

Control of a Thermoelectric Cooling System for Vehicle Components and Payloads - Theory and Test

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ABSTRACT

Hybrid vehicle embedded systems and payloads require progressively more accurate and versatile thermal control mechanisms and strategies capable of withstanding harsh environments and increasing power density. The division of the cargo and passenger compartments into convective thermal zones which are independently managed can lead to a manageable temperature control problem. This study investigates the performance of a Peltier-effect thermoelectric zone cooling system to regulate the temperature of target objects (e.g., electronic controllers, auxiliary computer equipment, etc) within ground vehicles. Multiple thermoelectric cooling modules (TEC) are integrated with convective cooling fans to provide chilled air for convective heat transfer from a robust, compact, and solid state device. A series of control strategies have been designed and evaluated to track a prescribed time-varying temperature profile while minimizing power consumption. The zone cooling device was tested within an enclosed cabin to provide localized convective zone cooling to a representative heat producing component. Mathematical models for the TEC's and overall system were developed and validated using these test results. The experimental results demonstrated that an amplitude modulated output from the tracking controller reduced power consumption by more than 78% as compared to a standard thermostat approach while reducing tracking error 19% during an 80 minute test. The findings of this power minimization study indicate excellent power savings opportunities for thermoelectric cooling devices, but also reveal a possible deficiency in current modeling techniques for thermoelectric devices.

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1. INTRODUCTION

The application of Peltier-effect thermoelectric devices continues to increase due to the solid state construction and their rising efficiency [1]. Much research has been devoted to the integration of these devices into computer and power electronics [2]. Using thermoelectric cooling devices (TEC) for electronics cooling requires careful attention to the heat rejection design [3]. Thermoelectrics are well-suited for refrigeration and low-capacity air conditioning since their solid-state design demonstrates a significant advantage over traditional vapor-compression refrigeration strategies [4]. Thermoelectric devices may be reversed, as governed by the Seebeck effect, enabling them to also be used for power generation (TEG). TEG's have been studied for potential use in waste heat recovery, but the technology still shows only modest returns due to the inefficiency of the devices and the weight of current heat exchanger designs [5]. Analyses on waste heat recovery using vehicle exhaust gas recirculation have shown the potential for a 3% improvement in fuel economy [6]. One recent feasibility study predicts up to a 6% improvement in fuel economy for passenger vehicles. [7]

With the development of these new applications for thermoelectric devices, the need arises for advanced control strategies. Above-ambient temperature control using TEC's was addressed as early 1983 and continues to be an important topic for electronic payload management [8,9]. Thin-film TEC's have been analyzed for processor cooling and discussed energy minimization methods leading to a 20% reduction in energy consumption [10]. Convection effects on thermoelectric devices have been studied under exposure to varying temperature fields and methods for limiting perturbation have been proposed [11]. An innovative pairing of thermoelectrics and heat pipes also analyzed power input reduction [12]. Varying set point tracking of TEC's was evaluated using linear feedback [13], while another study considered fixed set point tracking based on a modified classical controller with neural networks [14]. Thermoelectric device tracking of continuous and discontinuous temperature set points has been conducted using classical control to reveal the performance limits of the device [15].

In an innovative application of thermoelectric components, a thermoelectric heat exchanger capable of delivering localized cooling to an automotive child-safety seat was developed [16,17]. Based on this principle, it is proposed that thermoelectric systems may enable

controlled cooling of localized regions of a convection-dominated environment such as ground vehicle components and/or cabin similar to existing production technology for personal climate control in passenger automobiles [18]. This system has been developed as a directed-convection thermoelectric heat exchanger conforming to the characteristics of Figure 1. The device has been designed for portable, controllable thermal management with successful laboratory results. Mathematical models for the thermoelectric devices will be formulated so that a control law may be established to achieve precise temperature tracking. Using a predicted thermal saturation phenomenon observed with thermoelectrics, a control law will be designed to allow precise temperature tracking with a significant reduction in power consumption when compared experimentally to conventional control methods.

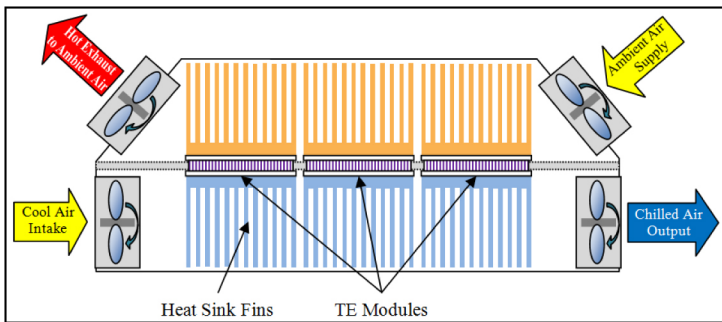


Fig. 1. Directed convective cooling system design with thermoelectric cooling modules (TEC) sandwiched between finned heat sinks subjected to forced convection cooling

The remainder of the paper is arranged as follows: Section 2 establishes a zone-cooling strategy which breaks down calm air environments into convective zones which may be used to control the temperature of objects of interest. The system model and control algorithms required to implement the zone cooling concept are developed in Section 3. Discussion of the numerical and experimental results is presented in Section 4. The research effort is summarized in the Conclusion followed by a full Nomenclature List in the Appendix.

2. CONCEPT OF ZONE COOLING

The zone cooling concept describes a convective air environment as a set of control volumes within an area of relatively calm air. The convective heat transfer between an object and surrounding air influences the local boundary layer so that this local air mass may be modeled as a temperature node. This approach allows modeling the overall convective system, resulting in a context for temperature regulation of the object of interest. To develop this concept toward a mathematical model, consider some object within a larger space as shown in Figure 2. Note that a pair of nested convective zones, described as control volumes CV1 and CV2, surrounds the object. The temperatures of these zones, T_{CV1} and T_{CV2} , are the states defined by the lumped-capacitance modeling method. In this approach, the temperature nodes should be assigned to key locations within the

thermal system. Convective heat transfer between the zones may be modeled as heat flow between the defined temperature nodes. Within the innermost zone, CV2, the object of interest is the temperature node T_{heat} which is subject to input heat, $\dot{Q}_{heat}$. This object's thermal interaction with T_{CV1} and T_{CV2} dictates that the control of T_{CV1} allows the cascaded control of T_{heat} with T_{∞} denoting the ambient temperature.

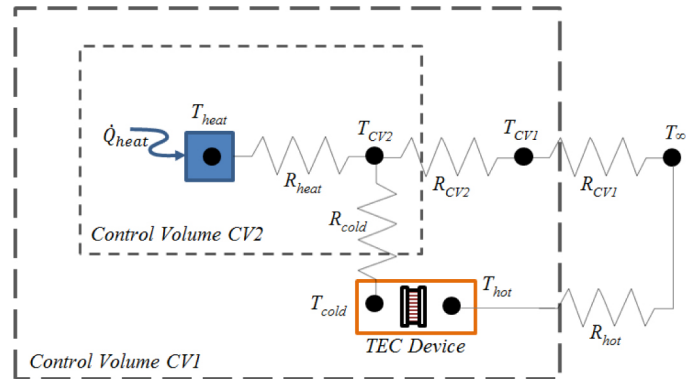


Fig. 2. Thermoelectric convective cooling system with two control volumes, CV1 and CV2, for temperature control of T_{heat}

Without loss of generality, the temperature of control volume CV2 may be considered to be provided by a cooling system influencing this control volume through its cooling output, $\dot{Q}_{removed}$, while receiving input heat, $\dot{Q}_{heat}$, due to external means. This system must be designed to reject heat to the ambient environment to achieve maximum performance. The cooling behavior described may be accomplished through air recirculation within control volume CV2 and ducting the removed heat to the surroundings, T_{∞} . Systems capable of producing this behavior include vapor compression systems (refrigerators and air conditioners) and thermoelectric devices. Power supplied to the system would regulate the control volume temperature, T_{CV2} , and thereby achieve temperature regulation of the object of interest, T_{heat} . In this study, the boundary of T_{CV2} will be specified by the wall of an enclosure, designated by the temperature node T_w .

3. SYSTEM MODEL AND CONTROLLER DESIGN

The zone cooling concept may be applied to a thermoelectric convective cooling system which directs a stream of cool air into a local convective environment, CV2, containing the object of interest whose temperature, T_{heat} , must be controlled to a given set point. The device may be ducted so as to recirculate the air from the local environment, CV2, back into the cold side of the TEC's, thus making system control easier due to the reduced air flow to and from the surroundings. Similarly, the heat rejected for the TEC hot side will be discharged to the ambient environment, T_{∞} , so that the temperature differential across the device can be maintained. The control architecture describing this methodology is shown in Figure 3.

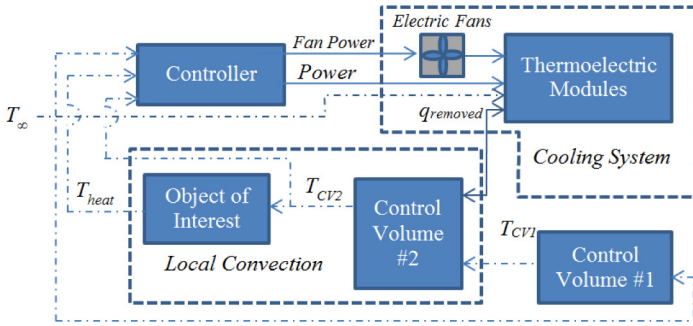


Fig. 3. Thermoelectric zone cooling control architecture and signal flow

3.1. Thermoelectric Behavior

Thermoelectric cooling systems utilize the Peltier effect which defines heat transfer to be dependent upon the current flow through the modules. The current flow, I , within the device is not only limited by electrical resistance, R_{elec} , but also upon the temperature differential between the hot, T_{hot} , and cold, T_{cold} , sides [16]. The current flow, I , may then be defined as

$$I = 1/R_{elec} (V - n\alpha(T_{hot} - T_{cold})) \quad (1)$$

The electric power expenditure by the unit follows the form $P = I^2 R_{elec}$ which may be expanded using equation (1) plus the fact $P = IV$ so that

$$P = n\alpha I(T_{hot} + T_{cold}) + I^2 R_{elec} \quad (2)$$

where n is the number of thermoelectric junctions, $n\alpha I(T_{hot} + T_{cold})$ represents the thermoelectric effect, and $I^2 R_{elec}$ represents electric power losses.

The heat absorption into the TEC cold side, $\dot{Q}_{cold}$, may be formed using the Seebeck effect, $n\alpha I(T_{hot} + T_{cold})$. The thermal conductivity of the device's junctions and electrical resistance must be considered so that the overall heat flow toward the hot side of the junction may be defined as

$$\dot{Q}_{cold} = 2n\alpha I T_{avg} - \frac{1}{2} I^2 R_{elec} - \frac{k_{tec}}{A_{tec}} (T_{hot} - T_{cold}) + f(T_{hot}, T_{cold}, I) \quad (3)$$

where $T_{avg} = 0.5(T_{hot} + T_{cold})$, k_{tec} is the thermal conductivity of the unit, A_{tec} is the conduction area of the unit, and hence $k_{tec}/A_{tec} (T_{hot} - T_{cold})$ represents thermal losses in the unit due to conduction. The expression $f(T_{hot}, T_{cold}, I)$ represents higher order thermal phenomena not captured by the conventional thermoelectric model. The first order thermoelectric phenomena used in this study are further developed in other publications [16, 19].

3.2. Thermal System Model

Localized cooling of a hot object within a larger environment may be effectively performed using thermoelectric devices. The thermal management problem may be modeled using a nodal network. This thermal network considers radiation heat transfer effects to be minimal since the nodes being considered remain within similar temperature ranges. Further, conductive heat transfer is only modeled within the thermoelectric device.

The thermal behavior of each component in the system may be described by a first-order differential equation; the resulting set of temperature nodes will compose the thermal network. In general, the dynamics of the j^{th} thermal node, T_j , is specified by heat capacitances, C_j and heat energy exchange specified by the temperature difference between T_j and T_i with the energy exchange limited by a thermal resistance $R_{i,j}$. The differential equation for the j^{th} node's temperature may be written as

$$C_j \dot{T}_j = \sum_{i=1, i \neq j}^l \frac{1}{R_{i,j}} (T_i - T_j) + \dot{Q}_j, (j = 1, 2, \dots, l) \quad (4)$$

where $\dot{Q}_j$ is the corresponding heat or cooling load. The variable j refers to the number of temperature nodes interacting with the j^{th} node. The thermal convective resistances may be expressed as

$$R_{i,j} = 1/h_{i,j} A_{i,j} \quad (5)$$

where $h_{i,j}$ is the convection coefficient, and $A_{i,j}$ is the convective surface area.

The thermal response of the local system is of specific interest within a modelling framework and demonstrates the lumped capacitance methodology. The dynamic temperature equations for the object of interest, T_{heat} , TEC cold and hot sides, T_{cold} and T_{hot} respectively, control volume CV1, T_{CV1} , and control volume CV2, T_{CV2} , are described as

$$C_{heat} \dot{T}_{heat} = \frac{1}{R_{heat}} (T_{CV2} - T_{heat}) + \dot{Q}_{heat} \quad (6a)$$

$$C_{CV2} \dot{T}_{CV2} = \frac{1}{R_{heat}} (T_{heat} - T_{CV2}) + \frac{1}{R_{CV2}} (T_{CV1} - T_{CV2}) + \frac{1}{R_{cold}} (T_{cold} - T_{CV2}) \quad (6b)$$

$$C_{cold} \dot{T}_{cold} = \frac{1}{R_{cold}} (T_{CV2} - T_{cold}) + \underbrace{\left[\frac{A_{TEC}}{k_{TEC}} (T_{hot} - T_{cold}) + \frac{1}{2} I^2 R_{elec} - n\alpha I (T_{hot} + T_{cold}) \right]}_{\dot{Q}_{cold}} \quad (6c)$$