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RESEARCH AND DEVELOPMENT ON INVESTIGATION
OF GALLING AND FRICTION CHARACTERISTICS OF
METALLIC MATERIALS AND SURFACE TREATED MATERIALS

By

Eber W. Gaylord

To

Watertown Arsenal
Watertown 72, Massachusetts

FINAL REPORT

July 1, 1956

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Carnegie Institute of Technology
Department of Mechanical Engineering
Pittsburgh 13, Pa.

RESEARCH AND DEVELOPMENT ON INVESTIGATION OF GALLING AND FRICTION
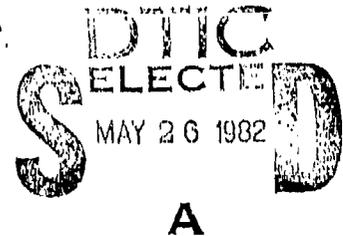CHARACTERISTICS OF METALLIC MATERIALS AND SURFACE TREATED MATERIALS

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OBJECT

To study the galling and friction characteristics of metallic materials
and surface treated materials and to investigate the thermal aspects of
friction.

ABSTRACT

This work is composed of three parts, which are:

1) A theoretical analysis of a dynamic thermocouple formed by the
junction of two metals sliding over each other.

2) A report on an experimental investigation of the dynamic
thermocouple.

3) The results of friction and galling tests are given for,
titanium rubbing on steel, titanium on titanium, surface coated
titanium on steel, surface coated titanium on uncoated titanium and
surface coated titanium on surface coated titanium.

SUMMARY

PART I   ON THE THEORETICAL ANALYSIS OF A DYNAMIC THERMOCOUPLE

The "Dynamic" thermocouple formed by the moving junctions of two dissimilar metals is analyzed theoretically. It is found that if the two leads from the cold junction, in series with a potentiometer are symmetrically placed in two bodies rubbing over each other the e.m.f. measured by the potentiometer satisfies Laplace's equation in terms of the positioning of the leads in the body. The boundary condition is that the potential at any contact area is the Seebeck e.m.f. corresponding to the contact area temperature.

It is shown that in the case of two semi-infinite rubbing bodies with many randomly distributed contacts, small in area compared to the distance between them, that the potential measured by thermocouple leads placed at an infinite distance away from the contact areas is the average of the Seebeck e.m.fs., corresponding to the contact temperatures, weighted by the square root of the areas.

PART II   EXPERIMENTAL RESEARCH ON THE DYNAMIC THERMOCOUPLE

An experimental investigation is made of the d.c. or time averaged e.m.f. produced by a dynamic thermocouple to establish its relationship to the interface temperature generated by friction. Tests are made to see if this d.c. e.m.f. is due entirely to thermal effects. Experiments to compare the dynamic thermocouple e.m.f. with theoretically estimated interface temperatures indicate that the real contact areas between two rubbing metals corresponds to the normal load and the yield pressure of the softest material.

PART III THE GALLING AND FRICTION CHARACTERISTICS OF SURFACE COATED
TITANIUM SPECIMENS

Rubbing tests are made to find the coefficient of friction and
galling characteristics of surface coated titanium rubbing on steel,
uncoated titanium, and coated titanium.

In resistance to galling considerable improvement over the use
of uncoated titanium is found for surface coated titanium rubbing on
steel, and on coated titanium, but not for coated titanium rubbing on
uncoated titanium.

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ON THE THEORETICAL ANALYSIS OF A DYNAMIC THERMOCOUPLE

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NOMENCLATURE

E	= Seebeck e.m.f.
π_{AB}	= Peltier coefficient
σ	= Thomson coefficient
γ	= electrical resistivity
J	= current flux density
P	= potential measured by a potentiometer placed in the thermocouple circuit
ϕ	= $P - P(\infty)$
$m_n(r)$	= intensity of the current source for the nth contact area
(x,y,z)	= coordinates in body B
(ξ, η, ζ)	= coordinates in body A
$T(x,y,z)$	= temperature in body B
T_o	= cold junction temperature
$(\ )_A$	= subscript referring to body A
$(\ )_B$	= subscript referring to body B

INTRODUCTION

The dynamic thermocouple, better known as the "Herbert Gottwein" thermocouple utilizes the junction between two dissimilar metals which are rubbing together as one junction of a thermocouple circuit, (Fig. 1). The e.m.f. generated in this circuit is used to estimate the temperature at the interface of two rubbing dissimilar metals.

In metal cutting research, the dynamic thermocouple, formed by the cutting tool and its moving tool chip, has been used to estimate the average interface temperature. In this application, the measured interface temperature has been found to agree rather well with the theoretically determined area averaged interface temperature[1].

The dynamic thermocouple has also been applied to the investigation of frictional heating between two metallic surfaces sliding over each other[2]. In this case the force between the two rubbing surfaces is much lower than it is between a tool and tool chip. Only a small fraction of the apparent contact area between the interface of the two rubbing metals is true contact area. Correlation between the measured thermocouple e.m.f. and theoretically determined temperatures has not been as conclusive as in the tool chip problem.

The following work is an analysis to determine how the instantaneously measured e.m.f. of the dynamic thermocouple is related to the instantaneous temperature distribution over the contact interface of the moving junction.

ANALYSIS

Consider as a model a large body of metal B rubbing on a large body of metal A at a given instant of time with axis (x,y,z) and (ξ,η,ζ) respectively; see Fig. 2. With a lead of metal B located at (x,y,z) in body B, and metal A at (ξ,η,ζ) in body A, the potential measured by the potentiometer will be some function $P(x,y,z,\xi,\eta,\zeta)$.

It will be assumed that the rubbing does not in itself generate any voltages in the system so that the potential is the same as it would be in a static system with the same temperature and contact situation. Assume that electrical time constants due to inductive and capacitive effects are zero so that electrically the system has reached steady state.

The interface between the two metals may be considered to consist of any number of arbitrarily small contact areas which in the limit become a continuous contact area with any arbitrary temperature distribution.

Next consider the thermoelectric effects in the thermocouple. Referring to an ordinary thermocouple, see Fig. 3, the Seebeck e.m.f. of a thermocouple is

$$ E \;=\; \pi_{AB} T_2 \;-\; \pi_{AB} T_1 \;+\; \int_{T_1}^{T_2} (\sigma_A - \sigma_B)\, dT \tag{1} $$

For the purposes of analysis the Seebeck e.m.f. is divided into two parts; the Peltier e.m.f., $\pi_{AB} T_1 - \pi_{AB} T_2$ which will be assumed to be generated at the junctions of the dissimilar metals and the Thomson e.m.f., $\int_{T_1}^{T_2} (\sigma_A - \sigma_B)\, dT$ which is assumed to be generated in the metal.

Let us now consider relations which must hold within the metal B.

First consider the one dimensional case, Fig. 4. Taking account of the Thomson e.m.f. and the Joule effect in the element.

$$V_{x+\Delta x} - V_x = \int_{T_x}^{T_{x+\Delta x}} \sigma_B \, dT - J \frac{\Delta x}{\gamma} \tag{2}$$

From which:

$$J_{(x)} = \gamma \sigma_B \frac{dT}{dx} - \gamma \frac{dV}{dx} \tag{3}$$

By similar reasoning for three dimensions

$$J = \gamma \sigma_B \nabla_B T - \gamma \nabla_B V \tag{4}$$

For steady state current flow

$$\nabla \cdot J = 0 \tag{5}$$

Combine (4) + (5)

$$\nabla_B^2 V = \nabla_B \cdot (\sigma_C \nabla_B T) \tag{6}$$

Referring to Fig. 2 note that:

$$V_{(x,y,z)} = P + \int_{T_0}^{T_{x,y,z}} \sigma_B \, dT + V_0 - \pi_{AB} T_0 \tag{7}$$

From 7 one finds that

$$\nabla_B^2 V = \nabla_B^2 P + \nabla_B \cdot (\sigma_B \nabla_B T) \tag{8}$$

Combining (6) and (8)

$$\nabla_B^2 P = 0 \tag{9}$$

By similar reasoning

$$\nabla_A^2 P = 0 \tag{10}$$

Along the boundaries with no current flow

Equation 4 becomes $\qquad \sigma_B \nabla_B T - \nabla_B V = 0 \qquad$ (11)

Combining (11) and (7) gives:

$$\frac{\partial P}{\partial n} = 0$$

(12)

For the boundary conditions where there is contact, the voltage drops around the circuit equal zero.

$$P = \int_{T_0}^{T} (\sigma_B - \sigma_A) dT + \pi_{AB} T - \pi_{AB} T_0 - B \int_{x,y,o}^{x,y,z} \frac{J}{\gamma} \cdot dS - A \int_{\xi,\eta,o}^{\xi,\eta,\xi} \frac{J}{\gamma} \cdot dS$$

(13)

or

$$P = E(T_{x,y,o}) - B \int_{x,y,o}^{x,y,z} \frac{J}{\gamma} \cdot dS - A \int_{\xi,\eta,o}^{\xi,\eta,\xi} \frac{J}{\gamma} \cdot dS$$

(14)

where $E(T_{x,y,o})$ is the Seebeck e.m.f. corresponding to the temperature on the boundary.

Since this boundary condition involves both blocks, A & B, the potential is a function $P(x,y,z,\xi,\eta,\xi)$, and equations 9 and 10 would have to be solved simultaneously.

This problem may be simplified by assuming that both blocks are identical with regard to boundary conditions and that the leads are symmetrically placed at $(x,y,z) = (\xi,\eta,\xi)$.

This assumption would be reasonable if both blocks were semi-infinite and the leads were placed an infinite distance away from the contact area. With this assumption

$$\nabla_A = \nabla_B = \nabla$$

(15)

By simultaneously placing both leads at (x,y,o) and (ξ,η,o), equation 14 gives the boundary condition

$$P = E(T_{x,y,o})$$

(16)

Hence we now have the problem reduced mathematically to a potential problem in a model the shape of one of the rubbing bodies with the boundary conditions that the potential is equal to the Seebeck e.m.f. on the boundary where rubbing contact is made, and $\frac{\partial P}{\partial n} = 0$ on the boundary where no contact is made.

In most practical applications of the dynamic thermocouple where the contact area is small compared to the size of the rubbing bodies one would be interested in solving for the potential at infinity in terms of the contact area temperatures and their corresponding Seebeck e.m.fs.

MANY RANDOMLY DISTRIBUTED SMALL CONTACT AREAS.

The foregoing theory will be applied to a case which might represent the frictional rubbing of two metal surfaces under a light load, where it will be assumed that there are many contact asperities and the total real contact area is small compared to the apparent contact area. The problem is to determine what temperature a potentiometer would indicate if the thermocouple leads are placed a large distance away from the contact areas.

In reducing this case to a potential problem as previously shown, consider a semi-infinite solid, Fig. 5. Near the origin and on the surface there is a distribution of area sources of current. These sources are held at constant voltage, each source being at some value. The size and value of the potential of each source varies.

It is desired to find the potential in the solid at some large distance removed from the localized region of sources. With $P(\infty)$ defined as the potential at this point define for convenience $\phi = P - P(\infty)$

As previously derived $\nabla^2\phi = 0$, $\frac{\partial \phi}{\partial n} = 0$ on surfaces with no contact and $\phi = E_n - P(\infty) = V_n$ on surface of the n_{TH} source area of a total of N source areas.

Contributions to $\phi(x, y, z)$ due to each source area can be added. Let $\vec{R}$ denote the point where potential is to be measured and $\vec{r}$ a point in the area source, then the contribution of the n'th source to the potential is:

$$\phi_n(\vec{R}) = \int_{A_n} \frac{m_n(\vec{r}_n)\, dA_n}{|\vec{R}_n - \vec{r}_n|} \tag{17}$$

where $m_n(\vec{r}_n)$ is the intensity of the source as a function of $\vec{r}_n$

For N sources

$$\phi(\vec{R}) = \sum_{n=1}^{N} \int \frac{m_n(\vec{r}_n)}{|\vec{R}_n - \vec{r}_n|}\, dA_n \tag{18}$$

and on the i_{th} source equation 18 expresses the boundary condition as:

$$V_i = \sum_{n=1}^{N} \int_{A_n} \frac{m_n(\vec{r}_n)}{|\vec{R}_i - \vec{r}_n|}\, dA_n$$

For large values of R_n equation 18 makes $\phi(\vec{R})$ go to zero for finite sources, which is consistent with the definition of ϕ.

The solution of the problem lies in finding the value of the current intensity $m_n(\vec{r}_n)$ in terms of the source potentials V_n. To simplify this step divide equation (19) into two parts

$$V_i = \sum_{\substack{n=1 \\ n \neq i}}^{N} \int_{A_n} \frac{m_n(\vec{r}_n)}{|\vec{R}_i - \vec{r}_n|}\, dA_n + \int_{A_i} \frac{m_i(\vec{r}_i)}{|\vec{R}_i - \vec{r}_i|}\, dA_i \tag{20}$$

Examine the first part of 20. If the areas, A_n, are small compared to values of R_i, one could write

$$\sum_{\substack{n=1 \\ n \neq i}}^{N} \int_{A_n} \frac{m_n(\vec{r}_n)}{|\vec{R}_i - \vec{r}_n|}\, dA_n = \sum_{\substack{n=1 \\ n \neq i}}^{N} \frac{\int_{A_n} m_n(\vec{r}_n)\, dA_n}{|\vec{R}_i - \vec{r}_n|} \tag{21}$$

where $|\overline{\vec{R}_l - \vec{r}_n}|$ is a mean value.

From continuity of currents

$$\sum_{n=1}^{N} \int_{A_n} m_n(\vec{r}) \, dA_n = 0 \tag{22}$$

On the basis of equation 22, if there is a random distribution of sources with respect to current flux, $\int_{A_n} m_n(\vec{r}_n) \, dA_n$, and if all values of $|\vec{R}_l - \vec{r}_n|$ are large compared to $|\vec{R}_l - \vec{r}_l|$ then the first part of equation 19 will go to zero as N, the number of sources becomes very large, giving

$$V_l = \int_{A_l} \frac{m_l(\vec{r}_l)}{|\vec{R}_l - \vec{r}_l|} \, dA_l \tag{23}$$

To evaluate equation 7 assume that all sources are circles of radius a. The current strength for such a source is given in reference 4

$$m_l(\beta) = \frac{K_l}{\sqrt{a_l^2 - \beta_l^2}} \tag{24}$$

where $\beta_l = |\vec{R}_l - \vec{r}_l|$ and K is some proportionality constant to be evaluated. Substituting 24 in 23 yields $K_l = \frac{V_l}{\pi^2}$ giving

$$m_l(\beta_l) = \frac{V_l}{\pi^2\sqrt{a_l^2 - \beta_l^2}} \tag{25}$$

Substituting 25 in 22,

$$0 = \sum_{n=1}^{N} \int_{A_n} m_n(\beta_n) \, dA_n = \sum V_n a_n \tag{26}$$

$$A_n = \pi a_n^2 \quad ; \text{ Hence } \sum V_n \sqrt{A_n} = 0 \tag{27}$$

Since $\quad V_n = E_n - P(\infty)$

$$P(\infty) = \frac{\sum E_n \sqrt{A_n}}{\sum \sqrt{A_n}}$$

<div align="right">(28)</div>

Equation 28 gives the conclusion, that if the Seebeck e.m.f. is linearly related to the temperature one would measure the square root area average temperature rather than the area average temperature.

In the friction problem the physical interpretation would be that if the.e were any correlation between the size of the protuberances and their temperature, then the temperature corresponding to the dynamic thermocouple e.m.f. would be weirhted more heavily to the temperature of the smaller protuberances.
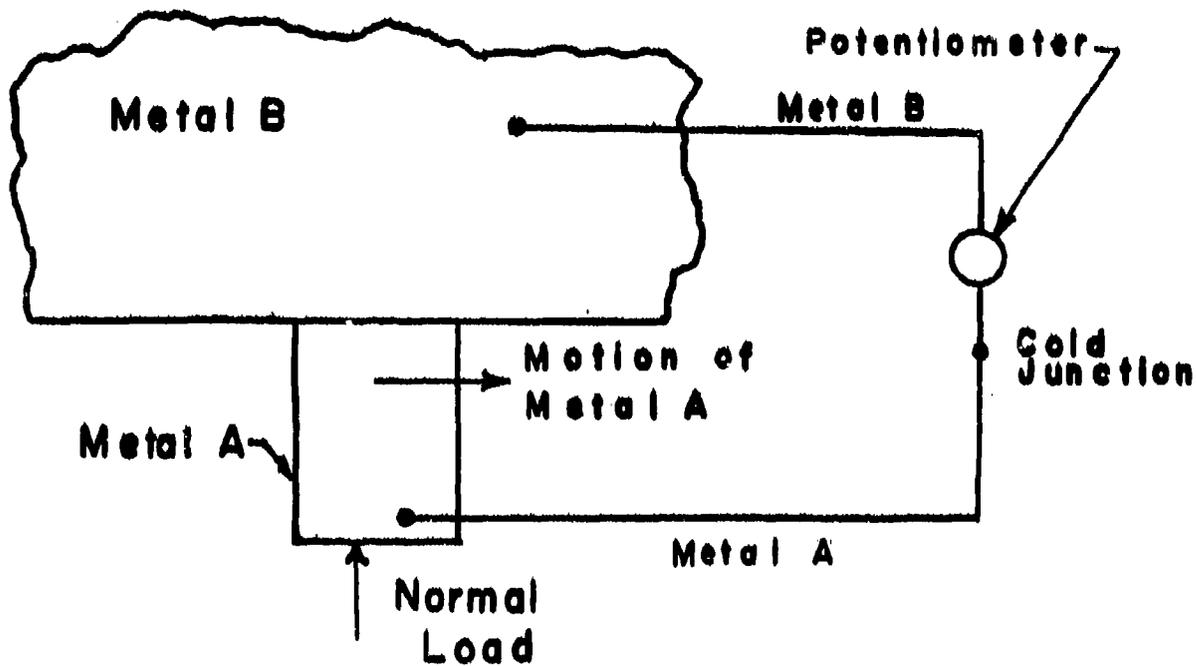
Fig. 1. Schematic of Dynamic Thermocouple.
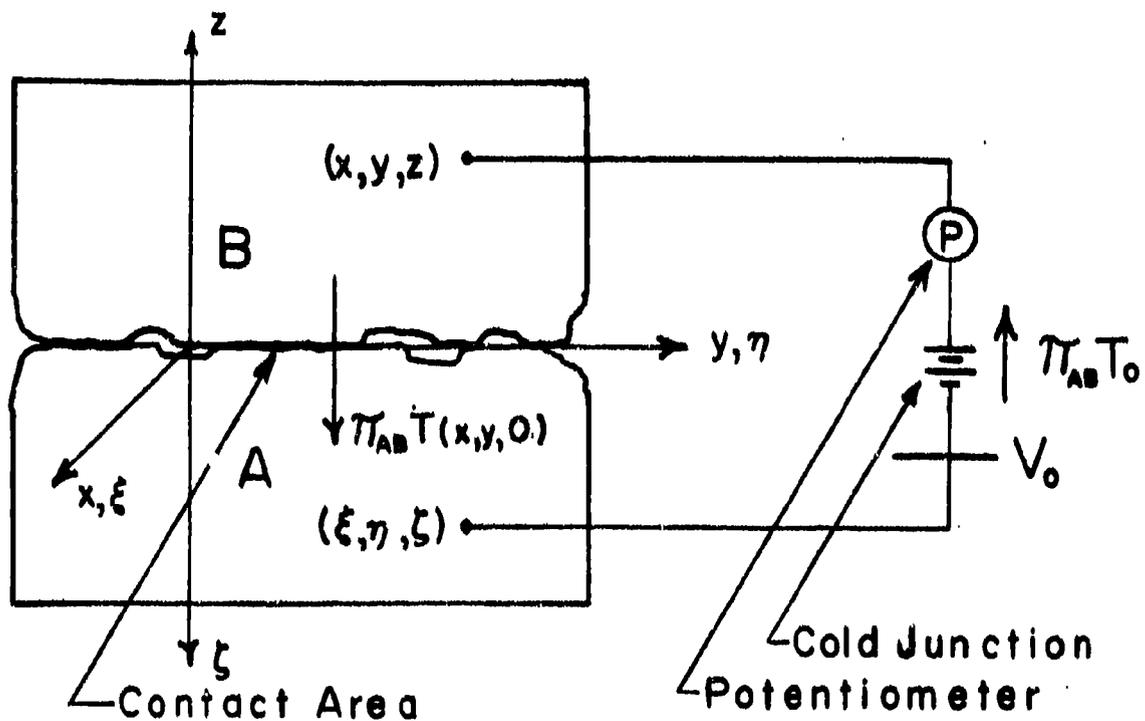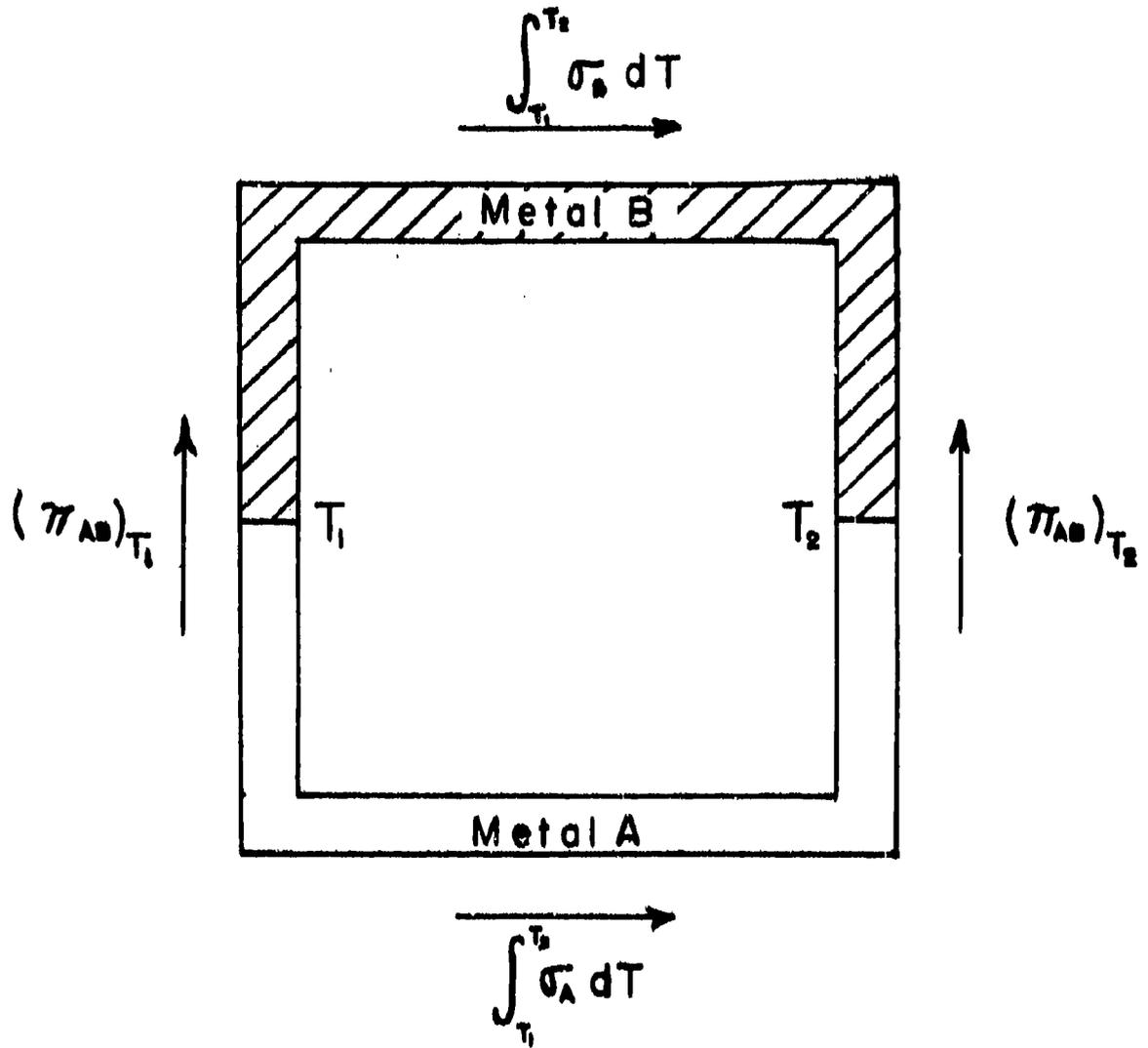
Fig. 2. Model for Dynamic Thermocouple Circuit.

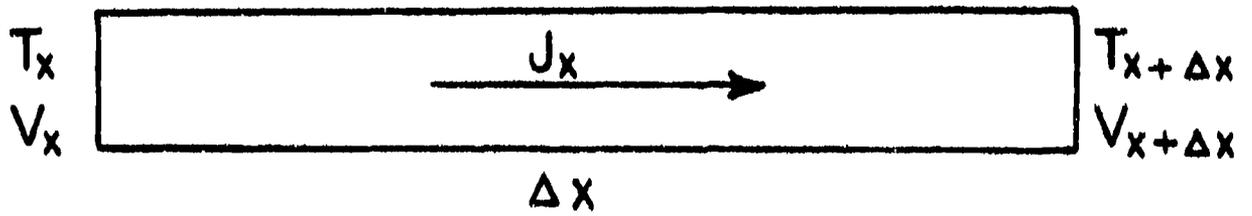Fig. 3. Illustration of Thermocouple Effects.

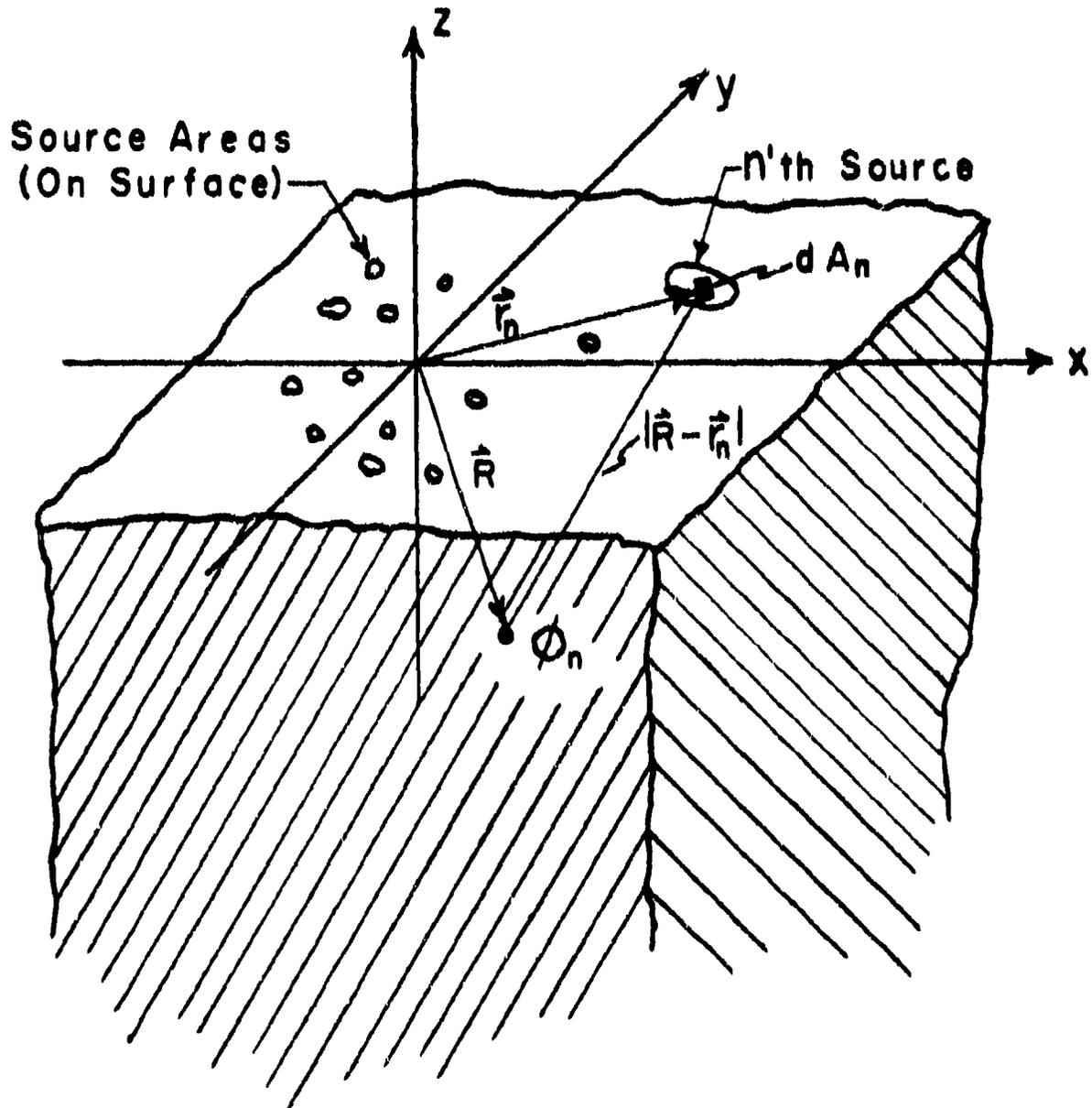Fig. 4. One Dimensional Current Flow in the Metallic Solid.

Fig. 5. Randomly Distributed Current Sources.

EXPERIMENTAL RESEARCH ON THE DYNAMIC THERMOCOUPLE

INTRODUCTION

In a rubbing process of one metallic surface over another, heat is generated at the interface by friction. This heat will cause a rise in temperature at the interface. A large change in temperature should cause a change in the properties of the material at the interface.

The measurement of the true interface temperature presents a problem because the temperature may fall off rapidly below the surface and, the exact nature of the contact area between the metals may not be known. If full contact area were made and the interface temperature were uniform over a sufficiently large area, then interface temperature could be determined by imbedding of thermocouples below the surface and computing the temperature at the surface.

In the friction problem however, uniform contact would require very large normal forces between the two rubbing metals such as occurs in metal cutting. With much smaller normal loads only certain small portions or surface asperities make contact. In the experimental work to be described this was the case.

DYNAMIC THERMOCOUPLE

The dynamic thermocouple discussed previously in The Theoretical Analysis, is described by Bowden and Tabor (2). When the e.m.f. generated by the thermocouple, during rubbing, is recorded on a cathode ray oscilloscope there is traced a pattern of random fluctuations of voltage with peaks that are very high compared to the time averaged or d.c. component of the thermocouple e.m.f. These voltage peaks were interpreted by Bowden and other investigators to represent very high temperature flashes at points on the interface of two rubbing metals. In some cases they were estimated to reach the melting point of the metal with the lowest melting temperature. However it remains to be proven whether these voltage peaks represent the temperature at the interface. On the basis of the preceeding analysis only contact asperities whose temperatures were very high could be in contact when such a flash occurred.

In this project the time average or d.c. component was investigated. This was done by electrically filtering the randomly fluctuating component of the dynamic thermocouple e.m.f. The problem is to determine if this e.m.f. corresponds to any sort of time space averaged temperature of the interface.

In being able to propose that any correlation between the d.c. component of the dynamic thermocouple e.m.f. and temperature exists the following assumptions have to be made and verified.

1) If the unfiltered e.m.f. produced by the dynamic thermocouple is different from what it would be as determined purely on the basis of the Seebeck e.m.f. then that difference should be a voltage that time averages to zero and is completely filtered.

2) The Seebeck e.m.f. or thermo-electric power characteristics of the rubbing pair of dissimilar metals should not be appreciably altered by the mechanical effect of rubbing.

APPARATUS

Two friction testing machines have been used for investigating the dynamic thermocouple. The older machine which will be referred to as the old friction apparatus is described in a previous report "Investigation of Galling and Friction Characteristics of Titanium Alloys"[5]. It consists of a plate revolving in a vertical plane with a slider or rod pressed against it by a normal loading spring. The assembly consisting of the slider and normal loading apparatus pivots on bearings in a plane parallel to the plate against a strain ring which measures the tangential load by means of strain gauges. The dynamic thermocouple circuit, see Fig. 6, is completed by connecting the moving plate to a mercury bath by means of a flexible cable.

The second machine, shown in Fig. 7, which will be called the new friction apparatus, was constructed during the present contract. With this apparatus, see Fig. 7, the plate revolves in a horizontal plane. The rubbing specimen, a slider, is held in a counter-weighted arm mounted on bearings which allow it to swing in a vertical plane. The normal load is applied by hanging weights on the arm. The tangential loading scheme is the same as that used by E. Rabinowicz[6]. The loading arm is made free to move in a horizontal plane by the deflection of a leaf spring.

A strain gauge ring measures the tangential load or frictional force. A dash pot is added to the system to prevent oscillations of the loading arm.

The new apparatus was built so that the two rubbing test specimens can be completely enclosed in a furnace. This makes it possible to calibrate the Seebeck e.m.f. of the pairs of metals used as dynamic thermocouples. The furnace also makes it possible to study the thermal effects of friction at elevated temperatures.

A number of refinements over the older apparatus are inherent in its design. They are--

1. The flexible cable and its possibility of generating electrical noise is eliminated.

2. A slot in the vertical shaft which turns the plate allows thermocouple leads to run down from the test specimen to the mercury bath at the base eleminating the possibility of having a temperature gradient in an intermediate metal in the thermocouple circuit.

3. Because there is no bearing friction in the tangential load measuring system the tangential load can be determined more accurately.

METHOD OF ESTIMATING INTERFACE TEMPERATURE FROM FRICTION MEASUREMENTS

The theory for estimating interface temperature from friction measurements is given in "An Investigation of Sliding Friction and Interface Temperatures Between Two Dry Metallic Surfaces"[7].

In brief the analysis is as follows:

Due to friction a uniform heat flux q is assumed to be generated over the area of the interface of a slider rubbing over a plate. A portion σQ of this heat flux is assumed to go to the slider and $(1-\sigma)Q$ to the plate. σ is assumed to be constant over the interface contact area.

Assuming that the contact area is a circle of radius "a" the average temperature rise at the interface above the bulk temperature of the sliding pairs of metals is

$$\bar{\Theta} = 0.731 \ \frac{Qa}{K_1} \qquad (29)$$ where K_1 is the thermal conductivity of the

slider.

Using an analysis by Block[8], is found by calculating the temperature of the plate surface due to $(1-\ )$ and equating the maximum temperature of the plate surface to the slider surface, giving

$$\sigma = \frac{1}{1 + \frac{K_2}{K_1}(2R)^{\frac{1}{2}}} \qquad R = \frac{Va}{4\alpha_2} > 5 \qquad (30)$$

$$\sigma = \frac{1}{1 - \frac{K_2}{K_1}} \qquad R = 0 \qquad (31)$$

The gap in between, $0 < \sigma < 5$, is given by F. Ling as

$$\sigma = \frac{1}{1 + \frac{K_2}{K_1} \frac{2\pi}{[1 + 0.414(1 - e^{-1.3K})] \ I(R)}} \qquad (32)$$

where
V = sliding velocity
K = thermal conductivity
ρ = density
C = specific heat
$d = K/\rho C$ = thermal diffusivity
$(\ )_1$ refers to slider
$(\ )_2$ refers to plate
$I(R)$ = Block's Function[7]

Values of σ versus R for the combinations of materials used in this project are shown in Figs. 8 through 10.

Using experimental friction data

$P = \mu N V$ = power dissipated by friction

where
μ = coefficient of friction
N = normal load

Letting
$$Q = \frac{P}{\pi a^2}$$

$$\frac{\theta}{P} = \frac{0.232\sigma}{a} \tag{33}$$

The radius a which is needed to solve the above equation as well as to determine σ is found by assuming that the contact area is determined by the yield stress of the softer material. That is

$$a = \sqrt{\frac{N}{\pi S}} \tag{34}$$

where S is the yield stress of the softer material.

The yield stress was determined from handbook values given for the materials used.

Equation 33 was evaluated for constantan on steel, steel on constantan and copper on constantan, see Fig (11) throught (12).

RUBBING TESTS OF DYNAMIC THERMOCOUPLES

Rubbing tests were performed on the new apparatus with the following pairs of metals:

1. Constantan slider on steel plate
2. Steel slider on constantan plate
3. Copper slider on constantan plate.

The normal load, tangential load, and rubbing velocity were determined. The dynamic thermocouple e.m.f. was electrically filtered and its d.c. component was measured by a potentiometer.

The dynamic thermocouple e.m.f. of the rubbing materials had previously been calibrated against a standard reference thermocouple by placing the apparatus in a furnace. Calibration curves for the materials used are shown in Figs. 13 throught 15.

Thermocouple measurements were made with the plate moving and with it stopped. It was found that after rubbing stopped the e.m.f. immediately fell back to the same value it had before rubbing began. This e.m.f. corresponds to the bulk temperature of the plate and slider. Using the calibration curves the increase in e.m.f. between the stationary and the running state was evaluated in terms of a temperature rise Θ. Friction measurements were used to evaluate the power dissipated in rubbing, P, and the experimentally determined values of $\frac{\Theta}{P}$ were plotted, see Figs. 16 through 19. The theoretically estimated values of $\frac{\Theta}{P}$ for corresponding rubbing speeds and normal loads are also plotted.

AGREEMENT OF ACTUAL VALUES OF $\frac{\Theta}{P}$ WITH THEORY

The experimentally determined points for $\frac{\Theta}{P}$ show considerable scatter and are generally lower than the theoretically estimated values for constantan sliders rubbing on steel plates or steel sliders rubbing on constantan plates. With copper sliding on constantan the experimentally determined values of $\frac{\Theta}{P}$ are higher than the theoretical values.

However, the results should not be expected to agree too well with the theoretically determined values because of the assumptions that were made. The significance is that they are of the same order of magnitude.

The assumptions that make the agreement between the experimental and theoretical values doubtful are:

1) It is doubtful that the real contact area is just one circular area.

2) The assumed yield stresses of the softer materials, 30,000 psi for constantan and 10,000 psi for copper may differ greatly from the actual yield stress of the materials during rubbing.

3) The portion of the heat flux dissipated to the slider, σ, is not constant over the entire contact area.

4) Some portions of the contact area may have been contaminated with surface oxides or other films. These contaminated areas may have supported some of the normal load with out producing any thermoelectric effects.

COEFFICIENT OF FRICTION AND DYNAMIC THERMOCOUPLE e.m.f.

With a constantan slider rubbing on a steel plate there was found to be some relation between $\frac{e}{P}$, the temperature rise divided by the power dissipated, and the coefficient of friction; see Fig. 20. Experimentally determined values of $\frac{e}{P}$ are considerably larger for small coefficients of friction than for larger values. The low coefficients of friction are generally associated with contamination of the plate surface.

DYNAMIC THERMOCOUPLE e.m.f. AND GALLING

When severe galling takes place it was difficult with the circuit and measuring instruments used, to determine a d.c. or filtered component of the dynamic thermocouple e.m.f.

The e.m.f. fluctuates randomly over a wide range of values. The coefficient of friction also fluctuates. It has not been determined whether the dynamic thermocouple e.m.f. in this case is due entirely to thermal effects.

VOLTAGE PRODUCED BY SIMILAR METALS RUBBING OVER EACH OTHER

Work done by F. Ling[7] on the old friction testing apparatus rubbing steel on steel showed a randomly fluctuating e.m.f. being produced, but when this was filtered there was no d.c. component. A similar test was run using the new apparatus and produced the same results.

These results help to strengthen the assummation that the d.c. component of the dynamic thermocouple e.m.f. is due entirely to thermal effects.

TESTS OF HEATED DYNAMIC THERMOCOUPLES

Using the old apparatus, a cone shaped constantan rod, Fig. 21, was rubbed on a heated steel plate. A static reference thermocouple was placed a short distance back of the rubbing surface. Temperatures calibrated from the e.m.f. of the plate rod junction are compared with those measured for the static thermocouple with the plate stationary, stationary and fused to the rod tip, and with the plate moving, see Fig. 22. The fairly small differences between the measurements for these three cases could have been due to different heat transfer characteristics between the rod and plate for the different types of contact, static, fused, and moving. The large temperature gradients in the cone shaped test sample are shown in Fig. 23.

With the new friction testing apparatus rubbing tests of constantan rubbing on steel and copper rubbing on constantan were made with the specimens enclosed in a furnace. The ambient temperature ranged from

about 80°F to 200°F. $\bar{\theta}$, corresponding to the difference in the d.c. component of the dynamic thermocouple e.m.f. when there was rubbing and no rubbing, was measured. Figs. 24 through 27 show values of $\frac{\bar{\theta}}{P}$ versus ambient temperature.

Due to a certain amount of non-uniform heating of the apparatus enclosed in the furnace and the resulting heat conduction, especially in the case of the copper slider, results were not as consistent as they were when the entire apparatus was at room temperature.

However the values of $\bar{\theta}$ are small and more or less of the same order of magnitude as they would be at room temperature for corresponding rubbing speeds and normal loads.

The results show that rubbing at higher temperatures did not cause any appreciably greater change in the dynamic thermocouple e.m.f. than it did at lower temperatures. These tests help strengthen the assumption that rubbing does not in itself appreciably change the thermoelectric power characteristics of the dynamic thermocouple.

CONCLUSIONS

1. Tests conducted so far have shown strong evidence that the time averaged or d.c.e.m.f. of a dynamic thermocouple is due to thermal effects.

2. Correlation of measured dynamic thermocouple e.m.fs. with theory, support the theory that the true contact area of two metals sliding over each other is of the same order of magnitude as the area that would be determined from the normal load and the yield stress of the softer material.
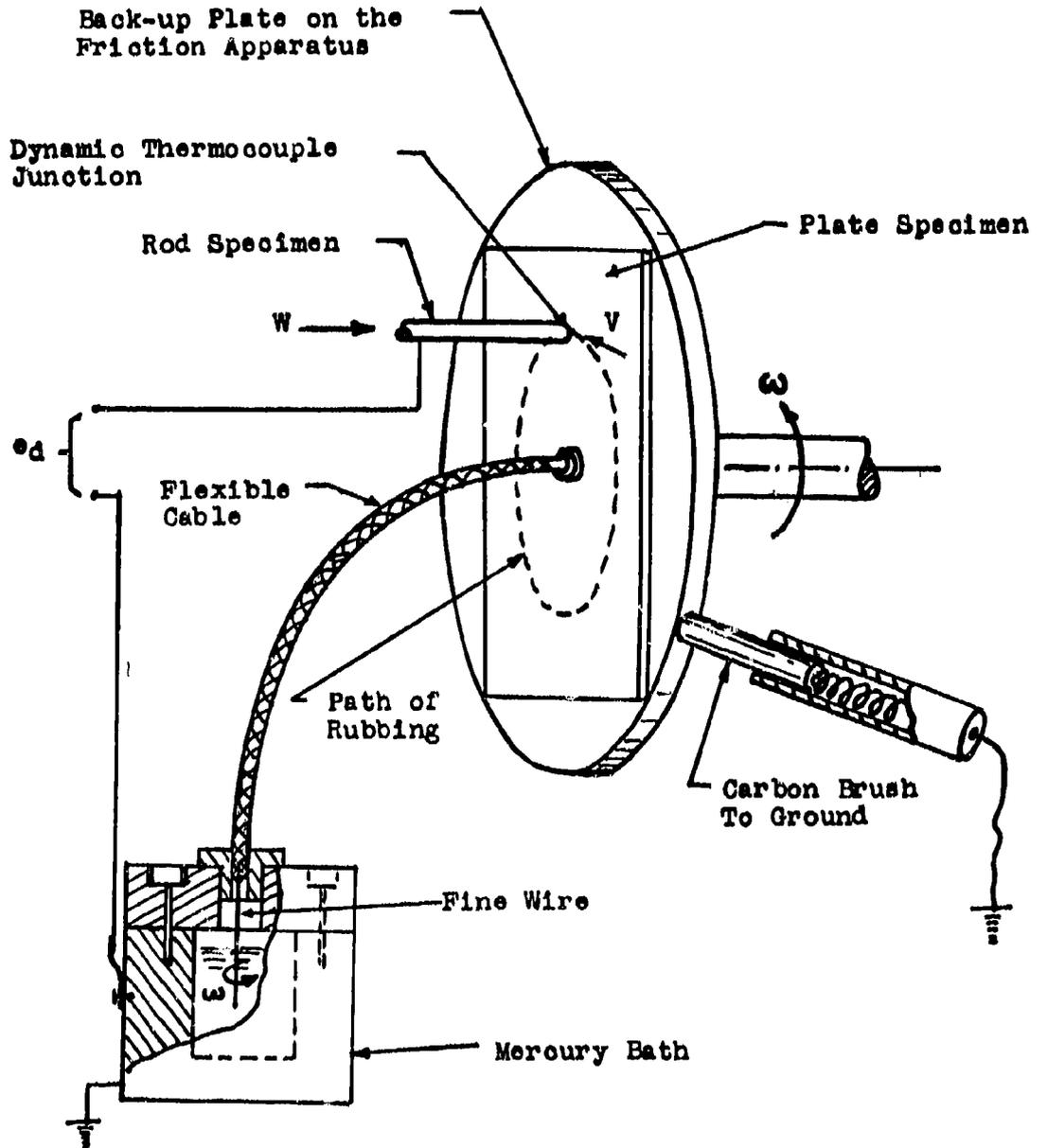
Fig. 6. Schematic Diagram of dynamic thermocouple, old apparatus.
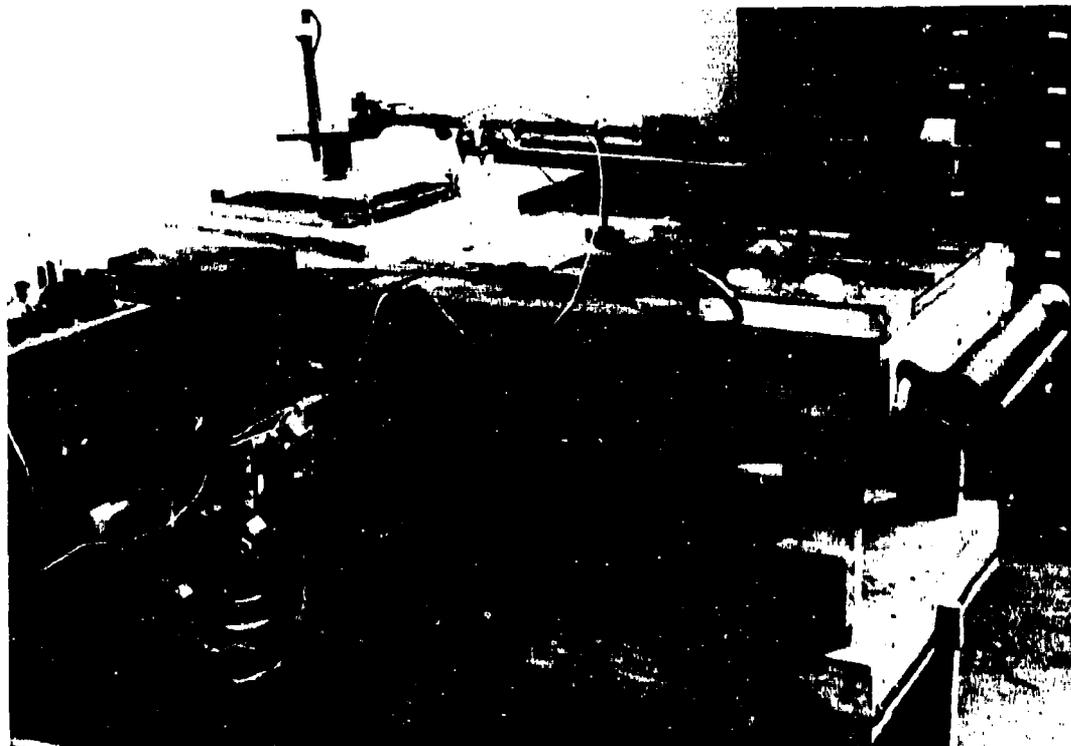
New Friction Testing Apparatus
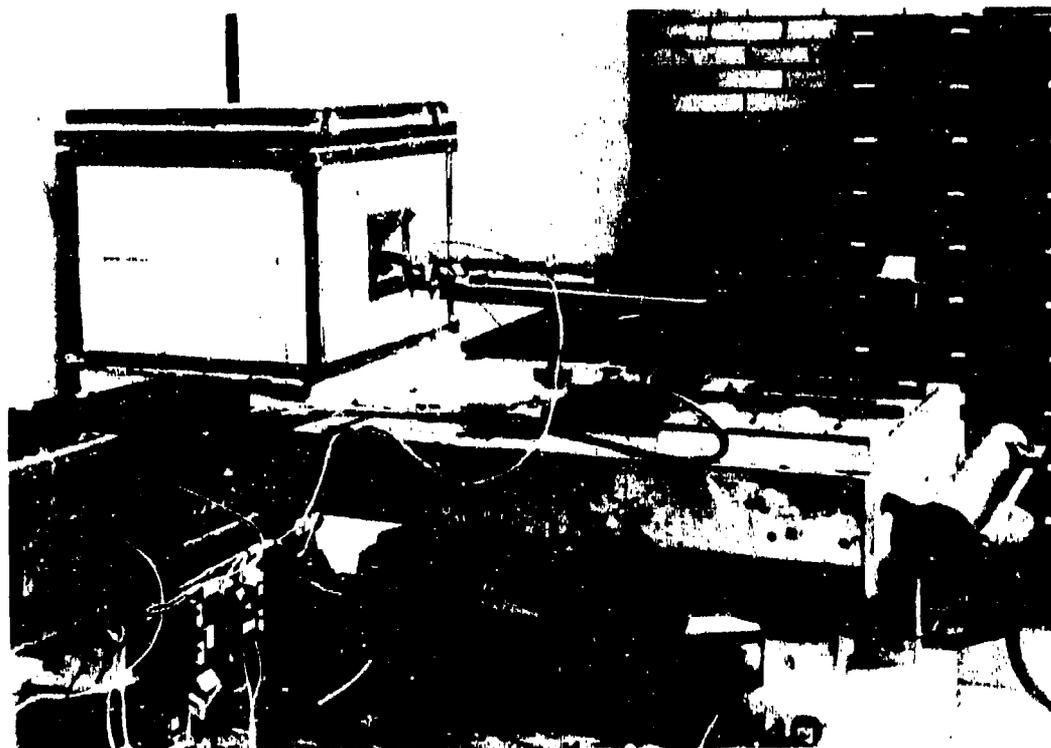
Friction Testing Apparatus Enclosed In A Furnace   FIG 7
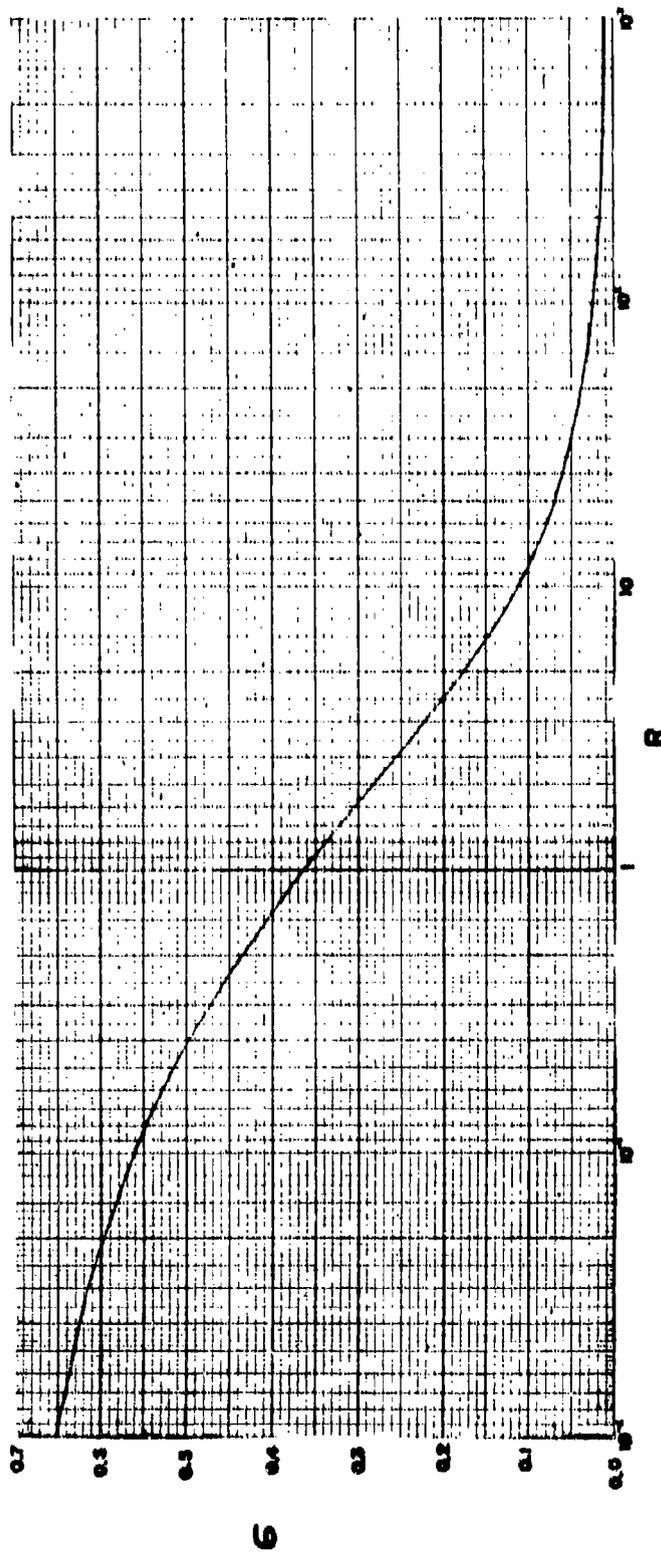
Fig. 8. CONSTANTAN SLIDER ON STEEL PLATE
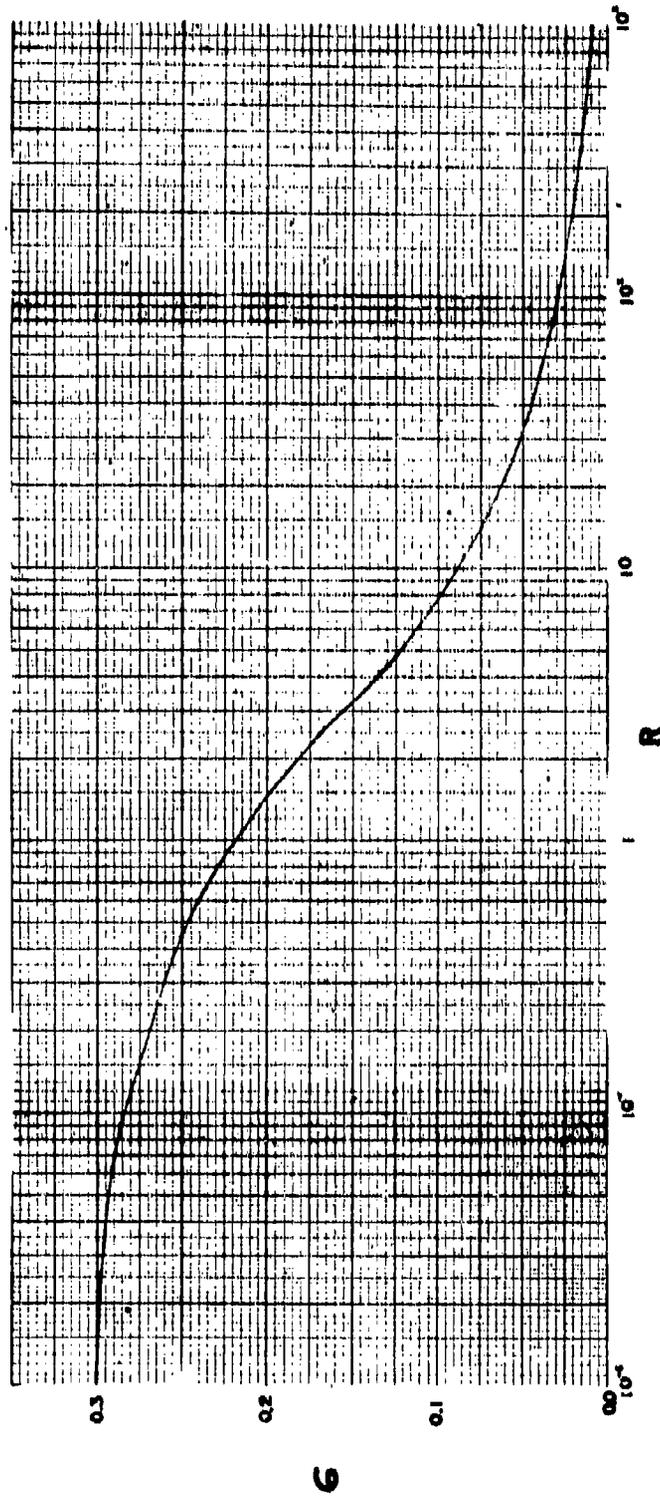
6 VERSUS R

Fig. 9. STEEL SLIDER ON CONSTANTAN PLATE
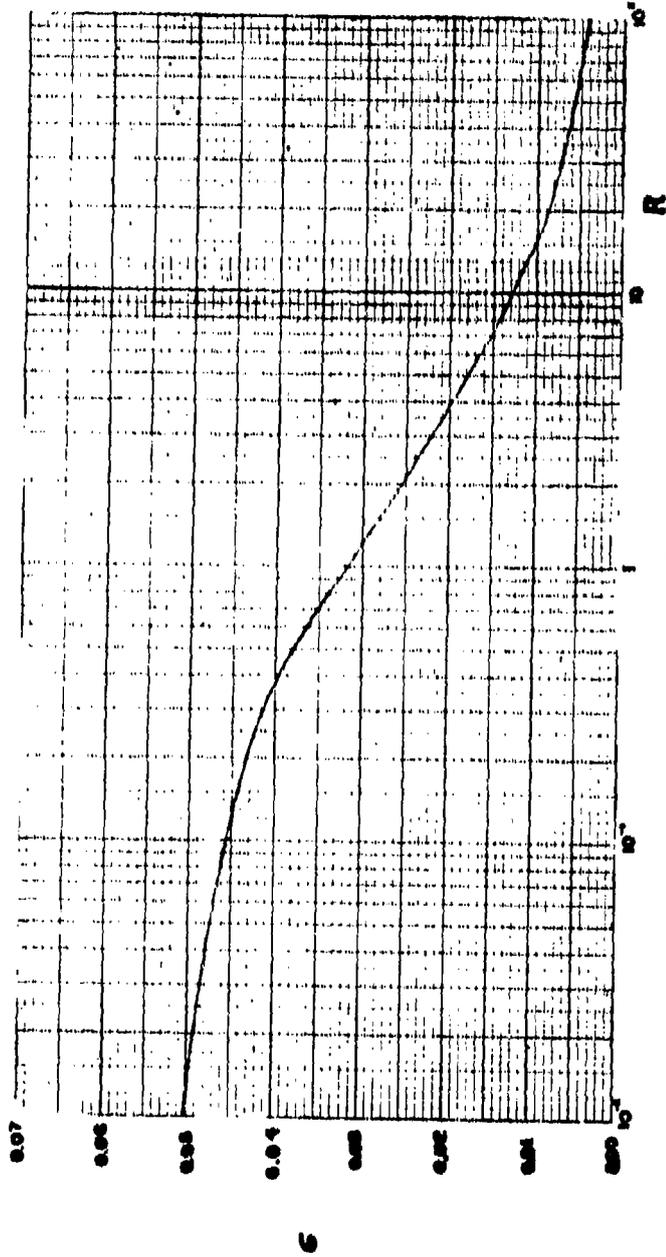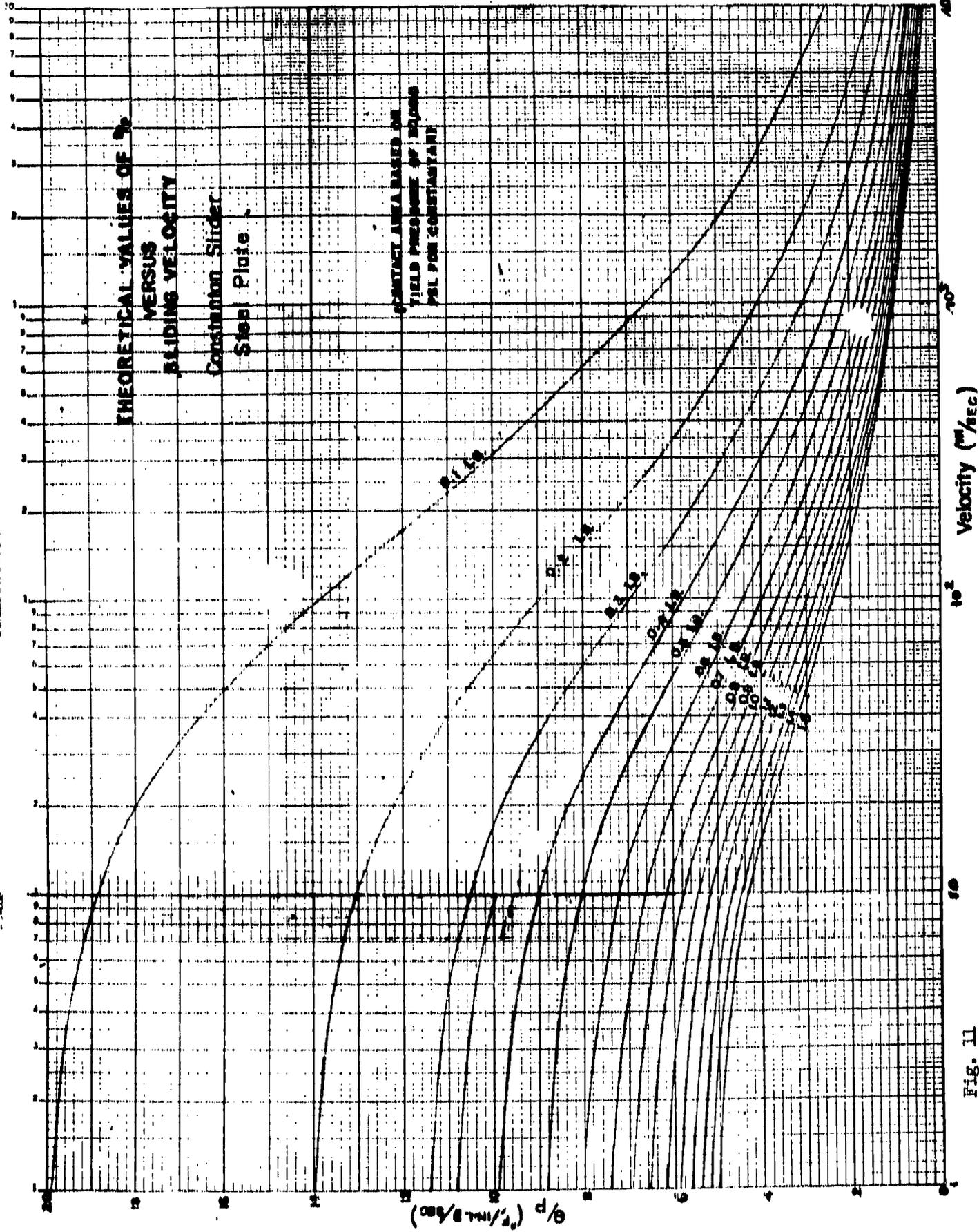
Fig. 10. COPPER SLIDING ON CONSTANTAN

Fig. 11

Fig. 12

Fig. 13

Fig. 14

Dynamic Thermocouple Calibration

Steel Slider on Constantan Plate

Cold Junction - 32°F

Static Thermocouple - 34.75 °F/mv

Dynamic Thermocouple - 34.75 °F/mv

Plate Temp. (mv) Based on Std. Fe-Constantan Thermocouple

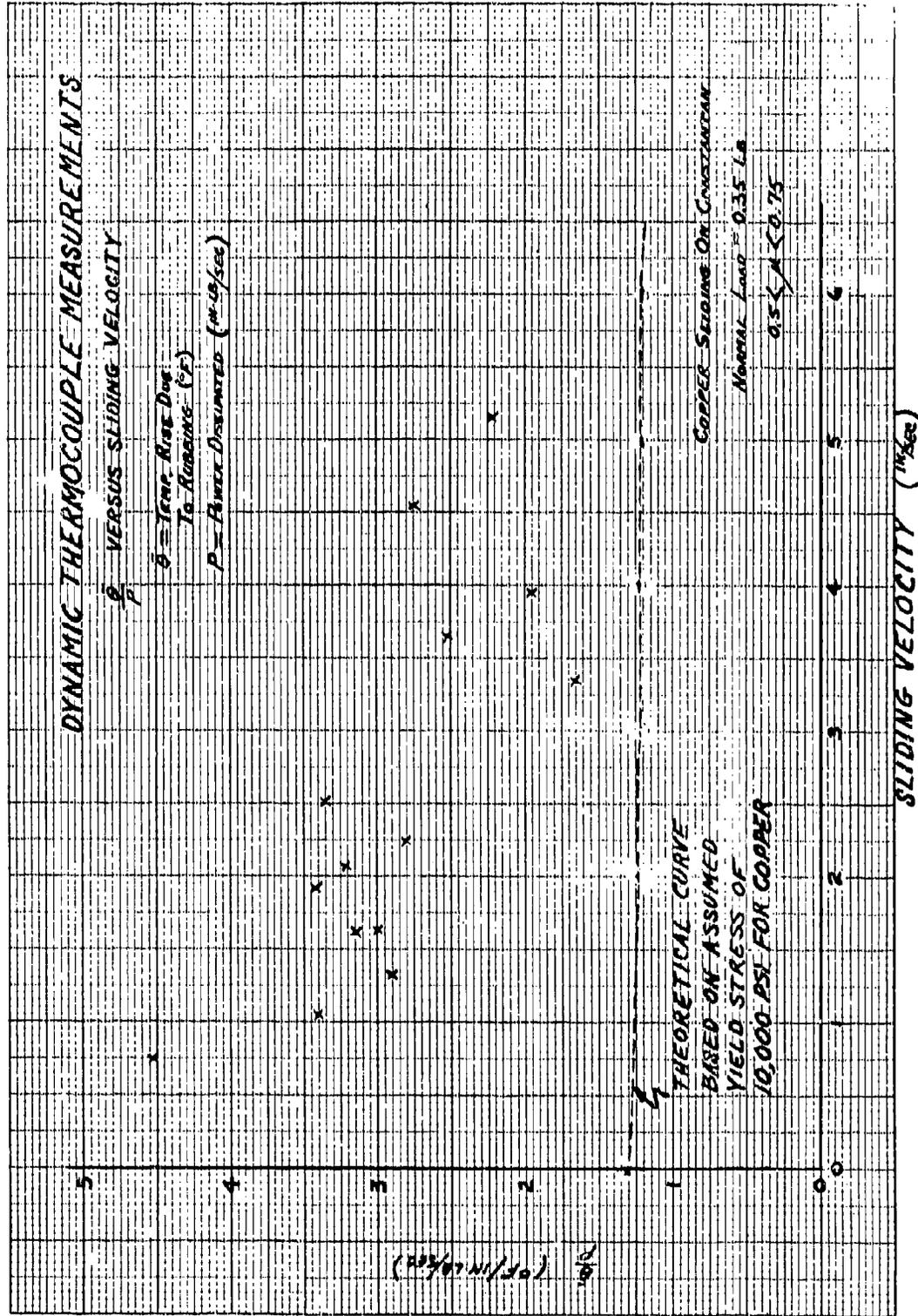Static or Dynamic Thermocouple (mv)
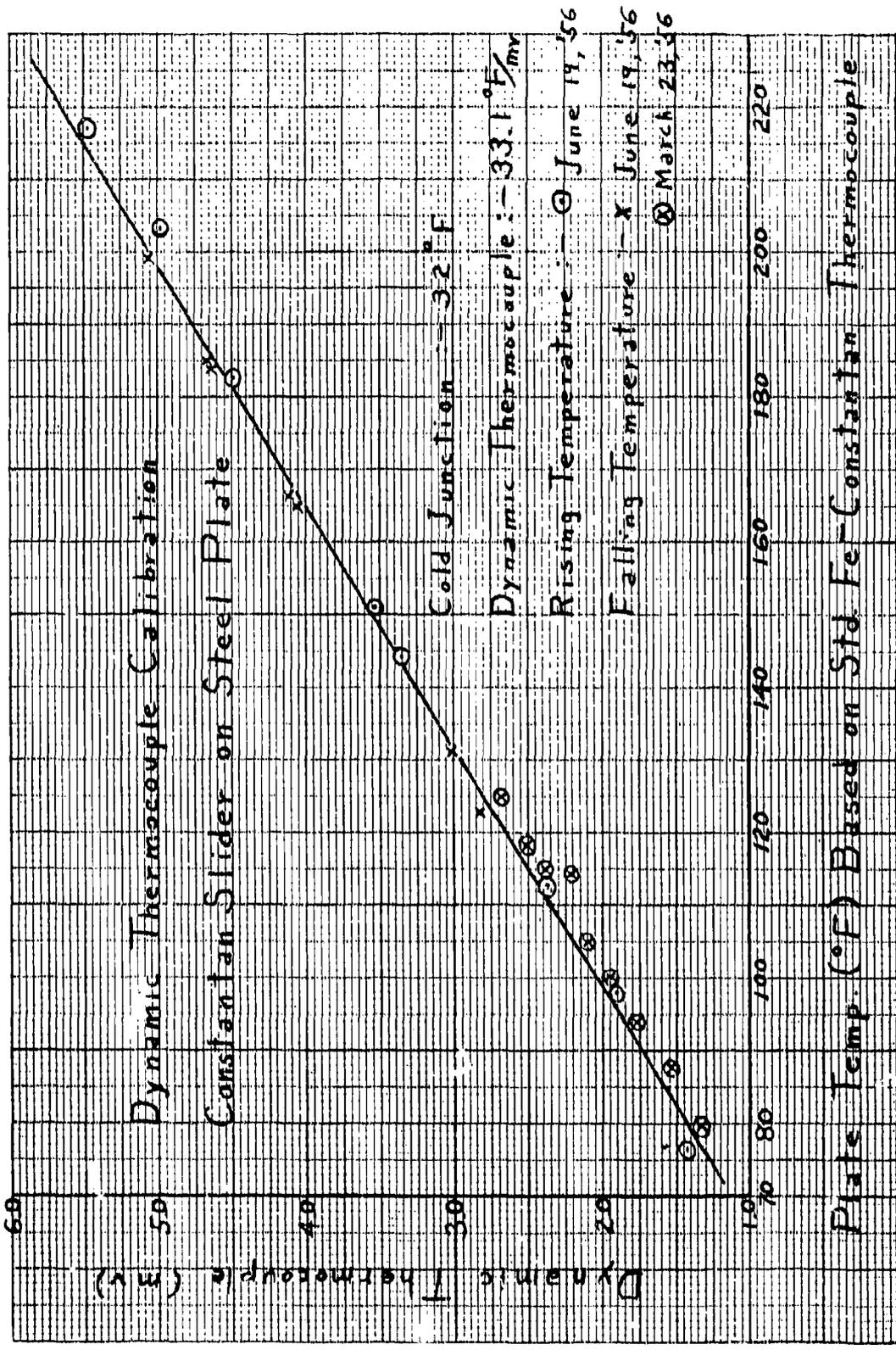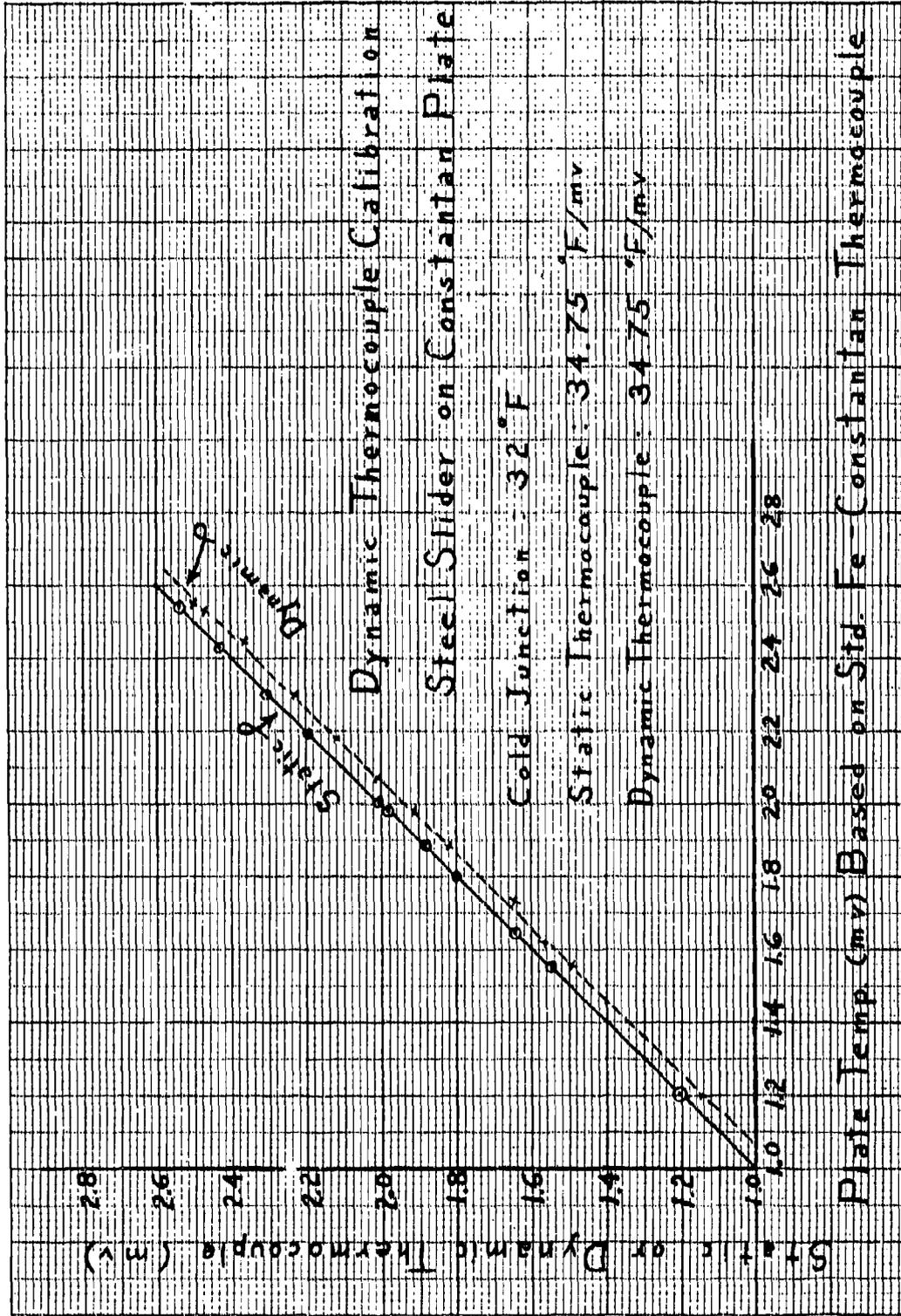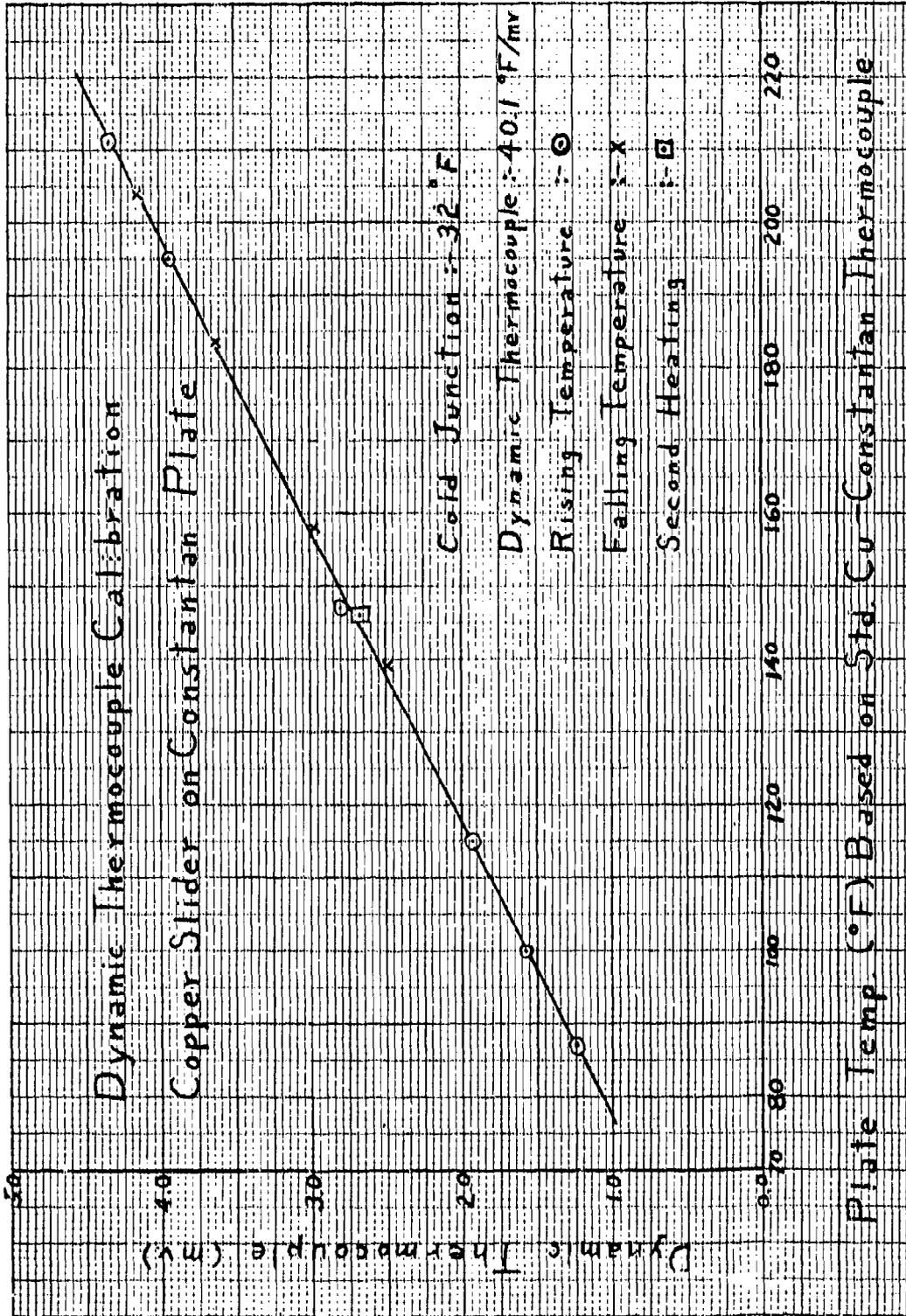
Fig. 15
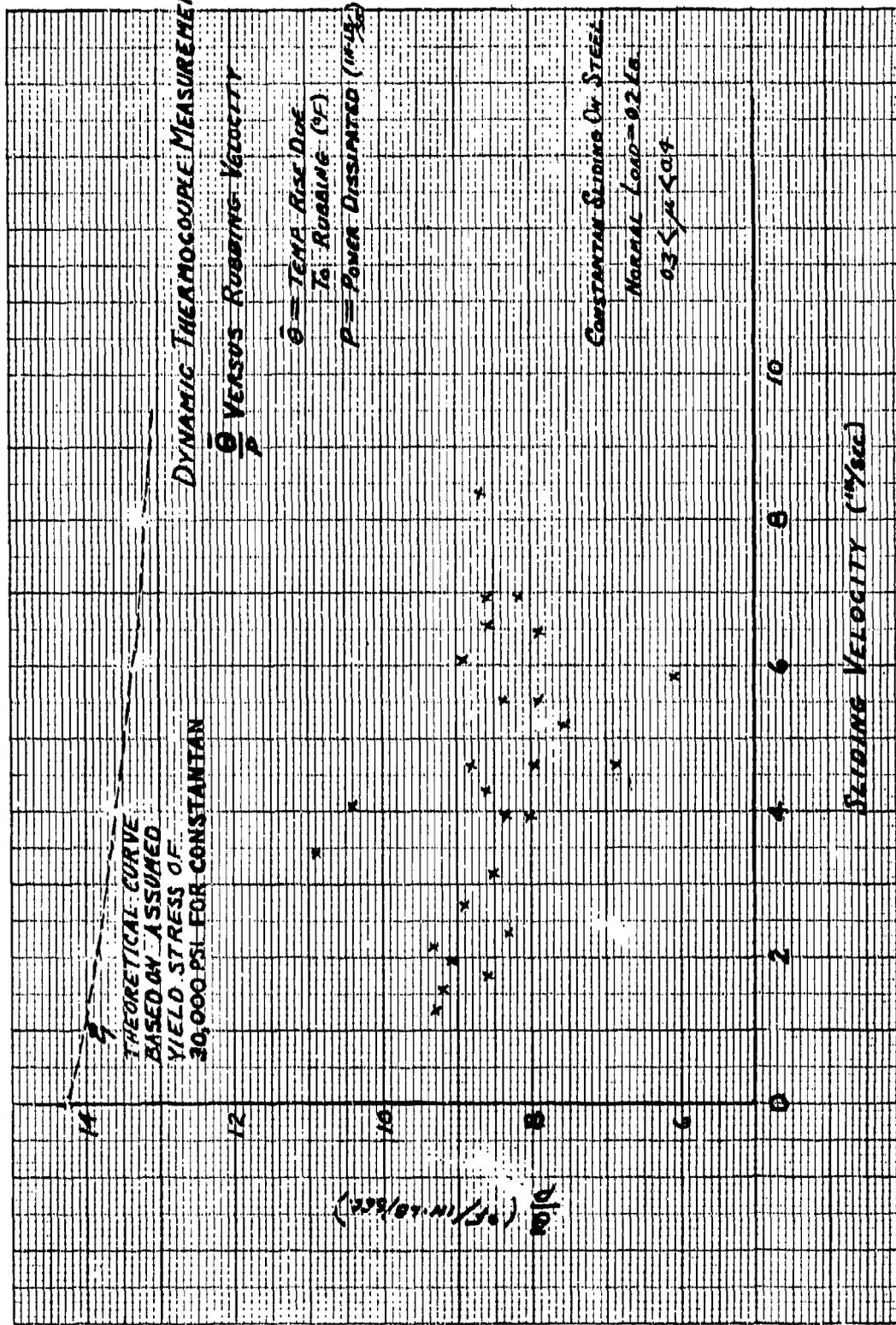
Fig. 16
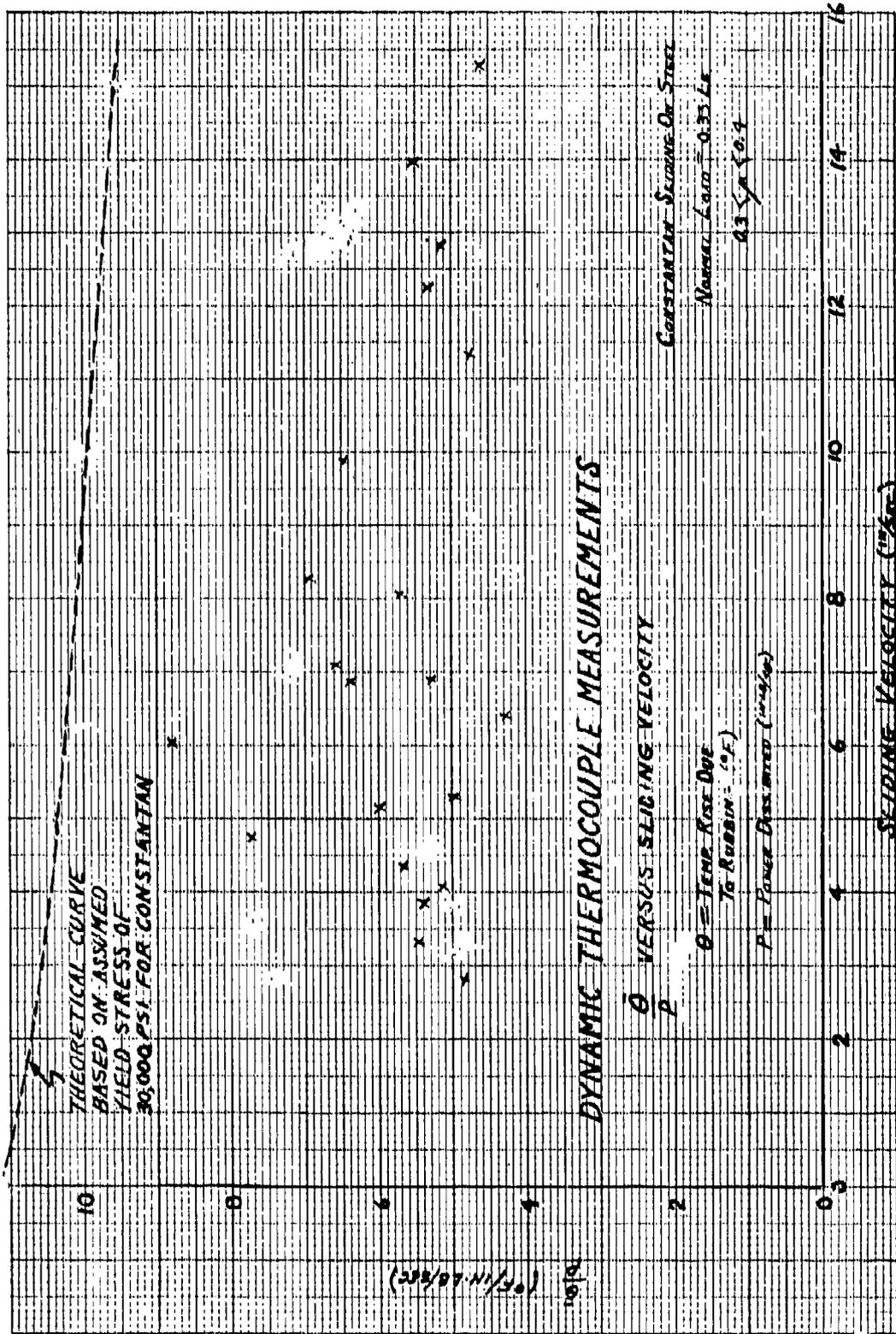
Fig. 17

Fig. 18

Fig. 19

DYNAMIC THERMOCOUPLE MEASUREMENTS
δ/P VERSUS SLIDING VELOCITY

δ = Temp. Rise Due To Rubbing (°F)   P = Power Dissipated (in·lb/sec)

⅓ lb. Load

Constantan Sliding On Steel

δ/P (°F/in·lb/sec)

Sliding Velocity (in/sec.)

Fig. 20

Fig. 21. Round Sample
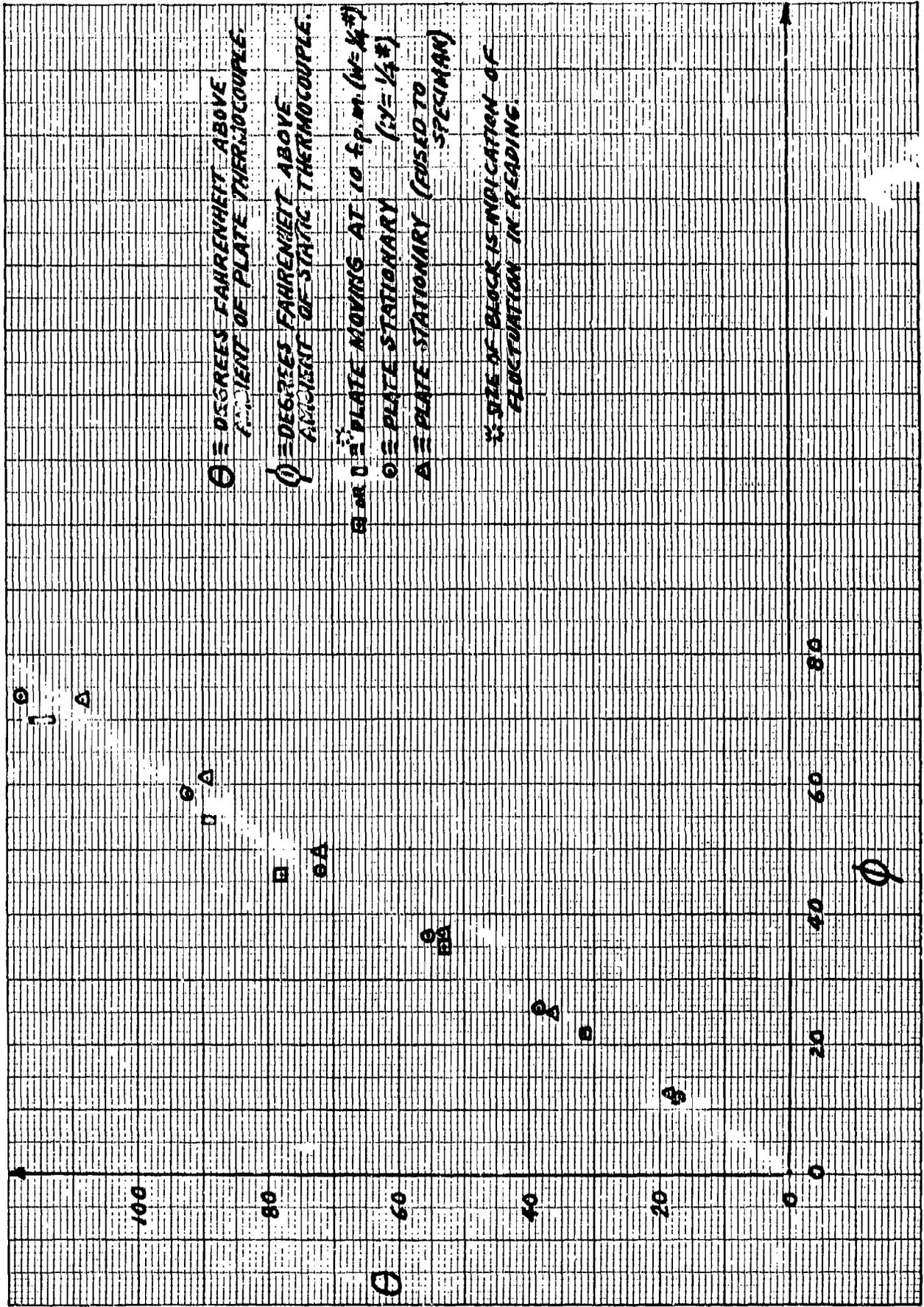
FIG. 22
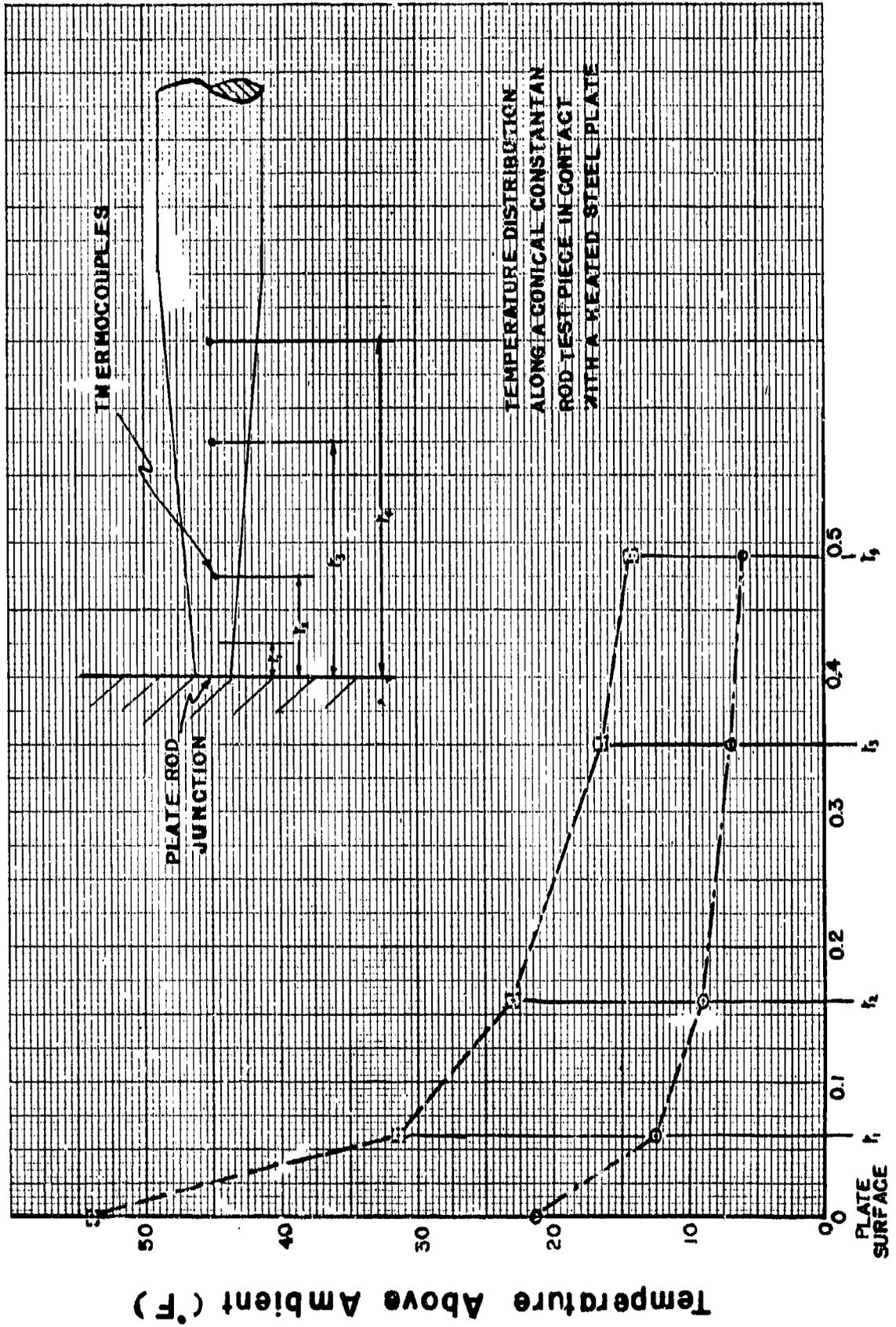
Fig. 23 Distance Of Thermocouple From Rod And Plate Junction (in)
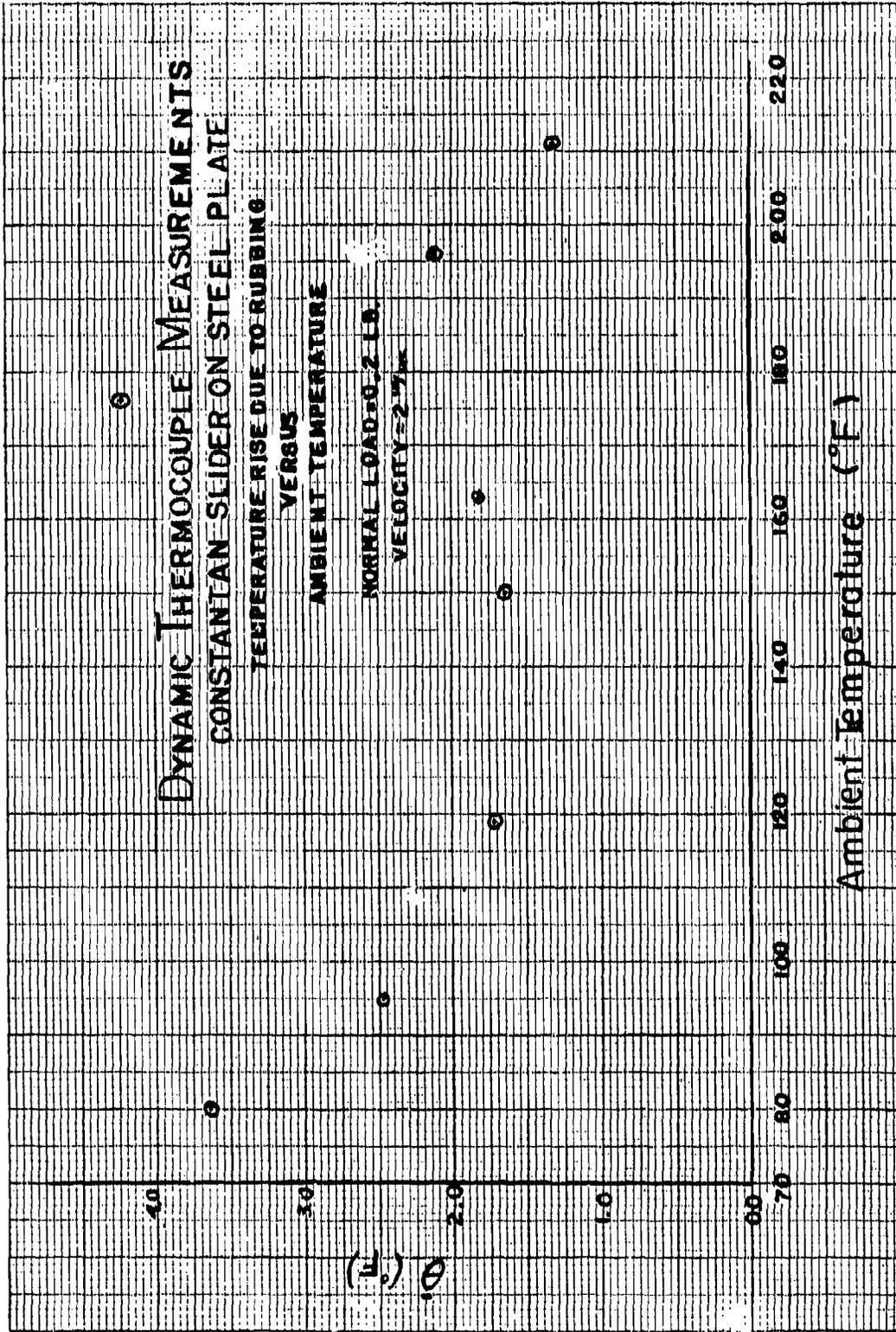
Fig. 24

Fig. 25

DYNAMIC THERMOCOUPLE MEASUREMENTS
CONSTANTAN SLIDER ON STEEL PLATE
$\frac{\theta}{\rho}$ DUE TO RUBBING
VERSUS
AMBIENT TEMPERATURE

NORMAL LOAD = 0.2 LB.
VELOCITY = 2.7 IPS

theoretical value

Ambient Temperature (°F)

$\frac{\theta}{\rho}$ (°F/IN·LB/SEC)

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Dynamic Thermocouple Measurements
COPPER-SLIDER ON CONSTANTAN-PLATE

TEMPERATURE RISE DUE TO RUBBING
VERSUS
AMBIENT TEMPERATURE

NORMAL LOAD = 0.2 LB
VELOCITY = 2.7 μ

$\mu = 0.66$

$\mu = 0.494$

$\mu = 0.573$

$\mu = 0.438$

$\mu = 0.436$

$\mu = 0.454$

θ (°F)

12.0
10.0
8.0
6.0
4.0
2.0
0.0

60   100   120   140   160   180   200   220   240
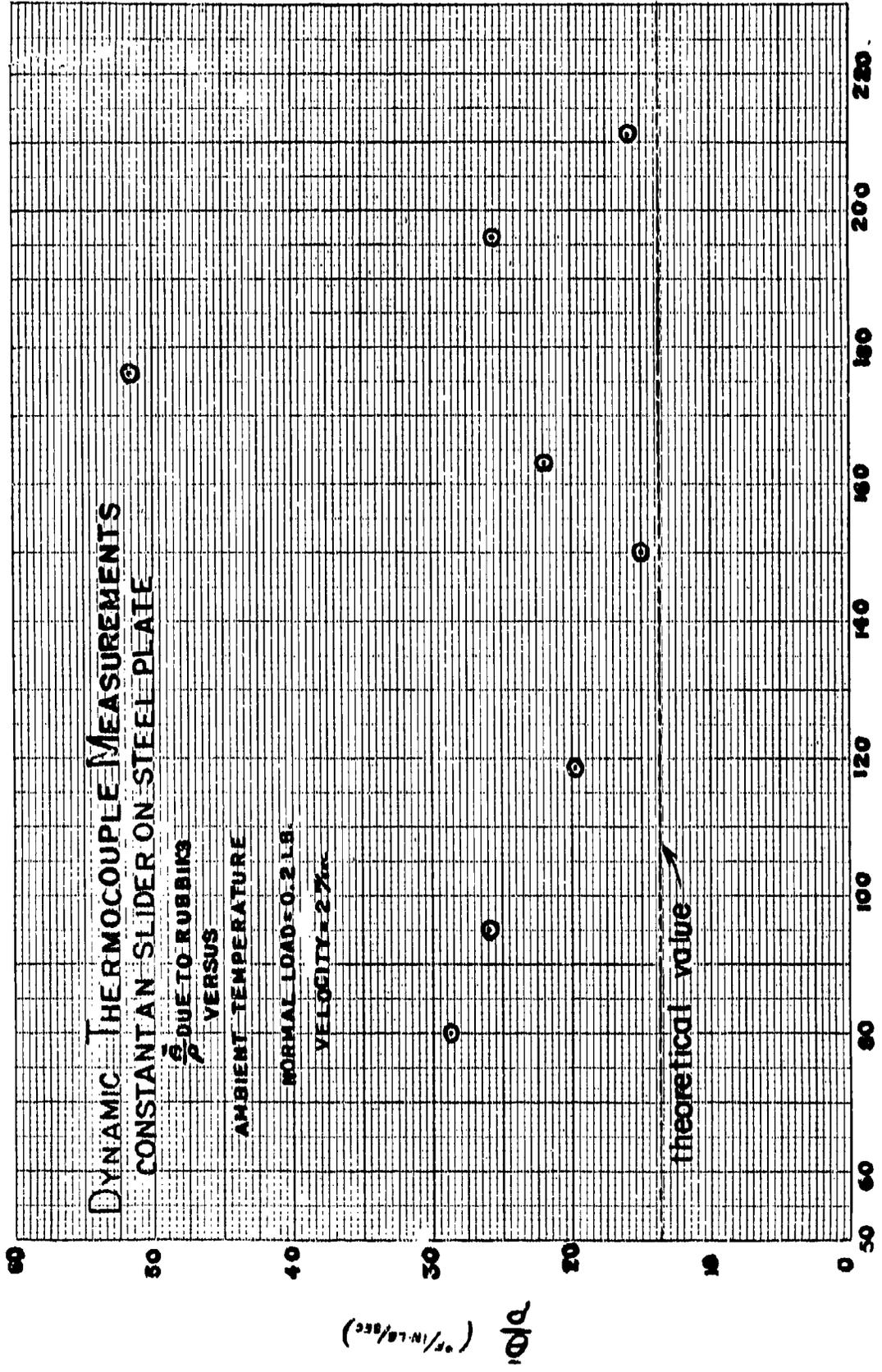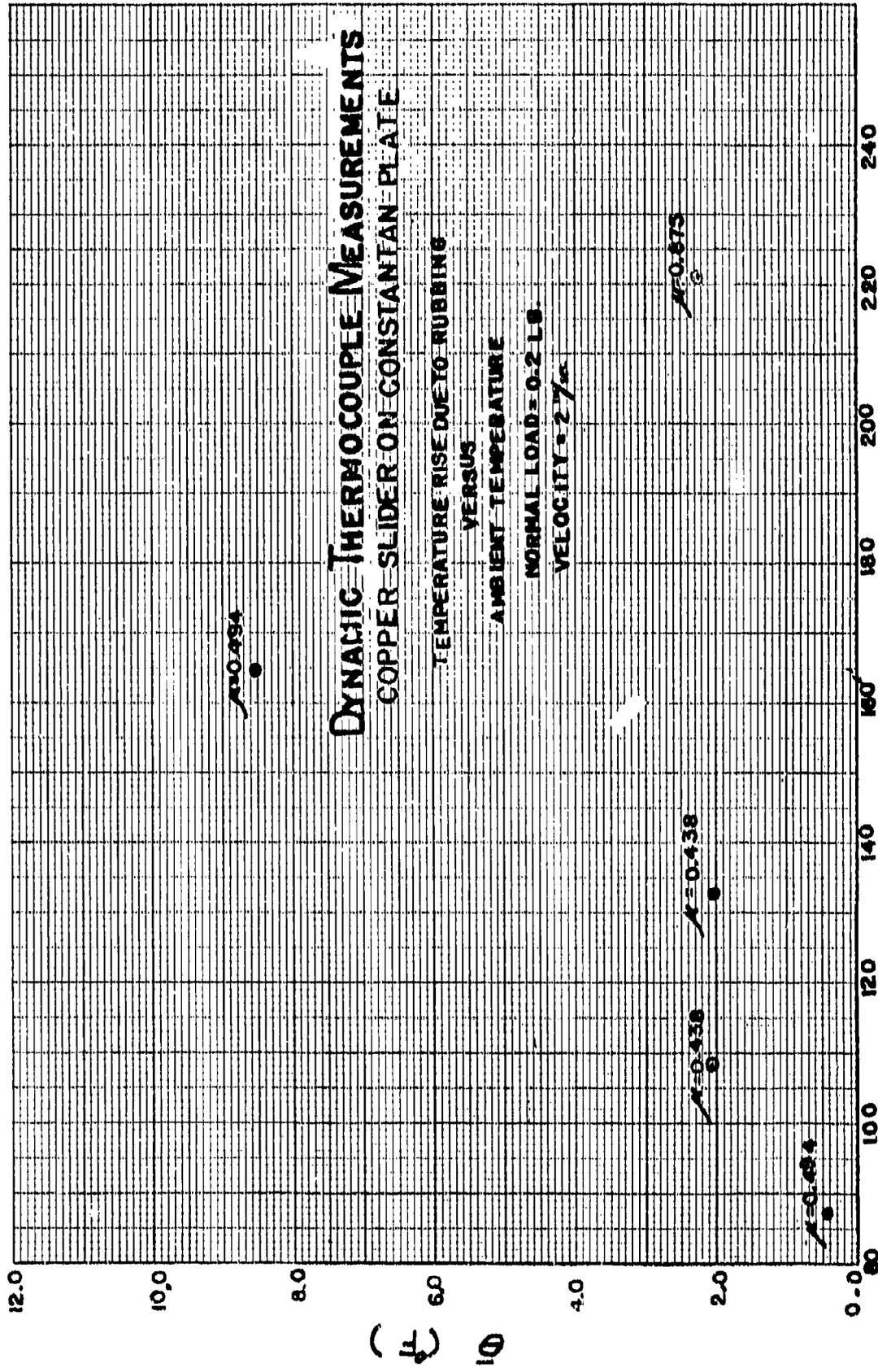
Ambient Temperature (°F)

Fig. 26
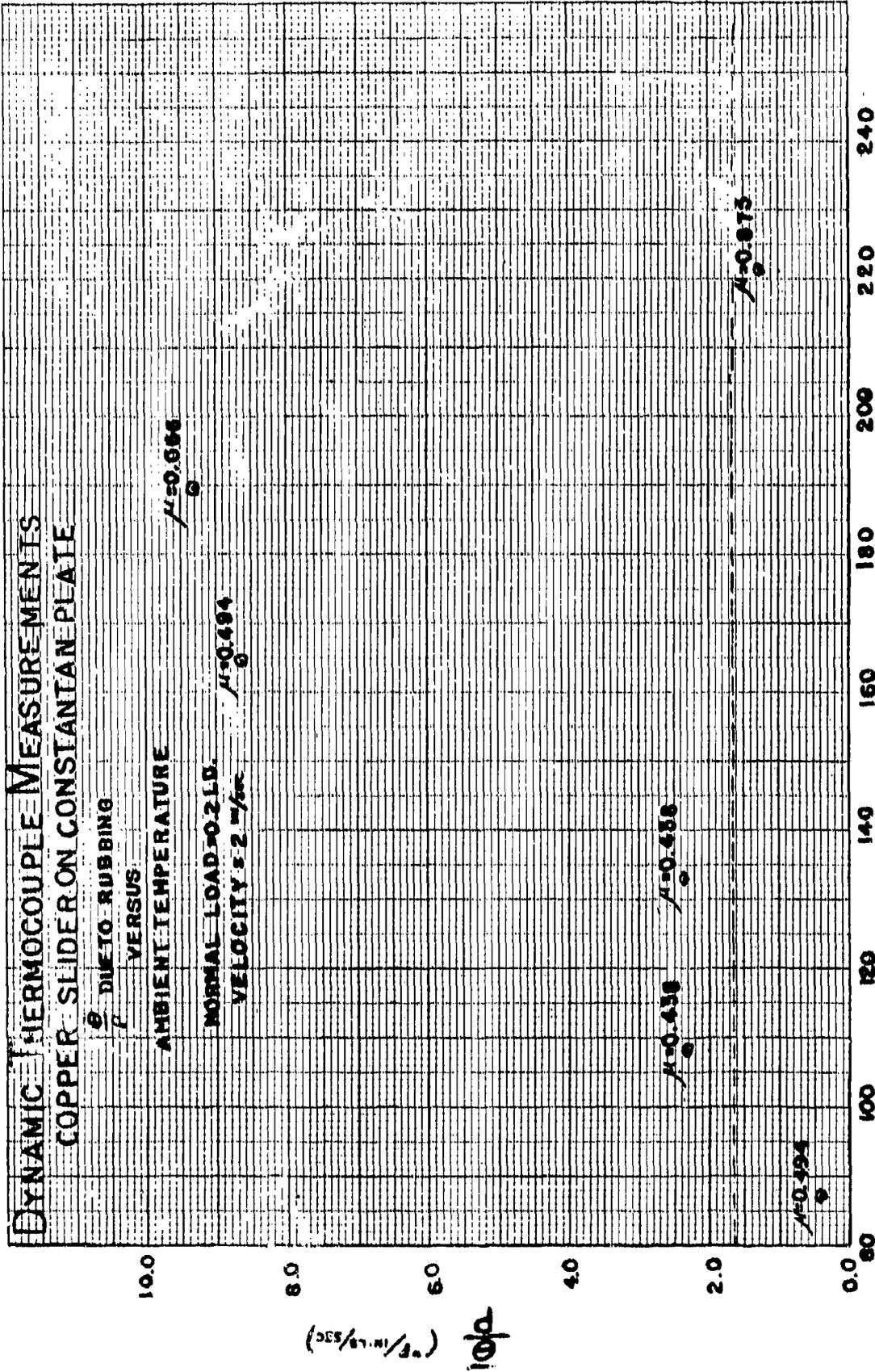
Fig. 27

Ambient Temperature (°F)

THE GALLING AND FRICTION CHARACTERISTICS OF SURFACE COATED TITANIUM

INTRODUCTION

The galling and seizing of titanium rubbing on titanium or other metals may be prevented or delayed by coating the titanium rubbing surfaces with a molybdenum disulphide or graphite impregnated resin. The coatings provide separation and possibly some lubrication between the metallic rubbing surfaces until the coating is worn off or destroyed.

This report includes the results of rubbing tests on several combinations of titanium and coated titanium rubbing on titanium, coated titanium, and on steel.

This report is preceded by the report "Investigation of Galling and Friction Characteristics of Titanium Alloys" Final Report October 31, 1954, Contract No. DA-36-061-ORD-361, Wal Report No. 401/65/37.

The work on the previous contract included tests of other surface treatments as well as for resin coatings. The results of tests of Titanium alloy RC 130 are included again in this report.

TYPE OF COATINGS

The following types of coatings have been tested and will be referred to in terms of the titles given them in this description.

Treatment A

Surface treatment.

(a) Fluoride Phosphate coated.
(b) Heat treated, 5 hours at 800°F.
(c) Molykote G Lubricant.

Treatment B

Surface treatment.

(a) Fluoride Phosphate coated.
(b) Treated with layer of MOS_2 and phenolic resin.
(c) Air dried 6 hours, cured 12 hours at 300°F.
(d) MOS_2 - Resin layer lopped to a thickness of 0.003" - 0.004".

DAG#213 Graphite - Epoxy resin formulation.

a. Spray coat with DAG#213. Before spraying the formulation was

diluted 1:1 with solvent consisting of equal parts of toluene and cellosolve acetate.

b. Air dry for 30 minutes.
c. Cure at 350°F for 30 minutes.
d. Cool in air.

Modified DAG#223 Molybdenum-disulphide-Epoxy resin formulation.

a. Spray coat with modified DAG#223. Before spraying the formulation was diluted 1:1 with solvent consisting of equal parts of xylene, toluene, methyl isobutyl carbinol and methyl isobatyl ketone.
b. Air dry for 30 minutes.
c. Cured at 350°F for 30 minutes.
d. Cool in air.

TYPES OF TESTS AND COMBINATION OF MATERIALS

Coated titanium specimens were tested with the following combinations of rubbing surfaces.

1. Uncoated titanium sliders rubbing on uncoated dry steel plates.
2. Coated titanium sliders rubbing on uncoated dry steel plates.
3. Uncoated titanium sliders rubbing on uncoated titanium plates.
4. Coated titanium sliders rubbing on uncoated titanium plates.
5. Coated titanium sliders rubbing on coated titanium plates.

RESULTS

Results of these tests are given in Table I. Coefficients of friction for the tests are shown on Figs. (28) through (37).

Coefficient of Friction.

The coefficient of friction of coated titanium rubbing on coated or uncoated metal has a range of values, 0.2 to 0.4; typical of the rubbing of dry unlubricated surfaces, but lower than the coefficient of friction for the dry rubbing of unlubricated metals where galling and seizing is taking place. As rubbing time increases and the coating tends to wear away the coefficient of friction gradually rises.

While no such tests were made it might be interesting to see what the coefficient of friction would be for a pure resin coating with no MO_2S or graphite added. Because the coefficient of friction for the coated surfaces is not very low, the added MO_2S or graphite does not appear to provide very much lubrication.

Combinations of Materials

Coated titanium sliders rubbing on uncoated steel plates showed a substantial improvement in resistance to galling over uncoated sliders rubbing on uncoated steel plates. Galling did not take place until the coating on the titanium sliders began to wear off.

Coated titanium sliders rubbing on uncoated titanium plates did not show a very good resistance to galling. Galling occurred in all tests. The uncoated titanium plates scratched easily and rapidly. These damaged parts of the plate appeared to develop protuberences which wore the coating off the slider.

Coated titanium sliders rubbing on coated titanium showed a substantial improvement in resistance to galling. The coating on the plate seemed to prevent the plate from forming large protuberences although it did not prevent the plate from scratching.

SUMMARY

The results show that when titanium surfaces are to rub on titanium surfaces, both surfaces whould be coated.

When titanium surfaces are rubbed on the surfaces of other metals, it appears that only the titanium needs to be coated if the other surface is not easily scratched or galled. The mechanism by which the steel surface did not gall was not determined. Perhaps the steel was harder than the titanium or perhaps it was able to maintain some lubrication due to iron oxide or other impurities on its surface.

Respectfully submitted,

CARNEGIE INSTITUTE OF TECHNOLOGY

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Department of Mechanical Engineering

D. W. Ver Planck

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Head, Department of Mechanical Engineering

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1. "Temperature Distribution at the Tool-Chip Interface in Metal Cutting", B. T. Chao and K. J. Trigger, Transactions ASME, Vol. 77, 1955, pp 1107-1121.

2. "The Friction and Lubrication of Solids", F. P. Bowden and D. Tabor, Oxford, 1950.

3. "Electricity and Magnetism", Francis Weston Sears, Addison-Wesley Publishing Company, Cambridge, 1953, pp 160-168.

4. "Foundations of Potential Theory", Kellog, Dover Press, 1953, p 188.

5. "Investigation of Galling and Friction Characteristics of Titanium Alloys", Eber W. Gaylord, Contract No. DA-36-061-ORD-361, WAL Report No. 401/65/37, Final report October 31, 1954.

6. "The Frictional Properties of Titanium and Its Alloys", E. Rabinowicz, Bureau of Aeronautics, U.S. Navy, Contract No. 53-213C, Annual Progress Report, January 1, 1954.

7. "An Investigation of Sliding Friction and Interface Temperature Between Two Dry Metallic Surfaces", by Fredric Ling, to Watertown Arsenal Laboratory, Contract No. DA-36-061-ORD-361, WAL Report No. 401/65/35, Interim Technical Report No. 2, May 31, 1954.

8. "Theoretical Studies of Temperature Rise at Surfaces of Actual Contact Under Oiliness Lubricating Conditions", by H. Block, Proc. the Gen. Diss. on Lubrication and Lubricants, Vol. 1, 1937, pp 227-245.

Combination of Rubbing Materials		Refer To Fig.	Load (Lbs.)	Speed (Ft/Min.)	Time For Galling (Min.)	Total Rubbing Time (Min.)	Remarks
Slider	Plate						
Titanium RC 130 A	Steel	28	0.5	100	Galling	10	All Loads Greater Than 0.5 lb. Cause Galling At 100 ft/min.
Titanium RC 130 A	Steel	29	4	100	No Galling	10	
Surface Treatment A	"	"	6	100	No Galling	10	
Titanium RC 130 A Treatment B	Steel	30	10	100	No Galling	10	No Higher Loads Or Speeds Tested
Titanium RC 130 DAG 213 Graphite	Steel	31	1	100	Galling 11 Min. Heavy Galling 58 Min.	58	30 Min. — Track Formed 48 Min. — Scratching
Titanium RC 130 A Uncoated	Titanium RC 130 A Uncoated	33	¼	100	Galling	10	Scratching - Begins At Once
°	°	52	½	100	Galling	10	°
°	°	33	¼	50	Galling	10	°
°	°	32	½	45	Galling	10	
°	°	33	¼	20	Some Galling	10	Light Single Track 6 Min.
°	°	32	½	25	Galling	10	
°	°	35	¼	12½	Some Galling	10	Light Tracks - 6 Min.

Table I

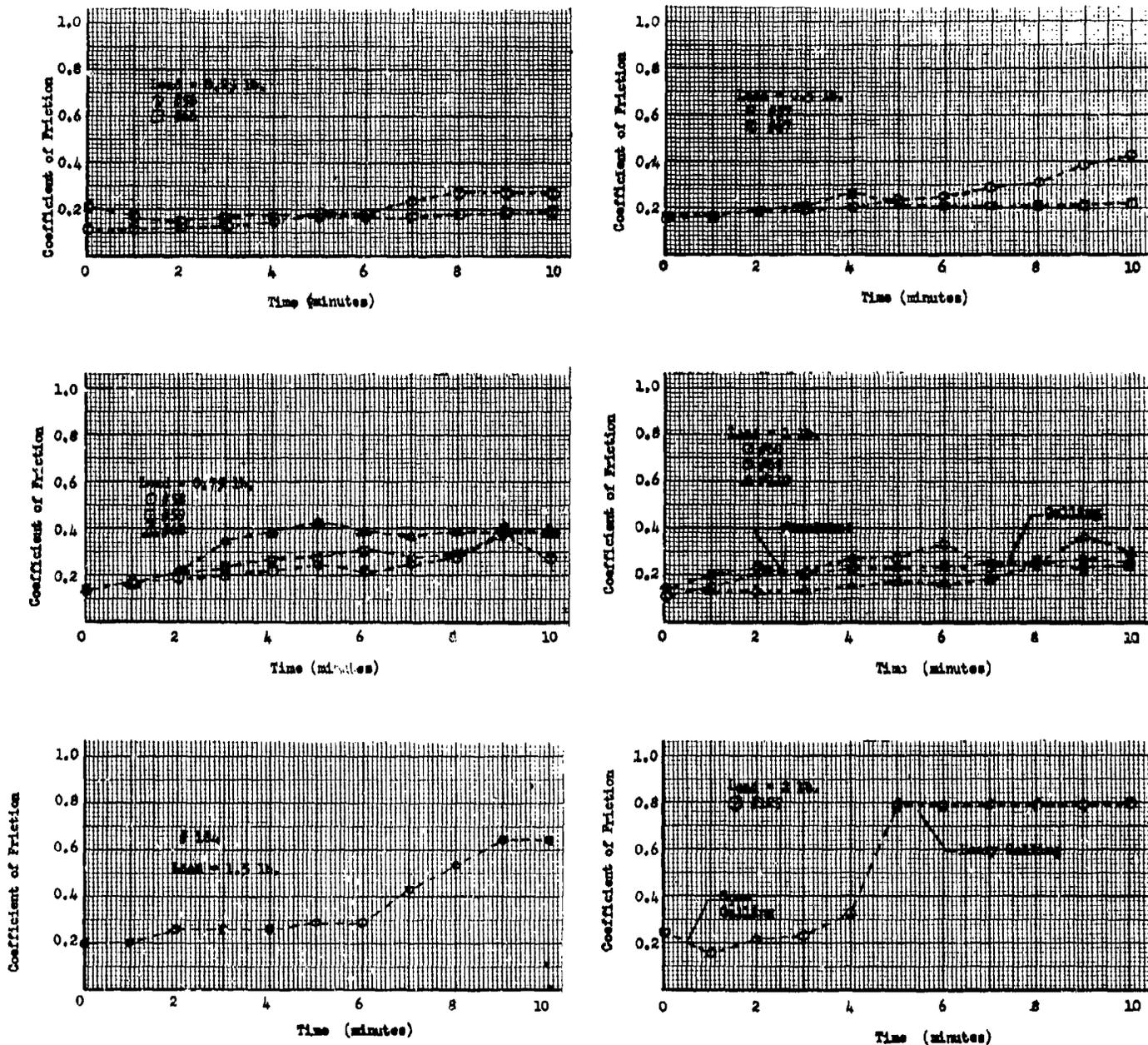| COMBINATION OF RUBBING MATERIALS		REFER TO FIG.	LOAD (LBS)	SPEED (FT/SEC)	TIME FOR TOTAL GALLING (HRS)	RUNNING TIME (HRS)	REMARKS
SLIDER	PLATE						
TITANIUM RC 130A UNCOATED	TITANIUM RC 130A UNCOATED	33	1/4	13	No Galling	10	SEVERE GALLING OCCURRED IMMEDIATELY. TEST STOPPED
TITANIUM RC 130A MODIFIED MoS₅	TITANIUM RC 130A UNCOATED	34	1	10	Galling	15	HEAVY GALLING ON EDGES OF SLIDER
DAG # 223	"	34	1	40	Galling 2 Hr.	20	THIN TRACK 1ˢᵗ HR. 2ⁿᵈ TRACK 14 HR.
"	"	34	1/2	35	Galling 11 Hr.	30	VERY HEAVY GALLING 18 HR.
"	"	34	1	25	Galling 4 Hr.	20	
TITANIUM RC 130A GRAPHITE DAG # 213	TITANIUM RC 130A UNCOATED	35	1/2	35	Galling 17 Hr.	30	2 HR. THIN LIGHT TRACKS 13 HR. TRACKS VERY HEAVY
"	"	35	1	25	Galling 5 Hr.	60	2 HR. LIGHT TRACKS 5 HR. TRACKS HEAVIER
TITANIUM RC 130A MODIFIED MoS₅ DAG # 223	TITANIUM RC 130A MODIFIED MoS₅ DAG # 223	36	1/2	10	1/2	60	1 HR. SCRATCHES FORMED IN CENTER FOLLOWED BY LIFE USED TRACK 4 HR. MUD LARGE THEREAFTER
"	"	36	1/2	20	No Galling	60	2 HR. SCRATCHES IN CENTER 5 HR. LIFE USED TRACK
"	"	36	1/2	40	No Galling	60	2 HR. SCRATCHES IN CENTER
"	"	36	1/2	80	Galling 30 Min.	60	2 HR. SCRATCHES 3 HR. CENTER BEGINS TO FORCE LIFE 30 HR. PARTIALLY FAILED AT END TEST ET
"	"	37	1	80	Galling BEGINS 13 MIN.	60	2 HR. SCRATCHES 10 HR. CENTER FORCE LIFE 13 HR. DEFICIENT
"	"	37	3	23	No Galling	60	13 MIN. FORMED CRED SLIDER LIFE USED TRACKS

Table I (continued)

Fig. 28

COEFFICIENT OF FRICTION VERSUS RUBBING TIME

Titanium RC-130A Sliders on Steel Plates
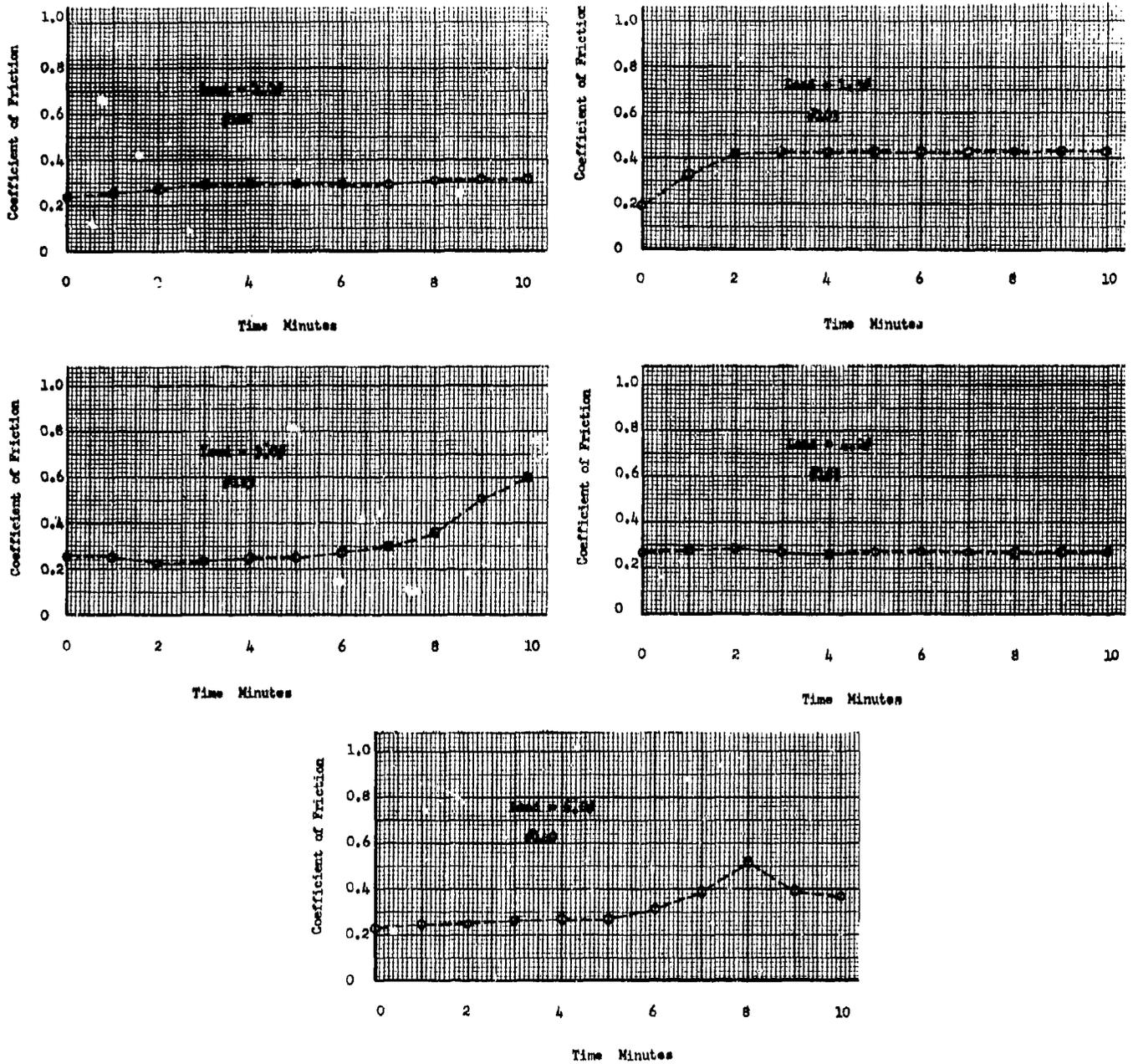Sliding Velocity: 100 Feet per Minute

Fig. 29

COEFFICIENT OF FRICTION VERSUS RUBBING TIME

Titanium RC-130A, Surface Treatment A,
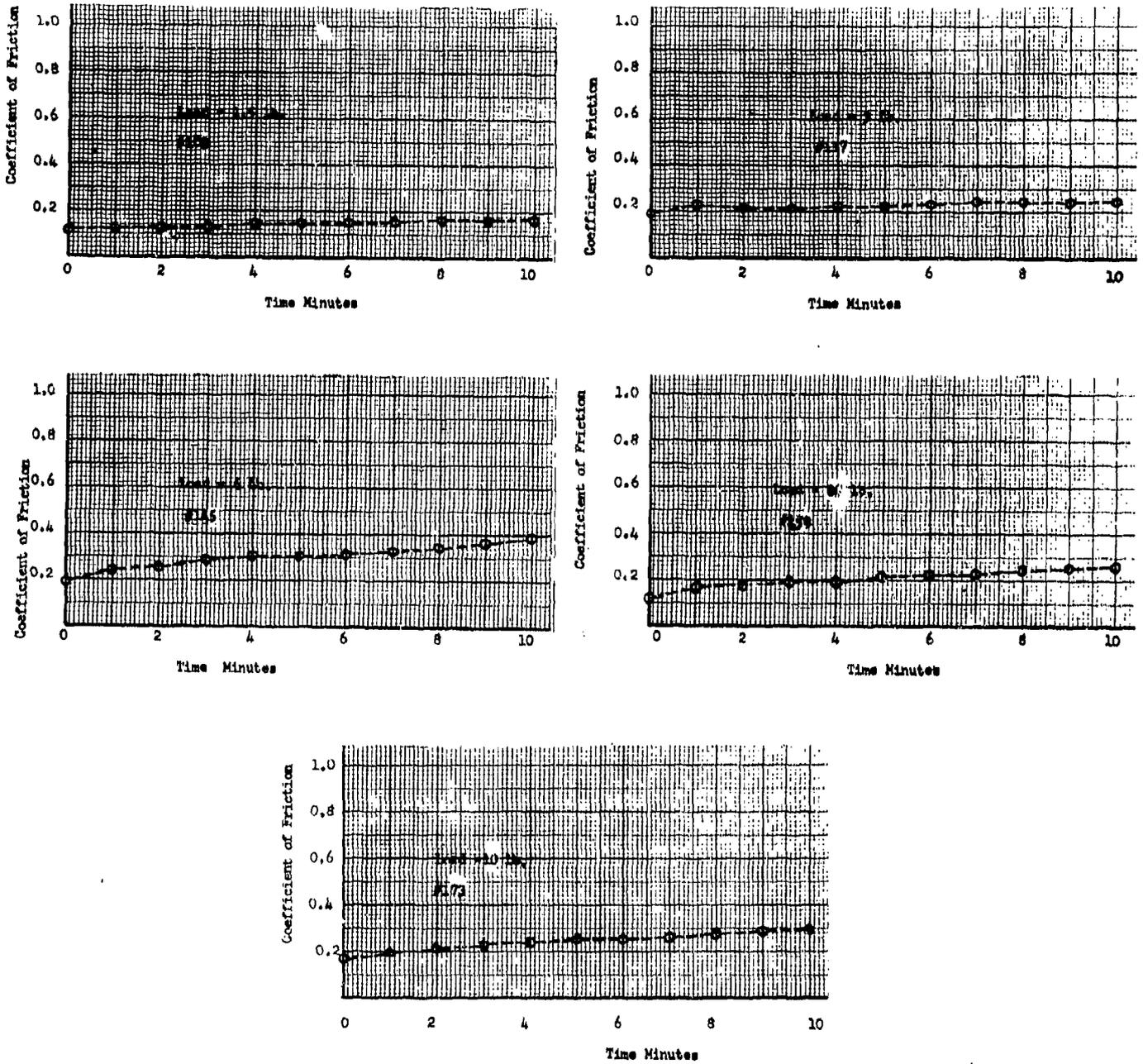Sliders on Steel Plates
Sliding Velocity: 100 Feet per Minute

Fig. 30

COEFFICIENT OF FRICTION VERSUS RUBBING TIME

Titanium RC-130A, Surface Treatment B,
Sliders on Steel Plates
Sliding Velocity: 100 Feet per Minute

Fig. 31

COEFFICIENT OF FRICTION VERSUS RUBBING TIME

Fig. 32   TIME (MINUTES)

Fig. 33

Fig. 34

Fig. 35

Fig. 36

COEFFICIENT OF FRICTION
VERSUS
RUBBING TIME

MODIFIED MO₂S DAG #223
COATED Ti RC 130 A
SLIDER AND PLATE

NORMAL LOAD ½ LB

80 FPM
40 FPM
20 FPM
10 FPM

Time—Minutes

Coefficient of Friction — f

COEFFICIENT OF FRICTION VERSUS RUBBING TIME
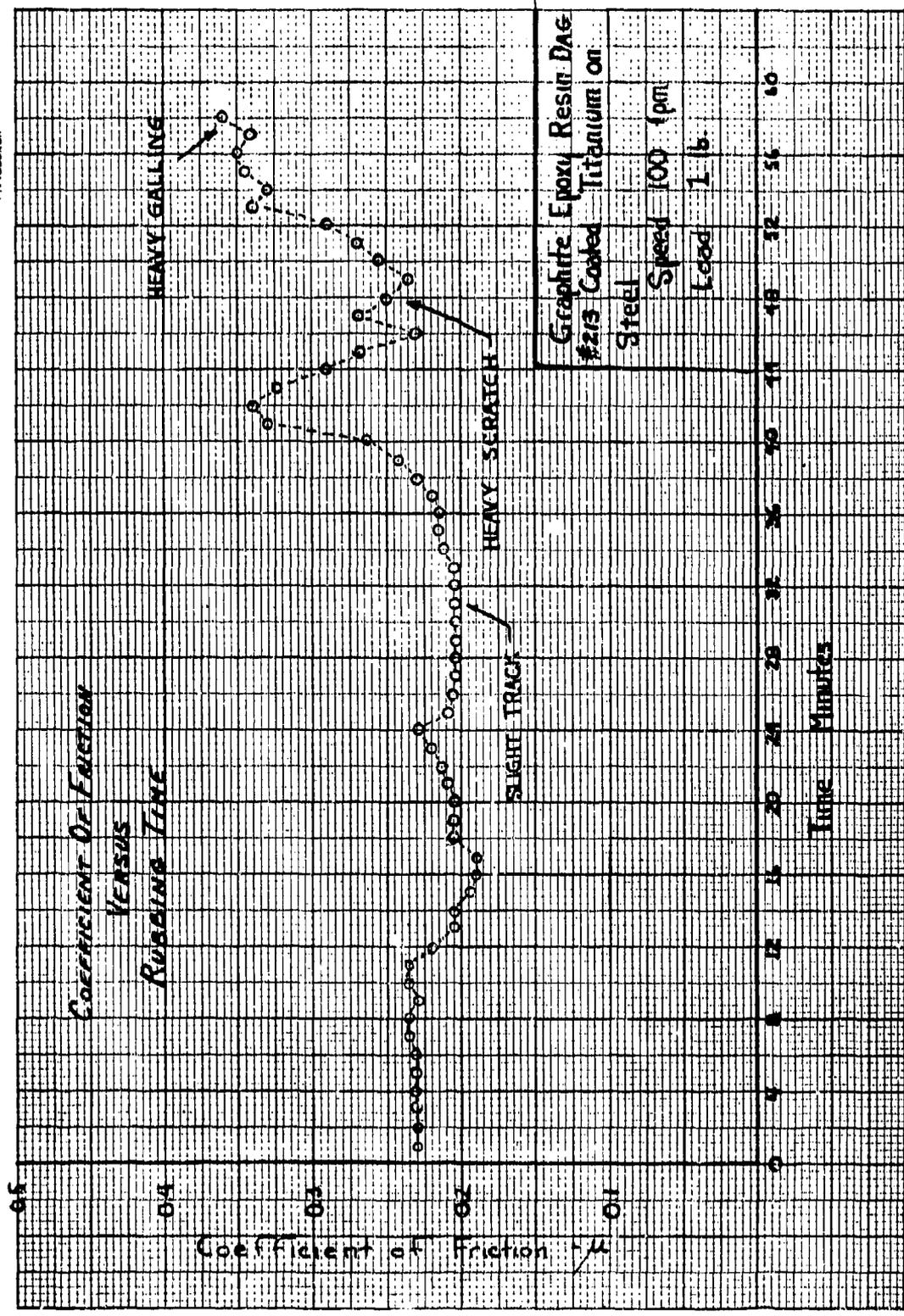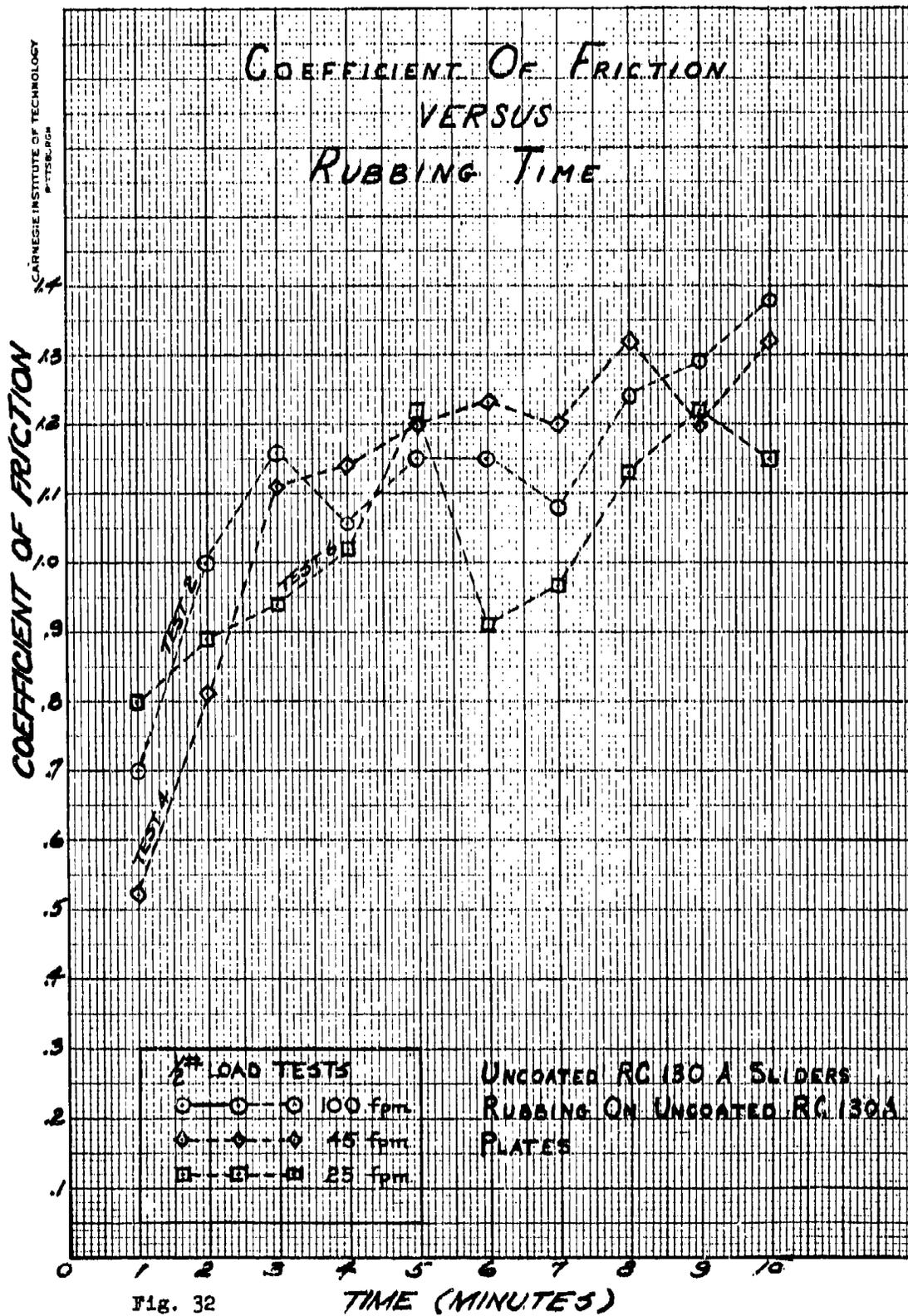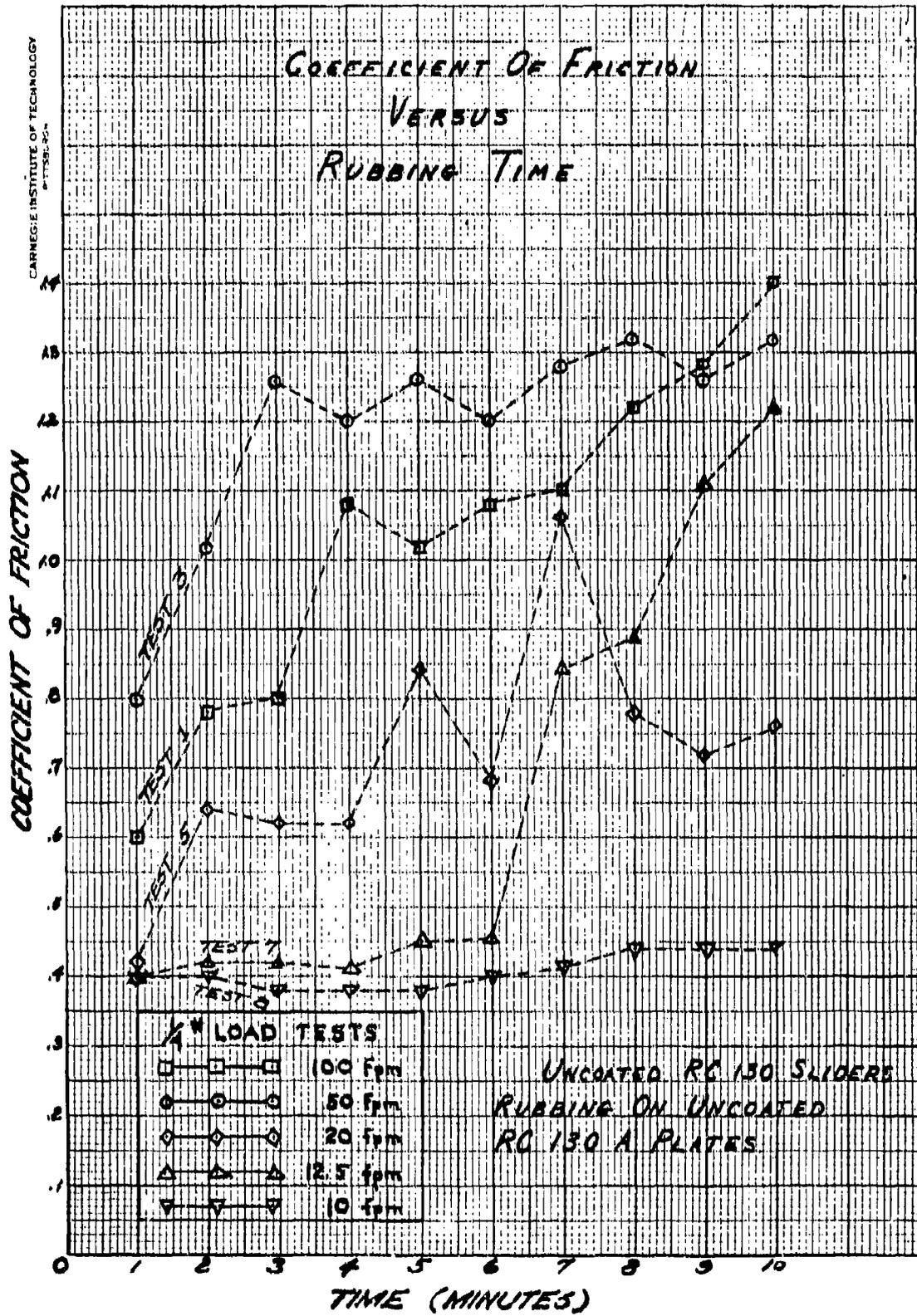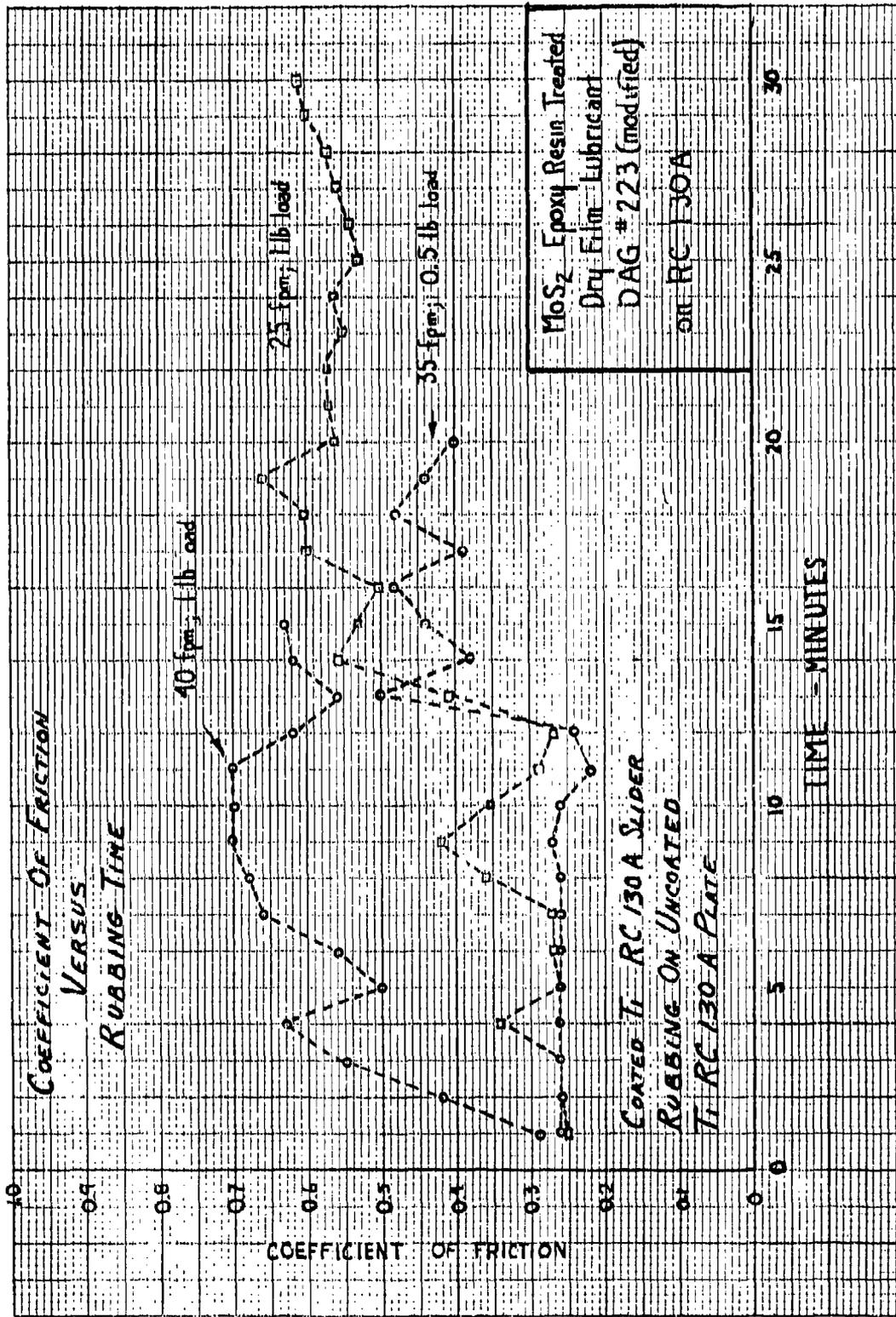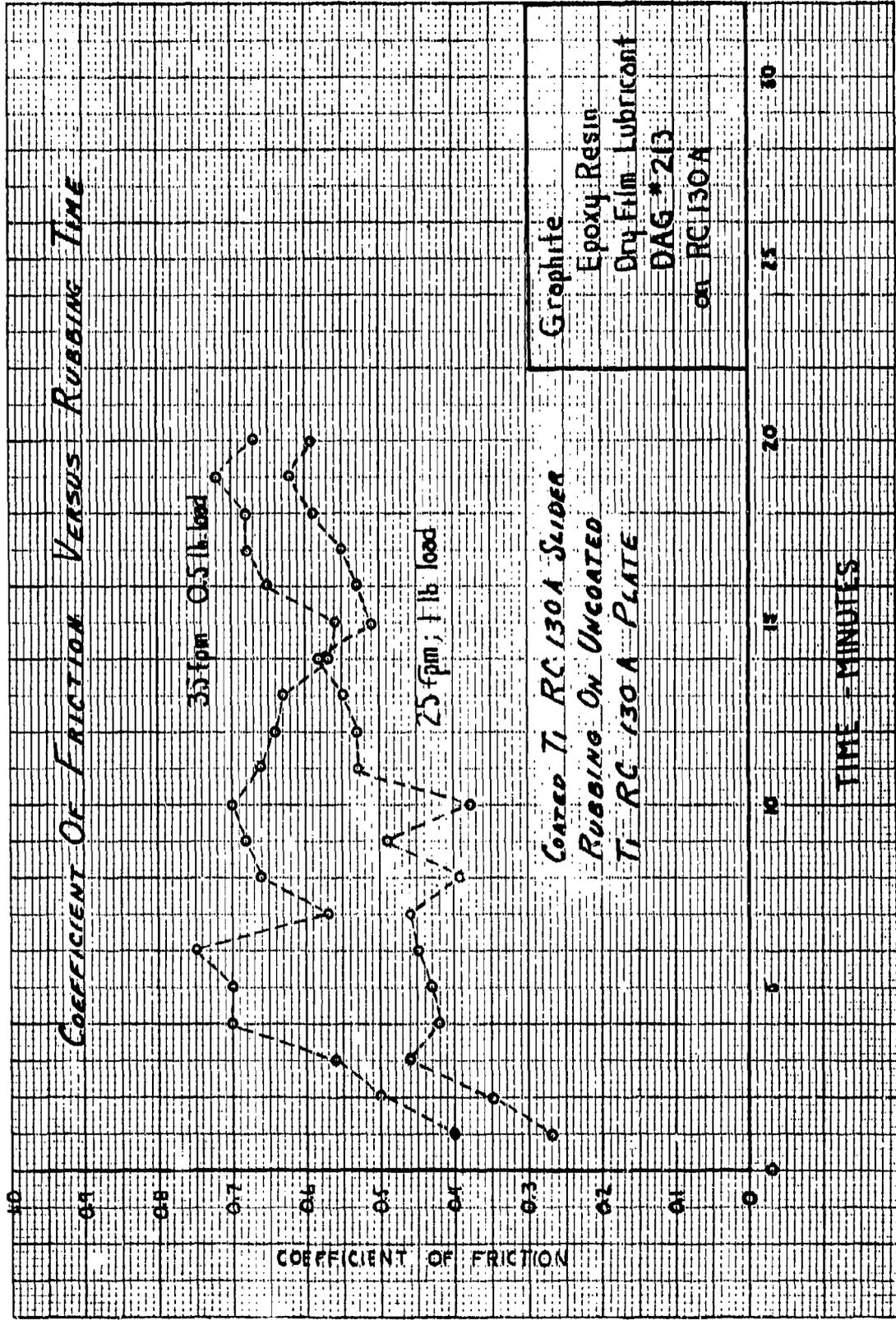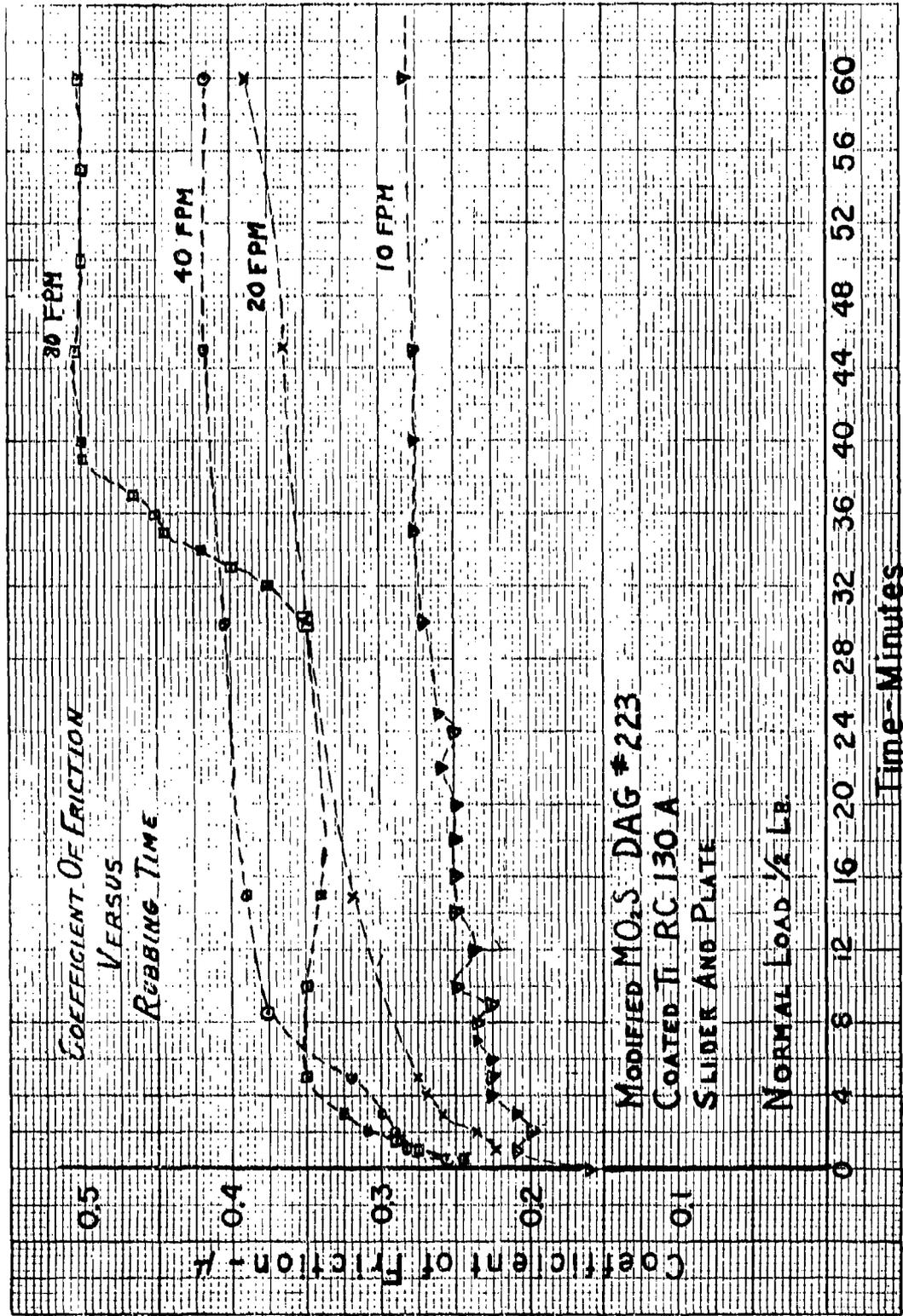
Fig. 37

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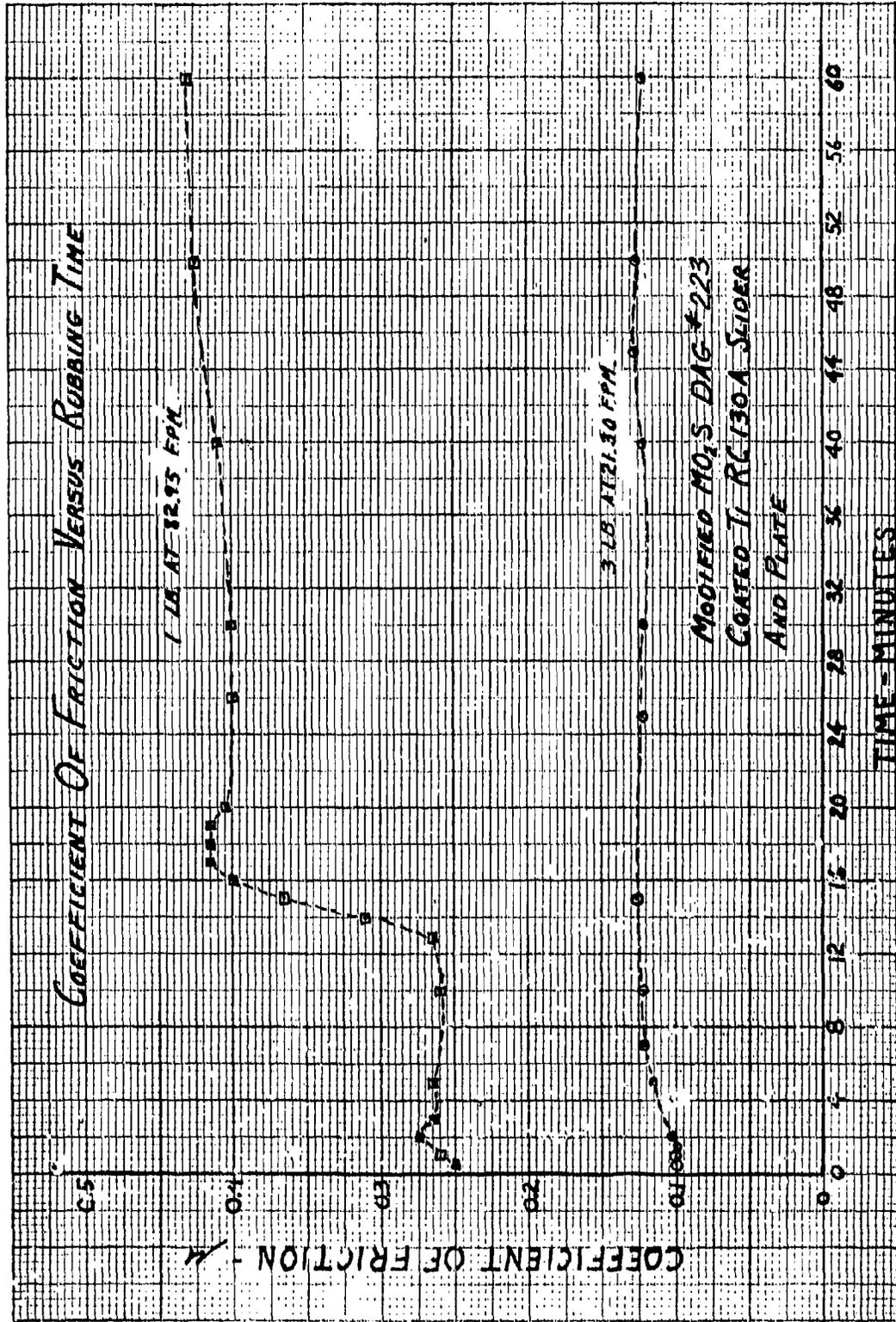| No. of Copies	To
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