

Investigating Thermal Testing Methods in the Frequency Domain

An Alternate Approach to Thermal Model Correlation

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by

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*Najeeba Murtuzapurwala
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I have approximate answers and possible beliefs and different degrees of certainty about different things, but I'm not absolutely sure of anything, and there are many things I don't know anything about.

Richard Feynman

Abstract

Thermal Mathematical Models (TMMs) are used to predict the thermal behaviour of satellite structures in orbit. However, due to inherent uncertainties in these models, physical testing is necessary to achieve reliable predictions. While these tests are critical, they often introduce uncontrolled uncertainties, such as heat leaks and measurement errors, making the correlation process complex and time-consuming. To address these challenges, this thesis proposes an alternative testing methodology that reduces uncertainties in thermal test data by using the phase shifts between temperature responses from oscillatory heat loads for TMM correlation. By comparing the measured and predicted phase shifts, thermal model parameters such as conductance or capacitance can be effectively correlated. The results demonstrate that this methodology is largely insensitive to uncontrolled conductive heat leaks, allowing the correlation process to focus only on key thermal parameters. This approach improves the reliability of thermal test data and streamlines the correlation process. Moreover, its potential application in ambient conditions offers a promising testing solution for early-phase model correlation.

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Acronyms

Abbreviation	Definition
ECSS	European Cooperation for Space Standardisation
FDTR	Frequency Domain Thermoreflectance
FFT	Fast Fourier Transform
GL	Thermal Conductive Link
GMM	Geometric Mathematical Model
GR	Thermal Radiative Link
MLI	Multi-Layer Insulation
NGN	Non Geometric Node
ODE	Ordinary Differential Equation
OTRT	Oscillatory Thermal Response Test
PCB	Printed Circuit Board
PCU	Power Supply Unit
RTD	Resistance Temperature Detector
SNR	Signal-to-Noise Ratio
STM	Structural-Thermal Model
TC	Thermocouple
TCS	Thermal Control Subsystem
TCC	Thermal Contact Conductance
TMM	Thermal Mathematical Model
TWA	Thermal Wave Analysis
VDA	Vacuum Deposited Aluminium

1

Introduction

Space technology has rapidly evolved since the launch of Sputnik-1 in 1957 [1], reaching a level of complexity that could not have been imagined in its early years. Modern satellites consist of intricate systems of multiple units that work together to ensure successful operation and mission fulfilment. These systems are broadly characterised based on their ‘payload’ and ‘bus.’ The satellite’s payload is responsible for fulfilling the primary space mission objective, be it antennas and a communication system for telecommunication or a cloud cover camera for weather monitoring [2]. On the other hand, the satellite bus or platform consists of several subsystems that support the mission. Figure 1.1 demonstrates the basic subsystems of a generic communication satellite.

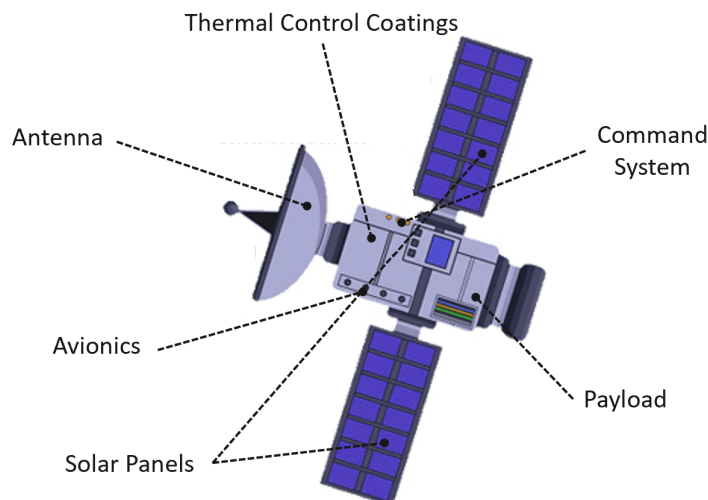


Figure 1.1: Subsystems of a generic communication satellites, adapted from [3].

These subsystems are integrated and coordinated with precision to ensure the functionality and longevity of the satellite. Lightweight, high-strength materials like aluminium alloys,

titanium, and composites are used to build the satellite structure, components, and other mechanisms. These parts are designed to withstand the mechanical stresses experienced by the satellite during launch and in space.

However, the space thermal environment is both extreme and complex. For Earth-orbiting missions, environmental factors are influenced by solar radiation, Earth's albedo¹, and infrared radiation emitted by Earth. These conditions fluctuate as the satellite orbits the planet, such as during transitions in and out of solar eclipses. These environmental swings affect the thermal behaviour of the subsystems. Therefore a robust thermal control subsystem (TCS) is needed to maintain all components within their permissible temperature ranges for survival during the mission and operational success. The purpose of TCS is to efficiently manage heat within the satellite using minimum spacecraft resources and minimise temperature gradients to guarantee temperature stability for components sensitive to temperatures [1]. In space, this becomes a challenge as heat transfer is limited to conduction, which allows for internal heat distribution and radiation, which serves as the primary mode of heat rejection. Different types of active and passive thermal control devices are used in combination to address the thermal challenges of a satellite. In every case, significant efforts are dedicated to verifying and validating the TCS design.

As the space sector has grown more dynamic, driven by budget constraints and shorter product development cycles [4], computational analysis has become standard practice in supporting TCS design and development. Thermal Mathematical Models (TMMs) are routinely used to verify TCS designs and predict thermal behaviour under space conditions [2]. These models are integral at every stage of the development cycle, from conceptual design to final in-flight predictions, as shown in Figure 1.2.

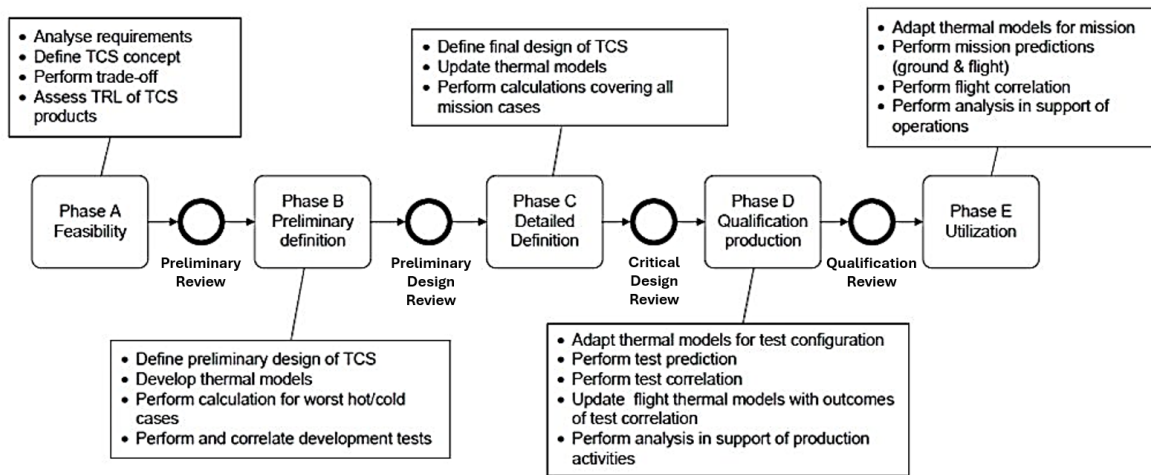


Figure 1.2: Thermal analysis in space project cycle, adapted from [5]

In some cases, thermal analysis remains the only feasible method to verify certain thermal requirements when physical testing is either too expensive or impractical [5]. Therefore, ensuring the highest quality in these models is important for the credibility of their predictions.

¹**Albedo** refers to the reflection of solar radiation from the Earth's surface [1]

1.1. Challenges with Current Methods

Ground thermal tests are conducted at various stages of the satellite development cycle to serve multiple purposes: they assess the flightworthiness of components, screen for defects, demonstrate the performance of the TCS in environments that closely resemble on-orbit conditions and verify the accuracy of the TMM [2]. The three primary thermal tests are the thermal vacuum test, thermal cycling test, and thermal balance test. Thermal vacuum tests confirm the operation and survival of the test article in a vacuum, simulating the harsh conditions of space, while thermal cycling tests assess performance under temperature fluctuations at ambient pressure [1]. The thermal balance test evaluates the performance of the TCS and validates TMM accuracy under steady-state conditions in a vacuum. These tests are strategically performed at different levels, unit², subsystem³, and system⁴, throughout the satellite's development phases, shown in Figure 1.2.

At the unit level, thermal cycling is often conducted early in the development process to detect workmanship defects and screen for thermal stress. At the subsystem and system levels, thermal cycling is combined with thermal vacuum and thermal balance tests to ensure comprehensive evaluation and model verifications in vacuum environments. When developing new designs or applying existing techniques to new applications, these tests are also conducted during the preliminary phase to provide early data that help inform the design process [1].

However, these tests are time-consuming and costly, as they require prolonged stabilization periods in a vacuum and the application of high, constant heat loads to accurately capture the temperature response throughout the unit. While the exact cost of conducting vacuum environment tests is significant, with costs ranging from hundreds of thousands to millions of USD, depending on the size of the satellite or spacecraft, they are an indispensable part of the validation process. Studies have proposed methods to reduce these costs, by exempting certain units from thermal vacuum tests or deferring these tests to the highest level of assembly [6, 7], but these approaches introduce additional risks later in the development cycle.

Additionally, with the growing dependency on simulations, thermal balance tests are also an important part of the product cycle. This is because there are considerable discrepancies and uncertainties between real-world results and mathematical models. Variations in material properties, electronic dissipation, and thermal contact conductance within the models often lead to inaccurate predictions and are inherent inaccuracies of a model. Furthermore, the test data can be compromised by uncontrolled heat leakages due to mounting and test support equipment, reflections from chamber walls, measurement errors, and thermal losses through wiring. These testing uncertainties make the correlation process more complex and time-consuming, as they are difficult to locate and quantify.

Given this context, an alternate testing methodology was explored. The primary objective was to establish an initial testing and correlation framework that generates temperature data for model correlation while reducing both costs and uncertainties associated with traditional testing methods.

²**Unit** is a function item viewed as a complete entity for manufacturing, for example, electronic boxes.

³**Subsystem** is an assembly of units that are functionally related, for example, an electric power subsystem.

⁴**System** refers to the complete integrated satellite structure

1.2. Thesis Outline

The thesis is outlined as follows:

Chapter 2: Literature Review

This chapter reviews existing literature on model correlations and frequency domain thermal testing. It highlights the current research being done and identifies a gap in the application of these methodologies within the space industry.

Chapter 3: Frequency Domain Thermal Analysis

This chapter conducts a theoretical study to evaluate the feasibility of using oscillatory heat loads as an alternative testing method. The analysis includes assessing the potential benefits, challenges and trends associated with this approach.

Chapter 4: Post-Processing and Data Analysis Methods for Thermal Testing

This chapter discusses the laboratory tests conducted to apply the methodology. It details the experimental setup, procedures, and the lessons learned from these tests, providing practical insights into the application of this methodology.

Chapter 5: Frequency Domain Testing and Correlation

This chapter presents the application of frequency-domain thermal tests and model correlation. The results are compared against traditional testing methods to confirm the reliability and accuracy of the new approach.

Chapter 6: Conclusions

The final chapter summarizes the findings of the study, discussing the implications of the results and potential areas for future research.

2

Literature Review

Before designing a correlation methodology, it is important to understand how thermal mathematical models are created, their basic physics, and the various sources of uncertainties in these models. Section 2.1 delves into the current modelling practices, the inherent uncertainties they introduce, and the practices used in the space industry to mitigate these uncertainties. Next, section 2.2 introduces an alternative testing strategy - thermal testing in the frequency domain. It discusses the methodology of frequency domain tests already employed in different industries for thermal property characterisation. Finally, the research objectives are outlined, and key questions are posed in section 2.3 to guide the study.

2.1. Thermal Mathematical Models and Correlations

Before the development of commercial thermal analysis software such as ESATAN, NASA's SINDA, Thermica, etc., thermal analysis was primarily carried out using manual calculations, simplified analytical models, and basic in-house computer programs [8]. The space environment was simulated using prototypes and performing extensive testing in thermal vacuum chambers. This process was time-consuming and required significant expertise and iterative adjustments. The introduction of thermal analysis software, in the 1980s, provided more precise and efficient tools for thermal modelling and simulation [8].

Today, ESATAN-TMS is widely used in the European space industry for modelling and simulating satellite thermal analysis. Thermal mathematical models (TMM) are a numerical representation of a satellite structure and its surroundings, used to predict its temperature distribution and thermal behaviour under various environmental conditions and operational modes. They are created using the satellite geometry, thermo-physical properties of the material, and the presence of internal heat sources and insulation layers [2]. They use the Thermal Lumped Parameter method, which divides the geometry into discrete isothermal nodes and assumes that the thermal properties of each sub-volume are concentrated at its centre [9]. Each node in the structure has a thermal capacitance, that accounts for heat storage effects. These nodes are connected by conductive (GL) and radiative (GR) links to nearby nodes, creating a thermal network that represents the pathways for heat transfer between different parts of the

structure. This simplifies complex geometries while accurately capturing its thermal behaviour. Figure 2.1 illustrates the discretisation of a simple 2D bar into nodes.

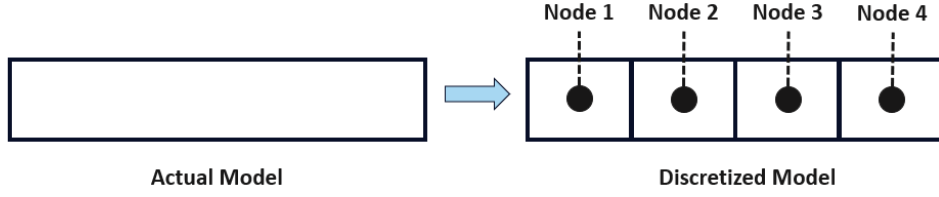


Figure 2.1: Nodalisation of a simple 2D bar.

In the lumped parameter approach, an energy balance at each node describes temperature changes over time, generating a system of nonlinear ordinary differential equations (ODE) that model the temperature evolution of the nodes [10]. The heat transfer equation represents the conduction and radiation in the model. Mathematically, it is a system of first-order ODEs, with an equation for each node. For a model with n nodes, the equation at node i is given by Equation 2.1.

$$C_i \frac{dT_i}{dt} = Q_i + \sum_{j=1}^n GL(i,j) (T_j - T_i) + \sigma \sum_{j=1}^n GR(i,j) (T_j^4 - T_i^4) \quad (2.1)$$

where,

- $GL(i,j)$ = Conductive links between node i and j [W/m]
- $GR(i,j)$ = Radiative links between node i and j [m²]
- C_i = Thermal capacitance of node i [J/K]
- T_i, T_j = Nodal temperatures of node i and j respectively [K]
- Q_i = Net heat flux at the node i [W/m²]
- σ = Stefan-Boltzmann constant, 5.67×10^{-8} [W/m²K⁴]
- dT_i/dt = Temperature increment in node i after time dt

The net heat flux (Q_i) is the sum of the heat loads from the external environment and the internal heat dissipation at node i . In Equation 2.1, the left side represents the rate of change in internal energy stored in the node, while the right side represents the net heat flow to the node, including the heat flowing into the node through linear and radiative conductance.

The radiative couplings within the model and from the external environment are computed and stored in the Geometric Mathematical Model (GMM) [2]. Additionally, it stores information related to the Gebhart factors¹ between the nodes, which can be tedious to calculate manually, especially for complex geometries. So in practice, ESATAN-TMS is used to automatically compute this information in the GMM. The most common approach to computing this is using a Monte Carlo ray-tracing algorithm [11]. The TMM, on the other hand, represents the conductive couplings between the nodes and their thermal capacitance. It is used, along with the GMM, to predict the temperatures within the system [2].

¹**Gebhart factor** gives the fraction of the total radiated energy from surface 1 that is incident on surface 2, including all reflections between them [2].

In a steady-state condition, a simplification can be done to Equation 2.1. The term dT_i/dt can be set to zero as the nodal temperatures remain constant over time. This reduces the energy balance to Equation 2.2, leaving a system of non-linear algebraic equations that can be solved for the steady-state temperatures.

$$0 = Q_i + \sum_{j=1}^n GL(i,j) (T_j - T_i) + \sigma \sum_{j=1}^n GR(i,j) (T_j^4 - T_i^4) \quad (2.2)$$

Thermal Modelling and Analysis Procedure

Figure 2.2 illustrates the process of building a thermal model. It begins with discretising the geometry and dividing the system into discrete isothermal nodes, each corresponding to various elements within the satellite. These nodes can take different forms [2]:

1. **Diffusive nodes** that possess finite capacitance, representing materials whose temperatures fluctuate due to heat flow into or out of the node.
2. **Arithmetic nodes** that are characterised by zero capacitance and serve a computational purpose, particularly in representing elements with minimal capacitance relative to others. This is physically unrealistic but helps with computational efficiency.
3. **Boundary nodes** that have infinite capacitance and symbolise boundaries with fixed temperatures that remain constant over time.

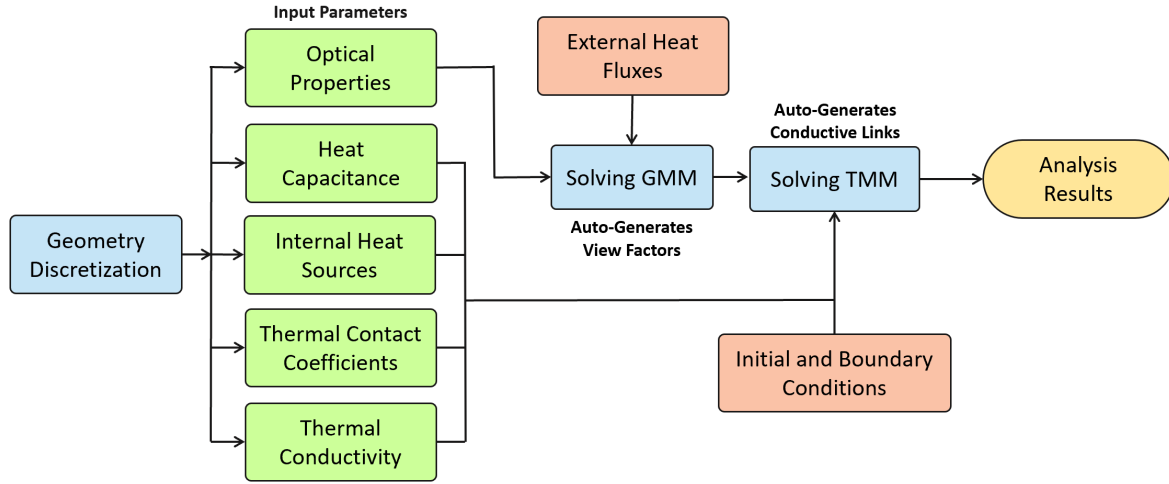


Figure 2.2: General thermal analysis procedure

Thermo-optical properties are then assigned as input parameters. After calculating and applying external heat fluxes² as boundary conditions, the GMM computes view factors and radiative conductances between nodes. Heat capacities are determined using the assigned density, specific heat, and volume of each node. Conductive couplings are calculated based on whether nodes are within a continuous solid medium or in contact. For the former, heat conduction is determined by thermal conductivity, while for the latter, it involves multiplying the

²External heat fluxes are calculated by defining the orbit and attitude of the space structure and can be generated using thermal analysis software like ESATAN or estimated from literature based on specific orbital parameters.

contact area by the thermal contact conductance coefficient. With known internal heat sources and established initial and boundary conditions, the TMM can then be solved for predicting the thermal behaviour in the orbit.

When built carefully, the thermal models can provide reliable predictions of the thermal behaviour of the satellite under various conditions. However, like any model, it is subject to errors stemming from various sources such as input parameters, simplifications made during model development, and inherent complexities within the system itself [9].

2.1.1. Sources of Uncertainties in TMM

Thermal Mathematical Models rely on precise knowledge of geometry, physical properties and boundary conditions [12]. However, in practice, various uncertainties can arise in the models due to inaccuracies in the environmental, physical and modelling parameters [11]. Studying these potential uncertainties helps establish proper design margins and improves the accuracy of the predicted temperatures.

Environmental uncertainties stem from inaccuracies in solar and planetary radiation and orbital parameters of the TMM. However, since space structures are designed based on worst-case environmental conditions, varying these parameters can sometimes result in over-engineering. Therefore, these uncertainties are evaluated on a case-by-case basis, depending on the requirements. Modelling uncertainties, on the other hand, are inherent and arise from three sets of thermal network parameters - GLs, GRs, and thermal capacitance. These stem from inaccurate or insufficient knowledge of thermal and optical properties or modelling inaccuracies.

Uncertainties in Radiative Links

A radiative conductance exists between two nodes coupled radiatively. This coupling depends on the surface area, the Gebhart factor (B) and the optical properties, i.e. solar absorptivity (α) and infrared emissivity (ϵ) of the node. In its simplest form, it is given by Equation 2.3.

$$GR = \sigma \epsilon \alpha AB \quad (2.3)$$

The first two properties are geometric and depend on the modelling quality. For complex geometries, the ECSS³ specify inaccuracies in these shape factors as high as $\pm 50\%$ [13]. The optical properties, on the other hand, are a more common source of uncertainty. Solar absorptivity measures how effectively a component can absorb solar radiation, while infrared emissivity measures how effectively a component can radiate infrared energy compared to a black-body⁴. These optical properties are determined experimentally using reflectance measurements, which require highly controlled environments. Additionally, both properties are influenced by the component's temperature and surface finish and can be altered by polishing or painting the surfaces [2]. Monte et al. [15] conducted experiments to determine the uncertainties in ϵ measurements but found that no general statements could be made due to its complex dependence on sample conditions, so, individual uncertainty estimation is needed.

³European Cooperation for Space Standardisation

⁴A **black-body** is an object that can absorb all radiation energy and emit it in all directions [14].

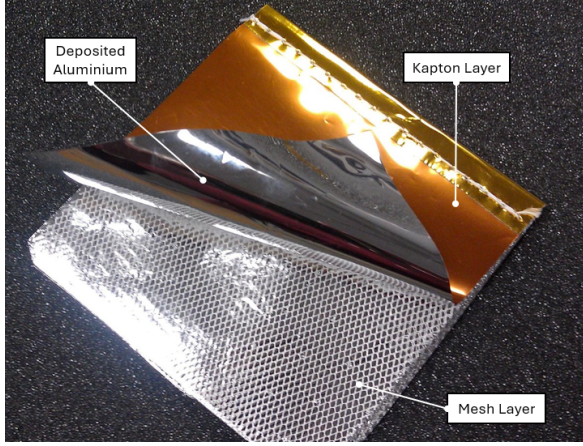


Figure 2.3: Multi-layer insulation [16]

mounting techniques of MLI on its ϵ^* and revealed a direct dependence on the manufacturing process. Similarly, Wei et al. [18] demonstrated that the shape and structure of the MLI also significantly influence its ϵ^* .

Space structures are complex in their geometries, and surface finishes and use intricate assemblies of MLI blankets. These parameters cannot be generalised and therefore introduce inherent uncertainties in a TMM that need to be correlated with actual test data.

Uncertainties in Conductive Links

The conductive links between two nodes can take two different forms, both of which can introduce inaccuracies in a TMM. The first type occurs as heat conduction between nodes within a continuous solid medium, governed by Fourier's law of conduction. Mathematically, this relationship is represented by Equation 2.4.

$$GL = \frac{kA}{L} \quad (2.4)$$

where k is the thermal conductivity [W/m.K] of the material, A is the cross-sectional area [m²] normal to the direction of heat flow and L is the distance [m] between the two nodes. The cross-sectional area and distance between the nodes are geometric parameters that depend on the quality of the model's discretization. While uncertainties in these parameters can be minimised by refining the nodal discretization of the geometry, this approach increases computational time. Therefore, such refinements are typically made only when necessary.

On the other hand, the thermal conductivity of the material, k , is often a source of uncertainty, as it is generally determined through experimental tests that inherently carry a margin of error. However, in components made of metal alloys, thermal conductivity is seldom considered a significant source of uncertainty, because of their homogeneity. But, in components made of composite, such as carbon fibre-reinforced sandwich panels, their effective thermal conductivity (k_{eff}) can be difficult to quantify due to their anisotropic nature and variability in manufacturing processes. Therefore they can introduce considerable uncertainties in the model. Pietrak et al. [19] reviewed the numerous models developed since the 19th century to predict k_{eff} in composites. However, these models fail to make an accurate prediction, as the

Another significant source of uncertainty in the radiative links arises from the effective emissivity (ϵ^*) of MLI, a common heat control system used to provide radiative insulation to space structures. They prevent both heat losses to outer space and excessive heating from environmental fluxes or rocket plumes [2]. The simplest MLI construction consists of multiple Kapton layers each with a vacuum-deposited aluminium (VDA) finish on one side, and mesh layers in between them, as shown in Figure 2.3. Kim et al. [17] investigated the impact of manufacturing and

parameter is not only a function of the thermal resistance between the composite layers but also depends on the material properties of the constituents, the shape and size of fibres used and the fibre volume fraction in the composite [20]. Therefore, k_{eff} is typically treated as an uncertainty in structures made of composite materials

The other form of conductive coupling exists between two nodes that represent a contact in the geometry. This is represented using the Equation 2.5.

$$GL = h_c A \quad (2.5)$$

where h_c is the thermal contact conductance (TCC) [W/m^2K] and A is the contact area [m^2].

A major source of uncertainty here lies in the thermal contact conductance parameter. All machined surfaces, regardless of their surface finishes, have some microscopic irregularities. The contact surfaces contain peaks and valleys, as illustrated in Figure 2.4, that differ at each contact interface. In a vacuum, heat transfer between the surfaces occurs through conduction at the contact points and radiation across the gaps. However, in ambient conditions, the presence of air in these microscopic gaps can increase heat transfer due to additional gaseous conduction. Therefore, accurately gauging this value is a complex task. Additionally, the sensitivity of TCC to factors like pressure, contact area, etc. makes it difficult to generalise the parameter. Parikh et al. [21] investigated an inverse heat transfer algorithm to accurately estimate TCC in different contact configurations. Their results are promising and experimentally validated. However, most space structures are highly complex geometries and often depend on thermal contact conductance for the effective internal distribution of heat in the structure. Therefore, in practice, predicted TCC values always need to be correlated with test data for higher certainty.

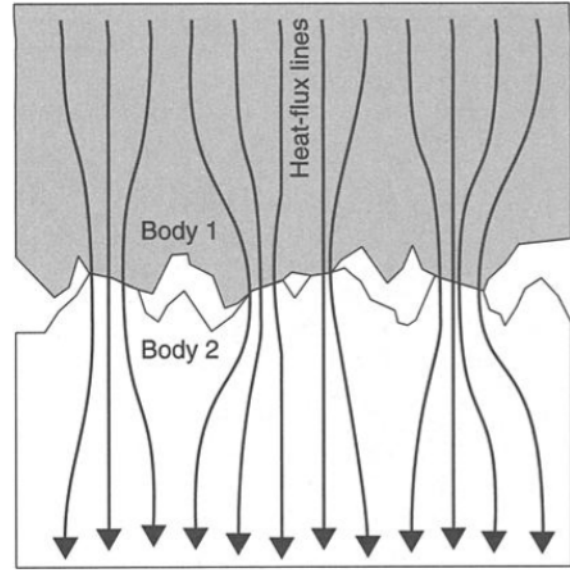


Figure 2.4: Microscopic view of a contact [2]

Uncertainties in Thermal Capacitance

When assessing the transient thermal behaviour of a component, it is important to consider its thermal capacitance. Thermal capacitance (C) measures the heat storage capacity of a component and is mathematically expressed as the product of specific heat and mass of the node, as shown in Equation 2.6.

$$C = mc_p = \rho Vc_p \quad (2.6)$$

where, c_p is the specific heat [J/kgK] of the material, m is the mass [kg] of the node, ρ and V are the density [kg/m^3] of the material and the volume [m^3] of the node, respectively. The modelling uncertainties here come from the simplification of the geometries [22]. This intro-

duces inaccuracies in the volume of the nodes, and consequently, the mass of components. However, these inaccuracies can be mitigated by adjusting the thermal capacitance of the nodes to match the actual mass of the components within the geometry.

The specific heat of metal alloys, similar to their thermal conductivity, is seldom considered an uncertainty. However, the specific heat of composite structures also depends on the constituents and the fibre orientations and is therefore considered an uncertainty. Experimental methods have been developed to estimate the specific heat capacity of composite materials [23–26] but these methods are limited to the type of composite under investigation, have uncertainties in measurements and are highly time-consuming. Thermal behaviour in orbits is always transient, therefore, the thermal capacities of composite structures need to be correlated with the test data.

Specific heat is also a major source of uncertainty in units containing numerous electronic components, such as the electronic box of a satellite, which includes various printed circuit boards (PCBs), wires, capacitors, resistors, and other elements, each with different heat capacities. These variations throughout the unit are typically not modelled individually in a TMM, leading to assumptions about their specific heats. Given that these units generate internal heat and influence the transient behaviour of the entire system, accurately assessing their specific heat is crucial. Therefore, these specific heats are treated as uncertainties and correlated with test results to ensure accurate modelling.

Significant research has been conducted to assess and optimise the extent of uncertainties in a TMM and study their impact on system performance. Thunnissen et al. [27] presented a method to quantify possible uncertainties in the conceptual design phase using a Probabilistic Subset Simulation method. Fu et al. [28] developed a method to calculate uncertainties using a chaotic polynomial expansion model, which accounts for both accuracy and efficiency in the analysis. While these methods help identify critical uncertainties in the design phase, they do not eliminate the need for ground thermal tests.

Table 2.1: Generic inaccuracies in parameters [11]

Inaccuracies in Physical and Geometric Parameters	Inaccuracy
Absorptivity	± 0.1
Emissivity	± 0.03
MLI Effective Emissivity	$\pm 50\%$
Composite Thermal Conductivity	$\pm 30\%$
Thermal Contact Conductance	$\pm 50\%$
Thermal Capacity	$\pm 25\%$
Inaccuracies in Test Facilities	Inaccuracy
Test hardware	$\pm 2\%$
Temperature sensor measurement accuracy	$\pm 1.5^{\circ}\text{C}$
Nodal/Sensor position error	$\pm 3^{\circ}\text{C}$

Additionally, the thermal tests themselves introduce uncertainties that are difficult to include in a TMM. ECSS standards specify a distinct set of uncertainty parameters tailored for model correlation tests, which address inaccuracies in temperature measurements, sensor positioning, and the thermo-optical properties of the test facility and hardware [11]. Uncertainties in the test hardware, such as heat leaks from harnesses and the absence of components in the test configuration, also need to be considered. These uncertainties vary based on the test setup and required accuracy, necessitating careful consideration during the correlation process [13]. An unexhaustive list of typical inaccuracies in the TMM parameters is provided in Table 2.1.

Due to the small margin of error in space projects, the reliability of a TMM is of utmost importance. Rigorous analysis is necessary to identify potential design issues and reduce the risk of failures in orbit. Ground thermal tests play a crucial role in this process by improving the robustness of these models and verifying the performance of the structures under test conditions. Therefore, various ground thermal tests are conducted, each tailored to suit specific testing objectives. The test data from Thermal Balance Tests is used to correlate the TMMs.

2.1.2. Thermal Balance Tests and Model Correlations

Thermal Balance Tests are performed with two objectives: to collect temperature data for TMM correlation and to verify the thermal control subsystem. They include dedicated hot and cold phases conducted in a controlled vacuum environment with predefined operational conditions to simulate the worst-case flight scenarios [2]. A generic thermal balance test typically consists of three distinct phases. These include a hot and a cold operational phase to verify the operational thermal limits, and a cold non-operational phase to demonstrate non-operational limits and validate heater capabilities. [2]. At each phase, the subsystem is thermally stabilised so that the unit can reach steady-state temperatures.

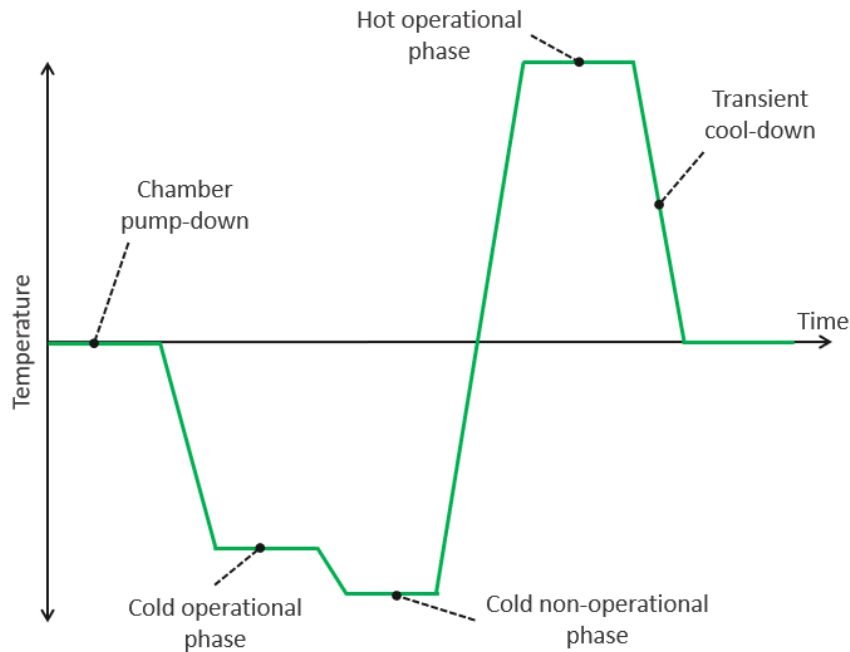


Figure 2.5: Generic thermal balance test profile. Adapted from [2].

Figure 2.5 illustrates a generic thermal balance test profile. At each phase, the survivability of the thermal control subsystem is verified. Strict stability criteria are also applied to ensure steady-state conditions are met. Although specific criteria are defined based on the test article, the general criteria follow a temperature change of 0.1 °C per hour for two hours. The success of the test depends not only on verifying the subsystem but also on accurately correlating the TMMs with the collected temperature data [2].

Correlation, however, begins before the tests are conducted, with the development of structural-thermal models (STMs), which are modified flight thermal models representing the chamber configuration and thermal test environment. This modification involves removing components not present in the test chamber, adding thermal nodes for test hardware, such as a baseplate or chamber walls, and adjusting power dissipation levels to reflect test conditions. Additionally, the GMM is updated to account for changes in view factors due to blockages caused by hardware and test equipment. These updated radiative couplings replace the original flight view factors, ensuring accurate test simulation and data correlation [2].

The modified TMM generates predicted temperature data for the different test phases. These predictions are compared with the measured temperature data for each phase. Steady-state phases are correlated using the corresponding temperature data, while transient phases are matched using transition phase data [2]. In regions of large discrepancies, the geometric and physical parameters of the TMM are adjusted till they match the measured thermal behaviour. Guidelines established by the ECSS, based on past project experiences, help determine successful correlation and are summarised in Table 2.2. A lack of successful correlation may indicate deficiencies in the thermal model, test setup, or subsystem hardware [2].

Table 2.2: Correlation success criteria [11]

Typical Temperature Deviation	
Internal Unit	<5K
External unit	<10K
Mean deviation	±2K
Standard deviation	<3K (1 σ)

Although critical to the design cycle, thermal balance tests and TMM correlation can be time-consuming and costly. This is especially true for complex models, where multiple parameters can influence the results. Significant efforts have been made to automate the process to address this challenge [29–31]. However, Garmendia et al. [32] in their research highlighted the risk of losing the physical meaning of modified parameters during automation. Thus, some manual oversight is necessary to ensure accuracy and reliability. This study does not focus on optimising the correlation process, as previous researchers have done. Instead, the aim is to design a different thermal test process that can be used for collecting data for model correlation.

Thermal balance tests use a constant heat load, applying a constant power input to the unit to accurately characterise the system's thermal behaviour in an equilibrium condition. However, in some cases, this requires the heat input to be high enough to characterise all possible heat paths. Additionally, achieving the stabilisation criteria can take as long as 12 hours or more, depending on the complexity of the test article. Since these tests are done in a vacuum environment, this wait time drives up the costs. Furthermore, keeping track of all the possible heat paths can be difficult, especially if there are uncontrolled heat leaks present in the test. Therefore, to address these issues, an alternative test methodology can be designed.

2.2. Thermal Testing in Frequency Domain

Based on fundamental heat transfer laws, two categories of thermal testing methods exist: steady-state methods and transient methods [33]. Each method has a range of applications depending on the material being tested, sample size, and operational temperatures.

Frequency domain thermal testing offers a different approach to analysing the thermal properties and behaviour of materials and structures. Unlike traditional methods, which examine temperature variations over time due to a constant heat source, this approach involves applying oscillatory or periodic thermal loads to the system and measuring the resulting thermal response. By analysing the amplitude and phase shift of the temperature responses, critical thermal properties, such as thermal diffusivity, heat capacity, and thermal conductivity, can be determined. A few popular frequency domain thermal tests are discussed below. It should be noted that although these tests use a common methodology, that involves oscillatory heat loads, the unique characteristics of each test article require tailored approaches to interpreting the temperature responses for material characterisation.

2.2.1. Thermal Wave Analysis Method

The Thermal Wave Analysis (TWA) Method is a popular frequency domain test. It is used to measure a material's thermal diffusivity, which is a measure of how fast heat diffuses through a material, and conductivity by applying a steady sinusoidal heat input to a sample. It was first developed in 1861 by Ångström [34] to measure the thermal diffusivity of metal rods. Figure 2.6 graphically illustrates his concept.

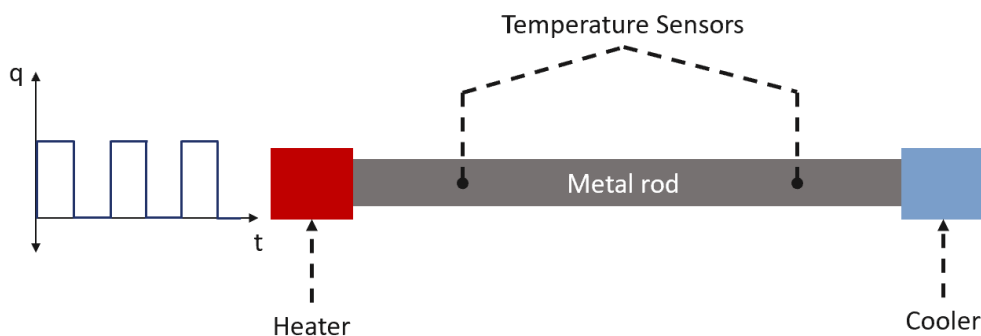


Figure 2.6: Ångström's thermal wave analysis concept

He applied a periodic heat load to one end of the metal rod while the other end was cooled. Temperature sensors were placed along the rod to measure the temperature responses. He then calculated the thermal diffusivity of the metal rod from the distance between the temperature sensors, the time taken by the thermal wave to travel between them and their amplitudes. Compared to the existing steady-state alternatives, his method offered a simple setup which was easy to reproduce. However, it assumed a one-dimensional heat flow, so its applicability was limited to thick samples with low conductivity, allowing heat losses through the side walls to be neglected [34].

Since then, significant research has been done to modify Ångström's method and expand its applicability. Mathematically, thermal diffusivity can be expressed as the ratio of thermal conductivity and the volumetric heat capacity of a material, at a constant pressure. Czarnecki et al. [35] and Bodzenta et al. [36, 37] used this relation to simultaneously measure the thermal diffusivity and conductivity from the amplitude gains and phase shifts in the temperature responses. They successfully measured these properties of both conductive and non-conductive materials. Nam et al. [33] also successfully adapted this method to measure the thermal diffusivity of porous insulation materials in a vacuum environment for space applications.

The method has also been used to measure thermal contact conductance in joints. Theoretical and experimental tests were conducted by Robinson et al. [38] to measure thermal contact conductance with an oscillatory heat input. They found a direct dependence of the thermal wave properties, like the amplitude gains and phase shifts, on the contact conductance, as well as the thermophysical properties of the materials in contact. Despite the small temperature amplitudes at sensors far from the heat load, the phase shift of the temperature profile, obtained using generic curve fitting methods, provided an accuracy comparable to steady-state methods. Similar results for a cylindrical joint were also presented by Benigni et al. [39], confirming the applicability of this method for quantifying the contact conductance between interfaces irrespective of their distance from the heat source.

2.2.2. Frequency Domain Thermoreflectance

Frequency Domain Thermoreflectance (FDTR) is another effective method that uses periodic heat waves to measure thermal properties. Unlike the TWA, which uses a heater in direct contact with the test article, FDTR generates periodic heating using a laser beam. The modulation in the thermal response in the sample is monitored using a photodetector. The delay in the phase between the laser beam and thermal response is used to extract the thermal properties of the sample [40]. The typical test setup is shown in Figure 2.7. The method is commonly used to measure the thermal properties of samples as thin as $10\mu\text{m}$.

In 1983, Opsal et al. [41] developed this method to measure the thermal properties of Al and SiO_2 films of thickness 500-2500 Å. Due to the flexibility of his approach, many variations of the methodology are still being explored. Schmidt et al. [42] proposed a methodology for measuring the thermal properties of homogeneous thin films. Rahman et al. [43] adapted it for anisotropic materials. Malen et al. [44] also successfully modified it to develop fibre-aligned FDTR to measure the thermal properties of electrically conductive samples.

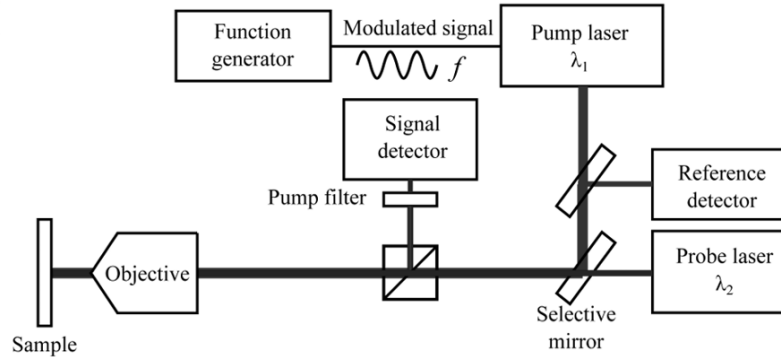


Figure 2.7: A typical FDTR test setup. Adapted from [42]

2.2.3. Oscillatory Thermal Response Test

Oscillatory Thermal Response Tests (OTRTs) also use oscillatory heat input but they are used to characterise the thermal properties of Earth's subsurface material. The subsurface thermal conductivity and thermal diffusivity are important parameters to design ground heat pump systems [45]. They were conventionally determined either from literature or from different laboratory analyses. But in 1987, Eskilson [46] investigated the use of sinusoidal heat injection to the ground and simultaneously estimate the subsurface thermal conductivity and diffusivity. He provided expressions to infer these properties from the amplitude attenuation and phase shift of the thermal responses to the sinusoidal heat. Using these findings, Giordano et al. [47] evaluated the effectiveness of OTRTs as a tool to measure subsurface thermal heat capacity for ground heat exchangers. They found that the phase shift of the temperature response is a function of only the thermal diffusivity, while the amplitude is a function of both thermal diffusivity and conductivity.

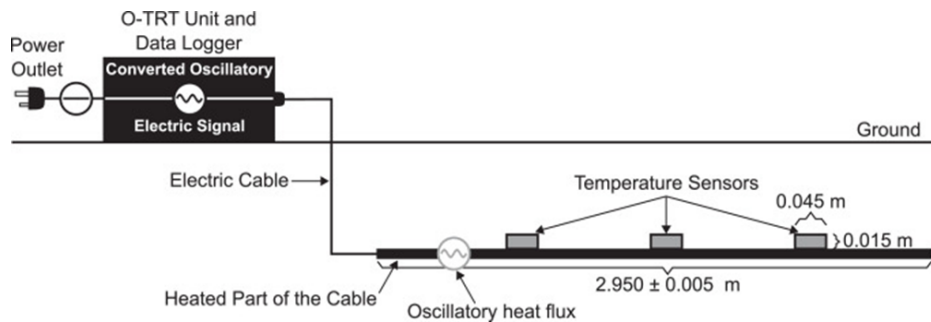


Figure 2.8: Oscillatory thermal response test diagram [45]

Langevin et al. [45] verified this analytical approach with experiments on shallow granular deposits. Their test setup, shown in Figure 2.8, produced ground diffusivity and conductivity values that matched the analytical results with less than 2 % deviation. An important conclusion from their tests was that longer oscillation periods, of a couple of hours, provide higher measurement accuracy at temperature sensors. Here the approach differs from the TWA method, which uses periods ranging from 2 to 100 seconds [37] only. This is because the test

articles and their sizes in TWA are much smaller compared to the scale of OTRTs.

Oscillatory heat loads offer significant advantages in thermo-physical property measurements. Their periodic application reduces the overall energy input required, making the method cost-effective. Additionally, the dynamic nature of oscillatory loads allows for the simultaneous measurement of multiple thermal properties, thereby reducing the test duration. While both amplitude gains and phase shifts in the temperature response can be used, the latter offers the advantage of reduced measurement errors. This approach can be extended to TMM correlations, making it valuable for thermal tests. The variation in thermal properties is often more pronounced under oscillatory conditions [33], making periodic loads particularly effective for capturing small changes that enhance model accuracy. A research gap has been identified where frequency domain thermal testing could be further explored for TMM correlations.

2.3. Research Objective and Research Questions

Ground testing for modal correlation is essential but often time-consuming and costly, particularly for unit tests in larger assemblies. Traditional methods rely on steady-state heat loads, which require lengthy stabilisation periods and high heat inputs. They are also prone to uncertainties such as undetected heat leaks that reduce the reliability of test data. To address these challenges, this research investigates the use of oscillatory heat loads as an alternative approach for unit-level thermal testing. The goal is to **reduce testing uncertainties by using sinusoidal heat loads to correlate TMMs based on the phase shift of temperature responses at critical locations**. This study aims to fill a key gap by applying transient thermal testing methodologies in the space sector, with the potential to enhance the accuracy and efficiency of early-stage thermal model correlations. The study is based on answering the following research question:

How can the phase shift between temperature responses due to oscillatory heat loads be used to correlate Thermal Mathematical Models (TMMs) and improve the reliability of the thermal test data?

To answer this main question, three sub-questions were formulated:

- Sub-Question 1 How sensitive are phase shifts in temperature responses to variations in thermal network parameters under oscillatory heat loads?
- Sub-Question 2 What is the impact of the amplitude and oscillation frequency of the heat load on the phase shift in temperature responses?
- Sub-Question 3 How do variations in test instrumentation and environmental conditions influence the phase shifts between periodic temperature responses?

3

Frequency Domain Thermal Analysis

Understanding the temperature response to oscillatory loads is crucial before designing a test using them. Oscillatory loads, by definition, involve heat inputs that vary with time, making it a transient heat transfer analysis, where the temperature at each location changes with time and position.

In this section, heat transfer and thermal analysis under oscillatory heat loads are examined. The sensitivity of a TMM to these oscillatory heat loads is then discussed, along with its potential application for model correlation. Finally, the steps necessary to design a test procedure for thermal testing and model correlation are summarised.

3.1. Thermal Response to Oscillatory Heat Loads

Consider a simple case of heat conduction through a semi-infinite¹ body with an oscillatory heat load at the boundary surface, as illustrated in Figure 3.1.

The general heat conduction equation for this one-dimensional system is expressed by Equation 3.1. The solution of this equation gives the temperature distribution $T(x, t)$ along the bar at different time intervals.

$$\frac{\partial T}{\partial x} = \alpha \frac{\partial^2 T}{\partial t^2} \quad (3.1)$$

where,

T is the temperature

t is time

x is the distance along the length of the bar

$\alpha (= k/\rho c_p)$ is the thermal diffusivity of the material of the bar

Assuming a sinusoidal heat load, of the form shown in Equation 3.2, is applied at the boundary surface of the semi-infinite body,

$$q = q_o \sin(\omega t) \quad (3.2)$$

¹Semi-infinite bodies are an idealisation with only one boundary surface and the other sides extend to infinity [14]. They are used to analyse temperature responses due to the boundary surface conditions. For example, it can be used to analyse temperature variations near the surface of the Earth due to seasonal changes.

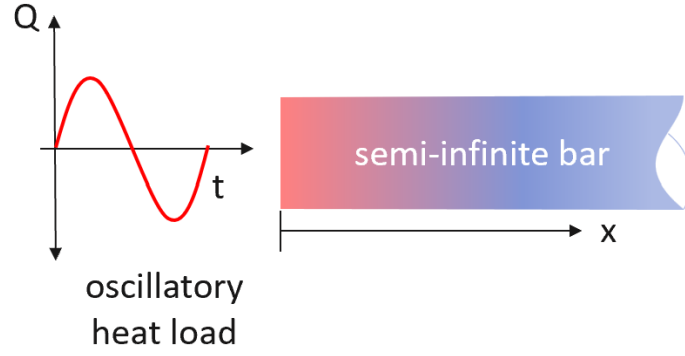


Figure 3.1: A semi-infinite bar with an oscillatory heat load at the boundary.

where,

q_o is the amplitude of oscillation

ω is the angular frequency of oscillation

and by applying the separation of variables method and Fourier series on Equation 3.1, the temperature distribution along the semi-infinite bar can be simplified to Equation 3.3 [48].

$$T(x, t) = T_o e^{-x\sqrt{\omega/2\alpha}} \sin\left(\omega t - x\sqrt{\frac{\omega}{2\alpha}}\right) \quad (3.3)$$

where T_o is a constant. This equation gives the temperature at any time instance and location along the bar from the boundary surface. Two important observations can be made from this equation :

1. The amplitude of the temperature response is a function of the frequency of oscillation, the thermal diffusivity of the material and the distance from the boundary surface.

$$T(x, t) \propto e^{-x\sqrt{\frac{\omega}{2\alpha}}}$$

Additionally, the amplitude reduces exponentially. Therefore, the further the distance from the boundary surface, the smaller the amplitude of the temperature response.

2. Similarly, the phase shift in the temperature response is also a function of the frequency of oscillation, the distance and the thermal diffusivity of the material.

$$\Delta\phi = x\sqrt{\frac{\omega}{2\alpha}} = x\sqrt{\frac{\omega}{2} \frac{\rho c_p}{k}}$$

So, at any distance from the boundary surface and with the knowledge of the oscillation frequency, a material's thermal diffusivity can be calculated from either the amplitude or the phase shift of the temperature response. This principle is used in the TWA methods to measure material thermal properties, as discussed in Section 2.2.1.

The objective of this thesis is not to characterise thermo-physical properties. However, these observations are important when designing the testing methodology for correlating a model. Both the phase shift and amplitude are functions of the material properties, but only the phase shift can be used at larger distances from the boundary surface. In a practical test setup, where an oscillatory heat load is applied to a space structure to collect temperature data, the location of sensors becomes very important. Sensors can not always be close to the heat source due to the convoluted shapes of the structures. Therefore, if the location of the sensor is far from the heat source, the amplitudes of the temperature response can be small.

More importantly, the amplitudes are sensitive to the accuracy of the temperature sensors and the quantity of heat flowing through the test article. Inaccuracies in these can produce amplitude offsets, which would affect the certainty of the correlation process. The phase shifts between the nodes, on the other hand, are theoretically insensitive to these errors. As the aim is to reduce the uncertainties from the model, only the phase shift of the temperature response will be considered for the correlation process.

To validate and further extend the understanding of heat transfer under oscillatory loads to its application for TMM correlation, thermal networks were analysed in the frequency domain.

3.2. Frequency Response of Thermal Networks

Thermal mathematical models are a network of thermal nodes that represent the structure's geometry. In traditional time-domain thermal network formulation, heat transfer in a structure takes the form of Equation 2.1. For a thermal network of n diffusive nodes and k boundary nodes ($m = n + k$, total nodes), the ODE for the i^{th} node can be written as Equation 3.4.

$$C_i \frac{dT_i}{dt} = Q_i + \sum_{j=1}^m GL(i,j) (T_j - T_i) + \sigma \sum_{j=1}^m GR(i,j) (T_j^4 - T_i^4) \quad \text{for } i = 1, \dots, n \quad (3.4)$$

With specified boundary conditions, various methods are available to solve the above non-linear equation in the time domain. However, mathematical manipulations are required to solve it in the frequency domain. Hailer et al. [49] proposed simplifying the system by approximating the model as a linear system near its equilibrium temperature. Their proposed assumption allows the linearisation of the radiative terms and represents the ODE in the standard time-invariant state space form. Laplace transformation of the linearised equation gives the transfer function that stores the phase shift and amplitude gain values of the nodes. Their methodology is explained in more detail in Appendix A.1. So, for a transfer function given by $G_{ij}(s) = x + iy$, its gain and phase shift are given by its absolute value (Equation 3.5) and argument (Equation 3.6), respectively.

$$abs(x + iy) = A = \sqrt{x^2 + y^2} \quad (3.5)$$

$$arg(x + iy) = \Delta\phi = \tan^{-1} \left(\frac{y}{x} \right) \quad (3.6)$$

Using the above results from Hailer et al.'s [49] study, the phase shift and amplitude gain between two adjacent nodes, due to oscillatory heat load on one of the nodes, were assessed.

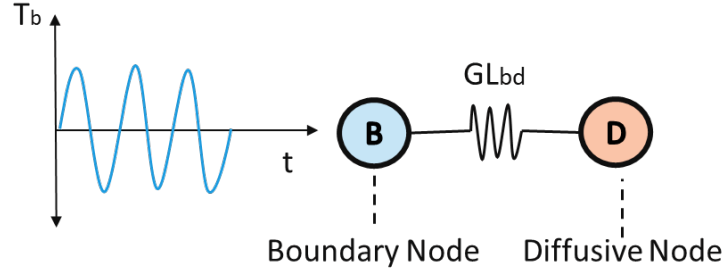


Figure 3.2: A purely conductive two-node thermal network

Assume a one-dimensional thermal network of two nodes, one diffusive and the other a boundary node. The gain and phase relation between these two nodes was derived. Figure 3.2 illustrates this proposed network. Only a conductive link between the nodes was modelled, making it a purely conductive heat transfer analysis. The gain and phase shift between them were found to be:

$$A = \frac{GL_{bd}}{\omega C_d} \frac{1}{\sqrt{1 + (GL_{bd}/\omega C_d)^2}} \quad (3.7)$$

$$\Delta\phi = \tan^{-1} \left(\frac{\omega C_d}{GL_{bd}} \right) \quad (3.8)$$

where,

- A = gain in the temperature response
- $\Delta\phi$ = phase shift between the nodes
- ω = angular frequency of oscillation
- C_d = thermal capacitance of the diffusive node
- GL_{bd} = conductive link between the two nodes

The derivation for these equations is detailed in Appendix A.2. Equations 3.7 and 3.8 provide the amplitude gain and phase shift of the diffusive node with respect to the boundary node, respectively. These equations illustrate the dependence of frequency response parameters on the oscillation frequency, the volumetric heat capacity of the diffusive node, and the conductive link between the nodes. These are similar to the parameters discussed in the previous section. Since the frequency of oscillation will be a known parameter, the thermal capacitance of the nodes and the conductive links between them can be correlated using the phase shift between the temperature responses. As discussed before, the conductive link can be represented as Equation 2.4 or Equation 2.5. So, the correlation will involve reducing the uncertainties in the thermophysical properties of the thermal network via the specific heat of the material (c_p), the conductivity of the material (k) or the thermal contact conductance (h_c) at interfaces or contacts.

To further explore the potential application of oscillatory loads in a TMM, a dummy model with four nodes was created. The purpose of this model was to study the effectiveness and feasibility of achieving successful correlation based on the phase shift of the temperature response.

3.3. Sensitivity Analysis Using a Simplified Thermal Model

The effects of the frequency of oscillation, the thermophysical properties of the material, and the nodal locations relative to the heat source on the observed phase shifts were studied in this exercise. The results were used to define guidelines that could be followed to design a test campaign using oscillatory loads. The trends found were useful for defining a model correlation strategy. The important questions addressed here are iterations of the sub-questions defined in Section 2.3:

1. What frequency of oscillation should be selected to get measurable temperature responses and phase shifts between them?
2. What is the sensitivity of the phase shift between temperature responses to different parameters of a thermal network?
3. What are the considerations for instrumentation placement in the test setup to ensure high phase shift measurability?

It should be noted that a “measurable” phase shift can be defined as a phase shift that can be detected and quantified using generic space-grade sensors and other measurement techniques. Quantifying this value is difficult as it will depend on the specific test unit and, therefore, may vary depending on the complexity of the test article.

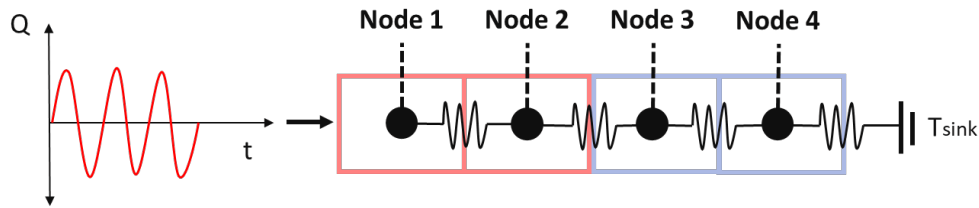


Figure 3.3: Thermal network of two bars in contact

The dummy model used for this study was an idealisation of two 10 cm long bars with a cross-sectional area of 2 cm^2 , in contact with each other. Figure 3.3 illustrates the model. The bars were made of two different materials, the thermo-physical properties of which are listed in Table 3.1. Since the bars were in contact, a thermal contact conductance of $500 \text{ W/m}^2\text{K}$ was assigned between them. A periodic heat load was applied at node 1, and the last node, i.e. node 4, was connected to a fixed temperature boundary. The initial and boundary temperatures were assumed to be 20°C .

Table 3.1: Dummy model material properties

Material	Node Nos.	$k [\text{W/mK}]$	$c_p [\text{J/kgK}]$	$\rho [\text{kg/m}^3]$
A	1, 2	211.6	921.6	2770.0
B	3, 4	391.2	385.2	8860.0

3.3.1. Evaluating Frequency of Oscillation

The phase shift in the temperature response is directly proportional to the oscillation frequency, f , therefore it is an important parameter to choose for the test.

$$\Delta\phi \propto \tan^{-1}(\omega) = \tan^{-1}(2\pi f)$$

Due to this direct proportionality, an increase in the oscillation frequency will also increase the phase shift between nodes. However, a very high frequency might not show any temperature response at all. Therefore, selecting an ideal frequency of oscillation is crucial. To examine the regions of maximum phase shift in this dummy model, a Bode plot was generated using ESATAN's frequency response library. These graphs are illustrated in Figure 3.4.

Bode plots illustrate the gain and phase of a system's response as a function of different frequencies [50]. In Figure 3.4, the plot shows the transient temperature response of the dummy model to unit heat perturbations at varying frequencies. The gain shows the magnitude of the temperature response of various nodes relative to the periodic heat perturbations, while the phase shift shows the delay in temperature response, in degrees, with respect to the sinusoidal heat input.

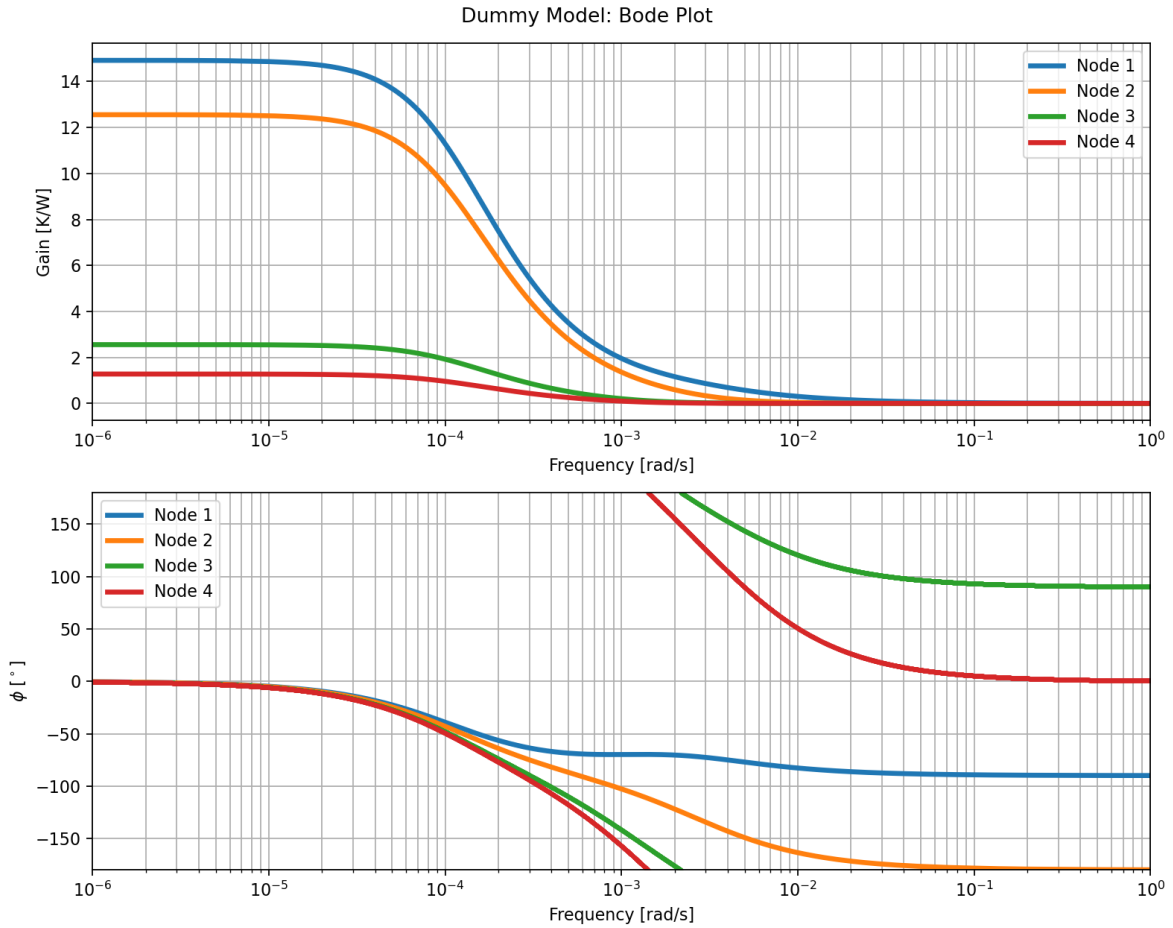


Figure 3.4: Bode plot for the dummy model

At lower frequencies, significant amplitude gains are observed, but the phase delay in the temperature responses is 0° . Since the focus is on using phase shifts for correlation, such frequencies cannot be considered. Additionally, lower frequencies, which are equivalent to periods of 100,000 seconds (≈ 28 hours), would defeat the purpose of making the test quick and cost-effective. Therefore, these lower frequencies are not of significance. Alternatively, higher frequencies, despite being quicker and having larger phase shifts, show no amplitude gains, indicating that the phase shift cannot be measured using standard temperature sensors. Therefore, it is important to select a frequency that provides some amplitude gain and phase shift in the temperature responses.

From the plots in Figure 3.4, it can be visually approximated that the ideal range of frequencies for the best measurability for this dummy model is between 2×10^{-4} rad/s and 4×10^{-3} rad/s, as these frequencies provide considerable phase shift in the nodes along with some amplitude gains. These frequencies correspond to oscillation periods of 5000 seconds (approximately 1.38 hours) and 1000 seconds (16.67 minutes) respectively. Further evaluations were conducted at frequencies within this range.

The phase shifts in Figure 3.4 are all relative to the sinusoidal heat input. However, for correlating the mathematical models, it is more practical to focus on the phase shift between two nodes. The phase shift in this context is governed by the conductive link and thermal capacitance between the nodes, as described in Equation 3.8. These parameters can then be adjusted by comparing the predicted phase shifts with the measured ones, which can facilitate more accurate model correlation. The phase shift between three nodal pairs was therefore compared.

- Node Pair 1-2 and 3-4: representing the phase shift within nodes linked by a conductive link in a continuous medium (represented by Equation 2.4).
- Node Pair 2-3: representing the phase shift between nodes linked by a conductive link between dissimilar materials or contacts (represented by Equation 2.5).
- Node Pair 1-4: representing the phase shift between nodes with both types of conductive links in series

The phase shifts between these pairs were assessed to understand the ideal sensor pair locations to observe during a test. This comparison is shown in Figure 3.5. The phases were calculated using the Fast Fourier Transform of the temperature responses of the nodes. They were evaluated in both degrees and seconds, using Equation 3.9.

$$\Delta t = \frac{\Delta \phi}{2\pi f} \quad [s] \quad (3.9)$$

A negative phase shift indicates a lag in the temperature response, which is why Figure 3.5 shows negative phase shift values when expressed in degrees. However, when expressed in seconds, the delays between temperature responses are positive, as they represent the actual time difference. Additionally, expressing the phase shift in seconds highlights more distinct differences between frequencies, making the data easier to interpret. Furthermore, since the oscillation frequency of 4×10^{-4} Hz gives the highest delays between node pairs, thermal parameter sensitivity was evaluated at this frequency using the time delay.

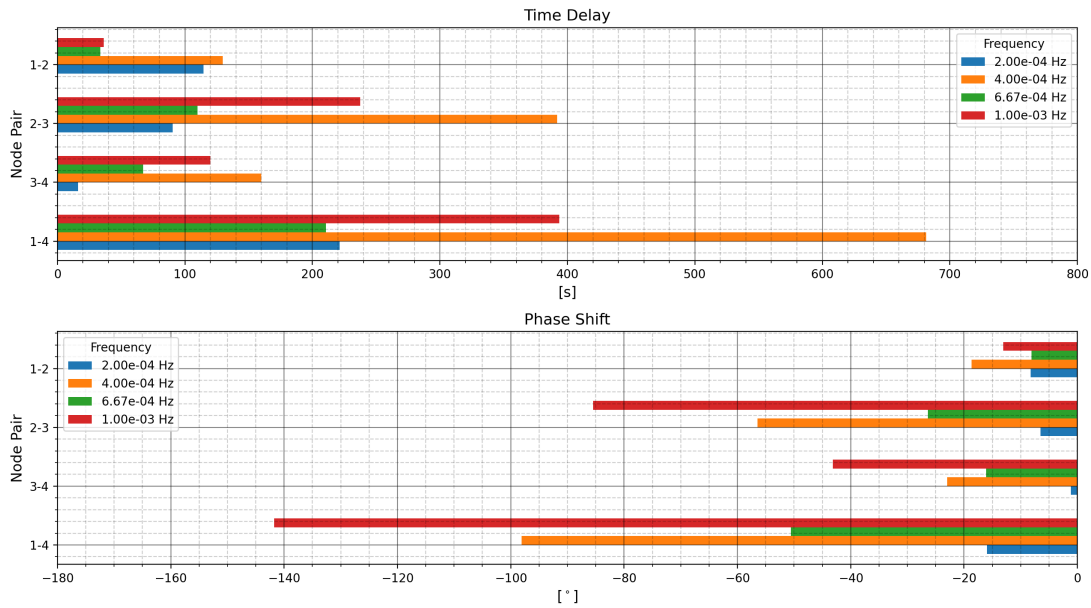


Figure 3.5: Comparing phase shift at different frequencies

3.3.2. Evaluating Sensitivity to Thermal Parameters

The thermal capacitance and conductive links between the nodes also influence the phase shift between them. Therefore, their influence needs to be evaluated. The assessment was conducted at a frequency of 4×10^{-4} Hz, corresponding to a period of approximately 2500 seconds or 42 minutes. Figure 3.6 shows the transient temperature responses of the four nodes over 5 hours. These were used to evaluate the phase shift between the Node Pairs.

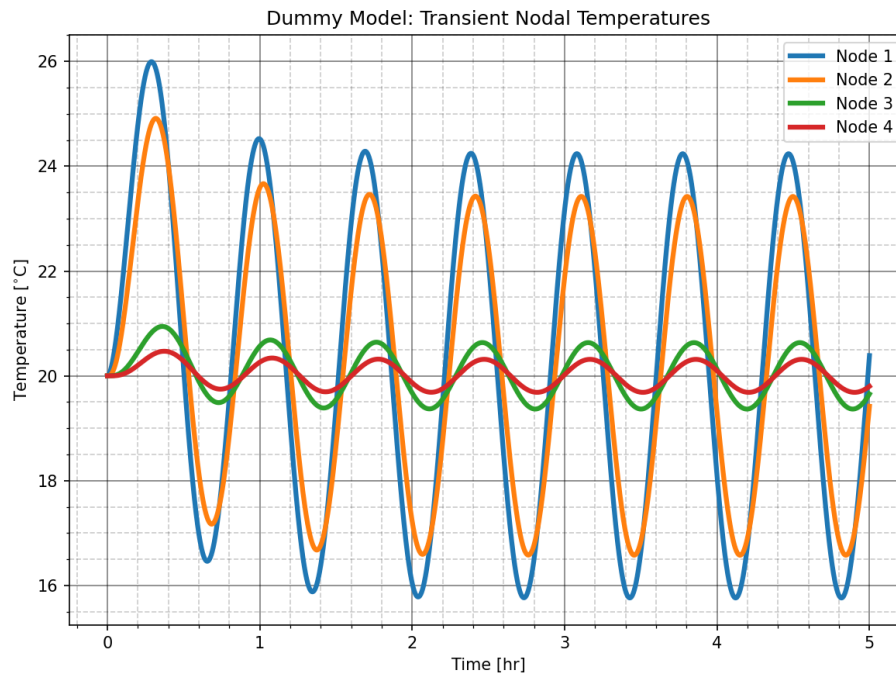


Figure 3.6: Transient nodal temperature

The sensitivity of phase shifts, measured in seconds, to variations in the parameters, was evaluated relative to the baseline model shown above. It is important to note that while all the differences in the phase shifts seem minor compared to the 2500 second cycle period, the correlation methodology relies on the relative phase shift between temperature responses. Therefore, the changes in the phases were assessed relative to the baseline model.

Conductive Link

From the thermal network in the frequency domain, it is known that the phase shift between two nodes is inversely proportional to the conductive link between them (Equation 3.10). An increase in conductance implies improved conduction through components, leading to greater heat flow and smaller phase shifts. Conversely, a decrease in conductance results in slower heat flow through the nodes, and therefore, a larger phase shift will be observed. This relationship has been validated in this section.

$$\Delta\phi \propto \tan^{-1} \frac{1}{GL_{bd}} \quad (3.10)$$

As discussed in the literature, this conductive link takes two forms, depending on whether the nodes represent a contact between two similar/dissimilar materials or a continuous solid medium. Uncertainties in the conductive link arise from uncertainties in the thermal contact conductance (h_c) or the thermal conductivity (k) of the material. The influence of both parameters on the phase shift between nodes will be assessed below.

The conductive link between Nodes 2 and 3, which represents a contact, is described by Equation 2.5. In a TMM, the thermal contact conductance typically carries an uncertainty of $\pm 50\%$. Therefore, the conductive link between Nodes 2 and 3 was varied by a factor of 0.5 and 1.5. The absolute differences in time delay relative to the baseline model are summarised in Table 3.2. Figure 3.7 shows the resulting percentage changes in time delay. This analysis provided insights into trends that could guide correlation practices.

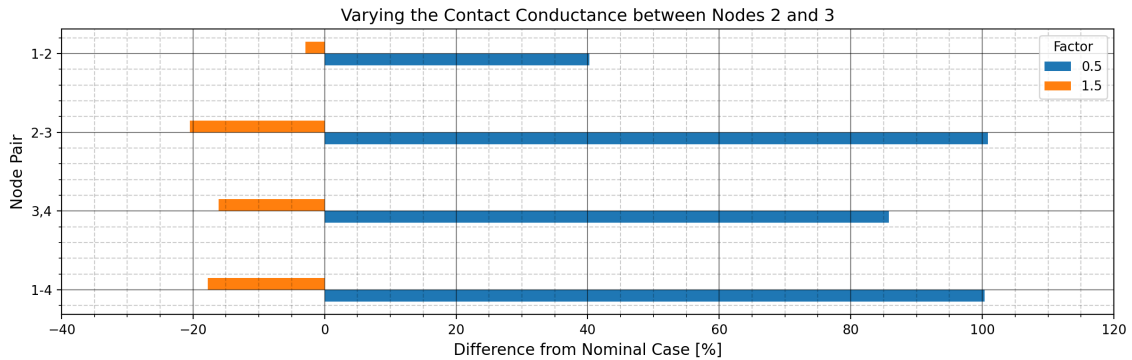


Figure 3.7: Relative percentage changes in time delay compared to the nominal case on varying the contact conductance

A reduction in contact conductance between Nodes 2 and 3 slows the heat flow between them, resulting in a 100% relative increase in delay, corresponding to a 90 second absolute increase from the nominal value. This also accounts for the 85% increase in delay observed

at Node Pair 3-4 and the 100% increase at Node Pair 1-4, as the heat flow between these nodes is also impeded. In contrast, the heat flow at Node Pair 1-2 remains largely unaffected, so, a relative increase in time delay of 40%, which is only a 12 second absolute difference, was observed. Increasing conductance accelerates heat flow, thus reducing the phase shift between nodes and leading to a negative change. The relative increase in delay for Node Pair 2-3 remains the most prominent, however, in both cases, the largest absolute delay is observed between Node Pair 1-4, as these nodes are the furthest apart, leading to a larger delay between their responses.

Table 3.2: Absolute differences in ϕ on varying contact conductance compared to the nominal case

Node Pair	Varying $GL_{2,3}$	
	$\phi_{0.5} - \phi_{1.0}$	$\phi_{1.5} - \phi_{1.0}$
1-2	12.24 s	-00.88 s
2-3	93.22 s	-18.91 s
3-4	52.60 s	-08.22 s
1-4	158.06 s	-28.01 s

These observations indicate the trends that can be used to adjust the contact conductance parameters during correlation. If the predicted phase shifts exceed the measured values, the conductance parameter should be decreased; otherwise, it should be increased.

Similar trends were observed when varying the conductance between Node Pairs 1-2 and 3-4, which represent the conductive link within solid materials as described by Equation 2.4.

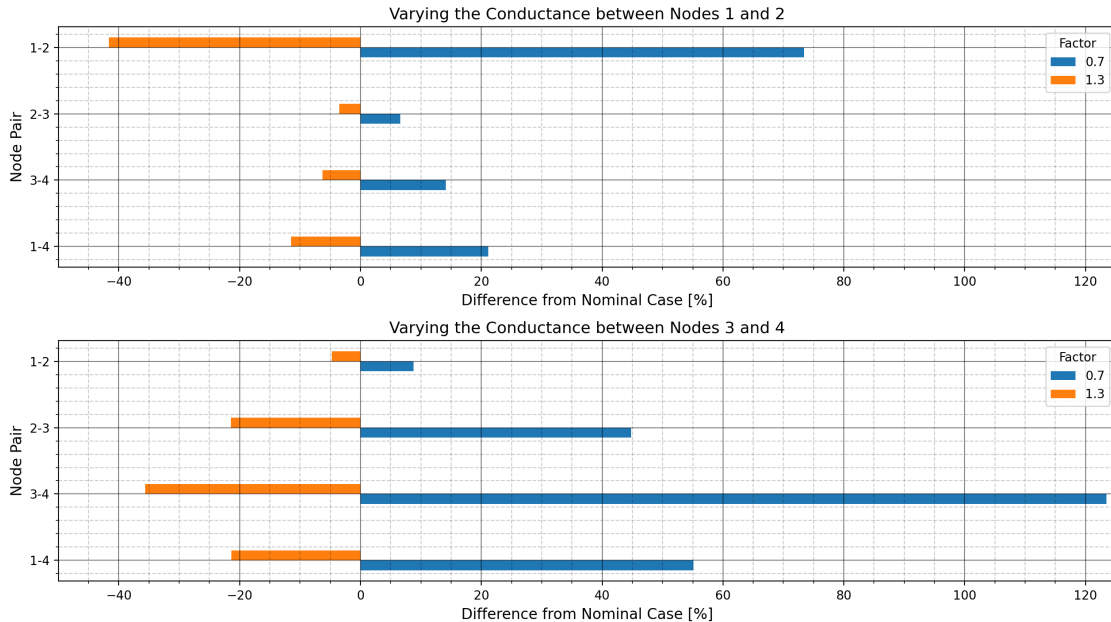


Figure 3.8: Relative percentage changes in time delay compared to the nominal case on varying the thermal conductance

The general uncertainty in material conductivity, as specified by ECSS standards, is approximately $\pm 30\%$. These variations in the material conductivity were applied to the conductive links between the Node Pairs and the changes were evaluated relative to the baseline factor of 1.0. The resulting percentage differences in phase shifts due to these variations are illustrated in Figure 3.8, with the absolute differences also summarised in Table 3.3.

The results indicate that the conductance between Nodes 1 and 2 has a limited impact on other node pairs. A reduction in conductance between Nodes 1 and 2 led to a 70% relative increase in time delay between them. However, the impact on Node Pair 2-3 was minimal, showing only a 6% relative change, corresponding to a 6 second difference. Similarly, increasing the conductance has little effect on it, with only a 3% relative decrease observed. The influence on Node Pair 3-4 and 1-4 was also relatively minor, showing a 15-20% increase when conductance was reduced and a 5-10% decrease when it was increased.

Following similar trends, varying the conductance between Nodes 3 and 4 had the most significant impact on that specific node pair. Reducing the conductance by a factor of 0.7 led to a delay increase of over 120% at Node Pair 3-4. In contrast, the effects on Node Pairs 2-3 and 1-4 were 45% and 55%, respectively. Conversely, increasing the conductance resulted in a 20% decrease in delay for both node pairs. The influence on Node Pair 1-2 was minimal, showing less than a 10% increase and a 5% decrease. As shown in Table 3.3 the largest absolute changes occurred at Node Pair 1-4, which is expected, as the phase shift between Nodes 1 and 4 depends on the conductance between all the nodes in the model, making it sensitive to variations in any of the conductive links.

Table 3.3: Absolute differences in ϕ on varying conductance compared to the nominal case

Node Pair	Varying $GL_{1,2}$		Varying $GL_{3,4}$	
	$\phi_{0.7} - \phi_{1.0}$	$\phi_{1.3} - \phi_{1.0}$	$\phi_{0.7} - \phi_{1.0}$	$\phi_{1.3} - \phi_{1.0}$
1-2	22.32 s	-12.66 s	02.67 s	-01.45 s
2-3	06.07 s	-03.25 s	41.41 s	-19.83 s
3-4	04.89 s	-02.17 s	42.76 s	-12.33 s
1-4	33.28 s	-18.08 s	86.83 s	-33.61 s

The study indicates that variations in conductive links primarily have a localized impact on phase shifts, influencing the temperature responses of nodes directly connected by the link. The temperature responses of the other nodes in the thermal network experience comparatively smaller deviations from the nominal profiles. This could be useful when correlating conductive links in the TMM, allowing for adjustments to one link at a time with minimal influence on other node pairs.

Thermal Capacitance

The thermal capacitance of a node defines the amount of heat required to produce a unit change in its temperature. A decrease in thermal capacitance results in faster heating of the node, leading to a smaller phase shift. On the other hand, a higher thermal capacitance causes slower heating, resulting in a larger phase shift. Mathematically, the phase shift is

directly proportional to capacitance, as shown in Equation 3.11.

$$\Delta\phi \propto \tan^{-1}(C_d) = \tan^{-1}(\rho c_p) \quad (3.11)$$

The thermal capacitance in the model was varied by $\pm 25\%$, based on the ECSS standards of expected uncertainty in thermal capacitance. In the first case, the thermal capacitance of Nodes 1 and 2 was varied by a factor of 0.75 and 1.25, followed by varying the capacitance of Nodes 3 and 4 by the same factors. The relative change between the time delay in temperature responses due to these variations is shown in percentage in Figure 3.9, and the absolute differences are summarised in Table 3.4.

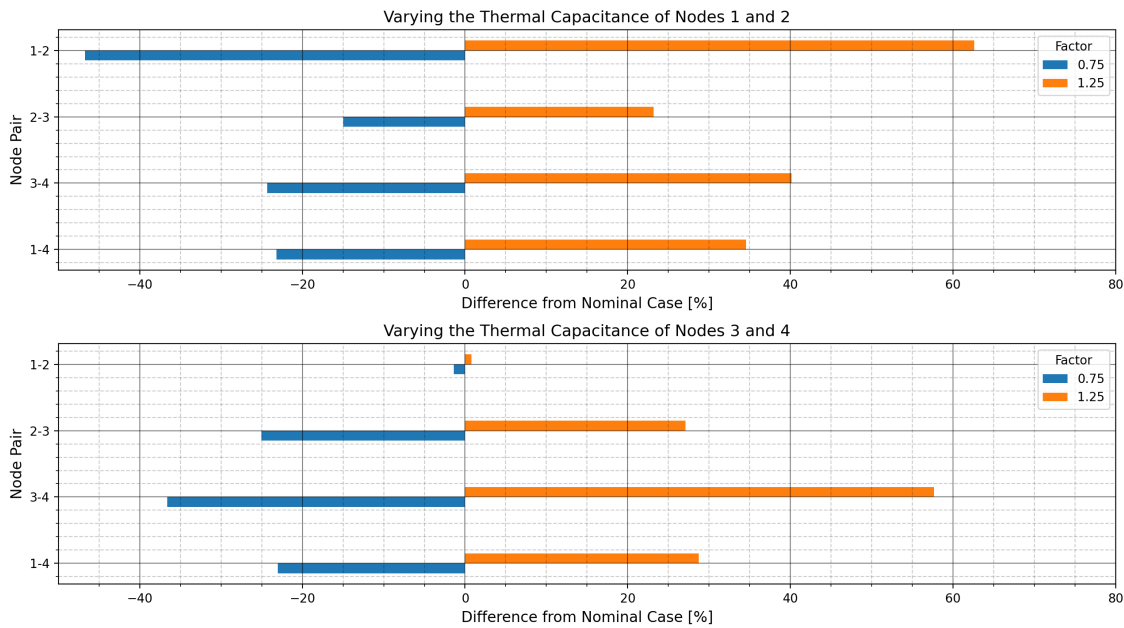


Figure 3.9: Relative percentage changes in time delay compared to the nominal case on varying the thermal capacitance

The trends observed here are opposite in sign to those seen when varying the thermal capacitance in Figures 3.7 and 3.8, indicating an opposite influence of conductance and capacitance on the difference in phases. This is consistent with the fundamental understanding of temperature response to oscillatory heat loads.

Similar to varying the conductance, varying the capacitance of Nodes 1 and 2 also impacts the delay between them. However, the effects on the other node pairs were slightly more significant in this case compared to before. All the other Node Pairs show almost a 20-40% increase on increasing the capacitance, which was less than 20% on varying the conductance. Similarly, on varying the thermal capacitance of Nodes 3 and 4, the most prominent change was observed in Node Pair 3-4, however, Node Pairs 2-3 and 1-4 had more than a 20% change as well. But the Node Pair 1-2 showed less than a 1% change, showing an absolute change of less than a second.

Table 3.4: Absolute differences in ϕ on varying thermal capacitance compared to the nominal case

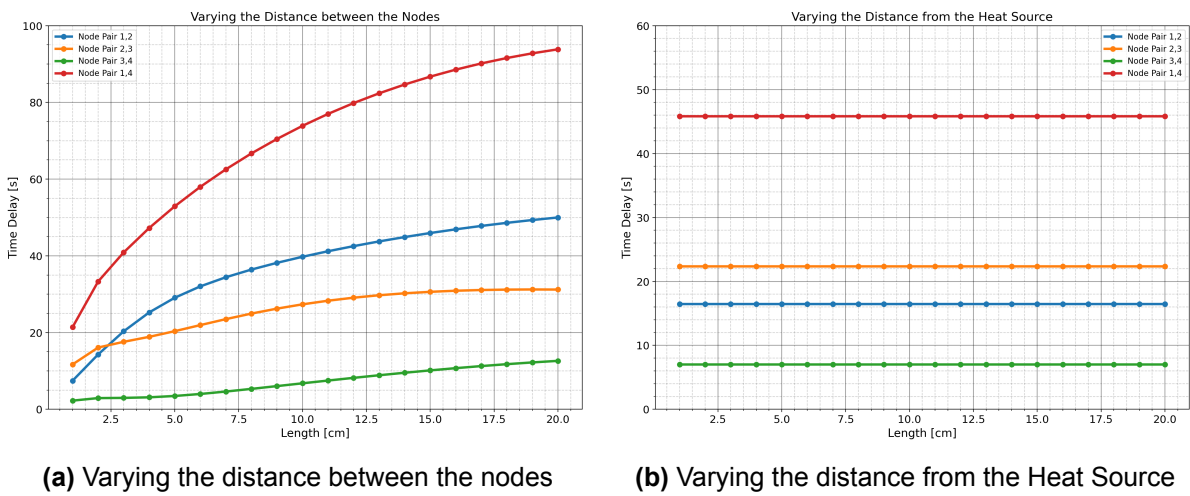
Node Pair	Varying C_1 & C_2		Varying C_3 & C_4	
	$\phi_{0.75} - \phi_{1.0}$	$\phi_{1.25} - \phi_{1.0}$	$\phi_{0.75} - \phi_{1.0}$	$\phi_{1.25} - \phi_{1.0}$
1-2	-14.22 s	19.04 s	-00.41 s	00.24 s
2-3	-13.83 s	21.46 s	-23.16 s	25.04 s
3-4	-08.43 s	13.90 s	-12.67 s	19.96 s
1-4	-36.48 s	54.40 s	-36.24 s	45.24 s

The analysis shows that on varying the capacitance of nodes, the phase shifts between all subsequent nodes along the heat flow path are affected. Therefore, if correlating the capacitance of a node, it is important to check the variations at all the nodes along the heat path. Additionally, consistent with earlier findings, the largest absolute differences were observed at Node Pair 1-4, making it the most sensitive Node Pair to changes in the thermal network.

3.3.3. Evaluating Instrumentation Placements

Studying the impact of the distance between the nodes can help determine the optimal heater and sensor placement for accurate correlation between test data and the TMM.

In the first case, the distance between the nodes was increased, which is equivalent to increasing the distance between the sensors. In doing so, the phase shift between the sensors also increases, as shown in Figure 3.10a. However, this increases the likelihood of accumulating uncertainties as additional conductive links are introduced between the two nodes. This was also evident in the absolute changes observed between Node Pair 1-4 before, where the nodes being the furthest apart, exhibited higher absolute changes in response to parameter variations compared to the other Node Pairs. To minimise uncertainties and capture the relationship between nodes accurately, it would be ideal to position sensors such that the conductive link between them is primarily influenced by the parameter being correlated.

**Figure 3.10:** Evaluating instrumentation placements by varying node distances and proximity to heat source

In the next case, the distance of the nodes from the heat source was varied. This is analogous to changing the location of the heater with fixed sensor locations. Figure 3.10b shows that the location of the heat source has little to no effect on the phase shift between the Node Pairs. This aligns with expectations, as phase shifts are measured based on temperature responses between two nodes, making them independent of the heat source's location. This is useful in a practical more industrial test setup, as existing heaters used for thermal control could be used to generate sinusoidal heat loads across the test unit without needing additional instrumentation. As long as temperature responses can be measured at the selected sensor locations, the heater's placement will not affect the phase shifts between the temperature responses.

3.4. Summarising Results

The results from this analysis can help with developing a testing methodology based on frequency domain testing. Key considerations for this include:

- The phase shift in temperature responses is influenced by the oscillation frequency of the heat load, so careful selection of the frequency is important.
 - The frequency should be chosen to ensure sufficient amplitude gain in the temperature responses, allowing the phase shift between nodes to be accurately measured.
 - The oscillation frequency should be low enough to produce noticeable temperature responses but not so low that the test takes too long. Therefore, a frequency analysis should be conducted to identify the optimal frequency range that satisfies both conditions.
- When preparing the test setup, attention should be given to the instrumentation.
 - The heater location is not critical, as long as adequate temperature responses are measured at all sensors to determine phase shifts between them.
 - However, sensors should be placed with care and as close as possible to the interface/parameter of interest. This ensures that in the test TMM, only a single conductive link is represented between them, that is the parameter being correlated.
- During the correlation process, careful consideration of the parameter being modified is necessary.
 - When adjusting the capacitance of nodes, the phase shifts across all node pairs from the modified node to the heat sink, along the heat flow path, will be impacted. Therefore the capacitance of the test unit should be modelled with care and correlated first.
 - When modifying the conductive links between nodes, the effects are more localised. As a result, they can be correlated one node pair at a time, without significantly affecting the phase shifts between temperature responses at other node pairs.

These results provide an answer for the Sub-Question 1, concerning the sensitivity of phase shifts to variations in thermal network parameters. Additionally, Sub-Question 2 and Sub-Question 3 have also partially been answered from these results.

Given these considerations, a test strategy for TMM correlation can be formulated. First, the ideal frequency of oscillation and heater power should be determined based on the preliminary TMM of the test setup. Next, the test setup should be instrumented, with heaters and sensors precisely positioned to ensure effective heat flow and accurate capture of critical interfaces for correlation. Testing can then be conducted at the selected frequency of oscillation, and the collected temperature data can be used for correlation, guided by the trends observed in the study above.

However, when working with real-life test data, post-processing is often necessary to filter out noise and address any discrepancies. Therefore, before fully defining and validating a testing and correlation strategy based in the frequency domain, it was important to conduct initial tests. Through these preliminary tests, the extent of post-processing required was understood and it helped gain insights into effectively handling and interpreting test data.

4

Post-Processing and Data Analysis Methods for Thermal Testing

A simple test campaign was carried out to assess the practical challenges associated with actual test data. The objective was to generate temperature responses in the test setup, shown in Figure 4.1, due to oscillatory heat loads and establish a foundation for post-processing methods before determining the phase shift between the responses. The experimental setup consists of a tin-coated copper braided strap bolted onto two aluminium, AL EN-AW-1050A, plates. The aluminium plates were 200 mm long, 100 mm wide, and 2 mm thick. The length, width, and average thickness of the strap were 152 mm, 25.4 mm, and 3 mm, respectively.

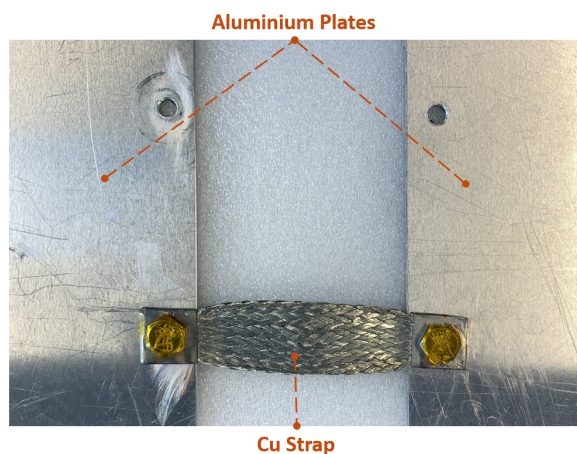


Figure 4.1: Simple test setup with copper strap

Therefore, the tests are conducted in a laboratory environment. This uncontrolled environment does not meet space industry standards, but, the tests helped establish useful post-processing practices. To gain full confidence in the testing methodology, it is crucial to compare it with existing methods and perform tests in a vacuum environment, which will be discussed in the next chapter.

In this chapter, the modelling of the TMM is discussed, followed by an examination of the selection of optimal heater power and oscillation period. The instrumentation of the test setup is then addressed, including the heater and temperature sensors that were used and their placement. Subsequently, the post-processing methodologies are outlined, along with the methods for measuring phase shifts. It is important to acknowledge that these are tests intended to build on and improve the understanding of the heat transfer under oscillatory heat loads compared to the simulations and theoretical expectations.

4.1. Developing the Thermal Mathematical Model

Based on the test setup, a Thermal Mathematical Model (TMM) was created using ESATAN Workbench. All components were modelled as primitive shell elements, as the thickness of the components was significantly smaller than their length or breadth. This approach also reduced the geometric and computational complexity of the model, enabling faster simulations. To capture the heat flow across the strap more accurately, it was modelled using three elements: two representing the fixtures and one representing the braided part of the strap. The holes on the aluminium plates and the bolts on the strap were not included in the idealisation of the setup, following general practice. Figure 4.2 shows the top view of the model.

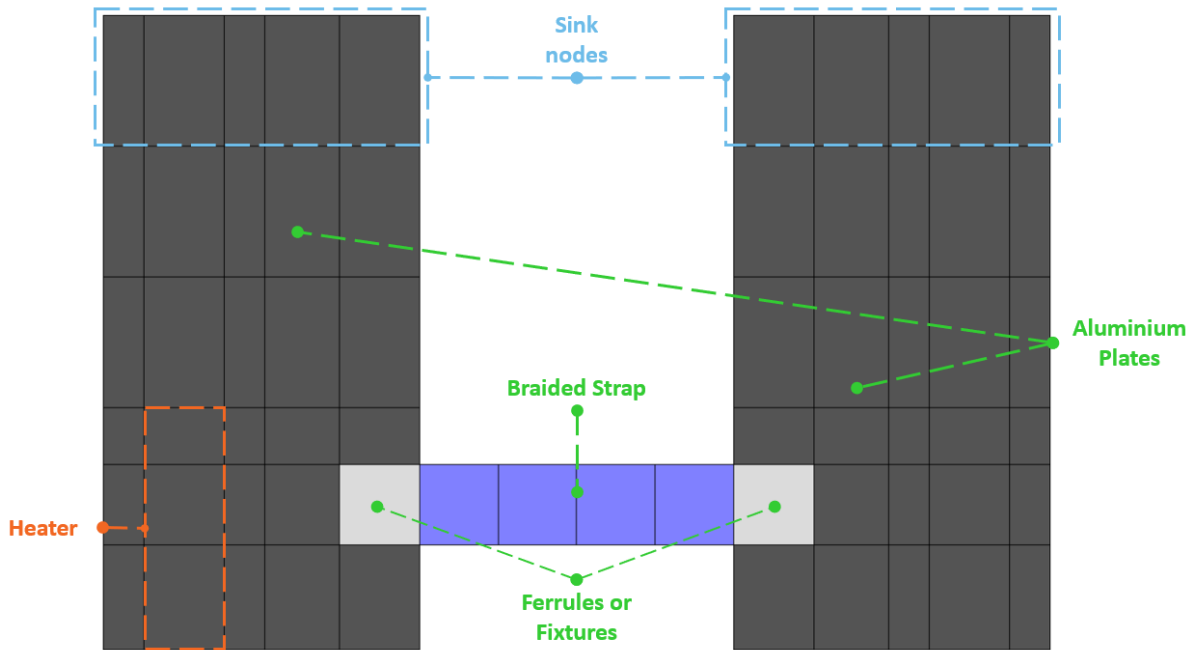


Figure 4.2: Preliminary TMM of the test setup

The aluminium plates were assigned the thermo-physical properties of AL EN-AW-1050A, while the strap and fixtures were assigned the properties of tin-coated copper. Table 4.1 lists these material properties.

Table 4.1: Material properties used in the TMM

	Density [kg/m ³]	Thermal Conductivity [W/mK]	Specific Heat [J/kgK]
AL EN-AW-1050A	2700.00	229.00	901.00
Tin-Copper	8912.93	387.00	385.19

Contact zones were defined at both fixture elements, mimicking the strap bolted to the aluminium plates, and corresponding to the conductive link representing a contact. A preliminary value of 500 W/m²K was assigned as the thermal contact coefficient for both interfaces. The contact area and other conductive parameters within the components were generated using ESATAN. Only conductive links were generated, as no radiation was considered in the simulation. Two boundary conditions were applied before solving the TMM. First, a heat load

was applied at the nodes “Heater”, also highlighted in Figure 4.2. Next, a fixed temperature boundary condition was applied at the top nodes of the aluminium plates, acting as the heat sink nodes. For an initial estimation, modelling the top nodes as the sink was considered an effective approximation of convection to air, which would be the primary heat dissipation mechanism in laboratory conditions.

A transient analysis was conducted on this model and the temperature responses at four nodes will be assessed: the Hot Plate, Hot Fixture, Cold Fixture, and Cold Plate. These correspond to the nodes on the aluminium plate and the strap’s fixture where the strap is attached to the plate. This schematic is also illustrated in Figure 4.3 which highlights the nodes of interest. In the figure, A and C refer to the contact between the two aluminium plates and the strap on either end and B refers to the conductance through the braided strap itself. The phase shift will be assessed at these interfaces. However, first, it is important to evaluate the required frequency of oscillation and the power profile of the heat load.

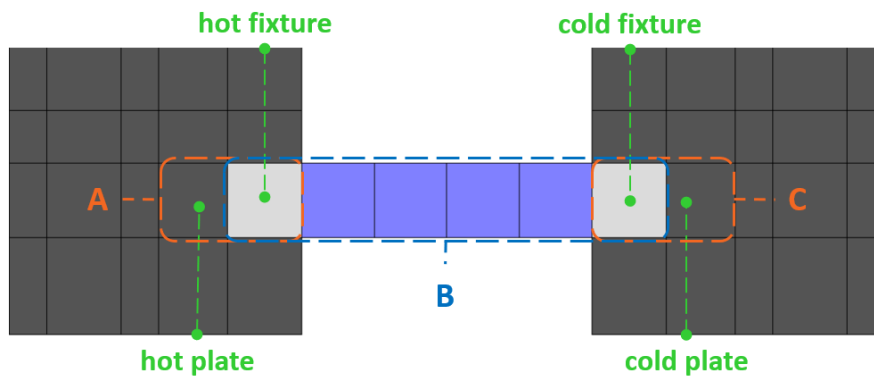


Figure 4.3: Laboratory test: Nodal nomenclature

4.1.1. Determining Testing Frequency and Heater Power Profile

To identify the frequency range that meets the criteria discussed in Section 3.3.1, a Bode plot was generated for the TMM using ESATAN's thermal frequency response function. From the plot shown in Figure 4.4, the suitable frequency range is determined to be between 2×10^{-4} Hz and 2×10^{-3} Hz, corresponding to periods of 5,000 seconds (1.38 hours) and 500 seconds (8.33 minutes), respectively. A heater power range of 1W to 6W was selected to ensure the voltages were within the operating limits of the test heater. Additionally, since this model is intended for laboratory tests, it was important to ensure that:

- the absolute temperatures in the model remained close to ambient conditions, thereby minimising the effects of convection in the uncontrolled laboratory environment
- the temperature response at the “cold” plate node should be at least in the order of 2°C (peak-to-peak) to ensure that the phase shifts in the test setup could be accurately measured

This threshold was established to account for standard temperature sensor inaccuracy of $\pm 1^{\circ}\text{C}$. It should also be noted that the amplitude of the temperature response is not the primary focus; however, this threshold was set to ensure that the oscillatory behaviour is clearly observable.

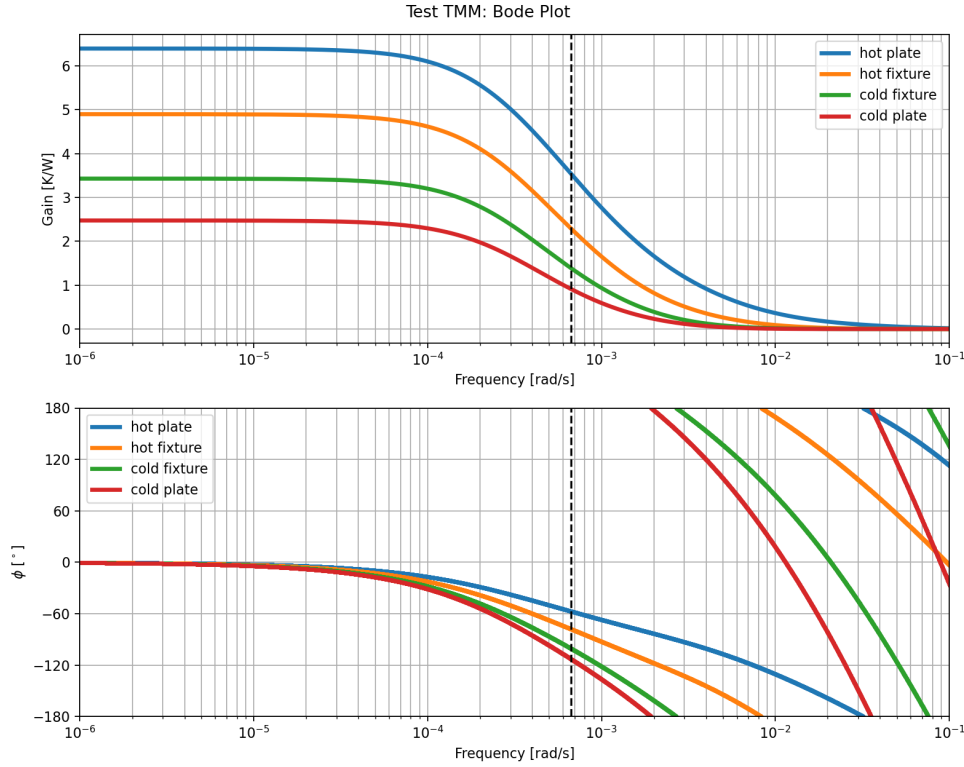


Figure 4.4: Bode plot and the selected frequency of 6.67×10^{-4} Hz

Multiple simulations were conducted to identify the frequency and heater power that best met the criteria. A frequency of 6.67×10^{-4} Hz, corresponding to a period of 25 minutes, and an average heater power of 3 W was selected. A sinusoidal heater profile, as described by Equation 4.1 was chosen.

$$q = q_o + q_o \sin \omega t \quad (4.1)$$

The heater power profile for the simulation duration of 2.5 hours is illustrated in Figure 4.5a.

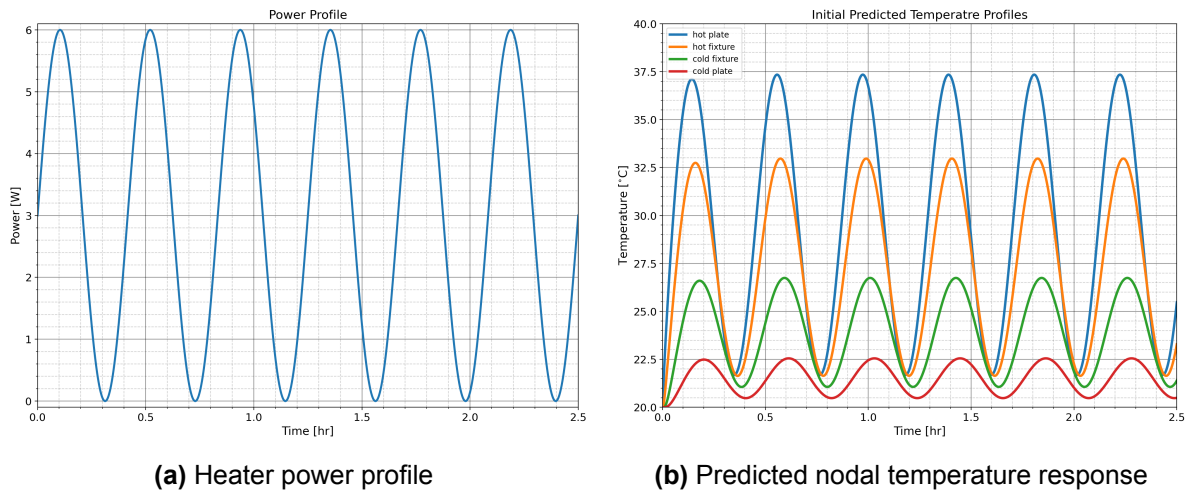


Figure 4.5: Input power profile and the resulting temperature responses

Figure 4.5b illustrates the predicted temperature profile of the nodes of interest, annotated in Figure 4.3, for the simulation duration. Once the thermal capacitance is accurately modelled, the phase shifts at the contact interfaces would correspond directly to the conductive properties of the system. The phase shift between the temperature responses of the “hot plate” and “hot fixture” at interface A, and between the “cold plate” and “cold fixture” at interface C, would represent the contact conductance between the Cu braided strap and the aluminium plates. Additionally, the phase shift between the hot and cold fixtures would represent the conductance through the strap itself. By correlating these three interface conductances the model can more accurately reflect the heat flow behaviour through the setup.

4.2. Test Setup and Instrumentation

Figure 4.1 shows the aluminium plates and Cu braided strap used in this test. A heater was mounted on one of the plates to simulate oscillatory heat loads. To measure the temperature response to these loads, temperature sensors were mounted at the interfaces, closely aligned with the nodes shown in Figure 4.2. The hardware used in the test setup is discussed below. Appendix B also provides a summary of the hardware components used and their purpose in these tests.

4.2.1. Heater: Polyimide Flexible Heater

The heating element used in this setup is a flexible polyimide heater with polyimide strain relief. This heater, OMEGA PLMLV-103/5 shown in Figure 4.6, was selected for its thermal stability, lightweight design and ability to provide precise uniform heating across the surface. It has a high-temperature performance range of -57°C to 232°C and a quick response time, which was useful to simulate oscillatory heat load for each time step.

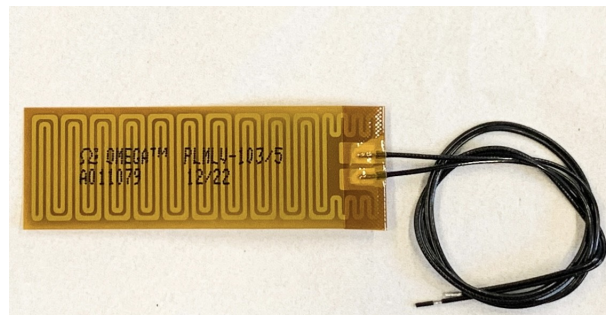


Figure 4.6: OMEGA PLMLV-103/5 flexible polyimide heater

The heater is 1 in (25.4 mm) wide and 3 in (76.2 mm) long with a watt density of 5 W/in^2 (0.008 W/mm^2). The heater has a resistance of 52.7Ω , and its maximum operating voltage was 28 V.

4.2.2. Temperature Sensors: Thermocouples and RTDs

Two types of temperature sensors were selected for measuring the temperature response: thermocouples (TCs) and resistance temperature detectors (RTDs). Although both are commonly used in thermal tests, they differ in operation, accuracy, and response time. Therefore, both were used to balance precision across different locations in the test setup.

Thermocouples (TCs)

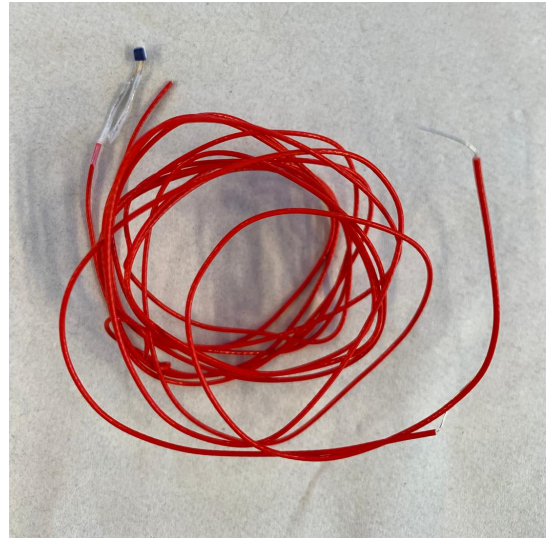
They operate on the principle of the Seebeck effect, where a voltage is generated at the junction of two dissimilar metals when exposed to a temperature gradient [51]. This voltage output is typically very small and must be converted to a temperature reading. The conversion pro-

cess is non-linear and complex [51]. Despite these challenges, thermocouples are widely used due to their simplicity, robustness, and suitability for a wide range of temperature measurements. Based on the metal compositions, different thermocouple types are available.

For this test, thermocouple type T was selected as it is commonly used in ground thermal tests. They consist of a positive leg made of copper and a negative leg made of constantan (a copper-nickel alloy). They offer a wide temperature range from -200°C to 350°C and provide the highest accuracy among the available variations, with a tolerance of $\pm 1^{\circ}\text{C}$ or $\pm 0.75\%$ [52]. The sensors were prepared by soldering the copper (brown) and constantan (white) wires together, as shown in Figure 4.7a, which shows the manufactured thermocouple.



(a) Thermocouple Type-T



(b) RTD - Pt100

Figure 4.7: Temperature sensors used in the test setup

Resistance Temperature Detectors (RTDs)

The electric resistance of certain metals changes predictably with temperature variations. This forms the operating principle of an RTD sensor [53]. RTD signals are processed using a quarter bridge configuration in a Wheatstone bridge, where resistance changes are converted into a measurable voltage that is then converted into temperature change [54]. RTDs are known for their high accuracy, typically $\pm 0.1^{\circ}\text{C}$, and for providing stable and repeatable measurements. However, they are more expensive than thermocouples, have a slightly slower response time, and are more fragile. Despite these drawbacks, RTDs were included in the test due to their higher accuracy, especially in measuring oscillatory responses far from the heat source.

The most commonly used RTD is the Pt100, a platinum-based sensor with a resistance of $100\ \Omega$ at 0°C . The platinum wire is wrapped around a ceramic core, which provides stability under extreme temperatures, offers good thermal insulation from ambient temperature changes, and protects the delicate platinum wires from damage [55]. This design ensures high accuracy, excellent stability and repeatability. Pt100 sensors are available in various configurations. Considering the application requirements and data acquisition limitations, a simple 2-wire configuration was selected. This configuration allowed for four Pt100 sensors to be placed on the test setup. Figure 4.7b shows the 2-wire Pt100 sensor used in the test.

4.2.3. Heater and Sensor Placement

To achieve a more linear and one-dimensional heat flow, all components were aligned along the length of the test setup, as shown in Figure 4.8.

1. **Heater:** The heater was mounted on one side of the strap using adhesive tape and positioned on the aluminium plate close to the strap. This ensured that most of the heat transferred from the heater to the strap while still leaving sufficient space for a temperature sensor to be placed near the interface between the strap and the aluminium plate.
2. **RTD:** The RTDs were strategically placed at key points where precise temperature measurements were critical, that is at the interfaces between the aluminium plates and the strap. A pair of Pt100 sensors were installed on the aluminium plate and at the bolt securing the strap on either side using Kapton tape. The bolts were chosen as sensor locations due to the ease of mounting. In total, four Pt100s were mounted.
3. **Thermocouples:** One type T thermocouple (TC-T) was mounted at the furthest point from the strap on the “cold” plate using Kapton tape to capture the linear heat flow across the whole setup. Additionally, two more thermocouples were installed on the strap’s fixtures as the temperature response at the bolts might not be the most representative of the heat flow through the strap. This is because, the bolts are made of galvanised steel, which is thermally much less conductive than the Cu braided strap. The thermocouples on the fixtures were secured using Kapton and Aluminium tapes to prevent sensor displacement. A total of three type T-TCs were used.

This configuration allowed for a comprehensive assessment of the temperature response at various locations on the test setup, ensuring that all critical points were monitored.

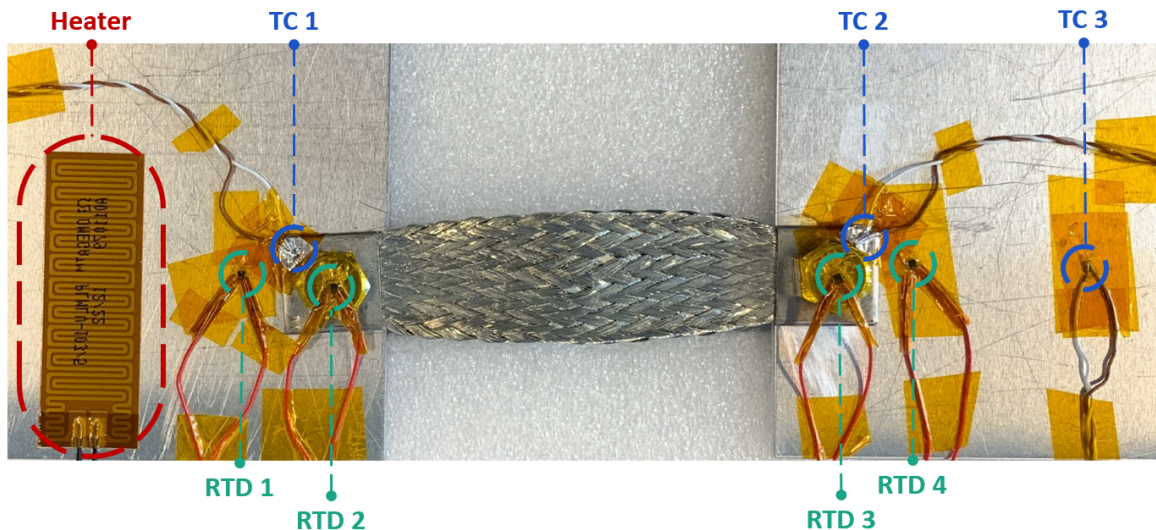


Figure 4.8: Test instrumentation and sensor nomenclature

4.2.4. Power Supply Unit and Heater Control

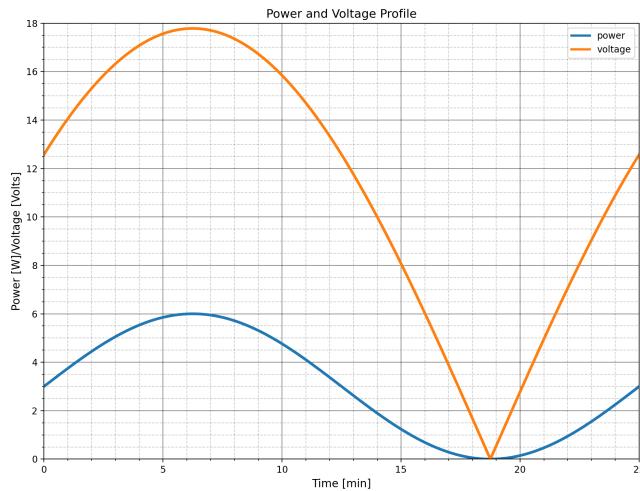


Figure 4.9: Power and voltage profile for one oscillation period.

The heater's power was controlled using a programmable power supply unit (PSU), model RSPD3303C, with a maximum output of 30V and 3A. The PSU was set to operate in voltage-controlled mode. The voltage required to achieve the desired power profile was calculated using the power-voltage-resistance relationship presented in Equation 4.2.

$$V = \sqrt{PR} \quad (4.2)$$

Given that the resistance of the PLMLV heater was 52.7Ω , the corresponding voltage profile was generated. Figure 4.9 shows the profiles for both, voltage and power, for one period of oscillation.

The voltage was remotely controlled and updated every second using a Python script.

4.3. Thermal Testing and Post-Processing Temperature Responses

As previously discussed, the tests were conducted in a laboratory environment to gather temperature responses to oscillatory heat loads. The heater was operated with a sinusoidal power profile of Equation 4.1 with an oscillation period of 1500 seconds. However, the test duration was extended to 3.5 hours, one hour longer than the simulated duration, to ensure that steady-state oscillations in the temperature responses were achieved. The test parameters are listed in Table 4.2.

Table 4.2: Test parameters - Laboratory environment tests

Test Parameter	
Oscillation Period	1500 s
Oscillation Frequency	6.67×10^{-4} Hz
Power Amplitude	3 W
Run Duration	3.5 hours
Initial Room Temperature	24.37°C

Temperature data from the sensors were recorded every second throughout the test to capture the thermal response with high precision. This high sampling rate was critical to ensure that the phase shift was captured precisely. The data acquisition was done using USB-Temp-AI and USB-Temp-TC for RTDs and thermocouples respectively. The voltage supplied to the heater was also monitored and stored using the USB-Temp-AI. This allowed for verification that the PSU delivered the correct voltage values and enabled adjustments for any discrepancies.

4.3.1. Raw Data and Initial Observation

The temperature data collected during the test provided an overview of the system's thermal response to the oscillatory heat loads. Initial observations indicated that the temperature responses closely followed the oscillation period of the applied heat load, with distinct phase differences observed between the various sensor readings. The raw temperature data from the seven sensors is displayed in Figure 4.10. The sensors, from near the heater to the “cold” aluminium plate, are labelled RTD 1 to RTD 4 and TC 1 to TC 3, as annotated in Figure 4.8. Initial observations of the temperature responses made were:

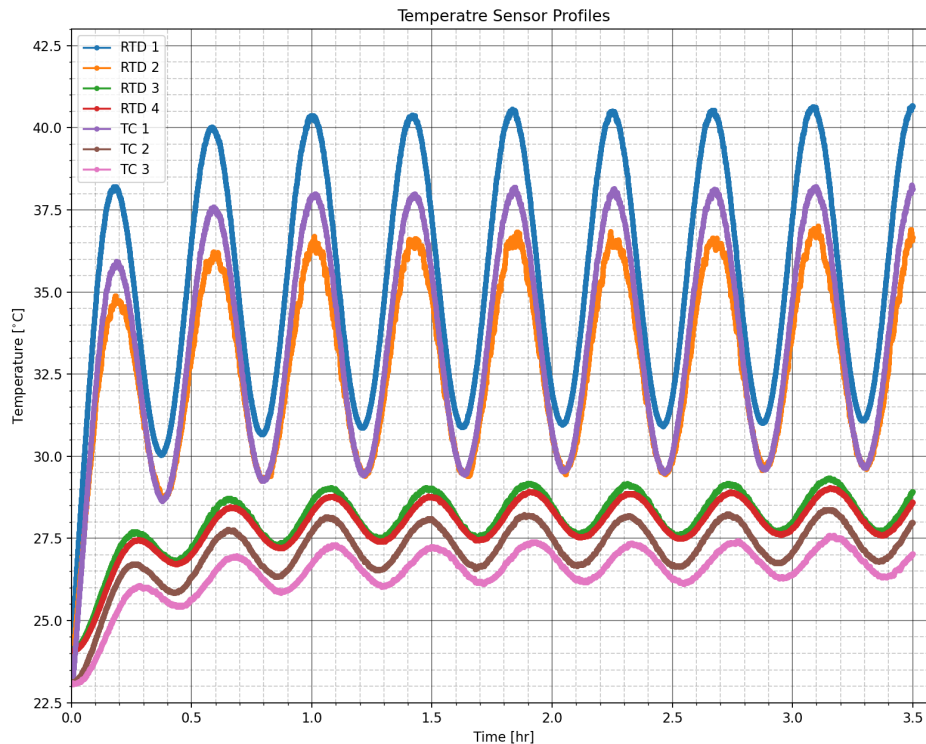


Figure 4.10: Raw temperature response of sensors under oscillatory heat loads

- RTD 1, which was closest to the heater, exhibits the highest temperature response amplitude and peak values of 2.5°C higher than the predicted temperature, shown in Figure 4.5b.
- The smallest amplitude gain is noticed in TC 3, which is the furthest apart from the heater, showing a peak value difference of 1°C less than the predicted temperatures. The peak-to-peak temperature at this TC is also less than 2°C.
- There is a significant drop in the amplitude between the sensors on the hot side (RTD 1, RTD 2, TC 1) compared to those on the cold side (RTD 3, RTD 4, TC 2, TC 3).
- An approximate 200-second delay between RTD 2 and RTD 3 was noted, potentially indicating either a delay in temperature propagation across the braided strap, possible heat losses through convection, or both.
- A slight difference in temperature response was noted between the RTDs on the bolts (RTD 2, RTD 3) and the TCs on the fixtures (TC 1, TC 2), likely because of their thermal capacities.

As previously discussed, the temperature responses of the bolts (measured by RTD 2 and 3) were not considered accurate representations of the heat flow through the strap. Instead, the temperature responses of the thermocouples on the fixtures (TC 1 and 2) were used to calculate the phase shift relative to the aluminium plates, as they were deemed more representative of the contact between the fixture and the strap.

Figure 4.11 shows the measured power data along with the intended power profile. The two curves closely match each other, confirming that the heat input to the test setup remains stable and does not introduce additional uncertainties. This consistency is essential for correlation.

The phase shifts between nodes from three sensor pairs (listed in Table 4.3) were calculated. They were selected to focus on key conduction paths and the uncertainties in them.

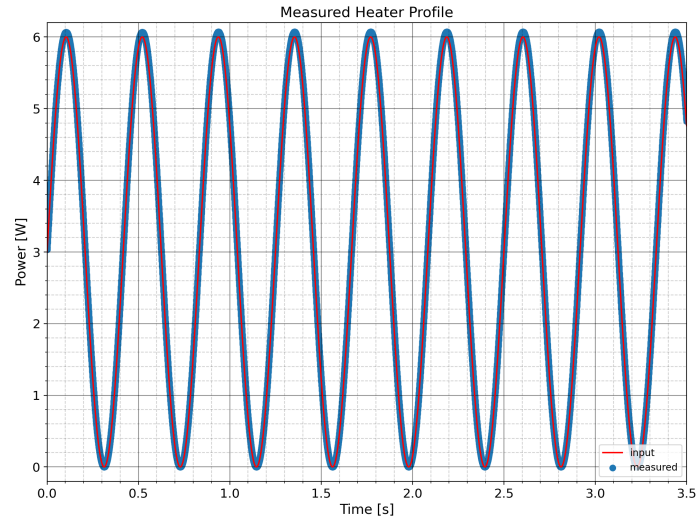


Figure 4.11: Heater power profile: input and measured curves.

Table 4.3: Interfaces and sensor pairs used for calculating $\Delta\phi$.

Interface	Sensor Pair	Representing
A	RTD 1 - TC 1	Contact between Al plate and Cu strap near the heat source
B	TC 1 - TC 2	Conduction through Cu strap
C	TC 2 - RTD 4	Contact between Al plate and Cu strap near the heat sink

4.3.2. Post Processing of the Temperature Responses

As seen in Figure 4.10, the raw temperature responses are noisy and exhibit slight drifts. To ensure accurate phase shift calculations, these responses must be post-processed. The first hour of the response, containing the system's transient behaviour, was excluded to focus on the stable oscillation periods, and then a curve fit was performed. Although various methods can be used to improve the quality of a curve fit, the steps followed in this study are outlined below.

Step 1: Smoothing and Detrending

Temperature responses often exhibit underlying trends and offsets that can interfere with the accuracy of curve fits. Therefore, to isolate their oscillatory behaviour, a moving average filter was applied. This filter averages a 'window' of points from the input data to produce the trends as the output signal. The '*rolling*' function in Python, with a window equal to the oscillation period, was used to identify these trends.

To remove the trends, detrending was performed by subtracting the moving average from the raw temperature data. This process enhances the accuracy and reliability of the curve fit by focusing on the true cyclic behaviour of the system. Figure 4.12 shows the stable temperature response, its trend, and the detrended temperature profile for RTD 1.

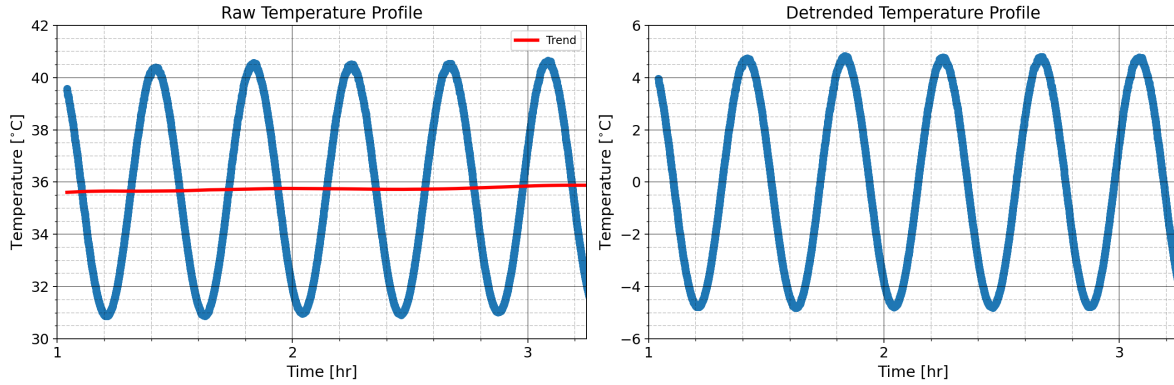


Figure 4.12: Raw and detrended temperature profile for RTD 1

Step 2: Curve Fitting

The periodic behaviour of the detrended temperature data was captured using a sinusoidal function. The curve fit was approximated using the function of the form given by Equation 4.3.

$$y = A \sin \left(\frac{2\pi}{p}x - \phi \right) \quad (4.3)$$

where,

- A = Amplitude of oscillation
- p = period of oscillation
- ϕ = phase of oscillation

The sinusoidal model was selected to match the heater power profile. For fitting the data to this model, the '*curve_fit*' function from the '*scipy.optimize*' library was used. This method uses a non-linear least square optimisation method to determine the fit parameters, A , p and ϕ , that minimises the sum of the squared residuals. The function adjusts the parameters to achieve the smallest possible discrepancies between the model and the observed data. The curve fits for the detrended temperature responses of the sensors are shown in Figure 4.13, along with its residuals.

Step 3: Quality Check

To avoid erroneous analysis and phase shift measurements, it is important to ensure a high-quality curve fit is achieved. A common method to assess this is by plotting the residuals of the curve fit, as shown in Figure 4.13. Residuals, defined as the differences between measured and fitted data points, indicate fit accuracy. As illustrated, the residuals are small, ranging from $\pm 0.05^\circ\text{C}$ to $\pm 0.1^\circ\text{C}$, they show random scattering without discernible patterns, and the root mean square error values are minimal. All of these factors indicate a good quality curve fit.

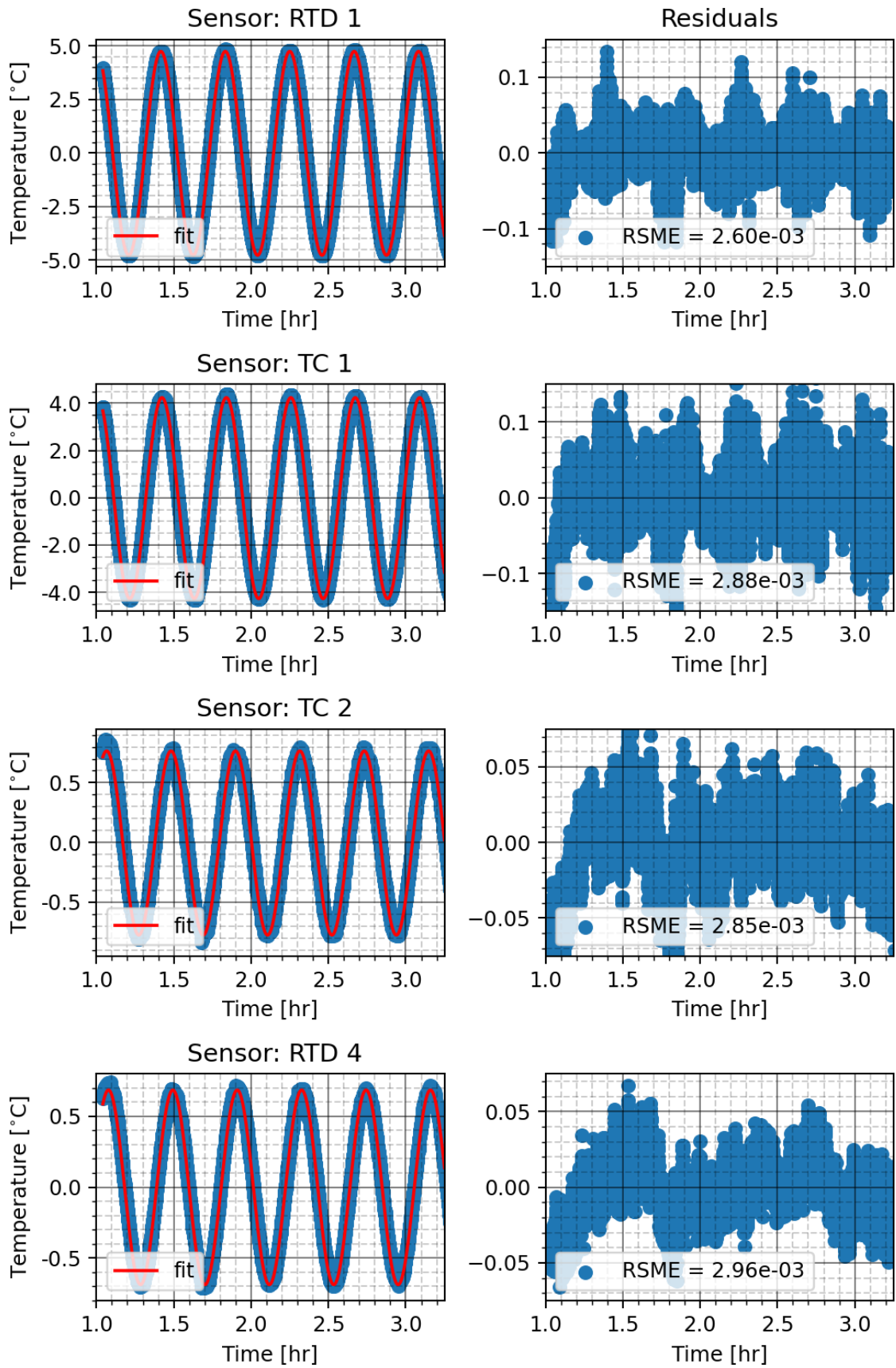


Figure 4.13: Curve fits and residuals of the detrended temperature data

4.4. Methods for Phase Shift Calculation

The ϕ between temperature responses at the interfaces, listed in Table 4.3, was calculated using the curve-fitted temperature data from above. Two methods of calculation were evaluated: the *find_peaks* function from SciPy and the Fast Fourier Transform (FFT).

4.4.1. Find Peaks

The *find_peaks* function identifies the local maxima in a signal based on parameters such as prominence and distance. It was considered for calculating time delays between temperature responses by detecting their peaks. However, it was observed that there were slight variations in the differences between the peak positions each cycle, leading to inconsistent time delay measurements depending on the selected peak. This variability potentially stems from ambiguous peak locations, inaccuracies in specifying correct peak parameters or curve fitting errors. Therefore, this method was considered to be less reliable for consistently calculating accurate phase shifts.

4.4.2. Fast Fourier Transform

FFT of the temperature responses can be used to calculate time delays by comparing phases at the frequency of interest. However, spectral leakage due to signal discontinuities at end-points can cause inaccuracies in the frequency domain. To mitigate this, a Hann window, shown in Figure 4.14 (top), was applied before performing the FFT. This tapering function reduces the signal amplitudes at the ends to minimise sharp transitions in the signal and improve the accuracy of the FFT. Although the Hann window introduces some phase distortion, it is linear and affects all frequency components equally. Therefore, the same window was consistently applied to all temperature responses ensuring that phase distortion effects would largely cancel out.

After the window was applied to the fitted temperature responses, FFT was performed, producing the spectrum shown in Figure 4.14 (bottom). The phase shifts were then determined by calculating the difference between the respective phase angles of each response. The corresponding time delay was calculated using Equation 3.9. This method proved to be robust and consistent, providing reliable time delay estimates. Table 4.4 summarises the measured time delays between the temperature responses of the sensors.

Table 4.4: Measured time delay [s] at the interfaces

Interfaces	Node Pairs	$\Delta\phi$ [s]
A	RTD 1 - TC 1	25.60
B	TC 1 - TC 2	207.81
C	TC 2 - RTD 4	41.95

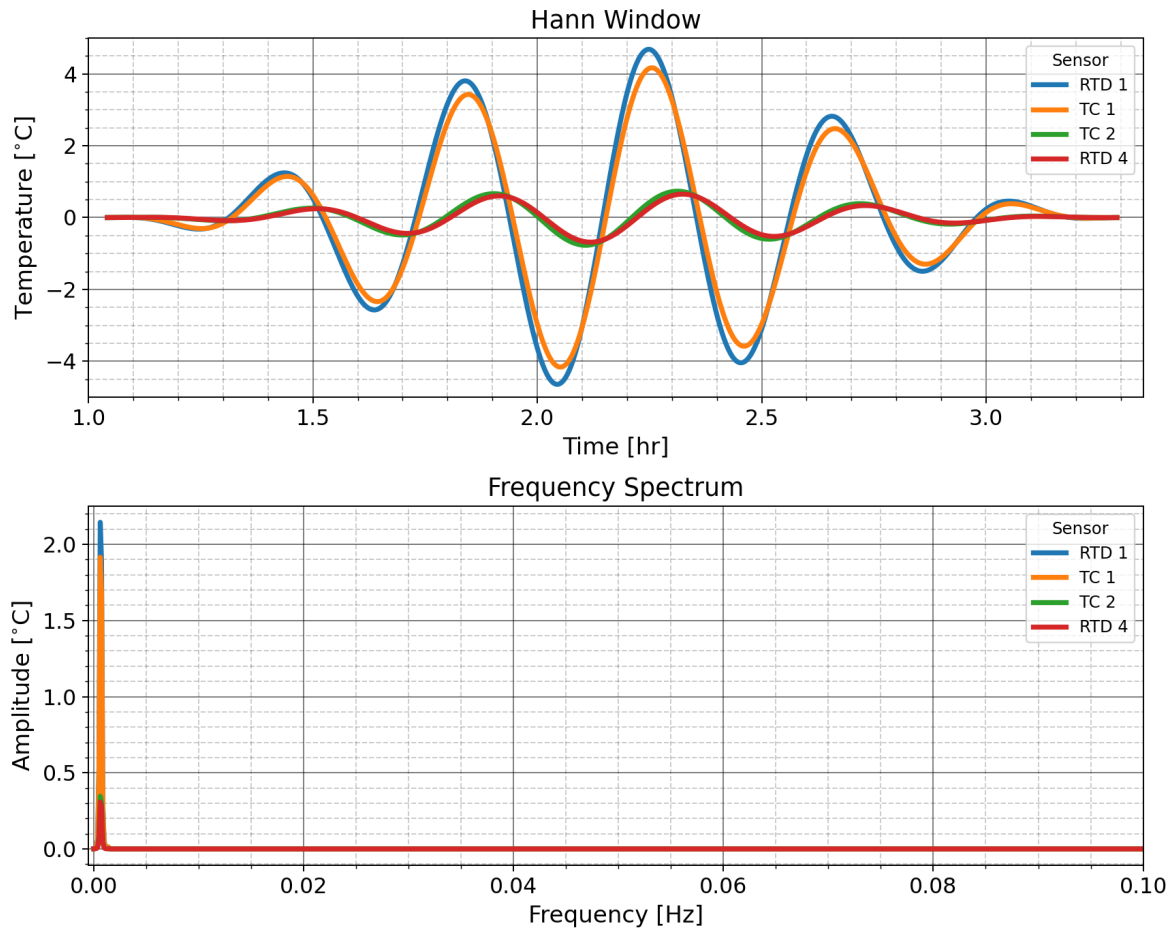


Figure 4.14: Hann window and FFT spectrum of sensor temperature responses

4.5. Additional Tests: Impact of Heat Input Power on Phase Shift

Using the same test setup, the influence of heater power input on the phase shifts at the interfaces was evaluated. The purpose was to determine whether varying the heat inputs would affect the phase shifts between the measured temperature responses. So, in each test, the heater power was varied while maintaining the ambient temperature at 24°C. Tests were conducted at power amplitudes of 3 W, 4 W, and 6 W. Power below 3 W was not considered, as that resulted in minimal temperature responses at the “cold” plate. The maximum power was limited to 6 W to maintain temperatures close to ambient conditions and minimise unintended convective effects. This also ensured that the voltages remained within the polyimide heater’s maximum operating range.

Signal-to-Noise Ratio

To verify the consistency of the tests, the signal-to-noise ratio (SNR) of the temperature data at each sensor was compared. The SNR was defined as the ratio of the variance of the detrended temperature data (signal) to the standard deviations of the curve fit (noise). Figure 4.15 shows the SNR results. A consistently high SNR across all sensors confirmed the stability of the temperature responses and ensured sufficient data quality for phase shift measurement.

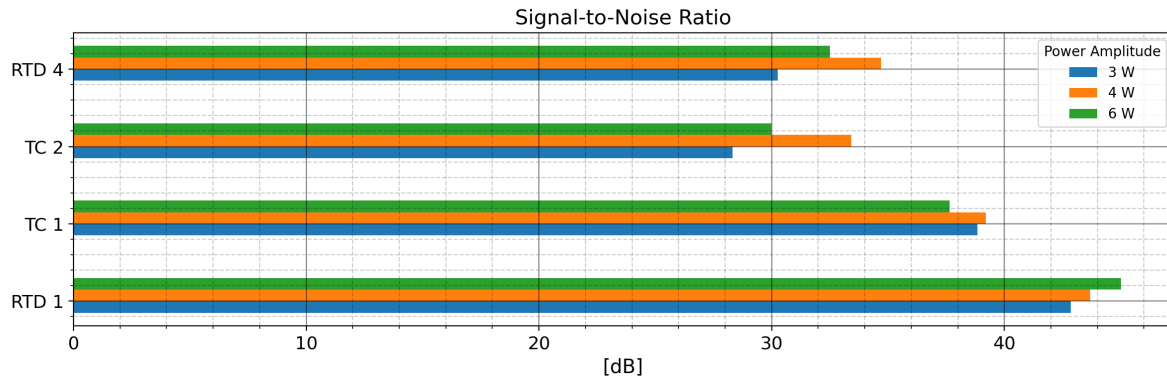


Figure 4.15: The Signal-to-Noise Ratio of the sensors

Phase Shift Comparison

Since the test setup was kept the same, the sensor placements and contact torques were consistent between the tests, making the heat power the only parameter that was varied. Therefore, any observed differences in the phase shifts could be confidently attributed to changes in the power input. Figure 4.16 shows the phase shifts between temperature responses at each interface for each test. The results indicate that the heater power has no significant impact on phase shifts. Differences at each interface are minor, corresponding to a 2-3 second absolute shift, with relative changes within 5% of the test results from Table 4.4. This indicates that phase shifts are largely independent of heat input magnitude.

By extension, it could be concluded that the phase shifts in temperature responses, due to oscillatory heat loads, are also unaffected by any potential heat leaks in the test setup. In other words, this method is possibly insensitive to the certainty of whether all the heater power is flowing through the interface of interest. This independence is significant, as it may remove a major source of testing uncertainty, quantifying uncontrolled heat leaks, from the process of TMM correlation, allowing the focus to remain solely on determining the thermal parameters of the system. However, these experiments did not account for radiative heat leaks and would require further assessment.

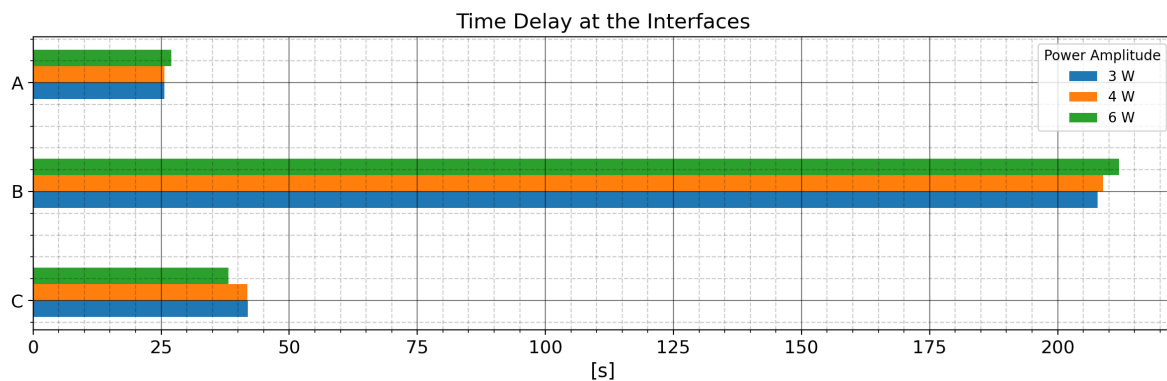


Figure 4.16: Phase shift between temperature responses at the interface with varied heater power

4.6. Chapter Summary and Testing Methodology

The purpose of this preliminary test campaign was to gain insights into effectively handling, post-processing and interpreting temperature responses for accurate phase shift calculation. Overall, the results contribute to refining the frequency domain thermal testing methodology. Key conclusions made were:

- Temperature responses inherently exhibit fluctuations and drifts; therefore, post-processing is essential to extract consistent and reliable phase shift data.
 - Detrending the temperature data significantly improves the accuracy of the fit, leading to more precise phase shift measurements.
 - Curve fitting is crucial to minimise the impact of drifts and ensure the phase shift calculations are based on the underlying periodic response.
- Using Fourier transforms on curve-fitted data proved to be a more robust phase shift calculation method.
 - Applying a Hann Window to the data to ensure reduced spectral leakages in the frequency domain.
- The tests show that heat input does not influence the measured phase shift at the interface, indicating that this methodology is insensitive to uncertainties from heat leaks and variations in power amplitude. This also answers the second half of the Sub-Question 2, concerning the impact of the power amplitude of the heat load on the phase shift.

4.6.1. Methodology for TMM Correlation Using Oscillatory Heat Loads

Based on these preliminary tests, the test strategy from Chapter 3 was modified, as outlined in Figure 4.17. To reiterate, first, the ideal frequency of oscillation and heater power should be determined from the preliminary TMM of the test setup. The setup should then be instrumented to ensure the critical interfaces are accurately captured for correlation. Testing can then be conducted at the selected frequency of oscillation. Detrending and curve fitting functions should be applied to the temperature data to ensure precise phase shift calculations before the final correlation, guided by trends observed in Chapter 3.

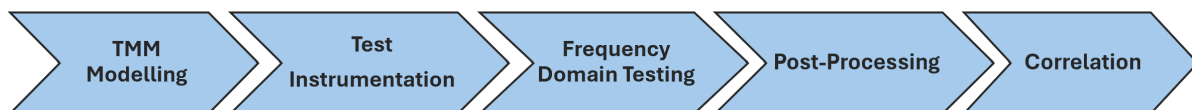


Figure 4.17: Schematic for frequency domain testing and model correlation

Implementing this methodology will help improve the accuracy of TMM correlation at an early stage. However, to fully align with space industry standards, testing must be conducted in a vacuum environment. This will enhance confidence in the methodology and identify any potential inconsistencies with industry benchmarks.

5

Frequency Domain Thermal Testing and Model Correlation

To fully validate the testing methodology using oscillatory heat loads, the test strategy outlined in Figure 4.17 was implemented. Building on the laboratory test setup, the tests were conducted again in a vacuum environment to align with space industry standards and ensure reliable thermal testing conditions. This also mirrors the environmental conditions of traditional thermal testing. Additionally, a conventional thermal balance test, typically performed in a vacuum to obtain data for model correlation, was conducted to verify that the results from the frequency domain testing method aligned with industry benchmark.

This chapter first outlines the modifications made to the test setup for use in a vacuum environment. Next, the tests conducted using both oscillatory and constant heat loads in vacuum and non-vacuum environments are discussed. The modifications made to the TMM are then addressed, followed by a presentation of the correlation results.

5.1. Test Setup for Vacuum Chamber

The vacuum chamber test setup retained the basic configuration from the laboratory tests, consisting of two aluminium plates connected via a braided copper strap bolted to the plates with M10 fasteners. The placement of the heater and resistance temperature detectors (RTDs) on the aluminium plate also remained the same as in the earlier tests. The hardware used is summarised in Appendix B. The other modifications made to the test setup are outlined below.

Heat Sink

Since convective heat transfer is absent in a vacuum environment, a dedicated heat sink was necessary to attain thermal equilibrium. For this, a metal block was used in conjunction with a Peltier device (TEC1-12706) as the heat sink. The heat sink block was sandwiched between the aluminium plate without the heater and the Peltier device. The heat was extracted from the test setup and transferred to the chamber. The use of the Peltier device allowed for control of the cooling process, essential for stabilising the system's thermal behaviour.

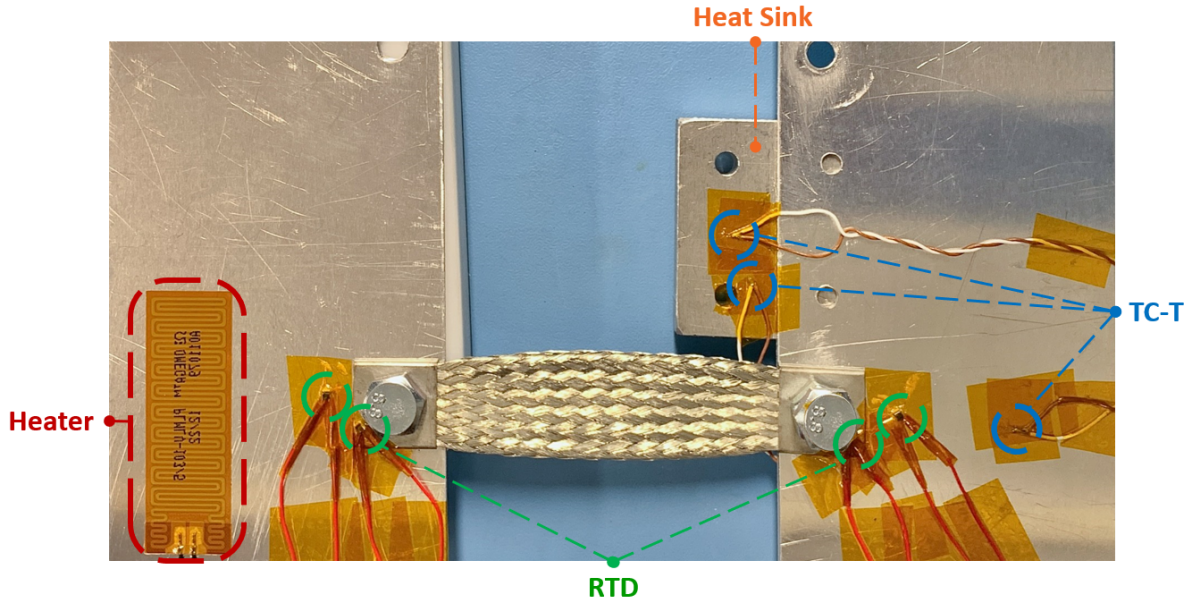


Figure 5.1: Setup and instrumentation for the vacuum chamber tests

Temperature Sensors Placement

In this test configuration, the placement of the temperature sensors on the braided strap was changed. Unlike earlier setups where the RTDs were placed on the bolts, here they were mounted on the fixtures of the braided strap, replacing the thermocouples previously used in this location. This adjustment was made to eliminate the effects of the steel bolts observed in the previous tests. In total, four RTDs were mounted to monitor the temperature responses at the key interfaces around the braided strap, ensuring high accuracy in the measured temperature data for model correlation. Two thermocouples were placed on the heat sink block to monitor its temperature, which also served as a boundary condition in the thermal model. A thermocouple was attached to the “cold” plate, at the same location as before, to capture the heat flow at the end of the setup. Figure 5.1 illustrates the test setup, including the sensor placements.

Multilayer Insulation (MLI)

Since the tests were conducted in a closed vacuum environment, it was necessary to account for radiative heat transfers. To minimise these effects, the setup was covered with MLI, made of Vacuum Deposited Aluminium (VDA) with an outer layer of Kapton. This configuration was selected due to its low emissivity, effective thermal insulation and common use in space applications. To fully cover the aluminium plates and the braided strap, three pieces of MLI, shown in Figure 5.2, were used.

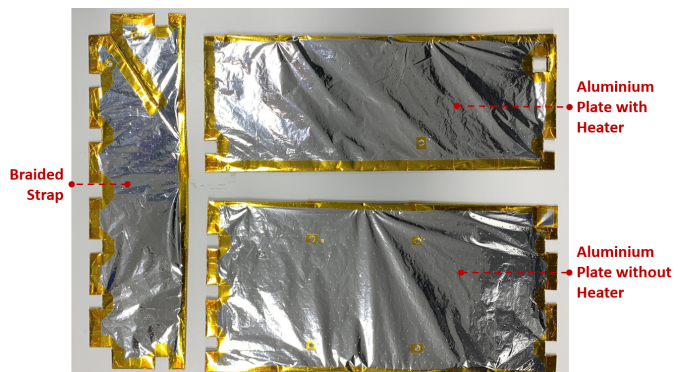


Figure 5.2: MLI pieces used in the setup.

Power Supply, Data Acquisition, and Chamber Feedthroughs

To precisely control the heater power and maintain consistent test conditions, two power supply units (PSUs) were used. The Delta Elektronika SM7020 regulated the oscillatory heat input via voltage control to the polyimide heater. The Delta Elektronika ES-030-10 was used to control the current to the Peltier device. The power to the device was maintained at a constant level equal to the heater's average power input. The consistent cooling ensured stable conditions for the temperature responses.

The temperature data acquisition was done by two devices: the USB-Temp-AI and National Instrument 9211. The former was used to store the RTD and heater voltage data while the latter was used to log the thermocouple readings. Both devices stored data at a high sampling rate of one reading per second. The connections to the vacuum chamber, including power and sensor feed-through, are illustrated in Figure 5.3, along with the test setup fully covered in MLI.

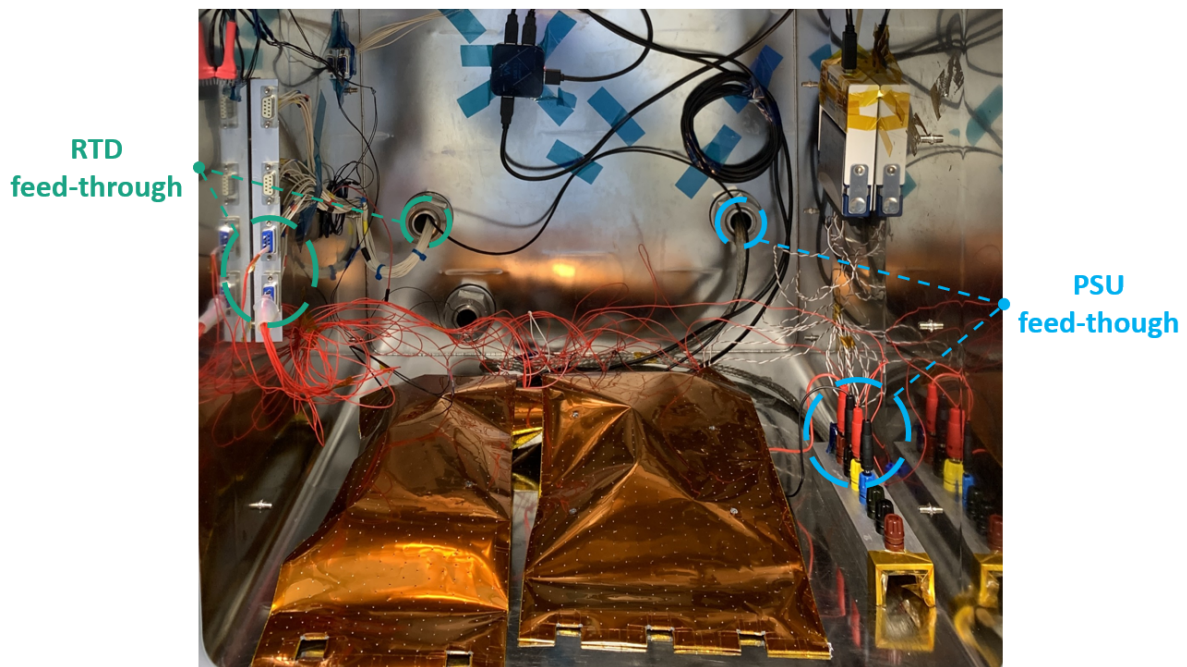


Figure 5.3: RTD and PSU feed-through and vacuum chamber test setup

Vacuum Chamber and Test Environment Control

The tests were conducted inside a Heraeus VT 6060M vacuum oven, which provided a stable, low-pressure environment for replicating space-like conditions. The chamber was maintained at a pressure of 0.083 mbar, replicating a medium vacuum environment. An additional thermocouple was mounted on the chamber walls, near the test setup, to store and use the temperature data of the chamber as a boundary condition for the TMM, since it was not being controlled.

Sensor Nomenclature

The nomenclature of the sensors used in the tests conducted are annotated in Figure 5.4. The thermocouple on the vacuum chamber was labelled 'TC 4'.

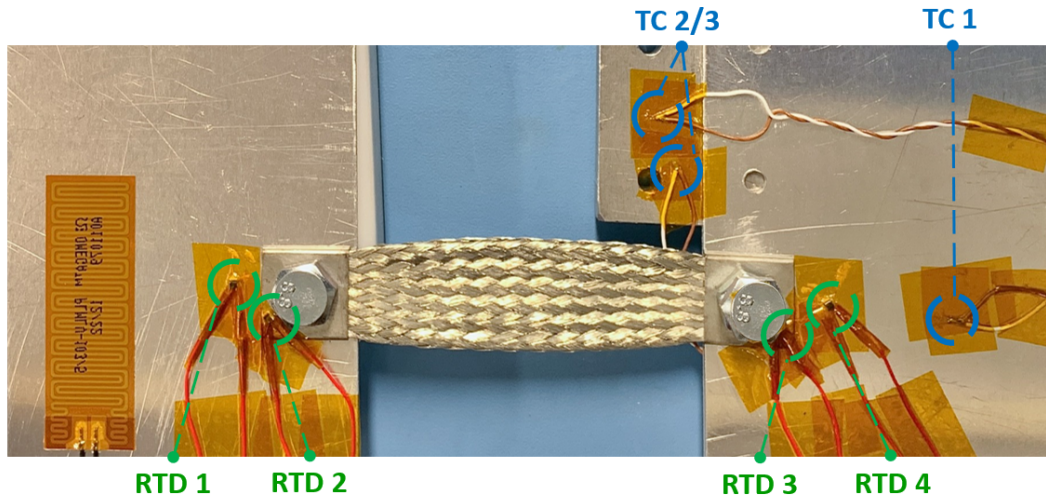


Figure 5.4: Chamber test: Sensor nomenclature

5.2. Frequency Domain Thermal Testing

The aim of the tests was to validate the testing methodology and evaluate its potential as an alternative to traditional thermal balance tests. Although the first step in the testing methodology (outlined in Figure 4.17) involved developing the mathematical model to determine the appropriate frequency and heater power for the test, the basic configuration of the test setup is the same as the tests conducted in the laboratory environment. Given this consistency, the heater power of 3 W and an oscillation period of 1500 seconds (or 25 minutes) were also applied to the tests conducted in the vacuum tests. Representing the same interfaces as before, the sensor nomenclature for phase shift calculation is outlined in Table 5.1

Table 5.1: Interfaces and sensor pairs used for $\Delta\phi$ calculations in chamber tests

Interface	Sensor Pair	Representing
A	RTD 1 - RTD 2	Contact between Al plate and Cu strap near the heat source
B	RTD 2 - RTD 3	Conduction through Cu strap
C	RTD 3 - RTD 4	Contact between Al plate and Cu strap near the heat sink

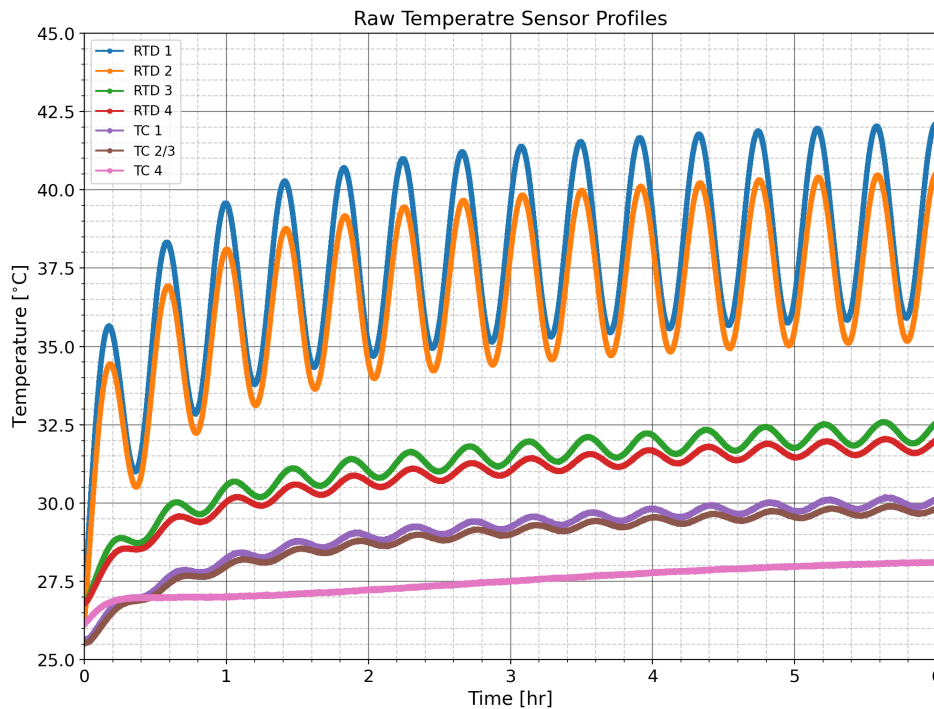
5.2.1. Controlled Environment

Before conducting the tests in a vacuum environment, an initial test was performed inside the closed vacuum chamber with air. This was done for two main reasons. First, testing in a controlled environment allowed for a more consistent assessment of the phase shift, as convective effects could be better controlled compared to a laboratory setting with fluctuating ambient conditions. Second, these tests served as a functional check before pulling a vacuum in the chamber, ensuring that both the heater and the Peltier device were operating as expected. Additionally, any unexpected instrumentation errors, such as improper sensor connections or data acquisition issues, could be identified and addressed before moving to a vacuum environment, where troubleshooting would be more challenging. Table 5.2 lists the parameters for the controlled environment test.

Table 5.2: Controlled environment thermal test - Test parameters

Test Parameter	
Oscillation Period	1500 s
Oscillation Frequency	6.67×10^{-4} Hz
Heater Power Amplitude	3 W
Run Duration	6.0 hours
Chamber Pressure	998 mbar
Initial Chamber Temperature	26 °C

The raw temperature data for the controlled tests are shown in Figure 5.5. The temperature oscillations seem to stabilise after approximately the first three hours, as indicated by the consistent amplitude of the responses. However, to ensure that any minimal drift doesn't influence the results, only the last two hours of the test responses were considered for the phase shift calculations.

**Figure 5.5:** Controlled environment thermal test - Raw temperature measurements

Similar to the laboratory tests, the raw temperature data was smoothed and detrended using the moving average filter before applying the '*curve_fit*' function to fit the data to the sinusoidal function described in Equation 4.3. Figure 5.6 shows the curve fits and their residuals. The quality of the fits is noticeably improved compared to the laboratory tests, likely due to the more stable conditions in the controlled environment. This improvement is clear in the residuals, which show smaller and more randomly distributed deviations, indicating a better fit than the laboratory tests.

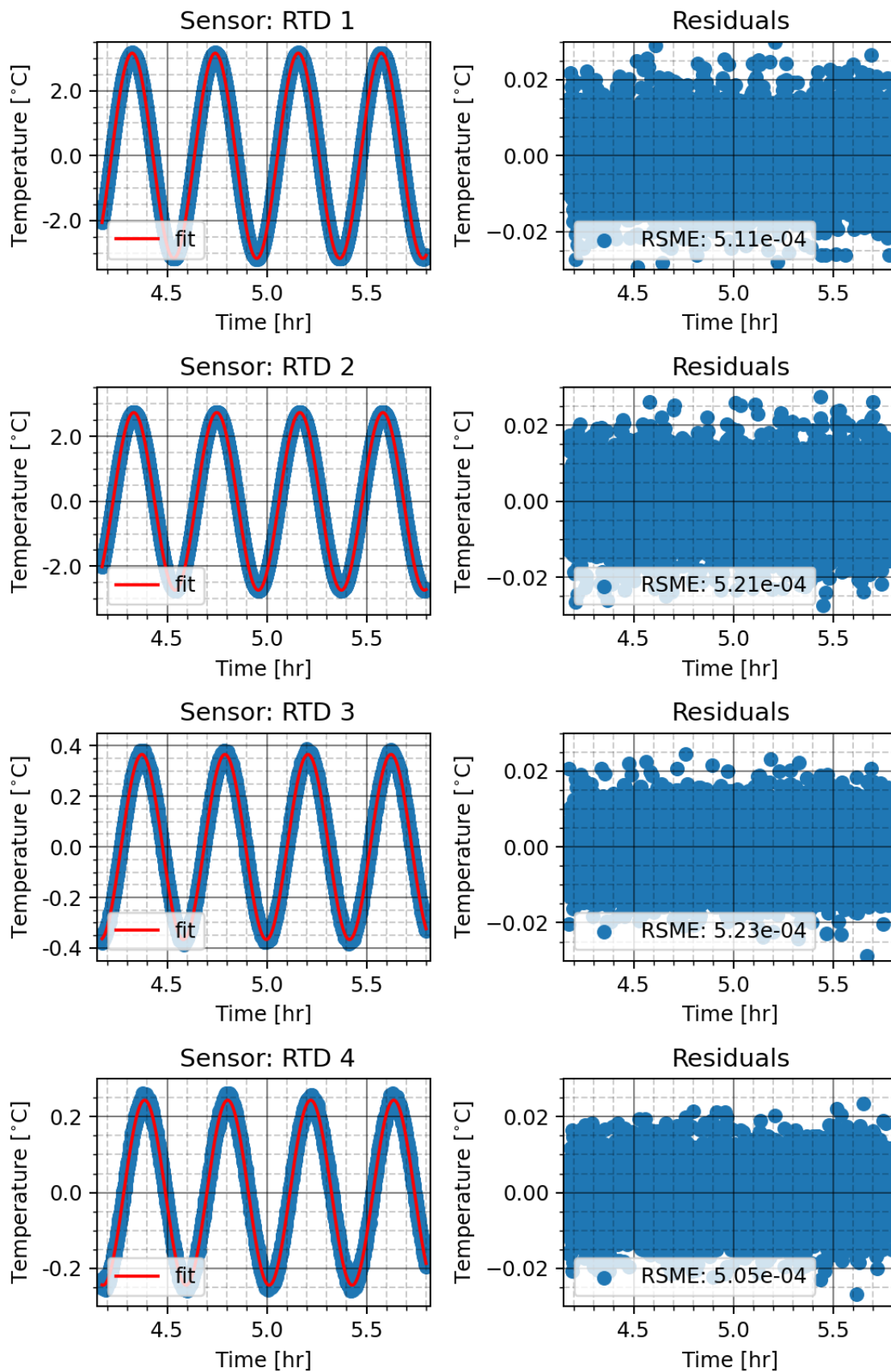


Figure 5.6: Controlled environment thermal test - Curve fits and residuals of the detrended temperature data

5.2.2. Vacuum Environment

For the vacuum environment test, same heat load profile was used, at a pressure of 0.083 mbar. The test parameters for the are outlined in Table 5.3.

Table 5.3: Vacuum environment thermal test - Test parameters

Test Parameter	
Oscillation Period	1500 s
Oscillation Frequency	6.67×10^{-4} Hz
Heater Power Amplitude	3 W
Run Duration	6.0 hours
Chamber Pressure	0.083 mbar
Initial Chamber Temperature	27.67 °C

The temperature responses are shown in Figure 5.7. The absence of convection in the vacuum environment implies that the heat transfer in the setup was governed solely by conduction and radiation. This shift in heat transfer mechanisms resulted in higher temperature amplitudes by almost 10°C than in the controlled environment. Additionally, there is a more prominent drift in temperature responses, which could be attributed to the gradual warming of the chamber walls. For the same reasons as before, only the last two hours of the temperature responses were used for assessment. The temperature responses were detrended and fit into the sinusoidal function, the residuals of which are plotted in Figure 5.8. The randomly distributed residuals and no discernible patterns indicate the goodness of the fit.

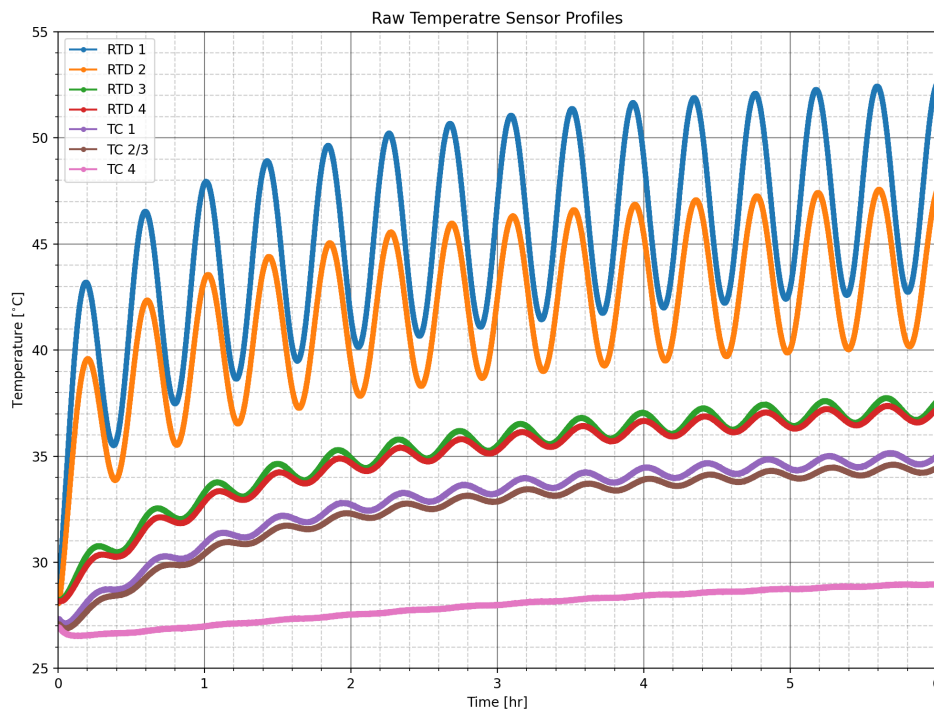


Figure 5.7: Vacuum environment thermal test - Raw temperature measurements

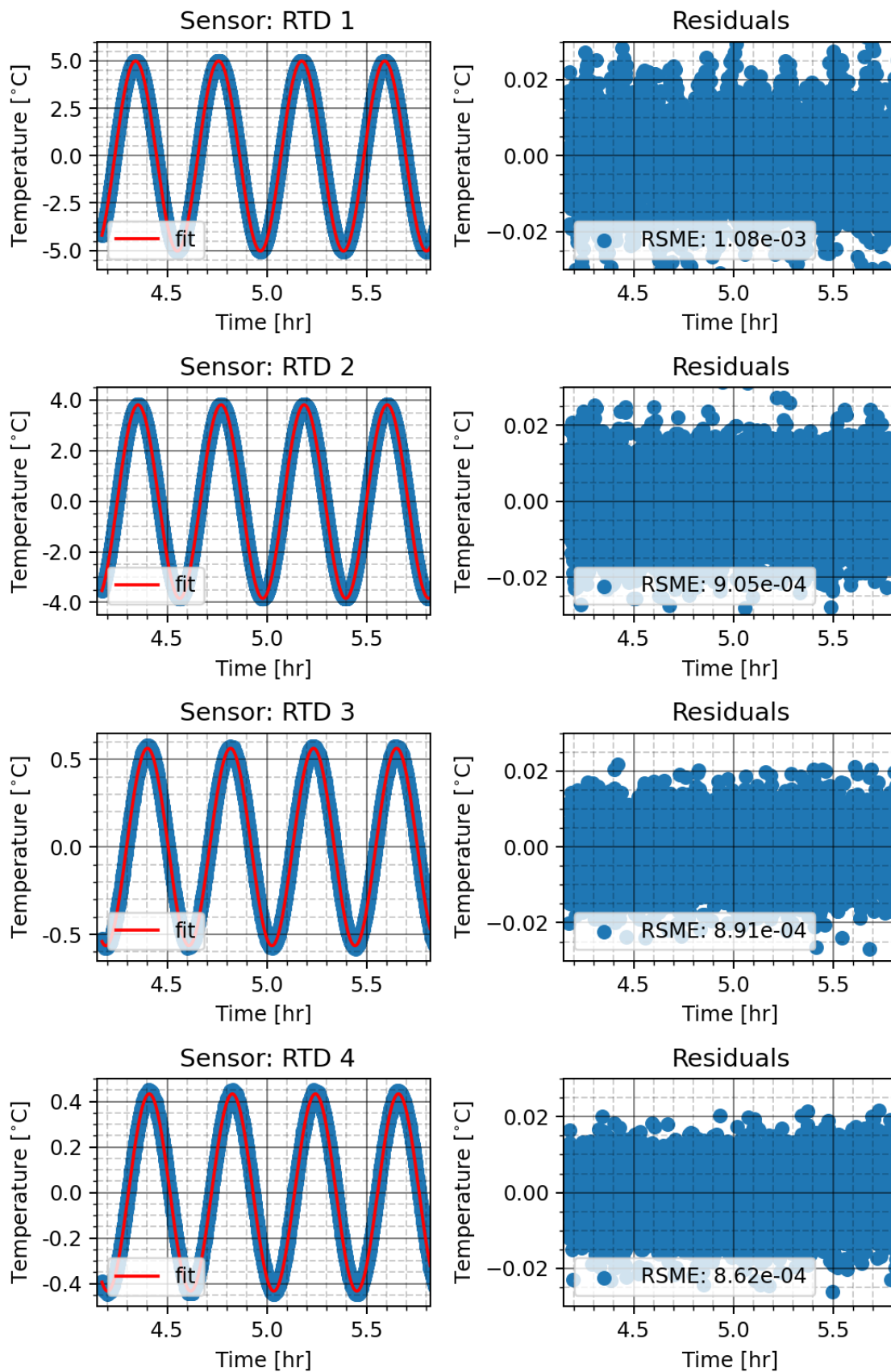


Figure 5.8: Vacuum environment thermal test - Curve fits and residuals of the detrended temperature data

5.3. Thermal Balance Test

The thermal balance test was also performed along with the frequency domain tests in the vacuum chamber. The test was used to help validate the testing methodology as it depicts the standard test used in the space industry. A constant heater power of 3 W, equivalent to the average power used in the oscillatory tests, was applied throughout the test. The key thermal balance test parameters are summarised in Table 5.4.

Table 5.4: Thermal balance test - Test parameters

Test Parameter	
Heater Power	3 W
Run Duration	10.0 hours
Chamber Pressure	0.085 mbar
Initial Chamber Temperature	27.67 °C

The system was monitored until thermal equilibrium was achieved. Figure 5.9 shows the measured temperature profiles during the test. The rate of stabilisation of the sensors was evaluated every 10 minutes over 10 hours, as also depicted in Figure 5.9. According to ECSS standards, a system is considered to be in a steady state when the rate of change in temperature is less than 0.1°C per hour. This criterion was met, confirming that thermal equilibrium had been reached, and the test was deemed successful. The steady-state temperatures were then determined by averaging the sensor readings from the last hour of the test.

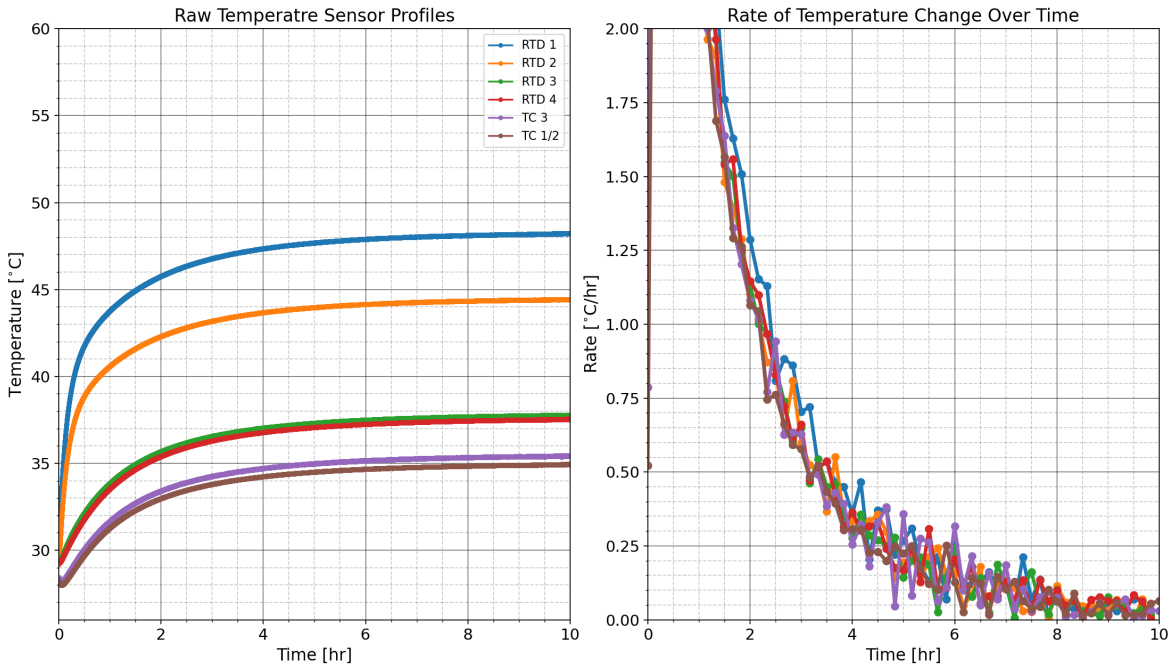


Figure 5.9: Thermal balance test- Temperature response and rate of stabilisation

5.3.1. Post Test Analysis: Unexpected Heat Leak

After the test setup was removed from the vacuum chamber, an unexpected heat leak was observed. It was found that the MLI on the aluminium plate with the heater had come into direct contact with the heater, resulting in a burnt heater and MLI (illustrated in Figure 5.10). This likely happened due to a mounting error in the placement of the MLI on the plate, which caused an uncontrolled heat path when air was pulled out of the chamber. Therefore, this heat leak likely exists in all the tests conducted in the chamber.

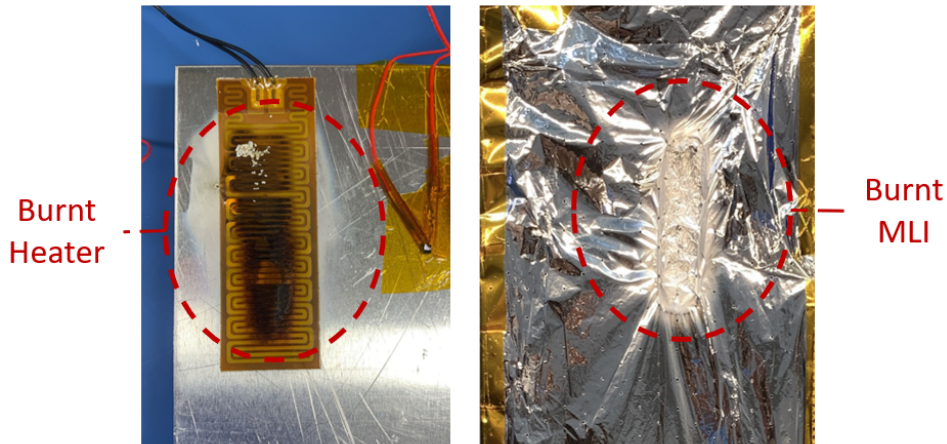


Figure 5.10: Heat leak: Heater to MLI

Although this unintentional heat leak introduced a deviation in the thermal environment, the insensitivity of the frequency domain testing methodology to such leaks has already been discussed with the earlier laboratory experiments. Therefore, it was decided not to account for this specific heat leak in the TMM during the frequency domain correlations. By comparing the correlated TMM results with the steady-state results from the thermal balance test—both with and without considering the heat leak, the robustness of this methodology can be further validated. This comparison aimed to provide additional insights into the sensitivity of the proposed testing approach to uncontrolled thermal paths.

5.4. TMM Correlation Process

To fully validate the frequency domain testing methodology and its application in thermal model correlation, it is important to show how the results from the tests align with a predictive thermal mathematical model (TMM). Model correlation involves iteratively updating the TMM until its predictions closely match the experimental data, thereby increasing confidence in the model's accuracy and predictive capability.

To perform this correlation, the TMM must first be modified to represent the test setup, including the vacuum chamber, MLI and any additional boundary conditions. Once this is done, the model parameters can be adjusted to get phase shifts that match the measured phase shift values from the frequency domain tests. The goal is to minimise the difference between the experimentally observed phase shifts and those predicted by the model, to get a correlated TMM. The sections below discuss the steps taken to correlate the model.

5.4.1. Updating the TMM for the Test Setup

The preliminary model outlined in Figure 4.2 was updated to more accurately represent the test setup used in the vacuum chamber. These modifications helped perform the correlation with a higher confidence in the test setup representation. The measured heater power (from the vacuum environment test) was used as the heat source in the TMM. The other TMM modifications are outlined below.

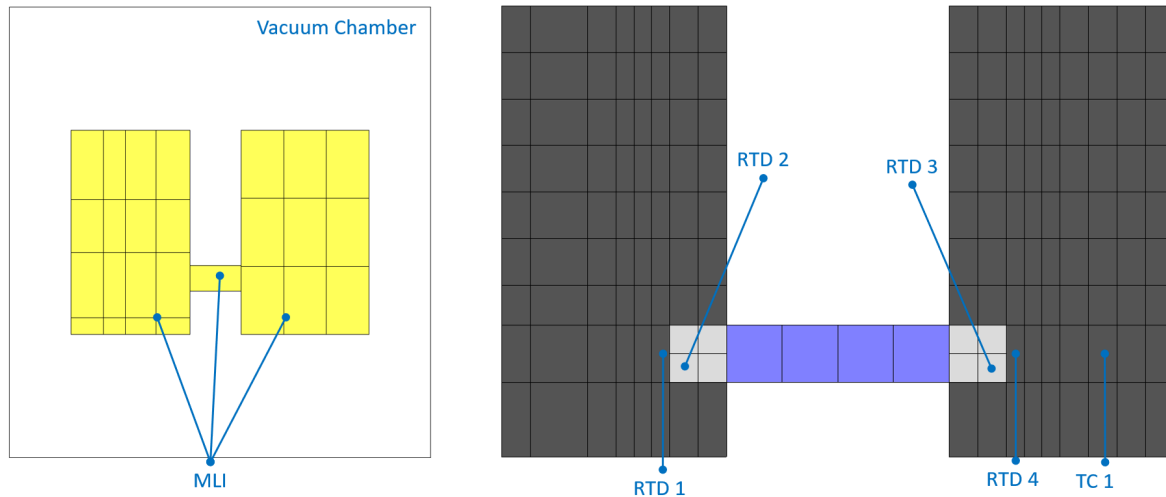


Figure 5.11: Modified TMM for the chamber tests

Vacuum Chamber

To accurately simulate the thermal environment of the test setup, the vacuum chamber was modelled based on its physical dimensions (49 cm × 52 cm × 48 cm) as a stainless steel enclosure. The chamber walls were assigned the bulk and optical properties of stainless steel 304, based on the manufacturer's specifications. These properties were assumed to provide an accurate representation of the chamber's radiative behaviour. Additionally, a boundary temperature condition was applied based on the readings from the thermocouple ('TC 4') mounted on the chamber walls, ensuring that the external environment was accurately captured.

Multi-Layer Insulation

The MLI was represented by three rectangular elements, as shown in Figure 5.11, which were assigned the optical properties of VDA on the inner face and Kapton on the outer face. These properties are summarised in Table 5.5. An emittance of 0.08 was used to model radiative heat transfer within the MLI, while conduction was neglected, consistent with typical assumptions in MLI modelling. This allowed the model to capture the dominant radiative effects with the chamber walls.

Table 5.5: MLI optical properties

	VDA	Kapton
Emissivity	0.06	0.51

Heat Sink

The heat sink was represented as a non-geometric node (NGN) to simplify the model while representing reality. The NGN was linked to the “cold” plate with a conductive interface of $800 \text{ W/m}^2\text{K}$ over the contact area annotated in Figure 5.12. The fixed boundary temperature for this node was the average of the two thermocouples (‘TC 2’ and ‘TC 3’) placed on the heat sink block.

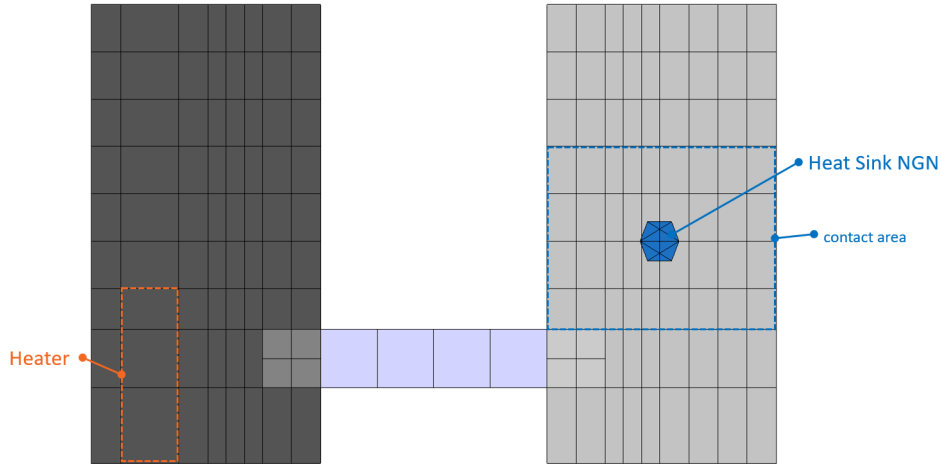


Figure 5.12: Heat source and heat sink nodes representative of the test setup

Copper Braided Strap

To improve the model's accuracy, the number of nodes along the strap fixtures was increased to better align with the locations of the temperature sensors (‘RTD 2’ and ‘RTD 3’). This ensured that local temperature gradients were more accurately captured. The strap's thermal capacitance was modelled based on its weight using Equation 2.6. As discussed before, bolts are typically excluded from TMMs as a general practice, as they are designed to be the least favourable heat path and rarely affect steady-state results. However, in this case, the large M10 bolts have significant thermal capacitance, which could notably alter the transient temperature results. Therefore, their capacitance was included in the model at the strap fixtures. Lastly, the contact conductance of the copper braid to the plate was set to $800 \text{ W/m}^2\text{K}$.

5.4.2. Frequency Domain Correlation

The correlation process began by calculating phase shifts for the predicted temperatures at the interfaces listed in Table 5.1 and comparing them to the measured values from the Vacuum Environment Thermal Test. The nodal locations equivalent to the temperature sensors are annotated in Figure 5.11.

The initial discrepancies between the predicted and measured responses are shown in Figure 5.13. The plot indicates that the phase shifts at interfaces A and C are larger than measured, suggesting that the contact conductance at these interfaces (that is between the aluminium plates and the strap) was underestimated. Additionally, the phase shift at interface B is smaller than expected across the strap, indicating that the actual conductance through the strap is lower than predicted. This could be due to the braided nature of the strap, where

the effective length may be longer than what was initially measured.

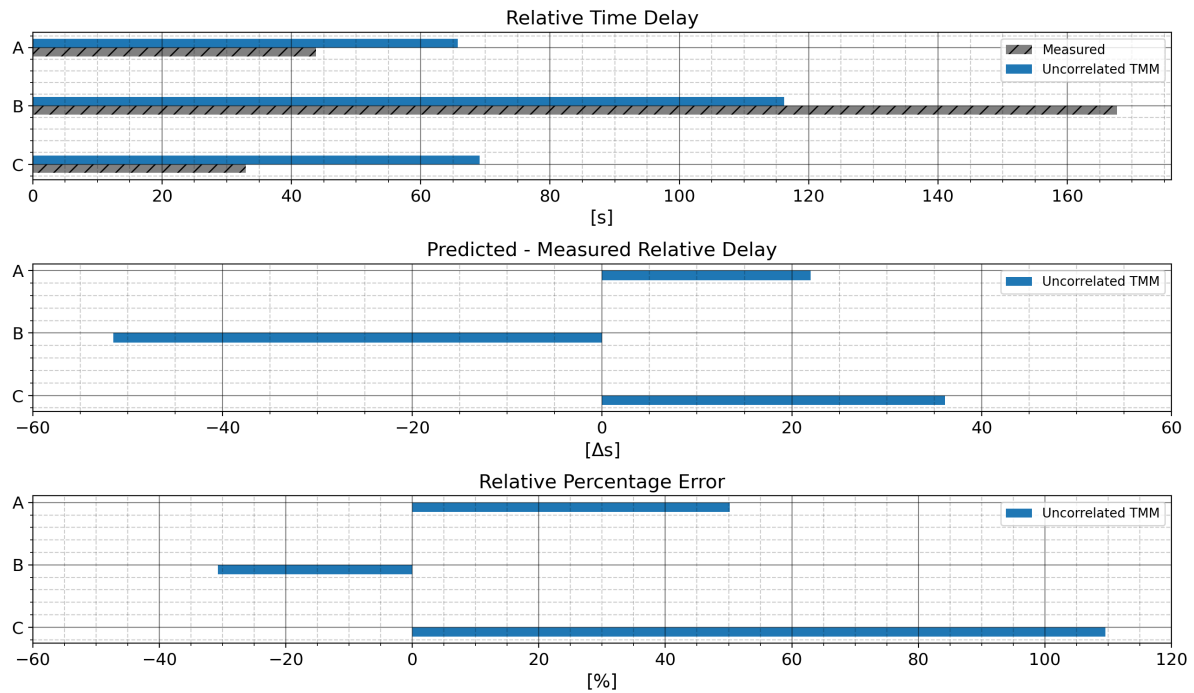


Figure 5.13: Uncorrelated TMM phase shift comparison

Given these observations, an iterative correlation process was initiated. The conductance values at each interface were adjusted individually, focusing on the contact conductance at Interfaces A and C, and the conductance through the strap at Interface B. While, the contact conductance values were adjusted directly, for the strap conductance a factor was applied. The uncorrelated model initially treated the braided strap as a solid copper piece, but during correlation, the factor was varied to account for the strap's braided structure. The goal was to align the phases at each interface while maintaining the model's overall physical consistency. Strategies outlined in Chapter 3, such as correlating one node pair at a time, were applied to increase the efficiency of the process.

Table 5.6: Modified TMM parameters

Thermal Contact Conductance [W/m ² K]	Uncorrelated	Correlated
Near heater (Interface A)	800	1656
Far from heater (Interface C)	800	2080
Heat sink	800	800
Conductance	Uncorrelated	Correlated
Strap Conductance (Interface B)	1.0	0.475

The results from the final correlated model are shown in Figure 5.14, with the modified parameters listed in Table 5.6. These parameters significantly improved the predicted phase shifts across all interfaces. The absolute difference at each interface was within 3 seconds.

This is 0.2% of the oscillation period of 1500 seconds and was therefore considered to be correlated.

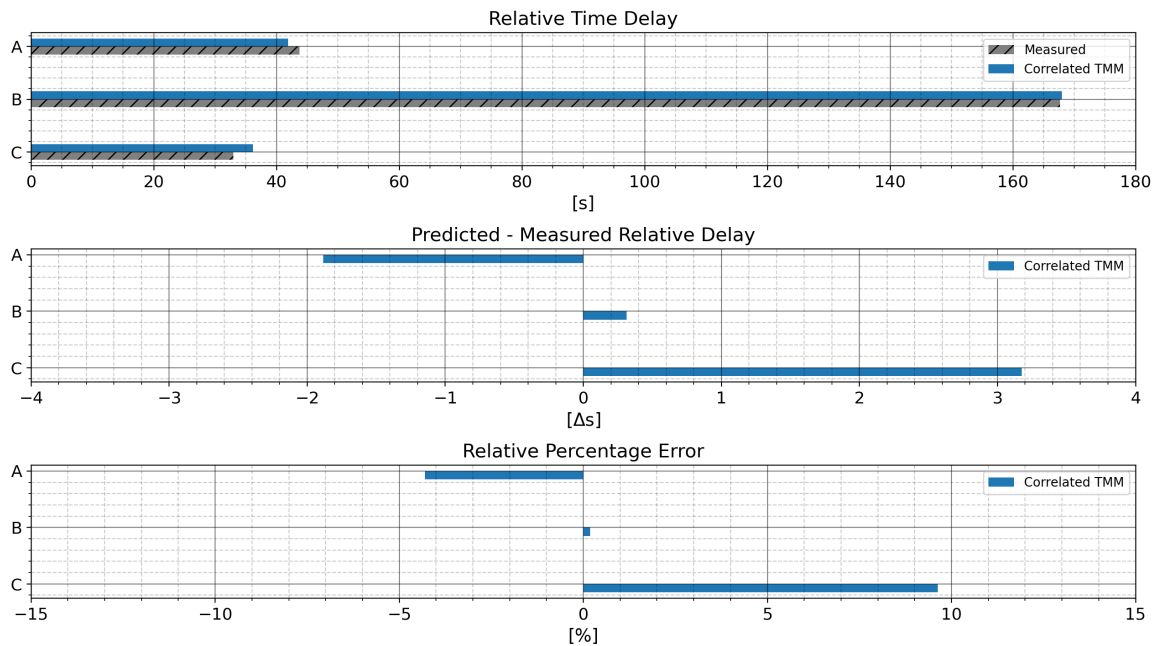


Figure 5.14: Correlated TMM phase shift comparison

With the correlation successfully achieved in the frequency domain, the results needed to be validated under steady-state conditions. The same TMM was used to check predictions with the steady-state results of the Thermal Balance Test. This would further validate the effectiveness and accuracy of the frequency domain testing methodology.

5.4.3. Validating the Correlated Model

To check the accuracy of the correlated TMM, the modified parameters listed in Table 5.6 were used for steady-state correlation using the measured thermal balance test results. A constant heat input of 3 W was set in the TMM at the "heater" nodes, consistent with the heater power used in the thermal balance test. Additionally, to account for the heat leak observed during the test, a direct conductive link was introduced between the aluminium plate and the chamber. This link represents not only the heat leak due to the MLI but also any other potential uncontrolled heat paths that might not have been identified before or after the test. This parameter was an uncertainty that needed to be correlated. The predicted and measured steady-state temperatures were compared, with the differences and absolute values plotted in Figure 5.15.

The results presented in Figure 5.15 demonstrate the impact of considering or disregarding the heat leak in the TMM. The correlated model without accounting for heat leaks correlates well with the temperatures measured by sensors located further from the heater, at 'RTD 3', 'RTD 4' and 'TC 1', with deviations within 2°C. However, the sensors near the heater, that is 'RTD 1' and 'RTD 2', read lower temperatures than predicted, indicating the presence of an alternate heat path that was not initially modelled.

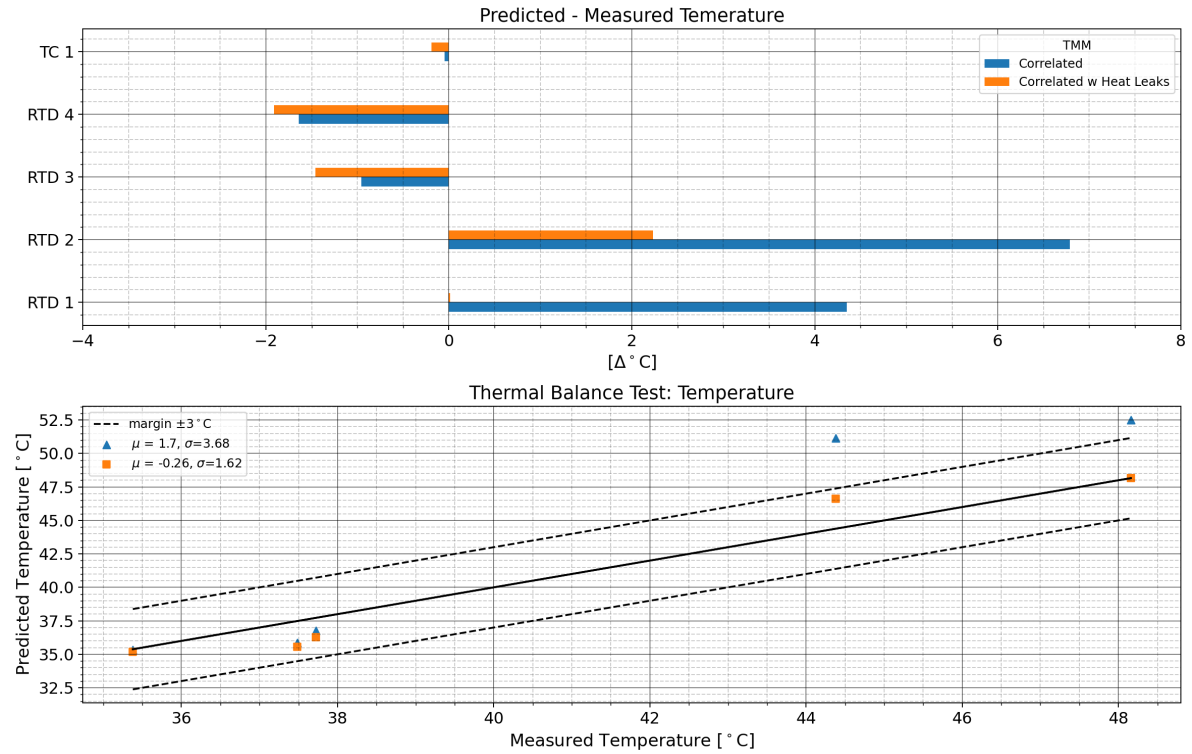


Figure 5.15: Thermal balance correlation results

On introducing a conductive link with a conductance of $1.85 \text{ W/m}^2\text{K}$, the correlated TMM exhibited much closer agreement with the measured temperatures for all the sensors. The mean deviation and standard deviation remained within the ECSS standards outlined in the literature, in Table 2.2. However, these results also show the sensitivity of traditional steady-state correlations to any form of heat leaks, as it was modelled as an extra correlation parameter.

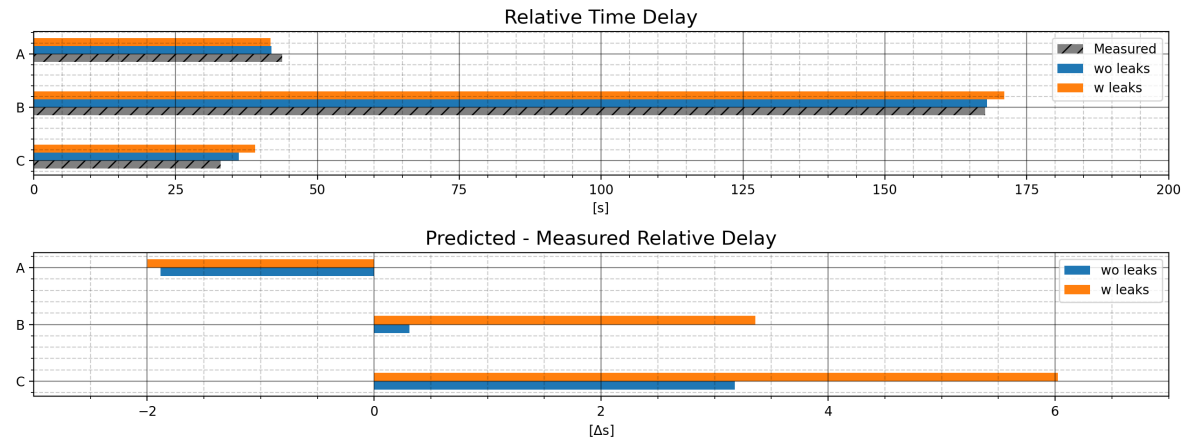


Figure 5.16: Effect of heat leak on predicted phase shift values

In contrast, when the same heat leak was included in the correlated TMM with oscillatory heat loads, the effect on the phase shift results was minimal. As shown in Figure 5.16, the differences in phase shift between the models with and without the heat leak are negligible,

ranging from 3 to 6 seconds. Given the overall phase shift duration, these variations are considered within the expected margin of error and therefore, insignificant.

The correlated TMM was thus successfully validated using the thermal balance test data. Although heat leaks were present, the frequency domain methodology proved to be robust and largely insensitive to such uncontrolled factors. Therefore, it can be concluded that this testing methodology holds significant potential for preliminary model correlation during the early stages of thermal design.

5.5. Additional Tests: Effect of Testing Environment on Measured Phase Shift

The oscillatory tests conducted in controlled and vacuum environments were compared to evaluate how environmental conditions affect the phase shift between temperature responses. This helped study the reliability and potential limitations of conducting tests outside a vacuum, especially when correlating thermal models that aim to replicate space-like conditions.

Identical test parameters were maintained across both environments, as discussed in Table 5.2 and 5.3, with the only difference being the chamber pressure. By isolating this, the results discuss how convection influences the observed temperature responses and phase shifts. The sensor pair selected to measure phase shifts corresponds to interfaces in the test setup, similar to those used in the laboratory tests. The sensor pairs and interfaces analysed were summarised before, in Table 5.1.

Signal-to-Noise Ratio Analysis

To verify the quality and consistency of the tests, the signal-to-noise ratio of the temperature responses was evaluated again. The same definition was used for signal and noise. A high SNR indicated a strong oscillatory signal relative to the noise, implying that the measurements are reliable. As shown in Figure 5.17, the tests from both environments exhibit consistently high SNR values across all sensors, indicating that the test quality was comparable.

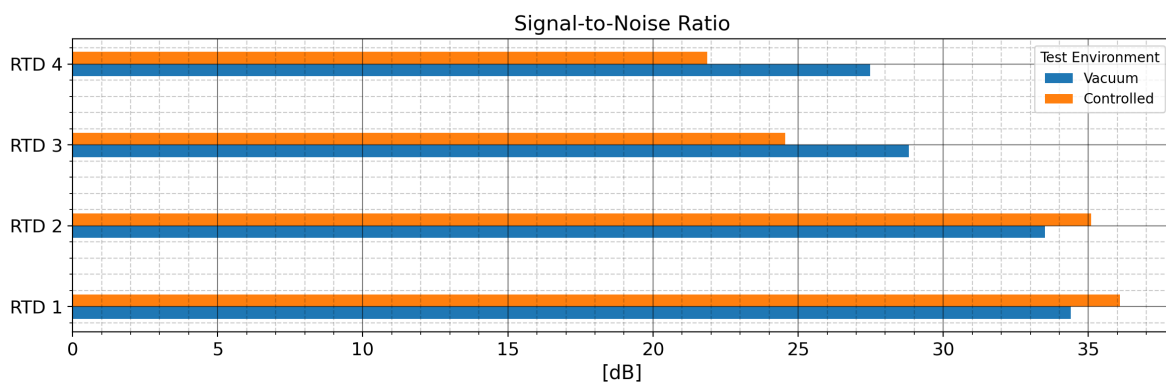


Figure 5.17: The Signal-to-Noise Ratio of the sensors

Phase Shift Comparison

The phase shifts between the sensor responses in the controlled and vacuum environments provide insights into the impact of environmental conditions on them. Figure 5.18 illustrates

these delays, where the results indicate that the vacuum environment shows larger time delays at Interface A and smaller time delays near the heat sink, at Interface C, in comparison to the controlled environment. The absolute difference ranges between 15-20 seconds across all the interfaces. This discrepancy can be attributed to convection in the controlled environment.

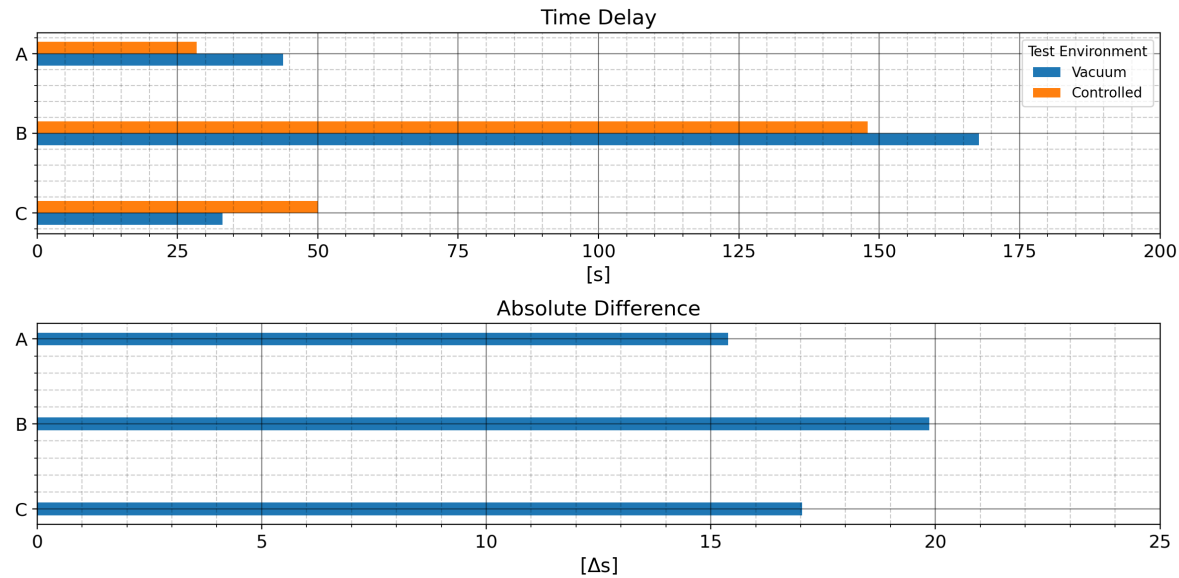


Figure 5.18: Difference in the phase shifts for different Test Environments

To assess the impact of the 20 second delay, correlations were conducted using the Controlled Environment Thermal Test. Convection effects were approximated using the convection equations and are given in Appendix C. The results, summarised in Table 5.7, indicate that the 20-second delay leads to approximately 25% variation in the thermal contact conductance parameters at Interfaces A and C, and a 5% difference in the conductance through Interface B.

Table 5.7: Comparison of TMM parameters in vacuum and controlled environments

Thermal Contact Conductance [W/m ² K]	Vacuum	Controlled	% Diff.
Near heater (Interface A)	1656	2100	23.64%
Far from heater (Interface C)	2080	1600	26.09%
Conductance	Vacuum	Controlled	% Diff.
Strap Conductance (Interface B)	0.475	0.5	5.13%

These findings suggest that frequency domain testing can be conducted in a controlled environment and used for TMM correlations, particularly during early development phases. As shown in Table 2.1, the generic inaccuracies reported in the literature can be reduced. If these tests are conducted at the coupon or unit level within the testing pyramid, the uncertainties in thermal contact conductance could be reduced by half.

Conclusions and Recommendation

6.1. Conclusion

Thermal balance tests and the correlation of Thermal Mathematical Models (TMMs) are critical steps in the development of thermal control systems for satellite structures. However, traditional methods present significant challenges, particularly for larger structures, due to their high cost and time-consuming nature, especially when maintaining vacuum conditions. Furthermore, testing uncertainties, such as uncontrolled heat leaks, measurement errors, and inherent model inaccuracies, complicate the correlation process, making it time-consuming.

Given these challenges, this research identified a potential gap in the development of an alternative testing methodology aimed at reducing the uncertainties associated with traditional thermal tests. The primary objective of this thesis was to establish a testing and correlation strategy based on the frequency domain, using oscillatory heat loads to generate temperature responses. The phase shift between these responses was then used as the basis for model correlation. Based on this objective, the following research question was formulated:

How can the phase shift between temperature responses due to oscillatory heat loads be used to correlate Thermal Mathematical Models (TMMs) and improve the reliability of the thermal test data?

To address this primary question, the following sub-questions were first considered:

How sensitive are phase shifts in temperature responses to variations in thermal network parameters under oscillatory heat loads?

The sensitivity analysis conducted on a dummy model demonstrated that phase shifts in temperature responses are indeed sensitive to variations in thermal parameters. Specifically, changes in conductive links resulted in localised variations in phase shifts, while changes in thermal capacitance affected all nodes along the conductive path from the point of variation to the heat sink. Based on these findings, a correlation strategy was developed, which allows for the focused correlation of conductive links between node pairs, one at a time. This approach offers a potential advantage over traditional steady-state-based correlation methods,

where any modification in conductance can lead to changes in temperature values in the entire model.

However, this sensitivity also highlights the necessity of accurately modelling the thermal capacitance of the test article. In contrast to traditional practices, where elements with small capacitance are often neglected, such omissions could lead to significant discrepancies in the frequency-domain approach. Therefore, the ideal application of this methodology would be for tests where only one type of thermal network parameter (e.g., thermal conductance) requires correlation, while the other parameters (e.g., thermal capacitance and radiative conductance) have a high assurance.

What is the impact of the amplitude and oscillation frequency of the heat load on the phase shift in temperature responses?

The impact of oscillation frequency on phase shift measurements is crucial. The results highlight the necessity of selecting an appropriate oscillation frequency to generate sufficient temperature responses at key locations, while also ensuring that the test duration remains manageable. For larger test articles, a lower oscillation frequency would typically be required to obtain measurable temperature responses across the entire structure. However, if time constraints prevent using a smaller frequency, the amplitude of the heat input can be increased. Laboratory tests demonstrated that phase shifts remain consistent regardless of variations in heater power, indicating that the phase shift in temperature response is independent of the heat input magnitude. Thus, increasing heater power can effectively generate the required temperature responses in the test article.

In practical applications, this methodology allows for a trade-off between heater power amplitude and oscillation frequency, depending on the sensitivity of the test article to thermal loads and the time available for testing. Additionally, the methodology's independence from heat input magnitude enhances its robustness against uncontrolled heat leaks, thereby improving the reliability of the thermal test data.

However, certain conditions may require the correlation of specific heat leaks, such as those from MLI or electronic components, to accurately model their effects. In other conditions, it may be required to correlate the quantity of heat flowing through the model due to power budget constraints. In these cases, this methodology may be limited. A potential solution could be to complement the frequency-domain tests with traditional thermal balance tests, enabling a more comprehensive correlation that addresses both the thermal conductance and any inherent heat leaks in the test.

How do variations in test instrumentation and environmental conditions influence the phase shifts between periodic temperature responses?

The sensitivity of phase shifts in temperature responses was assessed with respect to the location of sensors and heaters in the test setup. It is important to position sensors carefully, ideally in close proximity, to ensure they are represented by a simple thermal link in the TMM. In cases where access to optimal locations is challenging, selecting nearby locations that can be similarly represented in the TMM is important to maintain accuracy.

Interestingly, the results indicate that phase shifts are largely insensitive to the exact placement of the heater, provided that an oscillatory temperature response is generated at the critical locations. This insight offers practical flexibility in testing. One approach is to use the TCS heater to induce the required oscillatory responses for model correlation. Alternatively, if the TCS heater is unavailable or unable to generate sufficient heat, an additional heater can be placed near the critical locations of interest. This method is particularly beneficial in higher-level subsystem assemblies, where correlation is required only for specific locations.

Environmental conditions also play a significant role in phase shift behaviour. To assess this, frequency domain tests were conducted both in a vacuum and at ambient pressure. The results revealed a ± 20 second variation in recorded phase shifts between the two environments, translating to a 25% difference in the thermal contact conductance coefficient at ambient conditions. This finding suggests that if model correlation is performed in ambient conditions during the preliminary phases (Phase A or B, as shown in Figure 1.2), the parameters can be determined with a 25% certainty. This level of accuracy is half of the generic uncertainties specified by the ECSS (Table 2.1), indicating that early-stage TMM improvements can be made with reasonable confidence at ambient conditions.

To address the primary research question, phase shift correlations can be effectively performed by adjusting the thermal network parameters based on the oscillatory temperature response data. A simple test setup was used to carry out these correlations in a vacuum environment. The correlated model was then compared with steady-state results from a traditional thermal balance test to benchmark its accuracy against industry standards. The results demonstrated a strong alignment between the two, particularly after accounting for heat leaks in the steady-state correlation.

This finding highlights that while accurately modelling heat leaks is essential in traditional thermal tests, it becomes less critical in frequency domain testing. Even without explicitly factoring in heat leaks, the frequency domain method can still produce reliable estimates of key parameters, such as thermal contact conductance. This advantage enhances the efficiency and robustness of the correlation process, allowing for a more focused effort to eliminate inherent uncertainties in the TMM.

6.2. Recommendation for Future Work

To further validate and enhance the proposed testing and correlation methodology, the following steps are recommended:

1. **Reassessing Environmental Influence:** The influence of environmental conditions on phase shift measurements should be re-evaluated. The presence of a significant heat leak in the current tests may have affected the convection effects within the chamber. Therefore to gain more confidence in this observed 25% uncertainty margin, the tests should be performed again. Additionally, since this uncertainty may be more critical for systems with different thermal response times, it is necessary to conduct assessments on units with different thermal responses to determine if this uncertainty margin remains consistent under diverse conditions.
2. **Controlled Heat Leaks:** Introducing controlled heat leaks during testing could help to further validate the insensitivity of the measured phase shifts to variations in heat input. By systematically studying the impact of these controlled leaks, confidence in the robustness of the phase shift measurements can be strengthened.
3. **Addressing Composite Material Uncertainties:** One significant source of uncertainty lies in the inhomogeneity of thermal properties within composite materials. As the use of composites in the space sector expands, it is important to assess and correlate the uncertainties associated with these materials within the proposed methodology. Given that thermal capacitance in composite structures is not uniform, accurate modelling of the laminate's thermo-physical material properties will be essential for extending the applicability of this testing approach.
4. **Incorporating Radiative Effects:** Radiative effects, which were not considered in this work, are a critical aspect of the space environment. Future work should include the assessment and correlation of radiative effects with high certainty to ensure the comprehensive applicability of the testing methodology in space-like conditions.
5. **Exploring Test Unit and TMM Complexity:** The proposed methodology has primarily been validated for one-dimensional heat flow. As the complexity of test units increases, with heat flow occurring in multiple directions, it is important to assess the validity of this methodology in two-dimensional and three-dimensional scenarios. Expanding the methodology to handle multidirectional heat flow will significantly increase its scope and applicability. Additionally, the use of frequency decomposition in frequency domain analysis, using multiple heaters at different frequencies and locations, could be explored to correlate complex models more effectively.

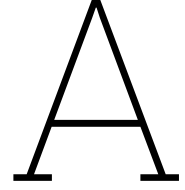
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Frequency Domain Thermal Network

The section discusses the transfer of thermal networks in the frequency domain as proposed by Hailer et al. [49] in their study of utilising the complex thermal transfer functions. It discusses using their methodology to derive the phase shift and amplitude gain between two nodes.

A.1. Thermal Modelling in Frequency Domain

A thermal network consisting of n diffusive nodes and k boundary nodes is considered, with the total number of nodes given by $m = n + k$. The ordinary differential equation (ODE) for the i^{th} node in this network is expressed by Equation A.1.

$$C_i \frac{dT_i}{dt} = Q_i + \sum_{j=1}^m GL_{i,j}(T_j - T_i) + \sigma \sum_{j=1}^m GR_{i,j}(T_j^4 - T_i^4) \quad \text{for } i = 1, \dots, n \quad (\text{A.1})$$

The initial temperature of the i^{th} node is denoted by $T_{i,0}$. The boundary nodes, on the other hand, are characterised by a sinusoidal temperature profile with an angular frequency of ω and an amplitude T_o .

$$T_i(0) = T_{i,0} \quad \text{for } i = 1, \dots, n \quad (\text{A.2})$$

$$T_k(t) = T_o \sin \omega t \quad \text{for } k = 1 + n, \dots, m \quad (\text{A.3})$$

To solve this network in the frequency domain, Hailer et al. [49] simplified the system by approximating the model as a linear system near its equilibrium temperature. It is assumed that the temperature can be represented as $T = T_E + \delta T$, where T_E and δT represent the equilibrium temperature and deviations from this temperature respectively. This assumption simplified the linearization of the radiative terms and allowed the ODE to be expressed in the standard time-invariant state space form. By applying these simplifications to the ODE, Equation A.1 was reduced to the linear form given by Equation A.4.

$$\text{diag}(C) \delta \dot{T} = \delta Q + \mathbf{K}(\delta T) + \mathbf{R}(\text{diag}(4T_E^3) \delta T) \quad (\text{A.4})$$

where,

- $diag(C)$ represents the vector of capacitance
 δT represents the temperature deviation from T_E
 $\mathbf{K}$ represents the symmetric matrix for conductive links between the nodes
 $\mathbf{R}$ represents the symmetric matrix for radiative links between the nodes

The conductive and radiative links were represented as symmetric matrices of size $(m \times m)$ and are defined by Equation A.5 and Equation A.6, respectively.

$$\mathbf{K}_{i,j} = GL_{i,j} \quad \mathbf{K}_{i,i} = - \sum_{j=1}^m GL_{i,j} \quad (\text{A.5})$$

$$\mathbf{R}_{i,j} = \sigma GR_{i,j} \quad \mathbf{R}_{i,i} = - \sum_{j=1}^m \sigma GR_{i,j} \quad (\text{A.6})$$

With this simplification, Equation A.4 could be expressed in the standard time-invariant form given by Equation A.7.

$$\delta \dot{T}_D = \mathbf{A} \delta T_D + \mathbf{B} \begin{bmatrix} \delta Q_D \\ \delta T_B \end{bmatrix} \quad (\text{A.7})$$

where,

- δT is the state vector
 $\mathbf{A}$ is the system matrix
 $\mathbf{B}$ is the input matrix

In Equation A.7, the subscript D denotes the diffusive nodes, and the subscript B indicates the boundary nodes. The system matrix $\mathbf{A}$ and the input matrix $\mathbf{B}$ are formed by a combination of the conductive and radiative links and the capacitance of the nodes.

$$\mathbf{A}_{i,j} = C_i^{-1} \left(\mathbf{K}_{i,j} + 4T_{E_j}^3 \mathbf{R}_{i,j} \right) \quad \text{for } i, j = 1, \dots, n \quad (\text{A.8})$$

$$\mathbf{B}_{i,i} = C_i^{-1} \quad \text{for } i = 1, \dots, n \quad (\text{A.9})$$

$$\mathbf{B}_{i,k} = C_i^{-1} \left(\mathbf{K}_{i,k} + 4T_{E_k}^3 \mathbf{R}_{i,k} \right) \quad \text{for } k = n + 1, \dots, m \quad (\text{A.10})$$

To convert Eq. A.7 from the time domain to the frequency domain, the system's transfer function was used. This involved applying Laplace transformation to the linearised Eq.A.7 to get the transfer function, given by Equation A.11.

$$(s\mathbf{I} - \mathbf{A}) \mathbf{G}(s) = \mathbf{B} \quad (\text{A.11})$$

where,

- $s = i\omega$ is the independent variable in frequency domain
 $\mathbf{I}$ is the identity matrix
 $\mathbf{A}$ is the system matrix constructed from Equation A.8
 $\mathbf{G}(s)$ is the transfer function matrix
 $\mathbf{B}$ is the input matrix constructed by Equation A.9 and Equation A.10

The transfer function matrix, $\mathbf{G}$, stores the phase shift and gain values at frequency $s = i\omega$ for each node. For a transfer function represented by $G_{i,j}(s) = x + iy$, the gain of the temperature response can be extracted from the absolute value (shown in Equation A.12) of the function and the phase shift from its argument (depicted in Equation A.13) [49].

$$abs(x + iy) = A = \sqrt{x^2 + y^2} \quad (\text{A.12})$$

$$arg(x + iy) = \phi = \tan^{-1} \left(\frac{y}{x} \right) \quad (\text{A.13})$$

Using the above equations, the phase and gain between two nodes can be formulated. The formulation of these parameters between two adjacent nodes will be presented in the next section.

A.2. Two Node Thermal Model

Consider the two-node thermal network as shown in Figure A.1.

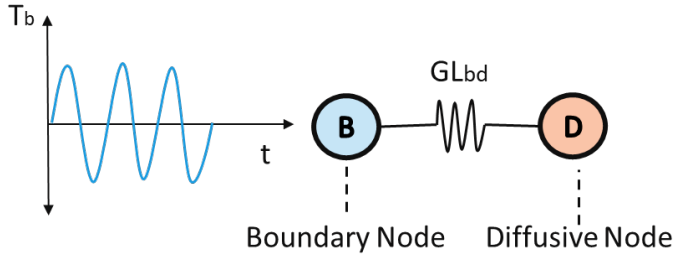


Table A.1: Nodes

Total	m	2
Diffusive	n	1
Boundary	k	1

Figure A.1: A two-node conductive thermal network under oscillatory boundary conditions.

The boundary node B has a sinusoidal temperature profile with an angular oscillation frequency ω . The conductive link between the diffusive node D and node boundary node B is denoted by GL_{bd} . No radiative links have been modelled. Assuming unit perturbations on the boundary nodes and no heat input to the system, Equation A.4 is reduced to

$$diag(C) \delta \dot{T} = \mathbf{K} \delta T$$

Here, the capacitance vector is,

$$diag(C) = \begin{bmatrix} C_b \\ C_d \end{bmatrix} \quad (\text{A.14})$$

the coefficient of the conduction matrix, using Equation A.5, is,

$$\begin{aligned} \mathbf{K} &= \begin{bmatrix} K_{bb} & K_{bd} \\ K_{db} & K_{dd} \end{bmatrix} \\ &= \begin{bmatrix} -GL_{bd} & GL_{bd} \\ -GL_{bd} & -(-GL_{bd}) \end{bmatrix} \\ \mathbf{K} &= \begin{bmatrix} -GL_{bd} & GL_{bd} \\ -GL_{bd} & GL_{bd} \end{bmatrix} \end{aligned} \quad (\text{A.15})$$

Using the above property matrices, and Equations A.8, A.9 and A.10, the system matrix **A** and the input matrix **B** are formulated as,

$$\mathbf{A} = \left[C_d^{-1} (\mathbf{GL}_{bd}) \right] \quad (\text{A.16})$$

$$\mathbf{B} = \begin{bmatrix} C_d^{-1} \\ C_d^{-1} (\mathbf{GL}_{bd}) \end{bmatrix} \quad (\text{A.17})$$

Using Equation A.11, with $s = i\omega$, the transfer function for the system is,

$$\begin{aligned} G(s) &= (s\mathbf{I} - \mathbf{A})^{-1} \mathbf{B} \\ &= (i\omega - C_d^{-1} \mathbf{GL}_{bd})^{-1} \begin{bmatrix} C_d^{-1} \\ C_d^{-1} (\mathbf{GL}_{bd}) \end{bmatrix} \\ &= \frac{1}{i\omega - C_d^{-1} \mathbf{GL}_{bd}} \begin{bmatrix} C_d^{-1} \\ C_d^{-1} (\mathbf{GL}_{bd}) \end{bmatrix} \\ &= \frac{i\omega + C_d^{-1} \mathbf{GL}_{bd}}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \begin{bmatrix} C_d^{-1} \\ C_d^{-1} (\mathbf{GL}_{bd}) \end{bmatrix} \\ G(s) &= \begin{bmatrix} \frac{C_d^{-2} \mathbf{GL}_{bd} + i\omega C_d^{-1}}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \\ \frac{C_d^{-2} \mathbf{GL}_{bd}^2 + i\omega C_d^{-1} \mathbf{GL}_{bd}}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \end{bmatrix} \end{aligned} \quad (\text{A.18})$$

So, the transfer function for the diffusive node is given by,

$$G_d(s) = \frac{C_d^{-2} \mathbf{GL}_{bd}^2}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} + i\omega \frac{C_d^{-1} \mathbf{GL}_{bd}}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \quad (\text{A.19})$$

The gain at the diffusive node is the absolute of the transfer function and is given as:

$$\begin{aligned} abs(x + iy) &= \sqrt{x^2 + y^2} \\ A &= \sqrt{\left(\frac{C_d^{-2} \mathbf{GL}_{bd}^2}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \right)^2 + \left(i\omega \frac{C_d^{-1} \mathbf{GL}_{bd}}{(i\omega)^2 - (C_d^{-1} \mathbf{GL}_{bd})^2} \right)^2} \\ &= \sqrt{\frac{(C_d^{-1} \mathbf{GL}_{bd})^4 + (-\omega)^2 (C_d^{-1} \mathbf{GL}_{bd})^2}{\left(-\omega^2 - (C_d^{-1} \mathbf{GL}_{bd})^2 \right)^2}} \end{aligned}$$

$$\begin{aligned}
A &= \sqrt{\frac{(C_d^{-1}GL_{bd})^4 + 2\omega^2 (C_d^{-1}GL_{bd})^2 + (\omega^2)^2 - \omega^2 (C_d^{-1}GL_{bd})^2 - (\omega^2)^2}{(\omega^2 + (C_d^{-1}GL_{bd})^2)^2}} \\
&= \sqrt{\frac{\left((C_d^{-1}GL_{bd})^2 + \omega^2\right)^2 - \omega^2 \left((C_d^{-1}GL_{bd})^2 + \omega^2\right)}{\left((C_d^{-1}GL_{bd})^2 + \omega^2\right)^2 - \left((C_d^{-1}GL_{bd})^2 + \omega^2\right)^2}} \\
&= \sqrt{1 - \frac{\omega^2}{(\omega^2 + (C_d^{-1}GL_{bd})^2)}} \\
&= \sqrt{\frac{(C_d^{-1}GL_{bd})^2}{\omega^2 + (C_d^{-1}GL_{bd})^2}} \\
A &= \frac{GL_{bd}}{\omega C_d} \frac{1}{\sqrt{1 + (GL_{bd}/\omega C_d)^2}} \tag{A.20}
\end{aligned}$$

The phase shift at the diffusive node is the argument of the transfer function and is given by,

$$\begin{aligned}
arg(x + iy) &= \tan^{-1} \left(\frac{y}{x} \right) \\
\phi &= \tan^{-1} \left(\frac{\omega C_d^{-1}GL_{bd}}{-\left(\omega^2 + (C_d^{-1}GL_{bd})^2\right)} \div \frac{(C_d^{-1}GL_{bd})^2}{-\left(\omega^2 + (C_d^{-1}GL_{bd})^2\right)} \right) \\
&= \tan^{-1} \left(\frac{\omega C_d^{-1}GL_{bd}}{(C_d^{-1}GL_{bd})^2} \right) \\
\phi &= \tan^{-1} \left(\frac{\omega C_d}{GL_{bd}} \right) \tag{A.21}
\end{aligned}$$

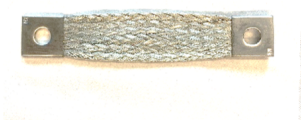
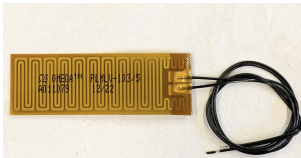
B

Test Documentation Sheet

B.1. Laboratory Test

The hardware and software used for the laboratory tests are summarised in Table B.1.

Table B.1: Test documentation sheet for laboratory test

Hardware Used			
Item	Images	Name	Purpose
Aluminium Plates		AL EN-AW-1050A	To attach the strap to the plates
Cu Strap		Ti coated Cu Strap	Test Item
Heater		PLMLV 103/5	To supply heat in the test setup
Power Supply Unit		RSPD3303C	Programmable power supply for a sinusoidal heat input.

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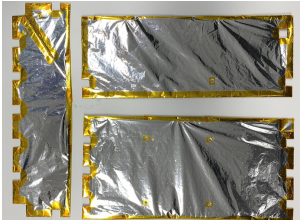
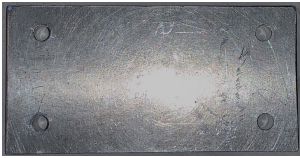
Table B.1: Test documentation sheet for laboratory test (Continued)

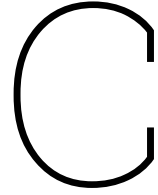
Temperature Sensors		PT100	For higher accuracy temperature measurements
		Thermocouple Type T	For additional temperature measurements in the setup
Data Acquisition System		USB - Temp - AI	Data acquisition device for the heater voltage and PT100s
		USB-TC	Data acquisition device for the thermocouples
Software/Python Packages Used			
Name	Purpose		
InstaCal	To configure the DAQ systems		
MCC	To read the DAQ data and store it in data frame using Python.		
PyVisa	To read/write the PSU using Python		

B.2. Vacuum Chamber Test

Most of the hardware from the laboratory tests was also used in the chamber tests. The additional equipment specific to the latter is detailed in Table B.2.

Table B.2: Test documentation sheet for chamber test

Hardware Used			
Item	Images	Name	Purpose
MLI		VDA with Kapton outer layer	To reduce the radiative effects in the chamber
Heat Sink Block			For heat stabilisation in the test setup
Peltier Element		TEC1 12706	To extract heat from the test setup
Power Supply Unit		Delta Elektronika SM 7020	Power supply for the heater
		Delta Elektronika ES 030-10	Power supply for the Peltier element
Data Acquisition System		NI 9211 DAQ	Data acquisition device for the thermocouples
Software/Python Packages Used			
Name	Purpose		
LabView	To control the PSU		



Modelling Convection

This section summarises the empirical equations used in the TMM to model the natural convection of the controlled air chamber tests. These equations were sourced from Cengel [14].

Natural convection on a surface is influenced by the geometry, orientation, and properties of the surrounding fluid. Despite a well-understood mechanism, the complexity of fluid motion often necessitates empirical correlations rather than simple analytical solutions. Therefore, this study applied widely accepted empirical correlations to model convection.

The rate of convective heat transfer, $\dot{Q}_{conv}$, from a hot surface to the surrounding fluid is calculated using Equation C.1.

$$\dot{Q}_{convection} = hA(T_s - T_\infty) \quad (C.1)$$

where,

- h = average heat transfer coefficient [W/m²K]
- A = heat transfer surface area [m²]
- T_s = temperature of surface [°C]
- T_∞ = temperature of fluid [°C]

The average heat transfer coefficient is determined from the empirical equation for the average Nusselt Number, Nu , as shown in Equation C.2.

$$Nu = h \frac{k}{L_c} = CRa_L^n \quad (C.2)$$

where,

- k = thermal conductivity of the fluid [W/mK]
- Ra_L = Rayleigh number
- L_c = characteristic length of the surface [m]
- C, n = empirical constants

The empirical constants in this equation depend on the surface geometry and flow regime, which is characterised by the Rayleigh number, Ra_L , a product of the Grashof number, Gr_L , and Prandtl number, Pr . It is given by Equation C.3.

$$Ra_L = Gr_L \times Pr$$

$$Ra_L = \left(g\beta (T_s - T_\infty) \frac{L_c^3}{\nu^2} \right) Pr \quad (C.3)$$

where,

g = acceleration due to gravity, 9.81 [m/s²]

β = coefficient of volume expansion [K⁻¹]

ν = kinematic viscosity of the fluid [m²/s]

In the test setup, since all components are rectangular plates, convection was modelled for a horizontal plate with a hot upper surface. The characteristic length, L_c , for this geometry and orientation, is calculated as the ratio of surface area A to perimeter p , as shown in Equation C.4. This value was used for all elements.

$$L_c = \frac{A}{p} \quad (C.4)$$

The empirical constants C and n were determined based on the Ra_L range, from the Equations below:

$$Nu = 0.54Ra_L^{1/4} \quad \text{when} \quad 10^4 < Ra_L < 10^7 \quad (C.5)$$

$$Nu = 0.15Ra_L^{1/3} \quad \text{when} \quad 10^7 < Ra_L < 10^{11} \quad (C.6)$$

In the model, two boundary nodes were added, each with a fixed temperature of 30°C to represent the air. All fluid properties were calculated at the film temperature, which is the average of the surface and fluid temperatures. Since surface temperature varies throughout the test, as shown in Figure 5.5, the peak amplitude was used to calculate the film temperature. This approximation was good enough as the fluid properties don't vary significantly depending on changing film temperature.

However, to account for the dependency of Ra_L on surface temperature, the model recalculated Ra_L , Nu , and subsequently h after each time step using the updated surface temperature. This iterative approach ensured that the convective link with the air node gave a representative approximation of the natural convective effects of the test.