A Heat Transfer Model for Thermal Fluctuations in an Ultra-Thin Air Bearing

Shuyu Zhang and David B. Bogy

Computer Mechanics Laboratory, University of California, Berkeley, CA 94720

Abstract

In this report, the momentum and energy equations are solved by applying, respectively, the slip and jump boundary conditions for velocity and temperature for an ultra-thin air bearing. With the temperature distribution known, the heat flux between the slider and the air bearing is obtained using Fourier's Law. It is found that the heat flux contains contributions from both heat conduction, which transfers heat from the slider to the air bearing when the slider has a higher surface temperature than the disk, and viscous dissipation, which transfers heat from air bearing to the slider. Whether an air bearing is a "coolant" or "heater" depends on which part, the heat conduction or viscous dissipation, dominates the heat transfer. Since the magnitude of viscous dissipation is relative small, the "heating" effect often plays a weaker role unless the temperature difference between the slider and disk is very nearly equal to zero. Simulation results also show that the heat conduction effect increases with the decrease of the flying height (or disk rotation speed), but the viscous dissipation effect decreases with the decrease of the flying height (or disk rotation speed). In other words, the "cooling" effect increases with the decrease of the flying height (or disk rotation speed).

Key words: heat transfer, air bearing, ultra-thin film.

1 Introduction

The flash temperature rise nearby the MR element due to a head/disk contact leads to a thermally induced disturbance in read-back signals. This phenomenon is referred to thermal asperities, and it can cause serious problems in data reading of a MR head. When a MR head flies very close to the disk surface, another type of thermally induced problem occurs in the MR signal as a result of flying height fluctuation (Tian, H., et al., 1997). Their experimental results showed that the air bearing has a cooling effect on the slider and makes the major contribution to this disturbance. In this report, we conduct a theoretical study of the heat transfer between a slider and the air bearing to find the mechanism of the "cooling" effect of the air bearing.

One difficulty in solving the heat transfer problem between a slider and the air bearing is that the traditional theory, which is based on the continuum assumption, is no longer valid when the air bearing is ultra-thin. For example, the flying height of a typical MR head is around 50 *nm* in today's hard disk drive, which means the Knudsen number $Kn=\lambda/h$, where λ is mean free path of the air and h is the spacing, is around 1. Air flow with such a Knudsen number is usually considered in the slip or transition regimes (Schaaf, S.L., et al., 1963). To solve the heat transfer problems in these regimes, we need to consider some special approaches such as the Boltzman equation or velocity slip and temperature jump conditions (Kennard, E.H., 1938). These methods have been used previously in solving for the velocity distribution in an air bearing by several researchers (Burgdorfer, A., 1959; Gans, R., 1985; Fukui, et al., 1988).

Another difficulty is that the continuity equation, momentum equation and energy equation need to be solved simultaneously because the physical properties of air depend on the temperature, which usually makes the problem more complicated, and leads to the need of more computation time in the numerical analysis. A simple approach is to assume the properties are constant if the temperature variation is not too great, so we can evaluate the properties at a certain reference temperature, say the average temperature of the two surfaces. With such an approximation, the momentum and energy equations can be decoupled for solution. Since the temperature difference between the slider and disk surfaces is expected to be very small, it is reasonable to apply a constant property assumption in an air bearing. Thus, we can solve the momentum and energy equations separately.

In this report, we first simplify the momentum and energy equations by dimensional analysis. Then we solve the reduced momentum equation to get the velocity distribution and solve the energy equation to get the temperature distribution in the air bearing. Using Fourier's law, we obtain an expression for heat flux between the slider and air bearing. A computer program is implemented to simulate the heat flux for several cases. The slider/disk system as well as the related coordinate system used in the analysis are shown in the Fig.1

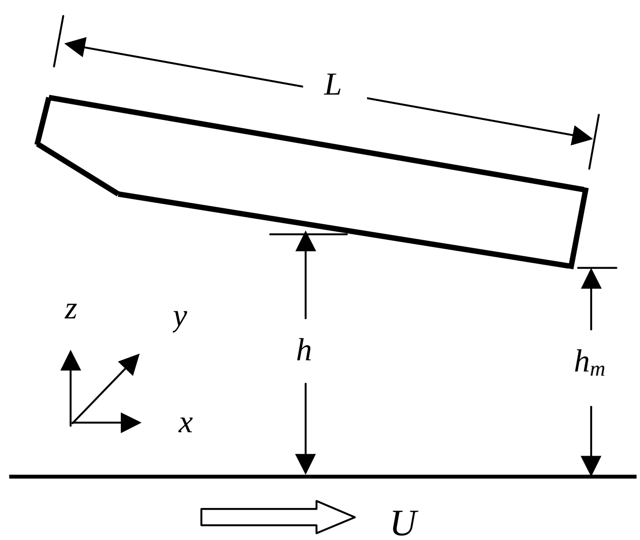


Fig.1 Slider/disk system and coordinates

2. Governing Equations in the Air Bearing

In the following analysis we focus on the steady case, so the time dependent terms in the related equations disappear. Using dimensional analysis, we reduce these equations to simpler forms.

(1) *Navier-Stokes (N-S) equation*:

The simplification of the N-S equation in an air bearing has been performed by many researchers. Here we only list the final results and do not present the detailed derivation (Gross, W.A., et al., 1980):

$$\frac{\partial p}{\partial x} = \frac{\partial}{\partial z}\left(\mu \frac{\partial u}{\partial z}\right), \tag{1-a}$$

$$\frac{\partial p}{\partial y} = \frac{\partial}{\partial z}\left(\mu \frac{\partial \upsilon}{\partial z}\right), \tag{1-b}$$

$$\frac{\partial p}{\partial z} = 0\,, \tag{1-c}$$

where u, υ are velocities in the x and y directions, p is the pressure and μ the viscosity of the air. Velocity w in the z direction is approximated to be zero. Clearly, pressure p remains constant across the thickness of the air bearing.

(2) *Energy equation*:

As in the N-S equation, the energy equation can also be simplified by using dimensional analysis in the air bearing. Assuming that the magnitudes $|\partial/\partial x| \sim |\partial/\partial y| << |\partial/\partial z|$, we write the energy equation as:

$$\rho c_p u \frac{\partial T}{\partial x} + \rho c_p \frac{\partial T}{\partial y} - u \frac{\partial p}{\partial x} - \upsilon \frac{\partial p}{\partial y} = \frac{\partial}{\partial z}\left(k \frac{\partial T}{\partial z} \right) + \mu \left(\frac{\partial u}{\partial z} \right)^2 + \mu \left(\frac{\partial \upsilon}{\partial z} \right)^2, \qquad (2\text{-a})$$

$$\underset{(I)}{|} \quad \underset{(II)}{|} \quad \underset{(III)}{|} \quad \underset{(IV)}{|} \quad \underset{(V)}{|} \quad \underset{(VI)}{|} \quad \underset{(VII)}{|}$$

where ρ is the density, c_p is the specific heat, k is the heat conductivity and T is the temperature of the air. To get this equation, we set $i=c_pT$ as the enthalpy of the flow.

As in simplifying the N-S equation, we use the characteristics of the air bearing to reduce the energy equation (2-a). Let's first look at the convection term (I) and conduction term (V). If we assume the magnitudes of $u \sim U$, $T \sim T_0$, $x \sim L$ and $z \sim h$, where U is the disk velocity, T_0 is the reference temperature (say the ambient temperature), L is the length of the slider and h is the thickness of the air bearing, then we can express (I)/(V) as $\rho c_p U h^2/Lk \sim PrRe(h/L)$, where Pr is the Prandtl number defined by $Pr=\mu c_p/k$ and Re is the Reynolds number defined as $Re=Uh/v$. For a typical air bearing, we can take $\rho \sim 1\ kg/m^3$, $c_p \sim 103\ W{\cdot}s/kg{\cdot}K$, $k \sim 0.03\ W/m{\cdot}K$, $U \sim 15\ m/s$, $L \sim 2\ mm$, and $h \sim 50\ nm$. Thus (I)/(V)$\sim 10^{-7}$ or the convection term (I) is much smaller than the conduction term (V), and therefore it can be neglected in the energy equation. Similarly, the convection term (II) is also negligible.

In a similar way, (VI)/(V)$\sim \mu U^2/k\Delta T$ where ΔT is the temperature difference between the slider and disk surfaces. If we take $\mu \sim 10^{-5}\ kg{\cdot}m/s$, $\Delta T \sim 10\ °K$, then (VI)/(V)$\sim 10^{-2}$. This implies that if ΔT appears in magnitude of 10 °K or smaller, the viscous dissipation term (VI) is comparable to the conduction term (V). Therefore, in the following analysis, we keep the viscous dissipation terms in the energy equation.

If we finally neglect terms (III) and (IV) because (III)/(V)~$p_0h^2/\mu UL$~$(p_0/\rho U^2)(h/L)Re$ ~10^{-3} by taking p_0~10^5 $kg/m\cdot s^2$, where p_0 is the ambient pressure, we can reduce the energy equation to:

$$\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\mu\left(\frac{\partial u}{\partial z}\right)^2+\mu\left(\frac{\partial \upsilon}{\partial z}\right)^2=0 \quad . \qquad \text{(2-b)}$$

In this reduced energy equation, the convection terms disappear and only the conduction and viscous dissipation terms remain. Strictly speaking, this equation is valid only when $PrRe(h/L)<<1$, $(p_0/\rho U^2)(h/L)Re<<1$ and $h/L<<1$. Fortunately, these conditions are usually satisfied in a slider/disk air bearing.

(3) *Boundary conditions*:

We assume that the disk has a non-zero velocity U in the x direction and zero velocity V in the y direction, which is the case of a slider flying at a middle radius of the disk. As for the temperature, considering the disk has a much larger size than the air bearing and rotates with high speed, we assume that it has a constant and uniform surface temperature that is the same as that of the ambient air flow. We also assume the slider's surface temperature is uniform. Introducing the slip condition for the velocity and the jump condition for the temperature at the boundaries of the air bearing for the cases of Kn around 1 (Schaaf, S.L., et al., 1963; Kennard, E.H., 1938), we can write the boundary conditions for velocity and temperature as (1st order):

$$u(0)=U+\frac{2-\sigma}{\sigma}\lambda\left.\frac{\partial u}{\partial z}\right|_{z=0} , \qquad \text{(3-a)}$$

$$u(h) = -\frac{2-\sigma}{\sigma}\lambda\frac{\partial u}{\partial z}\bigg|_{z=h}, \tag{3-b}$$

$$\upsilon(0) = \frac{2-\sigma}{\sigma}\lambda\frac{\partial \upsilon}{\partial z}\bigg|_{z=0}, \tag{3-c}$$

$$\upsilon(h) = -\frac{2-\sigma}{\sigma}\lambda\frac{\partial \upsilon}{\partial z}\bigg|_{z=h}, \tag{3-d}$$

$$T(0) = T_d + 2\frac{2-\alpha}{\alpha}\frac{\gamma}{\gamma+1}\frac{\lambda}{Pr}\frac{\partial T}{\partial z}\bigg|_{z=0}, \tag{3-e}$$

$$T(h) = T_s - 2\frac{2-\alpha}{\alpha}\frac{\gamma}{\gamma+1}\frac{\lambda}{Pr}\frac{\partial T}{\partial z}\bigg|_{z=h}, \tag{3-f}$$

where σ is the momentum accommodation coefficient and α is the thermal accommodation coefficient, γ is the ratio of c_p to c_v which are specific heats at, respectively, constant pressure and constant volume, T_s and T_d are , respectively, the slider surface temperature and disk surface temperature. For convenience, we write $a=(2-\sigma)/\sigma$ and $b=2(2-\alpha)\gamma/\alpha(\gamma+1)Pr$ in the following analysis.

3. Heat Transfer between the Slider and the Air Bearing

To get the heat transfer in the air bearing, we need to know the temperature distribution in it. This requires us to solve the N-S equation and the energy equation. Because of the approximation of constant properties of the air, we can decouple the N-S and the energy equations and solve them separately.

(1) *Velocity distribution*:

The velocity distribution can be obtained by integrating the reduced N-S equations (1-a)~(1-b) and applying the boundary conditions (3-a)~(3-f). The procedure is straight forward and was done by other researcher (Burgdorfer, A., 1959; Mitsuya, Y., 1993). Here we list the results of the solution:

$$u = -\frac{1}{2\mu}\frac{\partial p}{\partial x}\left(a\lambda h + hz - z^2\right) + U\left(1 - \frac{z + a\lambda}{h + 2a\lambda}\right), \qquad \text{(4-a)}$$

$$\upsilon = -\frac{1}{2\mu}\frac{\partial p}{\partial y}\left(a\lambda h + hz - z^2\right). \qquad \text{(4-b)}$$

In the RHS of equation (4-a), the first term is the contribution of the Poiseuille flow and the second term is the contribution of the Couette flow, while in (4-b) only the Poiseuille flow result is involved because we take the *y*-component of disk velocity *V*=0. Clearly, these results are not complete because we still do not know the pressure gradient in the *x* and *y* directions. To finish the solution we need to solve the Reynolds equation, which requires the integration of the continuity equation (Burgdorfer, A., 1959; Fukui, et al., 1988), to obtain the pressure distribution first. To get the solution, a numerical method is required (Cha, E.T., et al., 1995; Lu, S., et al., 1994).

(2) *Temperature distribution*:

We substitute the velocity solutions (4-a), (4-b) into the energy equation (2-b) and integrate it to obtain the temperature distribution in the air bearing:

$$T = T_d - \frac{1}{k}\left\{\frac{1}{12\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right]z^4 - \frac{1}{3}\left\{\frac{h}{2\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right] + \frac{\partial p}{\partial x}\frac{U}{h + 2a\lambda}\right\}z^3 + \right.$$

$$\left.\frac{\mu}{2}\left[\left(\frac{h}{2\mu}\frac{\partial p}{\partial x} + \frac{U}{h + 2a\lambda}\right)^2 + \frac{h^2}{4\mu^2}\left(\frac{\partial p}{\partial y}\right)^2\right]z^2\right\} + \left\{\frac{T_s - T_d}{h + 2b\lambda} + \right.$$

$$\frac{1}{k}\left[\frac{h^3}{24\mu}\left\{\left(\frac{\partial p}{\partial x}\right)^2+\left(\frac{\partial p}{\partial y}\right)^2\right\}+\frac{\mu U^2 h}{2(h+2a\lambda)^2}+\frac{Uh^3}{6(h+2a\lambda)(h+2b\lambda)}\frac{\partial p}{\partial x}\right]\Bigg\}(z+b\lambda). \qquad (5)$$

As in the velocity solutions, the temperature T also consists of contributions from the Poiseuille flow and Couette flow. In addition, extra terms exist which are the combined effects of both Poiseuille flow and Couette flow.

(3) *Heat transfer*:

Using Fourier's Law $q=-k\partial T/\partial z$ at $z=h$, and the temperature solution (5), we can obtain the heat transfer into the slider as follows:

$$q=-k\frac{T_s-T_d}{h+2b\lambda}+\frac{h^3}{24\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2+\left(\frac{\partial p}{\partial y}\right)^2\right]+\frac{\mu U^2 h}{2(h+2a\lambda)^2}-$$

$$\frac{Uh^3}{6(h+2b\lambda)(h+2a\lambda)}\frac{\partial p}{\partial x}. \qquad (6\text{-a})$$

We can also write the heat transfer equation (6-a) in a non-dimensional form as:

$$\frac{qh}{\mu U^2}=-\frac{T_s-T_d}{\left(\frac{\gamma-1}{2}\right)PrM^2T_0\left(1+2b\frac{\lambda}{h}\right)}+\frac{1}{2\left(1+2a\frac{\lambda}{h}\right)^2}+$$

$$\frac{1}{24}Re^2\left(\frac{h}{L}\right)^2\left(\frac{P_0}{\rho U^2}\right)^2\left[\left(\frac{\partial P}{\partial X}\right)^2+\left(\frac{\partial P}{\partial Y}\right)^2\right]-$$

$$\frac{1}{6}Re\frac{h}{L}\frac{P_0}{\rho U^2}\frac{1}{\left(1+2b\frac{\lambda}{h}\right)\left(1+2a\frac{\lambda}{h}\right)}\frac{\partial P}{\partial X}, \qquad (6\text{-b})$$

where P, X and Y are non-dimensional variables define as $P=p/P_0$, $X=x/L$ and $Y=y/L$, and M is the Mach number defined as $M=U/(\gamma RT_0)^{1/2}$.

4. Analysis and Discussion

In this section, we study the heat transfer for two special cases, Couette flow and Poiseuille flow between two plane plates, in order to reveal the physical meaning of each term in the heat flux equation (6-a). The velocity fields for the two types of flows are shown in Fig. 2 and 3.

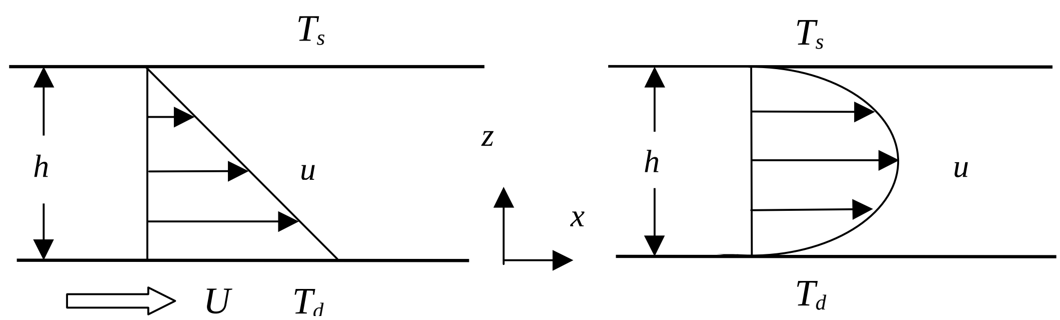


Fig. 2 Couette flow   Fig. 3 Poiseuille flow

(1) *Couette flow*:

Using the linear expression for Couette flow, in which the velocity is unidirectional (say in the x direction), and the boundary condition (3-a)~(3-b), we can obtain the velocity distribution as:

$$u = U\left(1 - \frac{z + a\lambda}{h + 2a\lambda}\right). \qquad (7)$$

Substituting this velocity solution into the energy equation (2-b) and integrating it, we obtain the temperature distribution and then the heat transfer between the upper plane and the air flow by Fourier's Law:

$$T = T_d - \frac{\mu U^2}{2k(h+2a\lambda)^2} z^2 + \left[\frac{T_s - T_d}{h+2b\lambda} + \frac{\mu U^2 h}{2k(h+2a\lambda)^2} \right](z + b\lambda), \qquad (8)$$

$$q = -k\frac{T_s - T_d}{h+2b\lambda} + \frac{\mu U^2 h}{2(h+2a\lambda)^2}\,. \qquad (9\text{-a})$$

We can also write this heat transfer equation in a non-dimensional form as:

$$\frac{qh}{\mu U^2} = -\frac{T_s - T_d}{\left(\frac{\gamma-1}{2}\right) PrM^2 T_0 \left(1+2b\frac{\lambda}{h}\right)} + \frac{1}{2\left(1+2a\frac{\lambda}{h}\right)^2}\,. \qquad (9\text{-b})$$

Comparing equation (9-b) with equation (6-b), we see that the second term in the RHS of (6-b) is the contribution from the viscous dissipation by Couette flow.

(2) *Poiseuille flow*:

The velocity field in the Poiseuille flow is also unidirectional and can be obtained by integrating equation (1-a) and applying the boundary condition (3-a)~(3-b) with U=0. The solution is:

$$u = -\frac{1}{2\mu}\frac{\partial p}{\partial x}\left(a\lambda h + hz - z^2\right). \qquad (10)$$

In a similar way as used with the Couette flow, we can express the temperature distribution and the heat transfer between the slider and the air bearing as:

$$T = T_d - \frac{1}{4k\mu}\left(\frac{1}{3}z^4 - \frac{2}{3}hz^3 + \frac{1}{2}h^2 z^2\right)\left(\frac{\partial p}{\partial x}\right)^2 +$$

$$\left[\frac{T_s - T_d}{h + 2b\lambda} + \frac{h^3}{24k\mu}\left(\frac{\partial p}{\partial x}\right)^2\right](z + b\lambda), \tag{11}$$

$$q = -k\frac{T_s - T_d}{h + 2b\lambda} + \frac{h^3}{24\mu}\left(\frac{\partial p}{\partial x}\right)^2. \tag{12-a}$$

Or in the non-dimensional form:

$$\frac{qh}{\mu U^2} = -\frac{T_s - T_d}{\left(\frac{\gamma - 1}{2}\right)PrM^2 T_0\left(1 + 2b\frac{\lambda}{h}\right)} + \frac{1}{24}Re^2\left(\frac{h}{L}\right)^2\left(\frac{P_0}{\rho U^2}\right)^2\left(\frac{\partial P}{\partial X}\right)^2. \tag{12-b}$$

Comparing equation (12-b) with equation (6-a), we see that the third term in the RHS of (6-b) is the contribution from the viscous dissipation of Poiseuille flow. Clearly, the fourth term is a combined contribution of both Couette flow and Poiseuille flow.

(3) *Heat conduction*:

The first term in the RHS of equation (6-b) is the contribution of heat conduction. The expression of this term is a modification of that for two plane plates by Fourier's Law: $q=-k\Delta T/h$. Due to the introduction of the temperature jump at the boundary, the effect of the heat conduction is reduced by a factor of $(1+2b\lambda/h)$.

It is interesting to note that the heat transfer between the slider and air bearing is not zero when the temperature difference between the slider surface and the disk surface vanishes, because of the effect of viscous dissipation.

5. Simulation Results

In this section, we compute several case studies for sliders flying close to the disk surface. We assume that the slider has a surface temperature either equal to that of the disk or higher than that of the disk because of an electrical current goes through the MR element (Tian, et al., 1997). For convenience, we choose a 50% (2*mm*×1.6*mm*) tri-pad slider with a load of 3.5 *g* and with taper length and angle of 0.2 *mm* and 0.01 *rad*, respectively. The slider is fixed at a radial position *r*=23 *mm*. The rail shape of this slider is shown in Fig. 4(a). For each case in the analysis, the Reynolds equation is solved by using the CML Air Bearing Simulator (Lu, S., et al., 1995).

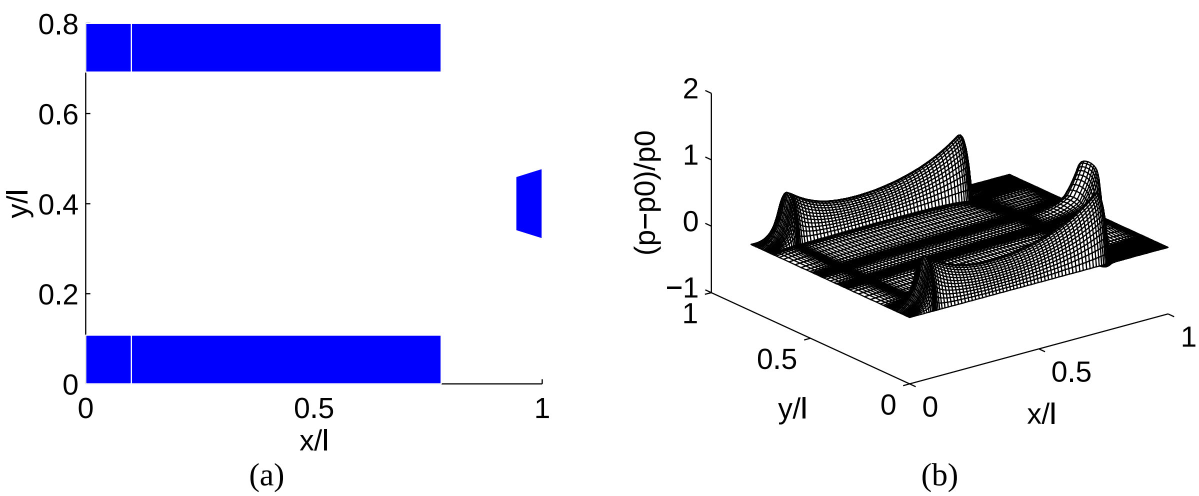


Fig. 4 A tri-pad slider and the pressure profile in the air bearing

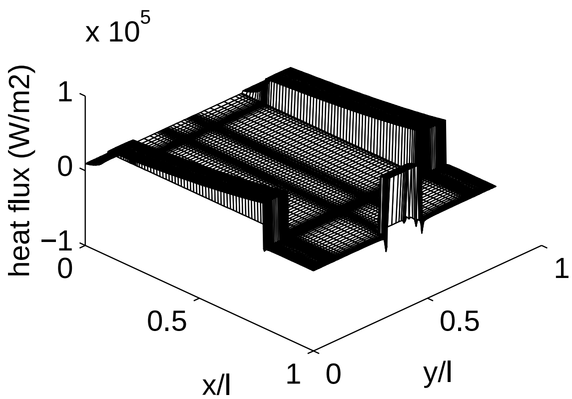
(1) "*Cooling*" *effects of the air bearing*:

In this case, we choose the disk rotation Ω=6400 *rpm*. With this rotation, the pressure distribution of the air bearing is calculated and shown as in Fig.4(b) and the flying characteristics are shown in Table 1. To calculate the heat transfer from the slider to the air bearing, we take T_s=301 °K and T_d=300 °K, or ΔT=1 °K. The heat flux for each point is shown in Fig.5, in which (a) and (b) are based on the same data but viewed from different perspectives. Note that positive values mean that heat is transferred from the slider to the air bearing.

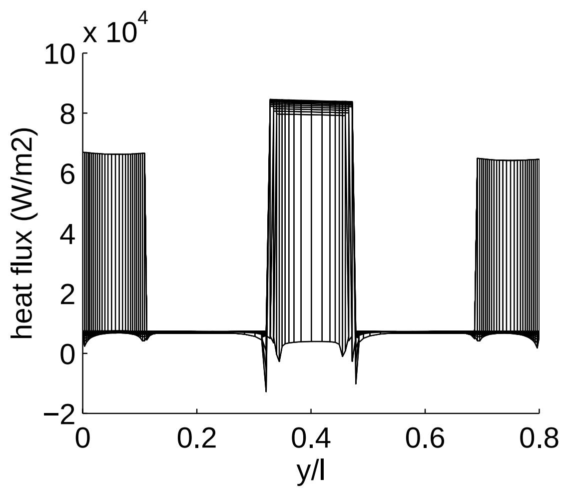
Table 1 Flying Characteristics

Disk rotation (*rpm*)	Pitch angle (μrad)	Roll (μrad)	CTE-FH† (*nm*)
6400	176	8	44

† Central trailing edge flying height



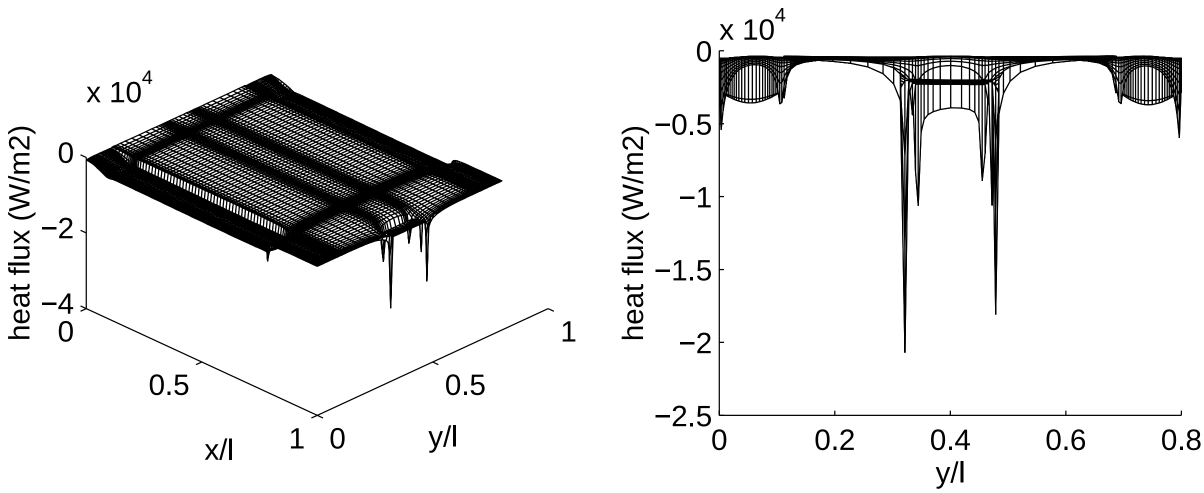
(a)



(b)

Fig.5 Heat flux between the slider and air bearing (T_s–T_d=1 °K)

It is seen that the heat is transferred from the slider to the air bearing in this case. In other words, the air bearing plays a “cooling” role under such a condition. But, this “cooling” effect does not always exist for the air bearing. From the discussion in the last section, we know that heat transfer between the slider and air bearing is a combined result of heat conduction and viscous dissipation. The former transfers heat from the slider to the air bearing if the slider has a higher surface temperature than the disk, while the latter transfers heat to the slider from the air bearing and plays a role of "heater". From Fig. 5(a, b), we see that the heat flux at some points at the trailing edge has smaller values, which implies that there exists stronger viscous dissipation at these points. Whether an air bearing is a "coolant" or "heater" depends on which part, heat conduction or viscous dissipation, dominates the heat transfer between the slider and air bearing. We can illustrate this point more clearly if we take T_s=T_d and obtain the heat flux as shown in Fig.6.



(a) (b)

Fig.6 Heat flux between slider and the air bearing (T_s=T_d)

It is clear that in this case heat is transferred to the slider because of the viscous dissipation, or the air bearing has a heating effect. Since the pressure has a sharp decrease close to the trailing edge, the magnitude of the heat flux increases sharply there (referring to equation (6-

b)). Comparing Fig.6 with Fig.5, we easily conclude that the viscous dissipation has a smaller magnitude than the heat conduction even for a small temperature difference.

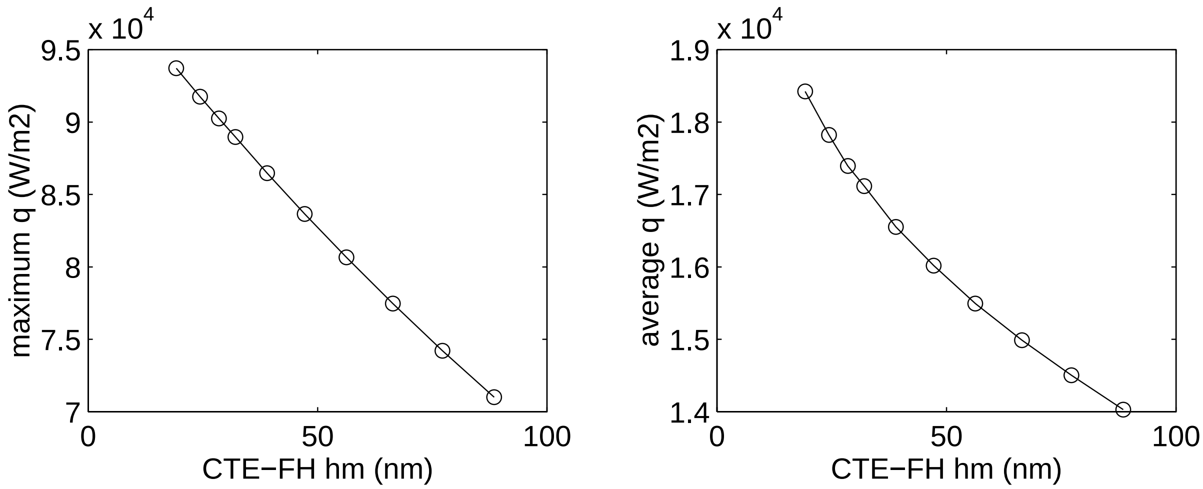
(2) *Effect of the flying height and disk speed*:

From Fig.5, we see that heat flux shows different values in the air bearing and recessed region, which implies that the heat flux changes with the head/disk interface (HDI) spacing. In the following cases, we study the relation of the heat flux to the CTE flying height *hm*. Note that to change the flying height, we have to change the disk rotation speed simultaneously if we keep the other parameters the same. Therefore, the heat flux is affected by both the disk rotation speed and the flying height. Table 2 shows the related flying characteristics for different cases. The results for heat flux with flying height for ΔT=1°*K* and 0°C are shown in Fig.7 and Fig.8, respectively.

Table 2 Flying characteristics for different cases

Disk rotation (*rpm*)	Pitch angle (μrad)	Roll (μrad)	CTE-FH (*nm*)
4400	136	5	19
4900	148	6	24
5250	155	6	28
5500	160	7	32
6000	169	8	39
6500	177	8	47
7000	184	9	56
7500	189	10	66
8000	194	10	77
8500	197	11	89

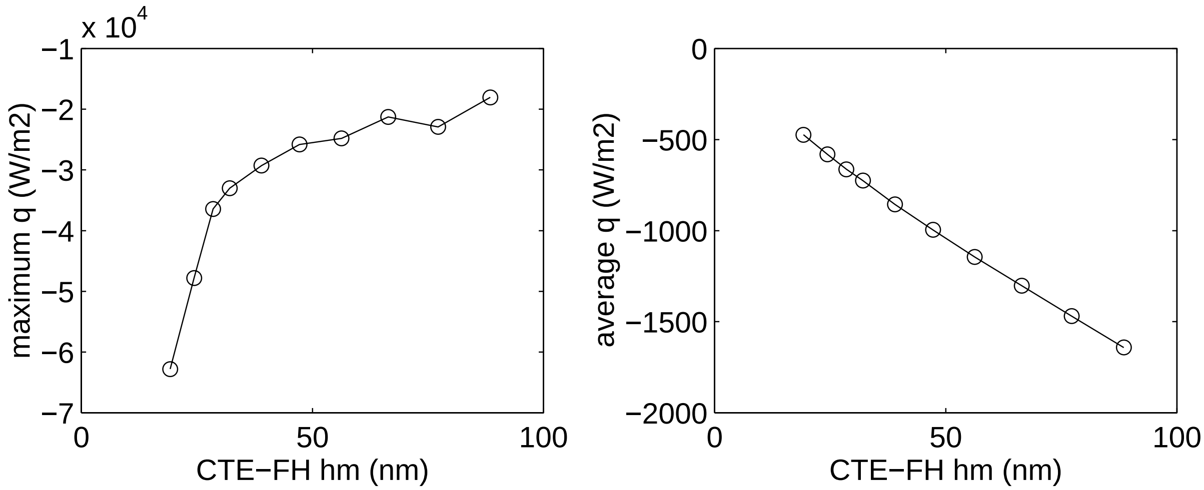
In Fig.7, for ΔT=1 °*K*, the maximum heat flux (Fig. 7(a)) and average heat flux (Fig. 7(b)) are determined over the whole slider surface. It is seen that both of them increase with the decrease of the flying height under the given temperature difference (T_s–T_d=1 °K). The calculation results also show that the maximum heat flux occurs at the trailing edge where the air bearing always has the smallest spacing.



(a) (b)

Fig.7 Heat flux vs. central trailing edge flying height (ΔT=1 °K)

In Fig.8, for ΔT=0 °K, the heat flux has negative values which means the heat is transferred to the slider because of viscous dissipation. Note that the "maximum" in Fig. 8(a) means the maximum magnitude, or maximum heat flux into the slider. It is seen that this maximum heat flux increases with the decrease of the flying height. A reason for it may be that the pressure profile at some points such as the trailing corners of the rear rail (Fig. 6(a)), where usually there exists a drastic pressure variation at the low flying heights, becomes smoother at the higher flying height. But for the average heat flux (Fig. 8(b)), its magnitude decreases with the decrease of the flying height. In other words, the "heating" effect of the air bearing decreases with the decrease of the flying height. Since the flying height is proportional to the disk speed (Table 2), we can also say the “heating” effect of the air bearing decreases with the decrease of the disk speed.



(a) (b)

Fig.8 Heat flux vs. central trailing edge flying height (ΔT=0 °K)

6. Conclusion

In this report, we solve the N-S and energy equations to get the temperature distribution in the air bearing and then the heat transfer between the slider and the air bearing. In solving these equations, we make an assumption that the properties of the air remain the same across the air bearing because the temperature variation is not significant, so we can decouple the momentum equation and energy equation and integrate them separately. The results show that the heat transfer between the slider and air bearing depends on both the heat conduction, which transfers heat to the air bearing if the slider has a higher surface temperature than the disk, and viscous dissipation, which transfers heat to the slider. In most cases heat conduction dominates the heat transfer, and therefore the net result is that heat is transferred from the slider to the air bearing. Under this condition, the air bearing is regarded as a coolant. But when the temperature difference is nearly equal to zero, viscous dissipation dominates the heat transfer and heat is transferred into the slider, so the air bearing acts as a heater. Since the magnitude of the viscous dissipation is not large, this heating effect is not significant. Simulation results also show that the heat conduction effect increases with the decrease of the flying height (or disk rotation speed), but the viscous dissipation effect decreases with the decrease of the flying height (or disk rotation speed). In other words, the "cooling" effect increases with the decrease of the flying height (or disk rotation speed).

Acknowledgement

This project is supported by the Computer Mechanics Laboratory, University of California, Berkeley.

References

Burgdorfer, A., 1959, "The Influence of the Molecular Mean Free Path on the Performance of Hydrodynamic Gas Lubricated Bearings", *J. of Basic Engr.*, Vol. 81, p.94-100.

Cha, E.T., Bogy, D.B., 1995, "A Numerical Scheme for Static and Dynamic Simulation of Subambient Pressure Shaped Rail Sliders", *J. of Tribology*, Vol. 117, p.36-46.

Fukui, S., Kaneko, R., 1988, "Analysis of Ultra-thin Gas Film Lubrication Based on Linearized Boltzmann Equation: First report-Derivation of A Generalized Lubrication Equation Including Thermal Creep Flow", *J. of Tribology*, Vol. 110, p.253-261.

Gans, R., 1985, “Lubrication Theory at Arbitrary Knudsen Number”, *J. of Tribology*, Vol. 107, p.431-433.

Gross, W.S., Matsch, L.A., Castelli, V., Eshel, A., Vohr, J.H., Wildmann, M., 1980, "Fluid Film Lubrication", *John Wiley & Sons, Inc*.

Kennard, A.H., 1938, “Kinetics Theory of Gases”, *McGraw-Hill Book Company, Inc*., New York and London.

Lu, S., Bogy, D.B., 1994, “A Multi-Grid Control Volume Method for the Simulation of Arbitrarily Shaped Slider Air Bearings with Multiple Recess Levels”, *CML report*, No. 94-016, UC Berkeley.

Lu, S., Bogy, D.B., 1995, “CML Air Bearing Design Program User’s Manual”, *CML report*, No. 95-003, UC Berkeley.

Mitsuya, Y., 1993, "Modified Reynolds Equation for Ultra-Thin Film Gas Lubrication Using 1.5-Order Slip-Flow Model and Considering Surface Accommodation Coefficient", *ASME Journal of Tribology*, Vol. 115, pp. 289-294.

Schaaf, S.A., Chambre, P.L., 1961, “Flow of Rarefied Gases”, *Princeton University Press*.

Tian, H., Cheung, C-Y., Wang, P-K., 1997, “Non-Contact Induced Thermal Disturbance of MR Head Signals”, *IEEE on Magnetics*, in press.