Research Journal of Applied Sciences 5 (3): 195-203, 2010
ISSN: 1815-932X

Experimental and Theoretical Analysis of a Beverage Chiller

A.A. Adeyanju, E. Ekwue and W. Compton
Department of Mechanical and Manufacturing Engineering,
University of West Indies, St. Augustine, Trinidad and Tobago

Abstract: The study shows the experimental and theoretical analysis of a fast beverage chiller based on the principle of a thermoelectric refrigeration and it was discovered theoretically that for a cooling time of 2 min and a beverage size of 474 mL (16 Oz), 6 TEC modules and hence 12 heat sinks (one for each side of the TEC module) were needed. However by increasing cooling time to 4 min and decreasing beverage size to 325 mL, 4 TEC modules and hence 8 heat sinks could be used. Comparison were also made between the beverage chiller's cooling time with cooling times obtained from the freezer space and cold space of a household refrigerator. All three tests were carried out on 325 mL of water in a glass jar. The result shows that for the refrigerator freezer space, the temperature of the water decreased linearly with increasing time. However for the beverage chiller, the water temperature decreased exponentially with increasing time.

Key words: Thermoelectric, module, heat sinks, sizing, beverage chiller, cooling, refrigerator

INTRODUCTION

A thermoelectric device is one that operates on a circuit that incorporates both thermal and electrical effects to convert heat energy into electrical energy or electrical energy to a temperature gradient. Thermoelectric elements perform the same cooling function as Freon based vapor compression or absorption refrigerators. Energy is taken from a region thereby reducing its temperature. The energy is than rejected to a heat sink region with a higher temperature.

Thermoelectric elements are in a totally solid state while vapor cycle devices have moving mechanical parts that require a working fluid. Thermoelectric modules (Fig. 1) are small sturdy, quiet heat pumps operated by a DC power source. They usually last about 200,000 h in heating mode or about 20 h if left on cooling mode. When power is supplied, the surface where heat energy is absorbed becomes cold; the opposite surface where heat energy is released becomes hot. If the polarity of current flow through the module is reversed, the cold side will become the hot side and vice-versa.

Thermoelectric devices can also be used as refrigerators on the bases of the Peltier effect (Cengal and Michael, 1998). To create a thermoelectric refrigerator (Fig. 1), heat is absorbed from a refrigerated space and than rejected to a warmer environment. The difference between these two quantities is the net electrical work that needs to be supplied. These refrigerators are not overly popular because they have a low coefficient of performance. Thermoelectric modules can be used as thermocouples for temperature measurement or as generators to supply power to spacecrafts and electrical equipment. Thermo electronic devices are used in a variety of applications. They are used by the military for night vision equipment, electronic equipment cooling, portable refrigerators and inertial guidance systems.

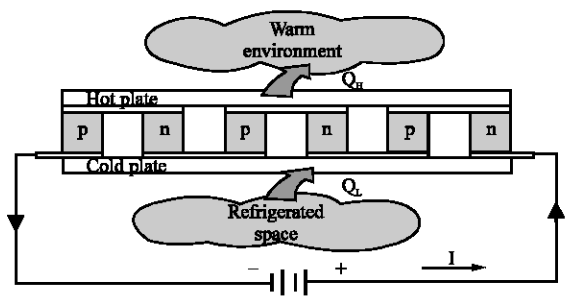

Fig. 1: A thermoelectric refrigerator based on the Peltier effect

These products are useful to the military during war and training because they are reliable, small and quiet. Another advantage to these thermoelectric products is that they can be run on batteries or out of a car lighter. The medical community uses thermoelectric applications for hypothermia blankets, blood analyzers and tissue preparation and storage. The main advantage of thermoelectric devices to the medical community is that

Corresponding Author: A.A. Adeyanju, Department of Mechanical and Manufacturing Engineering, University of West Indies, St. Augustine, Trinidad and Tobago

the devices allow doctors precise temperature control which is useful in handling tissue samples. Hypothermia blankets are pads that patients rest on during surgery to keep their body at a certain temperature. Thermoelectric devices are probably most well known for their contribution to powering spacecrafts like the Voyager.

Radioisotope thermoelectric generators provided all of the on board electrical power for NASA's Voyager. The thermoelectric devices proved reliable since they were still performing to specification 14 years after launch. The power system provided the equivalent of 100-300 Watts electrical power and multiples thereof. NASA is now requiring higher efficiency rates out of smaller units. The interaction between thermal and electric phenomena (Seebeck effect (1821), Peltier effect (1834), Joule effect (1841) and Thomson effect (1857)) was known since the 19th century (Rowe and Bhandari, 1983). In 1885 the English physicist J.W Rayleigh outlined the possibility of using thermoelectric devices as electricity generators but his development was totally stopped because of the low efficiency achieved. However, the major advance was made in the 1950s with the introduction of semiconductors as thermoelectric materials. It was observed that they had a high Seebeck coefficient, good electrical conductivity and low thermal conductivity. In those moments thermoelectric refrigeration began to look more promising and Peltier devices were developed for refrigeration applications mostly for the military field. Research on semiconductor thermocouples also led to the construction of thermoelectric generators with a high enough efficiency for special applications.

There was little improvement in thermoelectric materials from the time of the introduction of semiconductor thermo-elements until the end of the twentieth century. However, in recent years, several new ideas for the improvement of materials have been put forward and significant advances are being made (Goldsmid, 2009).

Nowadays, in the civil market, thermoelectric refrigeration has a place in medical applications and scientific mechanisms and devices where accurate temperature control is needed. Nevertheless, there are other applications with great potential in which companies are starting to show interest, e.g., dehumidifiers (Vian *et al*., 2002), domestic and automobile air conditioning systems, portable iceboxes, domestic refrigerators, devices to transport perishable products, computer processor coolers, etc. For these applications, thermoelectric refrigeration competes with conventional refrigeration systems like vapor compression refrigeration. For a typical conventional refrigeration system, a temperature difference between the ambient and the cabinet of about 25-30 K at T_h = 300 K is usually required to achieve satisfactory cooling performance. This indicates that the maximum COP of a thermoelectric refrigerator comprised of a commercially available module is around 0.9-1.2. However, the practical COP of a thermoelectric refrigerator is much lower than this because the temperature difference between the hot and cold side of the thermoelectric module is larger than the temperature difference between the ambient and the cabinet. In other words, the hot side temperature is higher than the ambient and the cold side temperature is lower than the cabinet temperature.

For a practical thermoelectric cooling system, the hot side heat exchanger rejects the heat produced on the hot side of the thermoelectric module to the ambient and so reduces the hot side temperature. The cold side heat exchanger removes the heat from the cold region to the cold side of thermoelectric module and so increases the temperature of the cold side. Because the thermoelectric module is very high heat intensity equipment, the high efficiency thermoelectric heat exchangers is necessity.

Use of a heat pipe will not be of benefit for natural convection because the dominant thermal resistance in this case is the convection resistance (Webb, 1998). Water-cooled forced convection heat exchangers have excellent performance. The main drawback of a water-cooled heat exchanger is that it needs a convenient source of cooling water. Without a source of cooling water, a forced convection water heat exchanger would require a pump and radiator and associated fittings and tubing. The added resistance of the radiator would increase the overall resistance.

Air-cooled systems are therefore often more desirable. Many heat exchange systems based on the afore-mentioned forced air convection exchangers and the use of heat pipes have been reported. Sofrata (1996) reported that using a double fan in an appropriate position could significantly increase the efficiency of the forced air exchanger compared to using the single fan in a refrigerator.

A long chimney for a natural-convection heat exchanger may also improve the performance of the refrigerator without the need to use fans that of course require the electrical power input. A novel, air-cooled thermosyphon reboiler-condenser system has been reported (Webb, 1998) and has been used as a heat exchanger of a thermoelectric refrigerator (Gilley and Webb, 1999).

This system is capable of providing very low heat sink resistance values with air cooling and a thermal resistance as low as 0.0194-0.0505 KW^{-1} was obtained for cooling a 45 mm^2 module. The system promises significantly higher COP for thermoelectric coolers than is possible using existing heat exchange technology. Riffat *et al*. (2001) have reported a thermoelectric refrigeration system which employs a Phase Change Material (PCM) as a cold side heat exchanger for cooling storage and improvement of the COP. The refrigeration system was first fabricated and tested using a conventional heat sink system (bonded fin heat sink system) at the cold heat sink. In order to improve the performance and storage capability, the system was reconstructed and tested using an encapsulated Phase Change Material (PCM) as a cold sink.

Both configurations used heat pipe embedded fins as the heat sink on the hot side. Results of tests on the latter system showed an increased performance. This was because the PCM had a large storage capacity allowing most of the cooling energy to be absorbed by the PCM and therefore, the cold side temperature fell more slowly than when the PCM was not used. During the phase change process, the temperature of the refrigeration system was almost constant until the phase change process was complete. This helped to keep the temperature difference across the thermoelectric module to a minimum, thus improving its performance.

In general, thermoelectric modules are very high heat intensity equipment which need high efficiency heat exchangers to lower the hot side temperature and increase the cold side temperature in order to improve the COP. Use of a greater number of modules would also improve the COP of the system. Use of more modules would reduce the heat load on each module and so lower the heat flux densities of both the hot and cold side of each module.

MATERIALS AND METHODS

Thermoelectric modules parameters selection:

- Desired temp. of cold compartment/beverage = 6°C/279.15 K
- Ambient temp., T_{amb} = 31°C/304.15 K
- Required time for desired cooling from T_{amb} to 4°C = 120 sec
- Volume of beverage to be cooled = 16 oz = 474 cc
- Mass of beverage to be cooled = 474×0.001 kg = 0.5 kg
- NB: Properties of water is used in place of beverage for calculations
- Using Newton law of cooling:

$$Q = mC_p\Delta T$$

$$C_{P(water)} = \frac{4186J}{kg.k}$$

$$Q_{beverage} = \frac{0.5}{120} \times 4186 \times (304.15 - 279.15) = 436.04\,W$$

Where:

$Q_{beverage}$ = Heat load supplied by beverage

- Accounting for heat transfer through panels of cold compartment
- Top and bottom panel dimensions = 0.32×0.155 m
- Vertical side panels dimensions = 0.44×0.155 m
- Front and back panel dimensions = 0.32×0.44 m
- Thermal conductivity (K) of materials:

$$K_{Al,6061} = \frac{167W}{m.K}$$

$$K_{Syrofoam} = \frac{0.033W}{m.K}$$

- To find hot-side temp. of TEC modules, T_{hot}
- Keeping heat-sink at 15°C above ambient temp.

$$T_{hot} = T_{amb} + 15°C = 31 + 15 = 46°C$$

- Temp. difference ΔT across TEC can now be calculated as follow:

$$\Delta T = T_{hot} - T_{cold} = 46 - 6 = 40°C$$

- About 40°C will be used for design calculations
- Parameters to be used for TEC module selection

$$Q = 75W, \quad \Delta T = 40°C$$

- From TEC module performance Fig. 2
- At Q = 75 W and ΔT = 40°C
- Voltage = 12 V and Current, A = 21 A
- Heat produced internally by each TEC module = VI = 12×21 = 252 W
- Total heat that must be dissipated by hot side heat sink = Heat produced internally by TEC module, VI+cold side cooling load, Q_{tec} = 252+75 = 327 W
- Thermal resistance of heat sink to be used

$$R_{hs} = \frac{T_{hot} - T_{amb}}{(VI) + Q_{tec}}$$

$$R_{hs} = \frac{46 - 31}{327} = 0.05\frac{K}{W}$$

TEC module parameters and performance graphs

Type	Couples	Imax (A)	Vmax (V)	Qcmax(w) ΔT = 0	ΔTmax (°C) Qc = 0	R (Ω)
CP1-12726	127	26	15.4	243.5	68	0.45±0.05

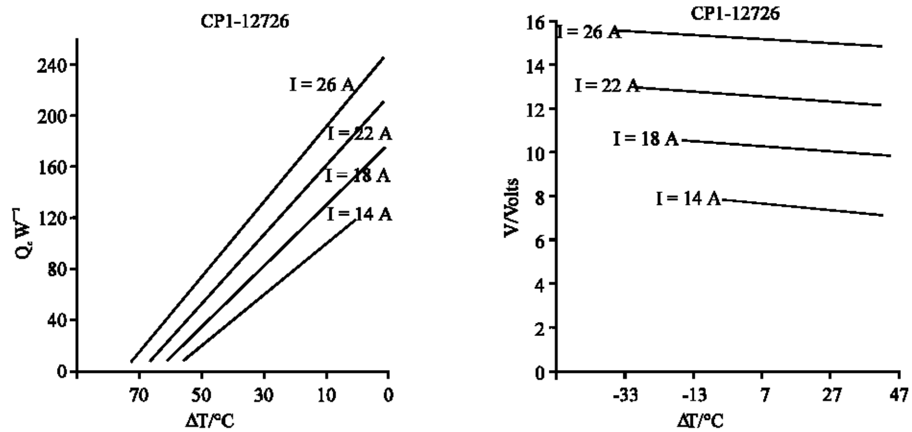

Fig. 2: TEC module performance graph (obtained from TEC module spec sheets)

- A heat sink or rating 0.05 K/W or less must be used with each TEC module

Heat sink sizing: Heat sink design parameters (Fig. 3) given as:

Heat sink hight (H)	=	0.085 m
Fin lenght (H_f)	=	0.08 m
Fin width (L)	=	0.135 m
Heat sink width (W)	=	0.12 m
Fin thickness (t_f)	=	2×10^{-3}
Number of fins (N_{fin})	=	25

Volumetric flow rate of heat sink fan, v = 300, cfm = 8.50 $m^3\,min^{-1}$ = 0.142 $m^3\,sec^{-1}$. Air velocity between fins:

$$V = \frac{v}{N_{fin}.b.H_f}$$

Where:

$b.H_f$ = Cross sectional area perpendicular to air flow:

$$V = \frac{0.142}{25\times2.917\times10^{-3}\times0.080} = 24.48\ m\ sec^{-1}$$

Spacing between fins:

$$b = \frac{W - N_{fin}\ t_f}{N_{fin} - 1} = \frac{0.12-(25\times0.002)}{25-1} = 2.917\times10^{-3}\ m$$

Properties of air @ 1 atm. and 28°C, $\mu = 184.6\times10^{-7}\ Ns\ m^{-2}$

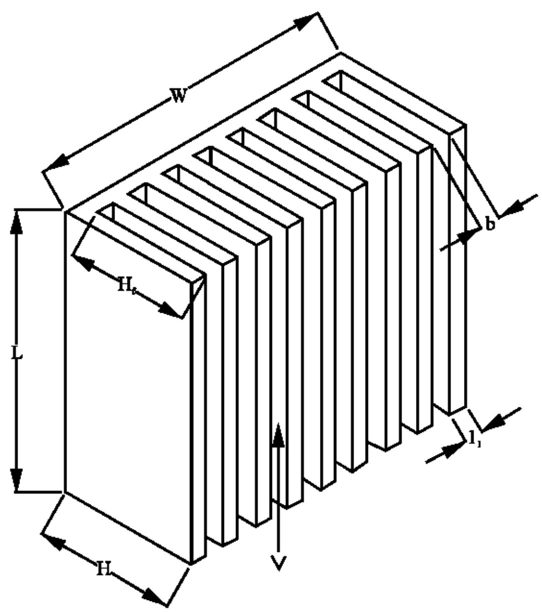

Fig. 3: Heat sink design parameters

$$C_p = \frac{1.007\ KJ}{Kg.K}$$

$$K = \frac{26.3\times10^{-6}\ KW}{m.K}$$

$$\rho = \frac{1.1614\ kg}{m^3}$$

$$\Pr = \frac{C_P \mu}{K} = \frac{1.007\times184.6\times10^{-7}}{26.3\times10^{-6}} = 0.707$$

The composite model for forced convection for the plate finheat sink is given by the equation:

$$Nu_i = \left(\frac{1}{\left(\frac{R_e^*.\Pr}{2}\right)^3} + \frac{1}{\left(0.664\sqrt{R_e^*}.\Pr^{0.33}\sqrt{1+\frac{3.65}{\sqrt{R_e^*}}}\right)^3} \right)^{-\frac{1}{3}}$$

where, Nu_i = Ideal Nusselt's number (equation developed by Teertstra *et al*., 2001).

Teertstra *et al*. (2001) modelled the heat sink as (N_{fin}-1) parallel plate 2 dimensional channels where channel width (space between fins, b) is used as the characteristic length. The Reynold's number, R_e is thus defined by:

$$R_e = \frac{\rho.V.b}{\mu}$$

The channel width, length and Reynold's number are combined to obtain the channel Reynolds number, R_e^*, analogous to the channel or Elenbaas Rayleigh number in natural convection:

$$R_e^* = R_e.\frac{b}{L}$$

$$R_e^* = \frac{1.1614\times24.34\times(2.9\times10^{-3})^2}{184.6\times10^{-7}\times0.135} = 96.51$$

$$Nu_i = \left[\frac{1}{\left(\frac{96.51\times0.707}{2}\right)^3} + \frac{1}{\left(0.664\sqrt{96.51}\times0.707^{0.33}\times\sqrt{1+\frac{3.65}{\sqrt{96.51}}}\right)^3} \right]^{\frac{1}{3}} = 6.67$$

Average heat transfer co-efficient,

$$h = \frac{Nu_i.K_{air}}{b} = \frac{6.67\times26.3\times10^{-3}}{0.0029} = \frac{60.11W}{m.K}$$

$$\text{Fin efficiency}(\eta_f) = \frac{\tanh(m.H_f)}{m.H_f}$$

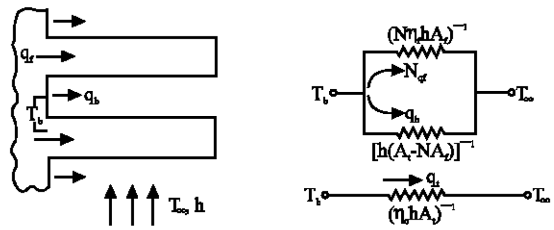

Fig. 4: Thermal resistance across heat sink fins

Incropera *et al*. (2007) where:

$$m = \sqrt{\frac{hP}{k_{fin}A_c}}$$

$$P = \text{Fin perimeter} = 2t_f + 2L$$

$$A_c = \text{Fin c/s area} = t_f L$$

$$P = (2\times0.002) + (2\times0.135) = 0.274 \text{ m}$$

$$A_c = 0.002\times0.135 = 270\times10^{-6} \text{ m}^2$$

$$m = \sqrt{\frac{60.11\times0.274}{167\times270\times10^{-6}}} = 19.11$$

Hence:

$$\eta_f = \frac{\tanh(19.11\times0.08)}{19.11\times0.08} = 0.60$$

Heat sink thermal resistance (Fig. 4), R_{hs} is then given by:

$$R_{hs} = \frac{1}{\eta_o hA_t}$$

where, η_o = Overall sueface efficiency which characteristic an array of fins and the base to which they are atteched (Incopera *et al*., 2007).

$$\eta_o = 1 - \frac{NA_f}{A_t}(1-\eta_f)$$

Where:

A_f = Fin surface area = $2H_f L$

A_t = Area associated with both the fins and exposed surface of the base (prime area)

$$A_t = Na_f + A_b$$

where, A_b = Prime surface/base area of spaces between fins:

$$A_b = (N_{fin} - 1).\ b.\ L = (25 - 1)\times0.0029\times0.135 = 0.009 \text{ m}^2$$

$$A_f = 2\times0.08\times0.135 = 0.022\ m^2$$

$$A_t = (25\times0.022) + 0.009 = 0.559$$

$$\eta_o = 1-\frac{25\times0.022}{0.559}(1-0.60) = 0.606$$

Hence:

$$R_{hs} = \frac{1}{\eta_o hA_t} = \frac{1}{0.606\times60.11\times0.559} = \frac{0.049K}{W}$$

0.049 K/W>>0.05 K/W (Maximum allowable heat sink resistance). Therefore, heat sink design meets requirements.

RESULTS AND DISCUSSION

Experimentation: The following tests were carried out on the fast beverage chiller: Two of the TEC modules were connected to two 12 V computer power supplies and the other two were connected to a variable voltage Lab power supply. The blower was connected to 110 V ac. The readings for the voltage and current are shown in Table 1. The airflow rate through the two warm compartments were also measured and the readings were shown in Table 2. Thermocouples were installed at various points in the beverage chiller to take the temperatures at these points. The placement of the thermocouples is shown in Fig. 5. Where, T1 indicates temperature at hot side heat sink base, $T_{base,hot}$; T2 measures temperature at cold side heat sink base, $T_{base\ cold}$; T3 measures cold compartment temperature, T_{cold} and T4 measures water temperature, T_{water}.

Table 1: The voltage and current readings

TEC module	Voltage/V	Current/A	Power/Watts
TEC 1	9.85	7.5	73.88
TEC 2	9.85	7.5	73.88
TEC 3	11.69	8.4	98.20
TEC 4	11.03	8.5	93.76

Table 3 and 4 shows the experimental results for the beverage chiller when allowed to reach operating temperature with nothing in the cold compartment. The same quantity of water in the same glass jar was then placed in: the freezer compartment of a household refrigerator, the cold space of a household refrigerator. Readings were taken until the desired temperature of 6°C was obtained results shown in Table 5. The following graphs were then plotted to observe the performance of the Beverage chiller.

The design power input obtained in the calculations for the TEC modules used was 98 W. Figure 6 shows the performance of a TEC module variation with input power: Performance in this case refers to the cooling rate Q_c. The Fig. 6 shows that as the input power increases,

Table 2: The air flow-rate through the two warm compartments

	Left side warm compartment		Right side warm compartment	
Position	Air flow (cfm)	Air speed (ft min^{-1})	Air flow (cfm)	Air speed (ft min^{-1})
Before heat sinks	530	1879	880	3114
After first heat sink	160	587	450	1593
After second heat sink	120	452	361	1278

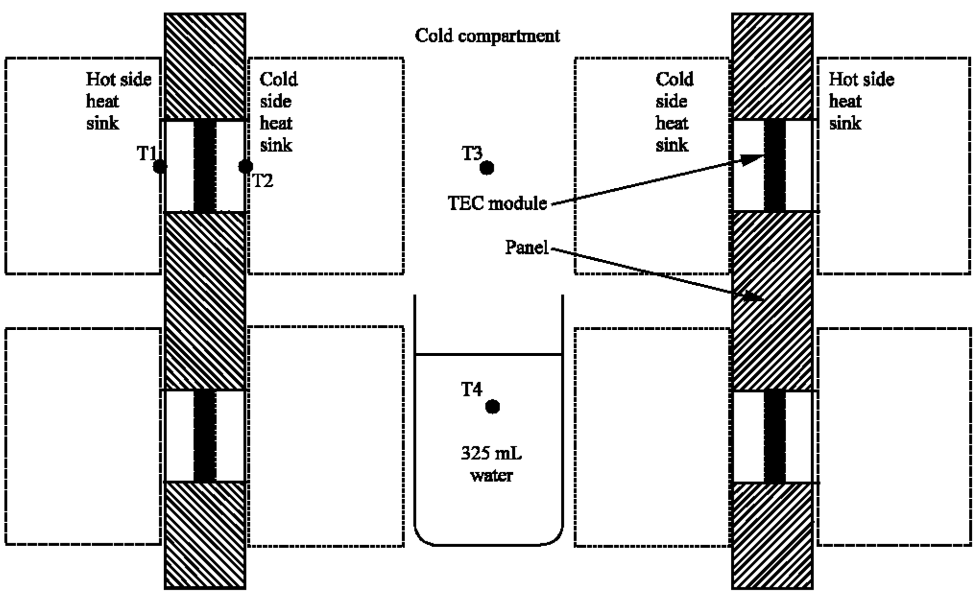

Fig. 5: The placement of thermocouples (T1, T2, T3 and T4)

Table 3: Temperature readings for the cold and hot compartment

Time/min	$T_{base,hot}$/°C	$T_{base\ cold}$/°C	T_{cold}/°C
0	32.7	32.7	32.7
5	61.7	21.6	20.6
10	60.9	14.3	12.8
15	60.4	11.4	10.1
20	59.7	8.4	6.9
25	58.9	6.1	4.8
30	59.0	5.2	3.8
35	58.8	4.4	2.9
40	58.8	3.8	2.3
45	58.7	3.2	1.7

Table 4: Temperature readings for the cold and hot compartment when water was tested

Time after water is placed inside/min	Total time chiller is on/min	$T_{base,hot}$ /°C	$T_{base\ cold}$ /°C	T_{cold} /°C	T_{water} /°C
0	46	59.0	4.0	5.7	31.5
5	51	59.2	5.5	5.6	25.9
10	56	59.4	6.0	5.6	22.4
15	61	59.4	6.0	5.5	19.4
20	66	59.5	5.9	5.3	17.0
25	71	59.4	5.7	4.8	14.8
30	76	59.5	5.5	4.6	13.2
35	81	59.5	5.2	4.1	11.7
40	86	59.5	5.0	3.9	10.4
45	91	59.5	4.8	3.7	9.4
50	96	59.4	4.6	3.4	8.3
55	101	59.4	4.5	3.2	7.7
60	106	59.3	4.3	3.0	7.0
65	111	59.5	4.2	2.9	6.2
69	115	59.4	4.1	2.7	6.0

performance also increases. Operating at close to the maximum is inefficient; most applications do operate at 40-80% of input maximum power of TEC modules.

For the selected TEC modules 40% input power max is 160.16 W. Therefore, the cooling rate of the beverage chiller would have been improved if the design requirement of 98 W for each TEC module was met. If the available power supplies supported supplying 160 W, this would have increased the cooling rate even more. Inefficient forced convective heat transfer within the cold compartment also adversely affected the cooling rate of the beverage chiller as shown in Fig. 7. A small fan was used to circulate air within the cold compartment. The fan had small cubic feet per minute (cfm) rating and its positioning also was not optimal. Having a fan set-up where the fans circulate air through the fins of all the cold side heat sinks and then directing a blast of cold air over the jar containing the water to be cooled would have resulted in better cooling times.

Figure 8 compares the beverage chiller's cooling time with cooling times obtained from the freezer space and cold space of a household refrigerator. All three tests were carried out on 325 mL of water in a glass jar. The result shows that for the refrigerator freezer space, the temperature of the water decreased linearly with increasing time. However, for the beverage chiller, the water temperature decreased exponentially with increasing time. In other words, cooling rate for the refrigerator was constant while for the beverage chiller it decreased exponentially. Figure 8 also shows that the freezer took 61 min to cool the water to 6°C while the beverage chiller took 69 min. It can be seen that for the majority of the

Table 5: Variation of temperature with time for 325 mL water placed in freezer and cold space of refrigerator

Freezer		Cold space	
Time (min)	Temperature (°C)	Time (min)	Temperature (°C)
0	31.7	0	31.70
10	26.3	15	28.10
20	21.2	30	24.30
30	16.8	45	21.20
40	13.0	60	19.50
50	10.1	75	19.00
60	6.3	90	17.50
61	5.9	105	16.15
		120	14.78
		135	13.20
		150	12.45
		165	11.70
		180	11.03
		195	10.50
		210	9.83
		225	9.25
		240	8.50
		255	7.30

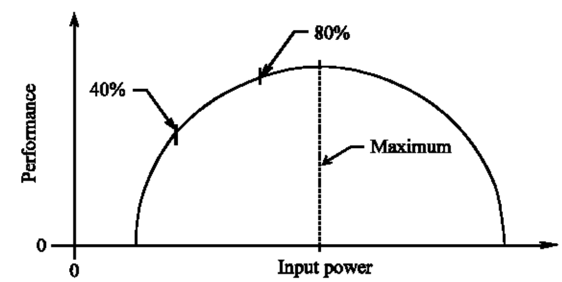

Fig. 6: Variation of performance with input power for a TEC module

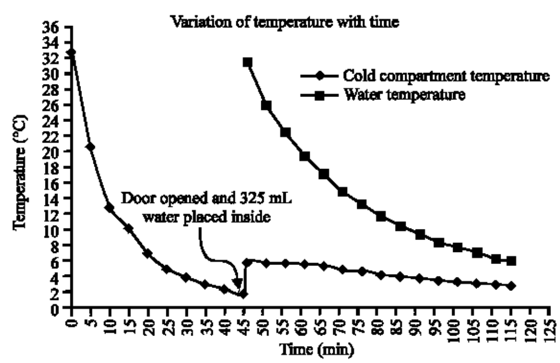

Fig. 7: Variation of temperature with time of the cold compartment of the beverage chiller and of 325 mL water placed inside the cold compartment

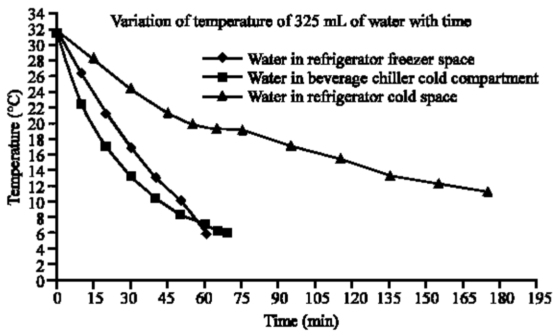

Fig. 8: Variation of temperature with time of 325 mL of water placed inside various cold spaces

cooling time, the beverage chiller was cooling at a faster rate than the freezer. But by virtue of the exponential cooling versus linear cooling, cooling rate for the beverage chiller was decreasing while the freezer cooling rate was constant throughout the cooling process. This caused the beverage chiller's cooling rate to eventually reach a point where it was lower than the freezer's cooling rate. This happened at around 7°C as shown in the Fig. 8 where the lines crossed. It must also be noted that the temperature within the freezer space was measured at -17.4°C while that of the beverage chiller's cold compartment was on average around 3.9°C (it started at 5.7°C and dropped to 2.7°C during the water cooling process).

Therefore, at the point in time at which the required water temperature of 6°C was attained, the temperature difference between water and cold space was 23.4°C for the freezer and only 3.30°C for the beverage chiller. This shows that although the beverage chiller took more time to cool the water, the heat transfer process from the water to the cold compartment was more efficient than in the freezer. The cold space of the refrigerator was measured at 5.1°C and took over 2 h to cool to 7.2°C which was very much slower than the beverage chiller.

The theoretical analysis showed a supply voltage of 7 V and current of 14 A obtained for the TEC modules from the TEC module performance curves. Which gives a wattage of (7×14) = 98 W. However, to gain this power rating of 98 W, 11.69 v had to be supplied which caused a corresponding current of 8.4 A in the TEC module. Design calculations stipulated a volume flow rate at the beginning of each heat sink of 150 cfm. This design requirement was met. For the left side warm compartment, the flow rate at the beginning of the first heat sink was 530 and 160 cfm at the beginning of the second heat sink. For the right side warm compartment, the flow rate at the beginning of the first heat sink was 880 and 450 cfm at the beginning of the second heat sink. The difference in the flow rates on the left and right side can be explained by the fact that the left side had a longer duct with an extra 45° angle than the right side. Therefore, pressure loss along the duct that supplies the left side would have been greater than that along the duct supplying the right side.

The dimensions and different airflows for heat sink thermal resistance was obtained from the mathematical analysis. The use of excel was instrumental in speeding up this process and the above calculations only represent the final result with the desirable values. Variables that were found to decrease the heat sink's thermal resistance are as follows: Increasing air flow-rate across heat sinks, Increasing Fin length (H_f), Increasing Fin width (L), Increasing number of fins (n_f), while keeping Heat sink width (W) the same, hence decreasing fin thickness (t_f) also; Increasing fin thickness (t_f), while keeping Heat sink width (W) the same, hence decreasing space between fins (b) also.

CONCLUSION

The theoretical determination of TEC module and heat sinks shows that for a cooling time of 2 min and a beverage size of 474 cc (16 Oz) mL, 6 TEC modules and hence 12 heat sinks (one for each side of the TEC module) were needed. However by increasing cooling time to 4 min and decreasing beverage size to 325 mL, 4 TEC modules and hence 8 heat sinks could be used.

REFERENCES

Cengal, Y. and B. Michael, 1998. Thermodynamics: An Engineering Approach. McGraw Hill, Hightstown.

Gilley, M.D. and R.L. Webb, 1999. Thermoelectric refrigerator with evaporating/condensing heat exchanger. United States Patent, No. 6003319, December 21. http://www.freepatentsonline.com/6003319.html.

Goldsmid, H.J., 2009. Introduction to Thermoelectricity. Springer Heodelberg Dordrecht, London, pp: 1-3.

Incropera, F.P., D.P. Dewitt and T.L. Bergman, 2007. Fundamentals of Heat and Mass Transfer. 6th Edn., Wiley, New York, pp: 152-155.

Riffat, S.B., S.A. Omer and M.A. Xiaoli, 2001. A novel thermoelectric refrigeration system employing heat pipes and a phase change material: An experimental investigation. Renewable Energy, 23: 313-323.

Rowe, D.M. and C.M. Bhandari, 1983. Modern Thermoelectric. Holt, Rinehart and Winston, London, pp: 7-25.

Sofrata, H., 1996. Heat rejection alternatives for thermoelectric refrigerators. Energy Convers. Manage., 37: 269-280.

Teertstra, P., M.M. Yovanovich and J.R. Culham, 2001. Analytical forced convection modeling of plate fin heat sinks. J. Elect. Manuf.., 10: 253-262.

Vian, J.G., D. Astrain and M. Dominguez, 2002. Numerical modeling and design of a thermoelectric dehumidifier. Applied Thermal Eng., 22: 407-422.

Webb, R.L., 1998. Advanced heat exchange technology for thermoelectric cooling. J. Elect. Pack., 120: 98-105.