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# Nanofluid Augmented Coolant Rail Thermoelectric Cooling of Electronic Systems—Modeling and Analysis

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*Abstract*—**Modern electronic systems have reached a dimensional complexity and power density which presents numerous cooling challenges. A robust thermal management system is needed to provide the required heat transfer, especially while operating in harsh conditions such as elevated ambient temperatures. While liquid cooling combined with phase change materials has been shown to provide significant improvements in cooling performance, it is desired to develop a system capable of cooling electronic devices without endurance restrictions. By combining a nanoparticle enhanced coolant rail with thermoelectric coolers, reliable heat transfer can be provided for the electronic equipment. Additionally, a remote multiple loop cooling system maintains cooling efficiency while providing packaging flexibility applicable to modern computational systems with space constraints. A mathematical system model has been created and numerically simulated to evaluate different cooling configurations to demonstrate the overall effectiveness while operating in elevated ambient temperatures. The results show that the proposed system maintains low computer chip temperatures while compartmentalizing the heat transfer process essentially away from the thermal load.**

## I. Introduction

The current microprocessor based electronic systems reflect a continued trend toward smaller and more powerful electronic packages. The small scale designs provide superb packaging flexibility, but the power requirements for this hardware have resulted in an increasingly difficult cooling problem. While adequate cooling system performance has been provided for standard operations, military electronics may be exposed to elevated ambient conditions which render conventional cooling strategies inadequate and a limiting factor to the continuous operation of the equipment. Such harsh applications require a reliable and efficient thermal management system with a minimal heat signature and packaging flexibility [1].

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Advanced automotive thermal management technology has been developed using electric cooling system components for improved engine temperature control and reliable cooling of vehicle payloads. Liquid cooling has been demonstrated in industry to be capable of cooling high performance computational equipment. The electric cooling components may be digitally controlled as needed to maintain steady temperatures while providing maximum effectiveness and power efficiency [2]. Heat exchanger designs recently developed for electronic systems can provide very high heat transfer coefficients capable of dissipating the heat loads imposed by new electronic systems [3]. One of these devices, the micro-channel heat exchanger, was mathematically modeled by Tan *et al.* [4]. Liquid cooling in itself, however, is limited in effectiveness by the temperature of the cold reservoir to which the heat is rejected.

One system which has demonstrated good performance in cooling electronics is the Peltier effect thermoelectric cooler (TEC). TEC devices provide a temperature differential below the ambient conditions without need for any moving parts or vapor compression cycle. Peltier devices use current flow across the junction of dissimilar conductive materials to provide a heat flux. Importantly, TEC's may provide sufficient cooling capabilities to increase the power densities of computer chips [5]. A properly sized TEC may even provide superior cooling to multiple computer chips as shown by Simons *et al.* [6]. In spite of these technical advantages, TEC's require an effective method to reject the heat transferred across the junction. Insufficient heat rejection will cause the heat flux to be overcome by the material's thermal conductivity.

The relatively low efficiency of TEC's presents a problem in both the power requirements and the heat rejection of the hot side. Specifically, a high capacity cold reservoir is required. When space is not an issue, passive technologies such as free convection skive (very thin) fin heat exchangers have been shown to provide adequate heat transfer capability [7]. A more conventional installation would use forced convection as studied by Chang *et al.* [8] which provides good performance at typical electronic heat loads but requires proper optimization in the context of the overall system. Due to the limitations of air cooled TEC's, combining advanced liquid cooling and TEC devices can maintain the strength of both technologies by providing cooling below ambient conditions while offering sufficient heat transfer to reject heat from the TEC module in space-limited applications. Such systems have been modeled to demonstrate their effectiveness and usefulness to optimize power consumption [9]. Additionally, air cooling may not be capable of satisfying the heat rejection requirements of new computational devices while liquid cooling has the opportunity to satisfy these demands.

Liquid cooling has been shown to be improved by the addition of highly conductive nanoparticles which enhance the thermal conductivity of the base fluid to allow lower flow rates and temperature differentials [10-14]. In the past, fluids were augmented with micrometer sized particles. These fluids were effective in enhancing the thermal conductivity, but at the cost of increased pressure losses and pipe wear. As observed by Godson *et al.* [11] and Choi [12], nanofluids have enhanced the heat transfer coefficients of a base fluid by 30% or more, with much lower pressure drops and pipe wear. These considerations when combined make nanofluids a very attractive option for solving the problem of heat removal in electronic systems. However, the mechanisms of this enhancement are still under debate due to the inconsistencies present in the available experimental data [10, 12]. Therefore, precise mathematical models of the thermally enhanced properties of nanofluids remain a research topic both analytically and experimentally.

Further expansion of this combined thermal management system may provide enhanced electronics packaging flexibility by moving the TEC device away from the heat source and using an inner liquid cooling loop to convect heat to the TEC module. The use of nanoparticle coolants and specialized heat exchangers renders this arrangement much more effective than traditional liquid heat exchangers. The remainder of the paper has been organized as follows. Section II explains the difficulties of cooling electronic equipment in high temperature environments. A thermal system model is introduced in Section III for an electronic system featuring computer chips and cooling components in various configurations. Representative numerical results will be provided and discussed in Section IV with the Conclusion presented in Section V. The summary is contained in Section VI. A complete Nomenclature List has been offered in the Appendix.

## II. Cooling Challenges

Demands for increasingly smaller electronic devices capable of similar, or greater, computational power than larger systems has produced a unique packaging challenge in which the cooling system has begun to dictate package size. Effective cooling, even with currently large heat sinks, may be inadequate to meet the thermal management demands of these small devices. In some instances, contact resistances must be reduced to the bare minimum. The cooling problem becomes even greater in elevated ambient conditions such as those encountered by military operations in desert climates. The convective and even liquid cooling systems may be unable to cope with the low temperature differential between the failure temperature of the equipment and the ambient temperatures. The implementation of thermoelectric cooling devices provides a unique opportunity in this area, but a remote cooling strategy which transfers heat to the ambient air would be even better due to the local packaging flexibility provided.

Peltier effect thermoelectric devices represent a cooling technology capable of driving the component temperatures below ambient conditions without the complication of sensitive vapor compression mechanical plumbing. This method of heat transfer may be produced by flowing electric current through the junction of two dissimilar materials. TEC units are relatively compact and have already been used in portable food coolers and electronic systems [15]. A key advantage of these small scale solid state solutions is that their external parts are limited to a fan heat sink or small liquid heat exchanger to provide hot-side heat rejection.

Current thermoelectrics operate at approximately 10% efficiency in most applications. While this efficiency does not significantly impact the proposed cooling application, it does demand an effective heat rejection system. Ideally, a liquid heat exchanger such as the coolant rail concept would be implemented. A typical TEC installation for cooling multiple computer central processing units (CPU) is shown in Fig. 1. This liquid cooling system may operate with, or without, the TEC's depending on the heat rejection demands. While this installation is effective, it does not fully address the footprint of most electronic cooling systems due to the amount of stacked equipment.

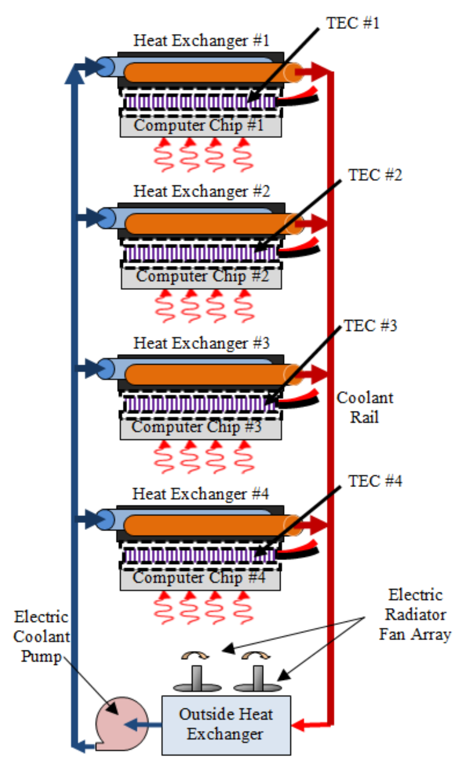


Fig. 1. Central processing unit cooler for multiple units in a single loop with liquid heat exchanger and optional TEC's.

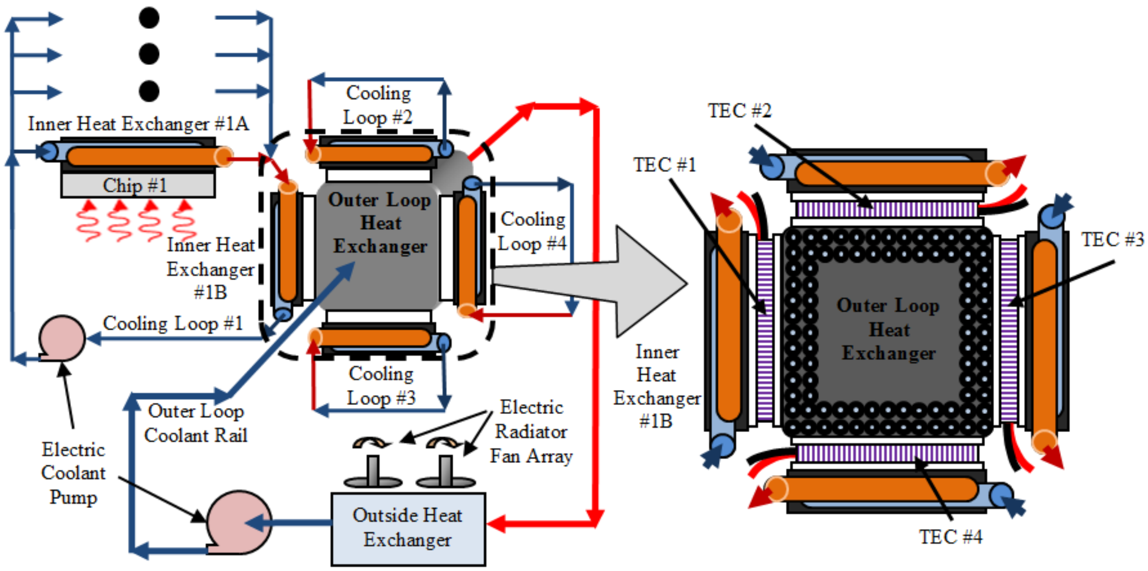


Fig. 2. Proposed remote cooling system arrangement with four TEC-cooled (multiple) loops served by a single heat exchanger.

The cooling system size constraint is thus addressed by implementing a separate cooling loop with moves the TEC device and hardware to another location. In this manner, the local footprint is minimized while offering the required cooling needs. As shown in Fig. 2, such a system can serve multiple processors or even whole separate devices. The increased heat transfer performance for nanoparticle coolants has prompted their integration into advanced cooling systems. Further, the enhancement of the thermal properties of the base fluid are temperature dependent. Wong and Kurma [13] have shown that the fluid's thermal conductivity is linearly increased with a rise in temperature while the viscosity is exponentially reduced. This is especially attractive since the fluid's heat transfer characteristics are further improved as the temperature increases, providing even more critical cooling. This technology is also particularly attractive for the remote cooling system concept as it helps to reduce component size. A smaller heat exchanger size is required due to the significant increase in convection coefficients.

## III. Mathematical Models

The thermal behavior of the electronic devices and cooling system will be mathematically described. The model represents the governing system dynamics and accommodates ambient conditions representative of the proposed system installations. A lumped capacitance approach will be used as developed in previous work [16]. In accordance with this methodology, the following three assumptions have been imposed:

A1. Convection is the primary means of heat transfer for the heat sinks.

A2. Heat conduction occurs through the material interfaces and may be considered one-dimensional.

A3. Radiation effects are negligible due to low temperature differentials.

### *A. Nanofluid Enhancment Model*

A subsystem model was developed to account for the heat transfer increase due to nanofluid enhancement. Since modelling techniques remain an unresolved issue to date, experimental data presented in the literature [11-13] was used to create an approximation. For a micro-channel heat exchanger, nanofluids may provide a 30% increase in heat transfer at room temperature. However, as reported in Wong and Kurma [13], the enhancement of the liquid coolant's viscosity decays exponentially as the temperature rises. In contrast, the nanofluidic thermal conductivity rises approximately linearly with temperature. Therefore, the heat transfer coefficient in the heat sinks were approximated as

$$h_{cool} = \begin{Bmatrix} 1.3h_0(1+\beta T_{cool})e^{\frac{T_{cool}}{\tau}} & ; & Nanofluid \\ h_0(1+\beta T_{cool}) & ; & Fluid\ Only \end{Bmatrix} \quad (1)$$

where $h_0$ is the coolant-only heat transfer coefficient at room temperature, $T_{cool}$ is the coolant temperature in °C, $\beta$ is the viscosity factor, and $\tau$ is the thermal conductivity factor of the nanofluid. It is important to note that in the temperature regimes considered, the convection coefficient rises exponentially with respect to the temperature.

The convection coefficient, $h_{cool}$, has been incorporated in the conventional manner as a thermal resistance, $R_{cool,i}$, on each heat exchanger such that

$$R_{cool,i} = \frac{1}{A_{cool,i}h_{cool,i}(T_{cool,i})} \quad (2)$$

where $A_{cool,i}$ is the convective surface area of the heat exchanger, and $h_{cool,i}(T_{cool,i})$ is the convection coefficient as a function of coolant temperature, $T_{cool,i}$.

### B. Thermoelectric Cooling Devices

Heat transfer through thermoelectric systems operating on the Peltier effect is governed by current flow through the module. The current, $i$, is dependent upon the unit's electrical resistance, $R_{TEC}$, supplied voltage, $V$, and the temperature difference between the two plates, $T_{hot}$ and $T_{cold}$ [17]. The algebraic relationship for the current, $i$, in the device may be expressed as

$$i = \frac{1}{R_{TEC}} V - \alpha (T_{hot} - T_{cold}) \tag{3}$$

where the parameter $\alpha$ denotes the Seibec coefficient of the thermoelectric material (note that $\alpha$ is negative). The electric power, $P$, consumed by the unit may be stated as

$$P = i^2 R_{TEC} - \alpha i (T_{hot} - T_{cold}) \tag{4}$$

The amount of heat removed from the cold side of the TEC, $\dot{Q}_{cold}$, is related to applied power so that [18]

$$\dot{Q}_{cold} = -2n(\alpha i T_{avg}) - \frac{1}{2} i^2 R_{TEC} - A_{TEC} k_{TEC} (T_{hot} - T_{cold}) \tag{5}$$

where $T_{avg} = \frac{1}{2}(T_{hot} + T_{cold})$ is in K, and n is the number of thermoelectric couples in the TEC. Ohm's law allows the TEC's behaviour to be modelled using a supply voltage so that the heat removal in equation (5) may be rewritten as

$$\dot{Q}_{cold} = 2n(\alpha T_{avg}) \frac{V}{R_{TEC}} - \frac{1}{2} \frac{V^2}{R_{TEC}} - A_{TEC} k_{TEC} (T_{hot} - T_{cold}) \tag{6}$$

Thus, the hot side has heat added which may be described as

$$\dot{Q}_{hot} = 2n(\alpha T_{avg}) \frac{V^2}{R_{TEC}} + \frac{1}{2} \frac{V^2}{R_{TEC}} + A_{TEC} k_{TEC} (T_{hot} - T_{cold}) \tag{7}$$

During system operation, the TEC is only powered when its associated CPU experiences an elevated temperature according to the digital thermostat control form

$$V_i = \begin{cases} V & ; \quad T_C \geq T_{inner,i} + T_{high} \\ V \frac{T_C - T_{inner,i} - T_{low}}{T_{high} - T_{low}} & ; \quad T_{inner,i} + T_{low} < T_C < T_{inner,i} + T_{high} \\ 0 & ; \quad T_C \leq T_{inner,i} + T_{low} \end{cases} \tag{8}$$

### C. System Thermal Model Formulation

The thermal behavior of each primary component in the electronic system may be described by ordinary differential equations which represent a set of temperature nodes. In a lumped capacitance thermal model, the heat transfer process is dependent upon the thermal resistances, $R_i$, and the heat capacitances, $C_j$ within the system. The differential equation for the temperature of the $j^{th}$ thermal node, $T_j$, may be written as

$$C_j \dot{T}_j = \sum_i \frac{1}{R_i} (T_i - T_j) + \dot{Q}_j, \; (j = 1,2,\ldots,n) \tag{9}$$

where $\dot{Q}_j$ is the corresponding heat load. The subscript n refers to the number of temperature nodes. In this study, the thermal resistances, $R_i$, are developed from standard convection and conduction expressions which, with the exception of the convective coolant interactions as given in equation (2), are constant with respect to temperature. As an example, the CPU and two TEC nodes may be formulated as follows

$$\begin{aligned} C_{C,i} \dot{T}_{C,i} &= \frac{1}{R_{C,i}} (T_{cold,i} - T_{C,i}) + \dot{Q}_C \\ C_{cold,i} \dot{T}_{cold,i} &= \frac{1}{R_{C,i}} (T_{C,i} - T_{cold,i}) - \dot{Q}_{cold,i} \\ C_{hot,i} \dot{T}_{hot,i} &= \frac{1}{R_{cool,i}} (T_{inner,i} - T_{hot,i}) + \dot{Q}_{hot,i} \end{aligned} \tag{10}$$

where the subscript $C$ denotes the $i^{th}$ CPU node and $T_{inner}$ is the average temperature of the associated coolant loop. Note that the thermal resistance of the TEC/heat exchanger interface is neglected since it was significantly smaller than the primary resistances in the system.

The standard formulations for the convective and conductive thermal resistances have been implemented and the nodes compiled into a suite of nodal equations. The state-space formulation has the form $\dot{\vec{x}} = A\vec{x} + B\vec{u}$ with the state and input vectors, $\vec{x} \in \Re^{17\times1}$ and $\vec{u} \in \Re^{7\times1}$, respectively.

## IV. Numerical Results

A thermal management system to accommodate the electronic heat load of four CPU's was considered. Specifically, a variable cooling demand was studied due to individual units coming "online" and "offline" at staggered intervals throughout the operational time interval. In other words, step changes in $\dot{Q}_{C1-4}$ were considered. The variable load profile allowed the investigation of system transients for performance comparisons of the proposed cooling designs. The model parameters have been listed in Table 1 with their corresponding units. These values are based on the representative material properties, system geometry, operating environment, and the thermal behaviors typical of the system. A Matlab/Simulink algorithm was created with an integration time step of $\Delta t$ = 0.10 s and executed for a 24 hour time period to demonstrate the effectiveness of the proposed system.

TABLE I
Summary of Model Parameters

| Symbol | Value | Units | Symbol | Value | Units |
|---|---|---|---|---|---|
| $A_{cool,HE}$ | 0.5 | $m^2$ | $T_0$ | 25 | °C |
| $A_{cool,in}$ | 0.1 | $m^2$ | $T_{low}$ | 0.5 | °C |
| $A_{cool,out}$ | 0.3 | $m^2$ | $T_{high}$ | 1.5 | °C |
| $A_{TEC}$ | 0.0016 | $m^2$ | $T_{var}$ | 12 | °C |
| $h_0$ | 1,000 | $W/m^2K$ | $T_{\infty,avg}$ | 40 | °C |
| $h_{HE}$ | 100 | $W/m^2K$ | $V$ | 12 | V |
| $k_{TEC}$ | 1.0 | $W/m^2K$ | $\alpha$ | -287 | μV/K |
| $n$ | 127 | - | $\beta$ | 0.0016 | °C |
| $\dot{Q}_C$ | 100 | W | $\Delta t$ | 0.10 | s |
| $R_{TEC}$ | 2.0 | Ω | $\tau$ | 2,800 | °C |

A cyclic ambient temperature profile, $T_\infty$, representative of desert conditions was implemented. The computer operating profile was set so that maximum heat load coincided with the day's highest ambient temperature condition as a check of the system's effectiveness. This thermal loading has been described in Fig. 3. Each computer chip was assumed to run continuously at full load during its operation period.

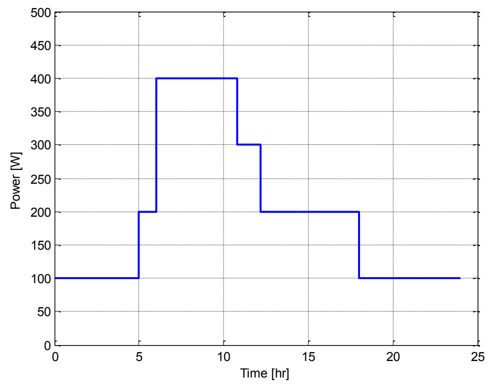


Fig. 3. Heat load for a 24 hour period in desert conditions – solar and computer cycling (t= 05:00, 06:00, 11:00, 12:00, 18:00 hours) contributions with resultant total heat generated.

Two different liquid cooling arrangements were considered—single-loop direct and multi-loop cooling systems For the latter case, a common central heat exchanger was served by a convective radiator. Five operating scenarios were investigated to validate the operational concept as summarized in Table II: Test 1 with a conventional single loop liquid cooling system (refer to Fig. 1) to provide base data; Test 2 with a single-loop nanoparticle cooling system to validate the cooling enhancement; Test 3 featuring a single-loop thermoelectric cooling system; Test 4 introducing a multiple loop cooling to remove heat from four electronic devices; and Test 5 which offers a multi-loop cooling system with thermoelectrics. The TEC controller was only used in Test 5 since the elevated coolant temperatures expected in Test 3 would have negatively impacted the overall cooling performance. The control strategy in equation (8) was replaced by directly linking the TEC control status to the CPU operation mode. The same computer operating profile was implemented in all tests. Parameters consistent with an outer loop coolant flow rate of $\dot{m}$ = 3 kg/s were applied with forced convection on the liquid-to-air heat exchanger, and the TEC's were assumed to each be supplied with 72 W of electrical power.

Comparing Tests 1 and 2 in Table III, the peak temperature reduction of 0.4°C demonstrates a slight improvement due to the nanoparticle cooling. However, this temperature value is disappointingly small given the operating temperature. The Test 2 results presented in Fig. 4 represent safe values for the given design operating regime but display temperatures in the neighborhood of the typical maximum operating temperature of 85°C, (e.g., a 14.4°C buffer exists).

TABLE II
DESCRIPTION OF OPERATING SCENARIOS TO EVALUATE PROPOSED COOLING SYSTEM DESIGNS.

| Test No. | TEC's Installed | Single Loop | Multiple Loop | Nanofluid |
|---|---|---|---|---|
| 1 | | x | | |
| 2 | | x | | x |
| 3 | x | x | | x |
| 4 | | | x | x |
| 5 | x | | x | x |

As shown in Fig. 5, the single loop thermoelectric system (Test 3) was observed to reduce the peak computer temperature by 17.3°C (e.g., 53.3°C versus 71. °C for Tests 3 and 1, respectively). The peak coolant temperature remains essentially the same, $T_{cool,out}$ = 68.3°C, due to the total heat load not changing significantly and due to the effective liquid-to-air heat exchanger parameters. Fig. 6 shows the temperatures of all four computers which demonstrate similarity of behavior among the elements. The much higher temperatures shown by the individual computer chips in Fig. 6 are due to the hot coolant actually heating up the computers even though they are shut down ($T_{C3,4} > 55$°C). This behavior can be eliminated by further restricting the flow of coolant to the units when they are not in operation. As expected, since the TEC's cool their respective CPU's below the coolant temperature during operation, the use of the thermostat controller in equation (8) would preclude the realization of the system's maximum performance.

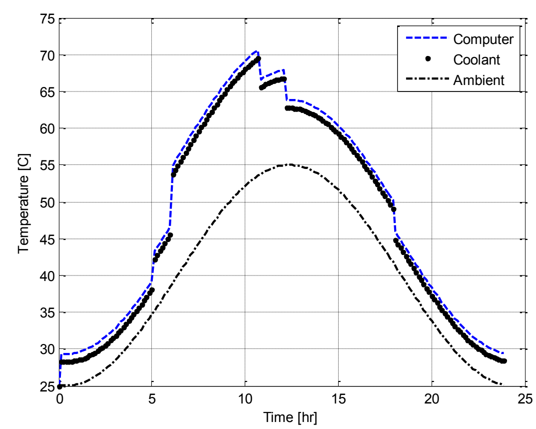


Fig. 4. Test 2 - Temperature response for the electronic equipment and coolant during a twenty-four hour period with single loop liquid cooling only.

A fourth test (Test 4) considered the multi-loop cooling system with nanofluids to establish a base test for the fifth and final test. Test 5 involved the complete implementation of multi-loop loop thermoelectric cooling system. The results for Test 5 demonstrate a very effective cooling system which showed a peak temperature only 2.2°C higher than Test 3 despite the system's increased thermal resistance. The results are thus very promising and maintain the CPU temperatures at a safe level. Additionally, during peak ambient conditions with less than three computers on-line, the operating CPU's remain below the ambient temperature, $T_\infty$, with temperature differences of up to 1.2°C at $t$ = 13.2 hrs.

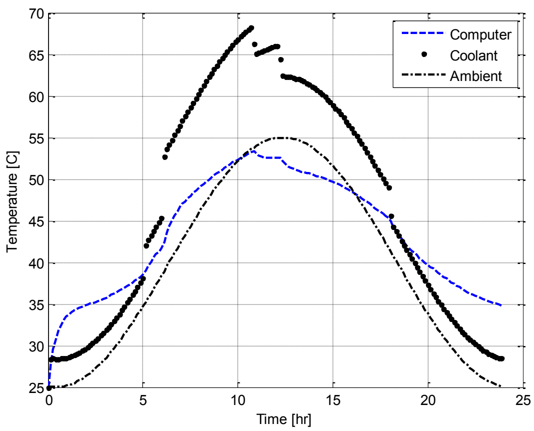


Fig. 5. Test 3 - Temperature response for the electronic equipment and coolant during a twenty-four hour period with operational TEC's and single loop liquid cooling.

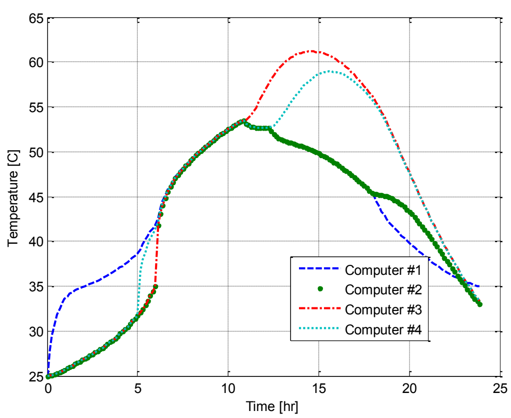


Fig. 6. Test 3 - Individual computer thermal responses for the 24 hour operating period with single loop cooling.

## V. Conclusions

To summarize the results, each test has been compared in Table 3 including the maximum temperatures for the coolant and CPU's, the maximum power consumption, and total power usage. These results clearly illustrate the cooling improvements available from the thermoelectric devices. For instance, a 15.1°C reduction in peak temperature may be observed for multiple-loop TEC cooling system (Test 5) when compared to liquid cooling alone, demonstrating the effectiveness and viability of this new technique for remote, modular cooling of electronic systems. However, the power consumption increased from 202 W-hr to 2102 W-hr with the TEC devices and multiple-loop cooling, and the peak current increased from 0.7 A to 27.4 A at 12 VDC. This increase in cooling system power consumption is acceptable for the intended applications due to the improvements in both CPU operating temperature and packaging flexibility. Additionally, the system uses considerably less power than the CPU's themselves, which consume a total of 4400 W-hr during the operating period. This figure also does not include the total power usage of an overall computer system, which could be several times larger depending on the type of computational equipment being considered.

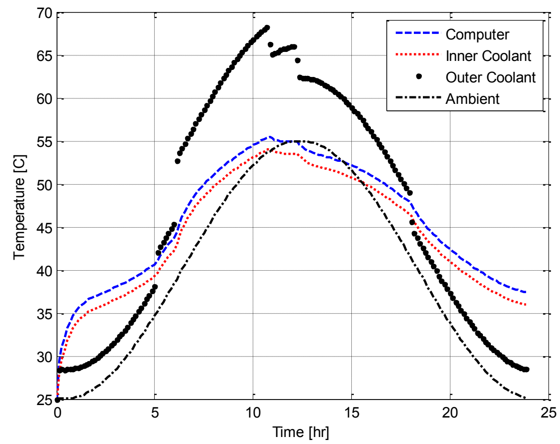


Fig. 7. Test 5 - Temperature response for the electronic equipment and coolant during a twenty-four hour period with operational TEC's and double loop liquid cooling with four separate cooling loops for the computers.

## VI. Summary

The growing power density of electronic devices presents a significant cooling challenge in electronic system design. Two liquid cooling system configurations, applicable to a thermoelectric cooler array with multiple cooling loops for the remote cooling of electronic components, have been presented in the paper. The system designs have been modeled and compared to demonstrate a high standard of heat rejection performance while achieving packaging flexibility. Future work should include the expanded testing of a remote cooling system system and analysis of various control strategies for precise temperature control and power usage minimization.

TABLE III
Summary of Numerical Results

| Test No. | Maximum Temperatures (°C) $T_C$ | $T_{cool,in}$ | $T_{cool,out}$ | Max. Power Consumption (W) | Total Power Usage (W-hr) |
|---|---|---|---|---|---|
| 1 | 71.0 | - | 69.5 | 8.4 | 202 |
| 2 | 70.6 | - | 69.5 | 8.4 | 202 |
| 3 | 53.3 | - | 68.3 | 296.4 | 1,912 |
| 4 | 72.5 | 71.1 | 69.3 | 40.4 | 392 |
| 5 | 55.5 | 54.1 | 68.3 | 328.4 | 2,102 |

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## APPENDIX: NOMENCLATURE LIST

| | |
|---|---|
| $A$ | area, [m$^2$], state space variable matrix |
| $B$ | state space input matrix |
| $C$ | heat capacity, [J/K] |
| $c_p$ | specific heat, [J/kg K] |
| $h$ | convective heat transfer coefficient, [W/m$^2$ K] |
| $k$ | conductive heat transfer coefficient, [W/m K] |
| $m$ | mass, [kg] |
| $\dot{m}$ | mass flow rate, [kg/s] |
| $n$ | number of temperature nodes |
| $P_{cool}$ | cooling system power [W] |
| $\dot{Q}$ | heat load electronics, [W] |
| $R$ | thermal resistance, [K/W] |
| $T$ | temperature, [ºC] |
| $\bar{u}$ | state input vector |
| $\bar{x}$ | state variable vector |
| $\Delta t$ | time step, [s] |
| $\alpha$ | Seebeck coefficient, [μV/K] |
| $\beta$ | viscosity factor, [ºC] |
| $\rho$ | density, [kg/m$^3$] |
| $\tau$ | thermal conductivity factor, [ºC] |

*Subscripts*

| | |
|---|---|
| *C* | computer chip |
| *cold* | TEC cold side |
| *cond* | conduction |
| *conv* | convection |
| *cool* | coolant |
| *CP* | computer peak |
| *CV* | computer convection |
| *HE* | coolant radiator |
| *high* | upper thermostat bound |
| *hot* | TEC hot side |
| *inner* | inner loop |
| *low* | lower thermostat bound |
| *outer* | outer loop |
| $\infty$ | ambient environment |
| *0* | base value |