic (PV) Regulator

To interface the fridge to the PV system, a power sharing PV regulator must be used. The problem was that there was no such regulator available in South Africa (1995). To design one, the following parameters should be set :

- Its power consumption should be as low as possible.
- It should be able to share power with the battery and the fridge.
- It should be able to measure the amount of amp-hours on discharge of the battery so that the charge cycle could replace exactly that amount of energy.

Different types of regulation were considered, but none had the power sharing capability, which made the writer ask the question whether such a regulator exists in RSA? If not, it could identify a possible need to develop such a regulator.

1.6 Research Method

The project can be divided into three requirements :

- Methods of designing a TEC heat exchanger.
- Design a simulator to test the TEC performance.
- Building of a 12VDC TEC coolerbox.

The approach in this dissertation has been to discuss certain options in constructing the TEC heat exchanger. Pure logic reasoning would decide the type of cooling on the hot side of the heat exchanger. The major thrust of the project was towards the design of the simulator to simulate the TEC coolerbox, the construction of the TEC coolerbox and the comparison with commercial types.

- A detailed theoretical study was made of the Peltier effect. The history and the working of the Peltier effect was noted and a mathematical theory that explains the behaviour of a TEC was then researched.
- A coolerbox simulator and power supply was then built to simulate the working of a TEC in conditions that are not as perfect as those in which the manufacturers' data are measured. The simulator then simulates more realistic conditions, and the data derived will help the design of the coolerbox so that expectations of performance could be predicted more correctly.
- A coolerbox was then constructed and the performance was compared to two production TEC coolerboxes and a small 40 litre compressor type refrigerator.

1.7 History of Thermo-electric Cooling

Since the last century it has been known that when a closed circuit of two dissimilar metals and two junctions is formed, a current will flow between the junctions. This happens when there is a temperature difference between the junctions or when the metals have different temperatures. The phenomenon is known as the Seebeck effect, and is the fundamental principal behind the thermocouple. Generally speaking, the greater the temperature differences, the higher the current. Also, the combination of metals that are used will affect the current flow.

In 1834, James C. Peltier discovered that using joined dissimilar metals, a heat pump can be constructed. He found that passing an electric current through a junction formed by certain types of dissimilar materials could cause an increase or decrease in temperature at the junction. Peltier also found that the direction of current flow dictated whether heating or cooling occurred, and that the magnitude of temperature change was determined by the type of material and the size of the junction. This effect is called the Peltier effect, and it is the fundamental principal behind a thermo-electric cooler.

Using different metals produced cooling devices that had very poor co-efficient of performances (COP). This was because materials with a high temperature conduction co-efficient were used partly because of excessive temperature conduction between the hot side and the cold side of the thermo-electric heat exchanger. Since the discovery of semiconductors, the co-efficient of performance of the TEC was drastically improved since materials could be used with low temperature conduction co-efficients, but by doping it, the semi-conductor could be made to conduct, exerting electrical conduction properties found in metals. Altenkirch[15] derived the mathematical theory of thermo-electrical cooling in 1911.

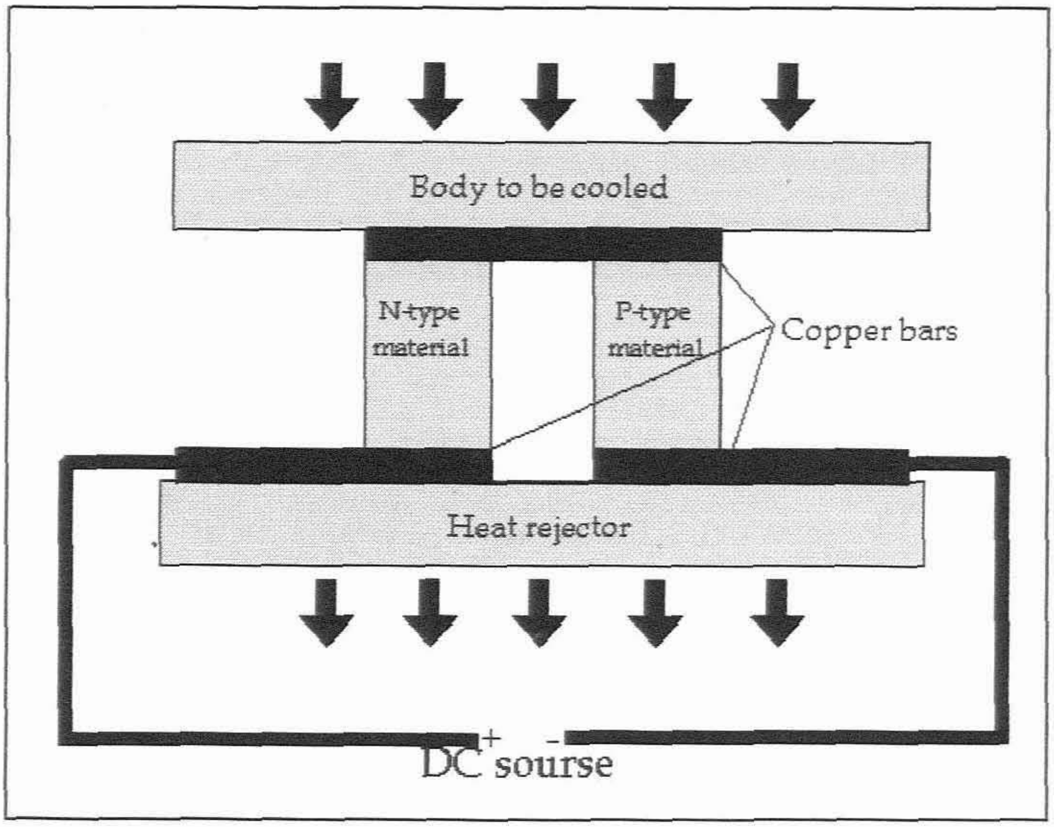

Figure 1.2 If the polarity of the DC source is connected as shown, then the heat is pumped from the top of the TEC, to the bottom. Should the polarity of the DC source be changed, the heat will be pumped in the opposite direction.

1.7.1 Description of the TEC

Fig 1.2 shows the arrangement of a simple TEC device. It consists of two pieces of semiconductor material; one is doped p-type material and the other doped n-type material. When current is applied, charge carriers move through the two materials, causing cooling of the top surface and heating of the bottom surface. This action is basically that of a heat pump. Heat is pumped from the top to the bottom of the device. If the applied current is reversed, then the top surface will be heated, and the bottom surface cooled. Most practical thermo-electric devices consist of many such elements. In those, the elements are connected electrically in series, and thermally in parallel.

CHAPTER 2

Theoretical Background and Design Parameters

2.1 Introduction

A thermo-electric cooler (TEC) is a solid state current device, which, if power is applied, move heat from the cold side to the hot side, acting as a heat exchanger. This direction of heat travel will be reversed if the current is reversed. It is a phenomena that is opposite to the Seebeck effect[16].

2.2 Materials used in a TEC

Although the theory existed in 1911, the materials available were not suitable for effective cooling. Metals have good electrical conduction but good thermal conductivity as well. This allowed for a very low COP (co-efficient of performance) of 1% due to the thermal conductivity of the metal from the hot side to the cold side of the TEC. It was only since the 1950's with the discovery of semiconductors, that the COP was increased. Semiconductors had the same electrical conductivity as metals but much lower thermal conductivity. This provided for a much improved COP of 20%. Typical material composition are alloys of the elements Bi, Cd, Sb, Te, Se and Zn. The standard alloy used today in manufacturing is the Bi_2Te_3 type. The typical composition of the material is shown in Table 2.1.

Table 2.1

Typical TEC p-n block material composition

N-type material	P-type material
$Bi_2Te_{2.4}Se_{0.6}$ doped with 0.1% AgT or CuBr	25% Bi_2Te_4 dissolved in 75% Se_2Te3 with 4% Te and 0.05% Ge

2.2.1 Operating Principle of the Thermo-electric Cooler

When a heat exchanger operates, a liquid or a gas acts as the cooling medium, carrying the heat from the cold side to the hot side of the heat exchanger. The liquid or gas acts as the heat carrier.

In a thermo-electric heat exchanger the electrons acts as the heat carrier. The heat pumping action is therefore a function of the quantity of electrons crossing over the p-n junction. Since the amount of current in a conductor represents the quantity of electrons flowing in a conductor, it can therefore be said that the cooling effect in a thermo-electric heat pump is directly proportional to the current flowing through the device.

The TEC consists of a p-n material junction. The operation can also be explained using the p-n junction theory used in diodes - the most simple semiconductor device. If a LED (light emitting diode) is analysed, it can be seen that the light that is produced by LED, follows the same principle as thermoelectric cooling. In p-material, the charge carriers are holes and in n-material the charge carriers are electrons. The free electrons in the n-material move in the conduction band, and movement will only take place under the influence of an applied voltage. This is known as electron motion. The hole transfer in p-material is a process in which energy levels in the valence band are not occupied by an electron. Holes, as with electrons, move in the valence band, but in the opposite direction. In any p-n junction where a voltage is applied, the holes and the electrons continually recombine (when current flows from the n-material to the p-material). In the case of the LED, electron flow is from the p-material to the n-material. The electrons move from the conduction energy level (n-material) to the valence band (p-material). The change in energy level from a higher energy level to a lower energy level, causes the electron to give off energy and this energy is given off in the form of visible light.

The same principle can be applied to thermo-electric cooling. As the electron moves from the higher energy state in the n-material to a lower energy state in the p-material energy is released (heating effect). If the electron moves from the p-material to the n-material, energy is absorbed because the electron moves from a lower energy level to a higher energy level, needing additional energy to cross the junction, which in the case of thermo-electric cooling, is obtained in the form of heat.

A thermo-electric cooler can be constructed by connecting many p-n junctions in series. Consider a p-n-p junction in a thermo-electric cooler; when the electron moves across the p-n junction, heat is absorbed due to the change from a lower energy level to a higher energy level, causing a cooling effect. As the electron moves across the n-p junction, the electron moves from a higher energy level to a lower energy level, causing the electron to give off energy, causing a heating effect. A heat pump is then formed, pumping heat (energy) from the cold side to the hot side. If the current is reversed, the cold side will change into the hot side and the heat will be pumped into the opposite direction.

The manufacturers change the composition of the n-material and the p-material continuously to improve the figure of merit[17] (Z) of the TEC. Since great amounts of money were spent on research and development, the exact composition of the n-material and p-material were not revealed to the writer by the TEC manufacturing companies. The maximum figure of merit reached today is $3.5 \times 10^{-3}K^{-1}$.

2.2.2 Construction of the TEC

The heat pumping of a single p-n junction is very low, typically in the milli-watt range. This problem was overcome, depending on the amount of heat required to pump, by connecting a series of p-n junctions in series and thermally in parallel.

If Fig 2.1 is observed, it can be seen that the p-n junction is split into a p-n block and electrically connected using a copper bar. The copper bar will also act as the surface where the heat is absorbed or rejected. If more than one junction is used, they are connected in series with the copper bars, forming the TEC.

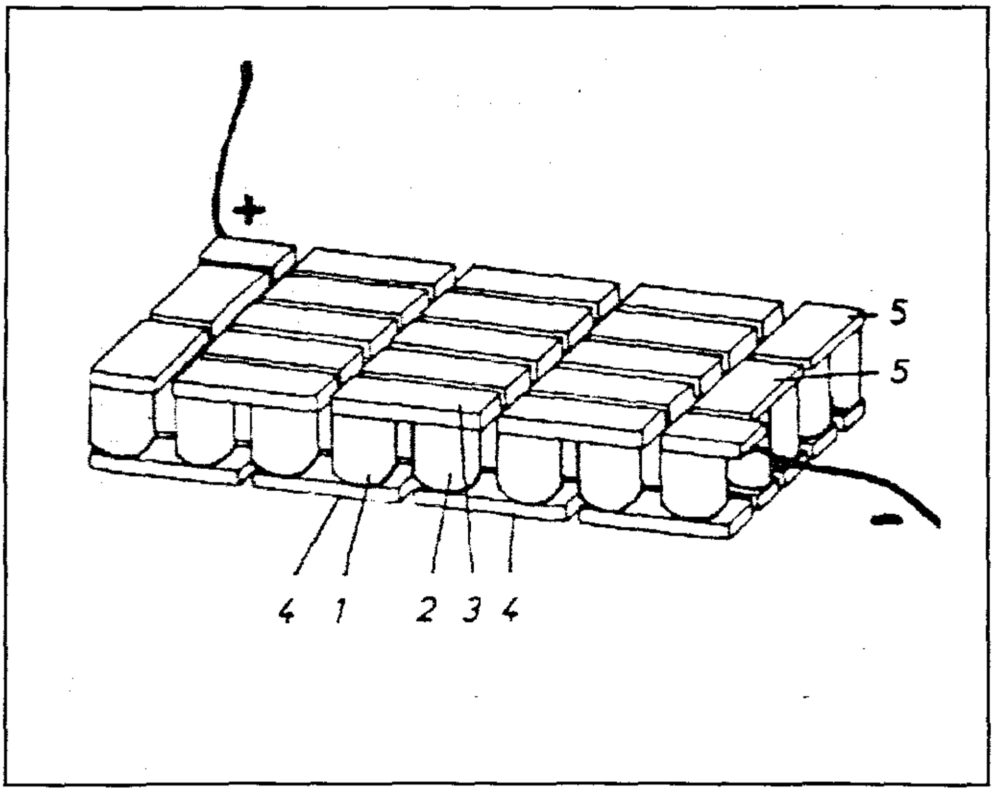

Figure 2.1 The basic construction of the TEC; 1,2 the p-n semiconductor legs; 3,4 copper bar contacts; 5, series electrical connection.

Due to the different voltages at the copper bars (because they are connected in series), the copper bars have to be isolated from the body that has to be cooled, to prevent any short circuit conditions. This is done using a heat conducting ceramic, which has no electrical conduction but proves to be an excellent heat conductor (to conduct the heat from the copper bars).

Some manufacturers (like Marlow Industries), isolate the TEC with a silicone seal to prevent any damp forming on the blocks or the copper bars due to the temperature gradient. This damp is detrimental, and due to the electrical flow of electrons can cause unwanted corrosion reducing the reliability of the TEC.

2.3 Types of Heat Conduction

When a thermo-electric cooler is used in a heat pumping application it is important to calculate the heat load for the heat exchanger. This is important, since the size of the TEC has to be determined to make sure that the heat exchanger operates satisfactorily under installation conditions.

2.3.1 Types of Heat Loads in a Coolerbox

Before setting out to determine the size (Q_c) of the TEC, the heat load has to be estimated. This is done by calculating the possible heat loads. The heat load may consist of four types[18], active or passive or combinations of the two:

- Active Heat load
- Radiation
- Convection
- Conduction

Active load is the power which is dissipated by the device. It is generally equal to the input power of the device. Passive heat loads (radiation, convection and conduction) are parasitic in nature. The heat load is measured is Watts.

2.3.1.1 Active heat load

$Q_{active} = V^2/R$ ..... (2.1)

where

Q_{active} = active heat load (W)

V = voltage applied to device being cooled (V)

R = device resistance (Ω)

I = current through the device (A)

2.3.1.2 Radiation

When two objects at different temperatures come within proximity of each other, heat is exchanged between them. This exchange occurs through electromagnetic radiation which is emitted from the one object and reaches the other object. The hot object will have a heat loss and the cold object will have a heat gain. This phenomena is called thermal radiation. This type of radiation is usually significant in systems with small active loads, and large temperature differences, especially when operated in a vacuum environment.

$Q_{radiation} = F.e.s.A(T_{ambient}^4 - T_c^4)$ .....(2.2)

where

$Q_{radiation}$ = radiation heat load (W)

F = shape factor (a worst case factor of 1 can be used)

e = emissivity (worst case value of 1 can be used)

s = Stefan-Boltzman constant (5,667 x 10^{-8} W/m^{2}K^4)

A = area of cooled surface (m^2)

$T_{ambient}$ = ambient temperature (Kelvin)

T_c = TEC cold side ceramic temperature (Kelvin)

2.3.1.3 Convection

When a fluid (in this case a gas) flows over an object while the temperature of the gas and the object are different, heat transfer takes place. The amount of heat transfer may vary depending on the rate (m^3/s) at which the fluid is flowing across the object. In the case of a coolerbox, the convection of air will be natural (or free) convection. This is caused by the density difference between the cold and warm gas.

$$Q_{convection} = h.A(T_{air} - T_c) \qquad ..... (2.3)$$

where

$Q_{convection}$ = convective heat load

h = convective heat transfer co-efficient (21,7 for air at 101.3kPa)

A = exposed surface area

T_{air} = temperature of surrounding air

T_c = temperature of cold surface

2.3.1.4 Conduction

Conductive heat transfer occurs when energy exchanges take place by direct impact of molecules from a high temperature region to a low temperature region. Conduction may occur through lead wires, mounting screws, etc. Anything may form a thermal path from the device being cooled to the heatsink or ambient environment.

$$Q_{conduction} = k.A.\Delta T/L \qquad ..... (2.4)$$

where

$Q_{conduction}$ = conductive Heat load (W)

k = thermal conductivity of the material (W/m°C)

A = cross-sectional area of the material (m^2)

L = length of the heat path (m)

ΔT = temperature difference across heat path (hot side - cold side)

2.4 Thermo-electric Cooler Heat Exchanger Installation Design

Since a TEC is a simple heat exchanger, it is also simple to install. To use a TEC as a high efficiency, reliable heat exchanger, a slightly more complex installation is required. The rest of this section will discuss these points.

2.4.1 Determining the TEC Hot Side Temperature

The heatsink is used to dissipate the heat pumped by the TEC, in addition to the heat dissipated internally by the TEC, to the surroundings (which may be air or liquid). To do this, the heatsink will be warmer than the surroundings. It is thus important to keep heatsink temperature as close as possible to ambient temperature.

The thermal resistance (efficiency) of a heatsink is measured, as the temperature rise of the hot side above ambient, per Watt of power dissipated into the heatsink. Therefore, the TEC hot side temperature may be estimated by multiplying the thermal resistance of the heatsink by the amount of heat dissipated at the hot side of the TEC. Three types of hot side TEC-to-air heat exchangers are usually used :

- heatsink with natural convection,
- heatsink with forced convection,
- liquid cooling with forced convection.

The choice depends on the requirements and constraints of each application. Typical performance values of 2.0°- to 0.5°C/W can be expected for natural convection, 0.5°- to 0.02°C/W for forced convection, and 0.02°- to 0.005°C/W for liquid cooling. Limiting the rise of the TEC hot side temperature to 5°C to 15°C above the surroundings is usually practical due to ΔT (64°C) between the hot side and the cold side.

For maximum reliability, it is important that the hot side temperature of standard TEC's be kept below 85°C. For this project water cooling was used. Cape Heat Exchange PTY (Ltd)[19], assisted in the design of the water-to-air heat exchanger. They used their experience to design the liquid radiator, using existing automotive designed cooling tubes. This is of great advantage, since a new tube design would have been costly.

2.4.2 Determining the TEC Cold Side Temperature

The temperature required at the cold side of the TEC is a very important factor. Here it should be checked if a single stage could be used or if a very low cold side temperature is required; a double stage or triple stage TEC could be used. The necessity to determine a suitable hot side temperature would be influenced by the temperature difference between the hot side and cold side of the TEC (see Annexure B on sizing a TEC).

2.4.3 Mains Supply to TEC

The current that will drive the TEC in this case will be DC (direct current) current from the PV supply and the battery. There will thus be no AC (alternating current) Joule heating added to the internal components.

If the TEC is supplied with rectified current (AC to DC), it should be noted that the manufacturers specify that the power ripple ($[I^2.COS^2\,\omega.t].R$) should be less than 10%. The smaller the ripple input power, the more efficiently heat is pumped since the AC internal Joule heating is limited. The degradation due to ripple[20] can be determined by:

$$\Delta T/\Delta T = 1/(1+N^2) \qquad ..... (2.5)$$

where

N = % ripple.

2.4.4 Temperature Control

When a temperature controller (thermostat) is installed, it should be mentioned that a TEC is not suitable for the on/off cycle type. This cycle causes the rapid expansion and contraction of the internal components, the copper bars and the applied pressure variation of the heatsinks. The end result is a much reduced life span of the TEC. The temperature controller should be a proportional action type, to regulate temperature increasing or decreasing current as accurately as possible, keeping the desired temperature as constant as possible.

2.4.5 Selecting and Installing a TEC Device

Before selecting a device the amount of heat to be pumped should be determined. A suitable TEC should then be decided upon to pump the amount of heat required at the temperature ΔT (difference between the hot side and the cold side). A maximum hot side temperature should be decided upon. Due to the 14.8V output voltage of a PV system, two TEC models were considered (see Table 2.2) due to their favourable maximum input voltage of 14.8V and 15.4V.

Table 2.2

Two models from two companies in the USA that manufacture TEC's.

Model	$T_h = 25°C$					Dimensions		
number	Q_{max}	V_{max}	I_{max}	ΔT_{max}	N	A	B	C
Marlow DT 1049	33.3W	14.8V	3.6A	64°C	127	30mm	30mm	3.7mm
Melcor CP 1.0-127-05L	33.4W	15.4V	3.9A	67°C	127	30mm	30mm	3.2mm

Table 2.1 should be understood the following way (see CP 1.0-127-05L):

- The heat pumped (Q_c) is 33.4W at $\Delta T = 0°C$ when $T_h = 25°C$.
- The maximum input voltage is 15.4V and the maximum input voltage is 3.9A.
- When ΔT_{max} is at a maximum difference of 67°C then the TEC pump zero Watts of heat.
- N, represents the number of thermocouples, i.e. 127 p-n couples (254 p,n blocks).
- The dimensions are the length, width and the diameter of the TEC block.

The TEC data for the DT 1049 can be interpreted in the same way.

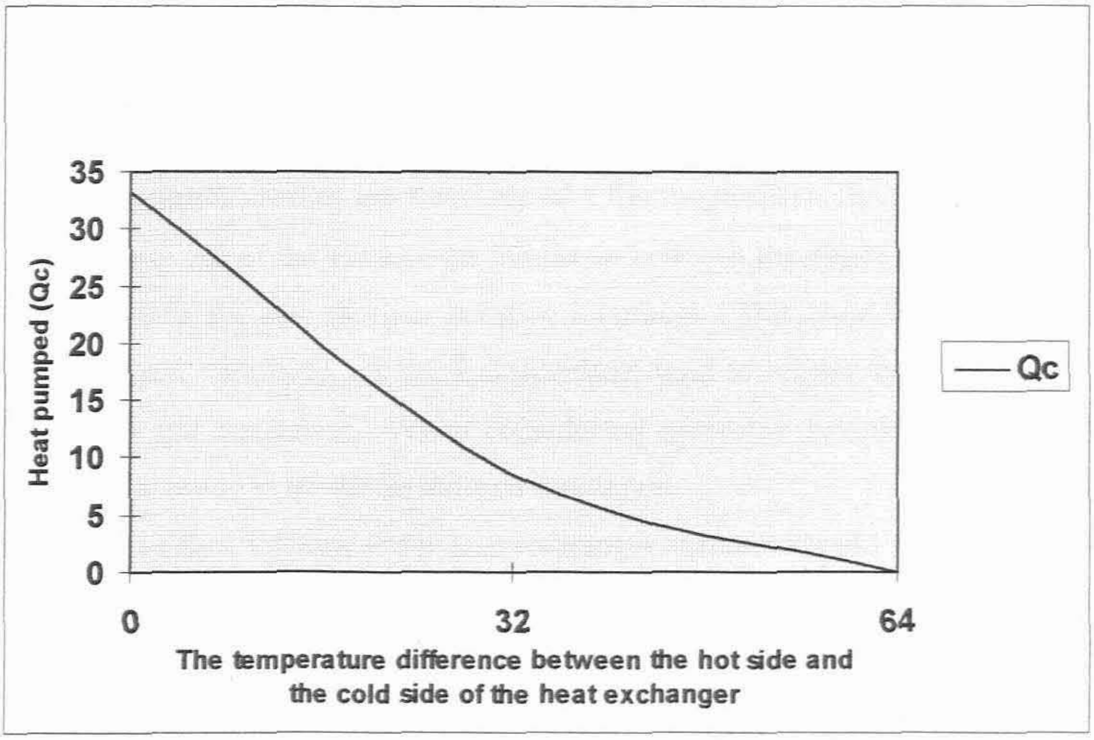

Figure 2.2 The non-linear decrease of the heat pumped (Q_c) as ΔT increases.

Fig 2.2 illustrates that the heat pumped (Q_c) decreases as the ΔT increases. It is very important to keep this in mind when working with TEC cooling because ΔT is so small.

- When a heatsink is designed, it should be noted that it must not first be designed for the 33.4W of heat to be pumped, but the I^2R heating effect generated by the TEC should also be taken into account. A sizing of the heatsink would be :

$$(I_{min} \times V_{max}) + Q_{max}.$$

Where I_{max} and V_{max} will be the supplied power (which need not be the I_{min} and V_{max} of the specification list but the operating or worst case input power to the TEC).

Example : $(I_{min} \times V_{max}) + Q_{max}$ (using values for I_{min} and V_{max} from Table 2.2)

$= (3,9 \times 15.4) + 33.4$

$= 93,46$ W

This value will give a good estimate and good overdesign value to make sure of efficient heat extraction from the hot side of the TEC, if the correct heatsink is used.

- The techniques used in the assembly of a thermo-electric (TEC) system are very important. All of the mechanical interfaces between the object to be cooled, and the hot side are also thermal resistive interfaces. This would include air heat conduction. Similarly, all thermal interfaces tend to inhibit the flow of heat or add thermal resistance. When considering assembly techniques every effort should be made to minimise thermal resistance.
- Contact surface tolerance[21] for heat exchanger surfaces should not exceed 25μm. Should there be a need to use more than one module between common plates the height variation between modules should not exceed $\pm$ 19μm. Most TEC assemblies utilise a thermal grease interfaces (aluminium oxide paste). The grease thickness should be held to 25μm with a tolerance of $\pm$ 13μm. Dirt, grit and grime should be minimised; this is very important when aluminium oxide paste is used, due to their affinity for these types of contaminants.
- A form of insulation/seal should be provided between the exchangers surrounding the modules. This material serves several purposes. Since the area

within the module, i.e., the element matrix, is an open DC circuit and a temperature gradient is often present, gas flow which may contain water vapour that could condense, should be minimised.

2.5 Theoretical Analysis

A TEC is constructed by using a number of thermocouples thermally in parallel but electrically in series. The theory in this section is done to explain the formulas[22] that are used to calculate the cooling power (Q_c), the maximum current (I_{max}), the optimum current (I_{opt}) for a specific ΔT, and the optimum voltage (V_{opt}) at I_{opt}. This analysis is thus done, by looking at only one of these thermocouples (one p-n junction). It is assumed that all the thermocouples in the TEC have the same properties.

2.5.1 Miscellaneous Expressions

The following miscellaneous expressions need to be explained :

α: Seebeck co-efficient (volts/°K)

ρ : Resistivity (ohm - cm)

χ : Thermal conductivity (watt/cm.°K)

G: The Geometry value is a value that is derived by dividing the area of the n-, p-block by the length (see Fig 2.3)

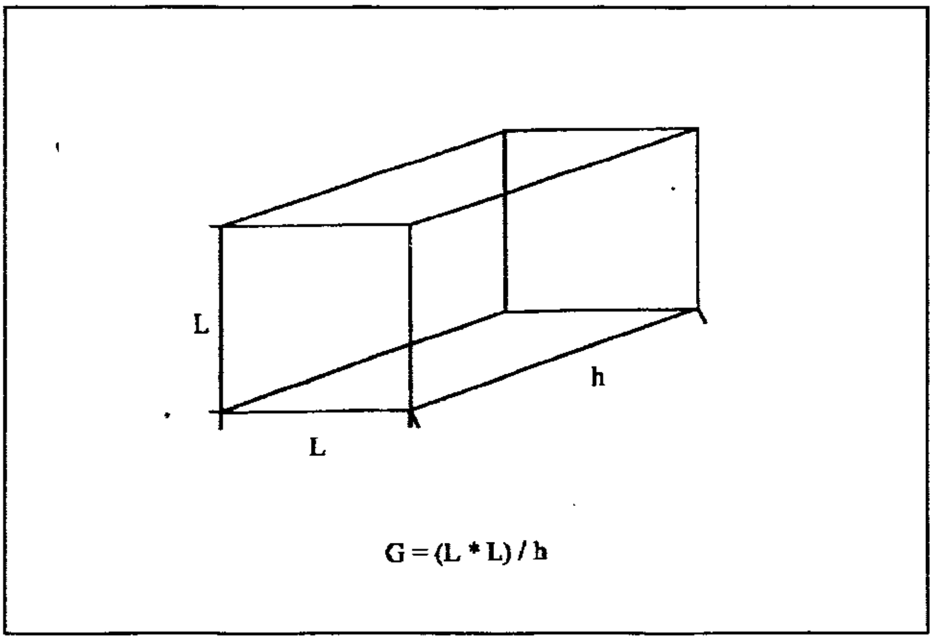

Figure 2.3 The geometry measurement of a p-block or n-block.

Z : The figure of merit (typically, state of the art figures of merit, see Section 2.1, are in the region of $350 \times 10^{-3} K^{-1}$). Temperature difference (ΔT) also has a influence on the Seebeck co-efficient (α), the resistivity (ρ) and the thermal conductivity of the thermocouple.

S : The device Seebeck voltage (volts/°K) is derived from the expression : $S = 2.\alpha.N$; where N is the number of thermocouples.

R : The electrical resistance (ohms) is derived from the expression : $R = 2.\rho.N/G$.

K : The thermal conductance (Watt/°K) between the cold side and the hot side of the TEC is derived from the expression : $K = 2.\chi.N.G$.

ϒ : The average of the hot side temperature (T_h) and cold side temperature (T_c).

2.5.2 Heat Pumped at the Cold side[23] (Q_c)

$$Q_c = \alpha.I.T_c - (I^2.\rho)/(2.G) - \chi.\Delta T.G \quad \text{..... (2.6)}$$

The value for Q_c is derived for only one thermocouple and should a TEC have 127p-n block pares, this value (Q_c) should be multiplied with 254 (127 p,n blocks but 2 × 127 p-n couples, i.e. resistive blocks).

2.5.3 Maximum Current[24] (I_{max}).

$$I_{max} = (\chi.G/\alpha)[\sqrt{(1 + 2.Z.Th)} - 1] \quad \text{.....(2.7)}$$

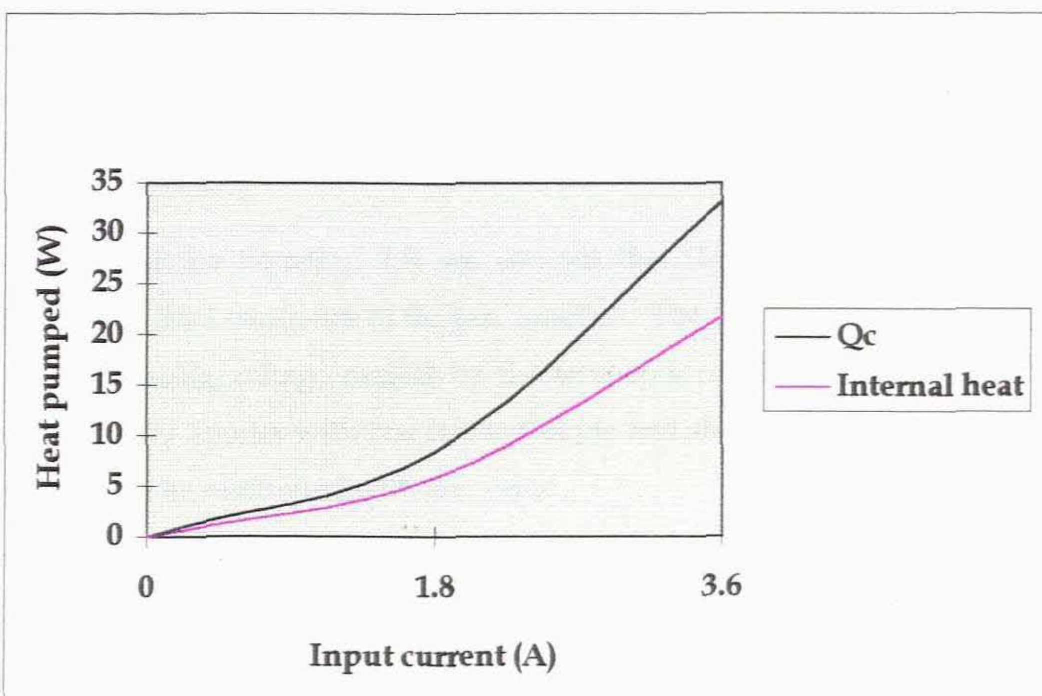

Figure 2.4 The maximum value of I_{max} compared to Q_c, and the negative effect of the I^2R Joule heating effect on Q_c.

The maximum current (I_{max}) is determined where (see Fig 2.4) the value of Q_c is a maximum point of the quadratic Equation 2.6. If the input current (I_{max}) is increased beyond this point, the I^2R Joule heating effect in Equation 2.6 $((I^2.\rho)/(2.G))$ would become to big, resulting in a deterioration of the COP and possible damage to the TEC, due to overheating of the p-n blocks.

2.5.4 Optimum Current[25] (I_{opt})

$$I_{opt} = [\chi.\Delta T.G(1 + \sqrt{(1 + Z.\chi)}]/\alpha.\chi \qquad ..... (2.8)$$

The optimum current is derived from the point where (see Fig 2.4), the total internal heat generated, the I^2R Joule heating component in Equation 2.6 $((I^2.\rho)/(2.G))$ coincides with the total heat pumped (Q_c) at the cold side of the TEC.

2.5.5 Optimum Voltage[26] (V_{opt})

$$V_{opt} = ((I.\rho/G) + \alpha.\Delta T) \qquad ..... (2.9)$$

When we look at the Equation 2.9, we can see that $(I.\rho/G)$ is the optimal current multiplied by the total resistance of the p-n junction. The second part of the equation is the Seebeck opposing voltage caused by the temperature difference between the p-n junction. This only applies with one thermocouple and should more thermocouples be used, V_{opt} should be multiplied with this value.

2.5.6 Co-efficient of Performance (COP)

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

$$COP_{electrical} = Q_c/(I_{opt} \times V_{opt}) \qquad ..... (2.10)$$

Q_c is the heat that is pumped at the cold side of the TEC. I_{opt} (Equation 2.8) is calculated to determine the optimal electrical COP of the TEC. V_{opt} (Equation 2.9) multiplied with I_{opt} to obtain the input power of the TEC for a specified ΔT. From Equation 2.10, $P = V_{opt} \times I_{opt}$ is in Watts and Q_c is in Watts resulting in a unit less value for the COP.

2.6 TEC Performance Handicap

There are two major reasons why a TEC is inferior to the compressor cooling when it comes to great amounts of heat pumping, namely size limitations and internal Joule heating.

The two major companies manufacturing TEC's in the U.S.A. are Melcor and Marlow. Both claim that a TEC is, when taking cost and power consumption into account, competitive for loads up to 200W (this figure is most probably calculated when ΔT = 0°C) providing compactness and the ability to pump small amounts of heat reliably, without moving parts, emitting zero electromagnetic radiation and under variable input power variations. This makes the TEC suitable for applications where a compressor system would not operate reliably. Cost is also an important factor, since in most applications cost plays a mayor role and a design aim is always to keep the cost of the heat exchanger as low as possible. Another factor that is very important is the fact that a TEC uses no gasses that may be harmful to humans or the environment.

The TEC looses its competitiveness due to three problems::

- Internal Joule heat generation
- Small temperature difference between the hot side and the cold side (see Section 3.6.2)
- Thermal conductance in the p-n blocks of the TEC between the hot side and the cold side.

2.6.1 Internal Joule Heating Effect

$Q_c = \alpha.I.T_c - (I^2.\rho)/(2.G) - \chi.\Delta T.G$ ..... (2.6)

The first part of the expression calculates the total amount of heat that is pumped. It is a linear function.

$P_T = \alpha.I.T_c$
$= (V/°K).I.°K$
$= V.I$ Watt
$= P_T$, Watts of heat that is pumped.

The second part of the expression calculates the total internal Joule heating generated in the TEC due to the resistance component in the TEC.

$P = (I^2.\rho)/(2.G)$
$= I^2.R/2$
$= P$, Watts internal heat generated

The last part of the equation is the heat flow between the hot side and the cold side of the TEC. The bigger ΔT becomes, the more heat is conducted from the cold side to the hot side.

$P = \chi.\Delta T.G$
$= W/(cm.°K).cm.°K$
= the heat that flows in the p-n block between the cold side and the hot side (W).
Q_c = Total heat pumped - Joule heating effect - heat conducted between p-n blocks.

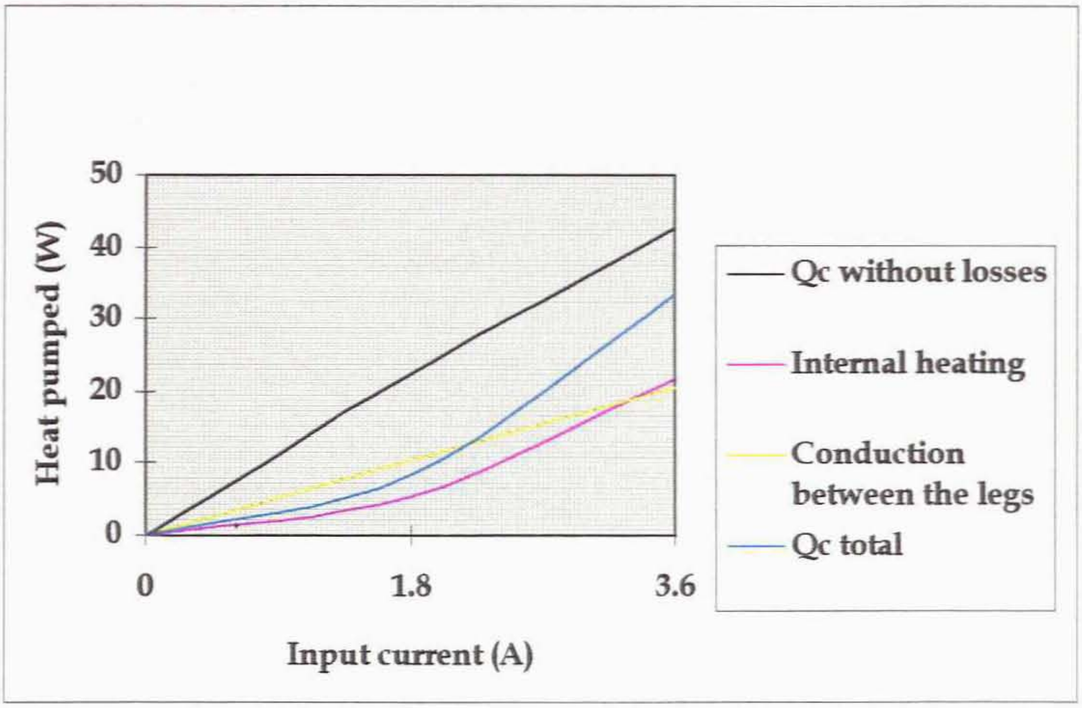

Fig 2.5 The different heat components in the TEC that result in the final quantity of heat (Q_c) that the TEC pumps.

2.6.2 Temperature Difference

The average temperature difference of a single stage TEC is between 64°C - 67°C. The same value for a compressor heat pump is about 110°C. To illustrate why a high temperature difference is important, see Fig 2.6 and Fig 2.7.

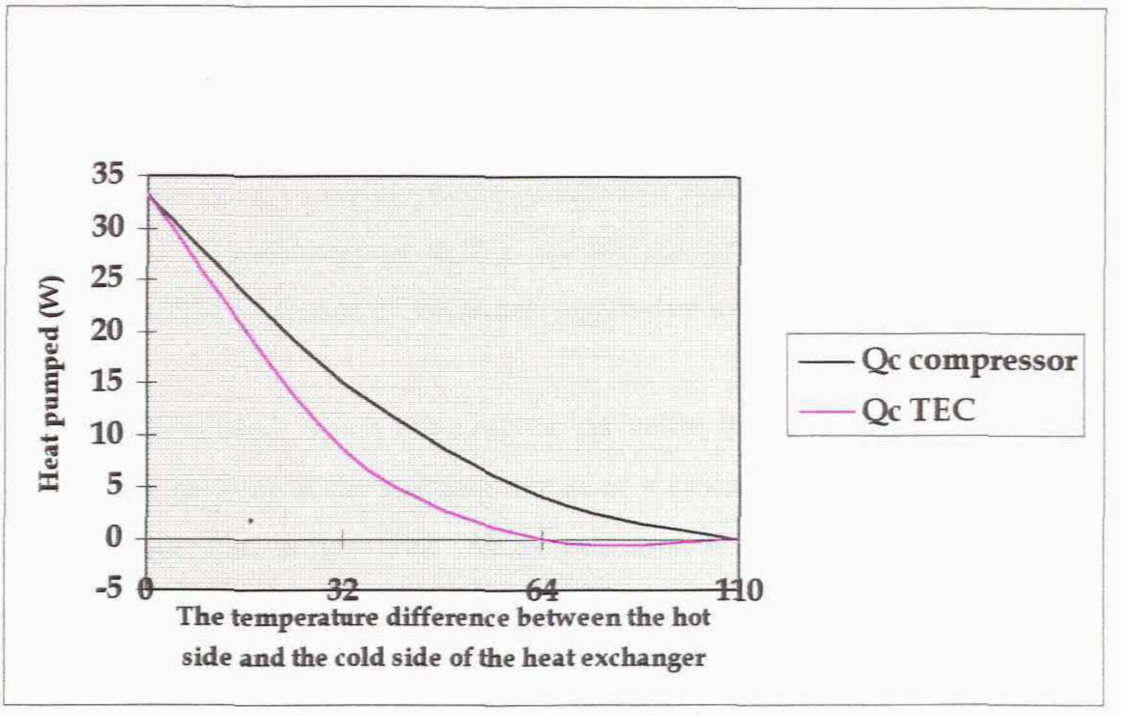

Figure 2.6 The advantage of a large ΔT compared to a lower ΔT in terms of Q_c

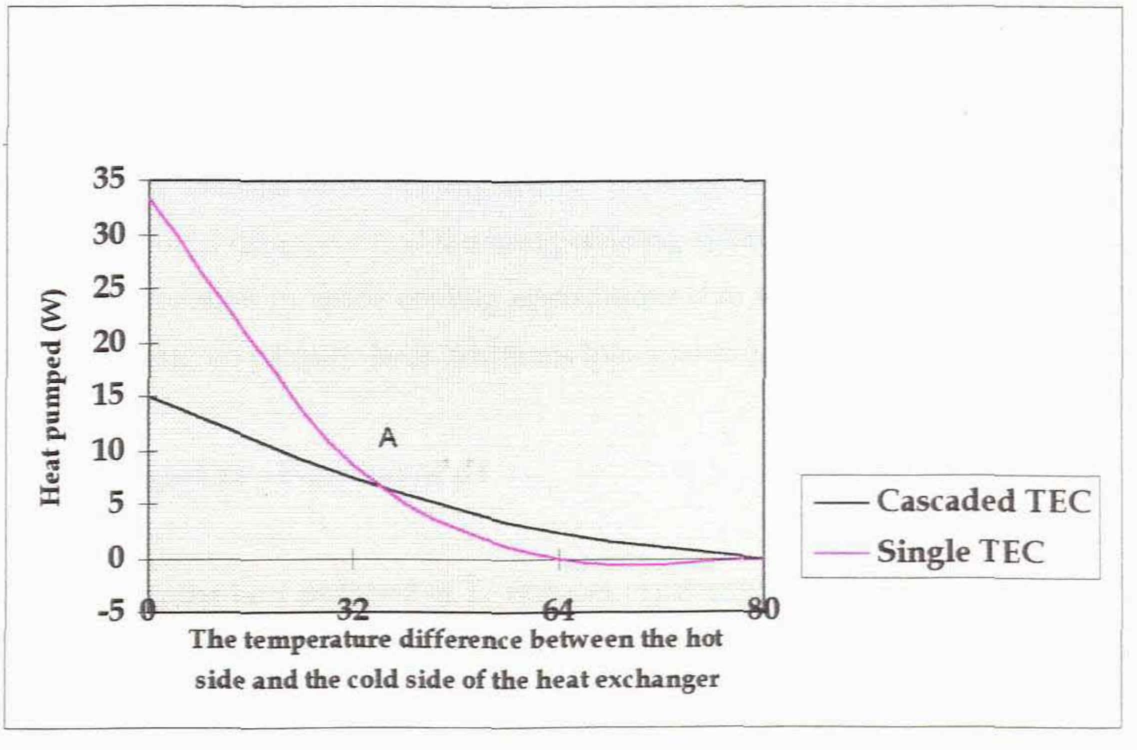

Figure 2.7 The single stage TEC compared to a double stage TEC. Both TEC's have the same input power.

In Fig 2.6, both the cooling systems can pump 33W of heat at $\Delta T = 0°C$ and both consume 60W of input power. When $\Delta T = 0°C$ both systems pump the same amount of heat and will have the same COP (see Equation 2.10). As ΔT increases, the TEC starts lagging in heat pumping capability to the compressor system. If we would look at the COP (co-efficient of performance) in Fig 2.6, the compressor system becomes more efficient than the TEC system as ΔT increases; again due to its superior ΔT.

If a cascaded TEC is used, which has a higher ΔT value, it will use the same amount of input power but because of the stacking nature of a cascade TEC less heat is pumped than a single stage TEC at $\Delta t = 0°C$. As ΔT increases, it will become more and more superior to the single stage TEC as shown in Fig 2.7, see point A. Should ΔT be increase from point A, the cascaded TEC becomes more efficient than the single stage TEC.

As is the case with the compressor system, the cascaded TEC costs about 4 times more than a single stage TEC, if a comparison is made on an equal input power basis.

Unfortunately superconductor technology has not yet advanced to the point where it retains its zero resistance properties at high temperatures. Resistance would be zero resulting in zero internal Joule heating effect. Looking at Equation 2.6, the I^2R Joule heating effect would disappear and the heat pumping ability would be linear, making a TEC more competitive in space cooling applications due to its greater heat pumping capabilities (taking no parasitic heat additions into account).

2.6.3 Heat Pumped as a Function of ΔT

As ΔT increases, the heat pumped at T_c reduces exponentially. It is also interesting to note the effect ΔT has on the COP of a TEC. Take $\Delta T = 25°C$, $15°C$, $5°C$ and consider Table 2.3 and Table 2.4:

Table 2.3

The calculated Q_c at a specific ΔT. Q_c increases as ΔT decreases.

ΔT	Q_c
25°C	12.2W
15°C	15.42W
5°C	18.63W

Table 2.4

The calculated COP for a TEC at ΔT = 25°C, ΔT = 15°C and ΔT = 5°C.

ΔT	COP
25°C	0.77
15°C	1.64
5°C	6.22

When cooling is considered, an advantage is that as ΔT becomes smaller, Q_c becomes bigger and the COP of the TEC increases, so that if the design had parameters designed for a specific ΔT and ΔT becomes smaller, the TEC, initially pumps more heat resulting in a higher efficiency, until the designed ΔT is again reached. This may seem to be a disadvantage but in a design where the heat pump is very small, it is important that the heat exchanger strives to reach the designed ΔT as quickly as possible.

2.7 Newton's Cooling Law

Newton's Law of cooling[27] states that the rate of change of temperature (T) in a cooling body is proportional to the difference between T and the temperature ($T_{surrounding}$) of the surrounding medium.

$$dT/dt = -k(T_{max} - T_{surrounding}) \ldots\ldots (2.11)$$

and $(T_{max} - T_{surround}) = -kt + c$

$$T_{max} - T_{surrounding} = A.e^{-kt} + c$$

where $A = e^c$ at Time = 0 s

2.7.1 Cool-down Curve

To design the TEC a suitable and realistic heat load have to be calculated. Firstly the maximum losses through the insulation of the coolerbox have to be decided. It was decided that these insulation losses should not exceed 1W at ΔT maximum. Since South Africa has a harsh summer, a realistic worst case ambient temperature for simulation and test was 30°C. The desired temperature of the water in the coolerbox was assumed to be about 10°C.

Another important factor is the cooling time. It was decided that a good design aim should be for the coolerbox to cool 1 litre of water within twelve hours from 30°C to 10°C. The heat that has to be removed per hour is thus calculated using Newton's law of cooling. The air temperature (cold side of the heat exchanger) in the coolerbox was assumed to be 8°C.

From the Equation 2.11:

$T_{max} - T_{surround} = A.e^{-kt}$ (negative kt indicates cooling)

at $T = 0$

$30 - 10 = A.e^{-0k}$

$A = 20$

After twelve hours the contents should be at 10 °C therefore

$10 - 8 = 20.e^{-12k}$

$2 = 20e^{-12k}$

$k = -1/12.\ln 2/20$

$k = -0.192$

To estimate the temperature as a function of time:

$T = 20e^{-0,192t} + 8$

2.7.2 Heat Pumping Demand

To estimate the power needed to reduce the 1 litre of water in twelve hours, Equation 2.12 is used[28] (This is the worst case since it is designed to keep foods cool and not cool them down).

$$Q_w = M.C.\Delta T \qquad \text{..... (2.12)}$$

where

M = Specific mass of the body

C = Specific heat of the body

For water

$M = 1kg/l$

$C = 4190\ W/kg°C$

The energy needed to reduce the temperature by 20°C is[29]:

$Q_j = 4190.1.20$

$= 83.8\ kJ$

To convert to Watts for twelve hour operations :

$$Q_w = energy/3600.t\ (W/h) \qquad \text{..... (2.13)}$$

$= 83.8 \text{ kJ}/(3600.12)$

$= 1.94 \text{ W/h}$ (extracted on average for 12 hours)

It can be concluded that twelve hours is needed to cool 1 litre of water from 30°C to 10°C extracting a mean of 1.94W/h for 12 hours. In this Section, it was assumed that the coolerbox is already cooled down and therefore the only additional heat that is introduced, is by the 1 litre of water. The amount of heat that is pumped, relating to time, would have the same gradient since the temperature difference is directly proportional to the ability of the TEC to pump heat. We know that as the temperature difference (ΔT) increases, Q_c decreases (see Section 2.6.3). This relationship is nearly exponential and would give a good estimate of the heat pumping capabilities of the coolerbox.

2.8 Total Heat Pumping Demand

For cooling, the heat exchanger has to pump the heat losses through the polystyrene insulation and the heat from the 1 litre of water. We estimated a heat loss though the insulation of no more than 1W for a worst case.

Total heat pumping demand = (heat loss + cooling + contact surface loss*)/h

= (1W + 1.94W + 2.61W)/h

= 5,55 W/h

(* see Section 3.5.5)

The amount of heat that has to be pumped by the coolerbox per hour for twelve hour is 5.55W. The heat pumping parameter for a TEC is 5.55 W/h, at $\Delta T = 20$°C for the water and $\Delta T = 25$°C for the heat exchanger when $T_h = 35$°C.

(It must be remembered that the hot side of the TEC is at $T_h = 35$°C but the ambient temperature may vary around 30°C).

2.9 Theoretical Estimation of the TEC Size

The size of the TEC was calculated after the heat load of the coolerbox was determined by the theory, calculating the V_{opt}, I_{opt}, Q_c and COP. The same heat load was used and the calculation the inputs of V_{opt}, I_{opt}, Q_c and COP were determined, using the Marlow guide (see Annexure B).

Table 2.5 illustrate that the theoretical results are the same as the results obtained using the Marlow Guide. The Marlow method of calculating the TEC size is graphical resulting in a less complicated calculation than the theoretical calculation. In Chapter 3, these results will be compared with the measured results of the tests done using a simulator. The simulator will simulate the TEC performance.

Table 2.5

The theoretically calculated results compared to the results obtained from the manufacturers guide

Theoretical calculated results				Manufacturers calculated results			
V_{opt}	I_{opt}	Q_c	COP	V_{opt}	I_{opt}	Q_c	COP
5.4V	1.24A	7.1W	1.1	5.6V	1.3A	7.1W	0.98

2.10 Conclusion

Table 2.5 clearly illustrate that there is very little difference in the values that were calculated for I_{opt}, V_{opt}, Q_c and the COP. The small differences could be due to small errors following the manufacturers steps where graphs are used as illustrated in Annexure B.

It can be concluded that allowing for a $\Delta T = 25°C$ for the heat exchanger, the TEC would be more than able to pump the required 5.55W (see Section 2.8) of heat with a very favourable COP of nearly one (see Tables 2.5). According to the results obtained, the TEC heat exchanger is quite efficient, operating at the same efficiency as a electrical heater.

CHAPTER 3

Construction of Test Apparatus and Tests

3.1 Introduction

To design a coolerbox, it is important to determine if the heat exchanger that is used would operate adequately under the designing conditions. For this project a TEC heat exchanger was used. In sizing the TEC (determining the heat pumping capability of the TEC), it was considered to be important to see if the manufacturers performance claims for the TEC were accurate.

This is important, because should the TEC be undersized, performance of the coolerbox under adverse conditions would suffer. Should the TEC be oversized, it would result in a high power consumption appliance that is not ideally suitable for small PV systems.

A simulator and a 20V variable DC power supply were constructed for this project. The intention was to see if the performance data on the TEC supplied by the manufacturers are measured under ideal conditions, thus resulting in elevated performance data that could lead to a poor heat exchange design due to undersizing. Secondly, it was important to see how the TEC coolerbox would operate in adverse conditions, so that its performance could be predicted under the varying temperature conditions and varying input power conditions associated with PV systems.

The purpose of the power supply that was constructed was to provide enough current to the water pump, cold side heater, hot side heater and the fans in the simulator. A second power supply with a very low ripple factor (2%) was used to power the TEC during the tests. This was done to minimise the Joule heating effect in the TEC caused by the AC component superimposed on the DC current. (See Annexure C and D for a detailed explanation on the design and operation of the powersupply and the simulator).

3.2 Method of Testing

A TEC was sandwiched between the hot side and the cold side heat exchanger and pressure was applied as described in Annexure D, Section 1.2. The exposed side of the cold side heat exchanger was covered so that it sat cocooned in polystyrene. Only one side of the cold side aluminium block was exposed to air. This side was the side on which the TEC would be connected. The hot side was exposed to ambient air temperature, as would be the case in the prototype coolerbox, and aluminium oxide paste was applied on the contact surfaces to improve the heat transfer. This set-up simulates as accurately as possible the application conditions in the coolerbox.

3.2.1 Calibration

The simulator, the two power supplies and the two multimeters were switched on for an hour before any measurements were taken. This was done so that the components of the devices could reach their operating temperatures and settle to prevent drift. A Roman X86 calibrator[30] was used to calibrate the op-amps on the simulator. The multimeters were checked to be in their operating range with tolerances of $\pm 0{,}1\ \%$.

The LM 35 temperature sensors[31] were guaranteed by the manufacturers to give a maximum error of ± ¾°C over their range of -55°C to +125°C. They are not adjustable but a comparison between them (testing them tied together) showed that they indicated the same temperatures with a 0.2% diviation for the worst sensor over the 0°C to 50°C range necessary for testing, and therefore were found to be accurate.

3.2.2 Testing Procedure

The full test was done on a Marlow[32] DT1049 TEC. Limited tests were done on a Melcor[33] manufactured TEC as illustrated in Table 3.3. This was done to see how the performance figures of the two manufacturers would compare.

In the final tests, the simulated hot side temperatures for which the Marlow product was tested were 25°C, 30°C, 40°C and 50°C. The TEC input current was varied from 0,5A to 2,5A, in increments of 0.5A (It should be noted that although the maximum input current specified by the manufacturer is 3.6A, this value could not be reached, since the input voltage would have been higher than the specified 14.8V (see Annexure E) and for fear of damaging the TEC, measurements were taken using a maximum upper input voltage of 14.8V). The cold side heater input current in the test started at 0A and was incremented by 0.5A until $\Delta T = 0°C$ between the hot side and cold side of the TEC, at the maximum input power supplied to the TEC. This was done due to possible use of a powersharing regulator where the power regulation would be based on a pulse width modulation technique, using the square wave to divide the current. The current is important, since the TEC input current is directly proportional to its heat pumping capability.

The test procedure was conducted as follows:

1. The hot side temperature of the simulator was adjusted at 25°C and allowed to settle.

2. No heat was induced in the cold side heat exchanger, in other words the cold side heater was not activated.

3. The TEC input current was adjusted to 0.5A and kept constant.

4. The temperature was monitored untill it stabilised and then it was recorded for both the hot side and cold side of the TEC.

5. The TEC input voltage was also recorded as well as the cold side heater voltage.

6. The following were calculated:
   a. Cold side heater input power (in other words the heat induced)
   b. TEC input power
   c. The COP was then calculated (COP = Q_c/input power)
   d. The temperature difference was calculated as $\Delta T = T_{hot} - T_{cold}$.

7. Steps 2 - 6 were repeated until ΔT = 0°C between the cold side and the hot side of the TEC. When ΔT = 0°C, the input power was incremented by 0.5A. Steps 2 - 6 were then repeated.

8. Steps 2 - 7 were repeated until the input voltage to the TEC reached 14.8V, which was the maximum input voltage specified by the manufacturer. The simulator hot side temperature was adjusted to 30°C, 40°C and 50°C respectively, as in step 1, and the whole process was repeated.

As will be seen in the results (see Annexure F), temperatures down to 0°C but not below, were measured. Since the coolerbox will operate at between 8°C to 10°C inside temperature, it was deemed not necessary to measure lower than 0°C. It was decided that since the coolerbox is not a freezer, it was not necessary to measure temperatures less than 0°C.

3.3 Measured Results

The experiments were done as described in the previous section. The results were tabled in the following columns:

Table 3.1

Table structure that was used to tabulate the TEC performance results.

TEC Current	TEC Voltage	TEC Power	Heater Cold Side Current	Heater Cold Side Voltage	Heater Cold Side Power	Temperature Cold Side	Temperature Hot Side	ΔT	COP
a.	b.	c.	d.	e.	f.	g.	h.	i	j.

A detailed explanation of Table 3.1 can be found in Annexure F.

3.3.1 Calculated results

The calculated results that were expected were determined by using the following equations as discussed in Section 2.5:

$Q_c = 2.N[\alpha.I.T_c - I^2.p/2.G - \chi.\Delta T.G]$ ..... (from 2.6)

$I_{max} = (\chi.G/\alpha).(\sqrt{(1 + 2.Z.T_h)} - 1)$ ..... (from 2.7)

$I_{opt} = [(\chi.\Delta T.G)(1 + \sqrt{(1 + 2.\Upsilon)}]/\alpha.\Upsilon$ ..... (from 2.8)

$V_{opt} = 2.N[(I_{opt}.p)/G + \alpha.\Delta T]$ ..... (from 2.9)

3.4 Performance Results

When the performance of a TEC is specified by the manufacturer, tests are conducted under ideal conditions. Typical test conditions are in nitrogen gas at 101.3kPa. This is done to isolate the TEC from the environment and air containing water vapour. Minimum performance losses occur under these conditions.

When a TEC is applied in a heat pumping capacity, it is exposed to all forms of parasitic losses (see Section 2.3.1) and these losses influences the performance of the TEC. A comparison between the manufacturer's specifications and the tested results follows in Table 3.2(a) for a DT 1049 manufactured by Marlow and Table 3.2(b) for a CP1-0-127-05L manufactured by Melcor (data is specified with T_h = 25°C).

Table 3.2(a)

Specified performance data compared to measured performance data.

(Marlow DT 1049)

Parameters	Specified	Measured	% Similarity
ΔT	64°C	44°C	69%
Q_{max}	33.3W	20.58W	62%
I_{max}	3.6A	3.18A	88%
V_{max}	14.8V	14.37V	97%
COP	0.63	0.45	71%
TEC power consumption	53.28W	45.7W	86%

Table 3.2(b)

Specified performance data compared to measured performance data.

(Melcor CP 1-0-127-05L)

Parameters	Specified	Measured	% Similarity
ΔT	67°C	49°C	73%
Q_{max}	33.4W	23.6W	71%
I_{max}	3.9A	3.79A	97%
V_{max}	15.4V	14.4V	94%
COP	0.56	0.43	77%
TEC power consumption	60.1W	54.6W	76%

As mentioned earlier (see Section 3.2.2), the maximum input current was limited due to the maximum input voltage of the TEC's. It is quite apparent, that ΔT and Q_{max} are values that are quite over optimistic and if these values were used in designing the coolerbox, it would result in a poor performing coolerbox. For this particular application, the tests were done with the most effective isolation method. This heat exchanger method (and the DT1049 TEC) will be applied in the coolerbox design, and therefore, the coolerbox design could be done using the data in Table 3.2(a). The percentage similarity for the Melcor CP 1-0-127-05L for ΔT and Q_{max} is 73% and 71% respectively. These figures are higher than the figures registered for the Marlow DT 1049. This is due to the fact that losses become less evident as a ratio, as Q_c becomes larger.

In general, the Melcor product proved to be closer to specification although the Marlow product had a better COP of 0,45 compared to the Melcor product COP of 0,43. The TEC with the highest COP was chosen for the coolerbox heat exchanger which is the DT 1049 from Marlow.

3.5 Discussion of Results

There are three major points of conflict between the calculated and measured values. They are the input voltage, heat pumping power and the COP.

3.5.1 Input Voltage

The input voltage to the TEC varied as ΔT increased due to the influence of the reverse Seebeck effect. As ΔT increases, so does the opposing Seebeck voltage having a decreasing effect on the input voltage in changing the resistance of the TEC, resulting in a slight drop in voltage if the input current to the TEC is kept constant.

Table 3.3

The percentage variation between the calculated and measured input voltages when T_h = 25°C, 30°C, 40°C and 50°C.

I_{TEC}	T_h = 25°C	T_h = 30°C	T_h = 40°C	T_h = 50°C
0.5A	17.2%	27.9%	31.5%	40%
1A	17.7%	29.3%	35.7%	42.3%
1.5A	18.1%	35.6%	36.2%	45.6%
2A	20.1%	37.7%	37.5%	46.7%
2.5A	22.2%	40.8%	40%	58.6%
2.68A	-	-	-	58.7%
2.95A	-	45.2%	43.8%	-
3.18A	25.7%	-	-	-

This change in input voltage was evident over the whole range of tests at T_h = 25°C, 30°C, 40°C and 50°C. As T_h increases above 25°C, a rapid increase of input voltage calculation error occurs. This may bedue to the fact that the equations assume T_h = 25°C and that no heat losses occur through atmosphere, damp and other factors. Another possible reason may be that ΔT will become greater as T_h (hot side temperature) increases and this is made possible by the fact that the temperature difference between T_c (cold side temperature) and the ambient temperature becomes less, allowing for a larger ΔT value and, also a larger reverse Seebeck voltage and thus a greater change in the error of the calculation. The values in Table 3.4 may decrease or increase for other types of installations where more or less heat losses are experienced.

3.5.2 Heat Pumped (Q_c)

The measured heat pumped compared to the calculated heat pumped showed the same variations (as was the case for the input voltage). As T_h increased and ΔT decreased to zero, the variation became bigger till it reached a peak at T_h = 40°C and ΔT = 0°C (a variation error of 58,8%).

Table 3.4

The percentage variation between the calculated and measured values of Q_c when T_h = 25°C, 30°C, 40°C and 50°C.

I_{TEC}	T_h = 25°C	T_h = 30°C	T_h = 40°C	T_h = 50°C
0.5A	5.7%	24.8%	18.9%	19.2%
1A	9.4%	22.9%	22.9%	18.7%
1.5A	12.2%	26.6%	27.7%	28.6%
2A	14.8%	41.4%	40.4%	33.1%
2.5A	15.9%	45.3%	38.7%	48.8%
2.68A	-	-	-	52.8%
2.95A	-	56.9%	58.8%	-
3.18A	24.7%	-	-	-

The heat pumped (Q_c) vs ΔT is not linear. This is due to the I^2R Joule internal heating effect in the TEC. The difference between the calculated values and the measured values is a result of:

- $T_h \geq 25°C$
- T_c will never reach the ΔT value of 64°C because of parasitic heat additions at the cold side (non-ideal conditions).

In the quadratic equation : $Q_c = ax^2 + bx + c$ (this is just an explanation using the same quadratic form as Equation 2.6), the gradient of the equation (a) is nearly equal for both calculated and measured values but the value of (c), is determined by the parasitic heat additions, contact losses and other losses. Therefore, for the best Q_c results, good cold side isolation (in the heat exchanger) is necessary from the ambient environment, limiting the amount of parasitic heat additions. This is important and it must be ensured that all the heat that is pumped is heat from the coldsink. This will ensure better system performance, increasing the COP of the heat exchanger.

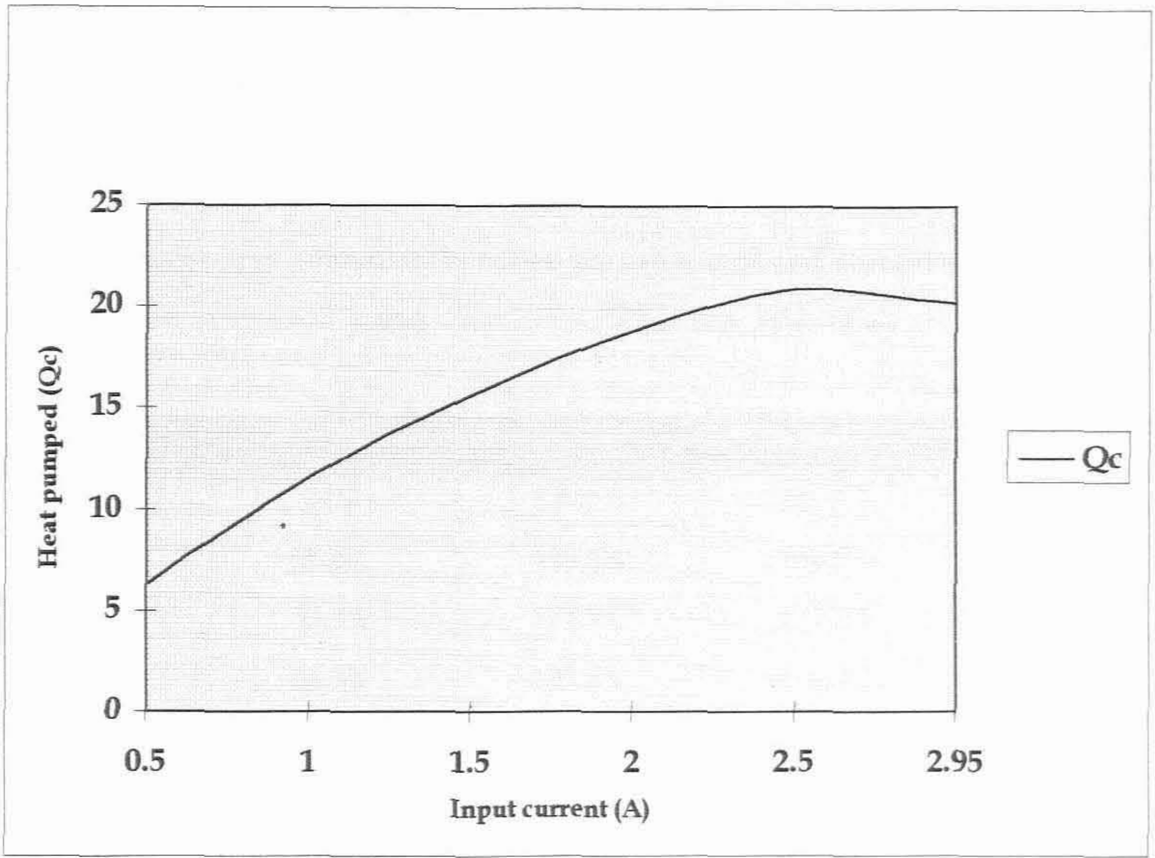

Fig 3.1 The amount of heat pumped (Q_c) by the TEC at T_h = 25°C and ΔT = 0°C.

When we look at Fig 3.1, it is interesting to note that the maximum heat that is pumped occurs at 2.5A and not at 2.95A. This is the effect of the internal I^2R Joule heating effect.. Another interesting point to note is that as Q_c increases from I_{TEC} = 2A to 2.5 A, the heat pumping capability increased by 13.4%, whereas the input current rose by 20%. Thus, more power was wasted in producing a little more cooling power.

3.5.3 Co-efficient of Performance (COP)

The COP of a system is the measure of efficiency of the system. For the sake of this experiment, the COP was taken as COP = Q_c / Input power. The same problem existed for the COP as for the V_{in} and Q_c. The COP values were optimistic. As mentioned earlier, if the cooling system was designed mathematically, the values reached for V_{in} and Q_c would produce an inaccurate COP for the cooling system. This is a huge problem since the result is usually an under designed system due to the over optimistic results for V_{in}, Q_c and COP. The following table illustrates this:

Table 3.5

The percentage over estimation between the calculated and measured values for the COP when T_h = 25°C, 30°C, 40°C and 50°C.

I_{TEC}	T_h = 25°C	T_h = 30°C	T_h = 40°C	T_h = 50°C
0.5A	24%	159.5%	156.5%	166.5%
1A	28.6%	159.4%	166.8%	169.1%
1.5A	35.5%	171.1%	173.2%	185.8%
2A	38.3%	194.8%	192.5%	175.5%
2.5A	41.6%	204.6%	194.1%	235.7%
2.68A	-	-	-	243.6%
2.95A	-	228.3%	227.7%	-
3.18A	56.8%	-	-	-

It is quite noticeable how the accuracy deteriorates as T_h increases. This complicates the design and just emphasises the fact that a coolerbox should be designed for a maximum temperature (T_h), of at least 35°C for a worst case condition.

When we look at the following Table 3.6 it can be seen that as ΔT increases, the measured and theoretical results differ by an even greater value.

Table 3.6

Comparison between measured and calculated COP at $T_h = 40°C$

ΔT	Measured COP	Calculated COP	Percentage over Estimation
49°C	0	0.38	∞
47°C	0.0231	0.38	1645%
39.3°C	0.092	0.39	423.9%
27.8°C	0.208	0.57	274%
12.1°C	0.38	0.85	223.6%
0°C	0.48	1.1	229.2%

It can be concluded that the closer to $\Delta T = 0°C$, the more accurate the calculated COP is, relative to the value at $\Delta T = 49°C$. At $\Delta T = 0°C$, theoretically it is possible to reach a COP of 1,1. The TEC in its installed capacity as a heat exchanger for a coolerbox would yield a measured COP of 0.48. This relates to an over optimistic heat pumping capability of 229.2%. When the calculated COP at $\Delta T = 40°C$ is used as a design parameter, the design would be under designed by a factor of 4.239 which would relate to a performance decrease factor of 0.236 of the expected or designed cooling capacity.

3.5.4 A comparison between the heat pumped at $T_h = 30°C$ and $T_h = 40°C$

Fig 3.2 illustrates the effect the ambient temperature has on the performance of the TEC. Q_c increased slightly as T_h increased. When $T_h = 30°C$ the most heat that was pumped occurred at 2.5A. When $T_h = 40°C$, more heat was pumped and this resulted in an increased input current. The most heat was pumped at 3.18A.

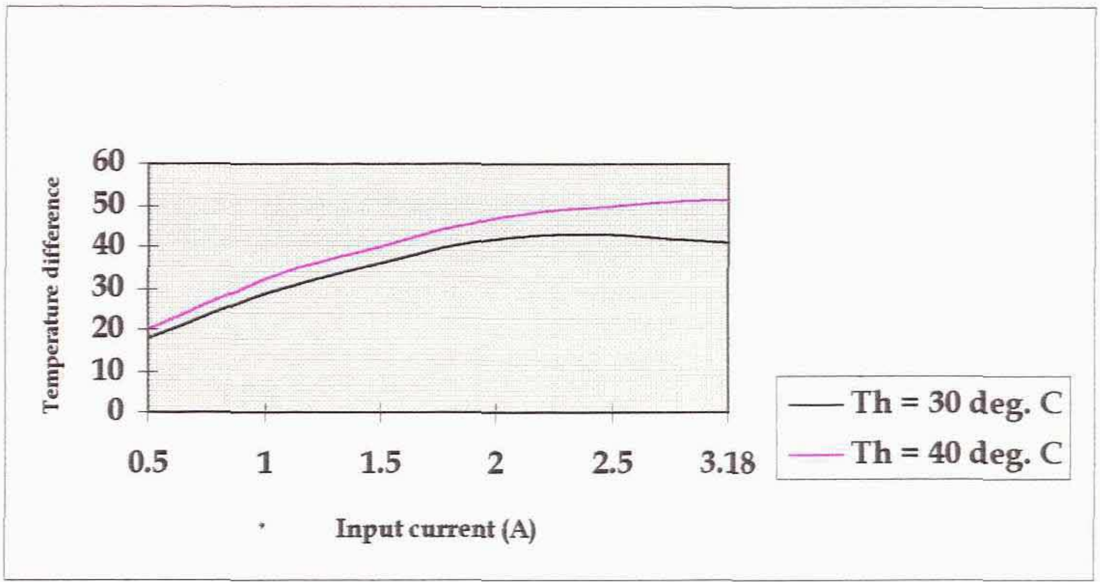

Fig 3.2 The effect the parasitic heat additions has as T_h increases.

The reason why the TEC could reach a larger ΔT at $T_h = 40°C$ was because when the tests where conducted, a hot side temperature was simulated by increasing the hot side cooling water temperature. The result was that the difference between the cold side and the ambient temperature (partly responsible for the parasitic heat additions) was less making it possible for the TEC to reach a slightly higher value for ΔT.

3.5.5 Average Heat Pumped per °C

As can be seen from Fig 3.3, there is a constant amount of power being pumped per degree Kelvin when I_{input} is constant. For example, at 0,5A, at $\Delta T = 0°C$, $Q_c = 6{,}22W$ (see Annexure F). For each degree ΔT increases 0,38W of $Q_{c\ max}$ is subtracted. The same applies to 1A, 1,5A, 2A, 2,5A and 3,18A.

It can therefore be assumed that when $Q_c = 0W$ (in this case $I_{input} = 0{,}5A$), $\Delta T = 16{,}5°C$. Each degree Kelvin would then represent 0,38W/°C of heat pumped.

Table 3.7

The W/°C that is pumped at different TEC (DT 1049) input currents.

TEC Input Current	Watt/°C Heat Pumped
0.5A	0.38W/°C
1A	0.39 W/°C
1.5A	0.43 W/°C
2A	0.44 W/°C
2.5A	0.46 W/°C
3.18A	0.51 W/°C

A good average of expressing one degree centigrade in terms of heat pumped (Watt), would be to get the average W/°C at different I_{inputs} supplied to the TEC. Therefore,

Average W/°C = (0.38 + 0.39 + 0.43 + 0.44 + 0.46 + 0.5)/6
= 0.435W/°C

The value of 0.435W/°C was used when the heat pumping losses for the coolerbox due to contact resistance were calculated.

3.5.6 Comparison between TEC Application Sizing Methods and Measured Results

The comparison was done using the parameters $\Delta T = 25°K$ and $T_{h.} = 35°C$.

Table 3.8(a)

Comparison between the calculated, manufacturers and measured performance data.

Theoretical Estimations				Manufacturers Estimates				Measured Results			
V	I	Q_c	COP	V	I	Q_c	COP	V	I	Q_c	COP
5.4V	1.24A	7.1W	1.1	5.6V (*)	1.3A	7.1W	0.98	9V	2A	7.1W	0.39

(*) The average voltage between minimum and maximum voltage was taken.

Table 3.8(b)

Comparison between the calculated, manufacturers and measured performance data but the parasitic losses were subtracted from the calculated and measured data.

Theoretical Estimations				Manufacturers Estimates				Measured Results			
V	I	Q_c	COP	V	I	Q_c	COP	V	I	Q_c	COP
5.4V	1.24A	7.1W (#)	1.1	5.6V	1.3A	7.1W (#)	0.98	5.5V	1.1A	3.5W	0.58

(#) For the sake of this comparison, the parasitic heat of 3.61W (see Section 2.8) was subtracted from 7.1W for Q_c. The COP was re-calculated accordingly.

From Table 3.8(a) it was quite evident that the theoretical estimation was quite optimistic compared to the measured values. In Table 3.8(b), the values for the calculated and the measures values of Q_c were adjusted by subtracting the parasitic heat losses to simulate a Q_c of 3.5W. It is now quite evident that the measured values were very close to the calculated values, suggesting that the calculated values should be manipulated to take the parasitic heat losses into account. Since $P = I^2 \times R$ and I is constant, then P is directly proportional to R. This R value could be due to power cable losses and unaccounted contact resistance losses or other parasitic heat additions.

This just emphasises the problem which arised from the fact that the TEC data from the manufacturers is optimistic (under ideal conditions) and highlights the need to determine parasitic cooling losses (or heat additions).

3.5.7 Revised Equations for the Optimum Input Current (I_{opt}) and the Optimum Input Voltage (V_{opt}) to Compensate for $Q_{paracitic}$

From Section 2.3 and 2.8 it is clear that many factors influence the Q_c of a TEC when it is used in any application, but mainly Q_c is influenced by ΔT. For any application Q_c should be calculated for optimum system performance. It is thus very important to determine the parasitic Q_c losses which are due to conduction, convection or radiation. The input power to cope with the amount of heat to be pumped and the losses is reflected in I_{opt} and V_{opt}. If the correct values of I_{opt} and V_{opt} can be determined, the installations should be performing at an optimal point.

The value of Q_c can be obtained by applying Equation 2.6. The additional heat that has to be pumped is compensated for by the input power of the TEC. The heat load is added to Equations 2.8 and 2.9. The same ratio of V/I from the TEC is used; in this case for the DT 1049 it is 14.8/3.6 = 4.11Ω (the voltage being the maximum specified input voltage and current supplied to the TEC). The heat load would use the same V/I ratio when the additional current and voltage is used.

3.5.7.1 New I_{opt} Incorporating the Parasitic Heat Additions

The parasitic heat addition is 3.61W (see Section 2.8). For the DT 1049 the ratio of V/I is 4.11.

The additional input current would be:

$P = I^2 \times R$

$I = \sqrt{(P/R)}$

$= \sqrt{(3.61/4.11)}$

$= 0.94A$

The same method provides a parasitic current for the CP1-0-127-05L of 0.96A. This value is now simply added to Equation 2.8.

Note: If I_{opt} plus the parasitic current is greater than the I_{max} of the specific TEC, a TEC with a larger heat pumping capacity should be chosen.

3.5.7.2 New V_{opt} Incorporating the Parasitic Heat Additions

The parasitic heat addition is 3.61W. For the DT 1049 the ratio of V/I is 4.11. The additional input voltage would be:

$P = V^2/R$

$V = \sqrt{(P \times R)}$

$= \sqrt{(3.61 \times 4.11)}$

$= 3.85V$

The same method provides a parasitic voltage for the CP1-0-127-05L of 3.78V. This value is now simply added to Equation 2.9.

3.5.7.3 Comparison to See if I_{opt} and V_{opt} Relates to the Tested Results

The value of the input power of the TEC is now adjusted to accommodate the additional heat additions. It is therefore clear that the value of Q_c is actually unchanged and

Equation 2.6 could be used to determine the amount of heat that is pumped. It should be mentioned that Equation 2.6 could still be optimistic but at least, the parasitic heat additions are now accounted for (or the total heat load).

Using Equation 2.7 to determine the I_{max} value:

$I_{max} = 3.53A$

Using Equation 2.8 to determine the I_{opt} value:

$I_{opt} = 2.6A$

Using Equation 2.9 to determine the V_{opt} value:

$V_{opt} = 11.18V$

Table 3.9(a)

Tested results of the CP 1-0-127-05L (Melcor) compared to the re-calculated results at $\Delta T = 49°C$ at $T_h = 25°C$.

Measured Values		Theoretical Estimations		New Estimations	
I_{opt}	V_{opt}	I_{opt}	V_{opt}	I_{opt}	V_{opt}
3.79A	14.4V	2.6A	11.18V	3.56A	14.96V

Table 3.9(b)

Tested results of the DT 1049 (Marlow) compared to the re-calculated results at $\Delta T = 44°C$ at $T_h = 25°C$.

Measured Values		Theoretical Estimations		New Estimations	
I_{opt}	V_{opt}	I_{opt}	V_{opt}	I_{opt}	V_{opt}
3.18A	14.4V	2.32A	9.99V	3.26A	13.84V

Table 3.9(c)

Tested Results of the DT 1049 compared to the re-calculated Results at $\Delta T = 25°C$ at $T_h = 35°C$.

Measured Values		Theoretical Estimations		New Estimations	
I_{opt}	V_{opt}	I_{opt}	V_{opt}	I_{opt}	V_{opt}
2A	9V	1.24A	5.4V	2.18A	9.25V

For the new estimated input values, the parasitic heat addition values for V and I have been calculated as in Section 3.5.7.1 and 3.5.7.2.

Table 3.10(a), 3.10(b) and 3.10(c) illustrate how realistically or accurately the input values for the TEC could be determined, by using the above method (see Section 3.5.7).

3.5.7.4 Calculating I_{opt} and V_{opt} for the Prototype Coolerbox

To determine the optimum input power for the coolerbox, the required 5.55W was calculated as the total heat load, and 3.61W as the parasitic heat load (see Section 2.8). The parasitic heat load was then converted to I_{opt} and V_{opt} values (see Section 3.5.7.1 and 3.5.7.2) and found to be 4.77V and 1.16A respectively. The theoretical estimations for I_{opt} and V_{opt} were then adjusted and found (see Table 3.10) to be 10.17V and 2.40A.

Table 3.10

The theoretical values compared to the prototype coolerbox adjusted values.

Theoretical Estimation		Prototype Optimum Input Values	
I_{opt}	V_{opt}	I_{opt}	V_{opt}
1.24A	5.4V	2.40A	10.17V

Using the method to determine I_{opt} and V_{opt} in Section 3.5.7, it is calculated that the input power to the prototype coolerbox will be 10,17V at 2.40A at its optimum point. The input current is the important value to comply with since it is responsible for the heat pumping capacity. Every type of TEC could have a slightly different resistance (R) which could make the voltage (V) change which in return will change the ratio of V/I but this will only be slight change which can be ignored.

3.6 Conclusion

3.6.1 Measurements

All the testing equipment was calibrated (see Section 3.2.1). The two multimeters showed a full scale accuracy of $\pm$ 0.1%. The amplifying and differential circuits were calibrated and accuracy varied by $\pm$ 1%. Other factors which were ignored were thermal contact resistance and the small losses (typically less than 0,1%) in the power cables. Voltage measurements were measured as close as possible to the device, to eliminate the error caused by the volt drop over the cable.

The test results could be seen to be representative of this type of installation as used in the prototype coolerbox (see Table 3.8(b)). The Q_c values are very realistic and the performance measured by the TEC (DT 1049) would give very accurate performance results of the TEC under the installation conditions of the prototype coolerbox.

3.6.2 Results

From the performance tests (see Section 3.4) it is quite evident that the TEC in an installed application, could prove to be a disappointment if the conditions under which the TEC operates are not taken into account. Parasitic heat additions and contact losses are the main problems and although the specifications suggest a ΔT = 64°C, this was impossible to obtain.

This method described in Section 3.5.7, proved to work well (see Table 3.9). The re-calculated values for I_{opt} and V_{opt}, came very close to the results measured in the simulator. It is therefore an easy way to estimate with relative accuracy, the optimal values for I_{opt} and V_{opt}.

From the tests done, it became clear that in designing and constructing a TEC heat pump the following are important:

i. Since ΔT is crucial (for the DT 1049, $\Delta T = 44°C$) and is a definite function of Q_c, the hot side temperature should be kept as close as possible to the ambient temperature.

ii. The thermal contact resistance between the TEC, the *coldsink* and *heatsink* should be as little as possible.

iii. A TEC should be chosen so that it will operate at about 60% - 75% of its maximum input power to ensure an optimum COP and cooling performance. In this case using a DT 1049, I_{opt} is 2.40A giving a value of 67% of the maximum specified input power (see Table 3.2(a) and Table 3.10).

iv. The theoretical estimation gives very close values for Q_c, (if the method in Section 3.5.7 is followed).

v. The heat load should be calculated. A very comprehensive analysis should be done allowing for as little error as possible. A worst case scenario should be chosen for the design of the heat exchanger. If the heat load is calculated the optimum input power (see Section 3.5.7) for the TEC can be calculated.

Thermal contact losses (see Section 1.5.5) between the TEC and the hotsink or coldsink surfaces is between 2°C - 3°C per contact surface. In the analysis 3°C (6°C for both surfaces) was used. It was calculated that the losses per contact surface on average (see section 3.5.5) were 0.435W/°C.

CHAPTER 4

Coolerbox Design and Testing

4.1 Introduction

A coolerbox is an appliance that keeps food cool. It is not designed to freeze. The writer decided that the average temperature inside the coolerbox should be about 10°C. A TEC is used as the cooling medium (12VDC), and a PV system is used to power it.

Figure 4.1 The prototype coolerbox, using a water-to-air hot side heat exchanger.

The external dimensions of the coolerbox were designed as small as possible, to optimise space utilisation in a dwelling. The bottom part of the coolerbox is utilised as a storage space for the batteries of the PV system. Because power consumption is a great factor, the cooling volume of the fridge should be small. The smaller the coolerbox, the less power it will consume. It is thus very important to design a coolerbox as small as possible and at the same time as functional as possible.

4.2 Design Parameters

Before the coolerbox was designed and built, basic important parameters had to be set. The most important aims that had to be met were:

- Low power consumption
- High efficiency
- Low cost

4.2.1 Low Power Consumption

Low power consumption was achieved by using a very low power cooling fan; a water pump and a DT 1049 TEC. The power consumption was calculated and a realistic figure at T_h = 40°C was determined:

TEC consumption	2.4A x 12,00V = 28.8 W
Pump & cooling fan	0,6A x 12,00V = 7.2 W
	Total consumption = 36.0 W

It should be realised that the coolerbox follows Ohm's law[34] ($V = I \times R$), which means as the battery voltage varies higher or lower, the power consumption of the coolerbox will increase or decrease. The coolerbox can be used in a PV system, with a regulator that is able to incorporate it. The regulator can operate in the following sequence; when the battery is in the charge mode (referring to a PV system), the power channelled to the coolerbox can, for instance, be limited to 50% or 75% (or even less than 50%), which is about 18W to 27W, (incidentally, near the point where the COP is at an optimal value (see Table 3.9). The TEC would have just enough power to keep the contents of the coolerbox cool (compensate for heat losses). Should the battery be charged, the additional solar power can be used to power the coolerbox at maximum power, at which point it will resume its cooling down function.

4.2.2 High Efficiency

The efficiency of the cooling system has been enhanced by using liquid to cool the hot side of the TEC.

The result is a slight increase in power consumption, but the coolerbox will be able to operate at higher ambient temperatures much more effectively, since liquid cooling tends to keep the hot side at temperatures as close to ambient temperature as possible.

The only negative point in using liquid cooling is the reliability factor using a cooling fan (see pg. 128) and a liquid pump. The life cycle of this water pump is more than 20000 hours (according to the sales person). By using a regulated supply, and operating at less than maximum power, the life cycle of this part will be increased. This will depend on the climate or environment where the coolerbox will operate.

4.2.3 Low Cost

A very important and challenging factor is the manufacturing cost of the coolerbox. It is crucial that the manufacturing cost be kept as low as possible, since the market target for the coolerbox is aimed at the poorer people in rural areas.

There are two ways of looking at the manufacturing process. One way would be using mass production; that is, a first world factory which is mechanised, requiring a small work force but a high initial capital investment. Another way would be a third world production line, which entails a design in which a coolerbox is built without complex machinery, is labour intensive and has a minimal investment capital. This would ensure job creation, a very important factor at this stage (1995) in South Africa.

4.3 Basic Construction

It was decided that the basic construction for the prototype coolerbox would be galvanised 1mm sheet metal. This was easy to handle, and to bend to the shape required. The insulation used was polystyrene (5cm thick) that could easily be cut into the correct shapes and glued together with silicone based glue.

4.3.1 Casing Design

1. To enhance the efficiency of the coolerbox, it was decided that a top loading entry door to the coolerbox would be used. This does not allow the heavy cool air to be drained from the coolerbox each time the door is opened, as is the case with vertical cooler doors (see Fig 4.2).

2. The coolerbox was designed so that the batteries (see Fig 4.2, point C) of the PV system could be housed underneath the coolerbox. This has two positive aspects:

   - Since the coolerbox is a top loader, the space created for the batteries makes for easier reaching into the coolerbox.
   - The PV system and power supply can be seen as a concealed system with difficult access to the regulators and batteries. This makes tampering with the system less tempting.

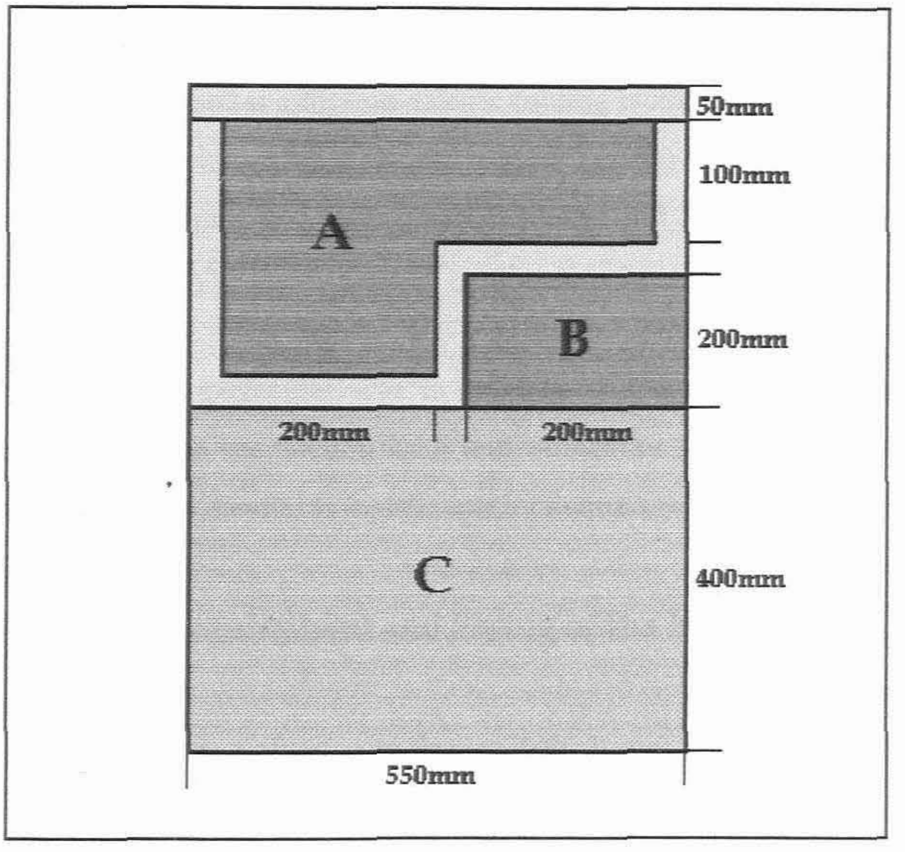

Figure 4.2. The design of the coolerbox. Point A is the cooling compartment, point B is the heat exchanger compartment and point C is the space where the PV system regulator and batteries are situated.

The cooling compartment has a capacity of 18,4 litre. Although this might sound small, emphasis has been placed on space utilization. In Fig 4.2, block A is the cooling compartment, and has been so designed that four 1liter Coke Cola sized bottles will fit into the space. The shallower part of the coolerbox is to discourage the stacking up of food. Smaller objects to be cooled can be placed here for easy retrieval. Usually stacked food leaves open spaces between the food and the lid, thus wasting space to be cooled. By inserting block B, a shallow space was created for the heat pump and its components. The end result is a very compact coolerbox giving moderate cooling power and space utilization in a dwelling such as a hut, a caravan or a hostel room.

4.3.2 Cooling Medium

The medium of cooling used is a thermo-electric cooling block (TEC): DT1049. This TEC block is supplied by Marlow, and is the same model as the one used in the simulated tests. The cost[35] of the unit is $17,50. Specifications and performance data may be retrieved from Annexure A.

The design of the heat exchanger was to enhance efficiency (no overlapping of the coldsink or heatsink, also see Section 2.4.5 and Annexure D), but also to reduce the parasitic losses due to the inevitable installation inadequacies.

4.4 Heat Pumping Requirement and Sizing of the Heat Exchanger

The sizing of the TEC was estimated in Section 2.9. It was assumed that the heat loss through the polystyrene would be below 1W and a compensation factor (in terms of °C) would amount to about 6°C for losses through contact resistance (see Section 2.8). To calculate the insulation losses[36] through the polystyrene, the following equation was used :

$$H = A.U.(t_1 - t_0) \qquad \text{..... (4.1)}$$

where

$U = 1/((1/f_1) + (x/k) + (1/f_0))$

H = heat transmitted (W)

A = area of exposed surface (m^2)

U = overall co-efficient of heat transmission ($W/m^2.K$)

t_1 = inside air temperature (°C)

t_0 = outside air temperature (°C)

x = thickness of material (m)

k = thermal conductivity of material (W/m.K)

f_1 = surface conductance of inside wall ($W/m^2.K$)

f_o = surface conductance of outside wall ($W/m^2.K$)

(f_1, f_o represents the small layer of air at the surface of the polystyrene).

For 'f', there are vertical and horizontal values of 0,12W/m².K and 0,107W/m².K. For the sake of simplicity, the values for 'f' were averaged and calculated to be 0.114W/m².K.

4.4.1 Total Inside Area of Coolerbox

The total inside area of the coolerbox was calculated to be 0,4935 m².

4.4.2 Capacity of the Coolerbox.

The volume of the inside area of the coolerbox (0,2 x 0,23 x 0,3) + (0,1 x 0,2 x 0,23)

$$= 0{,}0184m^3$$

The capacity of the coolerbox is 18.4 litre (calculated on the basis that a 1000 litre of water = 1m³).

4.4.3 Insulation Losses

To determine the losses through the insulation, Equation 4.1 was used. 'U', was calculated to be 0,0525W/m².K.

$$H = 0{,}4935 \,.\, 0{,}0525 \,.\, (40)/h \qquad \text{........(from 4.1)}$$

$$= 1.04\ W/h$$

This proves that the 5cm thick polystyrene was thick enough and only 1.04W of heat was leaked to the inside of the coolerbox (through the polystyrene) per hour. The calculation was done so as to ignore the insulation losses through the door seals, since when the door is closed, the insulation of the door and the casing is forced together.

4.4.4 Total Heat Pumping Capability of the Prototype Coolerbox

In the conclusion of Chapter 2, it became evident that the DT 1049 from Marlow Inc. was the most suitable TEC to use; this was verified in the conclusion of Chapter 3.

Insulation losses for the prototype coolerbox were calculated to be 1,04W/h; which is well within estimation and therefore, the original heat pumping estimate of 5,55W/h at $\Delta T = 25°C$ is still valid.

4.5 Coldsink Design

Generally the design of the coldsink had the following specifications set so that it would act as efficiently as possible:

- Must have as large a surface as possible for maximum heat absorption.
- Must take up as little space as possible.
- Must be easy to construct.
- Must act as a cold storage battery.

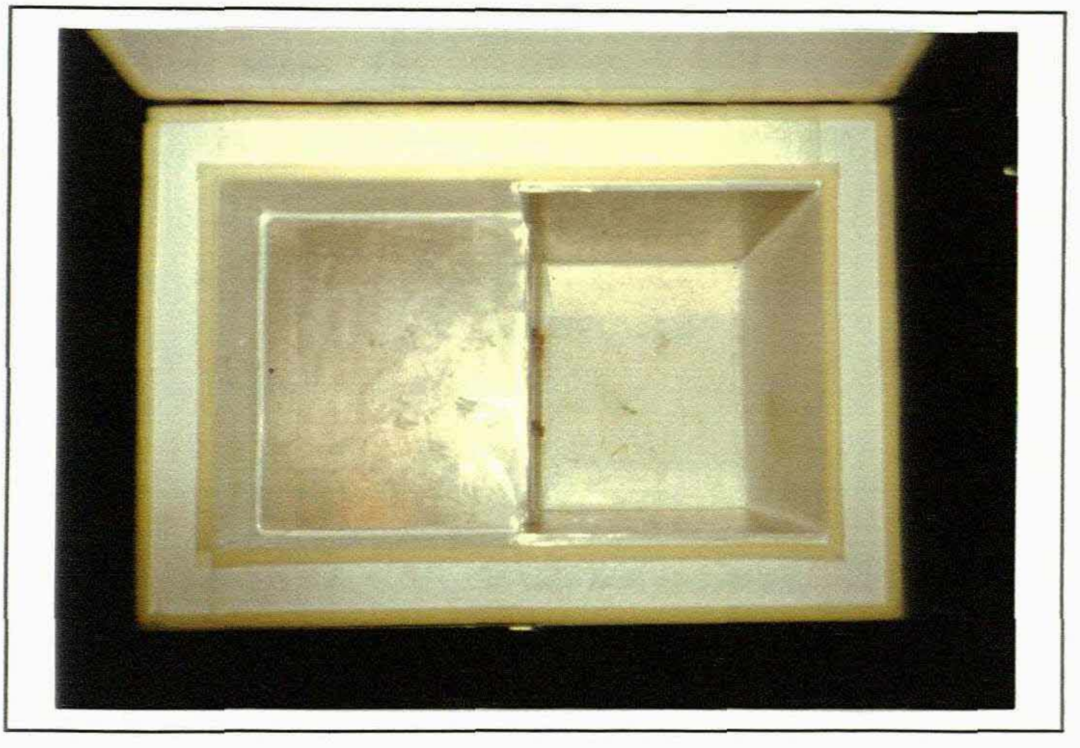

Figure 4.3 The inside view of the prototype coolerbox.

4.5.1 Physical Construction

An aluminium plate was used as a coldsink. It was constructed so as to cover the walls of the inside of the coolerbox. It was decided to use a 4,5 mm thick aluminium plate. This insured that the thermal path to the TEC is as efficient as possible without compromising on cost. The TEC is then mounted in the middle of the plate as illustrated in Fig 4.4. It should be mentioned that there was no real mathematical design because of the reasons mentioned in Section 4.5. If the guide lines are studied, it does not make any mathematical analysis necessary.

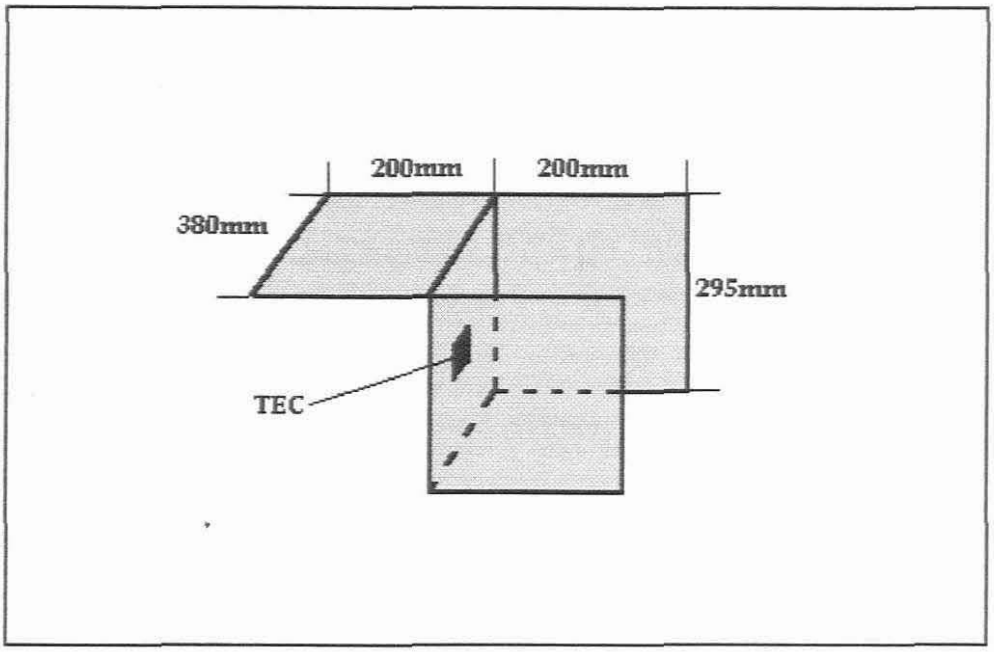

Figure 4.4. The construction of the cold side heat exchanger.

The purpose of a cold side heat exchanger, also acting as a cold storage battery, is that when warm objects are placed in the coolerbox, the cold storage battery can absorb the heat, giving a delayed heat pumping load for the TEC, which it can perform so that no additional heat needs to be pumped.

4.5.2 Heat Absorbing Capability

Since the coldsink is in direct contact with the TEC the heat transfer from the object to the coldsink is via convection and from the coldsink to the TEC via conduction. Heat transfer through radiation in this case is negligible and left out of the heat transfer calculations. This is even the case when, in Equation 2.2, a worst case scenario is used, assuming a factor of one (1) for both 'F' and 'e'.

For the convective heat transfer of the heatsink:

$Q_{conv} = A.h.(T_{water} - T_{coolerbox})$ (from 2.3)

The exposed area of the coolerbox $= (0.295 \times 0.2) \times 2 + (0.38 \times 0.295) + (0.38 \times 0.2)$

$= 0.306m^2$

The convective heat transfer for air = 21.7 W.m².°C (a typical value at 101,3kPa)

$$Q_{conv} = 0.306 \times 21.7 \times (30 - 10)$$

$$= 132.8W$$

For the convective heat transfer of the water container:

$Q_{conv} = A.h.(T_{water} - T_{coolerbox})$ (from 2.3)

The exposed area of the water container $= \pi.r.h$

$$= \pi \times 0.045 \times 0.165$$

$$= 0.0233m^2$$

The convective heat transfer for air = 21.7W.m².°C (a typical value at 101,3 kPa)

$$Q_{conv} = 0.0233 \times 21.7 \times (30°C - 10°C)$$

$$= 10.11W$$

Although the convective heat transfer of the coldsink looks very large, it should be remembered that the coldsink should act as a cold storage battery. Thus, it can be assumed that if the coldsink is at 10°C, it has a cold storage capacity of 132W (i.e. 132W of heat energy can be absorbed by the coldsink for its temperature to return to 30°C). This means that should the 1 litre container be placed in the coolerbox and the TEC pumps the parasitic heat losses only through the insulation so that the inside of the coolerbox is at a constant 10°C, the temperature rise of the coldsink (absorption) would be: $(Q_{heatsink} - Q_{canister}) = A_{heatsink} \times h.(T_{water} - T_{coldsink})$

therefore $T_{coldsink} = (-Q_{heatsink} + Q_{canister} + A_{heatsink} \times h \times T_{water}) \div (A_{heatsink} \times h)$

$= (-132.8 + 10.11 + (0.306 \times 21.7 \times 30)) \div (0.306 \times 21.7)$

$= 11.52°C$

If the one litre canister is placed inside the coolerbox, the heatsink temperature will increase by 1.52°C which is ideal since the buffer effect of a large coldsink compensates for the limitations of the heat pump.

4.6 Hot Side Heat Exchanger

The water-to-air heat exchanger was designed by Cape Heat Exchange (see Section 2.4.1 and Annexure E for the specifications of the components used on the hot side heat exchanger). They were approached because even if a suitable radiator was designed by the writer, the tubing would not be available and the writer would not be able to build it. It would be very expensive (in the case of manufacturing the radiator) to have the tubing specially made. It was decided that Cape Heat Exchange should design the heat exchanger with existing tubing. Although the radiator that was used in the prototype coolerbox is not of this design, it followed the same specifications. Cape Heat Exchange designed a radiator that would cool 80W of heat at 35°C, and the water temperature should be cooled as close as possible to ambient temperature. The design is a 120mm × 120mm, three core radiator and simulated using the following data from the coolerbox:

Cooling fan:
Power consumption: 1.1W
Max. air flow : 0.58m^3/min
Max. water pressure: 22 Pa

Water pump:
Power consumption: 3.6W
Flow rate : 1L/min

Pump pressure : 22Pa

Water-to-air heat exchanger:

Cooling capacity: 80W

Water inlet temperature: 30°C to 40°C.

Water outlet temperature: lower than 3°C of the inlet temperature at a flow rate of $0.04m^3/s$ and a 12VDC fan air velocity of 0.58m/s.

Fig 4.5 shows the visual design of the prototype coolerbox hot side heat exchanger (also see Annexure D, Fig D.5). The water pump is mounted inside the bottle (on the right side of the water-to-air heat exchanger) and the cooling fan[37] is situated on the inside of the radiator (see Annexure E for details on the design specifications for the water-to-air heat exchanger).

Figure 4.5 The heat exchanger used in the prototype coolerbox. The water pump is situated inside the plastic water container.

4.7 Performance Test of Prototype Coolerbox

The aim is to measure the cooling ability of three types of thermo-electric coolerboxes and one compressor type coolerbox, and to compare the compressor type with the TEC types in terms of cooling performance, as well as the performance of the prototype coolerbox compared to the production coolerboxes.

FIGURE 4.7 The different coolerbox performances being tested under the same conditions.

4.7.1 Procedure

1. All the coolerboxes were switched on twenty-four hours prior to the test being conducted. The TEC coolerboxes were connected to 12VDC power supply. The compressor type was connected to 220 VAC.

2. After twenty-four hours, four uniform canisters each with a capacity of 1liter, were filled with water and placed inside the coolerboxes. Temperature sensors were placed in the four water containers and the coolerboxes were closed (not to be opened again throughout the test procedure).

3. Each hour (the first hour on the table started at t = 0h) the temperature of the water was measured in each of the coolerboxes and recorded. The room temperature was also measured as well as the hot side temperature of the prototype coolerbox.

4. After the twelve measurements (that is after twelve hours) the cold side temperature of each coolerbox; the air temperature inside all the coolerboxes; the temperature at the hot side of the prototype coolerbox and the power consumption of the TEC coolerboxes were measured and recorded.

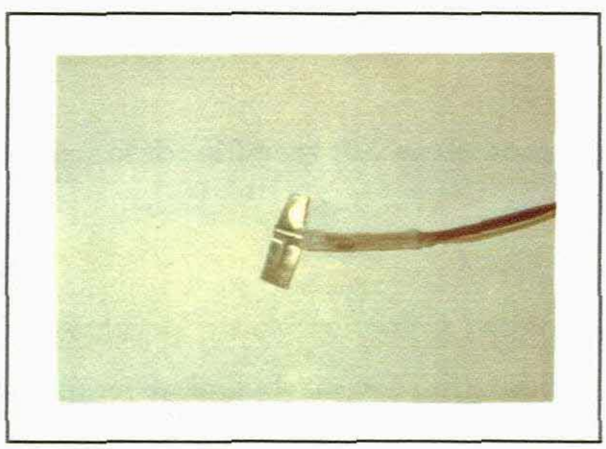

FIGURE 4.8 Temperature sensor that was used. It is a LM35 encapsulated in an aluminium plate, to ensure that a good average temperature of the water is measured.

4.7.2 Results

The prototype coolerbox out-performed the other two TEC coolerboxes. The test results are in Annexure F.

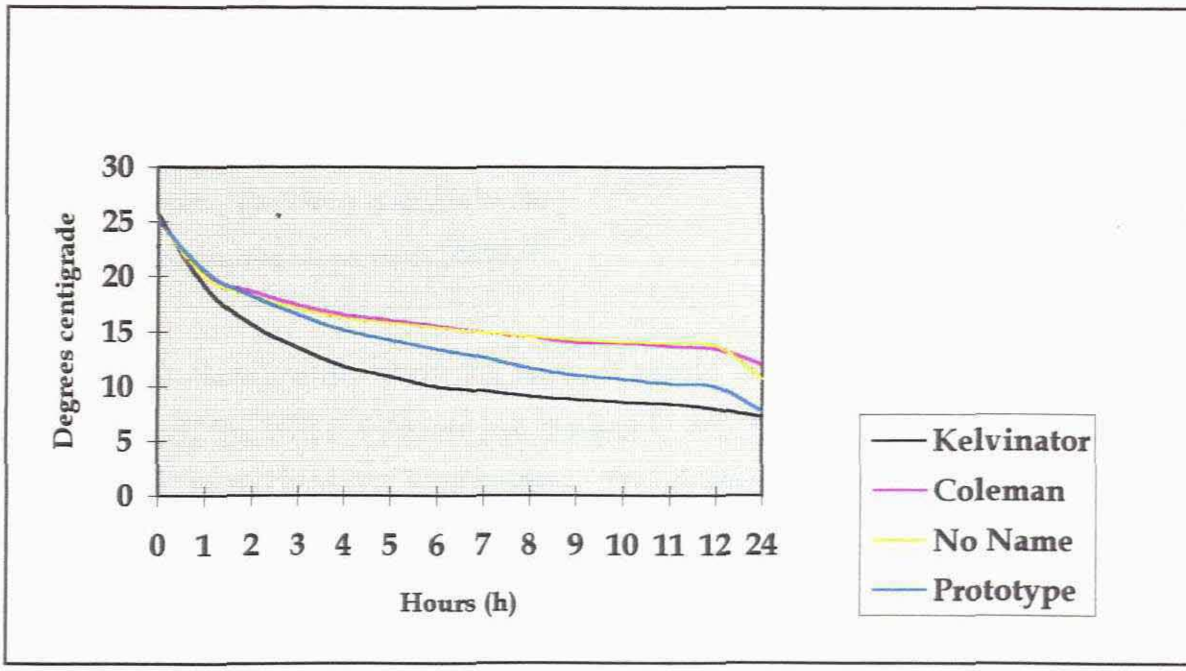

Figure 4.6. The performance of the different TEC coolerboxes and a compressor type coolerbox.

The prototype coolerbox could not match the cooling performance of the compressor type coolerbox which illustrates the importance of ΔT in a heat exchanger.

4.7.3 Comparison to Newton's Cooling Law

It was important that the coolerbox cooling capability should follow Newton's cooling law (see Section 2.7). If so, then it would imply that the coldsink does not heat up due to the work that is forced onto it, which is a good indication that it can cope with the heat load. In the test (see Annexure F, Table F.1), the following temperatures were used and recorded:

Air temperature inside coolerbox : 8°C

The liquid temperature : 25°C

Required temperature : 12 hours (the water temperature should reach 10°C after 12 hours)

Using Newton's cooling law, it was possible to calculate the cool-down curve gradient, using Equation 2.11. Two time brackets were used :

at time = 6 hours , the slope is -0.234W/h,

at time = 12 hours, the slope is -0.204W/h.

If the two calculated results were the same, it would have followed Newton's cooling law. However, the two results are not the same, which would suggest that there is a slight increase in the coldsink temperature. This is also calculated in Section 4.5.2 and the increase is 1.52°C. Refering to Section 2.7.1, the predicted cool-down curve had a gradient of -0,192W/h for a twelve hour period, which is very close to the measured value of -0.204W/h for the same period.

4.8 Cost Analysis

The cost analysis is divided into two categories (see Annexure E). The first category (2nd column) is the cost of the prototype and the second category (3rd column) is the predicted manufacturing cost of 100 units (1994).

It should however be borne in mind that the production unit cost could vary due to changing factors; for example, the exchange rate, inflation, labour costs. The cost of the prototype in production could be even lower if a proper production line is set up. The figure of R 351.22 is a conservative figure and by no means optimistic. Due to the easy construction of the coolerbox, the initial investment should be low. If a mass production line could be initiated (which inevitably would have a higher initial cost), and other types of casing materials are used, it will lower the unit cost due to the less expensive materials that are used and the shortening of production time. This decision would be a question of capital and production volumes.

4.9 Conclusion

In the performance test (after 12 hours), a $\Delta T = 18.5°C$ between the liquid temperature and the hot side temperature was measured. This resulted in conditions very close to the estimated conditions of $\Delta T = 20°C$. At $\Delta T = 17°C$ (see Annexure F, Table F.1), 1.65W of heat was pumped by the coolerbox which means that losses due to factors such as slow natural convection, dampen the cooling performance of the coolerbox.

The following conclusions were reached:

- The coolerbox was able to pump approximately 1.65W of heat.
- The losses that were estimated, proved to be accurate.

The prototype coolerbox outperformed the other two commercial type TEC coolerboxes, illustrating the advantages of using liquid cooling (see Fig 4.8). The commercial types could not even reach 10°C after twelve hours and their power consumption was a minimum of 23% to 35% more.

Producing the coolerbox proved to be easy but at least two components had to be imported. They are a 12VDC pump and the TEC module. The rest of the components could be obtained in South Africa. A modern production line could be used in order to produce the coolerbox cost effectively.

There is no doubt about the effectiveness of the prototype coolerbox and therefore the final conclusion is that although it would be slightly more complex to produce than a simple aluminium finned heat exchanger, the cooling advantages of the liquid cooling at the hot side, resulting in a good COP, are superior to the two tested commercial TEC type coolerboxes that used aluminium finned heat exchangers in terms of power consumption and cooling power.

Chapter 5

Final Conclusion and Future Research

5.1 Introduction

Developing and constructing the TEC coolerbox was difficult and the right approach had to be taken to ensure an efficient and usable coolerbox. The theoretical background of the TEC was researched as well as heat transfer characteristics in general. A simulator was constructed to simulate the TEC coolerbox, and to compare the TEC performance with the theoretical performance. The data received from the simulations were compared with the theoretical performance and because the theoretical performance proved to be too optimistic, a less complex method was derived to size a TEC for a heat exchanger application.

The coolerbox was constructed according to the experience obtained from the simulator, and compared to two production TEC coolerboxes and a compressor type coolerbox. The prototype coolerbox outperformed the two TEC coolerboxes and consumed less power.

5.2 Summary of Conclusions

The size of the TEC was calculated theoretically, using the requirements (see Annexure B). The same calculation was done using the manufacturers method. The results for I_{opt}, V_{opt} and Q_c were compared and found to be the same (see Section 2.9). It was decided that for the installation, the DT 1049 from Marlow (see Annexure A) would have more than adequate cooling power for this application.

However, from the tests conducted using the simulator, the values for I_{opt} and V_{opt} proved to be under estimated on average by nearly 61%, for the same value of Q_c (see Table 3.8(a)). Although the specifications indicated a ΔT of 64°C, this was impossible to reach and a ΔT of 44°C at T_h = 40°C, (although a ΔT of 50°C was reached when T_h = 50°C was achieved), at which point the TEC pumped 0W of heat. The maximum amount of heat that could be pumped at ΔT = 0°C is 20W. This is due to parasitic heat additions and thermal contact losses. This proves that the theoretical performance of the TEC is very deceiving compared to the performance as a practical heat exchanger. A new simple method of calculating the V_{opt} and I_{opt} were derived, to ensure that the TEC could be sized more correctly for its operating environment (see Section 3.5.7). These formulas were derived from the simulated results. They proved to give a more accurate sizing result for I_{opt} and V_{opt}.

The test results from the simulator indicated the importance of ΔT (temperature difference between the hot side and the cold side). The heat pumped (Q_c) is inversely proportional to ΔT (see Fig 2.2). The smaller ΔT could be kept, the higher Q_c would be and the more efficient the coolerbox would operate with the same input power. It was concluded that the hot side of the coolerbox should be kept as close as possible to the ambient air temperature. It was decided to use water cooling, to cool the hot side of the coolerbox. The hot side temperature was measured and found to be 3°C above the ambient air temperature.

The coldsink (heat exchanger) inside the coolerbox was made purposely as large as possible. It was supposed to act as a coldsink as well as a cold storage battery. In its function as a cold storage battery, it acts like a buffer between the TEC and the object to be cooled. It will also absorb more heat to enhance cooling speed of the object. The TEC could then remove this heat at its own pace at a later stage when no additional heat pumping is required.

A final comparison between the prototype coolerbox, two production type coolerboxes and a small (40 litre) compressor cycle coolerbox were done. Each one had to cool 1litre of water. The prototype coolerbox out performed (see Fig 4.6) the Coleman and the no name brand TEC type coolerboxes, using respectively 23% and 35 % more power. It was incidentally the only TEC coolerbox that could reach the target 10°C after twelve hours.

The compressor coolerbox outperformed all the TEC coolerboxes but after twenty-four hours, the prototype coolerbox water temperature came very close to the compressor type coolerbox.

The results from the comparison test confirmed that the tests done by the simulator helped to put the cooling performance of the Prototype TEC coolerbox into perspective. The prototype coolerbox, designed by using the simulator results and experience, outperformed the two production type coolerboxes and seems to be competitive with the small compressor type coolerbox. In addition, it is able to operate with a variable input power which makes it very suitable for a PV system.

5.3 Future Research

There are three points that could be researched in future:

- Optimising the water pump energy consumption.
- Using two TEC in different configurations.
- Developing a power sharing PV regulator.

5.3.1 Optimising the Water Pump Energy Consumption

In the prototype coolerbox, the water pump is current limited with a 10Ω resistor. This is done to regulate the flow of the water. The heat that is dissipated in the resistor is 3W. With a very simple pulse width modulated power supply, this power loss could be reduced, and this could result in an efficiency increase for the coolerbox by 8.3%.

5.3.2 Using Two TEC's in Different Configurations

Research can be done on two different configurations to determine the cooling performance, and the energy consumption of the coolerbox.

5.3.2.1 Using Two TEC's Totalling the Designed Q_c

Two smaller TEC's could be connected in parallel with a combined Q_c value equal to the design Q_c value.. The whole idea being that a two step coolerbox could be developed that can operate at half power or at full power, depending on one or both TEC's being powered. This would simplify the design of a power sharing regulator since the power to the coolerbox does not need to be regulated in conditions where it is required to operate at half of the maximum input power.

5.3.2.2 Using Two TEC's Sized to Double the Designed Q_c

Two TEC's should be connected in series resulting in each one operating at half of the input power. This would increase the COP of the coolerbox compared to the COP obtained for a single TEC coolerbox. It would be more costly and the power sharing regulator would be more complex (as would the case be if only one TEC is used) than the simple two step action described in Section 5.2.3.1.

5.3.2.3 Developing a Suitable Power Sharing Regulator

A power sharing regulator should be developed for the coolerbox. A suitable one could not be found in South Africa (1995). This regulator (see Section 1.5.6) should be able to regulate the amount of power received from the PV panel and distribute it to the battery and the coolerbox in such a manner that the system works at peak efficiency. This means that should the battery be fully charged, the excess power would be channelled to the coolerbox. At night, the regulator should provide the coolerbox with just enough current to maintain the inside temperature until the next day.

Further research should be done on the different possibilities to see if a better COP could be achieved. This is important since the power from a PV panel is limited due to cost. A power sharing regulator should be developed and the performance should be tested (depending on the TEC configuration that is used), and the whole system should be tested in real life conditions to see how effective it operates.

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Annexure A

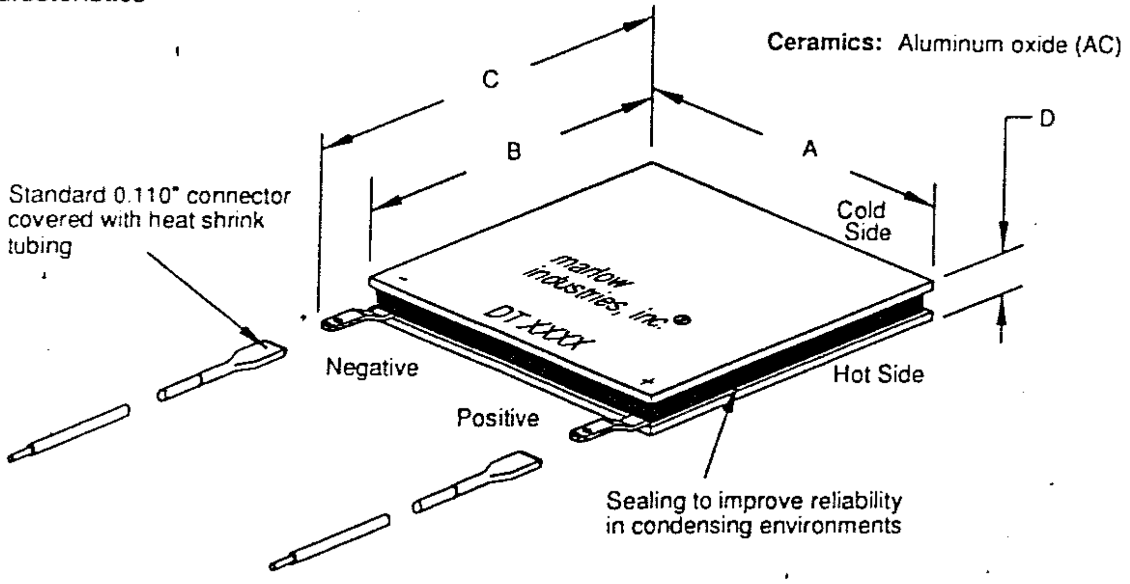
Catalog Number	I_{MAX} (Amps)	Q_{MAX} (1) (Watts) $T_H = 25°C$	V_{MAX} (Volts)	ΔT_{MAX} (°C)	N	Dimensions, mm A	B	C	D (2)
CP 0.8-7-06L	2.1	1.0	0.85	67	7	6	6	6	3.4
CP 0.8-17-06L	2.1	2.4	2.06	67	17	9	9	9	3.4
CP 0.8-31-06L	2.1	4.4	3.75	67	31	12	12	12	3.4
CP 0.8-63-06L	2.1	9.0	7.62	67	63	12	25	12	3.4
CP 0.8-71-06L	2.1	10.1	8.6	67	71	18	18	18	3.4
CP 0.8-127-06L	2.1	18.1	15.4	67	127	25	25	25	3.4
* CP 0.8-254-06L	2.1/4.2	36.2	30.8/15.4	67	254	50	25	50	3.4
CP 0.8-127-05L	2.6	22.4	15.4	67	127	25	25	25	3.1
* CP 0.8-254-05L	2.6/5.2	44.8	30.8/15.4	67	254	50	25	50	3.1
CP 1.0-7-08L	2.5	1.2	0.85	67	7	8	8	8	4.0
CP 1.0-17-08L	2.5	2.9	2.06	67	17	12	12	12	4.0
CP 1.0-31-08L	2.5	5.3	3.75	67	31	15	15	15	4.0
CP 1.0-63-08L	2.5	10.6	7.62	67	63	15	30	15	4.0
CP 1.0-71-08L	2.5	12.0	8.60	67	71	23	23	23	4.0
CP 1.0-127-08L	2.5	21.4	15.4	67	127	30	30	30	4.0
* CP 1.0-254-08L	2.5/5.0	42.8	30.8/15.4	67	254	60	30	60	4.0
CP 1.0-7-06L	3.0	1.4	0.85	67	7	8	8	8	3.6
CP 1.0-17-06L	3.0	3.4	2.06	67	17	12	12	12	3.6
CP 1.0-31-06L	3.0	6.3	3.75	67	31	15	15	15	3.6
CP 1.0-63-06L	3.0	12.7	7.62	67	63	15	30	15	3.6
CP 1.0-71-06L	3.0	14.4	8.60	67	71	23	23	23	3.6
CP 1.0-127-06L	3.0	25.7	15.4	67	127	30	30	30	3.6
* CP 1.0-254-06L	3.0/6.0	51.4	30.8/15.4	67	254	60	30	60	3.6
CP 1.0-7-05L	3.9	1.8	0.85	67	7	8	8	8	3.2
CP 1.0-17-05L	3.9	4.5	2.06	67	17	12	12	12	3.2
CP 1.0-31-05L	3.9	8.2	3.75	67	31	15	15	15	3.2
CP 1.0-63-05L	3.9	16.6	7.62	67	63	15	30	15	3.2
CP 1.0-71-05L	3.9	18.7	8.60	67	71	23	23	23	3.2
CP 1.0-127-05L	3.9	33.4	15.4	67	127	30	30	30	3.2
* CP 1.0-254-05L	3.9/7.8	66.8	30.8/15.4	67	254	60	30	60	3.2
CP 1.4-3-10L	3.9	0.8	0.36	70	3	5	10	5	4.7
CP 1.4-7-10L	3.9	1.8	0.85	70	7	10	10	10	4.7
CP 1.4-11-10L	3.9	2.9	1.33	70	11	10	15	10	4.7
CP 1.4-17-10L	3.9	4.5	2.06	70	17	15	15	15	4.7
CP 1.4-31-10L	3.9	8.2	3.75	70	31	20	20	20	4.7
CP 1.4-35-10L	3.9	9.2	4.24	70	35	15	30	15	4.7
CP 1.4-71-10L	3.9	18.7	8.60	70	71	30	30	30	4.7
CP 1.4-127-10L	3.9	33.4	15.4	70	127	40	40	40	4.7
CP 1.4-3-06L	6.0	1.2	0.36	67	3	5	10	5	3.8
CP 1.4-7-06L	6.0	2.8	0.85	67	7	10	10	10	3.8
CP 1.4-11-06L	6.0	4.4	1.33	67	11	10	15	10	3.8
CP 1.4-17-06L	6.0	6.9	2.06	67	17	15	15	15	3.8
CP 1.4-31-06L	6.0	12.5	3.75	67	31	20	20	20	3.8
CP 1.4-35-06L	6.0	14.2	4.24	67	35	15	30	15	3.8
CP 1.4-71-06L	6.0	28.7	8.60	67	71	30	30	30	3.8
CP 1.4-127-06L	6.0	51.4	15.4	67	127	40	40	40	3.8
CP 1.4-3-045L	8.5	1.6	0.36	67	3	5	10	5	3.3
CP 1.4-7-045L	8.5	3.8	0.85	67	7	10	10	10	3.3
CP 1.4-11-45L	8.5	6.0	1.33	67	11	10	15	10	3.3
CP 1.4-17-045L	8.5	9.2	2.06	67	17	15	15	15	3.3
CP 1.4-31-045L	8.5	16.8	3.75	67	31	20	20	20	3.3
CP 1.4-35-045L	8.5	19.0	4.24	67	35	15	30	15	3.3
CP 1.4-71-045L	8.5	38.5	8.60	67	71	30	30	30	3.3
CP 1.4-127-045L	8.5	68.8	15.4	67	127	40	40	40	3.3
CP 2-7-10L	9.0	4.2	0.85	70	7	15	15	15	5.6
CP 2-15-10L	9.0	9.1	1.82	70	15	15	30	15	5.6
CP 2-17-10L	9.0	10.3	2.06	70	17	22	22	22	5.6
CP 2-31-10L	9.0	18.8	3.75	70	31	30	30	30	5.6
CP 2-49-10L	9.0	29.7	5.93	70	49	36	36	36	5.6
CP 2-71-10L	9.0	43.1	8.60	70	71	44	44	44	5.6
CP 2-127-10L	9.0	77.1	15.4	70	127	62	62	62	5.6
CP 2-7-06L	14.0	6.6	0.85	67	7	15	15	15	4.6
CP 2-15-06L	14.0	14.2	1.82	67	15	15	30	15	4.6
CP 2-17-06L	14.0	16.0	2.06	67	17	22	22	22	4.6
CP 2-31-06L	14.0	29.3	3.75	67	31	30	30	30	4.6
CP 2-49-06L	14.0	46.2	5.93	67	49	36	36	36	4.6
CP 2-71-06L	14.0	67.0	8.60	67	71	44	44	44	4.6
CP 2-127-06L	14.0	120.0	15.4	67	127	62	62	62	4.6
CP 2.8-32-06L	24.0	51.8	3.87	67	32	40	40	40	5.0
CP 5-31-10L	39.0	81.5	3.75	70	31	55	55	55	5.8
CP 5-31-06L	60.0	125.0	3.75	67	31	55	55	55	4.9

Figure A.1 The specification list for the Marlow *DT* series TEC's.

marlow industries, inc.®
10451 Vista Park Road • Dallas, Texas 75238-1645 • USA
TEL 214-340-4900 • TWX 910-860-5161 • FAX 214-341-5212

DuraTEC™ Thermoelectric Cooler

Mechanical Characteristics



Typical Performance Values

	DT MAX DRY N2	QMAX (WATTS)	IMAX (AMPS)	VMAX (VOLTS)	CERAMIC WIDTH A		CERAMIC LENGTH B		TOTAL LENGTH C		HEIGHT D	
					INCHES	(mm)	INCHES	(mm)	INCHES	(mm)	INCHES	(mm)
DT1063	62	28.5	5.5	8.24	1.181	30.0	1.181	30.0	1.46	37.1	.162	4.11
DT1133	63	67.0	13.0	7.90	1.732	44.0	1.732	44.0	2.02	51.3	.166	4.22
DT1049	64	33.3	3.6	14.80	1.181	30.0	1.181	30.0	1.47	37.3	.147	3.73
DT1069	64	51.5	5.5	15.10	1.575	40.0	1.575	40.0	1.83	46.5	.162	4.11
DT1089	64	70.0	7.5	14.80	1.575	40.0	1.575	40.0	1.83	46.5	.144	3.66

DuraTEC™ Features

The DuraTEC™ series of thermoelectric coolers include:

- Sealing to improve reliability in condensing environments
- Quick and reliable standard spade lug connections
- The ability to withstand higher assembly processing temperature for short periods of time (<150°C)
- Dual Ni diffusion barriers for improved reliability

Storage, Installation, & Operation Cautions

For maximum reliability, storage and operation below 85 °C is recommended. Excessive power cycling and powering through thermostatic (on/off) control is not recommended. Consult Marlow Industries' Thermoelectric Installation Guide or Reliability Report for more details. For additional information, please contact one of our application engineers for technical support.

Figure A.2 The specification list for the Melcor *CP* series, higher current, larger heat pumping TEC's.

Annexure B

1.1 Theoretical Estimation of a TEC Size

A DT 1049 from Marlow was chosen because of the limited supply of current from a PV system, and because it was able to operate at input voltages of up to 14.8V.

Table B - 1. The spesifications of the Marlow DT 1049 TEC.

Model	ΔT_{max}	Qc_{max}	I_{max}	V_{max}
DT 1049	64°C	33.3W	3.6A	14.8V

The following design parameters were used:

$\Delta T = 25K$

$Th = 308K$

$Tc = 283K$

$\rho = 922 \times 10^{-6}$ ohm.cm

$\alpha = 199 \times 10^{-6}$ V/°K

$\chi = 0{,}018053$ W/cm.°K

N = 127 (number of couples)

$Z = 2{,}38 \times 10^{-3}$

$G = 0{,}07$

then

$$t = (T_h - T_c)/2 \qquad \text{..... (B.1)}$$

$$= 296K$$

1.1.1 Maximum Current (I_{max})

$I_{max} = (\chi.G/\alpha)[\sqrt{(1 + 2.Z.T_h)} - 1]$ (from 2.7)

$= 3.62$ A

1.1.2 Optimum Current (I_{opt})

$I_{opt} = [\chi.\Delta T.G(1 + \sqrt{(1 + Z.t)}]/\alpha.t$ (from 2.10)

$= 1.24$ A

1.1.3 Optimum Voltage (V_{opt})

$V_{opt} = (I.\rho/G) + \alpha.\Delta T$ (from 2.9)

$= 5.4$ V

1.1.4 Heat Pumped (Q_c)

$Q_c = \alpha.I.T_c - (I^2.\rho)/(2.G) - \chi.\Delta T.G$ (from 2.6)

$= 7.1$ W

Q_c shows that a DT 1049 is more than adequate for the cooling requirements of 5,55W at $\Delta T = 25°C$.

1.1.5 Co-efficient of performance (COP)

$COP_{electrical} = Q_c/(I_{opt} \times V_{opt})$ (from 2.10)

$= 1.1$

1.2 TEC Sizing Using the Manufacturers Method (Marlow Guide)

Marlow Industries provided an installation sizing guide[38]. The following sizing of a TEC is done by following the steps given by the guide.

1.2.1 Heat Loads

Total heat load on the TEC is 5,55W as already calculated, but for the comparison to the theoretical estimations we shall use the value for Q_c as calculated in Section 2.9 (also see Section 3.5.6), which is 7.1W.

1.2.2 Temperatures

TEC hot side (T_h)	308°K
TEC cold side (T_c)	283°K
ΔT (T_h - T_c)	25°C

1.2.3 Number of Stages Required

Since ΔT = 25°C it is far less than ΔT = 64°C, as specified by the manufacturer for a one stage TEC. Due to the small ΔT value of 25°C, it is not necessary to even consider a two stage TEC.

1.2.4 Selection of a One or Two Stage TEC

(a) Determine the ratio $\Delta T/\Delta T_{max}$:

ΔT = 25°C

ΔT_{max} = 64°C

$$\Delta T/\Delta T_{max} = 0.625 \approx 0.63$$

(b) On the performance graph (Fig B.1(a)), draw a horizontal line on the graph corresponding to $\Delta T/\Delta T_{max}$.

(c) Obtain the "Optimum" value of Q/Q_{max} at the intersection of the horizontal line just drawn and the diagonal "Optimum" Q/Q_{max} line. Interpolation between the curve may be necessary:

$$\text{Optimum } Q/Q_{max} = 0.23$$

(d) Obtain the "maximum" value of Q/Q_{max} at the intersection of the horizontal line (drawn in step 1.2.4 (b)) and the right vertical axis:

$$\text{Maximum } Q/Q_{max} = 0.38$$

(e) Divide the total heat load (from step 1.2.1) by the Q/Q_{max} ratios above to calculate the "Optimum" and "Maximum" Q_{max}.:

$$\begin{aligned}\text{Optimum } Q_{max} &= 7.1 \div 0{,}24\\ &= 29.6\end{aligned}$$

$$\begin{aligned}\text{Maximum } Q_{max} &= 5{,}85 \div 0{,}388\\ &= 11.6\end{aligned}$$

(f) Select a TEC from the table of "STANDARD TEC" (see Annexure A) with a Q_{max} greater than the "Maximum" Q_{max}, but less than "Optimum" Q_{max}. Within this range, a TEC with a Q_{max} close to the "Optimum" Q_{max} will provide for maximum efficiency, and Q_{max} close to "Maximum" Q_{max}, will yield smaller and possibly less expensive TEC's (For the purpose of cooling powered by a photovoltaic system, a TEC with a maximum voltage above 13,8V was selected).

A suitable TEC is a DT 1049 with, Q_{max} = 33,3W.

1.2.5 Determine the TEC's Performance

Now that a TEC is selected, the next step is to determine its performance.

(a) The specifications for a DT 1049:

$$\Delta T_{max} = 64°C$$
$$Q_{max} = 33{,}3W$$
$$I_{max} = 3{,}6A$$
$$V_{max} = 14{,}8V$$

(b) Determine the $\Delta T/\Delta T_{max}$:

$$\Delta T/\Delta T_{max} = 40/64$$
$$= 0.63$$

(c) On the performance graph, (see Fig B.1(b)) draw a horizontal line on the upper graph corresponding to $\Delta T/\Delta T_{max}$

(d) Divide the total heat load (from step 1) by the value of Q_{max} for the TEC selected (from step 1.2.5(a))

$$Q/Q_{max} = 5.85/33.3$$
$$= 0.176$$

(e) At the intersection of the horizontal line (drawn in step 1.2.5(c)) and the value of Q/Q_{max} (just calculated in step 1.2.5(d)), plot a vertical line on the performance graph.

(f) Record the value of I/I_{max} at the intersection of the vertical line just plotted and the bottom I/I_{max} axis.

$$I/I_{max} = 0.63$$

1.2.6 TEC Optimum Current

Using the value of I_{max} (obtained in step 1.2.5(a)) and the value of I/I_{max} (just obtained in step 1.2.5(f)), calculate the TEC current (I).

$$\begin{aligned} \text{TEC Current} &= I_{max} \times I/I_{max} \\ &= 3{,}6 \times 0{,}36 \\ &= 1.3\ \text{A} \end{aligned}$$

1.2.7 Calculating the Voltage

(a) Draw a horizontal line through each of the intersections of the vertical line (draw in step 1.2.5(c)) and the two curves in the lower half of the performance curve.

(b) Record the values of V/V_{max} at the intersection of the two horizontal lines drawn and the left vertical axis. These values determine the range of V/V_{max} for the TEC selected. For large ΔT's (small values of Q/Q_{max}), the voltage will correspond to the high end of the range, and for small ΔT's (large values of V/V_{max}), the voltage will correspond to the low end of the scale.

$$\begin{aligned} \text{High } V/V_{max} &= 0{,}45 \\ \text{Low } V/V_{max} &= 0{,}31 \end{aligned}$$

(c) Multiply V_{max} (obtained in step 1.2.6 (a)) by each value of V/V_{max} just obtained to get the range of TEC voltage:

High TEC voltage : 14,8 x 0,45 = 6.7V
Low TEC voltage : 14,8 x 0,31 = 4.6V

Average voltage = 5.6V

1.2.8 Calculate Maximum TEC Power

Multiply TEC current (from step 1.2.6) by high TEC voltage (from step 1.2.7(c)).

Maximum TEC power : 1.3A x 5.6V = 7.3 W

1.2.9 Calculate Power Dissipated into the Heatsink

The power dissipated into the heatsink is the sum of the total heatload and the input power to the TEC:

Maximum heat rejected = 7.3W + 5,55W
= 12.85 W

1.2.10 Calculating the TEC's COP

To calculate the COP, the average voltage value (in step 2.13.7(c)) is multiplied by the current (in step 1.2.6).

$$COP_{electrical} = Q_c/(I_{opt} \times V_{opt}) \qquad \text{(from 2.10)}$$
$$= 0.98$$

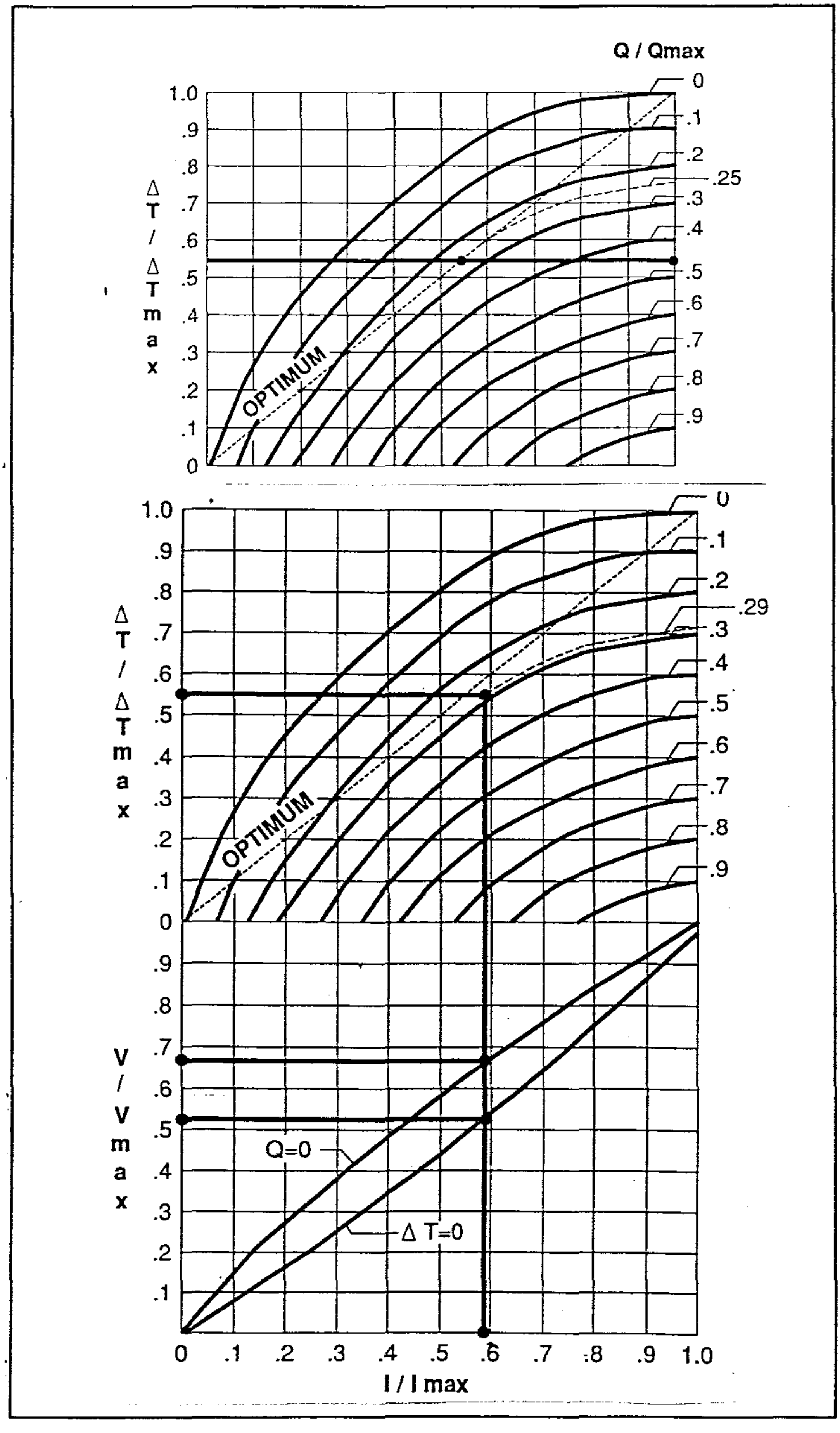


Figure B.1 The performance graphs of the TEC as supplied by Marlow.

Annexure C

1.1 20V Variable Power Supply

To deliver enough power to the simulator a 14A, variable 20VDC power supply was constructed. The requirements of the power supply were:

- Maximum current delivery of 14A.
- Variable voltage from 1,2V - 20V.
- A ripple voltage of less than 10%.

Figure C.1 The completed 20VDC powersupply.

1.2 Design of the Power Supply

To step down the 220VAC, a transformer was used. The transformer was designed to deliver 14A at 20V. It also had a 5V and 12V tap. The 20VAC was rectified using a 30A bridge rectifier. Due to the high current going through the rectifier, it tended to heat up, and was mounted on the heatsink. The rectifier operated at 48% of its maximum capability. The smoothing capacitors were calculated by the formula for a full wave rectifier[39]:

$$C = 1/(4.\sqrt{3}.fr.Rl.r) \qquad \text{..... (C.1)}$$
$$=1/(4.\sqrt{3}.50.1{,}42.0{,}04)$$
$$= 50823\ \mu F$$

The chosen capacitors' configuration were, five 10000μF, 50V capacitors in parallel to form a total of 50000μF at 50V (only 10000μF capacitors were available and due to financial reasons, were used to the same effect).

The voltage regulation was done by six LM 350T variable voltage regulators[40], connected in parallel. Each one of the LM 350T's has a current limit maximum of 3A. The maximum current, using six LM 350T's is 18A. These voltage regulators had a thermal shutdown as well as a short circuit automatic current limit of 3A each. The six LM 350T's outputs were connected in parallel, using six 0,1Ω/5W resistors. This was done to compensate for any differences in the output of each of the devices, since the devices are not identical and the outputs may vary slightly.

A power supply casing was constructed, and the heatsink was used as an integral part of the structure of the casing. The casing was made purposefully bigger for future expansion of the capabilities of the power supply. It must be noted that the heatsink size was not determined because there was no specification available and it was part of the casing construction. Forced cooling was used to cool the heatsink and this proved to be adequate. Two fans were used in the power supply. One fan cooled the outside of the heatsink while the second fan cooled the transformer and the components inside the power supply casing.

The output terminals of the power supply are (see Fig C.2):

- Variable 20V at 14A.
- Fixed 12V at 1.5A.
- Fixed 5V at 1.5A.

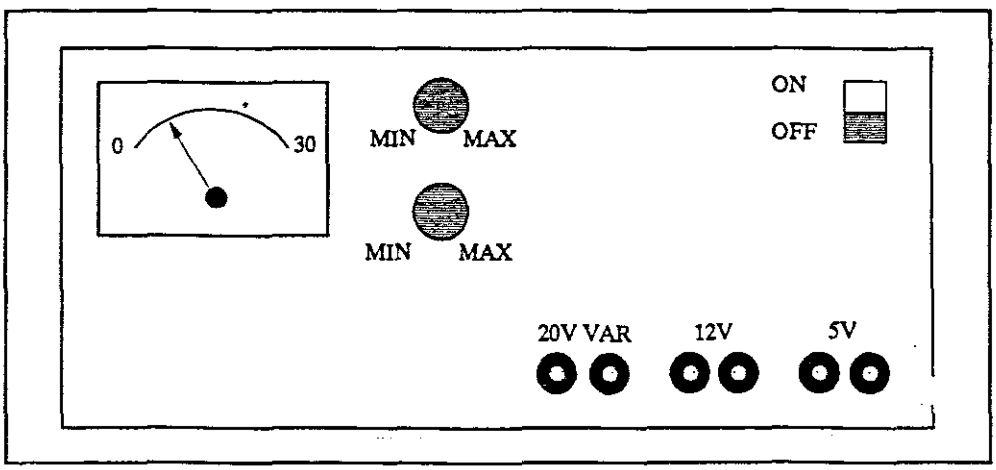


Figure C.2 The power supply illustrating the outputs of the power supply on the left and the adjustments on the right.

The voltage output is varied via two variable resistors. The lower adjustment, is used for rough adjustment. The fine tuning of the desired voltage is used by adjusting the upper fine adjustment. This function was added, since the accuracy of the desired voltage was very important in the tests.

Annexure D

1.1 Coolerbox Simulator

To specify a TEC for a certain application, the specifications of the manufacturer are used. This can sometimes lead to misleading sizing results (see Section 3.6.2). The installation of the TEC plays a big role in how well the coolerbox cools.

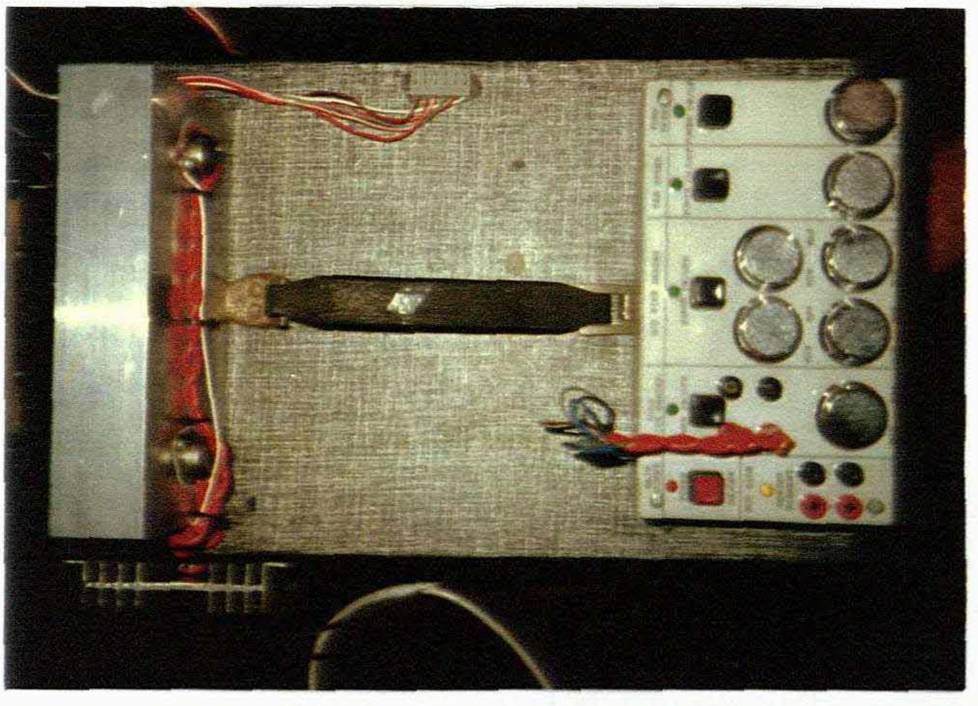


Figure D.1 The control panel mounted on top of the simulator.

The effect of the parasitic losses is too large to ignore and it is very important to design the heat exchanger using the specified Q_c value for the TEC (see the specification of the TEC in Table 3.2). The contact losses, exposure of ambient air to the cold side and other parasitic losses, influence the TEC differently. It is thus very important to test the TEC in conditions similar to the conditions of the installation in coolerboxes with liquid cooling. The test results can thus be used to simulate how the TEC will perform under given conditions.

A simulator was constructed to simulate a TEC as if it was installed in a TEC coolerbox. The expectation could be more easily evaluated in real life situations. By using the simulator and using the results, a figure of the behaviour pattern of the TEC in a coolerbox would give a very close realistic idea of how the coolerbox will perform under certain temperature conditions, given a specific TEC being used. This method of creating more realistic data on a TEC applied in a coolerbox with water cooling will prevent possible disappointment in the performance of the coolerbox, and help to assist with the specification of the TEC.

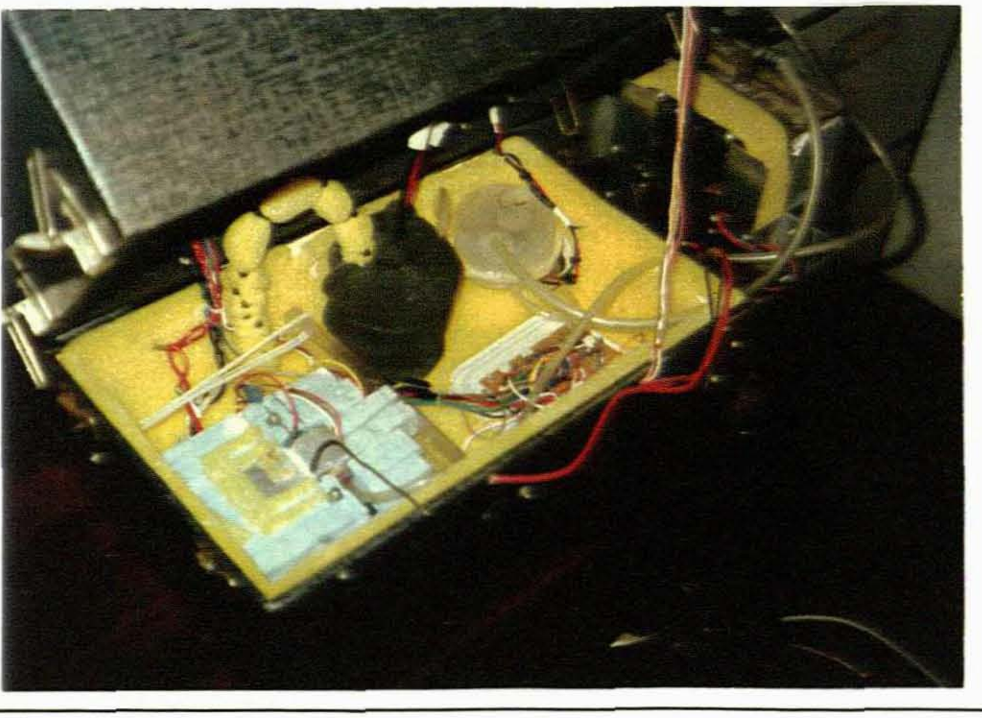


Figure D.2 The inside of the simulator where the TEC is mounted.

1.1.1 Description

The simulator was constructed with the following features:

- The ability to set the hot side temperature between 20°C and 50°C.

- The electrical heater mounted on the cold side could be set to produce variable joule heat induced in the cold side of the TEC from 0W to 50W.
- The power to the TEC is variable via an external power supply.
- The installation of the TEC is done in the same way as would be applied in a TEC coolerbox.
- Temperature sensors are fitted to the hot side and the cold side.
- The power to the cold side element and the TEC could be monitored.
- All the measurements can be done from one connector which is be connected to a computer with an A/D card, and recorded. (For this experiment, a calibrated multimeter was used since the A/D card on the computer proved to be unstable).

1.1.2 Cold Side Construction and Measurement

An aluminium block that is a 30mm × 30mm × 30mm cube is used for the cold side. It acts as a heat exchanger and is isolated from the atmosphere using 30mm thick polystyrene. One side is exposed so that the TEC could be attached to it. To act as a heater in the heat exchanger there are 4 × 15Ω/5W resistors. Four holes were drilled in the block, and the resistors were inserted in the holes using aluminium oxide paste for quick and effective heat transfer (see point A, Fig D.3).

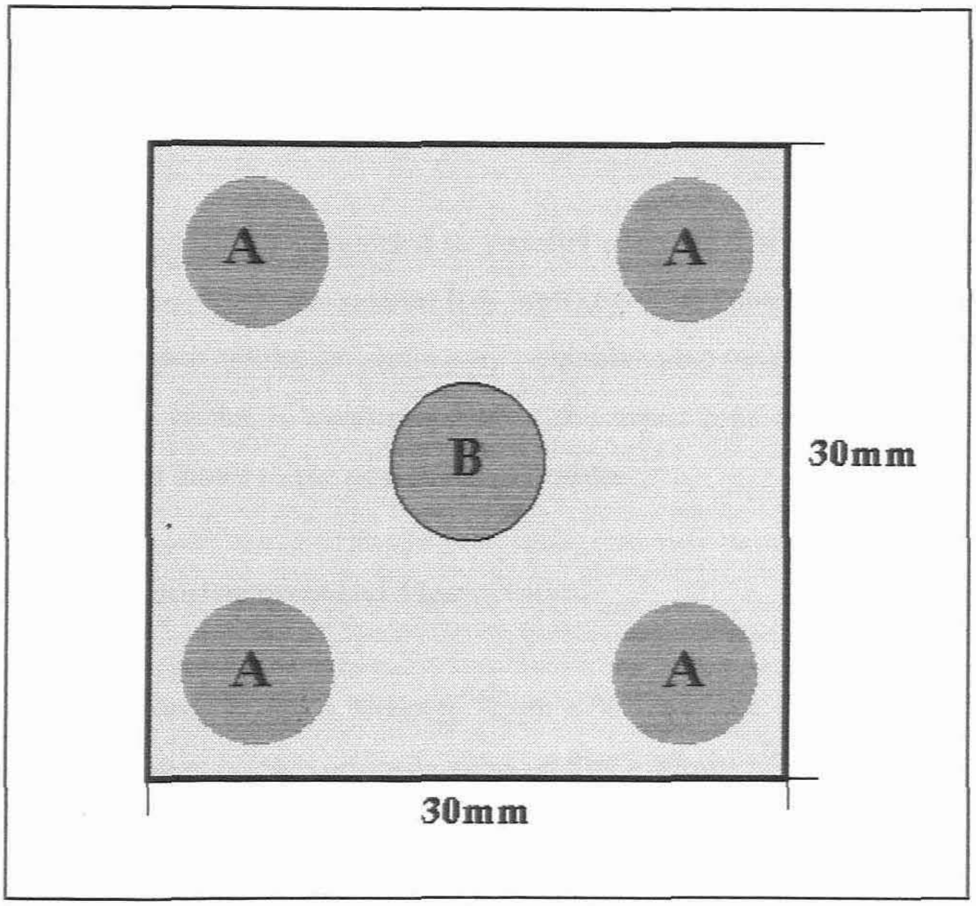


Figure D.3 The four resistors, acting as heaters, inserted at point A. Point B indicate the temperature measuring device.

An electronic thermometer (LM 35), to measure the temperature of the block, (see Fig D.3, point B) was inserted in the middle of the aluminium block to give a good average temperature of the block. The cold side block, will for the tests, act as the cold side on the inside of the refrigerator. The heater in the block will generate the heat that will be pumped.

The power that is fed to the heaters is monitored. This is done by using an extremely accurate $0{,}1\Omega$ resistor with a tolerance of $\pm 0{,}5\%$, in series with the cold side heaters. The $0{,}1\Omega$ resistor is as near as possible to the heaters to minimise losses over the power leads caused by the resistance of the wire. The potential difference across the resistor is an exact indication of the current that is supplied to the TEC.

$I = V/R$ D.1

where

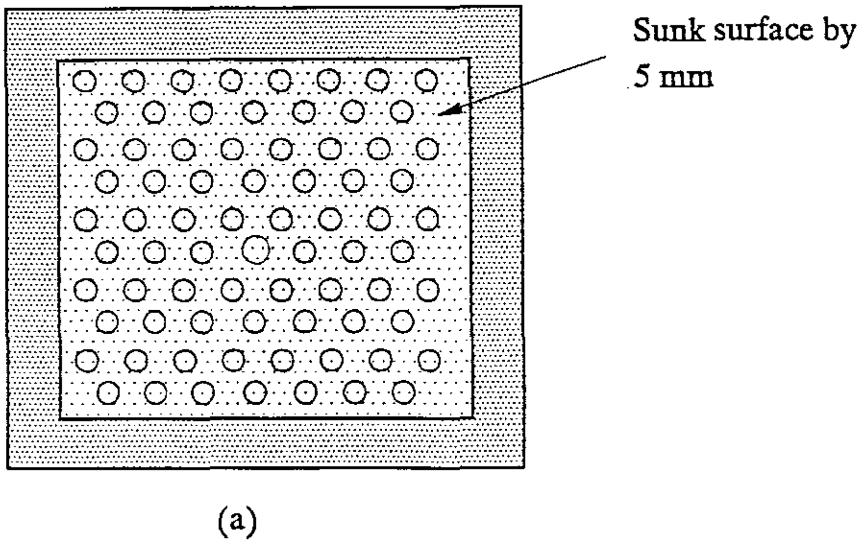
V = the potential difference across the resistance R

An op-amp comparator is connected in parallel over the 0,1Ω resistor to measure the potential difference across the resistor (see Fig D.6). This value is multiplied by a factor of 10 to give a higher resolution and a easy readable value for the current. The potential difference of the heater is measured across the input pins of the heater for greater accuracy, to avoid losses in the power supply cables.

1.1.3 Hot Side Constructions and Measurement

On the hot side there is also a 30mm x 30mm x 30mm block. This heat exchanger has been machined away on two opposite sides so that a 30mm x 30mm plate is attached to cover the depression machined in the block. These depressions act as a passage way for the water to flow through the holes in the block (see Fig D.4(b)).

Holes are drilled through the block (Fig D.4(a)) so that water can move through the holes and absorb heat. Two covers isolate the water inside the heat exchanger and water can only go into the block via an inlet and an outlet pipe.



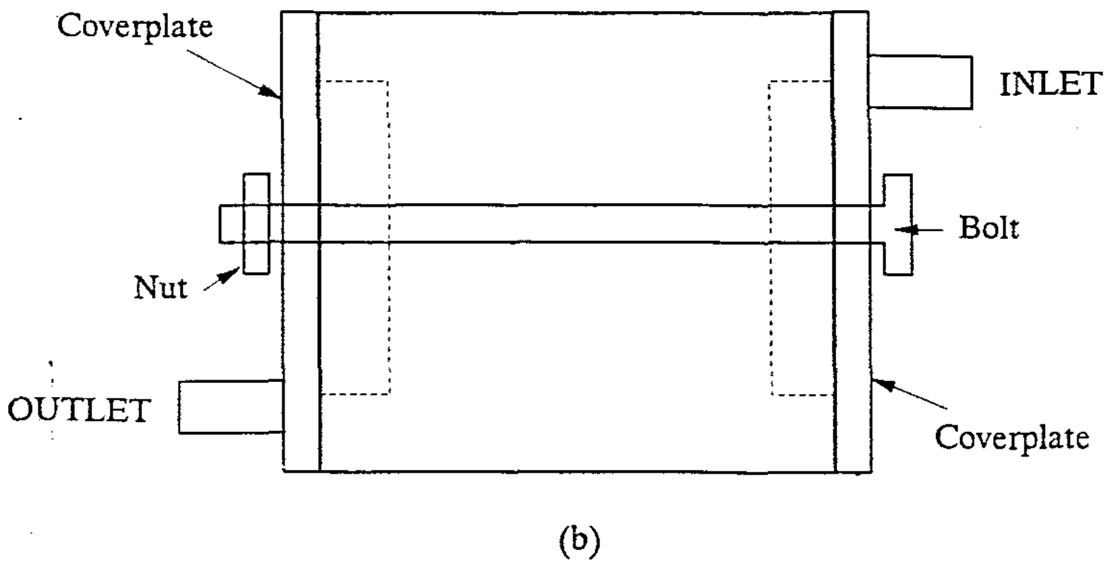


Figure D.4 (a) The water cooling hole that is drilled in the hot side block.
(b) The depressions that are machined in the hot side block.

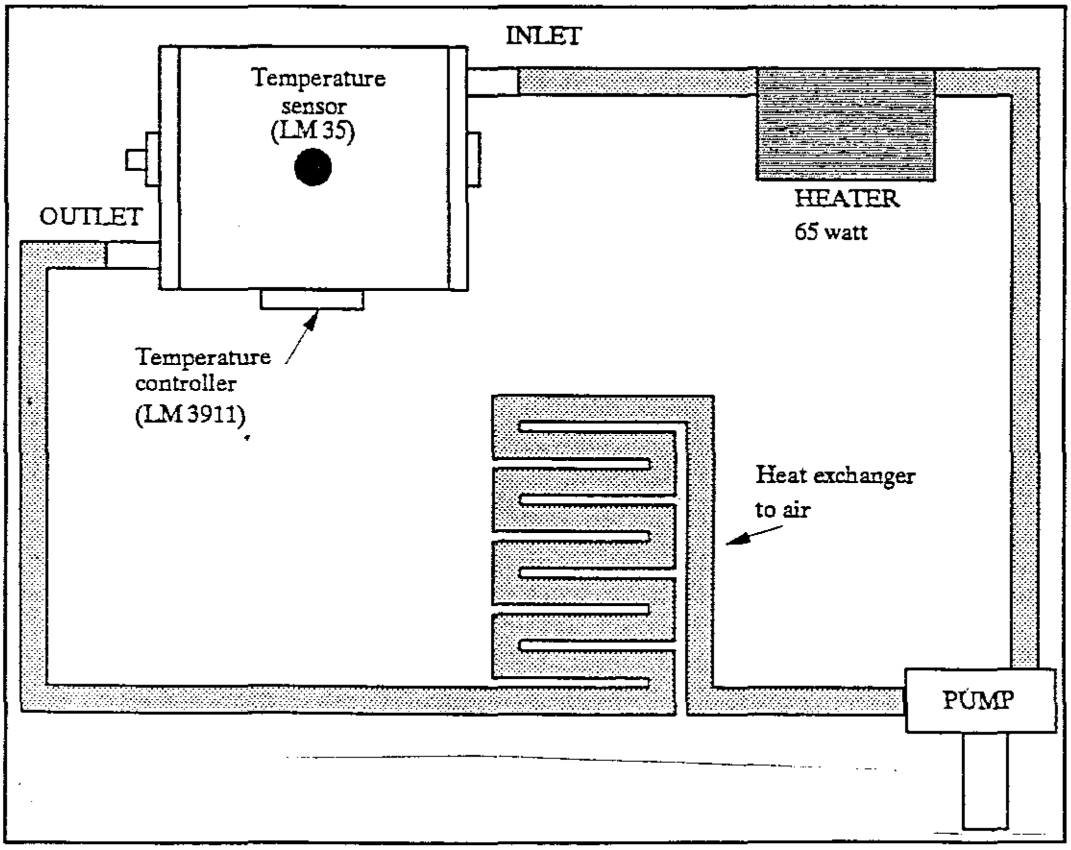


Figure D.5 The heat exchanger as used in the simulator.

The water is circulated through the cooling system, using a 12VDC water pump. The pump speed can be varied, which is useful if high hot side temperatures have to be simulated.

The power that is fed to the heaters is monitored. This is done by using an extremely accurate 0,1Ω resistor with a tolerance of ± 0,5 % in series with the heaters. The 0,1Ω resistor is as near as possible to the heaters to minimise power lead losses (see Section 1.3.2) caused by the resistance of the wires. The potential difference over the 0,1Ω resistor is an exact indication of the current that is supplied to the TEC (see Fig D.6).

$$I = V/R \qquad \text{(from D.1)}$$

where V is the potential difference over the resistor R

The comparator was connected in parallel over a 0,1Ω resistor to measure the potential difference over the resistor. This value is multiplied by a factor of 10 to give a high resolution and easily readable value for the current. The potential difference on the heater is measured over the input connectors to the heater.

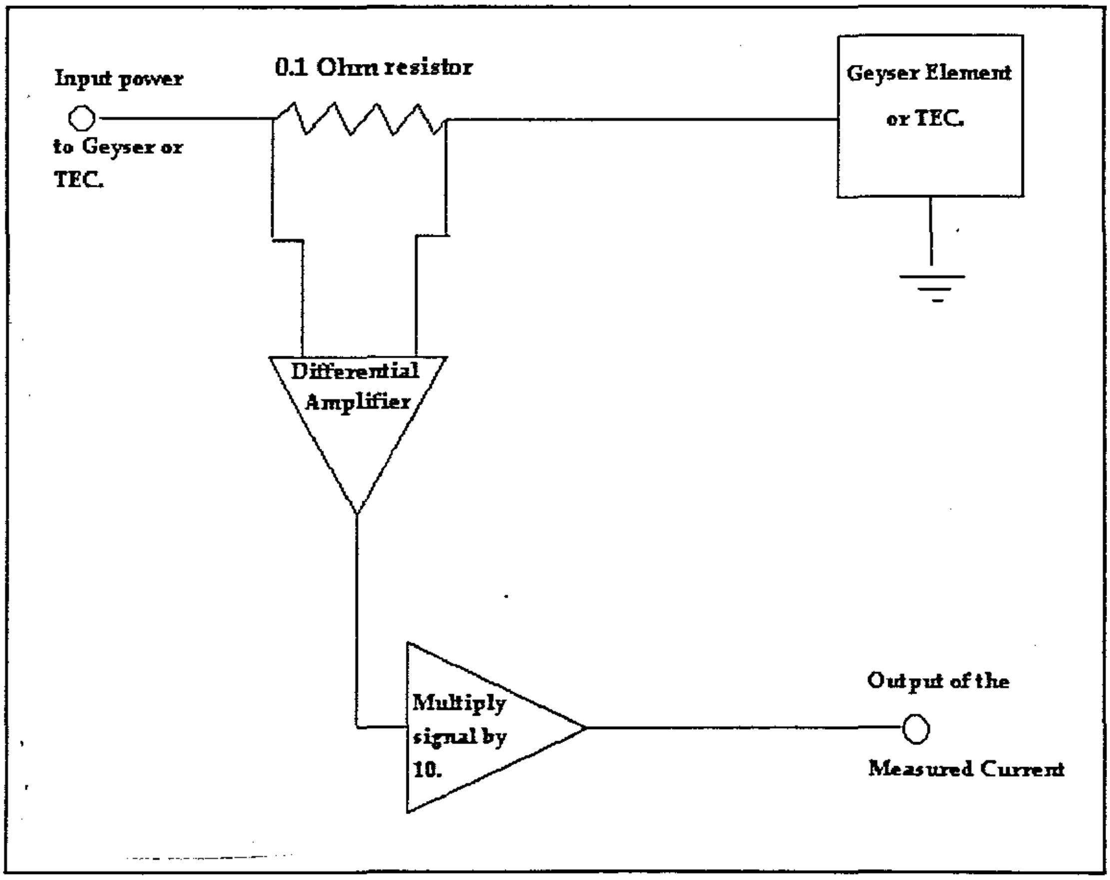


Fig D.6 Illustrating the current and voltage measurement.

The power to the cold side is controlled via a 10kΩ pot which drives a BC 237 transistor[41]. The BC237 transistor drives the base of a 2N3055 transistor. The 2N3055 transistor[42] regulates the power (heat) required for the cold side heat simulation, selected by adjusting the position of the 10kΩ variable resistor.

The water is pumped through a mini geyser which heats the water to a desired temperature at which the hot side of the TEC should operate (the simulated hot side temperature). The geyser is controlled by a temperature sensor on the hot side heat exchanger, to make sure the temperature of the heat exchanger is at the desired temperature. This is important since the hot side heat exchanger temperature is measured. If the water temperature at the geyser was used for measurement, then the heat induced by the TEC could cause the temperature to rise in the hot side heat exchanger, causing wrong readings for the hot side temperature. Since performance measurements must be done of fixed temperatures, this is not desirable.

A LM 3911 temperature controller[43] is used to control the temperature of the water supplied to the hot side heat exchanger. It has an external temperature measuring device which measures the temperature on the aluminium block, on which it is mounted. The internal temperature reference is compared to an external temperature reference. The temperature control is then done with an 2N3055 transistor regulating the temperature of the water in the geyser.

The water is pumped through the hot side heat exchanger where it absorbs the heat. An air cooled heat exchanger is used to cool down the water and the whole cycle is repeated.

The temperature controller is of the proportional bandwidth type, which is able to keep temperatures constant at ± 0,1°C of the desired temperature. The controller was specifically designed not to be of the on/off type so that temperature control would be extremely accurate for the tests.

1.2 The TEC Installation

The TEC is sandwiched between the cold side heat exchanger and the hot side heat exchanger. Aluminium oxide paste is also used on the contact surfaces to ensure maximum heat transfer. A plastic bar is used to position the external guard fast against the hot side heat exchanger, with constant pressure being applied via a pivotal system utilising a taut elastic band (see Fig. D.2). This is to ensure that the hot side, cold side and sandwiched TEC are under constant pressure forced together to reduce heat transfer losses. An electronic thermometer (LM 35) is placed in the middle of the heat exchanger. The location was decided upon because it would give a good average temperature of the aluminium block.

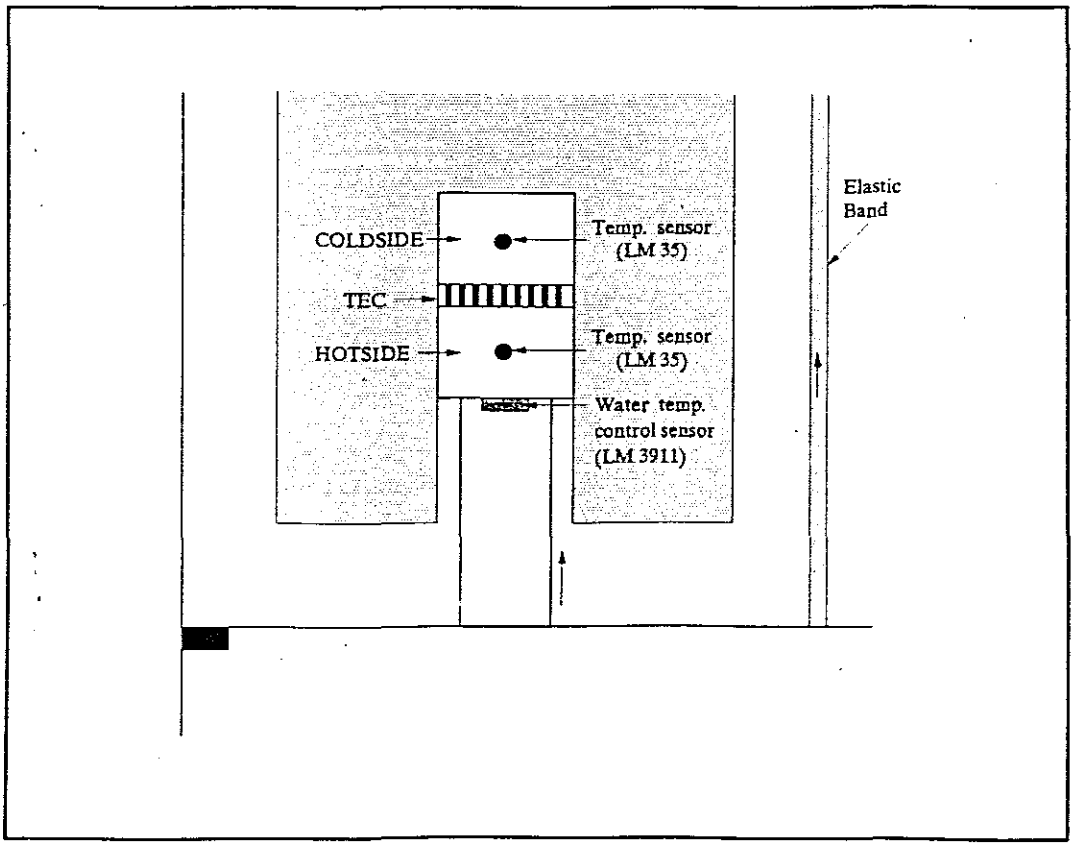


Figure D.7 The installation of the TEC.

1.2.1 TEC Power Control and Measurement

The TEC is powered by a variable voltage power supply. The power to the TEC is measured in the same way as the power measurement that is done for the cold side heater.

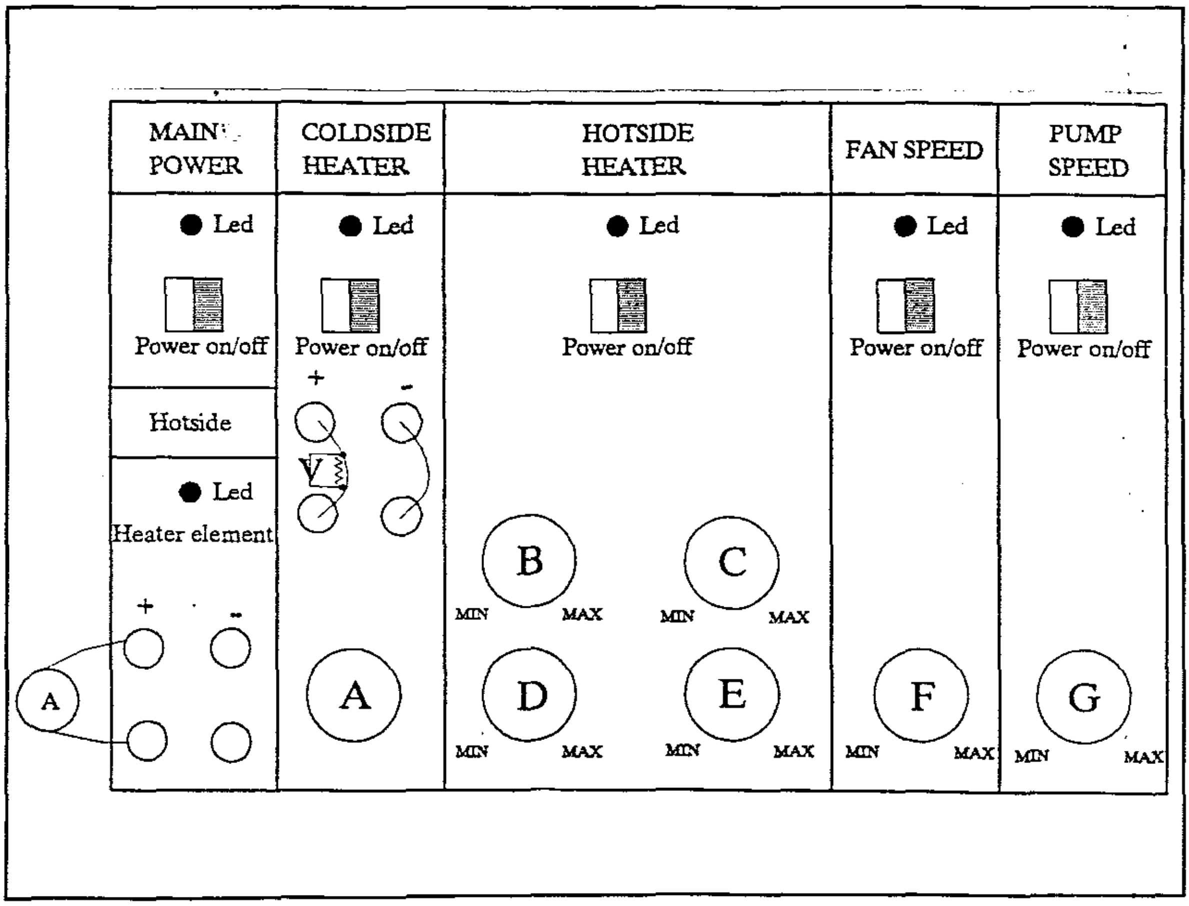


Figure D.8 The control panel of the simulator.

The front panel and features were designed to cater for as many as possible measurement functions and control over the tests as possible. It is also marked so as to be as user-friendly as possible. To aid the following explanation, the columns as in Fig D.8 were described:

be as user-friendly as possible. To aid the following explanation, the columns as in Fig D.8 were described:

1.2.1.1. Main Power Column

The main switch controls power to the whole control panel. Below the main switch marked hot side, is a yellow LED that lights up when the heater is switched on or off. As the heater switches on gradually the LED's brightness increases gradually and decreases if the heater switches off.

Below the yellow LED are 4 plugs. The one set is marked '+' (plus) and the other set is marked '-' (minus). These plugs, if bridged out, form the path for the current to the hot side element. The negative pair is shorted and the '+' (plus) pair is shorted by the ammeter. The ammeter can be used to set the temperature of the hot side. This can be done easily. When the desired temperature is reached, by looking at the ammeter reading, the reading should start to decrease. The same would apply if the temperature of the hot side drops below the set point, the ammeter reading should increase, as the heater switches on and draw more current.

1.2.1.2. Cold Side Heater Column

The cold side heater can be switched on and off using the cold side power switch. A green LED will light up when the power is switched on. There are 4 plugs. One pair, marked '+' (plus) and one pair marked '-' (minus). These plugs, if shorted out, form a current path to the cold side heater. The '-' (minus) pair is shorted. The '+' (plus) is shorted via a 0,1Ω resistor which is used to measure the current (see Annexure D, Section 1.1.3) supplied to the cold side. The power to the cold side heater is controlled via adjustment A, (see Fig D.7) to the desired value.

1.2.1.3. Hot Side Heater Column

The hot side heater main power is controlled by a switch. A green LED will light up if the power is switched on. The temperature is controlled by adjustment B, D and E. Usually only adjustment B and E are used (see Fig D.7).

1.2.1.4. Adjusting the Temperature at the Hot Side

The following steps can be used to adjust the desired hot side temperature:

1. Adjustment E is adjusted to maximum. Adjustment D usually adjusted 3/4 to maximum and B adjusted to minimum. Adjustment C is adjusted to maximum. The fan is switched off and the pump speed is adjusted to be half of maximum. The temperature is monitored.

2. When the desired temperature is reached, the fan is switched on to half power and Adjustment E is adjusted so that the heater is just switched on fully. Adjustment B is then used to do the fine adjustment so that the specified temperature is maintained.

3. Adjustment C adjusts the degree of proportional bandwidth for the temperature controller. If adjusted to maximum, the temperature controller will act as an on/off controller. Adjusting to minimum, the heater will react as an proportional bandwidth controller (the temperature controller causes less pronounced on/off switching until the controller logged of the heater element until the desired temperature).

1.2.1.5 Fan Speed

The fan, which supplies forced air to cool down the water in the water-to-air heat exchanger, can be switched on via an on/off switch. When the power is supplied to the fan-control circuitry, a green LED will glow. Adjustment F (see Fig D.7) controls the speed of the fan, and thus the amount of air through the water-to-air heat exchanger is controlled; resulting in a manual control of the water temperature.

1.2.1.6. Pump Speed

The pump can be switched on via the pump switch. If the pump circuits are switched on, a green LED will glow. The volume of cooling liquid pumped in the system is controlled via adjustment G (see Fig D.7). The temperature of the water, at extremes, can also be regulated changing the flow rate of the water.

Annexure E

The four main components of the coolerbox heat exchanger are the water-to-air heat exchanger (see Table E.2), the TEC (see Annexure A, Fig A.1), the water pump (see Section 4.6) and the cooling fan (see Table E.3). A cost analysis of the prototype coolerbox is done in Table E.1; also included is the projected unit cost for 100 units.

Table E.1

The cost analysis to produce the prototype coolerbox.

Coolerbox parts	Prototype coolerbox Cost (Rand)	Production of 100 Cost (Rand)
Casing	**Total: R 52.20**	**Total: R 35.55**
Glue (silicone based)	10.20	5.70
0.8mm galvanised steel ($2m^2$)	21.00	15.00
Two hinges	1.50	1.00
One lock-clip	2.50	2.25
Polystyrene (5mm thick)	15.00	10.10
Door seal	1.00	0.50
10 × 4mm self tapping screws	1.00	1.00
Hotside of heat exchanger	**Total: R 94.10**	**Total: R 104..42**
Water-to-air heat exchanger	20	50.00
0.5m, 5mm plastic pipe	0.60	0.60
12V computer fan	28.00	23.00
12V water pump	38.00	23.32
aluminium block 30mm × 30mm × 30mm	7.50	7.50
Coldside	**Total: R 293.25**	**Total: R 111.25**
TEC (DT 1049)	61.25	61.25
Coldsink	230	49.50
2 × 6mm bolts and nuts	2.00	0.50
Labour (R 50 per hour)	**Total: R100.00**	**Total: R100.00**
Two hours	0.00	100
Total cost	R 539.55	R 351.22

Table E.2
The design parameters used to develop the water-to-air heat exchanger.

Subject.	Design Parameter.
Process fluid	water
Atmospheric pressure	101.3kPa
Inlet air dry bulb temperature	25°C
Inlet air wet bulb temperature	17.85°C
Inlet air relative humidity	50%
Air outlet temperature	33.12°C
Air mass flow rate (dry)	0.01kg/s
Air velocity	0.58m/s
Process fluid mass flow rate	0.02kg/s
Process fluid volume flow rate	1 L/min
Process fluid inlet temperature	35°C
Process fluid outlet temperature	33.8°C
Cooling range	1.2°C
Cooling capacity	8W
Air side pressure drop	11.2kPa
Process fluid side pressure drop	0kPa
Type of tube used in simulation	Automotive design
Tube row pitch in air flow direction	19mm
Tube pitch across face of HX	10mm
Inner fouling co-efficient	6000W/m^2.K
Outer fouling co-efficient	2500W/m^2.K
Heat exchanger width (face)	120mm
Heat exchanger length (face)	120mm
Number of tube rows in air flow direction	3
Number of tubes per row	12
Serpentine method (pass layout)	Conventional
Tubular insert used in tubes	no
Number of passes	2
Process fluid flow velocity	0.04m/s
Process fluid side transfer co-efficient	924W/m^2.K
Air Side co-efficient	33.2W/m^2.K

The cooling fan is a low power consumption, computer power supply fan. In the coolerbox, it consumed about 1W and provided a air velocity of 0.58m/s through the water-to-air heat exchanger. According to the manufacturer, the life expectancy for the fan is 40000 hours[44].

Table E.3
The specifications for the cooling fan.

Model no.	Bearing	Rated voltage	Input voltage	Rated current	Rated input power	Maximum air flow	Maximum air pressure	Noise
SP8025121	Ball	12V	10.2V to 13.8V	0.1A	1.08W	0.58m³/s	2.2mmH20	28 dB

Annexure F

Table F.1

The test results obtained from the comparison tests done between the coolerboxes as described in Section 4.7.

Make	Kelvinator 401 (Compressor)	Coleman 301 (TEC)	No name 181 (TEC)	Prototype 181 (TEC)	Prototype Radiator Temp-erature (°C)	Ambient Temp-erature (°C)
Power Consumption	60W	47W	55W	36W		
Time (hours)	Water Temperature. (°C)	Water Temperature. (°C)	Water Temperature. (°C)	Water Temperature. (°C)		
0	25.7	25.1	25.1	25.2	32.8	29
1	19.1	20.1	20	20.5	32.9	29.2
2	15.6	18.6	18.2	18.2	33	29.2
3	13.5	17.4	17.1	16.5	32.6	28.9
4	11.8	16.5	16.2	15.1	32.9	30.1
5	10.9	16	15.8	14.2	33.2	29.5
6	10.2	15.5	15.4	13.4	32.9	29
7	9.6	14.9	14.9	12.6	32	28
8	9.1	14.5	14.5	11.6	32.3	29.5
9	8.8	14	14.2	11	31	28.6
10	8.6	13.9	14	10.6	32.4	29.5
11	8.4	13.6	13.9	10.2	31.5	29
12	7.9	13.4	13.7	9.9	31.4	29
24	7.2	11.9	10.7	7.7	31	28

Table F.2

The test results obtained from the comparison tests done between the coolerboxes as described in Section 4.7 (additional data recorded).

Description	Kelvinator 40l (Compressor)	Coleman 30l (TEC)	No name 18l (TEC)	Prototype 18l (TEC)
Cold Sink Temperature (°C) after 24 Hours	- 12.7	9.4	5	4.1
Cold Side Air Temperature - (°C)				
after 12 Hours	3.9	12.6	13.1	8.3
after 24 Hours	3.9	10.7	10.3	6.7
Hot Side Temperature (°C)	39	39	38	32
Power Consumption (W/h)	*	47	55	36

* The Kelvinator coolerbox is a AC device, and therefore it is more difficult to measure its power consumption. The AC motor is rated for 60W.

Table F.3

Table structure that was used to tabulate the TEC performance results.

TEC Current	TEC Voltage	TEC Power	Heater Cold Side Current	Heater Cold Side Voltage	Heater Cold Side Power	Tempe-rature Cold Side	Tempe-rature Hot Side	ΔT	COP
a.	b.	c.	d.	e.	f.	g.	h.	i	j.

(a) TEC input current : The current supplied to the TEC.

(b) TEC input voltage : The voltage supplied to the TEC.

(c) TEC input power : The product of voltage and current supplied to the TEC.

(d) Heater voltage (cold side) : Voltage supplied to the cold side heater.

(e) Heater current (cold side) : Current supplied to the cold side heater.

(f) Heater power (cold side) : Product of the voltage and current supplied to the cold side heater, i.e. the amount of heat pumped (Q_c) by the TEC.

(g) Temperature (cold side) : The temperature of the cold side as a function of the heater power and the TEC input power.

(h) Temperature (hot side) : The temperature as measured at the hot side. This is a pre-selected temperature (25°C, 30°C, 40°C or 50°C).

(i) ΔT : The temperature difference between the hot side and the cold side.

(j) Co-efficient of performance (COP) : Q_c/input power.

Table F.4
The performance measurements for a DT 1049 (Marlow) at T_h = 25°C.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temperature Cold Side (°C)	Temperature Hot Side (°C)	ΔT	COP
0.5	2.75	1.375	0	0	0	8.9	25	16.1	∞
0.5	2.67	1.335	0.5	2.02	1.01	11	25	14	0.757
0.5	2.39	1.195	1	4.03	4.03	18.8	25	6.2	3.373
0.5	2.11	1.055	1.23	4.96	6.101	25	25	0	4.701
1	5.18	5.18	0	0	0	-	25	-	∞
1	5.16	5.16	0.5	2.02	1.01	-	25	-	0.196
1	4.87	4.87	1	4.03	4.03	7.5	25	17.5	0.828
1	4.40	4.40	1.5	6.08	9.08	20.6	25	4.40	2.064
1	4.24	4.24	1.65	6.65	10.97	25	25	0	2.587
1.5	7.41	11.12	0	0	0	-	25	-	∞
1.5	7.36	11.04	0.5	2.02	1.01	-	25	-	0.092
1.5	7.18	10.77	1	4.03	4.03	0.1	25	24.9	0.374
1.5	6.83	10.25	1.5	6.08	9.08	11.1	25	13.9	0.886
1.5	6.38	9.57	1.92	7.74	14.86	25	25	0	1.55
2	9.53	19.06	0	0	0	-	25	-	∞
2	9.50	19	0.5	2.02	1.01	-	25	-	0.053
2	9.39	18.78	1	4.03	4.03	-	25	-	0.215
2	9.07	18.14	1.5	6.08	9.08	5.4	25	19.6	0.501
2	8.73	17.46	2	8.06	16.12	21.8	25	3.2	0.923
2	8.65	17.3	2.10	8.463	17.77	25	25	0	1.03
2.5	11.68	29.2	0	0	0	-	25	-	∞
2.5	11.66	29.15	0.5	2.02	1.01	-	25	-	0.035
2.5	11.60	29	1	4.03	4.03	-	25	-	0.139
2.5	11.39	28.48	1.5	6.08	9.08	1.2	25	23.8	0.319
2.5	11.1	27.75	2	8.06	16.12	16.3	25	8.7	0.581
2.5	10.99	27.48	2.23	8.99	20.05	25	25	0	0.73
3.18	14.80	47.06	0	0	0	-	25	-	∞
3.18	14.76	46.94	0.5	2.02	1.01	-	25	-	0.022
3.18	14.72	46.81	1	4.03	4.03	-	25	-	0.086
3.18	14.69	46.71	1.5	6.08	9.08	-	25	-	0.194
3.18	14.58	46.36	2	8.06	16.12	16.3	25	8.7	0.348
3.18	14.37	45.70	2.26	9.11	20.58	25	25	0	0.45

Table F.5
The performance measurements for a DT 1049 (Marlow) at T_h = 30°C.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temp-erature Cold Side (°C)	Temp-erature Hot Side (°C)	ΔT	COP
0.5	2.86	1.43	0	0	0	12.8	30	17.2	∞
0.5	2.76	1.38	0.5	2.02	1.01	15.4	30	14.6	0.732
0.5	2.41	1.205	1	4.03	4.03	24.8	30	5.2	3.344
0.5	2.14	1.07	1.20	4.84	5.81	30	30	0	5.43
1	5.36	5.36	0	0	0	1.6	30	28.4	∞
1	5.33	5.33	0.5	2.02	1.01	3.6	30	26.4	0.189
1	5.01	5.01	1	4.03	4.03	12.9	30	17.1	0.804
1	4.57	4.57	1.5	6.08	9.08	23.6	30	6.4	1.987
1	4.33	4.33	1.66	6.69	11.11	30	30	0	2.566
1.5	7.65	11.475	0	0	0	-	30	-	∞
1.5	7.46	11.46	0.5	2.02	1.01	-	30	-	0.088
1.5	7.52	11.28	1	4.03	4.03	3.6	30	6.4	0.357
1.5	7.15	10.725	1.5	6.08	9.08	17.2	30	12.8	0.564
1.5	6.78	10.17	1.94	7.82	15.17	30	30	0	1.491
2	10	20	0	0	0	-	30	-	∞
2	9.94	19.88	0.5	2.02	1.01	-	30	-	0.051
2	9.85	19.70	1	4.03	4.03	-	30	-	0.205
2	9.63	19.26	1.5	6.08	9.08	10.4	30	19.6	0.47
2	9.25	18.5	2	8.06	16.12	2702	30	2.8	0.871
2	9.21	18.42	2.05	8.26	16.93	30	30	0	0.919
2.5	12.36	30.9	0	0	0	-	30	-	∞
2.5	12.36	30.9	0.5	2.02	1.01	-	30	-	0.033
2.5	12.3	30.75	1	4.03	4.03	-	30	-	0.131
2.5	12.1	30.25	1.5	6.08	9.08	7.7	30	22.3	0.3
2.5	11.83	29.575	2	8.06	16.12	24.4	30	5.6	0.545
2.5	11.77	29.425	2.18	8.79	19.16	30	30	0	0.651
2.95	14.64	43.19	0	0	0	-	30	-	∞
2.95	14.63	43.16	0.5	2.02	1.01	-	30	-	0.023
2.95	14.57	42.981	1	4.03	4.03	-	30	-	0.094
2.95	14.53	42.864	1.5	6.08	9.08	7.6	30	22.4	0.212
2.95	14.36	42.362	2	8.06	16.12	24.0	30	6	0.381
2.95	14.33	42.274	2.20	8.87	19.51	30	30	0	0.462

Table F.6
The performance measurements for a DT 1049 (Marlow) at T_h = 40°C.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temperature Cold Side (°C)	Temperature Hot Side (°C)	ΔT	COP
0.5	3.01	1.505	0	0	0	20.2	40	19.8	∞
0.5	2.95	1.475	0.5	2.02	1.01	24.1	40	15.9	0.685
0.5	2.51	1.255	1	4.03	4.03	32.7	40	7.3	3.211
0.5	2.2	1.10	1.25	5.04	6.3	40	40	0	5.727
1	5.62	5.62	0	0	0	9.7	40	30.3	∞
1	5.54	5.54	0.5	2.02	1.01	10.8	40	29.2	0.182
1	5.33	5.33	1	4.03	4.03	17.4	40	22.6	0.756
1	4.75	4.75	1.5	6.08	9.08	34	40	6	1.912
1	4.54	4.54	1.69	6.81	11.51	40	40	0	2.535
1.5	8.09	12.135	0	0	0	0.06	40	39.94	∞
1.5	7.98	11.97	0.5	2.02	1.01	2.6	40	37.4	0.84
1.5	7.77	11.655	1	4.03	4.03	10.4	40	29.4	0.346
1.5	7.34	11.01	1.5	6.08	9.08	24	40	16	0.825
1.5	6.81	10.215	1.97	7.94	14.58	40	40	0	1.427
2	10.28	20.56	0	0	0	-	40	-	∞
2	10.28	20.56	0.5	2.02	1.01	-	40	-	0.049
2	10.12	20.24	1	4.03	4.03	4.1	40	35.9	0.199
2	9.82	19.64	1.5	6.08	9.08	16.5	40	23.5	0.462
2	9.29	18.58	2	8.06	16.12	35.1	40	4.9	0.867
2	9.2	18.4	2.1	8.46	17.77	40	40	0	0.966
2.5	12.52	31.3	0	0	0	-	40	-	∞
2.5	12.54	31.3	0.5	2.02	1.01	-	40	-	0.032
2.5	12.54	31.3	1	4.03	4.03	1	40	39	0.129
2.5	12.38	30.95	1.5	6.08	9.08	12.9	40	27.1	0.293
2.5	11.92	29.8	2	8.06	16.12	30.2	40	9.8	0.541
2.5	11.73	29.328	2.28	9.19	20.95	40	40	0	0.714
2.95	14.8	43.66	0	0	0	-	40	-	∞
2.95	14.8	43.66	0.5	2.02	1.01	-	40	-	0.023
2.95	14.8	43.66	1	4.03	4.03	0.7	40	39.93	0.092
2.95	14.77	43.57	1.5	6.08	9.08	12.2	40	27.8	0.208
2.95	14.39	42.45	2	8.06	16.12	27.9	40	12.1	0.34
2.95	14.19	41.86	2.24	9.03	20.23	40	40	0	0.48

Table F.7
The performance measurements for a DT 1049 (Marlow) at T_h = 50°C.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temp-erature Cold Side (°C)	Temp-erature Hot Side (°C)	ΔT	COP
0.5	3.28	1.64	0	0	0	28.6	50	21.4	∞
0.5	3.14	1.57	0.5	2.02	1.01	32.9	50	17.1	0.643
0.5	2.81	1.41	1	4.03	4.03	41.1	50	8.9	2.87
0.5	2.34	1.17	1.27	5.12	6.5	50	50	0	5.56
1	6.18	6.18	0	0	0	14.1	50	35.9	∞
1	6.02	6.02	0.5	2.02	1.01	17.8	50	32.2	0.168
1	5.64	5.64	1	4.03	4.03	26.3	50	23.7	0.715
1	5.11	5.11	1.5	6.08	9.08	40.9	50	9.1	1.777
1	4.76	4.76	1.75	7.05	12.34	50	50	0	2.803
1.5	8.64	12.96	0	0	0	7.1	50	42.9	∞
1.5	8.49	12.735	0.5	2.02	1.01	9.9	50	40.1	0.079
1.5	8.24	12.36	1	4.03	4.03	18.2	50	31.8	0.326
1.5	7.81	11.715	1.5	6.08	9.08	30.8	50	19.2	0.775
1.5	7.28	10.92	2	8.06	16.12	50	50	0	1.476
2	11.11	22.22	0	0	0	3.6	50	46.4	∞
2	11.2	22.4	0.5	2.02	1.01	6.5	50	43.5	0.045
2	11.01	22.02	1	4.03	4.03	13.6	50	36.4	0.183
2	10.8	21.6	1.5	6.08	9.08	26.5	50	23.5	0.42
2	10.4	20.8	2	8.06	16.12	43.8	50	6.2	0.775
2	10.27	20.54	2.2	8.87	19.51	50	50	0	0.950
2.5	13.76	34.4	0	0	0	1.2	50	48.8	∞
2.5	13.76	34.4	0.5	2.02	1.01	3.2	50	46.8	0.093
2.5	13.69	34.225	1	4.03	4.03	10.5	50	39.5	0.118
2.5	13.54	33.85	1.5	6.08	9.08	22.5	50	27.5	0.268
2.5	13.33	33.325	2	8.06	16.12	38.2	50	11.8	0.484
2.5	13.26	33.15	2.25	9.07	20.41	50	50	0	0.616
2.68	14.8	39.664	0	0	0	-	50	-	∞
2.68	14.78	39.61	0.5	2.02	1.01	2.8	50	47.2	0.026
2.68	14.68	39.34	1	4.03	4.03	9.3	50	40.7	0.102
2.68	14.55	38.994	1.5	6.08	9.08	21.7	50	28.3	0.233
2.68	14.37	38.51	2	8.06	16.12	38.3	50	11.7	0.419
2.68	14.23	38.14	2.27	9.15	20.77	50	50	0	0.545

Table F.8
The performance measurements for a CP 1.0-127-05L (Melcor) at T_h = 40°C.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temp-erature Cold Side (°C)	Temp-eratur e Hot Side (°C)	ΔT	COP
0.5	2.7	1.35	0	0	0	22.5	40	17.5	∞
0.5	2.62	1.31	0.5	2.02	1.01	24.6	40	25.4	0.771
0.5	2.28	1.14	1	4.03	4.03	32.5	40	7.5	3.535
0.5	1.95	.975	1.31	5.28	6.92	40	40	0	7.097
1	5.03	5.03	0	0	0	9.9	40	3.1	∞
1	4.96	4.96	0.5	2.02	1.01	12.1	40	27.9	0.204
1	4.75	4.75	1	4.03	4.03	19.9	40	21	0.848
1	4.23	4.23	1.5	6.08	9.08	32.1	40	7.9	2.15
1	3.92	3.92	1.77	7.133	12.63	40	40	0	3.22
1.5	7.11	10.67	0	0	0	1.9	40	38.1	∞
1.5	7.04	10.56	0.5	2.02	1.01	4.4	40	35.6	0.096
1.5	6082	10.23	1	4.03	4.03	11.7	40	28.3	0.394
1.5	6.48	9.72	1.5	6.08	9.08	23	40	17	0.934
1.5	5.90	8.85	2	8.06	16.12	38.3	40	1.7	1.821
1.5	5.83	8.745	2.05	8.26	16.93	40	40	0	1.936
2	9.15	18.3	0	0	0	-	40	-	∞
2	9.08	18.16	0.5	2.02	1.01	-	40	-	0.056
2	8.97	17.94	1	4.03	4.03	3.7	40	36.3	0.225
2	8.71	17.42	1.5	6.08	9.08	13.1	40	26.9	0.521
2	8.18	16.36	2	8.06	16.12	31.0	40	9	0.985
2	7.97	15.94	2.26	9.12	20.6	40	40	0	1.292
2.5	11.19	27.98	0	0	0	-	40	-	∞
2.5	11.15	27.88	0.5	2.02	1.01	-	40	-	0.036
2.5	11.02	27.55	1	4.03	4.03	.01	40	39.9 9	0.143
2.5	10.78	26.95	1.5	6.08	9.08	10.5	40	29.5	0.337
2.5	10.44	26.1	2	8.06	16.12	25.6	40	14.4	0.618
2.5	10.13	25.33	2.39	9.632	23.02	40	40	0	0.909
3	13.20	39.6	0	0	0	-	40	-	∞
3	13.17	39.51	0.5	2.02	1.01	-	40	-	0.026
3	13.14	39.42	1	4.03	4.03	-	40	-	0.051
3	12.95	38.85	1.5	6.08	9.08	7.6	40	32.4	0.102
3	12.65	37.95	2	8.06	16.12	21.8	40	18.2	0.425
3	12.41	37.23	2.48	9.99	24.78	40	40	0	0.666
3.5	15.39	53.87	0	0	0	-	40	-	∞
3.5	15.37	53.80	0.5	2.02	1.01	-	40	-	0.019
3.5	15.28	53.48	1	4.03	4.03	-	40	-	0.074
3.5	15.18	53.13	1.5	6.08	9.08	7.7	40	32.3	0.171
3.5	14.97	52.40	2	8.06	16.12	20.8	40	19.2	0.308
3.5	14.83	51.91	2.5	10.08	25.2	40	40	0	0.485

Table F.9
The performance measured for a CP 1.0-127-05L (Melcor) with different values for T_h and $\Delta T = 0°C$.

TEC Input Current (A)	TEC Input Voltage (V)	TEC Input Power (W)	Heater: Cold Side Current (A)	Heater: Cold Side Voltage (V)	Heater: Cold Side Power (W)	Temperature Cold Side (°C)	Temperature Hot Side (°C)	ΔT	COP
3.79	14.43	54.1	2.42	9.87	23.6	25	25	0	0.432
3.66	14.78	54.69	2.45	9.75	24.2	30	30	0	0.447
3.5	14.83	51.91	2.5	10.08	25.2	40	40	0	0.485
3.34	14.55	48.6	5.56	10.32	26.41	50	50	0	0.543