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EXPERIMENTAL EVALUATION OF LOW-BAND-PASS LANDING-GEAR
SHOCK ABSORBER FOR PULSE LOADINGS

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SUMMARY

An experimental investigation was conducted to determine the practicality of constructing a low-band-pass landing-gear shock absorber and to evaluate its effectiveness in reducing rapidly applied loads. The band-pass principle was first introduced in NACA Technical Note 3803 from a theoretical point of view. For the experimental investigation presented herein, a low-band-pass shock strut was designed, constructed, and tested in conjunction with a high-pressure tire which was used in order to develop and transmit steep pulses. Taxiing runs were made over individual bumps to determine the strut pulse rate response and over consecutive closely spaced bumps to determine strut recycling characteristics. Duplicate tests were made with a comparable shock absorber having a fixed main orifice area. Design details are presented for the shock struts tested and also for a proposed band-pass vibration absorber. The loads for the low-band-pass strut for single rapidly applied pulses were 50 percent of the loads for the fixed-orifice strut whereas for cyclic loading the low-band-pass loads were less than 25 percent of the fixed-orifice strut loads. Smaller reductions were achieved for rapidly applied pulses superposed on slowly applied pulses. There was no attempt in this investigation to develop an optimum control unit to minimize loads for both single and multiple superposed pulses over the entire spectrum of load application rates. The load reductions achieved, however, were accomplished by the addition of a $1\frac{1}{2}$ -pound control unit to the shock strut of a 5,000-pound airplane.

INTRODUCTION

This paper deals with the experimental evaluation of the band-pass principle as applied to a landing-gear oleo-pneumatic shock absorber hereinafter termed "shock strut." The application of this principle was originally considered from a theoretical point of view in reference 1 as a means of overcoming some of the limitations of conventional shock absorbers. For example, in the case of aircraft operation on a land or water runway of a given roughness, as the aircraft speed increases,

the rate of rise of the tire bottom over the bump increases with a corresponding increase of the load application rate. Since the force transmitted by a conventional landing-gear shock strut increases roughly as the square of the telescoping velocity, these struts tend to become very rigid; thus, severe shock loads are developed and transmitted to the aircraft for high rates of change of loading. The low-band-pass shock absorber considered herein was designed to overcome this limitation and still retain the characteristics of a conventional shock strut for low rates of load application such as the landing impact and taxiing over long hills or swells. This is accomplished in the following manner. First, the low-band-pass shock strut is rate sensitive and develops a pulse response function, with force attenuation occurring at the high rate end of the pulse spectrum. This strut also has the characteristic of being pulse-length sensitive; this property in combination with the rate sensitivity makes it have a frequency response type of characteristic since both the rate of rise and the duration of any given cycle in a vibration are necessary to define the frequency. The pulse-rate response of the band-pass strut alleviates rapidly applied loads whereas the pulse-duration response prevents this strut from excessive telescoping during the application of sustained rapidly applied pulses or during discontinuous velocity changes associated with passing from one steady slope to another.

Although band-pass shock absorbers can be made sensitive to rate and duration or frequency, depending on the application, it was decided that the theory could be checked by testing either type. Therefore, the former was chosen in the form of a landing-gear shock strut. Because very rapid action of mechanical parts is required in order for a low-pass landing-gear shock strut to attenuate rapidly applied pulse loads, some questions naturally arose as to whether such a device was practicable. In order to answer these questions and to obtain an idea of the problems involved in making such a device functional, a low-band-pass control unit was designed, constructed, and installed in a conventional shock strut. This modified strut has been tested in combination with a high-pressure tire by running it over bumps on the runway of the Langley impact basin. Similar tests were made with an equivalent fixed-orifice shock strut for comparative purposes. This paper presents the results of these experiments which were aimed at determining whether the band-pass principle could be successfully applied to a practical mechanical problem.

DESCRIPTION OF APPARATUS

The general plan followed in order to evaluate the merits of a low-band-pass shock absorber as a load-controlling device involved the

conversion into a band-pass absorber of a fixed-orifice landing-gear shock strut obtained from a propeller-driven military trainer of the 5,000-pound class. The behavior of this band-pass strut was then compared with a more or less conventional arrangement of this same strut having a fixed orifice area during a series of taxiing and landing tests over specific bumps. The overall test setup is described first, after which attention is focused on the design and operating principles of the band-pass shock strut, and then a brief description of the measuring instrumentation is given.

Test Setup

A photograph of the test setup showing the landing gear in the process of taxiing over a bump on the runway is presented as figure 1. This ramp-shaped bump, which is 1 foot high and 10 feet long, was used to duplicate the effects of the design landing without the complications of spin-up drag loads by merely taxiing the gear over it. The wheel and tire are from a 25,000-pound fighter airplane and the shock strut and fork are from a 5,000-pound trainer airplane. The landing-gear fitting is fastened to the carriage boom above it. The strut was rigidly mounted in this fitting and was adjusted for each pitch angle by inserting links of different lengths between the boom and fitting at the rear of the fitting. The front of the fitting was pivoted from the boom on both sides to give a three-point suspension. The carriage boom was free to rise and fall vertically as the carriage traveled forward on parallel rails over the runway. The entire upper mass weighed 2,250 pounds, whereas the effective weight associated with the entire lower mass was about 110 pounds. The tire was a type VII high-pressure tire with dimensions of 26 inches by 6.6 inches and was inflated to pressures from 75 pounds per square inch (soft condition) to 300 pounds per square inch (hard condition) during the tests. This hard tire condition was used in order to produce rapid load applications to the strut during the negotiation of short steep bumps. This high-pressure tire is not matched to the light strut, and it would have been desirable to utilize a complete stiff fighter landing gear because of the high natural ground frequency of the wheel and tire in the vertical direction; however, the impact basin carriage could not handle the loads transmitted by such a gear. The ground resonance frequency of the fighter wheel and tire was of the order of 60 cycles per second with the tire fully inflated. This stiffness caused the application of rapid transient loadings to the trainer strut in the same manner as if the fighter shock strut had been used. Although the loads were limited to lower values, the results have significance in that the important criterion satisfied was that the rates of application of loads were not limited to those of the flexible gear. With this combination, therefore, it was possible to investigate band-pass-strut behavior qualitatively and to determine whether the strut response is fast enough

to be used with a high-speed fighter airplane with small hard tires. The trainer-airplane fork used in these tests was not as stiff as might have been desired since, under the application of the structural design load of 12,500 pounds, it allowed the wheel to deflect upward relative to the strut about $3/8$ inch. The natural frequencies in the axial direction of the combination of the carriage boom and the landing-gear attachment fitting were roughly 100 cycles per second and 300 cycles per second. With the exception of the 1- by 12-inch and 2- by 12-inch rectangular plank bumps used in the landing-impact tests, the bumps utilized in the taxiing tests were shaped similar to the ramp shown in figure 1. Their heights and the lengths to the point of maximum height were as follows: 1 inch by 10 inches, 4 inches by 20 inches, 3 inches by 12 inches, and 4 inches by 10 inches. All were of solid wood construction with the exception of the 4- by 10-inch bump which was covered with a $1/4$ -inch metal plate, and all the bumps were about 2 feet wide. The instrumentation shown in figure 1 will be described subsequently.

Shock-Strut Design

General Construction.- The general construction of the band-pass shock strut is shown in figure 2(a) with the control unit in position. In order to convert the trainer shock strut to the band-pass configuration, it was necessary to unscrew the piston, drop out the original orifice plate and snubber, insert the band-pass control unit in place, and replace the piston. It was also necessary to remove the metering pin which was screwed into the base of the strut. The strut shown in the figure was filled with hydraulic fluid (type MIL-O-5606, AM.2) while in the completely compressed position after which it was extended fully and pressurized with air through the top. Although "O" ring seals have been substituted for the original chevron seal in figure 2(a), both types were used in the tests. In fact, in order to prevent the shock strut from failing because of rapid reextension with subsequent bottoming after the wheel rode off the far edge of a bump, the chevrons were used in many tests and were tightened to an excessive compression by screwing the bearing nut down hard in order to increase strut friction and to reduce the reextension rate. When the "O" ring seal was used, a rubber buffer (which is not shown) was installed between the "O" ring cage and the bearing immediately below it in order to reduce the forces arising from metal-to-metal impact at full reextension. For a more complete description of the original trainer shock strut arranged as a fixed-orifice strut, see reference 2.

Control-unit design.- An exterior view of the instrumented control unit is shown in figure 2(b) and the interior is illustrated in figure 3. Figure 3(a) shows a cross section of the assembly and figure 3(b) shows a photograph of the internal parts. The alinement vanes shown

along the sides of the control unit in figures 2(b) and 3(a) center this unit in the piston-supporting tube of the shock strut. The many holes visible in the sides of the control unit in figure 2(b) are for the purpose of transferring fluid rapidly from the outside to the inside of the unit in the neighborhood of the main spring. The small tube rising from the top of the unit contains a sealed cable for carrying the signals supplied by the instrumentation to the outside of the shock strut. The two displacement pickups shown in figure 2(b) are described subsequently.

The three principal functions of this control unit are to provide the shock absorber with the following:

(1) A variable main orifice size which is controlled by the rate of application and the duration of strut loading

(2) A strut dump valve, which in this case was combined with the main orifice valve, for allowing fluid to return rapidly from the upper strut cylinder to the lower cylinder to facilitate rapid recycling of the strut under repeated load conditions

(3) A frequency-discriminating snubber valve which snubs low-frequency rebounds such as for normal landings but does not snub shock-strut reextensions for high-frequency pulses; thus, a large part of shock-strut stroke is retained for each new bump encountered.

These three functions will be described in the following section with the aid of figures 3 and 4 beginning with the rate-sensitive variable orifice.

The most elementary type of variable-orifice valve which might be considered for the instantaneous relief of loads is the one shown in figure 4(a) which is simply an ordinary pressure-actuated relief valve. For this valve, as the strut telescopes, pressurized fluid in the lower hydraulic cylinder drives the plunger upward, opens the orifice wider, and thereby reduces the load. The disadvantage of this valve is that it cannot discriminate between different loading rates and dumps the load, even for low rates of load application such as those for the landing impact for which the aircraft vertical momentum would not be dissipated as desired.

In figure 4(b) is shown a simplified sketch of a rate-actuated valve, such as was incorporated in the low-pass strut tested. In this device, the plunger position is primarily controlled by the rate of change of pressure in the lower cylinder rather than directly by the magnitude of this pressure. Its action is as follows: For low rates of loading, as the pressure slowly increases in the lower cylinder, fluid is forced up through the main orifice, and also through the restricted tube, into the control cylinder. Since the rate of change

of pressure is small, fluid can pass through the restricted tube fast enough to fill the control cylinder, force the control piston down against the spring, and thus maintain the plunger balanced in its equilibrium position as shown. For this case, the strut behaves as a conventional fixed-orifice shock strut.

For high-frequency loads involving rapid rates of change of pressure, the fluid cannot flow through the restricted tube fast enough to move the control piston very far. As a result, the control piston remains substantially fixed, and the plunger is driven upward against the spring, opens the orifice, and reduces the load in much the same way as for the ordinary relief valve in figure 4(a). The plunger, main spring, control piston, and restricting tube shown in figure 4(b) may also be noted in figure 3.

The second function of the control unit is accomplished by the dump valve system which permits the rapid recycling of the strut between repeated load cycles. When the strut in figure 2(a) telescopes together during the application of a load pulse, the pressure in the lower main cylinder is greater than that in the upper cylinder or air chamber until the strut starts to reextend. From this point on, the pressure in the upper cylinder becomes greater than the pressure in the lower cylinder. When this condition occurs fluid is forced downward (see fig. 3) past the snubber disk, which is assumed to be in the up position, and then continues downward through the annular orifice formed by the plunger and the main orifice plate. Part of this fluid also passed inward around the main spring through the perforations in the control-unit cage, where it exerts a downward force on the plunger and an upward force on the control piston. In figure 3(a) it is seen that there are two openings to the control cylinder, one communicating to the lower main cylinder through the restricting tube and the other opening into the upper main cylinder through the top of the control cylinder. These orifices are so proportioned that the pressure in the control cylinder is adjusted by them to a value somewhere between the lower main cylinder pressure and the upper main cylinder pressure while the strut is telescoping closed. For this same condition the ratio of the control piston area to the plunger effective area is in turn adjusted so that, during the existence of a slowly applied pulse, the upward force on the plunger resulting from the lower cylinder pressure is slightly more than balanced by the downward force on the control piston resulting from the control-cylinder pressure communicated through the main spring.

During the reextension stroke of the shock strut when the lower main cylinder pressure is low, the high pressure in the upper cylinder exerts a net upward force on the plunger piston system, since the control-piston area is considerably larger than the plunger effective area. Therefore, the plunger is withdrawn from the main orifice during

strut reextension; this action allows fluid to flow rapidly downward from the upper to the lower main cylinder (assuming the snubber disk is in the up position); and the strut is thus allowed to reextend rapidly in readiness for the next compressive load pulse.

The third function of the control unit involves the operation of the frequency-discriminating snubber valve. The snubber valve in the proposed design was made a part of the control unit for easy shock-strut modification although it could have been located elsewhere in the strut as is often done in shock-absorber design. This particular snubbing system, shown in figure 3, consists of an annular snubber disk or valve connected by means of a spider to the snubber lag piston which is a close sliding fit in the annular cylinder formed between the upper section of the main orifice plate and the control-unit support cylinder. This snubber operates in the following manner.

When a loading pulse causes a pressure increase in the lower main strut cylinder, fluid flow upward through the annular main orifice impinges on the snubber disk, which in this case was made of lightweight aluminum, and immediately drives it upward against its seat at the base of the control-unit cage. Since the disk is connected to the snubber lag piston by means of the spider, the disk, spider, and piston are carried upward as a unit until the snubber is completely withdrawn from its annular cylinder. Thus a gap is opened at the top of this annular cylinder through which fluid enters and rapidly fills it. When the strut begins to reextend, the fluid in the upper cylinder is forced downward through the main orifice and drives the snubber disk back down to its closed position against the main orifice plate. The disk carries with it the snubber lag piston which is forced down into its cylinder and compresses the entrapped fluid which escapes primarily through the one or more snubber lag orifices shown in the main orifice plate. Since there is a certain time constant connected with emptying this cylinder, the strut is allowed to reextend rapidly during this short time after which the snubber closes and greatly reduces the amount of fluid returning to the lower main cylinder through the very small clearance annulus between the snubber disk and the plunger. Thus the rebound of the strut is effectively snubbed for the remainder of the extension stroke.

Thus, for high-frequency repeated impulses usually involving small strut displacements of, for example, an inch or so, the strut is unsnubbed on reextension so that the strut stroke lost during each compression is largely regained during each reextension. However, for pulses having low rates of application like the landing impact or during taxiing over long gently sloped bumps, both of which involve large strut displacements, most of the rebound stroke is effectively snubbed, the result being that the aircraft is not bounced off the runway by the potential energy stored in the air spring of the upper main cylinder.

In order to convert the band-pass shock strut to the identically comparable fixed-orifice strut, it was only necessary to lock the plunger down in its equilibrium position in which it appears in figure 2(a). This was done by inserting a rigid hollow cylinder inside the control cylinder extending from the top of this cylinder down to the control piston, and replacing the main spring by another rigid hollow cylinder which was as long as the uncompressed spring. Because of difficulties experienced with the snubber, such as sticking (due to imperfect machining of the assembly-line shock strut used in the tests and/or possibly from an inadequate length-to-diameter ratio of the snubber) and short circuiting of the snubber position switch (not described), this investigation was conducted with the frequency-discriminating snubber removed from the strut.

As a matter of information, the following control-unit specifications are pertinent. The effective flow area of the main orifice (plunger seated) was about 0.08 square inch. The control piston had a net cross sectional area of 0.65 square inch with effective orifice areas of 0.03 square inch for up flow and 0.04 square inch for down flow (to permit more rapid dumping of the control cylinder). The control-cylinder orifice had an effective area of 0.021 square inch. The plunger net cross-sectional area was 0.47 square inch. The main spring had a spring constant of about 1,150 pounds per inch with a full travel of 0.6 inch.

Instrumentation

Most of the external landing-gear instrumentation used in these tests is visible in figure 1. Since the landing-gear structure was very flexible, linear accelerometers were placed on the gear to measure the axial (approximately vertical) accelerations as follows. One of these accelerometers which measured the acceleration of the associated wheel mass was placed inside the axle on the center line of the wheel. This instrument had a natural frequency of 240 cycles per second and a range of 100g. Two accelerometers are placed at the base of the shock strut to measure the acceleration of the associated lower cylinder mass. One had a natural frequency of 250 cycles per second and a range of 25g, whereas the other had a natural frequency of 500 cycles per second and a range of 100g. These two instruments were intended to check each other for frequency response and structural oscillation errors. Two accelerometers, not visible, were also located on the top of the landing-gear attachment fitting adjacent to the strut. These had natural frequencies and ranges, respectively, of 85 cycles per second and 12g and 400 cycles per second and 50g. An additional pair of matched accelerometers were located at the base of the boom itself and were called boom accelerometers with natural frequencies and ranges of 85 cycles per second and 12g.

A slide wire (fig. 1) was used to measure the strut telescoping displacement and was actuated by the rod connected at the top of the fork. This rod was also cut as a gear rack to drive the telescoping velocity generator shown attached to the base of the slide wire. This slide wire was believed to have a flat response for frequencies up to 60 cycles per second. The telescoping velocity generator was estimated to have a flat response for frequencies up to 500 cycles per second.

The pressure gage utilized to measure the air pressure in the upper cylinder is visible near the top of figure 1 and is also shown in figure 2(a). This gage had a range of 1,500 pounds per square inch and was believed to give flat response to a frequency of 500 cycles per second. The pressure gage mounted at the base of the lower cylinder is not visible in figure 1 but can be seen in figure 2(a). This pressure gage had a range of 3,000 pounds per square inch and was believed to give flat response to a frequency of 1,300 cycles per second. Other external instruments, not shown, were as follows: a boom slide wire which measured the vertical displacement of the carriage boom and was believed to provide a flat response up to a frequency of at least 100 cycles per second, a velocity generator to measure the boom vertical velocity which was believed to be flat to a frequency of 500 cycles per second, a photoelectric cell which viewed tabs placed at a 1-foot interval along the track to provide a measure of the horizontal displacement of the main carriage. This displacement measurement was also used in conjunction with the timing lines provided on the oscillograph record to determine forward speed of the carriage.

The internal instruments may be viewed in figure 2(b). They consisted of two inductive linear displacement pickups, one of which indicated plunger position in the control unit and the other of which indicated the position of the control piston in this unit. These pick-ups were believed to possess a flat frequency response up to 600 cycles per second. They were checked for interactive errors due to the relative motion of the various metal parts in the control unit and for immersion in oil and were found to be unaffected by either. The electrical energy to and from these pickups was carried in a cable sealed in a metal tube which passed through the top of the strut down along the left inside wall of the piston supporting tube to the control unit. (See figs. 2(a) and 2(b).)

RESULTS AND DISCUSSION

Since the main purpose of this investigation was to demonstrate that the band-pass principle could be applied to the landing-gear problem, it was believed that the simplest means of evaluating the performance would be

through a comparison of the results for the band-pass shock strut with those for the fixed-orifice shock strut. These results are presented in time-history form and as maximum force values in figures 5 to 9. The maximum values of the more important measurements are also listed in table I for the low-pass strut and in table II for the fixed-orifice shock strut. It should be borne in mind that these maximums did not necessarily occur at the same instants of time in a given run. It might be mentioned that the lower cylinder maximum pressures given in the tables were obtained by fairing through the oscillations on the records. The main orifice area was calculated for the instant of maximum strut telescoping velocity by means of Bernoulli's equation. Values of main orifice flow area much smaller than 0.08 square inch (plunger-seated value) indicated that foam or air were passing through the orifice instead of oil although some variation above this value probably occurred with oil flow alone because of pressure recovery above the necked section in the orifice.

The different types of comparisons presented under this section are subdivided into separate subsections so that each property of the band-pass strut can be considered in its entirety before going on to the next property. First, results of taxi tests over single bumps are considered from which the pulse-rate-response comparison can be made. Next the time histories of the various parameters during passage over a single bump, from which a clearer understanding of the control mechanism functions can be achieved, are reviewed. The results of taxi tests over repeated bumps are then presented from which the recycling capability of the low-pass strut between pulses can be ascertained. Finally, the effects of landings on single bumps are discussed.

Pulse-Rate-Response Comparison

In order to test the pulse-rate response or band-pass characteristics of the low-pass shock strut, taxi runs were made for a range of speeds over several different bump sizes at several tire inflation pressures. The fixed-orifice strut was then taxied over these same bumps with an attempt made to cover the same speed variation until the speed was attained at which the design load for the strut was reached. In most cases the low-pass strut could be taxied over the bumps at the maximum carriage velocity whereas the fixed-orifice strut was limited to low speeds for the steep high bumps. An effective wing lift of 0.88g which might be considered to represent high aircraft taxi speeds was used on all taxi tests in this investigation. Actually, this was done to protect the equipment against damaging loads which might have been encountered if the strut ever bottomed during the tests. Several pulse-rate response plots drawn from these data are presented. The first of these is shown in figure 5 which presents maximum axial shock-strut force measured during taxiing over a bump 2 inches high by 10 inches long which was shaped as

an inclined plane and rounded at the far end to prevent cutting the tire. Although the inclined plane type of bump would at first glance appear to cause a low rate of load application, in effect it turned out to be rather severe and abrupt, since, when the tire contacted this bump, a large area of the tire came in contact with the bump almost at the instant of contact. A rectangular-shaped bump is not believed to be much more severe since it causes a progressive denting of the tire during passage over it and also because of the kinematics associated with the rolling of a rigid disk over a step-shaped bump.

The tire pressure in this test was 225 pounds per square inch which is considered to produce a fairly stiff tire. In figure 5(a) is presented a comparison of experimental maximum shock-strut forces derived from strut internal pressure measurements and in figure 5(b) are presented maximum forces derived from accelerometer measurements. In both cases conventional loads are denoted by the square symbols and low-band-pass loads by the circular symbols. As the taxi speed increases, the rate of increase of the applied load pulse increases since the time spent on the bump becomes smaller. Both plots show that the loads in the low rate of application regime are approximately the same for both struts up to about 20 or 30 feet per second as anticipated. These slowly applied loads were obtained at low taxi speeds and have rates of application equivalent to landing impacts. Actual landings made with both struts verified this agreement by developing equal loads for slowly applied pulses. The low-speed taxi runs actually were made over the 12- by 120-inch bump and were applied for the 2- by 10-inch bump by multiplying the actual forward speed of the test by the ratio of the slopes of the bump tested to the bump plotted to obtain the forward speed plotted. The reason for using the 12- by 120-inch bump instead of the 2- by 10-inch bump was that the carriage could not be taxied at a constant speed below about 20 feet per second as would have been required for the 2- by 10-inch bump. The equivalent speed for the 12- by 120-inch bump was higher and so could be used. An experimental check of this technique showed that it was allowable as long as maximum force was reached before the tire traveled more than 2 inches vertically up the 12-inch-high bump, which was the case in this test.

For the high rate of application regime where the band-pass action comes into play, the load developed remains constant as the taxi speed increases whereas the load developed by the fixed-orifice strut continues to grow larger. For the highest speed shown on the pressure derived plot, the low-pass strut developed only about 45 percent of the load for the fixed-orifice strut, whereas for the acceleration derived plot the low-pass strut developed only about 60 percent of the load for the conventional strut. Several reasons may be given to explain why the reduction in upper mass acceleration was not as large as the reduction in pressure. First, the effect of strut friction and binding loads was exceptionally high for this test and resulted from tightening the bearing nut (fig. 2(a))

on the strut to compress the chevron seal to prevent strut failure on complete reextension due to high-speed metal-to-metal impact as was mentioned previously. Second, the rigid-body acceleration of the boom was modified by the dynamic response of the flexible structure which had natural frequencies in the strut attachment fitting and boom combination of around 100 cycles per second which was in the same neighborhood as the pulse loads imposed on the conventional shock strut. The differences observed between the pressure-derived loads for the fixed-orifice and band-pass struts are believed to be more significant, however, than the differences observed for the accelerometer-derived loads because the pressure-derived loads are less dependent on the dynamics of the strut suspension and on the design of bearings and oil seals in any particular shock strut.

In figure 6 a set of plots similar to those of figure 5 are presented with the exception that a moderately low tire pressure of 75 pounds per square inch was used during taxi tests over a bump 4 inches high by 10 inches long. This case might be of interest for aircraft operating off of unprepared rough surfaces. In figure 6(a), a large difference of the pressure-derived loads at high velocity is again seen, the band-pass strut developing only about 50 percent of the load of the fixed-orifice strut at the highest velocity. A lesser reduction in the acceleration-derived loads is shown in figure 6(b), as expected, the maximum load developed by the band-pass strut amounting to about 75 percent of the fixed-orifice strut load. It is also evident, although only low-band-pass experimental data are available for the slowly applied loading regime, that an intersection point for the low-pass and fixed-orifice curves is indicated by the remainder of the data in the neighborhood of a taxiing velocity of 20 to 30 feet per second.

Control-Mechanism Functions

In order to furnish additional details of the band-pass action, figure 7 presents illustrative time-history records for one of the high-speed test conditions shown in figure 5.

The first significant point to note is that the plunger in the low-pass strut has opened the orifice wide (to effectively three times its original area) in only 2 milliseconds after the time when the first significant pressure rise is noted in the strut. This time corresponds approximately to only a 2-inch forward travel of the tire along the 10-inch bump length. This plunger response is shown subsequently to be adequate for reducing loads even during landings on step-shaped bumps or planks because of the gradual load buildup due to the kinematics of bump and wheel in spite of the hard tire. This rapid plunger action is believed to provide a satisfactory answer to one of the original questions regarding adequate response speed of the variable-orifice valve.

The band-pass action results in the reduction of lower cylinder pressure which to a large extent determines the hydraulic load developed by the strut. The reduction of boom acceleration for the low-pass strut is also evident in figure 7. Another interesting point is that the rates of increase of acceleration and of pressure (with oscillations faired out) are reduced by the band-pass strut action. It is believed that this automatic loading rate reduction should result in a reduction of spin-up drag loads during landings because of the longer spin-up time which results from the slower buildup of ground friction. Also, the higher harmonics of the landing-load pulse imparted to the airplane structure should be attenuated by this action and should result in beneficial alteration of the aircraft dynamic response during an abrupt landing impact.

From the velocity records it is evident that the band-pass strut telescopes considerably faster than the fixed-orifice strut which effectively allows the wheel for the band-pass device to follow the bump rather than to force the tire to deflect and load the gear. Although the design telescoping velocity of the fixed-orifice strut was of the order of 5 or 6 feet per second, the band-pass strut telescoped at the rate of over 25 feet per second during some tests with no evidence of the existence of foam in the lower cylinder and without exceeding the design load for the strut. From the displacement curve, it can be seen that the increased telescoping velocity results in an increased telescoping displacement for the low-pass strut as expected. It may be of interest to note that the telescoping limit before bottoming the strut was $7\frac{1}{2}$ inches and that the maximum displacement observed in the tests was $6\frac{1}{2}$ inches.

The 300-cycle-per-second oscillations apparent in the plunger displacement and the lower cylinder pressure curves for the low-pass shock strut are believed to result from excitation of the undamped plunger-spring combination in resonance with the strut mount which also had one natural frequency of approximately 300 cycles per second. Since every effort was made in the construction of this band-pass control unit to eliminate friction, it is possible that insufficient plunger damping was available and that, if the effort at friction reduction had been smaller or if a different strut mount frequency had been used, the vibration may not have appeared.

Another interesting observation concerns the behavior of the control piston and plunger combination. The control piston is seen to act in a delayed manner to compress the stiff spring between itself and the plunger. The sum of the instantaneous piston and plunger displacements gives an idea of the spring compression and shows how the piston forces the plunger closed by means of the spring. Therefore, the delay between the rapid plunger opening and its slow closing through

the downward motion of the piston determines more or less the time constant of the plunger opening for this control unit. This time constant can be adjusted by varying the relation of orifice sizes in the control piston and cylinder as well as the relation of effective areas of the piston and plunger and the main spring rate. Of course, other band-pass control units can be built in which the time-constant adjustment is made simpler by using an upper cylinder dump system as an auxiliary valve instead of combining it with the plunger variable-orifice valve as was done for this control unit.

The effect of the open-plunger time constant on strut behavior for rapidly applied bumps of different duration may be described somewhat as follows. For pulses having durations shorter than this time constant, the plunger opened up, allowed the strut to telescope rapidly to reduce the load, and then closed after the pulse was over. For pulses having durations longer than this time constant, the plunger opened up and allowed the strut to telescope rapidly so that the wheel mass achieved a high vertical velocity. Then when the plunger closed before the pulse was over, the load built up again to roughly the fixed-orifice-strut value because of the arrest of the extra high wheel mass velocity by the small orifice strut pushing against the large upper mass inertia.

Recycling Characteristics

In order to investigate the degree to which the band-pass shock strut was able to recover from a fast load pulse preparatory to experiencing a subsequent load pulse, some recycling tests comparing the fixed-orifice and low-band-pass shock struts were conducted. In these tests the landing gear was taxied over a series of three bumps in which the second and third bumps were spaced at equal intervals beyond the first bump. During these tests the bump spacing and the taxi speed were varied so as to produce different frequencies. Three ramp-shaped bumps were used each of which was 3 inches high by 12 inches long. For the band-pass strut, tire pressures of 75 and 225 pounds per square inch were used whereas, with the fixed-orifice strut, only the low tire pressure of 75 pounds per square inch was used in order to keep from failing the strut. The bump spacings were varied from 10 feet between bump center lines to 1.5 feet.

From the results of these tests, it was found that the band-pass shock strut was able to recycle for all bump spacings in this interval without exceeding the design loads. With the fixed-orifice strut, however, the design loads were exceeded, about the worst case occurring when the bumps were spaced about 5 feet apart. Time-history records comparing the band-pass and fixed-orifice shock strut for such a run are presented in figure 8 to illustrate recycling behaviors for both struts.

From the acceleration and pressure traces it can be seen that the load pulse for the first bump was slowly applied as evidenced by little alleviation from band-pass action. It is also evident from this first pulse that the pressure and acceleration are well within the design values. After this impact the action of the fixed-orifice strut is traced. As the strut reextended between bumps, foam was believed to have been introduced into the lower cylinder, both by passage downward through the orifice and by air coming out of solution in the lower cylinder when the pressure in it dropped. Contact with the second bump caused the strut to recompress very rapidly as seen from the telescoping velocity trace resulting in the transfer of some foam through the orifice and in forcing the air back into solution until liquid-to-metal contact occurred. At this point an abrupt reduction in telescoping velocity occurred with the resulting hammer blow shown on the pressure and acceleration plots. These hammer blows almost tripled the load by the time the third bump was contacted and exceeded the strut design acceleration by some 10 percent and the hydraulic pressure by some 40 percent. It may be noted that, although the strut was rapidly collapsing during the taxiing over the second and third bumps, the telescoping velocity was several times the design value while the bubble existed in the lower cylinder. The resulting arrest of the lower strut and wheel masses when the bubble disappeared produced the hammer blows measured. If a high tire pressure had been used in this experiment, the strut load probably would have been considerably larger and the strut would have failed.

Next, the band-pass strut is considered. Since the first impact was slowly applied, the load was not eased much as was observed previously. For the succeeding load pulses, however, the action of the control-unit components prevented the oil-hammer action mentioned before, and thus the maximum pressures and acceleration were reduced to less than one-fourth of the values for the fixed-orifice strut. The two control-unit actions which prevented the hammer blows may be described as follows. First, it can be seen that between bumps the piston and plunger were retracted upward and the main orifice was opened wide to allow the fluid to return rapidly to the lower cylinder; thus a moderately high pressure in this cylinder was maintained to prevent dissolved air from coming out of solution to form a bubble in the lower cylinder. Second, when the strut contacted the following bump, the plunger was forced to remain open by the rapid increase of pressure in the lower cylinder. Thus after the foam was driven out of the lower cylinder, the wide open orifice allowed rapid transfer of the oil into the upper cylinder so that the strut could continue to telescope at a rapid rate without the abrupt buildup of pressure which caused the hammer action in the conventional strut.

These recycling tests were made with about $2/3g$ effective wing lift to simulate an aircraft taxiing at a relatively high speed. Since this control action was effective for all the bump spacings used, it appeared that the band-pass strut could recycle itself for any bump spacing which the wheel and tire could be forced to follow. These recycling tests might also be interpreted to give an indication of the maximum load alleviation possible for this low-pass strut under the application of a step forcing function (the oil hammer action) for the case where a soft fork and tire cannot delay the application of load.

In order to determine whether the bump spacings used in these tests might be encountered in practical situations, one might envision a situation of a water-based aircraft with a shock-strut mounted ski, or foil taxiing over small closely spaced waves or a situation of a dispersed manned air force operating from rough unprepared fields with suitable landing gears. In such situations, small relatively closely spaced bumps might fail the landing gear if conventional struts were used, unless these struts are equipped with a means for rapid recycling which does not introduce foam into the lower cylinder.

Landings on Single Bumps

The case of a combined loading pulse made up of a rapidly applied, short-duration pulse superposed on a slowly applied, long duration pulse was discussed theoretically in reference 1. This condition is realized in practice during a landing, which constitutes the slowly applied load application, on a steep bump, which constitutes the rapidly applied pulse. Such a situation may occur, for example, during a design landing on an aircraft carrier in which the tire is bottomed and in this condition runs over an arresting gear cable. In the present investigation, landings were made with a hard tire on step-shaped bumps to compare the fixed-orifice and low-pass strut behaviors. A time-history record of such an impact on a rectangular bump 2 inches high by 10 inches long is presented in figure 9. A wing lift of 1 g was required to achieve a constant flight-path angle prior to ground contact.

From the acceleration and pressure curves it is evident that there is some reduction of load for the band-pass strut during the early stages of the impact. For impacts at lower forward speeds but with the same sinking speed, these reductions were not noted for the low-pass strut and both struts behaved similarly for such slowly applied impulses. These reductions of acceleration and pressure during the early stages of the impact in figure 9 may be explained as follows. On contact of the landing gear with the runway the plunger opens for a short time because of the discontinuous change from no load to finite load, after which time the plunger is closed by the control piston. These plunger and piston actions may be observed in figure 9. The fact that the

plunger is open results in a reduction in normal load, which, in turn, probably causes a reduction in the spin-up drag load. This lower drag, in turn, probably results in lower strut binding friction which maintains a lower normal load. This cycle of interdependency could not be definitely established by these tests since no measurements of spin-up drag were available. It is believed to have existed, however, for this case because of the fact that in impacts at lower forward speeds for which lower spin-up drags probably existed, load reductions were not encountered as was stated previously.

When the hard tire encountered the step-shaped bump during the landing impact, it is seen that the loads and pressures were boosted for both struts, although the values for the low-pass strut were only about 80 percent of the fixed-orifice strut values. Actually these reductions for this rapid pulse application during the impact could probably have been considerably greater if the control piston acted slower in forcing the plunger to close the main orifice. However, since the object of this investigation was only to find out whether the band-pass principle was practicable, no development work was carried out to adjust the orifice sizes in the control piston and cylinder to optimum values. Such adjustments would have modified the time constants of the plunger and piston motions which might have been balanced against each other to produce a more desirable action under combined pulse situations. From the results of figure 9, it is, however, evident that the control unit behaved in the proper manner from a qualitative point of view in reducing the high-frequency pulse loads. The displacement curves on this plot behave about as would be expected, the band-pass strut exhibiting greater telescoping deflections which resulted in greater vertical displacements of the upper mass representing the aircraft.

Possible Applications of Design

Some possible applications of the band-pass principle to practical designs are presented in this section.

For the single-acting low-pass shock absorber, several landing-gear applications have been suggested in reference 1 such as snow skis, hydro-skis, or high-pressure tires operating from rough surfaces. In connection with military aircraft dispersal it might also be considered that operation from rough unprepared fields might be possible. Reference 3 considers a band-pass strut in conjunction with a soft tire, probably on a lever suspension, for such operations. Another possible application might be for hypersonic aircraft in which aerodynamic heating of the enclosing structure converts normally used high-pressure pneumatic tires to potential bombs capable of wrecking an airplane. Because of this possibility, one manufacturer recently tested an aircraft equipped

with solid tires and conventional shock struts. These tests are believed to have been unsuccessful because of excessive ground loads. The use of a low-band-pass shock strut, which in some measure replaces tire resiliency, might however reduce the solid-tire loads to acceptable values.

Extension of the frequency range by use of a floating-piston design of low-pass shock strut has been considered. A double-acting vibration absorber of this type has also been designed and is described in the appendix. The areas of application of floating-piston band-pass absorbers might include vibration elimination (such as for spring-suspended helicopter rotors or blades) or vibration elimination (such as for flutter dampers).

CONCLUDING REMARKS

As a consequence of the experiments reported herein, it was demonstrated that the low-band-pass shock strut behaves as a fixed-orifice strut for slowly applied loads. For rapidly applied pulses of short duration achieved with hard tires on single steep bumps, loads for the low-pass strut were about 50 percent of the loads of the fixed-orifice strut, whereas for cyclic loading, the low-pass loads were less than 25 percent of the fixed-orifice strut loads. These load reductions were accomplished by the addition of a control unit weighing only $1\frac{1}{2}$ pounds to a shock strut which was designed for a 5,000-pound airplane. It is thus concluded that the band-pass principle offers a practical means for reducing rapidly applied, high intensity loadings. For rapidly applied pulses superposed on slowly applied pulses, load reductions of only 20 percent were achieved but these can probably be increased through further development of control elements. It was not attempted to improve this performance since optimumization of a shock absorber for any particular application was not an object of this investigation, the main object being instead to demonstrate that the band-pass principle could be applied to a practical problem. For the single-acting low-pass shock absorber, several landing-gear applications might be suggested such as with snow skis or hydro-skis, with high-pressure tires on rough runways, operation from rough unprepared fields, and for hypersonic aircraft using solid tires. The proposed design (discussed in the appendix) of a floating-piston vibration absorber may be used for spring-suspended helicopter rotors or for flutter dampers.

Langley Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., June 24, 1958.

APPENDIX

FLOATING-PISTON LOW-PASS VIBRATION ABSORBERS

Since completion of the design of the single-acting low-band-pass shock strut shown in figures 2, 3, and 4, consideration has been given to improving this design. The shock absorber dealt with in the main text of this paper utilized the principle of varying the orifice size as a function of the applied rate of loading to control the load developed and transmitted by the strut. For such an absorber, in order to reduce the damping force at, for example, high rates of loading, the orifice is opened wide. There is, however, a maximum flow rate at which this wide open orifice can be considered not to offer a flow restriction. Above this flow rate the shock-strut damping force increases to large values in spite of the large orifice opening. In order to extend the range of response of the strut to include steeper pulse rates, the "floating-piston" class of band-pass shock and vibration absorbers were designed and are discussed herein.

Designs for the floating-piston low-pass vibration absorber have been conceived for both the single-acting, landing-gear type of shock strut and the double-acting type of vibration absorber (an example of which is shown in fig. 10). The double-acting absorber was selected for illustration since single-acting low-pass shock struts have been discussed in reference 1 and in the present paper along with some of their practical applications whereas a design for a double-acting band-pass vibration absorber has not been presented previously. One of the essential features of the floating-piston absorber of figure 10 is that the main piston is not connected either to the main piston rod or to the main outer strut cylinder. This piston merely floats in the fluid and allows the main piston rod to slide through it within certain prescribed limits. For low frequencies or rates of load application, this piston is automatically coupled to the main piston rod by a type of fluid coupling and thus simulates a conventional vibration absorber. For high frequencies, on the other hand, this piston is merely left uncoupled from the main piston rod which slides freely up and down through the piston. The automatic fluid coupling consists essentially of a cylinder of fluid which becomes rigid (transmits load) when contained because of the inherent incompressibility of the fluid and becomes soft when the fluid is allowed to seep away. The method of operation of this vibration absorber is as follows:

The main shock-strut body is assumed to be attached to one element of a mechanism and the main piston rod to be attached to the other element in a system in which it is desired to absorb vibrations existing

between elements on a frequency-selective basis. Since the device of figure 10 is symmetrical about a horizontal plane through the main piston, alternating pulses are treated in exactly the same fashion by the appropriate half of the control unit involved. The parts in this control unit are mostly annular in shape so that the enlarged section in the circle gives an idea of the operation of the entire unit. Let us consider first the case of a rapidly applied force pulse exerted upward on the main piston rod with the main shock-strut cylinder assumed to be stationary. As the main piston rod rises, fluid is pressurized between the drive piston (shown in the enlarged view at the right) and the upper pump piston in the pump cylinder. This fluid is forced to flow through the upper supply port which increases the pressure in the upper half of the control unit, forces down the upper plunger, and thus allows the fluid to escape through the high-frequency inlet and escape ports into the main shock-strut cylinder in the region above the main piston. This pressure pulse also tends to close the upper check valve. Since the pulse is fast and of short duration, very little fluid is forced through the upper bleed orifice into the control cylinder to compress the upper spring by forcing the control piston upward toward the plunger. Thus, very little fluid pressure is required to operate the plunger against this uncompressed spring. It then follows that the pressures in the upper pump cylinder and control unit are relatively low; thus, the control unit and main piston are not forced away from their original position in the strut.

By the same token, since the lower section of the pump cylinder below the drive piston is expanded by the upward motion of the main piston rod, fluid is sucked into this cylinder through the fixed main orifice, into the overflow passage which is not closed by the drive piston that was displaced upward by the piston rod, through the lower half of the control unit, and finally through the lower supply port. If the lower plunger is slightly opened, fluid may also enter the lower half of the control unit through the lower high-frequency escape and inlet ports. Thus, for rapidly applied loads or motions, the main piston rod is effectively disconnected from the main piston so that only a small force is developed and transmitted by the absorber. After the upward pulse on the main piston rod, if no other pulse is applied, the centering spring will gradually raise the control unit relative to the main piston rod, until the main piston is again centered on the drive piston. If a reverse or downward pulse is next applied to the main piston rod, it is evident that an action, the reverse of that which has just been described, would occur.

For a low-frequency pulse having a low rate of application, a different action takes place. In this case the fluid in the upper pump cylinder is again pressurized as the main piston rod moves upward but at a slower rate; thus, an appreciable quantity of fluid is forced through the upper bleed orifice. The pressurized fluid also closes the upper check valve and compresses the plunger return spring by forcing the control

piston up toward the plunger. It is assumed that in this case, since the pressure rise is gradual, enough fluid can flow through the bleed orifice to balance exactly the downward pressure on the plunger in the closed position so that no fluid can leave the upper control cylinder through the high-frequency inlet and escape ports. Therefore, the pressure gradually increases in the upper pump cylinder and control unit with the result that the control unit and main piston are forced upward, at a slightly greater rate than the main piston rod moves upward, until the main piston is again centered on the drive piston. When the main piston just passes this point on the main piston rod, fluid can leave the upper control cylinder through the overflow passage and main orifice and can be dumped into the region below the main piston. Subsequent to this time, as long as the slowly applied pulse continues, the main piston is centered on the drive piston and the fluid from the pump cylinder is continuously exhausting through the overflow passage. Fluid from above the main piston, therefore, must flow through the main orifice into the region below the main piston; therefore, the shock strut will behave similar to a conventional fixed-orifice shock absorber for these low frequencies. The main piston is kept centered on the drive piston during slowly applied pulses in order to prevent the drive piston from bottoming on either end of the control cylinder either during large amplitude pulses or following a series of unsymmetrical pulses or motions.

For the case of a high-frequency pulse superposed on this low-frequency pulse, a rapid additional pressure rise occurs in the pump and control cylinders. Since the fluid cannot flow through the upper bleed orifice fast enough to increase the force on the main spring to the new level required by the high-frequency pulse, this instantaneous pressure rise will open the plunger momentarily and allow a burst of fluid to leave the control cylinder through the high-frequency inlet and escape ports. Thus, rapid pulses even though superposed on low-frequency pulses should be handled exactly as if they existed alone. In the case of the low-frequency pulse, as the load drops off, the pressure in the pump cylinder decreases and the plunger return spring forces the fluid out of the control cylinder through the upper check valve.

From this discussion, one can deduce that this floating-piston shock absorber is made up of a small pump which is driven by the main piston rod and which drives the main piston through the control unit (in this case, a frequency-controlled hydraulic motor). When the main piston is being driven, as for low-frequency pulses, the reaction on the main piston rod is the force resulting from the pressure differential above and below the main piston. This pressure differential arises from the fixed-main-orifice restriction as in a conventional fixed-orifice absorber.

The action of the main piston in recentering itself on the piston rod during a low-frequency pulse allows this shock absorber to attenuate many high-frequency load oscillations during the course of a single low-frequency load pulse. Thus the low-frequency load can be transmitted and damped, whereas the many high-frequency load oscillations are not transmitted. It can be seen that the recycling of the many parts of this shock absorber for both high- and low-frequency vibrations is automatically taken care of. It is believed that this shock absorber is applicable to much higher frequencies than the variable-orifice type of band-pass absorber. There is also analytical reasoning (not presented in this paper) which indicates that the high-frequency load-reducing characteristics of the floating-piston absorber are superior to those for the variable-orifice absorber.

The areas of application of either single- or double-acting floating-piston band-pass absorbers might include vibration isolation (such as for spring-suspended helicopter rotors or blades as mentioned in ref. 1) or in vibration elimination (such as for flutter dampers). In both cases, it is intended to remove the undesirable vibration occurring in one frequency range while preserving, without resonance, pilot-induced motions and airplane reactions occurring in another frequency range.

REFERENCES

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TABLE I.- TEST CONDITIONS AND RESULTING DATA MAXIMUMS - BAND-PASS SHOCK STRUT

[Dropping weight, 2,360 lb; initial strut air pressure, 40 psi]

Run	Tire pressure, psi	Forward speed, fps	Remarks code (**)	Strut inclination, deg	Axial acceleration, g, for -				Lower cylinder pressure, psi	Plunger displacement, in.	Orifice area, sq in.	Strut telescoping		Strut force, lb, determined from -	
					Boom	Attach fitting	Strut	Axle				Displacement, in.	Velocity, fps	Accelerometer	Pressure
Taxiing over bump 12 inches high by 120 inches long; wing lift of 0.88g															
74	75	19.8	c	6	0.44	0.46	5.2	3.3	150	0.018	0.114	3.20	1.89	1,094	1,093
*10	75	43.0	c	6	1.58	1.57	15.5	13.7	479	.040	.121	4.49	4.37	3,587	3,332
*11	75	49.8	c	6	1.51	1.67	9.1	13.7	426	.037	.124	4.58	4.63	3,466	2,973
12	75	55.3	c	6	2.22	2.39	13.1	15.7	581	.036	.127	4.32	5.16	5,081	4,029
13	75	67.1	c	6	2.62	2.66	15.4	16.0	738	.045	.133	4.87	5.76	5,957	5,100
14	75	76.9	c	6	2.71	2.81	17.2	19.8	758	.051	.147	5.18	6.49	6,169	5,234
15	150	49.0	c	6	1.97	2.13	14.3	13.8	484	.042	.144	3.83	5.49	4,509	3,368
68	150	55.0	c	6	1.70	1.73	30.3	26.7	704	.082	.172	5.82	7.45	3,867	4,867
73	225	22.6	c	6	.62	.54	3.9	3.9	209	.022	.098	3.28	2.28	1,391	1,494
71	225	37.6	c	6	1.07	1.09	17.2	12.3	434	.057	.132	4.38	4.67	2,434	3,179
72	225	54.6	c	6	1.48	1.43	27.0	14.6	493	.051	.153	5.07	5.46	3,350	3,431
69	225	85.5	c	6	2.73	3.06	87.2	29.6	920	.078	.173	6.36	8.44	6,270	6,334
48	300	22.8	c	6	.74	.69	8.9	7.2	242	.041	.109	3.05	3.05	1,670	1,646
49	300	41.7	c	6	1.16	1.23	22.7	11.1	386	.050	.130	3.80	4.30	2,649	2,735
50	300	55.0	c	6	1.69	1.66	30.3	18.4	509	.056	.159	4.41	6.09	3,832	3,573
51	300	66.2	c	6	2.15	2.22	49.3	22.3	585	.038	.153	4.84	6.49	4,897	3,978
52	300	80.7	c	6	3.08	3.38	72.4	29.8	763	.061	.131	5.50	6.29	7,058	5,357
Taxiing over bump 1 inch high by 10 inches long; wing lift of 0.88g															
84	300	44.4	c	6	0.89	0.97	12.4	9.4	130	0.022	0.154	0.81	3.48	2,217	955
Taxiing over bump 2 inches high by 10 inches long; wing lift of 0.88g															
80	225	36.2	c	11	1.81	1.66	23.8	15.9	484	0.052	0.156	1.44	5.76	4,228	3,980
81	225	54.6	c	11	2.07	2.27	34.9	23.8	510	.075	.194	1.44	7.75	4,799	4,231
82	225	70.9	c	11	2.29	2.27	38.9	31.0	510	.095	.220	1.38	9.04	5,259	3,774
79	225	85.5	c	11	1.84	2.20	64.5	43.4	623	.109	.227	1.64	10.43	4,304	5,097
31	300	71.9	c	11	2.49	2.44	53.6	40.4	792	.110	.216	1.84	10.79	5,712	5,466
32	300	80.7	c	11	2.57	2.45	63.2	44.8	724	.118	.225	1.76	11.12	5,870	5,003
Taxiing over bump 4 inches high by 20 inches long; wing lift of 0.88g															
20	75	23.0	c	11	1.42	1.48	10.7	6.8	386	0.031	0.098	2.45	3.11	3,284	2,698
21	75	49.0	c	11	2.48	2.35	62.1	23.1	1,013	.091	.150	2.62	8.01	5,665	6,968
22	75	61.4	c	11	2.82	3.07	59.1	29.8	1,018	.084	.152	2.45	8.41	6,522	7,002
23	75	71.4	c	11	2.94	3.24	59.9	31.9	1,114	.091	.173	2.48	10.06	6,809	7,660
75	75	85.5	c	11	3.04	3.28	56.9	37.6	1,018	.093	.209	2.76	12.11	7,037	7,010
76	225	36.9	c	11	2.08	2.04	43.2	25.0	695	.049	.205	3.14	8.84	4,771	4,806
77	225	54.4	c	11	3.14	3.22	81.0	31.3	976	.069	.203	3.23	10.82	7,232	6,720
78	225	70.2	c	11	3.95	4.43	76.6	41.8	1,409	.107	.203	3.23	12.61	9,178	9,666

*Initial strut air pressure of 60 pounds per square inch.
 **Code: c, chevron seal; d, bump separation; O, "O" ring seal; r, plunger retractor operative.

TABLE I.- TEST CONDITIONS AND RESULTING DATA MAXIMUMS - BAND-PASS SHOCK STRUT - Continued

[Dropping weight, 2,360 lb; initial strut air pressure, 40 psi]

Run	Tire pressure, psi	Forward speed, fps	Remarks code (*)	Strut inclination, deg	Axial acceleration, g, for -				Lower cylinder pressure, psi	Plunger displacement, in.	Orifice area, sq in.	Strut telescoping		Strut force, lb, determined from -	
					Boom	Attach fitting	Strut	Axle				Displacement, in.	Velocity, fps	Accelerometer	Pressure
Taxiing over bump 3 inches high by 12 inches long; wing lift of 0.88g															
85	300	54.6	c	13	3.56	3.76	62.2	44.9	1,157	0.118	0.250	2.56	14.30	9,010	7,949
33	300	71.4	c	13	4.13	3.94	74.5	67.1	1,304	.144	.233	2.74	15.09	9,518	8,954
34	300	75.8	c	13	4.37	4.83	85.7	75.1	1,373	.163	.249	2.85	16.29	10,202	9,423
Taxiing over bump 4 inches high by 10 inches long; wing lift of 0.88g															
41	75	29.2	c	22	2.20	2.34	30.0	25.3	908	0.073	0.184	2.74	8.74	5,376	6,256
42	75	44.2	c	22	2.52	2.63	50.3	33.2	1,066	.098	.194	2.48	10.92	6,124	7,332
53	75	54.4	c	22	3.19	3.51	76.5	67.5	937	.132	.369	2.59	21.75	7,790	6,492
43	75	55.0	c	22	2.99	3.38	59.1	42.6	1,142	.108	.222	2.48	13.17	7,321	7,849
45	75	61.0	c	22	3.38	3.81	69.8	54.2	1,044	.110	.243	2.65	15.16	8,273	7,183
46	75	65.4	c	22	3.67	3.84	85.3	68.7	1,123	.128	.270	2.71	16.81	8,920	7,723
47	75	70.9	c	22	3.60	3.86	87.9	68.8	1,165	.128	.269	3.11	17.01	8,771	8,012
54	75	70.9	c	22	3.79	3.94	116.0	94.4	1,190	.193	.430	2.82	24.63	9,207	8,179
55	75	75.2	c	22	3.98	4.03	116.0	101.7	1,193	.215	.375	2.82	25.32	9,646	8,279
56	150	48.1	c	22	3.92	4.26	66.7	48.5	1,264	.094	.347	2.79	20.65	9,562	8,735
57	150	60.6	c	22	4.30	4.62	73.8	55.8	1,224	.098	.349	2.88	23.34	10,477	8,502
58	150	65.8	c	22	3.93	4.40	84.7	59.7	1,235	.124	.376	2.71	23.83	9,613	8,585
59	150	70.9	c	22	4.21	4.94	84.9	63.5	1,274	.151	.471	2.68	25.62	10,345	8,752
60	150	76.3	c	22	4.05	4.39	83.7	64.9	1,357	.147	.463	2.62	25.82	9,877	9,309
61	225	48.1	c	22	4.15	4.73	71.0	59.3	1,302	.133	.265	3.05	16.09	10,169	8,976
62	225	54.6	c	22	4.56	5.08	81.0	60.0	1,413	.150	.270	2.88	16.98	11,149	9,699
63	225	60.4	c	22	4.64	5.05	80.4	64.6	1,466	.155	.276	2.76	17.68	11,320	10,057
35	300	28.9	c	22	3.38	3.44	60.0	38.5	1,180	.121	.211	3.02	11.52	8,185	8,104
37	300	38.4	c	22	4.68	5.16	79.4	56.4	1,535	.157	.225	3.02	14.43	11,427	10,521
Landings on flat runway; wing lift of 1 g															
171	225	44.3	c	15	2.12	2.47	54.4	18.6	509	0.055	0.160	4.98	7.15	5,020	5,651
172	225	84.0	c	15	4.32	5.69	63.1	30.3	1,103	.088	.207	6.45	10.92	10,120	9,959
Landing across rectangular plank 1 inch high by 12 inches long; wing lift of 1 g; sinking speed of 5.51 fps															
190	225	70.2	c; r	15	3.10	4.17	52.2	35.0	1,114	0.038	0.162	6.42	10.03	7,454	7,742
Landing across rectangular plank 2 inches high by 12 inches long; wing lift of 1 g; sinking speed of 5.63 fps															
191	225	70.9	c; r	15	4.42	6.12	73.4	54.4	1,702	0.086	0.207	6.19	15.49	10,437	11,766

*Code: c, chevron seal; d, bump separation; 0, "0" ring seal; r, plunger retractor operative.

TABLE I.- TEST CONDITIONS AND RESULTING DATA MAXIMUMS - BAND-PASS SHOCK STRUT - Concluded

[Dropping weight, 2,360 lb; initial strut air pressure 60 psi]

Run	Tire pressure, psi	Forward speed, fps	Remarks code (*)	Strut inclination, deg	Axial acceleration, g, for -				Lower cylinder pressure, psi	Plunger displacement, in.	Orifice area, sq in.	Strut telescoping		Strut force, lb, determined from -	
					Boom	Attach fitting	Strut	Axle				Displacement, in.	Velocity, fps	Accelerometer	Pressure
Recycling taxi runs over three consecutive bumps; wing lift of 0.65g															
173	75	40.0	c; d, 10 feet	10	1.69	2.05	21.2	17.5	717	0.064	0.160	3.20	6.75	3,873	4,969
173	75	40.0	c; d, 10 feet	10	1.90	2.01	22.0	19.1	819	.078	.172	3.46	7.75	4,375	5,664
173	75	40.0	c; d, 10 feet	10	1.94	2.10	19.0	15.8	876	.066	.162	4.84	7.25	4,477	6,083
174	75	44.3	c; d, 5 feet	10	1.88	2.53	29.6	22.4	815	.077	.180	2.42	7.84	4,257	5,612
174	75	44.3	c; d, 5 feet	10	1.04	1.90	31.9	28.9	562	.096	.177	2.22	7.15	2,495	3,907
174	75	44.3	c; d, 5 feet	10	1.80	3.38	57.6	37.6	870	.135	.243	2.68	10.53	4,180	5,998
175	75	48.5	c; d, 3 feet	10	1.76	1.79	23.8	21.8	864	.070	.170	4.52	7.94	4,050	6,001
175	75	48.5	c; d, 3 feet	10	.92	1.41	31.5	28.4	445	.082	.157	4.00	5.06	2,220	3,165
175	75	48.5	c; d, 3 feet	10	.88	1.62	31.5	28.4	411	.094	.270	3.97	6.85	2,069	2,916
176	75	65.8	c; d, 1.5 feet	10	2.06	2.54	36.4	29.8	982	.088	.225	5.70	9.83	4,724	6,870
176	75	65.8	c; d, 1.5 feet	10	1.07	1.42	28.1	24.4	285	.016	.080	5.62	1.79	2,196	2,196
176	75	65.8	c; d, 1.5 feet	10	.70	.67	13.7	15.2	206	.032	.038	5.44	.60	1,599	1,642
177	75	41.4	0; r; d, 10 feet	10	1.76	1.88	21.7	21.4	854	.107	.179	3.28	7.94	4,011	5,902
177	75	41.4	0; r; d, 10 feet	10	1.18	1.14	22.5	21.4	565	.253	.242	2.94	9.04	2,642	3,950
177	75	41.4	0; r; d, 10 feet	10	1.57	1.77	21.7	20.3	758	.138	.211	2.85	8.54	3,562	5,241
178	75	44.8	0; r; d, 5 feet	10	1.82	2.32	20.7	24.4	860	.112	.190	3.28	8.54	4,210	5,943
178	75	44.8	0; r; d, 5 feet	10	.71	2.98	15.5	18.1	307	.440	.300	3.40	8.84	2,109	2,179
178	75	44.8	0; r; d, 5 feet	10	1.11	1.45	41.4	44.6	436	.618	.342	2.74	13.01	2,612	3,038
179	75	48.3	0; r; d, 3 feet	10	1.88	2.21	24.7	23.9	877	.112	.192	3.48	9.04	4,306	6,062
179	75	48.3	0; r; d, 3 feet	10	.94	.75	53.2	45.2	375	.660	.309	3.66	10.72	2,113	2,647
179	75	48.3	0; r; d, 3 feet	10	.16	0	21.7	19.1	95	.640	.086	1.47	.79	302	731
180	75	65.8	0; r; d, 1.5 feet	10	1.66	2.25	35.2	34.1	801	.150	.222	3.43	10.53	3,799	5,545
180	75	65.8	0; r; d, 1.5 feet	10	.67	.95	24.0	20.1	199	.227	.015	3.25	.30	1,458	1,474
180	75	65.8	0; r; d, 1.5 feet	10	.55	.55	21.0	16.8	-21	.642	.020	2.59	.30	1,202	-39
181	225	41.4	0; r; d, 10 feet	10	2.14	2.67	35.2	34.7	916	.145	.242	3.72	11.52	4,915	6,327
181	225	41.4	0; r; d, 10 feet	10	1.83	1.57	42.7	42.8	608	.030	.359	2.65	13.21	4,136	4,213
181	225	41.4	0; r; d, 10 feet	10	1.67	1.96	34.4	36.9	665	.259	.318	2.74	12.41	3,828	4,604
183	225	44.9	0; r; d, 5 feet	10	2.75	2.83	46.0	39.6	969	.171	.233	3.54	11.42	6,146	6,686
183	225	44.9	0; r; d, 5 feet	10	1.19	1.17	40.5	37.4	434	.339	.324	3.00	11.02	2,701	3,033
183	225	44.9	0; r; d, 5 feet	10	1.83	2.32	68.5	64.5	597	.665	.514	1.99	16.09	4,297	4,140
184	225	48.5	0; r; d, 3 feet	10	2.71	3.31	45.5	40.6	1,070	.185	.231	3.69	12.41	6,238	7,375
184	225	48.5	0; r; d, 3 feet	10	1.59	1.60	110.7	82.3	589	.639	.338	3.89	15.29	3,604	4,090
184	225	48.5	0; r; d, 3 feet	10	.24	.28	42.4	39.0	108	.649	.248	3.89	1.99	508	808
182	225	66.2	0; r; d, 1.5 feet	10	2.82	3.57	59.1	58.8	1,221	.227	.275	3.80	16.38	6,514	8,403
182	225	66.2	0; r; d, 1.5 feet	10	.35	.08	25.5	13.7	186	.450	.148	2.94	-2.68	649	1,377
182	225	66.2	0; r; d, 1.5 feet	10	.35	.48	48.7	43.9	175	.660	.269	2.62	4.77	832	1,287

*Code: c, chevron seal; d, bump separation; 0, "0" ring seal; r, plunger retractor operative.

TABLE II.- TEST CONDITIONS AND RESULTING DATA MAXIMUMS - FIXED-ORIFICE SHOCK STRUT

[Dropping weight, 2,360 lb; initial strut air pressure, 40 psi]

Run	Tire pressure, psi	Forward speed, fps	Remarks code (*)	Strut inclination, deg	Axial acceleration, g, for -				Lower cylinder pressure, psi	Plunger displacement, in.	Orifice area, sq in.	Strut telescoping		Strut force, lb, determined from -	
					Boom	Attach fitting	Strut	Axle				Displacement, in.	Velocity, fps	Accelerometer	Pressure
Taxiing over bump 12 inches high by 120 inches long; wing lift of 0.88g															
87	225	23.2	c; p	6	0.74	0.82	3.0	6.8	245	----	0.075	3.34	2.18	1,846	1,737
88	225	37.8	c; p	6	1.31	1.26	10.0	6.7	428	----	.074	3.92	2.78	3,228	2,987
89	225	49.0	c; p	6	1.69	1.87	10.0	10.2	628	----	.075	4.18	3.48	4,216	4,349
90	225	60.6	c; p	6	2.26	2.35	8.8	11.2	872	----	.074	4.95	4.07	5,143	6,027
91	225	70.9	c; p	6	3.08	2.86	14.2	13.9	1,056	----	.077	5.27	4.77	7,568	7,276
92	225	85.5	c; p	6	3.67	3.81	12.9	16.0	1,281	----	.079	5.76	5.36	7,911	8,809
Taxiing over bump 1 inch high by 10 inches long; wing lift of 0.88g															
93	300	44.3	c; p	6	1.08	1.26	15.9	11.5	417	----	0.066	0.89	2.48	2,708	2,910
Taxiing over bump 2 inches high by 10 inches long; wing lift of 0.88g															
106	225	28.7	c; p	11	1.85	1.93	18.7	19.2	608	----	0.102	1.47	3.67	4,184	4,208
105	225	43.9	c; p	11	2.30	2.25	23.5	19.7	802	----	.080	1.32	4.27	5,218	5,527
104	225	59.9	c; p	11	2.55	2.95	32.9	27.5	1,011	----	.077	1.27	4.67	6,465	6,948
107	225	70.4	c; p	11	2.96	3.76	29.2	35.3	1,288	----	.070	2.36	4.77	6,650	8,849
102	225	75.8	c; p	11	2.85	2.85	34.1	34.6	1,235	----	.072	1.21	4.87	6,340	8,473
109	225	75.2	c; p	8	2.87	3.25	30.0	37.8	1,390	----	.070	2.71	4.97	5,565	9,549
141	225	74.1	c; p	8	3.55	3.55	46.7	44.4	1,421	----	.090	1.15	5.56	8,256	9,736
108	225	80.0	c; p	11	3.28	4.12	38.9	41.5	1,357	----	.073	2.56	5.16	7,387	9,321
103	225	84.8	c; p	11	3.58	4.59	51.5	42.8	1,373	----	.078	2.71	5.56	8,057	9,411
110	225	84.8	c; p	8	3.19	3.96	40.7	46.2	1,406	----	.075	2.79	5.36	8,080	9,659
94	300	54.2	c; p	11	2.32	2.22	20.5	20.9	909	----	.073	1.35	4.17	5,783	6,256
Taxiing over bump 4 inches high by 20 inches long; wing lift of 0.88g															
99	225	48.3	c; p	11	3.96	3.86	21.7	25.4	1,472	----	0.070	3.83	5.16	8,957	10,112
100	225	61.0	c; p	11	4.37	4.72	37.9	36.6	1,773	----	.074	2.36	5.96	9,776	12,138
101	225	70.9	c; p	11	5.06	5.52	38.5	36.2	1,921	----	.075	2.28	6.26	11,623	13,146
Taxiing over bump 3 inches high by 12 inches long; wing lift of 0.88g															
114	225	22.6	c; p	10	1.49	1.46	17.7	15.7	762	----	0.079	2.33	3.97	3,341	5,260
111	225	44.4	c; p	10	3.23	3.10	20.7	26.9	1,247	----	.073	2.16	4.87	8,032	8,561
112	225	60.2	c; p	10	3.62	3.61	33.0	40.9	1,669	----	.080	3.25	6.26	9,030	11,441
113	225	70.4	c; p	10	3.77	4.76	36.1	40.6	1,835	----	.077	3.00	6.36	9,614	12,570
134	225	70.4	c; p	10	3.87	3.88	45.5	43.1	1,822	----	.073	1.50	5.96	8,899	12,446
95	300	44.8	c; p	13	3.81	4.18	38.7	42.8	1,601	----	.074	1.70	5.66	9,675	10,965

*Code: c, chevron seal; d, bump separation; p, low-band-pass control unit with locked plunger; s, original trainer snubber and equivalent orifice area; 0, "0" ring seal.

TABLE II.- TEST CONDITIONS AND RESULTING DATA MAXIMUMS - FIXED-ORIFICE SHOCK STRUT - Concluded

[Dropping weight, 2,360 lb; initial strut air pressure, 40 psi]

Run	Tire pressure, psi	Forward speed, fps	Remarks code (*)	Strut inclination, deg	Axial acceleration, g, for -				Lower cylinder pressure, psi	Plunger displacement, in.	Orifice area, sq in.	Strut telescoping		Strut force, lb, determined from -	
					Boom	Attach fitting	Strut	Axle				Displacement, in.	Velocity, fps	Accelerometer	Pressure
Taxiing over bump 4 inches high by 10 inches long; wing lift of 0.88g															
121	75	28.4	c; p	22	2.12	2.13	22.0	19.5	937	----	0.068	2.16	3.97	5,592	6,450
124	75	36.8	c; p	22	2.67	2.91	25.0	24.1	1,165	----	.073	1.96	4.77	6,430	8,002
122	75	44.3	c; p	22	2.99	3.44	30.3	28.4	1,369	----	.073	1.70	5.16	7,979	9,387
126	75	48.8	c; p	22	3.73	4.33	41.2	30.3	1,755	----	.077	1.90	6.16	9,156	12,012
123	75	54.2	c; p	22	3.87	4.42	38.2	35.0	1,866	----	.077	1.67	6.36	9,334	12,769
144	75	59.5	c; p	22	3.79	4.41	33.8	34.2	1,907	----	.076	1.58	6.36	9,438	13,049
148	75	59.9	c; p	22	4.57	5.62	46.1	38.4	2,448	----	.073	1.70	6.95	11,284	16,728
96	300	32.9	c; p	22	4.60	4.55	38.2	42.5	1,565	----	.079	2.39	5.96	12,120	10,724
Landings on flat runway; wing lift of 1 g															
136	225	44.6	c; p	15	2.58	3.83	18.7	16.8	855	----	0.075	4.26	4.17	5,958	5,894
137	225	67.3	c; p	15	2.71	4.16	18.9	13.9	1,106	----	.081	4.38	4.67	6,425	7,611
138	225	85.5	c; p	15	4.82	6.95	37.4	29.8	1,756	----	.078	5.85	6.26	10,927	12,037
Landing across rectangular plank 1 inch high by 12 inches long; wing lift of 1 g; sinking speed of 5.63 fps															
196	225	70.9	0; p	15	3.73	5.98	23.4	28.4	1,420	----	0.075	3.95	5.36	8,629	9,778
Landing across rectangular plank 2 inches high by 12 inches long; wing lift of 1 g; sinking speed of 5.59 fps															
197	225	70.9	0; p	15	5.23	8.58	34.7	42.8	2,155	----	0.077	4.35	6.85	12,371	14,787
Recycling taxi runs over three consecutive bumps; wing lift of 0.63g															
**145	75	38.2	p; c; d, 10 feet	10	1.90	2.46	11.7	13.2	836	----	0.071	5.47	3.77	4,473	5,831
**145	75	38.2	p; c; d, 10 feet	10	1.82	2.21	15.6	17.7	766	----	.069	3.54	3.57	4,253	5,330
**145	75	38.2	p; c; d, 10 feet	10	2.11	2.88	19.4	18.9	848	----	.114	3.97	4.37	4,998	5,868
**146	75	43.5	p; c; d, 5 feet	10	1.77	2.26	10.4	12.1	848	----	.068	4.81	3.67	4,160	5,909
**146	75	43.5	p; c; d, 5 feet	10	5.57	7.92	38.4	38.2	2,082	----	.124	3.95	4.27	13,260	14,273
**146	75	43.5	p; c; d, 5 feet	10	6.40	8.68	39.2	38.2	2,400	----	.021	3.25	1.19	15,148	16,424
**147	75	47.5	p; c; d, 3 feet	10	1.91	2.26	12.3	14.1	887	----	.068	4.72	3.77	4,452	6,173
**147	75	47.5	p; c; d, 3 feet	10	2.36	2.52	33.0	30.6	853	----	.117	4.41	5.66	5,443	5,945
**147	75	47.5	p; c; d, 3 feet	10	2.57	3.85	33.0	28.9	1,002	----	.136	3.83	7.05	6,160	6,943
**198	75	39.7	0; s; d, 10 feet	10	1.80	1.87	14.2	13.6	842	----	.072	2.45	3.97	4,141	5,811
**198	75	39.7	0; s; d, 10 feet	10	2.01	2.32	26.0	22.6	675	----	.082	1.81	3.97	4,553	4,662
**198	75	39.7	0; s; d, 10 feet	10	2.01	2.03	11.0	8.5	875	----	.071	2.91	3.97	4,611	6,042
**199	75	44.3	0; s; d, 5 feet	10	1.85	2.11	13.1	12.4	872	----	.071	2.74	3.97	4,278	6,018
**199	75	44.3	0; s; d, 5 feet	10	6.72	7.09	86.1	33.3	2,036	----	.137	2.28	15.39	14,266	13,929
199	75	44.3	0; s; d, 5 feet	10	8.27	6.54	114.5	46.8	*	----	----	2.07	19.16	18,087	-----
200	75	46.5	0; s; d, 3 feet	10	1.99	2.15	10.6	13.1	*	----	----	3.31	4.57	4,595	-----
200	75	46.5	0; s; d, 3 feet	10	3.65	5.49	48.5	33.2	*	----	----	3.17	9.53	8,273	-----
200	75	46.5	0; s; d, 3 feet	10	3.36	5.77	37.9	31.6	*	----	----	2.88	10.13	7,628	-----

*Code: c, chevron seal; d, bump separation; p, low-band-pass control unit with locked plunger; s, original trainer snubber and equivalent orifice area; 0, "0" ring seal.

**Initial strut air pressure of 60 pounds per square inch.

***Failed pressure gage.

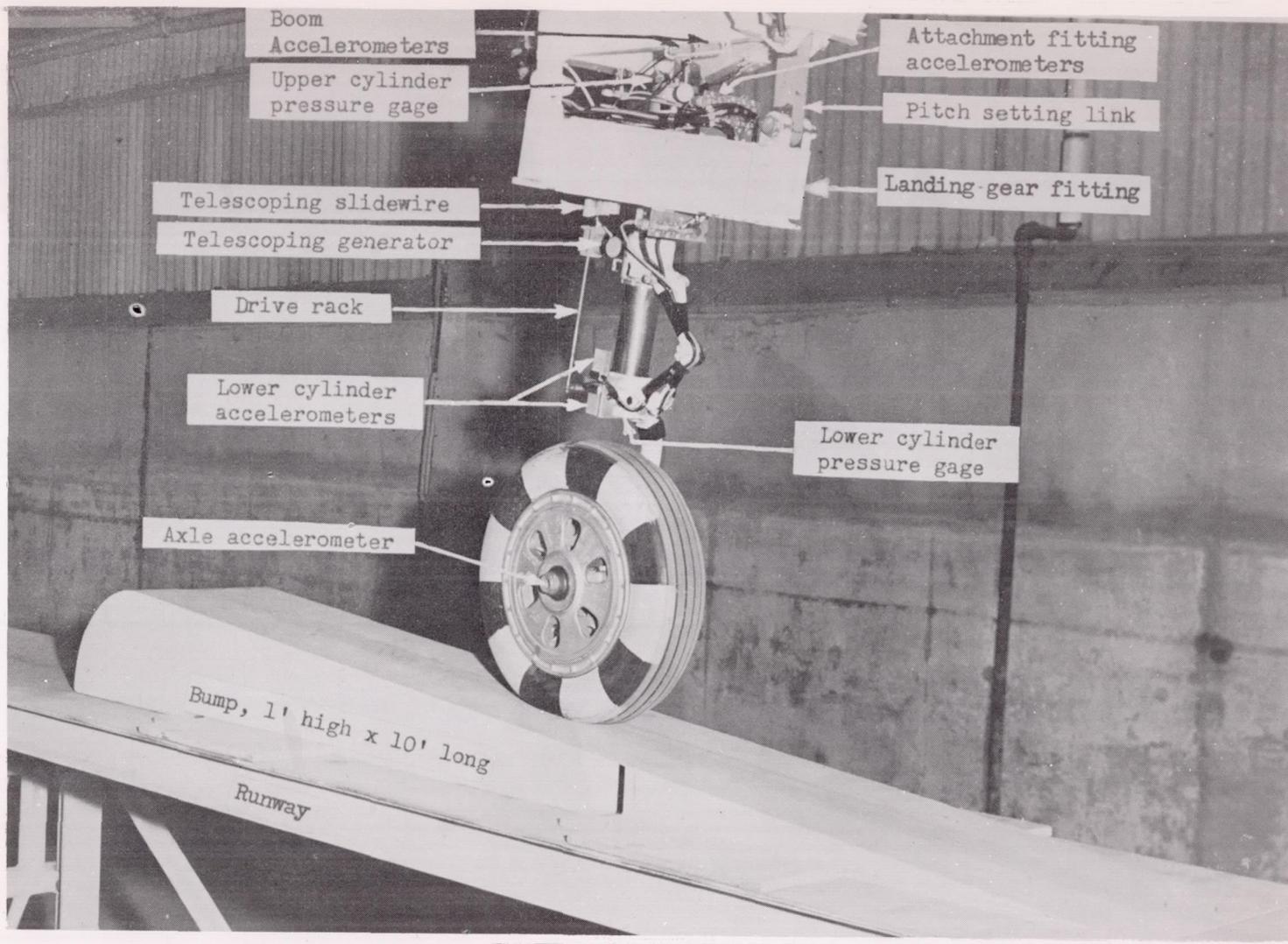
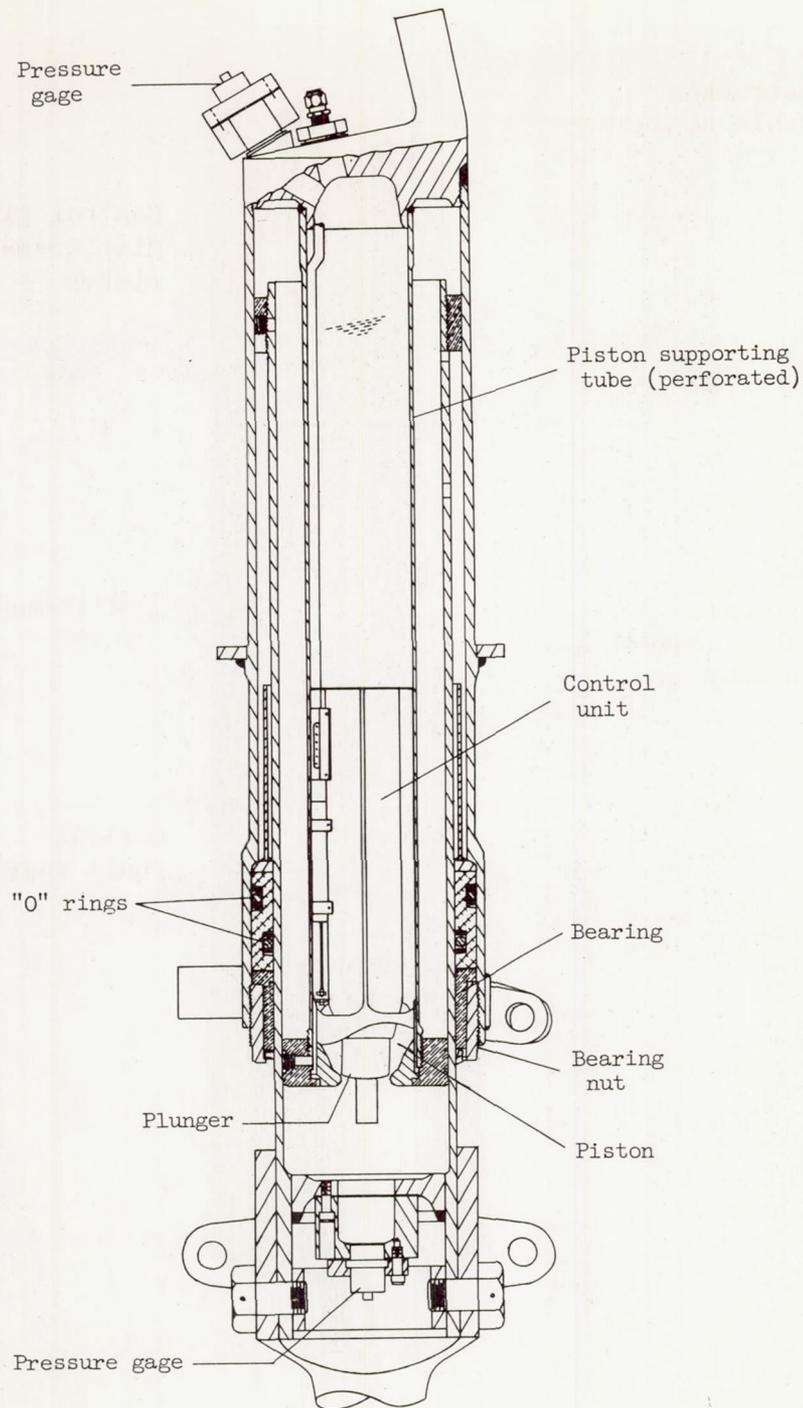


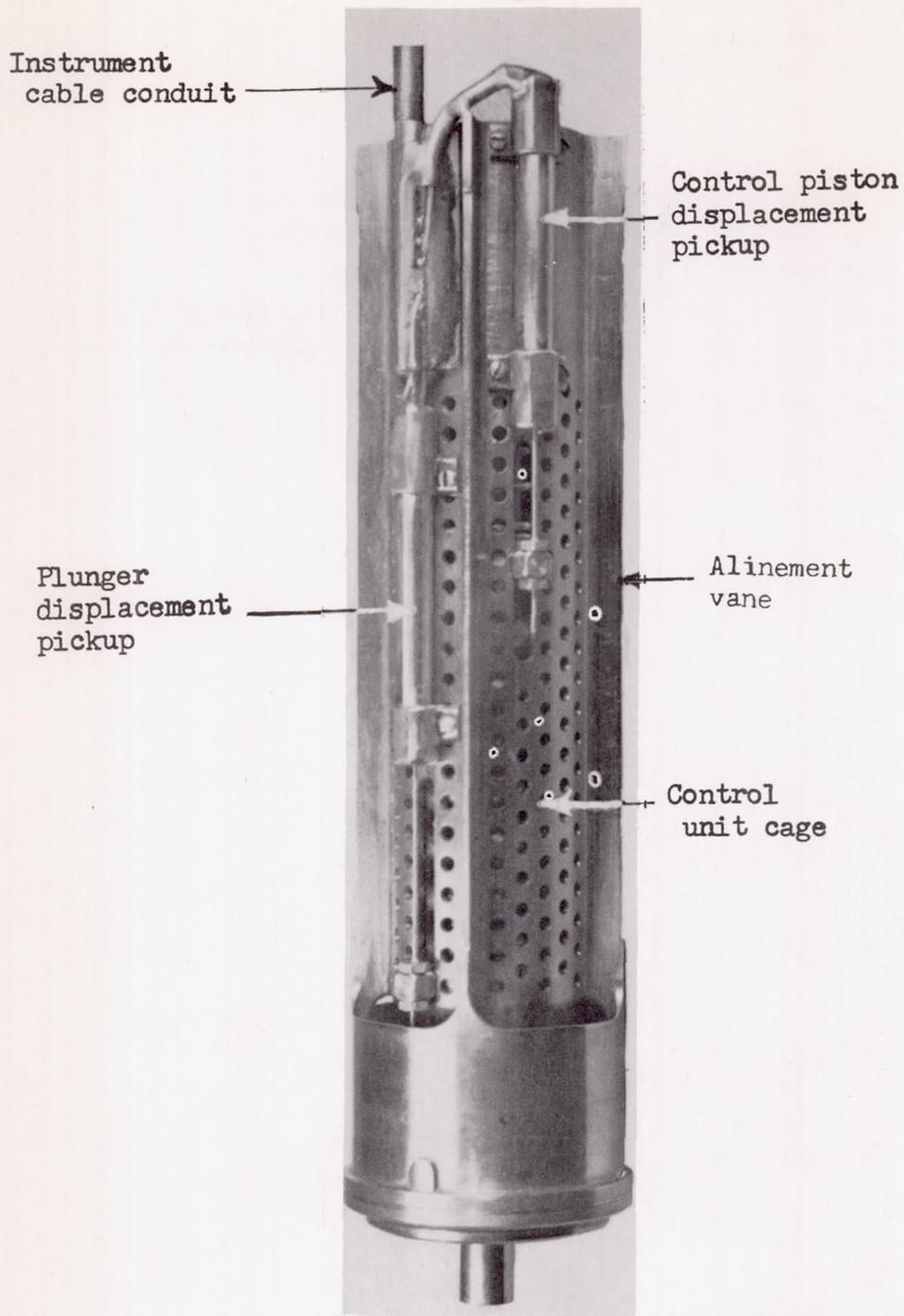
Figure 1.- View of landing gear taxiing over bump. Instrumentation is also shown.

L-57-5023.1



(a) Band-pass shock strut.

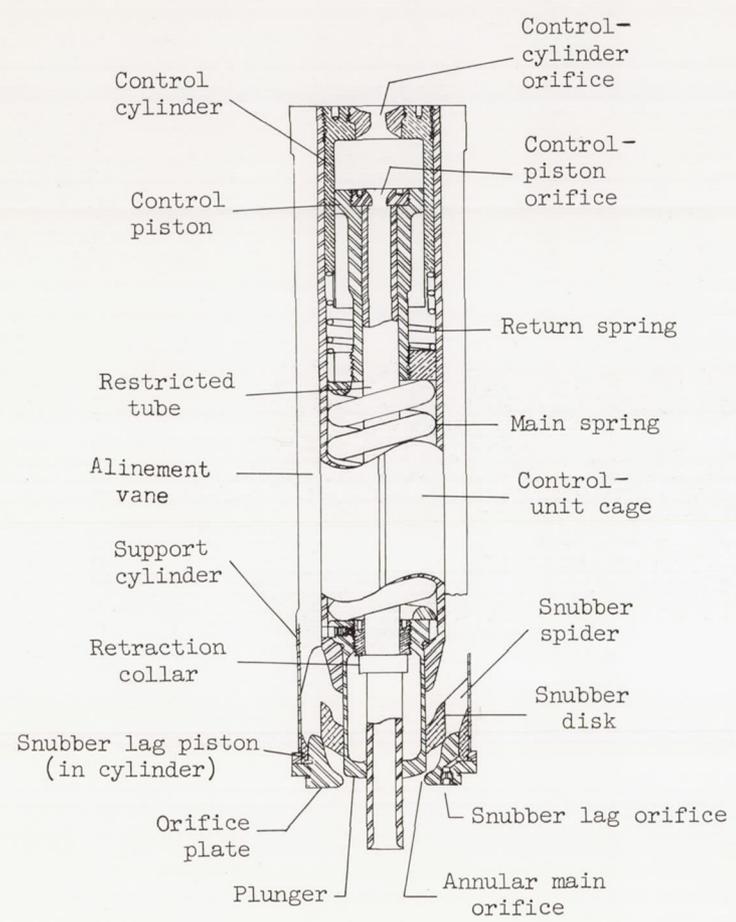
Figure 2.- Views of shock strut and control unit.



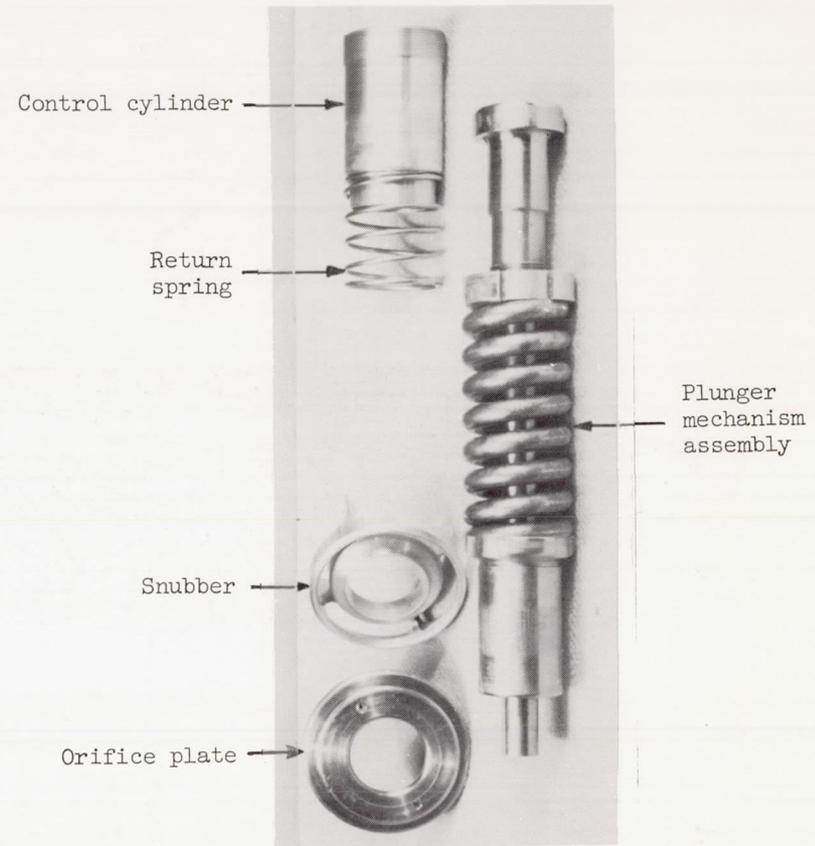
(b) Control unit.

L-58-1691

Figure 2.- Concluded.

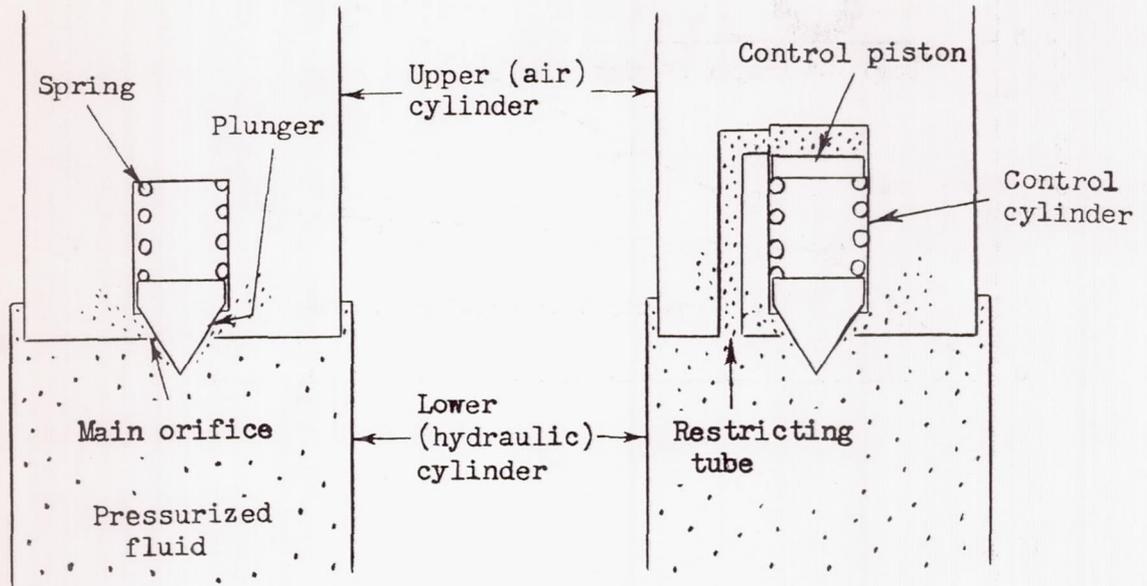


(a) Assembled cross section.



(b) Subassemblies. L-58-1692

Figure 3.- Views showing control-unit construction.



(a) Pressure-actuated relief valve. (b) Rate-actuated low-pass valve.

Figure 4.- Evolution of low-band-pass valve from relief valve.

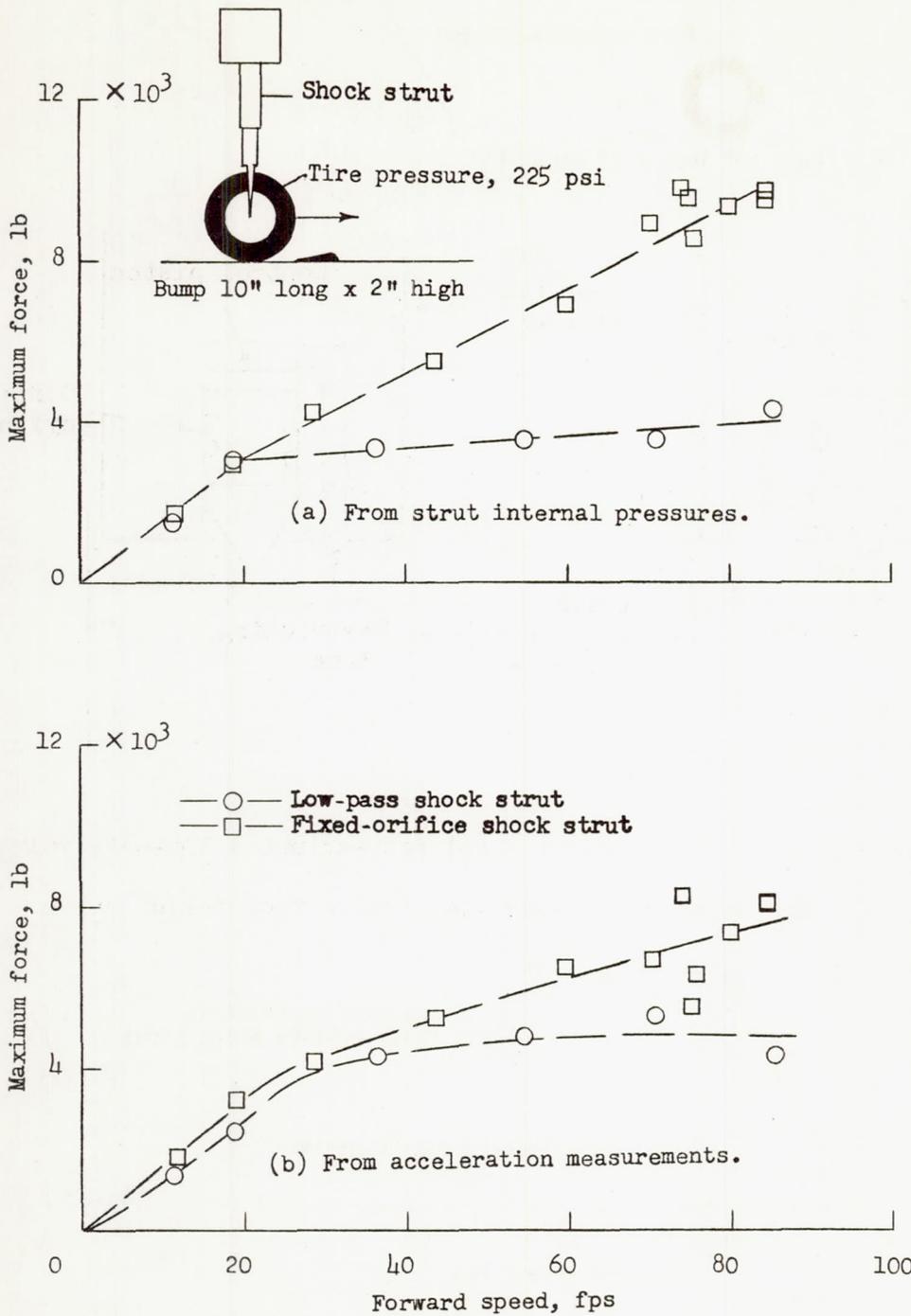


Figure 5.- Comparison of maximum axial shock-strut force for fixed-orifice and low-pass shock struts taxiing over a single bump with a hard tire. Wing lift of 0.88g.

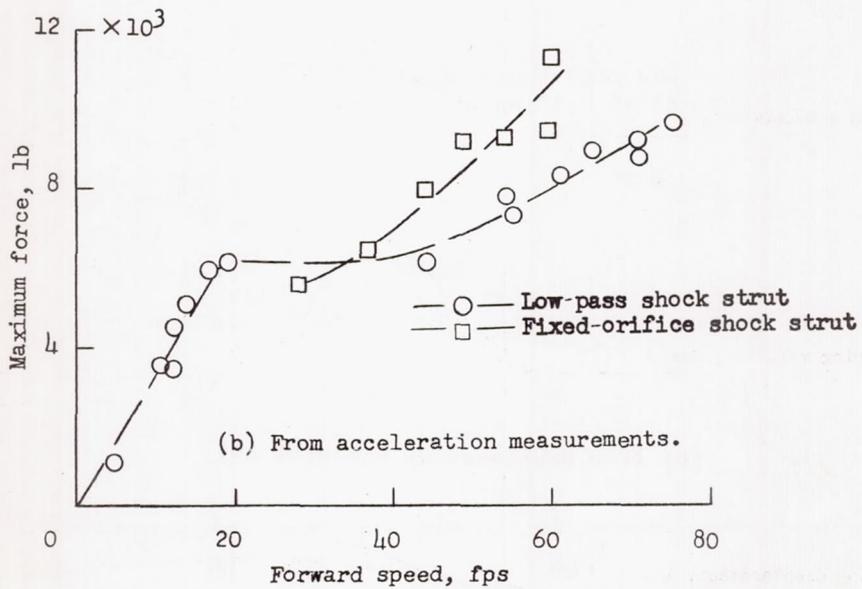
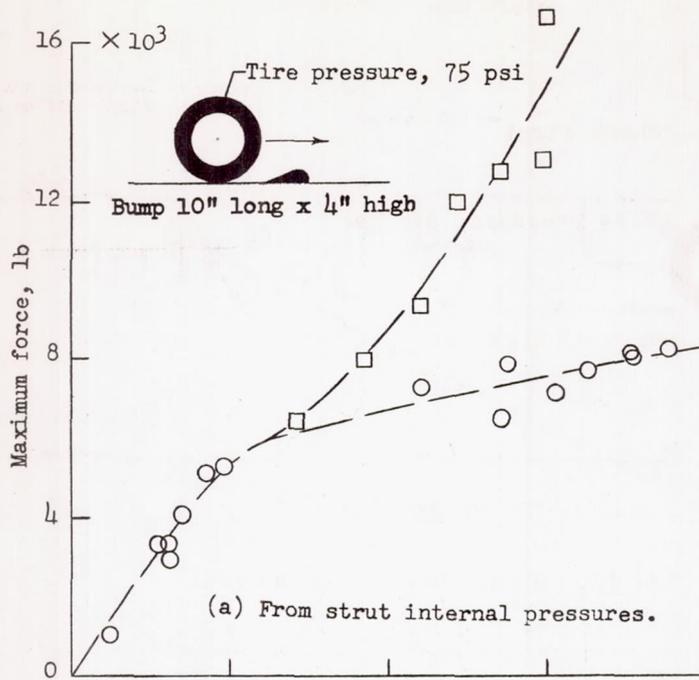


Figure 6.- Comparison of maximum axial shock-strut force for fixed-orifice and low-pass shock struts taxiing over a single bump with a moderately soft tire. Wing lift of 0.88g.

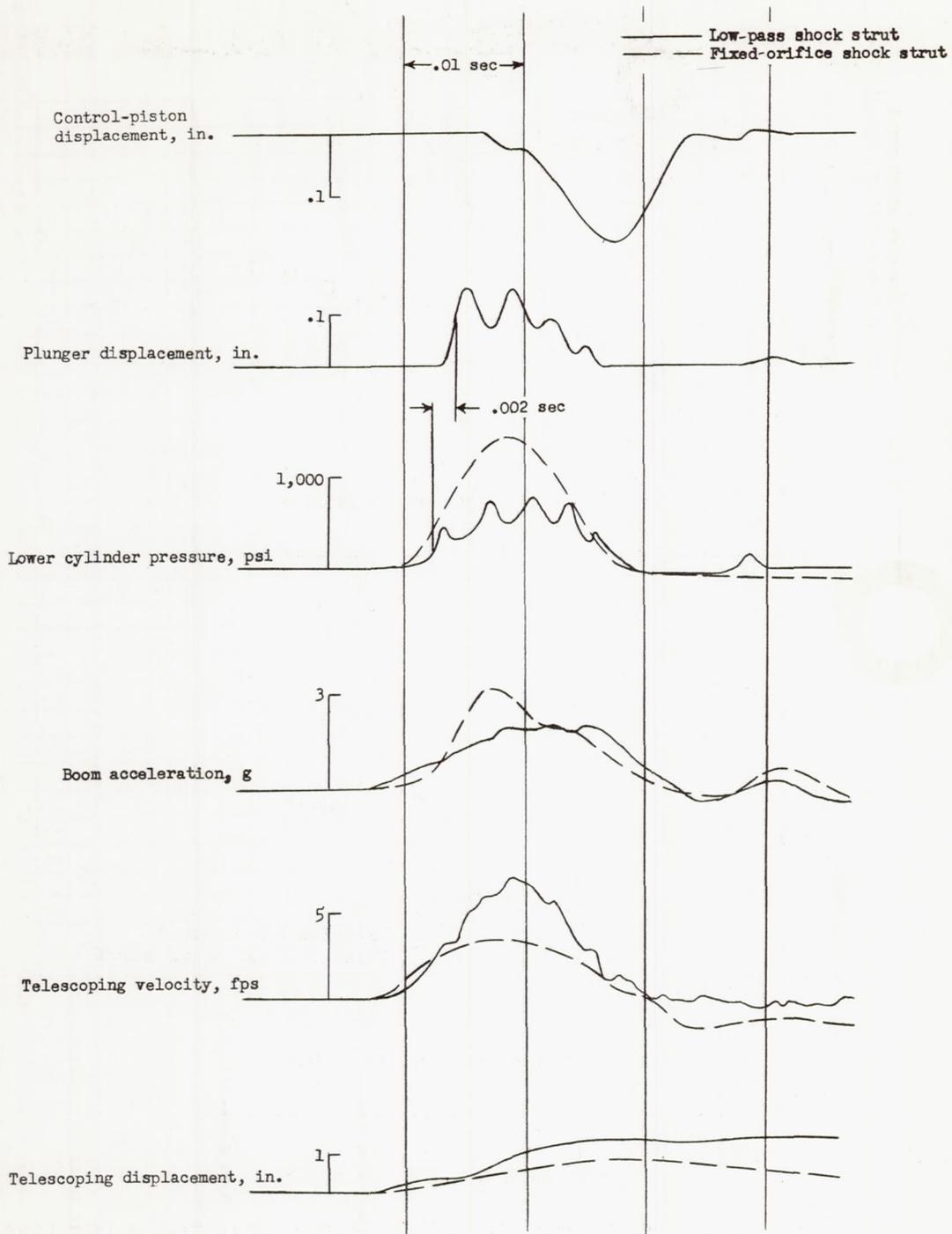


Figure 7.- Time histories for low-pass and fixed-orifice shock struts during taxiing at 85 feet per second over the 2- by 10-inch bump with the hard tire. Wing lift of 0.88g.

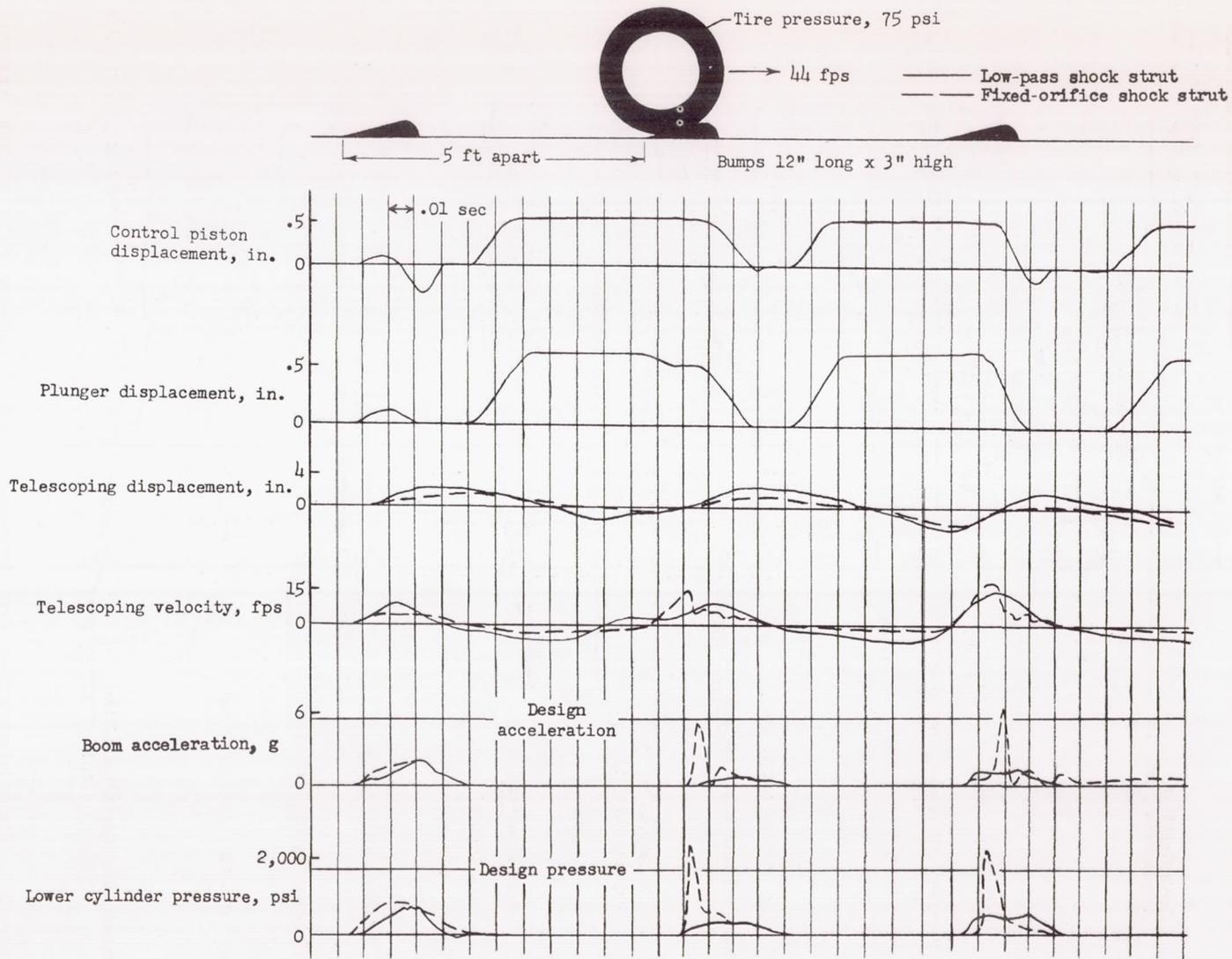


Figure 8.- Time histories for low-pass and fixed-orifice shock struts during taxiing over series of bumps. Wing lift of 0.63g.

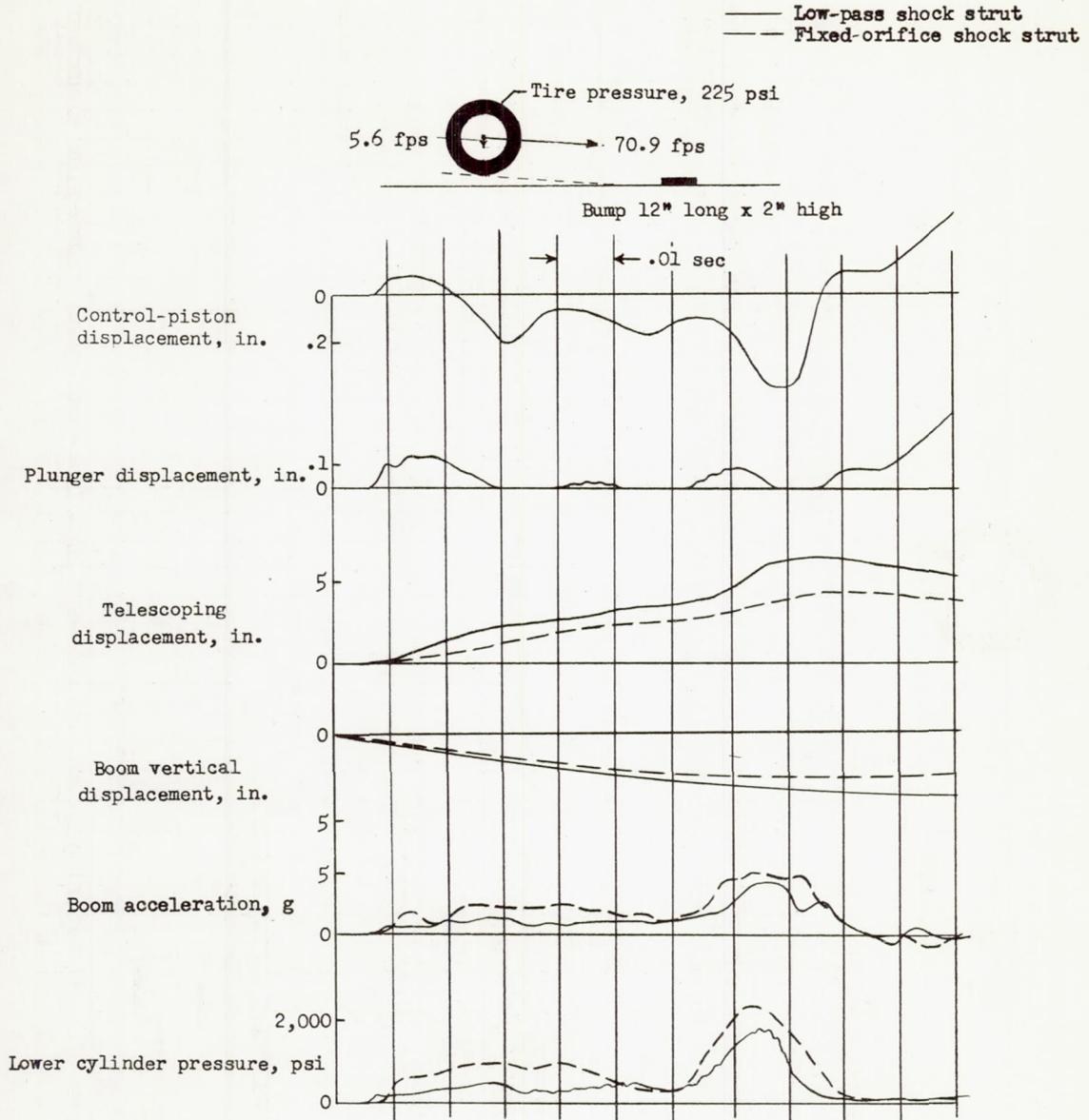


Figure 9.- Time histories for low-pass and fixed-orifice shock struts during impact on a step-shaped bump. Wing lift of 1 g.

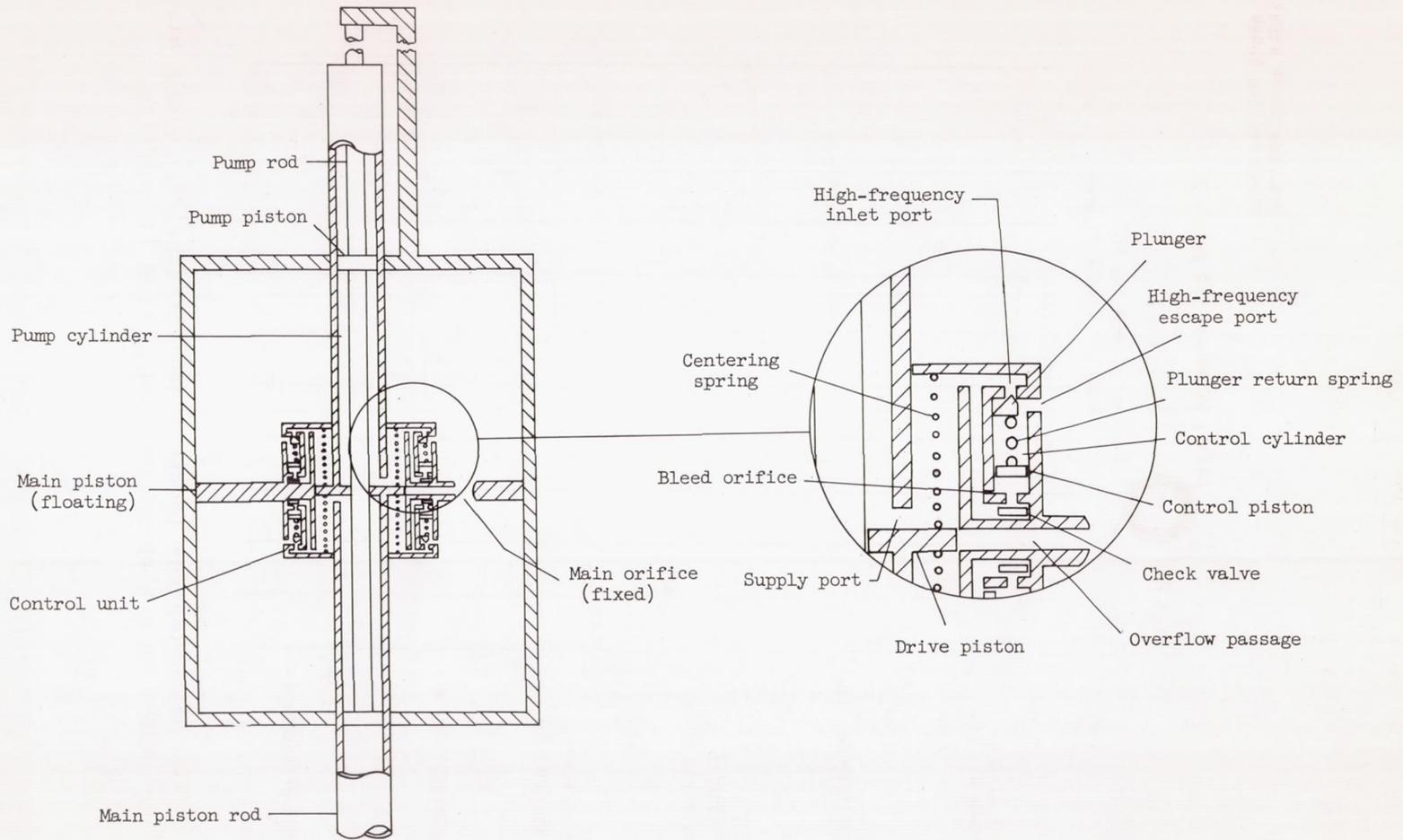


Figure 10.- Floating-piston, double-acting, low-band-pass vibration absorber.

1. The first part of the document discusses the general purpose of the project, which is to provide a comprehensive overview of the current state of the industry and to identify key areas for future research and development.

2. The second part of the document provides a detailed analysis of the market trends and forecasts, including a comparison of the current market with historical data and a projection of future growth.

3. The third part of the document discusses the various factors that are influencing the market, such as technological advancements, regulatory changes, and consumer behavior.

4. The fourth part of the document provides a detailed analysis of the competitive landscape, including a comparison of the strengths and weaknesses of the major players in the market.

5. The fifth part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

6. The sixth part of the document provides a detailed analysis of the various risks that are associated with the industry, and provides a number of recommendations for how to mitigate these risks.

7. The seventh part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

8. The eighth part of the document provides a detailed analysis of the various risks that are associated with the industry, and provides a number of recommendations for how to mitigate these risks.

9. The ninth part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

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11. The eleventh part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

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13. The thirteenth part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

14. The fourteenth part of the document provides a detailed analysis of the various risks that are associated with the industry, and provides a number of recommendations for how to mitigate these risks.

15. The fifteenth part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

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27. The twenty-seventh part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

28. The twenty-eighth part of the document provides a detailed analysis of the various risks that are associated with the industry, and provides a number of recommendations for how to mitigate these risks.

29. The twenty-ninth part of the document discusses the various opportunities and challenges that are facing the industry, and provides a number of recommendations for how to address these issues.

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