

To the Graduate Council:

I am submitting herewith a dissertation written by Vanshika Singh entitled “Additively Manufactured Nature Inspired Morphology for Redesign: Advancing Next Generation Energy Systems.” I have examined the final paper copy of this dissertation for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Doctor of Philosophy, with a major in Energy Science and Engineering.

Dr. Michael M. Kirka, Major Professor

We have read this dissertation
and recommend its acceptance:

Dr. Michael M. Kirka

Prof. Sudarsanam Suresh Babu

Dr. Stephanie C. TerMaath

Dr. Brett G. Compton

Accepted for the Council:

Carolyn R. Hodges

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(Original signatures are on file with official student records.)

Additively Manufactured Nature Inspired Morphology for Redesign: Advancing Next Generation Energy Systems

A Dissertation Presented for the
Doctor of Philosophy
Degree
The University of Tennessee, Knoxville

Vanshika Singh

December 2023

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|| Har Har Mahadev ||

To my lovely parents Mrs. Hema Singh & Mr. Rakesh Singh

To my world best siblings Shri & Sid

Acknowledgements

Research often involves quantifying things, while in the real world, the guidance, learnings, and help we receive for performing research is qualitative, which is immeasurable, and no particular words justify it. Firstly, I thank my advisors, Dr. Michael Kirka (Mike) and Prof. S.S. Babu (Suresh). Thanks to Mike for funding me, providing me with intriguing research projects, motivating me to go to conferences, providing me an environment to collaborate with multiple colleagues at ORNL, and asking me difficult questions. I am privileged and honored to be mentored by Suresh, as he is and will remain very influential in my growth as a researcher and person. I am thankful to him for the uncountable thorough meetings with deep technical discussions, for teaching me how to dig deeper, learn to get comfortable in the uncomfortable domain, take the lead in the research, and for pushing me to take risks and add calculations to those. Above all, Suresh has been very kind in supporting me through crests and troughs, no matter the situation. Secondly, I greatly thank my committee members, Dr. Stephanie TerMaath and Dr. Brett Compton, for always sparing their time for our periodic meetings and challenging me with difficult questions, making me a better student every day.

The first project of this research was sponsored by the U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Advanced Manufacturing Office, and Office of

Fossil Energy under contract DE- AC05-00OR22725 with UT-Battelle, LLC. The work was supported by the US DOE, Office of Fossil Energy, under the award DE-FE-0031831. The second project was sponsored by the US Department of Energy, Office of Energy Efficiency and Renewable Energy (EERE), Advanced Materials and Manufacturing Technologies Office (AMMTO) under contract DE-AC05-00OR22725 with UT-Battelle LLC. The research was performed partially at Oak Ridge National Laboratory's Manufacturing Demonstration Facility, an Office of Energy Efficiency and Renewable Energy user facility. I would like to thank the Bredesen Center for Interdisciplinary Research and Graduate Education (University of Tennessee - Oak Ridge National Laboratory) for the Energy Science and Engineering Fellowship. Additionally, a great thanks to the entire business administration of Bredesen Center and the Mechanical, Aerospace, and Biomedical Engineering Department (UT Knoxville).

A great thanks to Dr. Prashant Singh, who has been instrumental in the latter part of my Ph.D., for having multiple discussions related to heat transfer, guiding me through the right literature, and reviewing my experimental setup. Many thanks to Dr. Fred List III for helping me realize the experimental setup from paper to laboratory. He stopped me various times before making any uninformed decisions and taught me the fundamentals of hardware setup design. Thanks to Dr. Anand Kulkarni from Siemens Corporation for funding the first part of my Ph.D. and for being very helpful. Thanks to Dr. Ramanan Sankaran from ORNL for his time and several discussions on understanding the nuances of fluids from a computational standpoint.

I have big thanks to Chase Joslin, Jake Dvorak, and Paul Brackmann for helping me achieve my goal on time by listening to all my requests. I would like to thank Suresh's group students: Dr. Kevin Sisco - for having various conversations discussing each other's research,

providing feedback, and learning from each other; Stephen Eilertson - for providing me with critical and supportive feedback for all my presentations and being one of the most active members of our group discussions; Brandon Solsbee - being very kind and helping me with little things. I would thank Suresh's former students, primarily Dr. Sujana Chandrasekhar – I knew her the longest, helping me explore the basics of being a Ph.D. student, helping me with adjusting to the new country, and having so many long technical discussions. I would like to thank Dr. Sabina Kumar (she's really fun and helped me with staying on track and timeline for my Ph.D. – a great motivator), Michael Haines (who helped me Schlieren Optics experiments, having tech and non-tech discussions on various topics), Dr. Bryan Lim (his last-minute feedback on my writing and presentation helped me with refinement of my research communication). I thank Dr. Pikee Priya for going on long walks, listening to my daily research hurdles or achievements, and having brainstorming discussions. I would thank some of my friends in Knoxville, TN: Joshua Siktar, Rohit Ghanta, Anirudh Ramesh, Priyanshi Agarwal.

I hugely thank my family: my mum – for being the visionary, dreamer, decision-maker, and risk-taker; my daddy – one of the most intelligent men I know, my backup for everything; and I learn diligence and ethics from him; my brother Siddharth – my biggest support, our encyclopedia; and finally my special thanks to my elder sister Shrilekha – my path-maker, leader, teacher, and my source of positivity.

Abstract

To meet cleaner energy goals and increasing demand, energy systems such as gas turbines and power plants are required to be operated under harsh loading conditions like higher temperatures and pressures, fluctuating loads, and corrosive environments. Advanced manufacturing techniques such as additive manufacturing (AM) have put us on the trajectory for next-generation system designs, allowing complex geometries and high-temperature alloys with tailored material properties. We need new and systematic design philosophies to use AM's unique characteristics prudently. For a given functionality, nature tends to provide similar solutions in animate and inanimate structures. We propose to take inspiration from nature's repetitive solutions and implement them for real-world applications for the next-generation system.

In this dissertation, we first identify a structured approach and consequently categorize these repetitive solutions as a class of solutions for a given functionality. The class of solutions is devised and optimized while solving two real-world problems. The first problem is ash deposition, which is common for different components in power plants and gas turbines. To reduce deposition in power plants on superheater tubes, we propose to leverage an aerodynamically shaped tube, one of the solutions from the classes offered by nature, introducing a macro-level morphology change leveraging the laser powder bed fusion method.

The proposed shaped tube performed better than circular tubes in providing more laminar flow around the tube, thus reducing the propensity of ash deposition. Additionally, we also showed how to reduce the circumferential thermal stress, potentially improving the tube's creep life. The second problem is introducing cooling channels for high-pressure turbine applications. Similar to solutions existing in nature, where heat exchange is enhanced by modifying the surface texture, we propose to modify the surface morphology of the internal cooling wall by introducing multiscale micro features enabled through laser powder bed fusion to enhance the surface area and thus improve heat transfer. The proposed design demonstrates designs that are better than AM internal cooling channel designs previously discussed in the literature. We recommend emulating nature's governing approach for repeating its solution in varying-sized creatures through certain laws. We suggest generalizing solutions for differently sized components by identifying relationships between design parameters and component size. To our knowledge, this is one of the first works that attempts to codify and validate a nature-inspired design strategy for high-temperature applications.

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Chapter 1

Introduction

1.1 Motivation

1.1.1 Evolution of Energy Systems

“I sell here, sir, what all the world desires to have—POWER,” as said by Matthew Boulton, partner of James Watt to produce Watt’s innovative steam engines, quoted for the first time in a 1791 book, *The Life of Samuel Johnson* by James Boswell. Since then, the human desire to generate and distribute energy has led to four industrial revolutions and significant transformations in manufacturing tools and processes. Thermal efficiency has gone up to 62%-64% in the 600-1,700 MW electrical output range through gas turbine combined cycle (GTCC) power plants, which offer double the efficiency of a power plant available 70 years ago. In 2021, the GTCC fleet was responsible for 32% of total electricity generation, ranking first (1,326,278 gigawatt hours [GWh]), followed by coal-fired generation (22%) [14]. Natural gas accounted for 34% of total annual energy-related CO₂ emissions, whereas coal contributed 21% of total annual energy-related CO₂ emissions [15]. We aspire to monotonically enhance

the efficiencies of our energy systems while meeting climate control objectives. As a component of decarbonization efforts, fossil-based power plants for electricity generation will remain crucial in the next few decades to support renewable energy.

We need to design next-generation energy systems for lower emissions and improved efficiencies. These next-generation energy systems are expected to operate under harsher conditions such as high temperatures (600°C-1,500°C), pressure, fluctuating loads, and corrosive environments. To thrive under these conditions, the systems will require substantial changes in design such as geometrical shape, materials, mechanisms, and manufacturing capabilities. Advanced manufacturing techniques have enabled cutting-edge technologies. One technique that has explicitly emerged in the last decade is "additive manufacturing," also called 3D printing, which has enabled the development of promising high-temperature metallic alloys with complex design features that were otherwise impossible. Given such growing and unique capabilities, one might ask, "How can we leverage such a method to our benefit for the next-generation system?" Before we divulge the answer to this question, we must understand the current capabilities and demand from interested organizations and industries.

1.1.2 Role of Additive Manufacturing in Energy Systems

Additive manufacturing (AM), a layer-by-layer deposition process [16], may enable innovations resulting in low-emission energy systems. For varied functionalities and length scales, metal-based AM modalities such as 1) laser-powder bed fusion (L-PBF), 2) electron beam welding (EBM), 3) direct energy deposition (DED), and 4) binder-jet AM (BJAM) are used. These methodologies demonstrate promising solutions for industrial applications and have been deployed commercially [17]. The two key advantages of these techniques that lead to

their high demand in energy systems are 1) enabling complex geometrical features at different length scales and 2) processing of high-temperature alloy development, allowing complicated yet desired performance.

1.1.2.1 Structural Designs Leveraging Additive Manufacturing

We can exhaustively exploit AM-based techniques to produce complex geometrical parts to fully attain an efficient energy system. Complex design features to produce efficient and optimized metallic components have been explored by various industries such as Siemens, GE, and NASA’s Jet Propulsion Laboratory (as shown in Figure 1.1). Most of the interest in components that were additively manufactured relates to characteristics such as 1) part consolidation and reduced weight as well as material leveraging design for AM (DfAM), 2) novel heat exchanger designs with complex features that are otherwise not possible, 3) integrated cooling channels via AM that are traditionally difficult to produce.

1.1.2.2 Materials

There has been evidence of successful crack-free additive manufacturing of high-temperature non-weldable super-alloys, such as IN738, Mar-M 247, and Ti-based alloys [17, 18]. In addition, there are recent developments of refractory materials having high melting points ($> 6000^{\circ}\text{F}$), such as highly dense crack-free tungsten via the EBM process [5]. Also, as demonstrated by Plotkowski et al. [19], AM-based techniques empower site-specific tailored microstructure, leading to controlled material properties in high-temperature alloys. This material manufacturing was made possible by engineering novel scan strategies, manipulating AM process parameters, simulating the thermal model for the AM model, and using in situ non-destructive evaluation techniques. These advancements clearly illustrate growing

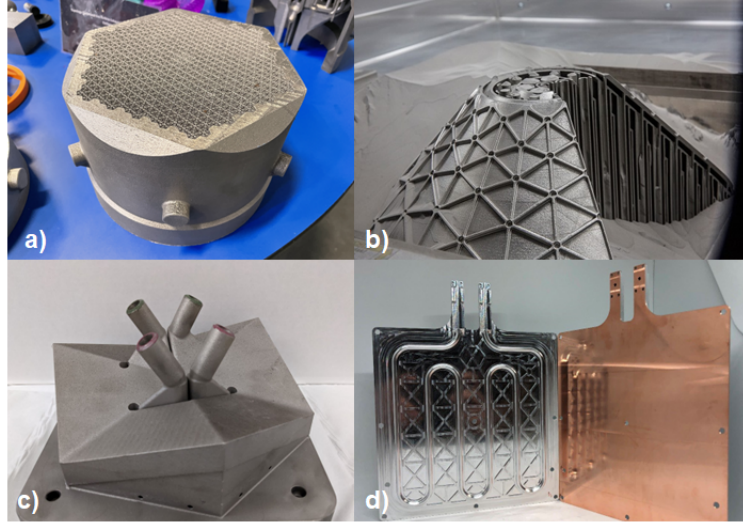


Figure 1.1: a) Topology-optimized heat exchanger by Siemens [1]; b) 3D printed oxygen-hydrogen micromixer by GE Additive and Shell [2]; c) GE Research's 3D printed subscale grape-shaped heat exchanger prototype [3]; and d) 3D printed radiator for a satellite that combines aluminum and copper for uniform heat spread across the face [4].

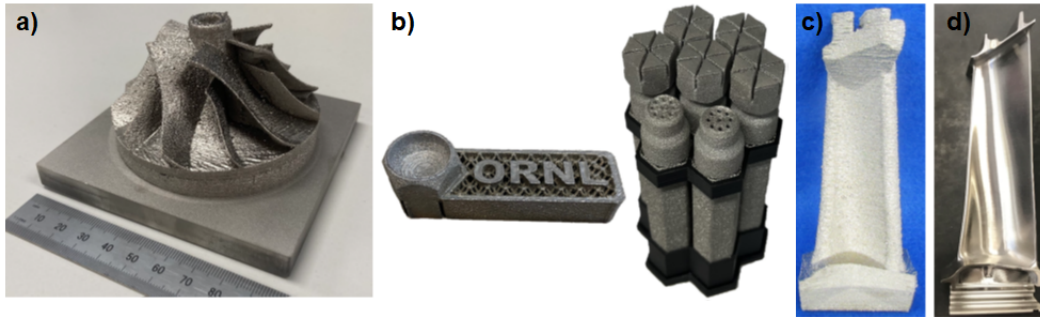


Figure 1.2: a) Additively printed turbo compressor impeller in Ni-based alloy using EBM; b) additively printed pure tungsten complex geometry using EBM [5]; c) additively printed IN738 turbine blades using EBM; and d) additively printed IN738 turbine blades after machining.

interest by industries and organizations in developing novel metallic alloys that can operate under harsh boundary conditions, inhibiting early failure. Here, early failure can be due to long-time exposure to high temperatures (creep), high thermal gradients inducing detrimental stresses, high internal pressures, and other extreme environments.

1.1.3 Design for Next Generation Energy Systems

With advancements in AM, we restate the question: how can one pivot around AM (flexibility with material and geometrical design features) such that we make efficient and better components operating under extreme environments? We must envision how to design components for next-generation energy systems. Lately, there has been limited progress in the advent of design strategies for such applications utilizing AM modalities. Nature evolved over millions of years and often demonstrates complex, non-trivial design solutions. Characteristics that make nature’s design intricate are shapes, sizes, distribution, length scales, materials, heterogeneity, dynamic movements, and many more. Despite the multitude of variables, implicit structuring may exist among the design solutions offered by nature. Through this dissertation, we identify and refine a new design strategy inspired by nature, redesigning components operating under high temperatures enabled through AM.

1.2 Aims and Scope

The primary objective of this research is discussed as follows:

1. To develop a new design strategy for high-temperature components to advance next-generation energy systems utilizing AM modalities.

- (a) Nature demonstrates unique and complex solutions in various animate and inanimate structures for given functionalities through intricate morphologies, materials, and dynamic movements. Can we identify the underlying structure in the solutions that exist in nature and utilize this knowledge further for engineering real-world design problems?
- (b) To apply and refine the proposed design strategy for component redesign operating under harsh boundary conditions for power plants and gas turbine applications. We study the following two problems:
 - i. Ash deposition issue: What geometrical variation might assist in avoiding early failure of superheater tubes subjected to undesired particle deposition on their surface? We aim to investigate design solutions through the proposed approach and conduct a detailed comparative analysis with the baseline design.
 - ii. Internal cooling channels: Integrated cooling channels needed for the efficient operation of gas turbines require advanced cooling schemes. Our objective is to explore design features through the proposed strategy, perform a detailed study producing final thermal-hydraulic characteristics, and compare with designs found in the existing literature.
- (c) Once we understand and implement the underlying commonalities of nature's design through real-world design problems, it is crucial to understand the reasons or laws governing those commonalities. Hence, can we recommend a structured approach based on the laws we identify for generalization or standardization of the proposed strategy for the engineering design we have put forth?

1.3 Dissertation Structure

The dissertation is structured into seven chapters, outlined as follows:

- Chapter 2: Briefly discusses the two redesign problems (i.e., ash deposition in superheater tubes and advanced cooling requirements for high-pressure turbine applications). Discusses the existing design philosophies and why we need to go beyond those. Proposes a new design philosophy inspired by nature.
- Chapter 3: Discusses the mechanism behind ash deposition issues on superheater tubes and the relevant literature review. Proposes a new geometrical design with variants based on the proposed design strategy. Investigates the effect of nonuniform ash presence on the loading conditions for hoop stress. Develops an understanding of the role of geometry through computational methods and Schlieren imaging for drag reduction. Develops an overall metric accounting for drag and hoop stress development.
- Chapter 4: Discusses traditional and additive convective design enhancement features. Using the proposed design approach, proposes a heat transfer enhancement feature. Discusses the coupon design, sequence of testing, and its manufacturing. Surface roughness characterization is performed using x-ray computed tomography and scanning electron microscopy.
- Chapter 5: Discusses the experimental setup development that can test up to 200°C. Discusses the effect of the unique surface characteristics enabled through AM for heat transfer and fluid flow behavior.

- Chapter 6: Discusses potential pathways for the current design problem using other solutions offered by the proposed design strategy. Generalizes the proposed design strategy and concludes the overall design philosophy.
- Chapter 7: Summarizes the key findings throughout the dissertation.

Chapter 2

Proposed Design Strategy for New Energy Systems

2.1 A Brief Introduction to Problem Statements

Two real-world problems have been studied in detail in the current research. These components belong to gas turbine and power-plant systems needing new cooling schemes and undesired deposition reduction. The following subsections briefly discuss these two problem statements requiring advanced design change:

2.1.1 Ash Deposition on Heat Exchanger Tubes in Power Plants

With increased alternative energy technologies, the coal fleet remains critical for electricity generation because it is reliable, cost-effective, and not dependent on real-time delivery. As the coal fleet repositions in the energy sector, it requires advanced development of existing components for harsher loading conditions. To optimize the carbon emissions arising from

coal-fired power plants, biomass, considered CO₂-neutral and a sustainable fuel, is co-fired with coal. Biomass has been shown to reduce CO₂, SO_x, and NO_x emissions [20]; however, it contains significant amounts of contaminants, including chlorine and alkali elements such as potassium and sodium. These contaminants increase the propensity of ash deposition on the shell and tube heat exchanger, which can be detrimental to the component’s life [21, 22]. One such tube is a superheater tube, stipulating considerations for its redesign for increased life.

The primary function of these tubes is to increase the temperature of the saturated high-pressure steam to superheated steam by increasing the steam temperature by 35°C-50°C. Because of high-pressure steam passing through, the tubes experience hoop stresses on their walls. With the emergence of new-generation fossil power plant systems, these tubes operate under temperature and pressure loading conditions that are higher than those of traditional boiler tubes (~610°C and 25-30 MPa) [23]. To superheat the saturated steam, the external surface of the tube is exposed to hot combustion flue gases with a temperature ranging from 800°C to 1,200°C [24]. These gases also contain undesirable ash particles, which deposit onto the tube surface (see Figure 2.1), and their shape and thickness gradually change over time. Owing to the presence of undesired chemicals (discussed in section 3.1), local high-temperature corrosion occurs externally on superheater tubes, causing thinning of the tube and leading to early creep failures.

Additive manufacturing that allows complex design features can mitigate these issues by providing geometric solutions to reduce ash deposition concurrently, improve heat transfer, and enhance creep life. However, the question remains: what design features can be implemented for such requirements?

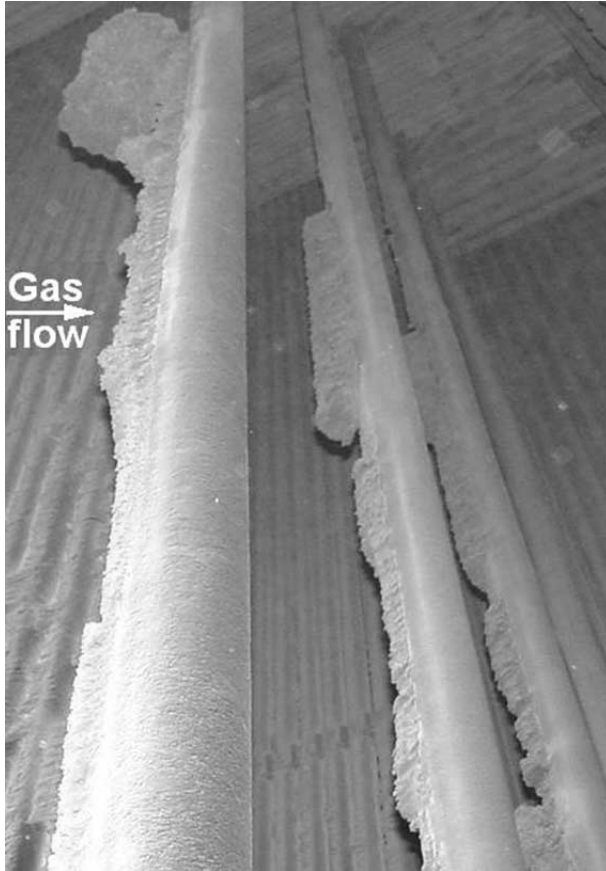


Figure 2.1: Ash deposits on real superheater tubes. [6]

2.1.2 Need for Advanced Cooling Schemes for HPT Components Using Additive Manufacturing

Gas turbines are crucial to energy systems and are used in land and marine-based power generation systems and aircraft propulsion. High-pressure turbine (HPT) inlet temperature must be maximized to increase a gas turbine’s thermal efficiency and power output. Figure 2.2 demonstrates the ideal performance curve for 100% efficiency with no leakage or cooling flows. The figure also shows that the HPT section needs high-temperature materials and enhanced cooling mechanisms to increase the specific core power. Conventionally, cooling channels are designed and manufactured in HPT rotating and nonrotating components such as nozzle vanes and turbine blades. Different cooling design strategies have been studied before, and their features have been summarized in section 4.2.

A full-scale AM-built rotating high-pressure turbine blade has been successfully demonstrated using high γ' Ni-alloy. Recently, Murray et al. [25] fabricated high-strength, defect-free Co-Ni-based superalloys through AM, laying a path for developing similar alloys operating under extreme conditions.

One of the components in the hot section, called the “tip shoe” or “tip shroud,” is a potential candidate to be printed additively. These are nonrotating segmented shrouds assembled circumferentially around the high-pressure turbine rotor to 1) provide a flow path to primary gas flow and 2) maintain a desired clearance between the rotating blade and itself. Based on the prior study for enabling AM-built cobalt-nickel-based superalloy, it is envisioned that the tip shoes will operate under higher temperatures. However, this comes with a penalty; with increased primary gas flow temperature, the tip side of a tip shoe is exposed to an even higher temperature than the other side of the tip shoe. This might cause

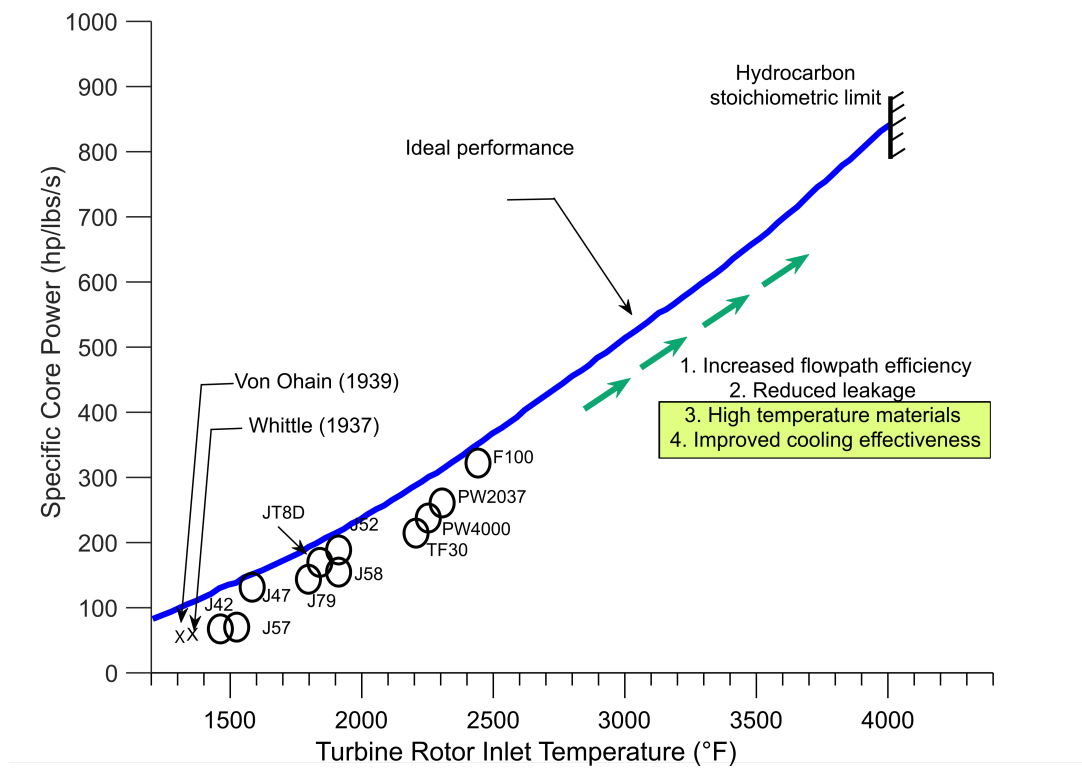


Figure 2.2: Increasing turbine rotor inlet temperature improves cycle power output. Reproduced from Han et al. [7]

a high thermal gradient radially and increased bulk temperatures of the component, inducing thermal stresses. This requires the development of integrated micro- or mini-cooling channel designs possible only through AM.

2.2 Current Designing Methods

Knowing the actual meaning of engineering design is essential before discussing the adopted methodologies. As described by Evbuomwan et al.[26], there is no one definition. Several authors suggested different definitions that highlight specific keywords and phrases sharing strong associations with design. Some include needs, specifications, constraints, scientific principles, functions, creativity, and manufacturing. Design methodologies use these features to solve problems encompassing routine designs, redesigns, nonroutine designs, and creative designs. This section discusses our current methodologies to design for real-world applications or explore newer designs at a low Technology Readiness Level (TRL). We further address the gap and available opportunities, proposing a design methodology (section 2.3).

2.2.1 Engineering Principles and Guidelines-Based Methodologies

Fu et al. [27] reviewed 66 research documents highlighting the three broad pillars leveraged in engineering design, which are 1) principles, 2) guidelines, and 3) heuristics. Principles, defined as "fundamental law or rule" derived from past experiences and empirical evidence, assist in achieving successful designs. Similarly, guidelines are established based on extensive past experiences for given criteria, leading to successful solutions. Cognitive skills, experience, expertise, fixation, and creativity influence heuristics or an intuitive approach. This leads to satisfactory solutions but not optimal solutions. Principles and guidelines

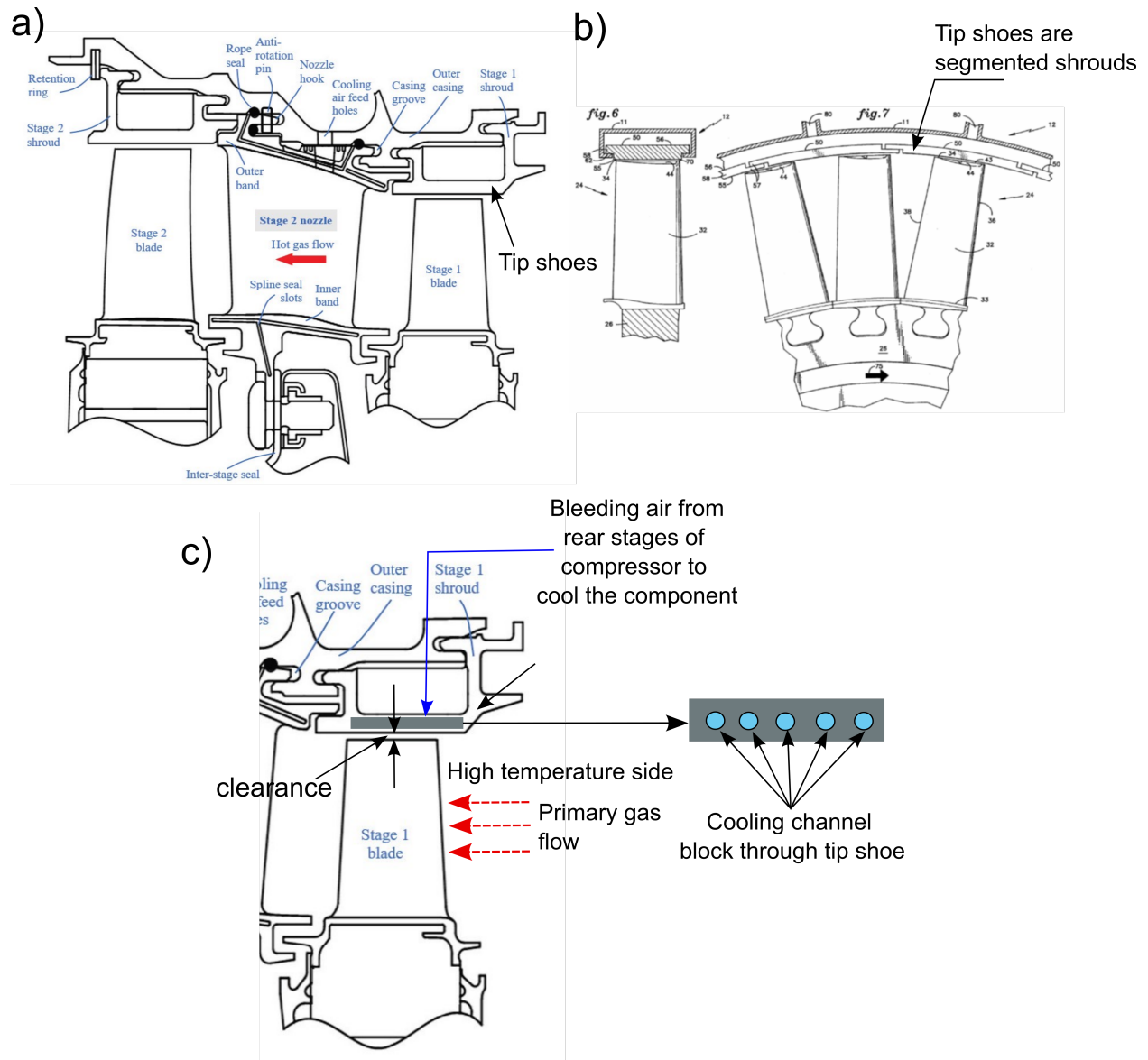


Figure 2.3: a) Partial view of a gas turbine highlighting the location of the tip shoe compared with adjacent components [8], b) shown in 3D demonstrating tip shoes as segmented shrouds (only for representation purposes) [9], and c) showing region of interest for introducing cooling channels within tip shoes [8].

in integration with an engineering analytical foundation lead to the formation of various "design practices" or "best practices." Such practices have been crucial in several generations of product developments in industries. Based on past experiences, failure studies, and case studies, industries such as General Electric (GE) [28] and Boeing [29] established broad design philosophies for their gas turbine and aircraft frame development, respectively. Some include 1) evolutionary approach, 2) scaling up or down, and 3) thorough preproduction development. Through the evolutionary approach, GE established a family of products (such as a set of axial compressors) over decades without compromising on the proven reliability of the existing designs for improved efficiency or increased capacity. This is primarily done by locally modifying the geometric parameters or material selection or by resizing the component. Another principle, the scaling of the product line, was leveraged by GE to produce 25-200 MW, maintaining the geometric similarity. Last, manufacturability and producibility using trial and error experimentation and feedback from field experience have been crucial in component design.

Back in 2005, when metal AM was not an established technique, Boeing published their design philosophy [29] proposing to leverage monolithic machined or welded integrated structures for reducing the part count, hence easing the assembly. With AM advancements, one can think of part consolidation by merging numerous parts into one through redesign. However, AM component design must account for the manufacturability constraints during printing. Because of AM's unique capabilities and manufacturability constraints, numerous works suggested design principles, heuristics, and guidelines for AM [30, 31, 32, 33, 34]. Whereas several of these focused on formalizing design knowledge for AM, Tang et al. [31] suggested a theorizing framework for function integration and structural optimization in the early design phase. It is proposed to leverage AM's ability to control design at

different length scales. However, this work remains theoretical and requires further real-world implementation.

In summary, engineering design principles and a foundational analytical framework allow us to meet the imposed target requirements over a period of time, although with a small derivative in performance improvement. Thus, we still require a methodology that can be used for real-world applications with significant benefits in performance.

2.2.2 Optimization-Based Computational Methodologies

In the second method, structural optimizations for size, shape, and topology are potential solutions to a significant change in design. The final geometry through topology optimization is reasonably complex and will require AM to produce the parts. There has been increasing interest in adopting topology optimization in industries. There is a surge in research funding of more than \$3.1M (USD) by ARPA-E only [35] for enhancing topology optimization coupled with AM for components that operate at extreme environments (operating temperature ranging from 800°C to 1,100°C and pressure ranging from 17 to 25 MPa). Interested parties such as Boeing, Siemens, Raytheon, and academic institutions propose to develop comprehensive multiphysics (accounting for structural, fluid flow, and thermal aspects) and multimaterial capabilities.

Though there is extensive interest in topology optimization for energy systems, some challenges exist, such as material property variation at the microscale with changes in geometry and surface finish roughness. These unknowns, as discussed in the National Academies of Science, Engineering, and Medicine workshop on exploiting advanced manufacturing capabilities [36], persist and are currently seen as insurmountable challenges. Current modeling software limitations must also be considered. For instance, capturing

AM roughness in computational fluid mechanics is currently computationally expensive [37]. Thus, it is not straightforward to model for designing additively manufactured heat exchangers or cooling channels. Therefore, topology optimization might provide theoretically optimal solutions without AM surface roughness modeling. However, it might not necessarily produce the same expected output after being additively manufactured. Apart from this, topology optimization poses another challenge regarding solving multiple physical processes. It is complex to combine different modeling packages into a single package that satisfies all physical modeling needs [36].

Because we plan to leverage AM operating under extreme environments for next-generation energy systems, it is challenging to integrate this design methodology in the early design phase.

2.2.3 Emerging Bio-Inspired Methodologies

The third emerging method, i.e., bio-inspired or biomimicked design, is also crucial to assess. The term "biomimicry" was coined in the 1950s by Otto Schmitt [38]; however, biomimicry has been historically used for over 3,000 years to produce improved solutions in various fields ranging from aircraft to adhesive tape. Vincent et al. [38] listed limited examples of such design translations from biology to technology.

With growing interest in biomimicry, there has been substantial research in formulating a systematic and organized approach in the form of a toolkit or software for identifying better analogies for bio-inspired designs. Numerous methods such as "BioTRIZ" [38], "design-by-analogy" [39, 40], "directed intuitive approach" [41], and many more as discussed by Fu et al. [39] and biomimicry pedagogy [42, 43, 44, 45] have been proposed. Further theorization of biomimicry was done by Ilieva et al. [46] and Kiakojouri et al. [47] with aims to produce

a design framework for sustainability and robustness, respectively. However, this research remains in the conceptual design phase and requires further real-world implementation for complex boundary conditions, including manufacturing.

Because energy systems perform under higher temperatures, pressures, and corrosive environments than nature, implementing bio-inspired solutions has been limited. As per Scopus and Web of Science search analysis, if we use keywords "nature-inspired designs in gas turbines" or "nature-inspired designs in power plants" and similar related terms, the number of publications is limited [48, 49]. Nevertheless, finding the best analogies for extreme boundary conditions is difficult or limited because nature operates under different boundary conditions than ours [50]. Despite this, industries such as GE and other research organizations have been interested in bio-inspired designs. GE proposed and tested two heat exchangers: 1) a grape-shaped 3D printed heat exchanger tested at 900°C [3], and 2) a lung-inspired 3D printed heat exchanger able to operate at a temperature exceeding the current state-of-art heat exchangers by 200°C [51].

In summary, bio-inspired design solutions have found successful applications in various fields; however, more research in high-temperature applications through AM is needed. The inability to identify the accurate analogy and translate it to our region of interest requires a new thought process that can provide a near-optimal solution for the early design phase.

From the above discussion, we conclude that though the existing design philosophies are beneficial in providing satisfactory solutions, they lack the usefulness to be leveraged exhaustively for next-generation energy systems. Thus, the overarching question is as follows: can we create a new design strategy that introduces a significant change in design, leading to complex geometry performing better than existing design solutions in terms of efficiency and/or life?

2.3 Proposed Design Strategy: "Class of Solutions"

Nature spent ~ 3.8 billion years to come to its current bio systems through the evolutionary process and often can sustain multiple and conflicting boundary conditions. The functionality often influences nature's design. Therefore, it exhibits repetitive solutions to various animate and inanimate systems for a particular function; for instance, a form of tree, snowflakes, river basins, and lungs are similar where flow is typical in each design-optimal criterion. For a given set of functionalities, one can designate solutions from nature into a set of classes called "class of solutions." Here, the class of solutions is defined as the categorized ways nature meets a particular or combination of objectives in different species. We propose the following: "Unlike design-by-analogy approaches, to get inspired by the class of solutions generated in nature in the context of the design parameters improvements for components pertaining to energy systems." The classified nature's solutions are likely to be a function of one or a combination of the following categories:

- Morphological solutions: The form (i.e., the shape and size and their distribution at different length scales) is crucial to designs in nature. The shape and surface characteristics are the results of optimizing a design parameter, which is important to the structure's functionality. Examples of morphological solutions are dendritic structures for optimized flow, such as in lungs or river basins [52]; hair or fur for insulation purposes in various-sized animals [10]; mammalian eyelashes protect the eye by minimizing airflow across the eye surface [53]; hierarchical surface characteristics for self-cleaning purposes [54]; and grooved structures or riblets for reduced drag [55]. It would be interesting to understand the morphological-based solutions, implement them for our design functionalities, and generalize one solution for varying-sized components.

- **Material design:** Besides shape, material systems for animate and inanimate structures are vital to meet design criteria. Nature exhibits unique mechanical properties for a given functionality, minimizing the material usage. Fraction and structure of the individual building block greatly affect the mechanical property. Examples include tendon, ligament, skin, cartilage, and bone containing the same fundamental components (collagen and elastin) but with varying fractions and arrangements leading to varying bulk mechanical properties [56]. Hybrid materials such as seashells, bones, and teeth are pervasive in nature [57]. Lately, efforts have been made to design metallic hybrid structural materials such as Co-Cr in IN718 matrix using blown-powder additive manufacturing [58]. This demonstrates one of the examples of efforts towards expanding the material design space beyond the existing metallic alloys. Additionally, in nature, local material property variation is available within one structure, which complements the shape. Inanimate structures such as paper wasps and termites show unique structures with varying local properties yet demonstrate a similar global output. Such localized property variation occurs mainly due to the construction algorithm followed by nature, leading it to meet local boundary conditions [13, 59]. The question remains: can we utilize the concept of construction algorithms to meet the local material property requirements by utilizing different individual building blocks for structural materials?
- **Compliant movements:** Last, the shape-changing ability in response to changes in boundary conditions in nature further optimizes the energy consumption and hence increases the system's efficiency. Animate materials or structures are sensitive to their environment and can adapt to fulfill their intended functions. A few examples include
 - i) human lungs changing air mass flow from 7 l/min to 68 l/min due to alveoli expansion

and contraction abilities, ii) cardiac muscles continuously working to activate the valves based on the pressure difference, iii) when a human walks, the muscle in the lower leg expands contracting the vein walls, leading to the valve's opening inside the veins [60]. However, engineering design has limited integration of such adaptive structures, which changes their shape or size with changes in boundary conditions.

In this dissertation, we will pursue morphological solutions offered by nature due to AM's current capabilities. The uniqueness of this research is identifying the morphological solution from the class of solutions and then implementing and studying them in detail for component redesign problems for high-temperature applications through metal AM.

Chapter 3

Modified Superheater Tubes through AM for Deposition Reduction

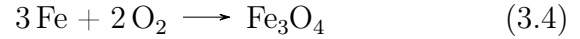
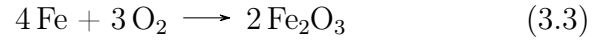
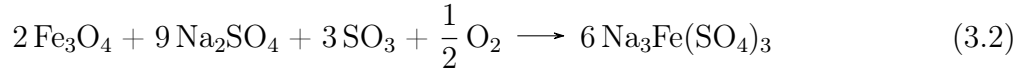
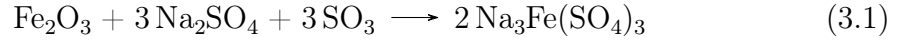
The work described in this chapter has been published in:

V. Singh, S. S. Babu, M. M. Kirka, and A. Kulkarni, “Novel Tube Design for Superheater Heat Exchanger Enabled Via Additive Manufacturing,” *Journal of Pressure Vessel Technology*, vol. 144, no. 4, p. 041407, Aug. 2022, doi: 10.1115/1.4054727.

3.1 Background and Theory

In section 2.1.1, we briefly discussed the deposition issue occurring in the superheater tubes in coal-biomass power plants. Here, we discuss how the non-uniform deposition of ash causes localized fireside corrosion and why it might lead to circumferential thermal loads across the tube. Understanding this phenomenon is essential because it might lead to non-uniform hoop stress distribution, causing localized stress concentration. Figure 3.1A shows the schematic representation of the superheater tube degradation mechanisms, leading to local thinning

of the tube and eventual failure. The flue gases, containing sulfates and chlorides such as Na_2SO_4 and NaCl , strike the tube surface and react with the ferric oxide or ferrous oxide layer to form low-melting alkali trisulfates in the presence of sulfur trioxide (SO_3). The melting temperature of these alkali trisulfates (554°C – 677°C) lies in the operating temperature of the superheater tube; therefore, liquid phase corrosion [61, 62] is enhanced. The overall chemical reactions are outlined in the following equations:



Reid et al. [63] suggested the presence of macro-galvanic couple across a tubular section due to SO_3 concentration gradients, hence setting up a gradient in corrosion potential (Figure 3.1D). A potential difference of 0.5 volts is calculated between metal in contact with solid sulfates at one point (just below ash deposits) and fused sulfates at another (around the 10 and 2 o'clock positions). This is consistent with the observed “wastage flat” near the 2 and 10 o'clock locations beneath the ash deposits (see Figure 3.1C) rather than around the 12 o'clock position. The location of wastage flats also supports the notion of high temperatures and easy access to SO_2 from the ambient [61, 63]. Around the 12 o'clock position, the deposit thickness is highest, and because its thermal conductivity is very low compared with the metal, it acts as an insulating layer. These boundary conditions should lead to lower temperatures at the 12 o'clock position compared with other locations where the ash deposit

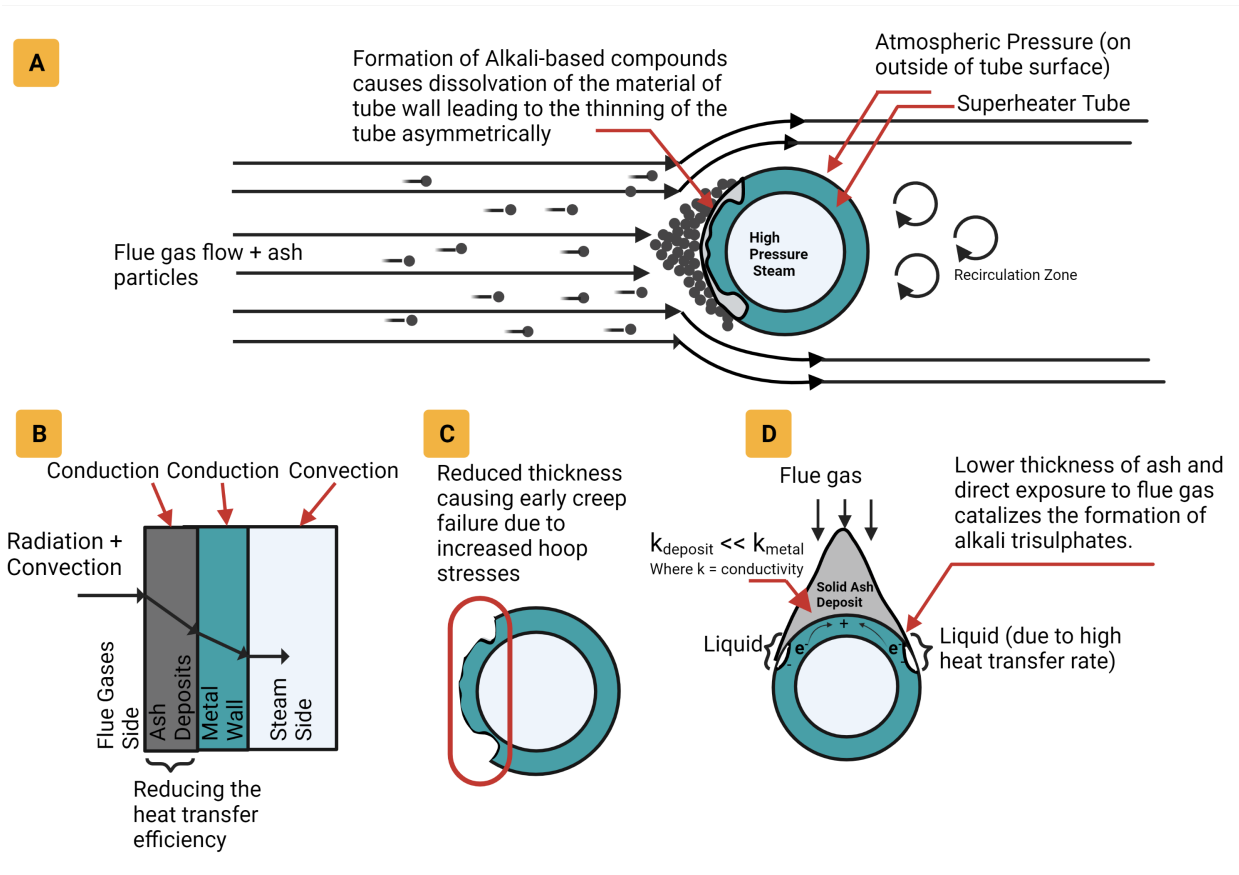


Figure 3.1: Schematic representation of a superheater tube containing high-pressure steam and getting exposed to flue gases and ash contaminants: (A) flow conditions that lead to ash deposition; (B) modes of heat transfer during operation; (C) typical degradation due to corrosion; and (D) mechanisms for preferential high-temperature corrosion due to ash deposition.

thickness is lesser [64]. These conditions also imply a circumferential thermal gradient around the tube. In addition to high internal pressure, the circumferential thermal gradient can further aggravate the tube’s hoop (circumferential) stress. Increased hoop stresses induce higher creep strain rates and reduced time to rupture. In fossil fuel power plants, failure due to creep or long-term overheating is the highest contributor ($\sim 23.4\%$ of failures) for forced outages [20]. Interestingly, 90% of the creep failures occur in the superheater and reheater tubes. In addition, the ash deposition and slagging reduce the heat transfer ability by 30 to $\sim 60\%$ (Figure 3.1B) [65]. Therefore, reducing the ash deposition on the tube surfaces is imperative while concurrently meeting mechanical design requirements. This overarching goal forms the motivation for this research.

3.2 Literature Review

Earlier studies for ash deposition on boiler tubes have primarily emphasized studying the chemical composition of ash, predicting ash growth through computational fluid dynamics (CFD) modeling, benchmarking the simulation model with experimental results, studying particle motion in flow and its interaction with the tube surface, and methods to reduce ash particles through external sources.

Constitution of Ash: Analysis of the chemical composition may help to predict the ash melting temperature. Vassilev et al. [66] showed the relationship between the chemical composition of coal ashes and the ash fusion temperature. Holubcik et al. [67] derived the empirical relationship to relate the chemical composition with the biomass ash melting temperature. Kim et al. [68] used the ASTM D1857 method and the TMA test to measure the ash fusion temperature to predict ash deposition characteristics.

Ash Growth Prediction Via CFD: CFD models have been used to predict the flue gases flow and concurrent particle deposition on the tubular surface [69, 70, 71]. Chapela et al. [72] performed CFD modeling using a full 3D-transient bed model and improved the accuracy of the prediction model by accounting for the boiler geometry. Riccio et al. [73] performed a CFD analysis of particle deposition and verified the calculations with experimental results with a 100 kW test rig facility. Further, particle-wall interactions have also been considered [74, 75]. In addition, based on the evaluation of ash compositions and validation of predictive models, external devices like soot blowers and monitors had been deployed for timely eroding or removal of the ash particles from the tubular surface [76, 77].

Tube Geometry Modification: Also, a few research publications focused on changing the tube shapes and their arrangement within the heat exchanger assembly for assessing fouling, heat transfer, and pressure drop. Bouris et al. [78, 79] and Tang et al. [80] proposed DDEFORM (Deposit DEtermined FOuling Reducing Morphology), elliptical, and circular-shaped tubes with different tubular arrangements for lignite utility power plants. Bouris et al. [79] studied these tubes for fouling, heat transfer rate, and pressure drop comparison employing numerical study and experimental methods like laser sheet visualization and laser Doppler anemometer (LDA). Tang et al. performed CFD analysis using Ansys® FLUENT software for examining ash fouling and thermal-hydraulic characteristics in a stacked elliptical tube arrangement. However, the authors did not study these tubes for their mechanical integrity. Also, the numerical study has been performed at a relatively higher Reynolds Number; for instance, Bouris et al., examined for Reynolds Number varying from 14,000 to 30,000, while Tang et al. studied for Reynolds Number ranging from 9,000 to 24,000. Li et al. [81] proposed a novel rhombic configuration for heat recovery steam generators for reduced ash deposition.

Research Gap: To our knowledge based on extensive literature review and patent search, limited work has been done on the superheater tube's geometry redesign to reduce ash deposition without affecting the mechanical hoop stresses' requirements or creep life.

3.3 Proposed Design & Hypothesis

Prior investigations explained many mechanisms for ash deposition, including inertial impaction, condensation, eddy impaction, thermophoresis, Brownian and eddy diffusion, and chemical reaction [64, 82]. Out of all these mechanisms, the dominant one is inertial impaction. This process causes larger particles ($>10 \mu\text{m}$) to overcome the drag force of the gas flow and deposit on the upstream of the tube surface. Earlier investigations have developed correlations between factors affecting the total mass deposition rate due to inertial impaction. Riccio et al. [73] presented a detailed mathematical formulation on the development of the total mass deposition rate:

$$m_{tot} = P_{tot} \frac{dm_p}{dA} \sin(\gamma) + m_{c,A} \quad (3.5)$$

where m_{tot} is total mass deposition flux ($\text{kg}/\text{m}^2\text{s}$), P_{tot} is the total fouling probability. Fouling probability is a function of the stickiness of particles (depending on particle chemical composition) and the stickiness of the tube surface (depending on vapor condensation). In the equation, m_p is the mass flow rate (kg/s) impacting the deposition area A (m^2), γ is the impaction angle at which the particle strikes the tube surface, and $m_{c,A}$ is the vapor condensation flux ($\text{kg}/\text{m}^2\text{s}$). Because of the condensation of particles on the tube surfaces, it forms an adherent layer on the tube surface, further enhancing the ash deposition. This depends on the characteristic length of the tube. In the circular tube, this length is equal to

the outer diameter of the tube. Because mass deposition due to condensation is relatively minimal compared with that due to inertial impaction [73], therefore, $m_{c,A}$ can be dismissed for further investigation of target geometry's role in ash deposition. From a geometrical standpoint, the impaction angle (γ) at which the ash particle strikes the tube could be varied. Since m_{tot} is directly proportional to the sine of γ , having a lesser impact angle is preferable.

3.3.1 Class of Solutions: Assisting Reduced Fouling

Like the deposition issue on heat exchanger tubes, fouling, adherence, and soiling of animals or plant surfaces are pervasive in nature because of undesired particles (or debris). Debris may be in the form of pollen, aerosols, bacteria, dust, or liquid droplets. In nature, there are several reasons to get rid of the particles. A few include 1) animal survival in highly dense particle regions [10] such as dew or fog, if accumulated in large amounts, can overwhelm small insects. For example, a single water drop of only 0.6 mm would weigh the same as the mosquito; 2) proper functioning of the body parts. For instance, if pollen grains are not regularly cleaned from the bees' surface, it would affect their flight control; and 3) hygiene purposes and longer life, for instance, disease transmission due to the accumulation of certain non-desirable particles such as mites and ticks. Previous work extensively documents how nature tackles fouling by removing and relocating accumulated debris. Bixler et al. [83] proposed classifying these solutions as physical or chemical controls. Amador et al. [10] broadly categorize the solutions as nonrenewable (animal uses its energy for cleaning) and renewable (relies upon its surroundings) cleaning strategies. It can be classified as below:

1. ***Shaking or Flicking:*** Certain mammals, birds, and mosquitoes violently shake their bodies to remove the accumulated particles. Additionally, certain animals, such as

elk and honeybee, have erect hairlike features that facilitate particle removal. The hairlike features initially store and then release the potential energy by catapulting the accumulated particles. This act of ejecting particles is called flicking. Through flicking and shaking, numerous creatures can remove most of the aerosols by generating acceleration of 100–1,000 **g**. However, they rely on different techniques to get rid of smaller and lighter particles (such as bacteria) because for this, shaking or flicking would require a large amount of 10^{12} **g** of acceleration [10]. Additionally, it is important to note that acceleration generated by organisms is inversely proportional to their body mass for both shaking and flicking, as shown in the following equations (3.6 and 3.7), where a_{fl} is flicking acceleration, a_{sh} is shaking acceleration, and M_b is body mass [10]. This also indicates that flicking is more effective for smaller organisms.

$$a_{fl} \sim M_b^{-1.1} \quad (3.6)$$

$$a_{sh} \sim M_b^{-1/24} \quad (3.7)$$

2. ***Aerodynamic Geometry—Redirection of Flow:*** Another important technique creatures utilize is diverting the airflow to minimize particle deposition. Eyelashes, usually considered for filtering the particle, diverts the flow to avoid particle deposition in the eye. Amador et al. [10] showed that the optimal length of eyelashes reduces the shear rate at the eye surface by 90%. The porosity in the eyelashes is multi-functional; it allows the sunlight to reach the eyes. Another example of redirection of flow is present in Shark. Sharks induce anti-fouling properties by combining low drag, riblets, flexion of scales, and mucous layers [83]. A major portion of drag reduction comes

from reducing pressure drag and consequent reduction in total shear stress through streamlining the geometry.

3. ***Surface Patterning or Texturing:*** As discussed earlier, reduced shear stress is desirable for reduced deposition. Sharks also employ riblets (microtexturing) on their body surface to constrain the naturally occurring vortices locally, which reduces the transfer of momentum and total shear stress, reducing the overall drag by 9.9% [83, 84, 85, 86]. Additionally, lower drag leads to higher velocity near the shark's surface, providing less settling time for microorganisms to settle on the shark skin. In addition, microtexturing and hierarchical structures could be observed on the lotus leaf for reduced deposition. Water droplets require more energy to adjust inside the nooks and crannies of the rough ridges present on the leaf and are thus forced to rest on the top of the surface features. The hierarchical structure of the lotus leaf causes minimal contact area between the particle and the surface, minimizing dirt deposition. Additionally, similar approaches for self-cleaning mechanisms are adopted by geckos, wings of cicadas, butterfly wings, and many more. Bixler and Bhushan discuss a detailed list of the same [83, 54].

3.3.2 Superheater Tube Modification Based on Class of Solution

Which Design Solution? Because we focus on pursuing morphological solutions in the current dissertation, we do not consider choosing a shaking- or a flicking-based solution. Additionally, as discussed previously, shaking and flicking acceleration produced by different organisms is inversely proportional to the body mass. This also indicates that superheater tubes, which are substantially long (3 m) and heavy (3.92 kg/m), would require more energy

to generate enough acceleration to shake off ash particles. However, one might quantify the acceleration needed to shed the particles on the order of microns and study it by creating unique surface geometries that interact with flow conditions and induce self-vibration to shed the ash deposits from the surface [77, 87].

Morphological-based solutions that are aerodynamically shaped tubes and surface microtexturing offer a potential solution. Koehl [88] discusses the non-linear relationship between morphology and performance and how small morphological changes can produce novel functions. Additionally, the effect of specific morphological changes depends on an organism's size, shape, stiffness, or habitat. For instance, for drag reduction, streamlining assists in reducing the pressure drag for large organisms such as sharks moving at higher Re (> 100) in contrast to small organisms such as benthic macroinvertebrates moving at low Re [88]. As an additional example, sharks employ streamlined geometry to reduce the pressure drag and further introduce riblets for form drag reduction.

Because our problem statement deals with macro-sized features, i.e., superheater tubes (39 mm), we deploy the macro-scale geometry change, i.e., **aerodynamic solution**, as the first-degree solution for tube redesign. This must assist in reducing the pressure drag, and thus, it might reduce the propensity for the ash particle to settle on the tubular surface. In the present study, variants of streamlined geometry for the tube's external surface in combination with circular and non-circular tubes' inner profiles have been studied in detail. For the context of this problem statement, we try to answer the following questions:

1. How does a modified tube geometric variation help in drag?
2. How do circumferential stresses redistribute in modified geometries, and how do they affect the overall life?

3. What does the overall design metric that accounts for fluid flow and mechanical strength characteristics suggest?

Geometric Design and Details: Based on this, an aerodynamic tube design that resembles an ogive shape or a teardrop shape is proposed that reduces the impaction angle, assisting in streamlined flow and hence reduced deposition. For simplicity’s sake, we further addressed our proposed tube as a “teardrop-shaped” tube throughout the work. Six variants of teardrop shape with circular and non-circular inner diameters were studied (see Figure 3.2). The geometries were created to reduce the impact angle, generating a streamlined flue gas flow and reducing the total ash deposition mass flux. While creating the geometries with a circular inner profile, the inner diameter is kept equal to the baseline circular tube design. The thickness on the semicircular side of the asymmetric tube is the same as the baseline circular design. The hydraulic diameter equals the circular inner profile in the geometries with a non-circular inner profile. However, the other parameters to design the tube are user-defined and could be optimized later. As an example, the dimension for the convex profile teardrop-shaped tube has been shown in Figure 3.2A. All the proposed tubes were additively manufactured on a Concept Laser M2 cusing machine in stainless steel 316L.

3.4 Role of Additive Manufacturing

Manufacturability: To enable the manufacturing of such tubes, AM-based techniques like laser-powder bed fusion (L-PBF) are proposed. Other manufacturing techniques, such as the extrusion-based method, are viable options to realize such tubes [89]. However, these superheater tubes are welded to a U-bend region in shell and tube heat exchangers. To avoid any sudden geometric aberration at the interface of the teardrop-shaped tube and the

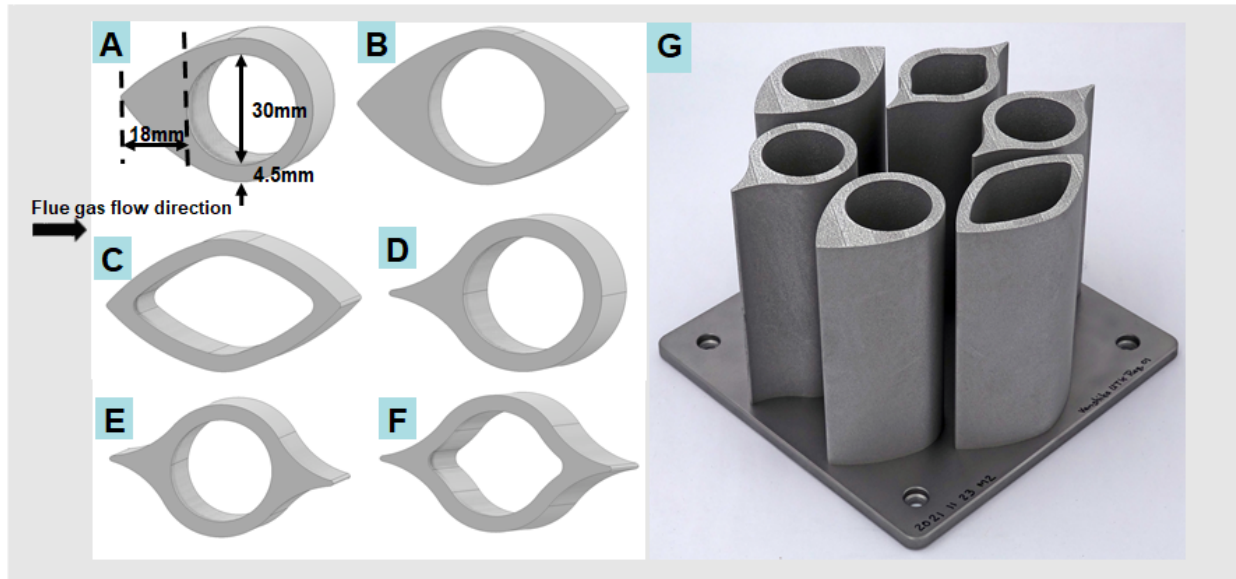


Figure 3.2: Teardrop shapes with (A) a convex profile and circular ID; (B) a symmetric convex profile and circular ID; (C) an asymmetric convex profile and non-circular ID; (D) a concave profile and circular ID; (E) a symmetric concave profile and circular ID; and (F) a symmetric concave profile and non-circular ID. (G) The proposed tubes were additively manufactured on the Concept Laser M2 cusing machine in SS316L.

U-bend, transitioning the teardrop-shaped tube to a circular cross-section toward the end is suggested. In such scenarios, AM-based techniques are significant in enabling such tubes. Additionally, AM methods also allow further modifications such as dimples, protrusions, or surface aberrations on the inner or outer surface of the tube [90]. Teardrop shapes with such surface modifications created with techniques other than AM can limit the sizing of these features, and multiple steps will be required to make these tubes [91]. This research evaluates the design using the L-PBF additive manufacturing and polymer-based additive manufacturing techniques.

AM-Built Material Properties: Besides geometric benefits, considering mechanical properties obtained via AM is equally essential. In the current study, SS316L is the chosen material for superheater tubes. Significant progress has been made for AM-built SS316L to minimize porosity and defects. Past studies have shown an increase in density in AM-built SS316L from $\sim 99.2\%$ [92] to 99.82% – 99.9% [93] through process parameters optimization. AM-built SS316L has a relatively shorter creep life benchmarked to conventionally built Type 316 SS [93]. As discussed by Li et al. [93], this might be attributed to the significant difference in carbon concentrations in Type 316 SS (0.048%) and AM SS316L (0.006%) affecting the creep rupture strength. It is also possible to enhance the creep properties of AM SS316L through the heat treatment process for specific parameters [93]. Finally, AM enables complex geometries and can allow tailoring of microstructure that can result in superior properties. Wang et al. [94] reported high-strength/high-ductile mechanical properties for AM SS316L compared with conventionally cast SS316L. Therefore, heterogeneity in AM properties, which is usually considered undesirable, can become a distinguishing benefit for next-generation component design.

3.5 Research Methodology

The overall research methodology to validate the proposed shapes is summarized in Figure 3.3. As part of the computational approach, a stress analysis was performed to predict the creep life of the proposed tube compared with the baseline tube design. Steps in the analysis include examination of the thermal and structural models of the tube. The second set of analyses was performed with a computational fluid dynamic model to understand particle motion around the tubes. A Reynolds-averaged Navier Stokes (RANS) $k-\omega$ shear stress transport (SST) model was built and coupled with the particle-tracing module in COMSOL. Schlieren imaging was performed for the qualitative experimental validation of the fluid flow behavior around the proposed and baseline designs.

3.5.1 Thermo-Mechanical Finite Element Analysis

Superheater tubes contain steam at high pressure (15–30 MPa) and temperatures exceeding 610°C. Such a critical operating condition elevates the hoop stress in the tube and is responsible for the stress rupture failure if extensive material waste occurs because of corrosion.

To study this, a 2D plane strain thermo-mechanical finite element (FE) analysis is performed using Ansys® Mechanical APDL for a typical service condition. Herein, thermal and pressure conditions were considered for loading conditions. Because the boiler tubes are long prismatic bars, and all the loads act in the in-plane direction, it is valid to assume plane strain analysis [95]. Stainless steel 316L is used as the material for the geometry.

The FE study has been performed in three steps: (1) The thermal FE model was created to calculate the circumferential thermal variation across a tube. (2) The average temperature

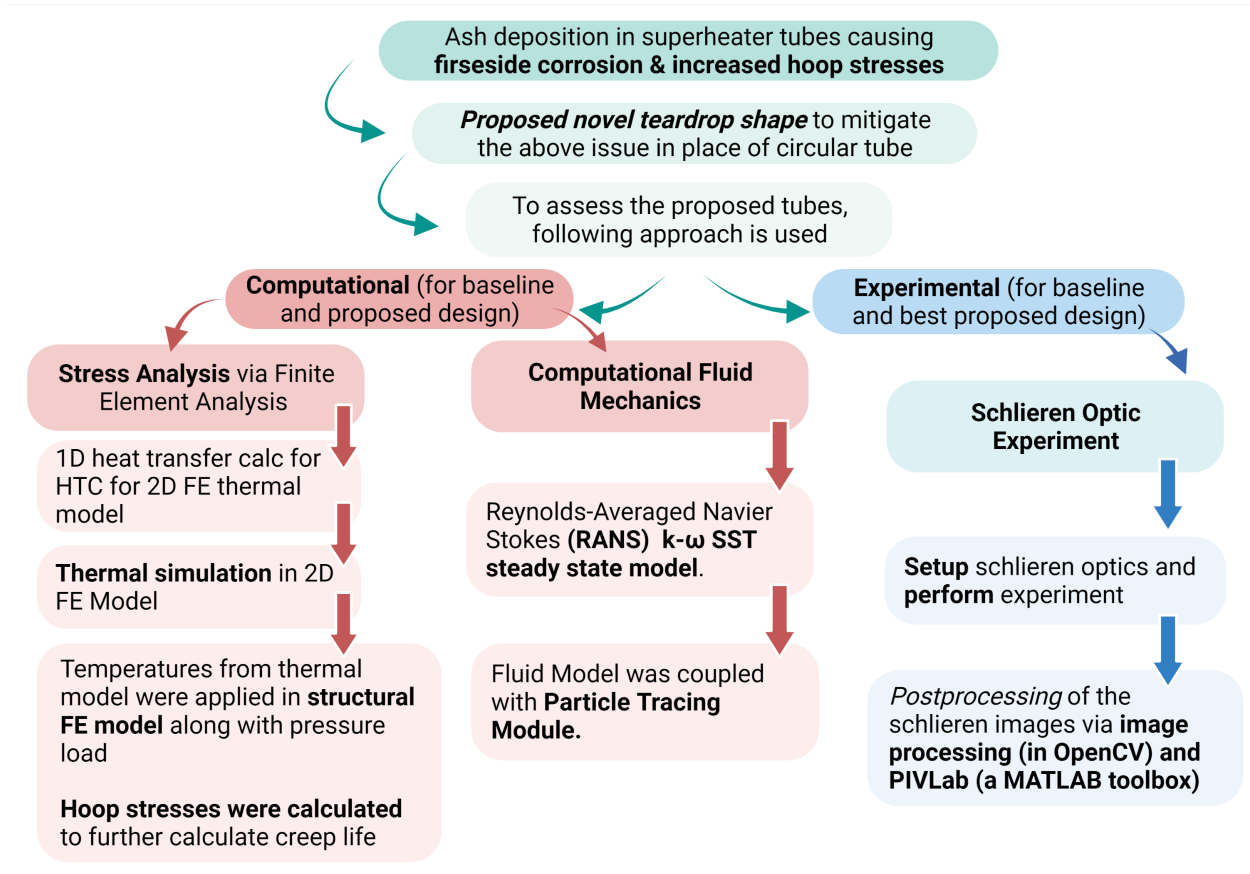


Figure 3.3: Methodology for the validation of the proposed design

calculated at different angular positions was used to perform the conduction run and calculate the intermediate temperatures. (3) The hoop stresses were obtained by applying these thermal temperatures as body forces for the structural FE model along with pressure loads.

3.5.1.1 Determination of Temperature Profile along Tube Surface

The localized ash deposition may induce temperature variations along the circumference of the tube surface. Such thermal variations might instigate or exacerbate stress-induced corrosion. Previous investigations have not considered the circumferential thermal variations in depth. For example, previous analyses have calculated the temperatures by using a 1D thermal resistance network among flue gases, ash deposits, the metal wall, and the steam side of the tube [96, 64]; calculated the thermal variation for ash deposits by assuming constant outer metal wall temperature [73, 97]; and calculated the temperatures for the stacked superheater tube arrangements [98]. Therefore, we need to calculate circumferentially varying surface temperatures across the tube by considering the non-uniform presence of the ash. In this study, a thermal FE model is developed for determining the temperature profile along the tube surface. During these thermal calculations, surface heat transfer coefficients were analytically estimated and provided as boundary conditions to the FE model.

Estimation of Heat Transfer Coefficient (HTC): In superheater tubes, heat transfer on the outer surface of the tube occurs due to the combination of radiation and convection. Therefore, the heat transfer coefficient (HTC) due to convection and radiation was calculated separately and superimposed for effective HTC. Through a detailed literature review for 1D heat transfer calculations for superheater tubes, Zbogar et al.[96] was found where the methodology and the input values used for 1D heat transfer were clearly discussed. Thus, for our HTC estimations, except the thermal conductivity of the metal wall, all input values for

thermal profile distribution were referred from Zbogor et al.[96] like the thermal conductivity of ash, the thickness of ash, flue gas temperature, steam temperature, and velocity of flue gas.

For calculating the HTC due to convection, a flow across the cylindrical tube has been considered. An average Nusselt number was calculated as opposed to the local Nusselt number for convective HTC. Studies by Zhou et al. [97] and Asgaryan [20] calculated local Nusselt number varying with the angular position of the tube, which was based on Sanitjai and Goldstein experimental calculations [99]. Though there have been efforts to understand the local Nusselt numbers, we calculated the overall average Nusselt number using the generalized empirical correlation available in literature [96, 100] (based on experimental values of the average heat transfer coefficient from a heated cylinder):

$$Nu = C.Re^m.Pr^{1/3} \quad (3.8)$$

where Nu is the Nusselt number, and the values of C and m vary as per the Reynolds number range. For flue gases, the density of the fluid (ρ_f) is 0.345 kg/m^3 , fluid velocity (μ_f) is 8 m/s , characteristics length (d) is 0.039 m , and dynamic viscosity (μ_f) is $4.41 \times 10^{-5} \text{ kg/ms}$. The calculated Reynolds number ($\frac{\rho_f \mu_f d}{\mu_f}$) was 2442.48; therefore, the chosen values of m and C were 0.683 and 0.466, respectively [96, 100]. Hence, the following relationship was used for Nusselt number outside the tube due to convection:

$$Nu = 0.683.Re^{0.466}.Pr^{1/3} \quad (3.9)$$

The Prandtl number was calculated to be 0.73 via $\frac{\mu_f C_p}{k_f}$ where C_p is the specific heat capacity at constant pressure (1,146 J/kg K), and k_f is the thermal conductivity of the flue

gas (0.069 W/m °C). The Nusselt number was calculated using equation 3.9 as 23.33. The effective heat transfer coefficient is calculated as below:

$$h_{eff} = h_{rad} + h_{conv} \quad (3.10)$$

$$h_{rad} = \frac{\epsilon \sigma (T_f^4 - T_d^4)}{(T_f - T_d)} \quad (3.11)$$

$$h_{conv} = \frac{Nu \cdot k_f}{d} \quad (3.12)$$

where ϵ is the emissivity of the surface, T_f is the flue gas temperature, T_d is the deposit temperature, σ is the Stefan-Boltzmann constant ($5.67 \times 10^{-8} \text{ Wm}^{-2}\text{K}^{-4}$), Nu is the Nusselt number, and k_{fluid} is the thermal conductivity of the flue gas. D is the characteristic length, the outside diameter (d_0) in this case.

Using equation 3.12, the average heat transfer coefficient due to convection h_{conv} was calculated to be 41.28 W/m² °C. For calculating the average heat transfer coefficient due to radiation h_{rad} , the emissivity of the deposit was taken as 0.5 [96]. The whole heat transfer process was initially solved as a 1D heat transfer problem to estimate the h_{rad} (See Appendix AA.1). The calculated h_{rad} value (159.36 W/m² °C) was finally used to estimate the h_{eff} . From 12 o'clock to 2 o'clock, the deposit is modeled in the 2D FE model, and thus h_{eff} is applied in that region. In the rest of the region, because the ash is not present in a significant amount, it is not modeled, and only h_{conv} is applied. However, in this region, the effect of a 1 mm thick ash deposit (radiation and conductance effect) was considered by reducing the bulk film temperatures by ~10%, equivalent to 1 mm thick ash. The sensitivity was performed for ~5% to 15% for bulk film temperatures equivalent to 0.05 to 1.5 mm of

deposit presence. The analysis showed that the impact was within $\sim 2\%$ on the outer wall temperature. Thus, assuming ~ 1 mm of wall deposit is acceptable. Both hand calculation and FE simulations verified this study. The heat transfer coefficient was calculated for every five-degree sector of the geometry, which ranged from $200.65 \text{ W/m}^2 \text{ }^\circ\text{C}$ to $41.28 \text{ W/m}^2 \text{ }^\circ\text{C}$. To maintain consistency with the work of Zbogor et al. [96], the HTC value was assumed based on their calculated thermal resistance values. In their work, 41% thermal resistance occurred from flue gas to outer tube wall and 11% thermal resistance from inner tube wall to steam. Hence, the heat transfer coefficient was assumed to be 4% higher on the steam side as compared to the flue gas side. The calculated heat transfer coefficients were then applied in the thermal finite element model (See Figure 3.4A).

Finite Element Modeling: For thermal modeling, the ash deposit shape has been modeled along with the circular tube shape. The ash deposit profile has been considered elliptical in shape, and a maximum thickness of 0.005 m is applied at the 12 o'clock position [96]. Ash deposits have been provided a thermal conductivity of $0.6328 \text{ W/m }^\circ\text{C}$ [96], whereas the SS316L has been provided a thermal conductivity of $19.9 \text{ W/m }^\circ\text{C}$. Plane 55 elements (2D thermal solid) were used to create the FE mesh within Ansys. The FE model has been chunked in a sector of five degrees circumferentially, and averaged HTCs have been applied in each sector's inner and outer boundaries. This simulation calculated the temperatures at the outer and inner walls. The average of the outer and inner walls has been used in the structural FE model (in Section 3.5.1.2) as the body force.

3.5.1.2 Structural Analysis Using Thermal and Pressure Loads

The 2D four-node solid elements were used for the plane strain analysis in the structural FE model. Figure 3.4B shows the boundary and loading conditions applied in the structural

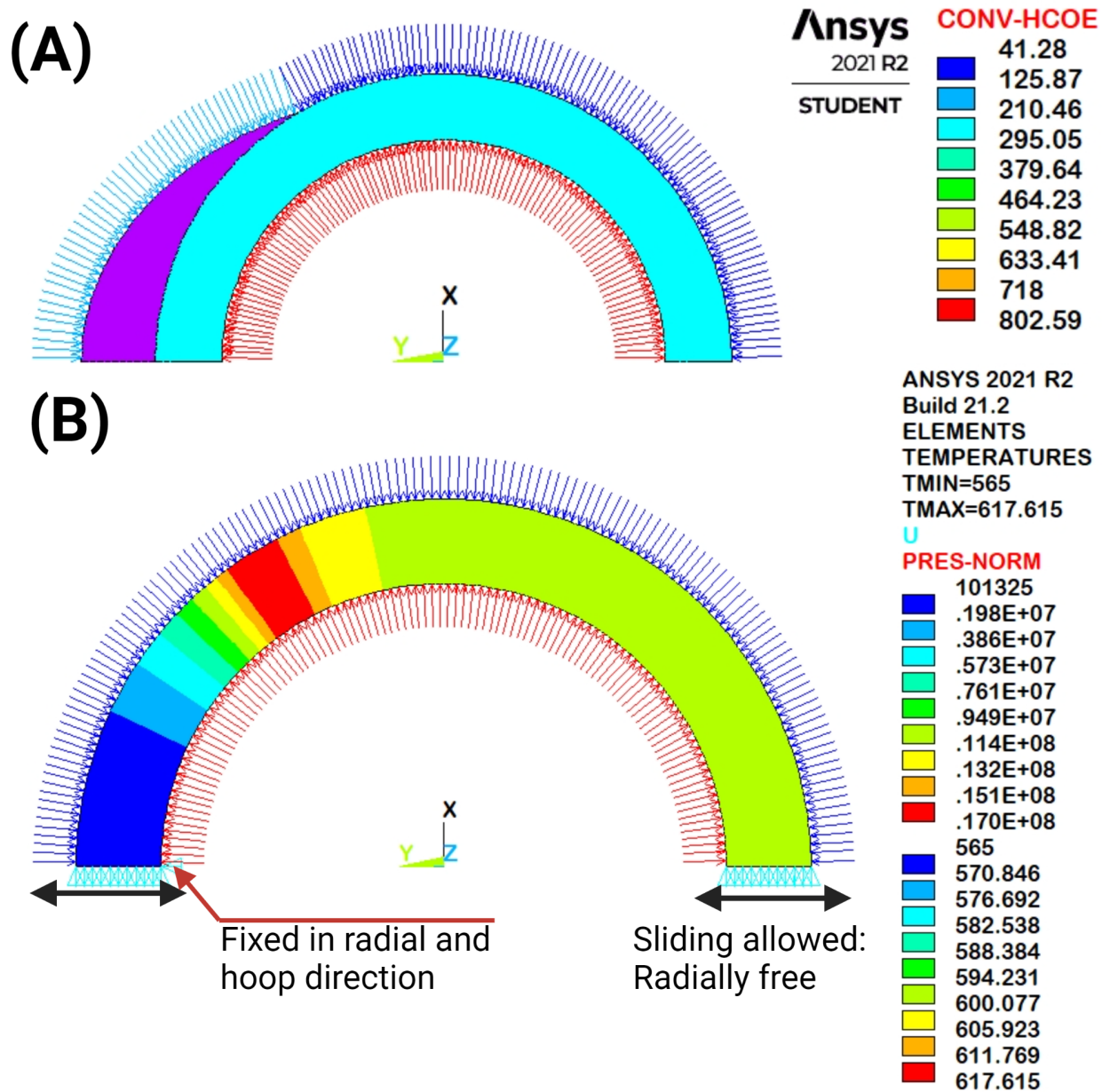


Figure 3.4: A) Heat-transfer coefficients in the finite element thermal model, and B) boundary and loading conditions applied in the structural finite element modeling of the tube.

FE model of the baseline tube. The calculated temperatures from thermal simulations were exported as body forces in the structural model of the tubes. In addition to temperature loads, pressure loads were applied as surface loads in the structural model. High pressure of 17 MPa was applied at the inner wall, and an atmospheric pressure of 0.1 MPa was applied at the outer wall. The model is fixed so that it is allowed to grow radially, constrained in the hoop direction (see Figure 3.4B). The exact boundary conditions have been assumed for the proposed tube designs. The thermal expansion coefficient for SS316L applied was 1.8E-05 cm/cm per °C.

3.5.2 Flow Study Around Differently Shaped Tubes

The primary motive of studying flow around differently shaped tubes is twofold: 1) The first motive is to quantify the likelihood of particles striking the tube surface for the proposed and baseline tubes. A computational fluid dynamics (CFD) study coupled with a particle tracking module is performed for this. The fluid flow is subsequently coupled with particles that follow the Stokes drag law. 2) The second motive is for Schlieren imaging to be performed and post-processed to visualize and qualitatively study the development of different flow regimes for differently shaped tubes.

3.5.2.1 CFD Modeling: RANS Flow with Particle Tracing in COMSOL

A multiphysics model was prepared to perform the comparative study of particle tracing in the tubes using COMSOL. A particle tracing module has been integrated with a single-phase flow module. For illustration purposes, Figure 3.5A shows the geometric details for circular and teardrop-shaped tube models. Figures 3.5B and 3.5C illustrate the mesh details around the circular-shaped tube and teardrop convex profiled tube. The geometric details

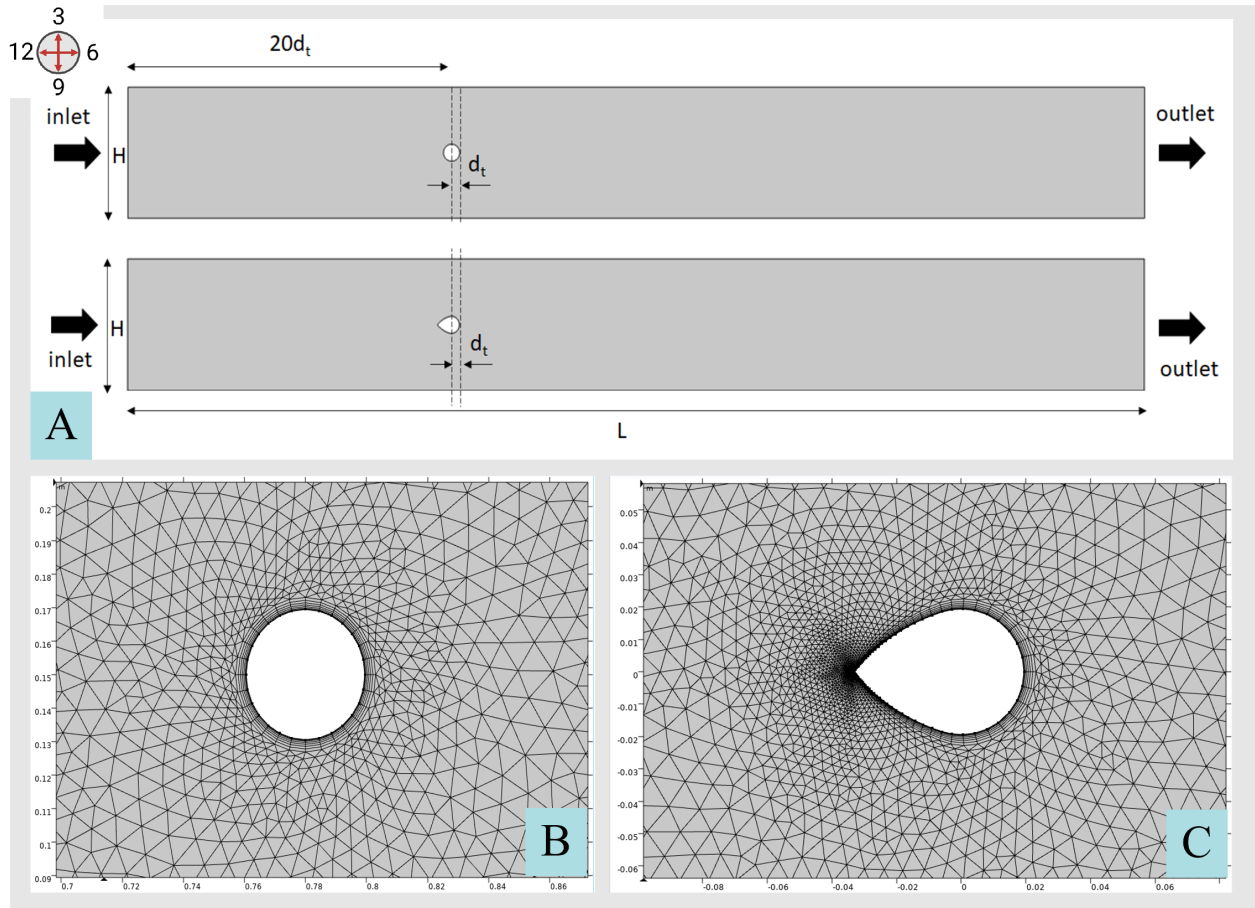


Figure 3.5: A) Geometric details for both circular and teardrop-shaped tubes; mesh details for RANS $k-\omega$ SST analysis for circular (B) and teardrop-shaped (C) tubes.

were inspired by Riccio et al. [73]. The current study uses a single tube for both circular and teardrop-shaped tubes to allow comparison.

The RANS approach has been used with the SST turbulent model for studying the single-phase flow. The SST model is a low Reynolds number model that considers $k-\epsilon$ in the free stream and $k-\omega$ model near the walls. For setting up the model, flue gas characteristics, particle density, particle mean size values, and particle mass flow rate were taken from the paper by Riccio et al. [73]. Table 3.1 summarizes the values for the geometric details, flue gas properties, and particle description for particle tracing. The grid was generated within the software, and the mesh was refined such that y^+ values remained near or below one. Additionally, the model was checked for mesh sensitivity. Because particle tracing aims to quantify and compare the propensity of particles striking differently shaped tubes, certain assumptions were made in the study. Particle-to-particle interaction was not considered. Uniform particle size (spherical, smooth particle size of $68\text{ }\mu\text{m}$ [73]) is considered. Once the particle impacts the tube surface, the particles are assumed to freeze onto the tube surface. Specifically, the particle velocity will come to zero, and the particle position will be fixed when it contacts the wall. Since rollover, sliding, and rebounding of particles were not considered in the study, temperatures were not applied to the fluid flow or the tubular surface, which might impact the shape of the particles. Stokes law has been used for calculating the drag force acting on the particles. Gravitational force and Van der Waals forces were not considered. For post-processing, drag coefficients and the number of particles encountering the tube surfaces will be compared for the circular and proposed tubes.

Table 3.1: Geometry details, flue gas properties, and particle description

Parameters	Values	Units
OD of baseline tube	39	mm
L	2.45	m
H	0.3	m
Inlet flue gas velocity	8	m/s
Flue gas density	0.345	kg/m
Specific heat capacity (flue gas)	1146	J/kg K
Thermal conductivity (flue gas)	0.069	W/m K
Dynamic viscosity (flue gas)	44.07×10^{-6}	kg/m s
Particle specific mass flow rate	1×10^{-3}	kg/s
Particle density	2200	kg/m
Particle mean diameter	68	m

3.5.2.2 Schlieren Optics for Visualizing Flow around the Tube

To visualize the flow around differently shaped tubes, Schlieren optics was used. Schlieren optics allows for visualizing the flow of gases or changes in the air refractive index via a concave mirror, point-source light, razor blade (light block), and a camera. It is based on the principle that the speed of the light changes in varying density mediums, and the path of the light will deviate from its original path (Figure 3.6). This technique has been abundantly used in various applications for flow visualizations, thermal variations, acoustic wave visualization, the study of materials, and other related studies. However, it has found limited applications in boiler tubes and heat exchanger assessments. In our work, we have used this technique due to its capability to allow the user to directly track fluid flow in contrast to tracking particles (for example, fluorescent particles in particle image velocimetry). Also, this technique is easy to set up and comprehend in modern fields of fluid analysis with emerging computer vision techniques and image processing tools [101].

Working and Setup: In the Schlieren optics setup, point-source LED light is projected in the field of view of the concave mirror. A razor blade (light block) is kept at twice the focal length of the concave mirror. A camera with a zoom lens is aligned behind the light block to capture the image of the fluid flow variation in front of the mirror. If the density of the fluid in the proximity of the concave mirror is varying, then the light reflected from the mirror will deviate from its original path. The razor is vertically aligned such that it barely cuts the converged image. If light rays are deflected toward the knife edge, image portions created from these deflected rays of light will be darkened when compared with image portions due to constant density. Alternatively, if light rays are deflected away from the knife edge, image portions due to these deflected rays of light will be brighter than

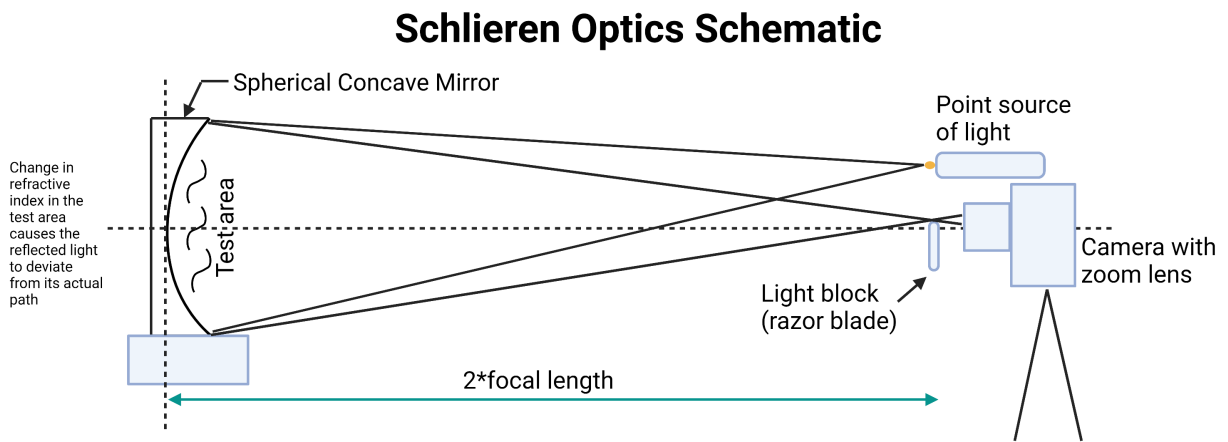


Figure 3.6: Schematic representation of a Schlieren optics setup.

the image portions [102]. These are the mechanisms for the enhanced visualization of the fluid flow that is not visible to the naked eye. An in-house setup was created for the same, described in detail in Appendix A.2.

Post-processing Technique: Post-processing was done using OpenCV (Python Library) and the open-source software PIVlab (Digital Particle Image Velocimetry for MATLAB Toolbox) [103]. OpenCV is used for image segmentation of the Schlieren images. These processed images were then used as inputs for PIVlab, which was originally designed for the calculation of displacement of particles based on cross-correlation of sub-images between two consecutive PIV images. However, Gena et al. [104] discussed in detail how PIVlab is useful for processing types of pattern displacements other than PIV particle images. This tool can be leveraged for Schlieren image post-processing. The current paper uses PIVlab to generate streamlines for the flow around differently shaped tubes.

3.6 Results

3.6.1 Thermal Simulation Results

The FE thermal model extracted temperature contour plots for the tube wall (See Figure 3.7A). From the model, averaged temperature values were fetched at every five-degree interval on the outer and inner walls of the circular tube. The thermal profile on the outer and inner walls with respect to the angular position of the tube is plotted in the polar plot form (see Figure 3.7B). This profile suggests higher temperatures around the 10 o'clock ($\sim 60^\circ$) and 2 o'clock (240°) positions relative to other locations. The temperature obtained is the lowest at the 12 o'clock (0°) position. This conforms with the hypothesis mentioned previously. As

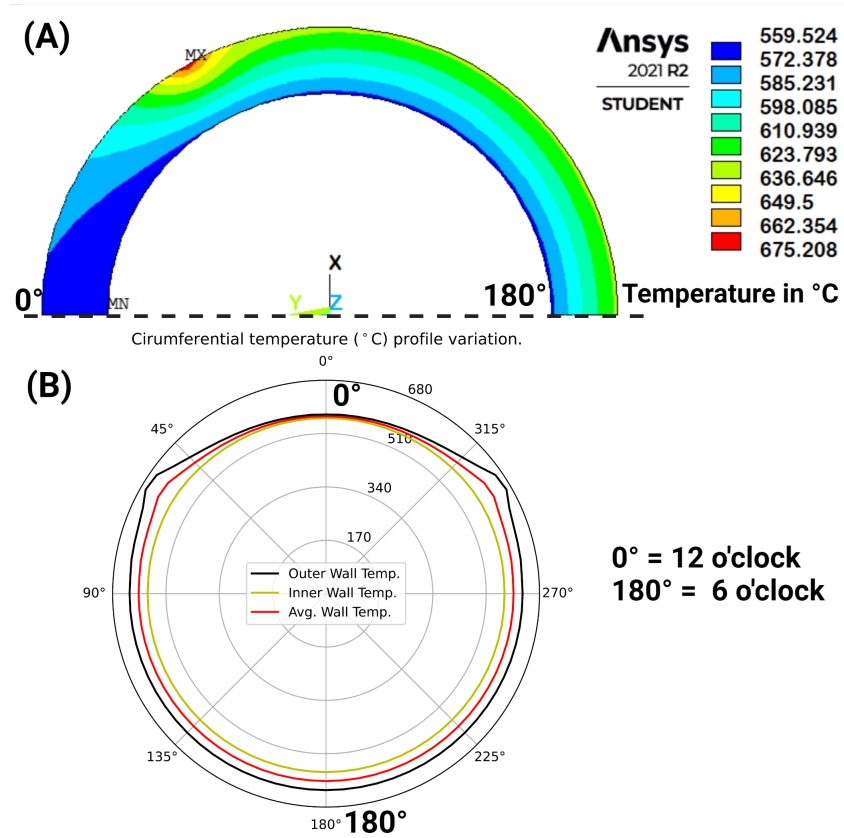


Figure 3.7: A) Thermal FE simulation results displaying temperatures (C) contours for the baseline tube wall. B) Polar plot for the tube wall temperatures on the outer wall, inner wall, and averaged wall temperature

noted earlier, the average temperatures of the inner and outer walls are used as input to the FE structural model.

3.6.2 Structural Simulation Results

The hoop stresses, i.e., stresses in the θ -direction in the cylindrical coordinate system (csys-1 and rsys-1 in Ansys), were extracted from the FE simulation results. The hoop stress contour plots have used overall maximum hoop stress and stress near the 2 o'clock position to compare the differently shaped tubes. The overall maximum hoop stress might or might not be near the 2 o'clock position. However, it would be interesting to understand the nature of stresses near the 2 o'clock location because these regions are known to be susceptible to material degradation. Tensile stresses around this location might aggravate stress-assisted corrosion and accelerate creep failure due to increased hoop stresses [105] (see Figure 3.8H). The tensile stresses would allow the liquid phase to penetrate the grain boundaries, thereby extending the crack into the sub-surface. Therefore, the design must consider reducing or removing tensile stresses around the 2 o'clock position. Figure 3.9 shows the hoop stress contours for the baseline tube (Figure 3.8A), proposed tube (Figure 3.8B), and its variants (Figures 3.8C, 3.8D, 3.8E, 3.8F, and 3.8G). A common scale is used, and tensile stresses (positive hoop stresses) have been plotted for comparison purposes. The grey portions in Figures 3.8B, 3.8C, 3.8E, and 3.8F indicate only compressive stresses, whereas the grey portions in Figures 3.8D and 3.8G contain both compressive and tensile stresses. The tensile stresses in designs D and G are relatively high compared with the other designs. The contour plots for hoop stresses clearly indicate the development of compressive stresses in the proposed tubes (designs B, C, E, and F) near the 2 o'clock position. In the region of interest ($\sim$ 2 o'clock position), average stress is marked in Figure 3.9 for all the tubes.

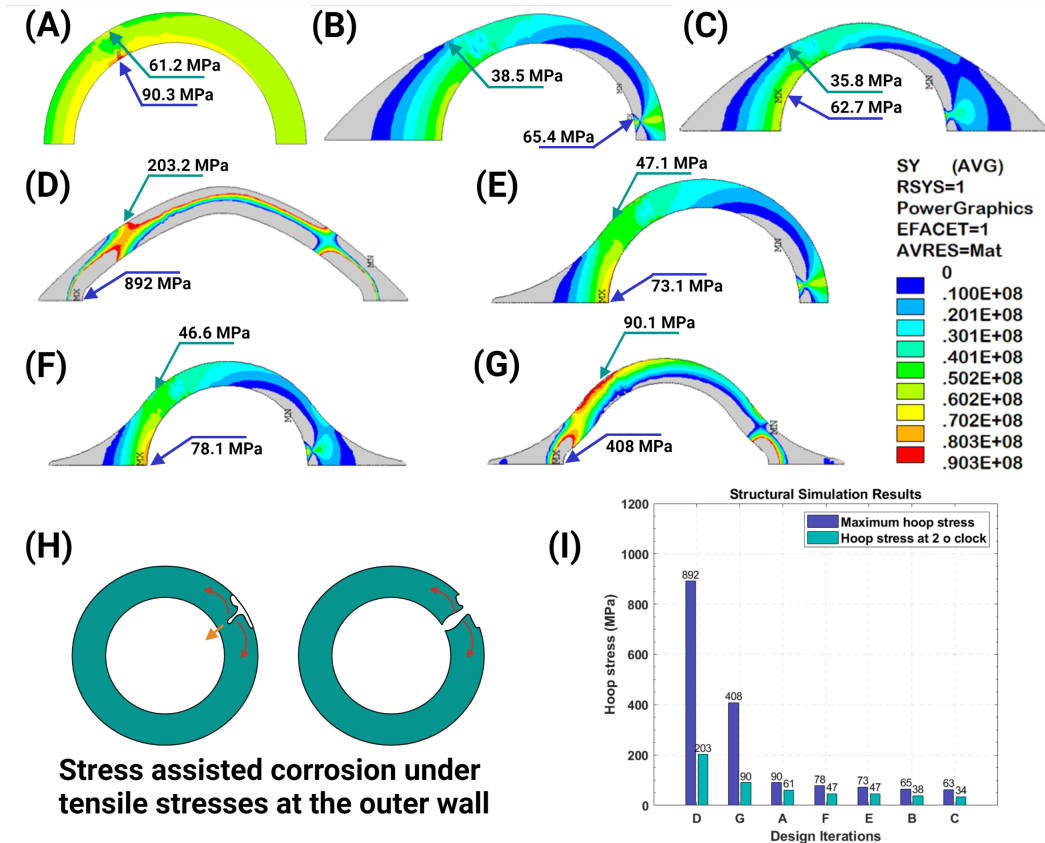


Figure 3.8: A-G) Tensile hoop stresses in a cylindrical coordinate system for the baseline design and the proposed designs; H) Schematic for stress-assisted corrosion under tensile stresses at the outer wall; I) Hoop stresses for each design iteration including baseline tube design. Maximum hoop stress indicates the largest magnitude of hoop stress in the tube. Hoop stress at the 2 o'clock position which is the region of interest.

For clear representation, overall maximum hoop stress and hoop stress at the 2 o'clock position have been plotted in Figure 3.8I. The overall maximum hoop stresses are highest in design D (892 MPa), followed by design G (408 MPa). Among designs A, B, C, E, and F, a circular-shaped tube (design A) has the highest overall maximum hoop stress (90.3 MPa). The teardrop convex profile shape with a circular inner diameter (design B) shows 37% lower hoop stress at the 2 o'clock position when compared with that of the baseline tube at the same location. Among the other proposed variants, the convex profiled shape provides 18–24% less stress in the region of interest than the concave profiled teardrop shape provides. Of all proposed and baseline designs with a reduction of mechanical stress, the prominence of designs B, C, E, and F appears to be suitable for our application.

3.6.3 CFD Modeling Results

In this subsection, velocity fields (refer to Figure 3.9), drag coefficients, and the number of particles encountering the tube surface for various geometries have been calculated and compared with a baseline tube design. The total drag force per unit length acting on the tube surface was calculated using the velocity field results. The drag force includes both pressure and skin friction forces. For calculating the drag force, local wall shear stress was integrated on all the boundaries of the tube shape. Further, the drag coefficient (a dimensionless number) for each tube was calculated using equation 3.13:

$$C_d = \frac{2F_d}{\rho u^2 A} \quad (3.13)$$

where C_d is the drag coefficient, F_d is the drag force (N), ρ is the density of the fluid (kg/m^3), u is the flow velocity of the object relative to the fluid (m/s), and A is the reference

area (m^2). The drag coefficients have been shown in Figure 3.10A. Design B has the lowest drag coefficient, 21.6% lower than the circular-shaped tube design, whereas the other variants of the tube design have higher drag than that of circular-shaped tubes. In addition to velocity fields, the number of particles striking the tube surface was studied. This study was performed for the different numbers of particles per release: 250, 500, 600, 750, 1,000, and 2,000 particles per release. Figure 3.10B shows that the number of particles freezing on the circular-shaped tube in all six cases was higher than in the teardrop shape convex profile tube design (design B). On average, the teardrop convex profiled shape (design B) has 13% fewer particles striking the tube surface than circular-shaped tubes have.

3.6.4 Schlieren Experiment Results

Flow around differently shaped tubes (designs A and B) was visualized using Schlieren optics set up in an .mp4 format. These videos were image-processed to get streamlines around the tube. The analyses were performed by extracting 20 frames per second of the movie using Python API. Further, post-processing was performed on images extracted from the measured data from teardrop and circular-shaped tubes. Because Schlieren images contain regions of different intensities, adaptive thresholding was performed using the OpenCV Python library. In this case, the different threshold values for different regions are calculated by taking the average of the neighborhood values. Figure 3.11A shows Schlieren images for four different frames and the corresponding images after adaptive thresholding. These pre-processed sets of images were then imported into PIVlab for image analysis and image post-processing. Averaging of the velocity vector field was performed on results from 80 images. The averaged information was then used to construct the streamlines (see Figures 3.11B and 3.11C), which have been covered in detail in the Discussion section.

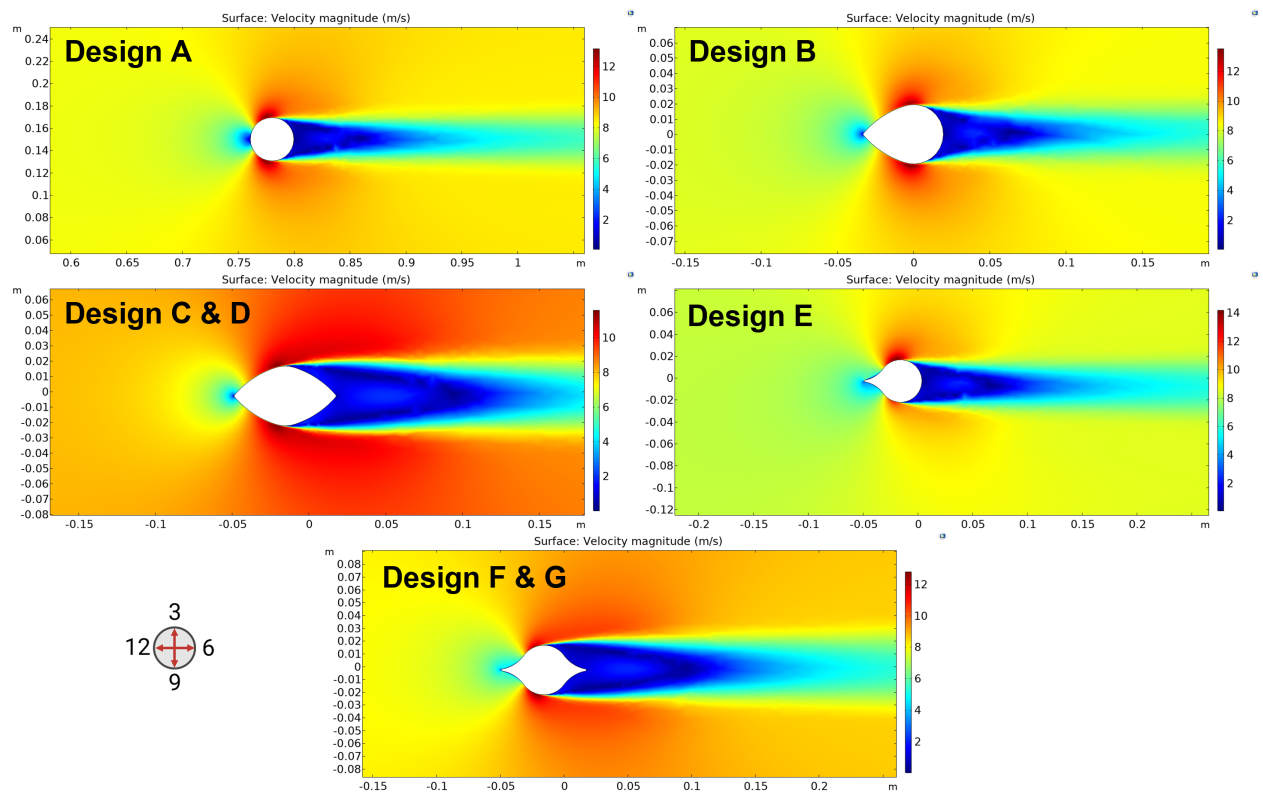


Figure 3.9: Velocity field for the baseline tube and proposed tube designs

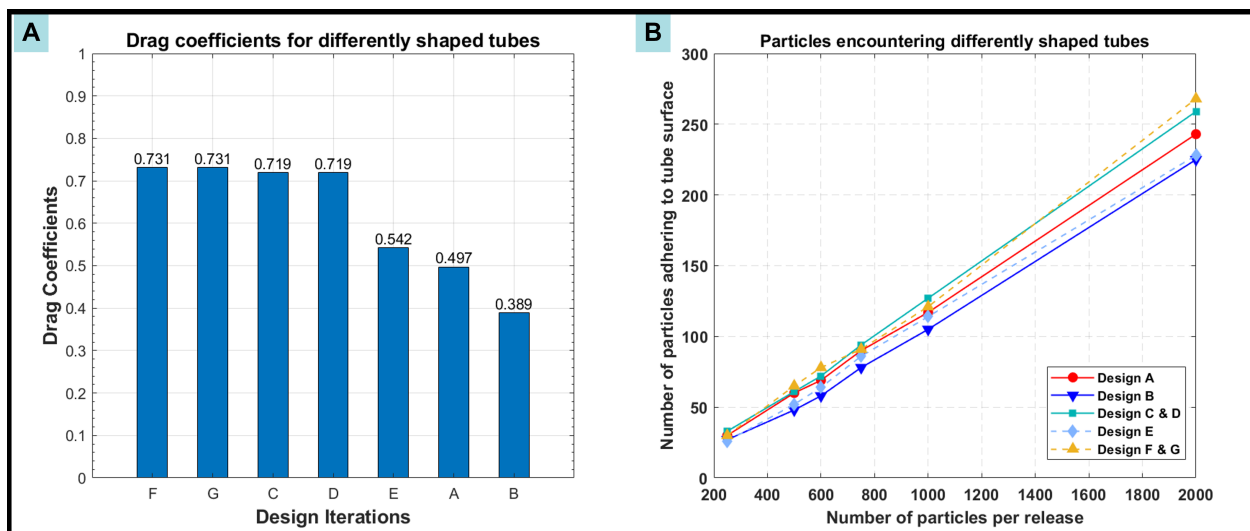
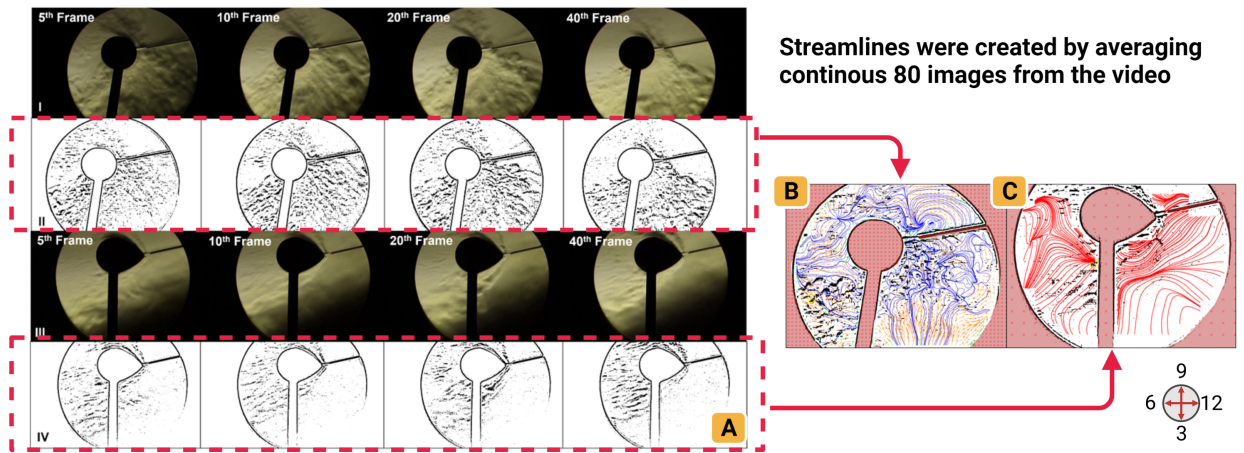


Figure 3.10: (A) Drag coefficient for each design iteration arranged in descending order; (B) Number of particles striking tube surface for differently shaped tubes



3.7 Discussion

3.7.1 Role of Geometry and Creep Life Calculations

Whereas past investigations used analytical methods to calculate hoop stress due to internal pressure [106, 20], we used finite element methods to calculate the hoop stresses by also considering varying circumferential temperatures in the tube cross-section. Circumferential thermal applications lead to non-uniform development of hoop stresses in the circular-shaped tube. However, in other differently shaped tubes, non-uniform hoop stress development is the consequence of the tube's shape and the circumferential thermal loads.

From the Results section 3.6.2, it can be inferred that the shape of the tube impacts the location of maximum hoop stress development. Designs D and G have the highest hoop stresses and perform more poorly than the baseline tube in terms of the tubes' mechanical integrity. This could be substantially attributed to the geometrical notch at the 12 o'clock and 6 o'clock positions on the inner side of the tube. However, other designs like B, C, E, and F are competitive for the circular-shaped tubes. As opposed to circular-shaped tubes, in all these designs, the maximum hoop stress did not develop in the region of interest (i.e., at the 2 o'clock position). In all four proposed designs, the overall maximum hoop stress formed near either the ~ 12 o'clock position or the ~ 6 o'clock position. Additionally, the magnitudes of the maximum hoop stresses are lower than those of the circular-shaped tube, which might lead to increased creep life of the superheater tubes.

To estimate the creep life of the baseline and proposed tubes, additively manufactured stainless steel 316L tubes have been considered. Li et al. [93] developed an empirical relationship between creep stress and Larson-Miller parameter (LMP) for AM SS316L

(manufactured through the L-PBF method on the Concept Laser M2 cusing machine) at different temperatures. For our case, we leveraged the same relationship at 600°C to estimate the LMP for differently shaped tubes at 0 hours and 10,000 hours of operation. Because designs D and G are incompetent for circular-shaped tubes, these tubes were dismissed for further investigation, including creep life calculations. Therefore, LMP was calculated for designs A, B, C, E, and F without and with degradation. For the degradation study, the material degradation of 0.2 mm after 10,000 hours of operation was assumed and applied in the geometry from 55° to 65° (See figure 3.12A). The LMP, $T(C + \log_{10}(t_f))$, where C is material specific constant, t_f is time to failure in hours, and T is the temperature in Kelvin, was calculated using $C = 17$ [93]. Figure 3.12B suggests that proposed tube designs B, C, E, and F have improved LMP and consequently improved life compared with the circular-shaped tube. This study also indicates that the rate of decrement of life after 10,000 hours for circular-shaped tubes is slower than that of the proposed tubes. The FE models with degradation developed for creep life calculations can be modified and extended to real-life conditions.

3.7.2 Significance of Fluid Flow Characteristics for Superheater Tubes

Primarily, the drag force acting on the tube surface is due to the pressure drag force. Figures 3.10A and 3.10B show that shapes with lower drag coefficients also have a lesser propensity of particles striking the tubular surface. Figure 3.10A displays higher drag coefficients for the symmetric tube designs. This might be mitigated by changing the geometry by reducing the width of the cross-section. However, this might lead to either a reduction in the hydraulic diameter, thereby causing the pressure drop inside the tube for the steam flow, or increased

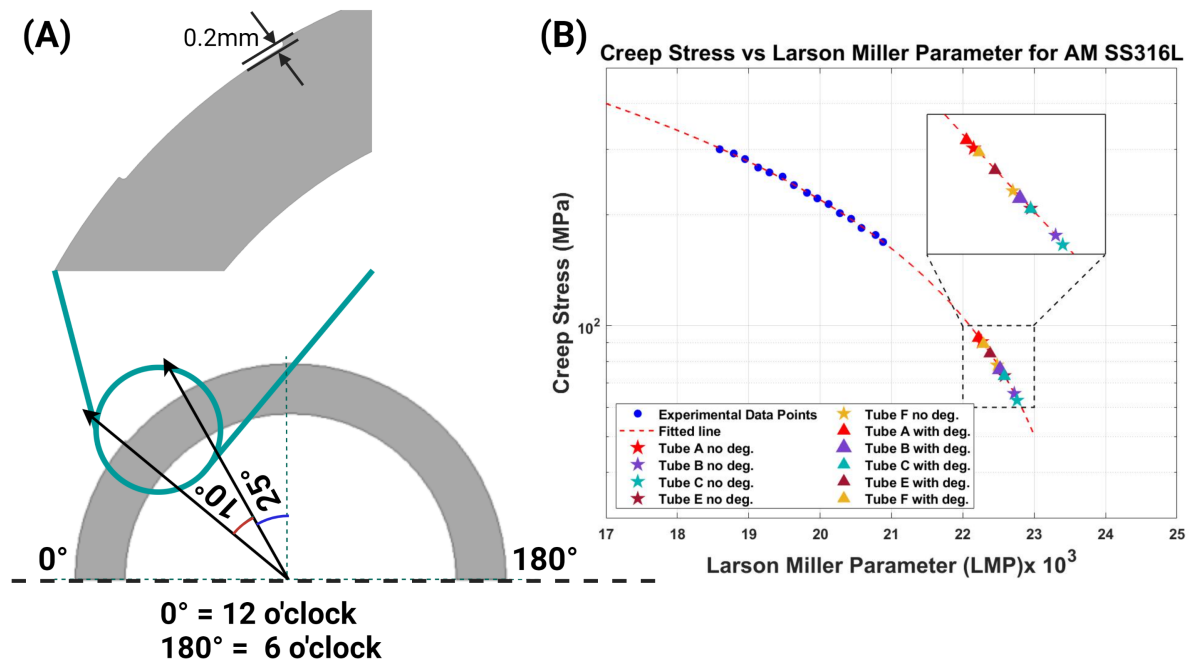


Figure 3.12: (A) Predicted LMP values for differently shaped designs; (B) Predicted time to rupture at 0 hours and after 10,000 hours of operation for each design iteration

hoop stresses due to stress concentration. Asymmetric tubes have lower drag coefficients than those of symmetric tubes, which could be attributed to their lower surface area. Also, Figure 3.10A implies higher drag coefficients for circular-shaped tubes than those of design B. This can be reasoned out via Schlieren imaging experimental studies. A lower drag coefficient should allow the streamline to follow the geometric curvature, whereas, in the case of a higher drag coefficient, the flow will separate early, causing turbulent wakes. Figures 3.11B and 3.11C indicate that the flow around the circular tube is more turbulent, whereas the teardrop shape is more streamlined. Also, in the case of a teardrop shape, the flow reattaches at the rear of the tube, signifying the lesser propensity of vortex shedding at the lee side of the tubes. The streamlined flow must also enable lower pressure to drop across the heat exchanger. Figure 3.11B also exhibits a high fluid mixing level in the circular tube compared with the teardrop-shaped tube. Reduced mixing of flue gas might lessen the ash deposition in the proposed shape compared with the current superheater tube design. However, reduced mixing and lower turbulence might impact the heat transfer efficiency at the tubular stack level. Further optimization of arrangements of tubes could be studied in the future.

3.7.3 Feasibility of Using Modified Tubes

Based on the previously discussed stress simulations and computational fluid dynamics performed for the proposed and baseline tubes, tube shapes with streamlined geometry and improved creep life can be leveraged as one tool kit to combat ash deposition. However, the weight and hydraulic diameter of the tubes must also be considered during the design. During the design phase, the hydraulic diameter is kept the same for all the tubes (30 mm), so it can be ruled out in the current case. Weight has been calculated using SS316L material for all the tubes per unit length. A metric table has been prepared and presented in Table 3.2

for all the tubes. All parameters—maximum hoop stress (MPa), hoop stress at 2 o'clock position (MPa), drag coefficient, and the weight of the tube (kg/m)—have been normalized. In Figure 3.13, normalized parameters are plotted. For normalization, the weightage factor applied for each parameter is kept at one, meaning that all the overall metric parameters are currently equally important. The x -axis shows the design type, whereas the y -axis represents the magnitude of the metric. The lower the sum of the overall metric, the better the design is for the superheater heat exchanger tubes. The metric clearly shows that design B, a teardrop convex profiled tube with circular ID, performs best among all, whereas design D, a teardrop convex profiled tube with non-circular ID (symmetric), performs worst.

3.7.4 Limitations and Future Work

This research proposes a novel-shaped tube design that can be realized via additive manufacturing. To implement these designs in real-time application in power plants, our proposed work unfolds areas for interesting research within the allowable limits defined by ASME codes. Because these designs are with oblique surfaces, the rebounding of the ash particles may increase. These rebounded particles can then hit other tubes and start depositing on them. Hence, it is required to study the tube design with the stack of tubes. Also, in the current study, heat transfer calculations were performed for a constant ash thickness, which might vary with respect to time. Additionally, the ash composition affects the conductivity of the ash and emissivity, and hence, might also alter the magnitude of the thermal calculations. In this study, for Schlieren optics experiments, the fluid has been treated as a jet to increase the sensitivity near the tube wall for visualization purposes. The jet speed is ~ 12 m/s, whereas the flow speed in CFD is 8 m/s.

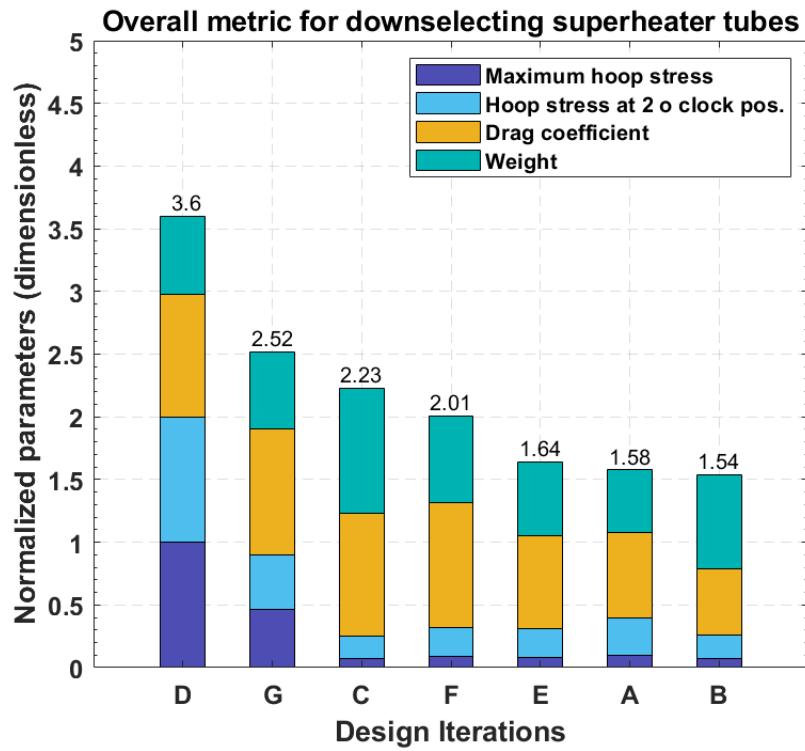


Figure 3.13: Overall metric for down selecting superheater tube shapes; equal weightage is applied for each parameter; lower overall metric represents better design.

Table 3.2: Overall metric for several tube designs for superheater tubes

<i>Design</i>	<i>Hydraulic diame- ter (mm)</i>	<i>Maximum tensile hoop stress (MPa)</i>	<i>Hoop stress at 2 o'clock position (MPa)</i>	<i>Drag coeffi- cient</i>	<i>Weight of the tube (kg/m)</i>
A: Baseline	30	90.3	61.2	0.497	3.92
B: Teardrop convex profiled circular ID— Asymmetric	30	65.4	38.5	0.389	5.92
C: Teardrop convex profiled circular ID— Symmetric	30	62.7	35.8	0.719	7.91
D: Teardrop convex profiled non-circular ID— Symmetric	30.42	892	203.2	0.719	4.93
E: Teardrop concave profiled circular ID— Asymmetric	30	73.1	47.1	0.542	4.7
F: Teardrop concave profiled circular ID— Symmetric	30	78.1	46.6	0.731	5.48
G: Teardrop concave profiled non-circular ID— Symmetric	29.99	408	90.1	0.731	4.91

3.8 Chapter Summary

This chapter focuses on the geometric modification of circular superheater tubes for assisting ash deposition reduction and improved creep life. We discussed classes of solutions for reduced fouling nature categorized as a) shaking or flicking, b) aerodynamic geometry—redirection of flow, and c) surface patterning or texturing. We deployed macroscale geometry change, i.e., aerodynamic solution, as the first-degree solution for the tube redesign. A teardrop-shaped tube design and its variants, which could be enabled via additive manufacturing techniques like L-PBF, were studied. These tubes were assessed along with the baseline tube for hoop stress development, creep life, and flow field characteristics around them, and an overall metric was prepared to compare different shapes. Among all studied tubes, the teardrop-shaped convex profiled asymmetric tube performs best.

1. Heat transfer calculation performed for a 2D model of the tube infers higher temperatures around the 2 o'clock position of the tube. The thermal boundary conditions coupled with the structural model show that the teardrop convex profiled asymmetric tube encounters $\sim 37\%$ lower tensile stresses at critical locations and $\sim 27.5\%$ lower maximum hoop stress compared with the baseline design. This leads to improved creep life for the superheater tubes.
2. The particle tracing analysis coupled with the single-phase flow turbulent CFD model showed that the teardrop convex profiled design has $\sim 2\%$ less drag than that of the baseline design. Also, the number of particles encountering the tube surface was lower in the case of the teardrop shape convex profiled tube than that of the circular tube design.

3. A Schlieren optics experiment was leveraged to visualize the flow around the differently shaped tubes. The streamlines generated through Schlieren videos indicated that the teardrop shape has a more streamlined flow (laminar flow) and reduced fluid flow mixing compared with the circular-shaped tube. Reduced mixing of flue gas might lessen the ash deposition in the proposed shape compared with the current superheater tube design.

Chapter 4

Advanced Cooling Channel Redesign for AM-HPT Components

4.1 Background and Theory

In section [2.1.2](#), we briefly discuss the requirement of cooling channel schemes for tip shrouds. Expanding on the same discussion, we comprehensively discuss 1) the rise of superalloy crucial to the HPT section and 2) why we need advanced cooling schemes, in general, for other turbine components that traditionally contain internal cooling channels.

4.1.1 Material Advancements for Turbine Application

Thermal efficiency and specific output of a gas turbine are primarily functions of turbine rotor inlet temperature (RIT) and compressor pressure ratio. However, the material's melting point of the HPT section is significantly lower than the stoichiometric (by burning 100% fuel) and, hence, a limiting factor to achieving maximum efficiency. Through the

material revolution of more than 60 years, superalloys have established a strong position in gas turbines, helping engines run hotter [107]. Superalloys refer to a class of nickel and cobalt-based alloys explicitly designed for high-temperature operation. Here, "super" refers to these alloy's excellent strength in terms of tensile, creep, and fatigue at high temperatures. Additionally, these alloys offer exceptional ductility and toughness at elevated temperatures (above 0.5 solidus temperature until ~ 0.75 solidus temperature) [107]. This made Ni-based alloys such as IN718 the most successful, comprising more than 50% of metal usage in GE Aviation engines [107]. With manufacturing and alloy development, we commercially stand at single-crystal airfoil casting, further improving the stress rupture life and increasing allowable blade temperature [108]. Yttria-stabilized, zirconia-based thermal barrier coating (TBC) is further deposited ($0.13\text{--}0.25\text{ }\mu\text{m}$) on the blades to increase the RIT [107].

However, most superalloy-based engine components require 8–10 lb of raw material for a 1 lb finished part, thus incurring high costs [107]. Additive manufacturing technology best fits such scenarios, saving material waste and additionally allowing site-specific properties and complex geometries. As discussed in sections 1.1.2 and 2.1.2, AM-based superalloys are emerging for similar applications to run engines hotter. Material development and TBC are not alone sufficient for overall increased efficiency. Integrated cooling channels traditionally built in turbine airfoils and turbine stator vanes (nozzle-guided vanes) are vital for efficient engine operation. AM allows the complexity that comes "free" and enables complex designs that weren't possible traditionally.

4.1.2 Heat Transfer Requirements

For a typical modern gas turbine, RIT can be as high as $1,500^{\circ}\text{C}$; cooling air of 650°C is extracted from the compressor region and fed into turbine components to lower its

temperature to approximately 1,000°C [109]. Turbine cooling air and leakage might be as high as 25–30% of the engine primary flow [110]. Thus, optimizing the coolant extracted from the compressor region is crucial. Han [111] discussed that, for future prospects, we might need to produce blades that can withstand higher thermal loads for engines to be run by alternative fuel sources such as hydrogen. Stimpson [112] also discusses that it is crucial to have a high thermal gradient from solid to fluid for higher heat transfer. Han and Stimpson suggested producing micro- or mini-channels near the airfoil skin to increase the thermal gradient and enhance heat transfer. Such channels are challenging to produce with conventional manufacturing. Thus, lately, AM-enabling complex geometries have been heavily studied for focusing on mini- or micro-channels.

This research thoroughly reviewed conventional convective heat transfer enhancement features. Next, we studied the existing exploration of AM-enabled newer geometrical concepts and the AM effect on traditional enhancement features.

4.2 Literature Review

4.2.1 Conventional Cooling Mechanisms

With respect to turbine vanes' and blades' internal cooling technologies, several enhancement techniques are used in different sections of the component, such as jet impingement, rib turbulator in serpentine channels, pin-fins, dimples in rectangular/circular channels, and compound cooling techniques [113, 114, 109, 115]. For current research goals, we reviewed the rib turbulator, pin-fins, and dimples. In the mid-chord section of the airfoil, the design rule calls for rib turbulators or dimples, whereas pin-fins as cooling features are implemented for the trailing edge. The evolution of these features over the last 30 years is summarized in detail

in Table 4.1. In the presence of rib turbulators, the boundary layer is tripped often, and then, because of the reattachment of the boundary layer that follows, the heat transfer coefficient is enhanced. Further enhancement can be achieved by skewing the ribs or through broken ribs, which triggers secondary flow generation. Dimples provide heat transfer enhancement without significant penalty on pressure drop. Additionally, the above designs have little dependence on Reynolds numbers. Pin-fins are heavily used in confined spaces, provide high surface area, and induce turbulence in the coolant and horseshoe vortices formations near the wall. This leads to high heat transfer, however, with an increased pressure penalty. Based on the literature review, one can conclude that there has been significant research and development with reference to the modification of shapes and geometries for gas turbine cooling.

4.2.2 Additively Manufactured Advanced Cooling

With the growing ability of AM to make components with complex geometries, the community has embarked on the fabrication of heat transfer components, including tip shoes, with integrated cooling channels and other innovative concepts. The research trajectories can be grouped under two categories: 1) the effects of AM roughness on similar conventional design features, and/or 2) investigating advanced complex shapes or features that were otherwise not possible to pursue with traditional manufacturing.

For related applications, in certain sections of the component, designers are limited by space and thus may resort to micro- or mini-cooling channels. Such geometries received substantial attention lately through AM, and the following options have been explored on the heat transfer augmentations and friction factor: 1) straight mini channels with various shapes, 2) rectangular wavy channels with various amplitudes and wavelengths, 3)

Table 4.1: Categorization and evolution of internal heat transfer mechanisms

<i>Internal Heat Transfer Mechanism</i>	<i>References</i>	<i>Key Outcomes</i>
Rib turbulator (used in the mid-chord of the airfoil)	Webb et al. [125]	• Rib turbulator oriented at 90 °to the coolant flow • Enhancement due to periodic tripping and reattachment of the boundary layer
	Han et al. [126] Han and Park [127]	• Skewed rib turbulator (angled to coolant flow) • Improved heat transfer enhancement compared with orthogonal ribs • Creates additional secondary flow patterns promoting fluid mixing
	Han et al. [128] Park et al. [129] Han and Zhang [130] Han et al. [131]	• V-shaped ribs provide more uniform heat transfer enhancement • Broken/discrete ribs provide improved heat transfer compared with continuous • Discrete ribs induce breakup of secondary flows
	Rallabandi et al. [132]	• Studied rounded rib corners as opposed to sharp corners • Reduced pressure drops with similar heat transfer enhancement
Dimple/concavity (alternative to ribs)	Chyu et al. [133]	• Nusselt number enhancement of up to 2.5 compared with the smooth channel • Reduced pressure drop compared to rib tabulators • Reduced weight • Structural weakening of airfoil
	Chyu et al. [134] Moon et al. [135]	• Heat transfer enhancement is independent of the Reynolds number • Heat transfer occurs outside of the dimple

Continued from **Table 4.1**

<i>Internal Heat Transfer Mechanism</i>	<i>References</i>	<i>Key Outcomes</i>
Dimple/concavity (alternative to ribs)	Borisov et al. [136]	• Proposed “banked” dimple design • 4.5 Nusselt number augmentation; however, with increased pressure penalty • Traditional hemispherical dimples provide the best overall thermal performance
	Khalatov et al. [137] Acharya and Zhou [138] Chang et al. [139] Liou et al. [140] Jordan and Wright [141]	• Different shaped dimples were studied • Square dimples provide better heat transfer augmentation compared with traditional hemispherical dimples • Teardrop dimples perform better than squares • V-shaped dimples provide enhancement up to 2.8 compared with smooth channels • V-shaped dimples benefit heat transfer at a high Reynolds number
Pin-fins (present at the trailing edge of the airfoil)	Metzger and Haley [142] VanFossen et al. [143] Brigham et al. [144] Armstrong et al. [145]	• Array of short, cylindrical pins • Increases surface area for heat exchange • Enhanced heat transfer due to increased turbulence of the coolant and horseshoe vortices formation at the junction of the pin and channel end wall

Continued from **Table 4.1**

<i>Internal Heat Transfer Mechanism</i>	<i>References</i>	<i>Key Outcomes</i>
Pin-fins (present at the trailing edge of the airfoil)	Chyu et al. [146] Siw et al. [147] Moon and Kim [148] Wright et al. [149]	• Various shaped pin-fins were studied • For a given heat transfer enhancement, half-diamond and hemispherical pin-fins are preferred, compared with full-diamond and cylindrical pin-fins because of reduced weight • Triangular pin-fins provide improved heat transfer enhancement compared with circular; however, with relatively increased pressure penalty • Fan-shaped pin-fins provide overall improved performance compared with round counterparts • Airfoil pin-fins provide enhanced surface area. Comparable pressure drops as round pin-fins with slightly decreased heat transfer enhancements

complex shaped pin-fin designs, 4) varied lattices infills, and 5) modification of inherent surface roughness through modification of AM process parameters. Wildgoose et al. [116] studied various shaped AM-built straight channels such as circle, trapezium, ellipse, and different polygonal shapes, suggesting the pentagon performed the best among all for heat transfer augmentation; it was comparable to that of the circle, however, with a reduced friction factor. Kirsch et al. [117, 118] proposed an AM-built rectangular wavy channel that provides more than 2 times heat transfer augmentation. This unique geometrical mini-channel design is considered state-of-the-art from an AM research perspective. Corbett et al. [119] extended the work on rectangular wavy channels by studying the impact of various wavelengths and amplitude. They concluded that with increasing relative waviness (A/λ), where A is amplitude and λ is wavelength, the friction factor increases significantly with a modest improvement in heat transfer augmentation. Kirsch and Thole [120], Ferster et al. [121], and Corbett et al. [122] studied additive pin-fins with varying shapes and spatial densities. Highly dense pin-fins result in a high friction factor with marginal improvements (>15) in heat transfer augmentations. Also, triangular-shaped pins with their tip facing upstream experience the least friction factor, whereas diamond-shaped pins seem to perform better in terms of heat transfer augmentation. Although these designs may lead to high-pressure drops, researchers have incorporated AM pin-fins in nozzle guide vanes in gas turbine engines [123, 118, 124]. Other design features such as ligament-based lattices and triply periodic minimal surface (TPMS) structures, are attractive owing to their high surface-area-to-volume ratio and because they can possibly be fabricated through AM.

Kaur and Singh [150] performed an extensive review of AM lattices, concluding that higher thermal conductivity can be achieved by modifying the porosity of the lattices. Also, AM lattices might have better thermal characteristics than metal foams; however,

from a fabrication standpoint, they are limited by the minimum member size. Parbat et al. [151] studied Ni-based lattices fabricated through metal AM, concluding there is a significantly higher pressure drop as opposed to other heat transfer augmentation methods [118]. Stimpson et al. [152, 153] and Thole et al. [118] explored the effect of AM roughness on heat transfer augmentation and friction factor, proposing a correlation for AM roughness with sand grain roughness. Also, Snyder and Thole [154] studied the tailoring of AM roughness of rectangular straight channels by manipulating build parameters, suggesting certain combinations of roughness lead to better thermal performance. However, as Stafford et al. [155] showed, AM roughness might locally enhance heat transfer, but we still require other methods to augment thermal performance for global thermal enhancements.

4.3 Proposed Design & Hypothesis

4.3.1 Class of Solution: Facilitating Heat Transfer

Like gas turbine engines, where we need effective cooling systems for efficient operation, nature also needs thermoregulation for the survival of organisms. Living organisms maintain their body temperature within a narrow range, mainly manipulating through physiological and behavioral methods. Different heat transfer mechanisms or their combination, namely 1) conduction, 2) convection, 3) radiation, and 4) evaporation, are adopted for different climates in different species maintaining a stable body temperature. In most species, such mechanisms occur through skin acting as a thermal medium or through their built structure [156]. Several work by Badarnah and Fernandez [156], Badarnah et al.[157], Mota-Rojas et al. [158], Tansey and Johnson [159], McCafferty et al. [160], Tan and Knight [161], and Zhu et al. [162] discuss overarching thermoregulation methods in detail in various organisms.

Based on these reviews and other related literature, broad categories of thermoregulation solutions are discussed below:

1. **Fractal branching:** Fractal nature for flow optimization and hence increasing the surface area is pervasive in nature. This could be in trees, lungs, blood vessels, and many more for the circulatory or respiratory system, minimizing pumping power requirements. This is discussed in detail by Bejan et al. [52]. As opposed to straight parallel microchannels, few studies explored fractal micro-channels in laminar and turbulent regimes, also suggesting correlation for fractal geometric parameters for electronic heat sink applications [163, 164, 165, 166].
2. **Insulation:** Conductance-based insulation methods are pervasive in numerous animals to survive in colder environments. These solutions can be either external (keratin matrix, e.g., feathers or furs) or internal (lipid-filled cells with low thermal conductivity, e.g., blubber) [156]. Birds' insulation material comprises multiple layers of downy and contour feathers forming a plumage. These feathers assist in trapping the still air closer to the skin, eventually reducing heat loss [160, 156]. Feather structure and plumage depth vary across the body, making some surfaces available for heat dissipation when needed. Evidence shows feature size variance across seasonal climate patterns [160]. In addition to fur or feathers, a thick subcutaneous blubber layer provides an added insulation function, specifically in mammals in the Arctic and Antarctica.
3. **Interconnected channels:** In addition to fractal branching, interconnected channels are commonly found in morphologies carrying blood or gas. Particularly, it can be found in alveoli; alveoli cover $\sim 90\%$ of the human lung's total volume [167]. Anatomy [167] demonstrates that these alveoli contain interconnected spheres, increasing the

overall surface area in addition to resembling grapes-like assembly. Further compliant motions (inflation or deflation) enhance the surface area and reduce pressure drop. Lung-inspired heat exchangers have received attention lately from industries and academia for different applications [168, 169, 170, 51].

4. ***Surface protrusions:*** Unlike warm-blooded animals, cold-blood animals must facilitate heat transfer from body to ambient or vice-versa. For instance, Bergles emphasized that the dorsal protrusion of large lizards provides an optimized extended surface for augmenting thermal contact [52]. Along similar lines, several studies support osteoderms evolving as the thermoregulation feature in various creatures such as Stegosaurus, Alligator, and Crocodylian [171, 172, 173, 174, 175]. However, Veenstra et al. [176] rejects this theory of osteoderms evolution for the thermoregulation function. Other examples of organisms leveraging surface texture for heat transfer purposes are elephants, which have a combination of wrinkly, rough, creased skin surfaces and sparse hair cover ($\approx 650 \text{ hair } m^{-2}$) [160], and butterflies contain nano- and micro-structures on its wing for thermoregulatory functions [177, 178]. Additionally, Termite mounds also demonstrate the usage of surface protrusions; it varies the wall thickness, mound surface design, or projecting surfaces [157, 156]. Mounds in warm environments (Savannah) have thinner walls with numerous ridges enhancing the surface area. In contrast, mounds found in colder climates (gallery forest) have thicker walls with few protrusions (reduced surface area) [157, 156].

5. ***Flapping of parts:*** Flapping of ears in elephants has also received significant attention for thermoregulation function augmenting the convective heat transfer [179, 180, 181]. However, the contribution of ear flapping remains low ($< 8\%$ of the

total heat loss). Despite this, there has been interest in studying the flapping of parts for convective heat transfer enhancement [182, 183] for engineering applications. This is further discussed later in section 6.1.2.

6. ***Self-expansion or contraction:*** A crucial method of thermoregulation in various species is self-expansion or contraction of the blood vessels to vary the heat transfer. The formal terms for this are vasoconstriction and vasodilation, allowing the variation in the blood flow amount near the skin and consequently either preventing or promoting heat transfer [157, 161, 159, 160, 158]. Calculation shows that blood flow in the capillaries is in the laminar or transition phase ($Re \in (1224, 1632)$). Since the capillaries deform their shape on demand, it changes the pressure inside the vessel.
7. ***Shivering:*** Shivering is another thermoregulation mechanism adopted by humans, birds, pigs, and many other organisms. Shivering is defined as involuntary, rapid, oscillatory contractions of skeletal muscles. During shivering, Adenosine triphosphate (ATP) is hydrolyzed, producing the energy that is released as heat [159, 161]. Shivering triggers when the maximum vasoconstriction (energetically inexpensive method) is achieved and responds to skin temperature as the stimulus. Shivering can induce heat production at approximately five times the basal metabolic rate [161]. In pigs, the primary mechanism for thermoregulation solely depends on shivering from the skeletal muscle [158].
8. ***Thermal hotspot:*** A few species also exhibits thermal windows or hotspot that becomes their major source of heat transfer location. For instance, duck's bill [184], elephant's ears (20% of the elephant's total surface area) [185, 186], and toucan bill [187]. The surface area of the toucan's bill is 25 - 40 times ($\sim 30 - 50\%$ of the body

surface) larger than the predicted value from scaling relationships. This plays a critical role in significant variance in heat loss from 25% to 400% of the resting heat production in adults [187].

4.3.2 Cooling Channel Design based on Class of Solution

Which Design Solution? Most of the class of solution for facilitating heat transfer in nature either maximizes the surface area or induces energetically expensive processes. Morphology emerges as one of the most critical characteristics adopted by nature for thermoregulation, where the surface area-to-volume (SA/V) ratio depends on the local requirements, such as heat rejection or heat insulation (varies with climates, body mass, and metabolic rate). Energetically expensive methods such as shivering and flapping of parts either trigger as a secondary solution after the morphological solution reaches its limit or do not contribute significantly to the overall thermoregulation. Self-expansion and contraction require compliant movements of the wall. Since the current focus of the dissertation aligns with the morphological solution, we do not pursue this concept. However, this concept can be studied for future purposes for intermittent heat transfer operation [60]. Unlike nature, in our application, we need to avoid thermal hotspots. We do not adopt this concept as it is a secondary step to heat transfer design (locally varying the heat transfer capabilities by changing the SA/V).

We considered the pressure penalty an additional requirement to select among the morphological solutions. Insulation-based features are unsuitable as they contradict our heat dissipation mission. Fractal branching in nature is often accompanied by compliant motions assisting in pressure penalty reduction. Drag, which consists of form drag and skin friction drag, must be minimized to reduce the overall pressure drop. It is beneficial to eliminate

macro-features, minimizing form drag and causing overall pressure drop reduction. Based on this, we choose **surface protrusion** as a potential solution. Similar to surface protrusion in nature enhancing surface area, can we modify surface texture, increasing wetted surface area, hence the expected increase in heat transfer?

Research Gaps: Revisiting the literature, many have studied pathways for enhancing heat transfer for additive micro-/mini- channels for high-temperature applications exhaustively. In contrast to channel-based or traditional designs, only a limited study explores surface texture modification by introducing micro-features to maximize the surface area for enhanced heat transfer.

Proposed Concept: We propose to introduce the surface texture by adding micro-features along with the roughness created by AM. The proposed design is motivated by our goal to increase the wetted surface area per unit volume and concurrently to avoid macro-features reducing the pressure drag and, hence, overall pressure drop. AM inherent roughness ($\sim 13 \mu\text{m}$), traditionally considered a non-desirable feature, can enhance the heat transfer characteristics of a cooling channel [153]. The 3D microfeatures with AM roughness will create *multiscale surface characteristics* that further increase the area available for heat exchange. Therefore, the current scope of the research focuses on studying micro-features with uniform shape and size, in addition to AM roughness for internal cooling purposes. For the context of this problem statement, we try to answer the following questions:

1. How much wetted surface area enhancement can one achieve through multi-scale rough features, and how much does it translate to heat transfer enhancements?
 - (a) How much geometrical deviation occurs during AM with reference to micro-features compared to the intended design features?

- (b) What is the isolated effect of AM inherent roughness in enhancing heat transfer?
- (c) Is there any upper limit on the increase in surface area for heat transfer enhancement?

4.4 Test Coupon Description

4.4.1 Test Coupon Geometry Description

4.4.1.1 Concept Design and Test Coupon Size Description

Amongst primitive shapes, tetrahedron provides the highest surface area to volume ratio of 7.21 per unit volume. Thus, for the micro-features, we choose 3D Tetrahedrons to generate the surface texture, which is tightly packed throughout the internal surface of the test coupons (shown in Figure 4.1a). After additively manufacturing the coupon, as expected, it provides the multi-scale regular rough micro-tetrahedrons due to AM inherent roughnesses (shown in Figure 4.1b). Figure 4.1c illustrates the clear image of each tetrahedron where h is the height of the tetrahedron, and a is the side length of the base triangle (Figure 4.1d). A flat rectangular coupon is used for fundamental research purposes and experimental testing. The dimensions were chosen based on the ceramic surface heater size available and previous literature for similar applications. The test coupon size was modified to accommodate the thermocouple holes for measuring the surface temperatures and assembly pins, as shown in Figure 5.3. Five thermocouple holes are placed axially to study temperature variation of the metal wall temperature.

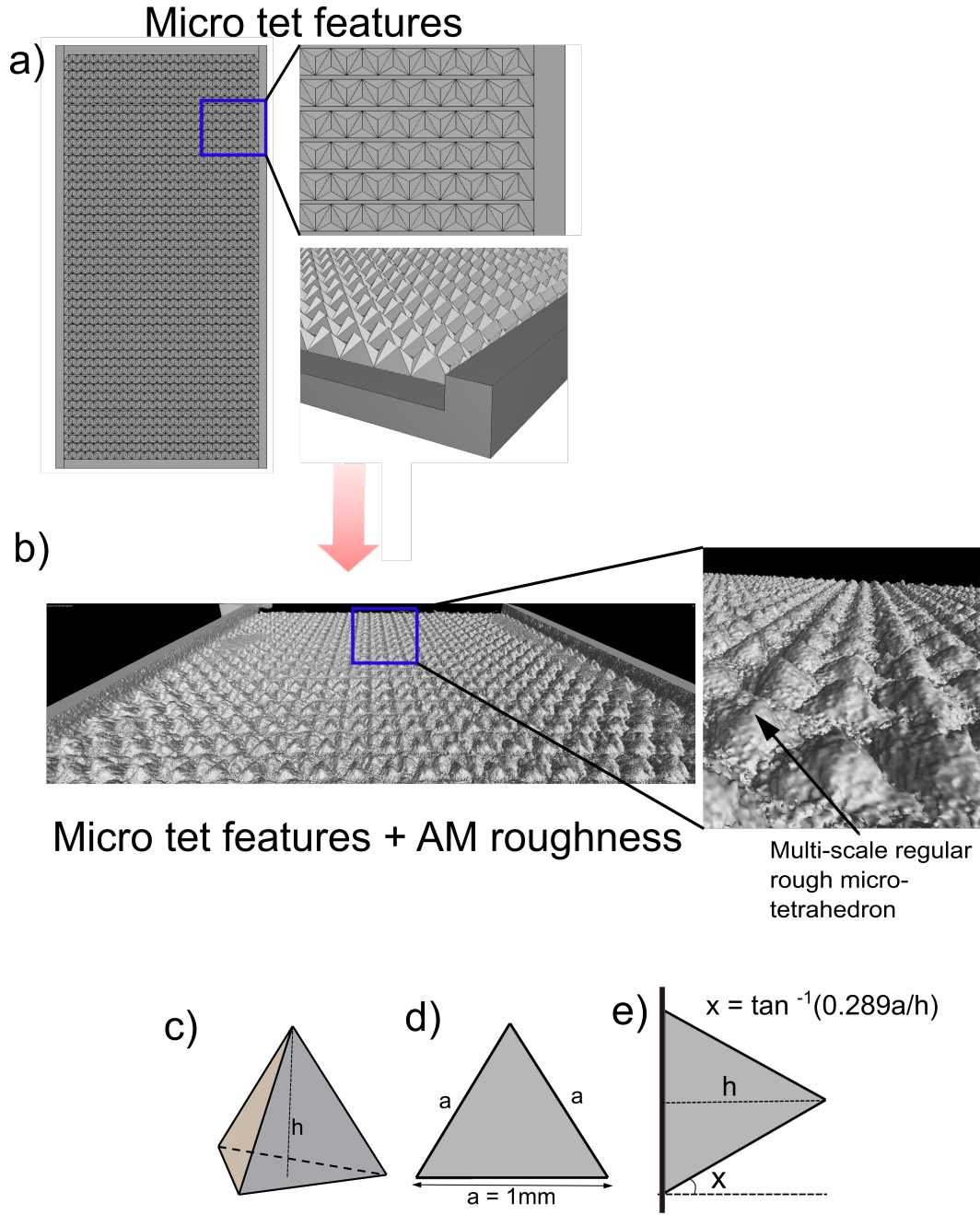


Figure 4.1: a) 3D Tetrahedron Features added on the surface, b) 3D Tetrahedron Features after Additive Manufacturing generating a multi-scaling regular rough micro-tetrahedrons, c) 3D image of each Tetrahedron, d) Equilateral triangle at the base of the Tetrahedron of 1 mm length for each case, e) Height of the Tetrahedron (h) is a function of overhang angle (x); three different h were chosen in the current research

4.4.1.2 Geometries of Test Coupons for testing

A total of five test coupons are tested and evaluated in the current research. We print these coupons additively using the Laser-Powder Bed Fusion method in a vertical build direction. We chose a vertical build direction to eliminate variability in roughness on different inner walls of the coupon due to other build direction angles. We perform the study by sequentially increasing the roughness, starting from machining roughness. The first test coupon was initially additively manufactured with a smaller flat rectangular slot, which was later machined to $23\text{ mm} \times 1.5\text{ mm}$ using EDM wire, eliminating AM roughness (see Figure 4.2a). The second coupon was printed with a flat rectangular slot of $23\text{ mm} \times 1.5\text{ mm}$; however, this contains as-AM roughness (see Figure 4.2b). The third, fourth, and fifth test coupon contains the same flat rectangular slot; however, it also contains micro-tetrahedrons packed exactly as shown in Figure 4.1a. The base triangle dimension (a) remains the same in all these three cases; h is the only variable and is varied as $167\text{ }\mu\text{m}$, $344\text{ }\mu\text{m}$, and $500\text{ }\mu\text{m}$, respectively (Figure 4.2a-e).

The relation between h , the surface area of the tetrahedron (SA), and the overhang angle (x) is presented in Table 4.2. The surface area of each tetrahedron linearly increases with height h . The maximum surface area, which can be accommodated in the test coupon, is $500\text{ }\mu\text{m}$. h values more than $500\text{ }\mu\text{m}$ also lead to a lower overhang angle (measured from horizontal), which is difficult to print and thus not considered for testing purposes. The h values of $167\text{ }\mu\text{m}$ and $344\text{ }\mu\text{m}$ are chosen further to observe the effect of a systematic increase in the micro-feature heights; also, the feasibility of 3D printing with various overhang angles (40° and 60° respectively) for each tetrahedron. For simplicity, throughout the paper, we will

Table 4.2: Relation between the height of tetrahedron, total surface area of each tetrahedron, and overhang angle for tetrahedron. Yellow shaded shows the chosen parameters for the current set of test coupons (Tet – 167, Tet – 340, and Tet – 500).

<i>Height of Tetrahedron (mm)</i>	<i>Surface area of each tetrahedron (mm²)</i>	<i>Overhang angle (degree)</i>
h	$SA = 0.433a^2 \left(1 + \frac{1}{\sin\left(\tan^{-1}\left(\frac{0.289a}{h}\right)\right)} \right)$	$x = \tan^{-1}\left(\frac{0.289a}{h}\right)$
1.64	2.92	10
0.79	1.70	20
0.50	1.30	30
0.34	1.11	40
0.29	1.04	45
0.24	1.00	50
0.17	0.93	60

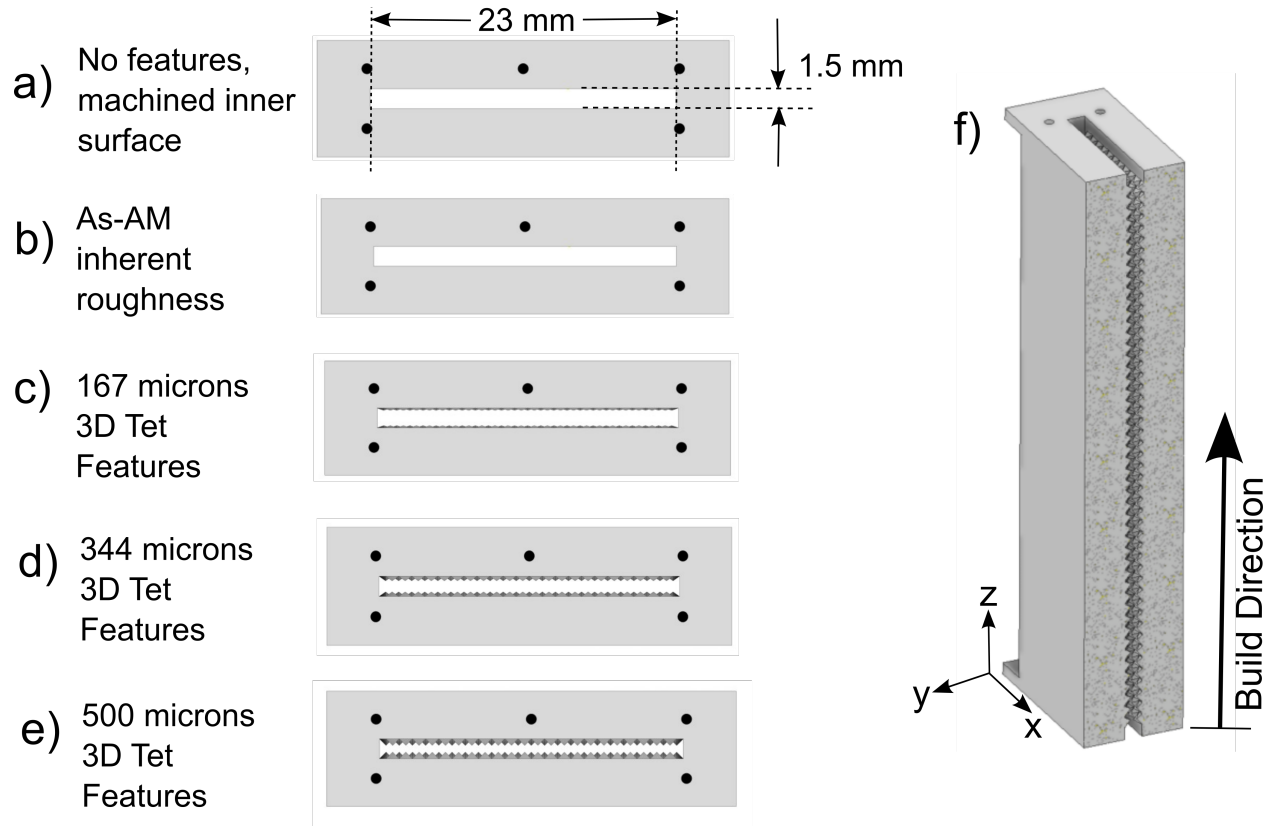


Figure 4.2: a – e) Geometries of test coupons for evaluation of thermo-hydraulic performances, f) shows a cross-section of the component to highlight the build direction (z-direction).

use EDM, As-AM, Tet-167, Tet-340, and Tet-500 to refer to these coupons 1, 2, 3, 4, and 5, respectively.

4.4.2 Additive Manufacturing of the Test Coupons

The test coupons were additively manufactured using Concept Laser M2 Cusing machine with SS316L. Though the end applications will be in Ni-based or Co-based superalloy, currently, we chose SS316L for feasibility demonstration purposes. Only one coupon was printed on EOS M290. The build parameter data is presented in Table 4.3. The energy density is defined as $\frac{P}{VHT}$ in each machine and is maintained approximately the same for similar AM roughness and material density. P is the laser power (W), V is the laser scan speed (mm/s), H is the hatch spacing (mm), and T is the layer thickness (μm). The contours and skins were kept on while the edge was turned off.

4.5 Roughness Characterization of the Test Coupons

Modifying the surface textures through engineering micro features affects the overall surface roughness of the coupon. Also, since AM provides an inherent roughness, the printed geometry is expected to deviate from the designed geometry. Thus, it is crucial that we understand the additively manufactured final geometry to be tested in terms of roughness, geometric deviations, and the effect of multiscale surface characteristics on the wetted surface area.

Due to the inaccessibility of the internal cooling channel, a non-destructive evaluation technique, 3D X-Ray Computed Tomography (XCT), was performed with a resolution of 8 μm per voxel for each coupon, including coupon#1 EDM. Since high-resolution XCT leads to

Table 4.3: Laser Powder Bed Fusion machine details used to print all the test coupons.

<i>Coupon</i>	<i>Machine</i>	<i>Power</i>	<i>Laser Speed (V)</i>	<i>Hatch (H)</i>	<i>Particle Particle Size</i>	<i>Laser Thick- ness (T)</i>	<i>Energy Density</i>
		<i>W</i>	<i>mm/s</i>	<i>mm</i>	<i>μm/s</i>	<i>μm</i>	<i>J/mm³</i>
EDM As-AM Tet-340 Tet-500	Concept Laser M2 Cusing	195	812	0.1	15-45	50	48
Tet-167	EOS M290	214	928	0.1	15-45	40	57

a lower field of view, a portion of each coupon was only studied for roughness characterization and is assumed to be similar throughout the coupon. The 2D image stack collected from XCT was 3D reconstructed and post-processed in a commercial tool called VGStudioMax software. A sub-voxeling of 2.5 μm was used for interpolating the data between the measured resolution. Further, XCT machine details and measurement settings are mentioned in the Appendix.

4.5.1 Surface Roughness Characteristics

Figure 4.3 represents the XCT images of each coupon's representative section (sampling area), using which surface roughness was calculated. ASME B46.1-2019 was used for reporting surface roughness, and S_a values were calculated. S_a is the arithmetic average of the absolute values of the measured height deviations from the mean surface taken from the sampling area, which is calculated as below in the Cartesian coordinates:

$$S_a = \left(\frac{1}{A_e} \right) \int_0^{L_y} \int_0^{L_x} |Z(x, y)| dx dy \quad (4.1)$$

Where A_e is the sampling area, L_x and L_y are the length and width of the surface area section, and Z is the height from the mean surface. Figure 4.3 shows that the roughness was increased by texturing the surface by ~ 23 times the surface having no features and ~ 11 times the As-AM roughness.

4.5.2 Overlay of CAD with the additively printed surface

To study the deviation of additively manufactured geometry with respect to the designed geometry, we overlaid the XCT 3D Tet-500 surface with the actually designed surface.

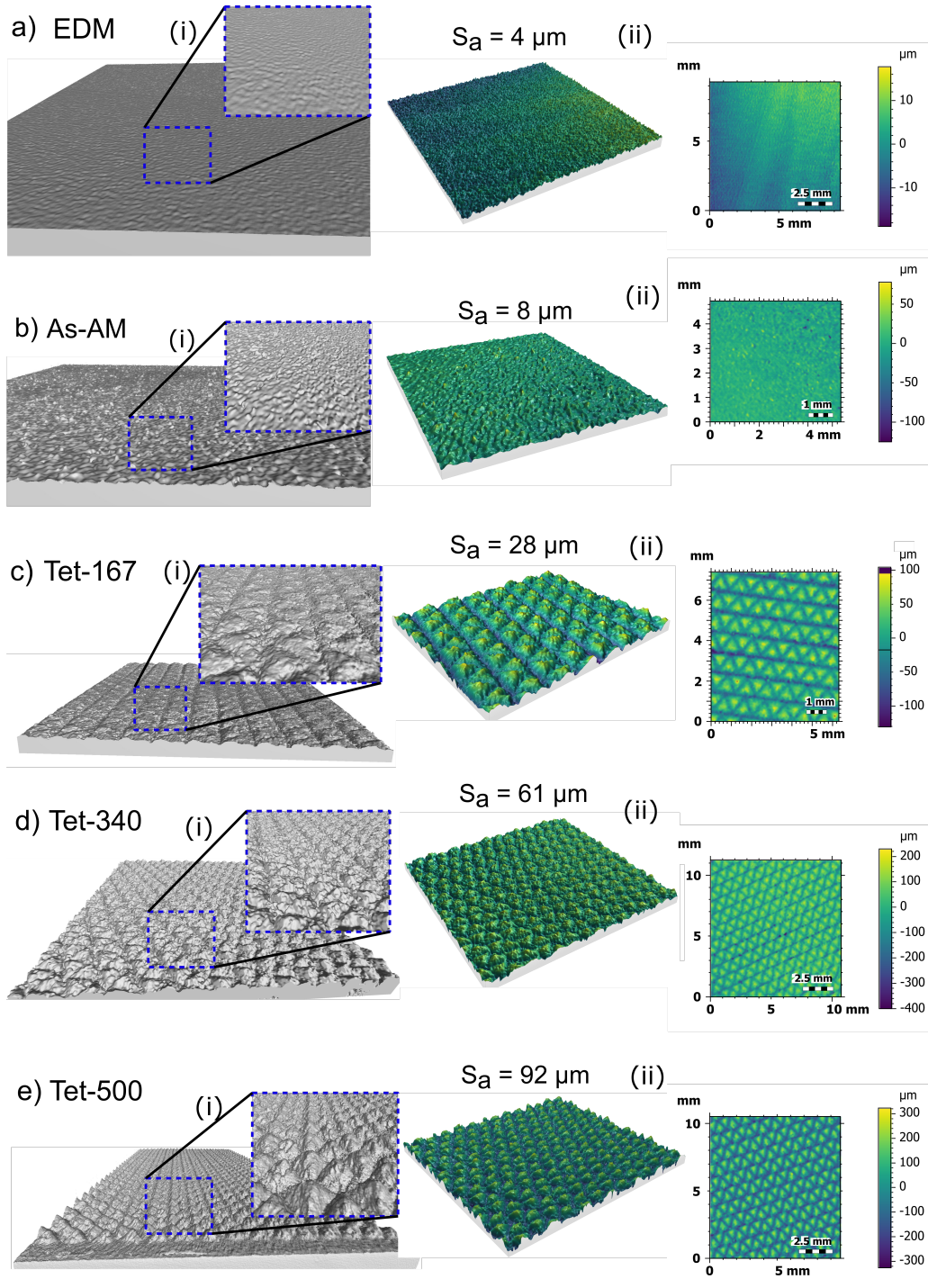


Figure 4.3: Demonstrates roughness characterization for each test coupon a) EDM, b) As-AM, c) Tet-167, d) Tet-340, and e) Tet-500 where (i) XCT image of a portion of the test coupon and (ii) corresponding height map of roughness in 3D and 2D respectively

Figure 4.4a represents the x-y plane (z is the build direction) of an overlay where the greyish geometric form is the XCT, and green lines indicate the CAD geometric form. Figure 4.4b shows the contour map of the magnitude of deviation from the actual geometry. The blue indicates the additively manufactured form was smaller than the CAD, red indicates larger, and green indicates close conformance with the CAD. It shows that the printed micro-Tet feature heights are smaller than the designed ones. However, this remains counterintuitive as one would expect printed features to be bigger than designed due to inherent AM roughness. This could be attributed to laser-powder bed printing settings. We provide a beam offset of 0.1 mm to compensate for the laser beam diameter to avoid the enlargement of part from the actual profile. Though this setting helps in conforming with the side profile of the geometry, it causes the elimination of the peak of tetrahedrons (see Figure 4.4b). This could be visualized through the schematic presented in Figure 4.4c where red lines are the scan strategy, the beam offset in green, and yellow is the region deviating from the designed geometry. Thus, one can infer that the final micro-Tet features were smoothed and shorter than the designed Tet features.

4.5.3 Multi-scale roughness features and wetted surface area

Using the XCT data, we calculated the wetted surface area for each coupon (see Table 4.4). The EDM surface was 4% higher than the actual CAD dimension. This indicates that even after finishing the surface using EDM wire, there are remaining surface burrs leading to higher surface area. Additionally, inherent AM roughness increased the surface area by 12%, while the engineered roughness nearly doubled in surface area (i.e., 192%) as opposed to having no features. Despite not printing the peak of each tetrahedron precisely, we obtained more

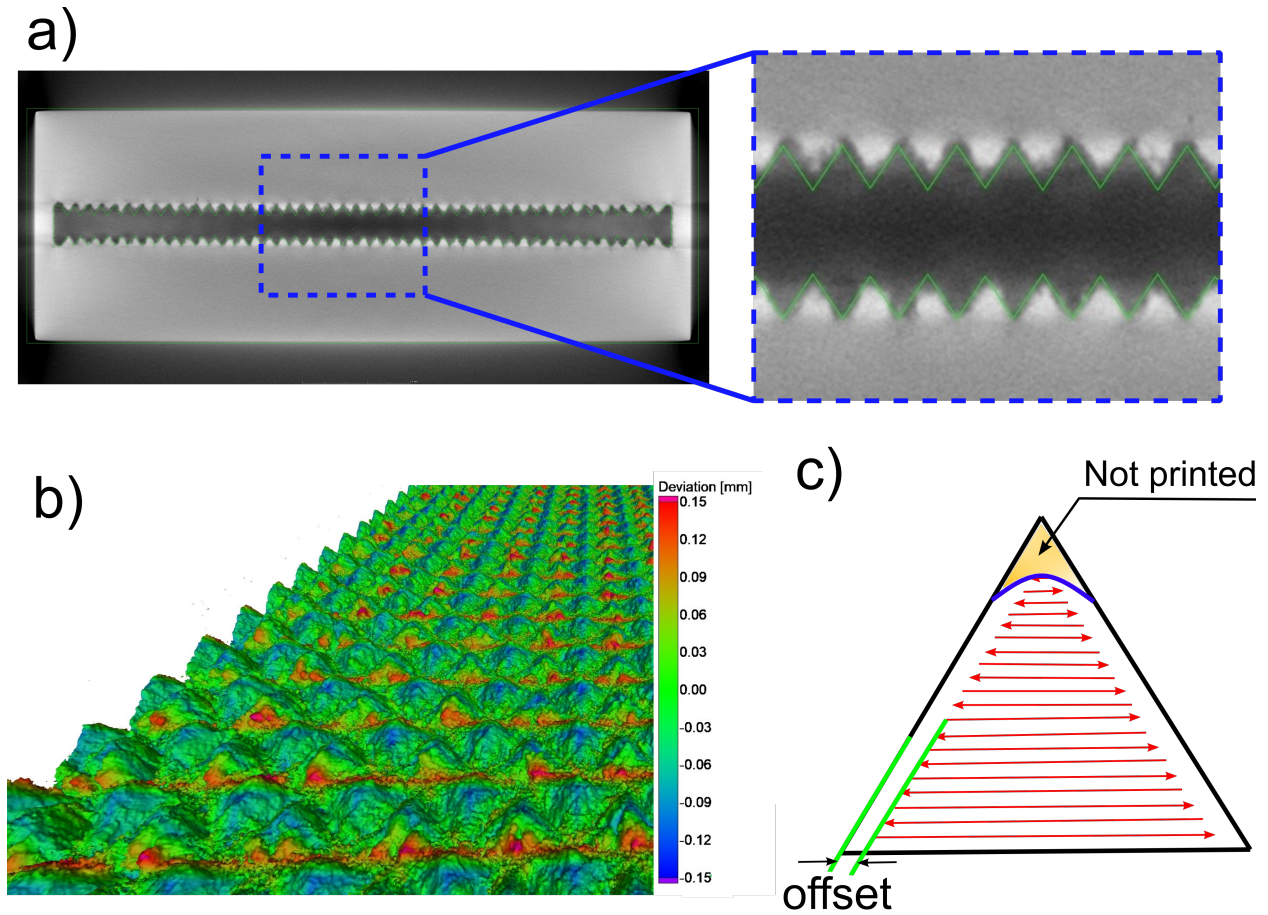


Figure 4.4: a) 2D overlay of XCT X-Y cross-section of Tet – 500 with corresponding CAD profile, b) 3D overlay of XCT of Tet – 500 with CAD profile reporting the deviation from actual profile. c) schematic for beam offset provided and hatch pattern stating the cause of AM geometry deviation from designed profile.

wetted surface area for micro-Tet featured coupons compared to the corresponding CAD-based wetted surface area (see Table 4.3). We attribute this additional surface area to the inherent AM roughness. Multi-scale surface features increase the surface area significantly without adding additional design features to the coupon. We hypothesize that this increase might benefit heat transfer by providing more surface area per unit volume for heat exchange without significantly increasing pressure drop.

Multi-scale surface characteristics were further visualized using a Scanning Electron Microscope (SEM) to distinguish between the sources of roughness due to inherent AM roughness and our micro-Tet design features. A Tet-500 coupon was sectioned and characterized in SEM. Figure 4.5a shows a section of the coupon presenting the micro-Tet features. The images (Figure 4.5b & c) show two different tetrahedrons. It is evident that the tetrahedron containing an overhang surface in the build direction leads to more irregular Tet features as opposed to the alternative row of Tet features. Also, with Figure 4.5d, one can infer the sources of AM inherent roughness that are as follows. 1) The power particles originally belonging to the powder batch partially melt and bonded to the surface. This roughness will be related to the powder particle size distribution (PSD) of the material feedstock used for the AM. 2) The roughness is also dictated by stochastic geometric features brought about by local melting conditions that are related to section geometry, process parameters such as hatch spacing, and scan strategy. 3) The roughness can also be traced back to surface undulations due to the solidification of melt pools under different melt pool sizes and shapes (e.g., humping or balling phenomena).

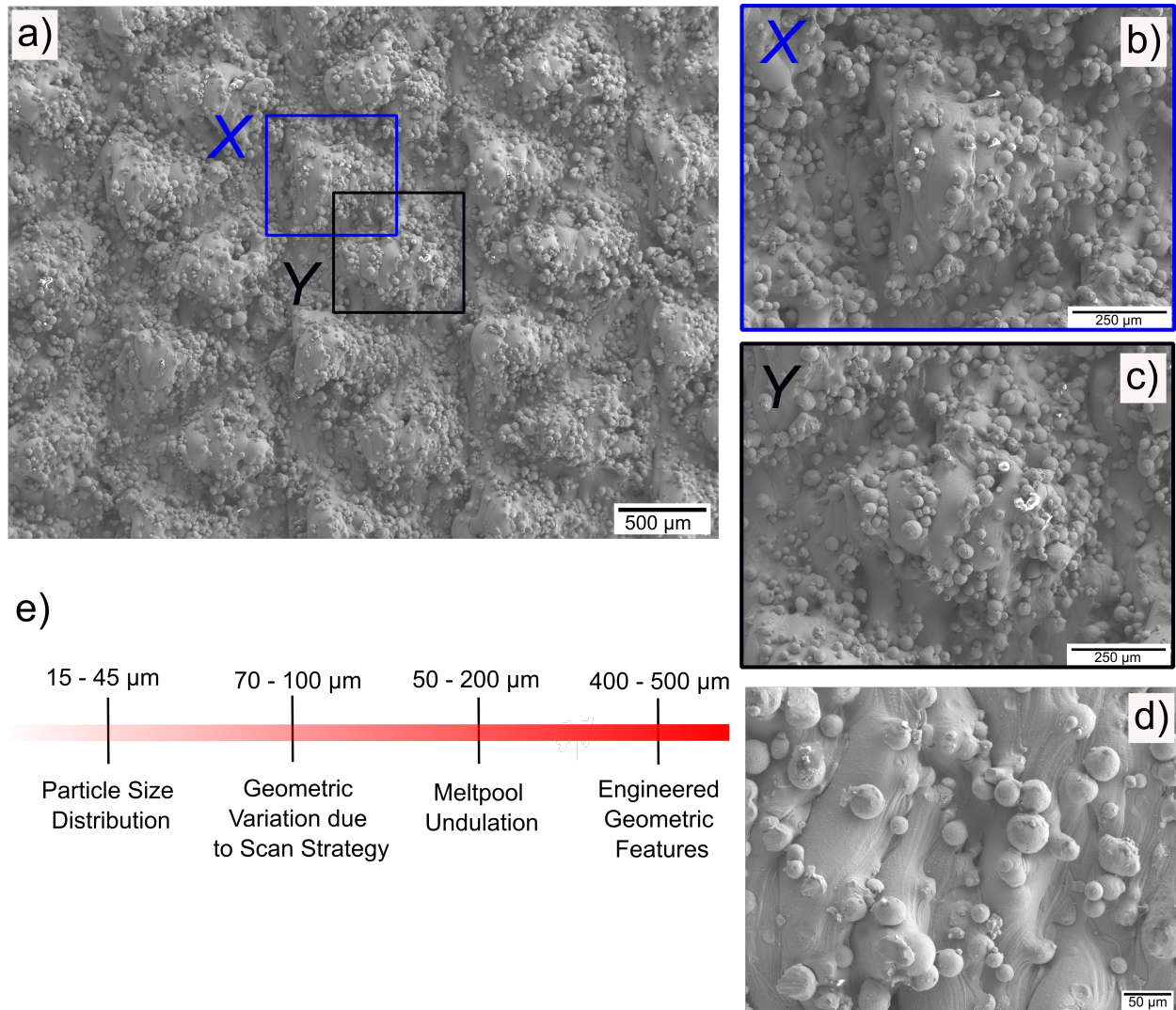


Figure 4.5: a) SEM image of a section of tet - 500 surface representing the actually printed tetrahedron feature; b & c) b represents one tetrahedron which conforms to the actually designed tetrahedron compared to an alternative set of tetrahedrons shown in c; d) SEM micrograph of AM induced roughness. This image is taken from the entry region of the coupon where no tetrahedrons were present, highlighting PSD and Melt-pool-based roughness features; and e) shows feature size ranges contributing to AM roughnesses and our contribution to further enhance roughness by adding geometric features

Table 4.4: Wetted Surface Area of each test coupon (CAD and AM printed geometry) and surface area enhancement compared to EDM surface.

<i>Description</i>		<i>Wetted Surface Area – CAD (mm²)</i>	<i>Wetted Surface Area – XCT (mm²)</i>	<i>Area Enhancement w.r.t. EDM CAD</i>
C01	EDM	2450	2553	1.04
C02	As-AM	2450	2733	1.12
C03	Tet-167	2739	2802	1.15
C04	Tet-340	3490	3706	1.51
C05	Tet-500	4281	4696	1.92

4.6 Chapter Summary

We explored the class of solutions from nature for the thermo-regulation function, which broadly relies on the SA/V ratio or energy-expensive processes. Various organisms, such as termite mounds' ridges, elephant wrinkly skin, butterfly scales, and protrusions in lizards and other cold-blooded animals, expand their SA/V through surface protrusions in different shapes and sizes at varied length scales for effective heat transfer. Based on this, we proposed the modification of surface texture by introducing multi-scale surface roughness features. This design was primarily motivated by 1) increasing surface area and 2) reducing pressure drag by avoiding macro-sized features. We designed test coupons for multi-scale surface characteristics and manufactured them additively. To study this, we created a total of five test coupons where we systematically increased the roughness: 1) EDM smooth surface and no feature, 2) Contains As AM roughness only, 3) 3D micro-tet with 167 μm height and AM roughness, 4) 3D micro-tet with 340 μm height and AM roughness, and 5) 3D micro-tet with 500 μm height and AM roughness. We performed surface characterization through X-ray computed tomography and Scanning Electron Microscope.

The following are the conclusions from this chapter:

1. We produced closely packed 3D micro tetrahedron features in addition to AM roughness-inducing multi-scale surface characteristics. We chose tetrahedron as a micro-feature, as it provides a maximum surface area to volume ratio amongst the primitive shapes.
2. XCT shows that through micro-roughness features, overall surface roughness could be increased by ~ 23 times machining roughness and ~ 11 times AM roughness.

3. L-PBF manufacturing machine setting (beam offset) causes non-conformance of the micro-Tet features compared to designed CAD. It could lead to $\sim 100 \mu m$ or more material smoothing for the Tet-500 peak.
4. Multi-scale roughness feature could lead to a surface area enhancement of up to 200% for the Tet-500 coupon. This significant rise in wetted surface area is only possible through Additive Manufacturing technology.
5. Scanning electron microscopy images distinguished the AM roughness and micro-features, demonstrating sources of roughness levels.

Chapter 5

Test Rig Setup Development and Design Validation

5.1 Experimental Test Rig Setup Development

Because these additively printed cooling channels will inevitably contain AM inherent roughness, using computational-based approaches to study the thermal-hydraulic characteristics might be challenging. In broadly used CFD – RANS–based computational approaches, equivalent sand grain roughness is used to account for actual roughness. Bons et al. [188, 189] emphasized that accurate modeling of roughness for profile loss and surface heat transfer is still out of range of present capabilities because the current model does not account for the different roughness effects on skin friction, heat transfer, and boundary layer. Other computational models that might provide better solutions, such as discrete-element and high-resolution CFD models, require more effort. Recently, Thole et al. [118] and Stimpson et al. [112] proposed a correlation between equivalent sand grain roughness with additive roughness; however, the absolute average error is up to 20%, and the maximum absolute

error is up to 71%. Additionally, capturing the detailed shape and pattern of the surface micro-elements to study the fluid interaction is extremely expensive computationally [37]. Hence, setting up an experimental test rig to study these additively manufactured micro/mini cooling channels is crucial. An experimental test rig was developed by Thole et al. [118] to study the effect of AM inherent roughness [118, 154, 153], AM builds direction and process parameters [190, 191], geometrical shapes [192, 193, 194], and pin-fins [121?] for heat transfer and friction factor characteristics of additively built micro-channels for similar applications. Because the experimental setup was built using polymers, it is expected to operate at lower temperatures while the components operate at relatively high temperatures ($\sim 800^\circ\text{C}$) in engine scale. A previous study by Shih et al. [195] suggests that for heat transfer ducts with enhancement features assuming wall-to-bulk temperature, T_w/T_b , near unity is still an open question. Thus, developing a high-temperature experimental rig facility to capture the near-real scenario closely might be preferable. The present study proposes developing a high-temperature experimental rig facility to test the AM cooling channel test coupons. Currently, the setup design is limited to attaining 200°C ; however, the same setup is planned to be an even higher temperature testing facility by introducing an extra cooling mechanism for the mass flow controller's operating temperature range.

The goal of this experiment is to calculate the following two parameters to evaluate different designs:

1. **Heat Transfer Coefficient:** For HTC, we need to measure the surface wall temperature and inlet air temperature entering the test coupon and applying constant heat flux on the test coupon.

2. **Pressure Drop:** To measure pressure drop across the channel, we must measure pressures at the inlet and outlet using a pressure transducer.

Thus, through the experimental setup, we need to measure the surface wall temperature of the test coupon at axially different locations and pressure measurements at the inlet and outlet of the test coupon.

5.1.0.1 Circuit Diagram

Figure 5.1 represents the schematic diagram of the setup, which is divided into three sections: 1) Room Temperature Zone—the compressed air system with pressure regulator and pressure relief valve operating at ambient temperature; 2) Heat Zone—test coupon section operating under high temperature; and 3) Low Temperature Zone—exit air zone where air is at relatively higher temperature compared with ambient temperature.

In this setup, the air is the working fluid entering the cooling channels in the test coupon at ambient temperature and a fixed inlet pressure of 80 psia. The test coupon walls are heated through surface heaters. The air will gain heat from the coupon acting as a cooling fluid and will exit the coupon at higher temperatures. A different mass flow rate was achieved by leveraging a mass flow controller present downstream of the test coupon. Congruent to the circuit diagram, the overall setup is shown in Figure 5.2. The following subsections will discuss the setup design in detail.

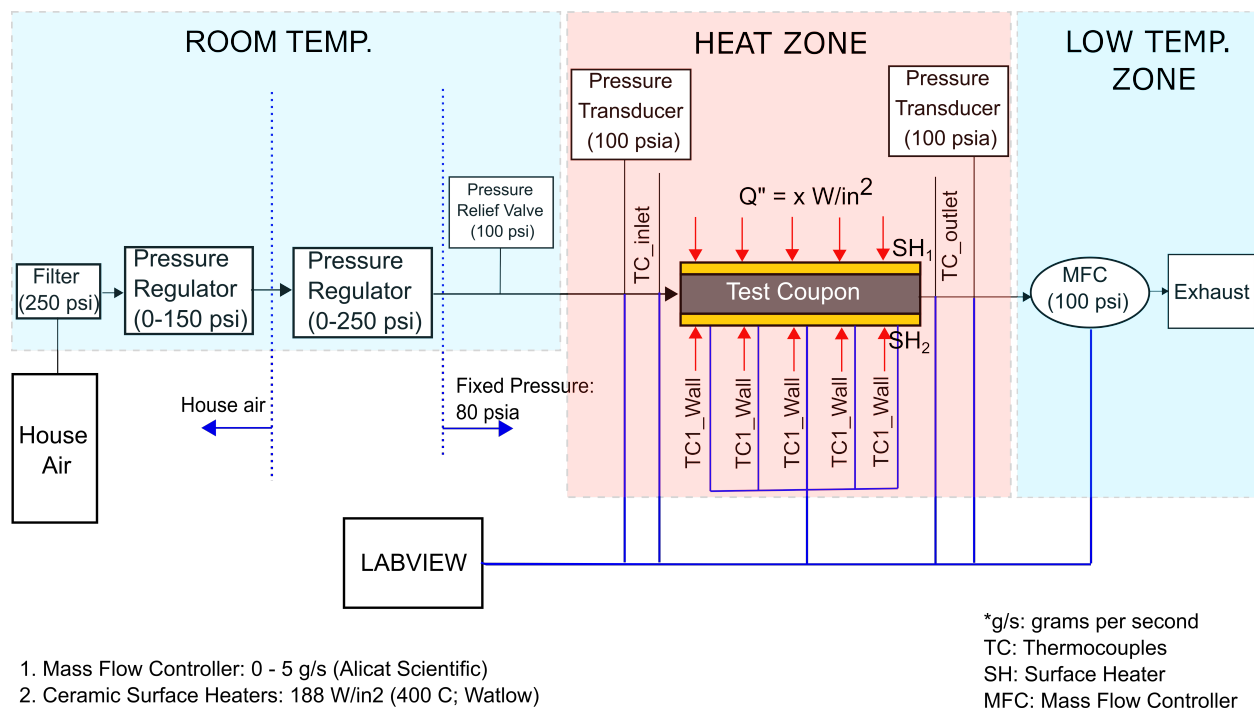


Figure 5.1: Schematic for the experimental setup.

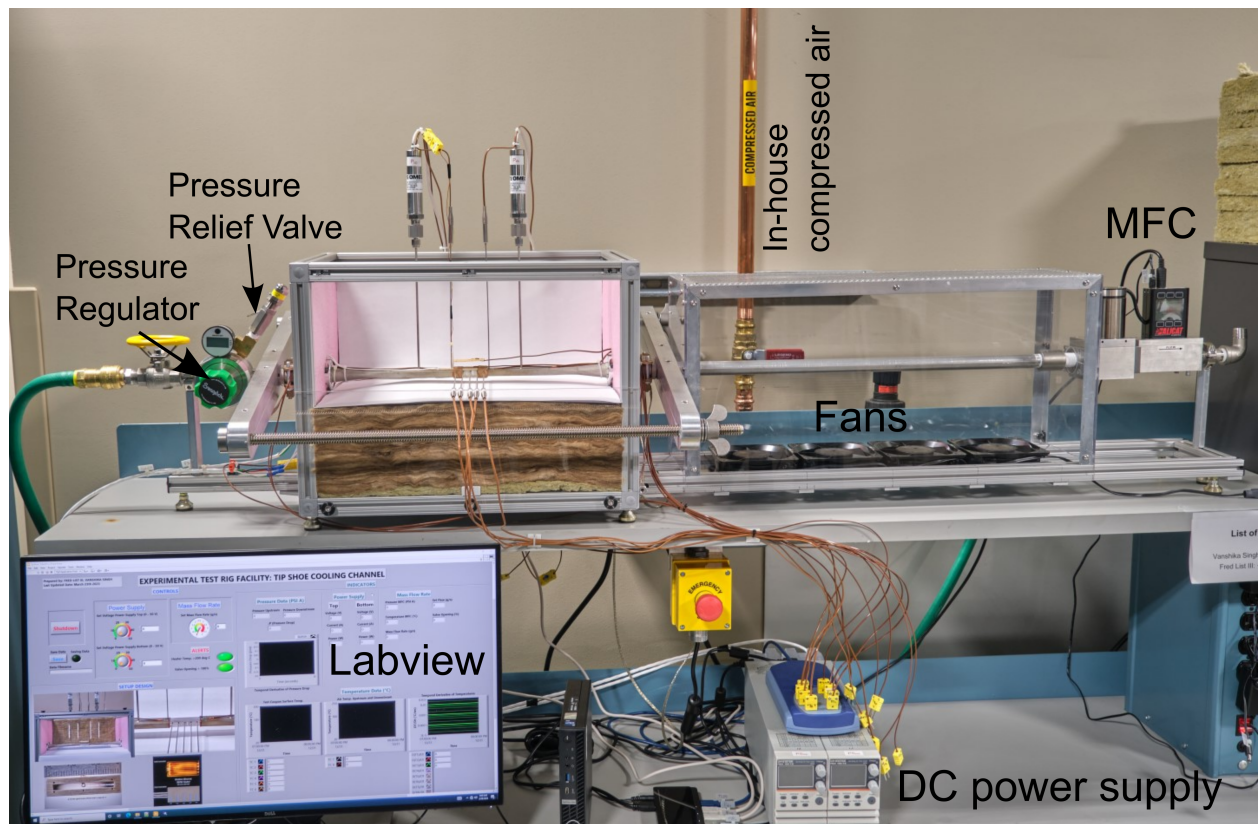


Figure 5.2: Overall setup developed congruent to the schematic shown in Figure 5.1.

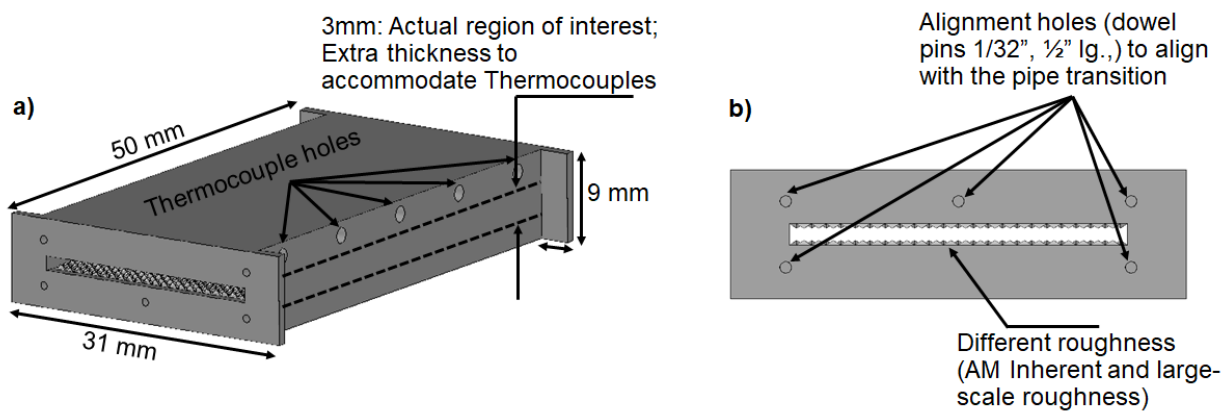


Figure 5.3: a) Isometric view of one test coupon with thermocouple holes, and b) top view of a test coupon showing alignment holes.

5.1.1 Test Coupon Section Design

5.1.1.1 Test Coupon Design

In actual applications, the parts are sector pieces; however, a flat rectangular coupon is used for conceptual research purposes. The test coupon size is $50 \times 25 \times 3$ mm. The dimensions were chosen based on the surface heater size available and previous literature for similar applications [119, 122]. This was modified to accommodate thermal instrumentation and assembly pins. A 3 mm thickness was added on both sides to accommodate 1/16" (1.58 mm) thermocouple holes. Five thermocouples were machined axially to measure the test coupon wall temperature in the axial direction. A detailed description of the test coupon geometry is given in Figure 5.3.

5.1.1.2 Transition Piece Design

Because the test coupons have a rectangular cross-section, a transition piece is required to bridge the circular pipes with the test coupons. Thus, we designed a 228 mm long transition piece wherein we provide an entry length of 100 mm that is ~ 37 times the hydraulic diameter of the test coupon, meeting the practical application guidelines (see Figure 5.3). These parts were manufactured using metal for two reasons: 1) we need material of high melting points for high-temperature applications, and 2) mechanical integrity is necessary because the inlet air needs to be entered at high pressure for the controlled velocity of air inside the coupon. In addition, the design must account for developing the fluid flow. The thickness of the part was maintained constant throughout the length, equivalent to 19.05 mm ($\frac{3}{4}$ " NPT pipe thickness). Therefore, these parts were additively manufactured on Concept Laser M2 using

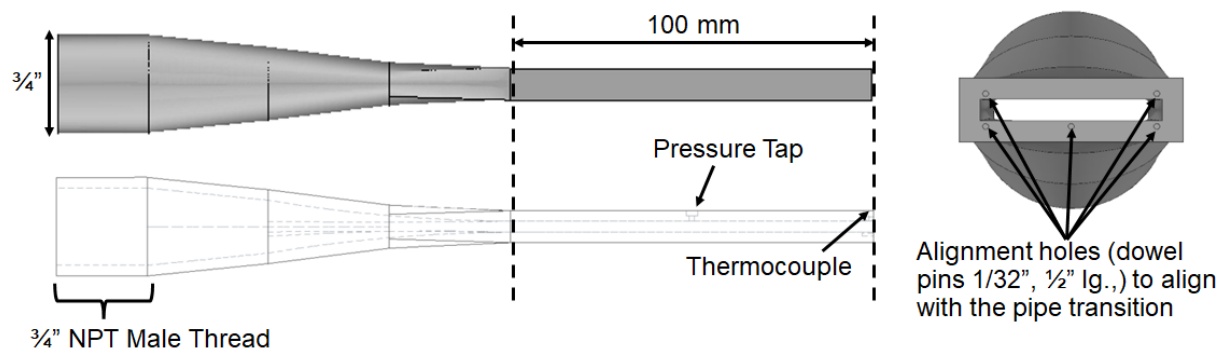


Figure 5.4: Front view of the transition piece describing the pressure instrumentation hole and assembly pin holes.

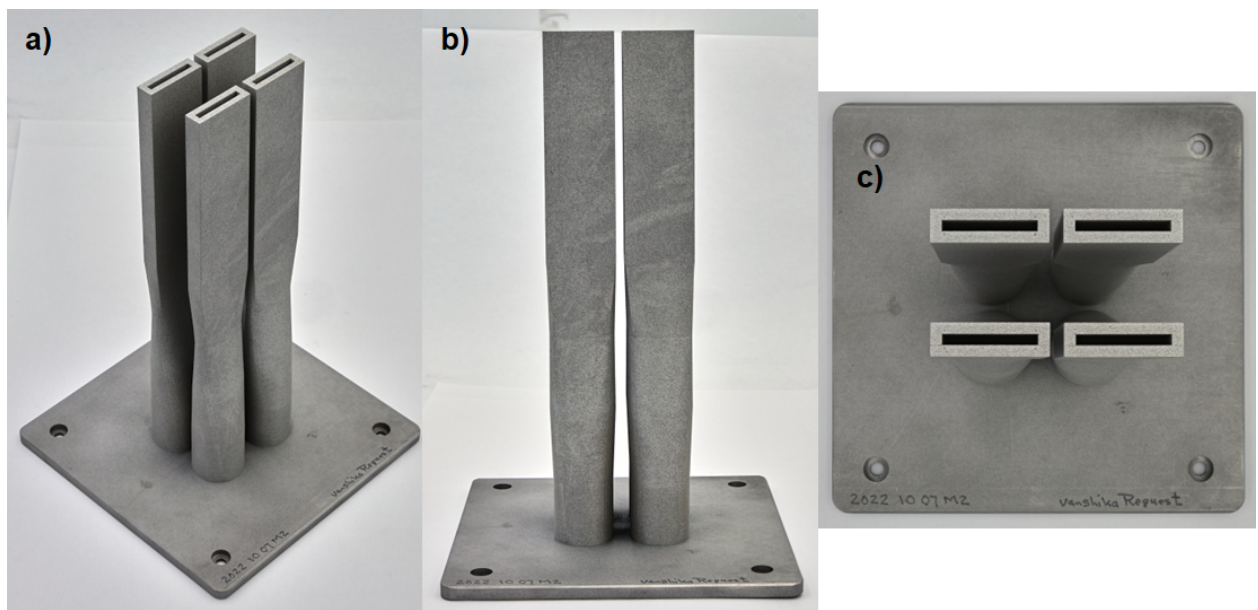


Figure 5.5: Additively printed transition pieces before machining: a) isometric view, b) front view, and c) top view.

machine (see Figure 5.4) using SS316L. The assembly pins, pressure taps, and NPT male threads were machined on the printed parts.

5.1.2 Detailed Experimental Facility Design

5.1.2.1 Thermal, Pressure, and Flow Instrumentation

Thermal instrumentation is required to measure the surface wall temperature of the test coupon at different axial locations. Five axial thermocouples are used to measure the average wall temperature of the test coupon displayed in Figure 5.6c. Inlet and outlet air temperature are also measured using the thermocouples. These temperatures are needed to quantify 1) the amount of heat added to the bulk cooling fluid and 2) the fluid bulk temperature. These thermocouples are brazed right before the entry and exit of the test coupon, measuring the midstream fluid temperature, as shown in Figures 5.6a and b. These temperature measurements are required to calculate the average heat transfer coefficient. For inlet, outlet, and surface wall temperature measurements, we used K-type thermocouples of 1/16" diameter (Omega: TJ36-CAIN-116E-12 T/C ASSEMBLY W/TRANS JOINT).

Two calibrated pressure transducers (Omega: MMA100USBHP1C0T8A4, 0–100 psi, Absolute, USB, 1/4" NPT Male, Cable, (-40–185°F)) are chosen so that each can be used to measure the inlet and outlet pressure across the test coupon. Pressure transducers were placed 50 mm away from the test coupon as shown in Figure 5.6a. These pressure measurements are required to calculate the pressure drop across the test coupon and, consequently, the friction factor for each test coupon. A pressure regulator (Swagelok KHF1GJM828A20000 0-250 psi), in combination with a pressure relief valve, is used at the inlet. The relief valve is calibrated for 100 psia. This pressure regulator is placed before

the inlet pressure transducer and manually adjusted so that the inlet pressure transducer reads ~ 80 psia. This high inlet pressure is used to maintain the Mach number ≤ 0.1 inside the test coupon. The current study's maximum Mach number reaches 0.07 with Reynolds numbers $\sim 20,000$. Pressure relief valves are chosen based on the pressure regulator ratings.

A calibrated mass flow controller (MCR-250SLPM-D-20X32), which can be used to adjust the mass flow rate from 0 to 5 g/s, was placed far downstream of the test coupon. The mass flow controller (MFC) adjusts its ball valve to maintain the set mass flow rate, so placing it downstream is beneficial. MFC needs to be maintained below its operating temperature (i.e., 60°C). Thus, a set of fans was used toward the exit of the setup for cooling the exit air entering the MFC. Its temperature is constantly monitored through the LabVIEW interface.

Owing to ORNL laboratory safety requirements, the additively built section, which includes the test coupon and the transition part (which does not conform with the standard tube dimensions), was enclosed in a plexiglass model setup. Laser-cut, fabric-reinforced silicone rubber gaskets were placed between the transition pieces and the test coupon to reduce air leakage. Two compressive tie rods were added outside the frame to further prevent minor leakage.

5.1.2.2 Heating Element and Power Source

Constant heat flux (W/in^2) would be provided to calculate the heat transfer coefficient, and maintenance of constant heat flux is critical to the experiments. Therefore, two surface ceramic heaters were chosen from Watlow. Heater wattage was decided based on 1) the average wall temperature to be maintained and 2) the mass flow rate. See the details of the chosen heater in Table 5.1. Heaters were chosen to perfectly fit the size of the test coupon. See Figure 5.7a for the illuminated image of the surface heater demonstrating the

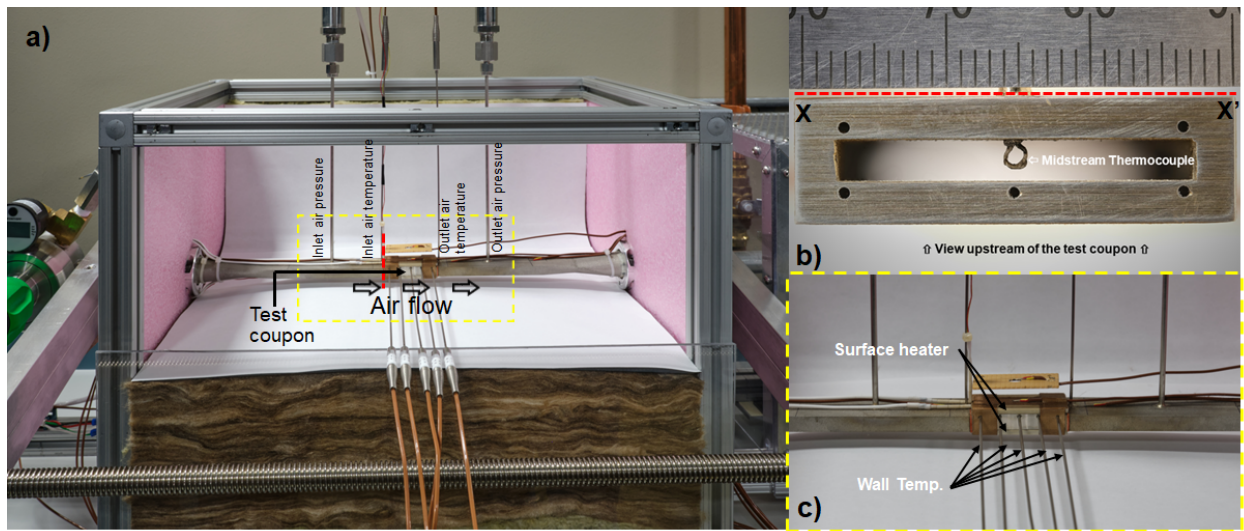


Figure 5.6: a) Test section showing the test coupon attached with the transition pieces, b) thermocouple positioning measuring the air midstream temperature, and c) surface heater placement on both test coupon walls and thermocouples measuring the wall temperature.

Table 5.1: Watlow surface ceramic heater specification

<i>Heater specification</i>	<i>Values</i>
Maximum heater temperature	400°C
Wattage	188 W/in ²
Dimension of heater	50 × 25 mm
Voltage	120 V

heating coil wrapped inside the ceramic. The coiling inside the heater suggests it might not provide uniform heat flux but will be close to ideal. Figure 5.7b shows the metal wall thermocouple position with respect to the heater. Surface heaters were placed on the upper and lower sides of the test coupon using a highly conductive silver thermal paste (CW7100, Chematronics, 100% silver filled silicone grease). The perfect bonding between the heater and the metal wall using the conductive paste is crucial to avoid overheating. LabVIEW monitors the surface heater temperature using the integrated K-type thermocouple so we do not overheat the heater (see Figure 5.7c). Similar heat flux is provided through the surface heater using the DC power supply. The heat flux magnitude varies with different mass flow rates and test coupons. Silicon-based insulation was used and packed around the test section to minimize heat loss to the environment. These two heaters were placed on top of the test coupon surface.

DC power supply provided to the surface heater is chosen per the applications' requirements: INS PSW 160-7.2 (23072.1 Instek DC Power Supply). Because the heater rating is 120 V, 363 W, it won't be able to draw a current of more than 3.025 A. The chosen DC power supply has a power rating of 360 W (160 V, 7.2 A), so if the current increases, the voltage will reduce to meet the rated power requirements [196]. Additionally, the power will be set up and controlled using a LabVIEW, where the upper limit for power is set to 100 VDC. Appropriate grounding has also been provided throughout the experimental setup.

5.1.2.3 Data Acquisition

A LabVIEW code was developed for setting the input and fetching the data from the instruments. Figure 5.8 presents the screenshot of the LabVIEW setup developed. This is primarily divided into two different sections: controls and indicators. We set each DC

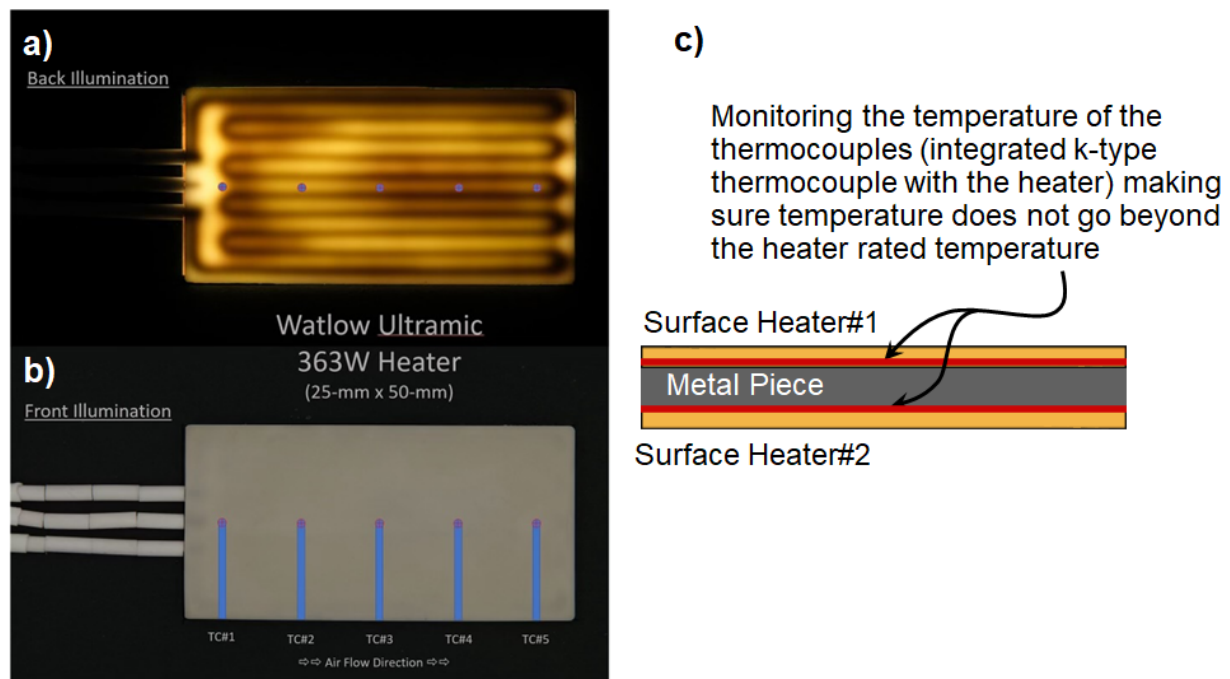


Figure 5.7: a) Back illumination of the surface heater highlighting the coiling of the heating element, b) metal wall axial thermocouple location with respect to the ceramic heater, and c) schematic of heater (yellow) placement and silver paste (red) on test coupon (grey).

power supply's voltage and mass flow rate in the controls section. We monitor the MFC temperature ($\neq 55^{\circ}\text{C}$), valve opening in MFC ($< 100\%$), and heater temperature ($\neq 350^{\circ}\text{C}$) for the alerts. In the top half of the indicators section, we read all the flow-related data; in the bottom half, we read all the temperature-related data. Data is fetched from the instruments at 1 Hz. For steady-state condition, flow and temperature are monitored through graphs: for flow steady state, $\frac{d(\Delta P)}{dt} \leq 0.002$ psi/s for temperature steady state, $\frac{d(T_s)}{dt} \leq 0.0005^{\circ}\text{C/s}$. Approximately 3–4 hours of operation were required for thermal steady state. Inlet and outlet air pressures and temperatures are averaged over 30 and 300 s, respectively. A second-order polynomial is fit to the time-averaged temperatures and pressures.

5.2 Test Procedure

5.2.1 Data Reduction Method

Two sets of experiments were conducted for each coupon. First, only pressure drop measurements were done without any power supply for mass flow rates varying from 1.5 to 4.5 g/s. Calibrated pressure transducers were used to measure inlet and outlet pressures. They were placed 50 mm upstream and downstream of the test coupon. The second set of tests includes power supply through a surface heater for varying mass flow rates ranging from 1.5 to 4.5 g/s where the metal wall temperature was maintained at approximately 100°C . The amount of heat added to the cooling air is calculated using equation 5.1:

$$q_{conv}'' = \frac{\dot{m}C_p(T_{dn} - T_{up})}{A_s} \quad (5.1)$$

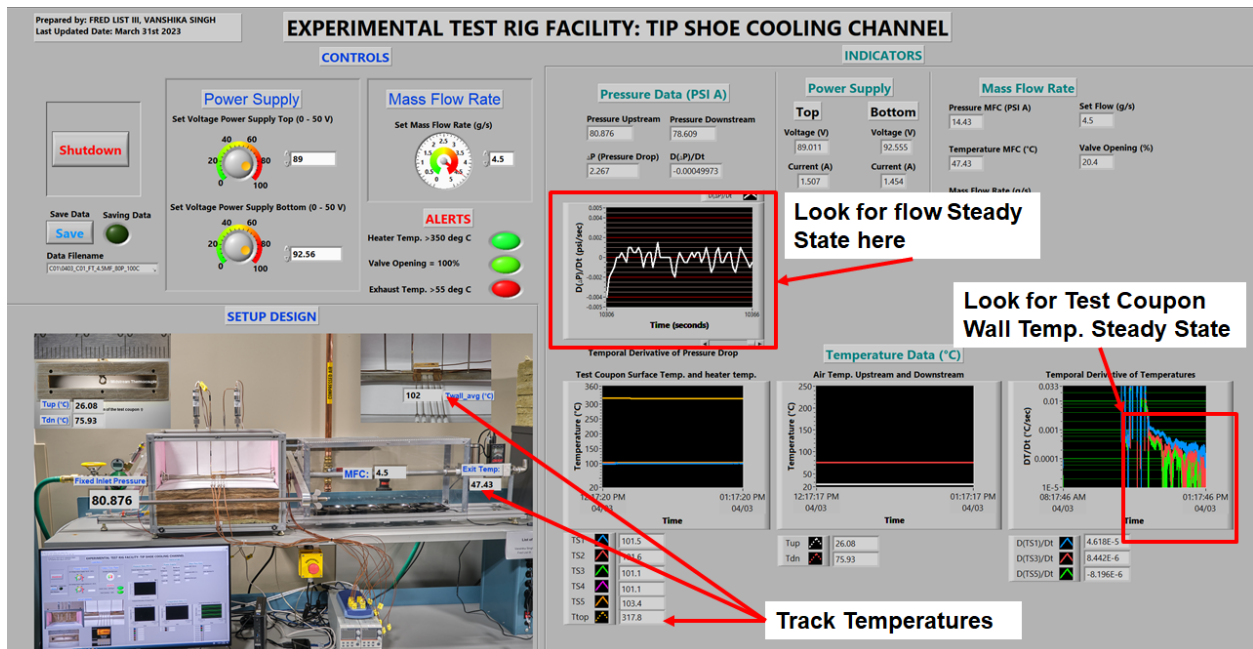


Figure 5.8: Screenshot for LabVIEW for setting mass flow rate and voltage and for reading temperature, pressure, power, and MFC data.

where q''_{conv} is the convective heat flux added in the air per unit area (W/m^2), m is the mass flow rate (kg/s), C_p is the specific heat capacity of air at constant pressure (J/kgK), T_{dn} is the outlet air temperature downstream of the test coupon ($^{\circ}C$), T_{up} is the inlet air temperature upstream of the test coupon ($^{\circ}C$), and A_s is the surface area of the test coupon wall in contact with the ceramic surface heater (m^2).

For the global heat transfer coefficient calculations, equation 5.2 is used:

$$h = \frac{q''_{conv}}{\bar{T}_s - T_b} \quad (5.2)$$

where h is the heat transfer coefficient (W/m^2K), $\bar{T}_s$ is the average wall temperature ($^{\circ}C$), and T_b is the fluid bulk temperature ($^{\circ}C$) calculated as the average of T_{up} and T_{dn} . For the accuracy of the experimental setup, Coupon #1, which is the flat slot machined through EDM, must provide heat transfer augmentation of 1, which means the experimental value should match or be closer to the theoretical value. This was validated and is discussed later in section 5.3.1.

5.2.2 Uncertainty Quantification

Reporting uncertainties involved in measured values and the error propagation to the calculated values is crucial for experimental results. Measurement uncertainties of the sensors are mentioned in Table 5.2 as per the manufacturer's specification sheet. Further, using the sequential perturbation method described by Moffat et al. [197], uncertainties are calculated for heat flux, Nusselt number, and friction factor. Uncertainty involving fluid and solid properties such as density, specific heat, and thermal conductivity are ignored. Uncertainty is calculated using equation 5.3:

$$u_r = \pm \left[\sum_{i=1}^N (\delta R_i)^2 \right]^{0.5} \quad (5.3)$$

where u_r is the uncertainty in the final output, δR_i is the approximated uncertainty contribution from each variable, and N is the number of independent variables. The uncertainty in calculated values is reported in Table 5.2. The uncertainty range involves data from each test coupon. Calculated uncertainty varies with the Reynolds number. The uncertainty was the highest for the lowest Reynolds number and lowest pressure drop; however, it gets better with our proposed design, which ranges from 2% to 13%. At the same time, our Nusselt number has a relatively lower uncertainty of $\sim 2.5\%$ for the range of Reynolds numbers with varying samples.

5.3 Results and Discussion

This section discusses each coupon's heat transfer and fluid flow characteristics in detail. It presents the convective heat transfer coefficients (h) and pressure drop (ΔP) calculated through direct measurements. Further, we calculate the non-dimensional numbers representing h and ΔP separately. This is mainly done for comparison purposes.

5.3.1 Heat Transfer Characteristics

Using equation 5.2, convective heat transfer coefficients were calculated for each coupon and compared. Following this, h is used to calculate the non-dimensional parameter Nusselt number:

Table 5.2: Uncertainty in measured and calculated values

<i>Measured values</i>	<i>Uncertainty</i>
Pressure transducers	$\pm 0.03\%$
Mass flow controller	$\pm(0.8\% \text{ of reading} + 0.2\% \text{ FS})$
Temperature	$\pm 0.5^\circ\text{C}$
<i>Calculated values</i>	<i>Mean uncertainty</i>
Q	2.4% (1.6%–3.1%)
Nu	2.5% (2.0%–3.0%)
f	21% (2%–40%)

$$Nu = \frac{hd_h}{k_f} \quad (5.4)$$

where d_h is the hydraulic diameter (m) at the inlet, and k_f is the thermal conductivity of fluid at bulk fluid temperature (W/mK). The experimental Nu values were further normalized with empirical Nu values for each coupon. Following are the two Nu empirical correlations valid for most of the tested Reynolds number ($Re = \frac{\rho u d_h}{\mu}$) values:

Gnielinski equation:

$$Nu_0 = \frac{\frac{f}{8}(Re - 1000)Pr}{1 + 12.7(\frac{f}{8})^{0.5}(Pr^{0.67} - 1)} \quad (5.5)$$

where f is the friction factor calculated using Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2 \log \left(\frac{\frac{\epsilon}{d_h}}{3.7} + \frac{2.51}{Re\sqrt{f}} \right) \quad (5.6)$$

Dittus Boelter equation:

$$Nu_0 = 0.023 Re^{0.8} Pr^{0.4} \quad (5.7)$$

In the above equations, Pr is the Prandtl number ($\frac{\mu C_p}{k_f}$), ϵ is the surface roughness (m), ρ is the fluid density at inlet absolute pressure and temperature (kg/m^3), μ is the dynamic viscosity ($Pa \cdot s$) at bulk fluid temperature, C_p is the specific heat capacity at constant pressure at bulk fluid temperature $J/(kgK)$, and u is the inlet velocity (m/s). Equation 5.5 is valid for all $Re \in [3000, 5 \times 10^6]$ and $Pr \in [0.5, 2000]$. While equation 5.7 is valid for all $Re \geq 10000$, $Pr \in [0.6, 160]$, and $\frac{L}{d_h} \geq 10$. Here, L is the length of the channel (m).

Both equations were used for comparing the Nusselt number enhancement for EDM, which must be close to one in an ideal scenario because it is EDM-wire machined with no features. Figure 5.9b shows that the empirical Nu is similar to experimental Nu using both equations. Normalized Nu is ~ 1.01 with the Dittus Boelter equation and ~ 1.1 with the Gnielinski equation. This also benchmarks our experimental setup and matches the conclusion of the previous study done for EDM wire-machined geometry of similar dimensions by Weaver et al. [198]. Because both empirical equations provide similar results, the Gnielinski equation is used for future discussions for Nusselt number enhancements.

5.3.1.1 Effect of Micro-Tet Features

Figure 5.9a shows the heat transfer coefficient (h) variation with a change in fluid velocity for each test coupon. Through this, it is observed that for a range of inlet velocity, h for Tet-500 is 3 to 4 times that of EDM. Also, Tet-340, Tet-167, and as-AM seem to follow the trend. As expected, with increasing surface area, the h increases sequentially. Figure 5.9b shows that the Nusselt number increases to 187 for Tet-500, 164 for Tet-340, and 107 for Tet-167 at around 20,000 Reynolds number. This leads to Nu enhancement of more than 4 for Tet-500, ~ 3.44 for Tet-340, and ~ 2.22 for Tet-167 (Figure 5.9c). We also calculated the Nu enhancements with respect to the EDM sample, as shown in Figure 5.9d, which still shows a high heat transfer enhancement of 3.72 for Tet-500.

Nusselt number for our highest performing proposed design, Tet-500, also demonstrates to be $\sim 56\%$ better compared with current state-of-the-art AM rectangular wavy channels for 20,000 Re and $\sim 83\%$ better for 10,000 Re [117, 119]. Also, with triangular AM pin-fins design (best shapes among pin-fins), we outperform Nusselt number by 44% at 15,000 Re

[121]. Additionally, compared with differently shaped AM straight channels, we achieved a 2.5 times better Nusselt number [118].

To explain the possible behavior of flow-inducing high heat transfer augmentation in our proposed design, we take inspiration from the work performed by Han et al. [131]. This is the closest design we found that studied broken ribs (delta-shaped and wedge-shaped geometries) with slight geometrical variation from ours with different spatial arrangements at higher length scales. Based on this work, we can infer that the heat transfer enhancement occurs because of a combination of flow separation and reattachment, as well as because of flow disturbances near the features. Additionally, there might be an impact of vortices occurring on both sides of the delta-shaped ribs. The overall vortex generation and propagation highly disturb the wall boundary layer and greatly enhance the heat transfer coefficients.

5.3.1.2 Effect of AM Roughness

Our as-AM test coupon showed improved Nusselt number enhancement by 27% compared with the EDM sample and 40% improvement compared with the empirical Gnielinski equation (Figures 5.9c and d). We can also observe the same through h values, calculated purely through measured experimental values (Figure 5.9a). In the current study, the EDM sample and as-AM roughness have the same geometry with the same intended dimensions; the enhancements show the clear contribution from AM inherent roughness only with standard process parameters and built vertically, eliminating any roughness variability on different walls. Our calculated heat transfer enhancement for inherent AM roughness might change based on different materials with varying PSD, process parameters, scan strategy, and build direction.

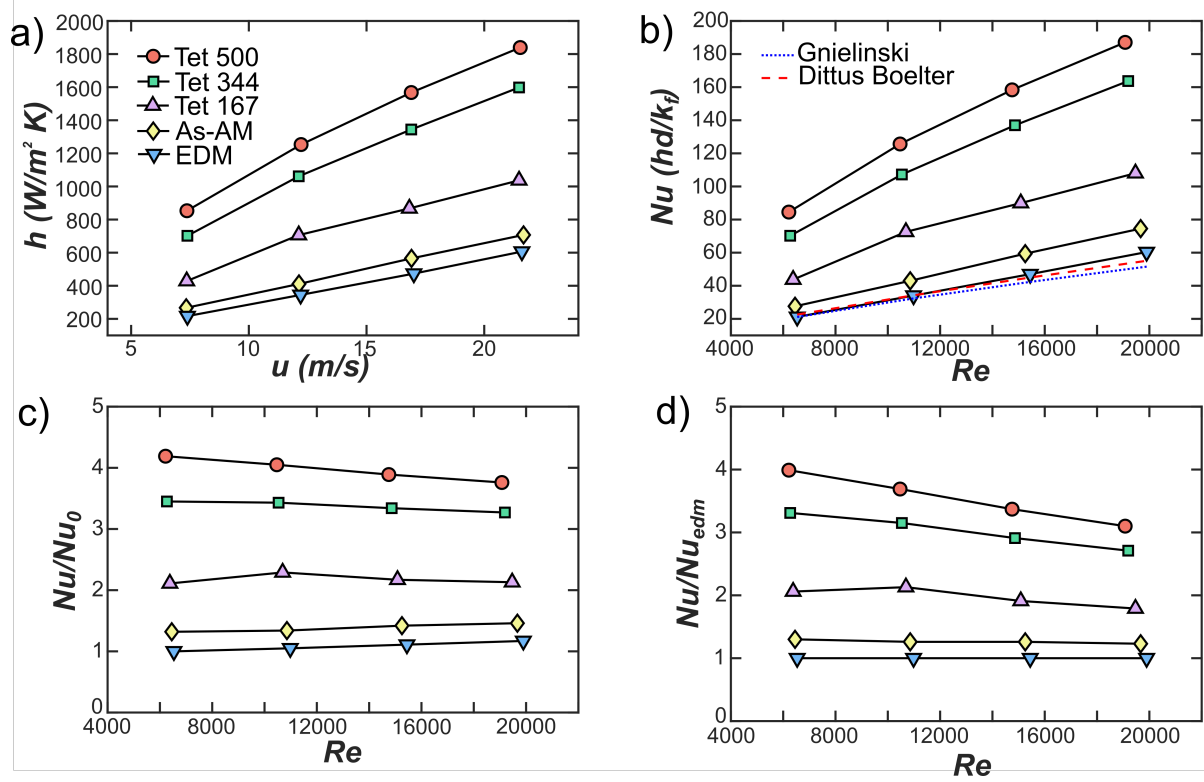


Figure 5.9: a) Heat transfer coefficient vs. inlet velocity, b) heat transfer enhancement vs. Reynolds number, c) heat transfer enhancement (normalized with empirical Gnielinski correlation) vs. Reynolds number, and d) heat transfer enhancement (normalized with EDM sample) vs. Reynolds number.

Stafford et al. [155] also generated near-AM roughness on an aluminum plate through milling. They studied isolating the effect of scaled-AM roughness on heat transfer augmentation for comparison with traditional design features such as ribs and dimples. They concluded that we require additional enhancement mechanisms for internal passages with AM roughness to achieve global thermal performance similar to traditional enhancement features. We concur with the conclusion that additional features along with AM roughness provide significant heat transfer enhancement.

5.3.2 Fluid Flow Characteristics

All heat transfer enhancement features have a pressure drop penalty across the channel. High-pressure drop signifies the high pumping power required to send the cooling fluid to achieve a certain amount of heat transfer. Using calibrated pressure transducers, we measured the inlet and outlet pressure at 50 mm upstream and downstream of the test coupon. Figure 5.10a represents the pressure drop measured for each test coupon. As expected, the pressure drops increase with incremental surface area, inducing high wall shear stresses. Figure 5.10b shows the friction factor variation with Reynolds number. The friction factor was calculated using Darcy friction factor as shown in the following equation:

$$f = 2\Delta P \cdot \frac{d_h}{L\rho u^2} \quad (5.8)$$

where ΔP is the measured pressure drop (Pa), L is the length of the test coupon (m), u is the inlet velocity (m/s), and ρ is the fluid density at inlet absolute pressure and temperature. The friction factor increases with an increase in roughness height features. Figure 5.10c shows the enhancement of friction factor with the smooth channel friction factor (f_0) calculated

using the Colebrook equation 5.6. Our smooth channel, EDM coupon, has approximately six times higher friction than that of the corresponding Colebrook friction factor. We also verified the friction factor enhancement through the ratio of fanning friction factor and Blasius friction factor correlation. This is not presented in the current charts shown in this paper; however, the friction factor enhancement through this correlation also remains the same. Because we are measuring the pressure 50 mm away from the sample, which is comparable to our test coupon length, the actual pressure drops, and consequently, the friction factor is expected to be lower than the calculated f values. Thus, in the current study, we focus on comparing f values with the friction factor of EDM (f_{edm}). The normalization of f with f_{edm} is presented in Figure 5.10d. It can be observed that the friction factor for Tet-500 is ~ 4 times that of EDM, whereas for Tet-340, it is 3 times our smooth channel. Friction factor variation with increased surface area is relatively linear compared with heat transfer enhancements.

5.3.3 Overall Performance

5.3.3.1 Heat transfer coefficient vs Pumping power

It is crucial to understand the overall performance, also known as efficiency index performance, which states the relative increase in heat transfer performance with respect to a unit increase in pumping power requirement [122]. To discuss the overall performance of the proposed design, we state the pumping power required to achieve a certain heat transfer coefficient, as shown in Figure 5.11. Through this, one can observe that for a given pumping power, the heat transfer coefficient can be increased by increasing the surface area.

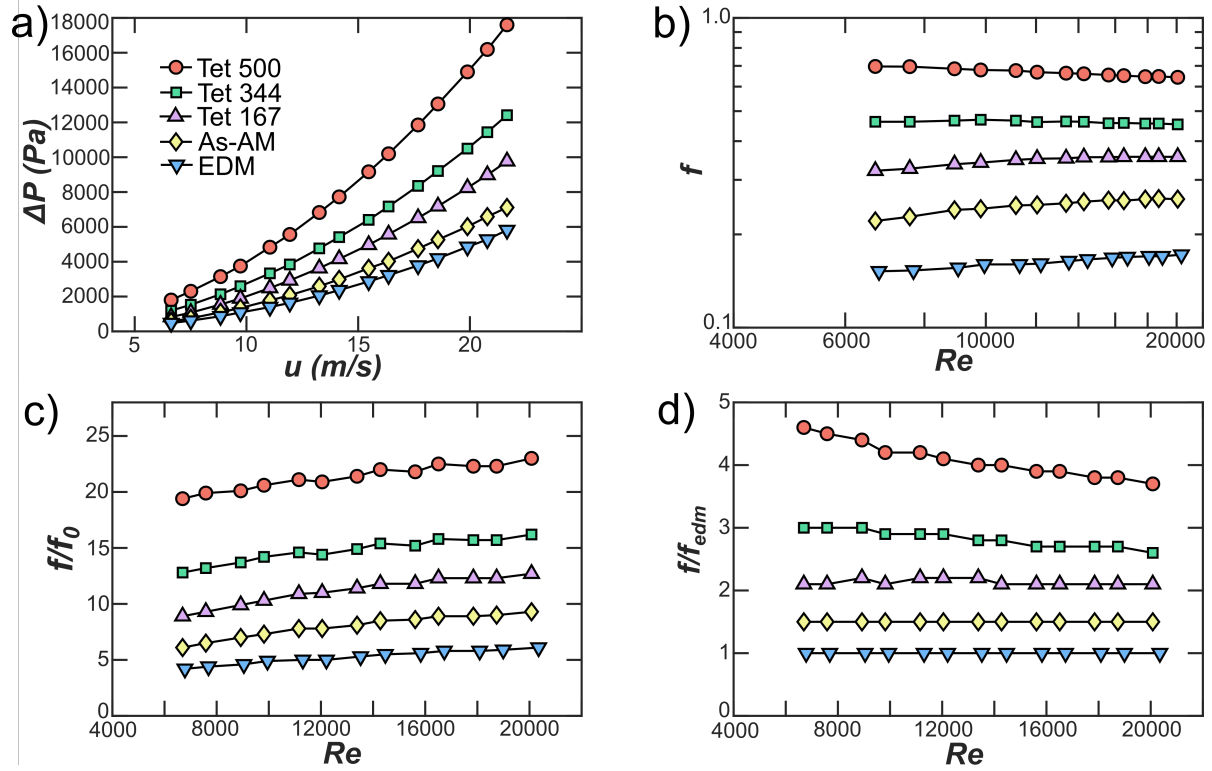


Figure 5.10: a) Experimentally measured pressure drop vs. inlet velocity, b) friction factor vs. Reynolds number, c) friction factor enhancement (normalized with empirical Colebrook equation) vs. Reynolds number, and d) friction factor enhancement (normalized with EDM sample) vs. Reynolds number.

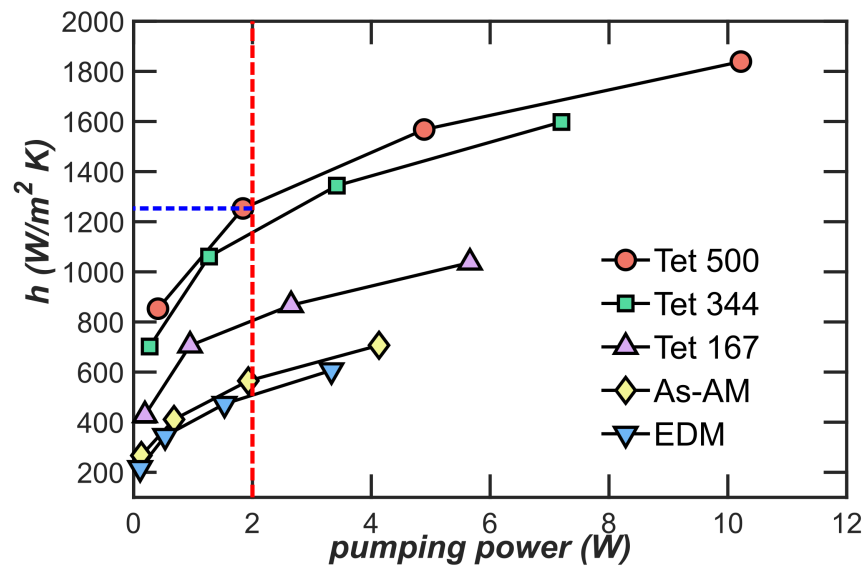


Figure 5.11: Heat transfer coefficient vs pumping power

For instance, if we supply 2 W of pumping power, we can achieve up to 2.5 more heat transfer coefficients by introducing micro-Tet features.

5.3.3.2 Thermal-Hydraulic Performance

Thermal hydraulic performance (THP) is also calculated for tested coupons. We can calculate THP with empirical correlations as well as with our smooth design (i.e., EDM) using the following equations 5.9 and 5.10:

$$THP_0 = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{\frac{1}{3}}} \quad (5.9)$$

$$THP_{edm} = \frac{\frac{Nu}{Nu_{edm}}}{\left(\frac{f}{f_{edm}}\right)^{\frac{1}{3}}} \quad (5.10)$$

Similar to f/f_0 , we expect our THP_0 to provide insufficient information about the actual overall performance of the tested coupon. This is due to the measurement of pressure drop 50 mm away from the component of interest. Therefore, THP_{edm} is more appropriate for understanding the actual overall performance. Figure 5.12 presents the THP_{edm} for proposed designs. The THP_{edm} is approximately 2.2. Comparing the overall thermal performance of proposed designs to AM pin-fins, AM straight microchannels, and AM rectangular channels requires further accurate measurement of THP_0 .

5.3.3.3 Importance of Surface Area in Designing Convection Enhancement Feature

Our initial hypothesis that the multiscale feature provides more area, hence improved heat transfer, matches the experimental behavior well. As discussed previously in section 4.5.1,

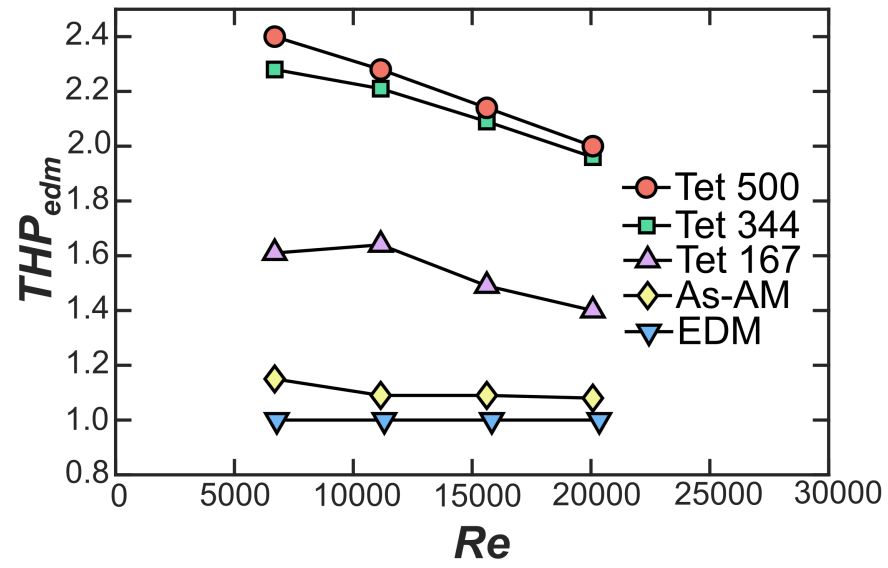


Figure 5.12: Thermal hydraulic performance normalized with our smooth channel i.e., EDM.

per unit area enhancement on a flat coupon due to AM roughness is 12%. Since we added micro-features increasing the surface area, providing more area for AM roughness to persist, the contribution of AM inherent roughness must also increase. However, no linear behavior is observed with increments of wetted surface area and heat transfer coefficients. This might be due to the influence of flow perturbation caused by roughness features inducing fluid mixing. It is also interesting to note in Figure 5.12 that micro-Tet features Tet-340 and Tet-500 seem to provide overall similar performance. The increase in heat transfer coefficient gets compensated with an increase in friction factor. This also infers that beyond a certain increase in surface area, there might not be much benefit in overall performance. However, we need to study the upper limit on the increase in surface area by enhancing the surface texture through the introduction of intentional deterioration of AM roughness.

5.4 Chapter Summary

We designed an experimental setup to study the heat transfer and fluid flow characteristics for the designed convective heat transfer micro-Tet features. The test setup is capable of testing up to $\sim 200^{\circ}\text{C}$; however, in the current study, we characterized heat transfer for all test coupons at $\sim 100^{\circ}\text{C}$. Following are the key outcomes of the study:

1. Approximately 4 times heat transfer enhancement in Tet-500 compared with EDM smooth surface can be achieved by introducing multiscale roughness features for Reynolds number up to 20,000.
2. We see friction factor enhancement of ~ 4 times in Tet-500 compared with EDM smooth surface. Because we measured the pressures away from the actual location, we observed

~ 20 times friction factor enhancement compared with the empirically smooth channel.

Thus, we emphasize focusing on f/f_{edm} for comparison purposes.

3. Heat transfer coefficient vs. pumping power is also calculated; Tet-500 demonstrates 2.5 times enhanced heat transfer for the same input power of 2 W compared with EDM smooth channel.
4. We calculate thermal-hydraulic performance (THP) with respect to our EDM smooth channel, which is calculated as THP_{edm} equal to ~ 2.2 .
5. We also observe that with increasing Reynolds number, Nusselt number enhancement weakly varies; hence, it does not demonstrate the strong function of Reynolds number in the turbulent regime as opposed to trends observed in rectangular wavy channels and pin-fins.
6. Last, we observe the overall thermal performance for Tet-340 and Tet-500 remains approximately the same. This demonstrates that there is no significant benefit beyond a limit of increasing the surface area for heat exchange as it gets compensated by increasing skin friction.

Chapter 6

Holistic View on the Design Strategy

6.1 Potential Solutions for Current Problem Statements

6.1.1 Ash Deposition

In the current research, we introduced an aerodynamic-shaped tube to reduce deposition so that the flue gas streamlines are laminar, reducing drag and particle deposition. Although we successfully demonstrated the macroscopic design change for deposition reduction, the question remains: Can we use the other two classes of solutions? Based on the class of solutions two and three (section 3.3.1) as a potential future direction, we propose to 1) modify surface texture and 2) generate dynamic movement of the tube to shake off the deposition.

Surface Texture Modification: Previous research by Bird et al. [199] and Girard et al. [200] demonstrated that we can reduce contact time and contact area between the liquid particle and solid surface by modifying the surface morphology. For their research, they designed a super-hydrophobic surface with an in-plane discontinuity redirecting the

momentum vertically upward. Though the application described in this research differs from the deposition issue in power plants, the physics behind the particle-to-surface interaction remain the same. So, the next question is, can we study modifying the surface morphology for deposition issues in high-temperature applications? Therefore, for future direction in tube design modification, one can study developing the surface morphology and manufacture it additively so that it assists in reducing the contact time or contact area between the ash particle and tube surface. This study can also be extended to other applications, such as ice accretion on aircraft wings.

External shaking: Among different methods used by nature for cleaning, shaking is most common in mammals, birds, and mosquitoes for removing particles [10]. In engineering, we generate vibration using fluid-structure interaction, which might also assist in deposition reduction. Energy harvester, used to power small electronic devices, is usually operated by flow-induced oscillations. Previous research studied vortex-induced vibration by passive means by introducing features on the tube [201, 202, 203]. Thus, the next question is, can we also modify the tube geometry to induce vibration through fluid-structure means? We suggest introducing the features at circumferentially different locations to introduce vibration analogous to shaking to remove ash deposits. However, we must also ensure that vibration occurs only after a certain ash deposition thickness to avoid continuous shaking.

6.1.2 Internal Cooling Channels

For the second problem of mitigating overheating in gas turbine components, we proposed modifying the surface morphology by introducing micro-Tet features and inherent AM roughness. This creates multiscale surface characteristics, increasing the surface area for heat exchange. Another primary component for enhancing heat exchange through the

class of solutions is integrating compliant movements of the walls discussed in the following paragraphs:

Intermittent cooling channel: Compliant movements of the walls (dilation and contraction) in living creatures allows controlled heat exchange. Developing walls that can alter their shape with change in thermal boundary conditions requires mechanisms capable of amplifying small changes in loading derivative into high strains. Singh et al. [60] suggested using an origami structure with a compliant mechanism to form lifelike structures for intermittent heat exchange. We can initiate with shape memory alloys (SMAs) to realize and test this concept. SMAs provide unique thermo-mechanical properties (superelasticity and shape memory effect) and can restore their original shape from thermally triggered excessive deformation [204]. High-temperature SMAs' progress allows them to operate in a range of 100–300°C [205]; thus, by leveraging SMAs, we cannot use such a concept in high-temperature applications (such as hot sections in gas turbines). However, this is still useful for studying compliant wall effects on heat transfer coefficient and pressure drops.

Flapping of parts: There is interest in studying the effect of flapping on convective heat transfer enhancements. Rips et al. [182] introduced a flag that experiences flow-induced flapping, enhancing the heat transfer mechanism. They studied this using 3D fully coupled fluid-structure-thermal simulations. They found that increasing the flag width significantly increases heat flux; a narrow flag width does not provide sufficient energetic flapping for a meaningful enhanced heat transfer. Ongoing efforts for efficient flapping include exploring and mimicking birds' micro-vascular systems and wing shapes to develop morphing structures [206]. The idea is to introduce SMA composite actuators on the aircraft wings' skin to actuate similarly to bird muscles. Though this work does not focus on the heat transfer aspect of flapping, it demonstrates concepts to achieve efficient flapping.

6.2 Generalized Extension of the Design Strategy

6.2.1 Developing Correlation for a Class of Solution

In the current research, we selected one class of solution among numerous possible pathways and studied that in detail for design improvements. We evaluated one profile function for a teardrop convex-shaped tube and one hydraulic diameter coupon with micro-Tet feature. Our natural next step is to answer this question: 1) How do these solutions generalize for the whole class? Here, generalization means the ability to implement the same solution for differently sized components for the same function for optimal performance. Frequent occurrence of generalization in nature provides the basis for the class of solutions we implemented for our study.

6.2.1.1 Scaling Laws in Nature

The same design feature for differently scaled creatures (ranging from insects to mammals) is implemented successfully in nature. The question here is: 2) How does nature generalize its one design solution for varying ranged organisms? Scaling law is the secret code to such complex-looking yet self-repeating morphologies. The scaling laws express the relationship between any parameter associated with the object with its length scale [207]. If the geometric similarity is maintained with the same scale factor for all its parameters, then it's called *isomorphic* or *isometric* scaling. In contrast, if different scaling factors are used for different parameters within the object (i.e., they do not scale similarly), it is called *allometric* scaling. The general equation representing scaling laws followed by living creatures in nature can be

expressed as [207]:

$$X = X_0 M^\lambda \quad (6.1)$$

where X is an observable and quantifiable parameter such as geometric features like length, radius, and density; X_0 is a normalizing constant; M is the organism's mass; and λ is an exponent. This equation spans a mass range of over 21 orders of magnitude (from a 10^{-13} g small microbe to 10^8 g of large mammals and plants) [207].

Examples of scaling laws include Rubner's law [207, 208], which states that basal metabolic rate (BMR), the rate at which the resting mammal consumes energy, is proportional to heat loss through the body. Heat generation rate (proportional to BMR) is a volume-dependent value, and heat loss is a surface area dependent feature; together, they lead to Rubner's law (i.e., $BMR \propto M_b^{\frac{2}{3}}$, M_b is the animal body mass). However, measurements of body mass and metabolic rate of several species show $BMR \propto M_b^{\frac{3}{4}}$, also known as Kleiber's law [208]. Though there is no convergence on which of these laws is an accurate representation [207, 208], they both demonstrate that the metabolic rate increases slower than the rate at which animal body weight increases.

Similarly, several such correlations have been studied for animal body size, design features, and function. Hair presence is common on various animal bodies, but how do its dimensions vary with differently sized animals? Amador et al. [10] developed scaling laws for the hair dimensions of various creatures. They studied 17 mammals and 10 insects to establish the correlations for surface area (A_{hair}) and animal body mass (M_b), hair thickness (h) and animal body mass, hair length (L) and animal body mass, hair density (η) and animal body mass, and number of hairs (η) and animal body mass. Through best fit from their collected data, they found: $h \sim M_b^{0.081}$, $L \sim M_b^{0.35}$, $\eta \sim M_b^{-0.34}$, $A_{hair} \sim M_b^{0.63}$, and

finally $N \sim M_b^{0.20}$ (see Figure 6.1). The developed correlation for hair thickness by Amador et al. [10] also matches well with the previously analytically found optimal hair thickness relationship for minimum convective heat loss by Bejan (i.e., $h \sim M_b^{0.083}$). Similarly, various scaling laws were also constructed for other functionalities and geometric parameters based on experimental data collected (discussed by Amador et al. [10]). The next question loops back to the first question we posed earlier in this section: 3) Can we also develop a similar correlation for our engineering design solution, forming a generalized solution for the class (family) of solutions?

6.2.1.2 Scaling Correlations for Engineering Design

Nature provides a crucial pointer for furthering the design philosophy for engineering component design (i.e., allometric laws). The typical practice of scaling for engineering design is based on geometric similarity (or isometric scaling). Later, its effect on design output is calculated (such as power, weight, stresses, efficiency, and other related outputs) [209, 210]. However, as in nature, for optimal performance, there might be nonuniformity in the scaling of each design parameter.

To discuss this concept, we expand on our micro-Tet features design for enhanced heat transfer and reduced pressure drop.

1. We chose a surface protrusion-based solution to enhance the heat transfer and reduce the pressure drop (see Figure 6.2a). We produced a point solution for one inlet hydraulic diameter of 2.8 mm and other inputs such as tetrahedron height (h), tetrahedron base triangle length (a), wetted surface area (SA_w), and coupon flow dimension of $23 \times 1.5 \times 50$ mm.

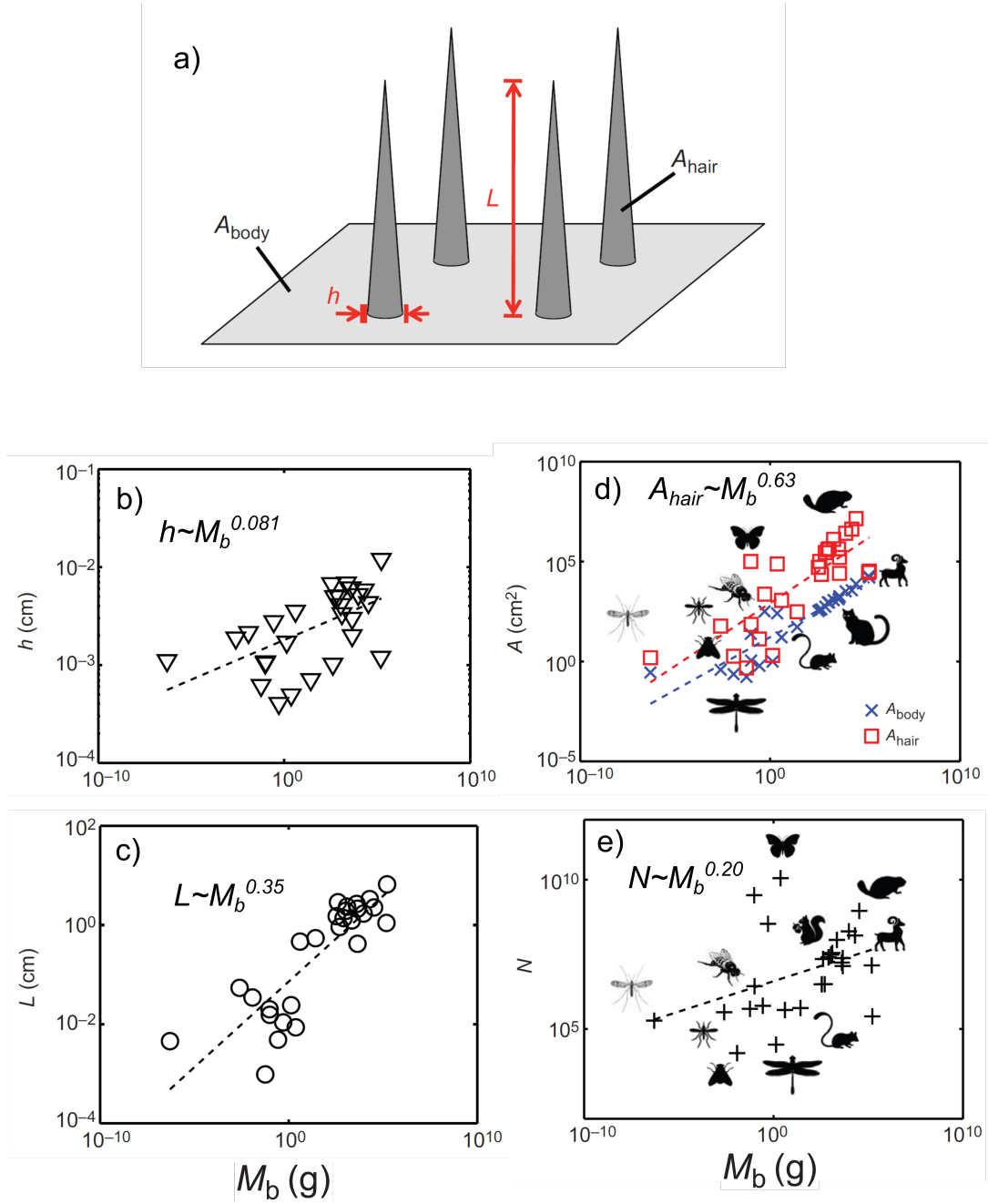


Figure 6.1: Figure reproduced from Amador et al. [10]. a) Schematic of animal skin (body) with hair showing hair basal diameter, h , and length, L . Using 27 mammals and insects, the best-fit relationship among feature dimensions and net surface area with animal body mass was developed. b) Relationship between hair thickness and animal body mass. c) Length and animal body mass relationship. d) Relationship between the area of hair and animal body mass (red boxes), and the relationship between the area of body and animal body mass (blue crosses). e) Relationship between the number of hairs and animal body mass.

2. We validated our proposed design through an experimental approach and calculated the overall THP. One of the conclusions states that there is an upper limit to the increase in surface area. Based on this, further increasing surface area could cause a reduction in the THP because of the increased friction factor (see Figure 6.2b). The blue dotted curve in the figure is a schematic, only for representation purposes.
3. If we scale up all input parameters isometrically, we find that $h \propto SA_w$, whereas $a \propto SA_w^2$. Consequently, $D_h \propto SA_w^2$ and $\frac{h}{D_h} = \text{constant}$. However, for optimal thermal-hydraulic performance (THP) with scaled-up hydraulic diameter, $\frac{h}{D_h}$ might vary under a turbulent regime. Also, with increasing D_h , we may need to increase the SA_w at a different rate lesser than quadratic for optimal THP.
4. Hence, we may have a nonuniform scaling of h and a for optimal THP. Therefore, for optimal thermal-hydraulic performance, similar to nature, we can develop allometric relationships such as $a = C_1 D_h^{\lambda_1}$, $h = C_2 D_h^{\lambda_2}$, and consequently $SA_w = C_3 D_h^{\lambda_2}$ (see Figure 6.2). When we derive allometric laws for design features in nature, we calculate the coefficients based on collected data from numerous species and the best-fit curve. We need to generate optimal data points through rigorous experiments and computational tools to calculate the exponents and proportionality constant for engineering design applications.

6.2.2 Consideration of Heterogeneous Morphology Design

Last, the solutions we find in nature, in the form of morphology (structures) or materials—or their combination—are global. Interestingly, these global solutions often result from local variance throughout the structure. Examples include elephant skin wrinkle variations (Figure

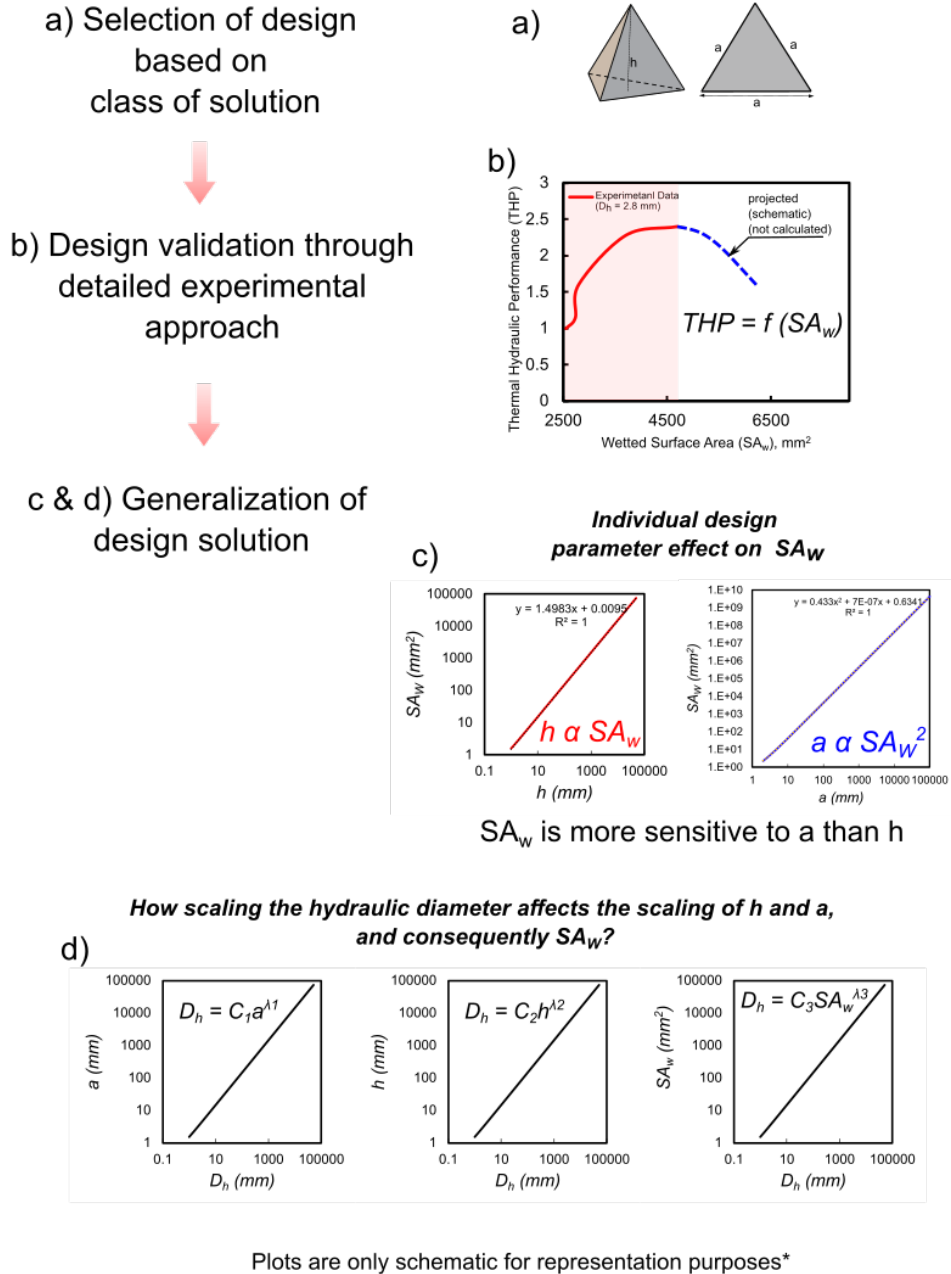


Figure 6.2: Flowchart for the overarching design philosophy: a) Selection of design for a given functionality from the class of solution. b) Validating proposed design using experimental and/or computational methods. Our experimental study showed that there might be an upper limit to an increase in SA_w . There is an optimal SA_w for micro-Tet features. c–d) Similar to scaling laws in nature, generalizing tetrahedron features with an increase in D_h . c) shows the effect of individual design features on SA_w . d) shows possible allometric scaling of h and a with D_h and consequent scaling of SA_w for optimal THP.

6.3a), osteoderm size variations in alligators (Figure 6.3b), termite mounds (Figure 6.3c), paper wasp nests, and many more. Paper wasps, whose nest looks similar globally, contain variability in material deposition to meet local strength requirements. In the absence of a centralized control system, termite mound morphology results from the individual termites modifying the design based on local boundary conditions [211]. Usually, such local geometric variations occur because of the simple rules followed by organisms meeting local design requirements [13, 59].

Can we bring nonuniformity locally based on the local boundary conditions for a given class of solution within a geometry? For example, it is crucial to accurately predict the thermal profile of the turbine blade to avoid thermal local hotspots. It is widely accepted that predicting turbine blade temperature at any local spot off by 30°C (50°F) reduces the blade life by half [114]. One aspect of solving this issue correlates with accurate measuring techniques; however, it also highlights the importance of maintaining the desired thermal profile for longer life. Thus, for the local thermal profile, the size of the micro-Tet features can be modified locally with the help of additive manufacturing. Figure 6.3d demonstrates a CAD drawing of tetrahedrons of varying heights that can be studied to attain desired thermal profiles.



Figure 6.3: a) Elephant wrinkle cell size varies locally. Image taken from [11] b) Osteoderm sizes on alligator skin vary based on the skin location. Image taken from [12] c) Termites demonstrate geometric irregularity and superior functionalities following a set of local rules leading to stochastic architectures locally [13]. d) side view of test coupon cut-section demonstrating variation of tetrahedron heights in two different halves. First half has 500 μm tetrahedrons and the later half contains 350 μm tetrahedrons

Chapter 7

Conclusion

In this research, we proposed and implemented a new design strategy based on nature for advancing next-generation energy systems using additive manufacturing. We studied this through two real-world design problems concerning high-temperature energy systems such as gas turbines and power plant applications. The following are the two design problems:

1. Ash deposition: Undesired ash particles deposit on superheater tube surfaces, inducing localized corrosion. This causes local thinning of the tube and hence early creep failure. Can we produce a design that reduces fouling and improves creep life?
2. Internal cooling channels: Improved rotor inlet temperature of the turbine in gas turbines requires integration of internal cooling channels. For next-generation engines, can we design micro-/mini-channels for components operating in the engine's hot section?

Proposed Design Strategy: We propose to get inspired by the class of solutions (repetitive solutions found in various animate and inanimate structures) for engineering design problems for a given set of functionalities. These solutions can be a function of a) morphology,

b) material-dependent solutions, or c) the inclusion of any compliant movements. We focused on morphological solutions that nature offers owing to AM's current capabilities. We implemented and refined the proposed philosophy using the stated design problems.

Design Solutions: Following are the design solutions implemented for each problem:

1. Superheater tube redesign: We propose introducing an *aerodynamic geometry* (teardrop-shaped and its variants) for reduced drag (mainly pressure drag). This is a macro-level geometry design modification based on the class of solutions nature offers for reduced fouling. Detailed study leads to these findings:
 - (a) Asymmetric-teardrop-shaped tube lowers drag by $\sim 21\%$. Schlieren imaging demonstrates more streamlined fluid around asymmetric-shaped tubes, indicating lesser fluid mixing.
 - (b) Nonuniform presence of ash causes nonuniform circumferential thermal variation in the tube.
 - (c) Circumferential thermal variation and pressure loads might induce localized hoop stresses, aggravating stress-induced corrosion. Asymmetric teardrop-shaped tubes experience $\sim 37\%$ lower tensile stresses at critical locations and $\sim 27.5\%$ lower maximum hoop stresses compared with the baseline design.
 - (d) Asymmetric-teardrop-shaped tubes outperform circular-shaped tubes in terms of overall design metric. This accounts for both drag and maximum hoop stress.
 - (e) Symmetric teardrop-shaped tubes lead to increased drag because of increased skin friction. Hence, they do not perform well compared with circular tubes.

2. Advanced cooling schemes for HPT: We proposed producing *multi-scale surface characteristics* through micro-tetrahedron features with AM roughness for the increased surface area. This design modification is based on the class of solutions for thermoregulation from nature; it generates protrusions-based designs at different length scales of varying sizes and spatial distribution. Design selection was driven by a) heat transfer enhancement and b) reduced pressure drop. Following are the detailed findings of the study:

- (a) We can achieve heat transfer enhancement of ~ 4 times with significantly lesser pressure drop penalty (friction factor enhancement of ~ 4 times) compared with our smooth channel.
- (b) Heat transfer coefficient vs. pumping power calculations show that Tet-500 provides 2.5 times enhanced heat transfer for 2 W of power compared with EDM smooth channel.
- (c) For micro-Tet-based designs, Nusselt number enhancement does not significantly vary with an increase in Reynolds number as opposed to trends observed in rectangular wavy channels and pin-fins.
- (d) We can significantly raise wetted surface area by combining engineered micro-Tet features and AM roughness. Our study shows up to $\sim 200\%$ surface area increase as opposed to no features, which helps achieve heat transfer enhancement.
- (e) We observe that for Tet-344 (SA $\sim 3,700 \text{ mm}^2$) and Tet-500 (SA $\sim 4,700 \text{ mm}^2$), THP is approximately the same. Thus, we can conclude that there is an upper limit to increasing the surface area because the overall performance will drop as

the friction factor enhancements overtake heat transfer enhancement and become a deciding factor.

Recommendation for generalizing the design solution: Following are two additional recommendations for further systematizing the proposed design philosophy, making it a complete approach:

1. In the present study, we studied the design for a given set of parameters for one input geometry. Nature that implements the same solution to various creatures follows allometric scaling laws (i.e., it follows a relationship between the design feature parameters and body size). Similarly, we recommend identifying the nonuniform scaling laws among the parameters of engineering design features with component size for optimal performance output to generalize the solution for differently sized components.
2. Within a class of solutions, nature provides a global solution that results from local design change based on local boundary conditions. This introduces nonuniformity in the designs locally. We recommend locally modifying the design features based on local design requirements.

Bibliography

- [1] S. Hendrixson, “3D Printed Heat Exchanger Illustrates Siemens’ CATCH and Release Approach,” *Additive Manufacturing*, 2023. [Online]. Available: <https://www.additivemanufacturing.media/articles/siemens-advances-additive-manufacturing-with-a-catch-and-release-approach> xv, 4
- [2] “Shell and GE Additive metal 3D print oxygen hydrogen micromixer at Energy Transition Campus Amsterdam,” GE Additive, Cincinnati, OH, USA, Tech. Rep., 11 2022. [Online]. Available: <https://www.ge.com/additive/press-releases/shell-and-ge-additive-metal-3d-print-oxygen-hydrogen-micromixer-energy-transition> xv, 4
- [3] P. Hanaphy, “GE’s Grape-Shaped 3D Printed Heat Exchanger Tested at 900°C,” 2021. [Online]. Available: <https://3dprintingindustry.com/news/ges-grape-shaped-3d-printed-heat-exchanger-tested-at-900c-191942/> xv, 4, 19
- [4] H. Everett, “Fabrisonic Deploys UAM for 3D Printed Heat Exchangers for NASA Spacecraft,” *3D Printing Industry*, 6 2021. [Online]. Available: <https://3dprintingindustry.com/news/fabrisonic-deploys-uam-for-3d-printed-heat-exchangers-for-nasa-spacecraft-191202/> xv, 4

- [5] C. Ledford, P. Fernandez-Zelaia, T. Graening, Q. Campbell, J. O. Rojas, A. M. Rossy, Y. Kato, and M. M. Kirka, “Microstructure and high temperature properties of tungsten processed via electron beam melting additive manufacturing,” *International Journal of Refractory Metals and Hard Materials*, vol. 113, 6 2023. [xv](#), [3](#), [4](#)
- [6] J. Tomeczek, H. Palugniok, and J. Ochman, “Modelling of deposits formation on heating tubes in pulverized coal boilers,” *Fuel*, vol. 83, no. 2, pp. 213–221, 1 2004. [xv](#), [11](#)
- [7] J.-C. Han, S. Dutta, and S. Ekkad, *Gas Turbine Heat Transfer and Cooling Technology*, 2nd ed. Boca Raton, FL, USA: Taylor & Francis Group, 2012. [xv](#), [13](#)
- [8] L. Rzeszutek and G. Kaczmarek, “Cooled Turbine Cantilevered Nozzle Mounting and Sealing System,” *Journal of KONES*, vol. 26, no. 3, pp. 197–202, 9 2019. [xv](#), [15](#)
- [9] S. B. Lattime and B. M. Steinetz, “Turbine Engine Clearance Control Systems: Current Practices and Future Directions,” NASA Glenn Research Center, Cleveland, OH, USA, Tech. Rep., 2003. [Online]. Available: <https://ntrs.nasa.gov/api/citations/20040000621/downloads/20040000621.pdf> [xv](#), [15](#)
- [10] G. J. Amador and D. L. Hu, “Cleanliness is next to godliness: Mechanisms for staying clean,” pp. 3164–3174, 10 2015. [xx](#), [20](#), [29](#), [30](#), [130](#), [133](#), [134](#), [135](#)
- [11] C. Corney, “Why do elephants have such wrinkly skin?” [Online]. Available: <https://www.sciencefocus.com/nature/why-do-elephants-have-such-wrinkly-skin> [xxi](#), [139](#)
- [12] W. Wise, “Alligator Osteoderm Armor,” 6 2022. [Online]. Available: <https://okefenokee.photography/2022/06/24/alligator-osteoderm-armor/> [xxi](#), [139](#)

- [13] K. Liu, R. Sun, and C. Daraio, “Growth rules for irregular architected materials with programmable properties,” *Science*, vol. 377, pp. 975–981, 2022. [xxi](#), [21](#), [138](#), [139](#)
- [14] M. Morey, “U.S. electric-generating capacity for combined-cycle natural gas turbines is growing,” 11 2022. [Online]. Available: <https://www.eia.gov/todayinenergy/detail.php?id=54539# 1>
- [15] “Energy and the environment explained.” [Online]. Available: <https://www.eia.gov/energyexplained/energy-and-the-environment/where-greenhouse-gases-come-from.php 1>
- [16] “Designation: F2792 12a.” [Online]. Available: <http://www.additive3d.com 2>
- [17] M. M. Kirka and P. Fernandez-Zelaia, “Additive Materials for High Temperature Applications,” in *Encyclopedia of Materials: Metals and Alloys*. Elsevier, 9 2021, pp. 529–536. [2](#), [3](#)
- [18] S. S. Babu, N. Raghavan, J. Raplee, S. J. Foster, C. Frederick, M. Haines, R. Dinwiddie, M. K. Kirka, A. Plotkowski, Y. Lee, and R. R. Dehoff, “Additive Manufacturing of Nickel Superalloys: Opportunities for Innovation and Challenges Related to Qualification,” *Metallurgical and Materials Transactions A: Physical Metallurgy and Materials Science*, vol. 49, no. 9, pp. 3764–3780, 9 2018. [3](#)
- [19] A. Plotkowski, J. Ferguson, B. Stump, W. Halsey, V. Paquit, C. Joslin, S. S. Babu, A. Marquez Rossy, M. M. Kirka, and R. R. Dehoff, “A stochastic scan strategy for grain structure control in complex geometries using electron beam powder bed fusion,” *Additive Manufacturing*, vol. 46, 10 2021. [3](#)

- [20] M. Asgaryan, “Prediction of the remaining service life of superheater and reheater tubes in coal-biomass fired power plants,” Ph.D. dissertation, Cranfield University, Cranfield, 2013. [10](#), [26](#), [39](#), [58](#)
- [21] X. Yang, “Development of Ash Deposition Prediction Models through the CFD Methods and the Ash Deposition Indices,” Ph.D. dissertation, University of Sheffield, Sheffield, 2016. [Online]. Available: <https://etheses.whiterose.ac.uk/17127/1/Thesis-x-y-2017.pdf> [10](#)
- [22] F. J. Frandsen, “Utilizing biomass and waste for power production - A decade of contributing to the understanding, interpretation and analysis of deposits and corrosion products,” in *Fuel*, vol. 84, no. 10, 7 2005, pp. 1277–1294. [10](#)
- [23] N. Simms, M. Seraffon, A. Pidcock, and C. Davis, “Assessment of Coating Performance on Waterwalls and Superheaters in a Pulverised Fuel-Fired Power Station,” *Oxidation of Metals*, vol. 88, no. 1-2, pp. 165–177, 8 2017. [10](#)
- [24] E. K. Vakkilainen, “Boiler Mechanical Design,” in *Steam Generation from Biomass*, E. K. Vakkilainen, Ed. Elsevier, 2017, pp. 167–179. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/B9780128043899000071> [10](#)
- [25] S. P. Murray, K. M. Pusch, A. T. Polonsky, C. J. Torbet, G. G. Seward, N. Zhou, S. A. Forsik, P. Nandwana, M. M. Kirka, R. R. Dehoff, W. E. Slye, and T. M. Pollock, “A defect-resistant Co–Ni superalloy for 3D printing,” *Nature Communications*, vol. 11, no. 1, 12 2020. [12](#)

- [26] N. Evbuomwan, S. Sivaloganathan, and A. Jebb, “A Survey of Design Philosophies, Models, Methods and Systems,” *Proceedings of the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture.*, vol. 210, no. 4, pp. 301–320, 1996. [14](#)
- [27] K. K. Fu, M. C. Yang, and K. L. Wood, “Design Principles: The Foundation of Design.” ASME International, 8 2015. [14](#)
- [28] D. E. Brandt and R. R. Wesorick, “GE Gas Turbine Design Philosophy,” Schenectady, NY, Tech. Rep. [Online]. Available: https://www.ge.com/content/dam/gepower-new/global/en_US/downloads/gas-new-site/resources/reference/ger-3434d-ge-gas-turbine-design-philosophy.pdf [16](#)
- [29] M. Mohaghegh, “Evolution of structures design philosophy and criteria,” *Journal of Aircraft*, vol. 42, no. 4, pp. 814–831, 2005. [16](#)
- [30] M. Mani and J. Jee, “Design Rules for Additive Manufacturing: A Categorization,” Tech. Rep. [16](#)
- [31] Y. Tang, S. Yang, and Y. F. Zhao, “Sustainable design for additive manufacturing through functionality integration and part consolidation,” in *Environmental Footprints and Eco-Design of Products and Processes*. Springer, 2016, pp. 101–144. [16](#)
- [32] M. U. Obi, P. Pradel, M. Sinclair, and R. Bibb, “A bibliometric analysis of research in design for additive manufacturing,” pp. 967–987, 5 2022. [16](#)
- [33] A. Blösch-Paidosh and K. Shea, “Design Heuristics for Additive Manufacturing Validated Through a User Study1,” *Journal of Mechanical Design, Transactions of the ASME*, vol. 141, no. 4, 4 2019. [16](#)

- [34] F. Valjak and A. Lindwall, “Review of design heuristics and design principles in design for additive manufacturing,” in *Proceedings of the Design Society*, vol. 1. Cambridge University Press, 2021, pp. 2571–2580. [16](#)
- [35] Z. Fang, “Topology Optimization and Additive Manufacturing for Performance Enhancement of High Temperature and High Pressure Heat Exchangers.” [Online]. Available: <https://arpa-e.energy.gov/technologies/exploratory-topics/topology-optimization> [17](#)
- [36] E. National Academies of Sciences, *Exploiting Advanced Manufacturing Capabilities Topology Optimization*. National Academies Press, 2022. [17](#), [18](#)
- [37] M. Kapsis, L. He, Y. S. Li, O. Valero, R. Wells, S. Krishnababu, G. Gupta, J. Kapat, and M. Schaenzer, “Multiscale Parallelized Computational Fluid Dynamics Modeling Toward Resolving Manufacturable Roughness,” *Journal of Engineering for Gas Turbines and Power*, vol. 142, no. 2, 2 2020. [18](#), [98](#)
- [38] J. F. Vincent, O. A. Bogatyreva, N. R. Bogatyrev, A. Bowyer, and A. K. Pahl, “Biomimetics: Its practice and theory,” pp. 471–482, 8 2006. [18](#)
- [39] K. Fu, D. Moreno, M. Yang, and K. L. Wood, “Bio-inspired design: An overview investigating open questions from the broader field of design-by-analogy,” *Journal of Mechanical Design, Transactions of the ASME*, vol. 136, no. 11, 11 2014. [18](#)
- [40] K. Fu, J. Murphy, M. Yang, K. Otto, D. Jensen, and K. Wood, “Design-by-analogy: experimental evaluation of a functional analogy search methodology for concept generation improvement,” pp. 77–95, 1 2015. [18](#)

- [41] M. W. Glier, J. Tsenn, J. S. Linsey, and D. A. McAdams, “Evaluating the directed intuitive approach for bioinspired design,” *Journal of Mechanical Design, Transactions of the ASME*, vol. 136, no. 7, 2014. 18
- [42] J. Louis Steinbacher, “Bio-inspired Design Workshoop Report_2232327_October 2022,” Tech. Rep., 2022. [Online]. Available: <https://www.pexels.com/photo/flying-fish-1661337/> 18
- [43] J. Kay Nagel, J. K. Nagel, C. Stewart Rose, and J. Madison University Cheryl Lea Beverly, “Enhancing the Pedagogy of Bio-inspired Design in an Engineering Curriculum Enhancing the Pedagogy of Bio-inspired Design in an Engineering Curriculum,” Tech. Rep. 18
- [44] J. K. S. Nagel, C. Rose, C. Beverly, and R. Pidaparti, “Bio-inspired Design Pedagogy in Engineering,” in *Design Education Today*. Springer International Publishing, 2019, pp. 149–178. 18
- [45] J. Emami, M. Tashakori, and Z. Tashakorinia, “Bionic Design in Industrial Design Education at University of Tehran,” Tech. Rep. 18
- [46] L. Ilieva, I. Ursano, L. Traista, B. Hoffmann, and H. Dahy, “Biomimicry as a Sustainable Design Methodology— Introducing the ‘Biomimicry for Sustainability’ Framework,” *Biomimetics*, vol. 7, no. 2, 6 2022. 18
- [47] F. Kiakojouri, V. De Biagi, and L. Abbracciavento, “Design for Robustness: Bio-Inspired Perspectives in Structural Engineering,” *Biomimetics*, vol. 8, no. 1, p. 95, 2 2023. 18

- [48] Z. Li, Y. Jin, J. Du, H. Zhang, and C. Nie, “Physical mechanisms investigation of sharkskin-inspired compressor cascade based on large eddy simulations,” *Journal of Turbomachinery*, vol. 143, no. 6, 6 2021. 19
- [49] J. Pike, “3D printed Purdue heat sink wins competition,” Tech. Rep., 2020. [Online]. Available: <https://engineering.purdue.edu/ME/News/2020/3d-printed-purdue-heat-sink-wins-competition> 19
- [50] H. Hashemi Farzaneh, “Bio-inspired design: the impact of collaboration between engineers and biologists on analogical transfer and ideation,” *Research in Engineering Design*, vol. 31, no. 3, pp. 299–322, 7 2020. 19
- [51] F. Guterl, “That’s Hot: This Lung-Inspired 3D-Printed Part For Cooling CO2 Could Take Power Generation To The Next Level,” General Electric, Tech. Rep., 7 2019. [Online]. Available: <https://www.ge.com/news/reports/thats-hot-this-lung-inspired-3d-printed-part-for-cooling-co2-could-take-power-generation-to-the-next-level> 19, 77
- [52] A. Bejan, “The tree of convective heat streams: its thermal insulation function and the predicted 3/4-power relation between body heat loss and body size,” *International Journal of Heat and Mass Transfer*, vol. 44, pp. 699–704, 2001. [Online]. Available: www.elsevier.com/locate/ijhmt 20, 76, 77
- [53] G. J. Amador, W. Mao, P. DeMercurio, C. Montero, J. Clewis, A. Alexeev, and D. L. Hu, “Eyelashes divert airflow to protect the eye,” *Journal of the Royal Society Interface*, vol. 12, no. 105, 4 2015. 20
- [54] B. Bhushan, Y. C. Jung, and K. Koch, “Micro-, nano- And hierarchical structures for superhydrophobicity, self-cleaning and low adhesion,” *Philosophical Transactions*

- of the Royal Society A: Mathematical, Physical and Engineering Sciences*, vol. 367, no. 1894, pp. 1631–1672, 5 2009. [20](#), [31](#)
- [55] B. Bhushan and Y. C. Jung, “Natural and biomimetic artificial surfaces for superhydrophobicity, self-cleaning, low adhesion, and drag reduction,” pp. 1–108, 1 2011. [20](#)
- [56] U. G. Wegst and M. F. Ashby, “The mechanical efficiency of natural materials,” *Philosophical Magazine*, vol. 84, no. 21, pp. 2167–2186, 7 2004. [21](#)
- [57] U. G. Wegst, H. Bai, E. Saiz, A. P. Tomsia, and R. O. Ritchie, “Bioinspired structural materials,” *Nature Materials*, vol. 14, no. 1, pp. 23–36, 1 2015. [21](#)
- [58] E. Cakmak, N. Sridharan, S. V. Venkatakrishnan, H. Z. Bilheux, L. J. Santodonato, A. Shyam, and S. S. Babu, “Feasibility Study of Making Metallic Hybrid Materials Using Additive Manufacturing,” *Metallurgical and Materials Transactions A: Physical Metallurgy and Materials Science*, vol. 49, no. 10, pp. 5035–5041, 10 2018. [21](#)
- [59] A. Honma and Y. Fukui, “Construction methods of three-dimensional shapes based on the nesting behavior of paper wasps,” *International journal of the Japan Society for Precision Engineering*, vol. 29, no. 4, pp. 361–366, 1995. [21](#), [138](#)
- [60] V. Singh, S. S. Babu, and M. M. Kirka, “Lifelike metallic structures using origami and compliant mechanisms,” *Manufacturing Letters*, vol. 34, pp. 34–37, 10 2022. [22](#), [79](#), [131](#)
- [61] I. G. Wright and S. C. Kung, “Possible scenarios for the causes of accelerated fireside corrosion of superheater tubes in coal-fired boilers,” pp. 316–326, 7 2018. [24](#)

- [62] A. Hendry and D. Lees, “Corrosion of Austenitic Steels in Molten Sulphate Deposits*,” Great Britain, Tech. Rep., 1980. [24](#)
- [63] William T. Reid, *External Corrosion and Deposits*, J. Beer, R. M. Tourin, R. Dolezal, and W. T. Reid, Eds. New York: American Elsevier Publishing Company, Inc, 1971. [24](#)
- [64] U. Kleinhans, C. Wieland, F. J. Frandsen, and H. Spliethoff, “Ash formation and deposition in coal and biomass fired combustion systems: Progress and challenges in the field of ash particle sticking and rebound behavior,” pp. 65–168, 9 2018. [26](#), [28](#), [38](#)
- [65] Y. Zhang, Q. Li, and H. Zhou, “Theoretical Foundation and Basic Properties of Thermal Radiation,” in *Theory and Calculation of Heat Transfer in Furnaces*. Elsevier, 2016, pp. 1–43. [26](#)
- [66] S. V. Vassilev, K. Kitanob, S. Takedab, and T. Tsurueb, “Influence of mineral and chemical composition of coal ashes on their fusibility,” *Fuel Processing Technology*, vol. 45, no. 1, pp. 27–51, 1995. [26](#)
- [67] M. Holubcik, J. Jandacka, and M. Malcho, “Ash Melting Temperature Prediction from Chemical Composition of Biomass Ash,” vol. 5, no. 3, pp. 119–125, 2015. [26](#)
- [68] J. H. Kim, G. B. Kim, and C. H. Jeon, “Prediction of correlation between ash fusion temperature of ASTM and Thermo-Mechanical Analysis,” *Applied Thermal Engineering*, vol. 125, pp. 1291–1299, 2017. [26](#)
- [69] W. Ai and J. M. Kuhlman, “Simulation of coal ash particle deposition experiments,” in *Energy and Fuels*, vol. 25, no. 2, 2 2011, pp. 708–718. [27](#)

- [70] Z. Zheng, W. Yang, Y. Cai, M. Xu, S. K. Boon, and P. Subbaiah, “Numerical investigation on ash deposition on the surface of tube bundle,” in *Energy Procedia*, vol. 143. Elsevier Ltd, 2017, pp. 336–341. [27](#)
- [71] C. Yongtie, Y. Wenming, Z. Zhimin, X. Mingchen, S. K. Boon, and P. Subbaiah, “Modelling of ash deposition in biomass boilers: A review,” in *Energy Procedia*, vol. 143. Elsevier Ltd, 2017, pp. 623–628. [27](#)
- [72] S. Chapela, J. Porteiro, M. A. Gómez, D. Patiño, and J. L. Míguez, “Comprehensive CFD modeling of the ash deposition in a biomass packed bed burner,” *Fuel*, vol. 234, pp. 1099–1122, 12 2018. [27](#)
- [73] C. Riccio, N. J. Simms, and J. E. Oakey, “Deposition prediction in a pilot scale pulverized fuel-fired combustor,” *Fuel*, vol. 253, pp. 1204–1213, 10 2019. [27](#), [28](#), [29](#), [38](#), [45](#)
- [74] L. M. Gibson, L. J. Shadle, and S. V. Pisupati, “Determination of sticking probability based on the critical velocity derived from a visco-elastoplastic model to characterize ash deposition in an entrained flow gasifier,” *Energy and Fuels*, vol. 28, no. 8, pp. 5307–5317, 8 2014. [27](#)
- [75] N. E. L. Haugen and S. Kragset, “Particle impaction on a cylinder in a crossflow as function of Stokes and Reynolds numbers,” *Journal of Fluid Mechanics*, vol. 661, pp. 239–261, 10 2010. [27](#)
- [76] N. Hare, M. Rasul, and S. Moazzem, “A review on boiler deposition/fouling prevention and removal techniques for power plant,” in *EE’10: Proceedings of the 5th IASME/WSEAS international conference on Energy & environment*. World

- Scientific and Engineering Academy and Society (WSEAS), 2 2010. [Online]. Available: <https://www.researchgate.net/publication/229034998> 27
- [77] A. Zbogar, F. Frandsen, P. A. Jensen, and P. Glarborg, “Shedding of ash deposits,” pp. 31–56, 2 2009. 27, 32
- [78] D. Bouris, G. Papadakis, and G. Bergeles, “Numerical evaluation of alternate tube configurations for particle deposition rate reduction in heat exchanger tube bundles,” Tech. Rep. [Online]. Available: www.elsevier.com/locate/ijh 27
- [79] D. Bouris, E. Konstantinidis, S. Balabani, D. Castiglia, and G. Bergeles, “Design of a novel, intensified heat exchanger for reduced fouling rates,” *International Journal of Heat and Mass Transfer*, vol. 48, no. 18, pp. 3817–3832, 8 2005. 27
- [80] S. Z. Tang, M. J. Li, F. L. Wang, and Z. B. Liu, “Fouling and thermal-hydraulic characteristics of aligned elliptical tube and honeycomb circular tube in flue gas heat exchangers,” *Fuel*, vol. 251, pp. 316–327, 9 2019. 27
- [81] J. Li, W. Du, and L. Cheng, “Numerical simulation and experiment of gas-solid two phase flow and ash deposition on a novel heat transfer surface,” *Applied Thermal Engineering*, vol. 113, pp. 1033–1046, 2 2017. 27
- [82] L. L. Baxter, “Ash Deposition during Biomass and Coal Combustion: A Mechanistic Approach,” *Biomass and Bioenergy*, vol. 4, no. 2, pp. 85–102, 1993. 28
- [83] G. D. Bixler and B. Bhushan, “Review article: Biofouling: Lessons from nature,” *Philosophical Transactions of the Royal Society A: Mathematical, Physical and Engineering Sciences*, vol. 370, no. 1967, pp. 2381–2417, 5 2012. 29, 30, 31

- [84] B. Dean and B. Bhushan, “Shark-skin surfaces for fluid-drag reduction in turbulent flow: A review,” *Philosophical Transactions of the Royal Society A: Mathematical, Physical and Engineering Sciences*, vol. 368, no. 1929, pp. 4775–4806, 10 2010. [31](#)
- [85] A. G. Domel, M. Saadat, J. C. Weaver, H. Haj-Hariri, K. Bertoldi, and G. V. Lauder, “Shark skin-inspired designs that improve aerodynamic performance,” *Journal of the Royal Society Interface*, vol. 15, no. 139, 2018. [31](#)
- [86] E. Barrette-Vanasse, A.-F. Laurin, R. Trana, and J. Xiao, “The Hydrodynamic Advantages of Shark Scales,” 5 2023. [Online]. Available: <https://bioengineering.dreamhosters.com/the-hydrodynamic-advantages-of-shark-scales/> [31](#)
- [87] S. Kaneko, T. Nakamura, F. Inada, and M. Kato, *Flow-Induced Vibrations: Classifications and Lessons from Practical Experiences*, first edition ed. Oxford: Elsevier, 2008. [32](#)
- [88] M. A. R. Koehl, “When Does Morphology Matter?” Tech. Rep., 1996. [Online]. Available: <https://about.jstor.org/terms> [32](#)
- [89] K. Harada and T. Yamakawa, “US 8,490,452 B2,” 7 2013. [33](#)
- [90] L. Zhang, S. Xie, Z. Liang, J. Zhang, Y. Wang, W. Chen, and C. Kong, “Numerical investigation of flow and heat transfer in enhanced tube with slot dimples,” *Heat and Mass Transfer/Waerme- und Stoffuebertragung*, vol. 55, no. 12, pp. 3697–3709, 12 2019. [35](#)
- [91] Y. Chudnovsky and A. Kozlov, “Development and Field Trial of Dimpled-Tube Technology for Chemical Industry Process Heaters,” *Gas Technology*

- Institute, Des Plaines, IL, Tech. Rep., 9 2006. [Online]. Available: <https://www.osti.gov/servlets/purl/894062> 35
- [92] C. Kamath, B. El-Dasher, G. F. Gallegos, W. E. King, A. Sisto, C. Kamath, B. El-Dasher, G. F. Gallegos, W. E. King, and A. Sisto, “Density of Additively-Manufactured, 316L SS Parts Using Laser Powder-Bed Fusion at Powers Up to 400W,” Lawrence Liver more National Laboratory, Livermore, CA, Tech. Rep., 12 2013. [Online]. Available: <https://www.osti.gov/servlets/purl/1116929> 35
- [93] Li and Meimei, “Progress Report on the Assessment of the Material Performance for TCR Applications Nuclear Science and Engineering Division,” Argonne National Laboratory, Tech. Rep., 9 2020. [Online]. Available: www.anl.gov. 35, 58, 59
- [94] Y. M. Wang, T. Voisin, J. T. McKeown, J. Ye, N. P. Calta, Z. Li, Z. Zeng, Y. Zhang, W. Chen, T. T. Roehling, R. T. Ott, M. K. Santala, P. J. Depond, M. J. Matthews, A. V. Hamza, and T. Zhu, “Additively manufactured hierarchical stainless steels with high strength and ductility,” *Nature Materials*, vol. 17, no. 1, pp. 63–70, 1 2018. 35
- [95] V. P. Nguyen, S. Lee, W. Choi, and S. T. Choi, “Effect of creep lifetime on geometric optimization of boiler tubes for thermal power plants,” *Materials at High Temperatures*, vol. 36, no. 5, pp. 379–387, 9 2019. 36
- [96] A. Zbogor, F. J. Frandsen, P. A. Jensen, and P. Glarborg, “Heat transfer in ash deposits: A modelling tool-box,” pp. 371–421, 2005. 38, 39, 40, 41
- [97] H. Zhou, P. A. Jensen, and F. J. Frandsen, “Dynamic mechanistic model of superheater deposit growth and shedding in a biomass fired grate boiler,” *Fuel*, vol. 86, no. 10-11, pp. 1519–1533, 7 2007. 38, 39

- [98] M. Trojan and J. Taler, “Effect of scale deposits on the internal surfaces of the tubes on the superheater operation,” *Archives of Thermodynamics*, vol. 34, no. 4, pp. 73–91, 2013. [38](#)
- [99] S. Sanitjai and R. J. Goldstein, “Forced convection heat transfer from a circular cylinder in crossflow to air and liquids,” *International Journal of Heat and Mass Transfer*, vol. 47, no. 22, pp. 4795–4805, 10 2004. [39](#)
- [100] Y. A. Cengel, *Heat and Mass Transfer: A practical approach*, 3rd ed. New York: The McGraw Hill Companies, Inc., 2007. [39](#)
- [101] A. Mazumdar, “Principles and Techniques of Schlieren Imaging Systems,” Coulmbia University, New York, Tech. Rep., 6 2011. [47](#)
- [102] B. Schmidt, “Schlieren Visualization Ae 104b,” Tech. Rep., 2015. [Online]. Available: https://shepherd.caltech.edu/T5/Ae104/Ae104b_handout2015.pdf [49](#)
- [103] W. Thielicke and R. Sonntag, “Particle Image Velocimetry for MATLAB: Accuracy and enhanced algorithms in PIVlab,” *Journal of Open Research Software*, vol. 9, pp. 1–14, 2021. [49](#)
- [104] A. W. Gena, C. Voelker, and G. S. Settles, “Qualitative and quantitative schlieren optical measurement of the human thermal plume,” *Indoor Air*, vol. 30, no. 4, pp. 757–766, 7 2020. [49](#)
- [105] George Y. Lai, *High Temperature Corrosion and Material Applications*. Ohio: ASM International, 11 2007. [51](#)

- [106] S. D. Kapayeva, M. J. Bergander, A. Vakhguelt, and S. I. Khairaliyev, “Remaining life assessment for boiler tubes affected by combined effect of wall thinning and overheating,” *Journal of Vibroengineering*, vol. 19, no. 8, pp. 5892–5907, 12 2017. 58
- [107] R. Schafrik and R. Sprague, “Superalloy Technology - A Perspective on Critical Innovations for Turbine Engines,” *Key Engineering Materials*, vol. 380, pp. 113–134, 3 2008. 68
- [108] S. Tin and T. M. Pollock, *Turbine Aerodynamics, Heat Transfer, Materials, and Mechanics*, T. I.-P. Shih and V. Yang, Eds. Reston, VA: American Institute of Aeronautics and Astronautics, Inc, 2014, vol. 243. 68
- [109] J. Han and L. M. Wright, “Enhanced Internal Cooling of Turbine Blades and Vanes,” College Station, Texas, Tech. Rep., 2006. [Online]. Available: <https://netl.doe.gov/sites/default/files/gas-turbine-handbook/4-2-2-2.pdf> 69
- [110] F. J. Cunha, “Heat Transfer Analysis,” Pratt & Whitney, United Technologies Corporation, Avon, CT, Tech. Rep., 2006. [Online]. Available: <https://netl.doe.gov/sites/default/files/gas-turbine-handbook/4-4.pdf> 69
- [111] J. C. Han, “Advanced Cooling in Gas Turbines 2016 Max Jakob Memorial Award Paper,” 11 2018. 69
- [112] C. K. Stimpson, “Cooling Performance of Additively Manufactured Microchannels and Film Cooling Holes, A Dissertation in Mechanical Engineering,” Tech. Rep., 2017. 69, 97

- [113] L. M. Wright and J.-C. Han, “Heat Transfer Enhancement for Turbine Blade Internal Cooling,” *Journal of Enhanced Heat Transfer*, vol. 21, no. 3, pp. 111–140, 2014. [69](#)
- [114] J. C. Han, “Fundamental gas turbine heat transfer,” *Journal of Thermal Science and Engineering Applications*, vol. 5, no. 2, 5 2013. [69](#), [138](#)
- [115] S. V. Ekkad, *Turbine Aerodynamics, Heat Transfer, Materials, and Mechanics*, T. I.-P. Shih and V. Yang, Eds. REXON, VA: American Institute of Aeronautics and Astronautics, Inc., 2014, vol. 243. [69](#)
- [116] A. J. Wildgoose and K. A. Thole, “Heat Transfer and Pressure Loss of Additively Manufactured Internal Cooling Channels With Various Shapes,” *ASME. J. Turbomach.*, vol. 145, no. 7, 7 2023. [74](#)
- [117] K. L. Kirsch and K. A. Thole, “Heat transfer and pressure loss measurements in additively manufactured wavy microchannels,” *Journal of Turbomachinery*, vol. 139, no. 1, 1 2017. [74](#), [118](#)
- [118] K. A. Thole, S. P. Lynch, and A. J. Wildgoose, “Review of advances in convective heat transfer developed through additive manufacturing,” in *Advances in Heat Transfer*. Academic Press, 1 2021, vol. 53, pp. 249–325. [74](#), [75](#), [97](#), [98](#), [119](#)
- [119] T. M. Corbett, K. A. Thole, and S. Bollapragada, “Amplitude and Wavelength Effects for Wavy Channels,” *Journal of Turbomachinery*, vol. 145, no. 3, 3 2023. [74](#), [103](#), [118](#)
- [120] K. L. Kirsch and K. A. Thole, “Pressure loss and heat transfer performance for additively and conventionally manufactured pin fin arrays,” *International Journal of Heat and Mass Transfer*, vol. 108, pp. 2502–2513, 2017. [74](#)

- [121] K. K. Ferster, K. L. Kirsch, and K. A. Thole, “Effects of geometry, spacing, and number of pin fins in additively manufactured microchannel pin fin arrays,” *Journal of Turbomachinery*, vol. 140, no. 1, 1 2018. [74](#), [98](#), [119](#)
- [122] T. M. Corbett, K. A. Thole, and S. Bollapragada, “Impacts of Pin Fin Shape and Spacing on Heat Transfer and Pressure Losses.” *ASME. J. Turbomach.*, vol. 145, no. 5, 5 2023. [74](#), [103](#), [122](#)
- [123] M. A. Hossain, A. Ameri, J. W. Gregory, and J. P. Bons, “Experimental investigation of innovative cooling schemes on an additively manufactured engine scale turbine nozzle guide vane,” *Journal of Turbomachinery*, vol. 143, no. 5, 5 2021. [74](#)
- [124] R. Krewinkel, A. Such, A. O. d. l. Torre, A. Wiedermann, D. Castillo, S. A. Rodriguez, J. H. Schleifenbaum, and M. Blaswich, “Design and Characterization of Additively Manufactured NGVs Operated in a Small Industrial Gas Turbine,” *International Journal of Gas Turbine, Propulsion and Power Systems*, vol. 11, pp. 36–44, 2020. [74](#)
- [125] R. L. Webb, E. R. G. Eckert, R. J. Goldstein, and W. Dbq, “Heat transfer and friction in tubes with repeated-rib roughness,” *International Journal of Heat and Mass Transfer*, vol. 14, no. 4, pp. 601–617, 1971. [71](#)
- [126] J. C. Han, L. R. Glicksman, and W. M. Rohsenow, “An investigation of heat transfer and friction for rib-roughened surfaces,” *International Journal of Heat and Mass Transfer*, vol. 21, pp. 1143–1156, 1978. [71](#)

- [127] J. Han, J. Park, and C. Lei, “Heat Transfer Enhancement in Channels With Turbulence Promoters,” *Journal of Engineering for Gas Turbines and Power*, vol. 107, no. 3, pp. 628–635, 7 1985. [71](#)
- [128] J. Han, J. J. Huang, and C.-P. Lee, “Augmented Heat Transfer in Square Channels With Parallel, Crossed, and V-Shaped Angled Ribs,” *Journal of Heat and Mass Transfer*, vol. 113, no. 3, pp. 590–596, 8 1991. [71](#)
- [129] J. S. Park, J. C. Han, Y. Huang, and R. J. Boyle, “Heat transfer performance comparisons of five different rectangular channels with parallel angled ribs,” Tech. Rep., 1992. [71](#)
- [130] J. C. Han and Y. M. Zhang, “High performance heat transfer ducts with parallel broken and V-shaped broken ribs,” Tech. Rep. 2, 1992. [71](#)
- [131] J. Han, J. J. Huang, and C.-P. Lee, “Augmented Heat Transfer in Square Channels with Wedge-Shaped and Delta-Shaped Turbulence Promoters,” *Journal of Enhanced Heat Transfer*, vol. 1, no. 1, pp. 37–52, 1994. [71](#), [119](#)
- [132] A. P. Rallabandi, N. Alkhamis, and J. C. Han, “Heat transfer and pressure drop measurements for a square channel with 45 deg round-edged ribs at high reynolds numbers,” *Journal of Turbomachinery*, vol. 133, no. 3, 2011. [71](#)
- [133] M. K. Chyu, Y. Yu, and H. Ding, “Heat Transfer Enhancement in Rectangular Channels with Concavities,” *Journal of Enhanced Heat Transfer*, vol. 6, no. 6, p. 429439, 1999. [71](#)

- [134] M. Chyu, Y. Yu, H. Ding, J. Downs, and F. Soechting, “Concavity Enhanced Heat Transfer in an Internal Cooling Passage,” in *International Gas Turbine & Aeroengine Congress & Exhibition*, 1997. [71](#)
- [135] H. K. Moon, T. O’connell, and B. Glezer, “Channel Height Effect on Heat Transfer and Friction in a Dimpled Passage,” Tech. Rep., 2000. [Online]. Available: http://asmedigitalcollection.asme.org/gasturbinespower/article-pdf/122/2/307/5636401/307_1.pdf [71](#)
- [136] I. Borisov, A. Khalatov, S. Kobzar, and B. Glezer, “Comparison of Thermo-Hydraulic Characteristics for Two Types of Dimpled Surfaces,” in *Proceedings of the ASME Turbo Expo 2004: Power for Land, Sea, and Air. Volume 3: Turbo Expo 2004.*, Vienna, Austria, 6 2004, pp. 933–942. [72](#)
- [137] A. Khalatov and V. Onishchenko, “Heat Transfer and Surface Friction Downstream of a Dual Array of Dimples of a Different Shape,” in *Proceedings of the ASME Turbo Expo 2008: Power for Land, Sea, and Air. Volume 4: Heat Transfer, Parts A and B.*, Berlin, Germany, 6 2008, pp. 1–13. [72](#)
- [138] S. Acharya and F. Zhou, “Experimental and Computational Study of Heat/Mass Transfer and Flow Structure for Four Dimple Shapes in a Square Internal Passage,” *Journal of Turbomachinery*, vol. 134, no. 6, 9 2012. [72](#)
- [139] S. W. Chang, T. M. Liou, K. F. Chiang, and G. F. Hong, “Heat transfer and pressure drop in rectangular channel with compound roughness of V-shaped ribs and deepened scales,” *International Journal of Heat and Mass Transfer*, vol. 51, no. 3-4, pp. 457–468, 2 2008. [72](#)

- [140] T. Liou, S. Chang, J. Chen, and C. Chan, “Fluid Flow Inside a Rectangular Duct With Two Opposite Walls Roughened by Deepened Scales,” in *Proceedings of the ASME Turbo Expo 2009: Power for Land, Sea, and Air. Volume 3: Heat Transfer, Parts A and B*, Orlando, Florida, USA, 6 2009, pp. 161–170. [72](#)
- [141] C. Neil Jordan and L. M. Wright, “Heat Transfer Enhancement in a Rectangular (AR = 3:1) Channel With V-Shaped Dimples,” *Journal of Turbomachinery*, vol. 135, no. 1, 10 2012. [72](#)
- [142] D. E. Metzger and M. Asme, “Heat Transfer Experiments and Flow Visualization for Arrays of Short Pin Fins,” Tech. Rep., 1982. [Online]. Available: <http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1982/79597/V004T09A007/2394228/v004t09a007-82-gt-138.pdf> [72](#)
- [143] G. VanFossen, “Heat-Transfer Coefficients for Staggered Arrays of Short Pin Fins,” *Journal of Engineering for Gas Turbines and Power*, vol. 104, pp. 268–274, 1982. [72](#)
- [144] B. Brigham and G. VanFossen, “Length to Diameter Ratio and Row Number Effects in Short Pin Fin Heat Transfer ,” *Journal of Engineering of Gas Turbines and Power*, vol. 106, pp. 241–244, 1984. [72](#)
- [145] J. Armstrong and Winstanley, “A Review of Staggered Array Pin Fin Heat Transfer for Turbine Cooling Applications ,” *ASME. J. Turbomach.*, vol. 110, no. 1, pp. 94–103, 1 1988. [72](#)
- [146] M. Chyu, C. Yen, and S. Siw, “Comparison of Heat Transfer From Staggered Pin Fin Arrays With Circular, Cubic and Diamond Shaped Elements,” in *Proceedings of the*

- ASME Turbo Expo 2007: Power for Land, Sea, and Air. Volume 4: Turbo Expo 2007, Parts A and B.*, Montreal, Canada, 5 2007, pp. 991–999. [73](#)
- [147] S. Siw, M. Chyu, and M. Alvin, “Heat Transfer Enhancement of Internal Cooling Passage With Triangular and Semi-Circular Shaped Pin-Fin Arrays,” in *Proceedings of the ASME Turbo Expo 2012: Turbine Technical Conference and Exposition. Volume 4: Heat Transfer, Parts A and B*, Copenhagen, Denmark, 6 2012, pp. 493–503. [73](#)
- [148] M. Moon and K. Kim, “Heat Transfer Performance of a New Fan-Shaped Pin-Fin in Internal Cooling Channel,” in *Proceedings of the ASME Turbo Expo 2013: Turbine Technical Conference and Exposition. Volume 3A: Heat Transfer*, San Antonio, Texas, USA, 6 2013. [73](#)
- [149] L. M. Wright, A. F. Chen, H. W. Wu, J. C. Han, C. P. Lee, S. Azad, and J. Um, “Heat transfer enhancement in a rectangular cooling channel with airfoil shaped fins,” *Journal of Thermal Science and Engineering Applications*, vol. 13, no. 4, 8 2021. [73](#)
- [150] I. Kaur and P. Singh, “Critical evaluation of additively manufactured metal lattices for viability in advanced heat exchangers,” 4 2021. [74](#)
- [151] S. Parbat, Z. Min, L. Yang, and M. Chyu, “Experimental and numerical analysis of additively manufactured inconel 718 coupons with lattice structure,” *Journal of Turbomachinery*, vol. 142, no. 6, 6 2020. [75](#)
- [152] C. K. Stimpson, J. C. Snyder, K. A. Thole, and D. Mongillo, “Scaling roughness effects on pressure loss and heat transfer of additively manufactured channels,” *Journal of Turbomachinery*, vol. 139, no. 2, 2 2017. [75](#)

- [153] —, “Roughness effects on flow and heat transfer for additively manufactured channels,” *Journal of Turbomachinery*, vol. 138, no. 5, 5 2016. [75](#), [80](#), [98](#)
- [154] J. C. Snyder and K. A. Thole, “Tailoring Surface Roughness Using Additive Manufacturing to Improve Internal Cooling,” *Journal of Turbomachinery*, vol. 142, no. 7, 7 2020. [75](#), [98](#)
- [155] G. J. Stafford, S. T. McClain, D. R. Hanson, R. F. Kunz, and K. A. Thole, “Convection in Scaled Turbine Internal Cooling Passages With Additive Manufacturing Roughness,” *Journal of Turbomachinery*, vol. 144, no. 4, 4 2022. [75](#), [121](#)
- [156] L. Badarnah and J. E. Fernandez, “Morphological configurations inspired by nature for thermal insulation materials,” Tech. Rep., 2015. [75](#), [76](#), [77](#)
- [157] L. Badarnah, Y. Nachman Farchi, U. Knaack, and U. Knaack, “Solutions from nature for building envelope thermoregulation.” WIT Press, 6 2010, pp. 251–262. [75](#), [77](#), [78](#)
- [158] D. Mota-Rojas, C. G. Titto, A. d. M. Geraldo, J. Martínez-Burnes, J. Gómez, I. Hernández-ávalos, A. Casas, A. Domínguez, N. José, A. Bertoni, B. Reyes, and A. M. Pereira, “Efficacy and function of feathers, hair, and glabrous skin in the thermoregulation strategies of domestic animals,” 12 2021. [75](#), [78](#)
- [159] E. A. Tansey, C. D. Johnson, and C. D. Johnson, “Staying Current Recent advances in thermoregulation,” *Adv Physiol Educ*, vol. 39, pp. 139–148, 2015. [Online]. Available: <http://advan.physiology.org> [75](#), [78](#)
- [160] D. J. McCafferty, G. Pandraud, J. Gilles, M. Fabra-Puchol, and P. Y. Henry, “Animal thermoregulation: A review of insulation, physiology and behaviour relevant to

- temperature control in buildings,” *Bioinspiration and Biomimetics*, vol. 13, no. 1, 1 2018. [75](#), [76](#), [77](#), [78](#)
- [161] C. L. Tan and Z. A. Knight, “Regulation of Body Temperature by the Nervous System,” pp. 31–48, 4 2018. [75](#), [78](#)
- [162] Z. Zhu, J. Li, H. Peng, and D. Liu, “Nature-inspired structures applied in heat transfer enhancement and drag reduction,” *Micromachines*, vol. 12, no. 6, 6 2021. [75](#)
- [163] D. Jing, S. Song, Y. Pan, and X. Wang, “Optimal fractal tree-like microchannel networks with slip for laminar-flow-modified Murray’s law,” *Beilstein Journal of Nanotechnology*, vol. 9, no. 1, pp. 482–489, 2 2018. [76](#)
- [164] J. Kou, Y. Chen, X. Zhou, H. Lu, F. Wu, and J. Fan, “Optimal structure of tree-like branching networks for fluid flow,” *Physica A: Statistical Mechanics and its Applications*, vol. 393, pp. 527–534, 1 2014. [76](#)
- [165] D. Zhuang, Y. Yang, G. Ding, X. Du, and Z. Hu, “Optimization of Microchannel Heat Sink with Rhombus Fractal-like Units for Electronic Chip Cooling,” *International Journal of Refrigeration*, vol. 116, pp. 108–118, 8 2020. [76](#)
- [166] A. Y. Alharbi, D. V. Pence, and R. N. Cullion, “Thermal characteristics of microscale fractal-like branching channels,” *Journal of Heat Transfer*, vol. 126, no. 5, pp. 744–752, 10 2004. [76](#)
- [167] L. Knudsen and M. Ochs, “The micromechanics of lung alveoli: structure and function of surfactant and tissue components,” pp. 661–676, 12 2018. [76](#)

- [168] B. Ahmadi, J. Cesarano, K. Nawaz, N. Ninos, and S. Bigham, “A high-performance lung-inspired ceramic 3D-printed heat exchanger for high-temperature energy-efficient systems,” *Applied Thermal Engineering*, vol. 219, 1 2023. 77
- [169] U. Puttur, M. Ahmadi, B. Ahmadi, and S. Bigham, “A novel lung-inspired 3D-printed desiccant-coated heat exchanger for high-performance humidity management in buildings,” *Energy Conversion and Management*, vol. 252, 1 2022. 77
- [170] B. Ahmadi and S. Bigham, “Performance Evaluation of hi-k Lung-inspired 3D-printed Polymer Heat Exchangers,” *Applied Thermal Engineering*, vol. 204, 3 2022. 77
- [171] J. O. Farlow, C. V. Thompson, and D. E. Rosner, “Plates of the dinosaur Stegosaurus: Forced convection heat loss fins?” *Science*, vol. 192, no. 4244, pp. 1123–1124, 1976. 77
- [172] F. Clarac and A. Quilhac, “The crocodylian skull and osteoderms: A functional exaptation to ectothermy? The crocodylian skull and osteoderms: 1 A functional exaptation to ectothermy?” 2018. [Online]. Available: <https://mnhn.hal.science/mnhn-02557596> 77
- [173] —, “The crocodylian skull and osteoderms: A functional exaptation to ectothermy?” *Zoology*, vol. 132, pp. 31–40, 2 2019. 77
- [174] J. O. Farlow, S. Hayashi, and G. J. Tattersall, “Internal vascularity of the dermal plates of Stegosaurus (Ornithischia, Thyreophora),” *Swiss Journal of Geosciences*, vol. 103, no. 2, pp. 173–185, 2010. 77
- [175] F. Clarac, F. Goussard, L. Teresi, V. Buffrénil, and V. Sansalone, “Do the ornamented osteoderms influence the heat conduction through the skin? A finite element analysis in Crocodylomorpha,” *Journal of Thermal Biology*, vol. 69, pp. 39–53, 10 2017. 77

- [176] L. Lissethe, I. Veenstra, and C. Broeckhoven, “Revisiting the thermoregulation hypothesis of osteoderms: a study of the crocodilian *Paleosuchus palpebrosus* (Crocodilia: Alligatoridae),” Tech. Rep., 2022. [Online]. Available: <https://academic.oup.com/biolinnean/article/135/4/679/6523009> 77
- [177] C.-C. Tsai, N. Shi, J. Pelaez, N. Pierce, and N. Yu, “Butterflies Regulate Wing Temperatures Using Radiative Cooling,” in *2017 Conference on Lasers and Electro-Optics (CLEO) : proceedings : San Jose, California, USA, 14-19 May 2017.*, 2017. 77
- [178] H. S. Horn, *Social Butterflies*, Princeton: Princeton University Press, 2021., 2021. [Online]. Available: <https://about.jstor.org/terms> 77
- [179] C. L. Myhrvold, H. A. Stone, and E. Bou-Zeid, “What Is the Use of Elephant Hair?” *PLoS ONE*, vol. 7, no. 10, 10 2012. 77
- [180] P. Wright, “Why do elephants flap their ears?” *South African Journal of Zoology*, vol. 19, no. 4, pp. 266–269, 1 1984. 77
- [181] T. M. Williams’, “Heat transfer in elephants: thermal partitioning based on skin temperature profiles,” Tech. Rep., 1990. 77
- [182] A. Rips, K. Shoele, and R. Mittal, “Heat transfer enhancement in laminar flow heat exchangers due to flapping flags,” *Physics of Fluids*, vol. 32, no. 6, 6 2020. 78, 131
- [183] S. Najafpour and M. Bahrami, “A biomimetic approach to improve convective heat transfer using self-excited flaps,” *International Journal of Heat and Mass Transfer*, vol. 211, 9 2023. 78

- [184] A. A. Hagan and J. E. Heath, “Regulation of Heat Loss in the Duck by Vasomotion in the Bill,” *J. therm. Biol.*, vol. 5, pp. 95–101, 1980. 78
- [185] P. K. Phillips and J. E. Heath, “Heat exchange by the pinna of the african elephant (*Loxodonta africana*),” *Comparative Biochemistry and Physiology – Part A: Physiology*, vol. 101, no. 4, pp. 693–699, 1992. 78
- [186] E. Ching, “How the Largest Land Animal Stays Cool – And What We Can Learn From It,” 2021. [Online]. Available: <https://sites.nd.edu/biomechanics-in-the-wild/2021/04/06/how-the-largest-land-animal-stays-cool-and-what-we-can-learn-from-it/> 78
- [187] G. J. Tattersall, D. V. Andrade, and A. S. Abe, “Heat exchange from the toucan bill reveals a controllable vascular thermal radiator,” *Science*, vol. 325, no. 5939, pp. 468–470, 7 2009. 78, 79
- [188] J. P. Bons, “A review of surface roughness effects in gas turbines,” *Journal of Turbomachinery*, vol. 132, no. 2, 4 2010. 97
- [189] J. P. Bons and S. T. McClain, “The effect of real turbine roughness with pressure gradient on heat transfer,” *Journal of Turbomachinery*, vol. 126, no. 3, pp. 385–394, 7 2004. 97
- [190] J. C. Snyder, C. K. Stimpson, K. A. Thole, and D. Mongillo, “Build direction effects on additively manufactured channels,” *Journal of Turbomachinery*, vol. 138, no. 5, 5 2016. 98

- [191] J. C. Snyder, C. K. Stimpson, K. A. Thole, and D. J. Mongillo, “Build Direction Effects on Microchannel Tolerance and Surface Roughness,” *Journal of Mechanical Design, Transactions of the ASME*, vol. 137, no. 11, 11 2015. 98
- [192] K. L. Kirsch and K. A. Thole, “Numerical optimization, characterization, and experimental investigation of additively manufactured communicating microchannels,” *Journal of Turbomachinery*, vol. 140, no. 11, 11 2018. 98
- [193] —, “Heat transfer and pressure loss measurements in additively manufactured wavy microchannels,” *Journal of Turbomachinery*, vol. 139, no. 1, 1 2017. 98
- [194] —, “Experimental investigation of numerically optimized wavy microchannels created through additive manufacturing,” *Journal of Turbomachinery*, vol. 140, no. 2, 2 2018. 98
- [195] T. I-P Shih, C.-s. Lee, and K. M. Bryden, “Scaling Heat-Transfer Coefficients Measured Under Laboratory Conditions to Engine Conditions,” in *Proceedings of ASME Turbo Expo 2017: Turbomachinery Technical Conference and Exposition*, Charlotte, NC, USA, 5 2017. [Online]. Available: <http://asmedigitalcollection.asme.org/GT/proceedings-pdf/GT2017/50879/V05AT16A010/4240224/v05at16a010-gt2017-64039.pdf> 98
- [196] GWINSTEK, “Multi-Range DC Power Supply: User Manual,” Tech. Rep. [Online]. Available: https://assets.testequity.com/te1/Documents/pdf/PSW_UserManual.pdf 110
- [197] R. J. Moffat, “Describing the Uncertainties in Experimental Results,” Tech. Rep. 114
- [198] S. A. Weaver, M. D. Barringer, and K. A. Thole, “Microchannels with manufacturing roughness levels,” *Journal of Turbomachinery*, vol. 133, no. 4, 4 2011. 118

- [199] J. C. Bird, R. Dhiman, H. M. Kwon, and K. K. Varanasi, “Reducing the contact time of a bouncing drop,” *Nature*, vol. 503, no. 7476, pp. 385–388, 2013. 129
- [200] H. L. Girard, D. Soto, and K. K. Varanasi, “Waterbowls: Reducing Impacting Droplet Interactions by Momentum Redirection,” *ACS Nano*, vol. 13, no. 7, pp. 7729–7735, 7 2019. 129
- [201] B.-F. Wang, Q. Zhou, and C. Sun, “Vibration-induced boundary-layer destabilization achieves massive heat-transport enhancement,” Tech. Rep., 2020. [Online]. Available: <https://www.science.org> 130
- [202] S. K. Kumar, C. Bose, S. F. Ali, S. Sarkar, and S. Gupta, “Investigations on a vortex induced vibration based energy harvester,” *Applied Physics Letters*, vol. 111, no. 24, 12 2017. 130
- [203] H. L. Dai, A. Abdelkefi, Y. Yang, and L. Wang, “Orientation of bluff body for designing efficient energy harvesters from vortex-induced vibrations,” *Applied Physics Letters*, vol. 108, no. 5, 2 2016. 130
- [204] B. Andrawes, “Shape Memory Alloys.” [Online]. Available: <https://publish.illinois.edu/shape-memory-structures/shape-memory-alloys/> 131
- [205] J. Ma, I. Karaman, and R. D. Noebe, “High temperature shape memory alloys,” *International Materials Reviews*, vol. 55, no. 5, pp. 257–315, 9 2010. [Online]. Available: <http://www.tandfonline.com/doi/full/10.1179/095066010X12646898728363> 131
- [206] P. B. C. Leal, J. Mingear, H. Stroud, J. Zamarripa, M. Cabral, and S. Murley, “Avian Inspired Morphing Aircraft.” [Online]. Available: <https://maestrolab.tamu.edu/research/avian-inspired-morphing-aircraft/> 131

- [207] A. Ghosh, “Scaling Laws,” in *Mechanics Over Micro and Nano Scales*. Springer New York, 2011, pp. 61–94. [132](#), [133](#)
- [208] M. Denny and A. McFadzean, *Engineering Animals*. Cambridge, MA, USA: The Belknap Press of Harvard University Press, 2011. [Online]. Available: <http://ebookcentral.proquest.com/lib/utk/detail.action?docID=3300969>. [133](#)
- [209] B. Spletzer, “Scaling Laws for Mesoscale and Microscale Systems,” Tech. Rep., 1999. [Online]. Available: <https://www.osti.gov/servlets/purl/10186> [134](#)
- [210] D. E. Brandt and R. R. Wesorick, “GE Power Generation GE Gas Turbine Design Philosophy,” Tech. Rep. [134](#)
- [211] S. A. Ocko, A. Heyde, and L. Mahadevan, “Morphogenesis of termite mounds,” *Proceedings of the National Academy of Sciences of the United States of America*, vol. 116, no. 9, pp. 3379–3384, 2 2019. [138](#)

Appendix

Appendix A

Superheater Tube

A.1 Heat Transfer Calculation Assuming 1D Case

The 1D resistance network in a superheater tube consists of a combination of radiation and convection (from flue gas to ash), conduction through ash, conduction through the metal tube wall, and forced convection from metal wall to high-pressure steam (see Figure A.1 for the thermal resistance network). The heat flux (W/m) can be calculated as below in equation A.1:

$$Q_t = \frac{(T_f - T_s)}{R_t} \quad (A.1)$$

The overall resistance can be calculated as below in equation A.2:

$$\sum R_t = R_1 + R_2 + R_3 + R_4 \quad (A.2)$$

$$\sum R_t = \frac{1}{\pi(h_{rad} + h_{conv})d_d} + \frac{\ln \frac{d_d}{d_o}}{2\pi k_{deposit}} + \frac{\ln \frac{d_d}{d_i}}{2\pi k_{metal}} + \frac{1}{\pi h_{steam}d_i} \quad (A.3)$$

Since,

$$Q_t = \frac{(T_f - T_d)}{R_1} = \frac{(T_d - T_o)}{R_2} = \frac{(T_o - T_i)}{R_3} = \frac{(T_i - T_s)}{R_4} \quad (\text{A.4})$$

where T_f is the flue gas temperature (900°C), T_d is the deposit temperature, T_o is the outer wall temperature, T_i is the inner wall temperature, and T_s is the steam temperature (550°C). R_1 is the thermal resistance from flue gas to ash, R_2 is the thermal resistance of ash, R_3 is the thermal resistance of the metal wall, and R_4 is the thermal resistance of the metal wall to steam. No inner and outer scales have been assumed. A maximum deposit thickness of 0.005 m has been assumed. Using the iterative method and equations A.1, A.3, and A.4, the individual thermal resistances were calculated as $0.032^\circ\text{C}/\text{W}$, $0.057^\circ\text{C}/\text{W}$, $0.002^\circ\text{C}/\text{W}$, and $0.013^\circ\text{C}/\text{W}$ for R_1 , R_2 , R_3 , and R_4 , respectively (see Figure A.2). This clearly shows that the presence of ash and heat transfer from flue gas to ash offers the highest thermal resistance and leads to decreased heat flux. The h_{rad} was calculated as $159.36 \text{ W}/\text{m}^2 \text{ }^\circ\text{C}$, which is further used for the 2D thermal simulation and circumferential thermal profile calculations. Before performing the 2D FE simulations for circumferential thermal gradient calculations, the 2D FE model was created with uniform ash deposition of 0.005 m thickness and was benchmarked with the above 1D hand calculations. FE results were within 2%–5% of the hand-calculated values. This indicates the goodness of the FE model for further analysis of thermal gradient with nonuniform ash distribution.

A.2 Schlieren Optics Experimental Setup

The following components were used and aligned to build up the Schlieren optics to achieve the overall setup. A concave spherical mirror (with aluminum protection) of 60 in. focal length and 6 in. diameter was procured from Edmund Optics (Figure A.3A). A mount for

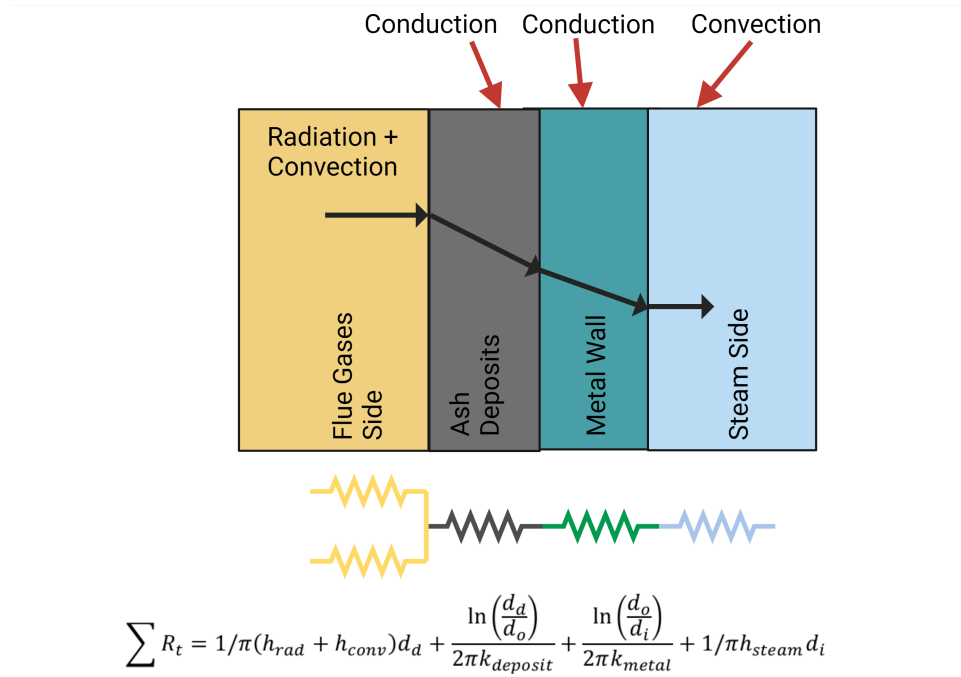


Figure A.1: One-dimensional resistance network in superheater tubes.

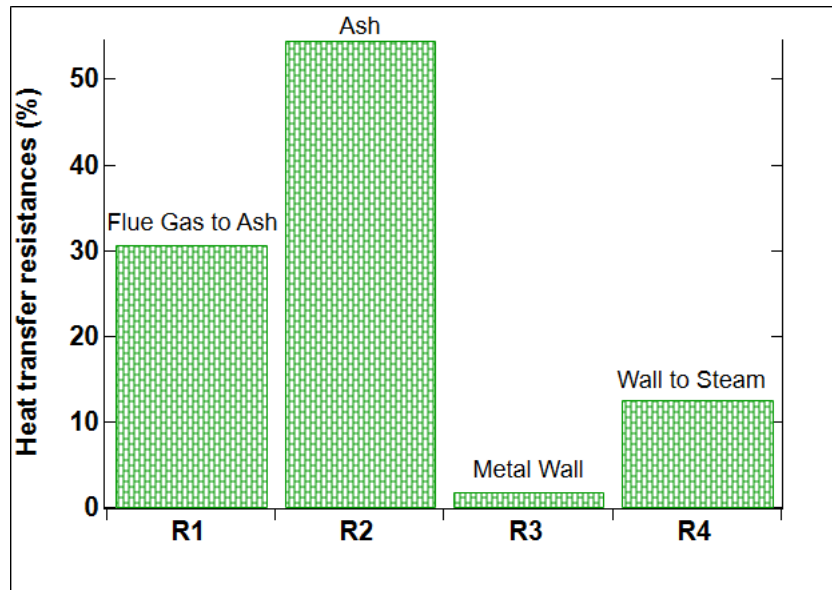


Figure A.2: Calculated thermal resistances for superheater tubes for 0.005 m thick ash.

holding the mirror was designed in CAD and was 3D printed using Fortus 400mc (acrylonitrile butadiene styrene material, grey color) (Figure A.3A). A razor blade is kept at a distance of ~ 120 in. from the mirror center, where the point LED image converges (Figure A.3C). A DSLR Canon camera with a zoom lens (75–300 mm focal length and the corresponding f-number 1:4.5–5.6) (Figure A.3B) is kept just behind the light block to capture the videos. A WWTAC A1S LED flashlight (1,150 lumens) covered with aluminum foil with a point hole (Figure A.3B) is used for the point LED source. Camera and point source LED alignment with the mirror is performed using a laser pointer (Figure A.3B). Differently shaped tube samples and mounts were designed in CAD and 3D printed using polylactic acid material (for tubes) and carbon fiber nylon (for mounts) on Makergear M3. These samples are placed in front of the mirror in the test region (Figures A.3D, A.3E, A.3F, and A.3G). Gas flow was generated around the tube surface using the refrigerant HFC 152a, which has a relative vapor density of ~ 2.3 (air = 1), thereby providing visible flow variation around differently shaped geometry via Schlieren imaging. The recorded video can be accessed at <https://youtu.be/XDHRqFlycz0>.

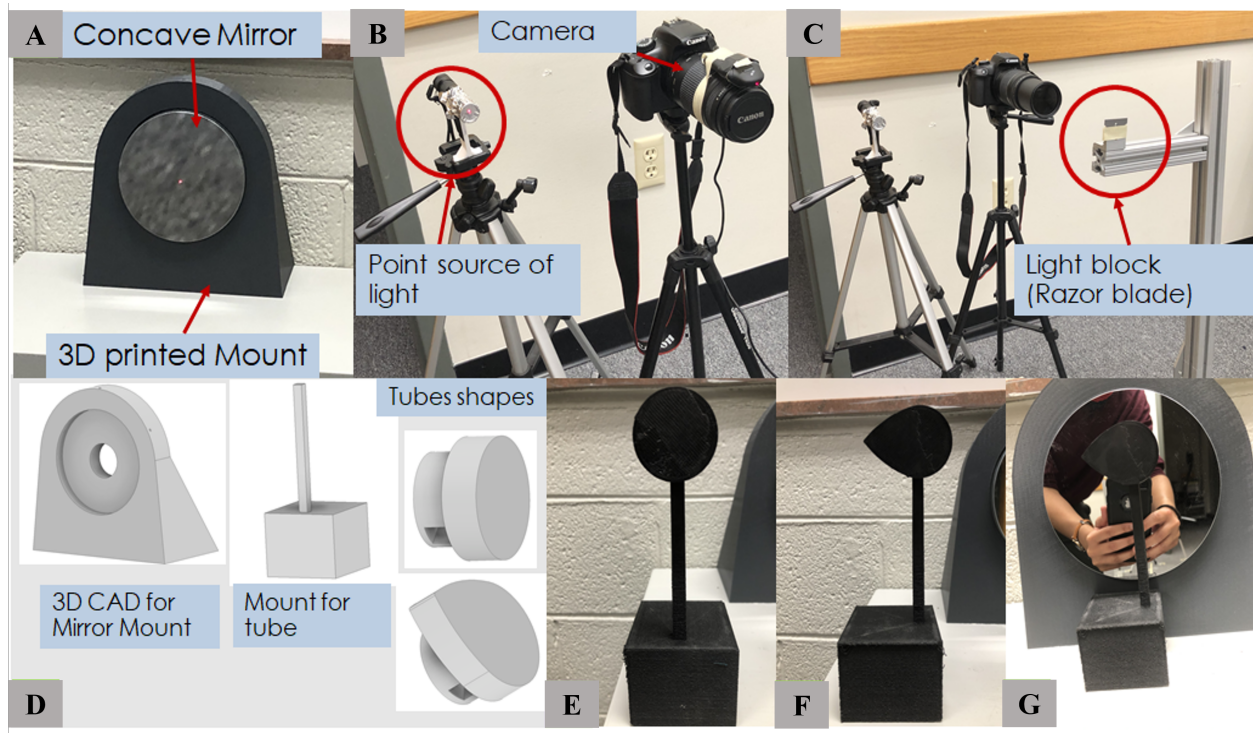


Figure A.3: Calculated thermal resistances for superheater tube for 0.005 m thick ash.

Vita

Vanshika Singh was born on July 11, 1994, in Varanasi, an ancient city in northern India, and was raised in New Delhi. She did her schooling at Central School (Kendriya Vidyalaya), Delhi, and graduated from her high school in 2012. Growing up, Vanshika had an interest in sports and was an avid cricket player. She did her Bachelor of Technology in Mechanical and Automation Engineering from Indira Gandhi Institute of Technology, Kashmere Gate, New Delhi, India (2012–2016). Following this, she worked at General Electric Aerospace, Bengaluru, India, finishing a three-year leadership program called the Edison Engineering Development Program (2016–2019). In 2019, she joined The Bredesen Center for Interdisciplinary Education and Research for the Ph.D. program in the Energy Science and Engineering Department. She performed her research under the guidance of Dr. Michael Kirka and Prof. Sudarsanam Suresh Babu. During her Ph.D. program, she leveraged metal additive manufacturing to develop a new design strategy inspired by nature and implemented for real-world high-temperature applications. After earning her Ph.D., she hopes to work in a national laboratory or an industry where she can design lower-emission energy systems for high-temperature applications. Additionally, she enjoys volunteering and mentoring students (from small kids to graduate students) and likes to transfer knowledge to the emerging generation.