DOE/ER/10471-1

Illinois Institute of Technology, Chicago, Illinois 60616

STUDY OF THE INTERACTIONS BETWEEN FRICTION, WEAR AND SYSTEM RIGIDITY

PROGRESS REPORT

FOR PERIOD JULY 1, 1979 - JUNE 30, 1980

V. ARONOV, A. F. D'SOUZA, S. KALPAKJIAN AND
I. SHAREEF

MARCH 1980

PREPARED FOR
THE U. S. DEPARTMENT OF ENERGY
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ABSTRACT

Progress to date in this project has consisted of the following activities. After an extensive study of various possible systems for friction, wear and vibration measurements, a pin and disk sliding system has been designed to be used on a rigid lathe bed. This versatile design has the capability of controlling the applied load, rigidity and damping of the total frictional system. The design and construction of the pin holding assembly has been completed with certain features to render it suitable for acquisition of appropriate data such as forces and displacements.

Special instrumentation has been obtained the major components of which are a tri-axial quartz piezoelectric force transducer, a tri-axial ceramic piezoelectric accelerometer for measurements of vibrations of the slider, charge preamplifiers with DC power supply, and monitoring equipment such as a spectral analyzer and an oscillograph.

Preliminary experiments indicate that the system, as designed and constructed, is appropriate for the type of study undertaken in this project. Some preliminary experimental results are included here.

The method of describing functions and harmonic balance is being employed for the study of friction induced self-excited vibrations. Some new developments of this method have been obtained to take into account the coupling between the degrees of freedom in the normal and frictional directions.

I. INTRODUCTION

The subject of friction and wear between sliding solid surfaces has been studied extensively for many years. Various theories have been proposed concerning mechanisms of friction and wear, and numerous attempts have been made to establish quantitative relationships between the many parameters involved.

A review of the extensive literature on this subject indicates clearly that the major factors involved in determining and controlling friction and wear are: the nature of the sliding materials, their surface condition, roughness, etc.; parameters such as load, speed, temperature, environment, and the presence of liquid or solid lubricants. A large amount of experimental data exists concerning the effect of these variables on friction coefficient and wear rate, although reproducibility of data has always been a problem due largely to the sensitivity of these parameters to the great number of variables involved [1].

In spite of the extensive data obtained on the effect of factors such as load, speed, surface roughness, and lubrication on friction and wear, there appears to be very little discussion in the open literature on the effect of system rigidity, i.e. dynamic characteristics of the test facility, equipment, machine, etc., on friction and wear. The only subject that has been studied to some extent is the stick-slip phenomenon where it has been recognized that this behavior is due to the fact that the friction force between two sliding surfaces is not a constant.

It is important to point out that stick-slip occurs at generally low speeds. Furthermore, interest in the subject does not appear to have extended beyond observations of the friction force and amplitude of oscillations of the bodies involved. A careful search of the literature has indicated that no studies exist on the effect of this phenomenon on wear, although it is recognized, naturally, that this can lead to not only objectionable noise levels but also it could result in surface damage and eventual failure of machine components.

It is further noted that discussions and experimental work to date on friction induced vibration involves generally a single degree of freedom only, in that the important parameters of spring stiffness and damping is taken along the direction of sliding velocity only. As pointed out by Tolstoi [2] it is only logical to expect that dynamic characteristics of the system normal to the sliding velocity also could play an important role, both on friction force and also on wear and vibrations.

It has now been clearly established that, even within the same system, more than one mechanism of wear can be operative as a result of a change in only one parameter. The traditional models of adhesive, abrasive, oxidation, and fatigue wear are commonly observed; yet the conditions under which transitions take place from one mechanism to another are still difficult to establish. The well known observation that even small variations in environmental conditions may have profound influence on wear rate is sufficient proof of the complexity of the subject.

The effect of the dynamic characteristics of the system on the mechanisms of wear and wear rate is a subject that requires careful study. It would appear that in studies on wear, it is not sufficient to vary only parameters such as load, speed, lubrication, etc. but also to vary parameters such as inertia, stiffness and damping in as many directions as practicable.

Accordingly, an interdiciplinary investigation has been undertaken combining the areas of friction, wear and dynamics. Specifically, this project has the following four objectives: 1) investigations of the effect of vibrations on frictional mechanisms; 2) model development of the vibratory structure, including contact stiffness; 3) analysis of self-excited vibrations caused by friction; and 4) investigation of wear processes including the interactions among friction, vibration and wear.

II. SUMMARY OF THE CURRENT STATE OF KNOWLEDGE

Although the nature of resistance to relative motion of two surfaces in contact is not completely understood at the present time, it is well known that friction is not entirely a steady state process. This frictional instability causes friction-induced vibrations of the mechanical systems which in turn influence the friction characteristics and the rate of wear of the materials of the surfaces in contact. The relevant technical literature in the subject was studied extensively as an initial part of this research project (2-30). For the sake of brevity the details of this survey will not be given here. The significant observations that have been made concerning the present state of knowledge is summarized below:

A) When a frictional contact is an element of a mechanical system, self-excited vibrations may be induced by different friction mechanisms, namely:
   a) Stick-slip
   b) Negative slope where the friction force is a decreasing function of the sliding velocity.
   c) Positive slope where the friction force is an increasing function of the sliding velocity. This mechanism is as yet unexplained and may be due to the coupling between the degrees of freedom in the tangential and normal directions.
   d) Random fluctuations in the friction force due to the mechanical and physical heterogeniety of the contact surfaces.

B) When self-excited vibrations are induced, the amplitude and frequency of vibrations depend on the nature and characteristics of the friction force and on the dynamic characteristics of both the structure and the frictional contact.

C) Any type of vibration causes a change in the magnitude and distribution of stresses and temperature on the real contact areas and, thus, affects the rate of wear as well as the friction force.

D) The dynamic characteristics of the frictional contact can vary depending on the temperature of the contact area and/or the mechanism of wear (which, in turn, depends on the characteristics of the system vibrations) causing a sharp transition from mild to severe wear and vice versa.

These observations show that there are complex interrelationships between friction, vibration and wear as shown schematically in Fig. 1. Most of the investigations carried out in the past are devoted to the study of vibrations in the tangential direction considering only the dependance of the friction force on the sliding velocity. The coupling between the degrees of freedom in the tangential and normal directions has not yet received its due attention.

The dynamic characteristics of both the structure and the frictional contact should play major roles in friction, wear and failure. However, to date the role of the contact stiffness has been neglected. A unified analytical method, based on energy considerations, is very useful for the investigation of conditions under which friction induces self-excited vibrations and also to determine their magnitude and frequency. This project is based on

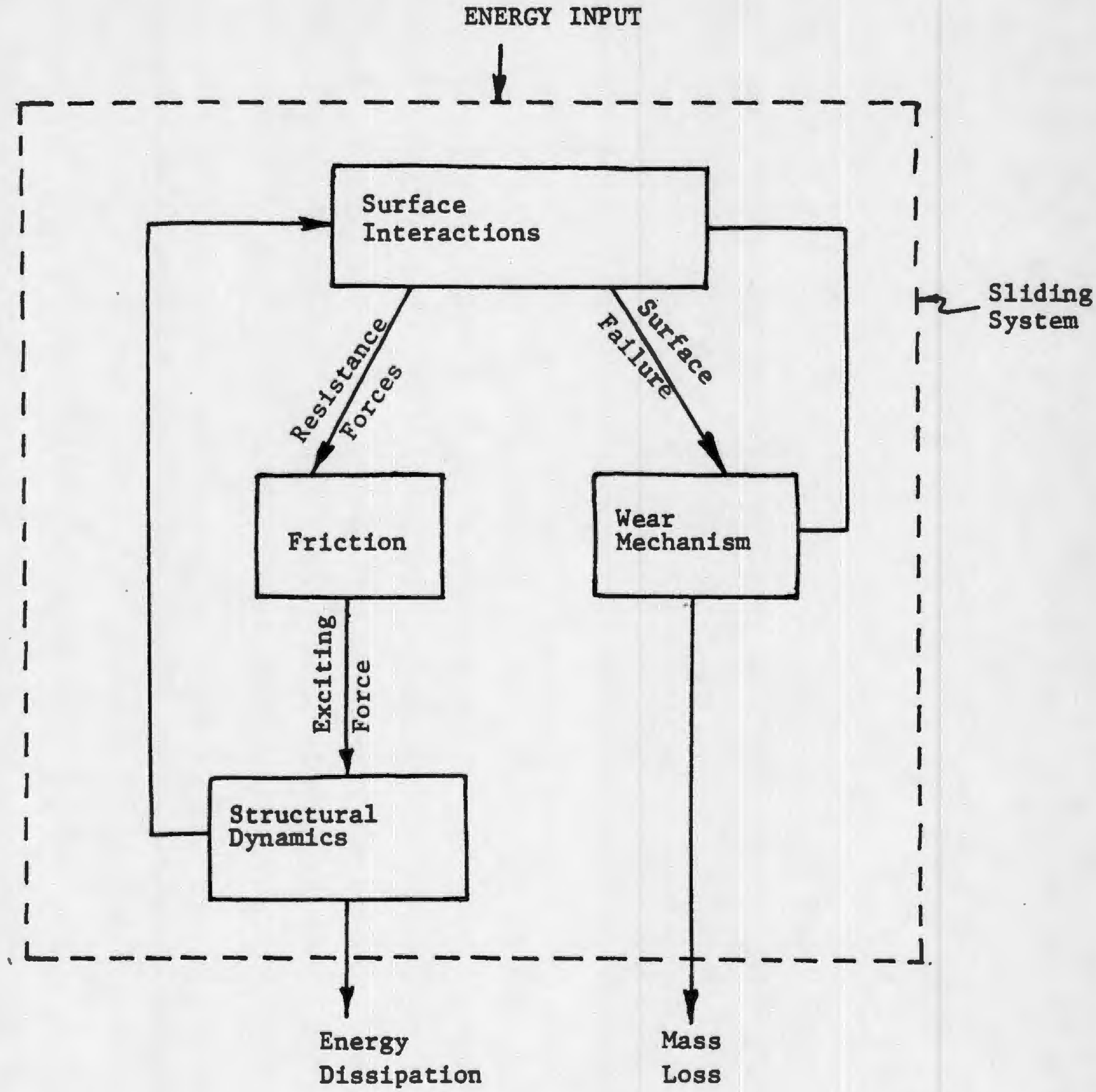


Fig. 1 Schematic of interactions between friction, vibrations and wear.

the very same ideas and goals which are being studied carefully in the light of the concept of describing function and its extensions. This concept, combined with investigation of the kinetics of the wear processes that is energy dependent, may contribute to our understanding of the friction and wear behavior of dynamic mechanical systems.

III. REVIEW OF DESIGN PRINCIPLES OF VARIOUS EQUIPMENT USED IN TRIBOLOGY RESEARCH

As it is evident from the review of the literature on tribology and as also outlined in Sec. II of this report, there are very complex interrelationships between friction, vibration and wear.

It is clear, therefore, that the dynamic characteristics of both the structure of the total system and of the frictional contact should play major roles in friction and wear. In view of the complexities involved in such dynamic characteristics of a tribological system, a great deal of effort was expended in the design of the equipment to be used in this present study. Furthermore, it was felt that a comprehensive study and evaluation should be made of all the various apparatus or equipment described in the available literature on the subject.

A great variety of friction and wear test devices have been developed over the past few decades, with a variety of characteristics (31). A majority of these designs are for somewhat routine testing of materials in sliding contact at ambient temperature and environment. More fundamental studies in the subject have necessitated the use of more sophisticated and instrumented apparatus as described below.

For a considerable period of time the most common friction and wear measurement apparatus has consisted of a pin and disk system. Friction force has been measured through the deflection of the arm supporting the pin. It is obvious that in order to be able to measure small magnitudes of friction force, the arm must not be too rigid. However, this in turn introduces a lack of stiffness in the tangential direction. The stiffness normal to the sliding surface is, in a sense, simply the weight of the arm. Dynamically, however, this mass is very significant because of the inertia effect when asperity movement is taken into account.

Such a system is, of course, useful in studying stick-slip phenomena in one direction, i.e. tangential, only where displacements can be measured by a transducer, as shown in Fig. 2 used by Brockley and Davis (32). Their apparatus was designed to study the time-dependence of static friction. It consists of a variable speed turntable (disk), a cantilever beam to provide both normal load on the specimen and spring force, and a displacement transducer to determine the friction force. The sliding speeds could range between 0.001 and 1 in/sec (0.025 and 25 mm/sec). The length of the cantilever beam was adjustable and thus the spring constant of the system could be varied. The normal load application, as shown in the figure, kept the vibrating mass constant regardless of the load.

Brockley et al. (5) used a similar system in another study on friction induced vibration, with the exception that the sliding was linear and damping of the slider was provided for by permanent magnets, Fig. 3. The driven surface was 6 ft. (183 cm) long with speeds ranging from 0 to 9 in/sec (0 to 230 mm/sec) with great rigidity and large mass. Different damping ratios were obtained by reducing the flux gap and tests revealed that damping was essen-

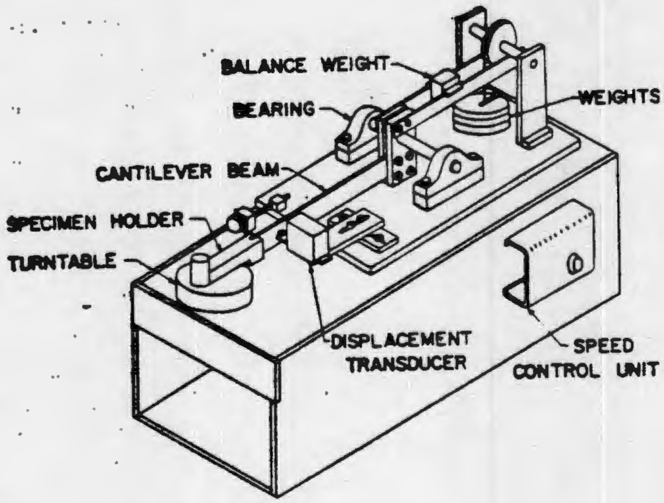


Fig. 2 Diagram of the apparatus used by Brockley and Davis (32).

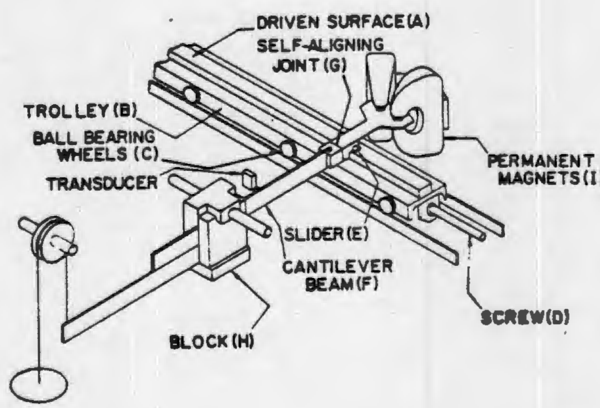


Fig. 3 Diagram of the apparatus used by Brockley, Cameron and Potter (5).

tially viscous in nature. The stiffness of the slider assembly could be altered by changing the length of the cantilever beam.

In further studies on friction-induced vibration by Ko and Brockley (33), special attention was paid to careful isolation of vibrations arising from the test apparatus or the surroundings. They also used accelerometers, velocity transducers and strain gauges for measurement of various quantities in the vibrating system, Fig. 4. One feature of this system was that the apparatus was placed on a massive table resting on a rigid foundation. Note also, in the insert to the figure, the design of the slider support mechanism which assures uniform contact with the rotating disk.

Even though the designs discussed above are among the most advanced, it will be noted that only the rigidity in the tangential (sliding) direction that can be changed. There appears to be no attempt to make provisions to make the stiffness normal to the sliding surface another variable in the system. In these apparatus, "resistance" to any transient vertical, i.e. normal to sliding surface, displacement is obtained through the inertia effect of the equivalent mass of the slider arm. It can easily be visualized, of course, that in addition to this mass there can also be a stiffness incorporated in the system.

Although not a part of their study, Swinnerton and Turner (34) used an apparatus which involved a compression spring between the slider and the point of load application, Fig. 5. This system was used to simulate sliding electrical contacts used in practice. It

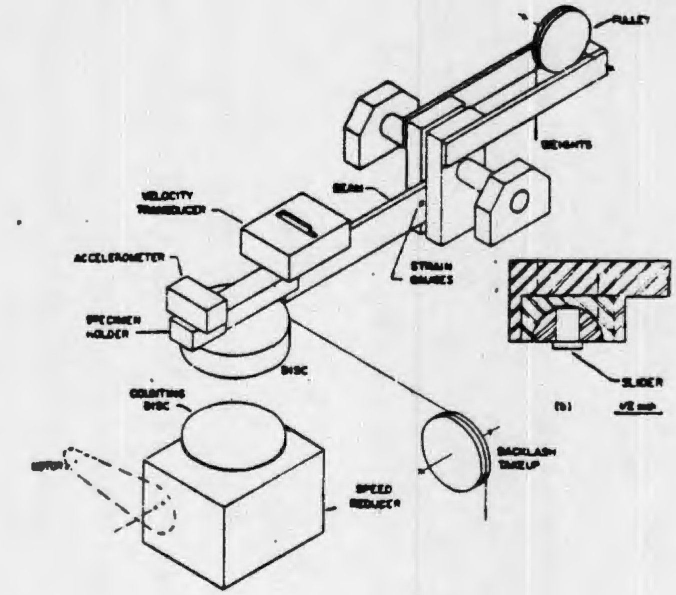


Fig. 4 Schematic diagram of (a) the apparatus and (b) self-aligning slider mount used by Ko and Brockley (33).

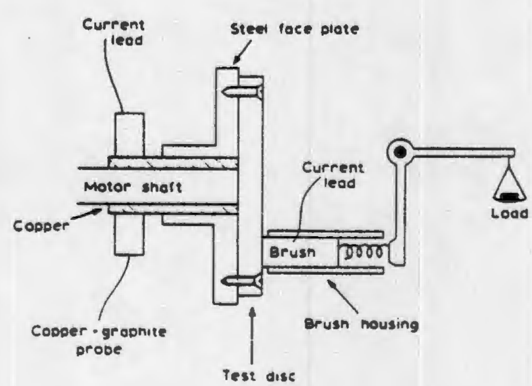
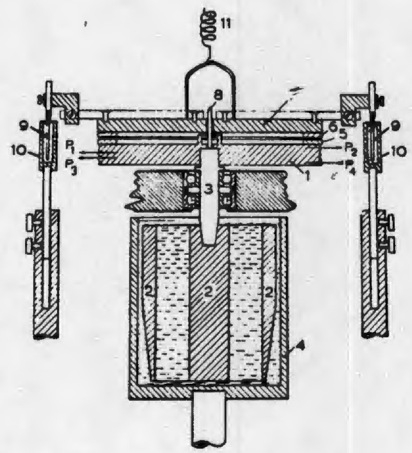


Fig. 5 Experimental apparatus used by Swinnerton and Turner (34).

is obvious, of course, that even though the load on the slider may be the same, this load could be transmitted through a spring with different spring constants, i.e. stiffnesses. Thus, the masses would remain the same but the stiffness could be changed over a wide range. The authors do not comment on the role of the stiffness of this spring.

Tolstoi (2) has pointed out the significance of the normal degree of freedom and natural normal vibrations in contact friction. Although conducted at very low sliding velocities of the order of 10^{-9} cm/sec, he showed that both negative friction-velocity slope and frictional self-excited vibrations are closely associated with the freedom of normal displacements of the slider. Whenever the latter is absent, self-excited vibrations disappear. Direct interferometric observations showed that, in self-excited vibrations, the tangential "darts" of the slider are invariably accompanied by simultaneous upward, i.e. normal, jumps.

His apparatus, Fig. 6(a), involves a slider (6) in the shape of a circular ring rotating over a disk (5) whose angular velocity was held steady by the damping system below the disk. The normal displacements of the slider were determined interferometrically by the device shown in Fig. 6(b). Damping of the slider in the normal direction is accomplished by the damping device (9,10) where bent blades are immersed in a viscous oil, thus causing damping in the vertical direction. Damping can be removed by simply sliding downward the containers (10) filled with the oil.



(a)

(b)

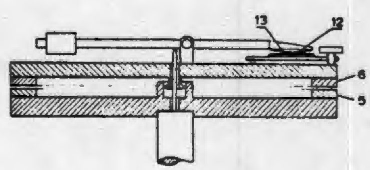


Fig. 6 Low-speed sliding friction apparatus (a) used by Tolstoi (2) and the interferometric device (b) for measuring vertical slider displacements.

IV. MECHANICAL DESIGN OF TEST FACILITY FOR PRESENT STUDY

A. Selection of Frictional Contact Geometry.

Seven different types of frictional contact geometry were considered for this project, Fig. 7 (a to g):

a) Line contact formed between two rollers rotated at different angular velocities.

b) Contact areas formed between a rotating roller and a brake shoe.

c) Line contact formed between rotating cylinder and a flat surface.

d) Point contact formed by two rotating cylinders placed orthogonally.

e) Flat area formed between rotating disk and a reciprocating pin.

f) Flat area formed between rotating disk and a stationary pin.

g) Flat area formed between rotating and stationary cups.

The following criteria were evaluated in order to select the proper design:

1. Ability to achieve constant apparent contact area.
2. Ability to achieve constant normal stress.
3. Simplest way to apply load.
4. Simplest way to control load, rigidity and damping.
5. Simplest mechanical design without loss of any essential features.

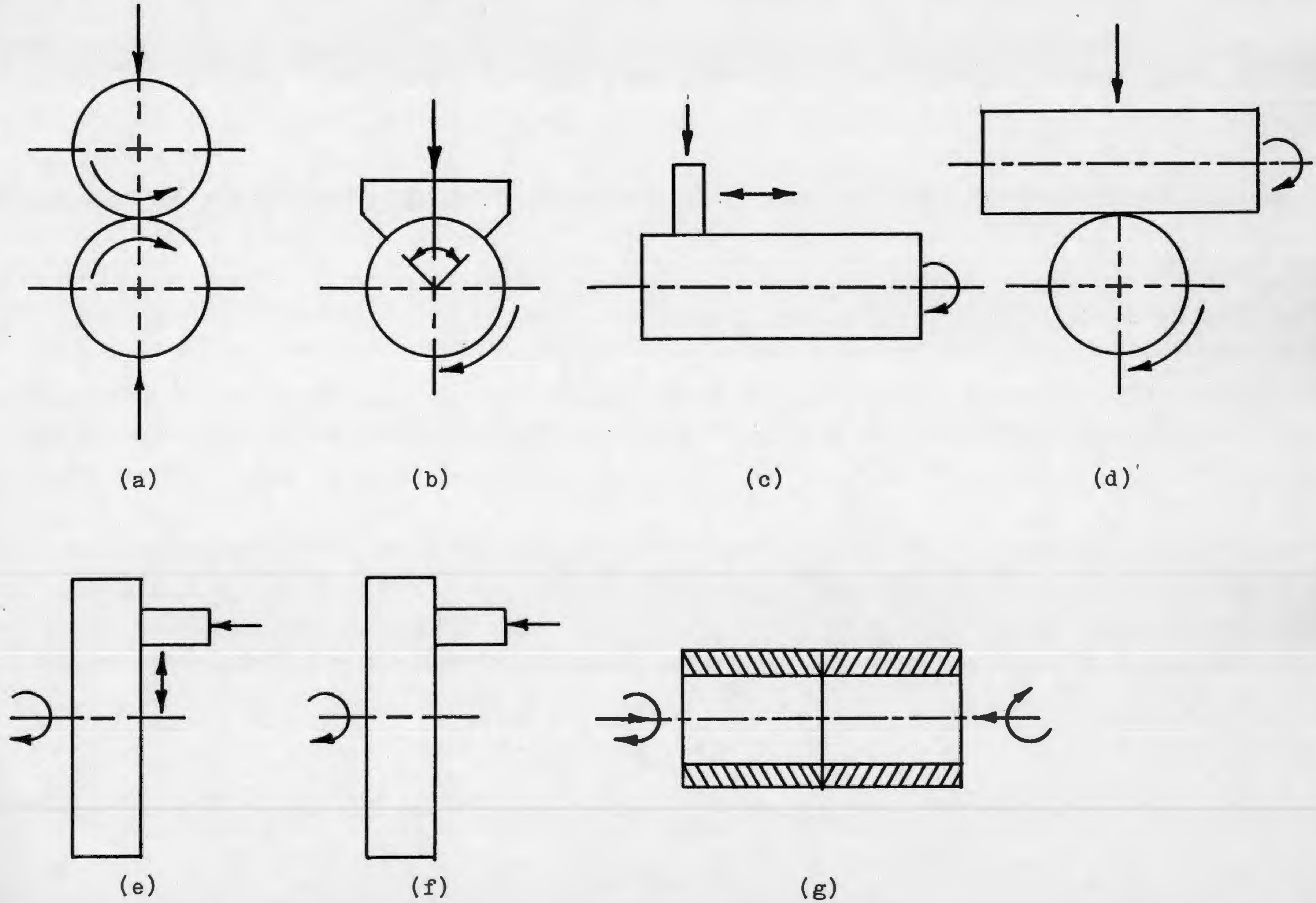


Fig. 7 Different types of frictional contact geometry.

It was decided to use a rotating disk and a stationary pin rubbing against it to obtain frictional contact. Various configuration of disk and pin contact were considered, as shown in Fig. 8 (a to e).

While evaluating the type of contact geometry, the possibility of using the spindle and bed of a lathe was checked. The lathe spindle was not vibrating from 0 to 1500 rpm when all gears from the gear box were taken apart. With the possibility of using the spindle and bed of a lathe as a part of the testing device and on the basis of the above mentioned criteria, the design shown in Fig. 9 was finally selected.

This particular design is the result of extensive discussions over a period of three months. Four different designs (not shown) varying in the degree of complexity were drafted. Out of these, the simplest was the one selected for fabrication. The design was made sufficiently versatile with respect to flexibility and ease in changing process parameters. For instance: a) The spring length could be changed from 3/4 to 6 inches (19 to 152 mm). b) The normal loads could be very easily and quickly changed. Linear bearings permit application of constant normal force without introducing inertia effects. c) The infinitely variable torque drive permits variation of sliding speed from zero to 240 inches/sec. d) Spring cross sections could be changed by just replacing the cup-spring assembly. e) Also the effect of damping can be studied by a change of damping fluids.

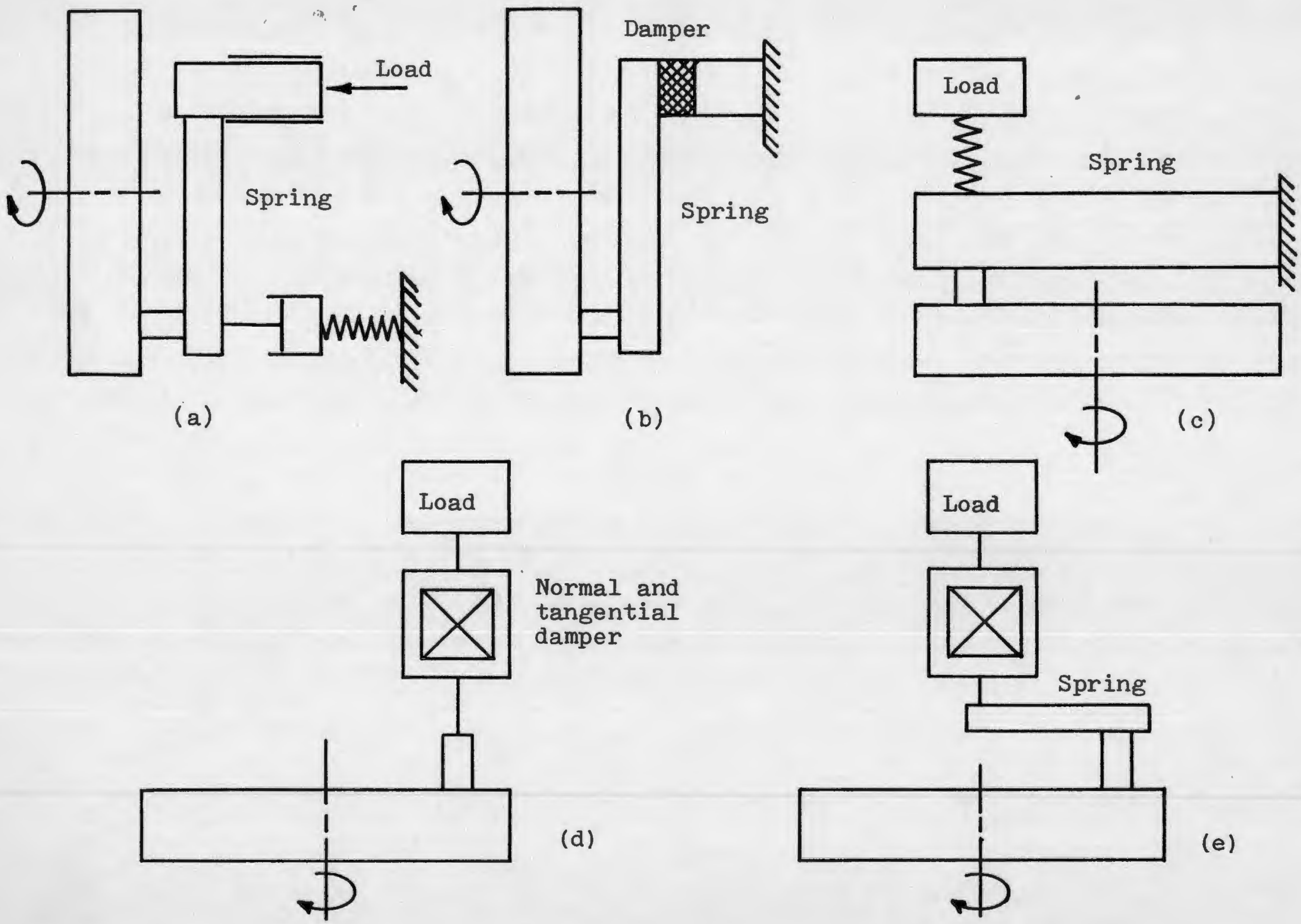


Fig. 8 Various configurations of apparatus design for sliding contact.

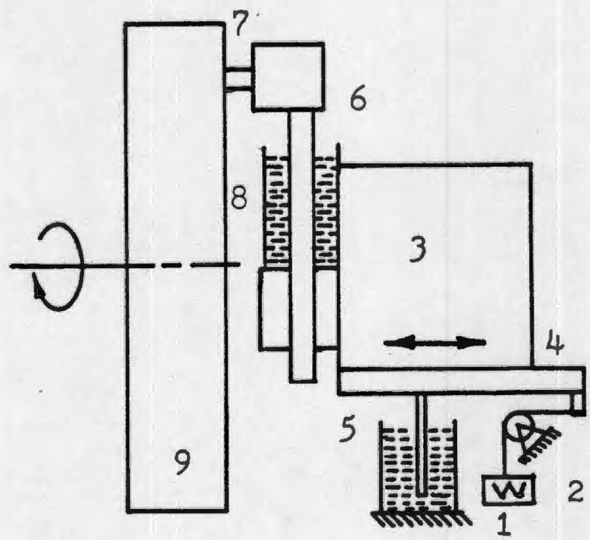


Fig. 9 Schematic of test apparatus for sliding friction and wear designed for the project.

In the early part of the discussions which eventually lead to the evolution of this design, it was arbitrarily fixed that a specimen of 5mm in diameter will be used as a sliding element. Since the pin is mounted on a spring whose deflection is dependent on the dead load applied through the linear bearings, it was observed that the contact area between pin and disk is not the same for different loads.

Since large variations in the apparent area of slider cannot be permitted, a change of 5% maximum in area of the pin was allowed. The max normal load to be used is 20 kg. Having fixed these values the cross-sectional area of the spring was calculated for the case of the most flexible configuration. That is maximum load, maximum deflection of spring and 5% change in contact area. This provided the minimum cross section of 0.625 in^2 (403 mm^2). The first cross section selected was square, since it provides the same spring constant in normal and tangential directions. The maximum cross section was then fixed at 1/2 x 1/2 in. square (13 x 13 mm) which permitted less than 1% change in area of contact for maximum load and maximum spring length.

After carefully lapping the pin and disk to a smooth finish (0.42 μm CLA) the preliminary experiments were started with 3 in. (76 mm) spring length and 1/2 inch (13 mm) square spring. Even with such a rigid configuration there was considerable chatter introduced. The spring length was then reduced to 1.25 inches (32 mm) which resulted in a smoother and more stabilized friction process. From these preliminary experiments it was inferred that springs with higher spring constants must be used. This will be done in forthcoming experiments.

B. Materials.

The materials were selected with the purpose of having a comparatively large range of mild wear in order to investigate the processes of mild wear, as well as the processes of transition to severe wear.

Two different materials were selected for the disk and slider. The material used for the disk is RY-ALLOY, an oil hardening AISI-SAE 01. The typical chemistry of this steel is 0.9% C, 0.5% Cr, 1.35% Mn, 0.5%w, 0.35% Si. An oil hardening steel was chosen for these experiments because of its property of combined high hardness, sufficient toughness and deep hardness penetration with minimum distortion, freedom from cracking and good wear resistance.

The heat treatment procedure adopted was to heat the steel between 1450 to 1500° F. The specimen was then soaked at this temperature for sufficient time to ensure uniform heating. Then it was soaked in an agitating pool of oil at 350° F. Finally, it was tempered at 400° F to obtain a through hardness of 59 Rockwell C.

Before the commencement of the experiment, the disk was ground and lapped. Talysurf measurements were then made and the surface roughness was found to be 0.42 μm CLA. A Talysurf record of surface is given in Fig. 10.

It was decided that a comparatively softer material than the disk be used for the slider. After careful scrutiny, 1045 special quality steel was selected for the slider. The typical chemistry of this material is 0.5% C, 0.85% Mn, 0.04% P,

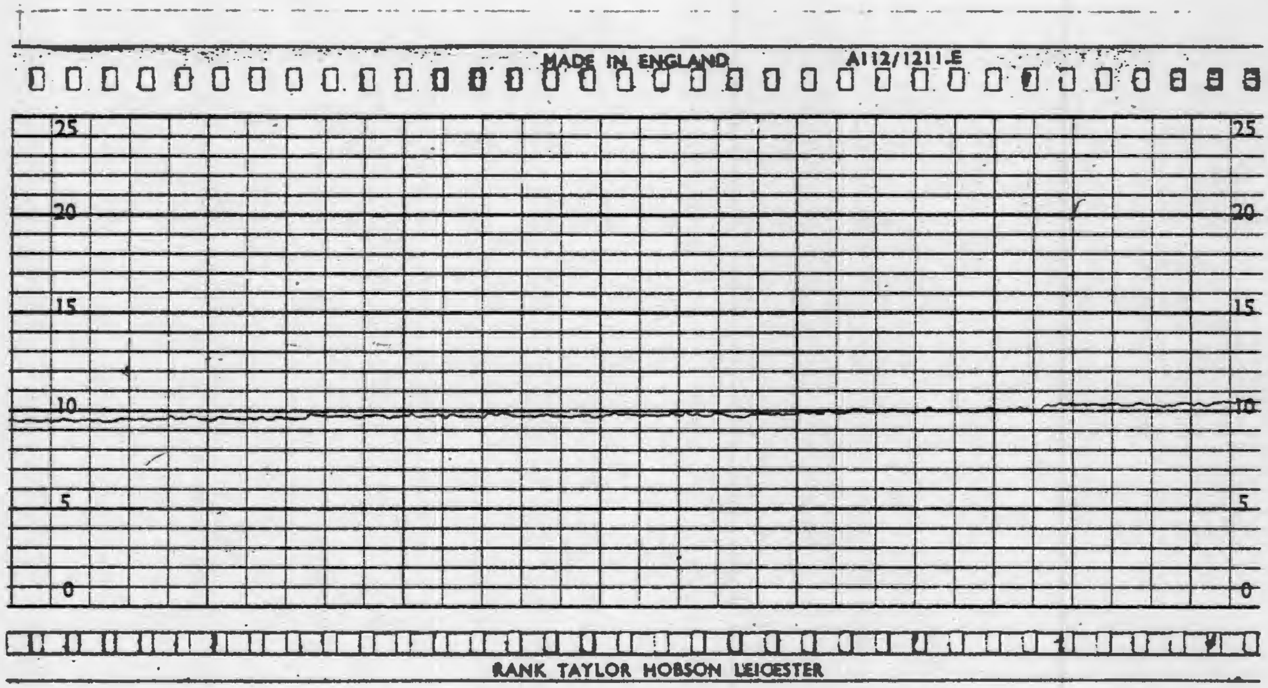


Fig. 10 Talysurf record of disk surface after lapping. Vertical: 400X, horizontal: 20X.

and 0.05 percent S. The pin material was analyzed for its composition by using optical and Scanning Electron Microscopes. A 1000x and 500x micrograph of the metal matrix is shown in Fig. 11.

The pin material is being used in the as rolled state with a hardness of 35 Rockwell C.

C. Sliding Speed and Load.

From the authors' past experience it is known that all possible modes of sliding friction and wear with selected materials can be achieved with sliding speed variation between 0 and 5 m/sec. The normal pressure is between 0 and 25 N/mm^2. To have this condition it was decided to make samples of 5 mm diameter and a disk of 203 mm diameter. In this case the angular velocity of the disk should range between 0 and 600 rpm.

D. Rigidity and Damping Control.

The most difficult problem encountered in the design of the test device was to prevent the transmission of friction induced vibration of the sample to the applied load. Of a few ideas that were considered the one selected was that shown in Fig. 9. This design consists of two systems: measuring and loading. In order to eliminate transmission of the sample vibration to the applied load, the natural frequency of the vibration of both systems should be far apart.

The loading system is comprised of the dead weight (1), wire rope system (2), massive block (3), two sliding ball bearings (4) mounted on the carriage of the lathe, and damper (5).

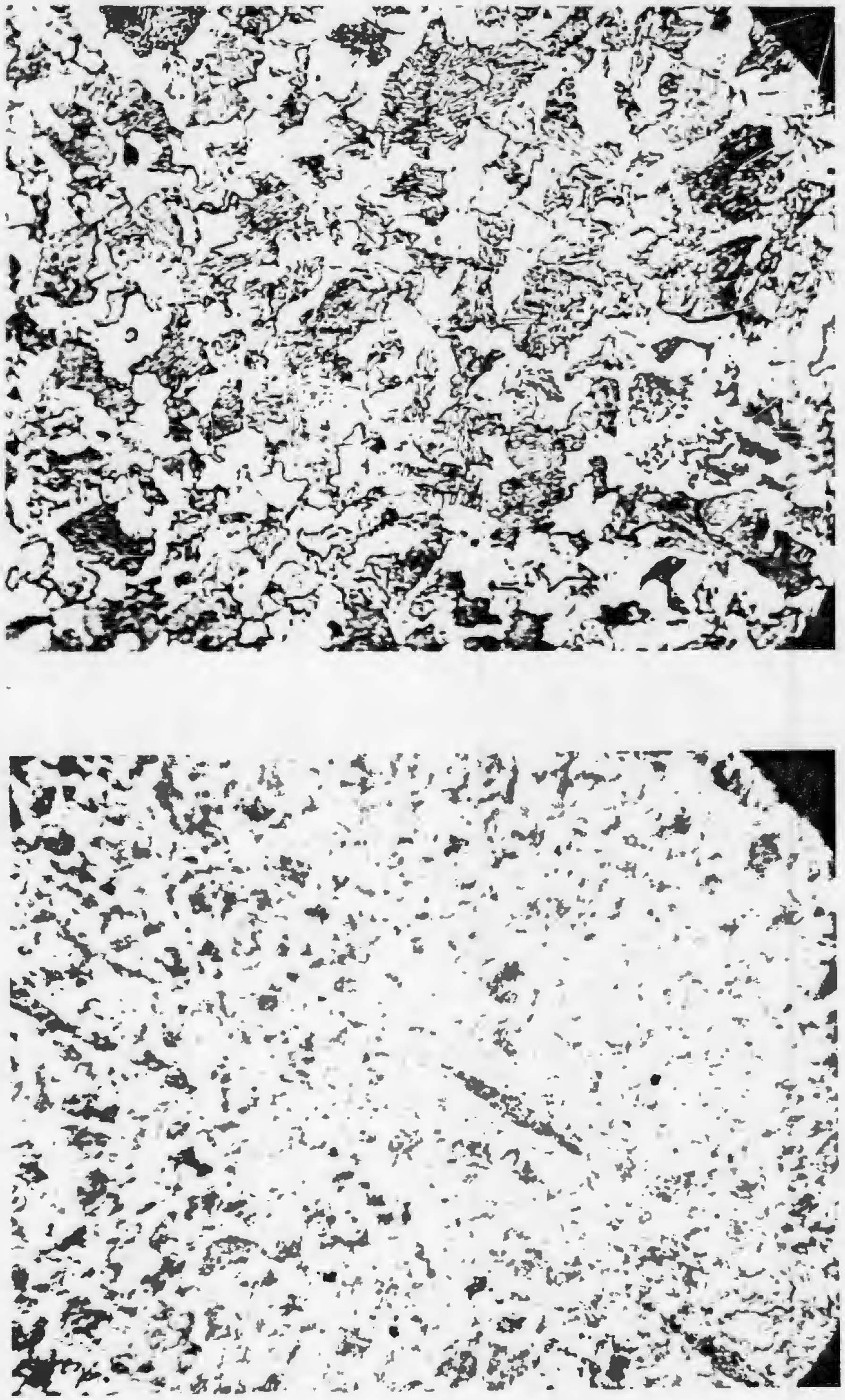


Fig. 11 Micrograph of pin material. Top: 1000X, bottom: 500X.

The measuring system contains a spring (6), attached to the beam (5), sample (7) and damper (8). As can be seen from Fig. 9 the weight (1) pushes block (3) and sample (7) against the rotating disk (9) attached to the spindle. The mass of the block (3) is two orders of magnitude larger than the total weight of spring (6) and sample (7). This causes a large resisting inertia force that, together with high damping characteristic (high oil viscosity) of damper (5), protects load (1) from vibration. In this case the load as applied to the spring (6) can be considered to remain constant. The vibration characteristics of the sample (7) can be varied by changing the spring constants and by damping of spring (6) and damper (8) in both normal and tangential to the friction surface directions.

E. Sample Holder and Spring.

The sample holder (Fig. 12) is manufactured together with a spring that has a rectangular shape. The stiffness of the spring can be changed by changing the dimensions of the rectangle cross-section, or by varying the length of the spring.

As shown in the figure, the holder contains the load cells, accelerometers, sample housing and nut. This design allows for the sample to be removed for weighing and placed back exactly in the same position.

F. Grinding System.

In order to achieve smooth operation, the rubbing surface of the disk should not oscillate with respect to the plane that is perpendicular to the axis of sliding bearings, at least within

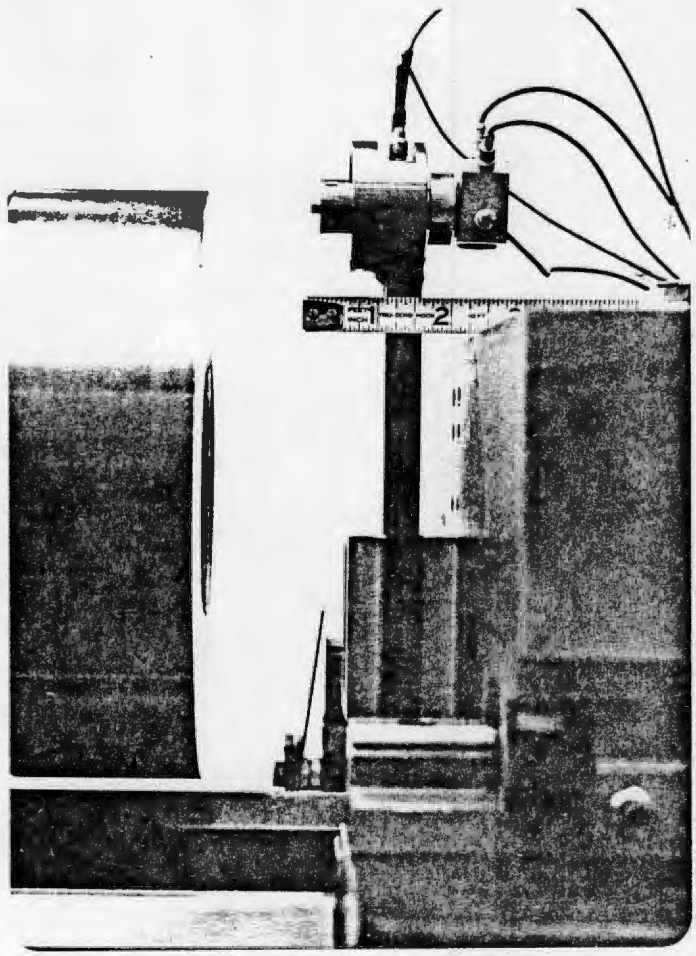


Fig. 12 Disk and pin attached to a cantilever beam, i.e., spring.

heights of asperities roughness of rubbing surfaces. In our case this oscillation should not exceed 2-3 μm. A special fixture for tool post grinder is designed so that disk can be ground directly on the lathe.

G. Power Drive.

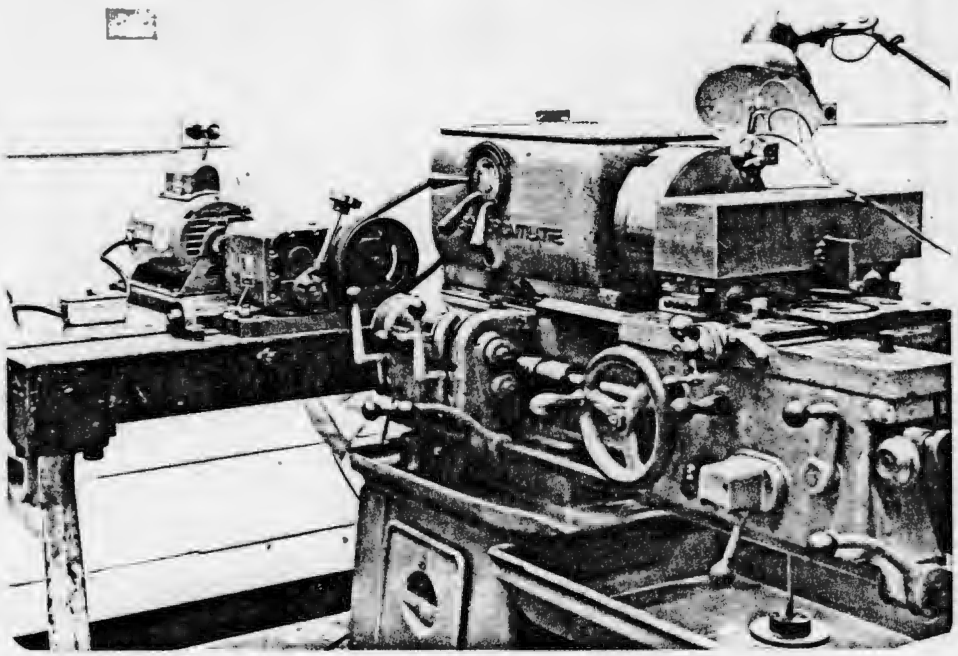
The power drive is installed separately from the bed of the lathe. This design was based on the following reasons: a) to vary the rotational speed of the disk the gears of the lathe head stock could not be used, because they cause vibration, b) the AC electric motor attached to the lathe base also would cause vibration.

The drive (Fig. 13) is mounted on a heavy table and consists of an 0.75 h.p. AC electric motor attached to a variable speed drive with a range from 0 to 400 rpm. In order to rotate the spindle at 600 rpm and to damp the vibration transmitted from the electric motor and variable speed drive, a v-belt drive was used with a speed ratio of 1.5 to 1.

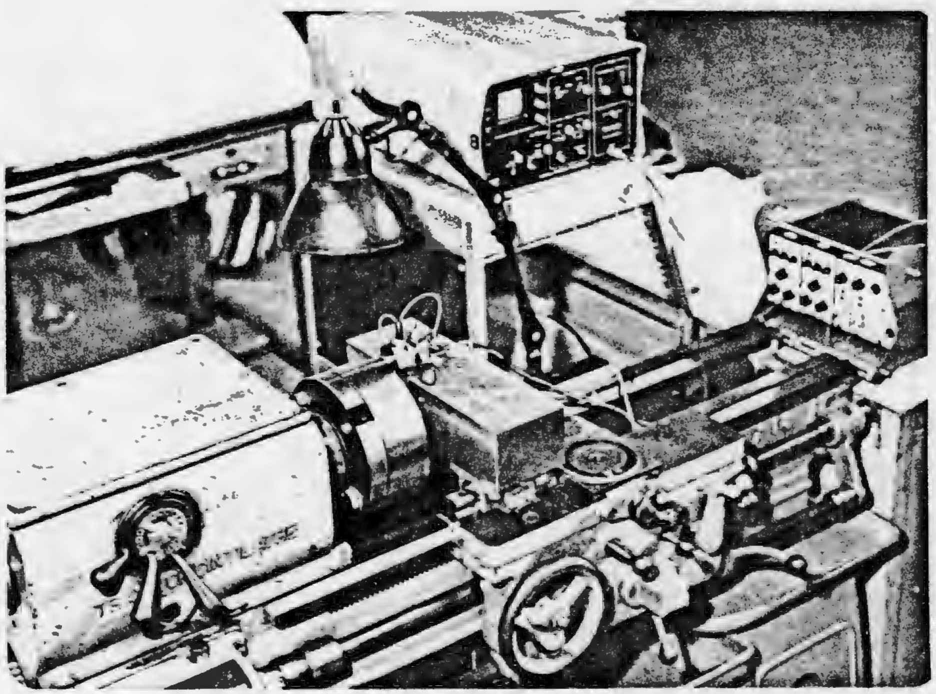
H. Preparation Technique.

The following technique is used for experiment preparation:

1. Grinding of the disk.
2. Lapping of the disk with abrasive paste and cast iron block.
3. Lapping of the disk and pin against each other using abrasive paste.
4. Thorough cleaning of the disk and sample with acetone.
5. Running in friction under a small load (0.5kg) until all the area of the pin is in full contact with the disk.



(a)



(b)

Fig. 13 General view of set-up showing various components.

V. INSTRUMENTATION AND MONITORING SYSTEM

A schematic diagram of the instrumentation, signal conditioning, recording and analyzing equipment for the forces and vibrations is shown in Fig. 14. The forces on the slider are measured by a tri-axial force transducer and the vibrations of the slider are monitored by a tri-axial accelerometer. The signals are conditioned by charge pre-amplifiers and analyzed by a spectral analyzer and, after further amplification, recorded on a four-channel oscillograph.

Force Transducer: A quartz piezoelectric force transducer, Kistler Type 9251, is employed for the measurement of the forces on the slider (Fig. 15). A force acting on the transducer can be decomposed into its three components in the x, y and z directions. It contains three quartz disk pairs, one pair being sensitive to compression in the z direction and two others to shear. As shown in Fig. 14, only two components, namely, the normal force in the z-direction and the frictional force in the x-direction, are measured simultaneously as only two charge preamplifiers are employed. In our case, there should be no force component in the y-direction i.e. the direction perpendicular to the plane of the normal and frictional forces. However, there is provision for monitoring this force but not simultaneously with the other measurements.

The resolution of the force transducer is 0.01 N (0.0022 lbs) and its resonant frequency in the normal direction is 8kHz. The charge sensitivities of the transducer are 3.87 pC/N in the z-direction, 8.07 pC/N in the x-direction, and 8.06 pC/N in the y direction. The cross influence is less than 5% and the linearity

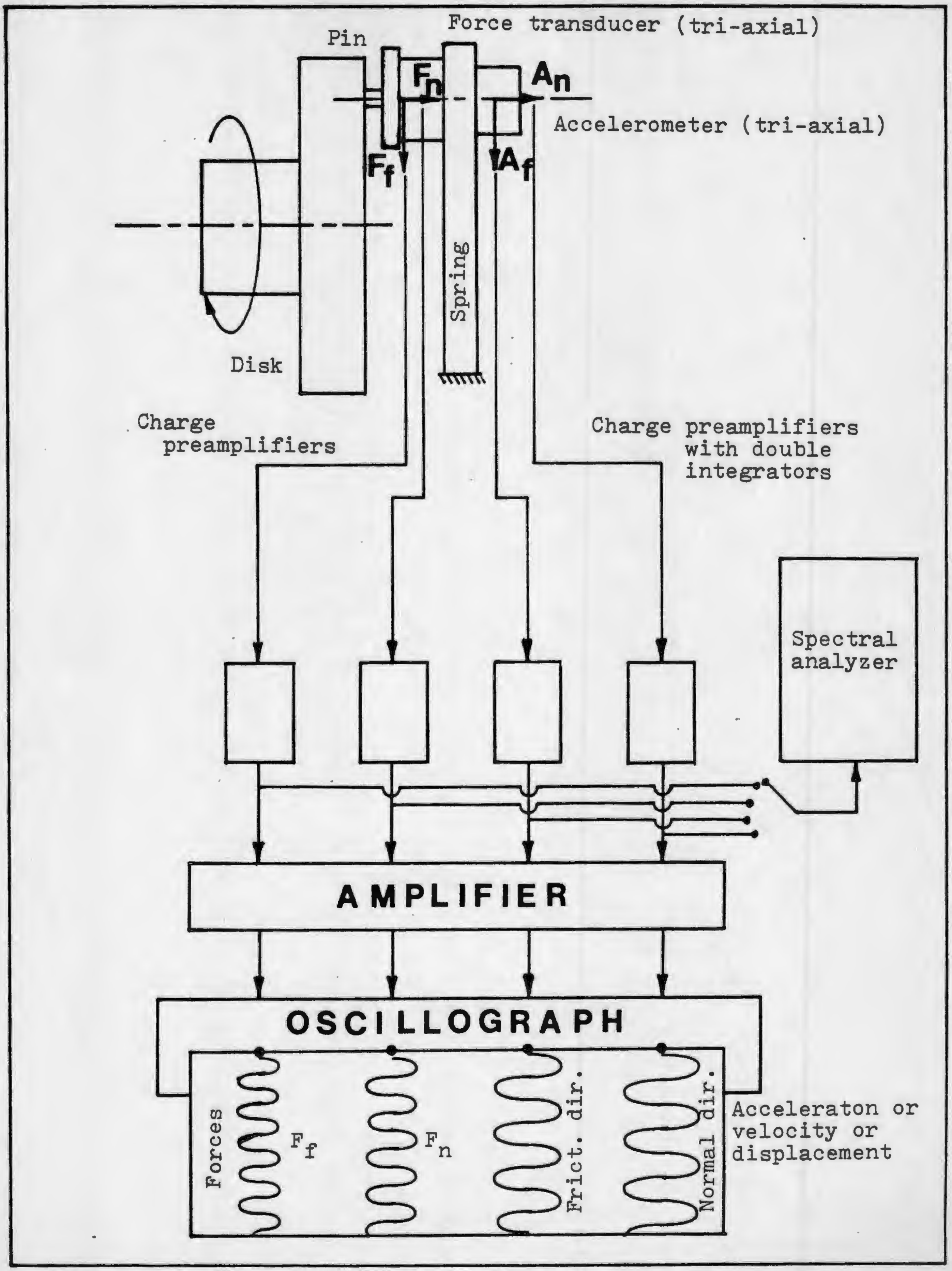


Fig. 14 Schematic diagram of the instrumentation, signal conditioning, recording and analyzing equipment.

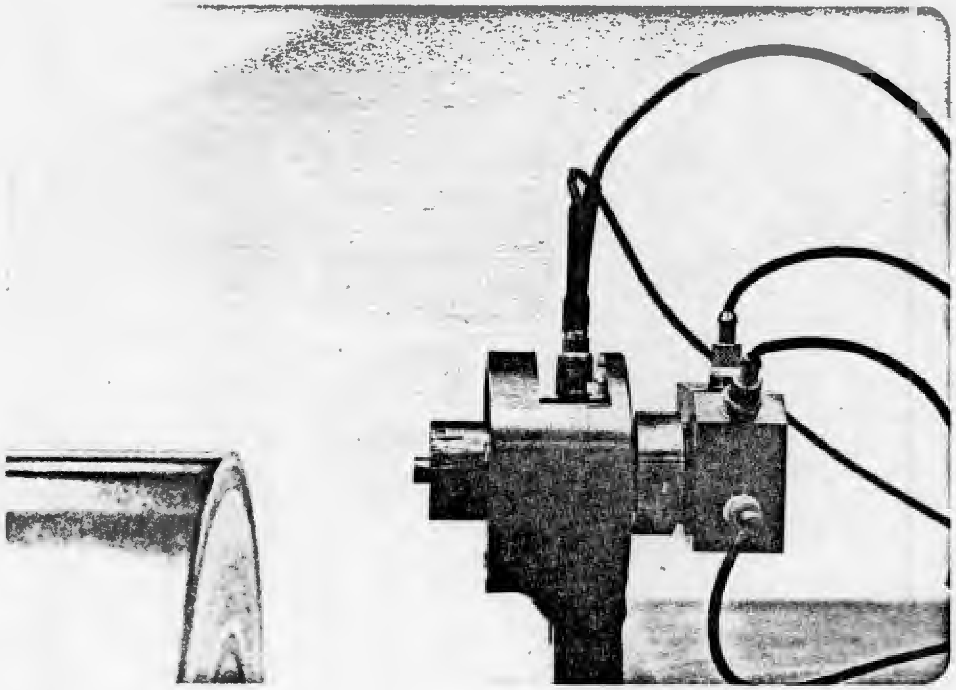


Fig. 15 View of slider with holder, tri-axial force transducer (left) and tri-axial accelerometer (right).

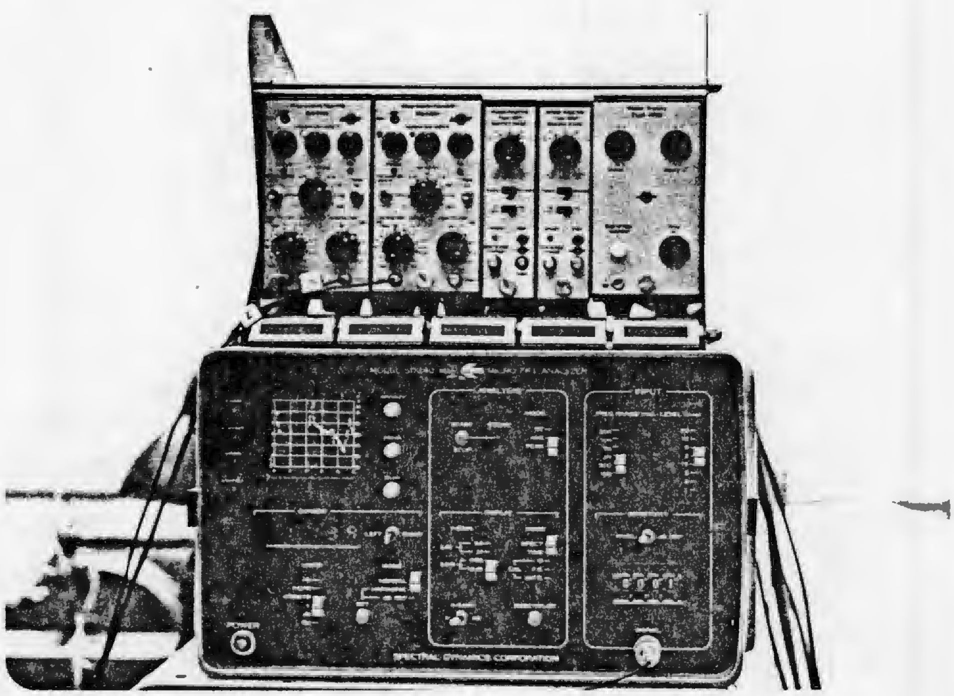


Fig. 16 View of charge preamplifiers (top) and spectral analyzer (bottom).

is within 1%. Due to the high rigidity of the transducer 1000 N/μm in the normal direction and 300 N/μm in the other two directions), its influence on the measuring arrangement is small. Quasistatic measurements and static calibration are possible.

Accelerometer: A ceramic Piezoelectric accelerometer, Bruel & Kjaer (B&K) Type 4321 is used for the measurement of vibrations of the slider, Fig. 15. It consists of three individual accelerometer elements with their principal axes mounted perpendicular to each other, so that it can detect vibrations in three mutually perpendicular directions. Each of the three transducers contained in Type 4321 are of delta shear construction where the piezoelectric slices are stressed in the shear mode when subjected to vibration. In our set-up, vibrations in the normal and frictional directions can also be measured but not simultaneously with the other measurements as only two charge preamplifiers are used for the vibration measurement.

The charge sensitivities of the transducer are 1.010, 0.992 and 1.020 pC/ms^{-2} in the z, x and y directions, respectively. The maximum cross sensitivities are 0.4, 2.0 and 2.4% in the z,x and y directions respectively. The frequency range is from 1Hz to 10kHz.

Charge Preamplifiers: Charge preamplifiers are introduced into the measuring set-up in order to convert the high output impedances of the force transducer and accelerometer to lower values and to amplify the relatively weak charge signals from the transducers to voltage signals.

The two charge preamplifiers (Fig. 16) used for the two channels of the force transducer are B&K Type 2651. They have a wide frequency range from 0.003Hz to 200 kHz and hence quasistatic measurements and short-term static calibration of the force transducer are possible. These charge amplifiers are powered by a DC power supply, B&K Type 2805. Two charge preamplifiers, B&K Type 2635, are used for the two channels of the accelerometer. Their frequency range is from 0.1 Hz to 200 kHz. This preamplifier has three switchable modes of operation: linear for acceleration, single integration for velocity, and double integration for displacement measurement. The amplifier also provides low frequency cut-off, switchable to either 1 or 10 Hz in the velocity and displacement modes in order to suppress low frequency noise.

Monitoring and Analyzing Equipment: For the purpose of monitoring and recording, the outputs of the four charge amplifiers are simultaneously fed to a four-channel DC amplifier, Honeywell Accudata Model 117-04, and then recorded on a four-channel oscillograph, Honeywell Visicorder Model 1508A, (Fig. 17). The gain of the Accudata amplifier can be independently adjusted for each channel. The paper speed of the visicorder can be adjusted from 3.75 mm/sec to 3 m/sec.

For the purpose of analysis, the output of a charge preamplifier can also be connected, one at a time, to a spectral analyzer, Spectral Dynamics Corporation Model SD 340. It is a microprocessor based FFT analyzer and is capable of time domain or frequency domain operation. Its frequency range is from 0.25 Hz to 20 kHz (Fig. 18).

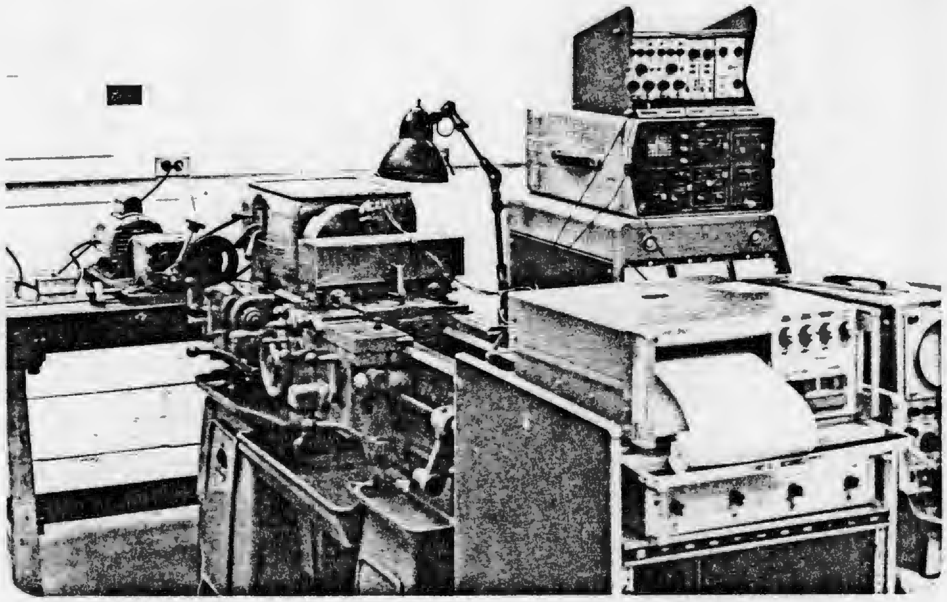


Fig. 17 General view of oscillograph and spectral analyzer.

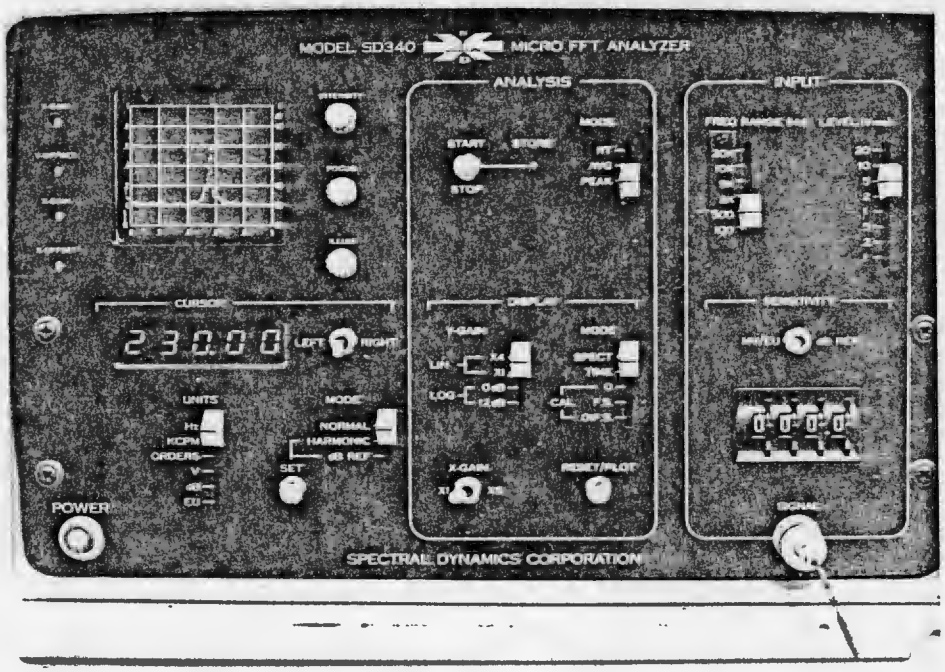


Fig. 18 Natural frequency viewed by the spectral analyzer.

VI. ANALYSIS OF FRICTION-INDUCED SELF-EXCITED VIBRATIONS AND PRELIMINARY EXPERIMENTAL RESULTS

In addition to random fluctuations, there exist three different types of self-excited periodic vibrations induced by friction. These are the following: 1) stick-slip vibrations; 2) self-excited vibrations occuring at the negative slope of friction-sliding velocity curve; 3) self-excited vibrations occuring at the positive slope of the friction-sliding velocity curve. The mechanisms that cause the first two types of self-excited vibrations are fairly well understood.

It is well established that stick-slip is caused by the fact that there is a difference between the static and kinematic coefficients of friction and the static friction varies with time during the period of sticking (5, 32, 33). The cause of the second type of self-excited vibrations is the negative slope of the friction versus sliding velocity curve (35). The mechanism that causes the third type of self-excited vibrations is not well understood. Our conjecture is that the mechanism is the coupling between the frictional and normal degrees of freedom. This coupling effect has been recognized by Tolstoi (2). The interaction between these two directions introduces feedbacks which along with the phase angle differences can supply energy to maintain the vibrations.

In fact, our preliminary experimental results indicate that the coupling between the frictional and normal directions is important for all types of self-excited vibrations. Hence, this interaction should be included in the analysis. The phase-plane method of analysis (5) is adequate only for a single degree of freedom model. The method of analysis being employed in our study is based on describing functions and their extensions. As shown by D'Souza and Caravavatna [36], the method of describing functions is closely related to the energy balance. When the conditions are right, some mechanism draws energy from a nonoscillating source. The system then seeks an energy balance and the values of the amplitudes, frequency and phase angles are such that this energy input is exactly balanced by the energy dissipated per cycle.

Any perturbation from steady state motion initiates transients. When the steady state is stable, the transients decay with time. When the steady state is unstable, the perturbations either tend to the limit cycle vibrations which are sustained around the loop or there is an excursion when limit cycles are not possible. The earlier applications of the method of describing functions, as discussed by Hsu and Meyer [37], are restricted to a single input - single output loop and a further restriction is that all nonlinearities be grouped together into a single nonlinear block and the other block consist of a linear transfer function.

These restrictions are not necessary and in the following we give the development of the techniques for the analysis of friction induced self-excited vibrations. We consider degrees of freedom in the frictional and normal directions. When the disk is rotating at a constant speed, the slider has an equilibrium position which may or may not be stable. Let x and z denote the perturbation displacements of the slider from its equilibrium position in the frictional and normal directions, respectively. Our preliminary experimental results using the spectral analyzer indicate that only the fundamental modes in those two directions are important and the higher order modes are negligible. The mathematical model representing the perturbations is expressed in the form of differential equations as

$$\ddot{x} + 2\zeta\omega_{n1}\,\dot{x} + \omega_{n1}^{2}\,x = f_1\,(x,z) \qquad (1)$$

$$\ddot{z} + 2\zeta_2\omega_{n2}\,\dot{z} + \omega_{n2}^{2}\,z = f_2\,(x,z) \qquad (2)$$

where w_{n1} and w_{nz} are the natural frequencies of the slider in the frictional and normal directions, respectively and ζ_1 and ζ_2 are the damping ratios in those directions. These parameters are evaluated from the pulse response of the slider by using the spectral analyzer. In equations (1) and (2), f_1 and f_2 are nonlinear functions of x and z and are caused by the frictional and normal forces at the disk-slider interface. They depend on the sliding speed and are evaluated by curve fits to the data of steady state friction force versus sliding speed for different values of the normal force. For the analysis of stick-slip vibrations, an additional model (32,5) is required to express the variation of the coefficient of static friction with time of sticking. The equations are shown in the form of a block diagram in Fig. 19.

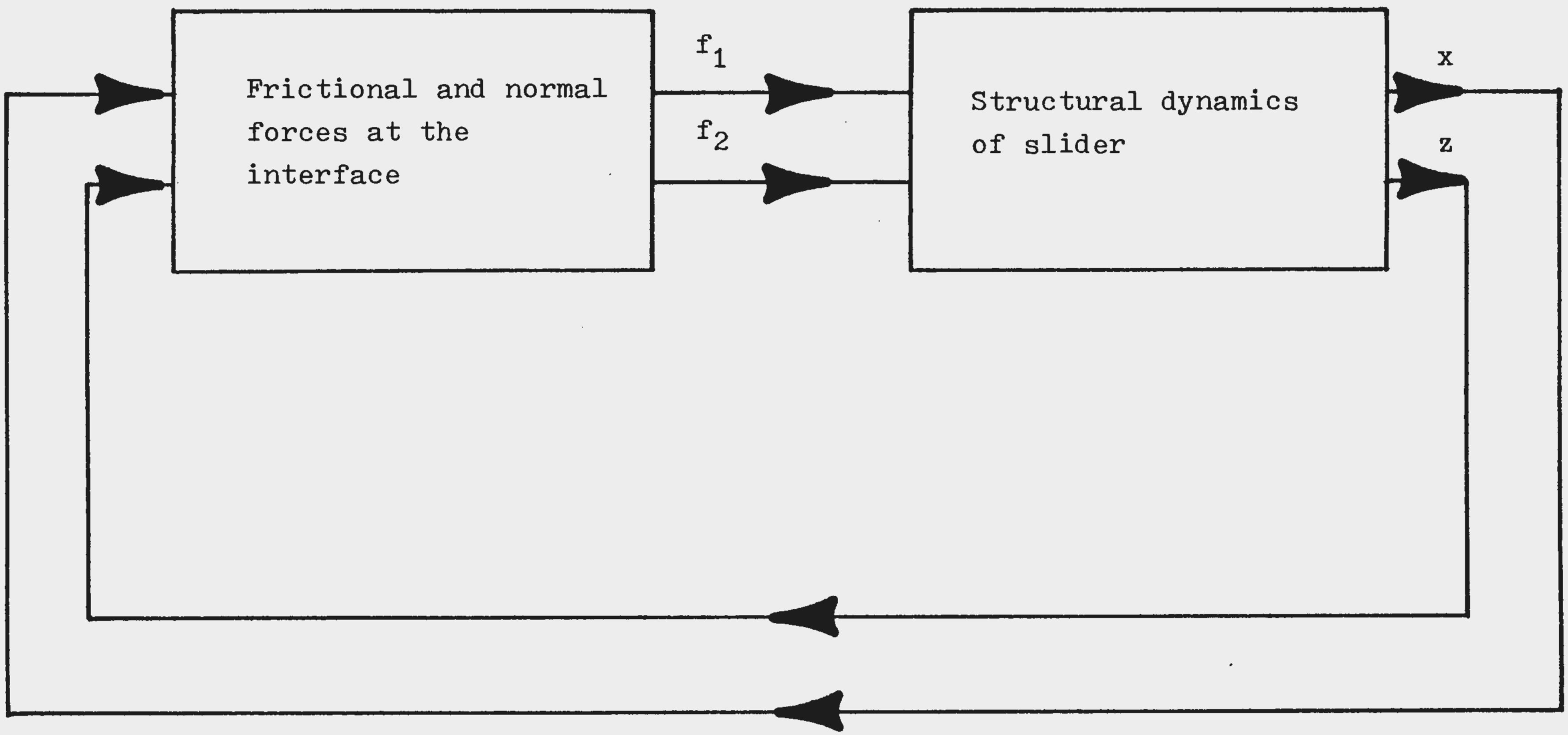


Fig. 19 Feedbacks for friction-induced vibrations.

The values of the natural frequencies depend on the stiffness of the cantilever spring supporting the slider and on the contact stiffness. Furthermore, the contact stiffness appears to depend on the nature of the surfaces in contact. For example, for a certain length of the cantilever spring and normal force, the measured values of the natural frequencies were ω_{n1} = 395 Hz and ω_{n2}= 400 Hz in the frictional and normal directions, respectively. Self-excited vibrations were observed whose frequencies were 750 and 745 Hz in the frictional and normal directions, respectively. In addition, the vibration in the frictional direction had a small component whose frequency was 220 Hz.

After about fifteen minutes of running time had elapsed, the measurements were repeated. It was found that the natural frequencies had now changed to ω_{n1} = 390 Hz and ω_{n2} = 395 Hz i.e. there was a decrease of 5Hz from the previous measurements. The frequencies of the self-excited vibrations also changed to 720 Hz with a 215 Hz component in the frictional direction and to 685 Hz in the normal direction. The closeness of the two frequencies in the frictional and normal directions may explain the beat type phenomenon that has been sometimes previously observed in friction-induced vibrations.

In any case, our preliminary results indicate that there is a strong coupling between the frictional and normal directions. This aspect requires detailed further study. It is necessary to employ the method of multiple-input describing functions [38] in order to include the different frequencies. Each frequency component is then balanced round the loop of Fig. 19. This method is discussed in the following.

The method of describing function assumes that there are self-excited vibrations. Then x and z are both periodic with fundamental frequencies ω_1 and ω_2, respectively. The nonlinear terms in equations (1) and (2) generate higher harmonics which are assumed to be filtered out as they pass around the loop [37]. Extension of the method is possible when this assumption is not satisfied. The fundamental components which are transmitted around the loop are $x=x_o \sin \omega_1 t$ and $z=z_o \sin (\omega_2 t+ \phi)$.

The nonlinear terms in equation (1) and (2) are replaced by their equivalent dual input describing functions. In the frequency domain, the derivative d/dt is replaced by $j\omega$. In this manner, the differential equations (1) and (2) are converted to nonlinear algebraic equations with real and imaginary terms. Equating separately the real and imaginary parts for each of the two frequencies, we obtain a set of coupled, nonlinear and real algebraic equations. The solution of these equations yields the amplitudes x_o and z_o, the frequencies ω_1 and ω_2 and the phase angle ϕ.

In case the set of algebraic equations has no solution with real values for the unknowns with further restriction that the amplitudes and frequency are postive real numbers, then self-excited vibrations do not occur. In case the high harmonics are not filtered out or subharmonics are generated, then the foregoing procedure requires modification.

When the method of describing functions yields the existence of self-excited vibrations, the results can be used further to determine the exact mechanism that supplies energy to the vibrations [36]. By balancing the energy, it can be shown that the net energy supplies is exactly balanced by the energy dissipated per cycle by the vibrations.

During the early part of the analytical study, the emphasis has been on the development of the foregoing methodology and techniques for the analysis of friction induced vibrations. The work now in progress includes the experimental determination of the functional relationships that will be incorporated in the analytical study.

VII. FUTURE PLANS

This progress report covers the progress made to date on this project. The future plans involve the following activities:

A. Friction and wear studies:

None

a) Determine the range of sliding speed for oxidative wear for the most rigid slider system.

b) Investigate the rate of wear as a function of sliding speed, normal load, system rigidity, observed frequency of the slider oscillation, and amount of energy dissipation for the oxidative wear range.

No

c) Determine the critical speed at which transition from mild to severe wear takes place for the most rigid slider system.

d) Investigate the critical speed and load behavior as a function of system rigity, frequency of slider oscillation, and amount of dissipated energy.

B. Analysis of Friction-Induced Vibration:

a) Both the spring stiffness and contact stiffness contribute to the overall system rigidity and natural frequencies. The spring stiffness is well understood from beam theory and is a constant for a particular configuration and length of the beam. The contact stiffness depends on the two surfaces in contact. It appears to vary with the nature of the surfaces in contact (i.e. the existence of oxide films, etc.), and is not well understood. Future plan is to study this relationship and its effect on the natural frequencies.

Coupled is

b) Investigate the coupling between frictional and normal directions. The objective is to determine the functional relationships which are discussed in more detail in the renewal proposal.

c) Continue the development of methodology and techniques for the analysis of different types of vibrations observed at different sliding speeds.

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