<技術講演>

The Effects of Surface Finish on Stick-slip with Cross-motion*

T. J. Takats**

Synopsis

With many of the automated machine tools, special purpose machines and alike the accurate positioning is normally done by an "error actuated" servo-mechanism.

Owing to slow down near to the "desired position of the positioning device, the dynamics of the lubricating film change; "sticktion" and subsequent "stick-slip" motion might take place.

When only a small "error" exist between the "desired" and actual position, the activating force of the servomechanism is often not enough to overcome the "limitting" friction thus the "small positioning error" will prevail,

In part I the tribological conditions are examined and in part II some methods are described to overcome or eliminate "stick-slip" in the required direction of motion.

Part I

The Problem of stick-slip

Consider an element of lubricant with dynamic lubrication.

Consider the element of lubricant shown in fig. 1

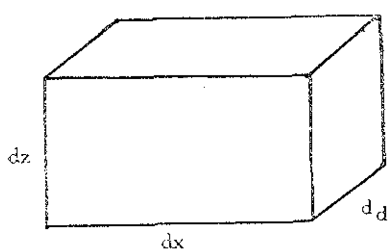

Fig. 1

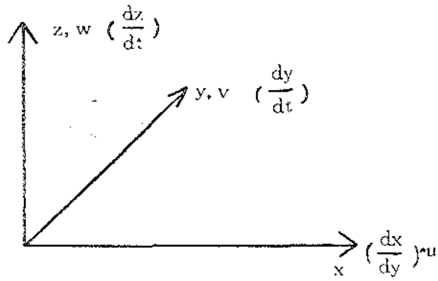

Fig. 2

and in the coordinate system-convention indicated in fig. 2. If relative motion exist, the rate of shear in the X direction is different between "top" and "bottom". Let the shear stess be on the "bottom" τ_x in the X direction The shear stress on the "top" be

$$\tau_x + \frac{\partial \tau}{\partial z} dz$$

Shear forces are shown in fig. 3.

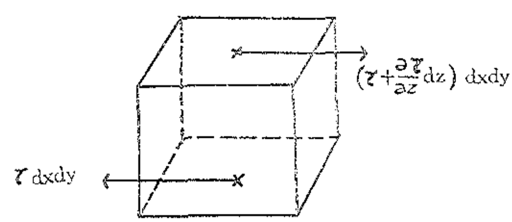

Fig. 3

The pressure varies along X, on the "left" face of the element it is $pdydz$ and on the "right" face of the element it is

$$\left(P + \frac{\partial p}{\partial x} dx\right) dydz$$

For dynamic equilibrium in the X direction the algebraic sum of these forces must be zero, thus:

$$\left(P + \frac{\partial p}{\partial z} dz\right) dydz + \tau dxdy - Pdydz - \left(\tau + \frac{\partial \tau}{\partial z} dz\right) dxdy = 0$$

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* 1975년 5월 30일 춘계 학술강연회에서 강연
** UNESCO Training Adviser, F.I.C.

or $\frac{\partial \tau}{\partial z}=\frac{\partial p}{\partial x}$ ....(i)

If one would consider the flow & shear in the Y direction, this, similarly would give

$$\frac{\partial \tau}{\partial z}=\frac{\partial p}{\partial y}$$

If Z is the direction of the "clearance" between slideway and slider, (i.e. the gap filled with lubricant, or the thickness of the lubricant) than, it is reasonable to assume that the pressure throughout the film thickness is the same, i.e.

$$\frac{\partial p}{\partial z}=0 \quad \cdots\cdots\cdots\cdots(ii)$$

Considering "Newtonian" lubricant, from Newton's viscosity equation, (shear stress=viscosity coefficient *x velocity gradient) one gets that:*

$$\tau_x=\eta\frac{\partial u}{\partial x} \quad \cdots\cdots\cdots\cdots(iii)$$

Where η=coefficient of viscosity

From (i) & (iii)

$$\frac{\partial\left(\eta\frac{\partial u}{\partial z}\right)}{\partial z}=\frac{\partial p}{\partial x}$$

Integrating gives

$$\eta\frac{\partial u}{\partial z}=\frac{\partial p}{\partial x}z+A_1$$

In η is constant in the film thickness, (h) than, after intergration

$$\eta u=\frac{\partial p}{\partial x}\ \frac{z^2}{2}+A_1z+B$$

Let velocity in X direction be u_1 at $z=h$

and u_2 at $z=o$

Obtaining values from these limits for A & B leads to

$$u=\frac{\partial p}{2\eta\partial x}(z^2-zh)+(u_1-u_2)\frac{z}{h}+u_2 \quad \cdots\cdots(iv)$$

Now, *considering the continuity of flow of an elementary column,* base dxxdy and hight h, as shown in fig. 4 one gets:

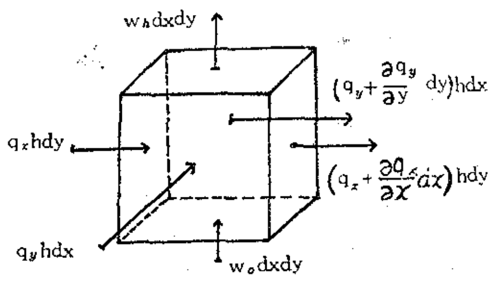

Fig. 4

From: flow entering=flow leaving (incompressible fluid)

$$q_xhdx+q_hhdx+w_odxdy$$
$$=\left(q_x+\frac{\partial q_x}{\partial x}dx\right)hdy+\left(q_y+\frac{\partial q_y}{\partial y}dy\right)hdx$$
$$+w_hdxdy$$

$$\text{or}\left[\frac{\partial q_x}{\partial x}+\frac{\partial q_y}{\partial y}+\frac{w_h-w_o}{h}\right]hdydx=0 \quad \cdots\cdots(v)$$

The average flow rate X direction will be given by:

$$q_x=\int_o^h udz \quad \cdots\cdots\cdots\cdots(vi)$$

Substituting for u from (iv)

$$q_x=\int_o^h\left[\frac{\partial p}{2\eta\partial x}(z^2-zh)+(u_1-u_2)\frac{z}{h}+u_2\right]dz$$

and partially differentiating the above with respect to ∂x yields

$$\frac{\partial}{\partial x}\left[-\frac{h^3\partial p}{12\eta\partial x}+(u_1+u_2)h/2\right]$$

The same argument *can apply in the* Y *direction, with* boundary velocities V_1 and V_2

Substituting these flow variation rates into the continuity aquation, (v) one gets

$$\frac{\partial}{\partial x}\left[-\frac{h^3\partial p}{12\eta\partial x}+(u_1+u_2)\frac{h}{2}\right]+\frac{\partial}{\partial y}\left[-\frac{h^3\partial p}{12\eta\partial y}\right.$$
$$\left.+(V_1+V_2)\frac{h}{2}\right]+\frac{w_h-w_o}{h}=O$$

This is keynold's equation in 3 dimension. Considering no transverse velocity, i.e. $V_1=V_2=O$ and

$U=U_1+U_2$ and using $\frac{dh}{dt}$ for (w_h-w_o)

one arrives at:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3\frac{\partial p}{\partial y}\right)$$
$$=6\eta\left(u\frac{dh}{dx}+\frac{2}{h}\ \frac{dh}{dt}\right) \quad \cdots\cdots\cdots\cdots(vii)$$

For a flat slide, for instance, one may consider that

$$\frac{\partial p}{\partial y}\longrightarrow 0$$

When U is low or approaches O, the flow in X direction will reduce drastically(neglecting inertia forces) which, when examining the above equation, indicates that h will also be reduced.

This will lead to a condition, known as boundary lubrication. Essentially, the continuity of the oil film separating the two surfaces breaks down, owing to reduction of the relative movement and the corresponding lack of lubricant flow. The film gets "punctured" (surface tension) by the "high spots" or "asperities" of the bearing surfaces.

These asperities from the mating surfaces come to contact and support the load. As the total area of the contacting asperities is very small, the stresses are in excess of the "yield stress" of material, thus "plastic deformation" of the surfaces take place, together with elastic, herzian momentarily deformation.

At such instances "cold welding" of the mating surfaces will take place. This represent and "instantaneous stop" or the "stick" portion of the "stick-slip" movement.

Now, if ∂y_1 is the yield stress of the slider and ∂y_2 is that of the sildeway and W is the total load on the bearing, than the total area of contact on which "cold welding" might take place is

$$A_1=\frac{W}{\partial_{Y1}} \text{ for the slider and}$$

$$A_2=\frac{W}{\partial_{Y2}} \text{ for the slideway}$$

Cold weld does not take place over the whole area owing to lack of time, presence of oxygen, etc. but, experiments show that to break i. e. to shear the so formed welded joints requires an accumulative force, equal to:

$$F=k_A\tau_L\frac{A_1+A_2}{Z}$$

If $A_1=A_2$ (i. e. $\sigma_{Y1}\simeq\sigma_{Y2}$) than, k_A, the "area coefficient of cold weld", is between 1/3 and 2/5, pending on the duration of the "stick" period. When the bearing remains in the "stick" position for prolonged time "sticktion" takes place.

When "sticktion" takes place K_A may be between 2/3 to 3/4 pending on conditions.

NOTE: "sticktion" in instruments is a very common occurrance. The actuation or motive force is very small charges of measured parameter; not enough to overcome "sticktion". As a practice, these instruments are slightly tapped by hand before reading, thereby removing the sticktion in the bearing surfaces of the movement of the instrument. In some cases, instrument panels were artificially vibrated, so as not to allow "sticktion" to take place.

When momentarly stick takes place and the accumulated motivating force is sufficient to break (shear) the cold-welded minute joints, a "slip" will follow.

During slip, complete dynamic lubrication conditions might be reestablished, but the forces are used up in supplying momentum to the sliding inertia, reducing eventually the speed of slip, (U), reducing the separation, (h) and stick might follow. The repetition of this condition is known as "stick-slip" movement.

In part II an investigation into the introduction of a "cross motion" will be described as a means, proved by experiments conducted by the writer, to reduce or eliminate "stick-slip" condition of machine tool slideways.

Part II

Description of the research apparatus

As no ready-made test equipment exist for the study of the effects of cross motion a new concept of experimental apparatus had to be developed.

Some researchers at a University were experimenting with textile fibre slip on cylindrical supports.

The fibre was applied over a cylinder under tension, then pulled horizontaly and the angle was observed when slip in the horizontal direction occured.

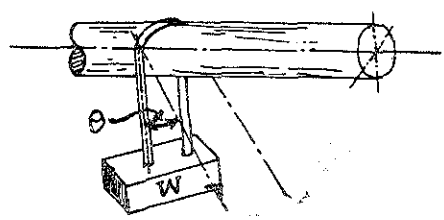

Fig. 5

Then the cylinder was rotated about it's axis and the experiment repeated. Slip occured at much lower angles–but results were not consistant as the fibre size was small and surface finish of the cylinder affected very much the results. Using a 2 inch diameter highly polished cylinder with copper wire was the next experiment. Here the results consistanty indicated that horizontal slip occur on the rotating cylinder at angles lower than on the stationary one. However, problems were experienced with the metal combination, as the annealed copper wire exhibited wear and the cylindrical surface was soon contaminated with copper.

The next experiment was with brass foil stretched over the cylinder. Though the wear problem was eliminated, no definite relationship could be established between the angle of slip and the rotational speed of the cylinder. For a very large span of rotalional speeds form a few rev/min. to several hundreds of rev/min. had little effects. At higher speeds the temperature increased appreciably and results became very unpredictable. Unpredictability also was true at very low cylinder speeds as well. The next apparatus was designed to eliminate the exponential contact pressure between the foil and cylinder. The foil tension over the cylinder varies in accordance with the exponential $e^{\mu\alpha}$, where α is the angle of lap (in this case π radian) and μ is the coefficient of friction but which coefficient? This question was puzzling and answers to date are not quite satisfactory.

The next apparatus featured a V-block sitting on the top of the shaft, which was rotating, as shown in **Fig. 6.**

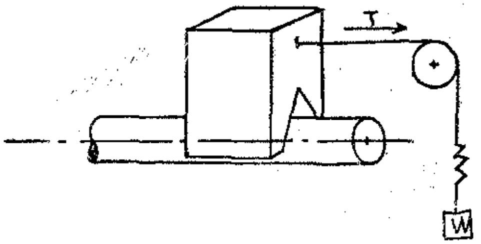

Fig. 6

A weight was attached to a string though a pulley, thus creating some tension, T, in the axial plane. The V-block did not move. The shaft was started, the speed increased continously until the V-block began to slide axially. The results indicated clearly that by the rotation of the cylinder an axial slip occured. The V-block often started to topple in the direction of the rotation before it began to slide axially. To balance it a long arm with adjustable weight and mechanical limiting was used, as shown in **Fig. 7.**

In spite of the balancing, at low rotational speeds, *at the commencement of the axial slide an interesting* oscillatory movement was noticed on the balancing arm.

To record this oscillation a mirror was attached to the side of the block and a lightbeam was reflected by this mirror and was observed on a screen. After

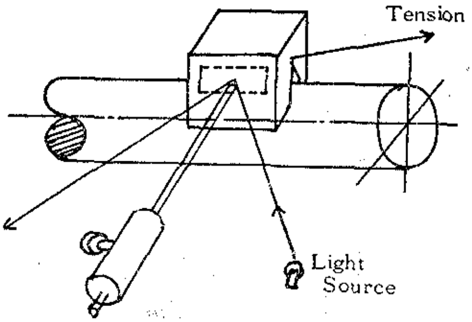

Fig. 7

several hours of running the experiment, the oscillation was reduced in magnitude, increased in frequency and developed a quasi-periodic pattern.

On examination of the V-block, the line of contact has been "run-in", so to speak, showing some sign of wear and a "smoother" surface than the rest of the V section. About 1/32 inch was milled off the V face and the experiment was repeated. The same general pattern was observed, however the results were so scattered that no firm conclusion could be reached.

A "dead load" was applied on the top of the V-block and the procedure was repeated. In general, the tension, T, necessary to produce axial slip varied with the vertical load and it was only slightly affected by the rotational speed of the shaft, provided that it was above a few rev/min. and below 200 rev/min.

An estimated relationship between vertical load and tension required to slide and rotational speed is shown in the graph in Fig. 8 below.

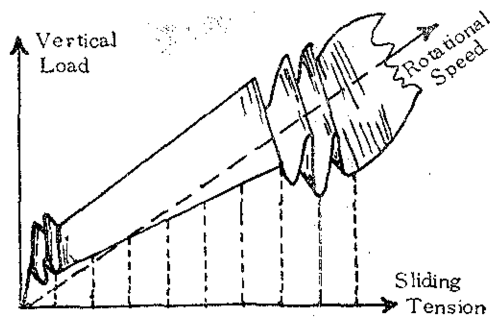

Fig. 8

At low speeds, the results are scattered and indicate a "transient" situation. At high speeds there appear to be also a "transient region", however, here the temperature component seem to have an influencing factor.

In order to study further the effects of temperature and the effects of surface wear on the sticktion coefficient, the experimental apparatus was completely redesigned.

The main frame was of heavy channel iron, "sandwitched" with polyester layers, which are good vibration absorbers. As the source of the oscillations at the onset of the slide might have come from the drive mechanism; in the new apparatus every effort was made to eliminate or to damp out any external source of vibration.

A large, 18 inch diameter steel drum was fine turned, hardened, and ground by J 5 grade wheel. The surface was inspected and mapped, using a portable TALYSURF, using a CLA micro-inch scale. (See Fig. 9)

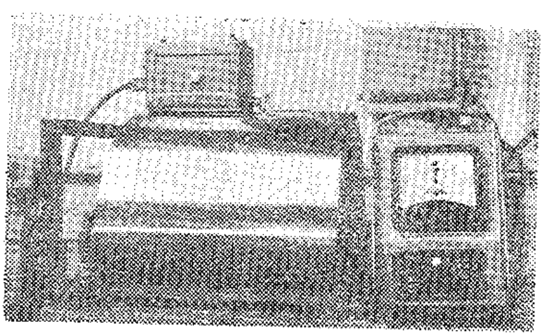

Fig. 9

In order to avoid the transmission of any vibration from the frame, the cylindrical drum assembly was held in two externally pressurised air bearings, using two jet rings at the quater planes with 8 radially equi-spaced 0.012 inch diameter jets. The radial nominal clearance was 0.0015 inch and the air supply at 60 psig was filtered by on 8 micron fabric filtering element. Considering the low rotational speed, a large bearing area of 5 in^2 projected section was used to support the 64.4 lbf dead weight drum.

The driving torque was transmitted via a "Constant torque" device, using two pairs of pegs with sleeves and an endless belt in a number 8 configuration. This was replaced later by a more positive methods, using two nylon rods as coupling pegs between the plates. This method was found very satisfactory.

The driving unit was a TASC No. 2 infinitely variable rotating magnetic field unit, geared 1:15 down. Constant and stable rotational speeds of 1/4 rev/min could easly be achieved. Without the reduction gear box and with special sets of pulleys, the high speed range was several hundred rev/min. The transmission from the prine mover unit was V-belt and pulleys, using a 2:3 or 1:1 ratio. The general arrangement is shown in **Fig. 10**.

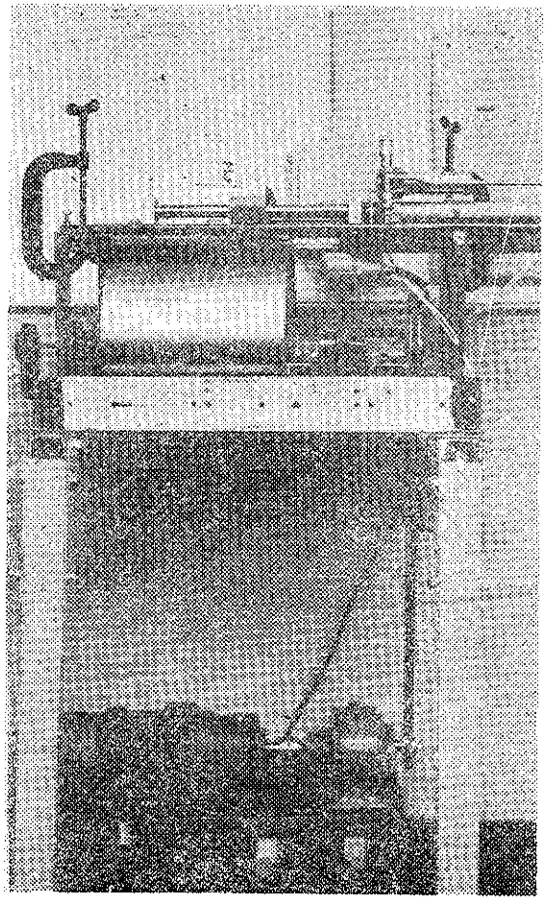

Fig. 10

The V-block was replaced by a special slider, or jockey, in the form of an inverted u bridge and a 1/2 inch wide metal foil stretched across the span, as shown in **Fig. 11A**.

The jockey was held by 2 unbonded strain gauge type force transducers (ETHER UFI TYPE DYNAMOMETER) for the measurement of X and Y components.

To avoid X-Y interaction, the connection force transmission element contained two natural hinges, formed out of 0.018 inch diameter spring-steel wire, bonded to the transmission rod. (see **Fig. 11 B**)

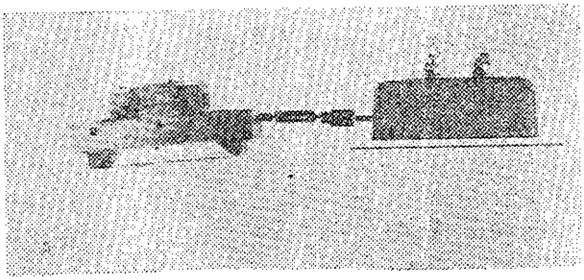

Fig. 11 A

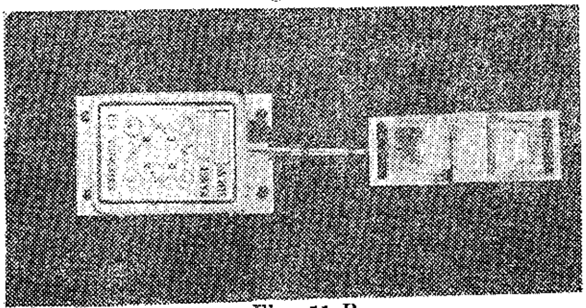

Fig. 11 B

The vertical load (Z component) was applied via a spring steel canti-lever with screw adjustment, using a 3 inch long vertical push rod, thus reducing the effects of any small misalignment, as the angle over the 3 inch long push rod yielded a cosine value about 0.9987 (see **Fig. 12**) the applied force was measured by a high gauge factor (200) ceramic type strain gauge, bonded to the spring steel canti-lever. Later, in order to double the output, a similar but negative output strain gauge was attached to the upper side of the spring-steel canti-lever. The combination was calibrated using a simple load-cell, replacing the jockey.

The jockey together with all the associated loading and instrumentation attachment was secured to a carriage assembly. This was guided by two parallel cylindrical guides and 3 recirculating ball-bushes. The cylindrical guides were mounted on a rigid 5/8 inch thick plate via compressed rubber bushes as not to give source of any vibration.

The linear prime mover in the axial direction was a $1\frac{3}{4}$ inch bore double acting phneumatic cylinder of 12 inch stroke, operating on 80 psig. air supply. To get a good speed control, a needle valve was used on the exhaust ports of the 5 part, 3 way control valve.

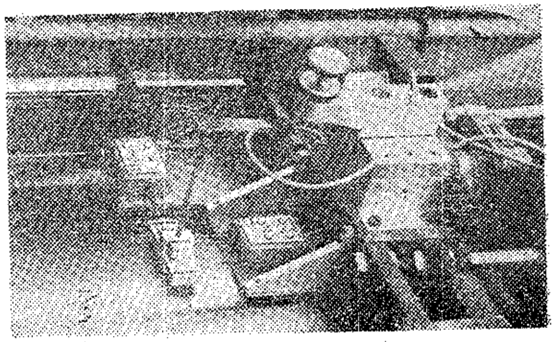

Fig. 12

To achieve axial speeds of 1/8 inch per minute, or lower a hydraulic check unit was opposing the motion of the phenumatic cylinder. The control action had a micrometer type needle valve. The temperature of the hydraulic check unit had to be kept constant to maintain a uniform oil viscosity. The check-unit was piloted by air valves, having "skip", "fast for ward", "fast return" actions thus facilitating initial warm up of the hydraulic oil.

The axial displacement was measured by a carbon type potentiometric linear transducer. An electronic differentiating circuit gave the speed values. Further, to have reference marking on the recording and not to have to rely on the "paper speed", a coarse pitch screw of 10 TPI and a micro-switch with roller follower as shown in **Fig. 13** gave a displacement mark at every 0.1 inch.

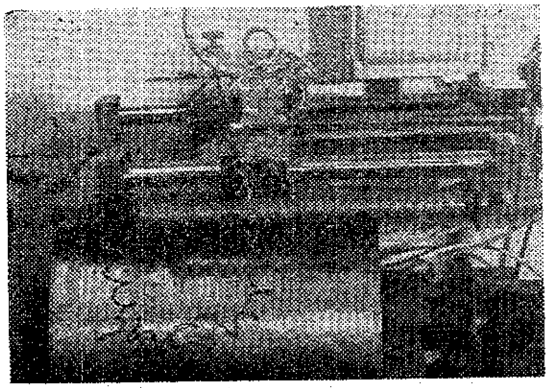

Fig. 13

The rotational speed of the drum was measured by a continuously rotating 355° active potentiometer via a differentiating electronic circuit. Again, as a double check, every 45° of rotation a proximity device gave intermittent marks. The accuracy of the method was confirmed by a strobo-flash applied at the reduction-gear-box input.

The plate with the carriage assembly and guides was aligned relative to the drum by using dial gauges, as shown in Fig. 14.

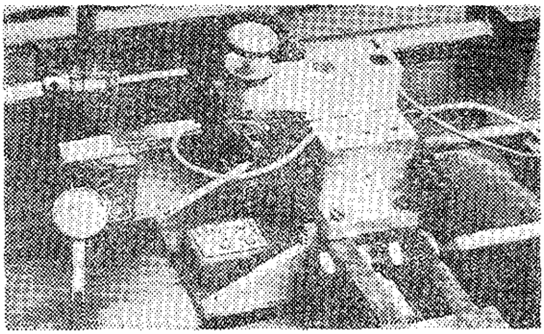

Fig. 14

All controls were housed in a purpose built console, using modular construction units.

The recordings were made using a 6 bank Ultra-Violet recorder and sub-miniature galvanometers of 25 cm optical arm. The strain gauges were exited by a stabilized D. C. supply and outputs were amplified by charge amplifiers. Speeds were recorded on the same chart. The intermittent check by microswitch (axial movement) and proximity device (rotational) were

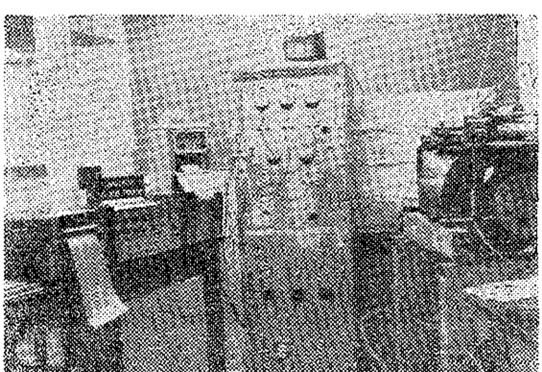

Fig. 15

recorded on a fast speed pen recorder (EVERSHED).

The general arrangement of the research apparatus is shown in Fig. 15.

Results & Conclusions

From the recorded observations and calculated results the following conclusions can be drawn:-

a) Under boundary type lubrication there is a sticktion between the two surfaces, at rest.

b) The sticktion coefficient is the limiting coefficient of friction; starting relative movement, depends on the load (contact stress) and on the general texture of the mating surfaces. The load is carried by the mutual metallic contact areas of the asperities; these contact areas depend on the vertical load. On rough surfaces there are fewer but larger contact areas; on smooth surfaces there are more, but smaller contacts areas. These contact areas are subject to stresses higher than yield stress; thus plastic deformation takes place which leads to "cold welding". On a smoother surface there is a larger distribution of small cold welded areas, so as motion begins there are fewer welded areas to be sheared at any one instant. The following stick-slip motion will have higher frequency and lower amplitude. Conversely, on rough surfaces the number of cold welded joints is less, but at any instance the cold welded joint is large, leading to a lower frequency, higher amplitude stick-slip motion.

c) The boundary lubricant itself has two functions:

(i) partical load supporting on a "stick" period

(ii) act as lubricant on a "slip" period

For the purposes of (i) the giant molecule type (eg. molybdium disulphade) lubricants serve very well as a few molecular layers are comparable in size with the rugosities of the surface. In some cases suitable lubricants will eliminate the need for very expensive surface fininsh, such as lapping, honing or superfinishing

d) "Running in" is a process where the "higher peaks" of surface asperities are worn off, thus stick-slip is less pronaunced after a running in period.

e) The "lay" or the direction of machining marks and general machining direction is important. The relative motion between two surfaces will take place easier in the direction of "lay" than across it (assuming no separation on account of lubricating film).

Omni-directional lay is generally a better "slipping" surface than uni-directional one, owing to the haphazard distribution of asperities.

f) The introduction of a moderate cross (Transverse) motion will eliminate stick-slip phenomenon beyond the detectable levels, assuring boundary lubrication and moderately well machined surface.

g) A prominent application at this conclusion is the application of rotating cylindrical guides to the servo-positioned platforms. (eg. N/C machine tools stable platforms, etc.)

강연자 소개

Mr T. J. Takats, 40 years of age, finished High School and University in Central Europe and then moved to England.

After working with ITT for nearly 4 years as a designer, joined University of London as a full-time student and graduated with honoors in mechanical engineering.

Following some experience as a designer and production engineer, become a lecturer and a senior lecturer at the London Technical Colleges for 4 years. During this time undertook research for a Master of Philosophy. In 1969 became an Expert of ILO in South America. There he graduated in a post graduate course at the University of Javariana, Between 72 and 74 he has worked in Mid East with ILO.

He is a Chartered Engineer (UK), Member of the Institution of Mechanical Engineers, (UK), Member of the Federation of European Higher Technical Professional Association, Member of Society of Engineers and many other Engineering Institutions in Eugland.

Currently Mr. Takats is engaged here in Korea with UNESCO as a Technical Training Adviser at the Fine Instruments Center.