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RESEARCH ARTICLE

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A review of heat and fluid flow characteristics in microchannel heat sinks

Turgay Coskun | **Erdal Cetkin**

Department of Mechanical Engineering, Izmir Institute of Technology, Izmir, Turkey

Correspondence
Erdal Cetkin, Department of Mechanical Engineering, Izmir Institute of Technology, 35430 Izmir, Turkey.
Email: erdalcetkin@gmail.com

Abstract

Heat transfer and flow characteristic in microchannel heat sinks (MCHS) are extensively studied in the literature due to high heat transfer rate capability by increased heat transfer surface area relative to the macroscale heat sinks. However, heat transfer and fluid flow characteristics in MCHS differ from conventional ones because of the scaling effects. This review summarizes the studies that are mainly based on heat transfer and fluid flow characteristic in MCHS. There is no consistency among the published results; however, everyone agrees on that there is no new physical phenomenon in microscale that does not exist at macroscale. Only difference between them is that the effect of some physical phenomena such as viscous dissipation, axial heat conduction, entrance effect, rarefaction, and so forth, is negligibly small at macroscale, whereas it is not at microscale. The effect of these physical phenomena on the heat transfer and flow characteristics becomes significant with respect to specified conditions such as Reynolds number, Peclet number, hydraulic diameter, and heat transfer boundary conditions. Here, the literature was reviewed to document when these physical phenomena become significant and insignificant.

KEYWORDS

axial heat conduction, continuum, entrance effect, microchannel heat sink, viscous dissipation

1 | INTRODUCTION

The size of electronic components decreases due to the trend of miniaturized and lightweight component trend, whereas computational and heat generation requirements increase. Therefore, the volumetric heat generation rate in electronic components increases steadily, and thermal management is vital due to capability of eliminating possible failures. Microchannel heat sinks (MCHS) are capable of enhancing heat transfer rate due to their high surface to volume ratio relative to the conventional heat sinks.[1-6] Tuckerman and Pease demonstrated that heat transfer rate in MCHS (up to 790 W/cm^2) are significantly greater than macroscale heat sinks (20 W/cm^2).[7] However, increase in the heat transfer surface area yields the penalty of increased pressure drop.[1,2,6,8-13] The feasibility of MCHS can be uncovered by finding a maximized heat transfer rate for acceptable pressure drop values.

In the literature, heat sinks are classified with respect to their channel length scales such as macro, mini, micro, and nano. Tuckerman and Pease[1] stated that a channel can be classified as microchannel if the hydraulic diameter range is in between 10 and 200 μm. Table 1 shows the classification of heat exchangers based on their channel sizes.[14,15] Gad-el-Hak[16] and Morini[17] also stated a device can be categorized as microdevice when its hydraulic diameter is between 1 μm and 1 mm.

Microchannel heat exchangers (HEX), that is, heat sinks, have been utilized in numerous fields of engineering applications, such as electronics, automotive industry, fuel cells, air conditioning, and some of them are shown in Figure 1.

A simple structure of the rectangular shape of MCHS is shown in Figure 2.[13] It is constructed with many channels (having characteristic dimensions of the order of micrometers) arranged parallel to each other and each channel contact with the boundaries on which heating load exists. Heat transfer in MCHS is carried out in two ways: conduction and convection. First, the heat sink (generally made from a high thermal conductivity material) absorbs the dissipated heat by conduction. After that, the absorbed heat emitted by the coolant (usually liquid) via forced convection. Hence, conjugate heat transfer is generally pronounced in the study of heat transfer in MCHS.

There are a number of publications on heat transfer characteristics of microscale heat sinks, and their performance comparison relative to the conventional (macro) length scales. However, the results of the publications are not consistent. Some studies show that heat transfer coefficient and pressure drop values can be calculated by using macroscale approach.[18,20-27] However, others state that is not the case.[28-33] There are various claims why there may be a mismatch in between micro- and macroscale calculations such as scaling effect, fluid viscosity, variable thermophysical properties, entrance effect, and conjugate heat transfer.[2,34-39] In addition, the effect of surface roughness,[2,30,40,41] electrical double layer (EDL),[42-45] axial heat

TABLE 1 Classification of heat exchangers (HEX) with respect to hydraulic diameter

Mehendale et al[14]		Kandlikar and Grande[15]	
Micro-HEX	$1\,\mu m < d_h \leq 100\,\mu m$	Transitional channel	$0.1\,\mu m < d_h \leq 10\,\mu m$
Macro-HEX	$100\,\mu m < d_h \leq 1\,mm$	Microchannel	$10\,\mu m < d_h \leq 200\,\mu m$
Compact HEX	$1\,mm < d_h \leq 6\,mm$	Minichannel	$200\,\mu m < d_h \leq 3\,mm$
Conventional HEX	$6\,mm < d_h$	Conventional channel	$3\,mm < d_h$

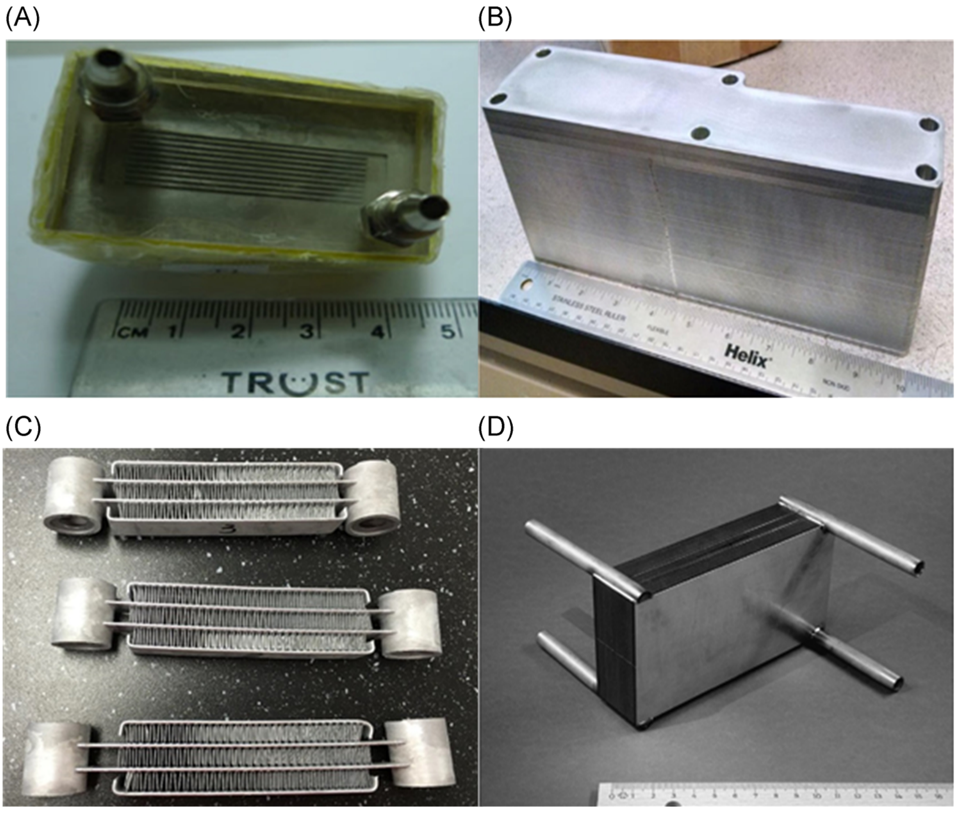

FIGURE 1 Microchannel HEXs utilized in (A) electronic cooling,[10] (B) natural gas cooling in automotive industry,[18] (C) air conditioning,[19] and (D) fuel cells.[20] HEX, heat exchangers [Color figure can be viewed at wileyonlinelibrary.com]

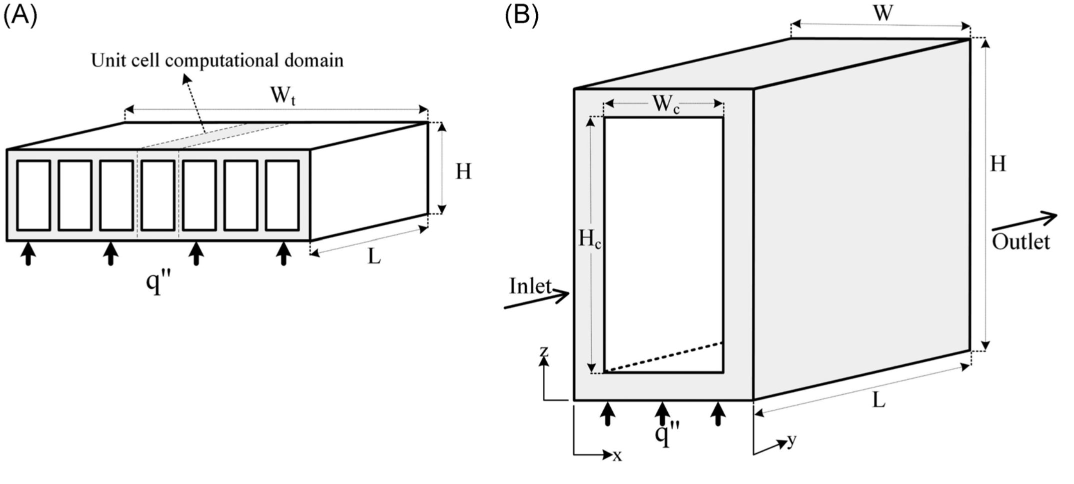

FIGURE 2 A simple structure of MCHS: (A) stacked model and (B) computational domain.[13] MCHS, microchannel heat sinks

conduction,[2,46,47] aspect ratio,[27,48,49] and viscous dissipation[2,50] are addressed as the possible reasons for the deviation from classical theory.

The documented critical Reynolds number values, which indicate transition from laminar to turbulent flow, vary also greatly in the literature.[28] Furthermore, the flow in MCHS is generally laminar due to relatively small hydraulic diameter comparison to the conventional scales.[26]

Here, the microchannel heat sink literature was reviewed with the focus of heat transfer and pressure drop characteristics. In addition, the aim of this paper is to document the significance of physical phenomena on heat transfer and pressure drop at microscale. Therefore, this review uncovers which physical phenomena should be considered or neglected at microscale when defining for the heat transfer and pressure drop calculations with respect to specified conditions such as Reynolds number, Peclet number, and hydraulic diameter.

2 | GENERAL OVERVIEW OF MCHS LITERATURE

There are numerous studies uncovering heat transfer and pressure drop characteristic of MCHS with analytical, numerical, and experimental methods. Experimental studies are dominant in the literature between 1980s and 2000.[50] There is no consistency between the results of microscale studies and predicted conventional length scale values in the published documents during this era. After year 2000, numerical models began to be used commonly in MCHS research; and variation between the results was diminished gradually by incorporating physical phenomena, which can be ignored at macroscales, such as axial heat conduction, viscous dissipation, surface roughness, rarefaction, EDL effect, and so forth.

Various channel geometries were studied in the literature such as circular, rectangular, trapezoidal, triangular, hexagonal, and many more.[51-56] Perret et al[51] numerically investigated the effect of channel shape on thermal resistance. They concluded that thermal resistance is smaller with rectangular microchannels in comparison to diamond- and hexagon-shaped channels. In addition, experimental and numerical studies with laminar and turbulent flows are documented in the literature. However, the majority of the cases are laminar because of relatively small hydraulic diameters in microchannels. Water and methanol are two of the most common liquids discussed in the literature. Generally, transition from laminar to turbulent flow occurs earlier than in conventional devices with macrolength scales ($Re_{cr} = 2300$).[2,33,57] Some of the obtained critical Reynolds number values for MCHS are given in Table 2.

TABLE 2 Transition from laminar to turbulent flow for MCHS

Study	Critical Reynolds number
Wu and Little[58]	400-900
Peng and Peterson[57]	1000
Harms et al[59]	1500
Yuan et al[33]	1500
Li et al[60]	1700

Abbreviation: MCHS, microchannel heat sinks.

MCHS literature can be categorized with respect to heat transfer mechanisms, fluid flow characteristics, and solution methods. The schematic representation of how the studies can be categorized is shown in Figure 3.

2.1 | Governing equations and boundary conditions for rectangular MCHS

In the literature, most of the studies treated flow (especially for liquids) as a continuum medium at microscale and utilized conventional correlations to solve continuity, momentum, and energy equations. Following assumptions are commonly used in the mathematical modeling of MCHS: (a) the flow is steady, single-phase, incompressible, and laminar; (b) constant thermophysical properties for both solid and liquid; (c) no gravitational force; and (d) no radiation and natural convection.[6,55,56,61-66] After that simplifications, conservation of mass, momentum, and energy equations become

$$\nabla \cdot \vec{v} = 0, \tag{1}$$

$$\rho_f (\vec{v} \cdot \nabla \vec{v}) = -\nabla P + \mu_f \nabla^2 \vec{v}, \tag{2}$$

$$\rho_f c_{p,f} (\vec{v} \cdot \nabla T) = k_f \nabla^2 T, \tag{3}$$

where ρ_f, $c_{p,f}$, k_f, and μ_f are the density, specific heat, thermal conductivity, and kinematic viscosity of the fluid, respectively. $\vec{v}$ is the velocity vector in the fluid domain.

To represent EDL effect, a body force term is added to the momentum equation and Equation (2) becomes

$$\rho_f (\vec{v} \cdot \nabla \vec{v}) = -\nabla P + \boldsymbol{F} + \mu_f \nabla^2 \vec{v}, \tag{4}$$

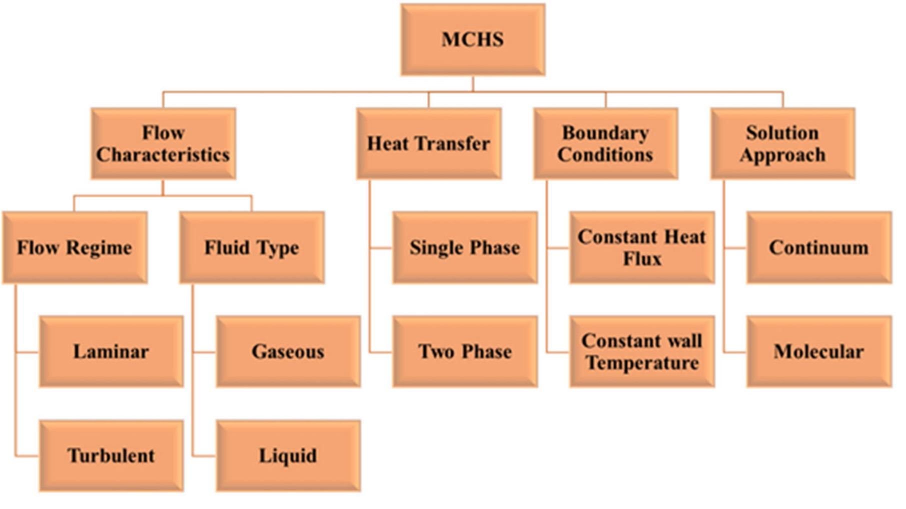

FIGURE 3 Schematic representation of MCHS categorization. MCHS, microchannel heat sinks [Color figure can be viewed at wileyonlinelibrary.com]

where $\boldsymbol{F}$ can be defined as the multiplication of induced electric field and net charge density.[43]

Viscous dissipation effect on the heat transfer at microscale is represented by a term Q_{vd} in the energy equation and Equation (3) becomes

$$\rho_f c_{p,f} (\vec{v} \cdot \nabla T) = k_f \nabla^2 T + Q_{vd}. \tag{5}$$

For solid regions, the energy equation reduces to

$$k_s \nabla^2 T = 0, \tag{6}$$

where k_s is the thermal conductivity in the solid domain (ie, the thermal conductivity of solid channel).

2.2 | Boundary conditions

The literature includes distinct boundary conditions; here, the most general boundary conditions are documented to give an insight. The coolant is driven by the pressure difference in between the inlet and outlet surfaces of the microchannel shown in Figure 2. The temperature and velocity of the coolant at the inlet boundary is generally described as

$$T = T_{\text{in}} \text{ and } V = V_{\text{in}}. \tag{8}$$

The outlet boundary is defined as pressure outlet and temperature gradient is equal to zero in any coordinate system at the outlet[52,63]

$$P = P_{\text{out}} \text{ and } \partial T / \partial n = 0. \tag{9}$$

At the left and right side of the domain, symmetry boundary conditions exist

$$\partial T / \partial x = 0. \tag{10}$$

The boundaries of the microchannel surrounded by the solid surface is defined as no slip wall boundaries with stationary wall

$$u = v = w = 0. \tag{11}$$

Generally, at the bottom wall uniform heat flux is applied

$$q'' = k_s \frac{\partial T}{\partial y}. \tag{12}$$

The remaining outside walls of the solid domain surrounding the microchannel are adiabatic. The continuity of energy at the interfaces of solid and fluid surfaces satisfy

$$k_s \frac{\partial T}{\partial n}\bigg|_{\text{wall}} = k_f \frac{\partial T}{\partial n}\bigg|_{\text{wall}}. \tag{13}$$

3 | SCALING EFFECTS IN MCHS

In the continuum approach (macroscale), some physical phenomena can be ignored while evaluating heat transfer and pressure drop characteristics. However, as length scale decreases from macro to micro, effect of the neglected physical phenomena become essential, and, therefore, they needed to be considered to evaluate heat transfer and flow characteristics accurately. Validity of continuum approach in microscale needed to be carefully investigated, especially for gaseous flow.[9] Knudsen number (*Kn*), (ratio of free molecular path over characteristic flow dimension), is utilized to characterize the flow regime of gaseous flow in MCHS. The flow is treated as continuum if *Kn* number is smaller than 0.001. Continuum assumption cannot be used as *Kn* number becomes greater than 0.001 and rarefaction effect becomes significant on heat transfer for gaseous flow.[67,68] Researchers[69-71] indicated that *Nu* number decreases as the rarefaction effect increases in MCHS. Flow regimes with respect to *Kn* number and the main solution approaches are listed in Table 3.[9,72,73] In slip flow regime, temperature and velocity profiles differ from continuum regime because of slip velocity and temperature jump.[57]

3.1 | Surface roughness

Relative surface roughness can be defined as the ratio between the roughness of the surface material (ϵ) and hydraulic diameter (d_h) of a microchannel. In macroscale (conventional) channels, the importance of surface roughness pronounced only in turbulent regime.[36,39] However, it is essential for microchannels even in laminar region[37]; although it is a debated topic in the literature.[2,30,31,33,39,40,74-83] Surface roughness along the microchannel[80] is schematically shown in Figure 4. ε is the roughness height, and the effect of surface roughness increases as the height increases.

Steinke and Kandlikar[39] stated that researchers should consider surface roughness effect in their studies. Most of the studies state that Nusselt number[2,33,75,79] and friction

TABLE 3 Flow regimes for gaseous flow at various *Kn* number[9,72,73]

Knudsen number	Flow regime	Solution method
$Kn \to 0$	Continuum (no molecular diffusion)	Euler equations with slip-BCs
$Kn \leq 0.001$	Continuum (with molecular diffusion)	NS equations with no-slip-BCs
$0.001 < Kn \leq 0.1$	Slip flow (slightly rarefied)	NS equations with slip-BCs
$0.1 < Kn \leq 10$	Transient flow (moderately rarefied)	Burnett equations with slip-BCs Moment equations Lattice Boltzmann Direct simulation Monte Carlo
$Kn > 10$	Free molecular flow	Collisionless Boltzmann Direct simulation Monte Carlo

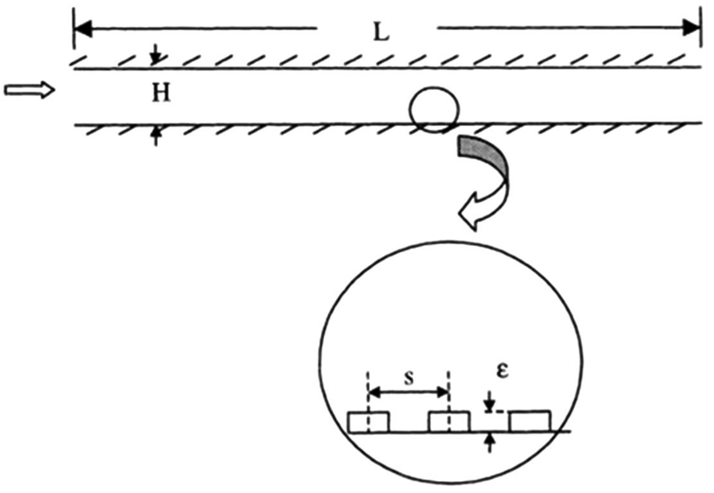

FIGURE 4 Schematic representation of the surface roughness in the microchannel[80]

factor[2,30,33,75,76,79,83] increase as surface roughness increases. Guo and Li[2] state that early transition from laminar to turbulent flow in MCHS is due to surface roughness. Some researchers[2,30,74] also state that the distinction in between experimental and theoretical values are due to surface roughness. Liu et al[79] stated that the effect of surface roughness cannot be neglected when Reynolds number is greater than 1500. In addition, Dai et al[84] stated that the roughness effect on friction factor and critical *Re* number need to be considered at high the relative surface roughness values (>1%). Moreover, researchers state that at low *Kn* number values (<0.02), the effect of surface roughness on friction factor becomes greater.[81] Kandlikar et al[82] experimentally investigated the effect of surface roughness on friction factor and heat transfer rate using two pipes with different inner diameters: 0.62 and 1.067 mm. They document that for 1.067-mm pipe diameter, the effect of surface roughness on heat transfer and pressure drop is insignificant when compared to 0.62-mm pipe diameter. Zhang et al[85] stated that *Nu* number and *Po* number in rough microchannels are not only related to shape of cross section of the channel but also related to the *Re* number of liquid flows. However, Croce and D'Agaro[76] indicated that the effect of surface roughness on heat transfer is insignificant (within experimental error limits) and it highly depends on tube geometry. In addition to that, some of the researchers stated that overall thermal performance of a HEX does not change with surface roughness.[33,79] Furthermore, Pelevic and Meer[86] indicated that surface roughness has minor effect (only 4% increase in heat transfer for 2.93% relative roughness) on heat transfer enhancement in their numerical study.

3.2 | Electrical double layer

EDL is related to electrostatic surface charge on the heat transfer surfaces, and it is formed on the heat transfer wall surfaces as reformation of charges on the solid surface and balancing charges in the liquid. It affects heat transfer and fluid flow due to interaction between the solid surfaces and aqueous solution.[9,42] EDL effect is essential for flow of liquids in microchannels.[42,45,87] Schematic representation of the EDL at the channel wall[80] is shown in Figure 5. ξ and ψ are electric potential at the boundary between the diffuse double layer and the compact layer (zeta potential) and electrostatic potential at any point in the electric double layer, respectively.

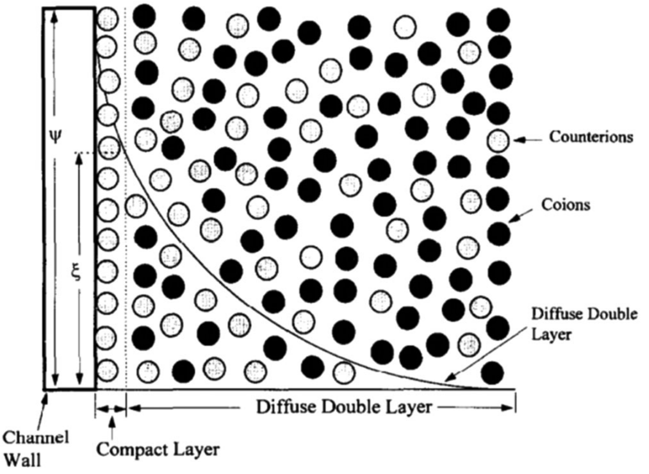

FIGURE 5 Schematic view of the EDL at the channel wall.[41,88] EDL, electrical double layer

Ng and Tan[45] stated that EDL effect decreases the effectiveness (actual heat transfer rate over maximum possible heat transfer rate) of the microchannel. They define EDL effect as a body force in the z-direction momentum equation.[44,87] Ren et al[89,90] and Li[91] showed that EDL effect increases pressure drop in microchannels. Mala et al[42] state that the heat transfer rate in microchannels decreases because of EDL effect, and it is dominant at the vicinity of walls. The literature documents that friction factor increase[42,43,92] and Nu number decrease due to EDL.[42,43] The mean increment in the friction factor and decrement in Nu number was recorded as 17% and 8%, respectively, in the study of Yang et al[43] for 200 mV zeta potentials and 10^{-8} M ionic concentration.

3.3 | Axial heat conduction

Maranzana et al[46] numerically and theoretically studied the effect of axial heat conduction on heat transfer of MCHS. They stated that the efficiency of heat exchanger is reduced in the presence of axial heat conduction. They point out that axial heat conduction is the reason why numerical and theoretical results do not match. They suggested that axial heat conduction needed to be defined during the numerical solution phase to overcome the mismatch. In addition, they defined a new dimensionless quantity, M axial heat conduction number. They indicated that under specific conditions ($M < 0.01$), the effect of axial heat conduction can be neglected. However, Lin et al[93] and Zhang et al[94] stated that M can be inadequate to judge whether the effect of the axial heat conduction on heat transfer if the uniform outside wall temperature boundary condition was existing. Moreover, Hetsroni et al[95] stated that the effect of axial heat conduction can be ignored when $Re < 150$ and $M = 0.01$. However, axial heat conduction should be considered when $Re > 150$ and $M > 0.01$. In addition, the effect of axial heat conduction is neglected in the studies of Cole and Cetin[96] and Yu et al[97] for high Pe number values ($Pe > 100$). Cole and Cetin[96] also stated that the thermal conductivity of the wall should be lower than the thermal conductivity of the fluid to neglect axial conduction. Furthermore, Barisik et al[98] stated that the effect of axial heat conduction can be ignored in pipe flow even for relatively small Peclet number values ($Pe < 100$) with the existence of viscous dissipation. The effect of axial heat conduction changes with respect to Knudsen number, Peclet number, thermal conductivity, and thickness of the separating wall.[99] An increment in *Kn* number, *Pe* number, and hydraulic diameter decreases axial heat conduction rate.[71,100-102] In contrast, the effect of axial heat conduction on heat transfer increases as *Re* number, thermal conductivity, and thickness of the separating wall increase.[96,102] At the entrance region, the presence of axial heat conduction increases heat transfer rate.[47,101] However, axial heat conduction result in the reduction in *Nu* number in fully developed region.[47] Lin and Kandlikar[103] stated that the effect of axial conduction can be neglected if the conductivity of channel material is lower than thermal conductivity of the fluid as indicated in the study of Cole and Cetin.[96] In addition, Kakac et al[104] proved that the effect of axial heat conduction can be neglected when *Kn* is number greater than 0.1 and temperature difference between the wall and fluid exceeds 1.667 ($\varkappa = 1.667$). In Figure 6, the effect of axial heat conduction on the heat flux distribution along the wall is represented schematically.[99] The figure shows that axial heat conduction yields heat flux distribution to vary along the solid-fluid interface even though applied heat flux to the solid region is uniform.

3.4 | Aspect ratio

Channel aspect ratio definition varies the literature. Generally accepted definition is the division of channel height to channel width. The orientation of the channel is not crucial for fluid flow due to fixed cross-sectional area.[39] However, channel orientation cannot be ignored for heat transfer and it is critical to define heat transfer boundary conditions.[28,36,56,105-107] In the rectangular cross sections, Nusselt number depends on the aspect ratio (α), and Nu increases from square channels ($\alpha = 1$) to deep rectangular channels (for parallel plates, $\alpha = 0$).[36] Zhimin and Fah[106] numerically studied the optimization of rectangular microchannels. They stated that channel aspect ratio in laminar flow region must be as high as possible to obtain minimum thermal resistance. They also indicated that the lowest thermal resistance can be achieved in turbulent region; however, this is not preferred because of high pumping power requirements. Furthermore, numerically showed that low thermal resistance (<0.1 W/mK) and high pressure drop (>250 kPa) in microchannels are obtained for high aspect ratio (20.333) and small hydraulic diameter (0.172). In addition, they concluded that the rectangular MCHS with the aspect ratio range of 8.904 to 11.442 have the best performance in terms of heat transfer and pressure drop. The change in friction factor with respect to aspect ratio was studied by Sahar

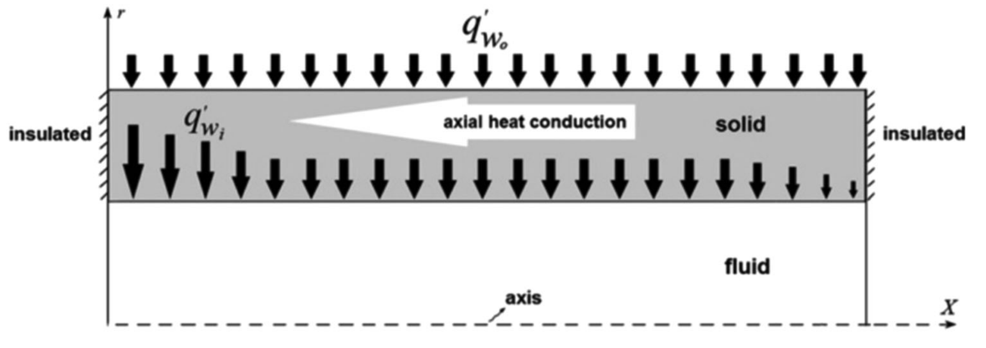

FIGURE 6 Schematic representation of the effect of axial heat conduction on the wall heat transfer per unit length[99]

et al[108] and Kim and Kim.[109] According to them, friction factor decreases as aspect ratio increases from 1 to 2, then it starts to increase as aspect ratio keeps increasing.

3.5 | Viscous dissipation

Pressure drop in MCHS is relatively greater than macroscale ones. Therefore, viscous heating effect cannot be neglected as in macroscale heat sinks. Brinkman (*Br*) number is the ratio of viscous heating over heat conduction in the fluid domain which flows along microchannel, and it is used to determine the effect of viscous dissipation on heat transfer mechanism in MCHS.[110] Kalyoncu and Barisik[71] numerically showed that viscous dissipation increases and decreases heat transfer rate for positive and negative *Br* cases, respectively. However, they indicated that developed *Nu* number values are the same for both cases. Morini[111] stated that decrease in friction factor as *Re* number increases can be explained with the reduction of viscosity because of viscous heating effect in MCHS. He also stated that viscous dissipation effect becomes significant for liquid flow when the hydraulic diameter is smaller than 100 μm. In addition, Koo and Kleinstreuer[112] documented that viscous dissipation should not be ignored for small size channels ($d_h < 50\,\mu m$). Chen[113] stated that *Nu* number increases in the fully developed region with viscous dissipation when the value of *Kn* number is small (<0.03). In contrast, Morini and Spiga[114] numerically indicated that heat transfer in MCHS is adversely affected by viscous heating due to reduction in *Nu* number. In addition, Mukherjee et al[115] stated that *Nu* number is inversely proportional to the *Br* number for constant heat flux boundary condition. Furthermore, Zhai et al[116] stated that the effect of viscous dissipation on temperature rise is insignificant when compared to convective heat transfer for water flow. In addition, they also stated that flow distribution and temperature field is affected by the length of entrance region. To eliminate entrance effect and obtain more uniform flow field, entrance length should be as long as possible. Morini[117] stated that at high *Re* number (>1000) entrance effect with viscous dissipation needed to be considered to define flow characteristic and temperature field correctly. In addition, Fani et al[118] stated that the effect of viscous dissipation becomes more important with increasing the *Re* number and volume fraction.

4 | UNCERTAINTIES IN EXPERIMENTAL STUDIES AND SUMMARY OF THE LITERATURE

The results of experimental studies in the literature are not consistent. There are three main reasons behind the different results from the experimental studies: (a) fabrication of microchannels[19,35]; (b) errors in measurement; and (c) misalignments in the experimental setups. Pfund et al[119] raise awareness to the uncertainties in the experimental studies, and they stated that to understand phenomena in microlevel correctly, experiments should be done precisely. Some researchers stated that experimental results are in good agreement with theoretical results, and conventional correlations can be used in microscales.[114-117] Missaggia et al[120] experimentally confirmed the utilization of microchannels in the cooling of a laser diode which dissipates 500 W/cm^2 heat flux. They also stated that experimental values are in good agreement with the theoretical calculations.

To sum up, the effect of some physical phenomena (ignored at macroscale) on heat transfer and fluid flow becomes significant in microscale studies. To evaluate heat transfer and fluid flow characteristics accurately, these phenomena should be considered. Table 4 documents the specification of the research papers based on the methodology, geometry, flow characteristics, and so forth. Furthermore, Table 5 summarizes which research papers focus on which specific phenomena such as surface roughness, EDL, and so forth.

5 | CONCLUSION

The literature documents heat and fluid flow characteristic in microscale heat sinks. According to the literature, heat and fluid flow characteristics are not fully understood and there is inconsistency between documents. To better understand heat and fluid flow characteristics at microscale levels, following should be considered.

1. There is no new physical phenomenon in microscale literature. Only difference between the macroscale is that the effect of surface roughness, EDL, axial heat conduction, aspect ratio, rarefaction effect, viscous dissipation, and so forth, are negligibly small in macroscales.
2. Experimental and numerical research is suggested to be conducted simultaneously to increase the accuracy of the results. Viscous dissipation, axial heat conduction, and rarefaction (for gaseous flow) effects should be considered while evaluating heat transfer and fluid flow in MCHS.
3. Continuum assumption in microscale is validated for liquid flow and generally problematic for gaseous flow. If *Kn* number is lower than 0.001, the gaseous flow can also be treated as continuum.
4. Viscous dissipation can increase or decrease heat transfer depending on *Br* number (heating or cooling). The friction factor reduces due to a decrease in apparent viscosity as temperature increases in the heating case.
5. The effect of axial heat conduction in MCHS becomes negligible when $\mathrm{Pe} < 100$.
6. The effect of surface roughness on heat transfer is generally insignificant; however, the change in friction factor with surface roughness is remarkable according to the literature. Surface roughness effect on the friction factor and heat transfer is generally pronounced with rarefaction effect for gaseous flow, and it becomes significant at low *Kn* number values.
7. EDL affects liquid flow in microchannels and creates resistance to the fluid flow at the vicinity of the solid wall, that is, rise in apparent viscosity. This increases pumping power requirements in MCHS.

Overall, geometrical parameters should be defined as certain as possible and manufacturing constraints needed to be defined clearly before the experiments. When comparing the results of experimental studies, all of the assumptions should be in agreement with the methodology. The literature shows that there is disagreement in MCHS literature, and the physics of fluid flow should be focused more. To increase consistency between the literature and to understand the physical phenomena at microscale accurately, experimental and numerical studies should consider all physical phenomena in their studies and state all assumptions and criteria to justify the reason of why some phenomena are neglected if applicable. Thus, would enable the audience to compare the microscale heat transfer literature and realize the effect of each physical phenomena on heat transfer and pressure drop accurately.

TABLE 4 Experimental and numerical studies in the literature

Study	Type	Geometry	Flow type	Coolant	d_h, µm	Heat flux, W/cm^2	Pressure drop, kPa	*Re* number
Tuckerman and Pease[1]	Exp.	R	L	H_2O	92-96	790	206	
Missaggia et al[120]	Exp.	R	T	H_2O	160	500	482	2000
Peng et al[29]	Exp.	R	...	CH_3OH	311-646			
Peng and Peterson[57]	Exp.	R	L/T	H_2O/CH_3OH	311-746			300-1000
Peng and Peterson[28]	Exp.	R	L/T	H_2O	133-367			
Harms et al[59]	Exp.	R	L	Deion. H_2O				173-12900
Qu et al[121]	Exp./Num.	Trap.	...	H_2O	62-169			
Weilin et al[30]	Exp.	Trap.	...	H_2O	51-169			
Xu et al[35]	Exp.	...	...		30-344			20-4000
Pfund et al[119]	Exp.	R	...	H_2O	128-521			60-3450
Qu and Mudawar[122]	Exp./Num.	R	L	H_2O	348-592	100-200	5-86	139-1672
Judy et al[21]	Exp.	C/S	...	H_2O/CH_3OH	15-150			8-2300
Gao et al[123]	Exp.	...	...	H_2O				
Lee et al[20]	Exp.	R	L	H_2O	55-85	109	55	
Wu and Cheng[75]	Exp.	Trap.	L					
Lee et al[26]	Exp.	R	...	H_2O	318-903			300-3500
Zhang et al[124]	Exp.	R	...	H_2O				
Shen et al[40]	Exp.	R	...	H_2O	300-800			162-1257
Park and Punch[125]	Exp.	R	L		106-307			69-800
Turgay and Yazicioglu[77]	Num.	R	L					
Gunnasegaran et al[52]	Num.	...	L	H_2O				100-1000

(Continues)

TABLE 4 (Continued)

Study	Type	Geometry	Flow type	Coolant	d_h, µm	Heat flux, W/cm^2	Pressure drop, kPa	*Re* number
Ma et al[126]	Exp.	R	L/T	H_2O		1.37-5.78		
Reyes et al[127]	Exp.	Square	L	H_2O	500			416-2600
Chiu et al[107]	Exp./Num.	R	...	Liquid				
Sharma et al[25]	Exp.	R	...	H_2O	610			375-2000
Huang et al[47]	Exp.	R	...	H_2O	102.3			
Hasan et al[102]	Num.	Tr.	L					
Liu et al[79]	Exp.	Square	...	Air	400		5-30	200-2100
Kim[48]	Exp.	R	L	H_2O	155-580	2.1		30-2500
Sahar et al[38]	Exp./Num.	R	...	H_2O -R134a				
Wang et al[56]	Num.	...	...					
Khan and Kim[54]	Num.	...	...	H_2O				
Sahar et al[108]	Num.	R	...					
Zhai et al[116]	Exp./Num.	R	...	H_2O				200-1000
Zunaid et al[128]	Num.	R	L	H_2O	348.9		5-35	200-1000
Raghuraman et al[129]	Num.	R	...	H_2O				
Hajmohammadi et al[130]	Num.	R	L	H2O		100	50	
Samal and Moharana[105]	Num.	S	L	H_2O	400	5-20		50-200
Zhang et al[85]	Exp.	C	L			6-13		800-1600
Tiwari and Moharana[131]	Num.	C	L	H2O	400			100-500
Rehman et al[132]	Num.	R	L	H2O	411	50-100		100-1000

Abbreviations: C, circular; L, laminar, R, rectangular; S, square; Trap., trapezoid, Tr., triangular; T, turbulent.

TABLE 5 Distribution of the studies with respect to the topics

Study	Coolant	Surface roughness	Electrical double layer	Axial heat conduction	Aspect ratio	Viscous dissipation	Rarefaction effect
Guo and Li[2]	Gas	✓					✓
Weilin et al[30]	Liquid	✓					
Zhou and Yao[32]	Liquid	✓					
Yuan et al[33]	...	✓					
Shen et al[40]	Liquid	✓			✓		
Mala and Li[41]	Liquid	✓					
Mala et al[42]	Liquid		✓				
Yang et al[43]	Liquid		✓				
Ng and Poh[44]	Liquid		✓				
Ng and Tan[45]	...		✓				
Maranzana et al[46]	...			✓			
Huang et al[47]	Liquid			✓			
Kim[48]	Liquid				✓		
Wang et al[56]	Liquid				✓		
Kalyoncu and Barisik[71]	Gas			✓		✓	✓
Wu and Cheng[75]	...	✓					
Croce and D'Agaro[76]	...	✓					
Turgay and Yazicioglu[77]	...	✓		✓		✓	✓
Rovenskaya[78]	Gas	✓					✓
Liu et al[79]	Air	✓					

(Continues)

TABLE 5 (Continued)

Study	Coolant	Surface roughness	Electrical double layer	Axial heat conduction	Aspect ratio	Viscous dissipation	Rarefaction effect
Sun and Faghri[80]	Gas	✓					✓
Kleinstreuer and Koo[81]	Liquid	✓					
Kandlikar et al[82]	Liquid	✓					
Tan and Ng[87]	...		✓				
Ren et al[89]	Liquid		✓				
Ren et al[90]	Liquid		✓				
Yang and Li[92]	Liquid		✓				
Cole and Cetin[96]	Liquid			✓			
Barisik et al[98]	Gas			✓		✓	✓
Rahimi and Mehryer[99]			✓				
Hadjiconstantinou and Simek[100]	Gas			✓			
Lin et al[101]	...			✓			
Hasan et al[102]	...			✓			
Lin and Kandlikar[103]	...			✓			
Kakac et al[104]	Gas			✓		✓	✓
Zhimin and Fah[106]	...				✓		
Chiu et al[107]	Liquid				✓		
Sahar et al[108]	Liquid				✓		
Kim and Kim [109]	...				✓		

TABLE 5 (Continued)

Study	Coolant	Surface roughness	Electrical double layer	Axial heat conduction	Aspect ratio	Viscous dissipation	Rarefaction effect
Cetin et al[110]	...			✓		✓	✓
Morini[111]	...					✓	
Koo and Kleinstreuer[112]	Liquid					✓	
Morini and Spiga[114]	Liquid				✓	✓	
Chen[113]	...					✓	
Mukherjee et al[115]	...					✓	
Morini[117]	Liquid					✓	
Qu et al[121]	Liquid	✓					
Raghuraman et al[129]	Liquid				✓		
Ng and Poh[133]	Liquid		✓		✓		
Vainshtein and Gutfinger[134]	Liquid		✓				
Tardu[135]	Liquid		✓				
Ren and Li[136]	Liquid		✓				
Ban et al[137]	Liquid		✓				
Renksizbulut et al[138]	Gas				✓		✓
Liu et al[139]	Liq./Gas			✓			
Cetin et al[140]	...			✓		✓	✓
Turner et al[141]	Gas	✓					✓
Bahrami et al[142]	Gas					✓	✓
Koo and Kleinstreuer[143]	Liquid	✓					

(Continues)

TABLE 5 (Continued)

Study	Coolant	Surface roughness	Electrical double layer	Axial heat conduction	Aspect ratio	Viscous dissipation	Rarefaction effect
Cao et al[144]	Gas	✓					✓
Adham et al[145]	Gas				✓		
Chen et al[146]	Liquid	✓					
Mishan et al[147]	Liquid			✓			
Natrajan and Christensen[148]	Liquid	✓					
Niazmand et al[149]	Gas				✓		✓
Zhu and Liao[150]	Gas				✓		✓
Jeong and Jeong[151]	...			✓		✓	✓
Hettiarachchi et al[152]	...				✓		✓

Abbreviations: C, circular; L, laminar, R, rectangular; S, square; Trap., trapezoid, Tr., triangular; T, turbulent.

NOMENCLATURE

Br	Brinkman number
d_h	hydraulic diameter
EDL	electrical double layer
HEX	Heat exchanger
Kn	Knudson number
M	axial conduction number
MCHS	microchannel heat sink
Nu	Nusselt number
Pe	Peclet number
Po	Poiseuille Number
q'	heat transfer per unit length, W/m
Re	Reynolds number

GREEK SYMBOLS

ξ	zeta potential, V
ψ	electrostatic potential, V
$\varkappa$	temperature jump parameter
α	aspect ratio
ϵ	surface roughness
ε	roughness height

SUBSCRIPTS

cr	critical
wi	inside wall
wo	outside wall

ORCID

Erdal Cetkin https://orcid.org/0000-0003-3686-0208

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