Friction Drive Dynamics of Surface Acoustic Wave Motor

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Abstract—Friction drive dynamics of a surface acoustic wave (SAW) motor is discussed through the direct measurement of the frictional interface and subsequent contact mechanics analysis. A slider of the SAW motor has projections at frictional surface for stable operation of the motor. The projections are the only contact points with the wave so that all the driving force is produced hereout. Since a stator of the SAW motor is transparent, we directly measured displacements of one projection through the stator by means of two laser Doppler vibrometers. The measurement was subsequently analyzed by means of contact mechanics formulae in quasi-static and dynamic way. The analysis indicated that the dynamics of the projection highly depends on the friction coefficient.

I. INTRODUCTION

A great deal of investigation of friction drive dynamics in ultrasonic motors (USMs) has been carried out to develop design/control method. To date, analytical modelling has led the way in the investigations due to the difficulty of measurement. However, evaluation of the models was not accurately accomplished only with comparison between calculated and measured mechanical output, such as torque–speed curve, since the mechanical output is a result of complex coupled dynamic system. Some kinds of measurements hence increase in importance [1], [2]. An objective surface acoustic wave (SAW) motor utilized lithium niobate for the stator, which is a transparent material, so that direct measurement of the frictional interface is somehow carried out. In this report, we discuss the friction drive dynamics of the SAW motor through measurement of the frictional interface by means of two laser Doppler vibrometers (LDVs).

II. FRAMEWORK FOR EXPERIMENTS

The SAW motor consists of a lithium niobate stator and a silicon slider. The stator generates the propagating Rayleigh wave at an IDT by the piezoelectric effect. The slider has many cylindrical projections on its contact surface to eliminate the squeezed air film disturbance in the contact with the Rayleigh wave. The surface geometry with projection is not only necessary for the stable operation of the SAW motor, but controls the contact pressure and the elastic deformation condition. A preload is applied to the slider in order to increase the driving force.

To reveal the friction/contact dynamics during the operation, we measured normal and tangential displacements of one projection on the slider by two LDVs. The projection seemed to be an appropriate target of the measurement, since the projections were the only contact points of the slider with the Rayleigh wave so that all the driving force was produced hereout. Moreover, since the projections were small therefore of high stiffness relative to the slider body, the internal or overall deformation of the projections was negligible to the ones of the slider body. Accordingly, we could regard the projection as a rigid body, namely the displacements of the projection were yielded by the normal and tangential deformations of the slider body. The rigid projection means any point of the projection moved simultaneously. That was very helpful for the measurement by using limited spot size of the laser beam, i.e., narrowing the spot sizes corresponding to the diameter of the projection would afford reliable measurements of the projections’ displacements.

The LDV system (Polytec GmbH, Waldbronn, Germany) used in the experiment comprised of an optical head OFV-353 and a controller OFV 3001 in which OVD-30 ultrasonic decoder (bandwidth: 50 kHz–20 MHz) was installed. The optical head provides minimum 10 µm spot size at the distance of 175 mm.

III. EXPERIMENTAL SETUP

The stator was a 62x14x1 mm³ rectangular plate of double-side polished 128° y-x lithium niobate. The IDTs were fabricated by vacuum deposition in which the dimensions of the IDTs were 400 µm in pitch, 100 µm in electrode strip and gap widths, and 9 mm in aperture. The IDT was composed of 20 electrode strip pairs, and the resonance frequency for the Rayleigh wave excitation was 9.6 MHz. Sticky tapes were attached as acoustic absorbers onto the outside of the driving IDT and the other side IDT so as to prevent reflection waves.

The silicon slider was made of a 5x5x0.3 mm³ square plate of silicon substrate. Number of 169 cylindrical projections was fabricated in a 4x4 mm² central square region of the surface by means of reactive ion etching technique. The cylindrical projections were 50 µm in diameter, 2 µm in height, and arranged at a 300 µm pitch.

A photograph of an experimental SAW motor is shown in Fig. 1. The stator was placed on a 10 mm thickness glass substrate so that the frictional surface of the slider was visible. The basement block had 48x8 mm² observation window underneath the track of the slider. A moving part of the setup was comprised of the slider, a steel hemisphere and a linear guide rail. The slider was glued with epoxy resin onto the steel
hemisphere. The hemisphere was put in a steel washer, which was glued to the linear guide stage.

The experimental SAW motor was fixed on a vibration isolating common bed. As shown in a photograph of the experimental setup in Fig. 2, two optical heads of the LDVs were fixed to the bed as well in a distance to achieve minimum spot size of the laser beam. The LDVs are hereafter called 'the left LDV' and 'the right LDV', respectively. The postures of the optical heads were set in such a way that the laser beams had 30° of incidence angles and 5° of elevation angles. The elevation angle was for the purpose that a forward scattering beam of a LDV would not incident on the other LDV’s optical head. A digital microscope (VH-5910 + ZH-Z25, Keyence Corp., Osaka, Japan) was put between the LDVs not to interfere with the optical path of the laser beams. While observing the slider surface by the microscope, focal points of the two LDVs’ laser beams were adjusted to a projection. The spot sizes were slightly larger than the projection’s diameter of 50 µm.

A. Data Acquisition

The LDV measures displacement in a direction parallel to the optical head by using the Doppler signals of back scattering beam. Hence, a measured displacement of the projection was a composition vector of normal and tangential displacement vectors as schematically illustrated in Fig. 3. Referring the figure, the measured displacement of the left LDV 'L' and the right LDV 'R' are represented by using the projection’s normal and tangential displacements 'V' and 'H' as follows:

\[
L = \frac{(V \cos 30^\circ - H \sin 30^\circ) \cdot \cos 5^\circ}{2 \cdot \cos 30^\circ \cdot \cos 5^\circ} \quad (1)
\]

\[
R = \frac{(V \cos 30^\circ + H \sin 30^\circ) \cdot \cos 5^\circ}{2 \cdot \sin 30^\circ \cdot \cos 5^\circ} \quad (2)
\]

where the angle 30° is the incident angles of the laser beams, and the factor cos 5° corresponds to the 5° of elevation angle.

The normal and tangential displacements are then calculated as follows:

\[
V = \frac{(L + R)}{2 \cdot \cos 30^\circ \cdot \cos 5^\circ} \quad (3)
\]

\[
H = \frac{(R - L)}{2 \cdot \sin 30^\circ \cdot \cos 5^\circ} \quad (4)
\]

Note that the reflective index of lithium niobate is not implemented into the equations, although the Doppler shift of laser beam conceptually caused at the stator-slider projection interface. We suppose that the reflection of the beam mainly happened at air-projection interface, since a real contact area
is significantly smaller than a nominal contact area due to roughness of surfaces.

Owing to the fact that the laser beams were incident with angles through the glass substrate and the stator, the reflected backscattering light from the projection to the optical head was of low intensity; the signal level was around 1/10 of the noise level. To elicit the displacement signal, we had to average 500 times of measurements. The high noise level, on the other hand, set a limit to the resolution of measurements, namely we could not efficiently use the full scale of a digital oscilloscope that was used for the data acquisition. The displacement quantization step was 0.78 nm in the measurements. Adding to that, since the decoder of the LDV has limited bandwidth as shown in Fig. 4, low frequency motion, such as the uplift motion or the travelling motion of the slider, was filtered out.

B. Time Response

The measurements of a slider projection’s normal and tangential displacements are shown in Fig. 5, in which the driving voltage was 100 \( V_{\text{peak}} \) and the driving waves were 100 cycles, and the preload was 15 N. The indicated data were the average of 500 measurements.

The Rayleigh wave reached the projection approximately 3 \( \mu s \) later. The Rayleigh wave amplitude on this driving voltage condition was 20 nm. The IDT was composed of 20 pairs of electrode strips so that it took 20 wave cycles (2.1 \( \mu s \)) for rise-up and fall-down of the amplitude as schematically illustrated in the figure. The measurement showed that the projection displaced both in normal and tangential direction followed to each wave vibration, which vibration amplitudes were related to the Rayleigh wave amplitude.

V. Analysis

In order to deduce the forces at the frictional interface from the measured displacements, the contact mechanics formulae [4] were introduced. To translate the experimental case into the contact mechanics terminology, we approximated this case as a rigid adhesive punch indenting to elastic half-space [5], [6]. More specifically, the projection and the slider body were approximated as rigid punch and elastic half-space, and the virtual boundary between them was assumed to be completely adhesive; that is schematically illustrated in Fig. 6. The contact mechanics formulae give the forces at the virtual boundary from the displacements. Then due to the rigid projection assumption, the total normal and tangential forces at the virtual boundary are identical with those at the frictional interface, thus we could deduce the forces at the frictional interface from the measured displacements.

A. Quasi-Static Analysis

1) Formulae: The coordinate for forces and displacements is that x-axis corresponds to the traveling direction of the slider and z-axis corresponds to the depth direction of the slider as shown in Fig. 6. Assuming the elastic half-space as an isotropic material whose Young’s modulus and the Poisson’s ratio are \( E \) and \( \nu \), the normal force \( P \) caused by a uniform normal displacement \( u_z \) of a circular region at an elastic half-space is given by [5],

\[
P = 4\eta a \ln(3 - 4\nu)/(1 - 2\nu) \cdot u_z
\]

where \( \eta = E/2(1 + \nu) \) and \( a \) denotes the radius of the projection. As well, the tangential force \( Q \) caused by a uniform tangential displacement \( u_x \) is given by [6]

\[
Q = \eta a(1 + 1 - 2\nu)/(16\pi\kappa) \cdot u_x
\]

where \( \kappa = \ln(3 - 4\nu)/2\pi \).
2) **Force Estimation:** The material of the slider 'silicon' is an anisotropic material. To use the above equations, we employed an approximated isotropic elastic constants $E$ and $\nu$ by means of Voigt average, namely $E=165.6$ [GPa] and $\nu=0.218$ [7]. The diameter of the projection was 50 $\mu$m, thereby $a$ is 25 $\mu$m.

Enlarged results under 15 N condition previously shown in Fig. 5 are indicated in Fig. 7 (a) in normal direction and (b) in tangential direction, in which time shifted data are overwritten in phase. Assuming that the normal deformation was zero at the bottommost position in (a), the normal force was calculated by using eq. (5) from the sinusoidal fitting result to the normal displacement in (a). The result is shown in (c). The force varied from 0 to 0.16 N. The mean value of the normal force was 10 % smaller than the mean preload per projection, 0.089 N, which is the given preload divided by the number of projection, namely 15 N/169.

The tangential force was calculated by using eq. (6) likewise, which is indicated in (d). For reference, the computed normal force $P$ multiplied by the friction coefficient $\mu$ i.e. maximum frictional force is depicted, where the coefficient between a silicon slider and a stator was reported as 0.18 [8]. Obviously, the maximum tangential force was much larger than the frictional force and the phase of the maximum tangential force was not coincident with that of the friction force. Two possible explanation of this phenomenon are conceived: one is a dynamic effect that may give rise to resonant response of the projection; the other is that the friction coefficient is much larger than the reported value. These two possibilities are discussed in the following subsections.

**B. Dynamic Analysis**

The rigid punch indentation problems can be solved in dynamic way by means of contact mechanics in which a sinusoidally oscillating displacement or force in the contact area at a certain frequency $\omega$ is a given condition [9]. To consider the response of the elastic half-space to an oscillating force, the term 'receptance' has important meaning. The receptance is defined as the ratio of the mean surface displacement within the loaded area to the total load. It is a complex quantity: the real part gives the displacement which is in-phase with the applied force; the imaginary part gives the displacement which is $\pi/2$ out-of-phase with force [4]. The inverse of receptance has the same form of the force-displacement system with a spring in parallel with a viscous dashpot. Then, the energy 'dissipated' by the dashpot corresponds to the energy radiated through the half-space by wave motion [4].

1) **Formulae:** The dynamic solution of a rigid adhesive punch problems in normal direction has, to the best of the author’s knowledge, not been reported. Thus the frictionless boundary case was used to estimate the dynamic response of the projection in the normal direction. In this boundary condition by using $\beta = \omega \sqrt{\eta/\rho}$ as a frequency parameter [9] where $\rho$ is the density, the inverse of receptance in the
rigid circular body indentation is expressed as \[9\], \[10\]:
\[
P = \frac{4\eta a}{w_0(1-\nu)} \cdot (p_1 + ip_2)
\] (7)
where \(w_0\) is the displacement amplitude in normal direction, and
\[
p_1 = 1 - a_{11}\beta^2 + a_{12}\beta^4, \quad p_2 = a_{21}\beta + a_{22}\beta^3
\] (8)
The coefficients \(a_{ij}\) are the function of Poisson’s ratio, and numerical values of some Poisson’s ratio are reported \[9\], \[10\].

To the author’s knowledge, the dynamic solution of a rigid punch problem that produce only tangential displacement has not been reported. Only the solution of the tangential displacement coupled with the rocking rotation has reported. Then, this solution was used to estimate the dynamic response of the projection in tangential direction. The inverse of receptance in tangential direction is \[9\], \[11\]
\[
\frac{Q}{d_0} = \frac{4\eta a}{1-\nu/2} \cdot (r_1 + ir_2)
\] (9)
where \(d_0\) is the displacement amplitude in tangential direction, and
\[
r_1 = 1 - c_{11}\beta^2 + \ldots, \quad r_2 = c_{21}\beta - c_{22}\beta^3 + \ldots
\] (10)
The coefficients \(c_{ij}\) are also the function of Poisson’s ratio, and some numerical values are reported \[9\], \[11\].

2) Inverse of Receptance: The numerical values of the coefficients \(a_{ij}\) and \(c_{ij}\) of Poisson’s ratio 0.25 is reported \[9\]; this Poisson’s ratio is the closest value to the approximated isotropic elastic constants of silicon. Thereby, by using these values the inverse of receptance in normal and tangential directions are computed. The results are indicated in Fig. 8 in such a way to represent the displacement amplitude ratio to the constant applied force (a) and the phase delay of the displacement to the applied force (b).

The displacement amplitude ratio at the driving frequency 9.61 MHz was 0.985 and 0.994 relative to the static case in normal and tangential direction, and the phase delay at 9.61 MHz was -12.8 and -9.6 degrees in normal and tangential direction, respectively. Presuming the inverse of receptance as a parallel spring-dashpot system, the mass of the projection, 9.62 ng, was attached to the system and the responses of the projection were computed, which results are also depicted in the figure. As obviously seen in the figure, the mass loading did not affect the amplitude ratio and the phase delay, since the mass was very small in relation to the spring constants.

The dynamic effect, namely the wave radiation, would delay the response of the projection to the loading force in which the delay in normal direction was slightly larger than that in tangential direction. This slight difference of the delays, however, did not explain the phase difference of the tangential force and the maximum frictional force indicated in Fig. 7 (d). Moreover, the dynamic effect would not amplify the displacement in both directions. Thus, we did consider the second possibility—the friction coefficient.

C. Friction Coefficient

To discuss the effect of friction coefficient, a stick-slip friction drive model \[8\] needs to be implemented with modification to the dynamic contact mechanics model of the projection. The friction drive model assumes a single point contact and takes into account the normal stiffness \(k_n\) and tangential stiffness \(k_t\) of the stator, and corresponding normal and tangential deformation, \(d_n\) and \(d_t\). The stiffness multiplied by the deformations give the normal and tangential force, \(P\) and \(Q\). Then if \(Q\) exceeds the limiting value \(\pm \mu P\), where \(\mu\) is the friction coefficient, slip occurs. This procedure of friction modeling is compatible with that of the contact mechanics modeling \[4\] in which the contact point extends to the area and the contact area has pressure distribution inside, which may cause partial slip; however the complete slip occurs in the same condition \(|Q| > \mu P\). This slip judgment can be applied to the impact in a same manner.

The contact mechanics modeling procedure is briefly explained as follows: the stator is elastic half-space that vibrates in the same way of surface particles of the wave, i.e., the phase of the wave is same everywhere, and the slider body is fixed half-space, which is schematically shown in Fig. 9.
The frictional state between the projection and the stator is complete stick or slip; the complete slip occurs if $|Q| > \mu P$.

We employed the dynamic modeling of the projection attached to the slider body. Since the forced displacement is given to the stator, it was not sure it was correct or not to model the stator in dynamic way. Then, the stator was modeled that only has stiffness components; the stiffness of the stator in relation to the projection indentation was calculated on the complete adhesive boundary condition.

Substituting the amplitude of the wave at 100 $V_{\text{peak}}$ condition to $a_{v}$ and $a_{h}$, namely 20 and 18 nm respectively, computations of the differential equation of the contact mechanics model was carried out under the condition of $\mu=0.18$ and 1. The computation results of the normal and tangential displacements of the projection are indicated in Fig. 10 together with the measurements. The time of computed results were manually adjusted to be in phase with the normal displacement. The result using the friction coefficient 1 seems apparently to be much closer to the measurements than that using the friction coefficient 0.18 that was the measured value between a silicon slider and a stator [8]: the amplitude of tangential displacements were much the same. However, the phase of the tangential displacement stayed away from the measurement.

**D. Discussion**

Friction only happens at real contact area thus the friction is a function of real contact area instead of nominal contact area (for example, see [12]). That is because that most of the engineering materials have roughness owing to the machining processes. The roughness of silicon substrates or lithium niobate substrates is much comparatively small due to the well-developed polishing process. We can imagine that in the contact between such substrates the real contact area could be much larger, and which could result a high friction coefficient. In the case of the silicon slider, however, the real contact area is, in a certain sense, artificially controlled owing to the fabrication of projections. Therefore, the macroscopically measured friction coefficient might be much smaller than that between the projection and the stator. We may need to use the value such like a friction coefficient per unit area to consider the friction drive of the SAW motor. Investigating the friction property will be the future work.

![Fig. 9. Schematic view of the friction drive model.](image)

![Fig. 10. Measurements and Simulations of the projection's displacements.](image)

**REFERENCES**