

Short communication

A fractional coverage model for gas–surface interaction in reciprocating sliding contacts

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Abstract

A model of fractional coverage in reciprocating sliding contact is developed. The sliding velocity profiles, the contact pressure distribution, the lengths of the slider or wear-track, and the periods of dwell where the slider is held stationary at the turn around locations are all variables. The model is evaluated for the condition of a constant fully reversing sliding speed with a uniform contact pressure and dwell. Plots are presented for surface area fractional coverage as a function of position illustrating the effects of individually varying vapor pressure, velocity, load, and dwell. The prediction of a steady state friction coefficient dependence on position is discussed, as well as the locations of maximum and minimum friction coefficient. The model predicts coverage to be enhanced by decreasing loads, decreasing sliding speeds, increasing gas pressures, and increasing periods of dwell.

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1. Introduction

Some of the early work on boundary lubrication used linear rule-of-mixtures in friction coefficient to model the process of fractional lubricant adsorption, Bowden and Tabor [1] and Kingsbury [2], with perhaps the earliest empirical modeling being done by Isemura [3]. Isemura found a nearly linear dependence on friction coefficient with the fractional area of a neutral-soap on glass plates. The first known modeling of the vapor lubrication (to these authors) is the work by Rowe [4] who modeled wear-rates assuming the surface coverage was a function of pressure and temperature (a Langmuir adsorption model [5,6] was used).

Since this early work, there have appeared a number of examples where a gaseous environment interacts with a bearing surface to form solid lubricous films that are being replenished as they wear away: using carbonaceous gases [7–20], phosphate containing vapors [21–27], hydrogen sulfide [28–31], and the vapor delivery of monomers [32]. When the interaction is favorable, it is often called ‘vapor-phase lubrication’, which was cursorily reviewed by Rao [33]. Similarly, there are a number of studies where a gaseous environment interacts with a material surface to af-

fect sliding friction [34–42] sometimes favorably and other times adversely.

Closed form analytical models are developed to predict the behavior of friction coefficient in steady-unidirectional sliding and steady combined rolling and sliding contacts [20] (shown in Fig. 1). These models were successfully applied to various vapor phase lubrication systems. The development of these models relies on a stationary location of contact under steady operating conditions. Thus, these models are not easily extended to the case of reciprocating motion. Reciprocating pin-on-disk tribometers are an extremely common laboratory apparatus, especially in studies of environmental friction dependence [20,26,27,30,31,36]. The analytical models developed for pin-on-disk and combined rolling and sliding contacts cannot be directly applied to these studies except at the mid-stroke location. The position dependent data, most likely being collected by researchers doing reciprocating pin-on-disk work, is not being published presumably because of the difficulty of developing models for this motion and interpreting the results.

2. Modeling

To bridge the gap between the models developed and validated for the pin-on-disk and combined rolling and slid-

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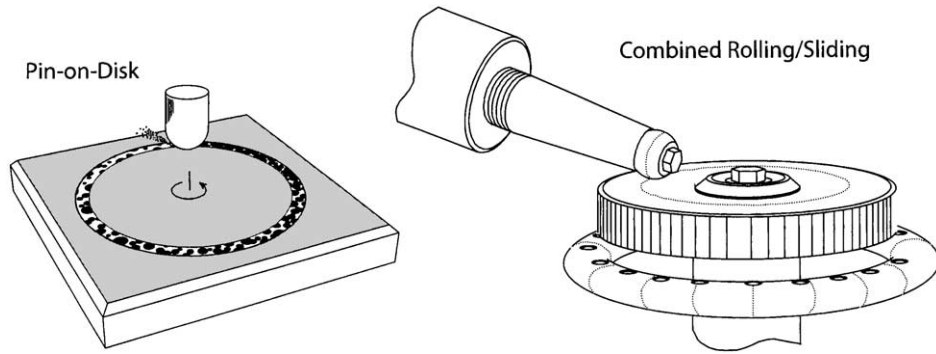


Fig. 1. Two different apparatuses for which steady-state solutions have been developed for fractional vapor-phase lubrication.

ing contacts with the experimental work on reciprocating tribometers, the model needs to predict friction coefficients at each location along the path under steady operating conditions. This is done by applying the previous models differentially through the entire path of the reciprocating tribometer. The reciprocating motion is assumed to have a sliding velocity that fully reverses during each cycle.

The model closely parallels a successful model for combined rolling and sliding contacts as well as pin-on-disk contacts [20]. New nomenclature for the reciprocating case

is defined: the superscripts *f* and *r* refer to the arbitrarily defined forward and reverse motion, respectively. A schematic of the deposition and removal process for reciprocating contacts is shown in Fig. 2. Fig. 3 illustrates variables unique to this modeling. The model can be non-dimensionalized following the techniques previously [20], but is not done for this short communication.

The model begins with the assumption that the deposition products are thin fractional films. Applying Archard's wear law differentially through the contact Blanchet and Sawyer

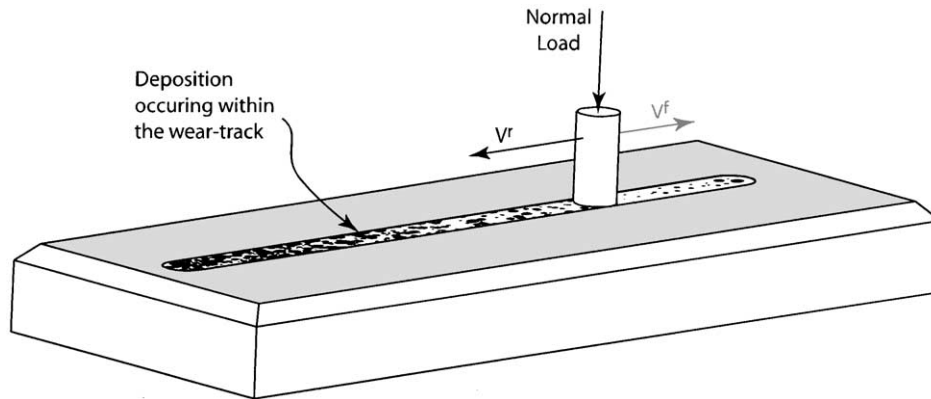


Fig. 2. A schematic of the reciprocating pin-on-disk contact.

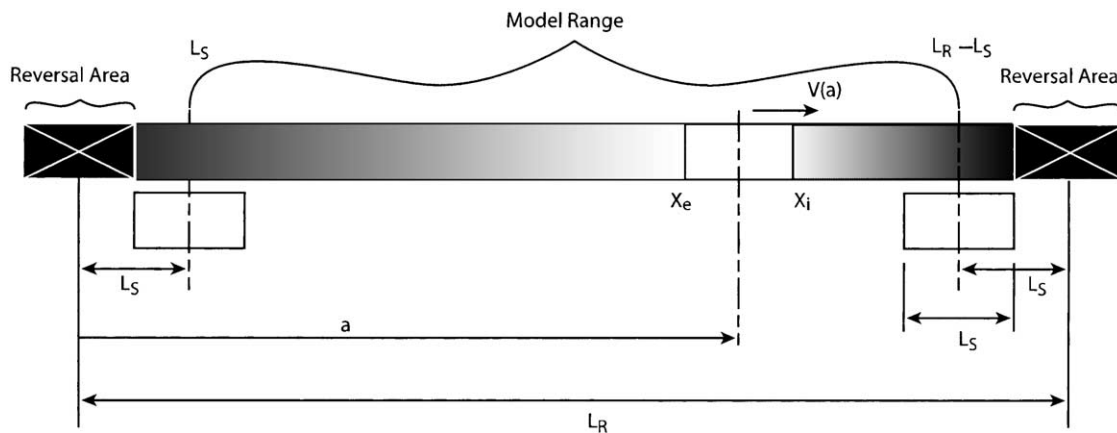


Fig. 3. A schematic of the nomenclature for the reciprocating pin-on-disk contact.

[43] developed a model to predict the surface area fractional coverage exiting the contact (X_e) as a function of the width of contact (w), the normal load (F_n), the dimensional wear coefficient of the film (K), and the entering (or inlet) surface area fractional coverage (X_i). Applying this model to a differential element that has a different inlet fractional coverage depending on the sliding direction (forward or reverse), gives Eqs. (1) and (2).

$$X_e^f = X_i^f e^{(-K/w)F_n} \quad (1)$$

$$X_e^r = X_i^r e^{(-K/w)F_n} \quad (2)$$

The Langmuir deposition model [5,6] is invoked to predict the entering (inlet) fractional coverage on a differential element [20]. The model is a function of the exiting surface area fractional coverage (X_e), a deposition constant (ν), the partial pressure of the gas (P), and the time of exposure (T').

$$X_i^f = 1 - (1 - X_e^f) e^{-\nu P(T^f)} \quad (3)$$

$$X_i^r = 1 - (1 - X_e^r) e^{-\nu P(T^r)} \quad (4)$$

Eqs. (1)–(4) are solved for the inlet fractional coverage (X_i). In Eqs. (5) and (6) the time of exposure is separated into a component of time during which the slider is moving (T^f or T^r) and the amount of time that the slider spends stationary at the reversal locations (dwell D).

$$X_i^f = \frac{1 + (e^{-F_n^*} - 1) e^{-\nu P(T^f + D)} - e^{-\nu P T} e^{-F_n^*}}{1 - e^{-\nu P T} e^{-2F_n^*}} \quad (5)$$

$$X_i^r = \frac{1 + (e^{-F_n^*} - 1) e^{-\nu P(T^r + D)} - e^{-\nu P T} e^{-F_n^*}}{1 - e^{-\nu P T} e^{-2F_n^*}} \quad (6)$$

A normalized load variable F_n^* is defined as, $F_n^* = (K/w)F_n$. The period T is defined as $T = T^r + T^f + 2D$. The return times T^f and T^r are specific to locations on the wear track (a), and are functions of the velocity profile ($V(s)$), the length of the reciprocating wear track (L_R), and the length of the slider (L_S). This is found as using Eqs. (7) and (8).

$$T^f = 2 \int_0^{a-(L_S/2)} \frac{1}{V(s)} ds \quad (7)$$

$$T^r = 2 \int_{a+(L_S/2)}^{L_R} \frac{1}{V(s)} ds \quad (8)$$

The fractional coverage of a differential element (X_a) that travels a distance δ through the contact is described by Blanchet and Sawyer [43]. The attached coordinate λ runs the length of the contact and P'' is the local contact pressure.

$$X_a = X_{ia} e^{(-K/w) \int_0^\delta P''(\lambda) d\lambda} \quad (9)$$

Using Eqs. (5)–(9), the fractional coverage on each differential element under the slider, and the average fractional coverage under the slider, can be found for any location of the slider.

3. Results for the case of uniform sliding speed, uniform pressure, and dwell

The case of uniform sliding speed and pressure are discussed, which mathematically implies infinite acceleration and deceleration of the counterface at the reversal locations. However, because the model neglects the areas of counterface under the reversal locations this steady sliding speed analysis assumes that the controllers are able to completely accelerate or decelerate the counterface during this period. Under the condition of uniform sliding speed (V_0), Eqs. (7) and (8) are simplified to give Eqs. (10) and (11), respectively.

$$T^f(a) = 2 \int_0^{a-(L_S/2)} \frac{1}{V_0} ds = \frac{2}{V_0} \left(a - \frac{L_S}{2} \right) \quad (10)$$

$$T^r(a) = 2 \int_{a+(L_S/2)}^{L_R} \frac{1}{V_0} ds = \frac{2}{V_0} \left(L_R - a - \frac{L_S}{2} \right) \quad (11)$$

$$T = T^r + T^f + 2D = \frac{2}{V_0} (L_R - L_S) + 2D \quad (12)$$

Under the conditions of a uniform-pressure P_0 the integral in Eq. (9) is simplified to:

$$\frac{-K}{w} \int_0^\delta P_0 d\lambda = -K \frac{F_n}{w} \left(\frac{\delta}{L_S} \right) \quad (13)$$

Starting from an arbitrarily defined ‘standard condition’ (Table 1). Predictions of the average fractional coverage under the slider at specific track locations to variations in the normal load, sliding speed, dwell time, and vapor pressure are plotted. Predictions of the inlet fractional coverage, average fractional coverage, and exiting fractional coverage for any slider position (a) for forward and reverse directions is shown in Fig. 4. The model shows that the local surface area fractional coverage (X) during the reverse motion is a mirror of the surface area fractional coverage during the forward motion, as described in Eq. (14).

$$X^r(a) = X^f(L_R - a) \quad (14)$$

The average fractional coverage in the forward direction of travel will be presented for the model predictions to changes in parameters from the ‘standard condition’.

Increasing load decreases fractional coverage, as shown in Fig. 5. As the load approaches zero, the average fractional

Table 1
Standard condition

K (m/N)	1.0×10^{-4}
F_n (N)	100
V_0 (m/s)	0.1
L_R (cm)	10
L_S (cm)	1
w (cm)	1
D (s)	1
ν (l/(Pa s))	0.001
P (Pa)	1000

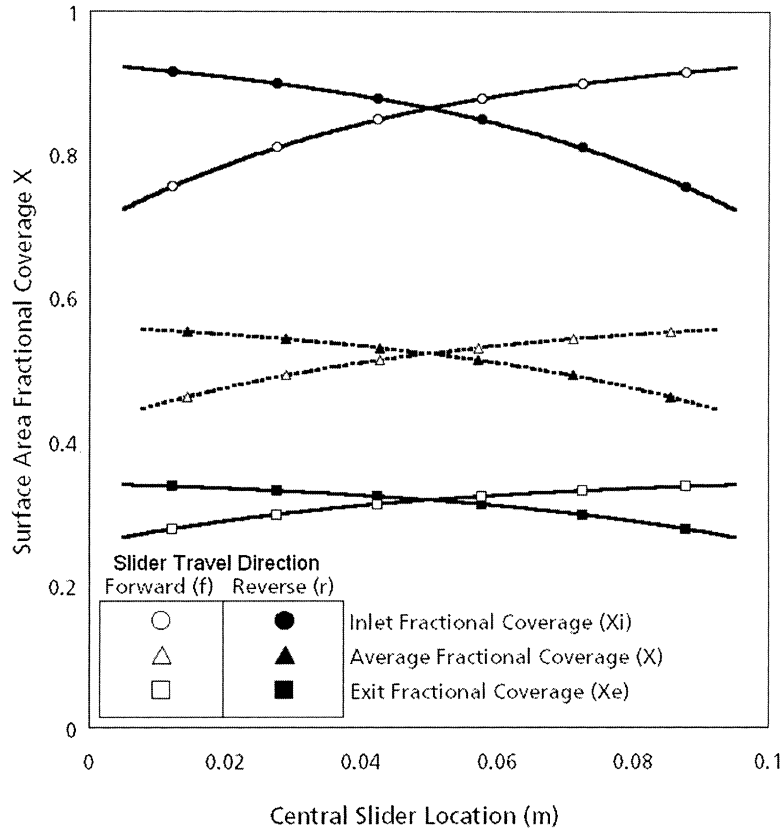


Fig. 4. The standard condition for the variation in surface area fractional coverage (X) with central slider location in the forward are reverse travel directions.

coverage approaches unity, and as the load approaches infinity, the average fractional coverage approaches zero.

Increasing sliding speed decreases the average fractional coverage, as shown in Fig. 6. At very low sliding speeds, the

inlet fractional coverage approaches unity and the removal of coverage approaches some constant value; thus, the average fractional coverage approaches some constant value. At very high sliding speeds, the time of sliding becomes negligible

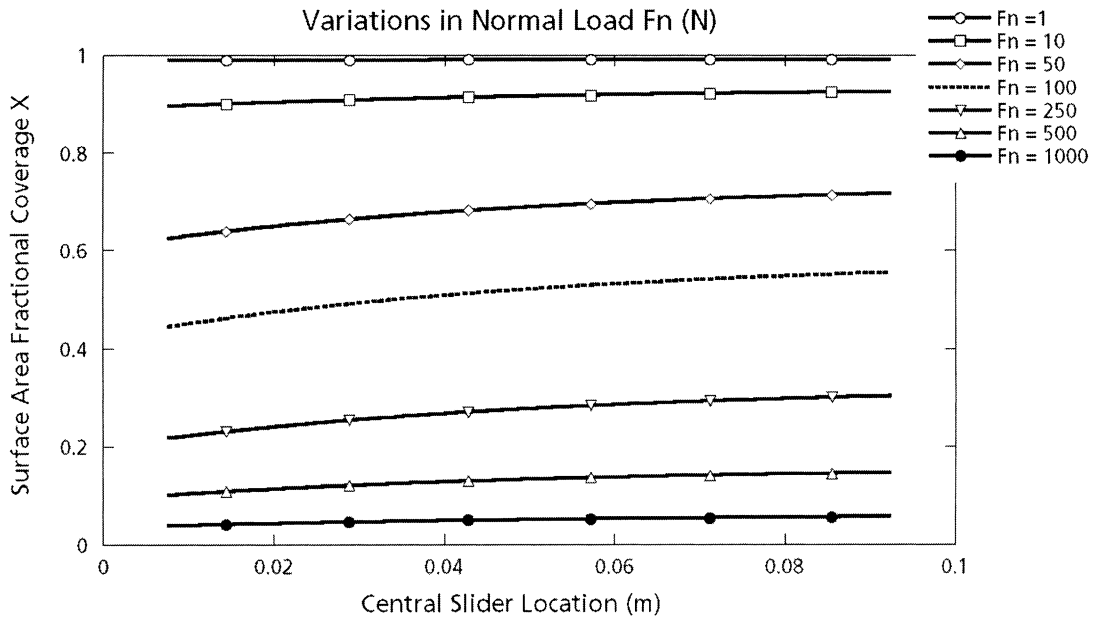


Fig. 5. Variations in average surface area fractional coverage X with position for the forward sliding direction varying normal load from 1 to 1000N.

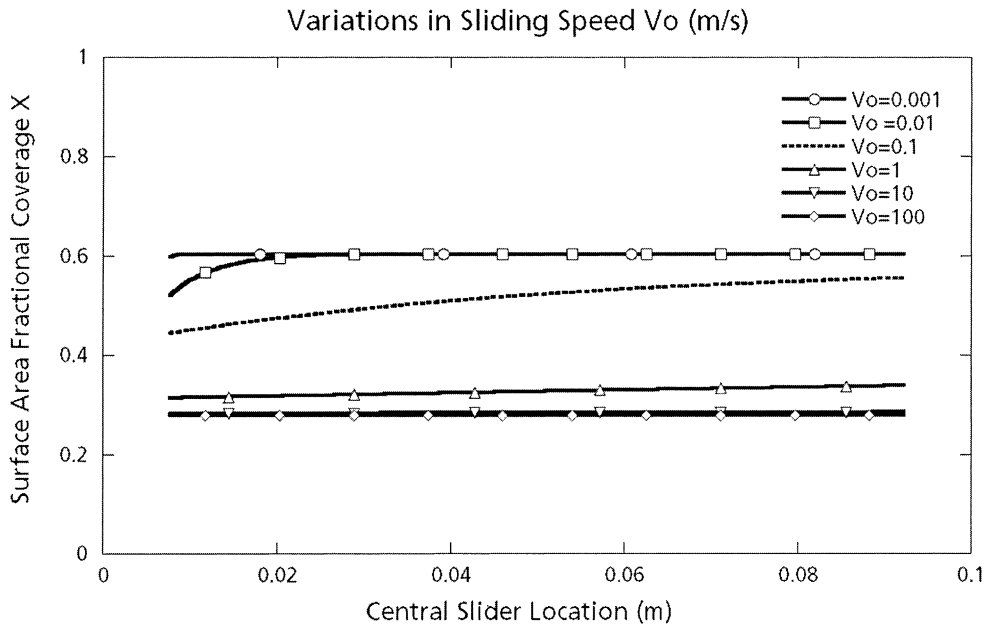


Fig. 6. Variations in average surface area fractional coverage X with position for the forward sliding direction varying sliding speed from 0.001 to 100 m/s.

compared to the dwell time; thus, inlet fractional coverage will approach a constant value determined by the deposition that occurs during the dwell periods.

Increasing the gas pressure increases fractional coverage, as shown in Fig. 7. At very high gas pressures, the inlet fractional coverage approaches unity and the removal of coverage approaches a constant value; thus, the average fractional coverage approaches a constant value. At zero gas pressure,

the average fractional coverage is equal to zero because there is no deposition.

Increasing dwell time increases the average fractional coverage, as shown in Fig. 8. As the dwell time approaches infinity, the inlet fractional coverage approaches unity and the removal of coverage approaches a constant value; thus, the average fractional coverage approaches a constant value.

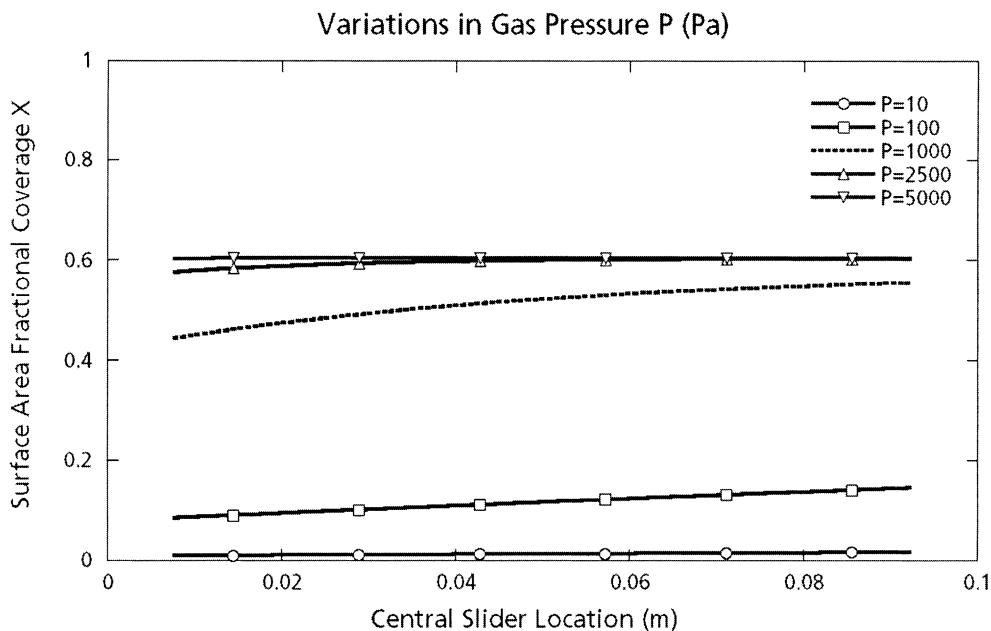


Fig. 7. Variations in average surface area fractional coverage X with position for the forward sliding direction varying the gas pressure from 10 to 5000 Pa.

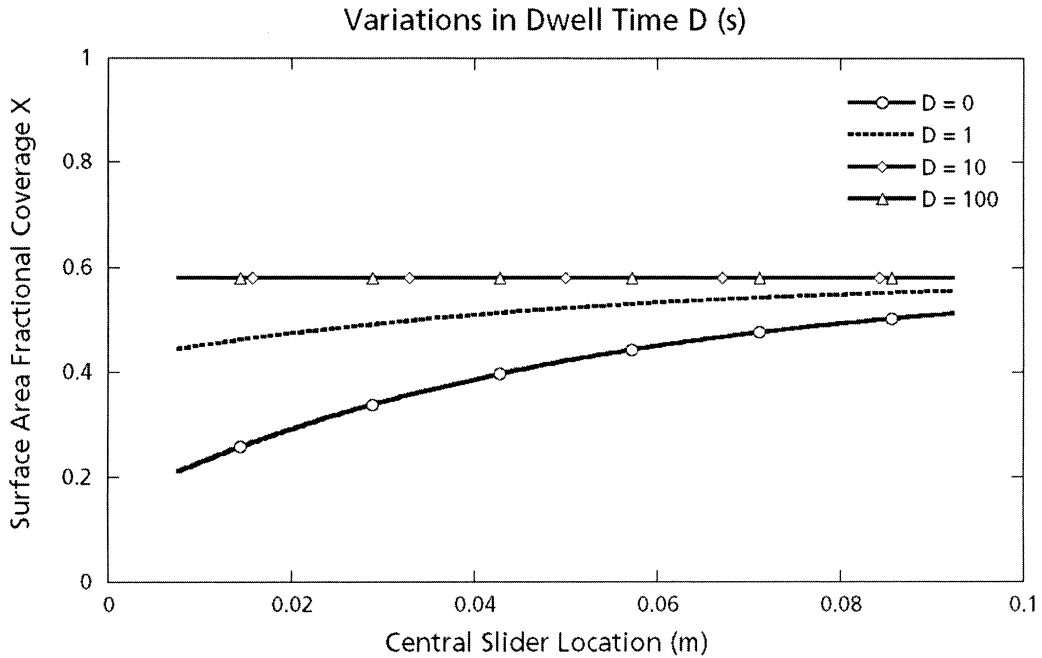


Fig. 8. Variations in average surface area fractional coverage X with position for the forward sliding direction varying dwell time from 0 to 100 s.

4. Discussion

The friction coefficient at any slider location is a function of the average surface fractional coverage (X) under the slider and the friction coefficients of the nascent material (μ_S) and the lubricant material (μ_L). Eqs. (15) and (16) give expressions for the average friction coefficient (μ) and

a normalized friction coefficient (μ^*), respectively.

$$\mu = \mu_L X + \mu_S(1 - X) \tag{15}$$

$$\mu^* = \frac{\mu - \mu_L}{\mu_S - \mu_L} = 1 - X \tag{16}$$

For the previously described “*standard condition*”, predictions of the friction coefficient history of a reciprocating

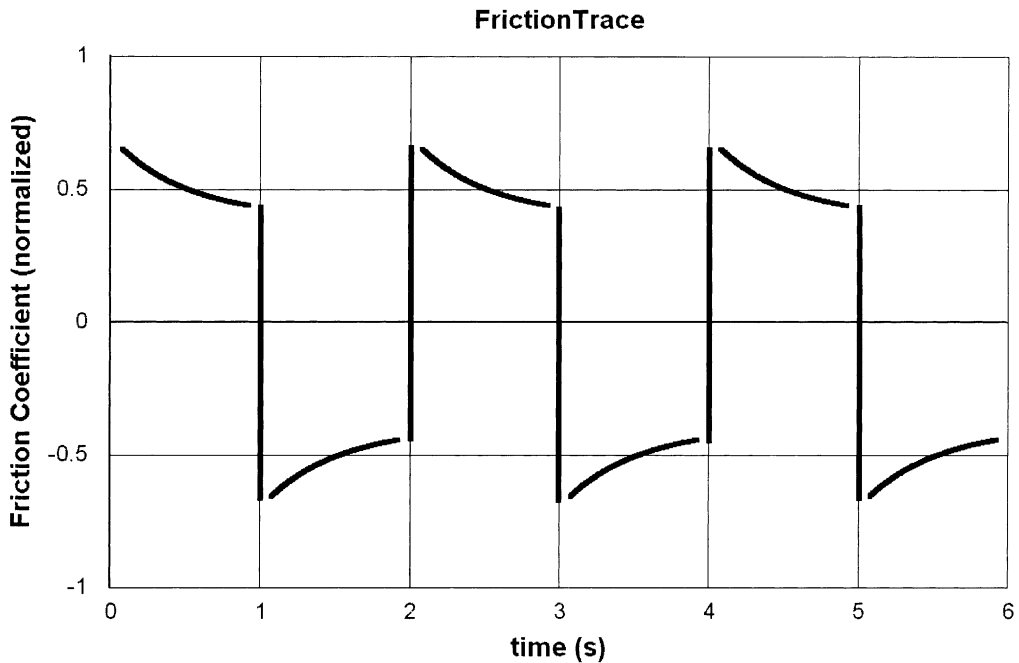


Fig. 9. A predicted normalized friction coefficient trace for the standard condition.

sliding test is shown in Fig. 9. The model offers insight into the occurrences of high friction near the reversal locations and the variation in friction along the wear track during a reciprocated sliding experiment.

Similar to models developed before for vapor phase lubrication, the use of this model requires values of the lubricant film depositions coefficient (ν) and island area wear-rate (K). Such parameters may be found through curve-fitting sets of spatial friction data from reciprocating sliding experiments.

5. Concluding remarks

1. An extension of a vapor-phase lubrication model is developed for reciprocating experiments. This model is based on the assumptions of a fractional film deposition.
2. The model predicts film coverage to be enhanced by decreasing loads, decreasing sliding speeds, increasing gas pressures, and increasing periods of dwell.

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