BICYCLE RIM BRAKE:

INVESTIGATION OF A PROMISING FRICTION MATERIAL

AND DEVELOPMENT OF PERFORMANCE SPECIFICATIONS

by

BRIAN DONALD HANSON

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Signature of Author . . Signature redacted . . .
Department of Mechanical Engineering, October 9, 1970

Certified by. . . . . Signature redacted . . . . .
Thesis Supervisor

Accepted by . . Signature redacted . . . .
Chairman, Departmental Committee on Theses

Abstract

The static and dynamic friction and wear characteristics of Lockheed-brake friction material were investigated under dry and wet conditions, and in transition between wet and dry conditions, to determine the suitability of this material for use in a bicycle rim brake. The nominal dry and wet coefficients of friction were found to be 0.35 and 0.10, respectively. Dry-to-wet transition is sudden; wet-to-dry transition is delayed, and progresses in a fluctuating manner. Specific energy for wear is 4 to 6 million foot-pounds per cubic inch. These results place restrictions on the design of simple calipers, servo-action brakes, antiservo brakes, and power-assist brakes for bicycles.

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Background

One approach to the problem of urban congestion and pollution may be to implement more widespread use of man-powered vehicles, which have the potential advantages of cheapness, compactness, simplicity, silence, and lack of polluting effluents.

The category of MPV's (man-powered vehicles) is confined mainly to the bicycle, which was developed in something approaching its present form in the latter part of the last century. This vehicle's current popularity is evidence of its attractiveness to many people in spite of its relatively poor reliability and maintainability, lack of weather protection, and serious rider exposure to collision dangers. If any of these drawbacks could be overcome, the increased attractiveness of the bicycle could result in its being used by enough more people so that auto congestion could be significantly reduced.

The problem

Safety is an important aspect of all modes of transportation, from walking to space-flight; certainly it should not be omitted in the case of the bicycle. Safety has two forms: accident prevention and crash protection. Current bicycles are lacking in both respects, but the former of the two is considered here.

The present most-popular bicycle brake, the caliper-and-rim type, is often only marginally effective when dry, and dangerously poor in the wet. Additionally, wear of present friction-block material is excessive in heavy use, and adjustment and replacement are difficult.

It is ultimately desired to develop a bicycle brake with significantly greater wet-capability than present varieties, at no sacrifice of reliability and maintainability (which are sufficiently poor already), and compatable with existing bicycle systems, i.e. capable of retrofit onto bicycles now in use. This thesis concerns itself with the initial stages of such a development.

Attacking the problem

An essential step in the design process is to define, as closely as practicable, the performance specifications the proposed mechanism must meet. These specifications should consist of three parts:

1. Anticipated inputs to the mechanism
    a. The nature of the command signal
    b. Available sources of power
    c. Noise and other environmental influences
2. Desired output produced by the mechanism
    a. The nature of the output
    b. Permissible fluctuation of output under extreme conditions
3. Priorities: Relative importance of cost, weight, appearance, fail-safety, environmental and social consequences, etc., and what characteristics of

the mechanism should be preserved at the expense of others

Some of these specificationscan be calculated theoretically; others must be determined on the basis of experiment; still others can only be estimated, and must be substantiated using a prototype of the mechanism itself.

Necessity for experiment

For a bicycle brake, it is necessary to determine the sliding characteristics of various friction-block materials on a bicycle-rim surface, both dry and water-wetted. These must be known, and the most promising of such materials identified, before the characteristics of the mechanism itself can be specified.

Friction characteristics for automotive and industrial friction materials are compiled under carefully-controlled conditions, dry and in oil, on cast-iron and uncoated steel surfaces.(1) No data are available from brake manufacturers involving either the presence of water or the use of plated or nonferrous surfaces.(2)

Other work

Asbell (3) investigated various brake-block materials and shapes with respect to their coefficients of sliding friction on bicycle-rim material, wet and dry. His investigation included normal forces to 150 lbf and pressures to 300 psi on the friction material. He found wet coefficients to be smaller, by a factor of 3 or greater, than corresponding dry coefficients for all materials and configurations tested. His conclusion was that Lockheed-brake friction material was the most suitable for a bicycle-rim-brake, from the standpoints of wear and of friction-retention when wet.

Goals of the experiment

1. A tentative decision has been made, that any new rim-brake mechanism will employ Lockheed-brake friction material. Before this decision can be accepted, however, Asbell's results for this material must be verified, i.e. more precise numerical determinations of coefficients of friction, dry and wet, must be made.

2. One guess as to the phenomenon underlying the dry-to-wet variation in sliding coefficients, is water-lubrication. This means the material rides on a thin film of water rather than directly on the rim material. It is desired to discover if there exists some pressure above which this water-film breaks down, re-establishing direct sliding contact, and restoring dry-friction behavior.

3. Little is known about the manner in which the friction characteristics change from dry values to wet values, with the introduction of water to the system; or about the manner in which recovery of dry-friction characteristics takes place as the wheel dries. Both of these dynamic transition phenomena should be observed to see if they impose

any additional performance requirements on the mechanism.

4. Some force-multiplication will probably be employed in any brake we might design, with consequent sensitivity of the mechanism to friction-material wear, from a geometric standpoint. The energy-absorption-versus-wear characteristic of this friction material should be determined, in order to estimate how frequently brake adjustments must be made, and consequent desirability of including a "self-adjust" mechanism into the brake.

5. Lockheed-brake friction material is planar-orthotropic, i.e. consists of compressed layers. It is desired to determine the effect (if any) of different orientations on friction and wear.

Experimental setup

The wheel used for the experiment was a front wheel from a lightweight touring bicycle. The wheel was of steel, and its surface finish had the appearance of nickel-chromium plate. Spokes and hub were left attached, and four small wood blocks were attached inboard of the rim at 90° intervals for mounting.

For positioning and spinning the wheel, the 30-inch lathe in Room 1-014 was used (see Fig. 1). Available speeds were 110, 190 and 330 rpm, corresponding to 10, 17 and 30 mph. Additionally, for the lathe sheave and chuck assembly, with drive gears disengaged, the equivalent linear inertia at the wheel radius was found to be 500 lbm.

Friction material specimens were cut from the same piece of Lockheed automotive brake material used by Asbell as his source. Four cylindrical specimens were cut, of diameter .357 ± .005 inch, for a face area of 0.10 in^2, and of varying lengths. The material is orthotropic, consisting of compressed layers. In three of the specimens the layers were normal to the axis of the cylinder; in the fourth, parallel.

A Cook-Smith three-component force dynamometer was obtained. Two of its outputs, corresponding to normal and tangential forces on the wheel, were fed into a Sanborn 311 amplifier-recorder, producing time plots of both outputs.

The dynamometer is very stiff (on the order of 10^4 lbf/in). If the friction material were mounted rigidly to the dynamometer, any irregularities in the rotating rim would result in large, sudden force fluctuations. Therefore a mounting was fabricated having a flex hinge and trailing arm, with provision for inserting compression springs, so that the stiffness of the mounting was stepwise variable from 50 lbf/in to 500 lbf/in to suit various experimental conditions (see Fig. 2).

Water, when required, was directed onto the wheel by a hand-held tube, permitting sudden application and removal of the wetting flow.

Experimental procedure

The experimental variables (inputs) took on the following values:

Input	Values
Rim speed	Low: 110 rpm Equivalent vehicular speed 10 mph. Rim surface speed 13 ft/sec Med: 190 rpm; 17 mph; 22 ft/sec Hi: 330 rpm; 30 mph; 39 ft/sec
Nature of run	Dry only Wet only Dry-wet Wet-dry Dry-wet-dry Dry wheel, wet friction material
Pressure on friction material	0-2000 psi
Orientation of orthotropic planes in friction material	z, θ, r corresponding to cylindrical coordinates with origin at hub of wheel. (z indicates planes normal to z-axis, etc.)

Each experimental run consisted of the following:

1. Pad configuration selected and mounted

2. Pad-protrusion measurement taken with scale, following an initial "seating-in" run

3. Wheel rotating at desired speed

4. Pad brought up close to rim (see Fig. 3), and recorder activated

5. Wetting supply activated

6. Pressure increased to desired value (see Fig. 4)

7. Wetting supply activated and/or secured as desired, and indications made on recording chart

8. Run continued until transients have died out, then

pressure decreased to zero, equipment secured, and another pad-protrusion measurement taken.

<u>Form of raw data</u>

The output of each experimental run was:

1. An annotated time-plot of two voltages, corresponding to normal and tangential* forces exerted by the friction material on the rotating wheel

2. Measurement of pad-material wear

<u>Processing of data</u>

Data reduction for a typical experimental run is shown in Appendix B.

Tabulation of finished data is given in Appendix C.

<u>Summary of experimental results</u>

1. Nominal coefficient of friction: 0.35 dry, 0.10 wet.

2. No significant dependence of coefficient of friction, wet or dry, on pad pressure, up to 2000 psi.

3. Transition from dry coefficient to wet coefficient occurs immediately upon introduction of water.

4. When water flow is halted, wet characteristics prevail for several turns of the wheel (4 to 15 in tests). Then the coefficient abruptly begins to increase. The <u>average</u> coefficient behaves roughly like the transient response

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* The dynamometer actually detected the moment about some horizontal axis, instead of tangential force; geometric corrections, which had some dependence on normal force, had to be introduced (see Appendix A).

of a first-order system, logarithmically approaching the dry value, reaching 90% of the dry value in 15 to 60 more turns of the wheel. The <u>instantaneous</u> coefficient may vary by as much as 0.10 around the wheel during the wet-to-dry transition.

5. When the friction material is mounted so that its orthotropic planes are parallel to the rubbing surface, the material can dissipate 4 to 6 million foot-pounds of energy for each cubic inch worn away. With the orthotropic planes oriented any other way, this figure decreases by at least half. Raybestos-Manhattan, Inc., a leading manufacturer of friction materials, give 5 to 10 million foot-pounds per cubic inch as the range of specific energies for their molded-organic friction materials, under slightly different test conditions on a cast-iron friction surface.(1)

<u>Remarks</u>

During Run No. 12, one side of the test wheel became seriously "rippled," forcing a change to the other side of the wheel. At the time, the average normal force exerted on the wheel was less than 200 lb. However, the flexible mount for the friction material is essentially an undamped spring-mass system. At one point on the circumference of the wheel there is a weld, ground quite smooth, but with a higher local stiffness than the rest of the wheel. This disturbance passing under the friction material was enough to stimulate oscillation at the natural frequency of the system, about 250 cycles/sec, building in amplitude over several runs, until Run 12, when the friction material actually bounced free of the wheel, and the sum of static and dynamic forces exceeded 500 lbf. This effect was most apparent during wet portions of runs, and may have to be considered when designing a brake mechanism.

Conclusions: The effect of experimental results on design specifications

1. Friction-block area

Since friction characteristics are independent of pressure on the friction material, the friction blocks should have as great a cross-sectional area as geometrical and other considerations permit, in order:

1. To minimize changes in block thickness for a given volume of wear, thus minimizing frequency of adjustment and replacement of blocks;
2. To reduce the risk of wheel-deformation; and
3. To reduce the variation of friction forces during each rotation of the wheel.

2. Force multiplication

Let us establish an altogether reasonable requirement, that the brake mechanism should produce a deceleration of 0.4g for a bicycle-and-rider of 250 lbm, braking the front wheel only. The required tangential force to be exerted on the bicycle rim is 100 lbf, or 50 lbf per block. To achieve this magnitude of force in the wet, the brake mechanism must exert a normal force of 500 lbf on each friction block. The maximum force exertable in squeezing a handgrip by 5th-percentile female subjects is 57-58 lbf (4). A nominal force-multiplication of 2 takes place at the handgrip; therefore the input (cable) force to the brake mechanism is of the order of 100 lbf, and a force multiplication of 5 must occur in the mechanism.

3. Servo action

The employment of servo action, or leading-shoe effect, in the brake mechanism has been considered. However, in a servo-action brake, the ratio of tangential (braking) force to applied force (known as Brake Effectiveness Factor or B.E.F.) is proportional to the quantity $\mu/(C - \mu)$, where μ is the coefficient of friction and C is a geometric parameter (see Appendix D). By comparison, the B.E.F. of the present caliper mechanism is directly proportional to μ.

In practice, a servo-action brake is designed so that the parameter C only slightly exceeds the maximum anticipated value of μ. Any decrease in μ results in a disproportionate decrease in B.E.F.; any increase in μ affects the stability of the system, increasing the tendency towards noise and "grabbing" of the wheel. Thus a servo-action brake is limited to applications where it is protected from environmental effects on friction. We, on the other hand, desire a mechanism whose behavior is affected minimally by friction variations; the rider should not have to adjust his grip for changes in friction, especially the rapid, complicated fluctuations occurring during the wet-to-dry transition.

4. Antiservo action

The opposite of servo action is antiservo action, or trailing-shoe effect. The B.E.F. for this class of mechanism is proportional to the quantity $\mu/(C + \mu)$, where the parameter C is the same as for an equivalent servo-action mechanism. The antiservo B.E.F. exhibits less of a dependence on u than that of existing calipers; the smaller the magnitude of C, the smaller the effect of changes in μ, on the performance of the antiservo brake mechanism(Appendix E).

5. Wear tolerance

The mechanical stroke or travel of the handgrip cable is 3/4 inch; a 5-fold force multiplication reduces the brake-block travel to 0.15 inch, whether the brake is a simple caliper or an antiservo mechanism. Allowances for play in the mechanism, and for compliance in the mechanism, friction material and rim, reduce to 0.05 inch the range of travel inside which the friction material must contact the rim. (An additional consideration is that, even if the brake were pivoted to follow the rim-wobble, some rubbing would occur with the brake fully released. This condition is intolerable to most riders, and must be somehow overcome.) As the point of contact approaches the lower limit of this range, an adjustment becomes necessary which will move the point of contact to the upper limit. That is, an adjustment must be made whenever the sum of the wear of both friction blocks in the brake approaches 0.05 inch since the previous adjustment. For friction blocks of area 1 in^2 this occurs after $2(10^5)$ ft-lbf have been dissipated.

6. Frequency of adjustment

A 250-lbm bicycle-and-rider stopping from 30 mph dissipates 7500 ft-lbf. The same rider braking down a 2-mile, 10% grade dissipates 264,000 ft-lbf. Adjustment of the brake mechanism described above would be required after 20 to 30 such stops, and after every such downgrade. Although actual brake-usage figures have yet to be precisely determined, either by survey or by experiment, a tentative conclusion is that a self-adjusting mechanism may be necessary.

7. Power-assisted brake actuation

A class of mechanisms is considered which uses wheel motion to progressively apply pressure to the friction blocks. This pressure would increase until the braking force reaches the desired value, proportional to an input signal (either a force or a displacement) from the handgrip cable. A representative system using a roller-and-screw is statically analyzed in Appendix E.

The advantages of this type of system are as follows:

1. Statically, the mechanism performs like an anti-servo, except that the parameter C can be made an order of magnitude smaller than the practical minimum for a simple antiservo. Thus the braking action can be made very nearly independent of coefficient of friction.

2. The system is unaffected by friction-material wear, except for a slight increase in the time it takes for the friction blocks to contact the rim; therefore, self--adjustment is not necessary.

The disadvantages of this type of system are:

1. Speed of braking action is directly proportional to the speed of the vehicle, i.e. system lag is in terms of wheel revolutions (or fractions thereof). The system must be made "quick" enough to be responsive at low speeds, which require brake application within one wheel revolution; yet the system must be controllable at high speeds, and free from overtightening due to inertial "overshoot" of the mechanism beyond the desired braking-force value, and consequent risk of wheel lockup.

2. The mechanism must release the brake on command, quickly and positively at all speeds, including at-rest. Such a feature is not inherent in this class of mechanism, and must somehow be added-on.

3. It is not known whether the mechanism is dynamically

able to handle the rapid, complicated fluctuations in friction characteristics which occur as a wet wheel dries.

4. It is not known how water affects the way in which power is transmitted from the rotating wheel to the brake-application mechanism. These effects may detract from the otherwise favorable-appearing wet-weather characteristics of this type of system.

Recommendations for further work

1. A study should be made of bicycle-braking demands in urban traffic, to verify adjustment-frequency and speed--of-response requirement estimates given above.

2. Raybestos-Manhattan, Inc., have supplied samples of two friction materials,which they believe may be suitable for rim-brake applications. These materials should be tested in an arrangement similar to the experiment described in this report.

3. Prototypes of a power-assist and at least one other mechanism type should be constructed, having variable geometries so that the effects of changing each parameter can be studied. If time permits,the debugged prototypes should be field-tested on an actual bicycle, with the idea of arriving at a mechanism which can be produced and sold.

References

(1) Raybestos-Manhattan, Inc., Engineering Bulletin No. 501.

(2) Raybestos Division of Raybestos-Manhattan, Inc., Paul Lee, Chief Sales Engineer, Letter of 23 September 1970.

(3) Asbell, O.D., Jr., S.B. Thesis, Department of Mechanical Engineering, M.I.T., 1969-70.

(4) Survey by Fisher and Birren, 1946; tabulated in Morgan, Cook, Chapanis, and Lund, Human Engineering Guide to Equipment Design, New York, McGraw-Hill, 1963, Table 11-105

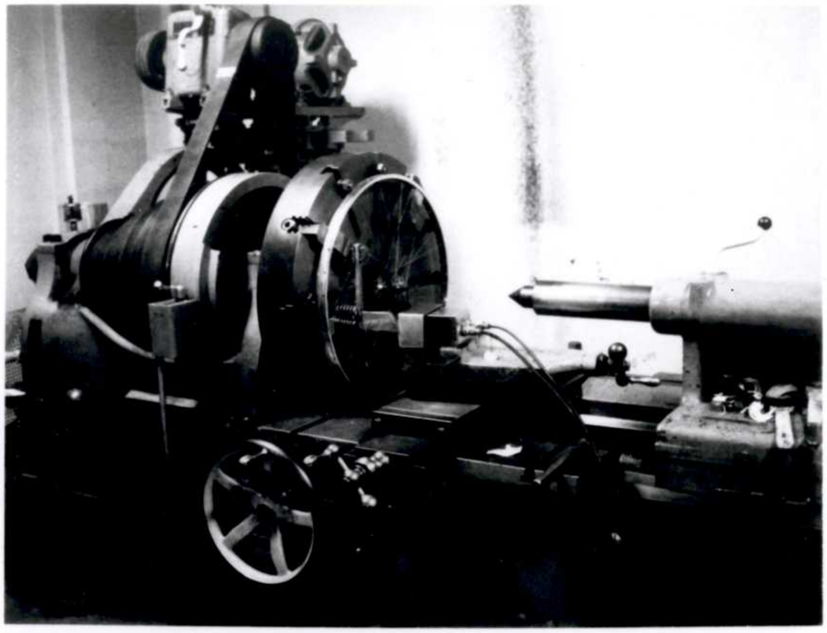


Fig. 1. Experimental setup in Room 1-014

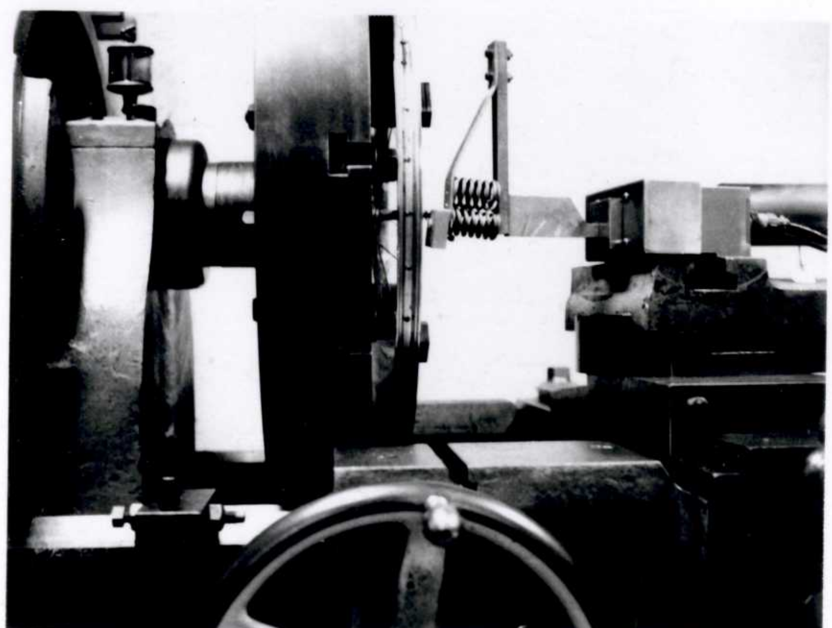


Fig. 2. Force dynamometer and flexible mounting for friction material, in position for experiment. As shown, the mounting has a stiffness of 405 lbf/in.

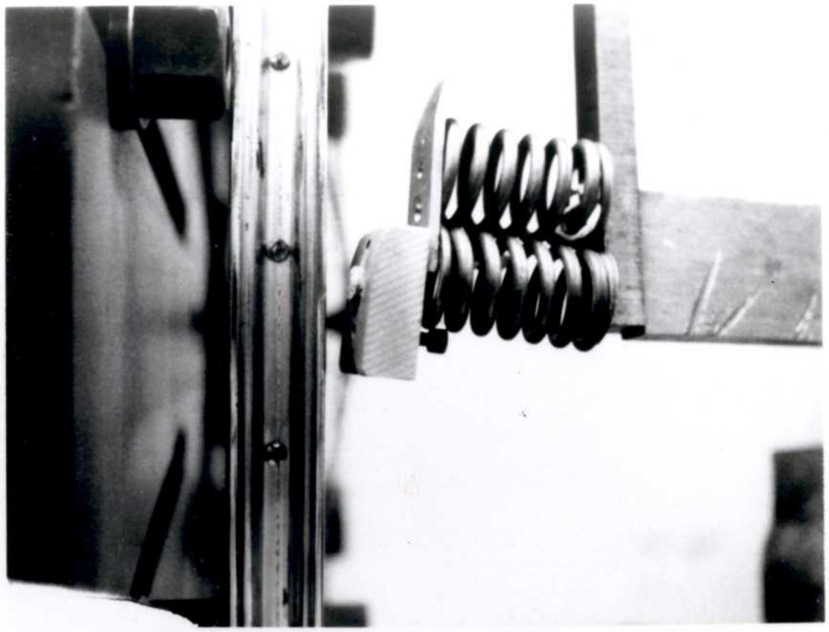


Fig. 3. Detail of friction material at start of experimental run

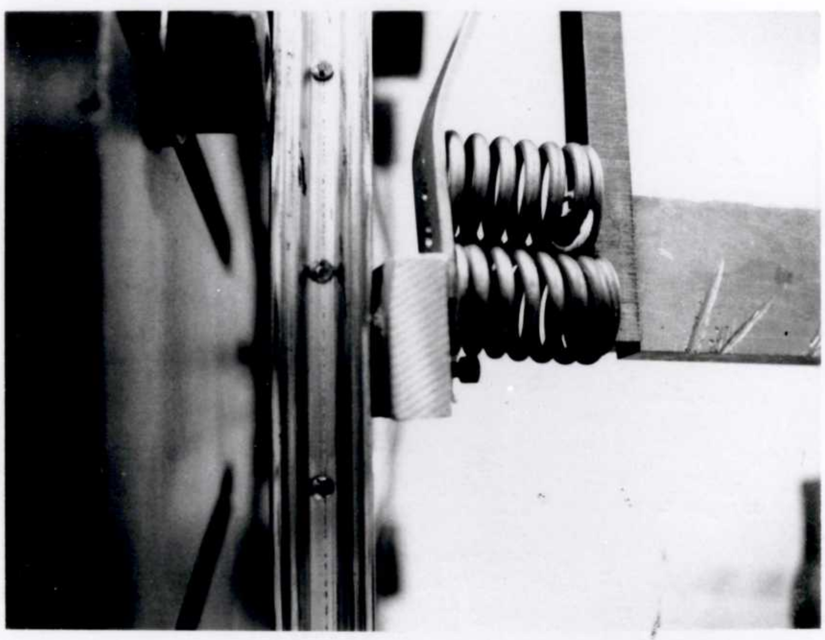


Fig. 4. Friction material in contact with rim, with F_n approximately 150 lbf

Appendix A. Correction for geometry of flexible mount

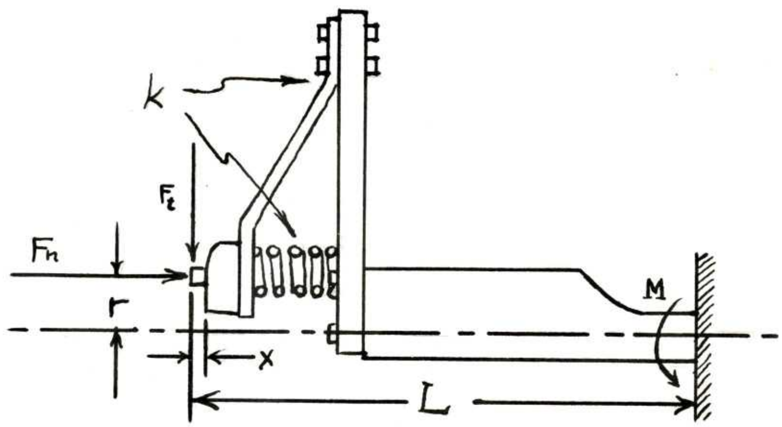


Fig. 5. Forces and geometry of flexible mount.

$$L = L_o + x - F_n/k$$

Total moment $M = (F_t \cdot L) - (F_n \cdot r)$

The quantities detected by the equipment are M and F_n.

$$F_t = f(M, F_n)$$

$$F_t = \frac{M + F_n \cdot r}{L}$$

$$F_t = \frac{M + F_n \cdot r}{L_o + x - F_n/k}$$

Spring(s) in place	L_o, in	k, lbf/in
1	7.65	213
2	7.65	245
1 and 2	7.70	405
3	8.0	503

r took on the following values: 1 in for Runs 1 - 8, 0.5 in for Runs 9 - 22, and zero for Runs 23 - 27.

x, the friction-material projection, was typically 1/8 in.

Appendix B. Processing of data for a typical run (No. 11)

Nature of run: Dry-wet-dry

Speed: Low, 13 ft/sec

Flexible-mount data: Springs 1 and 2 in place;

Offset $r = 0.5$ in

Pad protrusion: Initial, $11\frac{1}{2}/64$ or $.180 \pm .005$ in

Final, $10\frac{1}{2}/64$ or $.165 \pm .005$ in

Conversion factors: For F_n, 16.7 lbf/scale-division

For M, 11.2 lbf-in/scale-division

Geometric correction factors: $r = 0.5$ in; $x = 0.17$ in;

$L_o = 7.70$ in; $k = 405$ lbf/in.

Geometric correction formula: $F_t = \dfrac{M + 0.5\,F_n}{7.87 - F_n/405}$

Point (see Fig.6)	Observed		Converted		Calculated	
	F_n (div)	M (div)	F_n (lbf)	M (lbf-in)	F_t (lbf)	μ
1 (dry)	7.0	23	117	257	41	.35
2 (wet, at water cutoff)	7.5	4	125	45	14	.11
3 (5 turns later)	7.5	4	125	45	14	.11
(Recovery begins at 7th turn)						
4 (10 turns)	7.2	8(±4)	120	90(±45)	20(±6)	.17(±.05)
5 (15 turns)	7.0	10(±5)	117	112(±56)	22(±7)	.19(±.06)
6 (20 turns)	7.0	10(±2)	117	112(±22)	22(±3)	.19(±.03)
7 (30 turns)	6.8	18	114	201	34	.30
8 (40 turns)	6.5	20	109	224	37	.34
9 (50 turns)	6.5	21	109	235	38	.35

Peak power (point 1) = 1.0 hp; density 10 hp/in^2.

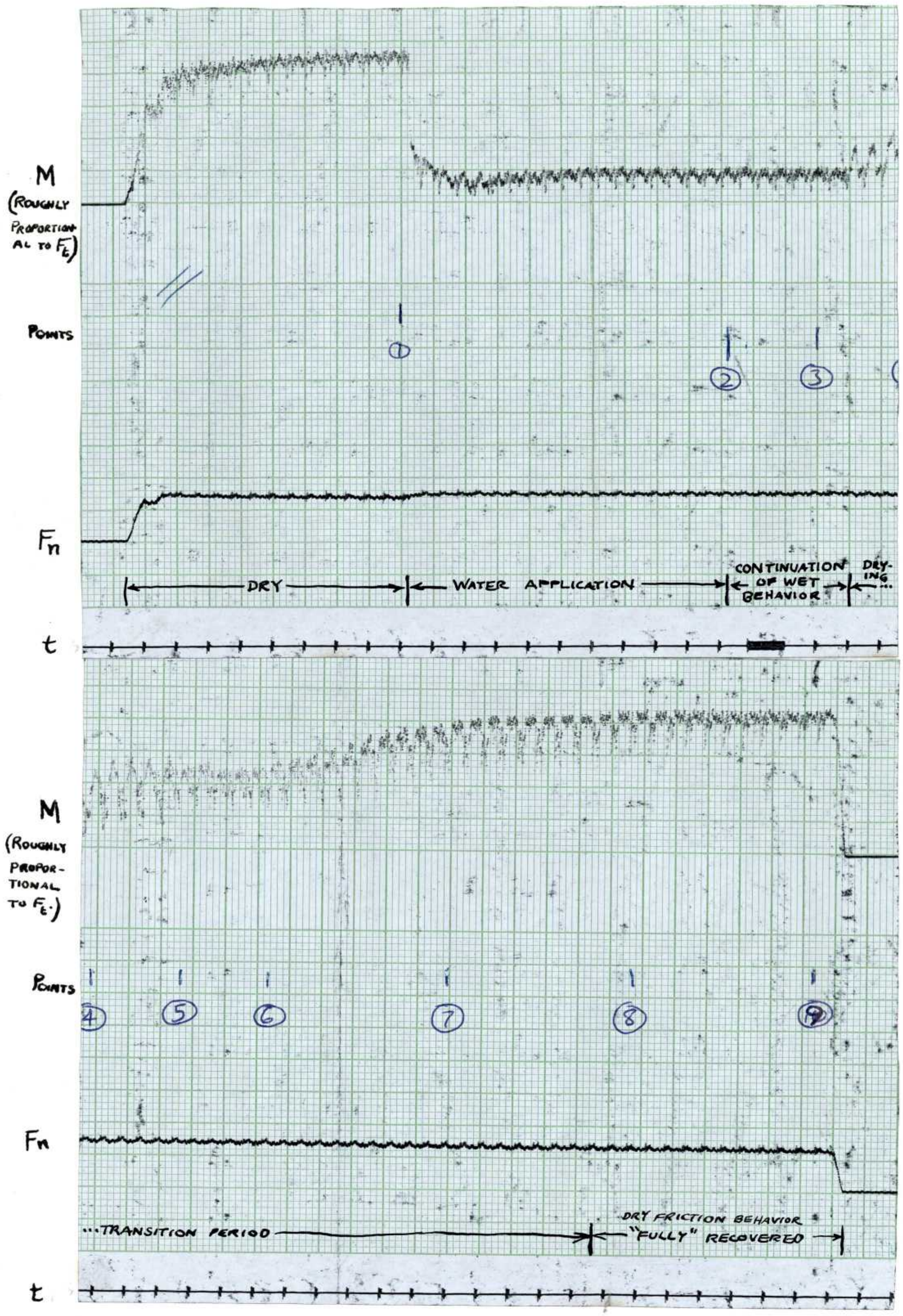


Fig. 6. Graphical output (raw data) for Run 11, with one-second time marks, and a signal triggered manually approximately 1 sec. after withdrawal of water stream.

Appendix B (continued) - Energy calculations for Run No.11

$$E = \int F \cdot dx = \int F_t \cdot V \, dt = V \cdot \int F_t \, dt \text{ (for } V = \text{const.)}$$

$$F_t = \frac{M + 0.5F_n}{7.87 \text{ in} - F_n/(405 \text{ lbf/in})}$$

Assume F_n = const. = 116 lbf.

Then $F_t = \dfrac{M + 58 \text{ lbf-in}}{7.68 \text{ in}} = (0.13M + 7.6)\text{lbf}$

$$\int_0^T F_t dt = 0.13\int_0^T M \, dt + 7.6T$$

Graphically, From Fig. 6, $\int M \, dt$ is found to be 600 scale-division·seconds, or 6700 lbf-in-sec.

T = 48 sec., therefore $\int F_t dt = 0.13(6700) - 7.6(48)$ lbf-sec, or 1240 lbf-sec.

Total dissipated kinetic energy $E = V \cdot \int_0^T F_t dt = 13(1240)$ ft-lbf, or 16,200 ft-lbf.

Volume of friction material wear: $v = A(x_i - x_f)$

$$v = 0.1 \text{ in}^2 (0.015 \text{ in})$$

$$v = 1.5 (10^{-3}) \text{ in}^3$$

Specific energy of friction material:

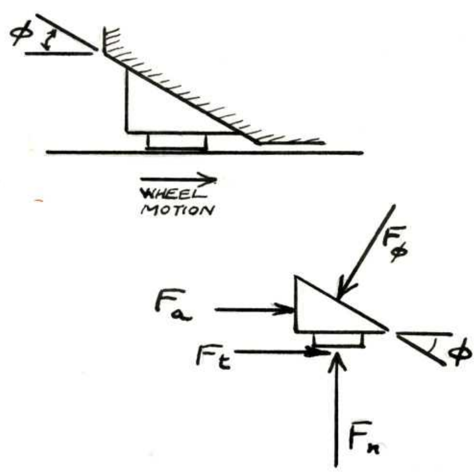
$$\frac{E}{v} = 1.1(10^7) \frac{\text{ft-lbf}}{\text{in}^3}.$$

Appendix C: Tabulation of experimental results

Run number	Friction material: orientation of orthotropic planes	Max. friction-material pressure, Kpsi	Equivalent vehicular speed, mph	μ dry	μ wet	Turns to onset of wet-dry recovery	Turns from onset to 90% recovery	Max. power density, hp/in^2	Dissipated energy, ft-lbf$\cdot 10^3$	Spec. energy, $\frac{\text{ft-lbf}}{\text{in}^3}\cdot 10^6$	Remarks
1	z	1.2	10	.34				11.2			)
2	z	2.1	10	.42				22.2			) Dry only; no wear measurements
3	z	2.1	17	.42				35.8			)
4	z	1.4	10	.38				13.5	7.5		
5	z	1.3	17	.38				17.8	6.8	5.7	
6	z	2.0	30	.40				57.5	8.8		Panic-stop from 30 mph
7	z	1.8	10		.07			3.7	16.2	5.4	Wet only
8	z	1.2	10		.07	8	60	8.7			Wet-dry
10	z	1.3	17	.35	.11	4	26	17.	17.5	4.4	
11	z	1.2	10	.35	.11	7	31	10.	16.2	11.0	
12	z	1.6	17	.35	.11						No wear meas.; Dry-wet only
16	z	2.0	30	.36				47.	12.0	4.0	Panic-stop from 30 mph
17	z	2.0	17	.35				29.	6.6	4.3	Pad presoak; μ initial =.19 μ =.35 after 15 turns.
20	θ	1.3	17	.38	.08	NR	15 25	18.	3.5	2.3	
21	θ	1.2	17	.30	.12	12	20	14.	7.3	0.8	Wet-dry
22	θ	1.3	30	.31				35.	2.6	0.2	
24	r	1.2	17	.35	.10			16.	3.2	2.0	Dry-wet only
25	r	1.5	17	.30				12.	3.2	0.4	Dry wheel, damp pad
27	z	0.8	17	.31	.09	15	30	10.	6.5	1.1	Pad presoak; μ final =.25

Appendix D: Analysis of a representative servo-action brake mechanism

Ramp servo with tangential applied force



Equilibrium (disregarding moments):

$$\Sigma F_x = 0; \quad F_a + F_t - F_\phi \sin\phi = 0, \qquad (1)$$

$$\Sigma F_y = 0; \quad F_n - F_\phi \cos\phi = 0. \qquad (2)$$

Elemental relation: $F_t = \mu \cdot F_n$.

Substituting for F_n in (2), we get

$$F_\phi = F_t/(\mu \cos\phi).$$

Substituting into (1),

$$F_a + F_t - \frac{F_t}{\mu}\cdot\left(\frac{\sin\phi}{\cos\phi}\right) = 0,$$

$$F_a = F_t\left(\frac{\tan\phi}{\mu} - 1\right).$$

Brake effectiveness factor B.E.F. $\triangleq F_t/F_a$,

$$\text{B.E.F.} = \frac{1}{\frac{\tan\phi}{\mu} - 1} = \frac{\mu}{\tan\phi - \mu}.$$

Thus the B.E.F. is of the form $\mu/(C - \mu)$, where C is the tangent of the ramp angle ϕ.

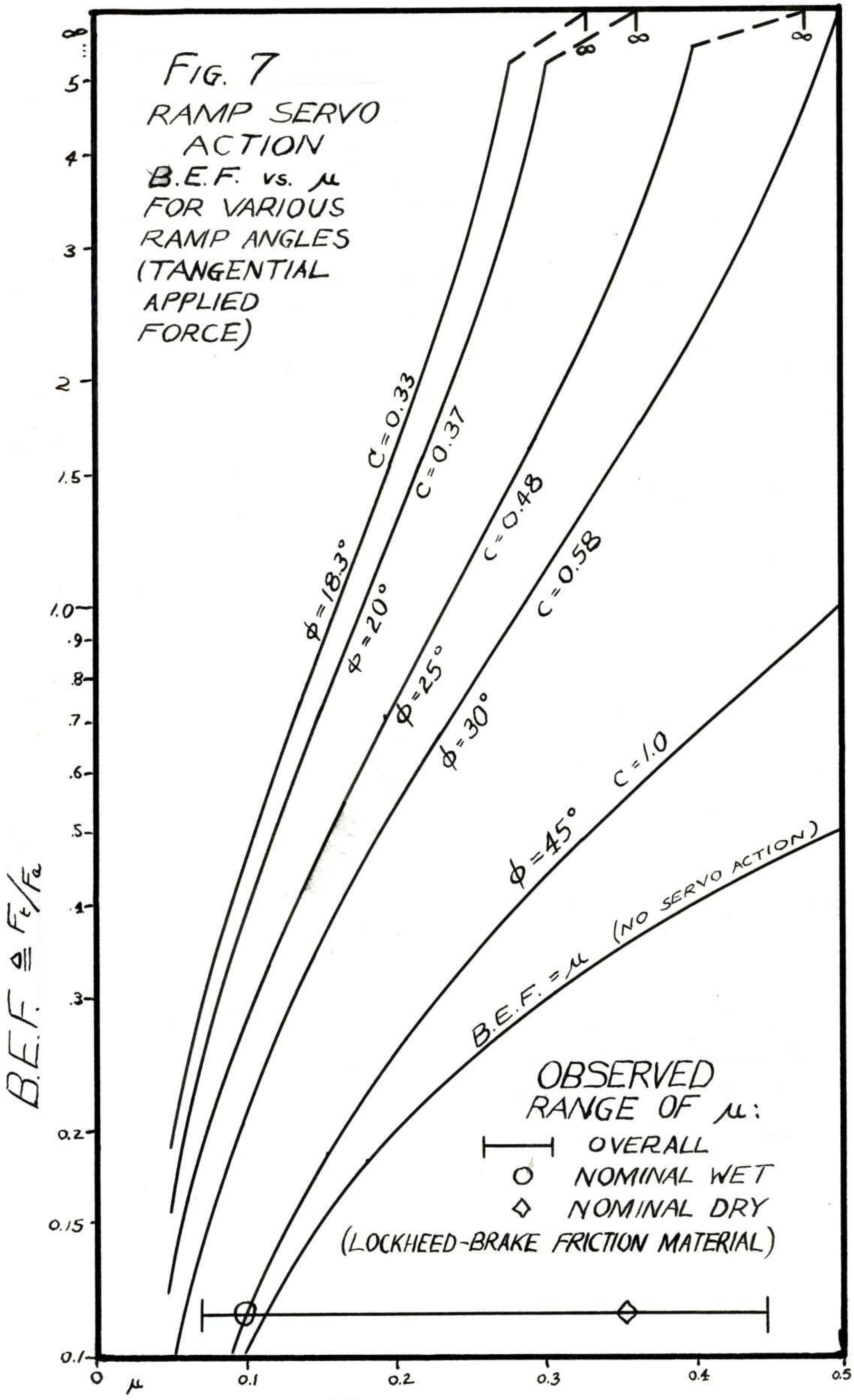
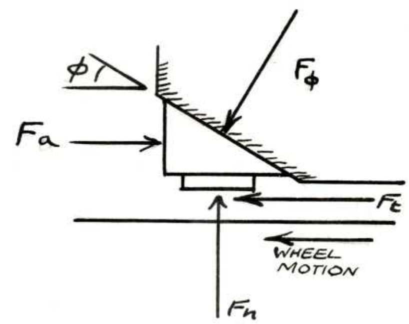


FIG. 7
RAMP SERVO ACTION
B.E.F. vs. μ
FOR VARIOUS RAMP ANGLES
(TANGENTIAL APPLIED FORCE)
C=0.33
C=0.37
C=0.48
C=0.58
C=1.0
ϕ=18.3°
ϕ=20°
ϕ=25°
ϕ=30°
ϕ=45°
B.E.F.=μ (NO SERVO ACTION)
OBSERVED RANGE OF μ:
OVERALL
NOMINAL WET
NOMINAL DRY
(LOCKHEED-BRAKE FRICTION MATERIAL)
B.E.F. ≜ Ft/Fa
μ
∞
5
4
3
2
1.5
1.0
.9
.8
.7
.6
.5
.4
.3
0.2
0.15
0.1
0
0.1
0.2
0.3
0.4
0.5

Appendix E: Analysis of a representative antiservo-action brake mechanism

Ramp antiservo



Equilibrium (disregarding moments):

$$\Sigma F_x = 0; \quad F_a - F_t - F_\phi \sin\phi = 0, \qquad (1)$$

$$\Sigma F_y = 0; \quad F_n - F_\phi \cos\phi = 0. \qquad (2)$$

Elemental Relation: $F_t = \mu \cdot F_n$.

Substituting for F_n in (2), we get

$$F_\phi = F_t/(\mu \cos\phi).$$

Substituting into (1),

$$F_a - F_t - \frac{F_t}{\mu} \cdot \frac{\sin\phi}{\cos\phi} = 0,$$

$$F_a = F_t \quad \frac{\tan\phi}{\mu} + 1$$

Brake effectiveness factor B.E.F. $\triangleq F_t/F_a$,

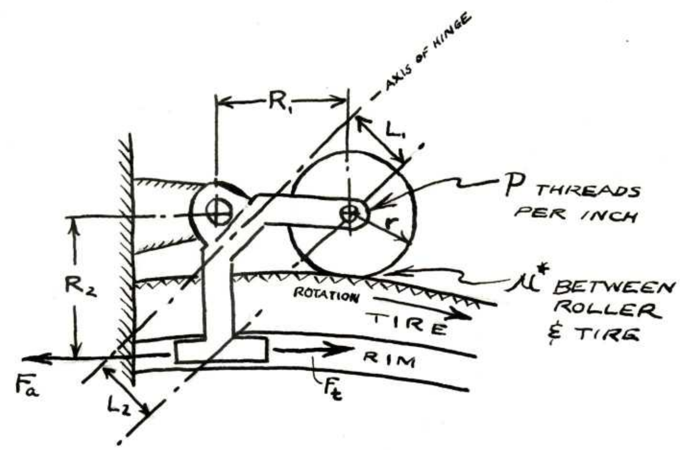
$$\text{B.E.F.} = \frac{1}{\frac{\tan\phi}{\mu} + 1} = \frac{\mu}{\tan\phi + \mu}$$

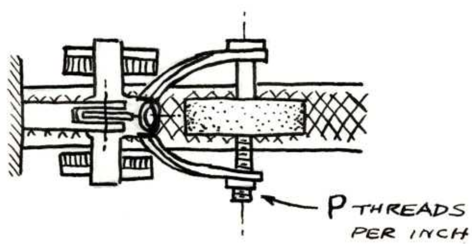
Thus the B.E.F. is of the form $\mu/(C + \mu)$,

where C is the tangent of the ramp angle ϕ.

Appendix E: Analysis of a representative power-assist brake mechanism

Roller and screw





Friction torque on roller: $T = \mu^* r(F_a - F_t)(R_2/R_1)$

Brake normal force is related to screw force F_r by the formula: $F_n = F_r(L_1/L_2)$.

But, from energy considerations, $2\pi T = F_r/p$

so that $F_n = 2\pi\mu^* pr(F_a - F_t)(L_1R_2/L_2R_1) = K(F_a - F_t)$.

Additionally, $F_n = F_t/\mu$, so that $F_t = \mu K(F_a - F_t)$;

$F_t(1 + \mu K) = \mu K F_a$, thus the B.E.F. $\triangleq F_t/F_a = \mu K/(1 + \mu K)$;

B.E.F. $= \mu/(C + \mu)$, where $C \triangleq \frac{1}{K} \triangleq \frac{L_2R_1}{(2\pi\mu^* prL_1R_2)}$.

Appendix E (continued)

Assuming some typical values for:

$\mu^* = 0.5$ (coarse abrasive grit surface against rubber in the wet),

$p = 10$ threads per inch,

$r = 1$ inch,

$L_1 = L_2$ and $R_1 = R_2$,

we obtain a C of 1/10 , or 0.03. This yields a drop in B.E.F. of 20% with a drop in μ from .35 to .10. (A 20% jump in force at a handgrip would hardly be noticed!)

If we require $\frac{1}{4}$ in. clearance on both sides of the rim with the brake fully released, Total free travel of the brake is $\frac{1}{2}$ in., or 5 turns of the roller. This corresponds to $5 \cdot \frac{\text{roller radius}}{\text{wheel radius}} = 5 \cdot \frac{1 \text{ in}}{15 \text{ in}} = 1/3$ revolution of the wheel -- a permissible lag for urban traffic situations.

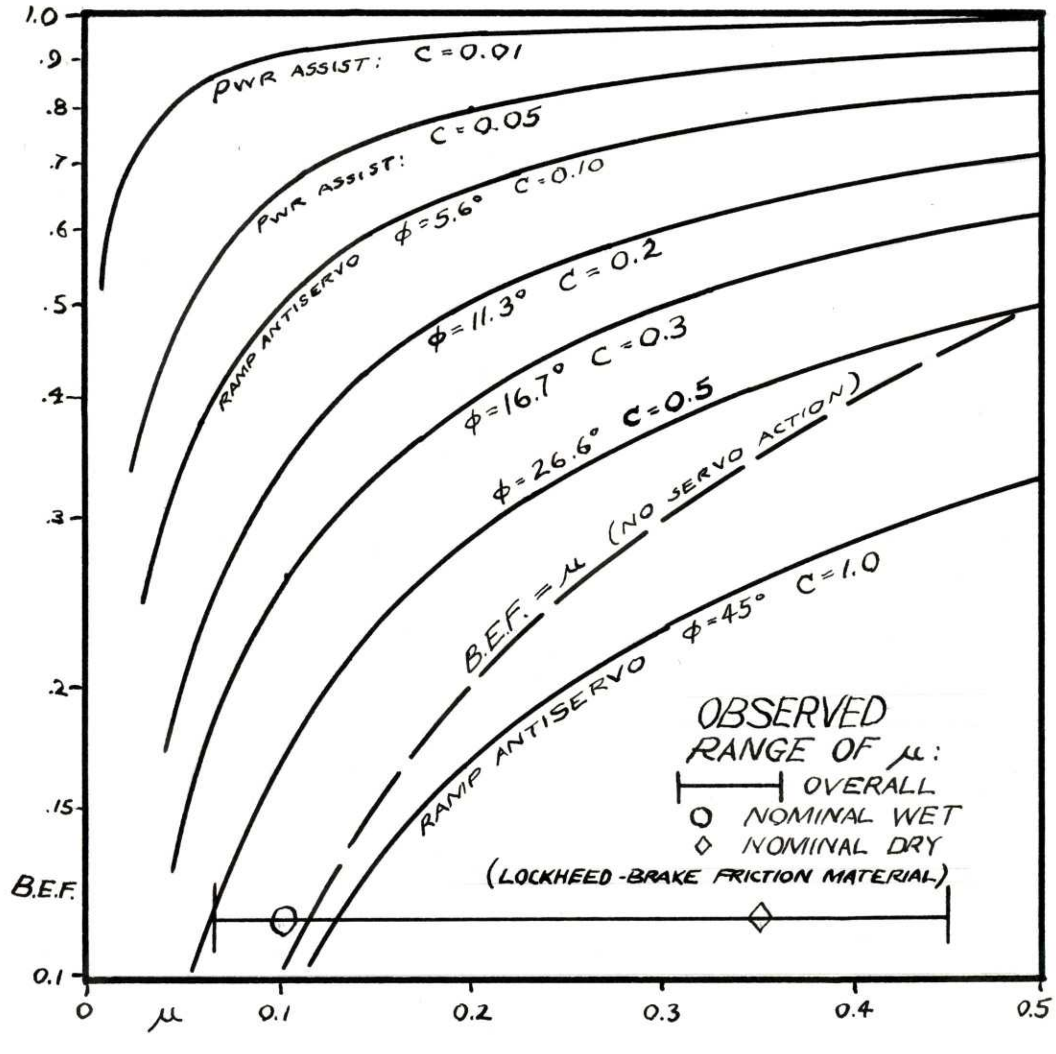


Fig. 8. Power-assist brake mechanism and ramp antiservo mechanism: B.E.F. vs. μ for various ramp angles (antiservo) and values of C (power-assist).