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Microscale Thermal Transport and Electromechanical Microfluidic Actuation

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This paper summarizes recent advances in fundamental research on microscale thermal transport in phase-change processes and developments in novel microfluidic actuation methods using electromechanical effects. The research topics covered include thin-film transport, pool boiling heat transfer enhancement, convective flow boiling in microchannels, microscale ionic winds, micropumping using electrohydrodynamic effects including ion induction and dielectrophoresis, and electrowetting-based control of liquid droplet motion. Progress in microscale flow characterization and non-intrusive temperature measurement is also discussed.

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Key words: microscale; thermal transport; microfluidic actuation; electromechanical actuation; micropump
1. INTRODUCTION

Thermal transport and fluidic actuation at the microscale present exciting possibilities for advances in several emerging fields of research including active liquid cooling of microelectronics and biomedical applications. Microchannel heat sinks, for example, have been recognized as being among the most promising thermal management technologies for dissipating the ultra-high heat fluxes encountered in the electronics industry; they are also amenable to on-chip integration [Garimella et al., 2006; Garimella, 2006]. However, the complex thermal transport phenomena that occur at the microscale, especially during liquid–vapor phase change, remain incompletely understood. Recent advances in the fundamental understanding of microscale phase-change heat transfer, including thin-film evaporation, enhancement of pool boiling heat transfer, and flow boiling in microchannels are reviewed in this paper.

In addition, conventional pumping methods are rarely suitable for the compactness and efficiency required for microfluidic actuation. Novel means of microfluidic actuation using electromechanical effects are introduced, and include examples of microscale ionic winds, electrohydrodynamic (EHD) pumping, dielectrophoresis (DEP)-induced pumping, and electrowetting (EW)-based control of liquid droplet motion. Developments in microscale flow characterization and temperature measurement are also summarized. This paper focuses on work conducted in the authors’ group at Purdue University.

2. MICROSCALE THERMAL TRANSPORT

Microscale thermal transport in thin evaporating films, pool boiling on enhanced surfaces, and flow boiling in microchannels is examined here with an emphasis on the physics of phase-change heat transfer.

2.1 Thin-Film Transport

When liquid wets a solid wall, the extended meniscus can typically be divided into three regions as illustrated in Figure 1: an adsorbed or non-evaporating region where liquid is adsorbed on the wall; a thin-film or transition region where effects of long-range molecular forces are felt; and an intrinsic-meniscus region where capillary forces dominate. Of these, the thin-film region is characterized by the highest heat transfer rates because of the very low thermal resistance across the liquid film. The intensive heat transfer in the thin-film region is an important mechanism in bubble growth during boiling [Wilson et al., 1999], film dryout [Oron et al., 1996], and spray cooling [Grissom & Wierum, 1981]. Thin-film evaporation occurs in a variety of phase-change systems including heat pipes, thermosyphons, vapor chambers, two-phase cold plates, and capillary pumped loops. The liquid pressure gradient due to interfacial and viscous forces, along with the pressure and temperature differences at
the liquid–vapor interface, control the phase-change heat transfer process in these microscale systems. In capillary structures, thin-film evaporation is reported to be a function of capillary flow, pore size, and thermal conductivity of the medium. But the exact nature of the relationship is not known and requires detailed experimental and analytical investigation to establish a functional relationship. A thorough understanding of the physical mechanisms occurring in these devices is critical for miniaturization of heat pipes, cold-plates, and other thermal solutions.

2.1.1 Microscale evaporating thin-film model

Porous or grooved surfaces can help to create thin films for enhanced operation of heat transport devices such as heat pipes and capillary pumped loops. The pressure and temperature distribution, the thin-film profile, and the heat and mass transfer in the three regions of the meniscus have been studied extensively in the literature [Deryagin et al., 1965; Deryagin, 1987; Potash & Wayner, 1972; Wayner et al., 1976; Schonberg & Wayner, 1990; Hallinan et al., 1994; DasGupta et al., 1993; Park et al., 2003; Wee, 2004; Wee et al., 2006; Freund, 2005; Stephan & Busse, 1992; Xu & Carey, 1990; Ma & Peterson, 1997; Morris, 2003; Chakraborty & Som, 2005].

Recently, an evaporating meniscus in a microchannel was investigated through an augmented Young–Laplace model and the kinetic theory-based expression for mass transport across a liquid–vapor interface [Wang et al., 2007]. The complete expression for mass transport was employed without any approximations, and with appropriate boundary conditions for the film profile. The thin-film and the intrinsic-meniscus regions were distinguished based on the disjoining pressure variation along the meniscus. The unique features of this model include: 1) heat transfer in the thin-film and micro regions is quantitatively compared and the relative contributions of the two regions to the overall heat transfer are delineated; 2) the kinetic theory-based expression for mass transport across a liquid–vapor interface [Schrage, 1953] is used instead of a simplified version often adopted in prior studies; the simplified expression is shown in this work to underpredict the total heat transfer at high superheats; 3) the influence of capillary suppression and channel size on heat transfer from the thin-film region is explored in detail.

The variation of disjoining pressure ($P_d$), capillary pressure ($P_c$), and liquid pressure change ($ΔP_l = P_l - P_l (x = 0)$) along the length of the liquid (octane) film are plotted in Figure 2 for two different channel radii (also the asymptotic intrinsic-meniscus radius), $R^* = 2500$ nm and $R^* = 200$ nm. The evaporated liquid is replenished by the liquid pressure gradient. For the first 40 nm along the thin-film region, Figure 2a shows that the increase in liquid pressure is solely supported by the reduction in disjoining pressure. Beyond this length, the capillary pressure also decreases and begins to contribute to the increase in liquid pressure, which is consistent with the observations of DasGupta et al. [1993]. In Figure 2b, it is seen that the capillary pressure is maintained at a high level and does not drop in the case of the much smaller channel size. The drop in capillary pressure is important for flattening the film thickness profile and enhancing liquid pumping.

The meniscus profiles for different levels of superheat from 20 K to 0.01 K are compared in Figure 3. It is seen that the apparent contact angle grows with superheat. The decrease in disjoining pressure causes liquid to be pumped into the thin film. When the superheat is increased, evaporation is strengthened and more liquid needs to be pumped into the thin film, decreasing the disjoining pressure faster and finally resulting in a larger apparent contact angle, which is consistent with previous studies [Hallinan et al., 1994; DasGupta et al., 1993; Park et al., 2003; Wee, 2004].

The interfacial evaporation heat transfer coefficient $h_{lv}$ is plotted in Figure 4 for wall superheats of 0.1 K and 1 K. It is clear that $h_{lv}$ is suppressed in the beginning due to the disjoining pressure as has been extensively discussed in the literature [Grissom & Wierum, 1981; Deryagin et al., 1965; Deryagin, 1987; Potash & Wayner, 1972; Wayner et al., 1976; Schonberg & Wayner, 1990; Hallinan et al., 1994; DasGupta et al., 1993; Park et al., 2003; Wee, 2004; Wee et al., 2006]. After a peak at $x = 400$ nm, $h_{lv}$ is seen to decrease again, this time because of the suppression due to capillary pressure. If the suppression due to capillary pressure were ignored, $h_{lv}$ would follow the dashed line in the figure and approach a higher constant value of $1.04 × 10^6$ W/m$^2$K, which is consistent with published values of $h_{lv}$ for situations that do not encounter suppression effects due to capillary pressure [Stephan & Busse, 1992]. The drop in $h_{lv}$ is more significant for lower $T_{lv}$. If the wall superheat is increased from 0.1 K to 1 K, the interface temperature is greater, and the restraining effect becomes less sig-
FIGURE 2. Variation of the different pressure components along the length of the octane meniscus (superheat 1 K) for two channel radii: (a) $R^* = 2500$ nm and (b) $R^* = 200$ nm [Wang et al., 2007].

FIGURE 3. Effect of superheat on the film thickness profile ($R^* = 2500$ nm) [Wang et al., 2007].
significant, as shown in the figure. It may be noted that even though the capillary pressure approaches a constant value beyond $x \approx 400 \text{ nm}$, $h_l$ continues to decrease because of the continuous drop in the interface temperature $T_l$. As $T_l$ decreases with increasing film thickness, the restraining effect of $P_c$ increases. It was also found that, while heat transfer in the thin-film region is relatively insensitive to channel size for channels larger than a few micrometers in radius, that in the intrinsic meniscus is quite sensitive to channel size. Compared to the relatively small contribution to overall heat transfer from the thin-film region, the micro region (defined here as extending from the non-evaporating region to a location where the film is 1-μm thick) is found to account for more than 50% of the total heat transfer.

2.1.2 Convection near an evaporating meniscus

More intense evaporation is expected to occur near the film tips than at other locations along the meniscus because of the film curvature [Yiotis et al., 2003]. The differential evaporation, in combination with buoyancy effects, gives rise to the buoyant-thermocapillary flow in the meniscus [Kirdyashkin, 1984; Villers & Platten, 1992; Buffone et al., 2005]. Dhavaleswarapu et al. [2007] and Chamarthy et al. [2007] systematically investigated steady buoyant-thermocapillary convection induced by the differential evaporation of methanol near a meniscus in horizontally oriented capillaries of sizes varying over a wide range from 75 to 1575 μm ID, using epi-fluorescent micro-particle image velocimetry (μPIV). The problem under consideration is more complex than those discussed in the literature due to the curvature of the free liquid surface, combined with the phase change prevalent and the buoyancy effects in play. A quantitative investigation of the 3D convection patterns is conducted at sub-millimeter length scales; 3D resolution at this scale has not been achieved in past studies.

Figure 5 shows the flow fields near the evaporating meniscus along the horizontal center plane for all five tube diameters investigated. The flow fields are seen to be similar for all five tubes and include two counter-rotating symmetrical vortices; only one vortex is seen for the largest tube since only half the plane could be included in the measurements for this tube diameter. The maximum velocity location is observed to be at the interface as is expected of purely thermocapillary-driven flow, and is midway between the center of the meniscus and the contact line (seen clearly in Fig. 5e). Be-

FIGURE 4. Interface heat transfer coefficient along the meniscus. Heat transfer is suppressed by the disjoining pressure to the left of the peak and by capillary pressure to the right. Lower superheat results in stronger suppression ($R^* = 2500 \text{ nm}$) [Wang et al., 2007].
FIGURE 5. Velocity vectors with vorticity contours obtained at the horizontal center planes for different tubes [Dhavaleswarapu et al., 2007].

FIGURE 6. Long-exposure particle streak images at the vertical center planes for different tubes (g is the acceleration due to gravity) [Dhavaleswarapu et al., 2007].
cause of the intensive evaporation near the corners, the
local liquid–vapor interface temperature is less than
that at the meniscus center. Since for methanol (and
for most liquids), surface tension decreases with an in-
crease in temperature, the surface tension is higher
near the corners than at the center. This drives flow
towards the corners, establishing the two counter-rotat-
ing symmetrical vortices observed at the horizontal
center plane for all the tube sizes.

The relative influences of buoyancy and thermodi-
capillarity on the flow were characterized with long-
exposure particle streak images obtained along the
vertical center plane for different tube sizes, as shown
in Figure 6. As evaporation occurs at the interface, it
renders the meniscus region relatively cooler than the
bulk liquid away from the meniscus. This difference
in temperature gives rise to buoyancy effects which
grow in strength with tube diameter. For the smallest
diameter investigated (75 μm), buoyancy effects are
negligible and a symmetrical flow pattern is observed.
For the 200-μm tube, asymmetry sets in and the coun-
ter-clockwise bottom vortex is found to be larger
than the clockwise upper one indicating the presence
of an asymmetrical toroidal vortex. As buoyancy ef-
fects increase with length scale, the counter-clockwise
vortex grows in size with an increase in tube size as
seen in Figures 6c and 6d. It may be seen that the
flow is divided into two regions — an upper region
where the motion is induced by surface tension and a
lower one driven by density differences. It was con-
cluded that for an evaporating meniscus in capillary
tubes and for the experimental conditions, a critical di-
ameter exists such that for smaller diameters, the flow
is driven primarily by thermocapillarity. Buoyancy ef-
ects set in beyond this critical diameter leading to
three-dimensionalities in the flow.

2.2 Enhancement of Pool Boiling Heat Transfer

Surface roughness has long been known to have a sig-
nificant impact on the boiling process [Jakob, 1936;
Bankoff, 1957; Clark et al., 1959; Griffith & Wallis,
1960]. In particular, the realization of the importance
of cavities in enhancing pool boiling heat transfer
rates spurred interest in using roughened surfaces to
increase the number and size of nucleation sites [Corty
& Foust, 1955; Gaertner & Westwater, 1960; Kurihara
& Myers, 1960; Bonilla et al., 1965; Marto & Rohsenow,
1966; Marto et al., 1968; Berenson, 1962; Vachon et al.,
1968; Kravchenko & Ostroukovskiy,
1979]. The consensus is that the surface roughness not
only affects the superheat required for incipience but
also the nucleate boiling performance. However, no
universally accepted method of characterizing surfaces
for nucleate boiling exists, and physics-based models
have yet to be developed that incorporate the important
characteristics of the surface and boiling fluid in de-
scribing the complex surface–fluid–vapor interaction.

An experimental setup was designed by Jones and
Garimella [2007] to study the effect of surface rough-
ness on the pool boiling of water and provide addi-
tional insights into the role of surface roughness on
nucleate boiling. Five aluminum surfaces of varying
roughness were prepared: one polished (0.062 μm
RMS) and four roughened by electrical discharge ma-
chining (EDM) with surface roughness of 1.37, 2.81,
7.37, and 12.53 μm RMS. All experiments were per-
formed in water at atmospheric pressure and saturation
temperature.

The boiling curve and heat transfer coefficient are
shown in Figures 7a and 7b, respectively. The rough-
est EDM surface (12.53 μm RMS) gave rise to the
lowest superheat and the highest heat transfer coeffi-
cient at a given heat flux, while the polished test piece
(0.062 μm RMS) was associated with the highest su-
perheat as well as the lowest heat transfer coefficient.
The other EDM test surfaces of intermediate rough-
ness (1.37, 2.81, and 7.37 μm RMS) performed very
similarly to each other; these surfaces showed an ap-
proximately 60% improvement in the heat transfer co-
efficient over the polished surface at the higher heat
fluxes while the 12.37 μm EDM surface showed ap-
proximately a 100% improvement. In addition, boiling
incipience was found to occur at a superheat of ap-
proximately 5°C for the roughest EDM surface (12.53
μm), 7°C for the other EDM surfaces (1.37, 2.81, and
7.37 μm RMS), and 11°C for the polished surface.

The boiling process was also observed using a
high-speed camera system. In Figure 8, the first row
shows the boiling process at the incipience superheat
(approximately 1.5 W/cm² for the polished surface and
between 1.0 and 1.2 W/cm² for the EDM sur-
faces). The 12.53 μm EDM surface has the smallest
bubble departure diameter and a high bubble emission
frequency. Conversely, the polished surface has the
largest bubble diameter and lowest bubble emission
frequency. At a heat flux of 5 W/cm² (second row of
Fig. 8), it is clear that the EDM surfaces offer many
more nucleation sites than the polished surface. At this
heat flux, the polished test piece forms isolated bubbles while the EDM surfaces show signs of bubble agglomeration. At a heat flux of 10 W/cm² (third row of Fig. 8), vapor slugs and vapor columns can be seen on all the surfaces. Again, the EDM surfaces show substantially more active nucleation sites. At 21 W/cm², large vapor columns and mushrooms are seen on the EDM surfaces. Vapor columns are also present on the polished surface, but of a smaller size than on the EDM surfaces. At $q_s^{\prime\prime} = 38$ W/cm², large vapor columns can be seen on all of the surfaces. However, there are still visible portions of the polished surface not covered by vapor while the EDM surfaces show much greater surface coverage.

It was also pointed out by Jones and Garimella [2007] that, despite its importance in understanding pool boiling heat transport, very little success has been achieved towards characterizing boiling surfaces.
Manually measuring cavity sizes and shapes using optical or electron microscopes [Yang & Kim, 1988; Cornwell, 1977; Wang & Dhir, 1993] is tedious and only tractable on smooth surfaces where the number of cavities is small; characterization of rough surfaces, where the cavities are often highly irregular in shape with small cavities embedded within larger cavities, presents a difficult challenge. Therefore, more sophisticated surface characterization and data processing techniques are needed along with a better understanding of the complex wetting phenomenon.

2.3 Flow Boiling in Microchannels

Flow boiling in microchannels has attracted much attention in recent years due to its potential for extremely high heat transfer rates in compact spaces. Additional advantages of utilizing the convective boiling process include a lower flow rate requirement and improved streamwise temperature uniformity than in single-phase microchannel heat sinks. In spite of these appealing attributes, the complex nature of convective boiling and two-phase flow in microchannels is still not well understood and has impeded their practical implementation [Sobhan & Garimella, 2001; Garimella & Sobhan, 2003; Kandlikar, 2002; Bergles et al., 2003; Thome, 2004]. Among the unresolved issues, of particular importance are a comprehensive experimental characterization of boiling flow regimes and heat transfer behavior and the quantitative prediction of flow boiling heat transfer in microchannels of different sizes.
sizes over a wide range of mass and heat fluxes with different working fluids [Liu & Garimella, 2007]. There is thus a clear need for additional systematic studies which carefully address the experimental characterization and physics-based modeling of flow boiling in microchannel flows. Recent studies in the authors’ group on flow boiling with water, a dielectric fluid (FC-77) and a refrigerant (R134a) [Lee & Garimella, 2008; Harirchian & Garimella, 2008; Bertsch et al., 2007] are reviewed here.

2.3.1 Flow boiling of water in silicon microchannels

A novel silicon microchannel test piece was designed and fabricated to include a $5 \times 5$ array of resistors and local temperature-sensing diodes as shown in Figure 9. Each of the 25 heater/temperature sensor elements incorporates a heating element and integrated diode sensors for on-die local temperature monitoring. The microchannels are cut into the top surface (back side) of the chip. The microchannel dimensions range in width from 102 $\mu$m to 997 $\mu$m, with a fixed nominal channel depth of 400 $\mu$m. Heat flux, temperature and pressure drop measurements were obtained during saturated flow boiling in these test pieces over a wide range of experimental conditions [Lee & Garimella, 2008].

A critical review of correlations in the literature showed that existing correlations developed for large channels do not match the experimental results obtained for two-phase pressure drop and heat transfer associated with flow boiling in microchannels. In order to improve the predictive capability for microchannels of various sizes, Lee and Garimella [2008] proposed new correlations to predict the two-phase pressure drop and the saturated boiling heat transfer coefficient. In the new pressure drop correlation, based on the approach of Mishima and Hibiki [1996], a more representative mass flux term was obtained using regression analysis of the comprehensive experimental data obtained. An additional channel size correction term was also introduced to more adequately account for the channel size effect. The asymptotic boiling heat transfer model of Steiner and Taborek [1992] was modified to account for the specific features of flow through microchannels. New correlations for the convective enhancement factor $F_{\text{conv}}$ and the nucleate boiling correction factor $F_{\text{nb}}$ were derived following the procedure by Steiner and Taborek [1992]. A channel size correction term was included in the nucleate boiling correction factor $F_{\text{nb}}$ to reflect the fact that the nucleate boiling component is influenced by the channel size for microchannel flow. Predictions of the two-phase pressure drop and the saturated boiling heat transfer coefficient showed good agreement with experimental data.
transfer coefficients are compared with the experimental data in Figures 10a and 10b and show very good agreement for the experimental conditions, with mean average errors of 11.4% and 14.7%, respectively; this agreement was significantly better than with the correlations in the literature.

2.3.2 Flow boiling of a dielectric liquid in silicon microchannels

Use of dielectric liquids in microchannel heat sinks has drawn recent attention since the working fluid in the microchannel heat sinks can then be in direct contact with the electronics. Although there have been a
number of investigations on flow boiling in microchannels using perfluorocarbon liquids [Lin et al., 1992; Chen & Garimella, 2006a,b; Pate et al., 2006], few have arrived at clear conclusions on the effects of mass flux and microchannel dimensions on the boiling heat transfer and pressure drop. In view of the absence of systematic studies on these issues in the literature, Harirchian and Garimella [2008] performed experiments with a dielectric fluid, FC-77, in microchannels with nominal widths of 100, 250, 400, 700, and 1000 μm (fixed depth of 400 μm) using a similar test setup as in [Chen & Garimella, 2006a,b]. The effect of microchannel size on the boiling curve for a fixed mass flux of 700 kg/m²⋅s is shown in Figure 11a. For microchannels of width 400 μm and larger, the boiling curves collapse to a single curve beyond the onset of nucleate boiling. As boiling starts in these microchannels, the wall temperature has a weak dependence on the heat flux and is relatively constant; however, at high heat fluxes the wall tem-

![Figure 11](image-url)
The wall temperature becomes more dependent on heat flux and the boiling curves deviate for different sizes. For the microchannels of width 100 and 250 μm, the wall temperature increases with increasing wall heat flux and the boiling curves do not collapse on those of the larger microchannels. This may be attributed to the early establishment of annular flow in microchannels of a very small diameter. The effect of microchannel size on the boiling curves seen in Figure 11 was also observed in the experiments conducted with other mass fluxes. The influence of microchannel size on the heat transfer coefficient is illustrated in Figure 11b. Interestingly, the heat transfer coefficient is independent of microchannel width for channels of width 400 μm and larger, and has a weak dependence on channel size for smaller microchannels. At the 250-μm width, the heat transfer coefficients are slightly lower than for the larger sizes at any wall heat flux. For the 100-μm-wide microchannels, the behavior is markedly different, with the heat transfer coefficient being rela-
tively higher at the lower heat fluxes. As the heat flux increases, the curve crosses over and is lower than for larger microchannels.

In Figure 12a, the pressure drop for different channel sizes is presented at a fixed mass flux as a function of wall heat flux. In both the single-phase and two-phase regions, the pressure drop increases with decreasing microchannel width at a given wall flux. In the two-phase region, the slope of the line also increases as the channel width decreases, resulting in much larger pressure drops for smaller channels at higher heat fluxes. Figure 12b shows the pumping power required to manage a required heat sink base heat flux with different microchannel sizes. In the single-phase region the pumping power required is almost constant, independent of heat flux, while in the two-phase region, the pumping power increases rapidly with heat flux. This figure also shows that for microchannels of width 400 μm and larger, the pumping power is not a strong function of microchannel width. For widths below 400 μm, however, the pumping power increases for a given base heat flux. Therefore, for a given pumping power, more heat can be removed from the heat source with larger microchannels, although, as discussed in the previous section, using the larger microchannels results in higher wall temperatures.

In ongoing work, these trends are being related to the two-phase flow regimes existing under the conditions of the experiment so that regime-based heat transfer and pressure drop models may be developed.

2.3.3 Flow boiling of refrigerant in microchannels

Refrigeration provides an effective means for greatly reducing the coolant temperature and therefore maintaining acceptable device temperatures when dissipating high heat fluxes. However, much of the published work [Mongia et al., 2006; Trutassanawin, 2006; Lee & Mudawar, 2006; Molki et al., 2004] on evaporative flow boiling heat transfer has dealt with the subcooled boiling of refrigerants. None of these studies investigated the local heat transfer coefficient of two-phase refrigerant up to saturated vapor conditions. This knowledge is essential to analyze evaporators in vapor compression systems since the fluid enters the evaporator as a two-phase mixture and remains in a superheated state. In addition, the currently available correlations for the two-phase heat transfer coefficient in microchannel heat exchangers predict unrealistic values or show large deviations from measurements.

Bertsch et al. [2007] conducted an experimental investigation of the local flow boiling heat transfer coefficient at different vapor qualities for refrigerant R-134a in a microchannel cold-plate evaporator used in small refrigeration systems with a target of dissipating approximately 200 W from an electronic chip. Measurements were carried out at several inlet flow conditions starting from subcooled liquid to two-phase flow with 80% vapor quality. Figure 13 shows the heat transfer coefficient at a fixed saturation pressure of 7.5 bar and four different refrigerant flow rates as a function of vapor quality [Bertsch et al., 2007].
is seen to increase with increasing flow rate. In the low-quality region of up to 40%, the heat transfer coefficient stays almost constant or increases slightly with quality, and then drops off as the quality increases beyond 40%. Measurement uncertainties, shown as error bands in the figure, were always below 13%. The variation of heat transfer coefficient with vapor quality for different saturation pressures (with the mass flow rate held constant at 0.5 g/s) is shown in Figure 14. It is seen that the heat transfer coefficient increases slightly with decreasing pressure, although the variation between the different measurements is almost within the measurement uncertainty.

3. ELECTROMECHANICAL MICROFLUIDIC ACTUATION

One of the important considerations in the implementation of microchannel heat sinks is to efficiently actuate flow at the microscale. Integrating the pumping action directly at the scale of the microchannels in a heat sink is an attractive alternative to the use of external pumps; a review of micropumps for electronics cooling was compiled in [Singhal et al., 2004]. On the other hand, handling fluids at the microscale presents significant challenges, as conventional fluid handling techniques do not translate well to the micro- and nanoscales. Electrical actuation for control of microscale fluid motion has emerged as an attractive option. The use of electric fields for fluid control obviates the need for moving mechanical parts, which are undesirable in microscale systems due to reliability, cost and fabrication concerns.

While electrically actuated fluid motion and the accompanying heat transfer have broad applications in MEMS and lab-on-a-chip devices, the resulting pumping action offers great potential for chip-integrated cooling. Promising advances in electronics cooling could be brought about by the use of site-specific, selective liquid cooling of devices on a chip. Chip-integrated liquid cooling [Garimella et al., 2006] could be harnessed to achieve increased localized heat spreading and reduced internal thermal resistances. Four representative microfluidic actuation technologies investigated in the authors’ group are discussed here: (i) microscale ionic winds; (ii) microscale pumping based on electrohydrodynamics (EHD); (iii) dielectrophoresis (DEP)-induced actuation of micro/nanofluids, and (iv) electrowetting-based control of liquid droplet motion.

3.1 Microscale Ionic Winds for Localized Enhancement

An ionic wind is formed when air ions generated by a corona discharge are accelerated by an electric field and exchange momentum with neutral air molecules, causing air flow. An alternative method is to create ions in the air from electrons generated by field emission. Because ionic winds can generate flow with no moving parts, they offer an attractive method for en-
hancing the heat transfer from a surface that would otherwise only be cooled by natural convection and/or radiation. In the presence of an external, flat plate flow, ionic winds distort the boundary layer such that local heat transfer is enhanced at the wall, and recent work has suggested that integrating such devices can be useful for cooling electronic components locally. In spite of the growing interest in using ionic winds to modulate external boundary layers in the aerospace community [Soetomo, 1992; Léger et al., 2002; Artana et al., 2002] it has received very little attention for heat transfer enhancement until recently [Schlitz et al., 2004; Hsu et al., 2006]. Miniature ionic wind devices have been studied at the microscale for on-chip thermal management of electronic devices. The authors’ group has modeled and studied the ionization process [Zhang et al., 2004; Go et al., 2006a, 2008] and demonstrated [Peterson et al., 2005] that, by using nanostructured carbon for geometric field enhancement, emission turn-on voltages can be greatly reduced.

Recently, Go et al. [2006b] investigated the efficacy of using ionic winds for local heat transfer enhancement in existing bulk flows. Experiments showed that ionic winds can increase the local heat transfer coefficient in an externally generated bulk flow over a flat plate by a factor of 2 for applied voltages of 2 to 4 kV. Additionally, the local heat transfer trends were predicted using a numerical model. Go et al. [2007a,b] documented mesoscale heat transfer enhancement along with parametric studies to identify enhancement trends.

Figure 15 shows a series of infrared, thermal contour images for a flat plate heater under three cooling conditions. In the first, the cooling is entirely by natural convection and radiation. In the second, a bulk flow is present, but the ionic wind is not turned on. Finally, in the presence of a bulk flow, when the ionic wind is turned on, temperatures over a wide region are seen to drop by over 20°C, compared to the condition where only a bulk flow is present. The local cooling effect is found to be greatest at larger corona currents, as expected. The peak value for heat transfer enhancement is greater than 200% for a current of 15 μA. The heat transfer enhancement occurs both upstream and downstream of the electrode pair, along the length of the wire, and peaks at the location of the corona wire. The upstream enhancement is not as great as the downstream enhancement. The enhancement downstream of the collecting electrode decreases steadily with distance as the effect of the ionic wind is dissipated.

FIGURE 15. Infrared temperature contours (°C) in the vicinity of an electrode pair under conditions of no bulk flow, a 0.28 m/s bulk flow, and ionic wind combined with this bulk flow.
3.2 Electrohydrodynamic (EHD) Micropumping

Microscale pumping based on induction electrohydrodynamics (EHD) has potential applications for chip-integrated liquid cooling of microelectronics. A new micropump concept was proposed by Singhal and Garimella [2005a]. This micropump integrates two existing pumping technologies, and thereby combines the operation of a vibrating diaphragm with that of induction EHD; this can lead to higher flow rates relative to their operation independent of each other. An early design of the micropump consisted of a valveless nozzle-diffuser pump with a series of thin, closely spaced parallel electrodes incorporated in the nozzle-diffuser elements. The two nozzle-diffuser elements were connected through a pumping chamber. The electrodes in the nozzle-diffuser elements were connected to a polyphase power supply, which, together with the piezoelectric diaphragm operation, drove the flow. The nozzle-diffuser elements have been eliminated and trapezoidal channels are used in the more recent design for easier fabrication as shown in Figure 16, without significant reduction in total flow [Singhal & Garimella, 2007].

Actuating the electrodes leads to pumping action due to induction of charges and the resultant Coulomb forces generated in the fluid. When the microelectronic component being cooled is operational, the fluid present in the microchannel experiences a temperature gradient across the height of the channel. This temperature gradient in turn causes a gradient in the electrical conductivity of the fluid. When an alternating voltage is applied to the electrodes, a traveling electric field propagates through the working fluid in the mi-

![Figure 16](image-url)

**FIGURE 16.** (a) Schematic representation of the EHD-based micropump design, (b) an array of microchannels, and (c) electrodes deposited on the bottom side of the lid covering the microchannels [Singhal and Garimella, 2007].
The traveling electric field results in an induction of charges in the bulk of the fluid. These charges are displaced due to charge relaxation and, hence, interact with the traveling wave, which leads to the application of Coulomb forces on the charges. These moving charges carry bulk fluid with them due to viscous effects, and this leads to an electrohydrodynamic (EHD) pumping action. Actuating the diaphragm shown in the figure causes a continual periodic increase and decrease in the volume of the pumping chamber, which superimposes an additional bulk flow. The combined operation of the vibrating diaphragm and induction EHD can lead to higher flow rates relative to their operation independent of each other. Since the channels can be fabricated directly in a silicon substrate, contact resistance is eliminated.

Experimentally validated numerical simulations indicate the promise of this novel micropump design. The net flow rates due to the action of induction EHD alone and from the combined action of the vibrating diaphragm and induction EHD, from a 200-μm-wide pump are shown in Figure 17, over a period during which the flow reaches a steady state. The steady-state net flow rate due to combined action of the vibrating diaphragm and induction EHD (1.75 × 10^{-10} m³/s) is 12% higher than the steady-state flow rate due to induction EHD alone (1.55 × 10^{-10} m³/s).

The flow due to the action of the vibrating diaphragm alone causes flow with a sinusoidal variation, even though the net flow is zero. It was shown in [Singhal & Garimella, 2005b] that when EHD is combined with the action of the vibrating diaphragm, a velocity enhancement is obtained due to an increase in the efficiency of EHD. Indeed, the increased flow in Figure 17 is solely due to the increase in the output of the EHD action, partially due to increased efficiency and partially due to increased power drawn from the electrodes.

The numerical model predicts that the micropump design can achieve a flow rate of 10.5 l/min of water for a pump of size 1500 μm × 200 μm × 50 μm. This corresponds to a total flow rate of 0.42 ml/min if the micropumps are integrated into forty parallel microchannels in a chip of size 1 cm × 1 cm. For a total fluid temperature rise in the microchannels of 50°C, this arrangement can remove 1.44 W/cm² of heat at a pumping power input of 0.32 mW [Singhal & Garimella, 2007]. The flow rate from the micropump can be significantly increased by decreasing the width and spacing of the electrodes (from the 20-μm dimension used in these simulations) and using fluids containing suspensions of nanoscale particles, which would enhance the EHD effect by offering additional interfaces in the fluid.

FIGURE 17. Comparison of flow due to combined action of vibrating diaphragm and induction EHD action to that from action solely of induction EHD for a pump of width 200 μm [Singhal and Garimella, 2007].
3.3 Dielectrophoresis-Induced Micropumping

Microelectrodes can be integrated into microfluidic devices to induce strong electromechanical interactions with flow fields, giving rise to electrohydrodynamic (EHD), electroosmotic (EOF), dielectrophoretic (DEP), and electrorotation (ROT) effects [Melcher, 1981; Probstein, 1994; Pohl, 1978; Jones, 1995]. Many unexplored avenues exist for microscale flow control and particle manipulation. A novel micropumping approach based on dielectrophoresis (DEP)-induced actuation of nanofluids has been proposed by the authors’ group, and involves the combined effects of DEP and EHD at the microscale, as shown in Figure 18.

Dielectrophoresis occurs when particles suspended in a medium are exposed to a non-uniform electric field. The interaction between the induced dipole and the electric field results in a net dielectrophoretic force on the particle [Probstein, 1994; Gascoyne et al., 1992]. The time-averaged dielectrophoretic force on a spherical particle of radius \( a \) can be computed as

\[
\langle \vec{F}_{\text{DEP}} \rangle = \pi a^3 \varepsilon_a \text{Re}[f_{CM}] \nabla |\vec{E}|^2
\]

\[
+2\pi a^3 \varepsilon_a \text{Im}[f_{CM}] \left( E_x \nabla \varphi_x + E_y \nabla \varphi_y + E_z \nabla \varphi_z \right)
\]  

(1)

in which \( f_{CM} \) is the Clausius–Mossotti factor defined as

\[
f_{CM} = \frac{\tilde{\varepsilon}_p - \tilde{\varepsilon}_m}{\tilde{\varepsilon}_p + 2 \tilde{\varepsilon}_m}
\]

(2)

in which \( \varepsilon_p \) and \( \varepsilon_m \) are the relative permittivities of the particle and the medium, respectively. The real and imaginary parts of the Clausius–Mossotti factor are denoted as Re\([f_{CM}]\) and Im\([f_{CM}]\), respectively. \( E_x, E_y, \) and \( E_z \) are the components of the electric field, and \( \varphi_x, \varphi_y, \) and \( \varphi_z \) are the phase angles of a traveling wave electric field. When Re\([f_{CM}] > 0\), positive DEP occurs and the particles move towards regions of high electric field strength, i.e., the electrode array, as shown in Figure 19a. On the other hand, negative DEP occurs if Re\([f_{CM}] < 0\). This leads to particles moving away from regions of high electric field strength, as shown in Figure 19b. A nonzero value of Im\([f_{CM}]\) and the presence of a traveling wave electric field gives rise to a traveling wave-induced DEP force which moves the particles in a direction opposite to that of the traveling wave propagation. The viscous drag resulting from particle motion leads to pumping of the surrounding fluid medium. For a sufficiently high particle concentration, the hydrodynamic interaction between the particles and the fluid could generate significant fluid flow and lead to cumulative pumping action.

DEP-based micropumping is being explored in the authors’ group [Liu & Garimella, 2008]. A traveling electric field is generated by applying a three-phase traveling wave voltage signal to an interdigitated microelectrode array, such as the one shown in Figure 18. The traveling wave DEP force on the particles has been estimated analytically, and the results show transverse motion of particles in regions above the elec-
FIGURE 19. Dielectrophoresis of micron-sized particles: (a) positive DEP and (b) negative DEP [Liu and Garimella, 2008].

FIGURE 20. DEP-induced flow velocity measurement in a polystyrene suspension using micro-particle image velocimetry: (a) velocity vector field and (b) average velocity at different voltages and frequencies [Liu and Garimella, 2008].
trode surface. The particle dynamics and particle–fluid interaction are being studied using the method of reflections which relates the contribution of the motion of individual particles to the enhancement of the overall flow field. The flow field in a microchannel resulting from traveling wave DEP is also being simulated. Preliminary micropumping experiments were performed using polystyrene microparticles (2.9-μm diameter) in deionized water with a volume concentration of 0.001%. The flow fields under different applied voltages and frequencies are analyzed using micro-particle image velocimetry (μ-PIV). Figure 20a shows a typical velocity vector field. The average measured velocities at different voltages and frequencies are shown in Figure 20b. A maximum velocity of 60 μm/s was obtained with this preliminary experimental setup.

3.4 Electrowetting-Based Control of Droplet Motion

The concept of EWOD (ElectroWetting on Dielectric) is founded on the premise of a reduction in dielectric–liquid interfacial energy upon the application of a voltage between a conducting droplet and an underlying dielectric layer [Pollack et al., 2002]. The electric field-induced reduction in interfacial energy causes the droplet to locally spread out. The resulting change in contact angles sets up a pressure gradient which drives the droplet towards the actuated electrode. EW-induced droplet pumping holds promise for application in the area of high-flux microelectronics cooling. A schematic illustration of a possible cooling device is shown in Figure 21. It consists of arrays of droplets moving along the length of the chip. The distinct advantage to this device is that the pumping arrays can be controlled independently by appropriate circuitry, unlike the less-selective forced convective flow in microchannels. In addition, the use of artificially structured surfaces in conjunction with electrowetting (as shown in Fig. 22) could be employed in the design of a novel cooling scheme for hot spots in microelectronic chips, as discussed later in this section.

Bahadur and Garimella [2006] developed a generic energy minimization-based modeling framework for EW systems, according to which the droplet minimizes its surface energy by transiting to the actuated electrode. The energy gradient is thus the driving force behind EW-induced motion of a fluid element. This model was used to estimate the EW actuation force on a droplet moving between two flat plates as illustrated in Figure 23. The actuation force has three components corresponding to the three regions of the droplet surface which undergo interfacial energy change under the influence of the applied voltage:

\[ T_{\text{act}}(x) = F_1(x) + F_2(x) + F_3(x), \]

where \( F_1(x) \) and \( F_2(x) \) are the force components arising from the dielectric–liquid interfacial energy reduction on the actuated and ground electrodes of the bottom plate, respectively. These forces have contributions from the rates of both area and voltage change of the associated capacitance. \( F_3(x) \) has only one contribution originating from the voltage change across the top plate dielectric.

**FIGURE 21.** Schematic illustration of an EW-based cooling device.
Figure 24 shows the transition time required for a 1-mm-radius droplet to move to the actuated electrode as a function of the applied voltage for three plate spacings. It is seen that droplet transition times strongly depend on the actuation voltage, which underscores the dominant influence of the voltage on the actuation force. No significant dependence of transition time on the gap spacing is seen, except in the low-voltage, low-velocity regime where a slight decrease in transition time is observed with increased gap spacing. While an increase in gap spacing leads to an increased droplet mass, the opposing viscous forces also change with gap spacing. The net effect is the absence of any substantial dependence of transition time on gap spacing. Average and peak velocities of 7 cm/s and 10 cm/s respectively were obtained for an actuation voltage of 50 V.

The energy-minimization approach can be extended to develop expressions for apparent contact angles of droplets on rough surfaces in the presence of an EW voltage [Bahadur & Garimella, 2007a]. The results indicate that an EW voltage can be used to dynamically change the Cassie and Wenzel contact angles of droplets and thus provides a powerful tool to manipulate droplet behavior on rough surfaces. The hot-spot cooling application can be understood by treating the Cassie and Wenzel droplet states as the "off" and "on" positions of a thermal resistance switch [Bahadur & Garimella, 2007b]. The droplet in its Cassie (off) state offers a high thermal resistance (between the base of
the substrate and the droplet) to heat transfer from the substrate due to the low contact area and the air gap resulting from the grooves. The thermal resistance in the Wenzel (on) state is dramatically reduced when the contact area is increased and the droplet wets the grooves. This concept of a thermal switch can be explored by calculating the change in thermal resistance between the substrate and the droplet resulting from the application of an EW voltage.

4. MICROSCALE DIAGNOSTICS

Recent advances in the field of MEMS and microfluidics have given rise to a growing interest in miniaturizing the scale of many thermal, chemical and biological transport processes. As the number of potential applications has increased, it has become necessary to develop new characterization tools. The small length scales have posed challenges for the measurement of flow and temperature fields with conventional techniques. The difficulties are especially pronounced for microfluidic systems, where accurate determination of velocity, shear stress, temperature, and heat flux is desired with micron-scale spatial resolution. Two recent developments in the authors’ group for nonintrusive velocity and temperature measurement are discussed.

4.1 Infrared Micro-Particle Image Velocimetry

Accurate measurement of fluid flows in microchannel heat sinks requires diagnostic techniques with micron-scale spatial resolution. One such technique that has grown in popularity is micro-particle image velocimetry (μPIV). The use of μPIV has been limited, however, to applications in which optical access to the flow field is available at visible wavelengths. In some applications such as stacked microchannel heat sinks, optical access to the fluid flow in the visible spectrum is unavailable. Since silicon is quite transparent to wavelengths between 1100 and 2500 nm, a diagnostic technique in the infrared range would be advantageous for measuring sub-surface flow fields in silicon microdevices.

A nonintrusive measurement technique, infrared micro-particle image velocimetry (IR-μPIV), was demonstrated by Liu et al. [2005] in which quantitative velocity measurements were made for flows inside (opaque) fused silica microtubes. Recently, Jones et al. [2008] further developed the IR-μPIV technique and used it to study flow maldistribution in a silicon microchannel heat sink with seventy-six 110-μm-wide x 371-μm-deep channels.

The flow distribution through the microchannel heat sink is shown in Figure 25. Velocity profiles in
various channels across the heat sink are shown in Figure 25a for Re = 10.2 and in Figure 25b) for Re = 102. Also plotted in Figures 25a and 25b are the mean theoretical velocity profiles evaluated from the known total flow rate through the heat sink and the channel dimensions. Very little flow maldistribution is observed at the lower flow rate (Re = 10.2), at which all of the measured velocity profiles fall very close to the mean velocity profile. At the higher flow rate (Re = 102), much higher velocities are observed in channels near the centerline than towards the lateral edge. Figure 25c shows the ratio of the area-averaged mass flux determined from the PIV measurements to the average mass flux determined from the known total flow rate and average channel dimensions. The maximum mass flux ratio is approximately 2.4% greater than the minimum for Re = 10.2 and 29.9% greater for Re = 102. For Re = 102, channels 1 through 15 carry approximately 55% of the mass flow rate despite representing only 39% of the channels. It may be noted that the minimum measured mass flux ratio for Re = 102 occurs in channel 30, with channel 38 having approximately a 1.7% greater mass flux ratio than channel 30.

4.2 Nonintrusive Microscale Temperature Measurement

Several techniques have been proposed for obtaining microscale temperature measurements [Childs et al., 2000; Yoo, 2006], such as microfabricated thermocouples and resistance temperature detectors (RTD), infra-
red thermography, molecular tagging thermometry (MTT), laser induced fluorescence thermometry, thermochromic liquid crystals (TLCs), micro-Raman thermometry, and thermoreflectance. More recently, it was recognized that the Brownian motion of sub-micron sized particles, used to seed the liquid in PIV measurements, can also be used to make temperature measurements. For small seed particles (<1 m) at low speed flows (<10 m/s), the Brownian motion was significant enough to cause a width-wise spreading of the correlation peak. This width-wise broadening of the correlation function was used to calculate the fluid temperature by Hohreiter et al. [2002] with an experimental uncertainty of ±3°C.

Chamarthy et al. [2006a,b] developed PIV thermometry as a non-intrusive whole-field diagnostic tool suitable for making simultaneous microscale measurements of temperature and velocity in steady flows. Three different methods, including cross correlation PIV, Single Particle Tracking Thermometry (SPTT) and Low Image Density (LID) PIV were used to analyze a set of experimental images to obtain temperature measurements. A standard epi-fluorescence PIV system was used for all the measurements, which were conducted using spherical fluorescent polystyrene-latex particles suspended in water. Temperatures ranging from 20°C to 80°C were measured. The PIV thermometry method was also used to simultaneously measure the velocity and temperature of a moving fluid. The average difference between the measured and predicted temperatures was found to be ±1.5°C, as shown in Figure 26.

5. CONCLUSIONS

Recent developments in microscale thermal transport and electromechanical microfluidic actuation from the authors’ work have been reviewed. Thin-film transport, boiling heat transfer (both pool boiling and convective flow boiling in microchannels) and a number of novel microfluidic actuation methods based on electromechanical effects are discussed, along with recent progress in microscale flow characterization and non-intrusive temperature measurement.

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REFERENCES


