NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM 1422

ON LANDING GEAR STRESSES

By A. Gentric

Translation of "Sur les Sollicitations des Atterrisseurs."
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SUMMARY

Vibratory phenomena, a new source of landing gear failures.

Preliminary study of test installations: Principle and application of the (landing gear testing) toboggan and of the vertical (drop test) machine. Excitation of the landing gears in the two cases.

Tangential forces applied to landing gears. Influence on these forces of the moment of inertia of the wheel assembly, of the inflation pressure, of the sliding velocity, of the nature of the rubber constituting the tire, and of the characteristics of the ground. Evaluation of the results obtained in the course of the test. Attempt at generalization. Uncertainty about the value of friction coefficient.

Fore-and-aft oscillations of landing gears. Some examples of forces in the drag strut and of displacements of the wheel axis. Study of the vertical forces. Influence of different factors. Consequence of fore-and-aft oscillations. Examination of various solutions: prerotation of the wheels before landing and longitudinal dashpot.

Examples of fatigue failures resulting from fore-and-aft oscillations.

Possible progress by extensive experimentation, the calculation of vibrations, improvement of existing test equipment, and utilization of new devices.

1. GENERAL

The present report is intended for the information of groups studying landing gears.

For a long time, and still even at present, manufacturers of landing gears have given attention, above all, to the problems of the absorption of the vertical energy during the landing impact of airplanes and to those of suspension during rolling. It must be remarked that the official regulations themselves place the emphasis on the two preceding points and

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arbitrarily relate the different stresses to the maximum vertical force by interpretation of more or less empirical relationships.

It seems plausible to think that up to 1945, at least in our country, the strictness of the official regulations for the various stresses was sufficient to lead to a dimensioning capable of masking the phenomena one encounters today.

In the course of these last years, the conditions for calculation have undergone a rapid evolution. A very pronounced tendency toward lowering of the load factors, combined with an increase in weight and speed, has given greater importance to vibratory phenomena and to the behavior of structures under cyclic loads.

Paradoxical failures, at least apparently paradoxical when referred to the regulatory requirements, manifested themselves.

A typical example, reported by J. F. McBrearty in an article entitled "A Critical Study of Aircraft Landing Gears" which was published in the Journal of Aeronautical Sciences of May 1948 gave information on the destruction of the main gear of the Lockheed Constellation by failure of the drag struts. These struts had been designed to operate in tension and were, according to the same source, particularly strong. Strain measurements made on the various members showed that they underwent stresses in tension and compression due to the longitudinal oscillations of the landing gear and that the forward acting stresses were equal to those acting toward the rear.

We had occasion to examine closely the same phenomena in 1949, during the landing-gear tests of the S.E. 2010 "Armagnac" and, subsequently, of all approved landing gears.

Before presenting results we shall briefly describe the test installations, which are little known in France except to a few landing-gear specialists.

2. THE TEST INSTALLATIONS

The study of the behavior of landing gears has been conducted on two test installations.

2.1. Toboggan

The installation which permits the heaviest loads, called the "toboggan," is represented schematically below. We shall quickly review its principles which M. H. Landwerlin developed in a communication to the Congress of Aviation in 1945.

The landing gear is fixed on a backstop inclined toward the horizontal at an angle α. Below this short backstop, there runs, on a horizontal track, a wedge-shaped carriage, the face AB (fig. 1) of which likewise forms an angle α with the horizontal. The landing gear is mounted on the backstop in such a manner that its position, relative to the face AB of the carriage, will be exactly the same as that it would occupy in the type of landing gear intended for study.

The carriage is hauled up on an inclined plane by means of a winch. It is then abruptly released, descends along the track on this inclined plane and, before striking against the landing gear, traverses a certain distance where the track is horizontal. The photos in figures 2(a) and 2(b) represent the complete installation.

The inclination α of the backstop and of the active force of the carriage is 30°. This angle requires the mass of the carriage to be one quarter of the reduced mass pertaining to the test landing gear, and requires the carriage to have a horizontal velocity twice the vertical velocity of impact.

The tests made so far tend to approximate actual conditions during which one assumes that the weight of the airplane is exactly supported by the aerodynamic lift. This condition implies that in a real impact the motion is uniform or, in other words, that the acceleration due to gravity is zero, with the lift balancing the weight. The horizontal part of the track permits realization of this type of operation. The speed is measured in the last centimeters preceding contact, and it is found experimentally that, for practical purposes, it does not vary in the horizontal part of the track. The accuracy of measurement is ± 1 percent.

During the impact, the following measurements are made:

Recording of the displacement of the carriage as a function of time

Recording of the flattening of the tire

Recording of the vertical and horizontal displacements of the wheel axis (trajectory described by a point on the spindle). Moreover, various strain measurements are made for studying the dynamic behavior of the structure. The devices used are electrical-resistance strain gages glued to the structure.

2.2. Vertical Machine

The other landing gear testing maching is designated, in contrast to the "toboggan," by the name "vertical machine." Figure 3(a) shows a photograph of it, and figure 3(b) a sketch of its operating principles.

A platform P under which the landing gear is fixed drops in free fall. The impact takes place either on a dynamometer anvil or on the rim of a flywheel.

Contrary to what happens on the "toboggan," the forces of gravity would operate during the impact if a device, called a compensator, would not balance them, thus representing the aerodynamic lift. This compensator is made up of two long-stroke pistons on which the platform is supported after a free fall of a height determined by the required impact velocity.

Following the establishment of contact with the compensator pistons, except for a few transitory oscillations due to the contact, the vertical motion takes place at constant velocity. The mass of the combination of the platform, the compensator pistons, and the landing gear is equal to the reduced mass of the airplane corresponding to the test landing gear.

2.3. Excitation of the Landing Gear

On the "toboggan". The initial impulse which subjects the landing gear to a vibratory motion depends evidently on, among numerous other factors, the position of the landing gear with respect to a line perpendicular to the ground passing through the wheel axis and, to an equally high degree, on the speed of the rolling band of the tire with respect to the ground.

The position of the landing gear with respect to the perpendicular to the ground is invariable during a test, and the parameter one can modify most easily is the relative speed of the tire with respect to the ground. On the "toboggan," the landing gear is fixed and the ground, represented by the carriage, possesses the desired relative vertical velocity, but the relative horizontal velocity is equal to $V_z\sqrt{3}$ where V_z is the relative vertical velocity. This horizontal velocity is too small to achieve the required horizontal impulse ("horizontal" here signifies parallel to the face of the carriage). We have therefore been induced to set the wheel in rotation before the impact in order to obtain a correct relative velocity. The wheel (or wheels) is given a rotational motion in the opposite sense to that imposed on it by the ground in a real landing. The photographs of figures 4(a) and 4(b) show the spin-up devices.

In figure 4(a), a wooden pulley connected with the wheel is driven by a belt.

At the impact the belt falls, and the motion is not disturbed.

In figure 4(b), the two tires are driven by a "bogie" placed behind the landing gear; the bogie contains two tires rotating at the same speed.

Immediately before the impact, the driving bogie is forcibly propelled backwards by an elastic return mechanism made of shock cords. The carriage then makes contact with the tires which have continued to rotate. In the two cases, the stopping of the wheel causes the impulse which excites the landing gear.

On the vertical machine.- Whereas on the "toboggan" the stopping of the wheel is obtained by dissipation of its kinetic energy in friction on the surface of the carriage and consequently by its absorption of part of the energy of the latter, on the vertical machine the wheel is stationary and, at the impact, makes contact with a flywheel driven at the required peripheral speed.

Here the representation of the phenomena is correct whereas on the "toboggan" the fore-and-aft excitation of the landing gear is obtained only at the price of a certain disturbance of the absorption of the energy stored in the carriage. However, this disturbance of the energy absorption does not, for practical purposes, influence the fore-and-aft oscillations of the landing gear.

3. STRESS APPLIED TO THE LANDING GEARS

The motions of the landing gears upon contact with the ground depend on the form of the stress applied to them. This stress can be broken up into a vertical oscillation corresponding to the classical absorption of the energy of descent and another horizontal one which stems from setting the wheels in rotation.

In the absence of rotation of the wheels, which would correspond to a landing with the wheels turning at a speed such that the sliding velocity would be zero from the instant of contact onward, it is easy to use the pneumatic tire as a dynamometer. Nevertheless, one must make preliminary static calibrations. The transposition of the static results into dynamically valid quantities by calculation does not permit predicting the forces with a precision greater then ±4 percent.

But the problem becomes more delicate when it is a matter of simultaneously measuring the vertical force and the tangential force due to either the stopping of the wheel on the "toboggan" or to its being set in rotation on the vertical machine. For want of more appropriate means, we made the rather daring hypothesis that the vertical force applied to the pneumatic tire by the ground is an unequivocal function of the flattening of the tire. Thus, this flattening has served us for measuring the vertical force. As to the tangential force applied to the tire, we had to resort, in the absence of an angular accelerometer of sufficiently high frequency, to recording angles of rotation as functions of time and to making a double differentiation; the moment of inertia of the wheel equipped with the rotating masses of the brake is determined by means of a bifilar pendulum.

3.1. Tangential Forces

We have related the tangential stress to the vertical force by means of the friction coefficient

$$\mu = \frac{X}{Z}$$

Z being the instantaneous vertical force
X being the instantaneous horizontal force

The accuracy of the measurements combined with the difficulty of reproducing exactly the same phenomena does not permit giving these friction coefficients with an approximation better than ±20 percent.

The following results have been obtained on four landing gears which differed by the weights of the airplanes they were intended for and likewise by their design. Even though these unsimilar testing devices did not permit us to make systematic comparisons, they have nevertheless clearly outlined the variations of the friction coefficients under the influence of certain parameters among which we shall mention:

(1) the moment of inertia of the wheel assembly

(2) the inflation pressure

(3) the sliding velocity

(4) the nature of the rubber of the tire

(5) the nature of the ground

Variations of μ with the moment of inertia of the wheel.- The curves of figure 5 show the variation of the friction coefficient as a function of the rotational speed for the tires of four landing gears.

Curve A corresponds to a wheel having an inertia equal to 0.23 mass kg meters square (wheel 12).

Curve B corresponds to a wheel having an inertia equal to 4 mass kg meters square (wheel 24).

Curve C corresponds to a wheel having an inertia equal to 6 mass kg meters square (wheel 22.5).

Curve D corresponds to a wheel having an inertia equal to 7.2 mass kg meters square (wheel 30).

The corresponding airplanes weigh, respectively, 6 t, 20 t, 50 t, and 75 t; the last two have bogey-type landing gears. The ground is the concrete slab attached to the carriage of the "toboggan". The rotational speed gives immediately the relative sliding velocity since the radius under load and the tangential component of the translational velocity of the carriage (4.35 mn in all cases) is known.

All the above tires are of natural rubber, with absolutely smooth contours. One notices a characteristic variation of the friction coefficient with the dimensions of the wheels. However, the curve C does not occupy the place which its moment of inertia would apparently assign to it. In this connection - and we shall see later the reason for it - it must be pointed out that the curve C corresponds to low-pressure tires (p_0 = 4.2 kg/cm^2), whereas the others correspond to inflation pressures:

$$\text{curve A, } p_0 = 3.5 \text{ kg/cm}^2$$

$$\text{curve B, } p_0 = 8 \text{ kg/cm}^2$$

$$\text{curve D, } p_0 = 8 \text{ kg/cm}^2$$

Variation of μ with the inflation pressure.- The influence of the inflation pressure on the friction coefficient has been studied on the landing gears corresponding to curves A and B. The impacts were successively made on a steel plate and on a concrete plate for inflation pressures of 1.5, 2.5, and 3.5 kg/cm^2 with natural-rubber tires, with the other test conditions remaining unchanged. Figure 6 shows the variation of μ on the steel plate.

It will be seen that μ increases with decreasing inflation pressure and, moreover, that it shows a rather clear tendency to increase when the wheel slows up, that is to say, when the relative sliding velocity decreases.

The same experiment was made on a concrete plate with wheel 12 and wheel 24. The curves of figure 7(a) show the variation of μ for wheel 12 and those of figure 7(b) for wheel 24. The same variation with the pressure is evident.

Whereas on steel μ increased when the sliding velocity decreased, in the case of figure 7(a), μ remains essentially constant with the velocity and, in the case of figure 7(b), μ decreases the more rapidly with the velocity as the inflation pressure increases.

Variation of μ with the initial sliding velocity.- As was to be expected, the variation of the coefficient μ depends on the initial value of the rotational speed. This effect is shown by the curves of figure 8 for the wheel 24 on a concrete slab.

The curves of figure 9 represent likewise the variation of μ on a steel plate. The curves of figures 8 and 9 show, moreover, that the effect of the variation in initial sliding velocity is very different according to the nature of the surface.

During the tests the tire, in rotating, throws off onto the impact surface particles of melted rubber.

The motion of the carriage is such that the tire slides on this viscous film (fig. 10) which has the effect of making the friction coefficient vary.

What can this variation be? In order to determine its order of magnitude, we gave to the landing gear the position symmetrical to that it occupied with respect to the plane normal to the ground and passing through the wheel axis; the sense of the rotation of the wheel remained unchanged with respect to the landing gear.

This modification which does not change, fundamentally, the geometry of forces, with the reservation that the translational velocity of the carriage must be taken into account, has the effect of throwing off the melted rubber backwards with respect to the direction of the relative displacement. The results follow from the curves given in figure 11.

One will note that, systematically, the last results obtained are by about 50 percent higher than those obtained in the initial position.

This aspect of the problem shows therefore a systematic error in our results which would thus be erroneous by default. It is interesting to compare these results with those given by McBrearty; these latter correspond to drops onto a fixed surface. Although the displacement of the carriage of the "toboggan" is rather small, nevertheless the friction is not produced always at the same point as in the American tests. One can only regret that this author has not given any details on his operating technique.

Variation of μ with the nature of the tire rubber.- We have been led to wonder whether the nature of the rubber modifies the values of μ. This concern arose due to the following reason: at every impact, the stopping of the wheel by the carriage dissipates into heat the entire energy stored up in the rotating wheel. This heat is dissipated in the cover of the tire and on the platen of the carriage. The temperature attained has not been measured but in every impact the rubber melts and burns at contact with the carriage. The tire rotates, therefore, during the time interval required for the stopping, on a layer of liquefied rubber. Thus the coefficient μ can only diminish. Under these conditions, the value of the melting temperature of the rubber must influence the obtained value.

The curves of figure 12 give simultaneously an account of the effect of the nature of the rubber and of the inflation pressure on μ; the tests in this case were made on a steel plate.

The solidly drawn curves have been obtained for synthetic rubber whereas the dotted curves were obtained for natural rubber. These tests have been performed only on the landing gear equipped with a wheel 12. The differences found to exist between the friction coefficients obtained are due to the fact that melting of the synthetic rubber has not been observed whereas the natural rubber did melt. Let us add also that, under the conditions of impact as closely similar as possible (same relative vertical velocity of carriage and landing gear, same sliding velocity of the tire), there has never been observed on tire 12 a melting of the rubber as pronounced as on tire 30 made of an identical rubber. In our opinion, this fact must be attributed to the difference in energy dissipated per unit of contact surface; on tire 12 the energy dissipated per unit surface has not led to melting whereas on tire 30 the dissipated energy is much higher (approximately in the ratio of 1 to 3). It seems reasonable to think that the melting temperature of the rubber constitutes a decisive threshold for each type of landing gear and, consequently, for each type of airplane; thus a variation in the vibratory behavior of the landing gears must result.

Variation of μ with the nature of the ground.- The fifth parameter studied was the nature of the ground. The concrete slabs of the carriages have been made with a medium grain similar to that of the runways used in France. The steel plates were covered by a black oxide layer resulting from a natural oxidation. The rust which covered them initially had been removed, after a few impacts, by the friction of the rubber. The curves in figure 12 illustrate the fluctuations of the friction coefficient with the nature of the ground. As we incidentally pointed out before, the variations of μ with the sliding velocity are not the same on steel as on concrete and, moreover, the values of μ are generally slightly higher on steel than on concrete. On steel, μ attains a maximum value of 1.35 whereas it does not attain more than 0.95 on concrete; this applies to the tire 12. Furthermore, it should be mentioned that the maximum value of 0.95 obtained on concrete was found for an inflation pressure of 1.5 kg/cm^2, as compared to 2.5 kg/cm^2 for the pressure corresponding to $\mu = 1.35$ on steel.

Evaluation of the results obtained.- We have emphasized before the systematic error which we have made due to the projection of powdered rubber on to the concrete strip on which the wheel slides. However, we cannot state that the magnitude of this discrepancy is the same for all wheels tested.

We shall review all causes of errors in order to show the uncertainties which still prevail regarding the admissible values for the friction coefficients.

Before dealing with these causes of errors, we want to stress the essential characteristics of the measurements made. The desired goal is the measurement of a friction coefficient under transient conditions, and such a measurement has nothing in common with the coefficients obtained under steady-state conditions. This may explain the considerable differences between the figures obtained by the methods we have applied and those to be found in the technical literature.[1] The stopping time is intimately linked with the moments of inertia of the wheels and, correlatively, with the heat generated by the sliding which modifies the friction coefficient.

The moment of inertia of the wheels has been measured experimentally by means of a bifilar pendulum. The accuracy of the method has been proven with a circular metal ring that had been weighed previously. Calculation and measurement did not differ by more than 1.5 percent. Unfortunately, the moment of inertia of the wheel is a quantity which is variable in the course of a test. The flattening of the tire brings a certain mass of rubber closer to the axis. The calculated variation of I attains 4 percent, and a measurement made on a statically deformed tire with the flattening observed in the course of the test gives 3 percent. We therefore operated with the hypothesis of a constant I. We have not taken into account the variation of I with the centrifugal force although this variation may be larger than the previous one. This last approximation is partly justified by the fact that at the establishment of contact with the ground the wheel does not yet rotate in a real landing.

We have related the tangential stress to the normal force by the friction coefficient. Even if the measurement of the vertical force by means of flattening of the tire is correct in the absence of sliding velocity, and thus of a tangential force, nothing is less certain in the case where the two stresses exist simultaneously.

The analysis has been made by implicit assumption that the equilibrium of the forces corresponds to figure 13(a). The flattening $(R - a)$ gives the vertical force Z which balances the forces of inertia P. The slowing-up couple of the wheel gives the tangential force, through knowledge of the quantity a. Under the action of the tangential force, the tire is deformed as indicated in figure 13(b). The resultant Z of the ground-pressure forces acting on the tire does no longer pass through the wheel axis and produces a couple with the same sign as that

[1]See the article of E. C. Pike in the Journal of the Royal Aeronautical Society, of December 1949, entitled "Coefficients of Friction."

which is due to the force X. Furthermore, the application of a tangential force to the tire produces, for a given vertical force, a reduction in the oblateness, and the distance from the axis to the ground becomes (fig. 13(b)) $a' < a$. The result is that:

The tangential force is overestimated because one attributes to it, and to it alone, the stopping of the wheel; the vertical force is equally overestimated since it is deduced from a flattening larger than that which one would obtain without tangential force.

Since both causes act in the same direction, it was not possible to determine either direction or magnitude of the error committed by the simplification which was made.

Static measurements with tangential force and with vertical force have been made, measuring the moment acting upon the axis of the wheel. These measurements showed us that the method of analysis applied did not lead to large errors in μ, within the limits where the flattening does not appreciably exceed half of the total flattening. However, if the value of μ is sufficiently correct, the error regarding the vertical force alone may attain an amount of 10 to 20 percent.

For these various reasons, in addition to the difficulty of making rigorously reproducible tests and to the inaccuracy of the graphical differentiation (mean error ±4 percent obtained graphically on a curve of known equation), one must not expect from the given curves an accuracy higher than ±20 percent.

Attempt at generalization.- We studied successively the friction coefficients for five landing gears, the wheels of which were characterized by the following moments of inertia, expressed in kilogram mass meters square:

Wheels	A	B	C	D
I	0.23	4	6	7.2

The tests were conducted for peripheral speeds in the neighborhood of normal landing speeds. The considerable discrepancies found between the different friction coefficients cannot be explained by errors steming from the test methods or the methods of analysis. The errors we have tried to evaluate do not change the relative positions of the curve on the different diagrams. The slightly irregular shapes of the curves plotted must be attributed to a great part to the landing-gear oscillations themselves; these oscillations modify the forces and, consequently, disturb certain measurements. In spite of that, each curve obtained point by point "is smooth" and is by no means only a probable curve. Thus we were led to search for the reason of the discrepancies that had been found. The fundamental observations are: the friction coefficients is the smaller the higher the energy stored up by the wheel; the friction coefficient diminishes when the inflation pressure increases; the variation of the friction coefficient on concrete and on steel with the sliding velocity is not the same.

We then condensed these remarks into a single one: the friction coefficient depends on the quantity of energy dissipated per unit of contact surface between tire and ground and, in addition, on the conductivity of the bodies involved and on their nature.

One is then greatly tempted to try to establish a relation which expresses the preceding principle and which permits the calculation of this coefficient so extremely useful for the study of the vibratory behavior of the landing gear in the design stage. Unfortunately, our results are too inaccurate, dependent on too large a number of parameters, so that our tentative hypothesis turned out to be a failure. Should one admit that, as we were told at Wright Field by an officer of the U. S. Air Force, the study of friction coefficients is of no interest because they depend on too many variables? We absolutely do not think so. We shall retain of the preceding results only the order of magnitude and the direction of the variations without neglecting to mention that the figures presented may, especially in the case of high-pressure tires, very well be underestimated by perhaps 50 percent. We make a point of the fact that the possible increase by 50 percent applies more particularly to the high-pressure tires because the energy dissipated per unit of surface is greater there and because in our tests their rubber melted, covering the sliding strip on the carriage with powdery rubber. With low-pressure tires, the tendency toward melting has always been less marked.

In spite of the 50 percent uncertainty involved in certain of our figures, our results still remain of an order of magnitude less than that of the American results. There would appear to be agreement for the values obtained with the low-pressure tire 12.

We have dwelt at some length upon the measurement of the friction coefficients because on their value and on their variation depends the form of the initial impulse which governs the longitudinal oscillations of landing gears.

3.2 Longitudinal Oscillations of Landing Gears

Some examples of oscillations.- In the course of tests of the forward landing gear of a heavy transport (see fig. 4(a)), very important longitudinal oscillations became evident. The main landing gear of the same airplane showed the same behavior. The symmetry of these two landing gears permitted obtaining rather clear recordings. In the case of wheels with eccentric mounting, a torsional oscillation, the axis of which is approximately that of the shock absorber, is superimposed on the longitudinal stress. This explains the somewhat deformed shape of the displacement curves of the wheel axis appearing below.

Figure 14 gives the displacement of a point connected to the wheel axis, parallel to the ground, as a function of the travel of the shock absorber and the curve of the variation of the forces in the drag strut, as a function of the time, with the rotational speed of the wheel as the variable parameter. The diagrams below are given as they were recorded. One may make them correspond point by point by means of the curve of the travel of the shock absorber as a function of the time; we do not show this curve because it does not offer any interest for the object of the present paper. One should note that the maximum displacements of the wheel are practically insensitive to the variation in rotational speeds, that is to say, of the sliding speeds of the tire on the ground. In all these tests, the axis of the shock absorber is normal to the ground and one will observe that the displacements are of the same order of magnitude toward the front and toward the rear; the forward displacements are always slightly larger than those toward the rear.

Besides, this slight difference is not to be found on the diagrams of the forces in the drag strut. As far as these last are concerned, one observes that the oscillations are essentially symmetrical with respect to zero load. On these diagrams, we have indicated the instant of stopping of the wheel, and this shows that the first impulse in this case is produced before the force in the drag strut changes its direction, that is to say, before the change in the direction of the oscillation.

Figure 15 shows the same diagrams as figure 14, but considers the variation of the vertical velocity only while the other parameters are maintained constant. The increase in vertical speed accounts for a slight tendency toward increasing the amplitude of the oscillations; this tendency is less apparent in the diagrams of forces in the drag strut. One may observe the form of the oscillations of the wheel axis in figures 14, 15, and 16. They represent, in addition to the coupling between the longitudinal and the torsional oscillations, binding of the tube sliding in the fixed part.

Figure 16 represents the same diagrams as figures 14 and 15, but here only the initial inflation pressure has been modified. The effect of this variation is more pronounced than that of the vertical speed and that of the sliding velocity.

In the case of the landing gear with which the results of figures 14, 15, and 16 were obtained, stopping of the wheel occurred before the maximum elongation of the first half-oscillation was attained. This stems from the small amount of energy stored in that wheel (wheel 12); the stopping of the wheel takes place in between two and seven hundredths of a second. In the case of heavy transports this is not so, there the wheels can be made to stop only after several oscillations.

Figure 17 shows the diagrams obtained with the landing gear equipped with wheels 30 (I = 7.2 mass kg times meters square).

The top curves correspond to an inclination of the shock absorber axis of 8.30° toward the rear with respect to a line normal to the ground and passing through the wheel axis. The bottom curves have been obtained with the landing gear normal to the ground. The comparison of these curves, recorded under identical conditions, demonstrates the considerable difference in amplitudes caused by the inclination of the shock absorber. The maximum amplitudes are in the ratio of approximately 1 to 2. One should likewise observe that the amplitude of the first half-oscillation increases with the vertical velocity of impact and that the frequency of the oscillations of the landing gear is influenced by the impact velocity. Numerous parameters, of which the friction is one, contribute to this variation but it seems that the shortening of the landing gear is predominant.

Other aspects of the oscillations.- For the landing gears possessing a drag strut, we have been tempted to determine the horizontal stress applied to those landing gears by the ground through measurement of the force in the drag strut. The complexity of the oscillating system with distributed stiffnesses and masses, and possessing free play, did not allow us to arrive at a concrete result. We have therefore adopted the following schematic representation: When the vertical force Z determined by the flattening of the tire and the force in the drag strut is known, it is possible to replace this system by another, equivalent one, applied to the wheel axis. Under these conditions Z always represents the vertical force, and the action of the drag strut is represented by a horizontal force X. The actual situation is certainly more complex, and this mode of representation has value only because of the picture which it gives of the stress.

The curves of figure 18 give an account of the oscillation of the resultant about the line normal to the ground for the landing gear appearing in figure 4(b).

Two vertical velocities correspond to the sliding velocities shown in these curves. One should note that for a low sliding speed (V = 15 m/s) the horizontal oscillations are extremely strong and that their amplitudes toward the front and the rear are of the same order of magnitude. The value of the horizontal load is close to that of the vertical force. On the other hand, for higher sliding speeds ($V = 40$ m/s), the oscillations are smaller. In this phenomenon one must see the influence of the friction coefficient of the tire and the ground; for low sliding speeds (wheėl set in rotation before the landing), the coefficient is higher and the horizontal impulse is stronger than for the high sliding speeds where the wheel is motionless at the instant of contact with the ground. Let us recall, furthermore, that in our tests the phenomena are reversed; the wheel in rapid rotation corresponds to the wheel which is motionless at the instant of landing, and the slowly rotating wheel corresponds to one put into rotation previously, before the instant of contact with the ground, but at a speed different from the translational velocity of the airplane.

The magnitude of the horizontal stresses at low sliding speeds thus has led us to study how the forces in the drag strut varied as a function of this speed. In fact, an examination of the curves of figure 18 seems to condemn devices for setting the wheels into rotation before landing, that is, those devices which may save wear on the tires but would impair the strength of the landing gear. The impact test at sliding speed zero has shown us that the oscillations are there less strong than at low speeds. It was therefore likely to expect an optimum sliding velocity which we then propose to determine.

For this purpose, the forces in the drag strut have been recorded for several values of the sliding speed and of the vertical velocity. From the oscillograms obtained we deduced the curves of figure 19.

The solidly drawn curves are the loci of the first maxima of the compression force in the drag strut, and the dotted curves correspond to the first maxima of the tension force. These two extremes thus correspond to the first total range of the force. Within the accuracy of the measurements one may state that these different curves pass essentially through a common point characterized by a negative sliding speed (wheels prerotated before landing at a peripheral speed higher than the translational speed of the airplane) of the order of 2.5 m/s. One should note likewise that the maximum force recorded lies between 15 and 25 m/s. This aspect of the problem stresses the unfavorable effect which would result from an insufficient amount of prerotation of the wheel before landing. It is, on the contrary, advisable to start the wheel rotating at a speed slightly higher than the translational speed. For this optimum value of the sliding speed, the force in the drag strut attains its final value in an aperiodic fashion. What is the explanation for this phenomenon? The reason which, we believe, can be given, is:

In the absence of an impulse normal to the axis of the shock absorber, there does not exist any reason for creating longitudinal oscillations since the coupling between the longitudinal motions and the stroke of the shock strut is zero. Since, on the other hand, the negative sliding velocity produces, at the impact, a first impulse forward, one must search whether there does not exist simultaneously an equal impulse of the opposite sign. Now, this last impulse does exist; while the tire is flattened at the impact, its radius diminishes and the ground applies to the wheel an angular acceleration. This angular acceleration results from the rearward impulse due to the ground friction of the tire. It is evident that there exists a certain sliding velocity, necessarily negative, for which the two impulses essentially cancel one another. The same phenomenon has been observed on other landing gears. The speed which gives the aperiodic motion lies in the neighborhood of -5 m/s and depends, evidently, on the relative variation of the radius of the tire during the absorption of energy.

Effect of the longitudinal oscillations of landing gears.- The experimental results which we have produced show the necessity of taking the longitudinal oscillations of landing gears into account in order to dimension them suitably. The failures due to fatigue which we have recorded in the laboratory and to which we shall come back later on indicate that the landing gears should have sufficient strength with respect to the cyclic stresses. The problem has therefore two aspects: (a) One must either have a landing gear capable of oscillating and possessing sufficient strength with respect to the cyclic forces, or (b) one must safeguard the landing gear from such oscillations.

In the first case, for the complete dimensioning of the members, the designer will have to carry out an extremely extensive calculation in order to determine the deformations and, consequently, the fundamental mode of vibration. Such a calculation will have to take into account the displacement of the semisupported masses and likewise the couplings between the axial oscillations of these masses, and the longitudinal and torsional oscillations of the landing gear. It seems to us that the regulations should prescribe these calculations rather than static calculations based on effective forces linked to the maximum vertical force. It appears that the maximum vertical force and the forces due to the longitudinal oscillations are not interrelated, and for this reason they must not be connected by empirical formulas.

In the second case, two solutions present themselves for preventing landing-gear oscillations: either to prerotate the wheel at a rotational velocity such that the sliding velocity at the instant of contact is the optimum value defined above, or to make use of a longitudinal dashpot as Lockheed did on the "Constellation." The first solution involves the risk of running against a serious obstacle; the accuracy with which the sliding velocity has to be obtained exceeds the possibilities of any devices which could be constructed, as a result of the variations in the relative velocity between the airplane and the ground, stemming from meteorological conditions. The second solution has already been proven on commercial airplanes and should, in our opinion, be used generally.

In this connection, let us stress that the "pile driver"-type shock absorbers in which the wheel has a recoil motion accompanying the deflection of the shock strut, do by no means solve the problem. The variables in this case are not separated, and the tests on such a shock absorber have shown that it was the source of longitudinal oscillations as large as those of a conventional landing gear.

The conclusion therefore seems to be a combination of the two procedures:

Use of a longitudinal dashpot

Prerotating the wheel before landing, but solely with the aim of saving the tires.

4. FATIGUE FAILURES, CONSEQUENCE OF THE LONGITUDINAL OSCILLATIONS OF LANDING GEARS

We have the example of landing gears which had successfully resisted static tests. These landing gears have been put into service and caused several incidents.

Figure 20 represents the body of a landing gear made of a light alloy broken in service.

Figure 21 shows the solution adopted in the junction of the lifting pivot and the body.

The weakness of the fillets in a highly stressed region should be noted.

On the other hand, figure 22 shows, at 10 times magnification, the presence of numerous cavities in the metal. The resistance to fatigue of such a piece is visibly impaired by the bad design of the junction of the pivot to the body and, in addition, by the dubious quality of the metal. Thus it is not at all suprising that the behavior in service should have proved deficient.

May we be permitted to stress at this occasion the risks inherent in the use of cast light alloys in the components of landing gears. It is known, in fact, that the conventional fatigue limit of these alloys is very low. For example, in the case above, the fatigue limit was about 6.5 kg/mm^2, the proportional limit 12 kg/mm^2, and the ultimate static rupture stress 30 kg/mm^2. In order to reduce risks of the foundry, the regulations prescribe increased loads in setting-up the calculations. This is contrary to the desired aim which is lightness.

As to the cost of the part, it loses all interest because of its inadequacy to resist the applicable stresses. We think that the cost is not the principal aspect of the problem. Above all, tough and light members, thus members with high strength characteristics are needed. This condemns cast parts. According to our opinion, one must resort to forged or stamped parts. Their higher cost will be offset by a saving which is hard to express in figures but which lies especially in an absence of losses: no suspension of flights, no structural failures in normal landings, no new studies and no new manufacture of replacements to consider, or at least to a much lesser degree.

We should not want to leave the foregoing example without mentioning the dynamic tests which followed the difficulties in service.

An identical landing gear was mounted on the vertical machine. It underwent 40 tests, following which it was dismounted to be placed in another landing position. Before remounting, it was examined and the examination revealed the presence of two cracks in the junction between the case and the pivot, thus prohibiting further testing.

One of the cracks is shown by the photos of figure 23. These photos were taken after cutting up the piece, and the last one corresponds to an enlargement of 7.

Attack with Keller reagent made the intercrystalline progression of the crack evident, which corresponds well to the characteristics of fatigue failures.

Following the tests, the dissection of test specimens taken from the case gave:

Mean rupture stress in static tension

$$= 32 \text{ kg/mm}^2$$

Mean elongation

$$A = 3 \text{ percent}$$

Fatigue limit

$$\text{of the order of } 6.5 \text{ kg/mm}^2$$

The value of the elongation in static tension, although low, is common for cast parts.

Following these tests, which were held to be too severe by the manufacturer, the tests were repeated on a new casing under less strict conditions ($Z = 2700$ kg instead of $Z = 3450$ kg). Resistance-wire strain gages were glued onto the junction of the pivot and the body of the casing In view of the strong curvature of this fillet, two strain gages were glued side by side. Their bases were 20 mm for one and 10 mm for the other. The difference in the readings obtained with the two strain gages reveals the concentration of stresses in the fillet.

Figure 24 gives the recordings of the stretch in these two strain gages. Everything allows the opinion that a strain gage shorter than the one with the 10-mm base and placed more judiciously would have given considerably higher values. The value of the stretch recorded by the strain gage with the 10-mm base (maximum of 3.1×10^{-3}) does, however, not permit a deduction of the value of the local stress. It would have been necessary to make, at the same point, at least one other measurement along a direction perpendicular to the first one; furthermore, this presupposes that the measurements taken would have been made along the principal axes of the deformation.

After several tests (three, to be exact), it became impossible to balance the measuring bridge on which these strain gages were mounted. This symptom is revealing and puts the blame on either the strain gage itself or the piece which begins to crack. The fact that the defect appeared simultaneously in two strain gages very close to each other tends to make one believe the second hypothesis.

These last tests were made without horizontal velocity (flywheel motionless). It is reasonable to assume that putting the wheel into rotation by means of the flywheel would give larger strains. Unfortuately, the cracking related above does not permit repeating the measurements made

$$\left(\frac{\Delta l}{l} = 7.1 \times 10^{-3}\right)$$

At the seventh test, after the one which gave the recording of figure 24, the casing was broken.

Figure 25 shows the fracture.

The mechanical characteristics of this last casing are a little weaker:

Mean rupture stress in static tension

$$= 26 \text{ kg/mm}^2$$

Elongation

$$A = 1.8 \text{ percent}$$

Fatigue limit-

approximately 6.5 kg/mm^2

The preceding example illustrates the disadvantages inherent in the use of cast pieces for strong stresses:

(a) Heavy pieces as a result of the conditions of calculation

(b) Pieces not well suitable for calculation because of possible variations in thickness due to the displacement of the cores

(c) Mechanical characteristics very variable from one point to the next, a defect mainly due to the phenomena of cooling after casting (thicknesses not uniform, coolers wrongly placed, mold poorly fed, insufficient casting risers, etc.), resistance to fatigue very weak

(d) Pieces not reparable in the case of even slight damage

(e) Sensitivity to notching higher than on wrought light alloys.

In the domain of susceptibility to notching, we shall limit ourselves to quoting a last example which is of interest because of the relations it has with a material in service. The drag strut which serves at the same time as a lifting jack is locked by a cylindrical piece with clamps. This piece is represented in its two versions in figures 26 and 27. In the course of the tests for adjustment of the shock absorber, without wheel rotation, on the "Toboggan," after about 40 tests, this piece broke, at right angles with the threads.

These threads, milled from a steel treated to 220 kg/mm^2, were joined to one another by a juncture, the radius of which was of the order of 1/10 mm. One may therefore speak of a connection at a sharp angle. The piece did not have any relief. Since no measurement of the forces in the drag strut had yet been made, one cannot exactly know the magnitude of the force which caused rupture of the clamp. The resistance of the steel, computed from a hardness test, and the broken section make one think that the static strength of the piece in this section exceeded 200 t.

A clamp identical with the preceding one was mounted on a pulsator adjusted to stress it in cyclic tension between 500 kg and 19,400 kg (usable maximum of the pulsator). After 43,000 cycles the clamp broke like the previous one, for an amplitude of stress variation, on the order of 1/10 of the rupture stress in static tension. Various cutouts led us to assume that the maximum force which the clamp had supported before rupture in the drag strut was of the order of 35 t - a figure which seems of the correct range with respect to the result obtained on a pul-pulsator, taking into consideration the much smaller number of cycles run through.

The photograph of figure 28 shows the characteristic appearance of the propagation of the crack. After this incident a new clamp was designed, but this time - taking into consideration the limited space in which the piece had to be contained - the design attempted to obtain the maximum reduction in any notching effect or stress concentration (fig. 27, bottom).

The tests were repeated, strain gages were glued on the drag strut, statically calibrated on a tension machine. After 23 tests with rotation of the wheels, this time under very different conditions, the drag strut gave way by bursting. The reason? A bronze ring forming a piston, screwed onto the previously mentioned clamp, broke due to fatigue like the first clamp (fig. 29).

The measurements made showed that this ring withstood tension forces as much as 45 t. These maxima correspond to sliding velocities between the tire and the ground of the order of 15 to 20 m/s - the values for which the oscillations have the largest amplitude, as we have already shown.

Since the locking of the piston was no longer assured by the locking of the clamp, the drag strut burst due to excess internal pressure. When the clamp had been modified, the bronze ring had been neglected although it presented a striking analogy with the clamp as far as the design is concerned.

We have gone to some length regarding details but our aim is to call the attention of the regulating agencies to the aspects of rupture due to fatigue. Certainly, these phenomena are known, but it seems to us that they have not been given the necessary attention. And this is true above all regarding stress concentrations created by bad detail design. Here, again, one knows the influence of a hole drilled into a stretched bar.

Unfortunately, it seems as if much more thought had been given to adaptation phenomena under static force than to the dangers due to cyclic oscillations. With respect to endurance, the adaptation is not involved. We advise the regulating agencies to refer to the reports and articles on fatigue failures because, until now, the detail-design requirements have not emphasized all the apprehension which endurance phenomena should still evoke.

5. CONCLUSIONS

In the course of this paper, we have attempted to show all the difficulties that arise if one wants to determine the stresses in landing gears. We have tried to measure the friction coefficients between tire and ground, but our measurements do not agree with the American results. We persist in believing that parallel measurements should be continued at the laboratory and in actual operation. We do not try to conceal the greater difficulty of the measurements taken in actual operation; this difficulty stems mainly from the impossibility of reproducing the test conditions. Nevertheless, one will gain from them a statistical knowledge of the landing conditions as McBrearty has done in his article quoted above. The envelope curves should permit setting up better rules for calculation. But as far as we are concerned, we no longer recommend a static calculation of the landing gear as is practiced at present but, after a first dimensioning, making a calculation of vibrations, with the imperative condition that no point in the structure exceed the endurance limit corresponding to the anticipated life of the machine. It is obvious that, with regard to this point, the requirements could not be the same for expendable military combat planes as for military or commercial transport planes.

It is equally evident that the conditions for the calculations should take into consideration the environment in which the airplane operates. The calculation conditions applicable to shipboard airplanes could not be suitable for those which use only concrete runways. A special condition must be mentioned for runways of grating, regarding which we do not know anything about the longitudinal stress.

Compared to foreign accomplishments, the French landing gears are heavy. A recent article of Messrs. Goulias, Bedet, and Le Gall analyzed the causes of this excess weight. The utilization of forged light alloys should permit a distinct improvement. But we think - in the hope of not always having to trail behind foreign work - that still numerous tests are necessary for real progress. Such tests will be possible only to the extent to which present equipment is completed and improved.

We have at present three test installations at our disposal: the "toboggan," the vertical machine, which we have summarily described, and a small vertical machine. This last one is, we believe, the first machine in the world to be equipped with a flywheel, the rolling band of which is of concrete. The peripheral speed of this flywheel, with a diameter of 1.40 m, has been increased to 140 km/h. We did not think it useful to continue beyond this speed because this machine is, by its very structure, intended only for landing gears of small airplanes for which a speed of 140 km/h is already more than enough. A fourth machine is in the process of being constructed and will be capable of accommodating landing gears of airplanes of 125 t and even more. Its striking peculiarity is having a flywheel of 6 m diameter the rolling band of which, 2 m wide, will likewise be of concrete. The anticipated peripheral speed is 300 km/h. Tests made on a concrete model flywheel of 1 m diameter, calculated and built exactly like the 6-m flywheel, led to a light cracking of the concrete at the angular velocity of 2000 rpm. This test permits us to count on a margin of approximately 10, with regard to holding the concrete on the large flywheel, for a peripheral speed of 300 km/h.

We have taken the trouble of constructing concrete surfaces for making impact tests, in order to represent as exactly as possible the stresses on landing gears. This makes it unnecessary, in particular, to investigate the possibility of replacing the concrete surfaces by others with equivalent characteristics, with respect to the behavior of landing gears. Such investigations have been carried out by the Americans and one may ask oneself to what point the experimental results will represent reality.

The machine described above will, moreover, be equipped with a test device to study the endurance of the wheels in rolling. This device will function independently of the landing-gear drop platform which will permit a more intensive utilization of the machine.

As far as the appearance of fatigue phenomena on airplanes is concerned, let us also mention the behavior of the wheels. The Adamson machine for which we have been waiting for so long will be in the process of being installed when this report is published. It will permit performance of these endurance tests jointly with the machine just discussed. Thus one may hope that from 1955 onward the test installations of Toulouse will permit satisfying, within a reasonable period of time, the needs of the French landing-gear industry.

We should not like to close this report without taking the opportunity of thanking Messrs. Aillères and Adde as well as their coworkers for the care and professional conscientiousness they have shown in conducting these tests. We want to be allowed to point out that these landing-gear tests have been conducted, all in all, by five persons, while an equal number of technicians were occupied with the tires, the wheels, and brakes.

Translated by Mary L. Mahler
National Advisory Committee
for Aeronautics

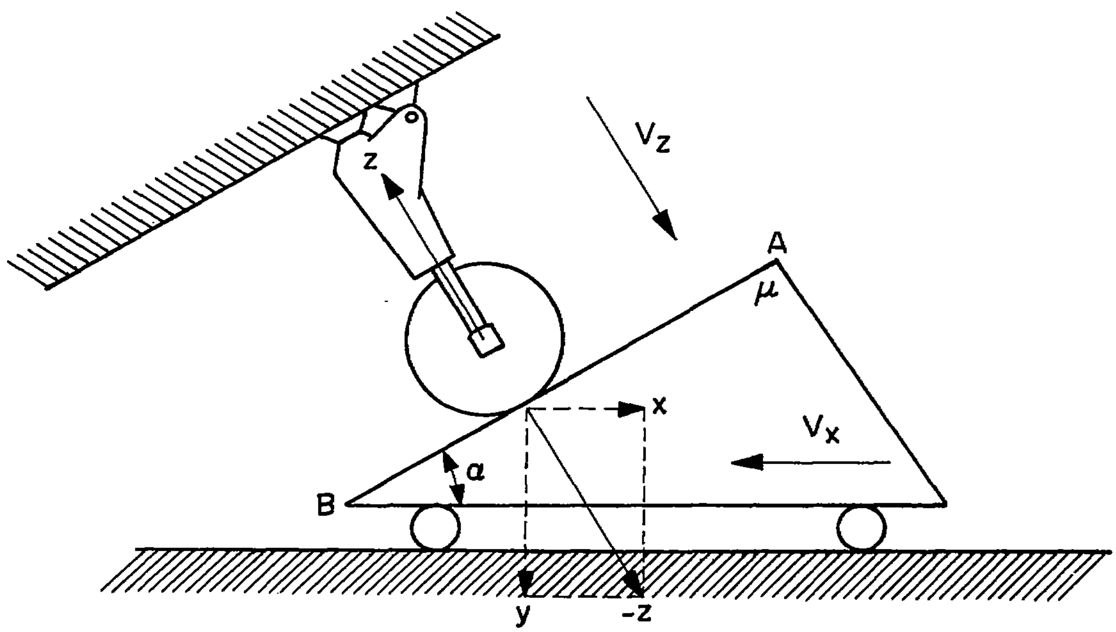


Figure 1.

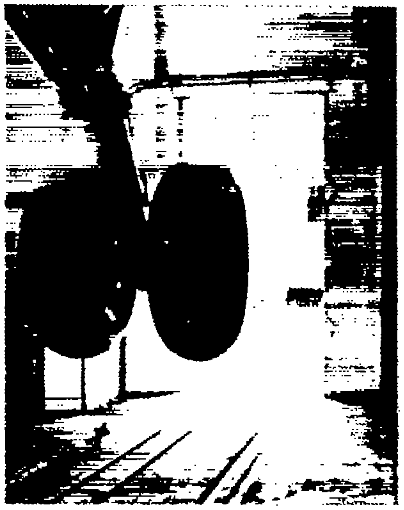


Figure 2(a).

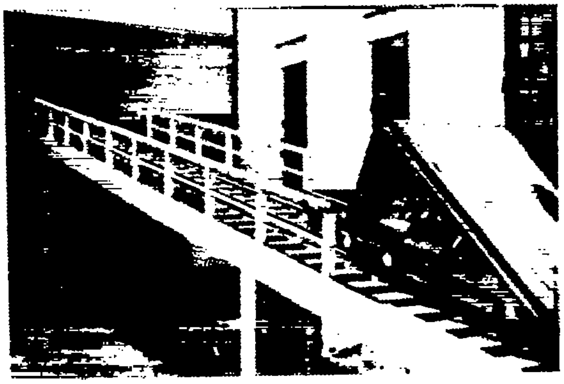


Figure 2(b).

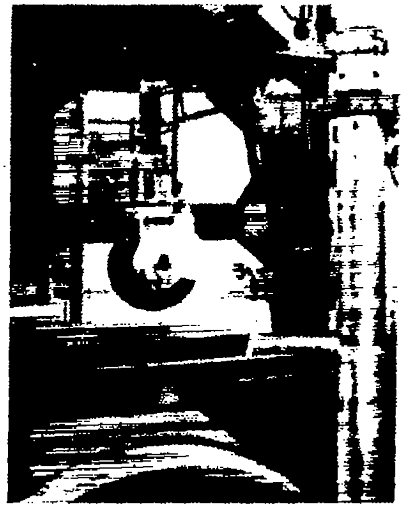


Figure 3(a).

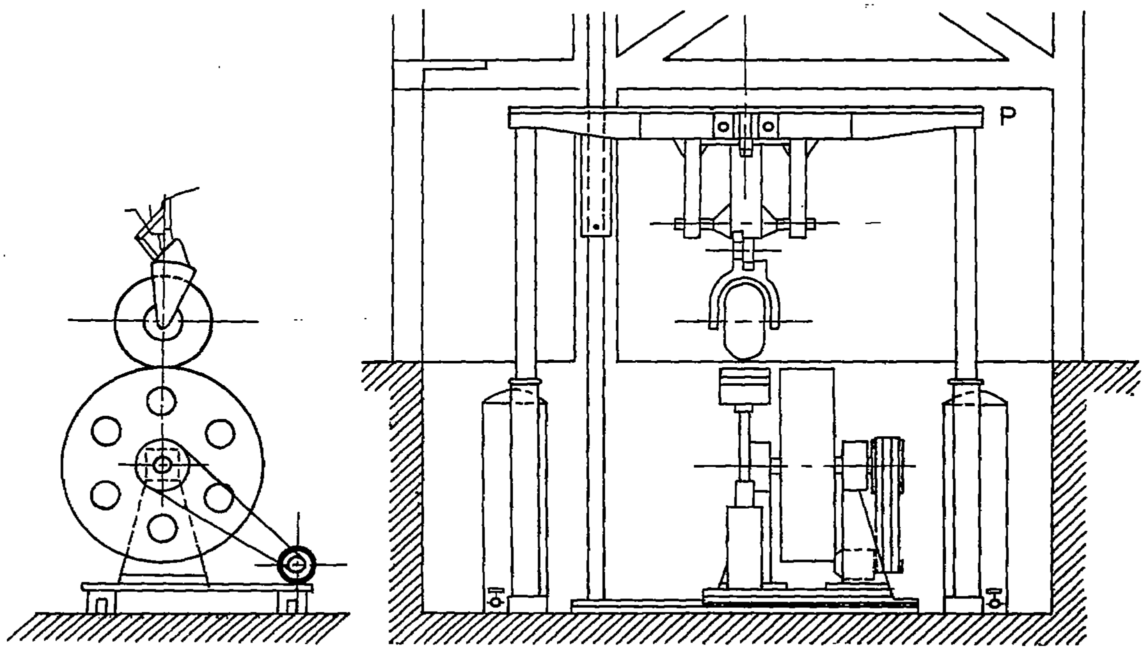


Figure 3(b).

Figure 4(a).

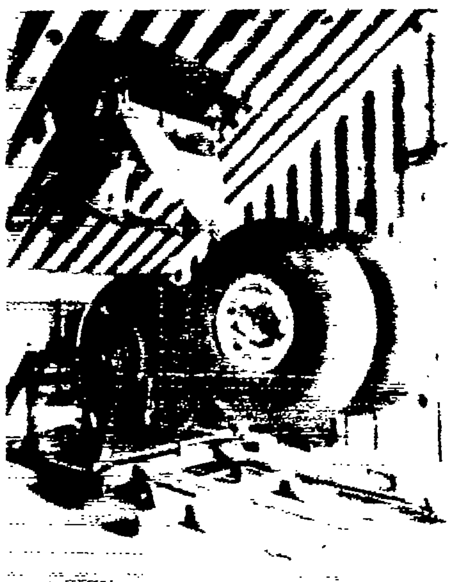


Figure 4(b).

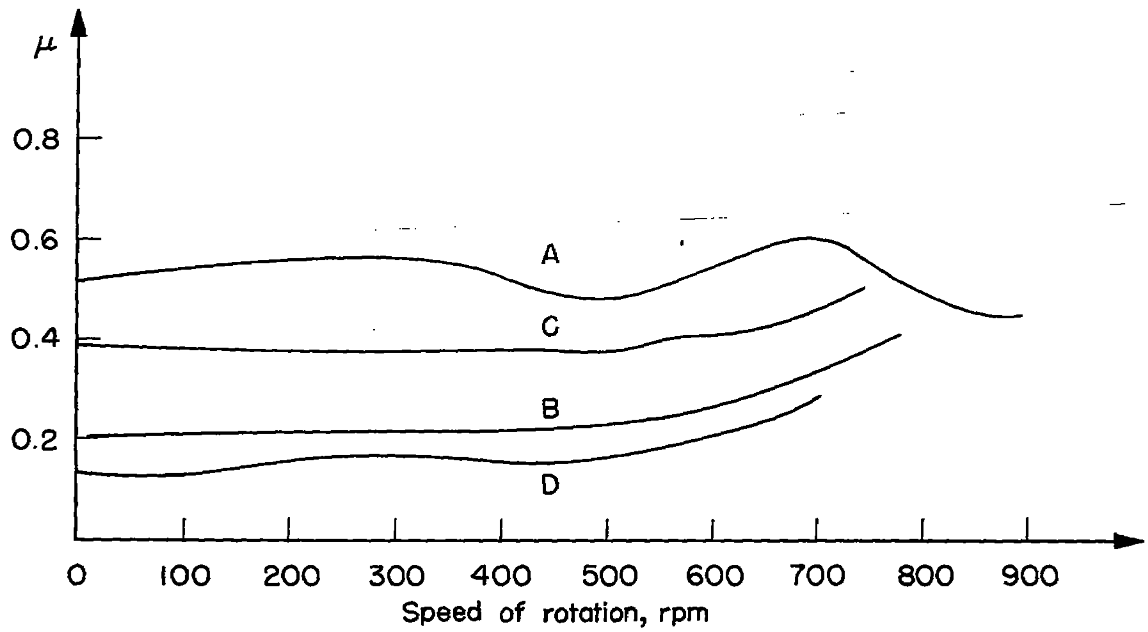


Figure 5.

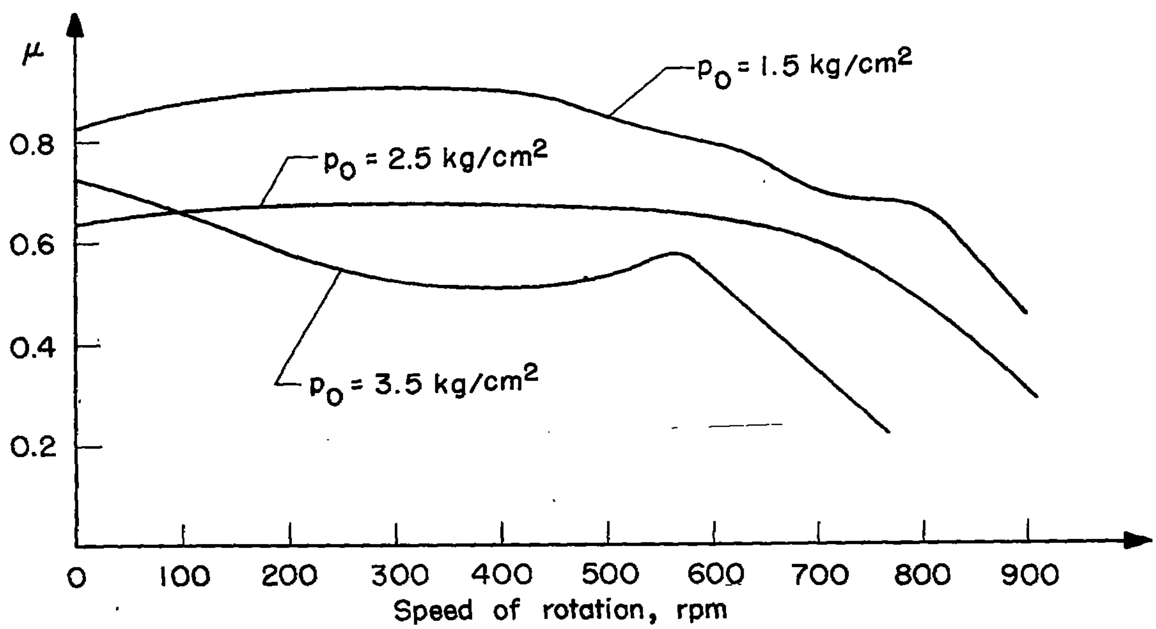


Figure 6.

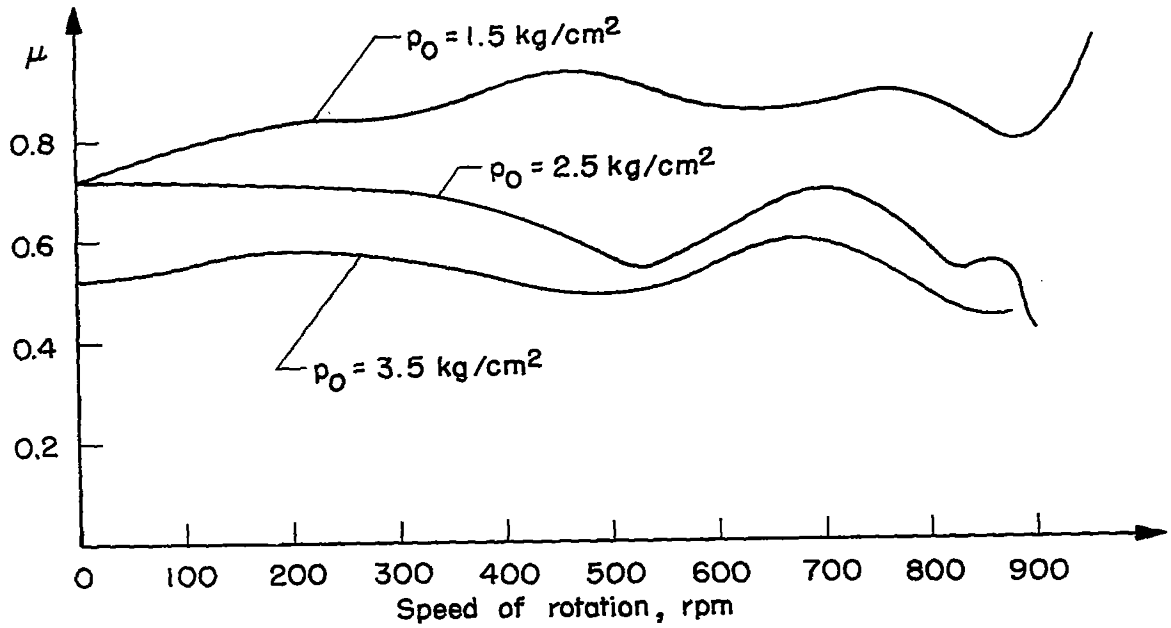
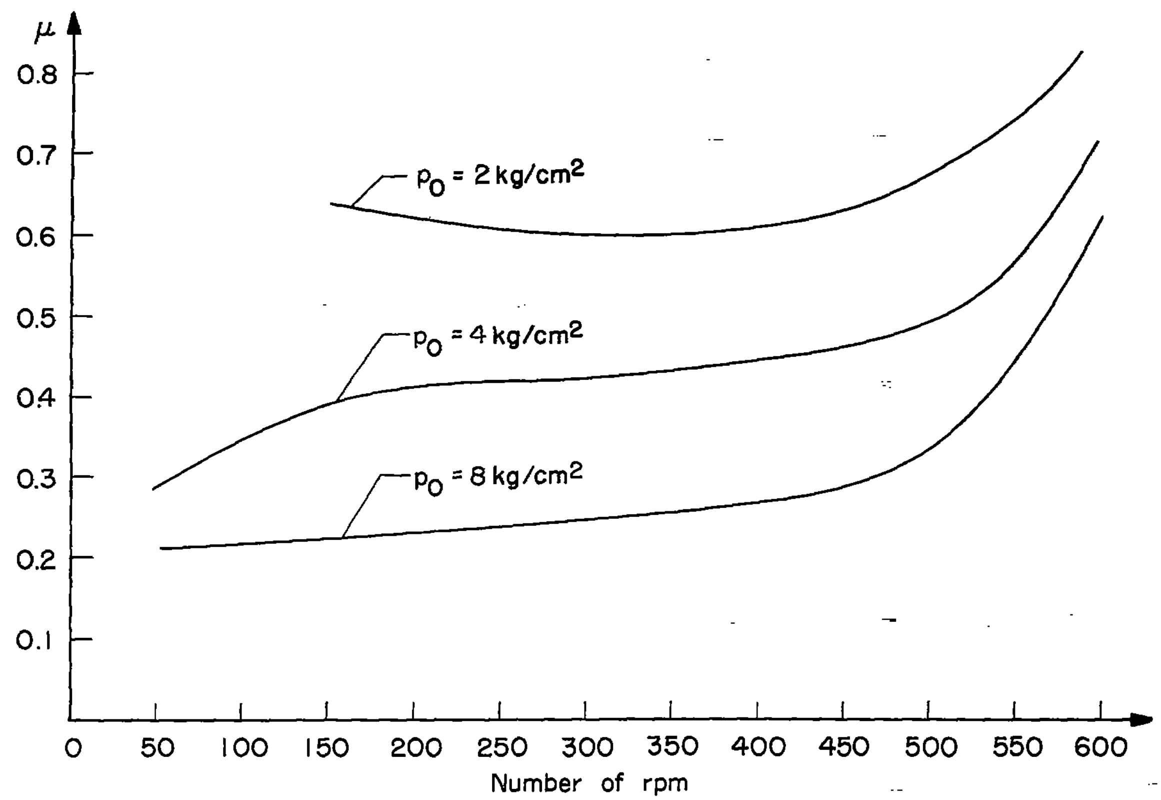

μ
0.8
0.6
0.4
0.2
$p_0 = 1.5$ kg/cm^2
$p_0 = 2.5$ kg/cm^2
$p_0 = 3.5$ kg/cm^2
0
100
200
300
400
500
600
700
800
900
Speed of rotation, rpm

Figure 7(a).



μ
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
$p_0 = 2\ kg/cm^2$
$p_0 = 4\ kg/cm^2$
$p_0 = 8\ kg/cm^2$
0
50
100
150
200
250
300
350
400
450
500
550
600
Number of rpm

Figure 7(b).

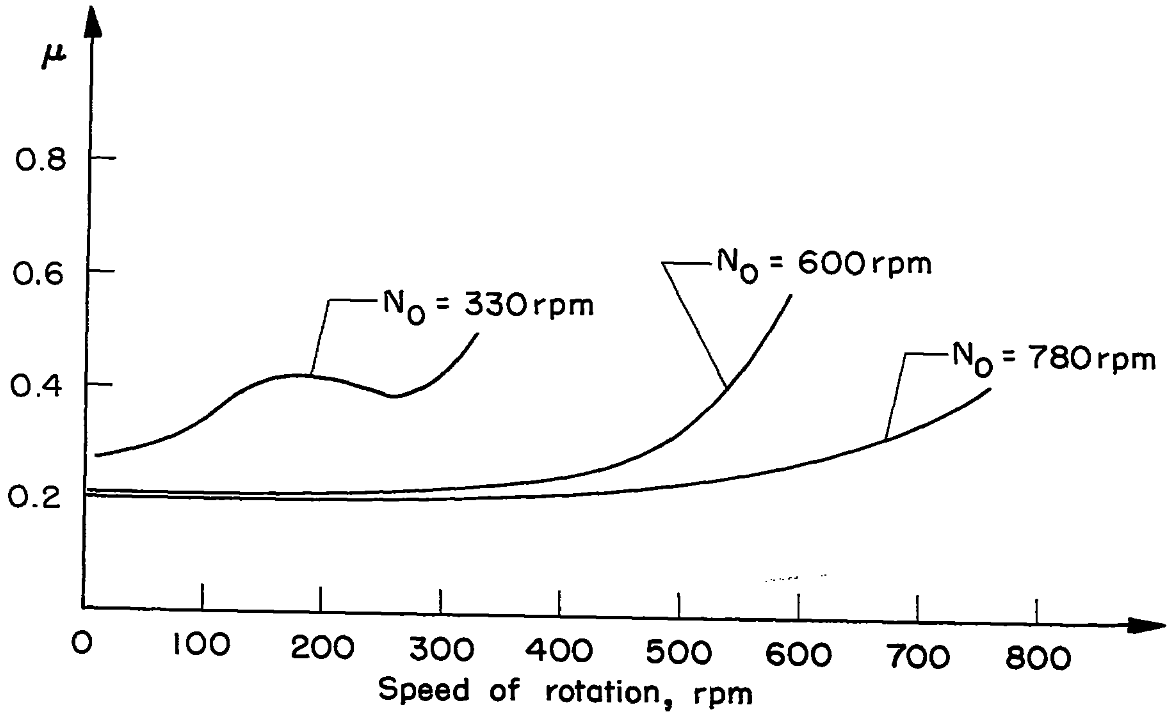


Figure 8.

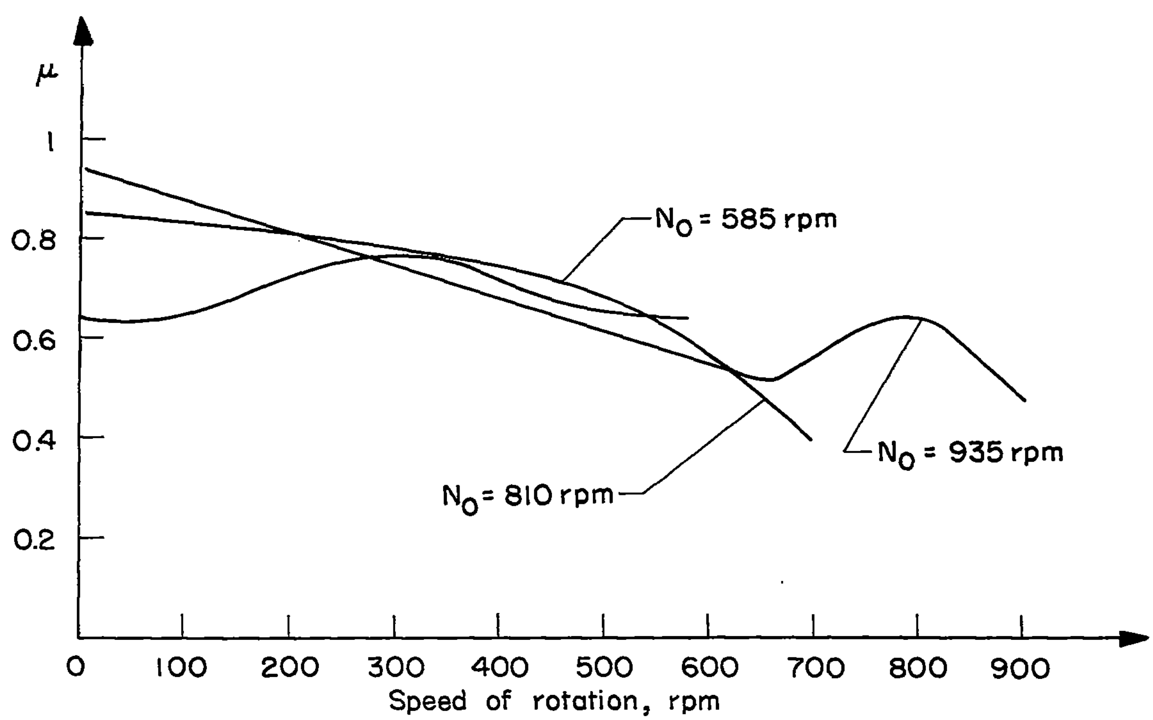


Figure 9.

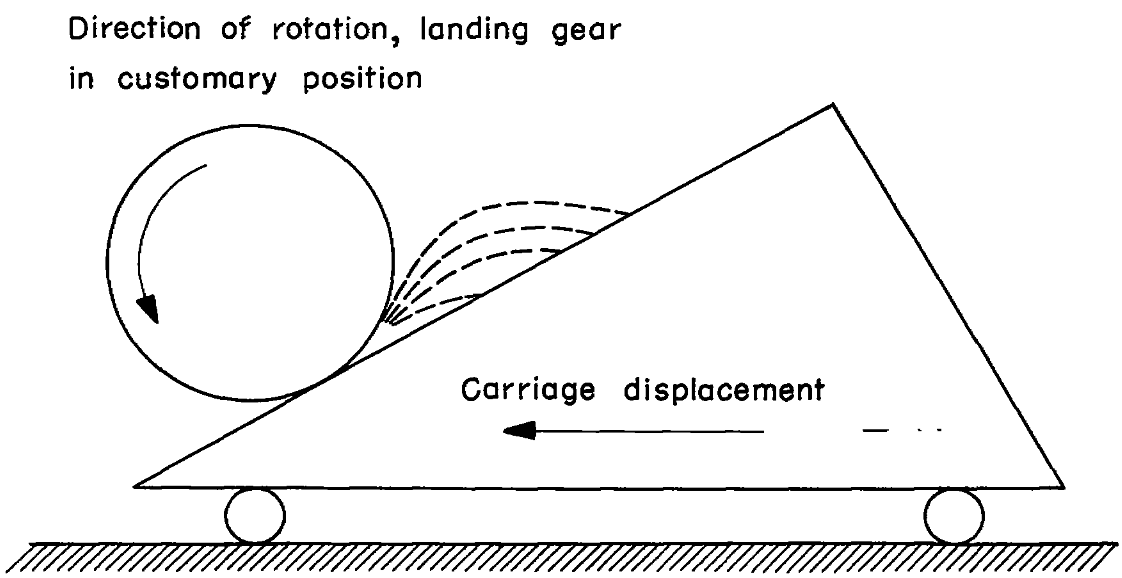


Figure 10.

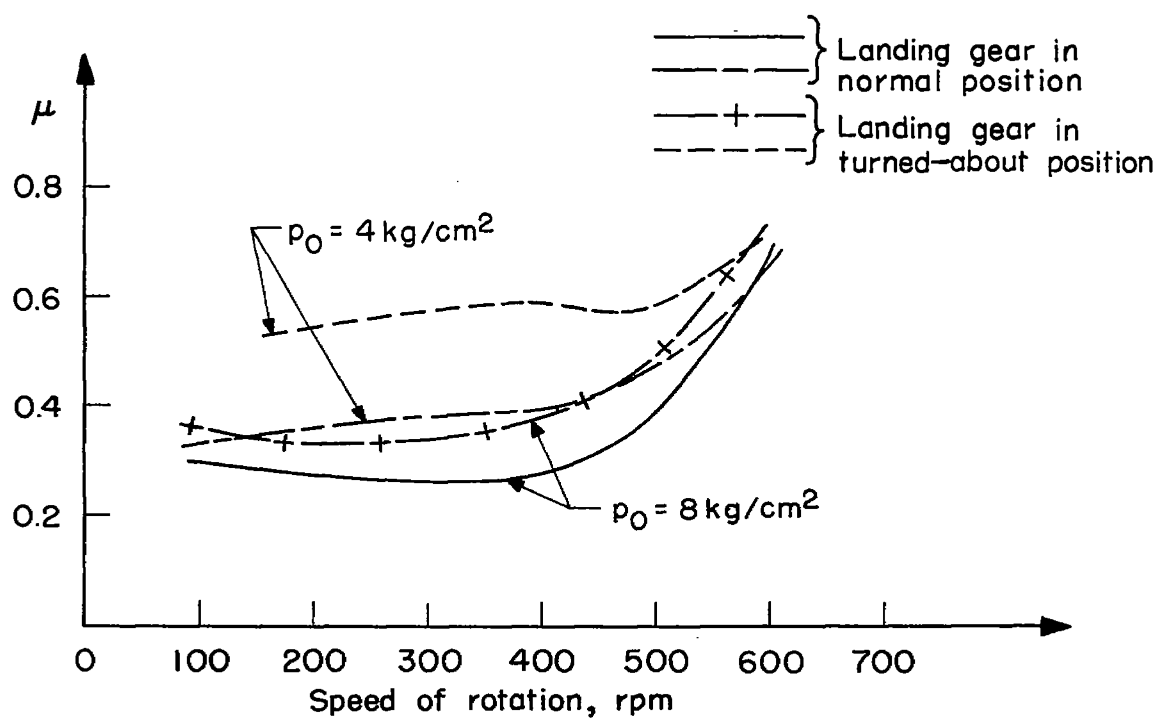


Figure 11.- Influence of throwing off melted rubber.

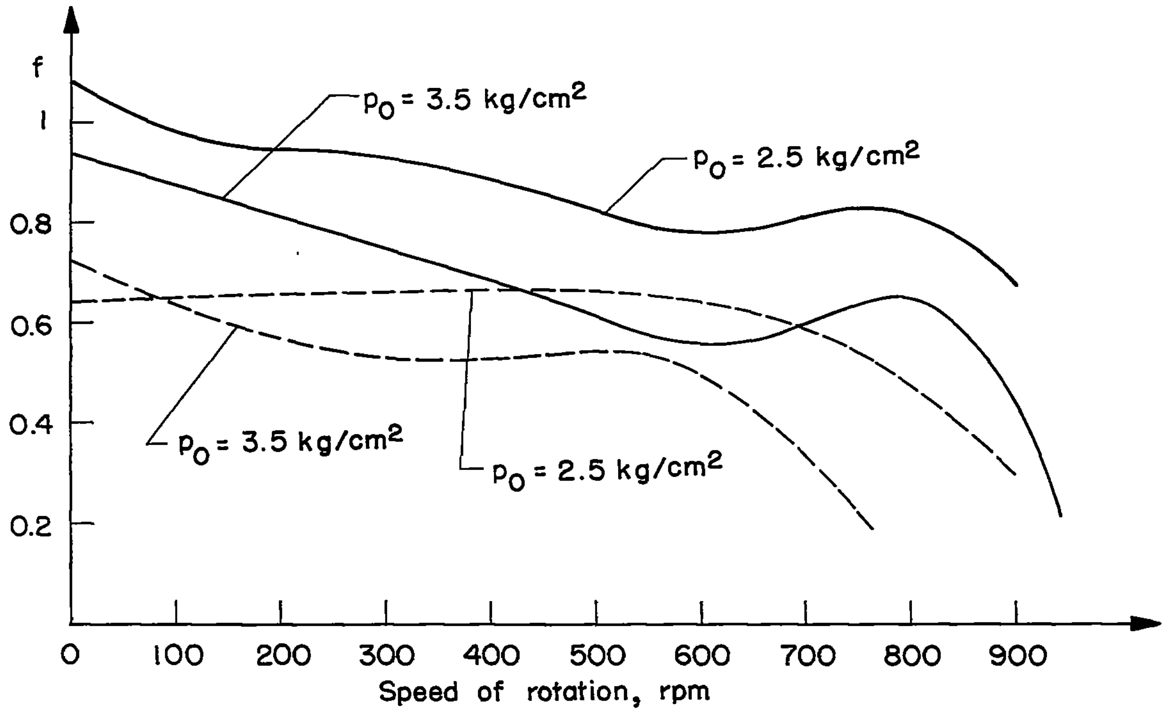


Figure 12.

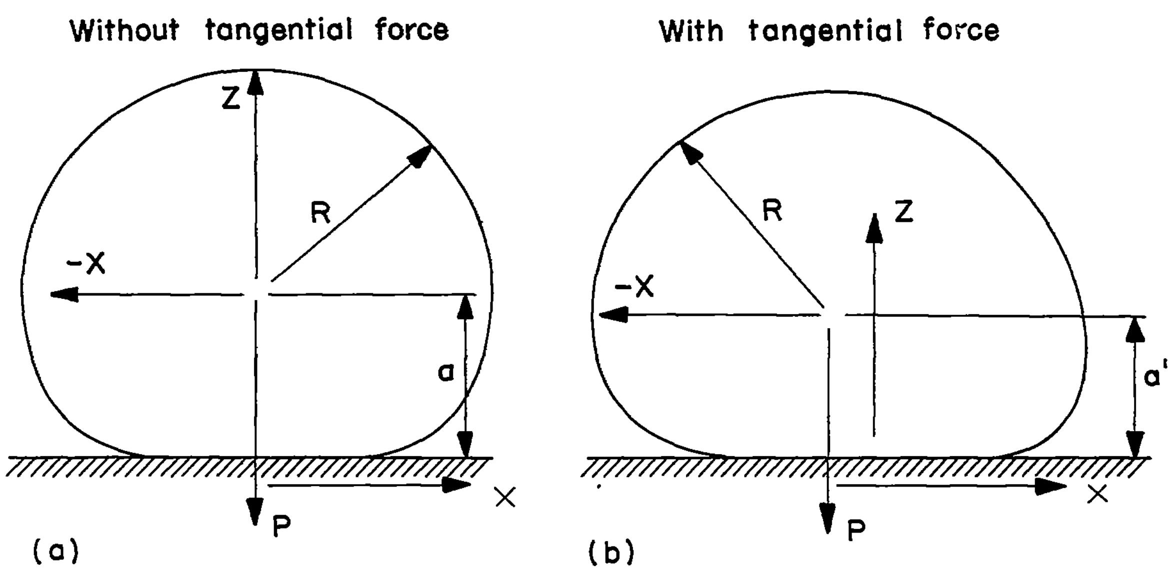
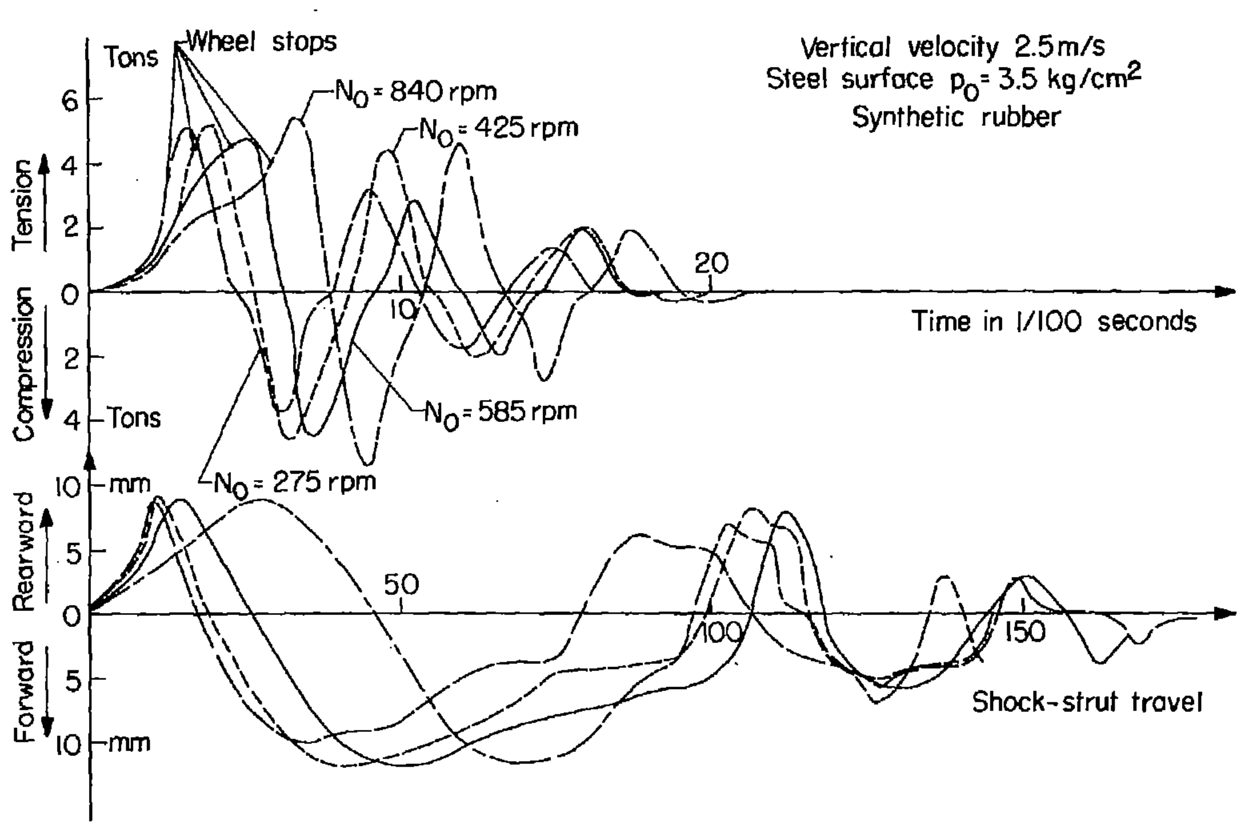


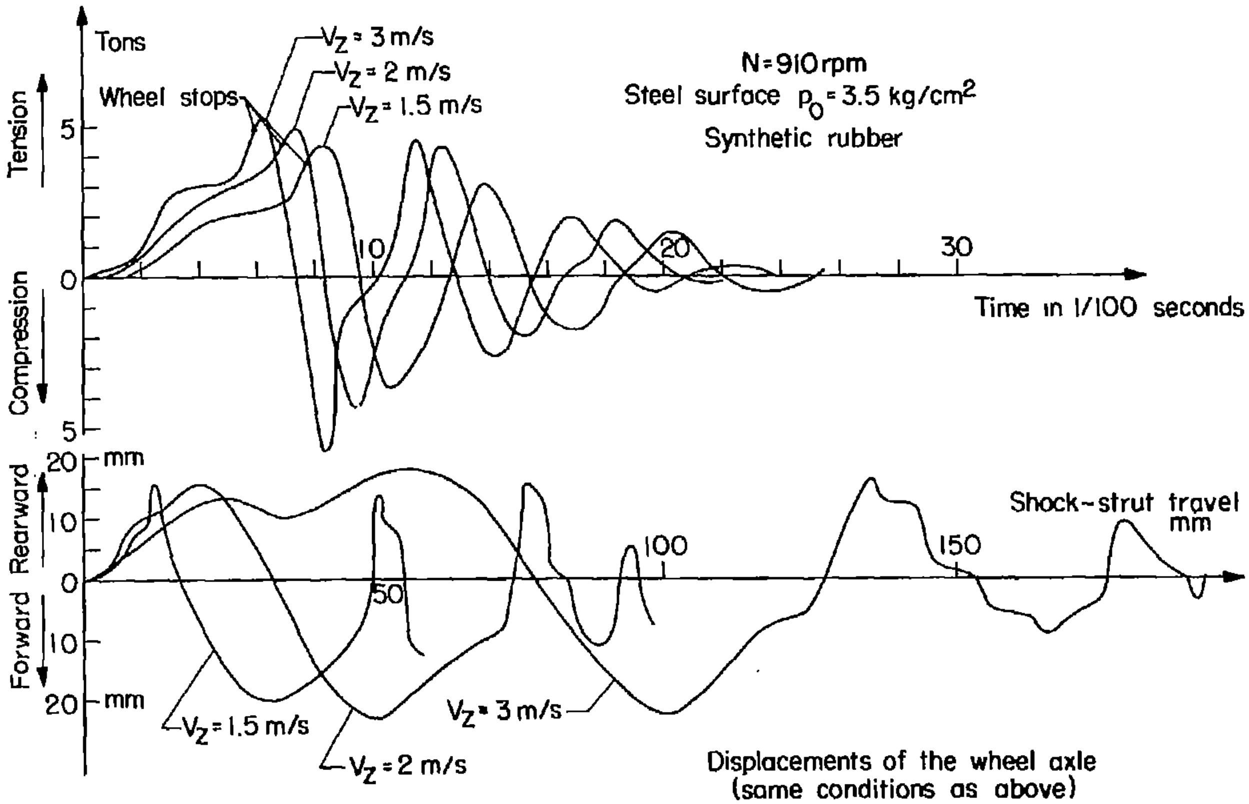
Figure 13.- Deformation of the tires.



(a) Forces in the drag strut as a function of time for different values of the speed of rotation.

(b) Displacements of the wheel axle as a function of the shock strut travel for the same values of the speed of rotation as above. (Steel surface, synthetic rubber: $V_z = 2.5$ m/s, $p_0 = 3.5$ kg/cm^2.)

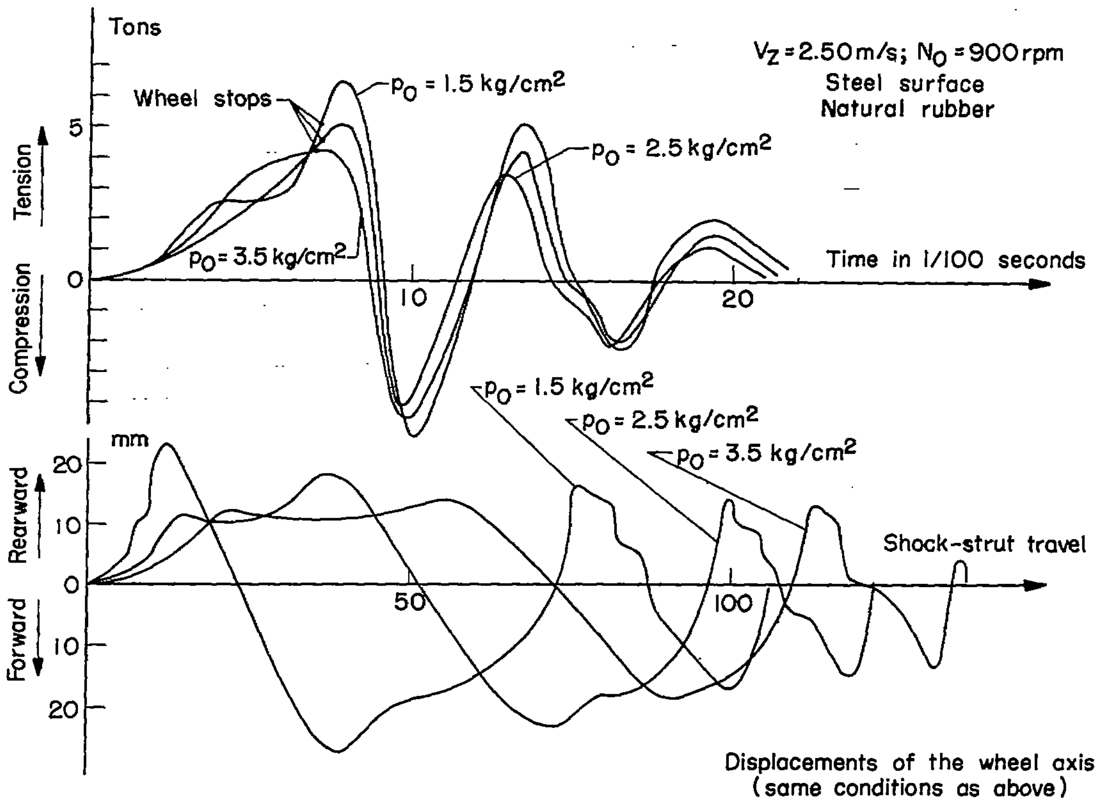
Figure 14.



(a) Forces in the drag strut as a function of time for different values of the vertical impact velocity.

(b) Displacements of the wheel axle as a function of the shock strut travel for the same values of the vertical impact velocity as above. (Steel surface, synthetic rubber: N_0 = 910 rpm, p_0 = 3.5 kg/cm^2.)

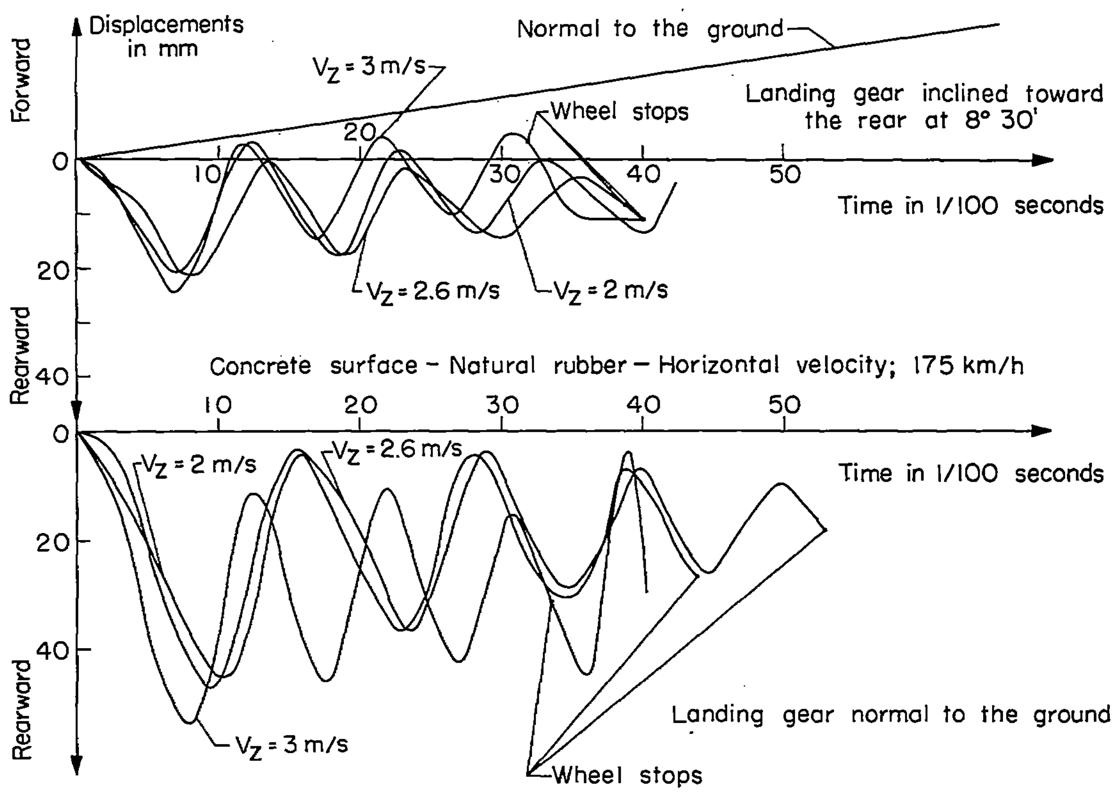
Figure 15.



(a) Forces in the drag strut as a function of time for different inflation pressures.

(b) Displacements of the wheel axis as a function of the shock strut travel for the same values of inflation pressure. (Steel surface, natural rubber: V_z = 2.5 m/s, N_0 = 900 rpm.)

Figure 16.



(a) Case of a landing gear inclined toward the rear at 8°30 .

(b) Case of a landing gear normal to the ground.

(a) Case of a landing gear inclined toward the rear at 8°30'.

(b) Case of a landing gear normal to the ground.

Figure 17.- Displacements of the wheel axis as a function of time for different values of the vertical impact velocity. (Steel surface, natural rubber: horizontal velocity, 175 km/h.)

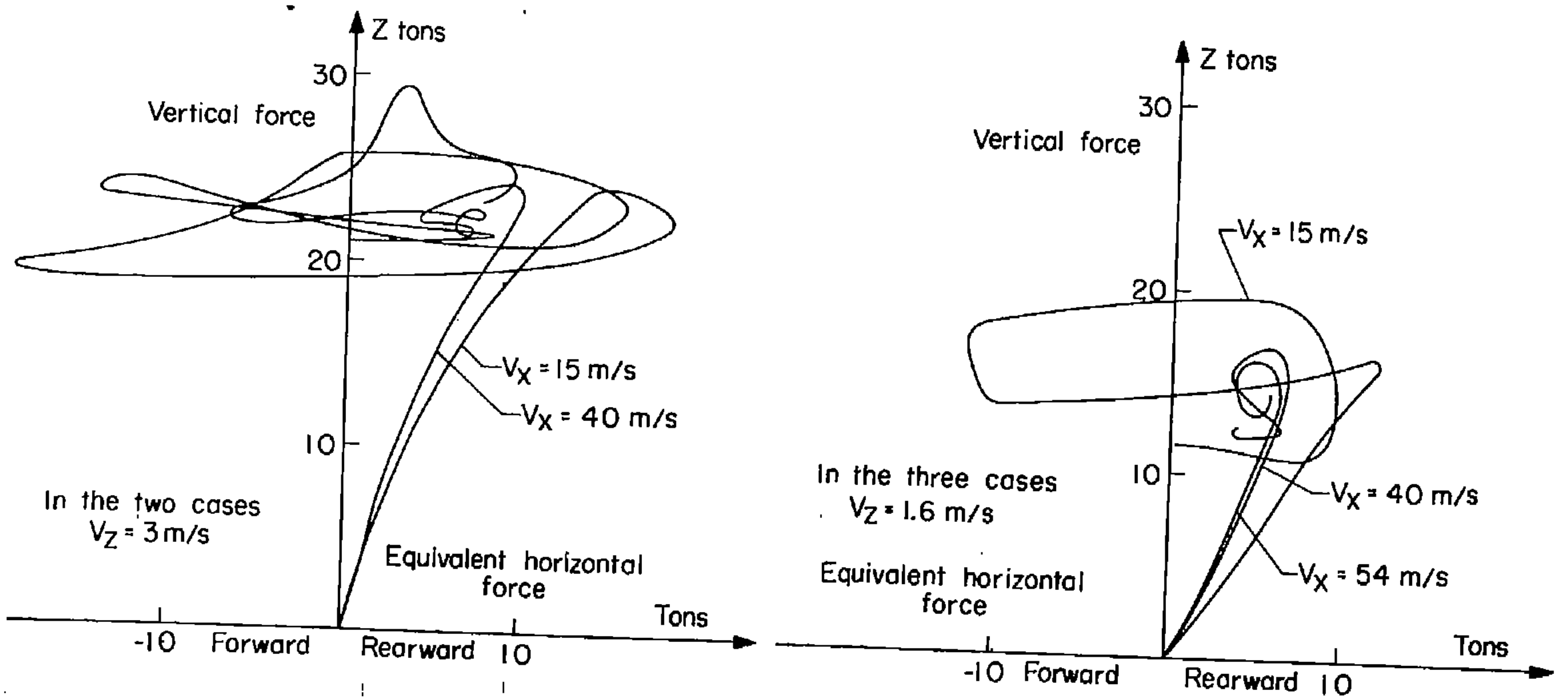

Z tons
30
Vertical force
20
$V_X = 15$ m/s
$V_X = 40$ m/s
10
In the two cases
$V_Z = 3$ m/s
Equivalent horizontal force
Tons
-10 Forward
Rearward 10
Z tons
30
Vertical force
$V_X = 15$ m/s
20
In the three cases
$V_Z = 1.6$ m/s
10
$V_X = 40$ m/s
$V_X = 54$ m/s
Equivalent horizontal force
Tons
-10 Forward
Rearward 10

Figure 18.

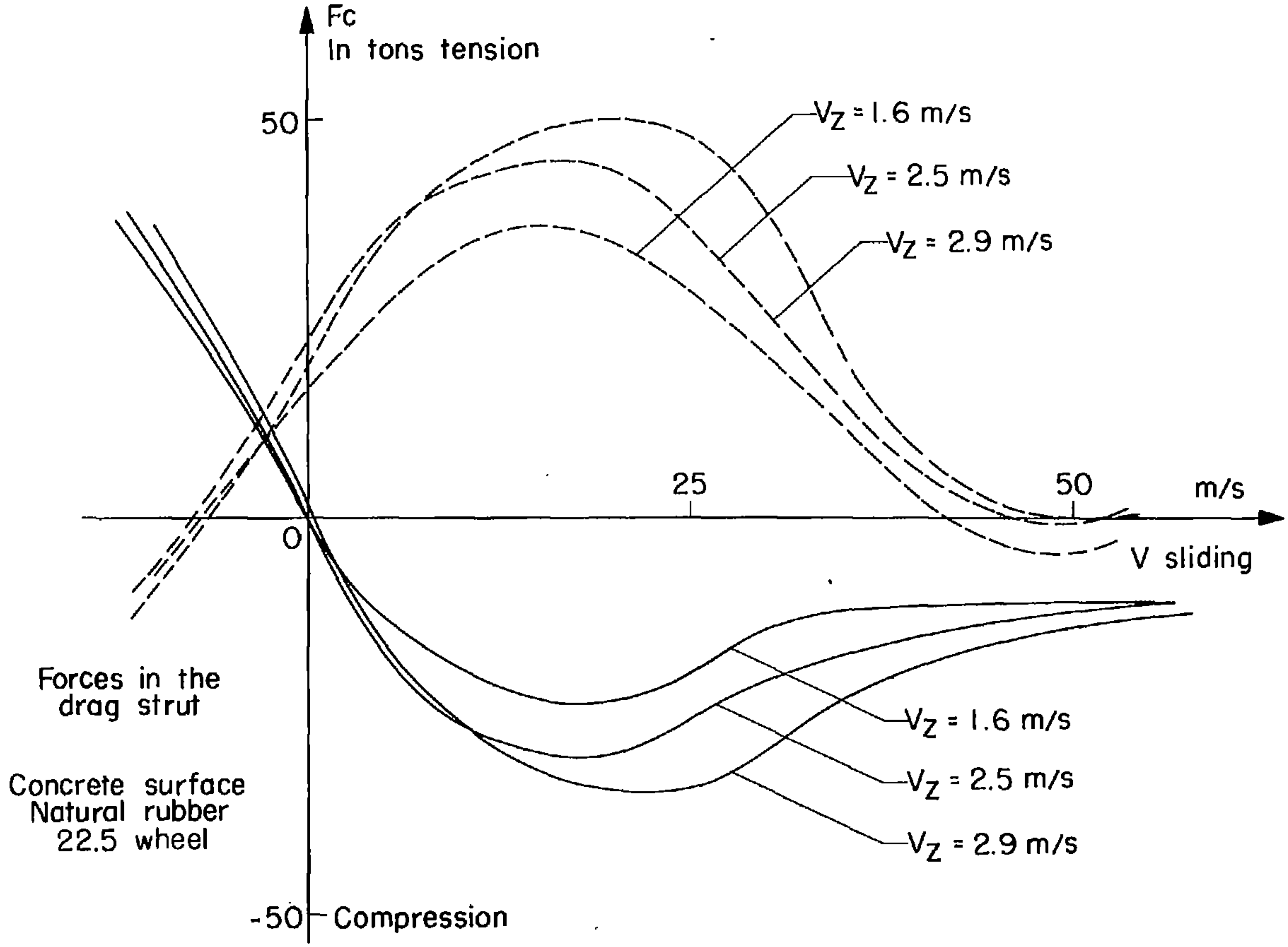

Fc
In tons tension
50
-50
Compression
0
25
50
m/s
V sliding
V_Z = 1.6 m/s
V_Z = 2.5 m/s
V_Z = 2.9 m/s
V_Z = 1.6 m/s
V_Z = 2.5 m/s
V_Z = 2.9 m/s
Forces in the drag strut
Concrete surface
Natural rubber
22.5 wheel

Figure 19.

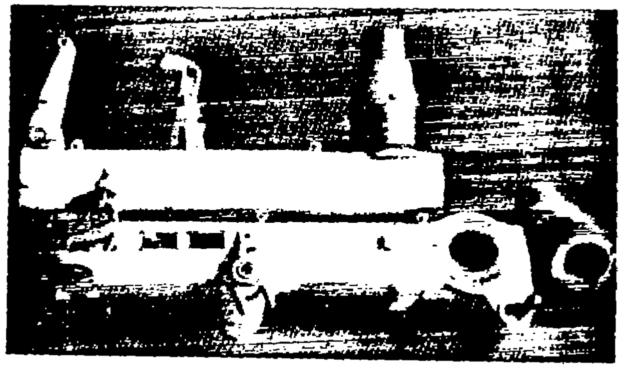


Figure 20.

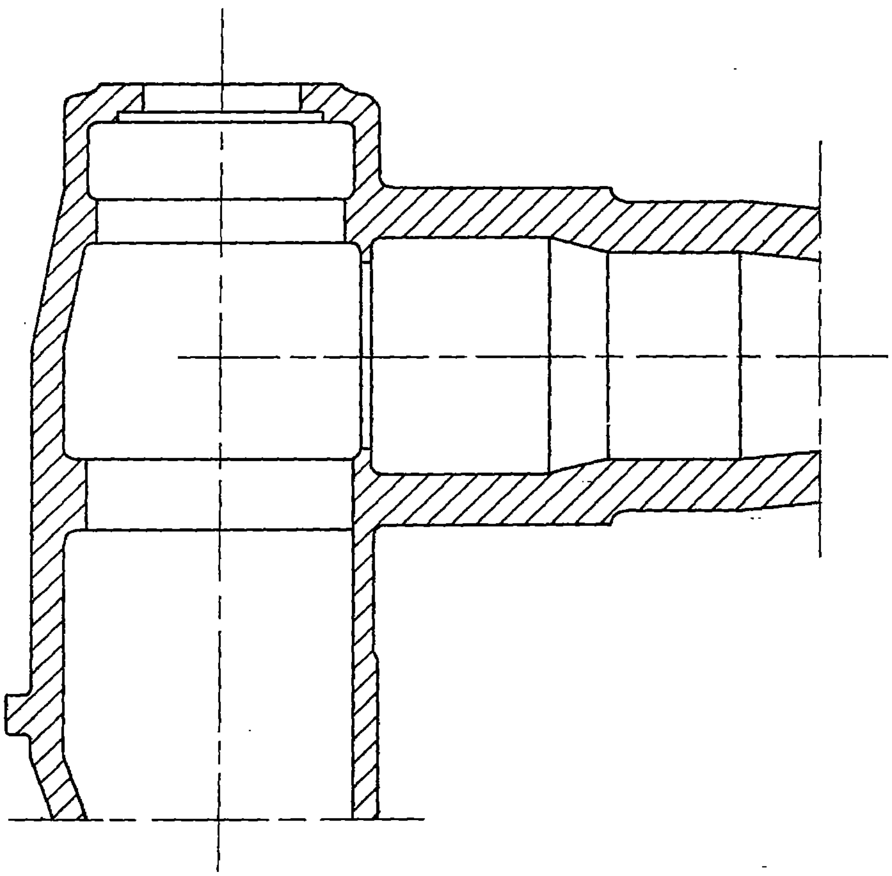
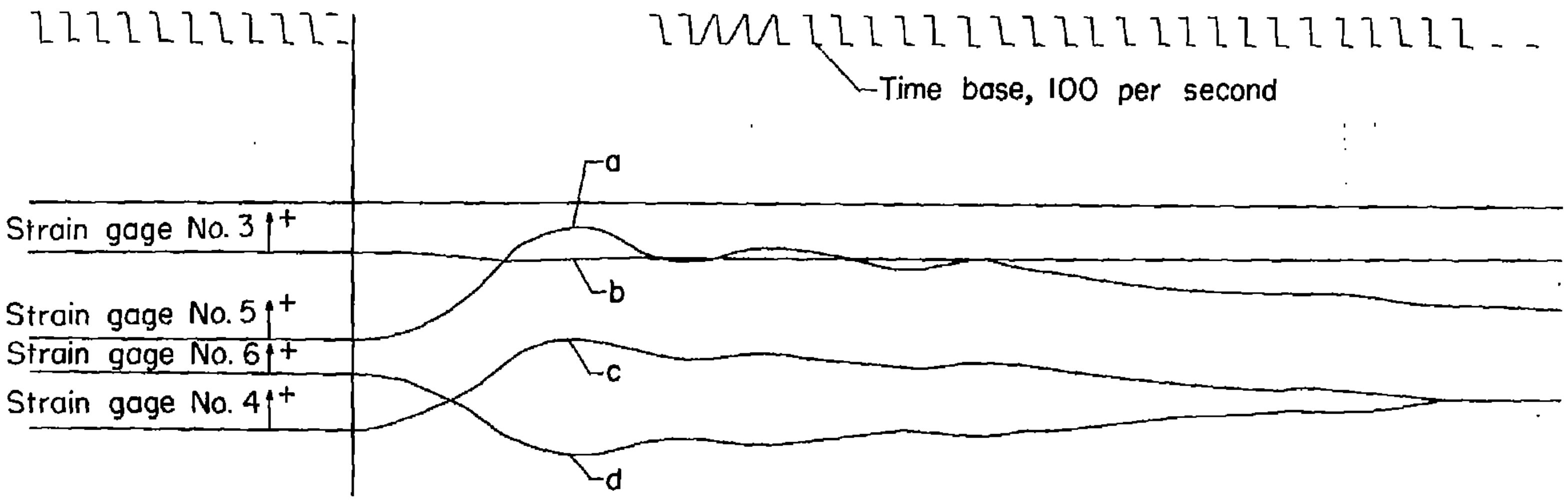


Figure 21.

Figure 22.

Figure 23(a).

Figure 23(b).



+Tension

-Compression $\frac{\Delta l}{l}$ $\begin{cases} a = +3.1 \times 10^{-3} \text{ (strain gage with 10-mm base)} \\ b = -0.17 \times 10^{-3} \\ c = +1.9 \times 10^{-3} \text{ (strain gage with 20-mm base)} \\ d = -1.7 \times 10^{-3} \end{cases}$

Figure 24.

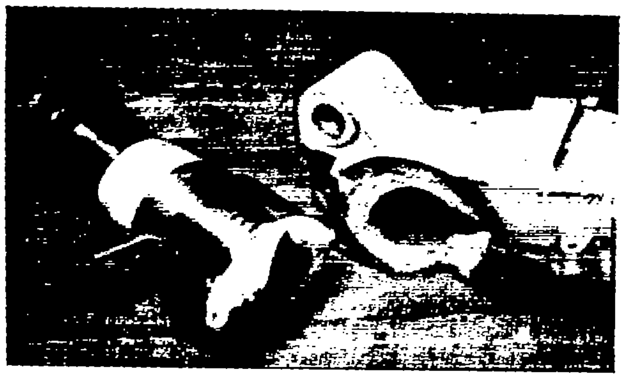


Figure 25.

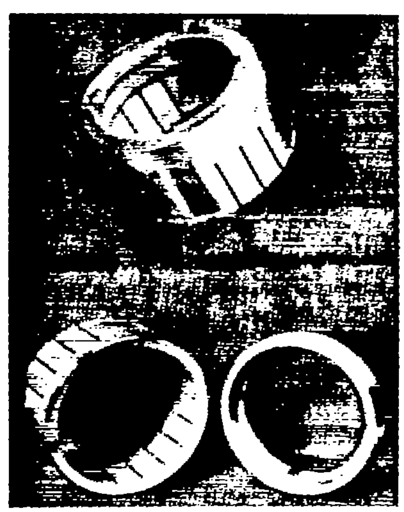


Figure 26.

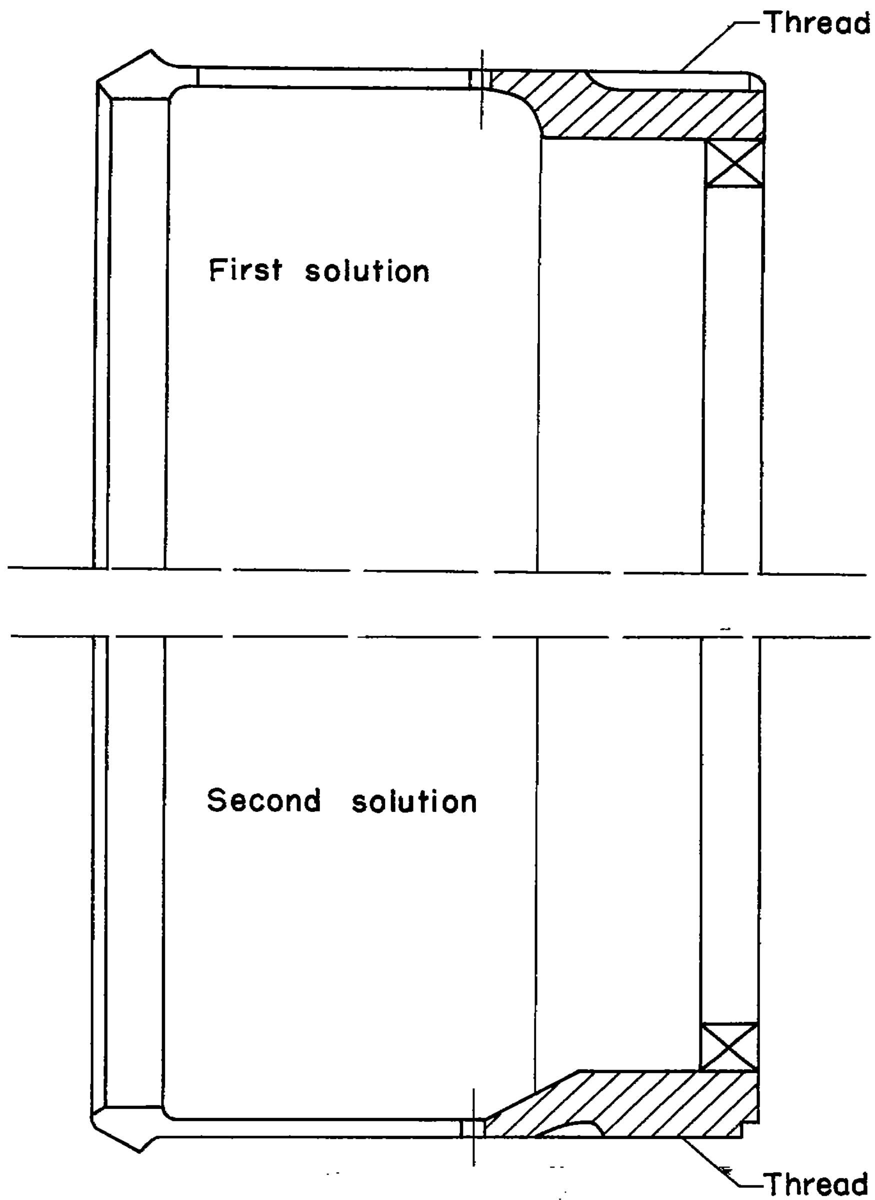


Figure 27.

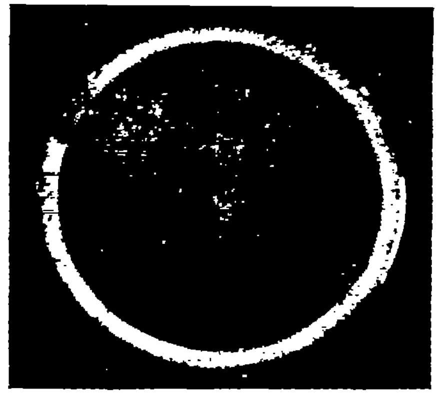


Figure 28.

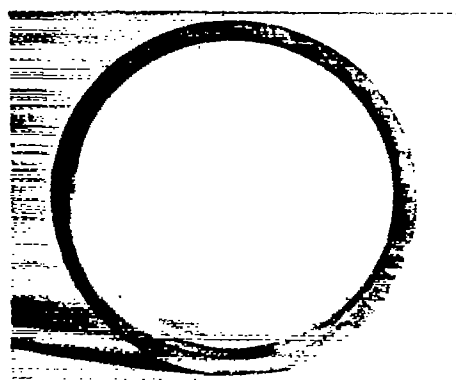


Figure 29(a).

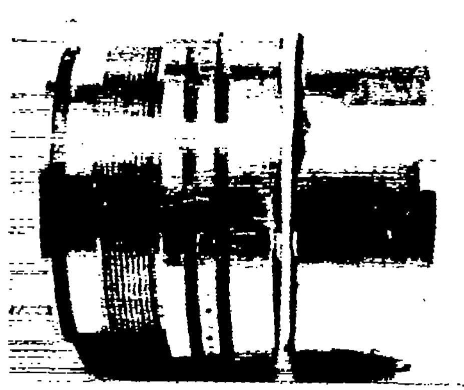


Figure 29(b).