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USAAVLABS TECHNICAL REPORT 68-16

40,000-POUND-CAPACITY HEAVY-LIFT HELICOPTER EXTERNAL CARGO HANDLING SYSTEM DESIGN STUDY

By

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April 1968

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

CONTRACT DA 44-177-AMC-468(T)
AAI CORPORATION
COCKEYSVILLE, MARYLAND

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

This report, which was prepared by the AAI Corporation under the terms of Contract DA 44-177-AMC-468(T), is the result of one of three contract studies of the same problem with varying technical approaches. This study is based on a previous investigation of problems associated with the mechanics of cargo handling by aerial-crane-type aircraft (USAAVLABS Technical Report 66-63).

The object of this contractual effort was to provide a design for an external load handling system which would be responsive to the specialized requirements and problems of the proposed heavy-lift helicopter.

The design solution presented in the report is a possible approach to the heavy-lift helicopter external load handling system, provided the associated airframe-hoist system interface problems can be resolved.

The conclusions contained herein are concurred in by this Command. This does not imply the practicality or endorsement of this external load handling system concept for the proposed heavy-lift helicopter.

Project 1X130901D332
Contract DA 44-177-AMC-468(T)
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40,000-Pound-Capacity Heavy-Lift Helicopter External Cargo Handling System Design Study

Final Report

AAI Report ER-4510 K

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ABSTRACT

The purpose of this program was to design a helicopter hoist system capable of lifting 40,000 pounds. Desired features of the hoist system included a weight limitation of 4,000 pounds and the ability to provide either a 1- and 2-point load suspension system or a 1- and 4-point load suspension system.

Major areas of investigation included a study of primary tension members; the relative advantages of multi-drum and single-drum hoist systems; single wrap and overlaying of cables on hoist drums; and the desirability of including vibration isolation components between the external load and the helicopter.

Through study and tests, it was determined that it was possible to provide a combined system which permits 1-, 2-, and 4-point load suspension. This feature is achieved by use of a single cable drum winding four separate cables. Varying the method of reeving the cables permits the selection of either a 1-, 2- or 4-point load suspension system. Load attitude and cable loading may be adjusted by means of four hydraulic control cyclinders included in the system. Using a single drum with four cables eliminates the complex synchronization problems associated with a multi-drum drive system.

The developed system was designed with a working load capacity of 40,000 pounds. It has been designed structurally using a 2.5-g design factor and a 1.5 ultimate factor. The drum is designed for 150 feet of lift cable. The system is designed to lift the design load at a normal rate of 50 feet per minute. The velocity is easily attained and may be exceeded by at least 30 percent if it should be desired. Four 5/8-inch-diameter cables are used. The cable drum is approximately 30 inches in diameter and 96 inches long.

The weight estimates for the systems considered are as follows:

For the 1-, 2-, and 4-point, 150-ft lift,	4,598 lbs
For the 1-, 2-, and 4-point, 80-ft lift,	3,745 lbs
For the 1- and 2-point, 80-ft lift,	3,220 lbs

The developed system meets the basic requirements and presents considerable redundancy with respect to safety. This design offers a cargo handling system that will be safe, have adequate control, and have minimum weight; it also provides a means, in a single system, to further investigate and compare the relative merits of 1-, 2-, and 4-point load suspension systems.

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A	area, square inches
b	width of plate, inches
c	damping coefficient
DP	diametral pitch
d_r	wire rope diameter, inches
d_r'	effective diameter, inches
E	modulus of elasticity, psi
F	face width of gear tooth, inches
F_{bru}	ultimate bearing stress, psi
F_{cu}	ultimate compressive stress, psi
F_{cy}	compressive yield stress, psi
F_{su}	ultimate shear stress, psi
F_{tu}	ultimate tensile stress, psi
F_{ty}	ultimate yield stress, psi
f_{bc}	combined bending stress, psi
f_{bh}	calculated bending stress in a horizontal plane, psi
f_{br}	calculated bearing stress, psi
f_c	calculated compressive stress, psi
f_n	calculated normal stress, psi
$f_{n(max)}$	calculated maximum normal stress, psi

LIST OF SYMBOLS (Continued)

f_s	calculated shear stress, psi
$f_{s(max)}$	calculated maximum shear stress, psi
f_t	calculated tensile stress, psi
f_w	calculated shear stress in welds, psi
h	thickness of plate, inches
I	moment of inertia of plane areas, $in.^4$
K	spring stiffness of hoist
k	kip, 1000 pounds
K_f	shaft key force, pounds
K_T	load on bolts due to torsional moment, pounds
L_B	static tooth load, pounds
l	length, inches
M	bending moment, inch-pounds
M_{max}	maximum moment, inch-pounds
M_H	weight of helicopter
M_h	bending moment in a horizontal plane, inch-pounds
M_L	weight of load
M_V	bending moment in a vertical plane, inch-pounds
M.S.	margin of safety
P	design load, pounds
P_b	load per bolt, pounds

P_R	resultant load, pounds
P_ω	force amplitude at frequency ω
P.D.	gear pitch diameter, inches
Q	direct load per bolt, pounds
R	beam reaction, pounds
r_i	inside radius, inches
r_m	mean radius, inches
r_o	outside radius, inches
S	section modulus, in.^{3}
T	torsional moment, inch-pounds
T_c	tension load in cable
T_d	maximum torque in system, inch-pounds
T_S	torque in drive shaft, inch-pounds
t	wall thickness, inches
y	gear form factor
τ_w	maximum shear stress in rubber, psi

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PROGRAM OBJECTIVES

This program was divided into two phases. Phase I was devoted to the study of the various possible hoisting systems, the relative merits of the components of these systems, and their effect on systems control, weight, safety and reliability. In the second phase of this program, a design configuration was developed based on the Phase I study and incorporating the most desirable features of the systems considered.

Performance and design objectives were included as a part of the contractual requirements for this program. These objectives are listed below to indicate the basic guideline used in the study and in the proposed helicopter hoist system. Figure 1 is included to illustrate 1-, 2- and 4-point load suspensions.

DESIGN OBJECTIVES

The heavy-lift helicopter external load-handling system shall be responsive to the design objectives enumerated below. These objectives are listed in order of decreasing priority to assist the resolution of possible design conflicts.

1. Every effort shall be made to minimize total system weight. A maximum target weight of 4,000 pounds for the total system is recommended.

2. A high degree of reliability is required for the system since malfunction may result in mission abortion and deadlining of the aircraft. Incorporation of system redundancies and the use of low mortality, standard components will assist in the achievement of the reliability goal.

3. The elimination of potential safety hazards in all operational phases, including load acquisition, flight, and load release. Of particular importance are those potential in-flight hazards adversely affecting the stability and control of the aircraft.

4. Efforts shall be made to maximize the maintainability of the proposed system. The following are some of the techniques to be effectively utilized:

   a. Convenient access shall be provided to components and parts that require periodic or frequent replacement or that may, by their nature or location, be subject to damage or corrosion.

   b. Maximum accessibility shall be provided to components and parts requiring frequent access for field adjustments, rigging and checking.

   c. The required preventive and in-storage maintenance procedures shall be minimized.

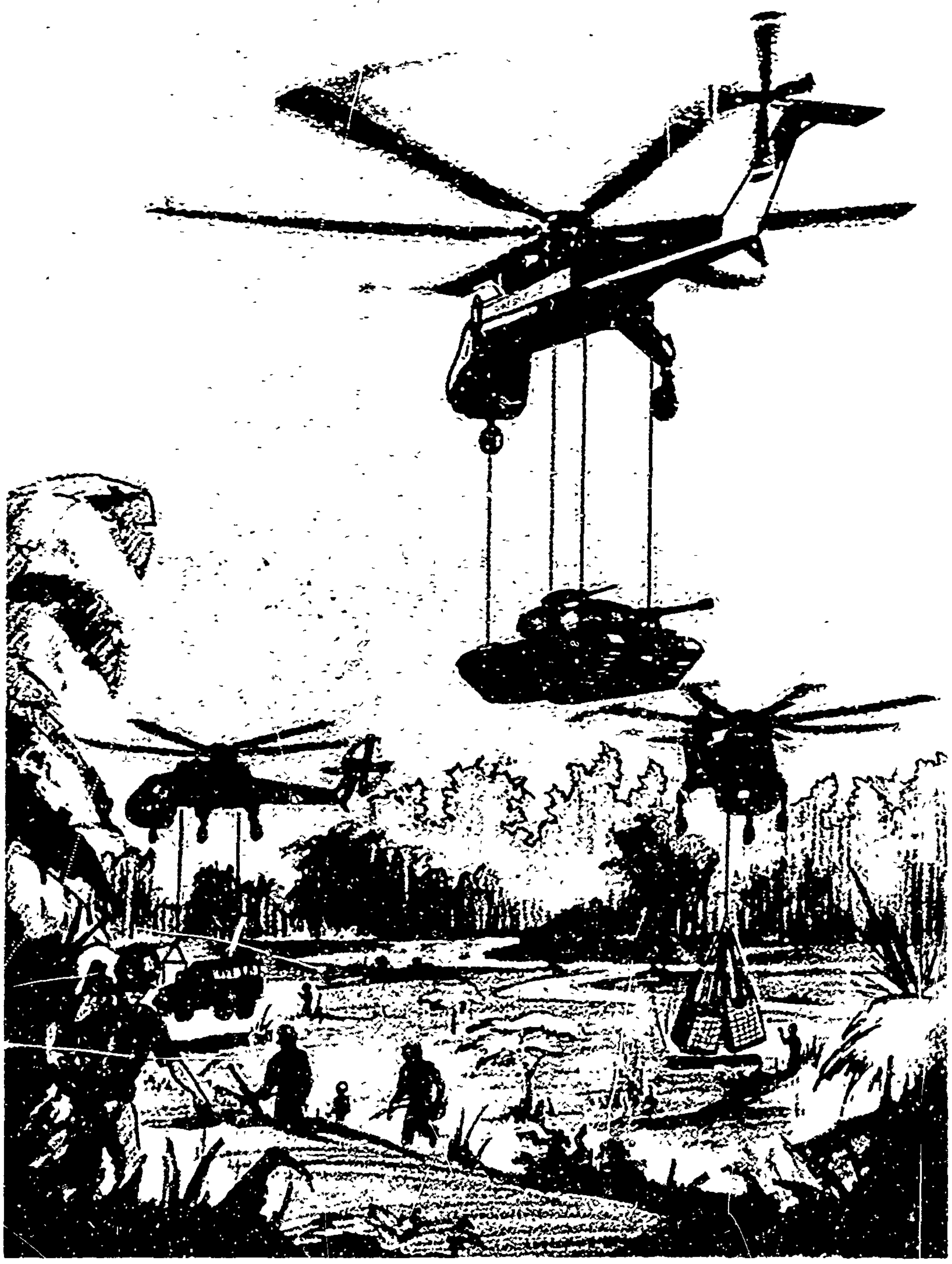


Figure 1. Illustration of 1-, 2-, and 4-Point Load Suspensions

d. Standard parts and components shall be used when possible.

e. Assemblies, components and parts shall be interchangeable and replaceable to the maximum extent possible.

f. Quick disconnects shall be provided for electical connection and hydraulic line couplings at all dynamic components.

g. Control rods, hydraulic lines and the like shall be positioned so that they are not exposed to damage by personnel moving in and about the helicopter during servicing and maintenance. Access and clearances shall be sufficient to permit maintenance and adjustment using standard tools.

h. The system shall incorporate a minimum number of dynamic components. Essential dynamic components shall be of the simplest possible design with a minimum maintenance and service life of 1200 cycles before removal or overhaul and 3600 cycles before retirement.

i. Dynamic components shall be protected from adverse effects of dust, sand, moisture, and the like, encountered in operation from unprepared areas.

j. A simple in-flight check of the status of each hoist mode shall be provided.

5. The size of the system and its components shall be kept as small as practical.

6. Power requirements shall be held to a minimum.

7. Total system cost shall be minimized.

PERFORMANCE OBJECTIVES

1. Both single-point and multi-point suspension modes shall:

a. Have a vertical lift capacity of not less than 40,000 pounds with a design load factor of 2.5 g's and a safety factor of 1.5 ultimate.

b. Minimize unsafe oscillations of a suspended load.

c. Be capable of load acquisition with manual ground assistance from a hover or landed position.

d. Facilitate rapid hook-up of loads. A 2-minute maximum hook-up time for previously rigged loads is allowed.

e. Provide for the positive locking of personnel pod loads relative to airframe with the cargo-handling system power off.

f. Provide for two methods of cockpit-controlled load release: electrical and mechanical.

g. Provide for manual release at load attachment points when the load is on the ground.

h. Incorporate provisions for automatic holding of suspended loads with the cargo-handling system power off.

i. Incorporate provisions for load jettisoning independent of the releases required by subparagraph f and the holding provisions required by paragraph h above.

j. Provide for attenuating shock forces imposed on suspension system.

k. Minimize cable backlash.

l. Provide a means for attaching suspension modes directly to lift points on items of cargo or through web slings without damage either to the attaching device or to the item engaged.

m. Be designed so that major components of each mode may be quickly detached and removed from the helicopter. The removal of components which function only during one mode of operation shall not negate the use or the functioning of the remaining mode of operation. Time for removal or installation of components shall not exceed one hour, including necessary checkouts. This removal or installation is to be accomplished by organizational level maintenance personnel using tools available in the Standard Aviation Mechanics Tool Kit.

n. Be capable of direct hook-up to two control stations permanently mounted in the cockpit, each of which shall have the capability for actuation and monitorship of the system under load and no-load conditions. Actuation shall include controls for raising, lowering, releasing, and jettisoning. Monitorship shall include measuring and displaying cable lengths, forces in each leg, and such other monitoring data as may be required for safe load-handling operations.

2. The single-point suspension mode shall also:

a. Provide for engaging and hoisting a maximum load while the aircraft is hovering above the load. Two maximum heights, 150 feet and 80 feet above the load, shall be considered.

b. Have a hoist speed rate with load of not less than 60 feet per minute with acceleration and deceleration rates which provide smooth hoisting and lowering operations.

c. Be usable for towing large ground vehicles.

3. The multi-point suspension mode shall also:

a. Provide for engaging and hoisting a maximum load while the aircraft is hovering at heights up to 50 feet above the load.

b. Have a normal hoist speed rate with load of not less than 30 feet per minute.

c. Be controllable collectively (simultaneously) from each control station.

d. Provide for synchronized load release to assure simultaneous load release at all suspension points.

e. Have a remote, plug-in type control station for use by a dismounted load master to actuate the multi-point mode (raising and lowering of hoist and opening and closing of load attachment points).

PHASE I DESIGN STUDY

This project was begun with a cursory review of existing equipment which might be used to accomplish the objectives described in the specification. The review indicated that major components capable of accomplishing the assigned tasks did not exist because of deficiencies such as excessive weight, insufficient capacity, or inflexibility of the components.

The next step taken was a detailed review of primary hoist components. These components were surveyed from the load end of the system, in order, toward the power source. A review of the availability of hooks having the capacity and characteristics required revealed that companies having both experience and products in this area are in the process of producing designs adequate for the capacity and performance required.

TENSION MEMBER ANALYSIS

The use of different tension members was then analyzed with respect to their potential adaptability. The work consisted of an investigation of the relative worth of 14 different tension members described by Table I, entitled "Hoisting Members".

Item 4, Flat Wire Rope was rejected because of its extreme width and because sufficient basic data are not yet available upon which firm designs can be based.

Items 5, 6, 7 and 8, all of which are chains, were too heavy to permit their use within the weight limitations imposed by the specifications.

Further considerations of Item 10, Stainless Steel Strap, was stopped because of the difficulties which would be encountered in handling a steel strap having these dimensions and because of the probability of easy damage from nicks and scratches.

Items 11 and 13, which are ropes, present difficult problems of storage. In both of these cases, a capstan would be required. It is further noted that the problem of unwinding the tension member was severe.

Only the major shortcomings have been referred to herein. Other minor difficulties not discussed herein contributed to the elimination of these items from further study.

Further consideration was then given to Items 1, 2, 3, 9, 12, and 14, in conjunction with various hoisting systems which were under consideration. The remaining members were then renumbered as they are listed on Table II and Table III. The charts were divided into "1-Point" and "4-Point" groups in order to permit a better analysis of their worth as they applied to either the 150-foot 1-point system or the 150-foot 2-point system. The use of the 1-point, 80-foot system received little consideration since effects appeared to be direct in terms of reduction in size and weight.

TABLE I. HOISTING MEMBERS

	TYPE	SIZE	WEIGHT PER FT. (LBS)	WEIGHT PER 150 FT.	CAPSTAN DRUM	STORAGE	EMERGENCY SHEAR CAPABILITY	RESONANCE	RESISTANCE TO CONTAMINATION	REMOTE CONTROL CAPABILITY	STANDARD ITEM	DURABILITY	HANDLING COMPATIBILITY	EASE OF MAINTENANCE
1 a.	ROUND WIRE ROPE (with Capstan)	1-1/4" DIA	2.89	434	15-3/4" DIA x 10" WIDE	DRUM 25-1/2"DIA x 18" WIDE 3 LAYERS, 11 REVS	EXCELLENT	UNKNOWN	FAIR	INTERNAL	YES	GOOD	POOR	SPLICE
1 b.	ROUND WIRE ROPE (without Capstan)	- -	- -	- -	NONE	- -	- -		- -	CONDUCTOR	- -	POOR TO FAIR		
2	SWAGED ROUND WIRE ROPE (without Capstan)	1-1/8" DIA	2.89	434	NONE	DRUM 33-1/2" DIA x 11" WIDE 4 LAYERS, 7 REVS.	EXCELLENT	UNKNOWN	FAIR (Better than unswaged)	SEPARATE CONDUCTOR	YES	FAIR		REPLACE
3	FLAT WIRE ROPE	9/16" x 5-1/4"	2.83	425	NONE	DRUM 42" DIA x 7" WIDE 37 LAYERS, 37 REVS.	FAIR	UNKNOWN	FAIR	INTERNAL CONDUCTOR	YES (NOT STOCK ITEM)	FAIR	POOR	REWEAVE RESEW
4	FLAT WIRE ROPE	.06" x 15"	1.155	174	NONE	DRUM 15" DIA x 17" WIDE	POOR	UNKNOWN	GOOD	SEPARATE CONDUCTOR	NO	GOOD	FAIR	
5	LINK CHAIN	7/8" DIA LINKS: 2-7/8" x 4-1/4"	8.5	1275	12" DIA x 4-1/2" WIDE (POCKET SHEAVE)	CHAIN LOCKER 28" x 28" x 28"	BAD	GOOD	EXCELLENT	SEPARATE CONDUCTOR	YES	GOOD	POOR	NEW LINK
6	ROLLER CHAIN (SINGLE STRAND) (RC-240)	3" PITCH LINKS: 2-11/16" x 4" WIDE	16.7	2505	10.649" P.D. 12.018" O.D. x 4"	RACK 15" x 44" x 70-1/2"	BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES (NOT STOCK ITEM)	POOR	POOR	NEW LINK
7	ROLLER CHAIN MULTIPLE STRAND	1-3/4" PITCH LINKS: 1-5/8" x 5-1/16" WIDE	15.0	2250	5.663" P.D. 6.437" O.D. x 6"	RACK 15" x 44" x 70-1/2"	BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES (NOT STOCK ITEM)	POOR	POOR	NEW LINK
8	LEAF CHAIN	1.625" x 3.813"	11.72	1760	DIFFICULT EXPENSIVE, 10" DIA x 6"	DRUM 64" DIA x 6" WIDE	BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES	POOR	POOR	NEW LINK
9	WIRE MESH	11/16" x 11-1/8"	3.647	547	NONE	DRUM 45" DIA x 13-1/2 WIDE 23 LAYERS, 23 REVS.	POOR	GOOD	EXCELLENT	MOLDED IN CONDUCTOR	YES & NO	EXCELLENT	GOOD	REPLACE
10	STAINLESS STEEL STRAP	15" x .050	2.54	382	NONE	DRUM 15" DIA x 18" WIDE 45 LAYER, 45 REVS.	GOOD		EXCELLENT	SEPARATE CONDUCTOR	YES	POOR	BAD	REPLACE REWORK
11 a.	ROUND NYLON ROPE	2-5/8" DIA	1.867	280	15-3/4" I.D. 23-1/2" O.D. x 17-1/2"	DRUM 34" DIA x 31" 3 LAYERS, 10 REVS.	EXCELLENT	UNKNOWN	GOOD	SEPARATE CONDUCTOR	YES	GOOD	EXCELLENT	SPLICE
11 b.	"				NONE	"								
12	FLAT NYLON WEBBING	.344 x 8"	1.1	165	NONE	DRUM 36" DIA x 9-3/4" 33 LAYERS, 33 REVS.	GOOD	UNKNOWN	GOOD	SEPARATE CONDUCTOR	YES	GOOD	EXCELLENT	REPLACE
13 a.	MANILA ROPE	4-3/4" DIA	5.55	833		DRUM 42" DIA x 42" WIDE	GOOD		GOOD	EXTERNAL CONDUCTOR	YES	GOOD	POOR	SPLICE
13 b.	"				NONE				(H_2O ABS)					
14	STAINLESS ROUND WIRE ROPE	CONSIDER WITH ITEM 1 WITH LOWER ALLOWABLE.												

Approximation of the size and type of device used for handling of the reeve end of the member

Approximation of the size and type of device used for handling of the reeve end of the member.

Ease with which tension member may be severed.

Natural resonant frequency to be below 2 cps.

Radio control can be used in all cases for low cost to weight and volume.

Basic member is obtainable on open market.

Heavy or large flat objects; dangerous.

Large flat sheets in airstream.

Refers to C.G. shift because of storage location causing a shift in location of C.G. of hoist and member.

How bad is stretch member for flight problems? Is it good for multiple hoisting? Is it good for estimating C.G.?

Refers to vertical smoothness of load when being hoisted. Chordal effect of some members causes oscillating motion.

A

TABLE I. HOISTING MEMBERS

EMERGENCY SHEAR CAPABILITY	RESONANCE	RESISTANCE TO CONTAMINATION	REMOTE CONTROL CAPABILITY	STANDARD ITEM	DURABILITY	HANDLING COMPATIBILITY	EASE OF MAINTENANCE	AERODYNAMICS	C.G. CONTROL	STRETCH 1 TO 4	POWER REQD. 1 TO 4	SMOOTHNESS	SPEED LIMITATION	INSPECTION RELIABILITY	LOAD LAND ORIENTATION	BACKLASH 1 TO 4	INJURIOUS NORMAL CONTAMINANTS	KINKING KNOTTING
EXCELLENT / -	UNKNOWN	FAIR / -	INTERNAL / CONDUCTOR	YES / -	GOOD / POOR TO FAIR	POOR	SPLICE	EXCELLENT	POOR / GOOD	2	3	3	NONE	2	FAIR	4	H_2O NaCl SAND	1
EXCELLENT	UNKNOWN	FAIR (Better than unswaged)	SEPARATE CONDUCTOR	YES	FAIR		REPLACE	EXCELLENT	POOR	1	4	3	NONE	2	GOOD	4	H_2O NaCl	1
FAIR	UNKNOWN	FAIR	INTERNAL CONDUCTOR	YES (NOT STOCK ITEM)	FAIR	POOR	REWEAVE RESIN	GOOD	GOOD	3	2	2	NONE	1	FAIR	2	H_2O NaCl SAND	1
POOR	UNKNOWN	GOOD	SEPARATE CONDUCTOR	NO	GOOD	FAIR		FAIR	GOOD	2	1	1	NONE	3	FAIR	2	ULTRA VIOLET	1
BAD	GOOD	EXCELLENT	SEPARATE CONDUCTOR	YES	GOOD	POOR	NEW LINK	GOOD	POOR OR BAD	2	1	4	NONE	3	GOOD	1	H_2O NaCl	1
BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES (NOT STOCK ITEM)	POOR	POOR	NEW LINK	GOOD	BAD	1	1	4 (1 WITH CAS)	NONE	3	GOOD	1	H_2O NaCl SAND DUST	4
BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES (NOT STOCK ITEM)	POOR	POOR	NEW LINK	FAIR	BAD	1	1	4 (1 WITH CAS)	NONE	3	GOOD	1	H_2O NaCl SAND DUST	4
BAD	UNKNOWN	BAD	SEPARATE CONDUCTOR	YES	POOR	POOR	NEW LINK	FAIR	BAD	1	1	4	NONE	3	GOOD	1	H_2O NaCl SAND DUST	4
POOR	GOOD	EXCELLENT	MOLDED IN CONDUCTOR	YES & NO	EXCELLENT	GOOD	REPLACE	FAIR	GOOD	2	2	2	NONE	4	FAIR	2	OZONE	1
GOOD		EXCELLENT	SEPARATE CONDUCTOR	YES	POOR	BAD	REPLACE REWORK	POOR	GOOD	1	4	1	NONE	2	GOOD	4		1
EXCELLENT	UNKNOWN	GOOD	SEPARATE CONDUCTOR	YES	GOOD	EXCELLENT	SPLICE	EXCELLENT	FAIR / GOOD	4	1	1	NONE	3	POOR	3	ULTRA VIOLET	2
GOOD	UNKNOWN	GOOD	SEPARATE CONDUCTOR	YES	GOOD	EXCELLENT	REPLACE	GOOD	GOOD	4	1	1	NONE	3	POOR	3	ULTRA VIOLET	1
GOOD		GOOD / (H_2O ABS)	EXTERNAL CONDUCTOR	YES	GOOD	POOR	SPLICE	GOOD	POOR / GOOD	4	2	1	NONE	3	POOR	1	HYDRO CARBONS	2

may be severed.

Natural resonant frequency to be below 2 cps.

Radio control can be used in all cases for low cost to weight and volume.

Basic member is obtainable on open market.

Heavy or large flat objects; dangerous.

Large flat sheets in airstream.

Refers to C.G. shift because of storage location causing a shift in location of C.G. of hoist and member.

How bad is stretch member for flight problems? Is it good for multiple hoisting? Is it good for estimating C.G.?

Refers to vertical smoothness of load when being hoisted. Chordal effect of some members causes oscillating motion.

The notation "NONE" means that a speed of 60 ft/sec would not have an injurious effect on the member.

Notation refers to the effect of stretch in a mechanical determination of distance from helicopter to load.

Some degree of protection may be added for all contaminants.

B

TABLE II. HOISTING MEMBERS 1-

NUMBER	TYPE	SIZE	WT/FT (LBS)	WT/150 FT (LBS)	STORAGE APPX. DIMS.	Mechanics					Flight-Chara		
						C. G. Control	Twisting-Side Load	Power Required	Smoothness	Speed Limitation	Resonance	Aerodynamics	Stretch
1	Round Wire Rope	1-1/4" Dia.	2.89	434	Drum=15-3/4" Core Dia. 1 Layer, 38 Revs. Drum=20"Dia x 56"	4	8	5	6	10	5	8	7
2	Swaged Round Wire Rope	1-1/8" Dia.	2.89 Apx.	434 Apx.	Drum=22-1/2" Core Dia. 1 Layer, 29 Revs. Drum=27"Dia. x39-1/2"	5	8	4	7	10	5	9	8
3	Stainless Round Wire Rope	1-1/2" Dia.	4.16 Apx.	624 Apx.	Drum=30"Apx. Core Dia. 1 Layer, 22.2 Revs. Drum=35"Dia x 40"	3	8	5	6	10	5	8	7
4	Flat Wire Rope	9/16" x 5-1/4"	2.83	425	Drum=12" Core Dia. 37 Layers, 37 Revs. Drum=42" Dia. x 7"	9	1	7	6	10	7	6	5
5	Wire Mesh	5/8" x 11-1/8"	3.65	547	Drum=12" Core Dia. 23.6 Layers, 23.6 Revs. Drum=43-1/2" Dia.x13-1/2"	8	1	7	8	10	8	5	9
6	Flat Nylon Webbing	.344" x 8"	1.10	165	Drum = 12" Core Dia. 33 Layers, 33 Revs. Drum = 36" x 10"	9	1	9	9	10	8	4	1

A

TABLE II. HOISTING MEMBERS 1-POINT

	Mechanics					Flight Characteristics						Control			Safety			Durability			Cost			
	C. G. Control	Twisting-Side Load	Power Required	Smoothness	Speed Limitation	Resonance	Aerodynamics	Stretch	Load - Land Orientation	Kinking & Knotting	Handling Compatability		Remote Control Compatibility		Emergency Shear Capability	Backlash	Inspection Reliability	Resistance to Contamination	Injurious Normal Contaminants	Durability	Standard Item	Ease of Maintenance (Repair)	Price Per Unit	Total Points
/4" ... ia "	4	8	5	6	10	5	8	7	6	4	6		5		5	6	6	3	5	6	9	4	8	126
/2" ... ia. 1/2"	5	8	4	7	10	5	9	8	8	5	7		2		6	5	6	4	6	6	7	4	6	128
px. ... . ia "	3	8	5	6	10	5	8	7	6	4	7		5		5	6	6	8	9	6	9	5	6	134
,	9	1	7	6	10	7	6	5	5	7	5		7		4	7	7	3	5	5	5	3	2	116
. rs, . /2" /2"	8	1	7	8	10	8	5	9	8	9	8		9		1	9	4	9	8	9	5	6	5	145
" , " 0"	9	1	9	9	10	8	4	1	2	6	9		2		8	4	2	7	7	7	9	2	8	124

L E G E N D

0. Impossible	5. Fair -
1. Very Bad	6. Fair +
2. Bad	7. Good -
3. Very Poor	8. Good +
4. Poor	9. Very Good
	10. Excellent

D

TABLE III. HOISTING MEMBERS 4-POINT

NUMBER	TYPE	SIZE	WT/FT (LBS)	WT/50 FT (LBS)	STORAGE DRUM	MECHANICS: C.G. Control	Twisting-Side Load	Power Required	Smoothness	Speed Limitation	FLIGHT CHARACTERISTICS: Resonance	Aerodynamics	Stretch	Load - Land Orientation	Kinking & Knotting	Handling Compatibility	CONTR… : Remote Control
1	Round Wire Rope	5/8" Dia.	.72	36	Drum - 15-3/4" Core Dia. 1 Layer, 11.3 Revs. Drum - 18" Dia. x 11" Quant. 4	6	9	5	6	10	5	8	7	6	4	6	5
2	Swaged Round Wire Rope	5/8" Dia	.72	36	Drum - 22-1/2" Core Dia. 1 Layer, 8.1 Revs. Drum - 26" Dia. x 10" Quant. 4	6	9	4	7	10	5	9	8	8	5	7	2
3	Stainless Round Wire Rope	3/4' Dia.	1.04	52	Drum - 30" Apx. Core Dia. 1 Layer, 6.1 Revs. Drum - 34" Dia. x 10" Quant. 4	6	9	5	6	10	5	8	7	6	4	7	5
4	Flat Wire Rope	3/8 x 1-3/4	.69	34.5	Drum - 12" Core Dia. 23.4 Layers, 11.5 Revs. Drum - 31" Dia. x 4 (Dbl. Wrapped) Quant. 2	9	8	7	6	10	7	6	5	5	7	5	7
5	Wire Mesh	5/8" x 4"	.91	45.6	Drum - 12" Core Dia. 20.8 Layers, 10.4 Revs. Drum - 40" Dia. x 6-1/2" (Dbl. Wrapped) Quant. 2	9	8	7	8	10	8	5	9	8	9	8	[illegible]
6	Flat Nylon Webbing	.344 x 2"	.30	15	Drum - 12" Core Dia. 21.5" Layers, 10.75 Revs. Drum - 29" Dia. x 4" (Dbl. Wrapped) Quant. 2	10	8	9	9	10	8	4	1	2	6	9	[illegible]

Legend

0. Impossible
1. Very Bad
2. Bad
3. Very Poor
4. Poor

5.
6.
7.
8.
9.
10.

A

III. HOISTING MEMBERS 4-POINT

MECHANICS				FLIGHT CHARACTERISTICS						CONTROL			SAFETY			DURABILITY			COAT			
Twisting-Side Load	Power Required	Smoothness	Speed Limitation	Resonance	Aerodynamics	Stretch	Load - Land Orientation	Kinking & Knotting	Handling Compatibility		Remote Control Capability		Emergency Shear Capability	Backlash	Inspection Reliability	Resistance to Contaminants	Durability	Standard Items	Ease of Maintenance (Repair)	Price per Unit	Total Points	
9	5	6	10	5	8	7	6	4	6		5		6	6	6	3	5	6	9	5	8	131
9	4	7	10	5	9	8	8	5	7		2		7	5	6	4	6	6	7	5	6	132
9	5	6	10	5	8	7	6	4	7		5		6	6	6	8	9	6	9	5	6	139
8	7	6	10	7	6	5	5	7	5		7		6	7	7	3	5	5	5	5	2	127
8	7	8	10	8	5	9	8	9	8		9		3	9	4	9	8	9	5	8	5	158
8	9	9	10	8	4	1	2	6	9		2		10	4	2	7	7	7	9	3	8	135

Legend

0. Impossible	5. Fair -
1. Very Bad	6. Fair +
2. Bad	7. Good -
3. Very Poor	8. Good +
4. Poor	9. Very Good
	10. Excellent

B

The mechanical advantage and the reduction in weight could be estimated in the reduction of drum sizes. The maximum permissible C.G. changes in one direction limited drum width and therefore permitted progression in drum diameter.only.

During this tension-member investigation, a laboratory test was made to determine the crushing loads imposed on a drum when a flat, flexible tension member is wrapped continuously upon itself. This test is described by Figure 2. Figure 3 is a photograph of the laboratory test setup. The loads were measured by strain gages placed inside the drum at 90° intervals to each other around the periphery of the drum under the area covered by a nylon tension strap. Indications were that the loads could become quite severe when a "stretchy" member is used and that this problem deserves serious consideration if a final design called for overlaying of any kind of flexible tension member on a hollow drum or a drum containing hoisting equipment.

From the analysis of the test data obtained, it was apparent that crushing loads imposed by "stretchy" hoisting members would be objectionable in designs in which equipment weights are as important as those used in helicopters.

Information obtained from the John F. Roebling & Sons Company of Trenton, N. J., showed that it is their custom, when overlaying cables, to use a series factor varying from a crushing load of 100 percent in the outermost layer toward 90 percent in the next layer and subsequently through succeeding layers in 10-percent increments to 70 percent in the fourth and all subsequent layers. It is believed that this method of load determination may be somewhat conservative. If an overlaying cable system is used, tests should be run in order to determine a more precise set of values. It was deemed advisable that the overlaying effects of any hoisting member considered for use should be checked through laboratory test. It is apparent, from the investigation done to date, that there is an appreciable difference in the behavior of different kinds of tension members when used in this manner.

INVESTIGATION OF SINGLE DRUM CONCEPT

An analysis of the problems encountered when simultaneously operating all of the tension members of either a 2- or 4-point hoisting system with sufficient precision to prevent overloading of a single member or "out of level hoisting", led to consideration of a single drum configuration. The use of a single drum system as described by the model photographs (see Figures 4 through 9) would allow a good degree of precision in the simultaneous reeling of different members. Control of this system could be accomplished with a hydraulic motor whose output torque and velocity are controlled by flow control and pressure metering. High impact forces normally introduced into the tension members during acceleration and deceleration could be modified by the damping action of the hydraulic valves.

Figure 4 demonstrates the versatility of the single-shaft system in the possible substitution of different tension members for testing purposes.

Illustration B of Figure 4 describes the use of 1-point and 2-point load suspensions in which either the side-by-side straps or the single cable is stowed on the drum when the other system is in use.

Illustration C of Figure 4 refers to the 1-point and 4-point load suspensions in which either one is drum-stored during use of the other.

Illustrations D, E and F of Figure 4 describe the possible combination of 2- and 4-point systems in which drum storage could be obtained as in the previous systems.

Illustration A describes a collection of all of the suggested methods which could be obtained in the suggested combination, with tension member substitution only, for flight test purposes.

A single drum system is readily adaptable to mechanical or electrical tension member jettisoning. Provisions can also be made for electrical opening of the hooks. However, mechanical opening of the hooks, as originally specified, was found to be impractical for the planned system, as the use of multiple hook opening lanyards would be complex and unreliable, and was therefore not considered in this design. Electrical hook opening, freewheeling of the cable drum, and cable cutting were considered instead. This results in a simple and reliable configuration, with only the load-carrying cables extending below the helicopter.

An electrical connection to the hook opening mechanism can be accomplished by providing an insulated stranded copper conductor in the center of each load-carrying cable. By coiling this wire around a soft center-core (cotton or nylon rope), it is protected from excessive cable compression or tension loads.

With respect to space required for the hoist, it is apparent that the smaller the cable involved, the smaller the drum diameter. In this direction, cable manufacturers recommend that the ratio of cable diameter to drum diameter be not less than 20 to 1. Again, it is pointed out that if, in the case of the single-point lift, a single cable was used, the drum would be quite large; i.e., a 40,000-pound-capacity stainless steel cable would be 1-1/2 inches in diameter and would require a 30-inch-diameter drum. On the other hand, if four separate cables were used to carry the 40,000-pound load, then four 3/4-inch-diameter cables would require a drum diameter of only 15 inches. This same relationship would also be translated into sheave-cable relationships. Moreover, if different size cables were used to handle the different kinds of loads (i.e., 1-, 2- or 4-point systems), different size drums would be necessary. For instance, in the case of the 1- and 4-point system, five separate drums with five separate drive systems would seem to be necessary, with the obvious increase in bulk, weight and control complexity.

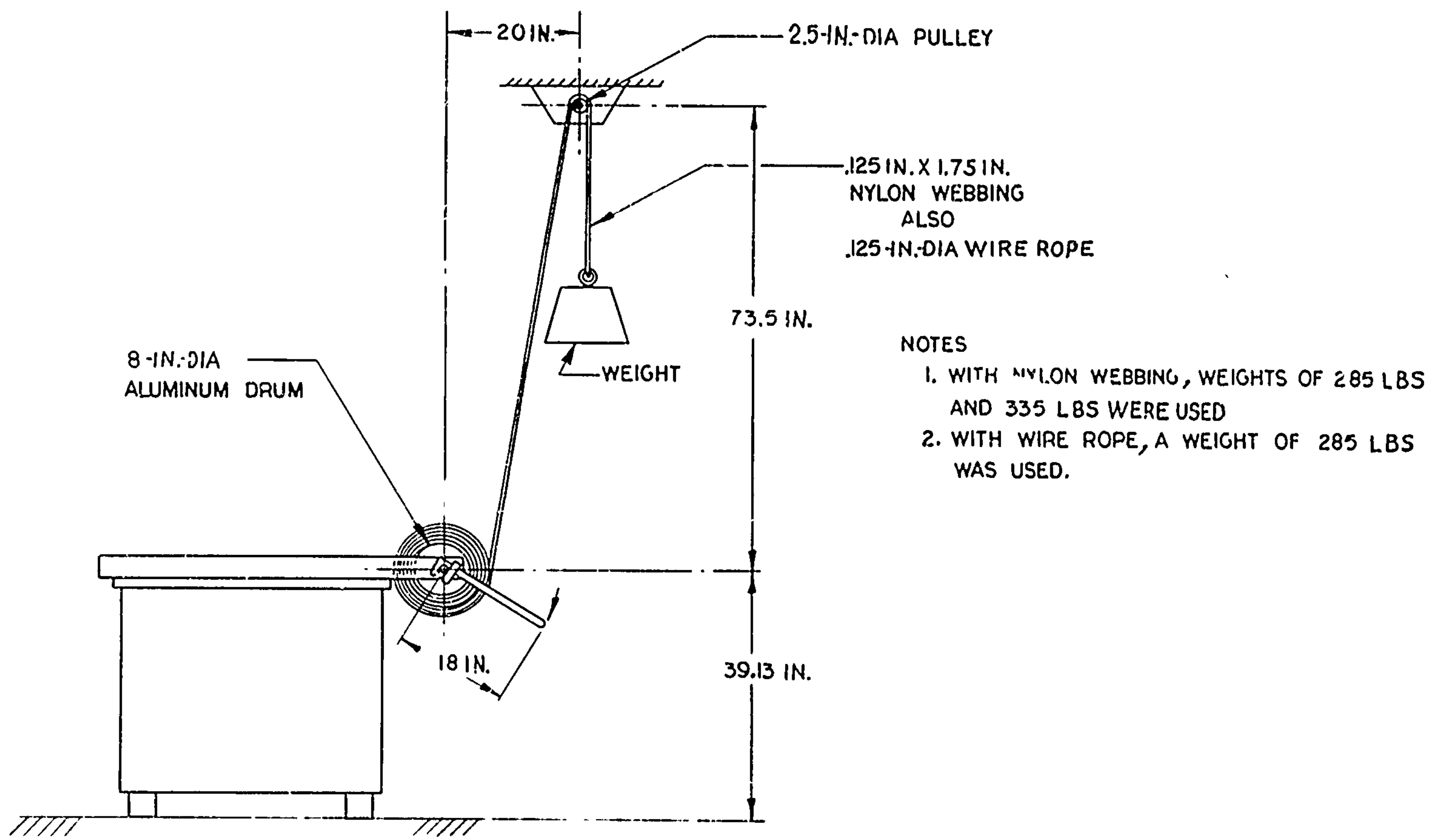


Figure 2. Drum Crushing Load Test

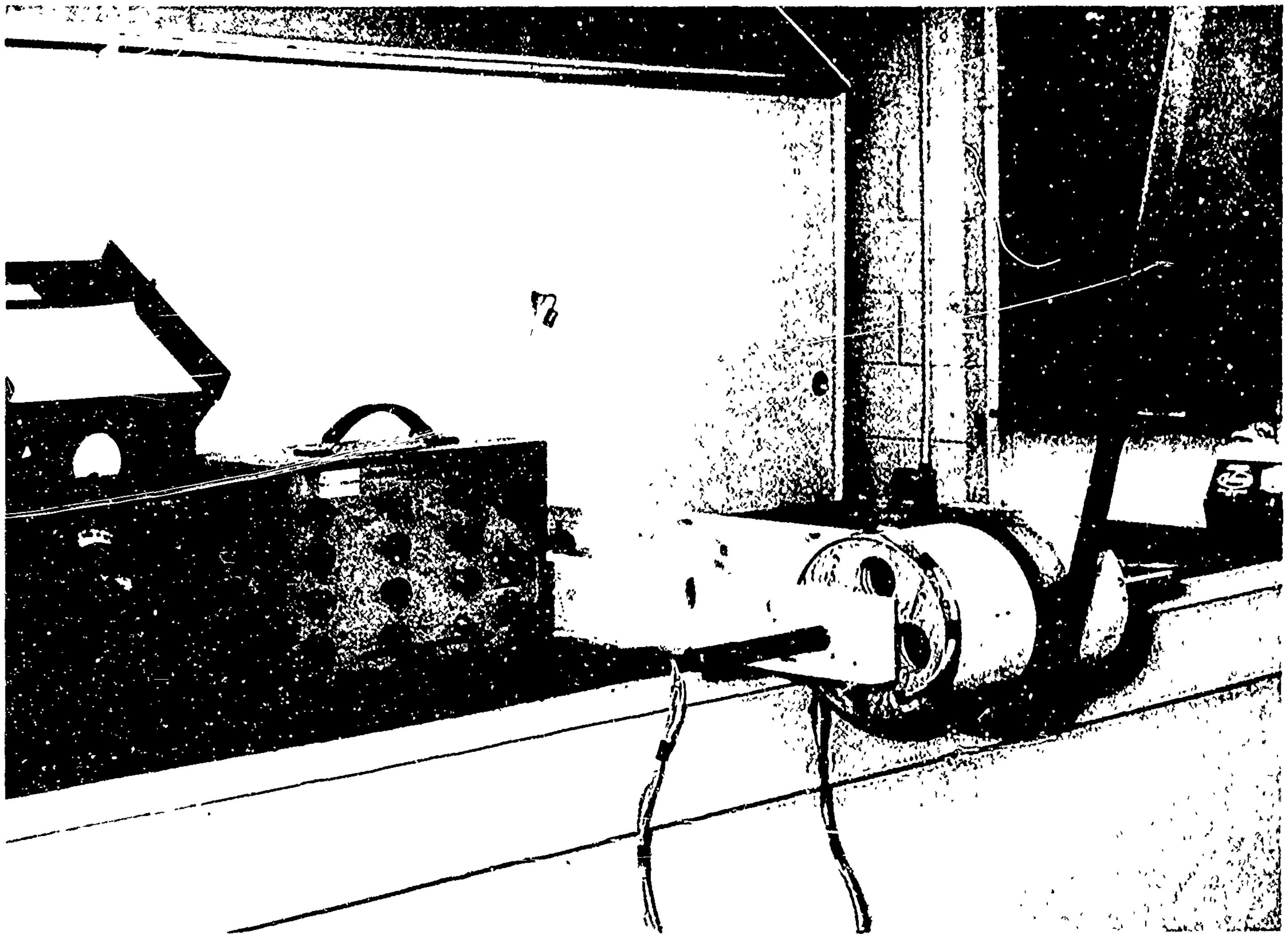
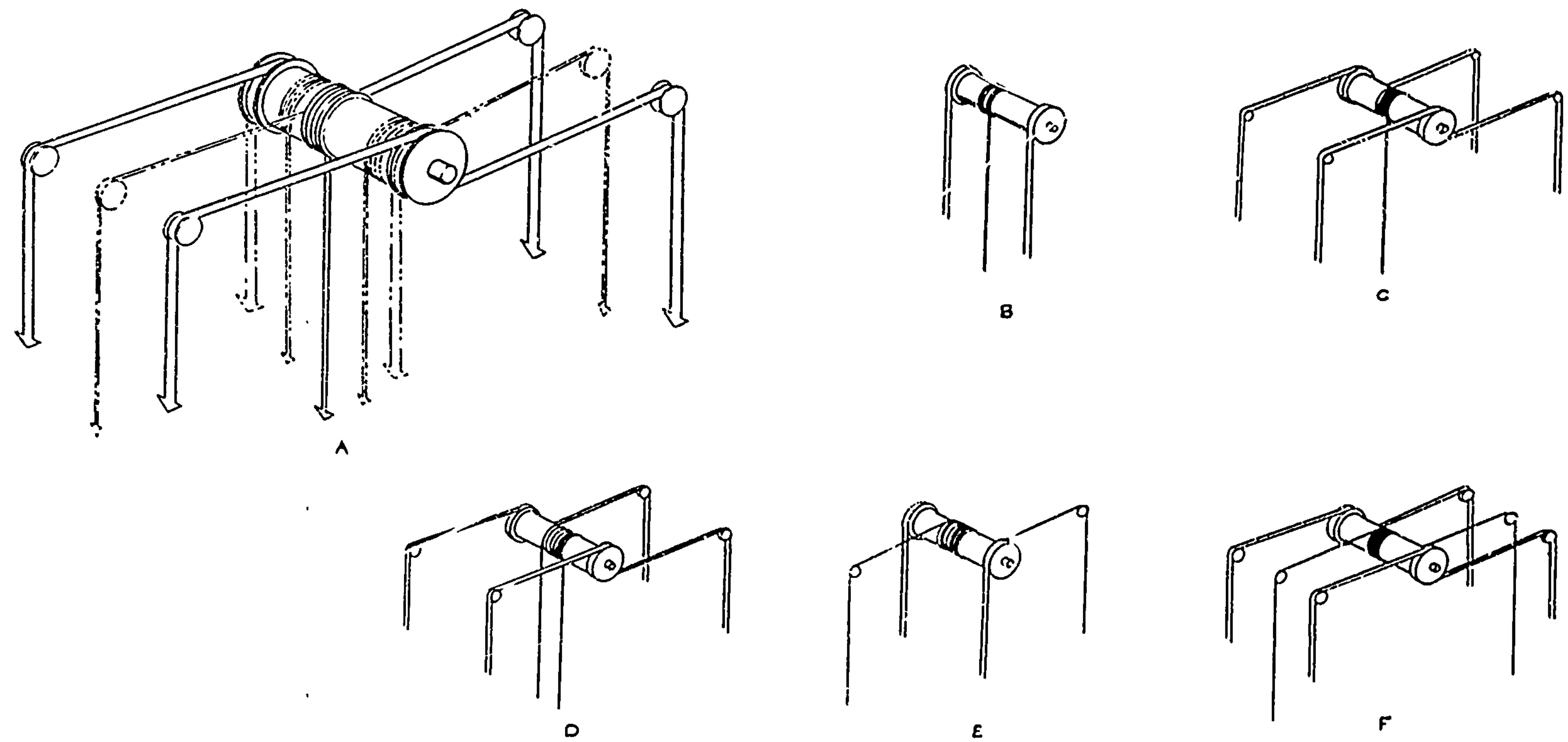


Figure 3. Drum Load Test



Note: See Text for Description of Function

Figure 4. Single Drum Reeving Configurations

Figure 5. Belts and Cable Stowed

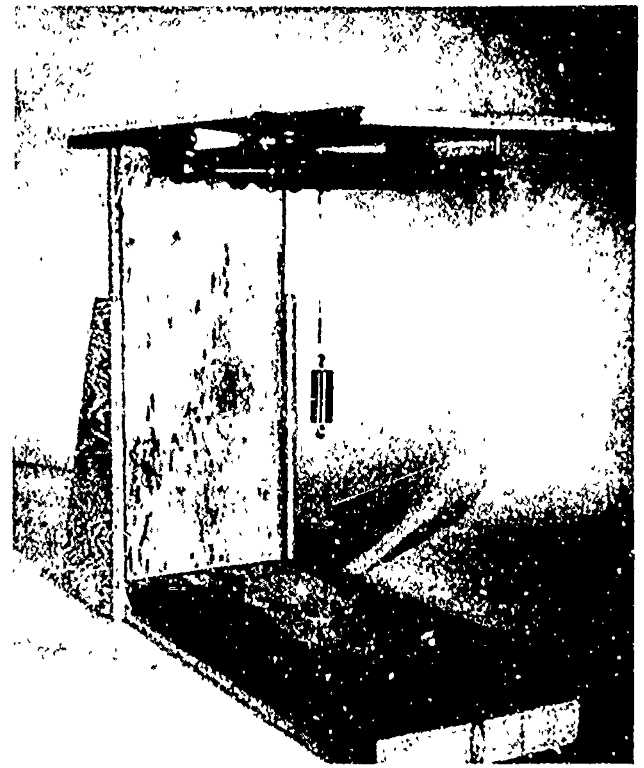


Figure 6. 1-Point Cable in Use; 2-Point Side-by-Side Belts Stowed

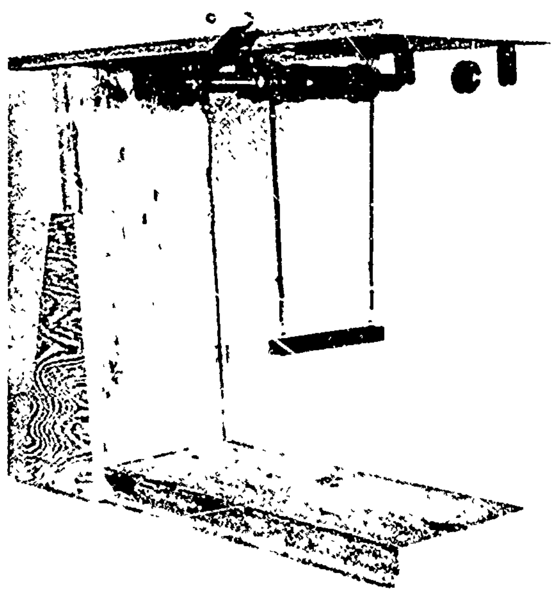


Figure 7. 2-Point Side-by-Side Belts in Use; 1-Point Cable Stowed

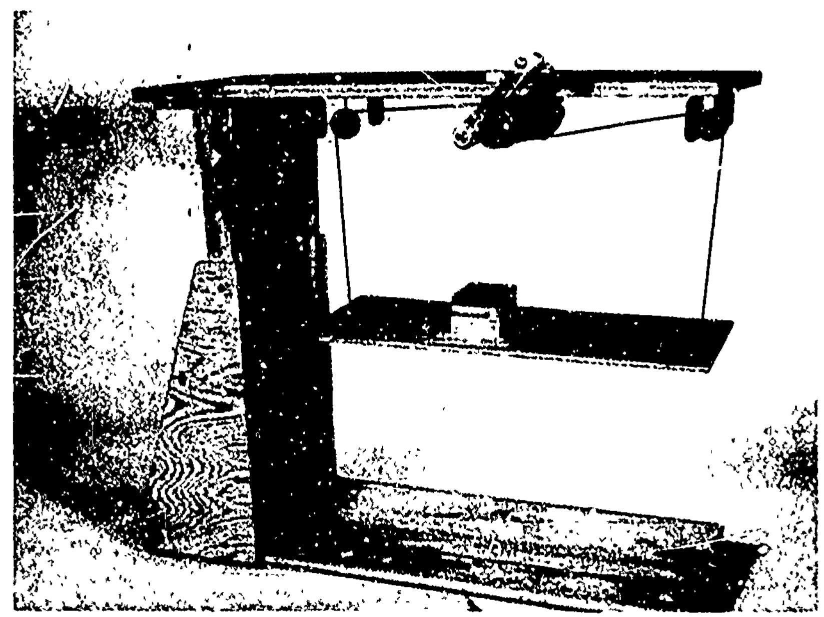


Figure 8. 2-Point Fore-and-Aft Cables in Use; 2-Point Side-by-Side Belts Stowed

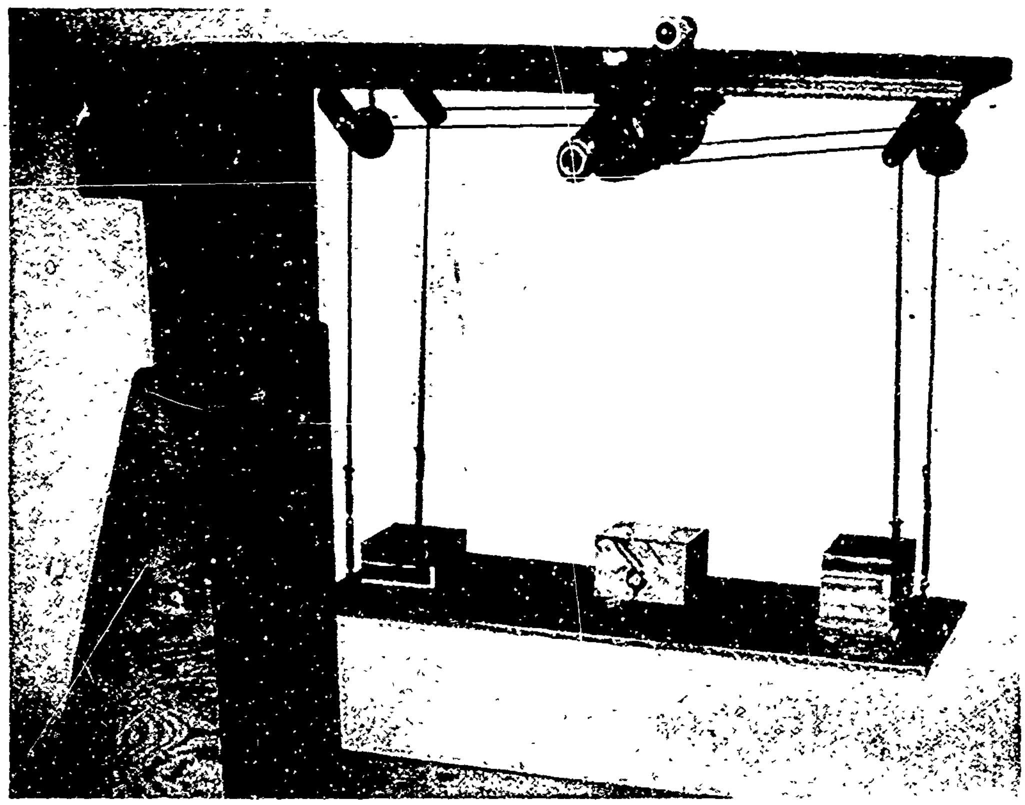


Figure 9. 4-Point Belts in Use; 1-Point Cable Stowed

The ability to change the single-shaft system from 1-, 2-, or 4-point lifts can be readily accomplished. The changeover of the system consisted of moving sheaves from one location to another and the replacement of the load hooks.

Simultaneous operation of separate drive systems (to keep cable forces adjusted) appeared to be extremely difficult. Because of the accuracy needed between line lengths and loads, complex hydraulic or electrical linkage between separately driven hoists would be required. Mechanical ties such as shafts, chains, belts, gears, etc., would have built-in inaccuracies as well as play and flexure. These systems become spread out, cumbersome, vulnerable, and difficult to monitor, maintain or replace. Moreover, inherent system redundancies cause weight and bulk increases.

AERODYNAMIC CONSIDERATIONS

If multiple hoists with inadequate control are used, there exists the possibility that in flight the load would not remain at the desired attitude with respect to the helicopter. This could induce the following problems:

1. The aerodynamic drag of the skewed load might be larger than the same load in the desired attitude, thus reducing the flight speed capability of the helicopter.

2. It is possible, because of aerodynamic unbalance and inadequate monitoring, that only two of the four suspension lines might carry the entire load, thus inducing failure of the cables.

3. The load may tend to "fly" due to an unfavorable angle of attack. This could induce a vertical oscillation of the cargo as the load alternately "flies" and "stalls" with probable dynamic coupling.

4. If one of the hoisting cables were to go slack, the individual motor controlling this cable might speed up due to relaxation of the line tension. As the motor speeds up and the cables become taut again, a "whump" may occur, which could cause an instantaneous shock to be generated and transmitted through the system. This could occur alternately on all of the multiple hoisting cables and could induce a resonant oscillation to be set up in the helicopter.

CONCLUSIONS

Considering the trade-offs of weight, bulk, dynamics, handling facility, safety, control, reliability, and cost, the following conclusions were drawn from this study:

1. Commercially available cargo hooks with electrical opening

features are excessively heavy. Cargo hooks can readily be designed for this application.

2. Only round steel wire rope and round stainless steel wire rope should be further considered for use as the primary tension member; since stainless steel wire rope is only slightly larger in diameter, it should be considered in the proposed design.

3. A single shaft drum reel should be used, as it presents the best possibility of arriving at a device which would be dependable and capable of best satisfying the specifications.

DESIGN ANALYSIS

During the proposal and Phase I work of this program, the relative merits of two different systems were investigated. This investigation compared a single-drum system and a system incorporating several drums. It became apparent during this analysis that the single-drum system has several significant advantages over the multiple-drum system.

CONTROL PROBLEMS OF MULTIPLE-DRUM SYSTEMS

Load balancing and leveling would be extremely difficult under flight conditions in rough air. A closed-loop servo control system using position and rate transducers can improve the stability of a multiple-drum system, but continuous interaction and hunting of the drives would lead to high power consumption, self-heating, and wear. The problem of synchronous driving of multiple hoists is very difficult. Efforts to accomplish the synchronous operation of separate hoists have not been successful in the past. If the required coordination could be achieved, it would increase costs, raise the system weight, increase the number of parts required, increase and complicate maintenance, add dynamic problems, and reduce safety.

WEIGHT

The weight of a single-drum system will be less than that of the multiple-drum system. The multiple drums in themselves will weigh more than the single drum. Moreover, the weight of the necessary structural supports, power sources, controls, etc., will be multiplied by the number of drums used. In the multiple-drum system, the use of different size cables will be necessary to meet the requirements of 1-, or 2-, or 4-point hoisting. When different cables are used for different loads, the necessary on-board storage of hoists and cables not in use will reduce the payload of the helicopter. In the single-drum system (with negligible exceptions), all of the on-board hoisting equipment is in use, regardless of which of the 1-, 2-, or 4-point systems is being used.

The weight penalty in the multiple-drum system will be even greater because each of the individual drive systems must have reserve capacity of its own, while a single-drum system has an equalizing feature which eliminates this requirement. For example, consider a 40,000-pound load suspended from a multiple-drum 4-point suspension system. In the idealized case each cable would carry 10,000 pounds. If two diagonally opposite cables were shortened, each of these cables would then carry 20,000 pounds. Each of the drive systems would then be required to operate with a 20,000-pound load. This would mean that the total system would have to be designed to operate with a lifting capacity of 80,000 pounds. In the single-drum system, under the same conditions, regardless of the load on the individual cables, the total load reflected to the cable drum and the drive mechanism would be the sum of the cable loads or 40,000 pounds. Thus, the total system would be designed to operate with a lifting capacity of 40,000 pounds.

RELIABILITY AND CONTROL

A multiple-drum system consists of three to five completely separate hoists, each with its own closed-loop servo control system. In order to balance and level the load, each one of the systems must continuously compare positions and rates of the other subsystems. This leads to a very complex control system, with relatively low reliability and a high maintenance burden. An added difficulty will appear when a 2-point system is used in a 1- or 2-point configuration, because now the drive systems must work in parallel, but at any instant must also exert the proper force to keep the load balanced between the two cables.

The single drive system, on the other hand, does not require an automatic, closed-loop control system. An all-manual control system, operated by a load master, will be simple, reliable, and flexible in operation. The only automatic devices in a single-drum design are UP-LIMIT and DOWN-LIMIT switches on the cable drum, plus overload relief valves on the hydraulic cylinders.

The precise simultaneous starting and stopping of four different drive systems would be extremely difficult. Again, the single-drum drive with a single shock absorber presents obvious advantages.

C.G. CONTROL

The problems of C.G. control in shifting from a single-point to a multi-point system are greatly simplified with the use of all of the on-board equipment, in each different mode of operation, and a single drum. If all four cables are used in either the 1-, 2- or 4-point configuration with a single drum reel, it is possible to make the C.G. shift negligible. This is also true because no equipment is removed from the aircraft and replaced with other equipment in adverse locations, as is likely in the case of multiple lifts or reels. Handling facility of the single shaft system in shifting from 1-, 2- or 4-point lifts is greatly simplified over others investigated. The changeover of the system consists only of moving sheaves from one location to another and interconnection of the hooks when required.

SAFETY

Safety requirements will be difficult to meet with multi-point lifts, because simultaneous cable roll-off or system locking becomes complicated and more vulnerable than would be the case with only one device controlling the single drive. It is also obvious that the single drive, with its reduced number of power feed lines, controls, etc., between drive and actuation source, results in greater operational safety.

In conclusion, the single-drum hoisting system has some fundamental advantages over a multiple-drum system as pointed out in the foregoing discussion. Secondarily, the single-drum system will permit easier maintenance because of fewer parts and less complicated controls. Onboard handling of the equipment will be easier because of the use of smaller cables. And finally, the use of fewer parts, a lower weight, and less control complication will result in considerably lower cost.

MECHANICAL DESIGN

GENERAL

The proposed system is a single-drum helicopter cargo hoist which is capable of providing a 1-point lift, or a 2-point lift, or a 4-point lift using the same hardware with minor re-rigging of the cables and sheaves. Figures 10, 11 and 12 are three illustrations showing the same system in use in the aforementioned order. The single-drum reel is driven by a hydraulic motor through a cyclocentric gear reduction (see Figure 13) and a pair of rubber torsilastic shock and vibration dampers. A disc brake is placed adjacent to the motor to provide for the required safety and control features. The drum (or reel) is provided with double helical cable grooves that progress inward toward the center of the drum from both ends. This grooving provides four separate slots for the cables. Two pairs of cables are fed simultaneously onto the drum through two lariat rope guides. The lariat rope guide is a device fabricated by AAI for their commercial Lo-Hed hoists. A typical system is shown in Figure 14.

Judicious placement of the sheaves in the proposed system results in no center-of-gravity change of the hoisting system itself when changing from a single-point suspension hoist to a multi-point suspension hoist or when the system is being used for hoisting purposes in any of the three configurations.

Four hydraulic cylinders are used in the system to prevent overloading of any individual cable assembly and to provide for adjustment of individual cables as required and for leveling of the load.

Cylinders with a 2-foot stroke have been tentatively selected, resulting in a 4-foot cable adjustment range. Cylinders with a shorter or longer stroke can be substituted without difficulty.

The sheave brackets are fully swivelling to permit their use in all three rigging arrangements. Ball or roller bearings are used throughout.

CAPACITY

The system will have a working load capacity of 40,000 pounds. It has been designed structurally using a 2.5-g design factor and a 1.5 ultimate factor.

The cable arrangement as shown permits the use of the system in 1-, 2-, or 4-point configurations at any distance up to 150 feet below the helicopter.

The velocities required are easily attained and may be exceeded by at least 30 percent using the equipment shown, should it be found desirable.

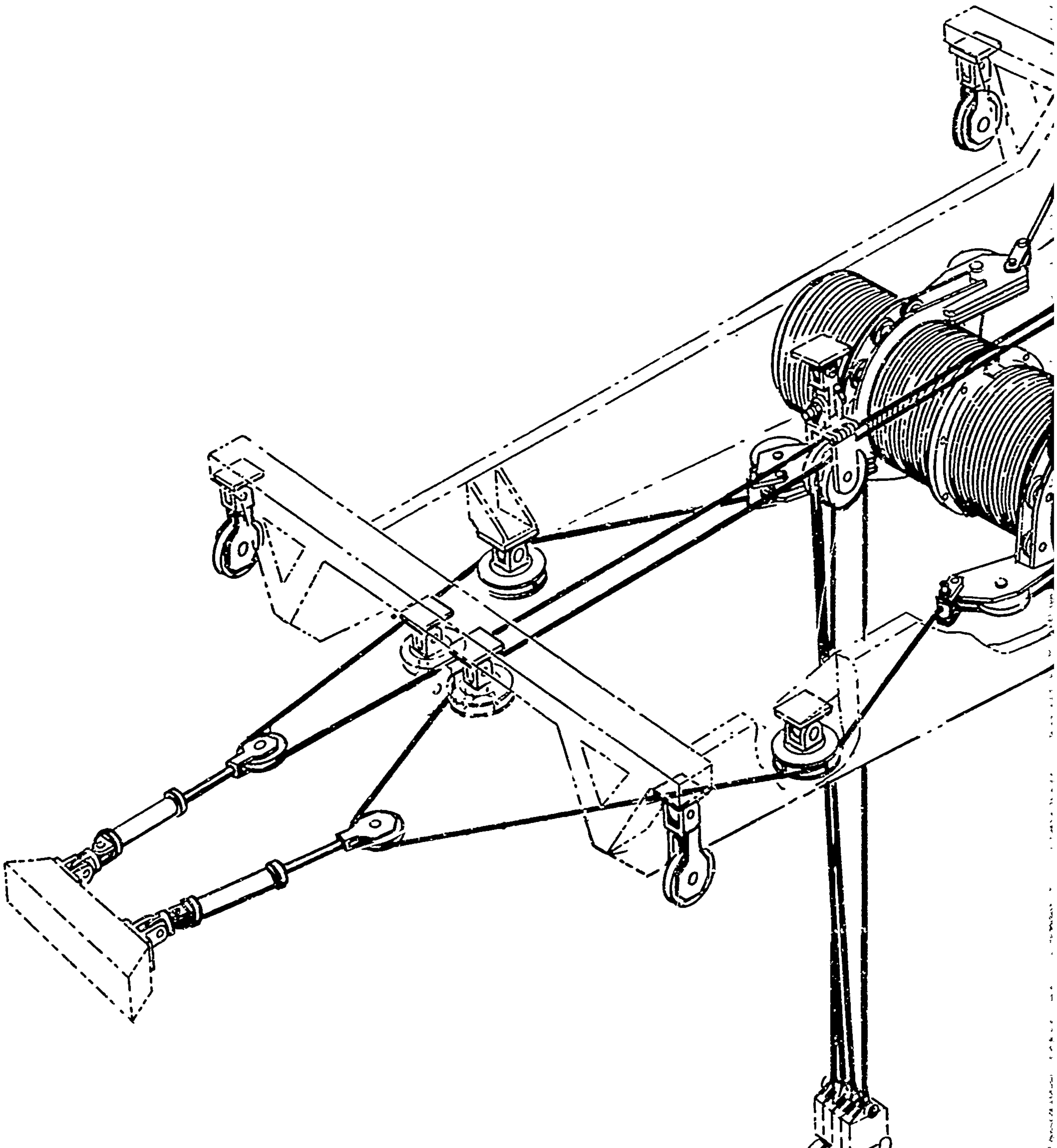
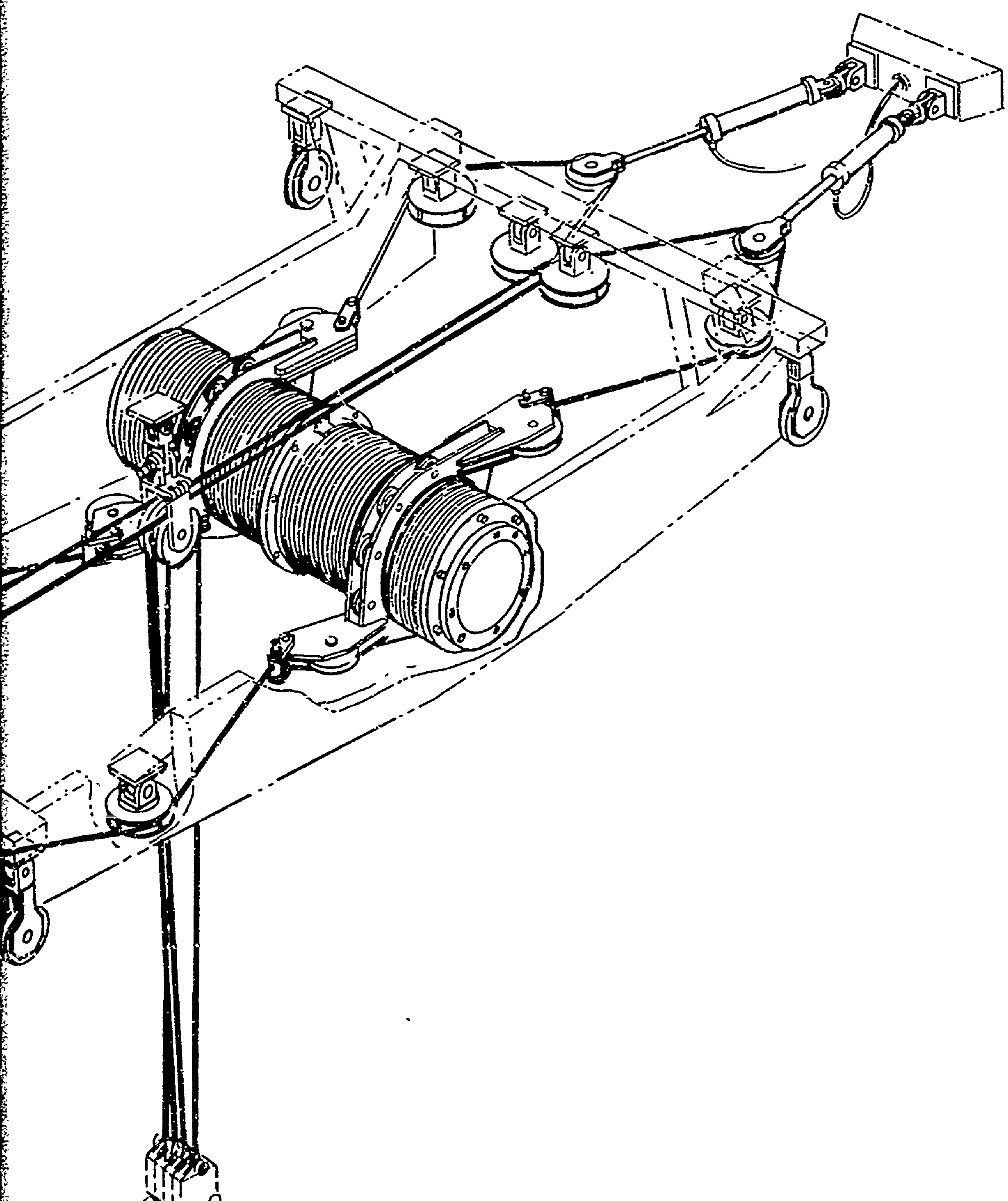


Figure 10. 1-Point Reeving System

A



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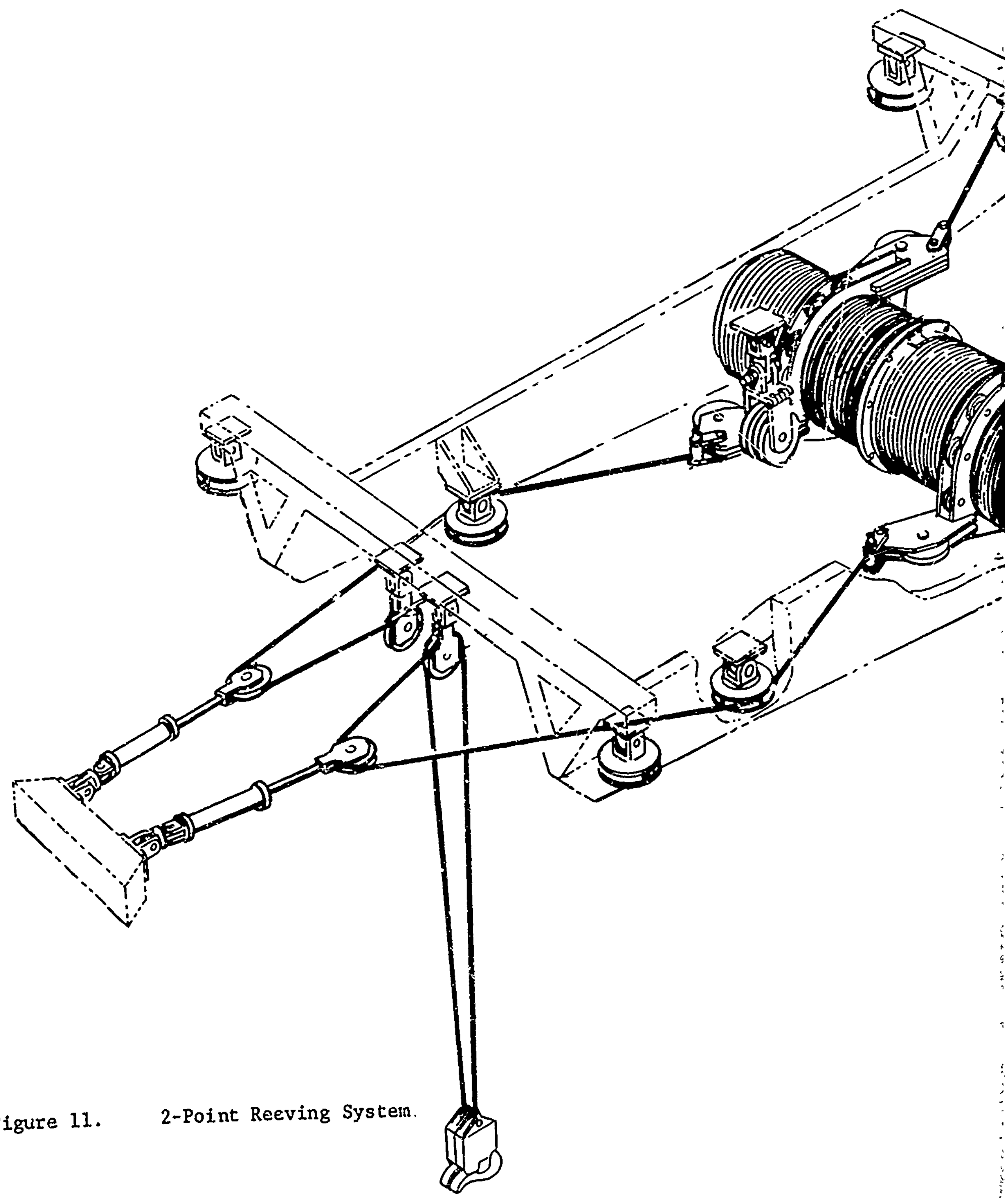

Figure 11. 2-Point Reeving System.

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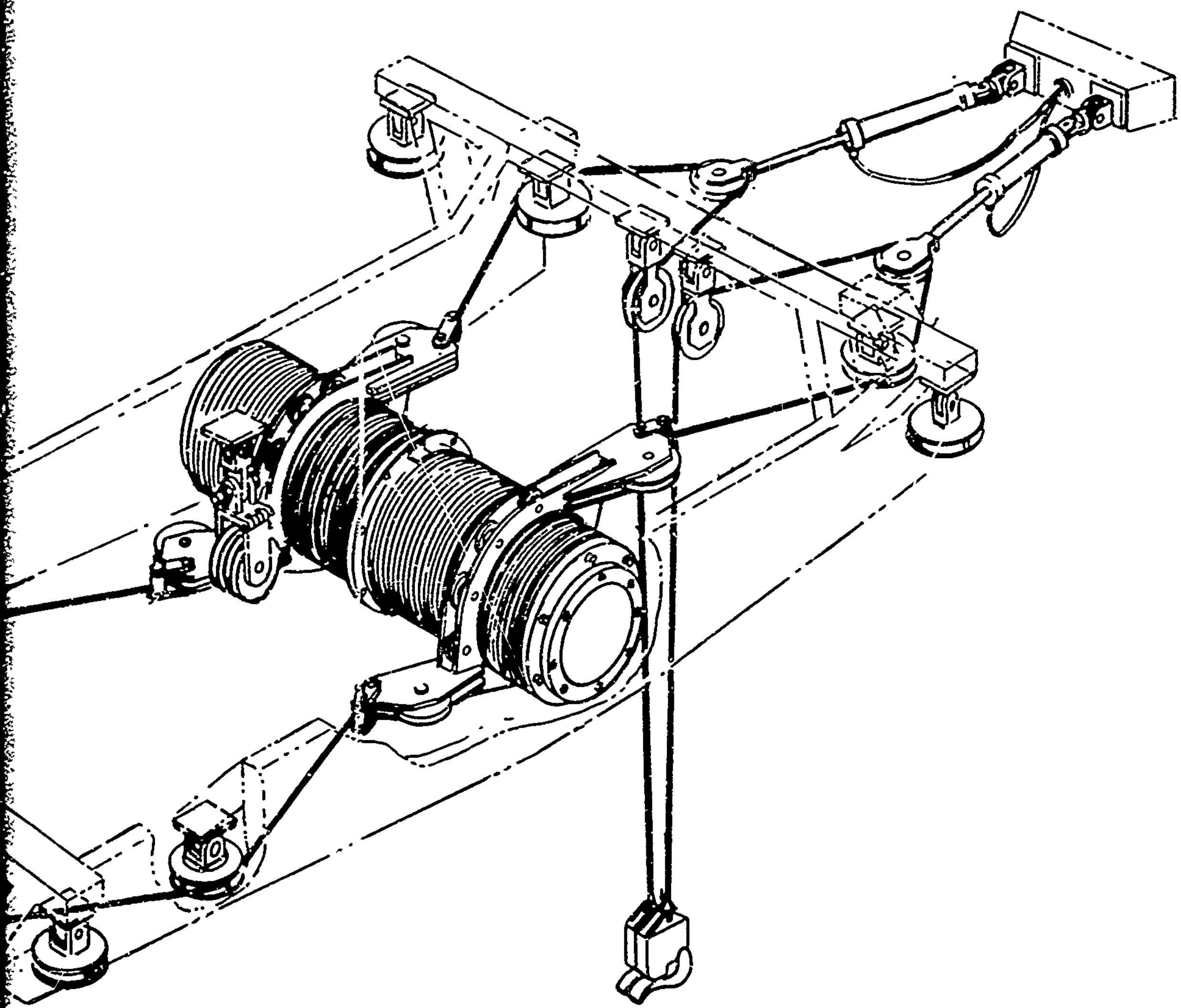

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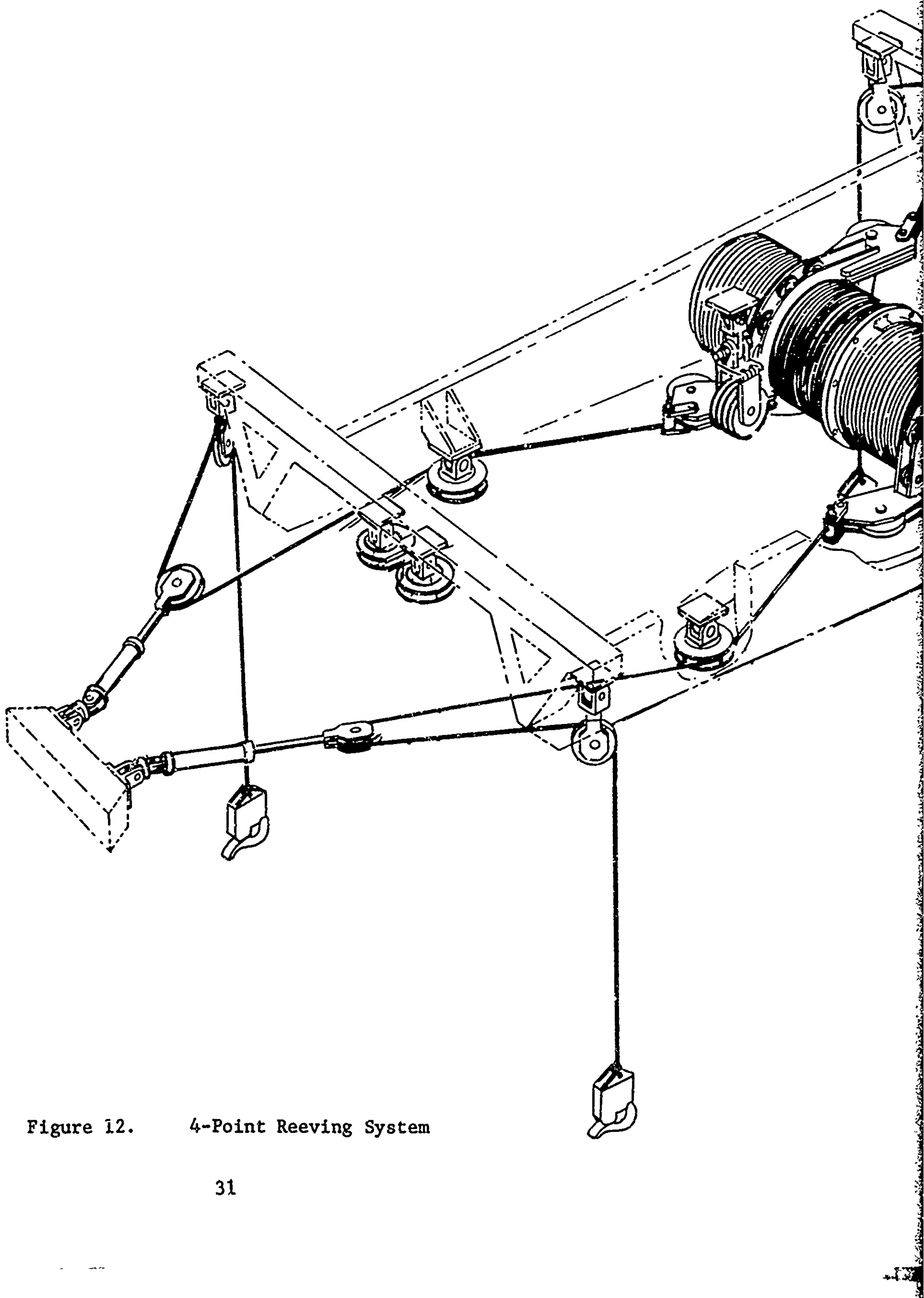
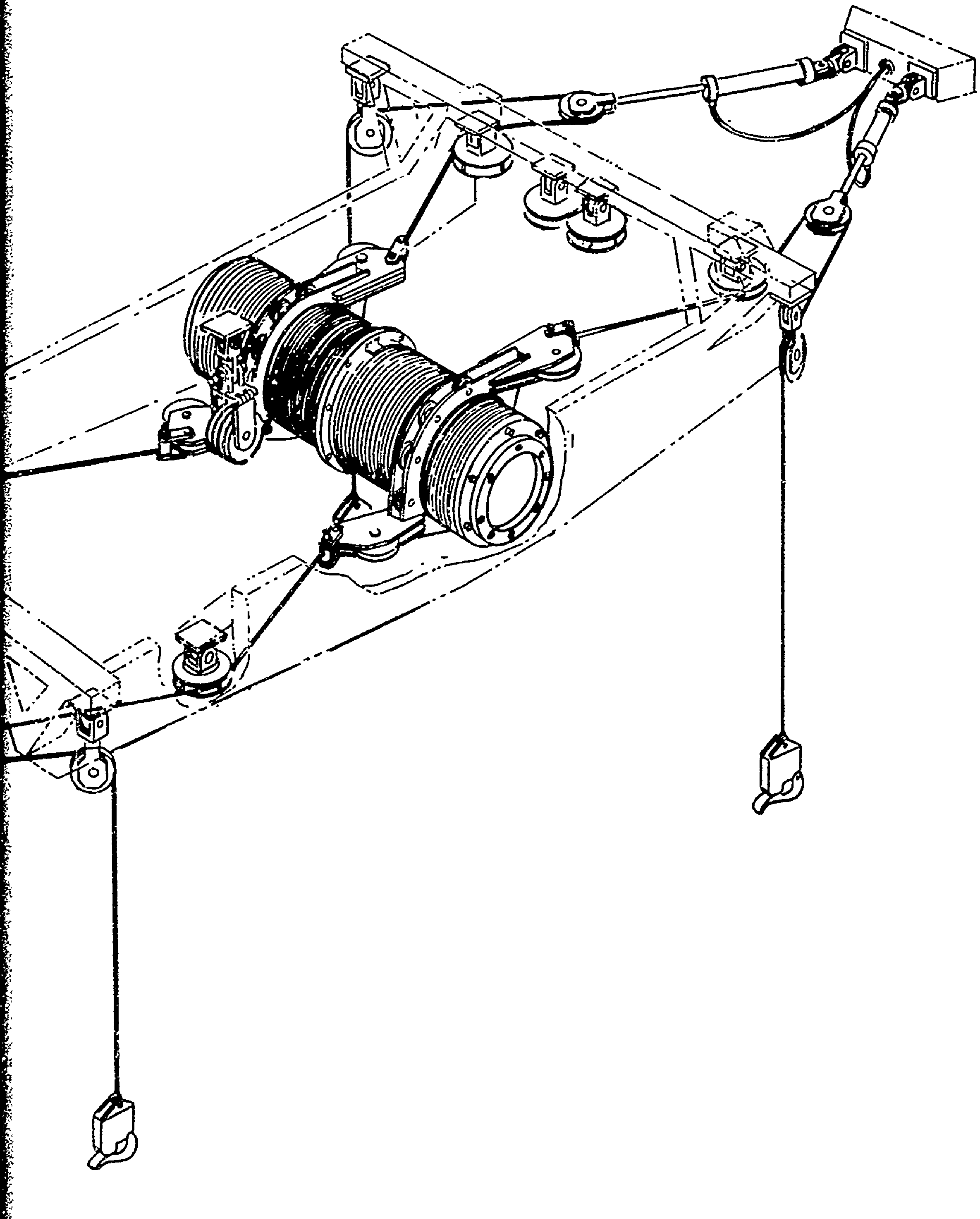


Figure 12. 4-Point Reeving System

A



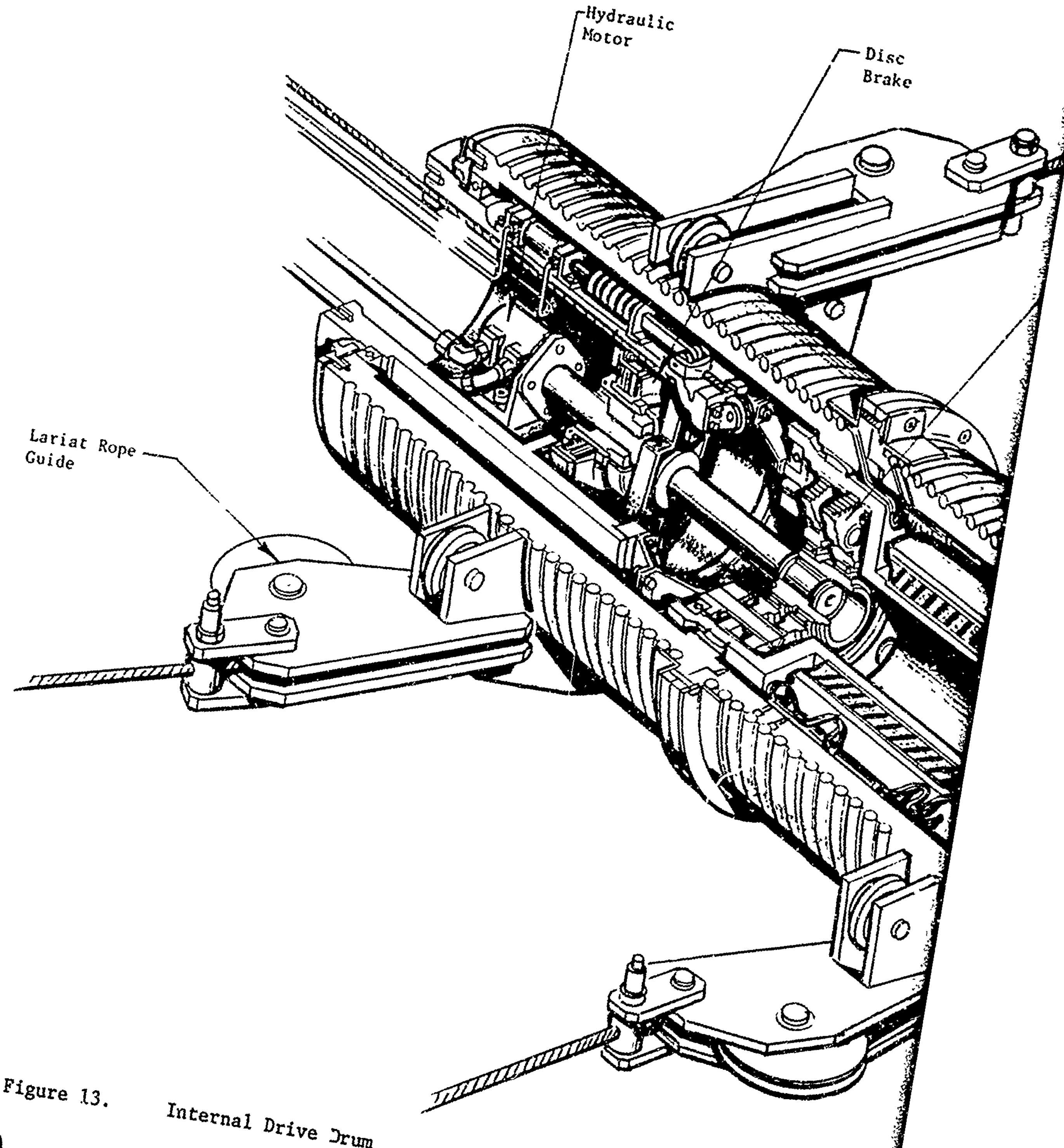
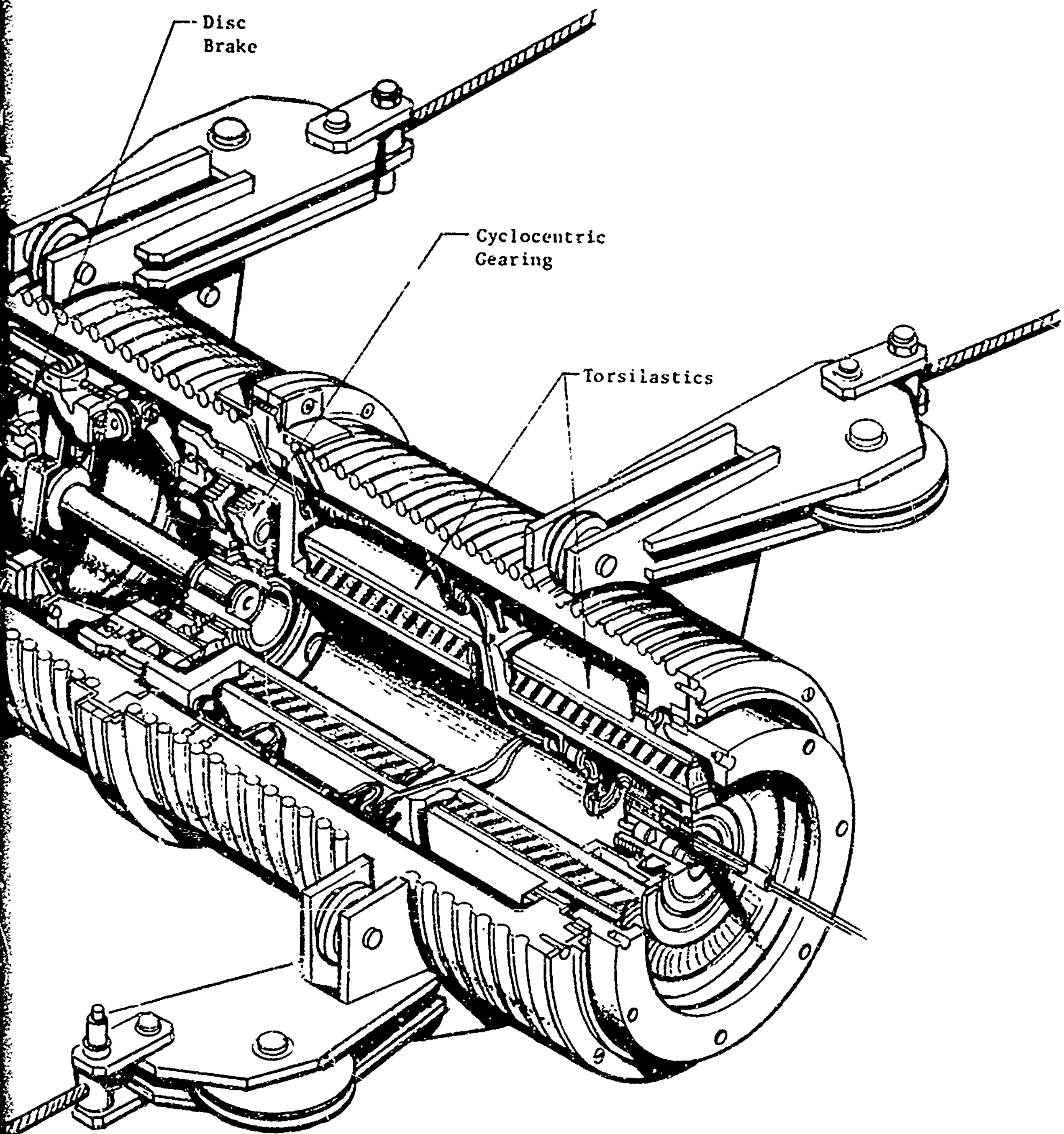


Figure 13. Internal Drive Drum

A



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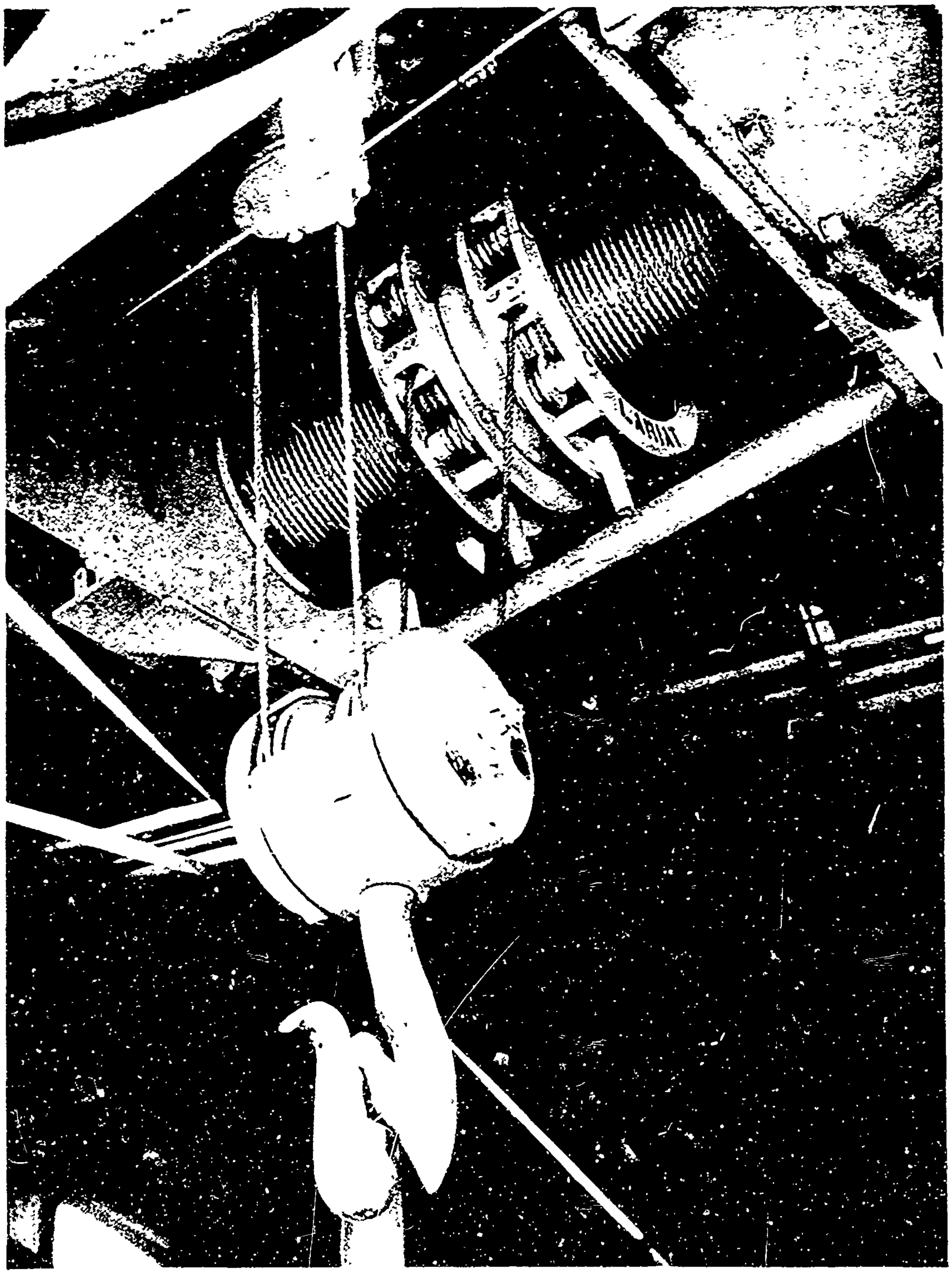


Figure 14. AAI "Lo-Hed" Hoist With Lariat Rope Guide

CABLE AND HOOK DESIGN

The cargo hooks to be used will be electrically controlled from both of the helicopter control stations as well as from the load-master station. Manual control, at the hook, is also provided in the design.

Load-master control is arranged by the use of a helicopter-installed line and control box which is designed to allow load-master electrical control of the hoist. This same device could be used to supply pilot-to-load-master communication if required.

No reasonable means of manual control of the cargo hooks from the helicopter could be found during this study. Three alternate modes of emergency release are described in the "Safety" section of this report.

Electrical release of the hooks is accomplished by the use of load-carrying cables equipped with electrical conductors in their cores. A dependable conductor for this application is obtained by using an insulated multiple-strand wire wrapped helically on a non-conducting synthetic center. This technique of fabrication permits extreme distortion or stretching of the conductor without fracture. It is pointed out that, where multiple cables are used for the single or double lift arrangements, redundancy of the conductor usage is possible by interconnecting the cables and hooks. See Figure 15.

The hooks will have provision for remote-electrical or direct-manual opening. Closing will be direct-manual only. No provision is made at this time for remote-electrical closing of the hooks, because of the resulting complexity; however, it can be incorporated if necessary.

Hooks with an electrical opening feature and with a nominal 10,000-pound load capacity are available commercially. However, these are designed for ground-based crane and hoist applications and are excessively heavy.

DISCUSSION OF 1- AND 2-POINT SUSPENSION SYSTEMS

An investigation was made to determine the effects of using the AAI proposed system as a 1- and 2-point suspension system only. The system incorporates both 150-foot and 80-foot lift capabilities.

Figure 16 illustrates such a system. In this system, 1-inch cable would be used and only two hydraulic cylinders. The central pulley would be a dual one only, and the 2-point usage would use single pulleys strategically located.

The drum would consist of one having a double helical groove progressing from one end of the drum completely through to the other. In this case, only one lariat rope guide would be required. The drum would be roughly 50 inches long and 30 inches in diameter. Although a 20-inch-diameter drum could be used for this application, a 30-inch diameter was selected to reduce the drum length.

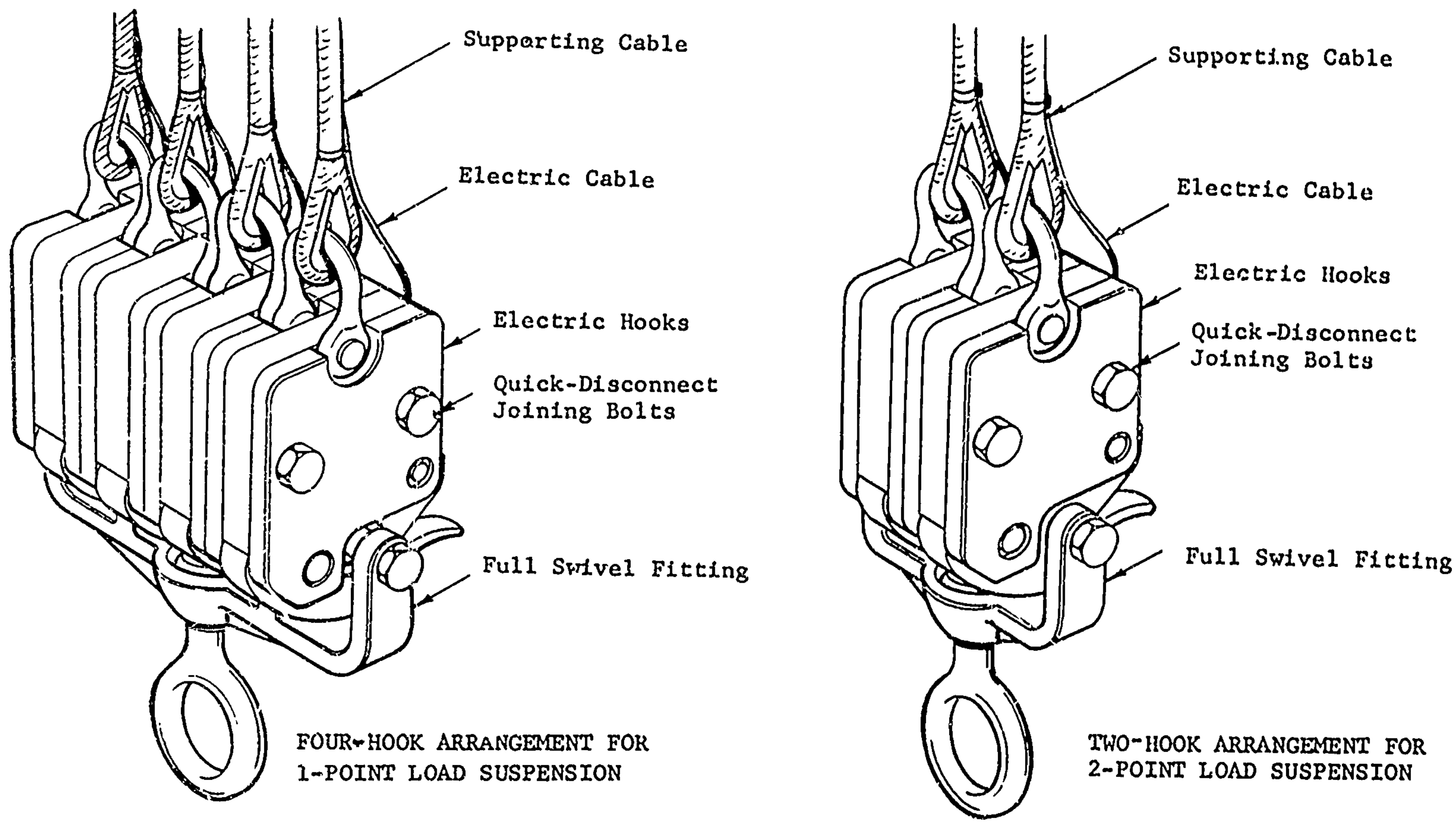


Figure 15. Hook Arrangements

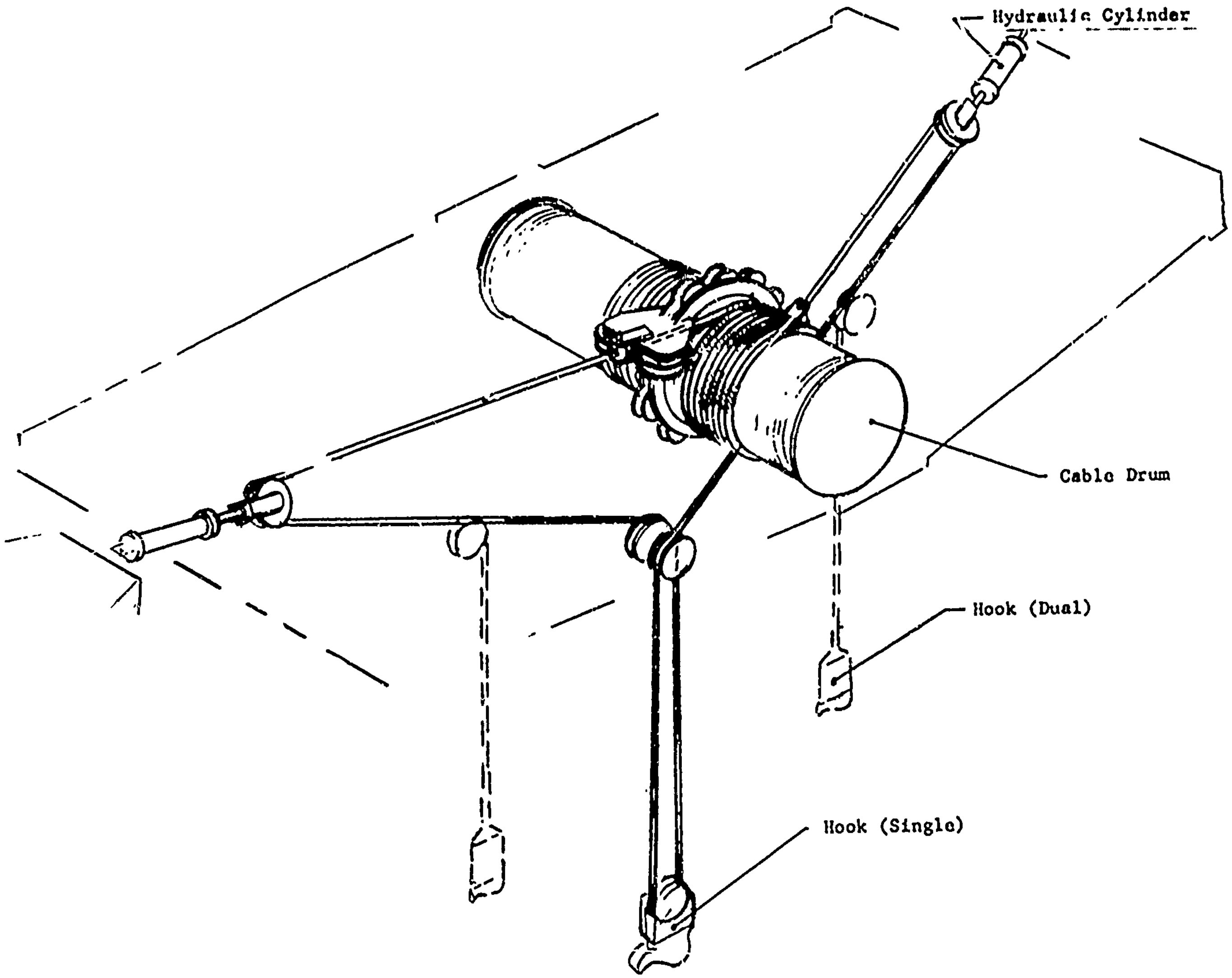


Figure 16. 1- and 2- Point Systems Only

The internal drive system would not differ from the one previously discussed. The electrical and hydraulic power requirements would not change.

The electrical and hydraulic control system would be simplified by the elimination of duplicate controls.

It would probably be necessary to design a coaxial torsilastic system to save space inside the drum. This could be accomplished without any particular difficulty.

There would be considerable savings in weight. They are as follows:

The 1- and 2-point 150-ft. lift will weigh - 3600 lbs.
The 1- and 2-point 80-ft. lift will weigh - 3220 lbs.

In conclusion, there would be significant savings in weight and space; however, there would be the obvious curtailment in the versatility discussed earlier in this report.

The basic 1- and 2-point systems are shown in Figure 16. The top and side views of the 4-point drum system are depicted in Figure 17.

EFFECTS OF EXTERNAL LOADS ON HELICOPTER DYNAMICS

DISCUSSION

The presence of external loads can have a detrimental influence on helicopter handling, stability and vibration. The effects of external loads are largely dependent upon the dynamic characteristics of the hoist for a particular helicopter/load configuration. It is the purpose of this section of this report to optimize the dynamic properties of the hoist so that problems associated with helicopter handling, stability and vibration are minimized.

A comprehensive discussion of the effects of external loads on helicopter dynamics is provided in Reference 1. The most serious problem involves the phenomenon of "vertical bounce". Divergent vertical oscillations have been observed on all major helicopter configurations in hover and forward flight. They consist of rapidly divergent vertical oscillations of moderately high frequency (3 to 4 cycles per second) which are completely uncontrollable by any deliberate maneuvering of the pilot's collective stick. Although this problem is common to all helicopters, it can be greatly aggravated if the natural frequency of the suspended load coincides with the frequencies of the vertical oscillations. As a consequence, it is recommended that the hoist be designed with a spring rate which will produce a natural frequency no higher than 2.5 cycles per second with the maximum design load attached.

A secondary problem associated with external loads is the aggravation of helicopter airframe vibration generated by the rotor system. Since airframe vibration has an important effect on pilot and crew comfort, as well as on structural fatigue, it is imperative that the airframe vibration be kept within tolerable limits. Levels of aircraft vibration are established by (1) the dynamic characteristics of the fuselage-cargo and (2) the vibratory load excitation. External cargo loads significantly change the fuselage dynamic system because basic aircraft modes of rigidity and flexibility are coupled to the cargo with additional degrees of freedom. Vibratory loads exciting the fuselage-cargo system are primarily attributable to rotor blade excitation. Since the rotor in its operation can have several harmonics contained in flapping and lagging motion, and since the rotor can create added frequencies in the transfer of forces to the fuselage, a large range of excitation forces is present. These excitation forces will range from the rotor speed, Ω, due to simple unbalance, to higher frequencies of 2Ω, 3Ω, etc., depending on the number of rotor blades. In the case of the heavy-lift helicopter rotor configurations described in References 2 through 4, the predominate rotor blade excitation frequencies fall in a range from 2 to 15 cycles per second.

In order to evaluate the dynamic characteristics of the hoist in terms of the severity of helicopter vibration, and stresses in the hoist, a mathematical model will be constructed which accounts for the important factors affecting the dynamic response of the helicopter-hoist-load system. To accomplish this investigation, a two-degree-of-freedom system will be used,

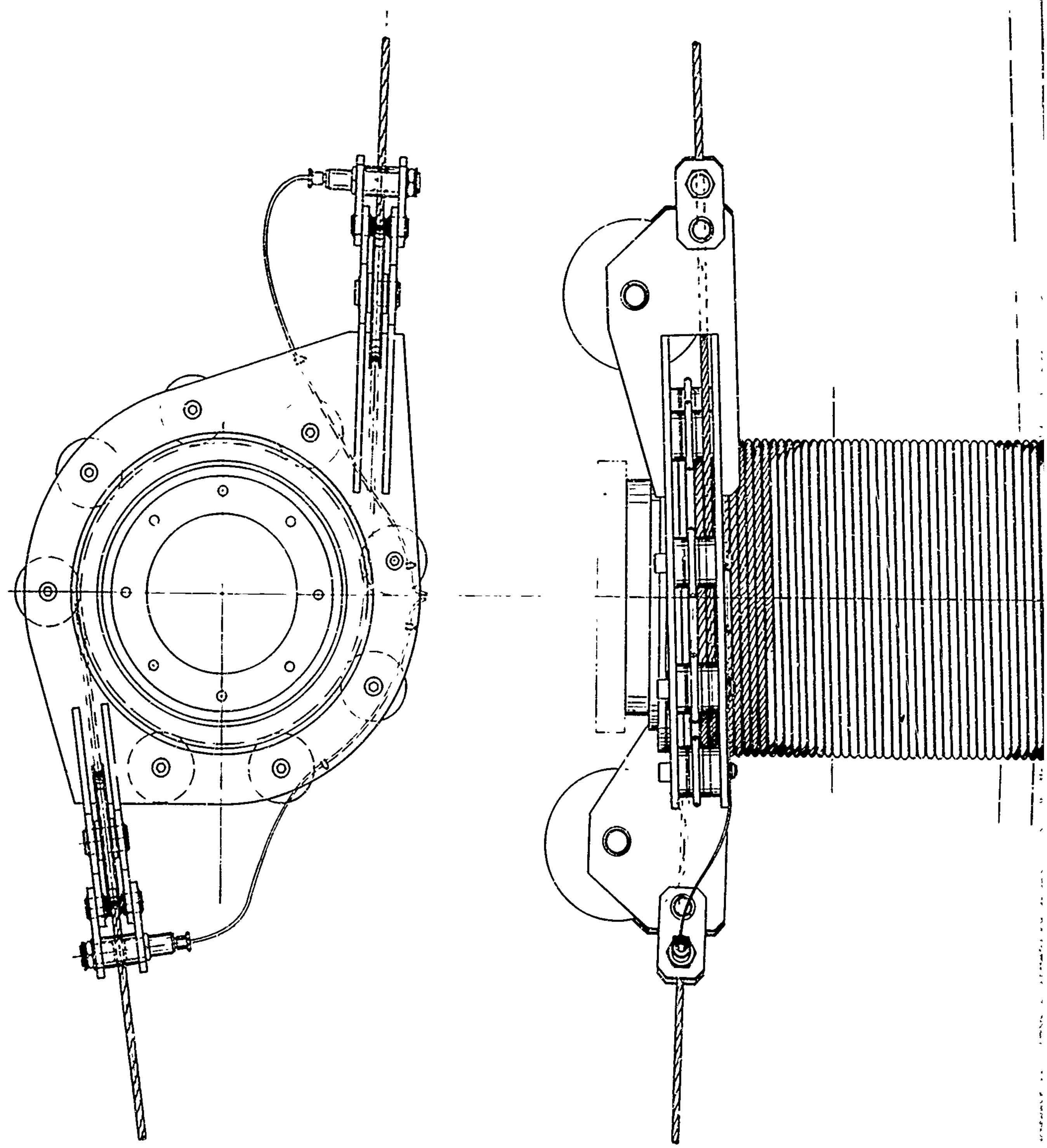
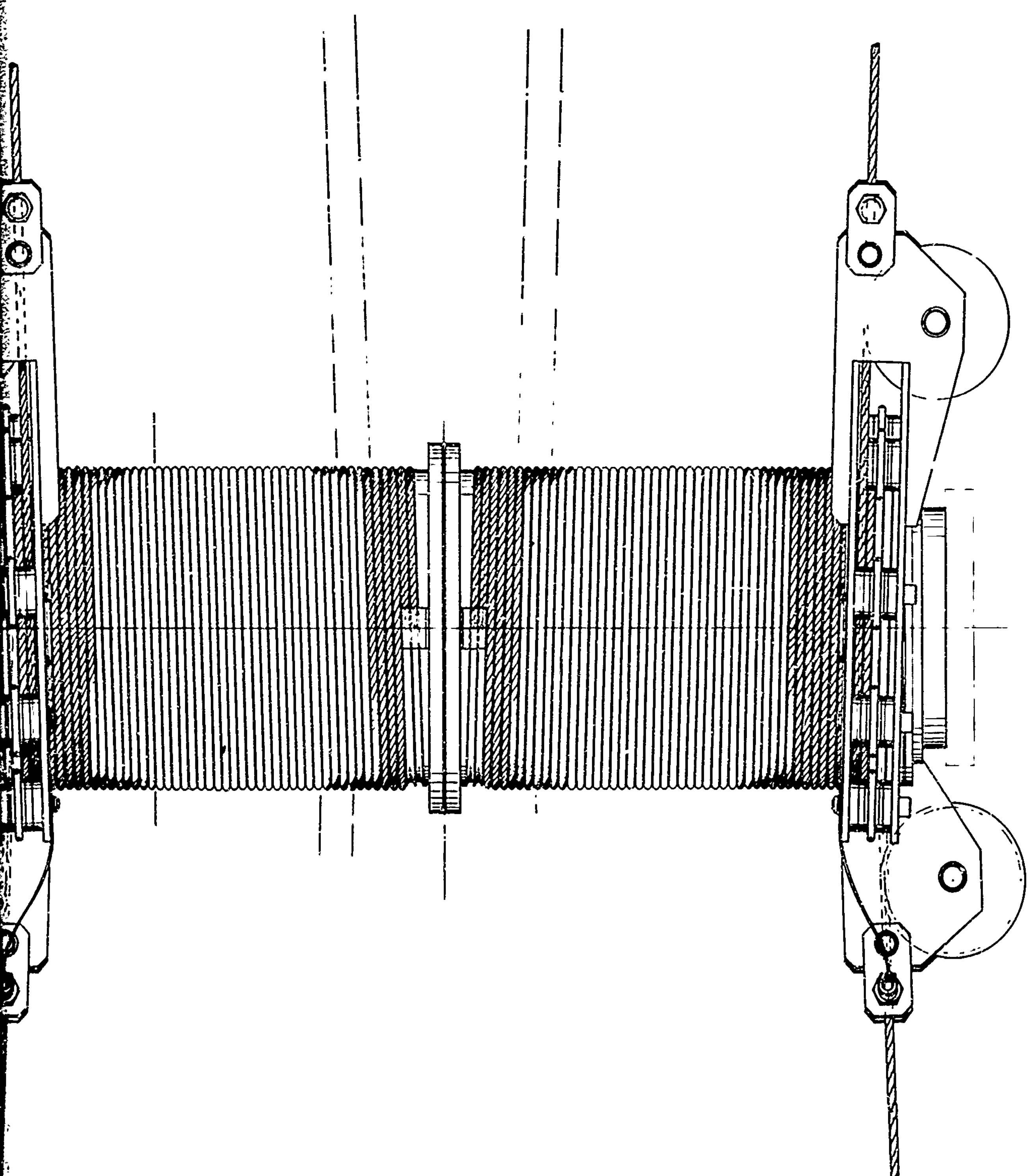


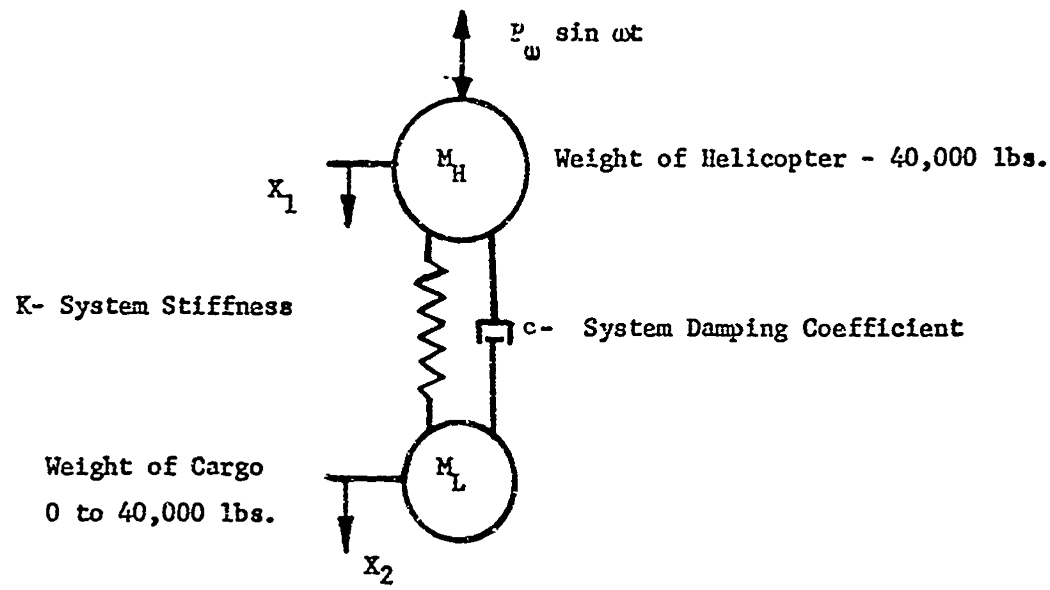
Figure 17. Top and Side Views of Drum System



B

consisting of a cargo mass attached by a flexible hoist cable to a rigid fuselage.

Mathematical Model



Both the helicopter and the load are treated as rigid masses. The weight of a helicopter capable of lifting a 40,000-pound payload is estimated to be approximately 40,000 pounds, based on heavy-lift helicopter studies performed in References 2 through 4. The weight of the load can vary from 0 to 40,000 pounds. For a constant hoist stiffness, the natural frequency of the system is a function of the load. Therefore, a range of payload weights will be examined to evaluate this effect on the helicopter vibration environment and on the load and stresses in the hoist system.

The spring joining the helicopter mass and the load mass represents significant flexible elements of the system which have been identified as the hoist cables and the vibration isolator.

The dash pot joining the helicopter mass and the load mass represents the system damping, which is assumed to be viscous. The damping characteristics of the isolator are used to approximate the damping characteristics of the system.

The vibrational forces applied to the helicopter could be computed from the rotor system characteristics. However, this involves a considerable amount of work, and a rotor system for the heavy-lift helicopter has not yet been finalized. As an alternate approach, these vibrational forces will be computed from the vibration spectrum presented in Figure 18. This spectrum is an envelope of vibration measurements made on numerous helicopters as indicated in References 5 through 7. Since this spectrum includes measurements from aircraft such as the CH-47, it should also be applicable to heavy-lift helicopter configurations proposed in References 2 through 4. The objective of the dynamic analysis is to determine the effect of the external load on the helicopter vibration loads. As a consequence, the absolute amplitudes of this vibration are of secondary importance.

The displacement spectrum shown in Figure 18 can be converted into a force spectrum using the following relationship:

$$P_{(\omega)} = M_H X_{(\omega)} \omega^2 \qquad (1)$$

where ω is the circular frequency of vibration in rad/sec

M_H is the helicopter mass

$X_{(\omega)}$ is the displacement amplitude at frequency ω

$P_{(\omega)}$ is the force amplitude at frequency ω

The resultant force spectrum is shown in Figure 19.

A solution to the differential equation of motion for the idealized model results in the following maximum displacement amplitudes for the helicopter and external load:

$$X_{1\ max} = P_\omega \sqrt{\frac{(-M_L\omega^2+K)^2 + (\omega c)^2}{(M_L M_H\omega^4 - M_L\omega^2 K - M_H\omega^2 K)^2 + (-M_L\omega^3 c - M_H\omega^3 c)^2}} \qquad (2)$$

$$X_{2\ max} = P_\omega \sqrt{\frac{(-K)^2 + (\omega c)^2}{(M_L M_H\omega^4 - M_L\omega^2 K - M_H\omega^2 K)^2 + (-M_L\omega^3 c - M_H\omega^3 c)^2}} \qquad (3)$$

The maximum relative amplitude of the two masses can be approximated from the following expression:

$$X_{Rel} = \frac{P_\omega X_{1(max)}}{c\omega} \qquad (4)$$

Neglecting the effects of damping, the circular frequency of vibration (ω) for the idealized system can be computed from the following expression:

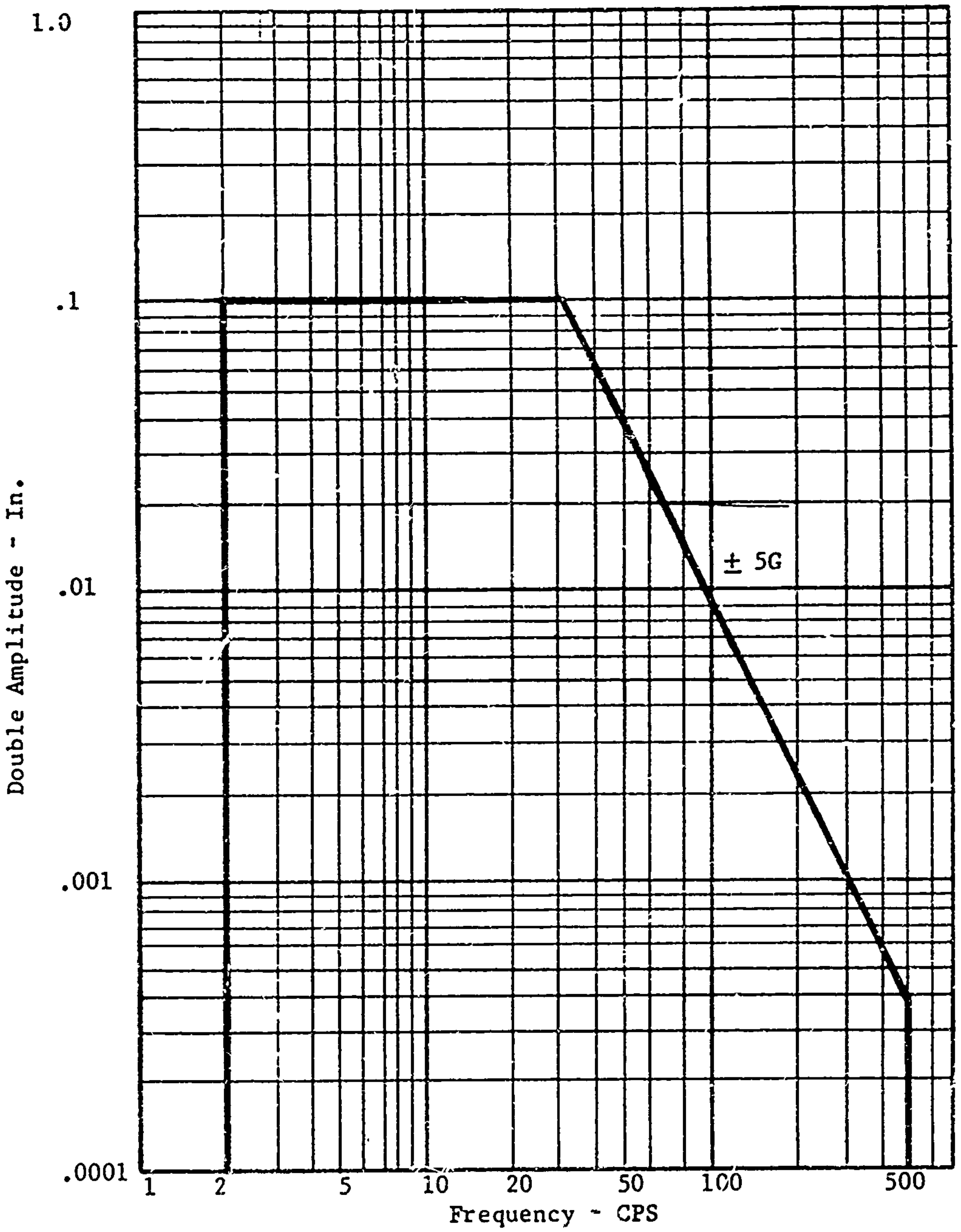


Figure 18. Vibration Spectrum for Helicopter in Flight

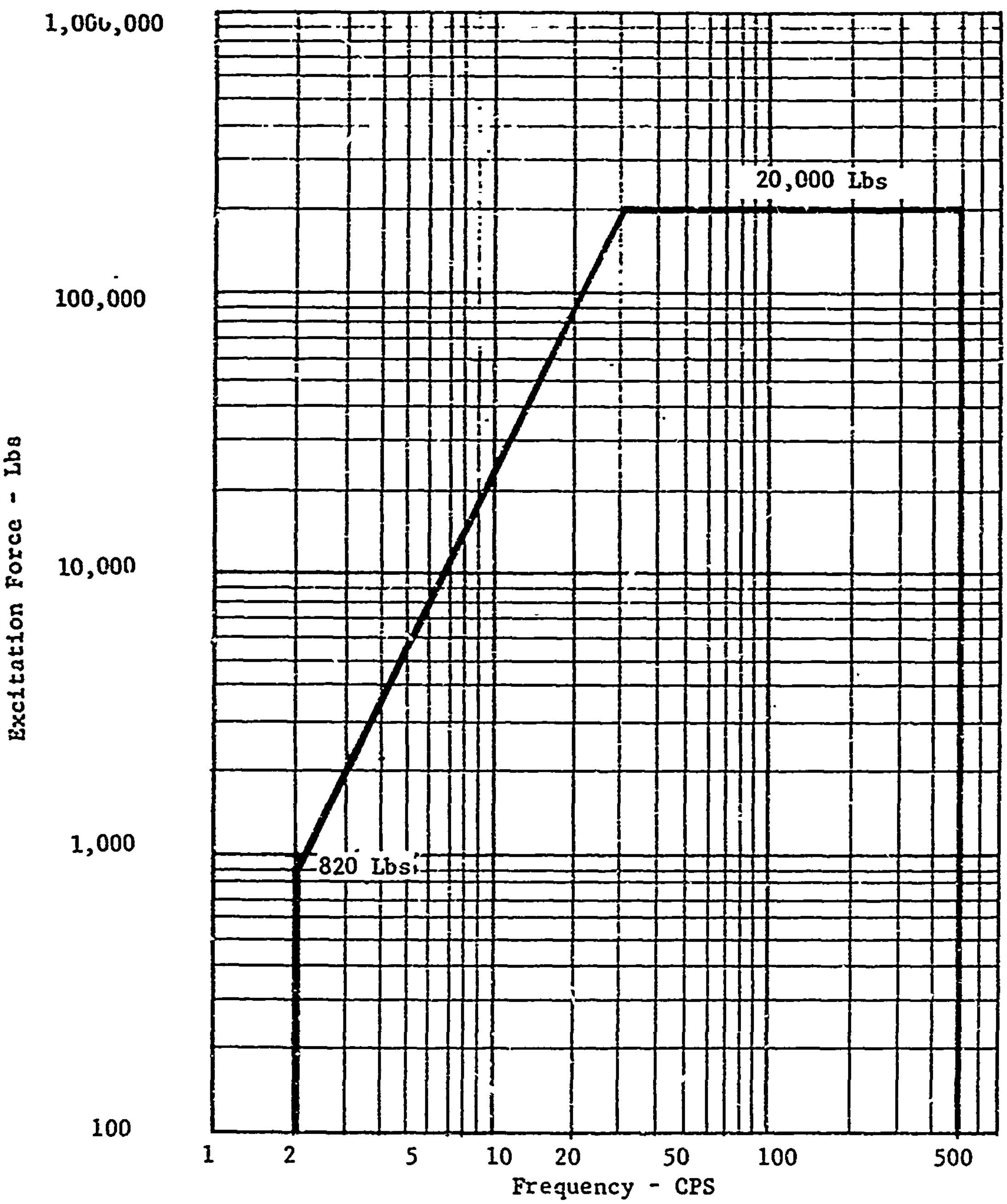


Figure 19. Force Spectrum for Helicopter in Flight

$$\omega = \sqrt{\frac{K\,(M_H + M_L)}{M_H\ M_L}} \qquad (5)$$

Efforts to optimize the dynamics of the helicopter-hoist load system centered primarily on two characteristics of the system.

The first characteristic was the natural frequency of the system. An effort was made to minimize the natural frequency of the fully loaded system so that its fundamental resonance would be below 2 cps, which is the lower limit of the excitation forces. From equation 5 it is seen that the natural frequency is a function of the stiffness of the system as well as the mass of the helicopter and load. Since the masses are fixed by design requirements, it is necessary to reduce the stiffness of the system in order to lower the natural frequency. The stiffness of the hoist cables is basically fixed by design requirements, leaving the stiffness of the isolator as the primary variable affecting the natural frequency of the system.

The second system characteristic considered was damping. Since amplification of the helicopter vibration at a resonance is inversely proportional to the amount of damping in the system, it is desirable to maximize this property of the system. Again, the isolator is the major contributor to the damping characteristics of the system.

The primary constraints on the system which limit the natural frequency and the damping characteristics are: the mass of the helicopter and load, the magnitude of the load and required safety factors, and the envelope and material property limitations of the isolator.

Within these constraints, numerous isolator configurations were examined. It was determined that two rubber springs in series would minimize both the vibration-induced loads in the hoist and amplification of the helicopter motion.

TORSILASTIC SPRING DYNAMIC ANALYSIS

Isolation of shock and vibration loads in the system will be accomplished by the use of a device termed a torsilastic, which is made by the B. F. Goodrich Company. The device consists of two coaxial metal cylinders, bonded together by a layer of rubber. Either cylinder can be connected to the load. In the present design, the outer cylinder is connected to the drum (load) and the inner one to the hydraulic motor, via the reduction gear. When a torsional load is introduced into the outer cylinder, the rubber deflects torsionally, damping the impact and carrying the torsion to the inner cylinder. High hysteresis loss reduces feedback of shock and at the same time dampens oscillation. Placement of the torsilastic in the system is such as to also dampen starting and stopping shock loads introduced by the hydraulic motor.

A typical torsilastic rubber-bushed spring is shown in Figure 20.

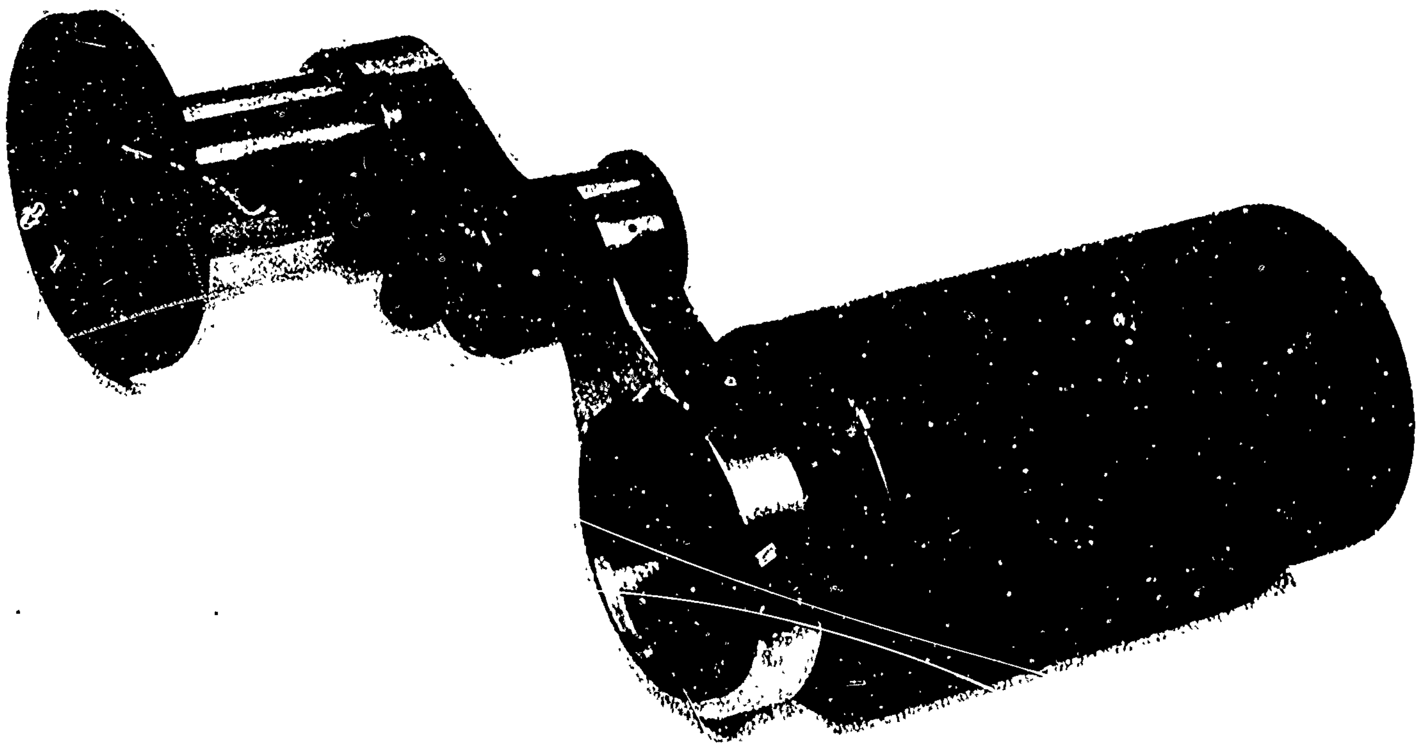


Figure 20. Torsilastic Rubber-Bushed Spring as Used in a Tank Suspension System

Torsilastic Spring Stiffness

The dynamic stiffness of the two torsilastic springs in series can be computed using the following equation (35.21 of Reference 8.):

$$K_{dt} = \frac{4 \pi L G_d}{n\left(\frac{1}{R_I^2} - \frac{1}{R_o^2}\right)} \tag{6}$$

where,

K_{dt} = dynamic stiffness in-lbs/radian

L = length of spring = 21 in.

G_d = dynamic shear modulus = 135 psi

R_I = inside radius = 6 in.

R_o = outside radius = 8 in.

n = number of springs in series = 2

The dynamic shear modulus G_d for a rubber with a durometer hardness of 60 is given in Table IX-1 of Reference 9 as 135 psi.

$$K_{dt} = \frac{(4)(\pi)(21)(135)}{2\left(\frac{1}{6^2} - \frac{1}{8^2}\right)}$$

$$K_{dt} = 1{,}470{,}000 \text{ in.-lbs/rad.}$$

Cable Stiffness

The cable stiffness is a maximum for the two-point suspension system when the load is full up. For this condition, the average effective cable length is approximately 300 inches. For 5/8-inch-diameter, 6 x 37 IWRC stainless steel cable,

$$K_c = \frac{AE}{L} \tag{7}$$

where,

K_c = cable stiffness, in.-lbs

A = area of cables = 1.22 in.^{2}

E = modulus of elasticity = 12×10^6 psi

L = length of cable = 300 in.

$$K_c = (1.22)(12 \times 10^6)/300$$

$$K_c = 49{,}000 \text{ lbs/in.}$$

The resultant stiffness of the torsilastic spring and cable can be computed from the following expression:

$$K_R = \frac{1}{\frac{1}{K_c} + \frac{r_d^2}{K_{dt}}} \qquad (8)$$

where,

r_d = drum radius, in.

$$K_R = \frac{1}{\frac{1}{49{,}000} + \frac{(15.38)^2}{1{,}470{,}000}}$$

$$K_R = 5{,}510 \text{ lbs/in.}$$

From equation 8, it is apparent that the torsilastic spring is considerably softer than the cables.

The natural frequency of the system from equation 5 is

$$f_n = 1/2\pi \sqrt{K_R(M_H + M_L)/\,M_H \times M_L} \qquad (9)$$

$$f_n = 1/2\pi \sqrt{(5510)(103.7 + 103.7)/(103.7)(103.7)} \qquad (10)$$

$$f_n = 1.65 \text{ cps or } 99 \text{ cpm}$$

This is well below the 2-cps lower limit of the assumed helicopter excitation forces.

The damping coefficient for the system can be computed from the following expression: (See Reference 10)

$$c = K\,\Delta/\pi\omega \qquad (11)$$

The logarithmic decrement (Δ) for a rubber durometer hardness of 60 is given as 0.39 in Figure 5.16 of Reference 10.

$$c = (5510)(.39)/3.14\omega = 685/\omega \text{ or } 685/2\pi f$$

Having defined the stiffness and damping properties of the system, equations 2 and 3 can be solved to determine the response of both the helicopter and suspended load, and subsequently the vibration-induced forces in the hoist cable. These equations will be solved for a range of suspended loads from 500 to 40,000 pounds.

The input data necessary to solve these equations are summarized in Table IV. For Cases 2 through 8, the frequency of the excitation forces is assumed to correspond to the natural frequency of the system, resulting in a condition of resonance. In Case 1, a maximum load of 40,000 pounds is suspended by the helicopter. For this condition the response of the system is computed for the minimum frequency of the excitation forces (2-cps). Since the natural frequency of the system is 1.65 cps for this case, a condition of resonance cannot exist.

TABLE IV. DATA INPUT FOR RESPONSE OF HELICOPTER AND LOAD TO EXCITATION FORCES

Case	Natural (1) Frequency (cps)	Forcing (2) Frequency (cps)	Exciting (2) Force - P_ω (lbs)	Lifted Load - W_L (lbs)	Damping (3) Coefficient c
1	1.65	2.00	820	40,000	54.50
2	2.00	2.00	820	20,329	54.50
3	2.50	2.50	1,280	11,000	43.60
4	3.00	3.00	1,840	7,047	36.40
5	4.00	4.00	3,280	3,680	27.30
6	5.00	5.00	5,100	2,280	21.80
7	6.00	6.00	7,360	1,556	18.17
8	10.00	10.00	20,500	547	10.90

where $K = 5{,}510$ in.-lbs (all cases)

$M_H = 103.6$ lbs-sec^2/in. (all cases)

(1) See Equation 5.
(2) See Figure 2.
(3) See Equation 11.

Equations 2 and 3 were solved on AAI's GE-415 computer for several sets of data. The results for the final configuration are summarized in Table V. It is apparent that the helicopter motion in maximized at a resonance of 2 cps. At this frequency the motion of the helicopter is increased from .050 inch to .140 inch as the result of suspending a 20,300-pound load. Actually, the lowest 1-revolution frequency for any of the rotor configurations of References 2 through 4 is 2.4 cps. At this frequency the response of the helicopter would be approximately .100 inch.

In the critical frequency range of 3 to 4 cps where vertical bounce is a problem, the helicopter motion is increased approximately 70 percent. Above 5 cps the suspended load has no appreciable effect on the level of vibration in the helicopter.

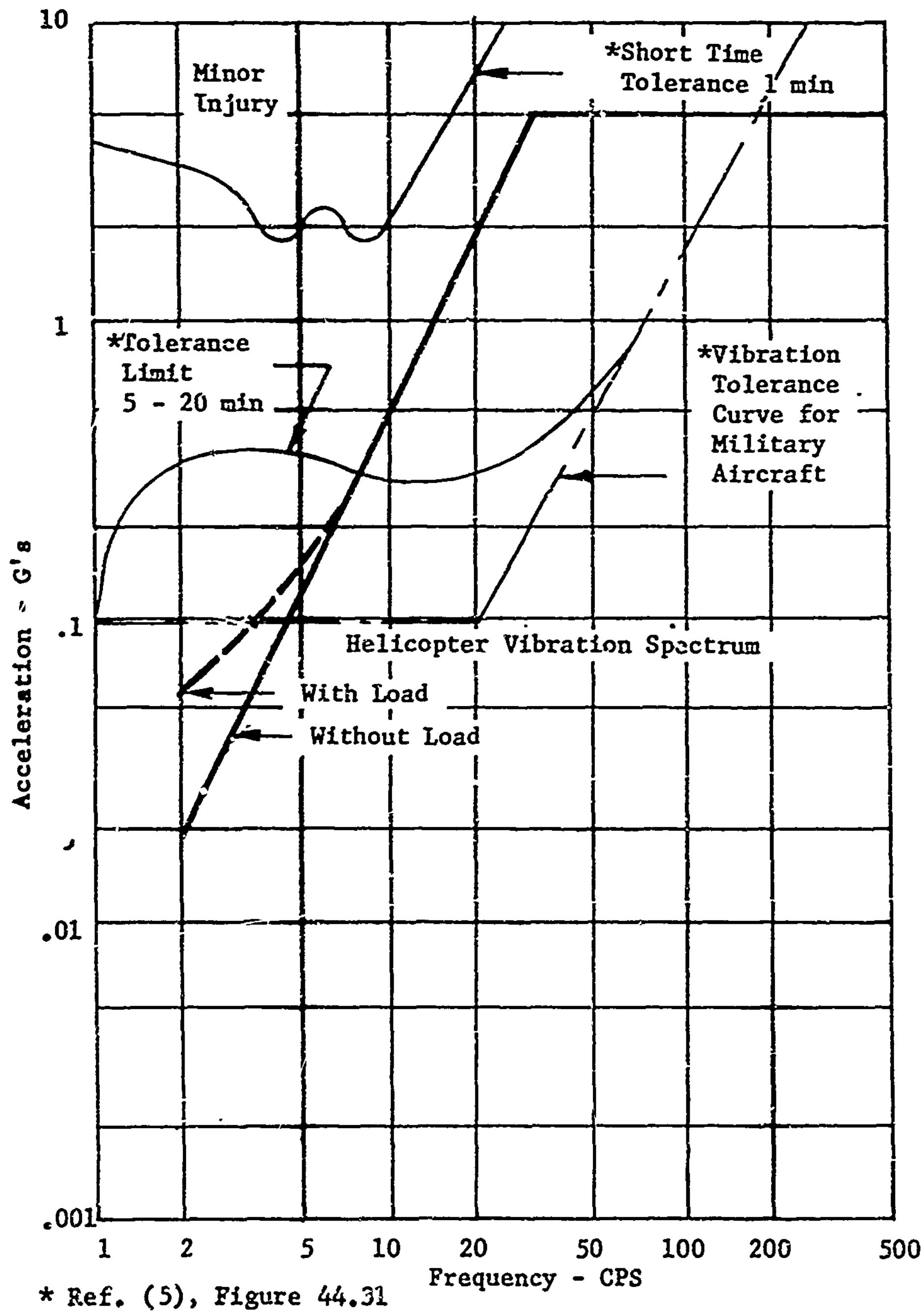
The acceleration level in the helicopter fuselage is plotted in Figure 21 with and without an external load. Below 4 cps the acceleration level is less than the vibration tolerance curve for military aircraft, and well below the 5- to 20-minute tolerance curve presented in Figure 44.31 of Reference 6.

When the helicopter is supporting the maximum load of 40,000 pounds, the increase in cable load due to vibration is less than 2 percent. With an ultimate safety factor of 3.75, it is quite apparent that vibration-induced loads on the hoist can be safely neglected in the stress analysis of the hoist components.

TABLE V. RESPONSE OF HELICOPTER AND LOAD TO EXCITATION FORCES

Case	System Natural Frequency f_n (cps)	Forcing Frequency f (cps)	Helicopter* Motion $X_{1(max)}$ (in.)	Load* Motion $X_{2(max)}$ (in.)	Relative* Motion X_{Rel} (in.)	Lifted Load W_L (lbs)	Cable Load Vibration KX_{Rel} (lbs)	Maximum Cable Load $W_L + KX_{Rel}$ (lbs)
1	1.65	2.00	.099	.050	.12	40,000	660	40,660
2	2.00	2.00	.140	.270	.17	20,300	940	21,240
3	2.50	2.50	.095	.319	.22	11,000	1,210	12,210
4	3.00	3.00	.074	.344	.27	7,050	1,490	8,540
5	4.00	4.00	.057	.372	.32	3,680	1,760	5,440
6	5.00	5.00	.052	.383	.39	2,280	2,150	4,430
7	6.00	6.00	.050	.390	.44	1,560	2,420	3,980
8	10.00	10.00	.050	.401	.45	545	2,480	3,025

* Single Amplitude



* Ref. (5), Figure 44.31

Figure 21. Helicopter Vibration Spectrum With and Without Suspended Load

STRESS ANALYSIS

DESIGN CRITERIA FOR HELICOPTER HOIST

The purpose of this analysis is to investigate the structural integrity of the most critical load-carrying areas of the heavy-lift helicopter hoist. The design criteria for the hoist are as follows:

(1) The rated load of the system is 40,000 pounds.
(2) The dynamic load factor is 2.5 g's.
(3) A safety factor of 1.5 is used, based on the ultimate strength of the material.
(4) The design load P is 40,000 x 2.5 x 1.5 = 150,000 lbs.

All margins of safety are based on the following criterion:

$$M.S. = \frac{\text{ultimate stress}}{\text{calculated stress}} - 1$$

DISCUSSION

For purposes of this analysis, the C.G. of the load is assumed to be at the geometrical center of the cable array. The load distribution under this assumption is such that the load is equally distributed to each cable. However, it is realized that this assumption is somewhat idealistic and that the cable geometrical center and the load C.G. will not coincide in many cases. This will result in unequal load distribution in the cables, but because of the lack of more definite information about the magnitude of this shift, it was decided to base the analysis on equal cable load distribution. When definite information on the maximum allowable shift becomes available, the capacities of the affected components can be adjusted to accommodate the new loads. It is estimated at this time that this will entail a relatively modest effort.

The 150,000-pound design load P is equally placed on the four cables, thus making the load per cable equal to 37,500 pounds. In order to make the analysis easy to follow, the components are investigated in sequence along the load path. This means that the loads are traced from the hook through the reeving and drum into the helicopter structure. Material allowables are presented for each component throughout the analysis.

The three systems to be examined are

(1) 1-point system
(2) 2-point system
(3) 4-point system

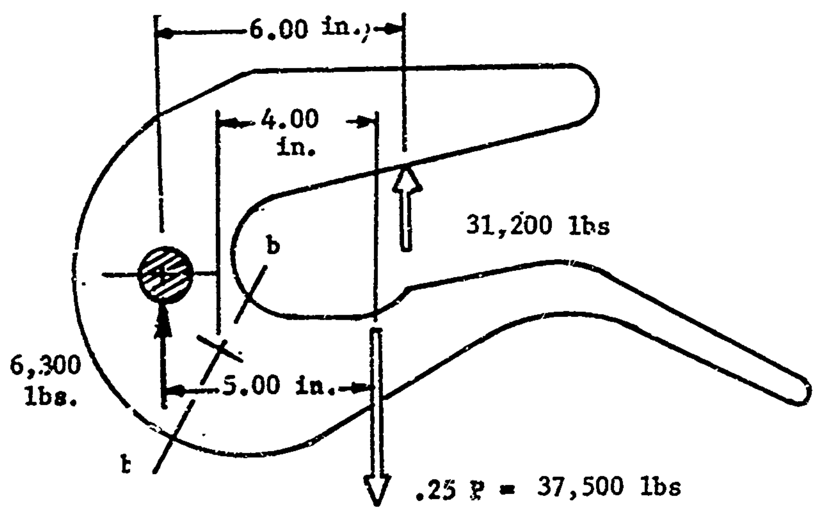
All three systems employ four ropes which carry the load to the drum. Each rope carries one-fourth of the 150,000-pound design load, or P/4 = 37,500 pounds. Use of Roebling 5/8-inch-diameter Royal Blue (IWRC 6 x 19) wire rope (rated 39,200 lbs) would yield a safety margin of 39,200/37,500, or 9 percent. This applies to ropes without an electrical conductor; hence, part of the extra margin will be used when the rope is made with an electrical center conductor.

Helicopter Hoist Analysis

The same hook is used in all three systems

(1) 1-point - four hooks joined together
(2) 2-point - two hooks joined together for each point
(3) 4-point - four single hooks

Load per hook, $P/4$ = 37,500 lbs



Bending moment at b-b, M_{b-b} = 150,000 in.-lbs

$A_{(1)} = 1.9 \text{ in.}^2$,

$A_{(2)} = 3.75 \text{ in.}^2$,

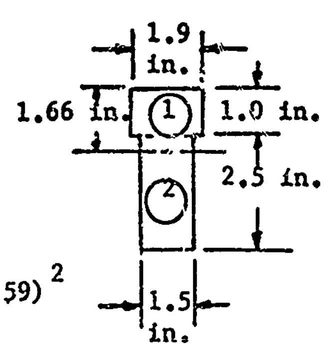
$I_{(1)} = .16 \text{ in.}^4$

$I_{(2)} = 1.95 \text{ in.}^4$

$I_{b-b} = .16 + (1.9)(1.16)^2 + 1.95 + (3.75)(.59)^2$

$I_{bb} = 5.76 \text{ in.}^4$,

$S_{b-b} = 3.06 \text{ in.}^3$



Section b-b

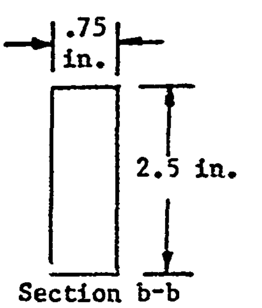
Bending stress in hook at Section b-b; $f_b = M_{b-b}/S_{b-b} = 150{,}000/\ 3.06 =$ 49,000 psi. If a low carbon steel hook is used, AISI 1015-1025 (Ref. 11) $F_{tu} = 55{,}000$ psi.

$$M.S. - (55{,}000/49{,}000) - 1$$

$$\boxed{M.S. = +.12}$$

If a high-strength alloy steel hook is used, weight can be reduced. AISI 4340 steel, heat treated to $F_{tu} = 200{,}000$ psi.

Section Modulus Required:



Section b-b

$$S_{b-b} = M_{b-b}/F_{tu} = 150{,}000/200{,}000$$

$$S_{b-b} = .75 \text{ in.}^3$$

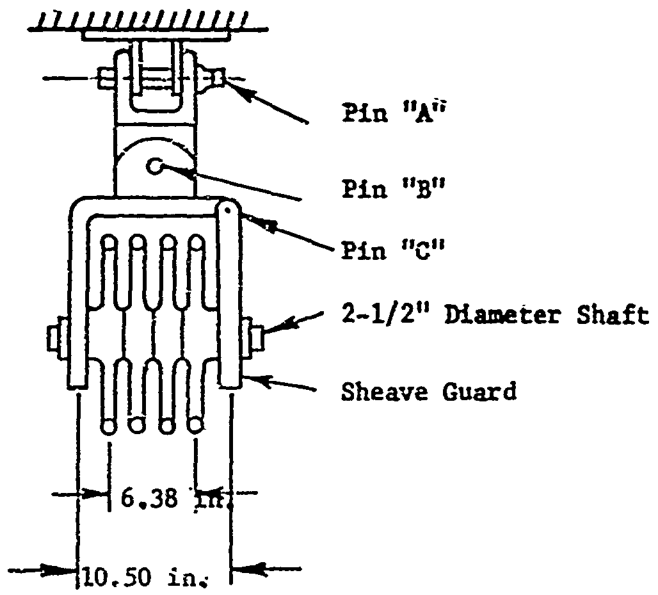
$$S_{b-b} = bh^2/6 = (.75)(2.5)^2/6 = .78 \text{ in.}^3$$

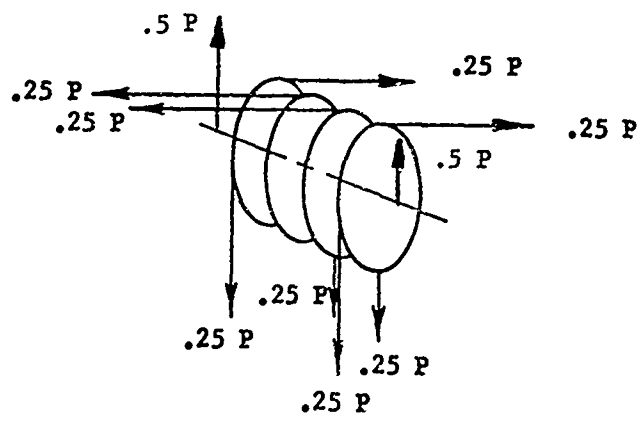
$$\boxed{M.S. = +.04}$$

The estimated weight savings on four alloy steel hooks and their housings is 300 pounds per hoist.

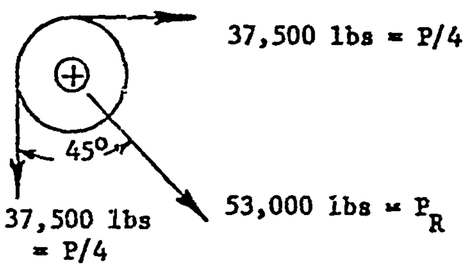
SHAFT ANALYSIS (SHEAVE)

Single-Point Yoke

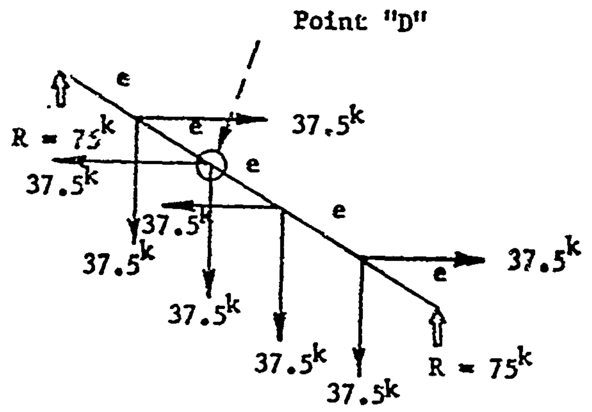




One quarter of the design load P applied by one sheave to the shaft results in the following load distribution.



Sheave Loads



Shaft Load Diagram

e = 2.12 in

Bending in Vertical Plane

$M_{v(max)}$ at point "D" = 239,000 in.-lbs

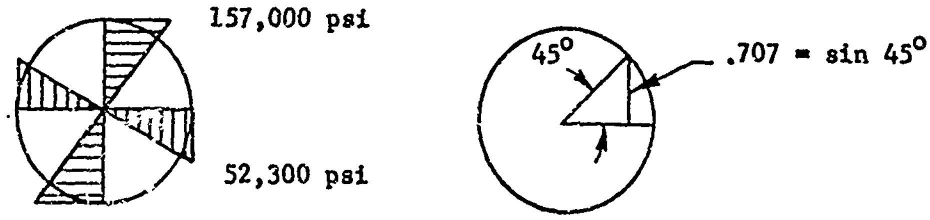
$S = 1.53 \text{ in.}^3$

$f_{b_v} = M_{v(max)}/S = 239,000/1.53 = 157,000$ psi

Bending in Horizontal Plane

$M_{h(max)}$ at point "D" = 80,000 in.-lbs

$f_{bh} = M_{h(max)}/S = 80,000/1.53 = 52,300$ psi



Combined Bending Stress at Point "D", f_{bc}

$$f_{bc} = \left[(.707\ f_{bv})^2 + (.707\ f_{bh})^2\right]^{0.5} = 117,000 \text{ psi}$$

Shaft Shear at Point "D"

Area of 2.5 in. diameter - A = 4.91 in.^{2}

f_s = 53,000/4.91 = 10,800 psi

Combined Bending and Shear at Point "D"

$$f_{s(max)} = \left[(f_{bc})^2/\ 2 + (f_s)^2\right]^{0.5} = \left[(58,500)^2 + (10,800)^2\right]^{0.5}$$

$$= 59,500 \text{ psi}$$

$$f_{n(max)} = f_{bc}/2 + f_{s(max)} = 58,500 + 59,500 = 118,000 \text{ psi}$$

F_{tu} = 160,000 psi M.S. = (160,000/118,000)-1

M.S. = + .35

Yoke and Clevises

Material HK31A - H24 - Magnesium

F_{tu} = 34,000 psi (Ref. 12)

F_{ty} = 25,000 psi (Ref. 12)

F_{su} = 27,000 psi (Ref. 12)

F_{bru} = 57,000 psi (Ref. 11)

E = 6,500,000 psi (Ref. 11)

Bearing Stress due to Shaft Load

Bearing load R = 75,000 lbs

Bearing area = 1.87 in.^{2}

f_{bru} = 40,000 psi

M.S. = (57,000/40,000)-1

M.S. = + .42

Shear in Pin "C"

Load R = 75,000 lbs - double shear

37,500 lbs - single shear

Area of .88 in. dia. pin = .601 in.^{2}

f_s = 37,500/.601 = 62,400 psi

F_{tu} = 125,000 psi, F_{su} = 82,000 psi (Ref. 11)

M.S. = (82,000/62,400) - 1

$\boxed{\text{M.S. = + .31}}$

Bearing in Magnesium Sheave Guard at Pin "C"

Load P/4 = 37,500 lbs

A = .88 in.^{2}

f_{bru} = 37,500/.88 = 42,600 psi

M.S. = (57,000/42,600)-1

$\boxed{\text{M.S. = + .33}}$

Shear in Pin "B" and Pin "A"

F_{tu} = 180,000 psi (Ref. 11)

F_{su} = 108,000 psi

Load on pin, P = 150,000 lbs.

Area of 1.38 in. dia. pin = 1.485 in.^{2}

In double shear,

f_s = 75,000/1,485 = 50,500 psi

M.S. = (108,000/50,500) -1

$\boxed{\text{M.S. = + 1.16}}$

Bearing in Plate "E"

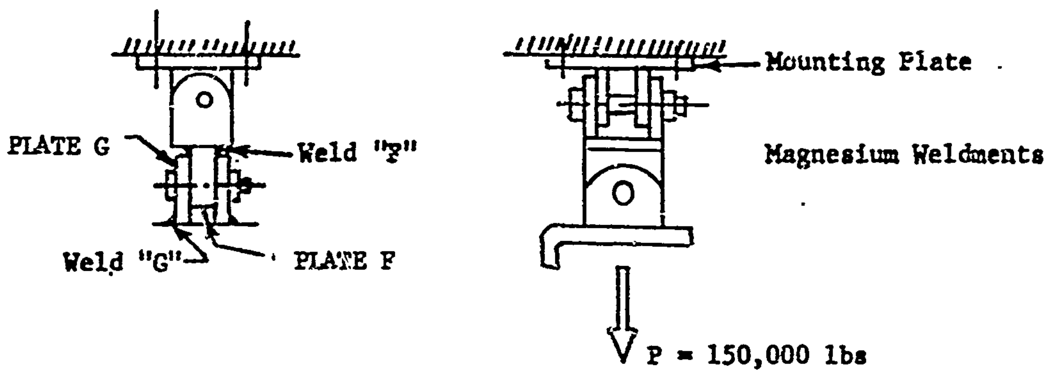


Plate thickness, 2.00 in Magnesium

$P = 150{,}000$ lbs

$A = 2.63$ in.^{2}

$F_{bru} = 57{,}000$ psi (Ref. 11)

$f_{bru} = 150{,}000/2.63 = 57{,}000$ psi

M.S. = (57,000/57,000)-1

$\boxed{\text{M.S.} = +.00}$

Therefore, make plate "F" and "G" 1 in. thick.

Weld "F"

Area of 8" long 3/4" fillet weld = 8 (.707)(.750) = 4.23 in^2

Weld efficiency 66%

Load 75,000 lbs

Stress in weld $f_\omega = 75{,}000/(.66)(4.23) = 26{,}900$ psi

$F_{su} = 27{,}000$ psi (Ref. 12)

M.S. = (27,000/26,900) -1

$\boxed{\text{M.S.} = +.003}$ Weld "F"

Weld "G"

Area of 24 in. long 1/2 in. fillet weld = 24(.707)(.500)

$A = 8.48 \text{ in.}^2$

Weld efficiency 66%

Load 150,000 lbs

$f_{tb} = 150{,}000/(.66)(8.48) = 26{,}800$ psi

M.S. = (27,000/26,800)-1

$\boxed{\text{M.S.} = +\ .007}$ Weld "G"

Welds "F" and "G" are the most highly stressed welds.

Yoke Mounting Bolts

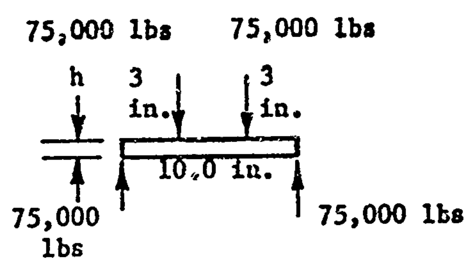
Each 3/4" diameter bolt must carry 37,500 lbs

Ultimate load 44,000 lbs on $F_{tu} = 125{,}000$ psi AN bolt (Ref. 11)

M.S. = (44,000/37,500)-1

$\boxed{\text{M.S.} = +\ .17}$ Mounting bolts in tension

Mounting Plate



Assume width of beam 10.00 in.

M_{max} = 75,000 (3)

= 225,000 in.-lbs

$F_{tu} = M_{max}/S_{req'd}$

$S_{req'd} = 225,000/34,000 = 6.62\ \text{in.}^3$

$b = 10$ in.

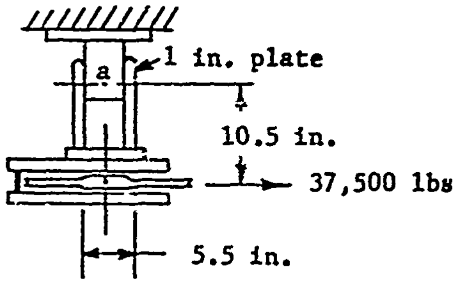
$S_{req'd} = bh^2/6$

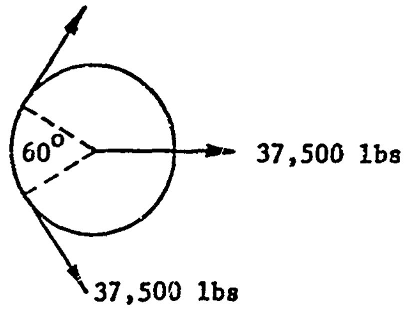
$6.62 = 10\ h^2/6$

$h = 1.998$

$\boxed{\text{Use 2 in. thick plate}}$

1-Point System





Moment at "a", T_a = 10.5 (37,500) = 394,000 in.-lbs

Reactions on pins = $T_a/5.5$ = 71,600 lbs

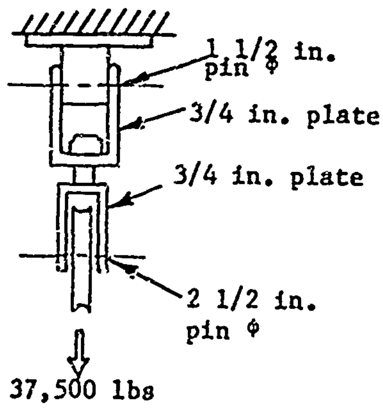
Bearing area 1.5 in.^{2}

f_{br} = 71,600/1.5 = 47,700 psi

F_{bru} = 57,000 psi M.S. = (57,000/47,700)-1

$\boxed{\text{M.S. } = +.19}$ 1-Pt Movable Sheave

2- and 4-Point System



Upper yoke critical in bearing

Load per plate - 18,750 lbs

Area 1.12 in.^{2}

f_{br} = 16,600 psi

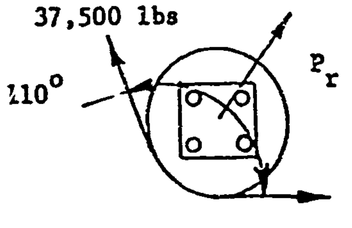
F_{bru} = 57,000 psi

M.S. = (57,000/16,660)-1

$\boxed{\text{M.S. } = +2.42}$ 2-Pt. on 4-Pt. Movable Sheaves

After the cables leave the movable sheaves, the remaining stresses are identical for all three systems.

Fixed Sheave



2-1/2" diameter tube in double shear, wall thickness 1/8"

Area .933 in.^{2}

f_s = 43,000/(2)(.933) = 23,100 psi

F_{su} = 55,000 psi 4130 steel seamless tubing (Ref. 11)

M.S. = (55,000/23,100)-1

$\boxed{\text{M.S. = + 1.38}}$ Shaft in shear

Mounting Bolts in Shear

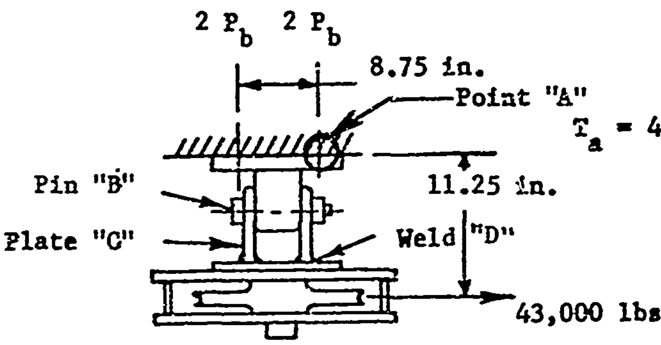
Four 3/4" diameter AN bolts carry 10,750 lbs each

Ultimate load is 33,150 lbs

M.S. = (33,150/10,750)-1

$\boxed{\text{M.S. = + 2.09}}$ Mounting bolts in shear

Fixed Sheave



T_a = 43,000(11.25) = 484,000 in.-lbs

Load per bolt = P_b

$(2\ P_b)(8.75)$ = 484,000 in.-lbs

P_b = 27,600 lbs tension load

Ultimate load - 44,000 lbs. (Ref. 11)

M.S. = (44,000/27,000)-1

$\boxed{\text{M.S.} = +\ .59}$ Mounting bolts in tension

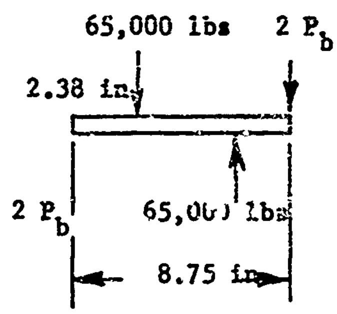
Combined Stresses

$$f_{s(max)} = \left[(f_t/2)^2 + (f_s)^2\right]^{0.5} = \left[(31,200)^2 + (24,400)^2\right]^{0.5} = 40,000 \text{ psi}$$

$$f_{n(max)} = f_t/2 + f_{s(max)} = 31,200 + 40,000 = 71,200 \text{ psi}$$

$\boxed{\text{M.S.} = +\ .75}$ Mounting bolts combined loading

Bending in Mounting Plate



$2\ P_b$ = 55,200 lbs

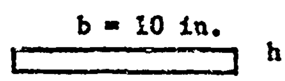
M_{max} = 126,000 in.-lbs

F_{tu} = 34,000 psi

Assume 10-inch wide beam

$S_{req'd} = M_{max}/F_{tu}$ = 126,000/34,000 = 3.7 $in.^3$

$S_{req'd} = bh^2/6$



$3.7 = 10h^2/6$

$h = 1.49$ in.

$\boxed{\text{Use 1.5 in. thick plate}}$

<u>Bearing due to Mounting in Shear</u>

Load = 10,750 lbs

Area = 1.125 in.^{2}

f_{br} = 9,550 psi

F_{bru} = 57,000 psi

M.S. = (57,000/9,550)-1

$\boxed{\text{M.S.} = +5.96}$ Plate in bearing

<u>Pin "B" in Shear</u>

.88 in. dia. F_{su} = 108,000 psi

P = 65,000 lbs

Area = .601 in.^{2}

f_s = 65,000/.601 = 108,000 psi

M.S. = (108,000/108,000)-1

$\boxed{\text{M.S.} = +.00}$ Pin "B" in shear

<u>Plate "C" in Bearing</u>

P = 65,000 lbs

Area = 1.312 in.^{2}

f_{br} = 49,500 psi

F_{bru} = 51,000 psi

M.S. = (57,000/49,500)-1

$\boxed{\text{M.S.} = +.15}$ Plate "C" in bearing

Weld "D"

Area of 15 in. long 3/4 in. fillet weld = (15)(.707)(.750) = 8.95 in.^{2}

Weld efficiency 66%

P = 65,000 lbs

F_{su} = 27,000 psi

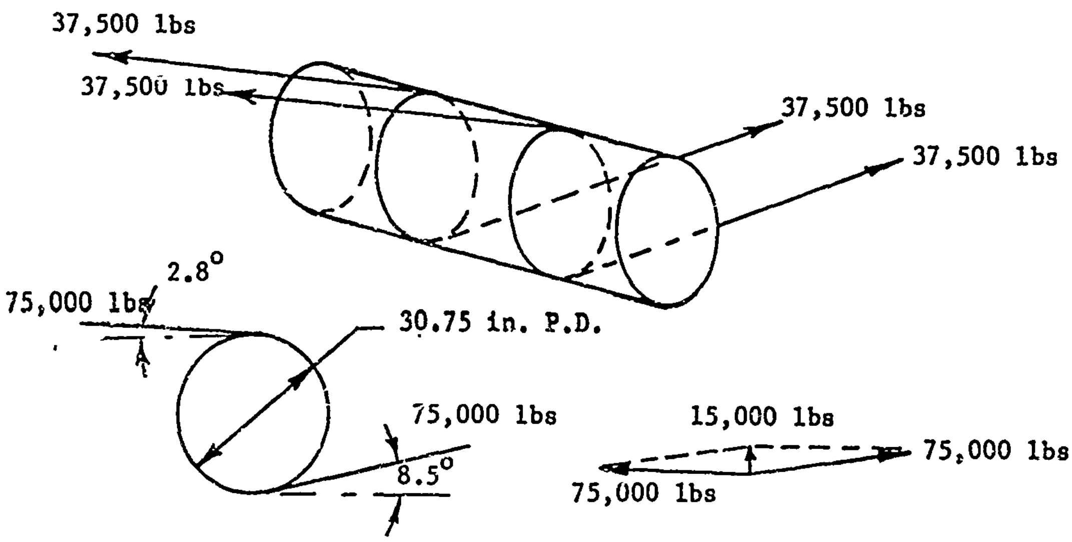
f_w = 65,000/(.66)(8.95) = 11,000 psi

M.S. = (27,000/11,000)-1

$\boxed{\text{M.S.} = +1.45}$ Weld "D"

DRUM MOUNTING BOLTS

Loading of All Reeving Systems



Resultant upward load on drum of 15,000 lbs is reacted equally at each end of drum. The 7,500 lbs reaction is carried in shear by eight 3/4-in.-diameter bolts at each end of the drum. The torque in the system, T_d = 30.75/2 (150,000).

T_d = 2,306,250 in.-lbs

Mounting Bolts at Motor End in Shear

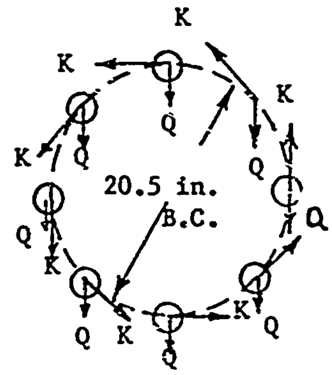
T_d = 10.25 (K) (8)

K = 28,200 lbs

Vertical reaction load per bolt

Q = 7500/8

Q = 940 lbs



Area of 3/4 in. dia. bolt = .442 in.^{2}

Maximum loaded bolt "A"
Load = K + Q = 29,140 lbs

f_s = 29,140/.442 = 65,900 lbs

F_{tu} = 180,000 psi

F_{su} = 108,000 psi

M.S. = (108,000/65,900)-1

$\boxed{\text{M.S. = + .64}}$ 3/4 in. hoist mounting bolts in shear

If 5/8 in. diameter bolts are used,

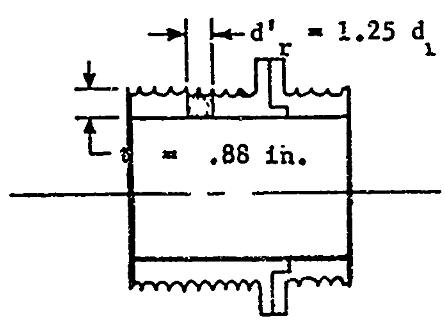
$\boxed{\text{M.S. = + .13}}$

DRUM

Crushing of drum

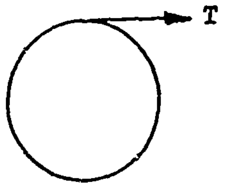
rope diameter d_r = .625 in.

effective diameter d'_r = .781 in.



Tension T = 37,500 lbs

Compressive stress $f_c = T/d'_r\ t$



f_c = 37,500/(.781)(.88) = 54,700 psi

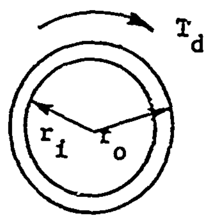
F_{cy} = 60,000 psi 5056 - H18 aluminum (Ref. 13)

M.S.= (60,000/54,700)-1

$\boxed{\text{M.S.= + .10}}$ Crushing of drum based on the compressive yield strength of 5056-H18 aluminum.

Drum in Combined Bending and Torsion

Torsion



Torsilastic spring half of drum is the critical section,

T_d = 2,306,250 in.-lbs

$f_s = 2T_d r_o/\pi(r_o^{\,4} - r_i^{\,4})$, r_o = 15.2 in., r_i = 14.3 in.

f_s = 1,760 psi

Bending

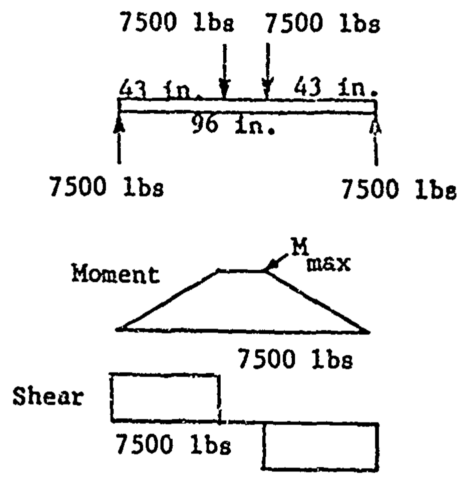
M_{max} = (43)(7,500)

M_{max} = 322,000 in.-lbs

S = 460 in.^{3}

$f_b = M_{max}/S$ = 322,000/460

f_b = 722 psi



Combined Stresses

$$f_{s(max)} = \left[(f_b/2)^2 + f_s^2\right]^{0.5} = \left[(361)^2 + (1,760)^2\right]^{0.5}$$

$$f_{s(max)} = 1,800 \text{ psi}$$

$$f_{n(max)} = f_b/2 + f_{s(max)} = 361 + 1,800 = 2,160 \text{ psi}$$

$$F_{tu} = 63,000 \text{ psi}$$

$$M.S. = (63,000/2,161)-1$$

$\boxed{M.S. = +\ 28.1}$ Drum in combined bending and torsion; therefore, crushing is critical.

Center Flange Bolts in Tension and Shear

These bolts must carry the M_{max} and one-half of T_d, or 1,153,125 in.-lbs

Load in bolt "A" = Q (tension)

$(3Q)(28) \approx 322,000$

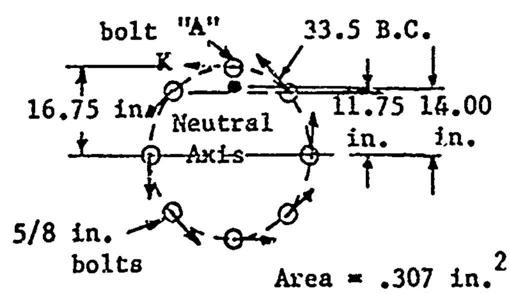
$Q \approx 3,840$ lbs

$f_t = 3,840/.307 = 12,500$ psi

Shear load due to torque K

$1,153,125 = 16.75\ (K)(8)$

$K = 8,600$ lbs



$f_s \quad = 8{,}600/.307 = 28{,}000$ psi

$f_{s(max)} = \left[(f_t/2)^2 + f_s^2\right]^{0.5} = \left[(6{,}250)^2 + (28{,}000)^2\right]^{0.5} = 28{,}700$ psi

$f_{u(max)} = f_t/2 + f_{s(max)} = 6{,}250 + 28{,}700 = 34{,}950$ psi

$F_{tu} \quad = 125{,}000$ psi

M.S. = (125,000/34,950)-1

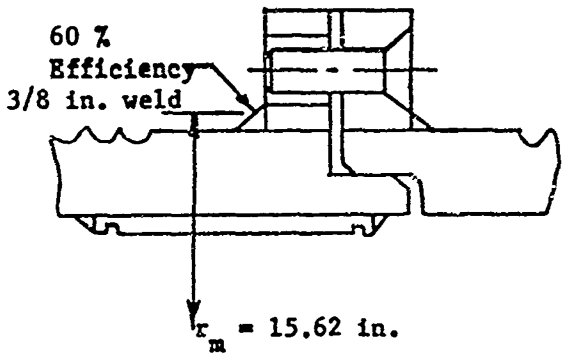
$\boxed{\text{M.S.} = +2.58}$ Center flange bolts

Center Flange Weld in Torsion Shear

$T \quad = T_d/2 = 1{,}153{,}125$ in.-lbs

Shear force $= T/r_m = 74{,}000$ lbs

Area $= 2\pi r_m\,(.707)(.375)(.60) = 15.6$ in.^{2}



$f_s \quad = 74{,}000/15.6 = 4{,}750$ psi

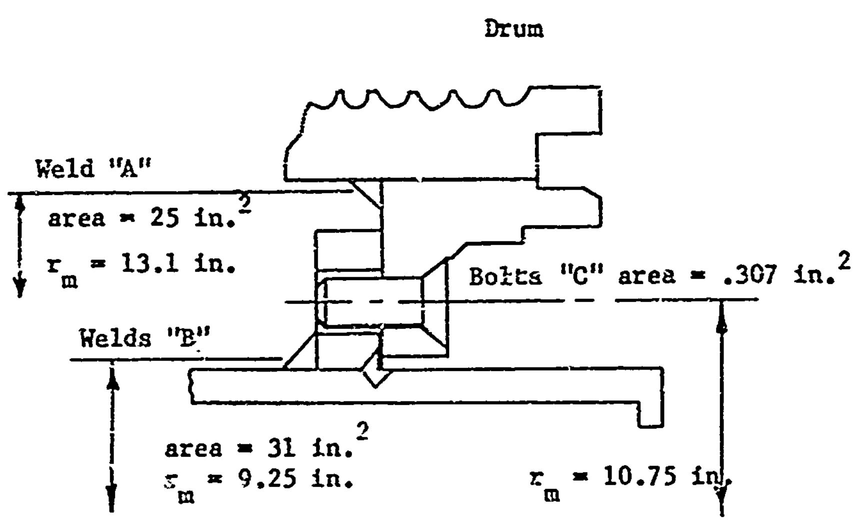
$F_{su} \quad = 24{,}000$ psi

M.S. = (24,000/4,750)-1

$\boxed{\text{M.S.} = +4.05}$ Center flange welds

Drum Torque Connection

Torque to be transmitted, T_d = 2,306,250 in.-lbs



Weld "A"

Shear force = 176,000 lbs

f_s = 176,000/25 = 7,050 psi

F_{su} = 24,000 psi

M.S. = (24,000/7,050)-1

$\boxed{\text{M.S. } = +2.55}$ Weld "A"

Weld "B"

Shear force = 250,000 lbs

f_s = 250,000/31 = 8,070 psi

F_{su} = 24,000 psi

M.S. = (24,000/8,070)-1

$\boxed{\text{M.S. } = +1.97}$ Weld "B"

Bolts "C"

8 bolts 5/8 in. dia

2,306,250 = (8) (load) (10.75)

Load = 26,800 lbs

f_s = 26,800/.307 = 87,400 psi

F_{su} = 108,000 psi

M.S. = (108,000/87,400)-1

$\boxed{\text{M.S.} = +.23}$ Bolts "C"

HYDRAULIC CYLINDER ANALYSIS

Pin Material AISI 4340 - Normalized

Pin "A" in Shear

$P/4 = 37,500$ lbs

$A(2\ 1/2 \text{ in. dia.}) = 4.91 \text{ in.}^2$

$f_s = 37,500/4.91 = 7,500$ psi

$F_{su} = 55,000$ psi (Ref. 11)

M.S. = (55,000/7,500)-1

$\boxed{\text{M.S.} = +\ 6.33}$

Material of Yoke HK31A-=24

$F_{tu} = 35,000$ psi (Ref. 12)

$A = 2.491 \text{ in.}^2$

$P/2 = 75,000$ lbs

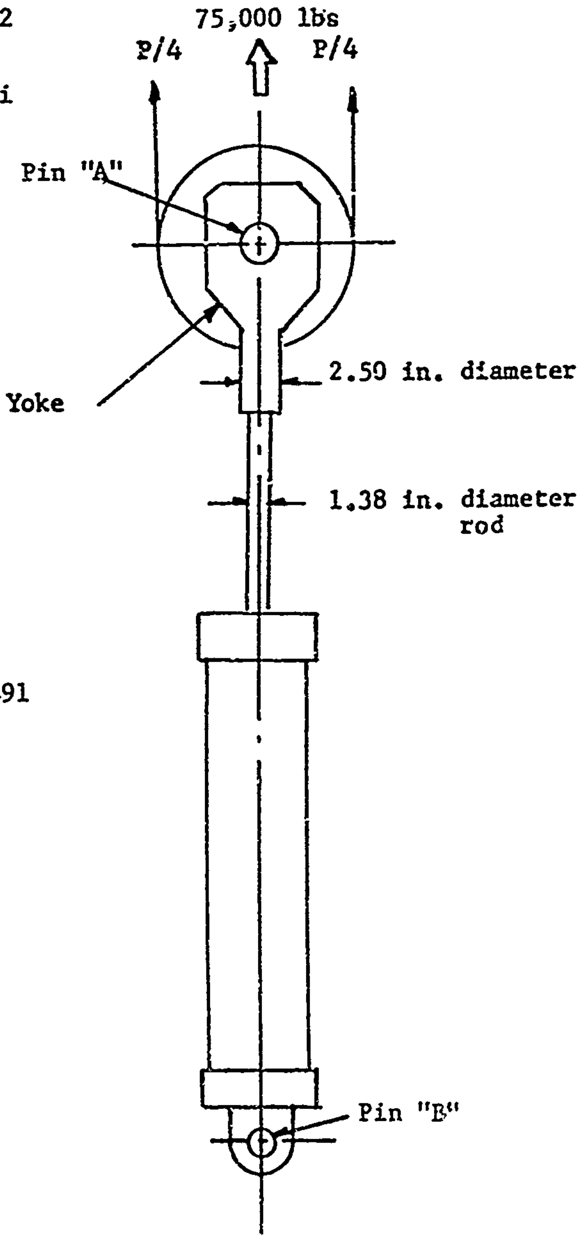
At Section, a-a

$f_t = P/2/A = 75,000/2,491$

$f_t = 32,100$ psi

M.S. = (35,000/32,100)-1

$\boxed{\text{M.S.} = +\ .09}$



Tension in Cylinder Rod

$A = 1.289 \text{ in.}^2$

$f_t = 7,500/1.289$

$f_t = 57,800$ psi

$F_{tu} = 125,000$ psi (A1S1 4340 Steel)

M.S. = (125,000/58,300)-1

$\boxed{\text{M.S.} = +1.14}$

A (.375 in. plate) = $.937 \text{ in.}^2$

$f_{br} = 37,500/.937 = 40,000$ psi

$F_{br} = 57,000$ psi

M.S. = (57,000/40,000)-1

$\boxed{\text{M.S.} = +.42}$

Pin "B" in Single Shear

P/4 = 37,500 lbs

A(7/8 in. dia.) = $.601 \text{ in.}^2$

$f_s = 62,300$ psi

$F_{su} = 75,000$ psi, $F_{tu} = 125,000$ psi

(A1S1 4340 Steel)(Ref. 11)

M.S. = (75,000/62,300)-1

$\boxed{\text{M.S.} = +.20}$

Clevis Plate in Bearing (Magnesium) HK31A-H24

$P/4 = 37{,}500$ lbs

A of .750 in. plate = 658 in.^{2}

$F_{br} = 57{,}000$ psi (Ref. 11)

$f_{br} = 37{,}500/.658 = 57{,}000$ psi

M.S. $= (57{,}000/57{,}000)-1$

$\boxed{\text{M.S.} = +.00}$

Hydraulic Cylinder

$P/2 = 75{,}000$ lbs

Hydraulic pressure = 3,000 psi

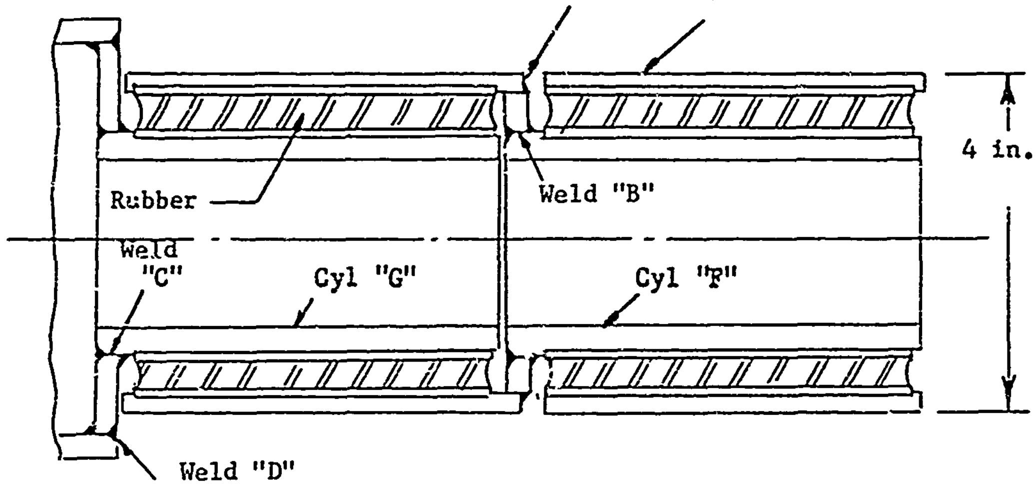
$A = 75{,}000/3{,}000 = 25$ in.^{2}

$d = 5.65$ in.

A commercial cylinder with a 6 in. bore, rated for 3,000 psi operation, will be suitable for this application.

TORSILASTIC SPRING

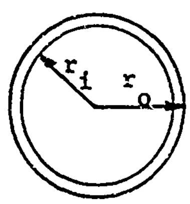
Torque, T_d = 2,306,250 in.-lbs



Cylinder "E"

$f_s = 2T_d r_o/\pi(r_o^4 - r_i^4)$

$r_o = 9.00$ in., $r_i = 8.75$ in.



$f_s = 4{,}612{,}500(9.00)/2040 = 21{,}000$ psi

$F_{su} = 27{,}000$ psi, magnesium (Ref. 12)

M.S. = (27,000/21,000)-1

$\boxed{\text{M.S.} = +.28}$ Cyl "E" in torsion

Bearing Load = 268,000 lbs

$A = 5.1$ in.^{2}

$f_{br} = 52{,}500$ psi

$F_{bru} = 57{,}000$ psi (Ref. 11)

$M.S. = (57,000/52,000)-1$

$\boxed{M.S. = +.08}$ Cyl "E" in bearing

Cylinder "F"

$r_o = 5.25$ in., $r_i = 4.25$ in.

$f_s = 4,612,500\ (5.25)/1040 = 23,700$ psi

$F_{su} = 27,000$ psi, magnesium

$M.S. = (27,000/23,700)-1$

$\boxed{M.S. = +.13}$ Cyl "F" in torsion

Bearing Load = 428,000 lbs

$A = 10.2\ in.^2$

$f_{br} = 42,000$ psi

$F_{br} = 57,000$ psi

$M.S. = (57,000/42,000)-1$

$\boxed{M.S. = +.35}$ Cyl "F" in bearing

Weld "B"

Shear Force = 385,000 lbs

$A = 22\ in.^2$

$f_s = 385,000/22 = 17,500$ psi

$F_s = 24,000$ psi

$M.S. = (24,000/17,500)-1$

$\boxed{M.S. = +.37}$

Weld "A"

Shear Force $= 275{,}000$ lbs

$A = 15.8 \text{ in.}^2$

$f_s = 275{,}000/15.8 = 17{,}400$ psi

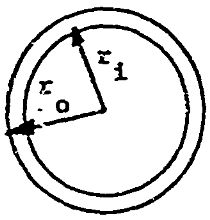
$F_s = 24{,}000$ psi

M.S. $= (24{,}000/17{,}400)-1$

$\boxed{\text{M.S.} = +.38}$

Cylinder "G"

$r_o = 5.25$ in., $r_i = 4.75$ in.



$f_s = 4{,}612{,}500(5.25)/440 = 55{,}000$ psi

AISI Steel $F_{tu} = 125{,}000$ psi (Ref. 11)

$F_{su} = 75{,}000$ psi (Ref. 11)

M.S. $= (75{,}000/55{,}000)-1$

$\boxed{\text{M.S.} = +.36}$ Cyl "G" in torsion

Weld "C" (Refer to Weld "D")

Weld "D"

Shear Force $= 220{,}000$ lbs

$A = 39 \text{ in.}^2$

$f_s = 220{,}000/39 = 5{,}650$ psi

$F_{su} = 75{,}000$ psi

M.S. $= (75{,}000/5{,}650)-1$

$\boxed{\text{M.S.} = +12.2}$

If cylinder "G" is made of magnesium, $r_i = 4.25$ in.

$\boxed{\text{M.S.} = +.13}$

Rubber

The maximum working moment applied to the torsilastic spring by the hoist cables is

$$M_{max} = (40{,}000)(15.38) = 615{,}000 \text{ in.-lbs}$$

The maximum shear stress in the rubber can be computed using equation 35.22 of Reference 9.

$$\tau_w = M/2\pi R_I^2 L$$

where R_I is inside radius of rubber in inches and L is length of rubber in inches.

$$\tau_w = 615{,}000/(6.28)(6.00)^2\ (21.0)$$

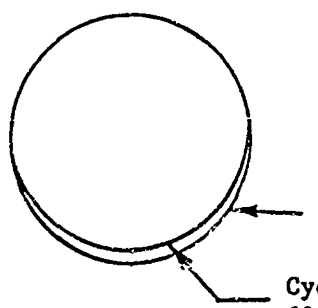
$$\tau_w = 129.5 \text{ psi}$$

This is in good agreement with the 130-psi working stress limit recommended by B.F. Goodrich (Reference 9).

$$\text{M.S.} = (130/129.5)-1$$

$$\boxed{\text{M.S.} = .003}$$

Cyclocentric Gear



33% of the teeth on each of the two cyloid discs are in engagement at all times. Number of teeth in engagement = (.33)(685)(2) = 452 teeth.

Gear ratio 229 to 1

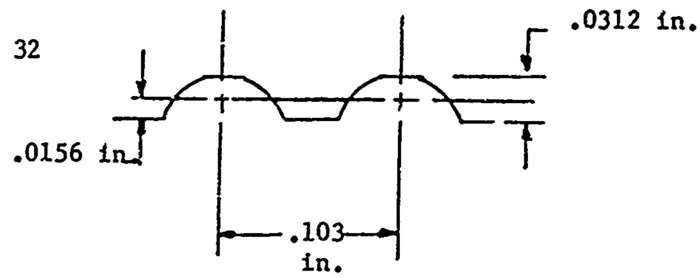
Input speed 1,710 rpm

Face width, F = 2.00 in.

Assume form factor, Y = .65

Diametral pitch D_P = 32

f_{allow} = 98,000 psi



Lewis equation

$$L_B = f_a FY/D_P = (98,000)(2)(.65)/32 = 4,000 \text{ lbs}$$

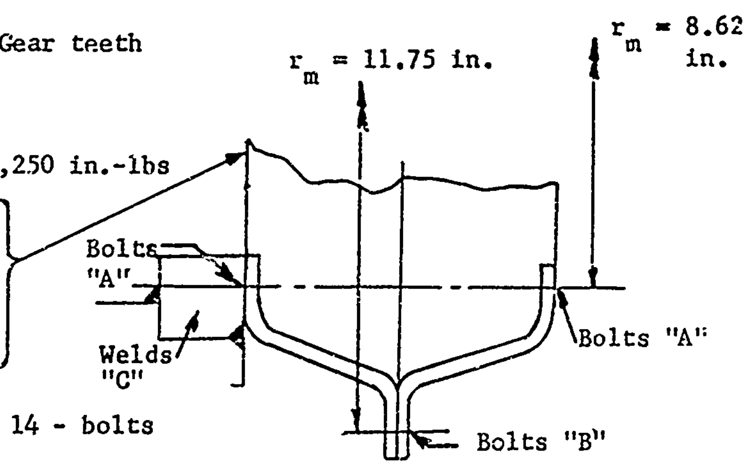
Total tangential force on teeth = 215,000 lbs

Load per tooth = 215,000/452 = 476 lbs

M.S. = (4,000/476)-1

$\boxed{\text{M.S.} = +7.40}$ Gear teeth

Coupling



Torque, T_d = 2,306,250 in.-lbs

Phosphor - Bronze

F_{tu} = 175,000 psi

F_{su} = 70,000 psi

Shear in Bolts "A" 14 - bolts

Shear force per bolt = 2,306,250/(8.62)(14) = 19,100 lbs

Area of bolt = .307 in.^{2}

f_s = 19,100/.307 = 62,300 psi,

F_{su} = 108,000 psi

M.S.= (108,000/62,300)-1

$\boxed{\text{M.S.} = +.73}$ Bolts "A"

Therefore, bolts "B" are ample.

Coupling in bearing

Maximum bearing load at Bolt "A" = 19,100 lbs

Phosphor - Bronze, F_{bru} = 180,000 psi

Bearing Area, A_{br}

F_{bru} = $19{,}100/A_{br}$ = 180,000,

A_{br} = .106 $in.^2$

$t_{req'd}$ for coupling = .170 in., $\boxed{\text{use .188 in. thick}}$

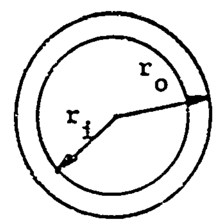
Torsion in coupling

T_d = 2,306,250 in.-lbs

r_o = 10.19 in.

r_i = 10.00 in.

$$f_c = 2\, T_d r_o/\pi(r_o^4 - r_i^4)$$



f_s = $4{,}612{,}500(20.38)/\pi(795)$ = 36,500 psi

M.S. = (70,000/36,500)-1

$\boxed{\text{M.S. = + .91}}$ Coupling in torsion

Weld "C"

Shear force = 243,000 lbs

Area = 26 $in.^2$

f_s = 9,350 psi

F_{su} = 24,000 psi

M.S. = (24,000/9,350)-1

$\boxed{\text{M.S. = + 1.57}}$ Shear Weld "C"

Drive Shaft

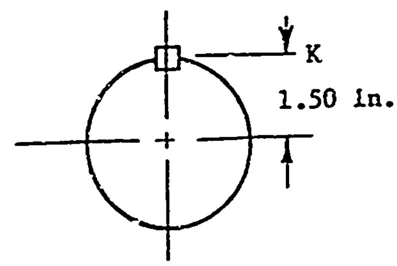
Cyloid Disc Key

1/2" x 1/2" x 3.5" long

T_d = 2,306,250 in.-lbs

Gear ratio 229 to 1

Torque in drive shaft, $T_s = T_s/229$



T_s = 10,070 in.-lbs

Key force K = 10,070/1.5 = 6,710 lbs

Shear area of key = 1.75 in.^{2}

f_s = 6,710/1.75 = 3,840 psi

C1020 key stock

F_{su} = 35,000 psi

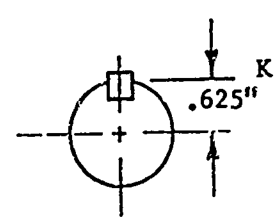
M.S. = (35,000/3,840)-1

$\boxed{\text{M.S.} = +\ 8.12}$ Cycloid disc key

<u>Motor Shaft Key</u>

Key force K = 10,070/.625

K = 16,120 lbs



3/8 in. x 3/8 in. x 1.25 long

Area = .468 in.^{2}

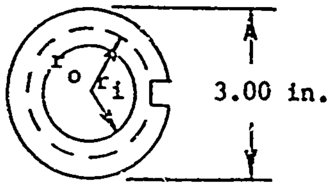
f_s = 16,120/.468 = 34,500 psi

M.S. = (35,000/34,500)-1

$\boxed{\text{M.S.} = +\ .01}$ Motor shaft key

Torsion in Shaft

T_s = 10,070 in.-lbs



r_o = 1.25 in.

r_i = 1.12 in.

$f_s = 2\ T_s r_o/\pi(r_o^4 - r_i^4) = (2)(10,070)(1.25)/\pi(2.44\text{-}1.60)$

f_s = 9,760 psi

Using low carbon steel, F_{su} = 35,000 psi

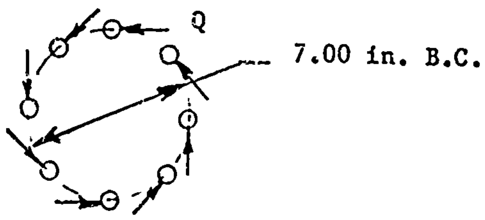
M.S. = (35,000/9,760)-1

| M.S. = + 2.48 | Drive shaft in torsion

Clutch Mounting Bolts

Shear force per bolt, Q (8 - bolts)

$Q = T_s/3.50(8) = 359$ lbs



Shear area per bolt = .049 in.^{2}

f_s = 359/.049 = 7,330 psi

F_{su} = 75,000 psi

M.S. = (75,000/7,330)-1

| M.S. = + 9.04 | Shear in bolts

Bearing on Plate

Force = 359 lbs

Thickness = 1/4 in.

Bearing Area = .0625 $in.^2$

f_{br} = 359/.0625 = 5,740 psi

F_{bru} = 57,000 psi

M.S. = (57,000/5,740)-1

$\boxed{\text{M.S.} = +\ 8.94}$ Clutch mounting plate in bearing

HYDRAULIC SYSTEM

The total horsepower required of the drive motor for raising a 40,000-pound load 150 feet in 3 minutes, assuming a 75 percent efficiency, is computed as follows:

$$1\ HP = 550\ \text{ft-lbs/sec}$$

$$N = \frac{W \times h}{550 \times t \times \eta}$$

where

N = Power required (HP)

W = Load (lbs)

h = Height (ft)

t = Time (sec)

η = Efficiency (%)

$$N = \frac{40{,}000 \times 150}{550 \times 180 \times .75} = 81\ HP$$

This amount of power will be required for about 3 minutes out of every 30 minutes, or a 10 percent duty cycle.

COMPARISON OF HYDRAULIC AND ELECTRICAL DRIVE

Both hydraulic and electrical systems can supply this power; but in airborne applications in the past, hydraulic systems have shown a distinct weight advantage. Hydraulic motors with 100-HP ratings are available commercially with weights of 30 to 80 pounds, while a 100-HP electrical motor would weigh in the order of 500 pounds.

To do justice to the electrical drive, however, the total system weight must be taken into consideration. This would include hydraulic valves, tubing, and accumulators, plus the associated mounting hardware, which weigh more than the contactors and wires of the electrical drive. Further, the electrical drives tend to be more reliable, with none of the oil contamination, wear, and leak problems that plague hydraulic systems.

At this time, the weight advantage is so clearly in favor of the hydraulic drive that the electrical drive was ruled out for the AAI 40,000-pound cargo handling system. However, considerable improvements are being made in electrical drives, and these will be closely followed by AAI for possible future use. (There would be space inside the cable drum to place a somewhat larger electrical motor, and the hydraulic cylinders could be replaced by electric-motor-driven jack-screws.)

Consideration was also given to a direct power takeoff from the main gas turbine. This might produce the lowest weight configuration, compared to an electrical or hydraulic system, because not only the motor, but also the generator or hydraulic pump, would be eliminated or reduced in size. Such a direct-drive system would, however, require gearboxes, bevel gears, magnetic clutches, and long transmission shafts. The problems encountered on tail-rotor drives in early helicopter designs have given this type of system a very poor reliability rating. An electrical or hydraulic hoist drive failure would not (with the exception of a pump or generator seizure) jeopardize the main-engine operation, while this is a danger with the direct drive. Therefore, no further consideration is given to direct-drive systems at this time.

A simplified schematic of the basic hydraulic subsystems is shown in Figure 22.

MOTOR AND VALVE CONSIDERATIONS

The down drive motor will work in three modes:

DRUM UP at full speed and high power
DRUM OFF, securely locked with no "creep"
DRUM DOWN at full speed and low power

The third mode would not be required for lowering a heavy load. When the hooks and cables alone are lowered, however, their pull is insufficient to overcome the friction of the gearbox and hydraulic motor; hence, a small amount of "DOWN" power must be applied.

When a heavy load is lowered, the hydraulic drive acts as a brake. Compression of the fluid and friction in the motor cause heating of the hydraulic oil. This heat must be removed by the oil cooler. The heat dissipated is theoretically

$$Q = \frac{h \times W}{\eta \times k} = \frac{150 \times 40{,}000}{.75 \times 778} = 10{,}200 \text{ BTU's}$$

assuming

h = height difference (ft)

W = load weight (lbs)

η = efficiency, 75%

k = conversion factor, ft/lbs to BTU = 778

These 10,200 BTU's represent a formidable amount of heat (enough to increase the temperature of 100 gallons of water by 12.2 degrees Fahrenheit) and must be dissipated into the air by the oil cooler. Because there is no forward airspeed during the load-lowering phase, a vertical airflow from the rotor may be preferable; if this should prove to be insufficient, an auxiliary blower or other heat exchanger may be required.

A Vickers Model 45 fixed-displacement motor was considered for the prototype system because of its high torque, its availability, and its low cost. It weighs 80 pounds. This weight could be reduced. However, it is only 50 pounds higher than that of comparable airborne units. Its low efficiency (70 to 75 percent), however, does present a serious problem when the heat problem in the load-lowering phase is considered.

The heat problem will influence the choice of a motor (and a pump) among the available aircraft types. A Vickers type PV3-205 motor, for example, with a displacement of 2.05 cubic inches per turn, can supply 100-HP shaft output power, but at a speed of 7,100 rpm. At this speed, the efficiency, because of turbulent flow losses, is only 79 percent, while at 3,750 rpm, it would be 91 percent, but the output would drop to 55 HP. The Vickers PV3-205 is the largest airborne motor/pump unit currently in production anywhere.

The only alternate is the AP12V pump/motor unit of the American Brake Shoe Corp., which has only 1.8 cubic-inch displacement-per-turn and about 80 HP at 7,250 rpm. It is used in the F-111 aircraft and could be used in the 40,000-pound hoist if the lifting speed specification were relaxed.

The next larger pump/motor unit of Vickers, Inc., is the 3.00 cubic inches-per-turn PV3-300, of which delivery is presently beginning for the C5A aircraft. Data for this unit are available in the pump mode, but they give an indication of the motor performance to be expected. At 5,000 rpm and 2,800 psi, it produces 100 HP and 1,500 foot-pounds of torque, at 62 gallons-per-minute flow and 86 percent efficiency. The PV3-300 appears to be best suited for both the power source and the motor tasks at this time.

HYDRAULIC POWER SOURCE CONSIDERATIONS

The specification for the cargo handling system stipulates that the contractor shall not concern himself with the hydraulic power source; a constant-pressure 3,000-psi supply will be provided.

A study of the hydraulics system and discussions with hydraulic component vendors indicate, however, that a 100-HP system will be more economical and efficient if the power source is included in the control loop. These are the reasons in favor of a controlled pump:

> The duty cycle of the hoist drive is very low. It is estimated that it will lift loads for less than 3 minutes every 30 minutes.
>
> The load factor will be less than 100 percent; i.e., some loads will weigh less than 40,000 pounds.
>
> A constant 3,000-psi power source would be used at capacity only 3 to 5 percent of the time, and would generate high pressure unnecessarily for about 95 percent of the engine-turning time.

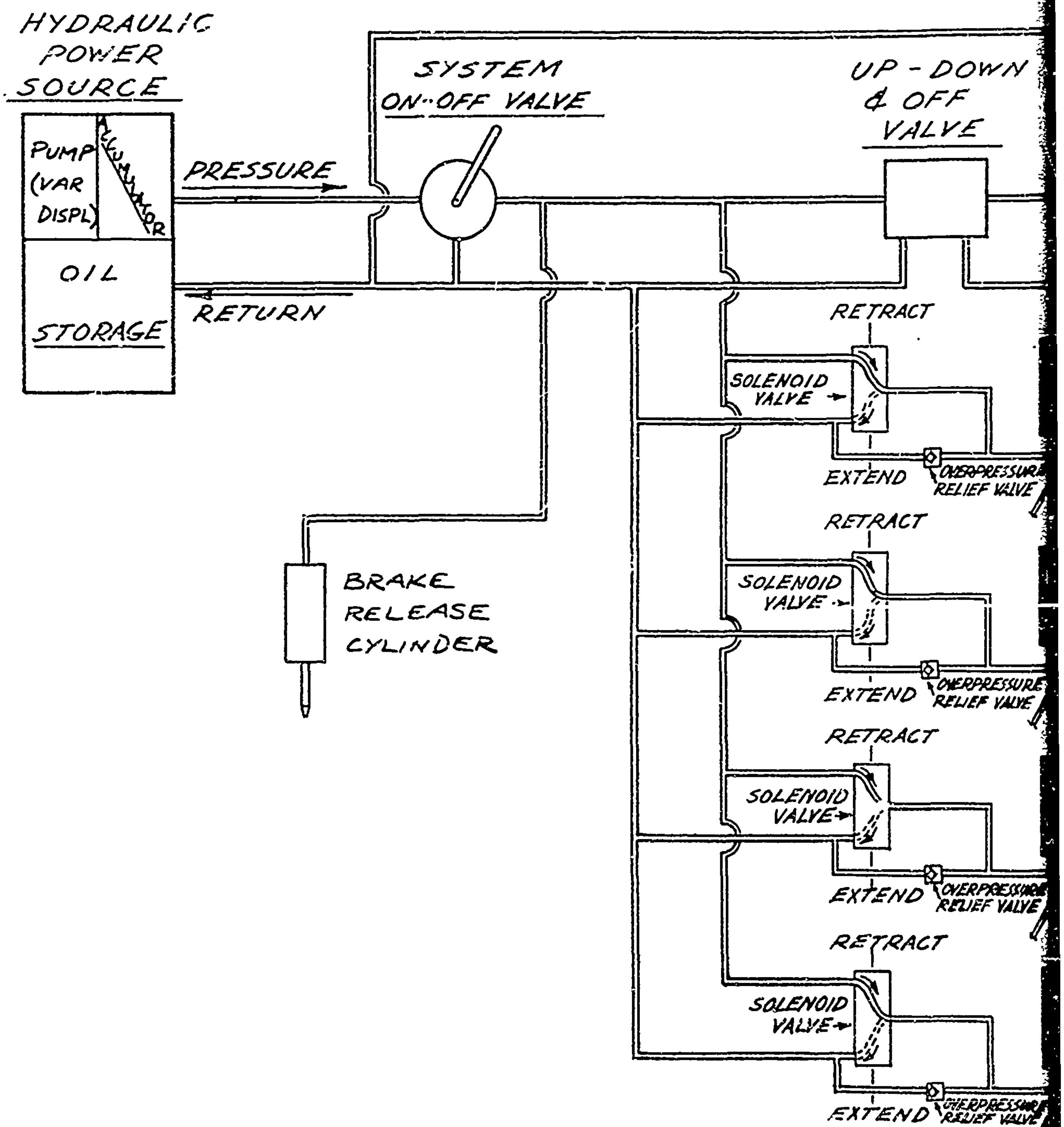
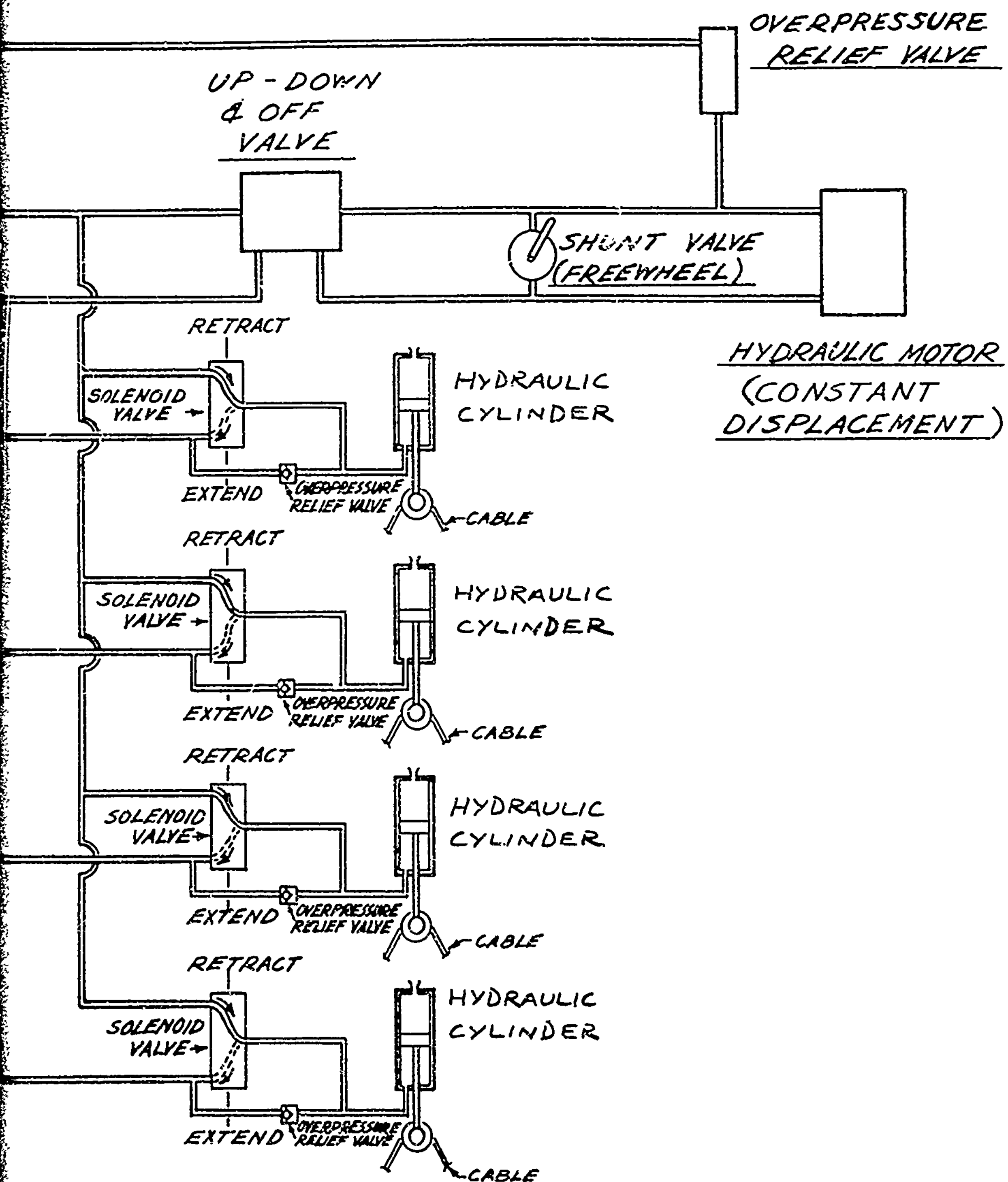


Figure 22. Hydraulic Schematic


OVERPRESSURE RELIEF VALVE
UP - DOWN & OFF VALVE
SHUNT VALVE (FREEWHEEL)
HYDRAULIC MOTOR (CONSTANT DISPLACEMENT)
RETRACT
SOLENOID VALVE
HYDRAULIC CYLINDER
EXTEND
OVERPRESSURE RELIEF VALVE
CABLE
RETRACT
SOLENOID VALVE
HYDRAULIC CYLINDER
EXTEND
OVERPRESSURE RELIEF VALVE
CABLE
RETRACT
SOLENOID VALVE
HYDRAULIC CYLINDER
EXTEND
OVERPRESSURE RELIEF VALVE
CABLE
RETRACT
SOLENOID VALVE
HYDRAULIC CYLINDER
EXTEND
OVERPRESSURE RELIEF VALVE
CABLE

B

A more practical approach would use a variable-displacement pump (practically all airborne pumps are of this type) and control its displacement from the copilot's and pilots DRUM UP/DOWN control. This would require a simple modification of a conventional constant-pressure pump, such as a Vickers PV3-300, and attachment of a small servo valve. The pump would then "idle" most of the time at zero displacement and would produce pressure only when called for.

There are, however, subsystems which cannot tolerate a variable-pressure source, such as the brake-release cylinder, load-equalizing cylinders and the helicopter's hydraulically controlled mechanism. Because the helicopter will most likely be equipped with two or more engines, it would be practical to mount the variable-pressure pump on one engine and a constant-pressure pump on the other. Figure 23 shows an arrangement that would probably result in longer component life, higher reliability, and reduced oil-cooler requirements.

In the DRUM OFF position, the braking action of the hydraulic motor is poor. About 4 percent leakage flow (of the 62-gpm rated flow) must be expected, which would cause an intolerable amount of "creep" during sustained flight with a heavy load. Hence, a mechanical brake <u>must</u> be used.

Reversal of the drum drive for UP and DOWN motion can be obtained in three ways:

Reversible pump flow
Switching of input and output lines with valves
Reversible motor

Using a reversible pump or motor appears very simple in theory, but it involves large costs for special units. The displacement varying yoke is tilted between a neutral and a maximum-stroke angle by a small hydraulic cylinder and piston assembly, which is part of the standard units. To make it reversible requires "over-the-center" operation with a second cylinder mounted on the opposite side. This requires a new pump housing, with extensive engineering and tooling cost, as well as long delivery. Hence, the reversible pump or motor approach appears to be less attractive than a line switching approach with valves.

A flow reversing valve for the motor must have a minimum pressure drop: a drop of less than 100 psi out of 3,000 psi at 60 gallons/minute flow is desirable. Unfortunately, MIL-approved valves for airborne use satisfying this requirement are not presently available. The only large MIL-approved airborne units, such as Moog or Sanders servo valves, have pressure drops of 500 to 1,000 psi at this flow rate. There are suitable industrial valves available with 1.0 inch pipe size and approximately 35 psi pressure drop which fulfill all basic requirements. It is contemplated to use such pilot-operated poppet valves for prototype development. For production, later on, military approval for these valves may be obtained, or a reversible hydraulic pump may become available.

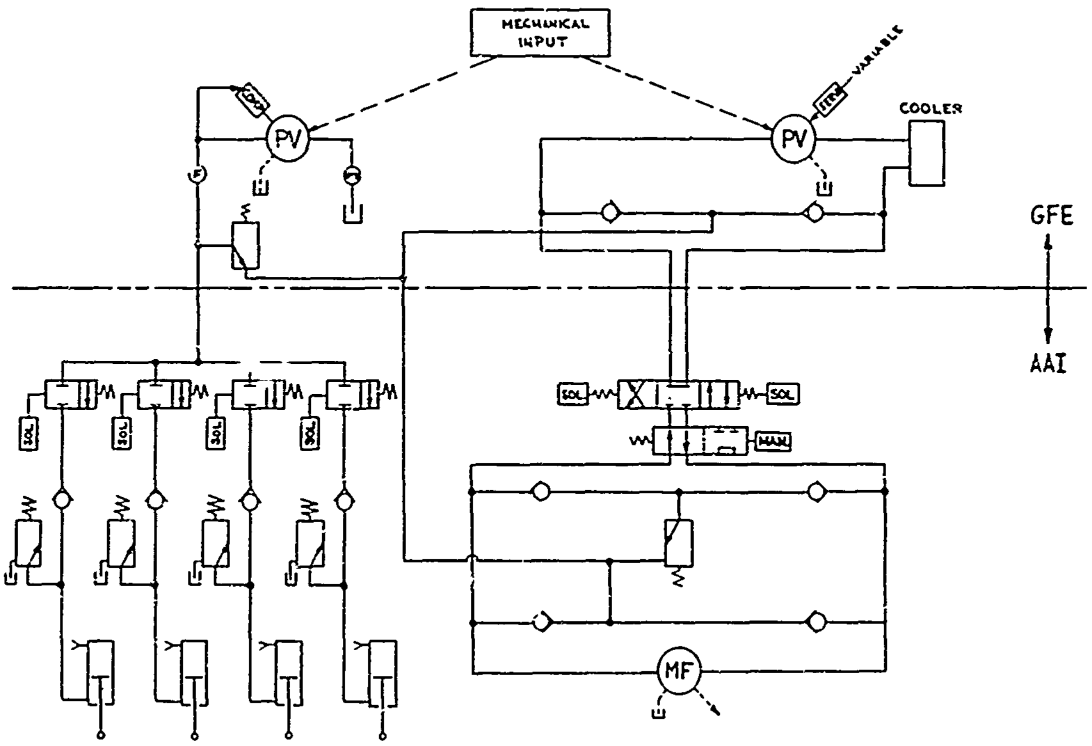


Figure 23. Dual Hydraulic System Schematic

For the FREEWHEEL emergency mode, the load must, in case of hydraulic failure, rotate the drum, gearbox, and motor. To remove the drag of the motor, the motor yoke can be moved to the neutral position, or a bypass valve can be opened to allow free fluid flow through the motor. The neutral position on the motor appears to be more economical and is preferred at this time. The displacement control cylinder will be operated from the cockpit with a lever and lanyard arrangement. A small hydraulic cylinder (similar to an automotive brake master cylinder) with its own fluid reservoir will be used as a link so that no mechanical changes need to be made on the motor.

SELECTION OF HYDRAULIC FLUID

MIL-H-5606 hydraulic oil is the standard hydraulic fluid for military vehicles, and will be used in the hoist system. There are non-flammable fluids available to MIL-H-7083A (Hydrolube), but their water content causes severe corrosion problems, and their use does not appear to be justified here because of the relatively small fluid volume involved.

CONCLUSIONS ON THE HYDRAULIC SYSTEM

Detail design of the hydraulic system will have to wait until the requirements are defined more closely. However, it can be stated here that the proposed hoist approach is straightforward and does not pose any problems that would not exist with other types of hoists. The single drum system is simple and flexible and can be adapted to any new requirements. Limitations of available hardware are strictly a function of the hoist capacity and lifting speed requirements.

INSTRUMENTATION AND CONTROLS

GENERAL

The cargo handling system can be controlled from three locations:

Pilot's seat (front left of cockpit)
Copilot's seat (right side of cockpit)
Load-master (on the ground)

All three will have access to the controls simultaneously, but OVERRIDE LOAD-MASTER switches on the pilot's and copilot's control panels can disconnect the load-master's control in an emergency. The pilot and co-pilot have independent control panels. This will enable the copilot to control the system in case the pilot is disabled, or vice versa.

The following functions will be controllable:

DRUM UP and DRUM DOWN buttons serve to raise and lower the load on all four cables simultaneously.

Four CYLINDER UP and four CYLINDER DOWN buttons enable the operator to raise or lower individual hooks in order to level the load and adjust the cable loads.

Individual cylinders can be retracted or extended (the load raised or lowered) by four sets of UP and DOWN push buttons, serving the NOSE LEFT, NOSE RIGHT, TAIL LEFT and TAIL RIGHT locations in the 4-point configuration.

In addition, the ALL CYLINDERS UP and ALL CYLINDERS DOWN buttons permit retraction or extension of all cylinders simultaneously. (This might be desired in the single-point configuration).

Emergency Functions (HOOKS OPEN, FREE WHEEL and CUT CABLES).

A LOAD-MASTER OVERRIDE switch on either the pilot's or co-pilot's control panel can disconnect the power to the load-master's control box, if this should become necessary in an emergency.

A HYD. PWR. ON/OFF switch actuates the main shut-off valve of the hydraulic power source.

An EL. PWR. ON/OFF switch disconnects all electrical power to the cargo handling system. Both the hydraulic and electrical power can be controlled from either one of the two control panels. (The cable cutter circuit has its own battery and remains on at all times. It can be actuated from either the pilot's or the copilot's panel by depressing the EMERGENCY and CUT CABLES buttons simultaneously.)

Two basic operating modes of the hoist system can be distinguished:

1. The pilot flies the helicopter; the copilot controls the hoist and instructs the pilot through the interphone system when he is to raise and lower the load. A control panel layout for this mode is shown in Figure 24.

2. The copilot, facing rearward to the load, controls both the helicopter and the load simultaneously. Because control of the helicopter requires both hands on the two control sticks at all times, the hoist control must be incorporated into the sticks. The configuration of the controls for the heavy-lift helicopter has not yet been established, but from existing designs it is anticipated that it will be difficult to add any controls to the sticks.

In all probability, only the DRUM UP and DRUM DOWN switches will be mounted on the control sticks, so that simultaneous control of the helicopter rotor and the hoist is possible. Detail design of this feature will have to await data on the cockpit configuration of the heavy-lift helicopter. The proposed hoist design is very flexible and can be adapted to any requirement.

An automatic load-balancing mode is considered for the "stick only" operation. The control valves supplying pressure to the hydraulic cylinders via the CYL. UP/DOWN switches will remain closed, and another set of solenoid valves will join the hydraulic lines to the four cylinders. This will permit the pistons in the cylinders to be moved up or down under the force of the load until the pressures balance. Within limits, this will tend to equalize the loads in the cables.

AUTOMATIC VS. MANUAL CONTROL

A manual control system, augmented by automatic limiting devices has been chosen instead of an automatic system for the following reasons:

It is inherently simple and reliable, using rugged, time-proven components.

It is more flexible in unusual situations when operated by an experienced load-master.

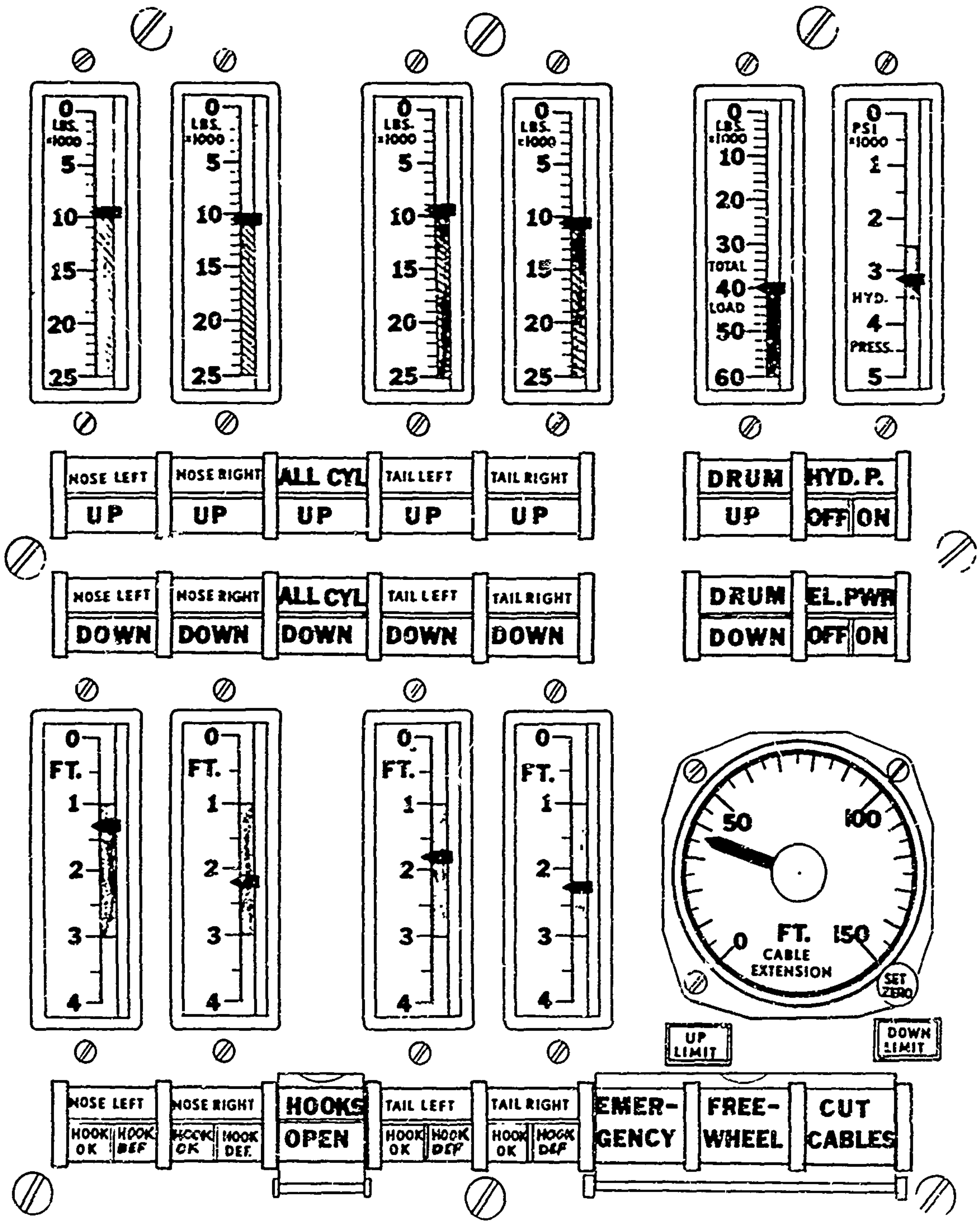


Figure 24. Control Panel

The redundancy of control panels and indicators offers a better probability that the system can be kept operating when damaged by enemy ground fire.

The cost is lower.

However, there will be two automatic functions incorporated into the system:

UP-LIMIT and DOWN-LIMIT switches will prevent accidental damage to the hoist system by the hooks' being retracted too far, and accidental running off of the cables from the drum if lowered too far.

Safety bypass valves will limit the load force on the pistons of the hydraulic cylinders. They can be preset to any overload value (typically +50 percent of max. rating, or 15,000 lbs) and will permit the hydraulic pressure to bleed off the accumulator. This will lower the piston and hence the load on the particular cable until the overload is reduced below the limit value. This would result in a load attitude change which must be considered before initiating this mode of operation.

INDICATORS

A human factors study has shown that conventional round aircraft indicators do not give a convenient side-by-side display of similar parameters, such as the four stress readings of the cables and the extension of the hydraulic cylinders.

Vertical-scale indicators have been selected instead, which are finding more and more use in manned space vehicles and high-performance turbojet aircraft. The Weston Type 1880, which has been tentatively selected, has a 3.0-inch scale in a hermetically sealed case measuring 1.3 inches wide, 4.6 inches high, and 6.0 inches deep. The instrument is ruggedized for aircraft application; additional protection will be obtained by mounting each control panel with vibration isolators (stainless steel mesh type).

Each control panel will contain 10 each vertical-scale indicators (4 each different scale types):

4 ea. cable load indicators, 0-25,000 lbs

4 ea. hydraulic cylinder extension indicators with scales calibrated 0 - 4' cable extension

1 ea. total cable load indicator, 0-60,000 lbs

1 ea. hydraulic power supply pressure indicator, 0-5,000 psi

In addition, there will be a circular indicator in a conventional 3-1/4-inch case, displaying cable extension 0 to 150 feet. The round indicator was chosen because there is only one such instrument; hence, no side-by-side comparison is required. Also, its 3.0-inch-diameter scale with 270° arc yields the highest accuracy of the 0- to 150-foot cable extension, and lends itself best to a synchro-readout.

When new cables have been installed, these indicators must be "zeroed in" once. This is simply performed by holding in a DRUM UP button until the drum stops and the UP LIMIT light goes on; the ZERO SET knobs on the CABLE EXTENSION indicators are then rotated to make the pointer read "0".

CONTROL SUBSYSTEMS

Momentary push buttons will be used to control the cylinder and cable drum UP and DOWN motions. (See Figure 25.)

All switches are connected in parallel. Should the pilot, for example, push the cylinder A UP button and the copilot simultaneously the DOWN button, the solenoid valve will remain in the OFF position until one of the buttons is released; therefore, no damage would occur.

A switch on both the pilot's and the copilot's panel will allow them to disconnect power to the load-master's control box in emergencies.

The emergency switches (FREEWHEEL, HOOK RELEASE and CABLE CUTTER) are shrouded with spring-loaded covers, as a precaution against accidental operation. All safety devices are connected in parallel; hence, simultaneous application by pilot and copilot will function without interference.

LOAD-MASTER'S CONTROL BOX

This box is connected to the helicopter through a multi-conductor cable, and permits the load-master to control the four rams as well as the cable drum. An interphone feature is added for communication with the pilot and copilot.

When not in use, the cable will be wrapped around two hooks on the helicopter's fuselage and the box will be deposited in a stowage compartment.

An electrically driven cable reel was considered for automatic retraction. This would require multiple slip rings for the cable, a gear motor, etc.; hence, it would add weight and complexity.

A remote radio control link was also considered, but its complexity does not appear to be justified for this cargo-handling system.

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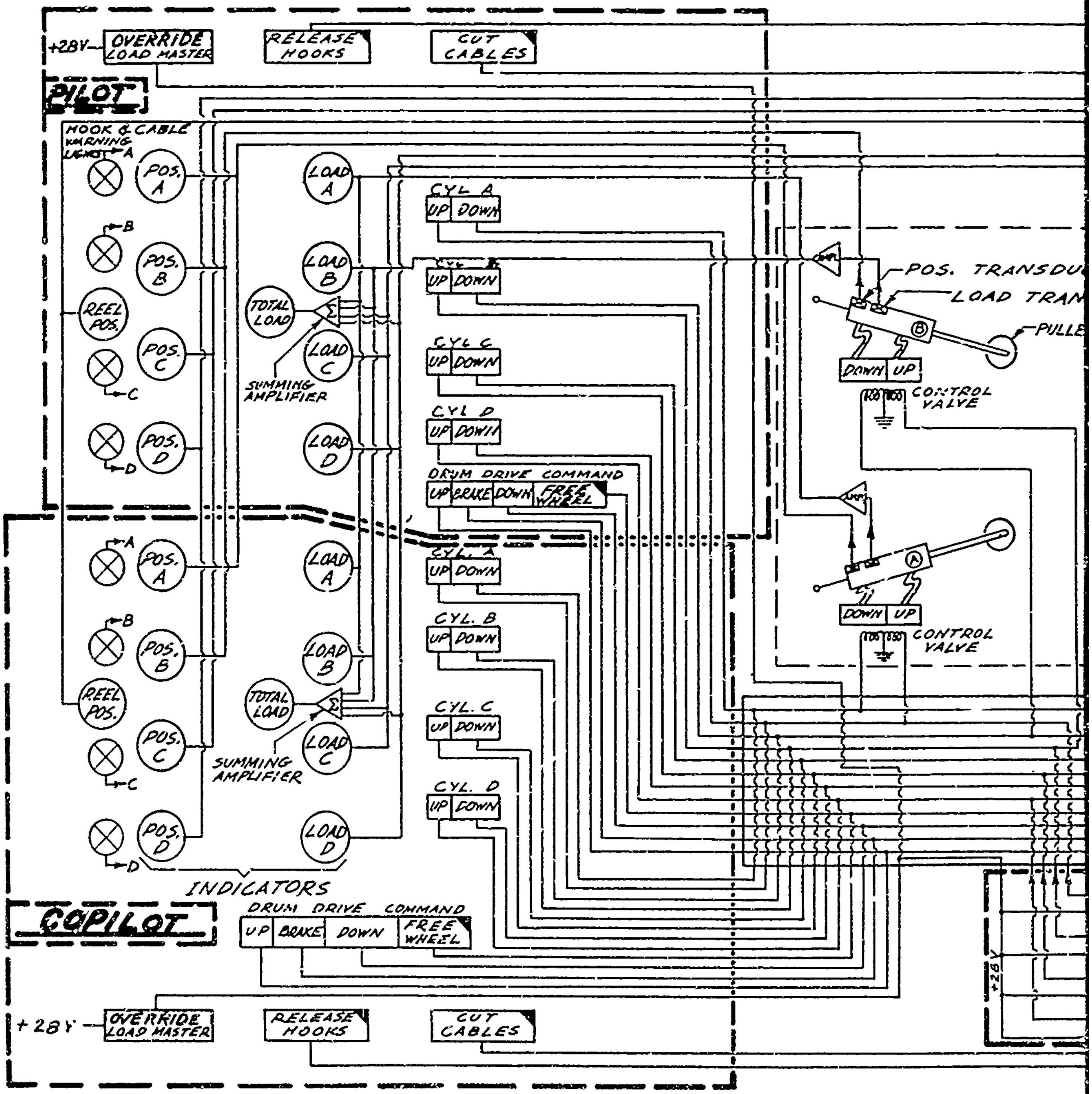
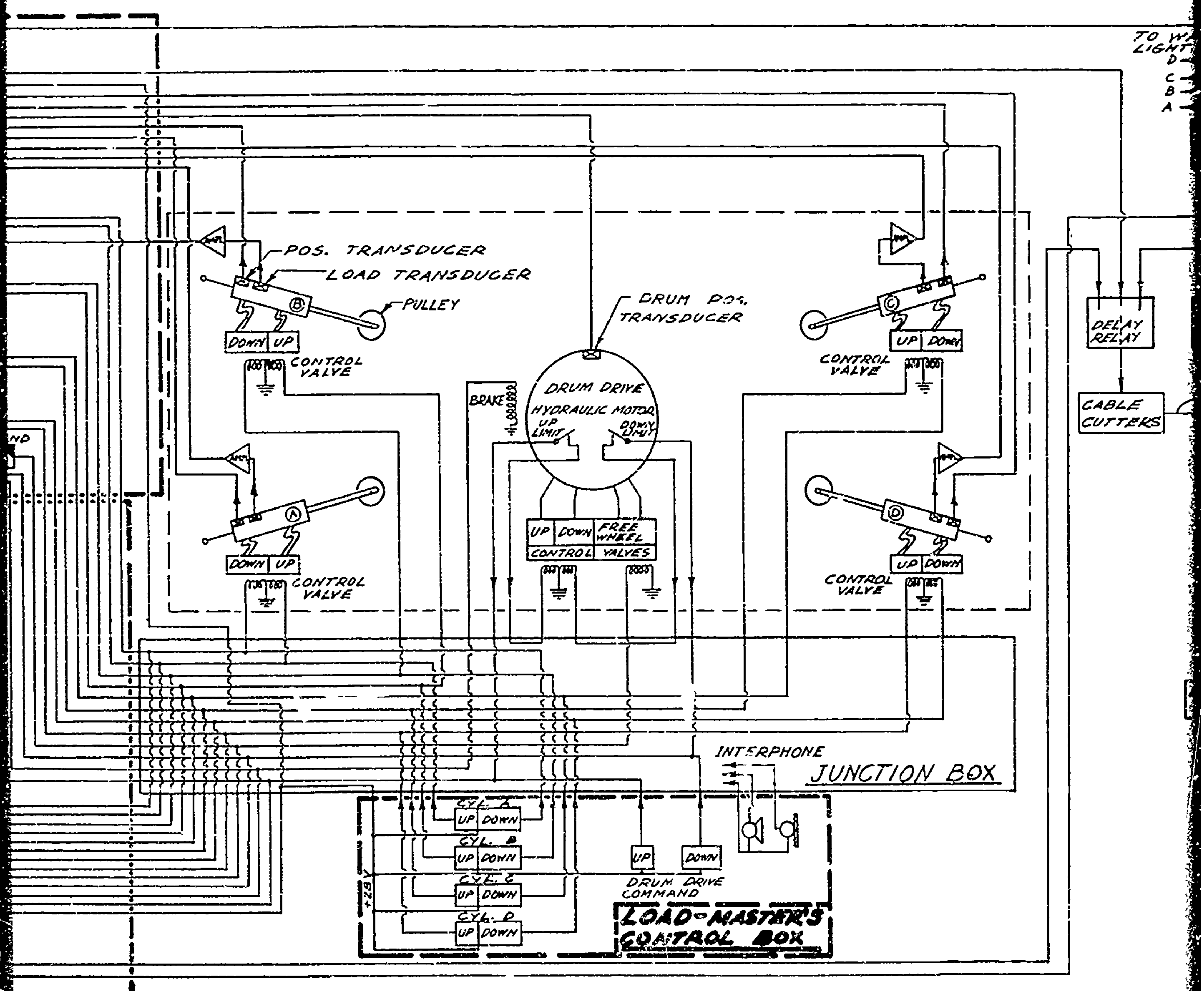


Figure 25. Electrical Schematic (Simplified)

A



ematic (Simplified)

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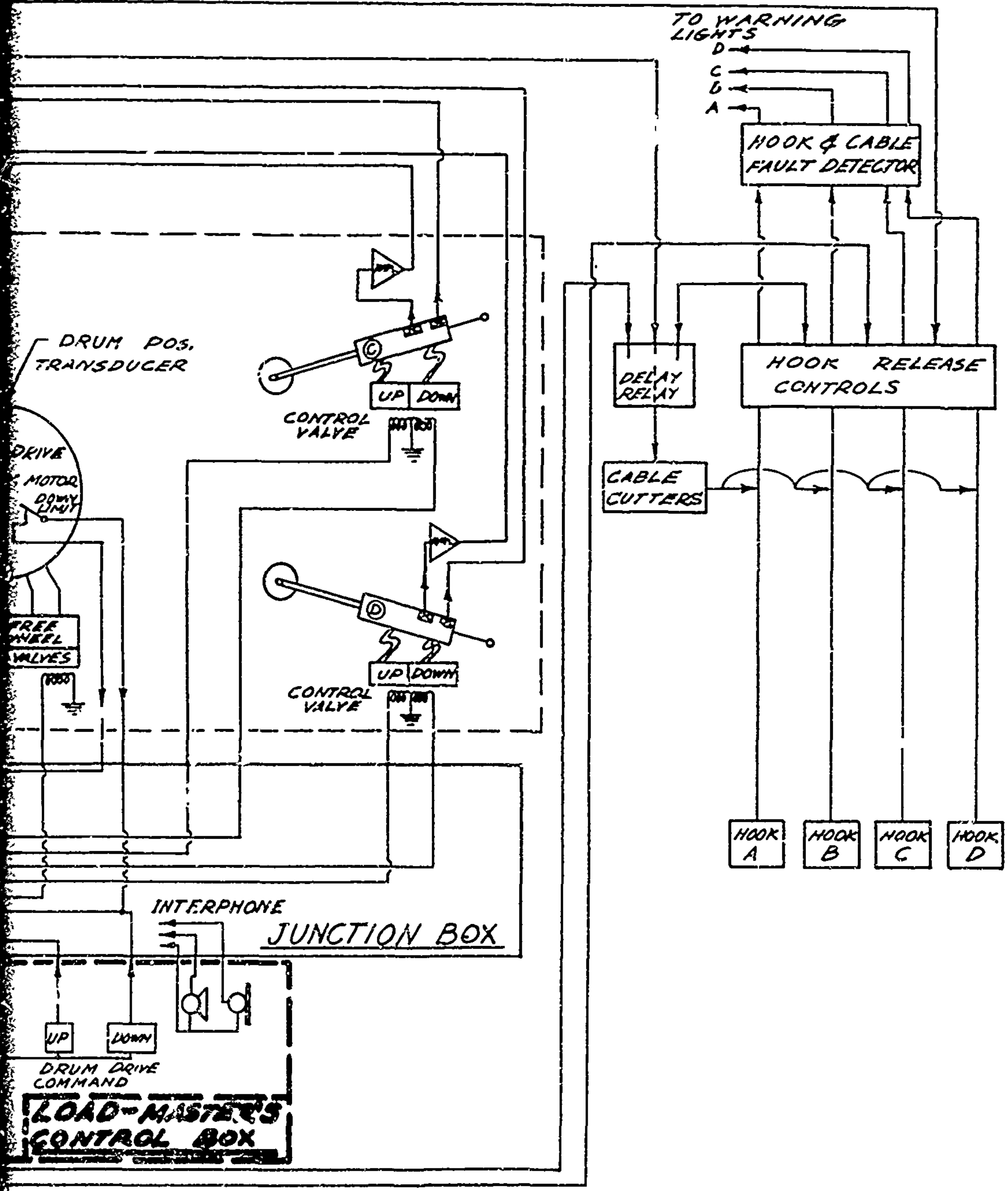


NOTES:

1. ALL SWITCHES ARE MOMENTARY PUSH BUTTON EXCEPT BRAKE AND OVERRIDE.

2. EMERGENCY BUTTONS HAVE SAFETY COVERS AGAINST ACCIDENTAL OPERATION. THEY ARE MARKED ▭ IN THIS DRAWING.

C

SELECTION OF TRANSDUCERS

CABLE EXTENSION below the helicopter fuselage can best be measured with a turns-counter on the cable drum, because the cables are fixed to the drum and run in deep grooves, thereby producing a repeatable relationship. A size 18 BU ORD type synchro transmitter (type 18 CX 4b) has been selected. In order to obtain a 270° scale deflection (at 1:1 synchro connection) for a 150-foot cable extension, a 30:1 speed reducer (PIC design Cat No. U4-6) is installed between the drum and the synchro. A slip ring assembly is also incorporated into the transducer assembly. (See cross-sectional drawing of cable drum, Figure 13.) The slip rings carry the hook-opening current to the cable ends on the surface of the drum, as well as the limit-switch connections.

DRUM - DOWN-LIMIT switches (Honeywell-Microswitch Type SS03B10) are mounted in recessed locations on the cable grooves, about 1-1/2 drum turns from the cable ends. Normally the plungers of the switches are depressed by the cables; if one or more of the cables lifts off from the plungers, the switch contacts close, thereby energizing the DRUM-INHIBIT relay in the control circuit. See Figure 26.

DRUM - UP-LIMIT switches will be mounted on arms below the helicopter fuselage, near the cable outlets. When the hooks are retracted fully, they strike the switch arms which in turn energize the DRUM-INHIBIT relay. (Mounting of the DRUM-UP-LIMIT switches directly on the drum, similar to the DOWN-LIMIT switches, is feasible. However, the UP-LIMIT would depend on the exact cable length used, and would vary between 1-, 2-, and 4-point configurations).

Hydraulic cylinder extension transducers are required to indicate to pilot and copilot how much piston travel UP and DOWN is still available in each cylinder. The following types of transducers have been considered:

1. Push-rod transducers, mounted alongside the hydraulic cylinders. Both inductive (linear variable differential transformer, LVDT) and resistive (potentiometer, slide wire) types are available. The LVDT type has no brushes and is the most reliable of the two types. See Figure 27.

2. A cable and pulley combination with rotary transducers (spring loaded) is more compact and permits use of conventional synchros or potentiometers. See Figure 27.

3. Transducers built into the hydraulic cylinders. Such transducers would be less vulnerable to snagging and damage, but they are not available commercially.

Alternates 1 and 2 were found to be unsatisfactory because of their exposed locations. Alternate 3 is the most desirable choice, but if conventional LVDT of potentiometer-type transducers are built in (available from G. L. Collins Co.), the length of the cylinder assembly will more than double.

To solve this problem, the contractor has designed a built-in transducer of the potentiometer type which does not increase the length of the cylinder. The only modification to adapt a standard cylinder is a 1/2-inch blind hole in the center of the pushrod, which receives a 1/4-inch-diameter resistance element as the piston is retracted. This element consists of a 1/4-inch-diameter fiber glass rod with a .08-inch-wide nichrome ribbon wound on it. This produces a 0-15 ohm resistance element, with a grounded set of contact fingers at the face of the piston. Electrically, it functions in a manner similar to potentiometer-type fuel gage indicator transducers. See Figure 28.

Cable Load Indicators

Measurement of cable loading is easily achieved by monitoring fluid pressure in each hydraulic ram, because this pressure is directly proportional to the cable load supported by the actuator. These pressures will be displayed to the pilot and copilot on dials calibrated directly in pounds of loads. Two basic types are available:

1. Hydraulic-mechanical pressure gages.

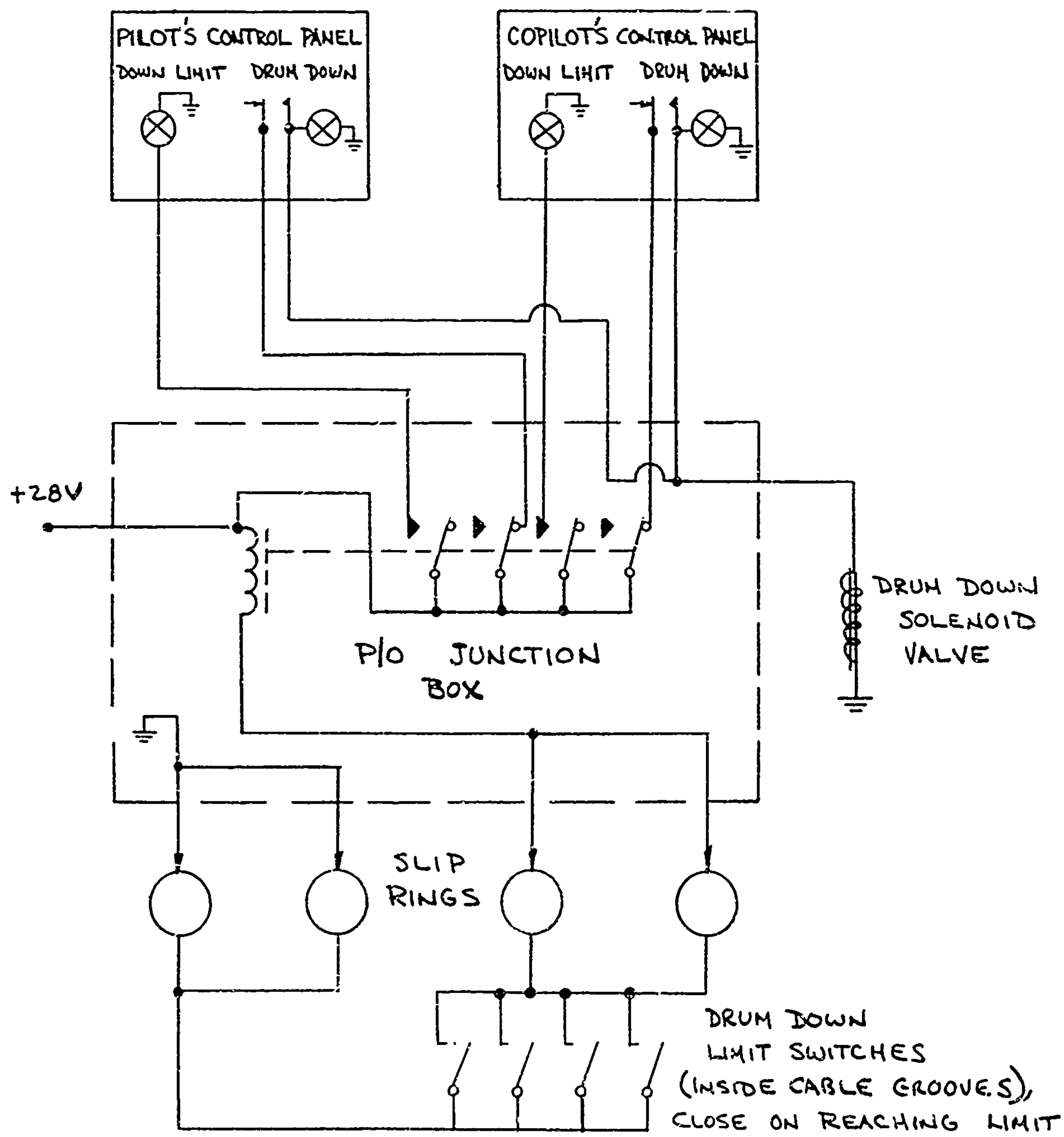
2. Electrical pressure transducers and indicators.

The first type is inexpensive but it is not desirable for this application because it would require routing hydraulic lines from the rams to the cockpit. The increased length of the hydraulic lines would increase their vulnerability to enemy small-arms fire. Also, it would require a special four-input indicator to display total load of the four cables.

Among the available electrical transducers are potentiometers, synchros, and strain gage types.

Strain gage types have tentatively been selected because of their small size and immunity to vibration. They will be mounted directly on the hydraulic cylinder pressure inlets.

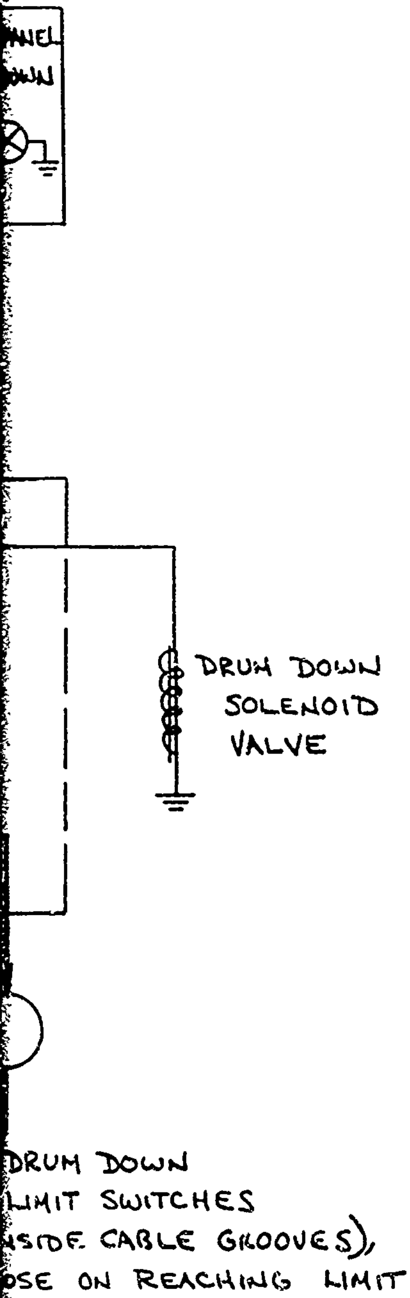
The signal output from strain gage transducers is too weak to drive indicators directly; they will be boosted (by small silicon transistor amplifier modules) to a level capable of driving conventional moving coil meters.

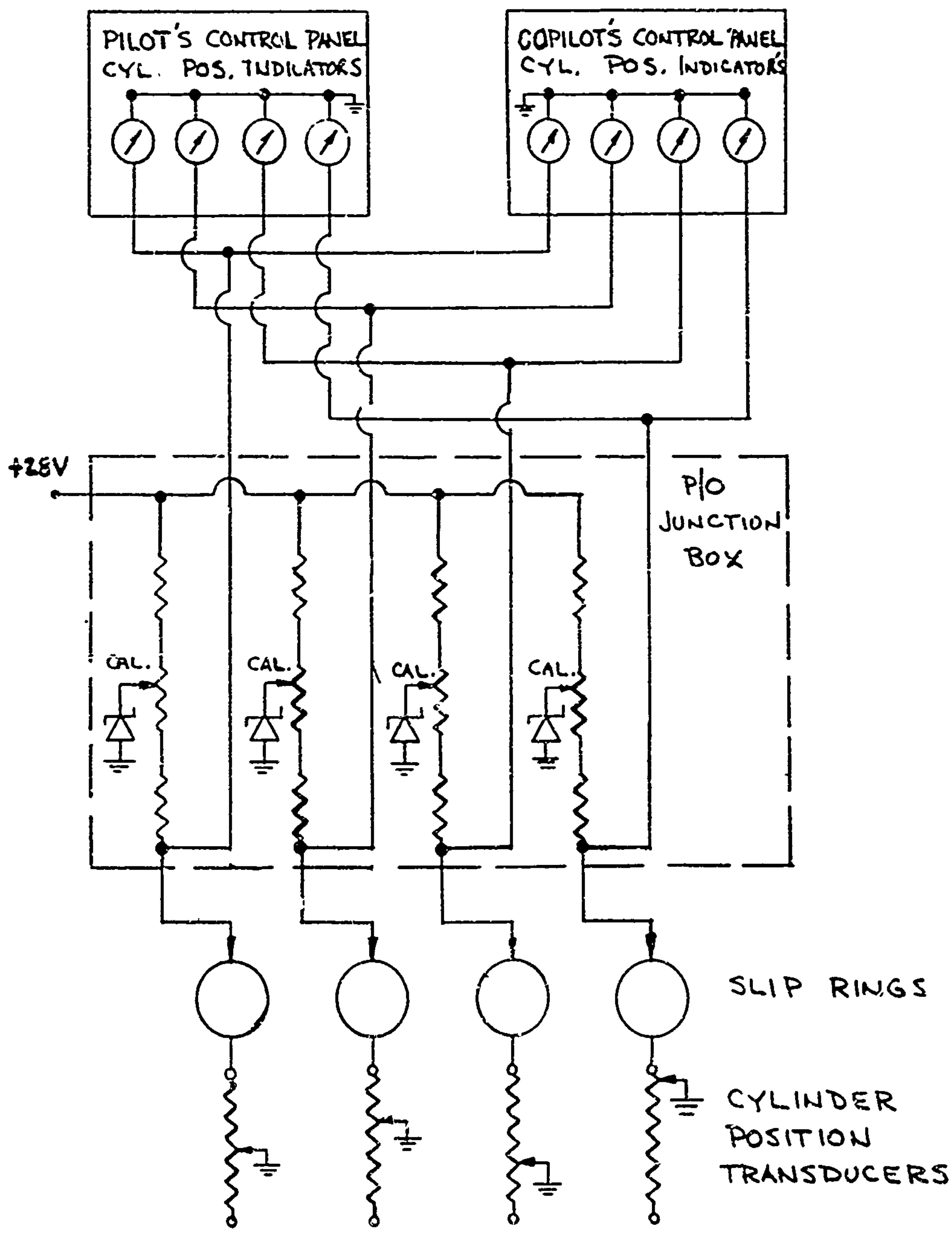


DRUM DOWN LIMIT CIRCUIT

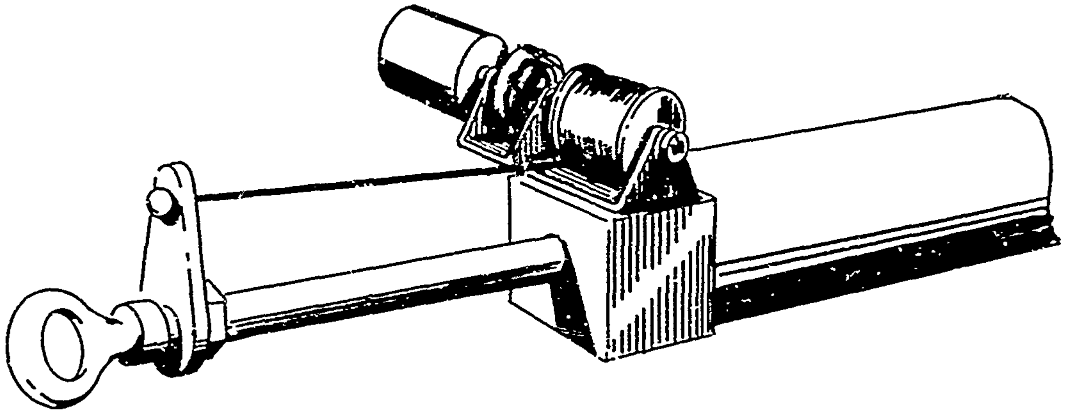
Figure 26. Cylinder Position Indicator Circuits

A

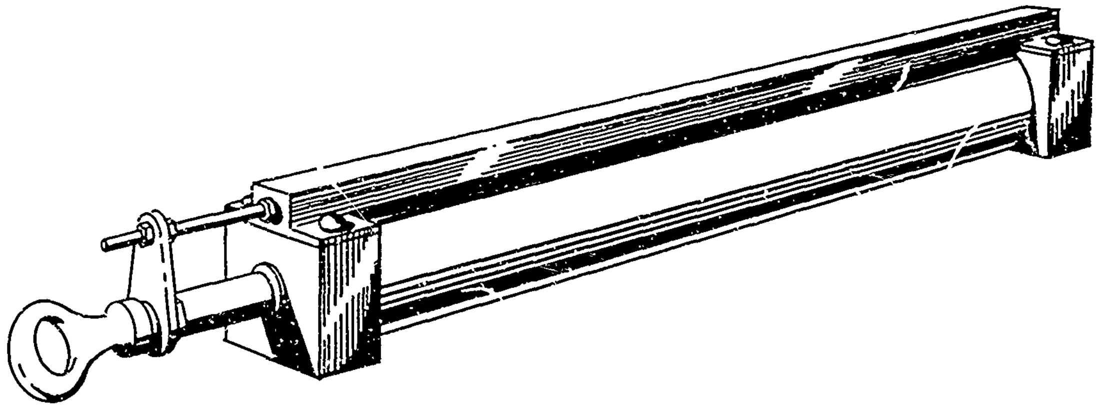




CYLINDER POSITION INDICATOR CIRCUITS



ROTARY TRANSDUCER



LINEAR TRANSDUCER

Figure 27. Hydraulic Cylinder Extension Transducers

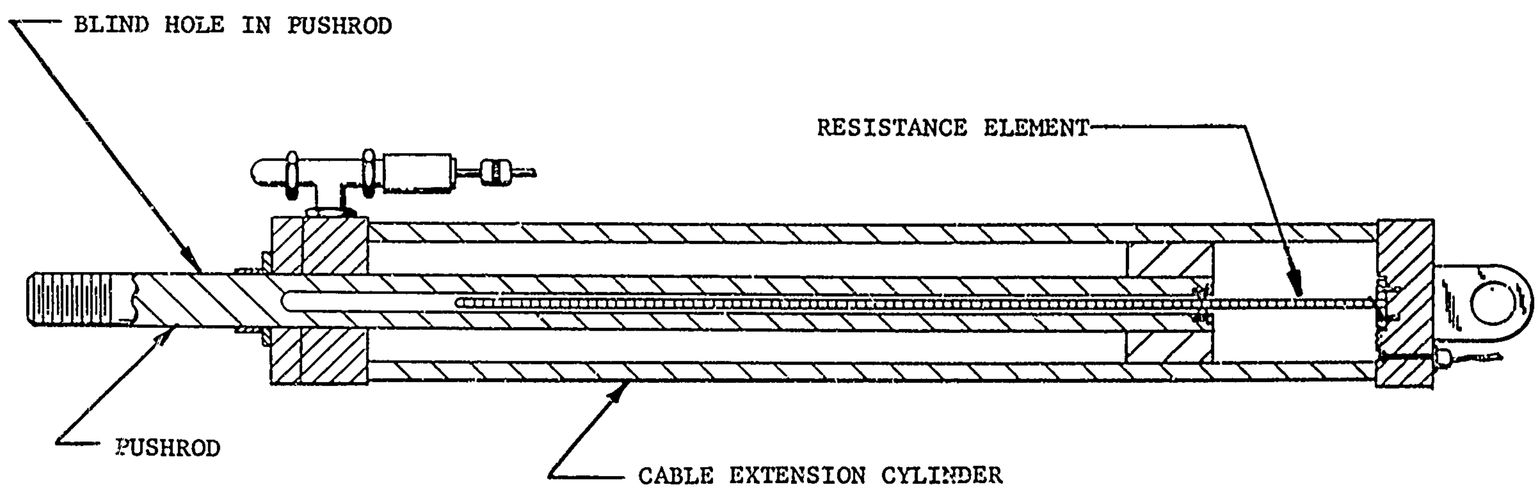


Figure 28. Built-In Transducer

An additional set of amplifiers, connected in a current summing mode, will add the load signals electrically and display them on a TOTAL LOAD indicator on both the pilot's and the copilot's panel.

CONTROL VALVES

Solenoid-operated three-way spool valves will be used to control the hydraulic power. Suitable valves are available commercially in a variety of configurations. A 1.0-inch valve size will be used to control the drum-driving hydraulic motor, while 0.5-inch valves are chosen for the hydraulic cylinders. They are discussed in the section "Hydraulic Systems."

Refer to Figure 22 for the schematic diagram of the hydraulic system.

ELECTRICAL POWER REQUIREMENTS

The electrical power consumption of the control system is relatively small, with hydraulic power performing the heavy work.

+28-volt DC power will be used for the major control functions, primarily the pilot-operated solenoid valves. These are short-term loads, occurring only during the load raising and lowering operations, and amounting to less than 1.0 kilowatt (36 amps at 28 volts) total. A short burst of power (approximately 1.0 KW) will also be required for opening the hooks. The other loads will be OFF during this 1-second operation. Instruments and panel illumination will consume little power, typically less than 0.2 KW total.

A small amount of 400 Hz single-phase power will also be required for operation of the drum-monitoring synchro transmitter with its associated cable extension indicator, as well as the cylinder load (strain gage) transducers. This total load will not exceed 0.1 KVA.

It is assumed that a 115-volt 3-phase 400 Hz power system will be available in the helicopter for operation of avionic and other equipment. If not, a small 26-volt 1-phase static converter, operated from the 28-volt DC system, can be incorporated.

OPTIONAL FINE/COARSE HOIST CONTROLS

The specification does not require a variable-speed or a two-speed control. The tentative control panel layout shows a DRUM UP and DRUM DOWN push button; hence, there are only three modes of drum drive: full speed up, off, and full speed down

The switches can, however, be furnished with a two-step arrangement: light pressure on the push button will turn the drum at low speed, and hard pressure will turn it at full speed. This would give a fine/coarse positioning capability which would ease the hoist operation during the last few inches of raising or lowering a load.

A continuously variable speed control is also feasible, but it would require replacement of the push button switches with levers or rotary knobs. If incorporation of the up/down control into the pilot's and copilot's control sticks is required, the placement of potentiometer or synchro command transmitters might present a space problem.

These details are, by necessity, tentative at this early stage of development; new requirements will be incorporated into the control system as they become necessary.

HOIST OPERATION

GENERAL

Hoist operation in the 1-, 2-, or 4-point mode consists of simple on-off operation of the hydraulic control of the single drum. Extend or retract motion of all hooks and cables is simultaneously accomplished by rotation of the drum in the desired direction. Individual hooks may be extended or retracted in the present design as much as 4 feet (this design dimension is tentative and may be increased or decreased). The individual control is accomplished by extending or retracting the cylinders hydraulically. The hooks may be remotely opened electrically, or opened and closed manually at the hooks. The load-master (ground) is provided with all the controls essential to the performance of his function (see "Instrumentation and Control" section of this report). The pilot and copilot are each provided with the controls and instrumentation essential to their operation of the hoist (See "Instrumentation and Control").

By adjustment of the individual cables, the hoist may be used in the 2- or 4-point mode to raise objects such as vehicles, etc., from an out-of-level attitude without introducing overloads in individual cables and a subsequently dangerous C.G. shift to the helicopter. This same operation is made relatively safe when depositing such loads on out-of-level surfaces. The helicopter crew is also provided with instrumentation to warn of a potential cable overload, should the event occur. As previously stated, this cable adjustment may also be used to change the flying attitude of the load and thereby provide greater aerodynamic stability.

CONFIGURATION CHANGEOVER

One of the most significant design parameters observed in the development of the three reeving configurations for this hoist system was the ability to change from one hoisting system to another within a short "turn-around" cycle. Figure 29 is an artist's sketch of the 1-, 2-, and 4-point systems superimposed to indicate the degree of complexity required for changeover.

A maximum of 20 sheaves is required for the single-point configuration, and a total of 16 sheaves is required in the 2- and 4-point reeving systems. This includes the sheaves contained in the lariat rope guides and the hydraulic cylinder yokes. Of these totals, a maximum of 4 sheaves must be moved to change from any one reeving configuration to another.

All structural connections are of the "PIP" pin type, so that no tools are required to effect a sheave change. Also, the sheaves that are relocated during configuration changeovers are held in their guards on "PIP" pin type shafts, so that it is possible to feed cables over these sheaves without any need of removing the hooks from the ends of the cables.

The four hydraulic cylinders are suspended on universal joints so that they orient themselves in all reeving configurations. Also, the piston rods rotate in the hydraulic cylinders, so that it will not be necessary to remove the cables from these sheaves during re-reeving operations.

The four-sheave yoke used in the center of the single-point configuration is hinged on one side; this permits easy placement of the cables on the four sheaves. The 5/8-inch-diameter steel cable and the 12-inch sheaves can readily be managed by a man of average strength and dexterity.

Sheaves not currently used in a particular configuration will be stowed in an out-of-the-way location in the hoist structure.

It is estimated that shifing of the system from the 1-, 2-, or 4-point arrangement to any other could be accomplished in about 30 minutes by two men.

In no case are the hooks ever removed from the ends of the cables. In the 1- and 2-point configurations, the hooks are joined by quick-disconnect bolts, as shown in Figure 15.

An exact procedure cannot be established until the configuration of the heavy-lift helicopter is known. However, using a scaled-up version of a Sikorsky CH-54A as an example, certain general statements can be made:

Each of the four load-carrying cables passes over three sheaves on the way from the drum to the hook:

Sheave I at the drum, as part of the lariat rope guide
Sheave II at the end of the load-balancing hydraulic cylinder
Sheave III at the cable outlet to the hook.

The sheave set III must be relocated for changing between configurations. This changeover can be accomplished if necessary by one person.

It is expected that a configuration changeover will normally be made at the home base of the helicopter before a mission. A self-propelled special-purpose gantry will ease the conversion.

In the field, the changeover can be made by one or two men, who enter the hoist frame by climbing the ladder rungs on the outside of the rear wall of the cockpit. Catwalks, handholds, and other support surfaces may be mounted on the hoist structure to provide a safe footing for the men.

A semiautomatic configuration changeover has been considered which would replace the 12 fixed sheaves with 4 movable sheaves and 4 track-like rails.

In this arrangement, each of the four load-carrying cables would always run over the same sheave; for a configuration change, each sheave could be moved along its rail-track between the outside corner (4-point), end center (2-point), and system center (1-point location).

From a weight aspect, the savings of eight sheaves would result, but four rail-tracks would have to be added. These tracks would be strong enough in the operating points to carry the rated loads, but they would be very light in the connecting portions.

REEVING CONFIGURATION CHANGEOVER - SHEAVE RELOCATIONS	
OPERATION	SHEAVE ASSEMBLY RELOCATION
1-POINT TO 2-POINT CONFIGURATION	SHEAVE(S) B RELOCATED TO POSITION 1 SHEAVE(S) A STOWED AT POSITION 2
2-POINT TO 4-POINT CONFIGURATION	SHEAVE(S) (B) RELOCATED TO POSITION 2 SHEAVE(S) (A) STOWED AT POSITION 1
4-POINT TO 1-POINT CONFIGURATION	NO SHEAVE RELOCATIONS REQUIRED

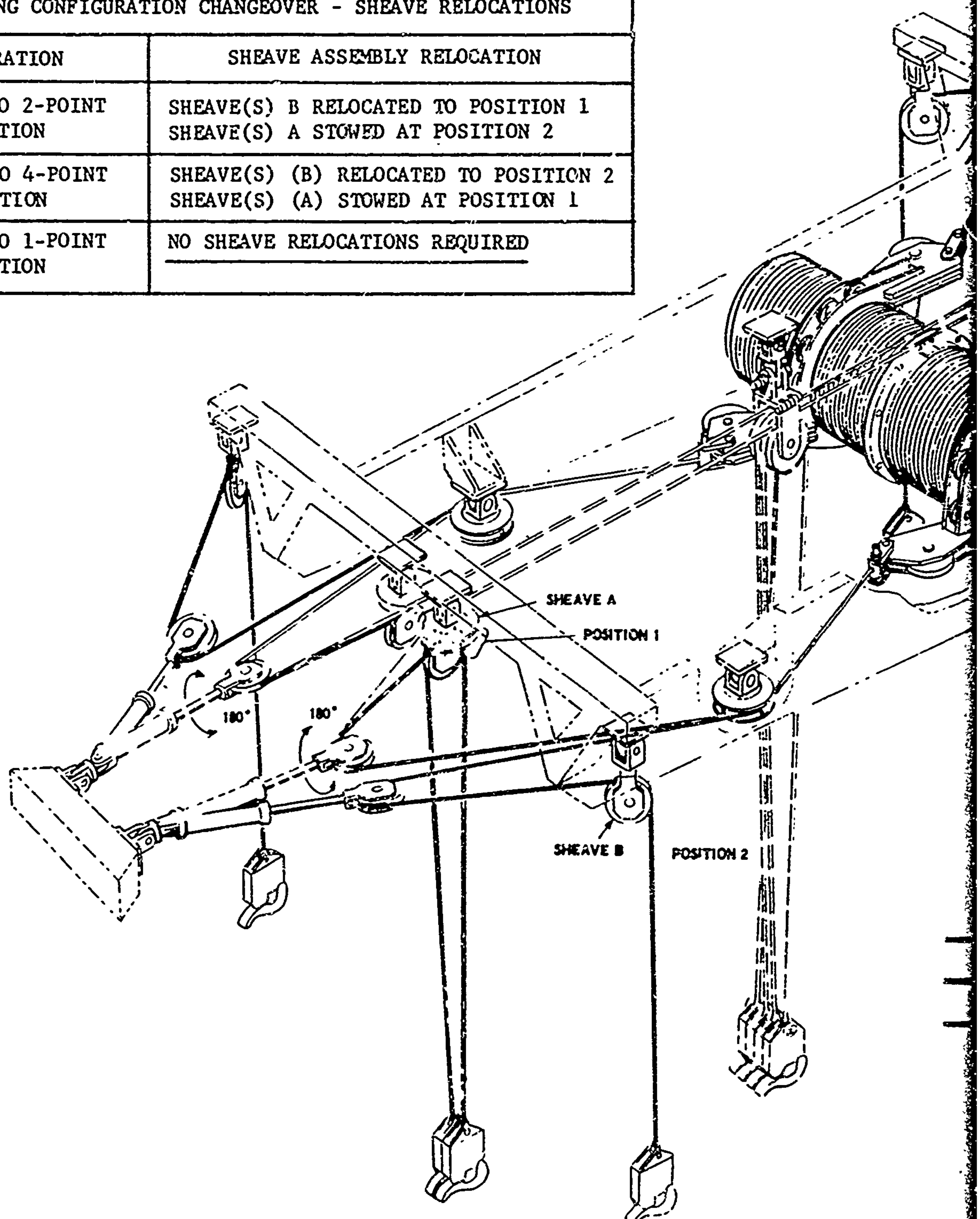
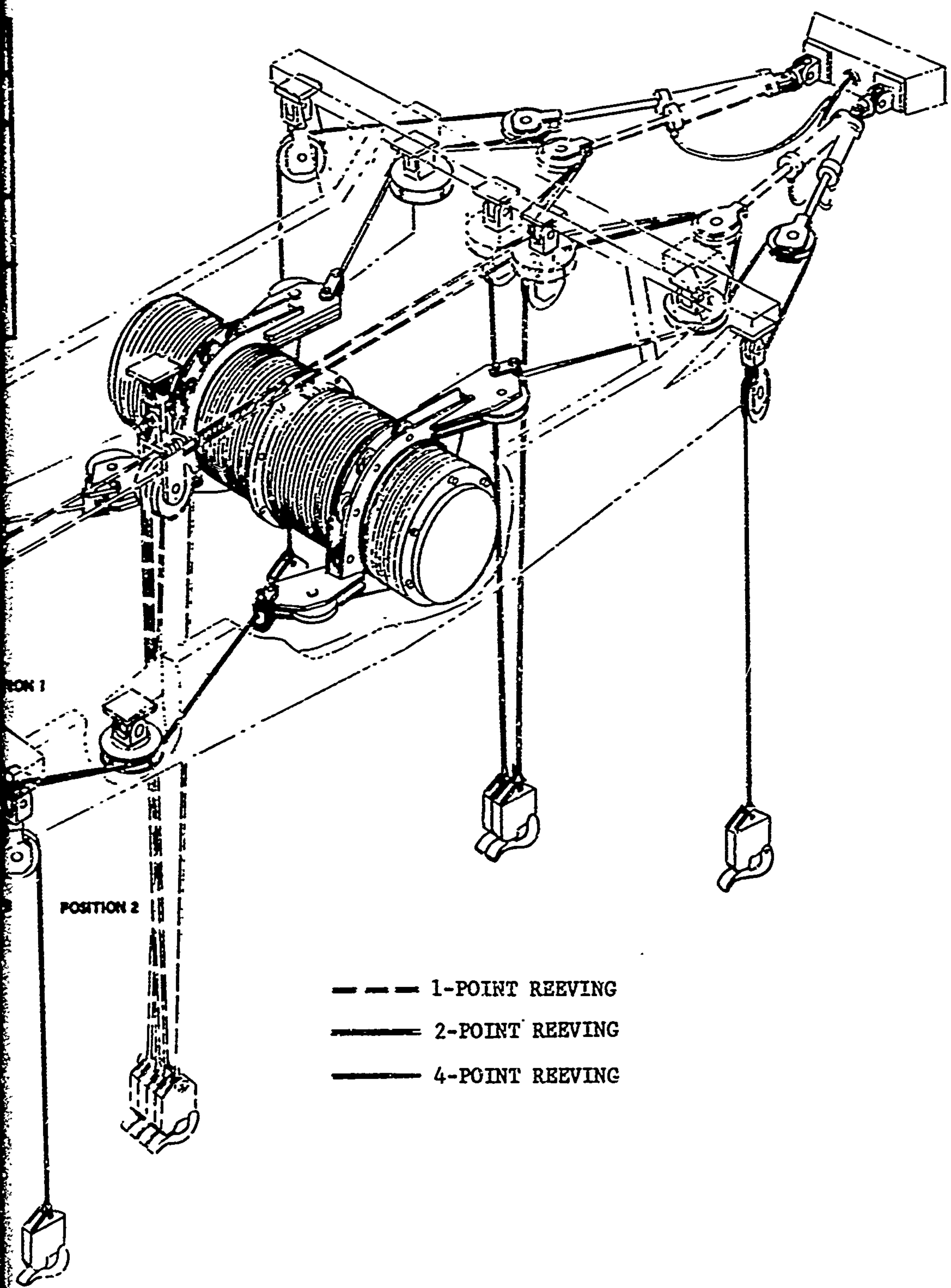


Figure 29. 1-, 2-, and 4-Point Reeving Systems Superimposed

A



B

Such an arrangement would have considerable merit for hoists in the 50,000-pound to 120,000-pound range, because the size of cables and sheaves would become too large to handle in the field without special tools. However, the weight penalty for a hoist in the 40,000-pound category would be prohibitive.

RELIABILITY

Reliability and safety of the cargo handling system are of paramount importance. To assure a high level of reliability, the following measures must be taken:

1. Use reliable, time-proven components.
2. Keep the number of components small and compatible with the required performance.
3. Design the system with low stress factors in all components; select a larger size component if necessary.
4. Isolate sensitive components from extremes of temperatures and vibration.
5. Keep close control of quality and workmanship in manufacturing and assembly.
6. Conduct environmental tests on prototype units.
7. Conduct an MTBF (mean time between failures) analysis in order to obtain a statistical estimate of the system's reliability. Modify those subsystems having high probability of failure to yield greater reliability.

MTBF COMPUTATION

The original formula, as established by D. R. Earles (see Ref. 14), is

$$M = MTBF = 10^6/n \times G_f \times K_{op}$$

where M = Mean time between failures (hours)

n = Number of components

G_f = Generic failure rate of the component/10^6 hours

K_{op} = Operating environment stress factor (50 for airborne equipment)

This simple formula applies only if there are identical components in the system, or different components of the same failure rate. For a practical system with a variety of different components, the system failure rate is the sum of the component failure rates:

$$MTBF = 10^6/(n_1 \times G_{f1} \times K_{op}) + (n_2 \times G_{f2} \times K_{op}) + (n_3 \times G_{f3} \times K_{op}) + (\ldots)$$

where n_1 = Quantity of servo valves

G_{f1} = Failure rate of servo valves

K_{op} = Operating environment stress factor

n_2 = Quantity of push-button switches

G_{f2} = Failure rate of push-button switches

n_3 = Quantity of relays

G_{f3} = Failure rate of relays

Tabulations of generic failure rates for electrical components are given in MIL-HDBK-217 A, dated 1 December 1965 (see Ref. 15). For many components, the product of generic failure rate and operating environment stress factor is presented in the form of graphs or tables. The values most closely applicable to the parts and environment of cargo handling systems have been selected from MIL-HDBK-217 A.

The major components in a manual-control system which could cause a system failure are described below.

Push-Button Switches

Push-button switches, using an actuator and sensitive switches, made to military specifications, are produced by Master Specialties, Inc., and the Microswitch Division of Honeywell, Inc. The Microswitch units are available with a greater variety of switch modules. The Microswitch (Honeywell) Series 2 has been chosen, equipped with momentary actuator and two V-3, SPDT 15-amp switch capsules; the V-3 is a rugged, high current capsule, and is more reliable than the subminiature types. MIL-HDBK-217A gives the failure rate of such switches as:

G_f = 0.3 failure / 10^6 operations

G_f = 0.03 failure/ 10^6 hours, with an operating environment stress factor of K_{op} = 50 for airborne use.

Relays

Relays, hermetically sealed, with balanced armature and over 1.0-cubic-ince size for missile and aircraft applications, are available as Military Standard part MS 25271. This is a four-pole double-throw unit with 10-ampere contacts. It is a miniature contactor which is QPL'd from Leach, Inc., Guardian, Inc., and others, and which has a long history of rugged, reliable operation. Its coil has a rating of 28 volts DC, 0.35 ampere.

MIL-HDBK-217 A gives its failure rate as:

G_f = 0.1 failure / 10^6 operations

G_f = 0.002 failure/10^6 hours, with an operating environment stress factor of K_{op} = 50 for airborne use.

The relay is made for 10 g's vibration to 1500 cps and 25 g's shock.

Connectors

Connectors will be the AN aircraft type (MS 3100 series) with a typical failure rate (Ref. 15) of G_f = 0.5 failure/10^6 hours, with an operating environmental stress factor K_{op} = 50 for airborne use.

Circuit Breakers

Circuit breakers will be the aircraft, magnetically tripped type with a failure rate of 0.14/10^6 hours.

Resistors, Capacitors, Silicon Diodes

Resistors, capacitors, and silicon diodes of the highest reliability will be used.

It is to be noted that transducers, synchros, and indicators have not been included in the reliability estimate. In case of a failure of one indicator, the remaining indicators will give sufficient data to complement each other. Also, the copilot can operate the hoist by observing the load through the rear window; hence, an instrument failure need not be considered a system failure.

While MIL-HDBK-217 and other sources give reasonably accurate data on electrical components, very little reliability information is available on mechanical and hydraulic components. The data compiled by D. R. Earles (see Ref. 14) between 1955 and 1960 on the Titan and other missile projects are the best available, but the generic failure rates and operational stress factor data are often contradictory and allow as much as 20:1 variation for some component categories.

Mindful of these limitations, a tentative reliability estimate has been compiled in Table VI. It yields a failure rate of 7893 failures per million hours, or an MTBF of:

$$MTBF = 10^6/G_f = 1{,}000{,}000/7893 = 125 \text{ hours.}$$

This is a comparatively good MTBF, with most of the failure probability in the hydraulic components. Proper contamination control (filtering) and periodic inspections will minimize the risk of an in-flight failure. In the AAI design, a defect in a hydraulic cylinder valve would not be serious, because proper control of the three other cylinders could compensate for the failing cylinder. Also, the presence of two identical control panels provides good redundancy against instrument or cable defects.

The reliability of the EMERGENCY system is high, because of the triple redundancy contained in the HOOK OPEN, FREEWHEEL and CUT CABLES circuits.

A more accurate reliability analysis will be conducted when the design of the hoist system is defined in more detail.

TABLE VI. RELIABILITY ESTIMATE

Component	G_f F/10^6 hrs.	K_{op}	Qty. Used	G_f Total
Hydraulic Pumps	16.000	50	2	1,600
Hydraulic Motor	4.300	50	1	216
Hydraulic Cylinders	0.120	50	4	24
Hydraulic Cylinder, Brake	0.500	50	1	25
Hydraulic Valves, Spool Type	6.900	50	6	2,070
Hydraulic Valves, Check	3.600	50	8	1,440
Hydraulic Valves, Relief	3.270	50	4	654
Hydraulic Pressure Hoses	2.300	50	10	1,055
Gearbox	0.360	50	1	18
Circuit Breakers	0.140	50	6	42
Push Button Switches	0.100	50	18	90
Relays, Sealed	0.160	50	8	64
Connectors, AN Type	0.500	50	12	300
Resistors	0.030	50	10	15
Capacitors	0.600	50	6	180
Silicon Diodes	0.200	50	10	100
				7,893 Failures/10^6 hrs

MAINTAINABILITY

DRUM SYSTEM

The internal drive drum is basically a cylinder with all machinery within itself. (See Figure 13.)

All parts with relative motion will be adjusted and prelubricated before final assembly. Bearings will be packed with lubricants per the manufacturer's recommendation. The "cyclocentric" gearing will be coated with a brush-applied or baked-on moly-disulphide type, dry, long-lasting lubricant.

Because of the low RPM of the large components, servicing is expected to be held to a minimum. The unit will function trouble-free for approximately 100 hours of operation, at which time a major inspection must be made.

To service the drive drum system, the mechanical, hydraulic and electrical lines must be disconnected and the drum removed from the helicopter.

To open the drum, the bearing retainers are taken off, the center flange screws are removed, and the right-hand half of the drum is withdrawn to expose the torsilastic dampers, etc.

To service the left-hand half of the machinery, the retaining ring fastening the outer ring bearing is removed and the left-hand half of the drum is slipped off.

Access to the centrally located units such as the motor, brake and gearing can then be made by disconnecting the flexible coupling.

REEVING SYSTEMS

The maintenance requirements of the reeving systems are minimal. "Fabroid"-type bearings, which require no lubrication, are used on the sheave assemblies. Thrust ball-type bearings are used in the 2-point and 4-point sheave yoke assemblies to permit swiveling of these yokes. These bearings will require oiling only at infrequent periods of time since, the rotation they experience in swiveling will be small as compared to the relatively high rates of rotation the bearings are designed to withstand.

Periodic inspections of the wire rope will be required to check for wear and broken strands in the cable, and to lubricate this cable as required. This procedure can be integrated into the regular operational maintenance schedule of the helicopter.

The only other operational maintenance required in the reeving systems will be lubrication of the pins in the hydraulic cylinder universal joints to insure free movement of the joints.

ELECTRICAL SUBSYSTEMS

All electrical subsystems and components are equipped with AN type aircraft connectors, for ease of replacement. The two control panels are mounted with 1/4-turn (DZUS, Comloc or equivalent) fasteners for instant access or replacement.

The electrical junction box (containing the relays, pressure gage amplifier, cable cutter, standby battery, etc.) is mounted on a bulkhead near the hoist. It is fastened with aircraft-type bolts and connected with AN-type (MS3100 series) connectors.

SAFETY

GENERAL

All through the design of the cargo handling system, safety has been treated as the most important parameter. Weight and cost considerations are also important, but nowhere has safety been traded off for weight and cost savings.

The mechanical, structural and hydraulic design of the hoist incorporates generous safety factors, and should not pose any problems under normal operation with loads up to and including full capacity.

The electrical control system incorporates double sets of instrumentation and control. The pilot and copilot will have separate, identical control panels, each capable of reading the cable loads and positions and of exercising control over them. Such a configuration is extremely reliable under normal operating conditions. Special consideration has, however, been given to limited war missions of the system, with the risk of damage by small-arms fire. The wire bundles from the two control panels to the hoist will be run separately. In case of bullet damage to one set of wires, the other probably will still be usable.

No hydraulic lines will be run from the hoist to the cockpit; instead, all pressure transducers will be located at the hoist, with electrical signals only to run to the pressure (load) indicators in the cockpit.

AUTOMATIC BRAKE

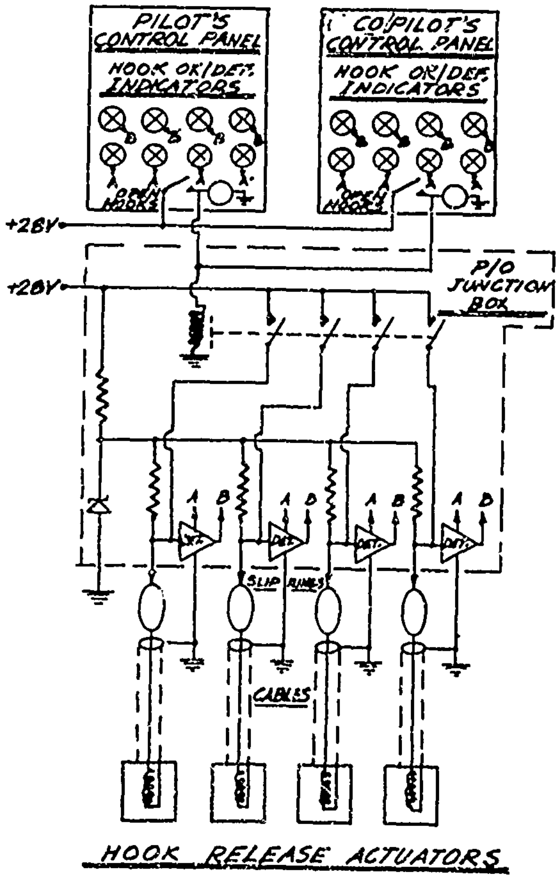
A brake is mounted in the drum. Under normal operating conditions, this brake is held open by a hydraulic cylinder, which in turn is energized by an electromagnetic solenoid valve. In case of an electrical or hydraulic system failure, a strong coil spring will apply the brake. A "BRAKE RELEASE" lever in the cockpit, connected through a lanyard to the brake, will enable the crew to override the automatic braking action mechanically.

EMERGENCY DEVICES

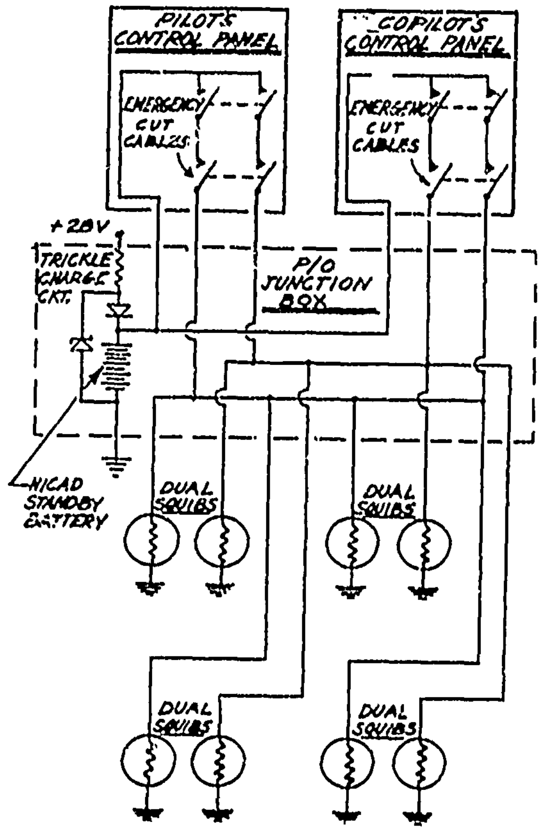
If an emergency should occur in flight, such as a broken load-cable or hook, the following actions can be initiated on one of the two control panels to jettison the load:

Opening the Hooks

(See Figure 30, which shows the basic hook circuits.) Pressing the "HOOKS OPEN" button on one of the control panels will open all four hooks within less than one second and drop the load. (A spring-loaded, transparent safety cover must be lifted in order to push the "HOOKS OPEN" button. This prevents its accidental operation.)
The electrical power for the hook-opening actuators is carried from the 28-volt DC source via the control panel switches and slip-rings



HOOK CONTROL CIRCUITS



CABLE CUTTER CIRCUITS

Figure 30. Electrical Control Circuits

(inside the cable drum) through an insulated center-conductor of the load-carrying cable down to the hooks. (The load-carrying cable itself serves as a return path for the actuator current.)

A set of warning lights on each control panel indicates whether the current path in each of the four load-carrying cables is in working condition. "HOOK OK" is indicated normally; if the current path is interrupted or short-circuited, a "HOOK DEF." light would be energized. Electrically, this as accomplished by continuously sending a small DC current (in the order of 5 percent of that required to open a hook) through the cable center-conductor, the actuator-coil and back, and measuring the voltage drop across this circuit. If it deviates by more than ± 30% from normal, the solid-state detector circuit will turn off the "HOOK OK" light and energize the "HOOK DEF." light in each panel.

Freewheeling the Cables

Lifting the transparent safety cover and depressing the EMERGENCY and FREEWHEEL buttons simultaneously will cause the cables to turn the drum under the pulling force of the load, causing the cables to drop free.

To avoid accidental operation of this emergency feature, momentary action push buttons have been provided. (The drum rotation would stop if the buttons were released before the cables have fallen free completely.)

In case of an electrical or hydraulic power failure, freewheel action can be obtained mechanically by pulling the BRAKE RELEASE lever described previously. The lanyard not only overrides the closing spring of the automatic brake, but also opens a bypass valve in order to permit the rotor of the hydraulic motor to turn easily under the torque exerted by the load on the drum.

Cable Cutting

The ultimate safety device for the helicopter and its crew in serious emergencies is the cable cutter.

The cable cutter system will quickly and reliably jettison the load. The copilot or pilot lifts the safety lid on his control panel, then presses the EMERGENCY and CUT CABLES buttons simultaneously. This energizes the hook opening mechanism and, after a short (approximately one second) time delay, fires the dual squibs of the cable cutters. Backlash of a severed cable cannot occur on the drum, as the lariat rope guide always covers the area of the cut cable and where it lies on the drum. Minimization of backlash away from the drum will be accomplished by opening the hook slightly in advance of the cable cutter. Thus, backlash in the system would occur only if electrical release of the hook does not function. If experimentation tests show other damages occurring from backlash, the cables could be screened.

The electrical circuits of the cable cutter system (see Figure 30) are completely isolated from the electrical power system and equipped with a small nickel-cadmium battery. This battery is kept charged by the main 28-volt DC system through a trickle-charge circuit and disconnect diode, but remains operational if the main electrical system should fail. Every cable cutter is equipped with two squib detonators for extreme reliability. Each squib is connected to each of the control panels by separate wires; hence, the probability of the cable cutter system's becoming inoperative due to wire damage caused by fire from the ground is extremely low. Automation of the cable cutters has been considered. This system would sense any possible load-cable break instantaneously and would jettison the load faster than a human operation would be capable of. Load transients (in very rough air) or electrical systems transients could, however, cause unwanted cable cutter operation when no emergency exists. Proper smoothing and integration of the signals is feasible, but this will tend to lengthen the reaction time of the system and make it as slow as that of a human operator, or slower. It is therefore proposed to use the cable cutters manually only.

CABLE STRESS RECORDING

The load-carrying cables will have a long, useful life if they are used within normal operating limits. The replacement of individual cables or complete sets will become necessary under the following conditions:

Severe wear, indicated by flattened areas on the outer cable strands, or broken strands.

Serious nicks or kinks which would occur if tangled cables are retracted by force; the latter might become necessary under combat conditions.

Cables which have been exposed to high stress, although they appear outwardly perfect. For example, a cable designed for 10,000 pounds nominal load, and with a theoretical yield strength of 27,000 pounds, should be replaced whenever a stress exceeding 20,000 pounds has occurred. This stress can be read on the cylinder load indicators of the control panels, but chances are that the overload is not observed accurately, if at all.

A more reliable method of registering excess cable loads would be to record them automatically.

For the prototype and early production units, it may be justified to make continuous strip-chart recordings of all the important parameters. A multi-channel graphic recorder (Sanborn, Brush, CEC, or Visicoder) would be most suitable.

In production machines, some type of peak-load registering device should be installed. This device could be a pointer-type instrument (similar to a 3-needle registering G-indicator), or a stylus-on-graph-paper type instrument (similar to shock-recorders shipped with delicate equipment cargos). It could also be an odometer (counter) type instrument with a simple potentiometer servo. In any case the recording instrument would not store the complete load time history, but only the peak stress value.

Such instruments would help to recognize overstressed cables which appear to be perfect but have broken strands inside or dangerously overstretched sections, and which might fail even under moderate loads in future missions.

In conclusion, it can be said that an automatic emergency detector and cable cutter system can be incorporated if desired; at this time, however, it appears that an all-manual system is to be preferred.

SYSTEM WEIGHTS

The comparative weight estimate, which was made in great detail, is summarized herein.

The weight estimate of the system presented resulted in the following totals:

For the 1-, 2-, & 4-point, 150-ft lift, the weight is	4,598 lbs
For the 1-, 2-, & 4-point, 80-ft lift, the weight is	3,745 lbs
For the 1- & 2-point, 80-ft lift, the weight is	3,220 lbs

The weights of the second and third methods above are calculated by subtracting only weight reductions from the first total. The major difference results from reduction in cable weight and drum size.

The following is a tabulation of the major weight items in the 1-, 2-, and 4-point 150-foot system. In every case, each major item includes all known details, down to nuts, bolts, screws, washers, etc. This estimate is conservative.

Weight Tabulation

1.	Drum Assembly	3,590 lbs
2.	Reeving System	858 lbs
3.	Hydraulic lines and valves not included above	100 lbs
4.	Electric system components not included above	50 lbs
	Total	4,598 lbs

CONCLUSIONS

The AAI single-drum cargo handling system is feasible and can be built with off-the-shelf components or minor variations of existing component designs.

The use of a single drum with an integral torsilastic vibration isolator, common to all four cables, will eliminate the risk of dangerous levels of vibration and oscillation. Also, this design yields a simple, easily controlled system with relatively low cost but high reliability.

The system has considerable versatility; only the placement of the last sheave is critical. The storage drum and drive, along with the balance of the rigging, can be located in places most adaptable to the helicopter itself. Providing for alternate locations of the last sheave would permit the system to handle loads having C.G. locations not suitable for handling with less versatile hoist systems.

The reliability will also be high, as no exotic or untried components will be used. All materials proposed for use are in common usage today. Standard hydraulic components and MIL-H-5606 hydraulic fluid have a long history of dependable operation.

It is concluded that the weight limit of 4,000 pounds can be easily arrived at, using the AAI design, if either a 1- and 2- or a 1- and 4-point suspension is used. If the 1-, 2-, and 4-point suspension is built, the 4,000-pound limit may be closely approached. The preliminary weight estimate (included in this report) is approximately 15 percent above the desired maximum weight. Since this is a conservative estimate based on incomplete detail design, it is anticipated that the finalized hoist will weigh no more than 4,300 pounds.

Purchasing costs of motors, pumps, valves, etc., can, in the AAI design, be held to a minimum, as serious effort has been made to locate and adapt existing components to the system. This effort results in the elimination of design and tooling charges incurred when special components are required.

The contractor has been unable to provide the design with a practical mechanical hook opening method which can be operated from the helicopter. The use of additional mechanical equipment extending from the helicopter to the hooks poses problems that are most severe. The weight increase is prohibitive, safety jeopardized, maintenance increased, and hoist operation complicated. The problem of maintaining positioning of a mechanical release coupled to a tension member which will be subject to different static loads coupled to dynamic conditions is a severe one. The problem is complicated by the difference in "stretch" in the tension member when its extended position is variable. This contractor has provided an alternate means of assuring the safety of the helicopter and crew by a combination of electrical hook opening, cable cutting and complete mechanical release of the cable from the helicopter, discussed under the "Safety" section of this report.

The problem of a realistic illustration of the support structure for the hoist has not been investigated to any depth. Such a study is not reasonable until such time as coordination of the helicopter and hoist designs is made.

The basic system, as described here, has the following capabilities:

1-point configuration with a 40,000 lb, 150-ft lift

2-point configuration with a 40,000 lb, 150-ft lift

4-point configuration with a 40,000 lb, 150-ft lift

The use of the 1-, 2-, and 4-point 150-foot system offers the best opportunity for exploring all of the methods deemed advisable for exploration, by Government personnel, on the same frame at the same time. Therefore, the additional cost of building and testing alternate systems to explore alternate methods will not be necessary.

The growth potential of the AAI system is excellent; all the cables, hooks, bearings, hydraulic cylinders and motors can be scaled up for cargo handling capacities of 60,000, 80,000 and 120,000 pounds. Hydraulic pumps and motors in the 100 to 300 HP range are presently being built for the Boeing 747, C5A and SST aircraft.

The only limiting factor in scaling up the size of the cargo handling system is the muscular strength of the men making the configuration changeover. While the 5/8-inch steel cable and 12-inch/13-pound sheaves of the 40,000-pound system are reasonably manageable, a practical limit for manual configuration changeover will be reached around 100,000 pounds capacity. Two means of extending the men's effectiveness are feasible: (a) use of special jigs, clamps and miscellaneous tools, and (b) use of a configuration of track-suspended sheaves which are moved from one point to another without the necessity of removing the cable (see "Hoist Operation" section).

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13. ABSTRACT

The purpose of this program was to design a helicopter hoist system capable of lifting 40,000 pounds. Desired features of the hoist system included a weight limitation of 4,000 pounds and the ability to provide either a 1- and 2-point load suspension system or a 1- and 4-point load suspension system.

Major areas of investigation included a study of primary tension members; the relative advantages of multi-drum and single-drum hoist systems; single wrap and overlaying of cables on hoist drums; and the desirability of including vibration isolation components between the external load and the helicopter.

Through study and tests, it was determined that it was possible to provide a combined system which permits 1-, 2-, and 4-point load suspension. This feature is achieved by use of a single cable drum winding four separate cables. Varying the method of reeving the cables permits the selection of either a 1-, 2-, or 4-point load suspension system. Load attitude and cable loading may be adjusted by means of four hydraulic control cylinders included in the system. Using a single drum with four cables eliminates the complex synchronization problems associated with a multi-drum drive system.

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Hoist, Helicopter, Heavy-Lift						
Hoist, 40,000 Pound						
Suspension; 1-, 2-, and 4-Point Load						
Cargo Handling System						
Helicopter Cargo Handling						