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REINFORCED PLASTIC LANDING GEAR FOR UH-1 HELICOPTER

By

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Robert D. Holmes

January 1966

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-120(T)

HAYES INTERNATIONAL CORPORATION
BIRMINGHAM, ALABAMA

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DEPARTMENT OF DEFENSE

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

This program was performed under Contract DA 44-177-AMC-120(T) with Hayes International Corporation, Birmingham, Alabama.

The data contained in this report resulted from research conducted to determine the feasibility of using glass-reinforced plastic for a UH-1 helicopter landing gear. The report contains a design study, a fabrication procedure, and a static and dynamic test evaluation.

These data are presented in the hopes of assisting workers in the same field, and to contribute to those who need design points in the use of glass-reinforced plastics in structural applications.

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REINFORCED PLASTIC LANDING GEAR FOR UH-1 HELICOPTER

Hayes International Corporation Report No. 1188

by

Leon R. Anderson

Robert D. Holmes

Prepared by

Hayes International Corporation
Birmingham, Alabama

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

SUMMARY

The primary objective of this program was to determine the feasibility of using reinforced plastic material for helicopter landing gears. A comprehensive design study was made, a fiber glass reinforced plastic landing gear system for the UH-1 helicopter was fabricated, and a full scale item was subjected to static and drop tests.

To adapt reinforced plastic materials to the UH-1 helicopter landing gear with minimum cost, the design utilized the same method of reacting loads to the aircraft body as is presently used. Reinforced plastic cross members were substituted for the aluminum members. The approach was desirable in that the existing aluminum skids could be used, and no change in helicopter internal structure was necessary.

Investigations were made of various configurations in an attempt to optimize the design and reduce weight. Two fundamental requirements of each design were (1) that the system have a preselected spring rate, which established magnitude of applied loads, and (2) that it possess sufficient strength to react these loads and the specified static test loads.

Numerous cross sections and geometric curves for the profile were investigated. The design selected was a segment of a circular arc with a rectangular solid cross section. The rectangle is considered the most efficient cross section for this application.

Materials used for fabrication of the landing gear were Owens-Corning S-994 HTS 901 twelve end glass roving and Union Carbide ER-2270 epoxy resin with methylnadic anhydride. Fabrication was accomplished by Cincinnati Testing Laboratories (CTL) by a wet winding process. Final design was based on the following properties: Flexural Strength = 183,000 psi; Interlaminar Shear Strength = 7,000 psi; Compressive Strength = 110,000 psi; Flexural Modulus = 8 x 10^6 psi.

The static and drop tests demonstrated a high degree of ruggedness and durability. One of the two reinforced plastic members of the landing gear systems was subjected to four ultimate design static tests and fifteen drop tests without damage or failure. The other member was subjected to all tests except one of the static tests. Load factors at all impact velocities were less for the reinforced plastic landing gear than for the present metal gear. Energy dissipation occurred quite rapidly and spring back was not as severe as anticipated. It was concluded that fiber glass reinforced plastics are feasible materials for this application.

FOREWORD

A program was conducted by Hayes International Corporation to investigate experimentally the feasibility of using reinforced plastic materials for helicopter skid type landing gear. A landing gear for the UH-1 helicopter utilizing fiber glass reinforced plastic was designed, fabricated, and tested. This program was accomplished under Contract DA44-177-AMC-120(T) for the U. S. Army Aviation Materiel Laboratories (formerly USA TRECOM), Fort Eustis , Virginia. The contract was initiated in December 1963. It was originally planned to be accomplished in three phases as follows: Phase I, Design; Phase II, Fabrication and Structural Test; Phase III, Fabrication and Flight Test. However, the program was revised to delete Phase III and a part of Phase II.

The program was conducted under USAAVLABS direction of Mr. J. N. Daniel, Chief, Aircraft Systems and Equipment Division; Mr. S. B. Poteate, Jr., Environmental Effects and Structures Branch; and Mr. D. P. Neverton, Project Engineer.

Principal Hayes' engineers were: L. R. Anderson, Project Engineer; R. D. Holmes, Lead Engineer; E. L. Moak, Analysis, Design, L. B. Wheeler, and J. Stanley; J. C. Cobern, Materials and Processes; and G. D. King, Test. Fabrication of the reinforced plastic components was accomplished by Cincinnati Testing Laboratories (CTL).

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SYMBOLS

A	in^2	Area
BL		Buttock Line
c	in	Distance from Neutral Axis to Extreme Fiber
δ	in	Deflection
Δ	in	Fore and Aft Deflection
E	lb/in^2	Modulus of Elasticity
F	lb/in^2	Allowable Stress
f	lb/in^2	Actual Stress
I	in^4	Moment of Inertia
L	in	Length
M	in-lb	Bending Moment
M. S.		Margin of Safety
P	lb	Axial Force
R		Ratio of Actual to Calculated Loads; Ratio of Actual to Allowable Stresses
S	in^3	Section Modulus
V	lb	Shearing Force
℄		Center Line
⊕		Center of Gravity
		Applied Forces and Moments
		Reacted Forces and Moments

SUBSCRIPTS

cr	Crippling
b	Bending
H	Horizontal
S	Shearing
t	Tensile
V	Vertical
X-X	Reference Axis
Y-Y	Reference Axis

INTRODUCTION

The skid type landing gear is a spring type shock absorber and is particularly suitable for utilizing the special properties of reinforced plastics. Fiber glass reinforced plastics are excellent energy absorbers because of their high usable strength and low modulus. They are extremely resistant to corrosive media, have a high strength to weight ratio, dissipate energy rapidly, and have a greater degree of damping than metal structures.

The skid type landing gear now in service on the UH-1 helicopter consists of two ground skids and two cross members, or struts. The skid and struts are fabricated from aluminum alloy tubing. A "yielding gear" concept is used in the design of the gear which permits permament deformation after sustained hard landings. The concept is current design practice used in several helicopters. It permits the cross member, or strut, to yield at loads well below limit landing loads, hence absorbing a large portion of the landing energy by plastic deformation. This design approach is based on the supposition that replacement of badly deformed cross tubes is acceptable to the user in lieu of the decreased helicopter performance associated with heavier "elastic" members. This practice has been approved by the FAA for civilian helicopters. However, no military specification has specifically approved the practice to date. The manufacturer of the UH-1 helicopter allows a certain amount of permanent set to accumulate in the present landing gear before the maximum required deflection can no longer be taken and replacement is necessary.

The reinforced plastic landing gear design was initiated with two basic goals: (1) a landing gear that would withstand "hard" landings without yielding, therefore eliminating the necessity of frequent replacement, and (2), a landing gear that would meet the requirements as presently specified in Military Specifications MIL-T-6053A, MIL-T-8679, and MIL-T-6898, References 12, 13, and 14. The landing gear system presently on the UH-1 helicopter does not meet the requirements of these specifications. In addition, the reinforced plastic landing gear should have reduced aerodynamic drag and be competitive in weight and cost.

The original program consisted of three phases as follows.

Phase I, Design, consisted of the following:

> Design and analysis of a glass fiber reinforced plastic skid type landing gear for a UH-1 helicopter.

- Fabrication and test of necessary test specimen to substantiate the design and fabrication techniques.

- Preparation of test agenda for static and drop tests.

- Design of drop test fixtures for the static and drop tests simulating the aircraft conditions of weight and center of gravity.

Phase II, Fabrication and Structural Test, consisted of the following:

- Fabrication of two UH-1 landing gear systems for use in static and drop tests.

- Fabrication of test fixtures for the static and drop tests.

- Static and drop tests conducted in accordance with the test agenda approved by the contracting officer.

Phase III, Flight Test, was deleted from the program by USAAVLABS

This report includes the results of an extensive design study, materials investigation, the necessary supporting and design analysis, description of the method of fabrication, and drop test program.

DESIGN REQUIREMENTS

The objective of this exploratory development program was to determine the feasibility of reinforced plastic materials for skid type helicopter landing gears. This was to be accomplished by designing, fabricating, and testing a landing gear system for the UH-1 helicopter utilizing reinforced plastic material. When compared to the existing landing gear, it was to be stronger, have reduced aerodynamic drag, and be competitive in weight and cost.

Design load criteria for the landing gear system were established by the specified test requirements, which call for static and drop tests generally in accordance with MIL-S-6053A, MIL-T-8679 and MIL-T-8698. Gross weight of the helicopter was specified to be 6, 600 pounds.

(a) A load equal to one-half of the maximum vertical reaction at each skid but not greater than 1.0W, shall be applied in a forward, aft, inboard, and outboard direction, each in combination with the vertical load considering each skid independently. The limit sinking speed shall be 8 feet per second in combination with 2W/3 rotor lift at design gross weight.

(b) Level landing - Basic Weight Condition. Design for an ultimate sinking speed impact with the weight distribution that is critical for the main gear and carry-through structure in the level or static attitude.

(c) Nose-up landing - Basic Weight Condition. Design for an ultimate sinking speed impact with the weight distribution that is critical for the main gear and carry-through structure in the nose-up attitude.

(c) Rolled landing - Basic Weight Condition. Design for an ultimate sinking speed impact with the weight distribution that is critical for the main gear and carry-through structure in the rolled attitude.

(e) Critical symmetrical - Overload Weight Condition. Design for a limit sinking speed impact with a weight of 1.15 times the basic landing design gross weight. The weight distribution to be critical for the main gear and carry-through structure in the most critical symmetrical attitude.

Because of the peculiarity of the skid-type landing gear and more particularly the one-piece cross members of the UH-1 gear system, it was difficult to clearly define design criteria as pertain to this type of landing gear.

Reference 14 was used as a guide in interpreting design strength requirements and establishing factors of safety. The ultimate strength of helicopter landing gears is specified by two requirements in Reference 14, stated as follows:

(a) The structure shall support, without failure, ultimate loads resulting from loading conditions incorporating an ultimate factor of safety of 1.5.

(b) During the reserve energy drop test demonstration, failure of the structure shall not occur at a vertical descent velocity equal to the limit vertical descent velocity times the square root of 1.5.

Of these two requirements, the first is specified as a factor of safety for the entire aircraft and,therefore, may be interpreted as a requirement for the landing gear. However, within the section on landing gear requirements, the second is also specified as an ultimate limitation. Since the landing conditions are determined from a contact limit vertical descent velocity and limit inertia load factor, the specification is interpreted to exclude the factor of safety of 1.5 in the landing gear system and to design the ultimate strength from the reserve energy requirements.

The requirements of Reference 14 also include maintaining a yield factor of safety of 1.0 based on limit loads. It was assumed that this requirement was intended for metal design since it contains ambiguity when associated with a material that has no yield point. Glass reinforced plastic is such a material. At all stresses less than its failure point, the landing gear will return to its original position.

Therefore, it has been assumed that a landing gear that will withstand the reserve energy tests will demonstrate the required structural integrity and fulfill specification requirements. This is the factor of safety criterion used in the design herein presented.

MATERIALS

The choice of feasible reinforced plastic materials was quite limited. The optimum material must have high flexural strength, low flexural modulus relative to metals, and high shear strength. Minimum weight and maximum efficiency can be obtained only with directionally oriented glass fiber reinforcements. The use of woven fabrics for this application is not feasible.

Preimpregnated unidirectional materials were given first consideration. It was believed that they are more adaptable to the feasible fabrication processes. Materials for wet lay-up were also investigated and were used in a winding process for the fabrication of the cross members of the landing gear.

Several materials covering a wide range of mechanical properties were investigated, and preliminary designs were made in order to evaluate their use. The final design was based on the following properties:

Flexural Strength = 183,000 psi

Interlaminar Shear Strength = 7,000 psi

Compression Strength = 110,000 psi

Flexural Modulus = 8×10^6 psi

Materials having lower properties result in increased weight of the landing gear. From investigations and tests it was concluded that these properties were as high as could be obtained with the limited development that could be accomplished in this program. They are appreciably higher than any known values that have been obtained in thick sections, and it is felt that they cannot be significantly increased with presently available materials. Newer developments in hollow fibers, specially shaped fibers, whiskers, and resins will undoubtedly result in somewhat better mechanical properties, especially in shear.

Materials used for fabrication of the landing gear for test were Owens-Corning S-994 HTS 901 twelve end glass roving and Union Carbide ER-2270 Bakelite* epoxy resin with methylnadic anhydride. Fabrication was by a wet winding process. Undoubtedly these properties can be obtained with other materials such as Stratoglas** 600St and 660St. "Scotchply"*** SP251S and other combinations of glass fibers and resins. A process using a wet winding instead of preimpregnated materials was used because it was felt that better control of the composite could

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*Registered Trade Mark, Union Carbide and Carbon Corporation.

**Registered Trade Mark, Air Logistics Corporation.

***Registered Trade Mark, Minnesota Mining and Manufacturing Company.

be accomplished for less cost for this experimental fabrication. Preliminary tests also indicated that better mechanical properties could be obtained by the fabricator using these materials. Fabrication for quantity production should consider preimpregnated materials, especially in the form of unidirectional tapes. Their use would require more elaborate tensioning and control devices, but could result in less fabrication time required and therefore a reduction in cost.

Some concern has been expressed of the possibility of a higher void content in a wet winding process than would be obtained if preimpregnated materials were used. This was investigated, and it was concluded that with close control in wet winding the void and mechanical properties of the fabricated parts would not differ from those fabricated with preimpregnated materials.

Following is a description of other materials that were considered in this study. Table I summarizes their mechanical properties as published in manufacturers' technical data sheets.

"Scotchply" Minnesota Mining & Manufacturing Company, St. Paul, Minn.

"Scotchply" brand reinforced plastic is a preimpregnated, high strength, moldable, laminated epoxy plastic reinforced with continuous nonwoven glass filaments that are straight and parallel, not crimped or woven. Individual plies may be oriented to meet specific stress requirements. It can be cured by the application of heat and pressure by matched metal dies, vacuum or pressure bag molding. Best mechanical properties can be obtained by molding at high pressure: 25 to 100 psi. It can be vacuum molded at 10 to 15 psi; however, there is a 10- to 25-percent reduction in mechanical properties.

"Scotchply"Type 1000 has a low exothermic reaction during the cure cycle, making it possible to mold parts in thickness up to 6 inches or more. The 3M* Company recommends it, especially where sections greater than one inch are to be molded. It produces high-strength mechanical properties at temperatures up to 200°F. It is chemically stable at higher temperatures and can be used in the temperature range of 200°F to 350°F; however, mechanical properties decrease rapidly with an increase in temperature (reference 3M Technical Data Sheet 3, dated 1 January 1958).

"Scotchply" Type 1002 is recommended by 3M for general purpose applications requiring high strength over a temperature range of -60°F to

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*3M Company is another name for Minnesota Mining & Manufacturing Co.

$+250^{\circ}F$. The exotherm developed by the resin during cure becomes a problem when the laminate thickness exceeds one inch. This material has a somewhat longer shelf life than Type 1000 (reference 3M Technical Data Sheet, dated 1 January 1963).

"Scotchply" Type 1002S is a modification of Standard Type 1002 reinforced with continuous nonwoven glass filaments of high-tensile-strength S-994 glass. Mechanical properties are improved over that of Type 1002, although fabrication methods and other characteristics are similar (reference 3M Technical Data Sheet 2, dated 1 October 1963).

"Scotchply" Type XP-251S is a high-strength unidirectional tape or roving made with epoxy resin having improved interlaminar shear properties. The tape, or roving, is reinforced with S-901 glass and the resin has improved shelf life and tack. Exotherm during cure does not limit thickness in use of this material (reference 3M Technical Data Sheet 2, dated 20 May 1964).

Stratoglas, Air Logistics Corporation, Foothill Blvd., Pasadena, Calif.

Stratoglas materials are epoxy-glass preimpregnated molding materials similar to "Scotchply", described above. Types considered were Types 300T, 600ST, and 660ST. The material is available in tape, roving, and sheet form in single thickness and in combinations of multi-ply configurations. The stratoglas 600 resin system is applied to wide widths or unidirectional glass such as HTSE, S-994 or 801.

NUF*, Ferro Corporation, Fiber Glass Road, Nashville, Tennessee

NUF is a nonwoven unidirectional fabric composed of parellel, continuous longitudinal strands of glass fiber, cross-bonded every three inches with resin-ladened fill or cross strands. Ferro Corporation produces NUF only in dry form. Other firms, such as Cordo Chemical Corporation, preimpregnate the material. Although NUF does not have as high mechanical properties as "Scotchply" and Stratoglas, it does have a significant price advantage and therefore was considered.

The preimpregnated materials that appear to be the most feasible are 3M "Scotchply" XP-251S, and Air Logistics Stratoglas. Extensive design studies were conducted to optimize designs for these materials. Very little information is available on mechanical properties of thick sections

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*Trade Name for nonwoven unidirectional fabric, Ferro Corporation

TABLE I

SUMMARY OF MECHANICAL PROPERTIES OF PREIMPREGNATED UNIDIRECTIONAL GLASS-EPOXY LAMINATES

T = 70° F

Item	"Scotchply" Type 1000 (1)	"Scotchply" Type 1002 (2)	"Scotchply" Type 1002S (3)	"Scotchply" Type XP251S (4)	Stratoglas 300T (5)	Stratoglas 600ST (6)	Stratoglas 660ST (7)	NUF (8)
Tensile Strength, psi x 10	98	160	195	270	135	238	183	120
Modulus in Tension, psi x 10	5. 3	5. 7	6. 4	8. 4	6. 5	8. 27	8. 0	-
Flexural Strength, psi x 10	113	165	175	200	155	275	200	130
Modulus in Flexure, psi x 10	4. 7	5. 3	5. 8	8. 0	6. 0	8. 27	8. 0	4. 0
Edgewise Compression, psi x 10	73	90	110	120	-	98. 7	-	50
Interlaminar Shear, psi x 10	-	4. 3	5. 4	11. 2	-	-	7. 0	-
1200 Edgewise Impact, ft-lb/in	64	60. 8	-	-	60	80	-	-
Cured Ply Thickness, in	0. 0083	0. 010	0. 010	0. 007	0. 005	0. 005	0 . 005	-
Resin Content by Weight, pct	34	36	34	22	22	22	28	35
Barcol Hardness	70	70	75	-	65 min	-	65 min	-
Specific Gravity	1. 8	1. 8	1. 8	2. 0	-	-	-	-
Density	0. 068	0. 068	-	-	0 . 075	0 . 078	0 . 078	-
Wet Strength Retention, 2 Hr Boil, pct	90	86	–	-	-	-	-	-
Shelf Life	2 mo 70°F	6 mo 70°F	6 mo 75°F	1 mo 70°F	1 yr 40°F	4 days, 68°F Indef, 0°F		-

(1) 3M Company Technical Data Sheet No. 3, 1 Jan. 1958
(2) 3M Company Technical Data Sheet, 1 Jan. 1963
(3) 3M Company Technical Data Sheet No. 2, 1 Oct. 1963
(4) 3M Company Technical Data Sheet No. 2, 20 May 1964
(5) Air Logistics Corp. Technical Bulletin No. 006, 11 Feb. 1963
(6) Air Logistics Corp. Technical Bulletin No. 007, 8 May 1963
(7) Air Logistics Corp. Technical Bulletin No. 005, 8 May 1963
(8) Ferro Corp., "Fiber Glass Reinforcements"

of these materials. Flexural strength such as shown in Table I are for relatively thin laminates, usually 0.125 inch thick, and are not necessarily representative of the strength that can be obtained from thick sections of the same material. There is some disagreement in the industry on whether the bending strength of thick laminates is limited by the compression strength or flexural stress or the compressive stress calculated from the flexure formula.

In an effort to obtain basic data on thick sections, a laboratory program using "Scotchply" was conducted. Several thick laminated solid sections using Type 1000 and hollow sections using Type 1002 were fabricated and tested in flexure, compression and shear. Testing was accomplished in accordance with Reference 5, Federal Specification LP-406b.

In the test program, thick fiber glass bars were fabricated from which specimens were cut. For some beam tests, however, the entire bar as fabricated was tested. The 3M Company provided Hayes with several thick laminated bars from which additional specimens were taken for testing.

In fabricating the fiber glass bars, a matched mold process was used. Two-inch-wide "Scotchply" tape was used to build up the rectangular bar laminated sections in a 36-inch length. Depth, or thickness, of the bars varied from 1-1/8 inches to 1-1/2 inches. Pressure was applied while the part was cured in an oven under high temperature (above 300°F). A post-cure cycle at 280°F was then performed during which pressure was removed from the part.

All specimens of Type 1000 material were of solid laminate construction. The specimens of Type 1002 material either contained a honeycomb core or were hollow. The cores were 1 inch wide by 1/2 inch deep.

The strength of these materials is somewhat less than required for the landing gear. The tests, however, did prove that flexural stresses considerably higher than the recognized compression strength of the material could be obtained.

A tabulation of specimens and test results is given in Table II. These results are not necessarily representative of maximum properties that can be obtained from the materials. They indicate that the fabrication process was not optimzied. Future testing of these materials was deleted when it became apparent that they could not be used and a sub-contractor (CTL) was found that would fabricate the components by a winding process and obtain the specific properties required. Some preliminary laboratory testing was accomplished in order to choose a material. Further testing was accomplished by full-scale windings. See the section on fabrication for results.

The strength of the specimens tested, including those fabricated by Hayes and those supplied by the 3M Company, was less than expected. However, higher values have been reported by other organizations, and it is reasonable to assume that values of mechanical properties for "Scotchply" of the magnitude used in design can be obtained. Additional experimentation with process control, using molds for the specific components, will be required.

"Scotchply" Types 1002 and 1002S have a high exotherm reaction and when used in thick sections tend to overcure. Although higher allowable strength can be obtained with Types 1002 and 1002S than with Type 1000, the former cannot be fabricated successfully in solid sections of the thickness required. Thin-walled hollow sections using Type 1002 were investigated. The low shear strength and fabrication problems make hollow sections and the 1002 series materials undesirable for this application. "Scotchply" Type 1000 can be laminated to the required thickness.

Air Logistics Stratoglas and 3M Type XP-251S appear to be feasible materials for this application. Their higher strength properties in a solid cross section result in the optimum reinforced plastic landing gear for the UH-1 helicopter. No laboratory tests were made by Hayes using these materials, but guaranteed properties were obtained from Air Logistics for Stratoglas and Cincinnati Testing Laboratories for XP-251S. CTL could also obtain the required properties by using S-994 glass fibers and one of several available high-strength resins.

TABLE II

SUMMARY OF LABORATORY TESTS OF THICK BEAMS

Test No.	Fabricated By	Type Material	Test Specimen			Type Test	Failure Stress (psi)	Remarks
			Number of Plies	Depth (in)	Width (in)			
1	3M	1000	90	0.746	1.611	Flexure	113,000	Outer edge cross plies failed at 10,000-psi range. Remaining net section used to determine final failure stress, Flatwise flexure.
2	3M	1000	90	0.745	1.602	"	109,000	
3	3M	1000	90	0.748	1.602	"	112,000	
4	3M	1000	90	0.815	0.745	"	103,000	Edgewise flexure
5	3M	1000	90	0.817	0.743	"	105,000	Edgewise flexure
6	Hayes	1000	136	1.261	1.981	"	75,000	Flatwise flexure
7	Hayes	1000	136	1.116	1.985	"	68,000	Flatwise flexure
8	Hayes	1000	136	1.126	1.990	"	90,000	Flatwise flexure
9	Hayes	1000	146	1.206	1.990	"	72,500	Flatwise flexure
10	Hayes	1002	127	1.274	1.998	"	81,300	Honeycomb core (open cells), Flatwise flexure
11	Hayes	1002	150	1.500	2.000	"	94,500	Honeycomb core (filled cells)
12	Hayes	1000	Shear Area =0.474 in^2			Interlaminar Shear	5,840	Notch method test with backup side plates
13	Hayes	1000	Shear Area = 0.519 in^2				5,700	
14	3M	1000	2.00 x0.745 x0.812			Compres-sion	68,800	
15	3M	1000	2.00 x 0.746 x 0.810				67,400	
16	Hayes	1000	1.04 x0.455 x0.490			"	73,400	
17	Hayes	1000	1.03 x0.495 x0.500			"	62,300	
18	Hayes	1000	1.03 x0.489 x0.466			"	57,900	
19	Hayes	1002	1.03 x 0.514 x0.491			"	75,500	
20	Hayes	1002	1.04 x 0.493 x0.473			"	71,000	
21	Hayes	1002	1.04 x0.482 x0.490			"	72,800	

CONFIGURATION STUDY

Many factors had to be considered in the design of a landing gear for the UH-1 helicopter using reinforced plastic materials. Some of the more important considerations leading to the final design were strength, weight, aircraft ground attitude, ground clearance, spring rate, aerodynamic drag, ease of fabrication, cost, and adaptation to the UH-1 with minimum modification.

A number of various configurations were investigated during the study phase. Early in the program it was concluded that the best way to economically fulfill the objectives of determining the feasibility of reinforced plastic materials for skid type landing gears was to use the same design concept as the present metal landing gear. This was accomplished by substituting reinforced plastic for the two cross members. This approach was desirable in that the existing aluminum skids could be used, and no change of helicopter internal structure would be necessary to react landing loads from the new gear system.

The fuselage structure of the UH-1 was designed to react only axial loads from the landing gear. Any landing gear system that would put torsion or bending moments into the fuselage would require a major redesign of the frame structure. Therefore, it was necessary to use a one-piece carry-through strut similar to the present aluminum member that was free to rotate and pivot at the fuselage attachment points.

The design selected is shown in Figure 1. It consists of two fiber glass reinforced plastic cross members or struts that are segments of circular arcs. The cross members were of rectangular cross section having a constant width of 2.2 inches and an inner radius of 60.8 inches. The thickness of the aft member was 2.0 inches and the forward member 1.6 inches. The present skid assemblies are used without modification. The skids were standard UH-1 helicopter parts constructed from 4.0-inch O.D. x 0.095-inch wall 2024-T3 aluminum alloy tubing, Bell Part No. FSN-1620-070-7848 and 7849.

Attachment of the cross members to the skids was made by means of the same aluminum saddle fittings used to attach aluminum struts to the skids. Aluminum shims were bonded to the fiber glass struts using 3M-EC-2216 A/B adhesive and then bolted through existing holes in the saddle fittings. These shims were of a half moon shape cross section and transitioned the rectangular cross section of the struts to the circular opening in the saddle fittings. The holes in the cross members were oversized to prevent load being transmitted to the bolts through bearing in the cross member. Axial load, therefore, was transferred from the cross member through the bonded joint to the shims, and then through the bolts to the saddle fittings. The saddles were stabilized for bending loads by ejecting an epoxy resin into the cavity after the struts were assembled to the skids.

Installation to the helicopter was accomplished in much the same way as the present landing gear system. This was done by clamping and bonding a two-piece steel ball type fitting around the rectangular member at B L 14 right and left. The ball fitting contacted a plate,and vertical loads were reacted by bearing along a radius. Flanges on the fittings react lateral loads. A retaining strap under each fitting was loaded only by the weight of the landing gear assembly. The joint was free to rotate in a vertical plane.

Comprehensive investigations were made of various configurations using this concept in attempts to optimize and reduce weight. Two fundamental requirements of each preliminary design, however, were (1) that the gear have preselected spring rate, which established magnitude of applied loads, and (2) that it possess sufficient strength to react the applied loads. This line of thought led to a multitude of strut profiles, cross sections and taper rates.

Several continuous polynomial geometrical curves were investigated for the profile of the new strut. The boundary restrictions of each curve were that it attach to each skid at BL 48, fit within the aircraft landing gear well, and have increased ground clearance over the aluminum gear system. The increased ground clearance was necessary for gears of lower spring rate in reducing load factors but still retaining the present aircraft static attitude. Segments of cubic, elliptical and circular equations were found to satisfy these boundary conditions. None of the cubic or elliptic curves were found to offer any advantage over a circular profile. Therefore, for the sake of simplicity in both analysis and fabrication, the circular segment was chosen for the profile design of the reinforced plastic.

In conjunction with selecting the strut configuration, an independent analysis of spring rate requirements was performed. Hypothetical landings were made to determine landing gear loads for various spring rates. Figures 2 and 3 present curves that can be used to determine maximum gear loads and deflections for various ratios of gear spring constants. Once the desired spring rate of the gear was determined, it was possible to calculate cross section stiffness requirements. With the landing load

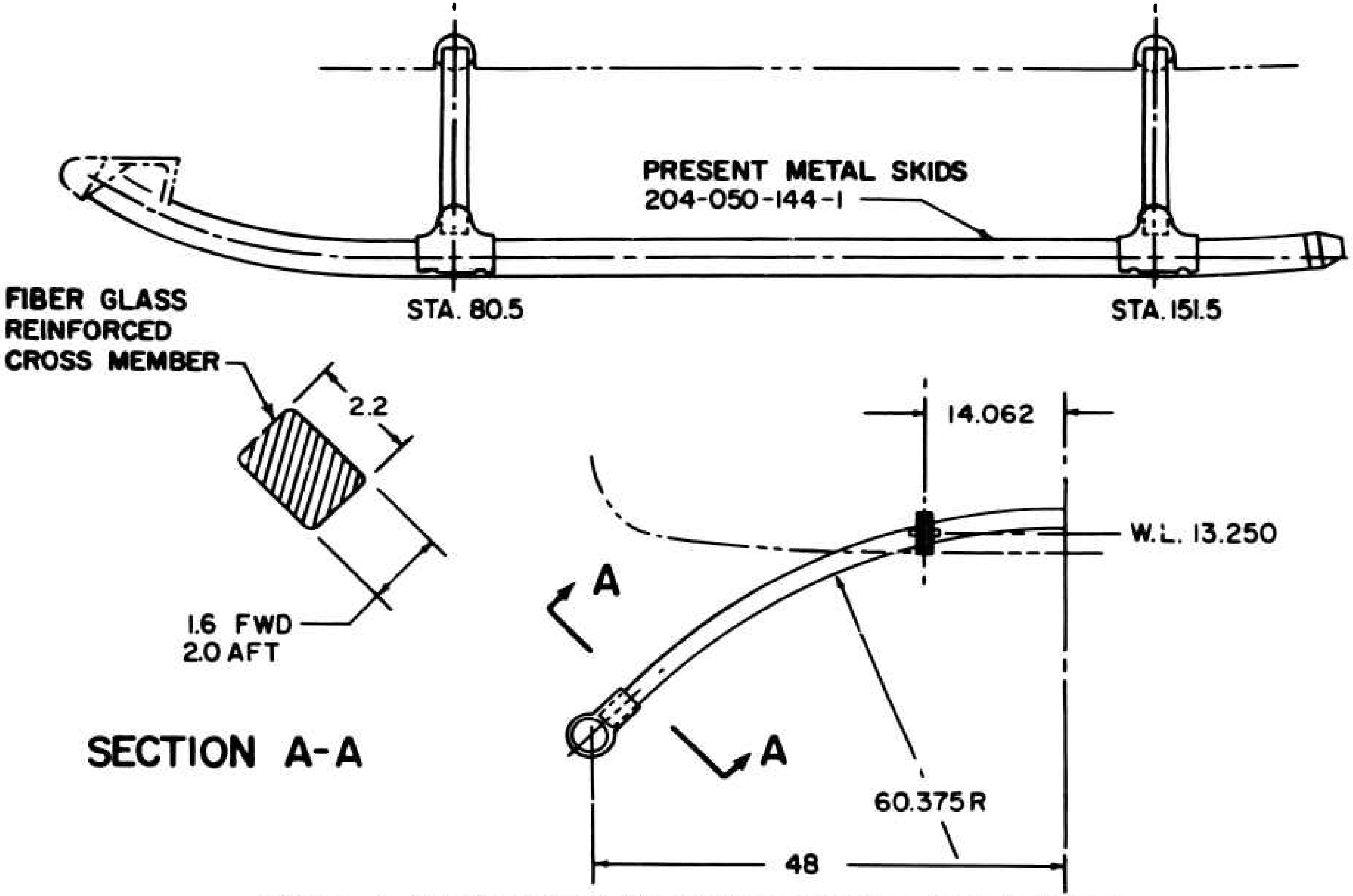
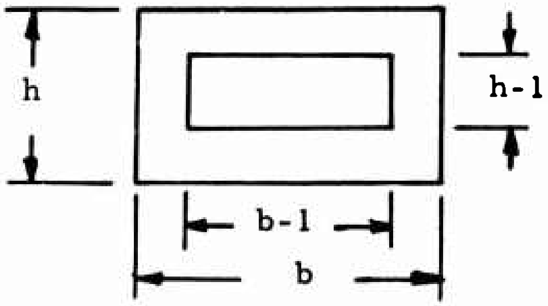


Figure 1. Reinforced Plastic Landing Gear For UH-1 Helicopter.

known and stiffness requirements determined, internal stresses could be calculated for comparison with the material mechanical properties. Various symmetrical sections were investigated using tubular, solid and composite construction from rectangular, oval, and circular cross sections.

It was soon discovered both from specimen tests and through industry inquiries that certain peculiarities of fiber glass reinforced plastic render the solid cross member more favorable than the tubular cross member. Flexural strengths of a solid beam are considerably higher than the usual allowable compressive stresses that cause failure. In addition, local crippling of the cross section is not a problem for the solid section. For the tubular cross section beam, failure stresses are more nearly equal to the allowable compressive stress. Further, the solid section reduces maximum interlaminar shear stresses - a very weak link characteristic of fiber glass plastics. Such a difference in failure stresses causes the solid member to be the more practical cross section. The importance of high flexural stress is illustrated in the following example. Assume that the section moment of inertia, I, necessary to give the required spring rate for $E = 8 \times 10^6$ psi is 1.47 in^4. Also assume that the landing load bending moment M_1 resulting from the given spring rate is 269,000 inch-pounds.

The hollow members required for two different allowable stresses are as follows. To avoid buckling problems and for required interlaminar shear strength, neglecting torsion, assume a constant wall thickness of 1/2 inch.



Design A

Let F = 183,000 psi (Flexural Strength)

c = FI/M = 183,000(1.47)/269,000 = 1.00 inch

h = 2c = 2.00 inches

$$I = \frac{bh^3-(b-1)(h-1)^3}{12}$$

Substituting $h = 2.00$ and $I = 1.47$,

and $b = 2.38$ inches

Design B

Let $F = 100,000$ psi

$c = 100,000(1.47)/269,000 = 0.546$

$h = 2c = 1.09$

$b = 13.50$ inches

Obviously, design B is unrealistic. Similarly, design A is also unreal since the high flexural stress of 183, 000 psi cannot be obtained in the hollow section design. Further, very high shear stresses are induced at the inside corners which make this design undesirable. Therefore, it is necessary to use a solid cross section where the high flexural stress could be obtained, and shear stress concentration is not a problem.

Design C

$c = FI/M = 183,000(1.47)/269,000 = 1.00$ inch

$h = 2c = 2.00$ inches

$$I = \frac{bh^3}{12} \quad \text{or,}$$

$$b = \frac{12I}{h^3} = \frac{12(1.47)}{(2.00)^3} = 2.20 \text{ inches}$$

The number in design C actually represents a lighter weight member when compared to hollow sections which have lower stress allowables.

Composite constructions investigated used a lightweight core such as honeycomb or wood in order to reduce weight. Shear stresses, however, are higher than permissible in lightweight cores, so the core could be used only as a nonstructural filler material. Therefore, the resulting design would be similar to the tubular design whose disadvantages were discussed above.

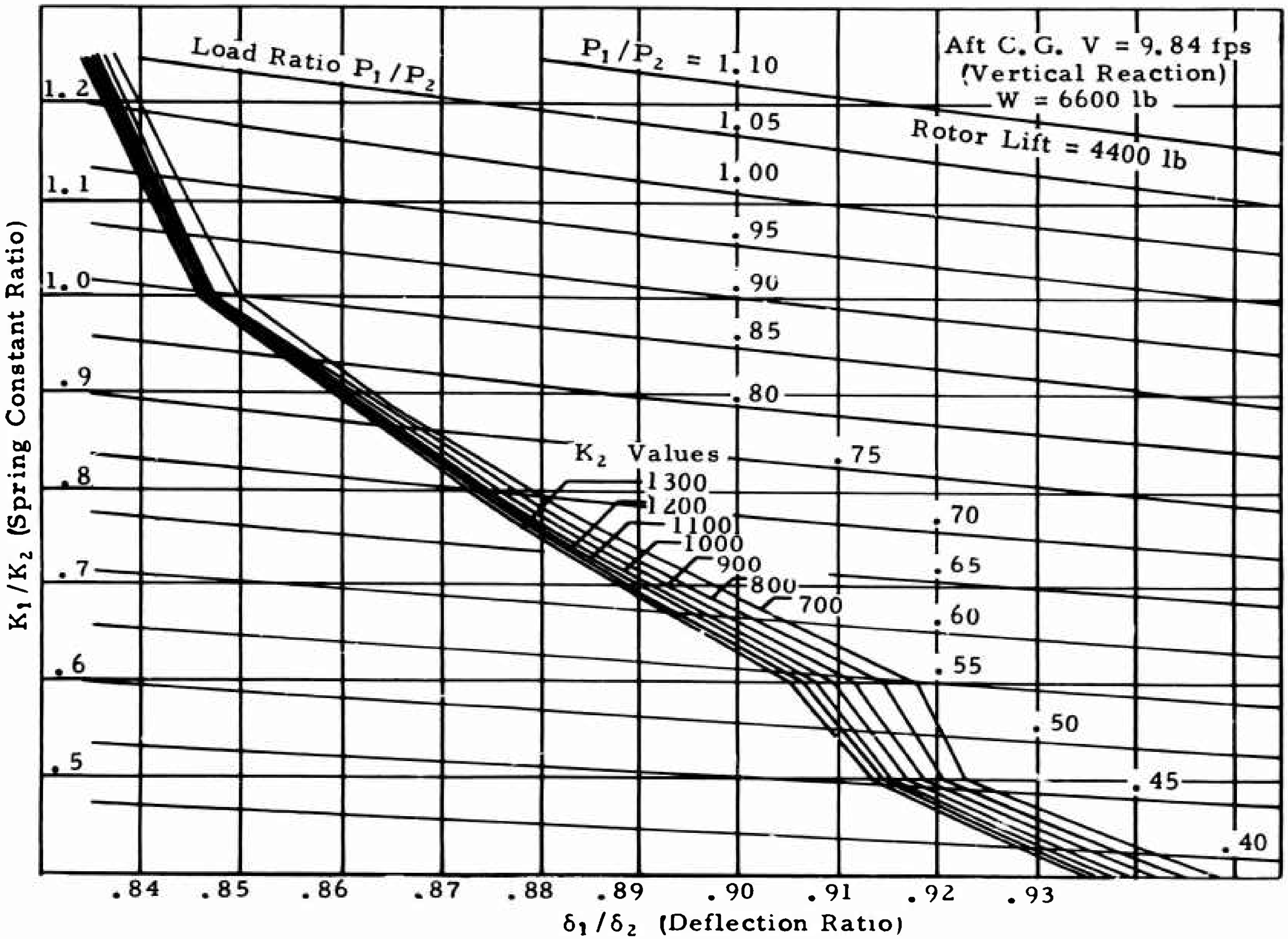


Figure 2. Ratios of Spring Constants Versus Loads and Deflections.

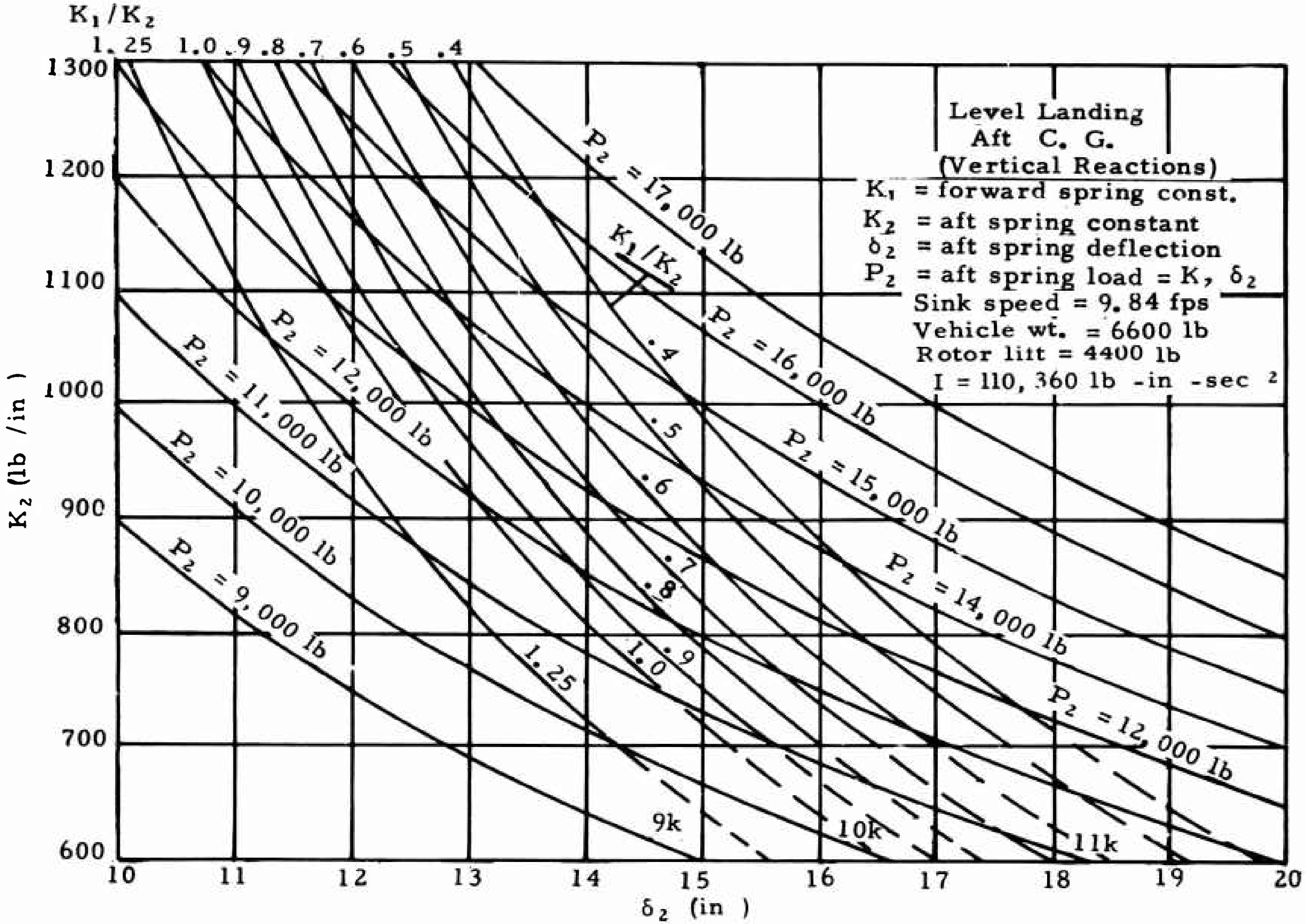


Figure 3. Spring Deflections for Various Spring Constants,

A solid elliptical cross section was studied as a means of reducing aerodynamic drag. The L/d ratio of an elliptical section that would offer significant drag reduction and still meet strength and spring rate requirements made the member too wide to pass through the helicopter landing gear well. The larger width contributes to stiffness requirements at a slower rate than does depth of the member, and the resulting strut is considerably heavier than a strut of shorter width and greater depth.

With increased complexity of fabrication added to the aforementioned disadvantages of various sections, it was decided that the best overall cross section would be a solid rectangular shape. The cross section having been finalized, it was then necessary to determine actual strut dimensions.

It was found that for a flat landing, loads applied in a vertical direction, the most efficient strut is a strut tapering in depth from a relatively thin section at the skids to greatest thickness at the point of attachment to the helicopter. Here, the strut is placed under transverse shear and bending loads about the strut horizontal axis.

Designs of the above type found adequate for flat landings were used to determine their adequacy for a case where loads act in a longitudinal direction in conjunction with vertical loads. Under large flexure from loads in a vertical and longitudinal direction, the cross member is placed under transverse shear and bending about the x and z axes, and, in addition, high torsion loads are introduced about the y axis. It is the torsion loads that tend to overstress the narrow end of a tapered cross section, thereby rendering the tapered strut inadequate for this loading condition. This is because of the very low interlaminar shear allowable of fiber glass reinforced plastics and because of the low torque-carrying capability of rectangular cross sections. Thus, in order to reduce torsion stresses at the skid ends of the strut, it is necessary to retain a thick cross section. Therefore, it follows that the required strut must not taper, but must remain a constant cross section throughout its entire length.

In view of the foregoing, a member of circular profile with a solid rectangular cross section was selected for the landing gear design. The actual width and depth dimensions were determined by the spring rate and stress requirements. To reduce aerodynamic drag, molded nylon leading edges were incorporated into the configuration. Being nonstructural, the nylon fairings do not continue over the unexposed portion of the strut, i.e., through the landing gear well area.

A solid rectangular cross section having been selected for the design, much concern was given to minimizing weight. Weight was found to be controlled by several parameters: density, modulus of elasticity, allowable flexural stress, allowable interlaminar shear stress, and design factors of safety.

With the exception of design factors of safety, a material's mechanical properties determine final weight of a design. The selected safety factor, for all practical purposes, causes weight changes that are independent of the material being used.

Table III presents a summary of various cross section designs that were investigated under Phase I of this program. The statements given under "Remarks" in Table III generally refer to the landing gear system of the UH-1 Helicopter and to the manner in which loads are distributed in this gear system. A cross section not feasible on the UH-1 might well be used on another type of landing gear.

TABLE III
SUMMARY OF CROSS MEMBER CONFIGURATIONS INVESTIGATED

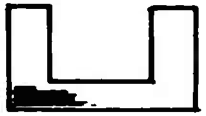
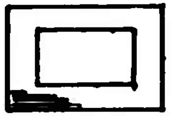
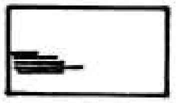
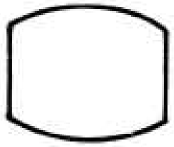
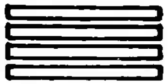
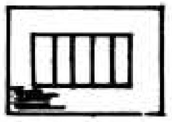
Cross Section	Description	Remarks
	Circular Solid	Stiffness cannot be controlled. The maximum radius permissible (R=c=FI/M) is not great enough to give required spring rate stiffness for elastic materials.
	Circular Tube	Same as above.
	Elliptical Solid	Stiffness can be controlled, but design is inefficient, heavy, and too wide to fit in gear well opening. Fabrication is difficult.
	I-Section	High flexural properties must be obtained with a low modulus material in order to make this a practical section; i. e., a large moment of inertia is required. Care must be given to flange buckling and web interlaminar shear. The smaller moment of inertia requirements for S-994 glass ($E = 8 \times 10^6$ psi) make the section impractical.
	Channel Section	Shear center location for fore and aft loads induces additional torsion which overstresses this type of design.
	Hollow Rectangle	Low flexural strength and high shear stresses at inside corners. Since height is controlled by $c = \frac{h}{2} = FI/M$ and I is required by spring rate, the width gets absurdly large.

TABLE III (CONT'D)
SUMMARY OF CROSS MEMBER CONFIGURATIONS INVESTIGATED

Cross Section	Description	Remarks
	Solid Rectangle	High flexural strength, easy-to-control stiffness and spring rate, simple fabrication, and weak in torsion.
	As Above With Curved Surfaces	Similar to solid rectangle, but more difficult to fabricate. Somewhat lighter than rectangular section. Aerodynamic drag reduction is negligible.
	Leaf Spring	Good for flat landings; very weak in torsion from fore and aft loads.
	Honeycomb Sandwich	Low flexural strength and high shear stresses. Shear stresses are higher than honeycomb allowable and, therefore, the core must be considered nonstructural.

AERODYNAMICS

The following analysis compares the drag contribution of the tubular landing gear with various shapes compatible with reinforced plastic construction.

Wind tunnel tests on the UH-1A, as reported in Reference 3, Bel Report No. 204-099-752, dated October 1956, show that the tubular landing gear drag coefficient varies from 0.00116 at negative angles of attack to 0.0014 at positive attitudes. This corresponds to an equivalent flat plate drag area, f, range of 1.77 to 2.14 square feet. Analytically, a drag estimate in good agreement with the tunnel data is obtained. Assume an average diameter of the tubular cross member of 2.75 inches. The equivalent flat plate drag area of the cross member and the drag for standard day conditions at 140 mph at sea level is estimated as follows.

Equivalent flat plate area = $f = C_D S$

where

C_D = drag coefficient

S = area (ft^2)

Total flat plate area is equal to the sum of that due to the cross members, skids, and interference drag. Therefore,

f due to cross members = $0.40 \times 1.9 = 0.760 \; ft^2$

where

$C_D = 0.40$ for $RN > 4 \times 10^5$

$S = 1.9 \; ft^2$ = exposed area of cross members

f due to skid = $0.200 \; ft^2$, Reference 8

f due to interference = $0.40 \times 2.29 = 0.916 \; ft^2$

where

$S = 2.29 \; ft^2$

Total flat plate area = $0.760 + 0.916 + 0.200 = 1.876 \; ft^2$

$$\text{Drag} = C_D Sq = 1481 \; C_D S M^2$$

$$= 1481 \times 1.876 \times (0.184)^2 = 95.6 \text{ pounds}$$

where

$C_D S$ = 1.876 ft^2

q = dynamic pressure

M = Mach number = 0.184 at 140 mph at sea level

Assuming the same conditions for the reinforced plastic configurations, the following relative flat plate drag areas result when using the drag data variation as found in Reference 6.

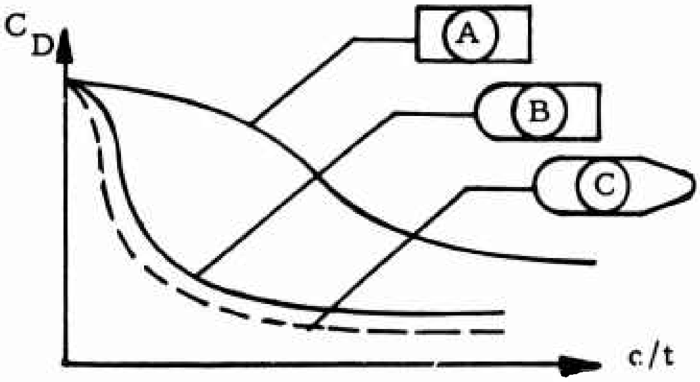


Figure 4. Comparison of Drag of Shapes with Parallel Sides.

Configuration A, Figure 4, represents the basic structure, which is rectangular in cross section with sharp corners having an average thickness of 1.80 inches, reference Figure 1.

This gives an exposed frontal area of 1.24 $feet^2$ for all the configurations, which is less than the present tubular gear which has 1.9 $feet^2$. The chord, c, of configuration A is 2.20 inches. The sharp-cornered leading edge, however, results in a drag coefficient of 2.0 for a c/t of 1.22, and a strut drag as follows.

f due to cross members = $C_D S$ = 2.0 (1.24) = 2.48 ft^2

f due to interference = 2.0 (1.415) = 2.83

f due to skid = $\underline{0.20}$

Total flat plate drag area = 5.51 ft^2

Thus, drag = 5.51 x 1481 x $(.184)^2$ = 281 pounds

Configuration B represents the most significant change in drag reduction through use of a nose fairing equal to one radius in length, making the average section thickness, t, equal to 1.80 inches and chord, c, equal to

3.1. This produces a drag coefficient of 0.70 for a c/t of 1.72. For this configuration,

f due to cross member = C_DS = .70(1.24) = 0.867 ft^2
f due to interference = .70(1.415) = 0.991
f due to skid = 0.200
Total flat plate drag area = 2.058 ft^2
Thus, drag = 2.058 x 1481 x $(.184)^2$ = 105 pounds

Configuration C further refines the drag by adding a trailing edge fairing to configuration B. This has the effect of reducing base drag by increasing the length, c. By adding a contoured trailing edge 1.5 inches in length and 1.80 inches in depth, a drag coefficient of 0.40 may be obtained for c/t = 2.55. This gives

f due to cross member = C_Ds = 0.40(1.24) = 0.496 ft^2
f due to interference = 0.40(1.415) = 0.565
f due to skid = 0.200
Total flat plate drag area = 1.261 ft^2
Thus, drag = 1.261 x 1481 x $(0.184)^2$ = 64.3 pounds

Hence, it is possible to achieve a cross member design through judicious fairing that produces considerably less drag than the aluminum tubular member.

Configuration B, which has only a semicircular leading edge added to the basic fiber glass gear, has drag characteristics very close to that of the tubular gear (105 versus 96 pounds). Therefore, it was recommended that only a leading edge fairing as in configuration B be used initially for flight test comparison of drag. A trailing edge could be added later, if necessary.

GROUND RESONANCE

A rotary-wing aircraft that employs a drag hinge in the rotor blades experiences two oscillating motions of the blade about the drag hinge - one of a higher frequency than the rpm of the rotor, and the other lower. In the fast oscillation, the blades remain at the same angular spacing from each other and cause periodic angular acceleration of the hub, but do not displace the hub laterally. The slow mode is the pendulus oscillation of one blade with respect to the others and, therefore, produces an unbalanced cyclic centrifugal force on the hub, which tries to displace the hub laterally. If the displacement of the hub is restrained, as when the aircraft is on the ground, the frequency of the slow oscillation will be increased and may coincide with the rotor speed while the rotor is being accelerated for takeoff or is decelerating after landing. This condition is known as ground resonance; and if the rotor speed is maintained at the resonant frequency,the hub with the rotor may fail structurally.

Although it is theoretically possible for ground resonance to occur as a result of blade lag bending in lieu of the presence of lag hinges, from a practical standpoint the necessary conditions for resonance do not occur. Therefore, since the UH-1 helicopter does not incorporate lag hinges in the rotor system, ground resonance is not considered to be a problem with this aircraft.

WEIGHT

A weight comparison of the fiber glass reinforced plastic landing gear and the present metal landing gear is shown in Table IV. An increase to aircraft weight of 29. 3 pounds is indicated.

Some weight savings can be realized with future material developments and a design that can be incorporated in the original design of the helicopter instead of adapting the landing gear design to existing structure. It should be noted that the reinforced plastic landing gear met military specifications for sustaining limit sink speed loads without yielding. The present metal landing gear does not meet this requirement. The reinforced plastic landing gear was subjected to load conditions more severe than its aluminum counterpart and retained its original shape after withstanding numerous limit and ultimate load tests. Therefore, the weight increase should be carefully evaluated.

TABLE IV

UH-1 LANDING GEAR WEIGHT COMPARISON

Item	Present Metal Landing Gear (lb)	Reinforced Plastic Landing Gear (lb)
Forward Cross Member	16. 7	28. 8
Aft Cross Member	26. 9	36. 1
Bearing Blocks	1. 5	9. 1
Retainer Straps	1. 8	2. 2
Skids	25. 6	25. 6
Saddle Fittings	14. 7	14. 7
Total Weight	87. 2	116. 5

NOTES:

1. The weight of the modification to the helicopter to accommodate the reinforced plastic landing gear was negligible.
2. The addition of nylon leading edges to the cross members will increase the weight 5. 5 pounds.

FABRICATION

The fiber glass reinforced plastic cross members were fabricated by Cincinnati Testing Laboratories (CTL) of the Missile/Space Technology Division of the Studebaker Corporation, Cincinnati, Ohio. A wet winding process was used in which a circular hoop having a width of 2.2 inches and an inner radius of 60.8 inches was fabricated, and the cross members and test speciment were cut from the hoop. Materials used were Owens-Corning S-994 HTS 901 twelve end glass roving and Union Carbide ER-2270 Bakelite epoxy resin with methylnadic anhydride.

The mandrel on which the hoop was wound was fabricated from aluminum alloy and had the appearance of a large spoked wheel with an open channel rim. The rim of the mandrel was partially detachable for removal of the hoop. Sixteen strands of twelve end roving were used. The roving was guided over rollers submerged in resin to provide thorough wetting, then through guides and tensioning devices adjusted individually to provide a constant tension of 4 pounds for each strand. Winding guides spread the 16 strands over the width of the part. Winding was accomplished in a horizontal plane at a speed of 175 inches per minute. The temperature of the part was maintained at 150° to 180°F during winding by radiant heaters located around the mandrel. Winding was continued until the thickness of the hoop was 1/8 to 1/4 inch greater than the thickness required for the part. After the winding was completed, the mandrel was allowed to rotate for approximately 24 hours with the radiant heaters in place to precure the glass resin system to a rigid state. The circular hoop was then removed from the mandrel and cut into oversized segments. These segments were then cured in an oven at 230 ± 10°F for 180-200 minutes and then at 320° ± 10°F for 18 to 24 hours. After this curing cycle, the segments were ground on the outside diameter surface to the finished thickness, the ends trimmed to size, and the bolt holes drilled.

A considerable amount of development work was accomplished by CTL prior to the final windings. Initially, Naval Ordnance Laboratory (NOL) Rings were fabricated to determine basic material properties and to establish cure cycles. Materials other than the above were investigated, but it was concluded that the Owens-Corning S-994 HTS901 glass and Union Carbide ER2270 resin and a wet winding process would give the most satisfactory results.

In accordance with design requirements as outlined in the section entitled Configuration Study, the objectives for material properties were

$F_b = 183,000$ psi

$F_s = 7,000$ psi

$E = 8 \times 10^6$ psi

The NOL ring specimens exhibited considerably higher strength and elastic moduli than the design requirements, but no information was available on expected properties of the large full-scale sections.

Some concern has been expressed about the possibility of higher void content in a wet winding process than would be obtained if preimpregnated materials were used. This was investigated and it was concluded that with close control in wet winding, the void content and mechanical properties of the fabricated parts would not differ from those fabricated with preimpregnated materials.

The first full-scale winding was scrapped because of small longitudinal cracks. A second winding was fabricated and mechanical properties were determined by tests of specimens 1 and 2, Table V, cut from the hoop. These were simple beam bending tests using specimens 32 to 40 inches long. This winding was rejected because of low mechanical properties.

The resin content of this first winding was 30 percent. The objective was 20 to 24 percent. In an effort to reduce the resin content and increase the mechanical properties, the tension of each strand in the winding was increased to 4 pounds per strand and two additional windings were fabricated. Test specimens 3 and 4 represent these windings. Shear strength was determined by NOL method using small rings, and the basic material strength was greater than 7,000 psi.

Although the properties of these members were slightly different from the design objective, it was recommended that they be used for the static and drop tests, since their use would result in only slightly different deflection characteristics than predicted. The intents and purposes of the tests would not be compromised. USAAVLABS accepted this recommendation, and the landing gear test program was initiated.

TABLE V
MECHANICAL PROPERTIES OF FABRICATED CROSS MEMBERS

Winding Number	Specimen Number	Resin Content (pct)	Failure Stress (psi)	Modulus of Elasticity (psi)	Remarks
1	-	-	-	-	Cracked - Rejected
2	1	30	181,200	6.22 X 10^6	Rejected
2	2	30	172,000	6.08 X 10^6	
3	3	24	191,000	6.79 X 10^6	Used for forward member
4	4	24	182,000	8.46 X 10^6	Used for aft member

TEST PROGRAM

DISCUSSION

Military specifications pertaining to static and drop tests of aircraft landing gears were written for application to conventional type landing gears. In some instances, the specifications appear quite vague and inadequate when applied to a skid type landing gear system. The test agenda briefly described here and in detail in Appendix III is considered to fulfill all contract requirements and is consistent with all intents and purposes of the several referenced specifications.

A complete UH-1 landing gear system as shown in Figure 1 was subjected to static and drop tests. Four static tests and twenty-two drop tests were planned using two landing gear assemblies. The rear reinforced plastic cross member failed while holding maximum load in Condition II and was replaced. The saddle fittings sustained damage in Condition IV and the saddle-skid assemblies were replaced. The rear cross member failed on the fifteenth drop test, maximum height drop for nose up attitude, and further testing was cancelled because of unavailability of additional components. Tests completed correspond to static test Conditions I through IV and drop test Conditions V and VI of the test agenda.

Static deflection tests were first performed to determine spring rates of the two cross members. These tests were repeated after the aft cross member was replaced. These spring rates were used to calculate loads for limit and ultimate sink speeds for maximum fore and aft center of gravity positions. These loads were then applied in the static tests.

The forward cross member was subjected to all tests, and the aft member was subjected to all but static Condition I. This is considered an extremely severe amount of testing on one landing gear, and it is noteworthy that the reinforced plastic members withstood this series of tests without failure or accumulative damage until failure in a very severe test.

Terminology used through the test program is defined as follows.

Sinking Speed (V)	- refers to the vertical component of velocity (fps) of the helicopter with respect to ground.

- Limit Sinking Speed = V_{limit} = 8.0 fps
- Ultimate Sinking Speed = 1.5 V_{limit} = 9.8 fps
- Reserve Energy Sinking Speed = ultimate sinking speed

Rotor Lift Factor (L) - the ratio of rotor lift to the design gross weight

Effective Drop Weight (W_e) - that weight of the drop test fixture that will give the same impact energy in free fall as the actual helicopter while undergoing rotor lift

$$W_e = W \frac{h+(1-L)d}{h+d}$$

where

W = 6,600 lb
L = 0.67
h = drop height
d = center of gravity displacement

Mass Moment of Inertia - a property of the drop test fixture in simulating the actual helicopter

- Pitching mass moment of inertia (I_{yy}) lb-in-sec ^{2} - the mass moment of inertia about the lateral axis of the helicopter
- Rolling mass moment of inertia (I_{xx}) lb-in-sec ^{2} - the mass moment of inertia about the longitudinal axis of the helicopter
- Yawing mass moment of inertia (I_{zz}) lb-in-sec ^{2} - the mass moment of inertia about the vertical axis of the helicopter

Load Factor - a factor used to combine inertia forces with gravity forces

Landing gear load factor ($N_{L.G.}$) - ratio of the maximum load on the landing gear to the weight of the upper mass

$$N_{L.G.} = \frac{W_e N_j}{W}$$

where N_j = drop test fixture load factor

W = weight of helicopter (6,600 lb)

W_e = effective drop weight

Helicopter load factor ($N_{C.G.}$) - upper mass or airplane load factor for equivalent airborne impact represented by reduced-mass drop test

$$N_{C.G.} = N_{L.G.} + L$$

STATIC TESTS

Static tests consisted of applying the computed vertical load resulting from a level landing sink speed of 8 feet per second in both fore and aft center of gravity conditions. Simultaneously, horizontal forces were applied in first inboard-outboard directions, and then in fore and aft directions. All loads were applied by hydraulic cylinders. Figures 5 and 6 show the static test setup for a vertical load acting in conjunction with an inboard side load. Figure 7 depicts the test of vertical load with a forward acting horizontal load. Figures 8,9 and 10 show the vertical load test with an aft-acting horizontal force.

DROP TESTS

The reduced mass method was used in performing all drop tests. This free-fall method consisted of raising the drop fixture assembly to the required height and attitude and, upon release, allowing the skids to impact the steel boiler plate drop surface.

In determining the required drop test fixture weight for the reduced mass method,

$$W_e = W \frac{h+(1-L)d}{h+d}$$

where

W = weight of helicopter

$h = V^2/2g$ = drop height

L = rotor lift factor

d = center of gravity displacement after impact

It is noted here that only the rotor lift factor, L, has direct effect on the ratio of W_e to W. For the drop tests reported herein, the desired value of the rotor lift factor was 0.67. The deflection characteristics of the landing gear must be estimated prior to calculating the effective fixture drop weight, W_e, for a particular drop condition. The magnitude of estimated deflection d, is based on the landing gear geometry and the material modulus, and represents the vertical travel of the helicopter center of gravity after ground contact. During the drop test, the actual value of d is measured; and if this value along with known values of W, W_e, and h are substituted into the formula for W_e, the rotor lift factor may be solved by the following equation:

$$L = 1 + \frac{h}{d} - \frac{W_e}{W} \left(1 + \frac{h}{d}\right)$$

By comparing the test rotor lift factor with the desired rotor lift factor of 0.67, the accuracy of the parameter W_e may be determined. If L = 0.67, the landing gear deflection and fixture effective weight was assumed correctly, and the proper landing load was applied during the drop test. If L is less than 0.67, the landing gear deflection was larger than calculated, and the effective weight dropped was larger than desired; i.e., the landing gear was subjected to a more severe condition than desired. If L is greater than 0.67, the converse is true. Figure 13 shows the drop test fixture and lead pigs placement for obtaining the desired effective weight and helicopter center of gravity and mass moments of inertia.

It was originally planned to perform four series of drop tests as follows, all in an extreme aft center of gravity condition.

- Condition V - 10 drops in a level landing with sink speeds from 2 to 9.8 feet per second (2 drops at 8 feet per second).
- Condition VI - 5 drops in a 12° nose-up condition with sink speeds of 4, 6, 8 (twice), and 9.8 feet per second.

Condition VII - 5 drops in a 14° rolled position with sink speeds of 4, 6, 8 (twice), and 9.8 feet per second.

Condition VIII - 2 drops in a level landing with sink speeds of 6 and 8 feet per second, with an overload condition.

Test Conditions V, VI, and VII were for a helicopter weight of 6,600 pounds, while test Condition VIII was for a helicopter weight of 7,590 pounds. The ten drops in Condition V and the first four tests in Condition VI were successfully completed. Failure occurred in the 9.8 feet per second drop of Condition VI, thereby halting the test program for lack of further cross members to test.

SUMMARY OF TEST RESULTS

The results of static and drop tests are given in the sequential order of the test program.

Static Test Condition I

This first static test consisted of applying a limit vertical load resulting from an 8-feet-per-second speed for a maximum aft center of gravity position with one-half this load applied to the skids in an inboard direction. No difficulty was experienced, and measured strains and deflections were relatively low. Table XIV in Appendix III gives the results of this test.

Static Test Condition II

The second static test was a repeat of the first test with the exception that the side load was applied in an outboard direction. Failure of the rear cross member outboard of the ball fitting at BL14L occurred just prior to the application of the maximum load. Failure was catastrophic in that there was a complete bending failure wherein approximately fifty percent of the fibers in the cross section failed in compression. Secondary failures consisted of severe longitudinal splitting throughout the length of the member. Table XV in Appendix III summarizes the results of this test. Figures 11 and 12 are pictures of the failed gear.

An investigation was made in an effort to determine cause of failure, but conclusive evidence for a specific cause was not found. It was believed, however, that a contributing factor was the presence of a relatively sharp edge on the ball fitting at the point of failure of the cross member. Using magnification, a slight cutting of the extreme fibers was detected at the location of the edge of the fitting at BL14R. The failure damage on the left side was too severe to determine if there was any initial cutting of fibers. The computed stress at failure was 160,000 psi, though the measured strain indicated 134,000 psi. A test specimen from the same hoop winding from which the rear cross member was cut had failed in a previous laboratory test at 182,000 psi. It was expected that some degree of stress concentration existed around the ball fitting during the static test that could have resulted in stresses higher than the value computed.

After reworking the ball fittings by machining larger radii at the points of metal-to-plastic contact, a new aft cross member was installed and static test Condition II repeated without incident. Table XVI in Appendix III gives the results of this test. It will be noted that the applied loads are somewhat different than in the previous tests. This was brought about by a re-evaluation of gear spring rates to take into account non-linearity which decreased the required vertical load. The side load was also reduced to agree with contract requirements. An excessive side load had been inadvertently applied in previous tests.

Static Test Condition III

This test employed the same vertical load as in the previous test, with the exception that the load was applied at a maximum forward center of gravity and with one-half the vertical load applied in a forward direction. The test was successfully completed without incident. Table XVII in Appendix III gives results of the test, and Figure 7 shows the landing gear under maximum load.

Static Test Condition IV

This test was similar to Condition III except that the vertical load was applied at the maximum aft center of gravity position and the horizontal load was applied in an aft direction. It was predetermined that this would be the most critical of the four static conditions; therefore, this test was performed last. The expected severity of this test was borne out by attachment failures of the saddle fittings joining the cross members to the skids. Attachment failures occurred while the gear assembly was sustaining 100-percent vertical load and 97.5-percent horizontal load. The gears continued to react these applied loads, but because of yielding in the saddle attachments, no additional horizontal load could be put into the system. The attachment failures consisted of a combination of rivet shear and hole elongation and tension tear-out at steel bolts attaching the gears to the skids. This test is shown in Figures 8, 9 and 10.

Test results are given in Table XVIII.

The saddle fittings used in the gear assembly were standard parts on the UH-1 helicopter gear system. Failure of these parts demonstrated the critical nature of this load condition. Each cross member was attached to the saddle fittings by two bolts that passed completely through the saddle fittings, shims, and cross member. The half-moon shaped shims were bonded to the cross member, permitting loads in the cross member to be sheared into the shims. Load was transferred from the shims to the saddle fittings by shear in the two bolts. Oversize holes in the cross member prevented load transfer directly from the cross member to the bolts. Upon disassembly of the gear system after static test Condition IV, surface hairline cracks 1/4 to 2 inches long were noted at the ends of each member in the vicinity of, and extending from, the bolt holes. The metal shims and saddle fittings had hidden these hairline cracks until disassembly. It was not known at what stage of static testing the cracks formed, but attachment failures in the saddle fittings very likely initiated some of the cracks.

It is noteworthy to mention that the static test conditions established the design criteria for the landing gear cross members, and zero margins of safety had been calculated for these loads, Reference 7. The same forward gear member was subjected to all four static tests, while the same aft member was subjected to the last three static tests.

Drop Tests

Following static tests, preparation was made for performing drop tests. The two skids and associated saddle fittings were replaced with new parts, and reassembly of the landing gear system was accomplished using the two cross members from the static test program. All drop tests were in a maximum aft center of gravity position. Figure 13 shows the fixtures used in these tests.

Documentation of drop tests consists of still and motion pictures and the time history oscillograph tracing for each drop. The oscillograph charts contain strain and deflection measurements of each cross member, and horizontal and vertical accelerations for computation of roll, yaw, and pitch rates. Movies were taken of all drops in the nose-up attitude, and of the higher drops in the level attitude. Table VI summarizes the results of all drop tests. Comparative values of aircraft load factors are given for the reinforced plastic landing gear and the present metal system where data are available.

Level Landing Test Condition V

This series of testing was performed with the test fixture dropping level onto the steel boiler plate surface from heights that gave impact sink speeds ranging from values of 2 to 9.8 feet per second. Ten drops were made in this attitude. The reduced mass method was used in simulating the UH-1 helicopter landing with a rotor lift of 0.67W. It will be noted in Table VI that some variation was made in the weight of the test fixture, W_e. This was done in order to obtain the required rotor lift. Previous spring rate characteristics were used to obtain an initial value of 4,430 pounds. After performing the first few drop tests, it was realized that this weight was excessive by solving for lift from the formula $L = 1 + \frac{h}{d} - \frac{We}{W} \ (1 + \frac{h}{d})$. A final fixture weight of 4,140 pounds was use used in the remainder of the drop tests.

Considerable distortion of the aluminum skids occurred in the area of the forward gear during the level landing drops at speeds greater than 8 feet per second. No damage nor permanent set was noted in the cross members, and it was decided to continue the series of drop tests. Figure 16 presents time histories of deflections and fixture load factors, Nj, for the level landing drop tests.

Nose-up Landing Condition VI

This series of tests consisted of raising the test fixture such that the helicopter center of gravity was the required height above the ground, tilting the fixture into a 12° nose-up attitude, and dropping it onto the steel boiler plate. Drops were successfully completed at 4-, 6-, and 8-feet-per-second impact velocity. Distortion continued to increase in the skids around the forward gear, but it was realized that since all tests were critical for the aft cross member, the distortion would not influence test results.

In the fifth nose-up drop test, V = 9.8 feet per second, complete failure occurred in the aft member at the same location as the previously described static test failure. Both failures were very similar in all respects: there was an apparent initial compression rupture of the upper surface, which then caused secondary failure throughout the member.

The maximum strain measured on the cross members was $18,200 \times 10^{-6}$ in/in as given in Table VI, measured in the direction of the fibers. It must be remembered that the member was subjected to shear, bending, and torsion, and the measured value can only give an indication of the

TABLE VI

SUMMARY OF DROP TEST RESULTS

Test Cond.	Impact Velocity V (fps)	Drop Height h (in)	Effective Weight W_e (lb)	Rotor Lift L	Maximum Deflection Fwd (in)	Maximum Deflection Aft (in)	Maximum Deflection C. G. (in)	Maximum Strain Fwd	Maximum Strain Aft	Fixture Load Factor n_j	Aircraft Load Factor (n) Hayes	Aircraft Load Factor (n) Bell
I	2.0	0.74	4430	0.355	8.5	9.0	8.9	6.500	8,400	1.65	1.47	2.00
I	4.0	2.98	4430	0.420	10.5	10.9	10.8	7,900	10,000	1.95	1.73	2.65
I	5.0	4.65			Malfunction in Recording Instruments							2.86
I	6.0	6.70	4430	0.472	13.5	15.3	15.2	9,500	11,900	3.15	2.59	3.53
I	7.0	9.12	4430	0.502	14.6	16.2	16.2	10,400	14,600	3.45	2.83	-
I	8.0	11.90	4110	0.673	14.5	14.9	14.8	10,100	14,300	3.95	3.13	3.38
I	8.0	11.90	4110	0.673	14.5	14.9	14.8	10,300	15,000	4.00	3.16	3.38
I	8.5	13.50	4140	0.700	14.7	15.1	14.9	10,600	15,100	4.05	3.24	-
I	9.0	15.10	4140	0.737	15.1	15.4	15.3	11,400	15,500	4.25	3.41	-
I	9.8	17.90	4140	0.769	16.3	16.7	16.6	12,800	-	4.40	3.53	3.79
II	4.0	2.98	4140	0.476	5.7	11.2	10.8	6,400	11,400	2.40	1.98	No
II	6.0	6.70	4140	0.567	7.7	13.3	12.8	8,300	14,100	2.65	2.23	Test
II	8.0	11.90	4140	0.637	9.8	17.1	16.5	10,500	17,500	3.50	2.84	
II	8.0	11.90	4140	0.635	9.9	17.2	16.7	11,100	18,200	3.50	2.84	
II	9.8	17.90	4140	-	7.6	15.7	-	7,400	16,100	3.20	-	↓

Test Condition I is for Level Landing, Test Condition II is for Nose-Up (12°) Landing

L = Calculated Rotor Life (Should Be 0.67)
values above 0.67 indicate W_e too small,
values below 0.67 indicate W_e too large

Fwd and Aft Refer to Fwd and Aft Cross Members

Strain is shown in in /in x 10^{-6}, E = 8.46 x 10^6 psi (Aft Cross Member) E = 6.79 x 10^6 psi (Fwd Cross Member)

n_j = Test Fixture Load Factor at C. G.

n = Aircraft Load Factor = $\frac{W_e}{W} n_j + L$ (W = 6600 lb)

Data for Condition II at 9.8 fps are at Failure stress in the anisotropic material of the cross member. Figure 17 shows time histories of deflections and fixture load factors for the nose-up drop tests.

During the drop tests, the reinforced fiber glass members performed quite satisfactorily with no sign of damage until sudden failure occurred. The nose-up drops were particularly critical for the aft gear, since in the 12° nose-up attitude, a side view would reveal the center of gravity force vector passing directly through the aft member saddle fitting. This caused a very high load to be induced on the aft member that caused bending, shear, and torsion in the member. At each initial impact in this attitude, the forward member was still 17 inches above the impact surface. Because of the large mass moments of inertia of the fixture and the small moment arm between the fixture center of gravity and point of impact, the aft member reacted the entire landing load. On rebound, the fixture would slowly pitch over and permit the forward member to begin taking load. It should be noted that on the UH-1 helicopter, a tail skid exists that would have prevented this condition from actually occurring.

SIGNIFICANT FINDINGS

The series of drop tests revealed several important findings. One of these was an answer to the question of springback and the time duration of energy dissipation. Springback was not as severe as anticipated, though at higher drops in the level landing attitude, 6.5 feet per second and above, the fixture was noted to completely clear the drop surface momentarily. However, motion subsided very rapidly; e.g., at sink speeds of 4 feet per second, negligible motion existed after 1-1/2 seconds.

Another significant finding was that aircraft load factors were always lower than those using the aluminum landing system. Table VI gives a comparison of these values for level drops. The manufacturer of the aluminum gear system, Bell Helicopter Company, accomplished tests only for conditions in which the two skids make simultaneous flat contact. The nose-up attitude is a more severe test condition, but no comparison of load factors can be made.

The durability and high strength of the fiber glass reinforced plastic struts were well demonstrated by the fact that the same forward cross member was used in all static and drop tests, while the same cross member was used in all tests except the first static test. Flexural strength and elastic moduli of such thick cross sections were not known until it was shown during the test program that flexural strengths as high as 190,000 psi and elastic moduli of 8.5×10^6 could be obtained.

The successful static and drop tests of the UH-1 landing gear system proved that glass fiber reinforced plastics are feasible materials for use in landing gears. It was shown that the gear system could withstand high landing loads from sink speeds of 9.8 feet per second without failing or yielding.

Although the reinforced plastic landing gear is somewhat heavier than the present UH-1 metal landing gear, it is considered competitive in weight. The metal gear would weigh considerably more than its present weight if it met the same design criteria or was designed as an "elastic" member rather than a yielding member.

CONCLUSIONS

The primary objective of this program was to experimentally determine the feasibility of using reinforced plastic materials for skid type helicopter landing gears. This objective was fulfilled, and it is concluded that fiber glass reinforced plastics are feasible for this application. Although the test program ended in a failure of one of the reinforced plastic members, this failure does not in any way negate the conclusion of feasibility. It does, however, show the importance of detail design of attachments and load input points of components fabricated from these materials.

The program has resulted in considerable knowledge and experience in the design fabrication, mechanical properties, and characteristics of thick high strength sections of glass fiber reinforced plastics. After careful review and appraisal of all the aspects, the following conclusions are made.

- The static and drop tests of the UH-1 fiber glass reinforced plastic landing gear system prove that these materials are feasible materials for use in landing gears.
- The use of reinforced plastics for landing gear shock components can reduce the aircraft load factors in all rates of descent.
- The reinforced plastic gear will withstand very hard landings without yielding or failing. The test landing gear withstood loads from impact velocities up to and including 9.8 feet per second. The present metal UH-1 landing gear starts yielding at an impact velocity of 5 feet per second.
- Static and drop tests demonstrated a high degree of ruggedness and durability. A single forward cross member was subjected to 4 ultimate design static tests and 15 drop tests without failing. A single aft cross member was subjected to all but 1 of these tests, with failure occurring in the last drop test.
- High-strength, high-quality, thick flexural sections can be successfully and economically fabricated.
- Bending strength as high as 190,500 psi and flexural moduli of 8×10^6 psi were obtained in thick members. Mechanical properties

of thick sections were not previously known.

- The high bending properties in the material were accompanied by rather high shear stress, although shear strength is still a problem in these materials.

- Energy dissipation occurs quite rapidly in the reinforced plastic landing gear, and springback was not as severe as anticipated.

- The hoop winding process presents an economical method of fabrication, and costs can be competitive with the metal gear.

- Weight of the reinforced plastic landing gear is somewhat heavier than the aluminum gear on the UH-1, but its performance requirements are greater than those for the metal gear. A metal gear designed to the same requirements would weigh considerably more.

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APPENDIX I - LOAD ANALYSIS

The helicopter landing gear is dynamically loaded at impact; hence, a proper load analysis must include an analytical study of the response to impulsive loading. Under impulsive loading, the maximum response is dependent on the relationship between the natural frequency of the structure and the duration and shape of the impulse function.

Military design specifications permit load factor determination for the landing gear through use of the formula

$$N = \frac{1}{3} + \left(\frac{1}{9} + \frac{2h}{\delta_s}\right)^{\frac{1}{2}}$$

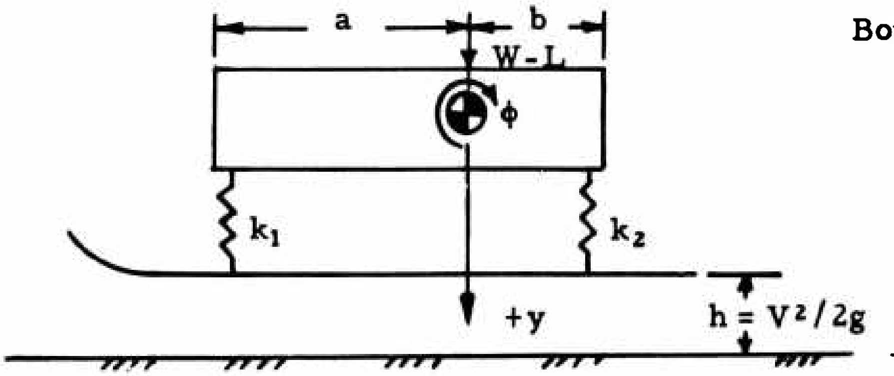
where

h = drop height
δ_s = static deflection

which is based on a rotor lift at impact of two-thirds the weight of the helicopter. The formula is derived from solving the energy equation at impact - assuming vertical translatory motion only. The load factor formula is particularly useful when applied at the aircraft center of gravity. For points not at the center of gravity(e. g. , the landing gear struts), effects of yaw, pitch, and horizontal motion contribute to the magnitude of load factor. Therefore, it is necessary to solve equations of motion for each landing condition to arrive at proper load factors for analysis of the landing gear.

METHOD OF ANALYSIS

The skid gear system of the UH-1 helicopter is analogous to a series of mechanical springs whose magnitude of deflection in landing determines load factors. The magnitude of deflection is governed by the gear location, gear spring rate, aircraft weight and moments of inertia, and sinking speed. This is demonstrated by the following sketch.



Boundary Conditions

@t = 0
$y = 0$
$\phi = 0$
$\dot{y} = V$
$\dot{\phi} = 0$

For the two-degree-of-freedom system shown, the equations of motion are

$$\begin{aligned} m\ddot{y} + (k_1 + k_2)y + (k_2 b - k_1 a)\phi &= W-L \\ I\ddot{\phi} + (k_2 b - k_1 a)y + (k_1 a^2 + k_2 b^2)\phi &= 0 \end{aligned} \qquad (1)$$

where

m = mass
I = moment of inertia
k = spring constant
W-L = weight minus lift
a & b = distance of spring from c. g.
y = vertical displacement
ϕ = rotational displacement

The nonhomogeneous equations of (1) may be solved in two parts: a solution of the homogeneous system and then a particular solution. Solving first the homogeneous system, assume solutions of the form

$$y = A\cos(\omega t + \phi) \text{ and } \phi = B\cos(\omega t + \phi) \qquad (2)$$

Substituting (2) into the homogeneous part of (1),

$$\begin{aligned} (-m\omega^2 + k_1 + k_2)A + (bk_2 - ak_2)B &= 0 \\ (bk_2 - ak_1)A + (-I\omega^2 + a^2 k_1 + b^2 k_2)B &= 0 \end{aligned} \qquad (3)$$

For (3) to have solutions other than the trivial solution, i.e., A=B=0, the determinant of the coefficients must equal zero.

$$\begin{vmatrix} -m\omega^2 + k_1 + k_2 & bk_2 - ak_2 \\ bk_2 - ak_1 & -I\omega^2 + a^2 k_1 + b^2 k_2 \end{vmatrix} = 0 \qquad (4)$$

Solving (4) yields two real roots of the natural frequency, ω_1 and ω_2. These roots are then substituted into (2) to obtain

$$\begin{aligned} y &= A_1\cos(\omega_1 t + \phi_1) + A_2\cos(\omega_2 t + \phi_2) \\ \phi &= B_1\cos(\omega_1 t + \phi_1) + B_2\cos(\omega_2 t + \phi_2) \end{aligned} \qquad (5)$$

From (3), we have $B = \dfrac{-(-m\omega^2 + k_1 + k_2)}{bk_2 - ak_1} A = \mu A$ (6)

Substituting (6) into (5),

$$y = A_1 \cos(\omega_1 t + \phi_1) + A_2 \cos(\omega_2 t + \phi_2)$$

$$\phi = \mu_1 A_1 \cos(\omega_1 t + \phi_1) + \mu_2 A_2 \cos(\omega_2 t + \phi_2) \qquad (7)$$

where

$$\mu_1 = \frac{-(-m\omega_1^2 + k_1 + k_2)}{bk_2 - ak_1} \qquad (8)$$

and

$$\mu_2 = \frac{-(-m\omega_2^2 + k_1 + k_2)}{bk_2 - ak_1}$$

Now obtaining particular solutions to (1) yields the final general solutions

$$y = A_1 \cos(\omega_1 t + \phi_1) + A_2 \cos(\omega_2 t + \phi_2) + C$$

$$\phi = \mu_1 A_1 \cos(\omega_1 t + \phi_1) + \mu_2 A_2 \cos(\omega_2 t + \phi_2) + D \qquad (9)$$

where C and D are the particular solutions,

$$C = \frac{(k_1 a^2 + k_2 b^2)(W-L)}{k_1 b_2 \ (a+b)^2} \quad \text{and} \quad D = \frac{-(k_2 b - k_1 a)(W-L)}{k_1 k_2 \ (a+b)^2} \qquad (10)$$

Boundary conditions can be applied to (9) for determining values of the four constants A_1, A_2, ϕ_1, and ϕ_2.

Solution of the above equations yields a time history of cross member deflection during the landing impact from which energy absorption, load distribution to the cross members, and load factors are determined.

STRUT DESIGN LOADS

A computer program was written in FORTRAN for use on an IBM 1620 computer in solving the equations given in the preceding section. From this program, load-and-attitude time history was readily obtainable for symmetrical landings.

Two landing conditions at basic design gross weight, 6600 pounds, were found to be critical in determining maximum vertical loads on the cross members: reserve energy with extreme aft center of gravity, and reserve energy with extreme forward center of gravity. The sinking speed for these conditions was 9.84 feet per second, or 1.5 V_{limit}, where V_{limit} equals 8 feet per second. These are conditions

A and B in Table VII. The center of gravity limits of the helicopter were taken from Reference 12 to be Fuselage Station (F. S.) 125 to F. S. 138. Loads and attitudes for these two conditions are given in Tables VIII and IX.

For static load conditions, the above center of gravity limits and weights were used in the computed program to determine vertical loads for a sinking speed of 8 feet per second, conditions C and D. The maximum vertical load was applied in conjunction with one-half the maximum vertical load applied alternately in an inboard, outboard, forward and aft direction on the gear system. See Appendix II, Stress Analysis. Tables X and XI give load results of these two conditions.

A fifth load condition, E, with loads as given in Table XII is for a landing at limit sink speed in an overload weight condition of 7590 pounds. This condition is required by Reference 1 to be demonstrated by static test. It is considered not critical for the landing gears.

TABLE VII

DESIGN CONDITIONS FOR LOAD ANALYSIS - LEVEL LANDING

Design Condition	Weight (lb)	Center of Gravity	Sinking Speed (fps)	Test Condition
A	6600	aft	9.84	V Test 10
B	6600	fwd	9.84	-
C	6600	aft	8.00	I through V
D	6600	fwd	8.00	I through V
E	7590	aft	8.00	VIII Test 2

An additional computer program for a flat, level landing was set up for parametric studies and optimization curves in selecting the best ratio of spring constants between the forward and aft gears in minimizing load factors on the helicopter. Results of this program were then used in determining if a given set of spring rates was satisfactory for the various other landing conditions, static load conditions, and physical limitations and requirements.

TABLE VIII

LANDING GEAR LOADS, LEVEL DESCENTS, AFT CENTER OF GRAVITY, DESIGN GROSS WEIGHT, ULTIMATE SINK SPEED, CONDITION A

A= 57.5 W= 2200.000 W1= 3.126
B= 13.5 M= 17.098 W2= 7.818
I= 110360.0 V= 117.600 K1= 308.000
K2= 726.0

TIME	C.G. DEFL.	PITCH ANGLE	FWD GEAR LOAD	AFT GEAR LOAD
0.020	0.2368211E+01	0.1128300E-04	0.7292093E+03	0.1719432E+04
0.040	0.4730618E+01	0.9036900E-04	0.1455430E+04	0.3435315E+04
0.060	0.7030205E+01	0.3044870E-03	0.2159911E+04	0.5106913E+04
0.080	0.9211499E+01	0.7185100E-03	0.2824417E+04	0.6694591E+04
0.100	0.1122192E+02	0.1393103E-02	0.3431678E+04	0.8160764E+04
0.120	0.1301305E+02	0.2382932E-02	0.3965817E+04	0.9470828E+04
0.140	0.1454184E+02	0.3735101E-02	0.4412738E+04	0.1059398E+05
0.160	0.1577163E+02	0.5487697E-02	0.4760476E+04	0.1150399E+05
0.180	0.1667306E+02	0.7668586E-02	0.4999493E+04	0.1217980E+05
0.200	0.1722477E+02	0.1029445E-01	0.5122916E+04	0.1260608E+05
0.220	0.1741393E+02	0.1337012E-01	0.5126705E+04	0.1277355E+05
0.240	0.1723653E+02	0.1688819E-01	0.5009761E+04	0.1267924E+05
0.260	0.1669751E+02	0.2082891E-01	0.4773952E+04	0.1232653E+05
0.280	0.1581060E+02	0.2516046E-01	0.4424074E+04	0.1172510E+05
0.300	0.1459803E+02	0.2983940E-01	0.3967737E+04	0.1089062E+05
0.320	0.1308992E+02	0.3481155E-01	0.3415182E+04	0.9844468E+04
0.340	0.1132360E+02	0.4001306E-01	0.2779036E+04	0.8613099E+04
0.360	0.9342664E+01	0.4537173E-01	0.2074007E+04	0.7227463E+04

TABLE IX

LANDING GEAR LOADS, LEVEL DESCENT, FORWARD CENTER OF GRAVITY, DESIGN GROSS WEIGHT, ULTIMATE SINK SPEED, CONDITION B

A= 44.5 W= 2200.000 W1= 3.172
B= 26.5 M= 17.098 W2= 7.798
I= 107790.0 V= 117.600 K1= 308.000
K2= 726.0

TIME	C.G. DEFL.	PITCH ANGLE	FWD GEAR LOAD	AFT GEAR LOAD
0.020	0.2368211E+01	-0.8081000E-05	0.7295196E+03	0.1719165E+04
0.040	0.4730616E+01	-0.6472900E-04	0.1457917E+04	0.3433182E+04
0.060	0.7030193E+01	-0.2180960E-03	0.2168289E+04	0.5099724E+04
0.080	0.9211446E+01	-0.5146530E-03	0.2844179E+04	0.6677608E+04
0.100	0.1122176E+02	-0.9978440E-03	0.3469977E+04	0.8127797E+04
0.120	0.1301265E+02	-0.1706845E-02	0.4031291E+04	0.9414349E+04
0.140	0.1454100E+02	-0.2675397E-02	0.4515297E+04	0.1050529E+05
0.160	0.1577002E+02	-0.3930777E-02	0.4911041E+04	0.1137341E+05
0.180	0.1667021E+02	-0.5492955E-02	0.5209710E+04	0.1199689E+05
0.200	0.1722003E+02	-0.7373877E-02	0.5404836E+04	0.1235988E+05
0.220	0.1740646E+02	-0.9576993E-02	0.5492453E+04	0.1245284E+05
0.240	0.1722529E+02	-0.1209697E-01	0.5471189E+04	0.1227283E+05
0.260	0.1668120E+02	-0.1491962E-01	0.5342299E+04	0.1182351E+05
0.280	0.1578771E+02	-0.1802210E-01	0.5109625E+04	0.1111515E+05
0.300	0.1456677E+02	-0.2137320E-01	0.4779506E+04	0.1016428E+05
0.320	0.1304831E+02	-0.2493399E-01	0.4360623E+04	0.8993365E+04
0.340	0.1126942E+02	-0.2865859E-01	0.3863777E+04	0.7630239E+04
0.360	0.9273549E+01	-0.3249508E-01	0.3301631E+04	0.6107424E+04

TABLE X

LANDING GEAR LOADS, LEVEL DESCENT, AFT CENTER OF GRAVITY, DESIGN GROSS WEIGHT, LIMIT SINK SPEED, CONDITION C

A=	57.5	W=	2200.000	W1=	3.126
B=	13.5	M=	17.098	W2=	7.818
I=	110360.0	V=	96.000	K1=	308.000
K2=	726.0				

TIME	C.G. DEFL.	PITCH ANGLE	FWD GEAR LOAD	AFT GEAR LOAD
0.020	0.1937950E+01	0.9220000E-05	0.5967254E+03	0.1407042E+04
0.040	0.3880483E+01	0.7394900E-04	0.1193879E+04	0.2817956E+04
0.060	0.5780716E+01	0.2494670E-03	0.1776043E+04	0.4199245E+04
0.080	0.7592810E+01	0.5893850E-03	0.2328147E+04	0.5518156E+04
0.100	0.9273079E+01	0.1144118E-02	0.2835846E+04	0.6743469E+04
0.120	0.1078106E+02	0.1959394E-02	0.3285867E+04	0.7846256E+04
0.140	0.1208051E+02	0.3074943E-02	0.3666341E+04	0.8800589E+04
0.160	0.1314026E+02	0.4523282E-02	0.3967092E+04	0.9584159E+04
0.180	0.1393499E+02	0.6328680E-02	0.4179895E+04	0.1017883E+05
0.200	0.1444585E+02	0.8506335E-02	0.4298674E+04	0.1057106E+05
0.220	0.1466091E+02	0.1106178E-01	0.4319656E+04	0.1075224E+05
0.240	0.1457545E+02	0.1399051E-01	0.4241466E+04	0.1071890E+05
0.260	0.1419207E+02	0.1727788E-01	0.4065168E+04	0.1047279E+05
0.280	0.1352064E+02	0.2089922E-01	0.3794233E+04	0.1002082E+05
0.300	0.1257804E+02	0.2482022E-01	0.3434469E+04	0.9374916E+04
0.320	0.1138773E+02	0.2899760E-01	0.2993874E+04	0.8551699E+04
0.340	0.9979234E+01	0.3337988E-01	0.2482447E+04	0.7572081E+04
0.360	0.8387357E+01	0.3790853E-01	0.1911946E+04	0.6460763E+04

TABLE XI

LANDING GEAR LOADS, LEVEL DESCENT, FORWARD CENTER OF GRAVITY, DESIGN GROSS WEIGHT, LIMIT SINK SPEED, CONDITION D

A= 44.5 W= 2200.000 W1= 3.172
B= 26.5 M= 17.098 W2= 7.798
I= 107790.0 V= 96.000 K1= 308.000
K2= 726.0

TIME	C.G. DEFL.	PITCH ANGLE	FWD GEAR LOAD	AFT GEAR LOAD
0.020	0.1937950E+01	-0.6601000E-05	0.5969792E+03	0.1406825E+04
0.040	0.3880482E+01	-0.5296500E-04	0.1195914E+04	0.2816211E+04
0.060	0.5780707E+01	-0.1786840E-03	0.1782907E+04	0.4193356E+04
0.080	0.7592766E+01	-0.4221610E-03	0.2344358E+04	0.5504226E+04
0.100	0.9272947E+01	-0.8195010E-03	0.2867300E+04	0.6716393E+04
0.120	0.1078074E+02	-0.1403472E-02	0.3339704E+04	0.7799817E+04
0.140	0.1207982E+02	-0.2202534E-02	0.3750773E+04	0.8727576E+04
0.160	0.1313893E+02	-0.3239976E-02	0.4091197E+04	0.9476530E+04
0.180	0.1393264E+02	-0.4533189E-02	0.4353384E+04	0.1002788E+05
0.200	0.1444194E+02	-0.6093055E-02	0.4531630E+04	0.1036763E+05
0.220	0.1465476E+02	-0.7923529E-02	0.4622265E+04	0.1048691E+05
0.240	0.1456618E+02	-0.1002136E-01	0.4623736E+04	0.1038225E+05
0.260	0.1417862E+02	-0.1237604E-01	0.4536640E+04	0.1005558E+05
0.280	0.1350173E+02	-0.1496983E-01	0.4363710E+04	0.9514254E+04
0.300	0.1255220E+02	-0.1777810E-01	0.4109744E+04	0.8770863E+04
0.320	0.1135330E+02	-0.2076972E-01	0.3781487E+04	0.7842910E+04
0.340	0.9934373E+01	-0.2390773E-01	0.3387466E+04	0.6752394E+04
0.360	0.8330068E+01	-0.2715002E-01	0.2937779E+04	0.5525290E+04

TABLE XII

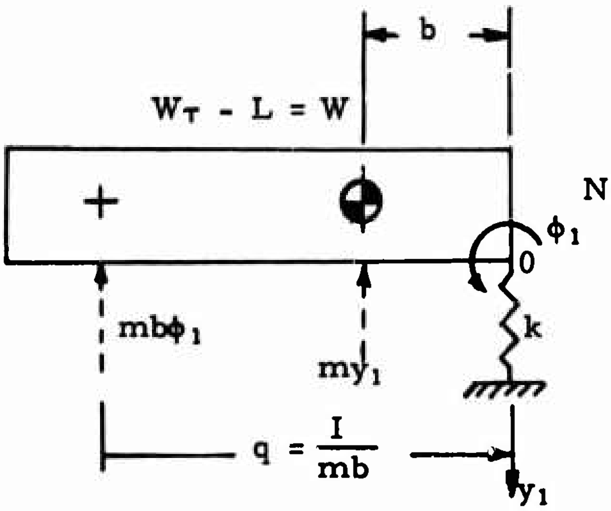
LANDING GEAR LOADS, LEVEL DESCENT, AFT CENTER OF GRAVITY, OVERLOAD GROSS WEIGHT, LIMIT SINK SPEED, CONDITION E

```
        A=         57.5      W=    2530.000      W1=        3.063
        B=         13.5      M=      19.663      W2=        7.296
        I=     114770.0      V=      96.000      K1=      308.000
        K2=       726.0

 TIME     C.G. DEFL.        PITCH ANGLE      FWD GEAR LOAD     AFT GEAR LOAD

0
0.020   0.1938964E+01     0.8869000E-05     0.5970439E+03     0.1407775E+04
0.040   0.3888595E+01     0.7115800E-04     0.1196427E+04     0.2823817E+04
0.060   0.5807964E+01     0.2402400E-03     0.1784598E+04     0.4218937E+04
0.080   0.7656799E+01     0.5682360E-03     0.2348231E+04     0.5564405E+04
0.100   0.9396328E+01     0.1104699E-02     0.2874505E+04     0.6832562E+04
0.120   0.1099012E+02     0.1895311E-02     0.3351390E+04     0.7997400E+04
0.140   0.1240483E+02     0.2980757E-02     0.3767897E+04     0.9035117E+04
0.160   0.1361092E+02     0.4395590E-02     0.4114318E+04     0.9924610E+04
0.180   0.1458331E+02     0.6167325E-02     0.4382436E+04     0.1064793E+05
0.200   0.1530186E+02     0.8315600E-02     0.4565702E+04     0.1119065E+05
0.220   0.1575182E+02     0.1085158E-01     0.4659379E+04     0.1154218E+05
0.240   0.1592416E+02     0.1377746E-01     0.4660642E+04     0.1169597E+05
0.260   0.1581572E+02     0.1708626E-01     0.4568644E+04     0.1164967E+05
0.280   0.1542932E+02     0.2076174E-01     0.4384540E+04     0.1140517E+05
0.300   0.1477365E+02     0.2477853E-01     0.4111458E+04     0.1096853E+05
0.320   0.1386314E+02     0.2910249E-01     0.3754442E+04     0.1034987E+05
0.340   0.1271759E+02     0.3369126E-01     0.3320345E+04     0.9563176E+04
0.360   0.1136179E+02     0.3849491E-01     0.2817686E+04     0.8625947E+04
```

SKID DESIGN LOADS

A drop loading condition which is critical for the existing aluminum skids is given in the Test Agenda, Appendix III. This condition allows the helicopter to make a nose-up landing at 12° angle, making initial ground contact on the aft ends of the skids. The following analysis determines the landing force from this condition for comparison of the strength of the skid with its induced bending moment.

The helicopter is assumed to rotate about the aft end of the skid, or point "O" in the sketch below. The spring constant, k, is considered to be the spring rate of the aft gear of 770 lb/in.



Note: ↑ denotes D'Alembert forces

For equilibrium,

$$mb\ddot{\phi}_1 + m\ddot{y}_1 + ky_1 = W \tag{11}$$

$$I\ddot{\phi}_1 + mb\ddot{y}_1 - Wb = O \tag{12}$$

From (12), $\ddot{\phi}_1 = \frac{Wb}{I} - \frac{bm}{I} \ddot{y}_1$

Substituting into (11) and simplifying,

$$\ddot{y}_1 + \left(\frac{kI}{Im-m^2 b^2} \right) y_1 = \frac{WI - mWb^2}{Im - m^2 b^2} \tag{13}$$

The solution to the reduced equation from (13) is

$$y_c = A \sin \omega t + B \cos \omega t$$

where

$$\omega = \sqrt{\frac{kI}{Im-m^2 b^2}}$$ A and B are integration constants.

The particular solution of (13) is

$$y_p = \frac{WI- mWb^2}{kI}$$

The complete solution of (13) is

$$y_1 = A \sin \omega t + B \cos \omega t + \frac{WI - mWb^2}{kI} \qquad (14)$$

The derivatives of (14) are

$$\dot{y}_1 = \omega A \cos \omega t - \omega B \sin \omega t \qquad (15)$$

$$y_1 = - \omega^2 A \sin \omega t - \omega^2 B \cos \omega t \qquad (16)$$

The boundary conditions are at $t = 0$, $y_1 = 0$, $\dot{y}_1 = V$.

From (14) $$0 = B + \frac{WI - mWb^2}{kI} \quad \therefore \quad B = - \frac{WI - mWb^2}{kI}$$

From (15) $$V = \omega A \quad \therefore \quad A = V/\omega$$

The final solution becomes

$$y_1 = \frac{V}{\omega} \sin \omega t + \frac{mWb^2 - WI}{kI} \cos \omega t + \frac{WI-mWb^2}{kI} \qquad (17)$$

For determining y_{max},

$$\dot{y}_1 = V \cos \omega t + \frac{\omega}{kI} (WI-mWb^2) \sin \omega t = 0 \qquad (18)$$

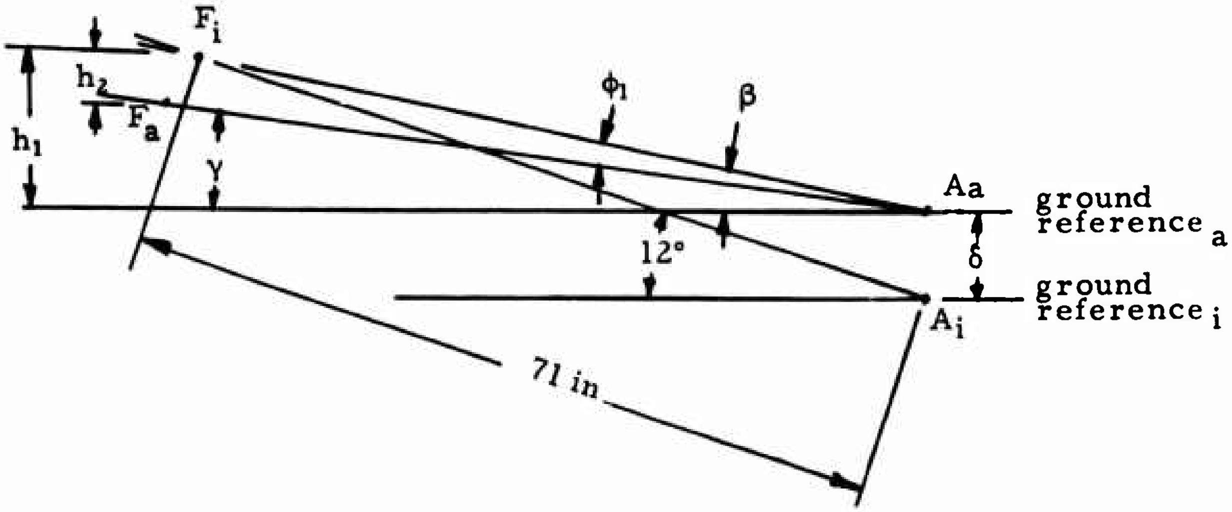
For the reserve energy drop condition with aft c. g.,

$$I = I_{c. g.} + mb^2 = 110,360 + 17.1 (13.5)^2 = 113,500$$

$$b = 13.5, \quad m = 17.1, \quad W = 2200, \quad V = 118$$

Substituting these values into (18), a value of $y_1 = 19.433$ inches is obtained. This indicates that the forward gear will be deflected before the aft gear can deflect through 19.433 inches. Thus, it is necessary to determine at what point the forward gear makes ground contact.

Geometry representing relative positions of the forward and aft cross members is shown in the sketch below.



where

Distance between gears = 71 inches
δ = deflection of aft gear after impact
h_1 = height of forward gear from ground when aft gear makes initial impact
h_2 = height of forward gear from ground after impact
A = aft gear
F = forward gear
Subscripts i and a refer to initial impact and after impact
ϕ_1 = rotation
γ = angle between forward gear and ground after impact

From the above sketch,

$$h_1 = 14.78 \text{ inches}$$

$$h_2 = h_1 - \delta = 14.78 - \delta$$

$$\gamma = \beta - \phi_1 = \sin^{-1}\frac{(14.78-\delta)}{71} - \phi_1$$

When $\gamma = 0$, the forward gear will contact the ground, as follows: and

$$0 = \sin^{-1}\frac{(14.78-\delta)}{71} - \phi_1 \;\; \underline{\text{or,}} \;\; \phi_1 = \sin^{-1}\frac{(14.78-\delta)}{71}.$$

The value "δ" is "y_1", $\therefore$ $\phi_1 = \sin^{-1}\dfrac{(14.78-y_1)}{71}$ (19)

From (12), $\ddot{\phi}_1 = \dfrac{Wb}{I} \quad \dfrac{bm}{I} \quad \ddot{y}_1$

Substituting (16) into (12),

$$\ddot{\phi}_1 = \frac{Wb}{I} + \frac{bm\omega^2}{I}(A \sin \omega t + B \cos \omega t) \tag{20}$$

Integrating twice,

$$\dot{\phi}_1 = \frac{Wbt}{I} - \frac{bm\omega}{I} A \cos \omega t + \frac{bm\omega}{I} B \sin \omega t + C_1 \tag{21}$$

$$\phi_1 = \frac{Wbt^2}{2I} - \frac{bm}{I} A \sin \omega t - \frac{bm}{I} B \cos \omega t + C_1 t + C_2 \tag{22}$$

Applying boundary conditions @ $t = 0$, $\phi_1 = \dot{\phi}_1 = 0$ (reference plane is 12° from horizontal),

$$C_1 = \frac{bmV}{I}, \; C_2 = \frac{bm}{I}\left(\frac{mWb^2 - WI}{kI}\right)$$

Equation (22) now becomes

$$\phi_1 = \frac{Wbt^2}{2I} - \frac{bmV}{I\omega} \sin \omega t - \frac{bm}{I}\left(\frac{mWb^2 - WI}{kI}\right) \cos \omega t$$

$$+ \frac{bmVt}{I} + \frac{bm}{I}\left(\frac{mWb^2 - WI}{kI}\right) \tag{23}$$

It is now necessary to substitute values of t into (17) and (23) until the following relation is satisfied: see (19)

$$\phi_1 = \sin^{-1} \frac{(14.78-y_1)}{71}$$

By trial and error, (19) is found to be satisfied at t = 0.13 sec. Thus, at t = 0.13 sec, the forward gear will make ground contact and begin absorbing landing energy. Substituting into (17), y_1 = 14.42 inches. For k = 770, the force applied on the skid is

$$P = ky_1 = 770\,(14.42) = 11,100 \text{ lb}$$

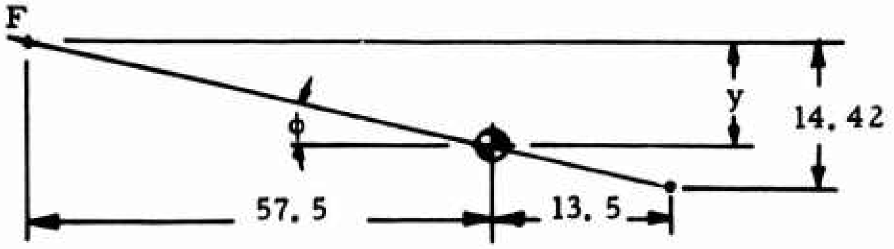
DYNAMIC SPRINGBACK LOADS

Dynamic springback loads may be defined as loads induced on the landing gear or aircraft as a result of "springing back" action of the system subsequent to initial impact and initial landing loads. The amount of springback is decreased by energy dissipated in the landing. It is therefore more pronounced in the case of an elastic landing gear system where all landing energy is returned to the system after first being absorbed by the landing gear.

Failure has been known to occur in the forward landing gear of a skid type arrangement even though initial impact was on the aft gear, Reference 17. This was caused by the added rotational velocity after impact acting with very nearly the initial vertical velocity and causing overloading of the forward gear.

The following is a continuation of the skid design loads analysis, whereby the analysis is extended into the time region after the forward strut makes contact. The effect this has in increasing or decreasing loads on the two struts is shown.

In order to continue the solution of this problem subsequent to the forward strut making contact, new geometry is obtained as follows, assuming rotation about the helicopter center of gravity:



$$y + 13.5 \tan\phi = 14.42$$

$$57.5 \tan\phi = y$$

$$\therefore 57.5 \tan\phi + 13.5 \tan\phi = 14.42$$

$$71 \tan\phi = 14.42$$

$$\phi = \tan^{-1} 0.203 = 11° 35' = 11.583°$$

$$y = 57.5(0.203) = 11.8 \text{ inches}$$

∴ For a new time reference of $t = 0$, $y = 11.8$ inches and $\phi = 11.583°$. Values of $\dot{y}$ and $\dot{\phi}$ must be determined from the previous references, (call previous references y_1 and ϕ_1) for $t = 0.13$ second.

$$y_1 = V \cos\omega t + \frac{\omega}{kI} (WI - mWb^2) \sin\omega t$$

$$\omega t = (6.82)(0.13) = 0.886 = 50.8°$$

$$= 118 \cos 50.8° + \frac{1660 \times 10^6}{800(113,500)} \sin 50.8°$$

$$= 74.6 + 14.65 = 89.25 \text{ inches per second}$$

From (11),

$$\dot{\phi}_1 = \frac{Wbt}{I} - \frac{bmV}{I}\cos\omega t + \frac{bm\omega}{I}\left(\frac{mWb^2 - WI}{kI}\right)\sin\omega t + \frac{bmV}{I}$$

$$\dot{\phi}_1 = \frac{(2200)(13.5)(.13)}{(113,500)} - \frac{(13.5)(17.1)(118)(.632)}{(113,500)} + \frac{13.5(17.1)(6.82)}{(113,500)} \left[\frac{17.1(2200)(13.5)^2 - 2200(113,500)}{(770)(113,500)}\right] 0.775$$

$$\dot{\phi}_1 = 0.034 - 0.152 - 0.0298 + 0.24 = 0.0922 \text{ radians per second}$$

The assumption is now made that these values of $\dot{y}_1$ and $\dot{\phi}_1$ represent motion about the center of gravity for the second part of the problem with the forward spring becoming effective

$$\therefore \text{at } t = o,\ y = 89.25 \text{ and } \dot{\phi} = 0.0922,\ y = 11.8,\ \phi = 11.58°$$

The final general solutions for a two-degree-of-freedom drop are

$$y = A_1 \cos(\omega_1 t + \theta_1) + A_2 \cos(\omega_2 t + \theta_2) + C \quad \text{(reference Eq. (9) and (10))}$$

$$\phi = \mu_1 A_1 \cos(\omega_1 t + \theta_1) + \mu_2 A_2 \cos(\omega_2 t + \theta_2) + D \qquad (24)$$

where $\mu_1 = \frac{-(-m\omega_1^2 + k_1 + k_2)}{bk_2 - ak_1}$ and $\mu_2 = \frac{-(-m\omega_2^2 + k_1 + k_2)}{bk_2 - ak_1}$

$$C = \frac{(k_1 a^2 + k_2 b^2)(W)}{k_1 k_2 (a+b)^2} \quad \text{and} \quad D = \frac{(-k_2 b + k_1 a)(W)}{k_1 k_2 (a+b)^2}$$

The computer program, reference Load Analysis, Appendix I, gave frequency values for the system of

$\omega_1 = 3.42$ radians per second and
$\omega_2 = 8.34$ radians per second

Substituting $k_1 = 396$, $k_2 = 770$, $a = 13.5$, $b = 57.5$, $W = 2200$, and $m = 17.1$ for the aft center of gravity position,

$$\mu_1 = \frac{-[(-17.1)(3.42)^2 + 770 + 396]}{13.5(770)-57.5(396)} = 0.0781$$

$$\mu_2 = \frac{-[(-17.1)(8.34)^2 + 770 + 396]}{13.5(770)-57.5(396)} = 0.00189$$

$$C = \frac{[396(57.5)^2 + 770(13.5)^2]\ 2200}{396(770)(71)^2} = 2.0748$$

$$D = \frac{-[(770)(13.5)-(396)(57.5)]\ 2200}{396(770)(71)^2} = 0.0177$$

Now, applying boundary conditions to the general solutions, the four constants A_1, A_2, θ_1 and θ_2 may be determined.

at $t = 0$ $y = 11.8$, $\dot{y} = 89.25$, $\phi = 11.58 = .202$ radians, and $\dot{\phi} = .0922$

$$\begin{aligned} y &= A_1 \cos\theta_1 + A_2 \cos\theta_2 + C \\ \dot{y} &= -\omega_1 A_1 \sin\theta_1 = \omega_2 A_2 \sin\theta_2 \\ \phi &= \mu_1 A_1 \cos\theta_1 + \mu_2 A_2 \cos\theta_2 + D \\ \dot{\phi} &= -\mu_1\omega_1 A_1 \sin\theta_1 = \mu_2\omega_2 A_2 \sin\theta_2 \end{aligned} \tag{25}$$

Substituting boundary conditions and known values of C, D, μ_1, and μ_2, and solving the four equations of (25) simultaneously, the four constants are found to be

$\theta_1 = -22°\ 38' = -0.395$ radians $\quad A_1 = 2.74$

$\theta_2 = -55°\ 2' = -0.96$ radians $\quad A_2 = 12.50$

The equations of (24) now become

$$y = 2.74 \cos(3.42t - 0.395) + 12.5 \cos (8.34t - 0.96) + 2.075 \qquad (26)$$

$$\phi = 0.214 \cos (3.42 - 0.395 - 0.0236 \cos (8.34 - 0.96) + 0.0177$$

By trial and error, the maximum deflection is found to occur at $t = 0.115$ second (actual time after initial impact is $0.115 + 0.13 = 0.245$). Therefore,

$$\text{at } t = 0.115 \quad y = 17.315 \text{ inches}, \quad \phi = 0.208 = 12°$$

The gear deflection is

$$\delta_{aft} = 17.315 + 13.5 (\tan 12°) = 20.195 \text{ inches}$$

$$\delta_{fwd} = 17.315 - 57.5 (\tan 12°) = 5.065 \text{ inches}$$

Load on aft gear is

$$k\delta = 770 (20.195) = 15,550 \text{ pounds}$$

Load on the forward gear is not critical.

Because of the large pitch angle ($\phi = 12°$) in conjunction with the large aft strut deflection, 20.195 inches, the aircraft tail skid would make ground contact before such high displacements could be obtained. Therefore, the foregoing analysis is conservative, since loads would be reduced on the aft strut.

APPENDIX II - STRESS ANALYSIS

This appendix presents the stress analysis of the UH-1 reinforced fiber glass plastic landing gear. The gear was analyzed for the load conditions specified under the section Design Requirements and as quantitatively presented in the Load Analysis, Appendix I.

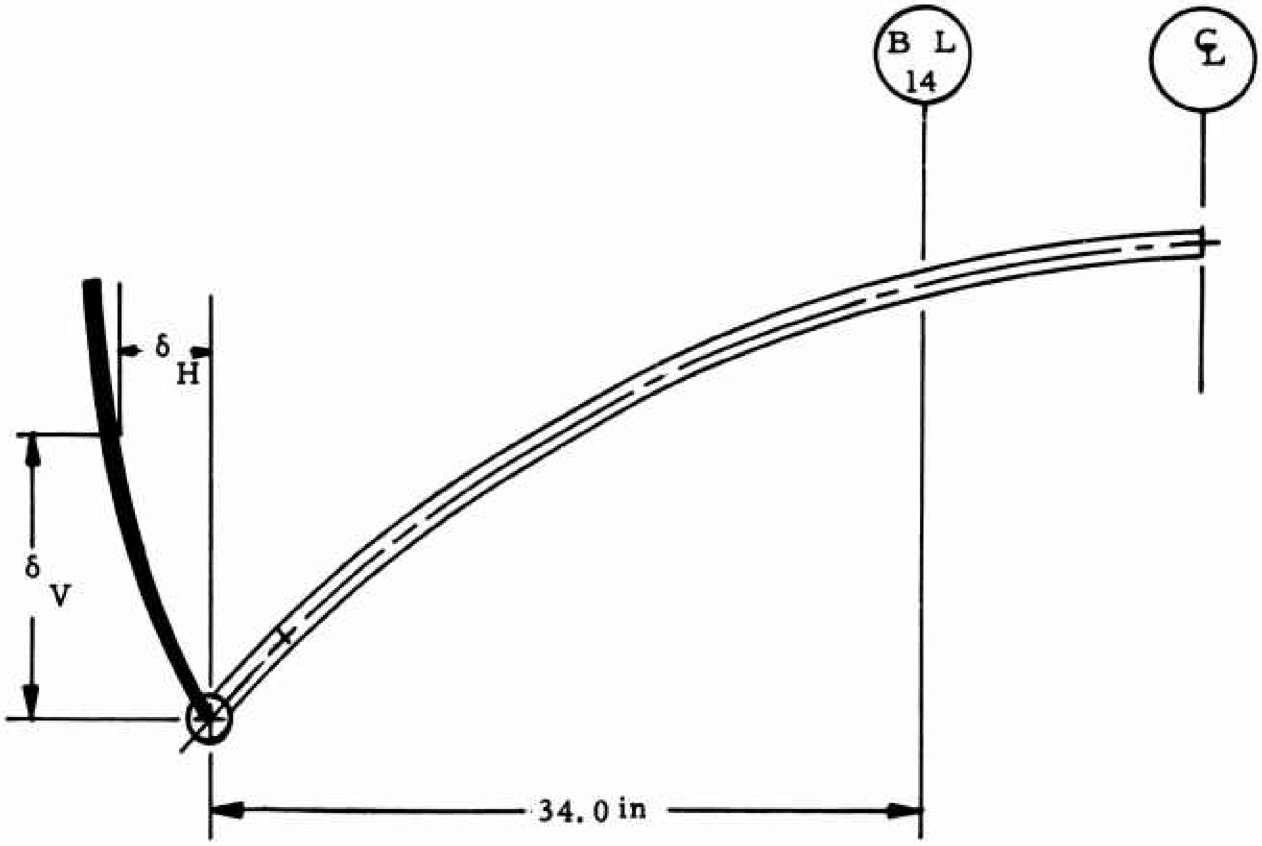
The design load criteria for the stress analysis were established from the loads expected in required drop tests and those loads to be directly applied in static tests. Preliminary analyses indicated that critical loads and stresses from drop tests at ultimate sink speeds of 9.84 feet per second are less severe than the static test requirements of applying horizontal loads equal to one-half the vertical loads from a drop at limit sink speed of 8.0 feet per second. Thus, the detailed stress analyses of this section are primarily for these latter conditions. A summary of the design conditions and loads is presented in Table XIII.

TABLE XIII
SUMMARY OF VERTICAL LOADS FROM LEVEL LANDINGS

Condition	Loads(2)			Deflection(3)	
(1)	Aft (lb)	Forward (lb)	Total (lb)	Aft (in)	Forward (in)
A	12,774	5,127	17,901	17.6	16.6
B	12,453	5,492	17,945	17.2	17.8
C	10,752	4,320	15,072	14.8	14.0
D	10,487	4,622	15,109	14.4	15.0
E	11,696	4,661	16,357	16.1	15.1

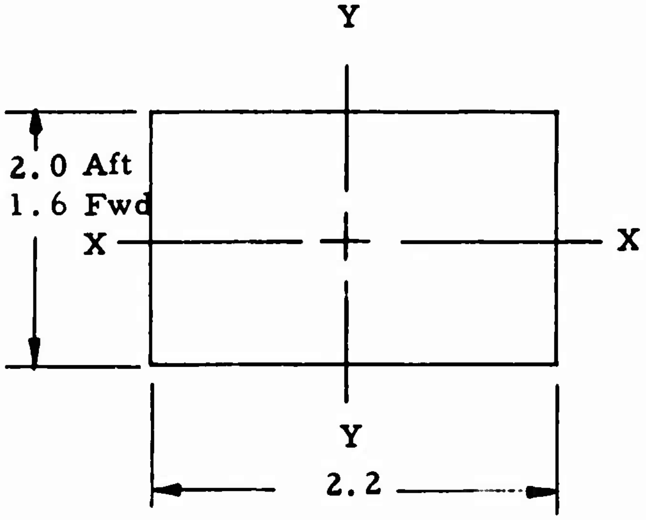
(1) See Table VII for description of conditions
(2) Forward and Aft refer to the two cross members
(3) Deflection equals P/k
k = 308 lb/in, forward cross member
k = 726 lb/in, aft cross member

Deflection geometry used in this analysis is as follows:



Moment arm after deflection = 34.0 + (δ_H at δ_V) = 39.2 inches.

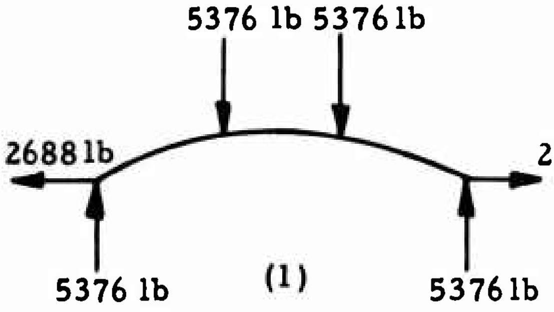
The reinforced plastic cross members have the following section properties:



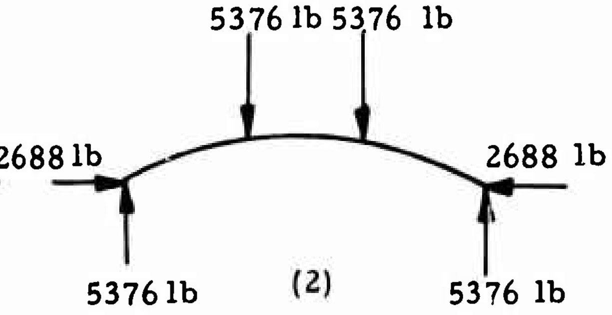
	Aft	Forward
I_{XX}	1.47 in^4	0.764 in^4
S_{XX}	1.47 in^3	0.955 in^3
I_{YY}	1.79 in^4	1.43 in^4
S_{YY}	1.63 in^3	1.30 in^3
A	4.40 in^2	3.52 in^2

AFT CROSS MEMBER

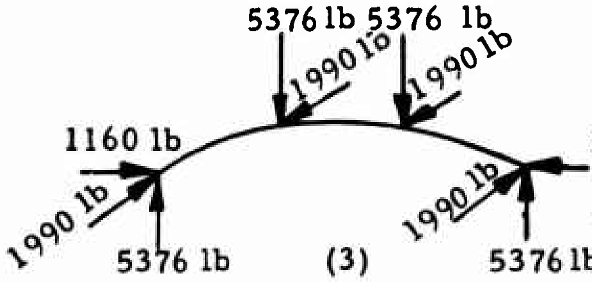
Condition C, loads from limit sink speed with aft center of gravity, is used to obtain the maximum vertical loads on the aft strut. Loads equal to one-half of this value are then allowed to act in fore and aft and inboard and outboard directions. The sketches below represent these loads on the strut. Sketches (1) and (2) are self-explanatory. Sketches (3) and (4) are loads determined from a moment distribution analysis.



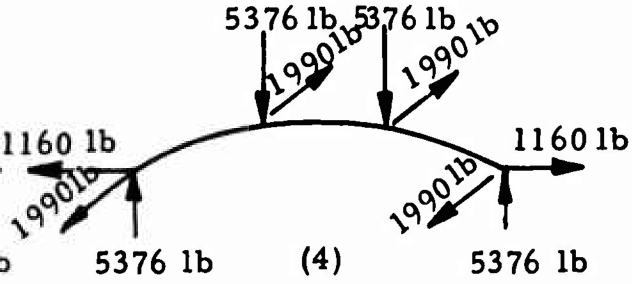
(1)



(2)



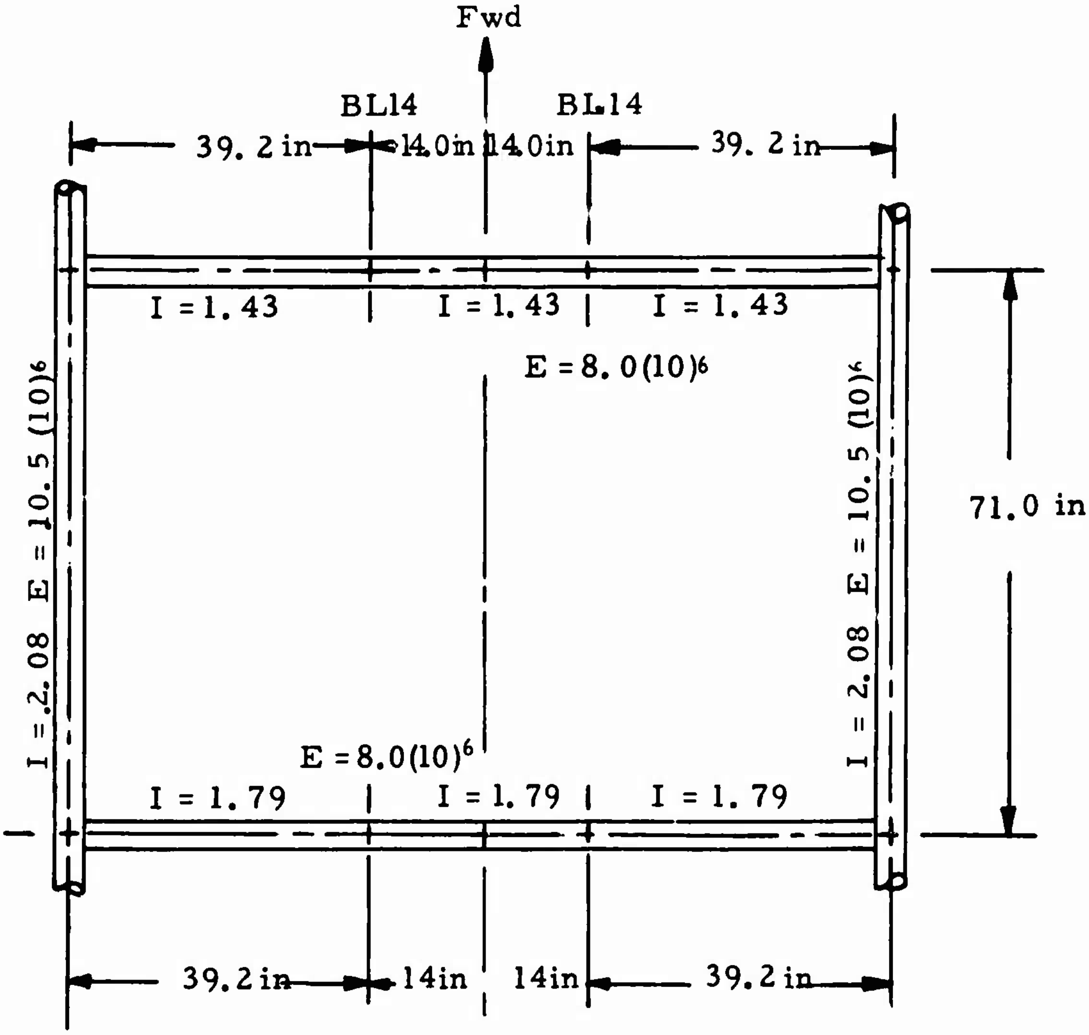
(3)



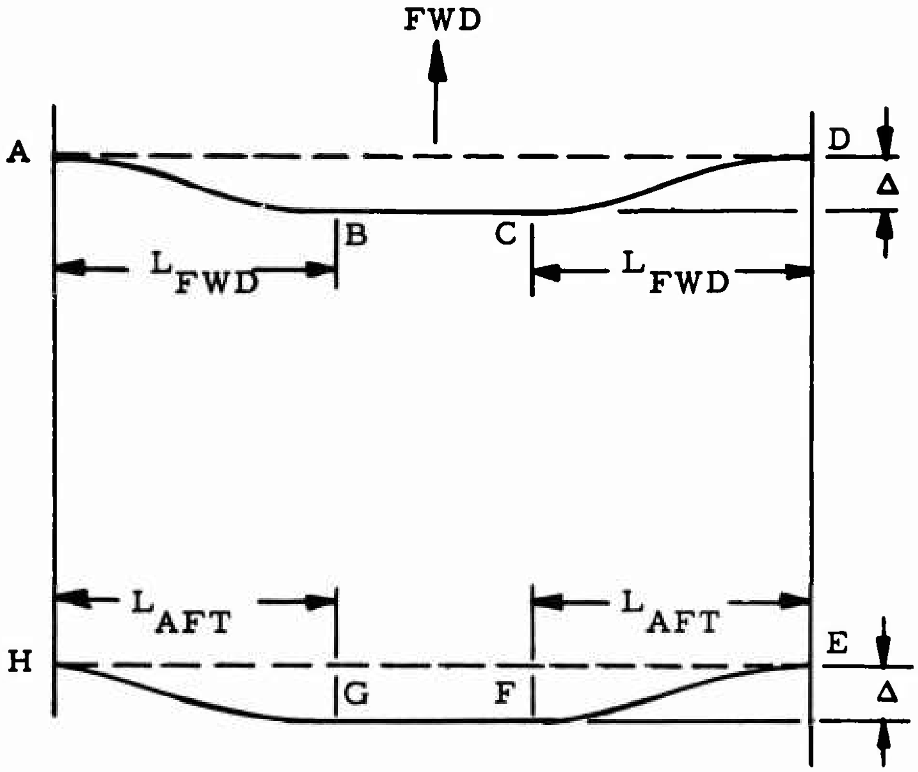
(4)

From observation, (1) and (4) produce greater bending moments than do (2) and (3). Since it is not so obvious which of (1) and (4) is the critical case, an analysis was performed which revealed that the existence of a moment about the Y-Y axis, makes (4) the design condition. The analysis of case (4) follows.

Distribution of Fore and Aft Loads



The above sketch represents a plan view of the gear system under load. The beam length of 39. 2 inches from BL 14 outboard is moment arm due to gear spread. The method of moment distribution was used to determine shear, axial and bending loads from the applied fore and aft loads.



To calculate fixed end moments, a deflection of $\Delta = 1.00$ inch is assumed. From the formula $M = 6EI\Delta / L^2$,

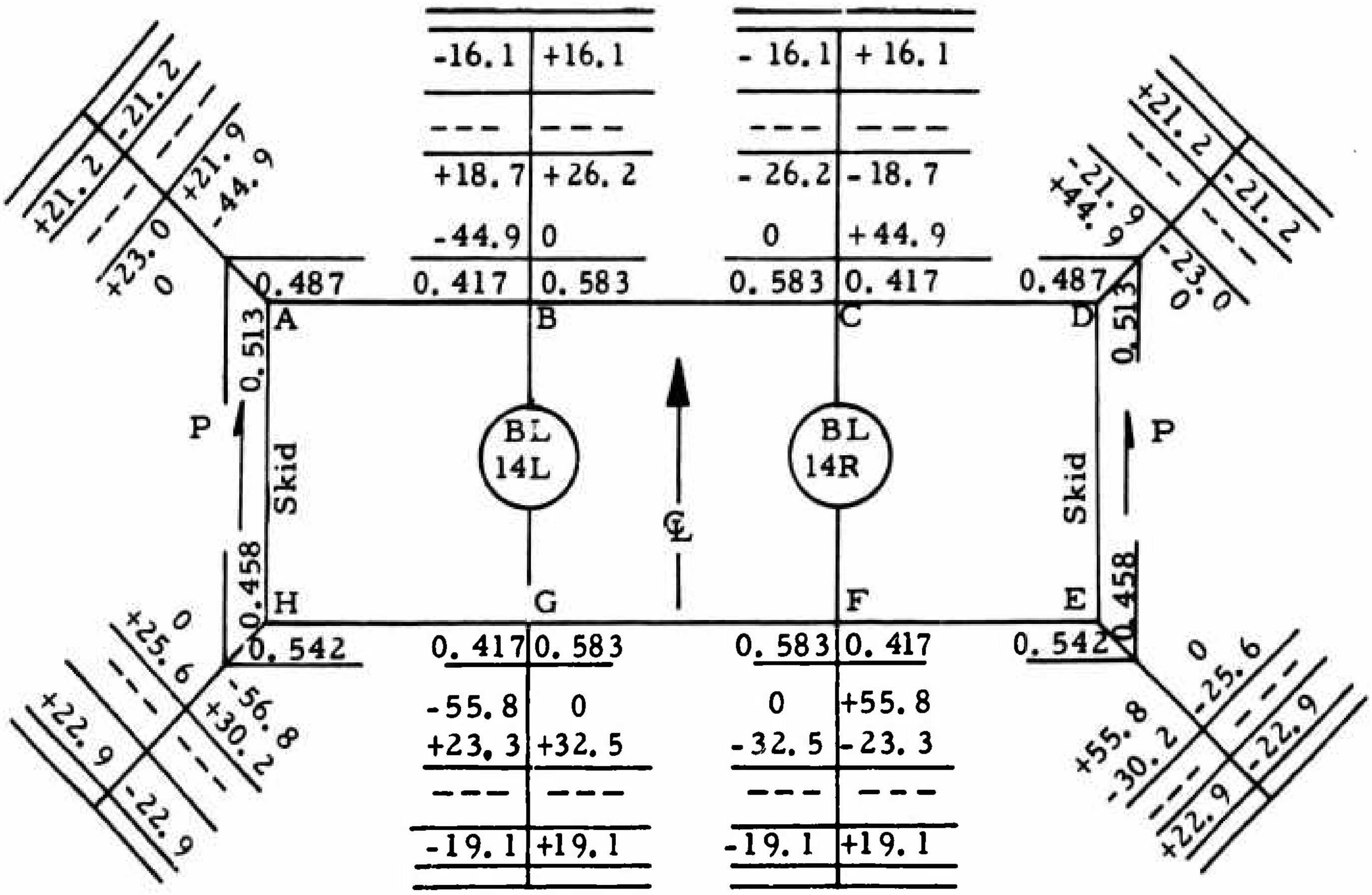
$$M_{FWD} = 6(11.5 \times 10^6) / (39.2)^2 = 44,900 \text{ in-lb}$$

$$M_{AFT} = 6(14.3 \times 10^6) / (39.2)^2 = 55,800 \text{ in-lb}$$

Distribution Factors:

$\overline{AB}$: EI/L = 0.293 ~ D. F. = 0.487
$\overline{AH}$: EI/L = 0.309 ~ D. F. = 0.513
$\overline{BA}$: EI/L = 0.293 ~ D. F. = 0.417
$\overline{BC}$: EI/L = 0.411 ~ D. F. = 0.583

$\overline{HA}$: EI/L = 0.309 ~ D. F. = 0.458
$\overline{HG}$: EI/L = 0.365 ~ D. F. = 0.542
$\overline{GH}$: EI/L = 0.365 ~ D. F. = 0.417
$\overline{GF}$: EI/L = 0.511 ~ D. F. = 0.583



P = Σ Shears on Cross Members

P = (21, 200 + 16, 000 + 22, 900 + 19, 000)/39. 2 = 2020 lb

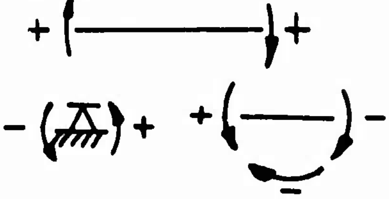
P_{Actual} = 1/4 x $P_{Vertical}$ (Cond 5)

= 1/4 x 15, 709 = 3920 lb

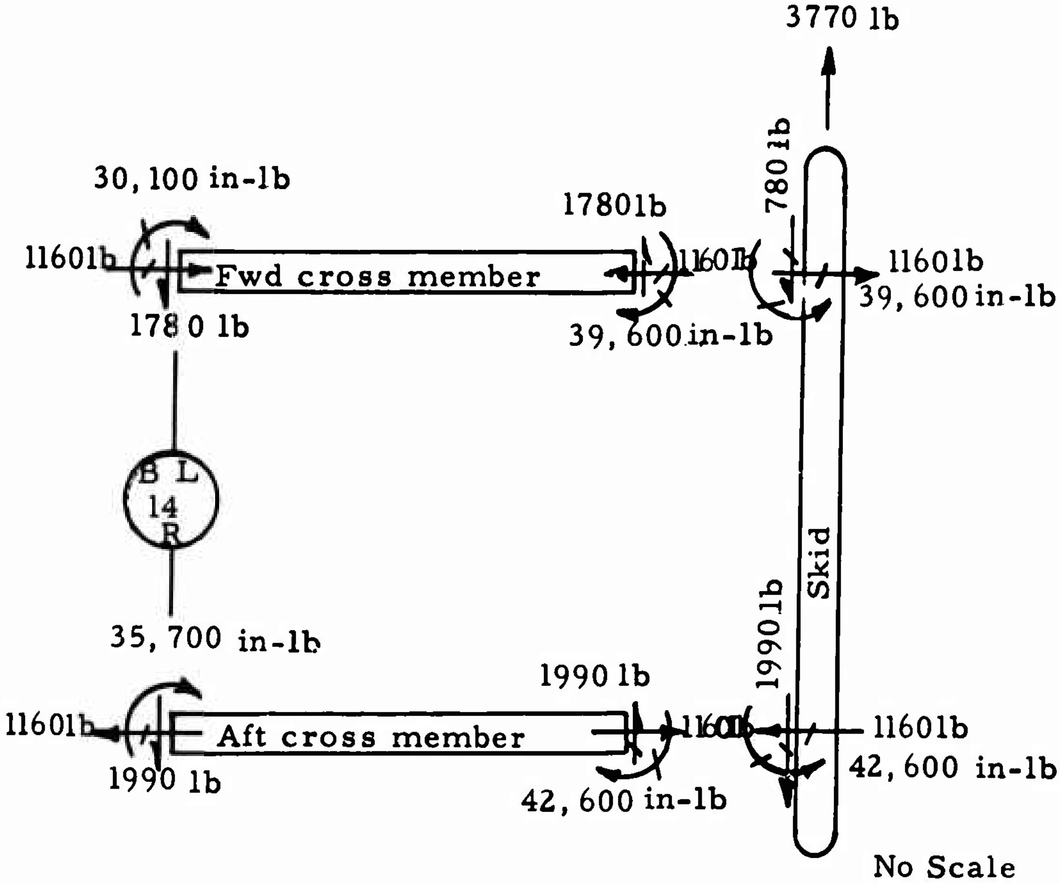
R = 3920/2020 = 1. 94

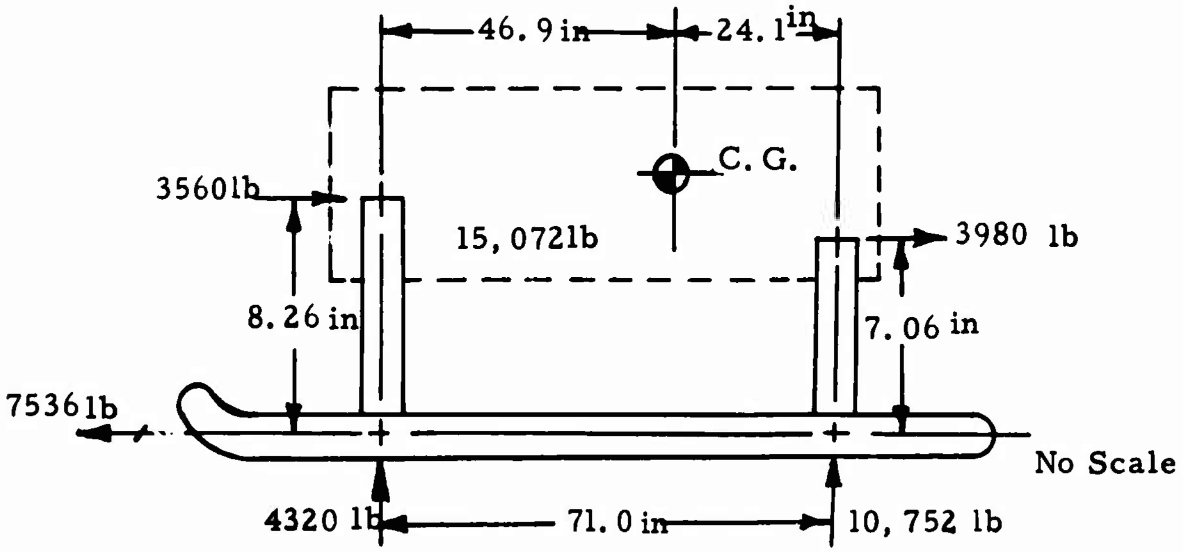
$M_{Actual} = RM_{Calculated}$

Sign Convention:



The distribution of the critical loads for the aft strut is as shown below. The center of gravity is shown relocated for static equilibrium.





Cross Member At BL 14

Bending and Tension:

$$M_{xx} = \frac{10,752}{2}(39.2) + 1160\ (7.06) = 219,200 \text{ in - lb}$$

$$M_{yy} = 35,700 \text{ in - lb}$$

$$\Sigma f = \frac{M_{xx}}{S_{xx}} + \frac{M_{yy}}{S_{yy}} - \frac{P}{A} \qquad \text{(At corner of cross section)}$$

$$= \frac{219,200}{1.47} + \frac{35,700}{1.63} - \frac{1160}{4.40}$$

$$= 171,160 \text{ psi}$$

$$M.S. = \frac{F}{f} - 1 = \frac{183,000}{171,160} - 1 = \underline{\underline{+0.07}}$$

Bending and Shear: (On Y-Y Neutral Axis)

$$f_b = \frac{M_{xx}}{S_{xx}} = \frac{219,200}{1.47} = 149,000 \text{ in - lb}$$

$$f_s = \frac{3}{2}\frac{V}{A} = \frac{3}{2}\left(\frac{1990}{4.40}\right) = 679 \text{ psi}$$

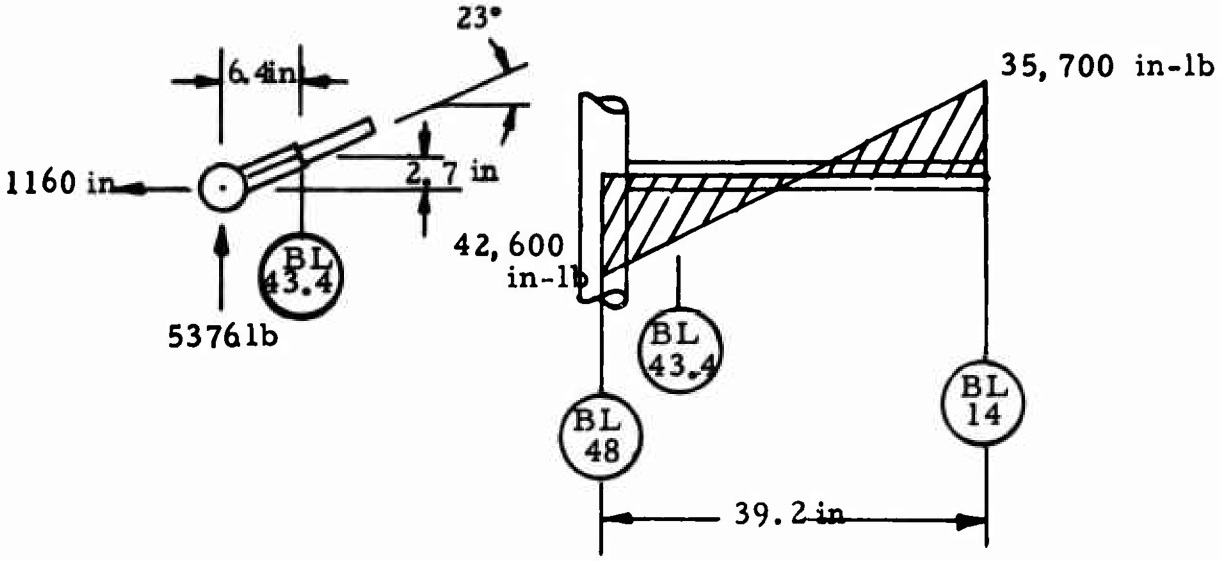
$$R_b + R_s = 1$$

$$R_b = \frac{f_b}{F_b} = \frac{149,000}{183,000} = 0.815$$

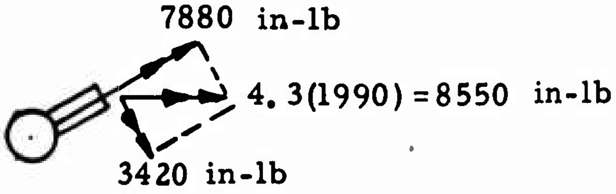
$$R_s = \frac{f_s}{F} = \frac{710}{7000} = 0.097$$

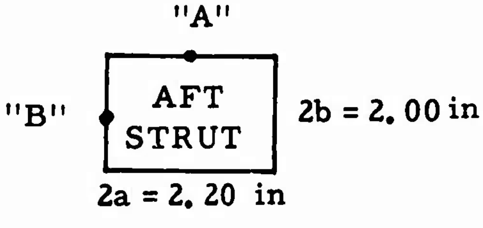
$$M.S. = \frac{1}{R_b + R_s} - 1 = \frac{1}{0.912} = +\ \underline{0.10}$$

Cross Member At BL 43.4



The centroid of the cross section at BL 43.4 is 4.3 inches below the point of fore and aft load application at BL 14. Thus, the longitudinal load of 1990 pounds results in torsion as follows:





CROSS SECTION

$$f_{\substack{st \\ (max)}} = \frac{T(3a + 1.8b)}{8a^2 b^2} = \frac{7880(5.1)}{8(1.21)} = 4150 \text{ psi}$$

Maximum torsion stress occurs at point "A". It is assumed that the same value of torsion stress acts at point "B".

Bending and Shear, Point "A"

$$f_b = \frac{M_{xx}}{S_{xx}} = \frac{5376\ (6.4)\ +\ 1160\ (2.7)}{1.47} = 25,500 \text{ psi}$$

$$f_s = f_{st} + \frac{3\ V}{2\ A} = 4150 + \frac{1.5\ (1990)}{4.40} = 4828 \text{ psi}$$

$$R_b = \frac{f_b}{F_b} = \frac{25,500}{183,000} = 0.139$$

$$R_s = \frac{f_s}{F_s} = \frac{4828}{7000} = 0.688$$

$$\text{M. S.} = \frac{1}{R_s + R_b} = 1 = \frac{1}{0.688 + 0.139} -1 = \underline{\underline{+0.21}}$$

Bending and Shear, Point "B"

$$f_b = \frac{\Sigma M_{yy}}{S_{yy}} = \frac{\left[42,600 - 1990\ (6.4)\right] - 3420}{1.63} = 16,200 \text{ psi}$$

$$f_s = f_{st} + \frac{3\ V}{2\ A} \doteq 4150 + \frac{1.5\ (5376)}{4.40} = 5980 \text{ psi}$$

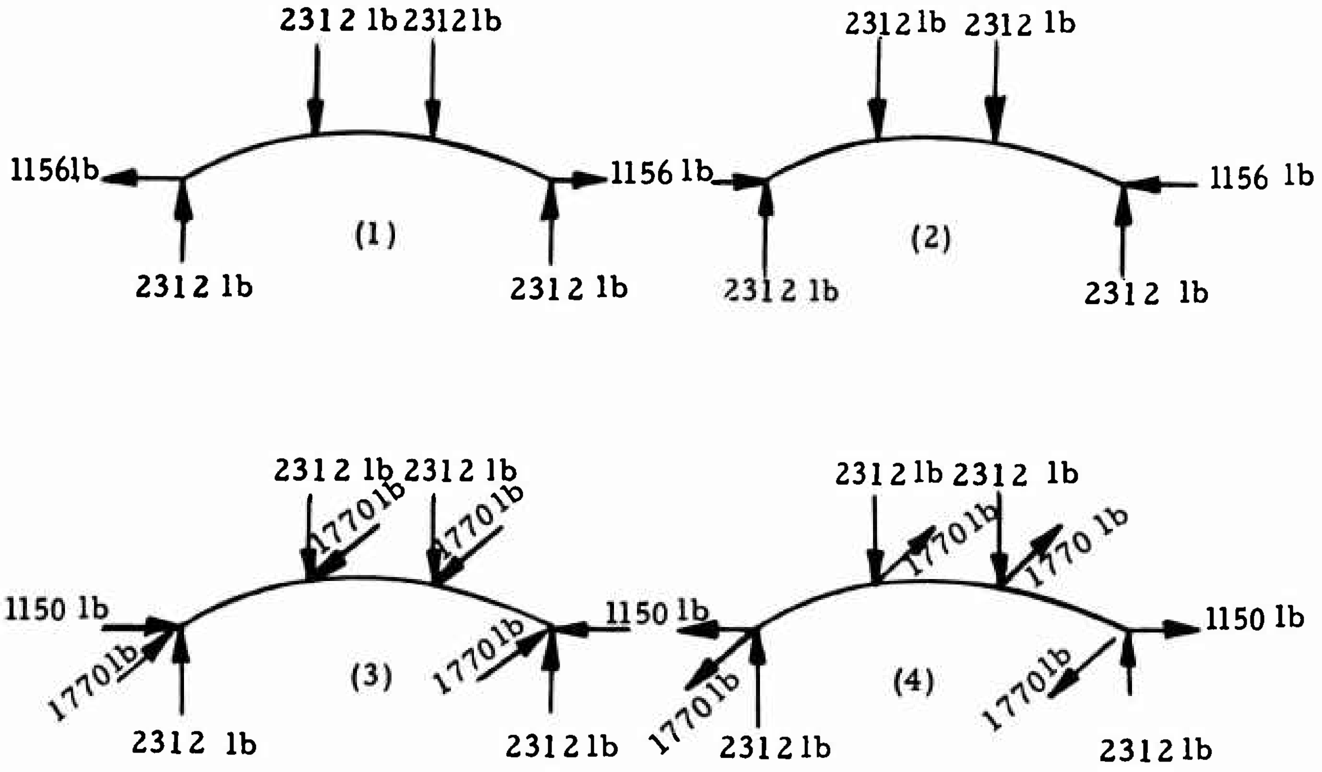
$$R_b = {f_b}/{F_b} = 16,200/_{183,000} = 0.088$$

$$R_s = {f_s}/{F_s} = 5980/_{7000} = 0.854$$

$$\text{M. S.} = \frac{1}{R_s + R_b} -1 = \frac{1}{0.854 + 0.088} -1 = \underline{\underline{+0.06}}$$

FORWARD CROSS MEMBER

The forward strut, like the aft strut, required preliminary investigation to determine the design loading. As in the aft strut, the most critical loading was found to be the peak vertical load of a limit drop in conjunction with horizontal loads equal to one-half this value. Condition D, for a forward center of gravity position, gives the maximum vertical load for this analysis. The possible combinations of loads for the static tests are shown below.



The horizontal loads of (3) and (4) were found from the moment distribution solution shown on page 71 and as follows. The detailed analysis is for conditions shown in (4).

$$P = 2020 \text{ lb}$$

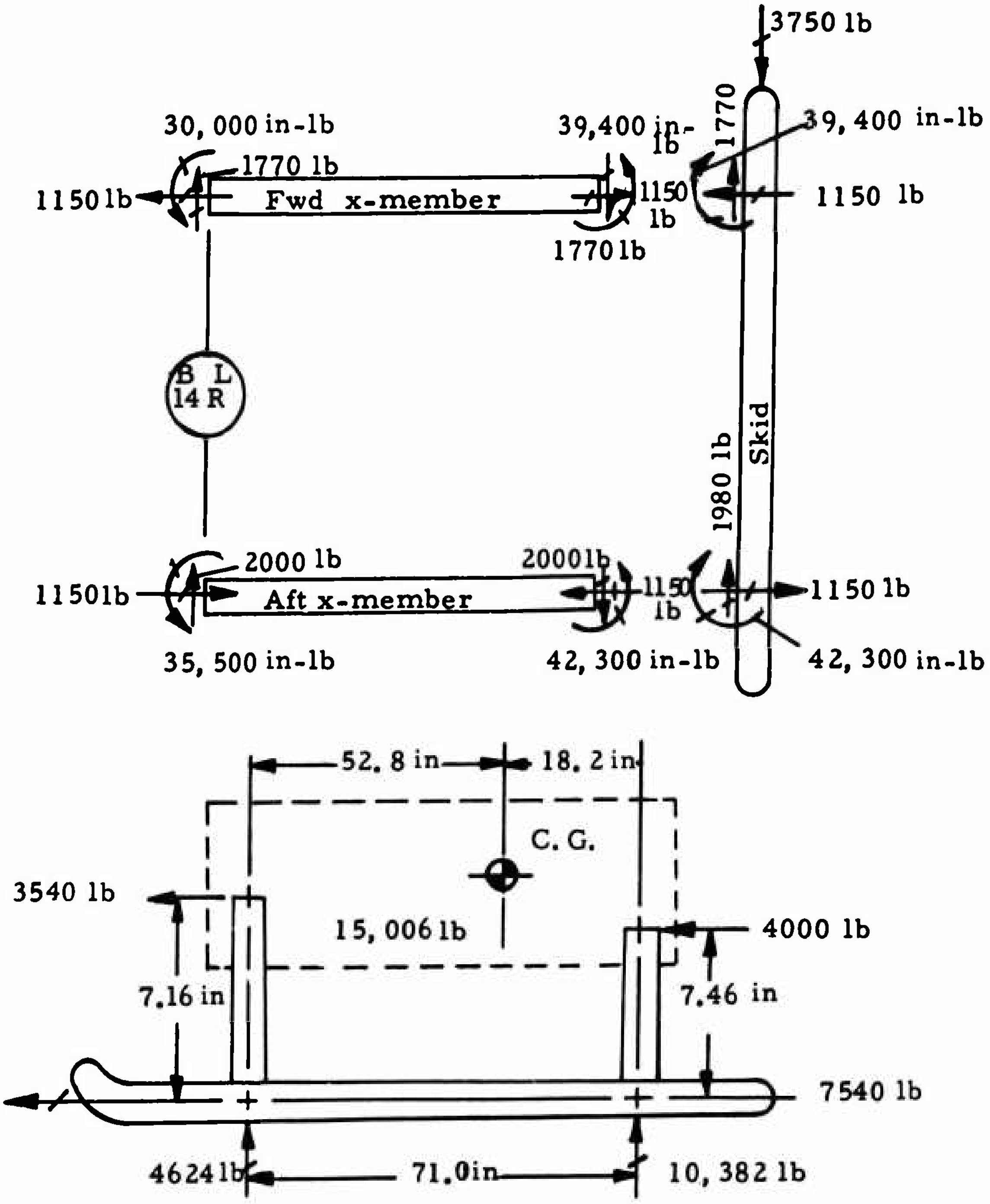
$$M_{actual} = 1/4 \times P_{vertical} \qquad (4)$$

$$= 1/4 \times 15,006 = 3750 \text{ lb}$$

$$R = 3750/2020 = 1.86$$

$$M_{actual} = RM_{calculated}$$

The distribution of the critical loads for the forward cross member is as shown below. The center of gravity is shown relocated for static equilibrium.



Cross Member AT BL 14

Bending and Tension:

$$M_{xx} = 2312\ (39.2) + 1150\ (7.16) = 98,700 \text{ in-lb}$$

$$M_{yy} = 30,000 \text{ in-lb}$$

$$P_{tens} = 1150 \text{ lb}$$

$$\Sigma f = \frac{M_{xx}}{S_{xx}} + \frac{M_{yy}}{S_{yy}} - \frac{P}{A} \quad \text{(At corner of cross section)}$$

$$= \frac{98,700}{0.955} + \frac{30,000}{1.30} - \frac{1150}{3.52} = 126,300 \text{ psi}$$

$$\text{M.S.} = \frac{F}{f} - 1 = \frac{183}{126} - 1 = +0.45$$

Bending and Shear: (On Y-Y Neutral Axis)

$$f_b = \frac{M_{xx}}{S_{xx}} = \frac{98,700}{.955} = 103,500 \text{ psi}$$

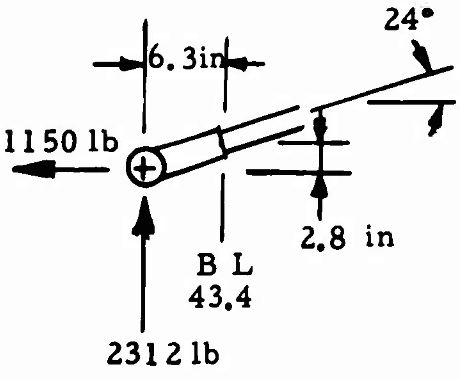
$$f_s = 1.5\frac{V}{A} = 1.5\ \frac{1770}{3.52} = 750 \text{ psi}$$

$$R_b = \frac{f_b}{F_b} = \frac{103.5}{183} = 0.566$$

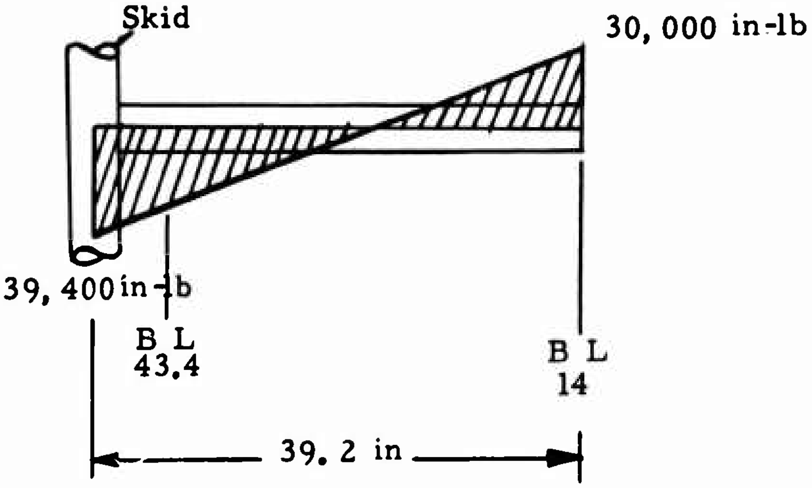
$$R_s = \frac{f_s}{F_s} = \frac{750}{7000} = 0.107$$

$$\text{M.S.} = \frac{1}{R_b + R_s} - 1 = \frac{1}{.673} - 1 = +0.49$$

Cross Member At BL 43.4 .

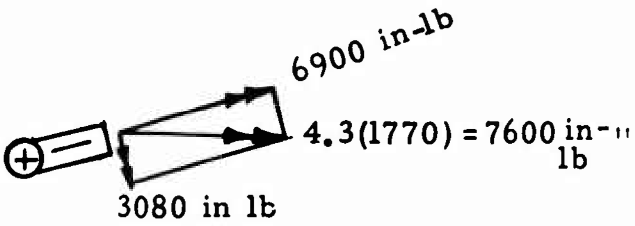


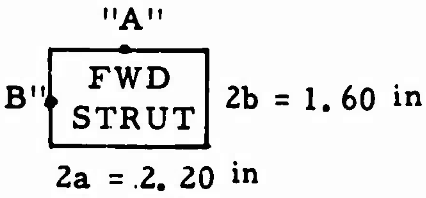
Vertical Loading



Fore and aft bending

At BL 43.4, a torque arm of 4.3 inches is associated with the fore and aft load of 1770 pounds.





Cross Section

$$f_{\substack{st\\max}} = \frac{T(3a + 1.8b)}{8a^2 b^2} = \frac{6900(4.74)}{8(1.1)^2(.80)^2} = 5250 \text{ psi}$$

(at point "A")

Bending and Shear, Point "A"

$$f_b = \frac{M_{xx}}{S_{xx}} = \frac{[1150\ (0.28) + 2312\ (6.3)]\ [\cos 24]}{0.955} = 17,000 \text{ psi}$$

$$f_s = f_{st} + 1.5\ \frac{V}{A} = 5250 + \frac{1.5(1770)}{3.52} = 6,000 \text{ psi}$$

$$R_b = \frac{17,000}{183,000} = 0.093$$

$$R_s = \frac{6,000}{7,000} = 0.857$$

$$\text{M.S.} = \frac{1}{R_s + R_b} - 1 = \frac{1}{.857 + .093} - 1 = 0.05$$

Bending and Shear, Point "B"

$$f_b = \frac{\Sigma M_{yy}}{S_{yy}} = \frac{[39,400 - 1770\ (6.5)] - 3080}{1.30} = 19,100 \text{ in-lb}$$

$$f_s = f_{st} + \frac{1.5V}{A} = \frac{1.60}{2.20}\ (5250) + \frac{1.5(2312)}{3.52} = 4800 \text{ psi}$$

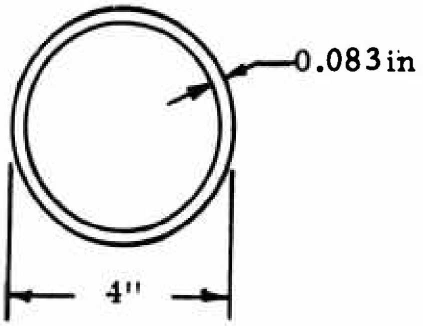
$$R_b = \frac{19,100}{183,000} = 0.104$$

$$R_s = \frac{4800}{7000} = 0.686$$

$$\text{M.S.} = \frac{1}{R_s + R_b} - 1 = \frac{1}{0.790} - 1 = +0.27$$

SKID

Section Properties



Ref. (2), Table 1002.b

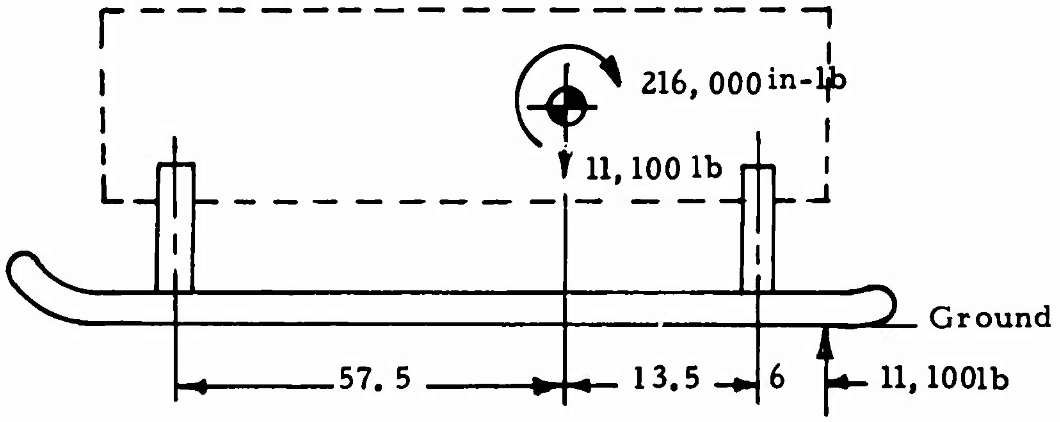
$A = 1.0214$ in ^{2}

$I = 1.9597$ in ^{4}

$S = 0.9799$ in ^{3}

Critical Loading

The critical load on the skid comes from Condition VI, page 62. The skid is analyzed here for Condition VI, with ground contact impending.



Maximum moment on skid is at the aft strut.

$$M = 1/2 \ (11,100) \ (6) = 33,300 \text{ in-lb}$$

$$f_b = 33,300/.98 = 34,000 \text{ psi}$$

$$F_{cr} = 43,200 \text{ psi}$$

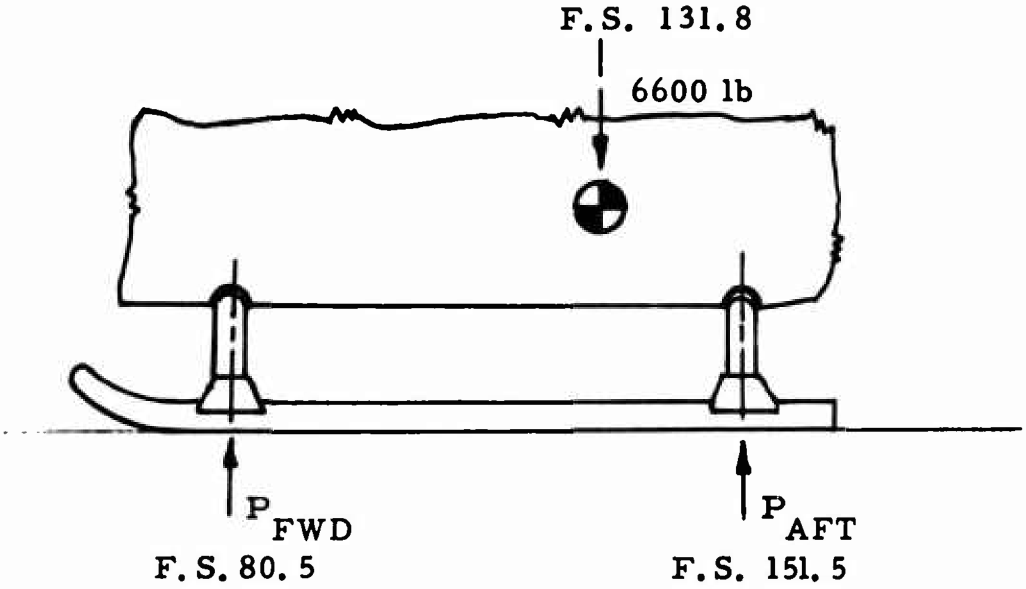
$$M.S. = \frac{43,200}{39,000} - 1 = 0.27$$

DEFLECTION

This section gives the determination of the UH-1 static position with RFP landing gears installed as compared with the static attitude with the existing aluminum gear system. Calculations are also shown for determining the gear spring rate, which, in turn, governs the static attitude.

Static Attitude

Static deflections are computed for the present aluminum and for the reinforced plastic landing gears in order to compare their ground attitude. It is assumed that the center of gravity is at fuselage station, F. S., 131. 8.



$$P_{FWD} = 19.7\,(6600)/71.0 = 1830 \text{ lb} = P_1$$

$$P_{AFT} = 51.3\,(6600)/71.0 = 4770 \text{ lb} = P_2$$

Aluminum Cross Members:

$$\delta_{FWD} = P_1/k_1 = 1830/900 = 2.03 \text{ inches}$$

$$\delta_{AFT} = P_2/k_2 = 4770/1810 = 2.64 \text{ inches}$$

The centerline of these gears at B L 14 is 19.96 inches above the ground surface at no load. Let h equal the distance from the ground at static load.

$$h_{FWD} = 19.96 - 2.03 = 17.93 \text{ inches}$$

$$h_{AFT} = 19.96 - 2.64 = 17.32 \text{ inches}$$

Reinforced Plastic Cross Members:

$$\delta_{FWD} = P_1/k_1 = 1830/308 = 5.94 \text{ inches}$$

$$\delta_{AFT} = P_2/k_2 = 4770/726 = 6.56 \text{ inches}$$

The centerline of these struts at B L 14 is 22.96 inches above the ground surface at no load.

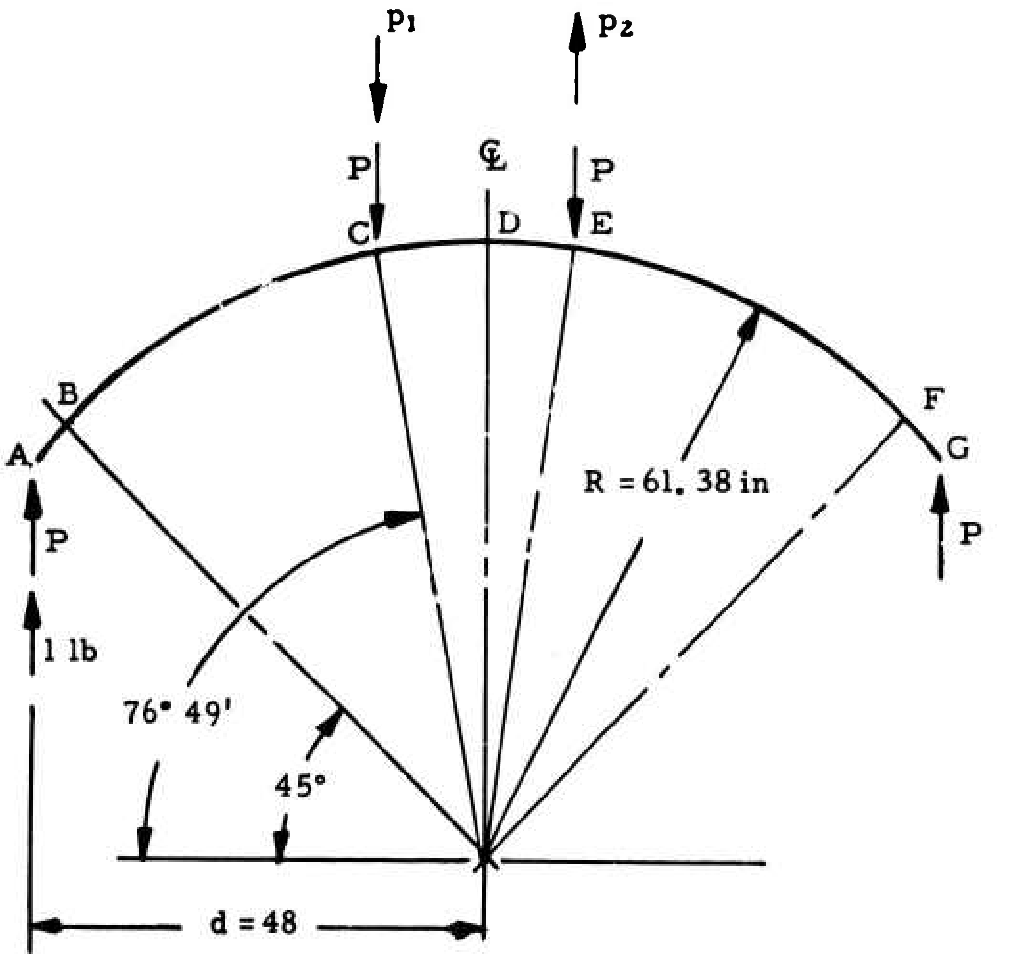
$$h_{FWD} = 22.96 - 5.94 = 17.02 \text{ inches}$$

$$h_{AFT} = 22.96 - 6.56 = 16.40 \text{ inches}$$

From the above, it is seen that only a small difference exists in the static attitudes of the two systems.

Spring Rate

The unit load method was used to calculate the spring constant for various landing gears. The following analysis shows the spring constant derivation for a solid rectangular cross section of constant width and depth.



$M_{AB} = Px \qquad m_{AB} = x$

$M_{BC} = Px \qquad m_{BC} = x$

$M_{CD} = 34P \qquad m_{CD} = x - p_1 (x-34) = 34-p_2(x-34)$

$M_{ED} = 34P \qquad m_{ED} = p_2 (x-34)$

$s = 2x$
$x = d-R\cos\phi$
$p_1 = p_2 + 1$

$$\delta_v = \int_A^E \frac{Mmds}{EI} = \int_A^B \frac{Mmds}{EI} + \int_B^C \frac{Mmds}{EI}$$

$$+ \int_C^D \frac{Mmds}{EI} + \int_E^D \frac{Mmds}{EI}$$

Substituting into the above expression and integrating, one obtains after considerable simplification:

$$EI = 15,300\ k$$

where

$$k = 2\,P/\delta_v$$

Thus, for a desired spring constant, the required stiffness, EI, for designing the member may be determined from the above equation.

APPENDIX III - TEST PROGRAM

A series of static and drop tests were accomplished on the UH-1 reinforced plastic landing gear. All planned static tests were completed. Some of the drop tests were deleted after failure of the aft cross member because of the unavailability of a new member in this program.

This appendix summarizes the planned program and describes methods, equipment, and instrumentation used for the tests. Detail results of all tests are included.

STATIC TESTS

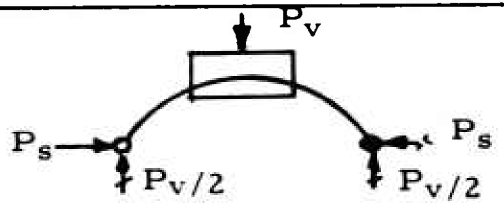
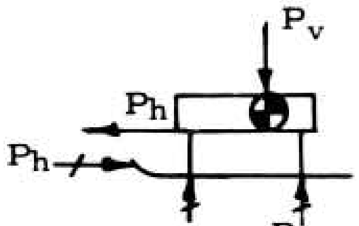
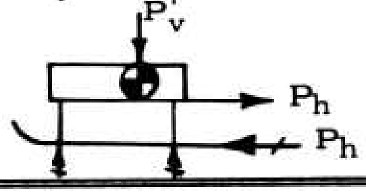
Static tests were accomplished by a special steel framework test fixture. Hydraulic cylinders were used to apply incremental loads in three mutually perpendicular directions. Dillon dynamometers were used to measure the load applied by the cylinders. Type C-40 strain gages, Budd Instrument Company, were located inboard of the ball joint fittings at the top and bottom adjacent to the edge of each cross member. This was a point of maximum stress due to bending. An attempt was made to determine shear stresses by the use of rosette strain gages. The results were of little value and are not included in this report. Strains were determined by the use of a Budd Instrument Company switch and balance unit, Model C-106, and digital strain indicator, Model A-110. Landing gear spread was measured using an extensional tape. Vertical deflection was measured with a transit.

Measurements of strains, deflections and loads were recorded for each increment of load. Four static tests were made. In each test a vertical load was applied at the aircraft center of gravity equal to the maximum vertical reaction from a level landing of 8 feet per second in combination with 2/3 W rotor lift at design gross weight, 6, 600 pounds. A horizontal load equal to one-half the vertical load, including the effects of friction of the skid on the steel plate, was then applied in an inboard, outboard, forward, and aft direction respectively.

Table XIV defines the static load conditions. The magnitudes of the vertical loads were obtained after emperically determining the landing gear spring constants and solving for reactions for an 8-feet-per-second sinking speed. The test fixture applies side load to one skid and is reacted by the opposite skid. Because of this method of loading, the correct maximum bending moments in the cross member are produced by applying a side load force of one quarter of the vertical load. In the first two

TABLE XIV

STATIC TESTS - UH-1 LANDING GEAR

Test Condition	Description	Center of Gravity	Sketch	Table Number
I	Vertical Load with Inboard Side Load	F.S. 138.0		XV
II	Vertical Load with Outboard Side Load	F.S. 138.0	Same as Condition I with P_s Acting Outboard	XVI & XVII
III	Vertical Load with Forward Long Load	F.S. 125.0		XVIII
IV	Vertical Load with Aft Long Load	F.S. 138.0		XIX

static tests, a side load greater than required was inadvertently applied. The friction between the skids and the steel plate surface of the test fixture was accounted for in determining the side load. The side load was increased by an amount necessary to overcome the friction due to a friction coefficient of 0.09. Side loads were applied in a manner such that each cross member reacted its proportional share of the load.

Because of method of loading, it was necessary to modify the vertical load for Conditions III and IV to account for the couple effect of the longitudinal load. Condition III was critical for the forward cross member. Therefore, the vertical load was modified to result in the same vertical load in the forward member as determined for Condition I. Condition IV was critical for the aft cross member. In a like manner, the load was modified to give the same vertical load in the aft member as determined for Condition I.

DROP TESTS

Drop tests of the landing gear were accomplished through the use of a special fixture simulating the weight, center of gravity, and mass moment of inertia of the UH-1 helicopter. The test conditions are as shown in Table XX. In these tests, the entire unit was raised at the required angle to a height such that the distance between the lowest part of the skid and the drop surface was the height necessary to result in the specified impact velocity. A bomb shackle was used for quick release. The drop surface consisted of a thick unpainted steel plate resting on packed sand. See Figure 13.

Acceleration load factors were measured with five Consolidated Electrodynamics Corporation (CEC) linear accelerometers, Type 4-202, located such that pitch, roll, yaw, and translatory accelerations could be determined. To accomplish this, three accelerometers were placed at the center of gravity in mutually perpendicular directions, and single accelerometers were placed to measure vertical acceleration at a distance forward and aft of the center of gravity.

Type C-40 strain gages were used to determine strain in the cross members. They were located inboard of the ball joint fittings on the top and bottom of the member adjacent to the edge. In these locations, maximum bending stresses for unsymmetrical bending cases could be determined. Deflection was measured at four points, the intersection of skids and cross members, by the flexure of thin aluminum straps placed accross the skids and attached to the test fixture frame. Strain gages on the straps were calibrated to determine deflection. A sharp probe at each of the ball fittings was allowed to penetrate a block of low-density foam as a means of ascertaining accuracy of maximum deflection measurements.

Data were continuously recorded during all drop tests through the use of a CEC Type 119 36-channel recorder, a CEC Type 118 amplifier, and a CEC Type 108 bridge balance unit.

Table VI summarizes the drop test results. Following is a more detailed presentation of acceleration and deflection data. No data were obtained from Condition I for an impact velocity of 5 feet per second because of a malfunction of recording equipment. Acceleration time histories are included for all other test conditions and deflection time histories for some.

TABLE XV

SUMMARY OF RESULTS, STATIC TEST CONDITION I
VERTICAL LOAD WITH INBOARD ACTING SIDE LOAD, AFT CENTER OF GRAVITY

Vertical Load (1)	Side Load	Vertical Deflection		Tread (2)		Maximum Strain Fwd (3)		Maximum Strain Aft (3)	
		Fwd	Aft	Fwd	Aft	Top	Bottom	Top	Bottom
(lb)	(lb)	(in)	(in)	(in)	(in)		(microinches)		
0	0	0	0	92.75	92.69	- 1	- 1	+ 1	- 2
4,000	0	2.64	2.43	96.37	95.63	-2,964	+2,688	-3,135	+3,148
6,000	0	4.47	4.38	98.18	97.56	-4,440	+4,130	-5,050	+5,170
6,800	3,400	5.08	4.93	98.75	98.13	-4,930	+4,560	-5,535	+5,700
8,000	4,000	4.69	4.76	98.37	98.00	-4,690	+4,130	-5,610	+5,620
10,000	5,000	4.87	4.91	98.50	98.00	-4,950	+4,400	-5,925	+5,930
12,000	6,000	4.97	5.02	98.50	98.06	-5,140	+4,570	-6,210	+6,180
13,500	6,750	5.22	5.11	98.75	98.13	-5,420	+4,970	-6,330	+6,370
14,500	7,250	5.44	5.19	98.81	98.18	-5,630	+5,230	-6,470	+6,560
15,750	7,875	5.65	5.27	99.06	98.25	-5,830	+5,470	-6,601	+6,710

(1) Vertical load shown does not include 390 pounds, the weight of the static test fixture.
(2) Dimension between inboard sides of skids.
(3) Strains measured inboard of ball joint fitting, maximum bending moment; $E_b = 8.46 \times 10^6$ psi (aft cross member); $E_b = 6.79 \times 10^6$ psi (forward member).

TABLE XVI

SUMMARY OF RESULTS, STATIC TEST CONDITION II, VERTICAL LOAD WITH OUTBOARD ACTING SIDE LOAD, AFT CENTER OF GRAVITY

TEST NO. 1

Vertical Load (1)	Side Load	Vertical Deflection		Tread (2)		Maximum Strain Fwd (3)		Maximum Strain Aft (3)	
		Fwd	Aft	Fwd	Aft	Top	Bottom	Top	Bottom
(lb)	(lb)	(in)	(in)	(in)	(in)		(microinches)		
0	0	0	0	93.31	92.69	+ 2	- 4	0	- 2
4,000	0	2.96	2.79	96.63	95.87	- 3,030	+ 2,970	- 3,350	+ 3,620
6,000	0	4.51	4.25	98.18	97.37	- 4,410	+ 4,100	- 4,900	+ 5,250
8,000	0	6.19	6.07	99.75	99.06	- 5,830	+ 5,410	- 6,740	+ 7,200
10,000	0	8.33	8.42	101.62	100.94	- 7,440	+ 7,050	- 8,930	+ 9,630
12,000	0	10.43	10.56	102.81	102.62	- 8,930	+ 8,660	-11,000	+11,700
13,000	0	11.53	11.53	103.31	102.94	- 9,600	+ 9,290	-11,900	+12,600
14,000	0	12.03	12.13	103.50	103.94	-10,200	+ 9,600	-12,400	+13,200
14,000	3,500	13.30	13.38	103.56	104.13	-10,900	+10,400	-13,300	+14,200
15,000(4)	7,500	16.10	16.17	-	-	-	-	-	+15,800

(1) Vertical load shown does not include 390 pounds, the weight of the static test fixture.

(2) Dimension between inboard sides of skids.

(3) Strains measured inboard of ball joint fitting, maximum bending moment ; $E_b = 8.46 \times 10^6$ psi (aft cross member); $E_b = 6.79 \times 10^6$ (forward member).

(4) Aft cross member failed outboard of L. H. ball fitting while strain gage data were being taken.

TABLE XVII

SUMMARY OF TEST DATA, STATIC TEST CONDITION II VERTICAL LOAD WITH OUTBOARD ACTING SIDE LOAD, AFT CENTER OF GRAVITY TEST NO. 2

Vertical Load (1)	Side Load	Vertical Deflection		Tread (2)		Maximum Strain Fwd (3)		Maximum Strain Aft (3)	
		Fwd	Aft	Fwd	Aft	Top	Bottom	Top	Bottom
(lb)	(lb)	(in)	(in)	(in)	(in)		(microinches)		
0	0	0	0	92.93	93.43	- 4	0	- 4	+ 1
4,000	0	3.29	3.20	96.81	96.81	- 3,500	+ 2,900	- 3,440	+3330
8,000	0	5.97	5.87	99.44	99.13	- 5,650	+ 4,980	- 5,650	+6780
12,000	0	11.21	11.52	103.00	103.00	- 9,520	+ 8,810	-10,200	Out
14,000	0	13.32	13.64	104.06	103.75	-11,200	+10,200	-11,800	Out
14,800	0	14.29	14.59	104.38	104.19	-11,850	+10,800	-12,500	Out
14,800	2200	14.76	15.00	104.62	104.38	-12,100	+11,100	-12,700	Out
14,800	4400	15.21	15.42	104.62	104.43	-12,400	+11,400	-12,900	Out

(1) Load shown does not include 425 pounds for weight of static test fixture.

(2) Dimension between inboard side of skids.

(3) Strains measured inboard of ball joint fitting, maximum bending moment; $E = 6.79 \times 10^6$ psi (forward member); $E = 8.43 \times 10^6$ psi (aft member) .

TABLE XVIII

SUMMARY OF TEST DATA, STATIC TEST CONDITION III VERTICAL LOAD WITH AFT HORIZONTAL LOAD, AFT CENTER OF GRAVITY

Vertical Load (1)	Aft Load (2)	Vertical Deflection		Tread (3)		Maximum Strain Fwd (4)		Maximum Strain Aft (4)	
		Fwd	Aft	Fwd	Aft	Top	Bottom	Top	Bottom
(lb)	(lb)	(in)	(in)	(in)	(in)		(microinches)		
0	0	0	0	93.18	93.06	0	- 4	- 5	0
4,000	0	2.86	2.58	96.56	94.81	- 2,970	+ 2,825	- 2,820	+ 2,770
8,000	0	6.20	6.77	99.50	99.81	- 5,670	+ 5,130	- 6,390	+ 6,870
12,000	0	9.13	10.15	101.69	102.18	- 8,000	+ 7,100	- 9,090	+10,000
14,600	0	12.44	13.58	103.63	103.75	-10,500	+ 9,400	-11,900	+12,900
14,600	7,300	12.90	14.48	-	-	-11,200	+11,800	- 6,200	+16,000

(1) Load shown does not include 425 pounds for weight of static test fixture.

(2) Intended load was 7,500 pounds, but structure stopped reacting additional load at 7,300 pounds due to yielding in the skid-to-cross member saddles.

(3) Dimension between inboard side of skids.

(4) Strains measured inboard of ball joint fitting, maximum bending moment; $E = 6.79 \times 10^6$ psi (forward member); aft gear, $E = 8.43 \times 10^6$ psi (aft member).

TABLE XIX

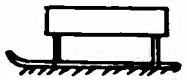
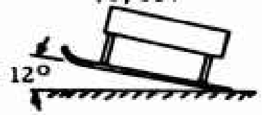
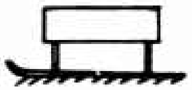
SUMMARY OF TEST DATA, STATIC TEST CONDITION IV, VERTICAL LOAD WITH FORWARD HORIZONTAL LOAD, FWD CENTER OF GRAVITY

Vertical Load (1)	Fwd Load	Vertical Deflection		Tread (2)		Maximum Strain Fwd (3)		Maximum Strain Aft (3)	
		Fwd	Aft	Fwd	Aft	Top	Bottom	Top	Bottom
(lb)	(lb)	(in)	(in)	(in)	(in)		(microinches)		
0	0	0	0	93.06	93.00	+ 3	+ 2	- 2	- 1
4,000	0	2.82	2.77	96.62	96.31	- 3,155	+ 3,020	- 2,940	+ 3,020
8,000	0	6.44	6.74	100.13	100.00	- 6,380	+ 5,860	- 5,930	Out
12,000	0	10.21	9.98	102.62	102.06	- 8,900	+ 8,500	- 8,500	Out
14,600	0	13.15	13.13	104.00	103.69	-11,000	+10,500	-10,400	Out
14,600	7,500	14.20	13.87	-	-	- 9,030	+12,100	- 9,200	Out

(1) Load shown does not include 425 pounds for weight of static test fixture.
(2) Dimension between inboard side of skids.
(3) Strains measured inboard of ball joint fittings, maximum bending moment; forward gear, $E = 6.79 \times 10^6$ psi; aft gear, $E = 8.43 \times 10^6$ psi.

TABLE XX

DROP TEST CONDITIONS - UH-1 LANDING GEAR

	V Level Landing Aft C.G.	VI Nose-Up Landing Aft C.G.	VII (2) Rolled Landing Aft C.G.	VIII (2) Overweight Level Landing, Aft C.G.
Gross Wt. (lb)	6, 600	6, 600	6, 600	7, 590
C.G.	F.S. 138.0 W.L. 60.5	F.S. 138.0 W.L. 60.5	F.S. 138.0 W.L. 60.5	F.S. 138.0 W.L. 60.5
Rotor Lift	0.67	0.67	0.67	0.67
I_{xx}(lb sec ^{2} in)	26, 716	26, 716	26, 716	27, 850
I_{yy}(lb sec ^{2} in)	110, 359	110, 359	110, 359	114, 770
I_{zz}(lb sec ^{2} in)	93, 657	96, 657	93, 657	97, 600
				
Test No.	V (fps)	V (fps)	V (fps)	V (fps)
1	2.00	4.00	4.00	6.00
2	4.00	6.00	6.00	8.00
3	5.00	8.00	8.00	
4	6.00	8.00	8.00	
5	7.00	9.84(1)	9.84	
6	8.00			
7	8.00			
8	8.50			
9	9.00			

(1) Failure of the aft cross member occurred in this test.
(2) These tests were not accomplished.

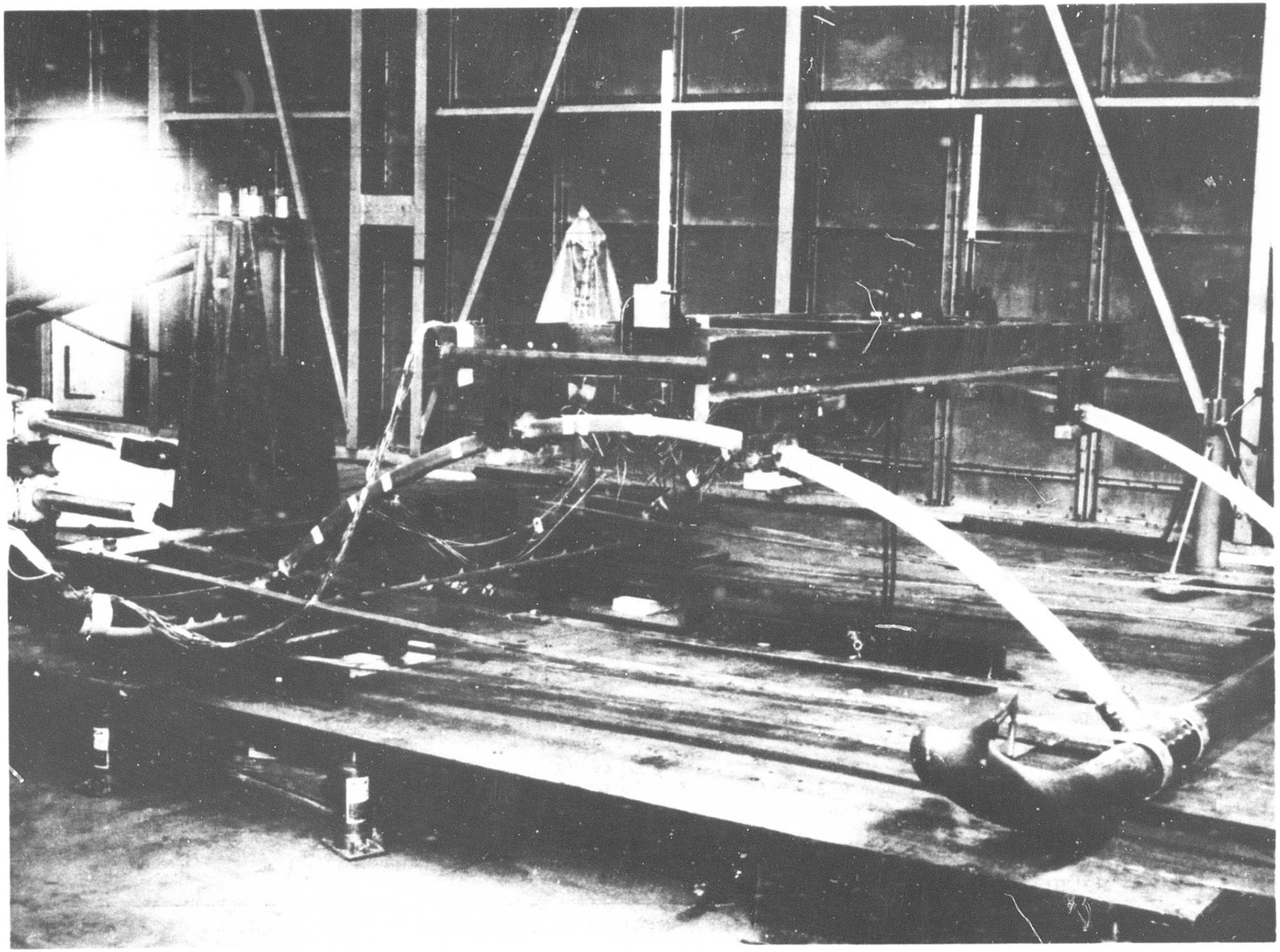


Figure 5. Landing Gear Assembly and Static Test Fixture.

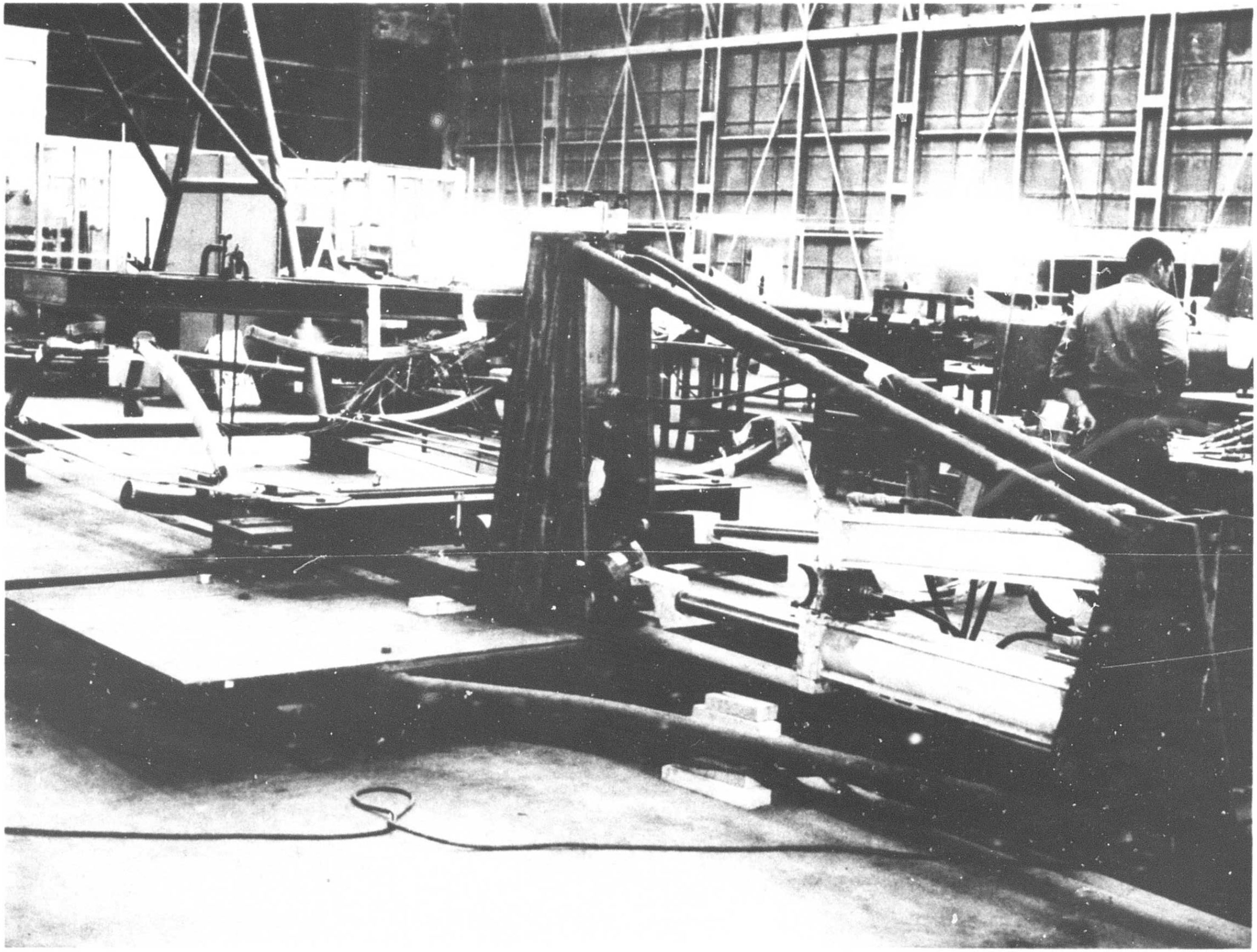


Figure 6. Static Test Fixture.

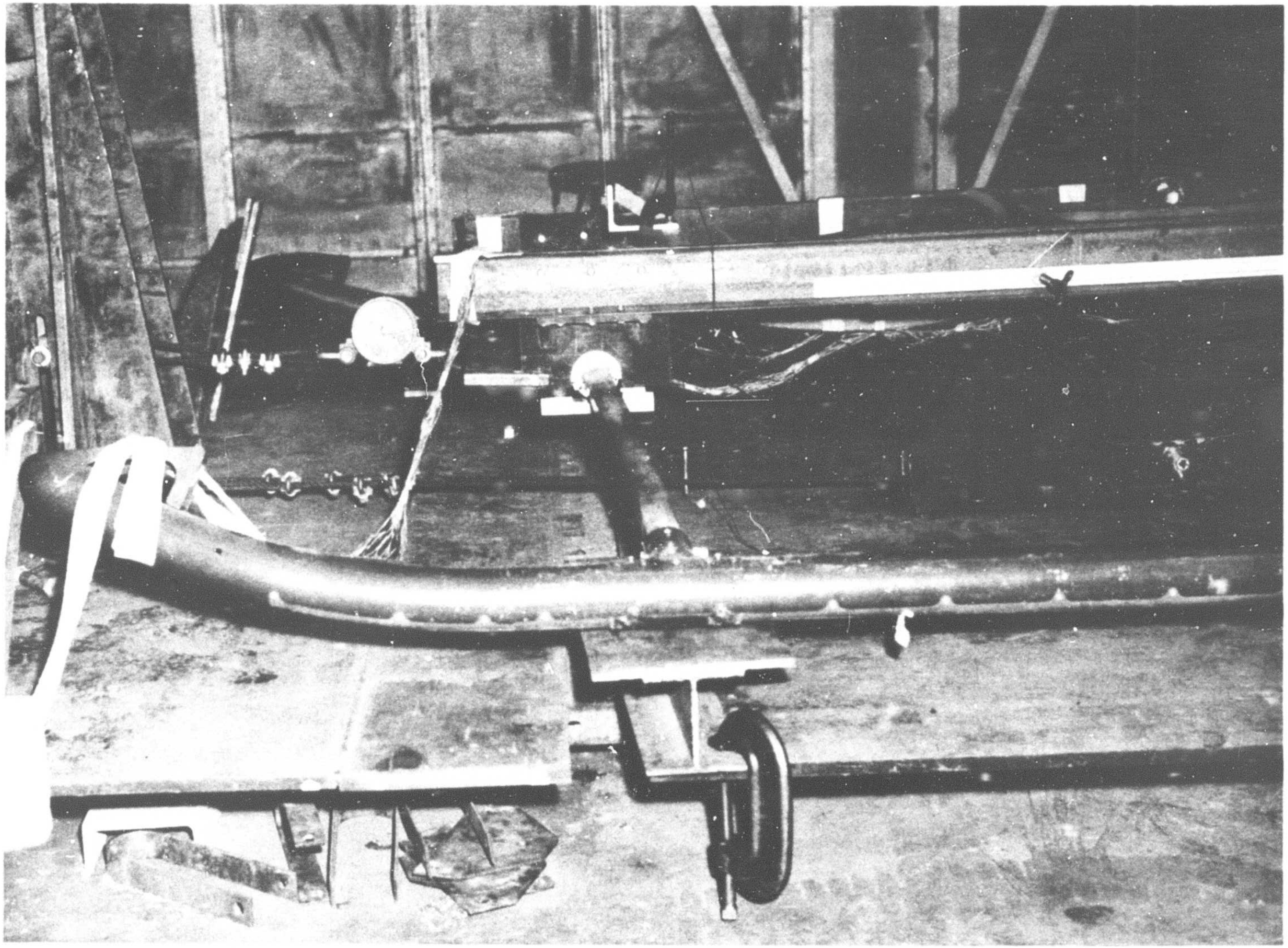


Figure 7. Landing Gear With Maximum Loads, Static Test Condition III.

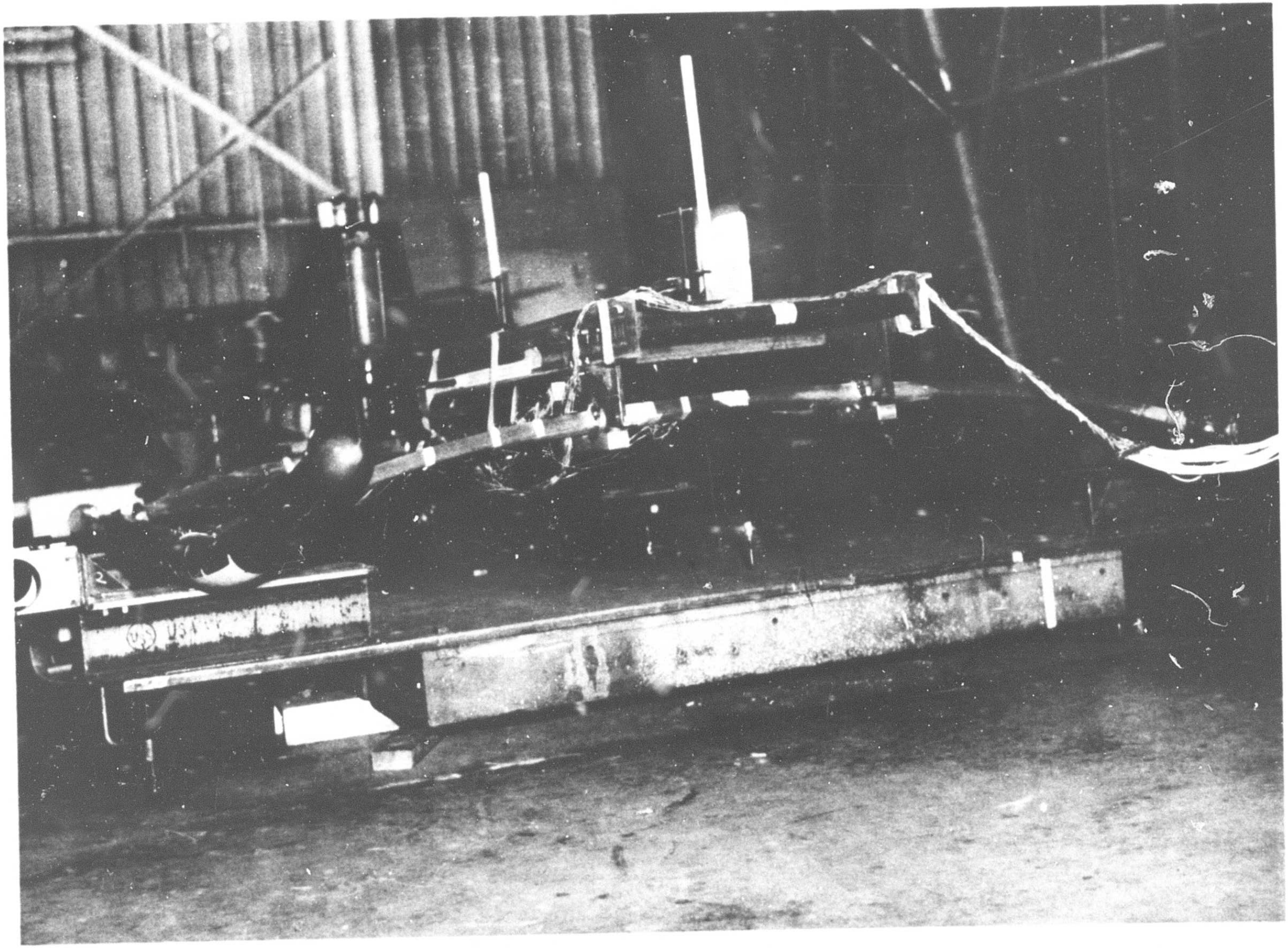


Figure 8. Landing Gear With Maximum Loads, Static Test Condition IV, Front View.

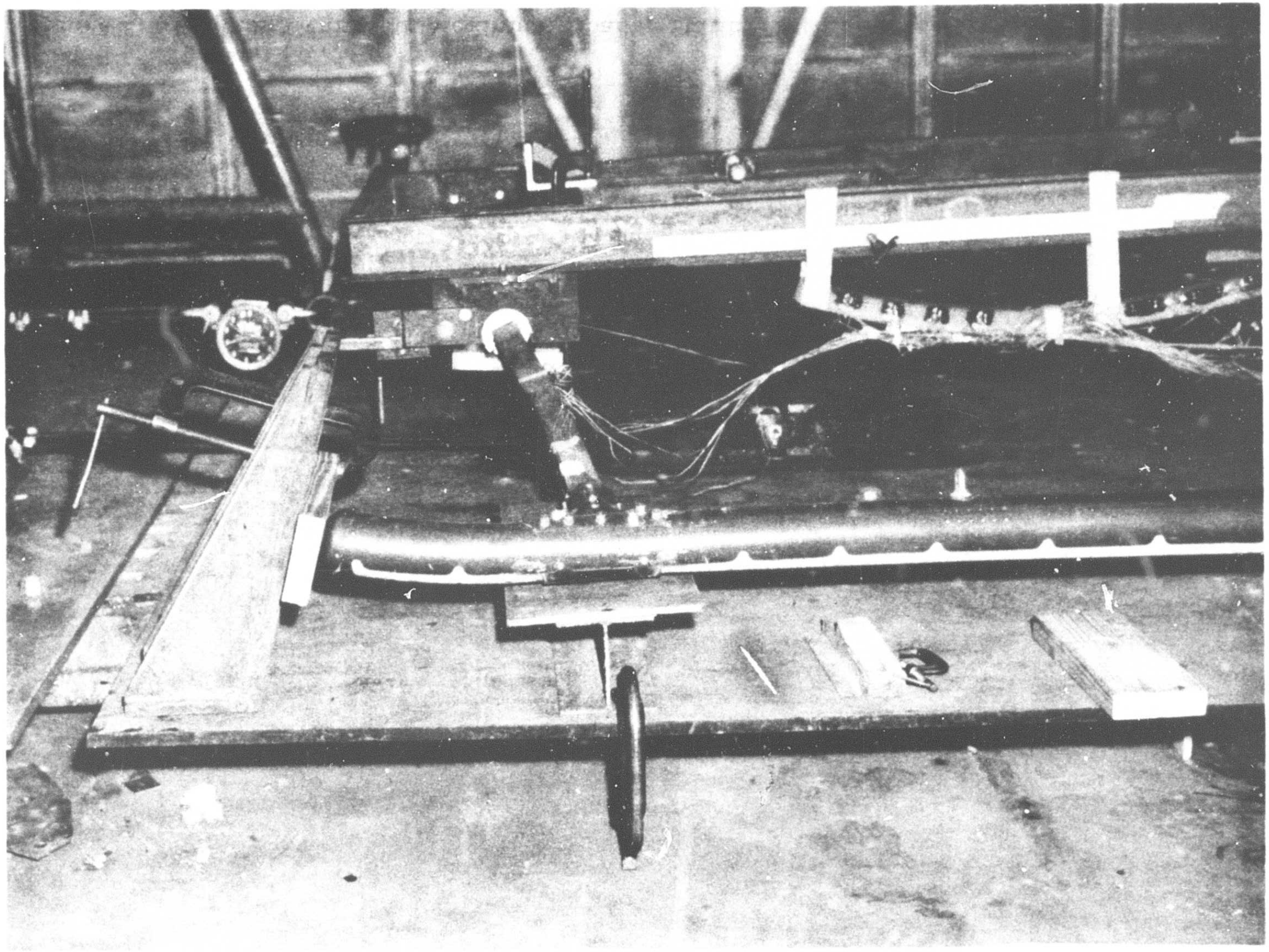


Figure 9. Landing Gear With Maximum Loads, Static Test Condition IV, Aft Cross Member.

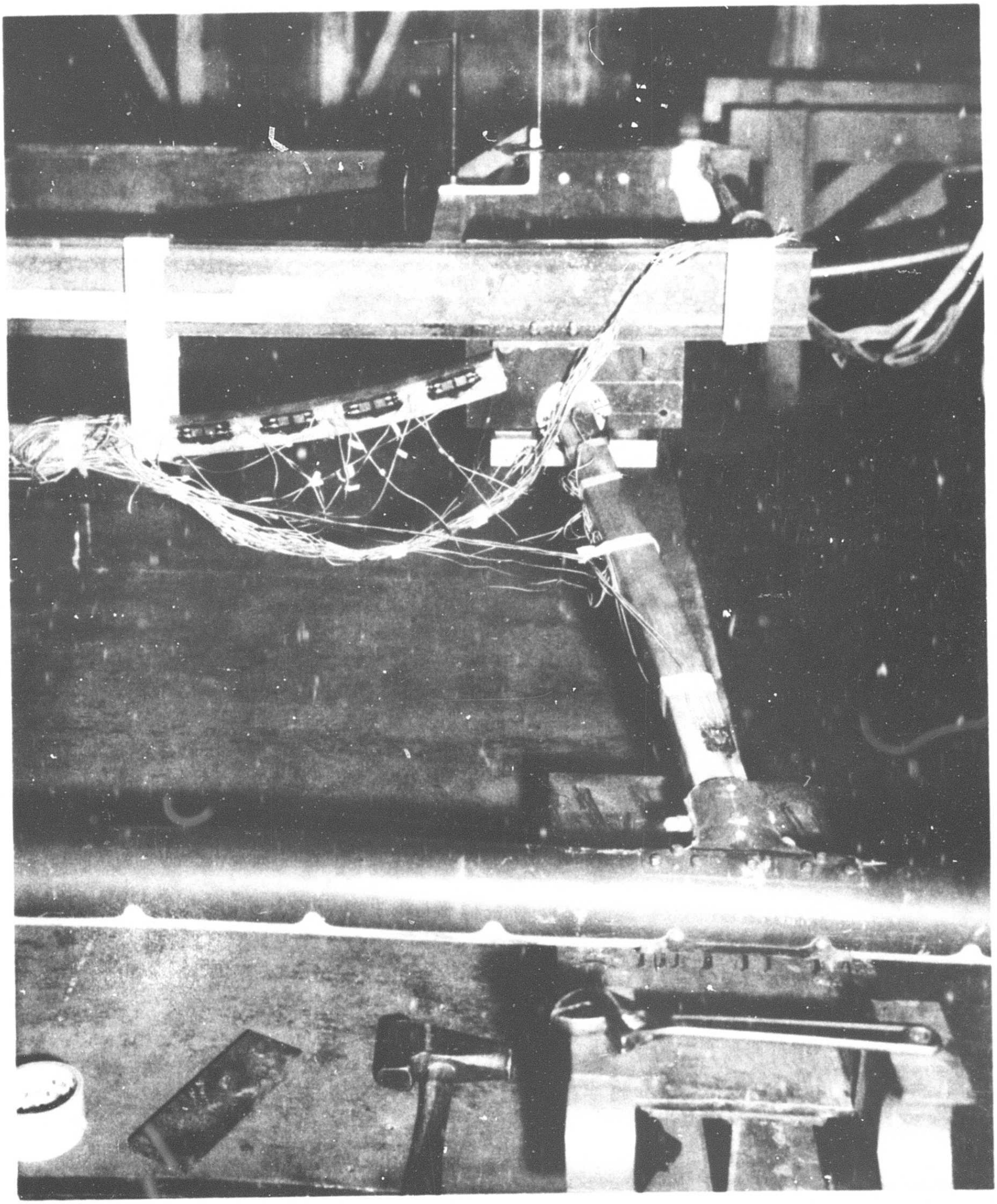


Figure 10. Landing Gear With Maximum Loads, Static Test Condition IV, Forward Cross Member.

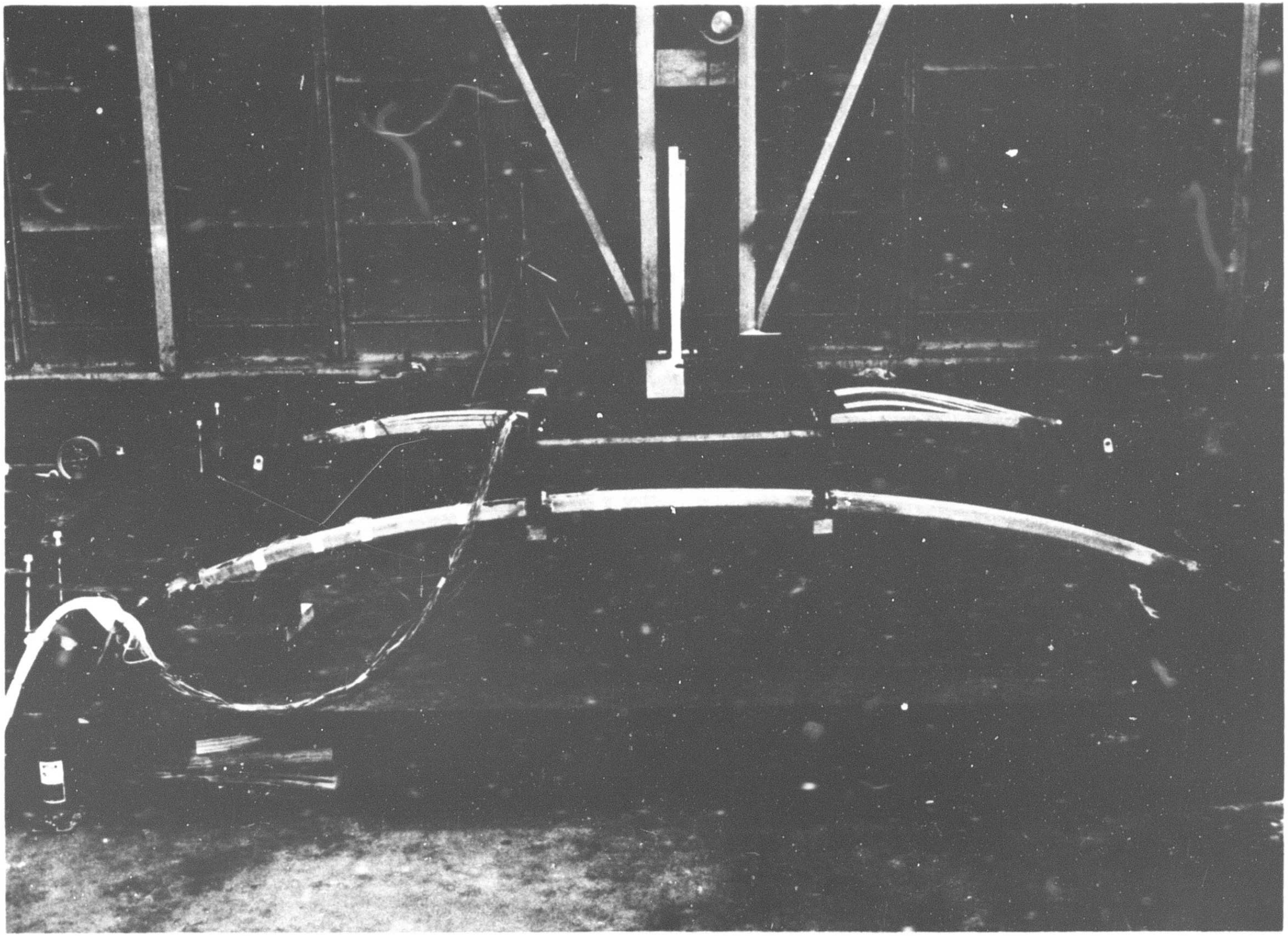


Figure 11. Static Test Failure, Condition II, Test 1 Overall View.

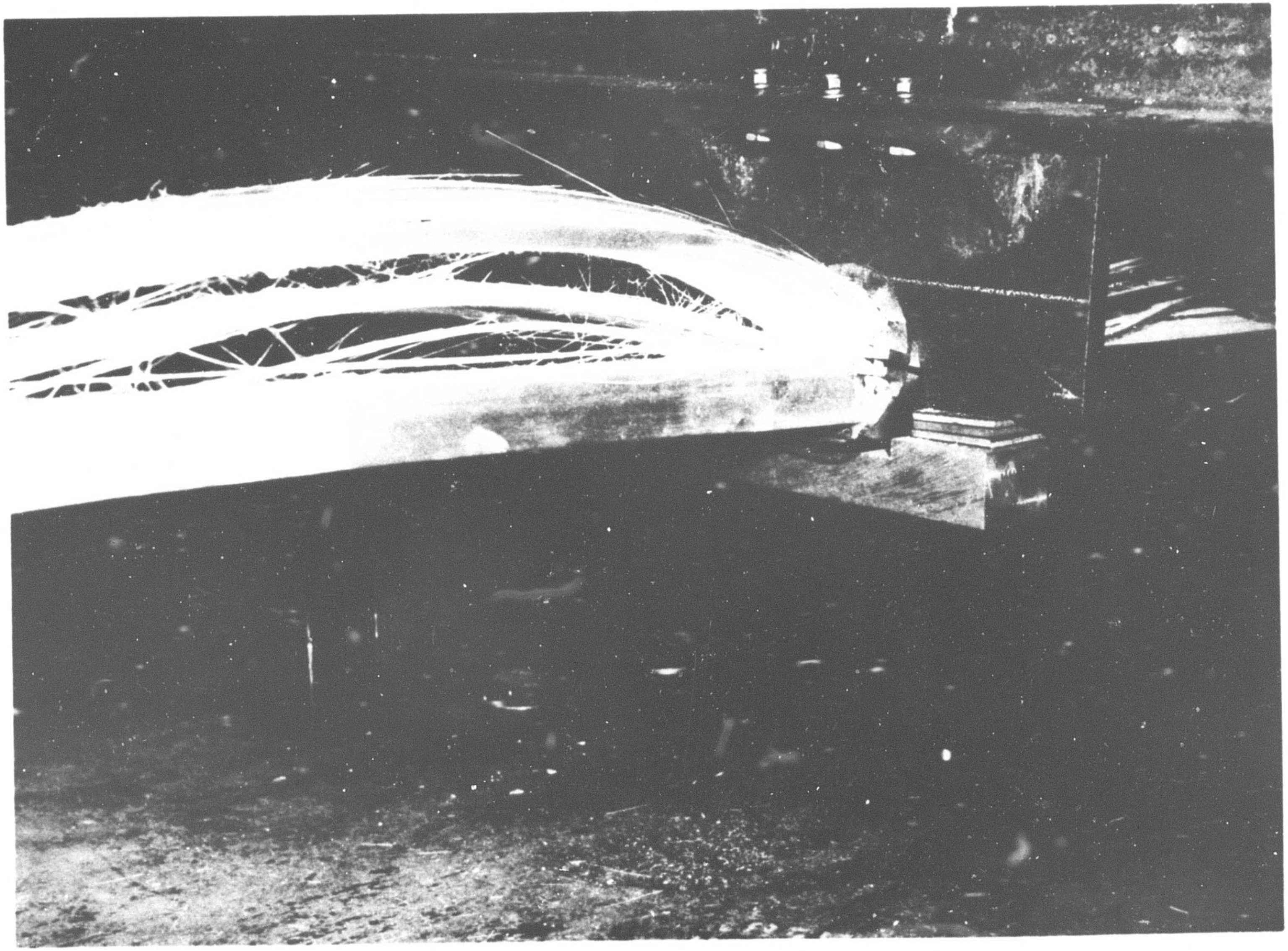


Figure 12. Static Test Failure, Condition II, Test 1 Aft Cross Member.

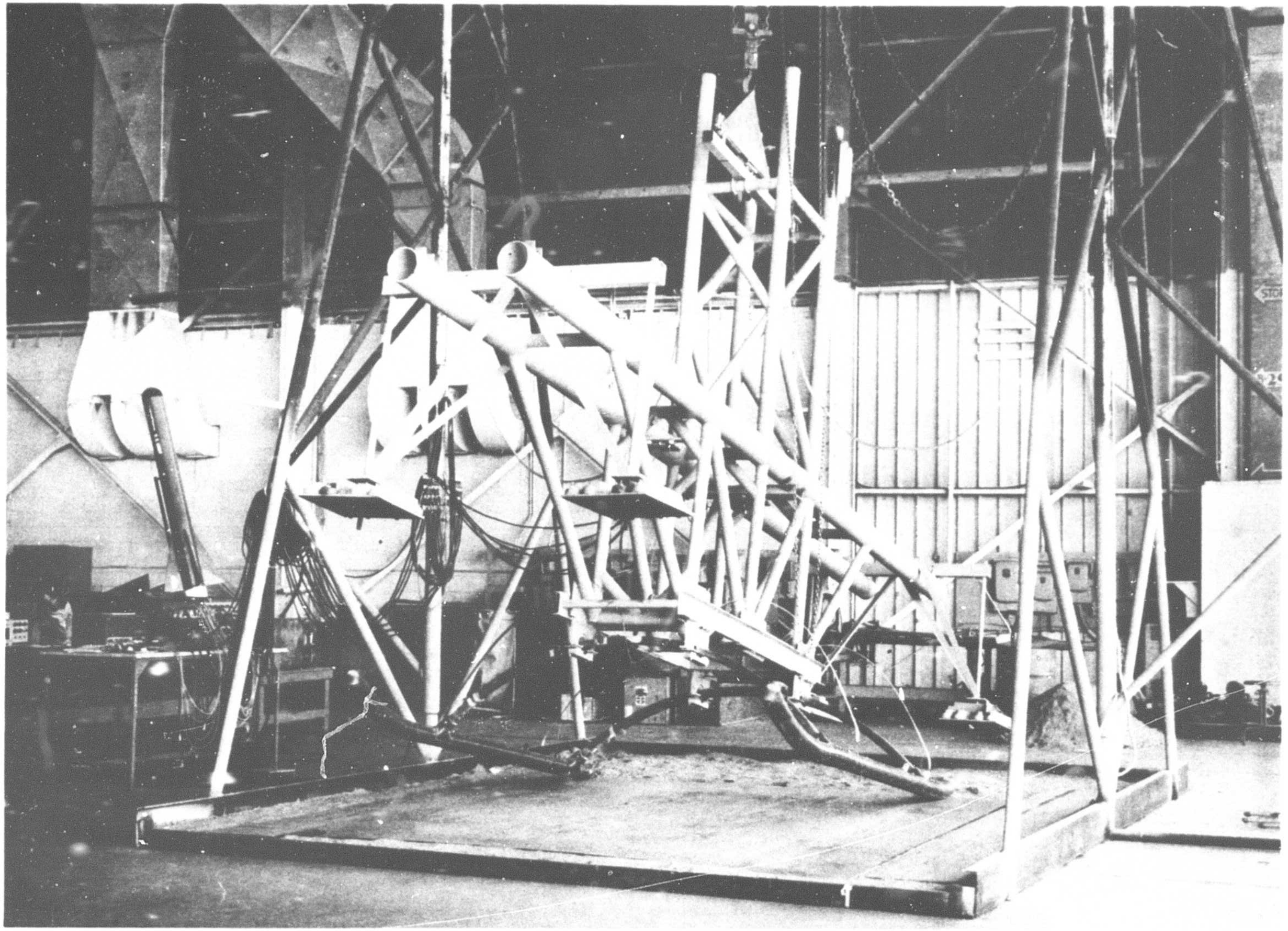


Figure 13. Drop Test Fixture.

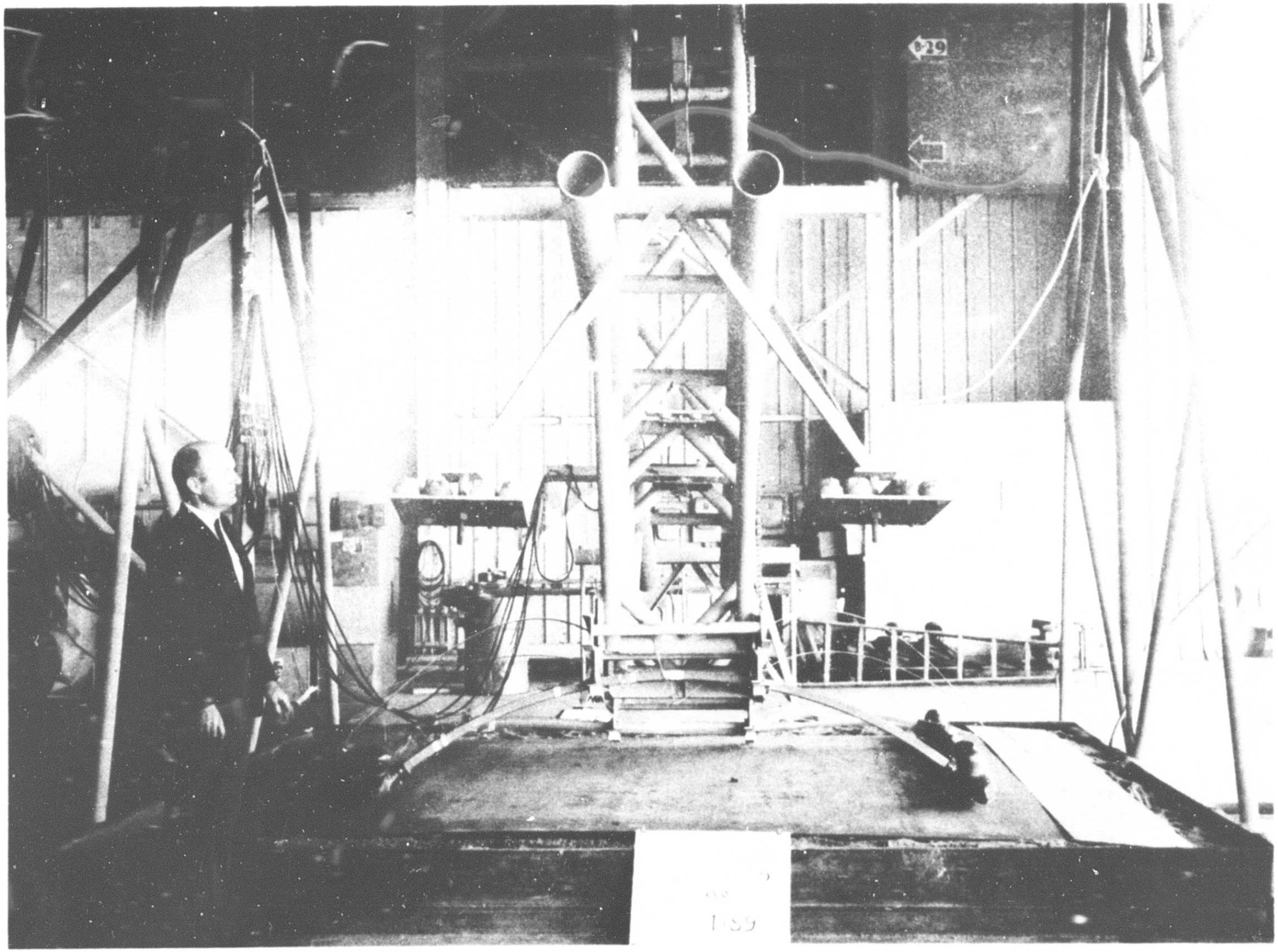


Figure 14. Landing Gear After Drop Test Failure, Condition VI, Test 5 Overall View .

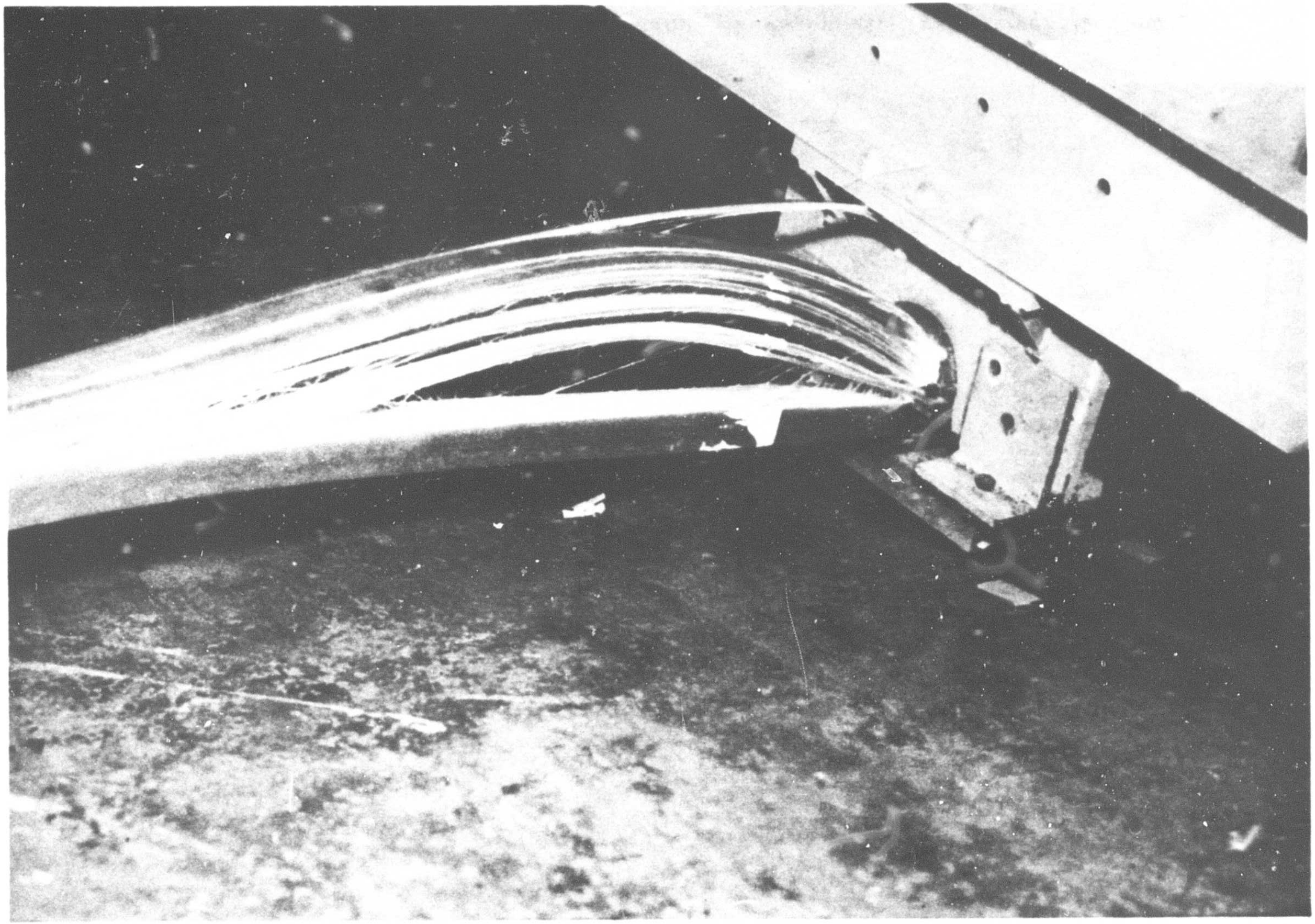
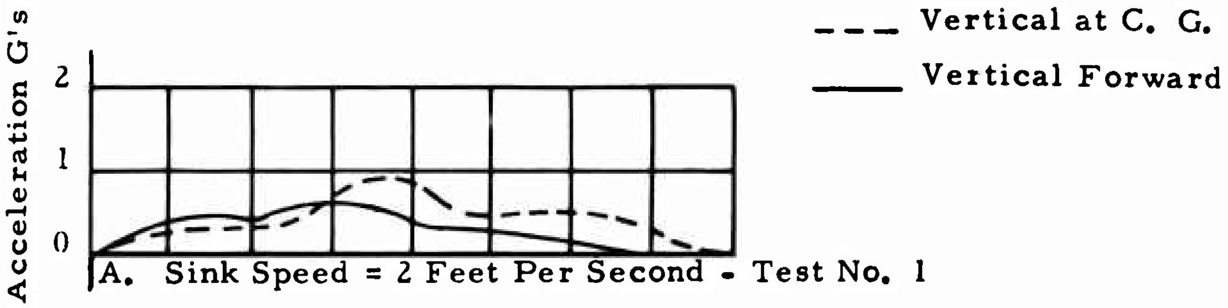
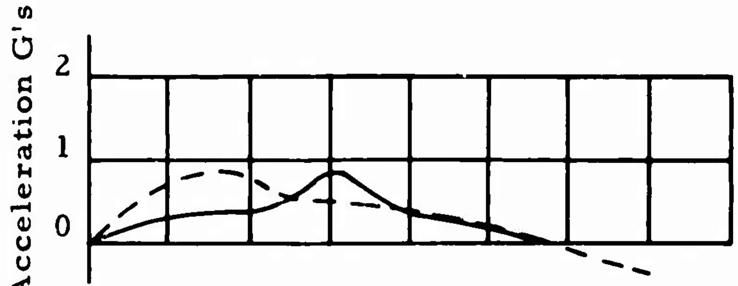


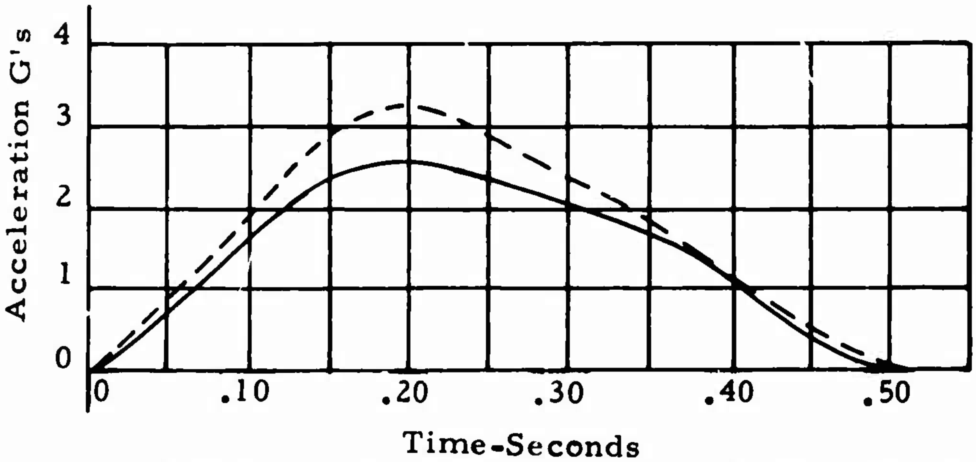
Figure 15. Aft Cross Member After Drop Test Failure, Condition VI, Test 5.



A. Sink Speed = 2 Feet Per Second - Test No. 1

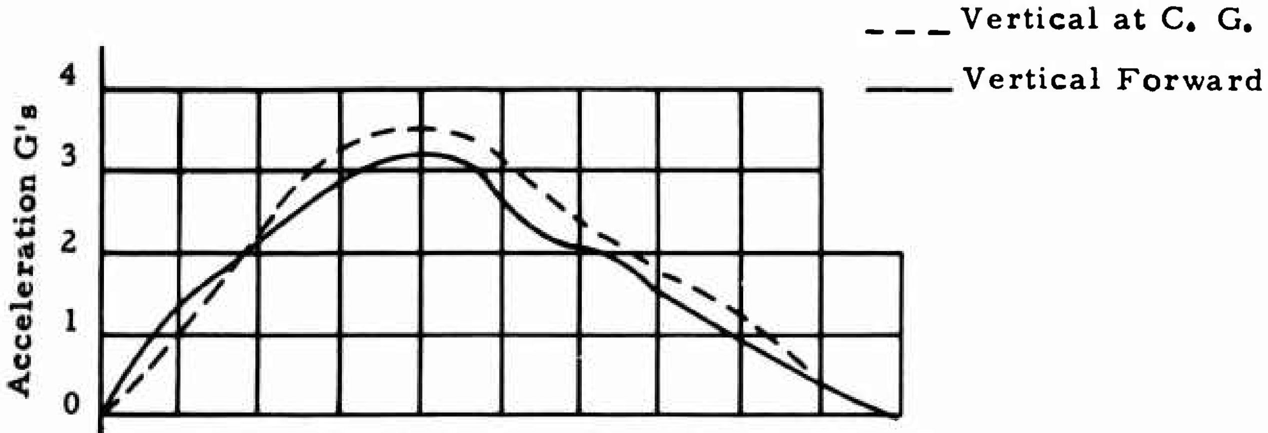


B. Sink Speed = 4 Feet Per Second - Test No. 2

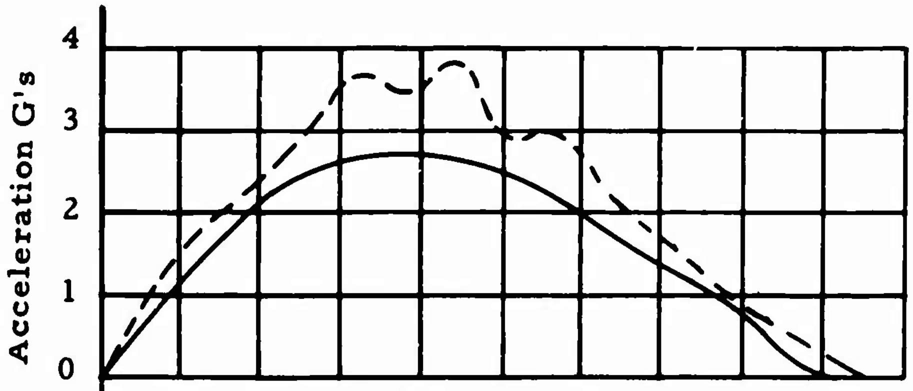


C. Sink Speed = 6 Feet Per Second - Test No. 4

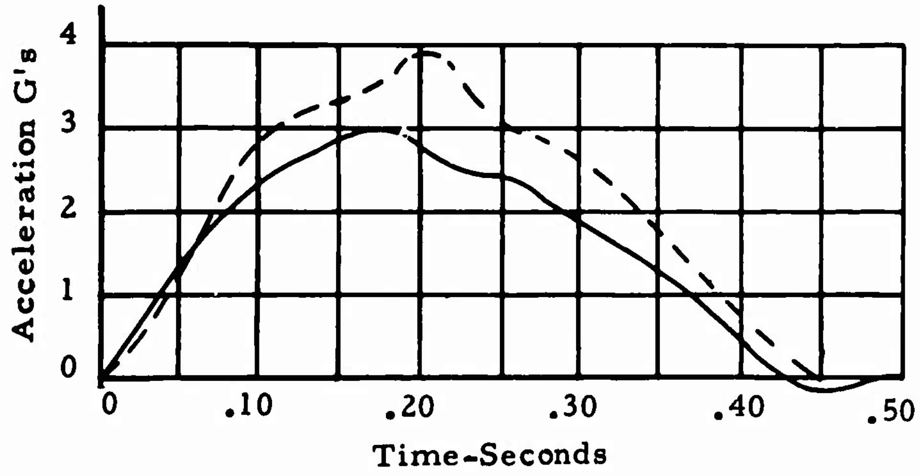
Figure 16. Acceleration Time Histories - Level Landings.



D. Sink Speed = 7 Feet Per Second - Test No. 5

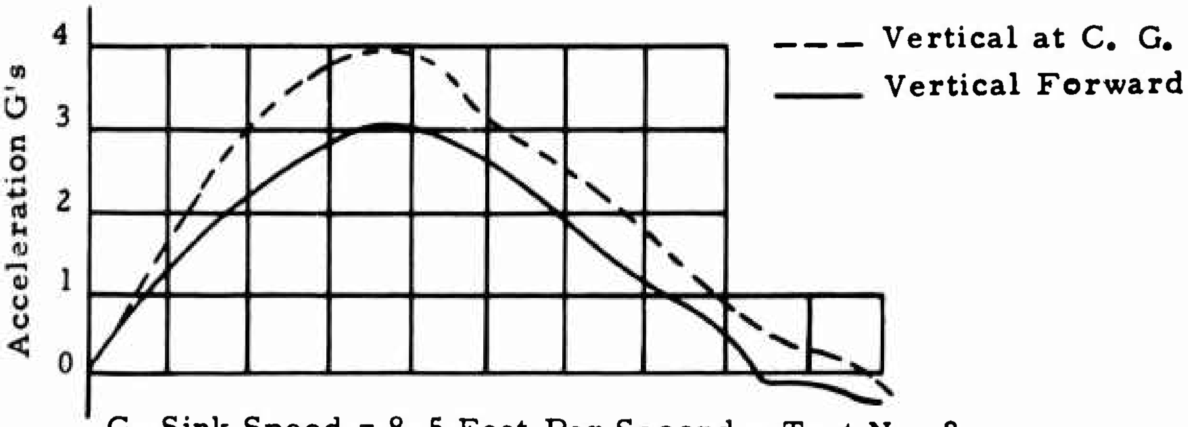


E. Sink Speed = 8 Feet Per Second - Test No. 6

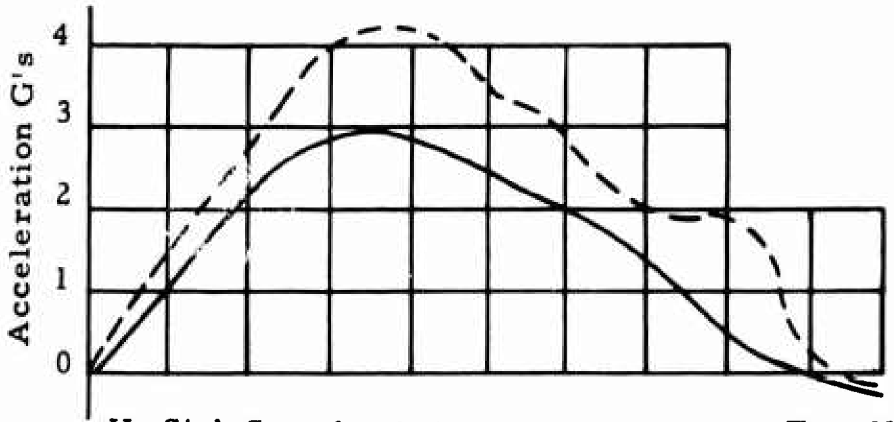


F. Sink Speed = 8 Feet Per Second - Test No. 7

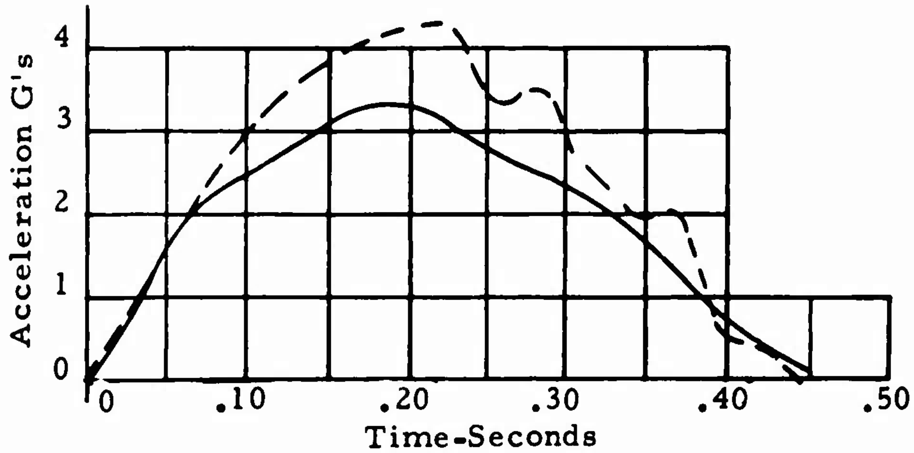
Figure 16, contd. Acceleration Time Histories - Level Landings.



G. Sink Speed = 8. 5 Feet Per Second - Test No. 8



H. Sink Speed = 9. 0 Feet Per Second - Test No. 9



I. Sink Speed = 9. 84 Feet Per Second - Test No. 10

Figure 16, contd. Acceleration Time Histories -Level Landings.

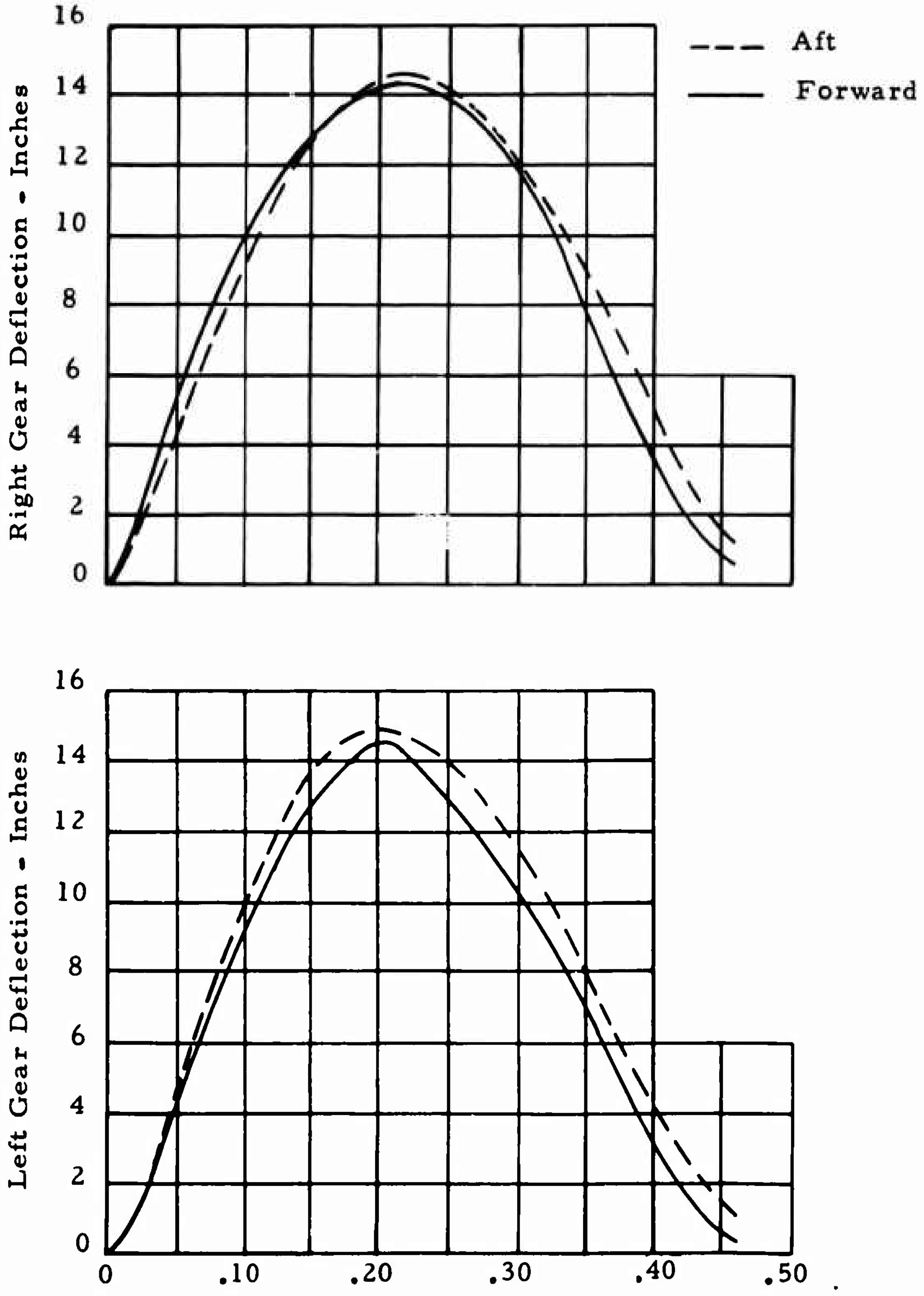
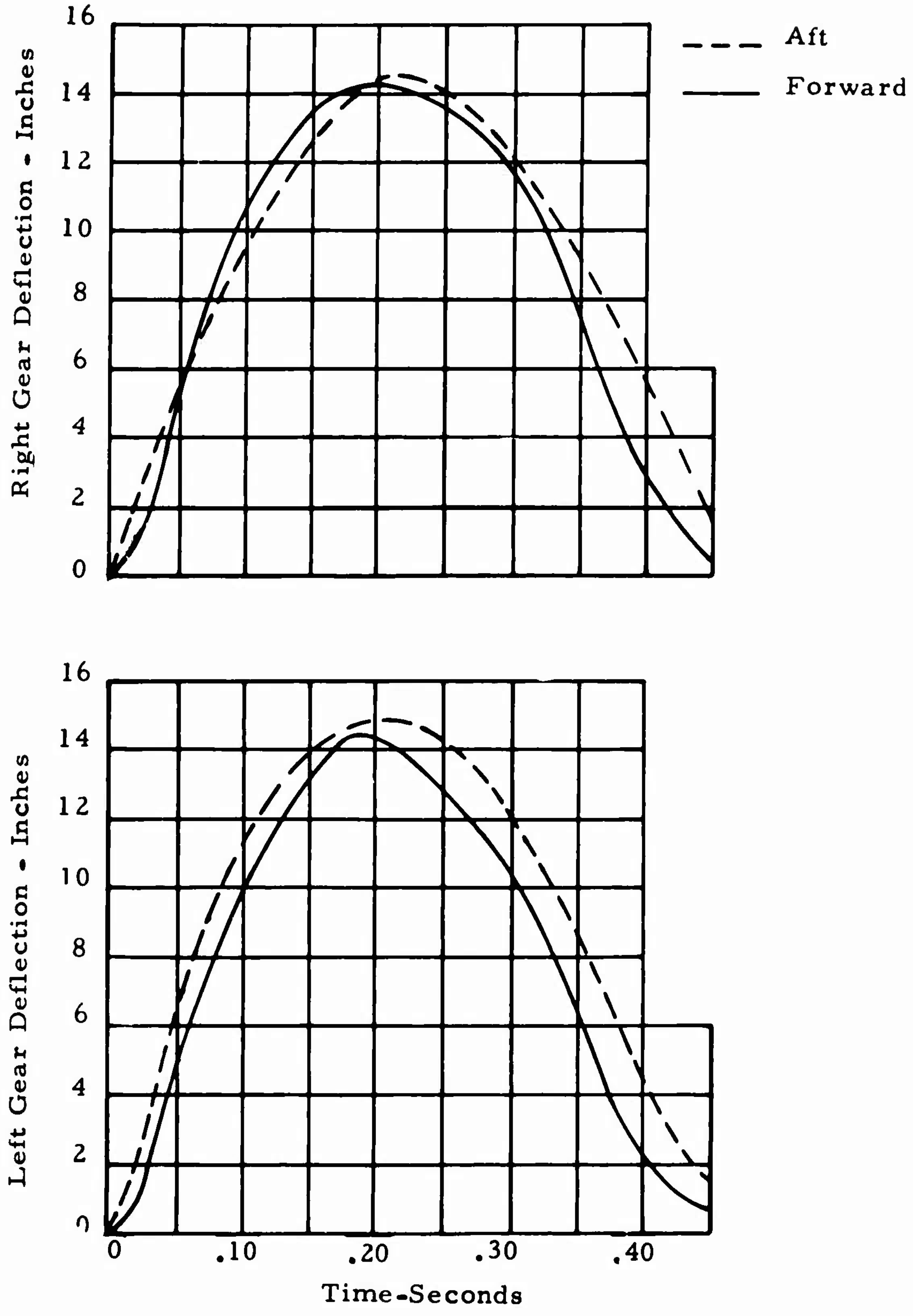
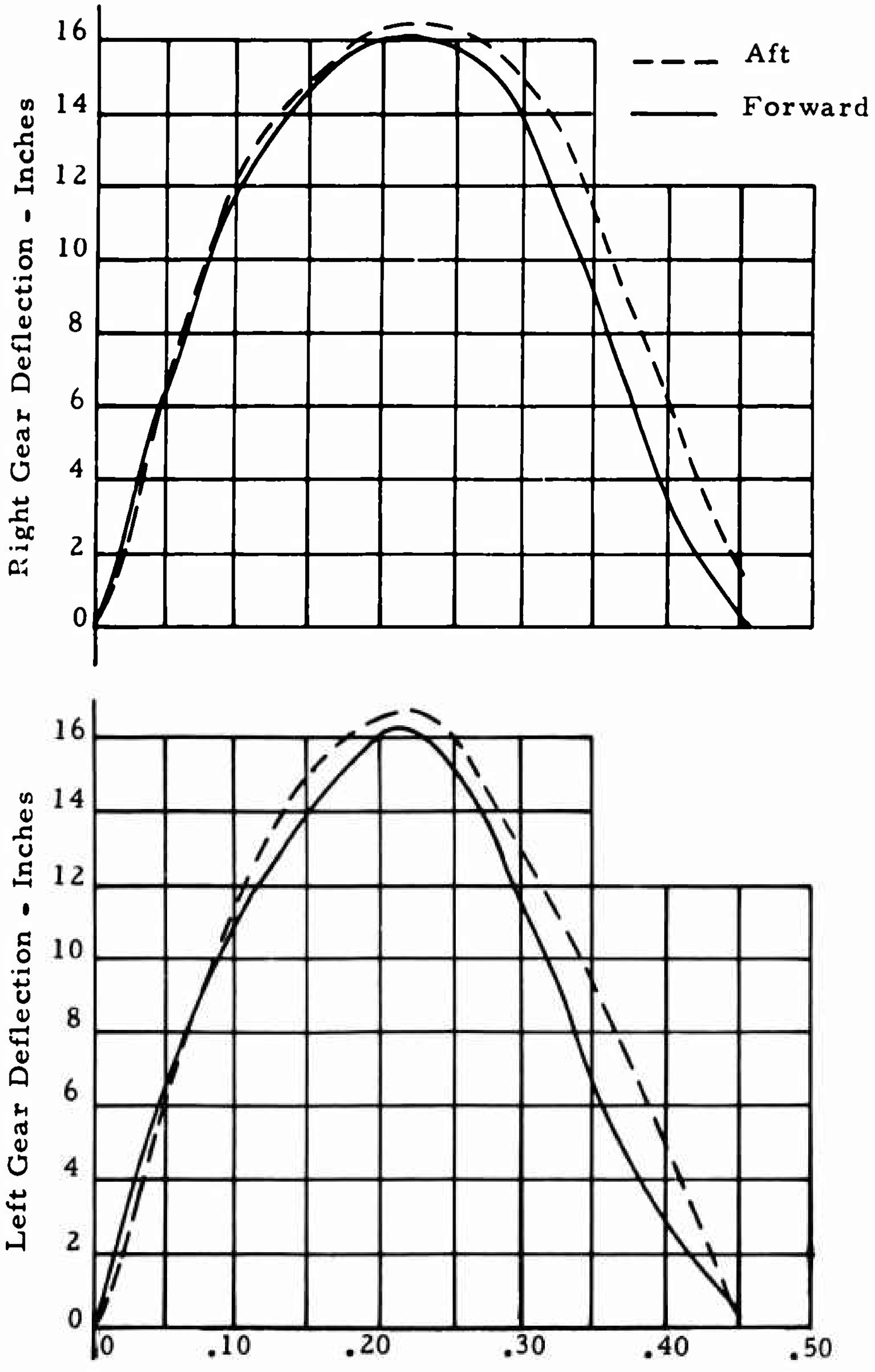


Figure 16, contd. Deflection Time Histories - Level Landings.



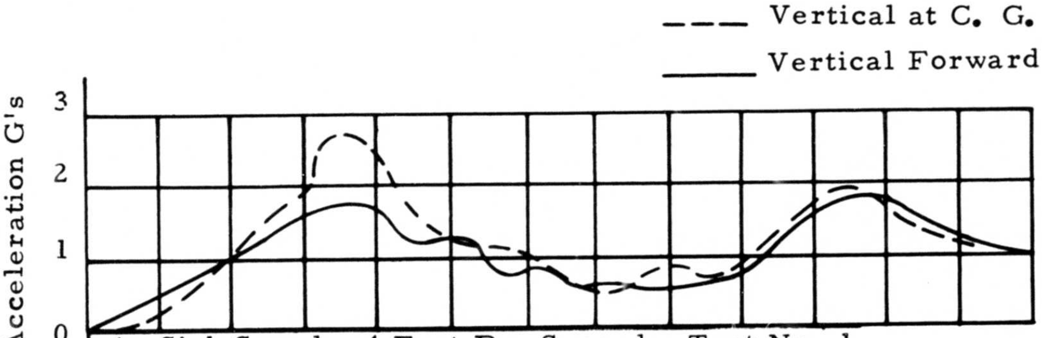
B. Sink Speed = 8 Feet Per Second - Test No. 7

Figure 16, contd. Deflection Time Histories - Level Landings.

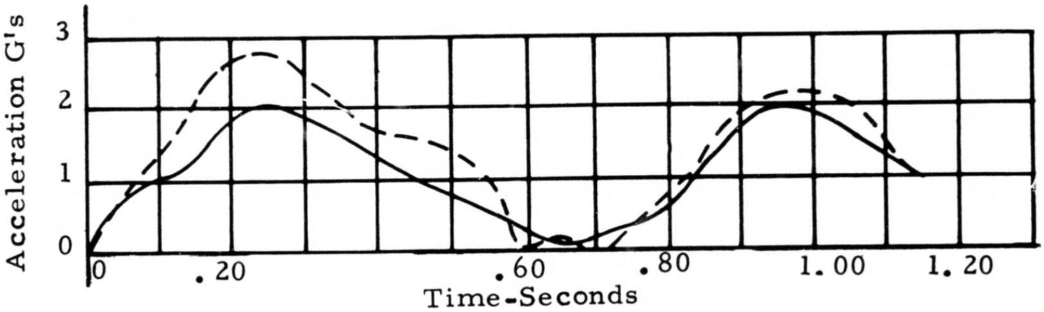


C. Sink Speed = 9.84 Feet Per Second - Test No. 10

Figure 16, contd. Deflection Time Histories - Level Landings.

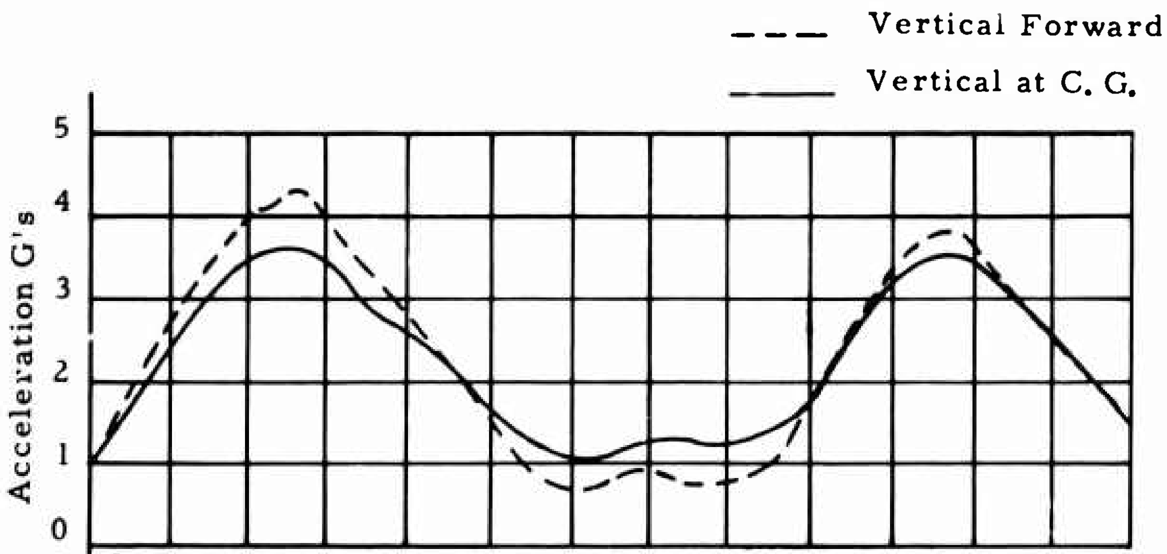


A. Sink Speed = 4 Feet Per Second - Test No. 1

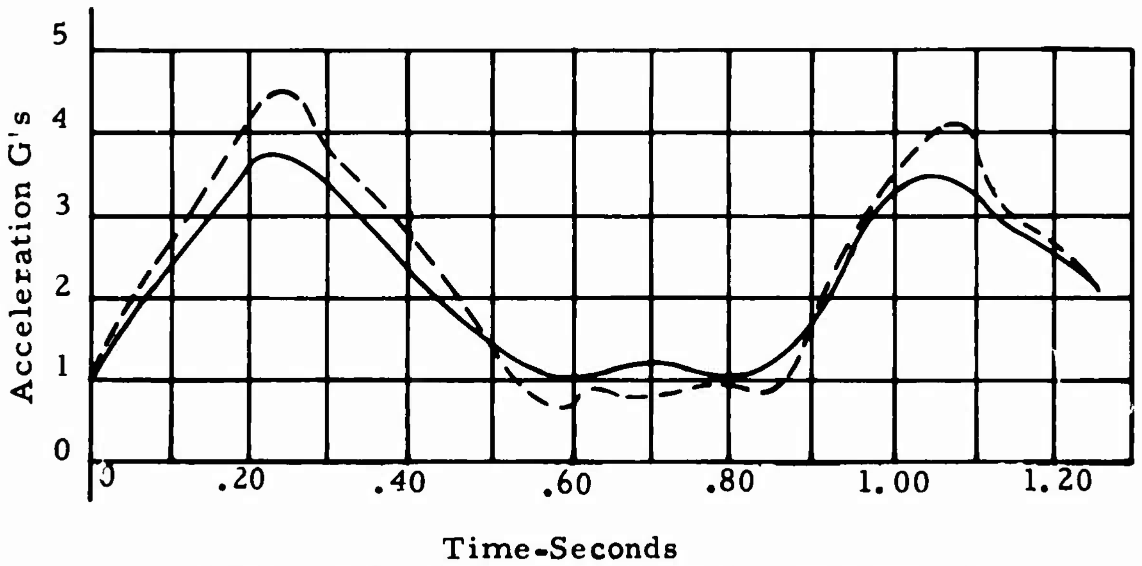


B. Sink Speed = 6 Feet Per Second - Test No. 2

Figure 17. Acceleration Time Histories - Nose Up Landings.

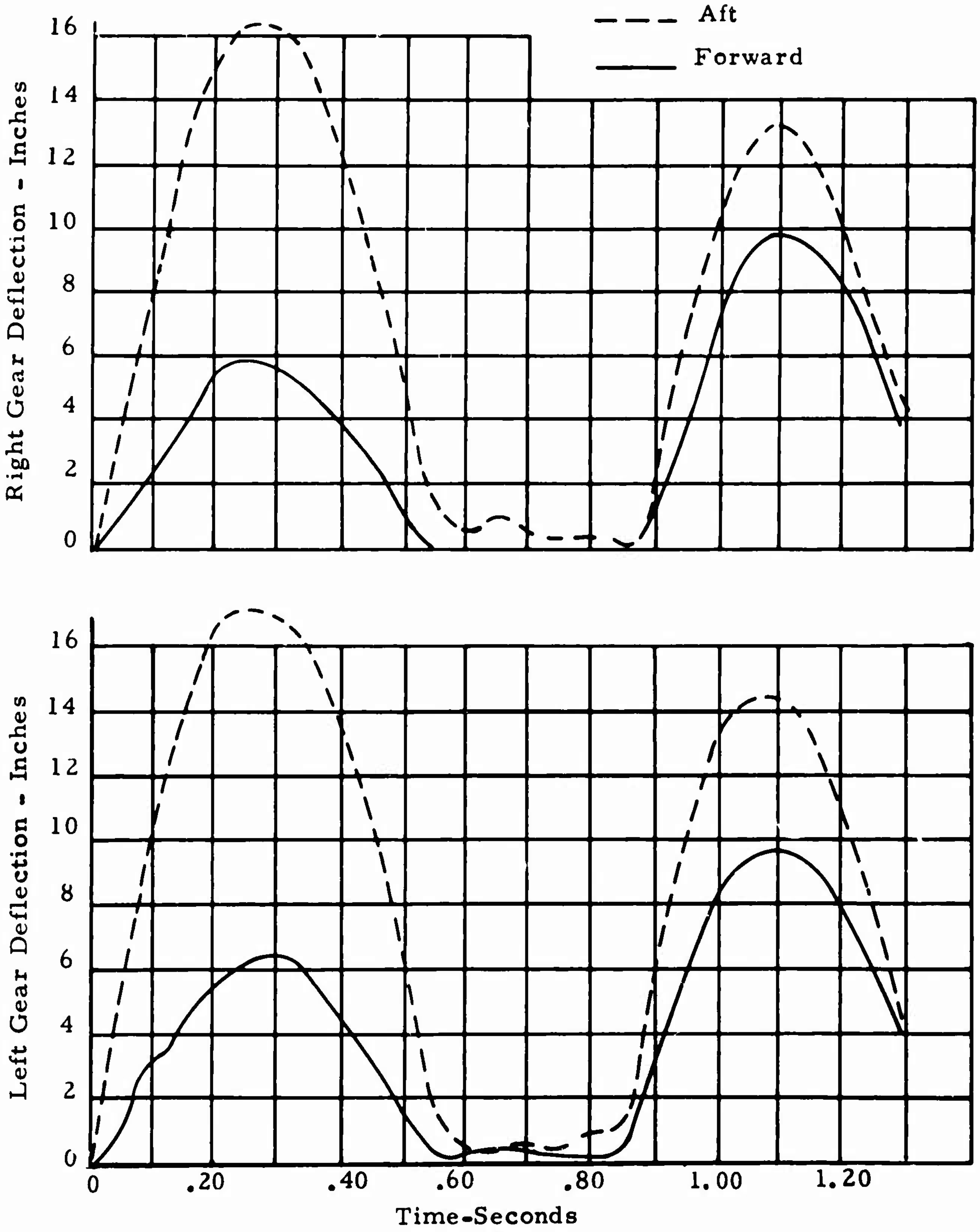


C. Sink Speed = 8 Feet Per Second - Test No. 3



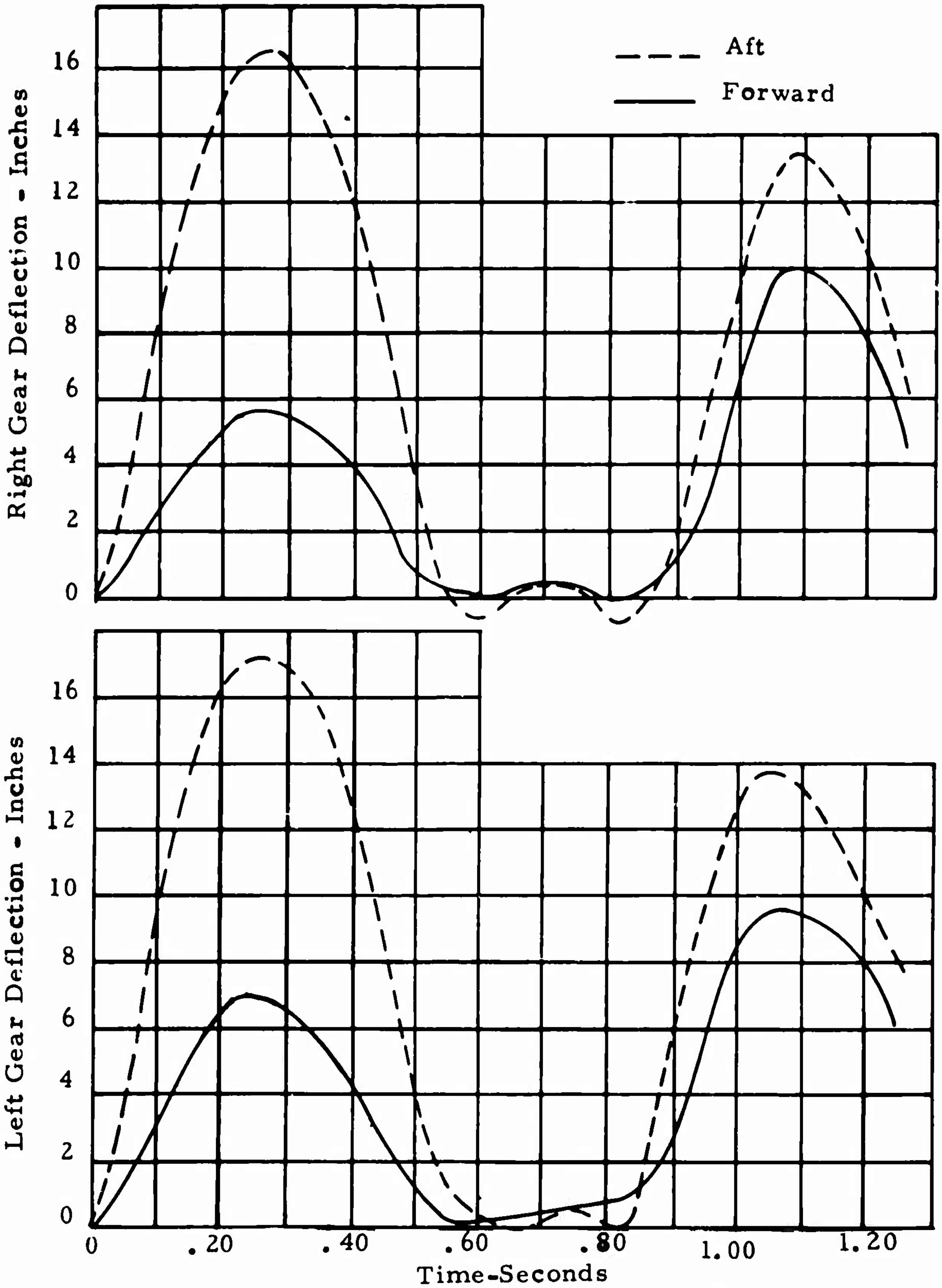
D. Sink Speed = 8 Feet Per Second - Test No. 4

Figure 17, contd. Acceleration Time Histories - Nose-Up Landings.



A. Sink Speed = 8 Feet Per Second - Test No. 3

Figure 17, contd. Deflection Time Histories - Nose Up Landings.



B. Sink Speed = 8 Feet Per Second - Test No. 4

Figure 17, contd. Deflection Time Histories - Nose Up Landings.

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13. ABSTRACT

The primary objective of this program was to determine the feasibility of using reinforced plastic material for helicopter landing gears. A comprehensive design study was made, a fiber glass reinforced plastic landing gear system for the UH-1 helicopter was fabricated, and a full scale item was subjected to static and drop tests.

Materials used for fabrication were Owens Corning S-994 HTS 901 twelve end glass roving and Union Carbide ER-2270 epoxy resin with methylnadic anhydride. Fabrication was accomplished by Cincinnati Testing Laboratories (CTL) by a wet winding process. Final design was based on the following properties: Flexural Strength = 183,000 psi; Interlaminar Shear Strength 7,000 psi; Compressive Strength = 110,000 psi; Flexural Modulus = 8 x 10^6 psi.

The static and drop tests demonstrated a high degree of ruggedness and durability. One of the two reinforced plastic members of the landing gear systems was subjected to four ultimate design static tests and fifteen drop tests without damage or failure. The other member was subjected to all tests except one of the static tests. Load factors at all impact velocities were less for the reinforced plastic landing gear than for the present metal gear. Energy dissipation occurred quite rapidly and spring back was not as severe as anticipated. It was concluded that fiber glass reinforced plastics are feasible materials for this application.

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14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Landing Gear						
Reinforced Plastics						
UH-1 Helicopter						
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