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Microchannel Heat Transfer

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1. Introduction

Microchannel Heat transfer has the very potential of wide applications in cooling high power density microchips in the CPU system, the micropower systems and even many other large scale thermal systems requiring effective cooling capacity. This is a result of the micro-size of the cooling system which not only significantly reduces the weight load, but also enhances the capability to remove much greater amount of heat than any of large scale cooling systems. It has been recognized that for flow in a large scale channel, the heat transfer Nusselt number, which is defined as hD/k, is a constant in the thermally developed region where h is the convective heat transfer coefficient, k is thermal conductivity of the fluid and D is the diameter of the channel. One can expect that as the size of the channel decrease, the value of convective heat transfer coefficient, h, becomes increasing in order to maintain a constant value of the Nusselt number. As the size of the channel reduces to micron or nano size, the heat transfer coefficient can increase thousand or million times the original value. This can drastically increase the heat transfer and has generated much of the interest to study microchannel heat transfer both experimentally and theoretically.
On the other hand, the lab-on-chip system has seen the rapid development of new methods of fabrication, and of the components — the microchannels that serve as pipes, and other structures that form valves, mixers and pumps — that are essential elements of microchemical ‘factories’ on a chip. Therefore, many of the microchannels are used to transport fluids for chemical or biological processing. Specially designed channel is used for mixing of different fluids or separating different species. It appears that mass or momentum transport process inside the channel is very important. In fact, the transfer process of the mass is very similar to the transfer process of the heat due to similarity of the governing equations for the mass and the heat (Incropera et al., 2007). It can be readily derived that the Nusselt number divided by the Prandtl number to the nth power is equal to the Sherdwood number (defined as the convective mass transfer coefficient times the characteristic length and divided by the diffusivity of the mass) divided by the Schmidt number (defined as the kinematic viscosity divided by the diffusivity of the mass) to the nth power. Understanding of the heat transfer can help to understand the mass transfer or even the momentum transfer inside the microchannel (Incropera et al., 2007).

However, the conventional theories, such as the constitutive equations describing the stress and the rate of deformation in the flow, or the Fourier conduction law, are all established based on the observation of macroscopic view of the flow and the heat transfer process, but do not consider many of the micro phenomena occurred in a micro-scale system, such as the rarefaction or the compressibility in the gas flow, and the electric double layer phenomenon in the liquid flow, which can significantly affect both the flow and the heat transfer in a microchannel. Therefore, both the flow and the heat transfer process in a microchannel are significantly different from that in a large scale channel. A thorough discussion and analysis for both the flow and the heat transfer process in the microchannels are required. In addition, experimental study to confirm and validate the analysis is essential. However, accurate measurements of flow and heat transfer information in a microchannel rely very much on the exquisite fabrication of both the microchannel and the microsensors by the MEMS techniques. Successful fabrication of these complicated microchannel system requires a good knowledge on the MEMS techniques. Especially, accurate measurement of the heat transfer inside a microchannel heavily relies on the successful fabrication of the microchannel integrated with arrays of miniaturized temperature and pressure sensors in addition to the fabrication of micro heaters to heat up the flow.
It appears that microfluidics has become an emerging science and technology of systems that process or manipulate small (10^{-9} to 10^{-18} liters) amounts of fluids, using channels with dimensions of tens to hundreds of micrometres (George, 2006; Vilkner et al., 2004; Craighead, 2006). Various long or short micro or nanochannels have used in the system to transport fluids for chemical or biological processing. The basic flow behavior in the microchannel has been studied in certain depth (Bayraktar & Pidugu, 2006; Arkilic & Schmidt., 1997; Takuto et al., 2000; Wu & Cheng, 2003). The major problem in the past is the difficulty to install micro pressure sensors inside the channel to obtain accurate pressure information along the channel. Therefore, almost all of the pressure information is based on the pressures measured at the inlet and the outlet outside of the channel, which is used to reduce to the shear stress on the wall. The measurements have either neglected or subtracted an estimated entrance or exit pressure loss. These lead to serious measurement error and conflicting results between different groups (Koo & Kleinstreuer, 2003). The friction factor or skin friction coefficient measured in microchannel may be either much greater, less than or equal to the one in large scale channel. Different conclusions have been drawn from their measurement results and discrepancies are attributed to such factors as, an early onset of laminar-to turbulent flow transition, surface roughness (Kleinstreuer & Koo 2004; Guo & Li 2003), electrokinetic forces, temperature effects and microcirculation near the wall, and overlooking the entrance effect. In addition, when the size or the height of the microchannel is much smaller than the mean free path of the molecules or the ratio of the mean free path of the molecules versus the height of the microchannel, i.e. Kn number, is greater than 0.01, one has to consider the slip flow condition on the wall (Zohar et al. 2002; Li et al. 2000; Lee et al., 2002). It appears that more accurate measurements on the pressure distribution inside the microchannel and more accurate control on the wall surface condition are necessary to clarify discrepancies amount different work.
The lack of technologies to integrate sensors into the microchannel also occurs for measurements of the heat transfer data. All the heat transfer data reported is based on an average of the heat transfer over the entire microchannel. That is, by measuring the bulk flow temperature at the inlet and the outlet of the channel, the average heat transfer for this channel can be obtained. No temperature sensors can be inserted into the channel to acquire

the local heat transfer data. Therefore, detailed information on the local heat transfer distribution inside the channel is not reported. In addition, the entry length information and the heat transfer process in the thermal fully developed region is lacking. Besides, the wall roughness inside the channel could not be controlled or measured directly in the tube. Therefore, its effect on the heat transfer is not very clear. This was attributed to cause large deviation in heat transfer among different work (Morini 2004; Rostami et al., 2002; Guo & Li, 2003; Obot, 2002). It appears that accurate measurements of the local heat transfer are required to clarify the discrepancies among different work.

Therefore, in this chapter, a comprehensive review of microchannel flow and heat transfer in the past and most recent results will be provided. A thorough discussion on how the surface forces mentioned above affect the microchannel flow and heat transfer will also be presented. A brief introduction on the MEMS fabrication techniques will be presented. We have developed MEMS techniques to fabricate a microchannel system that can integrate arrays of the miniaturized both pressure and temperature sensor. The miniaturized sensors developed will be tested to ensure the reliability, and calibrated for accurate measurements. In fact, fabrication of this microchannel system requires very complicated fabrication steps as mention by Chen et al. 2003a and 2003b. Successful fabrication of this channel which is suitable for measurements of both the local pressure drop and heat transfer data is a formidable task. However, fabrication of this complicated system can be greatly simplified by using polymer material (Ko et al., 2007). This requires fabrication of pressure sensor using polymer materials (Ko et al., 2008). The polymer materials that have a very low thermal conductivity can be fabricated as channel wall to provide very good thermal insulation for the channel and significantly reduce streamwise conduction of heat along the wall. This allows measurements of very accurate local heat transfer inside the channel. In addition, the height of the channel can be controlled at desired thickness by spin coating the polymer at desired thickness. The shape of the channel can be readily made by photolithography. All the design and fabrication techniques for both the channel and the sensor arrays will be discussed in this chapter. Measurements of both the local pressure drop and heat transfer inside the channel will be presented and analyzed. Therefore, the contents of the chapter are briefly described as follows:

1. Gas flow and the associated heat transfer characteristics in microchannels.
2. Liquid flow and heat transfer characteristics in microchannels including (a) the single phase and (b) the two phase flows.
3. MEMS fabrication techniques
4. Discussion on recent developments and challenges faced for MEMS fabrication of the microchannel system.
5. Working principle and fabrication of the miniaturized pressure and temperature sensors.
6. Fabrication of the complicated microchannel system integrated with arrays of either or both the miniaturized pressure and temperature sensors.
7. Local heat transfer and pressure drop inside the microchannels.

2. Gas flow characteristics in microchannels

Recent development of micromachining process which has been used to miniaturize the fluidic devices has become a focus of interest to industry, e.g. micro cooling devices, micro heat exchangers, micro valves and pumps, and lab-on-chips, more studies have been dedicated to this field. The fluid flows in micro scale capillary tube can be traced back to Knudsen at 1909. However, it has been very difficult to perform an experiment for micro scale flow and make detailed observation in a micro-channel due to the lack of techniques to fabricate a microchannel and make arrays of small sensors on the channel surface. Up to the present, most of the important information on micro scale thermal and flow characteristics inside the microchannel can not be obtained and measured. Instead, the flow and heat transfer experiments performed for micro scale flow in the past are mostly based on the measurements of pressures or temperatures at inlet and outlet of the channel and the mass flow rate, or the measurements on the surface of a relatively large scale channel. Therefore, some of peculiar transport processes which are not important in a large scale channel may play a dominant role to affect the flow and heat transfer process in the micro scale channel, e.g. the rarefaction effect of the gas flow. Therefore, the rarefaction of a gas flow in the microchannel should be taken into account in the analysis.

2.1 Theoretical analysis

In order to describe the rarefaction of gaseous flow, a ratio of the mean free path to the characteristic length of the flow called Knudsen number (Kn) has been used as a dimensionless parameter. The Knudsen number is defined as λ/D_c, where "λ" denotes the mean free path of gas molecules and "D_c" denotes the characteristic dimension of the channel. For convenience, it has been suggested (Tsien, 1948) that the rarefaction in gases can be typically classified into three flow regions by the magnitude of the Knudsen number, which are "the continuum flow regime", "the free-molecular flow regime" and "the near-continuum flow regime", as described as follows.

1. Continuum flow regime: This regime is defined for flow with $Kn < 0.001$. In this regime, the theories of the gas flow and fluid properties completely conform to the continuum assumption, and the Knudsen numbers approach to zero. In addition, the modified classical theories of the liquid flow are also suitable in this regime.
2. Near-continuum flow regime: this flow regime is defined in the range with $0.001 \le Kn < 10$. The Knudsen number in this flow regime is still large enough that the flow is subject to a slight effect of rarefaction. The flow can be considered as a continuum in the core region except in the region adjacent to the wall where a small departure from the continuum such as velocity-slip or temperature jump is assumed. For convenience, one can further subdivide the flow into two regimes, i.e. the slip-flow regime and the transition-flow regime. In the slip-flow regime, the macroscopic continuum theory, therefore, is still valid due to small departures from the continuum. However, in order to conform to the real-gas behavior, it is necessary to adopt some appropriate corrections for the slip of fluid at the boundary. The slip-flow regime is defined in the range of $0.001 \le Kn < 0.1$ while the transition-flow regime is defined in the range of $0.1 \le Kn < 10$. In the transition-flow regime, the intermolecular collisions and the collisions between the gaseous molecules and the wall are of more or less equal importance. The flow configuration can be regarded as neither a continuum, nor a free-molecular flow. There is no simplified approach to attack this problem. Some conventional methods, such as, directly solving the complete sets of Boltzmann equations or using the empirical correlations from the experimental data, have been adopted.
3. Free-molecular flow regime: This flow regime is defined in the region with $10 \le Kn$. The rarefaction effect dominates the entire flow field. The gas is so rarefied that

intermolecular collisions can be negligible. Hence, the flow characteristic is described by the kinetic theory of gas. Only interaction between gas molecules and boundary surface is considered.

Meanwhile, it has also been suggested (Tsien, 1946) that one can employ the kinetics theory of gases or the conventional heat transfer theory to study the gas flow in the continuum flow regime. When the gaseous rarefaction is within the range of the free-molecular flow regime, the kinetics theory of gases is suitable for use. However, in the range of the near-continuum flow regime, there has been no well-established method. In the slip-flow regime the gas flow can be considered as continuum. Hence, we can employ the macroscopic continuum theory to study the heat transfer in gases by taking account the velocity-slip and temperature-jump conditions at the wall. In the transition-flow regime the transport mechanisms in the rarefied gas are between the continuum and the free molecule flow regime, it is incorrect to consider the gas as a continuum or free molecule medium. Therefore, the theoretical study in the transition regime is very difficult. Many of the works (Ko et al., 2008, 2009, 2010; Bird et al., 1976a; Eckert and Drake, 1972; Yen, 1971; Ziering, 1961; Takao, 1961; Kennard, 1938) intend to develop some convenient methods to solve this problem, such as enlarging the validation of macroscopic continuum theory by using some corrections in boundary conditions or developing mathematical schemes to directly solve the highly nonlinear Boltzmann equation. However, these approaches are still not successful.

For theoretical study of the rarefied-gas flow, Kundt and Warburg (1875) have been the first to propose an important inference by experimental observation. They found an interesting phenomenon that the gaseous flow exhibits a velocity-slip on solid wall when the pressure in the system is sufficiently low. This phenomenon later has been confirmed by the analytical results from kinetics theory of gas by Maxwell (1890). In addition, Maxwell also defined a parameter "f_S" called tangential momentum accommodation coefficient to modify the departures from the theoretical assumptions and real-gas behavior in molecular collision processes. The value of f_S will presumably depend upon the character of the interaction between the gaseous molecules and the wall, such as the surface roughness or the temperature etc. In the observations of wall slip, Timiriazeff (1913) made the first direct measurements of wall slip. However, the most accurate measurements of velocity slip are undoubtedly made by Stacy and Van Dyke, respectively. Hence, a sound theory used to describe the rarefied gas behaviors has been established successfully. In the heat transfer studies, Smoluchowski (1910) has performed the first experiments for a heated rarefied gas flow and found the temperature-jump occurring on the solid wall.

Kennard (1938) has suggested that it could be analogous to the phenomenon of velocity slip and thus developed an approximate expression to describe this temperature discontinuity. In a flow field with a temperature of the gas flow different from the neighboring solid wall, there exists a temperature difference in a small distance "g", which is called temperature jump distance, between the gas and the solid wall. The jump distance "g" is inversely proportional to the pressure but directly proportional to the mean-free-path of the gas. Due to the very small jump distance, it looks as having a discontinuity in the temperature distribution between the gas flow and the neighboring solid wall. By using the thermal accommodation coefficient proposed by Knudsen (1934) and the concepts of heat transfer mechanism between gas molecules defined by Maxwell, a theory for the microscopic heat transfer occurred in the rarefied gas flows has been successfully established.

In addition, the gas flow in a micro-channel also involves other problems, such as compressibility and surface roughness effects. Therefore, other dimensionless parameters, such as the Mach number, Ma, and the Reynolds number, Re, should also be adopted. The relationship among these parameters has been derived and can be expressed as follows.

$$\mathrm{Re} = \sqrt{\frac{k\pi}{2}}\frac{Ma}{Kn} \tag{2-1}$$

where k is the specific heat ratio (c_p/c_v) of the gas. Since both Ma and Kn vary with compressibility of gas in the channel, the value of Re should vary according to the above equation. The full set of governing equations for two dimensional, steady and compressible gas flows can be written as follows (Khantuleva et al., 1982):

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0 \tag{2-2}$$

$$\rho u\frac{\partial u}{\partial x} + \rho v\frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + \mu[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{1}{3}(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 v}{\partial x \partial y})] \tag{2-3}$$

$$\rho u\frac{\partial v}{\partial x} + \rho v\frac{\partial v}{\partial y} = -\frac{\partial p}{\partial y} + \mu[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{1}{3}(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 u}{\partial x \partial y})] \tag{2-4}$$

$$\rho u C_p\frac{\partial T}{\partial x} + \rho v C_p\frac{\partial T}{\partial y} = u\frac{\partial p}{\partial x} + v\frac{\partial p}{\partial y} + k(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}) + \mu[2(\frac{\partial u}{\partial x})^2 + 2(\frac{\partial v}{\partial y})^2 + (\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y})^2 - \frac{2}{3}(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y})^2] \tag{2-5}$$

$$p = \rho RT = nkT \tag{2-6}$$

The boundary conditions for the velocity slip and temperature jump on the top and bottom walls are shown as follows (Wadsworth et al., 1993):

$$u - u_w = \frac{2-\sigma_u}{\sigma_u}\lambda(\frac{\partial u}{\partial y})_w + \frac{3}{4}\frac{\mu}{\rho T}(\frac{\partial T}{\partial x})_w; \quad y = \pm h/2 \tag{2-7}$$

$$T - T_w = \frac{2-\sigma_T}{\sigma_T}\frac{2\gamma}{(\gamma+1)}\frac{\lambda}{\mathrm{Pr}}(\frac{\partial T}{\partial y})_w; \quad y = \pm h/2 \tag{2-8}$$

where σ_u and σ_T are the momentum and the energy accommodation coefficient, respectively. λ, γ and h are the mean free path, the specific heat ratio and the height of the microchannel, respectively. Review of the recent literature indicates that compressible gas flow problems have been studied from the slip to the continuum flow regimes, however, different results are obtained in the micro-channels as described in the following paragraphs.

To analyze the rarefied gas characteristics in the near-continuum flow regime, the methods used (Takao, 1961; Kennard, 1938) in the classical kinetics theory of gas include (1) the small-perturbation approach, (2) the moment methods and (3) the model equation. The mathematical procedures of the small-perturbation approach are to use the perturbation

technique to linearize the Boltzmann equation. Since this method can be used in both the near-continuum regime and the near free-molecules regime, therefore, it is suitable for practical applications. The moment methods are first to make adequate assumptions in the velocity distribution f such as to express f in terms of a power series, i.e. $f = f_o(1 + a_1(Kn) + a_2(Kn)^2 + \ldots)$ as proposed by Chapman and Enskog. Then, substitute the assumed velocity distribution into the Boltzmann equation. The methods of the model equation are to construct a physics model, such as the B-G-K model proposed by Bhatnagar, Gross and Krook (1954), to simplify the expression of Boltzmann equation. Since the governing equation of the system is greatly simplified by the appropriate assumptions in the previous two methods, these approaches can be used for limited ranges of flows. In the numerical simulation (Bird, 1976a; Yen, 1971; Ziering, 1961), a very efficient computational scheme, i.e. DSMC (Direct Simulation Monte Carlo) method, has been developed. However, this method still suffers from the highly nonlinear behavior in the Boltzmann equation. Meanwhile, the use of different approach to solve even the same physical problem will encounter different difficulties due to the different advantages and limitations faced by each method. In addition, the predictions from the analysis should be confirmed by the experiments.
In the studies of numerical calculation, Beskok and Karniadkis (1994) have developed a scheme called "spectral element technique" to simulate the momentum and heat transfer processes of a rarefied gas subjected to either a channel-flow or an external-flow condition. The results have indicated that when the gas passes through a micro-channel at velocity-slip condition, it can cause a significant reduction in drag coefficient C_D on the walls. This is mainly caused by the thermal-creep effect when the Knudsen number increases significantly. Meanwhile, they have also addressed that the thermal-creep effect of the gas flow in a uniformly heated micro-channel can increase the mass flow rate, and the increase can be greatly enhanced by raising the inlet velocity. In addition, other effects, i.e. the compressibility and the viscous heating effects that may be occurred in the rarefied gas flow should also be considered. Chu et al. (1994) has used numerical analysis to evaluate the efficiency of heat removal when gas flows through an array of micro-channel under continuum or the velocity-slip condition. This numerical simulation is intended to study the cooling performance inside a micro-channel array that fabricated in a silicon chip. The numerical approaches have adopted the finite-difference methods incorporated with SOR (Successive over-relaxation) techniques to solve the problem with Neumann boundary conditions. The assumptions used include fully developed hydrodynamic condition, fully developed thermal condition and uniform heating on the bottom wall with the top wall well insulated. From the numerical results they have found that even though the temperature-jump causes decrease in Nusselt number that is contrary to continuum flow, the entire heat transfer performance were still higher than the case of continuum flow; this peculiar phenomenon is mainly due to the velocity-slip effects that induce greater mass flow per unit time into the channel. Therefore, the design of gas flow through a micro-channel array at the slip-flow regime as cooling is suggested. Fan and Xue (1998) have used the numerical method of the "DSMC" to simulate the gas flow in micro-channels at the slip-flow regime. They have assumed that the gas flow is simultaneously subjected to the effects of the velocity-slip and the compressibility. In addition, the effects of pressure ratio "P_o" between two ends of the micro-channel on the flow are also studied. Simulation analysis was carried out under different ratios of P_o, and the results indicated that the velocity-profiles of the flow near both ends of the channel are deviated from the parabolic profile. The mean flow velocity near the channel outlet increases greatly by increasing the ratio of P_o. The deviation

from the parabolic profile is caused mainly by both the entrance and the exit effect of the microchannel, only the flow field far from the end of the micro-channel can conform to the fully developed flow conditions. The second account of flow acceleration is not only significantly affected by the velocity-slip, but also induced by the compressibility of gas. Since the compressibility effect causes decrease in both the density and the pressure near the exit of channel, and the greater decrease in the exit pressure can accelerate the flow again to make up the density drop. Therefore, acceleration of the flow in a microchannel can be increased by increasing the pressure ratio P_o. Meanwhile the slip flow characteristics in the channel can be observed from the simulation results for the shear stress and velocity distributions near the wall region. The results further exhibit that the compressibility induced by the increase of P_o can greatly affect the gas flow behavior when the flow in the microchannel is at the slip-flow regime.

2.2 Experimental measurements

For experiments of gas flow in micro-channels, Wu and Little (1983) have measured the friction factors for both laminar and turbulent gas flows in trapezoidal channels. The widths of the channels are from 130 to 200 μm and the depths are from 30 to 60 μm, respectively. The working fluids used include nitrogen, helium and argon gases. The friction factors, f, obtained in his experiment are larger than the theoretical prediction for the critical Reynolds number less than 400. The deviations of the data form the prediction are attributed to the very high degree of surface roughness and measurement uncertainty. For a nitrogen gas flow in micro-tubes, the effects of wall surface roughness on the pressure drop or the friction factors are studied by Choi et al. (1991) for both laminar and turbulent flow. The micro-tube diameters are from 3 to 81 μm and the wall roughness is from 0.00017 to 0.0116. It is found that the Poiseuille number, Po, which is defined as f × Re, is 53 in the laminar region when the diameter of the tube is less than 10 μm. The Po of 53 in his experiment is lower than the theoretical value of 64 for fully developed laminar flow in the large scale tube, where the Po is kept as a constant. In the experiments of turbulent flow region, the results indicate that the Colburn analogy is not valid when the diameter of micro-tubes is less than 80 μm.

Some of pressure drop measurements have a good agreement with the predictions of the conventional theory. Acosta et al. (1985) has measured the friction factors in rectangular micro-channels, and the results are very close to the friction factor predicted by the conventional theory in small aspect ratios channels. Lalonde et al. (2001) has studied the friction factor of air flow in a micro-tube with a diameter of 52.8 μm. The experimental data has a good agreement with the predictions from the conventional theory. Turner et al. (2001) has performed an experiment to measure the friction factor with different working fluids, such as nitrogen, helium and air in microchannels with hydraulic diameters varying from 4 to 100 μm. The walls of the rectangular channels consider both the rough and the smooth wall conditions. The results indicate that the friction factors in laminar region for both the rough and the smooth wall conditions have good agreement with the conventional theory.

In contrast to the results that agree with the conventional theory, Pfahler et al. (1990a, 1990b) and Pfahler et al. (1991) have performed experiments to obtain the friction factor for working fluids of helium and nitrogen in micro-channels with the heights varying from 0.5 to 40 μm. The results indicate a significant reduction of C_f (Po_{exp}/Po_{theo}) which is a function of channel depth. The C_f decreases with decreasing Re in the smallest channel. Yu et al. (1995) has performed the experiments of gas flow in a micro-channel with either a trapezoidal or a rectangular cross section. The hydraulic diameter varies between 1.01 and

35.91 μm. They have observed a friction factor smaller than the prediction of the conventional theory, and conclude that the deviation may be caused by both effects of compressibility and rarefaction of the gas. Harley et al. (1995) has performed the experiments for subsonic, compressible flow in a long micro-channel. The working fluids used are nitrogen, helium and argon gases. The channels are fabricated by silicon wafer, and the dimensions of the channels are 100 μm wide, 10 mm long with depths varied from 0.5 to 20 μm. The experimental data have been presented in terms of the Po with hydraulic diameter from 1 to 36 μm. The measured friction factors agree with the theoretical prediction, but become smaller when the depth of channel decreases to 0.5 μm. The reduction in the friction factor is attributed to the occurrence of slip flow. The compressibility effects are also found by Li et al. (2000) who have performed an experiment of nitrogen gas flow in five different micro-tubes with diameters from 80 to 166 μm. The pressure drop along the tube became nonlinear when the Much number is higher than 0.3.

In order to understand more detailed pressure information inside a micro-channel, arrays of the pressure sensors should be integrated in the micro-channel for measurement of pressure distribution. Pong et al. (1994) are the first to present that a rectangular micro-channel can be fabricated with integrated arrays of pressure sensors for pressure distribution measurements. Both the helium and the nitrogen gas are used as the working fluid in his study. The channels are from 5 to 40 μm wide, 1.2 μm deep and 3000 μm long. The experimental results indicate that the pressure distribution is not linear and is lower than the prediction based on the continuum flow analysis in the micro-channel. The non-linear effects are caused by both effects of rarefaction and compressibility of the gas due to the high pressure loss. Liu et al. (1995) have used the similar channel as in Pong et al. (1994) but having different shapes to perform the experiments. The channel has a uniform cross section and has the dimensions of 40 μm wide, 1.2 μm deep and 4.5 mm long. The pressure drop distribution found is also nonlinear. For the channel with non-uniform cross section, sudden pressure changes are found at locations where variations of the cross section occur. In the mean time, analysis of the channel flow has also been performed with the assumptions of a steady, isothermal, and continuum flow with wall slip condition. However, the analysis can not explain the small pressure gradients measured near the inlet and the outlet of the channel.

Shih et al. (1996) has repeated the experiments of Pong by using a similar micro-channel with dimensions of 40 μm wide, 1.2 μm deep and 4000 μm long to measure the pressure distribution and mass flow rate for helium or nitrogen gas flow. The results of helium have a good agreement with the analysis based on the Navier-Stokes equations with slip boundary condition. The boundary condition of a slip flow on the wall is given by

$$u_w = \psi Kn(\partial u / \partial y) \tag{2-9}$$

where ψ is momentum accommodation coefficient. In general, $\psi = 1$ has been used for engineering calculation. All the experimental data indicate a non-linear dependence of the pressure drop with the mass flow rate. Li et al. (2000) and Lee et al. (2002) have performed experiments for channels with orifice and venture elements. The dimensions of channels are 40 μm wide, 1 μm deep and 4000 μm long. The working fluid used is nitrogen which has an inlet pressure up to 50 Psig. The mass flow rates are measured as a function of the pressure drop. The results indicate that the pressure distribution is non-linear and the pressure drop is a function of mass flow rate. The experimental data are used to compare with the

prediction from the Navier-Sotkes equation with a slip boundary condition. The friction factors for both channels with either the orifice or the venture are all lower than theoretical prediction.
It appears that contradictory results have been found in the previous studies. More accurate measurements of the pressure drop and heat transfer inside a microchannel are required. This requires fabrication of a micro-channel system, integrated with arrays of micro pressure sensors or temperature sensors, fabricated by surface micromachining process. However, the microchannel fabricated previously with arrays of pressure sensor is limited to a channel height of 1.2 μm due to the use of oxide sacrificial layer which is deposited by chemical vapor deposition (CVD) process. Much thicker deposition of the oxide layer is not possible with the current technology. In addition, the channel structure is very weak due to fabrication of the channel wall with a very thin film, only gas flow is allowed for the experiment. Therefore, in order to provide a channel which has a much greater height and is suitable for liquid flow conditions with a strong wall, an entirely new fabrication process for the channel should be considered and designed.

3. Liquid flow characteristics in microchannels

The liquid flow can be regarded as a continuum even in a very small channel. However, liquid flow can become boiling when the wall temperature is higher than the vaporization temperature of the liquid. Therefore, the liquid flow regimes can be divided into the single phase flow and the two phase flow regime. The real behaviors of heat transfer in the laminar or the transition flow (before turbulent) regime are deviated significantly from the prediction using the continuum theory due to the nonlinear terms of the surface forces in the Navier-Stokes equations. The surface forces play a major role in the micro-scale liquid flow, which can be significantly affected by the geometry, the electro-kinetic transport process, the hydrophilic or hydrophobic of the surface condition etc. inside the microchannel.

3.1 Experimental results

Single-phase liquid flow is considered incompressible in a micro-channel. However, the geometric configurations, such as the aspect ratio, the geometric cross-section of the channel or the surface roughness etc., can significantly affect the characteristics of the flow and the heat transfer process in a microchannel. Harms et al. (1997, 1999) have observed a friction factor well predicted by the conventional theory in the laminar region. Webb et al. (1998) have observed that the conventional theory is able to predict the single phase heat transfer and the friction factor for a rectangular channel. Pfund et al. (1998) have studied the water flow in rectangular micro-channels at Reynolds numbers between 40 and 4000. The friction factor has a good agreement with the conventional theory in the laminar flow region, but increase by the surface roughness in the turbulence region. Xu et al. (1999, 2000) have fabricated the rectangular micro-channels by bonding an aluminum plate or a silicon wafer with a Plexi glass. The channels were etched on a silicon or aluminum substrate. The hydraulic diameters of the aluminum channels are from 46.8 to 344.3 μm and for silicon channels are from 29.59 to 79.08 μm, respectively. The experimental results for liquid flow in micro-channels have very good agreement with the prediction from the Navier-Stokes equation for a Newtonian flow in laminar region. Qu et al. (2000, 2002) has performed experiments for water in silicon micro-channels with trapezoidal cross section having hydraulic diameter from 51 to 169 μm. The pressure drop measured has a good agreement with the prediction based on conventional theory. More experiments have indicated that the deviation from the prediction is attributed to the roughness of the channel wall and viscosity of the fluid. The friction factors obtained from these experiments are higher than the predictions from the conventional theory. Li et al. (2000, 2003) have fabricated different micro-tubes made by glass, silicon or stainless steel with diameters ranging from 79.9 to 166.3 μm, 100.25 to 205.3 μm and from 128.6 to179.8 μm, respectively. The results of the friction factor measured for DI water, in glass and silicon micro-tubes where tube wall can be considered smooth, has good agreement with the conventional theory. The deviation of the data in the stainless steel tube is attributed to the surface roughness. They have concluded that the relative roughness of the wall can not be neglected for micro-tube in the laminar flow region. Sharp et al. (2000) have considered laminar flow of water in micro-tubes with hydraulic diameters ranging from 75 to 242 μm. Their data agree with the conventional theory. Wu et al. (2003) have provided the experimental data of friction factor for DI water in smooth silicon micro-channels with trapezoidal cross section having hydraulic diameter from 25.9 μm to 291 μm. The results of their data have a good agreement with the prediction from the conventional theory. They conclude that the Navier-Stokes equations are still valid for laminar flow of DI water in microchannel with smooth wall and hydraulic diameters as small as 26 μm.

Some work reported the friction factors that are very different from the theoretical prediction. Yu et al. (1995) has performed experiments of water flow in silica micro-tubes with diameters ranging from 19 to 102 μm and the Reynolds numbers between 250 and 20000. The friction factors are lower than the theoretical predictions. Jiang et al. (1995, 1997) have studied water flow through rectangular or trapezoidal channels. The dimensions of the channels are 35 to 120 μm wide and 13.4 to 46 μm deep. The friction factor data are greater than the theoretical prediction, but become lower when the Reynolds numbers are between 1 and 30. It appears that the deviations of the friction factor measured from the prediction may be attributed to the surface behaviors of the liquid flow, especially the surface roughness of the channel wall, the surface potential and the electro-kinetic effect induced by the electrical double layer (EDL) etc. as discussed in the following section.

3.2 Analysis of electric double layer effect

If the liquid contains a very few amount of ions (ex. impurities), the electrostatic charges on the non-conducting solid surface will attract the counter-ions in the liquid flow. The rearrangement of the charges on the solid surface and the balancing charges in the liquid is called the electrical double layer. The thickness of the EDL is significantly affected by the ion concentration, the liquid flow polarity, the surface roughness and the surface potential. A thicker EDL possibly induced by a lower ion concentration, a polar liquid, a poor surface roughness or a higher surface potential could cause a larger friction factor and pressure gradient. This can significantly reduce the flow velocity, and the heat transfer of a liquid flow in the microchannel. This is true for infinitely diluted solution such as the millipore water, the thickness of the EDL is considerably large (about 1 μm). However, for solution with high ionic concentration, the thickness of the EDL becomes very small, normally a few nanometer. In this case, therefore, the EDL effects on the flow in microchannels can be negligible.

To account for the EDL effect for polar liquid flow in the microchannel, most of the work performed in the past is the theoretical simulation where the physical models can be formulated based on (1) the Poisson-Boltzmann equations for the EDL potential, (2) the

Laplace equations with the applied electrostatic field, and (3) the Navier-Stokes equations modified to include effects of the body force due to the interaction between electrical and zeta potential. However, the numerical results are always lower than the empirical data due to the unusual and complex surface behaviors described above. In addition, the aspect ratio and the geometric cross-section of the channels can also affect the thickness of the EDL. In general, the friction factor increases with decreasing the aspect ratio of the channels. A microchannel with a cross section of circular shape usually has the lowest friction factor. The friction factor in a silicon channel is larger than in a glass channel due to the different surface potential of the channel walls with millipore water.

The Poisson-Boltzmann equations for the EDL potential in a rectangular microchannel are described as follows (Beskok & Karniadakis, 1994):

$$\frac{\partial^2 \psi}{\partial y^2} + \frac{\partial^2 \psi}{\partial z^2} = -\frac{\rho_e}{\varepsilon_o \varepsilon} = \frac{2 n_\infty z e}{\varepsilon_o \varepsilon} \sinh(\frac{z e \psi}{k_b T}) \tag{3-1}$$

$$n_i = n_{i\infty} \exp(-\frac{z_i e \psi}{k_b T}) \tag{3-2}$$

$$\rho_e = ze(n_+ - n_-) = -2zen_\infty \sinh(\frac{ze\psi}{k_b T}) \tag{3-3}$$

where ψ and ρ_e are the electrical potential and the net charge density per unit volume. ε is the dielectric constant of the solution. ε_o is the permittivity in vacuum. $n_{i\infty}$ and z_i are the bulk ionic concentration and the valence of type i ions, respectively; e is the charge of the proton; k_b is the Boltzmann constant; T is the absolute temperature.

To account for the electric field effect, the Navier-Stokes equation describing the flow motion can be rewritten as following:

$$\frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} = \frac{1}{\mu}\frac{dp}{dx} - \frac{1}{\mu} E_x \rho_e \tag{3-4}$$

where E_x is an induced electric field (or called electrokinetic potential) and p is the hydraulic pressure in the rectangular microchannel.

At a steady state, the net electrical current is zero, which means:

$$I = I_s + I_c = 0 \tag{3-5}$$

$$I_s = 4\int_{h-1/k}^{h} \int_{w-1/k}^{w} u(y,z)\, \rho_e(y,z)\, dy\, dz \tag{3-6}$$

where I_s and I_c are the streaming and the conduction currents, respectively. In addition, the net charge density is non-zero essentially only in the EDL region whose characteristic thickness is given by $1/k$ (k is the Debye-Huckel parameter).

The conduction current, that is the transport of the excess charge in the EDL region of a rectangular microchannel, driven by the electrokinetic potential is given by:

$$I_c = 4\lambda_o E_x (h+w)\frac{1}{k} \tag{3-7}$$

$$k = [2z^2e^2n_\infty / (\varepsilon\,\varepsilon_o k_b T)]^{1/2} \tag{3-8}$$

where λ_o is the bulk electrical conductivity (1/Ω m). h and w are the height and the width of the microchannel, respectively. Substituting Eq.(3-6) and Eq.(3-7) into Eq.(3-5), the electrokinetic potential (E_x) can be written as follows:

$$E_x = -\frac{\int_{h-1/k}^{h}\int_{w-1/k}^{w} u(y,z)\,\rho_e(y,z)\,dy\,dz}{\lambda_o (w+h)(1/k)} \tag{3-9}$$

Both the Poisson-Boltzmann equation, Eq.(3-1) and Navier-Stokes equation, Eq.(3-4), can be solved numerically in order that both the EDL and the velocity fields in the rectangular microchannel can be determined.

3.3 Comparison with the data

Despite the theoretical prediction, some work presents occurrence of the electrical double layer of water flow in a micro-channel. Ren et al. (2001) have performed experiments to measure the interfacial electrokinetic effects of a liquid flow through rectangular silicon micro-channels with diameters of 28.1, 56.1 and 80.3 μm. Both the DI water and the KCl solutions with two different concentrations of 10-4 and 10-2 M are used as working fluid. The measured pressure drops for the pure DI water and the lower KCl concentration solution are significantly higher than that for higher concentration solution and the theoretical prediction. The authors have concluded that a significant increase in the friction factor is attributed to occurrence of the electrical double layer (EDL) which increases the pressure drop in the small micro-channels. Similar results have also been obtained by Li et al. (2001).

To compare with the experimental results, the analytical predictions for both the flow and the heat transfer developed from continuum assumption indicate large discrepancy when the characteristic length of the micro-channel becomes small enough. In the studies of liquid flow, many investigators (Ren et al., 2001; Fan et al., 1998; Chen, 1996; Chu et al., 1994; Choi et al., 1991; White et al., 1991; Pfahler et al., 1990, 1991) have concluded that even though the liquids can be regarded as a continuum in a very small system, the real behaviors of heat transfer at the laminar or the transition (before turbulent) condition are deviated from the predictions based on the conventional theory. Usually, for the data published, the uncertainties of flow rate measured and friction factor estimated are 2-5 % and 10-15 %, respectively. For most heat transfer studies, the uncertainties are under ± 20 %. In summary, the geometric effects, such as the aspect ratio, the cross-section shape or the surface roughness etc., can significantly affect the characteristics of both the flow and the heat transport in a microchannel. The onset of transition to turbulent flow in smooth microchannels does not occur if the Reynolds number is less than 1000. For a laminar flow, the Nusselt number varies as the square root of the Reynolds number. In turbulent flow, however, the numerical studies are not applicable and thus many empirical correlations have been proposed, but were not verified. However, satisfactory estimates of the heat

transfer coefficients can be obtained with sufficient accuracy by using either experimental results in smooth channels with large hydraulic diameter or conventional correlations.
Tso and Mahulikar (1998) have obtained the heat transfer for laminar liquid flow through a microchannel in both the thermal-developing region and the thermal-developed region. It is found that the Nusselt number decreases with increasing the Reynolds number not only in the thermal-developed region, but also in the thermal entry region. The results also indicate that the pressure distribution along the microchannel exhibits a non-linear profile. Despite much of the studies has addressed that the liquid flow appears a greatly complicated relation between Nusselt number and Reynolds number, however, all the results are very based on the assumption of continuum flow. Therefore, more detailed analysis combined with experiments is still required to clarify the role of the EDL and different results among different works.

3.4 Two-phase flow phenomenon in the microchannel

The two-phase flow or flow-boiling phenomenon in the microchannel exhibits some unusual characteristics. It is found that the bubbles are not rapidly generated even at a very high heat flux from the heated microchannel (Qu et al., 2000). Therefore, further experimental investigations on the flow boiling in microchannels were made by others (Ren et al., 2001; Peng & Wang, 1993; Lin and Pisano, 1991, 1994). In addition, the effect of microchannel scale, geometric configuration, liquid velocity, liquid sub-cooling and liquid concentration on the flow boiling were investigated. It is found that the heat transfer enhanced by a large more volatile component concentration is greater than the pure more volatile liquid. The heat transfer coefficient at the onset of flow boiling and in the partial nucleate boiling region was greatly influenced by the liquid concentration, the geometric configuration, the size of microchannel, and the flow velocity and sub-cooling, but not in the fully nucleate boiling region. Peng and Wang (2001), and Hu (1998) found the so-called "bubble extinction" behavior due to an induced vigorous nucleate boiling mode on a normal-sized heater or abnormal-sized channels. The normal bubbles could not successfully grow and form, if the channel height is less than a critical liquid space required. In order to interpret the unusual behavior observed in microchannel boiling, Peng and Wang (1994) proposed the concepts of "evaporating space" and "fictitious boiling". In fact, the small bubbles that can form initially in microchannel will eventually collapse since the size of the bubble could not grow up exceed the critical radius of bubble (r_c) formulated by conventional nucleation theory. The fictitious boiling occurred was attributed to the crowded tiny bubbles that grow and then collapse rapidly in a cyclic manner, and thereby mimicking a boiling state that can transfer large amount of heat. The observations suggest that close to bubble nucleation temperature the liquid will vigorously oscillate in the microchannel due to the emergence of tiny bubble embryos. More detailed explanations are given in (Jiang et al., 2001; Peng et al., 1998).
The experiments by Peng and Wang (1993) for flow boiling of water have been carried out in a stainless steel microchannel with rectangular cross-section of 600 μm×700 μm. In a much smaller channel array, with hydrodynamic diameter of 40 and 80 μm, made on a silicon substrate by wet etch, three stable phase-change modes, i.e. local nucleation boiling, large bubble formation and annular flow, were observed depending on the input power level (Qu & Mudawar, 2003). However, bubbly flow, commonly observed in macrochannels, could not be developed in the microchannels. A stable annual flow was also observed in a micro-

channel heat sink contained 21 parallel channels having a 231 μm × 713 μm cross-section (Lee et al., 2003).

Lee et al. (2003) proposed that a nearly rectangular microchannel heat sink with 14 μm in depth integrated with a local heater and array of temperature sensors on silicon substrate was made to investigate the size and shape effects on the two-phase patterns in microchannel forced convection boiling. It is found that when the heat input power increases, the downstream movement of the transition region increases the void fraction and causes a lower devices temperature. However, at the high flow rate, the transition region almost occupies the entire channel, the increase in the heat input power results in a higher devices temperature. An annular pattern induced by flow boiling appears stably in triangular microchannels, but not in rectangular microchannels. Two-phase boiling or superheated flow has numerous promising applications such as in cooling of electronic components. The principle advantage of two-phase flow lies in the utilization of latent heat absorbed by the working fluid due to phase change from liquid to vapor without increasing the flow fluid temperature. In fact, two-phase flow heat transfer in microcahnnel is a very important and interesting problem indeed.

However, much of the attention at later time has been given to the study of dynamic flow boiling instability in microchannels (Cheng et al., 2009; Wang et al., 2008; Wang et al., 2007; Kandlikar, 2006; Wu & Cheng, 2003, 2004; Brutin et al., 2003; Hetsroni et al., 2002; Hetsroni et al., 2001). A periodic annular flow and the periodic dry steam flow were observed for boiling of water in 21 silicon triangular microchannels having a diameter of 129 μm in (Hetsroni et al., 2001, 2002). However, two types of two-phase hydrodynamic instabilities, i.e. severe pressure drop oscillation and mild parallel channel instability were identified (Qu & Mudarwar, 2003) in the similar microchannels as in other work (Hetsroni et al., 2001). A simultaneous flow visualization and measurement was made on flow boiling of water in two parallel silicon microchannels of trapezoidal cross-section having hydraulic diameters of 158.8 μm and 82.8 μm, respectively (Wu & Cheng, 2003). The results shows that two-phase flow and single-phase liquid flow appear alternatively in microchannels, which leads to large amplitude/long-period fluctuations with time in temperatures, pressures and mass flux. The flow pattern map in terms of heat flux versus mass flux showing stable and unstable flow boiling regimes in a single microchannel has been identified (Wu & Cheng, 2004). It is found that stable and unstable flow-boiling modes existed in microchannels, depending on four parameters, namely, heat/mass flux ratio, inlet water subcooling, channel geometry, and physical properties of the working medium (Wang et al., 2007). In addition, the magnitudes of temperature and pressure fluctuations in the unstable flow-boiling mode depend greatly on the configurations of the inlet/outlet connections with the microchannels (Wang et al., 2008). By fabricating an inlet restriction on each microchannel or the installation of a throttling valve upstream of the test section, reversed flow of vapor bubbles can be suppressed resulting in a stable flow-boiling mode. Based on the exit quality of the flow from a microchannel, more detailed flow regimes are identified (Cheng et al., 2009).

In the past, however, a very important issue, i.e. the surface wettability effect, has been overlooked in the study of boiling flow heat transfer in a microchannel. The boiling flow phenomenon found in the microchannel is only for certain surface wettability. By changing the material of the microchannel or surface wetting property, the boiling flow phenomenon may be completely different. This may cause discrepancy of flow patterns observed in different channels made by different materials. Phan et al. (2009) have found that the wettability of a surface has a profound effect on the nucleation, growth and detachment of bubbles from the bottom wall in a tank. For hydrophilic (wetted) surfaces, it has been found that a greater surface wettability increases the vapor bubble departure radius and reduces the bubble emission frequency. Moreover, lower superheat is required for the initial growth of bubbles on hydrophobic (un-wetted) surfaces. However, the bubble in contact with the hydrophobic surface cannot detach from the wall and have a curvature radius increasing with time. At higher heat flux, the bubble spreads over the surface and coalesces with bubbles formed at other sites, causing a large area of the surface to become vapour blanketed.

The wettability of channel surface has been studied by Liu et al. (2011) who have fabricated three different microchannels with identical sizes at 105 x 1000 x 30000 μm but at different wettability. The microchannels were made by plasma etching a trench on a silicon wafer. The surface made by the plasma etch process is hydrophilic and has a contact angle of 36^o when measured by dipping a water droplet on the surface. The surface can be made hydrophobic by coating a thin layer of low surface energy material and has a contact angle of 103^o after the coating. In addition, a vapor-liquid-solid growth process was adopted to grow nanowire arrays on the wafer so that the surface becomes super-hydrophilic with a contact angle close to 0^o. Different boiling flow patterns on a surface with different wettability were found, which leads to large difference in temperature oscillations. Periodic oscillation in temperatures was not found in both the hydrophobic and the super-hydrophilic surface. During the experiments, the heat flux imposed on the wall varies from 230 to 354.9 kW/m^2 and the flow of mass flux into the channel from 50 to 583 kg/m^2s. Detailed flow regimes in terms of heat flux versus mass flux are also obtained.

4. Basic MEMS fabrication techniques

4.1 Chemical vapor deposition

Chemical vapor deposition (CVD) is a typical technique to fabricate a thin film on a substrate. In a CVD process, gaseous reactants are introduced into a heated reaction chamber. The chemical reactive gases diffuse onto and absorbed by the substrate. Then thermal dissolution reaction of the reactive gases occurs which lead to deposition of a thin solid film on the heated substrate surfaces. Depending upon the relative pressure and the temperature used the CVD processes are categorized as: (1) the atmospheric pressure chemical vapor deposition (APCVD), (2) the low pressure chemical vapor deposition (LPCVD), and (3) the plasma-enhanced chemical vapor deposition (PECVD). The process temperatures of APCVD and LPCVD are ranged from 500^oC to 850^oC. In PECVD processes, a part of thermal energy is shared from the plasma source. Therefore, the process temperatures of the PECVD are lower on the order of 100^oC to 350^oC. The silicon based thin films such as poly-silicon, amorphous silicon, silicon dioxide, tetraethoxysilane (TEOS, $Si(C_2H_5O)_4$) or silicon nitride film can be fabricated by using the CVD process. The chemicals used and the reaction occurred in the CVD process for different kinds of films are listed in Table 1. The poly-silicon film can be used for fabrication of pressure or temperature sensors or micro-heaters. The TEOS oxide layer is fabricated as insulator between each sensor layer. In addition, deposition of the silicon nitride film can be used to prevent penetration of moisture into the sensors during liquid flow experiments which may cause damage of the micro-sensors or micro electronics integrated in the micro-channel.

Films	Chemical reactions
Poly-silicon	$SiH_4 \rightarrow Si + 2\ H_2$
Silicon dioxide	$SiH_4 + O_2 \rightarrow SiO_2 + 2\ H_2$ $SiCl_2H_2 + 2\ N_2O \rightarrow SiO_2 + 2\ N_2 + 2\ HCl$
TEOS (tetraethoxysilane)	$Si(OC_2H_5)_4 \rightarrow SiO_2$ + by-products
Silicon nitride	$3\ SiH_4 + 4\ NH_3 \rightarrow Si_3N_4 + 12\ H_2$ $3\ SiCl_2H_2 + 4\ NH_3 \rightarrow Si_3N_4 + 6\ HCl + 6\ H_2$

Table 1. Chemical reactions used in the CVD process for different kinds of films.

4.2 Evaporation and sputtering deposition

Both evaporation and sputtering deposition are classified as physical vapor deposition (PVD) process which can form different kinds of films on a substrate directly from a source material. PVD is typically used for deposition of electrically conducting layers such a metal or silicide. Evaporation deposition of a thin film on a substrate is done by sublimation of a heated source material in a vacuum chamber. The vapor flux from the source can be condensed and coated on the substrate surface. The evaporation methods can be further categorized as the vacuum thermal evaporation (VTA), the electron beam evaporation (EBE), and the molecular beam epitaxy (MBE).

The simplest evaporator consists of a vacuum chamber with a crucible which can be heated to a high temperature, as shown in Figure 1(a) and 1(b) by a filament. The filament is used as a heater, which is made of Tungsten (W), a refractory (high temperature) metal. Evaporation is accomplished by gradually increasing the temperature of the filament until the source material melts. Filament temperature is then further raised to evaporate the source material from the crucible. The substrates are mounted on top of the crucible and are deposited with a thin film of evaporated material.

In the electron beam (E-beam) evaporation system, the high-temperature filament is replaced with an electron beam, as shown in Figure 1(c). A high-intensity beam of electrons, with energy up to 15 keV, is focused on the source material to be evaporated in a crucible. The energy from the electron beam only melts a portion of the source material, which eventually evaporates and condenses on the substrate to form a thin layer.

Sputtering deposition requires generation of plasma gas between high voltage electrodes, as shown in Figure 2, where positively ions can be accelerated and bombards on a target material (a cathode) so that flux of atoms can be sputtered and collected on the substrate. Usually, a physically inert gas, such as argon gas, is made into plasma by knocking out electrons of the molecules with high speed electrons emitted from the cathode. The sputtering deposition has the advantages of depositing various materials include not only for pure materials or metals, but also for compounds, alloys, refractory materials, or piezoelectric ceramics. In addition, puttering deposition has no shadowing effect as that occurred in evaporation deposition, which causes non-uniform deposition of a film. Therefore, sputtering deposition has been widely used for deposition of different kinds of films.