Elastic Contact Conditions to Optimize Friction Drive of Surface Acoustic Wave Motor

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Abstract—The optimum pressing force, namely the preload, for a slider to obtain superior operation conditions in a surface acoustic wave motor have been examined. We used steel balls as sliders. The preload was controlled using a permanent magnet. The steel balls were 0.5, 1, and 2 mm diameter, with the differences in diameter making it possible to change contact conditions, such as the contact pressure, contact area, and deformation of the stator and the slider. The stator transducer was lithium niobate, 128 degrees rotated, y-cut x-propagation substrate. The driving frequency of the Rayleigh wave was about 10 MHz. Hence, the particle vibration amplitude at the surface is as small as 10 nm. For superior friction drive conditions, a high contact pressure was required. For example, in the case of the 1 mm diameter steel ball at the sinusoidal driving voltage of 180 V_{peak}, the slider speed was 43 cm/sec, the thrust output force was 1 mN, and the acceleration was 23 times as large as the gravitational acceleration at a contact pressure of 390 MPa. From the Hertz theory of contact stress, the contact area radius was only 3 μ m. The estimation of the friction drive performance was carried out from the transient traveling distance of the slider in a 3 msec burst drive. As a result, the deformation of the stator and the slider by the preload should be half of the vibration amplitude. This condition was independent of the ball diameter and the vibration amplitude. The output thrust per square millimeter was 50 N, and the maximum speed was 0.7 m/sec. From these results, we conclude that it is possible for the surface acoustic wave motor to have a large output force, high speed, quick response, long traveling distance, and a thin micro linear actuator.

I. INTRODUCTION

USING high frequency vibrations over 1 MHz, the possibility of a high frequency ultrasonic motor has been demonstrated [1], [2]. However, these studies used fluid coupling to transmit the actuation force. Hence, the operational force was very tiny. The true merit of ultrasonic motors and piezoelectric actuators is their high output force. Therefore, a friction drive is superior to the liquid coupling for general actuators.

We have demonstrated the operation of the friction drive surface acoustic wave motor at 10 MHz [3], [4] and 20 MHz [5], [6]. The important point is the contact con-

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Fig. 1. Setup for experiments for the optimum preload conditions.

ditions between the slider and the stator because the squeezed air film disturbs the friction drive [7]. In high frequency operation, the effect of the squeezed film is significant due to the small vibration amplitude of displacement. For a friction drive with high frequency vibrations in the order of nano meters, a high contact pressure of 100 MPa is required. The performance, however, is not sufficient for an actual actuator. Hence, it was necessary to investigate the operation conditions in detail.

The surface acoustic wave (SAW) transducer is advantageous as a miniaturized actuator because of its high power density. At a driving frequency of approximately 10 MHz, a LiNbO₃ vibrator with dimensions of about 10 × 10 × 1 mm³ has the ability to transduce 100 W input electrical power to mechanical vibrations [8]. This means that the LiNbO₃ SAW device has an energy density of 200 kW/kg at 10 MHz. Such a high power density results in a high performance miniaturized actuator.

II. EXPERIMENTAL SETUP

The same stator transducer as that described in our previous paper [3] was used for the experiments. The piezoelectric wafer was 128 degrees rotated y-cut x-propagating LiNbO₃. Two interdigital transducers (IDTs) were arranged on a 3-inch wafer as shown in Fig. 1. The IDT pitch was 400 μ m, and the electrode strip width was 100 μ m. The electrode strip included 10 pairs for each transducer. The driving frequency was 9.6 MHz. The normal direc-

Fig. 2. Force of the preload in dependence of spacer thickness.

tion vibration amplitude of the surface particle was about 19 nm at a driving voltage of 180 V_{peak}. The vibration velocity in the tangential direction was 0.8 m/sec at the same voltage.

The sliders for this motor were small steel balls. In order to provide a preload, namely, a pressing force between the slider and the stator, a permanent magnet was installed beneath the wafer. The preload was changed by the spacer thickness between the magnet and the wafer. Due to the effect of the magnetization of the steel balls, the slider moved without rotation. The magnet was a 5 × 5 × 25 mm³ Neodymium permanent magnet which was magnetized in the z axis direction in Fig. 1. The uniformity of the magnetic attraction force was sufficient for the experiments within the operational area of the motor. The setup for the experiments is shown in Fig. 1. The RF power is supplied to one IDT to excite the traveling Rayleigh wave in the x direction. The steel ball moves to the IDT. The transverse motion of the steel ball was examined [9], [10]. We measured the transverse displacement and the speed of the steel ball in the x direction.

The attraction force between the permanent magnet and the 0.5, 1.0, and 2.0 mm steel balls was measured with an electric balance. The gap between the magnet and the ball was changed from 1 mm (thickness of the wafer) to 8.5 mm using several spacers. The results are given in Fig. 2. From these values, the contact pressure distribution between the stator (LiNbO₃ wafer) and the steel balls is calculated using the Hertz contact theorem. At the center of the contact region, the pressure is maximum. The maximum contact pressures are shown in Fig. 3. The contact pressure was able to be controlled up to 400 MPa for a 0.5 mm diameter ball and up to 600 MPa for a 2.0 mm diameter ball. In a previous paper [3], the contact pressure was reported to be below 140 MPa.

Because of the elasticity of the materials, the contact area of the slider ball is finite. The slider and the stator deform as illustrated in Fig. 4. The contact radius can be

Fig. 3. The maximum pressure of the ball sliders as a function of spacer thickness.

Fig. 4. Depression value of h due to the elastic deformation of the slider h_{sl} and the stator h_{st} by the preload.

calculated approximately with the Hertz contact theorem [11]. The actual contact radius was at most several micrometers when the steel ball sliders were pressed on the stator with the attractive force of the permanent magnet. The calculated values of the contact radii are indicated in Fig. 5. Making much comparison between the ball radii and the contact radii, we find that the slider ball radii are much too large for the tiny contact radii. However, because the purpose of the experiment is to estimate the conditions for the friction drive, we can ignore here the problem of the oversized slider dimensions in relation to the contact area. The depression of the slider ball into the stator, as indicated by h in Fig. 4, was calculated using the same procedure as that used for calculating the contact radius. The depression h is the sum of the elastic deformations of the stator h_{st} and the slider h_{sl} , as indicated in Fig. 4. This value was on the order of several nano meters as shown in Fig. 6. The deformation is on the same order as the stator vibration deformation.

Fig. 5. Contact radius of the steel ball slider as a function of the preload.

III. TRANSIENT RESPONSE OF THE SLIDER IN A SHORT TIME DRIVE

In an optimum preload condition, it is desirable to achieve both a high speed and a large thrust. In a light preload, the steady speed is high but the thrust is tiny; and in a high preload condition, the results are reversed. At an excessive preload, the slider can no longer move. Tradeoffs are necessary between a high speed and a large thrust to determine the optimum preload conditions. Moderate conditions should be investigated in order to enhance the performance. Because a real time measurement of the speed was not possible in this experiment due to the available equipment, we followed a simple method to estimate the speed.

A. Motion of the Slider

From the transient curve of the transverse displacement of the slider, we can obtain the transient response of the speed of the motor. The maximum thrust is deduced from the slope at the starting position. The speed without a mechanical load is obtained from the steady part of the motor speed. The relationship between the transient response and the steady state performance of the motor has been investigated [12]. Using the maximum thrust F , no-load speed v_o , and the mass of the moving part m , the transient response of the slider speed v is expressed as:

$$\begin{aligned} v &= v_o(1 - \exp(-pt)), \\ p &= F/(mv_o). \end{aligned} \quad (1)$$

We can obtain the transverse displacement d of the slider in the x direction by integrating the speed v with respect to the time t ,

$$d = v_o t - v_o(\exp(-pt) - 1)/p. \quad (2)$$

If the product pt is small, meaning that the driving period is short, using the Maclaurin's series expansion, (2) can be

Fig. 6. Depression value h as shown in Fig. 4 versus preload.

rewritten as:

$$\begin{aligned} d &= v_o t - v_o(1 - pt/1! + (pt)^2/2! - (pt)^3/3! + \cdots - 1)/p \\ &\sim v_o t - v_o(1 - pt/1 + (pt)^2/2 - 1)/p \\ &= v_o pt^2/2. \end{aligned} \quad (3)$$

In this case, the traveling distance d is approximately proportional to the square of the driving duration. If the product pt is large, meaning that the driving period is long, the traveling distance is proportional to the driving period. As is well-known, a large thrust and a high no-load speed are inconsistent with each other. By measuring the traveling distance and the driving duration, therefore, a balance between the thrust and the speed can be estimated. In other words, the traveling distance of the slider will become maximum at the optimum preload with an appropriately short drive time.

The transient response of the motor was determined by measuring the driving intervals for a 5 or 10 mm transverse distance of the slider. By changing the driving duration, the transverse distance was measured as shown in Fig. 7. The conditions of the experiment were as follows: 1 mm diameter steel ball slider with a preload of 350 MPa and a driving voltage of 180 V_{peak}. A square law increase with burst duration was observed for burst below 0.1 msec. The transient responses of the slider that were measured with a high speed video camera and an image processing equipment are also shown in Fig. 8. From these results, the slope was found to become moderate, and after 10 msec it became linear. The linear-to-square transition region was observed approximately from 0.1 to 10 msec driving time.

From the slope of the displacement data, as shown in Fig. 8, the transient speed curves were obtained as indicated in Fig. 9. Also from these data it is clear that the transient region was around 3 msec. We decided, therefore, to measure the displacement with a 3 msec drive burst for various operation conditions to estimate the optimum preload condition.

Fig. 7. Displacement d of the slider by short duration drive burst from 10 μ sec to 3 msec.

Fig. 8. Displacement d of the slider versus burst time measured with a high-speed camera and the resulting change in slope.

Fig. 9. Transient responses of the slider speed calculated from the displacement d data measured with a high-speed camera shown in Fig. 8.

Fig. 10. Displacement d of the steel ball sliders by 3 msec drive burst as a function of the maximum contact pressure.

Fig. 11. Displacement d of the steel ball sliders by 3 msec drive burst as a function of the depression h .

B. Optimum Preload Condition

At the driving voltage of 180 V_{peak} , the displacement d of the slider from a 3 msec drive burst was measured. The driving performance has been improved significantly in comparison with previous results [3], [4], as shown in Fig. 10. The sliders were 0.5, 1, and 2 mm in diameter. The contact pressure was changed up to 500 MPa. The maximum transverse displacement conditions depended on the diameter, as can be seen in Fig. 10. For the 2 mm diameter ball, the optimum preload was about 270 MPa. However, the optimum value was 350 MPa for the 1 mm diameter ball, which was much higher than that for the 0.5 mm diameter ball. The optimum pressure depended on the geometry of the contact radius of the sliders.

However, the depression h of the sliders, specifically the elastic deformations of the slider and the stator, gave a good result. Fig. 11 indicates the transverse displacement in a 3 msec drive burst as a function of the depression h due to the preload. The optimum depression h of the 1 and

Fig. 12. Displacement of the sliders by 3 msec burst for several driving voltages.

2 mm ball sliders was approximately 9 nm in both cases. For the 0.5 mm ball, the optimum condition seemed to be almost the same. The measured amplitude of the vibration displacement normal to the surface of the stator for a 180 V_{peak} driving voltage was 19 nm. Hence, the optimum depression of the slider was almost half the amplitude a of the vibration displacement of the surface acoustic wave.

To confirm the optimum depression condition at different vibration amplitudes, we made estimates in the same way at driving voltages of 120 V_{peak} and 60 V_{peak} using 0.5 mm and 1 mm diameter balls. The results shown in Fig. 12 indicate that half of the vibration displacement a was optimum for the depression h . The optimum depression h was about 6 nm for 120 V_{peak} and 3 nm for 60 V_{peak}, both of which were also half of the vibration displacement a . Therefore, it can be concluded that, to obtain a large thrust and high speed, which requires a quick response of the slider, the preload should be adjusted so that the elastic deformation of the depression h is half of the amplitude a of the vibration displacement.

IV. THRESHOLD AND STEADY SPEED

Usually an ultrasonic motor has a dead zone at a low driving voltage when the slider does not move. The threshold driving voltage or vibration amplitude a when the slider begins to move seems to depend on the preload. However, the quantitative relation between the preload and the threshold vibration amplitude has not yet been made clear.

The threshold driving voltage when the slider begins to move was measured versus the maximum contact pressure for the three different diameter balls as shown in Fig. 13. The threshold voltage depended on both the pressure and the diameter. However, as to the depression of the sliders, the threshold voltages were independent of the ball diameter, as can be seen Fig. 14. The threshold vibration displacement a was almost the same as the elastic de-

Fig. 13. Threshold driving voltages of the stator that gave the minimum vibration for the sliders linear motion.

Fig. 14. Threshold driving voltages as a function of the depression h by the preload.

mation h of the sliders. For example, when the depression h was 10 nm, the threshold voltage was approximately 100 V, corresponding to a 10 nm vibration displacement amplitude. At a low vibration amplitude, the threshold amplitude was slightly higher than the depression h , and at a high vibration amplitude a , the threshold was slightly lower than the depression h . This might be due to the influence of the roughness of the contact surfaces.

The steady slider speed v_s was estimated from the measurements using a high speed camera and an image processor unit. An example of the measurement is shown in Fig. 9. The steady slider speed was independent of the ball diameters, as shown in Fig. 15. In this experiment, the driving voltage was 120 V_{peak} and 180 V_{peak}, and the vibration velocities in the tangential direction were 0.54 m/sec and 0.81 m/sec, respectively. If the vibration displacement amplitude a , which is proportional to the driving voltage was the same, the steady speed v_s depended on the depression h . This means that the elastic

Fig. 15. Steady speed v_s of the sliders versus depression h . Fig. 16. Slip of the slider speed versus the velocity of the stator surface vibration v as a function of the ratio of the depression h to the vibration amplitude a .

deformation of the contact part, the depression h , is the important factor in fixing the steady speed v_s .

Fig. 16 indicates the ratio of the slider speed v_s to the tangential velocity of vibration v_t as a function of the ratio of the depression h to the normal displacement of vibration a . Specifically, the steady speed v_s and the depression h are normalized to the vibration components of the stator surface particles for comparison between the different driving conditions. The two different sets of driving conditions which are mentioned in Fig. 15 are also indicated in Fig. 16. The two sets of data show that the slider speed depends on the ratio of the depression h to the normal vibration displacement a . The slider speed was found to be almost 60% of the maximum tangential vibration velocity of the surface particles at the appropriate depression that gave the optimum condition. In other words, the depression h was half of the normal vibration displacement a . For a precise analysis of this, a detailed experiment and a simulation of the operation are required.

Fig. 17. Transient response of the 1 mm diameter ball slider.

V. OUTPUT FORCE OF THE FRICTION DRIVE

To estimate the output force of the surface acoustic wave motor, the available thrust and optimum conditions of operation were investigated. In the experiment described in Section III, the optimum condition was a balance of the acceleration and the steady speed. This was a simple way to roughly determine the appropriate conditions. However, for a quantitative evaluation, the dynamic response should be measured. For this purpose, the transient motions of the sliders were measured using a high-speed camera and an image processor unit. This method was effective in measuring the dynamic response of the motor, but it was not very efficient. The image processing took several hours, so the measured conditions were limited. The maximum frame rate was 4500 per second for full frame size of 256×256 pixels. In the case of a small frame, the maximum frame rate was 40,500 per second.

An example of the most rapid response using a 0.5 mm steel ball slider is shown in Fig. 17, in which the maximum contact pressure was about 430 MPa. The speed of the slider was calculated from the slope of the displacement d between the time of the frames. The steady speed was approximately 700 mm/sec at a driving voltage of 180 V_{peak}. For a quick response, the slope data varied widely due to the low resolution of the measurement. Figs. 8 and 9 show the same kinds of results for a 1 mm diameter ball slider at a driving voltage of 180 V_{peak}. The thrust was estimated from the starting acceleration and the weight of the slider. The rest of the data for a driving voltage of 120 V_{peak} were estimated in the same way, although the curves are omitted here. The measured results of each operation condition are summarized in Tables I and II and Figs. 18 to 21.

In Table I, data are shown for two different driving voltages and three preload conditions. The three preload values were 390 MPa, 310 MPa, and 190 MPa. The preload of 390 MPa was the optimum condition for the 3 msec drive at 180 V_{peak}. The rise time of the slider speed v was the shortest at this preload, thus the thrust was a maximum

TABLE I
EXPERIMENTAL RESULTS FOR THE 1 mm DIAMETER STEEL BALL SLIDER OBTAINED WITH THE HIGH-SPEED CAMERA.

Driving voltage	[V _{peak}]	180	180	180	120	120	120
Spacer thickness	[mm]	1.5	2.5	5	1.5	2.5	5
Steady speed	[mm/sec]	430	610	*800	270	340	460
Maximum acceleration	[m/sec ²]	230	170	42	100	108	42
Maximum thrust	[mN]	0.95	0.69	0.17	0.41	0.44	0.17
Thrust density	[N/mm ²]	35	40	26	15	26	26

*Expolarated value from the data.

TABLE II
EXPERIMENTAL RESULTS FOR THE 0.5 AND 2 mm DIAMETER STEEL BALL SLIDERS.

Diameter of ball	[mm]	0.5	0.5	2
Driving voltage	[V _{peak}]	180	120	180
Spacer thickness	[mm]	0.5	0.5	4
Steady speed	[mm/sec]	680	390	490
Maximum acceleration	[m/sec ²]	800	900	25
Maximum thrust	[mN]	0.41	0.16	0.81
Thrust density	[N/mm ²]	50	56	16

Fig. 18. Thrust of the steel ball slider in dependence of depression d to amplitude a ratio.

Fig. 19. Output force of the motor per unit contact surface.

Fig. 20. Equivalent frictional coefficient of the steel ball slider depending on d/a .

Fig. 21. Acceleration of the steel ball sliders; the driving voltages were 120 or 180 V_{peak}.

of 0.95 mN. In contrast, at the preload of 190 MPa, the steady speed was a maximum of 800 mm/sec, which was extrapolated from the transient curve of the slider speed. This speed v_s was almost the same as the peak value of the vibration velocity v_t of the stator surface particles in the tangential direction.

The preload of 310 MPa was the optimum at 120 V_{peak} driving voltage for the thrust. Again, for a higher preload of 390 MPa, the thrust as well as the steady speed decreased compared to the 310 MPa preload. The optimum preload was similar to that of the previous experiment with the 3 msec drive. However, the output force of the 310 MPa preload and 120 V driving voltage was smaller than that of the 180 V driving voltage; the preload was the same. The ratio of the depression h to the vibration amplitude a , which gave the maximum output force, was around 0.5, as indicated in Fig. 18.

From the view point of the thrust density, the output force per unit contact area, the optimum ratio of the depression h to the vibration amplitude a seems to be smaller than 0.5, as shown in Fig. 19. The maximum condition of the thrust density for the 1 mm diameter slider was around 0.3 (depression h)/(displacement a) ratio. This is because the equivalent friction coefficient, namely, the ratio of the thrust and preload decreased, if the (depression h)/(displacement a) ratio was larger than 0.3 as shown in Fig. 20. If the contact duration of the slider and the stator is less than a half cycle of the vibration, the thrust will increase in proportion to the preload. However, if the contact duration is more than a half cycle of the vibration, the thrust will not be proportional to the preload anymore, due to the backward driving force at the trough part of the wave.

We recognize from Figs. 19 and 21 that the smaller diameter ball slider was favorable for a high output force and a quick response slider. The smaller diameter slider caused a high thrust density because of the high contact pressure. An acceleration value of 900 m/sec² was the amazing result. Using a lot of small steel balls glued to a flat surface in order to make the slider contact at high pressure and at large normal force, we are able to create a high output force and a quick response actuator [13], [14]. If the slider dimension is 10×10 mm² and the actual contact surface is a thousandth of the slider area, the thrust will be 5 N, as deduced from the result with the 0.5 mm ball slider. An example of a new actuator design is shown in Fig. 22. By simply increasing the number of projections on the sliders, a larger thrust will be obtained. However, there will be a limit to the output power because the propagating energy of the stator transducer is finite. The energy conversion mechanism should be investigated to estimate the available thrust using a surface acoustic wave device [15].

Noteworthy is the fact that the elastic deformation h value of the stator and the slider by the preload is on the same order as the displacement of vibration a . A deformation is expected not only in the normal direction of the stator and slider surface, but also in the tangential direction due to the shear force. The shear deformation will

Fig. 22. Future design of a surface acoustic wave linear motor.

reduce the slip between the slider and the stator. As a result, the shear deformation will reduce the slip loss of the force transmission from the stator vibration to the unidirectional motion of the slider. This phenomena will be effective in improving the efficiency and extending the life of the actuator.

VI. CONCLUSION

We have successfully improved the friction drive conditions of the surface acoustic wave motor. The most important factor was the ratio of the elastic deformation of the slider and the stator by the preload to the normal displacement of the vibration. Notably, the thrust density was 50 N/mm², the maximum acceleration was 900 m/sec², and the maximum speed was 0.7 m/sec. We were able to demonstrate the applicability of the surface acoustic wave motor to a superior new linear micro actuator.

REFERENCES

- [1] R. M. Moroney, R. M. White, and R. T. Howe, "Ultrasonic micromotors," *Proc. IEEE Ultrason. Symp.*, 1989, pp. 745–748.
- [2] M. Takeuchi, H. Abe, and K. Yamanouchi, "Ultrasonic micro-manipulation of small particles in liquid using VHF-range leaky wave transducers," *Proc. IEEE Ultrason. Symp.*, 1994, pp. 607–610.
- [3] M. Kurosawa, M. Takahashi, and T. Higuchi, "Ultrasonic X-Y stage using 10 MHz surface acoustic waves," *Proc. IEEE Ultrason. Symp.*, 1994, pp. 534–538.
- [4] —, "Ultrasonic linear motor using surface acoustic waves," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, vol. 43, no. 5, pp. 901–906, 1996.
- [5] —, "Operation condition and output force of surface acoustic wave motor," *Tech. Rep. Inst. Elec. Info. Commun. Eng.*, US96-76, pp. 43–50, Dec. 1996 (in Japanese).
- [6] —, "A surface acoustic wave motor using V-shape groove guide," *Trans. Inst. Electron., Inform. Commun. Eng. A*, vol. J80-A, no. 10, pp. 1711–1717, 1997 (in Japanese).
- [7] T. Maeno and D. B. Bogy, "Effect of the hydrodynamic bearing on rotor/stator contact in a ring-type ultrasonic motor," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, vol. 39, no. 6, pp. 675–682, 1992.
- [8] M. Kurosawa, T. Watanabe, A. Futami, and T. Higuchi, "Surface acoustic wave atomizer," *Sens. Actuators*, vol. A 50, pp. 69–74, 1995.
- [9] M. Takahashi, M. Kurosawa, and T. Higuchi, "Direct frictional driven surface acoustic wave motor," *Proc. Int. Conf. Solid-State Sens. Actuators, Transducers 95*, Stockholm, pp. 401–404, 1995.

- [10] M. Kurosawa, M. Takahashi, and T. Higuchi, "Optimum preload of surface acoustic wave motor," *Proc. IEEE Ultrason. Symp.*, San Antonio, TX, pp. 369–372, Nov. 3–6, 1996.
- [11] S. P. Timoshenko and J. N. Goodier, *Theory of Elasticity*, 2nd Ed. New York: McGraw-Hill, 1970, pp. 409–414.
- [12] K. Nakamura, M. Kurosawa, and S. Ueha, "An estimation of load characteristics of an ultrasonic motor by measuring transient responses," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, vol. 38, no. 5, pp. 481–485, 1991.
- [13] M. Chiba, M. Takahashi, M. Kurosawa, and T. Higuchi, "Evaluation of a surface acoustic wave motor output force," in *Proc. IEEE Workshop on MEMS*, Nagoya, Japan, 1997, pp. 250–255.
- [14] M. Kurosawa, M. Chiba, and T. Higuchi, "Multi contact points slider for a surface acoustic wave motor," *Trans. Inst. Electr. Eng. Jpn., E*, vol. 117-E, no. 8, pp. 430–431, 1997.
- [15] M. Kurosawa, M. Takahashi, and T. Higuchi, "Contact and driving condition of a surface acoustic wave motor," in *Proc. IFAC Conf. Contr. Ind. Syst.*, pp. 396–401, May 20–22, 1997.

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