PERGAMON

International Journal of Heat and Mass Transfer 42 (1999) 1791–1800

International Journal of HEAT and MASS TRANSFER

A heat transfer model for thermal fluctuations in a thin slider/disk air bearing

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Received 7 January 1998; in final form 10 August 1998

Abstract

The signal readout of a magnetoresistive (MR) transducer is very sensitive to temperature. To study the thermal effects on the MR transducer when the slider flies close to the surface of the disk, we introduce a heat transfer model with discontinuous boundary conditions in the thin slider/disk air bearing and solve it numerically. It is found that the heat flux is primarily due to 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 the air bearing to the slider. Whether an air bearing acts as 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 relatively 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 show that the effect of the heat conduction increases with a decrease in the flying height, but the effect of the viscous dissipation decreases with a decrease in the flying height. In other words, the 'cooling' effect increases with a decrease in the flying height.

Key words: Microstructure heat transfer; Slider/disk air bearing; MR head; Thermal asperity

Nomenclature

c_p specific heat at constant pressure
c_v specific heat at constant volume
h air bearing space
k thermal conductivity of the air
Kn Knudsen number
L length of the slider
M Mach number
p air bearing pressure
P_0 ambient air pressure
p^* non-dimensional air bearing pressure, $p^* = p/P_0$
Pr Prandtl number
q heat flux between the slider surface and the air bearing
R gas constant
Re Reynolds number
T air bearing temperature
T_s temperature of the slider surface
T_d temperature of the disk surface
T_0 ambient air temperature
ΔT_0 temperature difference between the slider and disk surfaces
T^* non-dimensional air bearing temperature, $T^* = T/\Delta T_0$
u, v, w velocity components of the air bearing
U linear velocity of the disk at the slider location
u^*, v^*, w^* non-dimensional velocity components, $u^* = u/U$, $v^* = v/U$, $w^* = w/U$
x, y, z coordinates in the air bearing
x^*, y^*, z^* non-dimensional coordinates in the air bearing, $x^* = x/L$, $y^* = y/L$, $z^* = z/h$.

Greek symbols

α thermal diffusivity of the air
γ ratio of the specific heats, $\gamma = c_p/c_v$
λ mean free path of the air
μ viscosity of the air
ν dynamic viscosity of the air
σ_M momentum accommodation coefficient
σ_T thermal accommodation coefficient.

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0017–9310/98/$ - see front matter
PII: S0017-9310(98)00267-1

1. Introduction

The magnetoresistive (MR) transducer was developed using the principle that its resistance varies with the variation of the surrounding magnetic field [1]. Since its resistance is also temperature dependent, any temperature change will result in a noise in the MR readout signal. One such phenomenon, which is referred to as thermal asperities, is induced by the flash temperature rise when the slider contacts the disk near the MR transducer. A similar phenomenon was observed when a slider flies very close to the disk surface without contact [2]. Experimental results showed that when a slider carrying a MR transducer flies over an asperity that disturbs the steady flying condition, the readout signal fluctuates with the fluctuation of the flying height of the slider. They concluded that the air bearing has a cooling effect on the MR transducer, which makes a major contribution to the fluctuation of the readout signal. In this paper, we conduct a theoretical study of the heat transfer between a slider and the air bearing to determine the mechanism of the 'cooling' effect of the air bearing.

One important issue in solving the heat transfer problem between a slider and the air bearing is that the traditional lubrication theory, which is based on the continuum assumption, is not valid when the air bearing is very thin. For example, the flying height of a typical MR head is around 50 nm in today's hard disk drive. The Knudsen number $Kn = \lambda/h$, where λ is mean free path of the air and h is the spacing of the slider/disk interface, is between 0.02 and 1 under this condition. The air flow with such a Knudsen number is regarded as within the slip and transition regimes, and far out of the continuum region of $Kn < 0.01$ [3]. One approach to solving heat transfer problems in these regimes is to apply the Maxwell–Boltzman equation of the kinetic theory of gases. However, solving a complete Maxwell–Boltzman equation requires very much computation time. Another approach is to assume that the continuum governing equations such as the Navier–Stokes (N–S) equation and energy equation are still usable. As a modification, the discontinuous boundary conditions are applied [4]. These methods have been used previously in solving for the velocity distribution in an air bearing by several researchers [5–7].

Another important issue in solving heat transfer problems under these conditions is that the continuity equation, momentum equation and energy equation need to be solved simultaneously, because the physical properties of the air depend on the temperature, which usually makes the problem more complicated, and also requires more computation time in the numerical analysis. A simple approach is to assume that 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.

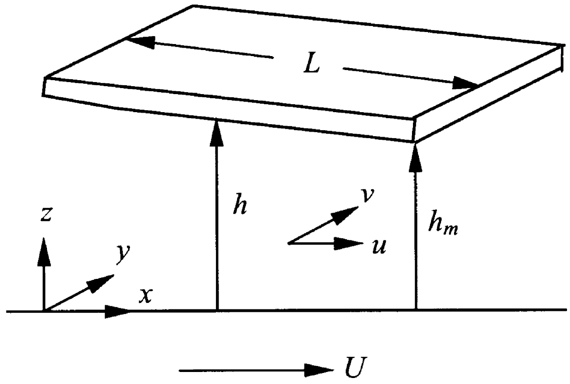

Fig. 1. Slider/disk system and coordinates.

In this paper, we first simplify the N–S and energy equations by dimensional analysis. Then we solve the reduced N–S equation with slip boundary conditions to get the velocity distribution and solve the energy equation with jump boundary condition 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 Fig. 1.

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.

2.1. Navier–Stokes (N–S) equation

The simplification of the N–S equation in an air bearing has been performed by many researchers [8]. Here we only list the simplified results and do not present the detailed derivation:

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial z^2}, \tag{1a}$$

$$\frac{\partial p}{\partial y} = \mu \frac{\partial^2 v}{\partial z^2}, \tag{1b}$$

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

where u, v are velocities in the x- and y-directions, p is the pressure and μ the viscosity of the air. For simplification, we assume μ is uniform in the air bearing. The velocity component w in the z-direction is approximated to be zero. Clearly, the pressure p remains constant across the thickness of the air bearing.

2.2. Energy equation

As in the N–S equation, the energy equation can also be simplified by using dimensional analysis in the air bearing. Since the magnitudes $|\partial/\partial x| \sim |\partial/\partial y| \ll |\partial/\partial z|$ and the air velocity w in the z-direction is approximately zero in a lubrication problem, we neglect the relatively small terms and write the energy equation as:

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

where ρ is the density, c_p is the specific heat, k is the thermal conductivity, assumed uniform in the air bearing, and T is the temperature of the air.

As in simplifying the N–S equation, we use the characteristics of the air bearing to reduce the energy equation (2a). Let's first define the non-dimensional variables: $u^* = u/U$, $v^* = v/U$, $T^* = T/\Delta T_0$, $p^* = p/\Delta P_0$, $x^* = x/L, y^* = y/L$, $z^* = z/h$, where U is the disk velocity, ΔT_0 is the temperature difference between the slider and disk surfaces and P_0 is the reference pressure (say the ambient air pressure), L is the length of the slider and h is the thickness of the air bearing. Substituting these variables into equation (2a) we obtain the following expression:

$$\frac{Uh^2}{\alpha L}\left(u^*\frac{\partial T^*}{\partial x^*} + v^*\frac{\partial T^*}{\partial y^*}\right) - \frac{P_0 U h^2}{k L \Delta T_0}\left(u^*\frac{\partial p^*}{\partial x^*} + v^*\frac{\partial p^*}{\partial y^*}\right) = \frac{\partial^2 T^*}{\partial z^{*2}} + \frac{\mu U^2}{k\Delta T_0}\left[\left(\frac{\partial u^*}{\partial z^*}\right)^2 + \left(\frac{\partial v^*}{\partial z^*}\right)^2\right], \tag{2b}$$

where $\alpha = k/\rho c_p$ is the thermal diffusivity.

For a typical head/disk air bearing, we can take $\rho \sim 1$ kg m^{-3}, $c_p \sim 10^3$ J kg^{-1} K^{-1}, $k \sim 0.03$ W m^{-1} K^{-1}, $\mu \sim 10^{-5}$ kg m^{-1} s^{-1}, $U \sim 15$ m s^{-1}, $L \sim 2$ mm, $h \sim 50$ nm, $\Delta T_0 \sim 10$ K, and $P_0 \sim 10^5$ kg m^{-1} s^{-2}. Thus $Uh^2/\alpha L \sim Pr\,Re(h/L) \sim 10^{-7}$, $(P_0 U h^2)/(kL\Delta T_0) \sim (Pr\,Re\,M^2)(T_0/\Delta T_0)(h/L)(p_0/\rho U^2) \sim 10^{-5}$, and $\mu U^2/k\Delta T_0 \sim Pr\,M^2(T_0/\Delta T_0) \sim 10^{-2}$, where Pr is the Prandtl number defined by $Pr = \mu c_p/k$, Re is the Reynolds number defined as $Re = Uh/v$ and M is the mach number defined by $U/(\gamma R T_0)^{1/2}$, and T_0 is a reference temperature (say the ambient air temperature). Therefore, compared with the conduction term in equation (2b), the non-linear terms on the LHS are small and can be neglected. The energy equation is thus reduced to:

$$k\frac{\partial^2 T}{\partial z^2} + \mu\left(\frac{\partial u}{\partial z}\right)^2 + \mu\left(\frac{\partial v}{\partial z}\right)^2 = 0. \tag{2c}$$

Generally, the viscous dissipation term is smaller in magnitude than the conduction term in equation (2c). But, when the temperature difference between the slider and disk surfaces is close or equal to zero, the warming effect of the viscous dissipation is not negligible. Therefore, we keep it in equation (2c) for future analysis.

Note that equation (2c) is valid only when $PrRe(h/L) \ll 1$, $(PrReM^2)(T_0/\Delta T_0)(h/L)(p_0/\rho U^2) \ll 1$ and $h/L \ll 1$. Fortunately, these conditions are usually satisfied in a slider/disk air bearing.

2.3. 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 that the disk has much larger size than the air bearing and rotates with high speed, we assume that it has a constant and uniform surface temperature, the same as the ambient air. We also assume that 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 [2, 4], we can write the boundary conditions for velocity and temperature as:

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

$$u(h) = -\frac{2-\sigma_M}{\sigma_M}\lambda\left.\frac{\partial u}{\partial z}\right|_{z=h}, \tag{3b}$$

$$v(0) = -\frac{2-\sigma_M}{\sigma_M}\lambda\left.\frac{\partial v}{\partial z}\right|_{z=0}, \tag{3c}$$

$$v(h) = -\frac{2-\sigma_M}{\sigma_M}\lambda\left.\frac{\partial v}{\partial z}\right|_{z=h}, \tag{3d}$$

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

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

where σ_M is the momentum accommodation coefficient and σ_T 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 $a = (2-\sigma_M)/\sigma_M$ and $b = 2(2-\sigma_T)\gamma/\sigma_T(\gamma+1)Pr$ in the following analysis.

3. Heat transfer between the slider and the air bearing

To obtain the heat transfer in the air bearing, we need to know its temperature distribution. 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.

3.1. Velocity distribution

The velocity distribution can be obtained by integrating the reduced N–S equations (1a)–(1b) with boundary conditions (3a)–(3d). The procedure is straightforward and was done by other researchers [5, 9]. Here we list the results of the solution:

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

$$v = -\frac{1}{2\mu}\frac{\partial p}{\partial y}(a\lambda h + hz - z^2). \tag{4b}$$

On the RHS of equation (4a), the first term is the contribution of the Poiseuille flow and the second term is the contribution of the Couette flow, while in (4b) 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 [5, 7], to obtain the pressure distribution first. To get the solution, a numerical method is required [10, 11].

3.2. Temperature distribution

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

$$\begin{aligned} T = T_d &- \frac{1}{k}\left\{\frac{1}{12\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right]z^4\right. \\ &- \frac{1}{3}\left\{\frac{h}{2\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right] + \frac{\partial p}{\partial x}\frac{U}{h+2a\lambda}\right\}z^3 \\ &\left. + \frac{\mu}{2}\left[\left(\frac{h}{2\mu}\frac{\partial p}{\partial x} + \frac{U}{h+2a\lambda}\right)^2 + \frac{h^2}{4\mu^2}\left(\frac{\partial p}{\partial y}\right)^2\right]z^2\right\} \\ &+ \left\{\frac{T_s - T_d}{h+2b\lambda} + \frac{1}{k}\left[\frac{h^3}{24\mu}\left\{\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right\}\right.\right. \\ &\left.\left. + \frac{\mu U^2 h}{2(h+2a\lambda)^2} + \frac{Uh^3}{6(h+2a\lambda)(h+2b\lambda)}\frac{\partial p}{\partial x}\right]\right\}(z+b\lambda). \end{aligned} \tag{5}$$

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

3.3. 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:

$$q = -k\frac{T_s - T_d}{h+2b\lambda} + \frac{h^3}{24\mu}\left[\left(\frac{\partial p}{\partial x}\right)^2 + \left(\frac{\partial p}{\partial y}\right)^2\right] + \frac{\mu U^2 h}{2(h+2a\lambda)^2} - \frac{Uh^3}{6(h+2b\lambda)(h+2a\lambda)}\frac{\partial p}{\partial x}. \tag{6a}$$

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

$$\begin{aligned} \frac{qh}{\mu U^2} = &-\frac{T_s - T_d}{\left(\frac{\gamma-1}{2}\right)PrM^2T_0\left(1+2b\frac{\lambda}{h}\right)} + \frac{1}{2\left(1+2a\frac{\lambda}{h}\right)^2} \\ &+ \frac{1}{24}Re^2\left(\frac{h}{L}\right)^2\left(\frac{P_0}{\rho U^2}\right)^2\left[\left(\frac{\partial p^*}{\partial x^*}\right)^2 + \left(\frac{\partial p^*}{\partial y^*}\right)^2\right] \\ &- \frac{1}{6}Re\frac{h}{L}\frac{P_0}{\rho U^2}\frac{1}{\left(1+2b\frac{\lambda}{h}\right)\left(1+2a\frac{\lambda}{h}\right)}\frac{\partial p^*}{\partial x^*}. \end{aligned} \tag{6b}$$

4. Analysis and discussion

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

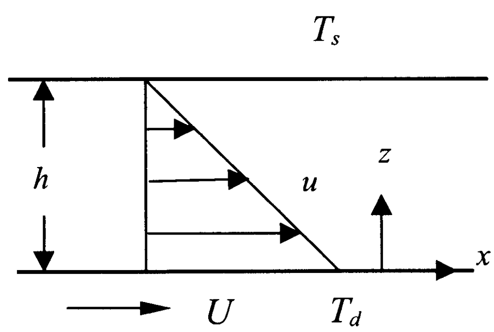

Fig. 2. Couette flow.

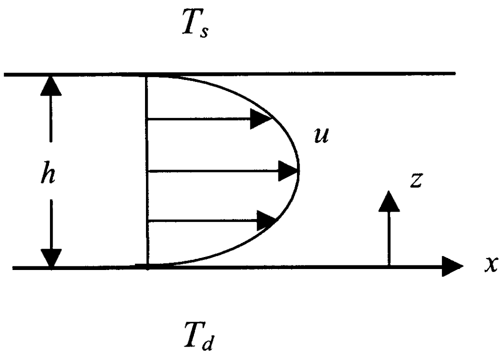

Fig. 3. Poiseuille flow.

4.1. Couette flow

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

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

Substituting this velocity solution into the energy equation (2c) and integrating the result, we obtain the temperature distribution and then the heat transfer between the upper plane and the airflow by Fourier's Law:

$$T = T_{\mathrm{d}} - \frac{\mu U^2}{2k(h+2a\lambda)^2}z^2 + \left[\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{h+2b\lambda} + \frac{\mu U^2 h}{2k(h+2a\lambda)^2}\right](z+b\lambda), \tag{8}$$

$$q = -k\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{h+2b\lambda} + \frac{\mu U^2 h}{2(h+2a\lambda)^2}. \tag{9a}$$

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

$$\frac{qh}{\mu U^2} = -\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{\left(\frac{\gamma-1}{2}\right)PrM^2T_0\left(1+2b\frac{\lambda}{h}\right)} + \frac{1}{2\left(1+2a\frac{\lambda}{h}\right)^2}. \tag{9b}$$

A similar expression to equation (9b) can also be found in [3], in which no detailed derivation is given. Comparing equation (9b) with equation (6b), we see that the second term in the RHS of equation (6b) is the contribution from the viscous dissipation by Couette flow.

4.2. Poiseuille flow

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

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

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

$$T = T_{\mathrm{d}} - \frac{1}{4k\mu}\left(\frac{1}{3}z^4 - \frac{2}{3}hz^3 + \frac{1}{2}h^2z^2\right)\left(\frac{\partial p}{\partial x}\right)^2 + \left[\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{h+2b\lambda} + \frac{h^3}{24k\mu}\left(\frac{\partial p}{\partial x}\right)^2\right](z+b\lambda), \tag{11}$$

$$q = -k\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{h+2b\lambda} + \frac{h^3}{24\mu}\left(\frac{\partial p}{\partial x}\right)^2, \tag{12a}$$

or in the non-dimensional form:

$$\frac{qh}{\mu U^2} = -\frac{T_{\mathrm{s}}-T_{\mathrm{d}}}{\left(\frac{\gamma-1}{2}\right)PrM^2T_0\left(1+2b\frac{\lambda}{h}\right)} + \frac{1}{24}Re^2\left(\frac{h}{L}\right)^2\left(\frac{P_0}{\rho U^2}\right)^2\left(\frac{\partial p^*}{\partial x^*}\right)^2. \tag{12b}$$

Comparing equation (12b) with (6b), we see that the third term in the RHS of (6b) 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.

4.3. Heat conduction

The first term in the RHS of equation (6b) is the contribution of the heat conduction. Due to the introduction of the temperature jump at the boundary, the effect of the heat conduction is reduced by a factor of $(1+2b\lambda/h)$ compared to the continuum case. Note that because of the effect of viscous dissipation, 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.

5. Simulation results

In this section, we compute several cases 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 that goes through the MR transducer [2]. For convenience, we choose a 50% (2×1.6 mm) tri-pad slider with taper length and angle of 0.2 mm and 0.01 rad, respectively, and with a recessed depth of 3 μm. The applied load is 3.5 g. The slider is fixed at a radial position $r = 23$ mm. The rail shape of this slider is shown in Fig.

4. For each case in the analysis, the Reynolds equation is solved by using the CML Air Bearing Simulator [12].

5.1. 'Cooling' effects of the air bearing

In this case, we choose the disk rotation speed $\Omega = 6400$ rpm. The pressure distribution of the air bearing is calculated and shown in Fig. 5, and the flying characteristics are given 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 $\Delta T = 1$ K. The heat flux for each point is shown in Fig. 6. Note that positive values mean that heat is transferred from the slider to the air bearing.

It is seen that even at a small temperature difference $\Delta T_0 = 1$ K, the heat conduction still dominates the overall heat transfer and results in a net heat flux from the slider to the air bearing, except at some points around the edges of end rail. Figure 7 shows the simulation result for $\Delta T_0 = 0$ K, in which only viscous dissipation exists. It is seen that the heat flux take negative values and has a large magnitude at the corners of the end rail. The reason for this may be that the pressure at these points has larger gradients (Fig. 5), which makes the magnitude of the heat flux increase sharply there (referring to equation (6b)). To reduce the warming effects of the viscous dissipation, it is recommended to avoid putting the MR sensor at the corners of the rails.

Comparing Fig. 6 with Fig. 7, we conclude that the viscous dissipation has a smaller magnitude than the heat conduction, except for when the temperature difference is very close to zero. Generally, when the temperature difference is non-zero and the air bearing surface (rail shape) is simple, we can use only the first term of the RHS of the equation (6a) to calculate the heat flux q in an air bearing with reasonable accuracy. But when the temperature difference is zero (or close to zero) and the air bearing surface is complicated, we need to include the viscous dissipation terms in calculating the heat flux.

5.2. Effect of the flying height and disk speed

From Fig. 6 we see that heat flux shows different values in the air bearing and recessed region, which implies that the heat flux changes with the slider/disk interface (SDI) spacing. In the following cases, we study the relation of the heat flux to the CTE-FT hm. Note that to change the flying height, we have to change the disk rotation speed if we keep the other parameters fixed. Therefore, the heat flux is actually 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 versus flying height for $\Delta T = 1$ K and 0 K are shown in Figs 8 and 9, respectively.

The heat flux for $\Delta T = 1$ K at a single point (5 μm inside the CTE) as well as the average heat flux over the surface of the end rail, which is important for the temperature variation of the MR transducer, are both plotted in Fig. 8. It is seen that both of them increases with the decrease of the flying height under the given temperature difference ($T_s - T_d = 1$ K). This means that more heat is transferred to the air bearing when the slider flies closer to the disk surface.

The same calculation results for $\Delta T = 0$ K are plotted

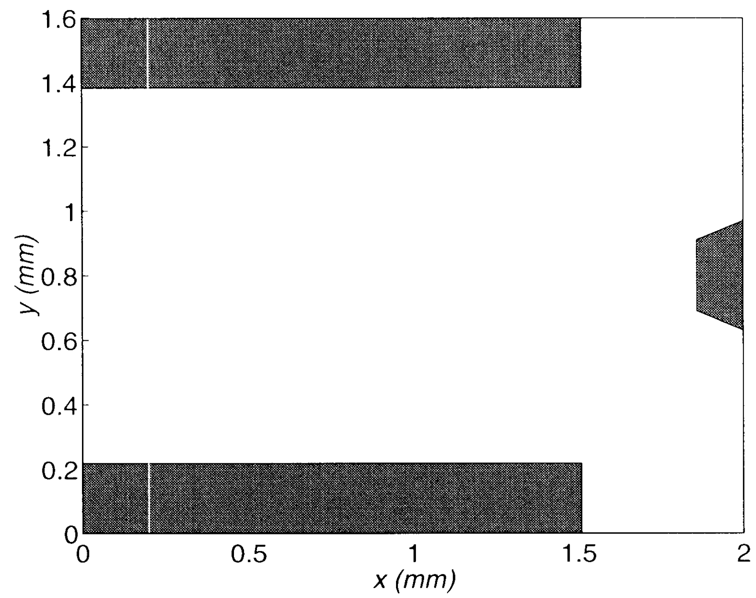

Fig. 4. The rail shape of the tri-pad slider.

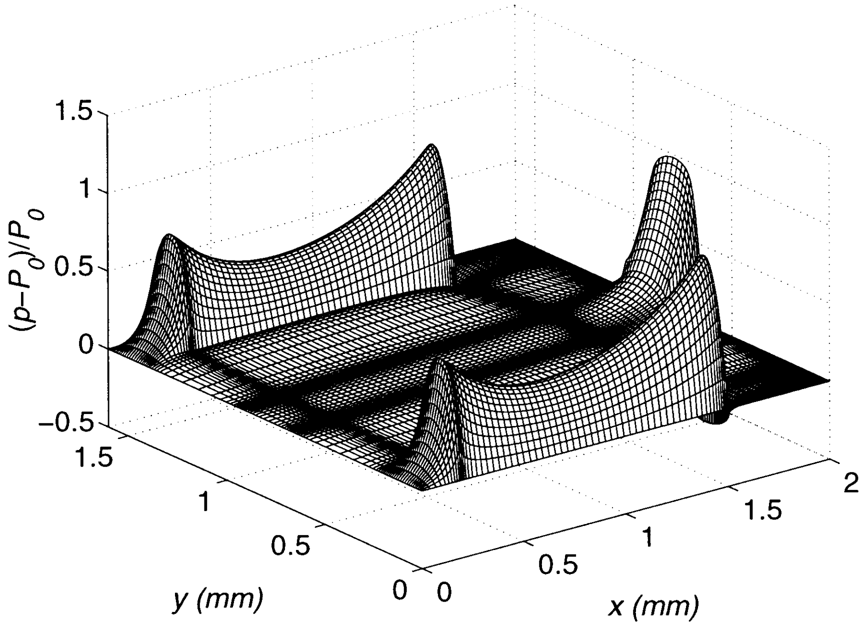

Fig. 5. The pressure profile in the air bearing of the tri-pad slider.

Table 1
Flying characteristics of the tri-pad slider

Disk rotation [rpm]	Pitch angle [μrad]	Roll [μrad]	CTE-FH* [nm]
6400	176	8	44

*Central trailing edge flying height.

Table 2
Flying characteristics of the tri-pad slider at different rpm

Disk rotation [rpm]	Pitch angle [μrad]	Roll [μrad]	CTE-FH [nm]
4000	126.5	4.8	15.7
4500	139.0	5.4	20.0
5000	150.4	6.1	25.2
5500	160.0	6.7	32.1
6000	169.3	7.5	39.0
6500	176.9	8.2	47.2
7000	183.7	8.9	56.3
7500	189.4	9.8	66.4
8000	193.9	10.4	77.2
8500	197.3	11.4	88.5

in Fig. 9, in which negative value means that the heat is transferred to the slider because of viscous dissipation. As in Fig. 8, the heat flux for both the single point and the averaged value over the surface of the end rail decrease in magnitude with the decrease of the flying height. In other words, if only viscous dissipation exists, less heat will be transferred to the slider when the slider flies closer to the disk surface. Combining this result with that for $\Delta T = 1$ K, we can say that the ‘cooling’ effect increases with the decrease of the flying height. This conclusion is identical to the experimental result by Tian et al. [2]. Since the flying height is proportional to the disk speed (Table 2), we can also say that the ‘cooling’ effect of the air bearing increases with the decrease of the disk speed.

6. Conclusion

In this paper, we solve the N–S and energy equations with discontinuous boundary conditions to obtain 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 constant across the air bearing because the temperature variation is not significant, so we can decouple the N–S 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

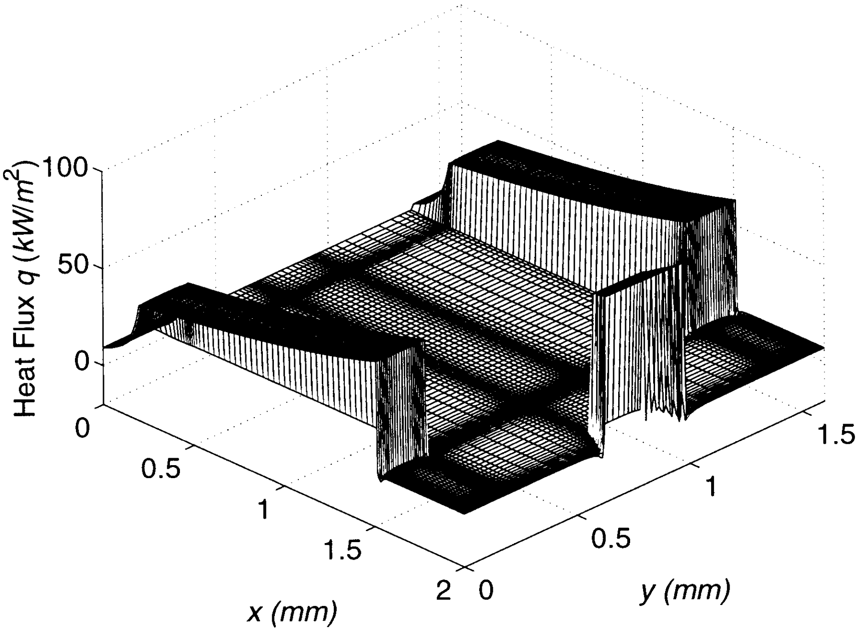

Fig. 6. Heat flux between the slider and air bearing at temperature difference $T_s - T_d = 1$ K.

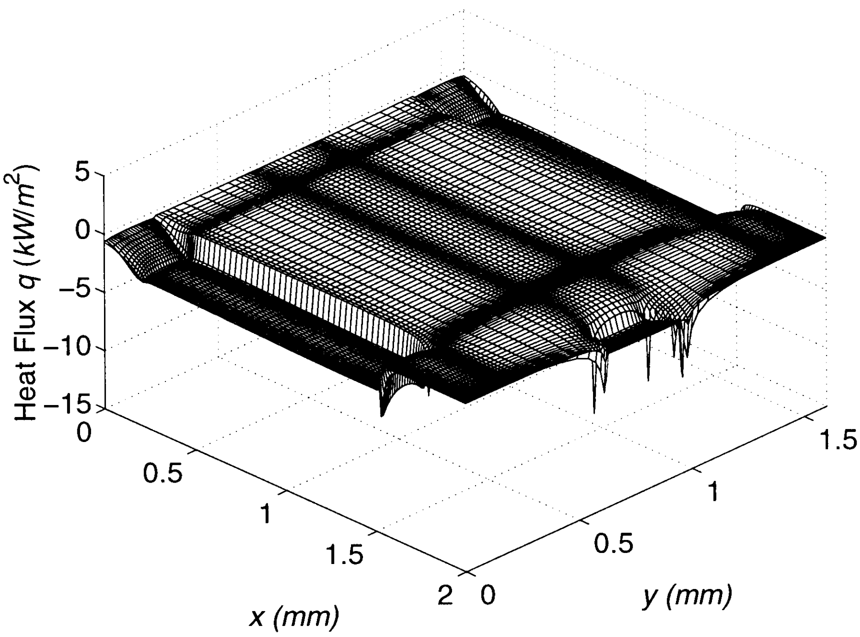

Fig. 7. Heat flux between the slider and air bearing at temperature difference $T_s - T_d = 0$ K.

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 the 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 situation, the air bearing is regarded as a coolant. But when the temperature difference is nearly equal to zero, viscous dissipation dominates

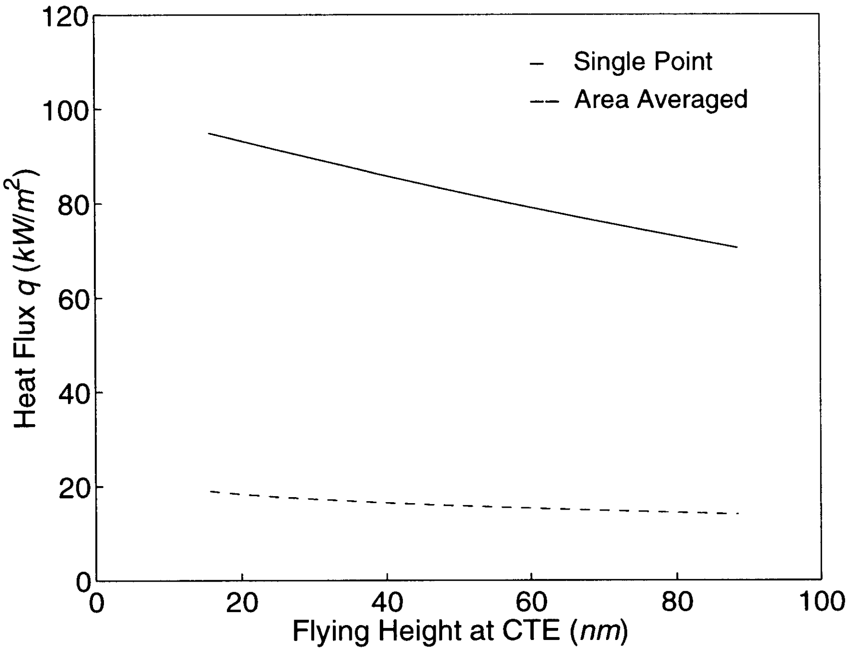

Fig. 8. Heat flux vs central trailing edge flying height at temperature difference $T_s - T_d = 1$ K.

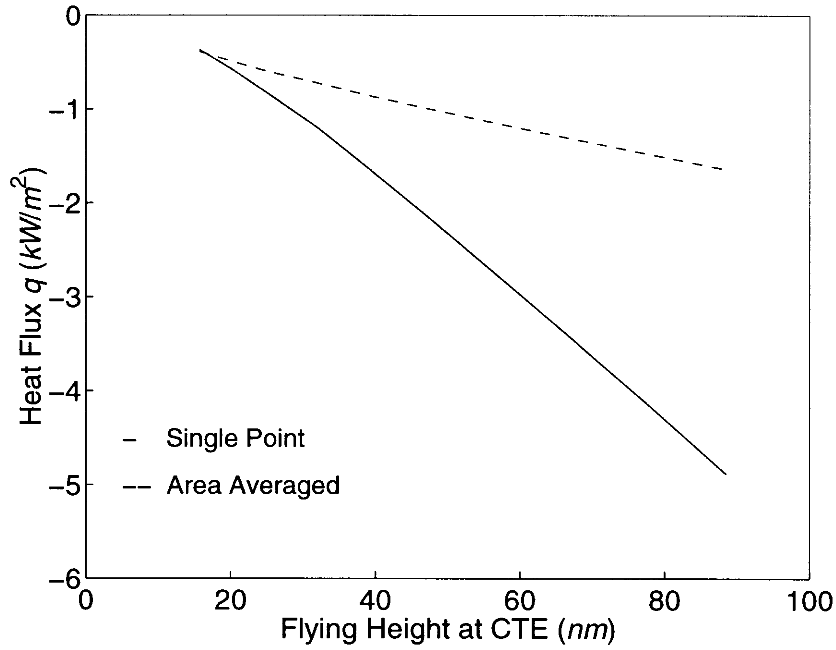

Fig. 9. Heat flux vs central trailing edge flying height at temperature difference $T_s - T_d = 0$ K.

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.

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