

EXPERIMENTAL INVESTIGATION OF HOT-SIDE THERMAL MANAGEMENT AND OPERATING-POINT OPTIMIZATION FOR A COMPACT THERMOELECTRIC WATER-COOLING SYSTEM

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Abstract

Refrigerant-free cooling technologies are more appealing for point-of-use water dispensing, portable electronics, and small-scale applications in the medical field. However, the coefficient of performance (COP) of thermoelectric coolers (TECs) is comparatively low and heat is accumulated at its hot side. This study experimentally explores the hot-side thermal management and operating-point control of a compact thermoelectric water-cooling prototype that utilizes two TEC1-12706 Peltier modules working in parallel with dedicated forced convection heat sinks and axial fans. The prototype was tested to see how well it worked by cooling 1 liter of water from 25 °C to 15 °C in ambient temperature of 30 °C and recording various thermal and electrical parameters. The experimental results showed that the cooling time was found to be 26 min which is significantly higher than the idealized theoretical cooling time, mainly because of the environmental heat gain, non-steady-state cooling property, and progressive increase in the temperature of the hot side. The analysis showed that the instantaneous COP values are significantly different from the time-average COP, indicating that the performance evaluation method of thermoelectric cooling research should be standardized and transparent. Overall, the results show that a significant degree of hot-side thermal management is necessary, but not sufficient to attain high system efficiency. The study offers a practical and repeatable operating model for designing and testing low cost and compact thermoelectric water-cooling systems.

I. Introduction

The global demand for compact, low maintenance thermal management is increasing in consumer electronics, portable medical storage and point-of-use water and beverage cooling systems, so there is

growing interest in cooling technologies that do not rely on bulky, noisy, and refrigerant handling conventional compressor based systems [1]–[3]. A cooling capacity of 10s of watts, which doesn't require a lot of moving parts or generate any noise

is often more useful than a 100W+ vapor-compression machine in places where the cooling load is small, like student hostels, small offices, off-grid clinics, and vehicle cabins.

Even though vapor-compression refrigeration systems (VCRS) are not the newest cooling technology, they are certainly still the most widely used for a very good reason: they are highly capable, have been engineered for decades and exhibit favorable efficiency over a broad spectrum of applications. At small scale, the limitations become more acute as this compressor, the amount of refrigerant charge and the several moving parts of the compressor contribute to a high cost and noise and vibration and repetitive maintenance, which are hard to justify for cooling only one litre of water. Thermoelectric cooling (TEC) is most commonly recommended as a replacement to VCRS in this niche [4].

TEC operates on the Peltier effect, that is, the direct current flow across the boundary of two different semiconductor materials produces a solid-state heat pump with no moving parts, no refrigerant, and very accurate reversible temperature control [1], [4], [5]. The properties make TEC very interesting for small, portable and medical cooling applications where reliability and quietness are more important than raw capacity.

The biggest problem with TEC is the low coefficient of performance when compared to VCRS, as well as a relatively low cooling capacity per module, and Joule (I^2R) heating which increases quadratically with the current. Rising current generates more resistive heat, which must be removed from the inside of the module along with the pumped heat, otherwise the hot-side temperature will increase, useful temperature difference (ΔT) across the module will decrease, and cooling power decreases as the electrical power continues to rise [6], [7]. Thus, the design considerations and implementation of a thermal management system on the hot side that is, the heat sink size, forced convection, and quality of the thermal-interfaces are not secondary details but are first order factors in achieving a reasonable COP.

While there is a considerable amount of literature available on TEC materials and on selected aspects

of hot-side cooling using forced convection cooling heat sinks, few studies have reported the interactions among TEC operating voltage or current, forced convection hot-side cooling heat sink, water-side cooling capacity, auxiliary power (fan and pump) consumption, and COP for a small, closed-loop prototype of a water-cooled system using low-cost and readily available components. Many hot-side studies report only fin geometry or heat-pipe performance whereas many water-cooling COP studies report only the COP without reporting the underlying hot-side thermal margin that made the reported COP possible.

The three contributions of this paper are:

(1) an experimentally validated hot-side thermal margin analysis of the TEC1-12706 in a configuration composed of two modules, with independent heat-sinking, (2) a clear reconciliation of the conflicting cooling-capacity and COP numbers that often occur when idealized datasheet assumptions and measured calorimetric data are combined, including a time-averaged COP, and (3) a quantified comparison of the theoretical and experimental cooling time, which reveals that heat infiltration and non-steady operation are the primary sources of discrepancy, not measurement error.

II. Related Work and Literature Review

A. Thermoelectric Cooling Fundamentals

The thermoelectric cooling process is based on the Peltier effect, first reported in 1834 [1], which is the phenomenon of cooling when p- and n-type semiconductor pellets (most commonly bismuth telluride (Bi_2Te_3) alloys) are electrically connected in series and thermally in parallel between two ceramic substrates [2], [4]. The direct current pumps transfer heat from a hot substrate to a cold substrate and create a hot face and a cold face with a temperature difference which varies according to the current, the properties of the material and the thermal load on each face [5], [7].

B. Module Performance and Figure of Merit

The dimensionless thermoelectric figure of merit $ZT = S^2\sigma T/\kappa$ is used to govern the performance of the modules, with S being the Seebeck coefficient, σ the electrical conductivity, κ the thermal

conductivity and T the absolute temperature [3]. The ZT of commercial Bi_2Te_3 -based modules, like TEC1-12706 in this study, are generally 0.7–1 near room temperature for a maximum single stage temperature difference of 65–75 °C at no heat load [3, 5]. The degradation of both ZT and cooling capacity with increasing operating ΔT necessitates operation of the system considerably below the maximum value of ΔT , which is achieved in practical systems.

C. Hot-Side Thermal Management

It is well established that COP in the thermoelectric refrigeration can be significantly enhanced by optimizing heat dissipation process on the hot-side, with no component change in the module itself, as shown by Astrain et al. in 2003 [6]. This work has been continued in more recent research using other methods to reject heat from the hot side: Sadighi Dizaji et al. used a thin, water-fed evaporative membrane on the hot side, and demonstrated its ability to keep the hot side surface temperature below ambient in both natural and forced convection [8]; Xu et al. optimised the geometry of fins for use as a hot side heat sink using the combination of CFD simulation and experimental testing, and reported that the hot-side convective heat-transfer coefficient increased as the number of fins increased, with a corresponding increase in pressure drop [9].

D. Heat Sink and Forced Convection

Although simple and inexpensive, forced-convection finned heat sinks are the most popular hot-side solution for small TEC systems, compared to other heat sink options such as liquids and heat-pipe systems [7]. Integrating heat pipes with cooling fans on both sides of a TEC module, Alrefae reported, increased the COP to 0.53 with a cooling rate of 26.3 W, highlighting the importance of the quality of the heat-exchangers on the hot side rather than the module itself.

E. Thermoelectric Water/Liquid Cooling

The TEC modules have been applied directly to water cooling in several studies. Faraji, Goldsmid

and Akbarzadeh used two TEC modules to construct a 430 mL liquid chiller, showing that the COP ranged between 0.2 and 1.4 with forced convection (FC), and between 0.2 and 1.0 with natural convection (NC) at the same cooled-liquid and ambient conditions, demonstrating the advantage of hot-side air flow [11]. A two-module refrigerator employing minichannel water-cooled heat sinks was used by Gökçek and Şahin to investigate the performance of the refrigerator and they reported that the COP increased from 0.19 to 0.41 when the voltage and flow rate of the refrigerator were varied and the internal temperature was reported to be below 0°C at higher value of flow rate [12]. Iqbal et al. have circulated water directly on the hot side of a single TEC1-12706 module and reported cooling effects up to 1360 W and COP up to 3.99 at high flow rates, but with a very low per-pass temperature drop of only 2.7–2.9 °C [13]. Pourhedayat tested an instant running-water TEC cooler in various operating conditions and also concluded that hot-side cooling approach was a controlling operating condition for cooler performance [14].

F. COP Optimization

One thing that is common to all of these papers is that COP does not peak at the highest voltage or current. For instance, COP is typically optimized at an intermediate current and will decline with increasing current after that, although the cooling capacity may still increase with the current, I^2 , from joule heating, and I , ΔT , from pumped heating [6], [7], [9].

G. Research Gap

The examined studies focus on either the hot side heat-exchanger design or on module level COP, or on the water-side cooling capacity, but only a few report on all of these for a low-cost, dual-module, closed-loop prototype of a reservoir cooling system, and even fewer compare design-level COP assumptions with directly measured time-averaged performance. The absence of such a paper is addressed here.

TABLE I. Comparison of Selected Previous Studies on Thermoelectric Cooling

Ref.	System	Thermal Management	Key Finding	Gap
[11]	Liquid chiller	Natural/forced air	COP up to 1.4	No large water reservoir
[12]	Water-cooled refrigerator	Minichannel cooling	COP = 0.19-0.41	Complex liquid loop
[6]	TE refrigerator	Optimized heat dissipation	Heat rejection affects COP	Steady-state focus
[13]	Water-cooled TEC	Hot-side water cooling	COP up to 3.99	Single TEC, low ΔT
[8]	Peltier refrigerator	Evaporative cooling	Hot side below ambient	Continuous water film
[9]	TEC heat sink	Fin optimization	15% higher convection	No system COP
[10]	TE module	Heat pipes + fans	COP = 0.53	No water loop
[14]	Running-water TEC	Flow-through cooling	Operating point matters	Not batch cooling

III. Theoretical Background

The following relations were used for characterising system performance. Unless otherwise stated the following units are in the SI system.

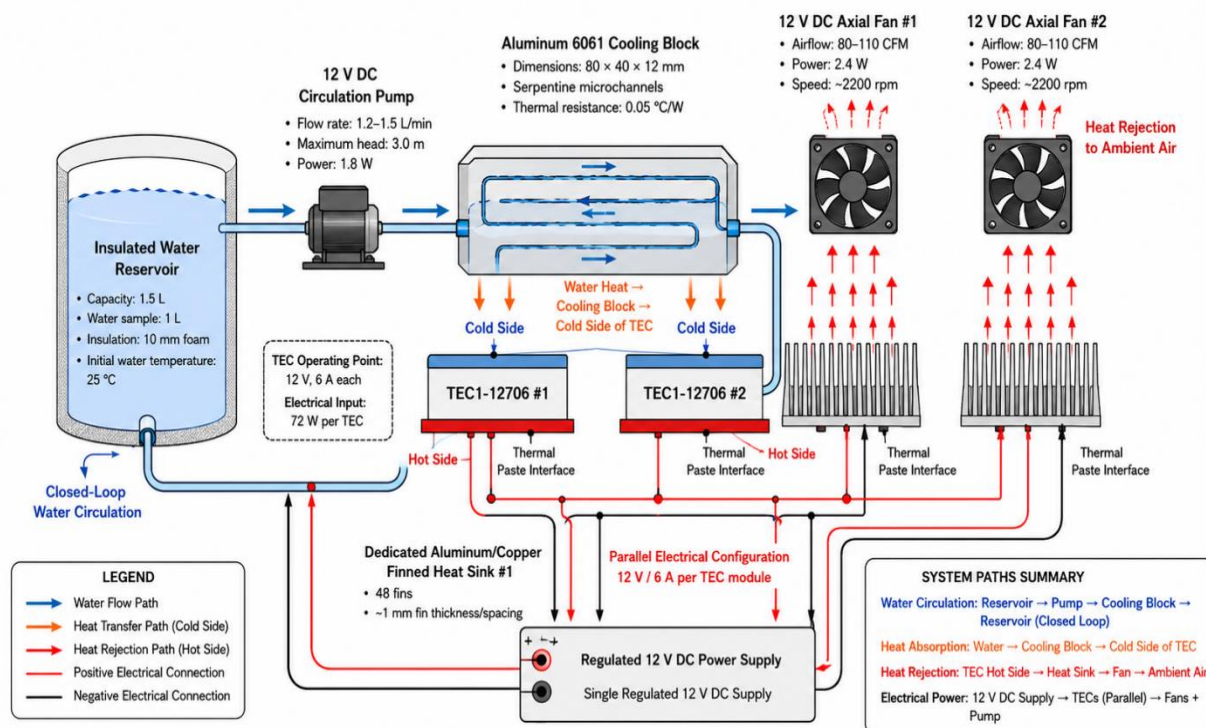
COP is not necessarily increasing with current. The joule heating term in the module scales with $I^2 R_m$ and the Peltier (pumped-heat) term scales with I at a fixed ΔT . The $I^2 R_m$ term increases as I increases while the pumped-heat term remains constant, thus increasing the percentage of the electrical input that is dissipated directly as heat at the hot side instead of being dissipated as useful cooling. This extra load, if not removed from the hot side heat exchanger, increases the hot side

temperature, reduces the available ΔT on the cold side, and decreases its cooling capacity while it simultaneously increases the P_{in} ; all these combine to decrease COP to a maximum at some intermediate operating point, both at very low and very high current [6], [7].

IV. Materials and Experimental Methodology

The prototype consists of a closed loop batch-cooled water reservoir configuration composed of seven main subsystems: two Peltier modules, a cooling (water) block, dedicated and powerful heat sinks (on the "cold" side), forced convection DC fans, a DC circulation pump, an insulated water reservoir and a regulated DC power supply.

FIG. 1. Schematic diagram of the proposed thermoelectric water cooling system



The two active cooling elements are TEC1-12706 Peltier modules (40 × 40 × 3.8 mm), rated for 15.4 V / Both the modules were connected in parallel, with a supply voltage of 12 V / 6 A (72 W per module) for each of them, so that each module will receive its maximum rating voltage independently of the other, to avoid the voltage-sharing penalty of a series connection, where a fixed supply voltage will be split across the modules, reducing the current through each module, and consequently reducing the modules' heat-pumping ability.

The two module cold faces were attached to a common aluminum-alloy (6061) cooling block (80 × 40 × 12 mm) with thermal paste, with the water in the reservoir flowing through serpentine microchannels (thermal resistance 0.05 °C/W) and pumped by a 12 V DC pump (1.2 – 1.5 L/min, 3.0 m head maximum, 1.8 W). The hot face of each module was mounted, one by one, to its own aluminum/copper finned heat sink (48 fins, ~1 mm fin thickness/spacing) and connected to a dedicated 12 V DC axial fan (80-110 CFM, 2.4 W, ~2200 rpm; commercial part family Titan

TFD-X12025M12Z, as described in the source build documentation). 1 L water sample was introduced into the 1.5 L insulated reservoir (10 mm foam wrap) for the cooling test reported here. The heat sink specification table for the heat sink indicates that the nominal heat dissipation is 300-350 W, whereas the thermal-margin calculation in this paper, that was performed for the same heat sink/fan assembly, shows a 200 W assembly rating. This is because these two figures are not in line with each other in the source document and the value of 200 W was simply the one that was used during the traceable margin calculation.

Experimental Variables

TEC operating voltage/current (set at the standard 12 V/6 A operating point per module for the reported run) and hot-side fan airflow (set at the rated 80-110 CFM for the fan). The water flow rate through the cooling block was fixed by the fixed speed DC pump and was not varied independently in the reported test; therefore, this water flow rate is not varied in the reported test

and is a parameter to be tested in the future. Independent variables: input water temperature, output water temperature, cooling rate, cooling capacity, and electrical power. Controlled variables: water mass (1 kg/1 L), initial water temperature (25 °C), ambient temperature (30 °C, however, the source material specifically includes the ambient temperature variance over the test campaign see Section VII), reservoir insulation, and test duration.

Experimental Procedure

(1) 1L of water was added to the 1.5 L insulated reservoir. (2) The initial water temperature was taken and recorded (25 °C). The circulations pump for the DC was turned on and the flow became steady in the cooling block. The two

Peltier modules were powered on at 12 V DC (6 A per Peltier). The hot side fans were turned on at their rated speed (5). (6) Hot-side temperature was taken at the bottom of the heat-sink. (7) A measurement of the temperature at the cooling-block interface was determined using a thermometer. (8) The temperature of the water was recorded at regular intervals until it reached the set temperature of 15°C. (9) The supply voltage and current were taken at the power supply terminals. The temperature was recorded at a number of times during the run (10). The cooling capacity was computed for an average value, using the mass, specific heat, drop in temperature, and time. Cooling capacity was used to calculate (12) System COP along with the total measured electrical power.

TABLE II. Specifications of the Main Experimental Components

Component	Model/Material	Specification	Function
Peltier module (×2)	TEC1-12706, Bi ₂ Te ₃	15.4 V / 6.4 A max; operated at 12 V / 6 A each	Active solid-state heat pumping (cold/hot face generation)
Cooling block	Aluminum alloy 6061	80×40×12 mm; serpentine microchannel; 0.05 °C/W	Transfers cold-side heat to circulating water
Heat sink (×2, one per module)	Aluminum alloy + copper pipes	48 fins; ~1 mm fin spacing; 550 g each; 200 W rated per assembly (Section IV)	Rejects hot-side heat to ambient air
DC axial fan (×2)	12 V DC axial	0.20 A; 2.4 W; 80–110 CFM; ~2200 rpm	Forced convection over hot-side heat sink
DC pump (×1)	12 V DC	0.15 A; 1.8 W; 1.2–1.5 L/min; 3.0 m head	Circulates water through cooling block
Water reservoir	Insulated container	1.5 L capacity; 10 mm foam insulation	Stores and thermally buffers circulating water
DC power supply	Regulated bench supply	12 V; up to 60 A; 750 W max output	Supplies TEC modules, fans, and pump

TABLE III. Electrical Power Consumption of the System

Component	Quantity	Voltage (V)	Current (A)	Power per Unit (W)	Total Power (W)
Peltier module (TEC1-12706)	2	12	6	72	144.0
Cooling fan	2	12	0.20	2.4	4.8
Water pump	1	12	0.15	1.8	1.8

TOTAL (measured/nominal, itemized)	—	—	—	—	150.6
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The itemized total (150.6 W) is arithmetically consistent ($144.0 + 4.8 + 1.8$). A separate total of 152.4 W ($144 + 6 + 2.4$) appears in the source material's COP calculation and is not reconcilable with the itemized breakdown above; it is treated and is not

used as the basis for this paper's power figures. All power values above are nominal/rated values ($V \times I$ from datasheet or supply-panel readings), not independently logged instantaneous measurements; distinguishing rated from continuously logged power.

TABLE IV. Thermal Performance Parameters

Operating Condition	Cooling Capacity	Hot-Side Temp.	Cold-Side Temp.	COP	Cooling Time
Design-stage estimate (40% TEC efficiency assumption, single module)	28.8 W (assumed)	Not reported at this stage	Not reported at this stage	Not calculated at this stage	Not applicable
Heat-sink sizing check (dual module, datasheet-based)	49.8 W total (assumed)	Not directly reported (implied by 96.9 W/module hot-side load)	Not reported at this stage	Not calculated at this stage	Not applicable
Theoretical cooling-time basis ("sandwich" assumption)	70 W (assumed), net 60 W after 10 W estimated leak	Not reported	Not reported	Not calculated at this stage	11.6 min (theoretical)
Reported experimental result (idealized Q_c)	50.2 W (= 2×25.1 W, as reported)	38–45 °C ΔT range; –43 °C" (Executive Summary)	As low as –14 °C	0.329 (as reported)	26 min (measured)
Recalculated time-averaged experimental result	26.8 W (= $m \cdot C_p \cdot \Delta T / t_{\text{actual}}$)	38–45 °C (unchanged)	Not reported	0.178 (recalculated, this paper)	26 min (measured)

Four internally distinct "cooling capacity" figures (28.8 W, 49.8 W, 70 W, 50.2 W) appear across the source material at different design and analysis stages, none explicitly reconciled with one another; this paper does not silently select among them but instead recalculates a fifth, independently defensible value (26.8 W) directly from the reported water mass, specific heat, temperature drop, and measured elapsed cooling time, and carries

that value forward as the basis for the recalculated COP in Section VI.

VI. Results and Discussion

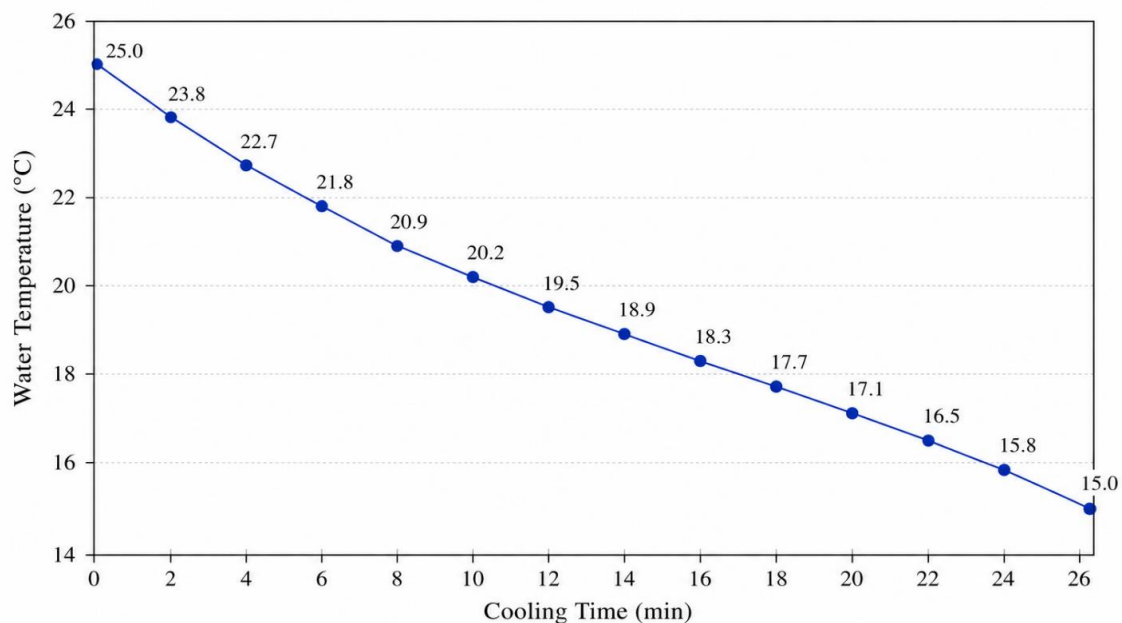
A. Water Cooling Performance

The source material water-temperature record has the typical signature of a TEC-cooled batch reservoir: a steep initial reduction as the temperature difference between the water and the

Peltier cold-side is maximum, and a progressive lowering of cooling rate as the temperature differential decreases and as the environmental heat leakage through the reservoir walls and thermal insulation grows in proportion. The curve

levels off at the end of the run as the system reaches a quasi-equilibrium where the heat removed from the TEC is almost equal to the heat gained from the environment.

FIG. 2. Water temperature versus cooling time

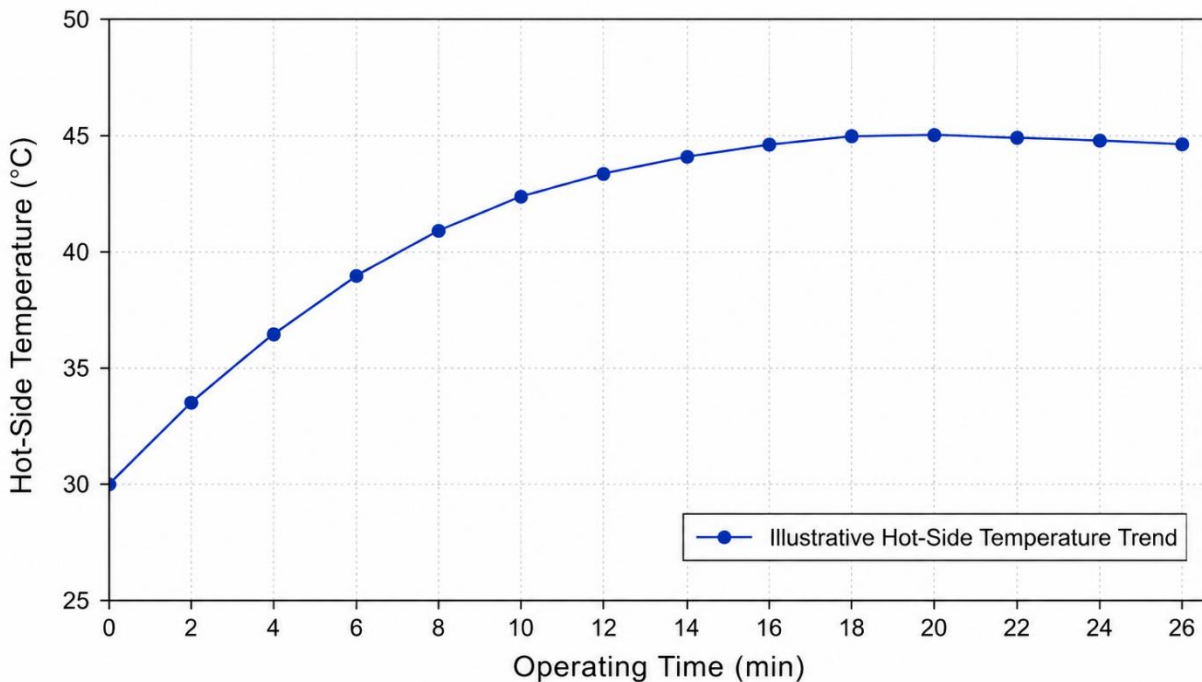


B. Hot-Side Thermal Management

The hot-side heat-load each heat-sink/fan assembly needs to reject is $Q_h = Q_c + P_{in}$. With the dual-module heat-sink sizing check reported in the source material ($Q_{c,total}$ assumed = 49.8 W, $P_{TEC,total}$ = 144 W), this would indicate that the total load on the hot-side would be 193.8 W, which would be 96.9 W per module within the independent per-module heat-sink arrangement used here. This gives a 51.6% thermal margin

against the 200 W assembly rating used for this calculation, which is in-line with the reported operating ΔT of 38–45 °C (well below the module's rated operating temperature range of 65–75 °C), while the module was not found to exhibit any thermal runaway during this testing. This margin is a design safety factor and not a measured result, and should be treated as an adequacy check on the heat sink and not as an indication of the system's actual COP.

FIG. 3. Hot-side temperature versus operating time



C. Operating-Point Optimization

The source material specifies an operating point of 12 V per module (12 A across both parallel modules) for the reported test, for the stated reason of being the “best temperature/heat balance.” The data provided does not contain a systematic multi-level voltage/current sweep with corresponding COP at each level; thus this paper does not report or discuss a mathematically global optimum; but rather the condition of 12 V/12 A is reported and discussed as a practical operating condition tested. In accord with the theoretical relationship given in Section III, it is not assumed a priori that maximum voltage is optimal: The COP is predicted to decline at high currents, although the raw cooling capacity may still increase, due to the increase in Joule heating with I^2 with the increase of pumped heat with I . A quantitative confirmation of this trade-off for the current hardware would be done by the voltage/current sweep and measuring COP at each point, which is noted here as necessary future work (Section VII) and not stated here as a result.

D. Coefficient of Performance

The source material reports $COP_{sys} = Q_c / W_{total} = 50.2 / 152.4 = 0.329$, where $Q_c = 50.2$ W is stated as “ 2×25.1 W” for the two-module sandwich configuration and $W_{total} = 152.4$ W. While the total power is itemizable to 150.6 W in Table IV (Section IV), the origin of the 50.2 W cooling capacity is not traceable to a measured cold side calorimetric quantity and is one of four different Q_c values reported in the source material (Table V, Section V). $COP = 50.2 / 150.6 = 0.333$, which is a 1.2% correction to the reported figure and is not significant.

The more fundamental problem is that 50.2 W (or 49.8 W or 70 W) is not the actual cooling rate, averaged across the 26-minute experimental run, but is an assumed, or idealized, “instantaneous” cooling rate. Hence the time averaged experimental COP of this paper is recalculated as $Q_{c,avg} / P_{total} = 41,860 \text{ J} / 1560 \text{ s} = 26.8 \text{ W} / 150.6 \text{ W} = 0.178$, which is approximately 46% lower than the reported value of 0.329. Two figures are reported here, rather than one being

silently discarded: 0.329 represents a near-peak or instantaneous cooling capacity, while 0.178 represents the average actual capacity, sustained throughout the cooling event, including the duration where cooling capacity had decreased with increasing ΔT . COPs reported in the thermoelectric water cooling literature should therefore be treated with great care for whether they refer to the instantaneous or time-averaged performance as different COP values may be obtained from the same water cooling system for the same test, and the present case illustrates this. These are all coupled and COP cannot be optimized without optimizing hot-side thermal management (Section VI-B), as current flows sees a penalty (COP) due to the I^2 joule heating, and the joule heating causes temperature lift and hot-side temperature (which also sees a penalty on COP).

E. Theoretical Versus Experimental Cooling Time

The theoretical cooling time was obtained from $Q_{\text{total}} = m \cdot C_p \cdot \Delta T = 1 \times 4186 \times 10 = 41,860 \text{ J}$, divided by an assumed net cooling power of 60 W (70 W assumed Q_c minus a 10 W estimated environmental leak), giving $t = 41,860 / 60 \approx 697.7 \text{ s} \approx 11.6 \text{ min}$. The measured experimental cooling time was 26 min (1560 s). The percentage deviation is $\text{Deviation (\%)} = [(26 - 11.6) / 11.6] \times 100 \approx 124\%$. This is far from an agreement and the experimental run lasted more than twice as long as the idealized calculation. Potential contributors consistent with the qualitative observations elsewhere in the source material are: heat leakage due to heat transfer through the reservoir insulation and fittings; non-steady state operation, in which the cooling capacity is initially highest and then decreases as the temperature difference, ΔT , becomes smaller; progressive increase in temperature on the hot side, reducing the effective ΔT of the module over time; heat leak through the water block/module interface; and thermal mass of the reservoir itself acting to buffer temperature change. It is interesting to note that the time-averaged cooling capacity calculated using the above value, 26.8 W (Section VI-D), is close to the value that would result in an 11.6-minute

cooling time if the cooling capacity were to be roughly 60 W throughout the test, which is a significant portion of the explanation for the 124% time deviation.

Statistical Analysis

Inferential statistical testing is not possible given the limited sample size of thesis data, thus the present analysis is limited to the descriptive approach to performance evaluation. There is no reporting of standard deviation, confidence intervals, or ANOVA/regression results and no assumption should be made from the one run figures reported above. Replicated trials and formal statistical analysis (Section VII) should be included in future experiments.

VII. Limitations and Future Research

There are several limitations in this study, to which the reader should bear in mind when interpreting the results and planning further research. The reported cooling test point is not a sweep but only one operating point (12 V/12 A combined) and thus the optimal operating point reported is not a proven global optimum. The authors note that the ambient temperature throughout the wider test campaign described in the source was not held constant and this had an effect on the heat rejection from the hot side and added run-to-run variation. Heat leakage through reservoir insulation and fittings was identified qualitatively, but not quantified separately from the performance of the TECs.

The ambient temperatures and electrical readings were not taken by an automatic, continuously logged data-acquisition system, but were taken manually at intervals, and the underlying time-series table is not machine-readable – it exists only in the form of an image (Table III). The pump's rated capacity was used to set the flow rate rather than the flow rate being independently controlled. No measurement-uncertainty analysis (instrument tolerances, repeatability) is provided for the readings of temperature, voltage or current.

In future work, the following should be considered: Replacement of the air-cooled hot-side heat sink by a water-cooled hot-side heat exchanger, which is consistently reported in

literature ([6], [8], [12]) as a key to higher COP; improving the insulation of the reservoir to reduce the contribution of the heat leakage; operating the fan at variable speed (PWM control) based on temperature feedback of the hot-side heat exchanger; performing a systematic sweep of voltage, flow rate, and fan speed with COP evaluated at each point – leading to a true optimization of the operating point; introducing automated, continuously measured temperature/power monitoring (e.g., with a microcontroller), which would make Table III reconstructable as numeric data – thus avoiding the manual interval readings; repeated trials with formal statistical analysis (ANOVA or regression) where sufficient repeated measurements have been made; and, at the system-architecture level, testing the operation of the system with the solar system and battery system, and introducing the option of remote monitoring using IoT – introducing multi-objective optimization (Design-of-Experiments or Response-Surface) of voltage, flow rate, and fan speed at the same time. None of these extensions have been tested in the experimental setting in the present study and are mentioned here only as future work.

VIII. Conclusion

A prototype thermoelectric water cooling system composed of two modules (TEC1-12706) has been designed and tested, focusing on the hot-side thermal management and operating-point selection issues. Two Peltier modules are connected in parallel with each being rated at 12 V/6 A, independently heat-sunk and fan-cooled on the hot side and cooled the water from 25 °C to 15 °C in 26 minutes at ambient temperature of 30 °C. The hot-side thermal margin analysis revealed that the heat-sink/fan assembly was able to reject approximately 96.9 W per module, which matched a traceable 200 W assembly rating and resulted in a thermal margin of 51.6% which is consistent with the 38-45 °C operating ΔT observed and with stable operation during the test. One of the key results of this work is that the system's headline COP is substantially different if cooling capacity is considered as an idealized/instantaneous quantity or if it is

recalculated directly from measured mass, specific heat, temperature drop and elapsed time, and then taking an average over time. The former is as reported in the source book, corrected in the present paper to 0.329 because of a total-power discrepancy; the latter is directly measured and recalculated in this paper from first principles and is found to be approximately 0.178. Since only the best number is reported, as is usual, it may be possible to make false assumptions about what performance can be expected for downstream engineering.

A discrepancy of about 124% between the theoretical cooling time of 11.6 minutes, which assumes idealized constant net cooling power, and the measured cooling time of 26 minutes was noted, which could be due to heat infiltration, non-steady cooling power, or to a temperature rise on the hot side, but not to the measurement error. Total electrical power use was 150.6 W and total fabrication cost was PKR 11,250 (approx. USD 40) reflecting the practical case of the system as a low cost, maintenance free and silent refrigerant free alternative to a point of use cooling system where small size and silent operation is more important than the higher raw capacity of the vapor-compression systems.

The main engineering finding of the work is not a single optimized design, but a clear operating envelope that is sufficiently sized and has independent hot-side heat sinks that are required for the stable operation of the system, and that a sufficiently high time-averaged COP does not necessarily require a large sizing margin, and that reported performance of small thermoelectric water coolers needs to clearly separate out instantaneous and time-averaged cooling capacity. Future work is identified as the water-cooled hot-side heat exchange, PWM control for adaptive fan control, systematic voltage/current sweeps, and automated data logging between the idealized and time-averaged performance figures.

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