THESIS

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DEVELOPMENT OF TEST SYSTEMS AND PROCEDURES FOR THE TESTING OF TYPICAL AUTOMOTIVE FRICTION CLUTCH LININGS

by

Mehmet Cemâl Beşkardeş

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M.S. Candidate : Mehmet Cemâl Beşkardeş

Mcemâl Beşkardeş

124136

SYNOPSİS

As a result of this study two new test systems have been designed and built for the testing of typical automotive friction clutch linings under the effects of varying pressures, rotational speeds, and rubbing velocities. The two test procedures developed may be used (1) in the determination of the friction coefficient and wear rates of the linings under various combinations of pressures and velocities, and (2) for the determination of the dynamic bursting behaviour of various types of disk-linings under varying rotational speeds and diameter ratios. These tests have been called (1) the Friction and Wear Test, and (2) the Disk-Bursting Behaviour Test. With the F and W Test, one can also study the variation of the maximum rubbing interface temperature under varying pressures and velocities.

The test procedures have been applied to specific molded and woven types of friction clutch linings manufactured by a Turkish plant.
Graphical presentation of experimental results indicate that the friction coefficients for both lining types drop linearly with increasing frictional energy dissipation per unit area, in the range of pressure and velocity levels applied.

The wear rates show a tendency to decrease with increasing frictional energy dissipation per unit area, at small values of pressure and velocity combinations. At high values of p and v, wear rates increase approximately in a linear way.

Woven types show better dynamic burst strength characteristics than molded types of same dimensions.

The maximum values of the frictional surface temperatures vary in an increasing exponential fashion with increasing pv-values for both lining types, in the range of p and v applied. The variation of surface temperature with the parameter $(\mu p v^{1/2})$ has been found to be linear.

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I - INTRODUCTION

A. Characteristics of Friction Clutch Linings

The purpose of a friction lining, whether fitted to a clutch or to a brake, involves the conversion of mechanical energy smoothly and under full control into heat, which must then be dissipated to the atmosphere or some other cooling medium by the clutch or brake unit.

Friction materials are designed to give a uniform and steady friction coefficient under the widest possible range of conditions. Since, however, a friction coefficient that remained uniform under all conditions colud only be obtained at the expense of other advantages, the design of a friction lining is usually a matter of compromise. The ideal friction lining would have the proper friction coefficient which would remain constant for the life of the lining under all operating conditions of pressure, speed, temperature and humidity. This ideal lining would not score the contact surface, would operate quietly, wear slowly, and be free of any offensive odour.

The ability of a clutch system to provide rapid engagement is proportional to the relative friction coefficient of the lining employed. The absolute frictional efficiency of the lining required for effective engagement is a function of the mechanical efficiency of the clutch mechanism.

The friction coefficient of automotive clutch linings generally will be in the range from 0,2 to 0,5 though usually manufacturers describe the linings

as having high, medium, or low friction values rather than specify the value numerically. Allining with a coefficient lower than 0,2 would normally be unsatisfactory since it would require excessive clutch pedal pressure during clutch engagement, whereas a lining with a coefficient as hight as 0,5 would be undesirable because it would be too sensitive or erratic and would wear out too quickly.

All linings disintegrate at the friction surface from the high temperatures developed when they convert energy of motion into heat energy. The chemical and physical changes that take place cause either an increase in friction, called "Build-up", or a decrease in friction, called "fade". Satisfactory linings will fade slightly upon each application, but will come back immediately to their initial state upon cooling. This property is called "good recovery".

The friction coefficient of a lining may change with age. When it decreases, a condition known as "hard pedal" is produced and, in some cases these linings have a tendency to score the contact surface and produce noise.

Linings must have a reasonable wear life. It is necessary that they also wear at a low uniform rate since such linings tend to produce a more consistent engagement action due to the fact that the friction surface is continually being renewed.

Three factors to be considered in the choice of a friction material are, therefore,

(1) the basic friction coefficient required, which determines the clutch operating force

(2) fade resistance, the importance of which varies with the effectiveness of the cooling available and the temperature reached in operation

(3) the wear resistance of the lining, which is considerably influenced by the operating pressure and velocity.

These three factors have to be considered for every example of clutch, each of which has its own peculiarities, such as the amount of energy to be absorbed, the amount of heat to be dissipated, the rubbing speeds between mating surfaces, and the applied pressure and friction.

For each particular application there will be one type of friction material which is the most suitable.

B. Characteristics of Automotive Friction Clutches

A clutch may be defined as a machine element which permits the connection and disconnection of shafts. The clutch is required to transmit motion from a power source to a stationary component until the two are moving at the same speed.

In automotive engines where the flow of mechanical power is controlled through clutches, efficient performance and flexibility of operation depend largely upon clutch characteristics. Currently, the trend toward increased power output and higher speeds impose increasingly critical requirements on these vital machine components.

The most obvious requirement of a friction clutch is that it should be capable of transmitting, without slip, the maximum moment that can be applied to it once full engagement has taken place.

When starting a vehicle from rest, the engine may be running at quite a high s a high speed, with the output shaft stationary. Engaging the clutch will, during the slipping period, dissipate part of the kinetic energy as heat. The metal mating plates and the size of the clutch have to be designed in line with the particular frequency of application to ensure that the temperature rise does not become excessive.

The typical axial friction clutch used in automotive applications consists essentially of two opposing friction surfaces (or mutliple sets of them), one attached to a driving member and the other to a driven member.

Fig. (1) shows a typical single plate clutch arrangement and the method of actuating it. The clutch consists of the following main components :

(1) driving plate, (flywheel) [illegible]

(2) pressure plate

(3) spinner plate

(4) operating springs

(5) throw-out bearings

(6) release toggles

(7) cover plate

The whole of the clutch mechanism is rotating with the input member. The driven shaft passes through the centre of the clutch shell and is splined at the end to take the spinner plate which is situated between the driving and pressure plates. The spinner plate is lined on both sides with friction linings (also called " facings ").

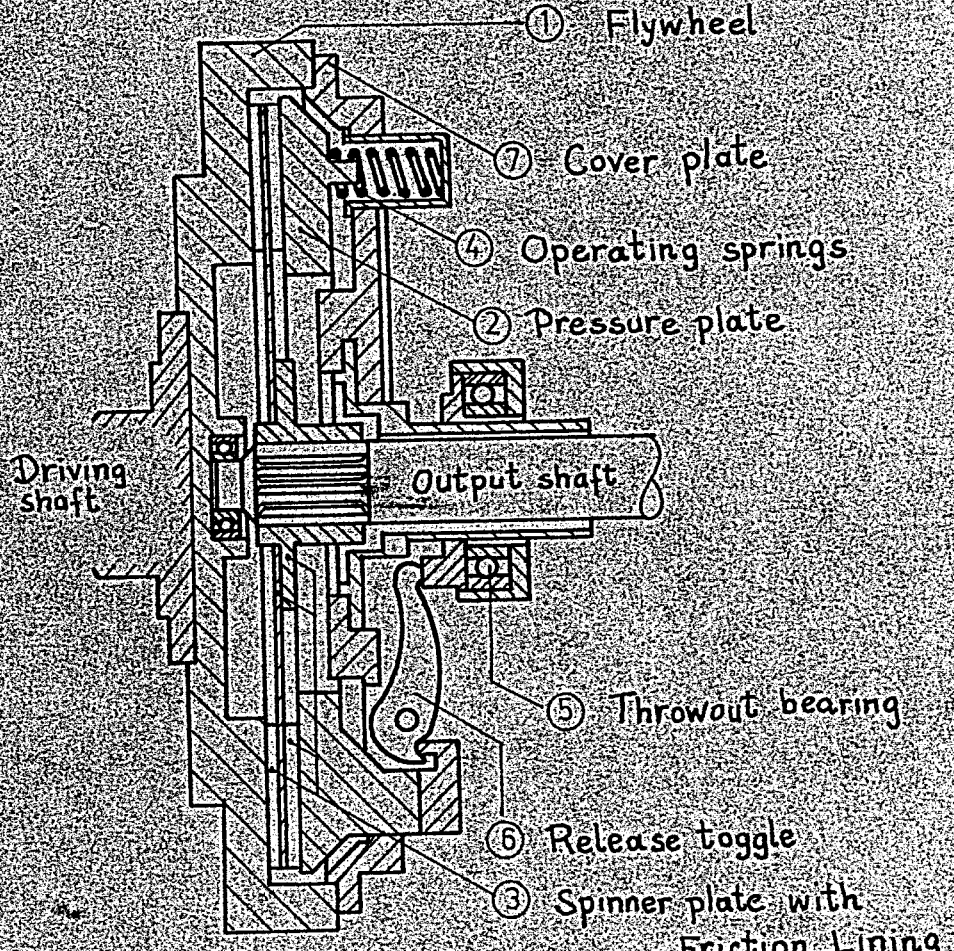


fig.1 - Typical Automotive Clutch

When the clutch is operative the pressure plate clamps the spinner plate against the driving plate by means of the springs, and the drive is then transmitted through the pressure and driving plate to the spinner and so to the transmission shaft.

To disengage the clutch, the pressure plate must be withdrawn to release the clutch plate. This is effected by depressing the throw-out bearing which, being coupled to the pressure plate by a series of toggles, pulls this plate of against the operating springs.

The actuating mechanism to force the friction surfaces into contact is of mechanical type in automotive applications, and the opposing force is inevitably in the form of a spring or springs, whether it be clamping or releasing.

Friction clutches will slip momentarily under shock loads, thereby providing a cushioning action, and they will slip during engagement, thus permitting a load to be coupled and accelerated smoothly even though the drive may be running at high speed.

C. Survey of Clutch Friction Lining Types

1 - Classification of Linings

Lining friction materials are classified according to their structure and bonding agents used in their manufacture into two main groups :

(1) organic (nonmetallic) base

(2) inorganic (metallic) base

(1) Organic Friction Materials :

Their ingredients may be grouped as follows :

(a) filler materials, fibrous and/or nonfibrous 20 to 80 per cent

mineral-base filler materials, i. e. asbestos, caolen, talc, mica, etc,

10 to 60 per cent organic-base filler materilas, i. e. cellulose, cotton, graphite, etc.

10 to 50 per cent metallic-base filler materials, i. e. steel wool, brass, copper, zinc, etc.

(b) bonding agents

20 to 40 per cent natural and/or synthetic rubber, thermosetting synthetic resins, duroplast, impregnating materials.

(c) special additives to provide desired characteristics (1)[x]

(2) Inorganic Friction Materials :

There are Sintered Metal Linings and Cermet Friction Materials.

(a) Sintered Metal Linings :

produced by the compression and partial fusion of mixtures of fine particles consisting mainly of powdered metals. Most common ingredients are copper, iron, tin, lead, graphite and silica, and some special additives for particular cases (2)

(b) Cermet Friction Materials :

similar to the sintered metal, but contain an appreciable proportion of ceramic particles (2)

x Numbers in parantheses refer to the Bibliography at the end of the thesis text.

Modern automative clutch friction facings manufactured from fibrous materials such as asbestos are classified into two basic categories :

(1) woven linings (wound or fabric molded facings)

(2) molded linings (molded non-oriented fiber facings)

It is recommended practice that the following requirements are made from type (1) and type (2) linings :

(A) for a temperature of $20^{o}C$ at rubbing interface :

(a) static friction coefficient at a clamping pressure of $p = 1\ kg/cm^2$

$\mu_s = 0.3$ for pure dry operation

(b) maximum kinetic friction coefficient at

$P = 1\ kg/cm^2$, $\mu_k = 0.5$

$P = 2\ kg/cm^2$, $\mu_k = 0.4$

(B) for a temperature of $200^{o}C$ and a slip velocity of 2 m/sec

$\mu_k = 0.35 - 0.40$

The sizes (basic dimensions) of clutch friction facings are standardized (see DIN 73455). The method of giving the basic dimensions of disk clutch facings is in the following form :

$$R_2 \times R_1 \times t$$

where

R_2 = outside disk radius

R_1 = inside disk radius

t = thickness of facing

The main requirements from clutch facings (recommended practice) are given in table form below (4) :

Clutch Lining type	Dry friction coefficient	Allow. Temp. (^{o}C) (trans. -steady)	Al. Clamp. pres. (kg/cm^2)	Al. Wear (cm^3/PSh)
(1)	0.5-0.3	300 - 2000	0.5 - 20 0.5 - 10	0.1 - 0.3
(2)	0.4-0.2	500 - 250	0.5 - 100	0.1 - 0.3

2. Manufacturing Methods of Facings

(1) Woven Facings (wound or fabric molded facings) :

The woven facings are made from asbestos roving or yarn which contains a wire core (either brass or zinc). The roving or yarn is treated with an adhesive composed of rubber, resin, filler, tackifier, and solvent; then dried and rewound on spools. Preforms are then wound into the shape of a ring. The treated asbestos must have the building tack necessary to hold the shape of the preform during the handling. The facings are wound in several different ways, i.e., convalute or concentric and fluted, after which the preforms are hot press cured. Rubber and resin are used as the binder, and resin content is generally high. Cashew nut shell oil modified phenolic resins are used primarily due to their good compatibility with rubber. These facings have high burst strength characteristics because of their design.

(2) Molded Facings (molded non-orieted fiber) :

Molded facings are generally made by cold preforming a dry mix and then hot molding. The resin used is either a simple phenolic or an oil modified phenolic two-step resin. A balance of performance and price dictates the

choice of resin. A second preforming method involves the extrusion into blanks of a mix composed of long asbestos fiber, powdered resin, and polyvinyl butyral (PVB) or rubber as an extrusion aid. The rings are then dinked out and hot molded. The scrap is returned to the next mix for immediate rework. The use of molded non-oriented fiber type of clutch facing is predicted to decline since the high torque, high hot burst strength (8,000 - 10,000 rpm at 250 - 300^{0}C) requirements necessary for high performance automotive engines cannot be met. However, this type of clutch facing still represents a good replacement market for older cars. (5).

D. Properties of Lining Ingredients

1 - Physical Caharteristics of Basic Components

(1) The basic asbestos :

Modern friction linings are wholly or largely nonmetallic, and asbestos in one form or another is the almost universal base material.

It is the "chrysotile" variety of asbestos which is mainly used in friction linings. Occurring as veins through a matrix of serpentine rocks, it is thought to have been formed following percolation of water through fissures in the rock formation. It is white-greyish when carded (straightened out). The chemical formula

(Mg_6. (Si_4 O_{11}). H_2O) indicates that the crystalline solid contains " water of constitution ", this asbestos is a crystalline "hydrate" (6). About 25 per cent of this water is free crystal water and evaporizes when asbestos is heated up to ~490°C. The remaining part of the water is the water of constitiution which dissociates only at about 700°C, together with the collapse of the chemical structure of asbestos. The average percentages of the components of asbestos are 43.5 per cent MgO, 43,5 per cent SiO_2, and 13-14 per cent H_2O. The specific gravity is 2.0-2.8 g/cm^3. The melting point is around 1550°C.

This mineral is unusual in being both fibrous and crystalline, elastic and brittle, and yet it can be carded, spun into yarn, and then woven into cloth and belting.

The length of individual asbestos fibres varies between 1 - 40 mm. Each individual fibre is a hollow tube with an outside diameter of about 0.02-0.04 μ, and an inside diameter of the order of 0.001 μ.

All types of asbestos are noted for their resistance to chemical attack as well as to the effects of high temperature. The chrysotile asbestos is affected by mineral acids, but is resistant to organic acids and hydroxides. Temperatures up to 400°C cannot change the physical characteristics of asbestos. At higher temperatures it

loses its strength, and since it also loses its water of constitution, the minerals in the crystal structure dissociate in the form of a fine powder. Especially this characteristic enables asbestos to be used as friction material component. Because of the loss of the water of constitution at temperatures around 700°C, asbestos layers on the friction material at the rubbing interface are dissociated and separated, and the new layers appearing show the same friction characteristics. This process is repeated continuously. (7).

To make linings from asbestos alone, woven or molded, is impracticable for a variety of reasons. They would be

(1) very deficient in mechanical strength

(2) deficient in resistance to wear

(3) would have a moredarate or low friction coefficient

(4) asbestos being a mild abrasive, would cause excessive damage to comparatively expensive contach surfaces

(5) impractical and costly

Therefore, a chemical impregnant or bond is introduced to combine with the asbestos base and make good these various deficiencies.

(2) Synthetic Resins :

Synthetic resins constitute the most important components of the bonding agents. Their molecular structure contains carbon, hydrogen and oxygen atoms. They don't dissolve in water, areslightly soluble

in alcohol, but are soluble in aceton, benzol and carbontetrachloride There are a great variety of synthetic resins being different in their chemical structure. The mostly used types are the phenolic resins. The reason why the phenolic resins are extensively used is that they are thermosetting, that is their ability at high temperatures. But since they get brittle after hardening, they are impregnated together with other materials such as rock powder, wood flour paper or cotton bands in order to give mechanical strength.

(3) Natural Rubber :

It is won from the cautschuck tree. Although it has good elastic properties it has a low mechanical strength and is sensitive to high temperatures. Therefore, it is used in light duty type friction linings.

(4) Synthetic Rubber (BUNA) :

The name Buna is derived from the first syllables of Butadien and Sodium (Na). By the polymerization process of butadien gas under the catalysis of sodium under high temperature and pressure, Buna is manufactured.

The heat and wear resistance of Buna which is used as bonding agent is more than those of natural rubber. It is sensitive to oils,

ruptures easily, dissolves easily in organic solvents, and has an ability to form homogeneous mixtures with other additives,

(5) Metallic Fillers :

The inclusion of metallic filler materials in fine particle or wool form serves to conduct some of the frictional heat away from the rubbing surface and so to reduce to a slight extent the maximum temperature reached by the clutch mechanism. The metallic particles have themselves high thermal conductivity, but since they are dispersed in the body of the friction lining, their effect on heat conduction is limited,

Their presence in the composition also helps to establish a more uniform and stable value for friction coefficient,

The mostly used metallic fillers are brass and soft-steel wool and chips,

The values of hardness of the metallic fillers are made compatible with that of the mating disk surface in order to prevent the scoring of the more expensive contact surface,

Summarizing our analyses of the properties of the lining ingredients, we can state the following :

The most common type of modern nonmetallic friction material is (organic), a heterogeneous mixture of rubber and/or resin binders,

carbon, sulfur, asbestos fiber, catalysts, metal particles, friction particles, mineral fillers, etc., about up to 10 to 20 components. These ingredients are needed to provide the great variety of properties which friction linings must possess. Among these are (8):

(1) appropriate friction coefficient

(2) uniform coefficient over wide temperature range and high fade resistance

(3) Low wear rate

(4) adequate mechanical strength

(5) resistance to oil and water

(6) absence of abrasive action on the contact surface

(7) low shrinkage or swelling

(8) absence of odor

(9) low cost and ease in manufacturing processes

2. Effects of Ingredients on Performance Characteristics

The functions of the more important lining ingredients in the determination of the performancecharacteristics may be summarized as follows :

(1) Asbestos fiber imparts heat resistance and mechanical strength. (Fibrous fillers like asbestos increase the stiffness, impact resistance and also tensile strength).

(2) Bonding agents - resin, rubber, or a combination :

(a) thermosetting phenolic resins provide mechanical strength and resistance to oil and heat;

(b) natural or synthetic rubber produces a more flexible material;

(c) rubber-resin combinations may provide the best features of both.

(3) Friction Particles provide wear resistance and high friction. They may consist of :

(a) hard-rubber dust (20 to 80 mesh) heavily loaded with inorganic fillers (barytes) to reduce plasticity,

(b) fully cured resins such as cashew nutshell oil,

(c) crushed coal and coke (low friction).

(4) Vulcanizing agents for rubber-sulfur, litharge, zinc oxide, accelerators - to give elasticity and mechanical strength.

(5) Carbon black may be used to increase tensile strength or as a coloring agent.

(6) Brass chips improve wear resistance and reduce plate scoring or abrasive action.

II - STATEMENT OF THE PROBLEM

A. The Need for Research on Automotive Friction Cluth Linings

Recent technological advancements in automotive industries show a remarkable trend toward decreasing the motor sizes and simultaneously increasing the motor power outputs and rpm's. Consequently, during the engagement of clutches, the clutch linings would be excessively stressed at the higher applied torques and rotational speeds under the effect of higher centrifugal stresses developed.

High values of rpm mean also high slip velocities during clutch engagement, which furthermore would cause high surface temperatures. Therefore, in addition to the conventional requirements of clutch friction materials, we also require a higher mechanical strength and a higher heat resistance for resisting the effects of the higher centrifugal stresses and surface temperatures, respectively.

For safe and satisfactory operation, the friction lining should not undergo deformation and not lose its smooth surface under the effects of the high rotational speeds.

The current trend toward increasing the motor speeds imposes critical requirements on the performance characteristics of the modern friction materials. Consequently, there has arisen a great need for analyzing the effects of the operating conditions of pressure and rubbing velocity on the

performance characteristics of typical modern automotive friction clutch linings.

B. Description of the Approach

In the analytical and experimental work, the author has studied the basic types of clutch friction facings. These two basic facing qualities are the following :

(1) molded or molded non-oriented fiber facings

(2) woven or wound or fabric molded facings

The operating conditions or the service variables whose effects on the performance characteristics of the clutch friction facings have been analyzed are listed as follows :

(1) slip velocity (sliding velocity, rubbing velocity of the contact surfaces), $\dot{v}^*$

(2) pressure of the mating plate (clamping pressure, pressure), p

(3) maximum value of the rubbing interface temperature, θ max.

(4) rotational speed of the output shaft of the engine, w

The performance characteristics of the facings have been investigated under the following three items :

(1) Friction and Wear Characteristics

(2) Bursting Behaviour

(3) Thermal Effects on the Rubbing Interface

In the experimental analysis, the author has run two different sets of experiments, namely Friction and Wear Tests and Disk-Bursting Behaviour Tests (F and W Test, D-B B Test). In the F and W experiments, the effects

(combined) of the pressure and velocity on the friction coefficient and specific wear rate on the basic lining types have been studied. In the D-B B tests, the effect of high rotational speeds on the (centrifugal) fracture stresses of the facings has been studied.

The studies have been carried out to assist the development of clutch friction facings manufactured by the Turkish Industries. Thus, the study has also taken the form of a technological research.xx

In addition to the abovementioned considerations by the approach of the problem, the author has also tried to develop a practical test procedures for effective testing of friction clutch materials, since an extensive survey of the SAE, ASTM, DIN and TSE standards with respect to clutch materials has shown the absence of this type of tests.

In summary, the aims of this study can be stated as follows :

(1) the development of testing systems for the investigation of the effects of the operating conditions on the performance characteristics of typical automotive clutch facings.

(2) the development of practical test procedures for the testing of automotive clutch facings.

x The words or expressions in the parantheses are the synonymous expressions of the foregoing ones.

xx The test specimens used in the experiments have been produced by the turkish plant, ERKA BALATACILIK KOL. ŞTİ.

III. THEORETICAL ANALYSIS OF FRICTION CLUTCH LINING PERFORMANCE

A. The effects of Operating Conditions on Friction and Wear of Linings

1 - Friction and Wear Phenomena

a - Friction and Friction Coefficient

The laws of friction of metals and polymers apply in general to brake and clutch linings and other friction materials, (9).

Definition :

The friction between two bodies is generally defined as the force which acts between them at their surface of contact so as to resist their sliding on one another. The frictional force F is the force required to initiate or maintain motion. If N is the normal reaction of one body on the other, the friction coefficient is defined as

$$\mu = \frac{F}{N} \qquad (3.1)$$

Static and Kinetic Friction :

If the force to initiate motion of one of the bodies is F_s and the force to maintain its motion at a given speed is F_k, there is a corresponding coefficient of static friction $\mu_s = \frac{F_s}{N}$, and a coefficient of kinetic friction $\mu_k = \frac{F_k}{N}$. In some cases these coefficients are approximately equal; but in most cases $\mu_s > \mu_k$.

Classification :

With respect to the type of motion, friction is divided into two categories as

(1) sliding friction

(2) rolling friction

If the relative motion is of sliding type, friction is classified into three types according to the characteristics of the contacting surfaces as

(1) dry friction

(2) semi-viscous friction (boundary friction)

(3) Viscous friction (hydrodynamic friction)

Our case is of dry-sliding friction type. Dry-sliding friction develops during the relative motion of nonlubricated surfaces and is primarily a very complex surface phenomenon. (10). Consequently it depends very markedly on surface conditions such as roughness, degree of work hardening, type of oxide film, and surface cleanliness. In general the roughness has only a secondary effect but surface contamination can hava a profound influence on the friction (and wear), particularly with surfaces that are nominally clean, Tables of friction must be used with caution, since very wide differences may be obtained in friction under apparently similar conditions, especially with unlubricated surfaces, (10).

Basic Laws of Friction :

The so-called " classical laws of friction " as formulated by Coulomb, 1785, are usually stated as follows :

(1) The friction force F between solid bodies is proportional to the normal force between the surfaces; i. e., is independent of N.

(2) The friction force F is independent of the apparent area of contact.

(3) The friction force F is largely independent of the velocity of sliding.

(4) The friction force F depends on the nature of the sliding surfaces, (11).

These laws of friction are reasonably well obeyed for sliding metals whether clean or lubricated; but with friction materials the laws are not so well obeyed; in particular the friction coefficient usually decreases with increasing velocity and load.

Excellent analyses and presentations on friction of clean metals, theories of metallic friction and various mechanisms of friction are given by Bowden and Tabor (12), by Archard (13), by Bowden, Moore, Tabor (14), by Bowden, Rowe (15), and by Tabor (16). We shall only summarize their findings very briefly.

Many experiments on the friction of metals and the mechanism of wear have shown that it may be explained in terms of the formation and subsequent breaking of metallic junctions (12). Since all carefully

prepared surfaces contain irregularities which are large compared to the molecular dimension, the real area of contact will be very small and the local pressure will correspond to the yield pressure. This intense pressure will cause the formation of localized welded junctions at the points of contact. The frictional resistance is largely due to these junctions but is also due somewhat to the displacement of the material from the path of the moving asperity. Bowden and Tabor (12) assumed this latter term to be negligible for most materials used in engineering applications and expressed the coefficient of friction by

$$\mu = \tau_s / \sigma_f \qquad (3.2)$$

where τ_s = shear strength of the weaker of the two contacting materials,

σ_f = flow strength of the softer of the two contacting materials

Since contacting surfaces, even carefully prepared, contain irregularities, the real area of contact is very small as compared to apparent contact area, and actual contact occurs at the points of irregularities. Therefore, under the normal forces these irregularities or asperities will be deformed elastically first, but later plastically exceeding the flow strength of the softer material. This

process is shown in Fig. (2). It continues until there develops a contact area sufficient to carry the applied load. The real area of contact, A_o, is found by

$$A_o = \frac{N}{\sigma_f} \qquad (3.3)$$

where N is the applied normal force. The friction force F required to shear the localized welded junction is given by

$$F = A_o \cdot \tau_s \qquad (3.4)$$

By the definiton of μ,

$$\mu = \frac{F}{N} = \frac{A_o \tau_s}{A_o \sigma_f} = \frac{\tau_s}{\sigma_f}$$

Thus, we have derived Eq. (3.2)

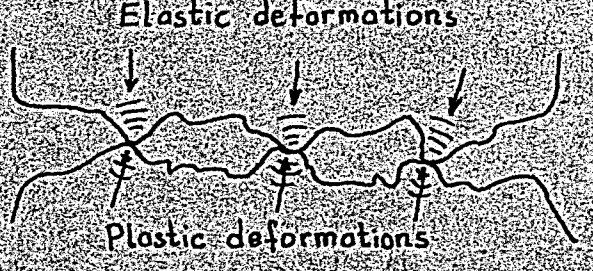
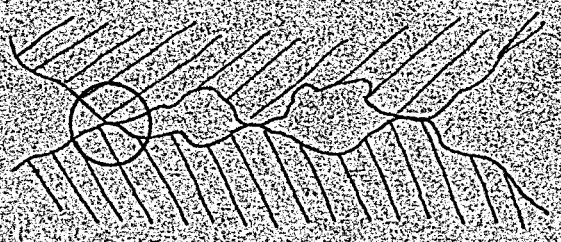


Fig (2) Real contact points at surfaces

For a given material, the values of σ_f and τ_s are constant. Therefore, μ should take a constant value. This result also verifies that the friction coefficient is independent of load, contact area, and sliding velocity, as stated in Coulomb's Laws. Because of the elastic deformations around the contact regions, the real contact area defined as $A_o = N/\sigma_f$ constitutes the theoretical lower limit of the real contact surface.

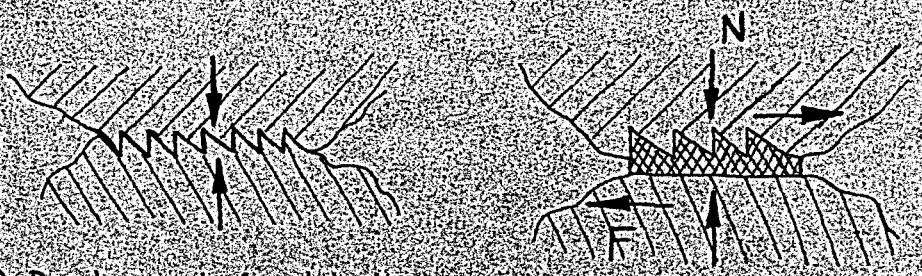
The above Adhesion Theory explains partly the friction between same kind of materials and metallic materials. Generally, it is not sufficient in explaining the friction phenomenon in the cases such as friction clutches and brakes where " organic-type " friction materials are used, becasue of the lack of the development of a metallurgical bond between the moving parts.

In order to overcome the deficiency of the adhesion theory, the Interlocking Theory was developed, which is based on plastic deformation in (crystalline) materials. Asperities are thought to be deformed plastically by abrupt movements which usually take place along several sets of crystallographic planes, causing roughening of the interfaces of contacting high spots. This tends to form a matched interlocking pattern during plastic deformation. The interlocking effect contributes substantially to the resistance to tangential motion. Thus, if a pair of asperities is strongly interlocked, a fracture tends to occur at the weakest section inside one of the asperities rather than at the interface. Fig. (3) shows in a schematic way the processes of interlocking. (17).



(a) Real contact points on surfaces

BOĞAZİÇİ ÜNİVERSİTESİ KÜTÜPHANESİ



(b) Development of matched interlocking patterns by plastic deformation

(c) Development of the wear particle

Fig. (3) The processes of interlocking

Looking at the phenomenon of friction in a bird's view and considering the laws and theories of friction in a panoramic manner, the author may formulate his own concepts as follows :

Friction is of a dual molecular-mechanical nature. On one hand, it depends on overcoming the bond between two surfaces by the forces of molecular interaction. On the other hand, it depends on the volumetric deformation of the material due to mutual penetration, geometric configuration of contacting elements, and a relation between mechanical properties. Friction may involve the following :

(1) Shearing of penetrated material of formation of a scratch with the separation of the material ;

(2) Formation of scratches with plastic displacement of the material;

(3) Elastic deformation of the material;

(4) No destruction of the material;

(5) Destruction at a certain depth.

In the first and last cases, each destruction of a bond causes breakdown of the material. Therefore, these two kinds should not be properly related to friction since the wear they involve is great. In the third and fourth cases no destruction of the material occurs, and hence they correspond to ideal friction.

In the second case, a breakdown of the material occurs due to the multiple destruction of the points in contact. The degree of destruction approximately corresponds to the number of cycles. This is characteristic of surface fatigue type failures in materials. This case is intermediate in severity; but it is actually the most typical kind of friction and wear, since the small value of the real area of contact leads to high stress, thus resulting in plastic deformation.

b- Wear

Laws of Wear :

Although the laws of friction are fairly well substantiated there are no satisfactory laws of wear. Ingeneral it is safe to say that wear increases with the time of running, and that with hard surfaces the wear is less than with softer surfaces. But there are many exceptions and the dependence of wear on load, nominal area of contact, speed, etc, is even less generally agreed upon. This is because there are many factors involved in wear and a slight change in conditions may completely alter the importance of individual factors or change their mode

of interaction, (17).

The complexity of wear is emphasized by a consideration of the number of factors required to describe it. The chief factors that influence wear may be listed systematically as follows:

(1) Variables connected with metallurgy

Hardness

Toughness

Constitution and structure

Chemical composition

(2) Variables connected with service (Operating Conditions)

Contacting Materials

Pressure

Speed

Temperature

Surface Finish

(3) Other contributing factors

Lubrication

Corrosion

Furthermore, the wear that occurs in practice is usually a combination of one or more of the elementary forms. Small particles resulting from galling may cause abrasive wear; or in a press-fitted assembly, frettage corrosion may result in wear and subsequent damage.

The concept of wear involves the presence of small particles between the contacting surfaces. It is probable that the abrading particles first penetrate the material and then cause tearing of the surface, introducing surface stress cracks which lead to the ultimate breakdown of the surface. A general discussion of the factors, or some of them, is given in a simplified way as follows. (17)

Hardness

Wear rate is generally known to decrease as the hardness of the material increases, providing other factors remain constant.

Load

Wear was found to increase almost proportionally with load for most materials. This relation, however, is obscured by the frictional heat involved.

Lipson (17) states as a rule of thumb that at stresses below the yield point, wear will be independent of the load.

Distance

Wear rate was found to be linearly proportional to distance run. Results of experiments by Nathan and Jones (18) show that the relationship between the volume of abrasive wear and the length of abrasive path is linear.

Temperature

Wear rate increases with temperature, because of a decrease in hardness, an increase in the chances of welding, plastic deformation, and corrosion by oxidation.

Sliding Velocity

The decrease of friction and wear with increased sliding velocity is probably due to the resulting steeper temperature gradient, the effective area of contact may be reduced, so there is less time available for yielding under the applied load.

Freitag (19) has found that, for certain material combinations, the rubbed surfaces undergo flow and melting. In other instances there was marked fragmentation of the surface layers.

Thus one can conclude that, at high sliding speeds, wear is influenced by two factors :

(1) high temperatures developed

(2) propagation of plastic deformation

Surface Finish

Generally, the rougher the surface, the higher the wear rate. However very smooth surfaces manifest hifh molecular interaction forces and lose the ability of storigg contaminants due to the absence of the "valleys" otherwise found between the relatively large asperities of a rough surface.

Contaminants

Contaminants might be beneficial, like lubricants and surface films, or harmful, like abrasives. The presence of low shear strength solid surface films or lubricants reduces friction and wear of rubbing surfaces.

Environmental Effects

The corrosive (and erosive) effects of chemicals and the erosiveness of fluids flowing across a surface promote wear in a manner corresponding to existing conditions.

For same materials, wear rates in dry atmosphere were found to increase " eightfold over those in normal air ". (20).

Types of Wear :

Generally, in practice, wear is usually a combination of one or more elementary forms. For example, small particles resulting from galling may cause abrasive wear; or corrosion may be entirely obscured by galling in a press-fitted assembly fretting (17). Thus, itiis not always easy in any given application to state which type of wear has taken place.

Lack of agreement on nomenclature and lack of good definition of the different manifestations of wear (such as scuffing, galling, abrasion, etc.) furthar complicate the subject. However, one logical method of

classifying wear would involve the nature of the contacting surfaces, as follows :

(1) metal-against-metal

(2) metal-against-nonmetal

(3) metal-against-fluid

In turn, this classification must be subdivided into the conditions of :

(1) lubricated wear

(2) nonlubricated wear involving one of the following friction types :

(1) sliding friction

(2) rolling friction

Moreover, conditions may change, and what was initially metal-against metal might become metal-against-nonmetal, or lubrication initially adequate might fail in further service. For this reason the above classification loses its usefulness in actual engineeringpractice, and grouping in commonly understood terms is preferred instead.

There are three basic types of wear encountered in friction material applications (1) :

(1) adhesive type of wear (galling, scuffing, scoring, and seizing)

(2) abrasive type of wear

(3) fatigue type of wear (pitting and spalling)

A very brief explanation of each type of wear encountered in friction material applications is given below :

(1) Adhesive Type of Wear :

Adhesive form of wear is probably the most basic type of wear. It is caused by a shearing action of micro-welds formed between the surface asperities that actually carry the load between two mating surfaces. This type of wear occurs because of the failure of the film which normally separates the two surfaces. In turn, film failure is caused by high temperatures, pressures, and sliding velocities. (The mechanism of adhesive type of wear is broadly explained under paragraph (III. A. 1. a)

(2) Abrasive Type of Wear :

Abrasive wear can be defined as wear that occurs at normal temperatures as a result of dynamic contact between material wear surfaces and abrasive particles or fragments. The highest wear rates occur, when the hardness of the abrasive is considerably greater than that of the abraded material.

Wear by abrasion can be divided into three major categories :

(1) gouging abrasion

(2) high stresscor grinding abrasion

(3) low stress scratching or erosion abrasion

All three may occur simultaneously on a wearing part, but usually the type which predominates can be recognized (17).

(3) Fatigue Type of Wear :

Pitting, pitting corrosion, spalling, and case crushing are all the same type of phenomenon, but spalling is probably the most common. (Spalling is generally considered a special form of destructive pitting. Large pits, formed by the joining of smaller adjacent pits due to failure of the adjoining material is a type of spalling).

Pitting is generally attributed to the cyclical repetiton of contact stresses between two mating surfaces. It is defined as surface fatigue failure of the material as a result of repeated surface or subsurface stresses that exceed the endurance limit of the material. It is characterized by the removal of material and the formation of cavities. The cavity thus formed is a pit, from which material frequently spalls out. These cavities can be small and can remain small. They may be small initially and then combine or increase in size by continued fatigue (17).

The destructive character of pitting involves a multitude of factors such as high contact stresses, possible formation of an elastic wave ahead of the instantaneous area of contact, surface flow, and subsurface fatigue. Incipient pitting, which may be erased by subsequent wearing action, and destructive pitting, which leads to fatigue failure, should be distinguished (17).

At this stage, the author would like to make the following remark with regard to the method of analyzing the friction and wear phenomena of the automotive friction clutch :

A survey of literature on various types of friction lining applications has indicated to the author a general tendency to disregard the type of metal, structure and even finish of mating surfaces of friction linings when considering practical friction and wear problems. It should always be remembered that there are always two contacting surfaces where solid friction is required, and both share in the resulting performance of the mechanism.

The mating surfaces of linings are generally chosen to be ferrous to resist the slightly abrasive nature of asbestos, and so confine wear to the lining fabric. Fine-grained high tensile east or alloy iron is the ideal metal surface in a kinetic friction application, whether the friction material contains a metal inclusion or not. Steel, whether cold pressed or forged, is satisfactory, provided the Brinell hardness value is 150 or over, when used with non-metallic or zinc wire linings. Steel surfaces having a Brinell of 200 and over are satisfactory for use with linings containing brass wire and when used with heavy-duty moulded linings (2).

Therefore, considering our friction and wear system as being composed of the cast/alloy iron or steel mating plate and the friction lining which rub against each other, we can assume that all of the above described basic types of wear occur simultaneously on each of the members, but in different quantity and severity.

c - Causes of Friction and Wear in Clutch Linings

The laws of friction of metals and polymers apply in general to brake and clutch linings and other friction materials (21). The performance of clutch linings is similar in many respects to that of metals. One reason is that the mating surface against which the clutch lining rubs is always a metal, usually the steel of the driving plate on one side and the steel of the pressure plate on the other side. Thus we have a metal-against-nonmetal type of friction and the two surfaces act together to determine the friction coefficient and its characteristics.

Five causes of lining friction may be considered :

(1) The lifting over surface asperities, because adhesion theory does not explain all frictional properties of linings (22).

(2) Welding of junctions, which may occur between metals does not occur between the clutch lining and mating plate. Nevertheless, adhesion and shearing of junctions occur, because the materials of the lining and the plate often become smeared over each other. The real area of contact varies markedly with the condition of the

lining surfaces. Examination with the optical microscope has shown that the pits and irregularities of the surface become filled with wear particles, and at high temperatures the entire surface becomes glazed with partially decomposed binder, plate metal, etc. As with metals, surface contamination has a marked effect on friction and wear. It is always necessary to run in a lining (please see the corresponding paragraph (IV. C. 1. c) of experimental procedure of teh F-W test under procedure 2, "Conditioning of the Test Specimen ") before a representative friction coefficient can be obtained under any given set of conditions (23).

(3) " Plowing ", which may be considered shearing below the surface helps to account for the high wear rates when the lining is abrasive (21).

(4) The elastic-plastic properties of the bulk material below the surface contribute to sliding friction, because the surface irregularities (always large compared with the molecular dimension) " work " the material below the surface (24).

(5) Triboelectrification (development of static electrical potential) accompanies the rubbing of linings as well as most other solids. Some observers believe this a cause of friction, whereas some experiments with the measurement of the frictional-electric potentials indicate that triboelectrification is an effect rather than a cause of friction (21).

The concept of wear involves the presence of small particles between the contacting surfaces. It is probable that the abrading particles first penetrate the material and then cause tearing of the surface, introducing surface, introducing surface stress cracks which lead to the ultimate breakdown of the surface. The effects of the factors influencing wear and the theories of wear have already been presented under paragraphs (III. A. 1. b) with due consideration of friction linings.

2 - Relationships among Operating Variables and Friction-Wear

a - The Clutch Model

Consider the arrangement or model shown in Fig. (4), in which it is desired to determine the torque M due to frictional forces existing at the interface of members A and B, and due to the axial force P.

From this diagram, the axial normal load acting on the differential strip of width dr is

$$dP = 2\pi r\, p\, dr \qquad (3.6)$$

so that

$$P = 2\pi \int_{R_1}^{R_2} p\, r\, dr \qquad (3.7)$$

where p is the pressure (clamping pressure).

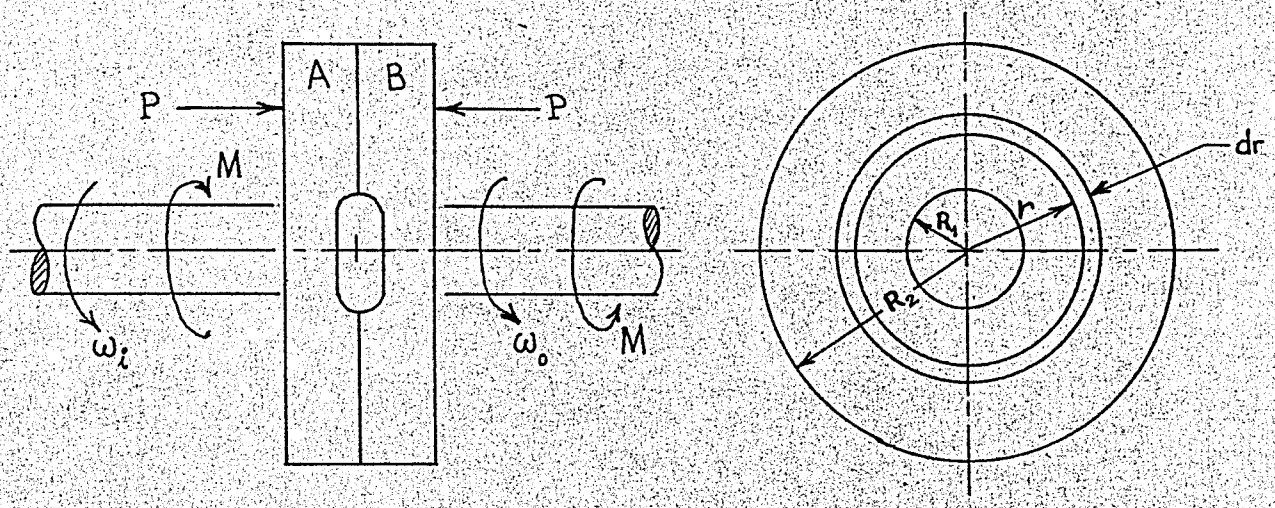


Fig. (4) Model of a friction disk-clutch

The frictional force acting on the strip dr is

$$\mu\, dP = 2\pi\, \mu\, p\, r\, dr \qquad (3.8)$$

Similarly, the associated frictional moment (torque), due to the tangential force acting on the elemental strip dr, is

$$dM = 2\pi\, \mu\, p\, r^2\, dr \qquad (3.9)$$

and the total frictional moment

$$M = 2\pi \int_{R_1}^{R_2} \mu\, p\, r^2\, dr \qquad (3.10)$$

Before Eq. (3.10) can be integrated, the dependence of μ and p on radius r, and the relationship connecting μ and p must be known.

b - Relationship between μ and v (or r)

For the moment, pressure may be assumed to remain sensibly constant during the engagement period. Also, the effect of transient temperature variation on friction may be considered to be included in the variation of friction with sliding velocity v. Fig (5) illustrates this variation for representative friction materials used by the automotive industry (9). The graph shows that friction coefficient decreases with relative sliding velocity v; the relationship can be written

$$\mu = f(v) = f(\omega_r r) \qquad (3.11)$$

where ω_r is the relative angular velocity between the members A and B of the Clutch in Fig. (4) and is given by

$$\omega_r = \omega_i - \omega_o \qquad (3.12)$$

where ω_i and ω_o are the angular velocities of input and output members A and B, respectively. The tangential relative velocity at radius r is

$$V = (\omega_i - \omega_o) r = \omega_r r \qquad (3.13)$$

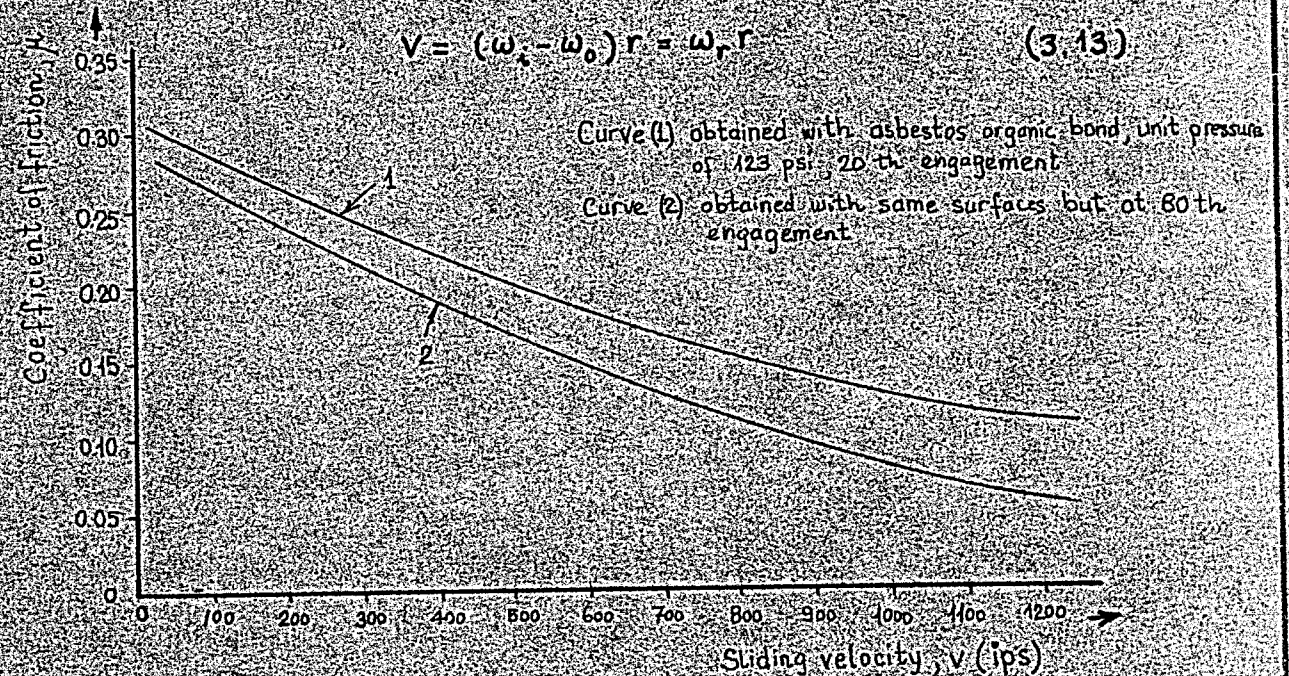


Fig. (5) Sliding friction coefficient obtained in flywheel deceleration tests

For some materials, the variation of μ with v may be approximated with a straight line having negative slope. This relationship, somewhat idealized, is shown in Fig. (6).

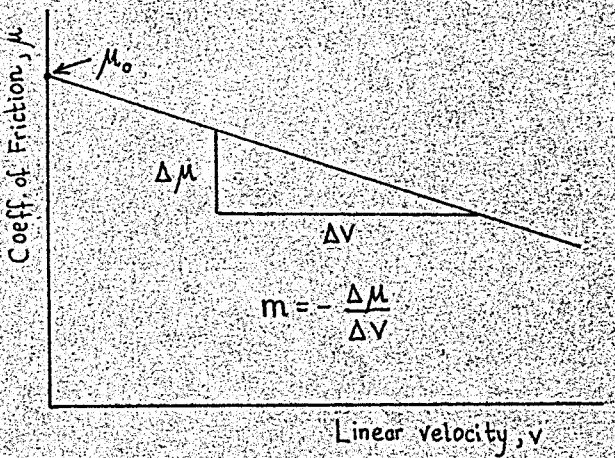


Fig. (6) Relationship of coefficient of friction, μ, and linear velocity, v, idealized as a straight line

The slope of the line, $m = -\Delta\mu/\Delta V$, indicates the dependence of friction on sliding velocity, and can be written

$$\mu = \mu_o - m \cdot v = \mu_o - m \cdot \omega_r r \qquad (3.14)$$

where μ_o = static coefficient of friction

m = slope of the friction-speed characteristic of the interface (depends on material combination)

c - Relationship between p and v (or r)

There is, in any practical situation, considerable uncertainty as to the distribution of axial load P over the contact area. Because of the difficulty of obtaining perfect geometrical fit between the two surfaces, and particularly with relatively hard materials such as rigid molded

linings, contact between the surfaces is far from uniform and pressure distribution highly irregular (2). In our case, the friction surfaces are made of resilient (resilience - potential energy stored up in a deformed body) materials, such as mixtures of asbestos (in the form of woven and flexible molded, or molded qualities), fairly uniform contact can be assumed. Therefore, the pressure is reasonably independent of v and r.

d - Relationship between Wear Rate and p and v (or r)

When clutch life is considered, the rate at which surfaces wear depends not only on the axial pressure but also on the sliding velocity between the surfaces.

Since wear on a new clutch is greatest at the periphery (largest circumferential velocity), older clutches exhibit a redistribution of pressure. The rate at which material is removed is a function of the pressure and the sliding velocity,

$$\text{Rate of Wear} = f(p, v)$$

Since there is little information available as to the exact relationship between the rate of wear, intensity of pressure, and sliding velocity, it is assumed that rate of wear is directly proportional to the product pv, representing the frictional energy dissipation per unit area. Thus,

$$\text{Rate of Wear} = k_1 \, p \, v \qquad (3.15)$$

and, since v varies directly with r,

$$\text{Rate of Wear} = k_2\, p\, r \qquad (3.16)$$

where k_1 and k_2 are constants.

There is experimental evidence that the product of the pressure in kg/cm^2 (or psi) and the velocity v in m/sec (or fpm) must be kept within certain limits (11). On this basis, the following values are given to be considered as an additional criterion for the design of clutches and friction facings :

(1) $pv = 30\ kg_f.\ m/cm^2.\ sec$ ($83,000\ lb.\ ft/in^2.min$) for continuous-load application and good heat dissipation, as in an oil bath.

(2) $pv = 20 kg_f.\ m/cm^2.\ sec$ ($55,000\ lb.\ ft/in^2.\ min$) for intermittent-load application, relatively long periods of test, and poor heat dissipation.

(3) $pv = 10\ kg_f.\ m/cm^2.\ sec$ ($28,000\ lb.\ ft/in^2.\ min$) for continuous-load application and poor heat dissipation.

Consequently, the assumption that rate of wear is uniform over the area of the frictional surfaces is made, that is,

$$p.r = K\ (\text{constant}) \qquad (3.17)$$

e - Relationship between μ and p

As previously indicated, μ may be a function of p. If p is invariant with r, the expression for the clutch frictional moment will not be effected by the variation of μ with p. However, the design of the

clutch (i. e., the number of plates required, etc.) will be different for different values of p.

If p varies with r, and μ is a function of p as in

$$\mu = \mu_p - a\,p \qquad (3.18)$$

and Eq. (3.17) is admitted, one obtains

$$\mu = \mu_p - a\,K/r \qquad (3.19)$$

Examination of the characteristics of those materials in which u varies with p shows that Eq. (3.18) and Eq. (3.19) can be expected to hold only over a limited range of values of p (9). A nonlinear representation of μ as a function of p will probably be necessary in most cases.

f - Derivation of Frictional Moment Expressions for Disk-Friction Clutch

In subsequent discussion, the variation of friction with sliding velocity is of importance. If the variation is linear, Eq. (3.14) may be used, and depending on which of the two assumptions, that of the uniform rate of wear or of the constancy of p over the frictional area, is taken upon which Eqs. (3.7) and (3.10) are integrated, one obtains two expressions for the frictional moment of the clutch.

Uniform Pressure Assumption :

In this derivation, p is invariant with r, and $\mu = \mu_o - m\omega_r r$. Eq. (3.7) is integrated

$$P = 2\pi p \int_{R_1}^{R_2} r\,dr = \pi p \left(R_2^2 - R_1^2\right) \qquad (3.20)$$

Values for μ and p from Eqs. (3.14) and (3.20) are substituted in Eq. (3.10). The integrated result is :

$$M = \frac{2}{3} P \mu_o \left(R_2^3 - R_1^3\right) / \left(R_2^2 - R_1^2\right) - \frac{1}{2} P m \left(R_2^2 + R_1^2\right) \omega_r \qquad (3.21)$$

For a clutch having n pairs of frictional surfaces in contact, the frictional moment is

$$M_n = M \cdot n \qquad (3.22)$$

Uniform Wear Assumption :

In this case, the integration of Eq. (3.7) results in

$$P = 2\pi K \int_{R_1}^{R_2} dr = 2\pi K \left(R_2 - R_1\right) \qquad (3.23)$$

Values for μ and p from Eqs. (3.14) and (3.22) are substituted in Eq. (3.10), and the integrated result is

$$M = 2\pi K \left\{ \frac{1}{2} \mu_o \left(R_2^2 - R_1^2\right) - \frac{1}{3} m \left(R_2^3 - R_1^3\right) \omega_r \right\} \qquad (3.24)$$

The expression for K, derived from Eq. (3.23), is substituted in Eq. (3.24), which becomes, for n pairs of friction surfaces,

$$M_n = \frac{1}{2} P \mu_o n \left(R_2 + R_1\right) - \frac{1}{3} P n m \frac{\left(R_2^3 - R_1^3\right)}{\left(R_2 - R_1\right)} \omega_r \qquad (3.25)$$

It is seen from Eqs. (3.22) and (3.25) that they have the form

$$M = M_o - \alpha \omega_r \qquad (3.26)$$

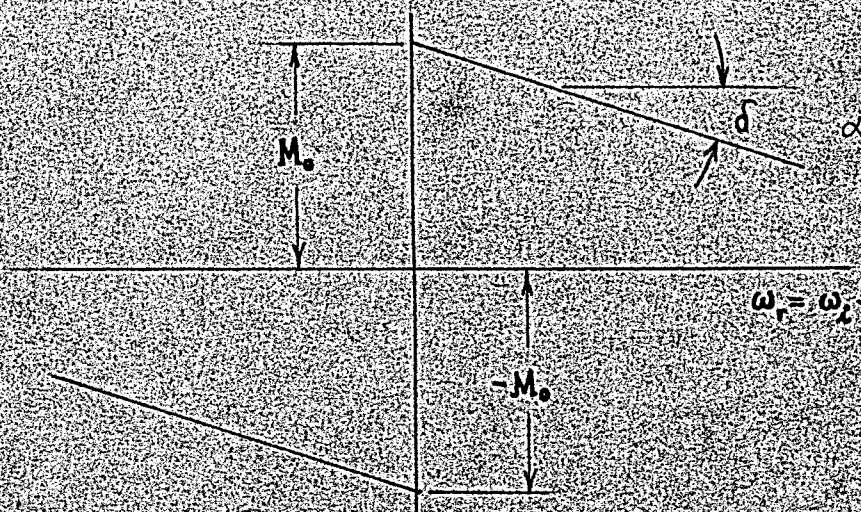


Fig. (7) Variation of frictional torque with slip velocity

where M_o is recognized as the static torque capacity of a friction clutch

$$M_o = \frac{2}{3} P \mu_o \frac{(R_2^3 - R_1^3)}{(R_2^2 - R_1^2)}$$

or

$$M_o = \frac{1}{2} P \mu_o (R_1 + R_2) \qquad (3.27)$$

depending on the assumption made in integrating Eqs. (3.7) and (3.10). Equation (3.26) is represented graphically in Fig. (7).

g - Comparison of Assumptions

Simple calculation will show that for clutches so designed that R_2 is not much larger than R_1 (in typical axial single-plate automotive disk-friction clutches; the ratio of inner to outer radius, R_1/R_2, varies between 0.6 and 0.7 in 90 per cent of the cases) both Eqs. (3.22) and (3.25) give values of M which, for all practical purposes, are identical.

In general, M obtained from Eq. (3.25) has a slightly lower value than that given by Eq. (3.22).

Let us examine the behaviour of the static torque capacities as $R_1 \to R_2$:

$$\lim_{R_1 \to R_2} M_o = \lim_{R_1 \to R_2} \frac{2}{3} \mu_o P \frac{(R_2^2 + R_2 R_1 + R_1^2)}{(R_2 + R_1)} = \mu_o P R_2 \quad \text{(uniform pressure)} \qquad (3.28)$$

$$\lim_{R_1 \to R_2} M_o = \lim_{R_1 \to R_2} \frac{1}{2} \mu_o P (R_2 + R_1) = \mu_o P R_2 \quad \text{(uniform wear)}$$

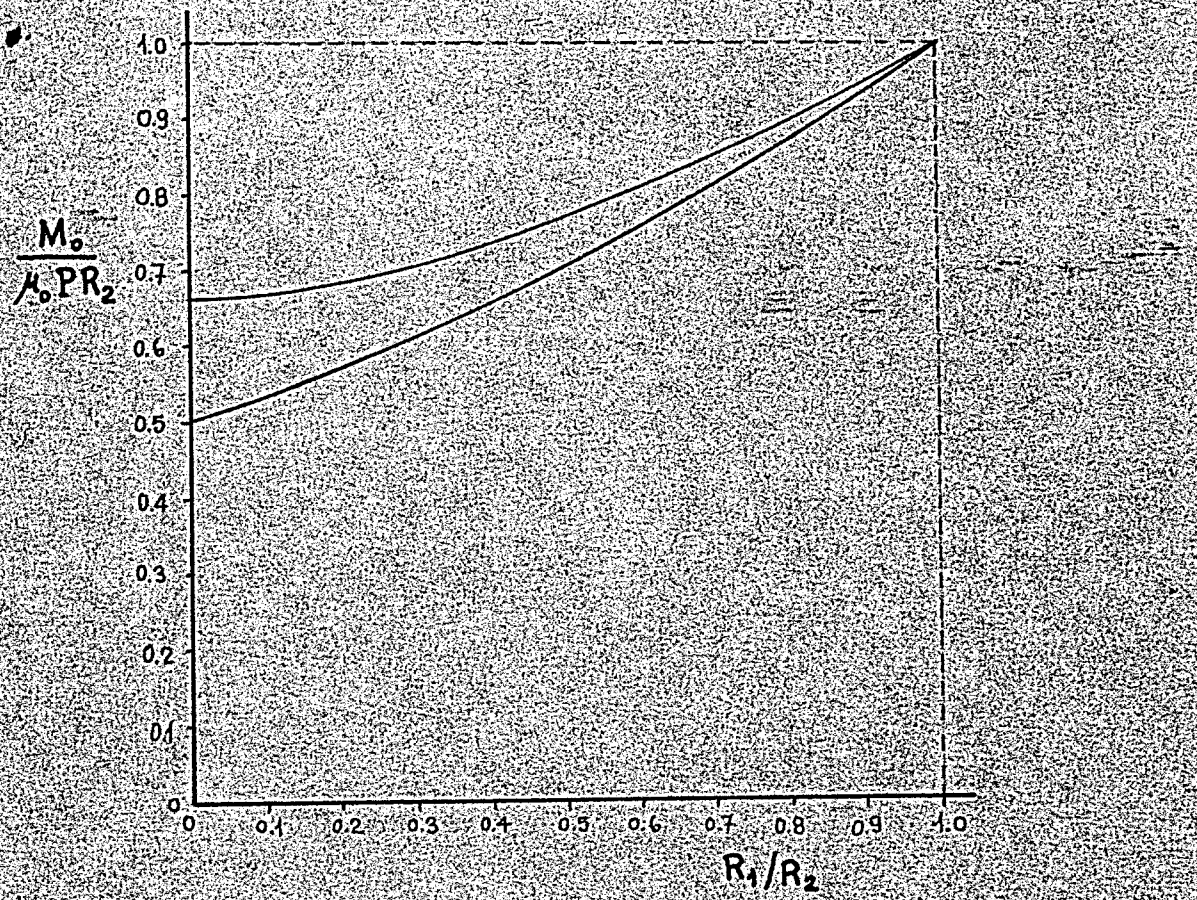


Fig. (8) Comparison of analyses of disk-friction clutch, based on different assumptions

Both solutions converge on the same value of $M_o = u_o R_2 P$, which is a reasonable limit, since all friction forces act at radius R_2, and the sum of all such forces is $u_o P$. Let us examine the behaviour of the solutions as $R_1 \to 0$:

$$\lim_{R_1 \to 0} M_o = \lim_{R_1 \to 0} \frac{2}{3} \mu_o P \frac{(R_2^2 + R_1 R_2 + R_1^2)}{(R_2 + R_1)} = \frac{2}{3} \mu_o P R_2 \quad \text{(uniform pressure)}$$

$$\lim_{R_1 \to 0} M_o = \lim_{R_1 \to 0} \frac{1}{2} \mu_o P (R_2 + R_1) = \mu_o P R_2 / 2 \quad \text{(uniform wear)} \tag{3.29}$$

To assure ourselves that the uniform wear assumption always predicts a smaller moment than the uniform pressure assumption, we place both equations in dimensionless form:

$$M_o / \mu_o P R_2 = \frac{1}{2} [1 + (R_1/R_2)] \quad \text{(uniform wear)}$$

$$M_o / \mu_o P R_2 = \frac{2}{3} [1 + (R_1/R_2)^2 / 1 + (R_1/R_2)] \quad \text{(uniform pressure)} \tag{3.30}$$

and plot $M_o/\mu_o P R_2$ versus R_1/R_2, as in Fig. (8).

Therefore, to be on the sage side, Eq. (3.25) being based on the uniform wear assumption seems better suited for computation of torque capacity.

B. The Effects of Rotational Speeds on the Bursting Behaviour of Disk-Linings

1- Stress Equations

Timoshenko (25) gives the following equations for the stress distribution in a rotating circular disk with a central hole:

$$\sigma_r = \frac{3+\nu}{8} \rho \omega^2 \left(R_2^2 + R_1^2 - \frac{R_1^2 R_2^2}{r^2} - r^2 \right)$$

$$\sigma_\theta = \frac{3+\nu}{8} \rho \omega^2 \left(R_2^2 + R_1^2 + \frac{R_1^2 R_2^2}{r^2} - \frac{1+3\nu}{3+\nu} r^2 \right) \tag{3.31}$$

where σ_r = radial stress component, kg/cm^2

σ_θ = tangential stress component, kg/cm^2

ν = Poisson's Ratio, dimensionless

ω = angular velocity of disk, 1/sec

R_1 = inside radius of disk, cm

R_2 = outside radius of disk, cm

ρ = mass density of disk material, kg-sec^2/cm^4

These equations have been derived under the following assumptions :

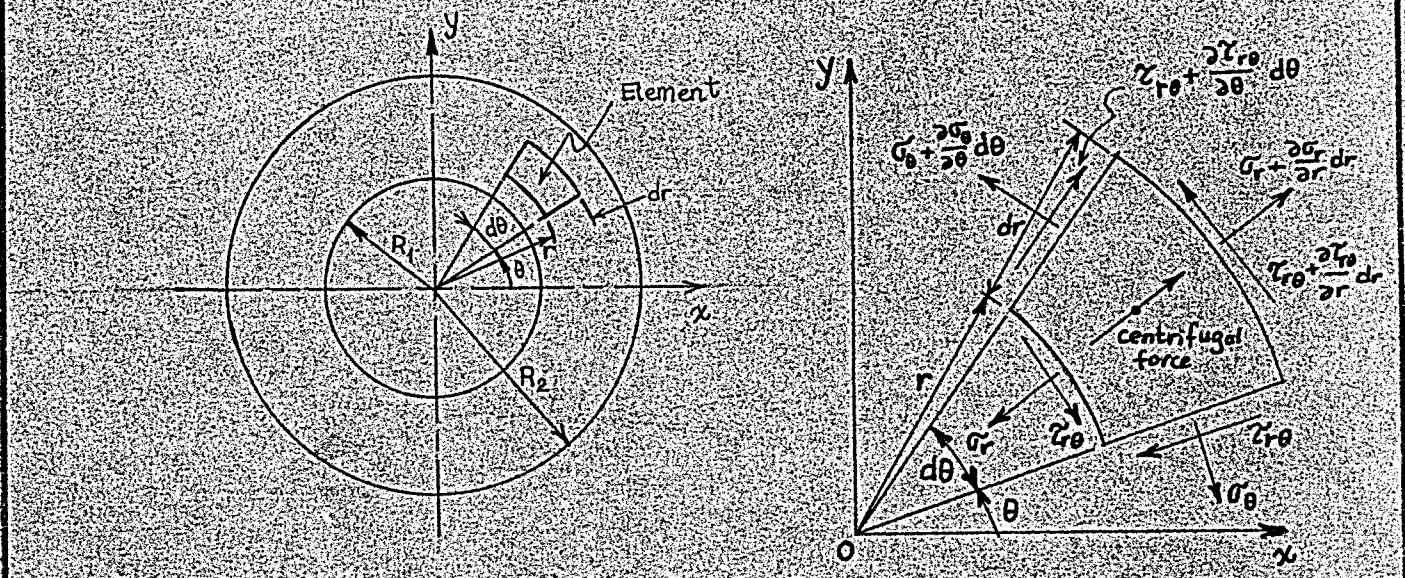


Fig. (9) Geometry of rotating circular disk with a central hole to show the use of polar coordinates to determine the stresses

(1) The disk material is isotropic and elastic,

(2) Since linearly elastic idealized material behaviour is assumed, Hooke's Law of Elasticity is applied,

(3) Relations between strains and stresses are completely defined by two constants E and ν, called the " elcastic constants ", (the modulus of elasticity in tension and the Poisson's Ratio, respectively),

(4) Thickness t of the disk is constant,

(5) Thickness of the disk is small in comparison with its radius, and variation of radial and tangential stresses over thickness can be neglected,

(6) Normal and shear stresses in the axial direction (axis of rotation or z-direction) are neglected, and the problem is said to be one of " plane stress ", since the only stresses of concern are those in the r, $\emptyset$ plane taken through the center of the disk

(7) The problem is a two-dimensional one,

Considerations for the justification of the above assumtions when applied to the case of the friction lining materials can be given as follows, in the order of the corresponding assumptions :

(1) Friction materials are two or more phase composites. The mix of linings usually contains disintegrated or woven fibres, together with resin powders and fillers, with a fine and perfect mixing. The composite material is often of an " elastic fibre - elastic matrix " type. Therefore, to an approximation, one can assume on a macroscopic scale the linings as being isotropic and elastic,

(2) The static tension test results for friction materials show a linear elastic stress-strain relationship with negligible plastic deformation before fracture. Therefore Hooke's Law may be applied to a fair approximation (1).

(3) The justification for this assumption follows from the above conclusions. The elastic constants may vary between the following values :

$$E \cong 2 - 3.5 \times 10^5 \ kg/cm^2 \ (3-5 \times 10^6 \ psi)$$

$$\nu \cong 0.04 - 0.06$$

(4) The clutch disk-facings are manufactured in the form of a constant-thickness disk of thickness t. The limits of t for automotive type facings are

$$3 \ mm \leq t \leq 8 \ mm$$

(5) The outside and inside diameters, D_2 and D_1, respectively, of disk-facings vary between the following limits in the automotive applications :

$$155 \ mm \ \phi \leq D_2 \leq 410 \ mm \ \phi$$

$$114 \ mm \ \phi \leq D_1 \leq 235 \ mm \ \phi$$

The ratio of inside radius to outside radius of disks varies between the limits

$$0.40 < R_1/R_2 < 0.85$$

We shall make a magnitude analysis for the determination of the variation of thickness to mean radius ratio of the facings. The mean radius can be expressed by

$$R_m = \frac{1}{2}(R_1 + R_2)$$

where R_m is the mean radius.

Considering the variation limits of R_1/R_2, the corresponding variation limits of R_m in terms of R_2 can be given by

$$0.7\ R_2 < R_m < 0.9\ R_2$$

Thus, the limits of R_m for about 95 per cent of automotive applications can be given as follows

$$110\ mm < R_m < 380\ mm$$

The variation limits of the t/R_m ratio given as

$$0.007 < t/R_m < 0.070$$

which is a very small ratio. Thus the assumption may be justified.

(6) and (7) Their justifications follow from the above analyses.

2 - Determination of the Bursting Stresses

Analyzing the equations of radial and tangential stresses in the rotating hollow circular disk with an inside radius of R_1 and an outside radius of R_2, one can determine that the maximum radial stress occurs at $r = \sqrt{R_1 R_2}$ and has a value

$$(\sigma_r)_{max} = \frac{3+\nu}{8} \rho \omega^2 (R_2 - R_1)^2 \qquad (3.32)$$

The maximum tangential stress occurs at the inner boudary or at $r = R_1$, and its value is

$$(\sigma_\theta)_{max} = \frac{3+\nu}{4} \rho \omega^2 \left(R_2^2 + \frac{(1-\nu)}{(3+\nu)} R_1^2 \right) \qquad (3.33)$$

A comparison of Eqs. (3.32) and (3.33) shows that

$$(\sigma_\theta)_{max} > (\sigma_r)_{max} \qquad (3.34)$$

Since ν is small for friction materials, we may write for $(\sigma_\theta)_{max}$ the following approximate equation :

$$(\sigma_\theta)_{max} \cong \frac{1}{4} \rho \omega^2 R_2^2 \left(3 + (R_1/R_2)^2\right) \qquad (3.35)$$

and since R_1/R_2 varies between 0.4 and 0.85

$$0.79\ \rho \omega^2 R_2^2 < (\sigma_\theta)_{max} < 0.93\ \rho \omega^2 R_2^2 \qquad (3.36)$$

We can express this also in the following way

$$(\sigma_\theta)_{max} \cong K \rho \omega^2 R_2^2 \qquad (3.36\,a)$$

where

$$K = 0.79 - 0.93$$

Winne and Wundt (26) give the following expression for the average tangential stress at bursting speed ω_b for a uniform thickness disk

$$(\bar{\sigma}_\theta)_b \cong \frac{1}{3} \rho \omega_b^2 R_2^2 (1 + X + X^2) \qquad (3.37)$$

where $(\bar{\sigma}_\theta)_b$ = average tangential stress at bursting speed, kg/cm^2

$\omega_b = 2\pi N_b/60$, 1/sec

N_b = rpm at bursting

ρ = mass density of disk material, $kg\text{-}sec^2/cm^4$

$X = R_1/R_2$, geometric factor

Considering the variation limits of X found in automotive practice

$$0.40 < X < 0.85$$

$(\bar{\sigma}_\theta)_b$ is expected to vary between the following limits :

$$0.5\ \rho \omega_b^2 R_2^2 < (\bar{\sigma}_\theta)_b < 0.9\ \rho \omega_b^2 R_2^2 \qquad (3.38)$$

which may be expressed also as

$$(\bar{\sigma}_\theta)_b \cong K' \rho \omega_b^2 R_2^2 \qquad (3.38\,a)$$

where

$$K' \cong 0.52 - 0.86$$

Comparing Eq. (3.38) to Eq. (3.33), we can calculate the variation limits of $(\bar{\sigma}_\theta)_b$ in terms of $(\sigma_\theta)_{max}$. Thus,

$$(\bar{\sigma}_\theta)_b \cong 0.66 \div 0.92 \ (\sigma_\theta)_{max}$$

Therefore, Eq. (3.37) gives safer values for the dynamic bursting strength of a rotating circular disk with a central hole.

In the most general case, the bursting stress is a function of ρ, ω^2, and $X=R_1/R_2$ and R_2^2. This can be expressed by the general functional relationship:

$$(\bar{\sigma}_\theta)_b = f(\rho, \omega^2, R_2^2, X)$$

One can define a bursting-speed parameter as $[\rho(\omega_b R_2)^2]$ and express $(\bar{\sigma}_\theta)_b$ in terms of it as follows

$$(\bar{\sigma}_\theta)_b = f\left([\rho(\omega_b R_2)^2], X\right)$$

C. The Effects of The Operating Conditions on Thermal Failure of Linings

1 - Thermal Problems of Friction Materials

When one solid body slides over another, most of the work done against frictional forces oqposing the motion will be liberated as heat at the interface. Consequently, the temperature of the rubbing sunfaces will increase and may reach values high enouh to destroy the clutch.

Even under moderate loads and slow speeds quite high instantaneous values of surface temperature are reached at points of actual contact of the two

surfaces. Thermoelectric methods employed in the measurement of these transient local temperatures indicate values as high as 1000°C, and their duration has been observed to be of the order of 10^{-4} seconds (9). The results are similar when the surfaces are lubricated with mineral oil except that peak temperatures are reduced (12). Also, it has been observed that, when large quantities of heat are liberated at the rubbing interface during a short time interval, high surface temperatures are generated while the bulk of the two bodies in contact remains relatively cool (9). This is because of the low thermal conductivity of the friction materials which varies between 0.35 - 0.50 cal/h-m-°C (3 - 4 Btu-in./hr-ft^2-°F) for a mean temperature of 50°C (125°F).

Most premature clutch failures can be attributed to excessive surface temperatures generated during slipping under axial load. In the case of asbestos-based nonmetallic clutch linings, which are used with steel or cast-iron mating plates, the friction material usually disintegrates because of high temperatures, and if the temperature penetration into the bulk of friction coating is appreciable, the material will be completely removed from the steel core of the plate (9).

Other effects of high surface temperatures are distortion of the shape of the linings, surface cracks on facings caused by thermal stresses,

appreciable fluctuation in the value of the friction coefficient ('fadign'),

The phenomenon of thermal failures of linings may be summarized as follows :

The fact, that perfect contact between the lining and the metal part seldom exists, and that the friction force is generated mainly at local areas of contact which move about the lining surface, leads to local high temperatures. These local high temperatures when associated with cooler surrounding metal can cause heat spots and thermal cracks, through the intense heating and rapid quenching of the metal.

This phenomenon is further analyzed in the next paragraph.

2 - Flash Temperatures

Even the smoothest-looking friction surfaces are sufficiently rough to be determining factors in the life of friction materials. Concentrated loads at high points cause flash temperatures well above the bulk temperatures of adjacent clutch components.

Very high values of transient surface temperature occur at the interface between two sliding bodies, these temperature flashes occurring at points of intimate contact between the two surfaces.

The effective or microscopic contact between the two surfaces is confined to surface asperities, and it can be shown that the relationship between the effective and the nominal or microscopic area of contact is

$$A_F / A_f = p / p_m \qquad (3.39)$$

where A_F = effective (microscopic) frictional area over which contact actually occurs

A_f = nominal (macroscopic) area of clutch face contact

p = nominal pressure (avergge pressure)

p_m = mean yield pressure of the softer of two friction materials

If contact occurs at n spots, and if these spots are assumed to be circular in shape and equal in size

$$A_F = \pi r^2 n \qquad (3.40)$$

where r=radius of the spot.

When relative motion exists between the surfaces, it has been observed that a number of luminous points appear at the rubbing interface and that the position of these spots changes rapidly as points of intimate contact wear away and new asperities come into contact (12), (9). Referring to Fig. (10) which represents a portion of a clutch facing, let S be the distance covered by such an isolated hot spot in the small time interval t_s. Then

$$t_s = \frac{S}{\omega R_m} \qquad (3.41)$$

where ω = angular velocity, rad/sec

R_m = mean radius of the facing, cm

Also let

$$S \propto r \qquad (3.42)$$

If a clutch with only one frictional surface is considered, the rate at which heat enters each spot is given by

$$q_r = \frac{M\omega}{\pi r^2 n}\,\frac{1}{J} \qquad (3.43)$$

where M = clutch frictional torque, kg-cm

J = conversion factor from kg-cm to kcal,

Because t_s is very small (of the order of 10^{-4} sec), q_r will remain essentially constant. Also, the effect of heat reflection from the other side of the plate can be neglected. This assumption allows the use of the simplest expression for the surface temperature, which is derived for the case of a semi-infinite solid receiving heat at a constant rate (27):

$$\theta_f(t) \propto \sqrt{1/k_s c_s \rho_s}\; q_r \sqrt{t_c} \qquad (3.44)$$

where θ_f = flash temperature (hot spot temp.)

k_s = thermal conductivity of steel pressure plate

c_s = specific heat of steel pressure plate

ρ_s = density of the steel pressure plate

t_c = time per cycle, sec.

From $M = \mu p A_f R_m$ and $A_f \propto R_m^2$, and Eqs. (3.39) through (3.44), the following expression is obtained:

$$\theta_f \propto \sqrt{1/k_s c_s \rho_s}\;(\mu p_m^{3/4} R_m p^{1/4} \omega^{1/2})/(n^{1/4}) \qquad (3.45)$$

We shall assume further that the maximum rubbing interface tempe-rature, θ_{max}, is proportional to θ_f. Thus

$$\theta_{max} \propto \theta_f \qquad (3.46)$$

From the previous analyses, we can take

$$\theta_{max} \propto M \omega^{1/2} \qquad (3.47)$$

This can be expressed in terms of μ, p, and v in the following manner

$$\theta_{max} \propto (\mu p v^{1/2}) \qquad (3.48)$$

This relationship will be analyzed in the experimental studies.

3 - Flash Temperature Considerations in Lining Selection

Eq. (3.45) indicates that, in order to decrease tha value of flash temperature occurring at the areas of intimate contact, the following points should be considered when selecting friction material (and clutch mating plate or pressure plate size) :

(1) Soft friction materials having good surface conformability are preferable. (p_m should be low.)

((2) The friction coefficient should be low. Since, in order to minimize clutch size, high values of are usually preferred, a material having sharp fade characteristics at high temperatures should be used.

(3) Contact between the mating plates should be as uniform as possible; i. e., the number of contact spots n should be as large as possible.

(4) For a clutch of given nominal frictional area A_f, the mean radius of the plate should be kept small. Since, for the plate of a given oiitside diameter, the frictional torque of the clutch is proportional to R_m, a compromise decision regarding the value of R_m should be made which takes into account both flash temperatures and the value of the

frictional torque desired.

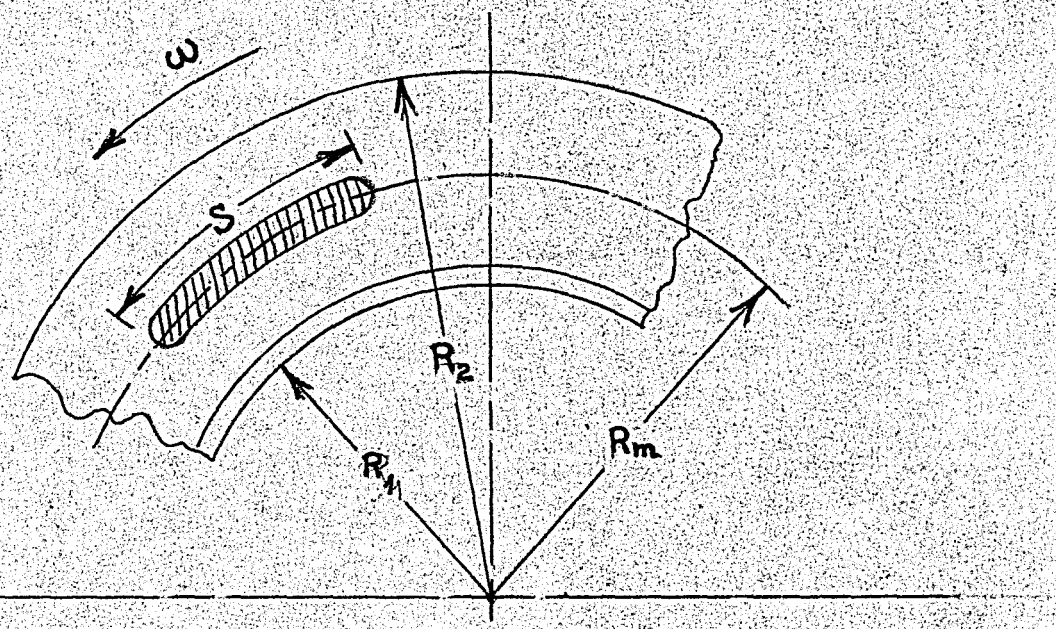


Fig. (10) A portion of a clutch facing showing the distance covered by an isolated hot spot.

IV - EXPERIMENTAL ANALYSIS OF FRICTION CLUTCH LINING PERFORMANCE

A. Experimental Methods of Friction Materials Testing

1 - General

There are two basic types of experimental methods used in engineering applications :

(1) applied tests under actual operating conditions

(2) laboratory experiments under similar conditions to actual practice

Frequently, the second type of experimentation is preferred, since the phenomena are easier to handle. Laboratory experiments should be preferred in the testing of friction materials because of the following reasons :

(1) In the study of friction and wear, since the knowledge of the actual physical phenomena (dynamic, chemical, thermal, etc.) is restricted, one has to simulate the phenomena.

(2) Slight changes in actual operating or service conditions may cause tr tremendous changes in performance characteristics.

(3) Difficulties (or impossibilities) in controlling the random changing patterns of performance characteristics.

(4) Applied tests are more time consuming.

(5) The results of the applied tests cannot be generalized even for the same type of experimental apparatus, i. e., engine or vehicle.

(6) Great difficulties in making measurements of variables.

(7) Experimental expenses are very high and only large scale industries can resort to them.

Therefore, laboratory experiments are carried out under similar conditions to actual case. We can classify the laboratory experiments into two main groups :

(1) The experimental unit or apparatus is equivalent to the applied unit, i.e., it has the same dimensions as the actual unit where the tested specimens are to be used.

(2) The experimental system is similar to the actual unit, but the design of the system may be in such a form as to permit the control of operating conditons in an efficient way. The test specimens may be smaller in size when compared to the dimensions of the applied material.

The results of the second type of lab experiment are in general more conveniently and more economically arrived at. The most important point in evaluating the results of this type of experiments is that one must be very careful by drawing conclusions about the actual performance.

2 - Friction Material Testing

Friction materials for clutches and brakes are tested by various test

methods and testing apparatus being developed according to the general principles outlined above. The test methods can be grouped as follows :

(1) vehicle test - (applied test)

(2) full-size sample lab test

(3) small-size sample lab test

a - Vehicle Test

The vehicel is equipped with new brake and clutch linings and is operated. This method is generally preferred by the automotive industries because it offers the following advantages :

(1) Since the friction linings are tested under actual operating conditions, one may obtain satisfactory results for performance characteristics.

(2) The important lining performance characteristics such as soft engagement, slipping while engaging, chatters or grabs when engaging, spins or drags when disengaged, noisiness, pedal stiff can be observed and detected easily, assuming the wear-in period has passed.

(3) Since the linings are tested with the vehicle they are are intended to operate, the factors influencing the functioning of clutch system are controlled quite efficiently.

The disadvantages of the vehicle test may be given as :

(1) In order to obtain satisfactory results about performance of the linings, a great number of vehicles should be tested under the

same operating conditions,.

(2) The experimental expenses amount to high values, and small scale industries will be unable to afford these tests.

(3) Since the vehicle is in motion, and since it has limited space available for measuring instruments, great difficulties would arise in making efficient measurements of the experimental variables.

(4) Since all external factors are operating in an uncontrollable manner, it is extremely difficult to evaluate the effects of these factors separately.

(5) Because of the difficulties of controlling the test conditions and of separating the effects of the operating conditions, it become nearly impossible to detect the causes of deficiecies in friction linings, in most of the cases.

(6) It is extremely difficult to compare the results obtained at different times and conditions, with different vehicles, even under similar operating conditions. Generalization of results with respect to lining performance will be misleading.

(7) The vehicle test gives results which help to evaluate the performance of the complete clutch system, but <u>not</u> of the friction pairs, especially of the lining.

All of the seven disadvantages listed above limit the value of the systematic research on friction materials by the applied vehicle test.

b - Laboratory Test - Full Size

In order to nemove some of the disadvantages met in the vehicle test, especially the difficulties in controlling the external effects, complete full-size clutches are tested in the laboratory.

Both the vehicle test and the full-size laboratory test yield results which give sufficient information about the characteristics of the complete clutch unit. But is is difficult to draw conclusions about the performance characteristics of clutch linings alone, since the lining serves only as an element in the complete unit. Especially, it is difficult to find out the friction coefficient which is the most important quantity characterizing the friction material. Furthermore, these test methods have limited value in determining friction material chaaacteristics such as friction coefficient and wear rate under the varying operating conditions of velocity, pressure and temperature developed.

For introducing ssome developments in friction material compositions, or for studying their behaviour under the varying operating conditions, a series of experiments are necessary. With regard to their usefulness and efficiency, these test methods are unpractical and expensive for the testing of friction linings.

c - Laboratory Test - Small Sample

Because of the shortcomings of the vehicle test and fullsize laboratory test in evaluating the performance characteristics of the friction materials, direct test methods have been established which would be helpful in the investigation of the lining material properties, only.

In order to minimize the effects of the variations of the material properties and the resulting differences in performance at different locations on the same lining and from lining to lining, the sizes of the test specimens are selected to be small. Generally, the test specimens have the form of small square blocks of about one inch sides.

In the testing of clutch friction facing materials the small square sample test may give useful results.

With the help of this test, the effects of the loading variables such as p, v, and θ_{max} on friction and wear may be studied efficiently. In order to specify the effects of these variables completely, two of them are held constant each time while the third is changed. For example, in order to study the effects of high temperature, the temperature is increased at constant pressure and rubbing velocity. In practice, generally, in the search for better friction and wear characteristics, the lining specimen is exposed to higher pressures and velocities as a result of which the interface temperature reaches

higher values.

For obtaining practical results from tests, the values of loading variables are increased simultaneously. The tests are started at small values of p, v, (and θ max) and are continued by increasing their values at the same time.

If the results of the friction and wear performance tests with small square block specimens, and the results of another type of test, namely the disk-bursting test at high rotational speeeds with full-size facing specimens, are obtained and evaluated, one can specify the performance characteristics of the lining to a good accuracy.

3 - Development of the Experimental Systems

a , Friction and Wear Test

The survey of American and German Standards, namely S. A. E., A. S. T. M., and DIN, has indicated to the author, that there is no standard quality control test procedure for clutch friction linings. This fact has led the author to the design of a test procedure which is similar to brake lining quality control test procedures, especially " SAE Recommended Practice - SAE J66la ".

Purpose

Purpose :

The purpose of this test method is to establish a uniform laboratory procedure for securing and reporting the friction and wear performance characteristics of clutch linings. The performance data obtained can be used for in-plant quality control by clutch lining manufacturers and for the quality assessment of incoming shipments by the purchasers of clutch linings.

Equipment :

A typical test machine, similar to Friction Materials Test Machine mentioned in SAE J 661a, has been developed for use in the friction and wear tests. This FMTM shall be equipped with suitable means for :

(1) measuring the mating plate temperature

(2) heating the mating plate

(3) controlling the mating plate heating rate

(4) cooling the mating plate

(5) controlling the mating plate cooling rate

(6) measuring the friction force

(7) measuring the rotational speed

Means shall be provided for measuring specimen thickness and weight.

The temperature measuring means shall incorporate an embedded thermocouple, coin silver slip rings, silver graphite brushes, and a pyrometer having a high input impedance.

The friction force and coefficient shall be measured by strain-gage techniques.

The temperature measuring system shall have ±2 per cent full-scale accuracy.

The friction force measuring system shall have ± 2 per cent full-scale accuracy.

The mating plate rotational speed measuring system shall have ± 2 per cent full-scale accuracy.

The mating plate diameter shall be between the limits of 360-365 mm (outside diameter).

b - Disk-Bursting Test

The survey of American and German Standards has shown to the author that there is no standard test procedure for testing disk-type friction clutch linings (facings) at high rotational speeds. This fact has led the author to the design of a simple testing system which will enable one to study the bursting behaviour of facings at two to three times higher rotational speeds than found

in practice.

Purpose :

The purpose of this test method is to establish a uniform laboratory procedure for reporting the bursting characteristics of facings. The rotational speeds applied in this test shall be up to three times higher than found in automotive practice in order to allow for safety requirements during actual operation. This method also enables one to obtain an idea about the magnitude of the upper limit of the centrifugal stresses the facing can withstand.

Equipment :

A simple test apparatus has been developed which is able to rotate up to 12,000 rpm with the facing attached to its specimen holder. The Disk-Bursting Test Apparatus shall be equipped with suitable means for :

(1) measuring the disk rotational speed

(2) specimen holder for factening the facings tightly and not allowing relative motion at high speeds.

(3) measuring the power of the prime mover, which is a Ward-Leonard Motor Unit with continuos variation of speed and power output, at any instant.

(4) safety guards for protecting the men and equipment from the bursting particles.

4 - Design of Experimental Apparatus

a - Friction and Wear Test

Principles of the Experimental Set-up :

A sketch of the Friction Materials Test Machine (FMTM) is shown in Fig. (11) [x]

The steel mating plate or disk is keyed to a horizontal shaft supported by two roller bearings. The test specimen being fixed to a specially designed arm mechanism is pressed against the disk with a known force. If the disk is rotated with the desired speed, the required relative slipping between the fixed specimen and the disk is achieved. The normal force with which the test specimen is pressed against the rotating disk (surface pressure) is varied by adding known weights to the load hanger end of a lever mechanism whose other end is supported at point A with a roller bearing and a horizontal shaft. The uniform contact of the specimen to the disk is secured with a joint-positioning mechanism (see Fig. (12)). The vertical arm carrying the specimen holder is designed in such a way that its deformation under the effect of the normal and friction forces will be negligibly small. It is connected to an edge member which is attached to the lever arm by means of a rectangular shaped thin member from spring steel. This member will undergo negligible deformation by the effects of the moments produced by the reaction against the pressing of the specimen,

[x] The test systems have been developed in the Laboratory of Machine Elements of the Faculty of Mechanical Eng'g, Istanbul Technical University.

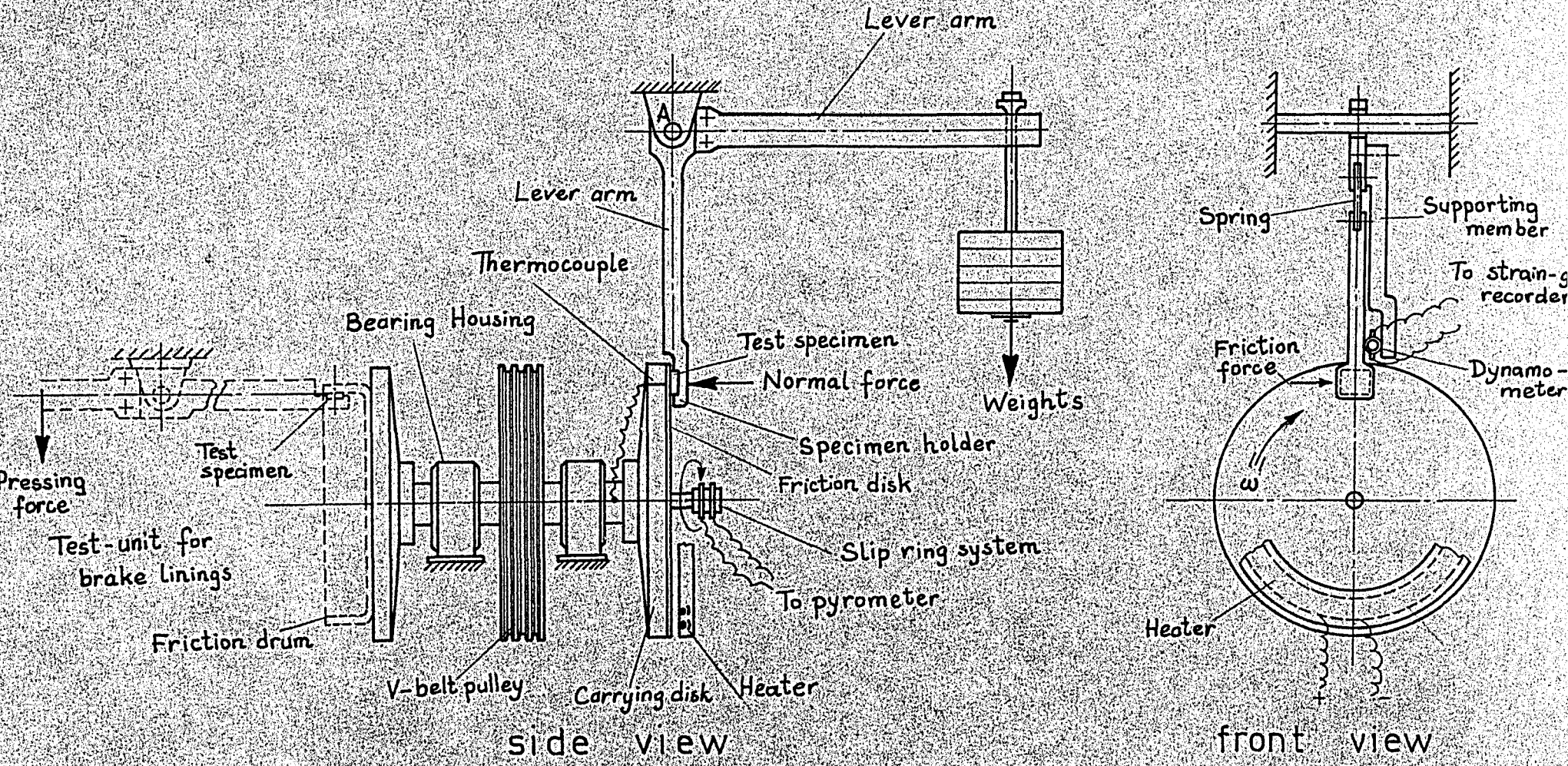


fig.11 – Schema of Friction and Wear Experimental Setup

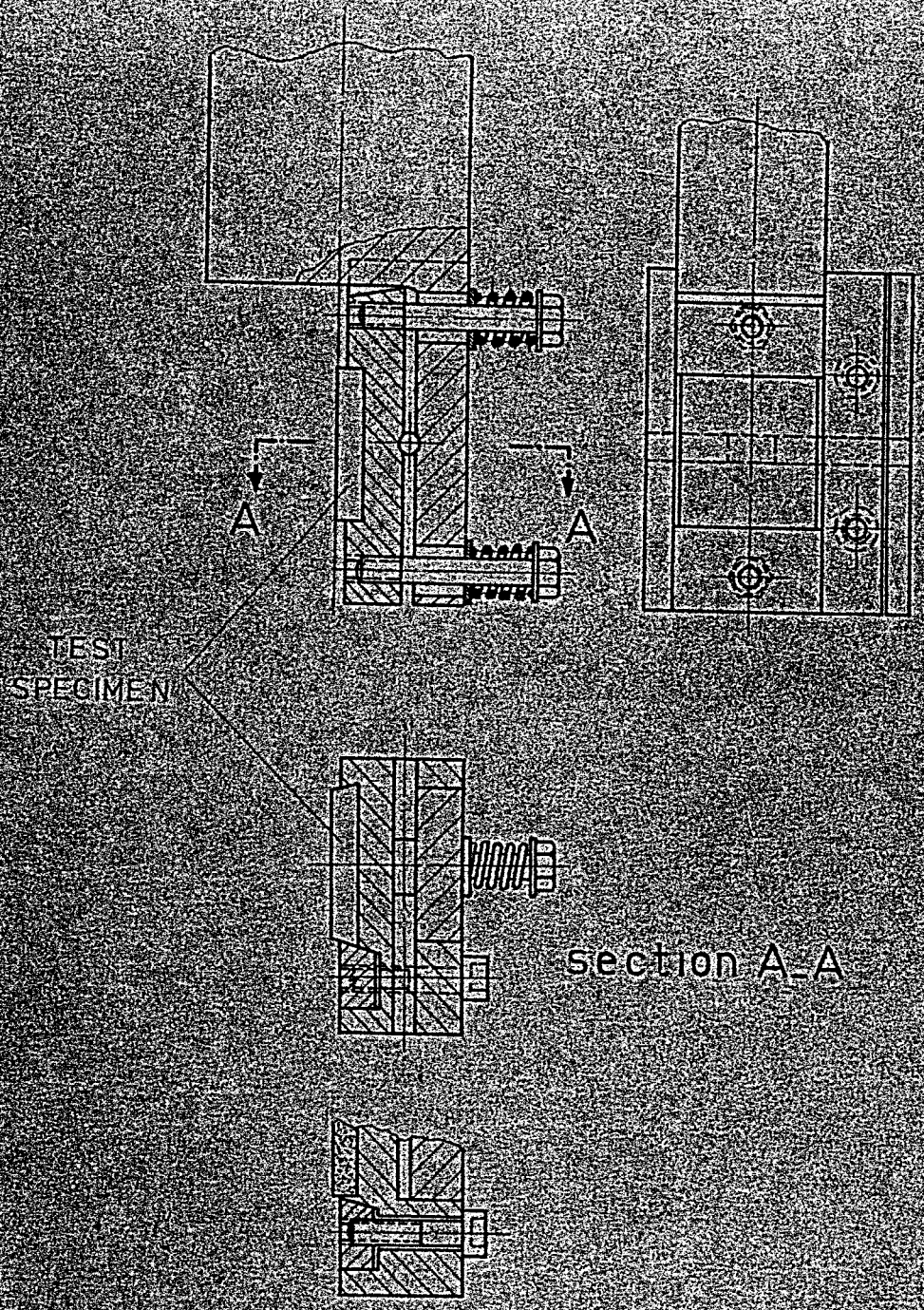


fig 12– Test Specimen Holder Details

since according to its position and cross-section its moment of inertia in this direction is very large. On the contrary, the springs will undergo relatively large displacements by small forces, since its moment of inertia of the cross-section in the direction of the friction force is small. The vertical arm being designed for rigidity, all the deformation will be taken up by this spring. Parallel to the vertical arm and rigidly dimensioned, a supporting plate is attached to the edge member at the top of the spring. Between the vertical arm and the supporting plate a dynamometer of tubular form is placed. On this dynamometer strain-gages are applied. Under the effect of the friction forces the vertical arm will press to the dynamometer and from the deformation of the dynamometer and with the help of the straim-gages the friction forces will be measured.

During the experiment, the temperature of the friction surfaces - both the disk and the lining specimen - will be measured by means of embedded thermocouples. The measuring junction of the thermocouple is brought to the disk friction surface through a hole drilled into the disk, whereas the other thermocouple end is passed through a central hole in the shaft carrying the disk, up to the silver-graphite brushes and slip rings mechanism placed at the end of this shaft. The thermocouple wire ends taken from the slip-ring system are

connected to a pyrometer with a high input impedance. The temperature of the test specimen is measured by means of another thermocouple embedded into it.

The system is driven by means of a V-belt driving mechanism.

The principles discussed above have been realized by the experimental system shown in Figs. (13) and (14).

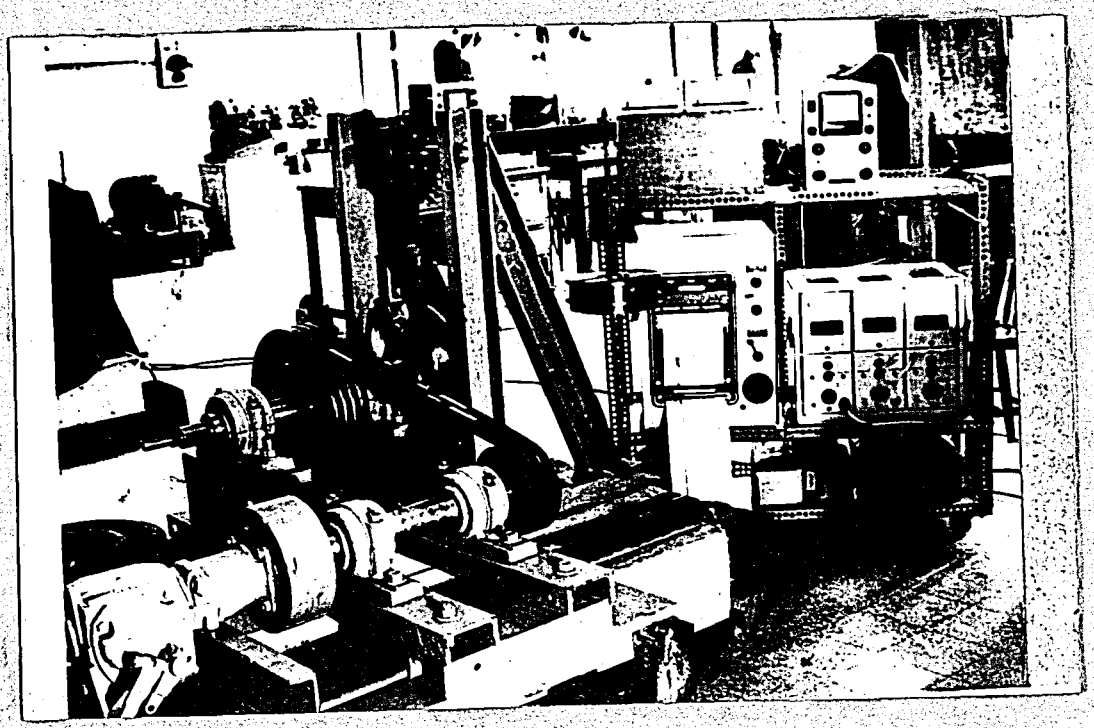


Fig. (13) Side-view of the F & W Experimental System

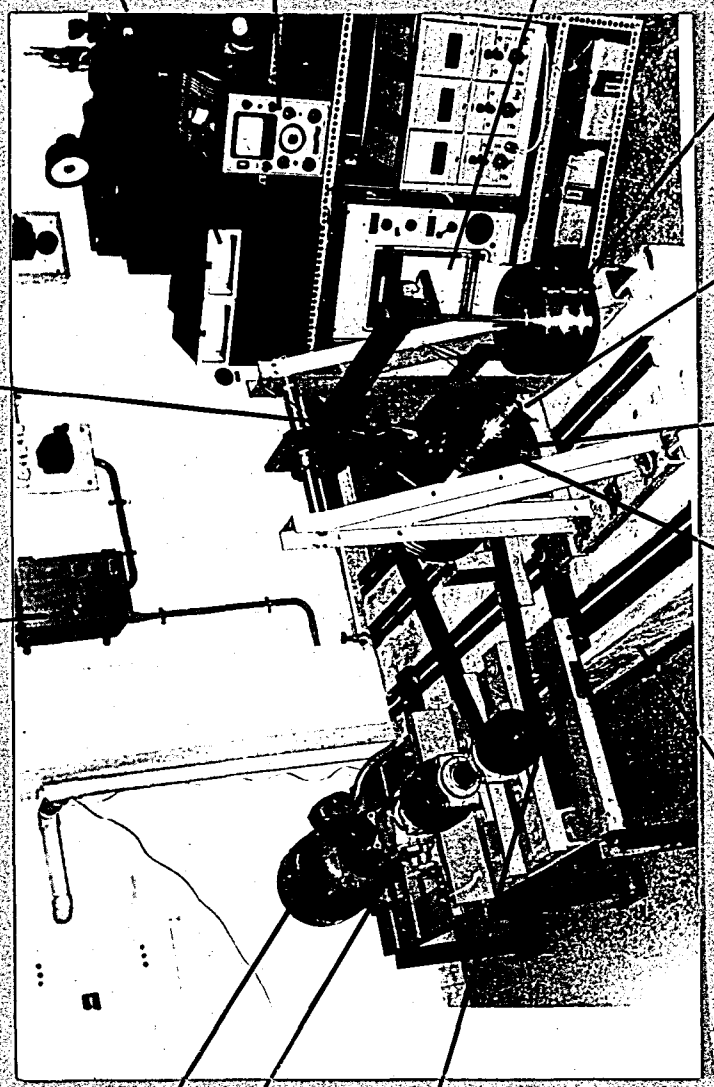


fig.14- Overall view of experimental setup

System Details :

The friction disk is 12 mm in thickness, has 360± 5 mm outside and 220 ± 5 mm inside diameters, and is connected to the main disk (being centered) by means of three bolts, as shown in Fig. (15).

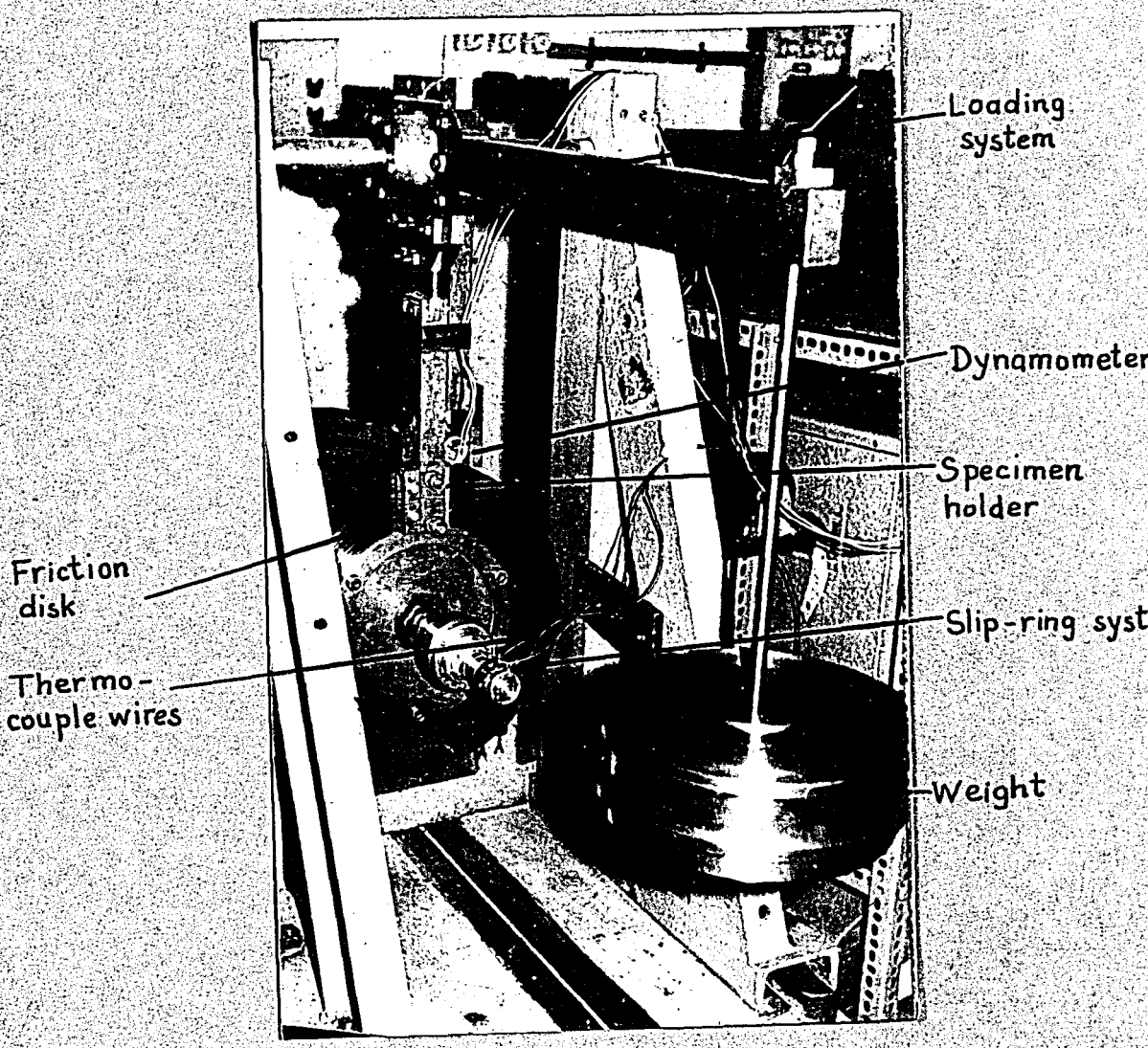


Fig. (15) Closeup of friction disk, dynamometer, and loading system

The disk friction surface has a mean diameter of 290 mm and an arc-strip width of 30 mm. The shaft carrying the friction disk and the V-belt pulley is horizontally supported by two self-aligning roller bearings.

Fig. (15) shows the specimen carrying system and the dynamometer for force measurement. The loading of the system is done by putting weights into the load hanger attached to the lever arm. The other end of the lever arm is supported on a knife-edge. The vertically acting force of the weights produces the pressing force or the contact pressure on the friction disk, by means of the horizontal lever arm and the vertical arm connected to each other at a right angle. In order to minimize the friction losses in this load transmission mechanism, the horizontal lever arm is supported on two single-row roller bearings. The cantilever spring attached to the vertical arm has a cross-section of 4x50 mm, the ratio of its sectional moment of inertias with respect to neutral axes is 1/150, and can be deflected easily in the direction of the frictional forces.

The details of the specimen holder are shown in Fig. (12). As shown in Fig. (11), in the system sketch, the vertically acting weight force is transformed into the normal force on the specimen by means of the perpendicularly connected lever arms and the turning joint at point A. In this case, the pressure distribution

on the specimen surface will s how a linear variation in proportion with the distance from point A, as shown in Fig. (16). By selecting a small size specimen and holding the distance to the axis of rotation relatively large compared to the specimen dimension the difference in pressure between the upper and lower part of the specimen is kept very small. In order to secure a homogeneous pressure distribution on the surface, the specimen holder is supported on a cylindrical pin-bearing located at the back center of the specimen, located parallel to the specimen and perpendicular to the vertical arm direction. Thus, a freedom of rotation about its central axis is given to the specimen. The applied normal force acts on the centre of the specimen since it is transmitted via this pin. Because of the rigid design of the specimen holder, the centrally applied normal force can produce a homogeneous pressure distribution.

By measuring the distance between the pin and the axis of rotation at joint A, we can calculate the normal force by the equilibrium equations and the known weights of the loading system.

The vertical arm carrying the specimen holder is bent in the region of the cantilever spring under the effect of the friction forces acting on the specimen surface in the direction of the rotation of the disk.

In order to avoid a complex force development by a torsional moment produced by the friction forces and to let the dynamometer work only for compression between the two pressing surfaces, the specimen is

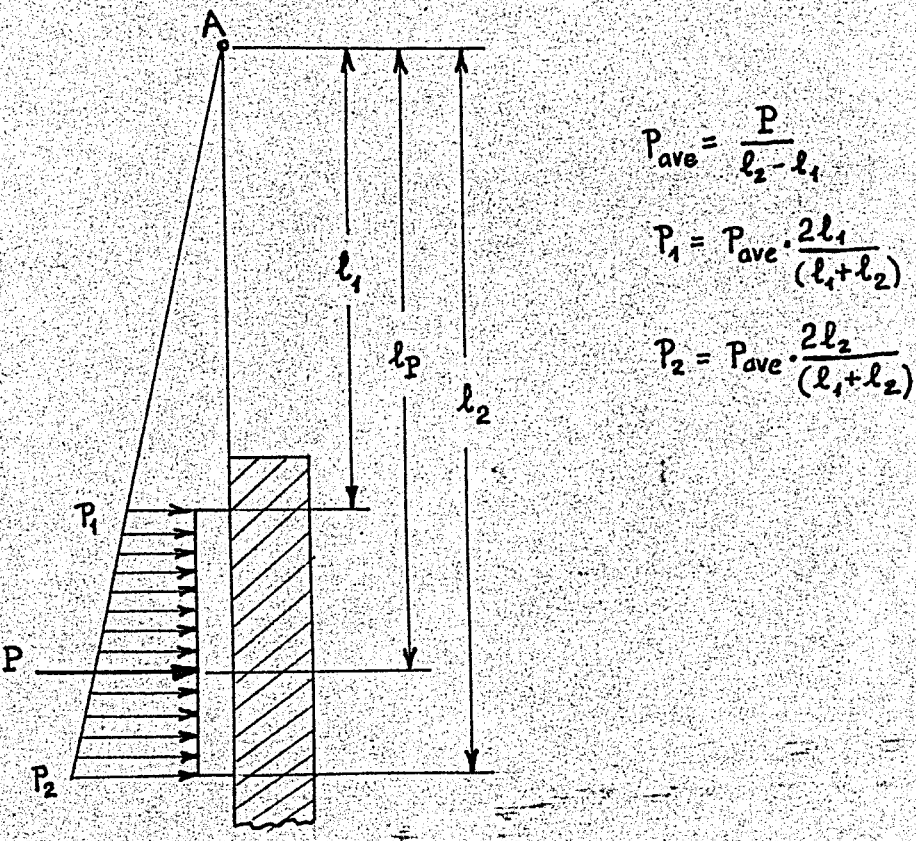


Fig. (16) Pressure distribution on the specimen (exaggerated).

located in such a way that the central symmetry axes of the test specimen, of the spring, and of the vertical arm coincide, Fig. (12)

The specimen cannot move about its vertical axis. The specimen holder is attached to the vertical arm and into a press-fitted housing. The friction forces acting at the rubbing disk-specimen interface are transmitted to the vertical arm by the reaction on the sides of this housing

The full contact (macroscopic) of the specimen to the disk is achived after measuring the space between the two by means of feelers up to 0,1 mm accuracy duning the mounting of the specimen, and by bringing them into parallel position.

The specimen holder is pressed against the vertical arm and the alignement pin by means of two bolts passing through two prestressed springs in order to fix the specimen to its housing.

The dynamometer placed between the vertical cantilever arm with the specimen holder and the support plate, is shown in Fig. (15). It has been manufactured from aluminum in the form of a tube of 30 mm outside diameter. On one side, it is bolted to the support plate with two bolts, on the other side it is pressed or squeezed by the vertical arm. In this way, it is deformed under the action of the friction forces. It is attached to its position after being prestressed. Onto the dynamometer, in the direction of the friction forces, an active strain-gage is attached. Below this, in the axial direction and in the perpendicular direction to the active gage, a dummy strain-gage is attached to the dynamometer in order to compensate the effects of the unavoidable thermal expansion. The system has been calibrated after the dynamometer has been placed into its position. The two strain-gages are coupled into a half-bridge circuit, thus allowing for thermal compensation. By using

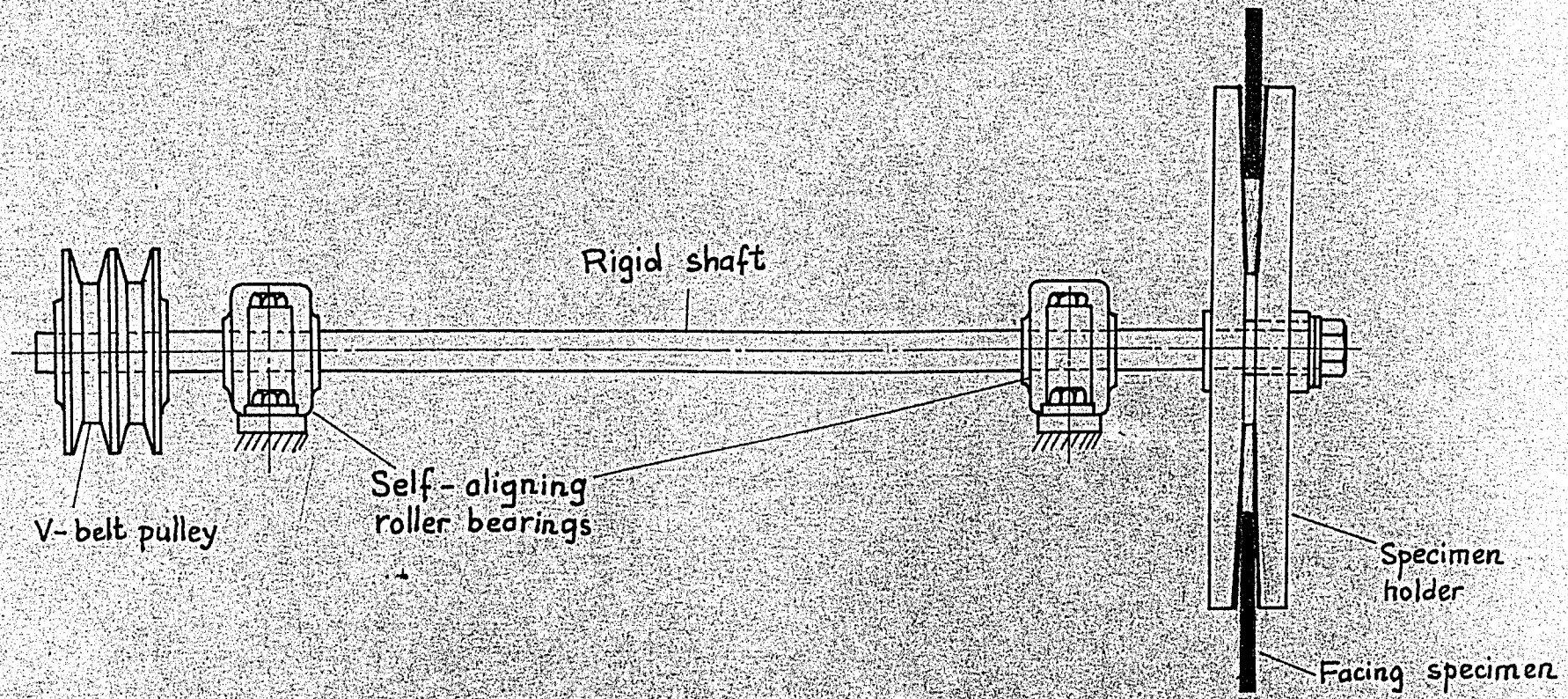


fig. 17 - Disk Bursting Test System Details

the dynamometer and the strain-gage amplifier together with the recorder, we can measure the friction force and the friction coefficient (see section (IV. B. 2).

The test system is driven by a non-synchronous electric motor (N=15 PS, n=1430 rpm) with a gearbox of three speed stages and a V-belt drive, Fig. (13). By changing the small V-belt pulley on the output side of the gearbox, and using a smaller one, we can obtain higher rotational speeds from the friction disk.

Table (1) below shows the circumferential speeds which can be obtained from the system.

TABLE (1) The circumferential speeds of the friction disk (mean dia. d_m = 290 mm, motor rpm n = 1430)

Gear	V-belt ratio	Disk rpm	Ave. exp. speeds (m/sec.)
1	1: 2.34	235	3.57
2		410	6.23
3		690	10.50

b. Disk-Bursting Test Apparatus

A sketch of the " Disk-Bursting Test Apparatus " developed by the author is shown in Fig. (17).

The system consists of a rigidly designed shaft carrying a disk specimen holder made of two clamping steel disks and a small V-belt pulley on both ends. The shaft is supported by two self-aligning roller bearings. The

system is driven by a V-belt drive mechanism, the prime mover being a Ward-Leonard Motor Unit with continuous speed variation. The large V-belt pulley is keyed and bolted at the outer end to the shaft of the motor as the driving pulley, whereas the small pulley is keyed to the rigid shaft of the system as the driven pulley. Thus, the V-belt drive gives a speed increase of 4 : 1. The rpm of the motor can be increased continuously from zero to 3,000 rpm, thus allowing the system to rotate at 12,000 rpm.

The system is enclosed in a safety-guard for protecting the man and equipment from the bursting particles.

The system details are shown in Fig. (18) through (21).

B. Measurement of Experimental Variables

1 - Speed Measurements

a - In the Friction and Wear Tests

The rotational speed of the disk has been measured for each velocity stage of the gearbox, both in loaded and disengaged position, by means of a tachometer and also with a stroboskobe for speeds above 600 rpm. Thi circumferential speed and the slip speed have been calculated for a mean friction disk diameter of $d_m = 290$ mm.

b - In the Disk-Bursting Test

The rotational speed of the Ward-Leonard motor is indicated on the

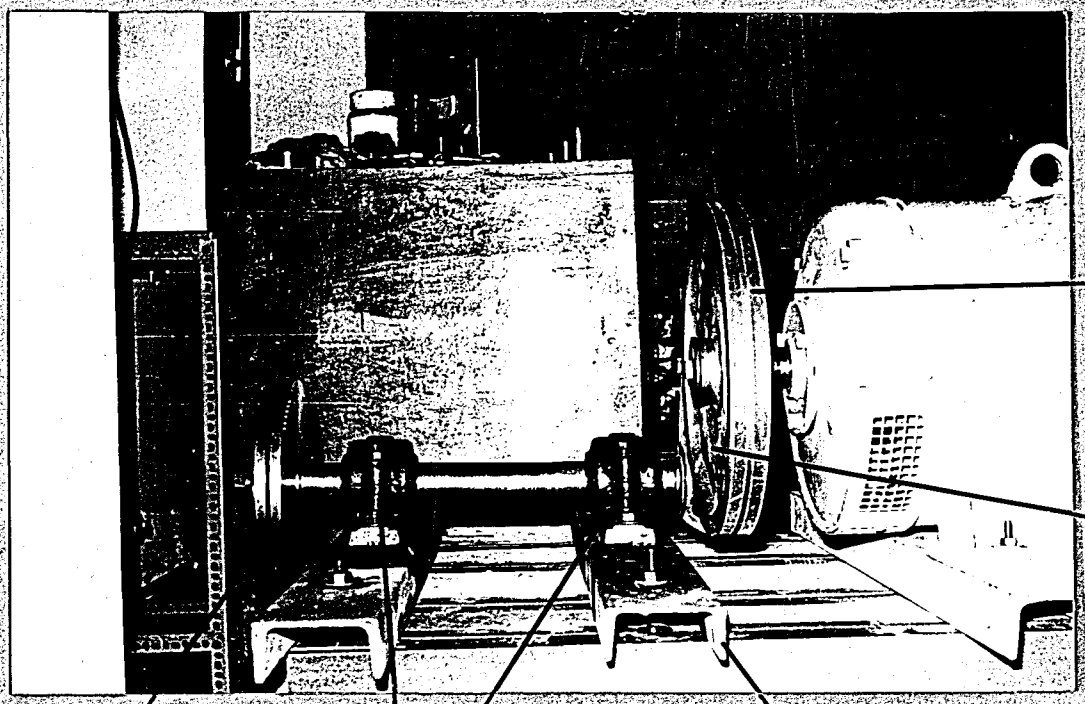


Fig. (18) Right side view of D-B system

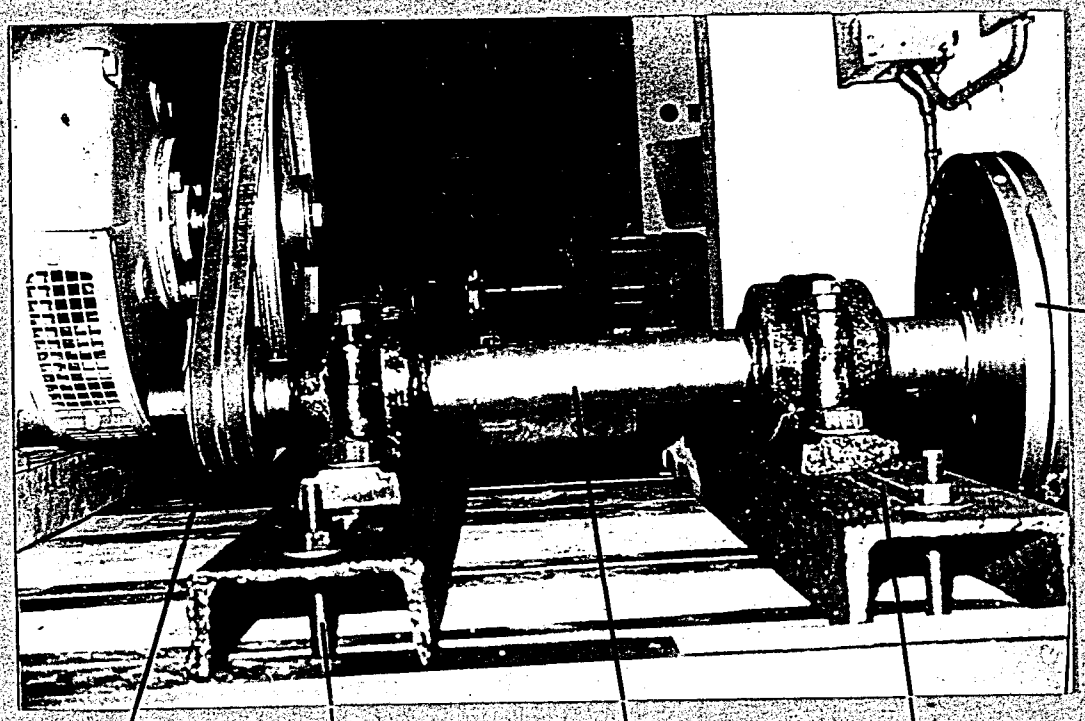


Fig. (19) Left side view of D-B system

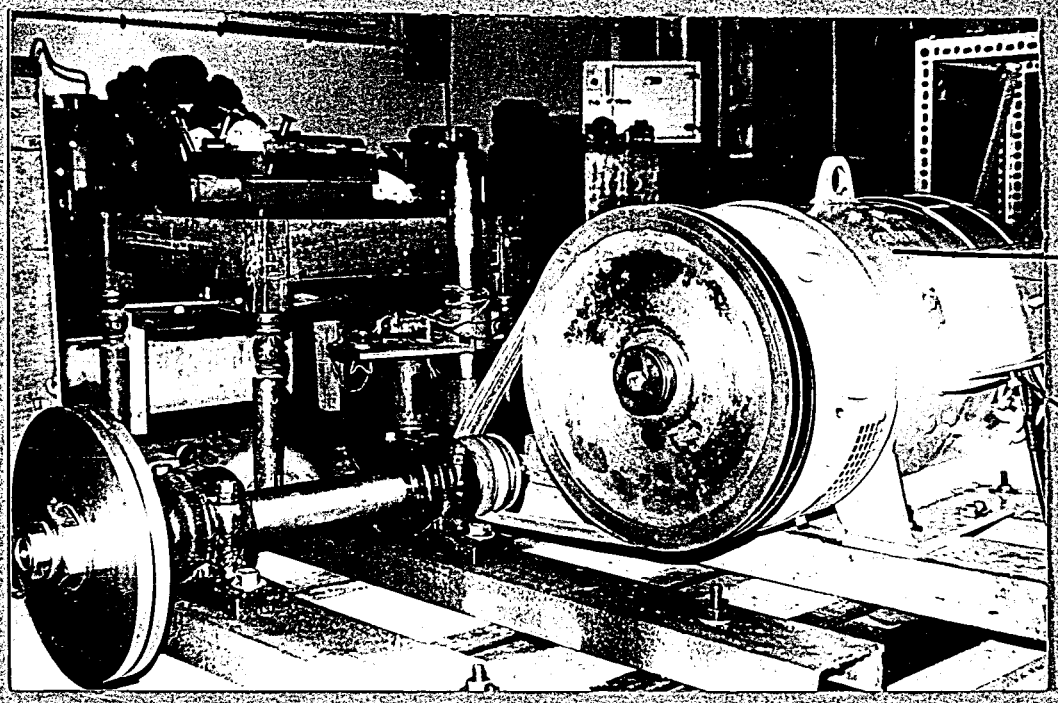


Fig. (20) Oblique front view showing details

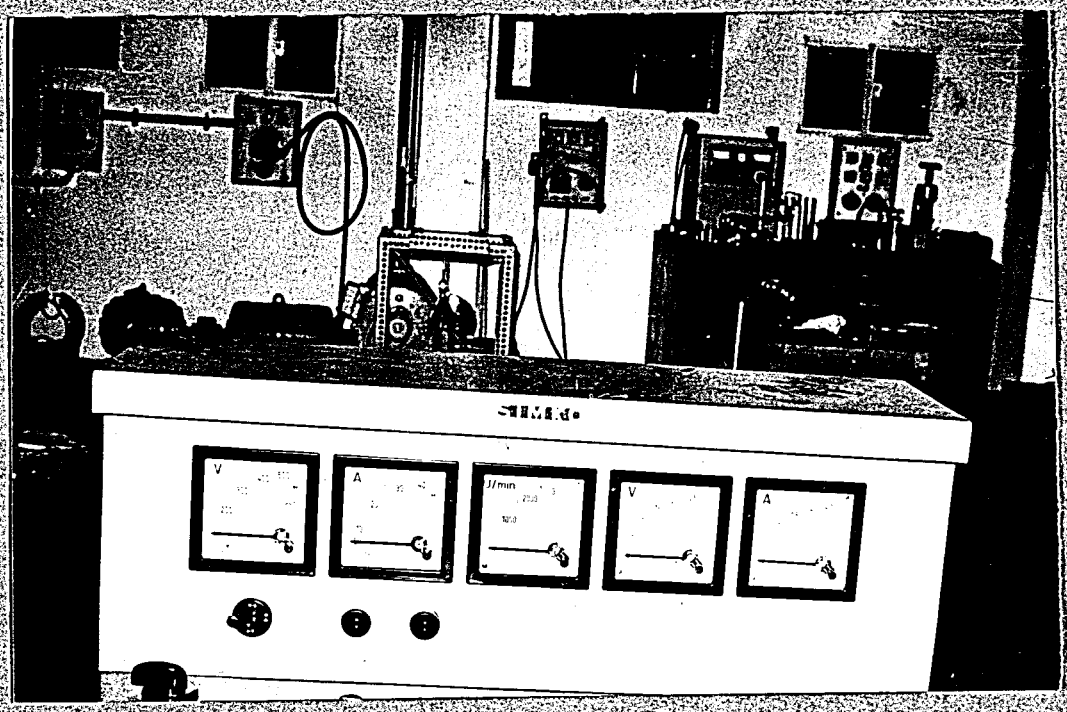


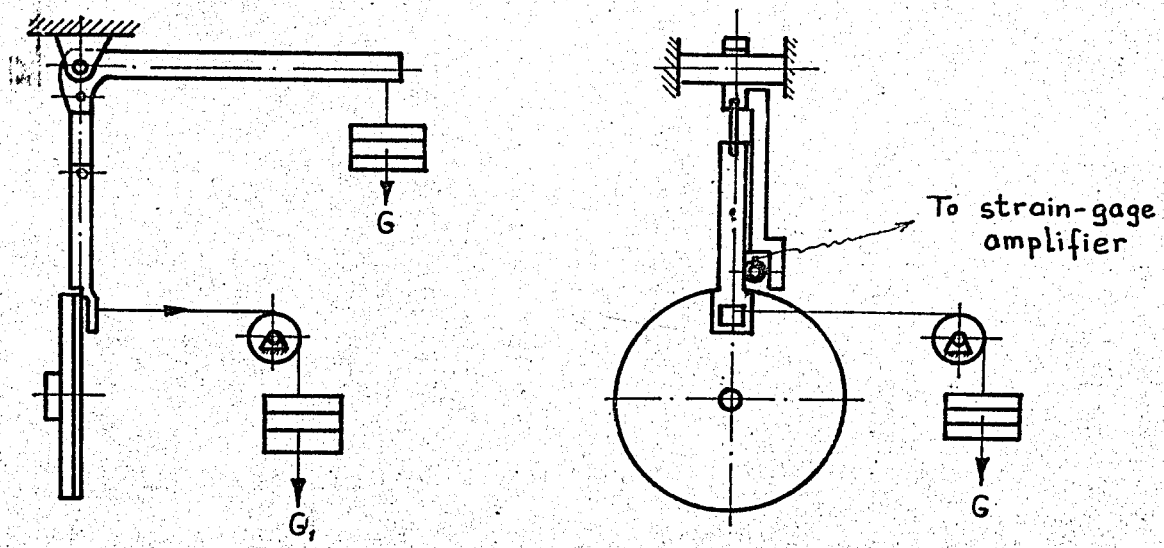
Fig. (21) Control panel of D-B system

tachometer of the system. The rpm of the specimen holder can be obtained by multiplying this value by four, or by means of a stroboskobe.

2 - Force Measurement

It is possible to calculate the normal force pressing the test specimen on the friction disk from the dimensions of the loading system. The loading system has been calibrated in order to take into consideration the actual lever arm dimensions and the friction losses in the bearings. To achieve this, a thin wire is connected through the backside of the specimen and is passed over a pulley in the horizontal direction, Fig. (22). This pulley is supported by two single-row roller bearings - in order to reduce the frictional effects to a minimum -, and the pulley diameter has been chosen as large in proportion to the pulley shaft diameter (1/20). The lever arm is loaded with known weights G and is balanced each time with weights G_1 hanged at the end of the wire passing over the pulley. From the weight ratio G_1/G, the ratio of the lever arms can be found. By this ratio, it will be possible to find the required weight G which will produce the normal force P on the disk ($P \simeq G$).

The active and the dummy strain-gages applied to the dynamometer



(a) Calibration of lever arm

(b) Calibration of the dynamometer

fig 22 Mechanism of dynamometer calibration

to be used in the measurement of the friction force are coupled into a half-bridge resistance circuit, and are connected to a strain-gage amplifier to be used in dynamic force measurements (HOTTINGER, Type KWS 11/5). The change of the friction force during the experiment has been followed on the amplifier, and is recorded simultaneously on a recorder (HELLIGE, Type He 86). The system has been calibrated with the dynamometer on the working position in order to be able to evaluate the accuracy of the instruments. The reasons for this calibration are twofold :

(1) The points of application of the friction forces and the axis of the dynamometer are not in line.

(2) To remove the disturbing effects arising from the bending of the (elastic) lever arms.

For this calibration, a horizontal wire connected to the backside center of the specimen holder is drawn in the opposite direction of the friction forces over a pulley, as shown in Fig. (22a). This system is loaded by hanging known weights at the end of the wire. For each position, the deviations on the instruments will be read. With the help of these readings the calibration curve shown in Fig. (23) has been drawn. By means of this curve, one can directly calculate the friction force from the values recorded on the

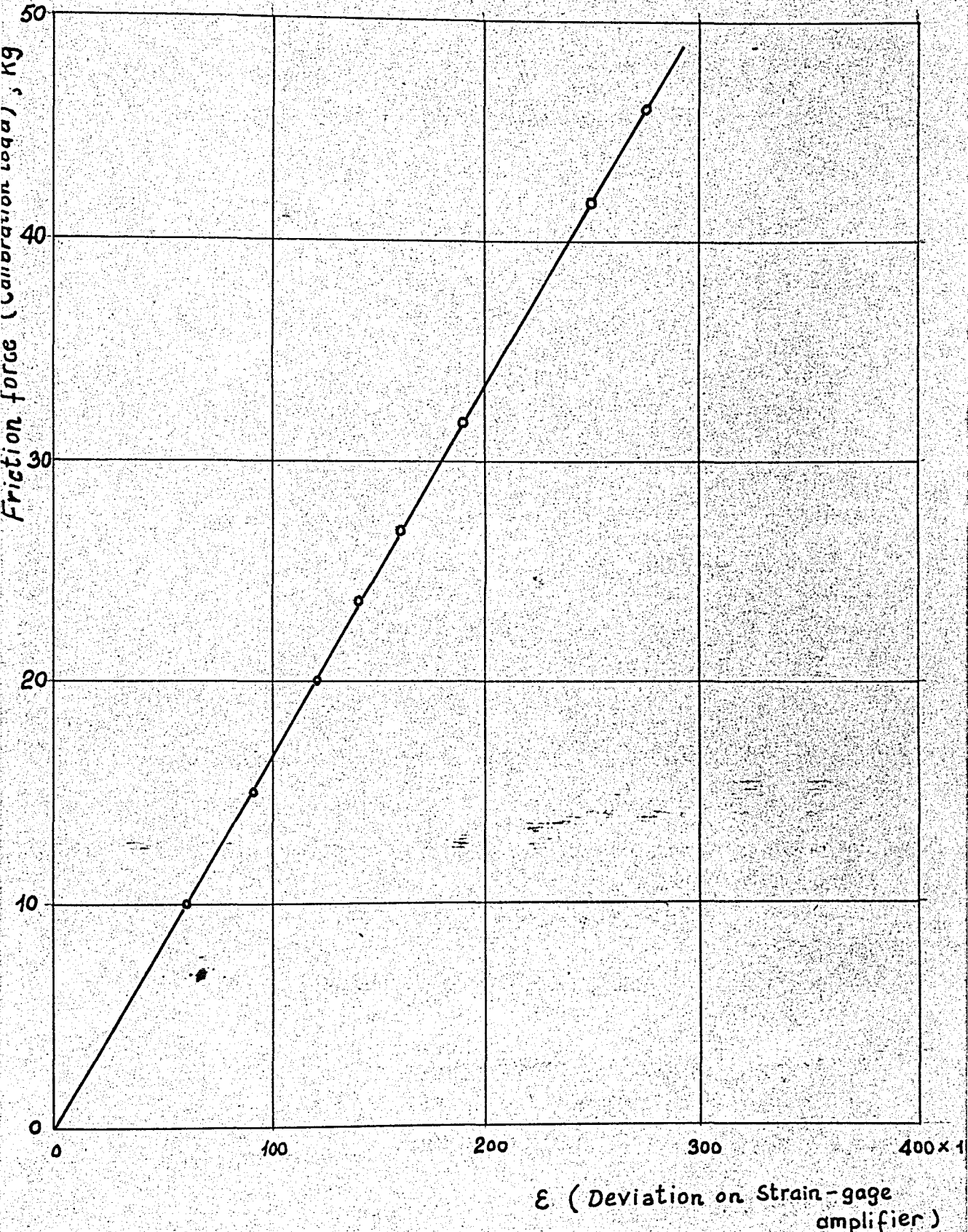


fig. 23- Dynamometer calibration curve

strain-gage amplifier. Since this curve is seen to be a straight line, one can conclude that the deformations of the system are within the elastic range, and after the loading and unloading of the system has been repeated for several times, it has been observed that the same curve is obtained for each time, leading to the conclusion that there is no permanent deformation in the system.

For controlling the sensitivity and accuracy of the measuring instruments, the calibration operation has been repeated with a strain-gage indicator for static measurements (BALDWIN-LIMA, Strain Indicator SR-4). The values obtained with this calibration are the same as in the first case.

3 - Temperature Measurement

Although the factors discussed in paragraph (III. C. 1) make the attempt of measurement of the rubbing interface temperatures very difficult, many methods have been developed for making measurements under similar conditions (28), (29).

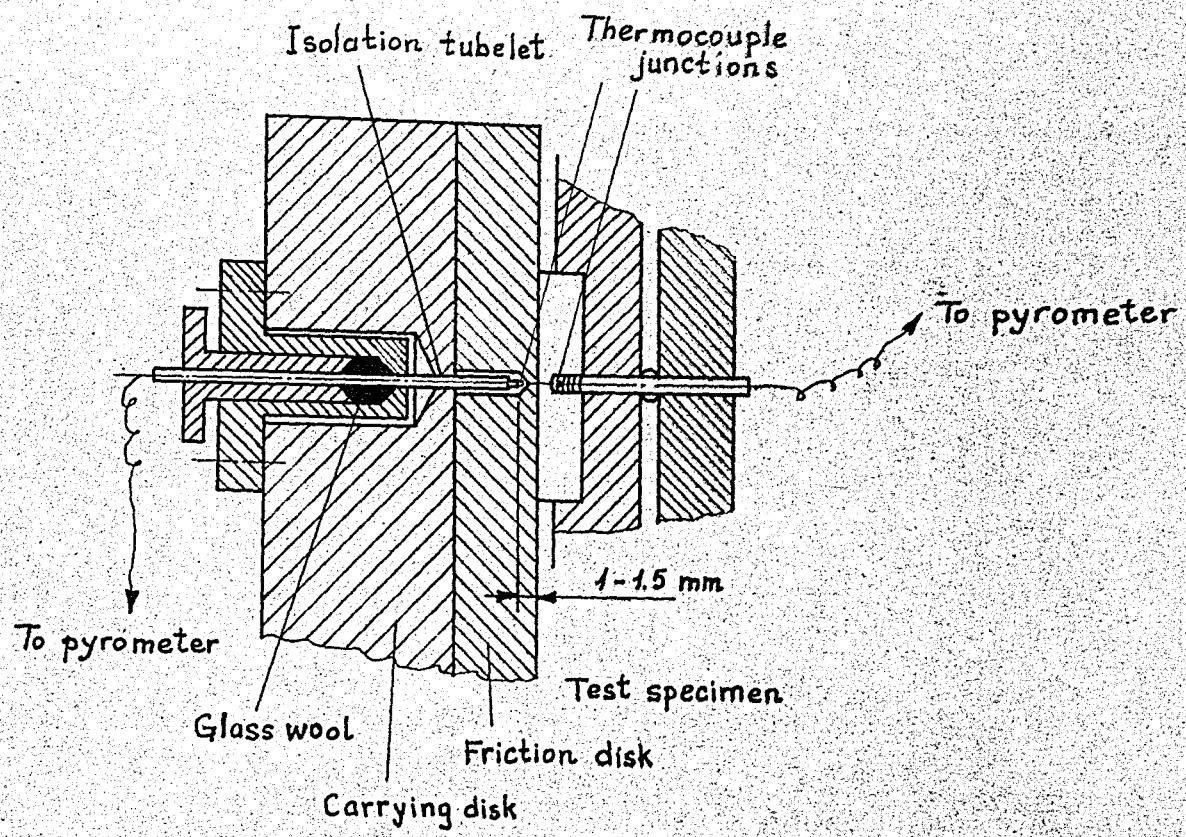
We have employed the " Bedded Thermocouple " technique in our experiments. The thermocouple is pressed into a minute hole which penetrates almost to the rubbing surface of the friction disk. This will indicate the true temperature at its point of location provided one discounts any disturbance to the temperature

gradients caused by the thermocouple hole.

As the gradients near the rubbing surface are extremely steep, the fact that the thermocouple stops even a few hundreths of a millimeter short of the surface can cause a low reading. This method of measurement involves a good deal of fitting work since slip rings are required. In Fig. (24), the method of placing the thermocouples into the disk is shown. A small isolation tube with two separate channels is passed through the hole opened in the disk and the thermocouple junction is brought to a 1 to 1.5 mm distance to the friction surface. The parts (1) and (2) which are bolted into the disk, fix the isolation tube into the disk tightly in order to prevent any motion of the tube during the experiment.

The other ends of the thermocouple wires are passed through a small cylindrical hole in the shaft carrying the disk and are connected to a special coin-brush slip ring system with a very low resistance (100 mohm). The ends of wires taken from the slip rings are connected to a pyrometer of sensitivity of $\pm 5^{o}C$. The specimen temperature is measured similarly with a thermocouple passing through it connected to another pyrometer.

Since the disk dimensions are large in comparison to the specimen dimensions, three thermocouples have been placed on the disk with equal arc distance from each other connected in parallel. Therefore,



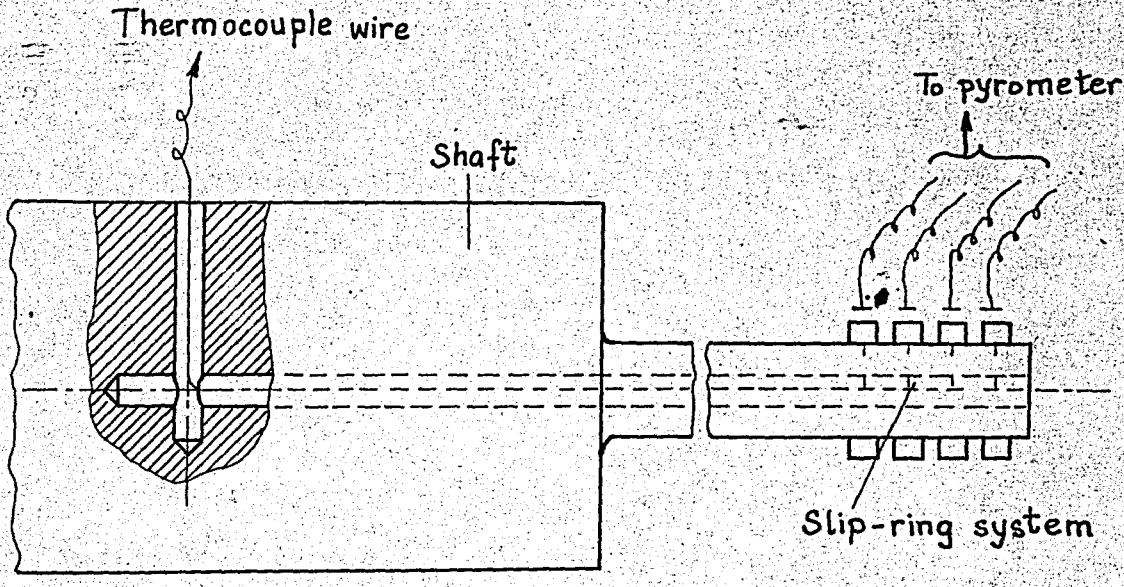


fig. 24 – Temperature measurement in the F & W test system

the largest value of the different temperatures will be measured.

The thermocouple pair has been selected as copper-constantan of 0.05 mm Ø diameter.

The temperatures recorded from the disk and specimen surfaces are compared.

4 - Wear Measurement

The quantity of worn material has been deterimend by weighing the test specimen before and after the experiment with a balance of ± 0.1 mg sensitivity (METTLER, Type B5). Only the wear rate of the test specimen has been considered. The quantity of wear of the disk has not been determined since it is negligibly small as compared to that of the specimen, and because of the difficulty of weighing the 14 kg disk within mg accuracy.

5 - Dimensions of Measured Quantities

a - Wear :

The quantity of wear can be determined by measuring the difference in weight of the test specimen in mg, before and after the loading stages of the experiment. Since the changes in dimensions of the

specimen due to wear are more important than the loss of weight, we find the volumetric wear rate (mm^3) by dividing the loss in weight by the density of the friction material. Thus,

$$V = G/\gamma \quad (mm^3)$$

In this way, the difference in the changes of dimensions of two materials, giving the same loss in weight, but whose densities differ from each other, can be determined accurately.

The quantity of wear is a function of the oppenating conditions and time. Therefore, the volumetric wear rate should be based on certain fundamentals.

The quantity of wear can be expressed as the decrease of weight of the material per unit surface per hour (g/cm^2-hr). But, this unit does not characterize wear perfectly, since it does not indicate the dimensional change, as well as the surface pressure, the slip velocity, the friction coefficient, and therefore the friction force (and the heat dissipated). These variables, i.e., the operating conditions, are included in the Volumetric Wear Work, as follows:

$$A_v = \frac{(\mu P v/75) \cdot z}{G/\gamma} \qquad (PS\text{-}hr/cm^3) \qquad (4.1)$$

where A_v = Volumetric wear work, $(PS\text{-}hr/cm^3)$

P = normal force, kg

v = slip velocity, m/sec

z = time, sec

G = weight, g

γ = density, g/cm^3

Some wear experiments have shown that, although the pressure and velocity have been increased, the volumetric wear work has diminished - mainly due to the rise in temperature - leading to the conclusion that this type of wear expression is also not sufficient. Alternately, the decrease of volume per PS-hr can be given as the reciprocal of the above quantity. Thus,

$$V = 1/A_v = \frac{G/\gamma}{(\mu P v/75)\cdot z} \qquad (cm^3/PS\text{-}hr) \qquad (4.2)$$

Since the operating conditions and the dimensions are taken into consideration, it is more practical and meaningful to use this formula in expressing wear.

By considering the friction surface A_f, the above formula takes the form :

$$V' = \frac{G/\gamma}{(\mu P v/75)\cdot z\cdot A_f} \qquad (cm/PS\text{-}hr) \qquad (4.3)$$

This expression shows the decrease in thickness per unit work.

By putting the pressure p instead of the normal force P, one obtains

$$q = \frac{G/\gamma}{(\mu p v/75)\cdot z\cdot A_f} \qquad (cm^3/PS\text{-}hr) \qquad (4.4)$$

This expression is identical to Eq. (4.2), but it contains the loading variable of pressure, p. This quantity q is called the specific wear rate.

In the American Federal Standards (30), the decrease in thickness (inches) of the specimen before and after the experiment are taken

as a measure of wear. This definition and dimension of wear is applicable only for experiments carried out under completely equal conditions, and the results may be compared only for identical experiments. The Eq. (4.3) may be taken as equivalent to this definition. But, since the reduction in thickness is related to the frictional work, it has a broader meaning and can be used to compare the results of different wear experiments.

b - Friction Coefficient :

The definition of μ is given as

μ = Friction Force/Normal Force

According to this definition, μ can be found by the applied normal force. It is a dimensionless quantity. The " Percentage Deviation of Friction Coefficient " (PDFC) is calculated by dividing the difference between the measured maximum and minimum values by the average value. Thus,

$$PDFC = (\mu_{max} - \mu_{min}) / \mu_{ave} \times 100 \quad \text{per cent} \qquad (4.5)$$

C. Experimentation

1 - Friction and Wear Test

a - Test Conditions

Actual tests for friction and wear performance should be started when preliminary preparations and lining " wear-in" have been completed as given in paragraph (b).

Conduct of test - All test stages should proced without interruption.

Disk speeds

All disk speeds are based on a 290 mm mean diameter friction disk with load applied to the specimen.

In order to determine the operating performance characteristics of the linings, the tests should be carried out under the stepwise increased loading. The test stages where the values of the friction coefficient and wear rate exceed the safety limits, are the limits of application of the lining type under consideration.

The most practical method for determining the theoretical limits of application of a certain friction material is to increase the values of the main loading or operating conditions or variables such as pressure and velocity by applying avarying set of combinations of the conditions (1).

Considering the maximum pressures found in automotive practice, we build three pressure stages as follows :

p (kg/cm^2) : 2 3 4 6 9 12

The velocity stages are :

v (m/sec) : 3.57 6.23 10.50

By combining these pressures and velocities in a proper way and increasing the value of their product parallelly, the test stages have been selected as shown below : (2) :

P [kg/cm^2]:	2	3	3	4	4	6	6	9	9	12	12
v [m/sec]:	3.57	3.57	6.23	6.23	10.50	6.23	10.50	6.23	10.50	6.23	10.50

b - Test Specimen

The test specimen shall be taken from the clutch friction facing from any place on the disk. The specimen shall be 3 x 3 cm square, flat on the bottom and top surfaces, with a thickness of 0,60 cm.

There is a problem in the selection of the specimen size : in order to secure uniform pressure, the specimen should have small dimensions, whereas because of the inhomogeneity of the friction material the specimen dimensions should be large.

In the US Federal Standards and SAE J 661a, 1 in. square, in the German Traffic Code 4 x 4 cm square specimens are to be used for the testing of brake linings. We have considered these values in the selection of our specimen sizes. In Fig. (12), the shape of the specimen is shown.

The working surface of the specimen should not be handled and should be kept free from foreign material.

C. Experimental Procedure

(1) Preparation of the Friction Disk Surface :

(a) New or Resurfaced Disk - After grinding the disk surface remove all grinder marks by polishing with abrasive paper of cloth. Final polishing shall be done with 320 grit (No. 0). Remove dust from disk with clean dry air and/or with clean cheesecloth or equivalent. Complete the surface

preparation by running a reference specimen continuously at $p = 2 kg/cm^2$ pressure and $v = 6.2$ m/sec velocity, and not over $\sim 90^oC$ until coefficient of friction has stabilized.

(b) Prior to Each Test : - Polish the disk surface with abrasive paper or cloth. Final polishing shall be with 320 grit. Remove dust from the disk with clean dry air and/or with clean cheesecloth or equivalent.

(2) Conditioning of the Test Specimen :

The specimen is run-in at $p = 2 kg/cm^2$ and $v = 3.5$ m/sec, and a maximum disk temperature of $\sim 90^oC$, for about 20 min., to obtain at least $\sim$ 95 per cent nominal contact.

(3) Initial Weight Measurement :

Weigh the specimen (in g) to third decimal place and record the value.

(4) Test Runs :

Initial and final temperature of the disk and specimen are to be measured and recorded for each test stage.

PROCEDURE NO.	P [kg/cm²]	V [m/sec]	t [min]
4.1	2	3.5	20
4.2	3	3.5	15
4.3	3	6.2	15
4.4	4	6.2	15
4.5	4	10.5	15
4.6	6	6.2	15
4.7	6	10.5	15
4.8	9	6.2	15
4.9	9	10.5	15
4.10	12	6.2	15
4.11	12	10.5	15

(5) Calculations :

(5.1) Friction Force : The values of the friction force F are found by reading the deviations on the strain-gage amplifier or the recorded readings, and by finding the corresponding friction force equivalents from the dynamometer calibration relationship.

(5.2) The average Friction Force per Test Stage : Find F as the arithmetic average of the recorded values.

(5.3) The max. min. and Ave. Values of the Friction Coefficient: μ_{max}, μ_{min}, and μ_{ave} are to be calculated by dividing F_{max}, F_{min}, and F_{ave} by P, the normal force on the specimen ($P = p \cdot A_f$), respectively.

(5.4) Average Friction Coefficient over the Test : the average value of u over the test is to be calculated by taking the average value of μ_{ave} found for each test stage.

(5.5) Percentage Deviation of The Friction Coefficient : PDFC is to be calculated by dividing the difference between μ_{max} and μ_{min} found over all test stages by μ_{ave}. The formula is

$$PDFC = (\mu_{max} - \mu_{min})/\mu_{ave} \times 100 \text{ per cent}$$

(5.6) Friction Coefficient of the Lining : This can be found by taking the total averages of μ_{ave} over ten test runs for the specific lining specimen.

(5.7) Wear Rate per Hour : For each test stage the weight loss W_s found by initial and final weighing, is divided by the test time, to find the Wear Rate per hour (g/hr).

(5.8) Frictional Horsepower : Frictional Horsepower N_f is calculated by the following formula

$$N_f = F_{ave} \cdot v/75 \text{ in PS}$$

where F_{ave} in kg, and v in m/sec.

(5.9) Specific Wear Rate : SWR q is calculated by the following formula

$$q = W/(N_f \cdot \gamma) \ (cm^3/PS\text{-}hr)$$

(5.10) The Average Specific Wear Rate : The ASWR q_{ave} is calculated by taking the average value of q found for each test stage.

d - Test Results

The test results are given in Appendix A and B, and their discussion is given in paragraph (V.A) under the title " Conclusions from Friction and Wear Characteristics Analysis "

2 - Disk-Bursting Test

a - Test Conditions

Since the tests are carried out at very high rotational speeds (n = 10 - 12,000 rpm), each component of the test apparatus should be checked for proper condition and location. The system should be dynamically- and statically balanced before it is operated. In addition to this, the system should be enclosed by means of a safety-guard in order to

protect the men and the equipment in the laboratory from the effects of the flying burst particles.

b - Test Specimen

The test specimens are disk-type standard friction clutch facings with an inside radius of R_1, an outside radius of R_2, and a thickness of t. These dimensions are given in the followin standard order :

$$R_2 \times R_1 \times t$$

c - Experimental Procedure

(1) The facing specimen is fixed to the disk-holder in a centered and tight manner.

(2) The Ward-Leonard Motor Unit is started and its speed is increased from zero to higher rpm, while the power input of the motor is varied from 0.017 Kw to 5.4 Kw.

(3) The values of the motor rpm and disk-holder rpm are followed. The rpm at bursting, N_b, is recorded.

(4) The value of the mean tangential stress at bursting, $(\bar{\sigma}_\theta)_b$, is calculated by

$$(\bar{\sigma}_\theta)_b \cong K \rho \omega_b^2 R_2^2 \qquad (kg/cm^2)$$

where K depends on the ratio $X = R_1/R_2$, and varies between 0.5 and 0.8 .

d - Test Results

The results are given in Appendix A and B

V. CONCLUSIONS AND RECOMMENDATIONS

A- Friction and Wear Performance

The test results have been evaluated by means of the " Stepwise Regression " technique (31). Computer results given in Appendix C indicate a rather strong correlation between the experimental variables. Values of the correlation coefficient (multiple) among the independent variables of p, v, and pv-product representing appropriate combinations of the levels (or values) of p and v, and the dependent variables of friction coefficient, μ, and wear rate, q, have been found to be more than 0, 85. This gives us a quantitative indication of the strength of the linear association between the variables, in this range of experimental conditions and for these specific types of friction clutch materials. Therefore, it makes sense to apply linear regression techniques to the variation of μ and q to find the straight lines that best fit the data. The straight line presentations of μ and q versus pv-product representing the frictional energy dissipation per unit area, for both molded and woven types of lining materials, are given in Appendix B.

The following conclusions may be drawn from the graphical results :

(1) For both types of linings, μ drops with increasing pv.

(2) For both types of linings, q increases with increasing pv-values for values of pv above ~ 40 kg-m/cm^2-sec. For smaller values of pv,

shows a tendency to decrease.

The equations of the straight-lines can be used as an empirical relation in analytical work.

Our findings in the determination of variation patterns of μ with v are in agreement with the previous experimental evidence in the friction materials testing. In general, the effect of increasing sliding velocity is to decrease the frictional force acting on two sliding bodies, together with the friction coefficient[x].

The friction force and friction coefficient decrease with increasing pressure (21),(9). Thus, μ was expected to decrease with the product pv increasing, as it is the case in our experiments.

Our findings in the investigation of the effects of operating conditions on wear rates are also in accordance with the existing limited experimental evidence. It is recommended that for molded and woven types of dry friction clutch lining materials q_{ave} should vary between 0.1 and 0.3 $cm^3/PS\text{-}hr$ for moderate pv-values. In our case of more severe operating conditions, we have found the following variation limits for q_{ave}:

$$0.34 > q_{ave} > 0.06 \quad \text{for woven linings}$$

$$0.28 > q_{ave} > 0.07 \quad \text{for molded linings}$$

Wear not only has an important effect on the life of the lining and mating plate, but also strongly affects the friction coefficient. Rising rubbing

surface temperatures tend to reduce μ (called " fade ") while wear tends to r. restore μ. This applies particularly to the extreme outer surface where temperatures are often excessive (see paragraph III. C.). A moderate amount of wear is therefore desirable to clean the lining and mating surface (21). With regard to this fact, the linings have given satisfactory performance values in this range of operating conditions.

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x The relationship between μ and v can be determined to a certain degree by

(1) changing the composition of the lining materials

(2) changing the types of the additives used

(3) changing the concentration of the materials

- Bursting Behaviour

The mean dynamic burst strength, $(\bar{\sigma}_\theta)_b$, has been plotted versus the square of the outside disk radius, R_2^2. The variation is linear as expected. Comparing the dynamic bursting behaviour of molded and woven types of linings tested, one can draw the following conclusions :

(1) Waven linings tested have higher dynamic strength values than molded ones, provided the X and t values are held constant.

(2) Therefore, a woven lining may be operated at higher service rotational speeds than a molded lining of same outside disk radius R_2, ratio of inner to outside disk diameters, X, and thickness, t.

C - Surface Temperatures

Graphical plots of θ_{max} versus $(\mu p v^{1/2})$ indicate a linear relationship between the two variables for both types of linings in the applied range of p and v values.

The empirical relationship between θ_{max} and pv may be best approximated by the exponential curves as shown in Appendix B. The graphs indicate that θ_{max} increases exponentially with pv, in the range tested, for these specific lining types.

The maximum values of temperatures reached are somewhat smaller for woven type than for molded one, due to the inclusion of metallic particles in the lining mix.

The frictional surfaces of the test specimens have been inspected by means of a stereo-microscope and also with the naked eye. On the rubbed surfaces, dark regions and black spots have been observed. These are probably disintegrated resins which have charred under the effect of high instantaneous local values of the actual contact surface temperature.

It was also expected that the synthetic resins would be subject to dissociation, recombination, cracking and charring, under the effect of high temperature in the interface region of contact developed as a result of pressure and velocity. The thickness of the interface layer is expected to be of the order of one molecule or less (32). Because this layer is so thin, very high temperatures can be reached in a few millimicroseconds. Thermoelectric methods employed in

the measurement of these transient local temperatures indicate values as high as 1000°C, and their duration has been observed to be of the order of 10^{-4} seconds (9).

Since our laboratory facilities were limited, we have been unable to make readings of the local hot spot temperatures.

D - Recommendations

(1) The Friction and Wear Test has been performed in a satisfactory manner. The effects of the operating conditions on the friction and wear characteristics of various lining types have been investigated in the pv-rating range representing high pressure and velocity, with the help of the test system developed at the ITU.

This test method may be used to establish a uniform laboratory procedure for securing and reporting the friction and wear characteristics of clutch linings. The performance data obtained can be used for in-plant quality control by clutch lining manufacturers and for the quality assessment of incoming shipments by the purchasers of clutch linings, such as by automative -(and also by machine - tool)- industries.

(2) The bursting behaviour of disk type clutch linings has been studied with the experimental system and procedure developed by the author at the ITU. In order to be able to see the effects of the higher rotational speeds and centrifugal stresses on the clutch linings more clearly - especially for

smaller size linings - the experimental system should reach up to 15,000 - 20,000 rpm in a relatively shorter time. Therefore, the capacity of the driving motor unit should be increased. Both the power output range and the speed range of the Ward - Leonard Group should be increased in order to be used efficiently for the testing of all possible sizes of disk friction facings.

(3) The microscopic and macroscopic analyses of the friction and wear specimens have have been carried out in order to investigate the effects of hot spot temperatures on the friction material surfaces. The effects have been clearly inspected. The microstructural details existing at the rubbing interface of the specimens were not investigated. This would be the subject of a future investigation aimed at offering guidance for future friction materials improvement.

(4) In order to arrive at somewhat more reliable conclusions about the performance of the clutch linings, the tests could be repeated more times. But the cost of research would go too high. An economic analysis with respect to the number of repetitions of the specific tests for each lining type should be prepared.

APPENDIX A

Experimental Data

Disk : GG 22 SAMPLE DATA SHEET Exp. No.: 1

P kg/cm²	V m/sec	time min	θ_{max} °C	Strain gage amplifier reading min	ave	max	F kg min	ave	max	μ min	ave	max	Wear g Total Weight	Loss	Loss/hr	PS	q g/PS-hr	cm³/PS-hr
													10.277					
2	3.57	0	20	37	39	41												
		5	30	39	40	41												
		10	40	37	38	39												
		15	45	40	42	44												
		20	50	39	41	43												
			50		40		6.18	6.63	7.32		0.37		10.251	0.026	0.073	0.48	0.162	0.085
3	3.57	0	50	57	59	61												
		5	56	53	55	57												
		10	62	55	56	57												
		15	70	56	58	60												
			70		57		8.83	9.48	10.18		0.35		10.224	0.027	0.108	0.77	0.141	0.073
3	6.23	0	55	53	56	59												
		5	63	56	58	60												
		10	82	51	50	53												
		15	90	48	50	52												
			90		54		8.00	8.82	10.00		0.33		10.196	0.034	0.136	1.06	0.128	0.067
4	6.23	0	70	66	67	68												
		5	98	63	66	69												
		10	110	68	69	70												
		15	120	73	74	75												
			120		69		10.50	11.57	12.50		0.32		10.140	0.050	0.200	1.57	0.128	0.067
4	10.50	0	64	60	64	66												
		5	100	64	66	68												
		10	155	61	63	65												
		15	175	66	67	68												
			175		65		10.18	10.81	11.35		0.30		10.055	0.089	0.338	1.93	0.175	0.091
6	6.23	0	80	86	88	90												

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Disk :

Exp No.:

P kg/cm²	V m/sec	time min	θ_{max} °C	Strain gage amplifier reading min	ave	max	F kg min	ave	max	μ min	ave	max	Wear g Total Weight	Loss	Loss/hr	PS	q g/PS-hr	q cm³/PS-hr
		5	125	93	99	105												
		10	164	95	98	101												
		15	190	92	95	98												
			190		95		14.43	15.78	17.50		0.28		9.963	0.092	0.370	2.24	0.165	0.086
6	10.50	0	90	85	90	95												
		5	160	78	81	84												
		10	185	98	103	108												
		15	205	95	98	101												
			205		93		13.00	15.53	18.00		0.29		9.841	0.122	0.490	2.43	0.202	0.105
9	6.23	0	92	128	130	132												
		5	165	152	155	158												
		10	195	135	140	145												
		15	220	150	154	158												
			220		145		21.40	24.20	26.40		0.30		9.729	0.112	0.447	2.04	0.219	0.114
9	10.50	0	105	120	123	125												
		5	174	138	140	142												
		10	215	133	134	135												
		15	245	125	127	129												
			245		131		20.00	21.80	23.60		0.27		9.550	0.179	0.716	3.65	0.196	0.102
12	6.23	0	100	148	151	154												
		5	182	172	176	180												
		10	218	178	180	182												
		15	260	165	168	171												
			260		169		24.60	28.10	30.30		0.26		9.305	0.245	0.980	3.14	0.313	0.163
12	10.50	0	100	143	146	149												
		5	190	133	135	137												
		10	275	134	136	138												
			275		139		22.20	23.20	24.80		0.21		8.928	0.377	2.260	4.52	0.500	0.260

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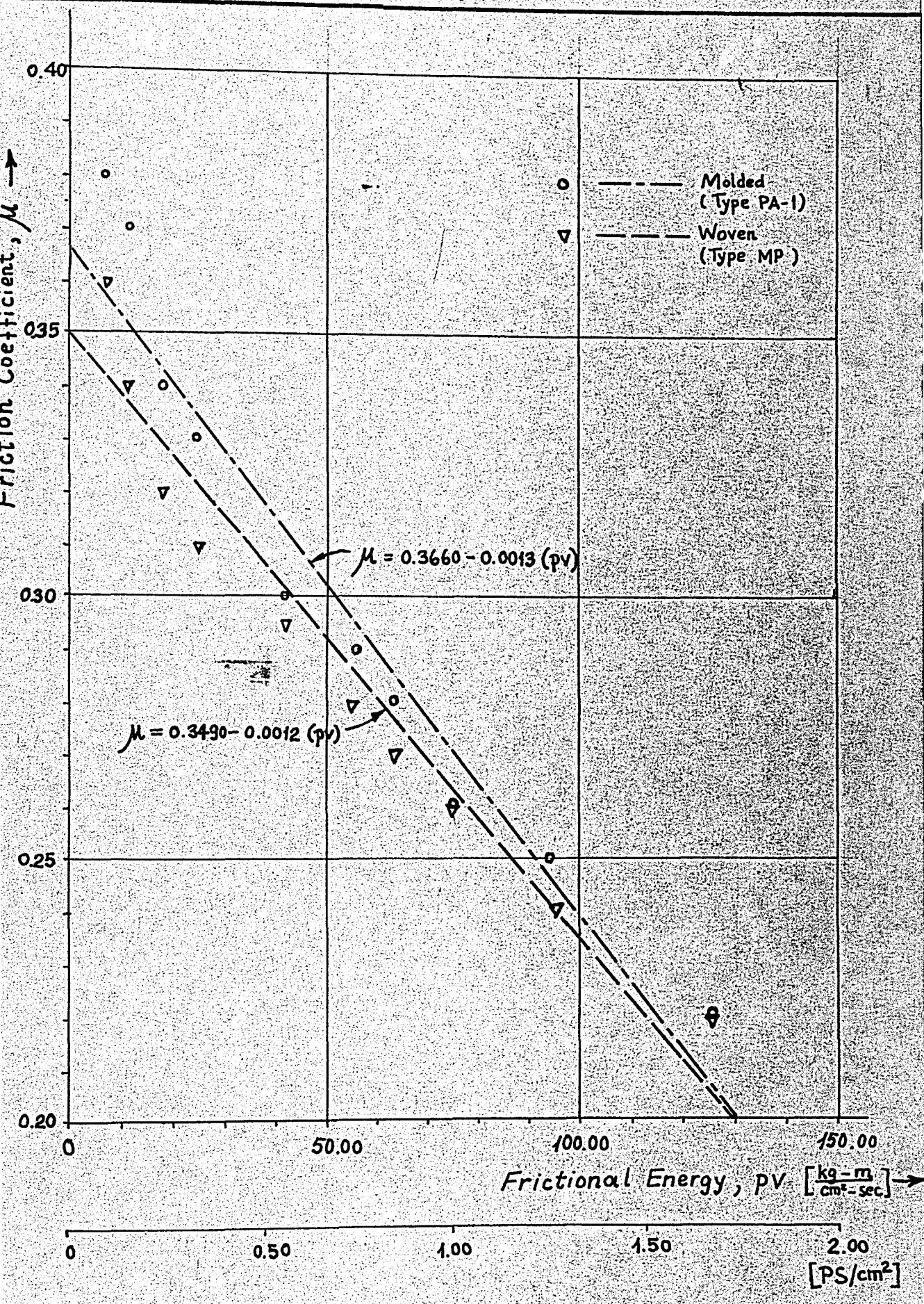
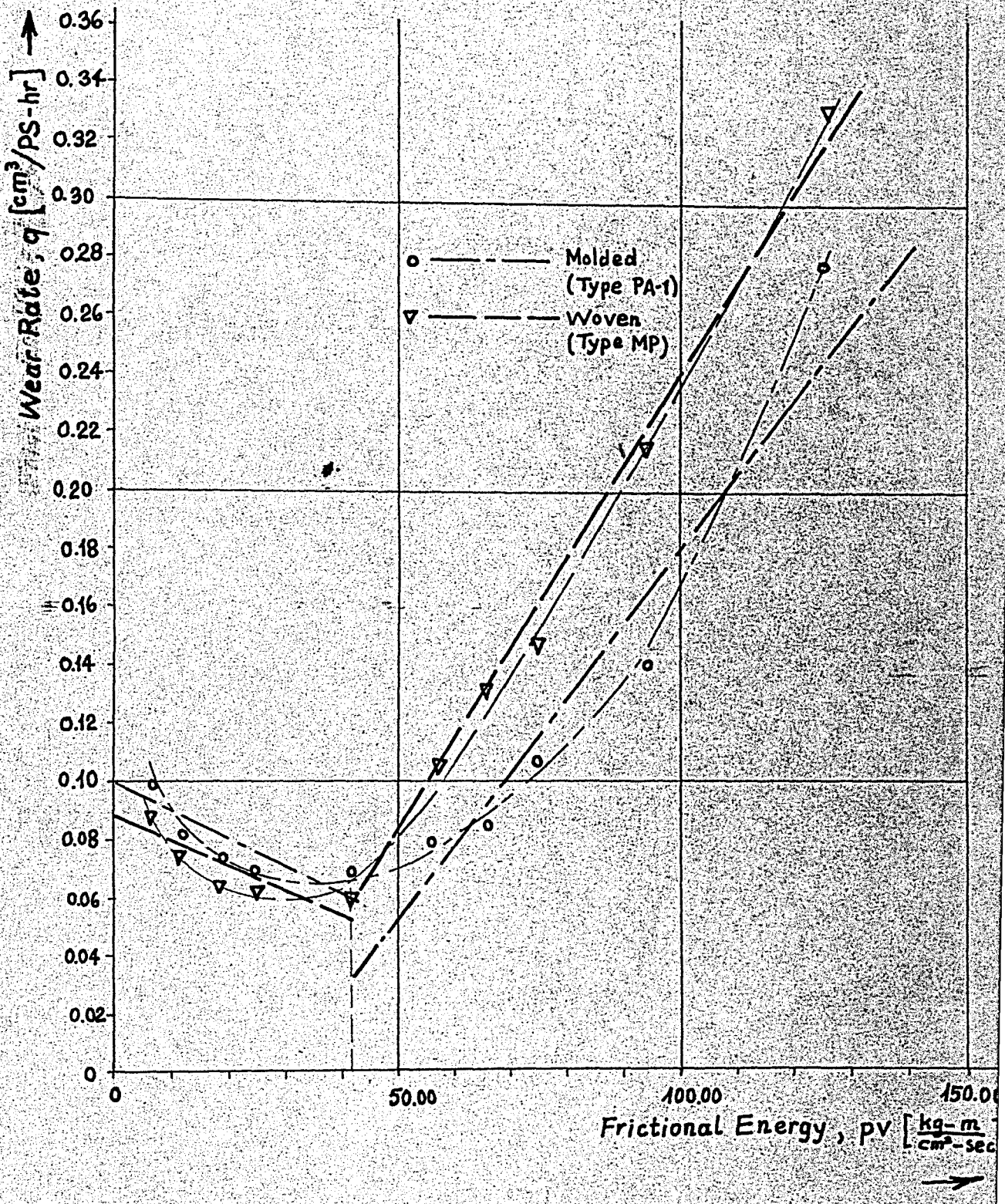
Specimen: PA-1 Molded Density: 1.92 g/cm³ Date:
Disk : GG 22 Exp No:

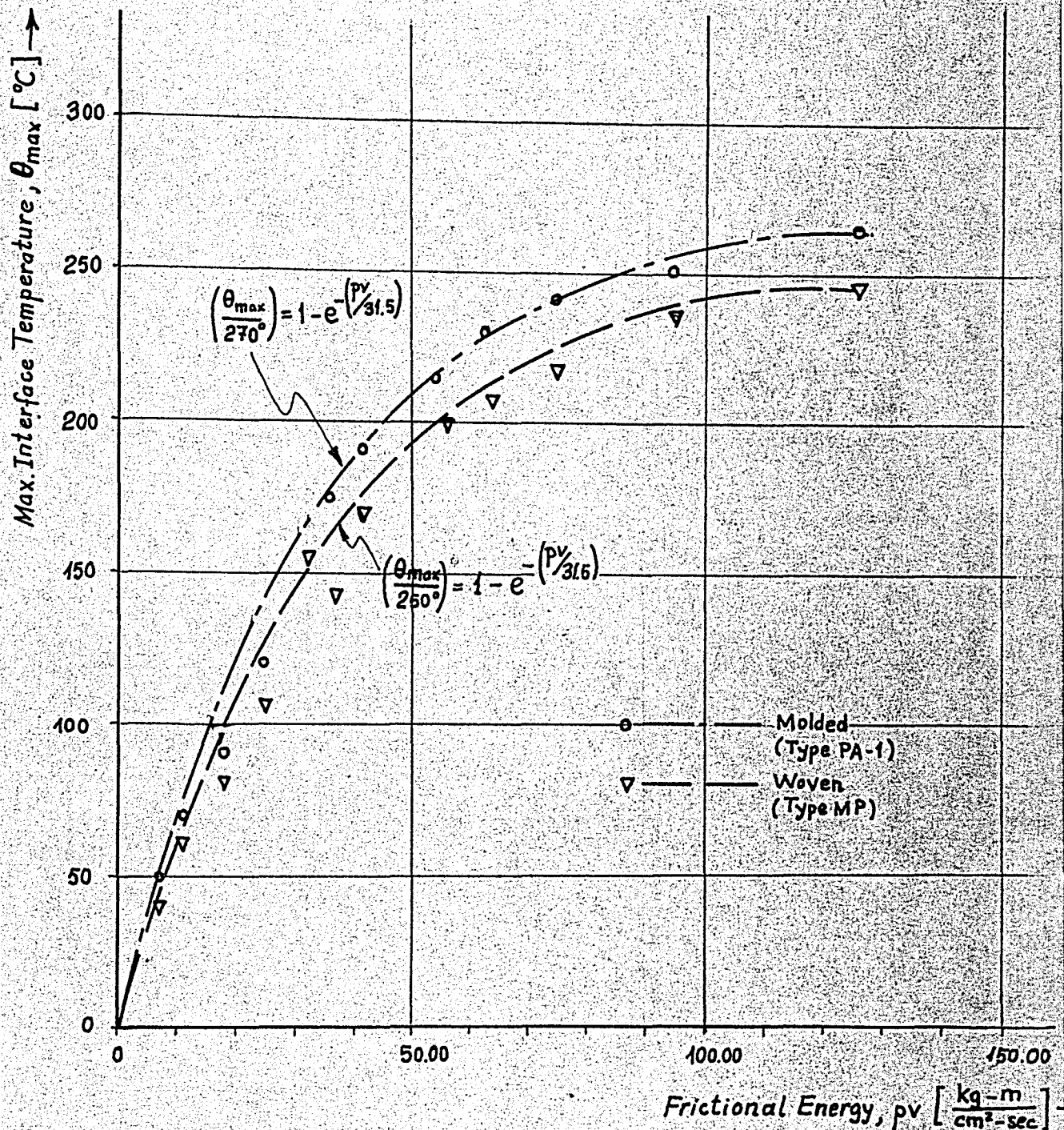
P [kg/cm²]	V [m/sec]	PV [kg-m / cm²-sec]	θ_{max} [°C]	μ min	μ ave	μ max	q [cm³/PS-hr]
2	3.57	7.14	50	0.36	0.38	0.40	0.100
3	3.57	10.71	70	0.35	0.37	0.39	0.080
3	6.23	18.69	90	0.32	0.34	0.37	0.074
4	6.23	24.92	120	0.31	0.33	0.36	0.070
4	10.50	4200	175	0.28	0.30	0.33	0.072
6	6.23	37.38	190	0.29	0.30	0.32	0.068
6	10.50	63.00	215	0.27	0.29	0.31	0.078
9	6.23	56.07	230	0.26	0.28	0.30	0.084
9	10.50	94.50	240	0.24	0.26	0.28	0.110
12	6.23	74.76	250	0.23	0.25	0.27	0.140
12	10.50	126.00	265	0.20	0.20	0.24	0.280
Specimen: MP-Woven Density: 2.00 g/cm³							
2	3.57		40	0.33	0.36	0.42	0.088
3	3.57		60	0.32	0.34	0.40	0.076
3	6.23		80	0.30	0.32	0.36	0.065
4	6.23		105	0.29	0.31	0.34	0.068
4	10.50		140	0.23	0.27	0.32	0.072
6	6.23		165	0.26	0.29	0.32	0.070
6	10.50		185	0.26	0.28	0.30	0.105
9	6.23		205	0.25	0.27	0.29	0.132
9	10.50		215	0.24	0.26	0.29	0.143
12	6.23		235	0.22	0.24	0.26	0.215
12	10.50		245	0.21	0.22	0.24	0.335

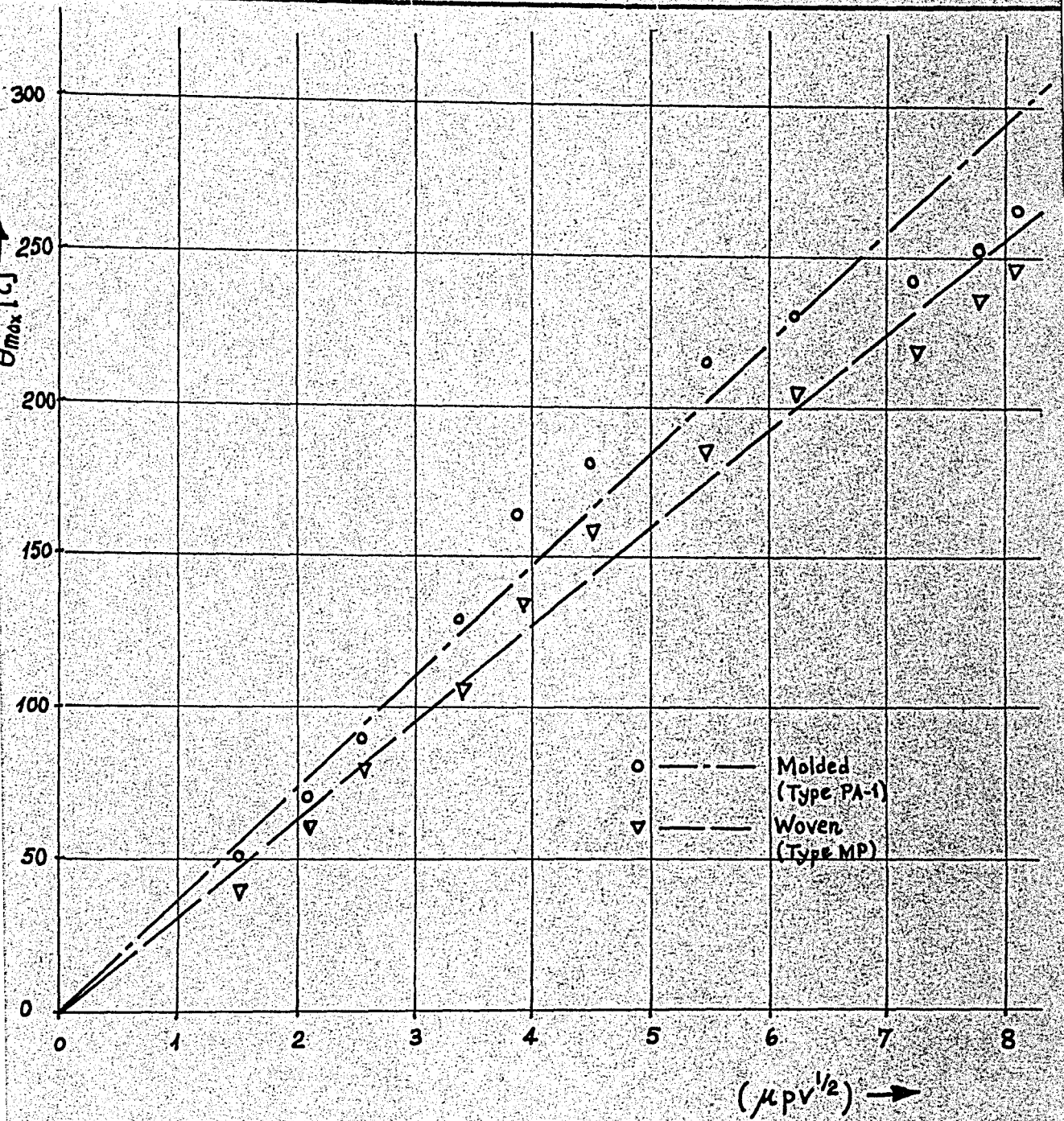
APPENDIX B

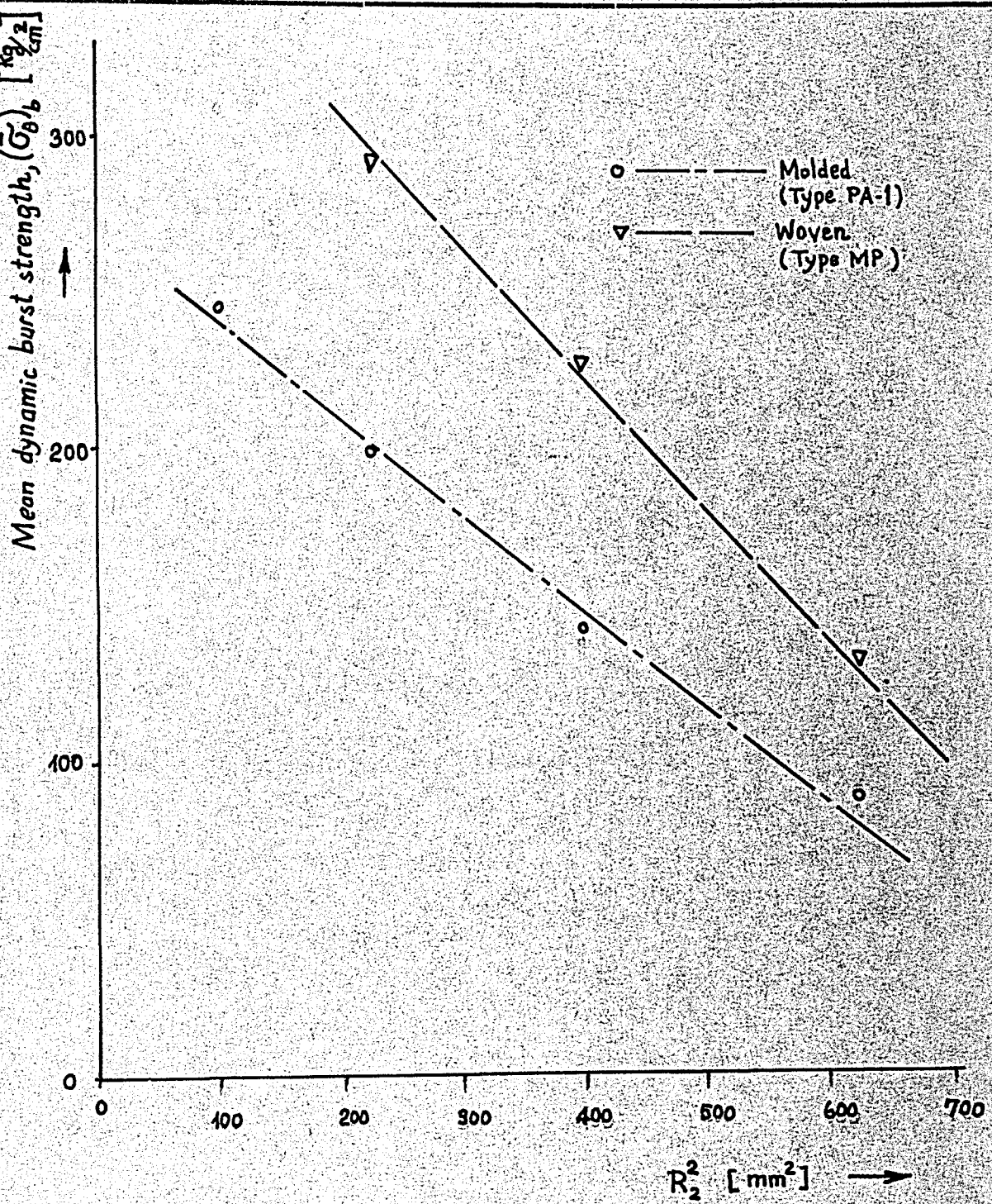
Graphical Presentation of Data


Friction Coefficient, μ
0.40
035
030
0.25
0.20
o —— Molded (Type PA-1)
∇ —— Woven (Type MP)
μ = 0.3660 - 0.0013 (pv)
μ = 0.3490 - 0.0012 (pv)
0
50.00
100.00
150.00
Frictional Energy, PV [kg-m / cm²-sec.]
0
0.50
1.00
1.50
2.00
[PS/cm²]


Wear Rate, q [cm³/PS-hr]
0.36
0.34
0.32
0.30
0.28
0.26
0.24
0.22
0.20
0.18
0.16
0.14
0.12
0.10
0.08
0.06
0.04
0.02
0
o —— Molded (Type PA-1)
▽ —— Woven (Type MP)
0
50.00
100.00
150.00
Frictional Energy, pv [kg-m / cm²-sec]


Max. Interface Temperature, θ_{max} [°C]
300
250
200
150
100
50
0
0
50.00
100.00
150.00
Frictional Energy, pv $\left[\frac{kg-m}{cm^2-sec}\right]$
$\left(\frac{\theta_{max}}{270°}\right) = 1 - e^{-\left(pv/31.5\right)}$
$\left(\frac{\theta_{max}}{250°}\right) = 1 - e^{-\left(pv/31.5\right)}$
Molded
(Type PA-1)
Woven
(Type MP)


300
250
200
150
100
50
0
σ_{max} [psi]
0
1
2
3
4
5
6
7
8
$(\mu pV^{1/2})$ →
Molded
(Type PA-1)
Woven
(Type MP)


Mean dynamic burst strength, $(\bar{G}_B)_b$ [kg/cm²]
300
200
100
0
0
100
200
300
400
500
600
700
R_2^2 [mm²]
Molded (Type PA-1)
Woven (Type MP)

APPENDIX C

Regression Analysis Results

AKLI REGRESYON

M NO. 10020

AYISI = 11

EN SAYISI = 4

ENIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

ENIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

ENLERIN TOPLAMI

X(1) =	70.0000
X(2) =	80.2900
X(3) =	555.1700
Y =	3.3200

RELER TOPLAMI

1) VS X(1) =	576.000000
1) VS X(2) =	555.170000
1) VS X(3) =	4736.690000
2) VS X(2) =	660.554300
2) VS X(3) =	4801.113100
3) VS X(3) =	41805.422000
1) VS Y =	19.450000
2) VS Y =	23.257500
3) VS Y =	150.017700
VS Y =	1.026800

ENLERIN ORTALAMA DEGERLERI

X(1) =	6.3636
X(2) =	7.2990
X(3) =	50.4700
Y =	.3018

KARELER TOPLAMI

1) VS X(1) = 130.545460

1) VS X(2) = 44.233640

1) VS X(3) = 1203.790000

2) VS X(2) = 74.510300

2) VS X(3) = 748.876900

3) VS X(3) = 13785.993000

1) VS Y = -1.677272

2) VS Y = -.975481

3) VS Y = -17.542700

VS Y = .024763

KORELASYON KATSAYILARI

1) VS X(2) = .448501

1) VS X(3) = .897327

2) VS X(3) = .738895

1) VS Y = -.932856

2) VS Y = -.718129

3) VS Y = -.949444

RININ STANDART HATASI = .049763

K NO. 1
ESYONA GIREN DEGISKEN 3
VIYESI = 82.3198
GERININ STANDART HATASI = .0164
YONLU KORELASYON KATSAYISI = .94944
T = .36604

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 3	-.00127	.00014

K NO. 2
ESYONA GIREN DEGISKEN 1
VIYESI = 4.1367
GERININ STANDART HATASI = .0141
YONLU KORELASYON KATSAYISI = .96697

```
=          .37723
     DEGISKEN          KATSAYI    KATSAYI STNDRT. HATASI

        X-  1          -.00571          .00281
        X-  3          -.00077          .00027


 NO.     3
SYONA GIREN DEGISKEN       2
IYESI =        26.2271
ERININ STANDART HAT ASI  =          .0069
ONLU KORELASYON KATSAYISI =          .99313
 =          .43194
     DEGISKEN          KATSAYI    KATSAYI STNDRT. HATASI

        X-  1          -.01307          .00199
        X-  2          -.00884          .00172
        X-  3           .00035          .00025


 NO.     4
SYONDAN CIKARTILAN DEGISKEN        3
IYESI =        -1.5887
ERININ STANDART HAT ASI  =          .0084
ONLU KORELASYON KATSAYISI =          .99130
 =          .41875
     DEGISKEN          KATSAYI    KATSAYI STNDRT. HATASI

        X-  1          -.01053          .00082
        X-  2          -.00684          .00109

NAN REGRESYON BASAMAKLARI SAYISI =      4


NAL ELEMANLAR
NO.       DEGER

          1.251805
          1.251804
           .053106


. VE HAKIKI DEGERLERIN KARSILASTIRILMASI

     OKUMA      HAKIKI          TAHMINI         FARK

    1.00          .38000          .37327          .00672
    2.00          .37000          .36274          .00725
    3.00          .34000          .34455         -.00455
    4.00          .33000          .33402         -.00402
    5.00          .30000          .30481         -.00481
    6.00          .30000          .31296         -.01296
    7.00          .29000          .28375          .00624
    8.00          .28000          .28136         -.00136
    9.00          .26000          .25216          .00783
   10.00          .25000          .24977          .00022
   11.00          .22000          .22056         -.00056
AKLI REGRESYON

M NO.      10030
```

YISI = 11

N SAYISI = 4

NIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

NIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

NLERIN TOPLAMI

X(1) =	70.0000
X(2) =	80.2900
X(3) =	555.1700
Y =	1.1560

ELER TOPLAMI

1) VS X(1) =	576.000000
1) VS X(2) =	555.170000
1) VS X(3) =	4736.690000
2) VS X(2) =	660.554300
2) VS X(3) =	4801.113100
3) VS X(3) =	41805.422000
1) VS Y =	8.892000
2) VS Y =	9.028880
3) VS Y =	76.029380
VS Y =	.159824

NLERIN ORTALAMA DEGERLERI

X(1) =	6.3636
X(2) =	7.2990
X(3) =	50.4700
Y =	.1050

ARELER TOPLAMI

1) VS X(1) = 130.545460

1) VS X(2) = 44.233640

1) VS X(3) = 1203.790000

2) VS X(2) = 74.510300

2) VS X(3) = 748.876900

3) VS X(3) = 13785.993000

1) VS Y = 1.535636

2) VS Y = .591131

3) VS Y = 17.686060

VS Y = .038338

KORELASYON KATSAYILARI

1) VS X(2) = .448501

1) VS X(3) = .897327

2) VS X(3) = .738895

1) VS Y = .686416

2) VS Y = .349748

3) VS Y = .769293

RININ STANDART HATASI = .061918

K NO. 1
ESYONA GIREN DEGISKEN 3
VIYESI = 13.0487
GERININ STANDART HATASI = .0416
YONLU KORELASYON KATSAYISI = .76929
T = .04034

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 3	.00128	.00035

K NO. 2
ESYONA GIREN DEGISKEN 2
VIYESI = 2.7821
GERININ STANDART HATASI = .0380
YONLU KORELASYON KATSAYISI = .83494
T = .09013

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI

X- 2	-.01092	.00655
X- 3	.00187	.00048

NO. 3
SYONDAN CIKARTILAN DEGISKEN 2
IYESI = -2.4343
ERININ STANDART HAT ASI = .0472
ONLU KORELASYON KATSAYISI = .76929
= .04034

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 3	.00128	.00040

NAN REGRESYON BASAMAKLARI SAYISI = 3

AL ELEMANLAR
O. DEGER

.194803
.454032
1.000000

VE HAKIKI DEGERLERIN KARSILASTIRILMASI

OKUMA	HAKIKI	TAHMINI	FARK
1.00	.10000	.04950	.05049
2.00	.08000	.05408	.02591
3.00	.07400	.06432	.00967
4.00	.07000	.07231	-.00231
5.00	.07200	.09422	-.02222
6.00	.06800	.08829	-.02029
7.00	.07800	.12116	-.04316
8.00	.08400	.11227	-.02827
9.00	.11000	.16157	-.05157
10.00	.14000	.13625	.00374
11.00	.28000	.20198	.07801

KLI REGRESYON

NO. 10010

YISI = 11

N SAYISI = 4

NIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

NIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

NLERIN TOPLAMI

X(1) = 70.0000

X(2) = 80.2900

X(3) = 555.1700

Y = 1675.0000

ELER TOPLAMI

1) VS X(1) = 576.000000

1) VS X(2) = 555.170000

1) VS X(3) = 4736.690000

2) VS X(2) = 660.554300

2) VS X(3) = 4801.113100

3) VS X(3) = 41805.422000

1) VS Y = 13120.000000

2) VS Y = 13521.200000

3) VS Y = 108993.150000

VS Y = 307175.000000

NLERIN ORTALAMA DEGERLERI

X(1) = 6.3636

X(2) = 7.2990

X(3) = 50.4700

Y = 152.2727

ARELER TOPLAMI

1) VS X(1) = 130.545460

1) VS X(2) = 44.233640

1) VS X(3) = 1203.790000

2) VS X(2) = 74.510300

2) VS X(3) = 748.876900

3) VS X(3) = 13785.993000

1) VS Y = 2460.910000

2) VS Y = 1295.223000

3) VS Y = 24455.900000

VS Y = 52118.190000

KORELASYON KATSAYILARI

1) VS X(2) = .448501

1) VS X(3) = .897327

2) VS X(3) = .738895

1) VS Y = .943452

2) VS Y = .657266

3) VS Y = .912367

RININ STANDART HATASI = 72.192925

K NO. 1
ESYONA GIREN DEGISKEN 1
VIYESI = 72.8950
GERININ STANDART HAT ASI = 25.2270
YONLU KORELASYON KATSAYISI = .94345
T = 32.31194

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	18.85098	2.20792

K NO. 2
ESYONA GIREN DEGISKEN 2
VIYESI = 13.2986
GERININ STANDART HAT ASI = 16.3987
YONLU KORELASYON KATSAYISI = .97914
T = -7.55188

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	16.22455	1.60582
X- 2	7.75130	2.12555

K NO. 3
ESYONA GIREN DEGISKEN 3
VIYESI = 6.7650
GERININ STANDART HAT ASI = 12.5016
YONLU KORELASYON KATSAYISI = .98944
T = -52.82453

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	24.97333	3.57951
X- 2	14.63574	3.10349
X- 3	-1.20173	.46203

ANAN REGRESYON BASAMAKLARI SAYISI = 3

NAL ELEMANLAR
NO. DEGER

10.702221
4.591813
18.829987

VE HAKIKI DEGERLERIN KARSILASTIRILMASI

OKUMA	HAKIKI	TAHMINI	FARK
1.00	40.00000	40.79136	-.79136
2.00	60.00000	61.47450	-1.47450
3.00	80.00000	90.81573	-10.81573
4.00	105.00000	108.30225	-3.30225
5.00	140.00000	150.27122	-10.27122
6.00	165.00000	143.27529	21.72471
7.00	185.00000	174.98[illegible]	10.01[illegible]
8.00	205.00000	195.73485	9.26515
9.00	215.00000	212.04616	2.9538[illegible]
10.00	235.00000	248.19441	-13.19441
[illegible]	[illegible]	[illegible]	[illegible]

KLI REGRESYON

NO. 10040

YISI = 11

N SAYISI = 4

NIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

NIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

NLERIN TOPLAMI

X(1) = 70.0000

X(2) = 80.2900

X(3) = 555.1700

Y = 3.1900

ELER TOPLAMI

1) VS X(1) = 576.000000

1) VS X(2) = 555.170000

1) VS X(3) = 4736.690000

2) VS X(2) =	660.554300
2) VS X(3) =	4801.113100
3) VS X(3) =	41805.422000
1) VS Y =	18.850000
2) VS Y =	22.537900
3) VS Y =	146.369300
VS Y =	.942700

NLERIN ORTALAMA DEGERLERI

X(1) =	6.3636
X(2) =	7.2990
X(3) =	50.4700
Y =	.2900

ARELER TOPLAMI

1) VS X(1) =	130.545460
1) VS X(2) =	44.233640
1) VS X(3) =	1203.790000
2) VS X(2) =	74.510300
2) VS X(3) =	748.876900
3) VS X(3) =	13785.993000
1) VS Y =	-1.450000
2) VS Y =	-.746200
3) VS Y =	-14.630000
VS Y =	.017600

ORELASYON KATSAYILARI

1) VS X(2) =	.448501
1) VS X(3) =	.897327
2) VS X(3) =	.738895
1) VS Y =	-.956600

2) VS Y = -.651614

3) VS Y = -.939223

ININ STANDART HATASI = .041952

NO. 1
SYONA GIREN DEGISKEN 1
IYESI = 96.9887
ERININ STANDART HAT ASI = .0128
ONLU KORELASYON KATSAYISI = .95660
= .36068

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	-.01110	.00112

NO. 2
SYONA GIREN DEGISKEN 2
IYESI = 21.6656
ERININ STANDART HAT ASI = .0070
ONLU KORELASYON KATSAYISI = .98848
= .38270

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	-.00965	.00069
X- 2	-.00428	.00091

NAN REGRESYON BASAMAKLARI SAYISI = 2

AL ELEMANLAR
O. DEGER

1.251804
1.251804
.053106

VE HAKIKI DEGERLERIN KARSILASTIRILMASI

OKUMA	HAKIKI	TAHMINI	FARK
1.00	.36000	.34810	.01189
2.00	.34000	.33844	.00155
3.00	.32000	.32705	-.00705
4.00	.31000	.31740	-.00740
5.00	.30000	.29911	.00088
6.00	.29000	.29808	-.00808
7.00	.28000	.27980	.00019
8.00	.27000	.26912	.00087
9.00	.26000	.25083	.00916
10.00	.24000	.24015	-.00015
11.00	.22000	.22186	-.00186

KLI REGRESYON

NO. 10050

YISI = 11

N SAYISI = 4

NIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

NIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

NLERIN TOPLAMI

X(1) = 70.0000

X(2) = 80.2900

X(3) = 555.1700

Y = 1.3660

ELER TOPLAMI

1) VS X(1) = 576.000000

1) VS X(2) = 555.170000

1) VS X(3) = 4736.690000

2) VS X(2) = 660.554300

2) VS X(3) = 4601.113100

3) VS X(3) = 41805.422000

1) VS Y = 11.248000

2) VS Y = 10.857980

3) VS Y = 95.427430

VS Y = .237800

NLERIN ORTALAMA DEGERLERI

X(1) = 6.3636

X(2) = 7.2990

X(3) = 50.4700

Y = .1241

ARELER TOPLAMI

1) VS X(1) = 130.545460

1) VS X(2) = 44.233640

1) VS X(3) = 1203.790000

2) VS X(2) = 74.510300

2) VS X(3) = 748.876900

3) VS X(3) = 13785.993000

1) VS Y = 2.555273

2) VS Y = .887422

3) VS Y = 26.485410

VS Y = .068167

ORELASYON KATSAYILARI

1) VS X(2) = .448501

1) VS X(3) = .897327

2) VS X(3) = .738895

1) VS Y = .856578

2) VS Y = .393761

3) VS Y = .863969

ININ STANDART HATASI = .082563

NO. 1
SYONA GIREN DEGISKEN 3
IYESI = 26.4951
ERININ STANDART HAT ASI = .0438
ONLU KORELASYON KATSAYISI = .86396
= .02721

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 3	.00192	.00037

NO. 2
SYONA GIREN DEGISKEN 2
IYESI = 8.6595
ERININ STANDART HAT ASI = .0322
ONLU KORELASYON KATSAYISI = .93714
= .10149

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI

X- 2	-.01629	.00553
X- 3	.00280	.00040

ANAN REGRESYON BASAMAKLARI SAYISI = 2

NAL ELEMANLAR

NO.	DEGER
I	.093438
2	2.202484
3	2.202484

I VE HAKIKI DEGERLERIN KARSILASTIRILMASI

OKUMA	HAKIKI	TAHMINI	FARK
1.00	.08800	.06334	.02465
2.00	.07600	.07336	.00263
3.00	.06500	.05241	.01258
4.00	.06800	.06989	-.00189
5.00	.07200	.04824	.02375
6.00	.07000	.10486	-.03486
7.00	.10500	.10718	-.00218
8.00	.13200	.15732	-.02532
9.00	.14300	.19558	-.05258
10.00	.21500	.20977	.00522
11.00	.33200	.28398	.04801

AKLI REGRESYON

I NO. 10060

AYISI = 11

EN SAYISI = 4

ENIN REGRESYONA GIRECEGI F SEVIYESI = 2.500

ENIN REGRESYONDAN CIKARILACAGI F SEVIYESI = 2.500

ENLERIN TOPLAMI

X(1) =	70.0000
X(2) =	80.2900
X(3) =	555.1700
Y =	1895.0000

RELER TOPLAMI

1) VS X(1) =	576.000000

(1) VS X(2) =	555.170000
(1) VS X(3) =	4736.690000
(2) VS X(2) =	660.554300
(2) VS X(3) =	4801.113100
(3) VS X(3) =	41805.422000
(1) VS Y =	14600.000000
(2) VS Y =	15308.300000
(3) VS Y =	121432.500000
VS Y =	386075.000000

KENLERIN ORTALAMA DEGERLERI

X(1) =	6.3636
X(2) =	7.2990
X(3) =	50.4700
Y =	172.2727

KARELER TOPLAMI

(1) VS X(1) =	130.545460
(1) VS X(2) =	44.233640
(1) VS X(3) =	1203.790000
(2) VS X(2) =	74.510300
(2) VS X(3) =	748.876900
(3) VS X(3) =	13785.993000
(1) VS Y =	2540.910000
(2) VS Y =	1476.523000
(3) VS Y =	25791.860000
VS Y =	59618.190000

KORELASYON KATSAYILARI

(1) VS X(2) =	.448501

(1) VS X(3) = .897327

(2) VS X(3) = .738895

(1) VS Y = .910791

(2) VS Y = .700555

(3) VS Y = .899651

ERININ STANDART HATASI = 77.212813

AK NO. 1
RESYONA GIREN DEGISKEN 1
EVIYESI = 43.7987
EGERININ STANDART HATASI = 33.6029
YONLU KORELASYON KATSAYISI = .91079

IT = 48.41221

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	19.46379	2.94101

AK NO. 2
RESYONA GIREN DEGISKEN 2
EVIYESI = 13.4152
EGERININ STANDART HATASI = 21.7840
YONLU KORELASYON KATSAYISI = .96763
IT = -4.77422

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	15.95960	2.13316
X- 2	10.34181	2.82356

AK NO. 3
RESYONA GIREN DEGISKEN 3
EVIYESI = 6.6779
EGERININ STANDART HATASI = 16.6598
YONLU KORELASYON KATSAYISI = .98357
IT = -64.71557

DEGISKEN	KATSAYI	KATSAYI STNDRT. HATASI
X- 1	27.54307	4.77010
X- 2	19.45687	4.13576
X- 3	-1.59110	.61571

LANAN REGRESYON BASAMAKLARI SAYISI = 3

ONAL ELEMANLAR

.NO.	DEGER
1	10.702221
2	4.591813

II VE HAKIKI DEGERLERIN KARSILASTIRILMASI

OKUMA	HAKIKI	TAHMINI	FARK
1.00	50.00000	48.47109	1.52890
2.00	70.00000	70.33390	-.33390
3.00	90.00000	109.39213	-19.39213
4.00	120.00000	127.02259	-7.02259
5.00	175.00000	182.92729	-7.92729
6.00	190.00000	162.28352	27.71648
7.00	215.00000	204.60014	10.39986
8.00	230.00000	215.17489	14.82511
9.00	240.00000	237.10941	2.89059
10.00	250.00000	268.06628	-18.06628
11.00	265.00000	269.61868	-4.61868

BIBLIOGRAPHY

1. Gediktaş, M. "Sürtünme Malzemelerinin Özelliklerinin Deneysel Tayini – Fren Kaplamaları" Istanbul : Teknik Üniversite Matbaası, 1968

2. Friction Materials for Engineers : A Design Manual by Ferodo Ltd, London : Chorley and Pickersgill Ltd., 1961

3. Twiss, S. B., and Basford P. R., "Properties of Friction Materials", parts I and II, ASME paper 57-SA-97, Transactions of ASME

4. Deutsche Verkehrsblatt, Buch III, 1965

5. Nutt, H., "Woven and Molded Friction Materials for Disk-type Clutches", Product Engineering, Vol. 17, 273-283, 1946

6. Berger, H., "Asbest-Fidel" Stuttgart : Verlag A. W. Gentner AG., 1961

7. Frank, K. "Asbest" Hamburg : Verlag Becker und Haag, 1952

8. Halstead, R. T., "The Fundamentals of Asbestos Friction Materials", Paper Trade Journal, Vol. 118, pp. 30-33, 1944

9. Jania, Z. J. "Friction Clutches", Mechanical Design and Systems Handbook, p. p. 28. 2-28. 38, New York : McGraw-Hill Book Co., 1964

10. Tabor, D. "Friction, Lubrication, and Wear", Metals Reference Book, London : Butterworth and Co., 1961

11. Spotts, M. F. "Design of Machine Elements" 3bb ed., Englewood Cliffs, N. J. : Prentice-hall, Inc., 1961

12. Bowden, F. P., and Tabor D. "The Friction and Lubrication of Solids" New York : Oxford University Press, 1950

13. Archard, J. F., "Contact and Rubbing of Flat Surfaces" Journal of Applied Physics, Vol. 24, pp. 981-988, 1953

14. Bowden, Moore, and Tabor, "Formation and Shearing of Junctions" Journal of Applied Physics, Vol. 14, p. 80, 1943

15. Bowden, F. P., and G. W. Rowe, "The Adhesion of Clean Metals" Procedings of the Royal Society, Vol. A233, pp. 429-442, 1956

16. Tabor, D., "Junction Growth in Metallic Friction : The Role of Combined Stresses and Surface Contamination ", Proc. Roy. SOc. (London), Vol. A 251, pp. 378-393, 1959

17. Lipson, C., "Wear Considerations in Design" Englewood Cliffs, NJ : Prentice-Hall, Inc., 1967

18. Nathan, G. K., and J. D. Jones, " The Empirical Relationsihp between Abrasive Wear and the Applied Conditions," Wear, Vol. 9, pp. 300-309, 1966

19. Freitag, E. H., "Wear and Friction of Metals at Very High Speeds", Conference on Lubrication and Wear, Paper No. 44, London Oct. 1-3, 1957

20. Hallyday, J. S., "Applications of the Reflection Electron Microscopy to the Study of wear Dots", Conf. on Lub. and Wear, Paper No. 40, London, 1957

21. Sinclair, D., "Friction Brakes", Mech. Design and Syetems Handbook, pp. 28, 38-28, 67, New York : McGraw-Hill Book Co., 1964

22

22. Bikerman, J. J., "Surface Roughness and Sliding Friction", Rev. Mod. Phys. Vol. 16, pp. 53-68, Jan. 1954

23. Spurr, R. T., "Frictional Behaviour of a Simple Rheological Material", Jour Appl. Phys., Vol. 32, p. 1450, Aug. 1961

24. Flom, D. G., "Rolling Friction of Polymeric Materials" Jour. Appl. Phys., Vol. 32, p. 1426, Aug. 1961

25. Timoshenko and Goodier, "Metals Design Handbook", Chapter 10 : Elasticity, ASME Series, 1968

26. Winne, D. H. and Wundt, B. M., "Application of the Griffth - Irwin Theory of Crack Propagation to the Bursting Behaviour of Disks", ASME Paper No. 52A-249, Dec. 1957

27. Carslaw, H. S., and J. C. Jaeger, "Conduction of Heat in Solids", 2nd ed., London : Oxford University Press, 1959.

28. Lindörf, H., "Technische Temperaturmessungen", Essen: Verlag Girardet, 1956

29. Lindorf, H., "Genaue Temperaturmessungen mit Thermoelementen," Frakfurt: Hartmann und Braun, 1963

30. "Brake Lining Quality Control", SAE Handbook, 1968

31. Efroymson, E. "Stepwise Regression", Esso Mathematics and Research, (Program adapted by Prof. Tümay)

32. Oetzel, J. G., "Characteristics of Molded and Woven Friction Materials", Machine Design, Vol. 23, p.128, Aug. 1961