Temperature Control Strategies in a Rectified Continuous Flow Loop for the Thermal Management of Large Structures

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ABSTRACT

Distributed loads are frequently encountered in large deployable structures used in space applications, such as optical mirrors, actively cooled sunshades, and focal plane electronics, or in zero boil-off cryogenic storage systems. An innovative mechanism for providing distributed cooling is via an oscillatory cryocooler such as a pulse-tube that is integrated with a fluid rectification system consisting of check-valves and buffer volumes in order to extract a small amount of continuous flow of cold gas. This continuous flow allows relatively large loads to be accepted over a long distance with a small temperature difference and has advantages relative to vibration and electrical isolation. Also, it is possible to provide rapid and precise temperature control via modulation of the flow rate. The same working fluid, for example helium, can be used throughout the entire system, reducing complexity and simplifying the contamination control process.

This paper describes a theoretical investigation of the ability of the rectifying interface to precisely control the temperature of a distributed load under dynamically changing conditions. The precise temperature regulation is enabled using temperature feedback control of a throttle valve placed in the loop. Flow modulation using the throttle valve is governed by a Proportional-Integral (PI) controller with gains that are selected to meet design temperature control criteria; specifically, a maximum temperature fluctuation and settling time. A linear thermal model based on experimental data is used to develop the control algorithm. The model demonstrates the ability to regulate the distributed load temperature under rapid load changes.

INTRODUCTION

Future space applications will require increasingly sophisticated cryogenic thermal control technology. No other cryogenic refrigeration system offers the same potential for low vibration, reliability, and efficiency as the pulse tube. However, regenerative coolers typically have small cold heads that must either be conductively coupled to heat loads or fluid dynamically linked using a warm or cold circulator. Thermal integration via conductive coupling is not ideal for

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distributed loads such as those associated by large, deployable structures and a circulator represents a substantial increase in complexity, mass and parasitic thermal load.

An alternative technique for thermally integrating a pulse-tube (or any regenerative cryocooler) with a distributed load using a rectifying interface is the Pulse Tube Rectifying Interface (PTRI) described in [1]. The PTRI utilizes a system of check-valves and buffer volumes to convert the oscillating pressure within the cryocooler to a quasi-steady pressure difference between the two buffer volumes. This pressure difference is used to provide a small, steady flow of cold gas that is capable of transporting the refrigeration capacity much more efficiently than a conductive strap and without the complexity of a cold circulator. An additional benefit of the PTRI concept is that the cryogen flow rate in the cooling loop can be rapidly modulated to provide thermal management of abruptly changing interface loads; the exploitation of thermal control aspect of the system is the focus of this paper.

Control algorithms have been developed to regulate the interface temperature using temperature feedback to appropriately adjust the cooling loop mass flow rate. The cold head temperature must also be controlled in order to maintain desired system temperatures. However, the cold head has a large thermal mass and subsequently changes temperature on a much longer time scale than the interface. The focus of this paper is therefore the control of only the interface temperature, and the cold head temperature is assumed to be constant over the time scales of interest.

A thermal model of the interface is described in [1] and is used in this paper to evaluate the control algorithms. The analysis predicts the interface temperature change (ΔT_I) to a rapid and stochastic change in the interface load. The controller gains are adjusted to achieve the desired temperature regulation with respect to the maximum allowable temperature fluctuation and settling time. For a step change in interface load, a non-dimensional maximum temperature rise $\left(\frac{\Delta T_I \ C_I}{\Delta \dot{q}_I \ t_s}\right)_{max}$ is defined and shown to be only a function of the damping ratio; this is convenient as it shows that for a given damping ratio, the thermal response is characterized by the interface thermal mass (C_I), interface load change ($\Delta \dot{q}_I$), and settling time (t_s). The result is useful to identify the optimal control gains and characterize the system performance potential, given thermal and hardware limitations.

PULSE TUBE RECTIFIED INTERFACE THERMAL MODEL

Interface Transient Model

A transient thermal model that considers the time evolution of the uncontrolled system temperatures under conditions corresponding to changing heat load, interface resistance, compressor stroke, etc. has been developed in the SIMULINK environment [2] and experimentally verified [1]. The system is modeled as two interacting and lumped thermal masses (see Fig. 1); the cold end of the pulse tube (C_c) with refrigeration ($\dot{q}_{pt}$) and temperature (T_c), and the interface (C_I) with temperature (T_I). The load interface heat exchanger provides the coupling between the distributed (interface) load, $\dot{q}_I$ (e.g., from a detector or a structure), and the cooling loop. The fluid is assumed to enter the interface heat exchanger at the cold head temperature and exit at the interface temperature.

An energy balance on the interface (shown as dashed lines in Figure 1) is used to develop the governing differential equation for the interface.

$$\frac{dT_I}{dt} = \frac{\dot{q}_I + \dot{q}_p - \dot{m}(f, R_I)\, c_p (T_I - T_C)}{C_I} \tag{1}$$

where t is time, $\dot{q}_p$ is the parasitic load from heat leak and c_p is the constant pressure specific heat capacity of helium. The mass flow rate through the interface ($\dot{m}$) is a function of the interface resistance R_I (i.e., the valve position), the stroke of the compressor (f), and, to second order, the cold head temperature. The subsequent section focuses on the analysis of a control strategy that pulse tube refrigeration.

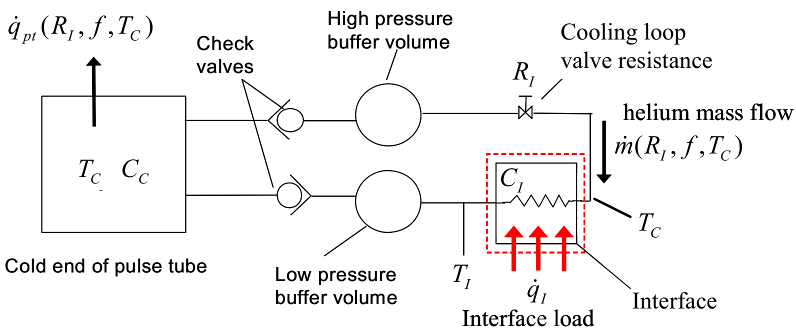

Figure 1. Schematic of transient thermal model highlighting the interface energy balance

is suitable for controlling the interface temperature in the face of small time scale disturbances in the interface load. It has been experimentally observed that adjusting the valve resistance provides a powerful and short time scale control authority for the interface temperature; therefore, the control gains will govern the adjustment of the valve resistance and the controlled parameter will be the interface temperature.

Linearized Interface Thermal Model

The equation governing the interface temperature Eq. (1) is non-linear; therefore, Eq. (1) is linearized in order to apply linear control theory to the problem. We have shown by comparison with a rigorous and non-linear model (Fig. 5) that the linearized model accurately predicts the response of interface temperature to changes in the valve resistance and the thermal load for conditions that are close to the nominal operating conditions (i.e., near the linearization point). Equation (1) is linearized using a Taylor series expansion about an experimental operating point that is summarized in Table 1. The parasitic heat load, compressor power, and cold head temperature as well as the thermal mass of the interface are all assumed to be constant for the purposes of the control model. The nominal operating condition is based on 80% compressor stroke in order to admit a control algorithm capable of both increasing and decreasing the mass flow in response to load perturbations. The mass flow rate is described as a linear function of the interface valve resistance:

$$\dot{m}_{80} = g_{80} R_I + h_{80} \tag{2}$$

where $\dot{m}_{80}$ is the mass flow at 80% compressor stroke and g_{80} and h_{80} are experimentally determined constants. The governing equation models the interface temperature as varying only with interface valve resistance and interface load.

$$\frac{dT_I}{dt} = \frac{\dot{q}_I + \dot{q}_p - (g_{80} R_I + h_{80}) c_p (T_I - T_C)}{C_I} \tag{3}$$

Table 1: Nominal operating values used to linearize the interface differential equation.

Parameter	Value	Parameter	Value
Initial interface temperature ($T_{I,0}$)	150 K	Mass flow constant (g_{80})	-3.94095e-5 kg/s
Cold head temperature (T_C)	140 K	Mass flow constant (h_{80})	2.31459e-4 kg/s
Initial interface load ($\dot{q}_{I,0}$)	2.64 W	Initial mass flow ($\dot{m}_{80}$)	1.353e-4 kg/s
Parasitic heat load ($\dot{q}_p$)	5 W	Interface thermal mass (C_I)	15 J/K
Stroke (f)	80%	Compressor frequency	60 Hz
Non-dimensional interface valve resistance ($R_{I,0}$)	2.44		

Note that the functional dependence of the mass flow rate on compressor stroke has been removed from Eq. (3); this is appropriate for the linearized model, as the control system will modulate the valve resistance and not the compressor stroke in order to maintain a fixed interface temperature. The Taylor expansion of differential Eq. (3) is performed using partial differentiation with respect to changes in the parameters that are allowed to vary: interface temperature (ΔT_I), interface load ($\Delta \dot{q}_I$), and interface valve resistance (ΔR_I). Only first order terms are retained, the higher order terms, as indicated as H.O.T. in Eq. (4), are assumed to be negligible. Therefore, the linear differential equation which describes the rate of interface temperature change is:

$$\frac{d\Delta T_I}{dt} = \Delta T_I \left(\frac{\partial}{\partial T_I} \frac{dT_I}{dt} \bigg|_{T_{I,0}, T_{C,0}, R_{I,0}} \right) + \Delta R_I \left(\frac{\partial}{\partial R_I} \frac{dT_I}{dt} \bigg|_{T_{I,0}, T_{C,0}, R_{I,0}} \right) + \Delta \dot{q}_I \left(\frac{\partial}{\partial \dot{q}_I} \frac{dT_I}{dt} \bigg|_{T_{I,0}, T_{C,0}, R_{I,0}} \right) + \cancelto{0}{H.O.T.} \quad (4)$$

where $T_{I,0}$, $T_{C,0}$ and $R_{I,0}$ are the initial nominal interface temperature cold head temperature and dimensionless interface valve resistance. After simplification, Eq. (4) yields the linearized governing differential equation:

$$\frac{d\Delta T_I}{dt} = \frac{-c_p (T_{I,0} - T_{C,0}) g_{80} \Delta R_I - c_p (g_{80} R_{I,0} + h_{80}) \Delta T_I + \Delta \dot{q}_I}{C_I} \quad (5)$$

where ΔT_I, ΔR_I, $\Delta \dot{q}_I$ are the perturbations from the nominal values of interface temperature, non-dimensional interface valve resistance, and interface load, respectively. For transient, frequency and root locus analyses, it is convenient to describe the system in the Laplace domain rather than the time domain. The Laplace transform of Eq. (5) yields:

$$\Delta T_I(s) = \frac{-c_p (T_{I,0} - T_{C,0}) g_{80} \Delta R_I(s) + \Delta \dot{q}_I}{C_I \left(s + \frac{c_p (g_{80} R_{I,0} + h_{80})}{C_I} \right)} \quad (6)$$

where s is the Laplace operator. Equation (6) shows that the interface system is first order (i.e., the denominator is a linear function of s) with a time constant of $C_I / c_p (g_{80} R_{I,0} + h_{80})$, or about 19 seconds. Using Eq. (6), it is possible to derive the transfer function that relates the non-dimensional interface valve resistance (the control parameter) to the interface temperature (the controlled parameter):

$$G_I(s) = \frac{\Delta T_I(s)}{\Delta R_I(s)} = \frac{-c_p (T_{I,0} - T_{C,0}) g_{80}}{C_I \left(s + \frac{c_p (g_{80} R_{I,0} + h_{80})}{C_I} \right)} \quad (7)$$

where $G_I(s)$ represents the interface plant with respect to the interface valve. Similarly, the interface plant with respect to a changing interface load is:

$$G_D(s) = \frac{\Delta T_I(s)}{\Delta \dot{q}_I(s)} = \frac{1}{C_I \left(s + \frac{c_p (g_{80} R_{I,0} + h_{80})}{C_I} \right)} \quad (8)$$

STATE FEEDBACK TEMPERATURE CONTROL

Proportional-Integral Control of Interface Temperature

The feedback control system makes corrective changes to the interface valve resistance (and consequently the cooling loop mass flow) based on the interface temperature error. The error is defined as the deviation of the interface temperature from its desired value ($\Delta T_{I,des}$); the desired value corresponds to no perturbation (i.e., we are assuming that the desired temperature is the nominal temperature listed in Table 1). Most thermal and fluid systems naturally exhibit stable first order transient behavior and therefore do not require derivative control [3]; therefore, a

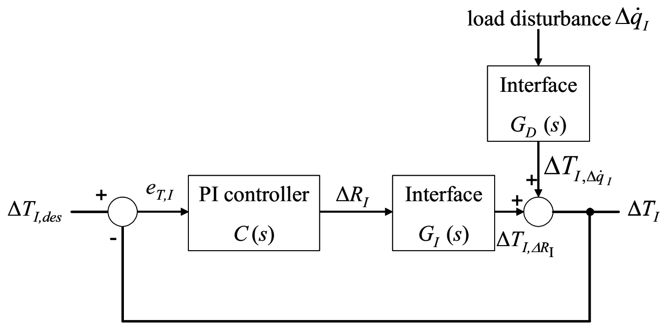

Figure 2. State block diagram of interface thermal model with PI control.

Proportional-Integral (PI) rather than a Proportional-Integral-Derivative (PID) controller is used to regulate the interface valve resistance according to:

$$\Delta R_I(s) = e_{T,I} \frac{k_{p,I}\, s + k_{i,I}}{s} \tag{9}$$

where $e_{T,I}$ is the interface temperature error defined as:

$$e_{T,I}(s) = \Delta T_{I,des}(s) - \Delta T_I(s) \tag{10}$$

and $k_{p,I}$ and $k_{i,I}$ are the proportional and integral gains that are set for the interface temperature controller. The interface temperature controller can be represented as a transfer function between the interface temperature error and the change in valve position, $C(s)$:

$$C(s) = \frac{\Delta R_I(s)}{e_{T,I}(s)} = \frac{k_{p,I}(s + k_{i,I}/k_{p,I})}{s} \tag{11}$$

The linearized controlled interface transient model in block diagram form is shown in Fig. 2. Note that in this configuration, the interface load fluctuation is depicted as a disturbance.

The Closed Loop Transfer Function (CLTF) for an interface load disturbance to the system shown in Fig. 2 is:

$$\frac{\Delta T_I(s)}{\Delta \dot{q}_I(s)} = \frac{G_D(s)}{1 + C(s) G_I(s)} \tag{12}$$

or:

$$\frac{\Delta T_I(s)}{\Delta \dot{q}_I(s)} = \frac{s}{C_I\left(s + \dot{m}_{80} c_p \big/ C_I\right)s - c_p (T_{I,0} - T_{C,0}) g_{80} k_p \left(s + k_I \big/ k_p\right)} \tag{13}$$

Determining PI Control Parameters – Step Response

Forming the system response in the Laplace domain facilitates the characterization of the system transient behavior in terms of its natural frequency and damping ratio. The temperature response in terms of these parameters is useful for developing a closed form solution for the maximum temperature fluctuation and settling time associated with rapid (i.e. stepwise) interface load changes. The corresponding controller gains required for a particular response can be readily calculated based on this analysis. This section describes the method for selecting the control parameters based on the desired temperature stability in terms of the maximum temperature fluctuation and settling time.

The PI controlled interface temperature is characterized by a 2nd order differential equation. The interface system response to a step change in load is studied in order to understand how the interface will respond to unpredictable, rapid load changes. An underdamped 2nd order system subjected to a step change in load exhibits a transient temperature response characterized by the product of a sinusoidal term and an exponential term that decays to zero (Eq. 23), indicating that the control system has restored the desired interface temperature. The maximum temperature error therefore occurs at the first peak of the sinusoidally varying response. The interface temperature response in the Laplace domain is calculated by multiplying Eq. (13) by an interface load function; therefore, the product of Eq. (13) and a unit step function (1/*s*) yields the interface temperature response to a unit step increase in interface load (note that other disturbances could be easily considered either analytically or numerically).

$$\Delta T_I(s) = \frac{1}{C_I\left[s^2 + \frac{c_p}{C_I}\left\{\left(\dot{m}_{80} - g_{80}k_p(T_{I,0} - T_{C,0})\right)s - g_{80}\,k_I(T_{I,0} - T_{C,0})\right\}\right]} \tag{14}$$

Figure 3 shows a controlled transient response and is referred to in order to clarify the transient response parameters. Note that the controller parameters used to generate Fig. 3 are selected so that the damping ratio is low (0.2) in order to highlight the oscillatory response.

The standard form of Eq. (14) is:

$$\Delta T_I(s) = \frac{1}{C_I\omega_n^2}\frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{15}$$

where ω_n is the natural frequency:

$$\omega_n = \sqrt{\frac{-g_{80}\,k_I\,c_p\,(T_{I,0} - T_{C,0})}{C_I}} \tag{16}$$

and ζ is the damping ratio:

$$\zeta = \frac{c_p\left(\dot{m}_{80} - g_{80}k_p(T_{I,0} - T_{C,0})\right)}{2\sqrt{-cp\,g_{80}\,k_I\,C_I(T_{I,0} - T_{C,0})}} \tag{17}$$

The maximum interface temperature fluctuation is determined by finding the first peak of the temperature response; the peak can be identified by setting the derivative of the temperature response equal to zero. The derivative of the temperature response given in Eq. (15) is:

$$\frac{d}{dt}\Delta T_I(s) = \frac{1}{C_I}\frac{s}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{18}$$

The inverse Laplace transform of Eq. (18) yields the derivative of the interface temperature response in the time domain [4]:

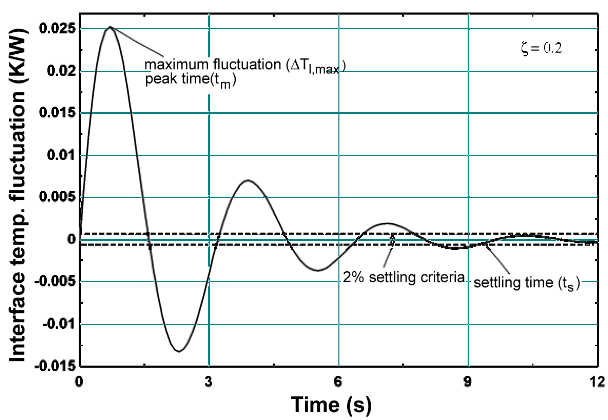

Figure 3 Thermal response of interface to step increase in thermal load which shows the important transient response parameters. The damping ratio is set relatively low to highlight the oscillatory behavior.

$$\frac{d}{dt}\Delta T_I(t) = \frac{-1}{C_I \beta} e^{\zeta\omega_n t} \sin\left(\omega_n \beta t - \phi\right) \tag{19}$$

where:

$$\beta = \sqrt{1-\zeta^2} \tag{20}$$

and

$$\phi = \tan^{-1}\left(\frac{\beta}{\zeta}\right) \tag{21}$$

The first peak of the oscillatory temperature response occurs at the peak time (t_p), when the sinusoidal term in Eq. (19) first becomes zero:

$$t_p = \frac{\phi}{\omega_n \beta} \tag{22}$$

The interface temperature in the time domain is calculated using the inverse Laplace transform of Eq. (15).

$$\Delta T_I(t) = \frac{\exp\left(-\zeta\, \omega_n\, t\right) \sin\left(\omega_n\, \beta\, t\right)}{C_I\, \omega_n\, \beta} \tag{23}$$

The peak interface temperature rise is then calculated by substituting the peak time, Eq. (22), into the time-domain representation of the interface temperature, Eq. (23):

$$\Delta T_{I,\max} = \frac{\exp\left(\dfrac{-\zeta \tan^{-1}\left(\dfrac{\beta}{\zeta}\right)}{\beta}\right) \sin\left(\tan^{-1}\left(\dfrac{\beta}{\zeta}\right)\right)}{C_I \beta \omega_n} \tag{24}$$

Equation (24) is useful as it provides a measure of the system stability given its physical characteristics as well as the controller gains. It is useful to describe the maximum temperature fluctuation using a dimensionless temperature. The dimensionless temperature fluctuation requires a characteristic time, which is taken to be the settling time (t_s); for a 2nd order system with a 2% settling criteria, the settling time is:

$$t_s = \frac{4}{\zeta \omega_n} \tag{25}$$

The dimensionless maximum temperature fluctuation, defined in Eq. (26), is a function only of the damping ratio:

$$\left(\frac{\Delta T_I\, C_I}{\Delta \dot{q}_I\, t_s}\right)_{\max} = \frac{\zeta \exp\left(\dfrac{-\zeta \tan^{-1}\left(\dfrac{\beta}{\zeta}\right)}{\beta}\right) \sin\left(\tan^{-1}\left(\dfrac{\beta}{\zeta}\right)\right)}{4\beta} \tag{26}$$

Note that Eq. (26) shows that, for a fixed damping ratio, the interface temperature response will scale linearly with the change in the load ($\Delta \dot{q}_I$). Also, a larger interface heat capacity or smaller settling time will tend to reduce the temperature fluctuation. Figure 4 shows the modeled prediction of the maximum interface temperature change for a step increase in interface load. Both the dimensional (a) and the nondimensional (b) interface temperature responses calculated using Eqs. (24) and (26) are shown in Fig. 4. The dimensional response in Fig. 4 (a) shows the response for a 1 W step increase in load. The interface temperature response is calculated using nominal parameters for the PTRI system (Table 1).

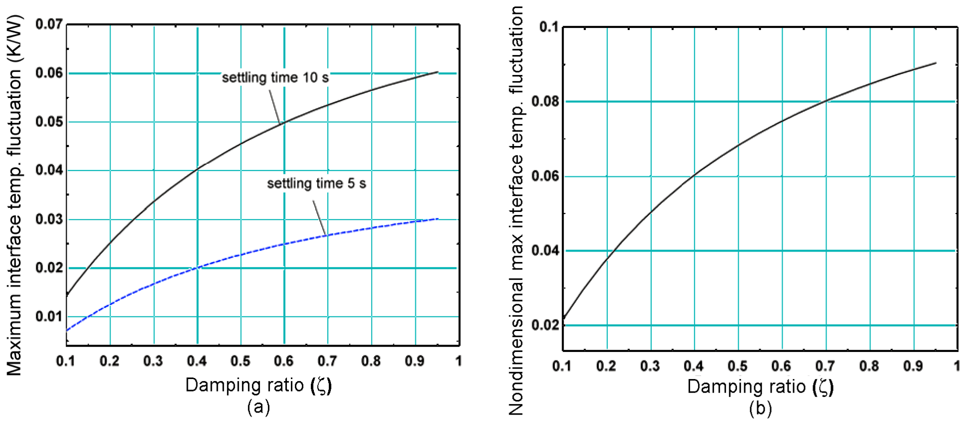

Figure 4. Maximum interface temperature fluctuation for a step increase in interface load using a PI controller. Both the dimensional (a) and nondimensional (b) maximum temperatue responses are shown.

Additionally, the temperature response is calculated assuming that the pertinent measurements required by the control loop have sufficient accuracy and time response to provide continuous feedback control.

Figure 4(a) shows that the maximum temperature fluctuation decreases with settling time; this relationship is advantageous as it shows that there is no tradeoff between settling time and maximum temperature. Additionally, the maximum temperature fluctuation decreases with damping ratio; this behavior can be explained by Eqs. (24) and (25). Equation (25) shows that for a given settling time, the natural frequency and damping ratio are inversely related. As shown by Eq. (24), the natural frequency in the denominator has a larger effect on the maximum temperature than the damping ratio, which is in the numerator. Therefore by decreasing the damping ratio but maintaining the settling time, the maximum temperature fluctuation will decrease.

In conclusion, a decrease in either the damping ratio or the settling time will allow the system designer to reduce the maximum temperature fluctuation. Note that decreasing the damping ratio is not necessarily the best way to minimize the fluctuations as systems with low damping are inherently less stable. Therefore, a reasonable damping ratio should be chosen (e.g., 0.5-0.8) and the maximum temperature fluctuation should be controlled by adjusting the settling time using the control gains, keeping in mind hardware limitations. Using Eqs. (16), (17) and (25) with a damping ratio of 0.707 and a settling time of 10 s, the control parameters required are $k_{p,I} = 5.51$ and $k_{i,I} = 2.36$. The corresponding maximum temperature fluctuation is 0.054 K; the dimensionless temperature fluctuation is 0.081. Table 2 summarizes these calculations.

VERIFICATION OF PEAK TEMPERATURE CALCULATION AND LINEAR MODEL

The analytic calculation of maximum temperature fluctuation that results from a step load is verified by comparing the linear model, Eq. (23), with the full, non-linear model discussed in [1]. Both models utilize the interface temperature feedback as well as PI control with the same controller gains. The transient temperature response for a 5 and 10 second settling time and damping ratio of 0.707 is shown in Fig. 5. The predicted maximum temperature fluctuation exactly matches the linearized model peak, and is very close to the non-linear model peak.

Figure 5 verifies that Eqs. (24) and (26) accurately predict the maximum temperature fluctuation for a step change in interface load; the discrepancy between the results grows as the deviation becomes larger (and thus the system moves further from the linearization point). Both the 5.0 and 10.0 second settling time responses show adequate agreement between the linear

Table 2: Parameters used to calculate controller gains and maximum temperature fluctuation.

Parameter	Value
Damping ratio (ζ)	0.707
Settling time (t_s)	10 s
Proportional gain ($k_{p,I}$)	5.51
Integral gain ($k_{i,I}$)	2.36
Interface load increase ($\Delta\dot{q}_I$)	1 W
Maximum interface temperature fluctuation ($\Delta T_{I,max}$)	0.05373 K
Dimensionless maximum interface temperature fluctuation	0.08059

and non-linear models, validating the use of the linear model for selecting control parameters. The mass flow adjustment associated with the control is readily achievable; for example the mass flow must be adjusted by approximately 10% over 8 seconds using a 10.0 second settling time and damping ratio of 0.707.

CONCLUSION

The PTRI system is an effective means of thermally managing distributed loads. The system offers advantages with regards to vibration and electrical isolation, does not require a cold circulator, can pick up multiple and highly separated loads through complex pathways, and can provide precise temperature regulation. The temperature regulation is achieved through modulation of the cooling loop mass flow which can be adjusted rapidly to respond to changes in thermal load; this advantage of the PTRI system was examined in detail in this paper. Experimental steady and transient data were used to create a thermal model that simulates temperature response of the PTRI to changing loads. A PI controller is integrated with the thermal model in order to evaluate and develop feedback control algorithms.

A linear version of the controlled thermal model is used to show that, with suitable hardware, the interface temperature can be precisely regulated under rapid and unpredictable load changes. The linearized thermal model is converted to the Laplace domain in order to facilitate the description of the temperature response in terms of its natural frequency and damping ratio. The governing interface temperature equation is then converted back to the time domain where the response is readily characterized by the maximum temperature fluctuation and settling time in terms of the damping ratio and natural frequency.

These useful design parameters can be directly used to select the appropriate control gains which govern the cooling loop mass flow. This approach was validated by comparing the

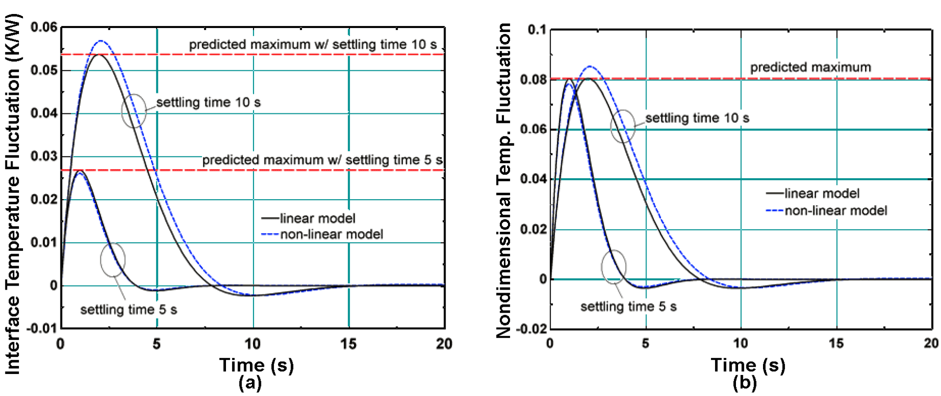

Figure 5: Linear and non-linear model prediction of controlled interface temperature response for a step increase in interface load. Both the dimensional (a) and nondimensional (b) temperature scales are shown.

transient thermal behavior predicted by the linear and non-linear models; their agreement validates the selection of control parameters based on the linear model.

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