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LIQUID VAPOR PHASE CHANGE FOR HIGHLY EFFICIENT AND SUSTAINABLE
COOLING

BY

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DISSERTATION

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ABSTRACT

Liquid-vapor phase change of water is an essential and complex phenomenon germane to many natural and industrial systems. Desirable characteristics such as high latent heat, wide availability, non-toxicity, and low cost, allow water to become the ideal fluid to extract heat in many powerplant and electronic related applications. This is especially important recently as effective thermal management has become a critical bottleneck that hinders further advancements in new technologies that continue to increase in functionality and heat generation. Typically, two-phase flow boiling and evaporation are the two most widely adopted techniques to extract heat. Given their ubiquitous importance in cooling applications, understanding the underlying physics that govern both is paramount to fully maximizing their potential. In doing so, we are able to attain more efficient systems that are more cost-effective and sustainable. To gain a fundamental understanding of thermal transport during evaporation, past approaches have utilized optical, shadowgraph, and infrared (IR) imaging. Although such methods can reveal rich information, they are unable to quantify the temperature of the surrounding gas phase. To overcome this challenge, others have used a micro-thermocouple to probe the temperature distribution around or near the liquid-vapor interface of a sessile droplet. However, in these past investigations, the spatial resolution of measured temperatures near the liquid-vapor interface was limited and the sessile droplet varied in size while the temperatures were acquired. To provide a better understanding of the liquid-vapor phase change phenomenon, a platform that would overcome past difficulties encountered during the study of droplet evaporation is required. This dissertation proposes we develop such a platform by coupling a piezo-driven droplet generation mechanism to a controlled micro-thermocouple to probe microdroplet evaporation. Although evaporation provides effective means for thermal management, it is only appropriate for sub-saturation temperatures. Alternatively, two-phase flow boiling allows heat extraction at higher temperatures. The rate of heat extraction during flow boiling is greatly influenced by the surface topology. Previous studies to enhance heat transfer utilized different techniques such as inserts, surfactants, dimples, extended surfaces, ribs, vortex generators, and many more. These techniques depend on geometric modifications to induce turbulence and create additional nucleation sites to augment the heat transfer. In order to extend on these augmentation techniques, this dissertation also investigates the feasibility of microstructures in enhancing heat transfer in copper and aluminum tubes.

To my family, for their endless love and support

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- Mohamed Hatem Mousa

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NOMENCLATURE

$\dot{m}$	Mass flow rate	(kg/s)
A	Surface Area	(m ²)
Bo	Bond number	
C_p	Specific Heat Capacity	(J/(kg·K))
d	Diameter	(m)
d_c	Depth of Wing Cut	(m)
e	Wire Diameter	(m)
f	Friction factor	
g	Acceleration Gravity	(m/s ²)
G	Mass Flux	(kg/(m ² ·s))
h	Heat Transfer Convection Coefficient	(W/m·K)
H	Height	(m)
h_{fg}	Latent heat of vaporization	(J/kg)
I	Current	(A)
k	Thermal conductivity	(W/m ² ·K)
L	Length	(m)
M	Calculated parameter	
n	Number of parameters	
Nu	Nusselt number	
o	Outer Diameter	(m)
p	Helical Pitch	(m)
Pr	Prandtl number	
q	Heat flux	(W/m ²)
Q	Heat input	(W)

$R3$	Performance Criterion	
Ra	Rayleigh number	
Re	Reynolds number	
rh	Rib height	(m)
St	Stanton number	
T	Temperature	(K)
tp	180-degree twist pitch	(m)
U	Average axial flow velocity	(m/s)
V	Potential Difference	(V)
w	Width of tape	(m)
x	Vapor quality	
y	Twist Ratio	
Y	Uncertainty	
Z	Primary Parameter	
C	Vapor concentration	($\mu\text{g}/\text{m}^3$)
D	Diffusion coefficient	(mm^2/s)
F	View factor	
Gr	Grashof number	
R	Radius	(m)
eff	Insulation effectiveness	
ΔP	Pressure Drop	(Pa)
$P_{i\tau-\frac{1}{2}}(\cosh \alpha)$	First-kind Legendre function	

GREEK SYMBOLS

θ	Droplet contact angle	($^\circ$)
$\dot{v}$	Volumetric Flow Rate	(m^3/s)

σ_k	Stefan-Boltzmann constant	(W/m ² ·K ⁴)
$\forall$	Volume of the bead	(m ³)
α	Void fraction	
β	Helical angle	(°)
Γ	Perimeter	(m)
δ	Liquid Film Thickness	(m)
λ	Apex Angle	(°)
μ	Dynamic Viscosity	(Pa·s)
ν	Kinematic viscosity	(m ² /s)
τ	Integration dummy variable	
ω	Thermal expansion coefficient	(K ⁻¹)
α	Void fraction	
η	Thermal performance	
ξ	Thermal diffusivity	(m ² · s ⁻¹)
ρ	Density	(kg/m ³)
σ	Surface Tension Coefficient	(N/m)

SUBSCRIPTS

∞	Ambient air
a	Augmented Tube
adv	Advancing contact angle
b	Droplet base
D	Dimpled tube
e	Total evaporation
f	Liquid (used as friction factor)
g	Vapor (Used as Acceleration due to Gravity)

h	Hydraulic
i	Inner
in	Tube inlet
in ph	Preheater inlet
in ts	Test section inlet
L	Liquid
lf	Liquid film
lv	Liquid vapor
ni	Plain tube
O	Smooth
OD	Outer diameter
ol	Outer liquid
out	Tube outlet
out ph	Preheater outlet
out ts	Test section outlet
p	Plain tube (used as p Helical Pitch (m))
pe	Pre-evaporator
ph	Preheater
pp	Pumping power
r	Root Diameter
rad	Radiation
ref	Refrigerant
s	Surface
sat	Saturation
sub	Substrate
T	Tip Diameter
t	With turbulator

tc	Thermocouple
ts	Test section
w	Water
x	Certain axial position from test section inlet

SUPERSCRIPT

app	Apparent contact angle
-----	------------------------

ABBREVIATIONS

<i>AC</i>	Alternating current
<i>AM</i>	Additive manufacturing
<i>AR</i>	Aspect Ratio
<i>CFC</i>	Chlorofluorocarbon
<i>CHF</i>	Critical heat flux
<i>CMC</i>	Critical micelle concentration
<i>CR</i>	Conical ring
<i>Cu</i>	Copper
<i>DDIR</i>	Discrete double inclined ribs
<i>DR</i>	Depth Ratio
<i>EF</i>	Enhancement Factor
<i>EHD</i>	Electrohydrodynamic
<i>EHT</i>	Enhanced heat transfer
<i>EV</i>	Electric vehicle
<i>FD</i>	Fin Depth
<i>FMCG</i>	Fast-Moving Consumer Goods
<i>FW</i>	Fin Width

<i>HCFC</i>	Hydrochlorofluorocarbon
<i>HTC</i>	Heat Transfer Coefficient
<i>HVAC</i>	Heating, Ventilation, and Air Conditioning
<i>ID</i>	Inner Diameter
<i>LMTD</i>	Logarithmic Mean Temperature Difference
<i>MHD</i>	Magnetohydrodynamic
<i>Ni</i>	Nickel
<i>OD</i>	Outer diameter
<i>O-DWT</i>	Oblique delta winglet tape
<i>ONB</i>	Onset of Nucleate Boiling
<i>PCM</i>	Phase change materials
<i>PCR</i>	Perforated conical rings
<i>PF</i>	Performance Factor
<i>PF</i>	Pressure Factor
<i>PFE</i>	Energy Performance Factor
<i>PFS</i>	Surface Performance Factor
<i>POE</i>	Polyol-ester
<i>PVC</i>	Polyvinyl chloride
<i>S-DWT</i>	Straight delta winglet tape
<i>SLS</i>	Selective laser sintering
<i>THPR</i>	Thermal Hydraulic Performance Ratio
<i>TT</i>	Twisted tape
<i>VR</i>	Vortex rings
<i>RH</i>	Relative Humidity

Chapter 1. Introduction

1.1 Background

Effective thermal management of processes is paramount to enhance the reliability of systems and prevent premature failure. Consequently, several methods have been presented in the past to remove large quantities of heat within geometrically constrained spaces. This is especially important in recent years with many industries aiming to increase system efficiencies in order to attain smaller and more sustainable devices characterized by lower material consumption, carbon footprint, and waste production. Additionally, in the electronics field, effective thermal management presents an obstacle to further advancements in the field. To that end, the main focus of this dissertation lies in understanding the thermal transport fundamentals that govern liquid-vapor phase change of water in order to utilize it in cooling applications more effectively and efficiently. Specifically, evaporative liquid-vapor phase change of water micro-droplets and two-phase saturated flow boiling are investigated.

For liquid vapor phase change of water micro-droplets, a micro-thermocouple is used to probe the temperature distribution around an evaporating sessile water droplet resting on different functional substrates. By mapping the gas phase temperature distribution, rich information regarding the fundamentals of thermal transport pathways is revealed enabling the validation of decades old frameworks and theories. This enables better understanding of how to utilize evaporation more efficiently in different cooling applications.

For two-phase flow boiling, a novel technique to enhance the heat transfer is presented. Specifically, the effect of microstructures on the heat transfer and pressure drop of copper and

aluminum is investigated and benchmarked against plain and commercially available enhanced tubing. By augmenting the thermal performance of metallic tubes, more power dense and compact heat exchangers can be designed that cater to sustainability and efficiency constraints.

1.2 Outline

In chapters 2-5, the gas phase temperature mapping of evaporating water micro-droplets is investigated. In Chapter 2, a literature review of past approaches used to infer information regarding evaporation is presented followed by how the proposed experimental work presented in this dissertation is able to cater to the deficiencies of past approaches. Subsequently, in Chapter 3, a detailed outline of the experimental setup is presented along with the methodological approach adopted to characterize the thermal transport of an evaporating water droplet. Different parameters including, surface temperature, surface wettability, surface thermal conductivity, and droplet size and how they affect evaporation dynamics are then discussed in detail in Chapter 4. Finally, Chapter 5 presents conclusionary remarks regarding the preceding chapters as well as future outlook.

In chapters 6-7, two-phase saturated flow boiling for high temperature applications is investigated. In chapter 6, a literature review of past approaches to augment heat transfer in single- and two-phase flow is presented along with critiques and remarks that describe when each approach is most suitable. Subsequently, in chapter 7, an outline on how to construct the experimental setup to investigate different heat transfer augmentation techniques with water as the working fluid is presented. Finally, a new passive technique that depend on microstructures to enhance heat transfer in metallic tubes is investigated.

Chapter 2. Review of Techniques to Study Evaporation

2.1 Background

Droplet transport and evaporation is an important phenomenon present in nature and industry[1]. Evaporation presents a simple yet effective way to regulate temperature and has thus been utilized across several applications ranging from power plant cooling to building thermal management[2] to nanofabrication[3,4]. More recently, droplet evaporation and transport have seen renewed interest in biological applications studying both direct and indirect pathogen and virus transmission[5,6], as well as utilization in electronics thermal management[7,8]. With the development of wide bandgap semiconductor materials, the power density of modern electronics has increased at a rate concurrent with heat generation. Evaporation presents an efficient mechanism to absorb high heat fluxes across small areas[9]. Alternate uses for droplet evaporation include DNA mapping[10] where water droplets containing DNA are stretched during evaporation and deposited on substrates creating optical maps of DNA molecules. Furthermore, evaporation of sessile droplets governs coffee ring dynamics[11,12] which are important to ink-jet printing, where inhibiting induced outward flows allows for greater uniformity of the deposits[13]. Uniformity is especially important for drug release applications, where polymer inkjet printing is utilized[14], as well as in the manufacturing of multicolor polymer light-emitting diode displays, polymer electronics, and 3-D printing[14]. Given its ubiquitous importance, understanding the dynamics governing liquid droplet evaporation on surfaces is paramount. Evaporation is known to be

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affected by several parameters including the type of surface on which the liquid resides, the nature of the liquid[15], and the surrounding environment[16,17]. To gain a fundamental understanding of thermal transport during evaporation, past approaches have utilized optical[18,19], shadowgraph[20,21], and infrared (IR) imaging[22–24]. Although such methods can reveal rich information, they are unable to quantify the temperature of the surrounding gas phase. Furthermore, unlike thermocouples, IR thermography and shadowgraphs are limited in their ability to measure the temperature at specific locations and depend on the liquid being studied. To overcome this challenge, others have used a micro-thermocouple to probe the temperature distribution around or near the liquid-vapor interface of a sessile droplet [25,26]. However, in past investigations, the spatial resolution of measured temperatures near the liquid-vapor interface was limited and the sessile droplet varied in size while the temperatures were acquired. Further investigations have introduced steady-state methods with bottom injection[27,28], top injection[29], and funneling[30,31]. However, in all past studies, the surfaces had high water affinities (were wetting). Moreover, although elucidating, previous studies have been limited in the spatio-temporal resolution for temperature data acquisition. In addition, the accessible droplet length scale ($Bo > 1$, where Bo represents the Bond number) was large, resulting in evaporating macroscale (> 1 mm) droplets. More recent work has employed the optical monitoring of evaporating droplets having different sizes to deduce the effect of surface thermal conductivity on evaporation. For a detailed review of past experimental techniques developed and available for the study of heat and mass transfer surrounding individual droplet evaporation dynamics, see Section 2.2.

2.2 Experimental Methods

In this section, a review of different experimental methods that study droplet evaporation is presented and critiqued. Furthermore, the experimental method presented in Chapters 3 and 4 is compared to these experimental methods in order to highlight the novelty of the work and how it addresses the gap in the field.

Pendant droplets and thermistors: One of the earliest experimental platforms to investigate evaporation dynamics involved pendant droplets [32–34]. Erbil and Dogan [34] investigated evaporation induced cooling of hanging droplets by dispensing 11 different liquids using a syringe at the crossover of two polypropylene fibers. The authors measured the changing mass of the evaporating pendant droplet using a microbalance and the temperature and pressure inside the chamber with a hygrometer. They concluded that significant surface cooling occurs for liquids with high vapor pressure. Although these investigations are the pioneering method to study evaporation, the method only considers heat conduction as the main energy transport mechanism in the absence of a solid substrate. Moreover, the method does not consider the substrate and does not determine the gas phase temperature distribution.

A second pioneering method involves the use of a thermistor to measure the temperature at the liquid-vapor interface [35,36]. The thermistors used in past investigations are of length scales considerably larger than that of micro-thermocouple beads (which has been adopted in recent studies to be discussed below) and therefore the ability to precisely measure the temperature with high spatial resolution and at certain locations was difficult. An example of investigations using the thermistor was the work of Jarvis[35]. The thermistor was used to predict the temperature of the surface of water by measuring the temperature of the water at several depths under the liquid-

vapor interface (within the liquid). Using extrapolation, the temperature of the water at the surface was predicted, while the temperature profile within the gas-phase was overlooked.

Infrared (IR) Thermography: One of noninvasive techniques used to study the temperature profiles generated during the evaporation dynamics of liquids is IR thermography[22–24,37–45]. By knowing the emissivity of the liquid, the measured IR spectra can be used to infer the temperature at the liquid-vapor interface or at the liquid-solid interface, i.e., at the droplet base area. Josyula *et al.*[23] applied IR thermography to image the evolution of the thermal patterns and the effect of Marangoni convection on evaporation kinetics of a sessile water droplet on a heated surface. It was noted that under high temperatures with local heating, Marangoni convection was more prominent than under uniform heating. However, the convection and thermal patterns faded at later stages of evaporation as a consequence of the decrease in the droplet volume during the dynamic evaporation process. Chandramohan *et al.*[22] developed an accurate method to characterize experimentally the physical mechanisms that drive the evaporation of 4 μl droplets generated using a pipette. Applying IR thermography imaging the side of the droplet, they investigated the cooling of the liquid-vapor interface how it might suppress the rate of evaporation on substrates with low water affinity, i.e., contact angle of approximately 140° , evaporating in the constant contact radius mode. The platform offered the means to validating numerical studies by providing the appropriate experimental evidence. Although IR temperature measurements are not invasive, using a thermocouple has distinct advantages:

- Thermocouples enable gas phase temperature mapping opposed to IR imaging which is limited to measuring liquid or substrate temperatures and cannot resolve the temperature of the surrounding gas phase.

- Thermocouples are able to measure micron length scales with high spatial and temporal resolutions. Using IR thermography instead would result in a poor resolution and pixelated imaging with a maximum resolution approach a few microns/pixel due to diffraction limitations.
- IR measures an average temperature over the optical path [28,40,44,46] whereas thermocouple measure temperature at specific locations.
- IR measurement values depend on the relation between material emissivity and temperature subjecting resulting values to inaccuracies. Thermocouple measurements are not liquid or gas optical properties dependent although they do require initial calibration.

Shadowgraph Imaging: Another optical technique that has been utilized to study evaporation fundamentals is shadowgraph imaging[20,21,47,48]. Although shadowgraphs have revealed rich information in the topic, similar to IR thermography, they have several important limitations when compared to micro-thermocouples:

- Thermocouples are able to measure micron length scales with high spatial and temporal resolution. Shadowgraphs have poor spatial resolution.
- Thermocouples enable gas phase temperature mapping while shadowgraphs cannot.
- Shadowgraphs measure an average temperature over the optical path [28,40,44,46] whereas thermocouple measure local temperature at a specific location governed by the user.
- Shadowgraphs measurement values depend greatly on the fluid properties such as the relation between density, index of refraction, differences in concentration and/or temperature (in turn influencing the density) making resulting values subject to high uncertainty. Thermocouple measurements are not liquid dependent and only require initial calibration.

More recently, Zhang and Chao[47] developed a new platform that used laser refracted shadowgraphs to measure the dynamic contact angle and spreading of sessile droplets simultaneously on opaque metal surfaces. This was achieved by measuring the sessile droplet's contact and refracted shadowgraphic image diameters. The technique was not used to probe gas phase temperature near the droplet, nor was it used to elucidate experimentally the energy balance for evaporating droplets.

Optical Imaging: Optical imaging[18,19,49–56] is another commonly and widely used technique to deduce information about evaporation. Lopes *et al.*[50] and David *et al.* [26] studied the effect of substrate thermal conductivities of glass and silicon, and aluminum, titanium, Macor and PTFE, respectively, on the overall evaporation time by monitoring the sessile droplet profile with a CCD camera looking from the side. It was deduced that having higher substrate thermal conductivity yielded higher evaporation rates. In both works, numerical and mathematical models were additionally proposed to validate the experimental results.

Although optical imaging from the side has proven to be extremely useful and is one of the preferred techniques to address the evaporation of sessile droplets evaporation trends, this technique is limited in its ability to quantify gas-phase temperature around an evaporating droplet. In consequence, coupling this method with thermocouples is necessary if temperature-related measurements were to be undertaken.

Evaporation Simulations: Thus far, experimental approaches have been limited to local measurements and a lack of fully resolving fully the spatial temperature distribution in the gas phase surrounding evaporating droplets. These limitations are inherently due to the lack of a platform that would enable gas phase temperature mapping of micron sized droplets as highlighted upon in above studies. To overcome these challenges, numerous numerical simulation approaches

and mathematical approaches[17,57–60] have been developed with specific focus on the gas phase heat and mass transfer, which have not been accessible through detailed experiments until the experimental work presented in Chapter 3. The majority of numerical simulation studies which focus on droplet evaporation are motivated by studying the experimentally unobtainable parameters measured in Chapter 4.

Thermocouple Measurements: As highlighted previously, thermocouples are preferred over IR thermography or shadowgraphy due to their ability to measure the actual temperature at the specific locations in the gas phase and their independency on the properties of the fluid being studied. As a result, the use of thermocouples provides a more accurate noninvasive methodology. Consequently, several measurement platforms that attempted to map temperature profiles in the gas phase using thermocouples exist in the literature. Although similar in concept, the approach presented in Chapter 3 differs greatly from previous ones in the way the microdroplet is generated allowing for the study of advanced functional surfaces that could not be accessed before as well as decreasing the length scales of the studied droplets when compared to earlier works. Below is a summary of the four distinct methods used to generate and study evaporating droplets:

- **Droplet Deposition Through a Funnel:**

A category of platforms dealt with constant sized “droplets” being generated through funneling water into an evacuated chamber allowing for the study of droplet evaporation with a steady state liquid-gas interface [30,31,61–63]. Fang and Ward[63] funneled water of constant temperature into an evacuated chamber through a funnel mouth with radius of 3 mm. With the water in the funnel, they measured, using a thermocouple, the temperature at five positions vertically away from the liquid-gas interface of the funnel-shaped droplets. Using the thermocouple, the temperature measurements revealed a discontinuity between the gas and the liquid phase. Duan

and Ward[64] followed this experimental method but with a funnel mouth radius of 3.5 mm in a vacuum chamber. Using a thermocouple, they measured the temperature at the centerline of the funnel and four different horizontal distances from it (0.7 to 2.8 mm) and different vertical positions. The temperature measurements revealed that the Marangoni number dictated whether thermal conduction to the interface alone was enough to cause evaporation or whether thermocapillary convection also played a role.

Briefly, this method along with the previously reported results using it differs substantially from the work in Chapter 3 in that a) the required droplet size is large ($Bo > 1$) in order to ensure stability of the liquid-vapor interface. The Bo number measures the importance of gravity forces with respect to surface tension forces. This relationship dictates the overall shape of the droplet b) the gas phase temperature distribution was not measured in the azimuthal direction but rather in the horizontal direction, with a main focus on the liquid-gas interface temperature and temperature jump at the interface, c) these studies were not able to decouple energy transfer mechanisms from the substrate, self-cooling, and surrounding ambient as proposed in Chapter 3, and d) these works could not address droplet phase-change under free evaporation conditions as per the spatial confinement within the chamber and the reduced pressures utilized..

Micron-scale droplets, opposed to millimetric scale droplets utilized in the studied above, undergo differing evaporation dynamics. For micron-sized droplets, similar to the ones investigated in Chapter 4, surface tension is the dominant force. Consequently, thermocapillary convection is prominent and the induced fluid motion and thermal stratification is intensified at the liquid-gas interface. For the previously reported results, the prominent force is gravity due to the larger radius size of the droplets. Therefore, buoyancy convection is intensified and the global fluid motion results in a more homogenously mixed temperature field. It is also important to

consider that the approach presented in Chapter 3 provides a platform that facilitates the versatile investigation of solid surfaces including the effect of substrate thermal conductivity, temperature, and wettability, all of which are absent in the above reviewed investigations utilizing a funnel, which otherwise should be functionalized directly on the funnel. Furthermore, the platform introduced enables easy investigation of different surfaces without damaging them as we are able to dispense micro-droplets from above the surface and do not require any hole to be drilled from the bottom. In fact, pumping water through the bottom of the chamber hinders accurate characterization of superhydrophobic surfaces due to the absence of air entrapped within the structures in the suspended state.

- **Droplet Injection From the Bottom through the Substrate:**

Another set of platforms allowing for the study of droplet evaporation of constant sized droplets, i.e., steady state liquid-gas interface generated droplets via a syringe that supplies water to the bottom of the substrate[28,65,66]. Ghasemi and Ward [65] pumped water from the bottom of the chamber to form a constant sized droplet of 9mm in radius ($Bo > 1$) on a copper substrate of temperature 4 °C to eliminate buoyancy-driven convection. Using a thermocouple, the temperature was then measured at 10 positions along the liquid-gas interface. In their experiment, the energy transport mechanism of an evaporating droplet was studied. By eliminating the buoyancy-driven convection, it was noted that 98.4% of the energy transport at the interface was by thermocapillary convection, with the rest being thermal conduction. However, the intensity of thermal conduction was shown to change along with the interface with the maximum being at the apex of the droplet.

Similar platforms attempted to measure the temperature around the interface of a water bed produced via a syringe that supplies water at the bottom of a drilled hole in the stage [27,67]. Duan and He[27] measured the temperature at the liquid-gas interface, gas and liquid phase using a

thermocouple and repeated the experiment by moving the thermocouples at different positions horizontally. From their measurements, it was concluded that a larger interfacial temperature gradient with heating at a flat surface can increase evaporation.

Similar to the funnel technique, these investigations were mainly concerned with droplets in the millimeter-scale and thus the dominant force is gravity. As highlighted previously, for micron-scale droplets, which are by far the most common sizes playing a role in both industrial applications and nature, the governing physics are different as the dominant force is surface tension. Gravitational forces induce the distortion of the droplet shape and density driven flows resulting in temperature field is more homogeneously mixed unlike surface tension forces that induce thermal stratification that is prominent at the liquid-gas interface. Further, these investigations do not describe and/or did not allow to address the effect of droplet size, substrate thermal conductivity, temperature, and wettability on evaporation dynamics, stressing the enhanced versatility of the technique presented in Chapter 3. For example, analysis of the heat pathway through the substrate and the surrounding gas and how it is affected by parameters such as droplet size and substrate thermal conductivity is absent. Moreover, the complete and accurate mapping of the gas phase temperature distribution is absent in these earlier investigations, which is also stressed as one of the main contributions of Chapters 3 and 4. Furthermore, the platform introduced in Chapter 3 facilitates the versatile investigation of different surfaces without damaging them as we are able to dispense the droplets from above the surface and do not require any hole to be drilled from the bottom. Finally, pumped water through the bottom of the chamber hinders the accurate characterization of advanced functional surfaces such as superhydrophobic surfaces and/or slippery lubricant infused porous due to the absence of air entrapped within the

structures in the suspended state and the induced displacement of the oil if pumping liquid from the bottom, respectively.

- **Droplet Injection From the Top:**

Another set of platforms, dispense the droplets from the top. David *et al.*[26] investigated the temperature of an evaporating droplet on four surfaces varying in thermal conductivity with the same order of roughness at ambient temperature conditions under dynamic droplet evaporation where the volume, i.e., contact radius and/or contact angle, decreased in time. In their work the temperature was measured via a 0.25 mm thermocouple at two different positions at the triple contact line and within the droplet bulk and at five different vertical heights parallel at the centerline of the droplet, and once in between. In this particular case, the one order of magnitude greater size of the thermocouple and absence of a micromanipulator (distances from the droplet interface are not provided) as well as the dynamics of the droplet shape changing in time, are some of the drawbacks, which have been overcome within Chapter 3.

Even though this study was able to overcome some of the limitations the preceding methods, the temperature measurements were mainly concentrated at limited positions such as the contact line and the apex of the droplet. In Chapters 3 and 4, the exquisite micrometer spatially controlled position of the thermocouple enables us to measure a wider range of points along the droplet periphery and in the radial direction and map the temperature of the gas phase nearer to the liquid-gas interface. Further, the above report does not characterize the heat pathway through the substrate or the surrounding gas and how it is affected by parameters such as droplet size and substrate thermal conductivity. Also, unlike David *et al.* [26] the droplets in Chapters 3 and 4 are steady and both the size and the curvature of the liquid-gas interface do not change in time during the experimental runs. Moreover, David *et al.* [26] investigated surfaces with the same order of

roughness while the investigation in Chapter 4 elucidates the evaporation dynamics of surfaces having several roughnesses. Lastly, the previous reports did not study the effect of surface wettability, and droplet size on the evaporation dynamics.

- **Liquid layers Injected From the Top:**

Gatapova *et al.*[25] used a syringe to deposit a liquid layer having 1.7 mm thickness on top of a PTFE cuvette with 35 mm diameter. Using three thermocouples of diameter size 4 μm , the temperature at three different horizontal positions above the liquid layer was measured. The cuvette is heated by a brass rod of a radius 1.6 mm that is inserted in the circular hole present at the center of the cuvette. Like preceding investigations, the authors detected a temperature jump that increases with increasing temperature difference between the heater and the ambient. In this study the size of the custom-made thermocouple allowed for an increased spatial resolution; however, it did not allow to address the temperature profiles of curved interfaces as is the case of sessile droplets.

Comparison Summary of Previous Work with the Study Presented in Chapter 3: While gas phase temperature mapping has proven to provide useful information and insights on the better understanding of droplet evaporation, no study exists to date that is able to map the temperature around droplets having micrometer length scales with the temporal and spatial control resolution we report. The proposed method in Chapter 3 is unique as it enables the gas phase temperature mapping with the precision and accuracy required to facilitate the quantification of the heat transfer pathways down to a micron-sized steady droplet from the substrate and the surroundings as opposed to previous studies focused on droplets with length scales in the mm range evaporating in a distinct regime when compared to the work Chapter 3. Moreover, the proposed technique in Chapter 3 allows the temperature mapping of the gas phase of droplets with increased curvatures

than earlier reported in the literature where investigations were limited to wetting substrates, i.e., contact angles below 90° .

In addition, we further demonstrate through experiments in Chapter 4, the outsized role played by the mass transfer resistance in the gas during evaporation when compared to the other thermal resistances. Furthermore, due to the ability of the novel platform in Chapter 3 to deposit microdroplets with controlled frequency, we are able to attain evaporating droplets of constant size, curvature, radius and contact angle, which enabled us to study the effect of droplet radius size on the pathway of heat transfer. Moreover, we were able to further extend earlier investigations to a wide array of surface roughness, thermal conductivity, and substrate temperature not accessible in earlier investigations.

Previous work that attempted to measure temperatures around a constant sized droplet that was deposited either through funneled water into an evacuated chamber or pumped it through the bottom of the chamber. In both cases, the accurate characterization of advanced functional surfaces such as superhydrophobic and/or slippery infused lubricant porous surfaces was hindered due to the absence of air entrapped within the structures in the suspended state and/or the displacement of the infused oil as a consequence of the water pumped from the bottom. Further, funneled water represents a different droplet geometry than the one studied in our investigation while the size of the droplets studied when pumping water from the bottom where above the capillary length for water, i.e., $Bo > 1$.

Another point to consider is that in all these previous studies, the droplets dispensed are in the mm scale and so the interfaces are shaped and affected by gravitational body forces. However, since length scales in Chapter 4 are in the micron range, the interface of the droplet is mainly governed by surface tension forces. The two mechanisms that induce natural convection are

gravitational buoyancy force which results in Rayleigh convection and surface tension forces which result in Marangoni/thermocapillary convection. Gravitational buoyancy is a body force that scales with L^3 and thus the induced fluid motion is global, and the resulting temperature field is more homogeneously mixed. Surface tension a surface force that scales with L^2 and thus the induced fluid motion and thermal stratification is intensified at the free surface.

The authors are aware of the study conducted by David *et al.* [26] that attempted to measure micron-sized droplets dispensed from the top with a thermocouple, however, in that study, the temperature measurements were mainly concentrated at the triple contact line and the apex of the droplet. In the method presented in Chapter 3, we are able, due to our exquisite control of the thermocouple, to measure more points along the droplet liquid-gas interface, in the normal to the droplet interface direction and near the contact line of a superhydrophobic surface with droplets evaporating in the steady-state, i.e., constant size, curvature, radius and contact angle, opposed to the work of David *et al.* where the volume of the droplet changes in time.

Chapter 3. Development of Novel Piezo-Driven Experimental Evaporation Setup

Gas-phase temperature measurements are not only key to understanding heat and mass transfer mechanisms governing heterogeneous evaporation of droplets, but they also form the basis for energy transfer pathway quantification which underpins numerous, and sometimes conflicting, theoretical frameworks describing the evaporation of droplets. In this chapter, we experimentally characterize the gas-phase temperature distribution surrounding evaporating water microdroplets (<1 mm). We develop a unique methodology to study steady droplet evaporation via the generation of finely controlled liquid-vapor interfaces coupled to a micrometer-controlled positioning micro-thermocouple. We use our technique to probe the effect of droplet size, surface temperature, apparent contact angle and substrate thermal conductivity on water droplet evaporation dynamics. Our experiments represent the primary validation of century-old theoretical and numerical models used to fundamentally understand the evaporation dynamics of submillimeter droplets, demonstrating experimentally the energy transfer breakdown during heterogeneous droplet evaporation. The method presented here is capable of characterizing both liquid and gas phase temperature, heat, and mass transfer behavior that can be used to study a broad set of materials and interfaces such as flat, structured, and chemically functionalized interfaces in a nonintrusive and non-destructive manner.

3.1 Droplet Generation

In order to form the sessile water droplets of different sizes, a micro-goniometer (MCA-3, Kyowa Interface Science) was utilized. The micro-goniometer consists of a 3D piezoelectric dispenser (that can move in the X–Y–Z direction) with voltage and frequency control and a stage that can be also moved spatially in 3D as shown in Figure 1. The stage is connected to a chiller which enabled us to heat the stage to the desired temperature with an uncertainty of $\pm 1^{\circ}\text{C}$. Since the surfaces of interest are placed on top of the stage, the surfaces attain the desired temperature through conduction. To verify the temperature of the surface, an E-type thermocouple (Chromega®-Alomega® CHCO-002, Omega, USA) having an uncertainty of $\pm 1^{\circ}\text{C}$ was attached on top of the sample (Fig. 1a, b).

To probe the gas-phase temperature distribution, we use a second E-type thermocouple (Chromega®CHCO-002, Omega, USA) having a diameter of $50 \pm 13\ \mu\text{m}$. Prior to utilization, the thermocouple was calibrated in a temperature-controlled bath having a class AA resistance temperature device (RDT), providing a calibrated uncertainty to the thermocouple of $\pm (0.1 + 0.0017*|T|)$ where $|T|$ denotes temperature in units of Celsius. Having the ability to move the stage in three dimensions enabled us to deposit the microdroplet at different positions on the surface.

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Once the desired surface temperature was attained, the piezoelectric dispenser was used to form steady microdroplets on the sample (Fig.1a, b). Microdroplets of $9\ \mu\text{m}$ in radius accrue to form a larger sessile droplet. Once the rate of evaporation is balanced by the rate of liquid addition, the sessile droplet ceases to grow, and a droplet having constant size is attained.

Therefore, control of the droplet size (Fig. 1c) was attained by changing the dispensing frequency of the piezoelectric dispenser. The larger the deposition frequency, the larger the droplet surface area needed to match the dispensing frequency with the evaporation rate.

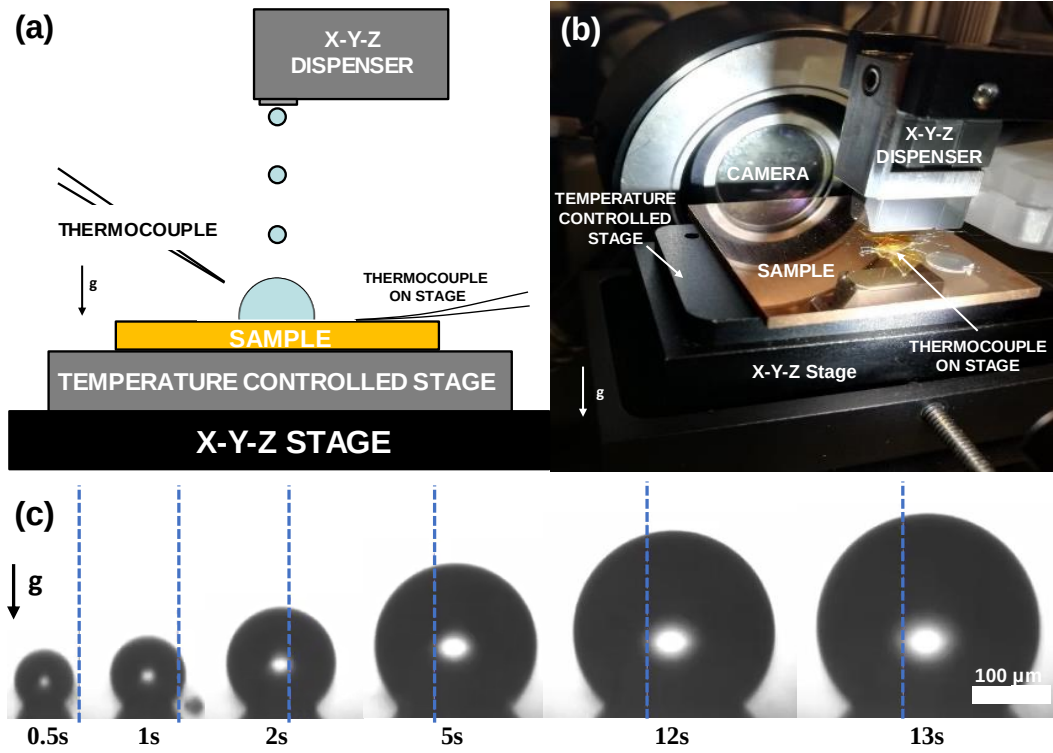


Figure 1. Micro-goniometer setup

(a) Schematic. (b) photograph of the micro-goniometer setup with micro-thermocouple attached to the surface. (c) Time-lapse optical microscopy images of water droplet growth with times labeled on the bottom of each image. The blue dotted vertical line represents a fixed spatial reference. Time zero represents when the dispenser was turned on.

Once we attained the sessile droplet of the desired size, the injection rate was modulated to ensure that the droplet shape and size are constant. The droplet was observed with a camera illuminated with a light-emitting diode (LED, AITECSYSTEM, TSPA22x8) for two minutes after reaching a steady-state in order to ensure a negligible change in droplet size occurs. For additional details related to the droplet generation and vibrational considerations due to the microdroplet impact, see Ref. 68.[68]

3.2 Experiment

We studied the steady-state evaporation of sessile water droplets having different radii (R) residing on functional surfaces having different affinities to water, substrate thermal conductivities, and substrate temperatures by making use of a piezo-driven droplet generation mechanism. Experiments consisted of generating small microdroplets having $R = 140 \pm 20 \mu\text{m}$ (Fig. 2a), $220 \pm 20 \mu\text{m}$ (Fig. 2b), and $295 \pm 20 \mu\text{m}$ (Fig. 2c). Following droplet deposition, a micro thermocouple of diameter $50 \pm 13 \mu\text{m}$ was used to measure the gas temperature around the droplet at different locations (Fig. 2d). During our experiments, caution was exercised to ensure that the deposited microdroplet was placed near (within 1 mm distance) of an additional thermocouple attached to the substrate surface. By doing so, we ensure that the temperature of the surface is similar to that of the surface on which the sessile droplet resides on. The experiments were conducted for four different substrate temperatures: $T_s = 20 \pm 1^\circ\text{C}$, $30 \pm 1^\circ\text{C}$, $40 \pm 1^\circ\text{C}$ and ambient temperature ($25.5 \pm 2.5^\circ\text{C}$) and relative humidity ($44.5 \pm 3.5\%$).

To obtain a high fidelity understanding of the gas dynamics, we measured the temperature around the sessile droplet at 14 different positions (Fig. 2d). The 14 positions are characterized by 5 radial lines normal to the droplet liquid-gas interface at angles $\approx 13^\circ, 27^\circ, 40^\circ, 52^\circ, 76^\circ \pm 4^\circ$ from the droplet base center and 3 different positions from the liquid-gas interface. The distance between the thermocouple and the liquid-vapor interface varied with the wettability of the substrate, the size of the droplet, and the extent to which the thermocouple could get close to the interface. Note that distances reported in Fig. 2(d) are for illustration and not to scale. On some occasions, the thermocouple could not be placed in the vicinity of the droplet without altering the droplet shape. The thermocouple measurement locations (intersections) are labeled alphabetically, where points

lying on the same radial line are denoted by the same letter. By using this arrangement, we were able to experimentally observe: 1) how the temperature varies along the droplet liquid-gas interface, 2) how the temperature varies at different distances away from the droplet, and 3) how the temperature gradient changes for different functional surfaces or surface temperatures, which to date have been mainly resolved via computational methods or theoretical approaches[17,60,69].

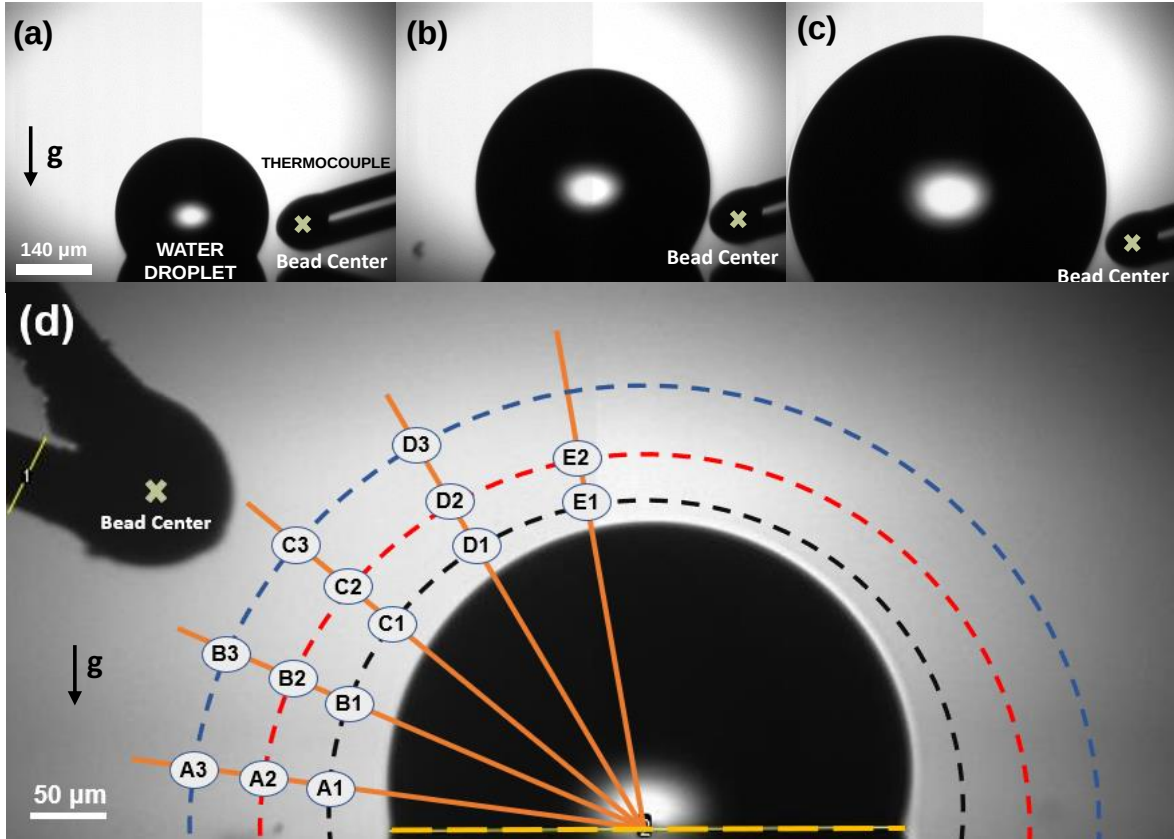


Figure 2. Steady-state evaporation of sessile

Different droplet sizes on a hydrophobic copper (Cu) sample investigated with the thermocouple (green cross marked as the bead center) in the vicinity of (a) small ($R \cong 140 \mu\text{m}$), (b) medium ($R \cong 220 \mu\text{m}$) and (c) large droplets ($R \cong 295 \mu\text{m}$). (d) Points where gas temperatures were measured at angles $\approx 13^\circ$ (A), 27° (B), 40° (C), 52° (D), and 76° (E). Uncertainty of the angular locations was $\pm 4^\circ$. Positions marked in (d) are for illustration purposes and are not to scale.

Given the axisymmetric nature of the generated droplets (Fig. 2), we measured the temperatures on one half of the droplet only. For every measurement point, the thermocouple was allowed to stabilize for 25 seconds. Subsequently, the average temperature, minima, and maxima for that

point were recorded as a function of time at an acquisition rate of 1 Hz before moving to the next point of interest. The acquisition software used to take measurements was DAQami™ v4.2.1.

3.3 Gas Phase Temperature Mapping

The steady size of the sessile water droplet generated via microgoniometry allowed us to probe the gas temperature distribution during evaporation by approaching the microscale thermocouple in the spatial vicinity of the liquid-vapor interface with high spatial and temporal control. Control of the thermocouple location was done using a custom-built X-Y-Z stage with a micro-thermocouple attached allowing for a spatial resolution of $\pm 1 \mu\text{m}$. The temporal resolution of the measurement technique was benchmarked by characterizing the thermocouple bead Biot number Bi_D , where D refers to the diameter of the thermocouple sphere. Here, we assume a spherical metallic chromium (Cr) bead having $D = 50 \mu\text{m}$. Estimates revealed that $\text{Bi}_D = hD/3k_{tc} \approx 0.00013 \ll 0.1$, where $h \approx 800 \text{ W}/(\text{m}^2 \cdot \text{K})$ represents the equivalent convection heat transfer coefficient surrounding the thermocouple bead, and $k_{tc} \approx 100 \text{ W}/(\text{m} \cdot \text{K})$ represents the thermal conductivity of the bead material assuming Cr as the main material (Chromega®CHCO-002, Omega, USA). The estimate of h was done assuming a Nusselt number of $\text{Nu}_D = hD/k_g = 2$, where $k_g \approx 0.02 \text{ W}/(\text{m}^2 \cdot \text{K})$ is the thermal conductivity of the gas, due to the small Rayleigh number characteristic of the thermocouple sphere[70] ($\text{Ra}_D = g\omega(T_s - T_\infty)D^3/\xi\nu^2$), representing steady conduction to and from the surrounding fluid. Here, ω is the air coefficient of thermal expansion, ν is the air kinematic viscosity, ξ is the air thermal diffusivity, T_∞ is the ambient temperature, and g is the gravitational constant. We note here that even for larger Nu_D , conduction to and from the surrounding fluid governs the heat transfer. Calculating the lumped capacitance

transient timescale (τ), we obtain a temperature measurement temporal resolution of $\tau \approx 30$ ms using $\tau = \rho c \forall / h A$ where ρ is the density of the thermocouple bead, c is the specific heat capacity, $\forall$ is the bead volume, and A is the bead surface area.

To characterize the effects of natural convection induced flows stemming from the heated horizontal substrate and its effect on the thermocouple bead, we calculated the Rayleigh number of the heated surface (Ra_L). Considering the case with the highest sample surface temperature ($T_s = 40^\circ\text{C}$), the Rayleigh number was calculated as $Ra_L = Gr_L Pr$, where Gr_L and Pr is the Grashof number and air Prandtl numbers, respectively. The product can be expressed as $Ra_L = g\beta(T_s - T_\infty)L^3/\alpha\nu^2$, where L is the horizontal sample characteristic length scale. Given that all substrates had square dimensions with side lengths of approximately $L' = 25$ mm, the characteristic length was calculated as $L' = A_s/P_s = L^2/4L = L/4$, where A_s and P_s represent the surface area and perimeter of the sample, respectively. The Rayleigh number was found to be $Ra_L = 170$. Since $Ra_L < Ra_{\text{crit}} = 2000$, natural convection induced fluid motion in the gas surrounding the droplet and thermocouple could be neglected in our experiments.

To characterize the effect of radiation from the heated sample to the thermocouple, we calculated the blackbody radiation from the sample at the highest surface temperatures considered ($T_s = 40^\circ\text{C}$) as $Q_{\text{rad}} = \sigma_k A_s F_{s-\text{tc}}(T_s^4 - T_{\text{tc}}^4)$, where $\sigma = 5.68 \times 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$ is the Stefan-Boltzmann constant, $F_{s-\text{tc}}$ is the view factor between the surface (s) and thermocouple (tc), and T_{tc} is the thermocouple temperature. Calculating the radiative heat transfer for a thermocouple position closest to the surface ($F_{\text{tc}-s} \approx 6 \times 10^{-6}$), we obtain $Q_{\text{rad}} \approx 100$ nW.

To characterize the effects of heat conduction away from the thermocouple due to the fin effect, we assume the length scale of the thermocouple wire leads to be reasonably larger compared to the bead dimension. The heat loss can then be calculated as: $Q_{\text{fin}} = 0.5(T_{\text{tc}} -$

$T_\infty)(h\pi^2 D_{tc}^3 k_{tc})^{0.5}$, where D_{tc} is the diameter of the thermocouple electrical leads leading to the bead, and $h \approx 80 \text{ W}/(\text{m}^2 \cdot \text{K})$ is the heat transfer coefficient of air. The estimate of h was done using a Nusselt number of $\text{Nu}_D = hD_{tc}/k_g = 0.2$, representing steady conduction of an infinite length cylinder to a semi-infinite medium (air).

The heat transfer mechanism governing the temperature equilibration of the thermocouple is governed by conduction from the surrounding air (*i.e.* $\text{Ra}_D < 2000$), hence the heat transfer magnitude is approximated as $Q = hA_{tc}(T_{tc} - T_\infty)$, where A_{tc} is the bead surface area ($A_{tc} = 4\pi R^2$) and $h \approx 800 \text{ W}/(\text{m}^2 \cdot \text{K})$ is the equivalent heat transfer coefficient of air as estimated above. Assuming a characteristic temperature difference of 5°C , we obtain $Q \approx 70 \mu\text{W}$.

Examining the different modes of heat transfer by analysis of T_{tc} as an unknown and solving the energy balance on the thermocouple, we obtained a value of T_{tc} approaching that of T_∞ . The parasitic effects account for approximately $\pm 5\%$ of the maximum temperature deviation for the conditions considered here. Consequently, the temperature measured by the thermocouple can be considered to represent the local gas temperature.

3.4 Surface Fabrication

Four different surfaces were fabricated for our experiments. Table 1 outlines all structural and wetting characteristics of our samples. The selection of surface was done to have control of both droplet wetting morphology, and substrate thermal conductivity, as both have been shown to govern the dynamics of water droplets evaporation. Borosilicate glass (3313, Thermo Scientific) and multipurpose polished copper (Cu, 110 Copper, McMaster-Carr) substrates were chosen due to their drastically different thermal conductivities: $k_{\text{glass}} = 1.2 \text{ W}/(\text{m} \cdot \text{K})$ and $k_{\text{Cu}} = 395 \text{ W}/(\text{m} \cdot \text{K})$,

respectively. To vary the apparent contact angle, the glass was either coated with a rough hydrophobic commercial coating (G-65 Glaco mirror court zero, soft99, Glaco) or functionalized directly with fluorinated self-assembled monolayer chemistry. To facilitate Glaco coating, the glass slide was first cleaned in an air plasma cleaner (Harrick Plasma PDC-001-HP) to activate surface radicals for good adhesion. The slide was then sprayed with the Glaco coating and left for 24 hours to ensure the Glaco was drying well and adhered to the surface. Fig. 3(a–c) shows the topographical characterization of the Glaco surface on a glass slide. The Glaco coated glass sample was able to entrap air molecules that aid in providing the superhydrophobic behavior desired [71].

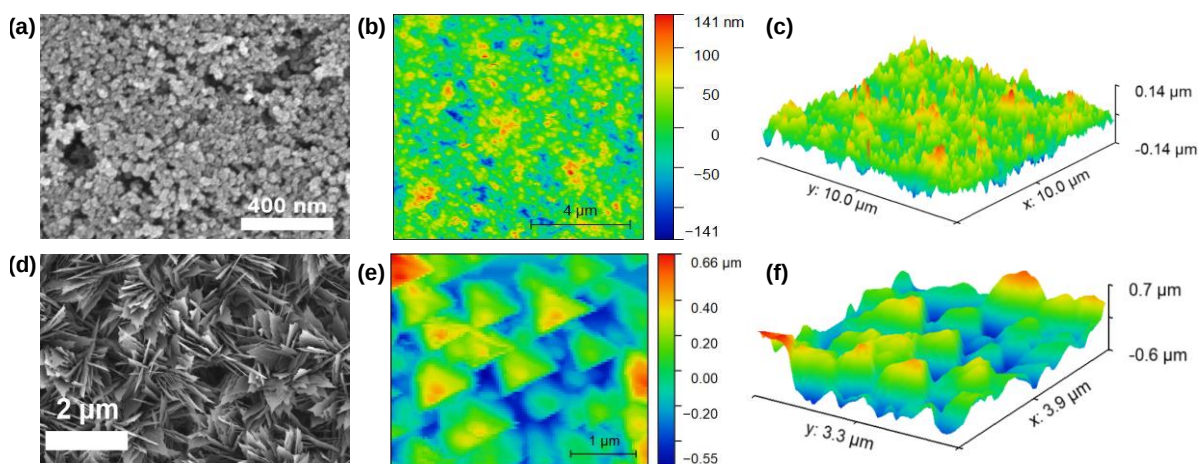


Figure 3. Characterization of Glaco coated glass

Images of glass coated with Glaco showing: (a) scanning electron microscopy (SEM) (b) top view atomic force microscopy (AFM) and (c) 3D isometric view AFM images. Images of copper-oxide nanostructures functionalized with hydrophobic HTMS showing: (d) top view SEM, (e) top view AFM and (f) 3D isometric view AFM images.

To fabricate hydrophobic Glass samples, a glass slide was first cleaned with air plasma, followed by careful placement in a large dish along with a solution of 0.95 mL Toluene (CAS # 108-88-3) and 0.05 mL of Heptadecafluoro (tetrahydrodecyl) Trimethoxysilane (HTMS, CAS # 83048-65-1, Sigma Aldrich) mixed in a small beaker[72,73]. The dish was sealed with aluminum

foil and placed in an air furnace (Lindberg Blue M) at 85°C for 3 hours. The furnace was then turned off and left to cool for one hour.

The same procedure used for functionalizing glass was also used to produce hydrophobic Cu. A clean Cu sample (110 Copper, McMaster-Carr) was degreased by washing with acetone (CAS # 67-64-1, Sigma Aldrich) followed by isopropyl alcohol (IPA, CAS # 67-63-0, Sigma Aldrich). The clean Cu sample was subsequently rinsed with water and washed again with IPA prior to drying with a clean stream of nitrogen. The degreased Cu sample was functionalized with HTMS in the same manner as described above for glass.

To fabricate superhydrophobic Cu substrates, copper oxide nanostructures functionalized with HTMS were produced[74–77]. We first carefully weigh 3.75 g of sodium chlorite (CAS # 7758-19-2, Sigma Aldrich), 5 g of sodium hydroxide (CAS # 1310-73-2, Sigma Aldrich), and 10 g of sodium phosphate (CAS # 7601-54-9, Sigma Aldrich) and add them to 100 mL of deionized water (DI, CAS # 7732-18-5, Sigma Aldrich) in a 200 mL glass beaker. We then used a magnetic stir bar to mix the solution. The solution was heated on a hot plate to a temperature of $95 \pm 5^\circ\text{C}$. A cleaned copper sample was first degreased using the same method outlined in the fabrication description for hydrophobic copper, and placed into the heated solution for ≈ 5 minutes to obtain CuO nanostructures. The oxidized Cu sample was then carefully removed from the beaker with tweezers, rinsed with DI water, and dried with clean nitrogen. Finally, the nanostructured Cu sample was functionalized with HTMS using the same procedure outlined for glass and Cu. The CuO coated with HTMS substrate had knife-like structures (Fig. 3d-f) that were able to entrap air similar to the Glaco coated glass (Fig. 3a-c) rendering it superhydrophobic (Table 1).

Table 1. Surface characterization for fabricated samples.

Surface characterization for all fabricated samples. The maximum standard deviation over 5 separate contact angle measurements on different locations on the same sample was $\pm 6^\circ$ for the apparent advancing state, and $\pm 10^\circ$ for the apparent receding state

Parameter	Bare		Functionalized or Structured			
	Glass	Cu	Glass with Glaco	Glass with HTMS	CuO with HTMS	Cu with HTMS
Thermal conductivity, k [W/(m·K)]	1.2	395	1.2	1.2	395	395
Root Mean Square (RMS) roughness [nm]	5	25	36	2	212	30
Coating thickness [nm]	-	-	< 100	≈ 1	≈ 1000	≈ 1
Apparent advancing contact angle, θ_{adv}^{app} [deg]	77	84	164	113	171	119
Apparent receding contact angle, θ_r^{app} [deg]	42	19	161	89	163	74
Solid fraction, ϕ	≈ 1	≈ 1	$\approx 0.05^*$ [71]	≈ 1	$\approx 0.04^*$ [78]	≈ 1
Roughness, r	≈ 1	≈ 1	≈ 4 [71]	≈ 1	≈ 10 [76]	≈ 1

* The solid fraction was calculated using the Cassie-Baxter equation and the apparent advancing contact angle.

Chapter 4. Effect of Functional Surface and Droplet Parameters on Evaporation Dynamics

4.1 Substrate thermal conductivity

Measurement results of the gas temperature distribution for an exemplary evaporating droplet having a fixed radius $R \approx 140 \mu\text{m}$ deposited on hydrophobic glass, structured superhydrophobic glass, hydrophobic copper (Cu), and structured superhydrophobic copper oxide (CuO) are shown in Figure 4 (see Methods subsection for sample fabrication as well as characterization details are shown in Table 1). The white areas surrounding the periphery of the sessile droplet (outlined in black near the ordinate) represents regions where the temperature could not be measured due to the size of the thermocouple bead.

In order to obtain the temperature contours, the 14 measured data points are input into the plotting software Origin and are enabled to linearize the data between measured points. This assumption relies on the fact that the measured temperatures are spatially close enough such that the linear approximation between them is appropriate. The method can then be validated simply by observing the contours and noting that discontinuities in the color maps do not exist, indicating that the granularity is high enough. To handle the boundaries of the plot, we set the far field temperature to the measured ambient temperature as obtained from the thermocouple measurements far ($> 1 \text{ cm}$) from the droplet, and use the same linear approximation to link these temperatures to the inner measured temperatures at the boundary.

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To check if the measurement resolution changes the result, additional experiments were conducted where we doubled the number of measured temperature points (28) by measuring local temperatures in between the initially reported measurements. Specifically, at different angular positions along lines 1, 2, and 3 using a droplet placed on the hydrophobic surface. Using the data processing method outlined above, the temperature contour was found to deviate by $< 1\%$ in terms of aggregated numerical difference from point-to-point between the two generated temperature maps. This low difference indicates that the approximation of linear temperature behavior between points is appropriate given the very small length scales between point measurements.

Figure 4 demonstrates a rather uniform temperature gradient surrounding evaporating water droplets in the radial direction for small-sized droplets. At the droplet liquid-vapor interface, the temperature was lowest due to evaporative cooling, gradually increasing as the distance from the liquid-vapor interface increases until it reaches the ambient temperature. It is important to note that larger droplets exhibited a bigger temperature mal-distribution having non-uniform temperatures along the interface. Past work has demonstrated that the thermal conductivity of the substrate has a direct effect on evaporation[79–81]. In order to gain the necessary heat for evaporation, the droplet molecules at the interface can either gain heat from the surface it resides on, or from the gas surrounding it. Since Cu has a higher thermal conductivity when compared to glass ($k_{\text{Cu}}/k_{\text{glass}} \approx 329$), heat is easily transferred through the substrate. Consequently, the heat required to evaporate sessile droplets on metals is mainly absorbed from the substrate instead of the gas. With that in mind, a portion of the heat required to initiate evaporation is also absorbed from the surroundings.

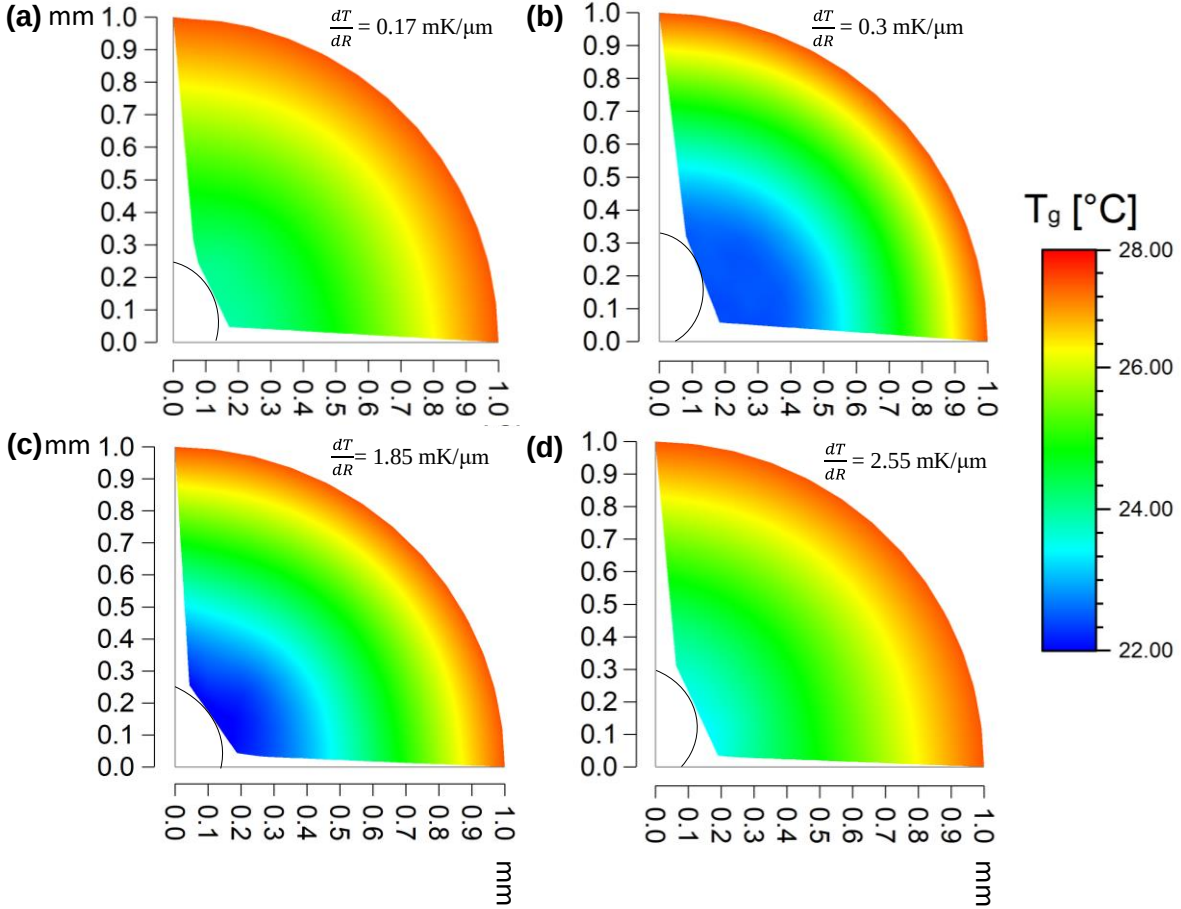


Figure 4. Temperature contours for microscale droplets

Temperature contours for microscale droplets with a $R \approx 140 \mu\text{m}$, and ambient temperatures of $27 \pm 1^\circ\text{C}$ on (a) hydrophobic Cu, (b) superhydrophobic CuO, (c) hydrophobic glass, (d) superhydrophobic glass. The x and y-axis scales are in units of mm. For surface fabrication and characterization details, see Methods. Each water droplet is depicted as the black border region near the ordinate with an apparent advancing contact angle drawn to scale. For more information regarding the temperature gradient quantification as well as plot generation methodology, see Appendix A.

Consequently, the temperature mal-distribution around the droplet is less pronounced for the Cu samples. When comparing Cu with glass substrates, the majority of the heat required for evaporation on the glass substrates is absorbed from the gas. This can be deduced from the higher heat flow from the ambient on glass samples which is quantified by the average gas-phase temperature gradient at the liquid/vapor interface. For example, for small-sized droplets residing on Cu samples, the temperature gradient $[dT/dR]_{r \approx R} = 0.3 \text{ mK}/\mu\text{m}$ for superhydrophobic CuO

and 0.168 mK/ μm for hydrophobic Cu. However, for small sized droplets residing on glass samples, the temperature gradient was $[dT/dR]_{r \approx R} = 2.55 \text{ mK}/\mu\text{m}$ for superhydrophobic glass and 1.85 mK/ μm for hydrophobic glass. Comparing the differences between droplets in hydrophobic and superhydrophobic states, a clearer picture emerges depicting the physical dependence on the interfacial area on the evaporation rate. For superhydrophobic state droplets, the gas phase temperature gradient was higher, almost two-fold when compared to the hydrophobic droplet on the metal surface as evidenced from the lower temperature near the liquid-gas interface when comparing Figs. 4a and b. This is simply due to the larger reliance on gas side heat transfer owing to the higher constriction thermal resistance at the base of the droplet and the larger heat transfer resistance path across the evaporating droplet[78]. The same trend is also observed for the glass substrates when comparing Figs. 4c and d.

To analyze the droplet size dependence and substrate temperature dependence on the gas temperature gradient, Table 15 shows a summary of the micro-droplet dispensing frequency for all experiments conducted function of the droplet size R , the substrate utilized, and its temperature. The droplet dispensing frequency is a direct indicator of the rate of evaporation[68]. The higher thermal conductivity of Cu, when compared to air, allows for a higher rate of heat supply from the substrate to the sessile droplets, therefore inducing a higher rate of evaporation. In fact, it was noted that for most of the droplet sizes across different temperatures, Cu had on average a higher rate of evaporation than glass except at the highest temperature where the evaporation rates approached each other. Here, the subcooling effect by the dispensed micro-droplets is neglected due to the sensible cooling effect being at least 200X smaller than the latent heat during evaporation. The evaporation heat transfer rate (Q_e) can be quantified from the frequency of the dispenser (f) when the droplet reaches a steady-state size, or $Q_e = (4/3)f\pi R_d^3 \rho h_{fg}$, where R_d is

the radius of dispensing droplets ($R_{\text{droplet}} = 9 \mu\text{m}$)[68] ρ and h_{fg} are the liquid density and liquid-vapor latent heat of phase change for water[68].

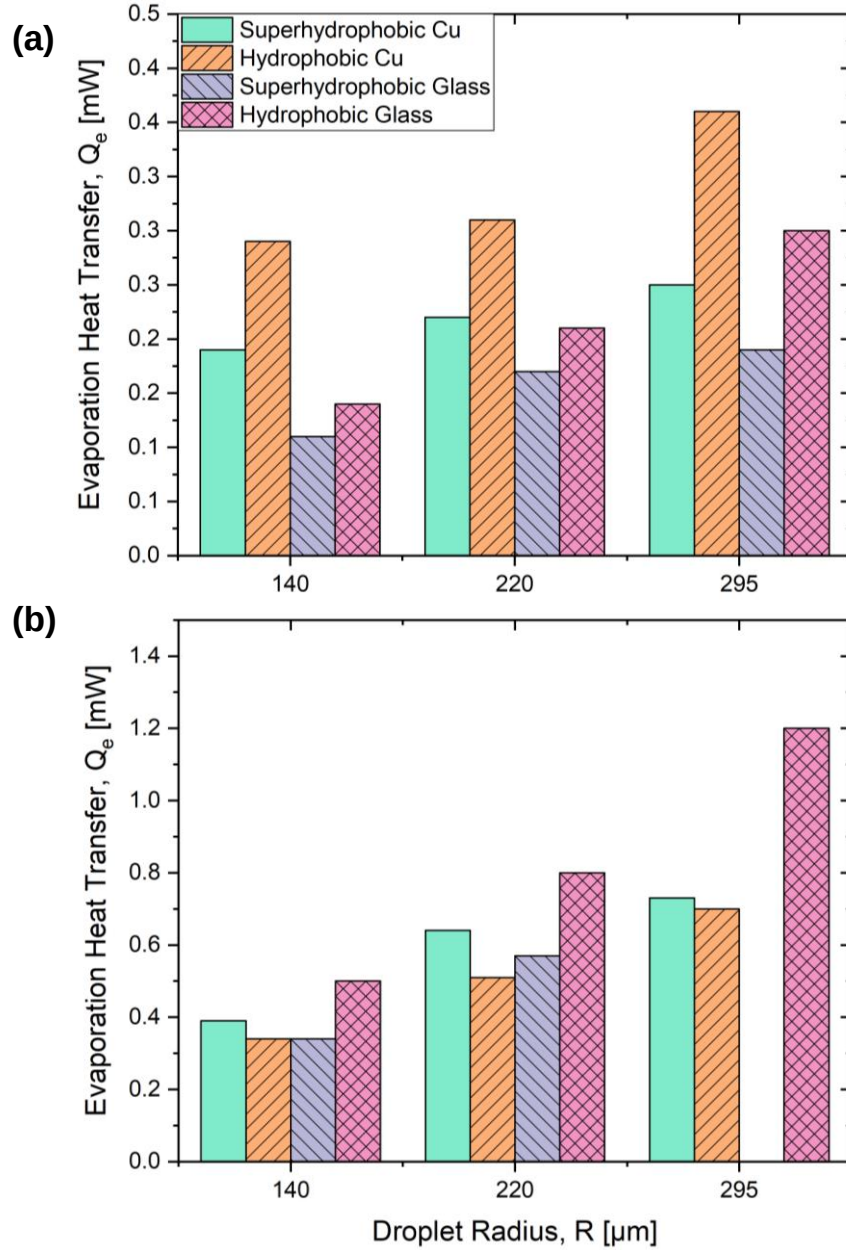


Figure 5. Evaporation heat transfer rate for droplets

Evaporation heat transfer rate for small ($R \cong 140 \mu\text{m}$), medium ($R \cong 220 \mu\text{m}$), and large droplets ($R \cong 295 \mu\text{m}$) at: (a) $T_s = 23^\circ\text{C}$ and (b) $T_s = 40^\circ\text{C}$ substrate temperatures. Legend shown in (a) is valid for results shown in (b). For tabulated results, see Table 15. Data for the superhydrophobic glass sample and $R = 295 \mu\text{m}$ is missing due to the thermocouple attracting the water droplet through triboelectric charging.

For ease of comparison, Figure 5 illustrates the evaporation heat transfer rate at $T_s = 23^\circ\text{C}$ and $T_s = 40^\circ\text{C}$ for the glass and Cu substrates.

For sessile droplets at ambient conditions, f and hence the evaporation rate for Cu samples was 56% higher on average when compared to glass samples. This was due to the lower gas temperature gradients, *i.e.* greater temperatures at the droplet interface with the consequent greater vapor pressure at the liquid-vapor interface responsible for evaporation along with the energy transferred from the solid.

To characterize the energy balance and quantify how much energy is conducted from the surrounding gas versus the substrate, we used the gas-phase temperature measurements and associated temperature gradients near the liquid-vapor interface to calculate the gas phase heat transfer (Q_g) by: $Q_g = -A_{lv}k_g[dT/dR]_{r=R}$, where A_{lv} is the droplet liquid-vapor interfacial area. After computing the gas heat transfer, we were left with the substrate heat transfer (Q_s) from an energy balance: $Q_s = Q_e + Q_g$. Figure 6 shows the breakdown of the energy flow for microdroplets on substrates having $T_s = 23^\circ\text{C}$. For a tabulated summary, see Table 16. For Cu, the substrate heat transfer (Q_s) is higher than that of glass, further quantifying and demonstrating the dominant role played by the substrate in driving evaporation for surfaces having higher thermal conductivity. As for glass substrates, it can be noted how the gas heat transfer (Q_g) is much higher than that of copper substrates. Further, as to be discussed later, the gas heat transfer (Q_g) in both substrates is more pronounced as the droplet radius increases due to the greater evaporation rates. Finally, since the balance of heat transfer is maintained, the total evaporation heat transfer (Q_e) also increases as the droplet size grows. However, the total evaporation heat transfer (Q_e) for Cu substrates is higher than that of glass.

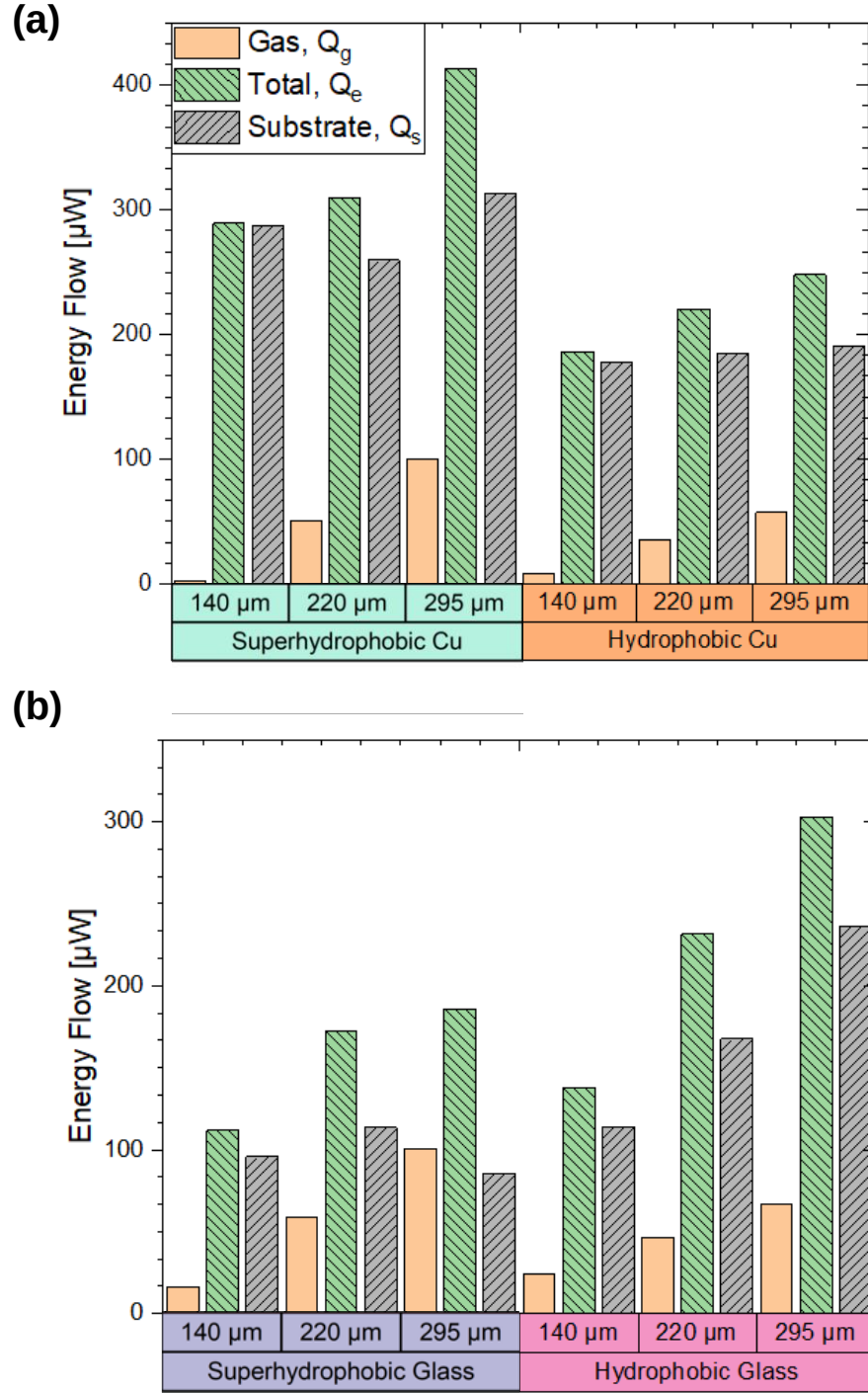


Figure 6. Energy flow breakdown for microdroplets

Energy flow breakdown for microdroplets having small ($R \cong 140 \mu\text{m}$), medium ($R \cong 220 \mu\text{m}$), and large ($R \cong 295 \mu\text{m}$) radii residing on (a) glass substrates with $T_s = 23^\circ\text{C}$, and (b) Cu substrates with $T_s = 23^\circ\text{C}$. Legend shown in (a) is valid for results shown in (b).

4.2 Droplet size

In addition to evaporation dependency on droplet contact angle and substrate material, the droplet size affects phase change evaporation rate and heat transfer significantly. Fig. 7 shows the temperature distribution around small-sized ($R \cong 140 \mu\text{m}$) and large-sized ($R \cong 295 \mu\text{m}$) water droplets evaporating at ambient temperature. For small water droplets on Cu (Fig. 7a), the temperature distribution around the droplet periphery varies minimally. As previously discussed, this is attributed to the fact that most of the heat absorbed for evaporation is taken from the surface instead of the surroundings. However, as we increase the size of the sessile droplet, we begin to see fluctuations in the temperature around the vicinity of the droplet. As shown in Fig. 7b, the temperature values along a given radial line are no longer similar and we start seeing a temperature mal-distribution similar to that of superhydrophobic glass.

To understand this phenomenon, we consider the droplet as a thermal resistor. For small-sized droplets, the thermal resistance to heat flow between the surface and the periphery of the droplet is relatively small, hence heat can favorably transmit from the surface, through the droplet and cause evaporation. However, as the size of the droplet starts to increase the thermal resistance to heat flow starts to increase and is no longer as dominant as that observed in small droplets. Consequently, the ratio of heat absorbed through the surroundings to the heat absorbed through the sample starts to increase (Table 17, Appendix A).

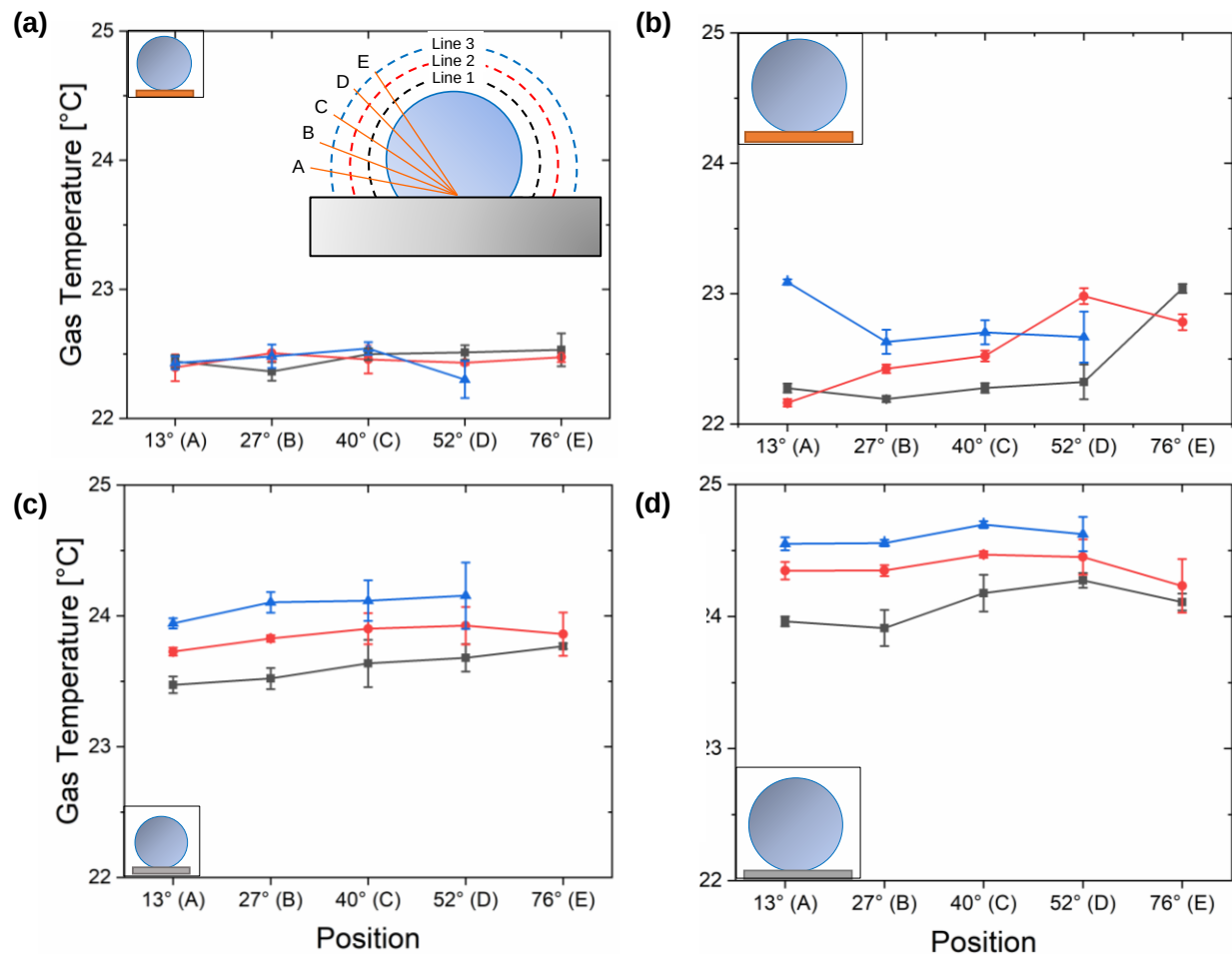


Figure 7. Local gas temperature against radial positions

Local gas temperature at ambient temperature conditions ($25.5 \pm 2.5^\circ\text{C}$) as a function of radial position on superhydrophobic CuO with (a) small ($R \cong 140 \mu\text{m}$) and (b) large ($R \cong 295 \mu\text{m}$) droplets and on superhydrophobic glass with (c) small ($R \cong 140 \mu\text{m}$) and (d) large ($R \cong 295 \mu\text{m}$) droplets, at ambient temperature. For a map of positions, see Figure 2. Insets: schematics of the droplet size and substrate type. The second inset in (a – top right) shows the line position explicitly as demonstrated in Figure 2d. Inset schematics not to scale.

For the glass substrate, small droplets with $R \cong 140 \mu\text{m}$ (Fig. 7c) develop a clear temperature distribution in the radial direction, where positions closest to the droplet periphery exhibit lower temperatures than those further away due to evaporative cooling. As we increase the droplet size to $295 \mu\text{m}$ (Fig. 7d), the temperature distribution becomes more distinct. This is because, in addition to the already existing thermal resistance stemming from the low thermal conductivity of glass, larger water droplets having higher thermal resistance to heat flow create

even higher thermal resistance. This reduces the amount of heat absorbed from the surface and increases the amount of heat absorbed from the gas (Table 18, Appendix A). The greater temperature gradient within the gas phase in the case of droplets evaporating on a glass substrate is evidenced by the larger differences between temperature values when comparing the profiles as we get further away from the droplet in Fig. 7 *i.e.* Line 1 and Line 3.

4.3 Surface Wettability

The evaporation rate of a sessile droplet is a function of the surface wettability. Previous work has demonstrated that surfaces with high wettability (contact angle $< 90^\circ$) show higher rates of evaporation near the triple contact line. This rate of evaporation gradually decreases to a minimum at the top of the droplet [59,60,82,83]. However, for hydrophobic surfaces with low wettability (contact angle $> 90^\circ$) the rate of evaporation near the triple contact line is minimized, with the rate gradually increasing to a maximum at the top of the droplet [59,60,84,85]. Since the rate of evaporation near the contact line for hydrophobic surfaces is minimum, it is expected that the temperature of the air near that region is maximum. Analysis of our experimental measurements (Table 19) does not provide a clear trend of how the profile varies around the periphery of the droplet. However, for hydrophobic glass substrates, the temperature around the periphery was maximum near the triple contact line and minimum near the top. We were not able to confirm the predictions of the numerical results of the previous work[59,60,83,85] due to limitations on the size of the thermocouple used.

The experiments conducted here reveal rich information about the effects of liquid-vapor interfacial curvature and droplet wettability (Tables 20 and 21). For superhydrophobic glass samples, the heat was mainly absorbed from the surroundings and not the substrate. As the size of

the droplet on the superhydrophobic glass increased, the heat transfer required to evaporate the larger droplet increased (Q_e , Table 15). Hence, accompanying the increase in absorbed heat from the gas (Q_g), the heat removed from the surface (Q_s) increases since the surface area of the droplet increases (Table 16). However, when the droplet increases from $R \cong 140 \mu\text{m}$ to $220 \mu\text{m}$, Q_g increases at a faster rate than Q_s . The non-linear balance between the heat transfer pathways for evaporation at the interfaces can be understood by considering how the thermal resistance of conduction through the droplet ψ_c (Table 22), interfacial thermal resistance ψ_i (Table 20), mass diffusion resistance in the gas phase, and spreading thermal resistance at the base of the droplet ψ_s (Table 21), scale with droplet radius (R). Droplet heat spreading from the substrate to the droplet base is a 3D conduction problem that can be analytically modeled by:

$$\psi_s = \frac{1}{4k_s R_b}, \quad (1)$$

where k_s is the thermal conductivity of the substrate and R_b is the radius of the droplet base in contact with the substrate. As can be noted from Table 21, ψ_s decreases as the size of the droplet increases from $R \cong 140 \mu\text{m}$ to $\cong 295 \mu\text{m}$ which is expected since R_b increases. For the same reason, ψ_s further reduces as the water affinity of the substrates increases. Moreover, since k_s of Cu substrates is several orders of magnitude higher than that of glass, ψ_s is several orders lower for the highly conductive metal. Consequently, the resistance to heat transfer is considerably lower for Cu surfaces. This demonstrates why the heat required to cause droplet evaporation on Cu substrates is mainly driven from the surface.

A second pathway for heat transfer is through the gas surrounding the droplet. The thermal resistance at the liquid-vapor interface ψ_i can be estimated using[86,87]:

$$\psi_i = \frac{1}{h_i 2\pi R (1 - \cos \theta_{\text{adv}}^{\text{app}})}, \quad (2)$$

where h_i is the liquid-vapor interfacial heat transfer coefficient, and $\theta_{\text{adv}}^{\text{app}}$ is the apparent advancing contact angle. The interfacial heat transfer coefficient can be computed from the kinetic theory as:

$$h_i = \frac{2\sigma}{2 - \sigma} \frac{1}{\sqrt{2\pi R_g T_s}} \frac{\rho_g h_{fg}^2}{T_s}, \quad (3)$$

where σ the accommodation coefficient (≈ 1), ρ_g is the gas density, and R_g is the specific gas constant. Table 20 summarizes the results obtained for ψ_i by making use of Eqns. (2) and (3). Compared to the substrate spreading resistance (ψ_s), the interfacial resistance (ψ_i) is several orders of magnitude lower and is neglected herein.

The third thermal resistance to be considered is the thermal conduction through the droplet, estimated as[87–90]:

$$\psi_c = \frac{\theta_a^{\text{app}}}{4\pi R k_w \sin \theta_{\text{adv}}^{\text{app}}}, \quad (4)$$

where k_w is the thermal conductivity of water. The two factors dictating the magnitude of ψ_c include the size of the droplet and the apparent advancing contact angle. Therefore, it is expected that surfaces with lower water affinities and greater droplet sizes have a lower thermal resistance ψ_c .

The last resistance to evaporation is the diffusional resistance in the gas phase. In this analysis we consider a well-validated fully-transient Fickian gas diffusion model to compute the mass transfer and evaporation rate[83]:

$$\dot{m} = -\pi C R D (1 - RH) \left(0.27 \theta_a^{\text{app}^2} + 1.3 \right), \quad (5)$$

where $\dot{m}$ is the droplet evaporation rate stemming from purely a gas-phase mass transfer transport analysis, D is the diffusion coefficient of water vapor in air ($\approx 26.3 \text{ mm}^2/\text{s}$), RH is the ambient far

field relative humidity, C is the saturated vapor concentration. In order to derive Eq. (5), the gas-liquid interface was assumed to be at saturation. As the coordinate system moves further from the liquid-vapor interface, the vapor concentration is considered to approach ambient conditions far-field from the droplet. The vapor concentration is assumed to change at a much smaller time scale when compared to the droplet evaporation timescale, hence the evaporation can be considered to be quasi-steady. Although Eq. (5) was derived for fully transient cases in which the contact angle ranges between 0 and $\pi/2$, when comparing the results of the fully transient model[83] with the values obtained experimentally (superhydrophobic Cu), the mass fluxes were $\approx 1.5 \times 10^{-10}$ kg/s and $\approx 1 \times 10^{-10}$ kg/s, respectively, in very close agreement. The ability of Eq. (5) to predict the evaporation rate for the majority of our experiments with good agreement lends credence to the notion that the mass transfer resistance in the gas phase plays an outsized role during evaporation when compared to the other thermal resistances.

The thermal resistance analysis demonstrates the importance of the limiting mass transfer resistance which is widely accepted for the evaporation of sessile water droplets on surfaces. Furthermore, our analysis provides insights into how the thermal resistance of heat conduction through the droplet (ψ_c), as the second most prominent resistance, affects the evaporation mechanism when compared to the interfacial or substrate spreading resistances. The substrate thermal conductivity and droplet size play a prominent role in the spreading thermal resistance (ψ_s), explaining why ψ_s plays an important role in experiments conducted on surfaces having lower thermal conductivities. We note that the mass transfer model used (Eq. 5) captures the effects of droplet size and geometry through R and θ . However, as noted earlier, for low thermal conductivity substrates, where the effect of the thermal resistances at the base of the

droplet is significant, the model is not as effective as it does not capture the relevant mechanisms occurring during the evaporation process.

The thermal resistance analysis demonstrates that for droplets residing on surfaces having poor thermal conductivity, heat absorption through the liquid-vapor interface from the gas may be significant when compared to the heat absorbed from the base of the droplet. Our analysis further cements the reason why we observe a high temperature gradients normal to the liquid-vapor interface for glass substrates. For Cu substrates, however, the substrate plays a less prominent role. Furthermore, we note that in both glass and Cu surfaces, for superhydrophobic state droplets, there is a larger reliance on gas side heat transfer owing to the higher constriction thermal resistance at the base of the droplet. Consequently, the gas phase temperature gradient was higher for superhydrophobic droplets when compared to hydrophobic ones.

4.4 Surface Temperature

One of the important parameters governing the evaporation dynamics of an evaporating sessile droplet is the temperature of the surface. Table 15 shows that with increasing temperatures, the dispensing frequencies keeping the size of the droplets constant must be increased, indicating that the evaporation rate of the sessile droplets increases. Furthermore, our measurements show that the increase in frequency with temperature is more significant for surfaces with higher thermal conductivity such as Cu since most of the heat required for evaporation comes from the surface instead of the surroundings. Moreover, as the temperature of the surface increases beyond ambient temperatures, the sessile droplet should also attain higher temperatures than that of the surrounding ambient air. Consequently, a temperature gradient where the temperatures decrease from the vicinity of the droplet as we move away from the droplet is expected at elevated temperatures,

while a temperature gradient where the temperature increase from the vicinity of the droplet as we move away from the droplet is anticipated at ambient or below ambient temperatures

4.5 Experimental Results Comparison with Theory

In order to compare our temperature measurements with past developed simulation-based models, a comparison between the experimental results and a closed form analytical solution that takes into consideration evaporative cooling effects is conducted. The model[91] is valid for a full range of different droplet apparent contact angles on a flat substrate of constant temperature. This specific model was chosen due to its non-isothermal assumption within the gas phase where change in the saturation vapor concentration as a function of temperature is accounted for as a consequence of interfacial cooling. The model solves for mass transfer in the gas phase using the quasi-steady state Fick's law in which the transient term is neglected, $\nabla^2 C = 0$ [12], where C is the local water vapor concentration. The heat transfer is also assumed to be quasi-steady state and the temperature T within an evaporating droplet is also solved by Laplace equation, $\nabla^2 T = 0$. The distributions for C and T are then solved in the toroidal coordinate system (α and β). The transformation from toroidal coordinates to cylindrical coordinates (r and z) is done by:

$$r = \frac{R \sinh \alpha}{\cosh \alpha - \cos \beta}, \quad (6)$$

$$z = \frac{R \sin \beta}{\cosh \alpha - \cos \beta}, \quad (7)$$

where, R is the droplet radius. The solution to vapor concentration is valid for a range of $0 \leq \alpha \leq \infty$ and $2\pi \leq \beta \leq 3\pi - \theta$ (θ is the droplet contact angle), and is expressed as: (8)

$$\tilde{C}(\alpha, \beta) = \frac{C(\alpha, \beta) - C_\infty}{C_i - C_\infty} = \sqrt{2 \cosh \alpha - 2 \cos \beta} \int_0^\infty E_c(\tau) P_{i\tau-\frac{1}{2}}(\cosh \alpha) \cosh((2\pi - \beta)\tau) d\tau,$$

where, C_i is the saturated vapor concentration of liquid at droplet liquid-vapor interface temperature, C_∞ is the vapor concentration far from the droplet at ambient air temperature, τ is a dummy variable for integration, $P_{i\tau-\frac{1}{2}}(\cosh \alpha)$ is the first-kind Legendre function of the complex half-integral degree and the argument of the hyperbolic cosine function, and

$$E_c = \frac{\cosh(\theta\tau)}{\cosh(\pi\tau) \cosh((\theta - \pi)\tau)} \frac{\tau F \coth(\theta\tau) - \frac{1}{3} \frac{dF}{d\theta}}{\tau F \coth(\theta\tau) - \frac{1}{3} \frac{dF}{d\theta} - E_0(\tau F \tanh((\theta - \pi)\tau) - \frac{1}{3} \frac{dF}{d\theta})}. \quad (9)$$

Temperature distribution inside the droplet is solved and is expressed as:

$$\tilde{T}(\alpha, \beta) = \frac{T(\alpha, \beta) - T_{\text{sub}}}{T_{\text{sub}} - T_\infty} = \sqrt{2 \cosh \alpha - 2} \int_0^\infty E_T(\tau) P_{i\tau-\frac{1}{2}}(\cosh \alpha) \sinh((\pi - \beta)\tau) d\tau, \quad (10)$$

where, T_{sub} is the substrate temperature, T_∞ is the ambient air temperature, and

$$E_T = \frac{E_0 \cosh(\theta\tau)}{\cosh(\pi\tau) \sinh(\theta\tau)} \frac{\tau F \tanh((\theta - \pi)\tau) - \frac{1}{3} \frac{dF}{d\theta}}{\tau F \coth(\theta\tau) - \frac{1}{3} \frac{dF}{d\theta} - E_0(\tau F \tanh((\theta - \pi)\tau) - \frac{1}{3} \frac{dF}{d\theta})}. \quad (11)$$

The term E_0 in Eq. (9) and (11) equals to 0.11 for water at room temperature and pressure. The functions F and $dF/d\theta$ are defined as:

$$F = \frac{\sinh(\theta\tau)}{\sinh(\pi\tau) \sin \theta}, \quad (12)$$

$$\frac{dF}{d\theta} = \frac{\tau \cosh(\theta\tau) \sin(\theta) - \sinh(\theta\tau) \cos \theta}{\sinh(\pi\tau) \sin^2 \theta}. \quad (13)$$

For Eq. (8) to be uniquely solvable, C_i must be determined first. Thus, the temperature at the droplet interface is initially solved using the temperature distribution solution (Eq. 10) and is converted to

vapor concentration using the ideal gas law. Then, vapor concentration and therefore temperature is solved locally and compared to experimental data. Experimentally measured temperatures at 14 spatially varying points are compared with the model. Except for one point, the differences between experimental and model values were less than 4%, with 60% of the data having less than a 2% difference when the substrate temperature is 30°C. When the substrate temperature is 23°C, the maximum difference was less than 6%. See Tables 23 and 24 in Appendix A for quantifiable experimental and model results.

Chapter 5. Discussion and Conclusion on Gas Phase Temperature Mapping of Evaporating Water Micro-droplets

5.1 Discussion

In this study, we develop a methodology to obtain gas temperature mapping during the liquid-vapor phase change phenomena. We probe the contour of evaporating droplet peripheries with exquisite spatial and temporal control in order to develop an understanding of the energy transfer dynamics governing heterogeneous droplet evaporation. Our experimental measurements represent the primary validation of numerous past theoretical and numerical models developed to gain a fundamental understanding of droplet evaporation on a variety of surfaces. Our method enables us to decouple the different modes of heat transfer from the substrate and gas-phases for steady evaporating droplets. In addition to validating past theory, our method enables the study of microscale droplet phenomena where the majority of heat transfer is known to occur during evaporation, condensation[92,93], boiling, and freezing[72]. The high spatial resolution of the micro-thermocouple enables the careful and accurate study of highly curved liquid-vapor interfaces for a plethora of applications. In the future, the extension of the developed method to study other phase-change phenomena where gas-side heat and mass transfer dominates the transport mechanisms (*i.e.* frosting[94]) is prudent. Furthermore, even more interesting would be the integration of the developed technique with pool-boiling studies where liquid-phase thermal gradients near the three-phase contact line facilitate thin film evaporation and microlayer and microlayer dynamics. While this study mainly focuses on surfaces with low wettability, this study

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could be extended for other surfaces exhibiting low contact angles. A challenge would be to restrict the spreading of the sessile droplets in order to attain a constant shape profile. This could be achieved via pinning the droplet using pillars etched into the surface[95].

Although the developed method is a powerful platform for the study of liquid-vapor phase change phenomena, the method has certain experimental limitations. In some circumstances, especially for the glass surface, we faced the problem of having the thermocouple attracting the sessile water droplet (Fig. 67). The attraction was attributed to the triboelectric charging of insulating surfaces. Triboelectric charging occurs when friction between insulating surfaces induces stable electric charges that contribute to electrostatic components which induce surface interactions[96–98]. In our case, the thermocouple could have acquired static charge from contact with other dissimilar materials. The charge on the thermocouple would attract the corresponding counter-ions in the water droplet distorting the shape of the water droplet as shown in Figure 67. In other cases, the water droplet would de-pin from the surface and attach to the thermocouple (Fig. 67b), making the measurement of the gas temperature near the droplet impossible. This effect was more pronounced for larger sized droplets where a larger availability for counter ions exists.

In the future, it would be interesting to focus on miniaturization of the thermocouple size. Available commercial E-type thermocouples were constrained to the 50 μm diameter, making it difficult to measure some temperature points near the triple-phase contact line (especially point A1, Fig. 2d). Future work is needed to identify smaller-scale thermocouples for use in the developed method[25,99,100], which may enable probing of evaporation dynamics near the three-phase contact line where the majority of evaporation occurs for certain conditions. In addition to greater spatial resolution, smaller thermocouples enable higher temporal resolution due to the lower Biot number of the thermocouple bead, and faster response time. For example, assuming

identical materials, reduction of the thermocouple diameter to $D = 20\text{ }\mu\text{m}$ results in a time constant reduction to $\tau = 0.02\text{ ms}$. Smaller time constants enable the study of highly transient processes that may govern evaporation in conditions devoid of non-condensable gases (NCGs) as well as other phase-change processes. Furthermore, it would be interesting to utilize the experimental method developed here to investigate the transport mechanisms of droplet evaporation whereby the contribution of the gas-phase transport mechanisms such as gas-phase convection and vapor diffusion can be qualitatively examined.

Although studied here for droplet evaporation, the developed method can and should be extended to other liquid-vapor phase change phenomena where gas-phase dynamics become important. Two promising future directions that need to be explored are the evaporation of pure fluids in saturated conditions where NCG effects are negligible and mass transfer in the gas side is not limiting. Furthermore, significant gas-phase dynamics play an important role in dropwise condensation in the presence of NCGs. The developed methodology can enable the probing of mass-transfer dynamics present in the counter-diffusion problem present during condensation with NCGs present. Indeed, recent studies have developed methodologies to enable the formation of non-axisymmetric droplets for enhanced evaporation[9,101,102], bringing forth the need for experimental quantification of evaporation physics in more complex systems than studied here.

The experimental studies conducted here focused on water evaporation on non-wetting surfaces. Future work is needed to utilize the developed technique to study evaporation dynamics of other working fluids of interest which may have non-polar molecular interactions or metallic bonding (low-temperature liquid metals) and hence evaporation physics. The studies conducted here are meant to benchmark the method and demonstrate the ability to quantifiably decouple the energy flows from the gas phase and substrate into evaporating droplets.

5.2 Conclusion

This work develops an experimental measurement platform for the study of local gas-phase temperature distributions near highly curved liquid-vapor interfaces during droplet evaporation. Unlike preceding investigations, our work experimentally characterizes the local gas temperature adjacent to microdroplets with exquisite spatial and temporal control. We develop a quantifiable understanding of heterogeneous droplet evaporation dynamics and how they relate to surface thermophysical properties and wetting characteristics, in addition to temperature and droplet size. By studying evaporation on hydrophobic and superhydrophobic glass and Cu surfaces across four different surface temperatures and three droplet sizes, we demonstrate high thermal conductivity surfaces (Cu) with small droplets show uniform radial gas temperature distributions due to substrate-mediated energy transfer. As the droplet size increased, temperature mal-distribution became more prominent due to increased droplet conduction resistance through the base. The opposite behavior was observed for glass substrates where temperature mal-distribution in the gas-phase was prominent, with an increasing gas temperature gradient as the droplet size increased. By experimentally measuring the gas temperature distributions approaching the liquid-vapor interface, we were able to experimentally delineate the magnitude of heat transfer rate coming from the substrate and the gas. Our results validate and show good agreement with past theoretical frameworks, indicating that the substrate plays a more important role during evaporation for smaller droplets with lower conduction thermal resistance and higher thermal conductivity substrates with lower heat spreading resistance. Our work not only experimentally quantifies and decouples energy flows into evaporating droplets to provide experimental validation of century-old frameworks, but it also develops an experimental measurement platform for the

characterization of liquid-vapor phase change phenomena for a range of phenomena such as evaporation, condensation, freezing, desublimation, and boiling.

Chapter 6. Review of Heat Transfer Enhancement Techniques for Single- and Two-Phase Flows

6.1 Single Phase Heat Transfer Enhancement Techniques

The thermal energy exchange between a flowing fluid and its confining channel is a ubiquitous process in modern society. To enhance the fluid-to-wall or wall-to-fluid heat transfer, several techniques have been developed to maximize the contact area between the fluid and the inner wall and/or disrupt the flow to enhance circulation or induce turbulence. Deployment of channels having features capable of enhancing heat transfer enables the reduction of heat exchanger size while maintaining performance. Reduction in equipment size is critical due to the ability to minimize the required volume of costly working fluids and to mitigate potential safety concerns associated with total system fluid volume. In this chapter subsection, a comprehensive review of single-phase heat transfer enhancement techniques is presented. The subsection provides a thorough comparison by analyzing the heat transfer rate, pressure drop, and other operational aspects. Single-phase heat transfer enhancement methods are divided into active and passive techniques. Active methods such as electrohydrodynamic (EHD), magnetohydrodynamics (MHD), or mechanical motion require external power to create enhancement. Passive methods such as dimples, fins, or tape inserts do not require external input and rely only on surface modification.

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Although active methods are more expensive and difficult to implement compared to passive techniques, it enables active control of heat transfer augmentation. This review develops and summarizes key learning data for design optimization enabled by additive manufacturing and machine learning algorithms, helping to inform these next-generation heat exchanger design methodologies for a plethora of modern applications such as electrification of vehicles, computing, and classical industries.

6.1.1 Introduction

Motivated by the rising concern of global energy sustainability, energy efficiency enhancement techniques have gained significant attention within the scientific community and industry. In 2016 alone, it was estimated that the total waste generated amounted to 2 billion tons [103]. Developing higher efficiency systems would not only help address the waste issue but would also reduce material consumption which has been reported to peak to a staggering 92.1 billion tons [103]. One of the main class of systems ubiquitous to many industries are heat exchangers. Heat exchangers find use in thermoelectric and nuclear power generation [104,105], food preparation [106], pharmaceutical drug development [107], building energy technology [108], refrigeration [109], transportation [110], agriculture [111], petrochemical processing [112], material processing [113,114], bulk materials manufacturing [115,116], and many more. To increase heat exchanger efficiency, one of the most common techniques is to utilize components such as channels or tubing with enhanced thermal performance. Utilizing ‘enhanced’ tubes not only decreases the consumption of material, as highlighted earlier, but also enables the manufacturing of compact heat exchangers that are efficient enough to cater to any spatial constraints on the system. Furthermore, having the ability to reduce the size of tubes for an equivalent heat transfer reduces material costs while increasing volumetric power density. This is especially important in modern

equipment given that the spread of COVID-19 has forced millions of employees to work online. As a result of the COVID-19 pandemic, stress on the data center infrastructure has risen due to the sudden increase in processing, memory, computational loading. Enhanced heat exchangers would play a pivotal role in enhancing the performance of such systems as they enable efficient thermal management which is needed to cater to computational power increases. Approximately 20 to 30% of the energy requirement for running a modern data center is used for thermal management [117]. Another instance in which enhanced heat exchangers could improve the performance of energy systems can be found in electric vehicles (EV). The performance of EVs is limited by energy consumption for thermal management of battery, power electronics, motor, and cabin comfort subsystems [118]. More efficient and compact heat exchangers would weigh less, reducing the overall weight of the EV, thus enhancing range. Moreover, efficient heat exchangers enable a higher percentage recovery of waste heat which enhances the overall efficiency and performance [119]. Approximately 20-50% of the energy input in industries at the United States is lost as waste heat. Around 60% of this loss occurs at low temperatures ($< 232^{\circ}\text{C}$) [120]. When investigating the main barriers to recovering this tremendous amount of energy, it was concluded that the inefficient heat exchangers play an important role [120]. Therefore, developing more efficient heat exchangers would not only enable the better utilization of energy, which provides financial benefits to the system operator, but it also cuts down on energy consumption (i.e. reduced fossil fuel use). For non-renewable energy sources, enhancing energy efficiency can lead to lower greenhouse gas emissions and hence slowing of climate change effects. In 2017 alone, CO_2 averaged to 405.5 ppm. This marks a 146% increase the pre-industrial levels [103]. Another low temperature application that requires heat exchangers can be found in heating ventilation air conditioning and refrigeration systems (HVAC&R). HVAC&R systems have been an area of heavy research as heating and

cooling alone consume up to 51% of energy produced by nations [121]. With the maximum efficiency of 1.2 billion currently operating air conditioning units clocking at just 14%, it becomes imperative that a dire need to enhance the efficiency of such systems exists before further units are installed. Last but not least, efficient heat exchangers are also utilized in mobile devices such as integrated circuits, consumer electronics, and power electronics. With the recent increase in the functionality of such devices, large quantities of heat are generated during peak computational operation. Even minute power losses cause local heating and damage to the devices [122] due to the high power density. As a result, it is important to utilize enhanced heat exchangers that can effectively cool these devices with minimal loss.

Here, a review of existing techniques that are used to enhance heat transfer for internal single-phase flows in channels and tubes is presented. This review is divided into two distinct categories, active and passive techniques. Passive methods do not require external input such as electrical power or mechanical actuation to operate. Such methods are the most common form of enhancement due to their low integration penalty with existing heat exchanger technologies. Passive techniques rely on altering the geometry of the tube or heat transfer component by adding inserts or by altering the walls of the tubing itself. In particular, techniques including dimples, wire inserts, and tape inserts are highlighted. Conversely, active methods require external input to operate. Typically, active methods are more complex and costly and are generally not as widely used as passive methods. However, active methods provide the ability to control heat transfer enhancement. This review focuses on techniques including electrohydrodynamic (EHD) active enhancements.

In addition to heat transfer, a second criterion that must be considered is the pressure drop that accompanies any heat transfer enhancement. For example, for single-phase internal liquid flow, increasing the heat transfer via wall surface enhancement enables for tube diameter reduction, leading to a higher pressure drop. Even in the absence of tube diameter reduction, any enhancement in heat transfer is paid for with pressure drop via the Colburn analogy which fundamentally links heat and momentum transfer. Most techniques usually attempt to disrupt the flow or increase the contact surface area which causes higher skin friction. Increasing pressure drop leads to the necessity for larger and more costly pumps to drive the flow. As a result, a delicate balance is required to ensure that the saving due to the heat transfer enhancement technique outweighs the costs of a larger pressure drop. To provide a fair comparison between different enhancement techniques, several efficiencies or factors have been developed and are discussed herein.

This review comes at a key time for a changing engineering landscape. The recent development and industrial push into electrification of ground-based (passenger and off-road) as well as aviation vehicles have placed a renewed focus on single-phase enhancement techniques and their optimization and integration. The mobile nature of these electrified applications necessitates high power density system development with a focus on gravimetric power density. Hence, recent advancements in additive manufacturing as a tool for ultra-high power density heat exchanger design and manufacture has been a focus in the literature to address these needs. Coupled with topology optimization and machine learning, this review represents a key and critical addition in the single-phase heat exchanger design literature to report past approaches and summarize past data capable of being utilized in these modern techniques

6.1.2 Passive Methods

6.1.2.1 Twisted Tape inserts

One of the most widely studied heat transfer enhancement techniques suggested by Manglik and Bergles is the insertion of a twisted tape (TT) into a horizontal tube [123]. Insertion of the insert has the potential to enhance heat transfer in several ways. Firstly, the fluid is restricted to flow in the twisted pathway, creating an effective increase in flow length. Moreover, provided that the TT is metallic and thick enough, it can act as a fin, effectively increasing the surface area in contact with the fluid. Finally, due to the inherent twisted shape of the insert, swirl mixing occurs which enhances agitation and induces turbulence causing heat transfer enhancement [124].

Previous studies have observed that in laminar flows ($Re_d = 83$ -2460 for water and 13-390 for ethylene glycol, where $Re_d = Gd/\mu$), the use of a TT insert inside a round tube can triple the heat transfer rate for a fixed pumping power [125]. For turbulent flows, follow up work [126,127] showed that depending on flow rate, heat transfer enhancement of up to 20% can be attained by using a TT insert. However, an increase in pressure drop was also observed.

In the study carried out by Manglik and Bergles [123], a yellow brass TT having a 4.57 m length was inserted into a 70-30 Cu-Ni (Copper-Nickel) heat exchanger tube having a 3.05 m length that was subjected to a uniform outer wall temperature [123]. The study was carried out for different twist ratios (y , Eq. 14) of $y = 3, 4.5$, and 6, utilizing single-phase water and ethylene glycol covering the range of $300 \leq Re_d \leq 35,000$ (where $Re_d = 4\dot{m}/\pi d\mu$). The twist ratio was defined as:

$$y = \frac{tp}{d}, \quad (14)$$

where tp is the 180-degree twist pitch and d is the tube inner diameter.

As numerically predicted by Date [128], as y decreases, the convection heat transfer coefficient (h) increases. A decreasing twist ratio leads to tighter twists, larger swirl flows, and longer flow paths. The maximum enhancement was found to occur with laminar flows ($Re_d < 2300$). The relative magnitude of the enhancement decreased for turbulent conditions. It was also observed that the Prandtl number ($Pr = \nu/\alpha$) influenced the enhancement, with ethylene glycol ($68 < Pr < 100$) reported to have 1.3 to 1.4 times higher Nusselt number ($Nu_d = hD/k_f$) when compared to water ($3.5 < Pr < 6.5$) [123].

A majority of past studies have concluded that inserts increase pressure drop when compared to smooth tubes with no enhancements. Several factors contribute to the higher pressure drop, including the increase in resistance due to the longer flow path [123,129], and swirl motion which increases wall shear stresses. Consequently, as the twist ratio (y) decreases, the pressure drop increases. It is also worthy to note that having higher Re_d induces larger pressure drop. For $y = 3$, it was reported that the friction factor (f) increased 3.9 times compared to that of smooth tubes at $Re_d = 100$ [123]. However, the pressure drop increase is 7.5 times that of smooth tubes when the flow rate was increased such that $Re_d = 2000$.

Due to the potential to enhance heat transfer, further investigations were carried out to study tape inserts in different conditions [130,131], with many focusing on enhancing the geometry of TTs either by introducing notches, perforations, serrations, or breaks to obtain an even higher heat transfer enhancement. One particular investigation conducted by Eiamsa-ard *et al.* [132] attempted to modify the shape of the insert by cutting its edges with an oblique and straight shape to obtain an oblique delta winglet twisted tape (O-DWT) and straight delta winglet twisted tape (S-DWT) respectively (Fig. 8). These cuts were performed periodically along the tape, where the distance between each cut was equal to the pitch length (H). The experiments were carried out

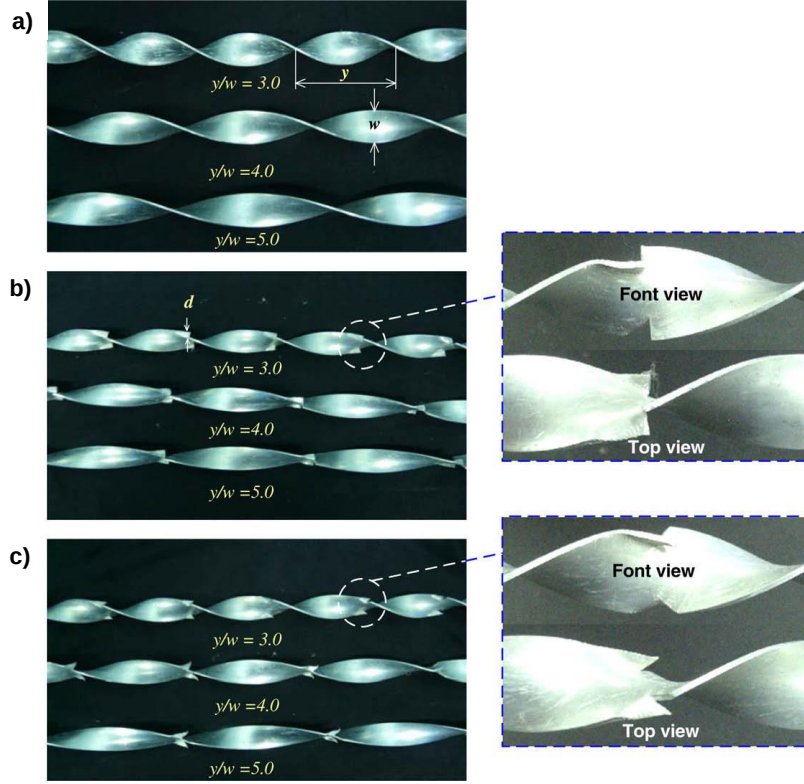


Figure 8. Optical images of twisted tape designs

Side-view optical images of twisted tape designs including: (a) conventional TT (b) S-DWT and (c) O-DWT [132].

The dimension w in (a) is 19.5 mm. Reprinted from Appl Therm Eng, 30, Eiamsa-ard et al., Heat transfer enhancement in a tube using delta-winglet twisted tape inserts, 310-8, 2010, with permission from Elsevier.

using an aluminum (Al) tape inserted into a 1 m long Cu tube having a 19.5 mm inner diameter with liquid water as the working fluid. The experiment was repeated for three different twist ratios ($y = 3, 4, 5$) and different depth of wing cut ratios ($DR = 0.11, 0.21, 0.32$) for flow rates corresponding to $3000 \leq Re_d \leq 27,000$. Note, in the work of Eiamsa-ard *et al.*, the twist ratio was defined as:

$$y = \frac{H}{w}, \quad (15)$$

where w is the width of the tape. The wing cut ratio was defined as:

$$DR = \frac{d_c}{w}, \quad (16)$$

where d_c is the depth of the wing cut.

Experimental results revealed that both the O-DWT and S-DWT provided higher Nu_d compared to conventional tapes, with Nu_d having a positive correlation with the Re_d [132]. The study found that the average Nu_d for both delta winglet tape designs were 1.02 to 1.64 times higher than conventional inserts and 1.1 to 2.55 times higher than tubes having no insert at all. The reason delta winglet tape designs have superior heat transfer compared to conventional tapes is due to vortex generation. Vortex generation induces further turbulent effects in addition to further increasing secondary flows near the tube walls [133,134]. However, due to the added disturbance to the fluid flow, delta winglet tapes have higher friction factors relative to conventional TTs. It was reported that delta winglet tapes have friction factors 1.08 to 1.95 times higher than conventional TTs and 2.5-7.02 times higher than smooth tubes.

Since the wing cuts cause a higher disturbance to flow which in turn causes an increase in turbulence providing a more effective heat transfer enhancement, it can thus be concluded that the larger the wing protrusion, the higher the disturbance. Since the protrusion is quantified by the depth of the wing cut ratio, the higher the depth ratio the higher the Nu_d and the friction factor.

Similar to conventional TTs, the smaller the twist ratio (γ) of tapes with DWTs, the higher the convection heat transfer coefficient obtained. As can be noted in table 2, the Nusselt number of DWT tapes is higher than that of a plain tube or conventional TT. Moreover, this increase is more prevalent for a twist ratio of 3 than that of a twist ratio of 5. Comparing O-DWT with S-DWT it can be observed from Tables 2 and 3 that for a given γ , the mean Nu_d and the friction

factor of S-DWT was slightly lower than O-DWT providing a lower effective heat transfer and pressure drop.

Table 2. Effect of twist ratio (y) on average Nusselt number (Nu).

Twist ratio, y	$\frac{Nu_{d,S-DWT}}{Nu_{d,smooth}}$	$\frac{Nu_{d,S-DWT}}{Nu_{d,conventional\ tape}}$	$\frac{Nu_{d,O-DWT}}{Nu_{d,smooth\ tube}}$	$\frac{Nu_{d,O-DWT}}{Nu_{d,conventional\ tape}}$
3	1.6	1.2	1.66	1.25
4	1.4	1.16	1.45	1.2
5	1.23	1.1	1.31	1.17

Table 3. Effect of twist ratio on friction factor (f).

Twist ratio, y	$\frac{f_{S-DWT}}{f_{smooth}}$	$\frac{f_{S-DWT}}{f_{conventional\ tape}}$	$\frac{f_{O-DWT}}{f_{smooth\ tube}}$	$\frac{f_{O-DWT}}{f_{conventional\ tape}}$
3	4.35	1.25	4.68	1.33
4	3.86	1.34	4.18	1.45
5	3.4	1.4	3.67	1.52

Furthermore, for a given DR and Re_d , it was observed that the Nu_d range and the friction factor range of S-DWT were also slightly lower than O-DWT providing lower effective heat transfer and lower pressure drop. For example, for DR = 0.32, the average Nu_d for S-DWT was 1.25 to 2.16 times higher than a smooth tube with no insert. The O-DWT was 1.35 to 2.55 times higher. For the mean friction factor, however, S-DWT was 3.22 to 5.85 times higher than smooth tubes with no insert while O-DWT was 3.62 to 7 times higher.

Generally, it was found that the mean Nu_d and f of S-DWTs were 4.2% and 7.8% lower than O-DWTs. To better understand the overall performance of TTs, the following efficiency relation was formulated:

$$\eta = \frac{h_t}{h_p} \Big|_{pp} \quad (17)$$

where η is the thermal performance factor, h_t is the heat transfer coefficient with a turbulator, while h_p is the heat transfer coefficient of the plane tube. The subscript pp indicates at a constant pumping power. If $\eta > 1$, the system performance is increased. The greater η is, the better the performance. However, it is important to note that this factor compares methods assuming a constant pumping power. This can be misleading as it does not take into account the change in the mass flow rate. For example, a technique can have the highest Nu_d augmentation for a given pumping power and be deemed more favorable, however, it alters the mass flow rate to an extent that is unfavorable for a given process downstream. Consequently, equation 4 must be used with caution.

For a fixed Re_d , η for both DWTs was greater than that of conventional TTs. However, similar to conventional TTs, η decreases with increasing Re_d . Since DWTs still have a η greater than that of conventional TTs at higher Re_d , this means turbulators are better implemented in high Re_d flows where conventional TTs are not as effective. In general, it was found that O-DWTs have a higher η than S-DWTs. Overall, the best thermal performance was $\eta = 1.24$, obtained for $Re_d = 3000$, $DR = 0.32$, and $y = 3$.

Several other interesting studies have been carried out on TTs. Similar to work on DWTs, Wongchareea and Eiamsa-ard [135] investigated the effect of changing the shape of the wings of the TT insert on heat transfer enhancement.

Wings having different shapes including triangles, rectangles, and trapezoids were inserted in a Cu tube (Fig. 9). All shapes provided an increase in both heat transfer rate and friction factor compared to a plain Cu tube with no insert. The experiments were repeated for wing chord ratios

of 0.1, 0.2, and 0.3. Trapezoidal wings provided the highest increase in Nu_d and f . The maximum reported an increase for Nu_d and f was 2.84 and 8.02 times that of a plain tube respectively.

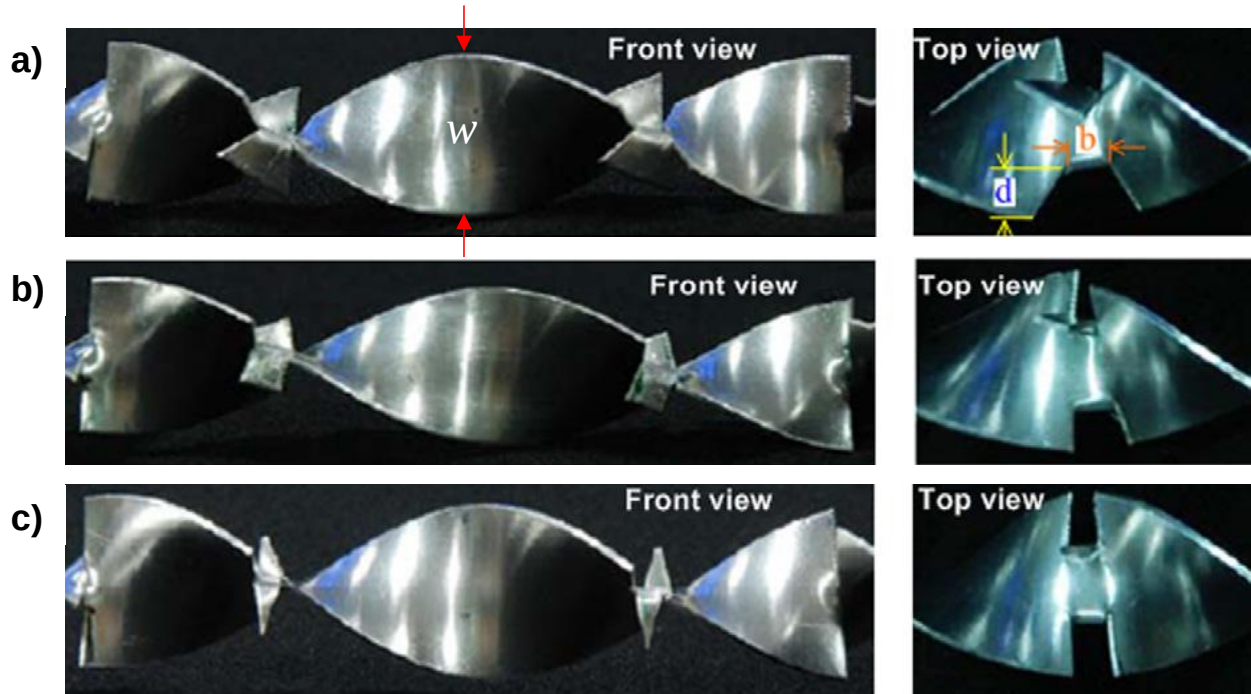


Figure 9. Images of twisted tapes

Side-view images of (a) trapezoidal, (b) rectangular and (c) triangular twisted tapes [135]. The dimension w in (a) is 20 mm. Adapted from Chem Eng Process Process Intensif, 50, Wongcharee and Eiamsa-ard, Heat transfer enhancement by twisted tapes with alternate-axes and triangular, rectangular, and trapezoidal wings, 211-9, 2011, with permission from Elsevier.

Esmailzadeh *et al.* [136] investigated the effect of varying TT insert thickness (0.5 mm, 1mm, 2 mm) inside a Cu tube on Nu_d and f in a laminar flow regime ($150 < Re_w < 1600$, where $Re_w = \rho U w / \mu$). Using a nanofluid as the working fluid, higher heat transfer was evident for thicker tapes. However, pressure drop also increased with TT thickness. Bhuiya *et al.* [137] investigated the effect of having two steel TT inserts in one brass tube (Fig. 10a, b). Four different twist ratios ($\gamma = 1.95, 3.85, 5.92, 7.75$) were studied in turbulent flow regimes ($150 < Re_d < 1600$). Using air as the working fluid, the study showed that the heat transfer rate increased by 60 to 240%, while f increased by 91 to 286% when compared to a plain tube. Furthermore, using the

Promvonge definition of thermal enhancement efficiency (defined in Table 6), the efficiency values were between 1.01 to 1.34. A later study [138] carried out by Bhuiya *et al.* noted that increasing the number of TTs, increases the Nu_d and f even further. In their investigation, it was reported that three TTs demonstrated an Nu_d and f increase of 3.85X and 4.2X, respectively, when compared to a plain tube. The efficiency was found to be 1.44. However, it is important to note the extent of the increase in Nu_d and f . A numerical study carried out by He *et al.* [139] compared the effect of adding an extra TT on the Nu_d and f . Using CuO-water nanofluid in different solid concentrations (1–4 vol%) as the working fluid, it was reported that, similar to Bhuiya *et al.*, increasing the number of TTs caused an increase in the heat transfer and pressure drop. However, the pressure drop increase overshadows the increase in heat transfer. Consequently, it is more favorable to use only one TT. Eiamsa-ard and Promvonge [140] investigated the effect of having a loosely fit twisted steel tape insert with and without a core-rod insert inside a Cu tube (Fig. 10c, d). Water was used as the working fluid and experiments were carried out over a range of Re_d ($2000 < Re_d < 12,000$). Although both demonstrated an enhancement in heat transfer, the enhancement of the case not having a rod insert was larger than that with a core by almost 50%. Furthermore, the friction factor was reduced by 50% less.

A similar study carried out by Bahiraei *et al.* [141] investigated double co-twisted tapes and double counter twisted tapes at different twist ratios. Similar to other TT investigations either orientations provided an increase in pressure drop and heat transfer augmentation. However, counter twisted tapes provided lower irreversibility and more turbulence and boundary layer disturbance than that of co-twisted tapes. For both cases, it was also noted that lower twisted ratios provided higher turbulence and interlayer mixing. Accompanying this increase in turbulence, however, would be an increase in the frictional pressure drop.

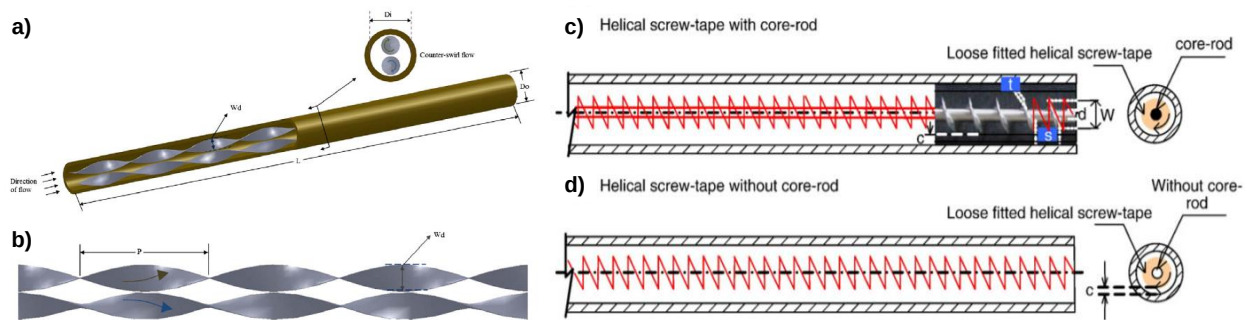


Figure 10. Schematic of double twisted tape

Schematic of a double twisted tape (a) fitted inside a tube and (b) viewed from the side with geometric parameters labeled [137]. Reprinted from Int Commun Heat Mass Transf, 50, Bhuiya et al., Performance assessment in a heat exchanger tube fitted with double counter twisted tape inserts, 25-33, 2014, with permission from Elsevier. (c) with core-rod and (d) without a core-rod [140]. Reprinted from Int Commun Heat Mass Transf, 34, Eiamsa-ard and Promvong, Heat transfer characteristics in a tube fitted with helical screw-tape with/without core-rod inserts, 176-85, 2007, with permission from Elsevier.

Other studies such as that conducted by Liao and Xin [142] incorporated TT inserts with modified tubes having three-dimensional (3D) internal extended surfaces (Fig. 11a). The study was conducted using water, ethylene glycol, and ISO VG46 oil for a range of flow conditions ($80 < Re_d < 50,000$, where d is measured from the bottom of the extended surface). The presence of two enhancements provided an increased heat transfer as well as pressure drop. For ISO VG46 oil, the average Stanton number ($St = Nu/Re.Pr$) increased by 580% while f increased by 650%. On a similar note, Thianpong *et al.* [143] studied the effect of tape inserts inside a dimpled Cu tube with air as the working fluid (Fig. 11b).

In their study, a range of Reynolds numbers ($12,000 < Re_d < 44,000$) and dimple pitch ratios were investigated. Due to the increase in the surface area combined with induced swirl motion, the combined effect of a TT inside a dimpled tube provided a higher heat transfer enhancement and f when compared to a dimpled tube alone or a plain tube with no dimples. It was also reported that as the pitch ratio (γ) decreases, the heat transfer enhancement is even more

pronounced. Ray and Date [144] conducted a study on TT inserts for square ducts. It was noted that local Nu_x , where x is the distance along the square duct, varied between 1.02 and 1.17 with the maximum being where the tape is aligned with the diagonal of the duct. Moreover, the square duct had a lower f when compared to its circular counterpart.

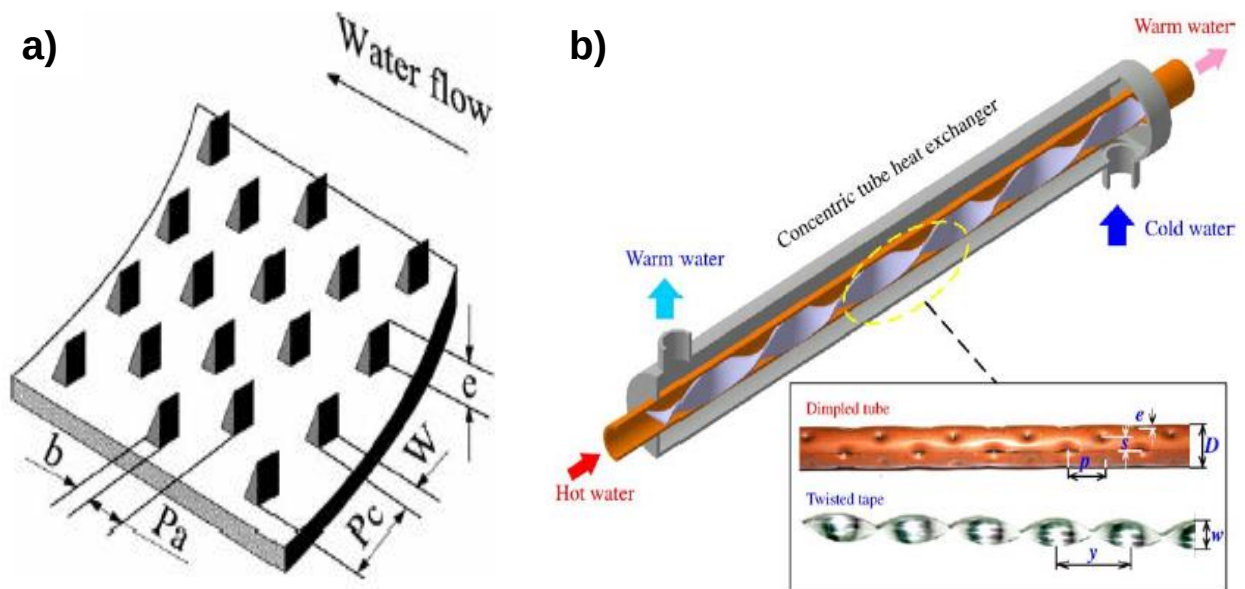


Figure 11. Schematic and photograph of tube insides

(a) Schematic of extended surfaces [142]. Reprinted from Chem Eng J, 78, Liao and Xin, Augmentation of convective heat transfer inside tubes with three-dimensional internal extended surfaces and twisted-tape inserts, 95-105, 2000, with permission from Elsevier and (b) Schematic and photograph (inset) of a dimpled tube with TT insert [143]. Reprinted from Int Commun Heat Mass Transf, 36, Thianpong et al., Compound heat transfer enhancement of a dimpled tube with a twisted tape swirl generator, 698-704, 2009, with permission from Elsevier

Although the TT insert methods are simple and provide an improvement in the heat transfer performance, the improvement is more prominent in laminar flow regimes over turbulent regimes [145,146]. Turbulent flows already have high agitation rates and thus more mixing in comparison

to laminar flows [147,148]. Consequently, the induced swirl motion created by the TTs does not add a significant improvement to mixing. Conversely, laminar flow is characterized by highly ordered streamlines and so any induced swirling motion will significantly enhance mixing. Therefore, the TT enhancement technique can find use in applications where the fluid is characterized as highly viscous or if the fluid flow velocity is low (laminar flow, $Re_d < 2,300$).

Furthermore, a common feature associated with TT inserts is the higher pressure drop when compared to smooth or plain tubes with no insert. Consequently, a larger pump would be required to obtain the equivalent desired flow rate.

Another point to also consider is that to reap the full benefit of the TTs, it is necessary to have the tape be made of thick metal and have it attached thermally or mechanically to the inside tube of the walls. Otherwise, the tape would not effectively act as an extended surface or fin [123]. Attaching the tape thermally to the tube could be challenging as well as difficult to undo if the tape needed to be removed. For long tubes (> 10 m), having a tape insert made of metal could be prohibitively expensive, especially if the tape was to be made from metals such as stainless steel as in the study conducted by Eiamsa-ard *et al.* [149]. Another consideration that must be accounted for is the need for tape replacement due to wear and tear after prolonged use. Depending on the application and the TT material, fouling is also a concern. For example, in the milk industry, fouling due to chemical reactions is prominent [150]. Finally, great care needs to be taken if the tube has a small diameter, as blocking due to solid contaminants may be an issue. It is important to ensure that the fluid flowing inside tubes with TT inserts is free of contaminants, this requires an additional filtration system that adds to the overall cost of the heat exchanger and system. To address the difficulty of enhancing heat transfer in turbulent flows with TTs, wire coil inserts are typically used. Due to the nature of its geometry, a coil would be more effective in turbulent

flow regimes because it enables agitation of the laminar sublayer near the tube walls more effectively than a tape [151]. The next subsection is dedicated to the review of coil inserts

6.1.2.2 Wire Coil Inserts

A study conducted by Garcia *et al.* [152] investigated the effect a helical steel wire coil (Fig. 12) has on heat transfer and pressure drop when inserted inside a tube of length 1.37 m and a diameter of 18 mm in different flow conditions. Like TTs, the coil generates swirl flow but mainly near the tube walls. The swirl juxtaposed with the mainstream flow generates centrifugal forces that cause high-temperature fluid in the boundary layer to move towards the center of the tube, subsequently enhancing heat transfer. Furthermore, as mentioned previously, the coil acts as a turbulence promoter in the laminar sublayer of turbulent flows. This is because the wire has the same effect as artificial roughness in turbulent flows. Previous studies have shown that wire coils have the same effect on flow as that of internally corrugated and ribbed tubes [153,154].

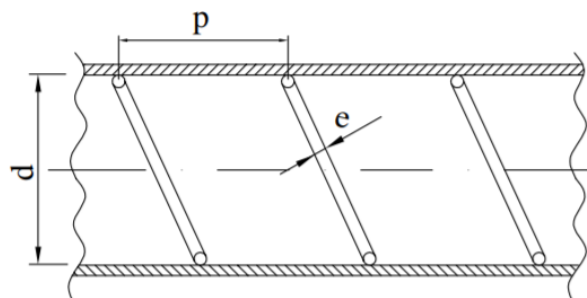


Figure 12. A tube with wire coil insert

Cross-sectional schematic of a horizontal tube having a wire coil insert [152]. Reprinted from Int J Heat Mass Transf, 48, García et al., Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers, 4640-51, 2005, with permission from Elsevier

Garcia *et al.* [152] used six coils having geometries with $1.17 < p/d < 2.68$, $0.07 < e/d < 0.1$, and $14 < p/e < 33$ (see Fig. 12 for schematic symbol definitions) in order to quantify the heat transfer enhancement and pressure drop. The coils were tested for $150 < Re_d < 90,000$ and $2.8 < Pr < 150$. Note, the definition of Reynolds number for tubes enhanced with wire coils is defined using the inner tube diameter. The fluids used were water and water-propylene glycol mix at 40°C and 65°C.

As suspected, in laminar flow regimes ($Re_d < 500$), the coil inserts provide negligible increases in Nu_d (less than 1.4 times that of smooth tubes). However, for turbulent flow, Nu_d increased significantly when the coil was introduced. The increase subsided as Re_d increased further [152,155]. For example, at $Re_d = 5,000$, the Nu_d increased by 2 to 2.8 times, but when Re_d was raised to 20,000 the Nu_d increased by 1.5 to 2.1 times. Coils having the lowest p/d showed highest Nu_d increase in the turbulent region.

The insertion of a coil insert trips the flow and accelerate the transition from laminar flow to turbulent flow. In fact, the transition to turbulent flow was observed to occur gradually at $Re_d \approx 700$ with no sudden transition. The smooth transition was attributed to the presence of swirl motion [156]. Since turbulent flow yields better mixing when compared to laminar flow, heat transfer is augmented. Consequently, the insertion of a wire coils helps to augment heat transfer because it ensures the transition to turbulence at an earlier stage compared to the plane tube [152].

As with TT inserts, wire coils increase f when compared to plane tubes. For laminar flow ($Re_d < 500$), f increased by 1.2 to 1.8 times. As the flow becomes turbulent ($Re_d = 80,000$), f increases anywhere between 2 and 9 times that of a plane tube depending on the wire geometry.

To compare the effectiveness of coil insert designs for different flow regimes, the R3 criterion was proposed [157], defined as:

$$R3 = \frac{Nu_{d,a}}{Nu_{d,s}}, \quad (18)$$

where the subscripts a and s represent ‘augmented’ (with wire coil) and ‘smooth’ (no wire coil) tubes. The R3 criterion is used to compare Nu_d for the case of identical pumping power and heat exchange surface area. The higher R3, the more significant the heat transfer augmentation. The definition of R3 is similar to Eq. 17, hence its limitations must be taken into consideration when using it.

For laminar flow regimes ($Re_d = 500$), R3 ranged between 1.3 and 1.4. For turbulent flow ($Re_d = 4,000$), $R3 = 1.6$ to 1.7 , decreasing to 1.3 to 1.4 when Re_d is raised to $10,000$. For $Re_d = 30,000$, R3 decreases to an extent that utilization of a wire coil has negligible effects on heat transfer and is not recommended. It was observed that the best modes of operation for coil inserts are flows residing in the transitional region where R3 can reach values approaching 2.

Behabadi *et al.* [158] studied the effect of wired coil inserts on the heat transfer and pressure drop of nanofluids dispersed in water. The study was carried out in turbulent flow regimes with different nanofluid particle weight concentrations of 0.05%, 0.1%, and 0.2%. At the highest Re_d value, the convective heat transfer coefficient was observed to increase, on average, by 85% compared to plain tubes with no coil and plain water as the working fluid. However, the pressure drop was observed to drop 4.75 times that of plain tubes. It was also observed that the increase in heat transfer increases with particle weight concentration. The enhancement was hypothesized to occur due to Brownian motion and micro-convection of the dispersed particles. A similar study carried out by Abdul Hamid *et al.* [159], investigated the effect of wired coil inserts on the heat

transfer and pressure drop of TiO₂–SiO₂ nanofluids dispersed in water. The experiment was conducted for a range of concentrations, p/d ratios, and Reynolds number. It was reported that having the nanofluid and wired coil combination would result in a maximum heat transfer enhancement of 254.4% when compared to water/ethylene glycol in a plain tube. Further, the frictional factor increased by 1.76-6.38 times. Although there is a considerable rise in the frictional factor, it was reported that the augmentation in heat transfer overshadowed the increase in the frictional factor across all volume concentrations. Gunes *et al.* [160] studied the effects of the wire coils on the heat transfer and pressure drop if the wire coil does not make mechanical contact with the circular tube walls. Carried out in a turbulent flow regime ($4105 < Re_d < 26,400$), the experiment was done with three different pitch ratios ($p/d = 1, 2, 3$) and two different wall-to-coil distance (1 mm and 2 mm). Both Nu_d and f increased with decreasing p/d and wall-to-coil distance. Using the efficiency definition by Promvonge [161] (Table 6), the highest efficiency was 50% (at $s = 1$ and $p/d = 1$). Emani *et al.* [162] attempted to study the effect of combining coils with corrugations and screw tapes in rectangular or square cross conduits with laminar flow regimes. It was reported that both the frictional factor and Nusselt number increased with the aspect ratio of the conduit, the helical angle, and the diameter of the wire coil. The highest augmentation was reported at 30° and 60°. Promvonge [161] studied the effect of changing the wire coil cross-section from circular to a square. The study carried over a range of Re_d ($5000 < Re_d < 25,000$), concluded that having a square cross-section provided an even higher heat transfer compared to circular cross-section wires. Furthermore, compared to plain tubes, the Nu_d was found to increase by 1.8 to 2.6 times. Sreenivasulu and Prasad [163] numerically investigated the convective heat transfer performance for an annulus having its inner cylinder wrapped with a helical wire of circular cross-section (Fig. 13). The simulation was for turbulent flow and varied the wire diameter

and annulus diameter ratios. The results showed that increasing the annulus diameter ratio and decreasing the p/d ratio causes the averaged values f and Nu_d to increase. Interestingly, the wire diameter had no significant effect on performance. The thermal hydraulic performance ratio (THPR) proposed by Fan *et al.* [164] (Eq. 19) was found to be considerably higher than unity.

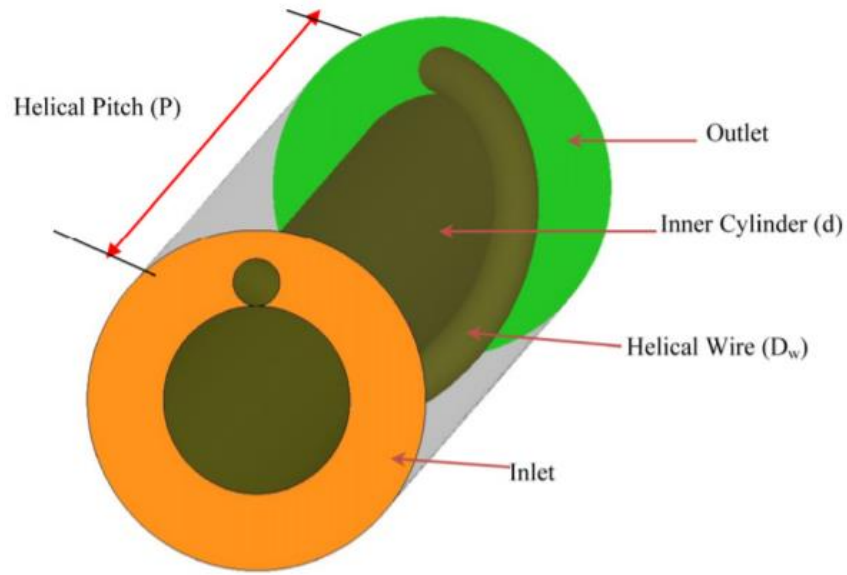


Figure 13. Annulus flow path with inner wire coil

3D schematic of an annulus flow path with a wire coil on the inner surface [163]. Reprinted from Int J Therm Sci, 48, Sreenivasulu and Prasad, Flow and heat transfer characteristics in an annulus wrapped with a helical wire, 1377-91, 2009, with permission from Elsevier

$$THPR = \left(\frac{Nu_w}{Nu_d} \right) \left(\frac{f}{f_w} \right). \quad (19)$$

There are many advantages to using wire coil inserts. Coiled wire inserts are cost-effective, easy to install in existing setups, and do not alter the mechanical strength of the tube [152]. Wire coil inserts are also less likely to experience blockage when compared to TT inserts due to geometry only obstructing flow at the tube walls. Furthermore, unlike other enhancement techniques, wire coil inserts have predictable performance in the transitional flow regime and are

more effective in turbulent flows. Conversely, in the laminar regime ($Re_d < 500$), wire coils fitted inside tubes have similar thermal-hydraulic performance as smooth tubes [151]. Moreover, as TT inserts, wire coil inserts cause high-pressure drop and increased f . Consequently, a larger pump would be required to obtain the desired equivalent flow rate to that of a smooth tube. However, the pressure drop for wired coil inserts is much less than that of TTs because disturbance to flow occurs only near the wall of the tube [165] with little introduction of swirl and added shear due to flow path elongation. Another disadvantage of wire coil inserts (like TT inserts), is their susceptibility to fouling and thus, the need for replacement. Finally, unlike tape insert design, design correlations for wire coil inserts are not as well established in laminar, transition, or turbulent flow regimes [154].

It is important to note that whether tape or coil inserts are used, fluid properties play a key role in the enhancement performance. For example, in laminar flow, fluids with higher Pr are more sensitive to the presence of inserts. The higher efficacy of inserts on higher Pr fluids is attributed to the fact that these fluids have thicker hydrodynamic boundary layer thicknesses, and therefore inserts have a greater capability to disturb the flow and enhance heat transfer [152,165]. In turbulent, however, a discrepancy exists. For example, Garcia *et al.*, [152] showed that the Prandtl number does not affect heat transfer enhancement, in contention with the findings of Wang and Sunden [165].

6.1.2.3 Wire Coil and Twisted Tape Inserts

In an attempt to reap the benefits of both wire coil and TT inserts, Promvonge [166] investigated the effects of using a coil and tape insert at the same time (Fig. 14) within a Cu tube having 1.5

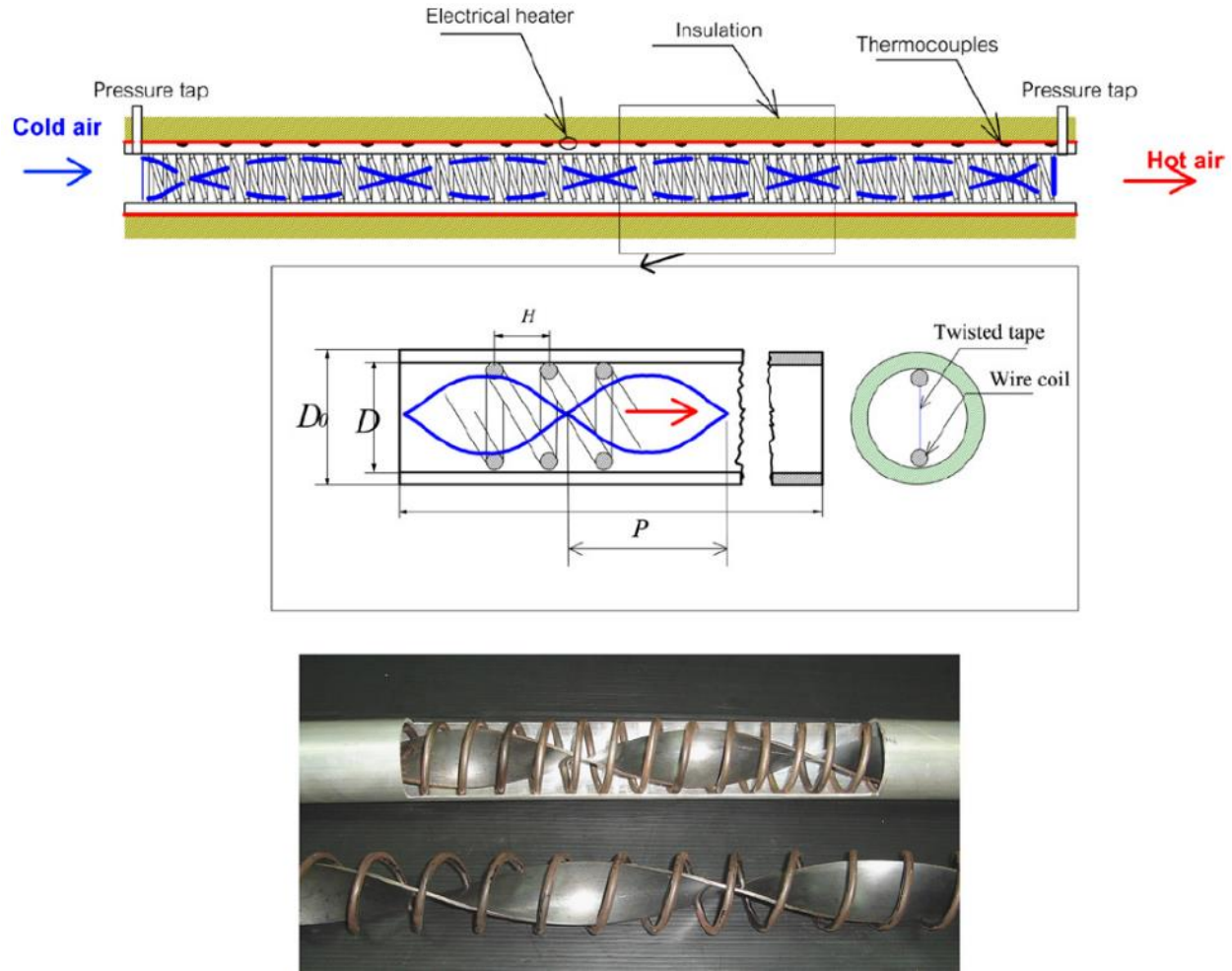


Figure 14. Coiled wire and twisted tape insert

Schematic (top) and pictures (bottom) of coiled wire and twisted tape insert in a Cu tube [166]. The inset schematic in the middle has all dimensional characteristics. The dimension D in the schematic is 47 mm. Reprinted from Energy Convers Manag, 49, Promvonge, Thermal augmentation in circular tube with twisted tape and wire coil turbulators, 2949-55, 2008, with permission from Elsevier

m length with inner and outer diameters of 47 mm and 50 mm, respectively. The air flowing inside had Re_d (where d signifies the inner tube diameter) values ranging between 3,000 and 18,000. The coils used were made of steel and had diameters of 5 mm with $p/e = 4, 6, 8$. The tapes used were also made of steel and had a thickness of 2 mm, a width of 35 mm, and $y = 4$ and 6.

The study reported that Nu_d increased tremendously by 300% to 685% depending on $p/e: y$ (coil spring pitch ratio: tape twist ratio) and Re_d value. The enhancement in heat transfer

was due to the joined effects of the wire and tape insert. The wire increases the interruption of the developing fluid boundary layer, which causes increased turbulence, while the tape induces swirl flow which aids in mitigating the amount of fluid being trapped between coil grooves. The added swirl flow prolongs the fluid residence time and increases momentum transfer at the tube walls. However, the augmentation in Nu_d occurs at the penalty of higher pressure drop. The study observed that the mean value of f increased by a factor of 30 to 68 when compared to an equivalent smooth tube. The higher f is due to the increase in surface area and flow restriction.

A comparison between the wire coil case, twisted tape case, and coil and tape combination case show that the best heat transfer rate is obtained when the combination was used. The second-best heat transfer rate, in general, is for the wire coil case. The same order is present when considering pressure drop, where the highest is observed in the coil-tape combination case, followed by the coil, and then twisted tape. Given that the study was conducted for the upper range of Re_d values *i.e.* turbulent, coils are better able to disturb the flow at the laminar sublayer, hence their higher heat transfer augmentation and pressure drop compared to tapes.

To compare the performance of the combined case, coil alone case, and tape alone case, an enhancement factor was introduced that considers the increase in Nu_d in addition to the increase in f , formulated as:

$$\eta = \left(\frac{Nu_{d,a}}{Nu_{d,o}} \right) \left(\frac{f_a}{f_o} \right)^{-1/3}, \quad (20)$$

where subscripts a and o represent augmented and smooth tubes, respectfully.

The enhancement factor η takes into account both the change in f and Nu_d , unlike Eq. (18) which only takes into account the change in the Nu_d , providing a clearer picture of the compromise between heat transfer enhancement and pressure drop penalty. If $\eta > 1$, the enhancement in heat

transfer is larger than the increase in pressure drop, making tube enhancement favorable compared to smooth or plain tube design. The higher η , the more favorable enhancement becomes. The enhancement factor was shown to decrease as Re_d increases.

Wang *et al.* [167] further cemented these results, where it was reported that wired coil inserts also provided much higher heat transfer enhancement and pressure drop compared to that of TTs. Further, it was reported that the combination of wired coils and TTs provided the highest enhancement and pressure drop than if each one was used separately. Using Eq. (20), it was also concluded that the highest enhancement factor corresponded to coil-tape combination, followed by tape inserts and then wired coil inserts.

Overall, it can be concluded that having the combination of TTs and coiled wires is better able to better cater to both laminar and turbulent flows. Although the increase in pressure drop is substantial, the pressure drop is overshadowed by the heat transfer augmentation rendering it a more viable solution. That being said, all the disadvantages presented previously in TTs such as blockage, fouling, and so on would also translate to this combination.

6.1.2.4 Baffles and Fins

Baffles, fins, and similar devices which increase the surface area of contact with the fluid, also induce turbulence and secondary flows to further enhance heat transfer. An investigation carried out by Sriromreun *et al.* [168] studied the effect of zig-zag shaped baffles (Fig. 15a) having a 45-degree inclination on the heat transfer inside a channel ($AR = 10$) subjected to isothermal heat flux on the top wall. Similar to fins, the baffles induce vortices that enhance turbulence and heat transfer. The study was carried out numerically and experimentally over a range of Re_h ($4,400 <$

$Re_h < 20,400$, where $Re_h = \rho U D_h / \mu$) with different baffle pitches and baffle-to-channel height ratios.

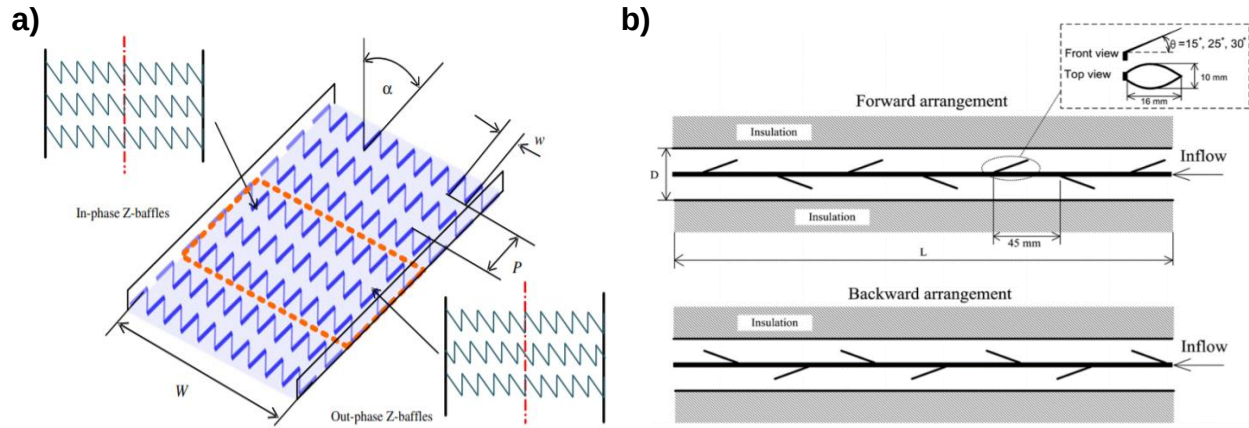


Figure 15. Zig-zag baffles and louvered strip inserts

(a) Schematic of zig-zag-shaped baffles [168]. Reprinted from Int Commun Heat Mass Transf, 39, Sriromreun et al., Experimental and numerical study on heat transfer enhancement in a channel with Z-shaped baffles, 945-52, 2012, with permission from Elsevier (b) Side-view cross-section schematic of louvered strips residing in a tube [169]. Reprinted from Int Commun Heat Mass Transf, 35, Eiamsa-ard et al., Turbulent flow heat transfer and pressure loss in a double pipe heat exchanger with louvered strip inserts, 120-9, 2008, with permission from Elsevier

The study showed that the presence of baffles increased the Nu_h and f considerably when compared to a smooth channel with no baffles. Another study by Eiamsa-ard *et al.*, [169] studied the effect of louvered strips (Fig. 15b) inserted inside a Cu tube for turbulent water flow ($6,000 < Re_d < 42,000$). Strips having inclinations of 15° , 25° , and 30° were studied and were arranged in forward and backward orientations. Having the strips in a forward orientation increased Nu_d and the f by 2.84 and 4.13 times, respectively when compared to plain tubes. Orienting the strips backward increased Nu_d and f by 2.63 and 2.33 times, respectively.

Torii *et al.*, [170] used winglets (Fig. 16) to induce vortices which in turn increase turbulence and enhance heat transfer. The investigation was carried out for laminar flow. Placed in such a way that reduces drag, the winglets were able to increase the heat transfer by 10 to 30% when placed in a staggered arrangement and 10-20% when placed in an inline arrangement. The

associated pressure drop decreased by 34% to 55% and 8% to 15% that of fin tube banks without a vortex generator.

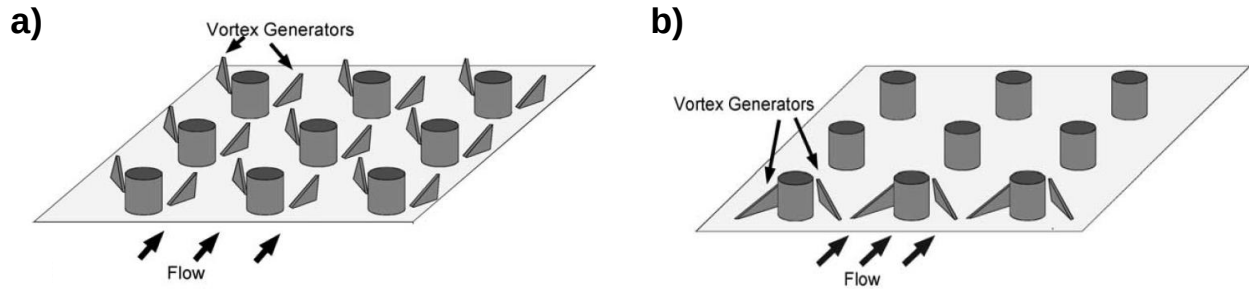


Figure 16. Winglets flow arrangements

Winglets: (a) Flow downstream arrangement (b) Flow upstream arrangement [170] Adapted from Int J Heat Mass Transf, 45, Torii et al., Heat transfer enhancement accompanying pressure-loss reduction with winglet-type vortex generators for fin-tube heat exchangers, 3795-801, 2002, with permission from Elsevier

Hosseinzadeh *et al.* studied the effect of internal copper fins in star [171] and hexagonal [172] shaped triplex tubes (Fig. 17a and 17b) on the solidification of hybrid nano-enhanced phase change materials (HNEPCM). Due to the increased surface area of contact provided by the fins in both orientations, the heat transfer is enhanced promoting the solidification rate.

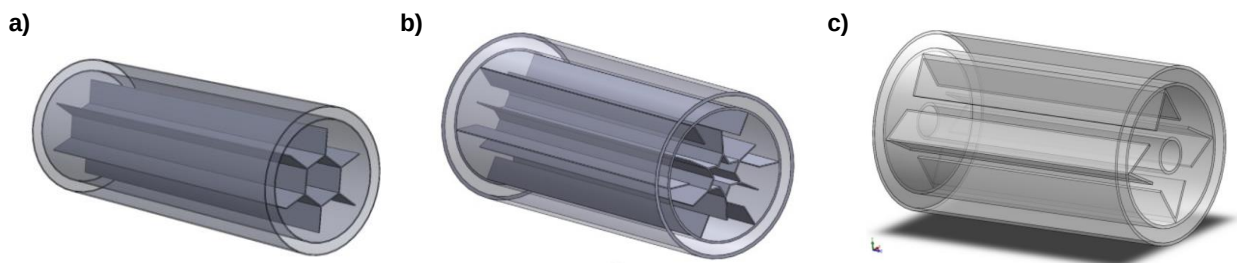


Figure 17. Fins in different shaped tubes

(a) Fins in a hexagonal shaped triplex tube [171]. Adapted from J Mol Liq, 300, Hosseinzadeh et al., Effect of fin and hybrid nano-particles on solid process in hexagonal triplex Latent Heat Thermal Energy Storage System, 112347, 2020, with permission from Elsevier, (b) Fins in a star-shaped triplex tube [172]. Adapted from Renew Energy, 154, Hosseinzadeh et al., Effect of internal fins along with Hybrid Nano-Particles on solid process in star shape triplex Latent Heat Thermal Energy Storage System by numerical simulation, 497-507, 2020, and (c) Fins in a V-shaped triplex tube [173] Adapted from Appl Therm Eng, 163, Alizadeh et al., Solidification acceleration in a triplex-tube latent heat thermal energy storage system using V-shaped fin and nano-enhanced phase change material, 114436, 2019

It was reported in both setups that adding fins increased the solidification rates by 9% and 8% for the star and hexagonal-shaped triplex tubes respectively. Further, it was identified that the fins were a better catalyst for solidification when compared to the hybrid nano-particles. Another study conducted by Alizadeh *et al.* [173] drew similar for V-shaped fins (Fig. 17c). Due to the fins length, the thermal penetration into the phase change material is facilitated, accelerating the phase change. It was reported the longest fins studied provided the optimum results

From the above studies, it can be concluded how advantageous fins or baffles can be to the heat transfer augmentation. Furthermore, it can be noted how the extent of this heat transfer augmentation can be controlled by simply varying the orientation, length, and angle of the fins. However, a common issue with fins is the relatively added difficulty of introducing it to already existing heat exchangers. Unlike other solutions presented such as twisted tapes, fins are more difficult to manufacture, remove or add to an existing tube. Moreover, maintenance becomes an issue if any blockage or damage occurs to the fins as the entire tubing would require

6.1.2.5 Ribbed Tubes

Similar to the coiled wire inserts, corrugated/ribbed tubes provide an added roughness that enhances heat transfer. Naphon *et al.* [174], studied the effect of helical ribs (Fig. 18a) on the heat transfer coefficient and friction factor. Using water as the working fluid, nine test sections of different helical rib pitch and helical rib height-to-diameter ratio were tested at varying flow rates. It was reported that having the helical ribs in the tubes increased the pressure drop and the heat transfer rate considerably. Li *et al.* [175], did a similar investigation, where discrete double inclined ribs (DDIR) were embossed on a tube (Fig. 18b). Using water as the working fluid, the study was conducted in a turbulent flow regime ($15,000 < Re_{OD} < 60,000$). Since DDIR induce vortices, the

heat transfer is augmented by 100-120% compared to that of a similar smooth tube. Accompanying this augmentation was an increase in pressure drop of 170-250%. This method proved to be more effective than other methods of inducing turbulence such as wired coils and corrugated tubes.

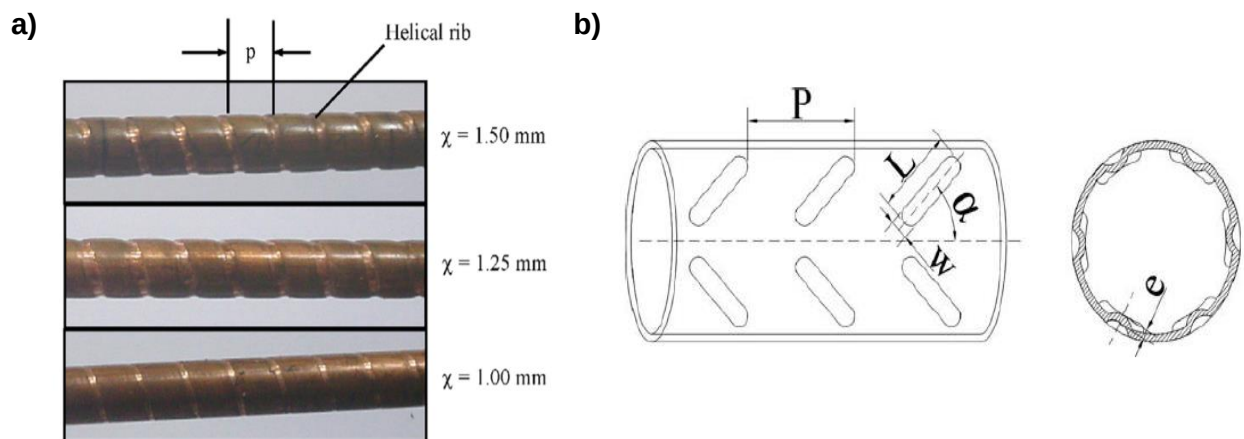


Figure 18. Helical ribs and DDIR tube

(a) Helical ribs [174] Reprinted from Energy Convers Manag, 47, Naphon et al., Tube side heat transfer coefficient and friction factor characteristics of horizontal tubes with helical rib, 3031-44, 2006, with permission from Elsevier and (b) DDIR tube [175]. Reprinted from Int J Heat Mass Transf, 52, Li et al., Turbulent flow and heat transfer in discrete double inclined ribs tube, 962-70, 2009, with permission from Elsevier

To better understand the effect of corrugation geometry on heat transfer and pressure drop, Kongkaiatpaiboon *et al.* [176] investigated tubes of different spiral numbers and depth ratios at different Reynolds numbers at turbulent flow regimes. Such spirals induce swirls near the tube's wall and core. Two competing factors that affect the overall heat transfer of the tubes are the positive effect of stronger swirl intensity which increases with spiral numbers and the negative effect of the difficulty of fluid motion into grooves which increases with depth ratio. Overall, the Nusselt number augmentation varied between 0.82-2.16 times that of a plain tube. Whether the augmentation had a positive or negative effect depended on the combined effect of spiral numbers and depths ratios at a given instance. Another attempt by Nuntadusit *et al.*, [177] used perforated

ribs to enhance heat transfer. Ribs of various perforation/hole inclination angle and hole locations were studied (Fig. 19). Performed at $Re_{rh} = 60,000$ (where $Re_{rh} = U r h / \nu$), it was reported that the perforated ribs enhanced heat transfer drastically. Moreover, it was noted that inclined perforated ribs provided a higher heat transfer immediately downstream from the ribs with 0-degree inclination or with no holes at all. This was attributed to the fact that inclined holes suppress recirculation more efficiently and enable flow reattachment nearer to the rib which produces strong shear flow on the surface behind each rib [178]. This, in turn, disrupts the thermal boundary layer and thus improves heat transfer [179,180].

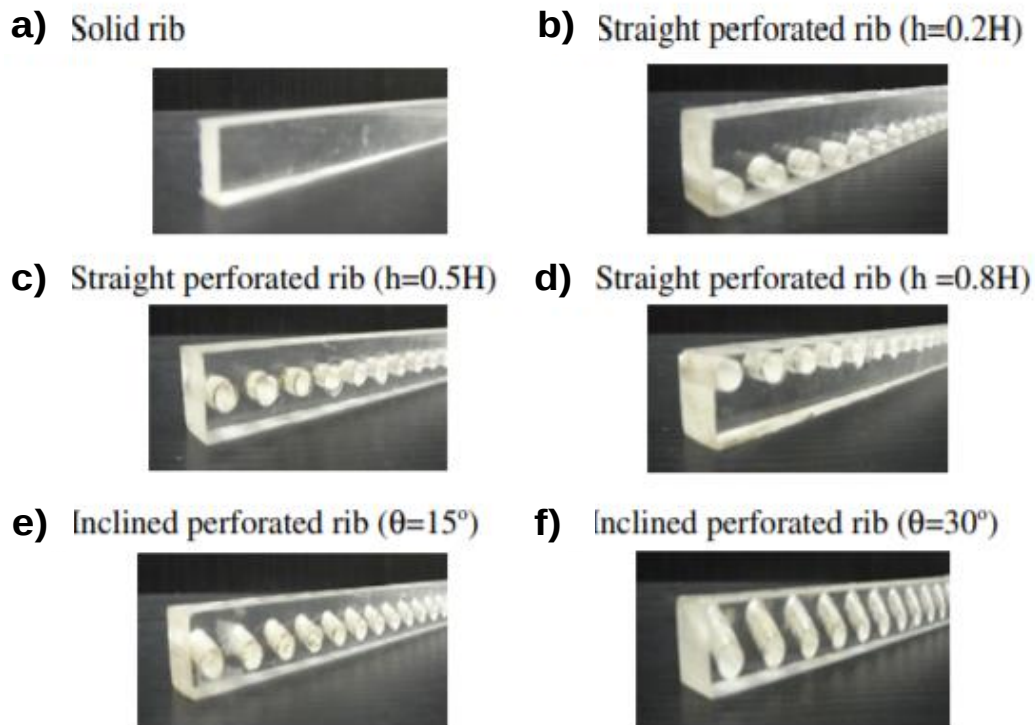


Figure 19. Perforated rib Images

Perforated ribs. Images of a (a) solid rib, (b) straight perforated rib, (c) straight perforated rib, (d) straight perforated rib, (e) inclined perforated rib ($\theta = 15^\circ$), and (f) inclined perforated rib ($\theta = 30^\circ$) [177]. Reprinted from Int Commun Heat Mass Transf, 39, Nuntadusit et al., Thermal visualization on surface with transverse perforated ribs, 634-9, 2012, with permission from Elsevier

Chen *et al.* [181] investigated numerically and experimentally the effect of outward corrugated tubes (Fig. 20) on heat transfer and pressure drop over a range of Reynolds numbers.

When comparing the tube side with the shell side, it was reported that the shell side had a higher Nusselt number reaching a maximum of 1.7. The maximum increment for the frictional factor and enhancement factor (as defined by Eq. (20)) was 1.13 and 1.26, respectively. This is due to the fact that vortex breaking occurs at the shell side. Moreover, the thickness of the boundary layer is smaller

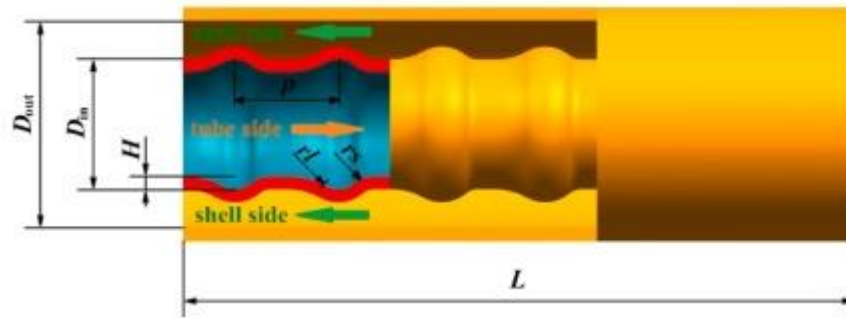


Figure 20. Outward corrugated tubes

Outward corrugated tubes [181]. Adapted from Nucl Eng Des,350, Chen et al., Turbulent heat transfer enhancement in a heat exchanger using asymmetrical outward convex corrugated tubes, 78-89,2019, with permission from Elsevier

From the above studies, it can be concluded how advantageous ribbed tubes are to the heat transfer augmentation. Corrugated tubes are also cheap and easy to manufacture. Furthermore, unlike other methods such as inserts, having corrugated tubes would not require as much frequent maintenance. However, it is important to note that other passive techniques may be able to provide higher heat transfer augmentation than corrugations.

6.1.2.6 Conical Rings

Another method to increase the heat transfer coefficient is through the use of conical rings (CRs). Such devices require no external power and increase turbulence by causing spiral flows which enhance mixing and heat transfer due to disruption of the boundary layer. A study of CRs was

carried out by Promvonge [182] where different ring-to-tube diameter CRs were mounted over a test tube. Air was used with $6,000 < Re_d < 26,000$. It was reported that Nu_d increased by 1.97 to 3.33 times depending on the way the CR was arranged (Fig. 21a-c).

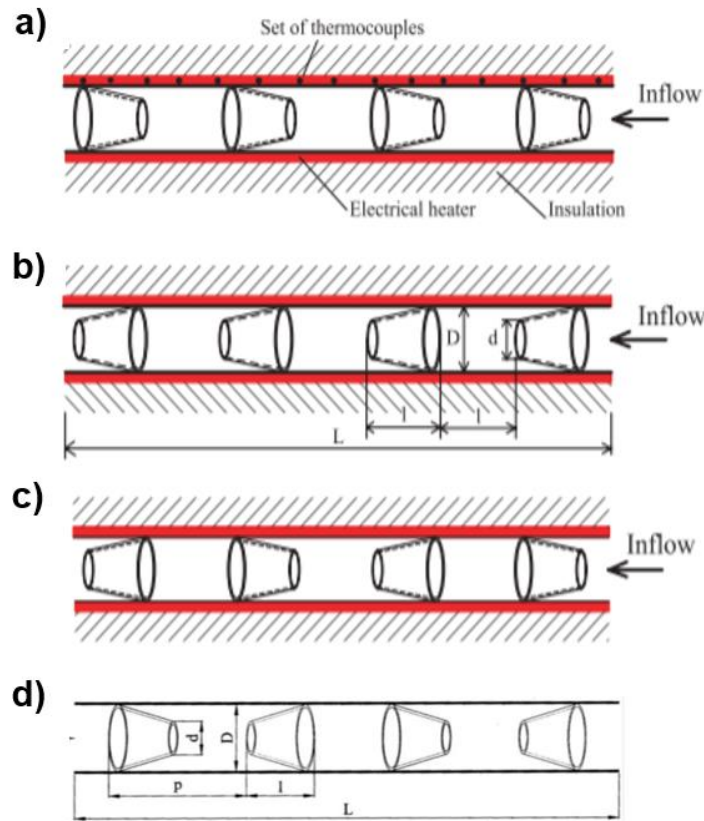


Figure 21 . Conical ring arrangements

Three conical ring arrangements are shown the (a) DR array, (b) CR array, and (c) CDR array [182]. Reprinted from Energy Convers Manag, 49, Promvonge, Heat transfer behaviors in round tube with conical ring inserts, 8-15,2008, with permission from Elsevier. (d) alternative CDR array orientation [183]. Adapted from Int J Therm Sci, 137, Ibrahim et al., A computational study of heat transfer analysis for a circular tube with conical ring turbulators, 138-60, 2019, with permission from Elsevier

A similar computational study conducted by Ibrahim *et al.* [183] investigated the effect of CRs at orientations as illustrated in Fig. 21a, 21b, 21d on the heat transfer enhancement, and frictional factor. Similar to the study by Promvonge, all the CRs were found to augment the Nusselt number. The augmentation increase with Reynolds number but decrease with increasing CR

diameter and pitch ratio. Nonetheless, the frictional factor was also found to decrease with CR diameter and pitch ratio. Out of the orientations studied, the authors reported that at a pitch ratio of 2, DR orientation had the highest Nu_d on average. Kongkaitpaiboon *et al.*, [179] utilized perforated conical rings (PCRs) to attempt to enhance heat transfer even further (Fig. 22a). The study was conducted over different pitch ratios and number of holes for $4,000 < Re_D < 20,000$ where D represents the inner tube diameter. Since the PCRs can disrupt the thermal boundary layer, the heat transfer rate rose by 137% compared to plain smooth tubes. Furthermore, when compared to conventional CRs with no perforations, it was noted that PCRs offered lower heat transfer enhancement. However, the friction factor was greatly reduced. As a result, the overall thermal performance of PCRs is superior. It was also concluded that the thermal performance increases with the number of holes and decreasing pitch ratio. A similar numerical study carried out by Nakhchi *et al.* [184] investigated the effect of different PCRs on heat transfer and frictional pressure drop. It was reported that increasing perforation holes from 4 to 10 caused an increase in the Nusselt number by 35.48%. This is because holes stimulate recirculating flows near the tube wall. This flow would, in turn, disrupt the thermal boundary layer and enhance heat transfer. However, with increasing hole numbers a decrement in the frictional factor. To reap the benefits of both TTs and CRs, Promvonge and Eiamsa-ard [185] incorporated these two methods together inside a single tube (Fig. 22b,c). The investigation was carried out in turbulent conditions ($6,000 < Re_D < 26,000$ where D represents the inner tube diameter) with air as the working fluid. Depending on the twisted tape ratio, it was reported that the Nu_d of the ring-tape system was 4% to 10% higher compared to the CR alone and higher 350% to 367% higher compared to a plain tube. However, the friction factor of the system was reported to be 5% to 12% higher than the CRs tube.

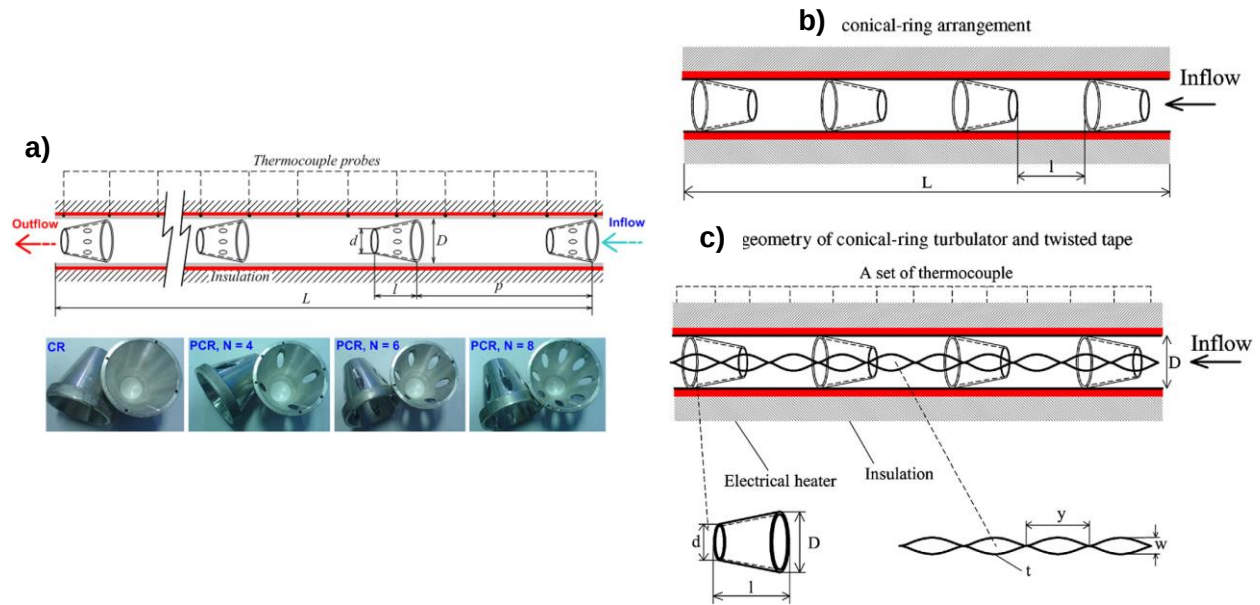


Figure 22. Perforated conical ring arrangements

(a) Schematic of perforated conical rings [179]. Reprinted from *Int Commun Heat Mass Transf*, 37, Kongkaiatpaiboon et al., Experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings, 560-7, 2010, with permission from Elsevier. Schematics of (b) CR and (c) CR with TT [185]. Reprinted from *Int Commun Heat Mass Transf*, 34, Promvonge and Eiamsa-ard, Heat transfer behaviors in a tube with combined conical-ring and twisted-tape insert, 849-59, 2007, with permission from Elsevier

Conical rings present yet another way to augment heat transfer. Although CRs are susceptible to the same advantages and disadvantages of other inserts discussed previously, the added benefit of CRs would lie in its maintenance. Instead of replacing an entire insert if damage were to occur, individual CRs could be instead replaced. However, installing and removing CRs would prove to be more demanding than other inserts.

6.1.2.7 Vortex Rings

Vortical devices such as vortex rings or cubes [186] augment heat transfer through generating counter-rotating vortices which enhance mixing and induce turbulence. Promvonge *et al.* [187] studied the effect of vortex rings (VR) of different ring width ratios and relative ring pitch ratios

at a 30-degree inclination over a range of Re values ($5,000 < Re_D < 26,000$ where D represents the inner tube diameter) as shown in Figure 23. Using air as the working fluid, it was reported that f and Nu_d increased by 35.1 to 36.5 and 3.6 to 4.3 times, respectively, when compared to smooth tubes.

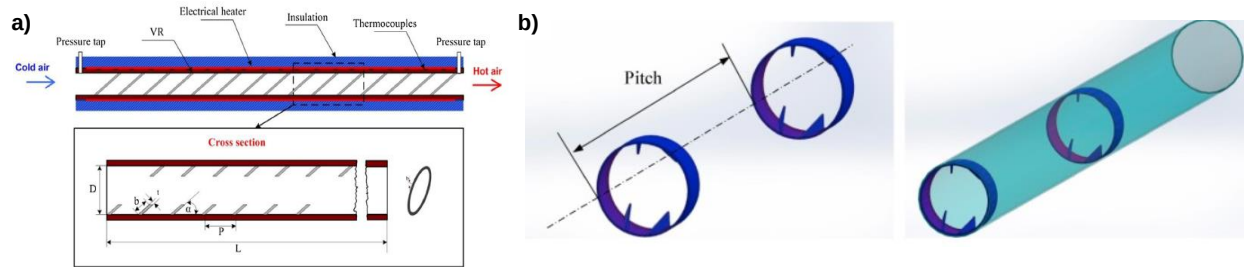


Figure 23. Vortex rings and generator pairs

(a) Vortex rings [187]. Reprinted from Appl Therm Eng, 62, Promvong et al., Thermal performance enhancement in a heat exchanger tube fitted with inclined vortex rings, 285-92, 2014, with permission from Elsevier and (b) Vortex generator pairs [188]. Reprinted from Appl Therm Eng, 153, Zhai et. al, Parametric study of major factors affecting heat transfer enhancement in a circular tube with vortex generator pairs, 330-40, 2019, with permission from Elsevier

Zhai *et al.* [188,189] conducted a similar study with vortex generators in the shape of the delta winglet. Using air as the working fluid, the investigation was conducted over a range of pitch ratios and winglet configurations at two different flow directions (downstream and upstream). It was noted that having the rings amounted to an increase in the Nusselt number and friction factor. However, the rise in Nusselt number overshadowed the increase in friction as the enhancement factor (as defined by Eq. (20)) reached a maximum of 1.44 for a pitch ratio of 9.6. It was also noted that the enhancement factor was higher for lower Reynolds number regions. Such regions are not characterized by high turbulence, therefore the effect of having the vortex generator pairs is more prominent. When comparing flow direction, it was further noted that upstream flow had an enhancement factor of 30-33% higher than that of downstream flow suggesting that the ring is more effective in generating turbulence in that orientation. Thianpong *et al.* [190], used metallic

twisted ring turbulators (Fig. 24). Twisted rings lengthen flow paths due to the helically twisted fluid motion and improve mixing by inducing secondary fluid motion.

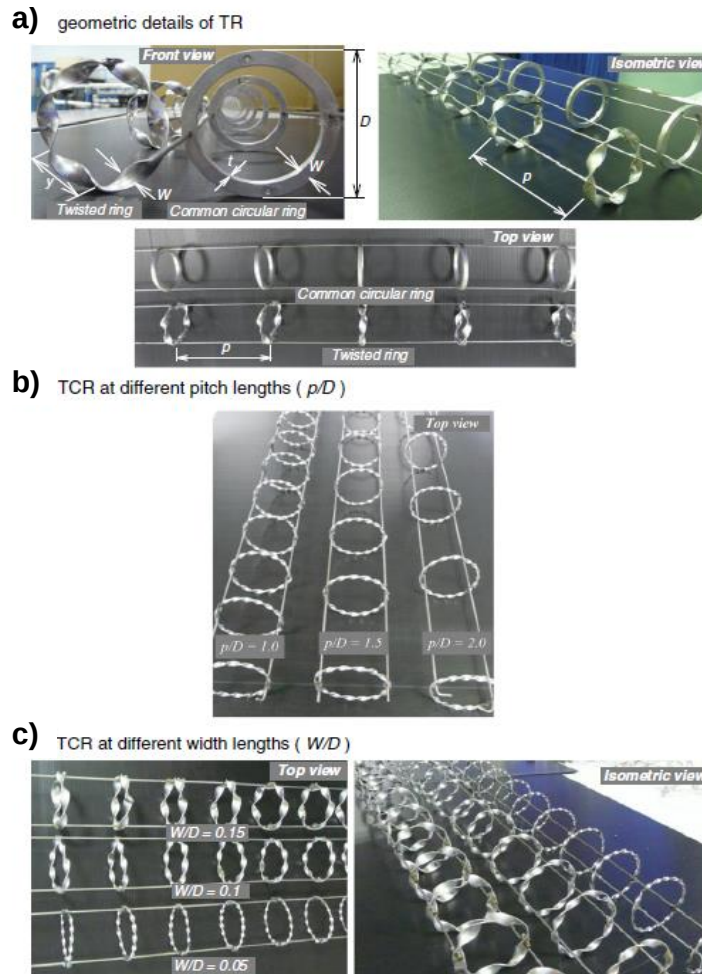


Figure 24. Twisted rings.

Twisted rings. Images showing (a) geometric details, (b) pitch lengths, and (c) width lengths [190]. Reprinted from Int Commun Heat Mass Transf, 39, Thianpong et al., Thermal performance evaluation of heat exchangers fitted with twisted-ring turbulators, 861-8, 2012, with permission from Elsevier

Different rings of varying pitch lengths and width lengths were studied. It was found that twisted rings provided lower heat transfer enhancement and friction factor than CRs. However, the performance was found to increase at higher width ratios and decreasing pitch ratios. Twisted rings are more feasible in terms of energy-saving at low Re and width ratio. Eiamsa-ard *et al.*, [191]

incorporated both TTs and turbulator rings inside one tube. Three pitch ratios were used for the ring with three different twist ratios for the tape. Using air as the working fluid, the experiment was conducted at a uniform wall heat flux conditions for turbulent flow regimes. For the range mentioned previously, the Nu_d and f increased by 25.8% and 82.8%, respectively, compared with tubes that had rings alone.

Overall, it can be observed that turbulator rings are effective in causing enhancement to heat transfer. However, similar to TTs, it can be noted that the effectiveness of this method is highlighted in less turbulent environments. Consequently, it may be worthwhile incorporating it with other passive techniques to better cater to a larger range of flow regimes.

6.1.3 Active Methods

Up until this point, all preceding enhancements can be categorized as passive enhancement techniques. As highlighted in the introduction, the main difference between the use of active and passive methods is the presence of external input (*i.e.* power, force, field, etc...). Although active methods have the advantage of enabling on-demand control, the added complexity and cost make these techniques less common [192]. Like passive methods, several active heat transfer enhancement techniques have been developed.

6.1.3.1 Deployment of External Forces (Fields):

6.1.3.1.1 Electric Field

Also known as *Electrohydrodynamics* (EHD), EHD uses an electric field to disturb the boundary layer of a dielectric fluid. Electric fields can be used to induce radial fluid motion causing the desired disturbance in the boundary layer. A study conducted by Fernandez and Poulter [193]

applied a DC voltage of 30kV to a wire introduced along the center of an annular tube containing electrohydrodynamically-enhanced oil (Shell Diala Oil “B”) to induce strong radial motion. The copper tube had an internal diameter of 25.4 mm and a length of 1 m. At the inlet, cold oil enters with high viscosity, impeding high electric interaction. However, as the oil progresses downstream, it gets heated and so electric motion starts to increase and dominate the heat transfer mechanism. This behavior continues until the oil reaches the same temperature as the wall of the tube, after which the heat transport rate starts to sharply reduce.

It was reported in the experiments carried out that the heat transfer increased by more than 20 times in fully developed laminar flow while the pressure drop increased by only 3 times. The higher pressure drop is due to the distortion of the axial laminar flow caused by strong outward velocities and sluggish return flow. Since the electric field converts electric energy into kinetic energy, it can be concluded that the higher the voltage, the higher the induced velocities and enhancement ratio.

To enhance the heat transfer coefficient of airflow, Tada *et al.* [194] used EHD with the addition of hollow glass particulates to the fluid. The solid particles suspended in the air were controlled using the electric field. The Coulomb forces acting on the particles induce collisions with the channel walls which enhance heat transfer due to the motion of the charged particles move from the hot tube wall to cooler regions of gas flow, carrying with them thermal energy. Consequently, it was concluded that the heat transfer coefficient increases with increasing applied voltage. Note, the force on the glass particles was generated via electrophoresis, rather than the more commonly observed EHD forces such as dielectrophoresis [195]. Wangnipparnto *et al.* [196], studied the efficacy of electric fields on thermosyphon heat exchanger performance. An observable increase in the heat transfer coefficient was only reported for applied voltages greater than 15.5

kV. At $Re_{OD} = 58$, and 17.5 kV, the heat transfer coefficient was found to increase by 15%. However, this dropped to 5% when Re_{OD} was increased to 230. The ratio of power consumption to heat transfer enhancement was found to be 0.5 to 20%. Lai [197] studied EHD enhanced natural convection in an air-filled conduit subjugated to a nonsymmetric electric field. It was observed that for a given aspect ratio of 5.0, the air filling the conduit was ionized by corona discharge from the wire. This forces air to be rushed towards the conduit wall which enhances heat transfer. Overall, it was found that the heat transfer enhancement increased with the supplied voltage. Further, it was noted that the optimum location for the electrode inside the conduit was near the leading edge of the heat transfer surface as this aids in disturbing the thermal boundary layer as soon as it starts to develop. Deylami [198] extended the study of EHD forces to rectangular corrugated channels. It was reported that, depending on the Reynolds number and the size of the corrugations, the enhancement factor (Eq. (20)) varies between 0.8-1.08 which demonstrates that EHD forces in a corrugated rectangular channel create very large pressure drops. This is due to the intense recirculating flow between corrugations.

From the above studies, it can be concluded that EHD has the potential to provide immense heat transfer augmentation. However, caution has to be exercised as the geometry of the channel and the position of the wire have a great effect on how the results pan out to be. Furthermore, the large potential difference required poses an electrical hazard that needs to be taken into consideration.

6.1.3.1.2 Magnetic Field

Also known as *Magnetohydrodynamics* (MHD), a magnetic field can be used to disturb the boundary layer. If the working fluid is a ferrofluid, magnetic fields can be used to induce fluid

flow causing the desired disturbance in the boundary layer. A study conducted by Li and Xuan [199] used a solenoid electromagnet to create a magnetic field in which a ferrofluid passes through. The fluid also passes over a heated wire and experiences an increase in heat transfer attributed to the Kelvin force causing magnetic nanoparticles in the fluid to collide with the heated wire. A similar study conducted by Goharkhan [200] analyzed the effect of a constant and alternating magnetic field on laminar forced convective heat transfer of water-based magnetite flowing inside a tube. Heat transfer enhancements of 31.4% and 18.9% were observed for alternating and constant magnetic fields of intensity 500 G. In the absence of magnetic fields, an enhancement of 13.5% was reported due to the presence of nanoparticles. The enhancement for alternating magnetic fields was attributed to the disruption of the thermal boundary layer and increased mixing. The enhancement in the constant magnetic field case can be attributed to the movement of the nanoparticles towards the walls of the tube, increasing its thermal conductivity and heat transfer. While the previous studies were conducted for horizontal tubes, Aminfer *et al.* [201] conducted a study on a vertical tube with the intent of investigating the effect of positive and negative non-uniform magnetic fields on heat transfer. Using kerosene and 4 vol% Fe_3O_4 as the working fluid, numerical simulations showed that having a negative magnetic field increases Nu_d due to the induced upward Kelvin force which acts analogously to Buoyancy. Furthermore, the negative magnetic field helps to suck the ferrofluid from the inlet, increasing the pumping power. Opposite effects were observed with positive magnetic fields. A point to consider is that the magnetic field is more effective in laminar conditions where a weaker magnetic field can disturb the flow. Lajvardi *et al.* [202], investigated the effect of magnetic fields on the heat transfer of a ferrofluid having Fe_3O_4 nanoparticles flowing through a copper tube. The experiment was conducted using different nanoparticle concentrations and was repeated with different magnetic field intensities. It

was noted that as the concentration of the nanoparticles increases so does the heat transfer augmentation. The same trend was also found with increasing magnetic field strength. The observed behavior was attributed to the fact that by either increasing the nanoparticle concentration or the magnetic field intensity or both, the chances of the nanoparticles interacting with the tube wall increases, which increases the effective thermal conductivity and thus heat transfer. In an attempt to directly utilize MHD in electronic applications, Izadi *et al.* [203] studied the impingement cooling of a porous metallic CPU cooling unit through water-alumina nanofluid under the effect of a magnetic field. The magnetic field is used to control the fluid flow of the nanofluid. A positive correlation was reported to exist between the strength of the magnetic field and impingement cooling demonstrating how effective MHD can be in electronics cooling. Abedini *et al.* [204] investigated numerically the effect of MHD on heat transfer in a baffled C-shaped enclosure with a heated surface. Using water- Fe_3O_4 as the nanofluid, it was reported that increasing either the aspect ratio of the enclosure or the volume fraction had a positive effect on the Nusselt number and heat transfer.

MHD in many ways is similar to the EHD heat transfer technique discussed in the previous subsection. It utilizes induced motion that aids in Nusselt number augmentation. Similar to EHD and other active techniques, MHD has the ability to control the extent of heat transfer enhancement. While passive techniques are able to control the amount of heat transfer enhancement through geometrical modification, once the geometry is set, the freedom to control heat transfer enhancement is no longer available. That being said, MHD does not lend itself to be used in all industries due to the need for ferroic fluid to be present. While this might not be an issue for electronic applications as highlighted earlier, it would pose more of an issue in FMCGs.

6.1.3.2 Fluid Vibration

Also known as the pulsatile flow method, fluid oscillation can cause a disturbance in the boundary layer. Nishimura and Matsune [205] numerically investigated the flow in a tube having sinusoidal wavy walls. The wavy geometry induces vortices that shrink and expand during acceleration and deceleration, respectively. This induces turbulence during deceleration reaching a maximum when the flow has zero velocity. It was observed that as the frequency increases, the vortex strength increases up until a certain threshold above which the opposite effect occurs. The pulsatile flow method of heat transfer enhancement is usually utilized for very viscous liquids. A similar study carried out by Khosravi-Bizhaem *et al.* [206] investigated pulsating flow through helically coiled tubes. Using distilled water as the working fluid, the effect of a range of different coil curvatures and Reynolds number on the heat transfer and pressure drop was investigated. It was reported that, on average, the pressure drop showed a 3%-7% increase to that of steady flow. Moreover, the highest augmentation to heat transfer was reported for lower Reynolds numbers since pulsating flows affected the laminar boundary layer immensely. Overall, the average heat transfer enhancement and the average heat transfer enhancement (Eq. (20)) at 4 Hz frequency was reported to be 19% and 1.16, respectively.

Jin *et al.* [207] studied the heat transfer enhancement of pulsating flow in a channel with triangular grooves (Fig. 25a) in laminar flow regimes ($270 \leq Re_{he} \leq 910$, where $Re_{he} = heU / \nu$). Heat transfer enhancement of 350% was reported at $Re_{he} = 270$ and $St = 0.34$ ($St = Nu/Re.Pr$). This enhancement is negatively correlated to Re_{he} . Furthermore, the observed enhancement is due to the mixing, vortex generation, expansion, and ejection caused by the pulsating flow. Another investigation by Jafari *et al.* [208] applied the oscillation to corrugated channels (Fig. 25b). The numerical study considered different Re_L ($Re_L = 50, 100, 150$) and $Pr = 3.1$. The simulation results

show that the oscillating velocity affects the flow configuration and thermal fields which lead to heat transfer enhancement. The heat transfer rate was reported to increase linearly with the amplitude of the oscillating velocity where the best performance was evident at higher Re_L . Akdag *et al.* [209] studied numerically the heat transfer enhancement of pulsating nanofluids in a laminar flow regime inside a wavy copper tube. The alumina (Al_2O_3) dispersed in water was simulated over a range of pulsating frequencies and amplitudes. Results showed that pulsating flow techniques do demonstrate the ability to provide large heat transfer augmentation and are relatively simpler to implement. In addition to showing a positive heat transfer enhancement, pulsating flow also had the advantage of preventing nanoparticle sedimentation. The heat transfer enhancement is found to increase with nanoparticle volume fraction and pulsation amplitude. Further, Akdag *et al.* [210] investigated numerically the heat transfer enhancement of pulsating nanofluids inside a triangular wavy tube. The CuO dispersed in water was simulated over a range of pulsating frequencies and amplitudes the heat transfer. Similar to previous investigations, it was shown that heat transfer performance increased with pulsating amplitude at low frequencies. This is due to the mixing between the main flow and the hot fluid near the tube walls.

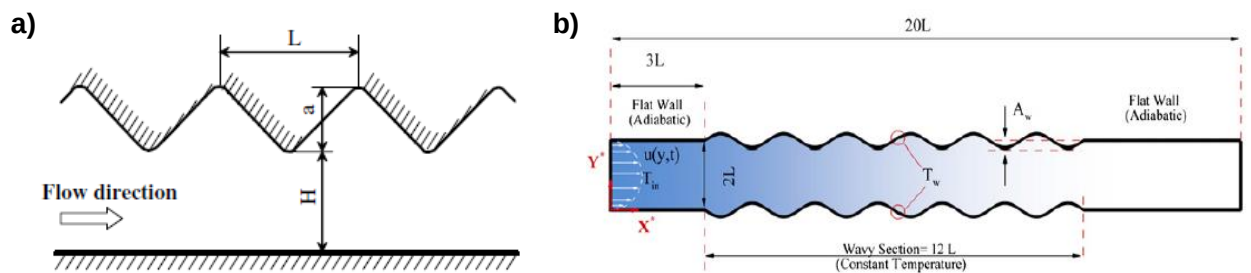


Figure 25. Triangular grooved channel and corrugated tube

(a) Schematic of a channel with triangular grooves [207]. Reprinted from Int J Heat Mass Transf, 50, Jin et al., Effects of the pulsating flow agitation on the heat transfer in a triangular grooved channel, 3062-71, 2007, with permission from Elsevier and (b) Schematic of a corrugated tube [208]. Reprinted from Int Commun Heat Mass Transf, 45, Jafari et al., Pulsating flow effects on convection heat transfer in a corrugated channel: A LBM approach, 146-54, 2013, with permission from Elsevier.

The above studies demonstrate how pulsating flow can show promise in providing an enhancement to heat transfer more simplistically than other active techniques. In addition, it emphasizes how nanoparticles dispersed in the fluid also affect heat transfer[186]. However, it is important to note that it is more geared towards laminar flow regimes. As a result, coupling such a technique with other passive or active techniques may be required. Unlike passive techniques, active techniques are rarely integrated. This could be an interesting area for future research.

6.1.3.3 Mechanical Power

The third active method uses mechanical means to disturb the thermal boundary layer. For example, an investigation carried out by Morris and Rahmat-Abadi [211], used rotation to induce turbulence inside the tube. A tube containing different types of internal ribs was rotated orthogonally to investigate the effect of convective heat transfer. The three different rib geometries were used with dimensions outlined in Table 4. Although, the bore diameters for each geometry is the same, the rib location, length, and distance between ribs were varied. The test section was made of machinable ceramic glass having a thermal conductivity (k) of $1.3 \text{ Wm}^{-1}\text{K}^{-1}$.

Table 4. The dimension of ribs for test geometries [211].

Dimension (mm)	Test geometry 1	Test geometry 2	Test geometry 3
Plain bore diameter	10	10	10
Bore at rib location	6	7	8
Length	130	97.5	65
Distance between ribs	20	15	10

When the test was conducted it was noted that some rib geometries induced a heat transfer enhancement by as much as 300% when compared to smooth tubes with zero rotation. It was also reported that the trailing edge had higher heat transfer enhancement compared to the leading one due to secondary flows induced by the Coriolis force.

A similar study by Wright *et al.* [212] used a rectangular cross-section (AR 4:1) with rotation. The channel contained different rib types including angled, discrete angled, V-shaped, discrete V-shaped, W-shaped, and discrete W-shaped. The ribs were placed at the leading and trailing ends of the channel with a 45-degree angle. Experimenting with turbulent conditions ($10,000 < Re_h < 40,000$ where h is the hydraulic diameter), it was found that W-shaped and discrete W-shaped provided the best heat transfer performance and the largest pressure drop. On the other hand, the V-shaped and discrete angled ribs resulted in the lowest pressure drop. Taking into account the increase in heat transfer performance and the rise in pressure drop, it was concluded that the best overall thermal performance was for discrete V-shaped ribs and discrete W-shaped ribs. Tao *et al.* [213] studied a rotating wedge-shaped channel with lateral fluid extraction. The experiment was conducted over $5,100 < Re_h < 21,000$ at five different rotational speeds between 0 to 1000 rpm and five different channel orientations. From the experiment, it was concluded that channel orientation has a significant effect on heat transfer characteristics. Another investigation by Qiu *et al.* [214] studied the effect of rotation of a square U-duct (Fig. 26) on the heat transfer performance and pressure drop. The experiment was carried between $10,000 < Re_h < 70,000$. It was observed that the rotation effects were most dominant in the bending/turning region. As a result, the Nu_h was found to increase the most in that location. The increase in heat transfer was found to be linear with increasing rotation speed. In an attempt to integrate passive techniques with active techniques, Abotsi and Kizito [215] studied the heat transfer enhancement in an axially

Pressure taps on outer surface

Mesh

24

24

24

258

307

0

X direction

292

220

184

74

102

520.5 to axis

Teflon support

copper plates

Unit: mm

24

20

3

20

1

blind holes for thermocouples

Schematic of the U-shaped rectangular duct [214]. Reprinted from Int J Heat Mass Transf, 66, Qiu et al., Pressure drop and heat transfer in rotating smooth square U-duct under high rotation numbers, 543-52, 2013, with permission from Elsevier.

complex structures with AM has not only enabled efficient boundary layer mixing causing a higher heat transfer coefficient but also provided unprecedented surface area to volume ratio while maintaining low-pressure drop. Overall, developments have been quite diversified in terms of specification of the flow behavior and application. For this review, the current section is focused on applications where the thermal-hydraulic performance of a single-phase flow is enhanced by deploying complex geometries using additive manufacturing. Collins *et al.* [216] studied the heat transfer in straight and manifold aluminum alloy (AlSi₁₀Mg) channels produced by laser sintering. Using water as the working fluid, Collins *et al.* [216] characterized the thermal and hydrodynamic performance of single-phase flow in channels with a hydraulic diameter of approximately 500 μm . The study showed that pressure drop can be determined by classical hydrodynamic theory despite the early transition to turbulent flow even at low Reynolds numbers due to surface roughness. However, the thermal performance was found to be 15-20% lower than that of the established values for channels of comparable sizes. This reduced thermal performance was due to the lower thermal conductivity of the AM parts from the presence of small scale voids in the material bulk, in contrast to fully dense materials produced by conventional manufacturing.

Many times, AM has been used to develop multi-functional components. As highlighted earlier, the resulting structure from the AM process results in relatively poorer thermal and mechanical properties due to higher porosity. Consequently, one potential application for AM is membrane development with a high heat transfer area. Collins *et al* [217], investigated the performance of a permeable membrane microchannel which functioned as a heat sink for electronics cooling. To increase the effectiveness, water as a working fluid was pumped through the thin and porous walls. The main advantage of AM is its ability to realize and create heat transfer channel configurations which are not possible using conventional manufacturing approaches. In a

separate analytical study, Kirsch and Thole [218,219] explored the potential of AM to produce wavy channels. The higher surface roughness of AM components was found to play a role in performance. While in many cases, such surface features can be disadvantageous to performance due to higher pressure drop, for heat transfer behavior, the presence of surface roughness facilitates local flow mixing and boundary layer distortion and thus augments the heat transfer coefficient. Multiple follow up studies have focused on the deployment of macro-scale turbulator or flow mixers. Calignano *et al.* [220], experimentally investigated the performance of gyroid structures with variable porosities deployed in channels. The deployment of such extended surfaces increases wetted surface area, enhancing heat transfer. These extended surfaces can vary in structure from simple pin fins to complex cross-sections and twisted tape designs manufactured in channels. [221,222].

Tiwari *et al.* [223], conducted an experimental study to analyze the performance of a unique shell and tube configuration (Fig. 27). Tubular manifolds produced by the filament deposition AM method enabled unique flow distribution which resulted in a maximized thermal performance with acceptable pressure drop.

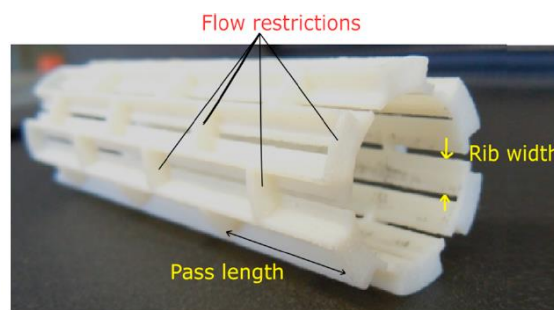


Figure 27. 3D printed manifold

Isometric view image of the 3D printed manifold used for the microchannel heat exchanger of pass length 25.4 mm [223]. Reprinted from Appl Therm Eng, 147, Tiwari et al., Development of an additive manufacturing-enabled compact manifold microchannel heat exchanger, 781-8, 2019, with permission from Elsevier

It is worth noting that along with the aforementioned sub-component developments where the AM process was used to realize enhanced heat transfer in individual channels, multiple studies have been focused on the development of full-scale components. Due to the potential increase in the complexity enabled with AM, various manufacturing approaches such as binder jetting, fused deposition, and laser and electron beam sintering have been used. Most heat exchanger developments have focused on air-to-liquid heat exchanger applications. Using water as the working fluid, Arie *et al.* [224] developed heat exchangers for air cooling or heating applications. To account for the effects of the base metal, the heat exchanger samples were produced using multiple metals. Due to manufacturing limitations, the authors were not able to achieve the desired level of geometrical accuracy in the printed samples. Regardless, an obvious improvement in the thermal-hydraulic performance was observed. In a related study, the same group evaluated the performance of polymer heat exchangers in air-cooled condensers produced by fused disposition AM. Polymers were used to lower the cost and weight of the heat exchanger [225,226].

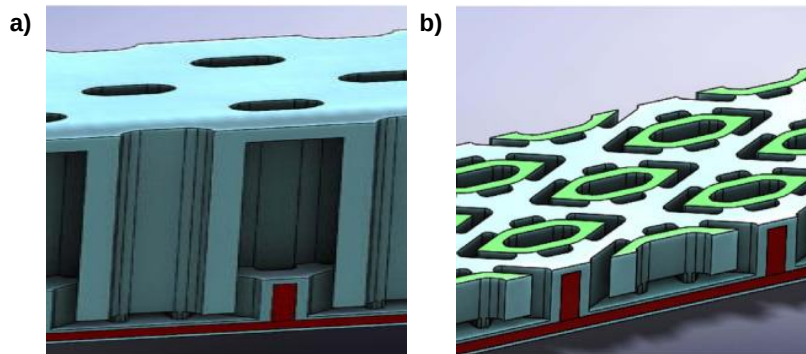


Figure 28. Single cell and micro-channel internal

(a) CAD images of a single cell in which liquid jets enters from the top and exit by moving sideways to a neighboring chamber and (b) Cutaway across the base of the internal microchannel. The jet orifices are $200 \times 70 \mu\text{m}$ diamond-shaped. Color scheme: Teal Green is cut section, Crimson Red is Copper and Aqua Blue is valloy [227].

Reprinted from Appl Therm Eng, 136, Robinson et al., A single phase hybrid micro heat sink using impinging micro-jet arrays and microchannels, 408-18, 2018, with permission from Elsevier

Another class of studies has considered AM to achieve multi-functional flows to enable higher heat transfer performance. Robinson *et al.* [227] developed a unique channel configuration where electrochemical AM was used to develop a hybrid heat sink (Fig. 28). The authors were able to assemble a novel system that successfully combined jet impingement and microchannel flow improvement in the heat sink.

The unprecedented compactness offered by AM is critical for many applications such as aerospace and marine systems where the volumetric and gravimetric power density [228] of the thermal management system is essential [229,230].

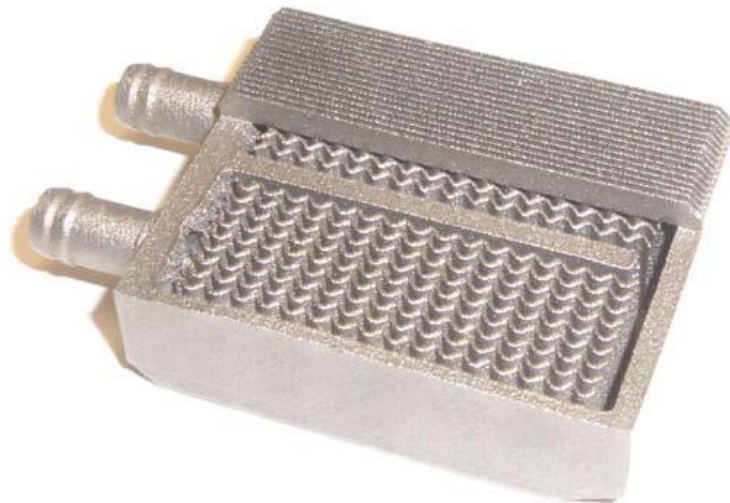


Figure 29. A miniature cooler

Picture of a miniature cooler developed by metal AM of dimensions 46x59x15 mm [231]. Reprinted from *Assem Autom*, 31, Neugebauer *et al.*, Additive manufacturing boosts efficiency of heat transfer components, 344-7, 2011, with permission from Emerald

To achieve a highly compact cooling solution, Neugebauer *et al.* [231] developed a miniature metallic heat exchanger using AM (Fig. 29). The authors conducted a comprehensive survey of various manufacturing options and concluded that AM processes such as laser beam

melting can achieve such highly complex structures. The method also manufactures prototypes at a reasonable cost and timescale when compared to conventional manufacturing approaches for metal parts.

In another relevant study, Kwon *et al.* [232], deployed a static mixer in channels to achieve higher heat transfer coefficients (Fig. 30). Using water as the working fluid, he noted that despite the noticeable increase in pressure drop, the heat transfer rate doubled when compared to flow without static mixers.

To conclude, AM techniques have the potential to revolutionize heat transfer component development and performance. Although the implications to system performance are profound, such processes have been limited to specific applications including electronic cooling, thermal management of turbomachines, and cooling in the power generation sector. Overall development in this regard can be broadly divided into three categories:

- (i) Localized enhancement - where the surface roughness causes higher heat transfer rates,
- (ii) Extended surfaces and mixers – where integrated microscale features in the channels provide an improved heat transfer area and higher heat transfer rate,
- (iii) Full-scale heat exchangers - where full-scale devices with unprecedented compactness can be used in applications including electronic cooling, HVAC&R, and process industries.

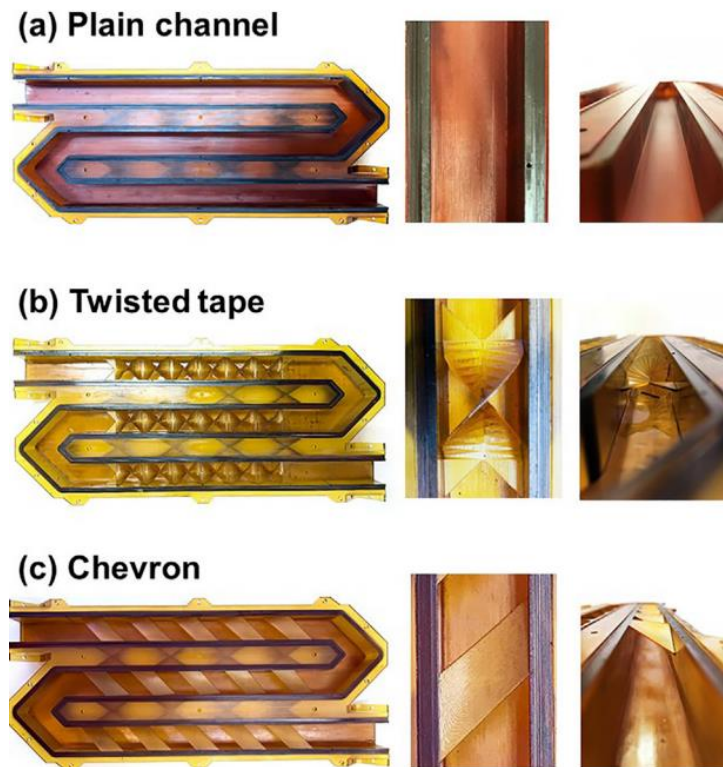


Figure 30. Plain, twisted tape and chevron mixers rectangular channel

Images of (a) rectangular channel with no static mixers (plain channel), (b) rectangular channel twisted tape mixers, and (c) rectangular channel with chevron mixers [232]. Reprinted from Int J Heat Mass Transf, 137, Kwon et al., Heat transfer enhancement of internal laminar flows using additively manufactured static mixers, 292-300, 2019, with permission from Elsevier

6.1.5 Summary of Single-Phase Enhancement Techniques

To summarize the work presented to this point, Table 5 highlights the most important findings. Furthermore, Table 6 includes the efficiency correlations used referenced previously along with proper citations.

Table 5. Summary of single-phase enhancement techniques.

No.	Type of enhancement	Passive /Active	Researchers	Fluid	Results*
1	Conventional TTs	Passive	Manglik & Bergles [123]	Water and ethylene glycol	Increase in heat transfer coefficient with the maximum being in the laminar region. Increase in the pressure drop by 3.9 to 7.5 times.
2	Delta winged TTs	Passive	Eiamsa-ard <i>et al.</i> [132]	Water	On average, DWTs provided 1.1 to 2.55 increase in Nu. On average, the friction factor increased by 2.5 to 7.02 times.
3	Triangular-Rectangular-Trapezoidal TT wings	Passive	Wongchareea and Eiamsa-ard [135]	Water	The maximum reported increase in heat transfer rate is 2.84. The maximum reported increase in friction factor is 8.02.
4	Varying TT thickness	Passive	Esmailzadeh <i>et al.</i> [136]	γ -Al ₂ O ₃ /water	Maximum heat transfer occurred at thicker tapes. Pressure drop increases with thickness.
5	Two TTs	Passive	Bhuiya <i>et al.</i> [137]	Air	Heat transfer rate increased by 60 to 240%. The friction factor increased by 91 to 286% times.
6	Three TTs	Passive	Bhuiya <i>et al.</i> [138]	Air	Increased Nu by 3.85 times. Friction factor increased by 4.2 times.
7	Loosely fit TT with and without core	Passive	Eiamsa-ard and Promvonge [140]	Water	Heat transfer enhancement caused by not having a rod insert is larger than that with a core by almost 50%. Having the rod increased the friction factor by 50%.
8	3-D extended surfaces with TT	Passive	Liao and Xin [142]	Water, ethylene, glycol and ISO VG46 oil	Increased heat transfer enhancement and pressure drop.
9	TTs in a dimpled tube	Passive	Thianpong <i>et al.</i> [143]	Air	Increased heat transfer enhancement and pressure drop.
10	TTs in rectangular channels	Passive	Ray and Date [144]	Water	Local Nu augmentation varied between 1.02 and 1.17 times.
11	Helically coiled wire	Passive	Garcia <i>et al.</i> [152]	Water and water-propylene glycol mix	Nu increased by 2-2.8 times for turbulent flows. Friction factor increased by 2-9 times for turbulent flows.
12	Coiled wire with nanofluid	Passive	Behabadi <i>et al.</i> [158]	MWCNT-water nanofluid	Convective heat transfer coefficient increased, on average, by 85%. Pressure drop increased on average by 475%.
13	Coiled wire, not touching tube walls	Passive	Gunes <i>et al.</i> [160]	Air	Nu number and the friction factor both increase with decreasing p/d.
14	Squared cross-sectional wired coil	Passive	Promvonge [161]	Air	Nu increased by 1.8-2.6 times.
15	Annulus with an inner cylinder wrapped with a helical wire	Passive	Sreenivasulu and Prasad [163]	-	Nu increases depending on the geometry. Friction factor increases depending on the geometry
16	Coiled wire and TT together	Passive	Promvonge [166]	Air	Nu increased by 3-6.85 folds. Friction factor increased by 30-68 times
17	Zigzag shaped baffles	Passive	Sriromreun <i>et al.</i> [168]	Air	Nu and friction factor increased considerably.

Table 5 Summary of single-phase enhancement techniques. (Continue)

No.	Type of enhancement	Passive /Active	Researchers	Fluid	Results*
18	Louvred strip	Passive	Eiamsa-ard <i>et al.</i> [169]	Water	Forward orientation: Nu increased by 2.84 times and friction factor increased by 4.13 times. Backward orientation: Nu increased by 2.63 times and friction factor increased by 2.33 times.
19	Winglets	Passive	Torii <i>et al.</i> [170]	Air	Staggered Orientation: Heat transfer increased by 10 to 30%. Pressure loss was reduced by 55% to 34%. Inline Orientation: Heat transfer increased by 10 to 20%. Pressure loss was reduced from 15% to 8%.
20	Helical ribbed tubes	Passive	Naphon <i>et al.</i> [174]	Water	Increased heat transfer and pressure drop considerably.
21	DDIR tubes	Passive	Li <i>et al.</i> [175]	Water	Heat transfer is augmented by 100-120%. Increased pressure drop by 170-250%.
22	Perforated ribs	Passive	Nuntadusit <i>et al.</i> [177]	Air	Increased heat transfer drastically.
23	CR with different arrangements	Passive	Promvonge [182]	Air	Nu increased by 1.97-3.33 times depending on the way the conical ring was arranged. CR increased the friction factor considerably
24	PCRs	Passive	Kongkaitpaiboon <i>et al.</i> [179]	Air	Increase in heat transfer by 137%,
25	TTs and CRs	Passive	Promvonge and Eiamsa-ard [185]	Air	Nu increased by 3.5-3.67 times, The friction factor of the system was reported to be 5%-12% higher than that of CRs alone
26	Vortex Rings	Passive	Promvonge <i>et al.</i> [187]	Air	Nu increased by 35.1-36.5 times. Friction factor increased by 3.6-4.3 times.
27	Metallic twisted ring turbulators	Passive	Thianpong <i>et al.</i> [190]	Air	Nu and friction factor increased.
28	TTs and turbulator rings	Passive	Eiamsa-ard <i>et al.</i> [191]	Air	Nu increased by 25.8%. Friction factor increased by 82.8%.
29	Wire along the center of the tube (EHD)	Active	Fernandez and Poulter [193]	Shell Diala Oil "B"	Heat transfer increased by more than 2000%. Pressure drop increased by 3 folds.
30	Gas-solid suspension flow by applying an electric field (EHD)	Active	Tada <i>et al.</i> [194]	Air with glass particulates	Heat transfer is enhanced.

Table 5 Summary of single-phase enhancement techniques. (Continue)

No.	Type of enhancement	Passive /Active	Researchers	Fluid	Results*
31	Electric fields for thermosyphon heat exchanger	Active	Wangnipparnto <i>et al.</i> [196]	Air	Increase in heat transfer coefficient (depending on applied voltage and Re)
32	MHD	Active	Li and Xuan [199]	Fe ₃ O ₄ -water	Heat transfer increased.
33	Effect of a constant and alternating magnetic field (MHD)	Active	Goharkhan [200]	Fe ₃ O ₄ -water	Heat transfer increased up to 31.4% for alternating magnetic field and up to 18.9% for constant magnetic field.
34	The positive and negative magnetic field on a vertical tube	Active	Aminfer <i>et al.</i> [201]	kerosene and 4vol% Fe ₃ O ₄	Negative magnetic field increased the Nu but decreased with positive magnetic field
35	Investigate enhanced ferrofluid under magnetic field effect	Active	Lajvardi <i>et al.</i> [202]	A mix of ferric chloride, tri-sodium-citrate, water, and hydrazine hydrate	Heat transfer augmentation increased with nanoparticle concentration and magnetic field strength
36	Sinusoidal wavy walls	Active	Nishimura and Matsune [205]	Water	Increased heat transfer enhancement
37	Pulsating flow in a channel with triangular grooves	Active	Jin <i>et al.</i> [207]	Water	Heat transfer enhancement of 350% was reported at Re=270 and St=0.34.
38	Pulsating flow in a corrugated channel	Active	Jafari <i>et al.</i> [208]	-	Increased heat transfer enhancement and pressure drop
39	Pulsating nanofluid inside a wavy copper tube	Active	Akdag <i>et al.</i> [209]	Al ₂ O ₃ -water-based nanofluid	The heat transfer enhancement and pressure drop increased.
40	Rotation of tube with internal ribs	Active	Morris and Rahmat-Abadi [211]	Air	Heat transfer increased up to 300%.
41	The rectangular cross section was rotated with internal ribs	Active	Wright <i>et al.</i> [212]	Air	Increased heat transfer with the best overall thermal performance was found to be for discrete V-shaped ribs and discrete W-shaped ribs.
42	Rotating wedge-shaped channel with lateral fluid extraction	Active	Tao <i>et al.</i> [213]	Air	Channel orientation has a significant effect on heat transfer characteristics.
43	Rotation of a square U-duct	Active	Qui <i>et al.</i> [214]	Air	Bending/turning region has the highest increase in Nu Rotation reduces friction resistance.

*Compared to a smooth tube with no inserts or geometrical modifications unless stated otherwise

Table 6. Efficiency correlations.

Authors	Correlation	Nomenclature
Eiamsa-ard <i>et al.</i> [132]	$\eta = \frac{h_t}{h_p} \Big _{pp}$	η : Thermal Performance factor h_t : Convective coefficient with turbulator h_p : Convective coefficient of plain tube
Garcia <i>et al.</i> [152]	$R3 = \frac{Nu_a}{Nu_s}$	R3: Performance Criterion Nu_a : Nusselt number augmented Nu_s : Nusselt number smooth
Promvonge [166]	$\eta = \left(\frac{Nu_a}{Nu_o} \right) \left(\frac{f_a}{f_o} \right)^{-1/3}$	η : Thermal Performance factor Nu_a : Nusselt number augmented Nu_o : Nusselt number smooth f_a : Friction factor augmented f_o : Friction factor smooth
Fan <i>et al.</i> [164]	$THPR = \left(\frac{Nu_w}{Nu_d} \right) \left(\frac{f}{f_w} \right)$	THPR: Thermal hydraulic performance ratio Nu_w : Nusselt number with coil wire f_w : Friction factor with coil wire

6.1.6 Conclusion

In this chapter, techniques used to enhance single-phase heat transfer are discussed, highlighting all advantages and disadvantages whenever possible. From the analysis of the enhancement mechanisms, it is evident that the method these enhancement techniques utilize to enhance heat transfer is through individually or a combination of disturbance of the boundary layer, enhanced mixing, and generation of secondary flows. The more prevalent these factors are, the higher the heat transfer enhancement. Concurrent with heat transfer enhancement, caution should be exercised to determine the flow regime prominent in a given system. For turbulent flow, devices that enhance mixing are not as effective as devices that disturb the laminar sublayer. Although the fundamental reasoning is similar for all techniques, the differences in real-life utilization of the proposed methods arise from the level of difficulty to implementation in already existing heat exchangers. For example, adding twisted tape or inserts is more integration-friendly than adding fins or baffles due to the requirement of replacing existing tubing. However, with new heat exchangers being manufactured constantly, integration is a major issue concerning existing units in operation. Furthermore, in addition to the integration penalty, cost-effectiveness and longevity are secondary contributing factors to deciding whether the benefit of enhancement outweighs the costs. For example, while twisted tapes might be cheaper to install, issues related to fouling and frequent maintenance may render the solution less beneficial. Although this review covers several of the most popular single-phase heat transfer enhancement techniques, it is not exhaustive. There are several other techniques and areas open for research or improvement. One such area includes the investigation of other possible combinations of passive techniques or even combining active and passive techniques. The current review represents a key summary at a critical time for heat exchanger design mainly for two reasons. The first is related to the recent advent and proliferation

of both polymer and metal additive manufacturing. The costs associated with additive manufacturing are approaching those of conventional legacy heat exchanger manufacturing, opening up the possible design space previously inaccessible with legacy techniques. The combination of additive manufacturing with the recent proliferation and wide utilization of topology optimization makes this review even more relevant as knowledge of existing topologies is key to the development of new designs. The second driver for a summary of existing techniques is the recent development and wide acceptance of machine learning-based neural networks for heat exchanger design. For machine learning to be successful, plentiful data is required to enable the learning step for successful design. This review represents a recent summary of all past data capable of enabling machine learning algorithms for next-generation heat exchanger design.

6.2 Two Phase Heat Transfer Enhancement Techniques

Two-phase internal flow is ubiquitous to many systems due to its ability to transfer large amounts of heat effectively. Recent advances motivated by sustainability have pushed towards augmenting the heat transfer between the tube and the two-phase working fluid. Augmenting heat transfer in two-phase flows enables process intensification, and compactness leading to reduced cost and material consumption. This review investigates the effect of heat transfer enhancement techniques on different fluids including R134a, R407C, R22, etc... The techniques are categorized into passive and active and focus on durable enhancement methods which do not include surface structuring or coating. Passive techniques utilize geometrical and surface modifications to induce better mixing. Several passive techniques are analyzed including fins, inserts, and more. In analyzing these techniques, the effect of changing geometric parameters is studied, and the limitations are highlighted. Further, active techniques that enable heat transfer enhancement through applying

power are also investigated. Electrohydrodynamic approaches are analyzed and compared to passive techniques. Additionally, the implications of heat exchange enhancement in the context of global energy are discussed. An energy analysis discretized by the use sector is conducted, with an emphasis on predicting the future potential for renewable and sustainable development. Moreover, although active approaches provide the highest heat transfer enhancement, due to implementation difficulties, passive techniques are more frequently adopted. The review ends with discussion of next-generation heat exchangers which will rely more on additive manufacturing due to its flexibility and reduced cost per part as the technology matures.

6.2.1 Introduction

With the continuous rise and rapid growth of global industrial output, concerns related to sustainability and energy efficiency have gained significant attention in recent years. For most of these industries, the continued growth in output and productivity is directly or indirectly limited by the effectiveness of the thermal management approach. In industry, one of the most adopted techniques over which large quantities of heat are absorbed, utilizes two-phase transition from liquid to vapor and back. While there are several thermal management approaches that make use of two-phase transitions such as evaporation [233], boiling and condensation are more widely adopted due to the high levels of superheat available in industrial operations. As such, heat exchangers are used instead, and serve to facilitate heat transfer between fluid streams at different temperatures. This exchange of thermal energy is ubiquitous in many industries such as automotive, aerospace, pharmaceutical, petrochemical, and food processing [234].

By improving the efficiency of heat exchangers through heat transfer enhancement techniques, systems that are characterized by a higher gravimetric and volumetric power density

can be attained. Additionally, by increasing efficiency, it is possible to produce more compact heat exchangers that are cheaper and require less raw material. This is especially vital today as overall material consumption has reached 92.1 billion tons [103]. Corollary, the carbon emissions due to transportation and production of raw material can also be reduced. This is important since the averaged mole fractions of CO₂ peaked at 405.5 parts per million in 2017 which is 147% higher than pre-industrial levels. Coupled with the decrease in material consumption, is also a decrease in waste production. This is intuitive as needing less material during production would reduce the waste produced at the end of the device lifecycle. Waste, in particular, was highlighted as a major issue in 2016, where it was reported that waste production peaked at a value of 2 billion tons [103]. Since heat exchangers are used heavily in many industries in their day-to-day operations, improving these devices would have a large-scale impact on sustainability.

Other applications for heat exchangers include air conditioning [72], where the evaporators and condensers that makeup air conditioners and HVAC&R systems are functionally similar to that of heat exchangers. Thermal management of buildings alone consumes 50% of the electrical energy production of developed nations (Fig. 31). Moreover, HVAC systems in buildings (commercial, residential, and industrial) in the United States consume around 75% of the electric power generated [235]. Furthermore, it is important to consider that even though more than one billion air conditioning units have been installed globally, the most efficient are operating at 14% of maximum theoretical efficiency. Since an additional 3.3 billion air conditioners are expected to be installed by 2050, it becomes imperative that increasing the operating efficiency is necessary. This can be achieved partly by improving heat exchanger efficiencies.

Aside from thermal management in buildings, heat exchangers also find their use in data centers where the heat generated by the electronics is absorbed by the coolant flowing in the heat

exchanger tubes. Thermal management using heat exchangers, is a critical bottleneck to increasing the computational performance of modern data centers in terms of density (Gflops/m³) and efficiency (Gflops/W)[236]. On average, about 20-30% of the energy used in modern data centers is dedicated to thermal management. In addition, server packaging density is limited by the spatial requirements of air or single-phase liquid cooling architectures [117]. By facilitating both faster clock speeds and efficient component packing than can be achieved with alternate single-phase cooling schemes, two-phase cooling approaches can improve computational density. As a result, two-phase cooling approaches have the potential to facilitate the deployment of data centers with next-generation graphical processing units for machine learning and big data analytics. This is especially important due to the persistence of COVID-19 across the globe. Many nations are requiring employees to work from home causing an increase in online information transfer and storage. The increased need for video conferencing has spurred the development and construction of multi-MW data centers which now consume up to 5% of the electrical energy generated in developed nations. This electrical energy consumption also translates into an annual increase of 6% in carbon dioxide emissions [237]. Therefore, the enhancement in thermal management would translate to less energy consumption and carbon dioxide emissions.

In addition to building energy, data center thermal management, and air conditioning, a major application requiring the use of heat exchangers is mobile systems, where radiators are the primary source for cooling. Figure 31, illustrates the breakdown of the energy consumption by the transportation sector, showing a 32% global energy consumption since 2016 [121].

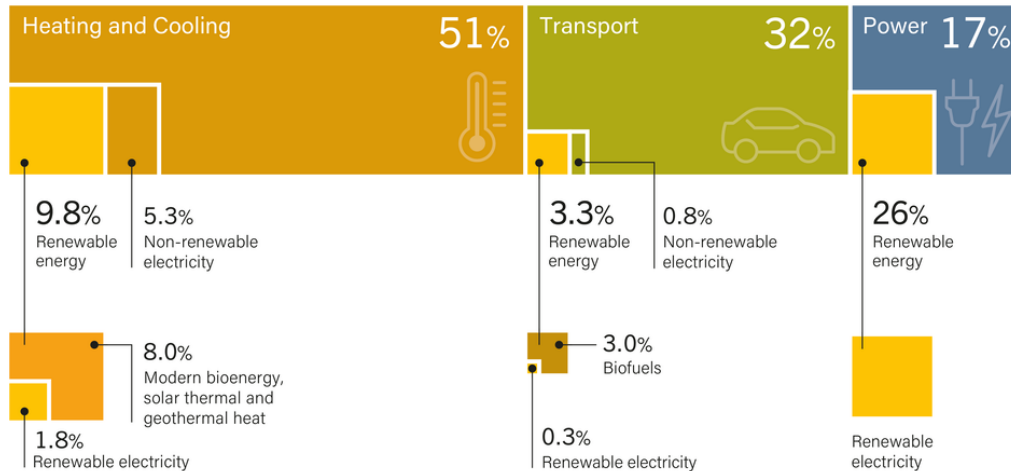


Figure 31. Global energy consumption breakdown

Total global energy consumption breakdown by sector as of 2016 [121]. Reprinted with permission from the Renewables 2019 Global Status Report, REN21, 2019, (Paris: REN21 Secretariat), ISBN 978-3-9818911-7-1. Retrieved from: <https://ren21.net/gsr-2019>

Since transportation greatly contributes to carbon emissions [238], the shift towards electrified transportation is highly sought due to the significant economic and environmental benefits. However, electric vehicles (EVs) are limited by energy consumption due to cabin comfort (HVAC) and battery thermal management[239]. Mobile HVAC systems play a very important role and can, depending on the ambient temperatures, consume up to 65% of the stored electrical energy of the vehicle's battery [118,240]. By increasing the efficiency of the heat exchangers, this energy consumption and associated carbon emissions can be reduced greatly. Consequently, key technologies are needed to design and build EVs that utilize advanced heat exchange, heat pumping, and thermal energy storage[241–243]. In addition to EV HVAC and battery thermal management [244,245], heat exchangers play an important role in large-scale water treatment applications. With 785 million people around the world lacking drinking water [103], it is imperative to develop simple water treatment methods cost-effectively. This can be achieved by highly efficient heat exchanger technology.

The recent focus on renewable and sustainable energy applications such as solar-thermal power generation [246], energy storage [247], transportation electrification [229], and low GWP HVAC systems has placed a renewed interest in two-phase heat exchanger development and design [248]. From an ideal thermodynamic performance perspective, the efficiency of a heat engine increases with increasing the hot-side cycle temperature. The effort to achieve higher efficiency requires higher hot-side temperatures. In the case of sustainable closed cycle power generation systems, recent interest in the use of supercritical carbon dioxide (sCO₂) has placed added focus on heat exchanger efficiency upstream and downstream of the turbine given the harsh environments (high temperature and pressure). Hence, the development of robust and cost-effective heat exchangers could lead to substantially higher power conversion efficiencies that yield reduced fuel consumption, footprint (lower capital and operation costs), and CO₂ and NO_x emissions.

Regardless of the application, the flow in an unmixed heat exchanger can be characterized as single-phase or two-phase. While heat transfer in single-phase flow is important and has been the classical subject of interest [234], two-phase flow is complex due to the transition from liquid to vapor or vice versa. This chapter focuses on two-phase flows specifically, condensation, and evaporation/boiling. While there are applications in which transition to and from the solid phase is important (e.g., reacting flows in internal combustion engines), the current review does not cover transitions to the solid phase to keep the discussion in the context of liquid and vapor phases. Furthermore, this review does not cover micro and nanoscale surface structuring[249], nor coating techniques[250] which maintain limitations owing to the lack of quantified robustness when used in realistic settings[93,251]. Table 7 presents a summary of the major applications and associated components where two-phase flow predominantly contributes to the performance of the overall system.

Table 7. Applications and components that utilize two-phase flows.

Use Category	Processes	Components	Significance of enhanced two-phase flow	Implications for energy efficiency and sustainability
HVAC&R	Air cooling	Condenser, Evaporator	Reduction in equipment size and refrigerants charge	1) Reduced cost 2) Enhanced process control 3) Improved coefficient of performance (COP) 4) Reduced environmental impact
	Air heating (heat pump)	Condenser, Evaporator		
	Refrigeration	Condenser/gas cooler, Low and medium temperature evaporator		
	Dehumidification	Evaporator		
Power Generation	Solar power generation	Energy storage system, condenser, boiler, recuperator, gas cooler	Compact system, reduction in working fluid	1) Enabling widespread implementation of modular systems 2) Reduced cost 3) Reduced environmental impact 4) Improved process control
	Fossil Fuel Based Power Generation	Condenser, boiler, recuperators, economizers		
	Nuclear power generation	Boiler, recuperators, intermediate heat exchangers for cascade cycles		
Electronics Cooling	Small scale electronics	Heat sinks, heat pipe, vapor chambers	Compact system	1) Improved performance 2) Reduced cost
	Data centers	Distributed two-phase heat exchanger, immersion cooling		
	Power electronics	Embedded heat sinks, heat exchangers		
	Electric storage, charging management	Heat sinks, heat spreaders		
Thermal Energy Storage	Thermo-chemical	Reactor for charging/discharging	Compact energy storage	1) Grid interactive systems enabled 2) Reduced cost
	Phase change materials	Embedded heat exchangers		
Miscellaneous	Waste heat recovery	Two-phase heat exchanger, heat exchangers with heat pipes	Reduced equipment size	1) Improved efficiency 2) Improved process control 3) Reduced environmental impact 4) Reduced cost 5) Enhanced fuel cell vehicle range
	Fuel cells	Integrated heat exchangers, gas reformers, vehicle radiators		
	Desalination	Evaporators/boilers, condensers		
	Renewables (wind, tidal, etc.)	Reactors (hydrogen production)		

Much of the research conducted in the past century focused on optimizing two-phase flow heat transfer in heat exchangers. This is achieved by improving the thermal performance of its heat transfer surfaces (e.g., tubes and channels). Generally, the main efforts focused on disturbing fluid flow to induce turbulence [252], enhance mixing [139], and/or maximizing the contact area between the fluid and tube walls [132]. These effects can be achieved by either geometrical modifications to the walls and/or the use of electric or magnetic fields to induce flows in certain directions [253,254].

Heat transfer enhancement techniques are classified into active and passive [254]. Passive techniques usually rely on geometrical modifications or inserts in an existing tube or channel to provide flow disturbance. Active techniques, however, rely on mechanical or electrical power input to generate oscillations or disturbances. Since passive techniques are easier and more cost-effective to implement, these techniques are more common in the industry. However, active techniques have the advantage of providing more significant increases in heat transfer compared to passive methods. Furthermore, since the enhancement is proportional to the power input, the amount of heat transfer enhancement is controllable for active techniques.

Whether active or passive, it is important to consider the increase in pressure drop associated with the heat transfer enhancement. All techniques covered in this chapter rely on disturbing the fluid flow which causes additional pressure drop along the tube. The larger pressure drop makes it necessary to use a bigger and more costly pump to compensate for the losses. If the heat transfer enhancement obtained by implementing a solution causes an excessive pressure drop, the cost penalty of using a bigger pump may overshadow the savings obtained by a more power-dense heat exchanger.

To enable a critical comparison and provide design guidelines, several enhancement factors have been developed and are summarized in this chapter. However, enhancement factors should be used with caution as some may not consider the pressure drop change. This review not only presents a comprehensive summary of past enhancement techniques for two-phase flows, but it also develops design guidelines for the development of advanced heat exchangers motivated by the recent push for renewable and sustainable energy system implementation. The critical review and summary of design guidelines presented here is an important contribution given the recent advancement in topology optimization [105], machine learning algorithms[255,256], and additive manufacturing techniques being pursued for two-phase flow enhancement in internal flows [257].

6.2.2 Data Reduction

To identify whether a particular enhancement technique is worth pursuing, the heat transfer coefficient and the pressure drop need to be quantified. Past work can be summarized by two main experimental approaches. In this section, these two approaches are summarized

6.2.2.1 Constant Heat Flux

The first and the most widely adopted approach in literature refers to subjecting the test section and pre-evaporator with a known and constant heat flux. In this approach, the first step is to calculate the outlet quality of the pre-evaporator by [258]:

$$Q_{pe} = C_{p,ref} \dot{m}_{ref} (T_{sat|pe} - T_{ref,in|pe}) + \dot{m}_{ref} h_{fg|pe} x_{out|pe} \quad (21)$$

where Q_{pe} is the heat provided to the pre-evaporator, C_p is the specific heat capacity of the working fluid, $\dot{m}_{ref}$ is the mass flux of the working fluid, $T_{sat|pe}$ is the saturation temperature at the pre-

evaporator, $T_{\text{ref,in|pe}}$, is the inlet temperature into the pre-evaporator, $h_{\text{fg|pe}}$ is the latent heat of vaporization of the working fluid in the pre evaporator, and $x_{\text{out|pe}}$ is the outlet vapor quality of the pre-evaporator. The inlet vapor quality of the test section, $x_{\text{in|ts}}$, is calculated assuming:

$$x_{\text{in|ts}} = x_{\text{out|pe}} . \quad (22)$$

The test section outlet vapor quality can then be computed using [258]:

$$Q_{\text{ts}} = \dot{m}_{\text{ref}} h_{\text{fg|ts}} (x_{\text{out|ts}} - x_{\text{in|ts}}), \quad (23)$$

where Q_{ts} is the heat provided to the test section, $h_{\text{fg|ts}}$ is the latent heat of vaporization of the working fluid in the test section, and $x_{\text{out|ts}}$ is the outlet vapor quality in the test section. The heat transfer coefficient can then be calculated locally at a distance, x , from the inlet of the test section using [259]:

$$\frac{1}{h_x} = \frac{T_{\text{wall}} - T_{\text{sat}}}{q_x} - \frac{\ln\left(\frac{d_o}{d_i}\right) d_i}{2k}, \quad (24)$$

where h_x , q_x , T_{wall} and T_{sat} are the heat transfer coefficient, heat flux, wall temperature, and saturation temperature at a distance x from the test section tube inlet, respectively. k is the metal thermal conductivity of the tube, d_{in} is the inner tube diameter, d_{out} is the outer tube diameter are.

To calculate the frictional pressure drop in horizontal tubes, the following relation is used [260]:

$$\Delta P_{\text{total}} = \Delta P_{\text{momentum}} + \Delta P_{\text{friction}}, \quad (25)$$

where the total pressure drop, ΔP_{total} , is measured experimentally, and the momentum pressure drop is calculated using the following correlation [261] :

$$\Delta P_{\text{momentum}} = G_{\text{ref}}^2 \left\{ \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g\alpha} \right]_{\text{out,ts}} - \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g\alpha} \right]_{\text{in,ts}} \right\}, \quad (26)$$

where ρ_f and ρ_g are the density of the liquid and vapor, respectively, G_{ref} is the refrigerant mass flux and α is the void fraction that can be calculated using the Steiner correlation [262].

6.2.2.2 Constant Temperature

The constant temperature experimental approach depends on heating the test section with an outer jacket of liquid. Although this method is not as widely adopted, it is important to many studies. To calculate the heat transfer in the pre-evaporator, Eq. (21) is used. However, to compute the heat provided to the test section, Q_{ts} , the temperature change of the outer liquid jacket is used[258]:

$$Q_{\text{ts}} = C_{p,\text{ol}} \dot{m}_{\text{ol}|ts} (T_{\text{ol,out}|ts} - T_{\text{ol,in}|ts}), \quad (27)$$

where $C_{p,\text{ol}}$, $\dot{m}_{\text{ol}|ts}$, $T_{\text{ol,out}|ts}$, $T_{\text{ol,in}|ts}$ are the specific heat capacity, the mass flux at the test section, the temperature at the outlet of the test section, and the temperature inlet of the test section, respectively. In this case, the test section denotes the outer single-phase liquid flow jacket.

The outlet vapor quality is then computed using Eq. (23). To calculate the internal heat transfer coefficient, h_i , of the test section, the following equation is used [259]:

$$h_i = \frac{Q_{\text{ts}} h_o A_o 2\pi L k}{A_i \text{LMTD} \ln\left(\frac{d_o}{d_i}\right)}, \quad (28)$$

where h_o , A_o are the heat transfer coefficient and surface area of the outer side of the tube, respectively. A_i is the inner surface area, and l is the length.

The log mean temperature difference (*LMTD*) is calculated using the following [259]:

$$\text{LMTD} = \frac{|(T_{\text{ref,out}|ts} - T_{\text{ol,in}|ts}) - (T_{\text{ref,in}|ts} - T_{\text{ol,out}|ts})|}{\ln\left(\frac{T_{\text{ref,out}|ts} - T_{\text{ol,in}|ts}}{T_{\text{ref,in}|ts} - T_{\text{ol,out}|ts}}\right)}. \quad (29)$$

To calculate h_o , the single-phase fully-developed correlations developed by Gnielinski [263] and Petukhov [264] are used if the outer tube wall is smooth. Otherwise, if the tube is not smooth due to the presence of dimples or grooves, an approximation using the Wilson plot method [265,266] is used.

6.2.3 Heat Transfer Enhancement Techniques

This section of the review focuses on analyzing existing heat transfer augmentation techniques for two-phase flows. Two-phase flow occurs when the liquid inside a tube turns gaseous, or vice versa. Two-phase flow finds its use in many applications such as microchannel flow boiling [267], which is used to provide stream-wise temperature uniformity [268–270]. Most of the techniques reviewed within this section are passive, with one active technique (electrohydrodynamic, EHD) due to its ability to augment the heat transfer coefficient (HTC) significantly. Figure 32 summarizes the techniques reviewed.

In both boiling and condensation, the heat transfer coefficient can be enhanced by increasing the area over which heat transfer takes place[271]. However, in the case of flow boiling, the thermal resistance to heat transfer of vapor is much greater than that of liquid, so a larger wetted area inside the tube walls is preferred. Therefore, several techniques attempt to increase the possibility of annular flow inside the tube. Other techniques undertake a different approach and attempt to modify the surface (roughness) to enhance nucleate boiling. It is important to note here that although the last decade has seen a drastic amount of research effort placed on developing surface structuring techniques to enhance two-phase heat transfer for boiling[272,273], and

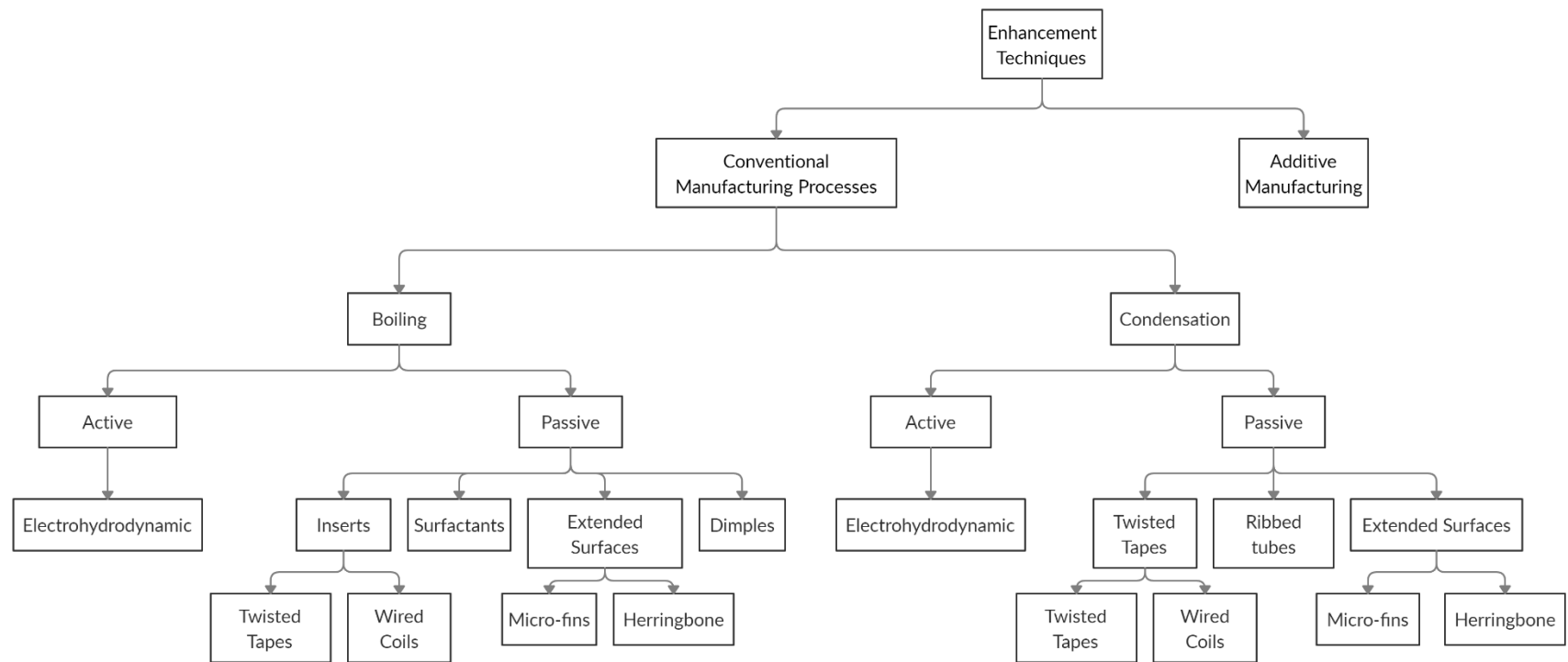


Figure 32. Summary of the enhancement techniques reviewed

condensation[274], the lack of demonstrated durability and long term operation (> 1 year) has precluded these methods from real industrial implementation, and hence these are not covered herein.

6.2.3.1 Flow Boiling

Flow boiling is heavily used in industrial processes to absorb heat generated from operations and effectively cool down the system or the necessary working fluid. Depending on the flow rate and heat flux, the vapor quality of the fluid varies, where different qualities correspond to different flow regimes. At low vapor qualities, the bubbly flow regime is observed and nucleate flow boiling dominates. At larger qualities, the annular flow regime is observed and convective flow boiling dominates. Typically, convective flow boiling regimes are characterized by higher heat transfer coefficients and are therefore sought out. To enhance flow boiling, a common approach is to alter the surface geometry in order to create artificial nucleation sites that initiate heterogenous boiling and encourage bubble departure. This subsection investigates the effect of enhancement techniques on the in-tube flow boiling HTC.

6.2.3.1.1 Surfactants

The first passive technique reviewed to enhance HTC in flow boiling applications is surface-active materials or surfactants. Surfactants are materials that reduce the surface tension of the liquids in which they are dissolved. Previous studies showed that small amounts of surfactant added to water can greatly enhance pool boiling [275–279]. A study carried out by Qiu *et al.* [280] investigated the effect of octadecylamine (ODA) on the flow boiling heat transfer of water in a vertical copper tube with 0.6 m length and 12 mm diameter. Using X-ray photoelectron spectroscopy, the mechanism that causes enhancement of boiling heat transfer was analyzed. When ODA is adsorbed onto the tube surface, it alters the heating of the surface which may create artificial sites for bubble

initiation and growth. Furthermore, the bubble departure diameter decreases. This increases the frequency of bubble departure and in turn, enhances the heat transfer for mass fractions in the range of 1×10^{-5} to 2.5×10^{-5} . However, once the mass fraction reaches the critical micelle concentration (CMC) at 3×10^{-5} , the convective HTC decreases [280,281]. At the CMC point, the surfactants saturate the liquid-vapor interface and a new third component called micelles form in the fluid bulk. Micelles form due to the hydrophobic attraction between the tails of the amphiphilic molecules (surfactant). When the CMC point is reached, the surface tension no longer changes and so the effect of having a higher surfactant concentration is negligible. Although HTC decreases at the CMC, there is still an enhancement when compared to flow boiling with pure water.

A study by Hetsroni *et al.* [282] investigated the heat transfer and pressure drop of water and a water-surfactant solution (*Habon G*) of 530 and 1060 ppm. The study was conducted for a fully developed laminar flow ($10 < Re_d < 450$ where $Re_d = Ud_i/\nu$, where U is the average fluid speed, d_i is the internal tube diameter, and ν is the liquid kinematic viscosity) in a capillary tube of $d_i = 1.07$ mm inner diameter. It was reported that adding the surfactant would increase the pressure drop in adiabatic and diabatic flows. Furthermore, it was reported that the HTC was higher when surfactants were present.

6.2.3.1.2 Dimples

Another passive approach taken by Shafaei *et al.* [283] highlights the heat transfer enhancement provided by dimples on a copper tube (Fig. 33). The test was carried out under specific conditions (mass flux (G) range: 155 – 470 kg/(m²·s); vapor quality < 0.8 ; heat flux: 15.8 kW/m²; saturation temperature: 56.5°C, tube 9.5 mm outer diameter and 1 m length).

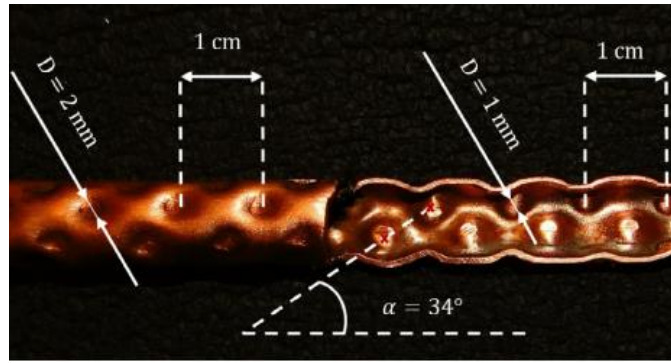


Figure 33. Horizontally oriented dimpled tube

Image of a horizontally oriented dimpled tube [283]. Reprinted from Appl Therm Eng, 107, Shafaei et al., Evaporation heat transfer and pressure drop characteristics of R-600a in horizontal smooth and helically dimpled tubes, p. 28–36, 2016, with permission from Elsevier

The working fluid used in the investigation was R600a, as it has a lower ozone depletion potential [284,285]. The dimples in the tubes were arranged with a pitch ratio of 1.21. The dimples were of two types: shallow and deep. The diameters of the shallow and deep dimples were 1 mm and 2 mm, respectively, while the depths of the shallow and deep dimples were 0.5 mm and 1 mm, respectively.

The results for both dimple types can be categorized into two distinct regions based on the vapor quality. In the low-vapor-quality region, as the vapor quality increased, the HTC decreased, reaching a minimum between $0.15 < x < 0.45$. In the high-vapor-quality region, the HTC increased with increasing vapor quality.

Analysis of the nucleate boiling mechanism can be used to explain the behavior exhibited in the low-vapor-quality regions. For lower vapor qualities, nucleate boiling was the dominant phenomenon. However, as the quality increased and the annular flow regime initiated, the liquid in contact with the tube walls started to thin (film effect). Since the high vapor velocity carried heat away by convection, the wall superheat was reduced to a value below the minimum threshold

required for bubble nucleation on the tube wall. As a result, nucleate boiling decreased, which in turn caused the HTC to decrease.

In these dimpled tubes, the range of vapor quality values over which nucleate boiling dominates was smaller than that in plain tubes. Further, the HTC increased during the low vapor quality regions. This response can be attributed to the protrusions that cause an increase in the superheated regions and thus the creation of more nucleation sites. The behavior in the high vapor quality region is due to the vaporization of the R600a refrigerant. As the refrigerant vaporized, its density decreased. The lower the refrigerant density, the higher the flow velocity, which subsequently led to liquid convection boiling dominance. Consequently, the HTC increased. For dimpled tubes, the range of high vapor quality region values over which convective boiling dominated was wider and the increase in HTC was more prominent than in plain tubes.

The experiments conducted by Shafae *et al.* [283] showed that the HTC of dimpled tubes was 1.29–2 times greater than plain tubes. Enhanced heat transfer was observed for the dimpled tubes because they promoted swirling near the wall, which disrupted the boundary layer and increased agitation, thus promoting mixing and turbulence. Furthermore, the surface area of the tube was increased, which promoted more nucleation sites.

Mass flux also played an important role in heat transfer. When it increased, the velocity of the refrigerant increased, increasing liquid convection and the HTC. However, this influence became marginal when the vapor quality was low.

Finally, it was noted that the correlation of the HTC with Reynolds number ($Re_{lf} = 4G(1 - x)\delta/(1 - \varepsilon)\mu_L$) was nonmonotonic. This is because, at low Re_{lf} , smaller eddies were more prevalent. Smaller eddies had more efficient momentum transfer than larger eddies at high Re_{lf} . Therefore, the heat transfer enhancement at lower Re was smaller [286,287].

As with the other enhancement techniques, the pressure drop increased as a direct implication. Since dimples increased the frictional surface area, the frictional pressure drop in dimpled tubes was always higher than in that plain tubes, by 7–103%. This large pressure drop range can be attributed to the large vapor quality range investigated in the study. As the vapor quality of the fluid increases, the amount of vapor generated increases. Since the gaseous phase is at a higher velocity than the liquid phase due to its lower density, it imposes elevated shear stress, and hence pressure drop increases. Consequently, the pressure drop increases with vapor quality due to the velocity of the vapor growing at higher qualities. To compare the effect of dimples on the pressure drop, the pressure drop at different mass fluxes was investigated. Table 8 summarizes the findings [283].

Table 8. Pressure drop comparison between dimpled tubes and smooth tubes.

Mass flux (kg/(m ² ·s))	Pressure drop of dimpled tubes normalized by the pressure drop of smooth tubes
420	1.1 – 1.44
305	1.07 – 1.7
205	1.27 – 1.47
155	1 – 2.03

To quantify the enhancement provided by dimpled tubes, the surface performance factor, PFS, is used [288]:

$$\text{PFS} = \frac{h_D}{h_p} \cdot \frac{A_p}{A_D}, \quad (30)$$

For all Re_{lf} values, the PFS was always greater than unity, indicating that the ratio of the convective coefficients (h_D/h_p) was larger than that of the surface area. Moreover, the convective

coefficient (due to flow separation, boundary layer disruption) and the internal surface area both increased with the addition of dimples.

Although the PFS ratio does offer means of exploring the enhancement provided by dimples, it does not consider the effects of pressure drop. Another ratio, the energy performance factor, PFE [289], takes pressure drop into account. It is defined as:

$$\text{PFE} = \frac{\frac{h_D}{h_p}}{\frac{\Delta P_D}{\Delta P_p}} \quad (31)$$

The PFE was calculated for different Re_{lf} values at $x = 0.173$ and $x = 0.45$. For 60% of the PFE values, the ratio was greater than unity, indicating that the presence of dimples was favorable. The best performance, however, was at $Re_{lf} > 18,000$. For other Re_{lf} values, the PFE values at $x = 0.45$ were greater than at $x = 0.173$. Consequently, annular regimes (i.e. at higher x values) are recommended.

Another commonly used correlation to quantify whether a technique is lucrative includes the thermal-hydraulic performance factor:

$$\text{PEC} = \left(\frac{Nu_D}{Nu_0} \right) \left(\frac{f_D}{f_0} \right)^{-1/3} \quad (32)$$

Similar to PFE, PEC provides means of comparison between heat transfer enhancement and frictional pressure drop. Even though not used by Shafee *et al.* [283], PEC is one of the more popular performance correlations adopted in the field.

The arrangement of the dimple pattern influences the thermal performance. Two common designs exist in open literature: staggered and inline. Numerical simulations conducted by Li *et al.*

[290] show that inline patterned dimples provide better heat transfer performance than staggered ones. The simulations also showed that the dimple shape, depth, and pitch had prominent effects on the heat transfer performance unlike other parameters such as dimple diameter. At $2300 < Re < 3100$, it was noted that spherical dimples had the highest PEC, whereas ellipsoidal-shaped dimples had the highest PEC for $Re > 4000$. It can be then noted that for lower Re ranges, turbulence induced heat transfer augmentation overshadows the increase in pressure drop. However, this gap is narrowed at greater fluid velocities. In all cases, it was noted that all dimple shapes have $PEC > 1$, showing that dimples provide an enhancement of heat transfer greater that overshadows the pressure drop increase.

A study by Zahid *et al.* [291] explored the effect of dimples on a 304 stainless steel tube with 19.05 mm outer diameter, 2 m length, and a 12.75 mm diameter PVC rod insert (Fig. 34). The PVC rod was fastened with screws to ensure that it was centered. The investigation was carried over specific flow conditions (inlet saturation temperature: 5°C; heat flux: 2.5 – 15 kW/m²; mass flux = 80 – 200 kg/(m²·s); inlet vapor quality: 0.12–0.72). The investigation used R134a as the working fluid.

As with annular flow in smooth tubes, it was noted that increasing the mass flow rate and vapor quality raised the heat transfer coefficient. Furthermore, for low mass flow rates and $x < 0.5$, there was a positive correlation between the heat flux and the HTC. However, this correlation diminished as mass velocities exceeded 80 kg/(m²·s), at which point nucleate boiling was suppressed. The lower mass velocity behavior agreed with the observations of Tibiriçá and Ribatski [292].



Figure 34. Dimpled tube with PVC rod insert

Photograph of a dimpled tube with PVC rod insert [291]. Reprinted from Int J Refrig, 75, Ayub et al., Two-phase pressure drop and flow boiling heat transfer in an enhanced dimpled tube with a solid round rod insert, p. 1-13, 2017, with permission from Elsevier.

Comparing the HTC of dimpled tubes containing inserts to smooth tubes with no inserts, it was noted that the HTC increased by up to threefold. Further, the pressure drop was significantly larger in tubes with inserts than that of a similar annulus region with a smooth tube. In addition, the heat transfer augmentation diminished as a mass velocity of $200 \text{ kg}/(\text{m}^2 \cdot \text{s})$ was approached. Consequently, as the mass velocity decreased, the HTC increased and the pressure drop decreased making the system ideal for low mass velocities. Although the insert was able to increase the HTC by up to threefold, this method had some restrictions that might render it inappropriate. The PVC insert limits the method to annular flows. However, this design could be useful for some industrial applications such as for direct expansion evaporators [291] or nuclear reactors. Moreover, to achieve the maximum heat transfer enhancement, low mass velocities are required. Otherwise, the pressure drop increases substantially and the heat transfer augmentation is reduced.

Another investigation carried out by Guo *et al.* [288], studied enhanced heat transfer (EHT) tubes produced by Vipertex (Fig. 35). These 2 m long tubes had an inner diameter of 11.5 mm with dimples/protrusions and secondary petal arrays. The modified EHT tube surface increased the surface area and flow separation, which created more nucleation sites and turbulence that disrupted the boundary layer and enhanced heat transfer. It was observed that for all three refrigerants (R22, R410A, R32), the enhancement ratio of heat transfer was 1.2 – 1.4 times larger than that for smooth tubes with a mass flux ranging between 55 and 142 kg/(m²·s). To quantify the effectiveness of the EHT tubes, the PFS was used. In all conditions, EHT tubes provided a PFS value greater than 1, except when R32 refrigerant was used at very low mass fluxes.

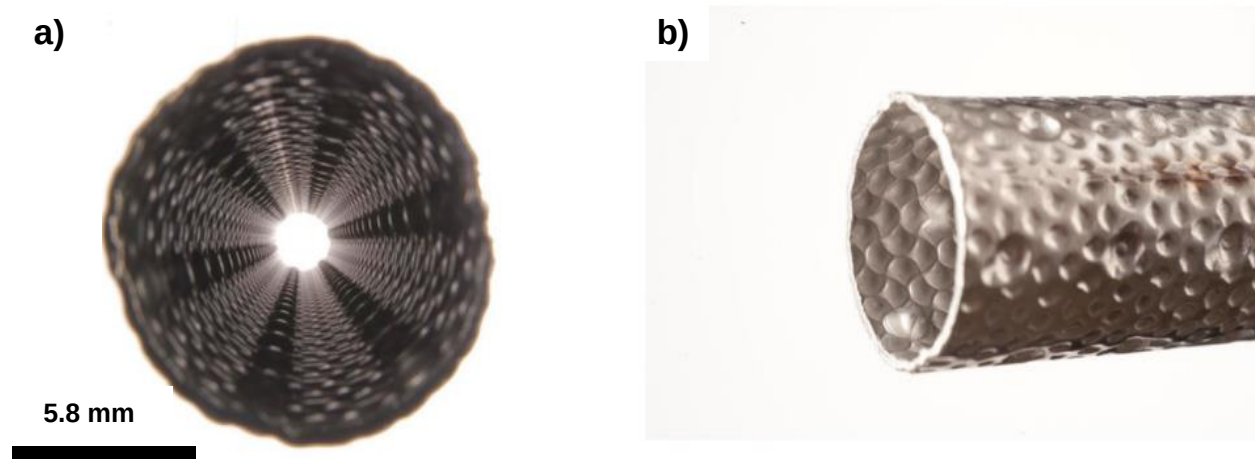


Figure 35. EHT tubes

Photographs of EHT tubes created by Vipertex showing the (a) internal geometry, and (b) the external geometry [288]. Reprinted from Int J Heat Mass Transf, 85, Guo et al., Condensation and evaporation heat transfer characteristics in horizontal smooth, herringbone and enhanced surface EHT tubes, p. 281–91, 2015, with permission from Elsevier

Similar research by Wei *et al.* [293] examined a 2EHT tube that had both its outer and inner surfaces enhanced by longitudinal grooves with an array of dimples. Using R410A as the working fluid, the HTC of the 2EHT tubes was reported to be higher than that of smooth copper tubes by

1.11 to 1.43 times, given a mass flux range between 80 to 200 kg/(m²·s). Using the PFS evaluation criteria, Wei *et al.* reported that during condensation at low mass flux values, PFS was less than unity, indicating that 2EHT tubes do not provide the desired heat transfer enhancement. However, for larger flux values, it was noted that PFS is higher than 1. Contrasting these results with evaporation, it was noted that during evaporation, 2EHT tubes provided PFS greater than 1 for all mass flux values. To conclude, the 2EHT tubes provide excellent performance overall independent of which mass flux condition is used. Similar to the other studies, the dimples/embossments increased the surface area and interfacial turbulence, leading to flow separation and heat transfer enhancement between the tube wall and the working fluid

6.2.3.1.3 Straight/Helical Micro-Fins

Micro-finned tubes, first developed by Hitachi Cable [294], are commonly used in evaporator or condenser tubes due to their ability to augment in-tube heat transfer with an acceptable pressure drop penalty. Several parameters dictate the performance of micro-finned tubes including the outer diameter (d_{OD}), root diameter (d_r), tip diameter (d_T), fin height (H), helix angle (β), and apex angle (λ). Furthermore, fins can take different shapes such as rectangular or trapezoidal. Depending on the application, the number of fins, fin height, and helix angle of the micro-fin tubes usually range between 50-70 fins, 0.075-0.4 mm, and 10°-35°, respectively [295].

By increasing the heat transfer area, micro-finned tubes can increase wetting inside the tube, which increases the rate of heat transfer. Several studies have investigated the effect of different fluids inside micro-finned tubing during flow boiling. Schlager *et al.* [296], studied the effect of 12.7 mm spiraled micro-finned copper tubes on the heat transfer and pressure drop with R22 as the working fluid. The heat transfer enhancement factor (Eq. 33) for evaporation ranged between 1.6 to 2.2. Moreover, as the HTC increased, the mass flux increased, with the HTC at 300 kg/(m²·s)

being 400% higher than that measured at $75 \text{ kg}/(\text{m}^2 \cdot \text{s})$. The pressure drop factor (Eq. 34) was lower than the EF, the maximum amounting to 1.4.

$$EF = \frac{(hA)_{\text{Enhanced tube}}}{(hA)_{\text{Smooth tube}}} \quad (33)$$

$$PF = \frac{(\Delta P)_{\text{Enhanced tube}}}{(\Delta P)_{\text{Smooth tube}}} \quad (34)$$

Using R417A, Zhang *et al.* [297] conducted flow boiling experiments in internally grooved tubes. The measured EF and PF values amounted to 1.26-2.8 and 1.8-2.6, respectively. Jiang *et al.* [298], investigated the flow boiling heat transfer and pressure drop for R22, R134a, R407C, and R410A in a 9.52 mm diameter smooth and micro-fin tube (Fig. 36). The EFs for R22, R134a, R407C, and R410A were 1.86, 1.80, 1.69, and 1.78, respectively, while the PFs were 1.42, 1.30, 1.45, and 1.40, respectively.

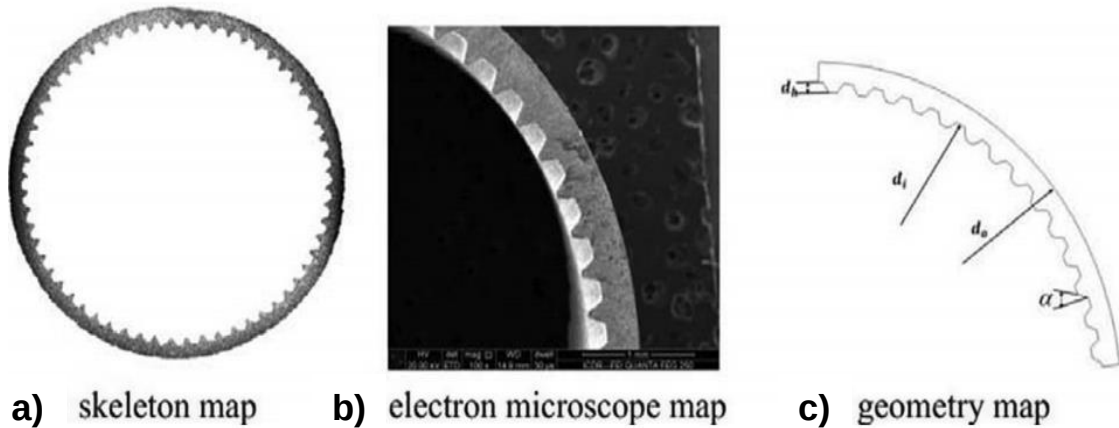


Figure 36. A micro-fin tube

(a) Cross-sectional skeleton map of a micro-fin tube. (b) Scanning electron microscopy image and (c) schematic of a micro-fin tube [298]. Reprinted from Int J Heat Mass Transf, 98, Jiang et al., Experimental study of boiling heat transfer in smooth/micro-fin tubes of four refrigerants, p. 631-42, 2016, with permission from Elsevier

It is worth noting that the augmentation in heat transfer provided by the fins decreased with mass flux. This can be explained by analyzing the flow regimes inside the tube. At lower mass

flux, the flow pattern in the smooth tube is stratified-wavy flow, while for the micro-fin tube, the flow regime is mainly annular. Consequently, the heat transfer in the low mass flux region is significantly enhanced. However, for higher mass flux, the annular flow pattern is present for both tubes. Therefore, the heat transfer enhancement at higher mass fluxes mainly comes from the increase in area. Several other studies were dedicated to investigating the effect of mass flux on the HTC and pressure drop. Yu *et al.* [299] studied heat transfer in a tube with helical fins having $\beta = 18^\circ$. Using R134a, the investigation was conducted over mass fluxes between 163 and 408 kg/(m²·s) and heat fluxes between 2,200 and 56,000 kW/m². The authors reported up to 200% increase in heat transfer at low mass fluxes and higher vapor qualities. Diani *et al.* [300] studied flow boiling inside micro-fin tubes with a fin height of $H = 0.12$ mm. Using R1234ze(E), a range of mass and heat fluxes between 375 and 940 kg/(m²·s) and 10 and 50 kW/m², respectively were studied. Compared to plain tubes, finned tubes showed an average EF of 2.4 with the minimal being 1.87. The enhancement decreased with increasing mass velocity and heat flux. However, the pressure drop increased as the vapor quality increased. Using the PF evaluation criteria, it was noted that the mean value of PF is 1.4. Filho and Barbieri [301], examined the heat transfer and pressure drop of two micro-fin tubes from two different manufacturers. The fins of the two tubes had essentially the same fin geometry. Both of these tubes were compared with an equivalent smooth tube. It was noted that both finned tubes obtained the same results and both provided a significant improvement in HTC compared to a plain tube. This improvement lasted through higher mass fluxes (500 kg/(m²·s)) and vapor qualities ($x = 0.65$). Since fins added flow resistance, the pressure drop in the micro-fin tubes was higher than in plain tubes. As a result, the enhancement factor for fluxes beyond 500 kg/(m²·s) did not justify adding fins. These conclusions were drawn using PFE as the evaluation criterion. For all mass flux values below 500 kg/(m²·s), PFE was

always greater than unity (reaching a maximum of ~ 3 at $100 \text{ kg}/(\text{m}^2 \cdot \text{s})$) for all vapor qualities. However, flux values equal to or above $500 \text{ kg}/(\text{m}^2 \cdot \text{s})$ started exhibiting PFE values less than unity. Kim and Shin [302] investigated seven micro-fin tubes to compare them with a smooth tube of similar geometry. For R22 and R410a working fluids, the authors calculated the PFE to be 1.86–3.27 times and 1.64–2.99, respectively, indicating that micro-fins are desirable. Dang *et al.* [303] carried out a similar investigation using carbon dioxide as the working fluid. Using a tube with fins having a helix angle of $\beta = 6.3^\circ$, the experiment was conducted over a range of heat and mass fluxes. Similar to the results of the preceding research, it was reported that using a finned tube increased both the HTC and the pressure drop compared to a smooth tube. The HTC and pressure drop increased by 1.9–2.3 times and 1.5–2.1 times, respectively. It was noted that increasing the heat flux increases the heat transfer augmentation while increasing the mass flux decreases the augmentation.

As stated previously, the helix angle (β) is critical in micro-fin tube performance. Numerous studies have demonstrated how it affects the HTC and pressure drop in two-phase flow. Schlager *et al.* [304], examined the heat transfer and pressure drop for R22 in horizontal micro-fin tubes (9.5 mm OD) with β ranging between 15° to 25° during evaporation and condensation. The optimal angle for peak performance was shown to be at $\beta = 25^\circ$. Oh and Bergles [305] examined the evaporative heat transfer in a 9.52 mm OD micro-finned tube with different helix angles ($\beta = 6^\circ, 12^\circ, 18^\circ, 25^\circ$, and 44°). Using R134a, it was reported that the optimal angle depended on the mass flux. The optimal helical angle for a flux of $50 \text{ kg}/(\text{m}^2 \cdot \text{s})$ was $\beta = 18^\circ$, while for a flux of $100 \text{ kg}/(\text{m}^2 \cdot \text{s})$ it was $\beta = 6^\circ$. Using R12, Kimura and Ito [306] studied the evaporative heat transfer in a 4.75 mm ID micro-finned tubes with $\beta = 4^\circ, 7^\circ, 15^\circ$, and 30° . Similar to other investigations, it was reported that the mass flux influenced the optimal β over which heat transfer is maximized.

In the stratified flow regime, the heat transfer coefficient increased with β . In the annular flow regime, the change in the HTC for β between 4° and 30° was not significant. However, within this range, the highest HTC was reported at $\beta = 15^\circ$. Yang and Hrnjak [307] further visualized the flow patterns, over different β , in a 3D printed transparent micro-finned tube. The authors found that β affected the transition from stratified flow to annular flow where annular flow regimes occurred at lower mass fluxes and lower vapor qualities as β increased.

Most of the previous discussion was mainly focused on helical micro-fin tubes. However, axial or straight micro-fin tubes ($\beta = 0^\circ$) are another economic option for heat transfer enhancement. Axial micro-fins are especially prevalent in aluminum tubes as they are made by extrusion. As a result, the aluminum axial micro-fin tube is considered as an alternative to copper helical micro-fin tubes in the HVAC industry. Although the heat transfer performance enhancement for axial micro-fin tubes is lower compared to helical designs, some results have shown very effective heat transfer performance in certain conditions. Chiang [308] experimentally investigated smooth, axial micro-fin, and helical micro-fin tubes having diameters of 7.5 mm and 10 mm. It was reported that the mass flux influenced the heat transfer performance in axial micro-fin tubes. When the flux exceeded $175 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the heat transfer in the axial micro-finned tube outperformed the helical micro-fin tube. However, as the mass flux decreased to below $175 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the helical micro-fin tube provided better heat transfer performance. Moreover, Wu *et al.* [309] concluded that the drag produced by helical angle fins had a significant effect on heat transfer at small tube diameters. Therefore, axial micro-fins had the potential for enhancing heat transfer in evaporation. Further, it was shown that, for R32, the axial micro-fin tubes increase heat transfer by 60% compared to smooth tubes.

Yang and Hrnjak [310] investigated the heat transfer and pressure drop for flow boiling in axial micro-fin tubes having a 6.3 mm ID. The tested tubes were different from previous literature (Fig. 37). In the manufacturing process, tube expansion is used to maximize the contact between the outside surface of the micro-finned tubes and the external air-side fins. This expansion process, however, changes the fin geometry and degrades the heat transfer performance. In their work, the average enhancement (EF) and pressure drop factors (PF) were 1.34 and 1.23, respectively. Yang and Hrnjak [311] further showed that the liquid refrigerant during stratified flow (especially at low vapor quality and mass flux) was easily trapped in the grooves. Moreover, since the thermal resistance of the liquid is lower than that of vapor, the heat transfer inside the tube is increased.

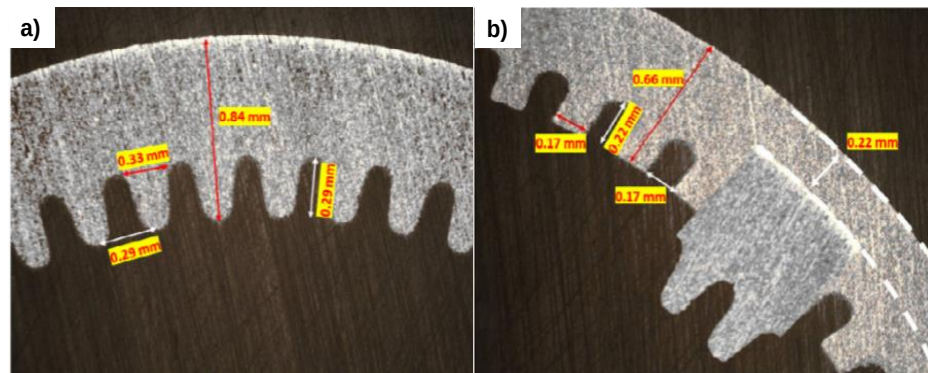


Figure 37. Unexpanded and expanded axial micro-fin tubes

Cross-sectional views of: (a) unexpanded and (b) expanded axial micro-fin tubes [310] . Reprinted from Int J Heat Mass Transf, 117, Yang et al., Effect of straight micro fins on heat transfer and pressure drop of R410A during evaporation in round tubes, p. 924-939, 2018, with permission from Elsevier.

6.2.3.1.4 Herringbone

Another way that extended surfaces can be arranged is the herringbone pattern. The pattern mainly constitutes two sets of extended surfaces forming a V-shape. Guo *et al.* [288] investigated the effect of a 2 m long herringbone tube having an 11.43 mm fin root diameter on the HTC. The tube had a V-groove shaped pattern (Fig. 38) that carried the fluid in two different flow directions,

which caused the film layer around the circumference to be nonuniform. Previous studies have proved that the structural orientation of the pattern affects the distribution of liquid around the walls and thus the HTC [312]. The study was carried out under low mass flux conditions using R22, R410A, and R32 as the working fluids.

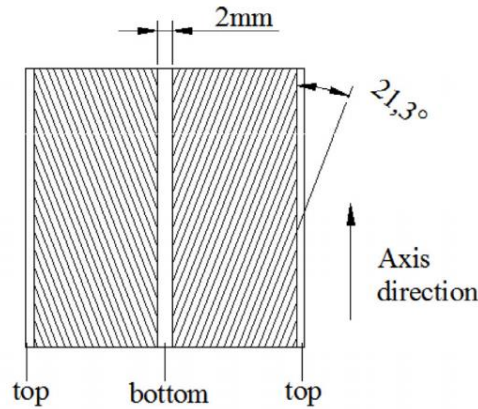


Figure 38. V-shaped herringbone

Schematic of a V-shaped herringbone [288]. Reprinted from Int J Heat Mass Transf, 85, Guo et al., Condensation and evaporation heat transfer characteristics in horizontal smooth, herringbone and enhanced surface EHT tubes, p. 281–291, 2015, with permission from Elsevier.

The results showed negligible heat transfer enhancement in the herringbone tube. It was noted that the EF ratio (Table 11) was far less than the surface area ratio, indicating that not all inner wall areas were participating in the heat transfer enhancement. This is because sections of the tube were exposed to the vapor and so were ineffective in heat transfer. Furthermore, since the saturation temperature was lower than the condensation temperature, liquid dry-out occurred which lowered the heat transfer enhancement. This finding was highlighted using Eq. (30), which determined that the performance factor (PFS) of the herringbone tube was lower than unity during evaporation.

Similar research conducted by Olivier *et al.* [313] investigated a micro-fin copper tube ($\beta = 18^\circ$) and compared it with a herringbone tube and a plain copper tube. The experiments were

conducted over a mass flux range between 400 and 800 kg/(m²·s) using R22, R407C, and R134a as the working fluids. It was reported that the pressure drop in herringbone tubes were 80% higher than plain copper tubes and 27% higher than micro-fin tubes. The higher pressure drop could be attributed to the fact that the surfaces extending from the tube walls prolonged the annular flow regime, which increased the vapor velocity and friction. However, unlike Guo *et al.* [288], the herringbone tube provided an enhancement in heat transfer. The average enhancement factor (PFS) was 70% higher than plain tubes and 40% higher than micro-fin tubes. Longo *et al.* [314] studied the effect of cross-grooved surfaces (Fig. 39) with a herringbone corrugation. The test was carried out with R22 as the working fluid and the results were compared to a plain tube. Repeating the experiment for a range of heat fluxes and Re_{dh} values demonstrated that the HTC increased by 1.3–1.4 times compared to a smooth surface.

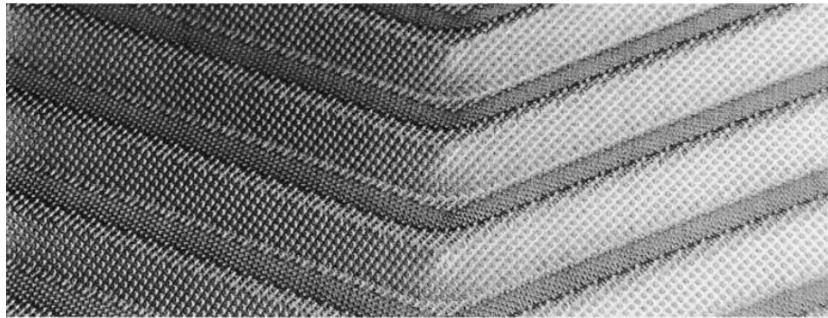


Figure 39. A cross-grooved plate

Image of a cross-grooved plate [314]. Reprinted from Int J Heat Mass Transf, 47, Longo et al., Experimental heat transfer coefficients during refrigerant vaporisation and condensation inside herringbone-type plate heat exchangers with enhanced surfaces, p. 4125–4136, 2004, with permission from Elsevier.

Filho and Jabardo [315] carried out experiments using R134a in herringbone and micro-fin copper tubes (Fig. 40). The experiment was conducted with a fixed heat flux of 5 kW/m² over a range of vapor qualities and mass flux values. It was reported that, over a wide range of mass fluxes, the herringbone tubes had superior thermal performance (PFE) compared to micro-fin

tubes. However, at low mass fluxes ($G = 100 \text{ kg}/(\text{m}^2 \cdot \text{s})$), PFE was the lowest. Furthermore, it was noted that the pressure drop of a herringbone tube was always higher than a micro-fin tube for all mass fluxes and vapor qualities investigated. The PFE for both micro-fin and herringbone tubes was greater than unity for mass fluxes lower than $200 \text{ kg}/(\text{m}^2 \cdot \text{s})$. For higher mass fluxes, both tubes had values lower than unity because of the significant pressure drops observed.

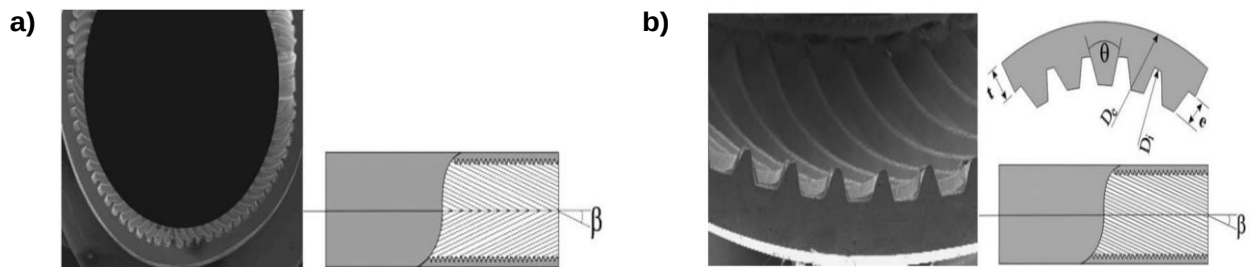


Figure 40. Herringbone and micro-fin tubes

Scanning electron microscopy images (left) and cross-section schematics (right) of (a) herringbone and (b) micro-fin tubes [315]. Reprinted from Int J Refrig, 29, Filho and Jabardo, Convective boiling performance of refrigerant R-134a in herringbone and microfin copper tubes, p. 81-91, 2006, with permission from Elsevier

6.2.3.1.5 Twisted Tapes

To increase the heat transfer coefficient in flow boiling applications, the heat transfer between the inner tube walls and the fluid must be optimized. This can be achieved by increasing the intermixing of the flowing fluid and increasing the skin contact area between the tube wall and the fluid. Twisted tapes effectively achieve both as they induce swirl flow inside the tube and act as an internal fin. To study the effect of twisted tapes on flow boiling heat transfer, Sarmadian *et al.* [316] investigated the effect of aluminum grade 6063 tapes of twist ratios 4, 10, and 15 on the heat transfer coefficient and pressure drop. Using R600a as the working fluid, the authors studied a range of vapor qualities and mass fluxes. The authors reported an increase in both pressure drop and heat transfer for tubes with inserts compared to plain tubes. Using PFE as a basis for comparison, the values ranged between 0.44-1.09. This indicated that for most of the mass flux

ranges studied, the pressure drop increase far overshadowed the increase in heat transfer coefficient rendering the tape as an unviable solution. Further, it was reported that the smaller the twist ratios, the higher the HTC and pressure drop increase. A similar investigation by Kanizawa *et al.* [317] compared aluminum tapes of twist ratios 3, 4, 9, and 14 on the heat transfer performance of tubes. Using R134a, the authors repeated the experiment over a range of heat and mass fluxes. It was reported that an improvement of up to 45% in the HTC was evident in tubes with tape inserts. Accompanying this augmentation is an increase in frictional pressure drop of up to 30 times. However, the increase in the heat transfer augmentation was found to be higher than the increase in pressure drop in the majority of the operating test conditions, indicating that twisted tapes insertion is recommended. Moreover, the study indicates that the vapor quality greatly affects the performance of tubes. Vapor qualities corresponding to intermittent and annular flow witnessed an augmentation in heat transfer, while vapor qualities corresponding to even higher vapor qualities had a degradation in heat transfer. Similar patterns were observed for pressure drops where higher vapor qualities corresponded to an increase in friction. This is due to the increase in fluid velocity that occurs during liquid-vapor phase change. Shishkin *et al.* [318] also conducted a similar investigation using R134a over a range of different twist ratios, tube diameters, mass fluxes, saturation temperatures, vapor qualities, and heat fluxes. The twisted inserts used are stainless steel and are covered by a thermostable powder paint in order to electrically insulate the tape from the tube walls. It was reported that having the tape insert induced an annular flow regime at lower than usual vapor qualities. In addition, it was noted that varying the mass flux had negligible effect on the HTC for mass and heat fluxes higher than $400 \text{ kg/m}^2\text{s}$ and 100 kW/m^2 , respectively. This is due to the fact that at much higher fluxes, turbulence is already dominant and so any fluctuations in the flow velocity reap minimal changes in fluid agitation and mixing. Yan

et al. [319] studied the effect of adding twisted tapes on the critical heat flux of highly subcooled water flowing through vertical tubes. Results show that having tapes, significantly increased the critical heat flux under high mass fluxes. Further, it was reported that the subcooled critical heat flux increased with decreasing twist ratios and vapor qualities. Additionally, it was found that increasing the mass flux increases the critical heat flux, however, this effect is dampened at regimes near the saturation point.

6.2.3.1.6 Wired Coils

Similar to twisted tapes, wired coils help agitate flow to enhance fluid intermixing. However, this agitation is more focused on the boundary layer near the tube wall. To study the effect of wired coils on the heat transfer and pressure drop, Behabadi *et al.* [320] inserted coils of different diameters and pitches in a 1.2 m long copper tube with an inner diameter of 7.5 mm. Using R134a as the working fluid, it was reported that having a coil insert can cause a pressure drop increase of up to 1000% when compared with plain tubes with no insert. Further, the enhancement of heat transfer was found to depend on the vapor quality, with the maximum enhancement in HTC peaking at 98%. Moreover, it was noted that the HTC increased with decreasing coil pitch up to a minimum of 8 mm. Further reduction in pitch would cause a decrease in the HTC as the insert resists heat transfer like a solid shell [321]. Since the pressure drop caused by the inserts is much higher than the increase in HTC, it was concluded that having wired coils is not favourable. Niezgoda-Żelasko [322] investigated wired coils of different pitches and diameters at a range of mass and heat flux values in vertical tubes. Using R507, R410, and R407C as the working fluid, the HTC for azeotropic fluids increased by 1.1-1.7 times, while the HTC for zeotropic agents increased by 1.1-1.3 times. Further, the pressure drop also increased by 1.8-4.5 times when

compared to plain tubes. Alimardani *et al.* [323] conducted a similar study but with R600a as the working fluid. Using horizontally oriented tubes, the authors noticed an increase in pressure drop by 90-958% when wired coils were inserted. However, this pressure drop is reduced when the wire diameter decreases, and the coil pitch increases. Additionally, the authors reported *PFE* values ranging between 0.13-1.4. Yun *et al.*[324] investigated wired coil inserts in tubes with nitrogen fluid. Using different wire pitches and diameters, the authors conducted experiments over a range of mass and heat fluxes. The wired coils augmented the HTC and pressure drop. It was noted that an increasing mass flux and decreasing heat flux caused an increase in the HTC. The pressure drop was also found to increase with decreasing mass flux. Generally, for the investigated range of mass flux values, Yun *et al.* reported *PFE* values greater than unity indicating that wired coils are an effective method to augment heat transfer. From the above studies, it can be concluded that due to the high pressure drop that characterize wired coil inserts, this heat transfer enhancement technique is only viable in very limited applications and test conditions.

6.2.3.1.7 Electrohydrodynamics

The final enhancement technique to be covered in two-phase flow boiling is an active method. In this method, dielectric fluid is subjected to an electric field to induce motions in certain directions. Like dimples and extended surfaces, electrohydrodynamics (EHD) can augment heat transfer in both single-phase flow and two-phase flow boiling.

For two-phase boiling, EHD can augment heat transfer in several ways. First, it is known that for low heat flux values, heat is transferred through forced or natural convection. As the heat flux or superheat increases, the *onset of nucleate boiling* (ONB) [260] occurs and some nucleation sites are activated. Applying EHD has been shown to initiate ONB at an earlier stage (*i.e.*, at a lower

wall temperature) and eliminate the hysteresis portion that occurs before ONB [325–328]. EHD also raises the critical heat flux (CHF) limit, and eliminates the heat flux drop between CHF and film boiling [325,329]. Moreover, when an electric field is applied, the surface tension of the dielectric fluid is lowered which increases its wettability on the surface [328,330] and enhances the heat transfer in a way similar to surfactants. Finally, since the frequency of bubble departure increases, the turbulence induced by the bubbles rising aid heat transfer [326,331,332]. Additionally, it is important to note that the inner walls of the tube surface have a prominent effect on heat transfer enhancement [325,333,334].

Cotton *et al.* [335] investigated the effect of EHD on the heat transfer of R134a in an annular channel under various mass fluxes and voltages. It was noted that at 24 kV and 60 Hz AC, the average HTC and pressure drop increased by 300% and 260%, respectively. This increase in heat transfer was attributed to the oscillation in flow which caused intermixing between phases and increased the interfacial area. Further, Cotton *et al.* [336] experimentally and numerically studied the impact of the electric field and the electrode position inside the tube. Using R134a as the working fluid, the experiment was carried out over a range of mass fluxes, vapor qualities, and voltages for three different electrode positions, mainly at the center and ± 2.73 mm from the centerline of the tube. It was found that heat transfer was enhanced by up to 60% compared to a tube with no electric field. Furthermore, the pressure drop increased by only 20%. There was also a positive correlation between the heat transfer augmentation and the applied voltage. However, this augmentation decreased with increasing mass flux. Using R134a, Salehi *et al.* [337] investigated EHD enhanced flow boiling inside a 1 mm round tube. The experiment was repeated twice for smooth and corrugated tubes over a range of voltages, Re_{dh} values (where $Re_{dh}=4\dot{m}/\Gamma\mu$), flow directions, and vapor qualities. It was reported that the technique was most effective at lower

Re_{dh} values. For $Re_{dh} = 500$, the boiling HTC increased tenfold when a 3 kV potential was applied over the corrugated tube with downward flow. Liu *et al.* [338] investigated the impact of electrode polarity on EHD heat transfer augmentation in a vertical tube. Using R123 as the working fluid, the electrode, carrying high negative or positive voltages, was placed concentric to the stainless-steel tube. The results showed that positive high voltages resulted in superior heat transfer enhancement and required lower electric field strength. The maximum enhancement factor amounted to 20.1 for an electric field strength of 1333 kV/m. Applying an electric field to a vertical tube created a dielectrophoretic force that pushed on the bubbles forming against the tube wall. This caused the bubbles to deform and spread along the surface which accelerated bubble growth by increasing the area of liquid film underneath the bubbles for evaporation. Furthermore, the force that caused thinning in the liquid film, increased the proportion of the bubbles in the superheated boundary layer, and increased the surface area per unit volume of the bubbles. However, if the intensity of the field increased above a certain threshold, the force acting on the bubbles would increase to the extent that bubbles would not be able to depart from the tube walls, preventing new liquid from replacing the gas and in turn reducing the HTC. Besides, a positive charge had the effect of increasing the body force on bubbles close to the surface causing an enhancement in heat transfer, while a negative charge had the opposite effect.

Compared with the passive techniques discussed above, a clear advantage that can be noted is the ability of EHD to enhance and affect every part of the boiling curve, in contrast to passive techniques that only aim to enhance the nucleate boiling region [325]. Furthermore, it can be noted how capable EHD is in enhancing HTC. However, active methods are not used as frequently as their passive counterparts due to the complexity such methods add.

6.2.3.2 Condensation

The main challenge in condensation heat transfer augmentation techniques is the removal of liquid condensate from the walls of the condensing surface as soon as they form [89,339]. As liquid condenses on the surface and the thickness of the liquid film increases [340], the thermal resistance to heat transfer from the vapor to the surface increases [88,341] which hinders heat transfer [195]. This subsection is dedicated to investigations that study the effect of tube enhancements on the HTC in the case of two-phase flow condensation.

6.2.3.2.1 Twisted Tapes

As highlighted, a twisted tape insert can enhance heat transfer through several means. Firstly, a twisted tape restricts the fluid to flow along the twisted pathway, which increases the flow length. Furthermore, a twisted tape serves as a fin, which increases the area of contact between the tube and the flowing fluid. Moreover, having a twisted tape also induces swirl mixing, which induces turbulence and vortices [124]. The increase in the axial velocity and the creation of vortices by the tapes aid in increasing the vapor shear effect and the condensate entrainment, which leads to further enhancement in HTC [342]. However, it is noted that using twisted tape inserts to enhance heat transfer adds obstruction to the fluid flow, and thus results in a significant pressure drop. Using R134a as the working fluid, Hejazi *et. al* [343] investigated the impact of a twisted tape on the in-tube heat transfer during condensation. The results revealed that twisted tape inserts enhance the HTC but also increases the pressure drop. Under the worst test conditions, the pressure drop experienced was 2.4 times more than that in a smooth tube. However, under the best test conditions, the condensation HTC increased by up to 40%. Overall, it was proven that the additional pressure drop attributed to the twisted insert was so large that it did not justify the use

its use. Akhavan-Behabadi *et al.* [344] examined the condensation heat transfer in tubes with twisted tape inserts of various twist ratios. Using R134a as the working fluid, it was reported that the HTC increased with the mass flux, while the twist ratios had a negligible effect. In addition, at low mass fluxes, the HTC was reported to increase by up to 38%. Agrawal *et al.* [345] studied the heat transfer augmentation provided by segmented and full-length twisted tape inserts. Using R22 as the working fluid, different mass fluxes were investigated. The enhancement in heat transfer during condensation was found to reach up to 25% for a full-length twisted tape insert. Although segmented twisted tapes also caused an enhancement in the HTC, it was reported to be relatively small. This enhancement improved with increasing the length of the segmented tape and was maximum for the full-length tape inserts. Furthermore, the authors reported that all the tape inserts were generally more effective at low mass fluxes. At higher mass fluxes, the axial velocity of the fluid is high and so the added enhancement provided by the tape in the flow field becomes inconsequential. In general, it is apparent that the HTC increases with the increase of segmental length and is the highest for the full-length tape inserts. However, it was reported that the full-length tape insert remained effective at the higher mass flow rates as well.

6.2.3.2.2 Wired Coils

Another type of widely used inserts is wired coils. Acting as added roughness, wired coils disturb the boundary layer of the fluid near the tube walls. This causes an enhancement in heat transfer in addition to pressure loss. Agrawal *et al.* [346] investigated the effect of a coiled wire on the condensation of R22 inside a horizontal tube. The investigation spanned over a range of wire diameters and coil pitches. Results reveal that having a wired coil insert could increase the condensing HTC by as much as 100%. Further, the HTC enhancement increased with wire diameter for high vapor qualities and vice versa for low vapor qualities. At high vapor quality, the

liquid film thickness on the inner walls of the tube decreased, which lowered the thermal resistance. Similar observations were exhibited for tubes with micro-fins and twisted tape inserts [347,348]. Furthermore, for a given wire diameter and low vapor quality, the heat transfer enhancement increased as the coil pitch decreased. To study the pressure drop associated with such an insert, Akhavan-Behabadi *et al.* [349] studied a double-pipe heat exchanger with R134a as the working fluid. Coils of different diameters and pitches were experimentally studied. It was reported that using a coiled wire insert increased the pressure drop by 260–1600% compared to a plain tube with no insert. Moreover, the pressure drop decreased with decreasing wire diameter and increasing wire pitch. A similar investigation of pressure drop due to coiled wires was conducted by Salimpour and Gholami [350]. Using R404A, the experiment was carried out over a range of mass fluxes and vapor qualities. A pressure drop increase of up to 1300% was observed compared to smooth tubes. This pressure drop was reported to increase with coil diameter and reduced coil pitch.

6.2.3.2.3 Ribbed Tubes

Ribbed tubes or corrugated tubes, like coiled wires, add roughness that disturbs and limit the growth of fluid boundary layers. Laohalertdecha and Wongwises [351] studied the effect of corrugation pitch (Fig. 41) on the HTC. The R134a condensation experiments were conducted with three tubes of different corrugation pitches (5.08, 6.35, and 8.46 mm) at different saturation temperatures, heat, and mass fluxes.

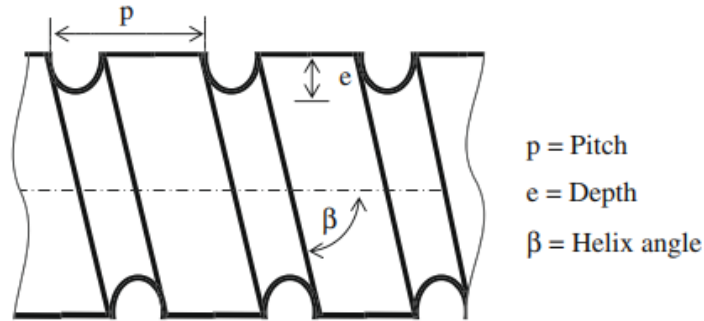


Figure 41. Corrugated tube

Side-view schematic of a corrugated tube [351]. Reprinted from Int J Heat Mass Transf, 53, Laohalertdecha and Wongwises, The effects of corrugation pitch on the condensation heat transfer coefficient and pressure drop of R-134a inside a horizontal corrugated tube, p. 2924-2931, 2010, with permission from Elsevier.

The study found that the average HTC and pressure drop increased with mass flux and vapor quality. Furthermore, increases of up to 50% and 70% in the HTC and frictional pressure drop, respectively, were shown for corrugated tubes. Khoeini et al. [352] studied the effect of inclination ($-90^\circ \leq \alpha \leq +90^\circ$) on the heat transfer of R134a inside a corrugated tube. The results showed that at lower velocities and vapor qualities, the highest heat transfer augmentation was achieved at an inclination of $+30^\circ$. This was 1.27–1.41 times greater than the worst achieved heat transfer augmentation occurring at -90° . The latter had the lowest heat transfer performance since the interfacial shear stress and the gravity forces both acted together. As a result, the flow remained annular. The thickness of the condensed liquid on the wall increased, increasing the thermal resistance, which caused a decrease in heat transfer. Fernández-Seara and Uhía [353] studied the condensation of ammonia on spirally corrugated tubes. The setup constituted of a heat exchanger with water inside the corrugated tube and ammonia surrounding the outside of the tube. The experiment was carried out over a range of wall subcooling values, and it was reported that the overall performance for corrugated tubes was 1.27 times higher than plain tubes. However, the

enhancement factor of the condensing HTC turned out to be close to unity over the range of wall subcooling temperatures tested.

6.2.3.2.4 Micro-fins

Fins enhance the condensation HTC because they increase the surface area as well as the tendency of thin-film condensate to form on their tips [254]. Jung *et al.* [354] examined the effect of micro-fin tubes on the flow condensation HTC for a range of mass and heat fluxes. Using R22, R134a, R407C, and R410A as the working fluids, it was reported that using micro-fin tubes caused a twofold to threefold rise in the HTC compared with smooth tubes. This enhancement, however, decreased with increasing mass flux. These results matched another investigation by Kwon *et al.* [355], who studied the condensation in a micro-fin tube. Using R22 and R410A, the setup consisted of a counterflow heat exchanger where the refrigerant flowed inside the finned tube and the cold water flowed outside of the tube. Carried over a range of mass fluxes, the enhancement ratio of the finned tube increased by 1.5 to 2.2 times based on the nominal heat transfer area. This enhancement was found to negatively correlate with the mass flux. Eckels and Pate [356] conducted their investigation with a tube containing 60 fins with 0.2 mm height and 17° spiral angle. Using R134a and CFC-12 as the flowing fluids, the study was conducted over a range of mass fluxes and wall temperatures. Using the fins provided a positive effect on the heat transfer for both refrigerants. For HFC-134a, it was noted that the HTC during condensation increased by 1.75–2.5 times that of smooth tubes. In the case of R12, the increase was 1.7–2.3 times. Accompanying this enhancement was also an increased pressure drop. However, it was noted that in almost all cases, the increase in HTC overshadowed the pressure drop increase. In consequence,

having micro-fins proved to be an effective heat transfer enhancement technique. Similarly, Naulboonrueng *et al.* [357] carried out their investigation using R134a as the working fluid. A 2.5 m long tube containing micro-fins was subjected to temperatures between 30°C and 40°C. It was reported that the HTC during condensation was 1.1–1.85 times higher than that of smooth tubes because the micro-fins promoted mixing. This HTC enhancement increased with higher quality and mass flux. Further, Naulboonrueng and Wongwises [358] went on to investigate the pressure drop in the same micro-fin tube using the same refrigerant. It was reported that the disturbance in the fluid flow caused by the micro-fins increased the pressure drop by 10–50% compared with a smooth tube. Moreover, the pressure drop increased with vapor quality and mass flux. Finally, Graham *et al.* [359] investigated, over a range of mass fluxes, the effect of axially grooved tubes on the condensation of R134a and compared it with the performance of helically grooved tubes ($\beta = 18^\circ$). The extent of heat transfer enhancement was dependent on the mass flux and quality ranges. For mass fluxes of 150 kg/(m²·s), axially grooved tubes provided better thermal performance than helically grooved or smooth tubes. However, the pressure drop for the axially grooved tubes was similar to that for helically grooved tubes, and the pressure drop for both was higher than that of plain tubes.

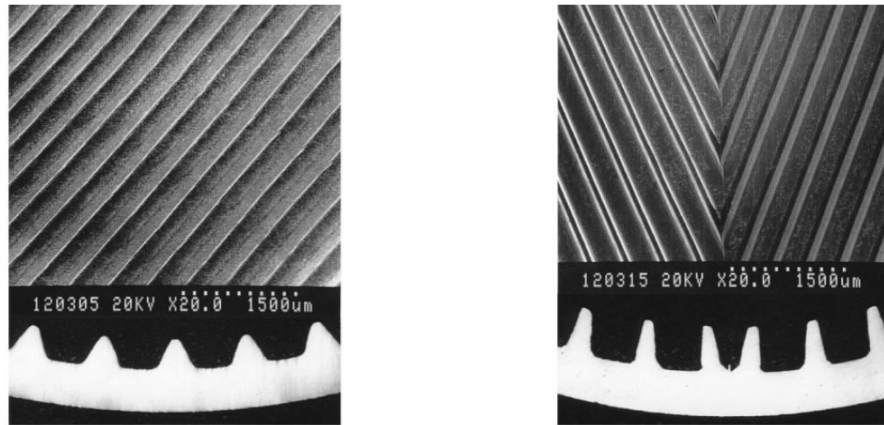
Eckels *et al.* [360] studied the use of 150 SUS pentaerythritol ester lubricants in a micro-fin tube. The tube contained 60 fins with 18° spiral angles. Using R134a, Eckels *et al.* reported that the condensation HTC and pressure drops were 50–200% and 10–50% higher, respectively than smooth tubes. The increase in heat transfer was adversely affected by higher lubricant concentration, while the pressure drop was unaffected. A similar investigation by Cho and Tae [361] used polyol-ester (POE) and mineral oils instead of the ester lubricant. R22 and R407C were used as the working fluids, and the experiment was extended to tubes with U-bend sections. The

experiment was conducted over a range of oil concentrations, mass fluxes, and inlet vapor qualities. For straight sections before the bend, it was reported that the oil concentrations adversely affected the heat transfer augmentation since it increased the viscosity of the flowing fluid. In the U-bend section, the heat transfer augmentation was reported to be at maximum value. The enhancement was attributed to the fact that the U-shape bend promoted secondary flows and flow disturbances. Moreover, the straight section following the bend showed a higher average HTC than the straight section preceding the bend. This is because the secondary flow and disturbances occurring at the bend extended to the straight section after the bend. Finally, the pressure drop was also reported to increase with oil concentration.

6.2.3.2.5 Herringbone

Miyara *et al.* [362] examined how herringbone patterns affected the condensation of R410A and R22 and compared it with conventional helical micro-fin tubes. The authors reported that for high fluid velocities, the performance of the herringbone pattern was superior to that of the helical micro-fin tube. However, at low velocities, the opposite was true. In general, herringbone tubes were able to collect liquid flowing at the top and bottom sections of the tubes. Consequently, the liquids left at the sides of the tubes were much thinner, enhancing heat transfer. Moreover, like conventional helical fins, herringbone tubes promote turbulence and mixing which enhances heat transfer. Finally, regardless of the velocity, the herringbone micro-fin tube always had a pressure drop larger than that of the helical fin tubes. Miyara *et al.* [363] went on to study the effect of the fin shape on the herringbone-patterned extended surface. The study was conducted using R410A. Results show that longer fins led to an increase in heat transfer. However, this enhancement had a plateau when the fins reached a length of 0.18 mm. Furthermore, it was reported that the HTC

increased with higher helix angles. This was attributed to the fact that more-helical fins increased the circumferential force acting on the liquid which affects the film thicknesses. Although changing the fin shape augmented the heat transfer enhancement, the pressure drop increased with the fin height or helical angle. Goto *et al.* [364] investigated herringbone grooves and spiraled grooves (Fig. 42) and their effect on the heat transfer coefficient.



a) Spiral grooved tube (C Tube) **b)** Herringbone grooved tube (W Tube)

Figure 42. Spiral and herringbone grooved tubes

Scanning electron microscopy images of (a) spiral grooves, and (b) herringbone grooves [364]. Reprinted from Int J Refrig, 24, Goto et al., Condensation and evaporation heat transfer of R410A inside internally grooved horizontal tubes, p. 628-638, 2001, with permission from Elsevier.

Using R410A and R22, the investigation was conducted over a range of mass fluxes. It was reported that herringbone grooves provided twice the increase in HTC compared with spiraled grooves. The heat transfer enhancement was due to the enhanced mixing of the liquid film of refrigerant in addition to the increased surface area for heat transfer. Further, Goto *et al.* [365] conducted a similar experiment with R410A and R22 at a different range of mass fluxes and pressures. The results of this investigation agreed with the results of the previous Goto *et al.* study showing twice the HTC in herringbone tubes compared with helical tubes [364].

6.2.3.2.6 Electrohydrodynamics

When condensation occurs, the least volatile substances condense first. As time passes, the concentration of volatile liquids in the vapor increases to the extent that a concentration gradient arises between the vapor near the free surface and the bulk flow. Consequently, diffusion starts to occur which obstructs the flow of any condensing vapor and so decreases the heat transfer due to the increase in interfacial resistance. EHD is used to reduce this resistance and augment the heat transfer [366]. Gidwani *et al.* [366] examined the effect of EHD using R134a, R407c, and R404a. The experiment was conducted with smooth and corrugated tubes. Using R404a, results showed that the use of an 18 kV electric field enhanced the heat transfer performance by 18.8 times for smooth tubes, and increased the pressure drop by 11.8 times. Repeating the experiment for corrugated tubes, the heat transfer enhancement reported was 5.8 times. For R134a, the maximum heat transfer augmentation inside a corrugated tube was reported to be 8.3 times higher than that in a smooth tube, and the pressure drop was 20.8 times higher. Finally, for R407c, the heat transfer increased by 3.9 times and 2.9 times for smooth and corrugated tubes, respectively. An increase in HTC was also reported by Choi [367]. Using R113 as the working fluid inside a vertical tube, a radial DC field was applied. The electric field caused increased the thinning of the liquid film, causing the condensing HTC to increase. For an applied voltage of 30 kV, an increase on the order of 100% was reported for the condensing HTC downstream of the tube. Didkovsky and Bologa [368] investigated the effects of electric fields of different strengths, frequencies, and uniformities on the condensation of stagnant water vapor inside a tube. It was reported that the HTC increased by 20 times compared with no electric fields. This was attributed to the thinning of the film layer in the recesses between transverse waves.

6.2.4 Advanced Manufacturing Processes

In the previous subsections, techniques to enhance two-phase flow in channels were presented. However, despite extensive developments in the active and passive enhancement techniques mentioned above, there are major limitations for the maximum achievable performance due to constraints in channel design that stem from production using conventional manufacturing processes. Recent developments in additive manufacturing, however, provide unprecedented solutions in the form of extremely complex configurations for thermal management [230,247,369]. For example, the deployment of a structure with a controlled feature size has been made possible due to three-dimensional printing technology[370]. As illustrated in Fig. 43, the diameter of the copper sphere can be varied providing an effective technique for enhancing the boiling process due to continuous contact of liquid on the hot surface [371]. To date, there are several experimental studies to enhance pool boiling heat transfer coefficient by deploying porous structures and structural modifications [372,373].

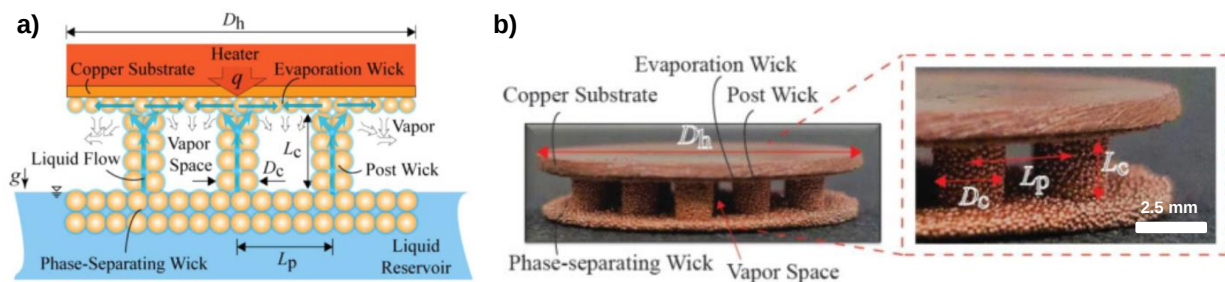


Figure 43. Wicking structure for enhanced boiling

Wicking structure for enhanced boiling heat transfer developed by sintering of copper spheres where: (a) Schematic and (b) Fabricated sample [371]. Reprinted from Int J Heat Mass Transf, 155, Egbo et al., Phase-change heat transfer of sintered-particle wick in downward facing orientation: Particle size and wick thickness effects, 119840, 2020, with permission from Elsevier

For example, Ho *et al.* [374] conducted an experimental investigation to enhance the pool boiling heat transfer performances of surfaces with microcavities and micro-fins fabricated by the SLM (selective laser melting) technique (Fig. 44a).

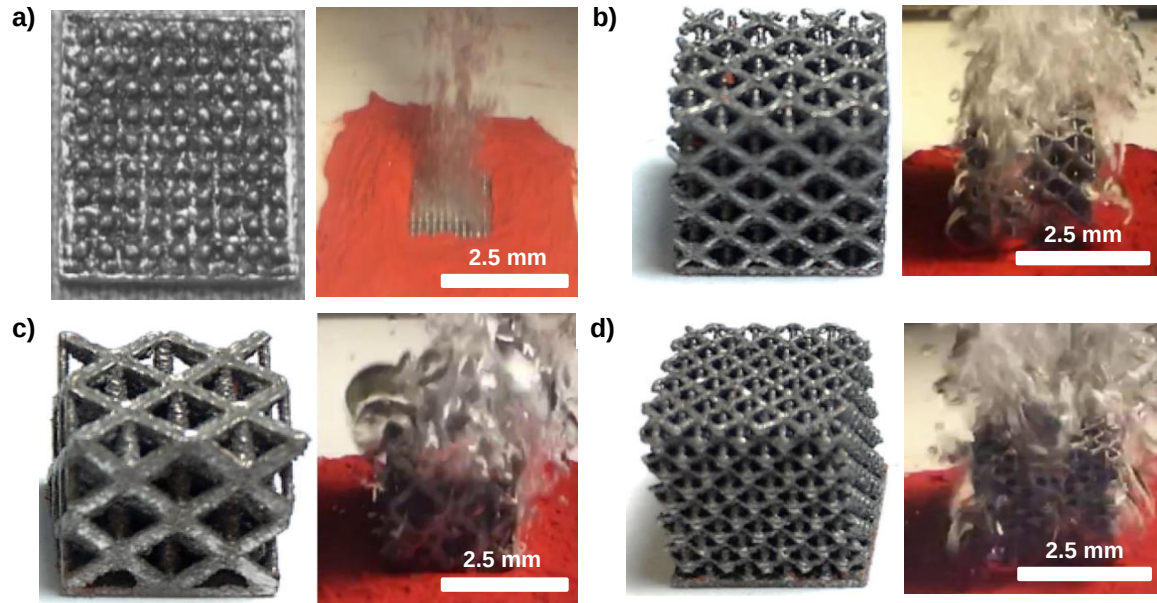


Figure 44. Micro-fin array with water boiling

(a) micro-fin array of fin size of total surface area size of 1.35 cm^2 with pool boiling [374]. Adapted from Int J Heat Mass Transf, 99, Ho et al., Saturated pool boiling of FC-72 from enhanced surfaces produced by Selective Laser Melting, 107-21, 2016, with permission from Elsevier. Porous structures in pool boiling setup where: (b) Unit cell size 2.5 mm with dry-out boiling, (c) Unit cell size 5.0 mm with dry-out boiling, (d) Unit cell size 10.0 mm with water replenishment [375]. Adapted from Int J Heat Mass Transf, 121, Wong and Leong, Saturated pool boiling enhancement using porous lattice structures produced by Selective Laser Melting, p. 46-63, 2018, with permission from Elsevier

Using FC-72, an improvement of around 70% was reported in the average HTC as compared to a plain Al-6061 surface. In another investigation, Wong *et al.* [375] studied the effect of porous structure thickness. Approximately threefold enhancement was shown in the HTC and an improvement of a similar order for the CHF compared to a plain surface (Fig. 44b-d). It is important to note that despite being an active area of research, there are obvious discrepancies in findings. For example, Rainey and You [376] prepared samples using additively manufactured samples and demonstrated noticeable improvement in the HTC. However, in another study with similar experimental arrangements, Kim *et al.* [377] observed a relatively lower improvement in the HTC. It was further reported that under certain conditions, the samples manufactured by additive manufacturing showed reduced performance compared to the baseline case. Overall, there is a consensus that when designed appropriately and fabricated with reasonable dimensional

accuracy, the sample produced by additive manufacturing can lead to a remarkable increase in thermal performance. Regardless of the extensive developments, it is important to note that most of the research and development work in the field of two-phase heat transfer using additive manufacturing has been focused on pool boiling.

The applications where advanced manufacturing has led to extensive enhancement in thermal-hydraulic performance for two-phase flow include heat pipes, heat sinks, and heat exchangers [378]. Heat pipes have been studied extensively due to the special feature of dispersing heat over large distances. Due to the nature of the operation, two major issues with conventional heat pipe technology include the dependence of dissipation rate on the design/characteristics of the wicking structure and the limited flexibility to adopt complex geometries

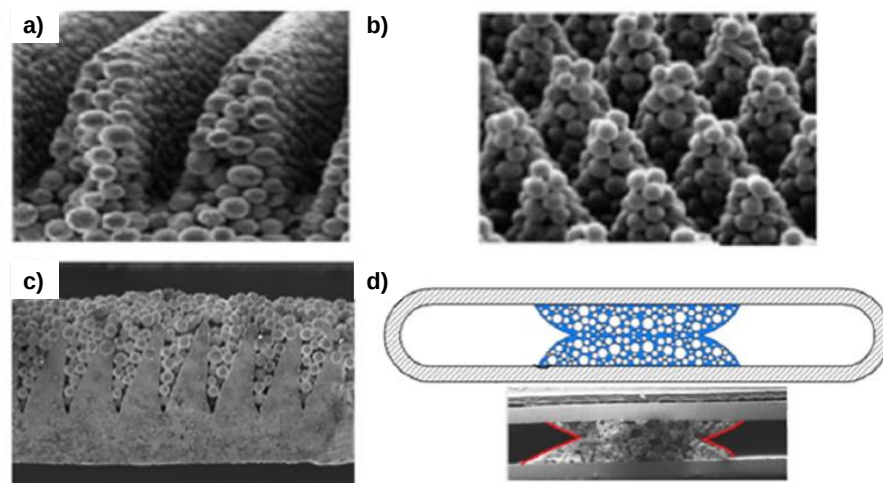


Figure 45. 2-D and 3-D modulated porous coatings

SEM imaging of 2-D and 3-D modulated porous coatings of height: (a) 1.7 mm and (b) 1.2 mm [379]. Reprinted from Int J Heat Mass Transf, 52, Min et al., Kaviany M. 2-D and 3-D modulated porous coatings for enhanced pool boiling, 2607-13, 2009, with permission from Elsevier. (c) SEM imaging of wick with sintered spherical powder of 75-110 μm diameter [380]. Reprinted from Int J Heat Mass Transf, 56, Deng et al., Characterization of capillary performance of composite wicks for two-phase heat transfer devices, p. 283-293, 2013, with permission from Elsevier. (d) Ultra-thin heat pipes and SEM imaging of the wick structure [381]. Adapted from Appl Therm Eng, 86, Li et al., Investigation of ultra-thin flattened heat pipes with sintered wick structure, p. 106-118, 2015, with permission from Elsevier

. Both issues have been effectively resolved by additive manufacturing since the advanced processes can produce porous wicking structures with varying levels of complexity and feature sizes [379–383]. Further, it has become possible to embed heat pipes to the applications under consideration regardless of the complexity of the shape. Some examples of the deployment of porous structures in heat pipes are presented in Figure 45.

Ameli *et al.* [384] carried out an investigation on ammonia-based heat pipes fabricated using AM of aluminum alloy. The investigation focused on the characterization and deployment of wick structures of different porosities, thicknesses, pore sizes, and permeabilities in different regions of a heat pipe to maximize the performance (Figure 46).

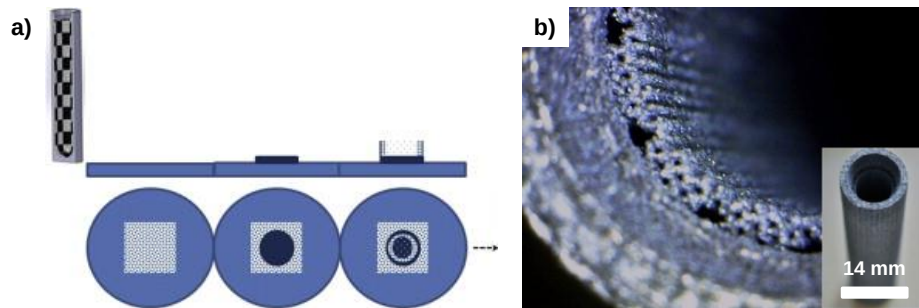


Figure 46. SLM fabricated HP and arterial wick

(a) Fabricated HP by SLM and (b) Close-up to arterial wick fabricated through SLM [384]. Adapted from Appl Therm Eng, 52, Ameli *et al.*, A novel method for manufacturing sintered aluminum heat pipes (SAHP), p. 498-504, 2013, with permission from Elsevier

In another experimental study, Esarte *et al.* [385] designed a linear heat pipe where selective laser sintering process (SLS) was used for the fabrication of the primary wick. The heat pipe was utilized to manage the heat produced from a Light Emitting Diode lamp. The hydraulic-thermal limits for such a system were established based on design parameters such as permeability, capillary pumping, and wettability of the system. Similarly, Thompson *et al.* [386] utilized an additive manufacturing process to manufacture a compact flat-plate oscillating heat pipe. The heat

pipe, made of Ti-6Al-4V, was shown to exhibit a thermal conductivity of approximately 110 W/m K at a power input of 50 W which was reported to be 400–500% higher compared to the thermal conductivity of solid Ti-6Al-4V. Follow-up work conducted by Ibrahim *et al.* [387], a multi-layered Ti-6Al-4V based oscillating heat pipe was fabricated with improved design (Fig 47).

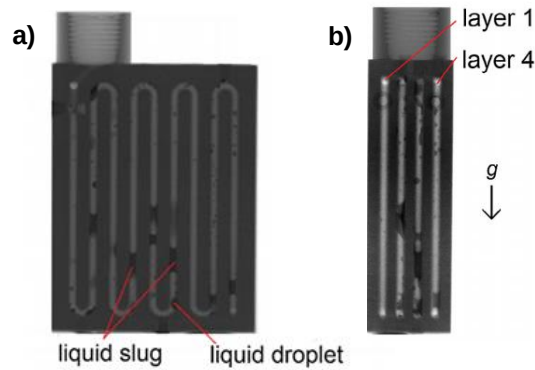


Figure 47. Four inter-connected layered heat pipe

Neutron radiographs of four inter-connected layered heat pipe ($50.8 \times 38.1 \times 15.75 \text{ mm}^3$) with residual water slugs and droplets where: (a) Front and (b) Side view [387]. Reprinted from Int J Heat Mass Transf, 108, Ibrahim et al., An investigation of a multi-layered oscillating heat pipe additively manufactured from Ti-6Al-4V powder, p. 1036-1047, 2017, with permission from Elsevier

The performance of the heat pipe was assessed at different power inputs where the neutron imaging facility at Oak Ridge National Lab was used to observe the behavior of two-phase flow in channels. Additionally, different fluids including n-pentane, Novec™ 7200, acetone, and water were used in the study.

Jafari *et al.* [388] manufactured a stainless-steel porous structure for heat pipe with an average pore size of 160 μm . The study focused on the establishment of the effective thermal conductivity of the wick structure under evacuated conditions and when it was filled with ethylene glycol or water.

Due to the development of compact electronic applications, thermal management of the heat generated has become an enormous challenge. Consequently, extensive research efforts have

focused on the development of heat sinks that can effectively disperse heat to mitigate hot spots. Advanced manufacturing methods have been critical to overcoming technological challenges for thermal management in high-power modules and packaging systems. Syed-Khaja *et al.* [389] noted that for high power applications, temperatures can reach elevated levels of 200 °C and higher at the interconnected level. As such, the design of the packaging of such a system becomes extremely critical. As a result, a heat sink was developed for high-power electronic systems and demonstrated that a significant reduction in volume, weight, and maximum chip temperatures can be achieved. Li *et al.*, [390] used additive manufacturing for the thermal management of batteries where unprecedented area and specific capacity was enabled by extrusion-based additive manufacturing. Similarly, Ferrari *et al.* [391], Fu *et al.* [392] and Sun *et al.* [393] investigated performance improvement by utilizing high-performance thermal management for micro batteries enabled by advanced manufacturing. Another application, where the design of improved heat sinks is critical, are advanced illumination systems. Kim *et al.* [394] and Lin *et al.* [395] reported that the lifetime of the LED-based illumination system is considerably shortened by hot spots. Consequently, the authors developed a heat pipe and heat sink based solutions using advanced manufacturing processes.

The development of highly efficient fuel cells is critical for clean energy production. Despite the extensive capital cost, there has been an increasing interest in various types of fuel cell technologies due to their higher efficiency and reduced environmental impact.

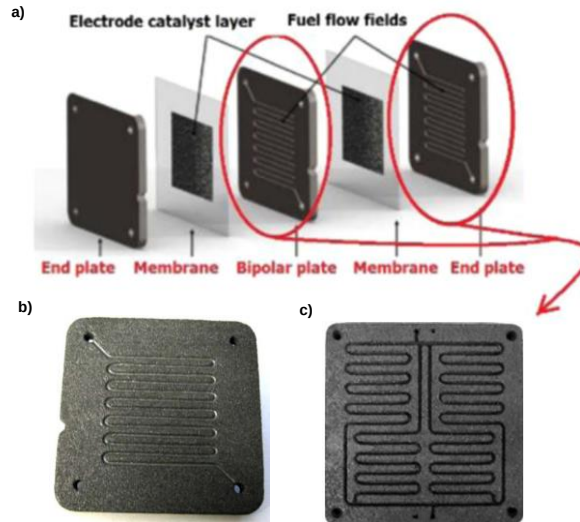


Figure 48. Fuel cell stack

(a) Fuel cell stack (b) midsection of bipolar plate ($50 \times 50 \times 5$ mm) [396]. Reprinted from Rapid Prototyp J, 16, Alayavalli and Bourell, Fabrication of modified graphite bipolar plates by indirect selective laser sintering (SLS) for direct methanol fuel cells, 268-74, 2010, with permission from Emerald Insight (c) Serpentine bipolar plate ($50 \times 50 \times 4$ mm) [397]. Reprinted from Int J Hydrogen Energy, 37, Effect of different graphite materials on the electrical conductivity and flexural strength of bipolar plates fabricated using selective laser sintering, p. 3558-3566, 2012, with permission from Elsevier

As a result, extensive research has been devoted to the development of thermal management of fuel cells using advanced manufacturing. Kaushik and Bourell [396], used additive manufacturing to fabricate electrically conductive bipolar plates for a direct methanol fuel cell application and demonstrated a significant performance improvement. Similarly, Guo and Leu [397], evaluated the electrical conductivity of bipolar plates produced by additive manufacturing to those produced by injection and compression molding. In another study [398], a bio-inspired “leaf” design was developed for bipolar plates which demonstrated a significant enhancement in the power density of the fuel cell. Figure 48 illustrates some of the designs for bipolar plates used in various studies produced by additive manufacturing.

Another relevant area that utilizes advanced manufacturing, is the design of efficient reactors for chemical production. Recent studies in this area have proved that enhancement of two-phase flow for such applications is possible by incorporating high surface area porous media [399].

Hutter *et al.* [399,400], evaluated the performance of tubular reactors deploying commercially available metal foams and free-form porous structure produced by the SLS process. Hutter *et al.* concluded that the fully sintered metallic porous foams provided twice as higher HTC compared to commercially available foams. Figure 49 shows the different porous structures inserted in the tubular reactor.

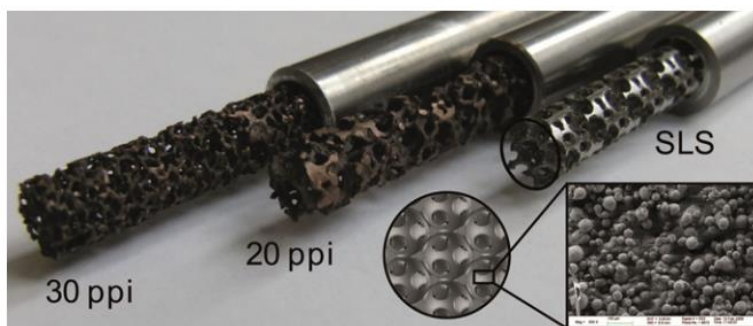


Figure 49. Inserted commercial foam elements

Image of inserted commercial foam elements (20 and 30 ppi) and fully sintered structure (SLS). The SEM image with x500 magnification of SLS indicates the roughness [400] . Reprinted from Chem Eng Sci, 66, Hutter et al., Heat transfer in metal foams and designed porous media, p. 3806-3814, 2011, with permission from Elsevier

While additive manufacturing has been studied in the past, their deployment in commercial applications is often a challenge especially when the equipment cost is a major consideration. However, in some special applications, such as the aerospace industry, the cost is not the decisive factor, and other factors such as the higher specific power and compactness play a more critical role. One of the areas where additive manufacturing has played a crucial role in such an industry is the development of advanced electro-thermal propulsion systems. To achieve a compact solution with high-temperature thermal management for aircrafts, Romei *et al.* [401] utilized the SLS process and achieved highly refined feature sizes using 316 stainless steel (Fig. 50a,b). Saltzman *et al.* [402] and Hathaway *et al.* [403], fabricated an oil cooling system for aerospace applications and reported a major reduction in the weight of the device with a noticeable performance

improvement. Vanapalli *et al.* [404] manufactured a highly compact flat-panel gas-gap heat switch (Fig. 50c-g) to provide cooling in cryogenic temperature ranges using helium, hydrogen, and nitrogen as the working fluids.

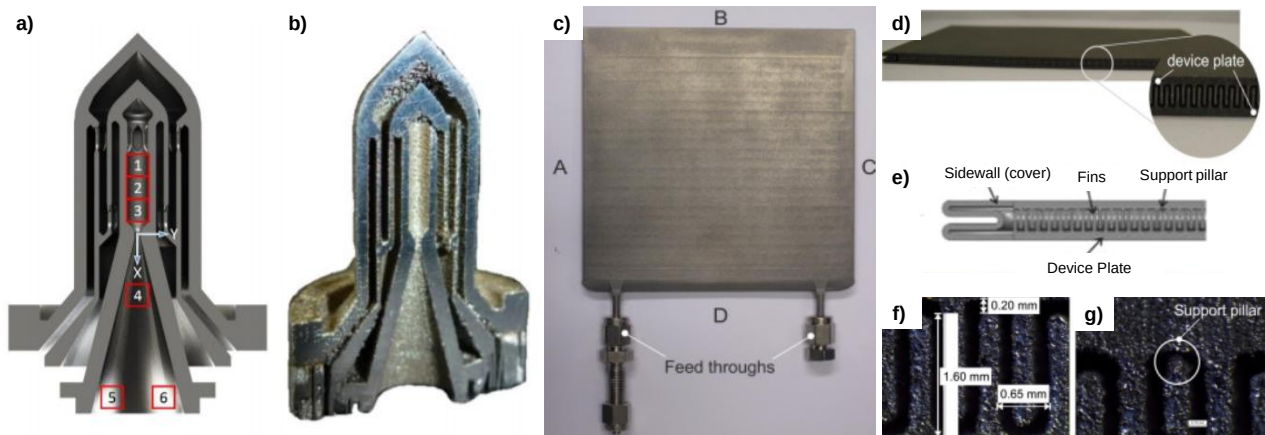


Figure 50. AM manufactured Heat exchanger half-section

AM manufactured Heat exchanger half-section with 0.42 mm nozzle throat diameter: (a) Schematic and (b) Manufactured part [401]. Reprinted from Acta Astronaut, 138, Romei et al., Manufacturing of a high-temperature resistojet heat exchanger by selective laser melting, 356-68, 2017, with permission from Elsevier. (c) Heat switch of size 11 cm × 11 cm × 3.2 mm (d) Cut section of the heat switch, (e) Fins in contact with device base (f) Image of fin dimensions, and (g) Image of support pillars [404]. Adapted from Cryogenics (Guildf), 78, Vanapalli et al., Cryogenic flat-panel gas-gap heat switch, p. 83-88, 2016, with permission from Elsevier

In conclusion, advanced manufacturing technology has a major impact on enhancing two-phase flow. The development of complex porous structures and embedded systems has opened a new field of ultra-compact and highly efficient solutions. While the majority of the existing literature focused on enhanced pool boiling, the developments in additive manufacturing have also provided opportunities to develop efficient two-phase flow systems such as heat pipes. Furthermore, several application-based solutions have been proposed for highly compact heat sinks, and heat exchangers including high power electronics, fuel cells, and aerospace.

6.2.5 Effect of Modern Low GWP Refrigerants

Rising concerns about the adverse effects of global warming and climate change has been the main driver to the phasing out of many refrigerants in recent history. Most notably, due to the Montreal Protocol, CFCs (chloroflourocarbons) has been banned from refrigeration cycles due to its ozone depleting characteristics [405]. Subsequently, a new generation of refrigerants HFCs (hydroflourocarbons) were introduced as viable substitutes due to their zero-depleting potential but still advantageous thermophysical properties [406–412]. However, these next-gen refrigerants were noted to contribute greatly to the production of greenhouse gases and are being phased out in recent legislations due to their high global warming potential [413]. As such, to abide by the increasingly stricter environmental registrations, natural fluids such as ammonia [414–416] , carbon dioxide (R744) [417–420], water (R718), amongst others have gained renewed interest. However, wide-scale adoption of these refrigerants in refrigeration applications are hindered due to their poor safety properties or their inability to operate in conditions relating to current refrigeration cycles [421]. Alternatively, new and synthetic alternative refrigerants with low GWP and zero ODP are being sought out namely, hydrofluoroolefins (HFOs) [422–424], hydrochlorofluoroolefins (HCFOs) [425–427], hydrofluoroethers (HFEs) [428–430]. Most notably, R1234ze(E) and R1234yf have gained significant interest due to their environmental and safety related characteristics. Even though they are costly and reduce system efficiency, R1234yf and R1234ze(E) has been adopted as a refrigerant for automobile applications [431,432] and commercial refrigeration cycles [433], respectively in order to replace R134a. However, in addition to lower performance, R1234yf and R1234ze(E) also has degradation concerns that could potentially have a negative impact on the ecosystem [433].

From the preceding discussion, it can be noted how environmental constraints pushed by different legislations overshadow the importance of system efficiency and cost reduction. As such, there is a continuous push to produce new synthetic refrigerants or adapting current cycles to operate under natural refrigerants. Currently, there is yet to exist a refrigerant that encompasses all desirable properties including low GWP, zero ODP, low cost, high performance etc. Furthermore, it is also important to note that different heat transfer augmentation techniques discussed in the aforementioned chapters are dictated by the thermophysical and fluid properties of the refrigerants. Consequently, when a new refrigerant is adopted into a pre-existing augmented heat exchanger, a separate study needs to be carried out to quantify how the fluid properties would affect the overall system efficiency. As a result, ascertaining the potential to use one refrigerant over the other in current systems would be dictated by the relative importance assorted to the different refrigerant characteristics while still abiding by the minimal acceptable environmental impact.

6.2.6 Discussion

This subsection presents a comprehensive review of heat transfer augmentation techniques for two-phase flows required in the design and development of next-generation heat exchangers. Generally, these enhanced heat exchangers can be manufactured by either conventional manufacturing techniques or other advanced manufacturing techniques such as AM. To summarize, Tables 9 and 10 highlight the most important findings in the flow boiling and flow condensation research fields, respectively. Furthermore, Table 11 summarizes efficiency correlations along with proper citations to aid future design. Several different techniques were reviewed, each of which has its strengths and limitations. The first method studied was the deployment of surfactants inside the flowing two-phase fluid. Surfactant additives are not intrusive

and would not require any change to the existing designs. Therefore, such a technique is relatively inexpensive to implement. However, the use of surfactants is limited to lower heat fluxes as well as for higher surface tension working fluids such as water or ethylene glycol. As the heat flux increases, the adsorbed surfactant molecules on the tube surface no longer adhere to the surface, allowing the tube to return to its original state [280]. Moreover, surfactant additives may not be appropriate for some industries, such as the food industry, in which surfactant could be considered as a contaminant. Although some surfactants were demonstrated to enhance HTC, others did not [281]. The second set of enhancement techniques include the deployment of inserts, namely coils, or tapes. Using such inserts is a cost-effective way to enhance in-tube heat transfer. Inserts are easy to install and additional treatment to the tubes is not required. However, it is important to keep in mind that such inserts would need to be replaced occasionally, as they are susceptible to wear or fouling[434]. Aside from inserts, other popular techniques such as extended surfaces, dimples, and ribs directly modify the tube wall geometry. Note that such techniques could also be used to augment heat transfer in single-phase flows [435]. Unlike other passive techniques, such as tape and wire coil inserts, dimpled tubes or tubes with ribs incur less maintenance cost as there is no need to replace tapes or inserts. Furthermore, the risk of blockage is minimal. However, implementing such tubing on existing systems is difficult because smooth tubes would need to be replaced with ones with dimples/ribs. Inserts (coils/tapes) may not have this issue. Finally, as shown in Table 8, dimples can cause significant pressure drops. As highlighted previously, extended surfaces can take many different geometries such as fins and herringbone. Micro-fins on tubes offer a solution to achieve heat transfer enhancement. Although it might be relatively more challenging to manufacture fins than to simply add a tape insert, fins provide in many cases much higher heat transfer enhancement than pressure drop increase. However, for evaporative two-phase

flows, the added benefit of micro-fins is mainly justifiable for lower mass fluxes. Therefore, if the existing system has a high flow rate, other enhancement techniques should be sought out. Another positive feature that fins provide in condensation applications is self-drainage. Since thin film condensate tends to form on the tip of the fin, the surface tension of the fluid pulls the film in the valley located in between the fins. This not only provides self-drainage, but it also has the added benefit of reducing the liquid film resistance [254]. Although herringbone tubes are similar in concept to micro-fin tubes, the herringbone pattern can provide a higher heat transfer enhancement. However, compared with micro-fin tubes, herringbone tubes result in much larger pressure drops and could be relatively more complex to manufacture. Similar to dimpled tubes, herringbone tubes incur fewer maintenance costs as there is no need to replace tapes or inserts except when the fouling of the tube itself occurs [436]. Finally, the risk of tube blockage is minimal. The last set of enhancement techniques reviewed are active techniques with a focus on EHD. Active techniques, in general, pertain to advantageous characteristics absent in all other passive techniques mentioned above. Active techniques can provide a much larger heat transfer augmentation and provide the ability to control the amount of heat transfer enhancement. This ability is granted by varying the output supply to the desired amount.

Active techniques are seldomly adopted in the industry due to how difficult it is to integrate them into pre-existing systems. Unlike passive techniques that just require a tube exchange, having an active technique such as EHD, would require an electric wire to run through the tube with a high potential difference. This may not always be suitable and poses an electric hazard during the operation. Furthermore, EHD only operates on dielectric fluids. This hinders the use of EHD in FMCGs since the fluid in the heat exchangers is the product being manufactured. Therefore, EHD is not suitable in applications where the fluid cannot be exchanged for another. Another point to

consider is the cost of operation. Passive techniques such as dimples, require only an initial investment to replace the tubes but do not incur any further costs during operation. However, for active techniques similar to EHD, the need for a constantly running power supply during operation translates into continuous expenses. Further, the precautions needed to ensure the system is electrically insulated incur further expenditures to set up the system.

From the aforementioned discussion, it can be noted that the most suitable technique for heat transfer enhancement is circumstantial. If the most substantial enhancement is sought, EHD would be most suitable, however, it is expensive and difficult to implement. As a consequence, active techniques are usually not adopted as much in the industry. If ease of implementation is the main concern, then changing the fluid behavior by using surfactants would prove to be one of the easiest and cheapest techniques. However, tampering with the liquid is not always permissible especially in FMCGs. A more balanced technique that is widely used in industries due to its aforementioned advantages and acceptable limitations is micro-fins [154]. That being said, the effectiveness of micro-fins is dependent on their shape, inclination, height, and so on. The optimal micro-fin geometry is also circumstantial and is application dependent. However, commercially available vendors exist to facilitate the decision-making process.

The previous discussion was limited to conventional techniques adopted to facilitate heat transfer enhancement. Recent developments, however, shifted towards additive manufacturing (AM) to cause heat transfer enhancement [437]. Utilizing the AM enhancement techniques reviewed in this chapter would not only enable more compact heat exchangers, but it also enables the fabrication of micro-heat exchangers that is needed in many electronic device applications [438]. Consequently, AM can overcome the manufacturing limitations of conventional manufacturing techniques. Furthermore, since AM relies on depositing layers one after the other

to manufacture components, this enables unprecedented flexibility in manufacturing complex geometries that would have been otherwise very challenging to implement using conventional manufacturing techniques [439]. Furthermore, this added flexibility enables the implementation of heat exchangers that are very customizable and situation specific. Moreover, AM does not require secondary processes such as machining to achieve required geometrical shapes. This cuts down production time and the cost of multiple manufacturing equipment [440]. Also, since AM is a layer-by-layer deposition technique, it can save on material because unlike conventional methods such as milling, it does not depend on removing material to achieve the desired geometry [440]. Another advantage AM possesses over other techniques, is its ability to use different materials, such as metals, plastics, or ceramics to manufacture a single part.

Even though AM revolutionized manufacturing and reflects recent advancements in the field, it has challenges that limit its adoption over conventional techniques. First, heat exchangers manufactured using AM are limited in size and must be essentially smaller than the 3D printer. For large-sized parts, large printers are needed, prompting a high upfront capital investment. Furthermore, surface finishing of AM parts usually needs to be followed up by post processes to obtain the desired tolerance [441,442]. Another challenge with AM processes is scaling [440]. For highly customizable heat exchangers with complex geometries, AM manufacturing is cheaper. However, limitations in production speed mean that for simple geometries AM would not be able to sustain large production rates and benefit from economies of scale that would have been realized otherwise by conventional manufacturing techniques

Table 9. Summary of flow boiling enhancement techniques.

No.	Passive/ Active	Type of enhancement	Author	Fluid	Results*
1	Passive	Surfactant	Qiu <i>et al.</i> [280]	Octadecylamine	Increase in heat transfer enhancement
2			Hetsroni <i>et al.</i> [282]	Water and water-surfactant (Habon G)	Increase in heat transfer enhancement and pressure drop
3		Dimples	Shafae <i>et al.</i> [283]	R600a	HTC increased by 1.29–2 times. Pressure drop was higher by 7 to 103%
4			Zahid <i>et al.</i> [291]	R134a	With rid insert: HTC increased threefold. Pressure drop increased
5			Guo <i>et al.</i> [288]	R22, R410A, R32	Heat transfer enhancement increased by up to 1.2–1.4 times
6			Wei <i>et al.</i> [293]	R410A	HTC increased by 1.11–1.43 times
7		Micro-fin	Schlager <i>et al.</i> [296]	R22	HTC increased by 1.6–2.2 times. Pressure drop increased
8			Yu <i>et al.</i> [299]	R134a	Heat transfer enhanced up to 200% at low mass flux and higher quality
9			Zhnag <i>et al.</i> [297]	R417A	EF amounted to 1.26-2.8 and PF amounted to 1.8-2.6.
10			Schlager <i>et al.</i> [304]	R22	Heat transfer enhancement ranging from 50 to 100% The maximum increase in pressure drop was 40%.
11			Oh and Bergles [305]	R134a	Optimal helix angle for maximum HTC depended on mass flux conditions.
12			Kimura and Ito [306]	R12	Spiral-grooved tube can increase HTC from 1.5-2 times for annular flow, and approximately 10 times for stratified flow
13			Yang and Hrnjak [307]	R410A	Annular flow occurred at lower mass flux and vapor qualities as the helix angle increased.

Table 9. Summary of flow boiling enhancement techniques. (Continue)

No.	Passive/ Active	Type of enhancement	Author	Fluid	Results*
14	Passive	Micro-fin	Chiang [308]	R22	Heat transfer augmentation in the axial micro-finned tube was larger than the helical micro-fin tube
15			Wu <i>et al.</i> [309]	R32	Increase the heat transfer by 60% and pressure drop by 40%
16			Yang and Hrnjak [310]	R410A	Increase the heat transfer by 1.34 times and pressure drop by 1.23 times
17			Jiang <i>et al.</i> [298]	R22, R134a, R407C, and R410A	Average HTC of R134a, R22, R410A, and R407C are 1.80, 1.86, 1.78, and 1.69 times higher. Average pressure drop of R134a, R22, R410A, and R407C are 1.30, 1.42, 1.40, and 1.45 times higher, respectively
18			Diani <i>et al.</i> [300]	R1234ze(E)	HTC increased by 2.4 times. Pressure drop increased by 1.4 times
19			Filho and Barbieri [301]	R134a	HTC and pressure drop increased
20			Kim and Shin [302]	R22 and R410a	HTC increased 1.64–2.99 and 1.86–3.27 times for R410a and R22, respectively
21			Dang <i>et al.</i> [303]	Carbon dioxide	HTC increased by 1.9–2.3 times. Pressure drop increased by 1.5–2.1 times
22		Herringbone tube	Guo <i>et al.</i> [288]	R22, R410A, and R32	No improvement
23			Olivier <i>et al.</i> [313]	R22, R407C, and R134a	Heat transfer enhancement 70% higher. Pressure drop 80% higher
24			Longo <i>et al.</i> [314]	R22	HTC increased by 1.3–1.4 times
25			Filho and Jabardo [315]	R134a	Herringbone has superior thermal performance and pressure drop compared with micro-fin tubes

Table 9. Summary of flow boiling enhancement techniques. (Continue)

No.	Passive/ Active	Type of enhancement	Author	Fluid	Results*
26	Passive	Twisted tape	Sarmadian <i>et al.</i> [316]	R600a	HTC and pressure drop increased. The PFE values ranged between 0.44-1.09
27			Kanizawa <i>et al.</i> [317]	R134a	HTC improved to a maximum of 45%. Pressure drop increased by a maximum of 30 times
28			Shishkin <i>et al.</i> [318]	R134a	Tapes induced an annular flow regime earlier. Mass and heat fluxes higher than 400 kg/m ² s and 100 kW/m ² , respectively did not affect HTC.
29			Yan <i>et al.</i> [319]	Water	Increased CHF significantly under high mass fluxes
30		Wired coil	Behabadi <i>et al.</i> [320]	R134a	HTC improved to a maximum of 98%. Pressure drop increased by a maximum of 1000%
31			Niezgoda-Żelasko [322]	R507, R410, and R407C	HTC for azeotropic fluids increased by 1.1-1.7 times, while the HTC for zeotropic agents increased by 1.1-1.3 times. Pressure drop increased by 1.8-4.5 times
32			Alimardani <i>et al.</i> [323]	R600a	Pressure drop increased by 90-958%. The PFE values ranged between 0.13-1.4
33			Yun <i>et al.</i> [324]	Nitrogen	HTC and pressure drop increased, with PFE values >1
34	Active	EHD	Cotton <i>et al.</i> [335]	HFC-134a	Average HTC increased threefold in annular channels while pressure drop increased 2.6 times
35			Cotton <i>et al.</i> [336]	HFC-134a	Heat transfer was enhanced 1.6 times while the pressure drop increased 1.2 times
36			Salehi <i>et al.</i> [337]	R134a	Boiling HTC increased tenfold at 3 kV
37			Liu <i>et al.</i> [338]	R123	Maximum enhancement factor amounted to 20.1 for an average electric field strength of 1333 kV/m in vertical tubes

*Compared with smooth tubes with no modifications unless stated otherwise.

Table 10. Summary of condensation heat transfer enhancement techniques.

No.	Passive/Active	Type of enhancement	Author	Fluid	Results*
1	Passive	Twisted tape	Hejazi <i>et al.</i> [343]	HFC-134a	Increase in HTC of 40% in best cases. Pressure drop increase of 2.4 times in worst cases
2			Akhavan-Behabadi <i>et al.</i> [344]	R134a	Heat transfer coefficients increase up to 38% at low mass fluxes
3			Agrawal <i>et al.</i> [345]	R22	Increased heat transfer by up to 25% for full-length twisted tape insert
4		Wired coil	Agrawal <i>et al.</i> [346]	R22	HTC increase of up to 100%
5			Akhavan-Behabadi <i>et al.</i> [349]	R134a	Pressure drop increase of 260–1600%
6			Salimpour and Gholami [350]	R404A	Pressure drop increase of 1300%
7		Corrugated tube	Laohalertdecha and Wongwises [351]	R134a	HTC increase of up to 50%. Frictional pressure drop increase of up to 70%
8			Khoeini <i>et al.</i> [352]	R134a	HTC increase of 1.27–1.41 times at an inclination of 30° compared with an inclination of –90°
9			Fernández-Seara and Uhía [353]	NH ₃	Heat transfer performance of corrugated tubes on the outside was 1.27 times higher
10		Microfin tube	Jung <i>et al.</i> [354]	R22, R134a, R407C, and R410A	HTC increase of twofold to threefold
11			Kwon <i>et al.</i> [355]	R22 and R410A	Enhancement ratio between 1.5 - 2.2 based on the nominal heat transfer area
12			Eckels and Pate [356]	R134a and CFC-12	For HFC-134a, HTC increase of 1.75–2.5 times. For CFC-12, HTC increases of 1.7–2.3 times. Pressure drop increase

Table 10 Summary of condensation heat transfer enhancement techniques (Continue)

No.	Passive/Active	Type of enhancement	Author	Fluid	Results*
13	Passive	Microfin tube	Naulboonrueng <i>et al.</i> [357]	HFC-134a	HTC increase of 1.1–1.85 times.
14			Naulboonrueng and Wongwises [358]	HFC-134a	Pressure drop increase of 10–50%
15			Eckels <i>et al.</i> [360]	R134a with ester lubricant	Heat transfer increase of 50–200%. Pressure drop increase of 10–50%
16			Cho and Tae [361]	R22 and R407C with oil	Before bend: Oil concentrations decreased heat transfer augmentation At bend: Maximum HTC After bend: Higher HTC than before bend
17			Graham <i>et al.</i> [359]	R135a	For mass fluxes around 150 kg/m ² s, axially grooved tubes provide better thermal performance than helically grooved or smooth tubes. Pressure drop for axially grooved tube and helically grooved tube are similar and both are higher than for smooth tubes
18		Herringbone groove tube	Miyara <i>et al.</i> [362]	R410A and R22	Thermal performance is superior to that of helical microfin tubes at high velocities and vice versa for low velocities. Pressure drop of herringbone tube is greater than that of microfin tube
19			Miyara <i>et al.</i> [363]	R410A	Longer fins increase heat transfer up to a height of 0.18 mm. HTC increases with helical angle. Pressure drop increase with height and helix angle
20			Goto <i>et al.</i> [364]	R410A and HCFC22	Herringbone groove provides twice the increase in HTC compared with the spiraled groove

Table 10 Summary of condensation heat transfer enhancement techniques (Continue)

No.	Passive/Active	Type of enhancement	Author	Fluid	Results*
21	Passive	Herringbone groove tube	Goto <i>et al.</i> [365]	R410A and R22	Herringbone groove provides twice the increase in HTC compared with the spiraled groove
22	Active	EHD	Gidwani <i>et al.</i> [366]	R134a, R407c and R404a	For R404a at 18 kV: Heat transfer was enhanced 18.8 times for smooth tubes and 5.8 times for corrugated tubes. Pressure drop increased 11.8 times for smooth tubes. For R134a at 18 kV: Heat transfer increased 8.3 times for corrugated tubes. Pressure drop increased 20.8 times for corrugated tubes. For R407c at 18 kV: Heat transfer enhancement increased 3.9 times for smooth tubes and 2.9 times for corrugated tubes
23			Choi [367]	Freon-113	HTC increase of 100% at 30 kV downstream in vertical tubes
24			Didkovsky and Bologa [368]	Water	Heat transfer increase of 20 times compared with tubes with no EHD in vertical tubes

*Compared with smooth tubes with no modifications unless stated otherwise.

Table 11. Efficiency correlations for flow boiling.

Authors	Correlation	Nomenclature
Shafee et al. [283]	$PFS = \frac{h_a}{h_p} \cdot \frac{A_p}{A_a}$	h_a : average heat transfer coefficient in augmented tubes h_p : average heat transfer coefficient in plain tubes A_p : total inner surface area of the plain tube A_a : total inner surface area of the augmented tubes
	$PFE = \frac{\left(\frac{h_a}{h_p}\right)}{\left(\frac{\Delta P_a}{\Delta P_p}\right)}$	h_a : average heat transfer coefficient in augmented tubes h_p : average heat transfer coefficient in plain tubes ΔP_a : Pressure drop in augmented tubes ΔP_p : Pressure drop in the plain tube
Promvonge [166]	$PEC = \left(\frac{Nu_a}{Nu_0}\right) \left(\frac{f_a}{f_0}\right)^{-1/3}$	η : Thermal Performance factor Nu_a : Nusselt number augmented Nu_0 : Nusselt number smooth f_a : Friction factor augmented f_0 : Friction factor smooth
Schlager et al. [296]	$EF = \frac{(hA)_{Enhanced\ tube}}{(hA)_{Smooth\ tube}}$	h : average heat transfer coefficient A : total inner surface area
	$PF = \frac{(\Delta P)_{Enhanced\ tube}}{(\Delta P)_{Smooth\ tube}}$	ΔP : Change in pressure

A dire need to begin to implement the reviewed enhancements in next-generation heat exchangers exists. Global energy consumption has steadily grown throughout the past century (Fig. 51) [443]. With 63% of the global generated thermal energy being lost across different temperature spectrums [444] as waste heat during heat transfer and conversion processes [445], it becomes essential to develop highly efficient systems that can capture the majority of the consumed energy resources. Since the majority of heat transfer processes utilizes a heat exchanger, an effort is needed to maximize their efficiency. The main approach to elevating the heat exchanger efficiency is through enhancing the HTC. Having a higher HTC increases the maximum heat flux that can be transmitted and reduces energy lost during conversion [446]. Moreover, being able to transfer a larger amount of heat per unit area allows for more efficient heat exchange and so longer tubes can be substituted for smaller ones while achieving the same goal.

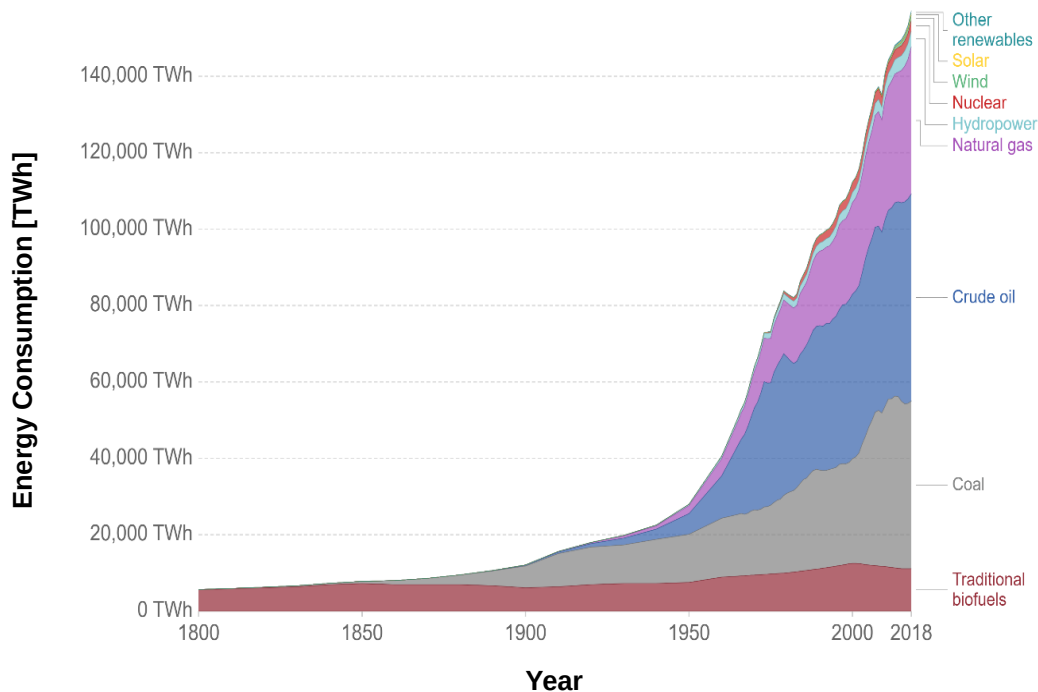


Figure 51. Global energy consumption (1800 to 2018)

Since more compact heat exchangers are enabled through the proposed techniques, the ability to decrease material consumption, which is projected to further increase as shown in Fig. 52, is facilitated. Decreased material consumption in turn aids in the development of more sustainable systems and industries. While it is relatively easy to develop new systems with the appropriate enhancement techniques, the greatest concern of adopting two-phase flow enhancement techniques is the ease of integration for already existing heat exchangers which cannot be retrofitted. As noted in the previous discussions and Tables 9 and 10, a wide array of cost-effective solutions exist which do not require major overhaul or disruption to already established processes. For example, dimples and twisted tapes can be implemented with minimal integration penalty when compared to other enhancements such as micro-fin tubes. The cost of the equipment required for integrating such enhancements is also another major challenge. A typical heat exchanger manufacturer would require major economic benefits before the infrastructure can be acquired to replace conventional technology. Consequently, compliance is always a major concern. For example, for tube and fin heat exchangers, replacing plain tubes with micro-fin tubes will require appropriate expansion technology which does not damage the fins during an expansion (pneumatic vs. bullet and rod expansion). Furthermore, durability and maintenance requirements are critical. For example, plain tubes have less potential for fouling compared to internally enhanced tubes.

Advanced manufacturing approaches such as additive manufacturing can bring remarkable performance improvements by invoking solutions that are not possible with conventional manufacturing approaches at both the design and material levels. However, for such processes, evaluation of materials is mandatory to meet specific thermal and mechanical properties. The manufacturing process has a major impact on these parameters and as such can influence the design of the device. For such processes, the design, material selection, and manufacturing are highly

interrelated and unlike conventional approaches, an additional cost is associated with such interrelation. Nonetheless, with the continued growth of greenhouse gas emissions worldwide [447] and the expansion of heavy industries to meet the demand of developing nations, building new heat exchangers is inevitable. Consequently, it is worthwhile to ensure that new thermal systems are being built with maximal efficiency by taking into consideration all the available enhancement techniques. This would not only aid in providing economic growth for industries but also aid in the responsible consumption of resources and mitigation of global warming for future generations.

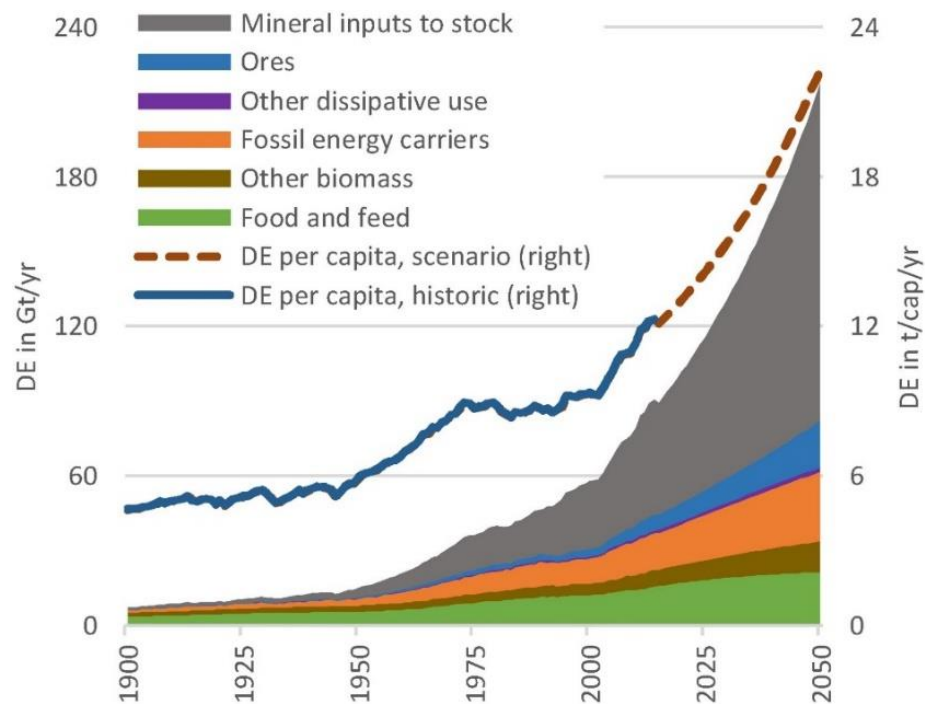


Figure 52. Extraction of materials (DE) in the US

Extraction of materials (DE) in the US from the years 1900 to 2050 (projected) [448]. Reprinted from Glob Environ Chang, 52, Krausmann et al., From resource extraction to outflows of wastes and emissions: The socioeconomic metabolism of the global economy, 1900–2015., 131–40, 2018, with permission from Elsevier.

6.2.7 Conclusion

This subsection reviews existing and emerging passive and active techniques to augment two-phase heat transfer. Passive techniques such as tapes, fins, and dimples are covered, demonstrating their benefit in enhancing heat transfer. Further, the use of electric fields (EHD) to augment heat transfer is discussed. Although EHD can be somewhat more involved in terms of cost and safety considerations, it provides the most capacity to increase the HTC. The EHD technique also provides control over the heat transfer by controlling the applied voltage and extent of HTC enhancement. However, whether active or passive enhancement is considered, a common trait for all techniques is the resulting additional pressure drop and consequent pumping power. Therefore, it is important that the savings from the use of smaller heat exchangers, enabled by enhancements, outweigh the added costs incurred from having to use a larger pump and more energy to operate the system. To capture this trade-off, several enhancement factors are presented to compare techniques. The value of these factors depends on the extent of the enhancement in heat transfer which, in turn, depends on several parameters. For example, for passive techniques such as coil inserts, the diameter, feature pitch ratio, and mass flux govern the heat transfer augmentation. For active techniques such as EHD, the position of the electric wire and the polarity impacts the rate of heat transfer. Furthermore, the flow regime and the fluid affect the heat transfer enhancement. For example, the use of surfactants to change the surface tension of the liquid facilitates evaporation. Although many techniques are reviewed for both condensation and boiling/evaporating flows, this work is not exhaustive, as the field continually develops.

Chapter 7. Investigation of Microstructured Surfaces on the Thermal Performance of Metallic Tubes

7.1 Introduction

Heat exchangers are widely used in different industries due to their ability to transfer large amounts of heat and regulate system temperatures effectively [234]. One of the major applications for heat exchangers is in heating, ventilation, air conditioning, and refrigeration (HVAC&R) systems. It is estimated that well over a billion air conditioning units have been already installed across the globe, with another 3.3 billion units to be installed by 2050 [449]. Given that air conditioning units are approximately 15% efficient, it becomes vital to improve their efficiency. In addition, by increasing heat exchanger efficiency, development of more compact heat exchangers is possible. Greater compactness results in lower material consumption and carbon footprint over the lifecycle of the system [103]. Other applications for heat exchangers are prevalent in powerplants where steam is utilized to run electric generators and produce power. Generally, water is crucial to powerplant operation as it possesses desirable characteristics such as high latent heat, wide availability, and low cost that make it the ideal fluid to extract heat from fossil fuels or other renewable sources of energy. In recent reports by EPA, it was noted that powerplant efficiencies in the United States peak at 33% due to heat losses to surroundings. As such, enhanced heat exchangers allow for more efficient extraction of heat enabling higher powerplant efficiencies.

To increase heat exchanger efficiency, several heat transfer augmentation techniques have been developed. Generally, these techniques increase the heat transfer coefficient (HTC) allowing for more effective heat transfer between the surface of the heat exchanger and the working fluid [234,449]. However, accompanying this heat transfer augmentation is an increase in pressure drop that mandates the use of larger, more costly, pumps to operate the system. Consequently, it is

necessary during thermal design to maintain a holistic view of how the augmentation technique affects both the heat transfer and pressure drop, to ensure that the savings attained from greater heat transfer efficiency are not overshadowed by capital or operating cost to maintain those enhancements [283,291,296].

Generally, heat transfer augmentation techniques can be categorized based on whether power input is required or not. Techniques that require power input such as electrohydrodynamics [335–337], are categorized as active and can generate high heat transfer augmentation. Additionally, since the heat transfer augmentation correlates with the power input, active techniques have the advantage of being able to control the extent of heat transfer augmentation. However, given the difficulty of implementation and relatively high costs of active techniques, adopted in industry has not been achieved. Instead, passive techniques that do not require power input are widespread. Examples of traditional passive techniques include the deployment of inserts [343,346], , dimples [288,291], extended surfaces [300,315], ribs [175,178], and vortex generators [187,450].

Passive techniques rely on geometric modifications to disrupt the fluid flow, increase the contact area between the surface and working fluid, and create additional nucleation sites. The effectiveness of these techniques is dependent on the flow regime, heat, and mass flux ranges. For example, twisted tape inserts are more effective at lower mass fluxes [145] as the increased disruption they induce due to swirl flow is more noticeable in the lower Reynold number spectrum. The effects of heat flux and liquid flow regime on heat transfer augmentation manifest themselves prominently in tubes with extended surfaces. Having exceedingly high heat fluxes can cause decreased heat transfer augmentation in finned tubes as it can promote film flow boiling at the root of the fins [306]. Moreover, the extent of heat transfer augmentation at a certain spiral angle is

fully governed by the flow regime present within the heat exchanger [304]. As a result, while designing enhanced heat exchangers, prior knowledge of the system operating conditions is necessary to fully maximize the potential of passive technique implementation.

While there are several passive techniques available in literature, micro-fins have been the most widely adopted in industry. However, to incorporate fins, tubes must be drawn, or internally grooved. Typically, these manufacturing techniques would limit fins to soft metallic tubes such as aluminum in order to avoid the wear of tools or allow drawing. While other manufacturing methods such as welding is possible for tubes of larger diameters, it is costly to undertake. In consequence, as with all other passive techniques, additional manufacturing techniques is necessary to incorporate geometrical enhancing features. This would require the overhaul of existing heat exchangers, as well as incur additional expenditures, and use of more material. To overcome these drawbacks, we present in this chapter a novel technique to incorporate microstructures to inner metallic tube walls, namely copper and aluminum. Our proposed technique has the advantage of scalability as well as ease of incorporation allowing for economies of scale and is applicable to wide array of metallic tube material.

7.2 Experimental Setup

To test the effect of microstructures on the heat transfer rate of internally structured metallic tubes, the setup shown in Fig. 53a was developed and used. The flow loop consists of a tank containing deionized (DI) water exposed to atmospheric temperature and pressure conditions. Immersed in the tank is an electric heater (Isotemp 2150, Fisher Scientific) to control and regulate the temperature of the water before it enters the pump. A 0.75 horsepower seal-less magnetic drive centrifugal pump (FTI DB9P-M202, Finish Thompson) is used conjunction with a system of

valves to control the flow rate flowing through the loop. An inline needle valve (SS-12NRS16 , Swagelok) located after the pump is used for fine control of the flow rate while a on/off valve (V591P-8-04, Parker) present on the bypass line is used for coarse control of the flow rate. In order to measure the flow rate, a Coriolis flow meter (CMFS015M, Emerson) is connected after the air vent. After the flow meter, a preheater (FLC-9, Watlow) rated at 3 kW is used to raise the temperature of water and preheat it to the desired temperature conditions before it enters the test section.

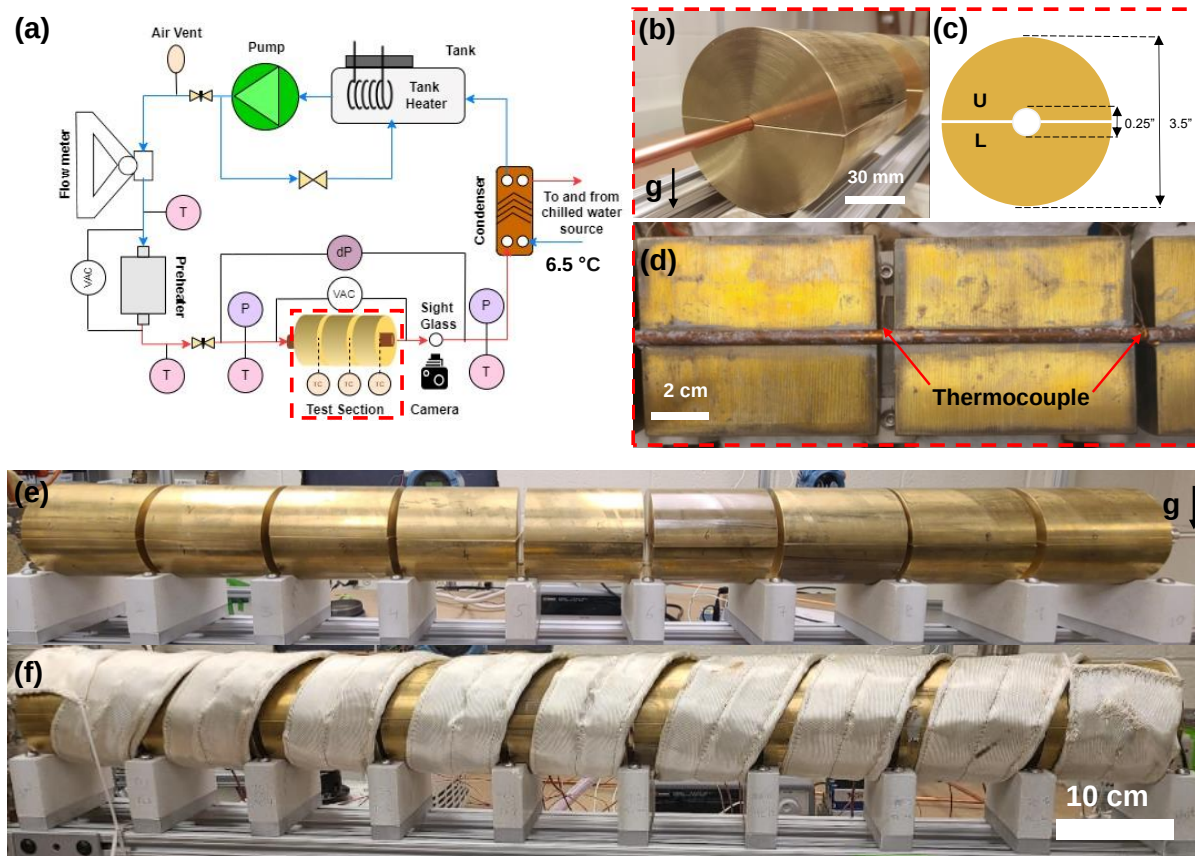


Figure 53. Test setup details

(a) Schematic of the test setup showing the location of resistance temperature detectors (RTD, T), thermocouples (TC), absolute pressure sensors (P), the differential pressure transducer (dP) and AC power supply (VAC). Schematic not to scale. (b) Isometric photograph of the horizontal brass block assembly without heaters or insulation wrapped. (c) Cross sectional schematic of the brass block assembly showing the upper (U) and lower (L) half sections. Schematic not to scale. (d) Photograph of the brass block assembly with the upper half (U) removed showing the tube sample (orange tube), thermal interface material (gray material), thermocouples welded to the sample, and brass blocks (yellow material). Photographs of the assembled heating section showing the glass-mica ceramic stands supporting brass blocks with (e) no heaters wrapped, and (f) heaters wrapped.

The test section tubing investigated consists of a 0.25 inch metallic tube having a length of 0.955 meters. Due to the large heat latent heat of vaporization of water, a high energy density fiberglass rope heater (SWH352-100LSE, Omega) is used to provide heat up to 3.135 kW. However, because the rope heater has a length spanning 10 feet and a width of 3.25 inches, it is necessary to increase the lateral surface area of the tube in order to avoid requiring an extended test section length. To achieve this function, the test section tube is enclosed within a brass block (360 Brass, McMaster-Carr) as shown in Fig 53b. Each brass block consists of two C-shaped parts with an outer diameter of 3.5 inches, inner diameter of 0.25 inches, and length of 4 inches. Further, for enhanced thermal contact between the brass block and the tube, thermal paste (OMEGATHERM OT-201-2, Omega) was applied between the tube samples and the brass blocks.

In total, nine brass blocks are used to cover the entirety of the test section. Small spacings (≈ 5 mm) are left in between every brass block such that two K-type thermocouples (TT-K-36-36, Omega) can be attached to the metallic tube as shown in Fig 53d. The thermocouple beads are also covered with thermal grease (OMEGATHERM OT-201-2, Omega) and are attached to the tube using copper tape (McMaster-Carr). In total, 18 thermocouples are attached to the top and bottom of the metallic tube every 4 inches along the test section with the final two thermocouples being attached after the ninth brass block.

Each of the brass blocks rests on a low thermal conductivity ($k \approx 1.33$ W/(m·K)) glass-mica ceramic to minimize heat loss (Figs 53e,f). Given that ceramics are brittle under tension, an aluminum slab having thickness of 1 cm is added under each ceramic support to carry the weight of the brass blocks. To mitigate additional heat losses to the surroundings, the heater on the test section (Fig 53f) is covered with fiberglass wool insulation (ECOBATT B24EV , Knauf). After the test section, a 0.25 inch diameter glass tube (sight glass, Fig 53a) is installed to provide optical

access to see the flow and capture it using a high-speed camera (Photron FASTCAM Mini AX 200, Vision Research or Phantom VEO 640L, Photron). Image capturing rates were set to 2500 frames per second.

The temperatures at the inlet and outlet of the test sections are measured using platinum RTDs (Reotemp Pt 385, 4 wire, Class AA). The absolute pressure before and after the test section is also measured using an absolute piezoresistive pressure transducer (PX 419-500AI, Omega). To quantify the pressure drop across the test section with high fidelity a differential pressure transducer (3051CD, Emerson) is connected to the test section inlet and outlet. The loop is completed by cooling down the saturated fluid exiting the test section to room temperature using a compact brazed plate heat exchanger (B&G BP400-20 LP) before being returned to the tank as liquid. Using these temperature and pressure values, the thermophysical properties of the fluid can be obtained.

Prior to measurement, the thermocouples attached on the test section were calibrated in a temperature-controlled bath (AP07, Polyscience) that consist of a coil heater immersed in ethylene glycol fluid. A pre-calibrated and highly accurate temperature probe (5615 PRT, Fluke) as well as the test section thermocouples are immersed in the water bath that is set to predefined temperature values that match the testing conditions during flow boiling. The temperatures measured by the thermocouples as well as the probe are recorded for the range of temperature values and a curve is then constructed in order to accurately calibrate the thermocouples. Full details of the uncertainty and instrumentation, see Table 12.

Table 12. Instrumentation used in the flow boiling setup depicted in Fig. 53a.

Parameter	Instrument	Uncertainty	Span	Model
Mass Flow Rate	Coriolis flow meter	$\pm 0.1\%$	35 - 1600 [g/min]	Emerson CMFS015M
Temperature	RTD	$\pm 0.1 [^{\circ}\text{C}]$	0 - 260 [$^{\circ}\text{C}$]	Reotemp Pt 385, 4 wire
	Thermocouple*	$\pm 0.57 [^{\circ}\text{C}]$	0 - 260 [$^{\circ}\text{C}$]	Omega TT-K-36-36
Absolute Pressure	Piezoresistive Transducer	$\pm 0.8\%$	0 - 3447 [kPa]	Omega PX 419-500AI
Differential Pressure	Differential Pressure Transducer	$\pm 99.5 [\text{Pa}]$	0 - 50 [kPa]	Emerson 3051CD
Heat Input-Preheater	Coil heater	$\pm 10\%$	0-3.0 [kW]	Watlow FLC-9
Heat Input-Heater	Fiberglass rope heater	$\pm 16\%$	0-3.135 [kW]	Omega SWH352-100LSE

* See Appendix B on thermocouple bias error calculation

7.3 Data Reduction

To quantify the heat transfer coefficient across the test section, we first calculate the amount of heat lost to the surroundings. This can be calculated for the preheater using the following expression:

$$eff_{ph} = \frac{\dot{m}_{ref}(h_{out|ph} - h_{in|ph})}{(VI)_{ph}}, \quad (35)$$

where eff_{ph} is the insulation effectiveness of the preheater, $\dot{m}_{ref}$ is the mass flow rate of the refrigerant, $h_{out|ph}$ is the refrigerant enthalpy at the exit of the preheater, $h_{in|ph}$ is the refrigerant enthalpy at the inlet of the preheater, and $(VI)_{ph}$ is the product of the voltage (V) and current (I) supplied to the preheater. A similar expression is also derived for the test section:

$$eff_{ts} = \frac{\dot{m}_{ref}(h_{out|ts} - h_{in|ts})}{(VI)_{ts}}, \quad (36)$$

where eff_{ts} is the insulation effectiveness around the test section, $h_{out|ts}$ is the refrigerant enthalpy at the exit of the test section, $h_{in|ts}$ is the refrigerant enthalpy at the inlet of the test section, and $(VI)_{ts}$ is the product of the voltage (V) and current (I) supplied to the test section rope heater.

By applying an energy balance on the preheater, the amount of heat required to raise the temperature of the refrigerant to a desired value can be calculated as:

$$Q_{ph} = eff_{ph}(VI)_{ph} = C_{p,ref}\dot{m}_{ref}(T_{out|ph} - T_{in|ph}), \quad (37)$$

where Q_{ph} is the heat provided to the refrigerant in the preheater section, $C_{p,ref}$ is the specific heat capacity of the refrigerant, $T_{ref,in|ph}$, and $T_{ref,out|ph}$ is the temperature of the refrigerant entering and exiting the preheater respectively.

Since minimal heat is lost to the surroundings, the bulk fluid temperature of the refrigerant at the inlet of the test section, $T_{in|ts}$, is almost identical to the bulk fluid temperature of the refrigerant exiting the preheater:

$$T_{in|ts} = T_{out|ph}. \quad (38)$$

By using Eqns. (2) and (4) and conducting an energy balance on the test section, the vapor quality at the exit of the test section is calculated as:

$$Q_{ts} = eff_{ts}(VI)_{ts} = C_{p,ref}\dot{m}_{ref}(T_{out|ts} - T_{in|ts}) + \dot{m}_{ref}h_{fg|ts}x_{out|ts}, \quad (39)$$

where Q_{ts} is the heat transfer provided to the refrigerant in the test section, $h_{fg|ts}$ is the latent heat of vaporization of the refrigerant in the test section, $x_{out|ts}$ is the vapor quality at the outlet of the test section, and $T_{out|ts}$ is the bulk fluid temperature of the refrigerant exiting the test section.

To calculate the local heat transfer coefficient at a distance, x , from the inlet of the test section, we consider the local wall temperature measurement:

$$h_x = \left(\frac{T_{x,\text{wall}} - T_{\text{sat}}}{q_x} - \frac{\ln\left(\frac{d_o}{d_i}\right) d_i}{2k} \right)^{-1}, \quad (40)$$

where h_x , q_x , and $T_{x,\text{wall}}$ are the local heat transfer coefficient, local heat flux, and local wall temperature, respectively, at a distance x from the inlet of the test section. Here, k is the tube wall thermal conductivity, d_{in} is the tube inner diameter, d_{out} is the tube outer diameter, and T_{sat} is the saturation temperature.

Assuming homogenous flow, the frictional pressure drop in tubes is calculated using the following relation[451] :

$$\Delta P_{\text{friction}} = \Delta P_{\text{total}} - \Delta P_{\text{momentum}} - \Delta P_{\text{static}}, \quad (41)$$

where ΔP_{total} is the total pressure drop, $\Delta P_{\text{momentum}}$ is the momentum pressure drop, and ΔP_{static} is static pressure drop. Since there is no change in elevation between the inlet and outlet of the test section tube, the static pressure drop is neglected ($\Delta P_{\text{static}} = 0$). In order to calculate the momentum pressure drop the following correlation is used[283] :

$$\Delta P_{\text{momentum}} = G_{\text{ref}}^2 \left\{ \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{\text{out,ts}} - \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{\text{in,ts}} \right\}, \quad (42)$$

where ρ_f and ρ_g are the density of the refrigerant when in liquid state and gas state, respectively, G_{ref} is the refrigerant mass flux and α is the void fraction. The void fraction, α , can be calculated using the Steiner correlation [262]:

$$\alpha = \frac{x}{\rho_g} \left[(1 + 0.12(1-x)) \left(\frac{x}{\rho_g} + \frac{(1-x)}{\rho_f} \right) + \frac{1.18(1-x)[g\sigma(\rho_f - \rho_g)]^{0.25}}{G_{\text{ref}}\rho_f^{0.5}} \right]^{-1}, \quad (43)$$

where g is the gravitational acceleration constant, and σ is the liquid-vapor surface tension of the refrigerant. Since ΔP_{total} is measured experimentally using the differential pressure transducer, $\Delta P_{\text{friction}}$ can be calculated using Eqns. (7) to (9).

Since the several tube samples investigated have different pressure drops and heat transfer coefficients, the performance factor (PF) is used to compare the thermal performance and gauge whether incorporating micro-structures is potentially beneficial:

$$PF = \left(\frac{h_a}{h_p} \right) \left(\frac{\Delta P_p}{\Delta P_a} \right), \quad (44)$$

where h_a is the tube-averaged heat transfer coefficient of a micro-structured tube, h_p is the tube averaged heat transfer coefficient of a plain tube, ΔP_a is the total pressure drop of a micro-structured tube, and ΔP_p is the total pressure drop of a plain tube. The performance factor defines a relation which represents the apparent ratio of pumping power (ω) to the averaged heat transfer coefficient (h) for augmented tubes compared to that of plain tubes [283]. Here, the pumping power is defined as [452]:

$$\omega = \dot{v} \Delta P, \quad (45)$$

where $\dot{v}$ is the volumetric flow rate of the refrigerant.

It is worth noting, that even though the PF provides means of indicating whether an enhancement technique should be adopted, there are many other factors introduced in literature as alternative indicators [449]. While there has not been a thorough investigation to develop a factor that takes the economical aspect into consideration, the factor that is adopted is generally dependent on the industry and the interest of the study. While we take into account here the effect of pressure drop, there are many industrial applications that are only interested in the heat transfer coefficient augmentation regardless of the increase in pressure drop.

Since key parameters, such as the local heat transfer coefficient, are calculated from inputs of other primary parameters that possess measurement uncertainty, it is expected that the uncertainty of the primary parameters would propagate into the calculated parameters. To determine the extent of this error propagation, the following expression is used [453]:

$$Y_M = \sqrt{\sum_{i=1}^n \left(\frac{\partial M}{\partial Z_i} \right)^2 Y_{Z_i}^2}, \quad (46)$$

where Y denotes the uncertainty of a parameter, M is the calculated parameter, Z is the primary parameter, n is the number of primary parameters and i is the index of primary parameter.

7.4 Effect of Microstructures on Copper Tube Thermal Performance

7.4.1 Surface Fabrication

Four copper tubes were studied: plain, microstructured, finned, and finned with microstructure. The finned tubes consisted of commercially available enhancements manufactured using welding supplied by Vipertex. Dimensional characteristics of the tubes as well as other parameters are summarized in Table 13. For each tube, a range of mass fluxes and heat inputs ranging between 255 - 485 kg/(m²·s) and 0 – 2.4 kW, respectively, were studied. To manufacture the microstructures on the inner tube walls, an alkaline mixture of sodium chlorite (CAS #7758-19-2), sodium hydroxide (CAS #1310-73-2), sodium phosphate tribasic dodecahydrate (CAS #10101-89-0) with weight percentages of 3.75:5:5:10:100, respectively, is prepared and poured into a glass beaker. To ensure the copper tube is clean before immersion into the alkaline solution, it is thoroughly rinsed with ethanol (CAS # 64-17-5), isopropyl alcohol (IPA, CAS #67-63-0), and

DI water (CAS # 7732-18-5). The copper tube is then immersed into the alkaline mixture for 30 minutes at a temperature of $90 \pm 5^\circ\text{C}$. The oxidation reaction ensues is described by the following:

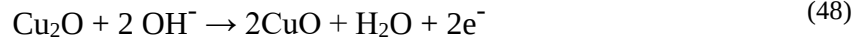
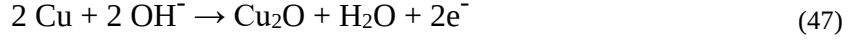


Table 13. Characterization of tubes prior to surface structuring.

Parameter	Value/Range
Smooth Tube	
Thermal conductivity (k_s)	395 [W/(m·K)]
Root-mean-square (RMS) roughness	25 [nm]
Tube Diameter (OD)	6.35 [mm]
Inner diameter - plain tube (d)	4.72 [mm]
Inner diameter - finned tube (d)	5.33 [mm]
Tube Length (L)	0.955 [m]
Enhanced Tube	
Thermal conductivity (k_s)	395 [W/(m·K)]
Root-mean-square (RMS) roughness	212 [nm]
Tube Diameter (OD)	6.35 [mm]
Inner diameter - plain tube (d)	4.72 [mm]
Inner diameter - finned tube (d)	5.33 [mm]
Tube Length (L)	0.955 [m]
Fin Width (FW)	0.3 [in]
Fin Depth (FD)	0.13 [in]
Helical angle (β)	20 °

The bare copper surface reacts with the hydroxyl group to form Cu_2O . Subsequently, Cu_2O reacts further with the hydroxyl group once more to form the desired copper oxide (CuO) product.

The resulting copper oxide structure emulate a thin knife-like form as illustrated by figure 54. The solid fraction, ϕ , and the roughness are ≈ 0.023 and 212 nm respectively.

After immersion, the tube was cleaned again DI water, isopropyl alcohol and dried with a clean nitrogen air stream.

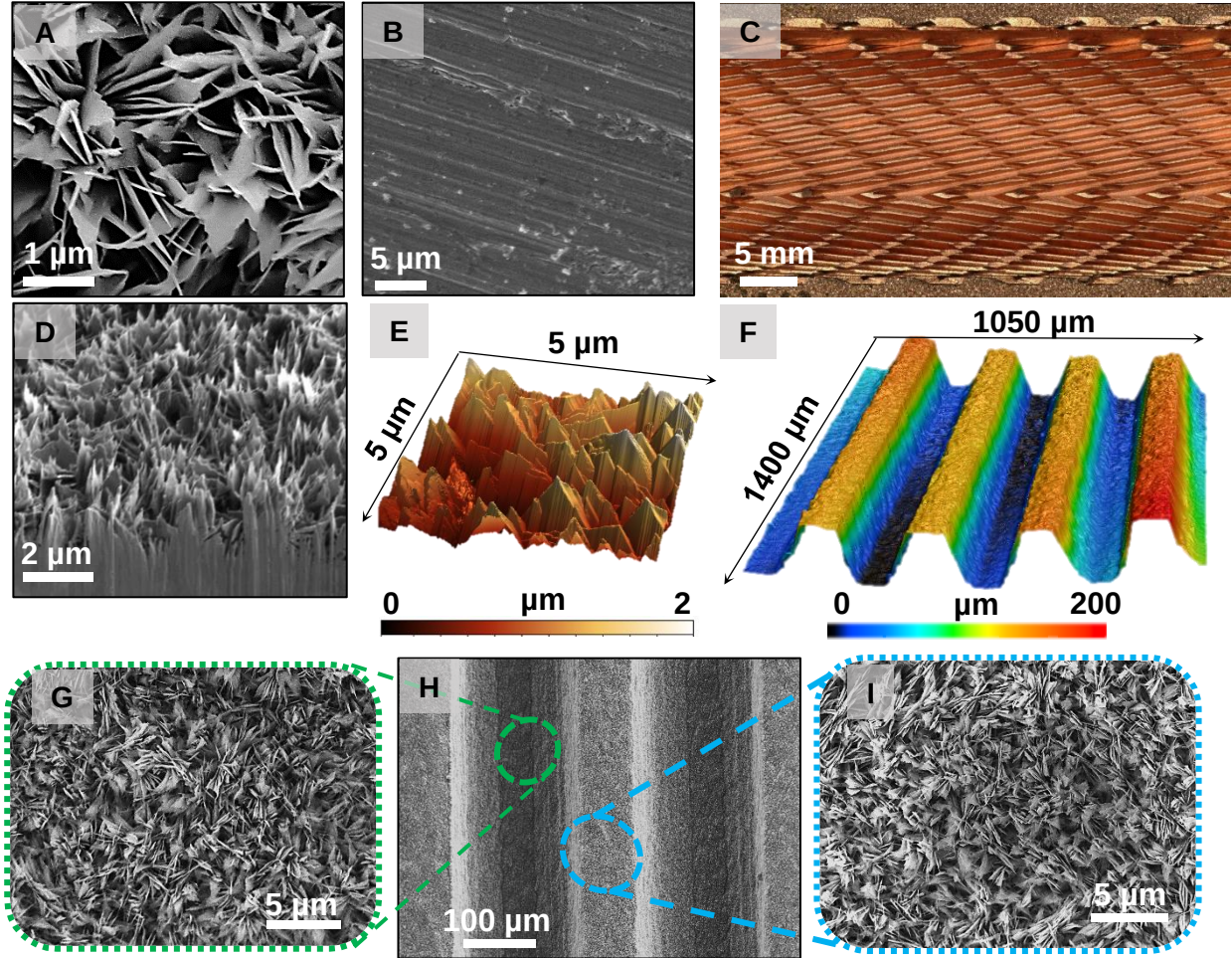


Figure 54. Microstructures formed on plain Cu tube

Top View scanning electron microscopy (SEM) image of (A) copper oxide (CuO) nanostructured surface and (B) plain copper surface inside the tube. (C) Confocal optical microscopy image showing finned enhancement inside the finned copper tube. (D) SEM image of the copper oxide nanostructure cross section prepared by focus ion beam (FIB) milling. (E) Atomic force microscopy (AFM) image showing the topography of the CuO nanostructured surface. (F) 3D laser scanning confocal microscopy image showing the interior topography of the finned enhanced copper tubes. (G) Zoomed in SEM view of the valley, (H) zoomed out SEM image of peak-valley-peak, and (I) zoomed in SEM view of the peak of the CuO nanostructured fins showing conformality of the CuO nanostructures.

7.4.2 Results and Discussion

Prior to conducting two-phase experiments, single phase validation of the flow loop was conducted using tap water. Single phase experiments were conducted for plain copper tubes over a range of

mass and heat flux as well as Reynolds number values ranging between $740 \text{ kg}/(\text{m}^2 \cdot \text{s}) < G < 900 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $42 \text{ kW}/\text{m}^2 < q'' < 105 \text{ kW}/\text{m}^2$, and $4500 < Re < 7300$ respectively. Figure 55 details the results of the heat transfer coefficient values obtained experimentally. To compare the measured quantities, we used the Gnielinski correlation [258]:

$$h_x = \frac{k_{ref} \left(\frac{f}{8} \right) (Re - 1000) Pr}{d_i \left(1 + 12.7 \left(\frac{f}{8} \right)^{\frac{1}{2}} (Pr^{2/3} - 1) \right)} \quad (49)$$

The results are within 20%, providing validation of the experimental setup [258]

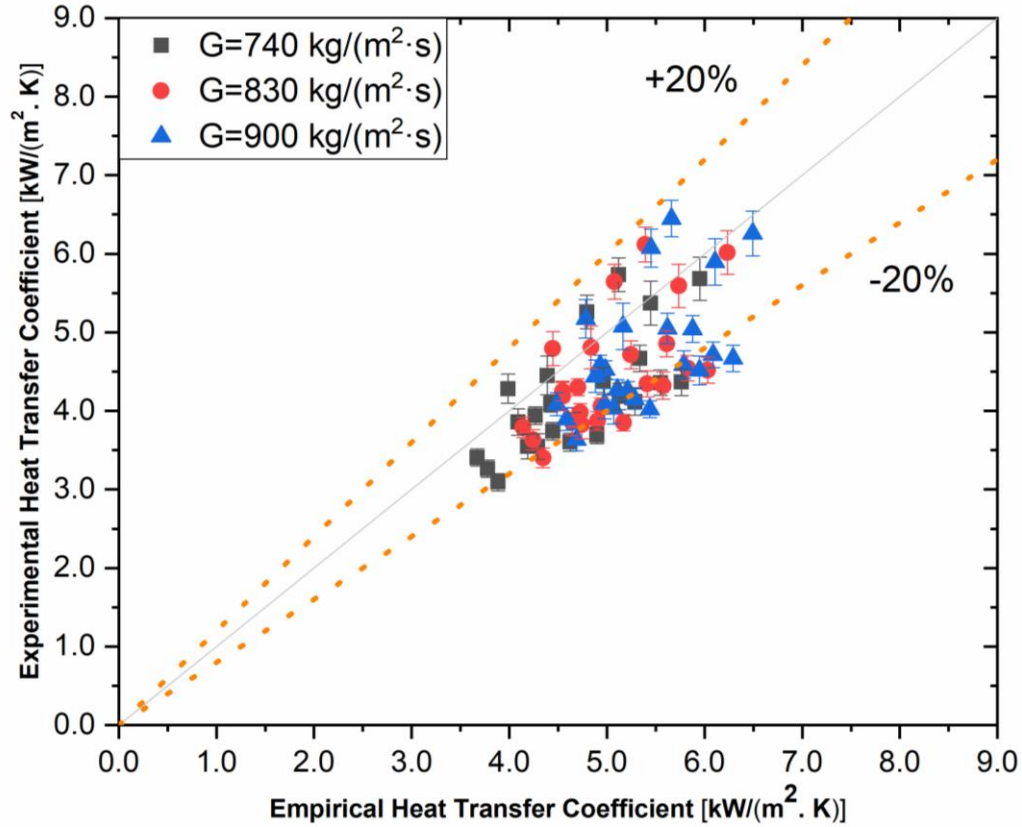


Figure 55. Experimental versus empirical heat transfer coefficients

Measured experimental heat transfer coefficient as a function of the empirical heat transfer coefficient correlation as predicted by Gnielinski [258].

7.4.2.1 Effect of Microstructures on Plain Copper Tubes

We began our two-phase experiments by investigating the effect of microstructures on the local and tube averaged heat transfer coefficient and pressure drop of plain copper tubes. To quantify the effect of adding microstructures to copper tubing, we compare the heat transfer coefficient of the microstructures tubes to plain copper tubes across different vapor qualities and mass fluxes as shown in Fig. 56. It is important to note that the vapor quality range varies for different mass flux values due to limitations on the maximum heat flux that can be provided by the test section heat and preheater. The vapor qualities shown in Fig. 56 represent the condition at the exit of the test section and is controlled by varying the heat provided by the test section heater.

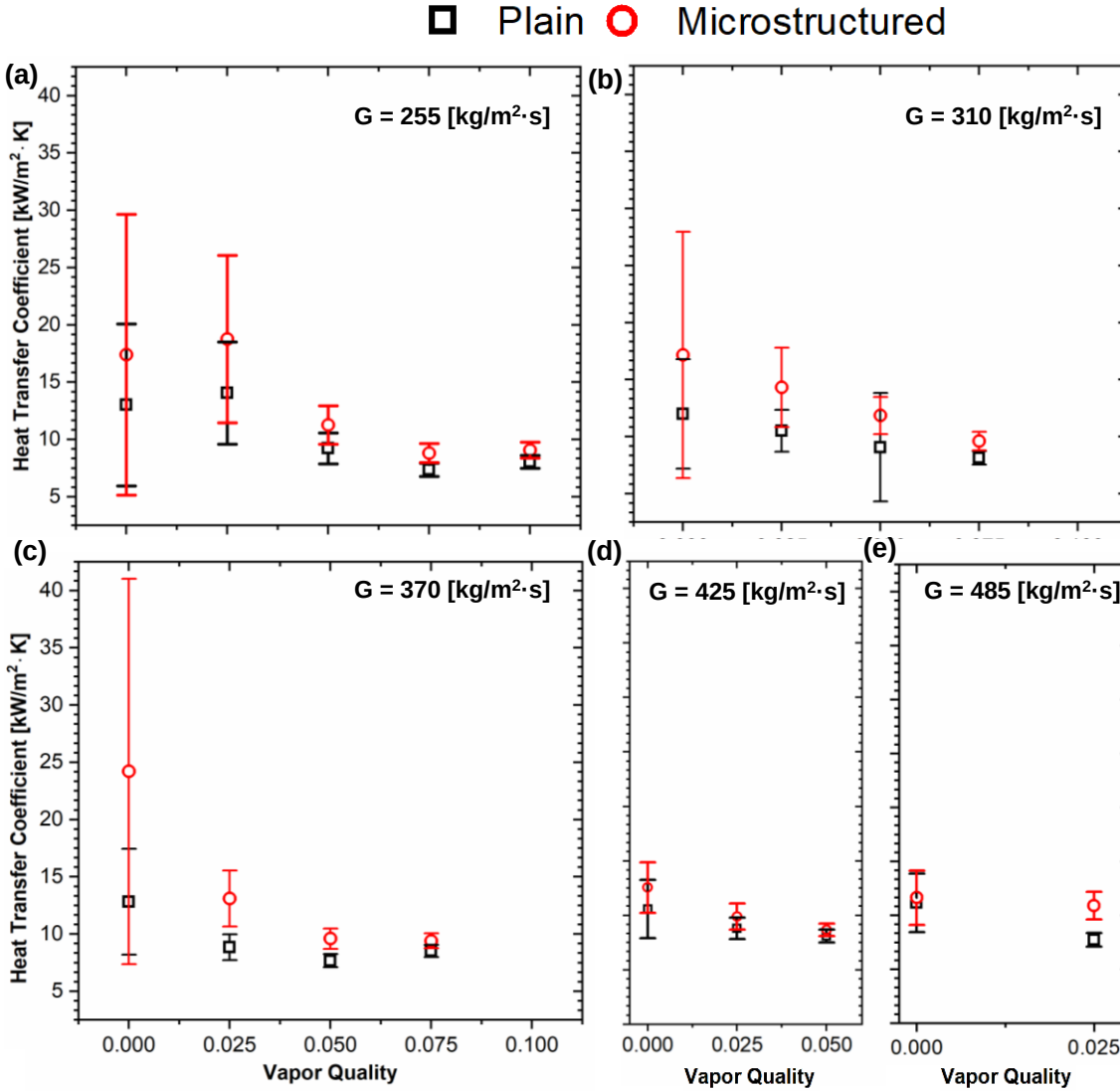


Figure 56. Water flow boiling heat transfer coefficient

Water flow boiling heat transfer coefficient as a function of exit vapor quality at mass fluxes of: (a) $G = 255$ kg/(m²·s) , (b) $G = 310$ kg/(m²·s) , (c) $G = 370$ kg/(m²·s) , (d) $G = 425$ kg/(m²·s), and (e) $G = 485$ kg/(m²·s).

From Fig. 56 we note that microstructured copper tube consistently augments the tube-averaged heat transfer coefficient by 10% to 90% across vapor qualities and mass fluxes when compared to plain copper tubes at the same test conditions. For mass fluxes of $G = 255$ kg/(m²·s), 310 kg/(m²·s), 370 kg/(m²·s), 430 kg/(m²·s), 480 kg/(m²·s), the heat transfer coefficient of the

microstructures tube increased by 13 to 35%, 18 to 44%, 10 to 90%, 7 to 19%, and 4 to 41%, respectively, when compared to the smooth tube.

The observed augmentation in heat transfer from the microstructures is attributed to the added nucleation sites. When fluid flows through the tube walls, these nucleation sites effectively trap air bubbles. Vaporization can then occur at lower wall superheats at the liquid-vapor interface of these trapped bubbles, augmenting heat transfer [451]. As vapor quality increases, the heat transfer coefficient decreases in both plain and augmented copper tubes due to the suppression of nucleation sites [283]. At vapor quality of $x = 0$, the nucleation sites are activated, and bubbly flow exists. However, as vapor quality increases and the void fraction inside the tube is higher, transition to plug and annular flow occurs where convective vaporization at the liquid vapor interface becomes more prominent. As the liquid film covering the inner tube wall becomes thinner, only smaller sized nucleation sites which require higher superheats become active [451]. Consequently, as the vapor quality increases the superheat required to maintain active nucleation sites is higher than that supplied by the rope heater, suppressing nucleate flow boiling. This is the case for both plain and augmented copper tubes, however, since the augmented copper tubes have more nucleation sites than plain copper tubes, the suppression of nucleate boiling is reflected more in the heat transfer coefficient in augmented copper tubes.

Following our experimental observations, it becomes apparent that the saturated flow boiling regime is nucleate flow boiling dominant [454]. This can be further cemented by investigating the relation between the average heat transfer coefficient and mass flux. In nucleate flow boiling dominant regime, the convective vaporization at the interface is not the main mode governing the heat transfer coefficient when compared to boiling due to the nucleation sites [455]. As such, the mass flux has negligible effect on the heat transfer coefficient as evident in Fig. 56.

As the vapor quality is increased, transition to the convective flow boiling dominant regime occurs. Here, the effect of mass flux on the heat transfer is clearly evident. Consequently, as the mass flux increases the effectiveness of vaporization at the liquid-vapor interface causing an increase in the heat transfer coefficient [451].

To study the effect of micro-structures on the pressure drop characteristics of plain copper tubes, Fig. 57 plots the frictional pressure drop as a function of sample exit quality over a mass flux range between $255 < G < 480 \text{ kg/(m}^2\cdot\text{s)}$ for both plain and microstructured copper tubes. A positive correlation between the pressure drop and vapor quality can be noted for both tubes across all mass fluxes. As the vapor quality increases, bubbles rising due to activated nucleation sites increase the fluid disturbance and intermixing causing an increase in pressure drop [451]. As the void fraction and vapor quality increase further, the added vapor causes an increased interaction with the surrounding liquid. Due to the higher vapor velocity relative to the liquid velocity, shear stresses increase as the vapor inside the tube grows leading to an increased pressure drop at elevated vapor qualities [283]. This phenomenon explains the increased pressure drop at higher vapor quality for both plain and microstructured copper tubes.

However, when comparing microstructured copper tubes with plain copper tubes, we note that the pressure drop exhibited in microstructured copper tubes is higher by 0.5% to 16%. For mass fluxes of $255 \text{ kg/(m}^2\cdot\text{s)}$, $310 \text{ kg/(m}^2\cdot\text{s)}$, $370 \text{ kg/(m}^2\cdot\text{s)}$, $430 \text{ kg/(m}^2\cdot\text{s)}$, $480 \text{ kg/(m}^2\cdot\text{s)}$, the pressure drop of the microstructured tube increased by 4 to 16%, 0.3 to 12%, 0.4 to 10%, 0 to 38%, and 1 to 2% , respectively, when compared to the smooth tube. The increased pressure drop is mainly due to the increase in skin friction stemming from the added roughness of the microstructures which, in turn, increases pressure drop. Further, due to added nucleation sites, the

bubble departure frequency for microstructured copper tubes at lower vapor qualities overshadows that of plain copper tubes, causing increased pressure drop across the test section.

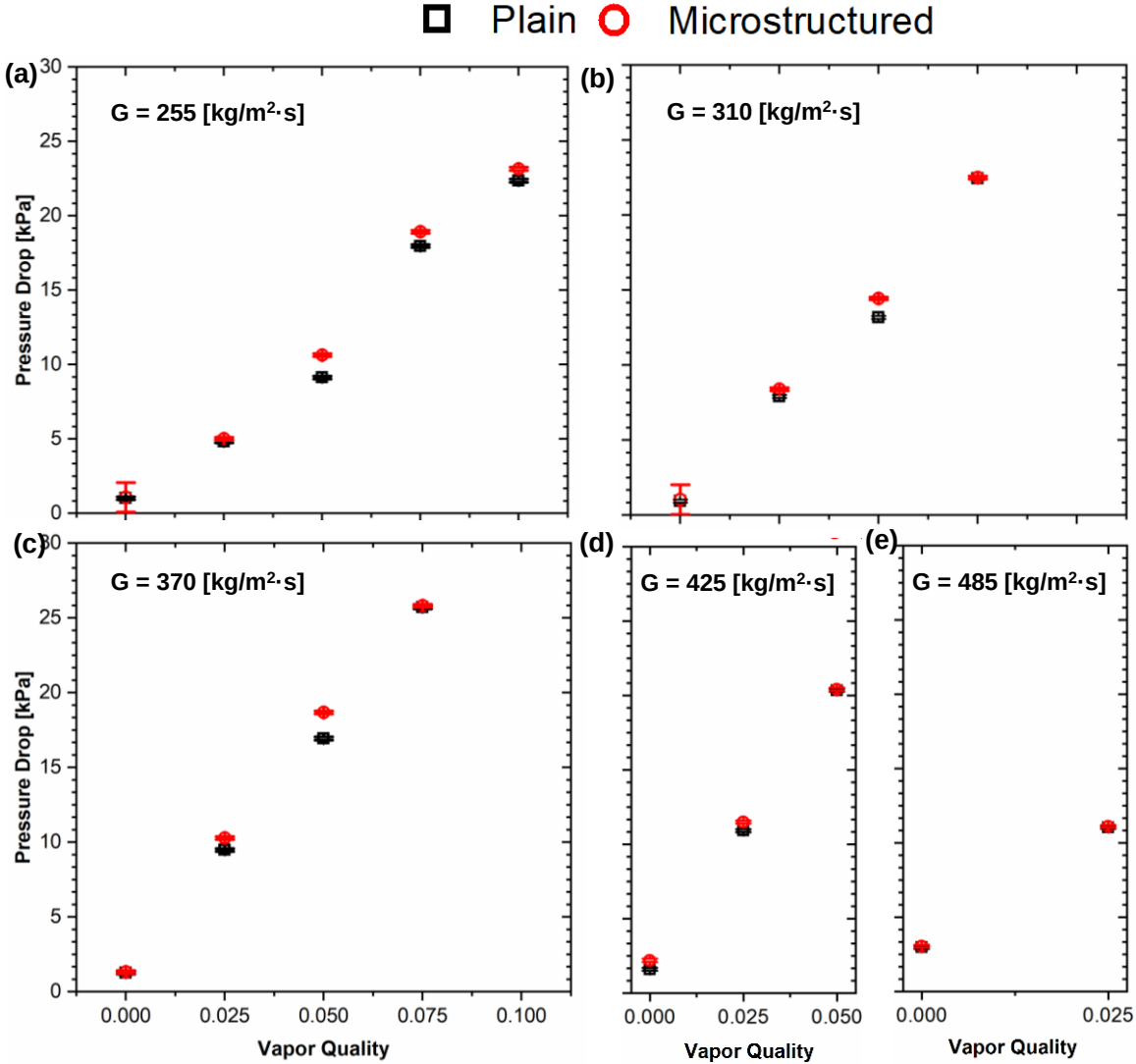


Figure 57. Flow boiling pressure drop

Flow boiling pressure drop as a function of exit vapor quality for mass fluxes of: (a) $G = 255 \text{ kg/(m}^2\cdot\text{s)}$, (b) $G = 310 \text{ kg/(m}^2\cdot\text{s)}$, (c) $G = 370 \text{ kg/(m}^2\cdot\text{s)}$, (d) $G = 425 \text{ kg/(m}^2\cdot\text{s)}$, and (e) $G = 485 \text{ kg/(m}^2\cdot\text{s)}$

Overall thermal-hydraulic performance of the plain and microstructured tubes

From the experimental results presented in the aforementioned discussion, it can be concluded that imparting microstructures to plain copper tubing increases the heat transfer coefficient due to the

presence of added nucleation sites. However, accompanying this augmentation in the rate of heat transfer is added fluid disturbance that increases pressure drop. The larger pressure drop requires larger pumps in order to operate the system at the required flow rate. In order to gauge whether the increase in the rate of heat transfer overshadows the pressure drop increase penalty, Eqns. (44) and (45) were used. Performance factors greater than unity (horizontal dotted line in Fig. 58) gives an indication that microstructuring can be viable to pursue, as the HTC increase is more than the pressure drop increase

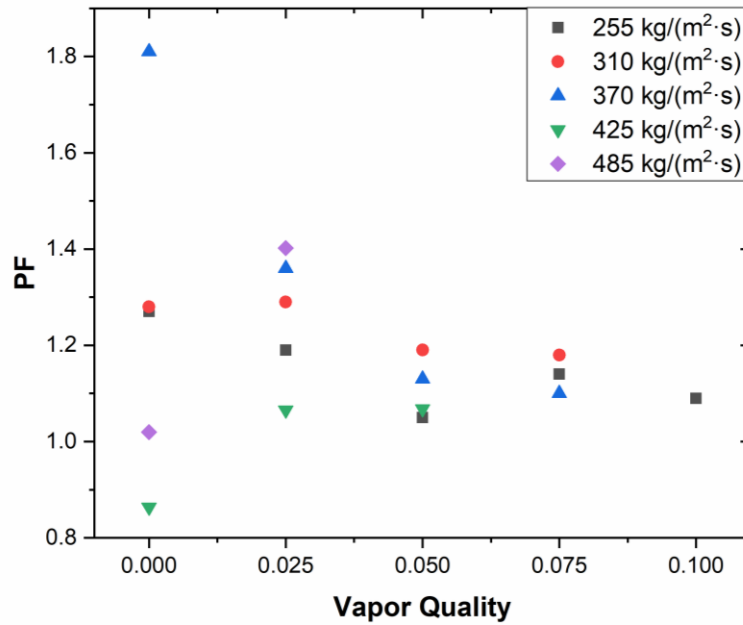


Figure 58. Performance factor (PF)

Performance factor (PF) of microstructures as a function of vapor quality (x) for a range of different mass fluxes.

Figure 58 illustrates the effect of the added microstructures on the overall thermal-hydraulic performance. As can be noted, having micro-structures causes the HTC increase to overshadow the pumping power increase across all mass flux and vapor qualities. Generally, it can be observed the PF decreased with vapor quality. Initially, all nucleation sites are active

contributing to bubble nucleation and causing heat transfer enhancement. However, as the vapor quality increases, suppression of nucleation sites starts to occur and the augmentation in the HTC start to diminish. Juxtaposed to this deterioration in heat transfer, is an increase in pressure drop. This is attributed to the higher vapor velocity (at larger vapor qualities) that contribute to larger shear stresses.

7.4.2.2 Effect of Microstructures on Finned Copper Tubes

In this subsection, we investigate the effect of micro-structures on the HTC and pressure Drop of finned copper tubes. Introducing fins to the inner walls of metallic tubes augment the rate of heat transfer between the flowing fluid and the tube due to increased fluid agitation, surface area, and introduction of secondary flows. However, unlike the case of plain copper tubes, adding micro-structures to a finned tube diminishes the rate of heat transfer. To quantify this effect, the heat transfer coefficient of micro-structured finned tubes and plain finned tubes are plotted against the outlet vapor quality as shown in Fig. 59. We note that in both tubes, the heat transfer coefficient is initially highest at vapor quality of 0 where bubbly flow regime manifests itself. However, as the vapor quality and the vapor void fraction inside the tube increases, the liquid film covering the inner tube wall becomes thinner which effectively deactivates large cavities present on the tube surface. As such, only smaller cavities which require larger superheats to remain active can contribute to the heterogenous nucleate flow boiling. Eventually, the liquid film inside the tube decreases in thickness (due to the continuously increasing void fraction) up to a threshold where nucleate boiling is suppressed. Further, as noted from Fig. 59, the heat transfer coefficient is unaffected by mass flux. Consequently, from the previous observations, it can be deduced that the flow boiling regime is nucleate dominant.

We note from Fig. 59 that micro-structured plain copper tube consistently diminishes the HTC by 14 – 75% across different vapor qualities and mass fluxes when compared to plain finned copper tubes for the same testing conditions. For mass fluxes of 255 kg/(m²·s), 310 kg/(m²·s), 370 kg/(m²·s), 425 kg/(m²·s), 485 kg/(m²·s), the HTC of the micro-structured tube decreased by 14-59%, 25-50%, 33-75%, 14-22%, and 24-59%, respectively. A possible explanation to this phenomenon could be attributed to partial film boiling. In micro-structured finned copper tubes, cavities that exist along the fin's crest and trough encourage the entrapment of air bubbles and act as nucleation sites during bubbly flow. However, due to the tube fin surfaces being in close proximity at the trough, coupled with the increase in the nucleate bubbles that form due to the added surface micro-structures, the probability of bubble coalescence near the tube surface increase. This flow rate of vapor away from the tube would eventually prevent the wetting of the inner tube walls at the trough causing patches of vapor to be in contact with the tube walls and effectively drying out portions of the surface. This would diminish the rate of heat transfer locally as the vapor patch has lower thermal conductivity and the surface area of the tube exposed to liquid is lessened. Further, the dry vapor patches effectively deactivate nucleation sites at the troughs. Since we note previously that the dominating mechanism for flow boiling is nucleate rather than convective, this further diminishes the heat transfer rate observed as only the cavities on the crest of the fin contributes to bubble nucleation.

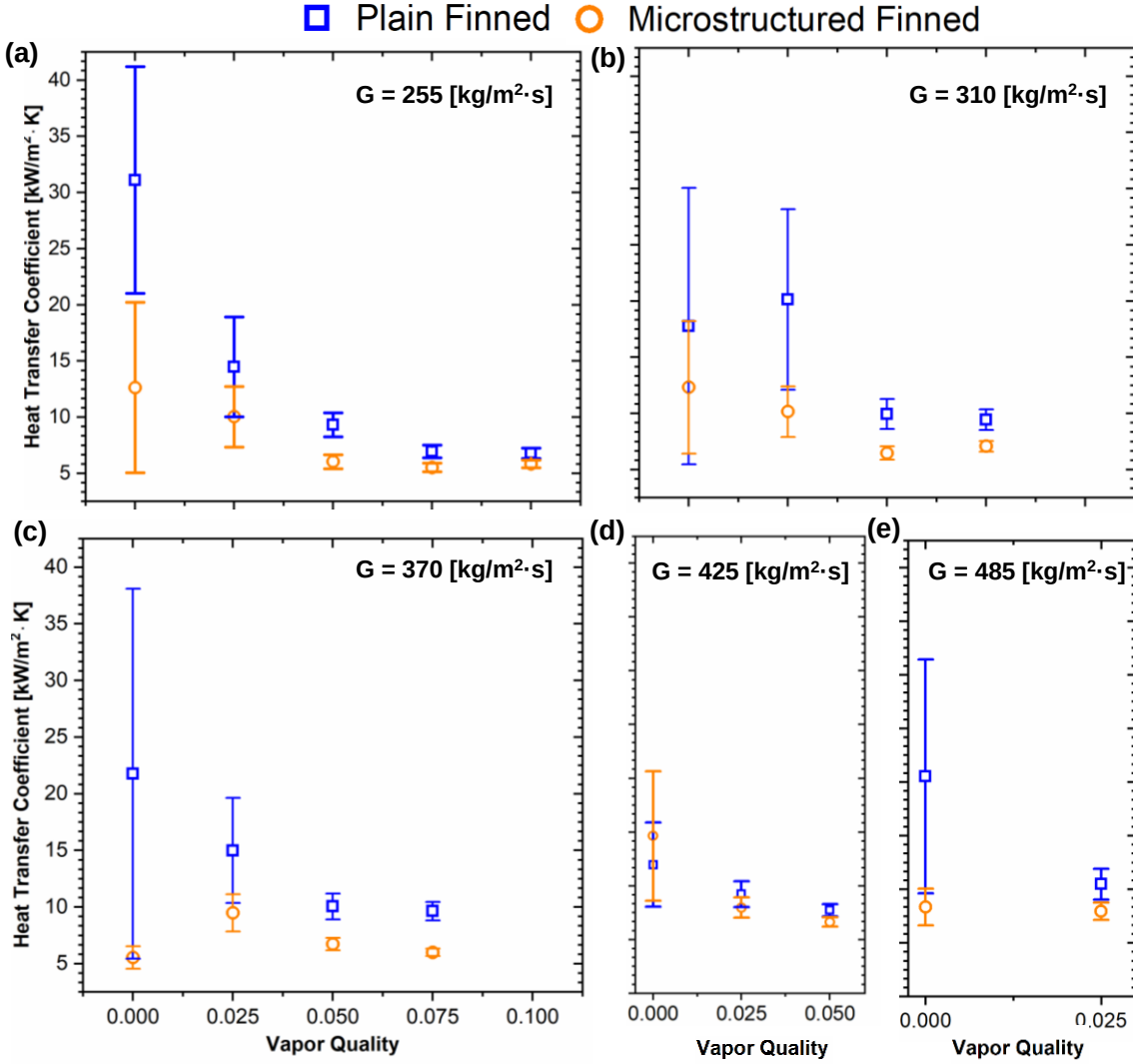


Figure 59. Heat transfer coefficient across vapor qualities

Heat transfer coefficient across vapor qualities for mass flux of: (a) $255 \text{ kg/(m}^2\cdot\text{s)}$, (b) $310 \text{ kg/(m}^2\cdot\text{s)}$, (c) $370 \text{ kg/(m}^2\cdot\text{s)}$, (d) $425 \text{ kg/(m}^2\cdot\text{s)}$, (e) $485 \text{ kg/(m}^2\cdot\text{s)}$

To demonstrate the effect of micro-structures on the frictional pressure drop of finned tubes, Fig. 60 shows the pressure drop of plain fin tubes and micro-structure fin tubes across a range of mass fluxes and vapor qualities. It can be noted that increasing the vapor quality causes an increase in the frictional pressure drop for both tubes, due to the increased fluid disturbance caused by the rising nucleate bubbles. Moreover, the increased void fraction inside the tube further

increases the shear stresses between the vapor and liquid flowing leading to an increased pressure drop.

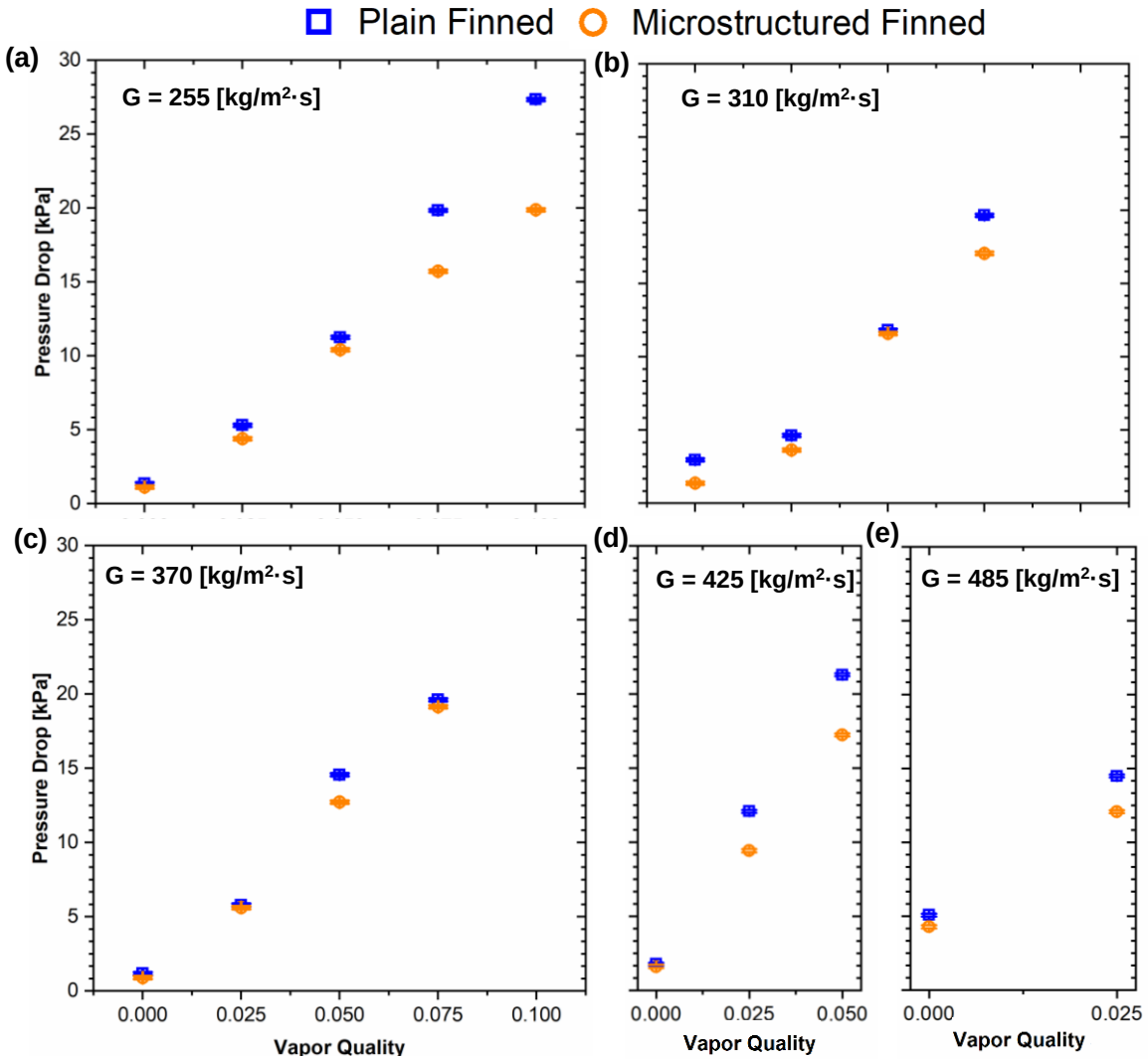


Figure 60. Pressure drop across vapor qualities

Pressure drop across vapor qualities for mass flux of: (a) 255 kg/(m²·s) , (b) 310 kg/(m²·s) , (c) 370 kg/(m²·s) , (d) 425 kg/(m²·s), (e) 485 kg/(m²·s)

However, when comparing micro-structured finned copper tubes with plain finned copper tubes, it can be noted that the frictional pressure drop decreases by 2-96 % when introducing the added roughness to the tube surface. For mass fluxes of 255 kg/(m²·s), 310 kg/(m²·s), 370

kg/(m²·s), 425 kg/(m²·s), 485 kg/(m²·s), the pressure drop of the micro-structured tube decreased by 8-27%, 2-54%, 2-93%, 9-22%, and 16-17%, compared to plain finned tubes respectively. This could be explained by the added nucleation sites that are present in micro-structured tubes. As highlighted previously, the increase in bubble nucleation sites along the micro-structured tube fins could cause partial film boiling at the troughs of the finned tubes. This is due to vapor patches that form due to the coalescence of nucleate bubbles. As such, this inhibits liquid from wetting portions of the tube walls and effectively minimize the skin friction as only partial wetting occurs.

Overall Thermal Performance

From the aforementioned discussion, it can be noted that the benefits of having micro-structures added to finned copper tubes is equivocal. While the heat transfer coefficient deteriorates, the pressure drop is found to decrease. Therefore, even though the power density of the heat exchange devices decreases, the pumping power required to operate the system at accustomed flow rate levels decreases. In order to gauge whether the benefits outweigh the drawbacks, Eq. (30) is used. In Fig. 61 the results are outlined for different mass flux and vapor quality ranges. Overall, we note that the PF is consistently lower than unity, indicating that the deterioration drawback in the HTC far outweighs the enhancement in pressure drop.

While micro-structured fin tubes have PF lower than unity, it is important to acknowledge that different variables could cause more favorable results. The dimensional parameter of fins including the helical angle, space width, fin width, and fin depth can greatly affect the physics governing flow boiling [449]. Different fin dimensions could lead to more favorable bubble nucleation and increased heat transfer augmentation. Additionally, different vapor quality ranges manifest different flow boiling mechanisms. In this study, we are mainly concerned with nucleate flow boiling dominant regimes (up to low vapor quality annular flow regime). However, in higher

vapor quality ranges, the dominating flow boiling mechanism is convective. Since the added micro-structures increase surface roughness, the water affinity inside the tube increases. Consequently, during high vapor quality annular flow regimes, the thickness of the liquid layer covering the inner tube walls decreases which enhances the effectiveness of vaporization at the liquid vapor interface as the thermal resistivity associated to the liquid layer decreases. This could potentially lead to increased heat transfer augmentation. Finally, the flowing refrigerant characteristics can greatly affect the overall PF. A thorough review of the effect of different refrigerants on the heat transfer rate is available in Ref. [449].

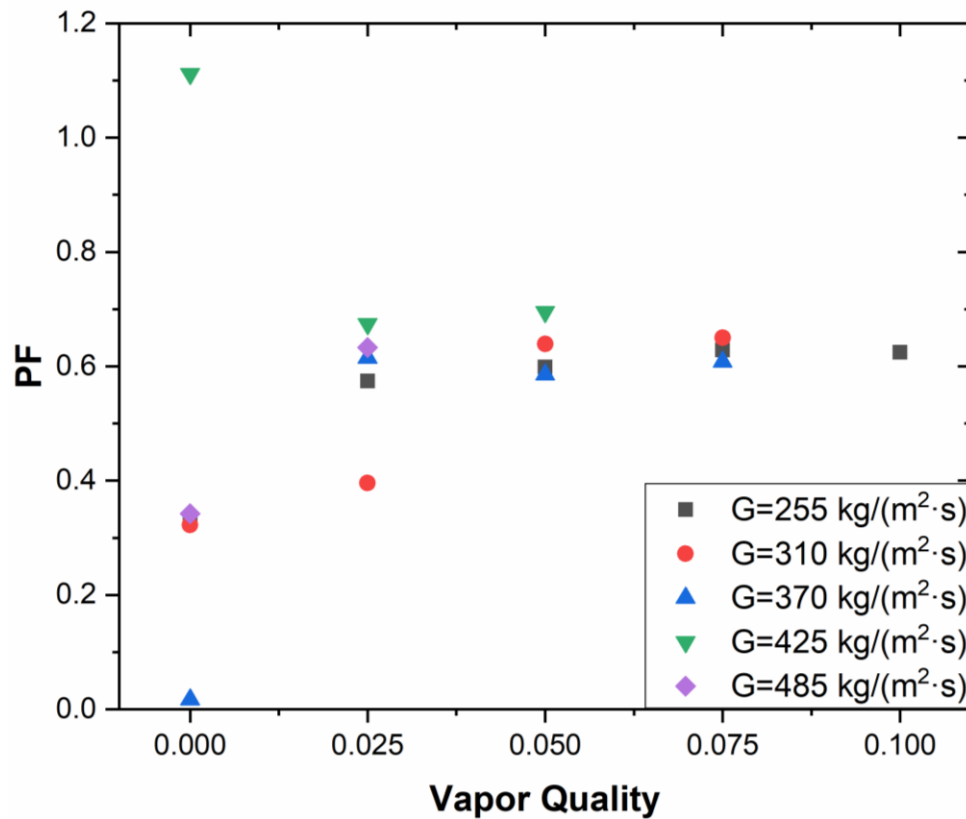


Figure 61. Performance factor of micro-structure finned tubes

Performance factor of micro-structure finned tubes across vapor quality range for different mass flux values

7.5 Effect of Microstructures on Aluminum Tube Thermal Performance

7.5.1 Surface Fabrication

Three aluminum tubes were studied: plain, herringbone, and etched plain tubes. The herringbone tube is commercially available and manufactured by Vipertex. The tube dimensions as well as other parameters are presented in Table 16. In this study, a mass flux ranging between 200 - 380 $\text{kg}/(\text{m}^2\cdot\text{s})$ was investigated. To obtain different vapor qualities at output, the test section heater power was varied between 0-2.4 kW. Before etching the aluminum tube for investigation, the tube was first cleaned by dipping it in acetone (CAS# 67-64-1) to degrease it and remove organic materials. The tube was then cleansed with ethanol (CAS# 64-17-5), then isopropanol (CAS# 67-63-0) before rinsing it with DI water (CAS# 7732-18-5). In order to etch the aluminum tube, it was first wrapped by 1-mm thick Kapton tape to cover its outer wall before being inserted in a 1-inch diameter stainless steel pipe that is plugged on one end with a rubber stopper. A 500 mL 2 M HCl (CAS# 7647-01-0) solution is then poured into the steel pipe containing the aluminum tube and was allowed to react for 15 minutes. Since the outer tube wall is covered by tape, only the inner tube wall surface was etched by the HCl solution while the outer surface remained untreated and smooth. The tube is then removed from the pipe and cleaned by DI water and isopropanol to remove any remaining residues. The resulting structure is illustrated in the figure 62 below.

Table 14. Characterization of tubes prior to surface etching

Parameter	Value/Range
Smooth Tube	
Thermal conductivity (k_s)	167 [W/(m·K)]
Root-mean-square (RMS) roughness	25 [nm]
Outer Diameter (OD)	6.35 [mm]
Inner Diameter (d)	5.33 [mm]
Tube Length (L)	0.955 [m]
Herringbone Tube	
Thermal conductivity (k_s)	167 [W/(m·K)]
Outer Diameter (OD)	6.35 [mm]
Inner Diameter (d)	5.33 [mm]
Tube Length (L)	0.955 [m]
Fin Width (FW)	0.11 [mm]
Fin Depth (FD)	0.26 [mm]
Helical angle (β)	38 °
Etched Tube	
Thermal conductivity (k_s)	167 [W/(m·K)]
Root-mean-square (RMS) roughness	12.6 [μ m]
Tube Diameter (OD)	0.25 [in]
Inner Diameter (d)	0.21 [in]
Tube Length (L)	0.955 [m]

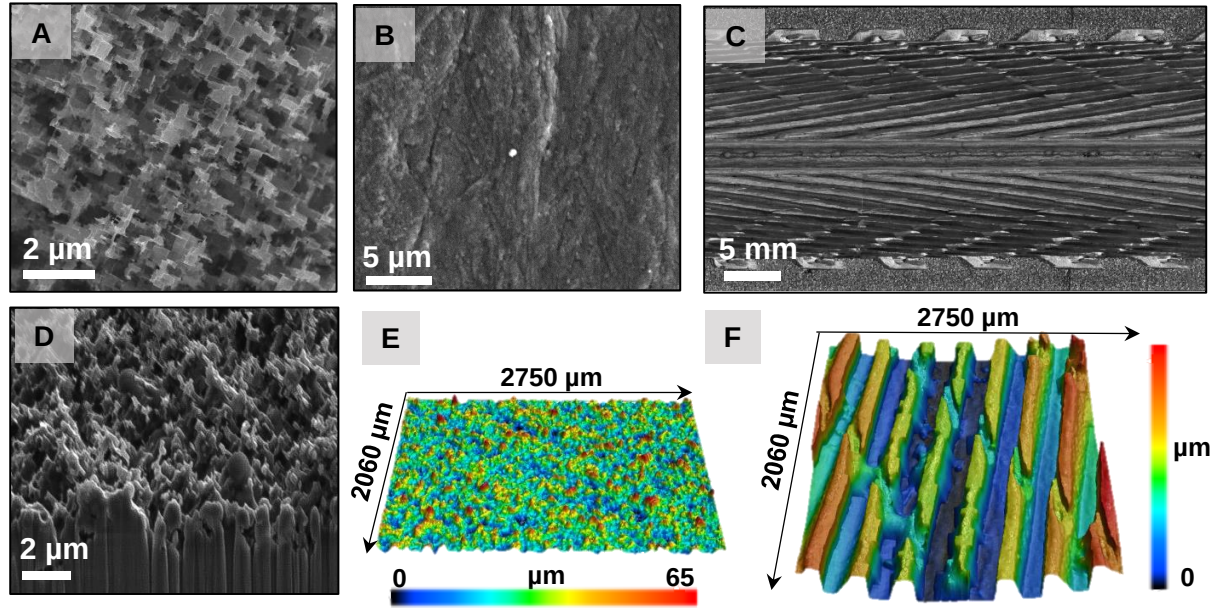


Figure 62. Microstructures formed on plain Al tube samples

Top view scanning electron microscopy (SEM) image of (A) etched aluminum (grade 6061) microstructured surface and (B) plain aluminum surface inside the tube. (C) Confocal optical microscopy image showing finned enhancement inside the finned aluminum tube. (D) SEM image of the etched aluminum microstructured surface cross section prepared by focus ion beam (FIB) milling. (E) 3D laser scanning confocal microscopy image showing the topography of the etched microstructured aluminum surface (root-mean-square roughness is $\pm 12.6 \mu\text{m}$) and (F) the interior topography of the finned enhanced aluminum tubes.

7.5.2 Results and Discussion

To validate the flow loop setup, single phase testing is conducted on a plain aluminum tube for flow rates ranging between $7000 < Re_D < 10000$ and $630 \text{ kg}/(\text{m}^2 \cdot \text{s}) < G < 740 \text{ kg}/(\text{m}^2 \cdot \text{s})$. Subjecting the DI water to a heat flux ranging between $60 \text{ kW}/\text{m}^2 < q'' < 125 \text{ kW}/\text{m}^2$, the experimental heat transfer coefficient (Eq. 40) is compared to the empirical heat transfer predicted by Gnielinski's correlation (Eq. 49) [258]. As shown in the figure 63 below, the experimental results lie within 20% of the values predicted by Gnielinski's correlation, thus validating the flow loop.

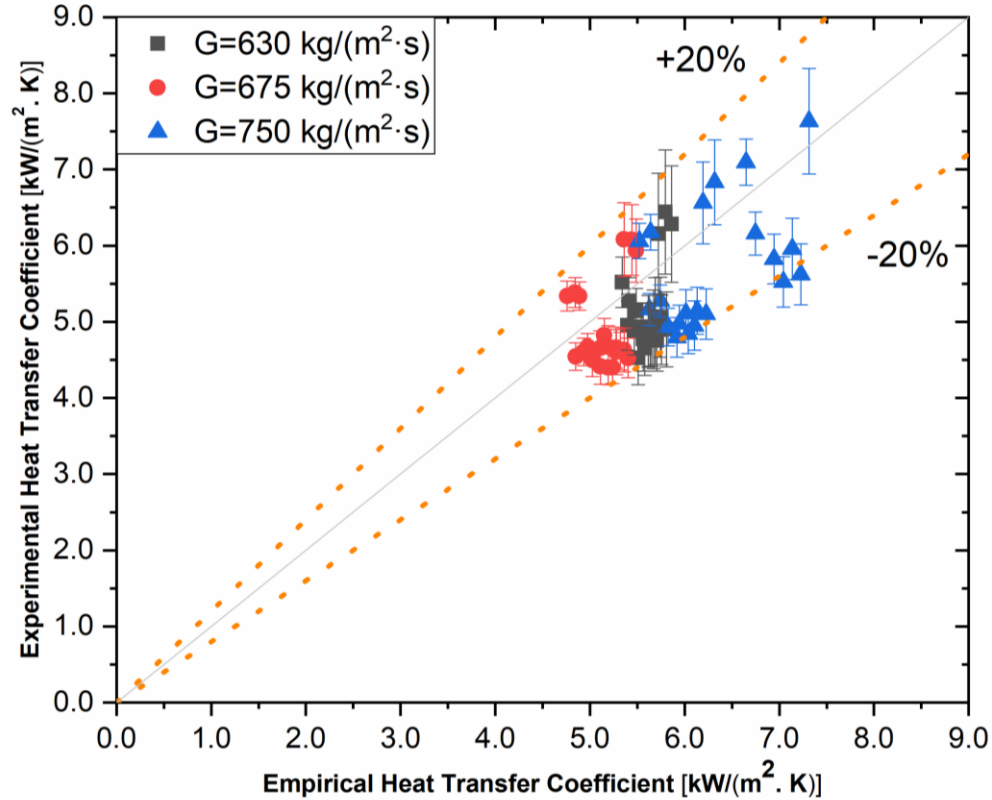


Figure 63. Experimental versus empirical heat transfer coefficient

Measured experimental heat transfer coefficient as a function of the empirical heat transfer coefficient correlation as predicted by Gnielinski [258]

Effect of microstructures on the overall thermal performance of aluminum tubes

To quantify the performance of microstructures in two phase flow boiling applications, the heat transfer coefficient and pressure drop of the tube was measured for a range of different mass flux and vapor qualities as outlined in the figure 64 below. Further, the performance is benchmarked against a plain tube of similar diameter as well as a commercially available herringbone tube of similar meltdown diameter [456]. It is important to note that the vapor quality range attained is limited by the flow rate conditions and the maximum heat flux that can be provided by the test section heater and the multiple preheaters. As such, with increasing mass flux values, the vapor quality range becomes more restricted. Additionally, the vapor quality range demonstrated in the

figure 64 below refer to the conditions at the outlet of the test section and can be controlled by varying the power of the test section heater.

Across the mass flux ranging between $200 \text{ kg}/(\text{m}^2 \cdot \text{s}) < G < 380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, heat transfer coefficient increases by 0% - 104% and 1.9%-53% for etched and herringbone tubes respectively when compared to plain tube. For etched tubes of mass flux values $G = 200 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $250 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $290 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $335 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the heat transfer coefficient of the enhanced tube increased by 3.6 to 25.4%, 0 to 66.4%, 0 to 104%, 7.8 to 45%, and 5.1 to 12%, respectively, when compared to the smooth tube. While for herringbone tubes of mass flux values $G = 200 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $250 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $290 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $335 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the heat transfer coefficient of the enhanced tube increased by 28.5 to 53.2%, 5 to 15.2%, 10.4 to 46.4%, 4.8 to 13.4%, and 1.9 to 21.5%, respectively, when compared to the smooth tube.

Saturated flow boiling across a range of different vapor qualities can be subdivided into two phases namely, nucleate flow boiling dominant and the convective flow boiling dominant [454]. When saturated flow boiling starts at $x = 0$, the dominant phase is nucleate flow boiling which is manifested by the bubbly flow regime and is characterized by small nucleate bubbles forming at the inner tube walls that release into the fluid flow. Typically, these bubbles are released from cavities that entrap air when fluid first wets the surface. Vaporization occurs at the liquid-vapor interface of these entrapped bubbles at relatively lower superheats causing the bubbles to release. Once the bubble releases, it is usually replaced by another air bubble and the process is repeated numerously. The released air bubbles cause agitation and intermixing as it rises which further contributes to the enhancement of the heat transfer. Consequently, it is intuitive to conclude that more cavities in the inner tube walls would translate to an increase in the heat transfer coefficient during the nucleate boiling dominant flow phase. As a result, since etched tubes increases the

overall tube roughness and the density of the entrapped air bubbles in the cavities, we observe an increase in the heat transfer coefficient.

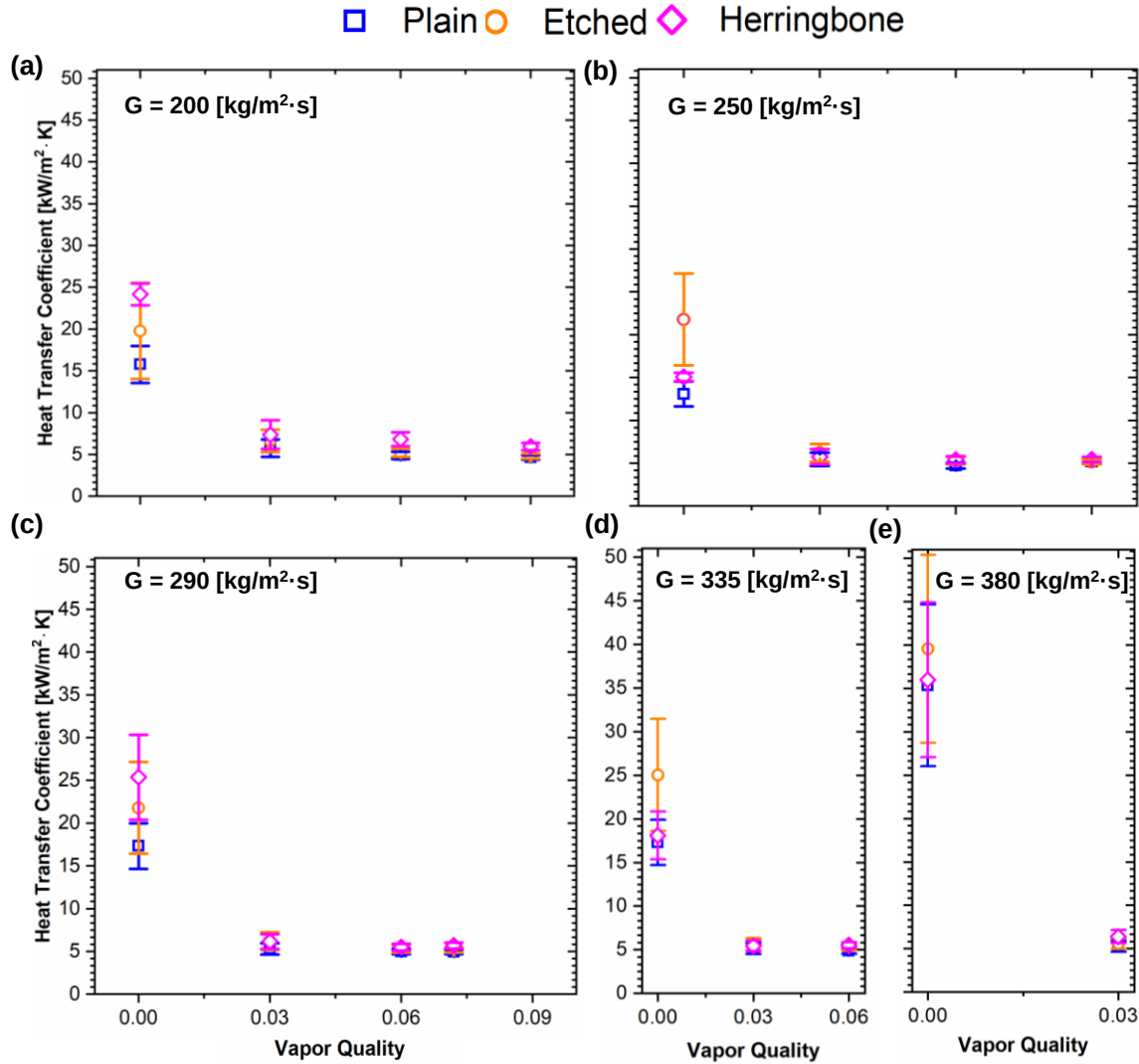


Figure 64. Water flow boiling heat transfer coefficient

Water flow boiling heat transfer coefficient as a function of exit vapor quality at mass fluxes of: (a) $G = 200 \text{ kg/(m}^2\cdot\text{s)}$, (b) $G = 250 \text{ kg/(m}^2\cdot\text{s)}$, (c) $G = 290 \text{ kg/(m}^2\cdot\text{s)}$, (d) $G = 335 \text{ kg/(m}^2\cdot\text{s)}$, and (e) $G = 380 \text{ kg/(m}^2\cdot\text{s)}$.

However, as the vapor quality increases, this increase in heat transfer coefficient diminishes in both plain and etched tubes due to the suppression of the nucleation sites. The quick transition to annular flow regime from bubbly flow regime is dependent on the discrepancy between the density value of the liquid compared to vapor. For water, the discrepancy is substantial and so the transition

to annular flow regime occurs at very low vapor qualities. In such annular flow regimes, only small cavities which require high superheats are active. As the vapor quality increases, the liquid film covering the inner tube wall becomes thinner and so the active cavities become even smaller in size until eventually the superheat of the inner tube walls required to keep these smaller cavities active is much higher than what can be supplied by the rope heater and the preheaters. This effectively deactivates the all cavities and causes the diminishing heat transfer coefficient we observe. It is important to note that even though this phenomenon occurs in both plain and etched tubes, the effect of cavity suppression is reflected more on etched tubes due to the presence of more nucleation sites.

Similar to etched tube, herringbone tube enhanced the heat transfer during two phase flow boiling. This is because the internal extended surface structures of the tube act as fins that increase the surface area of contact between the inner tube wall and the flowing fluid over which nucleation could occur as well as cause agitation and secondary flows. Additionally, the chevron shaped grooves that characterize the herringbone tube carries the fluid flow in two opposing directions, which result in the film layer around the circumference to not be uniform [288]. Similar to the etched tube, this heat transfer coefficient enhancement due to the suppression of the active nucleation sites.

It is worth noting that in both herringbone and etched aluminum tubes, the heat transfer coefficient was not affected by the mass flux. This can be attributed to the fact that nucleation is the dominating enhancement factor affecting the heat transfer coefficient in the vapor quality range investigated. As the vapor quality increases in annular flow regime is observed, it is projected that eventually the heat transfer coefficient would increase again. This is because with increasing vapor quality, the internal vapor velocity inside the tube would increase causing the convective

vaporization at the liquid vapor interface to increase. The increase in heat transfer coefficient due to convective vaporization would eventually overshadow that due to nucleate vaporization causing the overall heat transfer coefficient to increase even though all cavities have been suppressed. In this vapor quality range, etched aluminum tubes are expected to enhance the heat transfer. The added roughness of the cavities causes the surface to be wetting and the tube to become hydrophilic. When annular flow regime is observed, the liquid layer that covers the inner tube walls would become thinner due to the increased hydrophilicity of the tube. Since this reduces the thermal resistance of the liquid layer, the effectiveness of the vaporization would increase compared to plain aluminum tube which augments heat transfer.

Even though both the herringbone tube and etched tube caused an enhancement in heat transfer compared to the plain tube, the pressure drop observed in both tubes were also higher as evident in figure 65. Across the mass flux ranging between $200 \text{ kg}/(\text{m}^2 \cdot \text{s}) < G < 380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, pressure drop increased by 0% - 65.4% and 31%-168.8% for etched and herringbone tubes respectively when compared to plain tube. For etched tubes of mass flux values $G = 200 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $250 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $290 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $335 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the heat transfer coefficient of the enhanced tube increased by 11.1 to 29.9%, 10.7 to 65.4%, 0 to 30.7%, 0 to 8%, and 0 to 21%, respectively, when compared to the smooth tube. While for herringbone tubes of mass flux values $G = 200 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $250 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $290 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $335 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $380 \text{ kg}/(\text{m}^2 \cdot \text{s})$, the heat transfer coefficient of the enhanced tube increased by 31.9 to 137.1%, 44.2 to 168.8%, 54.9 to 130.6%, 30.9 to 50.7%, and 75.5 to 80.6%, respectively, when compared to the smooth tube.

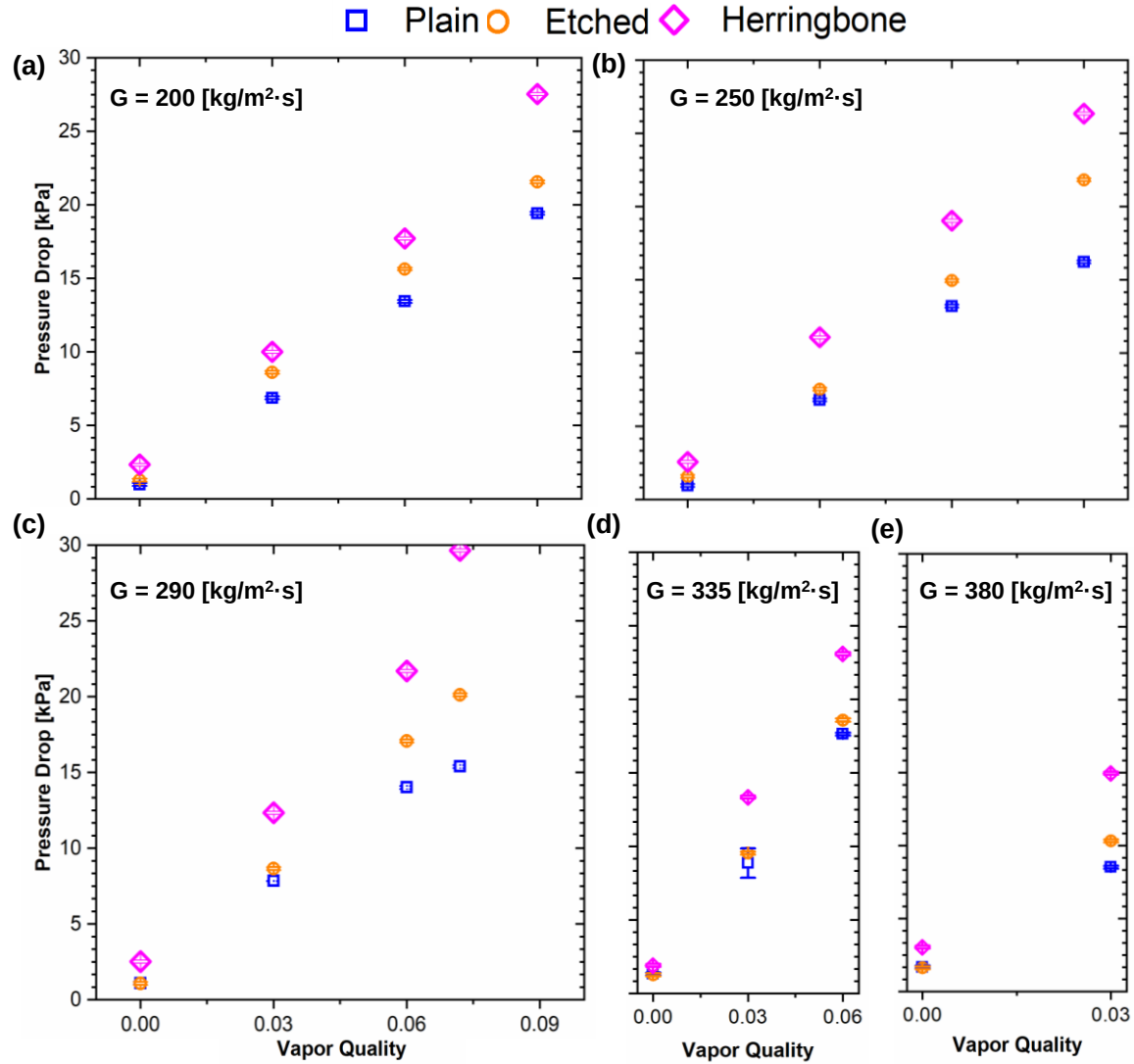


Figure 65. Flow boiling pressure drop

Flow boiling pressure drop as a function of exit vapor quality for mass fluxes of: (a) $G = 200$ $\text{kg}/(\text{m}^2 \cdot \text{s})$, (b) $G = 250$ $\text{kg}/(\text{m}^2 \cdot \text{s})$, (c) $G = 290$ $\text{kg}/(\text{m}^2 \cdot \text{s})$, (d) $G = 335$ $\text{kg}/(\text{m}^2 \cdot \text{s})$, and (e) $G = 385$ $\text{kg}/(\text{m}^2 \cdot \text{s})$

The increase in pressure drop in the etched tubes can be attributed to the increased skin friction that is caused by the added roughness presented by the microstructure cavities. Further, the increased bubble departure frequencies that happen due to the added nucleation sites cause more agitation and intermixing that further attribute to the increase in pressure drop. Similarly, for herringbone tubes, the welded fins cause an increase in the surface area which increases the skin friction as well as disrupt fluid flow. As a result, the overall pressure drop in herringbone tubes

was substantially higher than plain tubes. It can be noted that in both herringbone and etched tubes, when the vapor quality increases, the overall pressure drop increases. This is because with higher vapor quality, the void fraction inside the tube increases. As a result, the added vapor causes an increased interaction with the surrounding liquid layer covering the tube walls. Since the velocity of the vapor is relatively higher than that of the liquid, the shear stresses increase with increasing vapor qualities.

Overall Thermal Performance

In order to gauge whether the etched and herringbone tubes could be of interest, we use the performance factor (PF) to study how the heat transfer coefficient augmentation compares to the increase in pressure drop. In the figure 66 below, the PF was investigated for a range of different vapor qualities and mass fractions. As can be concluded from Fig. 66a, even though in almost all testing conditions, an augmentation in the HTC can be observed, some testing conditions have PF lower than unity indicating that the HTC augmentation is overshadowed by the pressure drop increase. Generally, it can be noticed that the PF was higher at lower vapor qualities for etched aluminum tubes. This is because at lower vapor qualities, most nucleation sites are active and the speed of the fluid is generally lower. As the vapor quality increases, suppression of nucleation sites occurs which deteriorates the HTC and the velocity of the vapor inside the tube increases causing higher pressure drops. Consequently, the PF drops with increasing vapor quality.

Contrasting the results with herringbone aluminum tubes, it can be concluded from Fig. 66b that in almost all testing conditions the PF is lower than unity. This is attributed to the very large pressure drop in herringbone tubes that occur due to the increase in surface area. In fact, the chevron pattern inside the tube causes the internal lateral surface area of the herringbone tube to

be 2.7 times more than the plain aluminum. However, it is important to note that this does not mean that herringbone tubes are not useful in heat exchanger applications where pressure drop is not the deciding factor. As discussed previously, herringbone tubes can augment heat transfer substantially. Furthermore, it is important to acknowledge that with different testing conditions, results for herringbone tubes could be more favorable. Such conditions include the vapor quality range, the fluid utilized, the tube orientation, the tube diameter etc...

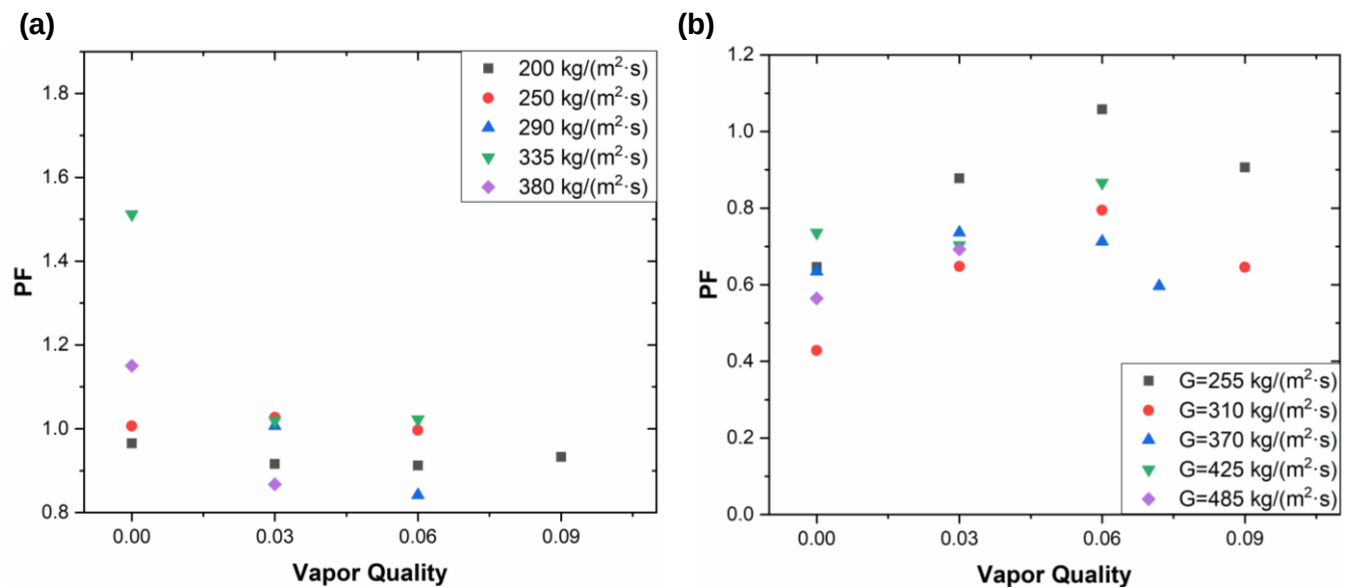


Figure 66. Performance factor of aluminum tubes
Performance factor of different aluminum tubes investigated where: a) PF of etched aluminum tube, and b) PF of herringbone aluminum tube

7.6 Conclusion

In this chapter, an experimental setup was constructed to investigate the effect of microstructures on the heat transfer and pressure drop characteristics of water flowing inside of aluminum and copper tubes. The study was conducted over a range of mass flux and vapor quality ranges in order to evaluate the effect of the microstructures across different operating conditions. Results show that having microstructures in plain copper and aluminum tubes have an overall positive enhancement in the rate of heat transfer. It was noted that the heat transfer coefficient increased up to 90% in copper and up to 104% in aluminum due to the added nucleation sites. However, accompanying this heat transfer augmentation is an increase in pressure drop. An increase of up to 38% was reported in copper tubes and up to 137% in aluminum tubes. However, these microstructures had opposite effect on finned copper tubes. Studying the heat transfer coefficient and pressure drop across different mass flux and vapor qualities it was concluded that adding these structures decreased the heat transfer by 14 – 75% and decreased the pressure drop by 2-96 %. Since the drop in heat transfer outweighs the decrease in pressure drop, using microstructures to finned tubes is not desirable.

Due to the potential applications of microstructures, future work could be devoted to investigating the fouling potential of microstructured tubes and thermal stresses that arise during tube expansion. Furthermore, durability studies of these structures across different operating conditions are paramount to compare the longevity of this enhancement technique to others. Additionally, the effect of different structure sizes coupled with different refrigerants can provide useful insight on how to reap the full potential of this enhancement technique. Finally, it is important to extend these studies to include different tube orientations, diameters, materials, and

testing conditions. This would provide a useful bank of information required for machine learning and topology optimization to be successful.

Chapter 8 Conclusion

In this dissertation, novel platforms were developed to investigate liquid vapor phase change of water during evaporation and two-phase flow boiling. The main findings can be summarized as follows:

- For sessile droplets residing on flat surfaces, as the radius of the droplet increased the temperature mal distribution around the droplet increased. This is attributed to the increased conduction resistance at the base of the droplet. Consequently, for relatively larger sized droplets more heat is absorbed from the liquid vapor interface rather than the substrate in order to cause evaporation
- The thermal transport pathway is dependent on the thermal conductivity of the substrate. For droplets residing on substrates of relatively higher thermal conductivity, the temperature mal distribution around the droplet is relatively lower. This indicates that more heat is absorbed from the substrate to cause evaporation compared to droplets residing on substrates of relatively lower thermal conductivity.
- Droplets residing on substrates with higher thermal conductivities exhibit higher evaporation rate increase with temperature than substrates with lower thermal conductivities
- For droplets residing on substrates with similar thermal conductivity, the lower the surface wettability the larger the temperature gradient surrounding the droplet
- Two phase flow boiling can be divided into nucleate dominant and convective dominant depending on the vapor quality range. In nucleate dominant, the HTC diminishes with

increasing vapor quality with the maxima being at $x=0$. This is attributed to suppression of the nucleation cavities

- Having an increased surface roughness enhances the HTC in plain metallic tubes as the surface roughness acts as cavities that entrap air bubbles. Vaporization occurs at the liquid vapor interface of these entrapped air bubbles at relatively lower superheats. It was concluded that having microstructures increased the HTC in copper and aluminum tubes by 10-89% and 0-104% respectively when compared to their smooth metallic counterparts.
- Since microstructures add to the overall skin friction, it was observed that microstructured copper and aluminum tubes exhibited a 0.5-16% and 1.9-53% increase in pressure drop when compared to their smooth metallic counterparts.
- Microstructured finned copper tube was found to have an HTC and pressure drop decrease of 14-75% and 2-96% respectively when compared to their smooth finned copper tube counterpart. This is due to the vapor at the valleys of the fins partially blocking water from wetting the surface.

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Appendix A: Gas-Phase Temperature Mapping

Supplementary Information

Table 15. Dispensing frequency (f) for water droplets at different surface temperatures. Values in the bracket represent the evaporation heat transfer rate (Q_e).

Surface Temperature	Radius [μm]	Superhydrophobic Cu [Hz]	Hydrophobic Cu [Hz]	Superhydrophobic Glass [Hz]	Hydrophobic Glass [Hz]
23°C	140	27 (0.19 mW)	42**(0.29 mW)	16.2 (0.11 mW)	20 (0.14 mW)
	220	32 (0.22 mW)	45 (0.31 mW)	25 (0.17 mW)	31 (0.21 mW)
	295	36 (0.25 mW)	60 (0.41 mW)	27 (0.19 mW)	44**(0.3 mW)
20°C	140	24.7 (0.17 mW)	16.3 (0.1 mW)	16.5** (0.11 mW)	13.5 (0.1 mW)
	220	39** (0.27 mW)	25 (0.17 mW)	23.7** (0.16 mW)	14.1 (0.1 mW)
	295	46** (0.32 mW)	42 (0.29 mW)	29.5** (0.20 mW)	16 (0.11 mW)
30°C	140	35 (0.24 mW)	35.7 (0.3 mW)	21.5 (0.15 mW)	20.5 (0.1 mW)
	220	57 (0.39 mW)	56 (0.39 mW)	31 (0.21 mW)	36.9 (0.3 mW)
	295	70 (0.48 mW)	70 (0.48 mW)	52 (0.36 mW)	41 (0.28 mW)
40°C	140	56 (0.39 mW)	50 (0.34 mW)	49 (0.34 mW)	69.6 (0.5 mW)
	220	93 (0.64 mW)	74 (0.51 mW)	83 (0.57 mW)	112 (0.8 mW)
	295	106 (0.73 mW)	108 (0.7 mW)	-*	174 (1.2 mW)

*Could not record measurements due to attraction between thermocouple and droplet (Video S1).

**Larger than expected due to contaminants in the nozzle partially blocking the dispensing mechanism.

Table 16. Energy flow breakdown for microdroplets residing on substrates with $T = 23^{\circ}\text{C}$.

Radius [μm]	Superhydrophobic Cu			Hydrophobic Cu		
	Q_g [μW]	Q_e [μW]	Q_s [μW]	Q_g [μW]	Q_e [μW]	Q_s [μW]
140	1.92	289.34	287.41	8.37	186	177.63
220	50.34	310	259.66	35.38	220.45	185.07
295	100.42	413.34	312.92	57.14	248	190.86
Radius [μm]	Superhydrophobic Glass			Hydrophobic Glass		
	Q_g [μW]	Q_e [μW]	Q_s [μW]	Q_g [μW]	Q_e [μW]	Q_s [μW]
140	15.96	111.6	95.64	23.9	137.78	113.84
220	58.79	172.22	113.44	46.03	213.56	167.53
295	100.37	186	85.64	66.94	303.12	236.18

Table 17. Temperature values for superhydrophobic CuO coated with HTMS at different positions for droplets having $R \cong 140 \mu\text{m}$ and $R \cong 295 \mu\text{m}$.

Position	Temperature at $R \cong 140 \mu\text{m}$ [$^{\circ}\text{C}$]	ΔT [$^{\circ}\text{C}$]	Temperature at $R \cong 295 \mu\text{m}$ [$^{\circ}\text{C}$]	ΔT [$^{\circ}\text{C}$]
13° (A), Line 1	22.4	-0.01	22.3	0.81
13° (A), Line 3	22.4		23.2	
27° (B), Line 1	22.4	0.12	22.2	0.44
27° (B), Line 3	22.5		22.6	
40° (C), Line 1	22.5	0.04	22.3	0.42
40° (C), Line 3	22.6		22.7	
52° (D), Line 1	22.5	-0.21	22.3	0.35
52° (D), Line 3	22.3		22.7	

Table 18. Temperature values for superhydrophobic glass coated with Glaco at different positions for droplets with $R \cong 140 \mu\text{m}$ and $R \cong 295 \mu\text{m}$.

Position	Temperature at $R \cong 140 \mu\text{m}$ [$^{\circ}\text{C}$]	ΔT [$^{\circ}\text{C}$]	Temperature at $R \cong 295 \mu\text{m}$ [$^{\circ}\text{C}$]	ΔT [$^{\circ}\text{C}$]
13° (A), Line 1	23.5	0.47	24.0	0.59
13° (A), Line 3	23.9		24.6	
27° (B), Line 1	23.5	0.58	23.9	0.65
27° (B), Line 3	24.1		24.6	
40° (C), Line 1	23.6	0.48	24.2	0.52
40° (C), Line 3	24.1		24.7	
52° (D), Line 1	23.7	0.47	24.3	0.35
52° (D), Line 3	24.2		24.6	

Table 19. Temperature for different surfaces at different radial positions across line 1 for a droplet having $R \cong 140 \mu\text{m}$.

Position	Temperature for superhydrophobic Cu [$^{\circ}\text{C}$]	Temperature for hydrophobic Cu [$^{\circ}\text{C}$]	Temperature for superhydrophobic Glass [$^{\circ}\text{C}$]	Temperature for hydrophobic Glass [$^{\circ}\text{C}$]
13° (A)	22.4	23.9	23.5	22.1
27° (B)	22.4	24.0	23.5	22.0
40° (C)	22.5	24.0	23.6	22.0
52° (D)	22.5	24.0	23.7	22.1
76° (E)	22.5	24.1	23.8	22.0

Table 20. Interfacial thermal resistance (ψ_i).

Surfaces	$R \cong 140 \mu\text{m}$	$R \cong 220 \mu\text{m}$	$R \cong 295 \mu\text{m}$
Superhydrophobic Glass	10.3	6.57	4.90
Hydrophobic Glass	14.6	9.26	6.91
Hydrophobic Cu	13.6	8.67	6.47
Superhydrophobic CuO	10.2	6.48	4.83

Table 21. Thermal resistance at the base of the droplet (ψ_s).

Surfaces	$R \cong 140 \mu\text{m}$	$R \cong 220 \mu\text{m}$	$R \cong 295 \mu\text{m}$
Superhydrophobic Glass	3.21×10^3	2.85×10^3	2.21×10^3
Hydrophobic Glass	1.50×10^3	9.39×10^2	7.06×10^2
Hydrophobic Cu	4.71	3.14	2.33
Superhydrophobic CuO	15.6	9.55	8.30

Table 22. Thermal resistance of heat conduction through the droplet (ψ_c).

Surfaces	$R \cong 140 \mu\text{m}$	$R \cong 220 \mu\text{m}$	$R \cong 295 \mu\text{m}$
Superhydrophobic Glass	9.84×10^3	6.26×10^3	4.67×10^3
Hydrophobic Glass	2.03×10^3	1.29×10^3	9.63×10^2
Hydrophobic Cu	2.25×10^3	1.43×10^3	1.07×10^3
Superhydrophobic CuO	1.81×10^4	1.15×10^4	8.58×10^3

Table 23. Comparison between experimental and analytical measurements for a droplet having $R \cong 140 \mu\text{m}$ and substrate temperature of 30°C ($T = 30^\circ\text{C}$).

Position	Experimental Measurement [$^\circ\text{C}$]	Analytical Calculation [$^\circ\text{C}$]	Difference [%]
13° (A), Line 1	28.0	27.8	0.7
13° (A), Line 2	28.1	27.3	2.9
13° (A), Line 3	28.0	26.9	3.9
27° (B), Line 1	27.9	27.8	0.19
27° (B), Line 2	27.7	27.3	1.4
27° (B), Line 3	27.8	26.8	3.7
40° (C), Line 1	27.6	28	1.4
40° (C), Line 2	27.8	27.3	1.9
40° (C), Line 3	27.6	26.7	3.1
52° (D), Line 1	27.4	28.1	2.4
52° (D), Line 2	27.8	27.7	0.29
52° (D), Line 3	27.8	26.6	4.3
76° (E), Line 1	27.9	28.1	0.63
76° (E), Line 2	27.7	27.1	1.2

Table 24. Comparison between experimental and analytical measurements for a droplet having $R \cong 140 \mu\text{m}$ and substrate temperature of 23°C ($T = 23^\circ\text{C}$).

Position	Experimental Measurement [$^\circ\text{C}$]	Analytical Calculation [$^\circ\text{C}$]	Difference [%]
13° (A), Line 1	22.1	23.3	5.4
13° (A), Line 2	22.3	23.4	4.7
13° (A), Line 3	22.5	23.6	5.1
27° (B), Line 1	22.0	23.3	5.7
27° (B), Line 2	22.3	23.5	5.3
27° (B), Line 3	22.4	23.6	5.2
40° (C), Line 1	22.0	23.3	5.9
40° (C), Line 2	22.3	23.5	5.5
40° (C), Line 3	22.4	23.6	5.1
52° (D), Line 1	22.1	23.4	5.7
52° (D), Line 2	22.3	23.5	5.5
52° (D), Line 3	22.4	23.6	5.2
76° (E), Line 1	22.0	23.3	5.8
76° (E), Line 2	22.2	23.4	5.2

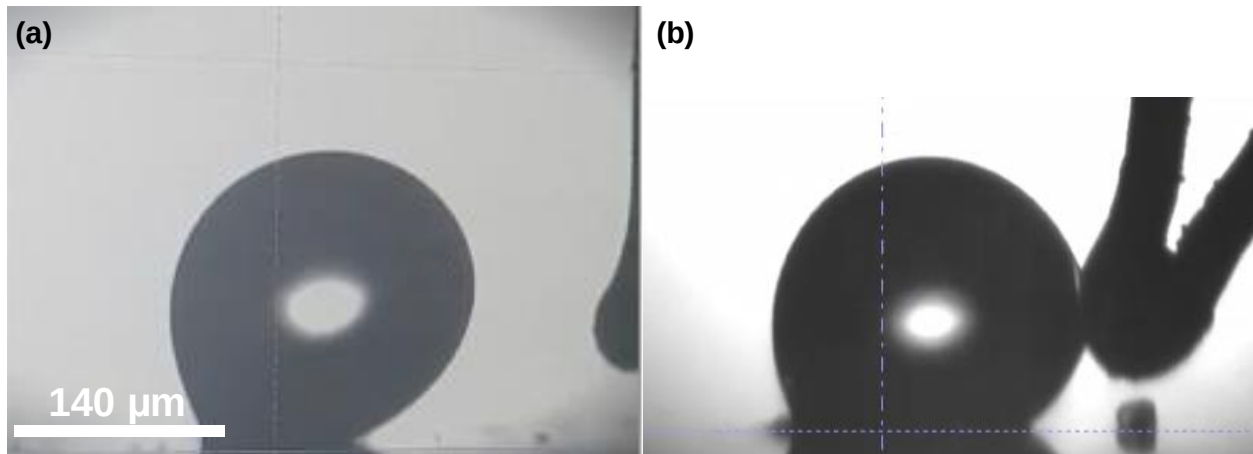


Figure 67. Attraction between the thermocouple

Side view optical image showing attraction between the thermocouple (object on the right) and droplet (center) with (a) attraction causing distortion in the spherical droplet profile, and (b) the droplet attaching to the thermocouple.

Appendix B: Bias Error

In order to determine the uncertainty of a temperature reading, it is necessary to consider the associated errors with a temperature reading. Typically, the total measurement error can be subcategorized into precision and bias errors. While precision errors reflect errors associated with statistical random variations in repeated measurements, bias errors do not relate to statistical data and are typically more involved to accurately predict. To calculate the bias error in the case of the thermocouples used in the two-phase flow boiling experiments, we consider the following system:

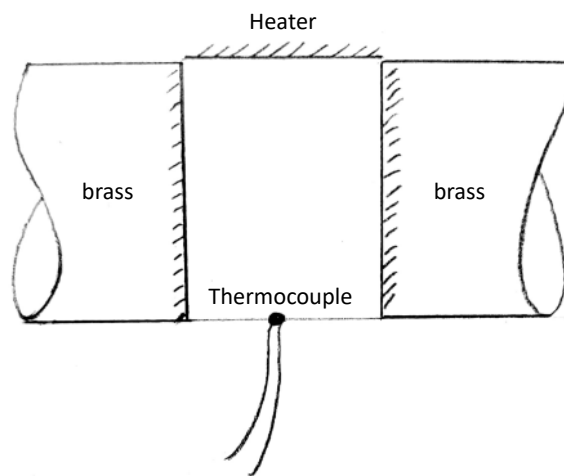


Figure 68. Thermocouple System

In the above system, the thermocouple is in temperature equilibration with the other surrounding constituents. Considering the energy balance on the thermocouple, heat is transferred to the surrounding air through convection and heat is further exchanged with the surrounding brass blocks and heater via radiation. To calculate the heat exchange through radiation the following network is considered:

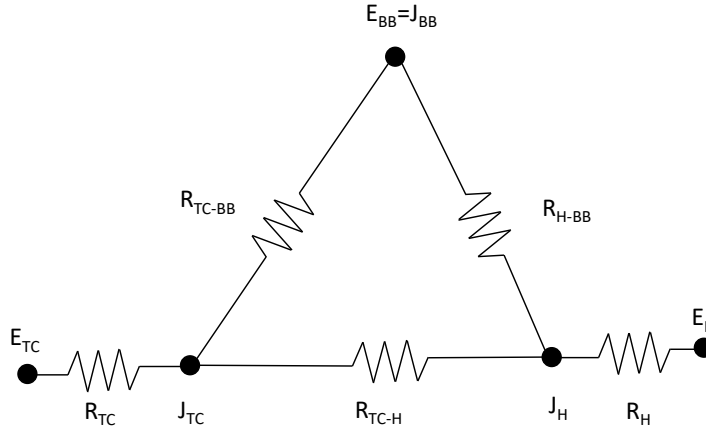


Figure 69. Radiation resistance diagram

The surface resistances are:

$$R_{TC} = \left(\frac{1 - \varepsilon_{TC}}{A_{TC}\varepsilon_{TC}} \right)$$

Where R_{TC} is the surface resistance of the thermocouple, ε_{TC} is the emissivity of the thermocouple, and A_{TC} is the surface area of the thermocouple.

$$R_H = \left(\frac{1 - \varepsilon_H}{A_H\varepsilon_H} \right)$$

Where R_H is the surface resistance of the heater, ε_H is the emissivity of the heater, and A_H is the surface area of the heater.

The space resistances are:

$$R_{TC-H} = \frac{1}{A_{TC}F_{TC-H}}$$

Where R_{TC-H} is the space resistance between the thermocouple and the heater, A_{TC} is the surface area of the thermocouple, and F_{TC-H} is the view factor between the thermocouple and the heater.

$$R_{H-BB} = \frac{1}{A_H F_{H-BB}}$$

Where R_{H-BB} is the space resistance between the brass block and the heater, A_H is the surface area of the heater, and F_{H-BB} is the view factor between the heater and the brass block.

$$R_{TC-BB} = \frac{1}{A_{TC} F_{TC-BB}}$$

Where R_{TC-BB} is the space resistance between the thermocouple and the brass blocks, A_{TC} is the surface area of the thermocouple, and F_{TC-BB} is the view factor between the thermocouple and the brass block.

The view factor correlations used were obtained from Ref. [457,458]

For node J₁:

$$\left(\frac{E_{bTC} - J_{TC}}{R_{TC}} \right) + \left(\frac{J_H - J_{TC}}{R_{TC-H}} \right) + \left(\frac{J_{BB} - J_{TC}}{R_{TC-BB}} \right) = 0$$

For node J₂:

$$\left(\frac{E_{bH} - J_H}{R_H} \right) + \left(\frac{J_{TC} - J_H}{R_{TC-H}} \right) + \left(\frac{J_{BB} - J_H}{R_{H-BB}} \right) = 0$$

where

$$J_{BB} = E_{bBB} = \sigma(T_{BB}^4)$$

$$E_{bTC} = \sigma(T_{TC}^4)$$

$$E_{bH} = \sigma(T_H^4)$$

Where σ is the Stefan-Boltzmann constant, T_H is the temperature of the heater, T_{TC} is the temperature of the thermocouple, T_{BB} is the temperature of the brass block.

To calculate the total heat gained through radiation:

$$Q_{radiation} = - \frac{E_{bTC} - J_{TC}}{R_{TC}}$$

To calculate the heat transfer due to convection:

$$Q_{convection} = h A_{TC} (T_{TC} - T_{air})$$

Where T_{air} is the temperature of the air surrounding the thermocouple, the heat transfer coefficient was calculated using $Nu_D=2$, given the conduction limit for a sphere

Since the heat gained through radiation must equal to the heat loss due to convection, the temperature measured by the thermocouple is calculated. Knowing that the bias error is the difference between the thermocouple reading and the actual temperature of the surrounding air, the bias error was found to be ± 0.52 °C