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FRICTION AND WEAR MECHANISM AT HIGH SLIDING SPEEDS

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A new melting wear theory of the slider is proposed based on the non-steady heat conduction equation. The friction coefficients calculated with the theory agreed well with those reported by previous workers. The estimated values of the slider wear also agree well with those obtained experimentally. The melting wear theory is confirmed to estimate the melting wear quantitatively from thermal properties of sliders. Based on the theory, a practical selection method of slider material has been established.

NOMENCLATURE

a_B	thermal deffusivity of bore (m^2/s)
a_S	thermal deffusivity of slider (m^2/s)
h	melting thickness of slider (m)
H_S	latent heat of slider (J/kg/K)
p	contact pressure (Pa)
q_B	heat flux to bore (W/m^2)
q_S	heat flux to slider (W/m^2)
Q_B	thermal energy to bore in an unit area (J/m^2)
Q_S	thermal energy to slider in an unit area (J/m^2)
Q_T	total thermal energy in passage of slider in an unit area (J/m^2)
t	elapsed time from slider contact (s)
t^*	corrective factor of time (s)
t_{Bal}	elapsed time from shell body started (s)
t_{ms}	total time interval defined as the sum of solid- and partial melting-contact time (s)
t_{se}	time interval of solid contact condition (s)
T_{0S}	initial temperature of slider (K)
T_{0B}	initial temperature of bore (K)
T_{MS}	melting point of slider (K)
v	velocity of shell (m/s)
W_S	slider width (m)

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WR	wear rate in an unit time (m/s)
λ_S	thermal conductivity of slider (W/m/K)
λ_B	thermal conductivity of bore (W/m/K)
μ_{mean}	mean friction coefficient (--)
μ_s	friction coefficient in solid state contact condition (--)
μ_l	friction coefficient in melting state contact condition (--)
ρ_S	density of slider (kg/m^3)

INTRODUCTION

The traveling shell is rotated as the slider goes along the bore groove. If the slider is worn away, an adequate spin rate could not be attained. Recently, the excessive wear of the slider has been reported [1], because higher shell velocity has been required. Wear at high sliding speeds was experimentally studied by R. S. Montgomery [2]. No one has reported theoretical treatment on the slider wear. Our previous study revealed that the surface of the slider melts within a few centimeters from the onset of shell movement [3], demonstrating that the slider wear occurs mostly in the molten condition.

The present paper describes the theory of slider wear by applying one-dimensional non-steady heat conduction equation. Thermal properties of copper-based materials were carefully measured at higher temperatures. Friction coefficients between slider and bore surface were estimated and contact conditions are discussed. Based on the discussions above an adequate material for slider is presented.

THEORY

Fig. 1 shows a relationship between the contact condition, the surface temperature and heat flux when the slider moves at an arbitrary position in a bore. In this figure, the following two assumptions are made; (1) temperature of the entire slider surface has reached its melting point, (2) the contact condition changes from solid state contact to partial melting state contact and finally to melting state contact as direction goes to the rear end.

In the solid state contact condition, the surface temperature of the front area of the slider has reached the melting point, but the surface melting is restricted by the latent heat. In this condition, the solid state friction causes larger heat flux than those flowing in to the slider as the result the surface temperature of bore increases.

In the partial melting state contact condition, the generating heat flux decreases to a level to maintain the bore surface temperature at the melting point of the slider. Decrease in heat flux is brought by the decrease in friction coefficient due to the partial melting.

In the melting state contact condition, the generating heat flux becomes constant with the melting lubricated friction. In this condition, heat flux to the bore decreases because time interval from the onset of contact increases with direction toward rear end of slider. Accordingly, the thickness of melting layer increases as distance goes to the rear end.

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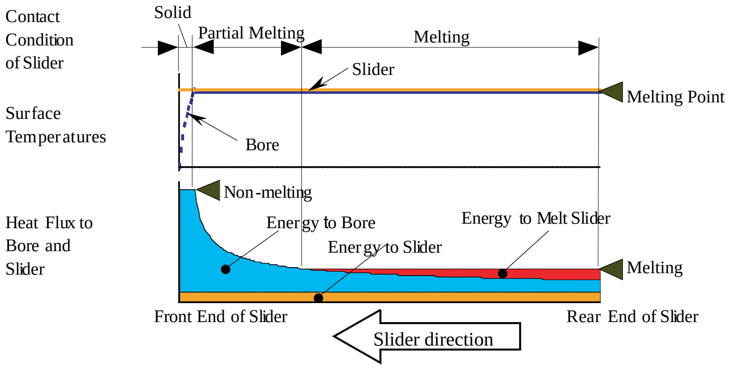

Figure 1 – Schematic of contact condition of slider, surface temperatures and heat flux to slider and bore sliding at a constant contact pressure and velocity.

Based on this model and an one-dimensional non-steady heat conduction equation, a theory of the slider wear for an arbitrary section of the bore is proposed. The nomenclatures used in the following equations are described above. The heat flux to the slider (q_S) is defined in eq (1), under assumption that the surface temperature of the slider reaches its melting point at the onset of shell movement. This assumption could be provable, because the surface melting was found to occur shortly after a shell starts to move [3].

$$q_S = \frac{\lambda_S (T_{MS} - T_{0S})}{\sqrt{\pi \cdot a_S \cdot t_{Bal}}} \tag{1}$$

A time interval of solid contact condition (t_{se}) is defined in eq (2). It is equal to the time to raise the bore surface temperature to the melting point of slider. The heat flux to the bore (q_B) is the result of subtracting heat flux to the slider from generated heat flux and defined in eq (3).

$$t_{se} = \frac{\pi}{4} \cdot \frac{\lambda_B^2}{a_B} \cdot \left(\frac{T_{MS} - T_{0B}}{q_B} \right)^2 \tag{2}$$

$$q_B = \mu_s \cdot p \cdot v - q_S \qquad \text{at} \quad t \le t_{se} \tag{3}$$

After the surface temperature of bore reaches the melting point of the slider, the surface temperature remains constant. A non-steady heat conduction equation for a constant surface temperature is adapted by introducing a corrective factor of time (t*) as defined in eq (4). So, heat flux to the bore (q_B) can be defined in eq (5).

$$t^* = \frac{1}{\pi} \cdot \frac{\lambda_B^2}{a_B} \cdot \left(\frac{T_{MS} - T_{0B}}{q_B} \right)^2 \tag{4}$$

$$q_B = \frac{\lambda_B (T_{MS} - T_{0B})}{\sqrt{\pi \cdot a_B (t - t_{se} + t^*)}} \qquad \text{at} \quad t > t_{se} \tag{5}$$

A total time interval (t_{ms}) is defined as the sum of solid contact time and partial melting contact time in eq (6).

$$t_{ms} = \frac{1}{\pi} \cdot \frac{\lambda_B{}^2}{a_B} \cdot \left(\frac{T_{MS} - T_{0B}}{\mu_l \cdot p \cdot v - q_S} \right)^2 + t_{se} - t^* \qquad (6)$$

Thermal energy transfered to the bore and solid slider in an unit area can be calculated by integrating each heat fluxes. Thus the generated thermal energy (Q_T), the thermal energy to the slider (Q_S) and thermal energy to the bore (Q_B) are described in eq (7) to (9).

$$\begin{aligned} Q_T &= \mu_s \cdot p \cdot v \cdot t_{se} + \mu_l \cdot p \cdot v \cdot \left(\frac{W_S}{v} - t_{ms} \right) \\ &\quad + \frac{2 \cdot \lambda_B (T_{MS} - T_{0B})}{\sqrt{\pi \cdot a_B}} \cdot \left(\sqrt{t_{ms} - t_{se} + t^*} - \sqrt{t^*} \right) - q_S \cdot (t_{ms} - t_{se}) \end{aligned} \qquad (7)$$

$$Q_S = \frac{\lambda_S (T_{MS} - T_{0S})}{\sqrt{\pi \cdot a_S \cdot t_{Bal}}} \cdot \frac{W_S}{v} \qquad (8)$$

$$\begin{aligned} Q_B &= \left(\mu_s \cdot p \cdot v - \frac{\lambda_B (T_{MS} - T_{0S})}{\sqrt{\pi \cdot a_S \cdot t_{Bal}}} \right) t_{se} \\ &\quad + \frac{2 \cdot \lambda_B (T_{MS} - T_{0B})}{\sqrt{\pi \cdot a_B}} \cdot \left(\sqrt{\frac{W_S}{v} - t_{se} + t^*} - \sqrt{t^*} \right) \end{aligned} \qquad (9)$$

When the generated thermal energy is larger than the total energy that flows to the solid slider and to the bore, the slider may be melted with the excess energy. The melting thickness (h) is defined in eq (10) under the assumption of uniform melting on the surface of the slider.

$$h = \frac{Q_T - (Q_S + Q_B)}{H_S \cdot \rho_S} \qquad (10)$$

The wear rate (WR) shown in eq (11) is obtained by dividing the thickness with the passing time of the slider.

$$WR = h \cdot v \big/ W_S \qquad (11)$$

A total wear of a slider could be estimated by integrating the wear rate in the whole process with time. In the estimation process described above, reliability of several thermal properties would be essential to obtain accurate estimation results.

THERMAL PROPERTIES

Table 1 shows the thermal properties necessary for the estimation of slider wear. The thermal diffusivities were measured with the laser flash method and the specific heats with the differential scanning calorimeter (DSC). The melting points and the latent heats were obtained with the differential thermal analysis (DTA). The thermal conductivities were calculated from the corresponding thermal diffusivity, specific heat and density.

Since red brass has the highest thermal diffusivity and thermal conductivity in these three copper-based alloys, red brass can be estimated to have the best resistance to the slider wear, because of its high heat diffusivity, melting point and latent heat.

Table 1 Thermal Properties of Slider Materials

Material		Red Brass	Brass	Al-Bronze
Density*	(kg/m^3)	8710	8370	7750
Specific heat**	(kJ/kg/K)	0.461	0.507	0.532
Thermal conductivity**	(W/m/K)	262	153	88
Thermal diffusivity**	(mm^2/s)	65.6	36.2	21.3
Melting point	(K)	1310	1161	1298
Latent heat	(kJ/kg)	154	105	143

* Room temperature ** 873K

DISCUSSION OF RESULTS

Friction Coefficients

Fig. 2 shows the relationships between sliding travel and friction coefficient calculated with the theory described above. This calculation was carried out on the condition that the contact pressure is 150 MPa and the sliding velocity is 200 m/s. These conditions are equivalent to those of the pin-on-disk test reported by R. S. Montgomery [2].

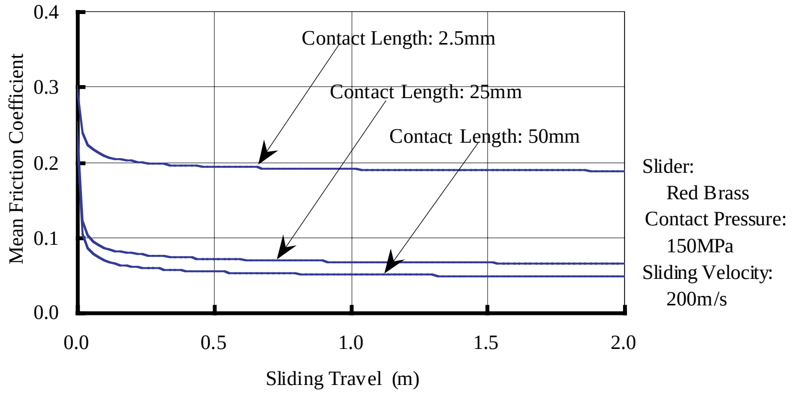

Figure 2 – Relationship between slider travel and mean friction coefficient.

The mean friction coefficient is an average value in the direction of the slider length, which is defined in eq (12).

$$\mu_{mean} = \frac{Q_T/(W_S/v)}{p \cdot v} \tag{12}$$

The mean friction coefficient rapidly decreased shortly after the onset of shell traveling and leveled off in the range of the sliding travel between 0.5 m and 1.0 m. The value of mean friction coefficient at higher sliding travels was about 0.2 in the case of contact length 2.5 mm, which agreed well with those reported by R. S. Montgomery [2].

Contact Conditions

Fig 3. shows the change in the contact condition for slider lengths ranging from 0.1 mm and 1.0 m. As the slider travels, the borders of contact conditions shifted toward the front end of the slider. It is natural to think that the heat flux into slider could decrease as the sliding travel (or sliding time) increases.

It should be noted that the partial melting contact might appear in the reported pin-on-disk test using 2.5 mm length specimen [2].

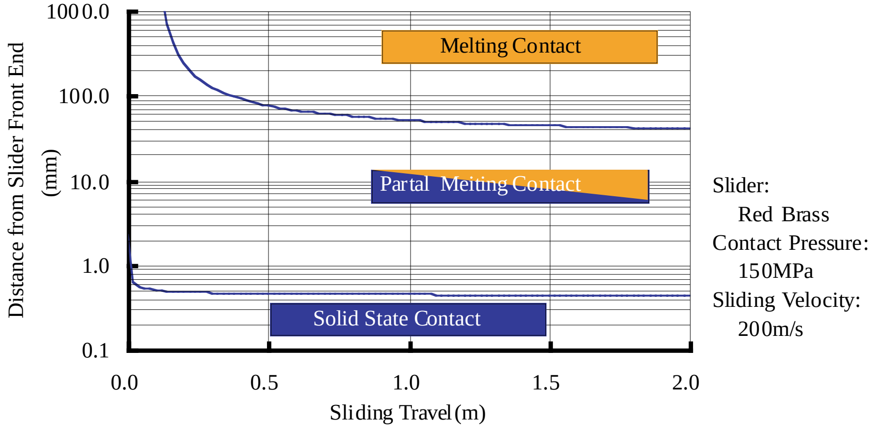

Figure 3 – Relationship between sliding travel and contact conditions.

Estimations of slider wear

Fig. 4 shows the calculated results of the melting wear versus sliding travel for several copper alloys. The values of the calculated wear of red brass, Al-bronze and brass increased in this order in the whole travel range. This estimated order agrees well with observed order in our laboratory. The value of the slider wear became constant above a slider travel of 4 m, whose value ranged from 1.5 mm to 3.7 mm depending on slider materials.

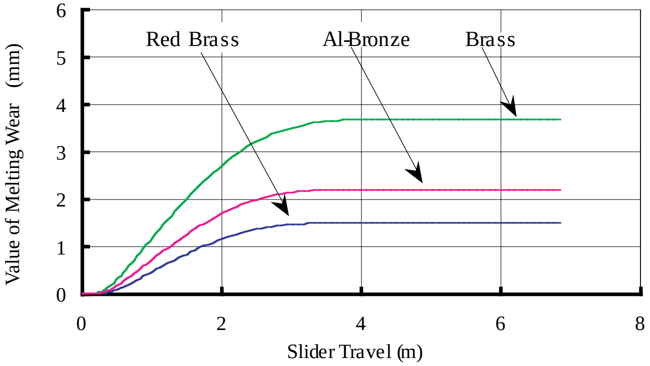

Figure 4 – Relationship between Slider Travel and Melting Wear.

According to our theory, a material with higher values of thermal properties (thermal diffusivity, thermal conductivity, melting point and latent heat) decreases slider wear, i.e. red brass slider exhibited better resistance in the melting wear compared to Al-bronze- and brass-slider.

CONCLUSIONS

(1) Our melting wear theory enables us to discuss the friction and wear phenomena quantitatively by using thermal properties of the slider materials.
(2) The mean friction coefficient decreases as the slider length increases.
(3) Within candidate materials, red brass which has higher values of thermal properties (thermal conductivity, thermal diffusivity, melting point and latent heat) was proved to be the most suitable material for the slider.

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