Optical measurements of real contact area and tangential contact stiffness in rough contact interface between an adhesive soft elastomer and a glass plate

Satoru MAEGAWA*, Fumihiro ITOIGAWA* and Takashi NAKAMURA*
*Nagoya Institute of Technology
Gokiso-cho Showa-ku Nagoya, Aichi 466-8555, Japan
E-mail: maegawa.satoru@nitech.ac.jp

Received 26 April 2015

Abstract
This study demonstrates optical measurements of a real contact area and tangential contact stiffness in a rough contact interface between a soft adhesive elastomer and a glass plate. A friction tester developed in this study employs a rough contact interface between a rough rubber plate that is made of cross-linked poly-dimethyl siloxane (PDMS) and a smooth optical glass (BK7) hemisphere. This friction tester also equips a transmission optical system that is composed of a white light-emitting diodes (LED) light source, an objective lens system, and a charge coupled device (CCD) camera, for the in situ observation of the space distribution of the real contact area in the apparent contact region. Based on a simplified analysis, in accordance to the measurements of the transmitted light intensity, the amount of the real contact area was determined without any complicated calibrations. As a result, it was found that the amount of the real contact area linearly increased with normal load. Furthermore, using the digital image correlation (DIC) method, the time changes in the surface displacement field within the apparent contact region were visualized from the onset of the sliding motion to the onset of the steady sliding. These results provided a direct estimation of the tangential contact stiffness that resulted from the tangential deformation at the contacting asperity layer.

Key words: Real contact area, Tangential contact stiffness, Soft elastomer, Digital image correlation, Shear strength, Transmission optical system

1. Introduction

Studies on the friction between a soft elastomer and a smooth rigid surface have been focused on important topics in practical applications of tribology, e.g., in rubber pad processes, vibration control systems, seals, and power transmission systems, including shoes and the grip of robot hands. In these processes and products, static and kinetic friction forces, and the value of the contact stiffness, are important factors for determining the performance of the entire system.

To understand the mechanisms of the contact and sliding processes of rubber friction, many research studies have been performed over the past several decades. In particular, the in situ observation of the contact interface has been focused on one of the prior published powerful methods (Schallamach, 1971; Dieterich and Kilgore, 1994; Rubinstein, et al., 2007; Maegawa and Nakano, 2007, 2010; Chareauminois and Fretigny, 2008; Maegawa, et al., 2010; Nguyen, et al., 2011; Krick, et al., 2012; Prevost, et al., 2013). These results provided beneficial information for improvement of the theories of the contact and sliding mechanisms for elastic materials. For example, Schallamach (1971) first discovered a surface instability, such as the propagation of the detachment waves across the contact zone during sliding motion. In recent years, as presented in the references by Chareauminois and Fretigny (2008) and Prevost et al. (2013), the digital image correlation (DIC) method has been focused on visualizing the surface displacement and stress fields within the contact interface. These research studies will be helpful for the understanding...
of the relationship between the local friction law or the surface deformations on the contact interface, and the macroscopic tribological properties of the system.

The aim of this study is to present a novel observation method for rough contact interface, in which the real contact and non-contact region can be visualized simultaneously. Using a developed friction tester with a transmission optical system, which has a point contact between a rough rubber plate—made of cross-linked poly-dimethyl siloxane (PDMS)—and a smooth optical glass (BK7) hemisphere, the time changes in the friction force and the images of the contact interface were recorded. Based on the contact interface measurements, a simple technique to quantify the total amount of real contact area was developed. In addition, direct measurements of the surface displacement at real contact and non-contact regions provided an experimental evidence for the effectiveness of the method to estimate the tangential compliance in a rough contact interface based on the tangential deformation at a contacting asperity layer.

2. Experiment

2.1 Experimental setup

Figure 1 shows a schematic of the developed friction tester. This friction tester employed the point contact between an optically transparent rubber plate made of cross-linked PDMS and a smooth glass hemisphere lens. The thickness of the PDMS plate and the radius of the glass hemisphere were 3 mm and 5 mm, respectively. Additionally, this tester also equips a transmission optical system for the in situ observation of the spatial distribution of the real contact area within the apparent contact region. The glass hemisphere was fixed directly below an objective lens (LEICS Z16 APO ×6.3) system. The PDMS plate was connected to a double-cantilever spring that was mounted on the X-directional motorized stage via a lever arm. An LED white light source was placed under the contact region, and a charge coupled device (CCD) camera with a frame size of 1920 pixels × 1440 pixels was attached to the objective lens system. To measure the deflection of the double-cantilever spring, an eddy current gap sensor was installed on its side.

The right side of Fig. 1 schematically illustrates the transmission optical system. One of the advantages of the use of the transmission optical system for in situ measurements of the contact interface is that it can visualize contact and noncontact regions. The contact distribution was captured as the intensity distribution of the transmitted light (see Fig. 1). The captured light intensity at the contact regions is relatively larger than that in the noncontact region. In contact regions, the light source passes through with a small refraction because the refractive indexes of the PDMS plate and the glass lens are similar. In contrast, in the noncontact regions, the light is reflected and scattered owing to the surface irregularity and the large refractive index difference between the contacting surface materials (PDMS and glass) and air. In this study, based on the advantage that the transmission optical system can visualize the contact and non-contact region simultaneously, we tried to measure the surface displacement field of both the real contact and noncontact regions.

2.2 Specimens

A rubber plate made of cross-linked PDMS (Dow Corning’s SYLPOT 184) was prepared and used as the specimen. The mixture was composed of a PDMS melt and a cross-linker agent with a compounding ratio of 10:1 that were poured into a rectangular steel mold. The bottom surface of the mold was roughened by a sandblast process with an arithmetic average roughness $R_s = 0.77 \mu m$, and rms roughness is $0.92 \mu m$. The roughness of the mold surface was transferred onto the contacting surface of the PDMS plate. The thickness of the plate was 3 mm and its surface dimensions were 25 mm × 25 mm.

To obtain the Young’s modulus of the PDMS plate, a JKR test (Johnson, et al., 1971; Johnson, 2003) was performed between its smooth rear face and a smooth glass hemisphere. The details of the method used to obtain the Young’s modulus are outlined in Maegawa and Nakano (2007). The Young’s modulus of the PDMS plate used in this study was 1.1 MPa. The Poisson’s ratio of such PDMS elastomers is close to 0.5 (Mark, 1999).

2.3 Procedure
In stationary tests, in which the motorized stage was not moved, normal loads $F_Z$ with values equal to 0.03, 0.06, 0.09, 0.12, and 0.15 N, were applied by a lever mechanism using five types of weights. After waiting for 300 s, the intensity distribution of the transmitted light was imaged. A specific waiting time was necessary to eliminate the effects of the changes in the real contact area with contact time (Dieterich and Kilgore, 1994).

In dynamic tests, similar to the case of the stationary tests, five types of normal loads were applied. After a period of 300 s, the PDMS plate was driven by the motorized stage that moved along the $X$-direction with a constant speed of 0.1 mm/s, while the temporal variation of the tangential load $F_X$ and the images of the contact interface were simultaneously recorded. The tangential load $F_X$ was measured at a sampling rate of 10 kHz by monitoring the deflection of the double cantilever using the eddy current gap sensor. The intensity profile of the contact interface that continuously changed with time was imaged at a flame rate of 26 frames per second (fps). All experiments were performed in an air-conditioned room where the temperature and relative humidity were approximately 25 °C and 30 %, respectively.

3. Results and discussion

3.1 Typical results

Figure 2 shows typical results for snapshots of the contact interface and the time changes of the tangential load $F_X$ under three different normal loads, i.e., for $F_Z = 0.03, 0.09, 0.15$ N. The white and black regions indicate real contact and noncontact regions, respectively.

Focusing on the temporal changes in $F_X$, it is found that under all conditions, $F_X$ increases gradually with time after the beginning of the lateral motion of the $X$-directional motorized stage, i.e., at $t = 0$ s. Subsequently, $F_X$ reached its maximum value at the maximum static friction force $F_{\text{max}}$. In $F_Z = 0.03$, and 0.09 N, after a relatively small drop, $F_{\text{max}}$ reached its steady value at the kinematic friction force $F_k$. In contrast, in $F_Z = 0.15$ N, $F_X$ reached the steady value $F_k$ after a damped vibration. It has been known that stick-slip vibration tends to occur under highly normal load conditions (Nakano, 2006). Thus, the normal load dependence of the vibration is consistent with the occurrence condition of stick-slip motion. In addition, as a general rule, it was found that the values of $F_{\text{max}}$ and $F_k$ increased with increasing normal loads $F_Z$.

In accordance to the well-known classical theory of sliding friction (Bowden and Tabor, 1950), the increases in the friction forces for $F_Z$ are attributed to the increase of the real contact area. Based on the contact images, it is clear that the apparent contact regions increase with the normal load $F_Z$, leading to increases in the friction forces $F_{\text{max}}$ and $F_k$. In addition, it is found that the real contact regions stay unchanged before and after the onset of the global slip. It has been shown that the propagation of Schallamach waves (Schallamach, 1971) or a large macroscopic deformation of an apparent contact region (Barquins, 1985) occurs during the sliding motion in the case of the sliding friction, based on which a smooth elastomer slides on a rigid smooth surface. However, in this study, these surface deformations were not observed since the surface of the elastomeric specimen used was rough.

3.2 Measurement of real contact area

The central part of the apparent contact regions at $t = 2.0$ s in Fig. 2(a) is magnified in Fig. 3. It is found that the magnified image has an adequate spatial resolution to clearly distinguish the real contact and noncontact regions dispersed within the apparent contact region. The lower graph shows the profile of the transmitted intensity along the white dashed line drawn horizontally in the upper image. As shown in the lower graph, the contact and noncontact regions are distinguished by the boundary value of the transmitted light intensity $I_B = 104$, which is denoted by the dashed horizontal line illustrated in the lower graph. Figure 4 shows the binarized image of Fig. 3. All pixels in the image were binarized using the threshold $I_B$. When $I \geq I_B$, the position of the pixel indicates the contact region (i.e., $I = 255$), and it is depicted in white color. In contrast, when $I < I_B$, the position of the pixel indicates the noncontact region (i.e., $I = 0$), and it is depicted in black color.

The value of $I_B$ should be carefully determined because the estimated real contact area strongly depends on $I_B$. For example, Persson et al. (2008) pointed out that the real contact area depends on the lateral resolution (or magnification) of the instrument’s optical system. Thus, the measured contact area decreases with increasing...
In general, this area is determined based on an appropriate model that considers some effects of the optical configuration, and on the evanescent light effects. In this study, however, we approximately estimated the value of $I_b$ as described below. The estimated value of $I_b$ will be validated in Fig. 6.

To find an appropriate value of $I_b$, histograms of the intensity distribution within the apparent contact region were calculated, as shown in Fig. 5. The ordinate and abscissa represent the normalized number of pixels and the transmitted intensity, respectively. Two different peaks are clearly seen in the histogram curves, i.e., peak 1 and peak 2. Note that the histogram curves are normalized by the height of peak 1. Considering the attributes of the transmission optical system, the intensity values of these peaks, i.e., $I_{p1}$ and $I_{p2}$, indicate the feature quantities of the transmitted intensities. In turn, these indicate the real contact and noncontact regions. The normalized height of peak 2 increased with the normal load, but the position of $I_{p2}$ remained constant. This agrees with the experimental results, where the total area (or total number) of the real contact regions increased with the normal load. In this study, we estimated the value of $I_b$ approximately, using the relationship $I_b = (I_{p1} + I_{p2}) / 2 = 104$. Thus, the developed visualizing method for real contact and non-contact regions allows to obtain both information at the contact and non-contact reasons, leading to provide a simple and easy method to determine $I_b$.

Fig. 6 shows the relationship between the total amount of the real contact area $A_{real}$ and the normal load $F_Z$ [Fig. 6(a)], and $A_{real}$ and $F_X$ [Fig. 6(b)]. Note that $A_{real}$ was calculated by the sum of the pixels for real contact regions of the binarized images in accordance to $A_{real} = A_{pix} \Sigma / 255$, where $A_{pix}$ is the area of the single pixel, i.e., $1.286 \times 10^{-6}$ mm$^2$. In Fig. 6(a), a linear dependence between $A_{real}$ and $F_Z$ can be observed. As described above, the value of $F_X$ was estimated by the tangential load $F_X$ under the steady sliding. The linear dependence under a low $F_Z$ agrees with the contact theories derived by Archard (1957), Greenwood and Williamson (1966), and Persson (2001, 2008). Similarly, the relationship between $A_{real}$ and $F_X$ is also linear. It is also well known that the above relationships are the origin of Amonton’s friction law, whereby the friction force linearly increases with an increasing normal load. However, it should be noted that in the higher $F_Z$ regime, the relationship slightly deviates from linearity [see Fig. 6(a)]. This results from the transition from the asperity summit contact to the Hertzian (overall) contact. In accordance to the theoretical derivation of Persson (2001, 2008), under the higher loading condition the overall Hertzian contact is formed, and the real contact area does not linearly increase with the normal load (Maegawa, et al., 2015).

The friction force $F_X$ in an elastomer/rigid interface is in general considered to arise from the two components, the adhesion term $F_{adhesion}$ and the hysteresis term $F_{hysteresis}$ (Tabor, 1960).

$$F_X = F_{adhesion} + F_{hysteresis} \quad (1)$$

The former term relates to the intermolecular interaction between the two contacting surfaces (Roberts and Thomas, 1975), and the latter term relates to the energy lost through the bulk deformation process (Roberts, 1992). Fuller (1975) and Persson (2001) pointed out that in a tyre/road contact, the hysteresis friction $F_{hysteresis}$ has a major contribution to the total friction. In contrast, as explained by Roberts (1975) and Persson (2006), when an elastomer slides on a very smooth surface, the adhesion force $F_{adhesion}$ exerts the main influence in determining rubber friction. Thus, in this study, the friction forces $F_X$ can be described as listed below:

$$F_X = F_{adhesion} = \tau A_{real} \quad (2)$$

where $\tau$ is the shear strength that acts on the interface between the PDMS and BK7 surfaces.

The value of the shear strength $\tau$ can be estimated as the value of the slope in Fig. 6(b). Thus, $\tau$ is determined to be 0.151 MPa. The estimated value is in a good agreement with the results by Wu-Bavouzet et al. (2007) and Lorenz et al. (2013). The agreement provides the validity of the measurements of the real contact area in this study. However, it should be noted that the measured real contact area strongly depends on the magnification of the visualized system (Persson, 2008). Therefore, for a more detailed discussion, other theoretical and experimental approaches are needed. In addition, determining the physical meaning of $I_b$ estimated above method is difficult. In fact, when $I_b$ is set to be 100, $\tau$ is determined to be 0.144 MPa. On the other hand, when $I_b = 108$, $\tau$ is 0.159 MPa. These values are also consistent with the experimental values obtained in previous researches above. Thus, it is difficult to make a quantitative discussion for the accuracy of this method. However, the aim of this study is not to make a quantitative discussion of the proposed method, rather, to highlight the usability and simplicity of the developed method. In fact, the
current method can estimate the total area of real contact without any complicated calibrations, and it has adequate accuracy.

3.3 Surface displacement field within the apparent contact region

Using the DIC method, we performed in situ measurements of the surface displacement field in the apparent contact interface (Chareauminois and Fretigny, 2008; Prevost, et al., 2013). The typical results from the surface displacement measurement in the X- and Y-directions, $u_X$ and $u_Y$, are shown in Fig. 7. Here, the surface displacement was determined as the averaged value which includes the tangential deformation at the non-contact regions and non or small displacement at the real contact regions. In the case in Fig. 7, the averaged area is $13 \times 13 \mu m^2$. This figure shows the displacement from $t = 0$ to $t = 0.5$ s at $F_X = 0.03$ N. The color bars represent $u_X$ and $u_Y$. The white dashed circles in the figure denote the apparent contact region in a stationary condition. Thus, it indicates the contact edge position of the apparent contact area before the sliding motion. The white curve represents the $u_X$ profile on the horizontal line at $X = 0$ mm, i.e., $u_{YX} = 0$. Even in the apparent contact area, a small displacement along the X-direction was observed. The two arrows indicate the radius of the stick region, i.e., the value of $r_c$. The value of $u_X$ was nonzero outside of the circle with radius $r_c$. This indicates that microslip occurs within the apparent contact region. From the classical theory derived by Mindlin (1949), it is known that an annular microslip region surrounding a central stick region gradually decreases as the shear load is increased. The development process of microslip agrees well with previous studies for visualizing the Mindlin slip (Prevost et al., 2008). The DIC method used in this study has an adequate resolution for discussing the partial slip within the apparent contact regions.

In Fig. 7, inside the circle with radius $r_c$, no displacement is observed. However, further to a closer observation, we can find a small displacement in the central stick region. Fig. 8 shows time changes in $u_X$ at the central position of the apparent contact region. It is found that $u_X$ gradually increases with time even though the global slip has not occurred yet. Here the global slip means the macroscopic entire slip, in which the slip occurs at the entire apparent contact region. Thus, a small displacement is observed inside the stick region during the static friction regime. Similar results were observed in the experiments conducted by Scheibert et al. (2008). They proposed a possible mechanism for this as is described next. The asperity layer of the PDMS plate can deform along the driving direction, as illustrated in Fig. 9. Thus, the measured displacement arises from the partial slip (local slip at each real contact regions) rather than the lateral deformation of the contacting asperities. In this study, this contact model was clearly confirmed in Fig. 10 that shows the surface displacement measurements at $F_X = 0.03$ N under a higher spatial resolution. The two images show the intensity profile at $t = 0$, and for a $u_X$ field generated from $t = 0$ to $t = 0.5$, at the same position. All the displayed regions in the figure were located inside the stick region. Upon comparison of both images, it is found that the typical noncontact regions, which are surrounded by dashed circles, had larger $u_X$ values than contact regions. This result corresponds to the illustration shown in Fig. 9. Thus, the developed optical method first observe the existence of the tangential deformation of the asperity layer in rough contact interfaces. The results suggest that the DIC measurements based on the transmission optical system used in this study have the potential for analyzing the in-surface plane displacement and the tangential deformation of the asperity layer in rough contact interfaces.

3.4 Measurement of tangential contact stiffness

As described above, the lateral deformation of the contacting asperities is an important factor to determine the tangential contact stiffness. Barthoud and Baumberger (1998) and Medina et al. (2013) described an approximate expression for the total tangential contact stiffness as indicated below:

$$\frac{1}{K_{total}} = \frac{1}{K_{asperity}} + \frac{1}{K_{bulk}}$$

(3)

where $K_{total}^T$, $K_{asperity}^T$, and $K_{bulk}^T$, are the total tangential contact stiffness, tangential stiffness at the asperity layer, and tangential stiffness at the bulk region, respectively. Both the values of $K_{bulk}^T$ and $K_{asperity}^T$ are important for determining the total tangential contact stiffness of the contact surfaces. An important manifestation of this statement refers to the studies of Nakano and Maegawa (2009), Fuadi et al. (2010), and Maegawa et al. (2013) that pointed out that the ratio of
$K_{\text{bulk}}^T$ to $K_{\text{asperity}}^T$, i.e., $K_{\text{bulk}}^T/K_{\text{asperity}}^T$, is an important factor for determining the stability of sliding systems and the value of static friction.

The value of $K_{\text{bulk}}^T$ in this study can be estimated by Mindlin’s theory (Mindlin, 1949), in accordance to $K_{\text{bulk}}^T = 4aE/(1+v)(2-v)$, where $a$ is the contact radius of the apparent contact region, $E$ is the Young’s modulus of PDMS, and $v$ is the Poisson’s ratio of PDMS. Thus, $K_{\text{bulk}}^T = 8.3 \times 10^5$ N/m in $F_Z = 0.03$ N. Furthermore, as described above, this study roughly estimates the value of $K_{\text{asperity}}^T$ based on the results shown in Figs. 8 and 9. From the theoretical and experimental approaches performed by Bureau et al. (2003) and Prevost et al. (2013), the small displacement due to the lateral deformation of the contacting asperities was derived in accordance to the following equation:

$$\frac{\sigma_N}{\sigma_Z} = \mu \left(1 - e^{-\frac{X}{L(\mu \sigma_Z)}}\right)$$

where $\mu$ is the friction coefficient, and $L$ is an elastic length that is determined by the rms roughness of the interface $h$, i.e., $L = h(2-v) / 2(1-v)$. Figure 11 shows the relationship between $\sigma_N/\mu \sigma_Z$ and $\mu Z$, where $\sigma_N$ and $\sigma_Z$ were calculated based on the Hertzian contact theory and Mindlin’s theory (Johnson, 2003). Here, Eq. (4) can be changed to $\sigma_N/\mu \sigma_Z = 1-\exp(-\mu Z/\mu L)$. The solid line in Fig. 11 is the fitting curve based on this changed Eq. (4). In the fitting process, the least-squares method was used to determine the value of $L$; the value of $L$ was changed only the fitting parameter. Consequently, $L$ was determined to be 1.02 µm. Considering the roughness of this system, it is found that the experimentally determined value of $L$ is validated with the theoretically determined values. Thus, in this study, the value of $L$ can be directly measured based on the developed transmission optical system. It should be noted that the maximum values of $\sigma_N/\mu \sigma_Z$ under $F_Z = 0.09$ N and $F_Z = 0.15$ N is larger than unity, although Eq. (4) asymptotically increases to unity. The difference results from that Eq. (4) does not consider the difference between static and kinetic friction coefficients. As shown in Fig. 2, the observed results in this study, the static friction coefficient is larger than the kinetic friction coefficient. Additionally, the value of $K_{\text{asperity}}^T$ was derived by Barthoud and Baumberger (1998) as follows:

$$K_{\text{asperity}}^T = \frac{F_Z}{L}$$

Therefore, the tangential contact stiffness $K_{\text{asperity}}^T$ for a normal load of 0.03 N can be estimated to be $2.9 \times 10^5$ N/m.

As described above, the value of $K_{\text{asperity}}^T$ is much larger than the measured $K_{\text{bulk}}^T$. Thus, in this system, the effect of $K_{\text{asperity}}^T$ is neglected in Eq. (3). The value of $K_{\text{bulk}}^T$ has a major contribution to the determination of the total tangential contact stiffness. It means that the method based on the DIC analysis can quantify the value of $K_{\text{asperity}}^T$, even if $K_{\text{total}}^T$ is mainly characterized by $K_{\text{bulk}}^T$. As used in the study by Barthoud and Baumberger (1998), in usual method focusing on the mechanical relationship between an external tangential load and a deformation of an elastic slider, $K_{\text{total}}^T$ is only obtained. Thus, in these methods, when $K_{\text{asperity}}^T$ is much larger than the measured $K_{\text{bulk}}^T$ it is difficult to quantify the value of $K_{\text{asperity}}^T$.

In addition, it is found that, as shown above, $K_{\text{asperity}}^T$ depends only on normal load and surface roughness, and does not depend on the Young’s modulus. Thus, for materials with higher Young’s moduli, the value of $K_{\text{bulk}}^T$ will approach the value of $K_{\text{asperity}}^T$. In this case, the total tangential contact stiffness at the asperity layer will be an important factor for the determination of the performance of sliding systems.

4. Conclusion

This study presents an optical method that can visualize both real contact and non-contact regions. From a model experiment, in which a rough contact interface between a cross-linked PDMS plate and a BK7 hemisphere lens is used, it was found that the method provides a simple and easy measuring method for the real contact area and the tangential contact stiffness at a rough contact layer without any complicated calibrations. In addition, it was found that the DIC analysis focusing on the displacement field within a rough contact interface is effective method to quantify the tangential contact deformation at a contacting asperity layer of elastic materials.
Fig. 1 Schematic of the experimental setup.
Fig. 2 Typical results for captured images of the contact interface and changes in the tangential load $F_X$: (a) $F_Z = 0.03$ N, (b) $F_Z = 0.09$ N, and (c) $F_Z = 0.15$ N.
Fig. 3 (Upper image) Magnified image at the central part of the apparent contact region under $F_Z = 0.03$ N at $t = 2.0$ s. The lengths of each side of the image are 0.1 mm. (Lower graph) Intensity profile along the white dashed line in the upper image.

Fig. 4 Binarized image of Fig. 3. The white region is the real contact region, and the black region is the noncontact region.
Fig. 5 Histogram of the intensity distribution within the apparent contact area.

Fig. 6 Relationship between the (a) real contact area $A_{\text{real}}$ and the normal load $F_Z$ under stationary tests, and (b) kinetic friction force $F_k$ and real contact area $A_{\text{real}}$.

Fig. 7 Surface displacement fields $u_X$ and $u_Y$ at $F_X = 0.03$ N. White circles depict the apparent contact regions under stationary conditions. The white curve represents the $u_X$ profile on the horizontal line at $X = 0$ mm.
Fig. 8 Changes in the surface displacement $u_X$ with surface stress $\sigma_X$ (open circles: $F_Z = 0.03$ N, dotted circles: $F_Z = 0.09$ N, solid circles: $F_Z = 0.15$ N, vertical dashed lines: the onset of the global slip that is determined based on the measurements of time changes in $F_X$ shown in Fig. 2).

Fig. 9 Schematic illustration of the tangential deformation of the roughness layer.

Fig. 10 Magnified intensity map (left) and displacement map (right).
Fig. 11 Relationship between shear stiffness and normal load (solid line: fitting curve based on Eq. (4), open circles: $F_Z = 0.03$ N, dotted circles: $F_Z = 0.09$ N, solid circles: $F_Z = 0.15$ N).
References

Archard, J. F., Elastic deformation and the laws of friction, Proceedings of the Royal Society A, Vol.243, No.1233, (1957), pp.190–205.

Barquins, M., Sliding friction of rubber and Schallamach waves – A review, Material Science and Engineering, Vol.73, (1985), pp.45–63.

Barthoud, P. and Baumberger, T., Shear stiffness of a solid–solid multicontact interface, Proceedings of the Royal Society of London, A, Vol.454, No.1974, (1998), pp.1615–1634.

Bureau, L., Caroli, C. and Baumberger, T., Elasticity and onset of frictional dissipation at a non–sliding multi–contact interface, Proceedings of the Royal Society of London, A, Vol.459, No.2039, (2003), pp.2787–2805.

Bowden, F. P. and Tabor, D., The Friction and Lubrication of Solids, Clarendon Press, Oxford, 1950.

Chareauminois, A. and Pretigny, C., Local friction at a sliding interface between an elastomer and a rigid spherical probe. The European Physical Journal, Vol.27, No.2, (2008), pp.221–227.

Dieterich, J. H. and Kilgore, B.D., Direct observation of frictional contacts: New insights for state-dependent properties. Pure and Applied Geophysics, Vol.143, No.1–3, (1994), pp.283–302.

Eguchi, M. Shibamiya T, and Yamamoto, T., Measurement of real contact area and analysis of stick/slip region. Tribology International, Vol.42, No.11–12, (2009), pp.1781–1791.

Fuller, K. N. G. and Tabor, D., The effect of surface roughness on the adhesion of elastic solids, Proceedings of the Royal Society of London, A, Vol.345, No.1642, (1975), pp.327–342.

Fuadi, Z. Maegawa, S., Nakano, K. and Adachi, K., Map of low-frequency stick-slip of a creep groan, Proceedings of the Institute of Mechanical Engineers, Part J: Journal of Engineering Tribology, Vol.224, No.12, (2010), pp.1235–1246.

Greenwood, J. A. and Williamson, J. B. P., Contact of nominally flat surfaces. Proceedings of the Royal Society of London, A, Vol.295, No.1442, (1966), pp.300–319.

Johnson, K. L., Kendall, K. and Roberts, A. D., Surface energy and the contact of elastic solids, Proceedings of the Royal Society of London A, Vol.324, No.1558, (1971), pp.301–313.

Johnson, K. L., Contact Mechanics, Cambridge University Press, Cambridge, 2003.

Krick, B. A., Vail, J. R., Persson, B. N. J. and Sawyer, W. G., Optical in situ micro tribometer for analysis of real contact area for contact mechanics, adhesion, and sliding experiments, Tribology Letters, Vol.45, No.1, (2012), pp.185–194.

Lorenz, B., Krick, B. A., Rodriguez, N., Sawyer, W. G., Maniagalli, P. and Persson, B. N. J., Static or breakloose friction for lubricated contacts: the role of surface roughness and dewetting, Journal of Physics: Condensed Matter, Vol.25, No.44, (2013), 445013.

Maegawa, S. and Nakano, K., Dynamic behaviors of contact surfaces in the sliding friction of a soft material, Journal of Advanced Mechanical Design, Systems, and Manufacturing, Vol.1, No.4, (2007), pp.553–561.

Maegawa, S. and Nakano, K., Mechanism of stick-slip associated with Schallamach waves, Wear, Vol.268, No.7–8, (2010), pp.924–930.

Maegawa, S., Suzuki, A. and Nakano, K., Precursors of global slip in a longitudinal line contact under non-uniform normal loading, Tribology Letters, Vol.38, No.3, (2010), pp.313–323.

Maegawa, S., Itoigawa, F., Shinoyoshi, T., Suzuki, A., Tadokoro, C. and Nakano, K., Design criteria on effective static friction coefficient of elastomers, Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.79, No.803, (2013), pp.2622–2634 (in Japanese).

Maegawa, S., Itoigawa, F., and Nakamura, T., Effect of normal load on friction coefficient for sliding contact between rough rubber surface and rigid smooth plate, Tribology International, Vol.92, No.12, (2015), pp.335–343.

Medina, S., Nowell, D. and Dini, D., Analytical and numerical models for tangential stiffness of rough elastic contacts. Tribology Letters, Vol.49, No.1, (2013), pp.103–115.

Mark, J. E. (Editor), Polymer Data Handbook, Oxford University Press, Oxford, 1999.

Mindlin, R. D., Compliance of elastic bodies in contact, ASME Journal of Applied Mechanics, Vol.16, No.3, (1949), pp.259–268.

Nakano, K., Two dimensionless parameters controlling the occurrence of stick-slip motion in a 1-DOF system with Coulomb friction, Tribology Letters, Vol.24, No.2, (2006), pp.91–98.

Nakano, K., Two dimensionless parameters controlling the occurrence of stick-slip motion in a 1-DOF system with Coulomb friction, Tribology Letters, Vol.24, No.2, (2006), pp.91–98.
Nakano, K. and Maegawa, S., Stick-slip in sliding systems with tangential contact compliance, Tribology International, Vol.42, No.11–12, (2009), pp.1771–1780.

Nguyen, D. T., Paplino, P., Aurdy, A. C., Chateauauminois, A., Fretigny, C., Chenade, Y., Portigliatti, M. and Bartel, E., Surface pressure and shear stress fields within a frictional contact on rubber, The Journal of Adhesion, Vol.87, No.3, (2011), pp.235–250.

Persson, P. N. J., Theory of rubber friction and contact mechanics, The Journal of Chemical Physics, Vol.115, No.8, (2001), pp.3840–3861.

Persson, B. N. J. and Volokitin, A. I., Rubber friction on smooth surfaces. The European Physical Journal E, Vol.21, No.1, (2006), pp.69–80.

Persson, P. N. J., Sivebaek, I. M., Samoilov, V. N., Zhao, K., Volokitin, A. I. and Zhang, Z., On the origin of Amonton’s friction law, Journal of Physics: Condensed Matter, Vol.20, No.39, (2008), p.395006.

Prevost, A., Scheibert, J. and Debregeas, G., Probing the micromechanics of a multi-contact interface at the onset of frictional sliding, The European Physical Journal E, Vol.36, No.2, (2013), pp.17–29.

Roberts, A. D. and Thomas, A. G., The adhesion and friction of smooth surfaces, Wear, Vol.33, No.1, (1975), pp.45–64.

Roberts, A. D., A guide to estimating the friction of rubber, Rubber Chemistry and Technology, Vol.65, No.3, (1992), pp.673–686.

Rubinstein, S. M., Cohen, G. and Fineberg, J., Dynamics of precursors to frictional sliding, Physical Review Letters, Vol.98, No.22, (2007), p.226103.

Schallamach, A., How does rubber slide?, Wear, Vol.17, No.4, (1971), pp.301–312.

Scheibert, J., Debregeas, G. and Prevost, A., Micro-slip field at a rough contact driven towards macroscopic sliding, (2008), e-print, arXiv:0809.3188.

Tabor, D., Hysteresis losses in the friction of lubricated rubber, Rubber Chemistry and Technology, Vol.33, No.1, (1960), pp.142–150.

Wu-Bavouzet, F., Clain-Burckbucher, J., Buguin, A., De Gennes, P. G. and Brochard-Wyart, F., Stick-slip: wet versus dry, The Journal of Adhesion, Vol.83, No.8, (2007), pp.761–784.