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EXPERIMENTS WITH AIRPLANE BRAKES

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The attempts to check the length of run of an airplane by brakes go back to the earliest days of flying. At one time it was essayed to produce the necessary arresting force by means of a hook fastened to the landing gear which, it was hoped, would bury itself in the ground upon landing. But the experiments with these more or less abortive designs were doomed to failure because of the uncontrollable magnitude of the prevailing braking force, and it is to the credit of American aviation that the problem of brakes has at last been seriously considered in connection with aircraft. Disregarding the prejudice against brakes prevailing in Europe, where their use had always been associated with a tendency for pitching of the airplane on its nose, the United States experimented with accepted automobile brakes for aircraft use, and the results were exceedingly gratifying.

In Germany the question of brakes received a new impetus by the abnormal lengths of run of large airplanes when using tail-skid shoes and wheels which were specially designed for turf protection. The development of German wheel brakes for airplanes is primarily due to the efforts of the Junkers Airplane Company, the Knorr Brake Company, and the Elektron Metal Company.

Whenever the expression "airplane brake" is used in the following, it means braking of the airplane by the ground friction upon landing or rolling. It does not include aerodynamic braking devices which, by changes in air resistance or propeller thrust, produce a deceleration of the airplane prior to touching the ground.

The replacement of the already strongly decelerating tail skid by a special tail wheel, in particular, was but one step removed from equipping this tail wheel with a

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*"Versuche mit Flugzeugbremsen." Zeitschrift für Flugtechnik und Motorluftschiffahrt, May 28, 1931, pp. 302-312, and June 15, 1931, pp. 338-344.

brake. Proposals of this nature have, in fact, been frequently made, but had to be rejected upon closer examination. Efficient braking effect of the tail wheel would, by virtue of the small amount of airplane weight resting on the tail, require such an abnormally high coefficient of ground fiction, that it would hardly be possible not to injure the turf, which was precisely the object of the special tail wheel.

Based upon the experiences gained with automobile brakes, the first attempts consisted in equipping the wheels of small and medium size airplanes with direct-acting mechanical brakes operated by hand or foot. Another positive-acting brake is the much used oleo-pneumatic brake, particularly in foreign countries. To overcome oil leakage the pilot has a brake lever which operates a small hydraulic pump. A few backward and forward movements of this lever take up all the brake clearances, and any further pressure on the lever applies to the brake. Since the transmission may not be raised arbitrarily, while on the other hand low actuating force is desirable, the wheel size for direct-acting brakes is limited to 1020 X 175 mm (40.16 X 6.89 in.) wheels for mechanical brakes, and slightly higher for oleo brakes.

For greater braking energies, as necessitated on large airplanes, the application of indirect-acting brakes is recommended, and this report deals primarily with brakes of this kind. The principal advantage of the indirect-acting brake is that it releases the pilot; he merely engages the brake and sets it to the desired pressure, which calls for no special efforts. A very simple exponent of this type of indirect-acting brake is the compressed air brake, which is therefore specially suited for very high braking forces. The advantages in using compressed air for actuation lie in the ease with which the braking effect may be fitted to a particular airplane, in the sensitiveness of the gradation, the precise limitation of the highest permissible braking pressure, and the suppression of any difficulties when applying the brakes. In addition, the compressed air brake ensures balancing of the energizing forces of the brake shoes of several wheels by prescribed equivalent braking. When the energizing forces are equal no exact balance is necessary, because the different wheels always will show some slight discrepancies in brake effect. But it is absolutely necessary that the energizing forces of the brake shoes balance one another.

The chief disadvantage of this type of brake is that a supply of compressed air or else an air compressor must be carried on the airplane. When the compressed air containers or cylinders are of light metal it is possible to carry 4 liters (244.1 cu.in.) of air at 150 atmospheres, weighing 4 kg (8.82 lb.) - a supply ample for more than 100 braking operations. If the airplane is large enough to carry an auxiliary power plant it is advisable to include a small air compressor which automatically keeps the pressure at a constant minimum. The compressed air cylinder must be large and tight enough to ensure satisfactory braking, especially when needed most, that is, in case of a forced landing, when the engine, and through it, the compressor, stops.

The construction of brake wheels began with wire wheels which were often simply modified to receive a brake drum. There has been a persistent attempt in Germany to abandon the wire wheel for the disk wheel, particularly for the larger sizes where the elektron castings offer special advantages. For several years the use of cast elektron wheels has gained considerably. Recently such wheels were also equipped with brakes. Thus, Figure 1 shows a cast elektron wheel 1300 × 300 mm (51.18 × 11.81 in.) with the brake shoe arrangement of the Elektronmetall company (original type). The brake drum is 500 × 50 mm (19.69 × 1.97 in.); operation is by compressed air at 3 atmospheres. (The shoe arrangement is type I of Figure 3.) Figure 2 shows the same wheel mounted with a brake of the Knorr Brake Company, which operates at 6 atm. (shoe arrangement VI in Figure 3). However, since the use of compressed air for brake actuation still tends to make installation somewhat complicated, it was imperative to make the mounting in the wheel as simple as possible. For that reason no radically new types were resorted to, but the internal expanding types were used which, in order to gain an idea of their effect and treachernousness, are in need of some explanatory notes, particularly as concerns the "servo action."

In the automobile industry the internal expanding type of brake with servo action is an arrangement in which the one brake shoe is assisted by the subsequent one, with the result that the frictional forces of the first brake shoe raises the energizing pressure and through it the braking effect of the second shoe. In principle it pertains, therefore, to the utilization of the friction on brake shoes for creating normal forces and through it new frictional forces. With this as starting point, it is advisable to go into the conception "servo action" a little more in detail. When a small piece of brake lining is pressed at a given constant normal force against a brake drum, a braking torque corresponding to the friction coefficient is produced as soon as the drum rotates. If the coefficient of friction rises, the braking torque, because of the constant normal force, increases precisely proportional to it, or in other words, the frictional force does not influence the normal force; there is no servo action. In the following, we speak of a "positive servo action" when by constant actuating force the retarding effect of a brake by increasing friction coefficient μ of the lining rises stronger than proportionately to μ, and of a "negative servo action" when by increasing μ the rise of the braking torque is less than proportionately to μ.

In order to gain an insight into the method of operation of different types, a series of brake-shoe dispositions (fig. 3) are mathematically investigated:

I Two shoes with positive servo action, separate, but equal actuating force;

II Two shoes, with positive servo action, the first shoe assisting the second;

III Three shoes with positive servo action;

IV Internal expanding brake with angle of contact = 300°;

V One shoe with positive, the other with negative servo action, both shoes separate, but with equal actuating force;

VI One shoe with positive, the other with negative servo action, balanced by cam operation;

VII First and second shoes with positive, and third shoe with negative servo action; first and third shoe separate but with equal actuating force; the first shoe energizes the second;

VIII First and second shoe with positive, the third with negative servo action, partially balanced by cam operation;

IX Two shoes with negative servo action and separate, but equal actuating force.

The difference in the stationary pins from the movable pins is readily noted in Figure 3.

The determination of the servo action of these brake-shoe assemblies for arbitrary operating conditions is hopeless because of the excessive number of variables affecting the transaction. However, it is possible to gain a clear conception of the merits of the divers arrangements for a few limiting cases by very simple calculation. In the comparison the brake drum diameter was kept constant.

We began by examining the forces acting on the brake shoes. For the determination of the load distribution over the shoes it was assumed that the brake linings follow Hooke's law, are neatly fitted and bedded in by wear. The assumption of Hooke's law, that is, the proportionality between compression of the lining and the absorption of force, is fulfilled to a certain extent for the loading, as becomes apparent from the load tests described further on. But there is a material discrepancy at unloading. From the load distribution we merely defined the position of the normal force resultant, while for the rest, the effect of the distribution was disregarded in the comparison of the different shoe dispositions.

TABLE I. Forces on the Brake Shoes (fig. 3)

Assumption 1. Braking torque M, brake drum diameter d, and friction coefficient μ of the brake linings are constant.

2. The fit and the elastic behavior of the linings are so efficient that the load per unit area of the lining corresponds at every point to the energizing path of the brake shoes.

All forces are expressed in per cent of the sum of normal forces N on the shoes, whereby

$$\Sigma N = \frac{M}{\mu \frac{d}{2}}.$$

Coefficient of friction	Arrangement	I	II	III	IV	V	VI	VII	VIII	IX
μ = 0.3	Σ actuating forces	32	12	8	8	40	49	25	26	37
	normal force 1. shoe	50	34	18	-	70	50	28	27	50
	normal force 2. "	50	66	30	-	30	50	47	46	50
	normal force 3. "	-	-	52	-	-	-	25	27	-
	resultant free force	0	34	37	42.5	42.5	0	35	30	0
μ = 0.5	Σ actuating forces	22	6	3.5	3.9	26	49	17	20	78
	normal force 1. shoe	50	24	10	-	84	50	23	22	50
	normal force 2. "	50	76	25	-	16	50	61	56	50
	normal force 3. "	-	-	65	-	-	-	16	22	-
	resultant free force	0	58	62	56	75.5	0	58	54	0

In addition, the same friction coefficient for all shoes and the same braking torque for each brake was assumed, thus making for an ideal condition. The forces obtained for $\mu = 0.5$ as plotted in Figure 5 at an arbitrary scale, are valid for all forces. If the sum of the normal forces, which is the same for all arrangements, equals 100%, and all forces are expressed in per cent of this normal force summation, the result is a comprehensive comparison of the various brake-shoe arrangements with respect to the forces under ideal operating attitude. These values are appended in Table I for $\mu = 0.3$ and $\mu = 0.5$. The discrepancies between the individual shoe arrangements are enormous in part as, for example, a coefficient of friction $\mu = 0.5$ in the three-shoe type (III in fig. 3) yielded only 3.5% of the required actuating force, while in arrangement IX (fig. 3), it rose to 78% as a result of the negative servo action. Another remarkable feature is that arrangements I, VI, and IX possess no "free forces" which must be diverted into the landing gear, for which reason the braking torque in these arrangements is preferably transmitted without free forces by a couple of forces (flange) to the landing gear.

The distribution of the braking effect over the various shoes is readily seen in Table I. As the coefficient of friction decreases the discrepancies due to servo action become smaller.

In the determination of the forces a constant operating attitude for each brake was used as basis. Now, if for any reason the friction coefficient changes while the pilot actuates the brake with constant pressure, that is, while he expects a constant braking effect, a change in braking torque occurs which again is an index of the extent of the servo action of a brake-shoe arrangement.

Figures 4 and 5 show the variation in braking torque plotted against μ for the divers arrangements, with the braking torque for $\mu = 0.3$ (fig. 4) and $\mu = 0.5$ (fig. 5) equal to 100%, that is, corresponding to the accepted braking torque, for which the brake is designed.

In order to bring out the effect of the servo action more clearly the M/μ curves in Figures 6 and 7 were plotted against μ, with precisely the same assumptions as in Figures 4 and 5. The attitude of missing servo action in Figures 6 and 7, is shown by the horizontal $\frac{M}{\mu}$ = constant = 100%. Positive servo action denotes a rise in the curves when μ increases, hence $\frac{d(M/\mu)}{d\mu}$ is positive. By negative servo action, accordingly $\frac{d(M/\mu)}{d\mu}$ is negative (arrangement IX, fig. 3). The brakes with large positive servo action show the greatest fluctuations. For instance, a change in μ from $\mu = 0.5$ to $\mu = 0.6$ in the three-shoe arrangement (III, fig. 3) resulted in an increase of almost 100% in braking torque. (Fig. 5.) As the servo action diminishes, the effect becomes smaller until at arrangement VI, by missing servo action the braking torque varies only proportional to μ, although even in this arrangement the shoes operate separately with servo action. But due to the cam operation the energizing paths of the two shoes are no longer independent of one another. The equilibrium demands equal normal forces along both shoes, with the provision, of course, that the elastic conditions in both shoes are exactly alike, that they wear the same and have the same amount of clearance.

If the initial attitude $M = 100\%$ is at $\mu = 0.3$ (fig. 4), the changes in braking torque by equal variations in friction coefficient (expressed in percentage) are slightly smaller, but in the arrangements with large servo action they always are very pronounced.

As second important disturbance of the ideal operating attitude, ~~we~~ examined the effect of a changed load distribution along the brake lining. Figure 8 shows an arbitrarily shaped shoe whose load distribution along the brake lining is very conducive to ideal operating attitude. The normal force resultant is almost in the center of the brake lining, and the change in load per unit area along the lining is moderate. Now, any disturbance set up in this distribution through uneven wear or clearance along the lining shifts the normal force resultant upward or downward. This may cause a very essential change in braking effect, and constitutes the principal danger in shoe brakes embodying the otherwise excellent servo action. In order to estimate the highest overload which may occur here we determined the magnitude of the normal force resultant for different points on the brake lining and plotted them in the middle position in per cent of the normal force resultant. The diagram denotes a very pronounced rise in normal force resultant and, through it, in braking effect, as soon as the resultant approaches the upper extremity of the brake lining. In order to remedy this, a displacement of the pivot was effected which lowered the peak considerably. On the other hand, it should be noted that shifting the pivot entails a change in load distribution in the case of the above designated ideal operating attitude, and that the here discussed investigation must be made for the possible maximum friction coefficient of the utilized brake lining. The result is a compromise which must be effected in order to ensure a favorable solution.

Moreover, the investigations prove the significance of the snug fit of shoes and lining to the brake drum. In brakes with high servo action especially, the lining should be worn in gradually before attempting to apply the maximum braking pressure. Another possible improvement in brake lining fit, may be accomplished by reducing the wheel hub clearance, say, by using ball bearings. Attempts should be made to equip the wheel hubs with needle bearings and to subject them to a series of tests.

The calculations made for several limiting cases reveal, although far from conclusive information, considerable data as to the merits and demerits of the servo action. To make this information complete, we would have to include the effect of the load distribution and the shoe length, the deformation of shoes and drums due to mechanical and thermal stresses, etc. But this is impossible unless a very detailed arrangement is available for investigation at the same time that actual brake measurements are made. Now the question arises whether or not servo action is at all to be recommended on airplane brakes. If direct-acting brakes are used for high braking pressures, the servo action becomes almost a necessity. As a matter of fact there is no objection to the use of servo action if the dangers are suitably reduced by corresponding design of the brake shoes. One advantage of indirect-acting brakes is that they permit very high braking forces easily even without resorting to servo action. But unfortunately this is at the expense of increased weight, because the actuating forces become considerably higher. One may even go a step further and use indirect-acting brakes with negative servo action, thus approaching the very desirable attitude of minimum effect of friction coefficient from variations. But unfortunately, a limit in still greater extent is encountered, as becomes evident in the air consumption of compressed air brakes.

The braking effect of the landing wheels was estimated from a number of pull-up tests made with a Junkers A 24 (owned by the D.V.L.) at landing and rolling. These measurements were preceded by a series of wear-in experiments of brake linings and of a check on the various instruments. Mr. Biechteler of the D.V.L. flight section was at the controls during all these flight tests, while the readings and the paper work were in the hands of my associates, Grabarse and Schumacher.

The gross weight of 5100 kg (11,243.56 lb.) was constantly maintained for all flights, and no changes in trim were effected. Figure 9 shows the airplane with its respective C.G. position, so as to facilitate the estimate of the possible braking effect (danger of nosing over). The friction coefficients of the various brake linings were not confirmed by laboratory tests. Such measurements are of great value in all problems of brake lining development, and when it is attempted to effect some improvement by relatively simple comparison tests as, for example, in behavior under different temperatures. But the conditions under which the brake linings actually work are difficult to simulate in the laboratory, and the results of such tests are in most cases to be interpreted relatively. In order to obtain some data on the conditions as they actually occur in the wheel brake, even at the risk of not always being able to separate the individual factors affecting the procedure, the flight tests were preferred.

The primary object of these brake tests was to define:

1. The coaction of wheel braking with different forms of tail skids and the obtainable reduction in run;

2. The deceleration of the airplane and its variation during the run;

3. The existing braking torque and the effect of the ground condition on the possible braking effect;

4. The behavior of various brake linings under different operating attitudes.

The simplicity of the test instruments is of primary importance, so that the braking effect may be estimated as soon as one test has been completed. For measuring the actuating forces the air pressure was kept constant after engaging the brake and the pressure was read for each wheel individually on precision pressure gauges mounted in the cockpit, as indicated in Figure 10. The precision gauges c are at the left, the switchboard d for optograph and dynamometer, at the right; a denotes the brake lever of the wheel brake of Figure 1, which is coupled to the rudder, b shows the brake lever of wheel brake according to Figure 2 (temporary installation). The lever serves for two- and for one-side braking.

At the beginning of the brake tests, it was attempted to measure the decelerations direct during rolling; but since they are comparatively small, and in addition, since much greater ones may be set up perpendicular to the direction of rolling due to landing shocks, which make the determination of the component in the direction of rolling more difficult, this method was soon abandoned for path-time diagrams even though they are not quite as accurate.

Among the equipment shown in Figure 11, is a light test wheel with electric lights, and which during rolling is pressed against the ground by rubber cord. A D.V.L. optograph is mounted below the third cabin window, in addition to a film camera set for the top edge of the test wheel rotation. For each wheel rotation the lights on the test wheel pass once through the focus of the optograph which records the test-wheel revolutions with respect to the time interval.

Figure 12 is an optograph record of a landing without brakes with 144 test-wheel revolutions from the time the landing wheels touched the ground to pull-up. Figure 13 shows the corresponding optograph record for a landing with full braking effect with 59.5 test-wheel revolutions and consequently, 42% shorter run than in the landing without brakes.

Aside from the deceleration as defined by the path-time curves, it was held desirable to measure the braking torque direct on the wheel by means of a dynamometer mounted in the transmitting rods of the braking torque and observed by means of a mirror from the cockpit. Subsequently this was modified and the record mechanically plotted with respect to time interval.

Figure 14 shows the dynamometer with recording mechanism which prevents the brake disk from turning with the landing wheel. The pointer of the dynamometer is replaced by a thin paper-covered disk which executes rotary deflections corresponding to the forces introduced. A recording pen, actuated by an electric motor, moves radially and at constant speed over the paper.

The rolling and landing tests were partly executed on the Berlin Adlershof flying field and partly at the Berlin Tempelhof airport. As far as possible the tests included the effect of the conditions of the ground and of the weather on the braking effect, which were made on grass-covered fields and on concrete runways.

At the beginning of the brake tests the tail skid was fitted with the conventional shoe. But since this type has been ruled out as unsatisfactory, the remaining experiments were made with turf-protecting tail-skid shoes or tail wheels, as shown in Figures 15 and 16, both of which were steerable. The wheel is of elektron and fitted with a solid rubber tire. These two types of tail skids and a wheel brake conformal to Figure 1 were used to make a series of comparative landings with the primary object of defining the taxying run and the rate of deceleration of the airplane. The reading of the braking torque served to get a rough estimate of the average occurring in braking. (The automatic recording dynamometer in Figure 14 was not used until later.) This test series also included experiments with tail wheel and wheel brake conformably to Figure 2. So as not to cause too much delay the experiments were not restricted to measurements in still air or at a constant wind velocity. In fact, it was desirable to test them in bad weather and in a strong wind also. The results of these tests are shown in Tables II and III.

TABLE II. Landing Tests with Tail Skid and Tail Wheel
(Wheel brake as shown in Figure 1)

With tail skid						
Landing No.	Kind of braking	Brake pressure atm.	Wind velocity m/s	Time of rolling[1] s	Landing speed[2] m/s	Pull-up m
11	Without brake	-	4.4	26	21.5	253
39	"	-	5.5	35	22.5	322
61	"	-	0.5	38.5	23.0	426
62	"	-	1.2	33.5	27.0	410
8	With brake	1	(5.5)[3]	11	21.0	128
12	"	1	4.3	14	20.6	173
17	"	1	(8)	12	20.1	133
9	"	2	(6.8)	10	20.4	112
13	"	2	3.8	10.3	25.0	139
18	"	2	(8)	10.5	21.3	136
10	With brake	3	5.5	10	20.0	127
14	"	3	3.0	11	24.6	149
19	"	3	(8)	9.5	22.9	136
63	"	3	0	15.5	26.6	237
64	"	3	0	13.0	26.8	174

(m X 3.28083 = ft.)

[1]Determined by stoppage of airplane independent of optograph record.

[2]Speed over ground.

[3]The bracketed figures are averages in gusty wind.

TABLE II (Cont.)

Landing Tests with Tail Skid and Tail Wheel
(Wheel brake as shown in Figure 1)

With tail wheel						
Landing No.	Kind of braking	Brake pressure atm.	Wind velocity m/s	Time of rolling[1] s	Landing speed[2] m/s	Pull-up m
20	Without brake	-	2.8	36.5	24.0	250
27	"	-	(5.5)	30	22.0	280
31	"	-	3.8	36	22.0	299
35	"	-	3.8	41	20.2	318
68	"	-	0.5	45.5	24.5	460
79	"	-	2.8	42	24	415
28	With brake	1	(5.5)	15.5	24.2	179
32	"	1	3.8	14	21.0	163
36	"	1	3.8	14	25.4	149
29	With brake	2	(5.5)	11	21.0	140
33	"	2	3.8	12	21.3	128
37	"	2	3.8	12	22.7	136
34	With brake	2.5	3.8	12	22.8	139
38	"	3.0	3.8	10	24.2	125
65	"	2.9	1.0	13.5	29	179
66	"	3.0	0.7	12.5	25.1	190
67	"	3.0	2.0	12.5	24.2	153

[1]Determined by stoppage of airplane independent of optograph record.

[2]Speed over ground.

(m × 3.28083 = ft.)

TABLE III. Landing Experiments with Type of Brake Shown in Figure 2

With tail wheel					
Landing No.	Braking pressure atm.	Wind velocity m/s	Time of rolling s	Landing speed m/s	Length of run m
82	2	2 to 3	17	25.8	194
83	5.6	2 " 3	10	21.5	123
87	4 to 6[1]	5 " 6	10	25.0	116
88	4 " 6	5 " 6	9	22.2	115
90	5 " 6	4.4	8.5	25.0	116
91	4	4.7	10	23.0	112

[1]Brake pressure applied progressively.

(m × 3.28083 = ft.)

Figures 17-21 show the path-time curves of the landings at different degrees of braking effect. Because of the uneven landing speeds (Compare Tables II and III) and of the effect of the wind, the runs, and particularly those after landings without brakes are scattered considerably. To ensure a better comparison for estimating the braking effect the measured pull-up curves were again tabulated in Figures 19 and 20, but beginning at 17 m/s (55.77 ft./sec.) (61.2 km/h = 58.03 mi./hr.) rolling speed. This reduces the scattering considerably in spite of the fact that the run without brakes but with tail wheel is still very much affected by wind, condition of ground, etc. The run of the airplane without brakes and with tail wheel was 10 to 20% greater in still air than when the special tail-skid shoe was used. These figures are valid for grass-covered fields. An airplane with idling engines and equipped with tail wheel is at times impossible to stop in a slight tail wind, whereas the tail skid still creates enough frictional resistance for braking. The use of tail wheels makes absolute dependability of the brake imperative.

A comparison of the different measurements yields the figures for shortened runs by wheel braking, as appended in Table IV.

TABLE IV. Runs Shortened by Wheel Brake

	Total length of run (m)			Run, beginning at v = m/s	
Run with	tail skid shoe	tail skid	tail wheel	tail skid	tail wheel
Without brakes %	100	100	100	100	100
With brakes %	40	35	30	30	20

(m × 3.28083 = ft.)

These figures are obtainable without noting any tendency of the airplane to nose over in ordinary landing terrain. From the figures, one requirement may be deduced, namely, that a good brake should ensure a shorter run by at least 60%. Another important result, according to these curves is, that even a moderate braking effect shortens the landing run considerably, whereas the application of greater brake pressures does not result in conformably shorter runs. The reason for this is that during the first few seconds of landing the brake is either not yet energized or, if previously engaged, the lack of evenly strong ground pressure of the airplane prevents it from becoming fully effective. Because of the high speed of the airplane during these first few seconds, the rolling distance not utilized for brake action is considerable and practically unaffected by the amount of applied brake pressure.

In principle the brake should not be engaged before the airplane touches the ground. Several kinematographically recorded landings with a different airplane of the same type and mounted with the wheel brake of Figure 2 are shown in Figure 22. They were made with "blocked" wheels at the Tempelhof airport and evince only slight discrepancies from the other pull-up measurements, more particularly as the airplane was fitted with a conventional tail skid. Thus engaging the brake prior to setting down is of no particular advantage. In most cases it merely gives an airplane a greater tendency to bouncing and roll erratically.

This brings us to the deceleration of the airplane and its variations during the run. The deceleration records of the landing (by optograph) are plotted in Figures 22 to 24. The slowing up began as the airplane touched the ground with a speed of about 1 m/s². In the landings without brakes but with tail skid, it then dropped to about 0.5-0.6 m/s² (1.64-1.97 ft./sec.²), and to about 0.3-0.4 m/s² (.984-1.312 ft./sec.²) when a tail wheel with solid rubber tire was used.

By small braking effect the deceleration remained approximately constant after the rise of engaging; but full braking effect was in most cases accompanied by a maximum deceleration in about the middle of the time of rolling followed by a drop toward the end of the run. The highest deceleration according to our measurements, amounted to 4 m/s² (13.12 ft./sec.²) in normal landings, and about 5 m/s² (16.4 ft./sec.²) for the landings with blocked wheels. The drop in deceleration after exceeding a maximum is due in part to the drop in air resistance, and very likely due to a decrease in coefficients of friction of the brake linings during the run. At any rate, a pronounced drop on the dynamometer pointer toward the end of the run could be noted. The first experiments were made with Ferodo fiber brake linings, which have a very high coefficient of friction, but are also more susceptible to higher temperatures. As a result of the above experiences, various other kinds of brake linings were selected for the other tests.

By normal braking the highest deceleration figures occurred at rolling speeds ranging between 70 to 85% of the landing speed; that is, speeds accompanied by pronounced landing gear stresses in bad landings.

In a subsequent series of rolling and landing tests with a wheel brake according to Figure 1, we defined the braking torque from the dynamometer records, one of which is reproduced in Figure 25. The angular displacement corresponds to the force in the rods transmitting the braking torque, while the distance of the curve point from the center represents the time of rolling. The braking torque is plotted against the time of rolling in Figures 26 to 28. The braking torques with wheel brake No. 2 were defined in the same way but, being very similar to the others, were not reproduced. The fixed values pertaining to the braking torque curves may be obtained from Table V, which shows that, save for a few exceptions, the braking pressure could be kept exactly constant after the brake was engaged, so that any variation in torque is to be ascribed to fluctuations in braking effect in the individual brake shoes. In general, but particularly at low braking pressures, the torque was constant to a greater extent than the experiences gained from the deceleration measurements had led us to believe; a proof that the employed linings were less susceptible to temperature effects than the Ferodo fiber brake lining. At the same time, the decelerations showed a more uniform to constant behavior. The runs, beginning at v = 17 m/s (55.77 ft./sec.) rolling speed, by full braking effect again ranged between 50-70 m (164.04-229.66 ft.) (Compare Figures 19 and 20), so that the determination of run and deceleration could be forgone in the subsequent experiments.

One remarkable feature was the disturbed behavior in braking torque of all linings under maximum braking pressure. At any rate the variations are still within admissible bounds. This result is of particular significance because of the selection of brake shoe arrangement 1 (fig. 3) which, according to previous experiments, revealed an appreciable servo action, that is, a weakness to variations in the coefficient of friction of the brake linings.

In Figure 29 the braking torque is plotted against the braking pressure, that is, the energizing pressure of the brake shoes for various brake linings. The scattered range differs but little for the different linings. One interesting feature is that the variations during a landing are still completely superposed by the scattering of the values in the different landings in which the same braking pressure is applied. According to the curves greater braking effect is still possible by raising the braking pressure, for there is no sign of approaching a maximum. The discrepancies in the different linings is discussed in detail in a subsequent paragraph.

The maximum braking torque of the 1300 × 300 mm (51.18 × 11.81 in.) wheel amounted to about 500 mkg (3616.49 ft.-lb.) at pull-up, according to Figures 26-28. And the question now is pertinent whether this braking torque is ample for the size of the wheel, because the experiments were made with a lower airplane gross weight than the permissible loading of the airplane wheel warranted. Thus it becomes incumbent to examine somewhat more closely the effect of the surface of the ground on the possible braking effect.

The coefficient of friction of the tire on the ground may be determined approximately, since the impact factor of the landing wheel loading and the distribution of the gross weight over landing gear and tail skid were not measured at the same time. If the airplane is disturbed during rolling, a shock produces an increase in the coefficient of friction necessary to set up the constant braking force, till at last the sliding limit is reached and the wheel is blocked. This was especially noticeable on the wheel track when the ground was frozen. Landing on a soft layer of snow, our grass-covered ground showed the coefficient of friction to be very small. Even a moderate braking effect made the airplane slide forward with blocked wheels for a distance of 80 m (26.2 ft.) without rooting up the soil. To reduce the danger of sliding it is recommended to modify the tires and to use a simple antiskid tread. Attempts in this direction have already been made by several tire manufacturers, but unfortunately no acceptable comparative test data has been published as yet. Likewise it might perhaps be timely to modify the inside construction of the tire to conform to the higher stresses through the supplementary braking forces.

On the other hand, the introduction of wheel brakes makes certain exigencies on the quality of the ground. In several landings the sod failed to withstand the stress, but separated from the sandy subsoil beneath it and was simply pushed aside by the blocked wheel. As a result the wheel dug into the sandy soil, the sod in front of the wheel kept piling up till at last the airplane began to nose over, even though the brake was released. Fortunately, this occurred at the end of the run, so that the airplane did not nose over, however. The plausible reason for this soil destruction is that no sufficiently long effective braking force was available save at low rolling speed. For that reason the sod should be well rooted in the subsoil. On the other hand, the profitable ground friction of an airplane running on sod is indubitably lower than of automobiles on dry streets, where coefficients of friction up to $f = 0.75$ are no exception.

In order to get a general idea, let the mean ground friction coefficient with respect to dead wheel load be computed for a maximum braking torque of 500 mkg (3616.5 ft.-lb.) (Compare fig. 9), with the airplane with tail wheel as basis, predicated on the assumption of absolutely level ground and an attitude of rolling toward the end of the run without any rotation of the airplane around a lateral axis.

TABLE V. Braking Torque for Different Brake Linings
(Wheel brake as in fig. 1)

Brake lining	Ground	Test No.	Braking pressure atm.	Braking torque* mkg maximum	mean	minimum
Ferodo Asbestos	concrete	153	1.2 to 1.4	126	100	90
		152	1.9	200	170	140
		145	2.7	300	250	200
		146	4.0	422	405	390
	sod	154	1.0	90	85	80
		155	1.7	180	160	160
		148	2.0	240	200	170
		147	2.8	300	295	270
		151	2.8	270	270	270
		156	2.8	290	265	240
		157	3.9	415	385	340
		150	4.0	435	390	320
Bremskerl	concrete	117	2.2	185	175	160
		114	3.0	370	330	280
		115	3.0	340	335	325
		118	3.9	400	370	330
	sod	122	0.8	130	110	100
		123	1.5	215	200	200
		121	2.0	270	245	235
		120	3.0	310	295	280
		116	3.1	315	295	290
		119	3.8	450	420	400
Jurid Hydraulic	concrete	110	1 to 1.3	160	125	110
		109	2 to 2.2	230	210	200
		106	3.0	260	255	250
		108	3.0	320	300	275
		111	3.0	320	295	290
		107	3.1	280	270	260
	sod	101	1.0	125	112	100
		105	1 to 1.2	155	140	100
		98	1.7	245	230	210
		100	2.2	350	325	310
		99	2.6	405	395	380
		102	(2.5) to 3	445	425	400
		103	(2.5) to 3	500	460	435
		112	3.0	425	410	400

*Since the maximum and minimum figures occurred only for very short periods, we included averages, which really decide the amount of braking during the whole run.

(mkg X 7.23298 = ft.-lb.)

An effective rolling radius of the airplane wheels of about 0.6 (1.97 ft.) yields for the whole airplane a braking force on the ground of

$$P_B = \frac{2 \times 500}{0.6} = 1670 \text{ kg } (3681.72 \text{ lb.})$$

The resultant of this braking force and the load on the wheel yields a total wheel force which approaches the C.G. of the airplane and thereby relieves the tail-skid pressure considerably. The unloading already amounts to 62% of the available tail-skid pressure for a skid pressure of

$$P_S = 550 - 340 = 210 \text{ kg } (462.97 \text{ lb.}),$$

for an ideal landing in this case. The load on both airplane wheels is 4890 kg (10,780.59 lb.) by a utilized ground friction coefficient of

$$f = \frac{P_B}{P_R} = \frac{1670}{4890} = 0.342$$

To facilitate conversion to other airplane wheels, we include the ground friction coefficient reduced for full wheel load (tail skid raised) and full wheel radius:

$$f = 0.342 \times \frac{4890}{5100} \times \frac{0.6}{0.65} = \sim 0.3.$$

This figure may serve as index in dimensioning suitable brakes for airplane landing wheels, just as the above figure for the obtained tail skid unloading represents by more than 60% an empirical value for the suitable location of the landing gear ahead of the center of gravity. Since the total wheel load, for which the 1300 × 300 mm (51.18 × 11.81 in.) wheel - 500 × 50 mm (19.69 × 1.97 in.) brake drum - was designed, is 3250 kg (7165.02 lb.), it ensures for this wheel a maximum braking torque of

$$M = 640 \text{ mkg } (4629.11 \text{ ft.-lb.}),$$

which is ample for effectively breaking the full gross weight of the airplane. If the landing wheel is intended for higher gross weights, a larger brake drum is recommended.

Taking advantage of the test apparatus used for the determination of the braking torque, we extended the brake experiments to include various brake linings. We used one brand of soft brake band material and two kinds of hard-pressed linings. Each firm furnished the brake shoes with the lining installed. We examined

a) Ferodo asbestos of the German Ferodo company, Töpken & Co., Berlin-Mariendorf;

b) Bremskerl, of the North German Brake Band Co., Nienburg-Weser;

c) Jurid hydraulic, of the Kirchbach company, Kirchbach & Co., Coswig-Dresden.

The friction coefficients obtained with the different kinds of brake linings being of greatest interest, we began with an interpretation of the braking torque measurements in order to define μ. This presumes a certain operating attitude of the shoe brake to make the calculation of μ at all possible when servo action is included.

We used the ideal operating attitude as defined on pages 3 and 4 of this report, as basis. In order to simulate it as nearly as possible, all brake linings were repeatedly braked in rolling before the actual measurements took place. In addition, a check was kept on the wearing qualities during the tests. Strictly speaking, the computed friction coefficients are valid only for those brake-shoe arrangements which were used in the measurements. Lastly, the designer of brakes is not so much interested in the friction coefficient arrived at in a laboratory under special conditions, but rather in that friction coefficient on which he must base his calculation of a certain given brake shoe in order to arrive at the actually produced braking torque.

The brake drum was of cast iron and stood up well during these experiments. Its disposition in the elektron wheel casting is very rigid as became evident in the numerous strength tests with different elektron landing wheel types. The rate of slippage on the linings amounted to about 8 m/s (26.2 ft./sec.) under the first full braking effect.

Since braking periods, braking torques, and cooling conditions are of the same order of magnitude in all experiments with equal braking pressure, it was not attempted to examine the temperature effect separately. Satisfactory comparison is obtained by simply plotting the coefficients of friction of the brake linings against the surface pressure. Figures 30 to 33 reveal the computed coefficients μ based upon the maximum and minimum figures of the braking torques (Compare Table V). It should be noted here that the surface pressure, as a result of the servo action in the brake-shoe arrangement aside from the set braking pressure, is still dependent on the friction coefficient itself, so that corresponding maximum and minimum values of μ pertaining to braking with constant pressure are oblique to one another in the scattered field of the curves, as becomes readily apparent in the diagrams.

The comparison of the individual linings yields the following picture: all linings reveal, as already seen from the braking torques, smaller variations in μ within one landing than the total scattering zone of different landings with equal braking pressure. The soft Ferodo asbestos showed the least variation in the μ coefficients, the average was $\mu = 0.37$ and was unaffected by the surface pressure. The Bremskerl lining showed about the same amount of scattering as the Ferodo asbestos but a decrease in μ as the surface pressure increased. The average was about $\mu = 0.5$ by small surface pressure and $\mu = 0.38$ by full pressure. The highest μ by full braking pressures were obtained with the Jurid hydraulic lining, whose scattered zone was likewise somewhat smaller than for the other brands. One peculiar feature was the difference between braking on sod and braking on concrete. Whereas the coefficients of friction for Ferodo and Bremskerl, according to tests on concrete, still fall within the zone of the values measured on sod, the corresponding μ for Jurid hydraulic are materially lower and wholly outside of the scattering zone (Compare figure 32). Contrary to expectation, the braking effect when rolling on concrete was less, although the airplane shakes less than when rolling on sod. The mean μ for braking on sod amounted to $\mu = 0.48$ for the Jurid hydraulic, and was practically unaffected by the amount of surface pressure. In the other two brands the coefficient of friction was therefore higher than the laboratory tests foretold. High μ coefficients are desirable to avoid high surface pressures.

In order to confirm the effect of oily brake linings on the deterioration of the friction coefficient, we made several other measurements with oiled linings. To produce the worst possible condition, the shoes were removed and soaked in Voltol F motor oil; 40 hours for Ferodo asbestos and Jurid hydraulic, and 15 hours for Bremskerl. Then the shoes were replaced and the tests continued. The results were as follows: In the Ferodo brand the braking effect was practically gone; at least, it was so small that the airplane could no longer be maneuvered by using the brake for steering. Jurid was slightly better at first, but after a few brakings, became just as bad as the first. Bremskerl still showed some considerable braking effect, so that we made various measurements on it, after which its shoes were again removed and again soaked in oil for 40 hours. The results were the same as before.

Figure 33 depicts the coefficients of friction computed from the measurements with respect to the surface pressure. With the exception of a very few points the oiled linings still show satisfactory μ coefficients, even though the scattering is slightly greater. The average is $\mu = 0.33$.

Undoubtedly the hardness of the brake lining has some effect on its behavior in internal expanding brakes with servo action. To gain some insight into the elastic behavior of these three brands of brake lining, a number of compression tests were made with a set of new linings. (See fig. 34.) The loads were applied in stages of 50 kg/cm^2 (711.18 lb./sq.in.) surface pressure, each stage consuming about 20 seconds. (Compare fig. 35.) This test revealed a pronounced deviation from Hooke's law in all three linings for the first and second stages of loading, accompanied by the remarkably great hardness of Jurid and the high energy absorption in the softer linings. The actual case of braking is more nearly simulated when loading in stages with subsequent unloading to zero, as appended in Figure 36. Here the rate of loading was 20 kg/cm^2 (284.47 lb./sq.in.) surface pressure, and the proportionality between work absorbed and compression is much better expressed, although the unloading curve still shows a pronounced curvature. The range of compressibility for repeated loading is seen in Figure 37. According to it the Bremskerl lining uses the highest load factor before a balanced attitude is obtained.

The appearance of all brake linings was good during the tests. There was no sign of local thermal overstressing. Figures 38 to 40 show the used linings after completing the experiments. The piece broken off the corner of the Jurid lining attests to its hardness. The break was most likely caused by the adjacent rivet hole. Ordinarily, Jurid is furnished with hydraulically drilled holes, while in these particular shoes the linings were attached afterward. The wear was slight, and since the number of brakings in an airplane are so much less than in automobiles, the conventional brake linings should present no difficulties in this respect.

In connection with these experiments it might be of interest to point out, in brief, the fundamental difference between an airplane and an automobile brake. In comparison to the importance of brakes for automobiles and to the frequency of use their significance for airplanes is decidedly less. Yet the requirement of minimum structural weight of the airplane brake is much more stressed than in automobiles, because the whole landing gear presents nothing but ballast in flight. Moreover, the airplane brake is used for steering and maneuvering on the ground, and in normal flight operations the use of one or the other wheel brakes for steering is much more important than braking after landing. It is only in an emergency landing that the wheel brake fulfills its real purpose. Since there is no occasion for using a brake when rolling backward, the designer of airplane brakes, - of servo brakes, for instance - is decidedly less restricted than the designer of automobile brakes. Whereas the automobile brake undergoes its highest stresses as speed brake on down-hill grades, the airplane brake assumes a special importance as locking brake, when slowing down the engine at high static propeller thrusts and where perhaps greater braking forces are required than the admissible retardation during rolling.

If we compare the obtained braking effect with the figures encountered in automobile practice, we find for automobile brakes and pneumatic tires, the following figures for braking, beginning at 60 km/h (37.28 mi./hr.) speed (Schenck, "Höchstwerte der Fahrbahnreibung." Automobiltech. Z., Vol. 33, No. 3, 1930).

	Admissible figures	Averages from 4 wheel brake tests	Top figures
Braking distance s m	27.1	22.6	15
Coefficient of ground friction f	0.52	0.63	0.945
Deceleration b m/s^2	5.1	6.2	9.25

(m X 3.28083 = ft.)

From the preceding experiments with airplane brakes the figures are (that part of the run, beginning at 60 km/h rolling speed):

	Averages	Top figures
Braking distance s m	55	40
Coefficient of ground friction f	0.26	0.355
Deceleration b m/s^2	2.55	3.5

The figures are, in conformity with automobile practice, arrived at by assuming the braking force on the wheel periphery as constant and the absence of any other resistance during the run. These presumptions are less admissible for an airplane (air resistance, for example). The already cited top figure for coefficient of ground friction $f = 0.342$, computed directly from the braking torque measurement, is therefore below the above figure, while the absolute top figure of the deceleration for normal landing with 4.0 m/s^2 (13.12 ft./sec.^{2}) is higher than conforms to the shortest "braking distance."

As in automobile design, so also, but in greater degree, the amount of attainable braking effect is a question of ground condition, to which must be added the danger of nosing over in smaller airplanes. So any material raise in the above figures is not likely to occur. Moreover, there is no such great need for it as in automobiles where a matter of a few feet may become very vital, indeed.

In computing automobile brakes the contracting firms of brake linings usually furnish friction coefficients as standards which are even below the averages so as to ensure that the braking effect, for which the brake was built, is really obtained. In contrast to this, it is interesting to know the possible peak in friction coefficients for airplane brakes, so that the braking effect, principally in servo brakes with regard to nosing over and landing gear stresses, does not increase abnormally high while the pilot expects normal braking effect.

As concerns the thermal stresses, the braking periods are much more favorable for airplane brakes than for automobile brakes. Full brake application after landing always begins with a cold brake drum.

To illustrate the conditions, let us compute the amount of energy to be decelerated per 1 cm² brake lining in one second brake period. With q = surface pressure, v = rate of slippage, and μ = coefficient of friction of the brake lining, the energy amounts to

$$A = q \, \mu \, v \; \frac{m\ kg}{cm^2\ s} .$$

For about $v = 8$ m/s maximum rate of slippage at the beginning of braking, the top figures for A are:

	Ferodo asbestos	Bremskerl	Jurid hydraulic
μ	0.4	0.42	0.5
q kg/cm²	12.2	12.0	11.3
$A \frac{m\ kg}{cm^2\ s}$	39	40.3	45

(kg × 2.20462 = lb.) (cm ×.155=sq.in.) (mkg×7.23298=ft.-lb.)

For the cooling conditions existing in automobiles, a specific friction of from 25 to 35 mkg/cm² s is considered admissible,* which, however, is frequently exceeded with low pressure wheels. If we reflect that the cooling in the airplane is perhaps somewhat better, but that, on the other hand, the above figures will perhaps be still higher with a gross weight of 6500 kg (14330.03 lb.) and the same rate of deceleration, it becomes apparent that the stated limit has perhaps been already reached. Improvements will entail smaller surface pressures and subsequently greater braking surfaces, such as has already been applied to this same wheel for 8.5 ton gross weight.

The primary purpose of the experiments on airplane brakes was to acquaint the brake designer with the special conditions under which airplane brakes have to work, and to give the air-

*Kollinck, "Mehr Bremsfläche." Der Motorwagen, Vol. 31, 1928, No. 18, p. 399.

plane designer some points on the effect of brakes on the construction and characteristics of the airplane as a whole. Moreover, it was to show the airplane owner what the requirements of a good airplane brake should be. Our aim herein was to encourage the use of tail skids or tail wheels with landing wheel brakes and thereby protect the landing fields of airports and at the same time increase the safety of commercial aircraft in forced landings. The most recent tendencies to improve the elastic properties of landing gears by using very low air pressure in tires will eventually result in much smaller wheels. This has been carried so far in low pressure tires* with internal pressures of from 0.5 to 1.0 atmosphere, that the wheel body is merely a thick hub. This means greater difficulties in ensuring sufficient braking area and heat dissipation.

Lastly, these experiments were not intended to furnish conclusive estimates as to the merits and demerits of any definite type of internal expanding shoe brake or brake lining. That would entail decidedly more research than the scope of any study on wheel brakes and their application to aircraft embraces.

Translation by J. Vanier,
National Advisory Committee
for Aeronautics.

*While this report was being printed the static test section of the D.V.L. started experiments with American Goodyear low-pressure airplane wheels and brakes.

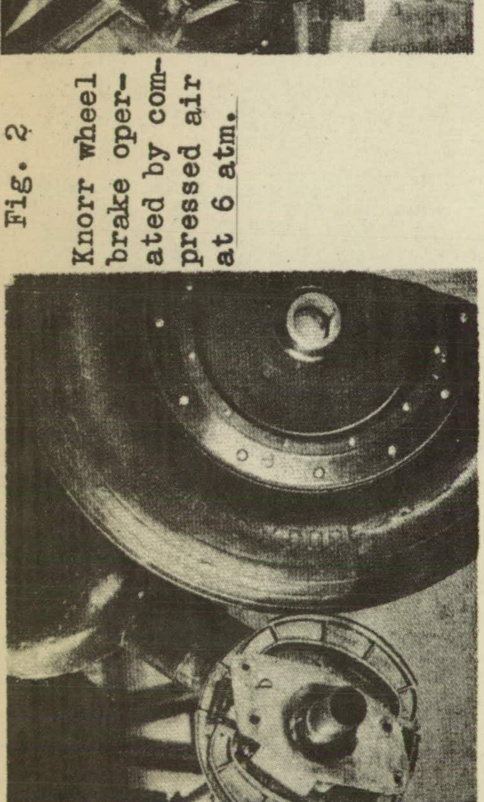


Fig.1

1300 x 300 mm (51.18 x 11.81 in.) wheel brake actuated by compressed air at 3 atm.

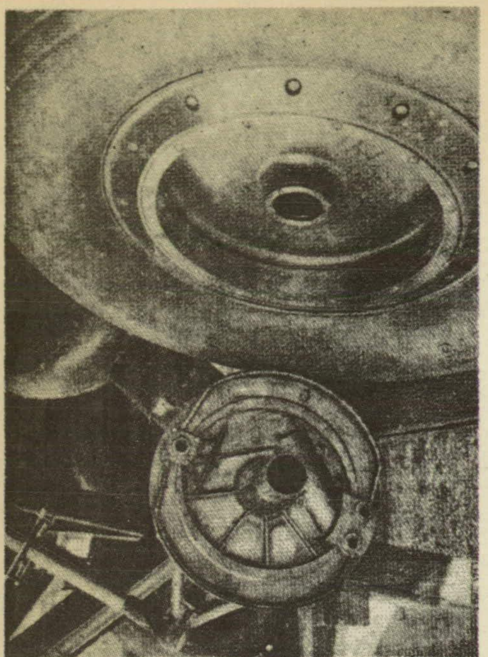


Fig. 2

Knorr wheel brake operated by compressed air at 6 atm.

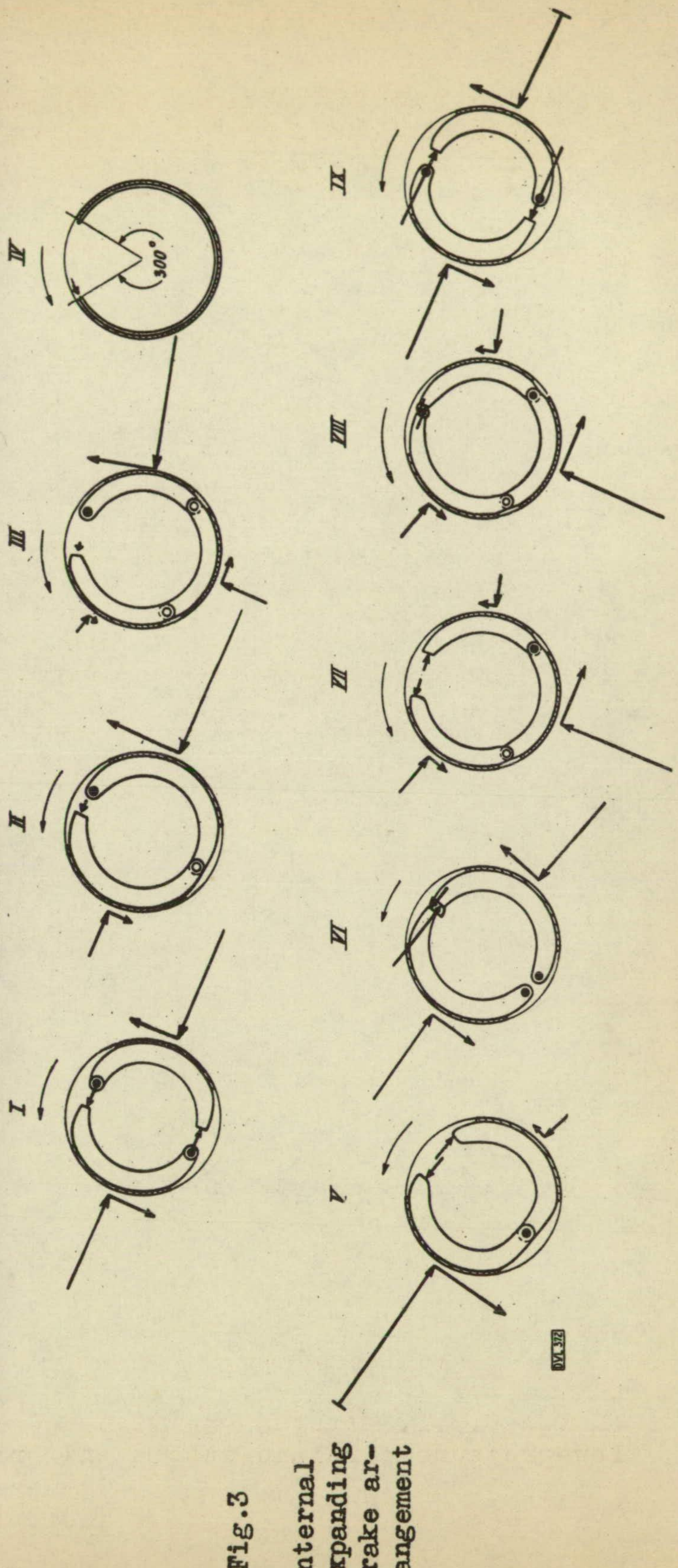


Fig.3

Internal expanding brake arrangement

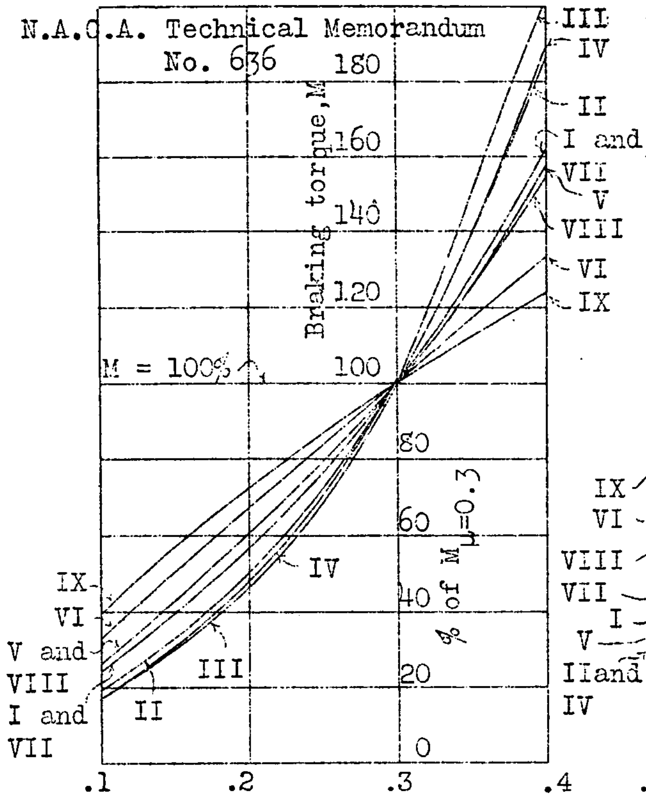


Fig. 4 Braking torque M plotted against coefficient of friction μ for the different shoe arrangements in Fig. 3. The braking torque for μ=0.3 is equal to 100% for all arrangements.

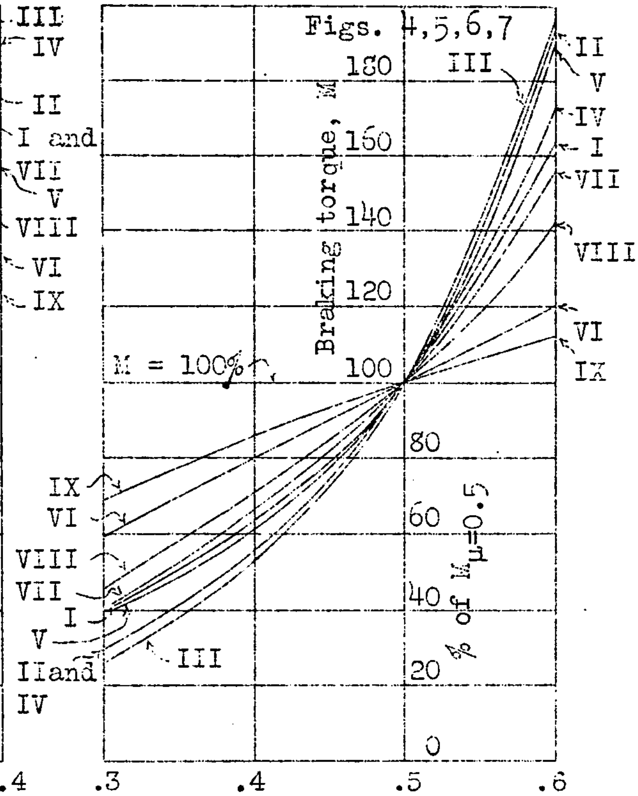


Fig. 5 Same as Fig. 4 except μ=0.5

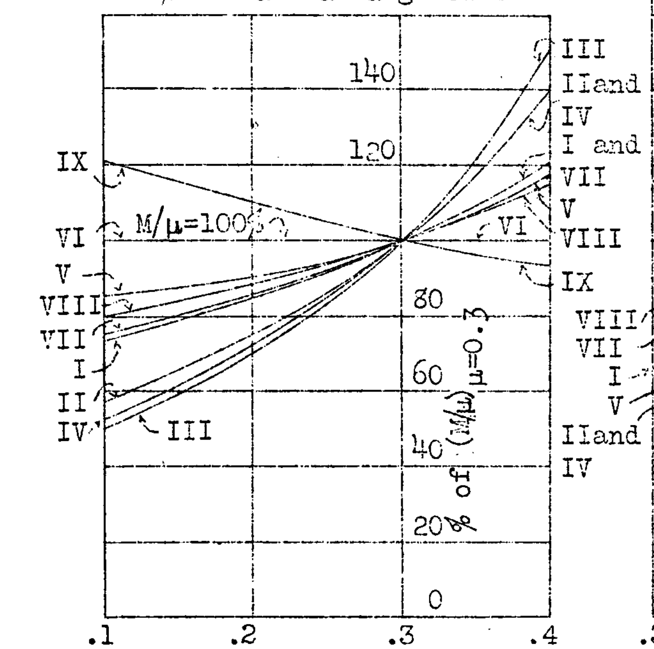


Fig. 6 Effect of μ on M/μ; M/μ=100% for μ=0.3 100%

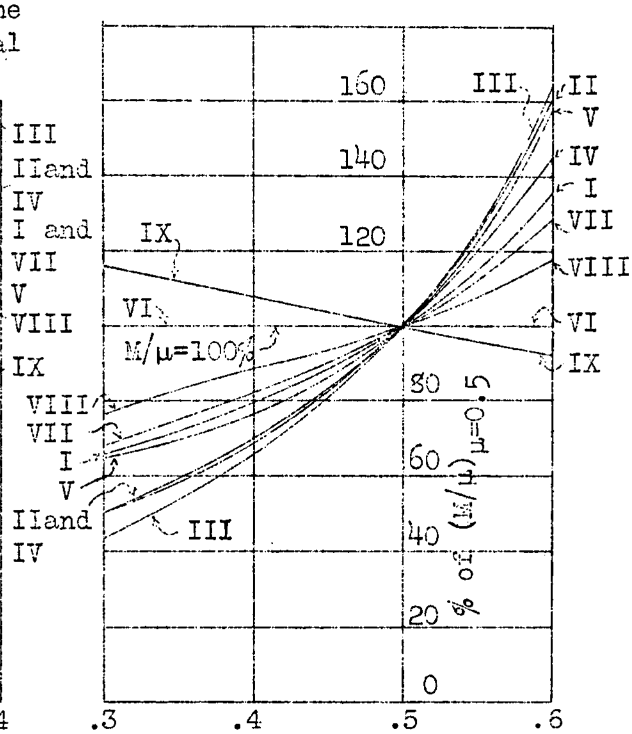


Fig. 7 Effect of μ on M/μ; M/μ=100% for μ=0.5 100%

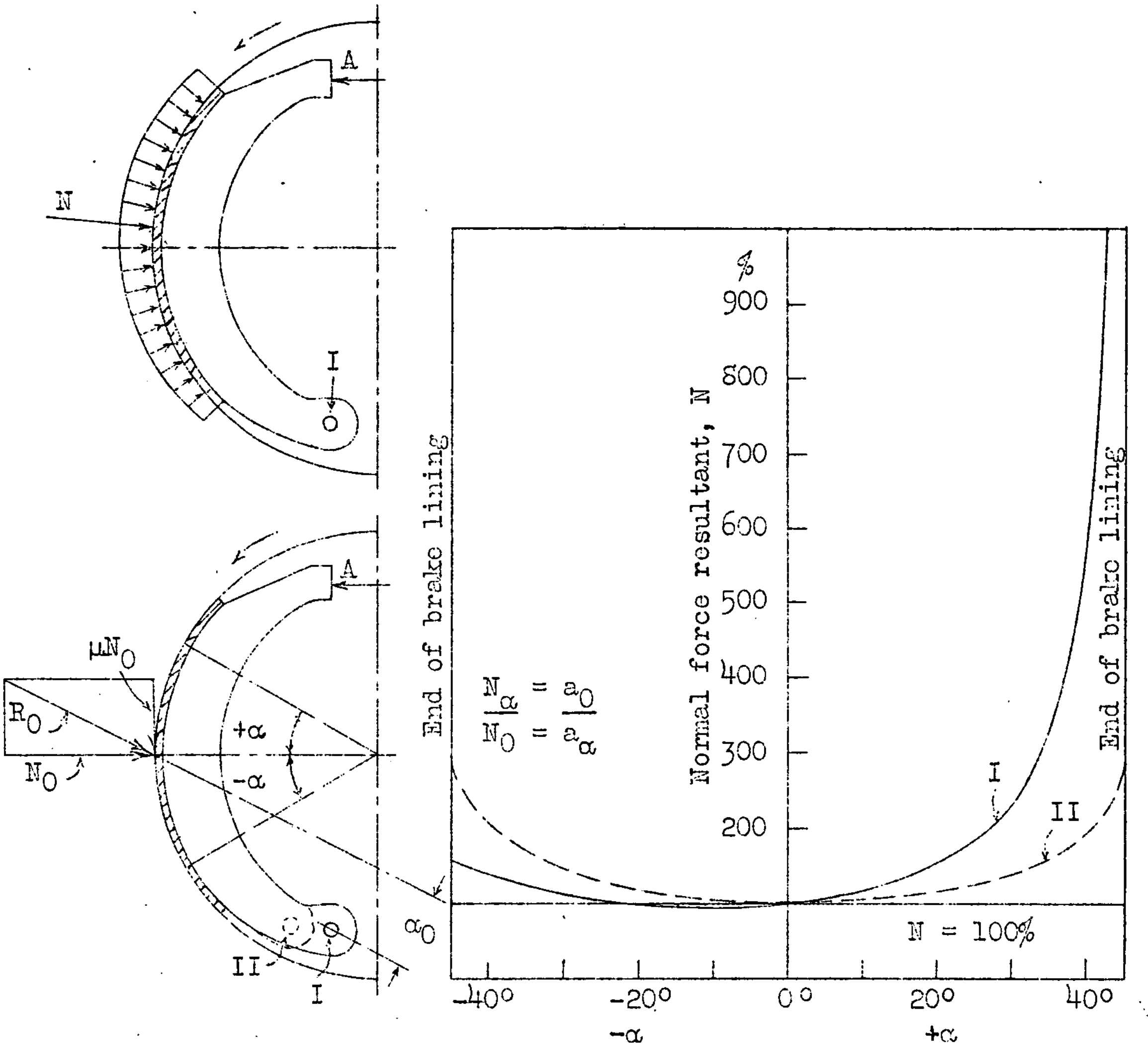
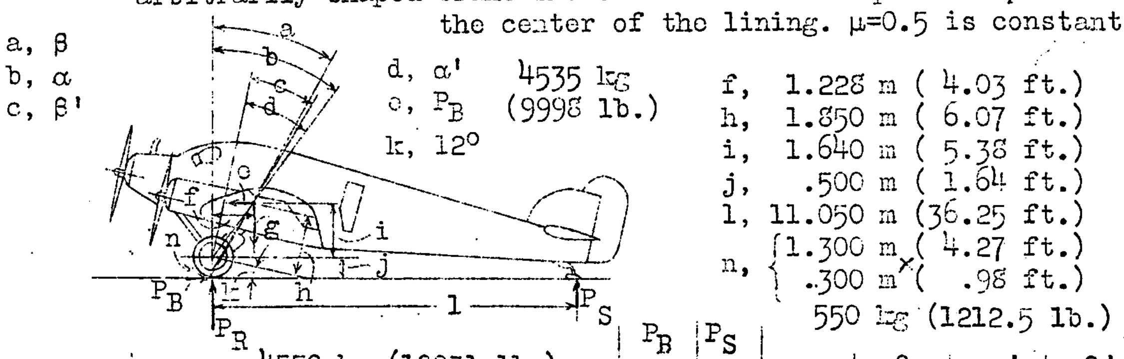


Fig. 8 Effect of location on maginitude of normal force resultant for arbitrarily shaped brake shoe. The resultant equals 100% in the center of the lining. μ=0.5 is constant.



	P_B kg	P_S kg	α	β	α'	β'
With tail skid	4535	565	19.1°	28.5°	24.7°	36.7°
With tail wheel	4550	550	18.4°	27.8°	23.8°	35.8°

Fig. 9 Junkers G 24. P_B = braking force.

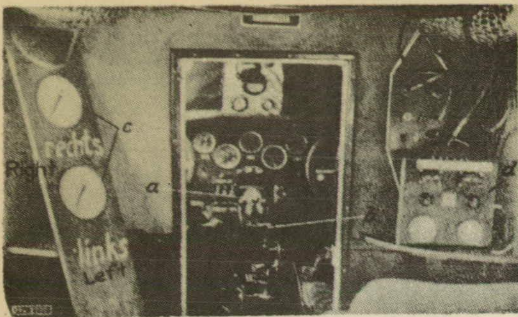
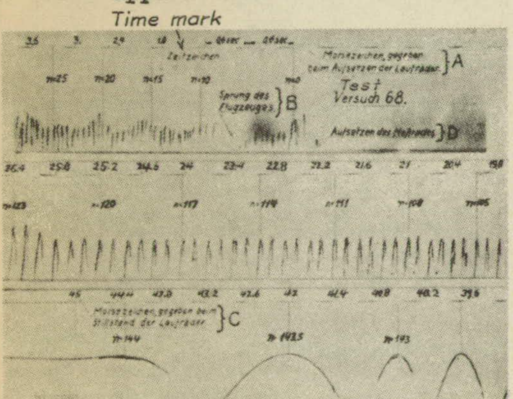


Fig. 10 Cabin of experimental airplane.

Fig. 11 Junkers G24 of the DVL equipped with brake test apparatus.



A, When landing wheels touch ground Morse signal given.
B, Airplane bounces.
C, Morse signal at stoppage of landing wheels.
D, Test wheel contact.

Fig. 12 Optograph record of landing without brakes.

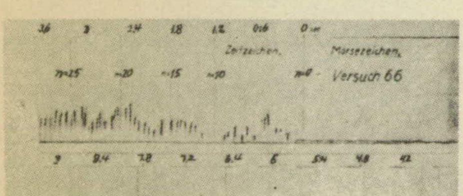


Fig. 13 Optograph record of landing with full braking effect.

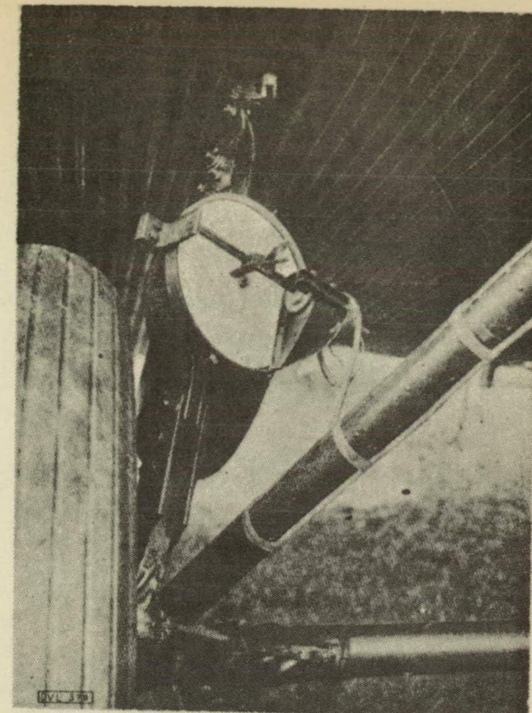


Fig. 14 Dynamometer recording the braking torque.

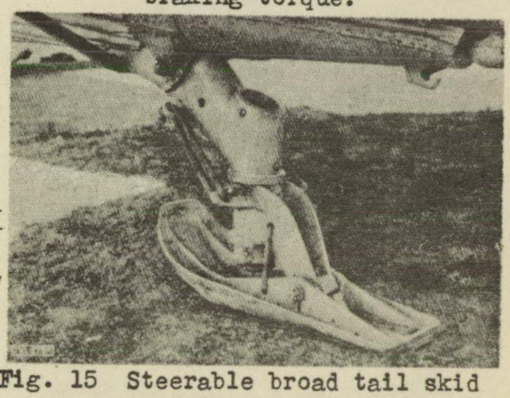


Fig. 15 Steerable broad tail skid used for brake experiments. Ground pressure about 1.2 kg/cm^2. (17.07 lb./sq.in.)

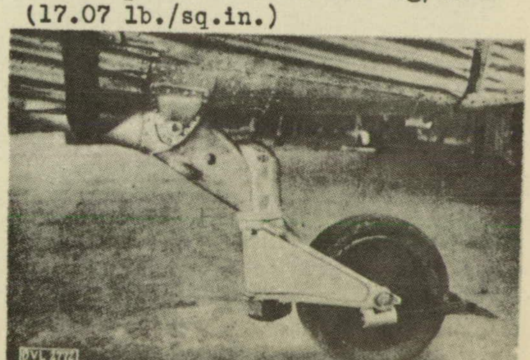


Fig. 16 380 x 130 mm (14.96 x 5.12 in.) steerable tail wheel of elektron with solid rubber tire and recoil spring.

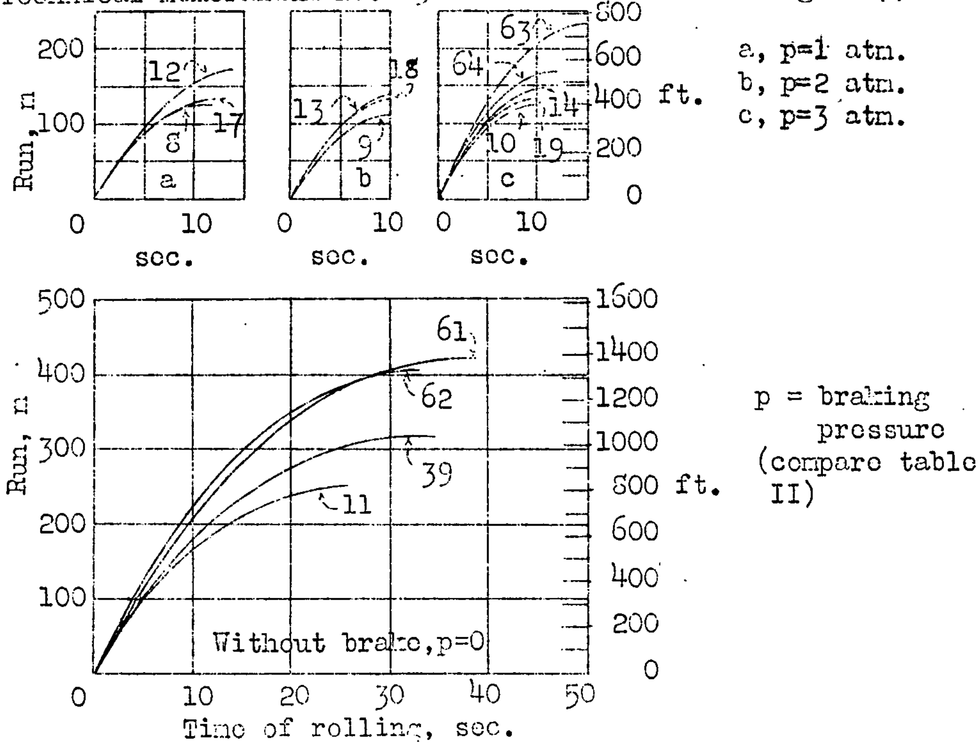


Fig. 17 Landing run with and without brakes plotted against time of rolling, airplane equipped with tail skid and wheel brake according to Fig. 1

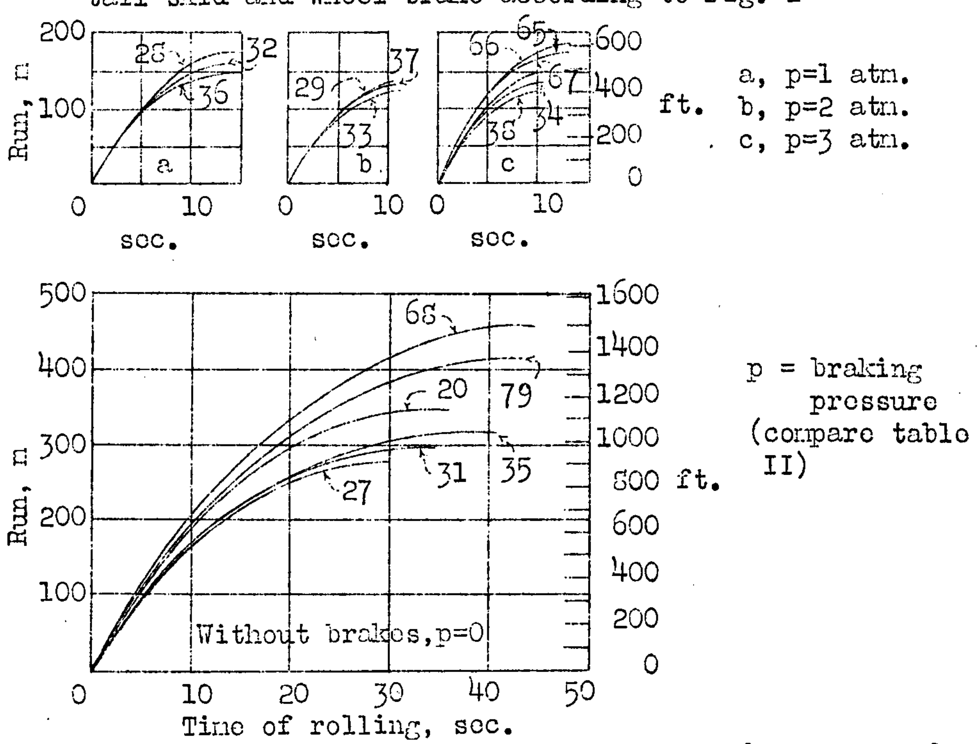


Fig. 18 Landing run with and without brakes plotted wheel against time of rolling, airplane equipped with tail

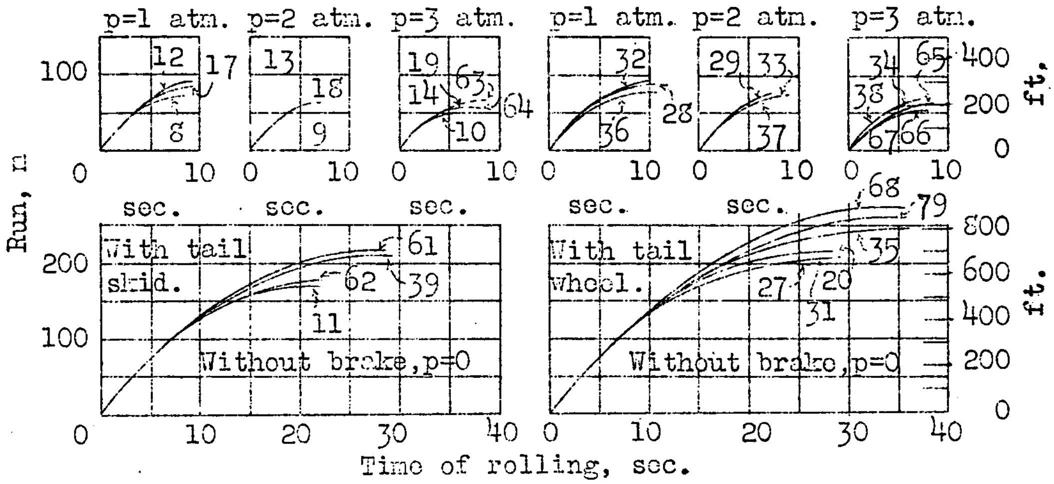


Fig. 19 The runs of Fig. 17 and 18 plotted against time of rolling, but begining at v = 17 m/s (61.2 km/h)(38 mi./hr.) rolling speed.

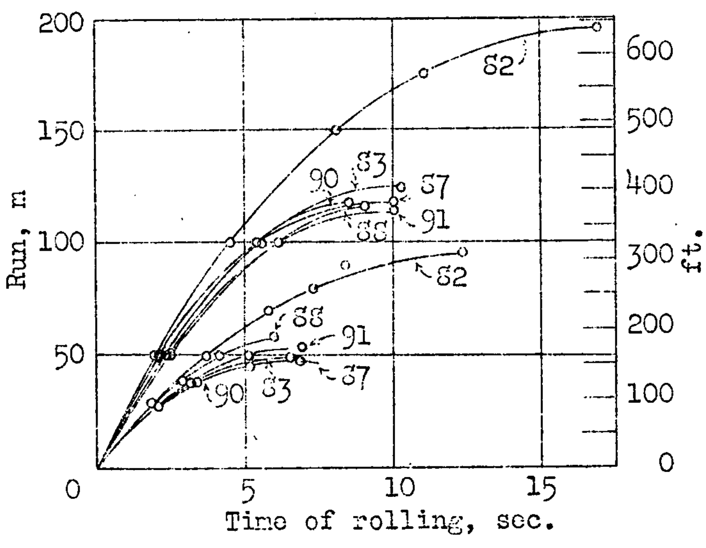


Fig. 20 Landing runs with and without brakes plotted against time of rolling, tail wheel and wheel brake according to Fig. 2. The upper curves denote the total run, the lower as begining at v = 17 m/s (61.2 km/h)(38 mi./hr.) rolling speed. (compare table III)

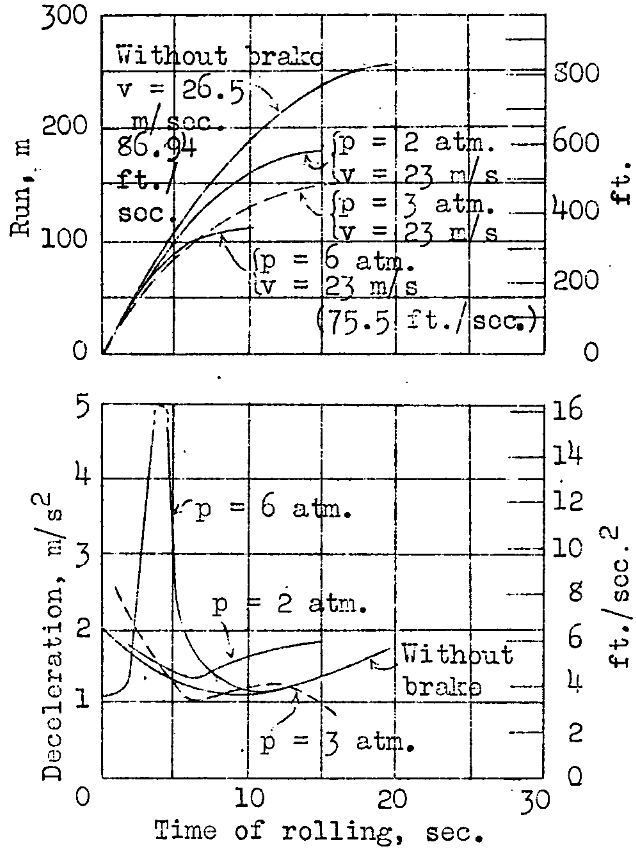


Fig. 21 Run and deceleration plotted against time of rolling for landings with blocked wheels. Airplane equipped with tail skid shoe and wheel brake according to Fig. 2. p = braking pressure, v = landing speed, wind = 2.5m/s (8.2 ft./sec.)

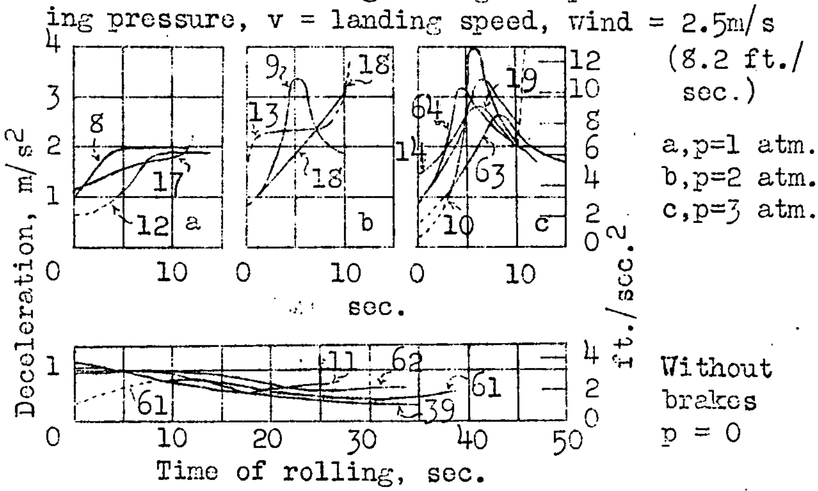


Fig. 22 Deceleration in landings of Fig. 17 plotted against time of rolling.

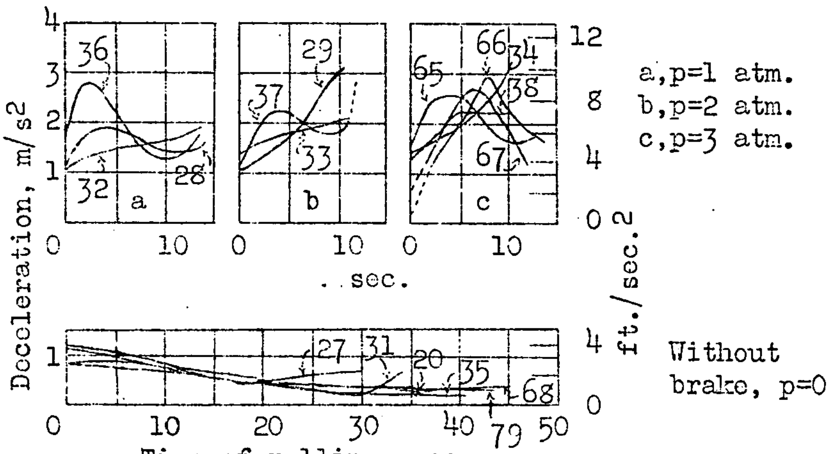


Fig. 23 Deceleration in landings of Fig. 18 plotted against time of rolling.

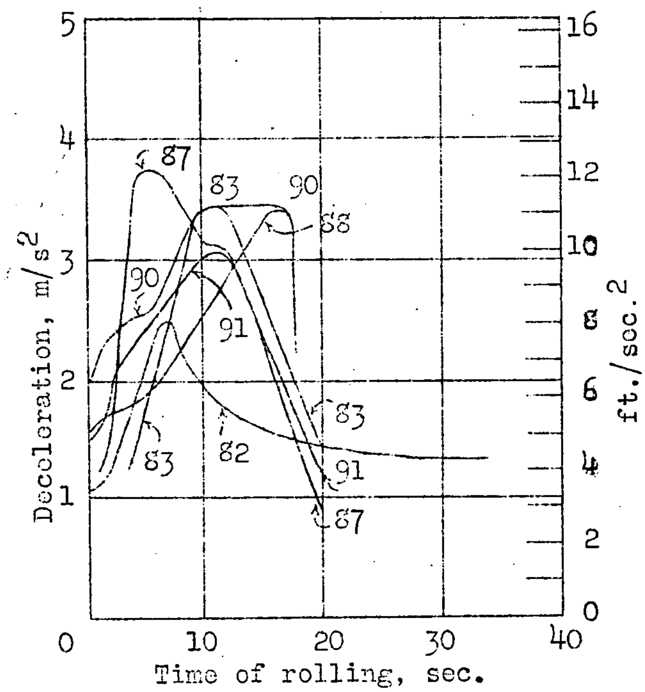


Fig. 24 Deceleration in landings of Fig. 20 plotted against time of rolling.

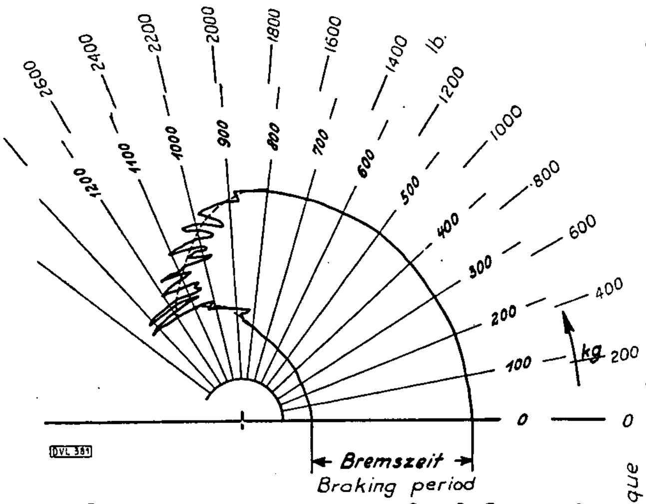


Fig. 25 Dynamometer record of force in transmission rod of the braking torque.(compare Fig. 14.)

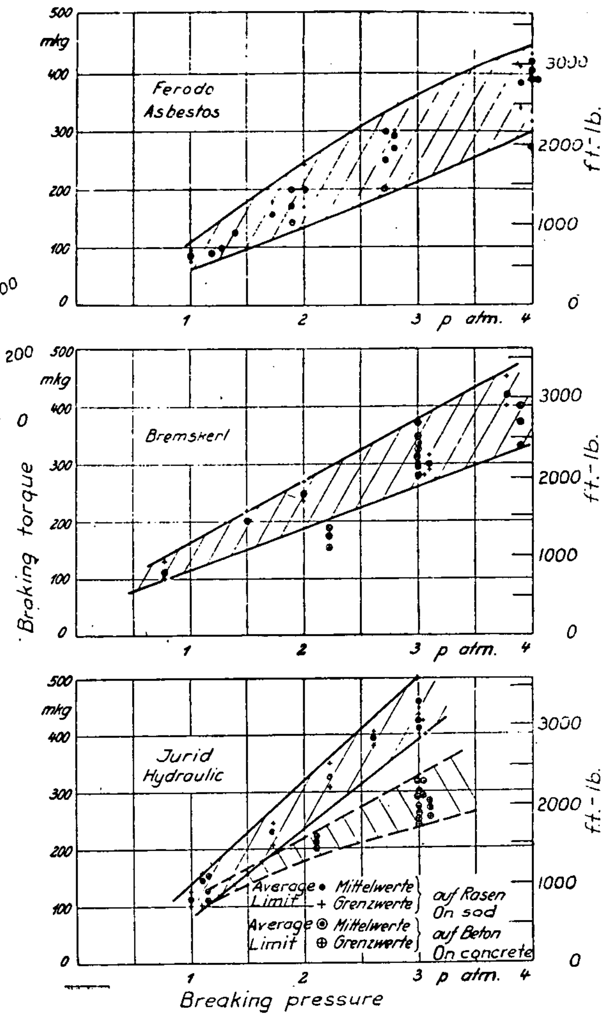


Fig. 29 Braking torque against braking pressure for different brake linings.(Wheel brake of Fig. 1)

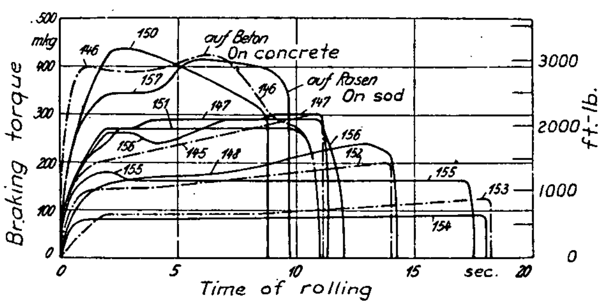


Fig. 26 Braking torque against time of rolling.Wheel brake as in Fig. 1. Lining: Ferodo Asbestos (Compare table V)

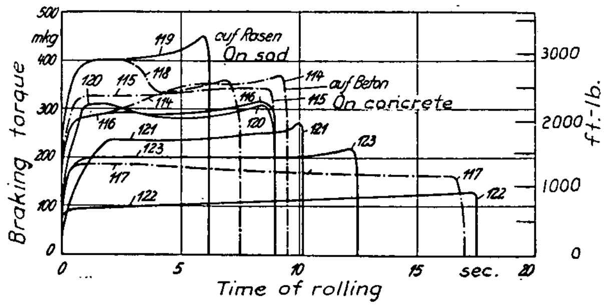


Fig. 27 Braking torque against time of rolling. Wheel brake as in Fig. 1. Lining: Bremskerl moulded shape (Compare table V)

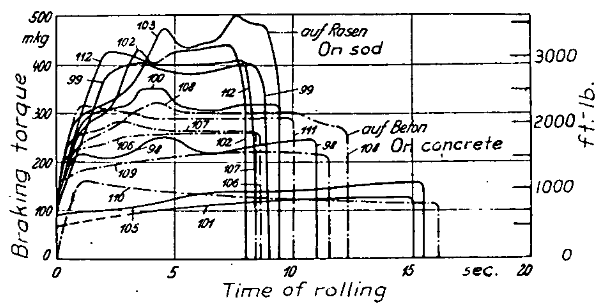


Fig. 28 Braking torque against time of rolling. Wheel brake as in Fig. 1. Lining: Jurid Hydraulic moulded form (Compare table V)

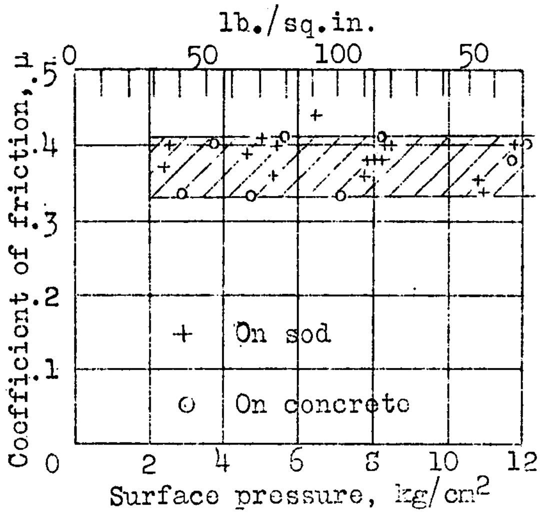


Fig. 30 Coefficients of friction for Ferodo asbestos computed from the braking torques of Fig. 26 plotted against the surface pressure of the lining.

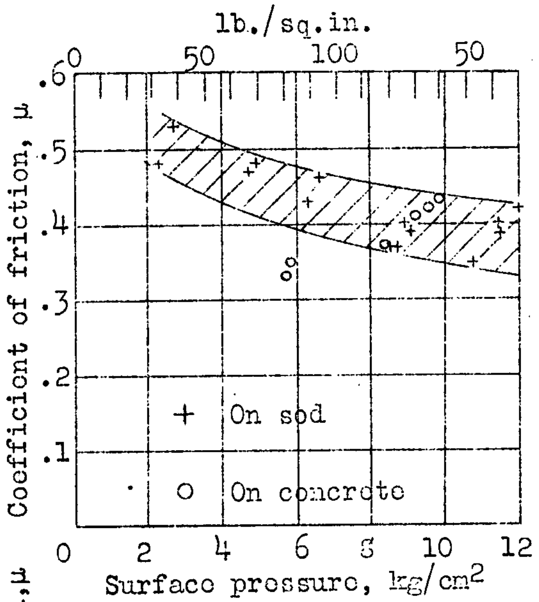
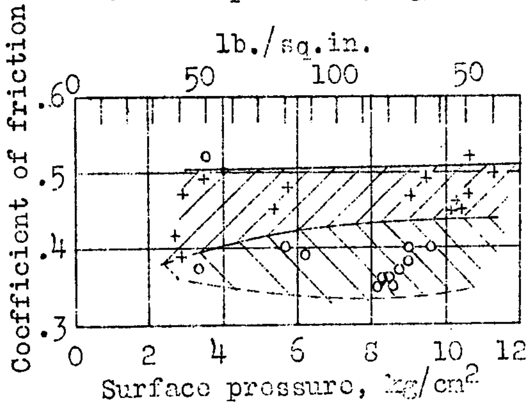


Fig. 31 Coefficients of friction for Bremskerl computed from the braking torque of Fig. 27 plotted against the surface pressure of the lining.



+ On sod
o On concrete

Fig. 32 Coefficients of friction for Jurid hydraulic computed from the braking torques of Fig. 28 plotted against the surface pressure of the lining.

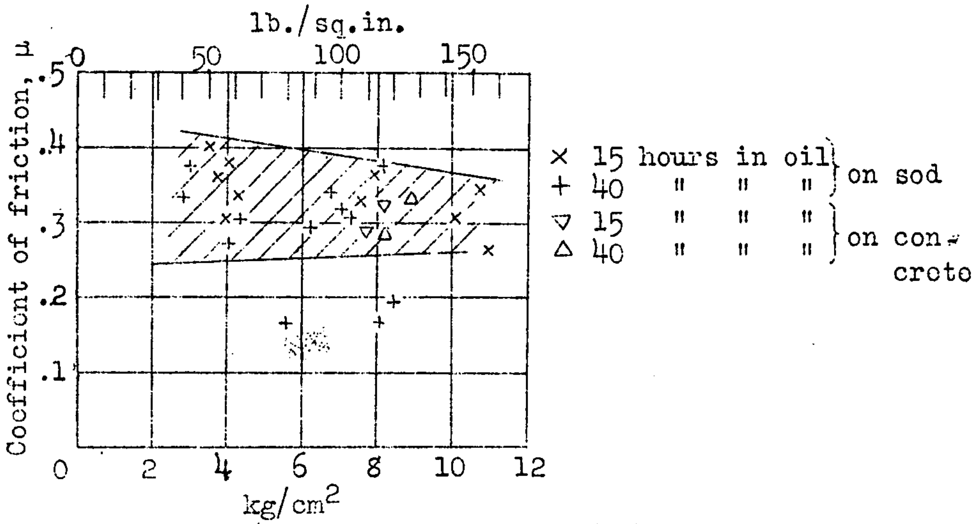


Fig. 33 Coefficients of friction for "oiled" Bremskerl plotted against the surface pressure, as computed from tests after 15 and 40 hours soaking in oil.

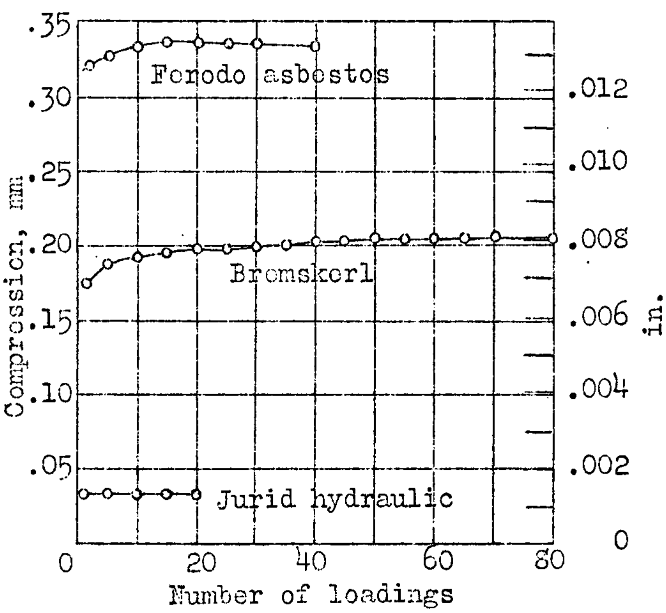


Fig. 37 Compression plotted against number of loadings. Repeated loadings to 10 kg/cm² (142.24 lb./sq.in.) surface pressure without stages.

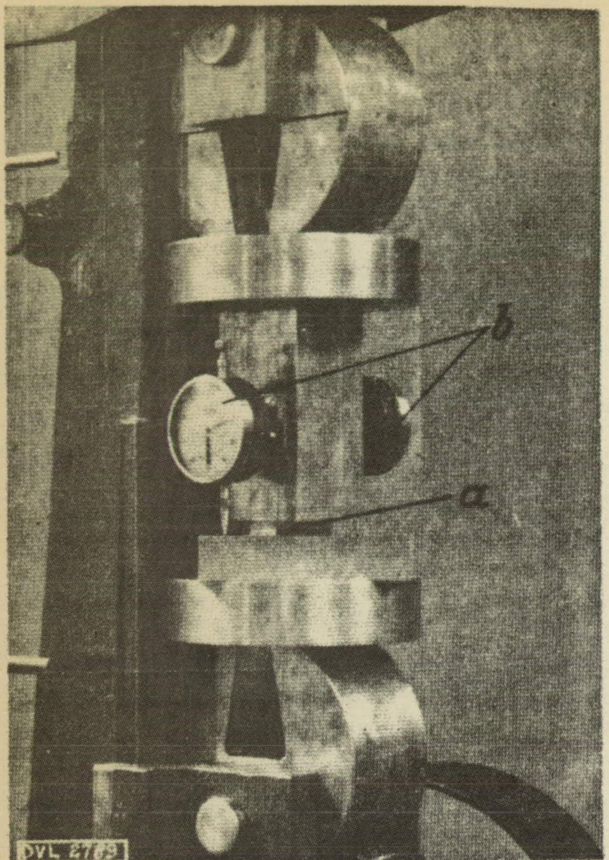
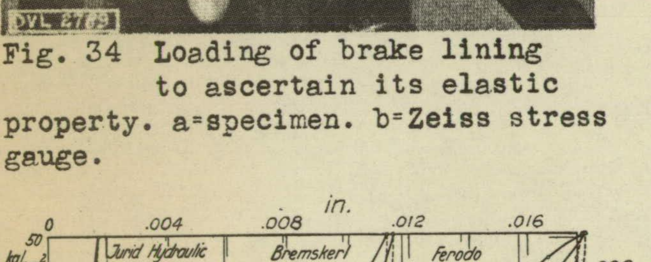
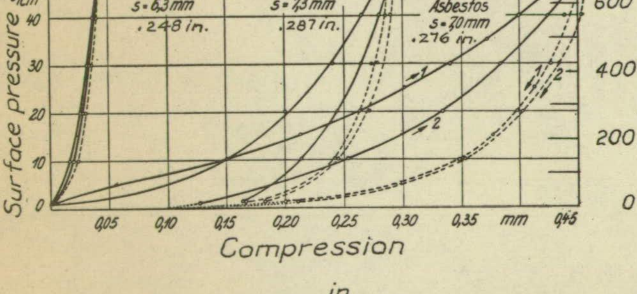


Fig. 34 Loading of brake lining to ascertain its elastic property. a=specimen. b=Zeiss stress gauge.





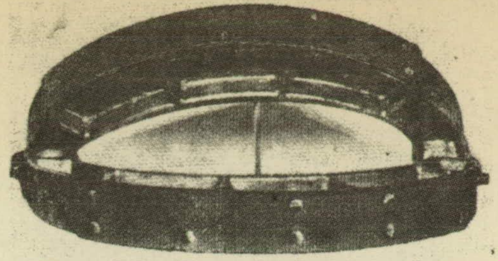


Fig. 38 Ferodo brake shoes after tests showing even wear.

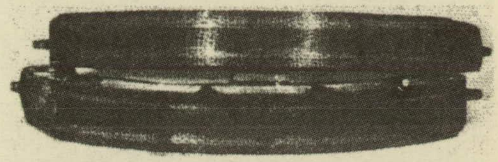


Fig. 39 Bremskerl brake shoes after tests.

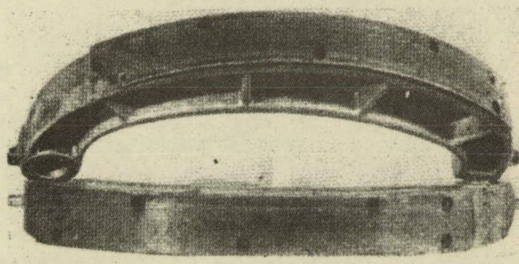


Fig. 40 Jurid hydraulic brake shoes after tests. The type of brake at the corner attests to the hardness of the material.

Fig. 35 Surface pressure against compression after twice loading to 50 kg/cm^2 (711.17 lb./sq.in.) surface pressure. The reduction to equal strengths in the linings was foregone, since it does not alter the conditions to any extent.

Fig. 36 Surface pressure against compression for different brake linings by 20 kg/cm^2 (284.47 lb./sq.in.) load stages. The scale is not the same as in Fig. 35