VARIATION OF THE HEAT FLUX BETWEEN A SLIDER AND THE AIR BEARING WHEN THE SLIDER FLIES OVER AN ASPERITY

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Abstract

In this report, we introduce a quasi-steady model for the heat transfer in an air bearing combined with a dynamic air bearing simulator. Using this heat transfer model, we study the mechanism of the variation of the heat flux between the slider and air bearing when the slider flies over an asperity without contact. The simulation results show that the heat flux is related to the air bearing thickness. When a slider flies over an asperity, its flying height fluctuates, which causes the air bearing thickness to fluctuate, and this in turn causes the heat flux between the slider and the air bearing to fluctuate. Corresponding to a decrease of the flying height, the heat flux out of the slider increases and vice versa. This result explains why a MR head read-back signal, which is related to the MR element's temperature and thus to the heat flux in or out of the MR sensor, follows the variation of the flying height when the slider flies over an asperity without contact. The variation of the MR signal output due to the variation of the heat flux is estimated to be about 10% of its normal output.

Key words: heat transfer, air bearing, asperity.

Introduction

Thermal noise has a significant effect on the read-back signal of a magnetoresistive (MR) head because the resistance of the MR sensor is temperature dependent (Gorter, Potgiesser and Tjaden, 1974). A well-known thermal disturbance is induced by the temperature rise nearby a MR sensor due to a head/disk contact, which is referred as the “thermal asperity” phenomenon. Using a theoretical model, Hempstead (1974) studied the thermal response of a MR head for friction heating between the head surface and dust particles or other asperities on the recording medium surface during relative motion between the head and medium. In his model, he assumed that the heat source produced by the interaction between the slider surface and a particle or an asperity was a point heat source moving across the surface of a semi-infinite solid, and the heat transfer in the air bearing was negligible and the solid surface could be regarded as insulated except for a moving point heat source. Thus, the transient temperature distribution inside the solid could be obtained by integration (Carslaw and Jaeger, 1986).

Jander, et al. (1996) proposed a simplified geometric model for analyzing the heat conduction in MR heads. They assumed that the current in the MR sensor is uniform, resulting in a uniform heat generation. By neglecting the heat transfer in the air bearing, they simplified the problem to that of a planar rectangular heat source (MR head) embedded in an infinite stratified medium consisting of the gap dielectric, shields, underlayer and overcoat. With these assumptions, the temperature distribution around such a source could be found by integrating the heat conduction equation (Carslaw and Jaeger, 1986).

A common point in the works mentioned above is that the heat transfer in the air bearing was neglected. But a recent work (Tian, Cheung and Wang, 1997) showed that such a simplification is questionable. They found experimentally that as a slider flies over an asperity without contact, the MR signal fluctuates similar in waveform to the fluctuation of the slider's flying height. Since no contact was observed in the experiment, they concluded that the signal fluctuation, which is related to the temperature variation of the MR sensor, was caused by the fluctuation of the heat transfer in the air bearing, and that the air bearing acted as a "coolant".

To investigate the "cooling" effect of the air bearing, Zhang and Bogy (1997) proposed a model for the steady heat transfer in an air bearing, and they obtained an expression for the heat flux between the slider and the air bearing. According to their analysis, the heat flux is comprised of two parts: heat conduction due to the non-zero temperature difference between the slider and disk surfaces, and viscous dissipation of the airflow within the air bearing. After simulating various cases, they concluded that the heat transferred to the air bearing increases with a decrease of the flying height (or disk rotation speed) for the case when the slider has a higher surface temperature than the disk.

In this report, we expand Zhang and Bogy's (1997) result to the case of a slider flying over an asperity, and we introduce a dynamic model that determines the fluctuation of the heat transfer between a slider and the air bearing. With this dynamic heat transfer model, we study the mechanism which causes the fluctuation of the read-back signals of a MR head observed by Tian, et al. (1997).

Model

The heat flux between a slider and the air bearing is influenced by the slider’s flying height, pressure distribution in the air bearing and the disk rotation speed (Zhang and Bogy, 1997). When a slider flies over an asperity or a bump, its flying height fluctuates, which causes the pressure and the heat transfer in the air bearing to also fluctuate. To study the relationship between the variation of the heat transfer and the variation of the flying height, we first need to obtain the pressure and velocity distribution in the air bearing for each transient flying state. As an approximation, we assume that the physical properties are constant in the following analysis.

(1) *Dynamics of the Slider*

The two-dimensional equations of the motion of a slider flying over a rotating disk are:

$$m\frac{d^2z}{dt^2} = F_s + \int_A (p - p_a)dA, \tag{1}$$

$$I_\theta \frac{d^2\theta}{dt^2} = M_{s\theta} + \int_A (p - p_a)(x_g - x)dA, \tag{2}$$

$$I_\phi \frac{d^2\phi}{dt^2} = M_{s\phi} + \int_A (p - p_a)(y_g - y)dA, \tag{3}$$

where m is the slider’s mass, z is the slider’s vertical displacement, θ and φ are the slider’s pitch and roll angles, I_θ and I_φ are the slider’s moments of inertia, x_g and y_g are the coordinates of the slider’s center of gravity. F_s, $M_{s\theta}$ and $M_{s\varphi}$ are the force and moments exerted on the slider by the suspension, p is the pressure distribution in the air bearing, and p_a is the ambient air pressure.

Clearly, to solve equations (1)~(3), we need to know the pressure distribution p in the air bearing, which can be obtained by solving the modified Reynolds equation expressed as follows:

$$\sigma \frac{\partial}{\partial T}(PH) = \frac{\partial}{\partial X}\left[\hat{Q}PH^3 \frac{\partial P}{\partial X} - \Lambda_x PH\right] + \frac{\partial}{\partial Y}\left[\hat{Q}PH^3 \frac{\partial P}{\partial Y} - \Lambda_y PH\right]. \qquad (4)$$

Equation (4) is a generalized unsteady Reynolds equation in the non-dimensional form, where, $X=x/L$ and $Y=y/L$ are non-dimensional coordinates in the air bearing, $H=h/h_m$ is the non-dimensional air bearing spacing, $P=p/p_a$ is the non-dimensional pressure in the air bearing, $T=\omega t$ is the non-dimensional time, and L, h_m and ω are, respectively, the slider's length, initial given flying height at the central trailing edge and the disk rotation speed. $\sigma=12\mu\omega L^2/p_a h_m^2$ is the squeeze number, $\Lambda_x=6\mu UL/p_a h_m^2$ and $\Lambda_y=6\mu VL/p_a h_m^2$ are the bearing numbers in the x and y directions, and $\hat{Q}$ is the Poiseuille flow factor (Ruiz and Bogy, 1989).

(2) *A Quasi-steady Heat Transfer Model in the Air Bearing*

When a slider flies over an asperity, its flying height fluctuates with time. The same is true of the pressure and heat transfer in the air bearing. Therefore, strictly speaking, the heat transfer in the air bearing in this case is an unsteady problem.

Extending the energy equation for the air bearing (Zhang and Bogy, 1997) to the unsteady case, we obtain the result:

$$\rho c_p \frac{\partial T}{\partial t} = k\frac{\partial^2 T}{\partial z^2} + \mu\left(\frac{\partial u}{\partial z}\right)^2 + \mu\left(\frac{\partial \upsilon}{\partial z}\right)^2, \qquad (5)$$

where ρ, c_p, k and μ are, respectively, the density, specific heat, heat conductivity and viscosity of the air, T is the temperature, and u and υ are velocity components of the airflow.

Note that an important characteristic of a slider air bearing is that its thickness is ultra-thin. For example, a typical dimension of the thickness is about 50 *nm* or less for a current MR head. For such a small thickness, we can expect that any small thermal disturbance may cause a transient change in the temperature distribution, or the temperature distribution may shift to a new equilibrium very quickly. To justify this view, let's look at the transient term (LHS) and the conduction term (1st term in RHS) in Eq. (5). If we assume the magnitude of the temperature variation in the conduction term is $\Delta T_{cond} \sim T_s - T_d$, where T_s and T_d are the temperatures of the slider and disk surfaces, then the magnitude of the temperature variation in the transient term can be approximated as $\Delta T_{tran} \sim (\Delta h/h)\, \Delta T_{cond}$, where Δh is the variation of the flying height. Usually, $\Delta h/h$ is smaller than 1 for a flying slider. If we further assume that $t \sim L/U$ and $z \sim h$, we can write the ratio of the transient term to the conduction term as: $(\Delta h/h)(h^2 U/L\alpha) \sim Pr Re_h h/L$, where $\alpha = k/\rho c_p$, $Pr = \nu/\alpha$ and $Re_h = Uh/\nu$. For the air bearing and slider studied in this report, $Pr \sim 0.7$, $h \sim 10^{-8}\ m$, $L \sim 10^{-3}\ m$, $U \sim 10\ m/s$, $\nu \sim 10^{-5}\ m^2/s$. Thus $Pr Re_h h/L$ has a magnitude of 10^{-7}, or the transient term is negligible compared with the conduction term. Dropping the transient term in (5), we can write the energy equation as:

$$0 = k\frac{\partial^2 T}{\partial z^2} + \mu\left(\frac{\partial u}{\partial z}\right)^2 + \mu\left(\frac{\partial \upsilon}{\partial z}\right)^2 . \qquad (6)$$

Equation (6) is actually a quasi-steady expression of the energy equation. In a similar way, the N-S equations can also be simplified. Solving them by applying the slip condition for velocity and the jump condition for temperature, we obtained the temperature distribution and then the heat flux between the slider and the air bearing using Fourier's Law (Zhang and Bogy, 1997), which can be expressed as:

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

$$\frac{Uh^3}{6(h + 2b\lambda)(h + 2a\lambda)}\frac{\partial p}{\partial x}, \qquad (7)$$

where q is the heat flux, λ is the mean free path of the air, $a=(2-\sigma_m)/\sigma_m$ and $b=2(2-\alpha_T)\gamma/\alpha_T(\gamma+1)Pr$, σ_m and σ_T are momentum and thermal accommodation factors, and γ is the ratio of specific heats.

Solution Approaches

The solution of the heat flux variation is decoupled from the solution of the dynamic slider air bearing because of the introduction of the quasi-steady heat transfer approximation. In solving the dynamic slider air bearing problem, we need to solve the slider motion equations (1~3) and the Reynolds equation (4) simultaneously. The Reynolds equation is discretized using Partanka's control volume method and solved by an ADI method combined with a multi-grid control volume method, and equations (1~3) are integrated directly. At each iteration, the pressure profile is obtained first by solving the Reynolds equation for a given flying height. Then the pressure profile is used to solve equations (1~3) to obtain the new displacements of the slider. The new displacements are compared with the previous ones to check if further iteration is needed. The detailed description of these approaches can be found in the related documents (Cha and Bogy, 1995; Lu and Bogy, 1994; Hu, 1996) and will not be presented in this report.

With the pressure distribution and the air bearing spacing obtained, the pressure gradient can be calculated and then the heat flux between the slider and air bearing is obtained using equation (7) for each flying state (or each iteration step). Note that the air bearing thickness used in solving the Reynolds equation is evaluated by considering the height of the asperity, bump, or any other roughness on the disk surface. This thickness is also used in solving the heat transfer in the air bearing. The whole solution procedure is implemented by using a thermal analysis code combined with the CML Air Bearing Dynamic Simulator (Hu and Bogy, 1995).

Simulation Results and Discussions

For convenience, we choose a 50% (2*mm*×1.6*mm*) tri-pad slider (Fig. 1(a)) as an example in the analysis. The slider has taper length and angle of 0.2 *mm* and 0.01 *rad*, respectively, and is loaded by 3.5 *g*. It is fixed at a radial position r=23 *mm* with a disk rotation speed 6400 *rpm*. Under these given conditions, the steady state flying height is 44 *nm*.

Figure 1(b) shows the heat flux distribution between the slider and the air bearing under the steady flying state for $T_s – T_d$=0 °C. Under this condition, the only heat transfer comes from the viscous dissipation in the air bearing which acts to heat the slider (Zhang and Bogy, 1997). For convenience, we plot as positive values the heat flux from the air bearing into the slider in this figure. It is seen that a relatively large heat flux resulting from the viscous dissipation occurs at the points around the trailing edge rail where usually there exists a steep pressure gradient.

In the following analysis, unless otherwise stated we use the new convention that a positive heat flux means the heat is transferred out of the slider.

(1) *Slider Flying Over a Square Asperity*

In this section, we study the heat transfer between a slider and the air bearing when the slider flies over a rectangular asperity. The asperity used is 30 *nm* in height, 150μm in length and 300μm in width. Since the slider usually has a higher temperature than the ambient air or disk, we take $T_s - T_d = 20$ °C and we assume that these temperatures remain constant through the whole process. Figure 2 shows the variation of the air bearing thickness (Fig.2(a)) and the heat flux (Fig.2(b)) at a single point close to the central trailing edge (about 5 μm away from it). We also plot the heat conduction (Fig.2(c)) and viscous dissipation (Fig.2(d)), components separately to see their relative contribution to the total heat flux. From these two figures it is clear that the heat conduction dominates the heat transfer in this case.

When the tri-pad slider flies over the asperity located along its centerline, the air bearing thickness does not change until the asperity reaches and passes under the trailing edge rail (TER) (Fig.2(a)). This is because the slider's flying state is affected by the pressure profile in the air bearing, which does not change much when the asperity goes through the recessed region. When the asperity gets close to the TER, the air bearing thickness first slightly increases, which is caused by the increase of the slider's flying height, and then decreases sharply as the TER passes over the asperity. Note that the air bearing thickness here is a combined result of the increase of the flying height and the reduction of the asperity height. When the asperity leaves the trailing edge, the air bearing thickness increases sharply and then decreases and starts to vibrate around its steady value.

Corresponding to the variation of the air bearing thickness, the heat conduction decreases slightly at first and then increases sharply (Fig.2(c)) when the asperity reaches the TER. When the asperity leaves the trailing edge, it goes down sharply and then goes up and starts to vibrate afterwards. Comparing Fig.2(c) with Fig.2(a), we see that the variation of the heat conduction follows inversely the air bearing thickness almost exactly. That is, the heat conduction increases with a decrease of the air bearing thickness and decreases with an increases of the air bearing thickness. In other words, more heat is transferred out of the slider when the air bearing thickness is smaller. This is a natural conclusion because the smaller the thickness, the smaller the thermal resistance for conduction.

From the results of the viscous dissipation (Fig.2(d)), we draw the same conclusion as for the heat conduction. But here the heat flux takes negative values, which means the heat is transferred from the air bearing to the slider. Therefore, less heat is dissipated into the slider with the decrease of the air bearing thickness.

The total heat flux is the sum of the above two portions of heat transfer (Fig.2(b)). Its profile is the same as that in Fig.2(c) and 2(d). Thus, we can say that the “cooling” effect increases with a decrease of the air bearing thickness. This conclusion is similar to our previous results (Zhang and Bogy, 1997), in which the variation of the heat flux was obtained for various steady cases by changing the flying height through changing the disk rotation speed.

In the above analysis, we studied the mechanism of the heat flux variation for a single point. But a MR sensor is actually affected by the heat flux over a finite surface area around it. To study this overall thermal effect, we use the same case as above but we focus on the average values of the heat flux determined over the air bearing surface of the TER (Fig.3(b), 3(c) and 3(d)). We also shift to compare them with the flying height at the central trailing edge (FH-CTE) (Fig.3(a)) instead of with the local air bearing thickness at a point. It is seen that the FH-CTE does not change before the asperity reaches the TER (Fig.3(a)). When the asperity reaches the TER and goes through the air bearing, the FH-CTE increases sharply and then decreases when the asperity leaves the trailing edge, after which it vibrates around the steady state flying height. Corresponding to the variation of the FH-CTE, the response of the average heat conduction can be divided into two intervals as shown in Fig.3(c). In the interval I, the average heat conduction increases when the asperity begins to occupy the air bearing of the TER, which leads to a decrease of the average air bearing thickness. After that, the heat conduction decreases due to the increase of the average air bearing thickness contributed by the asperity leaving the TER air bearing. In the interval II, since the effect of the asperity height vanishes, the variation of the air bearing thickness directly follows that of the flying height. Therefore, the average heat conduction increases with decreases of the FH-CTE, and decreases with increases of the FH-CTE.

It is interesting to note that the average viscous dissipation decreases when the asperity begins to occupy the air bearing of the TER, or more heat is dissipated into the slider when the average air bearing thickness decreases. This result is opposite to that for a single point analyzed above (see Fig.2(d)). In our previous report (Zhang and Bogy, 1997), we pointed out that the maximum viscous dissipation increases in magnitude as the FH-CTE, or the average air bearing thickness, decreases. The reason for this may be that the pressure profile at the corners of the TER, where there exists more drastic pressure variations at the smaller air bearing thickness, becomes steeper at the smaller air bearing thickness, which increases the pressure gradient and also the magnitude of the viscous dissipation at these points. Due to the contribution of these points, the magnitude of the average viscous dissipation increases when the asperity goes through the air bearing under the TER. After the asperity leaves the trailing edge, the average viscous dissipation fluctuates with a small magnitude.

The average heat flux is shown in Fig.3(b). Its profile is close to that of the heat conduction because the heat conduction dominates the heat transfer in this case. Clearly, more heat is transferred out of the slider when the FH-CTE becomes lower, except for during a small period in interval I.

(2) *Initial Impulse*

An often-met case for a working hard drive is that the drive is acted on by a sudden external force, which causes the flying height to fluctuate drastically. To study the corresponding heat flux variation, we give an initial impulse to a steadily flying slider. For simplification, we only give a non-zero value $w_{g0}=0.001$ m/s to the initial vertical velocity at the gravity center of the slider. The simulation results are shown in Fig.4.

It is seen that the flying height has a deflection away from its steady value at the beginning, then it vibrates and damps to its steady flying state (Fig.4(a)). Since there is no effect of asperities, the variation of the flying height reflects the variation of the air bearing thickness. Following the variation of the flying height, the heat flux at a single point (same point as in the asperity case) increases with the decrease of the flying height (Fig.4(c)). Similarly, the average heat flux also increases with the decrease of the FH-CTE (Fig.4(b)).

(3) *Effect of the Heat Flux Variation on the MR Output*

We know that the MR read-back signal is very sensitive to the temperature variation in the MR sensor, which is affected significantly by the heat transferred in or out of the MR sensor. Therefore, the variation of the heat flux caused by the fluctuation of the air bearing thickness will affect the temperature in the MR sensor, and hence its signal output. To evaluate the magnitude of this effect, we introduce a simple heat transfer model for the MR sensor, in which we assume that the MR sensor, together with its shields, is a thin plate (Fig. 5). The plate has an uniform temperature caused by balancing the heat generated by the current through it and the heat transfer between the plate and the air bearing. When the slider flies over an asperity, the heat flux between the plate and the air bearing changes, which in turn changes the thermal balance in the sensor and causes a variation of the temperature in it. This change can be approximated as:

$$\rho_s c_s h_s \Delta T_s \approx \Delta q \Delta t \qquad (8)$$

where ρ_s, c_s and h_s are, respectively, the density, specific heat and thickness of the MR sensor, ΔT_s is the temperature variation in the MR sensor, Δq is the variation of the heat flux between the plate and air bearing and Δt is the heating time.

For a typical MR sensor, we can take ρ_s~8000 kg/m^3, c_s~400 $J/kg\cdot K$ and h_s~2 μm. We also have Δq~7×10^4 W/m^2 and the related Δt~10^{-5} sec (Fig. 2) for the slider flying over an asperity. Thus, we obtain the temperature variation ΔT_s~0.11 °C by equation (8). Based on a correlation between the temperature and resistance (Tian, et al, 1997), we obtain the resistance variation in the MR sensor as: $\Delta R_s=\alpha R_s \Delta T_s \approx 0.00239\times30\times0.11\approx0.008\ \Omega$, where R_s is the resistance of the MR sensor which is about 30 Ω and ΔR_s is its variation, and α is a coefficient obtained by experiment. Note that the typical resistance variation during MR readback is about 0.5% (Waldera, 1997). So the flying height variation caused by a slider flying over an asperity can result in about 5% of the variation of the MR readback signal.

Conclusions

In this report, we introduced a quasi-steady model for the heat transfer in the air bearing combined with a dynamic air bearing design model. Using this model, we studied the mechanism of the variation of the heat flux when a slider flies over an asperity and when a slider is given an initial impulse. The simulation results show that the heat flux is related to the air bearing thickness. That is, a decrease in the air bearing thickness will increase the heat transferred out of the slider. When a slider flies over an asperity, its flying height fluctuates, which causes the air bearing thickness and the heat flux between the slider and the air bearing to fluctuate simultaneously. Since a decrease of the flying height usually causes a decrease of the air bearing thickness, it therefore increases the heat transferred out of the slider.

A similar result is obtained for a flying slider given an initial impulse, which causes the fluctuation of the flying height and then causes a variation of the heat flux between the slider and the air bearing.

Based on a simple heat transfer model for the MR sensor, we estimate the temperature variation caused by the fluctuation of the slider's flying height to be about 0.1 °C, which causes the MR signal output to vary by about 10%.

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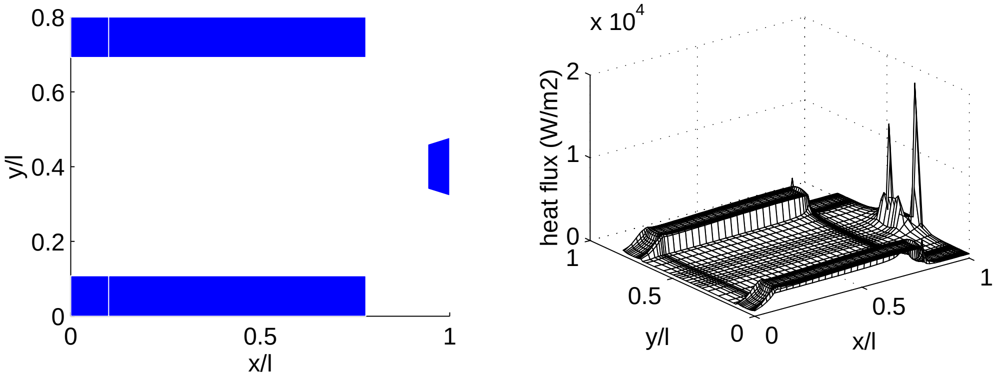
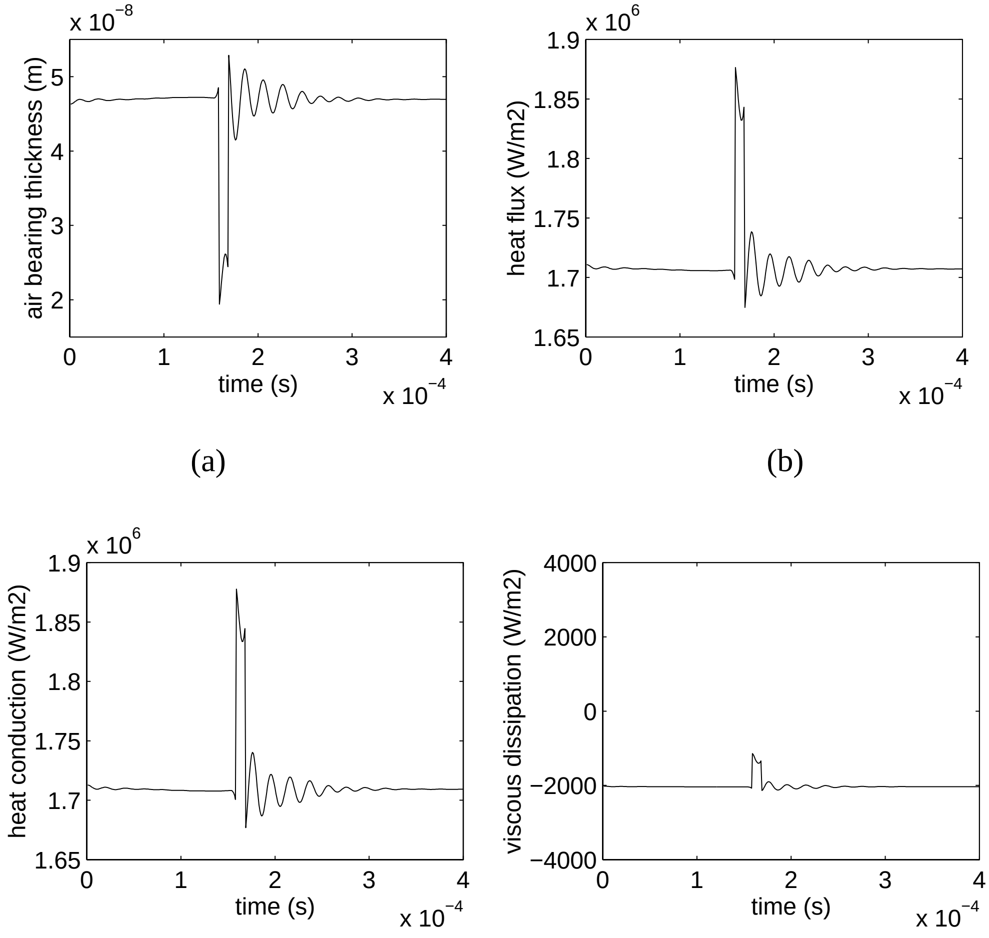


(a) (b)

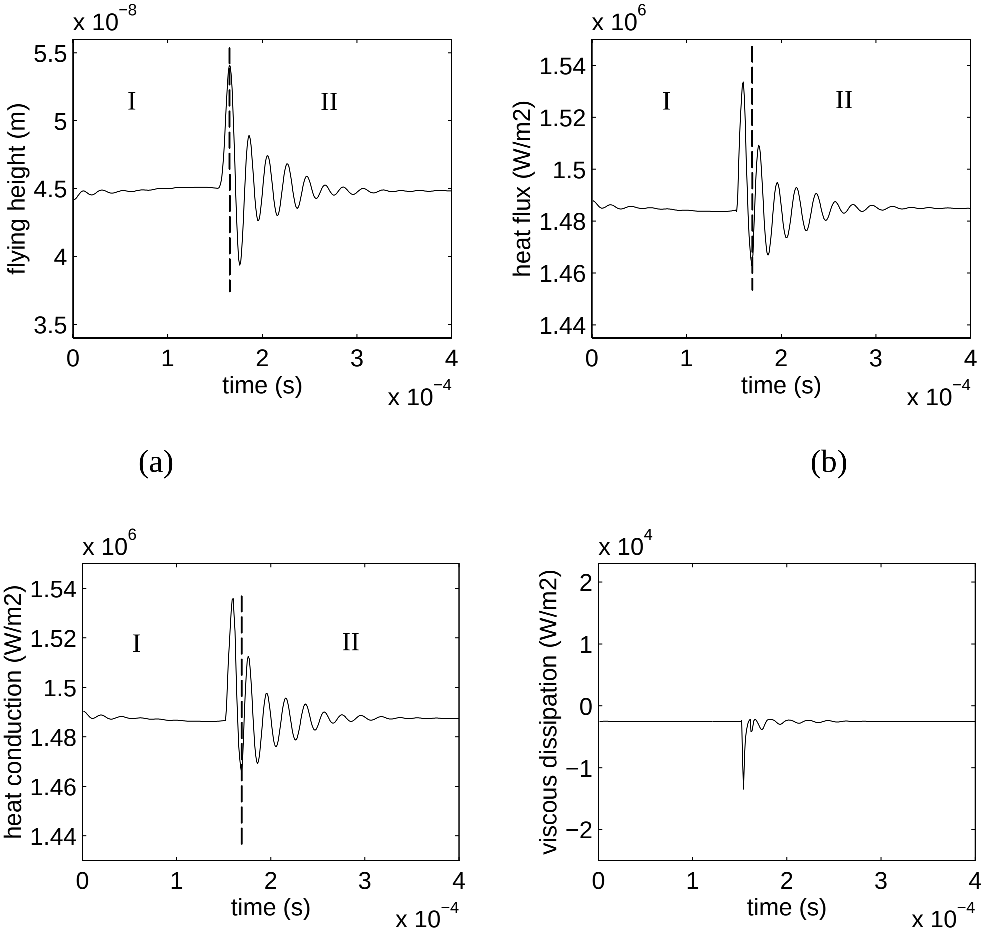
Fig.1 A tri-pad slider and the heat flux profile in the air bearing (ΔT=0°)



(a) (b)

(c) (d)

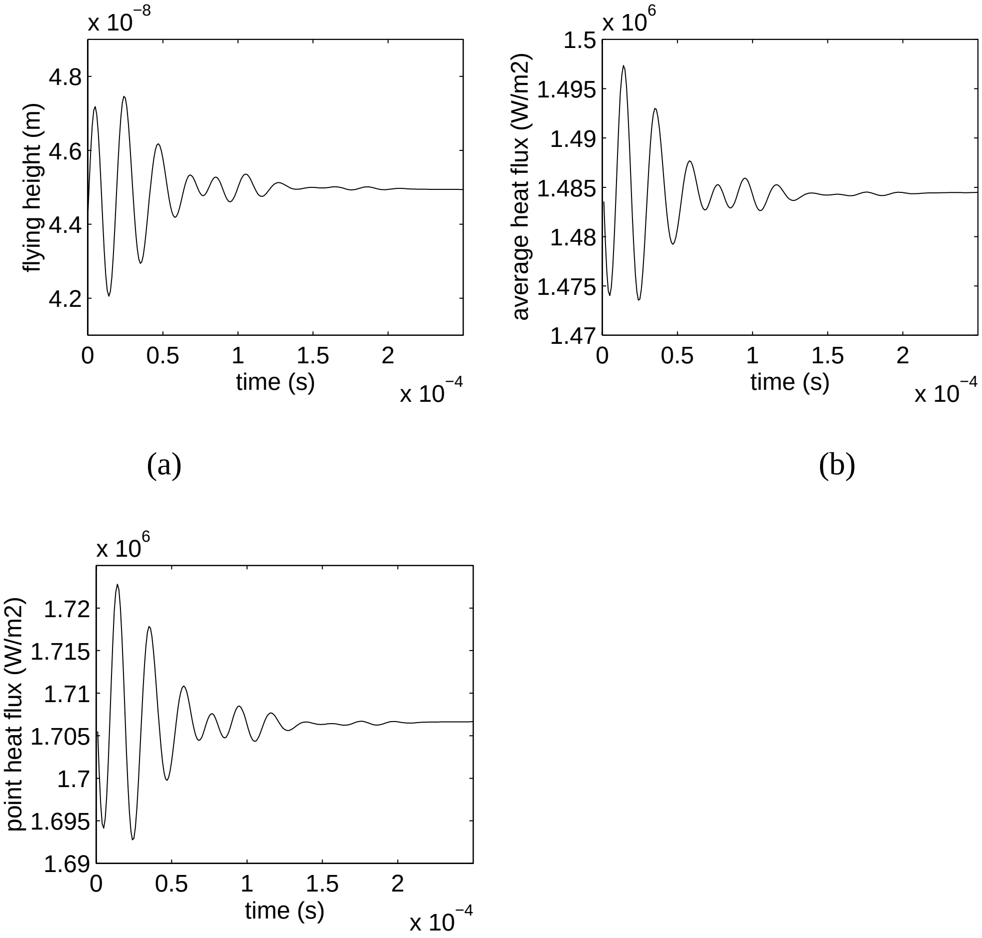
Fig.2 Air bearing thickness and heat flux (single point located at 5 μm from the CTE) for a slider passing over an asperity



(a) (b)

(c) (d)

Fig.3 Flying height and heat flux (averaged value over the TER) for a slider passing over an asperity



(a)

(b)

(c)

Fig.4 Flying height and heat flux for a slider given an initial impulse

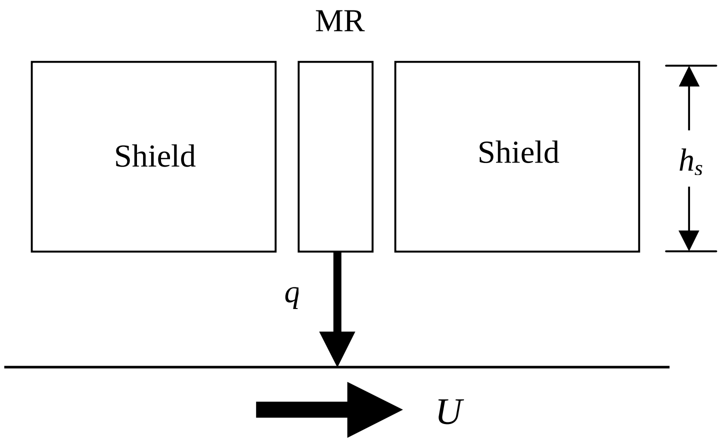


Fig. 5 Heat transfer model for the MR sensor