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USAAVLABS TECHNICAL REPORT 67-51

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

AD826531

By

T. Lancashire

R. Kalpas

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

CONTRACT DA 44-177-AMC-435(T)
VERTOL DIVISION
THE BOEING COMPANY
MORTON, PENNSYLVANIA

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This design study was based on previous investigations of problems associated with the mechanics of cargo handling by aerial-crane-type aircraft (USAAVLABS Technical Report 66-63). This report is one of three contract studies of the same problem with varying technical approaches.

In some respects, this proposed system concept departs from established helicopter hoist system practices; thus, further evaluation from a system concept will be necessary to determine its ultimate value as the heavy-lift helicopter external load handling system.

In general, this command concurs with the results of this preliminary design study. The preliminary designs developed by the contractor are complete, accurate, and in sufficient detail to provide a basis for component development programs.

Future work to be considered by this activity relative to this area includes an analysis of the three preliminary contract designs so as to define an optimum system based on the best features of each. This may be followed by component development and test for critical items as appropriate and the detail design, fabrication, and test of an experimental system.

Project 1X130901D332
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October 1967

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

Final Report

D8-0491

by

T. Lancashire
R. Kalpas

Prepared by

VERTOL DIVISION
THE BOEING COMPANY
Morton, Pennsylvania

For

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

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SUMMARY

The purpose of this two-phase investigation is to define the optimum design of a heavy-lift helicopter external cargo handling system of 40,000 pounds capacity. This report covers the results of an interface subsystem configuration analysis and defines the load handling winches and hoist systems required for a feasible, overall cargo handling system (including those aircraft-related systems necessary for control -- actuation and monitorship, load attachment, suspension, hoisting and shock dampening).

PHASE I - DESIGN ANALYSIS

The analytical study considered all elements which would ultimately be integrated into a complete airborne external hoist system. They include: flight safety during all phases; the effects of center of gravity on system design (aircraft with and without load); cost and time estimates for system design, fabrication, and qualification testing; and finally, the overall feasibility, efficiency and reliability of concepts for handling external cargo by heavy-lift-type helicopters.

The two-point hoist system which provides the required load restraint proved to be the lightest, most reliable and least expensive of the methods investigated and best satisfied the multi-point hoisting mode. Although feasible, the four-point hoist system, which is heavier, more expensive and requires more development time than the two-point system, requires hoist synchronization to meet flight safety criteria.

A beam system integrating the cargo hooks provided a common base for mechanically connecting the hooks and releasing them simultaneously, while the use of a reeved cable system was dictated when lift capacity exceeds the 1-inch-diameter wire rope limit. Also, capstan-type winches are considered to be superior to drum-type winches, because of their zero-moment feature and their adaptability to different aircraft configuration installations.

The load isolation concept as a solution to vertical bounce phenomena was analyzed, and various design principles were described. Ultimate selection of load isolators is dependent upon selected installation criteria.

PHASE II - PRELIMINARY DESIGN

The preliminary design effort presents a two-point, two-winch hoist system which appears to be capable of providing both multi- and single-point lift modes.

These preliminary design drawings, provided in sufficient detail, should aptly support a basis for future detail design, fabrication and qualification of an experimental hoist system.

This experimental hoist system is a two-winch, two-point system consisting of two capstan-type winches, the arrangement of which results in a winch system free to pivot about the prime mover longitudinal axis. The suspension system (one wire rope from each winch) terminates at the ends of a beam to which are secured two cargo hooks. The beam forms a common base for the interconnection of the hook release mechanisms, permitting simultaneous release of the hooks through a common electromechanical actuator, thus satisfying the specified normal mode of release operation. Manual release of the system (emergency mode) utilizes the same basic linkage except that a manual input is applied in lieu of an electrical one.

The preliminary design studies also include supporting analyses of load and stress, maintainability and reliability as well as safety and system growth in order to confirm the feasibility of the system.

Planning cost and development time estimates for both the design analysis (Phase I) and preliminary design (Phase II) are included in this report.

LEADING PARTICULARS OF HOIST SYSTEM (Designed in Phase II)	
Winches (2)	Capstan Type
Dynamic Capacity (per winch)	25,000 lbs
Lift Rate	60 ft per min
Wire Rope	1-in. dia, type 6 X 37 Warrington Searle
Wire Rope Dia/Drum Dia Ratio	1:18
Lift Height	150 ft
Cargo Hooks (2) - Type	Mechanical Release
Release Normal Mode	Electromechanical
Release Emergency Mode	Mechanical
Indicators	Hook Open/Hook Closed
Electrical Conductor Reel	
Normal Operation	Electrical Torque Motor
Emergency Release	Mechanical
System Control	
Winch Control	Independent variable speed control of each winch plus integrated control through one four-way switch
System Emergency Release	Dual cartridge type cable cutters on each winch - electrically initiated through separate circuits

Rope Length Indicator	Provision on winches for driving a rope footage indicator pot plus limit switches to control max & min rope lengths
Braking System	Automatic load brake system, in which the brake unit is an integral portion of the drum drive mechanism

System Weight	
Winches (2 X 1198.09)	2396.18
Beam assembly	182.20
Release unit (with 160' of rope)	54.30
Winch supports (2 sets)	85.16
Load isolators/Load cells	80.00
	2797.84
Single Point mode hook & ropes	210.00
	3007.84 Pounds

FOREWORD

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SYMBOLS

A	area, square inches
A_e	effective area
C_d	aerodynamic drag
C_{LA}	specific heat, light alloy
C_{ST}	specific heat, steel
C_O	specific heat, oil
CG	center of gravity
D	load aerodynamic drag, kips, pounds
D_e	effective diameter
D_p	pitch diameter, inches
E	modulus of elasticity in tension and compression
f	friction coefficient
F	force
F_c	elastic primary compressive stress at failure, pounds per square inch, thousands of pounds per square inch
F_{cc}	crippling stress, pounds per square inch, thousands of pounds per square inch
F_{cy}	compressive yield stress
F_{su}	ultimate shear allowable
F_{tu}	ultimate tension allowable
F_{ty}	tension yield allowable, pounds per square inch, thousands of pounds per square inch
g	acceleration of gravity
G	modulus of elasticity in shear
GPM	gallons per minute
H	heat

I	moment of inertia, inches ^{4}
I_g	geometry factor for bevel gears
ω_I	load isolator natural frequency, radians per second
K	kips, 1,000 pounds
K_1	load isolator
K_2	sling spring rate
K_R	resultant spring rate
L	load
L	column length, inches, feet
m	external load mass $(\frac{lb\text{-}sec^2}{in.})$
M_{LA}	mass, light alloy
M_{ST}	mass, steel
M_O	mass, oil
M	moment
MS	margin of safety
ΣM_p	O = Aa + Ba - Ca - Da
ΣM_r	O = Ab - Bb + Db - Cb
M_x, M_y, M_z	bending moments in x, y, z planes respectively
N	number of teeth, rate
ω_n	resultant natural frequency, radians per second
P_d	diametrical pitch
P_{I_G}	operating pressure
PSIG	pounds per square inch gauge
q	dynamic pressure, pounds per square foot, pounds per square inch

Q	BTU , flow
R	radii
R_M	mean radius
RPM	revolutions per minute
S	body frontal area, feet square
S_c	hertz stress, pounds per square inch
S_t	tension stress, pounds per square inch
t, t_1, t_2	thickness, temperature
T	torque
V	internal volume
ΣV	O = A + B + C + D = 40,000
W	weight of external load, pounds, kips
W_t	tangential component of gear load, pounds
X	resulting moment arm
x, y, z	rectangular coordinates - longitudinal, lateral, and vertical axis respectively
Y	the offset
Y_k	tooth form factor including stress concentration
ρ	air density, slugs per cubic feet
ϕ	gear tooth pressure angle, degrees
ω	natural frequency, radians per second

INTRODUCTION

One of the most essential elements which must be considered in the development of a heavy-lift helicopter is its external cargo handling system. Operationally, the heavy-lift helicopter will be employed as an aerial hoist and transporter for various heavy loads, including combat vehicles and other large, bulky items which cannot be lifted by other helicopters. The heavy-lift helicopter's attributes must include the ability to deliver its cargo by landing and by winching down while hovering.

Prior research in the field of airborne external load handling winch and hoist systems indicates that major gaps exist in load-lifting technologies, particularly in the capacity ranges of proposed heavy-lift helicopters. Typically, these gaps result in excessive hoist component weights, incompatible configurations, poor reliability and maintainability, inadequate in-flight safety provisions, complex control, and other undesirable factors.

The basic element contributing to these discrepancies has been the fact that in a majority of cases, cargo handling systems have been considered to be a secondary function of the complete weapons system.

The philosophy of building helicopters and then designing a system for carrying external cargo has resulted in system compromise. Since it is proposed to build aircraft whose prime function will be to acquire, transport, and deliver cargo externally, the ideal plan should be to design the cargo handling system and then to design a helicopter to fit the system. In order to accomplish this ideal design, it must first be ascertained just what limitations the available technology and design techniques impose upon design of the elements which comprise an external cargo handling system.

Cargo handling techniques have developed in an evolutionary manner to the current state of the art. Interest by the Army in a heavy-lift helicopter with a lift capacity of between 12 and 20 tons required that cargo handling techniques for external loads be further evaluated. This interest led to a request for proposal from USAAVLABS to perform a study of a heavy-lift helicopter external cargo handling system of 40,000-pound

capacity and to the subsequent award of a contract for such a study to The Boeing Company, Vertol Division.

PHASE I - DESIGN ANALYSIS

The objective of the Phase I, design analysis part of the heavy-lift helicopter external cargo-handling system study is to conduct investigations and studies to identify and define equipment, subassemblies and components which are applicable to multi-point and single hoist systems concepts. The equipments are defined in sufficient detail to facilitate a qualitative and quantitative design appraisal. Such an appraisal will thus permit the selection of a hoist system considered best to satisfy the external cargo-handling requirements of future heavy-lift helicopters.

THE INFLUENCE OF WIRE ROPE CONSIDERATIONS ON HOIST SYSTEM DESIGN

The element which influences the design of a hoist system more than any other is the hoist system cable. The majority of hoist systems use wire rope, although some types are in use which use roller chain, link chain or tapes. Roller chain hoist systems are special-purpose systems, usually employed where the lift height is small or where there is no tendency for the suspended load to swing (i.e., a roller chain is primarily a power transmission medium designed specifically to take tension; it is not designed to take any bending due to side loads). Link chain is used in low-speed (usually hand-operated) hoists. A comparison of the relative weights of wire rope, link chain and roller chain, shown in Table I, points out the major reason for wire rope being the obvious choice, where minimum weight is a major consideration.

TABLE I
COMPARATIVE WEIGHTS OF HOIST MEDIA FOR THE SAME LOAD

Type	Weight per 150 Ft (lb)
Wire Rope (Type 6 x 19)	495
Roller Chain - Double Width	2,670
Roller Chain - Single Width	3,100
Link Chain	4,200

The use of tape winches is relatively new, and investigation shows that such a system shows promise of satisfying the requirements of the heavy-lift helicopter. Tape winch technology will be discussed under a separate section of this report.

In order to appreciate fully the application of wire ropes to hoist systems for the proposed heavy-lift helicopter, contact was established with the leading wire rope manufacturers in this country. Over a long period of usage, certain qualification criteria have been established for wire rope applications. What qualification testing has been accomplished has been on ropes of up to 1-inch diameter; little or no testing has been done on rope sizes larger than this. Hoist system manufacturers, being aware of this lack of qualification, get around the discrepancy by using reeved cable systems when designing for loads which exceed the capacity of 1-inch diameter rope.

Effects of Bending

All wire ropes, except stationary ropes used as guys or supports, are subjected to bending around sheaves or drums. The service obtained from a wire rope is, to a large extent, dependent upon the proper choice and location of the sheaves and drums about which it operates (see Figure 1). A wire rope may be considered as a machine in which the individual elements (wires and strands) slide upon each other when the rope is bent.

Therefore, as a prerequisite to the satisfactory operation of wire rope over sheaves and drums, the rope must be properly lubricated. With this in mind, the effects of bending may be classified as:

1. Loss of strength due to bending.

2. Fatigue effect of bending.

Loss of strength due to bending is caused by the inability of the individual strands and wires to adjust themselves to their changed position when the rope is bent. Tests made by the Bureau of Standards and reported in Technologic Paper No. 229 show that the rope strength decreases in a marked degree as the sheave diameter grows smaller with respect to the diameter of the rope. The loss of strength resulting from the bending of wire ropes over the sheaves found in common use will not

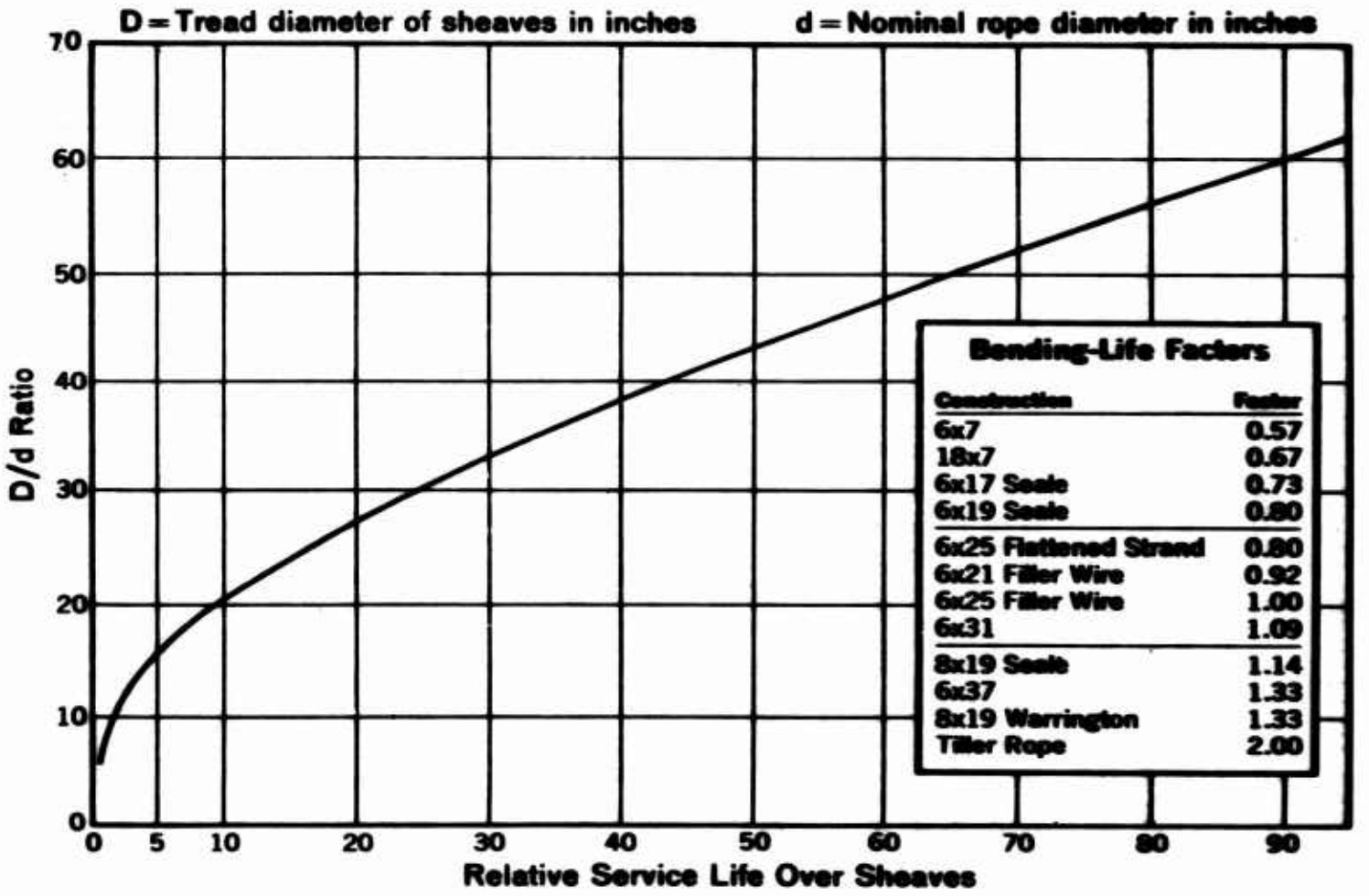


Figure 1. Relative Service Curve for Different Sheave/ Rope Ratios.

exceed 6 percent and will usually be about 4 percent.

The bending of a wire rope is accompanied by readjustments in the positions of the strands and wires and results in actual bending of the wires. Repetitive flexing of the wires sets up points of stress concentration, even though the bending loads may be well within the elastic limit of the wires.

The fatigue effect of bending appears in the form of small cracks in the wires at these foci of overstress. These cracks propagate under repeated stress cycles until the remaining sound metal is inadequate to withstand the bending load. This results in broken wires, showing no apparent contraction of cross section.

Experience has established the fact that, from the service viewpoint, a very definite relationship exists between the size of the individual outer wires of a wire rope and the size of the sheave or drum about which it operates. Sheaves and drums smaller than 200 times the diameter of the outer wires will cause permanent set in a heavily loaded rope.

Reverse Bends

Bending a rope first in one direction and then in the opposite direction causes excessive fatigue and should be avoided whenever possible. When a reverse bend is necessary, larger sheaves are required than would be the case if the rope were bent in one direction only. A comparison of minimum tread diameters of sheaves and drums is given in Table II.

In recent years there has been a trend toward the use of 6 x 37 class wire rope for overhead crane service, and away from the use of 6 x 19 class. The general specifications listed below have been established. When followed, they have been very successful in eliminating a variety of operating problems.

Specification for Crane Service

Wire Rope Diameter (In.)	Specification
1/4 to 5/16	6 x 19 class
3/8 and larger	6 x 37 class

TABLE II

MINIMUM TREAD DIAMETERS OF SHEAVES AND DRUMS IN INCHES

Mild Plow Steel, Plow Steel, and Monitor Steel Wire Ropes

Rope Dia.	6x7	6x19 Seale Pat. 6x17 Type L 18x7 Non-Spin.	6x19 Warrington 6x25 Type B 6x30 Type G 6x21 Type M	6x19 Type N 8x19 Seale Pat.	6x29 Type P	6x23 Type R 8x19	6x37 Type S 5x19 Marlin Clad	6x6x7 Tiller
1/4	10 1/2	8 1/2	7 1/2	6 1/2	.....			3 1/2
5/16	13 1/4	10 3/4	9 1/2	8 1/4	.....			4 1/2
3/8	15 3/4	12 3/4	11 1/4	9 3/4	8 3/4	8	6 3/4	5 1/4
7/16	18 1/2	15	13 1/4	11 1/2	10	9 1/4	8	6 1/4
1/2	21	17	15	13	11 1/2	10 1/2	9	7
9/16	23 1/2	19 1/4	17	14 3/4	13	11 3/4	10 1/4	8
5/8	26 1/4	21 1/4	18 3/4	16 1/4	14 1/2	13 1/4	11 1/4	8 3/4
3/4	31 1/2	25 1/2	22 1/2	19 1/2	17 1/4	15 3/4	13 1/2	10 1/2
7/8	36 3/4	29 3/4	26 1/4	22 3/4	20 1/4	18 1/2	15 3/4	12 1/4
1	42	34	30	26	23	21	18	14
1 1/8	47 1/4	38 1/4	33 3/4	29 1/4	26	23 3/4	20 1/4	.....
1 1/4	52 1/2	42 1/2	37 1/2	32 1/2	28 3/4	26 1/4	22 1/2	.....
1 3/8	57 3/4	46 3/4	41 1/4	35 3/4	31 3/4	29	24 3/4	.....
1 1/2	63	51	45	39	34 1/2	31 1/2	27	.....
1 5/8	.....	55 1/4	48 3/4	42 1/4	37 1/2	34 1/4	29 1/4	.....
1 3/4	.....	59 1/2	52 1/2	45 1/2	40 1/4	36 3/4	31 1/2	.....

These Minimum Tread Diameters are based on factors of 400 times the diameters of outer wires for all except the 18x7 Non-Spinning Rope, for which a factor of 500 is used.

Fleet Angle

Where a wire rope leads over a sheave and onto a drum, the rope will not remain in alignment with the sheave but will deviate to either side, depending on the width of the drum and its distance from the first fixed sheave.

The angle between the centerline through the sheave and the centerline of the rope leading to the drum is called the fleet angle (see Figure 2).

To avoid excessive wear on the sheave and to prevent excessive chafing of the oncoming rope against previous wraps on the drum, it is desirable to keep the fleet angle as small as possible.

Where space limitations are unrestricted, such as on mine hoists, the fleet angle is sometimes as small as half a degree.

This is equivalent to a distance of 115 feet between the drum and the first fixed sheave for each foot of drum width. It represents the minimum below which the rope will not properly wind back from the drum flange after completing one layer.

Most installations do not permit so great a distance between the drum and the sheave; for average conditions it is considered good practice to keep the fleet angle within 2 degrees for a grooved drum and 1-1/2 degrees for a drum with a smooth face.

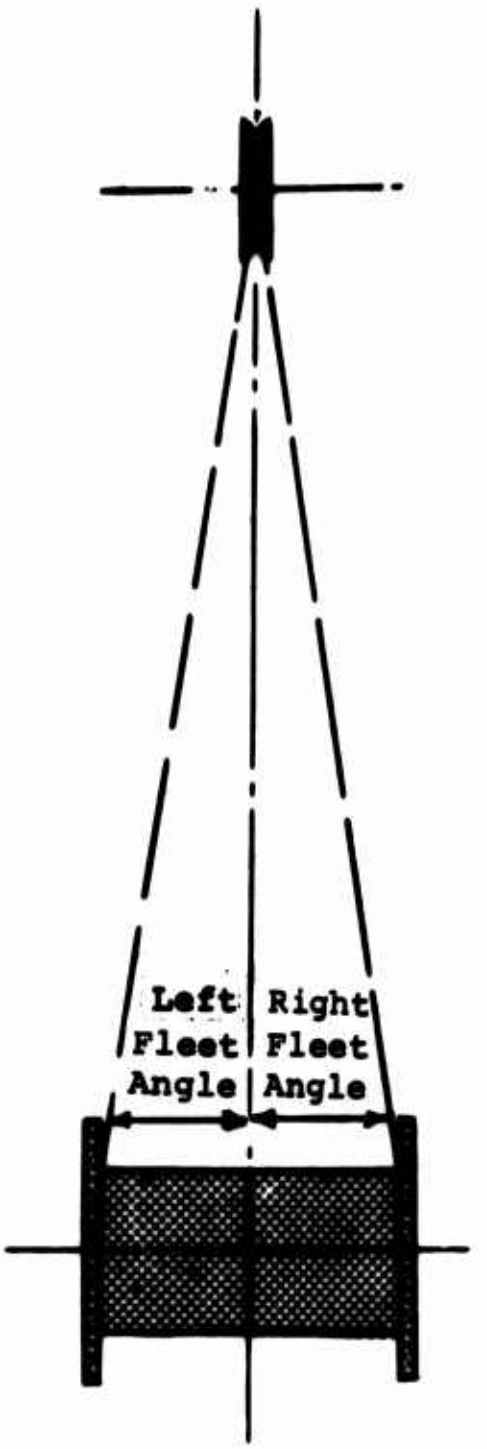


Figure 2. Rope Fleet Angle.

Rope Selection

The selection of the rope for the airborne hoist system application is a fundamental decision which will affect the overall characteristics of the hoist system. Factors in the selection of the rope are: (1) rope life expectancy, (2) rope construction, and (3) type of rope loading. These factors are not listed in order of their relative importance. It must be understood that the primary objective in the design of the system is to produce the ultimate in lightweight equipment consistent with high reliability, low maintenance, maximum maintainability and reasonable cost. In addition, the hoisting equipment must be integrated into the aircraft to provide an overall aircraft/hoist system which will be the ultimate in machine efficiency and will thus establish the optimum effectiveness of the aircraft. Efforts aimed at using so-called "standardized" or "all-purpose" equipment should be discouraged because this approach will detract from the overall effectiveness of the aircraft/hoist complex. Less experienced investigators may be attracted by the fallacious promises of economic advantages to be derived from an "all-purpose" helicopter hoisting system. The cargo hoisting system must be fully coordinated to provide the ultimate performance for cargo operations. To achieve the ultimate performance in the aircraft cargo system (helicopter plus cargo-hoisting system), the problems of maintenance, reliability, weight, power, and flight characteristics of the aircraft must be considered in the design of the cargo hoisting system.

Rope Life Expectancy

The determination of life expectancy of a rope is based upon many complex factors, such as rope construction, loading, lubrication, rigging, operating fleet angle, compatibility of rope and hoist component materials operating in interface, rope stresses due to dynamic loading, and thermal stresses developed between hoist components and the rope. These factors, which have a profound effect upon the life of the rope, will be discussed in design recommendations set forth in this report.

Projections of rope life predicated upon fatigue and wear criteria experienced in certain commercial cable applications can lead to the establishment of unrealistic values in deter-

mining the design of vital components in the hoisting system. Experience based upon the examination of many ropes replaced in aircraft hoists reveals that fatigue failure is very rarely the reason for rope replacement. The primary cause for rope replacement is damage due to mishandling, kinking, and abrasion, as evidenced by broken wires in the cable strands.

It is recommended that the life goal for the rope be coordinated with the maintenance functions of the aircraft. If we were to assume a normal periodic maintenance for the aircraft after X hours of operation and if the hoist were to be operated for approximately 2 cycles per aircraft flight hour, then the normal life goal for the rope would be computed at 2X operational cycles. This rope life expectancy is predicted on normal or working-load cycles. It is understood that the rope life would be proved at full-working-load cycles as part of the qualification testing program.

In actual field operations, the probability of encountering a full schedule of full working loads is not likely; this would mean that the rope would normally be replaced periodically with ample safety margin. In addition, it is customary to tensile-test the ropes after they have been subjected to life testing to prove that residual rope strengths exceed the normal working loads by a factor of two, as a minimum standard. The establishment of the rope life goal will permit the designer to proceed.

Rope Construction

The type of rope construction selected has an important influence on hoist design. Cable strand formation is a deciding factor in rope performance. If we were to consider either resistance to wear or flexibility as the determining characteristic in rope performance, wear and abrasion resistance could be obtained by using a small number of large wires in the construction or, conversely, flexibility could be ensured by using a large number of small wires. Flexibility of a rope is not synonymous with fatigue resistance. In many cases, a well-designed 19-wire rope would have longer life when subjected to repeated reversal cycles than a rope with many more wires. It is evident that secondary bending or flexing of individual wires within the cable strands is more damaging than the primary bending due to flexing of the rope around the drums or sheaves. The secondary bending is induced in the wires by

using a cross-wire construction in the rope. Modern state-of-the-art techniques would utilize an equal-laid type of construction, i.e., a construction in which the wires lie in the interstices of the under layment of wires and do not cross over to form bending pressure points. Such a construction is represented by a 6 x 37 Seale Warrington equal-laid wire rope.

Rope Loading

The manner in which the rope is loaded is important to (1) the selection of the rope construction and (2) the design of hoisting equipment on which the rope is to be used. The requirement to lift unguided loads dictates the selection of the cable and also establishes the details of the cable hook.

It may be of interest to discuss the spin mechanics of hoist ropes in order to afford an understanding of the criteria governing ultimate rope selection for a particular hoist application. The principal feature of a nonspinning, non-rotating or, perhaps more correctly, a spin-resistant wire rope is a construction which comprises many strands of wire in the outer layers, with inner and outer strands of wire laid in opposite directions. Under load, the inner strands tend to rotate in one direction while the outer strands tend to rotate in the opposite direction. These opposing tendencies reduce the spin reaction. The spin-resistant types of construction generally employ crossed wires and are subject to secondary bending stresses which reduce the fatigue life of the rope. In addition to secondary bending stresses developed in the rope as a result of loading the cable, it should be noted that when the cable is wound on a multi-layered drum, crossed wires are susceptible to crushing, which further reduces the life of the rope.

In analyzing the mechanics involved in a loaded wire rope, it is important to understand the concern given the problem of reducing the spin tendency of the applied wire rope. If, in lifting an unguided load, a wire rope is permitted to spin, or if the characteristics of the wire rope or of the hook termination induce spinning under load, certain wire strands in the rope are lengthened as a result of the spin. The load imposed on the lengthened strands is transferred to the contacting wire strands, and the overall strength of the cable is reduced. If the rope is heavily loaded, this transfer of loads from one layer of strands to another layer can result in failure of the rope.

It must be noted that the above phenomenon applies both to spin-resistant types and to conventional-construction ropes when an unguided load is applied.

It is recommended that the equal-laid-construction wire rope be used in the hoisting system application under study, for, in addition to the improved fatigue resistance, it provides better resistance to crushing, lower loss in breaking strength for a given amount of wear on the outer wire, and higher elastic moduli (i.e., reduced stretch under load).

Effect of Wire Rope Selection on Hoist Design

The ultimate hoist design is influenced by the selection of the wire rope, since vital hoist components including the drum or drums must be designed around the wire rope. The minimum diameters for sheaves or wire rope drums is determined by three primary factors: (1) rope material, (2) diameter of rope, and (3) rope construction.

Rope material and diameter of the rope are generally specified. The choice of material may be between corrosion-resistant steel Type No. 302 and high-carbon steel, with negligible difference in weight or strength in the sizes under consideration. Rope diameters are dictated by loading; the hoist under consideration will have a 20,000/40,000-pound working capacity which will require rope diameter in the range of 3/4 inch to 1-3/8 inches, depending on the type of wire rope construction chosen. The minimum economical drum diameter is also largely determined by the choice of wire rope construction. Experience has indicated that the designed drum diameter may vary as much as 400 percent, as a function of wire rope construction.

Experience accrued in the design and production of a multitude of both drum and capstan hoists/winches for aircraft applications calls into question the validity of so-called design factors used to determine sheave and drum diameters relative to rope diameter and type of rope construction. These factors provide a rough method of determining the required size ratios, but current commercial experience with wire rope tends to favor the use of empirical data, conservatively applied. Generally, the ratio of rope diameter to minimum sheave diameter for the 6 x 37 Seale Warrington equal-laid wire rope construction is 18 to 1.

Factors determining required wire rope size in terms of design load are: working load (including weight of cable and hook), safety factor, and bending factor.

The safety factor is determined, from data supplied in FAA Federal Aviation Regulations Part 133 - Rotorcraft External-Load Operations, to be 3.75. The factor determined from aircraft structural practice would include an ultimate load factor of 1.5. The yield factor of safety shall be 1.15, and other safety factors and allowable loads and stresses shall be in accordance with MIL-A-8629 (AER), paragraph 3.21.

Bending factor is empirically determined; experience has indicated that bending will produce a deterioration in cable strength of up to 6 percent of the working load.

CARGO HOOKS FOR HEAVY-LIFT HELICOPTER APPLICATION

A review and design appraisal of existing cargo hooks indicates that present hardware has been developed to an acceptable degree of reliability. Hooks of 40,000-pound capacity do not exist today; however, their manufacture presents no problems since they can be designed and fabricated using existing design philosophy as a base. The principle of operation is illustrated in Figure 3. Essentially, the hooks are electromechanical devices permitting both electrical and manual release when actuated from remote stations on the aircraft (i.e., pilot's and/or loadmaster's positions). Provision for manual release by ground crew is also embodied.

Electromechanical hooks of two types are illustrated in Figure 4, one having an open throat and the other a closed throat. The closed throat has been superseded by the open type since it permits easier loading. The closed throat type may have to be reintroduced if in multi-point systems the slings have a tendency to ride on the hook keeper.

Mechanical hooks considered are (1) a remote control type which is essentially the same as the electromechanical type but with the electric release feature and (2) a plain hook with keeper which must be released by physically removing the hook from the load sling. In hoist systems, remote control is impractical, thus dictating the use of (2). The use of such hooks in airborne hoist systems does not satisfy safety of flight requirements.

In cases where electrical conductors for hook actuation are embodied in a single-point system in which the hook is required to swivel to prevent hoist cable windup, the use of a slipring-type connection between hook and cable becomes necessary. This type of system has been produced successfully, although it must be noted that slip rings have been a source of trouble in the past. The complementary requirements for electrical conductors in the hoist cable and for slip rings in the winch add further sources of possible trouble.

Hook Release on Single-Point Hoist System

The cargo hooks illustrated in Figures 3 through 5 are all equipped with electrical release (normal) and manual release (emergency). However, when the hook is suspended from a single-point hoist system, direct coupling to the emergency release

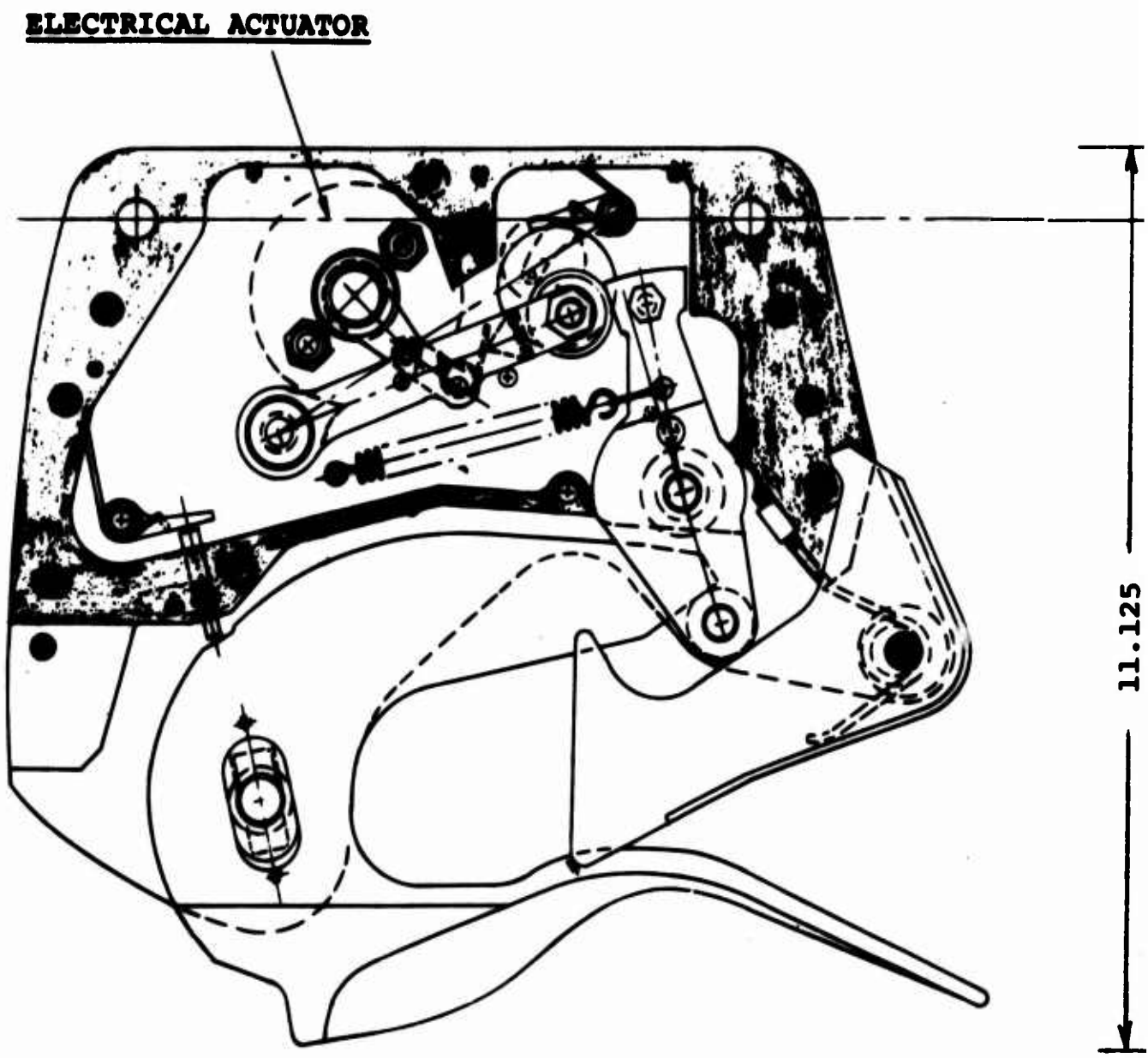
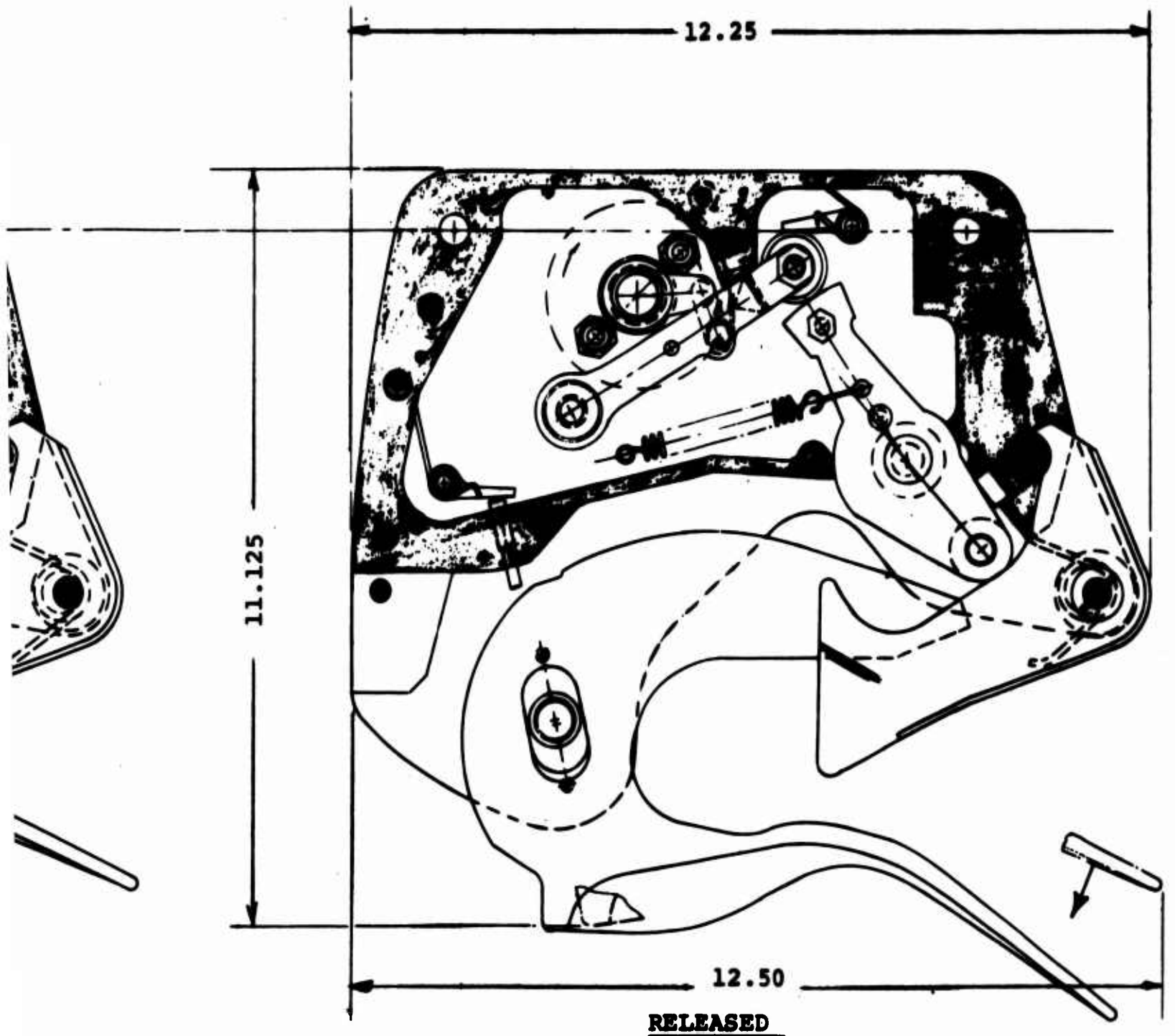


Figure 3. Cargo Hook Showing Principle of Operation.



RELEASED

: Operation.

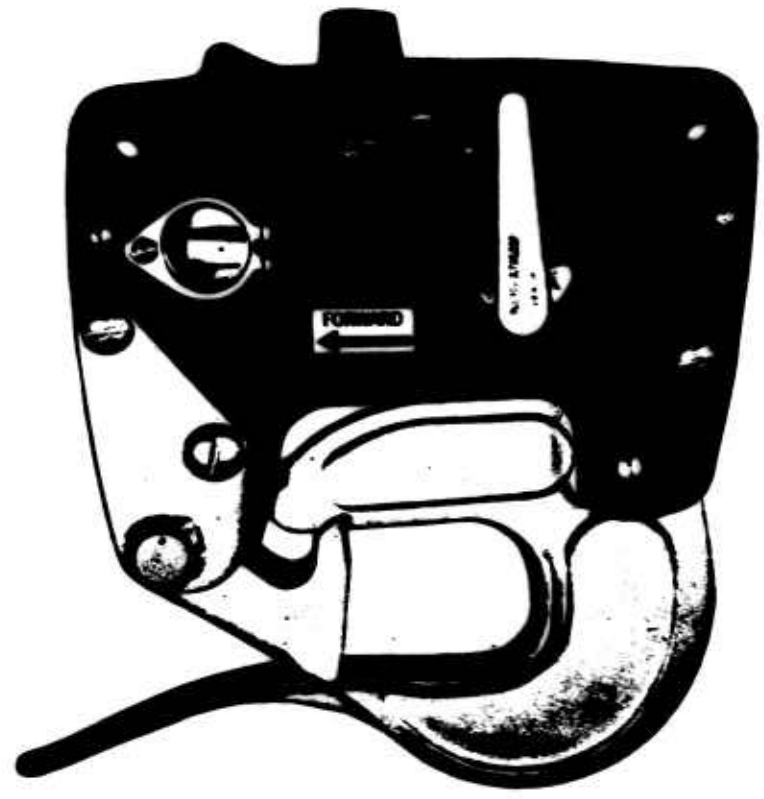


Figure 4. Typical Cargo Hooks.

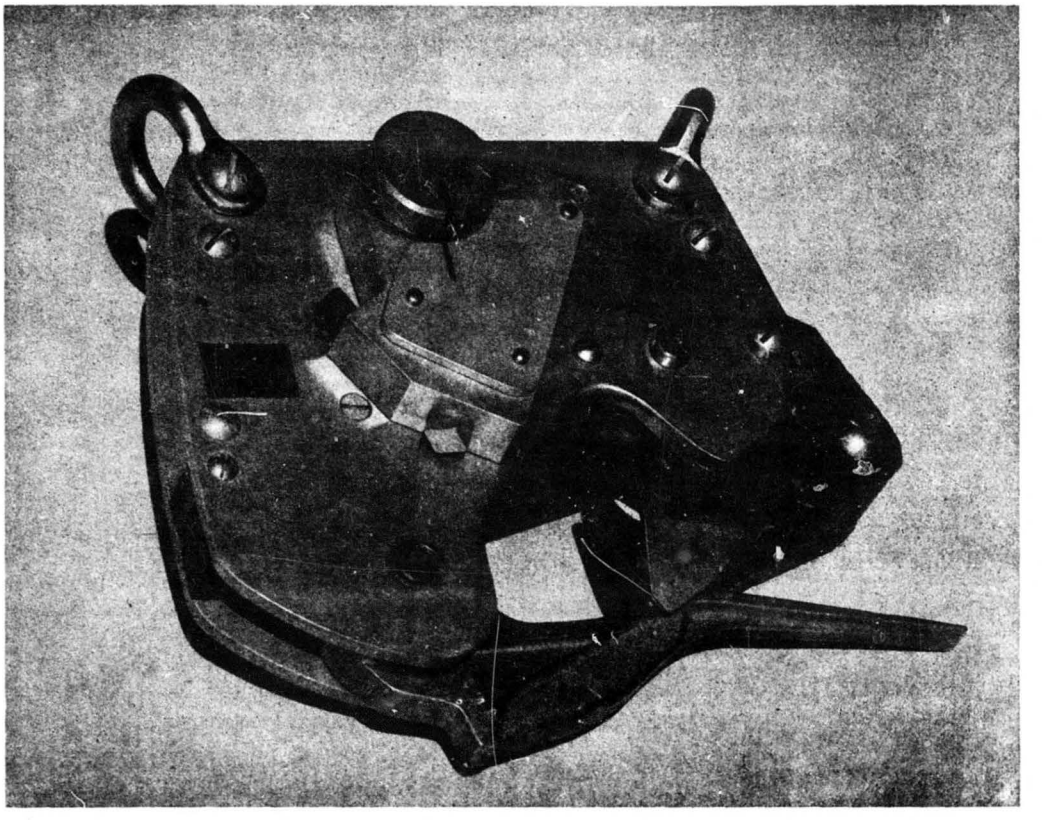


Figure 5. Hook with Cartridge-Type Emergency Release.

feature is impossible. One solution to this problem is to install a gas-operated device which by means of a piston imparts sufficient force to operate the hook manual release mechanism. The gas is generated by a cartridge which is electrically fired. This requires electrical conductors which must travel the same path as those installed for normal electrical release. Figure 5 illustrates a typical hook with this emergency feature incorporated.

Conclusions - Cargo Hooks (Refer to Table III)

The following are the conclusions drawn from the review of cargo hooks for the heavy-lift helicopter application:

1. The present state of the art in hook technology is considered adequate to meet contemplated HLH requirements.

2. Every endeavor should be made to develop systems which do not rely upon electrical circuits through the winch-hoist- rope hook system.

3. The problem of providing manual hook release on single-rope systems has not been effectively resolved. Although hook-mounted electrically initiated gas generators have been successfully developed to provide power for manual release, their actuation systems require electrical conductors from the aircraft to the hook.

TABLE III
RELATIVE MERITS OF CARGO HOOKS

Type	Wt.	Cost	Reliability	Flight Safety	Productivity
Mechanical Release (Remote)	2	3	2	6	6
Mechanical Release (Ground)	1	1	1	10	10
Electro-mechanical	3	4	4	1	1

For element definition, see the section entitled "Relative Merits of Hoist System Components and Complete Hoist Systems."

INFLUENCE OF AIRCRAFT CG TRAVEL ON HOIST SYSTEM DESIGN

All helicopters have a certain inherent allowable cg travel in both the lateral and longitudinal directions. This feature is good in that it permits some freedom in positioning the load within the limits imposed by the allowable aircraft cg range. The tandem-rotor helicopter, which can be represented by the concept of a beam supported at its ends, has a larger longitudinal cg travel range than the single-rotor helicopter, which can be represented by a beam supported at its center. Lateral cg travel is the same for a tandem- or a single-rotor helicopter, and is a specific requirement of MIL-H-8501, Helicopter Flying Qualities, Requirements For.

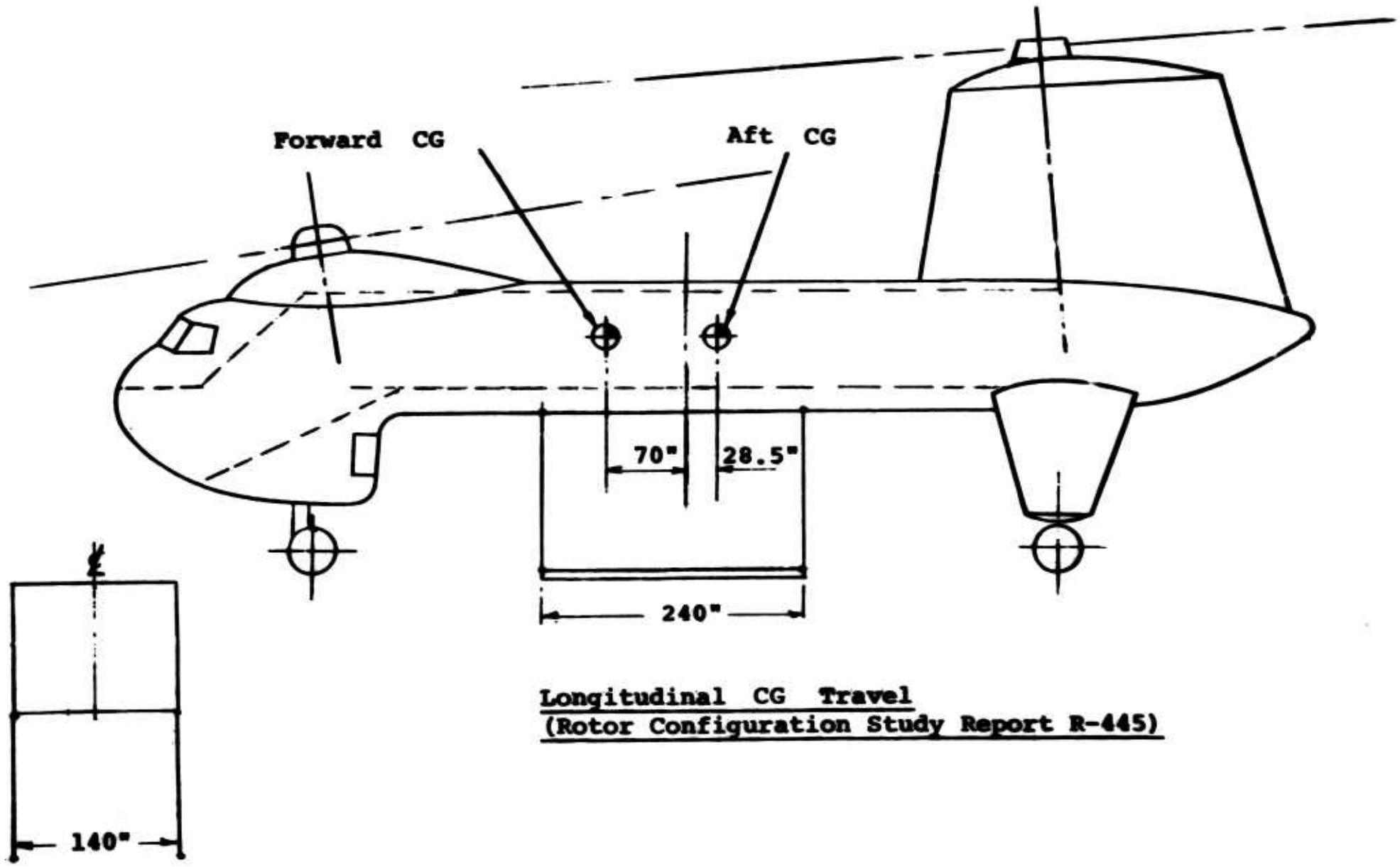
The cg travels used in this study are those illustrated in the HLH rotor configuration studies performed by Vertol Division, The Boeing Company, and Sikorsky Aircraft Division, United Aircraft Corporation, under contract with USAAVLABS. The longitudinal cg travel of a tandem-rotor-configured helicopter, being greater than that of a single-rotor aircraft, will be used in ascertaining the influence of cg on hoist design. Figure 6 illustrates the relationship between cg travel and proposed suspension systems. The longitudinal distance between suspension points has been assumed to be 20 feet (240 inches). The lateral distance between suspension points (four-point system) has been assumed to be 11 feet 8 inches (140 inches). This suspension geometry has been derived from the above-mentioned rotor configuration studies. It must be appreciated that any future change in these dimensions will have an effect on winch loading criteria.

Two-Point Hoist System

Longitudinal cg limits for the two-point hoist system are located on 98.5-inch centers; lateral cg limits do not apply.

The distance between suspension points is 240 inches, symmetrical about longitudinal cg limits; laterally the system is mounted on the longitudinal axis of the aircraft.

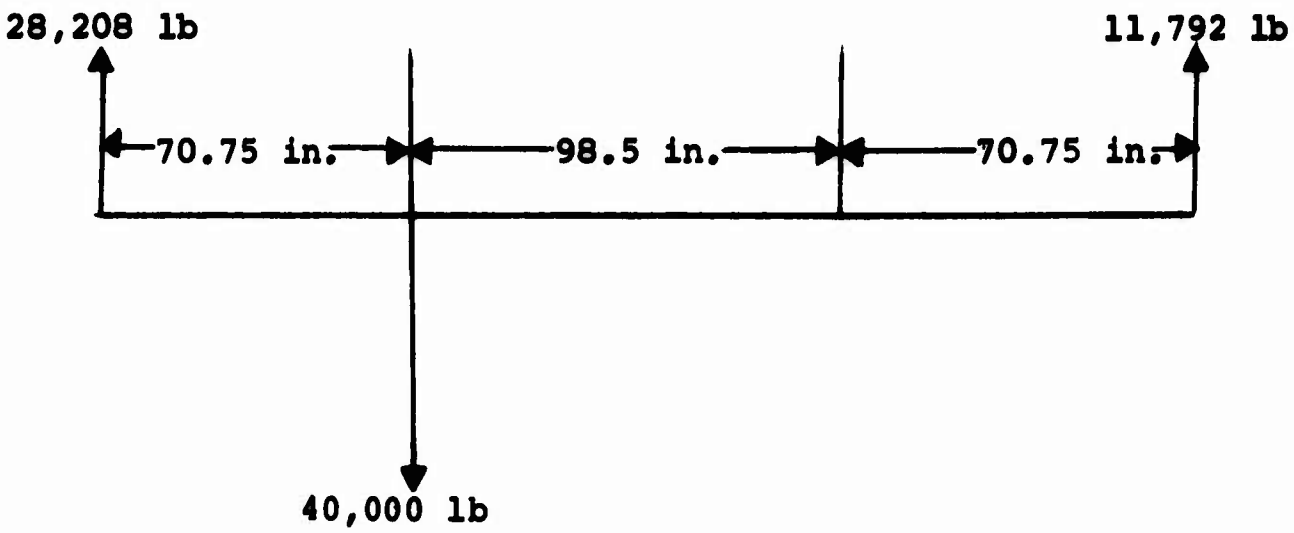
Since an external load of 40,000 pounds can be suspended from any point within the cg range of the aircraft, the individual loads in each suspension cable will be dependent upon load cg.



Longitudinal CG Travel
(Rotor Configuration Study Report R-445)

Figure 6. Assumed Relationship Between Aircraft CG and Multi-Point Suspension System.

Thus, with this load acting at the forward limit of aircraft cg, the magnitude of load carried by each suspension cable will be as shown in the diagram below.



Required winch and cable capacity is therefore 28,208 pounds, assuming a single suspension cable from each point.

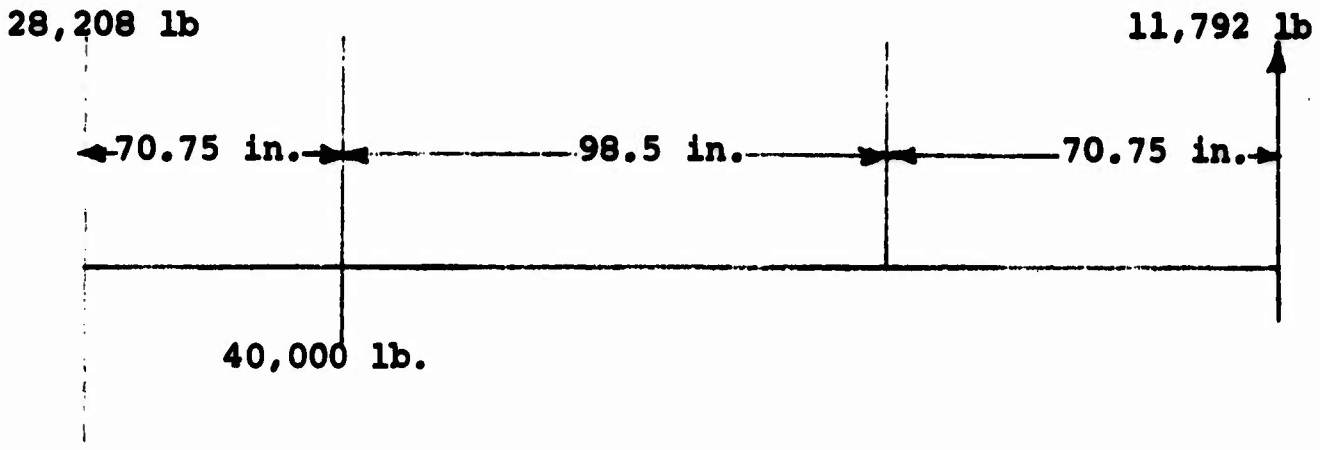
Four-Point Hoist System

Longitudinal cg limits for the four-point hoist system are located on 98.5-inch centers. The longitudinal distance between suspension points is 240 inches, symmetrical about longitudinal cg limits. The lateral distance between suspension points is 140 inches, symmetrical about the aircraft centerline. The lateral cg limits of load are assumed to be ±7 inches from the load centerline. This 7-inch dimension is based upon experience gained from random loading of CH-47A helicopters; i.e., the random loading results in a lateral cg shift of approximately 10 percent of fuselage width. However, it must be appreciated that the allowable aircraft lateral cg travel exceeds this and that it is permissible to load within this travel limit. A lateral cg limit of ±30 inches is considered realistic, although precise figures have not been determined for proposed heavy-lift helicopters.

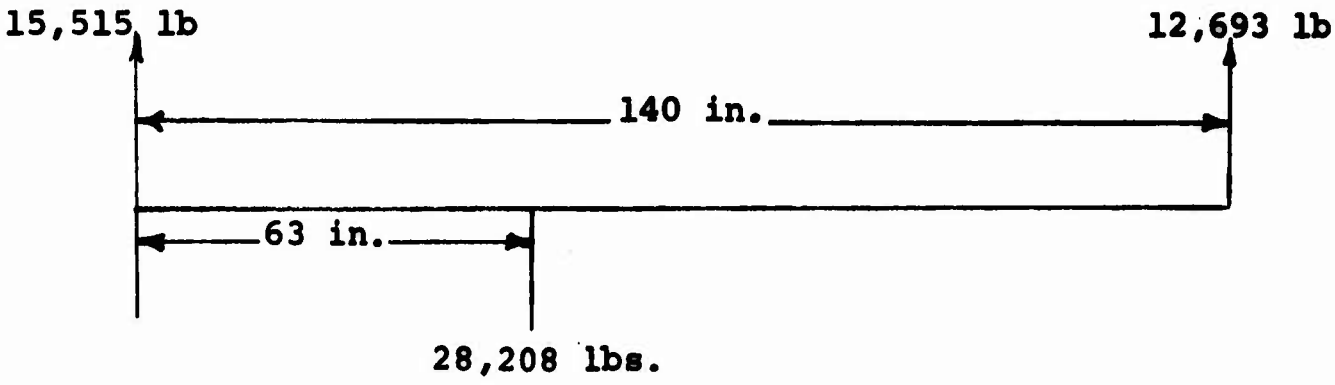
If this ±30-inch figure should be considered a limiting design value for the establishment of winch capacity for the four-point-hoist system, the required individual winch capacity would be approximately 20,000 pounds, resulting in a large

increase in system weight. We consider this to be unrealistic, since some control of lateral cg position of the load must be assumed and, as already stated, ±10 percent of width is considered acceptable. This latter cg travel figure would result in an individual winch capacity of 15,515 pounds, as shown in the diagram for the four-point system below.

CASE I - LONGITUDINAL



CASE II - LATERAL



The above analogy is shown only to indicate that the load cg will influence the winch capacity requirements. The results shown are comparative (i.e., two- as against four-point) and are not to be considered as being specific.

WINCHES FOR AIRBORNE HOIST SYSTEMS

The two types of winches considered acceptable for application to the proposed airborne hoist systems are the drum type and the capstan type.

Drum-Type Winches

The drum-type winch, which is the one most commonly used today, consists of a drum onto which the wire hoist rope is wound. Providing the recommendations for the use of wire rope are adhered to, a high degree of reliability can be achieved. (Refer to the earlier section of this report entitled "THE INFLUENCE OF WIRE ROPE CONSIDERATIONS ON HOIST SYSTEM DESIGN.") Figures 7 and 8 illustrate typical drum-type winches in use today. Since the wire rope must be progressively wound on the drum, it is essential to provide means of preventing the rope from winding upon itself; this rope overlap causes unacceptable crushing damage. To ensure that rope overlap does not occur, a conventional level-wind system is usually embodied; some of the characteristics of such a device are shown in Figure 9. Another method which achieves the same result is to employ a rope-follower level-wind system. This system consists of a collar assembly containing three rollers which engage a spiral rope groove formed on the drum. This assembly follows the unwinding rope in much the same way that a nut runs down the thread of a bolt. Figure 10 shows some of the characteristics of a rope-follower level-wind system. This method can only be used when not more than one layer of rope is wound on the drum, as will be the case on drum-type winches considered in this study.

Another design requirement of drum-type winches is that the rope be restrained from jumping on the drum when the cable springs back upon release of load. This requirement is usually met by feeding the rope through pinch rollers as it leaves the drum. When a rope-follower type of level wind is employed, the follower rollers can be designed so that they bear down on the rope and hold it in the drum grooves, thus eliminating the need for pinch rollers. When large-diameter rope is used, the rope also has a tendency to bunch up on the drum unless the pinch rollers are driven, thus providing power reel-out of the rope. A roller rope-follower, by retaining the rope on the drum at three equally spaced circumferential positions around the drum, shows promise of effectively achieving the same result without the necessity of using power reel-out.

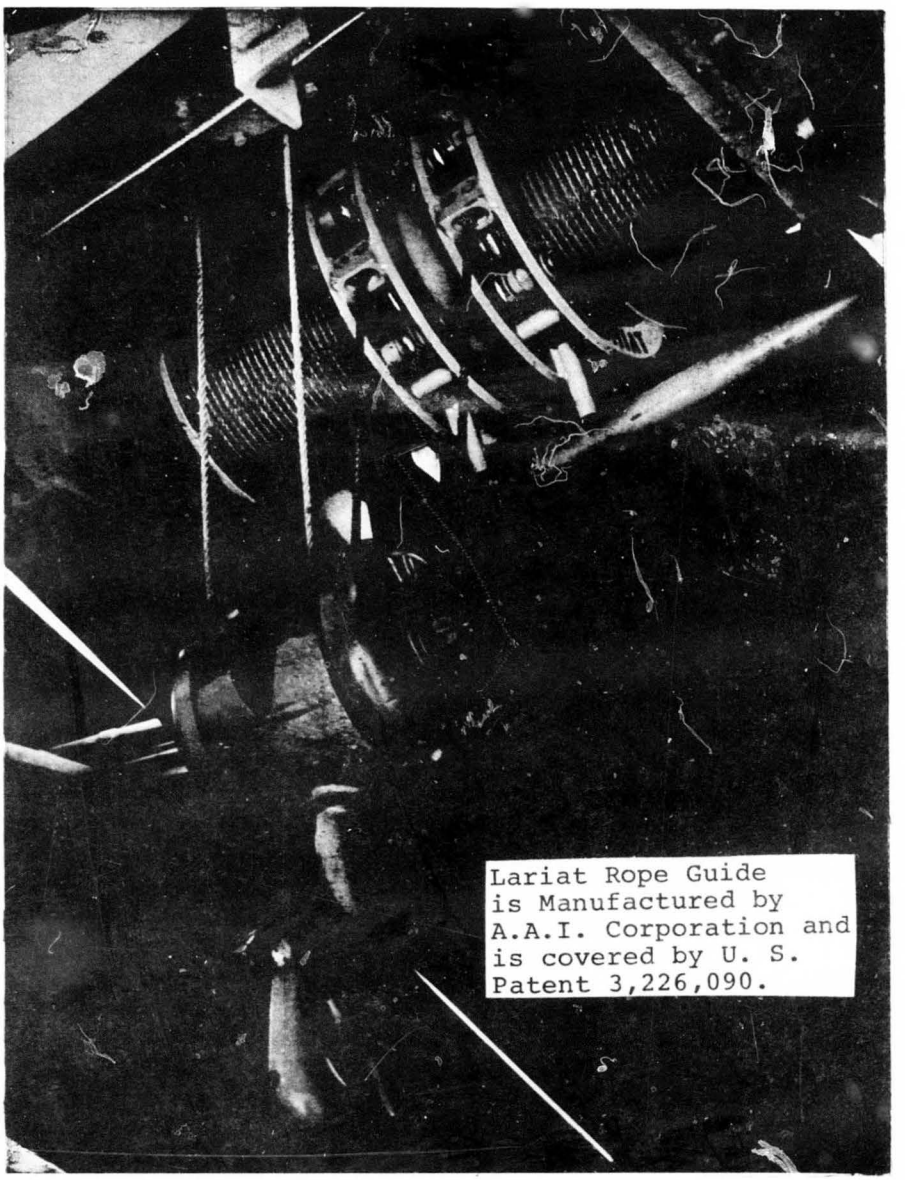


Figure 7. Twin-Drum Type Winch with Rope Follower.

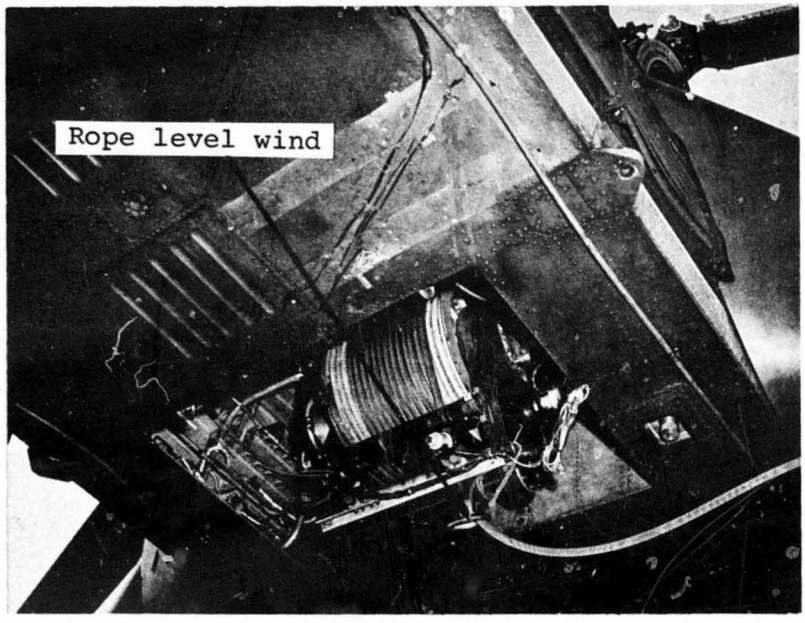
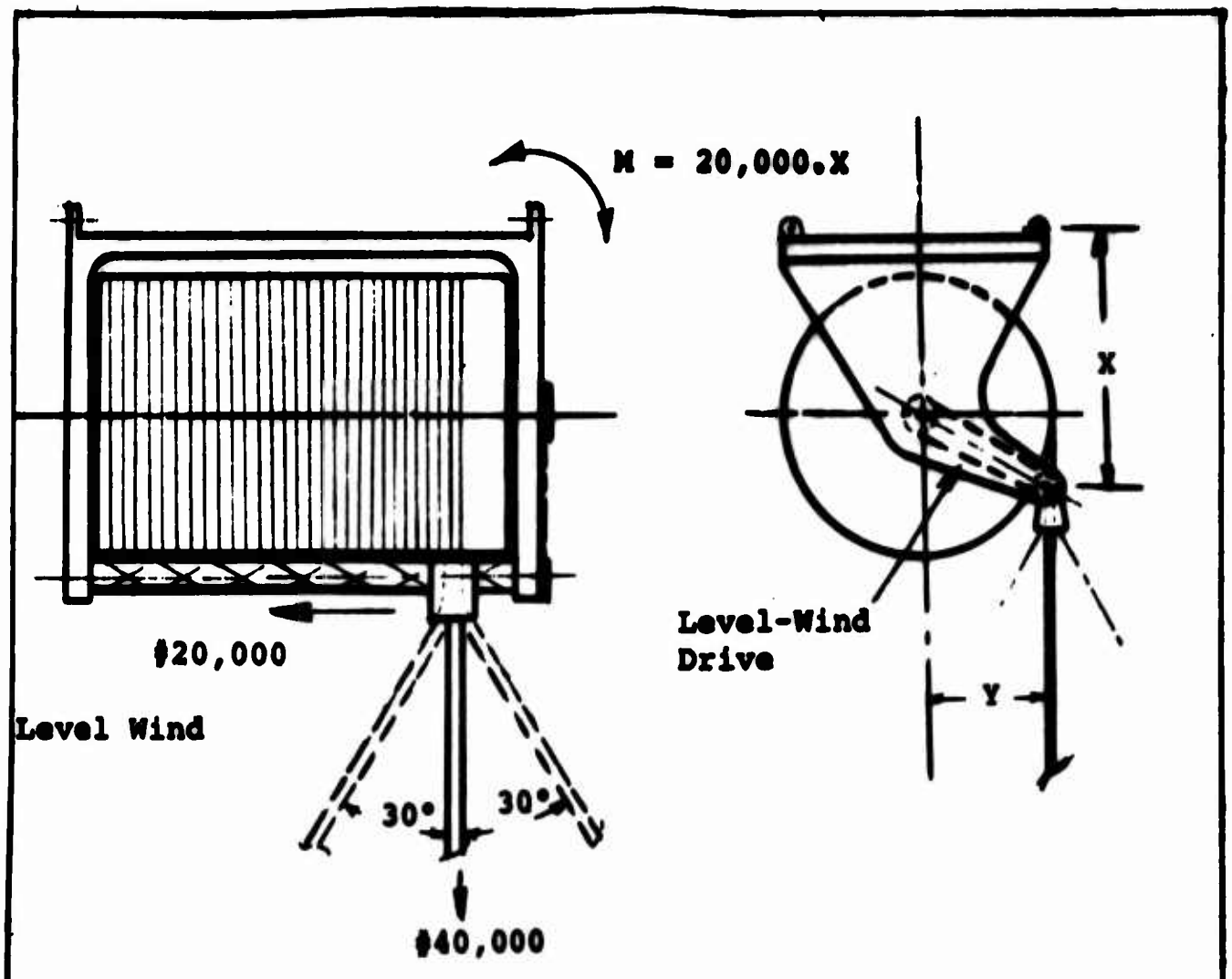


Figure 8. Drum-Type Winch with Level Wind.

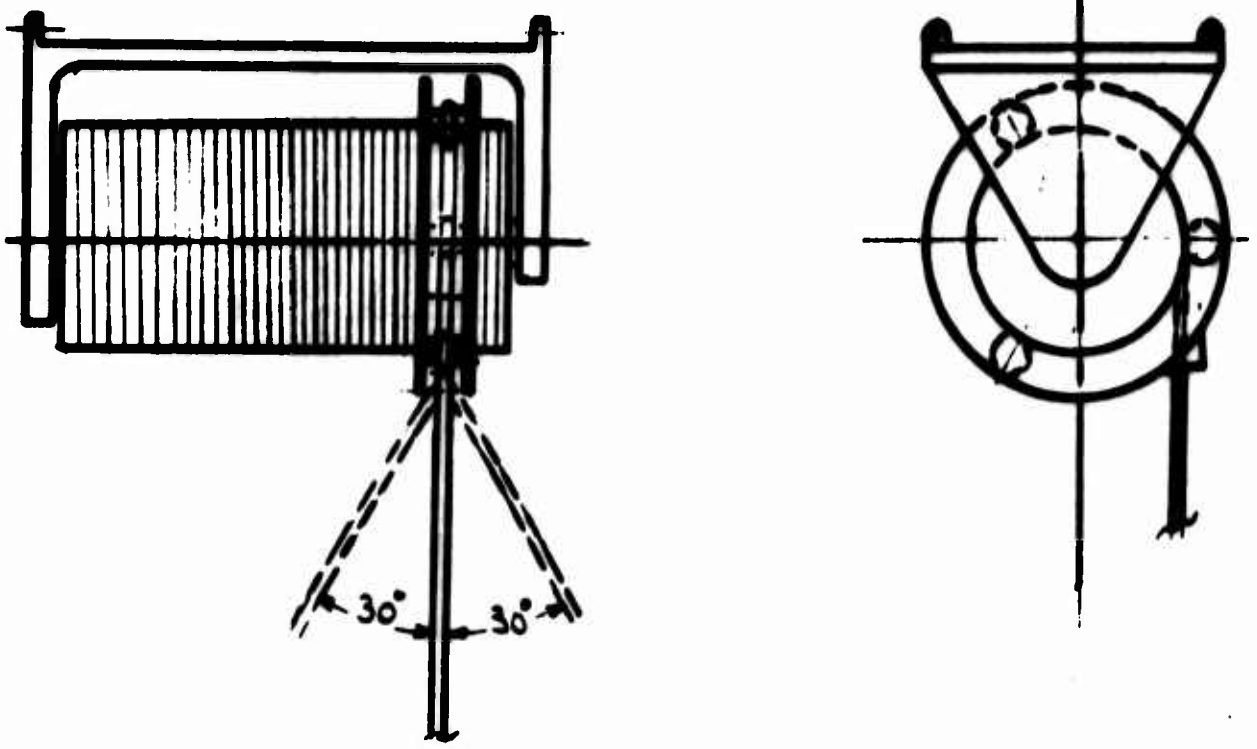


Conventional level wind (two-way thread) requires power drive.

End loads due to swinging load are reacted by the level wind, resulting in the generation of high bending moments. These moments must be taken out in bending through the winch structure. In a single-point system, the resulting moment arm X is proportionally large. The offset Y results in a torsional moment which must also be taken out by the winch structure.

This type of level wind does not retain rope on drum grooves during load release springback unless additional pinch rollers are added for this function.

Figure 9. Effect of Conventional Level Wind on Drum-Type Winch Structure.



Rope - follower-type level wind does not require power drive; it chases the rope groove in the drum.

A small increase in drum length is required to allow for follower overrun.

Loads due to rope swing are transferred directly to the cable drum, then to the winch structure, resulting in smaller moments.

Lateral rope swing causes the follower to run around the drum, thus reducing localized bending at the level wind bell-mouth.

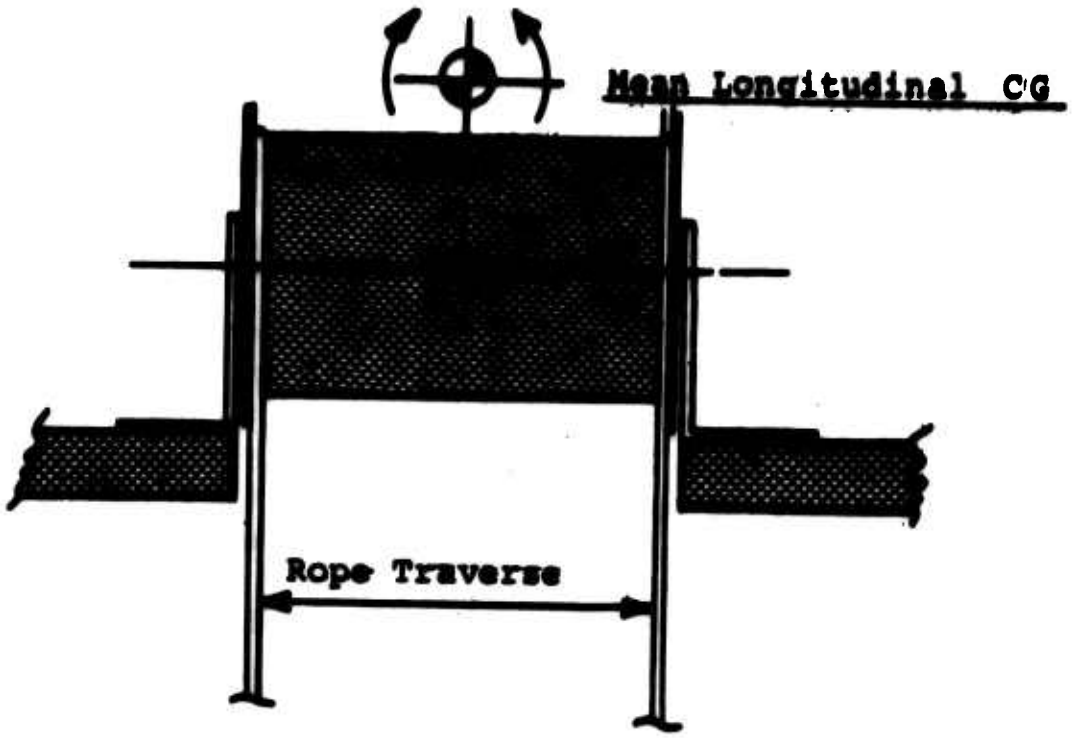
Follower rollers retain the rope in drum grooves during load release springback.

Figure 10. Rope-Follower-Type Level Wind on Drum-Type Winch.

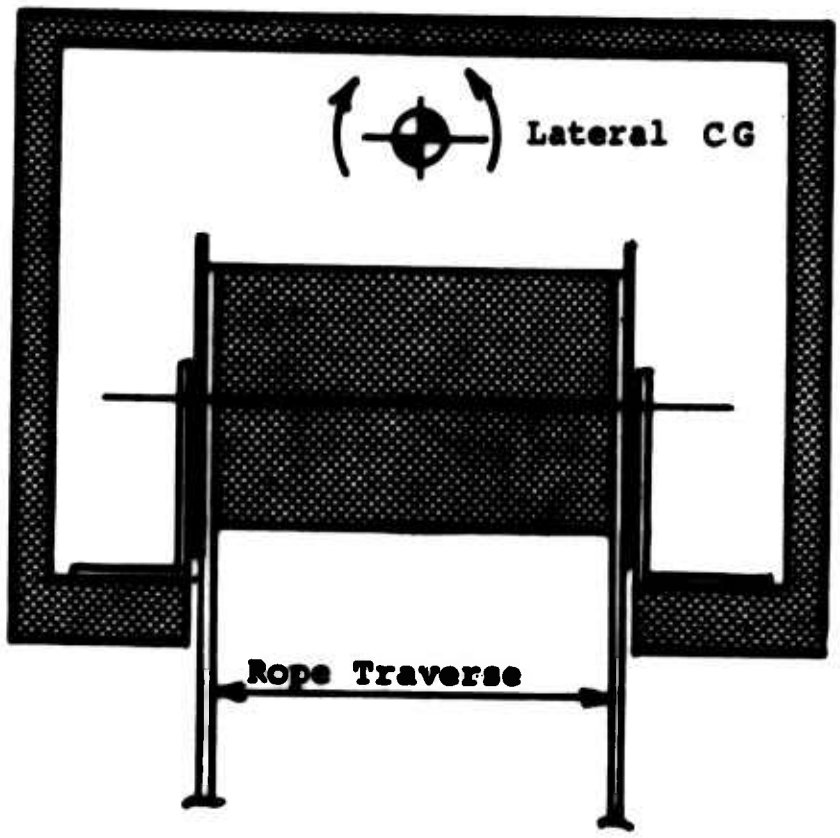
An undesirable feature of a drum-type winch is that as the load unwinds, it traverses the drum, with the result that the load can be applied at any position on the drum, dependent upon how much rope is deployed. In an aircraft installation, this rope traverse will create a variable moment about the aircraft cg which must be considered in arriving at the installation configuration (see Figure 11). When reeved hoist systems are used, the rope length is greater; i.e., in a single-reeved system,we have twice the rope length, compared with a single-rope lift. This is reflected in greater drum length, which in turn causes a misalignment of the two ropes as the rope unwinds (see Figure 12). To eliminate the long drum, the solution is to increase the drum diameter; this results in fewer turns of rope for a given length. The optimum relationship of drum length to drum diameter considering envelope and cable traverse has been established as 1:1.

When a load isolator, which must be installed in series between the load and the suspension point on the aircraft, is mounted as an integral part of a drum-type winch, the winch attachment points to the aircraft have to be designed in the form of a hinge,which allows the winch to sway and thus impart linear motion to the load isolator (see Figure 13). This means that the torque generated by the load on the winch drum is reacted through the isolator into the aircraft structure. Since this torque will be reacted at the power input end of the winch, consideration must be given to the case where, due to rope traverse, the load will be suspended from the opposite end of the drum. Imposition of side loads on the drum resulting from a swinging load will create a torsional moment at the end of the winch where no provision is made to react it. It is true that two load isolators can be used, but their effectiveness could be compromised. Another solution would be to design the winch structure to transfer the resulting torsion to the end designed to react torsion into the aircraft structure. In either case, the solution requires a greater winch weight than would result from a winch secured to the aircraft at four points,which would allow any moments generated by a swinging load to be reacted directly into the aircraft structure.

An alternative method of installing the load isolator directly into the winch system is shown in Figure 14. Although this method is somewhat simpler than the previously discussed method, the result is an open-type configuration. This means that there will be no protection for the aircraft structure



Effect of Longitudinally Mounted Drum Winch



Effect of Laterally Mounted Drum Winch

Figure 11. Effect of Rope Traverse on Drum-Type Winch Installation.

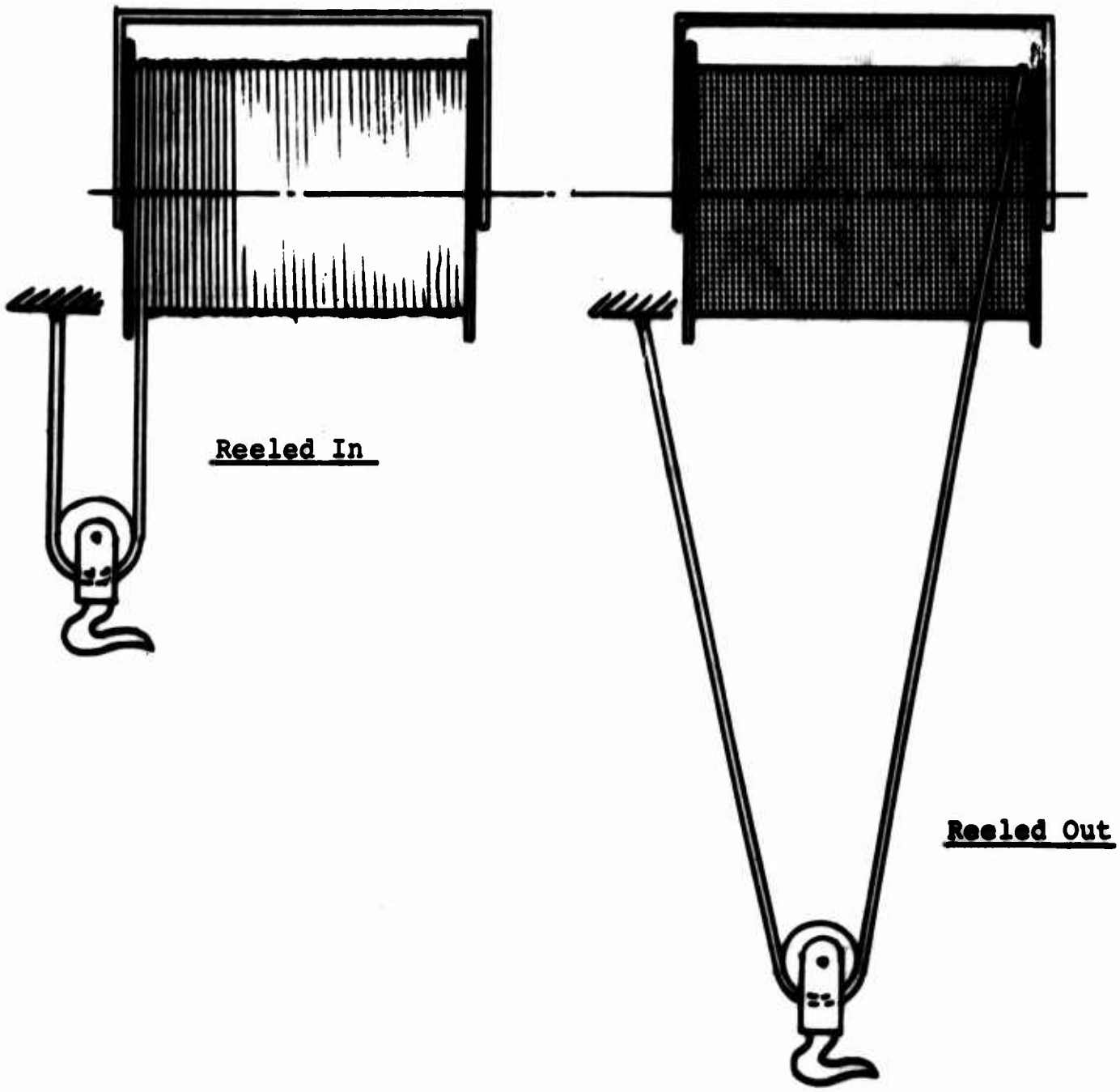


Figure 12. Effect of Drum Length on Reeved Rope System.

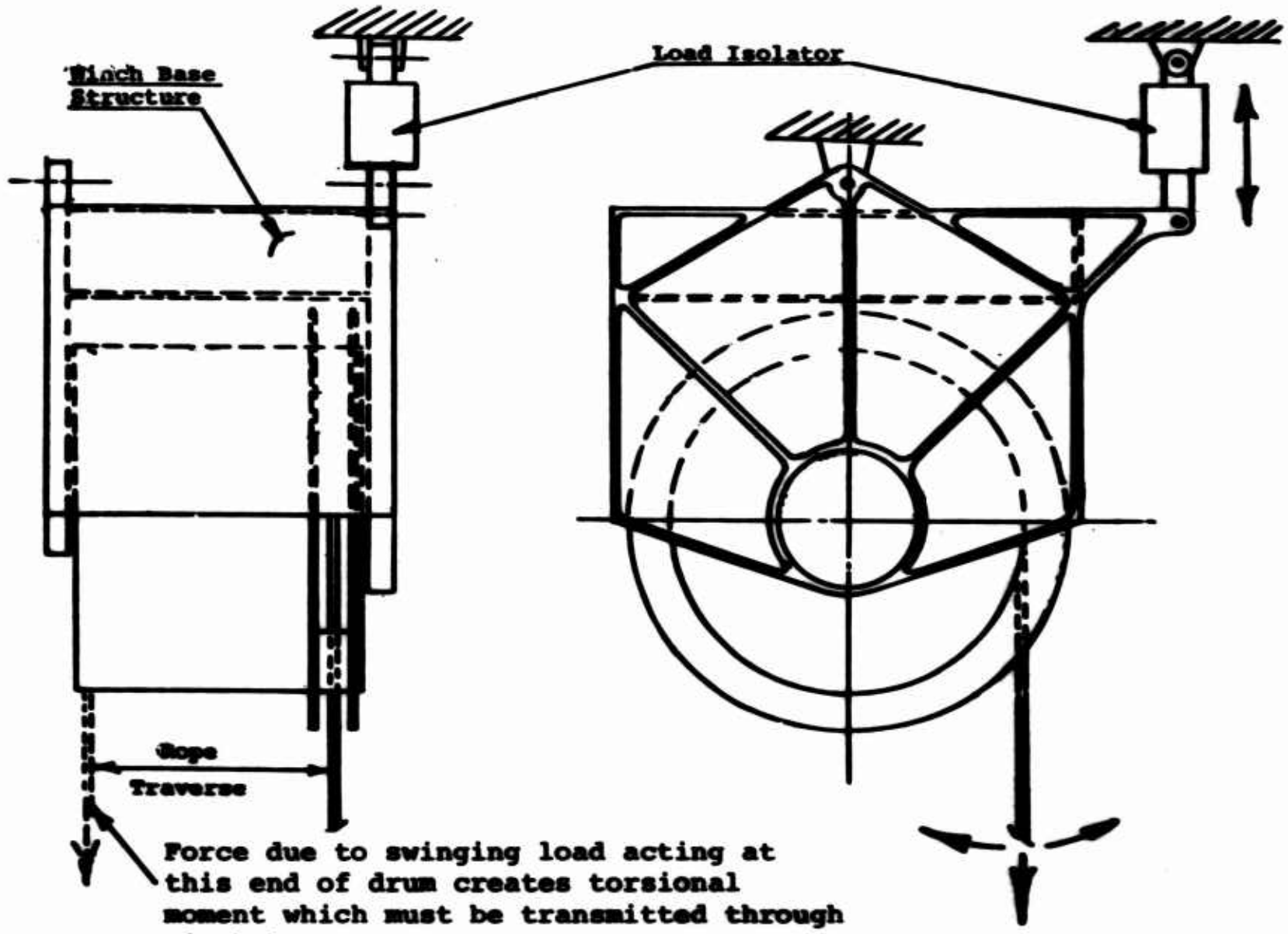


Figure 13. Suspension of Winch from Hinge Point in Order to Generate Linear Motion to Isolator.

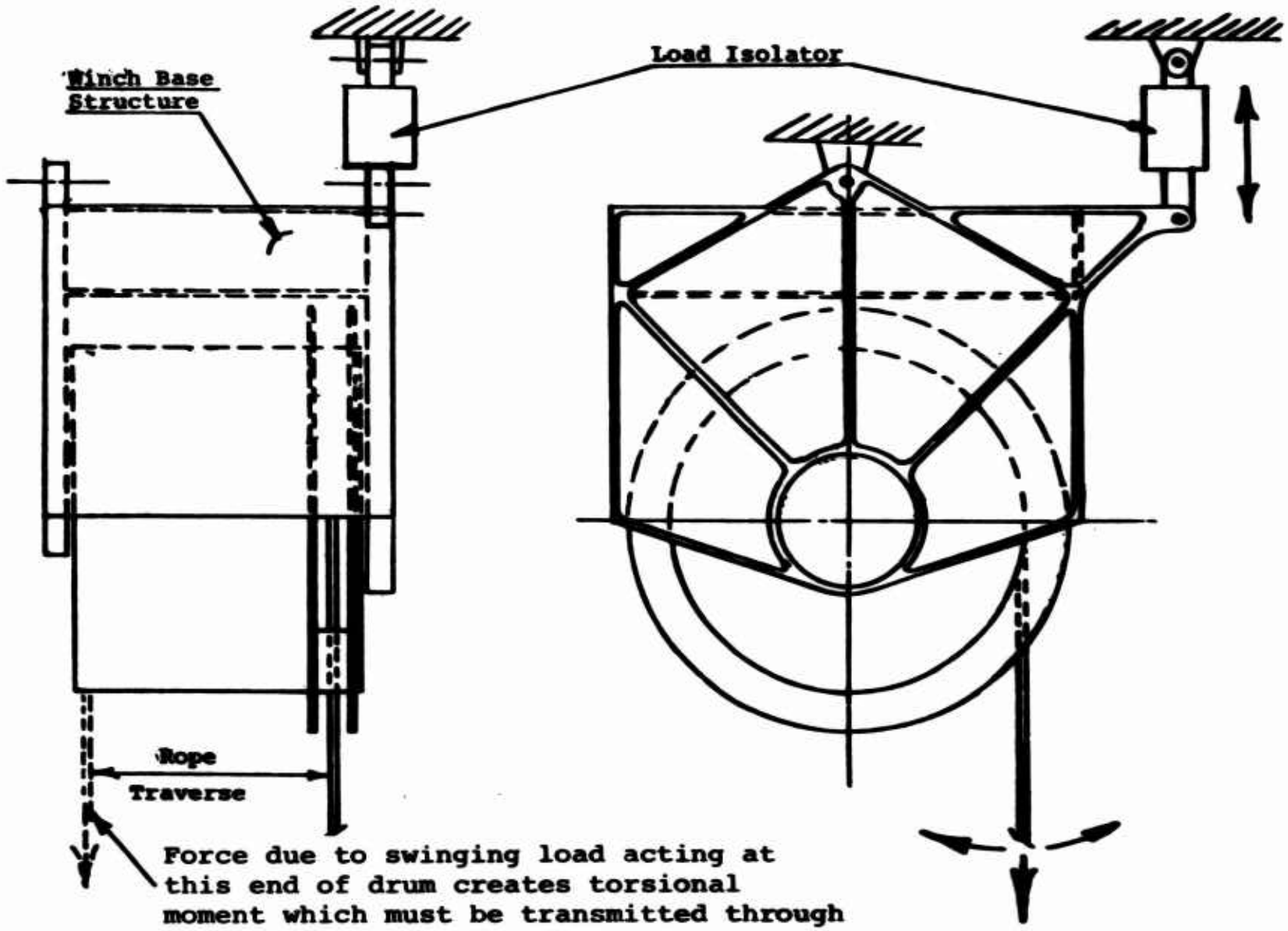
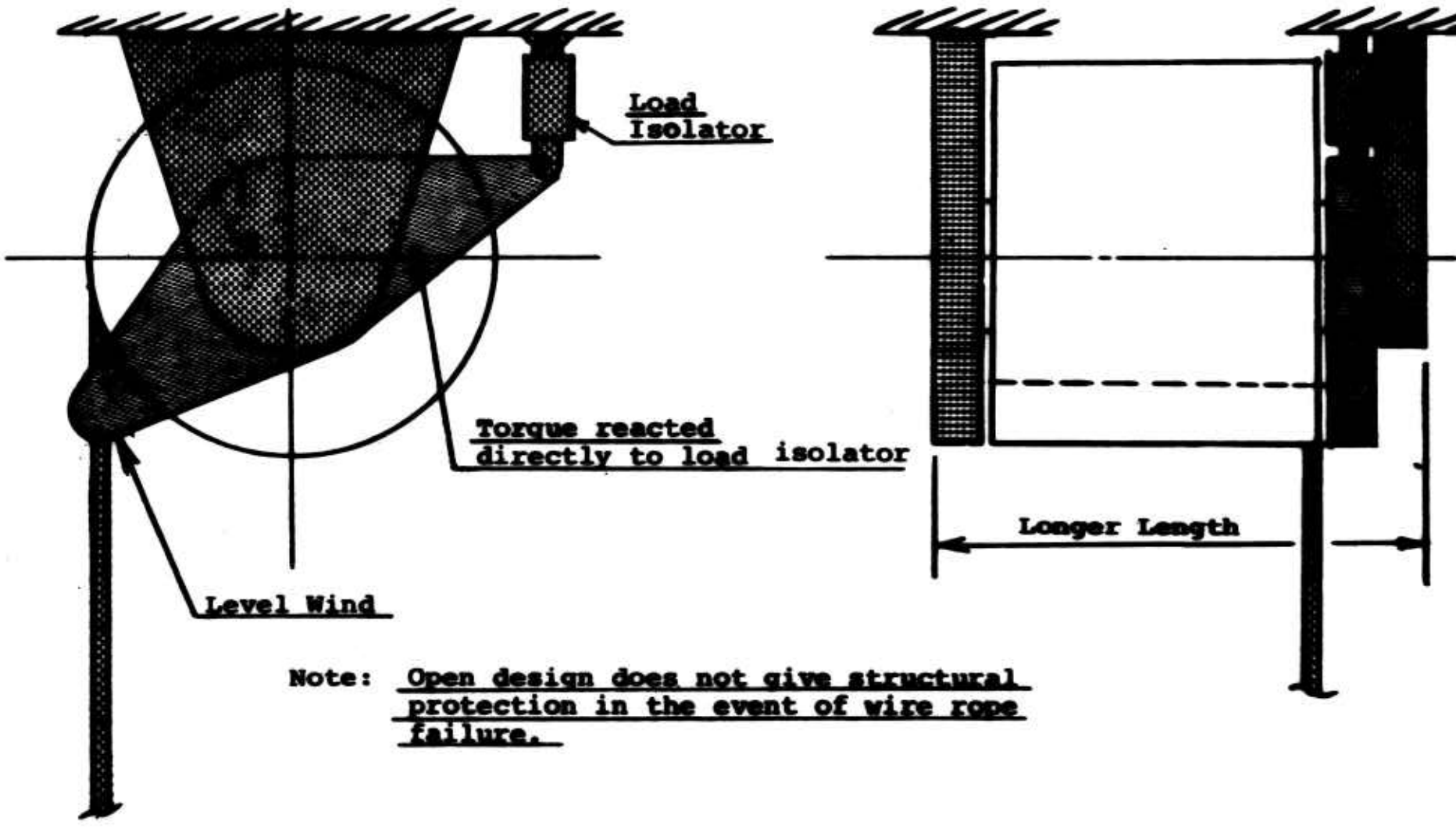


Figure 13. Suspension of Winch from Hinge Point in Order to Generate Linear Motion to Isolator.



Note: Open design does not give structural protection in the event of wire rope failure.

Figure 14. Alternative Drum-Type Winch Configuration.

in the event of wire rope breakage. To satisfy this requirement, the drum would have to be partially or totally enclosed in order to contain a wild rope; this could result in a weight penalty.

If a single-reeved system is employed, then the problem resolves itself as shown in Figure 15. It will be noted that the load isolator is mounted on the free end of the rope system and that the winch can be rigidly mounted to the structure.

The installation of load cells for monitoring rope load follows the same principles used for load isolators. In the ultimate system, it appears feasible to integrate the load cell into the load isolator unit when a reeved rope system is employed.

Figures 16 through 20 illustrate the configuration and relative sizes of drum-type winches which we consider meet the requirements of a 40,000-pound hoist system.

Optimum Envelope for Drum-Type Winches (Refer to Table IV)

It will be noted from Figures 17, 19, and 20 that when reeved rope systems are used in conjunction with a drum diameter-to-rope-diameter ratio of 18:1, the drum length becomes excessive. The effect of this excessive length is illustrated in Figures 11 and 12. Figures 21 through 26 illustrate the effects of correcting this situation by making drum diameter approximately equal to drum length. The use of this optimum envelope results in drum diameter-to-rope-diameter ratios well in excess of the minimum considered to be good design practice by the wire rope manufacturers. The net benefit would be better fatigue life for the rope. On the negative side, the resulting weight of such an optimum-envelope system will be greater, although subsequent structural requirements for winch installation may be lighter.

A further envelope consideration is that of protecting the aircraft structure; to guard against damage resulting from a wild-running broken wire rope, the drum must be partially enclosed. Such enclosure should be made part of the basic winch structure in order to hold down winch weight. Examples of such configurations are shown in Figures 21 through 26. Total drum enclosure is not practical, since provision must be made for load swinging and rope traverse. Investigation shows that a sheet metal cover with brush-sealed openings will

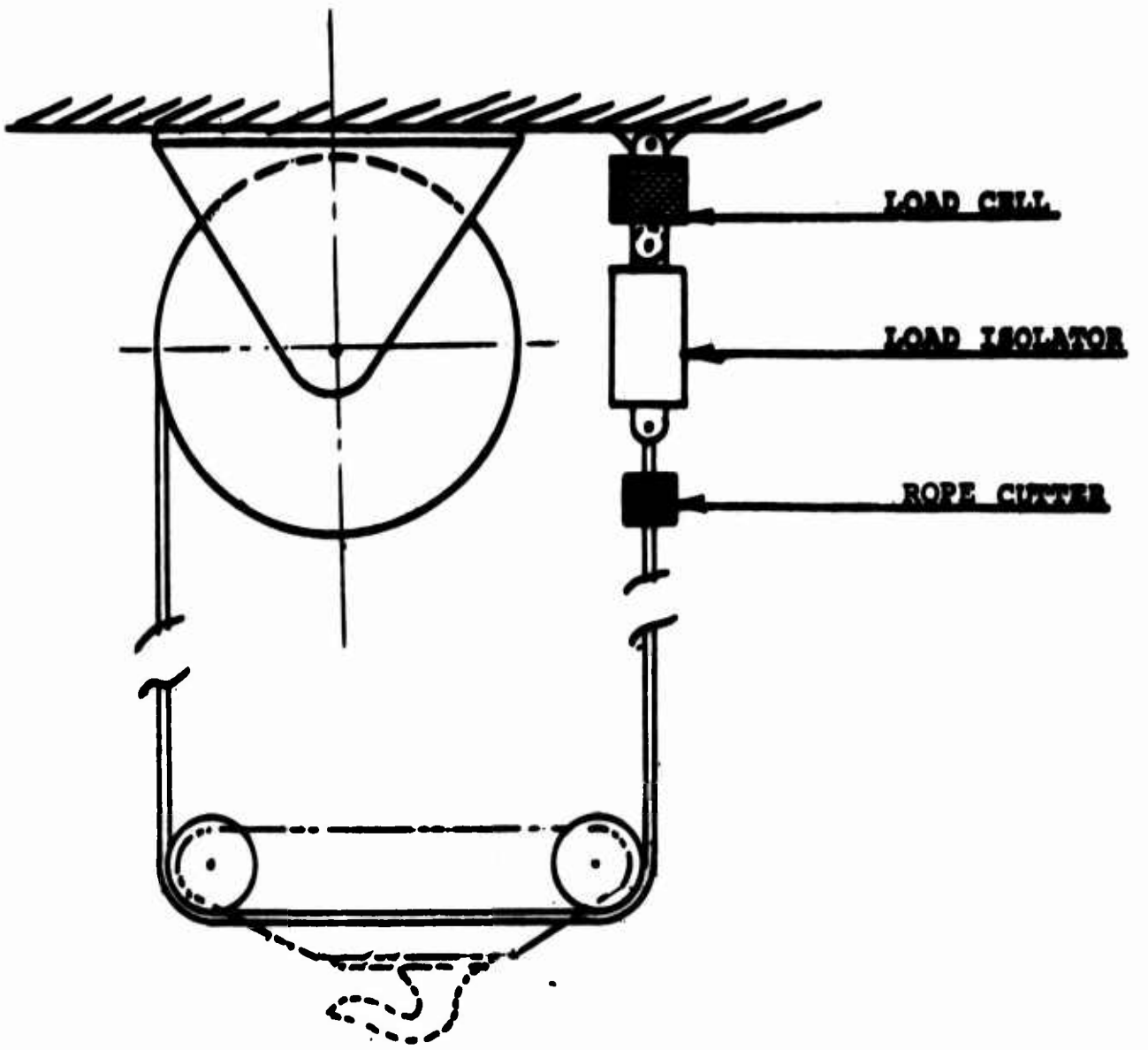
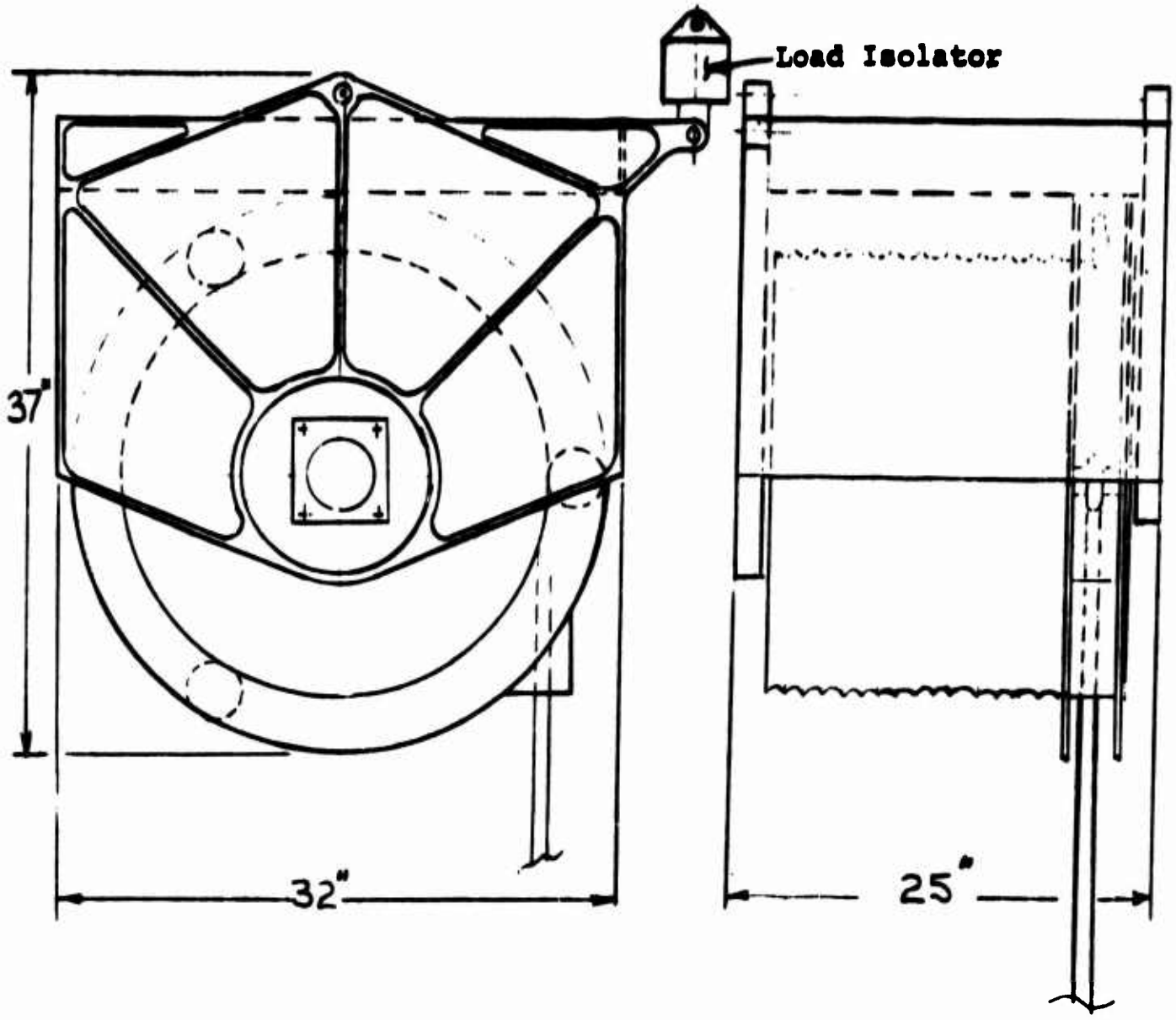


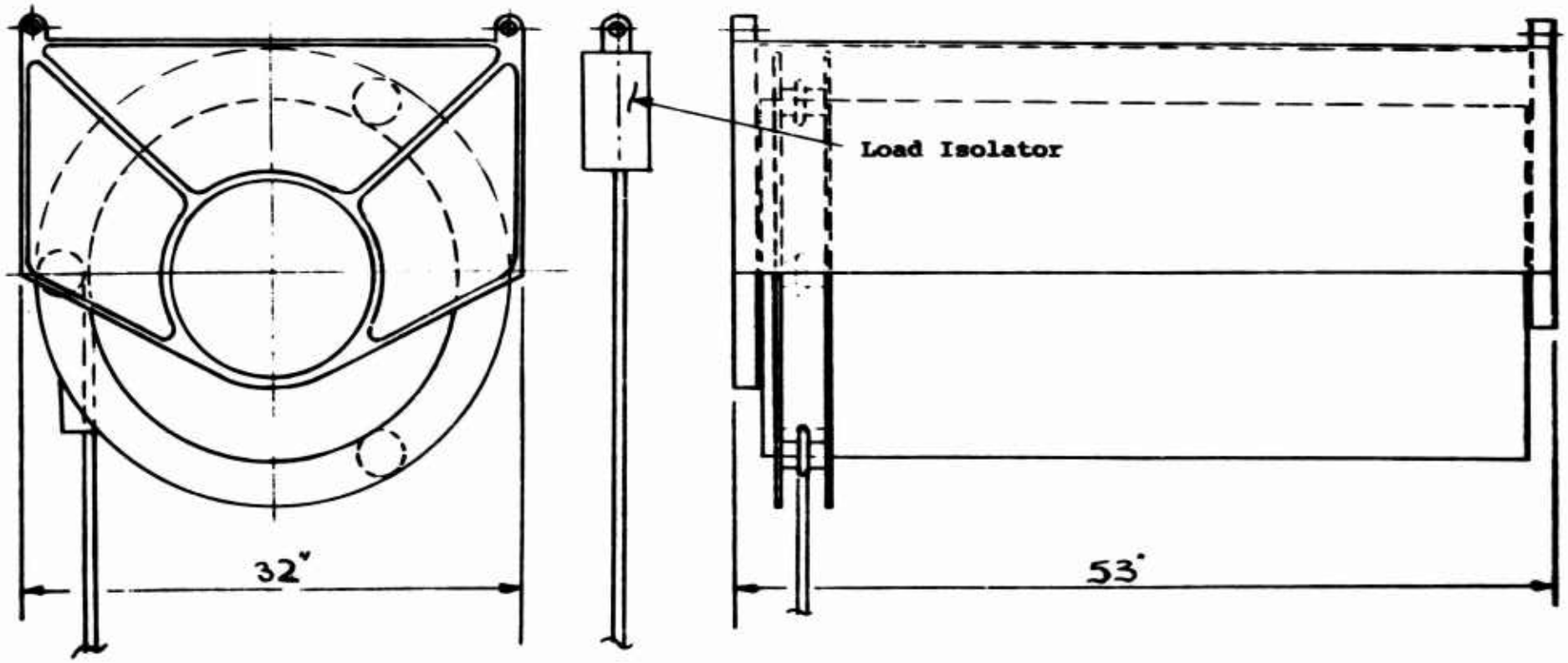
Figure 15. Location of Load Cell, Load Isolator and Rope Cutter on a Single-Reeved Winch Installation.



Load 40,000 x 3.75	=	150,000 pounds
+ 6% Bending Factor	=	9,000 pounds
+ Rope Weight (640 x 3.75)	=	2,400 pounds
+ Hook Weight (150 x 3.75)	=	562 pounds
Rope Design Load		161,962 pounds

Lift Height = 150 feet
Rope Size=1.375 in.dia.
Rope Break Strength = 183,400 pounds

Figure 16. Single-Point, Single-Rope Winch.

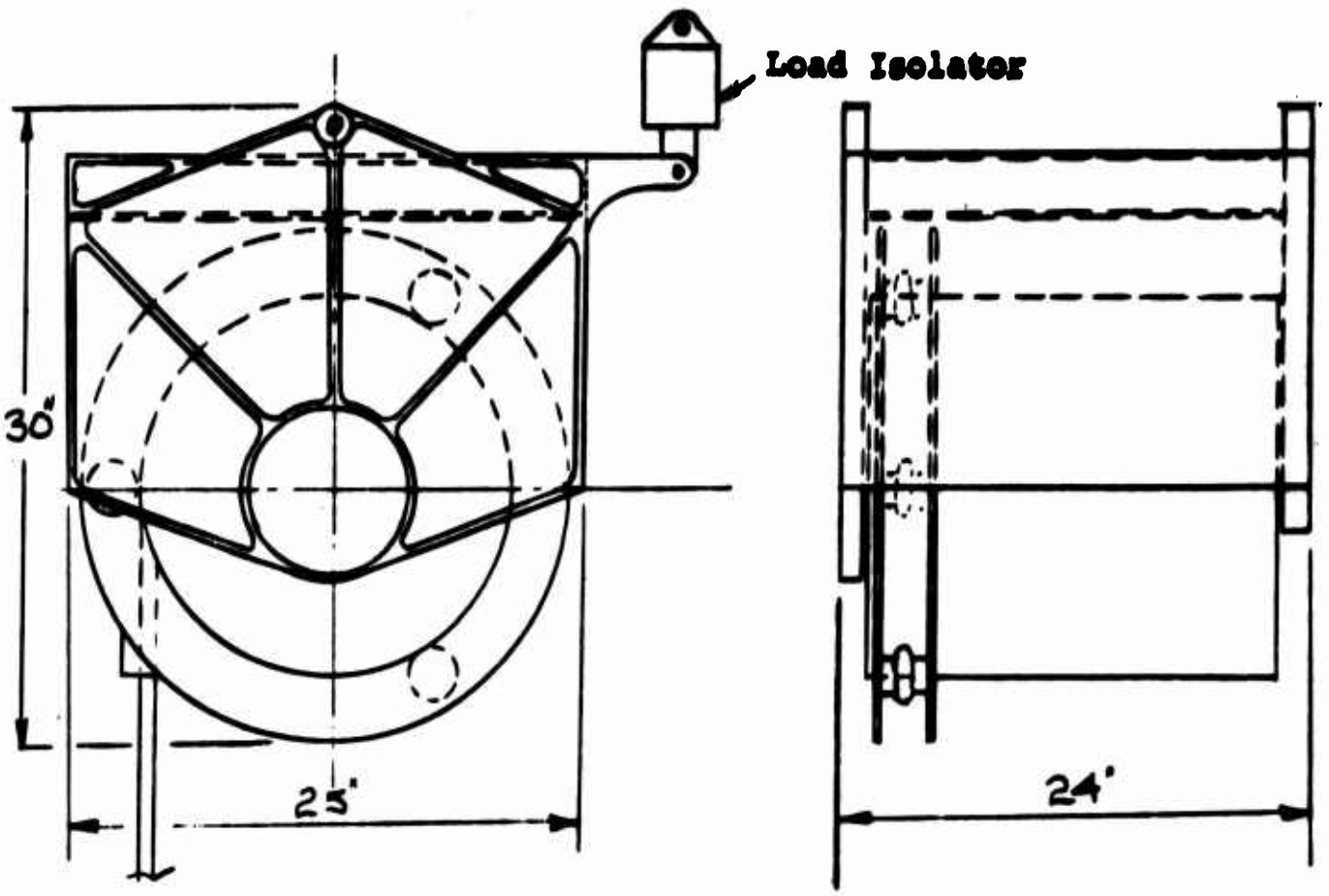


Load	40,000 x 3.75 =	150,000 pounds	* Drum Ratio = 24:1
6% Bending Factor	=	9,000 pounds	Drum Dia. = 24"
Rope Weight (630)	=	2,362 pounds	Lift Height= 150 Ft.(300 ft.rope)
Hook Weight (150)	=	563 pounds	
Rope Design Load	=	161,925 ÷ 2 = 80,962 pounds	

Rope Size = 1.0 in.dia.
Rope Break Load = 98,200 pounds

* A ratio of 24:1 is used since with an 18:1 ratio the rope drum length becomes excessive (approx. 75")

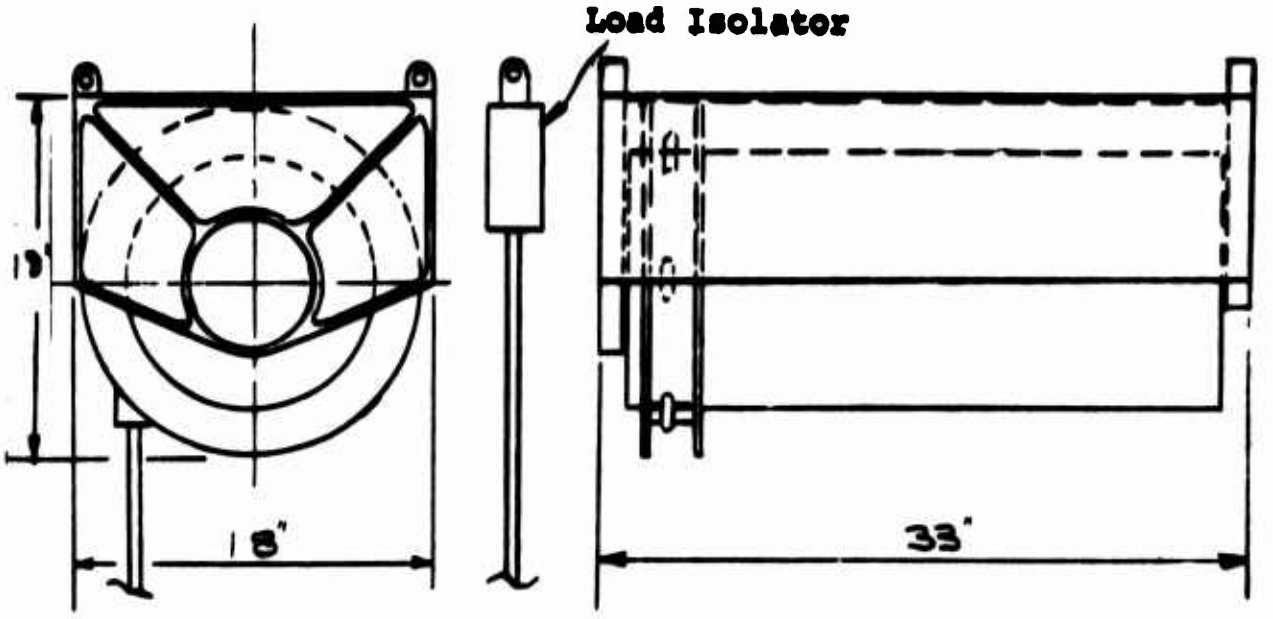
Figure 17. Single-Point, Single-Reeved Winch.



Load per Winch	15,515 x 3.75	=	58,181 pounds
+ 6% Bending Factor (3490)		=	3,490 pounds
+ Rope Weight (92 x 3.75)		=	345 pounds
+ Hook Weight (40 x 3.75)		=	150 pounds
Rope Design Load		=	62,166 pounds

Lift Height = 50 feet
Rope Dia. = 7/8 in.
Rope Break Strength = 17,760 pounds
Drum/Cable Dia. Ratio = 18:1, Drum Dia. = 18 inches

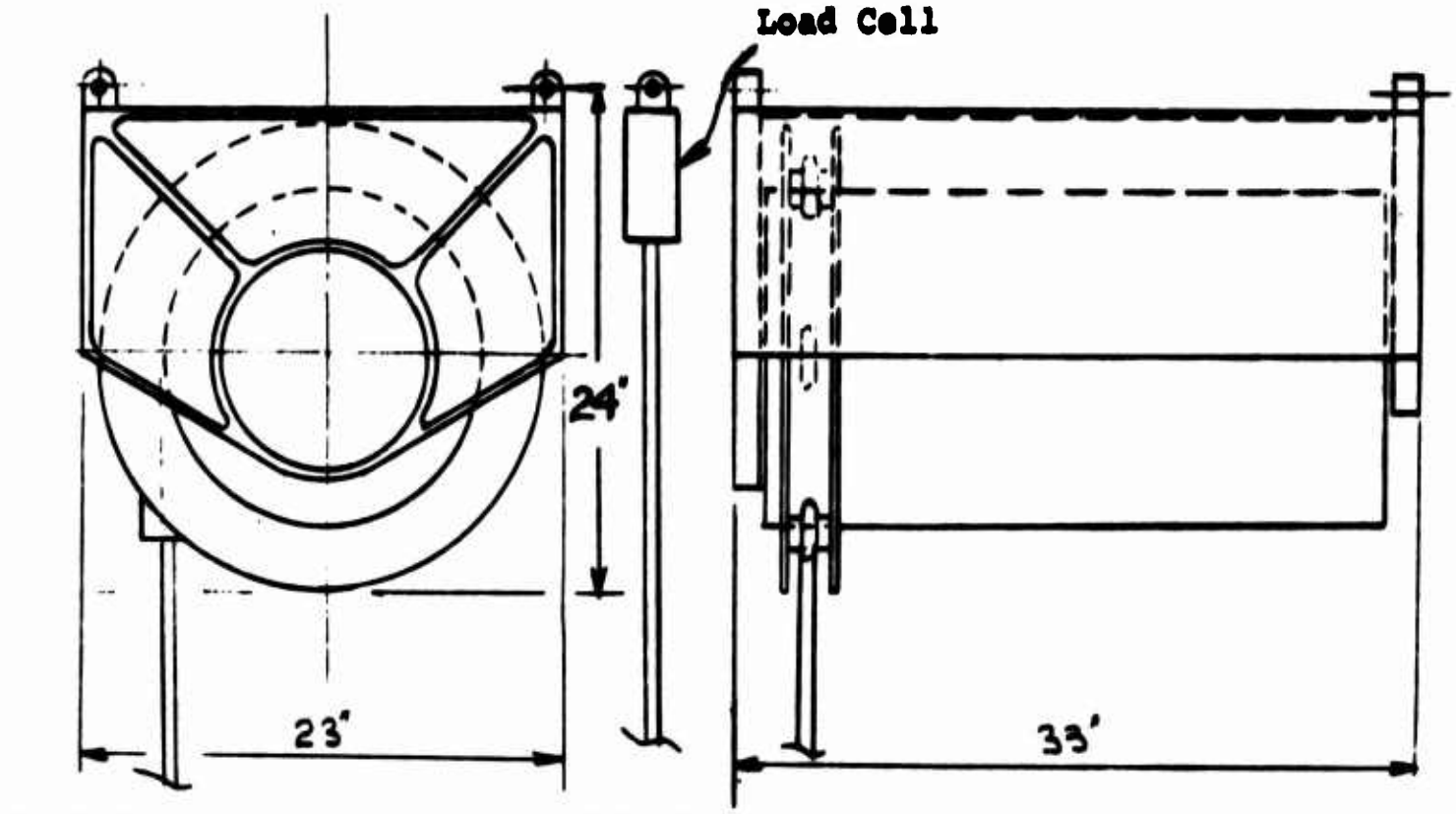
Figure 18. Winch for Four-Point Hoist System.



Load	$\frac{15,515 \times 3.75}{2}$	=	29,090 pounds
+ 6% Bending Factor (3133)		=	1,745 pounds
+ Rope Weight 80 x 3.75 x 2		=	600 pounds
+ Hook Weight 50 x 3.75		=	187 pounds
Rope Design Load		=	31,622 pounds

Lift Height = 50 Feet (120 ft cable) Rope Drum Dia = 18:1
Rope Size = .75in dia Drum Diameter = 13.5 in.
Rope Break Strength = 39,200 pounds

Figure 19. Four-Point Hoist System Using Four Winches and Single-Reeved Rope.

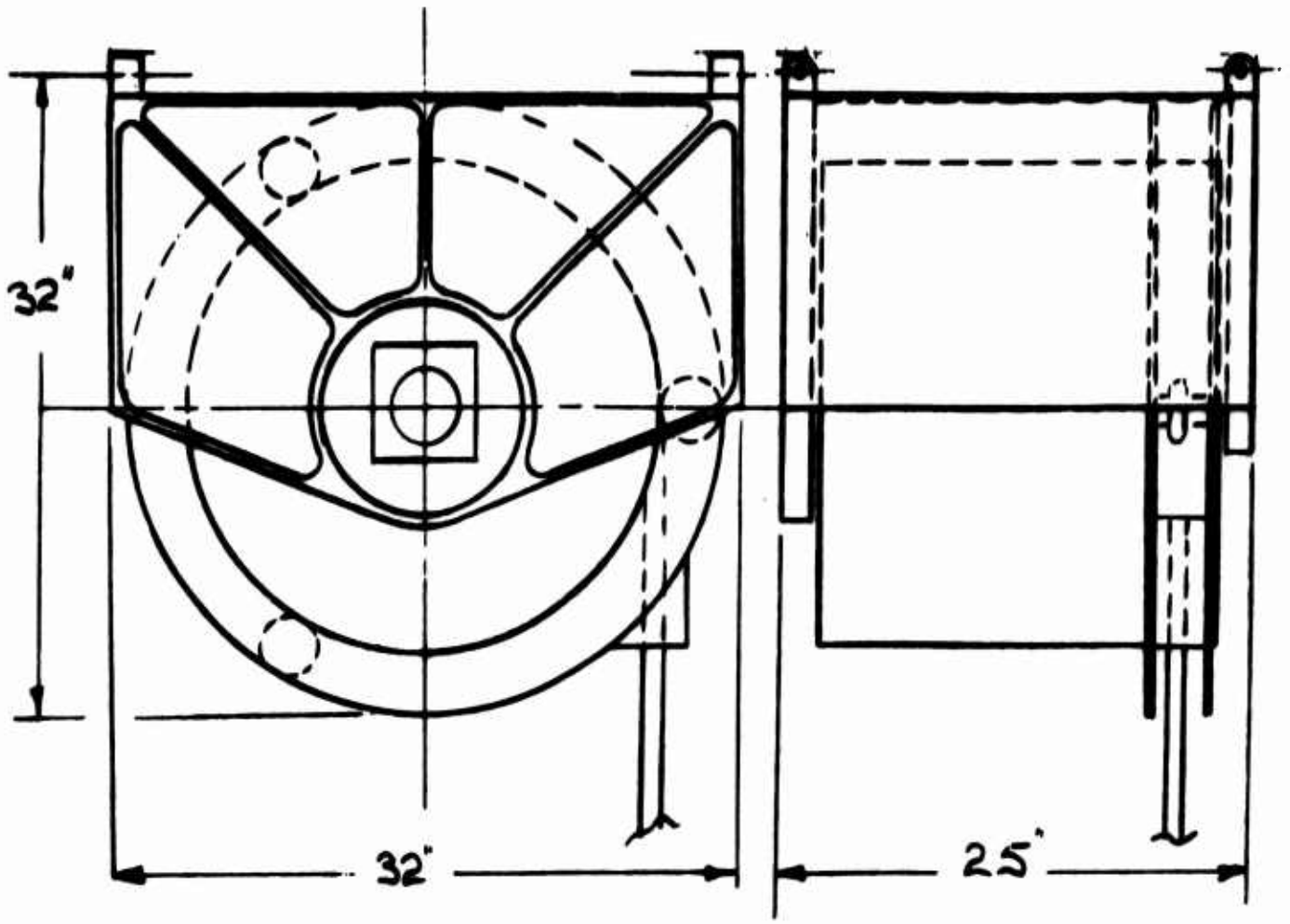


14,104 14,104

Load	28,208 x 3.75	= 105,780 pounds
+ 6% Bending Factor		= 6,347 pounds
+ Rope Weight	160 x 3.75	= 600 pounds
+ Half Beam Assembly Weight		
	300 x 3.75	= 1,125 pounds
System Design Load		= 113,852 pounds
Winch Design Load		
	113,852 ÷ 2	= 56,926 pounds

Lift Height= 50 Ft. (100 ft. cable)
Rope Size = .875 in.dia.
Rope Break Strength= 75,760 pounds

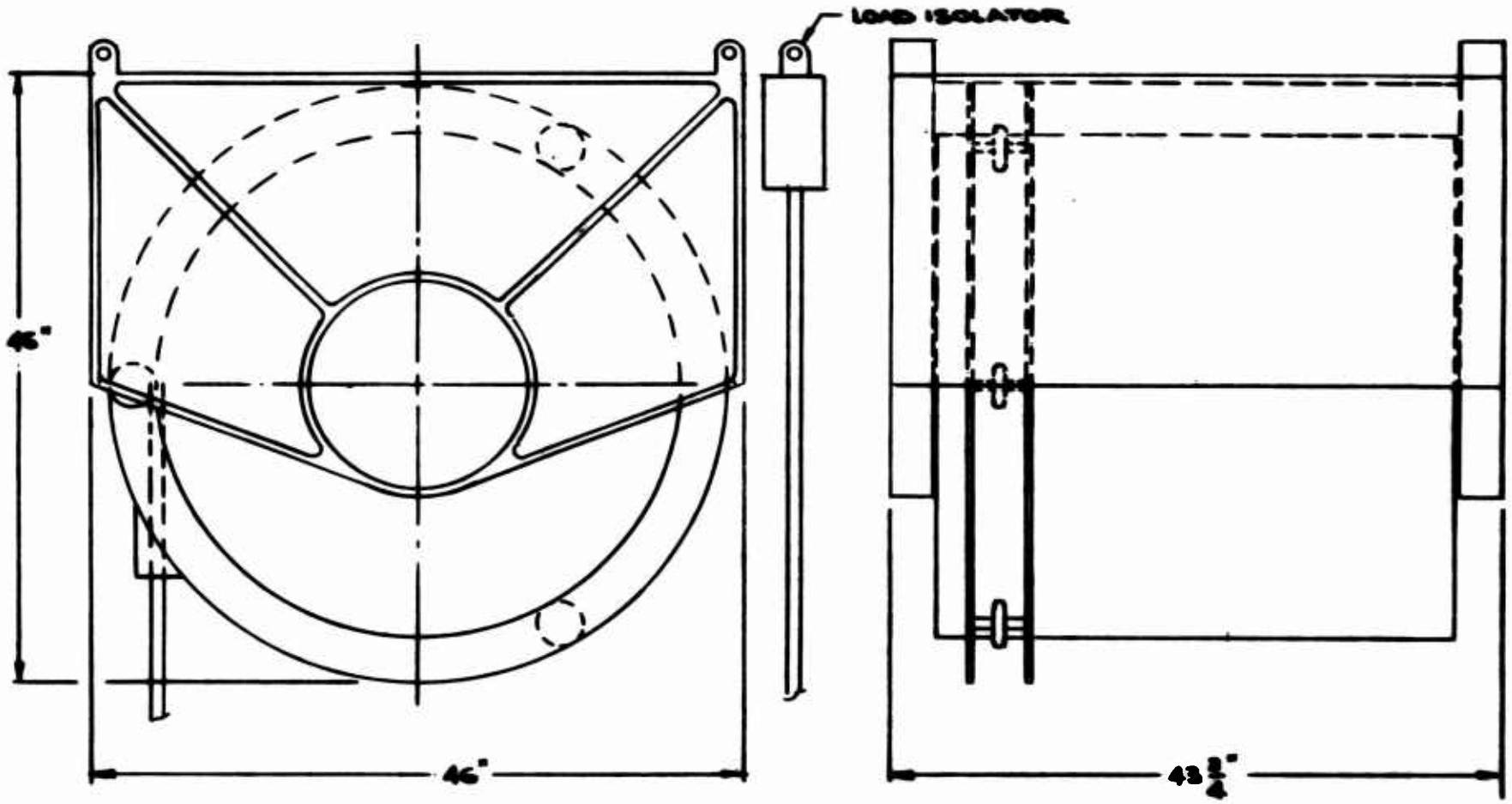
Figure 20. Two-Point Hoist System Using Two Winches and Single-Reeved Rope.



Load 40,000 x 3.75	=	150,000 pounds
+ 6% Bending Factor	=	9,000 pounds
+ Rope Weight (640 x 3.75)	=	2,400 pounds
+ Hook Weight (150 x 3.75)		562 pounds
Rope Design Load	=	161,962 pounds

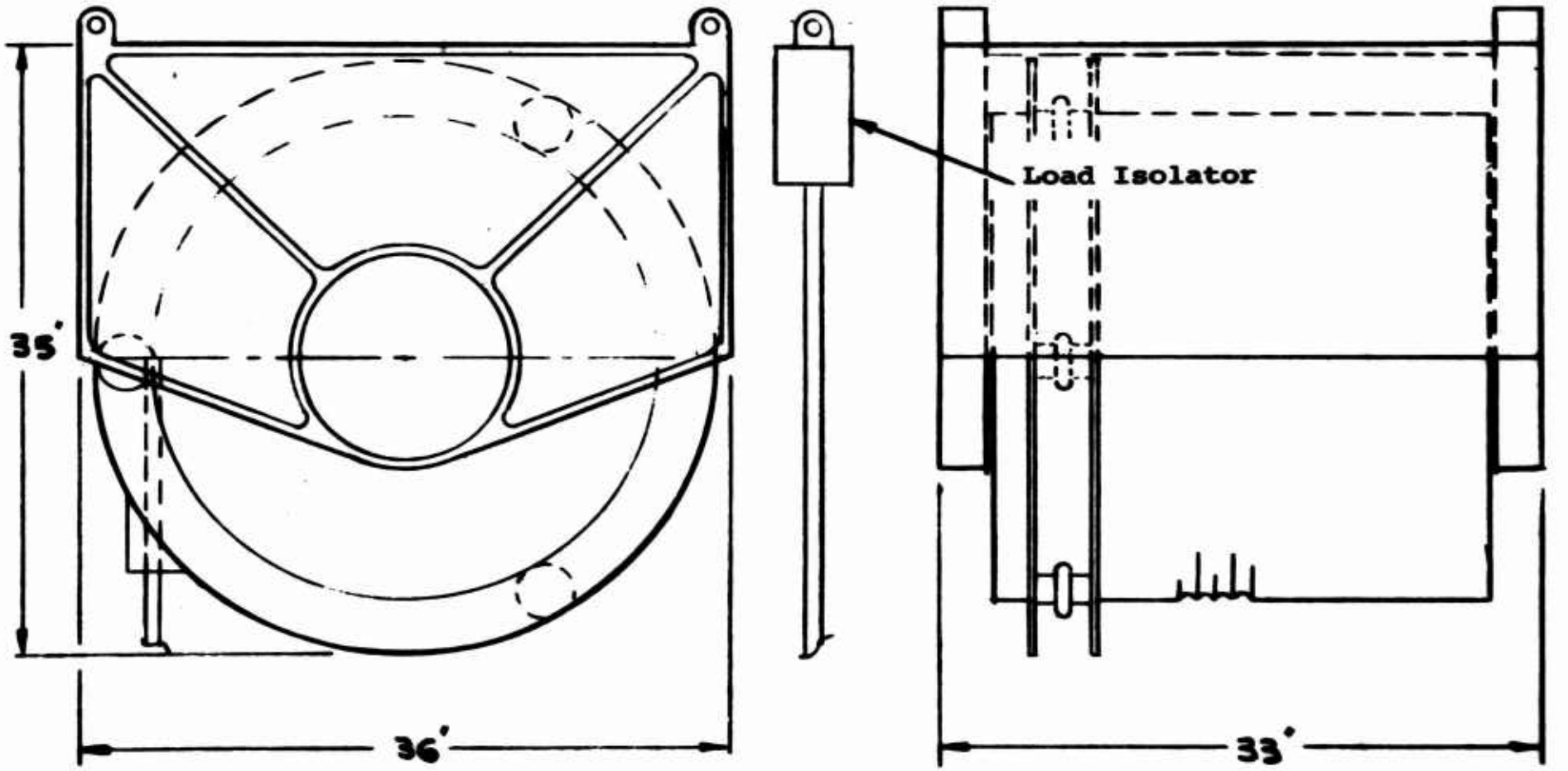
Lift Height = 150 feet
Rope Size = 1.375 in. dia.
Rope Break Strength = 183,400 pounds

Figure 21. Single-Point, Single-Rope Winch (150-Foot Lift).



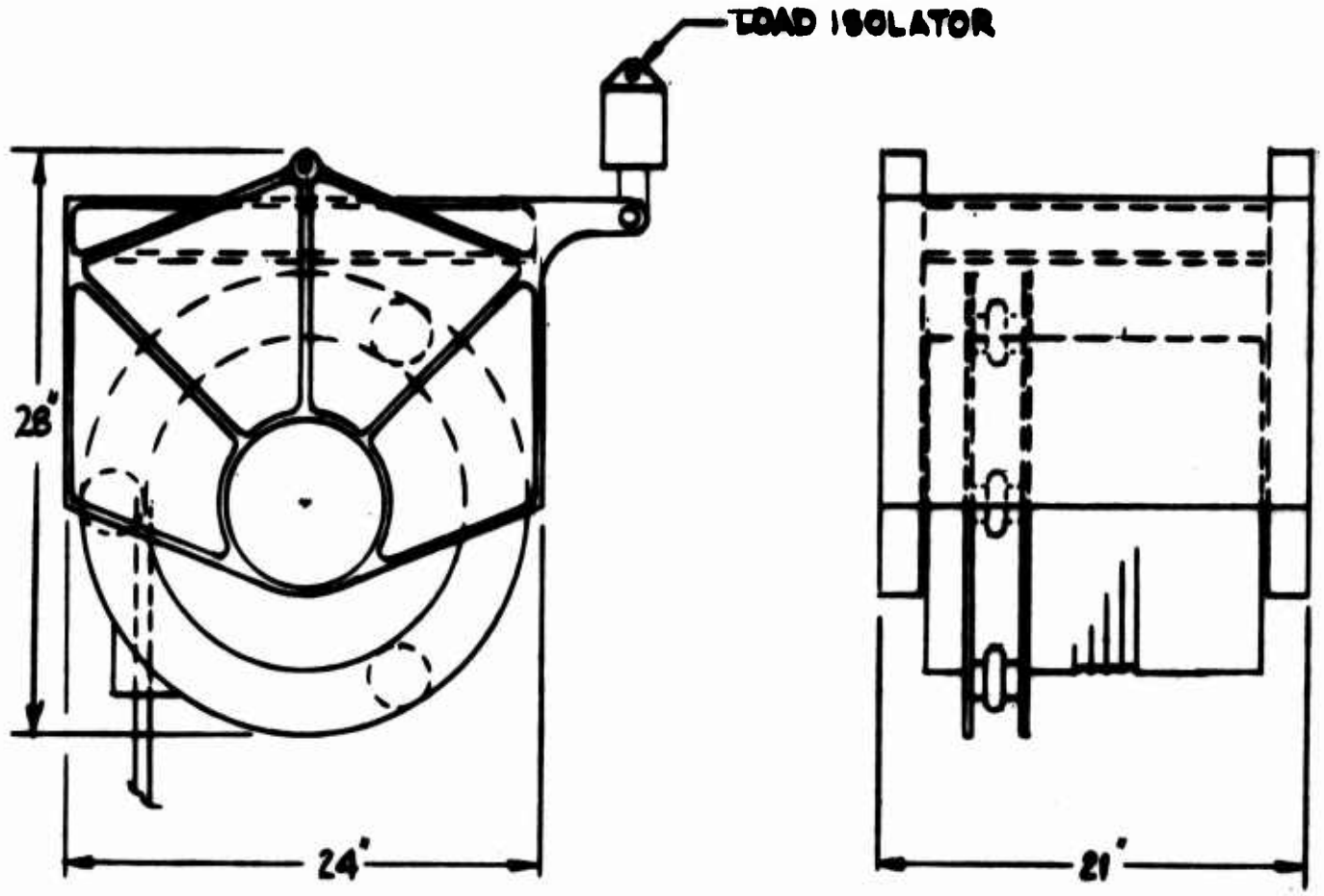
Load 40,000 x 3.75	= 150,000 pounds	Drum Ratio =	37:1
+ 6% Bending Factor	= 9,000 pounds	Drum Dia. =	37.00 in.
+ Rope Weight (630 x 3.75)	= 2,362 pounds	Lift Height =	150 ft (300 ft rope)
+ Hook Weight (150 x 3.75)	= 563 pounds		
Rope Design Load	= 161,925 pounds ÷ 2 = 80,962 pounds		
Rope Size	= 1.0 in. dia.		
Rope Break Strength	= 98,200 pounds		

Figure 22. Single-Point, Single-Reeved Rope (150- Foot Lift)



Load 40,000 x 3.75	= 150,000 pounds	Drum Ratio =	28:1
+ 6% Bending Factor	= 9,000 pounds	Drum Dia. =	28.00 in.
+ Rope Weight (352 x 3.75)	= 1,321 pounds	Lift Height =	80 ft (160 ft Rope)
+ Hook Weight (150 x 3.75)	= 563 pounds		
Rope Design Load	= 160,884 pounds		
Rope Size	= 1.0in.dia.		
Rope Break Strength	= 98,200 pounds		

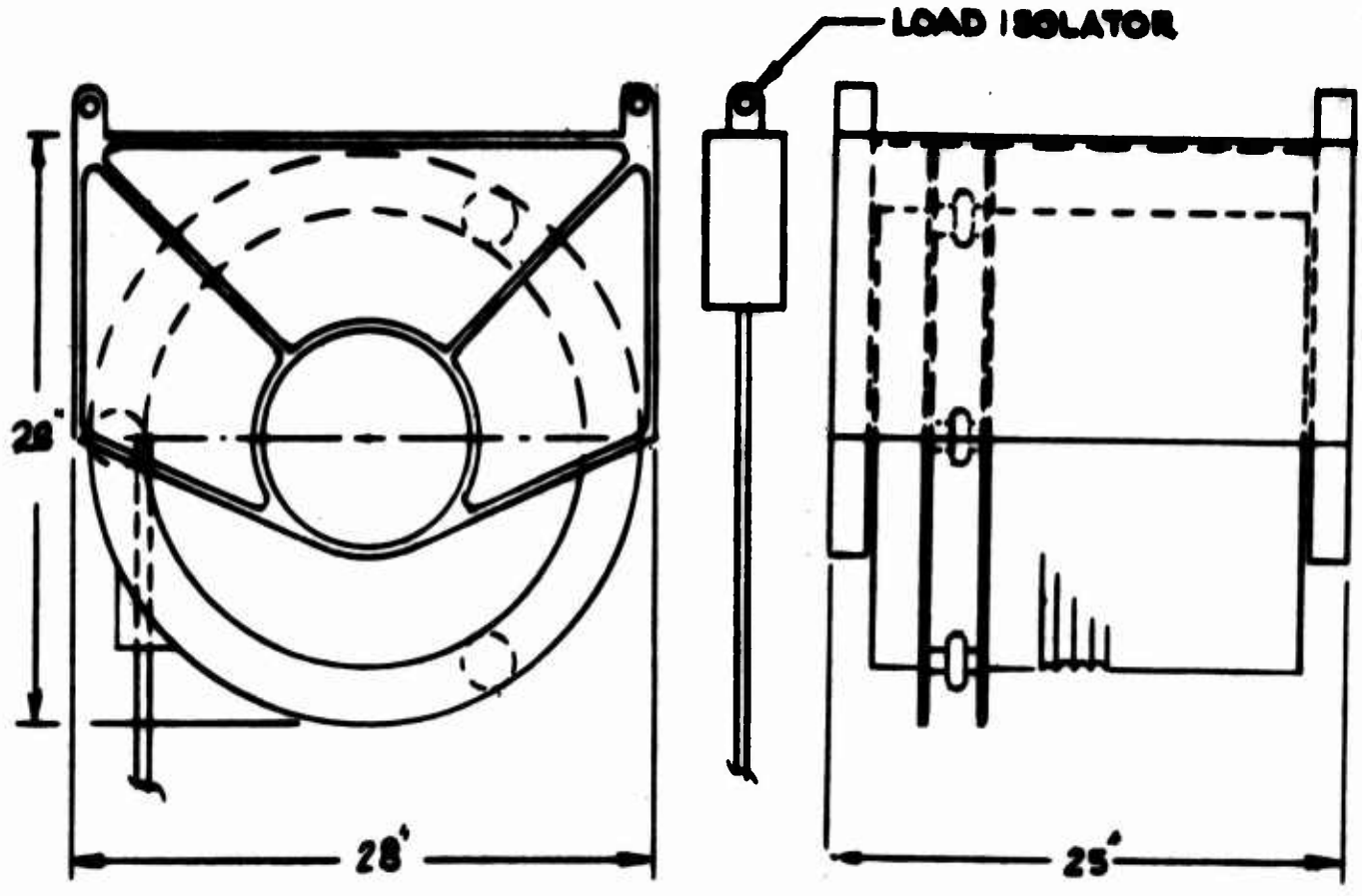
Figure 23. Single-Point, Single-Reeved Rope (80-Foot Lift).



Load per Winch 15,515 x 3.75	=	58,181 Pounds
+ 6% Bending Factor 4533	=	3,490 Pounds
+ Rope Weight (92 x 3.75)	=	345 Pounds
+ Hook Weight (40 x 3.75)	=	150 Pounds
Rope Design Load	=	62,166 Pounds

Lift Height = 50 feet
Rope Dia. = 7/8in.dia.
Rope Break Strength = 75,760 Pounds
Rope /Drum Ratio = 18:1
Drum Dia. = 16.00in.

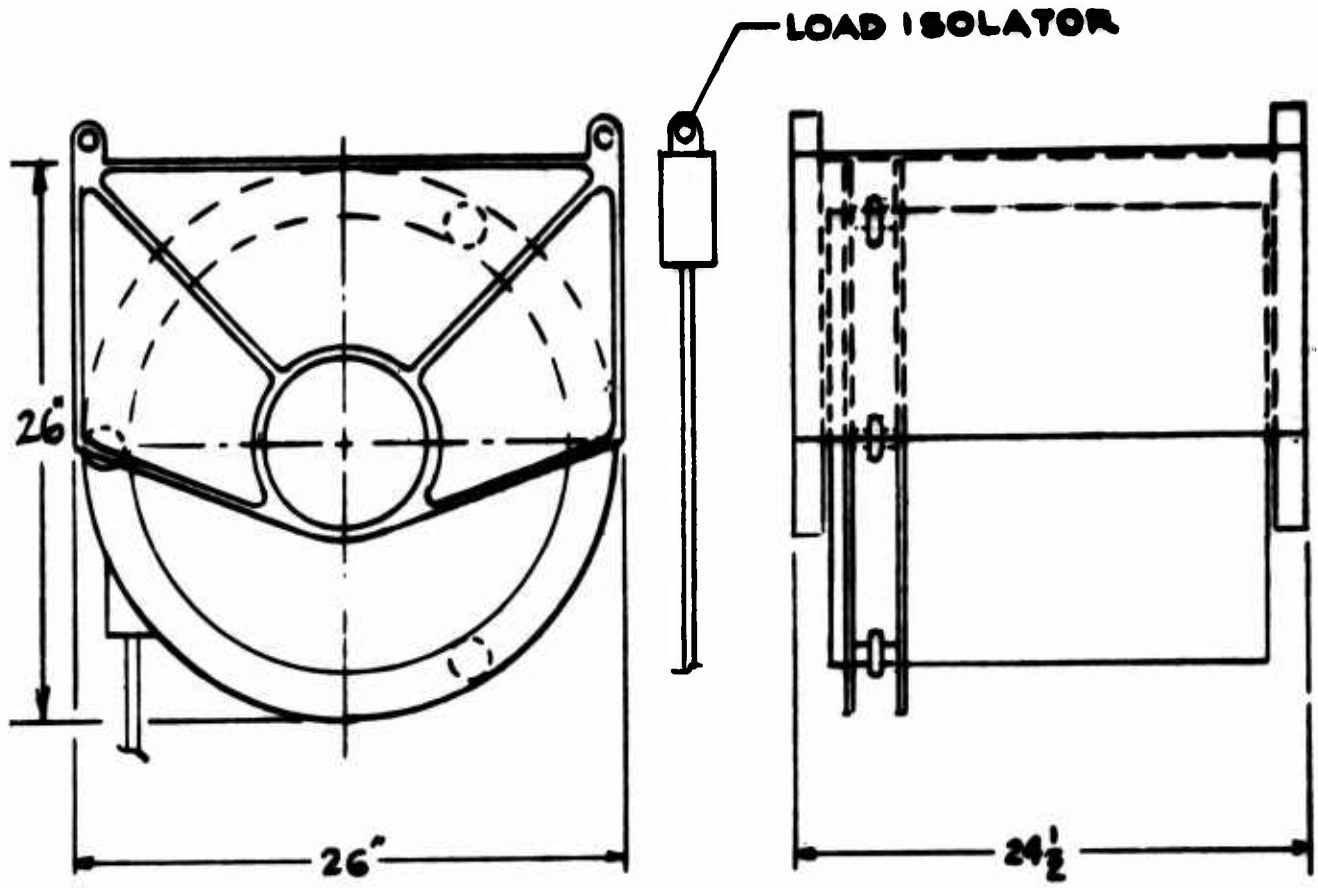
Figure 24. Winch for Four-Point Hoist System.



Load 28,208 x 3.75	=	105,780 Pounds
+ 6% Bending Factor	=	6,347 Pounds
+ Rope Weight (160 x 3.75)	=	600 Pounds
+ Half Beam Assy Weight (300 x 3.75)	=	1,125 Pounds
Rope Design Load =		113,852 Pounds
Winch Design Load 113,852 ÷ 2 =		56,926 Pounds

Lift Height = 50 feet (100 ft Cable)
Rope Size = .875 in. dia.
Rope Break Strength = 75,760 Pounds
Rope Drum Ratio = 24:1
Drum Dia. = 21.00 in. dia.

Figure 25. Two-Point Hoist System Using Two Winches and Single-Reeved Rope.



Load	$\frac{20,149 \times 3.75}{2}$	=	37,778 Pounds
+ 6% Bending Factor (3133)		=	1,566 Pounds
+ Rope Weight (67 x 3.75) + 2		=	126 Pounds
+ Hook Weight 50 x 3.75		=	187 Pounds
Rope Design Load		=	39,657 Pounds

Lift Height-50 feet (120 ft Cable)
Rope Size-.75 in. dia
Rope Break Strength-55,800 Pounds
Rope Drum Dia.-28:1
Drum Diameter-21.0 in.

Figure 26. Four-Point Hoist System Using Four Winches and Single-Reeved Rope

give partial protection against the elements. The configurations shown in Figures 21 through 26 are all adaptable to this closure method. Figure 27 illustrates such a cover installed.

The winch base is considered to be the optimum location for the attachment pickup points. This location is not considered mandatory, and subsequent installation requirements may dictate that the pickups be located on the winch body sides.

Capstan-Type Winches (Refer to Table IV)

This type of winch is often called a "zero-moment" winch; i.e. one in which the rope enters and leaves the drum from a fixed point. If such a winch is suspended from a point directly in line with this payout point, no moment is imposed as the rope is layered on the storage drum. The capstan principle allows the initial load of the rope to be absorbed in a few wraps on the driving (capstan) drums (see the system schematic illustrated in Figure 28). The cable is then taken from the upper driving drum through a level wind to a storage reel. A minimum cable tension is always maintained on this storage reel by driving through a slip clutch at a peripheral speed slightly in excess of that of the driving drums, thus allowing the storage drum to overrun. Looseness or slack in the rope is prevented by causing the rope to drag through a tension roller as it is wound onto the driving drum. The driving drums are grooved to minimize chafing of the heavily loaded rope. It is not necessary to groove the storage reel, since the rope is wound on it in a lightly loaded condition.

The winch design configuration is dictated by recommended design practice for wire rope, and the principal item which influences the design of capstan-type winches is the rope fleet angle (refer to the section entitled "THE INFLUENCE OF WIRE ROPE CONSIDERATIONS ON HOIST SYSTEM DESIGN" on page 3). The rope drive is taken through two driven capstans which consist of two drums suitably grooved to guide the rope. In traversing from one groove in the first drum to the adjacent groove in the second drum,the rope must make a "fleet" angle, which means that the rope is out of alignment with the grooves (see Figure 29). If this angle is excessive, the rope, which is under heavy load, is subject to severe chafing with subsequent reduction in service life. A similar situation exists when the rope leaves the capstan drums and is wound onto the storage reel through a level wind system. Design investigation indicates that an acceptable configuration can be found but

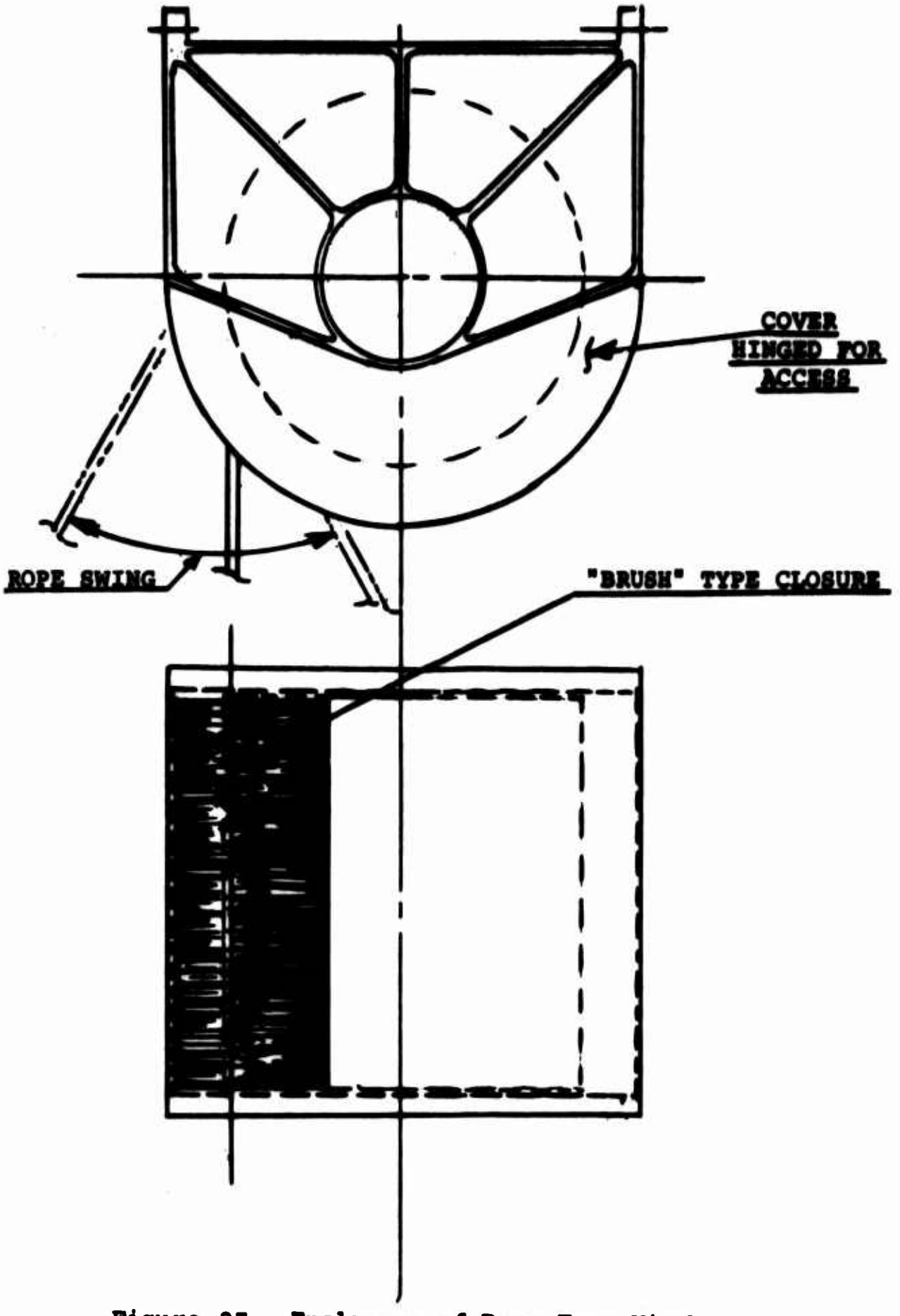


Figure 27. Enclosure of Drum-Type Winch.

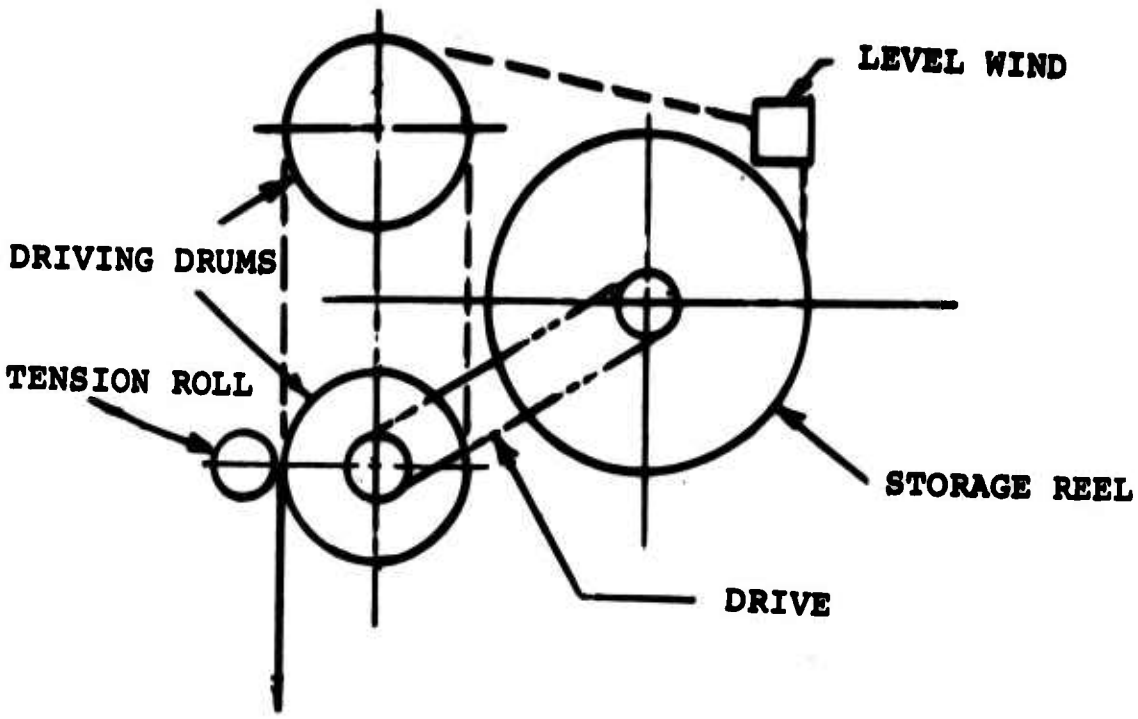


Figure 28. Capstan Winch System.

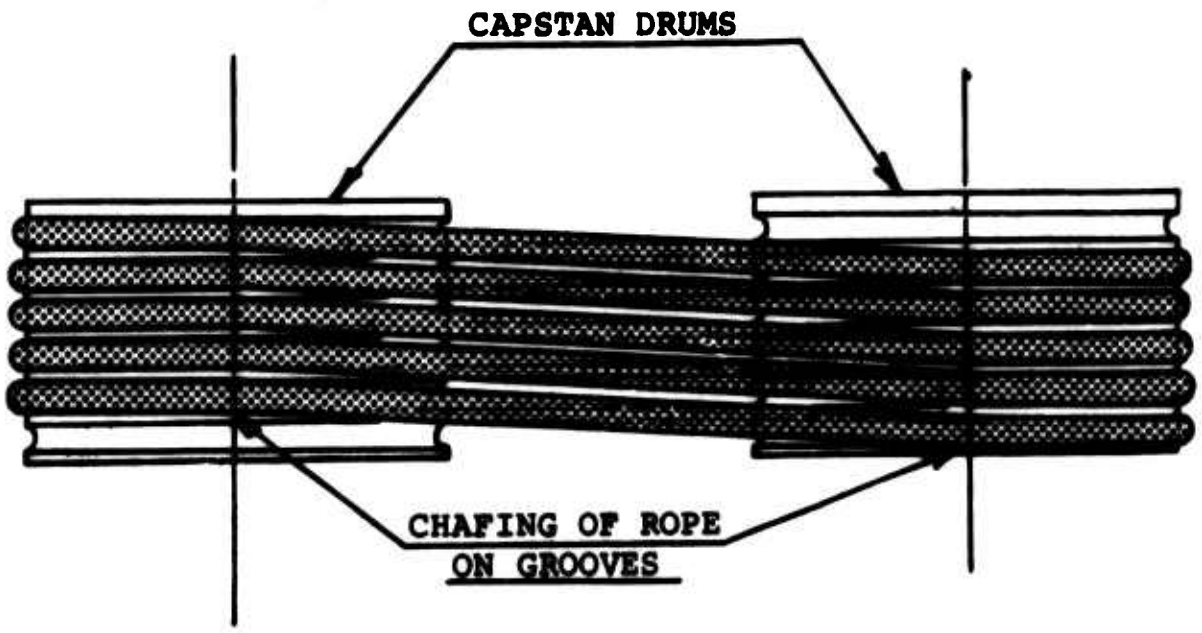


Figure 29. Effect on Rope of Traverse Between Drums of Capstan-Type Winch.

will result in a larger envelope. Figures 30 through 39 illustrate the various configurations investigated, Figure 39 being the one considered best for meeting the established wire rope criteria while allowing design envelope variation.

Criteria used in selecting the configuration were as follows:

1. Drum-diameter/rope-diameter ratios to be the same as used in the configuration review made of drum-type winches, except that the storage drum diameter was established by evolving the ratio which was optimum for rope stowage in relation to rope fleet angle

2. The cg on the unloaded winch to be such that whether rope was reeled on or off its storage drum, the winch would hang vertically

3. Fleet angle of 1-1/2 degrees

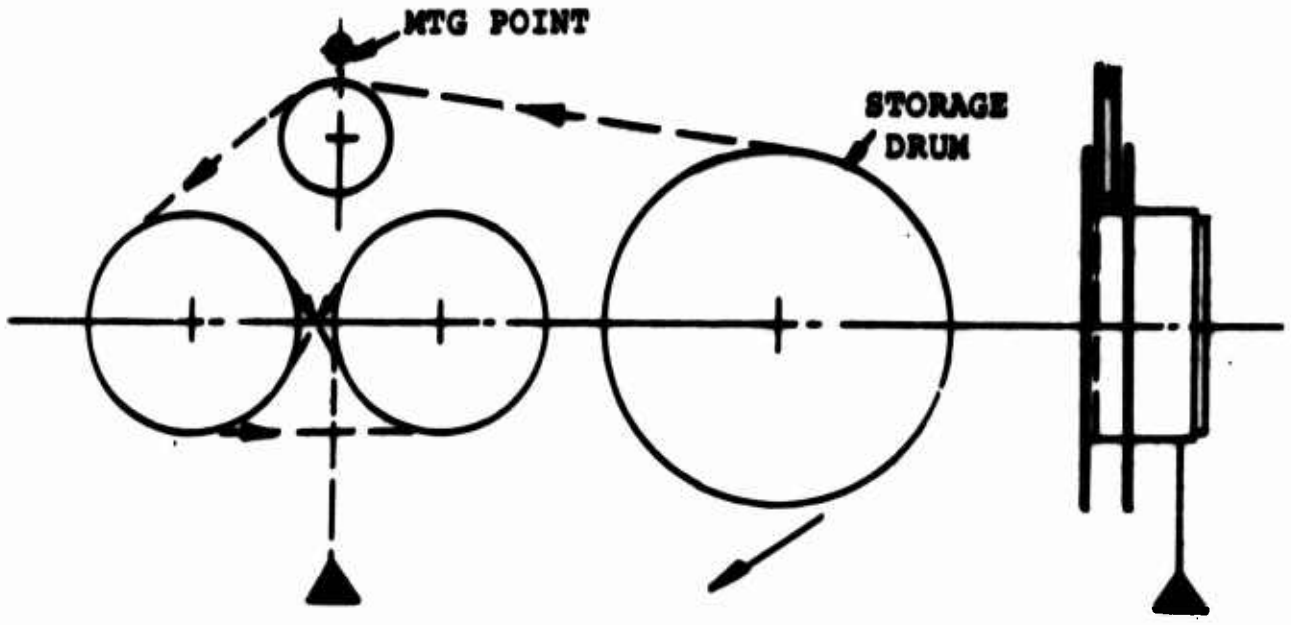
4. No reverse rope bends

5. Compactness of envelope

It will be noted that the chosen configuration differs from that shown on Figure 40, which is a capstan-type winch designed by Breeze Corporation for the CH-54A four-point system. The principal difference lies in the location of the storage reel, which is mounted to the side of the capstan drums; this results in a more compact envelope at the expense of putting reverse bends in the rope. It can be argued that the rope at this point has only nominal tension in it, but the fact remains that, regardless of load, any reverse bends in wire rope are considered undesirable.

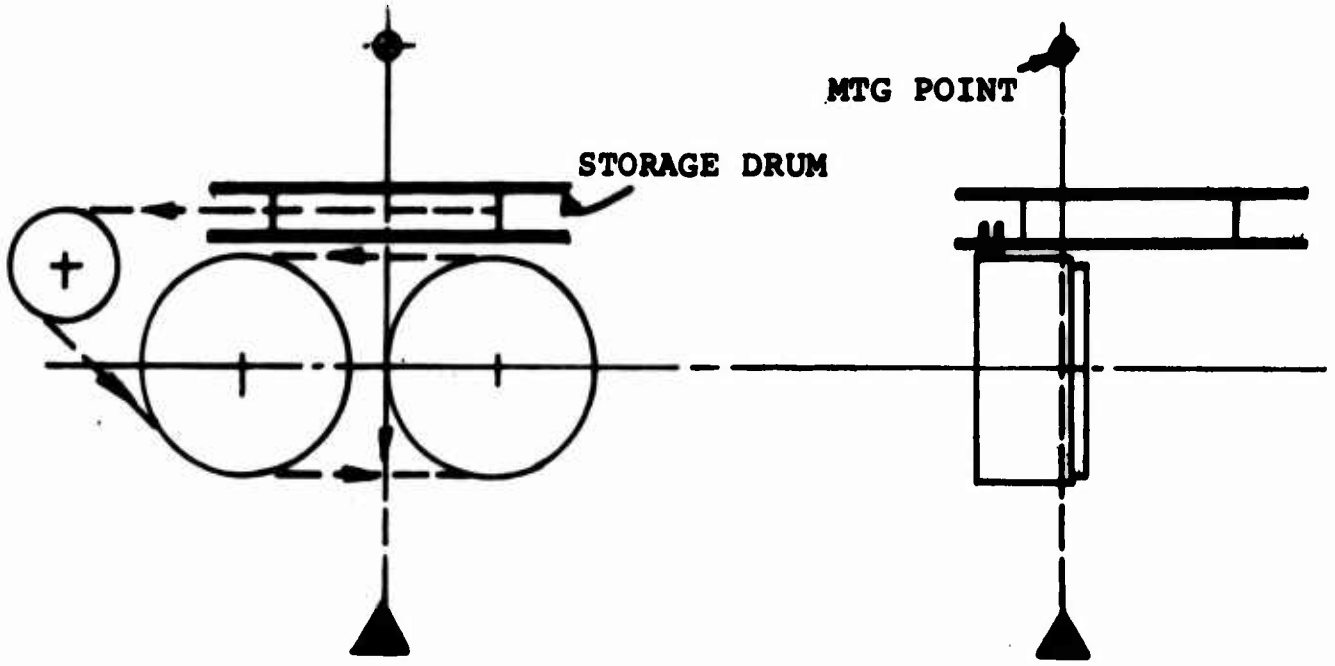
A great advantage of the capstan winch is that the rope length can be varied by changing the capacity of the rope stowage reel. Little increase in envelope results when this type of winch is used in a reeved system. Further, since the winch can be suspended from one point with rope traverse eliminated, no misalignment takes place with a reeved system as is the case with drum types (refer to Figures 11 and 12).

The single-point suspension is also conducive to the installation of load isolators and load cells, since they can be installed in series with the pickup (universal) point.



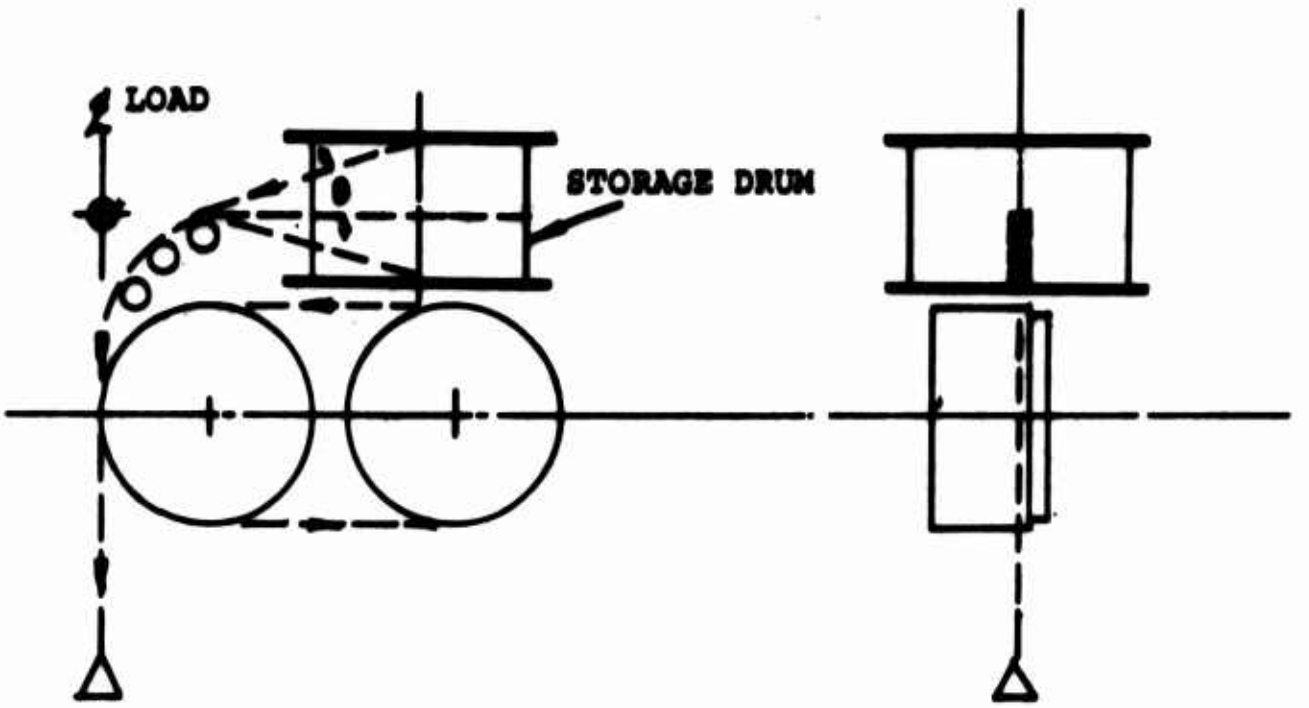
(a) System rotates under no-load condition.
(b) 6% bending factor increased considerably due to reverse bending of rope on load drums.

Figure 30. Capstan-Type Winch Configuration.



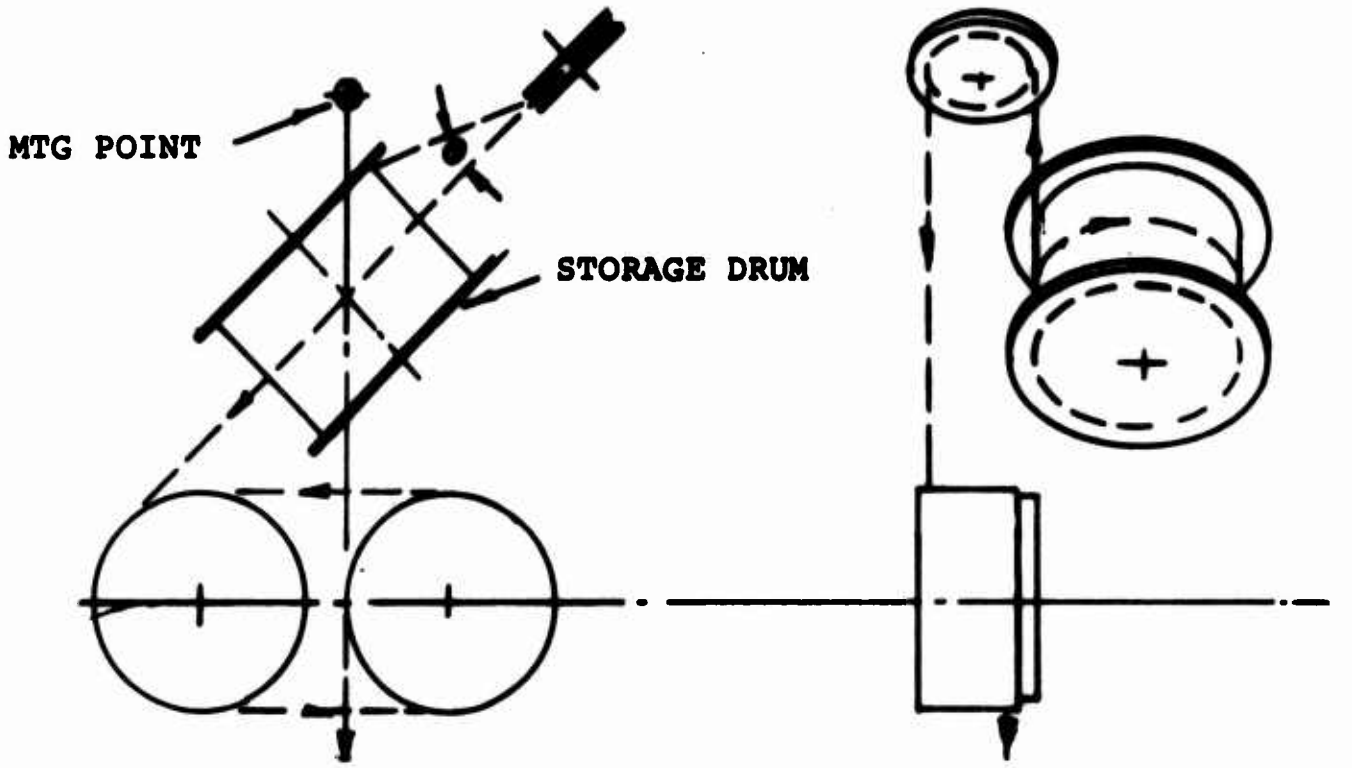
(a) System rotates under no-load condition.
(b) Undesirable right-angle bend of rope between idler pulley and storage drum.
(c) Requires an elaborate and costly mounting configuration.

Figure 31. Capstan-Type Winch Configuration.



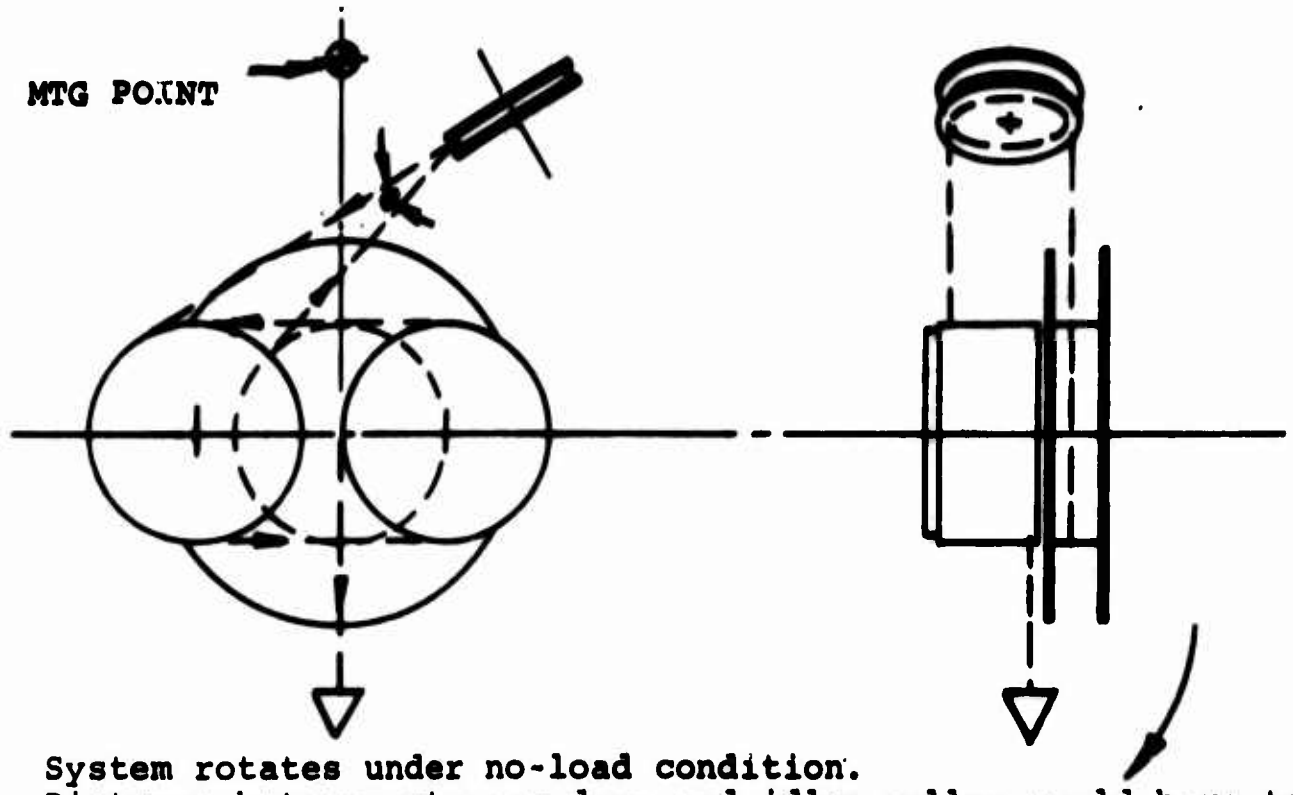
(a) System rotates under no-load condition.
(b) Fleet angle ● considerably exceeds 1-1/2°.

Figure 32. Capstan-Type Winch Configuration.



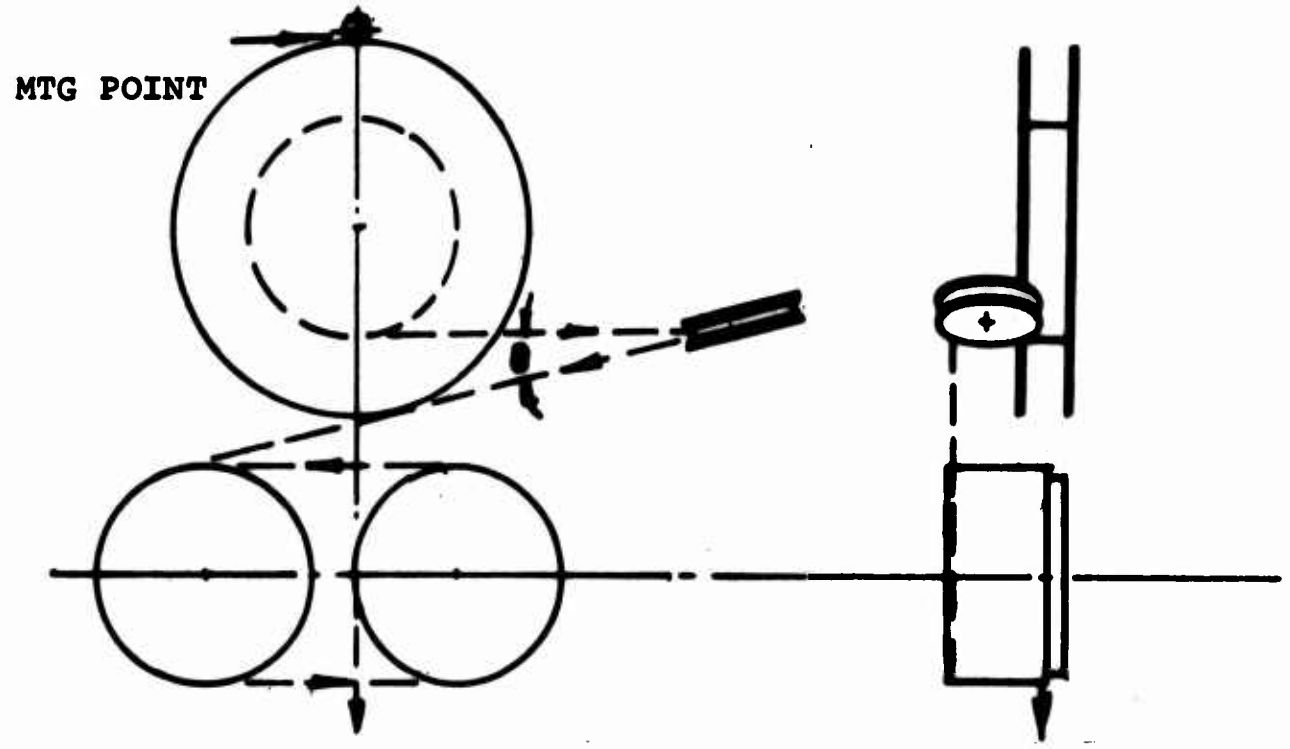
(a) System rotates under no-load condition.
(b) Fleet angle ● considerably exceeds 1-1/2°.
(c) Requires an elaborate and costly mounting configuration.
(d) Distance between storage drum and idler would have to be increased out of proportion to decrease fleet angle ● to 1-1/2°, plus conditions (a) and (c) would remain.

Figure 33. Capstan-Type Winch Configuration.



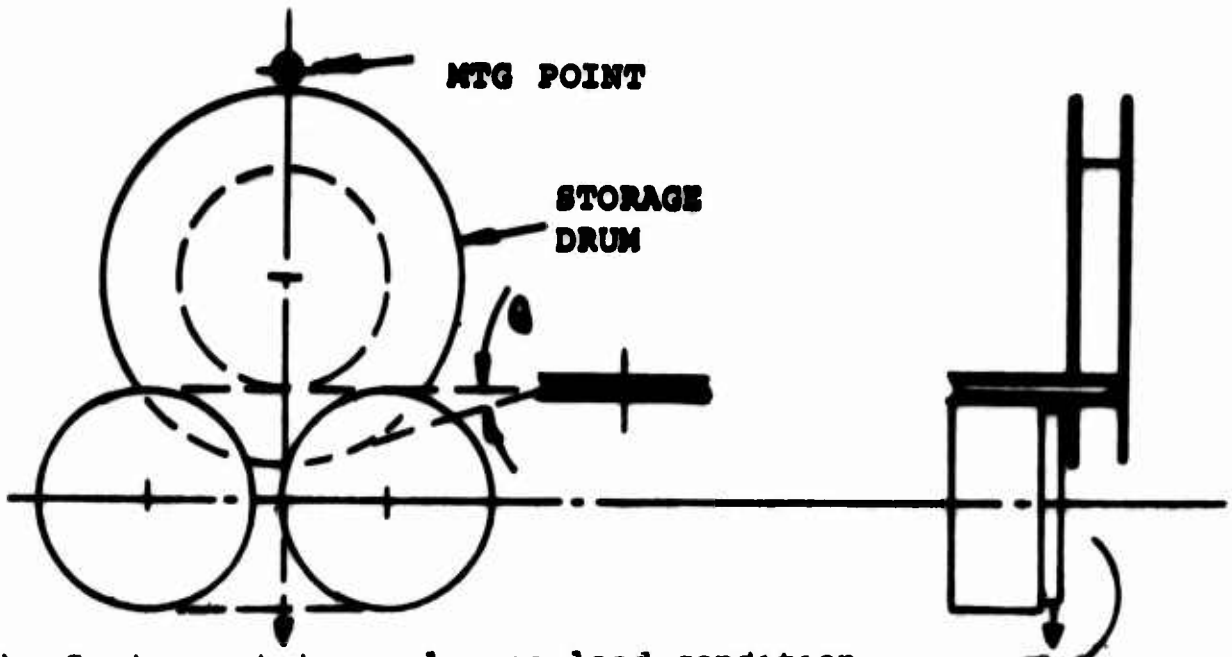
(a) System rotates under no-load condition.
(b) Distance between storage drum and idler pulley would have to be increased out of proportion to decrease fleet angle θ to 1-1/2° plus condition (a) would remain.

Figure 34. Capstan-Type Winch Configuration.



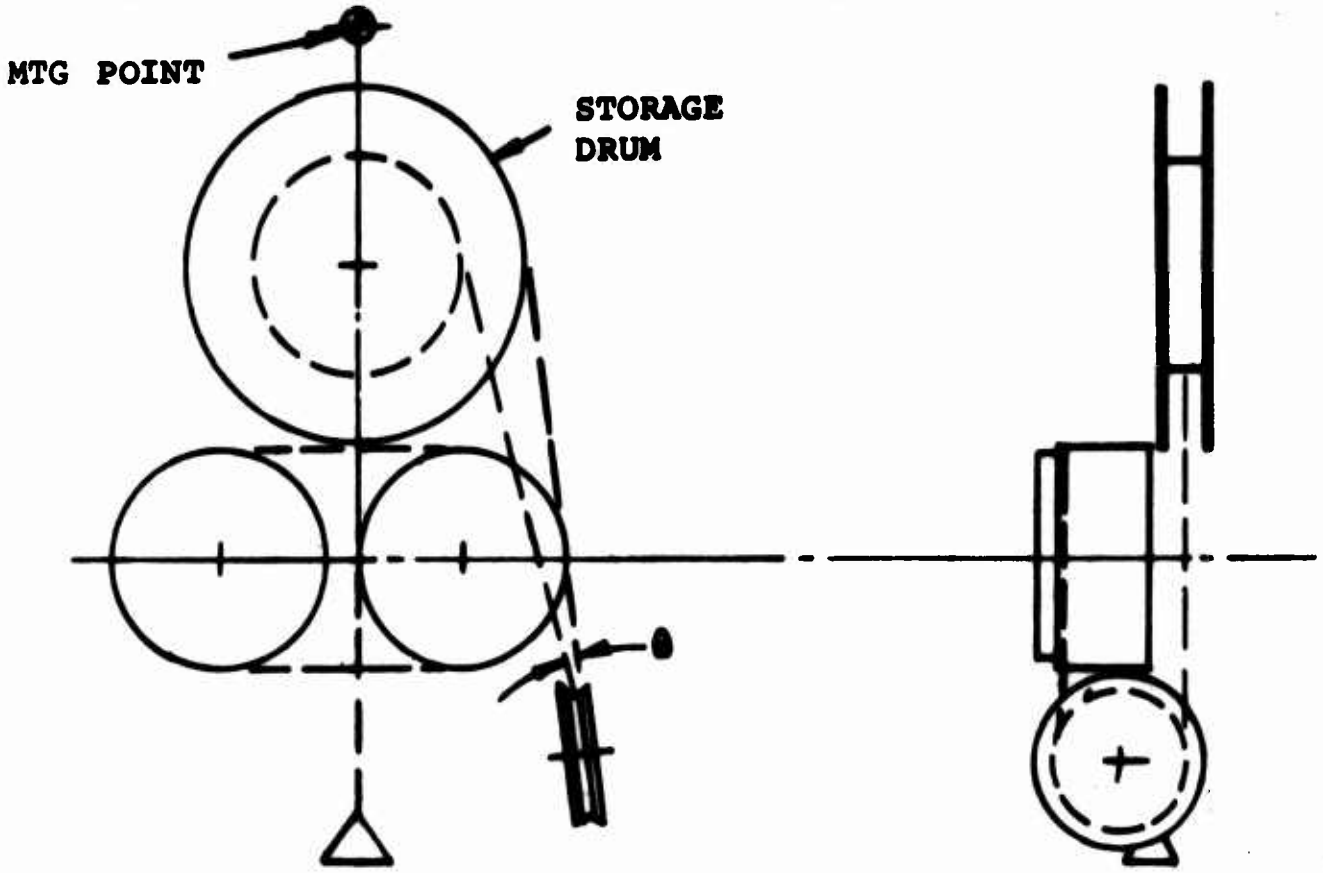
(a) Would require a 7-7/16 in. dia. idler pulley for 1-3/8 in. dia. rope, which would increase 6% bending factor considerably.
(b) Same as (b) in Figure 34.

Figure 35. Capstan-Type Winch Configuration.



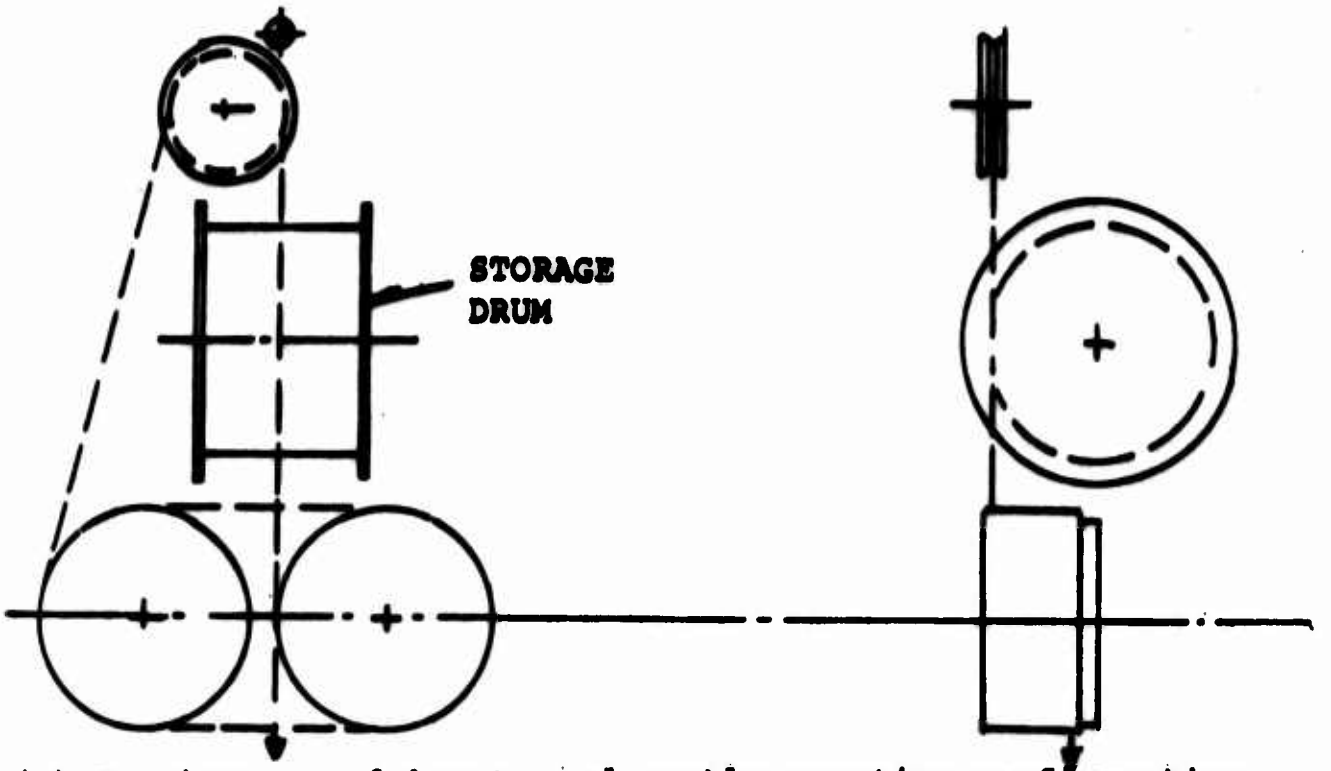
(a) System rotates under no-load condition.
(b) Undesirable right-angle bend of rope between idler pulley and storage drum.
(c) Requires an elaborate and costly mounting configuration.

Figure 36. Capstan-Type Winch Configuration.



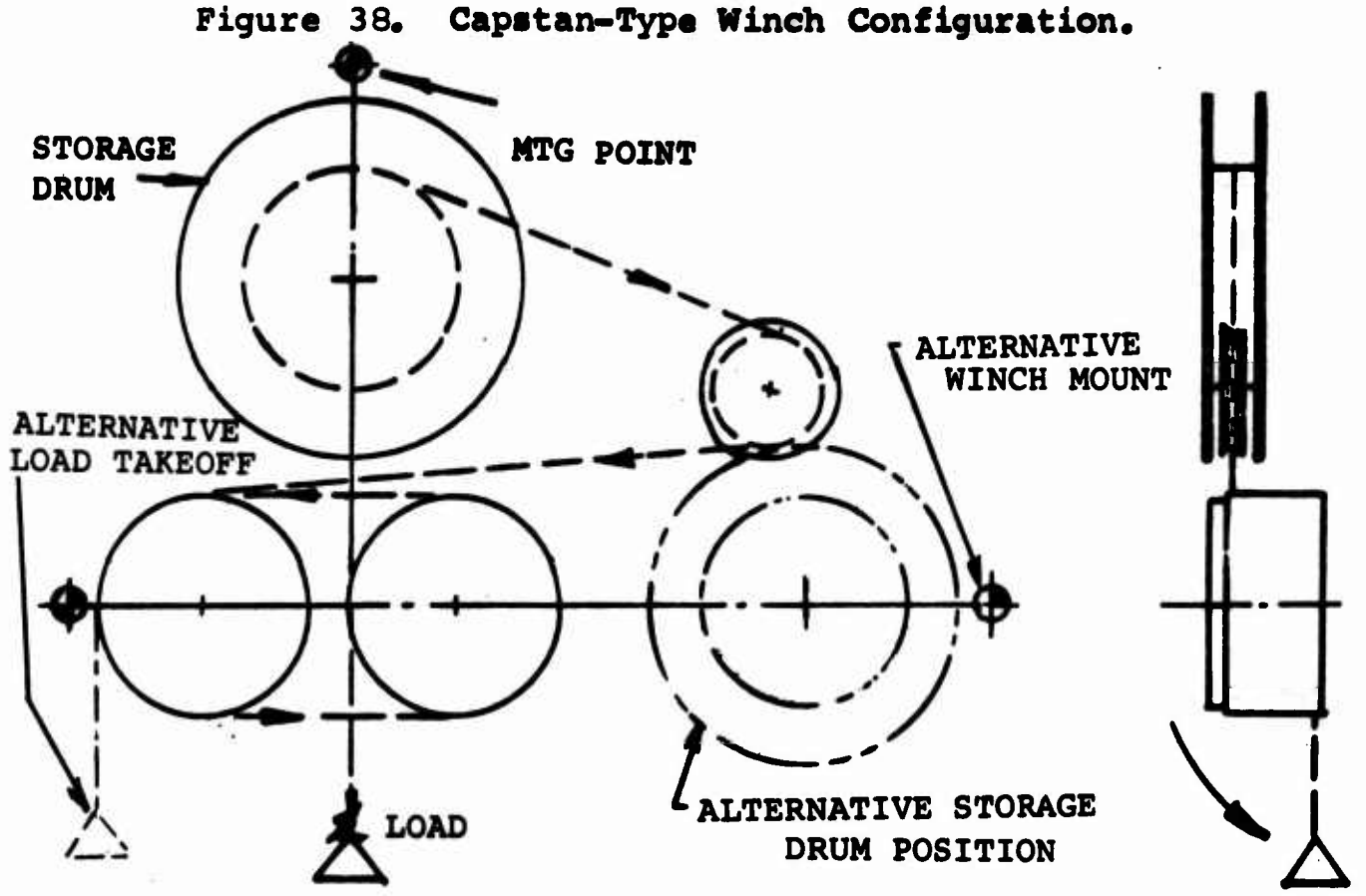
(a) System rotates under no-load condition.
(b) Idler pulley would have to be moved toward load to decrease fleet angle ● to within 1-1/2°.

Figure 37. Capstan-Type Winch Configuration.



(a) Requires an elaborate and costly mounting configuration.
(b) Distance between center line of load drums and universal mtg joint would have to be exceedingly long to be practical.

Figure 38. Capstan-Type Winch Configuration.



(a) System rotates under no-load condition.
(b) Idler pulley must be in extended position shown to maintain 1-1/2° fleet angle.

Figure 39. Capstan-Type Winch Configuration.

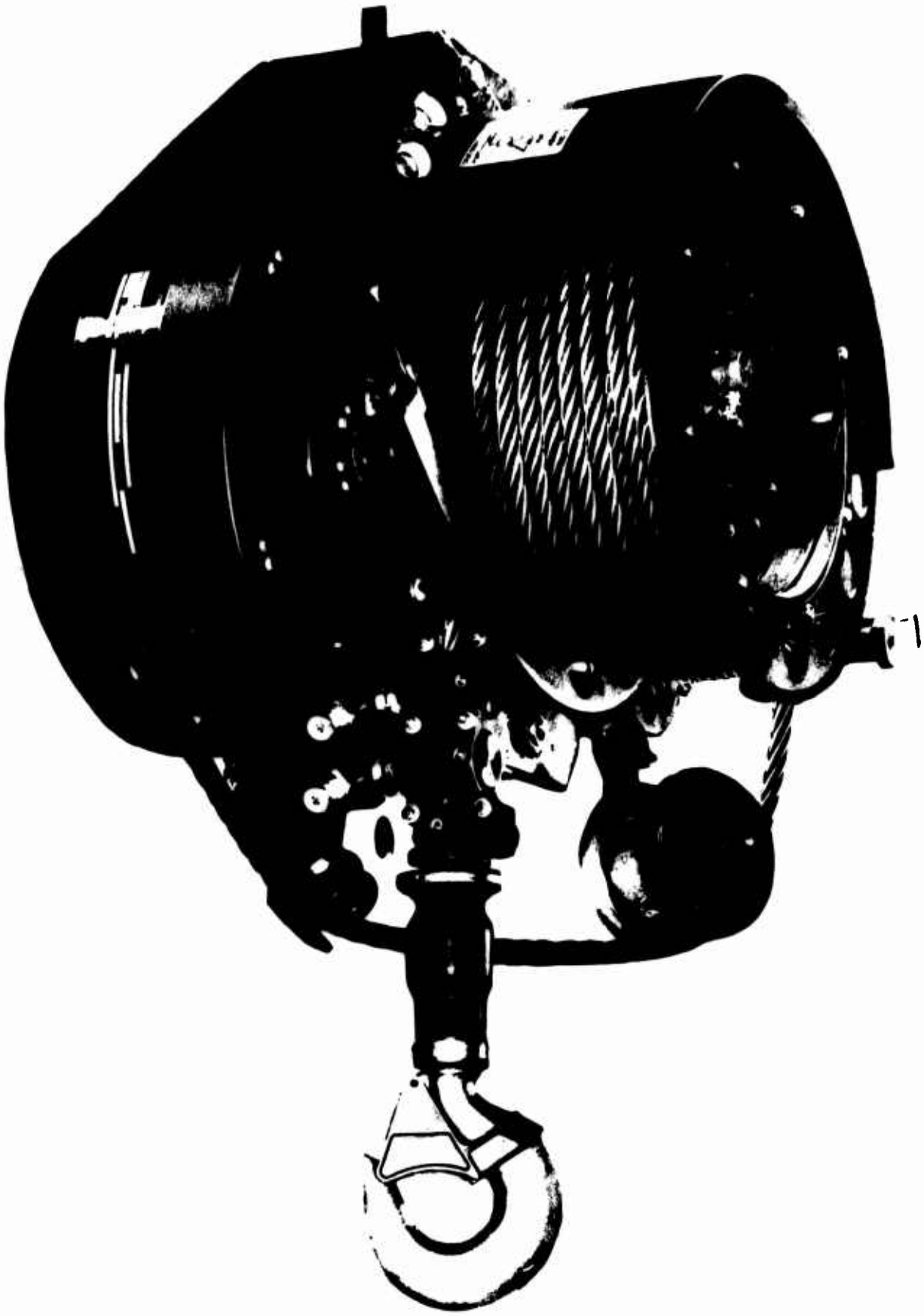


Figure 40. Capstan-Type Winch for a Four-Point Hoist System.

TABLE IV
RELATIVE MERITS OF WINCH TYPES

Item	Weight	Cost	Maintenance	Reliability	Installation
Capstan Type	2	2	2	2	1
Drum Type	1	1	1	1	5

For element definition refer to "Relative Merits of Hoist System Components and Complete Hoist Systems" on page 178.

NOTE: In the above table, in all elements except "installation" the relative degree of merit difference is only in the order of 5 percent, and aircraft installation design may reduce this figure. The flexibility of the capstan-type winch to meet varying installation requirements more than offsets any disadvantages implied in the above table.

Figures 41 through 45 illustrate the various sizes of capstan-type winches which meet the requirements of a 40,000-pound hoist system. The 40,000-pound single-lift winch is illustrated, but this does not imply that such a size is recommended.

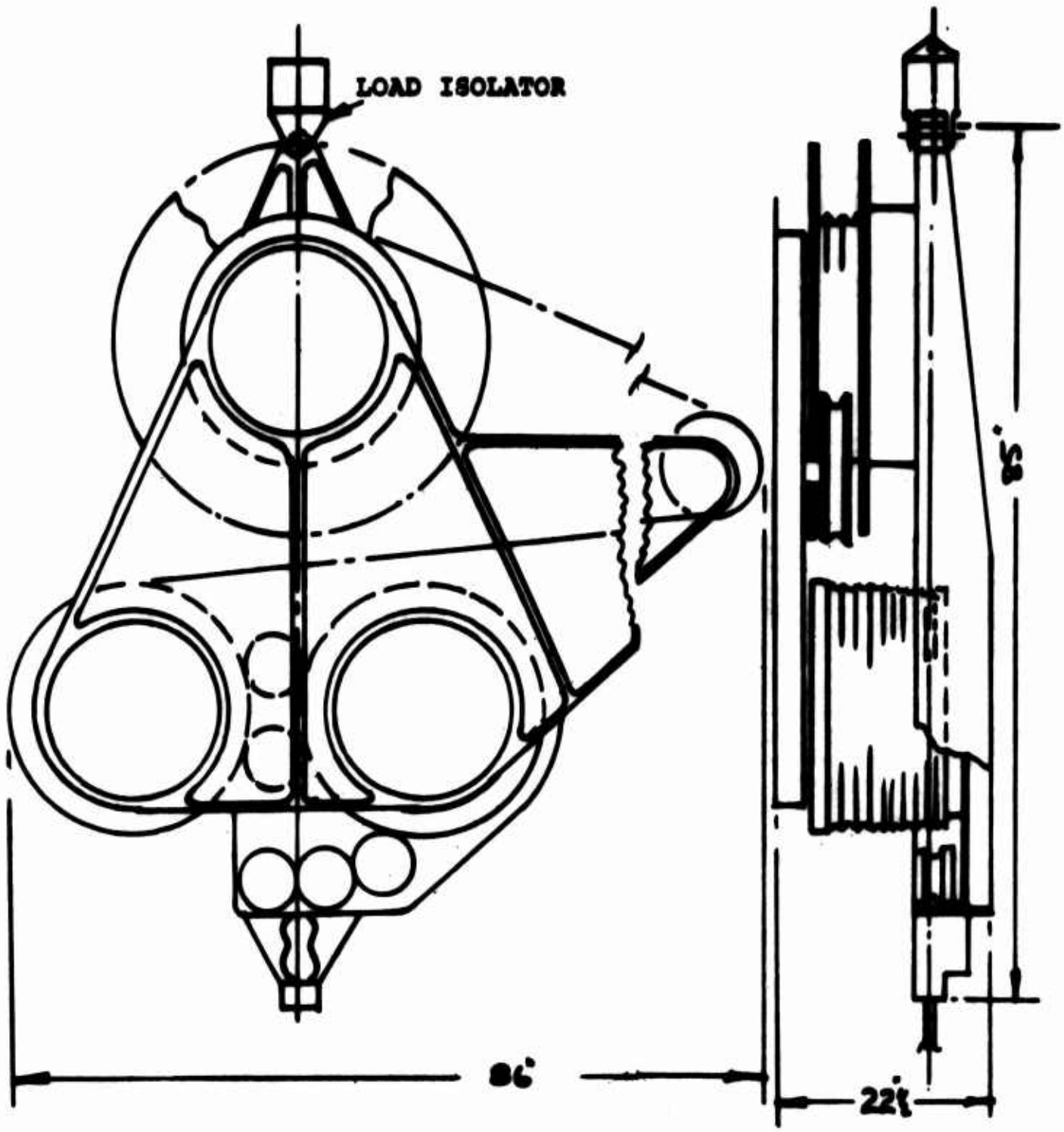
Comparison of Drum-Type with Capstan-Type Winches

Comparison of the characteristics of drum-type and capstan-type winches reveals the following:

1. Both winch types can be effectively built into the proposed 40,000-pound hoist system. More experience exists with the drum type. The greater height of the capstan is a product of satisfying rope criteria, and it could be argued that, since the specified life requirement of 3600 cycles is relatively low, configurations embodying reverse bends could still meet this figure. However, in the absence of any testing of rope in this configuration, such assumptions are not valid. On the other hand, ample testing already performed proves that reverse bending halves rope fatigue life. Attempts to extrapolate these test results (in association with rope manufacturers) have given negative results.

2. The capstan-type has a more complex drive system. This is due to the necessity of driving the storage drum as well as the capstan pulleys.

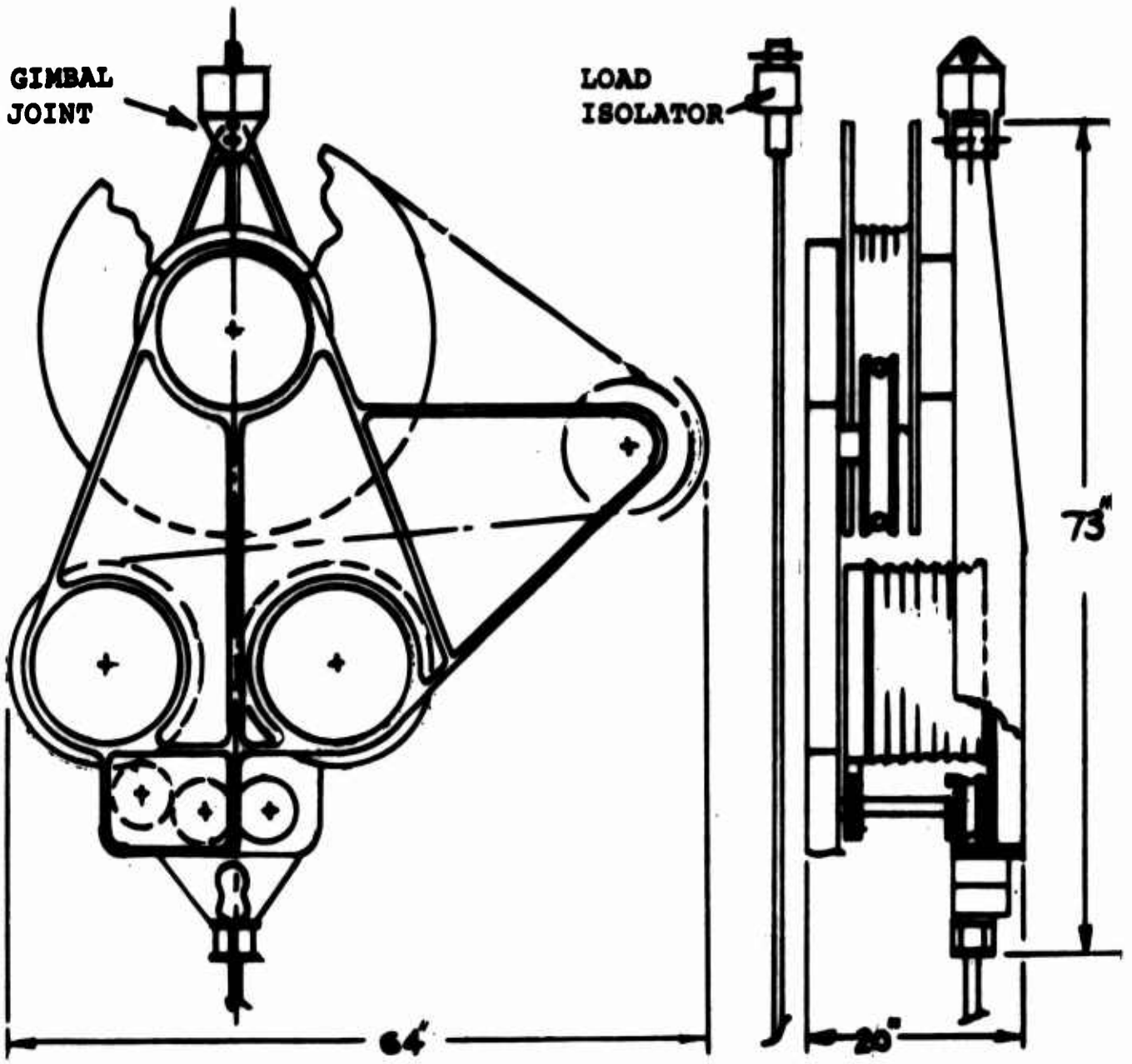
3. The zero-moment feature of capstan-type winches gives them a distinct advantage over the drum types; in installations where this feature is not required (for example, where winches are mechanically coupled for synchronized operation), the winch can be installed on a rigid mounting. It must be appreciated that the configuration for capstan winches shown relate only to a suspended winch. Installation criteria may dictate a design arrangement which will be different from that shown and, providing the drum relationship (i.e., distance between drums and lateral relationship of drums) is maintained, then they may be rearranged to suit. For example, the three drums may be mounted on a common center line and the resultant winch mounted parallel to or recessed in the aircraft floor.



Load 40,000 X 3.75	= 150,000 pounds
+ 6% Bending Factor	= 9,000 pounds
+ Rope Weight (840 x 3.75)	= 3,150 pounds
+ Hook Weight (150 x 3.75)	= 562 pounds
TOTAL DESIGN LOAD	162,712 pounds

Lift Height = 150 feet
Rope Size=1.375india.
Rope Break Strength = 183,400 pounds

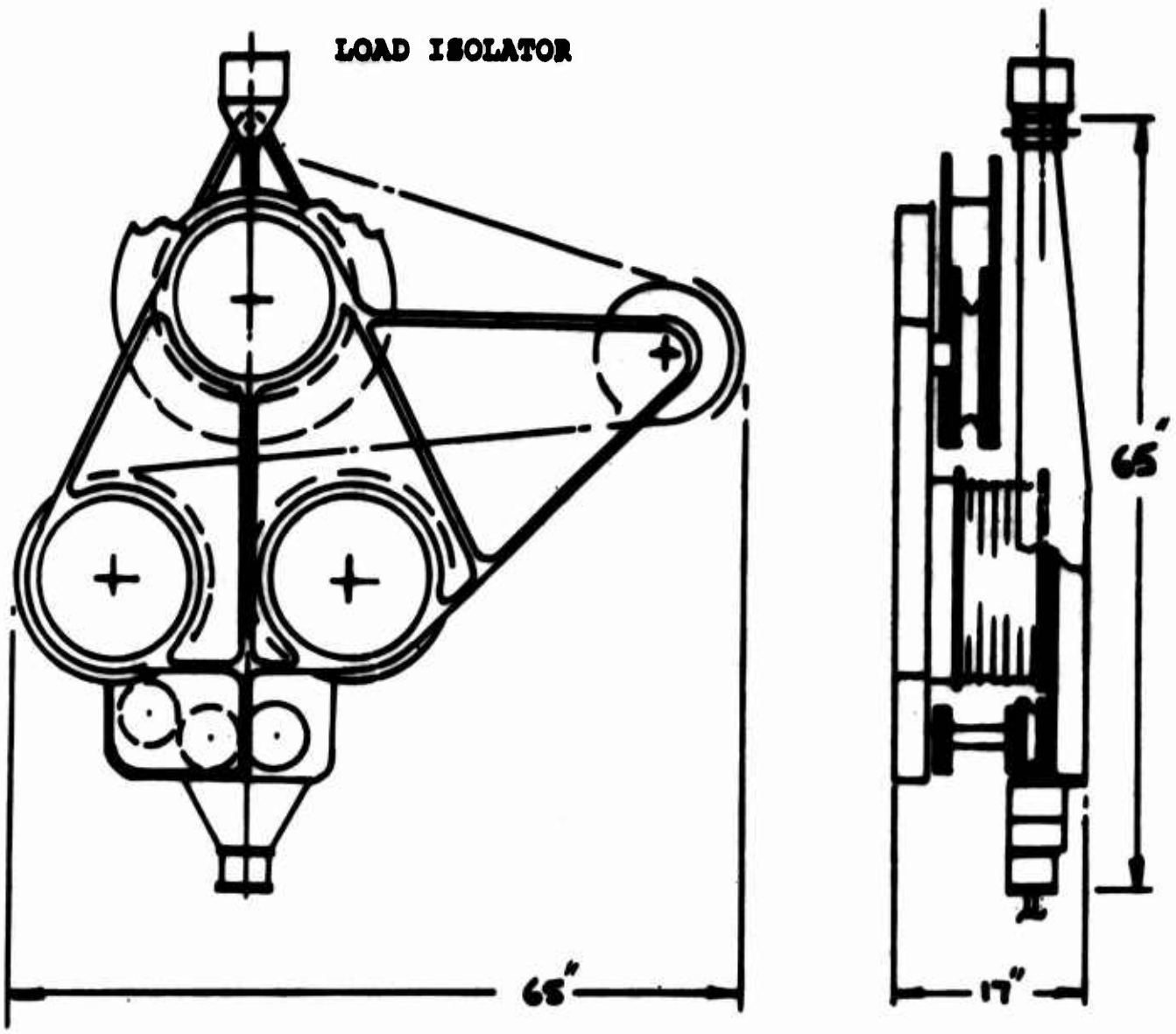
Figure 41. Single-Point, Single-Rope Capstan (150-Foot Lift).



Load 40,000 x 3.75	= 150,000 pounds
+6% Bending Factor	= 9,000 pounds
+ Rope Weight (681 x 3.75)	= 2,554 pounds
+ Hook Weight (150 x 3.75)	= 563 pounds
TOTAL DESIGN LOAD	≥ 162,117 pounds

Lift Height=150' (300 ft Rope)
Rope Size=1.0indiam

Figure 42. Single-Point, Single-Reeved Capstan (150-Foot Lift).

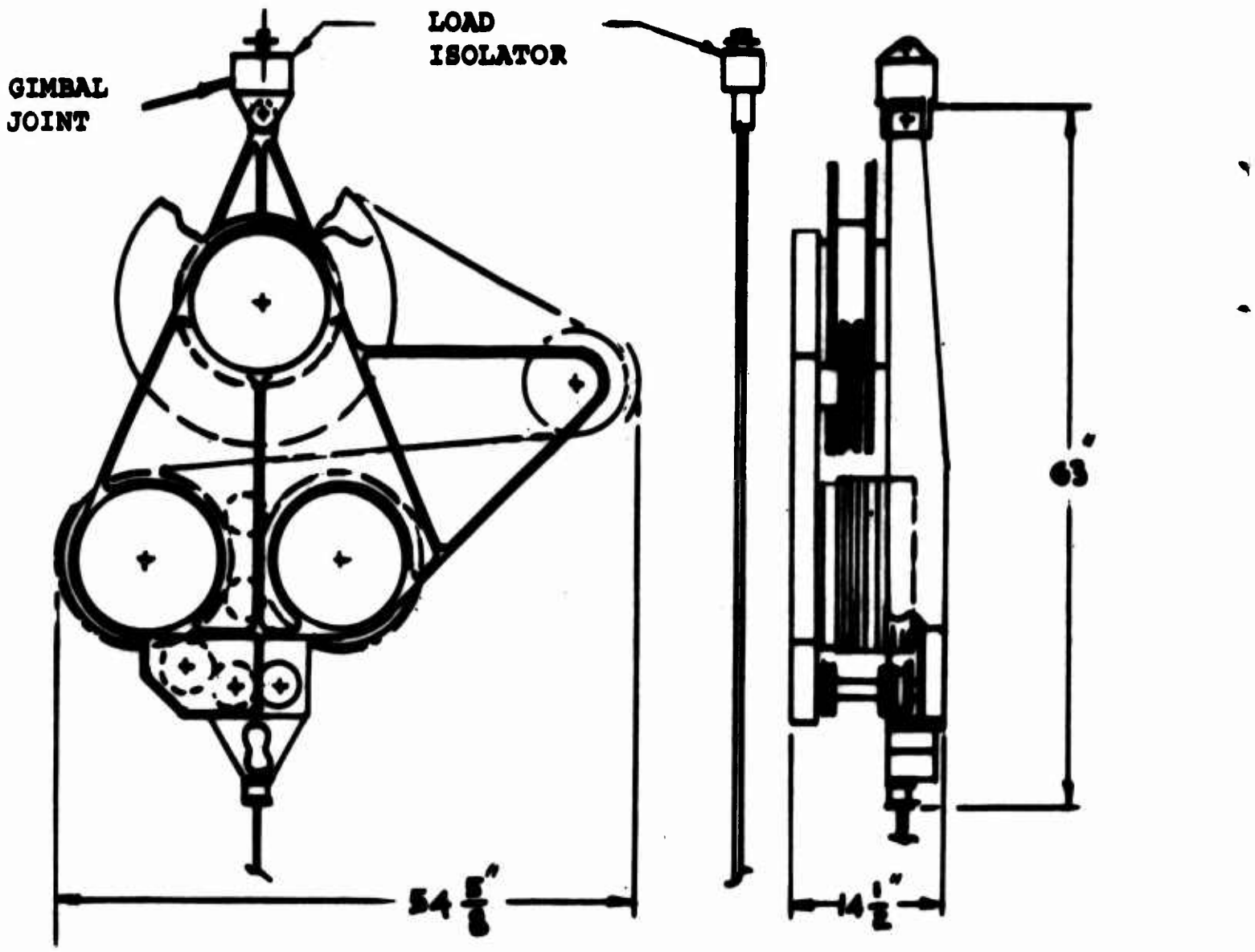


Load per Capstan 28,208 x 3.75	= 105,780	pounds
+6% Bending Factor	= 6,347	"
+Rope Weight (100 x 3.75)	= 375	"
+Hook Weight (40 x 3.75)	= 150	"
Total Design Load	= 112,652	pounds

Lift Height = 50 feet
Rope Dia. = 1.0 in
*Rope Break Strength = 98,200 pounds

*Although design load exceeds the catalog value of 98,200, rope can be supplied 1 in.dia.with a break strength of 117,000 pounds.

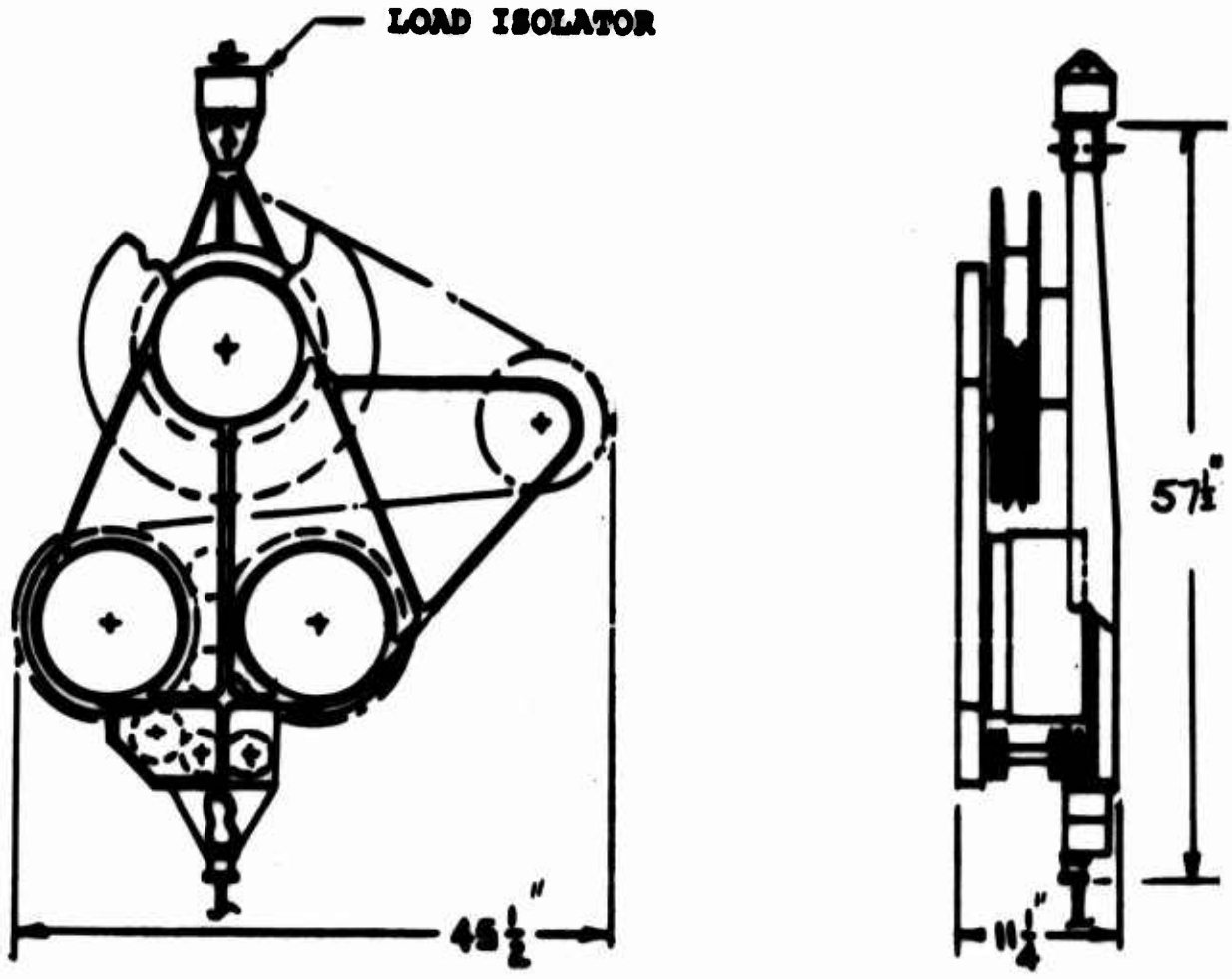
Figure 43. Two-Point Hoist System Using Two Capstans (No Reeves).



Load per Capstan (28,208 x 3.75)	= 105,780 pounds
+ 6% Bending Factor	= 6,347 pounds
+ Rope Weight (222 x 3.75)	= 833
+ Half Beam Assy Wt (300 x 3.75)	= 1,125 Pounds
Total Design Load	= 114,085 Pounds

Lift Height=50 feet (100 ft rope)
Rope Dia=.875 in.
Rope Break Strength=75,760 Pounds

Figure 44. Two-Point Hoist System Using Two Capstans and Single-Reeved Rope.



Load Per Capstan 15,515 x 3.75	=	58,181 pounds
+6% Bending Factor	=	3,490 pounds
+ Rope Weight (92 x 3.75)	=	345 pounds
+ Hook Weight (40 x 3.75)	=	150 pounds
Total Design Load	=	62,166 pounds

Lift Height-50 feet
Rope Size-7/8in. dia.
Rope Break Strength-75,760 pounds

Figure 45. Four-Point Single-Rope Winch.

4. The lack of rope traverse permits precise cg prediction. Even on installations which use a capstan-type winch as a fixed installation, this feature is still an advantage since the sweep of the rope on a drum type winch (fleet angle) in "buried" installations requires structural clearance. Any rope sweep on a capstan winch is performed within the winch envelope and does not influence aircraft structure.

5. Relative life of the drum type appears better than that of the capstan type; however, adherence to rope design criteria in design of the capstan type could eliminate this difference.

6. The capstan type permits more flexibility in varying rope lengths to fit conditions.

Summary of Winch Analysis (Refer to Table IV)

The optimum design for a 20,000/40,000-pound cargo hoisting system for use on a heavy-lift aircraft is most likely to be realized by using a capstan-type winch. An analysis of wire rope criteria reveals that as rope design loads approach 75 tons and thereby dictate rope diameters approaching 1-3/8 inches, these wire rope parameters preclude the application of a one-rope, one-part hoisting system. Perhaps, with the development of new materials, wire rope diameters can be reduced to sizes more compatible with airborne hoisting equipment. It is recommended that wire rope diameters for use on optimum-design aircraft cargo hoisting systems be limited to the 7/8- to 1-inch range. It is suggested that reeved systems be utilized. This would reduce the breaking strength requirements of the wire rope and thus permit the use of rope diameters not exceeding 1 inch.

THE USE OF TAPE AS AN ALTERNATIVE TO WIRE ROPE

Although wire rope is still under a continuous process of development, it can be argued that it is approaching a development limit, whereas tape technology is in its infancy. The material development upon which tape technology is dependent has shown very rapid improvements over a short time span. This positive development trend has led to consideration of the use of tapes for hoist system application.

The use of tape in place of rope in the intended application to HLH hoist systems shows promise of evolving a system which would have simpler hardware and hardware which would be more compatible with installation on helicopters than conventional wire rope systems.

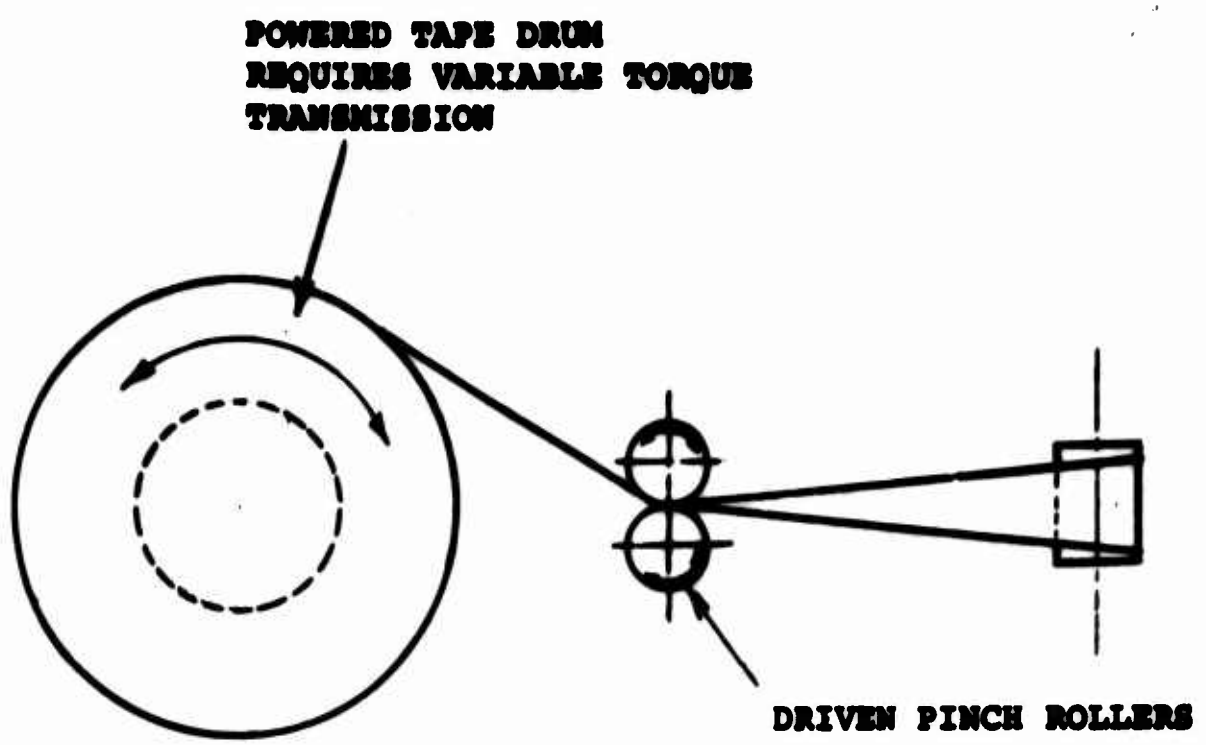
Influence On Winch Design

Wire rope wound around drums requires the use of a level-wind mechanism to ensure that the cable traverses the drum without the rope overlapping. It is also desirable practice in airborne winches to have only one layer of rope on the drum, in order to keep crushing forces from causing cable damage. This condition results in a long drum (dependent upon cable length), especially in reeved hoist systems. Unless capstan-type winches are used, the traverse movement of the cable affects the moment of the suspended load relative to the cg of the aircraft; this is an undesirable feature.

Tapes can be wound upon themselves with no traverse; this results in a drum width little greater than the tape width (see Figure 46).

Drum diameter, when wire rope is used, is dictated by rope diameter, i.e., bending stiffness of the rope. Tape, by its nature, can be bent around smaller diameters; however, tape which wraps upon itself will result in drums having overall flange diameters greater than for rope drums (dependent on tape length), but any auxiliary pulleys will be relatively smaller.

Another feature of tape is that it can be designed into systems where change of direction is desirable to obtain the optimum system installation. Since tape twists of 90 degrees are allowable, the relatively narrow drums could be mounted flat within the confines of the helicopter floor structure



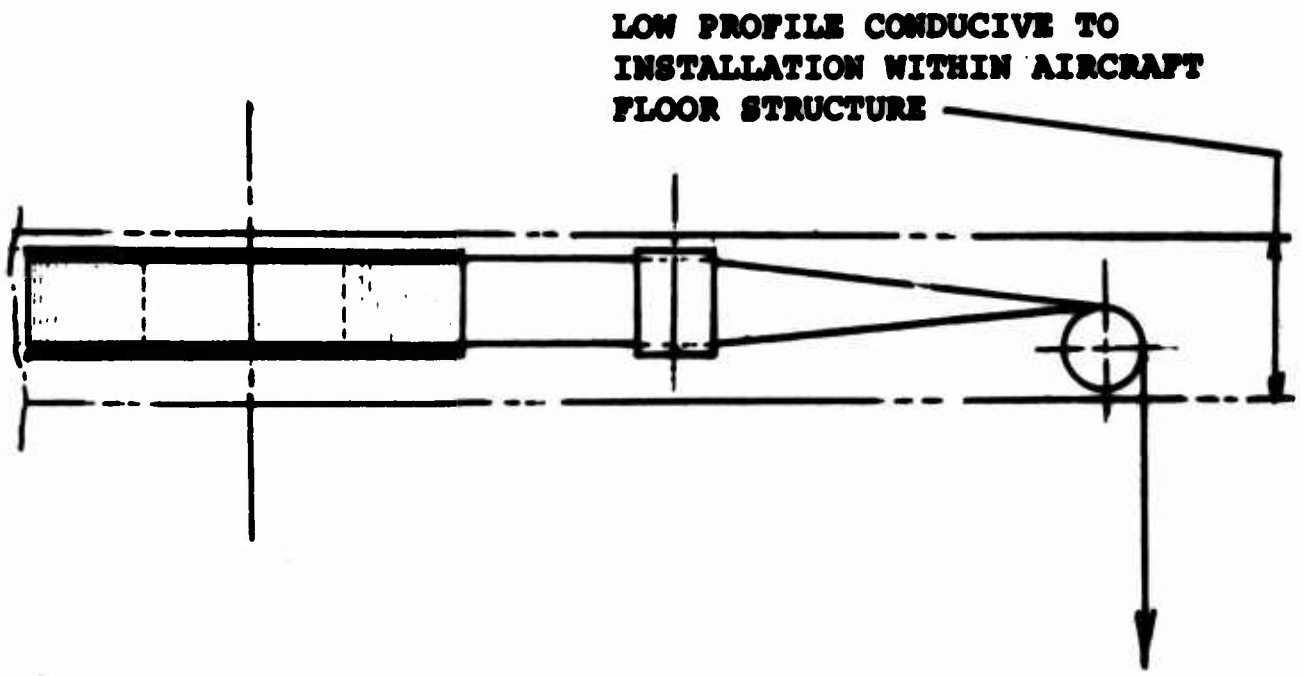


Figure 46. Tape Winch.

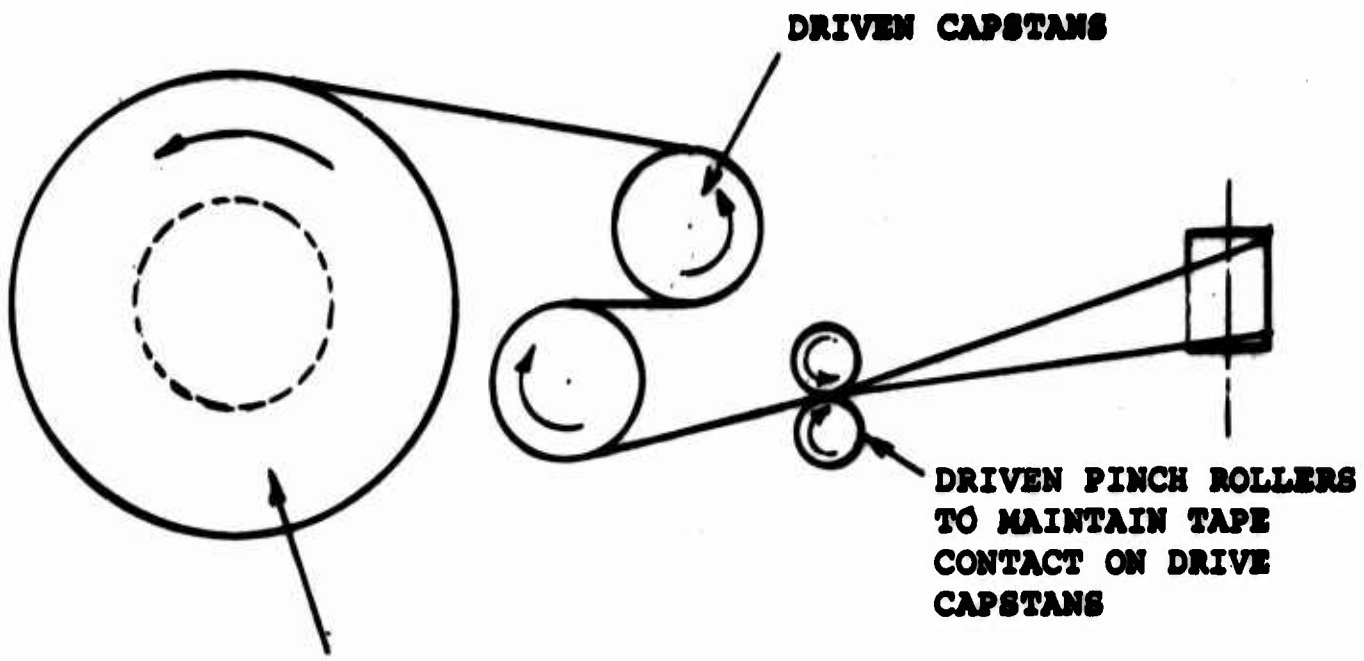
(see Figures 46 through 48).

The drive system required for a tape winch differs from that of a single-layer drum-type rope winch in that a variable torque drive is required. This is due to the fact that the tape progressively builds up on the drum. This will result in a heavier drive system since it must be designed for the fully wound drum case (i.e., maximum radius), as shown in Figure 46. The use of variable-displacement hydraulic motors to give variable torque requirement could, however, be designed out of the winch by taking the drive through capstan pulleys (Figure 47). This again will add weight, and the end result might well prove to be a weight increase greater than that required for a variable-torque drive.

One other unknown factor which must be considered is that when the tape is progressively wound onto its drum under load, the load may cause unacceptable distortion of the tape by crushing. The employment of a capstan drive system would solve this problem, since the tape on the drum would then be only lightly loaded.

Conclusions - Tape

The use of tape in place of rope shows promise of evolving a hoist system for the future HLH. Only limited testing of tape for winch applications has taken place to date, and it is recommended that a complete review of tape technology be authorized with the object of laying the foundation for a qualification test program which will establish the limiting design criteria for tape application.



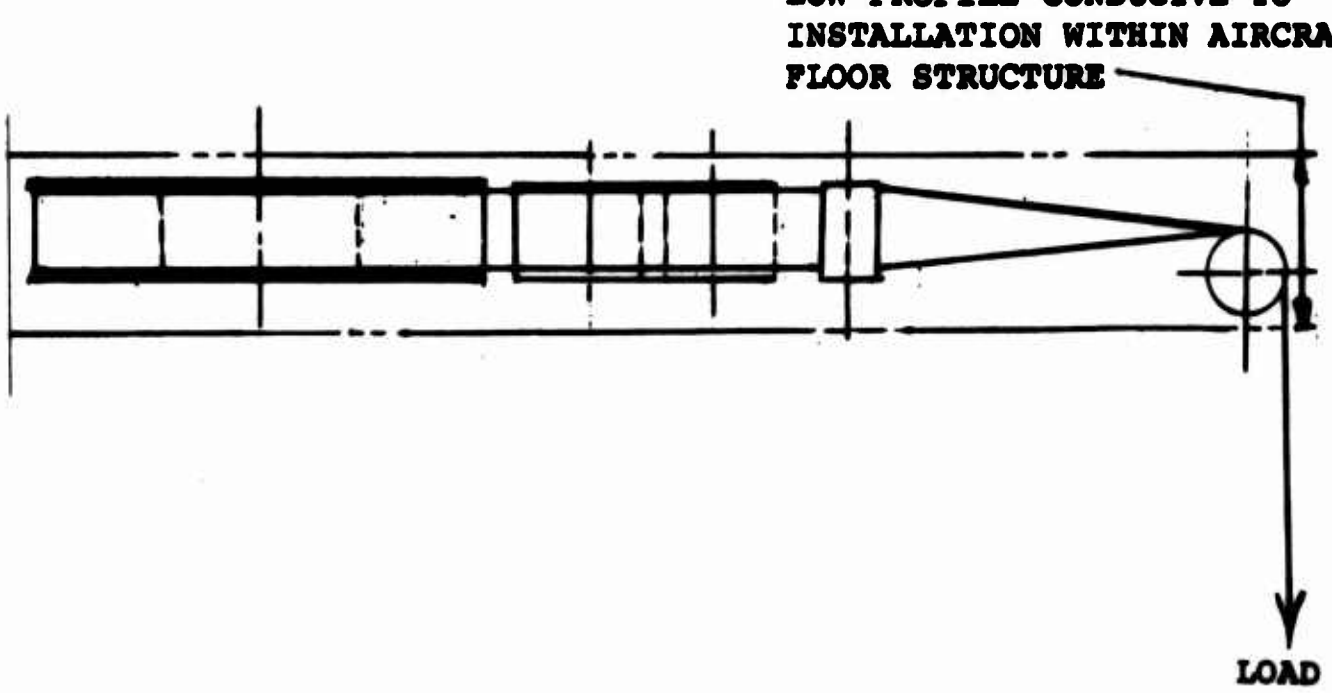


Figure 47. Capstan-Type Tape Winch.

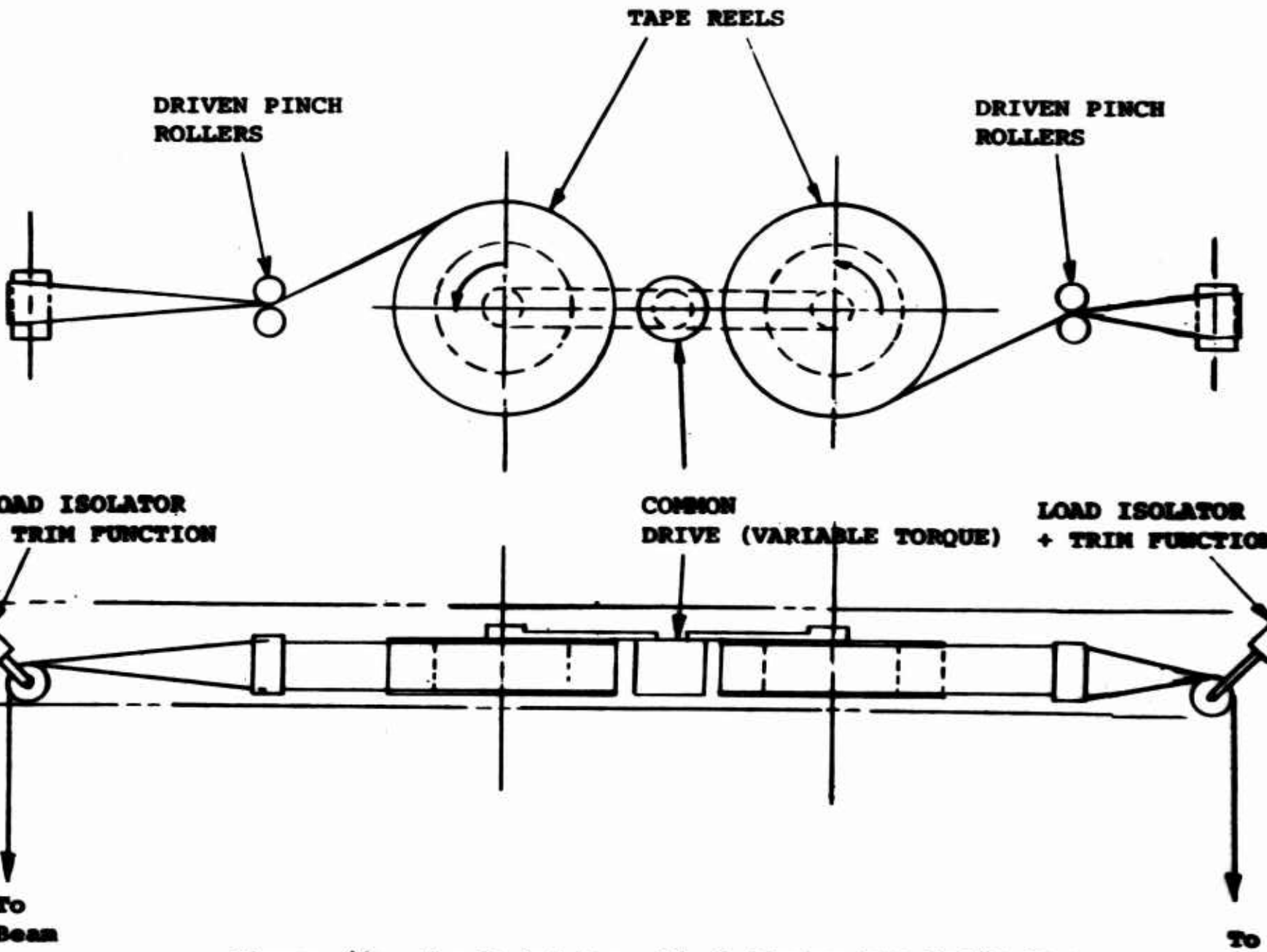


Figure 48. Two-Point Tape Winch Having Two Reels on Common Variable-Torque Drive.

SUSPENSION METHODS FOR EXTERNAL LOADS

Single-Point Suspension

Single-point suspension (see Figure 49) is the simplest and certainly the commonest way of carrying external loads. Single-point suspension systems can be divided into two different types: one incorporating some torsional restraint of the load, and the other having a swivel intentionally introduced into the system. Torsional restraint of the load is achieved by attaching the load sling to the cargo hook through a large ring. The ring serves to provide a base for the sling, as shown in Figure 50, and reacts the torque into the cargo hook in a manner similar to the torque reaction of two interlocking rings under tension. If the ring was not used, the sling legs would twist on themselves, thus limiting the centering action to the load and possibly damaging the sling.

Occasionally it is necessary to lift a load on a very long sling, or a very long load on a short sling. In either case, when the torsional restoring moment afforded to the load is insufficient to guarantee that rotation will not occur, it is common practice to install a swivel into the system to minimize the risk of damage to the sling. It is also necessary to install a swivel on systems that use a single cable or pendant between the sling and helicopter, because a torsional load applied to the cable reduces the ultimate tensile load by changing the distribution of load between individual stands. A hook equipped with such a swivel is shown in Figure 51.

If it is necessary to carry aerodynamically unstable loads on long suspensions, the load must be either aerodynamically stabilized or torsionally restrained. An example of an aerodynamically stabilized load is given in Figure 52, which shows a CH-47A transporting a damaged CH-47A fuselage, with the load stabilized by use of a drag parachute. Stabilization of the load by drag parachute takes time and increases the total drag of the load. It is not, therefore, a suitable technique for use in the transportation of tactical loads, where aircraft productivity is important. Loads may be stabilized with only a small drag penalty by the addition of a vertical stabilizer surface, but this technique is unwieldy due to the size and difficulty of attaching a surface of sufficient size.

A major factor in determining sling and suspension length is the phenomenon known as vertical bounce. Vertical bounce is

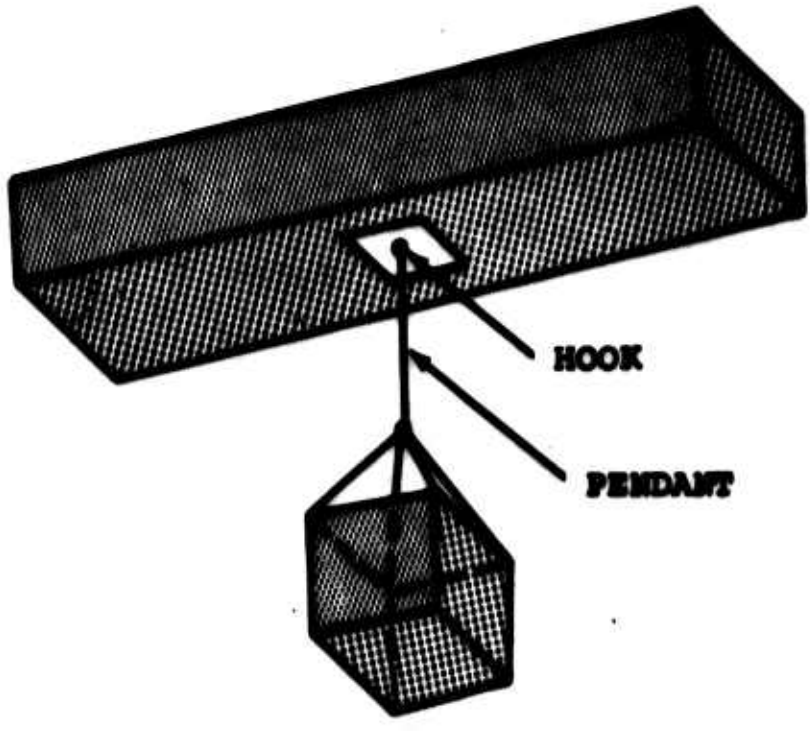


Figure 49. Single-Point Suspension.

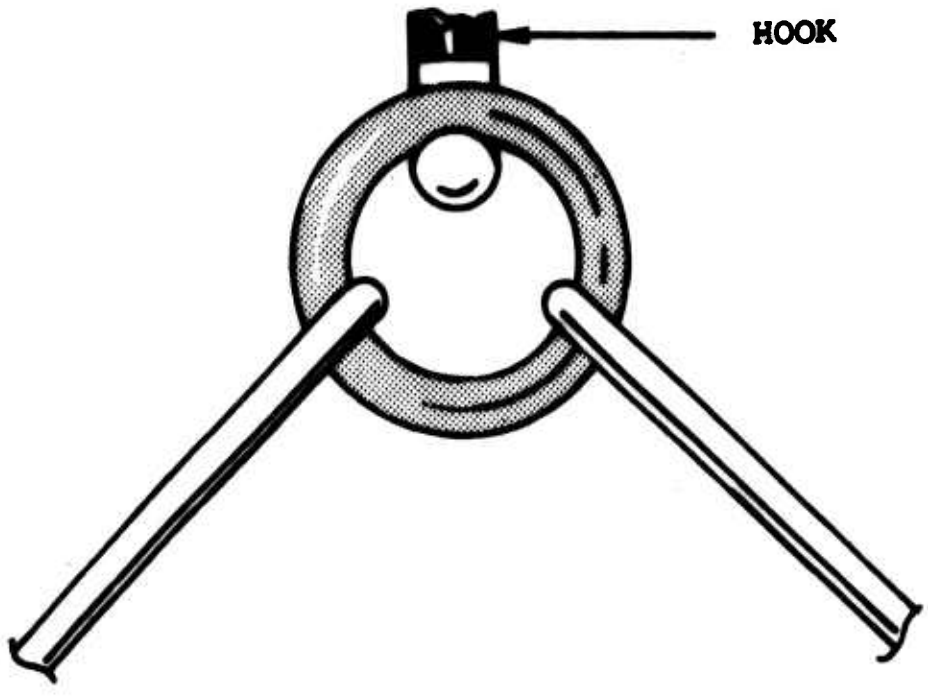


Figure 50. Cargo Ring.

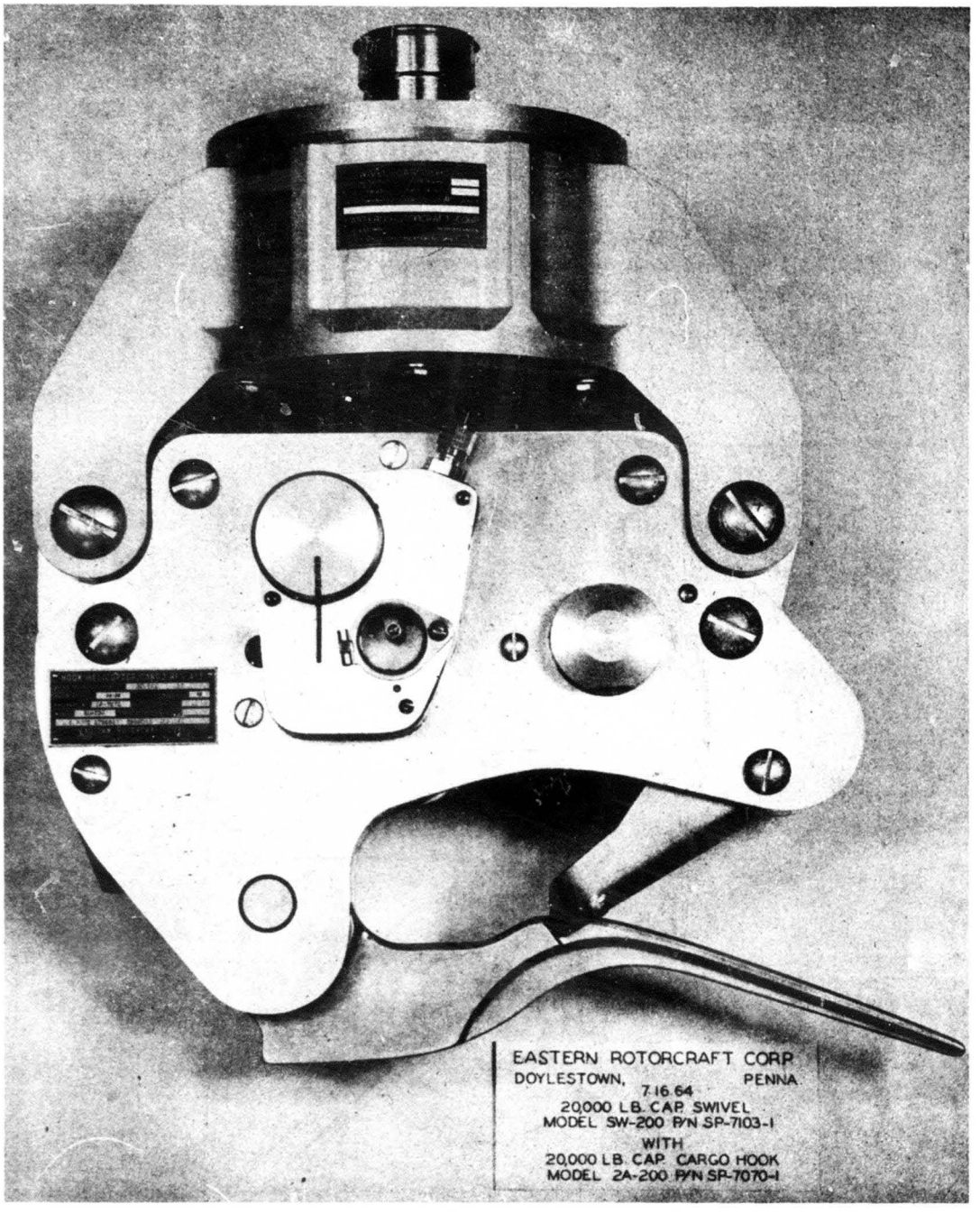


Figure 51. Hook with Swivel.

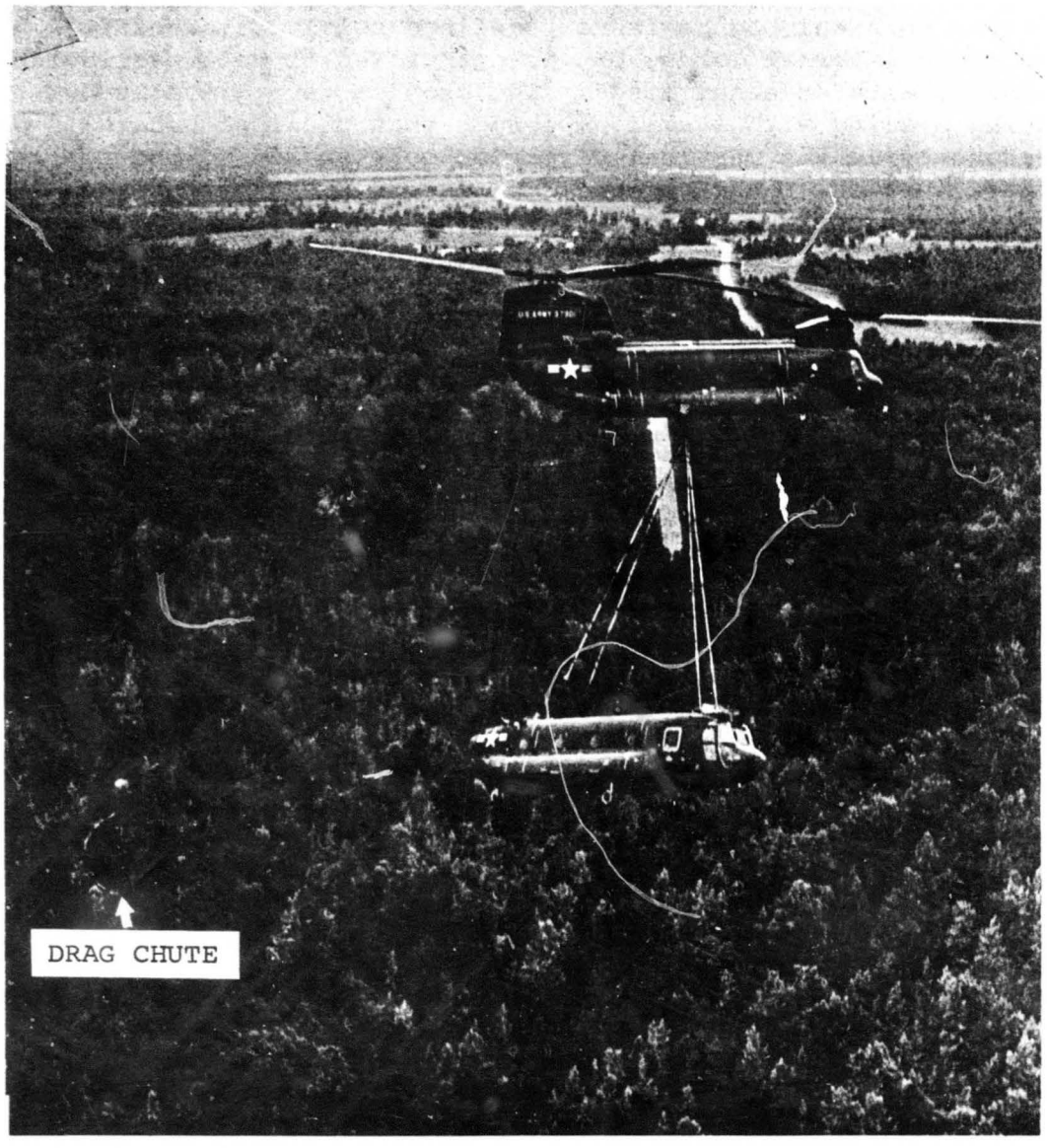


Figure 52. Use of a Drag Chute to Stabilize a Load.

a vertical vibratory motion of the fuselage and load at approximately 3 to 4 cycles per second; this may increase to such intensity as to cause the pilot to jettison the external load. Vertical bounce is not confined solely to aircraft carrying external loads, but it has proved to be a particular problem with external loads. The problem is aggravated by using a sling with a stiffness such that the load natural frequency in the vertical direction corresponds to the aircraft vertical bounce frequency. In the past, the problem has been minimized by selecting sling materials and lengths that cause the load bounce frequency to mismatch with the fuselage bounce frequency. Army experience with the CH-47A has indicated that, when using steel rope , suspension bounce becomes a problem over a wide range of lengths. This finding resulted in the recommendation that only nylon slings be used.

Another approach to the solution of the vertical bounce problem was taken on the CH-54A single-point hoist system. Since it was necessary to operate with variable-length suspension, it was not possible to guarantee that the suspension frequency would not match the bounce frequency. The solution was to fit a soft spring in series with the hoist; this maintained a low spring rate over the entire hoist travel. This system, known as a load isolator, has minimized the problem. Vertical bounce has not been encountered by the Army CH-54A pilots; it is important to note, however, that these pilots are all very experienced, and they feel that the system may be unacceptable to an inexperienced pilot. Vertical bounce and load isolators are discussed in detail in Reference 1 in this document.

Multi-Point Suspension

The advantage of a multi-point suspension is in the stability that it affords to the load. This type of suspension was introduced on the CH-54A crane-type helicopter shown in Figure 53. The system on the CH-54A has four hard points to which a load leveling and lifting system is fitted. The system assembly consists of four servo cylinders mounted in pairs to stationary fittings on each side of the aircraft. In this design, the lower ends of the servo cylinders are attached to a hinged beam assembly to which cargo lashing reels are mounted. Actuation of the servo cylinders raises or lowers the beam assemblies and the attached cargo lashing reels,

Figure 53. Multi-Point Suspension of Armored Vehicle.

thus raising or lowering any attached load. Each cargo lashing reel has a mechanical lock. The four-point system was designed to be compatible with a removable pod, the four points being available for suspending cargo when the pod is not fitted (see Figure 54).

In practice, it is difficult to realize the benefits of multi-point suspension with the CH-54A four-point system. The four-point suspension system is structurally redundant because of the near impossibility of achieving equal loading in the legs. Unequal loading results in one leg being loaded substantially less than the other three legs, or worse, in one leg being slack. The CH-54A system has the capability of "beeping" small extensions of the legs individually by means of the servo cylinders, but precise control cannot be maintained. Furthermore, a suspension that has been statically adjusted is not necessarily in balance in flight. The result of uneven loading is manifested as a coupling of the load motion in flight. That is to say, when the load moves aft due to aerodynamic drag forces, it may also move laterally and yaw. Such a load is difficult to control, since the pilot must make simultaneous corrections at different rates in all axes.

It is necessary to examine the requirements of load aerodynamic stability in order to determine the necessary features of a suspension system for stable high-speed flight. The discussion in the section entitled "STABILITY OF SLUNG LOADS" located in Reference 1 of this document indicates that the primary restraining modes required from a suspension system are yaw and pitch.

The single-point suspension, discussed earlier in this section, offers only limited yaw and pitch restraint. Four-point suspension gives greater yaw and pitch restraint, together with increased roll restraint. Three-point suspension gives comparable yaw restraint without structural redundancy. Two-point suspension, oriented in a fore and aft fashion as shown in Figure 55, provides the desired pitch and yaw restraint, and is structurally statically determinate.

Multi-point suspensions offer advantages in load stability, but incur problems in load release. Each leg of a multi-point suspension is equipped with a cargo hook. During release of the load, should any one hook fail to open and the pilot attempt to take off, the aircraft could be lost. Figure 56 illustrates the upsetting moment that would be created in

Figure 54. Four-Point Suspension Cargo-Lashing Reels and Leveling System.

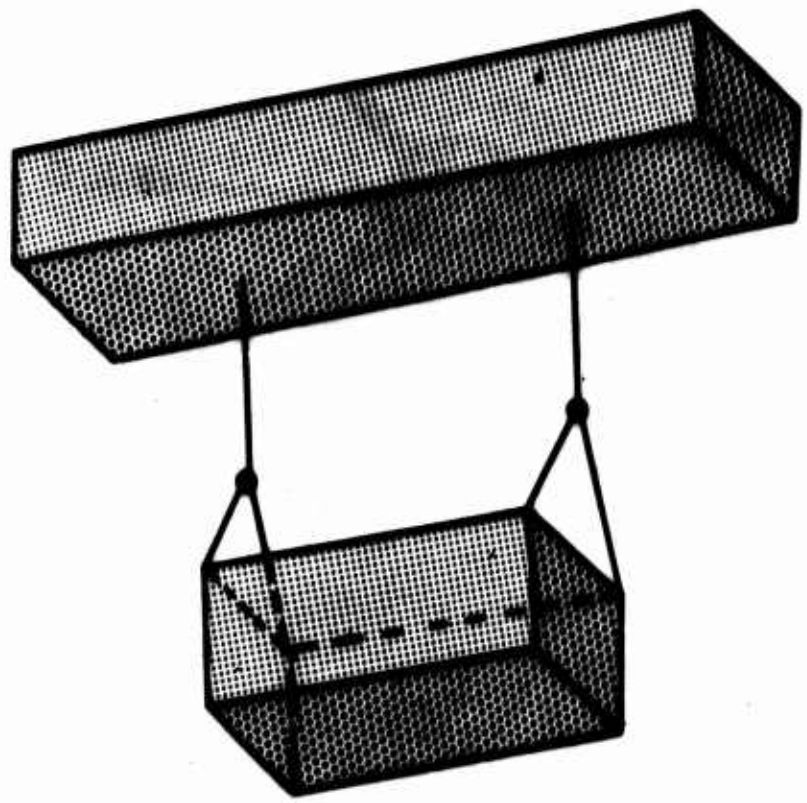


Figure 55. Simple Two-Point Suspension.

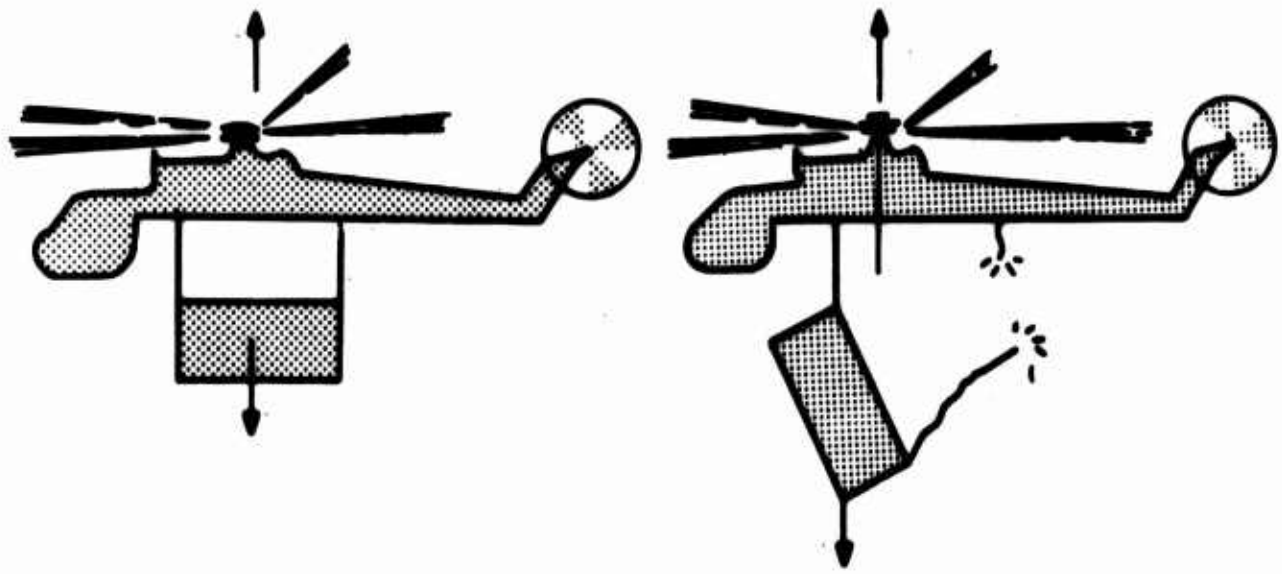


Figure 56. Effect of Partial Failure on Multi-Point Suspension.

such an eventuality. It might be argued that the likelihood of this happening is extremely rare. However, if a hook should become hung up, the pilot would probably not notice it until he had pulled away from the load. At that time he would sense the upsetting moment, but normal correcting procedures might well compound the problem; such an experience has occurred with the original cargo sling on an H-21 helicopter.

Experience with current cargo hooks has shown that these are not 100-percent reliable in releasing the load on command; hence, they are fitted with secondary release mechanisms. A warning system could be devised to indicate when one hook is not released; but if the aircraft is hovering while releasing the load, a gust could raise the aircraft sufficiently to cause the suspension line to pick up the load with the consequence described above. Therefore, without a technique for the positive coupling of hook release mechanisms, multi-point suspensions constitute a safety hazard. A two-point suspension with a connecting beam, as shown in Figure 57, shows promise for a practical solution to this problem.

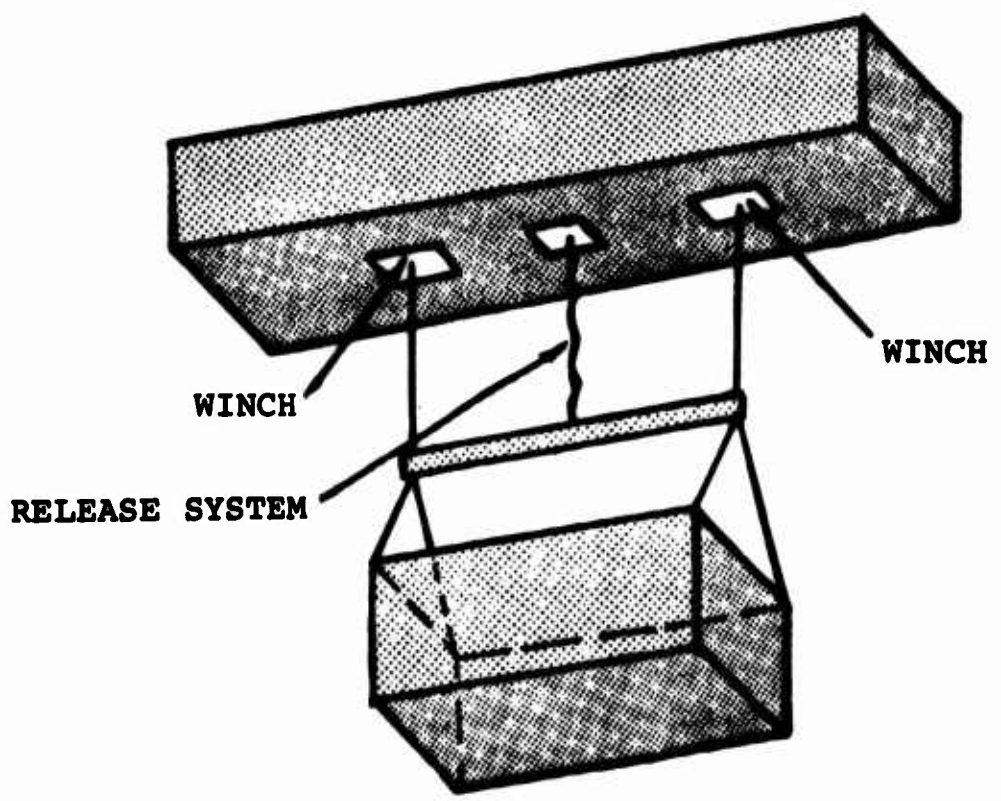


Figure 57. Two-Point Suspension with Connecting Beam.

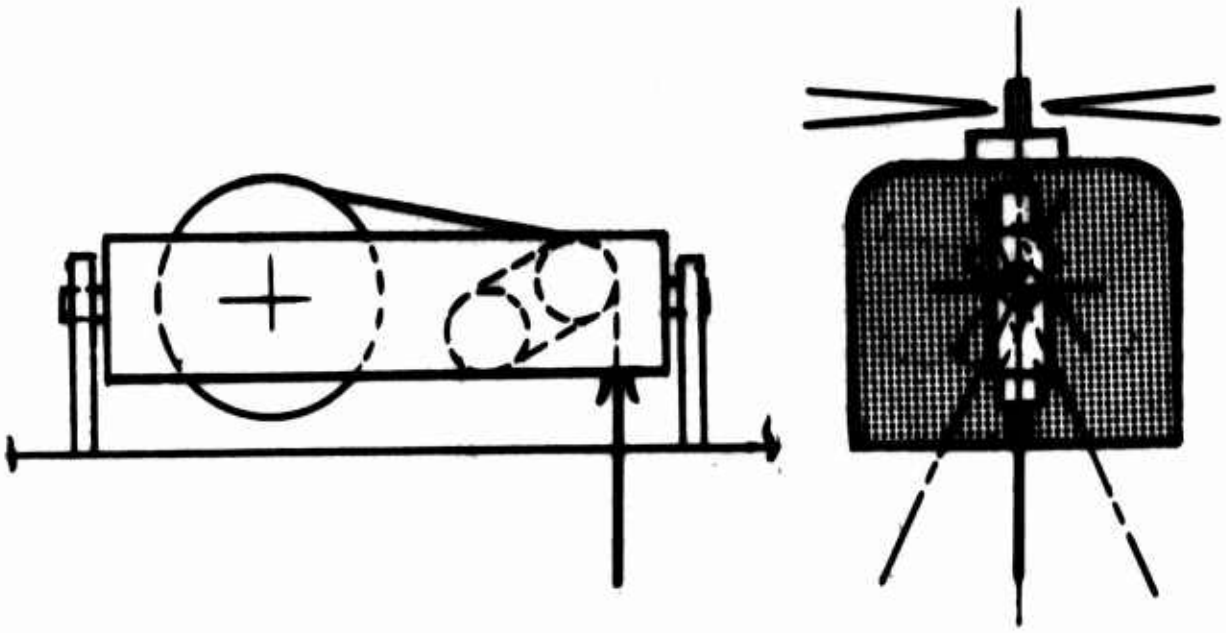
It has been said that the only difference between internal and external loads is that it is possible to jettison external loads. It could be further argued that emergency release is not necessary, but experience has shown many times that dangerous conditions can result from load instabilities. Therefore, an emergency release capability must be included in any load suspension system that does not rigidly attach the load to the aircraft (such as a pod or platform).

As in the case of the pilot-controlled electrical load release, a two-point suspension with a connecting beam appears to offer the simplest means of positively coupling emergency release mechanisms.

The requirements for ground personnel to attach a load depend on the method of load acquisition. If the pickup is made with the helicopter on the ground, one man may connect all the hooks. However, if a hovering pickup is made, one man is required to ensure attachment of each hook, in order to prevent the pilot from inadvertently picking up a partially attached load, which could result in loss of the aircraft. In general, the availability of a multi-point suspension will not lessen the need to sling the load, since the majority of loads are not equipped with compatible pickups points. It is not usually possible to provide such pickup points, because of the diverse shapes of the loads.

Lateral swing of multi-point suspended loads creates the same aircraft rolling moment experienced with single-point suspension. Consequently, unless it is possible to attach the suspension at a waterline passing through the center of gravity (as in the CH-54A), a curved beam (as used in the CH-47A) or longitudinally pivoted winch should be fitted (Figure 58). Curved-beam mechanisms are not feasible for four-point suspensions; such systems are not, therefore, suitable for mounting on the bottom of a transport helicopter. Two-point suspensions may be fitted to curved beams in exactly the same manner as single-point suspension systems (see Figure 59).

A helicopter equipped with a two-point suspension or hoist system is able to operate in a single-point mode by connecting both legs as shown in Figure 60. The angularity of the legs increases leg loading and is contained within the known factor for uneven loading. The aforementioned technique actually provides a single-point system which is an improvement over a system using a single pendant, because the tor-



Longitudinally pivoted winch.

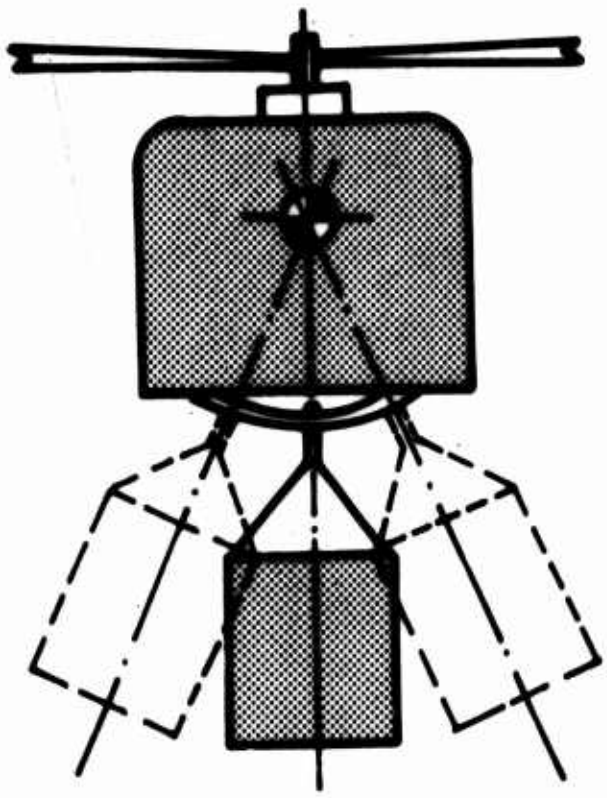


Figure 58. CH-47A Cargo Beam.

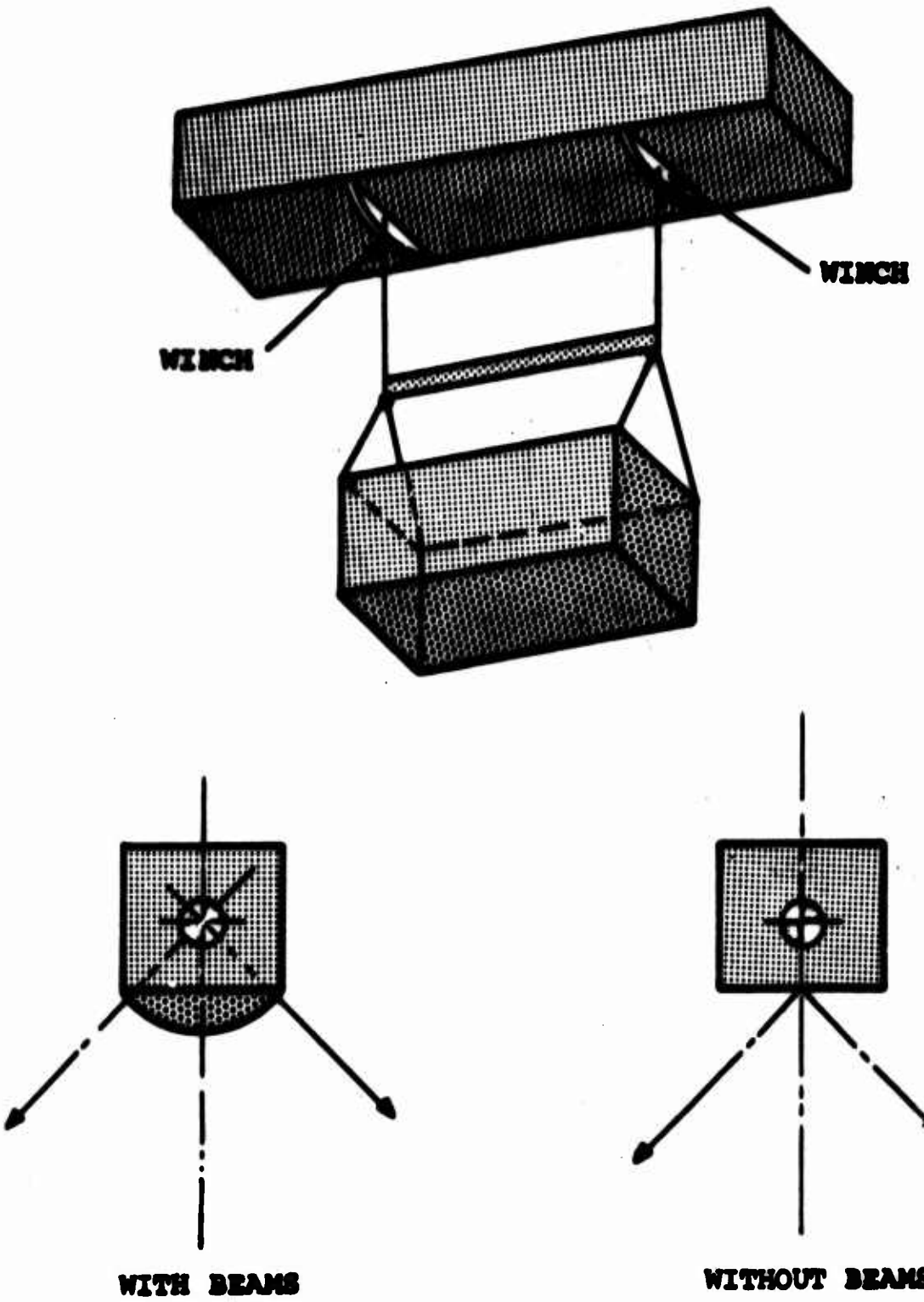


Figure 59. Two-Point Suspension System with Curved Lateral Beams.

sional restoring moment afforded the load is increased.

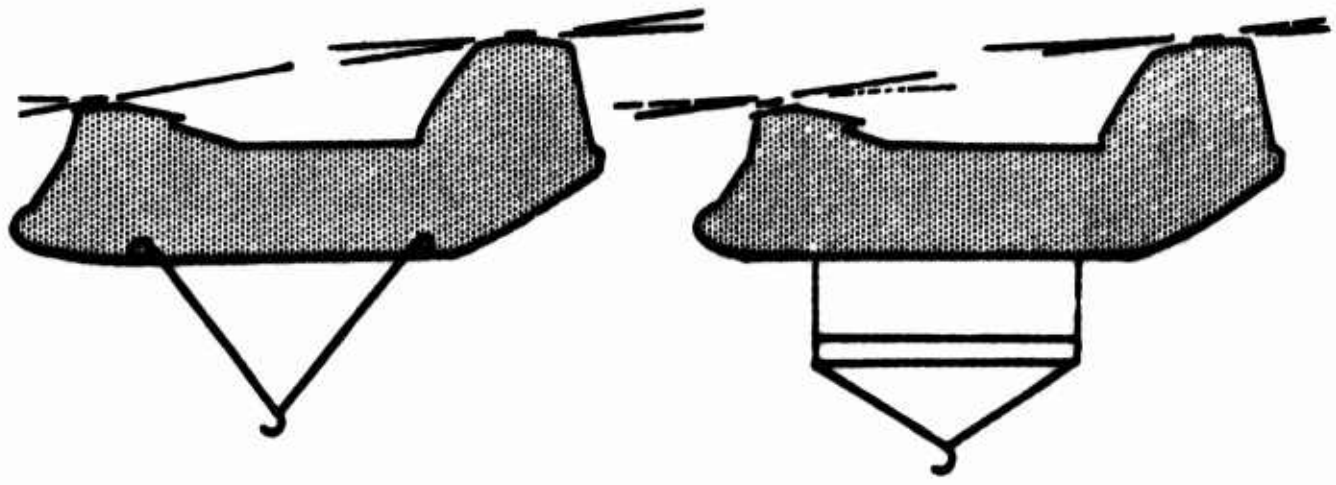


Figure 60. Single-Point Lift from Two-Point System.

Sling Load Isolators

One of the methods which is available for dealing with the problem of sling load vertical bounce is the concept called the load isolator. Schematically, the system consists of a soft spring placed in series with the external load sling, between the helicopter and the external load sling. Natural frequency of this spring is designed to be well below the once-per-rotor revolution bounce excitation frequency of the helicopter. This dynamically soft link in the external

load system prevents the external load/sling combination natural frequency from ever coinciding with the rotor excitation frequency.

This can be seen by considering the equation for the resultant spring rate of a series spring combination

where

K_R = Resultant spring rate (lb/in.),

K_1 = Load isolator spring rate,

K_2 = Sling spring rate.

When a load isolator is used, K_1 is essentially a constant. The spring rate of the sling (K_2) is a variable and can be any value. The resultant spring rate (K_R) then can have a range of

$$0 < K_R < K_1 .$$

Using only the linear theory, the relationship between spring rate and natural frequency is

$$\omega = \sqrt{\frac{K_R}{m}}$$

where

ω = Natural frequency (rad/sec),

K_R = Resultant spring rate (lb/in.),

m = External load mass $\left(\frac{\text{lb-sec}^2}{\text{in.}}\right)$.

Natural frequency is thus proportional to spring rate and will vary in a similar manner:

$$0 < \omega_N < \omega_I$$

where

ω_N = Resultant natural frequency (rad/sec),

ω_I = Load isolator natural frequency (rad/sec).

This shows that the resultant natural frequency of any external load and sling in series with a load isolator will always be less than, or at most equal to, the natural frequency of the load isolator alone.

The above analysis indicates the way in which a load isolator operates. However, there are many considerations in defining an operationally acceptable system, such as:

1. Configuration - The load isolator must be capable of supporting the full static and dynamic weight of the external load, in addition to possessing the required low natural frequency. Obtaining the proper natural frequency is a challenge, since most helicopters required a load isolator natural frequency of between two and four cycles per second over a wide temperature range and with adequate damping. At present, both pure liquid springs and air springs are being investigated as means of providing the required characteristics.

2. Adaptation - Of necessity, the load isolator becomes an integral part of the helicopter. This is dictated by the requirement that it be in series with the external load and yet not interfere with the hook release system.

3. Weight Penalty - Present load isolator designs with capacities up to 25,000 pounds weigh between 50 and 150 pounds. The addition of a mass of this magnitude to the weight of a helicopter must be completely justified, particularly if it is carried during missions where external cargo is not transported.

Load isolators have the potential of becoming the complete solution to sling load vertical bounce. It is premature at present to predict the success which will attend efforts to convert theory to reasonable hardware, to meet reliability requirements, and to overcome resistance to the idea of adding any more heavy accessories to helicopters.

Hydropneumatic Load Isolators

Figure 61 illustrates a self-contained hydropneumatic spring for damping the vertical bounce of the load suspension cable on the Boeing-Vertol CH-47 helicopter. Similar devices can be effectively designed into the future heavy-lift systems discussed in this report.

A hydropneumatic system was chosen since only this type of unit can meet the self-contained requirements of the spring rate and temperature change specifications. The major factor in using gas is the fact that the change curve for gas pressure versus temperature is 30 times better than the curve for liquid pressure versus temperature. In fact, the problem of the liquid's contraction or expansion in response to temperature changes is so critical as to render the liquid spring concept impracticable. Another point in favor of using gas is the consideration that the gas system operates at a much lower pressure than a liquid system. This increases reliability and makes for an extremely serviceable unit.

The hydropneumatic spring concept makes use of the compressibility of gas to obtain the spring forces and employs the liquid for damping control. The gas and liquid are separated in the accumulator by a floating aluminum separator piston. The unit works as a tension spring, with the preload force obtained by the gas pressure acting on the liquid through the separator piston which, in turn, acts against the spring force piston area. This area is the difference between the cylinder body inner diameter and the piston rod outer diameter. As the piston rod is extended, the liquid is pushed back into the accumulator, thereby compressing the gas. Different spring rates can be obtained by varying the gas pressure and the liquid volume. Figures 62 and 63 illustrate the minimum and maximum spring rates that can be obtained. However, after initial testing, it may be possible to increase the maximum spring rate by an additional 15 percent. An orifice disc inserted between two flanges will provide a true damping calibration. Different-size orifice plates can be utilized for

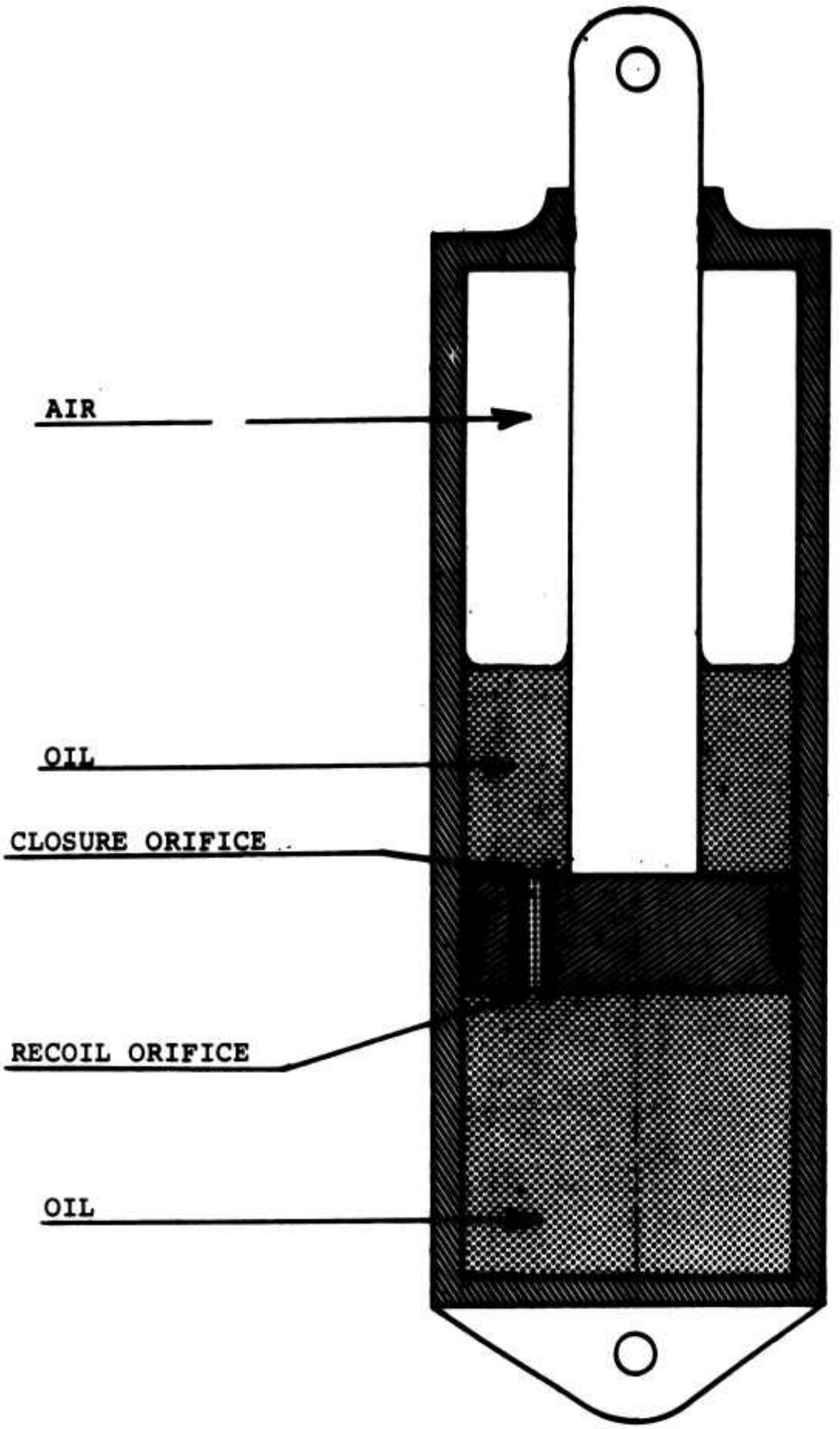


Figure 61. Hydropneumatic Load Isolator.

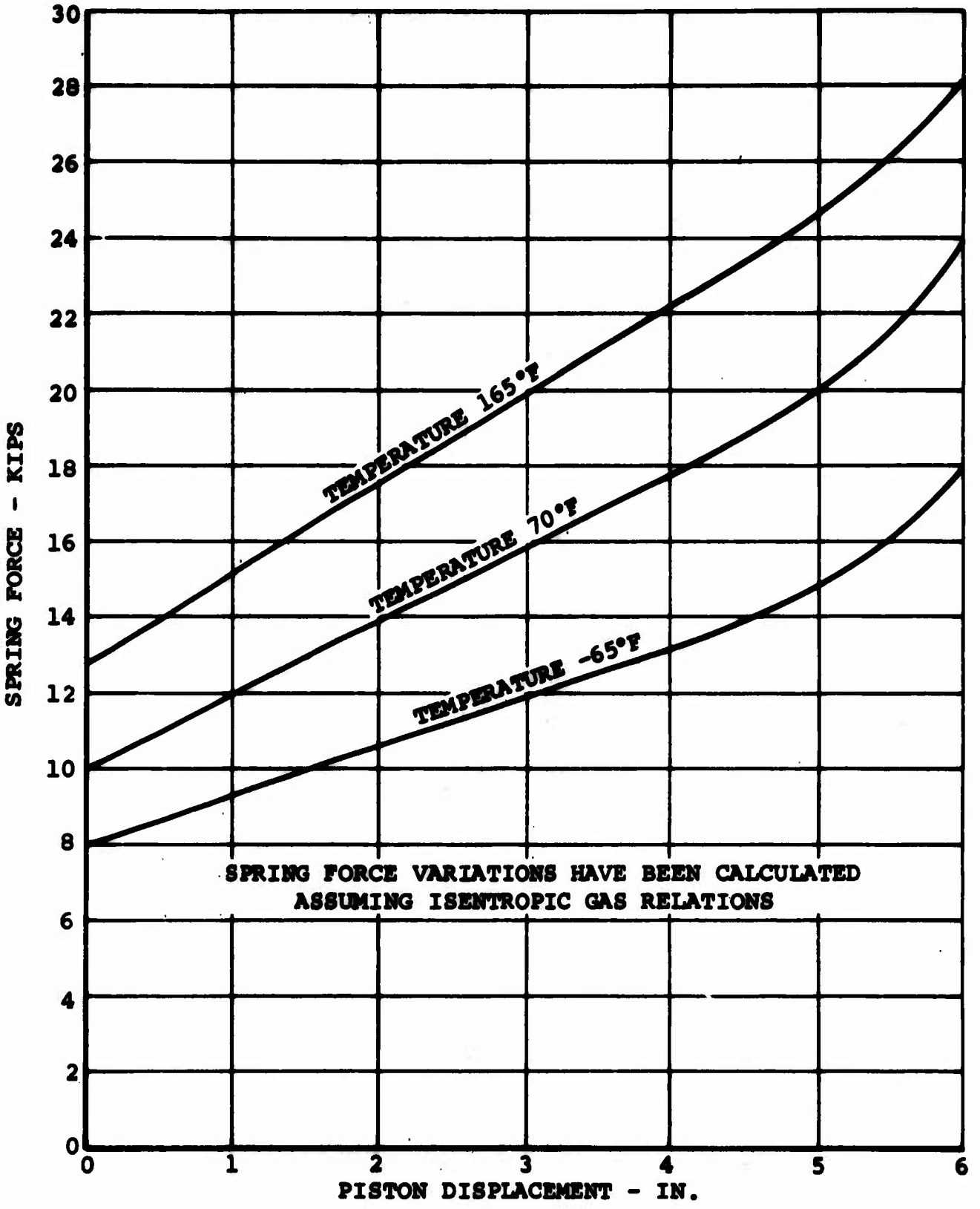


Figure 62. Low-Rate Spring Curves.

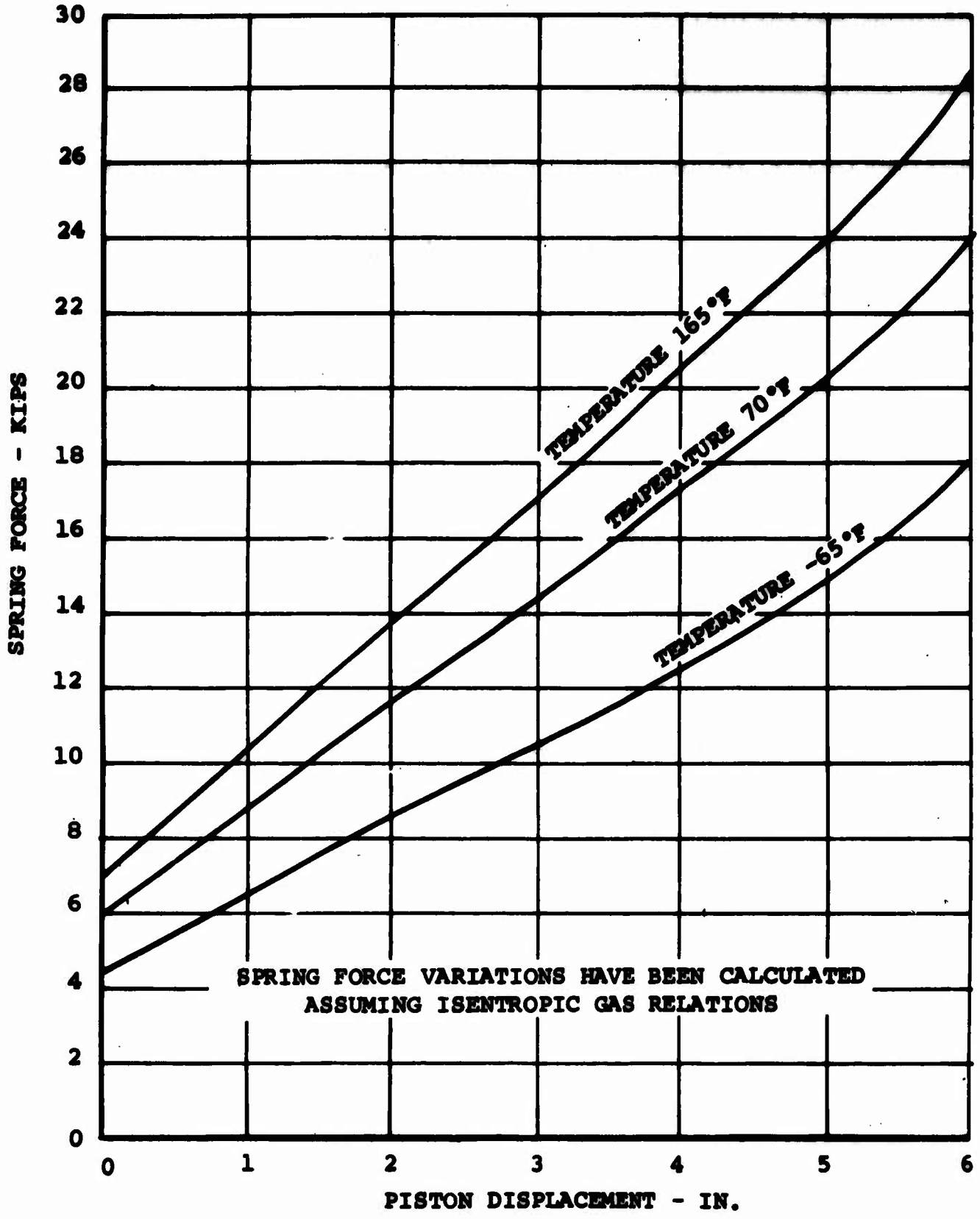


Figure 63. High-Rate Spring Curves.

damping variation.

Air-Spring Load Isolators

Air provides the softest, most efficient, most easily controllable vibration isolation medium known. An air spring offers the following specific advantages:

1. A lower natural frequency than can be practically attained by any other means

2. Essentially constant natural frequency - the same effective static deflection under all conditions of loading

3. Leveling - an additional feature which can be included by using an external air pressure source to provide the same static height under all conditions of loading

4. Economy - virtually no maintenance and relatively low initial cost

5. Lowest weight

The major disadvantages of using an air spring as a load isolator in aircraft hoist system installations are:

1. The physical size, for a given set of conditions, is large.

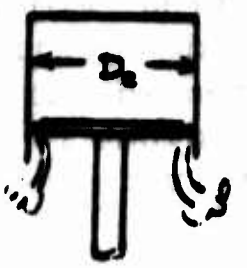
2. It is essentially a compressive device, whereas the intended helicopter utilization requires a tension device. The air spring can be adapted to a tensile installation but only at the expense of an envelope increase.

In applications where a winch system is mounted on the aircraft floor, the air spring can be used to great advantage. However, its bulk precludes its use in a suspended system.

Operation of Air Springs

The simplest kind of air spring is a piston operating within a cylinder. Such springs were patented and actually used in railroad applications over a century ago. Unfortunately, the

sliding seals on the piston not only leaked but also introduced considerable sliding friction that largely defeated the soft ride that the air springs were intended to give. Such air springs also have a rate characteristic that leaves something to be desired.



A nylon-tire-cord-reinforced rubber airide or airmount spring has no sealing problems and operates with practically no friction. Since the effective area (discussed below) can be varied, the rate characteristics can be carefully controlled.

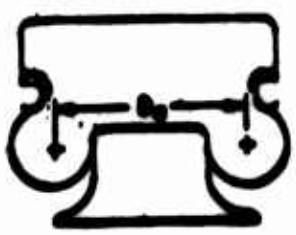
The important thing to remember is that the rubber does not support the load. The load is supported by a column of compressed air. An airide or airmount is a carefully designed package which contains a cushion of air.

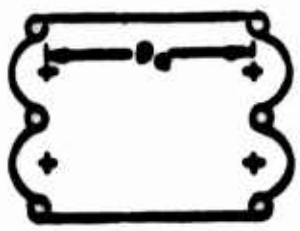
Three parameters control the rate characteristics of airide and airmount springs. These are: (1) effective area, (2) internal volume, and (3) operating pressure.

Effective area (A_e) is determined by the effective diameter (D_e).

$$A = \frac{\pi D_e^2}{4}$$

The effective diameter of two typical airides or airmounts is shown in the sketch below and on page 93.





Note: Airide and airmount are trade names of the Goodyear Tire and Rubber Company, Akron, Ohio.

Internal volume (V) is determined by actual measurement of the airide or airmount. The volume increases as the spring is extended and decreases as the spring is compressed. The rate of volume change can be reduced by adding displacement or reservoir volume to the spring system.

Operating pressure (P_{1_g}), measured in PSIG (pounds per square inch gauge), is determined by the load (L) and the effective area (A_e) of the airide or airmount.

$$P_{1_g} = \frac{L}{A_e}$$

As the airide or airmount is compressed, its volume (V_1) decreases to V_2 and its pressure (P_{1_a}) increases to P_{2_a}. (P_{1_a} and P_{2_a} are expressed as absolute, not gauge pressure.)

$$P_{1_a} = P_{1_g} + 14.7$$

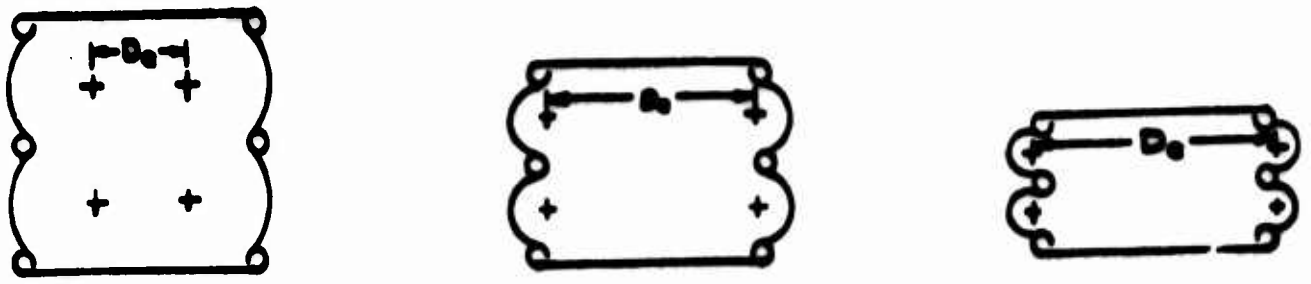
When an airide or airmount extends, its volume increases and the pressure decreases. The new pressure (P_{2_a}), after a known volume change, can be calculated according to the General Gas Law,

$$P_{2_a} = P_{1_a} \left(\frac{V_1}{V_2}\right)^{1.38}$$

In plotting a load-deflection or rate curve, in order to calculate the new spring <u>load</u> after a volume change due to spring movement, one need only multiply the new <u>gauge</u> pressure (P_{2_a} - 14.7) by the effective area. In the case of airides or airmounts, however, the effective area also changes as the spring moves. Thus the instantaneous effective area at each position must be used in calculating the new load after a spring movement. This is precisely the advantage of airides and airmounts over constant-area air springs. The manner in which the effective area changes for the one type of spring is illustrated on page 93.

It should be borne in mind that the rate of the airide or airmount will increase as the static load increases. As a result, the static deflection (L/R) and frequency remain practically the same. This is precisely the advantage that airides and

airmounts afford in a situation where they are required to support a varying load. They give essentially the same isolation and ride characteristics under all conditions of loading. However, it does necessitate calculating the rate for each load involved.



Effective diameters

Liquid Spring Load Isolators

The application of liquid springs (i.e., springs formed by compressing oils volumetrically) to aircraft systems is firmly established. They are used extensively in aircraft landing gear struts. Although essentially compressive devices, they can easily be adapted to tension application.

The basic mechanism of a liquid spring is merely a cylinder into which some form of plunger is introduced. The flow across the piston affects the shock absorption proper and, since areas are large, the damping pressure difference is not great. Recoil control requires a valve device on the piston. Figure 64 illustrates the principle.

Liquid Spring Performance Problems

There is little doubt about the reliability of the liquid spring concept, as demonstrated by millions of hours of service on aircraft landing gears. One feature which is not satisfactory, however, is the sensitivity of the units to temperature changes, as reflected in volumetric changes of the liquid. Initial pressurization of the oil to 2,000 psi can eliminate

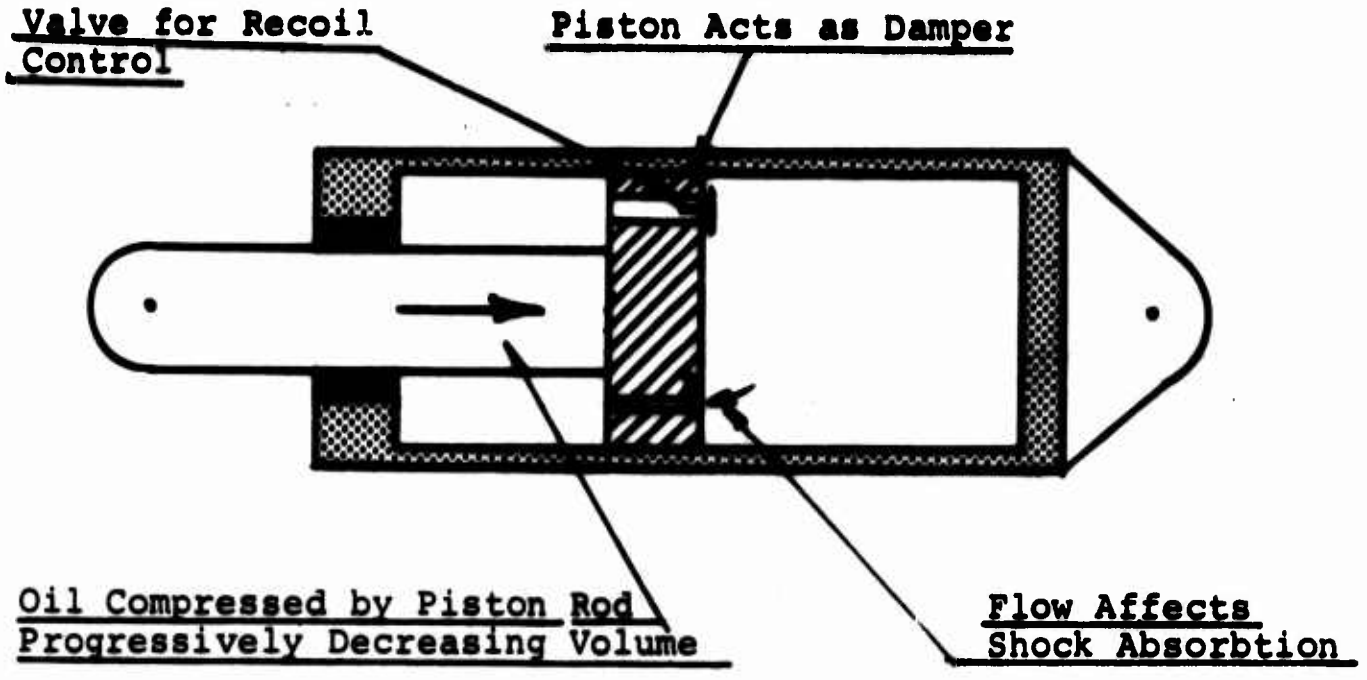


Figure 64. Liquid-Spring Shock Absorber.

the backlash or free play which will develop in the unit at low temperature. Unfortunately, this need for pressurization neutralizes some of the inherent maintenance advantages of the design. However, load isolator constructions which combine a liquid spring with a low-pressure air chamber integral withthe isolator have the advantage that the pressurization can be easily checked. Alternatively, existing aircraft hydraulic system pressure can be used to effect the required initial pressurization. Such a method has been proposed for the load isolators currently being developed for the CH-47 helicopter. However, in this case it was decided that the hydropneumatic system previously described would be used. The reason for choosing hydropneumatic instead of a liquid spring was that the pressures developed in the liquid spring are in excess of 10,000 psi as against 5,000 psi in the case of the hydropneumatic unit. Further, the hydropneumatic spring did not require an external hydraulic pressure supply. Figure 65 shows a simple tension load isolator schematic.

Summary - Sling Load Isolators

Any of the load isolator principles described are considered to be acceptable for airborne hoist system application. The

installation on the helicopter will be the major element controlling the choice. The use of capstan-type winches suspended from a single point dictates the use of either a liquid spring or a hydropneumatic type, whereas installation of winches inside a transport-configured helicopter would tend to favor the air-spring type.

Table V indicates the relative merits of the three types of isolators for the elements shown.

TABLE V*
RELATIVE MERITS OF THREE TYPES OF ISOLATORS

Type	Weight	Cost	Reliability	Installation Complexity	Efficiency	Maintainability
Hydropneumatic	2	1	1	1	2	1
Hydraulic	1	1	2	1	2	2
Air	3	2	2	4	1	3

*For element definition see "Relative Merits Of Hoist System Components And Complete Hoist Systems" on page 178.

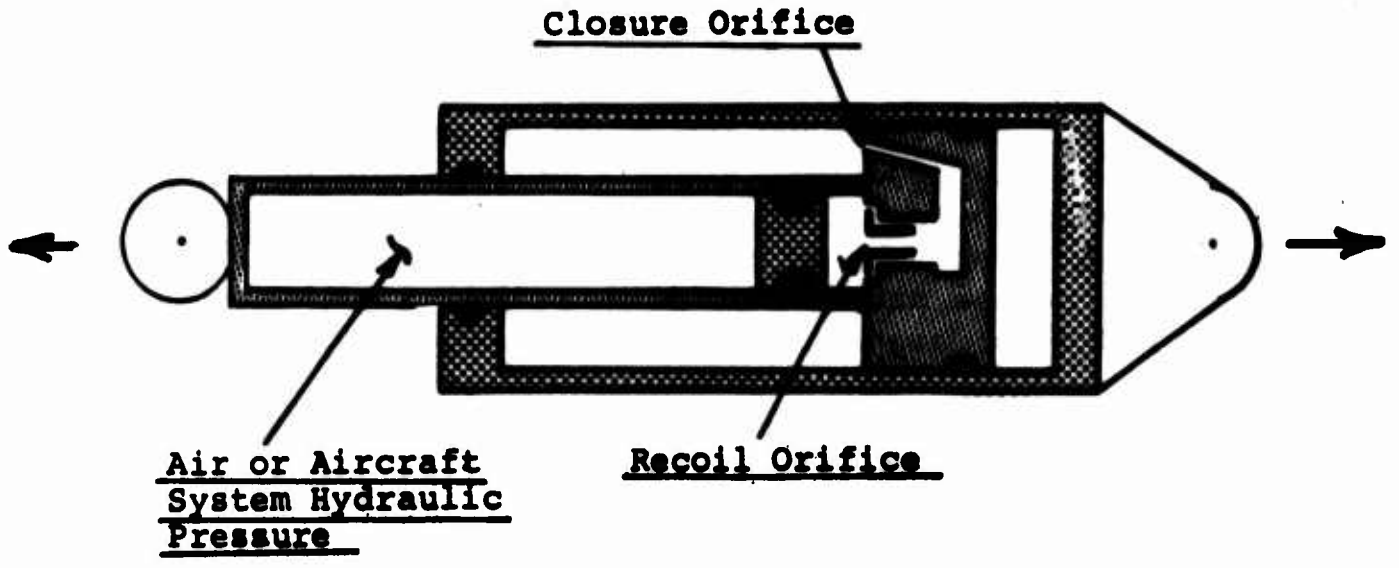


Figure 65. Tension Shock Absorber.

HOIST SYSTEM CONSIDERATIONS

Single-Point Hoist Systems

The basic purpose of a hoist system is to enable the helicopter to acquire a load from an inaccessible place, or to stand off from the load to alleviate downwash problems. Another use occurs in the vertical replenishment of warships operating in high-sea conditions where the load is heaving and rolling with the ship. Such a system is fitted to the UH-46 helicopter shown in Figure 66. It may be argued that these tasks can be performed by using a long fixed pendant attached to a belly hook. There are two reasons why this is not done. First, in many cases, the crew of the helicopter does not know in advance how much pendant is required. Second, and more important, it is sometimes difficult to fly with loads on very long suspensions.

Analysis of the Vertol 107 helicopter hovering with large loads on long suspensions (for a logging operation) indicated an unstable swaying mode in hover (refer to the STABILITY IN HOVER AND FORWARD FLIGHT section of Reference 1 of this report). This instability has a long periodic time which, while making it possible for the pilot to fly the helicopter, still produces an unpleasant sensation. Stability in forward flight depends on the actual load. If the load is aerodynamically stable, the long suspension is not usually detrimental; however, for unstable loads, the cruise speed may be severely limited. One problem with long suspensions is at least partly psychological; this is the tendency for loads to make large excursions at the end of the pendant. Operational helicopter crews are not accustomed to seeing this and tend to fly at reduced speeds, even though the actual angular excursions of the pendant may be the same as those experienced with a short pendant and considered acceptable (see Figure 67).

An important difference between long and short pendants, however, is that although the loads on the helicopter may be of the same magnitude, the frequency of the oscillations is different. Long suspensions give rise to low-frequency motions which tend to be very uncomfortable. This is discussed under STABILITY IN HOVER AND FORWARD FLIGHT, referenced above.

One of the most useful attributes of a hoist system is to enable the helicopter to acquire a load on a long line and

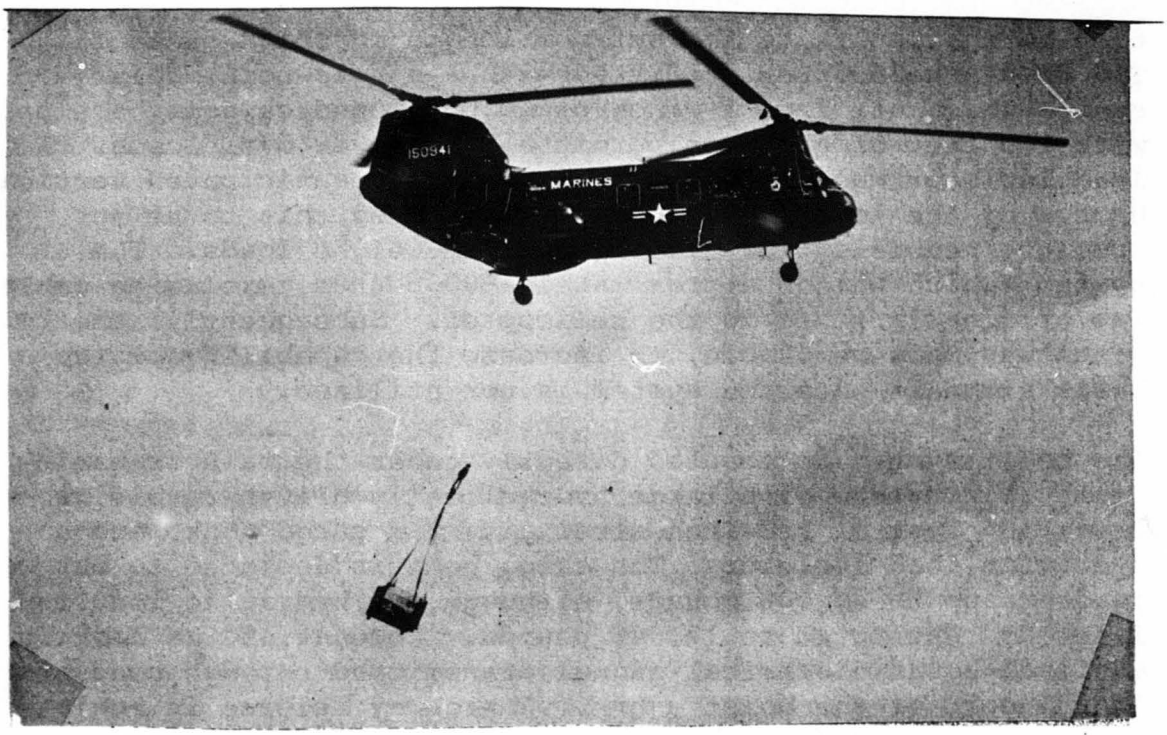


Figure 66. UH-46-Type Single-Point Hoist System.

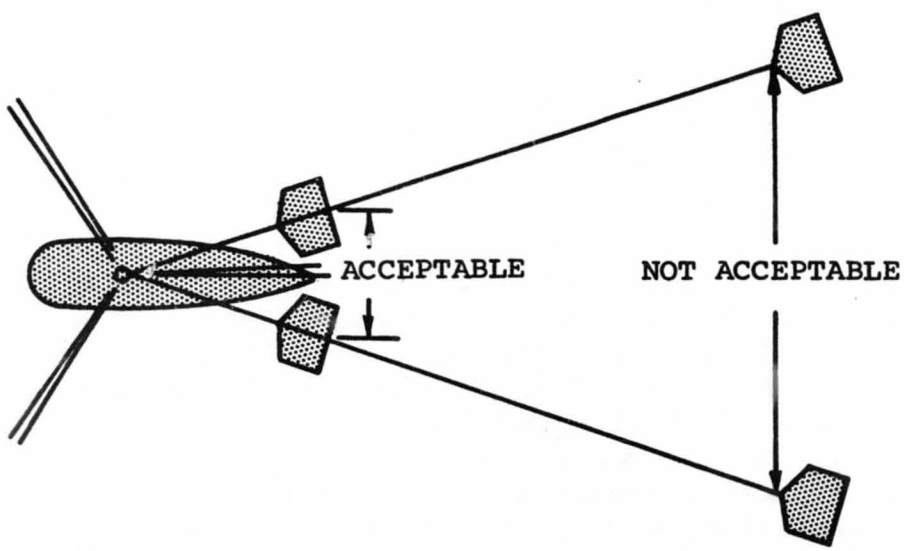


Figure 67. Effect of Suspension Length on Load Oscillation.

then to shorten it to the optimum length for high-speed flight. The CH-54A helicopter is fitted with a single-point hoist system which utilizes a winch of 20,000-pound capacity equipped with 100 feet of cable to permit a high hover over loads. A load isolator was fitted to compensate for anticipated vertical bounce of the load; subsequent tests proved this to be an absolute necessity for lifting maximum-weight loads. The system (which weighs approximately 980 pounds, excluding cable) was originally fixed to the helicopter. Subsequently, the winch was made removable, to increase the capability of the basic aircraft when the system is not utilized.

The hoist system is mounted directly under the main transmission. It consists of a large, hydraulically driven, revolving drum; 100 feet of 7/8-inch steel rope; a cargo hook; and a hydraulic load isolator. The cargo hook is designed to support loads of up to 20,000 pounds, although the hydraulic load isolator is restricted to 17,640 pounds. Release of the hook is achieved by an electrical signal transmitted through conductors in the core of the hoist rope. Emergency release is achieved by rope cutters (electrically triggered cartridge cutters). The single winch, which is of the conventional type, is rigidly mounted to the airframe and has a single layer of rope, only, to avoid chafing.

A single-point hoist system has load stability characteristics similar to those of the fixed single-point suspension as discussed in sections entitled "SUSPENSION METHODS FOR EXTERNAL LOADS" and "STABILITY OF SLUNG LOADS" located in Reference 1 of this document.

Consideration of single-point hoist systems with a capacity of 40,000 pounds for the heavy-lift helicopter indicates that further development of current techniques is not the answer. Discussion of cable and winch sizes in the section entitled "WINCHES FOR AIRBORNE HOIST SYSTEMS" of this document indicates that such a system would require a 1-3/8-inch-diameter rope. Another extremely important consideration is the effect of rope flexibility which, in the case of a 1-3/8-inch-diameter rope approaches that of a 1-3/8-inch-diameter lead pipe.

It is of interest to consider the practice used by commercial crane operators. Investigation of the cranes used in the shipbuilding industry (capacity to 100 tons) shows that regardless of capacity, the maximum-size rope used is 3/4 inch in diameter. The required weight capacity is achieved by using a

reeved rope system. A typical reeved rope system is illustrated in Figure 68. A reeved system requires less load for straightening (headache ball) than an equivalent single-rope system. The weight of the lower block contributes to the straightening load and so involves no weight penalty.

Reeved rope hoist systems have not been used in the past because there have been no problems with rope size. A factor to be considered when using reeved rope systems for single-point suspension is that if the lower block is allowed to rotate, the ropes may chafe against each other. Provided a swivel is used and the extension required is not excessive, the inherent restoring moment in the system will inhibit this tendency. To establish feasibility, it is possible to compute the maximum length of a given system that will not wind up on itself. Reeved rope hoist systems are used in pairs and are fastened to a common beam. Winding-up problems are minimized by the restraint provided by the beam.

Multi-Point Hoist Systems

The only airborne multi-point hoist system in existence is the four-point system built for the CH-54A. Multi-point hoist systems are a logical development, based on multi-point suspensions and single-point hoists. The objective is to provide the load stability of the multi-point suspension, while allowing pickup from hover and subsequent hoisting of the load to an optimum position.

Although the CH-54A four-point hoist system has yet to be evaluated, it is apparent that it will be subject to problems similar to those encountered in previous four-point suspension systems; namely, structural redundancy in the legs and synchronization of the release system.

The structural redundancy could be eliminated by coupling two adjacent legs to a single winch, as shown in Figure 69, but this would require the use of two different types of winches. This would be expensive and would compound the synchronization problems.

Release system synchronization is the overwhelming problem on four-point suspensions. The CH-54A four-point hoist system is not equipped with a pilot-controlled normal release. Normal release must be made by the ground crew, and this is very dangerous. One advantage of a hovering pickup is that outsize

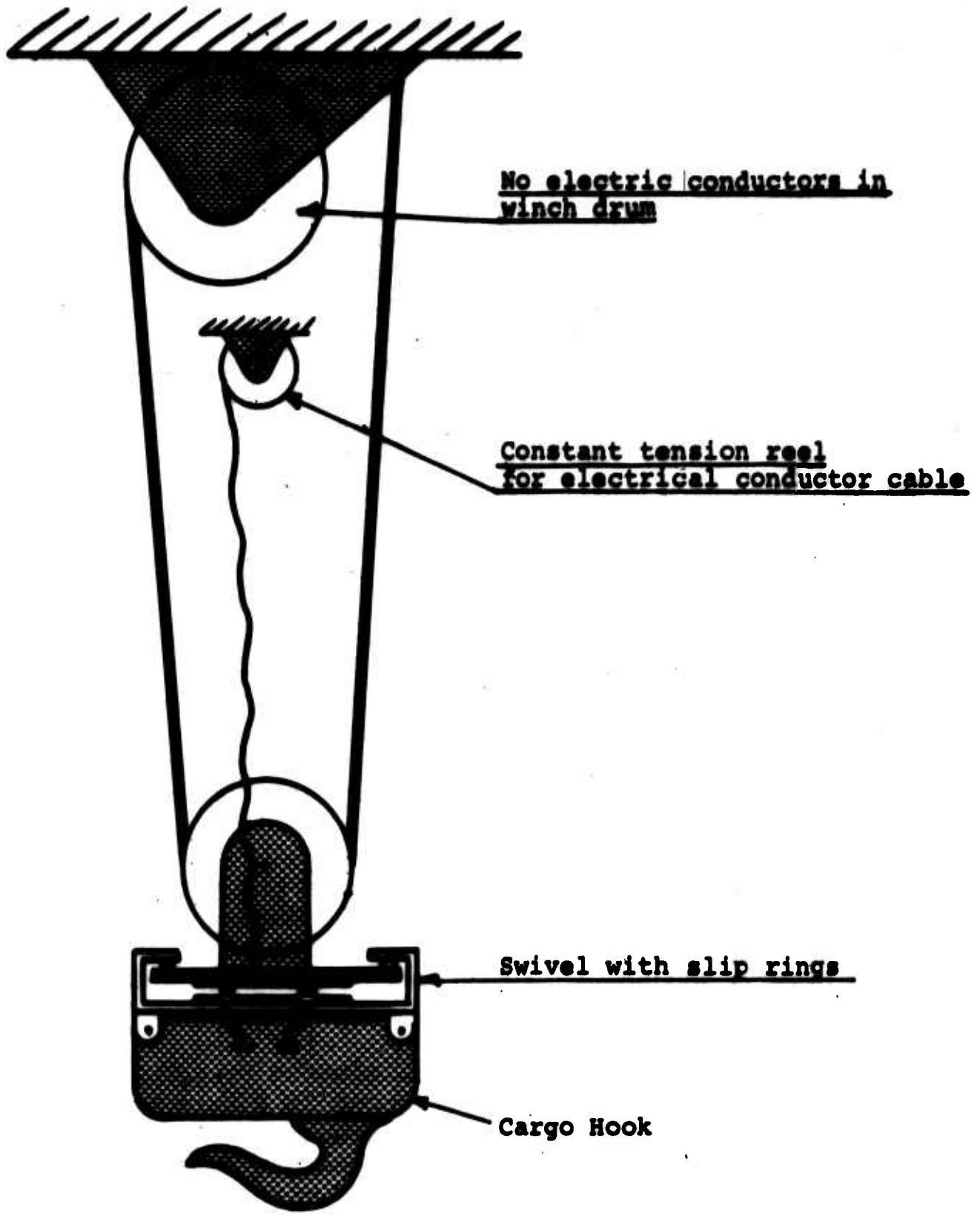


Figure 68. Single-Reeved, Two-Rope, Single-Point Hoist System.

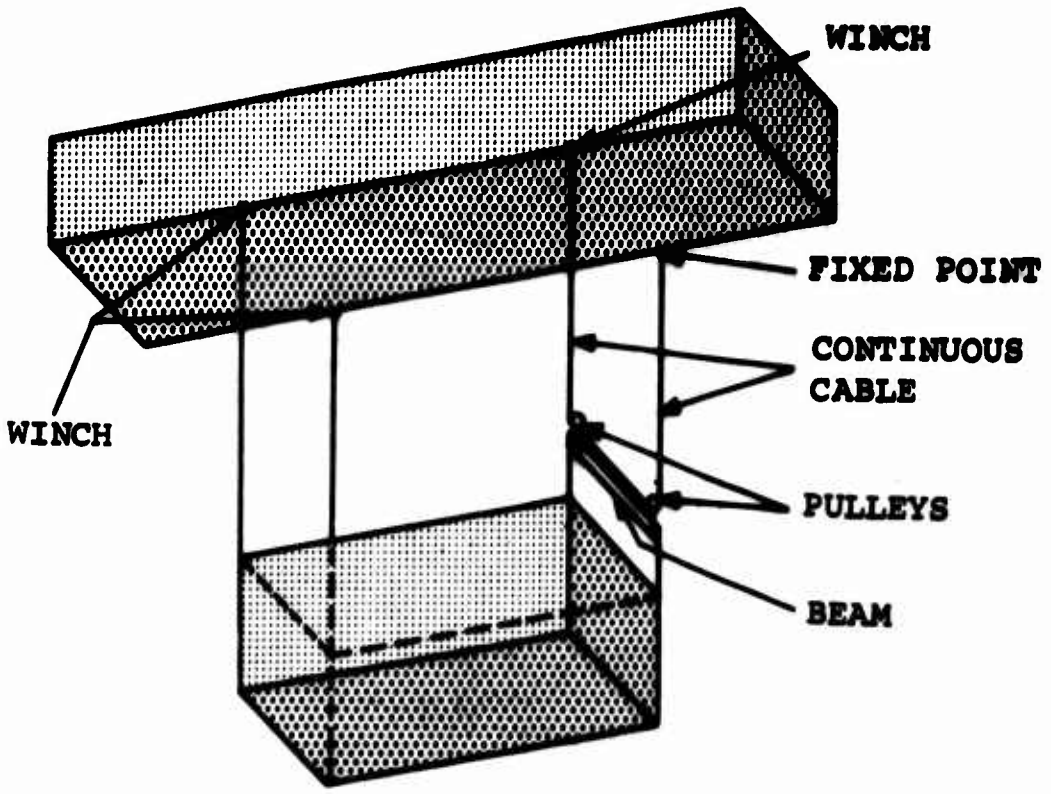


Figure 69. Elimination of Redundancy in Four-Point Hoist System.

loads can be acquired which the aircraft cannot straddle. Obviously, the release must also be made in hover (unless the lines are sufficiently long to allow alongside landing). If the load is only partially released and, because of a gust, the load is taken up in the hooked legs, the helicopter may overturn. Emergency release of the load on the CH-54A four-point hoist system is accomplished by rope cutters mounted in the winches.

As in the case of multi-point suspension, analysis of various multi-point hoist systems indicates the two-point hoist to be the simplest configuration. An elementary two-point hoist system is shown in Figure 55, and a more versatile system with a connecting beam is shown in Figure 70. Use of a connecting beam bestows several advantages, such as the ability to lift multiple loads. It also allows the hook release mechanisms to be mechanically interconnected, which ensures hook release synchronization. Electrical conductors for the hook release mechanisms are brought to the center of the beam where they are less likely to become snagged by the ropes . Since the ropes do not then have to contain conductors, the winch drums may be less complex and the overall system more reliable.

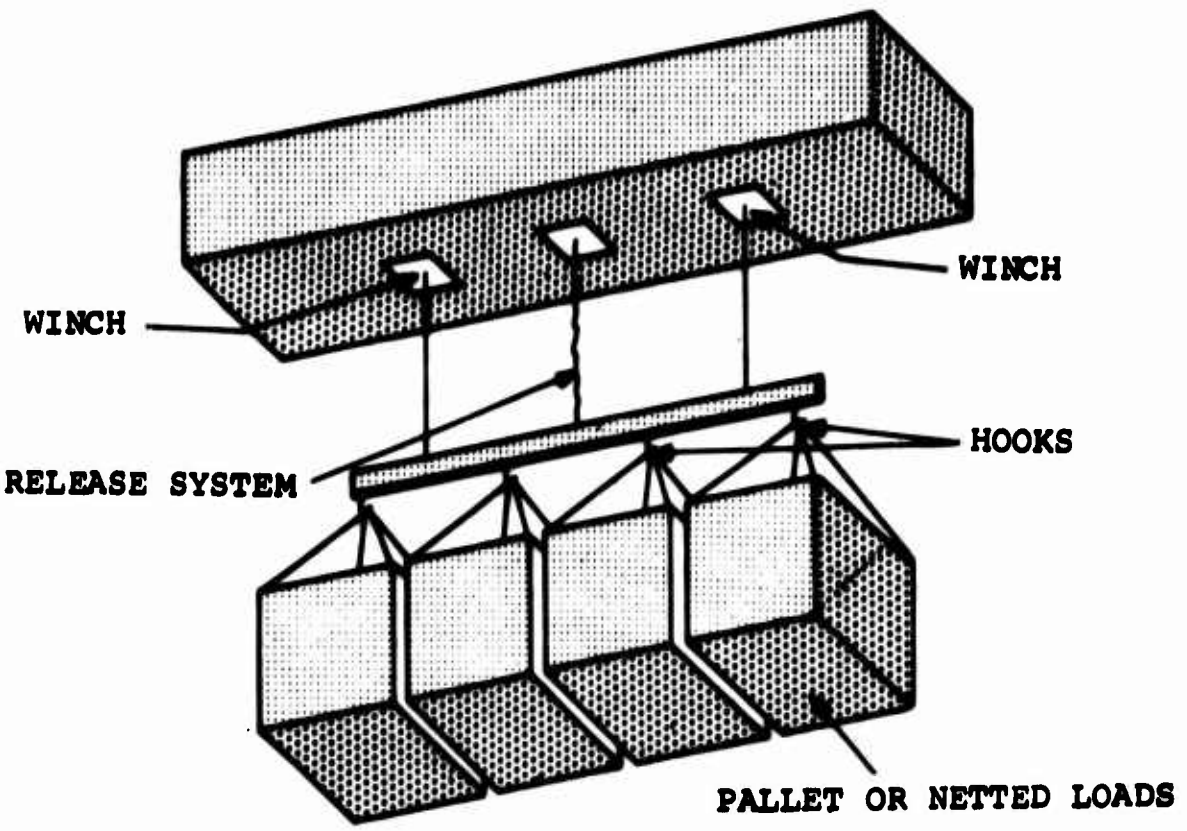


Figure 70. Two-Point Hoist System with Connecting Beam and Four Hooks.

When determining the capacity of individual hoist legs in a multi-point suspension, it should be remembered that, in general, the load will not be equally divided. Aircraft cg travel and, in the case of the four-point system, the system's inherent redundancy will dictate winch design capacity (refer to the section entitled "INFLUENCE OF AIRCRAFT CG TRAVEL ON HOIST SYSTEM DESIGN").

PODS CARRIED AS EXTERNAL CARGO

The word "pod" as applied to aircraft was coined to describe a container which was positively secured to the aircraft structure (first used on the C-120 aircraft). The philosophy was that such pods could be prepacked and so increase aircraft cargo transfer productivity. The word was later used to describe a module for the transportation of personnel on flying-crane-type helicopters and became known as a "people pod." This pod was necessary to convert a basic crane-type-configured helicopter into a personnel carrier. As in the first application, the pod was positively secured to the helicopter and possessed no jettison capability. The jettison capability was not required since, having a tall landing gear, the helicopter could land with its pod.

Thus the pod, as it is known today, was dependent upon the prime mover's having a tall landing gear and a built-in mechanism for acquiring the pod with the helicopter on the ground. This technique is known as a straddle pickup and requires the pod to be maneuvered under the helicopter. This type of pickup was practical provided the aircraft and pod were on level ground; however, on unprepared terrain, difficulty has been experienced in acquiring the pod. This is in part due to the fact that the alignment of four points on an unwieldy, relatively flexible container is difficult unless sufficient adjustment (vertical, lateral and longitudinal) is provided. This adjustment capability has been found to be inadequate on crane-configured helicopters in service today, unless the pod is acquired on level, prepared terrain.

Since this method of attachment requires the helicopter to land for pod acquisition and delivery, any operations requiring use of the pod concept in its present form will be restricted to areas where the helicopter may land on terrain level enough for pod acquisition. Such a situation is incompatible with the principles of true air mobility. To eliminate this operational limitation, one approach is to treat the pod as a suspended load, with acquisition being achieved by the helicopter from hover, and the pod being subsequently hoisted up to the aircraft. Multi-point suspension to give flight stability to the load will be mandatory for maximum productivity. However, if the pod were designed with acceptable aerodynamic characteristics and its cg maintained centrally within tight limits, there appears to be no reason why such loads could not be carried on a single-point system.

Special types of pods such as radar units, medical units and communication centers can be classified as high-value loads; if such a pod is acquired as a suspended load, some method of positively locking it to the structure may be considered necessary. An aircraft configured with a tall landing gear would allow delivery of the load from hover or by landing; also, in the event of an emergency, it could land with the pod. However, if the aircraft is configured as a transport (i.e., with short gear), then embodiment of emergency jettison capability for the pod becomes mandatory.

A four-point hoist system used for pod acquisition will be subject to the requirements discussed in the section entitled "SUSPENSION METHODS FOR EXTERNAL LOADS." Essentially, this calls for synchronized release of four cargo hooks by elec-

trical mode (normal) and also by manual (emergency) mode. During acquisition from hover, the physical attachment of the pod to the hoist system will require four men (one to each point) in order to ensure simultaneous attachment of hooks. This simultaneous attachment is necessary in the interest of flight safety, since the helicopter is subject to movement in all directions if gust conditions are present. Further, considerations of safety will also dictate that sufficient wire rope slack be present to compensate for helicopter movement. (For normal loads, this working slack is obtained by the load sling.)

Conclusions - Pods Carried as External Cargo

The following conclusions are relevant to the carrying of pods as external cargo:

1. Hover pickup of pods by means of a multi-point hoist system will provide optimum operational capability.

2. A four-point system will require twice the number of ground personnel required by a two-point system.

3. A single-point pickup is possible, but the system suffers from the flight limitations characteristic of this mode.

4. When pods are carried as suspended loads on transport helicopters, jettison capability should be mandatory.

5. If personnel are carried in the pod or if the pod is considered to be of high value, then provision must be made for positively locking it to the aircraft structure.

6. The simplest acquisition will be achieved by treating the pod as a slung load.

THE INFLUENCE OF HELICOPTER SAFETY OF FLIGHT CRITERIA ON THE DESIGN OF AIRBORNE HOIST SYSTEMS

Operational experience gained during the evolution of external suspension systems for helicopters has resulted in certain requirements becoming mandatory in the interest of flight safety, as detailed in the following paragraphs.

Release of the Suspended Load

The release of the load shall be under the prime control of the pilot and copilot, and shall be accomplished by an electrical signal to either an electrical or a hydraulic actuator; this release shall take no more than one-half of one second from the moment of initiation to release of the cargo (refer to Reference 9). This mode of release is called "normal release." Also, an emergency release under pilot or copilot control shall be installed to provide for electrical system failure. The actuating device and the system for transmitting this emergency release command to the hook mechanism shall be independent of their counterparts in the normal release system. The emergency release system may be a completely independent system. Lanyards shall not be employed for emergency release. A manual means of release shall be provided on the hook for convenient use by ground personnel.

The following controls shall be provided for operation of the cargo hook system:

1. Control Panel - A control panel for the use of the pilot and copilot shall be provided and shall contain controls and indicators as set forth below. The panel shall be in accordance with Specification MIL-C-6781.

2. Release Mode Switch - The release mode switch shall be a guarded, three-position switch in accordance with MIL-S-3950; it shall provide for selection of the type of release and shall act as the master switch for the system. When in the center or SAFE position, the normal hook operating system shall be de-energized through the upper or normal release switches. In the lower or AUTOMATIC position, the hook system shall be in the automatic-release condition. The guard shall not affect the switch when opened, but shall return it to the SAFE position

from either side when closed. The guard shall be in accordance with Specification MIL-G-7703.

3. Remote Station Select Switch - This switch shall be a two-position switch which will permit normal release by the pilot or copilot when set in one position, and by the pilot, copilot, or crew chief in the cargo compartment when set in the other position. The switch shall be in accordance with Specification MIL-S-3950.

4. Emergency Release Control - An emergency release control, to actuate the emergency release system, shall be provided. This control shall require a positive action to actuate it and shall not be liable to inadvertent actuation through being bumped or snagged by any personnel or their equipment.

5. Indicators - The control panel shall contain two indicator lights. The first shall be an amber light, illuminated when the hook system is in the automatic touchdown condition; the second shall be a red light, illuminated when the hook load beam is in an unlatched condition. These indicators shall be in accordance with Specification MIL-L-7961 and shall have a dimming feature.

6. Normal Release Switches - Normal release switches shall be provided for the pilot, copilot and crew chief. The switches for the pilot and copilot shall be of the pushbutton type and shall be suitable for location on their cyclic sticks. These switches shall be in accordance with Specifications MIL-S-6744 and MIL-S-6743. The normal release switch for the crew chief shall be a hand-firing key, or pickle, in accordance with NAF1174-1 or equivalent.

7. Interconnection Provisions - Interconnecting cabling, piping, or other provisions, including all connectors and other components, shall be provided as required for complete system operation. These provisions shall permit the hook to be located at any suitable distance from the control panel. Provisions for connection to the crew chief's firing key shall be made at a position near the hook. All electrical

wiring and cabling shall be in accordance with Specification MIL-W-5088. All electrical connectors are to be in accordance with Specification MIL-C-5015. Other systems shall be in accordance with applicable MIL specifications or equivalent.

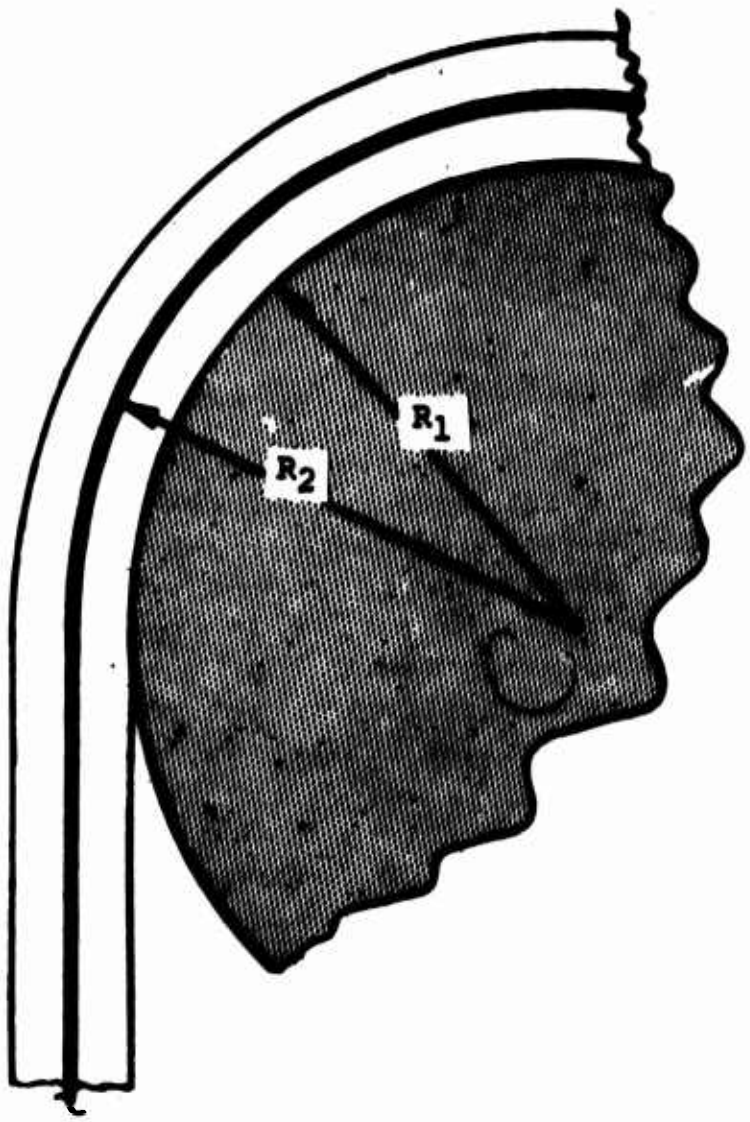
The requirements of the foregoing criteria have been successfully met and embodied on helicopters already in the military inventory which employ a single-point suspension system; e.g., the CH-34, CH-46, and CH-47.

Single-Point Hoist Requirements

The introduction of a single-point hoist system compounds the hook release problem. First, electrical conductors must chase the hook up and down; this has been resolved by having the electrical conductors integral with the hoist rope. For their protection, these conductors are buried in the core of the wire rope during manufacture, which means that they are subject to alternate bending and straightening as the rope winds on and off its storage drum - a fatigue condition, as shown in Figure 71.

Second, the hook (now suspended on a single rope) has no torsional restraint; this can result in rope unwinding. However, to prevent this condition, a swivel is fitted between the hook and rope which prevents rope windup but creates electrical conductor discontinuity. This problem is resolved by incorporating slip rings in the hook swivel. The same method is used to provide continuity between the rope storage drum and the aircraft electrical system (see Figure 72).

Direct manual emergency release is impractical on a single-rope hoist system. However, a compromise employing a cartridge-activated gas generator system has been introduced by Eastern Rotorcraft, Incorporated, Doylestown, Pennsylvania. In this system, gas is used to drive a piston which in turn effects "manual" release of the hook. However, the cartridge is electrically fired, the electrical conductors having to travel over the same route as those required for hook release. As an additional safety measure, cartridge-type cable cutters are usually embodied on the wire rope adjacent to or integral with the winch; this system is electrically actuated under pilot/copilot control. Use of the cutters destroys the hoist rope, preventing the helicopter from carrying external loads until the rope is replaced.



Different radii, R_1 & R_2, cause alternating loads in conductors during winding in and out on winch drum.

Figure 71. Effect of Winding on Drum a Wire Rope Containing a Core of Electrical Conductors.

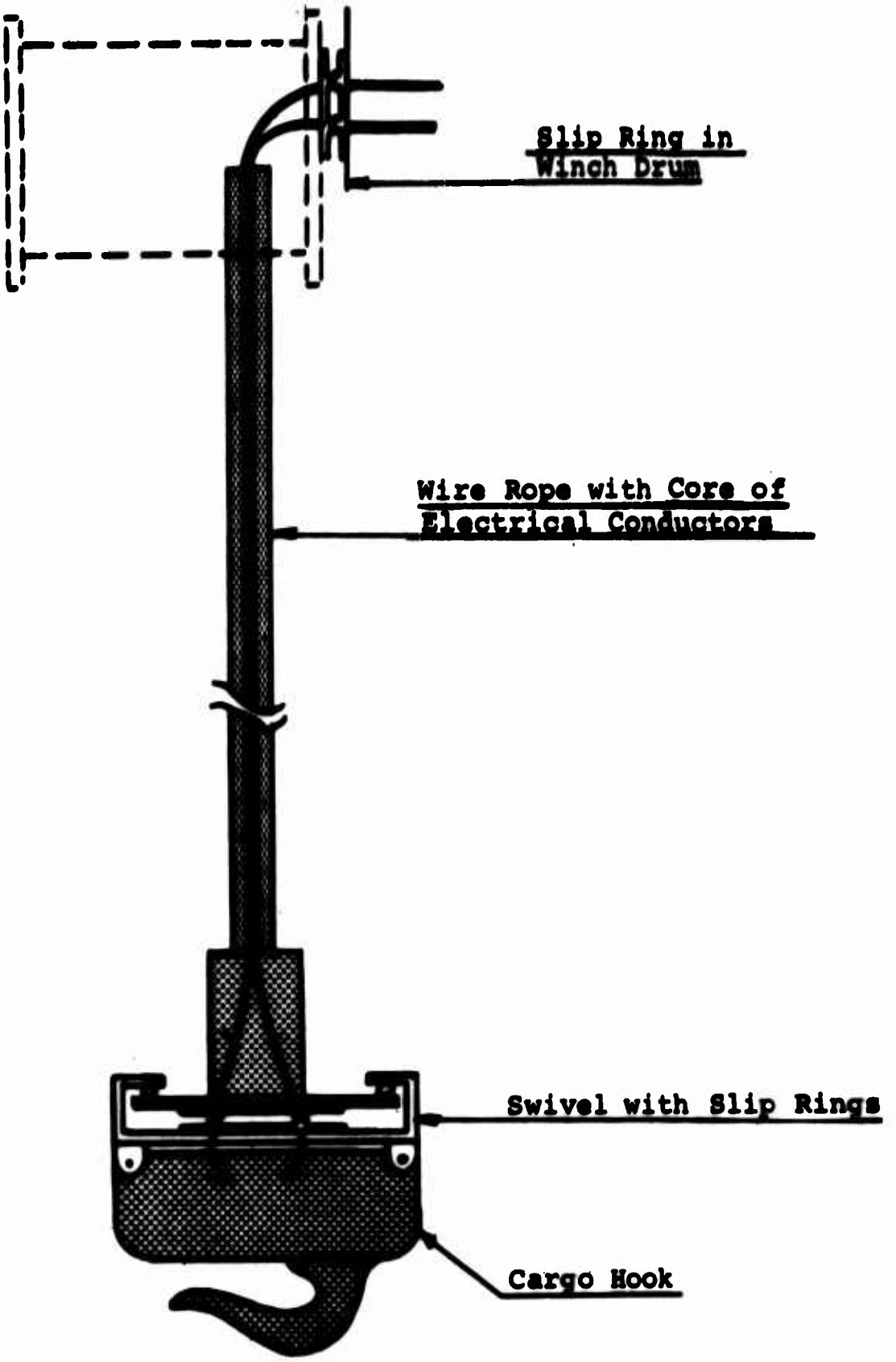


Figure 72. Single-Rope Hoist System.

If the recommendation that wire rope for airborne hoist systems not exceed 1 inch in diameter is adhered to, then it will be necessary to resort to a reeved rope system for those cases which require hoisting capacity in excess of that provided by rope of this size.

In a reeved system, there is no positive continuity between the wire rope and the hook (i.e., the continuity is via a pulley); this negates the use of electrical conductor circuits integral with the hoist rope. The required electrical conductors must therefore be run separately from the wire rope. This can be achieved by the use of a constant-tension storage reel which automatically allows the electrical conductor cable to chase the hook as the hoist is raised or lowered. Because winding up of the rope system is not acceptable, the use of a slip ring between the hook and the lower rope reeve pulley block (Figure 68) becomes mandatory. Since the electrical circuits are now divorced from the hoist rope and a requirement for slip rings in the winch is eliminated, a more reliable system will result.

Multi-Point Hoist Requirements

The problem of load release in multi-point suspension systems is further compounded by having to release all hooks in the system simultaneously while still satisfying the requirements for normal and emergency release under pilot/copilot command. The reason for this specific requirement can be justified by consideration of the purpose of multi-point suspension as against that of single-point. The single-point system, due to load instabilities, restricts the helicopter to slow, relatively unproductive forward flight. To stabilize the suspended load in forward flight, it becomes necessary to use multi-point suspension. The relative efficiency of a multi-point system is influenced by the base length; the larger the base, the more stable the load. The resultant base length will spread outside the acceptable limits of helicopter cg travel, so that unsynchronized release of the load will cause an upsetting moment to act upon the helicopter - a serious hazard to flight safety.

Since release synchronization in the specified modes is mandatory, it follows that some form of mechanical interconnect will be required between all hook release systems. This can only be achieved by terminating the lower ends of the multi-point suspension system in a structural element which will interconnect the hook release systems mechanically. As in the case

of a reeved single-point hoist system, the electrical conductors for hook release will be rigged independently from the hoist ropes. Such beam systems are shown in Figures 73 and 74. The release coupling on a two-point system is shown in more detail in Figure 75. It will be noted that both normal and emergency modes can be achieved, and the number of hooks on the beam is not necessarily restricted to two. The application of a similar feature to a four-point system is shown on Figure 76. The four-point beam system, because of its physical spread, will be approximately twice the weight of the two-point system for the same load capacity.

The introduction of a beam feature into multi-point systems permits the use of reeved ropes and eliminates their tendency to wind up on themselves, thus making the use of swivels unnecessary.

The problem of ensuring flight safety in the event of breakage of a hoist rope or runaway of one winch can also be resolved by using the beam feature. The electrical conductor cable reel can be designed to have an inertia-activated lockout so that in the event of rope failure all hooks will be automatically opened, thus releasing the load before upsetting moments are transmitted to the helicopter (see Figure 77).

An undesirable feature of a two-point system with beam is that if the total helicopter load acquisition system comprises a single-point system plus a two-point system, the beam of the two-point will interfere with the single-point system (see Figure 78). To eliminate this interference, the two-point beam will have to be designed to provide clearance for the single-point system, as shown in Figure 79. This modification may also be used to provide lateral restraint for the beam if it is decided that the beam should be physically locked to the aircraft structure in its up position, in order to provide suspension system redundancy.

Serious consideration should be given to the fact that a two-point suspension hoist system can be used for the single-point pickup, as shown in Figure 80. By adopting this method, the normal electrical release can be achieved both at the single-point hook and at the two-point hook. Since emergency release will also be available at the two-point hooks, the flight safety requirements for load release by pilot/copilot will be exceeded. The elimination of a separate single-point system means a reduction of approximately 2000 pounds in helicopter

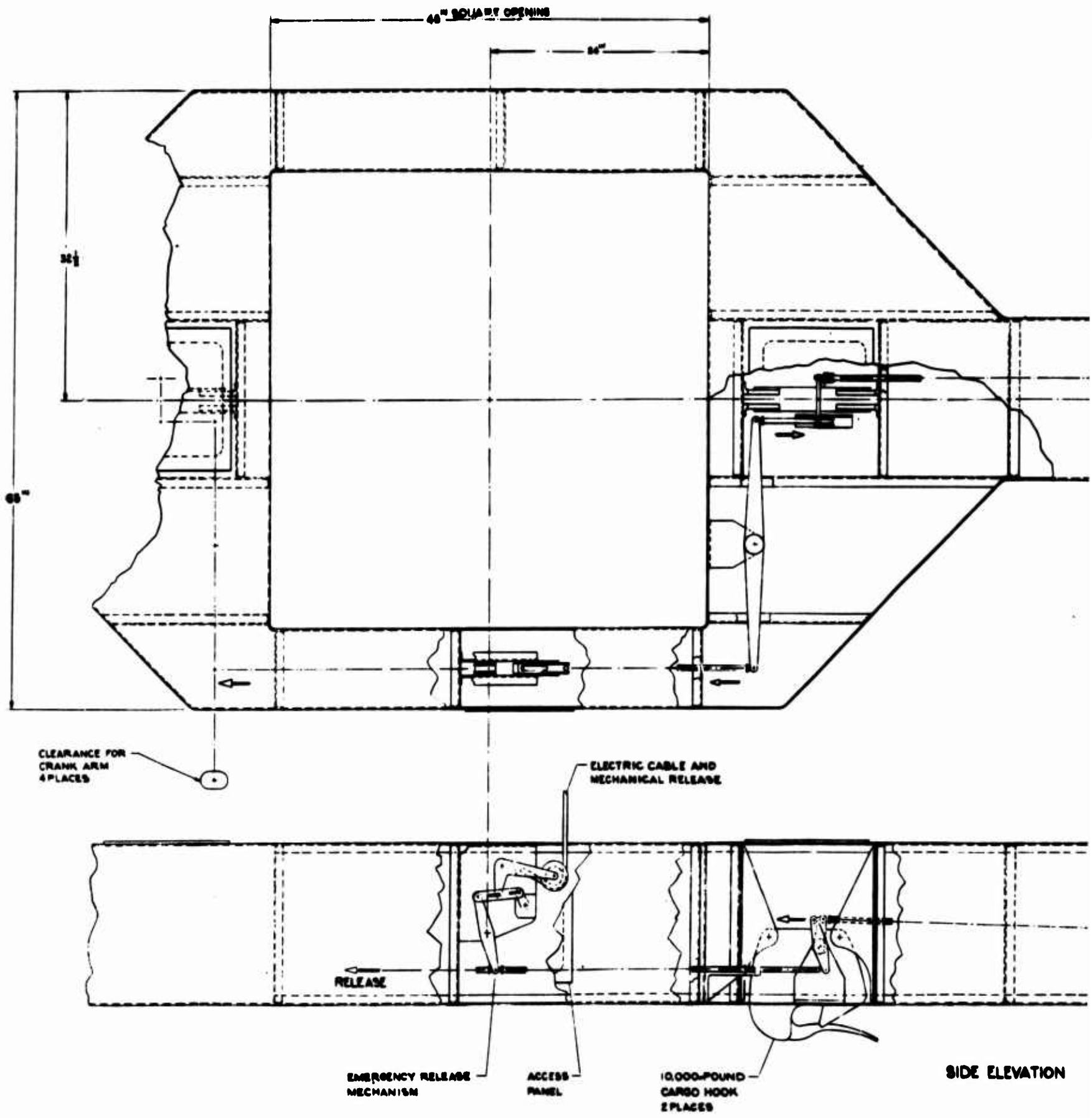
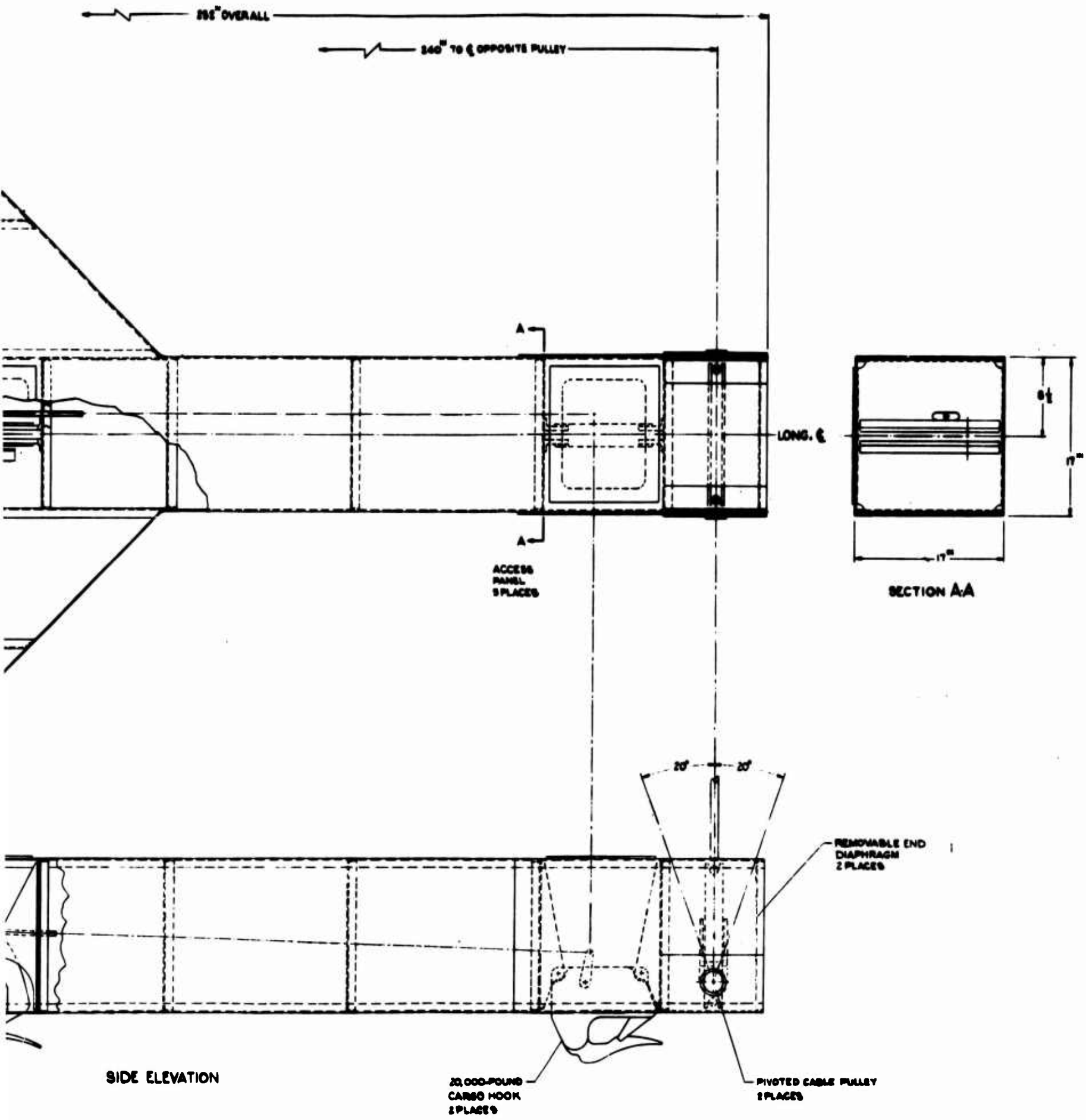


Figure 75. Two-Point Beam Showing Hook Release Mechanism.



SIDE ELEVATION

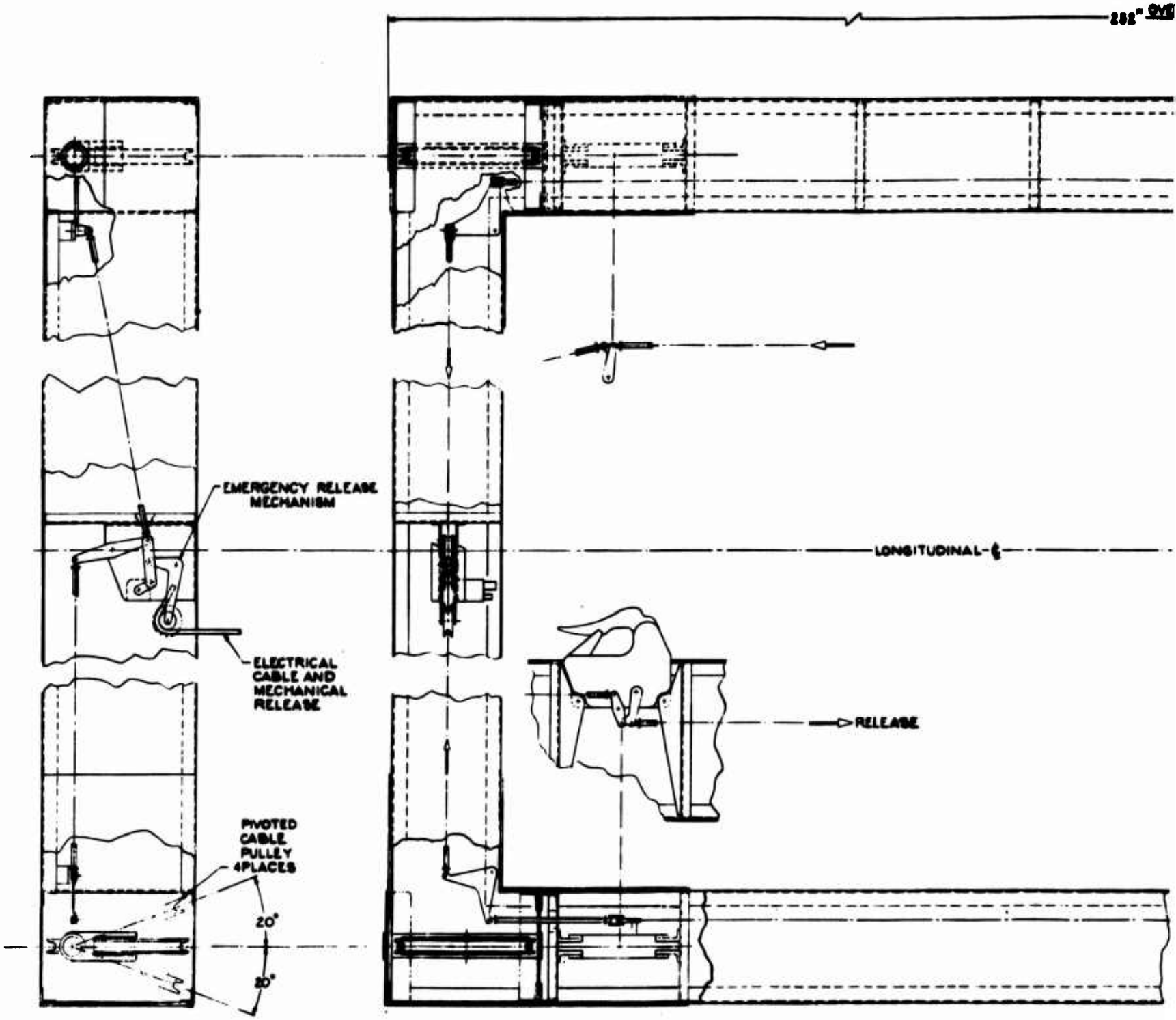
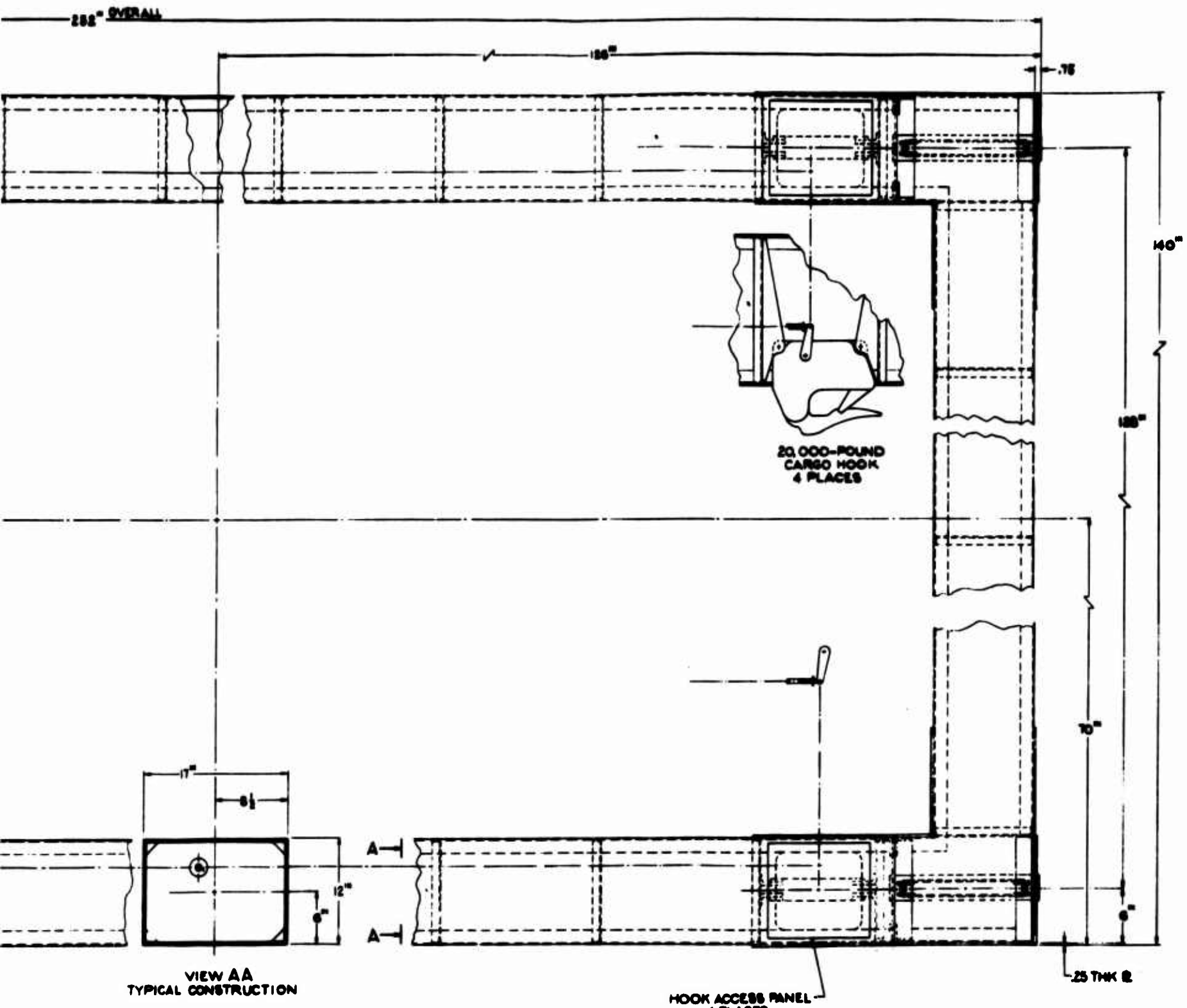


Figure 76. Four-Point Beam Showing Hook Release Mechanism.


252" OVERALL
156"
.75
140"
120"
70"
9"
20,000-POUND
CARGO HOOK
4 PLACES
17"
8½
12"
6"
A
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VIEW AA
TYPICAL CONSTRUCTION
HOOK ACCESS PANEL
4 PLACES
.25 THK ®

2

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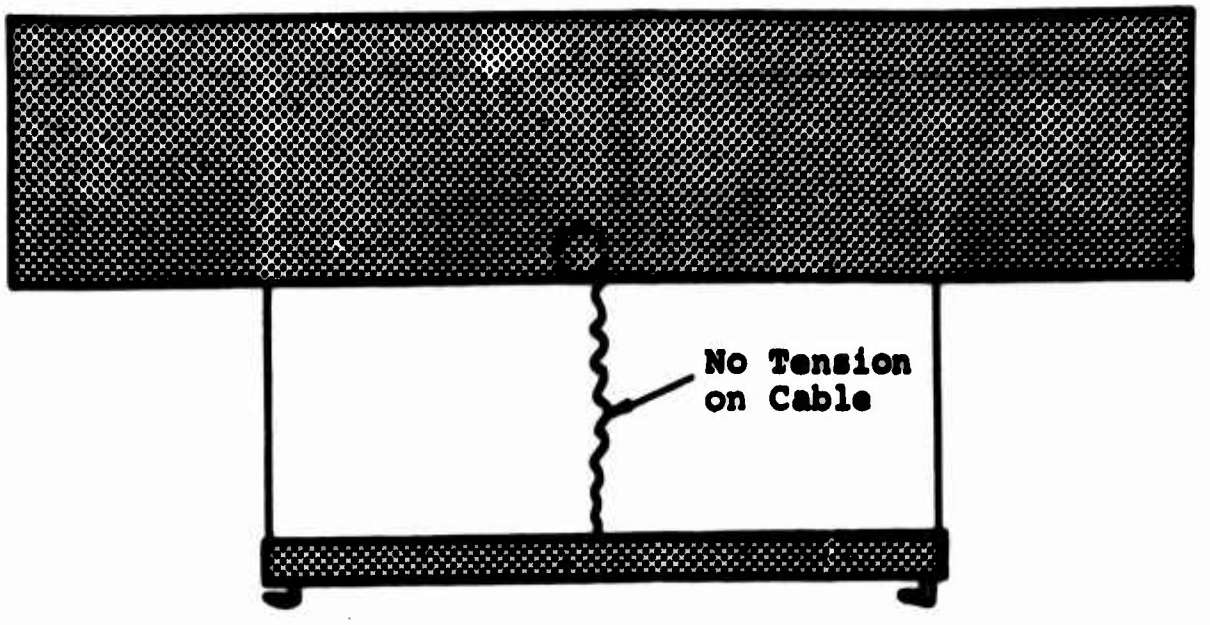


Two-Point Normal Mode - No Tension on Electrical Cable

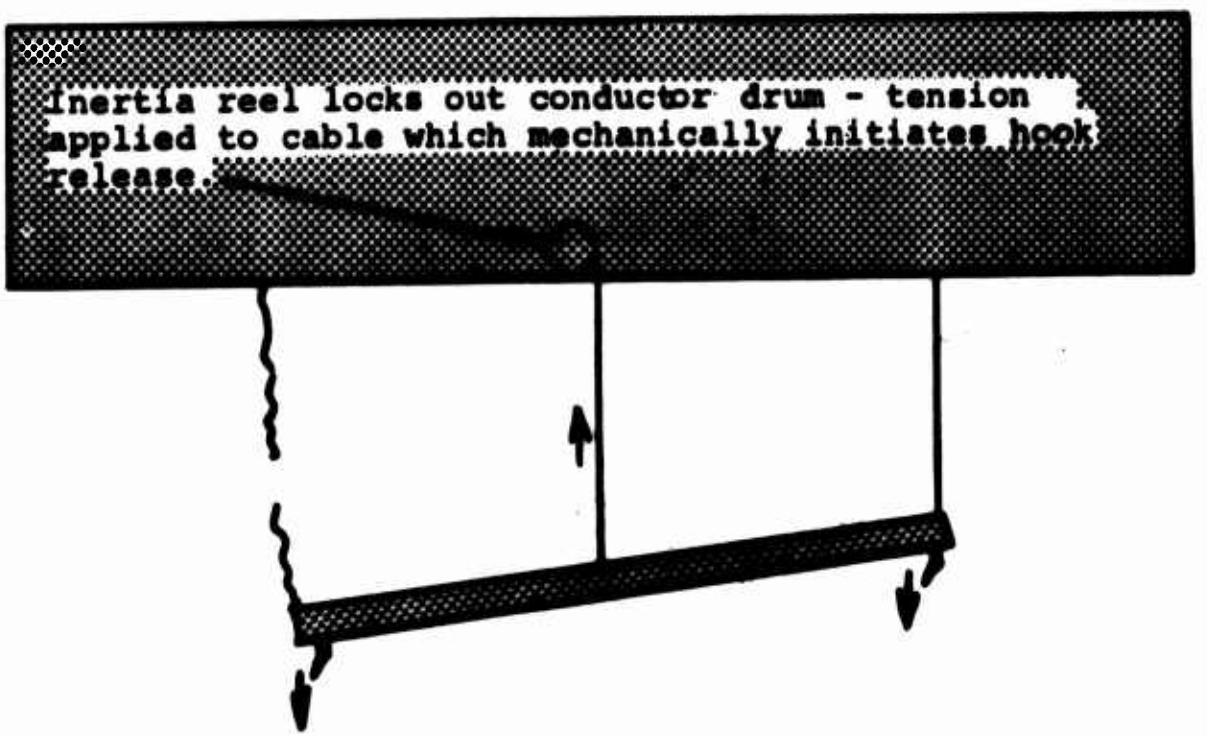


Figure 77. Emergency Due to Rope Breaking.

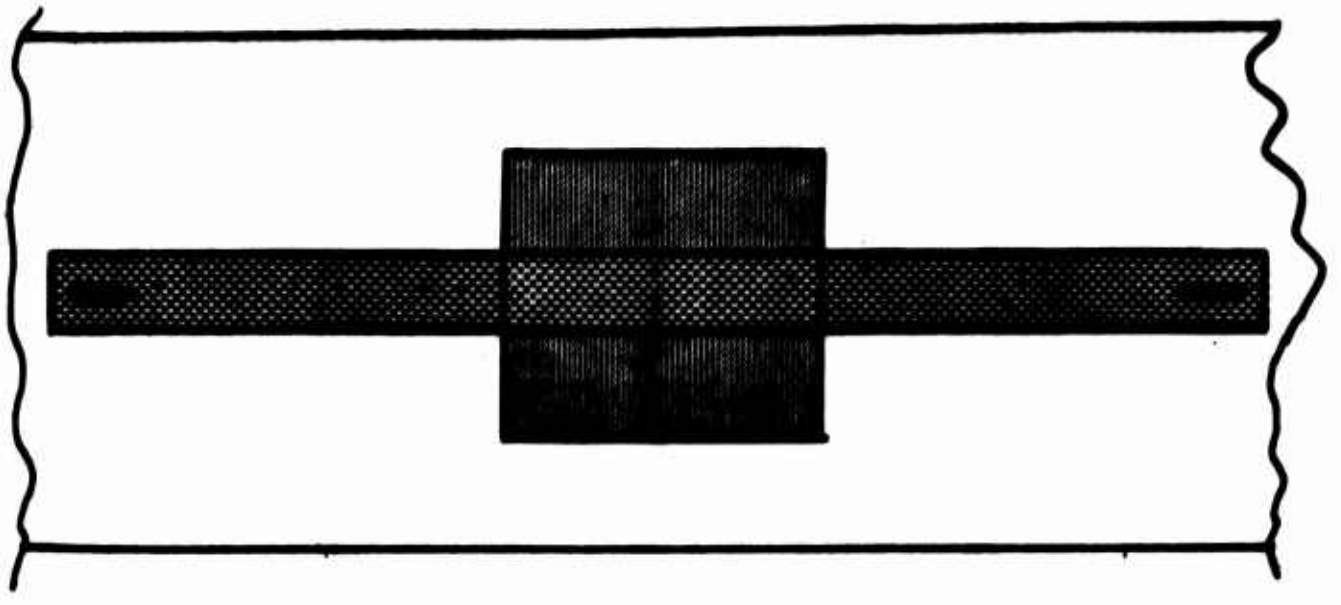


Figure 78. Bottom View of Helicopter Showing Two-Point Beam Interfering with Single-Point System.

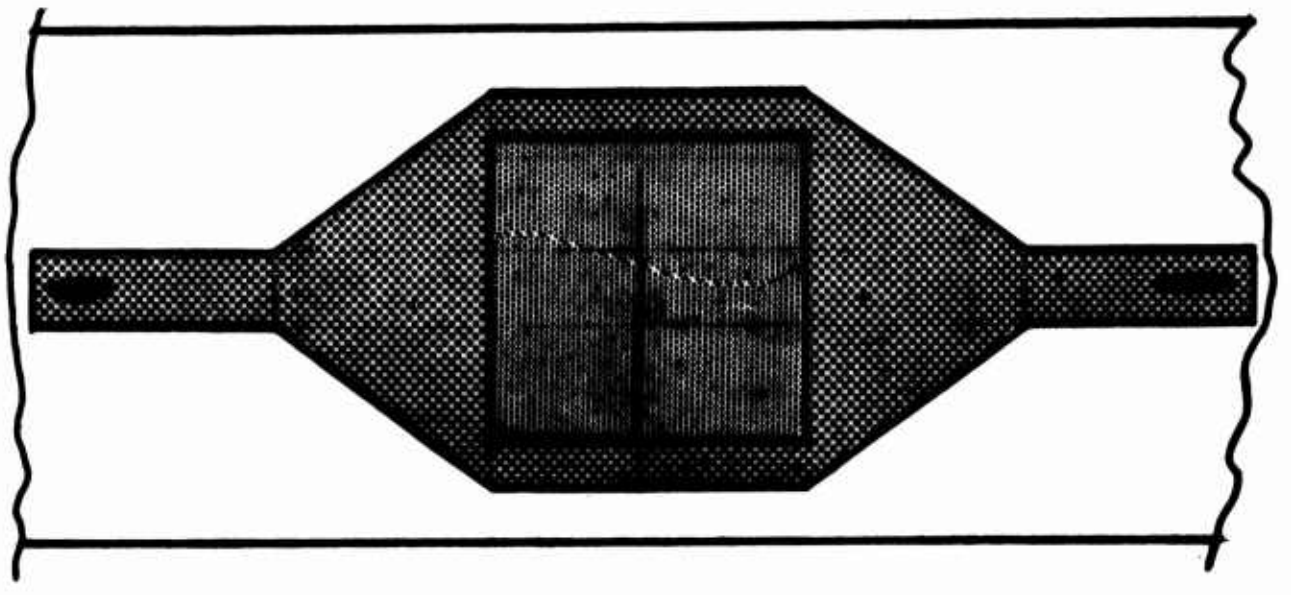


Figure 79. Bottom View of Helicopter Showing Two-Point Beam Modified to Clear Single-Point System.

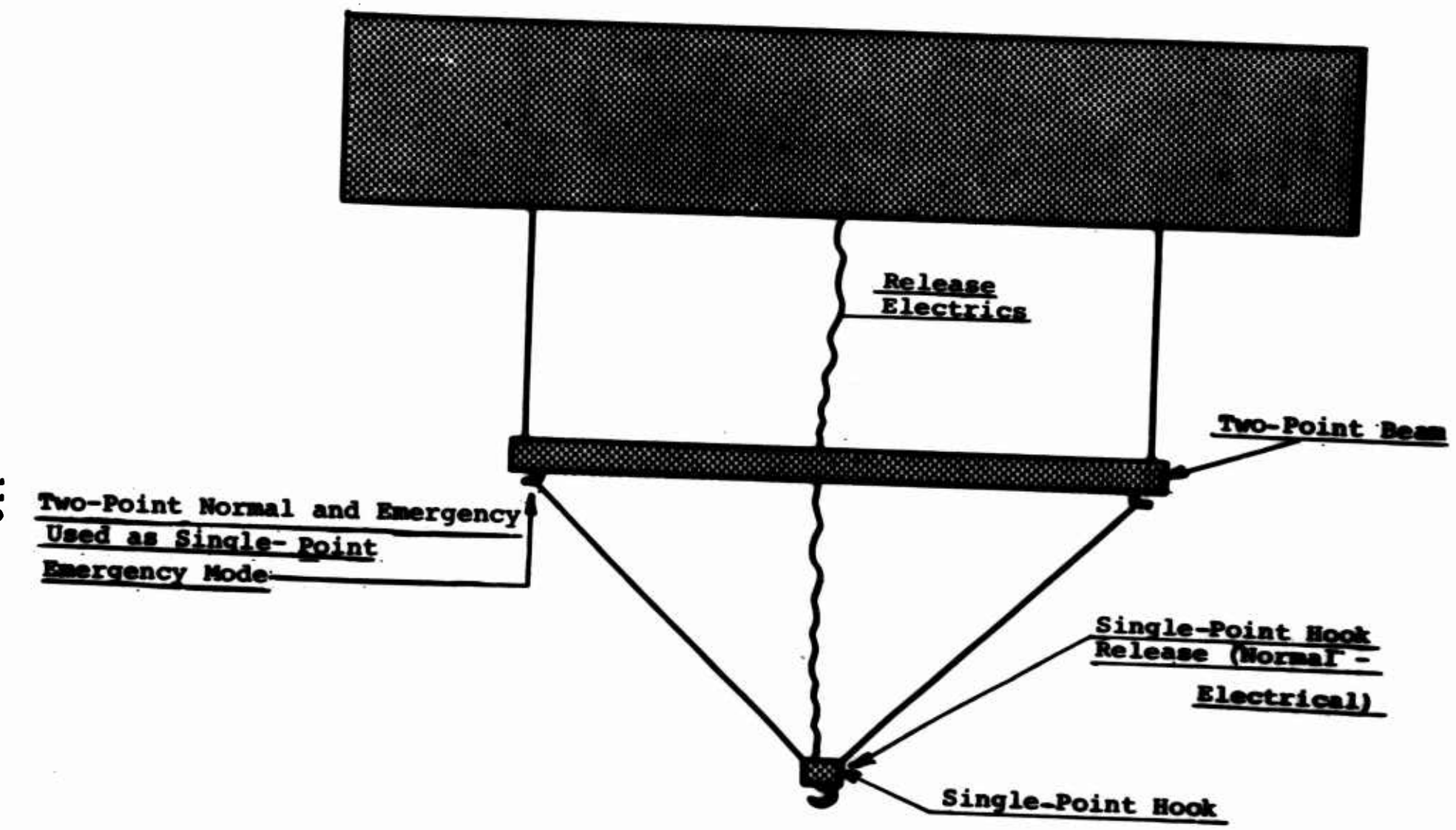


Figure 80. Single-Point Mode from Two-Point System

hoist system weight plus the elimination of a complete hoist system from inventory. However, if two separate systems are embodied, i.e., single-point plus multi-point, the helicopter can still be utilized as an external load carrier in the event that one system becomes inoperative. The question is, therefore, whether the weight and cost of two different hoist systems can be justified. A four-point system with beam could also be used in a single-point mode.

SYNCHRONIZATION OF MULTI-POINT HOIST SYSTEMS

Four-Point Suspension Background

Helicopter-mounted four-point suspension systems originated with the CH-54A crane-type aircraft, to meet the requirement to attach a pod-type load. Known as a "load leveler," this system can be operated only while the aircraft is on the ground.

The load leveler system consists of four hydraulic jacks, each one fixed at its upper end to the aircraft structure and at its lower end to an individual cargo-attaching device. Each cargo attaching device has two uses: first, it embodies lugs to which a pod may be rigidly bolted and, second, it mounts a lashing reel from which cable may be pulled and attached to the load. Excess cable may be wound back onto the reel, but this device is not used to hoist the load.

The load is attached while the aircraft is on the ground and is then lifted clear of the ground by the aforementioned hydraulic jacks (additional lift is obtained by means of a kneeling landing gear). When the cable from the lashing reels is used to attach a vehicle or other irregularly shaped load, it is possible for different lengths of cable to be required at each corner. In this case, the load is lifted by the four jacks and any slack in one cable is taken out by "beeping" the applicable jack control.

However, operational experience (reported by D. Harding and G. Wilson on a trip to Ft. Benning - 478th Flying Crane Company) with this system has indicated a need to continually trim the load suspension in flight. This is due to twisting of the airframe and, possibly, the load under different flight conditions. If the suspension is not trimmed and one cable is allowed to remain slack in flight, experience has shown that undesirable stability and flight safety conditions ensue. The load has a tendency to teeter about the diagonally opposite tight cable, giving rise to shock loads in the slack cables.

If the suspension cables are not parallel (and this generally will be the case, since most loads will have different hoisting point locations), aerodynamic-drag-induced longitudinal motions will couple with and induce motions in other directions, such as lateral swing, yaw and pitch motions. Generally, these individual oscillations will occur at different frequencies; thus, when added together, their net effect on the helicopter

is a random upsetting force. While it is possible for a pilot to control a simple periodic upsetting force, it is extremely difficult for him to control a random upsetting force. In a heavy lift helicopter, where the weight of the external load may approach that of the helicopter, it is imperative that the load dynamic behavior be both stable and predictable.

Quantitative analysis of the above phenomena is very complex and would be of questionable value without the inclusion of a pilot in the loop. Simulation of the condition would also be of questionable value unless the pilot could be subjected to the aircraft acceleration. It would seem, therefore, that the operational experience obtained with the CH-54A is the only reliable information that currently exists on handling qualities of helicopters with unevenly loaded four-point suspensions.

As previously stated, this experience indicates that such a loading condition is undesirable and should be avoided.

Mechanics of Four-Point Suspension

One-, two-, and three-point suspensions are statically determinate structures. That is to say, the load in each suspension leg is fixed and may be determined from knowledge of the load and the suspension geometry. Four-point, or more-than-four-point, suspensions are structurally redundant and therefore statically indeterminate. This means that the loads in the individual suspension legs may not be determined from knowledge of the load and suspension geometry. All that can be said of the loading is that the total load rolling and pitching moments may be established together with the equations governing the relationship between loads in the individual legs. Total solution of the problem and, hence, definition of the individual loads can be performed only by analysis of the system stiffness and compatibility conditions in the same way that a typical redundant aircraft structure is analyzed. This requires that the length, stiffness, and position of each structural element be known - a condition that is not easily satisfied for this problem.

It can be shown, however, that there are an infinite number of load distributions that can satisfy the problem boundary conditions, which are:

$$\Sigma V = o = A+B+C+D-40,000$$
$$\Sigma M_P = o = Aa+Ba-Ca-Da$$
$$\Sigma M_R = o = Ab-Bb+Db-Cb$$

This conclusion must be modified in the case of a cable suspension; since the rope cannot carry compression loads, the limiting case is when one cable becomes slack. Loadings that satisfy the above boundary conditions may vary as shown in Examples 1, 2, and 3, contained in Figures 81, 82, and 83 respectively.

It must be emphasized that <u>all loading distributions</u> within the range shown <u>entirely satisfy the equilibrium conditions.</u>

Complete understanding of the above analysis is a prerequisite to the design of four-point suspension systems.

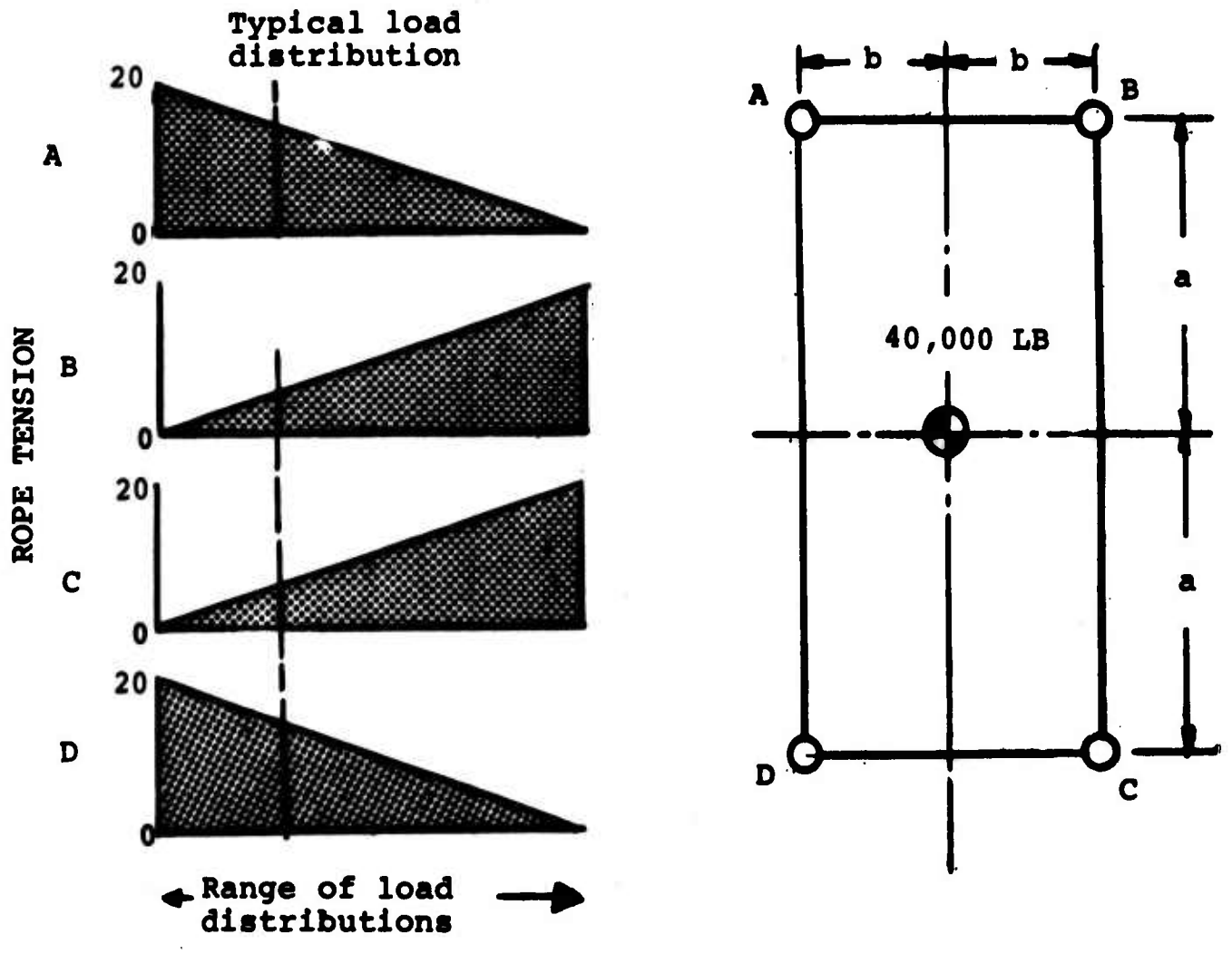


Figure 81. Load Distribution on Four-Point System (Example 1).

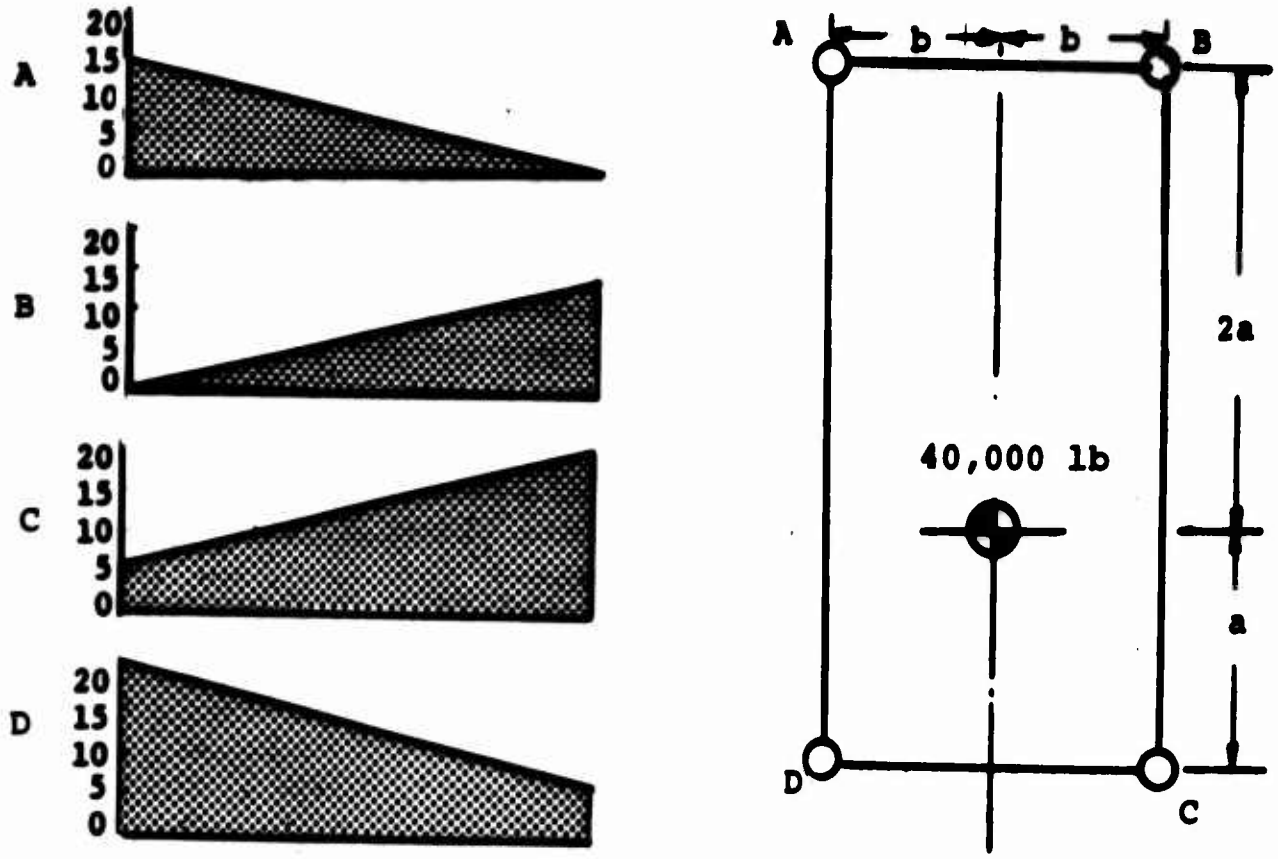


Figure 82. Load Distribution on Four-Point System (Example 2).

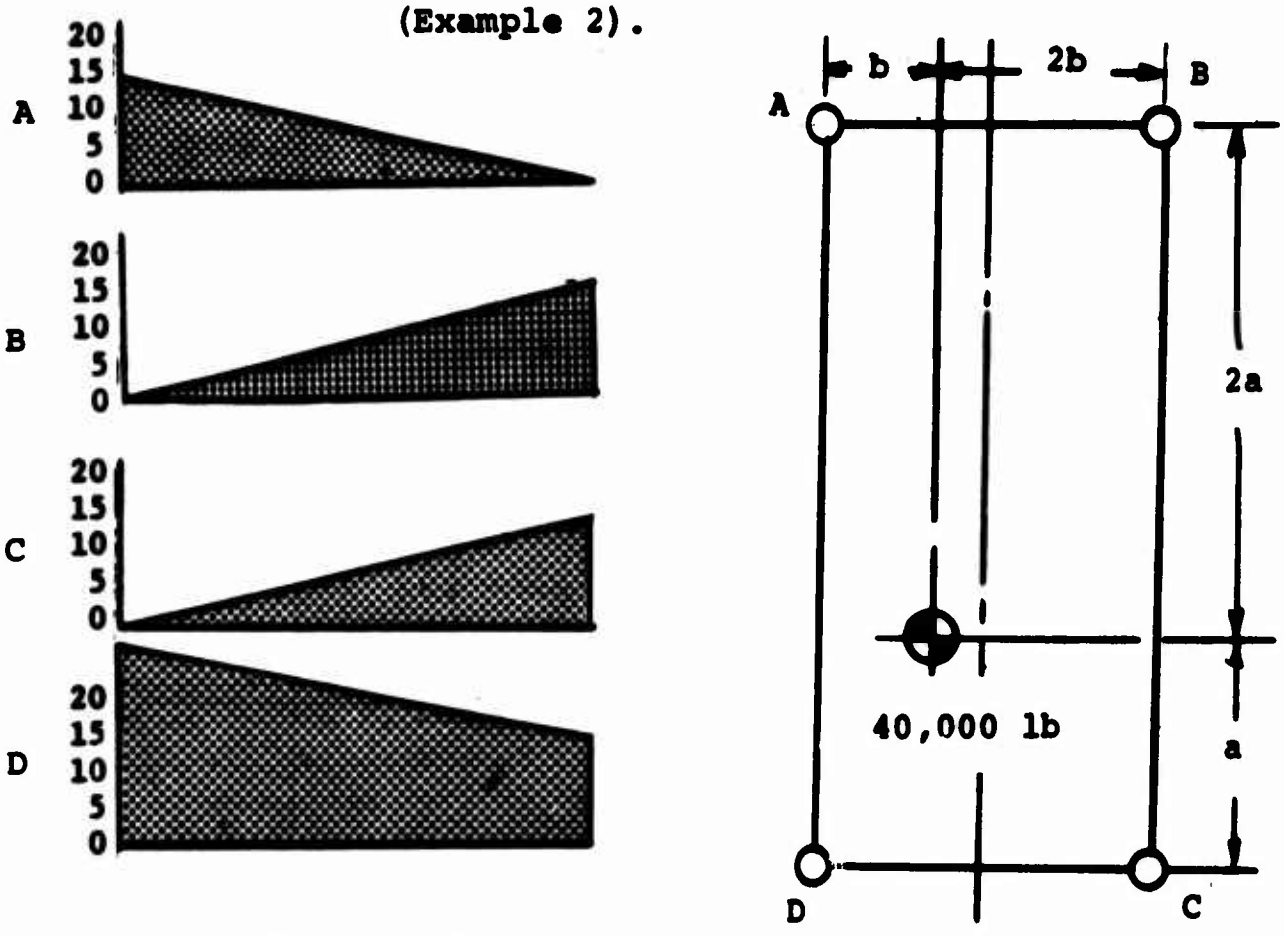


Figure 83. Load Distribution on Four-Point System (Example 3).

Changes of the load distribution within the possible extremes are effected by altering the length of one leg. The amount of load change for an incremental change in length is a function of system stiffness. If the system is very stiff, a small change in length will result in a large load change and vice versa. As previously stated, operational experience with the CH-54A load leveling system has indicated that it is difficult to maintain a trim condition that ensures that all cables remain tight. This is probably because the system is stiff, and changes in leg length, due to airframe windup in flight, result in slack ropes .

There are two solutions to this problem. The first is to make the system sufficiently soft to minimize the effect of airframe windup. However, if this is accomplished by adding equal linear springs in series with the individual legs, coupled bounce-roll-pitch vibrations can result. Provided the spring rate is sufficiently low, these will not be divergent in nature (they should not be confused with the vertical bounce phenomena), but they will give rise to undesirable aircraft-disturbing forces.

An acceptable solution may be found by using constant-height or self-leveling air springs, as fitted in typical truck and bus air suspension systems. The advantage in this technique is that the bounce frequency is maintained substantially constant irrespective of load or load distribution; thus, bounce-pitch-roll coupled motions due to varying spring rates are eliminated. Manual control of individual leg lengths is still required with this system, as the basic load distribution is still a function of leg length, although the trim sensitivity is reduced.

The analogy between four-point load suspension and road vehicle air suspension is interesting, since the problem of redundancy affects the vehicle in a similar way. In the early days of self-leveling air suspensions, each spring was fitted with a separate air supply and leveling valve. This resulted in a varying load and stiffness distribution, as described above for load suspensions. This situation was unacceptable to vehicle designers, so the solution they devised was to interconnect either the two front or the two aft springs.

The technique of equalizing the loads in two adjacent legs is the only logical way of automatically maintaining cable tension. It has been suggested that it is only necessary to ensure that all legs are carrying at least a nominal load and that this may be achieved by putting a minimum initial pressure into the

load-leveling jacks. No logical method exists, however, of deciding just what this nominal load should be. The fact that the lifted load may vary from zero to 40,000 pounds make this problem particularly acute.

The problems of an automatic load-leveling system are not quite as simple as those connected with road vehicle suspension since, in general, the vehicle designer knows which end of the vehicle is most heavily loaded. The importance of this is illustrated in Example 3 (Figure 83) above, where the cg is offset laterally. Load equalization can be achieved only at the lightly loaded end. Attempts to equalize the load at the heavy end may result in overturning of the load. It can be concluded, then, that if an automatic load leveling system is desired, it must include load-sensing devices and a control system which will enable the load to be lifted with the load equalization locked out in order to determine first the least heavily loaded end. The equalization mode may then be automatically selected. Operationally, this could take the form of a RAISE button at the loadmaster location which would first be pushed and held to acquire the load and then released to effect load equalization.

Four-Point Hoist System Synchronization

While the foregoing discussion was based on four-point load levelers, it applies equally to four-point hoist systems. In the following paragraphs three alternative systems for synchronized operation of a four-point hoist system are described. These are: (1) hydromechanical, (2) hydraulic, and (3) electrical winch drive and control systems.

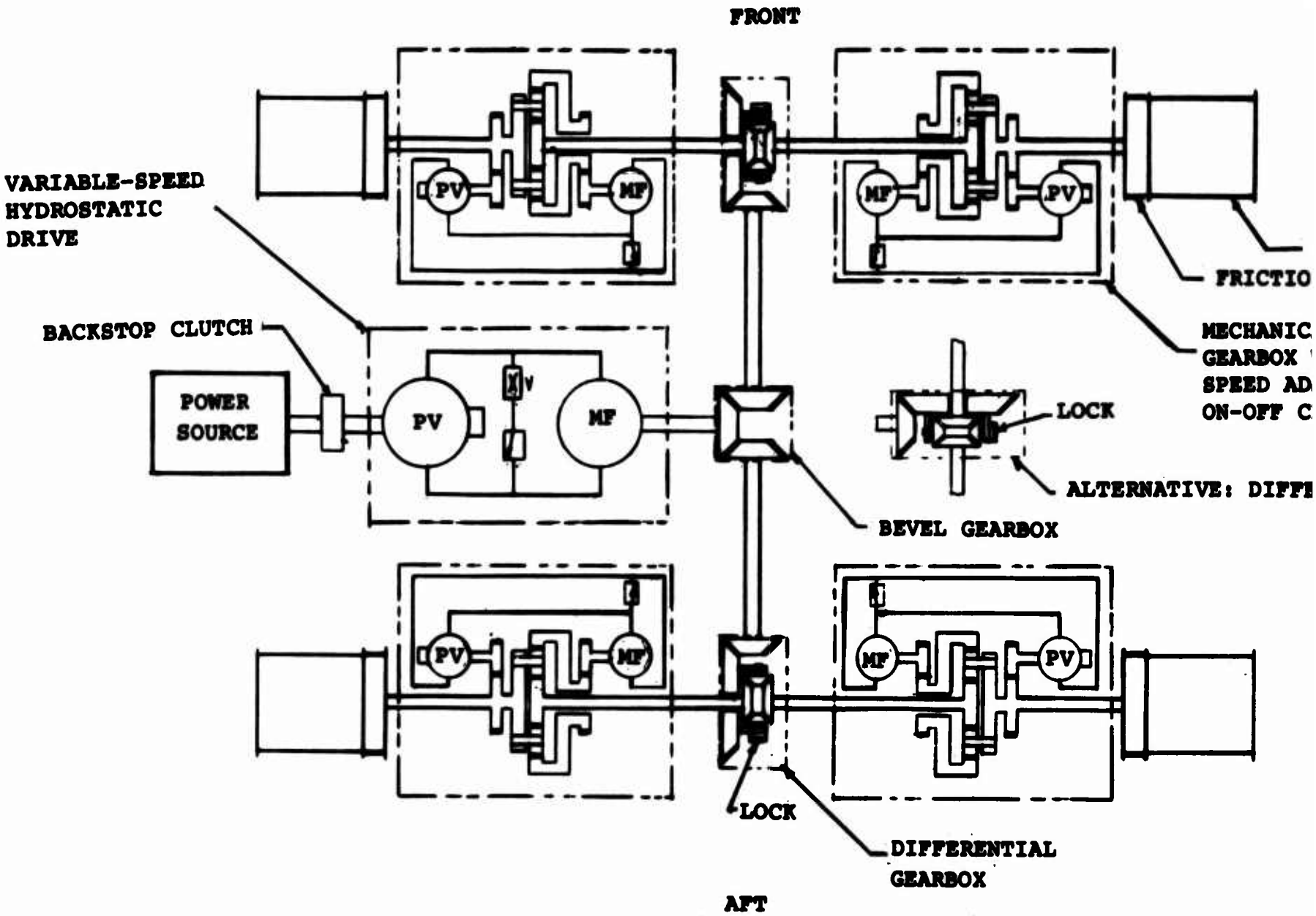
Hydromechanical Winch Drive and Control System for a Four-Point Hoist System

Two versions of a hydromechanical winch drive and control system suitable for a four-point hoist system are shown in Figures 84 and 85; these are:

1. A mechanical planetary gearbox with hydraulic speed adjustment and on-off clutch action.

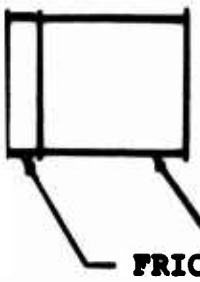
2. A differential gearbox with hydraulic differential speed adjustment and locking action.

In both cases, the object is to provide a drive from a common power source to all four winches through mechanical



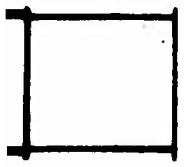
VERSION 1.

Figure 84. Hydromechanical Winch Drive and Control System for a Four-Point Hoist System.



SEE NOTE

RNATIVE: DIFFERENTIAL GEARBOX



ERSION 1.

NOTE: WHEN THE HYDRAULIC PUMP IS SET TO ZERO DISPLACEMENT, THE FLUID IN THE CIRCUIT IS TRAPPED AND THE HYDRAULIC MOTOR IS STATIONARY. THIS KEEPS FIXED THE RING GEAR OF THE PLANETARY GEARING AND THE SET WORKS AS A FIXED-RATIO GEARING. ANY OTHER PUMP STROKE SETTING WILL CAUSE ROTATION OF THE MOTOR AND SPEED INPUT TO THE RING GEAR (IN EITHER DIRECTION OF ROTATION WHEN THE PUMP IS OF OVER-CENTER TYPE). THIS CHANGES THE GEARING RATIO AND ALTERS THE OUTPUT SPEED.

WHEN THE BYPASS VALVE IS OPEN, THE HYDRAULIC MOTOR DOES NOT TRANSMIT TORQUE, THE RING GEAR IS FREE TO ROTATE AND SO IS THE WINCH DRUM WHEN ITS OWN BRAKE IS RELEASED.

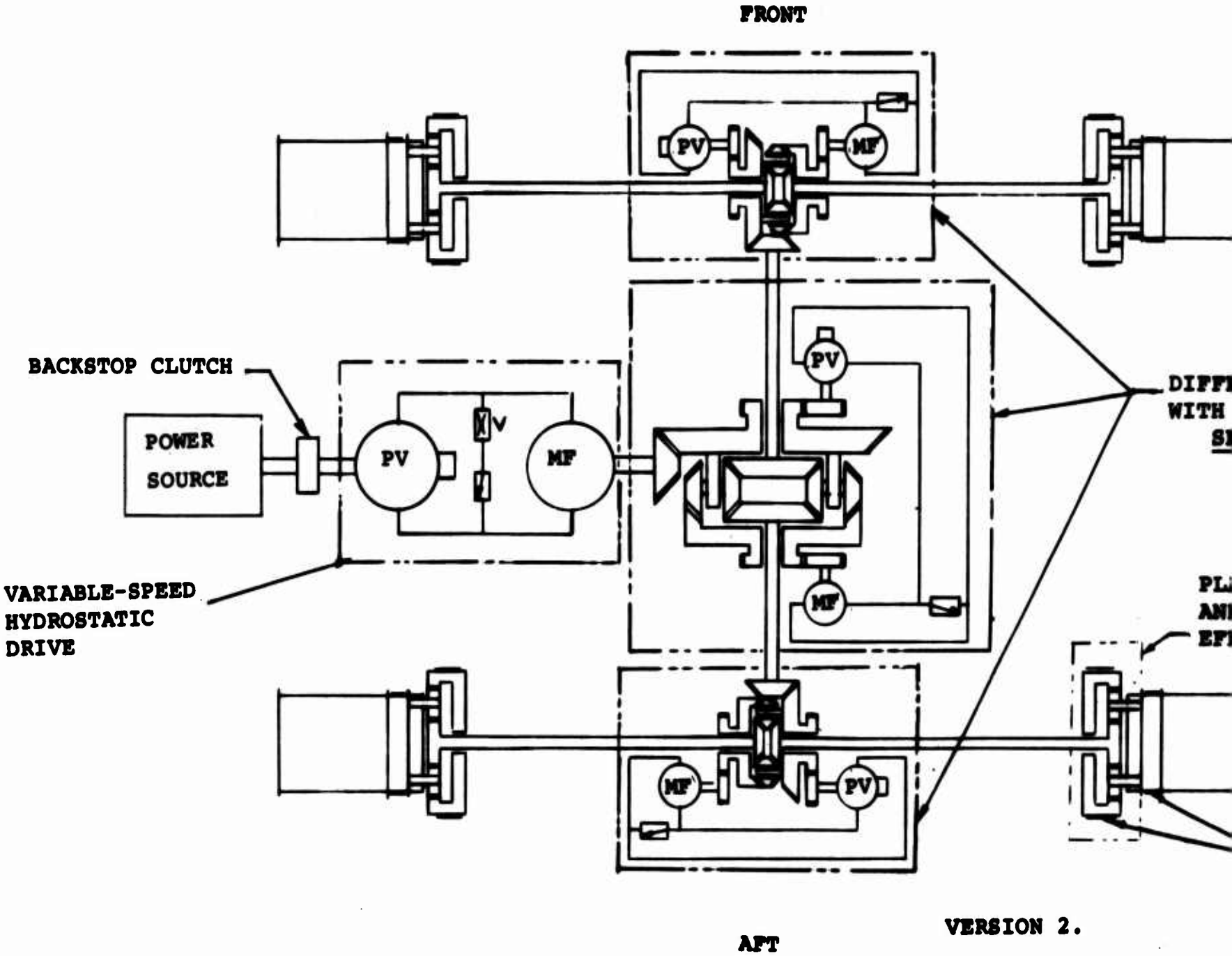
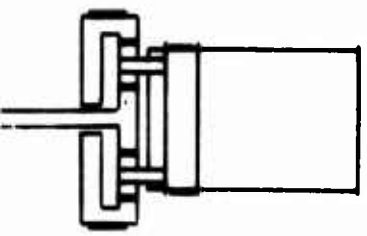
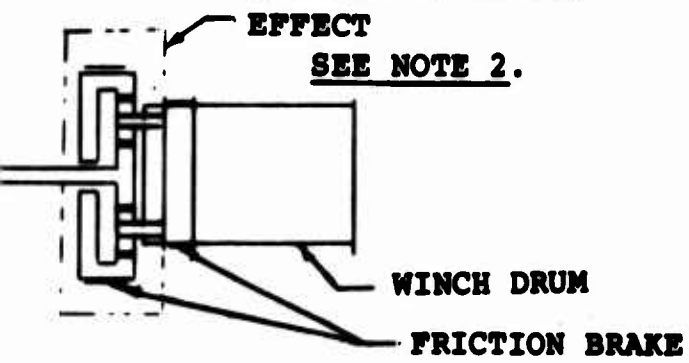


Figure 85. Alternative Scheme of a Hydromechanical Winch Drive and Control System for a Four-Point Hoist System.



DIFFERENTIAL GEARBOX
WITH HYDRAULIC CONTROL
SEE NOTE 1.

PLANETARY GEARING
AND ON-OFF CLUTCH
EFFECT
SEE NOTE 2.



N 2.

NOTES: 1. AT ONE PARTICULAR PUMP STROKE CONTROL SETTING, THE EFFECT OF THE HYDRAULIC BIAS IS SUCH AS TO HAVE THE BEVEL RING GEAR (IN MESH WITH THE PLANETS) AND THE PLANET CARRIER TO ROTATE AT THE SAME SPEED. THE DIFFERENTIAL GEARING IS THUS HYDRAULICALLY LOCKED AND BOTH OUTPUT SHAFTS ROTATE AT THE SAME SPEED. AT ANY OTHER PUMP STROKE SETTING, THE OUTPUT DIFFERENTIAL MOTION WILL BE INDUCED. THIS CAN BE PROGRESSIVELY CHANGED. THE HYDRAULIC CONTROL HAS NO EFFECT WHEN THE BYPASS VALVE IS OPEN.

2. WHEN THE RING GEAR IS KEPT STATIONARY BY THE BRAKE, THE PLANETARY GEARING WORKS AS A FIXED RATIO GEARING. WHEN THE BRAKE IS RELEASED, NO OUTPUT TORQUE IS TRANSMITTED TO THE CABLE DRUM, WHICH IS FREE TO ROTATE WHEN ITS OWN BRAKE IS RELEASED.

2

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gearing of equal ratios which will normally give synchronized motion but which will have the superimposed capability of adjusting the ratio of any of the winch drives by hydraulic means wherever required. The power source in both cases consists of:

1. A power source
2. A backstop clutch
3. A variable-displacement (overcenter) hydraulic pump
4. A fixed-displacement hydraulic motor
5. A hydraulic circuit interconnecting pump and motor, including a single-flow-path valve (normally closed) and a variable restriction
6. An auxiliary hydraulic circuit, including a hydraulic control-pressure pump and servo valve used as a means of powering the variable-delivery pump stroke-control cylinder. (This circuit is not shown on the schematic diagrams.)

The power input provides means for variable-speed drive in both directions. Moreover, the motor bypass with restrictor (in conjunction with the backstop clutch) provides means for dynamic braking of the winches in addition to that provided by the mechanical friction brakes located on all winch drums.

The two different versions of this drive concept are described in the following paragraphs:

1. <u>Mechanical Planetary Gearbox Drive</u>

The output of the common variable-speed hydrostatic drive hydraulic motor is conveyed to two fore-and-aft branch shafts through a bevel gearbox (or differential gearbox). The torque of each fore-and-aft branch shaft is transmitted, in turn, into two transverse shafts by differential gearboxes. Each transverse shaft drives its winch drum through a mechanical planetary gearbox with an additional hydrostatic drive coupled by means of spur gears

between the ring gear and the planet carrier of the planetary gearing. A variable-displacement pump is driven from the planet carrier, and a fixed-displacement motor is driven from the ring gear.

If the pump is set to zero-stroke position, the fluid in the circuit is locked and the motor is then stationary. The ring gear is fixed and the output motion of the planetary gearing is strictly in accordance with the mechanical gear ratio. When the hydraulic pump is set to a position other than zero-stroke, fluid flow will cause motor rotation and also rotation of the ring gear and alteration of the output shaft speed.

By this means, the speed of any of the drums can be altered (increased or decreased) with respect to the other drums. Speed is controlled through a beep control.

When the hydraulic bypass valve (normally closed) is open, no torque would be transmitted by the motor. The ring gear would thus be free and no torque would be transmitted by the shaft driving the drum.

As mentioned before, the bypass valve and the variable restrictor in the prime mover hydrostatic transmission are used for dynamic braking; however, they can also be used for limiting the output torque of the hydraulic motor for initial pre-tensioning of winch cables prior to load lifting.

When the load is lifted and the bevel planets are unlocked in the differential gearbox on the transverse shaft, the torque can be kept equalized on both right and left branch shafts, thus contributing to the load stability.

2. Differential Gearbox Drive

In this case three differential gearboxes with an additional hydrostatic coupling in each are provided. The first is connected to the power source system hydraulic motor output shaft and drives the two fore-and-aft branch shafts. Each of the fore-and-aft shafts drives, in turn (through its differential gearbox), two

transverse branch shafts used to drive the individual winches.

The hydraulic coupling of each differential gearbox is such that a variable-displacement pump is driven by a pair of spur gears from the large bevel gear (planet carrier), and the fixed-displacement motor is similarly driven from a bevel ring gear which is in mesh with bevel planets.

By adjusting the pump displacement relative to the fixed motor displacement, it is possible to make (1) both output shafts operate at synchronized speed or (2) one of the shafts run progressively faster or slower than the other.

This enables both front winches to be driven at a speed equal to, higher than, or lower than that of the two aft winches; also, any of the right winches can be driven at the same speed, or at higher or lower speed than that of the corresponding left winch.

Additional simple planetary gearing is coupled to each winch drum so that the planet carrier is fixed to the drum and the sun gear is keyed to the drive shaft from the differential gearbox. The ring gear outer cylinder serves as a drum for a friction brake. When the ring gear is kept stationary by application of this brake, the planetary gearing acts as a mechanical reduction gearing.

When the brake is free, no torque is transmitted to the winch drum. If the winch is holding a load, the load can be lowered by releasing both friction brakes; i.e., the one on the ring gear drum and the one on the winch drum. By means of the foregoing arrangement, it is thus possible to operate any one of the four winches separately.

Pre-tensioning of cables prior to load hoisting can also be accomplished with this arrangement as described above under Mechanical Planetary Gearbox Drive. Similarly, the equalization of torque on corresponding transverse branch shafts is possible, as mentioned above. Moreover, the fore-and-aft branch shafts can also have their torque balanced through the differential gearbox. In this case, the unlocking of the differential shafts

is achieved, under static conditions, by opening the hydraulic motor bypass valve.

The auxiliary hydraulic circuit for control purposes is not shown on the diagram. It includes at least one hydraulic pump and four servovalves. In operation, any one of the stroke-control cylinders can be actuated on each of the variable-displacement pumps. Remote control of the bypass valves and friction brakes will be provided. The entire control system will also include means of obtaining a closed-loop control circuit.

Hydraulic Winch Drive and Control System for a Four-Point Hoist System

In this drive system (see Figure 86), each of the four winches is individually driven by a hydraulic fixed-displacement motor through a reduction gearbox. A suitable hydraulic circuit supplies the fluid to power the motors. Fluid flow to each motor is controlled by an electrohydraulic servo valve used in conjunction with a closed-loop control circuit.

The power spool of the servo valve is of the four-way type, capable of directing the fluid pressure selectively to either of the two motor ports, to have power operation in both the lifting and lowering modes. A one-way restrictor may be used in that motor port which serves as the inlet port in load lifting, to prevent motor cavitation by restricting the return flow during the lowering of the load. In addition, a safety control circuit (not shown on the schematic) will be used; it will operate the friction brakes (provided on each drum) when required.

The input to the servo valve is in the form of an electric signal generated by a linear variable differential transformer (LVDT) unit, operated by the pilot or loadmaster, in which a mechanical displacement is translated into a proportional electrical signal. The feedback signal is generated by a tachometer unit driven from the winch drum shaft through suitable gearing; the feedback signal is, thus, proportional to the drum rotary speed (lifting or lowering rate). The direction and the speed of drum rotation are related to the direction and magnitude of the pilot's control lever displacement from its neutral position.

1. CABLE
2. FRICTI
3. REDUCT
4. HYDRAÚ
5. ELECTR
6. ELECTR
7. TACHOM
8. TORQUE
9. LOGIC

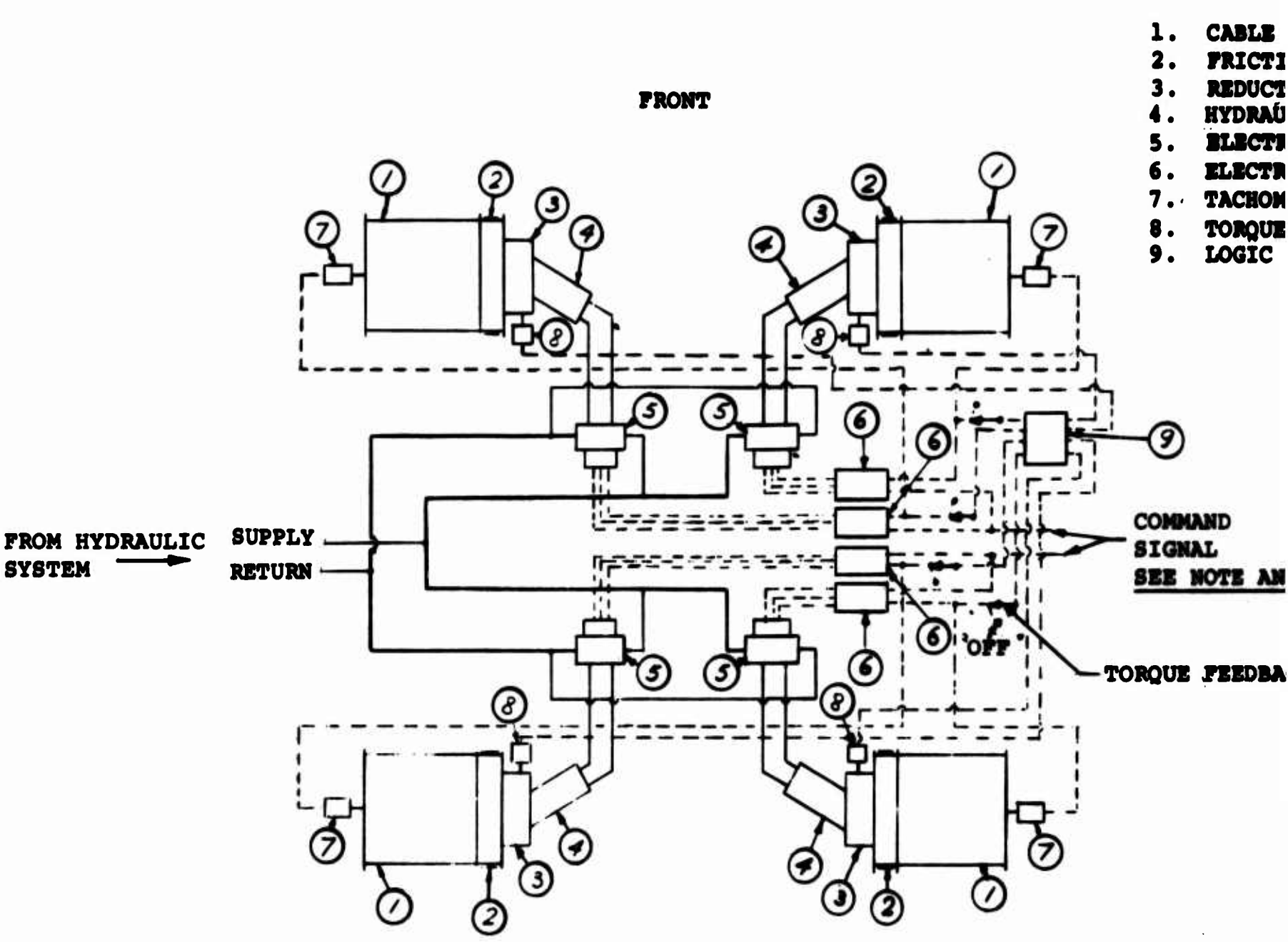


Figure 86. Hydraulic Winch Drive and Control System for a Four-Point Hoist System.

1. CABLE DRUM
2. FRICTION BRAKE
3. REDUCTION GEARBOX
4. HYDRAULIC MOTOR, FIXED-STROKE
5. ELECTROHYDRAULIC SERVO VALVE
6. ELECTRONIC AMPLIFIER
7. TACHOMETER
8. TORQUE SENSOR
9. LOGIC BOX

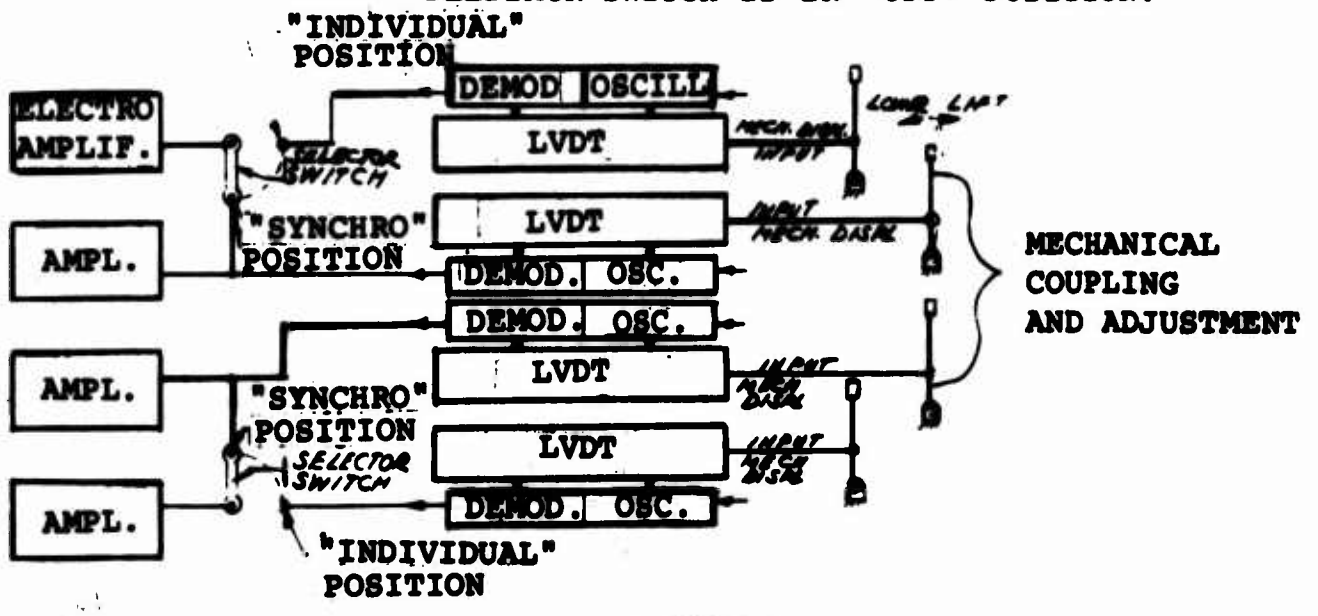
(9)

COMMAND
SIGNAL
SEE NOTE AND DETAIL

ORQUE FEEDBACK SWITCH

NOTE: THE FOLLOWING CONTROL MODES ARE AVAILABLE:

1. SYNCHRONIZED OPERATION
   IN THIS MODE THE TWO SERVO VALVES OF THE FRONT WINCH DRIVE CIRCUIT RECEIVE A COMMON INPUT SIGNAL FROM ONE SIGNAL SOURCE. SIMILARLY , THE TWO SERVO VALVES OF THE AFT CIRCUIT RECEIVE A COMMON INPUT SIGNAL FROM SECOND SIGNAL SOURCE. THE SYNCHRONIZATION OF FRONT AND AFT PAIRS OF WINCHES IS BY MECHANICAL COUPLING OF THE CORRESPONDING CONTROL LEVERS (SEE DETAIL): THIS IS LESS CRITICAL.
2. SYNCHRONIZED IN PAIRS ONLY
   TWO SYNCHRONIZED FRONT WINCHES CAN BE OPERATED INDEPENDENTLY FROM TWO SYNCHRONIZED AFT WINCHES (UNCOUPLED CONTROL LEVERS).
3. INDIVIDUAL WINCH OPERATION
   ANY ONE WINCH CAN BE OPERATED SEPARATELY (CONTROL LEVERS UNCOUPLED AND SELECTOR SWITCH ON "INDIVIDUAL"). IN THIS MODE THE FEEDBACK IS LIMITED TO RATE FEEDBACK SIGNAL; TORQUE FEEDBACK SWITCH IS IN "OFF" POSITION.



DETAIL

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In the four-point hoist system, it is mandatory to have the following operation modes:

1. Synchronized operation of all four winches
2. Synchronized operation of the front two winches and synchronized operation of the aft two winches, but with each pair controlled independently of the other
3. Independently controlled operation of each one of the four winches

The described system uses an identical input signal to the servo valves of both front winches and another identical signal to the servo valves of the aft two winches. This is to comply with the second requirement (synchronized pairs). The synchronization between both pairs of winches is not so critical as between the winches in a given pair and can be achieved by a mechanical coupling of the two control levers (one for each pair) and by operating the two controls together. Some means of adjustment could be provided to locate one control relative to the other so as to have equal signals prior to coupling them together. For individual operation, a separate LVDT unit is used for each winch.

In order to obtain the necessary loading distribution between winches, as described on page 158 of this section, it is necessary to provide a feedback control system to effect the functions shown in the logic box of the schematic on Figure 86; these are as follows:

1. Summing of the torque values of the two front winches
2. Summing of the torque values of the two aft winches
3. Determining the more lightly loaded end
4. Determining the torque differential
5. Applying the torque differential signal to the rate signal with appropriate sign, according to conditions

Function 5 is achieved by superimposing a differential torque feedback signal to the winches at the lightly loaded end so as to cause torque equalization.

For individual winch control, two additional LVDT units are switched on by positioning two selector switches to INDIVIDUAL. The differential torque feedback is not required in this case and is eliminated by positioning the four feedback switches to OFF; the rate feedback is maintained.

Electrical Winch Drive and Control System for a Four-Point Hoist System (Refer to Table VI)

Whether an electrical adjustable-speed drive for the hoist system (see Figure 87) becomes a practical proposition or not depends on the availability of high-power low-weight components specially developed for aircraft application. Moreover, solutions must be available to such basic problems as the capability for a gradual torque build-up, as required when starting to hoist a load, and the capability of holding steady an acquired load. Furthermore, the system selected must be able to meet the following operational requirements:

1. Synchronized operation of all four winches

2. Synchronized operation of the front two winches and synchronized operation of the aft two winches, but with each pair controlled independently of the other

3. Independently controlled operation of each one of the four winches

The fact that a four-point hoist system is a redundant system imposes an additional requirement on the control system. This is the prevention of slackness in one rope . When one end of the hoist system (front or aft) is more loaded than the other and, at the same time, one side (left or right) is also more loaded than the other, slackness in one of the cables at the lightly loaded end can occur, even if all the winches are synchronized. This condition is undesirable and can be prevented either by using an intricate control system or by using transverse shafts between the winches with a differential gearbox in the middle of each shaft. The system shown on the schematic diagram (Figure 87) uses the transverse shaft method.

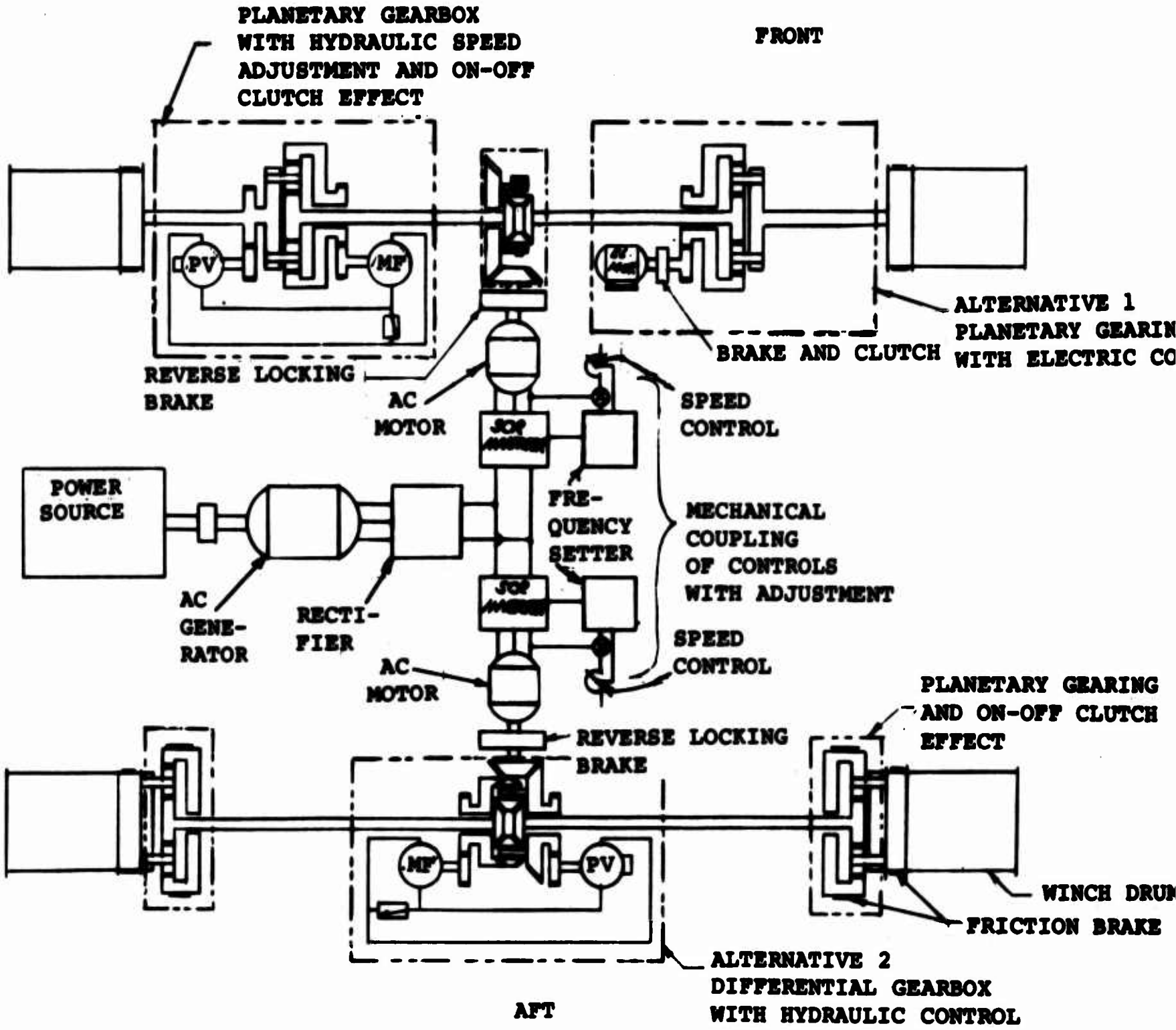
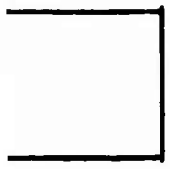


Figure 87. Electromechanical Winch Drive and Control System for a Four-Point Hoist System.



ALTERNATIVE 1
PLANETARY GEARING
:H WITH ELECTRIC CONTROL

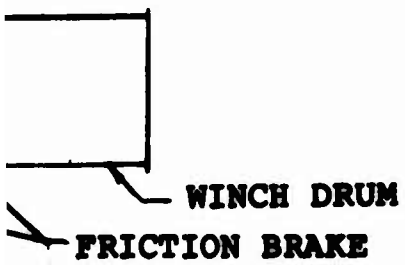
NOTE: THE FOLLOWING CONTROL MODES ARE AVAILABLE:

1. SYNCHRONIZED OPERATION
   IN THIS MODE THE TWO SPEED CONTROLS OF AC MOTORS ARE MOVED TOGETHER USING A MECHANICAL COUPLING. HYDRAULIC (OR ELECTRIC) CONTROLS OF MECHANICAL GEARING ARE SET SUCH AS TO HAVE FIXED GEARING OF EQUAL RATIOS FOR BOTH HALVES OF EACH TRANSVERSE SHAFT.

2. SYNCHRONIZED IN PAIRS ONLY
   TWO SYNCHRONIZED FRONT WINCHES CAN BE OPERATED INDEPENDENTLY FROM TWO SYNCHRONIZED AFT WINCHES. TWO AC MOTOR SPEED CONTROLS ARE UNCOUPLED. HYDRAULIC (ELECTRIC) CONTROLS AS IN POINT 1. ABOVE.

3. INDIVIDUAL OPERATION
   ANY ONE OF FOUR WINCHES CAN BE OPERATED OR STOPPED SEPARATELY AND ITS SPEED CAN BE ADJUSTED RELATIVE TO THE OTHER WINCH IN PAIR. HYDRAULIC (OR ELECTRIC) CONTROLS ARE USED TO CHANGE THE GEARING RATIO AND TO PRODUCE THE ON-OFF CLUTCH EFFECT.

PLANETARY GEARING
AND ON-OFF CLUTCH
EFFECT



ARBOX
CONTROL

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The purpose of the illustrated scheme is to show how an electrical adjustable-speed drive can be conveniently applied to the four-point hoist system, rather than to state which of the many existing and possible electrical drives is the most suitable for the given application. For instance, two static cycloconverters might be considered in lieu of the rectifier and the two silicon control rectifier (SCR) inverters shown on the diagram. However, an electrical system using AC induction motors and static solid-state frequency converters was considered preferable to the system using dc motors.

In the described system a power source drives, at a constant speed, an electric AC generator which supplies power to two AC induction motors, one of which serves to drive the two front winches and the other the two aft winches. A static solid-state frequency-adjusting system with one separate speed-setting rheostat for each AC motor is used as a control means. A mechanical coupling of the rheostat controls (with provision for pre-adjustment) is used for synchronized operation of all four winches. The controls are uncoupled for separate operation of each pair of winches. Additional control means are provided for individual adjustment of the speed of one winch relative to that of the other in the same pair. This is achieved by separate hydraulic or electrical control of mechanical gearing on a transverse drive shaft. A planetary gearbox and a differential gearbox are shown as alternatives. The planetary gearbox also provides the means to have on-off clutch effect. Thus, selective operation or immobilization of any of the four winches is possible.

A reverse locking brake is shown immediately downstream of each of the two AC motors. The purpose of this brake is to assure the load holding capability of the system, which is power operated for both lifting and lowering of the load. This brake is in addition to the friction brake on each cable drum; it can be dispensed with if the electrical drive system finally selected proves to have sufficient load-holding capability.

TABLE VI

ESTIMATED RELATIVE WEIGHT AND COST OF FOUR-POINT WINCH DRIVE AND CONTROL SYSTEMS

	Relative Weight %	Relative Cost %	Within State of Art	Equipment Available
Hydromechanical	365	100	Yes	Partly
Hydraulic	100	150	Yes	Partly
Electrical	475	200	Yes	Partly

Component Availability

Hydromechanical Drive - A lightweight variable-delivery, overcenter hydraulic pump, withservo control, in the 100-horsepower class, and suitable for aircraft application is not available; industrial pumps are available but are heavy.

Hydraulic Drive - A very intricate control system is required for satisfactory performance. It is not available and will have to be developed.

Electrical Drive - An AC induction motor in the 50-horsepower class, for aircraft application, is not available. For solid-state cycloconverters, in which power rating depends on ambient temperature, a forced-air circulation system for cooling is required.

Summary - Four-Point Winch Drive and Control Systems

The hydromechanical four-point winch drive system is preferable from the point of view of performance characteristics, safety, and ruggedness.

Two-Point Hoist System Synchronization

Synchronization of a two-point hoist system is not mandatory since any misalignment occurring during hoisting can be trimmed out by means of an individual variable-speed control to each winch. However, embodiment of a synchronized system

could result in lower acquisition times with resultant increase in system productivity.

The following paragraphs contain descriptions of three alternative systems for synchronized operation of a two-point hoist system. These consist of (1) hydromechanical, (2) hydraulic, and (3) electrical winch drive and control systems.

Hydromechanical Winch Drive and Control System for Two-Point Hoist System

Two versions of a hydromechanical winch drive and control system for a two-point hoist system are:

1. A mechanical planetary gearbox with hydraulic speed adjustment and on-off clutch action

2. A differential gearbox with hydraulic differential speed adjustment and locking action

In both cases the object is to provide a drive from a common power source to the two winches through mechanical gearing of equal ratios which will normally give synchronized motion but which will have the superimposed capability of adjusting the ratio of either winch drive by hydraulic means whenever required.

The power source in both cases consists of:

1. A power source

2. A backstop clutch

3. A variable-displacement (overcenter) hydraulic pump

4. A fixed-displacement hydraulic motor

5. A hydraulic circuit interconnecting pump and motor and including a single-flow-path valve (normally closed)

6. An auxiliary hydraulic circuit, including a hydraulic control-pressure pump and servo valve as a means of powering the variable-delivery pump stroke-control cylinder. This is not shown on the

schematic diagrams.

The power source provides means for variable-speed drive in both directions. Moreover, the motor bypass with restrictor (in conjunction with the backstop clutch) provides means for dynamic braking of winches in addition to the mechanical friction brakes located on all winch drums.

Two different versions of this drive concept are shown in Figure 88 and are described in the following paragraphs.

1. Mechanical Planetary Gearbox Drive

The output of the common variable-speed hydrostatic drive hydraulic motor is conveyed to two fore-and-aft branch shafts through a differential gearbox. Each branch shaft drives its winch drum by means of a mechanical planetary gearbox, with an additional hydrostatic drive coupled by means of spur gears between the ring gear and the planet carrier of the planetary gearing. A variable-displacement pump is driven from the planet carrier, and a fixed-displacement motor is driven from the ring gear.

If the pump is set to zero-stroke position, the fluid in the circuit is locked and the motor is then stationary. The ring gear is fixed and the output motion of the planetary gearing is strictly in accordance with the mechanical gear ratio. When the hydraulic pump is set to a position other than zero-stroke, fluid flow will cause motor rotation and also rotation of the ring gear and corresponding alteration of the output shaft speed.

By this means, the speed of each drum can be increased or decreased with respect to the other drum. Speed is controlled through a beep control.

When the hydraulic bypass valve (normally closed) is open, no torque would be transmitted by the motor. The ring gear would thus be free and no torque would be transmitted by the shaft driving the drum.

The bypass valve and the variable restrictor in the prime mover hydrostatic transmission are used for

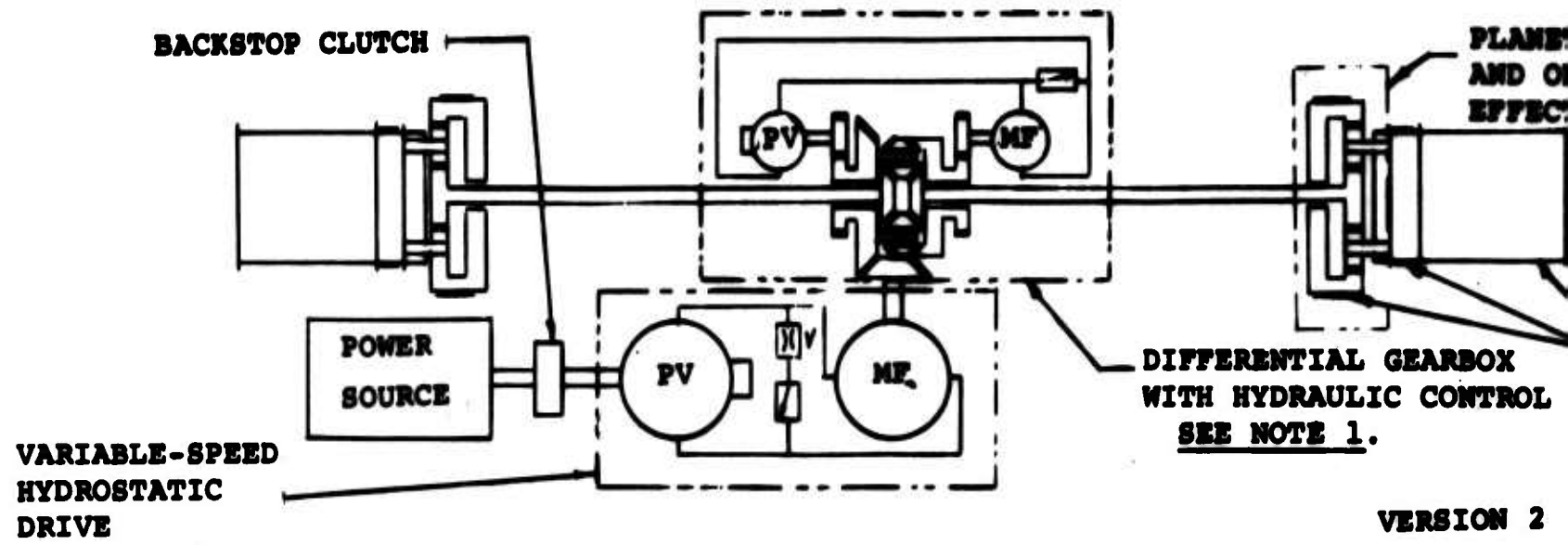
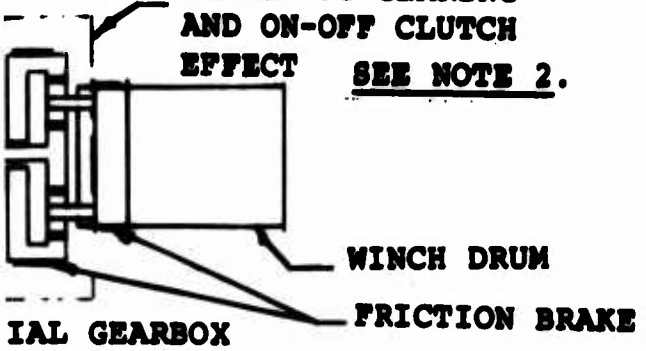


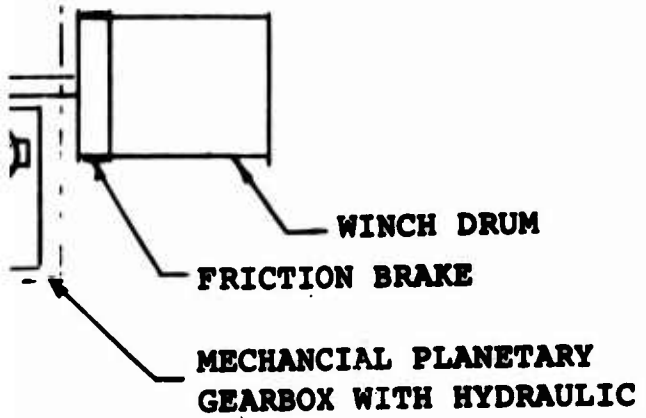
Figure 88. Hydromechanical Winch Drive and Control System for a Two-Point Hoist System.



IAL GEARBOX
AULIC CONTROL
E 1.

VERSION 2

X



VERSION 1

NOTES:

1. AT ONE PARTICULAR PUMP STROKE CONTROL SETTING, THE EFFECT OF THE HYDRAULIC BIAS IS SUCH AS TO HAVE THE BEVEL RING GEAR (IN MESH WITH THE PLANETS) AND THE PLANET CARRIER TO ROTATE AT THE SAME SPEED. THE DIFFERENTIAL GEARING IS THUS HYDRAULICALLY LOCKED AND BOTH OUTPUT SHAFTS ROTATE AT THE SAME SPEED. AT ANY OTHER PUMP STROKE SETTING THE OUTPUT DIFFERENTIAL MOTION WILL BE INDUCED. THIS CAN BE PROGRESSIVELY CHANGED. THE HYDRAULIC CONTROL HAS NO EFFECT WHEN THE BYPASS VALVE IS OPENED.

2. WHEN THE RING GEAR IS KEPT STATIONARY BY THE BRAKE, THE PLANETARY GEARING WORKS AS A FIXED-RATIO GEARING. WHEN THE BRAKE IS RELEASED, NO OUTPUT TORQUE IS TRANSMITTED TO THE CABLE DRUM, WHICH IS FREE TO ROTATE WHEN ITS OWN BRAKE IS RELEASED.

3. WHEN THE HYDRAULIC PUMP IS SET TO ZERO DISPLACEMENT, THE FLUID IN THE CIRCUIT IS TRAPPED AND THE HYDRAULIC MOTOR IS STATIONARY. THIS KEEPS FIXED THE RING GEAR OF THE PLANETARY GEARING, AND THE SET WORKS AS A FIXED-RATIO GEARING. ANY OTHER PUMP STROKE SETTING WILL CAUSE ROTATION OF THE MOTOR AND SPEED INPUT TO THE RING GEAR (IN EITHER DIRECTION OF ROTATION WHEN THE PUMP IS OF OVERCENTER TYPE). THIS CHANGES THE GEARING RATIO AND ALTERS THE OUTPUT SPEED.

   WHEN THE BYPASS VALVE IS OPEN, THE HYDRAULIC MOTOR DOES NOT TRANSMIT TORQUE, THE RING GEAR IS FREE TO ROTATE, AND SO IS THE WINCH DRUM WHEN ITS OWN BRAKE IS RELEASED.

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dynamic braking, as mentioned before; however, they can also be used for limiting the output torque of the hydraulic motor for initial pre-tensioning of winch cables prior to load lifting.

When the load is lifted and the bevel planets are unlocked in the differential gearbox, the torque can be kept equalized on both fore-and-aft branch shafts; this will contribute to load stability.

2. Differential Gearbox Drive

In this case, a differential gearbox with an additional hydrostatic coupling is used. The input shaft of the differential gearbox is coupled to the shaft of the power unit hydraulic motor. Each of the two output shafts is used to drive its respective winch through additional planetary gearing, as described below.

The hydraulic coupling of the differential gearbox is such that a variable-displacement pump is driven by a pair of spur gears from the large bevel gear (planet carrier), and the fixed-displacement motor is similarly driven from a bevel gear which is in mesh with bevel planets.

By adjusting the pump displacement relative to the fixed motor displacement, it is possible (1) to make both output shafts operate at synchronized speed or (2) to make one of the shafts run progressively faster or slower than the other.

Additional simple planetary gearing is coupled to each winch drum so that the planet carrier is fixed to the drum and the sun gear is keyed to the drive shaft from the differential gearbox. The ring gear outer cylinder serves as a drum for a friction brake. When the ring gear is kept stationary by application of this brake, the planetary gearing acts as a mechanical reduction gearing. When the brake is free, no torque is transmitted to the winch drum. If the winch is holding a load, the load can be lowered by releasing both friction brakes; i.e., the one on the ring gear drum and the one on the winch drum.

By means of the foregoing arrangement, it is thus

possible to operate either of the winches separately.

Pre-tensioning of cables prior to load hoisting can also be accomplished with this arrangement, as described above under Mechanical Planetary Gearbox Drive. Similarly, the equalization of torque on branch shafts is possible, as was mentioned above. In this case, the unlocking of the differential shafts is achieved, under static conditions, by opening the hydraulic motor bypass valve.

The auxiliary hydraulic circuit for control purposes is not shown on the diagram. It includes one hydraulic pump and one servo valve. It works by actuating the stroke control cylinder on the variable-displacement pump. Remote control of the bypass valve and friction brakes will be provided. The entire control system will also include means of obtaining a closed-loop control circuit.

Hydraulic Winch Drive and Control System for Two-Point Hoist System

In this drive system (see Figure 89), each winch is individually driven by a hydraulic fixed-displacement motor through a reduction gearbox. A suitable hydraulic circuit supplies the fluid to power the motors. Fluid flow to each motor is controlled by an electrohydraulic servo valve used in conjunction with a closed-loop control circuit.

The power spool of the servo valve is of the four-way type, capable of directing the fluid pressure selectively to either of the two motor ports, to have power operation in both the lifting and lowering modes. A one-way restrictor may be used in that motor port which serves as the inlet port in load lifting, to prevent motor cavitation by restricting the return flow, during the lowering of the load. In addition, a safety control circuit (not shown on the schematic) will be used; it will operate the drum friction brakes when required.

The input to the servo valve is in the form of an electric signal generated by a linear variable differential transformer (LVDT) unit, operated by the pilot or load-master, in which a mechanical displacement is translated into a proportional electrical signal. The feedback signal

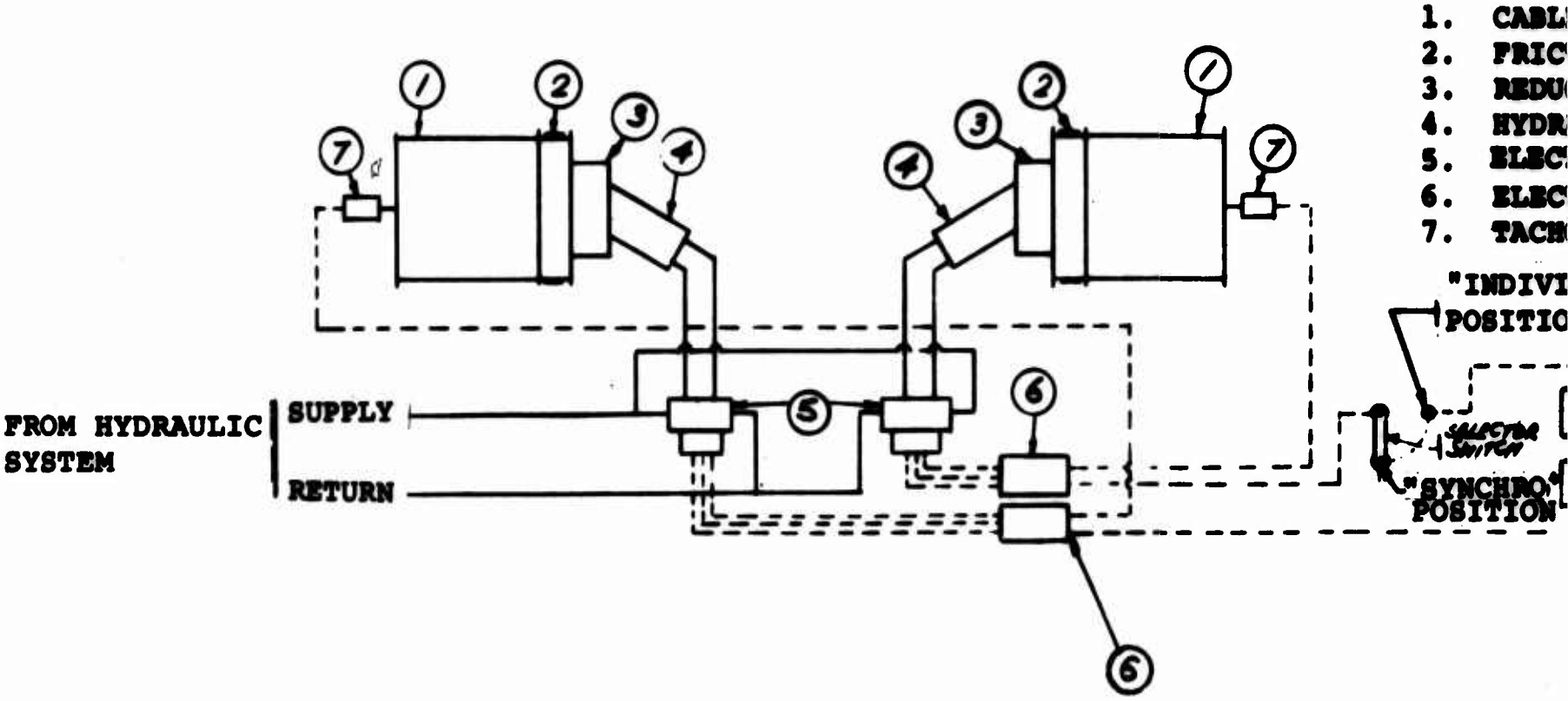
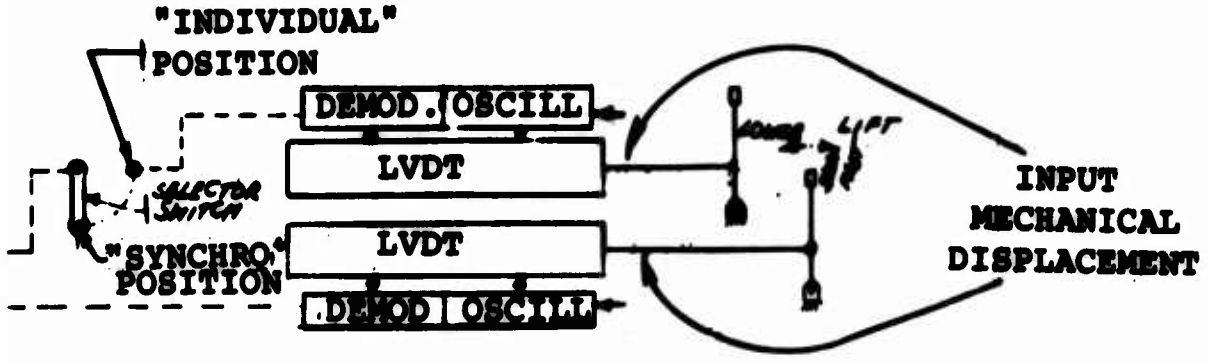


Figure 89. Hydraulic Winch Drive and Control System for a Two-Point Hoist System.

1. CABLE DRUM
2. FRICTION BRAKE
3. REDUCTION GEARBOX
4. HYDRAULIC MOTOR, FIXED-STROKE
5. ELECTROHYDRAULIC SERVO VALVE
6. ELECTRONIC AMPLIFIER
7. TACHOMETER

NOTE: THE FOLLOWING CONTROL MODES ARE AVAILABLE:

1. <u>SYNCHRONIZED OPERATION</u>
   IN THIS MODE BOTH SERVO VALVES RECEIVE A COMMON INPUT SIGNAL FROM ONE SIGNAL SOURCE.
2. <u>INDIVIDUAL WINCH OPERATION</u>
   EACH WINCH OPERATED SEPARATELY BY ITS OWN CONTROL



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is generated by a tachometer unit driven from the winch drum shaft through suitable gearing; thus, the feedback signal is proportional to the drum rotary speed (lifting or lowering rate). The direction and the speed of drum rotation are related to the direction and magnitude of the pilot's control lever displacement from its neutral position.

In the two-point hoist system, it is mandatory to have two operation modes:

1. Synchronized operation of both winches

2. Individual operation of each winch

In the synchronized operation, an identical input signal is sent to each servo valve simultaneously from one LVDT unit. The selector switch is positioned to SYNCHRONIZED and only one control lever is used (the other is inoperative).

For individual operation of the winches, the selector switch is positioned to INDIVIDUAL. Two control levers are used; each one operates a corresponding LVDT unit which generates a separate input signal to the related servo valve. Similarly, two independent feedback signals are generated by two tachometer units. Thus, each winch control circuit is entirely independent of the other.

Electrical Winch Drive and Control System Two-Point Hoist System

Whether an electrical adjustable-speed drive (see Figure 90) for the hoist system becomes a practical proposition or not depends on the availability of high-power low-weight components specially developed for aircraft application. Moreover, solutions must be available to such basic problems as the capability for a gradual torque build-up, as required when starting to hoist a load, and the capability of holding steady an acquired load. Furthermore, the system must meet the following operational requirements:

1. Synchronized operation of both winches

2. Independent operation of each winch

In the described system a power source drives, at a constant speed, an electric AC generator which supplies power to two AC induction motors, one for each winch. A static solid-state frequency-adjusting system with one separate speed-setting rheostat for each AC motor is used as control means. A mechanical coupling of two rheostat controls (with provision for pre-adjustment) is used for synchronized operation of the two winches. The controls are uncoupled for separate operation of each winch.

A reverse locking brake is shown on the schematic immediately downstream of each AC motor. The purpose of this brake is to ensure the load-holding capability of the system, which is power operated for both lifting and lowering of the load. This brake is in addition to the friction brake on each cable drum; it can be dispensed with if the electrical drive system finally selected proves to have sufficient load-holding capability.

Component Availability

Hydromechanical Drive - A lightweight variable-delivery, overcenter hydraulic pump, with servo control, in the 100-horsepower class, and suitable for aircraft application is not available; industrial pumps are available but are heavy.

Hydraulic Drive - All equipment is available.

Electrical Drive - An AC induction motor in the 50-horsepower class, for aircraft application, is not available. For solid-state cycloconverters, in which power rating depends on ambient temperature, a forced-air circulation system for cooling is required.

Summary - Two-Point-Winch Drive and Control Systems

A hydraulic winch drive system is preferable for the two-point drive system.

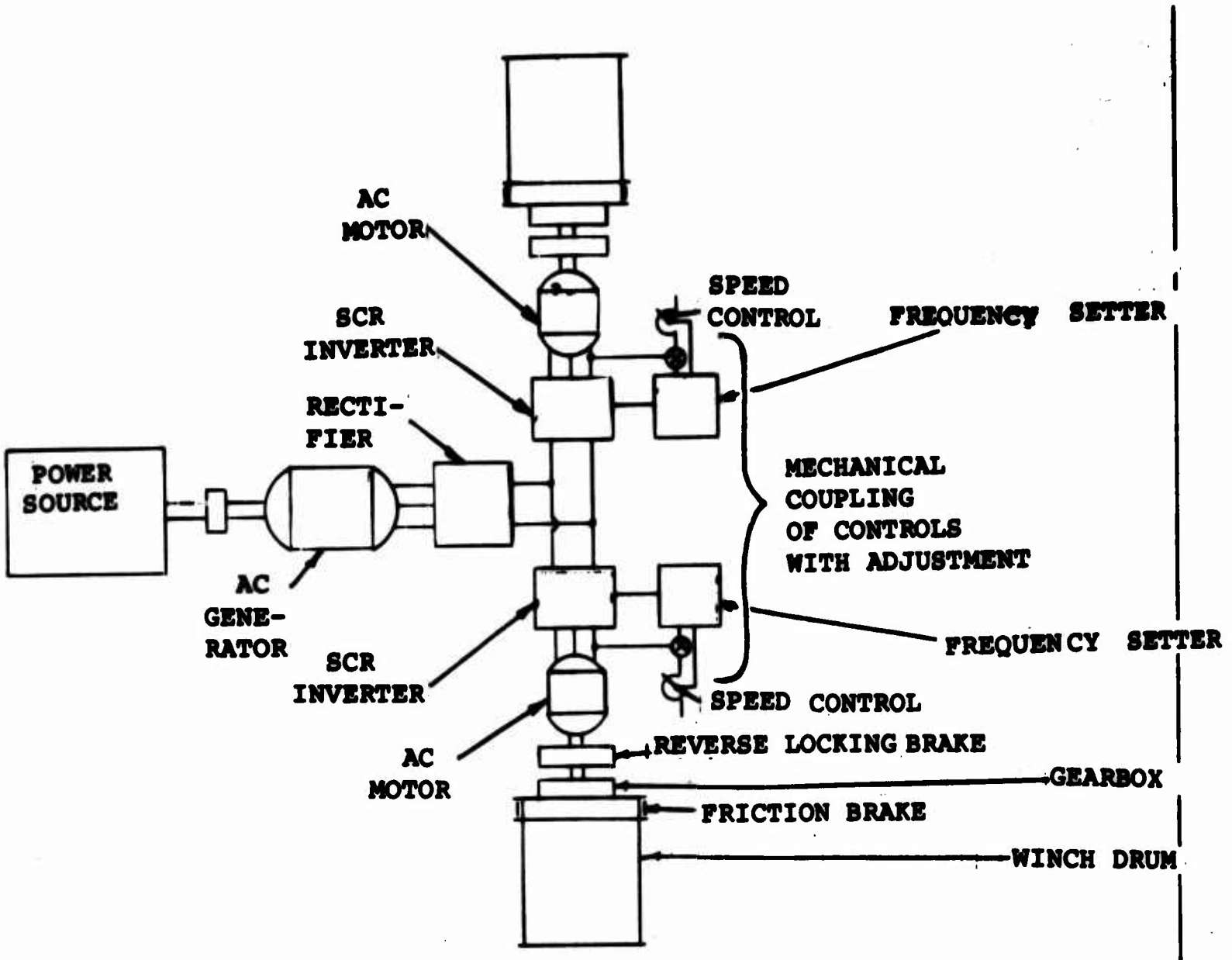
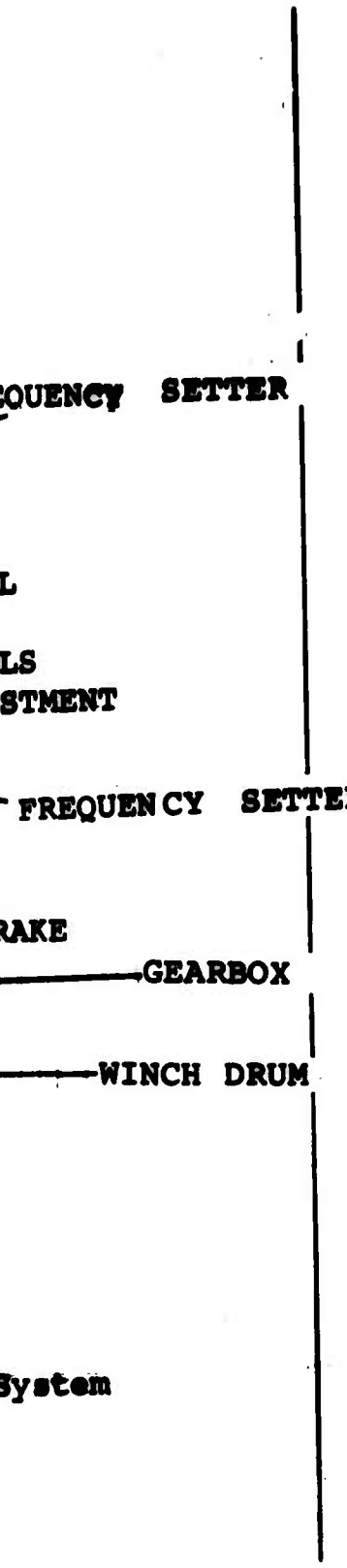


Figure 90. Electrical Winch Drive and Control System for a Two-Point Hoist System.



NOTE: INDIVIDUAL WINCH CONTROL IS AVAILABLE BY PROVISION OF SEPARATE SPEED CONTROL MEANS FOR EACH MOTOR.

FOR SYNCHRONIZED WINCH OPERATION BOTH SPEED CONTROLS ARE SIMULTANEOUSLY AND EQUALLY OPERATED BY MECHANICAL COUPLING HAVING PRE-ADJUSTMENT MEANS

2

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HOIST SYSTEM DESIGN FEATURES INCLUDING ESTIMATED SYSTEM WEIGHTS

The system capacities required by the 20,000/40,000-pound nominal working loads indicate that the optimum hoist design would employ a two-rope, two-part single-reeved system for the single-point lift and two single-rope winches for the two-point system, the philosophy being that in both modes the same diameter of rope would be used. This would mean that the same basic winch components could be employed in both winch types, thus contributing to a minimum-cost system.

The criteria used in the following examples are based on the contractual performance objectives.

An aircraft cargo hoisting system using a 6 x 37 Seale Warrington wire rope of approximately 1.0-inch diameter would have the following characteristics:

Total Design Load (Single-Point)	161,962 pounds
Total Design Load (Two-Point - Each Winch)	113,852 pounds*
Lift Height (Single-Point)	80 feet
Lift Height (Two-Point)	50 feet

Normal Working Loads:

1. 40,000-pound (Single-Reeved,Two-Rope)

2. 20,000-pound (Single-Rope)

Rate: 60 feet per minute (Single-Point System)
30 feet per minute (Two-Point System)

Limit Switches:

1. Rope Fully Extended

2. Rope Fully Retracted

3. Rope Intermediate Position

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*Standard 1.0-inch-diameter rope has a 98,200-pound break strength; however, 1.0-inch-diameter rope to special order can be obtained with a break strength of 117,000 pounds.

Emergency Jettison: An electrically initiated shaped charge to sever the wire rope.

Footage Counter: A direct readout of the extended length of wire rope.

Load Brake: A positive-acting mechanical type, to operate independently in event of failure of the hydraulic system.

EXAMPLE 1

Estimated weight of total system consisting of one single-point-mode system plus one multi-point-mode system of two points.

Wire Rope	490	pounds
Motor (Single-Point)	20	"
Motors (Multi-Point) (2 x 15)	30	"
Hook & Beam Assy (Two-Point)	400	"
Hook (Single-Point)	150	"
Winch (Single-Point)	1,500	"
Winches (Multi-Point) (2 x 1100)	2,200	"
Total Estimated Weight	4,790	pounds

The above example is based on a lift height of 80 feet for the single-point mode. For a 150-foot lift, the weight increase would be:

Additional Wire Rope	240	pounds
Additional Winch Weight (Assuming Capstan-Type)	30	"
Total	270	pounds

EXAMPLE 2

If a total system consisting of one single-point-mode system plus one multi-point-mode system of four points, unsynchronized, is assumed and if the criteria established in this study relating to the effect of aircraft and load cg travel upon winch design and the satisfying of flight safety requirements are adhered to, winches to satisfy a four-point system will have to be the same as those designed for a two-point system. Therefore, if the same ground rules used above for a single-plus two-point

system are used, the resulting weight of a single- plus four-point system will be:

Wire Rope	640 pounds
Motor (Single-Point)	20 "
Motors (Multi-Point) (4 x 15)	60 "
Hook & Beam Assy	500 "
Hook (Single-Point)	150 "
Winch (Single-Point)	1,500 "
Winches (Multi-Point) (4 x 1,100)	4,400 "
Total Estimated Weight	7,270 pounds

This weight assumes that the four-point system is <u>unsynchronized</u>. If synchronization is embodied, then the individual winch capability can be reduced from 28,000 pounds to 15,500 pounds. (Refer to the section entitled "INFLUENCE OF AIRCRAFT CG TRAVEL ON HOIST SYSTEM DESIGN.") This will introduce two different winch rope sizes into the system, one for the single-point-mode system and one for the four-point-mode system. There will be no change for the single-point system. The resulting system weight will then be as shown in Example 3, below.

<u>EXAMPLE 3</u>

Wire Rope	605 pounds
Motor (Single-Point)	20 "
Motors (Multi-Point) (4 x 10)	40 "
Hook and Beam Assy	400 "
Hook (Single-Point)	150 "
Winch (Single-Point)	1,500 "
Winches (Multi-Point) (4 x 900)	3,600 "
Total Estimated Weight	6,315 pounds

Examples 2 and 3 are based on a lift height of 80 feet for the single-point mode. For a 150-foot lift, the weight will increase by 270 pounds.

<u>WEIGHT SUMMARY</u>

	<u>Pounds</u>
Single-Point (80-foot lift) + Two-Point	4,790
Single-Point (150-foot lift) + Two-Point	5,060
Single-Point (80-foot lift) + Four-Point (Unsynchrz'd)	7,270
Single-Point (150-foot lift) + Four-Point (Unsynchrz'd)	7,540
Single-Point (80-foot lift) + Four-Point (Synchrz'd)	6,315
Single-Point (150-foot lift) + Four-Point (Synchrz'd)	6,685

CONSIDERATION OF THE CONTRACTUAL PERFORMANCE OBJECTIVES WITH AN APPROACH TO THEIR RESOLUTION

Performance objectives applicable to both single-point and multi-point modes shall be to:

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

   This resolves into an ultimate design factor of 3.75. Wire rope manufacturers disagree with this factor as applied to wire rope and recommend an ultimate factor of 5. However, investigation has shown that their factor is based on fatigue life of the rope, and since the duty cycles specified for heavy-lift hoist system are only a fraction of commercial usage, a factor of 3.75 is considered acceptable. However, this acceptance is based on the assumption that all other recommended design criteria established for wire rope are adhered to. Deviation from the recommended design criteria should be compensated for by an increase in design factor.

2. Minimize unsafe oscillations of a suspended load.

   In a single-point system, oscillations become a function of external forces acting on the suspended load. The only controls over these forces are regulation of flight speed, aerodynamic characteristics of the load, and suspension length. Negative aerodynamic characteristics can be compensated for by the use of stabilizing agents such as drag chutes or fins attached to the load. In cases where such methods are considered unacceptable, multi-point suspension must be used. From a system aspect, the moments created about the aircraft cg by the oscillating load must be kept within the design limits of aircraft control system. This is achieved by mounting the suspension point close to the cg. This becomes a requirement for system installation into the aircraft. In a multi-point system the same rule applies.

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

In hover pickup, flight safety is the most important element. On this basis, a single-point pickup is the optimum, since one man, only, is required to physically engage the load sling with the hook. It is important to realize that the load sling is engaged with the hook, not the hook with the load. This stems from the fact that the hook is integral with the helicopter and so is subject to continuous movement. This fact dictates the use of load slings for hover pickup in order to compensate for helicopter movement. It can be argued that a slack hoist rope will achieve the same result, but this will require the ground crewman to lift and maneuver a hook system which weighs in excess of one hundred pounds; this is considered impractical.

In hover pickup with a multi-point system, the ability to meet flight safety requirements degenerates. In a two-point system, two men will be required for load attachment (one to each hook) and, as with a single-point system, loads must be slung to give the necessary compensation for helicopter movement. Another problem arises in that if a beam system is embodied as part of the two-point system (this is considered necessary to effect hook release synchronization), it will be necessary to align the beam with the load. This can be accomplished by giving the ground crew beep control of the winch system by embodying electrical circuits between the beam and winch system through the same conductor circuit cable used for hook release. The inherent danger during acquisition arises from the fact that if only one hook is engaged and the aircraft lifts off, it can result in upsetting moments being transferred to the helicopter. Therefore, in this pickup mode the pilot or copilot must be constantly alert and ready to release the load instantly. The employment of a loadmaster who would visually monitor physical acquisition of the load and stand ready to take instant action in an emergency would alleviate this potentially dangerous situation.

In a four-point system, the danger to flight safety is compounded since coordination of four ground crew members (one to each hook) will be required. The beam trim function will be essentially the same as for the two-point, except it will be more complex

since trim in two planes will be necessary as against one plane for a two-point system.

The foregoing discussion stresses the use of slings in load acquisition from hover. This means that if flight safety requirements are to be met, then pods or containers acquired from hover must also be treated as slung loads.

With the aircraft in a landed position, the satisfaction of flight safety criteria can be assured, since system checkout may be accomplished before takeoff. The use of slings in this mode is not mandatory since the suspension system may be adjusted to suit the load. The direct coupling of pod-type loads to aircraft structure is also practical. However, this method of attachment dictates that the aircraft be able to land at the delivery site in order to unload. Personnel pods presently in use are secured directly to aircraft structure at four points; with this arrangement, pod and aircraft structure flexibility makes the fitting of the pod difficult. Preliminary investigation indicates, however, that the attachment could be accomplished at two points, thus eliminating four-point misalignment due to structural movement of the pod and aircraft.

4. <u>Facilitate rapid hook-up of loads. A two-minute maximum hook-up time for previously rigged loads is allowed</u>.

   This is related to Item 3, above. Two minutes is considered to be rather long as an objective. A one-minute maximum is considered to be practical for acquisition of previously rigged loads. It must be appreciated that the longer time may well constitute a greater hazard to flight safety.

5. <u>Provide for the positive locking of personnel pods to the airframe with cargo handling system power off</u>.

   This is covered in Item 3, above, where it is implied that the attachment of personnel pods to the structure should be independent of the aircraft external hoist system.

6. Provide for two methods of cockpit-controlled load release, electrical and mechanical.

This requirement can be met as described in the text of this study.

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

This requirement can be met by extending the mechanical linkage within the beam system on the two- or four-point system to a level accessible to ground personnel. Such a system would have a safety lockout to prevent inadvertent operation. For the single-point system, a similar feature is already available on cargo hooks in military use.

8. Incorporate provisions for automatically holding suspended loads with the cargo handling system power off.

All winch brake systems will be failsafe; that is, when the power to the winch is off, the brake will automatically go on. Such a feature ensures that the winch will not reel out in the event of system power failure.

9. Incorporate provisions for load jettisoning independent of the releases required by Item 6, above, and the holding provisions required by Item 8, above.

This will be accomplished by installing dual cartridge rope -cutter devices or shaped charges, electrically initiated; each cutter will be on an individual circuit with each circuit having a separate power source. It should be pointed out that the reliability of such devices will be a function of the number of wire ropes being severed; one is the optimum; four, the least reliable.

10. Provide for attenuation of shock loads imposed on suspension systems.

The embodiment of load isolators in the system will meet this requirement.

11. Minimize cable backlash.

This will be accomplished by designing into the winches devices which will hold the wire rope in contact with the drum grooves under all conditions. In situations where a slack rope exists before the helicopter lifts the load, then either the slack should be winched out before lift-off, or lift-off should be performed slowly.

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

This objective is directed to sling technology and contradicts the Statement of Work, Paragraph 2.b, which states:"it excludes the power source and any provisions necessary to rig the load for acquisition by the systems." Load acquisition by the suspension system is discussed in Items 3, 4, 6, and 7, above.

13. Be designed so that major components of each mode system 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 Mechanic's Tool Kit.

This requirement can be met with the exception that, because of the weight of the components (notably winches), an auxiliary hoist will be necessary for winch installation or removal. Investigation has shown that small, light hand hoists are available which will satisfy this requirement. This, however, will be an aircraft installation problem, and the only provision to be made on the proposed hoist system will be the embodiment of hoist points on the winches.

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

    Similar aircraft systems have already been designed and qualified. Investigation has indicated that existing practices can adequately meet these requirements.

Additional objectives for the single-point suspension mode shall be to:

1. 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.

    The hoist system vertical lift capability (height) has a major influence on winch weight and, especially in the case of drum-type winches in a reeved rope system, extended lifting heights can result in the drum length becoming excessive. Although this length can be compensated for by increase in drum diameter, it will still result in a large winch envelope. Long rope lengths are better accommodated on capstan-type winches; even on this type of winch, however, rope length is a major contributor to winch envelope. However, any rope length can be accommodated if we will accept the attendant weight/envelope penalty.

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

    System rate is a major contributor to system weight; if we double the rate, we double the power requirement. It follows that the lower the rate, the lower the overall system weight. Unlike rope length which influences only winch weight, rate will affect all items

in the complete system from winch to power source. The 60-feet-per-minute rate specified was derived from a requirement to restrict hover time to a minimum. It is of interest to note that during feasibility trials of an airborne hoist system on the Navy UH-46 helicopter, a rate of 50 to 60 feet per minute was considered adequate. If we assume a system efficiency of 80 percent, then such a rate would require the following power:

$$\frac{40{,}000 \times 60}{33{,}000} \times \frac{100}{80} = 90.9 \text{ horsepower}$$

Control of acceleration and deceleration will be a function of the winch brake system. Results of investigation into the different types of brakes available have shown two types to be compatible with the winch application:

a. The single-disc type
b. The multiple-disc type

A typical single-disc brake consists of a spring-applied, hydraulic-release "failsafe" unit designed for installation on the input end of a helicopter winch. For a given brake volume, the multiple-disc brake is capable of taking higher torques, because of its multiple friction surfaces, and has higher heat capacity, because of the increased mass available for heat sink, than the single-disc brake. However, a single-disc configuration is considered superior for winch applications for the following reasons:

a. Positive disc clearance is easily attained. This is important at the high-speed, low-torque end.

b. A failsafe (spring applied, hydraulically released) design is easily and economically achieved.

c. Cost is generally lower.

d. It is easier to maintain.

The final choice, however, will be dependent upon available space within the winch envelope.

Investigation has shown that precise control of braking can be achieved if advantage is taken of some suitable automatic braking device which would control the braking torque in such a way as to maintain or change the rate of load descent in accordance with the value of input signal, which in turn is a function of the setting of pilot's or loadmaster's winch control. Such a system is within today's state of the art.

3. Be usable for towing large ground vehicles.

Towing from a single-point winch system is not recommended for the following reasons:

a. The drawbar pull capacity of a helicopter is in excess of the winch lift capacity. An unacceptable system weight penalty would result if the winch were designed for the drawbar capacity.

b. Rope damage would result unless the winch installation was designed to swivel so as to orient the rope always in a direct line between tow and winch.

c. Optimum location of single-point winch for vertical lift may not be compatible with the tow requirements, which could constitute a flight hazard.

d. Towing is best achieved by locating a hard point on the fuselage at the optimum location for towing. Towing would then be accomplished with a tow rope attached at this location. This tow point would be designed with a tension-indicating device, emergency cutters, and a tow rope lateral-angle indicator. The latter would indicate to the pilot his alignment with the tow. It should be appreciated that a large degree of misalignment constitutes a flight hazard.

Additional requirements for the multi-point suspension mode shall be to:

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

The acquisition height will be influenced by the rotor downwash velocity. Available data tends to indicate that 50 feet will be acceptable both from a pilot visibility and a hazard-to-ground-personnel point of view. It must be appreciated that precise data for the proposed heavy-lift helicopter is not available. Further, if an alongside pickup technique is resorted to, an effective rope length of 50 feet appears to be inadequate.

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

As in the case of the single-point mode, the multi-point system rate was derived from a requirement to restrict acquisition time to a minimum. In a multi-point system, acquisition time is not just a question of how much rate you build into the system but is also a matter of how well you can control the rate of all the suspension points. In a two-point system, small discrepancies in hoist length between front and rear points cause no hazard and may be trimmed out when the load has been winched up. In a four-point system, regulation of four points without having some form of synchronization is virtually impossible; one rope at least will always be slack, resulting in additional load being carried by the remaining suspension ropes. If it is conceded that the load on a nonsynchronized four-point system might, in fact, be supported during acquisition by only two of the suspension ropes, then the system should be designed for this condition. This would result in a system having four winches of the same capacity as the winches in a two-point system, a 100-percent weight penalty.

3. Provide collective (simultaneous) control of all hoisting elements from each control station.

Synchronization of the winches in a two-point system is not critical; however, as a feature which will give minimum load acquisition time, its embodiment is considered justified. Further justification for synchronizing a two-point system is that it permits the integration of a two-point into a single-point system, eliminating the need for a single-point winch. This will effectively increase the payload of the helicopter

by approximately 2000 pounds. As a function of load acquisition rate (Item 2 above), synchronization of a four-point system, as discussed in this report, dictates that such a requirement be mandatory, unless the resultant weight penalty is considered acceptable.

4. Provide simultaneous load release at all suspension points.

   This can be achieved on either multi-point system by integrating the cargo hooks into a common beam; the penalty is weight. In the case of a two-point beam carrying a hook at each end, the addition of further hooks along its length will create bending moments in the beam which will further increase the beam weight.

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

   This is an aircraft systems requirement. Investigation indicates that no problem exists in its embodiment. It is recommended that if a beam system is used, controls for monitoring the system be provided on the beam for use of ground personnel, so that the system may be trimmed at the point of load acquisition. This would be in addition to a remote plug-in facility for the loadmaster. The reason for this is that the loadmaster's external facility will be used when the aircraft is on the ground, whereas system trim function by ground personnel will be necessary during hover pickup.

RELATIONSHIP OF WINCHES TO AIRCRAFT CONFIGURATION AND ITS INFLUENCE ON HOIST SYSTEM DESIGN

Although the contract calls for no specific aircraft configuration to be taken into consideration during the performance of this study, some appreciation of the relationship of the major components of the proposed systems to a prime mover configuration is considered to be of assistance in appraising the various elements analyzed in this study. Future heavy-lift helicopters may be a pure crane or transport configuration or a combination of both. The configuration used in this study is a combination of both configurations and is one of those shown in Report No.

R445, Reference 12.

Three sketches are presented to show the relationship of various winch configurations to the aircraft structure. (Reference Figures 91, 92 and 93.)

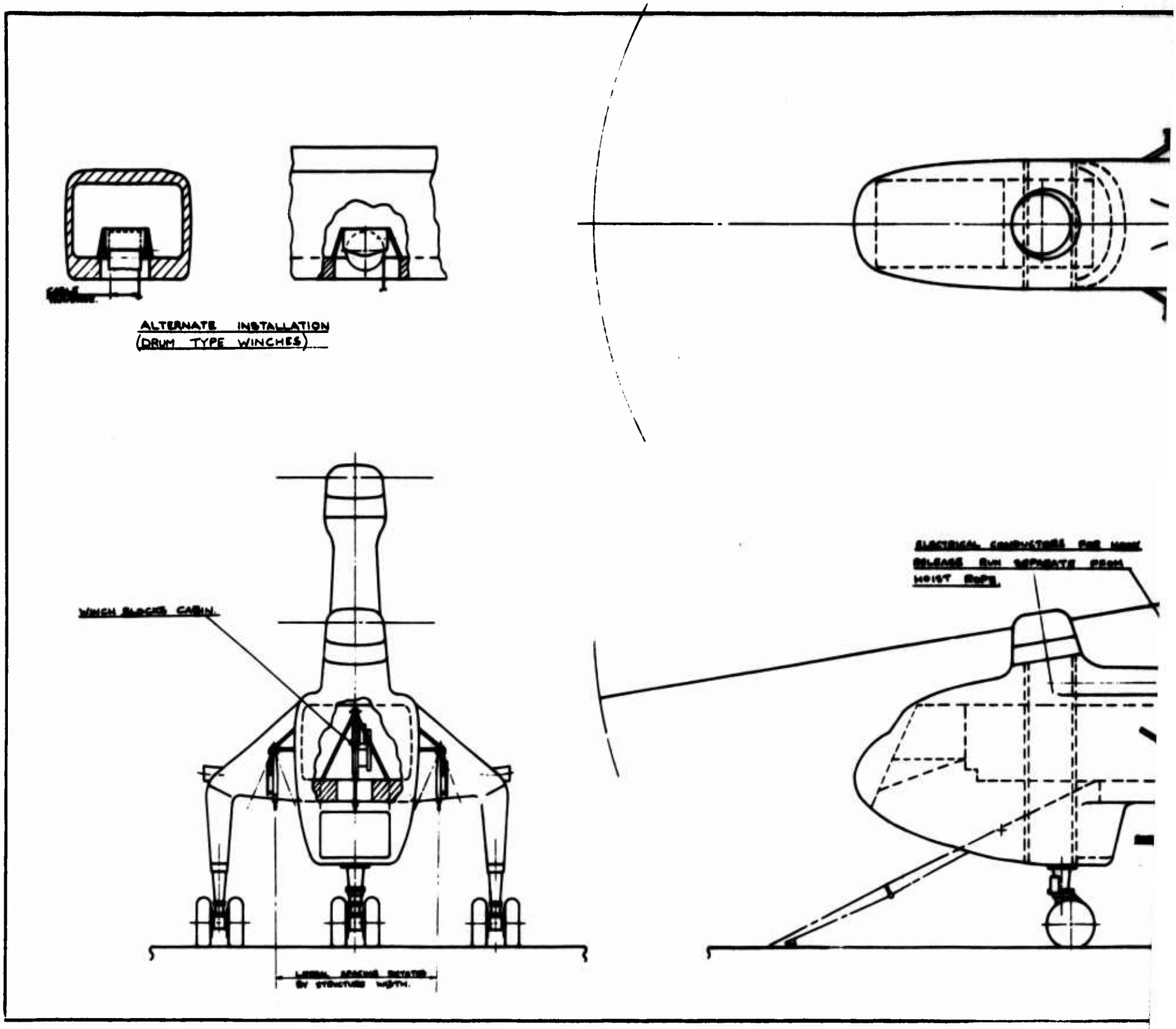
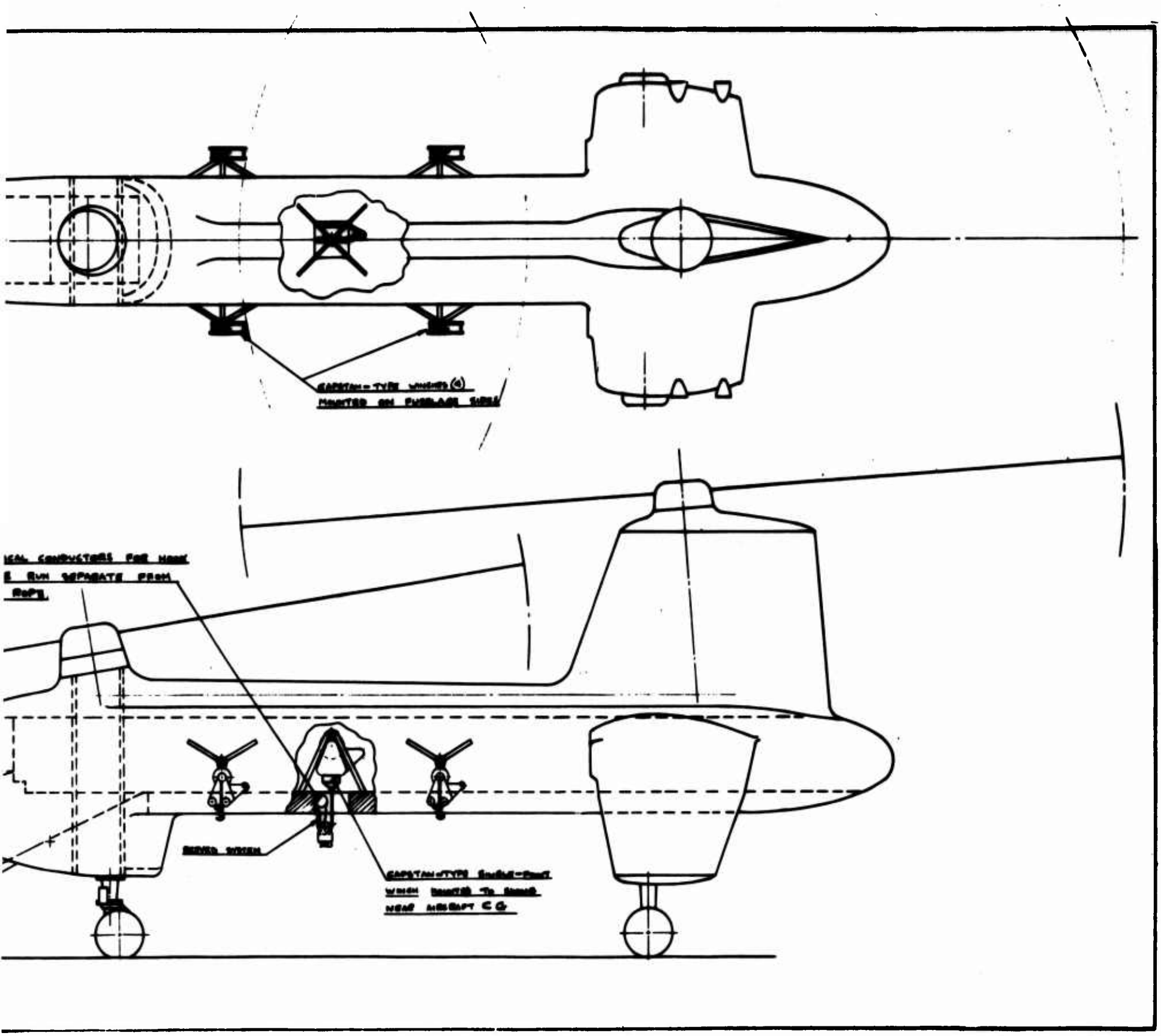
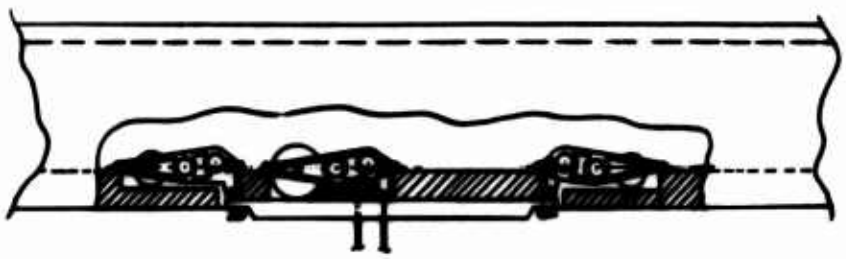


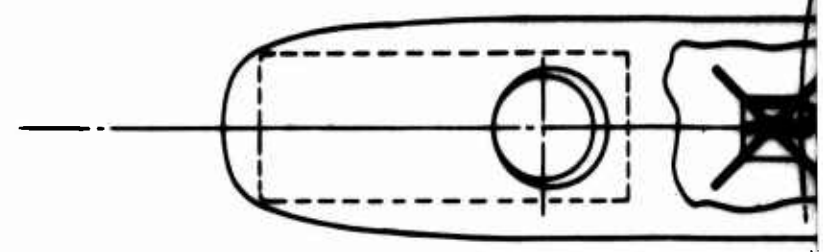
Figure 91. Single-Plus Four-Point System.


CAPSTAN-TYPE WINCH (4)
MOUNTED ON FUSELAGE SIDES
ELEC. CONDUCTORS FOR HOOK
& RUN SEPARATE FROM
ROPE
BOWEN WINCH
CAPSTAN-TYPE SINGLE-SPEED
WINCH MOUNTED TO SUPPORT
NEAR AIRCRAFT C.G.

2



ALTERNATE WINCH DESIGN
CAPSTAN WINCHES DESIGNED AS 'FIXED' INSTALLATIONS.



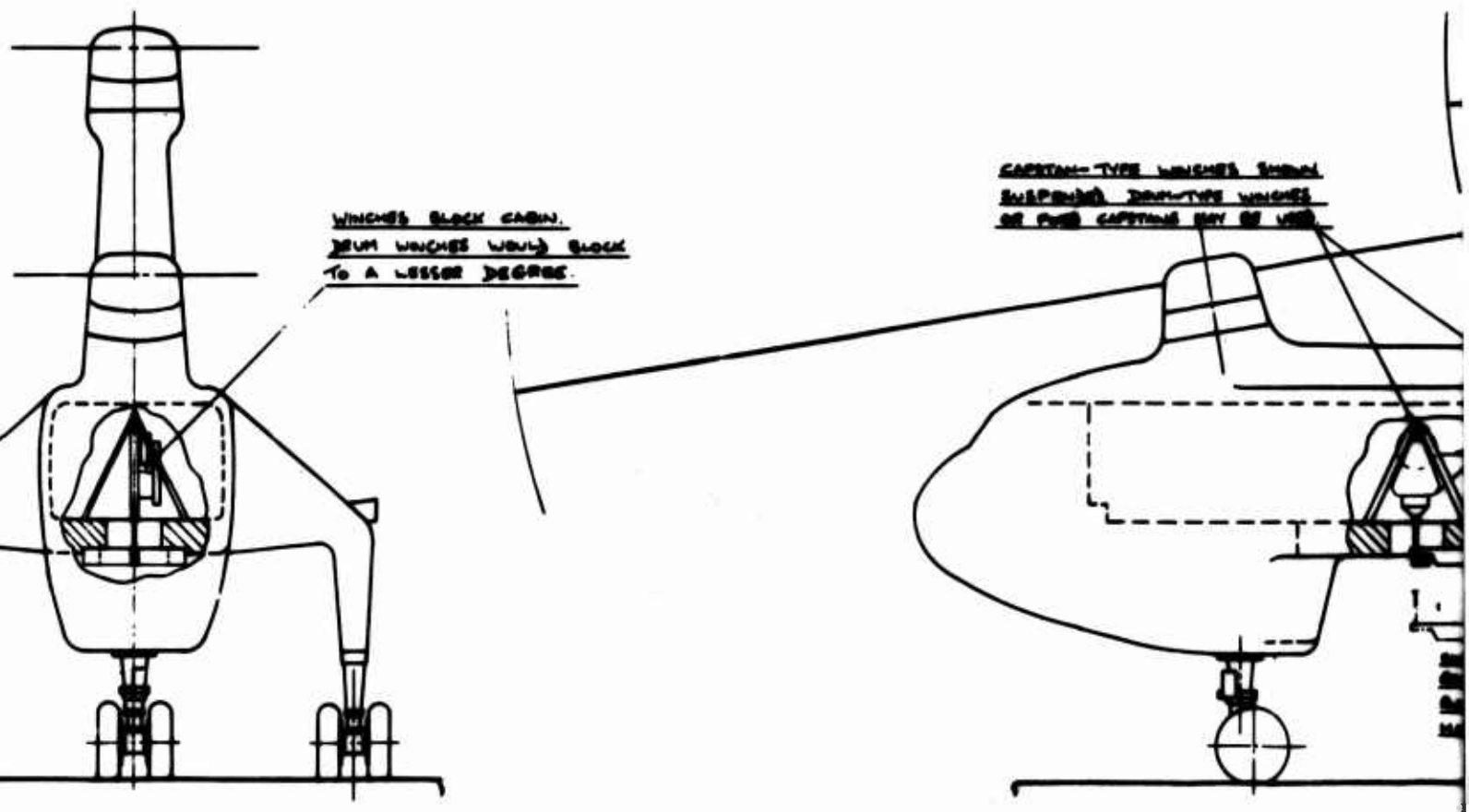
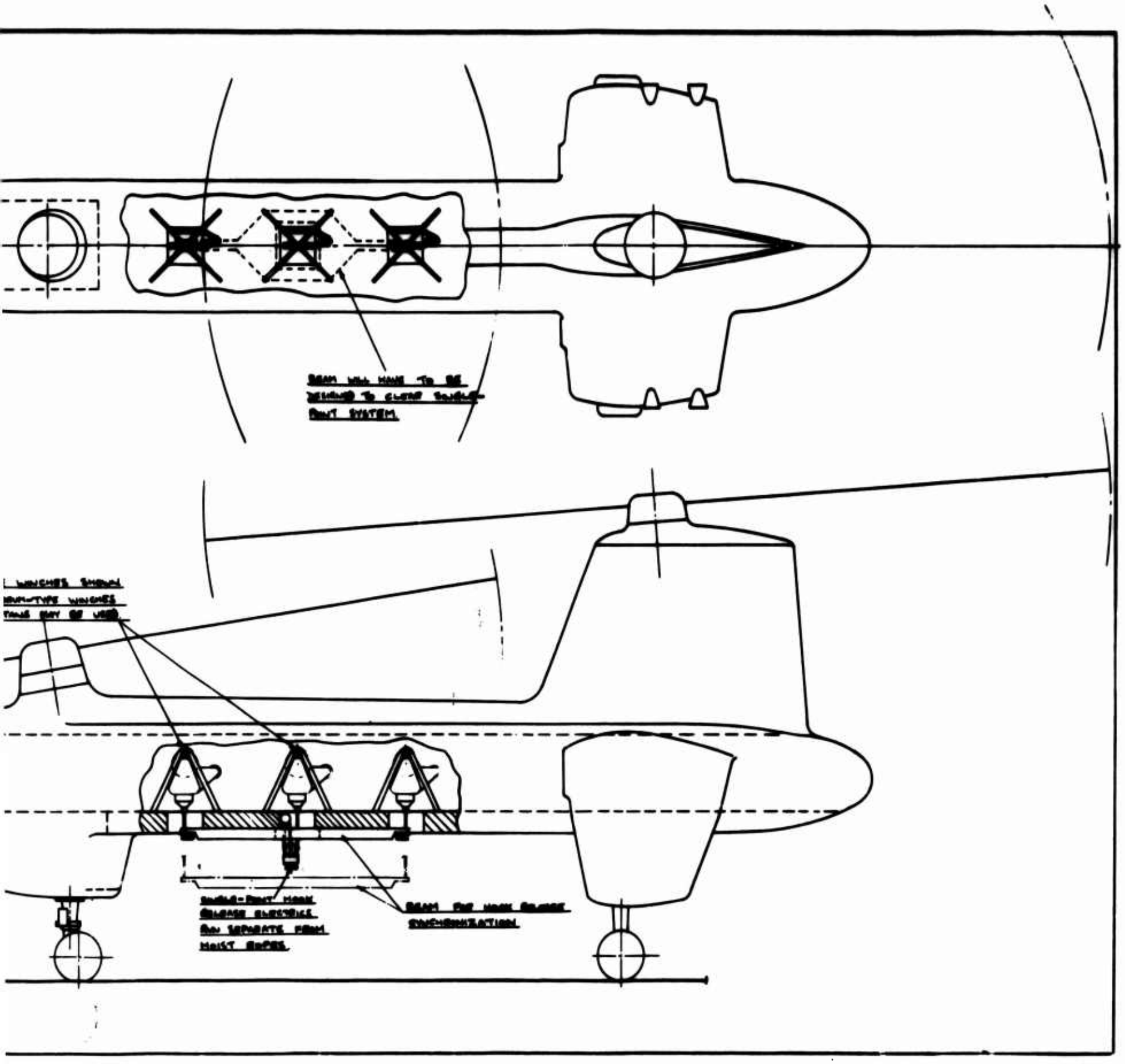


Figure 92. Single- Plus Two-Point System.


BEAM WILL HAVE TO BE
STRENGTHENED TO CARRY TWO-POINT
HOIST SYSTEM.
WINCHES SHOWN
HERE-TYPE WINCHES
THAT MAY BE USED.
TWO-POINT HOOK
RELEASE ELECTRICALLY
AND SEPARATE FROM
HOIST SUPPORT.
BEAM FOR HOOK RELEASE
SYNCHRONIZATION.

2

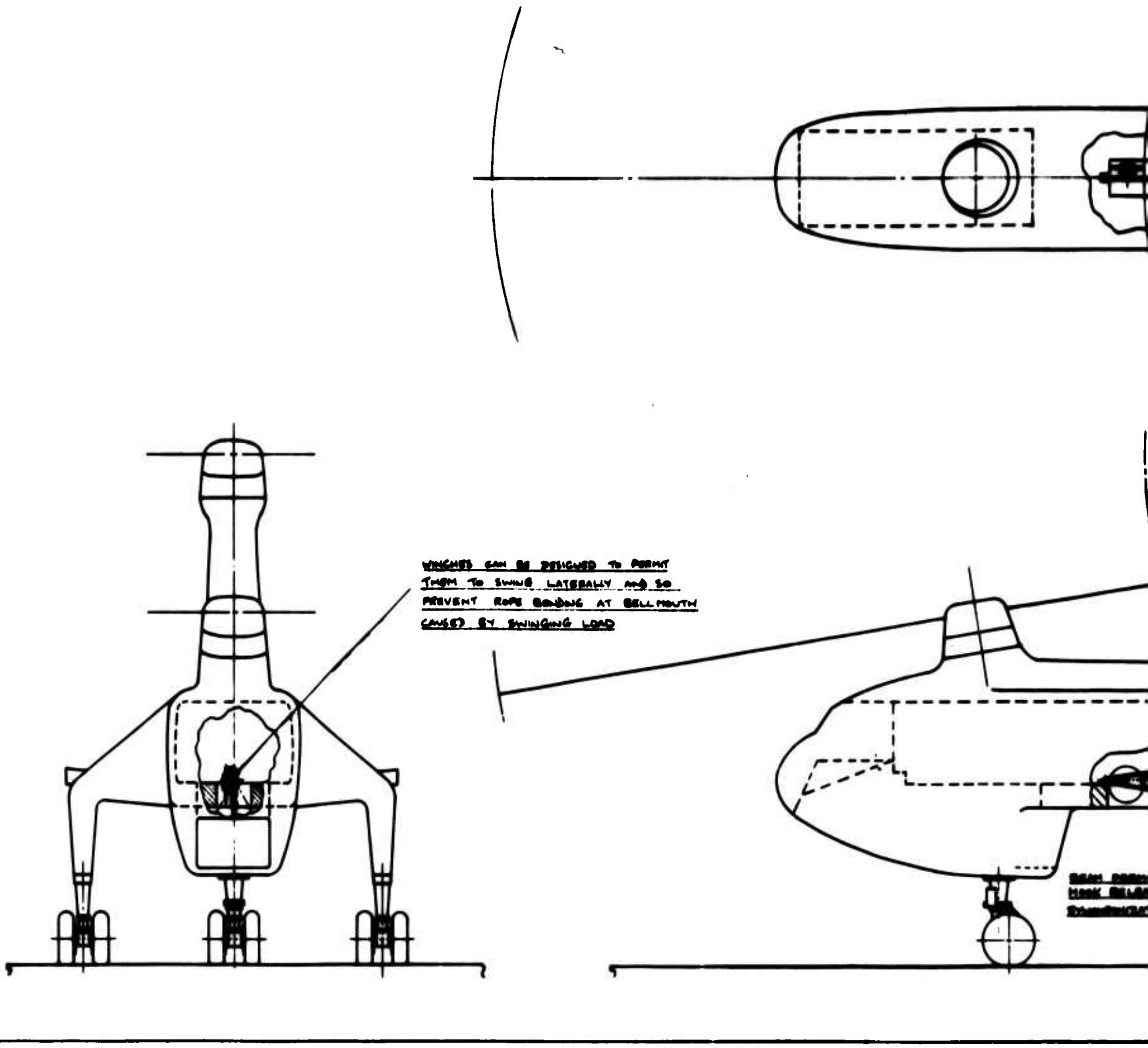
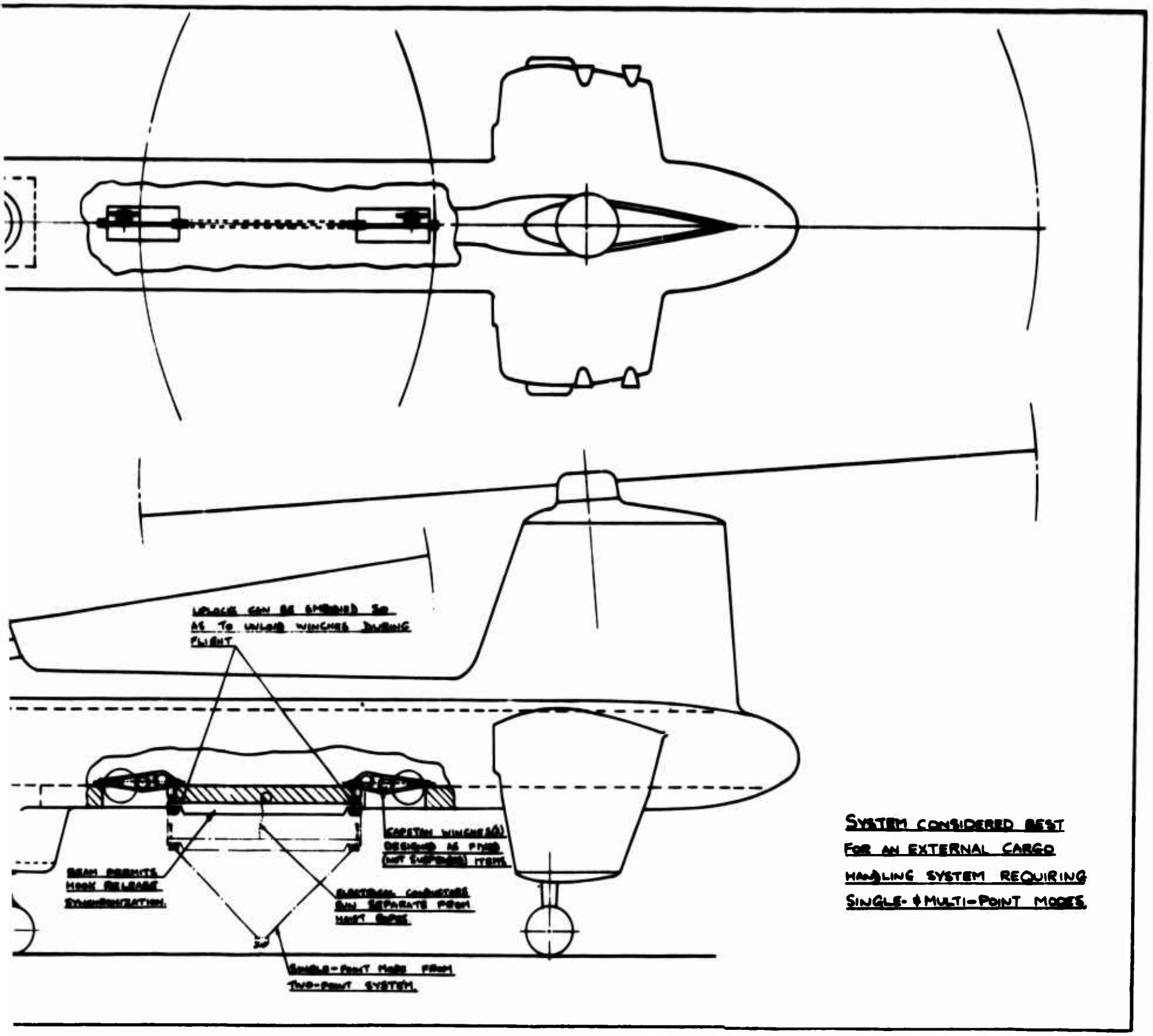


Figure 93. Two-Point System with Single-Point Mode.



SYSTEM CONSIDERED BEST FOR AN EXTERNAL CARGO HANDLING SYSTEM REQUIRING SINGLE- & MULTI-POINT MODES.

2

PHASE I - CONCLUSIONS AND RECOMMENDATIONS

1. The limiting diameter for wire rope is 1 inch; in cases where the lift requirement exceeds the capacity of 1-inch wire rope, a reeved rope configuration will be used.

2. It is recommended that the wire-rope-to-winch-drum diameter ratio be not less than 1:18. The ratio for sheaves should not be less than 1:15. This assumes a construction as represented by a 6 x 37 IWRC Seale Warrington equal-laid wire rope. Fleet angles should not exceed 2 degrees for a grooved drum.

3. Electrical conductors with their complementary slip ring systems should be divorced from the winch/rope system. It is recommended that separate constant-tension cable reels be used for the purpose of conveying electrical conductors from the helicopter to the hoist system cargo release media.

4. The capstan type of winch is considered to be superior to the drum type for helicopter external hoist systems principally because of the zero-moment feature of the capstan type; i.e., the ability to suspend the winch from a single point and thus eliminate undesirable cg change during reel-in or reel-out.

5. To satisfy flight safety criteria, it is mandatory that a four-point hoist system have synchronized control of the four winches. The development of such a system will be expensive both in time and in dollars. Before the design of a four-point synchronization system can be initiated, a study of the comprehensive system dynamics must be performed.

6. For an installation consisting of single point plus two point suspension, a capstan-type winch having a single-reeved two-rope suspension is recommended for the single point system. The wire rope recommended for this system is a 1-inch-diameter 6 x 37 IWRC Seale Warrington rope of equal-laid construction. The two-point system recommended consists of two winch systems identical to that recommended for the single point except for wire rope length. Since rope length influences only the storage drum capacity and since approximately 95% design commonality will exist between winch types, the wire rope would also be the same

size and type for both systems. In order to ensure synchronized release of cargo hooks, the two-point suspension will terminate in a beam, thus providing a common base for interconnection of cargo hook release mechanisms. The cargo hooks recommended are of the electromechanical type presently in use, modified to suit installation into the beam. The electrical conductors for hook release will be run separately from the hoist ropes and winches via a constant-tension cable reel from aircraft to cargo beam. The load isolators recommended are of the hydropneumatic type because of their small size, low weight and high degree of reliability. Control of the system consists of an individual variable rate for each winch plus integration of both winch controls. This permits system control to be achieved through a four-way control, permitting raise-lower-forward trim-aft trim. The above recommendation is based on the following for a single- plus two-point system:

a. Lowest system weight (single- plus multi-point)

b. Lowest system cost

c. Versatility of capstan principle which permits design flexibility to suit differing aircraft configurations

d. High degree of parts commonality between a single- and a two-point system

e. Lowest cost and development time for a single- plus multi-point system

f. System control relatively simple (i.e., similar to existing qualified control systems)

g. Maintenance requirements least for a single- plus multi-point system

h. Cargo release system that satisfies established flight safety criteria

i. Suspended beam system that forms a base for acquisition of loads (i.e., pods, containers, etc.)

j. All items designed within today's state of the art (see sketch).

7. For an installation consisting of a single-point plus a four-point suspension, the single-point system recommended is identical to the single-point system in 6, above. The recommendation for the four-point system is four capstan-type winches, the hoist ropes of which terminate in a beam frame to which are secured four cargo hooks. The frame will form a base for interconnection of the hook releases so as to ensure hook release synchronization. The cargo hooks recommended are of the electromechanical type presently in use, modified to suit installation into the beam frame. The electrical conductors for hook release are run separately from the hoist ropes via a constant-tension cable reel from aircraft to beam frame. The load isolators recommended are of the hydropneumatic type because of their relatively small size, low weight and high reliability. In order to meet flight safety requirements, full synchronization of the four winches is recommended based on the following for a single-point plus a four-point system:

   a. Versatility of capstan principle to permit design flexibility to suit differing aircraft configurations

   b. Flight safety criteria satisfied

   c. Suspended beam system that forms a base for acquisition of loads (i.e., pods, containers, etc.)

   d. Lowest system cost and weight which satisfies safety-of-flight criteria for acquisition and transporting externally slung loads

   e. All components except synchronization control system within today's state of the art

8. The above systems (6 and 7) exceed the design weight objective of 4000 pounds (6 being 4790 pounds, 7 being 6415 pounds). In order to meet the 4000-pound objective, Vertol submits for consideration the following system consisting of a two-winch system which will satisfy both single- and multi-point suspension modes. Such a system would be similar to that described above (6) for a two-point system. The single-point mode would be achieved by integrating the lift point with the cargo beam into one hook either mounted in the center of the beam or suspended from the two beam end hooks to form one common hook. This

system would satisfy all the requirements required for a single- and multi-point mode system without the disadvantage of installing a 40,000-pound-capacity winch, resulting in a system weight reduction of approximately 2100 pounds. Such a system is estimated to weigh 3000 pounds. Any subsequent requirement for a single-point hoist system of the same capacity could be met by using the same winch with a reeved cable. This system is Vertol's recommendation for a system satisfying the requirements for a single- and multi-point mode external hoist system because it:

a. Is the lightest of all systems investigated which meets multi-point and single-point modes.

b. Is the least-cost system.

c. Needs no complex control system.

d. Meets flight safety requirements for load release.

e. Is capable of employing higher hoist rates than a four-point system, since it is more controllable.

f. Uses winches that would be adaptable to both a two-winch system and a single-winch (reeved) rope system for the same capacity.

g. Weighs 1000 pounds less than the design objective of 4000 pounds.

h. Is the least complex installation, resulting in a weight saving on the prime mover.

SUMMARY OF THE RELATIVE MERITS OF HOIST SYSTEM COMPONENTS AND COMPLETE HOIST SYSTEMS INVESTIGATED IN THIS REPORT

Below, in Tables VII and VIII, are compiled the relative merits of components and complete hoist systems. Only the major elements contributing to the most effective hoist system are considered; prime mover configurations including power system requirements are not considered.

Hoist system elements considered are as follows:

1. Weight - Hoist system components, excluding the weight of basic aircraft structural provisions for their installation;

such changes are considered in installation.

2. Cost - The costs involved to design, fabricate and qualification test.

3. Maintenance - The relative maintenance required based on inspection requirements, component replacement and component accessibility.

4. Reliability - The relative reliability assessed as a function of the number and complexity of components involved within a given system.

5. Flight Safety - The relative degree with which each component (when applicable) or system contributes to achieving optimum safety of the prime mover and ground personnel during load acquisition, flight with load and load delivery.

6. Installation - The relative adaptability of the basic design to varying installation requirements, plus the relative complexity of structural changes to basic airframe structure to accommodate the various system components.

7. Productivity - The relative contribution of the component or system to the achievement of optimum system (helicopter plus hoist system) productivity, including acquisition time, quantity of ground support required and load delivery.

8. Efficiency - The relative ability of the component or system to perform its intended function, considering the design and performance objectives specified.

Note: Ten degrees of acceptability are used, one being considered to be the best for the element under consideration; i.e., a 1 under weight indicates the lightest component or system, and a 10 under flight safety indicates that flight safety requirements have not been met.

TABLE VII

RELATIVE MERITS OF COMPLETE HOIST SYSTEMS

Complete System Type	Mode of Control		Weight	Cost	Flight Safety	Reliability	Maintenance	Productivity
	Hooks	Hoists						
Single-Point System	-	-	1	1	2*	1	1	8**
Single- Plus Two-Point	U.S.	U.S.	3	3	10	3	3	5
Single- Plus Two-Point	S	U.S.	4	4	1	4	4	3
Single- Plus Two-Point	U.S.	S	5	5	10	4	4	4
Single- Plus Two-Point	S	S	6	6	1	5	5	2
Single- Plus Four-Point	U.S.	U.S.	7	7	10	6	6	6
Single- Plus Four-Point	S	U.S.	8	8	5	7	7	4
Single- Plus Four-Point	U.S.	S	9	9	10	7	7	5
Single- Plus Four-Point	S	S	10	10	10	8	8	3
Two-Point With Single-Point Mode	S	S	2	2	1	2	2	1***

U.S. Indicates Unsynchronized
S Indicates Synchronized

* Hook has no manual mode of release (Except for cutters).
** Flight speed restricted with high percent of loads.
*** Loss of single-point system increases payload approx. 2000 pounds.

TABLE VIII

RELATIVE MERITS OF HOIST SYSTEM COMPONENTS

Item	Weight	Cost	Maintenance	Reliability	Flight Safety	Installation	Productivity	Efficiency
Capstan-Type Winch	2	2	2	2	1	1	1	1
Drum-Type Winch	1	1	1	1	1	5	1	1
Hook-Electro-Mechanical (Two Modes)	3	3	2	4	1	2	1	1
Hook-Mechanical (Ground Release)	1	1	1	1	10	1	6	5
Load Isolator Hydro-Pneumatic	2	1	1	1	1	1	1	2
Load Isolator Hydraulic	1	1	2	1	1	1	1	2
Load Isolator Air	3	2	3	2	1	4	1	1
Hook Release Unsynchronized	1	1	1	1	10	1	6	1
Hook Release Synchronized		5	2	1	1	2	1	1
Electric Conductors In Hoist Rope	1	1	5	5	1	1	1	1
Electric Conductors Separate	2	2	1	1	1	2	1	1
Hoist Controls Unsynchronized (Two-Point)	1	1	1	1	1	1	2	2
Hoist Controls Synchronized (Two-Point)	3	3	4	4	1	3	1	1
Hoist Controls Unsynchronized (Four-Point)	1	1	1	1	10	1	10	10
Hoist Controls Synchronized (Four-Point)	4	6	4	4	1	4	1	1

PHASE I - PLANNING SCHEDULE AND COST ESTIMATE FOR THE DESIGN, MANUFACTURE, AND QUALIFICATION TESTING OF A 40,000-POUND EXTERNAL HOIST SYSTEM

PLANNING SCHEDULE

The schedule given in Figure 94 covers a planning estimate of the time span required for the development of an external hoist system of 40,000-pound dynamic capacity for use on future heavy-lift helicopters. The system will consist of either:

1. A single-point hoist system plus a two-point hoist system or

2. A single-point hoist system plus a four-point hoist system.

These systems are covered by Event No. 1 of the schedule.

In addition, Events No. 2 and No. 3 cover the development planning estimate of the time required to develop a system capable of achieving synchronized control of the multi-point systems.

Event No. 1 Time Span

The element dictating the time span for this event is the design and manufacture of the winches. Checks with established airborne winch designers and manufacturers have confirmed that fifteen months is a realistic figure. Other items of equipment procured from outside sources by winch manufacturers all fall within this time span and include:

1. Hydraulic Motors (12 months)

2. Load Isolators (6 months)

3. Tensiometers (6 months)

4. Cable Cutters or Shaped-Charge Devices (5 months)

5. Cargo Hooks (9 months)

6. Cable Tension Reels (10 months)

Event No. 2 Time Span

A total estimated time span of fourteen months for this event assumes that the system is fully synchronized. Such a system could be built with existing technology. If a nonsynchronized system was considered acceptable for this mode, the development time span could be reduced to six months. In either case, the development time span is less than that required for Event No. 1.

Event No. 3 Time Span

A total estimated time span of twenty-four months for this event assumes that the system is fully synchronized. The major element dictating this is the estimated six months required to establish the basic criteria before system design can begin. The long development time of this event exceeds that of Event No. 1, which results in an extension of the overall system qualification time to thirty months.

It should be noted that this schedule does not include any flight testing.

PLANNING COST ESTIMATE

Planning estimates for Events No. 1 and No. 2, which assume that two complete sets of hardware will be manufactured, have been derived from estimates of similar systems. The estimates include full qualification by simulation of flight parameters but exclude an operational evaluation (i.e., flight testing). The estimates which follow do not include the cost of a test facility:

1. Single-point-mode plus two-point-mode systems complete with control system giving synchronized control of the two-point mode:

   $400,000 to $600,000.

2. Single-point-mode plus four-point-mode systems complete with control system giving synchronized control of the four-point mode:

   $700,000 to $800,000.

In addition to the above, it is estimated that a test facility having a capability to test the above systems would cost:

$80,000 to $100,000.

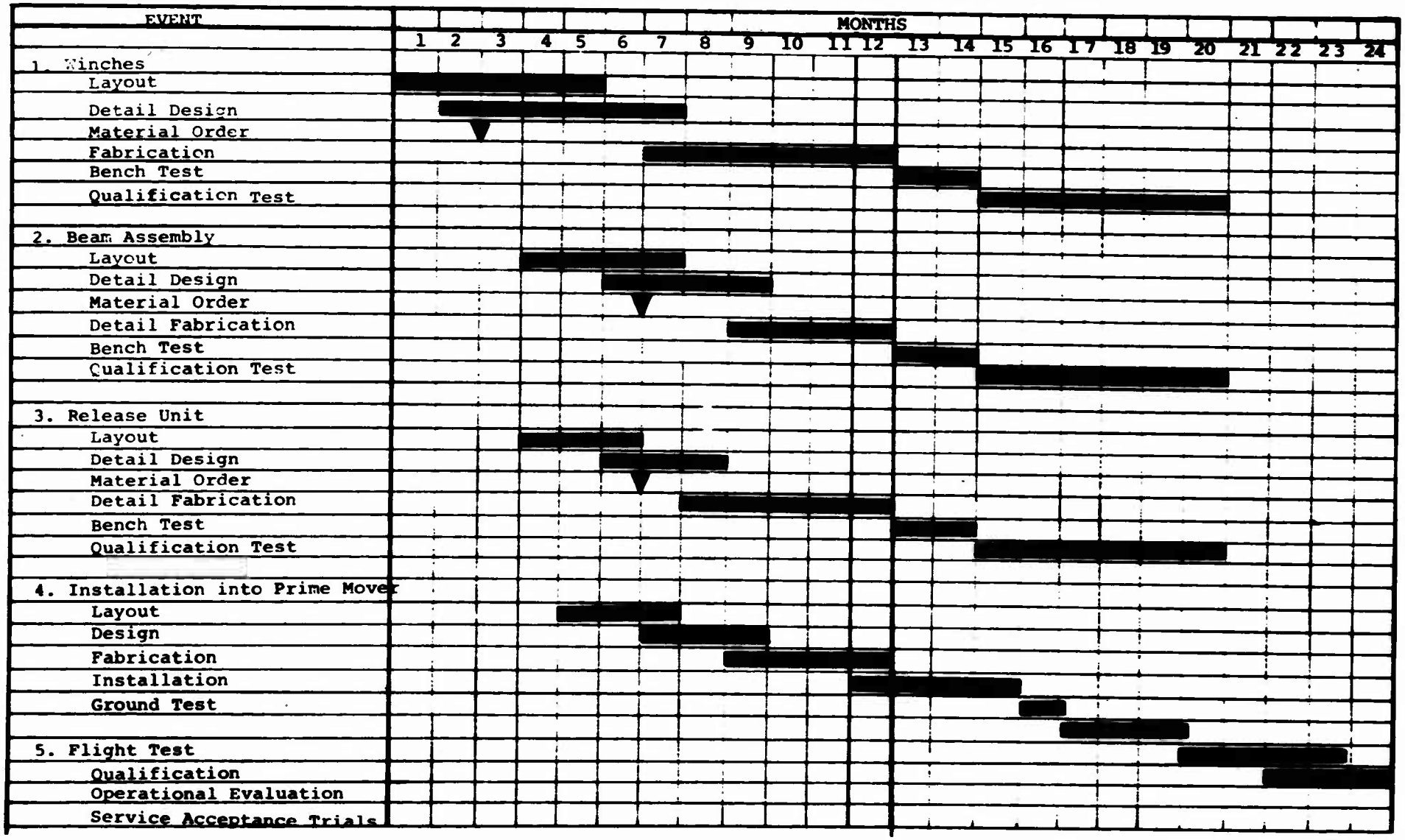


Figure 94. Planning Schedule for a 40,000-Pound Hoist System.

PHASE II - PRELIMINARY DESIGN

The objective of the Phase II, preliminary design part of the heavy-lift helicopter external cargo handling system study is to produce preliminary design drawings in sufficient detail to provide a basis for future detail design, fabrication and qualification of an experimental hoist system. In support of the preliminary design, a preliminary load and stress analysis, a maintainability analysis and a reliability analysis are provided. These supports will confirm the feasibility and practicality of the hoist system and will pinpoint those areas where independent detail development may be necessary to eliminate any "back to the drawing board" philosophy during system development.

GENERAL DESCRIPTION OF SYSTEM

The system is based on Vertol's recommendation (refer to Paragraph 8) in the Conclusions and Recommendations section of the Phase I Design Analysis section of this report. The experimental hoist system is a two-winch, two-point system consisting of two capstan-type winches, the arrangement of which results in a winch system free to pivot about the prime mover's longitudinal axis. The suspension system (one wire rope from each winch) terminates at the ends of a beam to which are secured two cargo hooks. The beam forms a common base for the interconnection of the hook release mechanisms, permitting simultaneous release of the hooks through a common electromechanical actuator, thus satisfying the specified normal mode of release operation. Manual release of the system (emergency mode) utilizes the same basic linkage except that a manual input is applied in lieu of an electrical one.

The electrical conductors with integral manual release input cable are run from the beam to the prime mover, where they are stored on a constant torque reel, automatically deploying in or out as the beam rises or falls.

DESCRIPTION OF SYSTEM COMPONENTS (Refer to Table IX)

Winches

Two identical winches are utilized. The winches are of a capstan type configured as illustrated in Figure 95. It will be noted that the winch is mounted so as to be free to pivot

TABLE IX
LEADING PARTICULARS OF WINCH SYSTEM

Winches (2)	Capstan Type
Dynamic Capacity (per winch)	25,000 lbs
Lift Rate	60 ft per min
Wire Rope	1-in. dia, type 6 X 37 Warrington Searle
Wire Rope Dia/Drum Dia Ratio	1:18
Lift Height	150 ft
Cargo Hooks (2) - Type	Mechanical release
Release Normal Mode	Electromechanical
Release Emergency Mode	Mechanical
Indicators	Hook open/hook closed
Electrical Conductor Reel	
Normal Operation	Electrical torque motor
Emergency Release	Mechanical
System Control	
Winch Control	Independent variable-speed control of each winch plus integrated control through one four-way switch
System Emergency Release	Dual-cartridge type cable cutters on each winch - electrically initiated through separate circuits

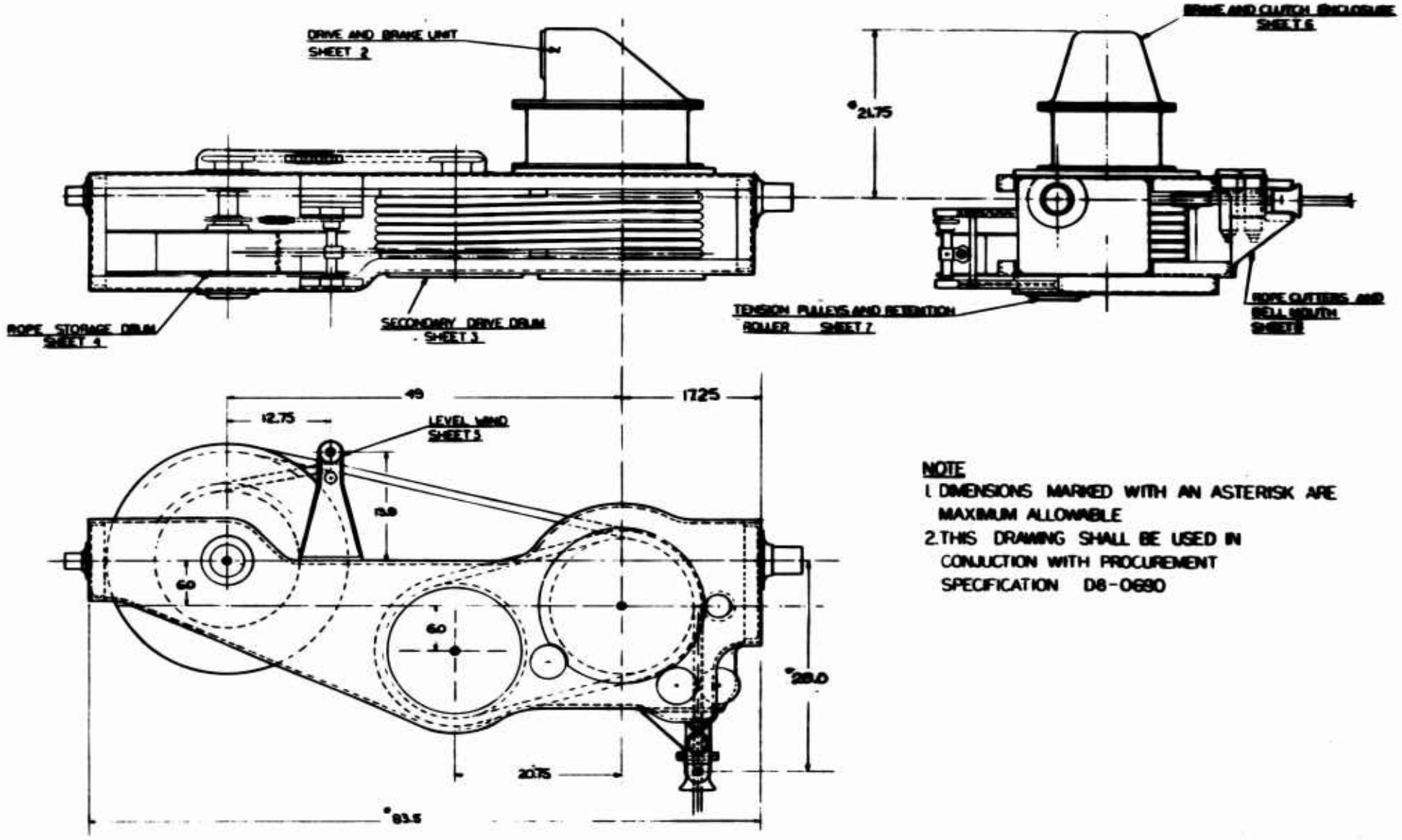
TABLE IX Continued	
Cable Length Indicator	Provision on winches for driving a cable footage indicator pot plus limit switches to control max & min cable lengths
Braking System	Automatic load brake system, in which the brake unit is an integral portion of the drum drive mechanism

on its longitudinal axis; such an arrangement eliminates localized bending of the wire suspension rope due to lateral swinging of the load.

Essentially the winch consists of two side beams, each end of which terminates in a self-aligning ball-type support bearing. The side beams support the two capstan drive drums and the cable storage reel. The relationship of the drums and storage reel is such that wire rope fleet angles are kept within acceptable limits. (The angles on the design shown in Figure 95 are 1.8 degrees between drum and drum and 2.4 degrees from drum to storage reel. The 2.4 degrees, while being in excess of the recommended practice of 2 degrees, is considered acceptable since the wire rope is subjected to a nominal tension load only between drum and storage reel.)

Capstan Drums

Both drums have a tread diameter of 18 inches (i.e., an 18:1 drum-wire rope diameter ratio). The "main" drum houses the three-stage planetary reduction gearing, which is the final portion of the winch drum drive, and automatic brake mechanisms. The secondary drum is driven from the main drum through an idler gear (see Figure 96). An external drive takeoff is provided from this idler gear for the installation of a cable footage indicator and limit switch system (see Figure 96). A chain sprocket is mounted on the drum to provide power to the storage drum.



NOTE
1. DIMENSIONS MARKED WITH AN ASTERISK ARE MAXIMUM ALLOWABLE
2. THIS DRAWING SHALL BE USED IN CONJUCTION WITH PROCUREMENT SPECIFICATION D8-0680

Figure 95. Winch Assembly, 20-Ton Hoist System.

Both drums are designed to be fabricated from light alloy castings. However, light alloys used to facilitate manufacture and to maintain an optimum strength-weight ratio are considered to be unacceptable as a wire rope tread (groove) material because of the high localized radial bearing pressure imposed by the wire rope. To accommodate this high radial bearing load and to ensure acceptable limits, wire rope life steel treads, made from alloy steel, have been embodied on the drum rims. The steel treads are keyed to the basic drum to provide shear transfer, clearance and flexibility to compensate for the differential expansion of the light alloy inner drum and steel tread rim. (See Figure 97.)

The grooves, for wire rope guidance on the drums, are shown with a profile having a flat bottom tread. Since the wire rope must traverse from grooves on one drum to adjacent grooves on the other drum, it would tend to climb out if the groove were circular in profile, thus imparting additional tension to the rope between drums. Since no qualification for this eventuality exists, it is recommended that specimens of different groove profiles be tested to obtain the optimum profile. It will be noted, however, that the basic design shown (refer to Figure 97) provides for changing the tread rims without redesign of the basic drums. (Such philosophy also permits the changing of rims during winch overhaul without the necessity of changing complete drums.)

Drum Drive and Brake System

Figure 98 presents a schematic diagram of the main drum drive and automatic brake mechanism which is comprised of the following main components:

1. Hydraulic motor
2. Initial reduction gearing
3. Reverse-locking clutch
4. Intermediate gearing
5. Drive shaft with Acme screw thread at one end

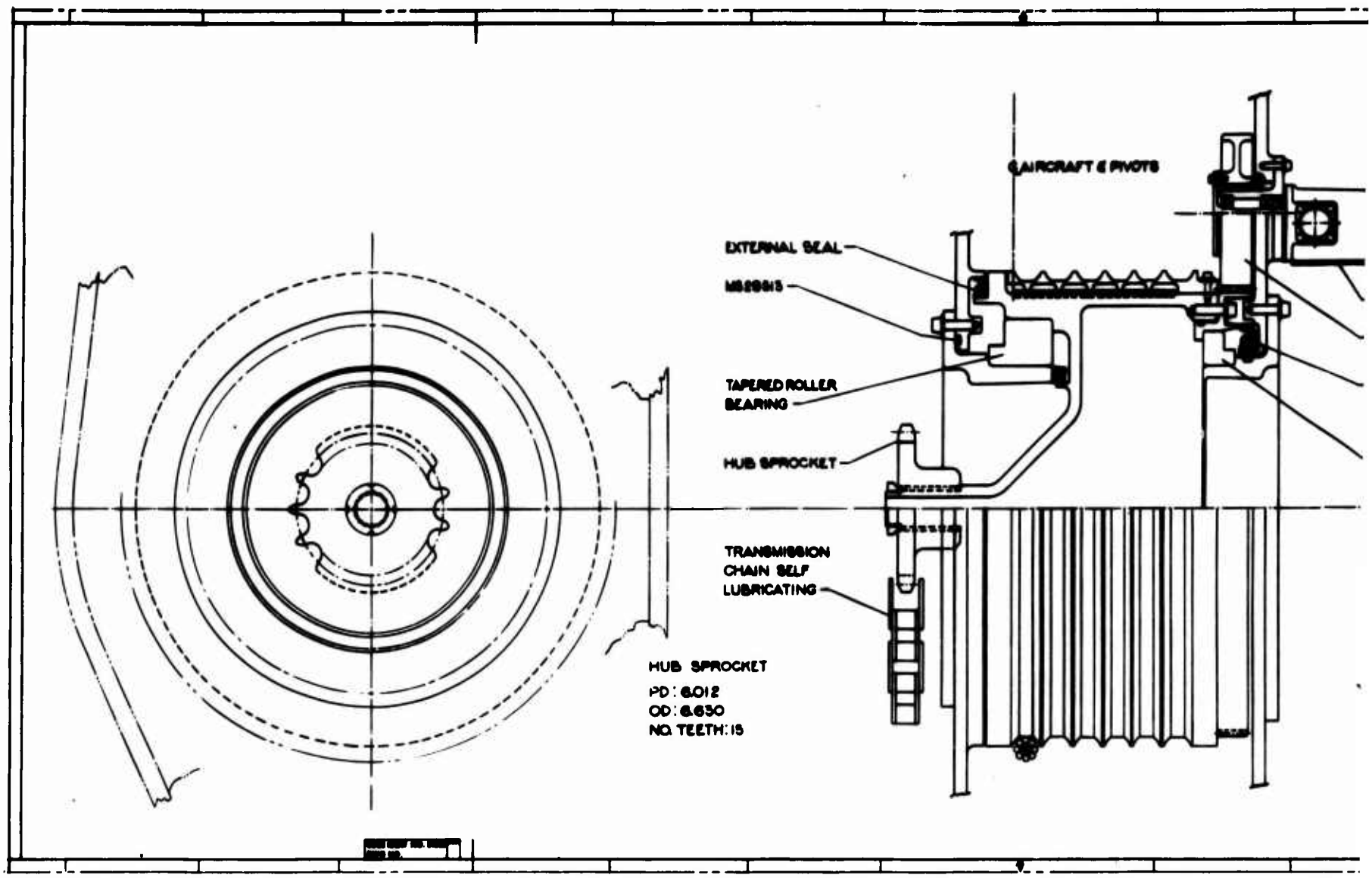
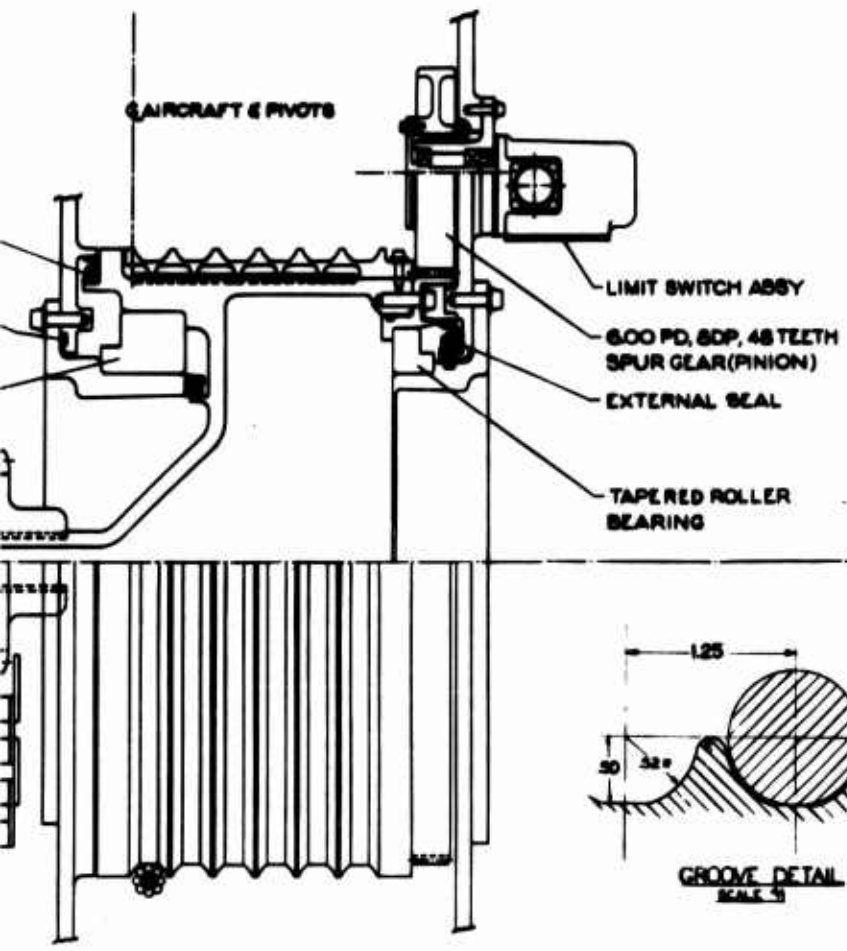


Figure 96.Detail of Secondary Drum Winch, 20-Ton Hoist System.


AIRCRAFT & PIVOTS
LIMIT SWITCH ASSY
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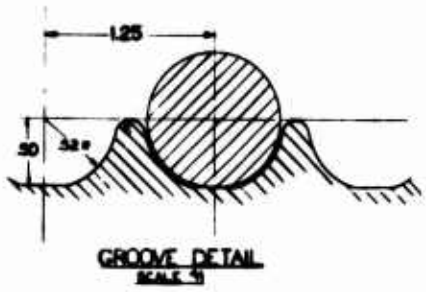
EXTERNAL SEAL
TAPERED ROLLER BEARING


1.25
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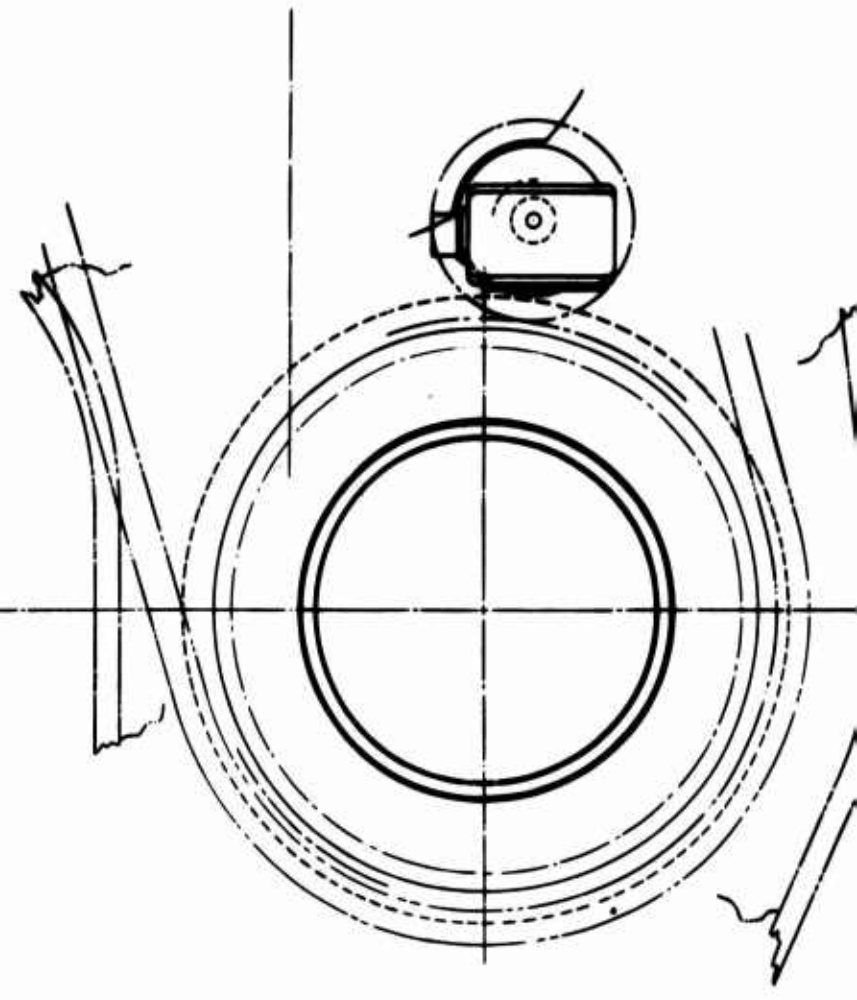
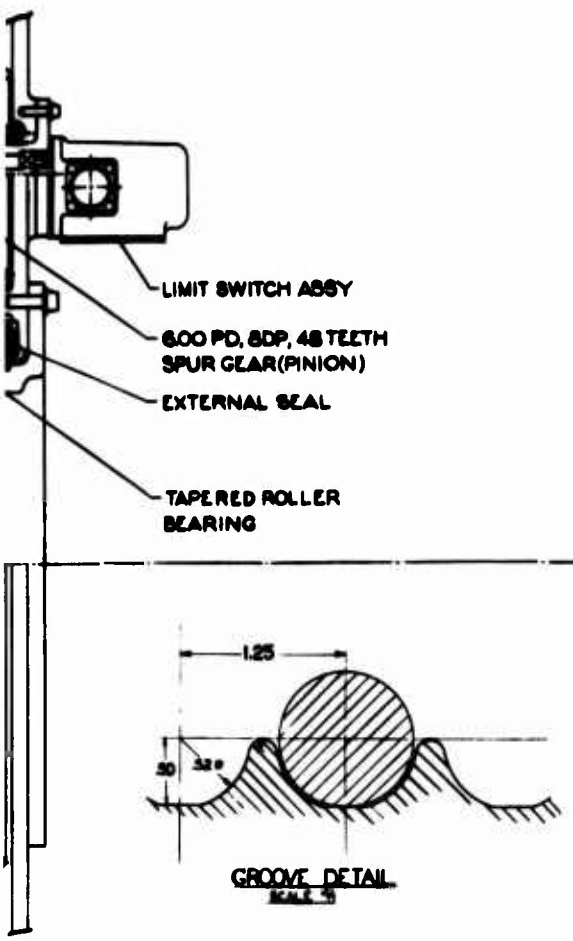
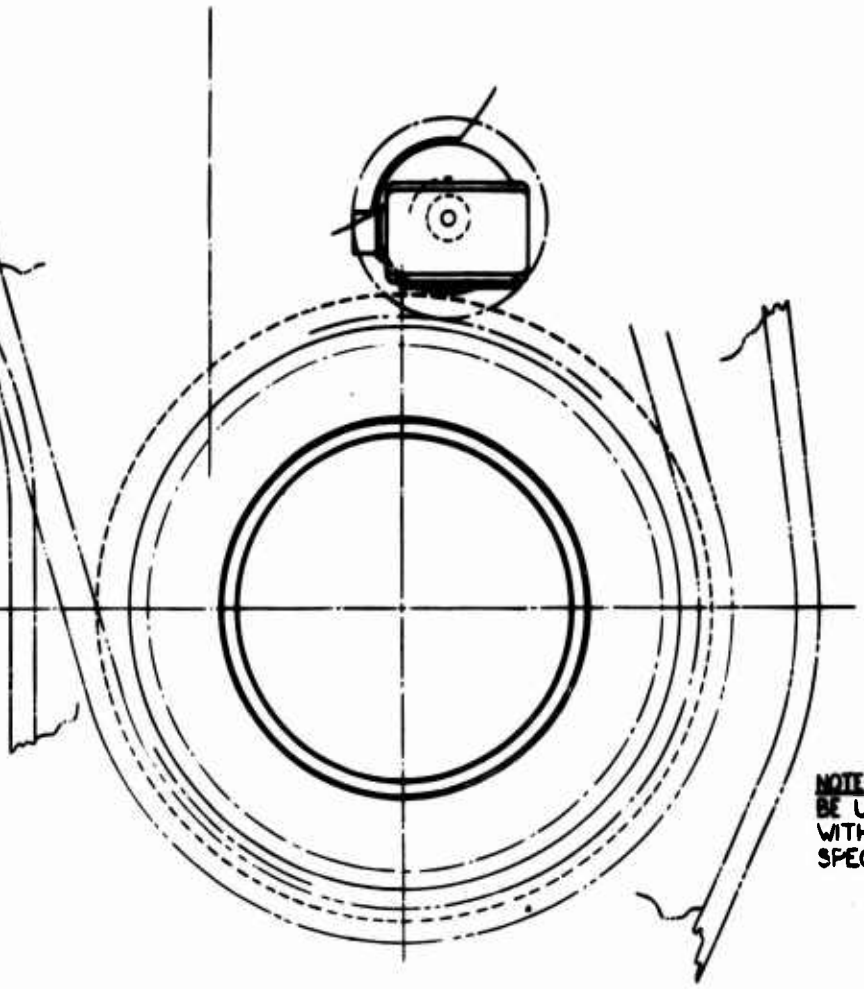
32°
GROOVE DETAIL



st System.



GROOVE DETAIL
SCALE 2/1



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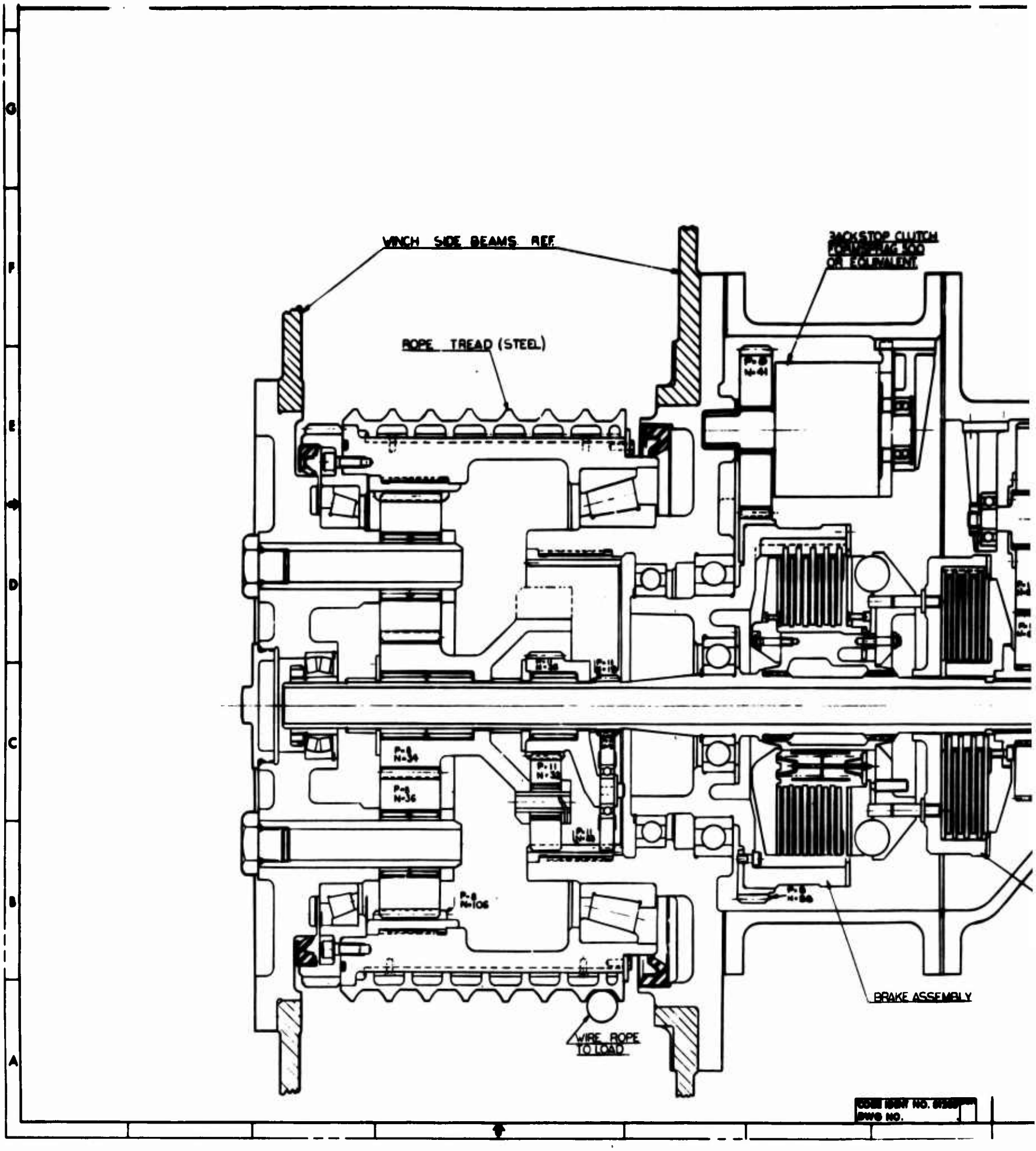
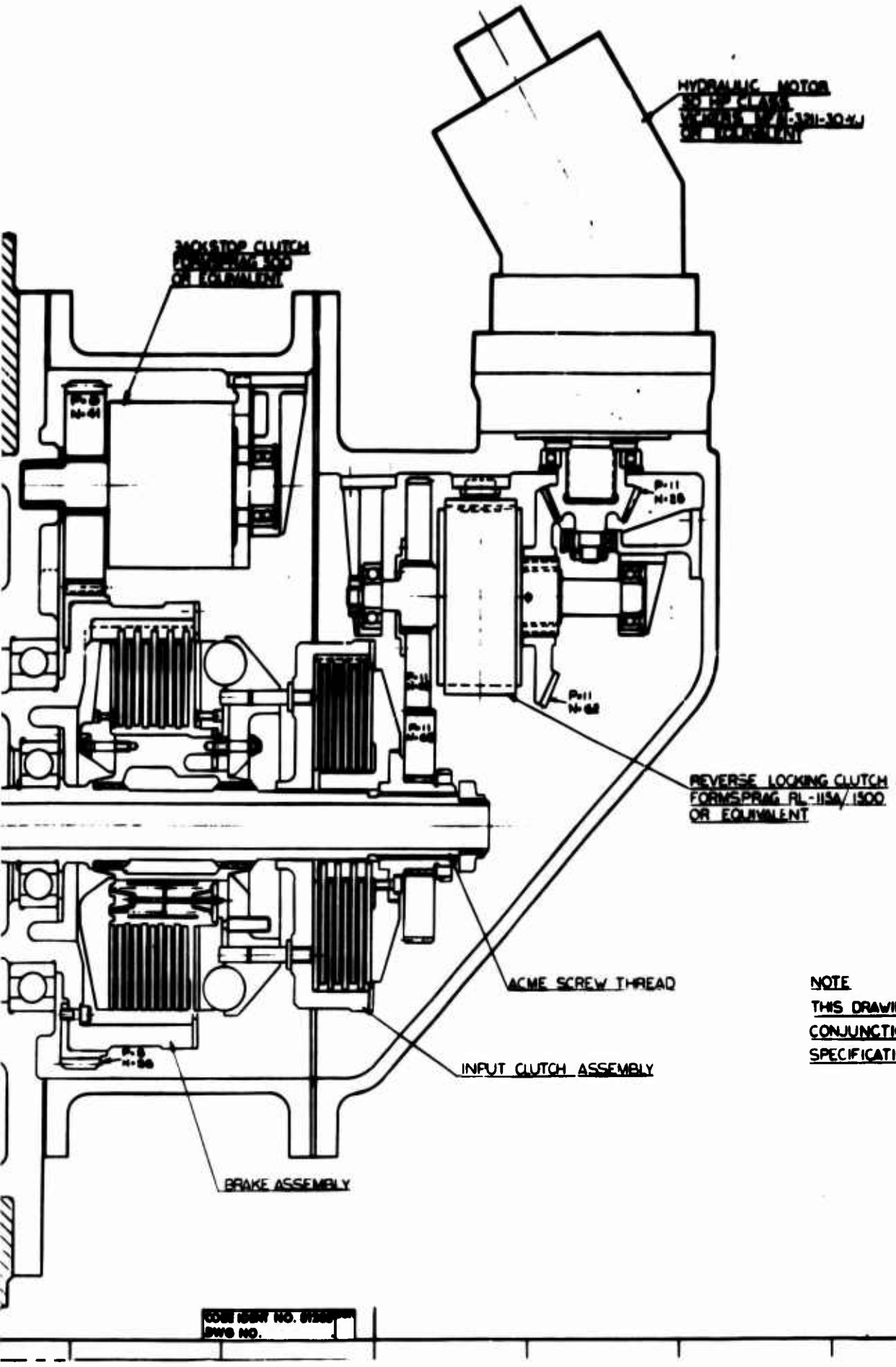


Figure 97. Winch Drum Drive and Automatic Brake, 20-Ton Hoist System.



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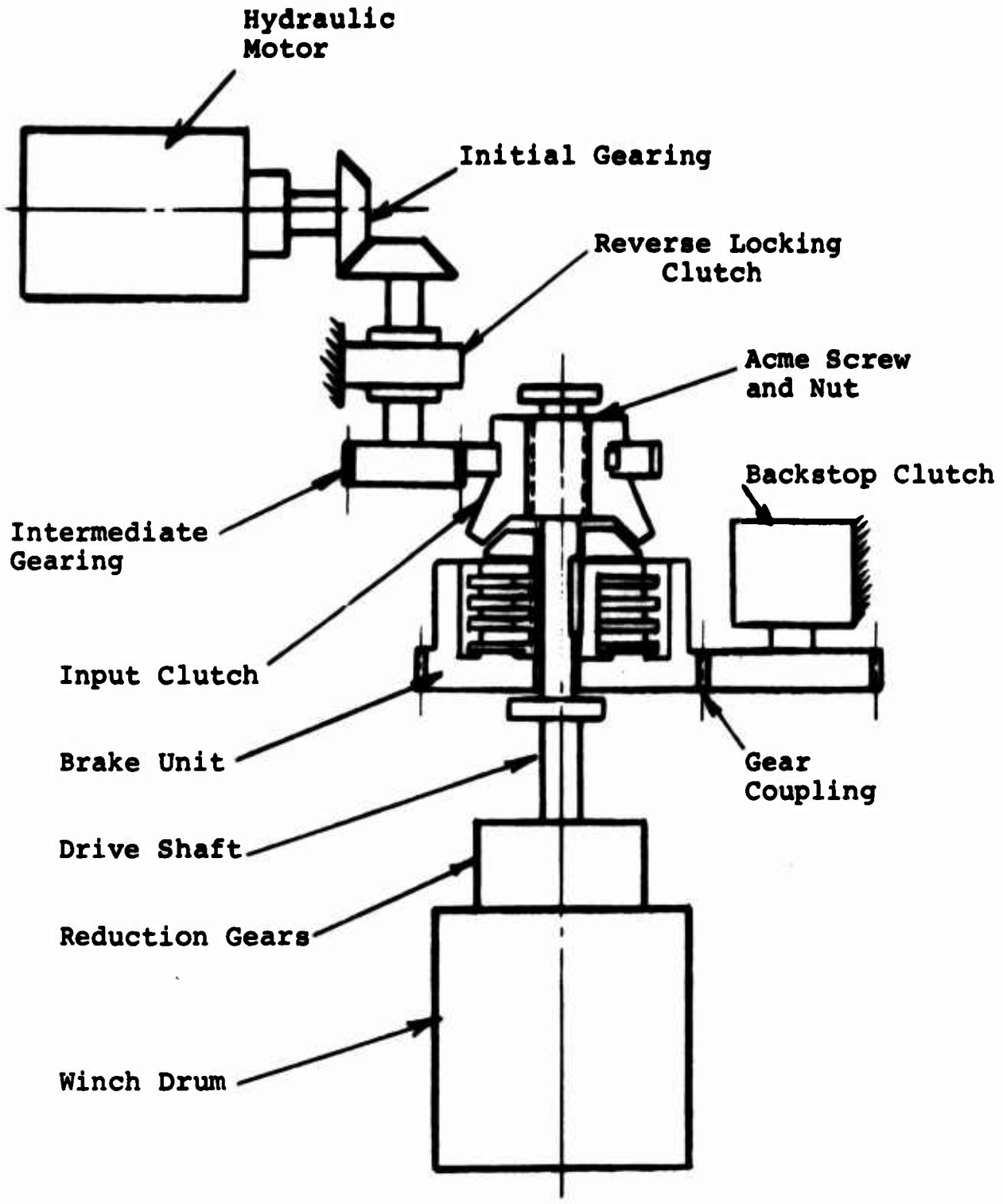


Figure 98. Winch Drum Drive and Automatic Brake Schematic Diagram.

6. Screw nut which incorporates an input clutch

7. Brake assembly

8. Backstop clutch

9. Drum reduction gearbox

System Functions

1. Hoisting the Load

When fluid power is supplied to the motor and the motor starts to rotate in the load lifting direction, rotary motion is transmitted through the initial gearing, the reverse locking clutch and intermediate gearing to the screw nut, to which one of the gears is keyed. Before the drive shaft can be rotated, sufficient friction torque in the input clutch must be built up. Thus, the sequence of events is such that the screw nut first starts to rotate on the shaft screw and by so doing displaces itself axially towards the brake assembly. Next, when the brake and the input clutch are sufficiently energized, the drive shaft also starts to rotate; then, through the gearbox, it overcomes the load reaction torque, and the load is raised. The fact that the brake is applied does not create an obstacle to the load lifting action because the backstop clutch, geared to the brake housing, permits the entire brake assembly to rotate freely in the load lifting direction. The backstop clutch acts in the same way as the ratchet and pawl mechanism, permitting rotation in one way and arresting the motion in the opposite way.

2. Holding the Load

As the brakes have already been energized on load lifting, the situation at the end of hoisting is such that the winch drum, gearbox, and drive shaft are all torsionally fixed through the brake action to the brake housing. Thus, when hydraulic power is released, any tendency by the drum to rotate in an opposite direction will be transmitted to the brake housing, which, in turn, will be immediately arrested by the backstop clutch and the load will be held steady.

If we now assume that the brake torque was less than that which is required to hold the load, and that brake slippage occurred, then again a relative motion will take place between the screw end of the drive shaft and the nut. The nut will be drawn towards the brake and will increase the brake loading force until the brake slippage stops. The reverse-locking clutch will not permit the nut to rotate with the screw without axial displacement and thus assures the brake loading action as a result of the excessive load torque.

3. Lowering the Load

To lower the load, it is necessary to provide hydraulic motor torque in the direction opposite to load hoisting. The effect of motor input motion on the screw nut is such as to start unscrewing the nut away from the brake. This will diminish the brake loading force to such an extent as to obtain descent of the load at a rate determined by the motor speed. The screw nut acts as a governor and seeks a balanced position corresponding to torque equilibrium and load steady motion. If the winch drum picks up speed in excess of the equilibrium condition, the nut is drawn towards the brake and increases the braking torque, thereby slowing down the load.

If the drum speed starts to lag below the balance speed, the braking torque will be decreased by the nut outward motion, and the load speed will be correspondingly increased, to return to the balanced condition.

4. Lowering the Beam Hook System at No-Load Condition

The case is now considered when the weight of the hook (or beam with hooks) is not sufficient to overcome drum drive mechanism friction torque to unwind the rope. As previously described, the load lowering motion was initiated by providing hydraulic motor input motion to permit release of the brake. The drum motion was caused by the load torque. We now assume that when the brake is released, the drum will not start to rotate. In this case the screw nut will travel away from the brake until it reaches a stop collar. Further rotary motion applied to the nut will create an axial force, and the resulting buildup of torque is sufficient to rotate the shaft. Figure 97 represents the preliminary layouts of the drum drive and brake mechanism.

Storage Drum

Figure 99 shows in detail the build-up of the wire rope storage drum. Essentially, it is of light alloy construction suitably supported from the main winch beams. The drive to this drum is by drive chain from the second capstan drum. Since the effective storage diameter of the wire rope changes with rope length, and since a nominal tension is required in the wire rope to the capstan drums, the storage drum, during the lifting cycle, is overdriven through a slipping clutch (Figure 99). During the lowering cycle, the drive to the storage drum is severed by a one-way drive, torque reaction from the storage drum being reacted by a backstop device. Adjustment for the clutch is provided in order to set it at optimum breakout torque during qualification and to provide a means for adjustment to compensate for wear during service. In order to ensure the even wrapping of wire rope on the drum, a level wind has been embodied, this being chain driven from the storage drum. All chains are of the self-lubricating type, and all chain drives are encased to exclude foreign objects (see Figure 100).

All bearings used in the storage drum assembly are of prelubricated or sealed type or have provision for relubrication.

The capacity of the drum is 150 usable feet of wire rope, plus three complete turns for wire rope retention.

Wire Rope Cutters

Dual cable cutters are provided (Figure 101). These are of the cartridge type similar to those designed by Holex Incorporated, Holister, California.

Miscellaneous Details

1. Wire Rope Reel-Out

   Provision is made for powered reel-out of the wire rope (see Figure 102), the drive being taken through an idler gear from the main driving drum.

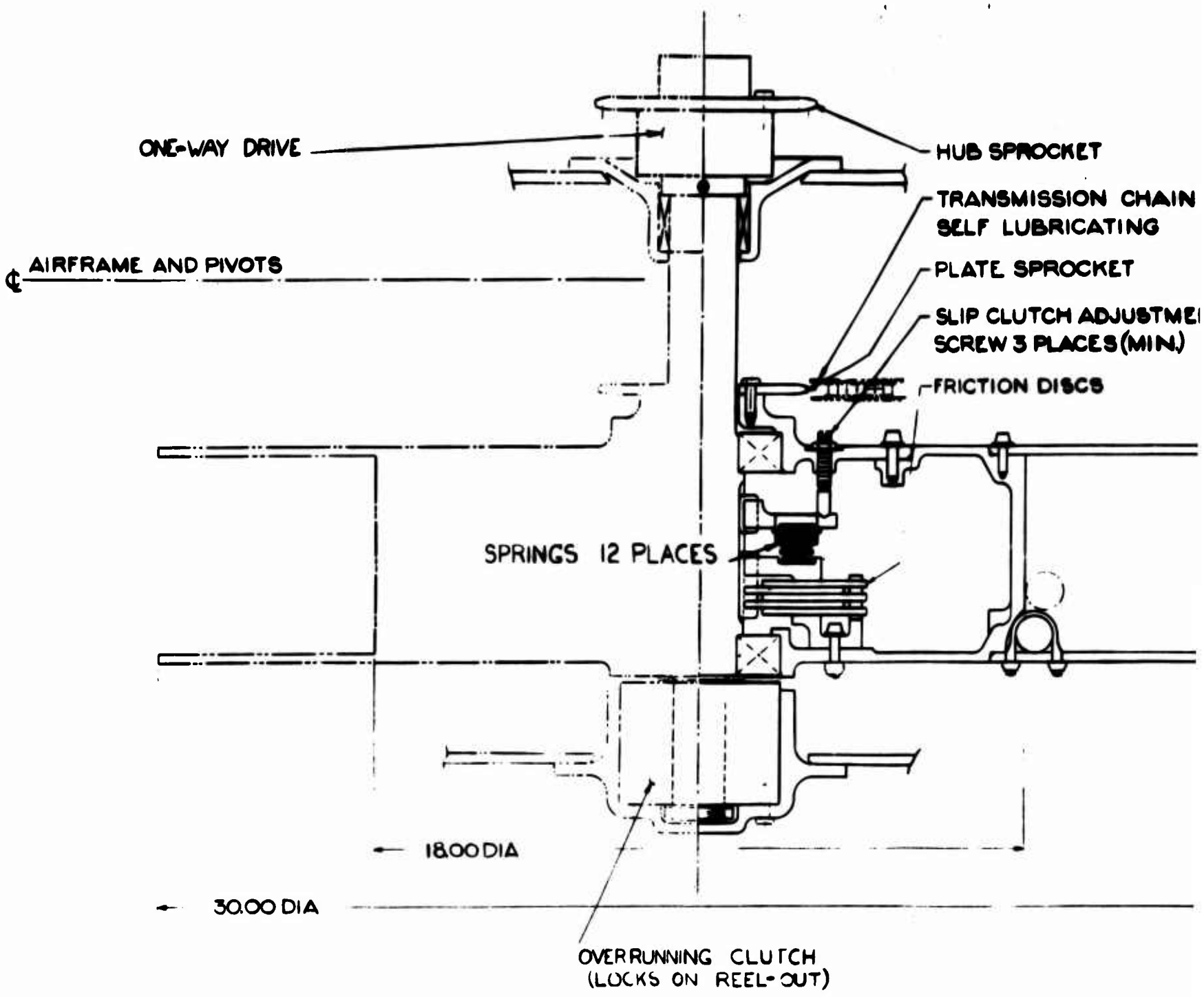
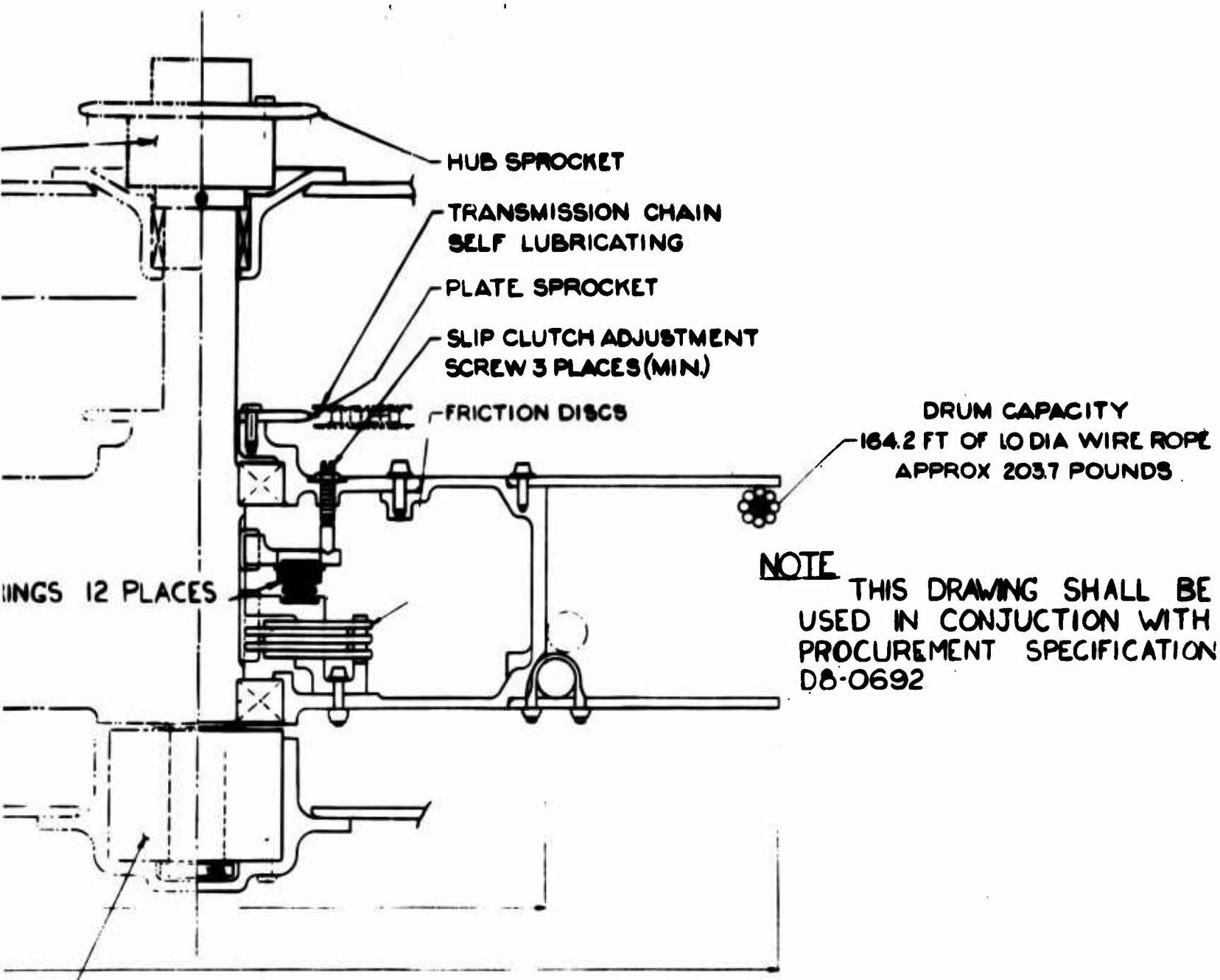


Figure 99. Detail of Storage Drum Winch, 20-Ton Hoist System.



-Ton Hoist System.

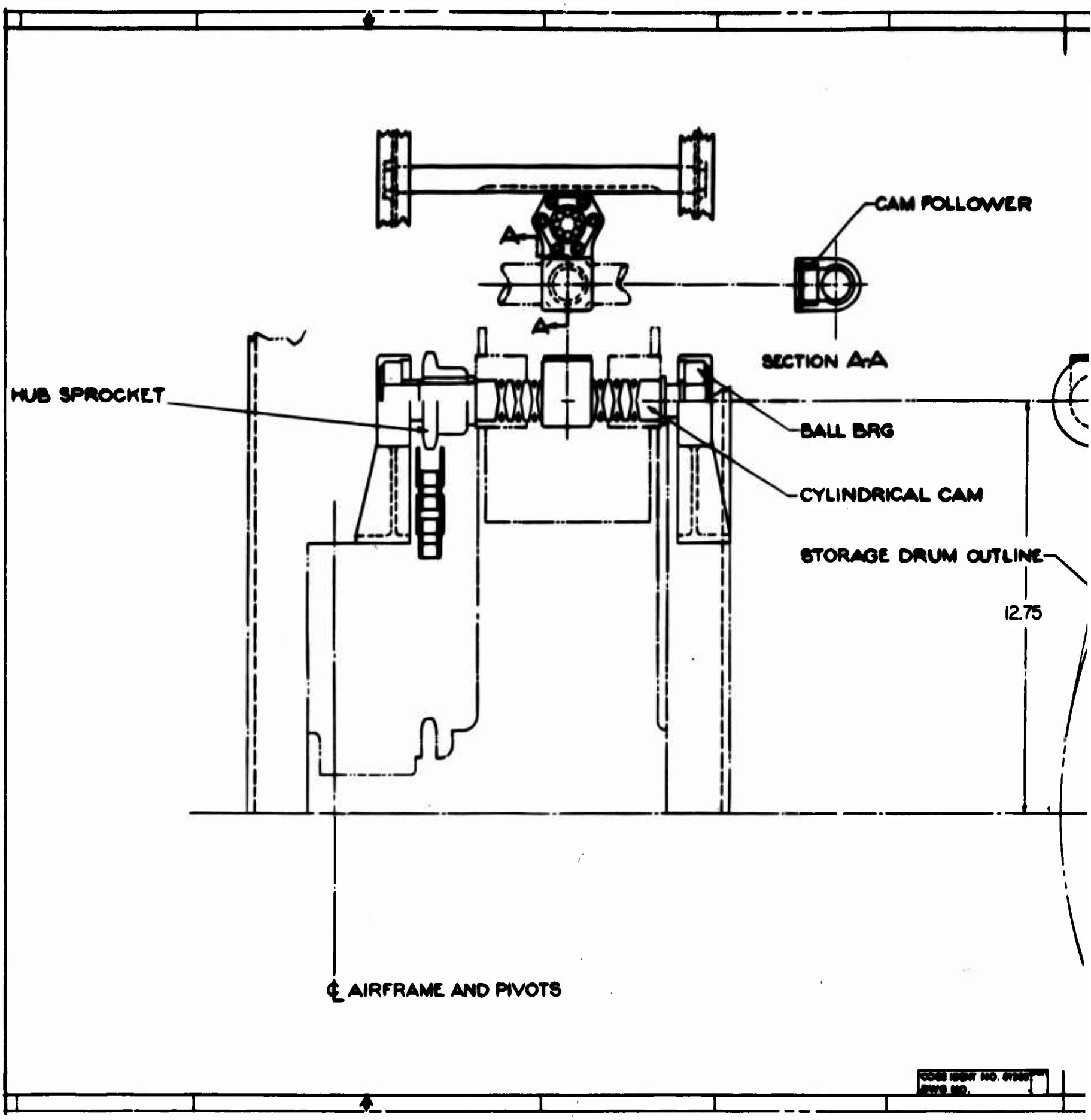
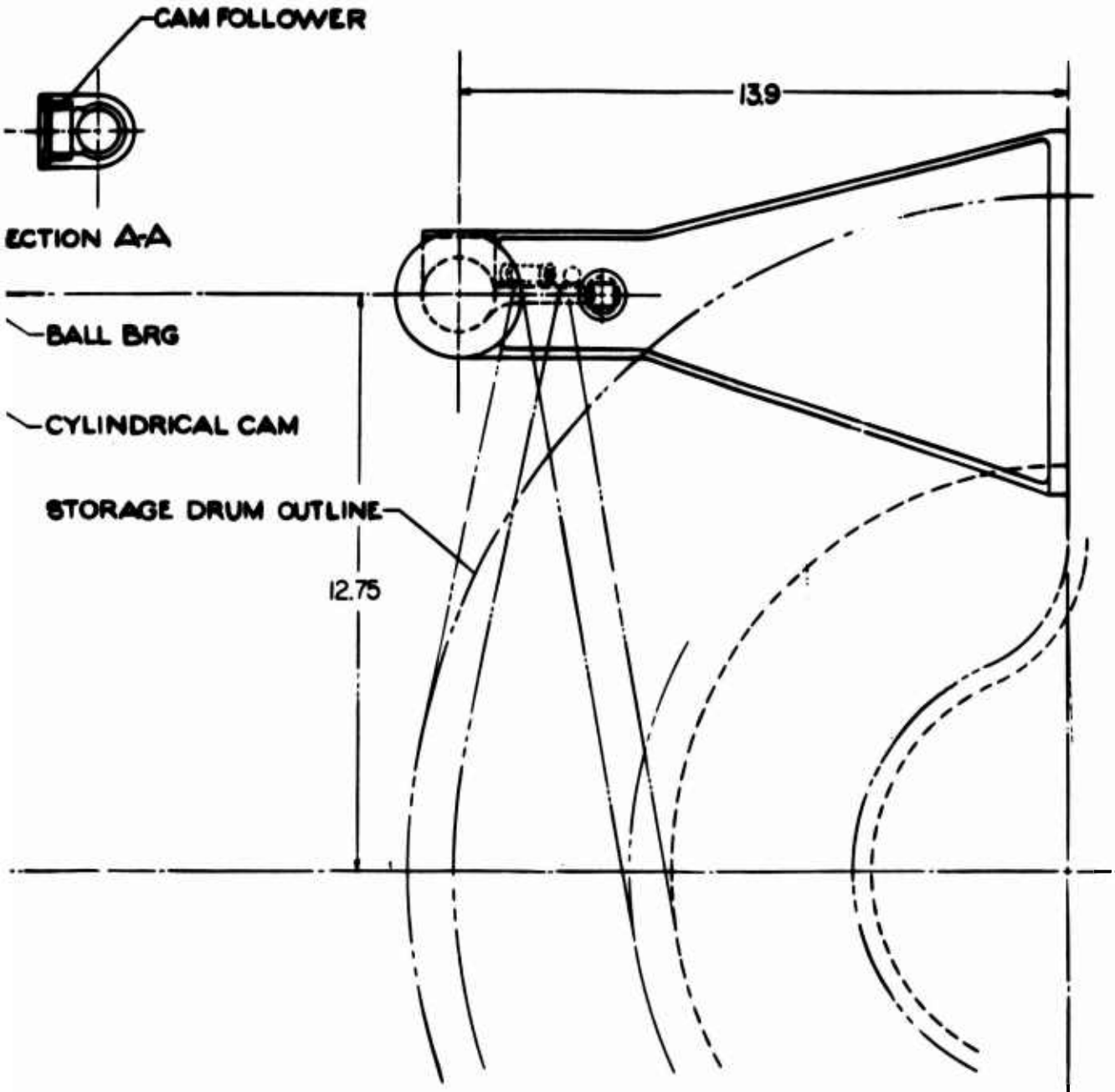


Figure 100. Detail of Level-Wind Winch, 20-Ton Hoist System.



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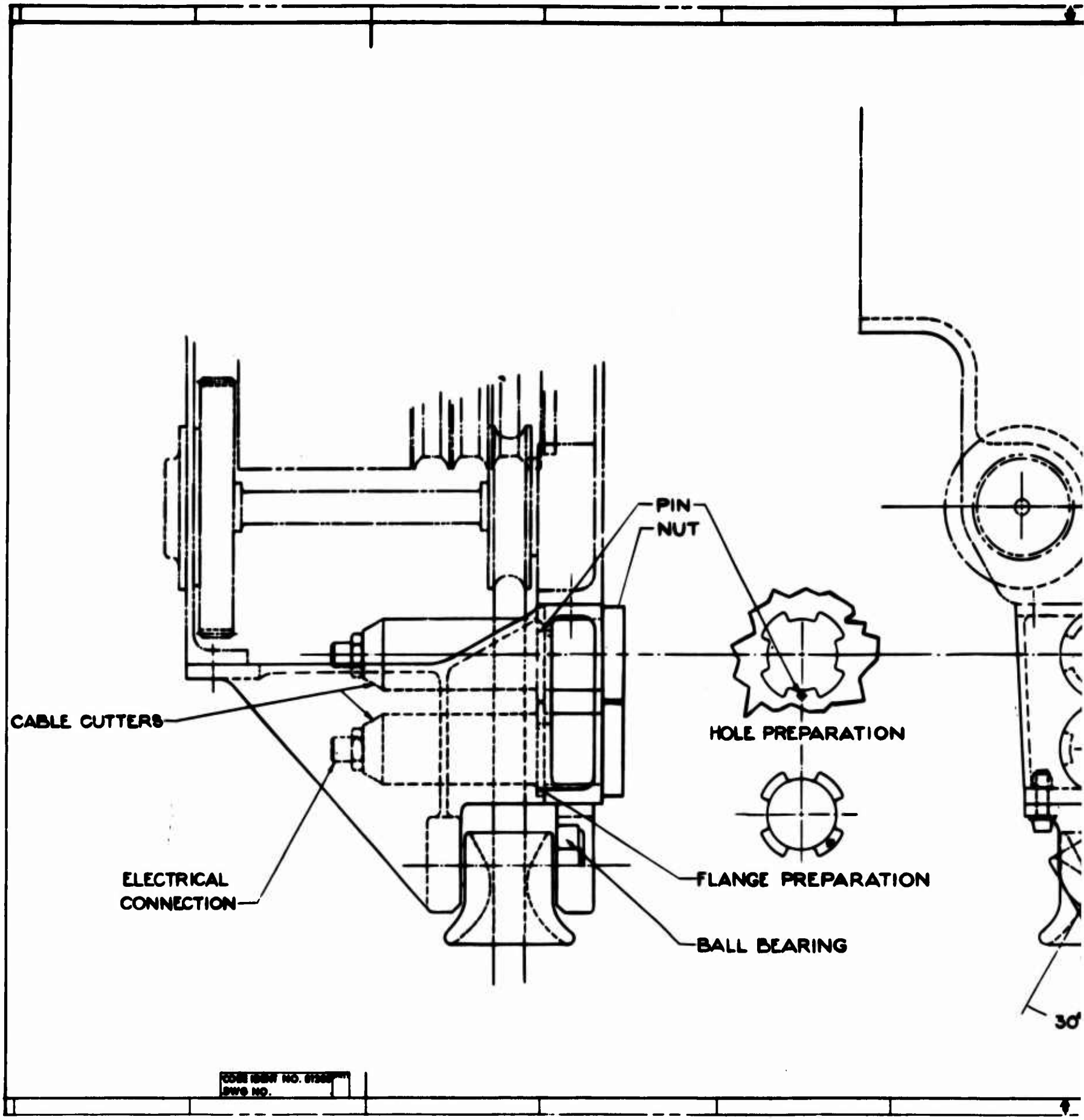
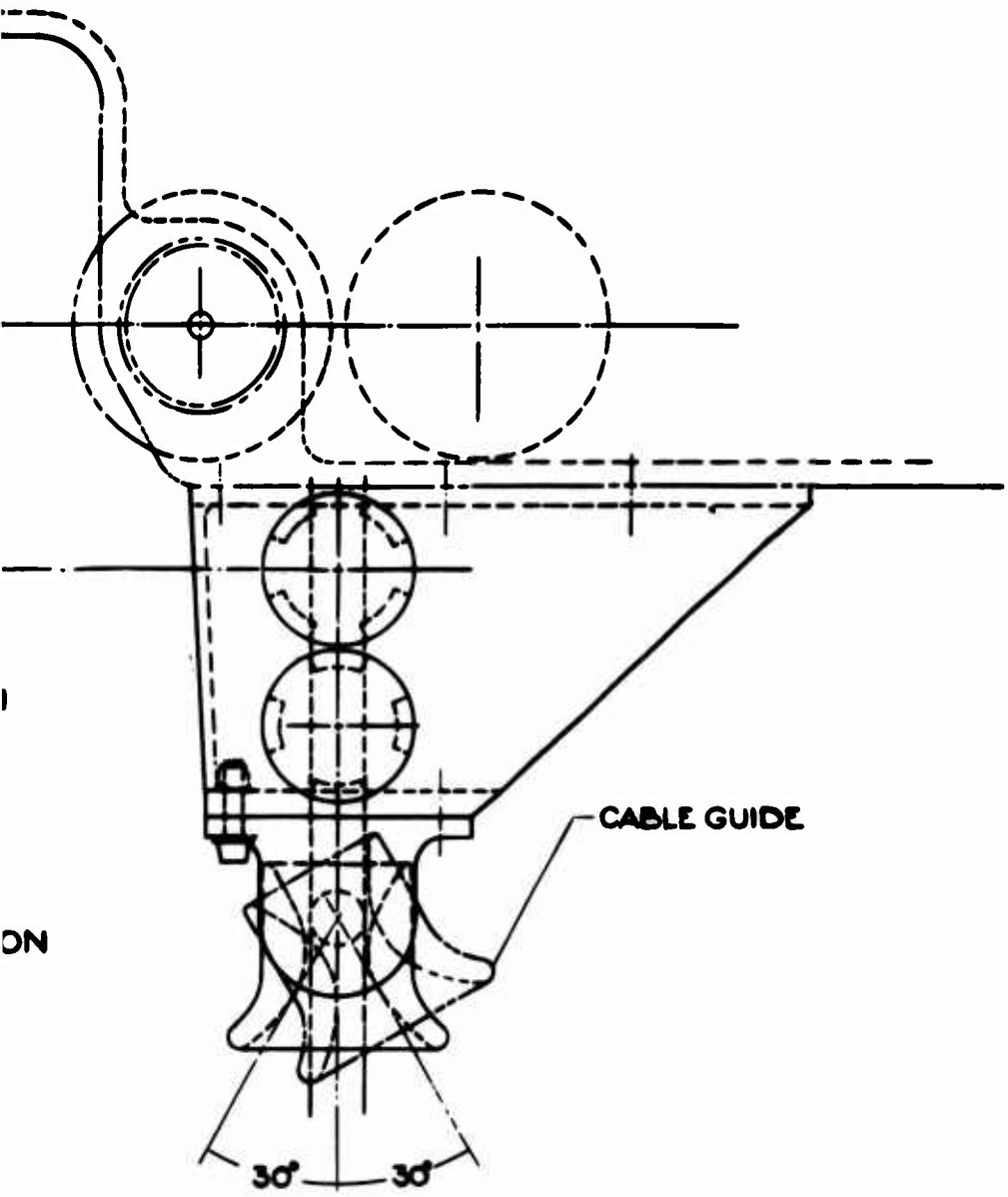


Figure 101. Detail of Rope Cutters and Bellmouth Winch, 20-Ton Hoist System.



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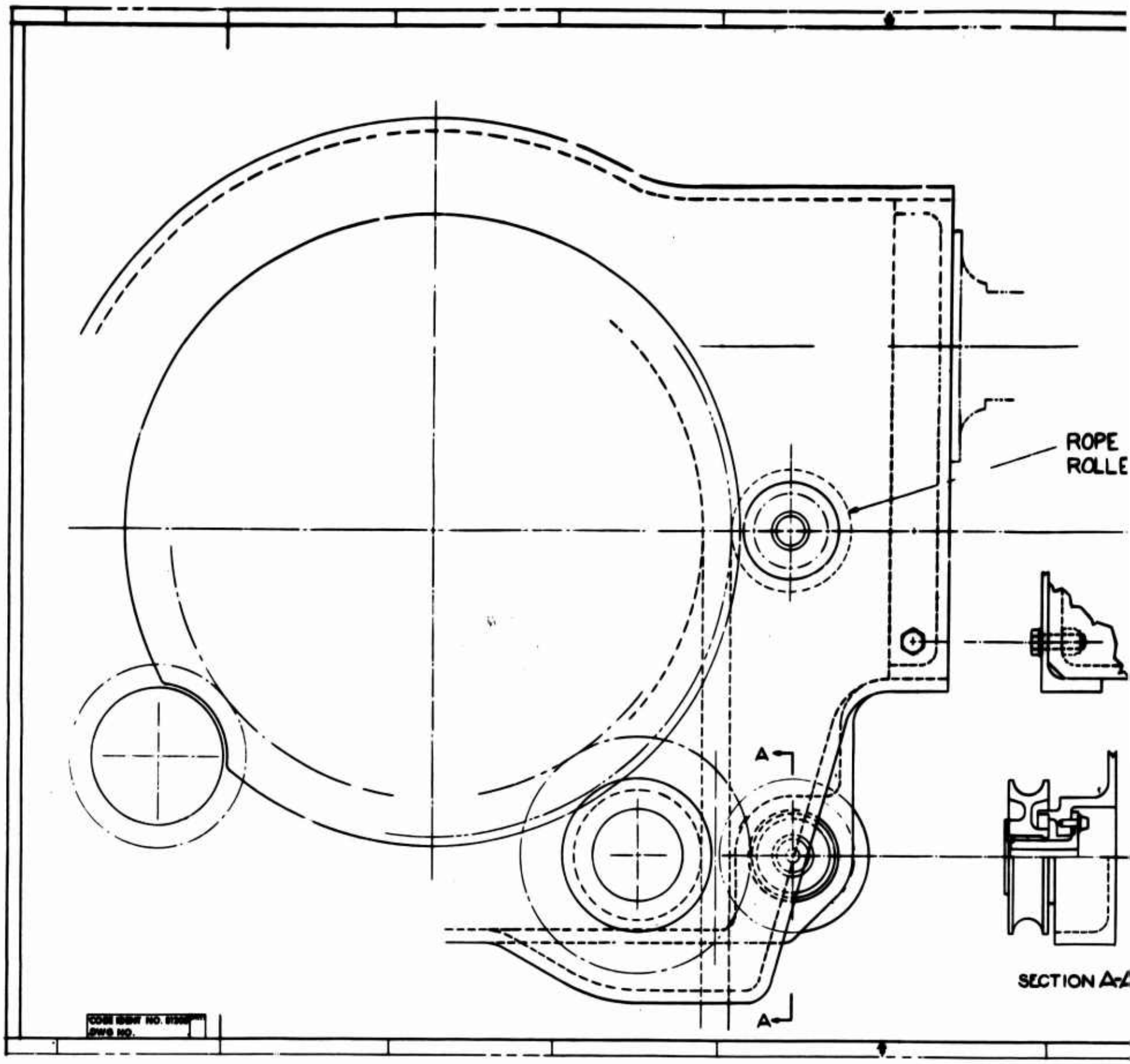

Figure 102. Detail of Tension Pulleys and Retention Rollers, 20-Ton Hoist System.

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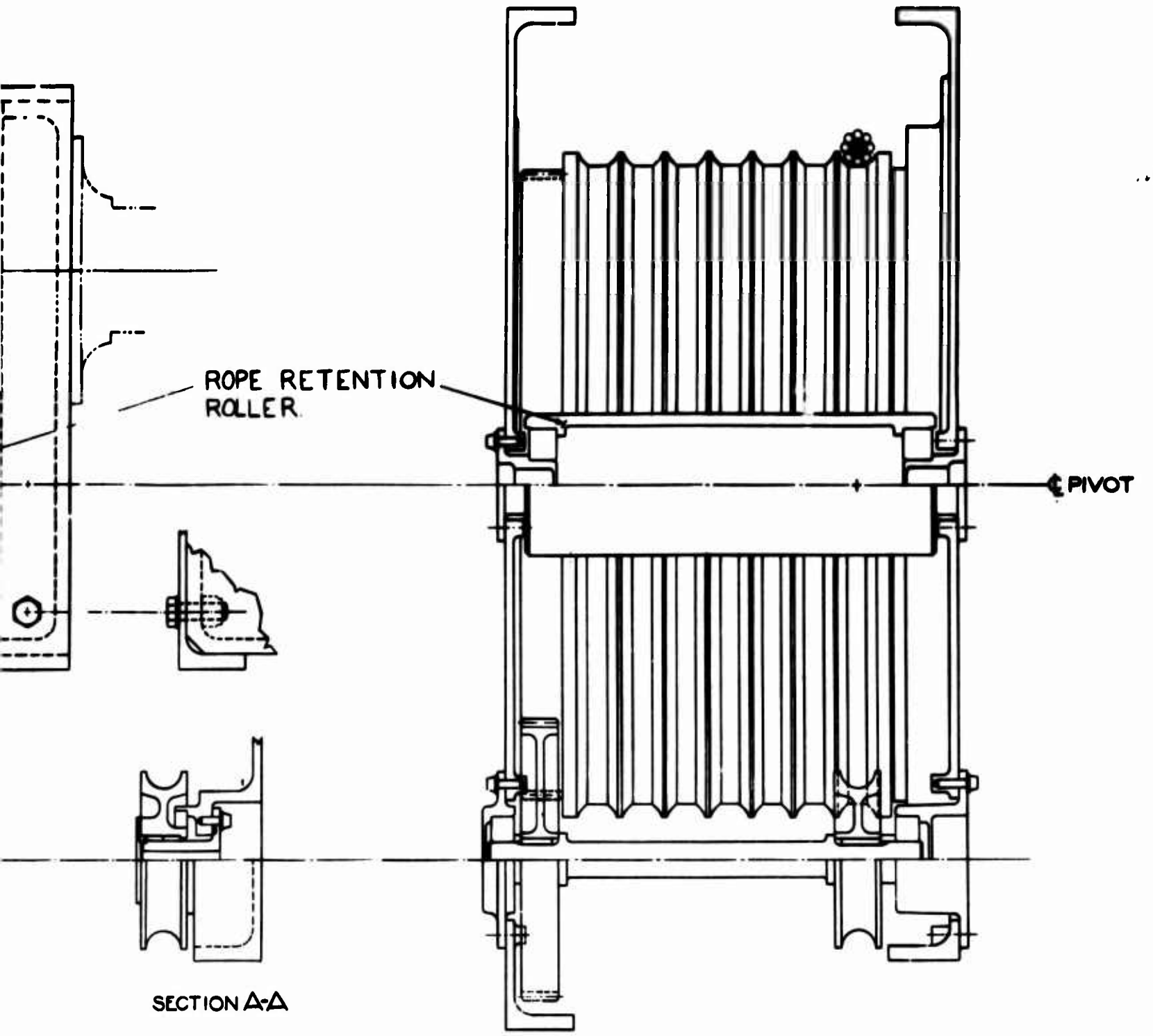

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2. Compensation for Wire Rope Springback

To prevent the wire rope from jumping out of the drum grooves due to wire rope springback , a roller is provided to retain the wire rope in the grooves (see Figure 102).

3. Bellmouth

The wire rope exit bellmouth is designed to rotate about the lateral axis of the winch. This reduces localized rope bending due to longitudinal swinging of the load. (Lateral load swinging is compensated for by the winch being free to pivot about the winch longitudinal axis, reference Figure 101.)

4. Closure of Open Areas of Winch

No provision has been shown on the basic design of the winch to protect the structure and personnel in the event of a wild rope, the reason being that this could be integrated with any complete winch "cover" during installation design.

5. Winch Mounts

Although an installation requirement, the establishment of winch mounts was necessary during the performance of this design study in order to establish the installation criteria for the load isolator. It will be noted that the main mount in the installation shown houses the load isolator and also carries into the prime mover structure any axial loads felt by the winch due to a swinging or trailing load. The remaining mount is pin-jointed to the prime mover structure in order to facilitate winch installation and removal (see Figures 103 and 104).

The winch installation shown in Figure 105 assumes mounting in a transport-type-configured prime mover, which places the load isolator in compression.

Beam Assembly

The purpose of the beam, shown in Figure 116, is to form a common base to permit mechanical interconnection of the cargo

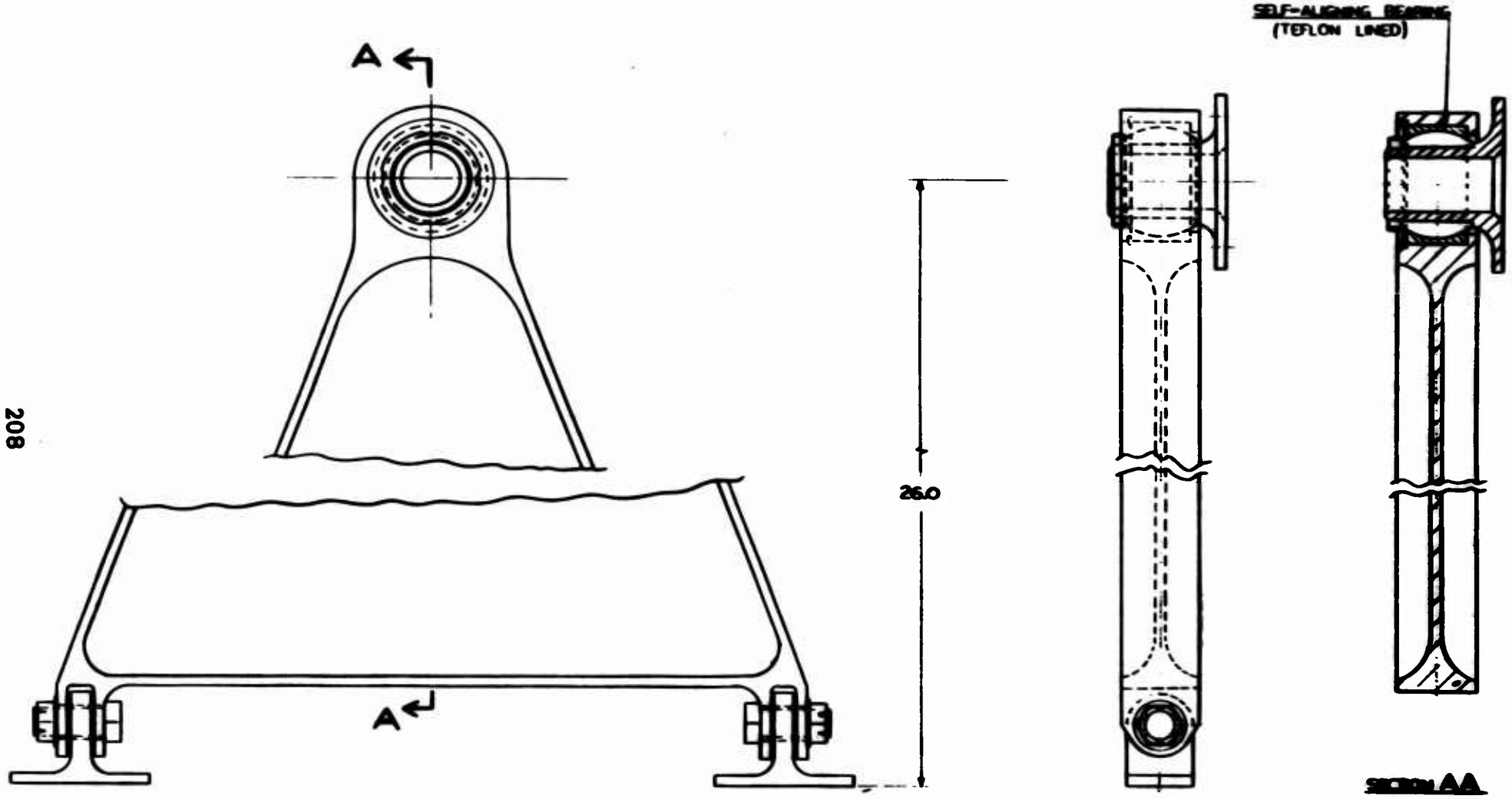

Figure 103. Support Winch, 20-Ton Hoist System.

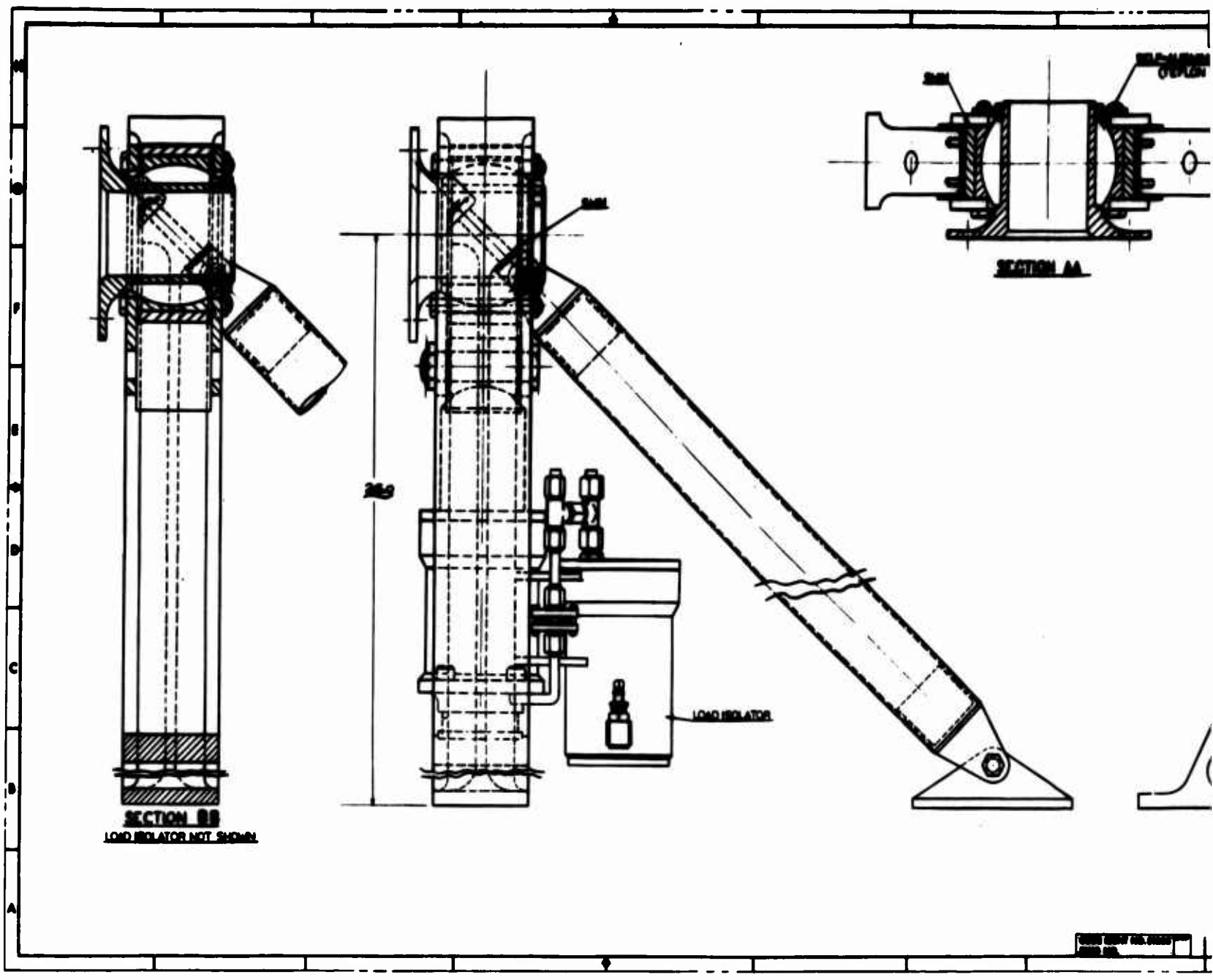

Figure 104. Main Winch, 20-Ton Hoist System.

SELF-ALIGNING BEARING
(TEFLON LINED)

SECTION AA

B

A

A

NOTE
STRUTS NOT SHOWN
IN THIS VIEW

GUIDE BEARING
SURFACE

LOAD
ISOLATOR

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CONJUNCTION WITH PROCUREMENT
SPECIFICATION DS-0888

B

LOAD CELL

2

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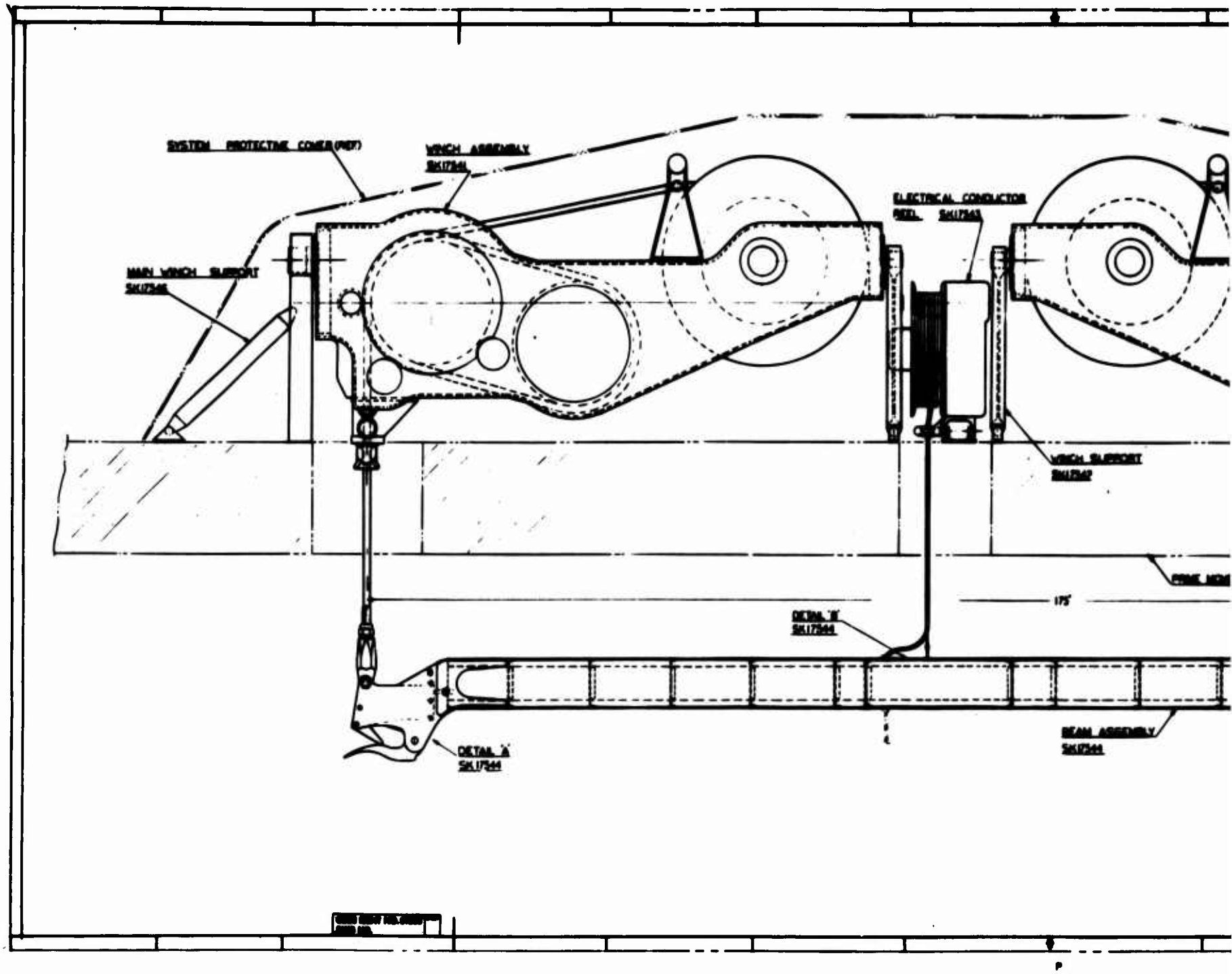
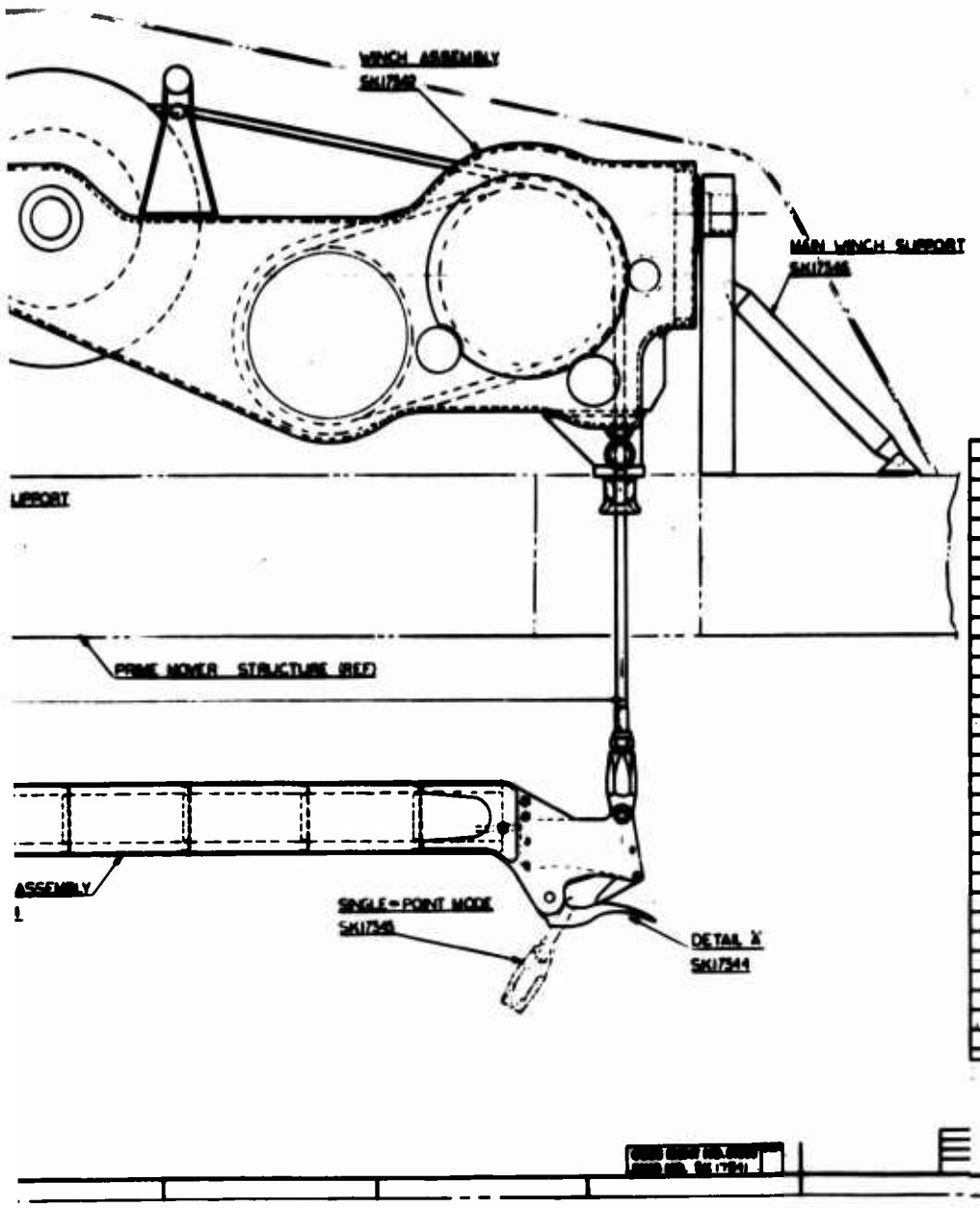


Figure 105. Component Relationship, 20-Ton Hoist System.



NOTES:

1 THE 175-INCH SUSPENSION ROPE SPREAD IS BASED ON AN ANALYSIS OF ANTICIPATED LOADS. A SPREAD OF 170 TO 180 INCHES IS CONSIDERED ACCEPTABLE. SUBSEQUENT DETAIL DESIGN OF SYSTEM COMPONENTS WILL ESTABLISH EXACT (MINIMUM) FIGURE.

2 INSTALLATION SHOWN IS IN A TRANSPORT-CONFIGURED HELICOPTER FOR INSTALLATION IN A CRANE CONFIGURATION. THE WINCHES WILL BE SUSPENDED, REQUIRING A REDESIGN OF THE MAIN SUPPORT (SK17546).

3 THIS DRAWING SHALL BE USED IN CONJUNCTION WITH PROCUREMENT SPECIFICATION D8-0888.

SK17547	WINCH SUPPORT
SK17546	
SK17545	
SK17544	ANTI-SKID LINK
SK17543	
SK17542	WINCH
SK17541	HOIST SYSTEM

THE BOEING COMPANY

SK17541

hook release mechanisms, necessary to ensure simultaneous release of hooks.

The beam is of conventional aircraft construction, being fabricated from standard aluminum alloy extrusions and aluminum alloy sheet. The cargo hooks are based on an established design. Those shown on Figure 116 have operating features identical with the Eastern Rotor Craft Corporation Model 2S-200-type military airborne cargo hook. The hook bodies have been modified to make them compatible with their installation onto the cargo beam assembly. The hooks are mechanically operated and are self-cocking (i.e., upon release the falling load winds up a torsion spring which stores up sufficient energy to return the hook to its "ready to load" status automatically). Electrical circuits (microswitches) for hook status indication are provided. No touchdown release is called for; however, the basic hook design is compatible should such a requirement be justified during future system development. To achieve the optimum strength-weight ratio, the hook assemblies have been made an integral part of the beam. This feature could be changed to permit the hooks to be mounted to the beam by two bolts (i.e., conventional hooks have two-point mounting holes as a standard method). The result of this change would be an increase in weight.

It will be noted that the hooks are "stepped" down from the beam centerline, the reason being that when the hoist system is used in a single-point lift mode, the beam is subjected to compression forces. The design shown allows these forces to be reacted on the ℄ of the beam, thus ensuring optimum weight-strength ratio for the beam. (Refer to Figure 107 for single-point mode.) However, the installation of the hoist system into a prime mover may well dictate that the beam system be recessed into the fuselage structure, in which case it may prove advantageous to design the beam structure such that the top surface of the beam is relatively flat, thus minimizing the cut-out depth in the prime mover structure.

Load Isolator and Load Cell

The load isolator shown is of the oleo-pneumatic type and is so designed that it will be installed as a self-contained system (i.e., no separate external pressure source or oil reservoir will be necessary for its operation). Figure 103 shows the load isolator mounted within the main winch mounting bracket. This type of installation calls for a compressive-type load isolator (compressive operation is considered

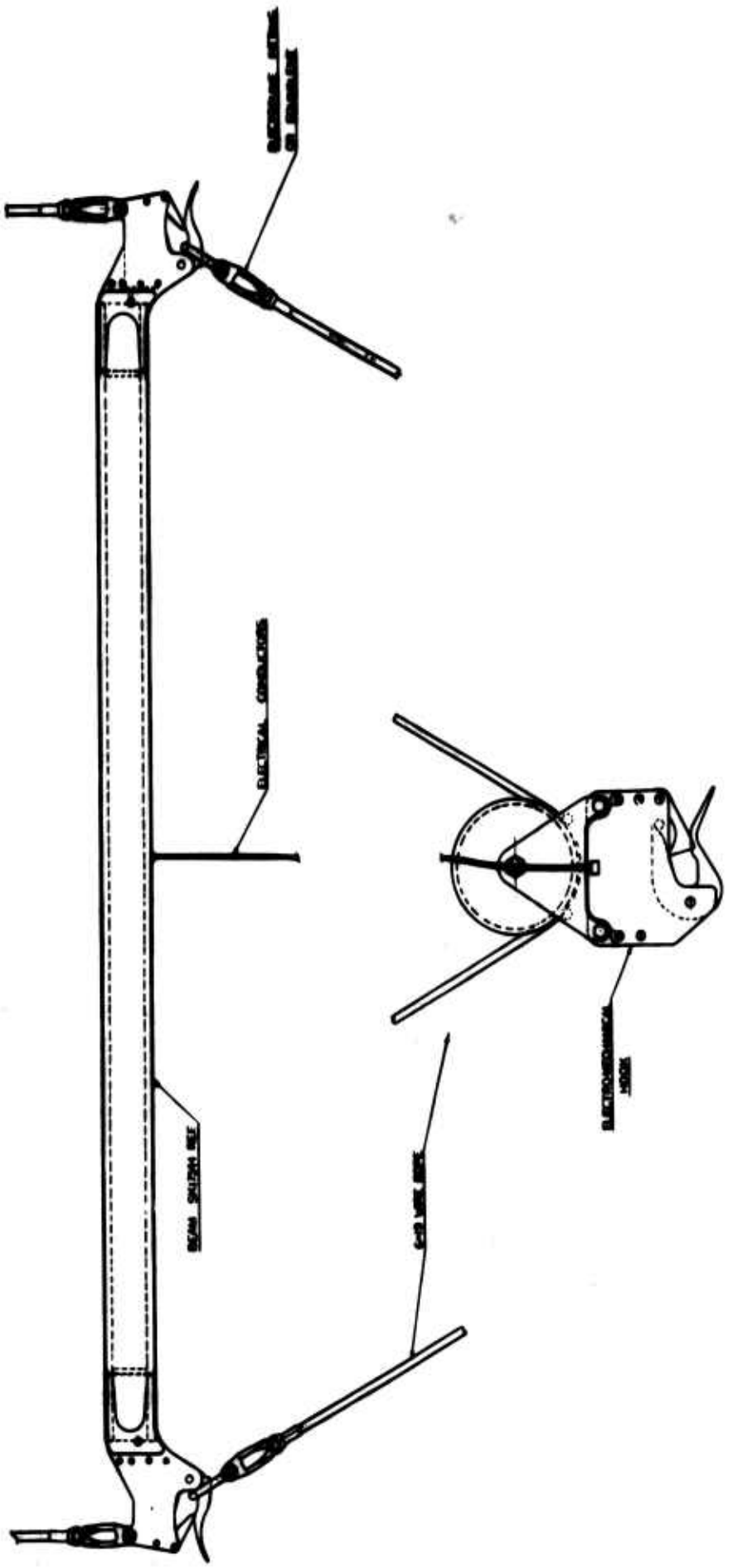
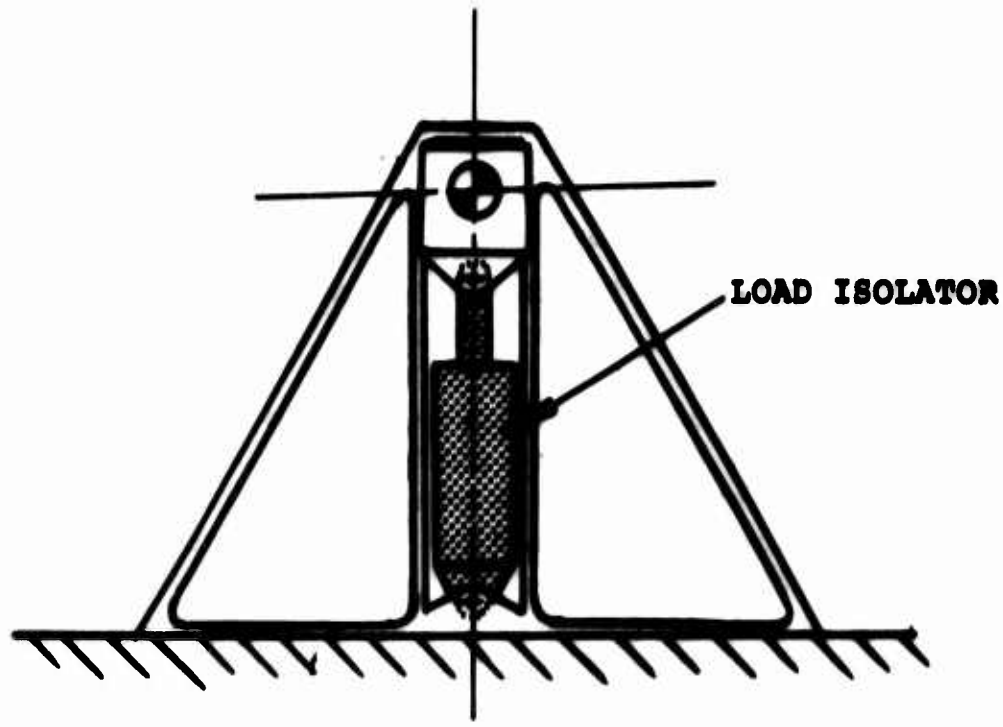
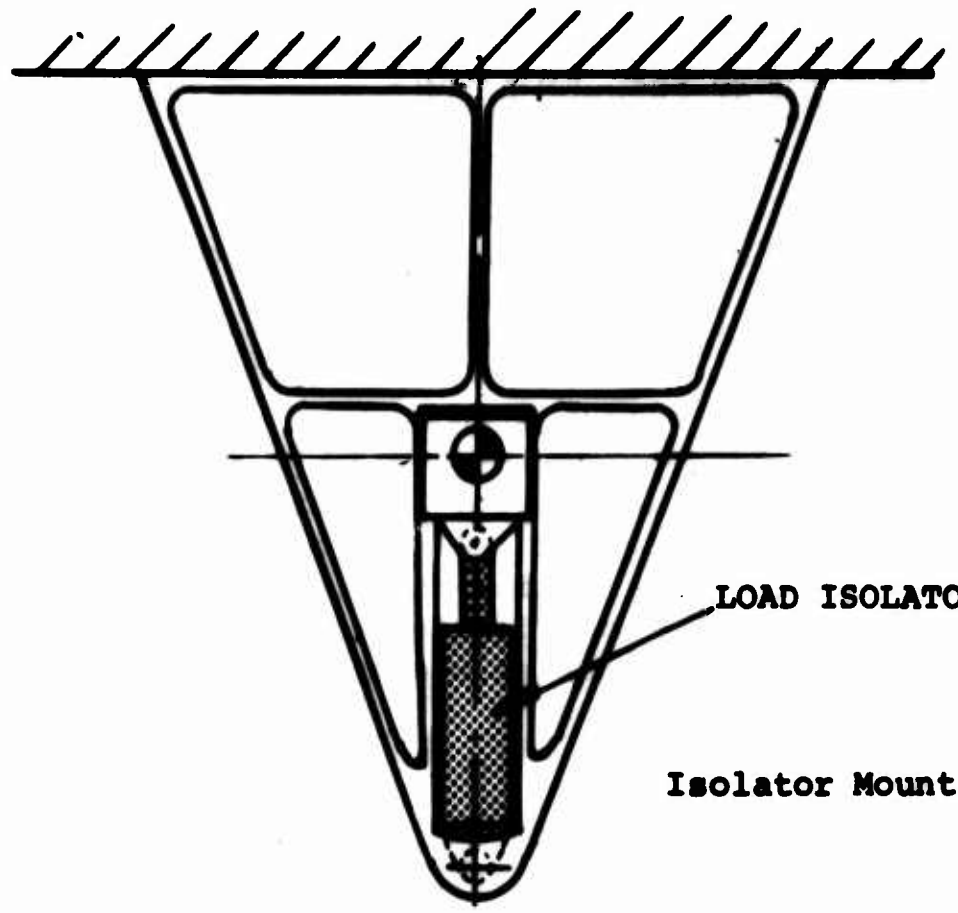


Figure 106. Single-Point Lift Mode, 20-Ton Hoist System.



Isolator Mount (Transport)



Isolator Mount (Crane)

Figure 107. Isolator Mounts.

to be the most efficient). The isolator, which in principle is the same as a landing gear strut, is considered to have a service life in excess of the specified life of the hoist system.

The load cell is installed in series with the load isolator. Its purpose is to display to the pilot and crane operator the actual load in each suspension rope.

The system component relationship, shown in Figure 105, assumes the prime mover to be transport configured. If the hoist system were to be installed in a crane-configured prime mover, the component relationship would be similar except that the winch supports would be rotated 180 degrees (i.e., the winches would hang from the aircraft structure). The result of such an arrangement would cause the load isolator and the load cell to become inoperative. To utilize the same load isolator/load cell design, the main winch support would require redesign such as is shown in Figure 107.

DESIGN AND DEVELOPMENT PROBLEM AREAS

General

No components or subcomponents called for in the preliminary design shown in this report are beyond today's state of the art. Recommended design practices in the application of wire rope have been rigidly adhered to. The drive system is of conventional design, and all bearings shown are of standard size and available as stock items. The specific source of the bearings is not indicated so as to permit the designer a choice during detail design of the system. The brake system has been designed around existing units; adaptation in some cases will require minor modification to off-the-shelf items, but in no instance will redesign become necessary.

Problem Areas

Although the recommended practices in the application of wire rope have been adhered to, the application of wire rope in capstan-type winches poses problems for which little qualitative data exists. The problems are as follows:

Rope Load Transfer to Drum

In a capstan-type winch, the wire rope is wound around two grocved driven drums. Load in the wire rope is transferred by friction into the driven drums. The use of a standard formula to determine the optimum amount of drum surface area contacted (by the rope) is dependent on the coefficient of friction of the rope and drum. Recognized standard methods of obtaining rope and drum coefficient of friction in this particular application are unreliable. Since this results in an unknown in the preliminary design shown (see Figure 97), the drums have been overdesigned; e.g., the maximum number of grooves that the available drum width will allow have been provided so that the number of wraps may be reduced during qualification testing of the winch if necessary. It should be noted that the least number of turns required to effect load transfer must be determined, since any "extra" turns, although they may be considered an additional assurance against rope slippage, reduce rope fatigue life (e.g., more bends).

2. Groove Form

The form of the groove is important since the wire rope, in transferring from drum to drum, must progressively advance one groove for each complete wrap around the drums. It has been reasoned that the side load on the rope (under load) induced by the rope fleet angle (drum to drum) causes the rope to climb out of the groove, assuming the groove to be semicircular as displayed in Figure 110. This induces more load into the rope between drums. To reduce this tendency of the rope to climb out of the groove, the groove form recommended and shown in the preliminary design (Figure 111) has a flat tread. Since no qualitative data exists to substantiate this, the design of the treads shown (Figure 97) allows tread replacement by treads having a different form without necessitating redesign of the drive drums. This feature also has the advantage of allowing replacement of the treads (due to wear) during the service life of the winch without the necessity of replacing the drums.

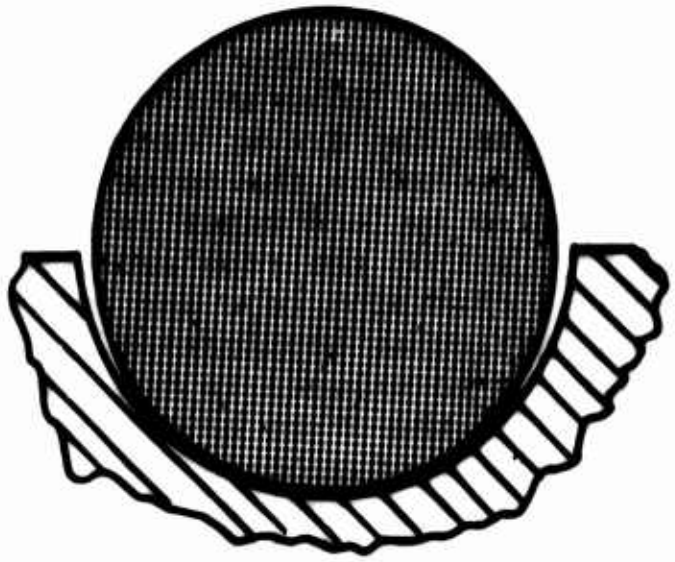


Figure 108. Semicircular Rope Groove.

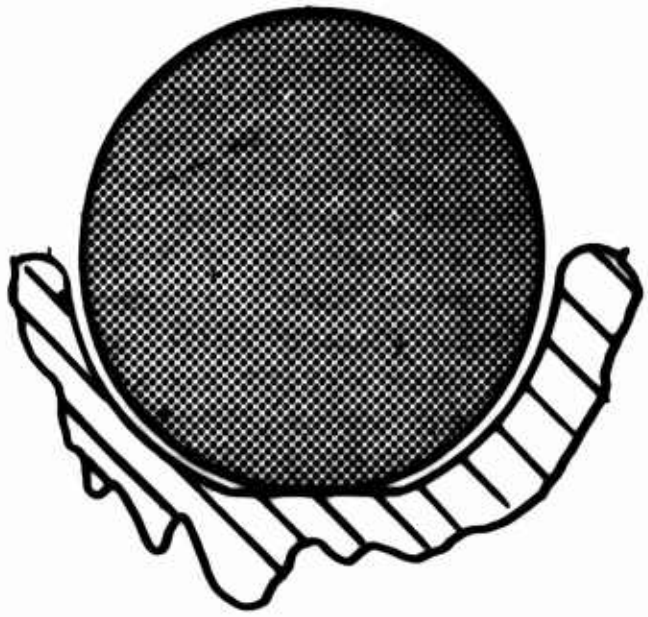


Figure 109. Recommended Rope Groove.

3. Storage Drum Capacity

The wire rope storage drum has been designed for a lift height of 150 feet. If, during subsequent evaluation of the hoist system, a greater lift height is required, the storage drum assembly could be increased in width to accommodate up to 300 feet of rope. This modification can be accomplished without any winch redesign other than that of the storage drum assembly. This feature will be of great value if the winch is used as a single-point lift system, that is, as a reeved rope system with a lift capacity of 50,000 pounds (see Figure 110). The use of the basic (25,000-pound) winch in a single-reeved system, while doubling the capacity, reduces the lift height to 75 feet at half the original lift rate (i.e., 30 instead of 60 feet per minute). The replacement of the storage drum assembly with one having a 300-foot rope capacity will restore the original 150-foot lift height. Conversely, a requirement for a lower lift height could be achieved by reducing the amount of rope on the existing drum.

4. Cargo Hook Release Unit

The cargo hook release unit (refer to Figure 116), consisting of three separate functional units, namely, torque motor, mechanical emergency release unit and inertia device, has been designed using established design principles. However, the integration of the three unit functions must be rigidly coordinated. Since the emergency release and inertia device functional criteria can be achieved in more than one way (for example, the emergency device could be operated by a spring motor or a cartridge generator device, and the inertia device could be a flywheel or centrifugal type), the precise combination required to achieve optimum integration of the three functions could justify an independent development program for this unit.

Specific Design Objectives

The design conforms to the following specific design objectives, in addition to the standard design requirements for aircraft-quality equipment:

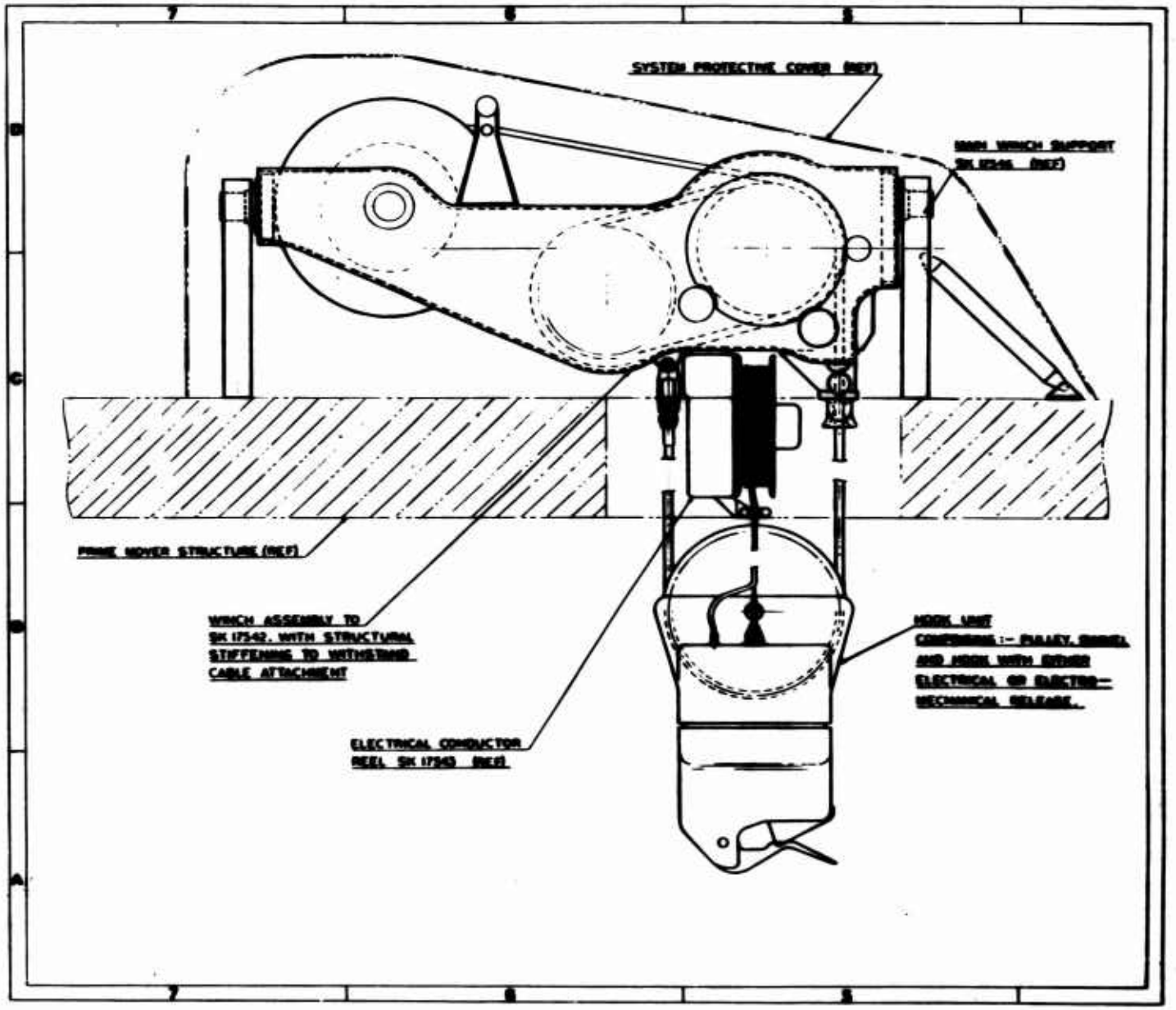


Figure 110. Single-Point Hoist System.

1. Power Requirements

Power input shall be by means of a hydraulic fixed displacement motor working with a 3000-psi hydraulic system and with rpm not to exceed 2500. Power developed shall be sufficient to obtain hoisting of a 25,000-pound load at a rate of 60 feet per minute.

2. Control Requirements

Only one control element, e.g., an electrohydraulic servo valve, shall be used for motor speed control (drum speed control), both in hoisting and in lowering, to facilitate winch synchronization in the two-point hoist system. In particular, no additional rate of load descent control, such as a separate brake control, shall be required.

3. Brake Torque Capacity

The brake torque capacity shall be such as to permit holding a 25,000-pound load under the following conditions:

a. 2.0g load factor in vertical direction.

b. No power input to drum drive.

Reference the paragraph on "Drum Drive and Brake Mechanism Calculations" on page 222.

4. Energy Absorption and Dissipation Capacity

The winch drive and brake system shall be of such self-contained energy absorption capacity as to be able to absorb heat energy generated on lowering the load of 25,000 pounds from a 150-foot height, at a rate of 60 feet per minute. The capability of the system to dissipate the energy shall be such as to permit the repetition of the work cycle after one half-hour interval. No reliance shall be made on dynamic or regenerating braking using aircraft hydraulic or electric system, in order to have the self-contained energy absorption capability of the drive and brake system not dependent on any particular aircraft installation. Reference "Energy Absorption and Dissipation

Capacity of Winch Drive and Brake System Calculation" on page 227.

5. Mechanism Housing

The entire mechanism, except for the hydraulic motor, shall be enclosed in a sealed housing and submerged in an oil bath.

6. Design Criteria

The criteria used in the winch system design shall be in accordance with procurement specification D8-0692. (See page 376)

DESIGN CALCULATIONS

Drum Drive and Brake Mechanism Calculations

Given:

Load:	25,000 lb
Hoisting Rate:	1 ft per sec (60 ft per min)

The required net power is:

$$\frac{25{,}000 \times 1}{550} = 45.5 \text{ hp}$$

Selected winch drum effective diameter: 19 inches

Required drum speed to obtain given hoisting rate is:

$$\pi \times \frac{19}{12} \times N = 60 \text{ feet per minute}$$

$$N = \frac{60 \times 12}{\pi \times 19} = 12.05 \text{ rpm}$$

Selected gearing ratios:

Initial (Bevel) Gears $\frac{1}{2.2}$

Intermediate (Spur) Gears $\frac{1}{1}$

Planetary:

First Stage: $\frac{1}{6.23}$

Second Stage: $\frac{1}{3.83}$

Third Stage: $\frac{1}{3.11}$

Total Planetary Ratio: $\frac{1}{74.2}$

Total Drive Mechanism Reduction Ratio: $\frac{1}{2.2 \times 1 \times 74.2} = \frac{1}{163}$

Required Hydraulic Motor Speed: $12.05 \times 163 = 1970$ rpm

Required Hydraulic Motor Torque: $\frac{25,000 \times 9.5}{163 \times .935} = 1560$ lb/in.

Where: .935 Drive Mechanism Overall Efficiency Coefficient:

(.99 x .99 x .985 x .985 x .985 = .935,)

.99 = one pair of gears efficiency factor

.985 = one stage planetary gearing efficiency factor

9.5 = load moment arm, inches

Select hydraulic motor having the following characteristics:

Displacement (theoretical), cubic inch/rev	3.671
Theoretical torque at 3000 psi operation pressure, lb/in.	1752.5

Maximum operating pressure for continuous duty, psi	3000
Maximum speed recommended for continuous duty, rpm	2080
Volumetric efficiency at maximum operating speed and pressure, pct	96

Example of available motor:

Vickers Incorporated MF81-3921-30YJ Motor Fixed Displacement.

Hydraulic motor theoretical power:

$$hp = \frac{T \times N}{63{,}025}$$

$$hp = \frac{1752.5 \times 2080}{63{,}025} = 58.0$$

Assume power coefficient = .868

(.96 x .905 = .868)

Where .96 is volumetric efficiency factor and

.905 is assumed mechanical efficiency factor

Estimated available hydraulic motor power:

58 x .868 = 50.3 hp

Estimated required power:

$$\frac{45.5}{.935} = 48.7 \text{ hp}$$

.935 = drive mechanism efficiency coefficient

50.3 - 48.7 = 1.6 hp

This is power available to cover other losses such as cable bending on drums. It amounts to:

$$\frac{1.6}{45.5} \times 100 = 3.5\% \text{ of net lifting power required}$$

Hydraulic motor flow required:

$$Q = \frac{D \times N}{231 \times .96} = \frac{3.671 \times 2080}{231 \times .96} = 34.4 \text{ GPM}$$

Input Clutch and Brake Calculations

Hydraulic motor torque is transmitted to the drum drive shaft through the input clutch and brake. The input clutch comprises four rotor discs keyed to the drive shaft nut and four intermeshing discs keyed to the clutch housing, which, in turn, provides torque input (via input bolts) to the drive shaft and simultaneously provides an axial force to load the brake assembly.

The disc contact surface lining has: 8.00 in. OD and
6.80 in. ID

There are seven working contact surfaces.

Clutch effective mean radius is:

$$R_M = \frac{1}{3}\left(OD + ID - \frac{OD \times ID}{OD + ID}\right)$$

$$= \frac{1}{3}\left(8.00 + 6.80 - \frac{8.00 \times 6.80}{8.00 + 6.80}\right) = 3.71 \text{ in.}$$

Load torque on drive shaft is:

$$T = \frac{25{,}000 \times 9.5}{74.2} = 3{,}200 \text{ lb/in.}$$

and the maximum input speed is:

$$\frac{2080}{2.2} = 945 \text{ rpm}$$

Required axial force on clutch to transmit the above torque is:

$$F = \frac{T}{f \times R_M \times 7}$$

$$= \frac{3,200}{.08 \times 3.71 \times 7} = 1,545 \text{ lb}$$

f = .08 is the friction coefficient for clutch discs submerged in oil.

There are 7 friction surfaces.

Note: Although the input torque is transmitted to the drive shaft through the input clutch only, and not jointly through clutch and brake, this is so only so far as the rotary motion and torque are concerned. In the axial direction, the functions of clutch and brake are intermingled such that the clutch cannot be axially loaded (to develop the torque) before the brake is loaded, and thus the axial reacting force is created to squeeze the clutch discs.

The brake assembly comprises five discs keyed to the drive shaft and five intermeshing discs keyed to the brake housing. The disc lining has:

9.31 in. OD and 7.91 in. ID

There are 10 working surfaces.

The effective mean radius is:

$$R_M = \frac{1}{3}\left(9.31 + 7.91 - \frac{9.31 \times 7.91}{9.31 + 7.91}\right)$$

$$= 4.32 \text{ in.}$$

The brake axial load required to develop braking torque sufficient to balance (via gearbox) the load torque applied to the winch drum, under steady load conditions, is:

$$N = \frac{T}{f \times R_M \times 10} = \frac{3,200}{.08 \times 4.32 \times 10}$$

$$= 930 \text{ lb}$$

Where f = 0.08 is the disc friction coefficient for discs in oil, and 10 is the number of friction surfaces.

Due to the incorporation of the self-energizing balls and wedge-shaped grooves into the brake design, the actual required input force to load the brakes, such as to react 3,200 pounds in torque, can be less than 930 pounds.

If the effective ball ramp angle $\alpha = 17°$, ($\tan_\alpha = .3$), then the required input force to generate 3,200 pounds in brake torque is:

$$F = \frac{T\ (\tan\alpha - f)}{f \times R \times 10 \times \tan\alpha}$$

$$= \frac{3,200\ (.3 - .08)}{.08 \times 4.32 \times 10 \times .3} = 680 \text{ lb}$$

However, it is considered here that the brake torque capacity should be more than the 3,200 pounds in torque (required under steady load condition) and should be capable of reacting temporary load torque increases, such as in the 2g vertical load condition.

The corresponding numerical values are:

$N' = 2N = 1,860$ lb and

$F' = 2F = 1,360$ lb

It is also considered that the input clutch energizing force should not be less than the value of $F' = 1,360$ pounds.

Its present design value is 1,545 pounds.

<u>Energy Absorption and Dissipation Capacity of Winch Drive and Brake System Calculations</u>

The 25,000-pound load, when lifted to a height of 150 feet, has a potential energy of:

25,000 x 150 = 3,750,000 ft-lb

On lowering the weight, this energy will be absorbed by the brake and transformed into heat energy.

To lower the load, it is necessary to have an input rotary motion from the hydraulic motor. The work produced by the motor will also be absorbed by the brake.

Assume that the motor input work to lower the load amounts to 10% of the work required to lift the load; then total energy to be absorbed is:

$$3{,}750{,}000 + 375{,}000 = 4{,}125{,}000 \text{ ft-lb}$$

The corresponding heat energy is:

$$H = \frac{4{,}125{,}000}{778} = 5{,}310 \text{ BTU.}$$

The above heat energy is generated on brake discs; however, as the entire system is submerged in oil, it will also be transferred to the drive mechanism, drive housing, drum gearbox, drum itself, drum base plates, and winch beam assemblies.

In total, the heat will flow to the following three groups of masses:

Steel Components: 348 lb, Specific heat $C_{ST} = .112$

Light Alloy Components: 304 lb, Specific heat $C_{LA} = .230$

Oil: 83 lb, Specific heat $C_O = .40$

Assuming uniform temperature of the entire system, the temperature increase following the release of the heat energy into the system is calculated from the equation:

$$t_2 - t_1 = \frac{H}{M_{ST}C_{ST} + M_{LA}C_{LA} + M_OC_O}$$

$$= \frac{5{,}310}{348 \times .112 + 304 \times .230 + 83 \times .40}$$

$$= 37.3°F$$

Assuming now that the oil temperature safe limit is 240°F and that the work cycles follow each other without interval (no time for heat dissipation), then the number of cycles possible, prior to reaching the temperature limit, is for different

initial temperatures as follows:

Initial Temp. °F	0	75	110	160
Number of Cycles	6	4	3	2
Final Temp. °F	223	224	222	233

The data listed above correspond to the conditions where the maximum load of 25,000 pounds was lifted to the maximum hoisting height of 150 feet. However, it has to be noted that the most frequently used hoisting height will not be 150 feet, but rather of the order of 60 feet. The potential energy would then be:

$$25{,}000 \times 60 = 1{,}500{,}000 \text{ ft} - \text{lb}$$

and the total energy released in brakes:

$$1{,}500{,}000 + 150{,}000 = 1{,}650{,}000 \text{ ft} - \text{lb}$$

Where 150,000 is the energy contributed by the hydraulic motor (10% assumed), the corresponding heat energy is:

$$\frac{1{,}650{,}000}{778} = 2{,}120 \text{ BTU.}$$

and the system temperature increase per one work cycle is:

$$37.3 \times \frac{2{,}120}{5{,}310} = 14.9, \text{ say, } 15°\text{F}$$

The number of possible work cycles prior to reaching the limit temperature is now in accordance with the data below:

Initial Temp. °F	0	75	110	160
Number of Cycles	16	11	8	5
Final Temp, °F	240	240	230	235

<u>Heat Dissipation</u>

The heat dissipation study which follows should be treated only as approximate, due to simplified assumptions made in it.

Heat Dissipated Through Radiation

$$Q_r = .174\ AE\ \left[\left(\frac{T_2}{100}\right)^4 - \left(\frac{T_1}{100}\right)^4\right] \quad \text{BTU per hour}$$

Where:

A = Surface of emissivity, sq ft

E = Emissivity factor (assumed = .67)

T_2 = Absolute temperature at the emissive surface

T_1 = Absolute temperature of the surrounding objects

Assuming the temperature of the surrounding objects to be 75°F, so that:

$T_1 = 75 + 460 = 535°$ Abs.,

and the system temperature to be 240° F, so that,

$T_2 = 240 + 460 = 700°$ Abs.

and having A = 35.1 sq ft,

$$Q_r = .174 \times 35.1 \times .67 \left[\left(\frac{700}{100}\right)^4 - \left(\frac{535}{100}\right)^4\right]$$

$$= 6{,}480 \text{ BTU per hour}$$

Heat Dissipated Through Convection

$$Q_C = .22\ (t_2 - t_1)^{1.33}\ A \text{ BTU per hour}$$

Assuming temperature conditions as above, so that

$t_1 = 75°F$ and

$t_2 = 240°F$

and having A = 35.1 sq ft

$$Q_C = .22 \times 165^{1.33} \times 35.1 = 6{,}850 \text{ BTU per hour}$$

Total Heat Dissipation

$$Q_t = Q_r + Q_C = 6{,}480 + 6{,}850$$

$$= 13{,}330 \text{ BTU per hour}$$

In the same way, the rate of heat dissipation can be calculated for the condition where the system temperature is $t_2 = 240 - 37.3 = 202.7$, say, 203°F, and the temperature of the surrounding objects is $t_1 = 75$°F, as before. 37.3°F is the temperature increase per one work cycle for maximum hoist height of 150 feet.

$$Q_t = Q_r + Q_c = 4,590 + 4,890$$

$$= 9,480 \text{ BTU per hour}$$

$$Q_{av} = \frac{13,300 + 9,480}{2} = 11,405 \text{ BTU per hour}$$

is the average rate of heat dissipation for the temperature range of 240°F to 203°F.

The time required to cool the system from 240°F to 203°F, so that another work cycle could be performed, is:

$$\frac{5,310}{11,405} = .467 \text{ hr} \quad \text{(or 28 minutes)}$$

At 75°F system initial temperature and also surrounding objects temperature, the winch system is capable of performing four consecutive work cycles, with 150 feet as hoisting height, and with no intervals for cooling off. A period of the order of a half-hour would be required to cool the system sufficiently to perform each additional cycle.

Similarly, eleven reduced-height (60 feet) work cycles are possible without cooling-off intervals; then an interval of the order of 10 minutes would be required prior to each additional 60-foot-height cycle.

If the system is to be able to perform an unlimited number of work cycles to 150 feet hoisting height without cooling-off intervals, then the rate of heat removal from the system should be such as to be not less than the rate of heat generation on lowering the load.

Considering that the time required to lower the load from a height of 150 feet at the rate of 60 feet per minute is 2.5 minutes, the rate of heat generation is:

$$\frac{5,310}{2.5} = 2,125 \text{ BTU per min}$$

One way to obtain the required system cooling rate is to recirculate the system oil through a heat exchanger (e.g., fan cooled).

Assuming a 100°F oil temperature drop in the oil cooler, the required rate of oil flow is calculated as follows:

$$MC_O\ (t_2 - t_1) = 2{,}125 \text{ BTU per min},$$

where M is mass flow in lb per min and C_O is the specific heat of oil.

$$M = \frac{2{,}125}{.4 \times 100} = 53.2 \text{ lb per min}$$

This is equivalent to the volumetric flow of:

$$Q = \frac{53.2}{7.5} = 7.1 \text{ GPM}$$

Storage Drum Slip Clutch Calculations

The clutch contains two rotors and thus four working surfaces of the following dimensions:

7.875 in. OD and

3.375 in. ID

Effective Mean Radius:

$$R_M = \frac{1}{3}\left[OD + ID - \frac{OD \times ID}{OD + ID}\right]$$

$$= \frac{1}{3}\left(7.875 + 3.375 - \frac{7.875 \times 3.375}{7.875 + 3.375}\right)$$

$$= 2.96 \text{ in.}$$

Twelve pairs of concentric springs provide the disc loading force.

Outer Spring:

D = .812 in. mean dia

d = .112 in. wire dia

Inner Spring:

D = .512 in. mean dia

d = .112 in. wire dia

Outer Spring Force

$$P = \frac{\pi S d^3}{8D}$$

$$P = \frac{\pi \times 110{,}000 \times (.112)^3}{8 \times .812} = 74.5 \text{ lb}$$

Inner Spring Force:

$$P = \frac{\pi \times 110{,}000 \times (.112)^3}{8 \times .512} = 119 \text{ lb}$$

Spring Pair Force = 74.5 + 119 = 193.5 lb

The clutch is subjected to an adjustable load of a value up to:

12 x 193.5 = 2,310 lb

Clutch Torque Available:

$T = N \times f \times R_M \times 4$

f = .25 (Friction Coefficient)

T = 2,310 x .25 x 2.96 x 4

= 6837 lb - in.

Pull on rope in inner layer of radius = 9.5 in.

$$P = \frac{6837}{9.5} = 720 \text{ lb}$$

For outer layer of radius, 14.5 in.,

$$P = \frac{6837}{14.5} = 472 \text{ lb}$$

TABLE X
ENERGY ABSORBED BY CLUTCH IN ONE WORK CYCLE

Rope Layer Radius, In.	Layer Circum-ference, In.	Length of Rope in Layer, In.	Rope Load, Lb	Rope Travel Corresponding to Clutch Slip, In.	Work Increment, Ft-Lb
9.5	59.6	298	720	0	0
10.5	65.9	330	652	32	1755
11.5	72.1	361	595	63	3130
12.5	78.1	392	548	94	4260
13.5	84.9	424	506	126	5320
14.5	91.2	456	472	158	6200

Sum of the work increments is 20,665 ft - lb, and this represents the work produced on either load lifting or lowering.

Work for one work cycle is 2 x 20,665 = 41,330 ft-lb

$$\frac{41,330}{778} = 53.0 \text{ BTU}$$

Disc mass is 3.40 lb,

Specific Heat: $C_{ST} = .112$

Temperature rise in discs per one work cycle is:

$$t_1 - t_2 = \frac{H}{MC_{ST}} = \frac{53.0}{3.40 \times .112} = 139°F$$

TWENTY-TON HOIST SYSTEM - CONTROL DESCRIPTION (ELECTRICAL)

(Refer to Figure 111)

General

The basic control voltage of the 20-ton hoist system is 28 VDC nominal from the DC primary bus. The release unit torque motor is operated on a 400-cps, 3-phase, 115/200-VAC voltage, from a No. 1 primary AC bus. The control schematic indicates that the pilot and/or crane operator has complete control of the hoist system, as well as provision for an external "plug-in" source to enable hoist system control from outside the aircraft when the aircraft is on the ground. It is pointed out that giving the pilot full control of the hoist system could create problems when hoisting loads from high hover, since monitoring by ground crew will be ineffective. Such operations dictate that synchronization of the winches be mandatory, with effective limit stops to automatically stop the hoist system at a predetermined distance under the aircraft.

Circuit functions are described in Table XI;functions and control nomenclature and location are included.

Other associated auxiliary functions are shown in the illustrations listed below:

Figure 112 shows the strain gauge bridge and servo indicator-type of rope tension indicator system.

Figure 113 shows the voltage-regulated, multi-turn type of a rope footage indicator.

Figure 114 shows the hook arming and release function for pilot and loadmaster.

Figure 115 illustrates the rope cutter system.

TABLE XI CIRCUIT FUNCTIONS			
NOMENCLATURE	**LOCATION**		**CONTROL FUNCTION**
Pilot Arming Switch	Cockpit Overhead Panel	OFF	Power off complete system. Release unit brake "on", torque motor de-energized, winches inoperative, brakes "on".
		PILOT	Relay K4 energized, release unit brake "off", torque motor energized, crane operator's controls de-energized, pilot's winch control armed, relay K1, K5 remain de-energized, assigning control of both winches to pilot.
		CRANE OP.	Relay K4 energized, release unit brake "off", torque motor energized. "Ground operation transfer switch" armed, "Crane operator arming trigger" armed. Relay K1 and K5 energized, assigning control of both winches to crane operator (contacts K1A, K1B, K1C, K1D, K5A, K5B).
Pilot Winch Control Lever	Cockpit Center Console		See Item 1 for arming conditions. Lever spring loaded to neutral, horizontal position. Pilot or copilot depresses lever for both winches down, raises lever for both winches up. Limit switch closes as lever moves out of neutral, either direction, releasing brakes. Progressive

TABLE XI Continued

NOMENCLATURE	LOCATION		CONTROL FUNCTION
			movement of lever increases voltage on winch control units, increasing winch speed. Decrease degree of lever movement to decrease speed, back to neutral, where voltage at winch control units = OV and limit switch opens, braking winch.
Crane Operator Arming Trigger	On Crane Operator's Grip	OFF	Winch control circuits in crane operator's grip de-energized.
		ON	Trigger closed by natural force of pilot grasping the grip. Winch control circuits in crane operator's grip energized.
Crane Operator's Grip	In Crane Operator's Capsule		A single, thumb control lever on hand contoured grip. Lever controls No. 1 (fwd) and No. 2 (aft) winch simultaneously, with thumb up, thumb down motion spring, loaded to center neutral. Also has thumb left, thumb right motion with two detents each direction. Thumb up - winch up, thumb down, winch down, winch speed increasing with thumb displacement. Otherwise similar in operation to item 2. Moving lever left to first detent slows No. 2 winch (aft) to 20% less than No. 1 winch (fwd) and vice versa, for in-motion trim.

TABLE XI Continued

NOMENCLATURE	LOCATION	CONTROL FUNCTION	
			Moving lever left to second detent stops No. 2 winch for final trimming, and vice versa.
Ground Operation Transfer Switch	In Crane Operator's Capsule	CRANE OPERATOR	Ground operator's controls de-energized. Relays K3, K2 de-energized, assigns control of No. 1 and No. 2 winch to crane operator (contacts K2A, K2B, K2C, K2D, K3A, K3C).
		GROUND OPERATOR	Ground operator winch control grip energized, relays K2, K3 energized, assigns control of No. 1 and No. 2 winch to ground operator.
Reel-In and Reel-Out Limit Switches			Remove power from winch control unit when pre-determined travel limit of winches is reached. (Load limit switch removes power from winch control unit when allowable load is exceeded.)
Reel-In Range Select Switch			Bypasses partial reel-in limit switch to allow load to be winched up beyond partial limit set point to full reel-in position.

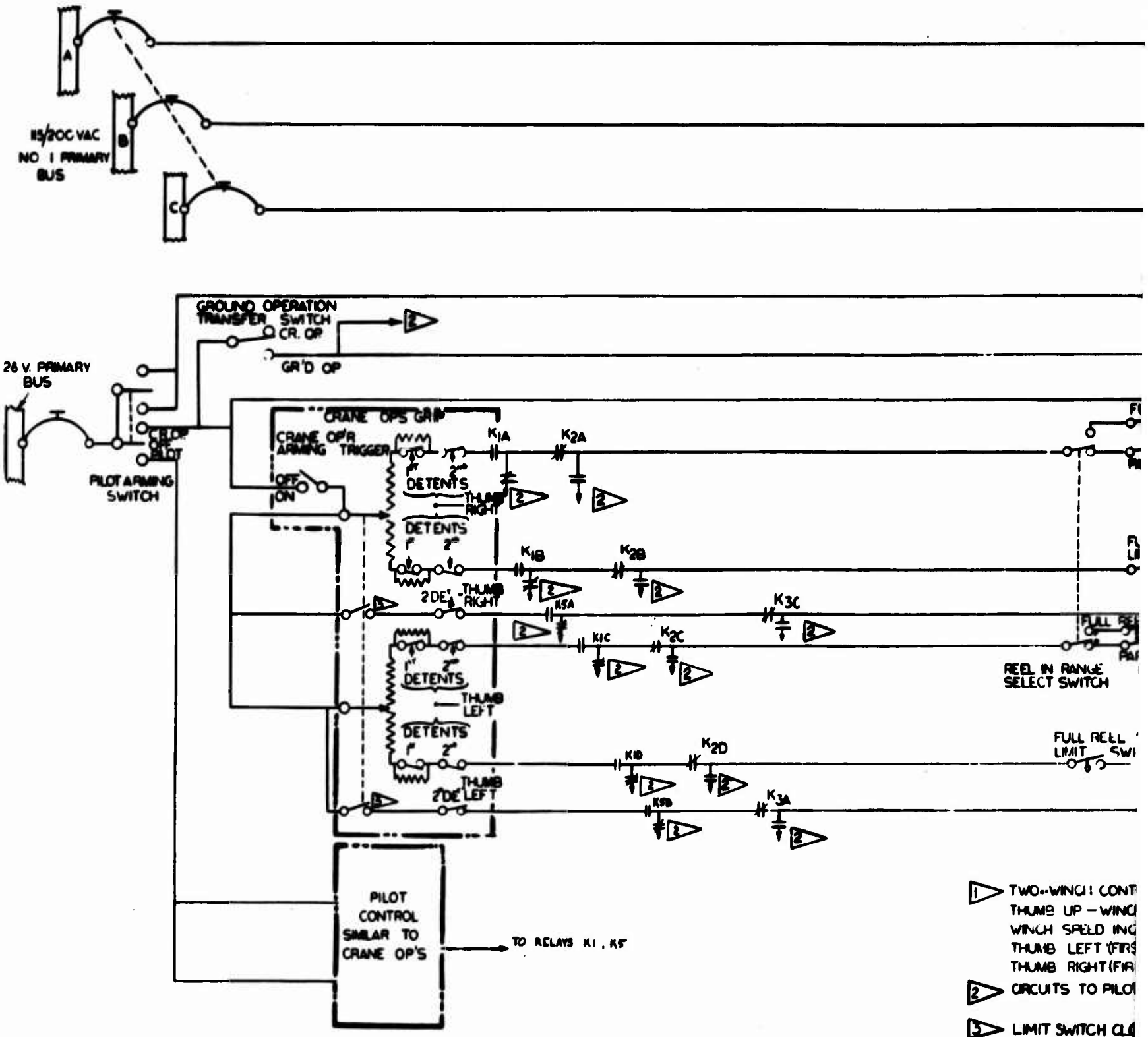
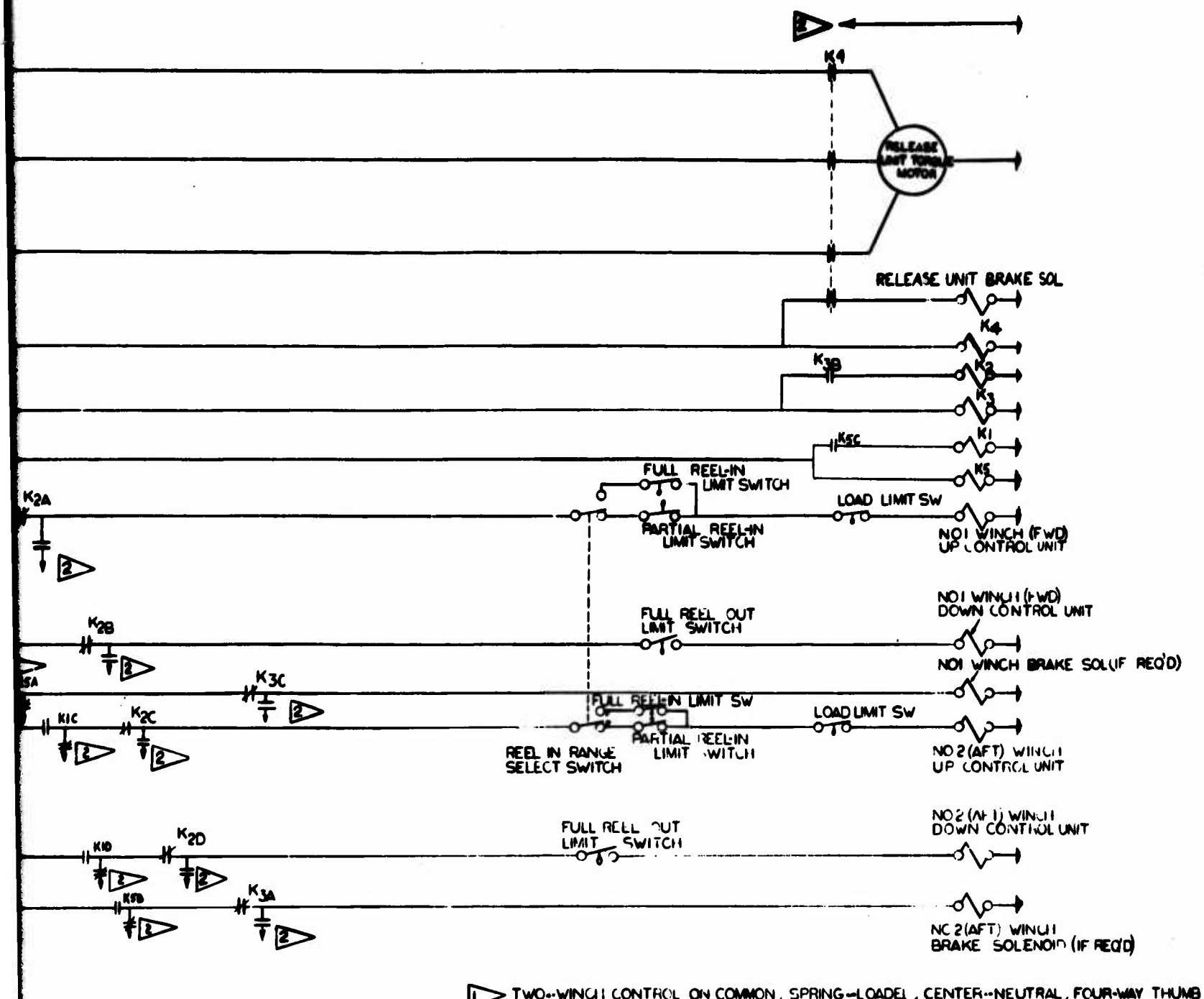


Figure 111. Electrical Circuits, 20-Ton Hoist System.



1. TWO-WINCH CONTROL ON COMMON, SPRING-LOADED, CENTER-NEUTRAL, FOUR-WAY THUMB LEVER.
   THUMB UP - WINCH UP THUMB DOWN - WINCH DOWN
   WINCH SPEED INCREASES WITH DISPLACEMENT
   THUMB LEFT (FIRST DETENT) NO.2 WINCH SPEED DECREASES 20% SECOND DETENT #2 WINCH STOPS.
   THUMB RIGHT (FIRST DETENT) NO.1 WINCH SPEED DECREASES 20% SECOND DETENT #1 WINCH STOPS.
2. CIRCUITS TO PILOT AND GROUND OPERATORS STATIONS.
3. LIMIT SWITCH CLOSES WHEN SPEED CONTROL MOVES OUT OF CENTER POSITION

WS K1, K5

its, 20-Ton Hoist System.

2

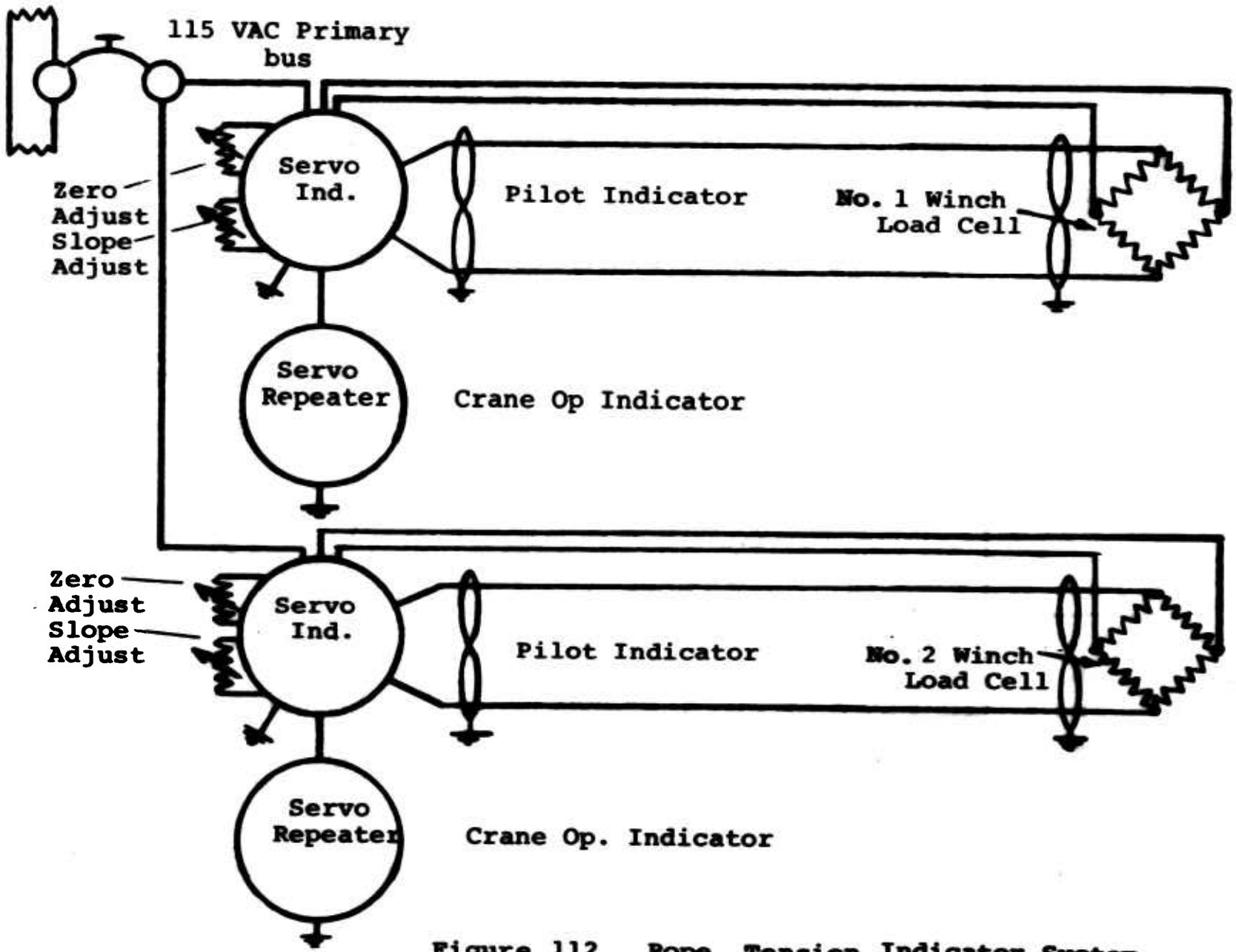


Figure 112. Rope Tension Indicator System.

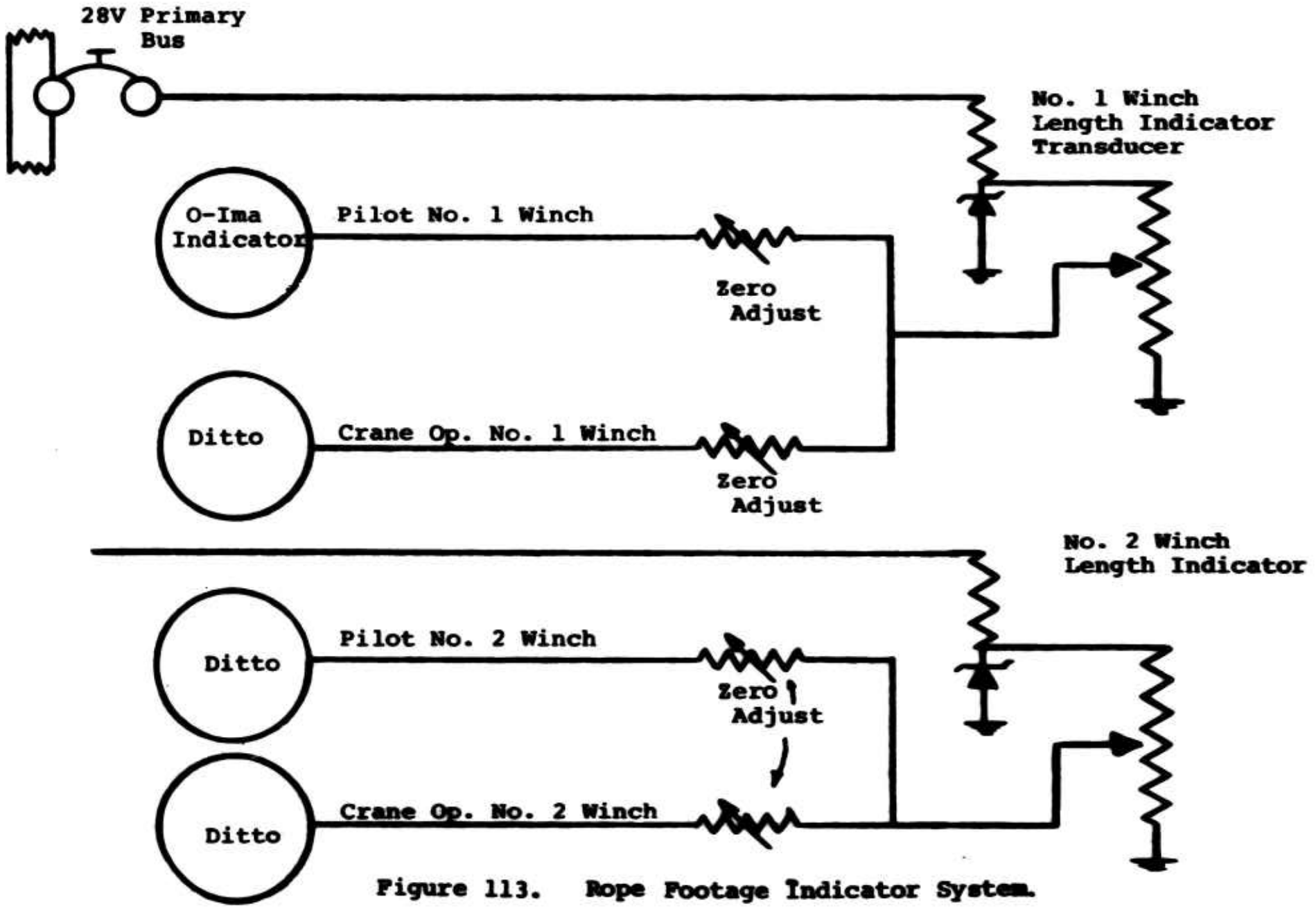


Figure 113. Rope Footage Indicator System.

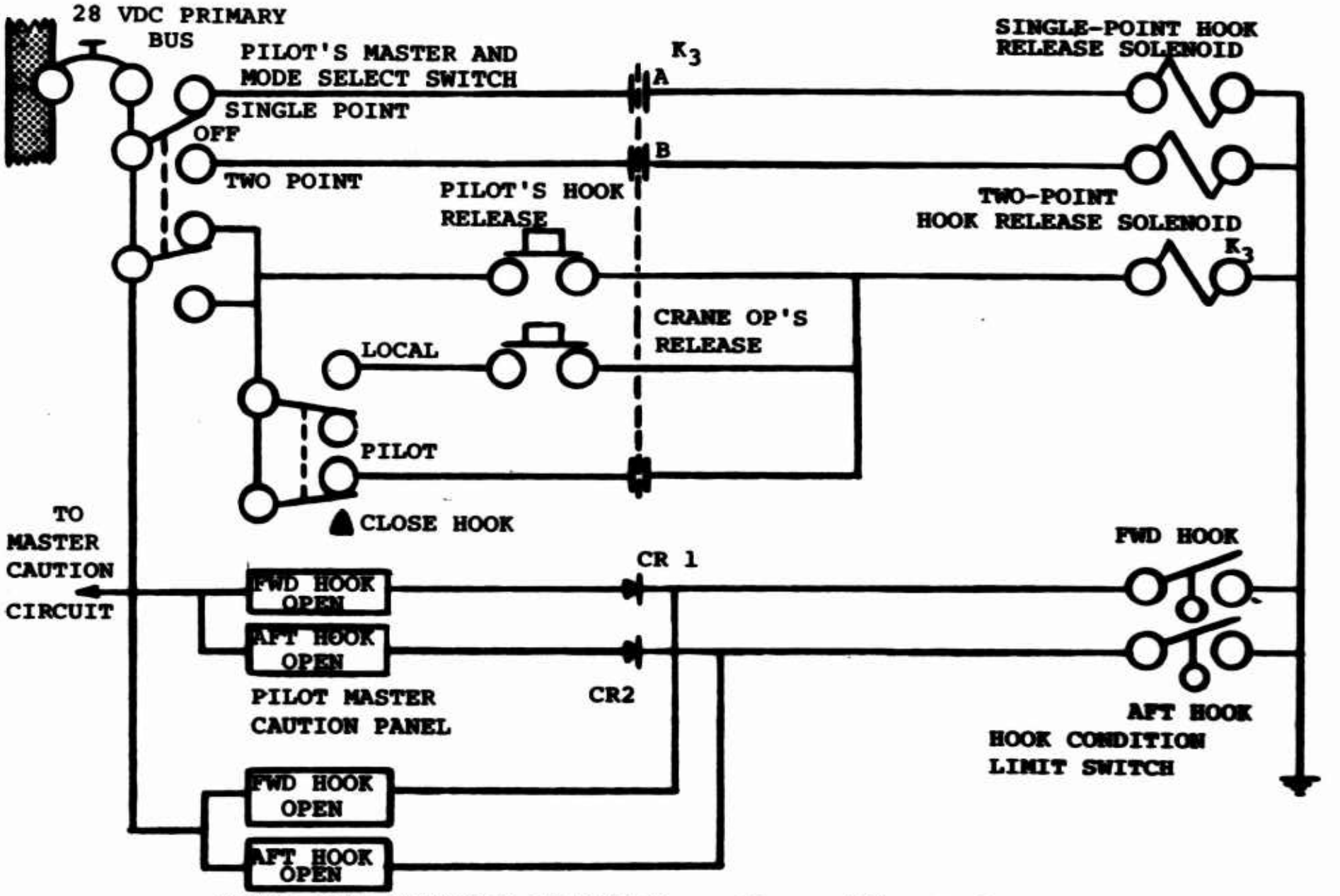


Figure 114. Hook Release System.

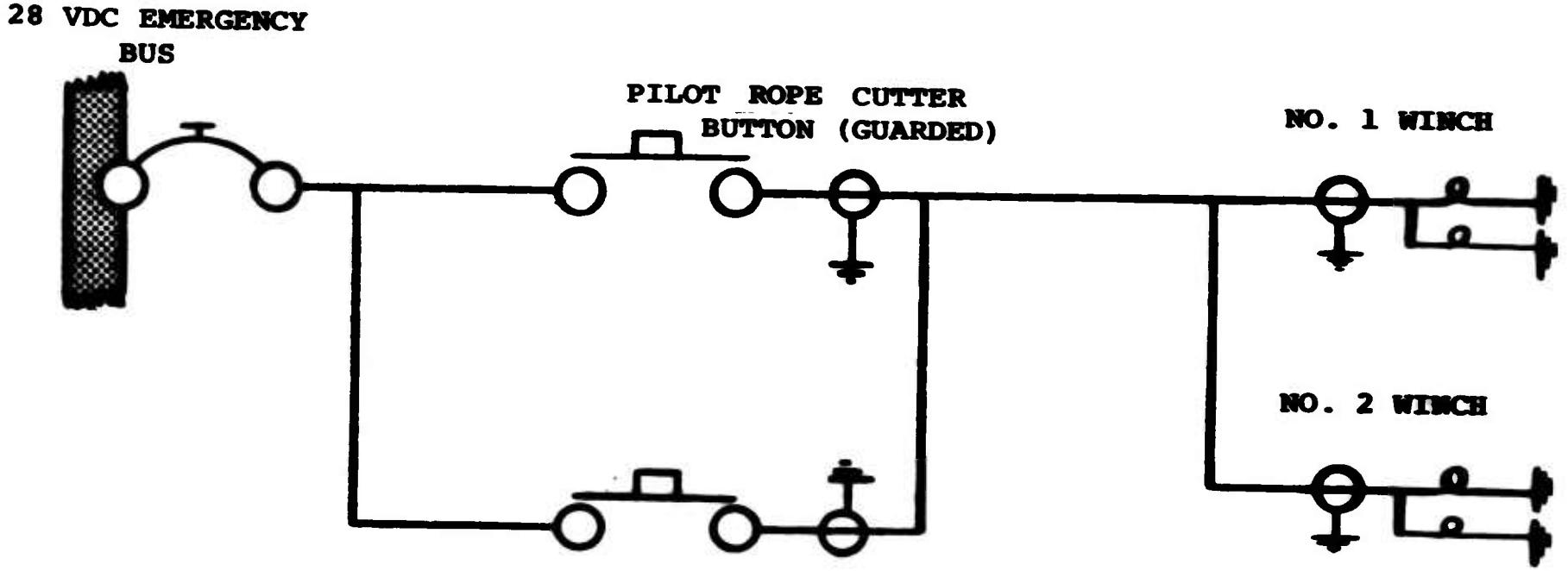


Figure 115. Rope Cutter System, 20-Ton Hoist System.

LOAD RELEASE SYSTEM

The two cargo hooks are connected to a common lever system at the center of the beam. In turn, this system is coupled to an electromechanical actuator. Initiation of the actuator by an electrical signal from the pilot or the crane operator results in a tensile force on the linkages to the cargo hooks. Since these are connected to a common lever, simultaneous release of the hooks is assured. The actuator automatically returns to its original position after releasing the cargo hooks. (Refer to Figure 116.)

In the event of electrical failure, provision is made for the manual operation of the release linkage and the subsequent simultaneous release of the cargo hooks.

Since the beam assembly rises and falls with the hoist system, provision must be made to interconnect the beam assembly and the prime mover both electrically and mechanically. This is accomplished by an electromechanical rope, which consists of a center core of stainless-steel flexible rope surrounded by electrical conductors. The resulting assembly is covered by a protective sheath. To facilitate automatic reel-in/out of the rope as the beam moves, the rope is stored on a drum which is driven by an electrical torque motor, the torque motor being activated automatically while the hoist system is being operated. To prevent the self-unreeling of the rope due to its own weight when the torque motor is deactivated, a brake is incorporated in the torque motor. Electrical continuity is achieved via a slip ring assembly in the drum system. This electrical system satisfies the normal release mode of the system. Emergency mode, i.e., manual release of load by pilot and/or crane operator, is accomplished by a pilot- and/or crane operator-initiated mechanical motor mounted in parallel with the drum torque motor. In principle, the emergency release motor, upon initiation, first positively engages the drum and then induces rotation which results in a tensile pull in the electromechanical rope. This pull is sufficient to overcome a safety device in the beam assembly, permitting simultaneous release of the cargo hooks. After use, the emergency system motor must be re-energized. The motor has the capability to overcome the torque motor brake in the event that the emergency release is used with the hoist power off. In order to accomplish automatic release of the load in the event of a suspension rope failure, the release assembly is fitted with an inertia lock-out device. A suspension rope

break will cause one end of the beam to fall, which in falling would create a reel-out acceleration in the electromechanical rope. The inertia device, responding to this acceleration, will lock out the drum and the resulting tension felt by the rope will operate the emergency release system, which results in the load being automatically dropped.

An additional flight safety emergency mode is provided by dual cartridge-type wire rope cutters installed on both winches at the rope exit bellmouth. The cutters will be so designed that, having cut the wire ropes, the free ends will be retained, eliminating wild ropes.

Criteria used in the design of the beam assembly and release unit shall be in accordance with Procurement Specification D8-0691.

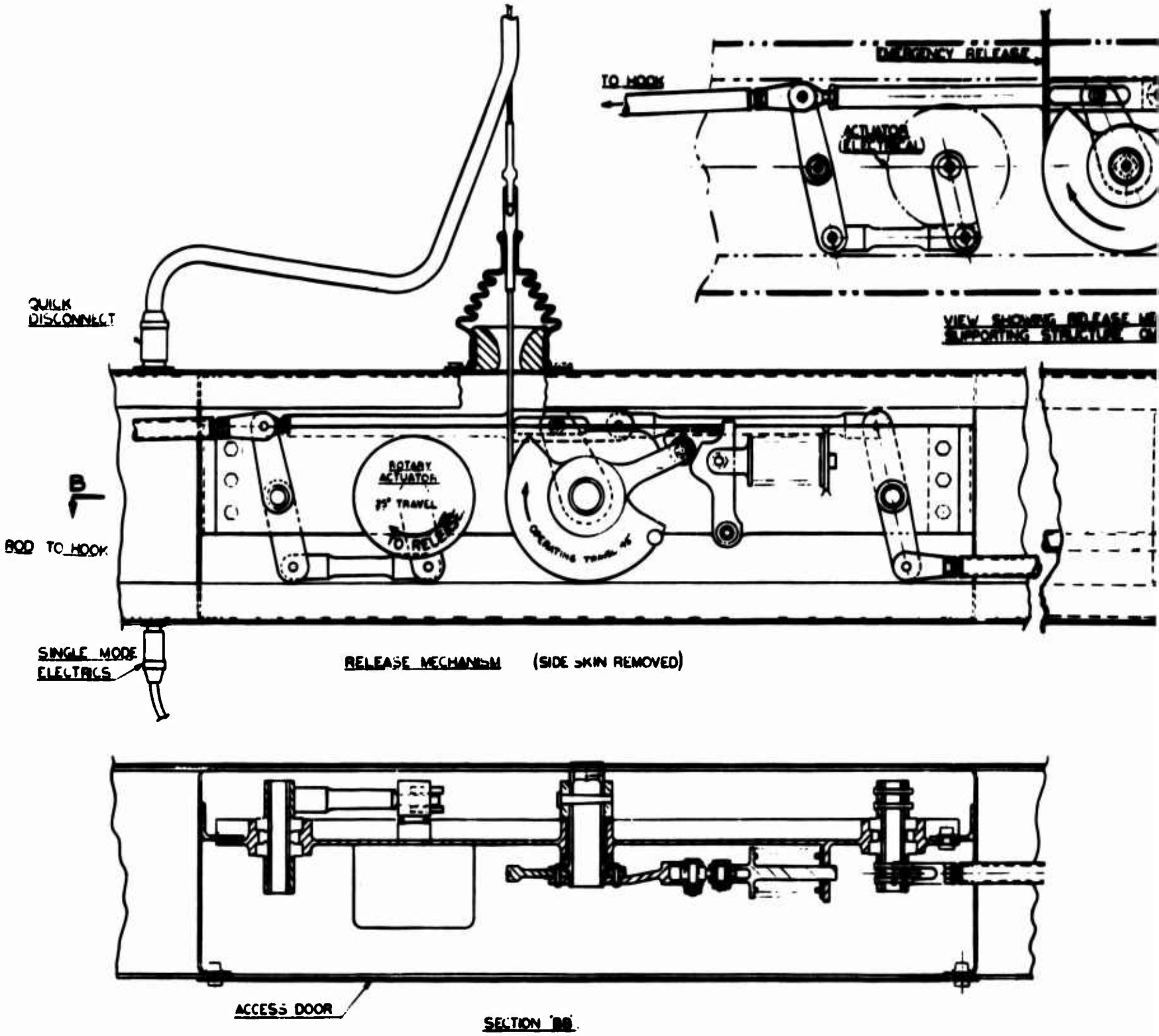
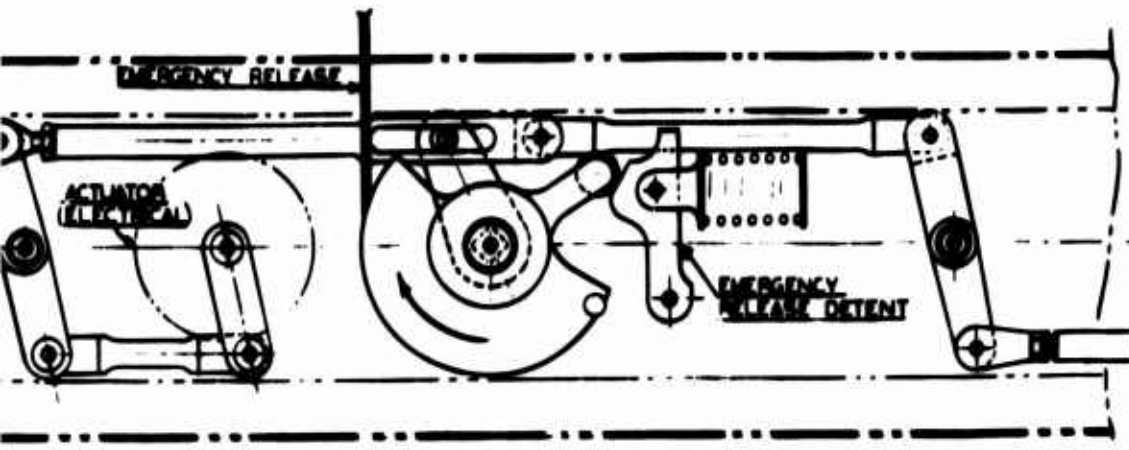
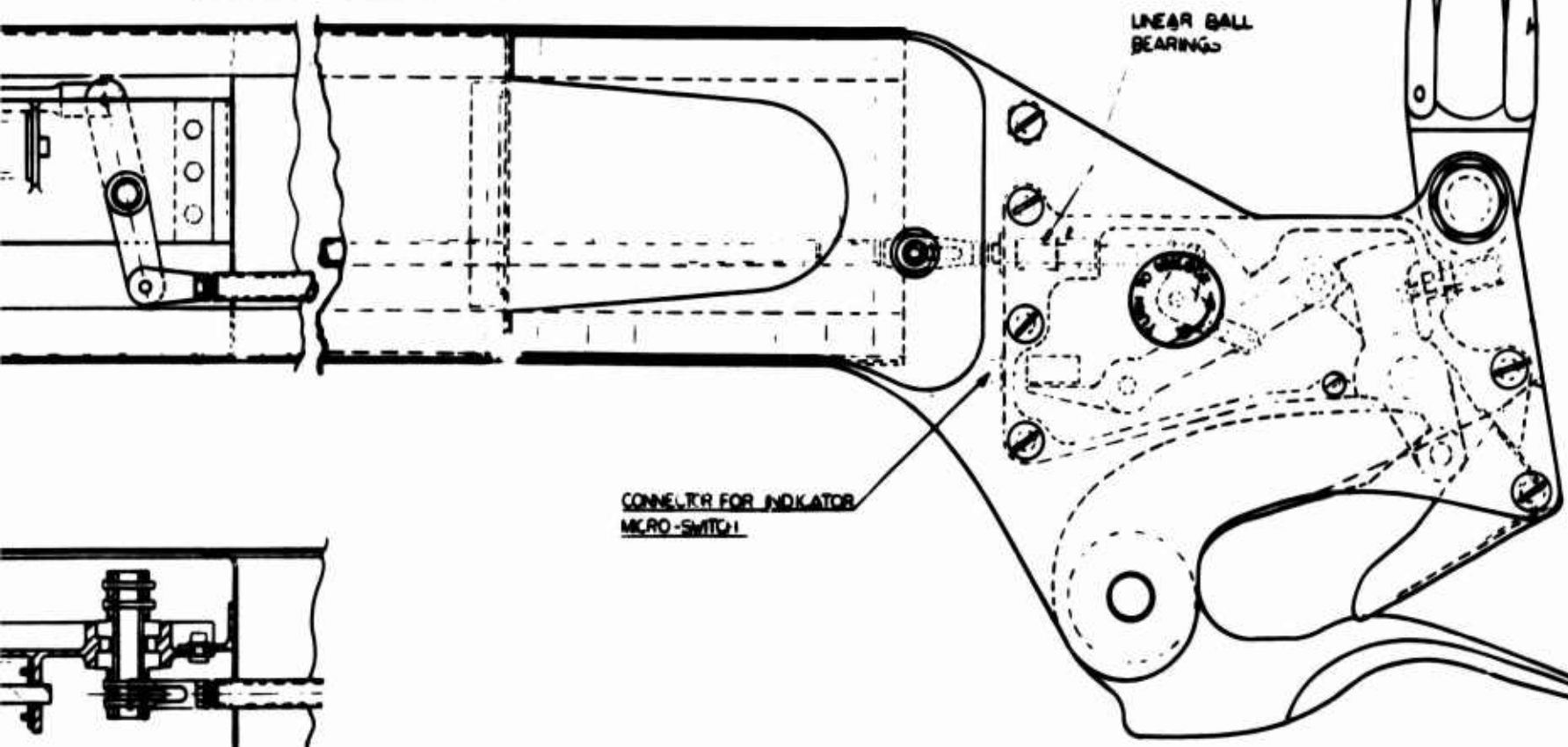
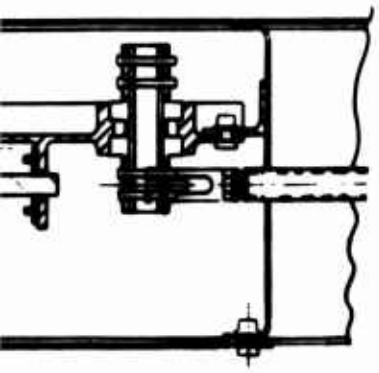


Figure 116. Beam Assembly, 20-Ton Hoist System.



VIEW SHOWING RELEASE MECHANISM SUPPORTING STRUCTURE OMITTED





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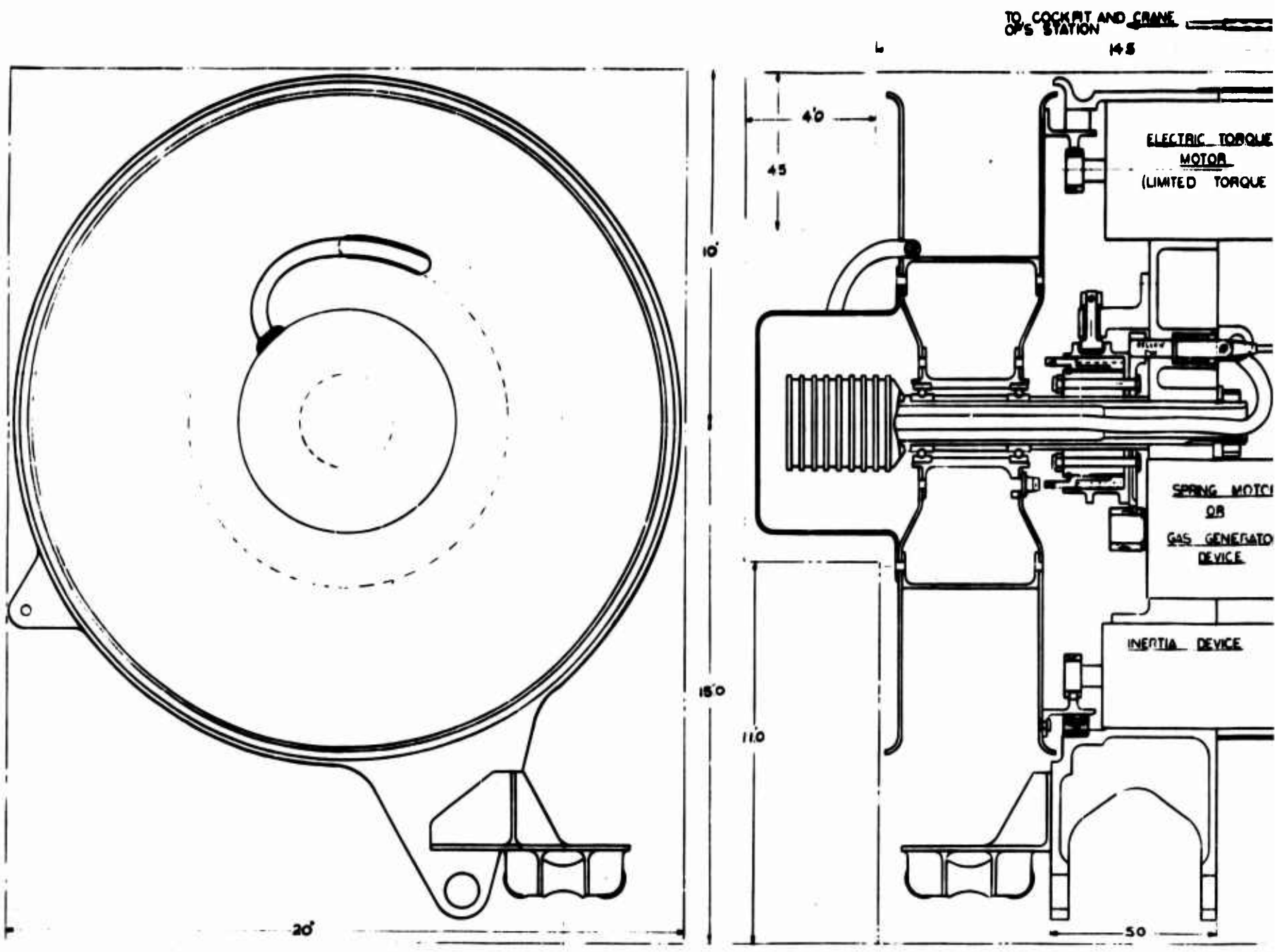
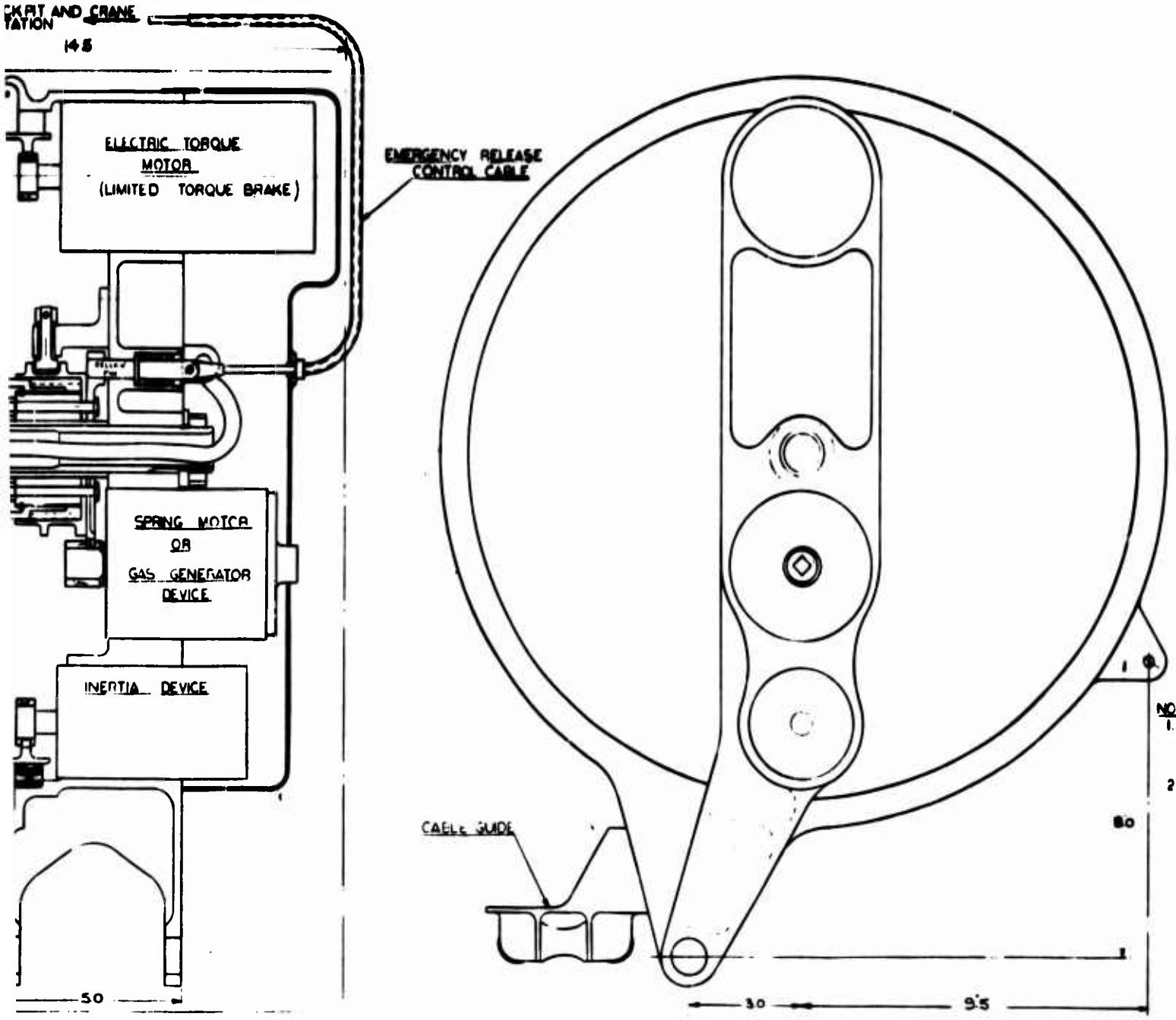


Figure 117. Release Unit, 20-Ton Hoist System.



NOTE

1. DIMENSIONS SHOWN ARE MAXIMUM ALLOWABLE COMPONENT ENVELOPE
2. THIS DRAWING SHALL BE USED IN CONJUNCTION WITH PROCUREMENT SPECIFICATION D8-0691

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Emergency Release System Installation in Helicopter

The installation of the hook manual release (emergency) system in the helicopter, regardless of whether or not the helicopter configuration be a transport, crane-transport, or pure crane-type, creates no new installation problems since the system will be similar to those presently installed in military aircraft (e.g., CH-46). Essentially, this type of system consists of a mechanical push-pull element from the hook emergency release system to a "pull to operate" tee-handle mounted on the cockpit floor. In Navy helicopters (CH-46), both pilot and copilot are provided with individual tee-handles, whereas, on the Army CH-47 helicopter the emergency release is the responsibility of the crew chief at the hook station, the pilots having no control over this function except by voice contact with the crew chief.

It is of interest to note that the foregoing illustrates the different philosophies regarding the operation of cargo emergency release systems on helicopters. On the one hand, it is assumed that the pilot requires both hands to fly the helicopter during an emergency, and removal of one hand from the controls to operate the emergency hook release system is considered (during a flight emergency) to be a hazardous procedure. On the other hand, providing independent emergency release to both pilot and copilot provides two more hands in the cockpit for emergency release operation. This relies upon voice contact between pilots to effect coordination, the same as voice contact between pilot and crew chief.

The introduction of a crane operator creates additional emergency criteria which are at present not defined. The crane operator on the CH-54A helicopter has limited flight control authority, which means that both hands are occupied for flight control. The hoist system control functions (electrical) are integrated into the two control grips, there being no emergency (manual) hook release system on this aircraft. Introduction of a manual hook release system will create the same hazardous condition as described in the preceding paragraph. It can then be reasoned that the emergency (manual) cargo release system should be installed in the pilot's cockpit with provision for both pilot and copilot operation. It is not, however, implied that this arrangement be specific, since any future helicopter proposals having a crane operator station will be subject to a human factors appraisal of cockpit layouts and arrangements to ensure the optimum

1. HOOK STATUS
2. HOOK ARMING SWITCH
3. SUSPENSION ROPE LOAD INDICATOR
4. ROPE LENGTH INDICATOR
5. WINCH SYSTEM HYDRAULIC PRESSURE INDICATOR
6. WINCH LUBRICATING OIL TEMPERATURE INDICATOR
7. MODE SELECTION INDICATOR
8. LOAD CELL
9. LOAD ISOLATOR
10. SUSPENSION ROPE CUTTER

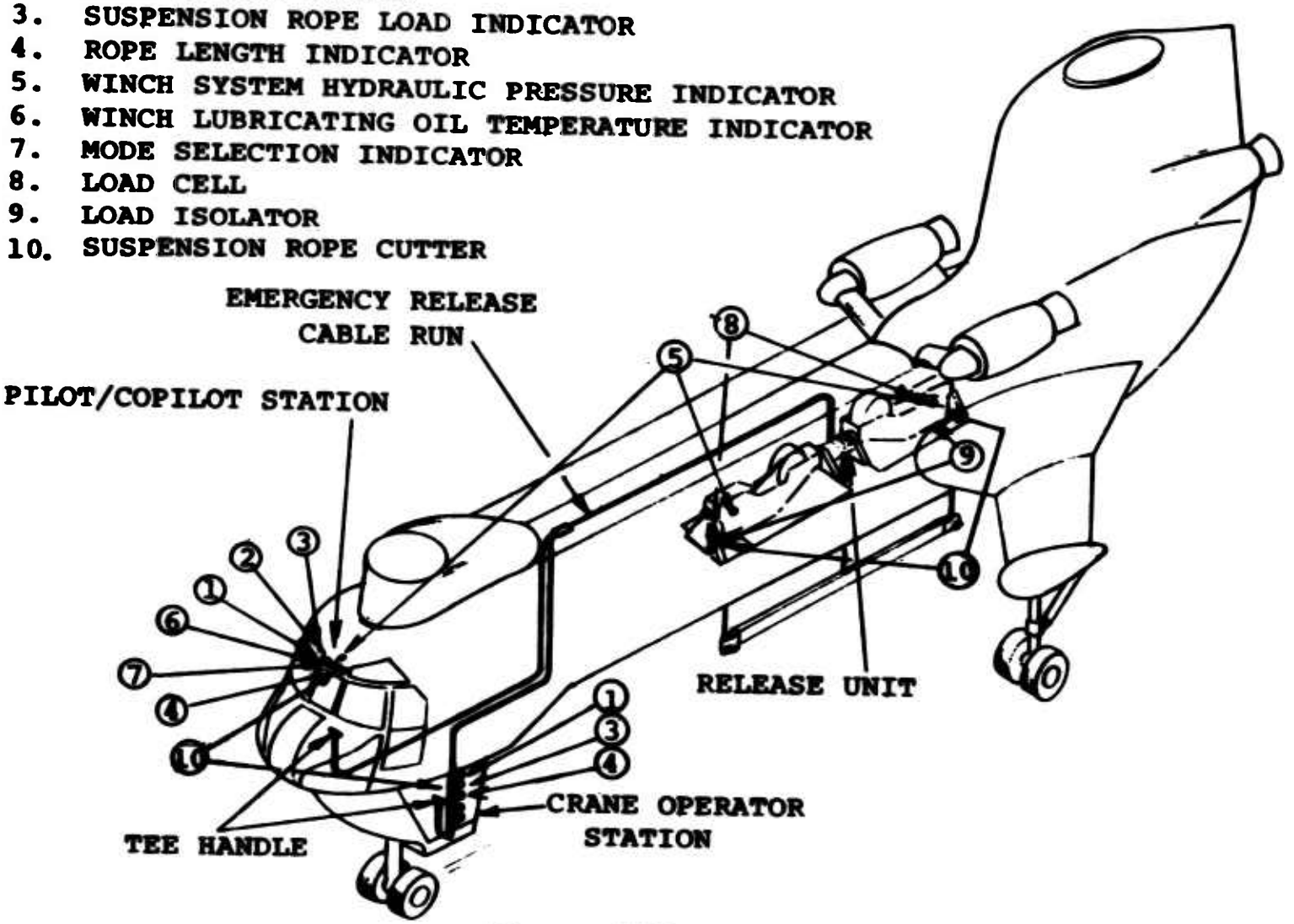


Figure 118. Emergency Release System Installation and System Monitorsnip Locations.

integration of all system (flight & hoist) functions encompassing all modes of operation. Additionally, the percentage of flight control authority given to the crane operator and its integration into the primary flight control system must be established. Experience gained on the CH-54A helicopter to date is small and inconclusive, and opinion ranges from giving the crane operator no authority to giving him full authority.

Figure 118 shows an arrangement of a manual emergency release system into a crane-transport-configured helicopter. In this instance a manual release is led to both pilot and crane operator stations.

SYSTEM MONITORSHIP

Included in Figure 118 is the approximate location of the hoist system monitoring instrumentation and its relative position in an assumed configuration. Although various elements of system monitorship are included in the subsystem component descriptions in this report, the following summary groups them together for reference.

TABLE XII

SYSTEM MONITORSHIP

Monitor Element	Location	Remarks
Hook Status	Pilot/Copilot plus Crane operator stations. GO (green light)-NO GO indicators	Each hook has switching circuits to indicate hook open (RED), hook closed (GREEN)
Hook Arming	Pilot/Copilot station	Release circuits may be placed inoperative during flight to prevent inadvertent release
Suspension Rope Load Monitoring	Pilot/Copilot station plus Crane operator station	Load cells mounted in series with load isolators on winch mounts. Readout in 1000-pound increments per rope

TABLE XII Continued		
Monitor Element	Location	Remarks
Rope Length Monitoring	Pilot/Copilot station plus Crane operator station	Potentiometer on each winch. Readout in feet out or in on one instrument (dual needle)
Winch System Hydraulic Pressure	Pilot/Copilot plus winch hydraulic manifold	Pressure transducer installed in pressure side of winch hydraulic supply. Readout in P.S.I.
Winch Lubricating Oil Temperature	Pilot/Copilot	Temperature transducer in gear case of each winch. Readout could be either by instrument showing temperature or by red warning light.
Mode Selection (i.e., Single- or Two-Point Mode)	Pilot/Copilot	Selection of one mode automatically disconnects the remaining mode. NOTE: This can also be accomplished by having the single-point mode plug on the cargo beam automatically de-energize the two-point electrical release system when the single-point mode plug is inserted.

WINCH DRIVE HYDRAULIC SYSTEM

General

The following hydraulic system description and schematic are not intended to be specific in all details since the prime mover into which the hoist system will be installed is not defined. However, the hydraulic system shown is considered practical and of simple design and applicable to either a synchronized or an unsynchronized control funtion. All major hoist system control elements (hyraulic and complementary electrics) are shown to be integrated into a single (one per winch) manifold which will be mounted directly to its winch and form an integral part thereof. This feature is recommended since the entire winch package, i.e., winch, winch mounts, load isolator, load cell, and control package, can now be easily removed as a complete assembly from the aircraft. This will allow maximum weight strippage from the aircraft when alternate missions are required. Additionally, the integration of the hydraulic and electrical winch control elements with the winch simplifies the ground test equipment requirements, since all that will be required will be an appropriate hydraulic electric power source. Further, the winch plus all its control elements can be bench-tested as a complete package before installation.

Hydraulic System Description

Hydraulic power to operate the two-winch hoist system is supplied by an assembly of two variable-displacement pumps, each of 50-hp and 3000-psi class. Both are mounted on a common gearbox, which provides the mechanical power input to the system.

A vented-type fluid tank supplies the fluid to each pump through a separate suction line and a separate filter. Each pump output is supplied via a check valve to a common output point in a manifold, from which it branches off again to each of the two hydraulic motors, one on each winch. Due to the above common output point, each of the pumps can supply fluid power simultaneously to both motors. Thus, in case of one pump malfunction, the other pump would continue to power the entire system.

The hydraulic motors are of fixed-displacement type and are mounted on drive pads provided one on each winch drive

mechanism housing (see Figures 119 and 120). Each motor is controlled by an electrohydraulic servo valve mounted on an adjacent manifold fixed to the winch beam assembly. The servo-valve electronic input circuit has been shown in Figure 89.

Each motor manifold also comprises a solenoid operated "On-Off" valve, a filter, a relief valve and a connection via a snubber to a pressure transmitter, having electrical connections to two pressure indicators: one on the manifold and the other on the hoist system control panel. Each motor manifold has one supply port and one return port. Each of the ports is connected to a fixed portion of a self-sealing quick-disconnect coupling, the moving portion of which is mounted on the end of the corresponding (supply or return) hydraulic hose.

Two additional self-sealing quick-disconnect couplings are used for ground test purposes. The fixed portions of the couplings belong to the winch drive hydraulic system, and the hose-mounted portions remain with the ground test pump unit (which has independent power drive). When the ground test unit is connected to the winch hydraulic system and put into operation, it draws fluid from the system tank via a common input point to which all pump suction lines are interconnected. Similarily, the test pressure line leads to a common output point. This enables the ground test pump to simultaneously supply fluid power to both hydraulic motors, unless any one of the solenoid valves was operated to shut the fluid supply off and thus to immobilize the hydraulic motor.

SYSTEM WEIGHT SUMMARY

The following weight estimate is based on the use of such standard materials as steel, aluminum alloy and magnesium alloy. Additionally, only standard size bearings and hardware are used. The use of exotic materials, custom-made parts and more complex manufacturing techniques would result in a lower system weight. The estimates shown in Table XIII do not include the prime mover hydraulic system or the structural modifications required to accept the hoist system. It does, however, include the components required to mount the winches into the prime mover. All figures shown are in pounds.

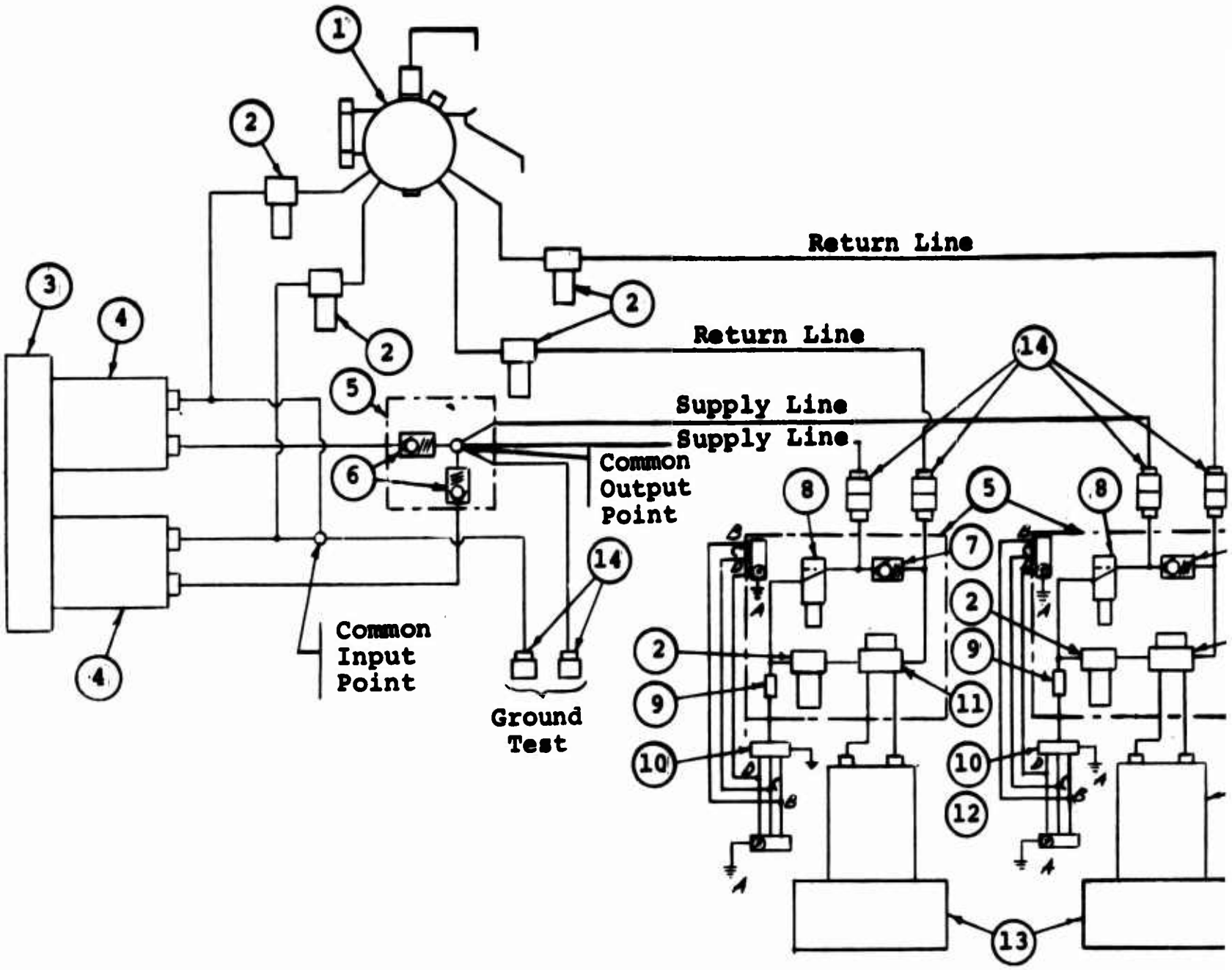
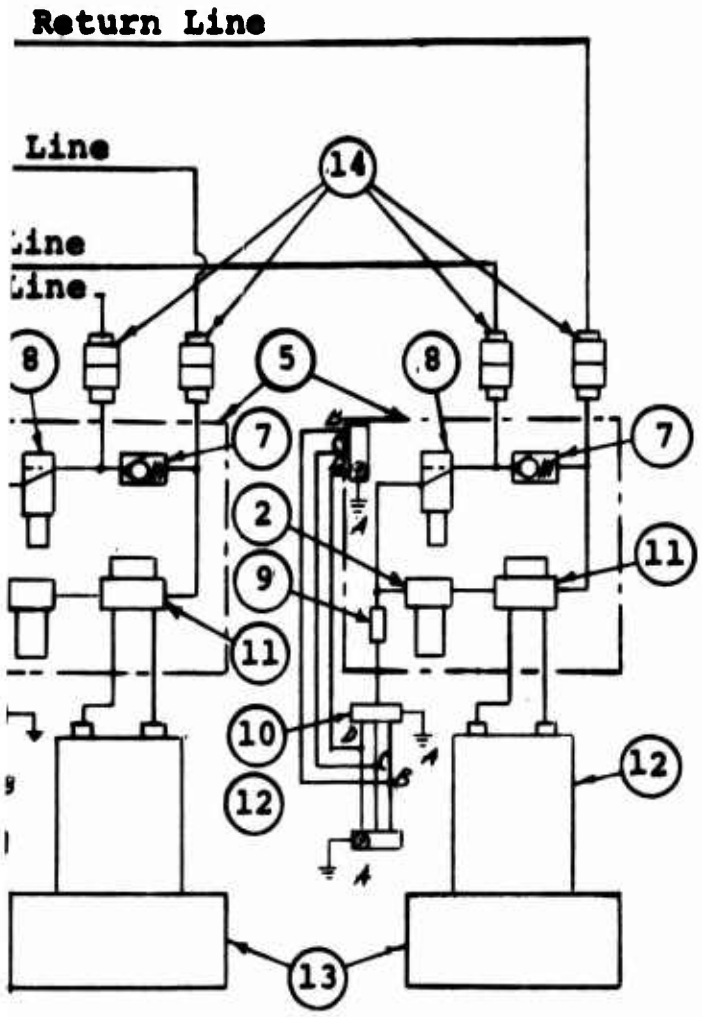


Figure 119. Winch Drive Hydraulic System Schematic Diagram.



1. FLUID TANK ASSEMBLY
2. FILTER
3. GEARBOX
4. PUMP VARIABLE DELIVERY
5. MANIFOLD
6. CHECK VALVE
7. RELIEF VALVE
8. SOLENOID VALVE
9. SNUBBER
10. PRESSURE TRANSDUCER & INDICATOR
11. ELECTRO HYDRAULIC SERVO VALVE
12. MOTOR FIXED DISPLACEMENT
13. WINCH DRIVE MECHANISM
14. QUICK-DISCONNECT COUPLING

gram.

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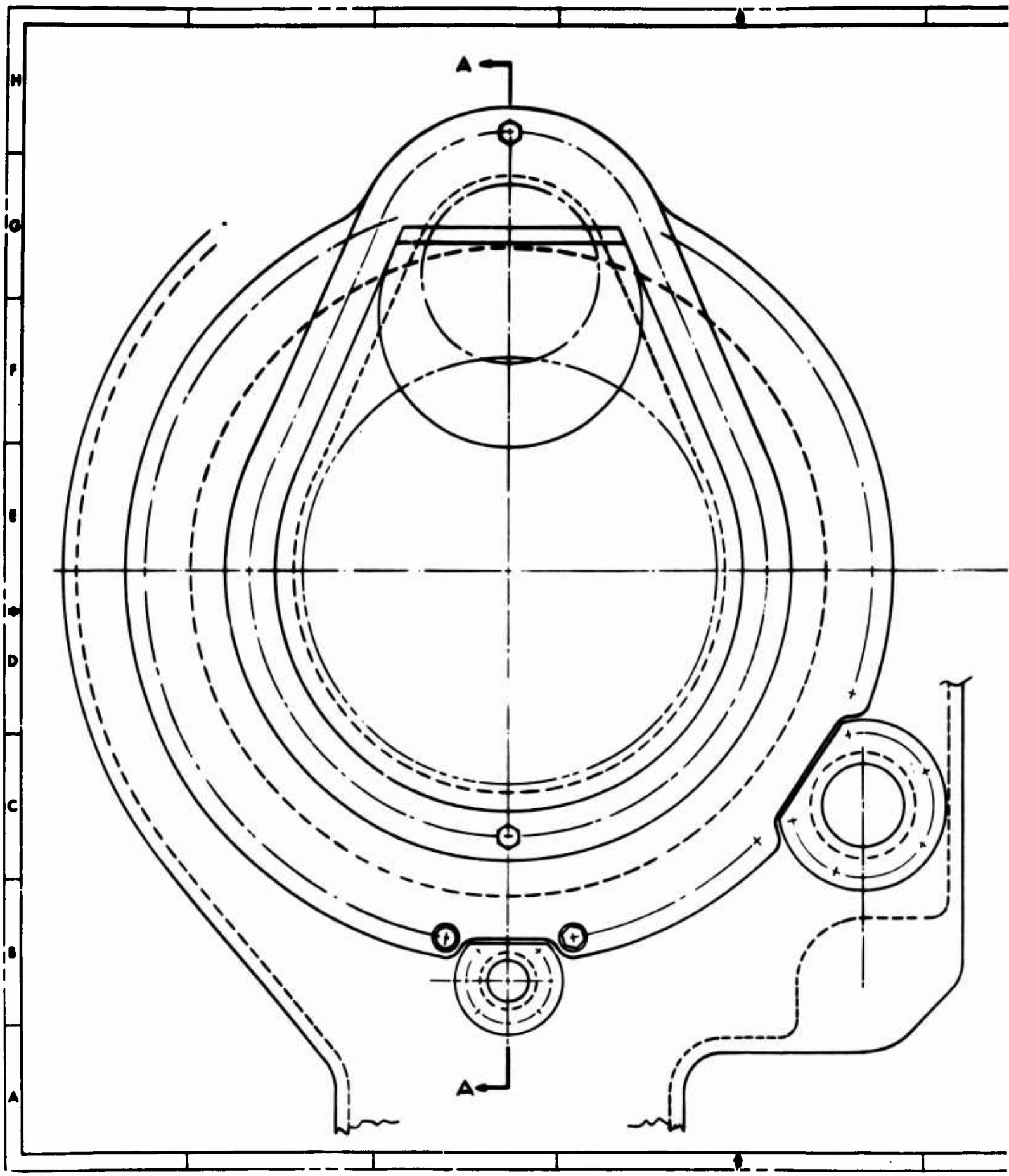
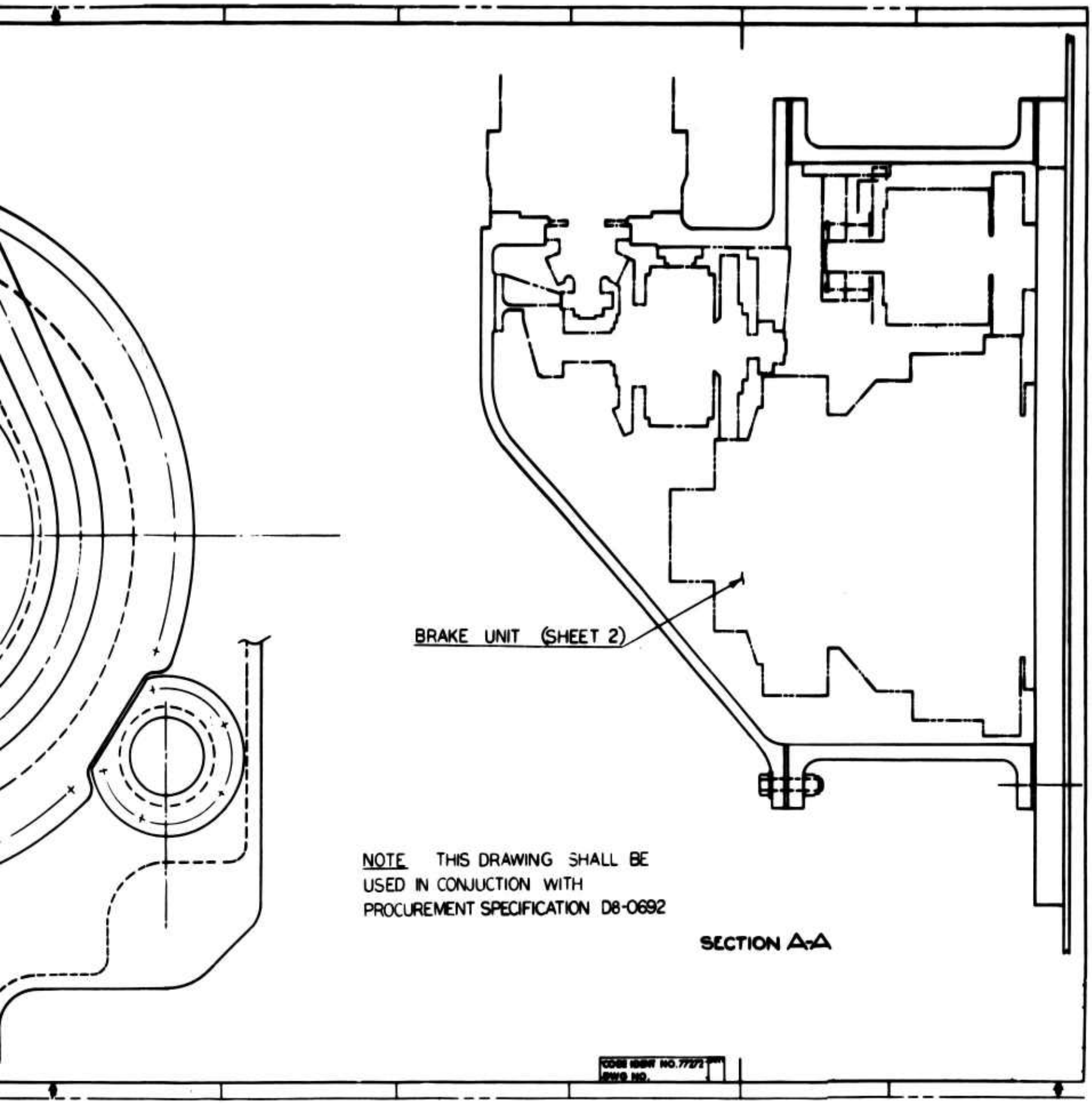


Figure 120. Detail of Clutch and Brake Enclosure Winch 20-Ton Hoist System.


BRAKE UNIT (SHEET 2)
NOTE THIS DRAWING SHALL BE
USED IN CONJUCTION WITH
PROCUREMENT SPECIFICATION D8-0692
SECTION A-A

osure Winch

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TABLE XIII
SYSTEM WEIGHTS (BREAKDOWN)

Main Drive, Drum, and Brake Assembly		
Drive Housing	29.5	
Initial Gearing	3.0	
Reverse Locking Clutch	3.0	
Spur Gears	5.5	
Clutch Assembly	23.0	
Drive Shaft	6.0	
Brake Assembly	49.5	
Spur Gears to Backstop Clutch	10.0	
Backstop Clutch Assembly	11.5	
Main Gearing	70.0	
All Bearings	87.5	
Drum Base Plate Inner	35.0	
Drum Base Plate Outer	20.0	
Drum Outer (Tread)	49.5	
Drum Inner	32.5	
Drum Gear	24.5	
Bolts, Nuts, Spacers, etc.	15.0	
		475
Oil (11 gals.)	83	
Hydraulic Motor	33	
		116
Wire Rope (200 ft)		342
	Total	933
Secondary Capstan Drum		
Drum Outer (Tread)	49.5	
Drum Inner	14.78	
Sprocket	1.69	
Drum Mounting Plates	7.64	
Drive Gear	8.00	
Idler Gear	1.90	
Bearings	9.00	
Idler Gear Support	.13	
	Total	92.64

TABLE XIII (continued)

Bellmouth Support Housing		
Main Casting	4.80	
Bearings	.72	
Bearing Housing	.56	
Bellmouth	1.80	
Bolts, Nuts, etc.	1.00	
Cable Cutters	7.00	
	Total	15.88
Rope Storage Drum		
Drum Flanges	6.26	
Drum	7.00	
Shaft	3.65	
Drive Sprocket	1.69	
Bearings	3.25	
Clutch Plates	9.70	
Clutch Plate Retainers	1.90	
Clutch Plate Support	.55	
Level-Wind Sprocket	1.35	
Rope Tension Clamp	.15	
Bearing Housings	.65	
Clutch Springs & Retainers	.75	
Bolts, Nuts, etc.	2.00	
Drive Chain	3.00	
One-way drives	8.00	
	Total	49.90
Rope Storage Drum Level Wind		
Support Bracket (2)	3.30	
Level-Wind Cam Shaft	2.33	
Sprocket	.78	
Cam Follower Assembly	.90	
Bearings	.72	
Drive Chain	1.50	
Bolts, Nuts, etc.	.25	
	Total	9.78

TABLE XIII (continued)

Rope Reel-Out Pulleys and Rope Retention Roller		
Drive Gear	1.87	
Dirve Shaft	1.58	
Bearings	.72	
Bearing Housings	2.52	
Roller Pulleys	2.25	
Roller	2.00	
Bolts, Nuts, etc.	.75	
Total		11.69
Main Winch Structural Elements		
End Plates	12.55	
Side Beams	62.38	
Main Winch Support Fitting	5.78	
Minor Winch Support Fitting	3.14	
Bolts, Nuts, etc.	1.35	
Total		85.20
Winch Main Support (To Aircraft)		
Main Structure	13.20	
Main Bearing Retaining Plates	1.80	
Main Bearing Housing	2.50	
Oilite Bearing Surfaces	1.70	
Main Bearing & Nut	7.70	
Struts	4.15	
End Fittings	.88	
Bolts, Nuts, etc.	2.00	
Total		33.93
Winch Support (To Aircraft)		
Main Structure	5.65	
Bearing Plus Retaining Nut	3.00	
Total		8.65

TABLE XIII (continued)

Summary of One Winch, Including the Internal Lubricant, Motor and Wire Rope		
Main Drive, Drum & Brake Assembly	475	
Oil (11 gals.)	83	
Hydraulic Motor	33	
Wire Rope (200 ft)	342	
Secondary Capstan Drum	92.64	
Bellmouth Support Housing (Including Cutters)	15.88	
Rope Storage Drum	49.90	
Rope Storage Drum Level Wind	9.78	
Rope Reel-Out Pulleys & Rope Retention Roller	11.69	
Main Winch Structural Elements	85.20	
Total		1198.09
Release Unit Assembly		
Storage Drum Side Plates	4.80	
Storage Drum	1.52	
Storage Drum Side Support	.65	
Bearings 2 AW20AK	.71	
Spacer Tube	.10	
Slip Ring Assembly	1.50	
Slip Ring Assembly Cover	.50	
Center Shaft	1.52	
Main Gear Ring	2.00	
Manual Release Unit Details	2.50	
Main Support Casting	2.50	
Torque Motor	3.00	
Manual Release Motor	3.00	
Inertia Device	2.50	
Protective Casing	2.00	
Fasteners, etc.	.50	
		29.30
Electromechanical Cable (160 feet)		25.00
Total		54.30
Beam And Hook Assembly		
Side and Top Skins	24.48	
Corner Extrusions	22.83	

TABLE XIII (continued)

Bulkheads	2.69	
Levers and Pivots	.25	
Cam Support Shaft	.13	
Hook Release Rods	1.70	
Bearings	.18	
Rod Ends	.42	
Electrical Connectors	.25	
Electrical Wiring	.30	
Hook Assemblies	100.00	
Cam Wheel	.50	
Cam Follower	.17	
Spring Assembly	.33	
Support Plate	.85	
Support Plate Angles	.12	
Wire Rope Termination Fittings	17.00	
Actuator	10.00	
Total		182.20

Weight Summary of One Complete Aircraft System

Two Winches Complete (1198.09 x 2)	2396.18
Beam Assembly	182.20
Release Unit (Complete with 160' Cable)	54.30
Winch Main Support (Two)	67.86
Winch Supports (Two)	17.30
Load Isolator/Load Cells	80.00
	2797.84
Single Point Mode Hook & Ropes	200.00
	2997.84

SYSTEM GROWTH ANALYSIS

Lift Capability

The two-winch, two-point system shown in this report has been designed with two suspension lines, each having a lift capacity of 25,000 pounds. The 25,000 pounds is necessary to allow for longitudinal cg tolerances in the load. However, if the load cg is in the center of the beam as will be the case when a single-point lift mode is made, providing a vertical lift is made and assuming that the prime mover has the lift capability, then loads up to 50,000 pounds may be made, i.e., 10,000 pounds over the design hoist system load of 40,000 pounds. However, if this lift is undertaken, the forward flight of the prime mover will be restricted, since drag

forces on the load cause a redistribution of load in the suspension ropes.

Single-Point Winch Capability (Reference Figure 110)

As previously mentioned, a winch designed for the two-point system may be used as a single-point winch of 50,000 pounds capacity by reeving the wire rope into a two-rope, single-point system. This will require the introduction of a ball bearing swivel between the hook and rope pulley as shown in Figure 110. Provision has not been made for the receipt of the rope "free" end on the winch structure. The winch side beams, which will pick up the load in this rope, will have to be beefed up locally and the winch support bearing size increased to transfer the additional load to the aircraft structure. Also, since one winch will now pick up twice the lateral and longitudinal loads due to load swinging, the winch supports will have to be strengthened.

Two-Point Increased Capacity

Similarly, the capacity of the two-point system, if the winch ropes are reeved, can be doubled (i.e., 80,000 pounds). The changes discussed above will also be necessary by this modification. This increased capacity will require a new beam and hook system having:

1. Hooks of 50,000 pounds capacity each.

2. The introduction of sheaves on the beam for wire rope guidance.

3. Modification to the release unit to compensate for the system rate change (i.e., reeving the rope will halve the original rate of 60 feet per minute to 30).

Note: A system rate of 30 feet per minute for lifting a 40-ton load is considered to be adequate.

The above growth without major redesign means that a hoist system developed and qualified to the criteria specified in this report can be used on future heavy-lift helicopters at minimum cost to a future heavy-lift program; i.e., loads of 40 tons could be lifted on the two-point mode and 50 tons on the single-point mode.

System Growth

Additional growth by increasing the capability of the winch within the same winch envelope up to 25% is considered feasible. The dynamic components would have to be uprated. The major element would be the wire rope. This would mean the procurement of a nonstandard rope of approximately 1.1 inches diameter. The drum diameter to rope diameter ratio, assuming that the retention of the existing winch envelope is maintained (16.36:1), is considered acceptable. (A loss in winch life would result.) The rope level-wind drive ratio would also have to be changed to compensate for the new rope diameter. Such a change is considered feasible but is not recommended unless instituted as a limited-life system.

Growth Summary

The following summary indicates the versatility of the two-winch system design:

1. The System as Designed

Capacity - Two-Point Mode	40,000 pounds
- Single-Point Mode	50,000 pounds
Rate	60 feet per min
Lift Height (both modes)	150 feet

2. The System as Designed but with Reeved Ropes

Capacity - Two-Point Mode	80,000 pounds
- Single-Point Mode	100,000 pounds
Rate	30 feet per min
Lift Height (both modes)	75 feet

3. The System as Designed plus 25% Increase

Capacity - Two-Point Mode	50,000 pounds
- Single-Point Mode	62,500 pounds
Rate	60 feet per min
Lift Height (both modes)	150 feet

4. The System as Designed plus 25% Increase, but with Reeved Ropes

Capacity - Two-Point Mode	100,000 pounds
- Single-Point Mode	125,000 pounds

Rate	30 feet per min
Lift Height (both modes)	75 feet

Note: The lift height of 75 feet in Systems 2 and 4 above may be increased by replacing the rope storage drum on the winch with one of the required capacity.

PRELIMINARY RELIABILITY ANALYSIS

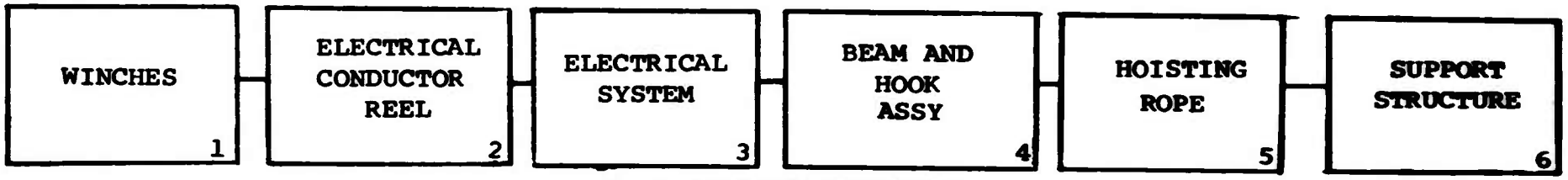
The preliminary reliability analysis of the heavy-lift helicopter external load is based on the reliability block diagrams and mathematical models shown in Figures 121 and 122.

Failure rates were obtained from the tri-service Failure Rate Data Handbook (FARADA).

The results of the analysis are presented as Mission Reliability R_M to a 1-, 2-, and 3-hour load handling system mission. (This is not to be construed as a 1-, 2-, or 3-hour helicopter mission.)

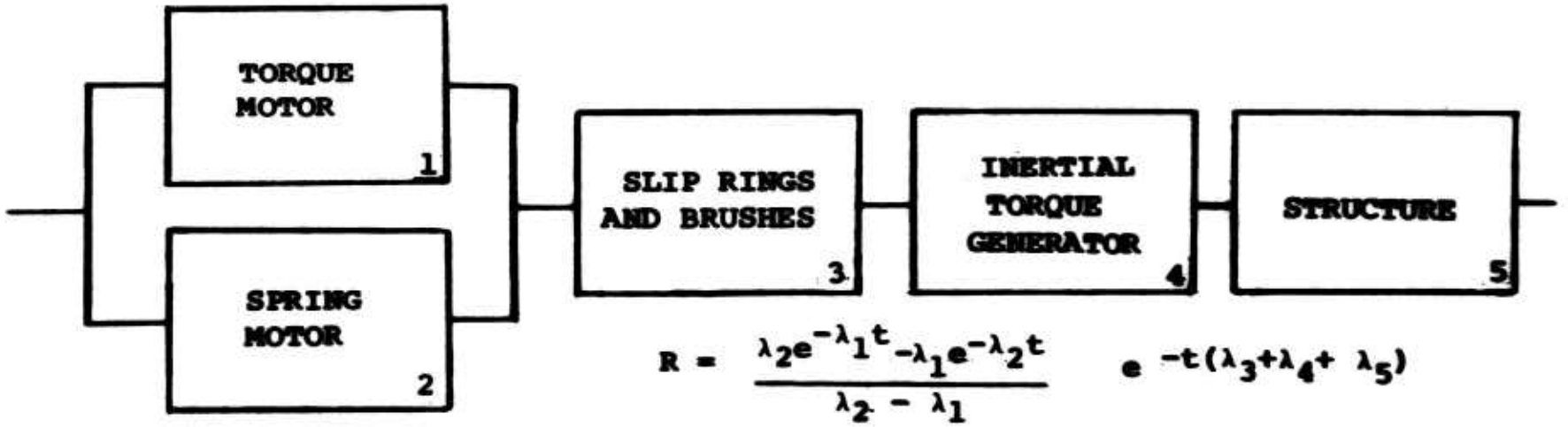
TABLE XIV
MISSION RELIABILITY

Component	1 Hr.	2 Hr.	3 Hr.
Winches	.9999514	.9999028	.9998542
Electrical Reel	.999952	.999904	.999856
Electrical System	.999932	.999864	.999796
Beam & Hook Assy	.999990	.999980	.999970
Hoisting Ropes	.999514	.999028	.998542
Supports	.999969	.999938	.999907
R_M	.999309	.998617	.997926



$$R_M = R_1 \cdot R_2 \cdot R_3 \cdot R_4 \cdot R_5 \cdot R6$$

Figure 121. External Load Handling System.



$$R = \frac{\lambda_2 e^{-\lambda_1 t} - \lambda_1 e^{-\lambda_2 t}}{\lambda_2 - \lambda_1} \quad e^{-t(\lambda_3 + \lambda_4 + \lambda_5)}$$

Figure 122. Electrical Conductor Reel.

PRELIMINARY MAINTAINABILITY STUDY ANALYSIS

The technique presented in this report has been developed to provide an engineering tool for designing equipment to fulfill specified maintainability requirements. Maintainability analysis methods will be established to complement the prediction technique.

Study Objectives

1. Identify maintainability factors that affect equipment availability and support.

2. Determine relationship of equipment design to maintainability.

3. Determine the relationship of maintainability to support environment and maintenance personnel.

The use of the prediction technique to evaluate equipment's maintainability is approached from a sampling basis. In this instance,comparative studies will be made of the CH-46 and CH-47 helicopters.

Analysis Method

Data derived from the prediction technique will be used to specifically determine areas within the equipment in which to effect maintainability improvement.

1. Maintenance time data evaluation to determine parts within the equipment requiring improvement.

2. Isolation of deficiencies for various design features.

The establishment and measurement of maintainability consist of analyzing existing data, expanding the existing data acquisition on operational CH-46 and CH-47 helicopters, and making appropriate comparison.

Hoist System Maintainability Checklist

1. Accessibility of components and parts requiring periodic or frequent replacement.

2. Accessibility of components and parts requiring frequent inspection for damage or corrosion due to their nature or location.

3. Accessibility of components and parts requiring frequent field adjustments, rigging and checking.

4. Minimum requirement for preventive and in-storage maintenance procedures.

5. Utilization of standard components and parts.

6. Interchangeability and replaceability of assemblies, components and parts.

7. Incorporation of quick disconnects for electrical connection and hydraulic line couplings at dynamic components.

8. Routing of control rods, hydraulic lines and like hardware to prevent damage by personnel performing maintenance tasks in area.

9. Systems designed to enable maximum utilization of standard tools.

10. Protection provided dynamic components from adverse effects of environment (dust, sand, moisture).

11. Simplified in-field check of status of each hoist mode.

12. Size of system and components of practical nature to allow ease of handling.

Results

1. As data evolved during the subject study, and available cost comparisons were derived, it became evident that there was little or no correlation between CH-47 and CH-46 winches and the proposed HLH winch. The preliminary analysis points to the need for more detailed maintainability data utilizing detail installation drawings of a final configuration.

2. The preliminary design critique and drawing reviews have been tested; they conform to the maintainability checklist and reflect all maintenance considerations. The areas considered to need more detailed explanation for a final analysis are: inspection access panels, provisions for external circuitry checks, and provision for functional check of safety systems.

3. As part of the preliminary analysis, we find that the following will be required:

   a. Daily inspections for security and condition.

   b. Preintended use inspection and functional check.

   c. On-condition replacement of cable, grooved drum treads, hooks and external fittings.

   d. Periodic installed maintenance and inspection of winch and cable assemblies every 50 hours or 50 cycles, to include lubrication and cleaning of cable, brake, and drum treads, and check of condition.

   e. Periodic removal and inspection of major components, i.e., brake, clutch, drum gears, and hydraulic motor, to include proof load test.

4. The current estimate of scheduled maintenance requirements is:

a.	Daily Inspection	.25 Man-hours
b.	Preintended use and functional check	.50 Man-hours
c.	50-hr/50-cycle inspection	2 Man-hours

5. Close monitoring on the detail design of the complete system will reduce the estimated times in paragraph 4 above.

GROUND HANDLING AND INSTALLATION OF WINCHES IN THE PRIME MOVER

Each complete winch assembly, weighing approximately 1300 pounds, creates a handling problem during installation. The hoist system is installed with the use of a dolly (see Figure 123). The dolly, consisting of a tubular framework with fully castering wheels, would have a capability of lifting the winch, complete with supports, off the aircraft floor structure and of subsequently transporting it out of the aircraft. The same dolly would be used to store the winch assembly when not installed in the aircraft and would also facilitate routine maintenance testing of the winch system (see section entitled "Winch Drive Hydraulic System" in this document).

The above referenced dolly assumes that the helicopter configuration will be a transport with a rear loading ramp, thus permitting the removal of the winch on its transport and storage dolly.

In the case where the helicopter configuration will be a crane transport with high landing gear, then provision will have to be made for lowering the winch from the aircraft. Hard-points will have to be provided in the fuselage cabin ceiling structure to receive a manually operated auxiliary hoist similar to those at present in use as bomb or external stores hoists on Navy and Air Force aircraft. Figure 124 illustrates the proposed method.

In the case where the helicopter configuration will be a crane (i.e., without cabin), provision will have to be made in the basic aircraft fuselage structure for a rope/pulley system which will allow the raising or lowering of the winch by means of a manual auxiliary hoist. (Figure 125)

In all cases above, a dolly will be required to receive and transport the winch when removed from the helicopter (Figure 123).

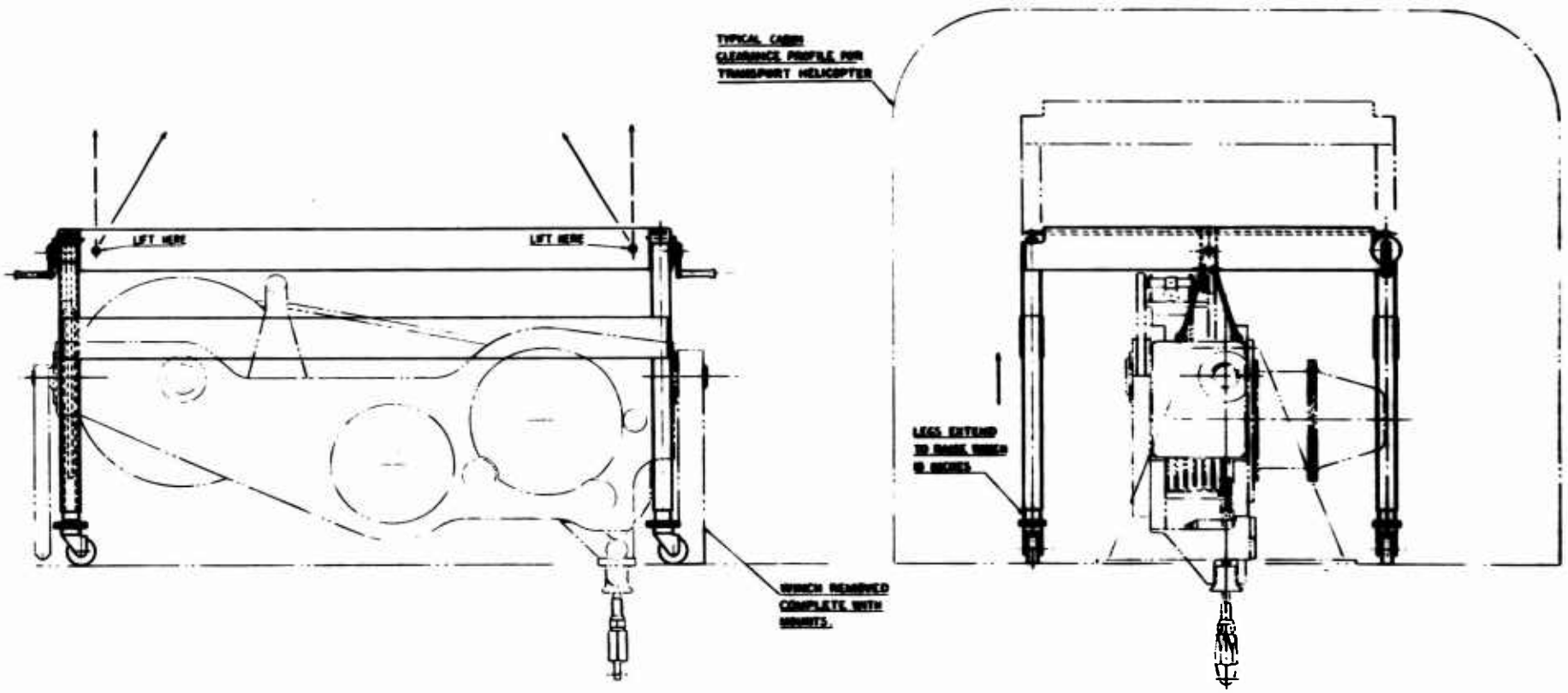


Figure 123. Installation of Dolly Winch, 20-Ton Hoist System.

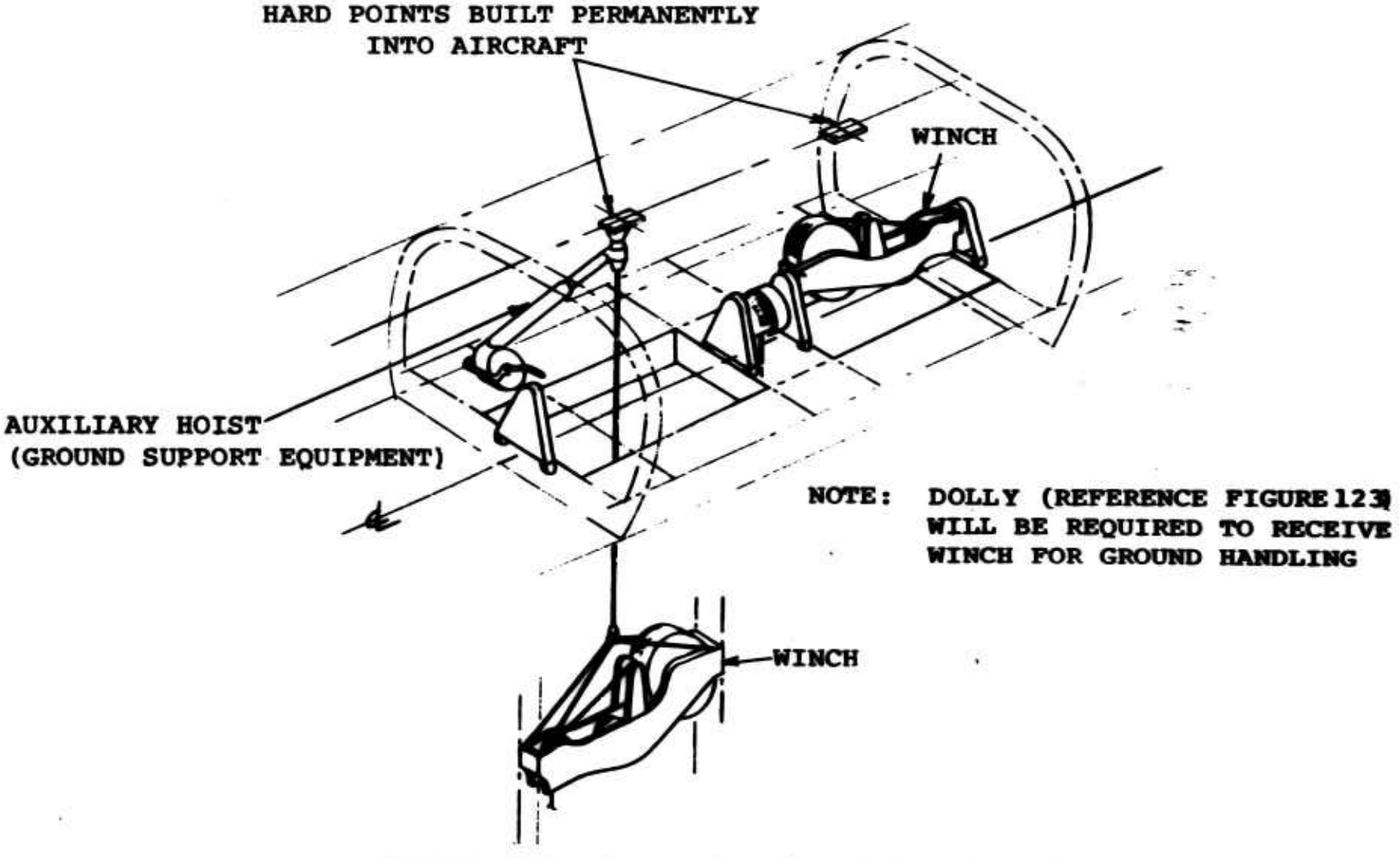


Figure 124. Removal and Installation of Winches in a Crane/Transport-Configured Helicopter.

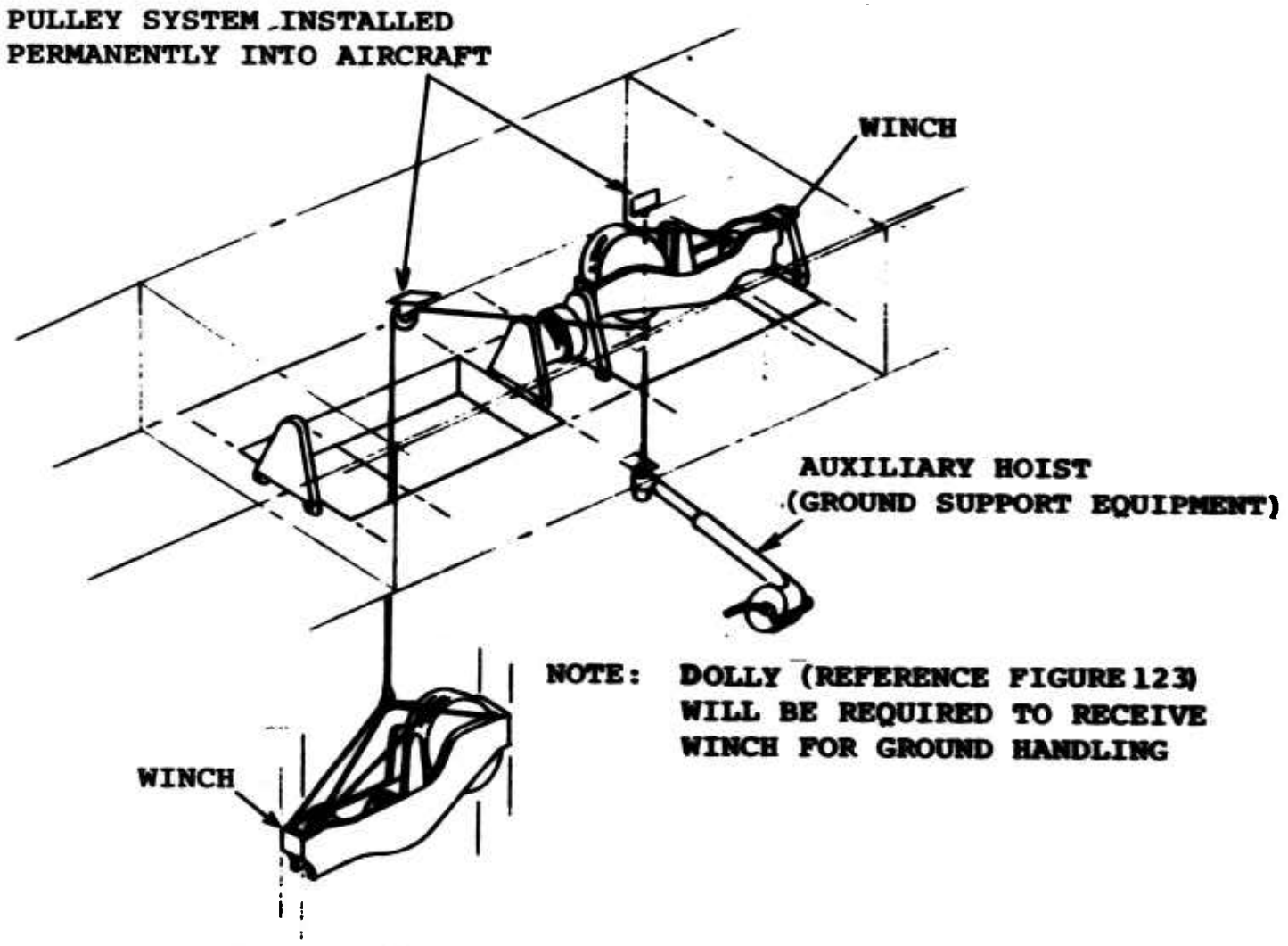


Figure 125. Removal and Installation of Winches in a Crane-Configured Helicopter.

SYSTEM SAFETY ANALYSIS

Introduction

A systems safety review was made of the two-winch, two-point, external-cargo handling system by using the "fault tree analysis" technique. The fault tree analysis technique has been effectively employed by the Boeing Company in other programs, and has been refined and standardized by Boeing so that this analysis can be virtually applied to any system, design, or procedure.

Fault tree analysis presents the interrelationships between individual subsystems, the operator, and the operational environment. A properly constructed fault tree readily identifies hazards, which result in a undesired event, that are not visible in a single thread analysis such as the failure modes, effects, and criticality analysis. Fault tree, therefore, is a tool to identify the total effect of any single failure on the hoist system.

Fault Tree Construction

A fault tree is constructed by properly relating all possible sequences of events that, upon occurrence, result in the undesired event. The fault tree graphically depicts the progressive paths that lead to each succeeding (higher) level of the display. This does not imply that each ascending fault path has a "higher probability of occurrence"; in fact, in many instances, the opposite may be the case. However, a series of "little things", each with a relatively low probability of occurrence, may trigger an event at the next higher level. This is depicted in the fault tree as a progression of events through the logic gates (see Figure 126).

The two basic logic gates used in constructing a fault tree are the "AND" gate and the "OR" gate. The "AND" gate performs the logic function that requires the coexistence of all gate input events in order to realize an output event. The "OR" gate performs the logic function that requires any one of the gate input events in order to realize an output event. A legend for the fault tree symbols used in this analysis is shown on the fault tree diagram shown in Figure 126.

Analysis

Step 1 defined the most undesired event; i.e., the event which must be kept from happening. This event was defined as "Aircraft destroyed during cargo handling operation". Next, the cargo handling operation was separated into the various operational phases; i.e., still applying the same philosophy that the top-most undesired event must be kept from happening during each of the operational phases. The phases selected were:

1. A/C* hovering over load. Ground crew securing load to A/C.

2. A/C hovering over load. Initial lift-off of load.

3. A/C hovering with load.

4. A/C hovering. Load simultaneously being reeled in.

5. A/C hovering. Load simultaneously being reeled out.

6. A/C cruising.

7. A/C hovering. Unloading.

*A/C = Aircraft

As an example of fault tree analysis, the first phase is examined in detail. Starting at the top of tree and working down, we proceed through the following course of events:

1. The most undesired event is defined as "aircraft destroyed during cargo handling operation".

2. The operating condition during which we do not want this event to occur is: "A/C hovering over load. Ground crew securing load to A/C".

3. An event which can cause the top output event, "destroyed A/C", is an "unstable A/C".

4. For the output event "unstable A/C" to occur, the <u>co-existence</u> of input events "unwanted movement of A/C" and "asymmetrical load" are required (this is indicated by the AND logic gate).

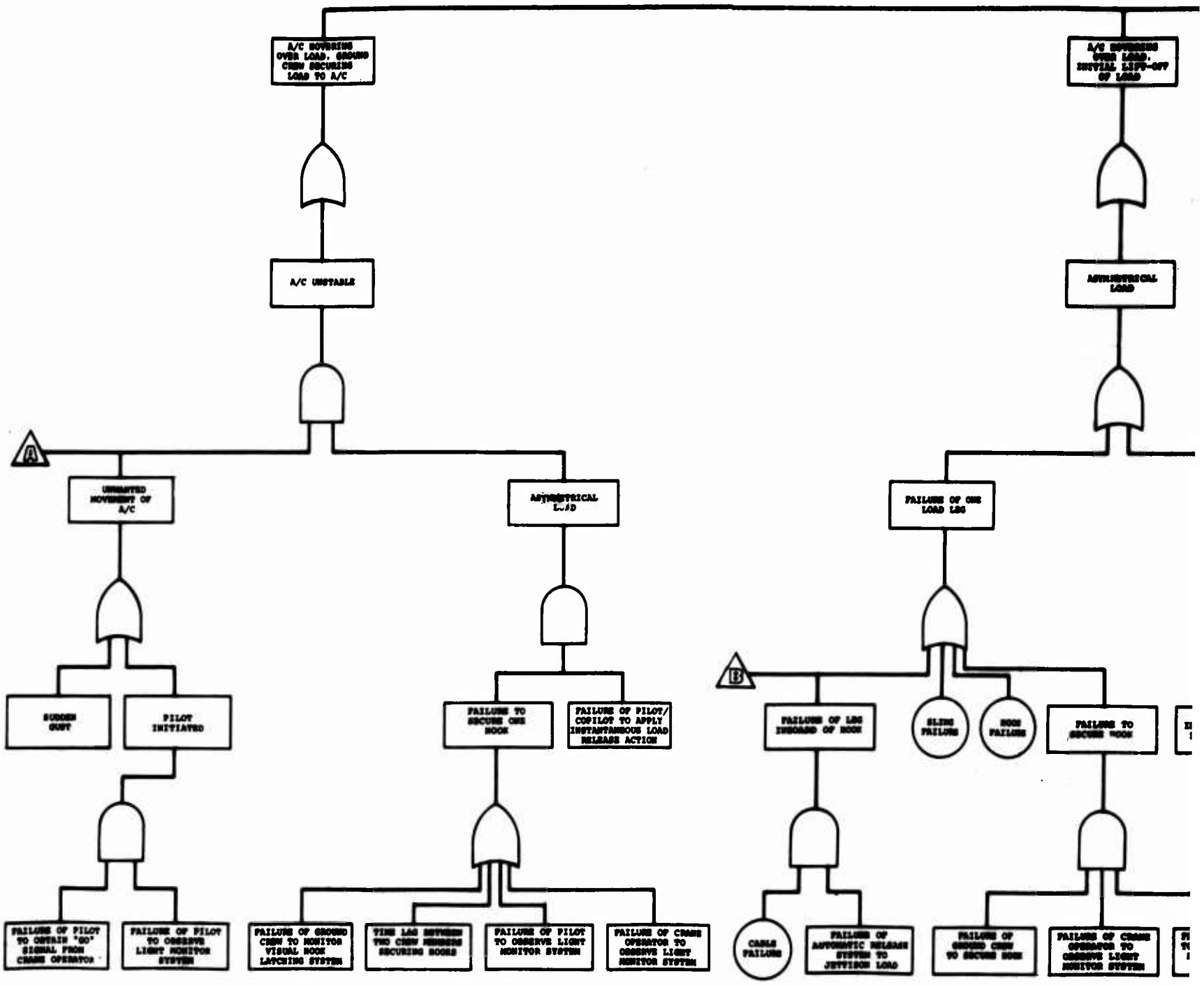
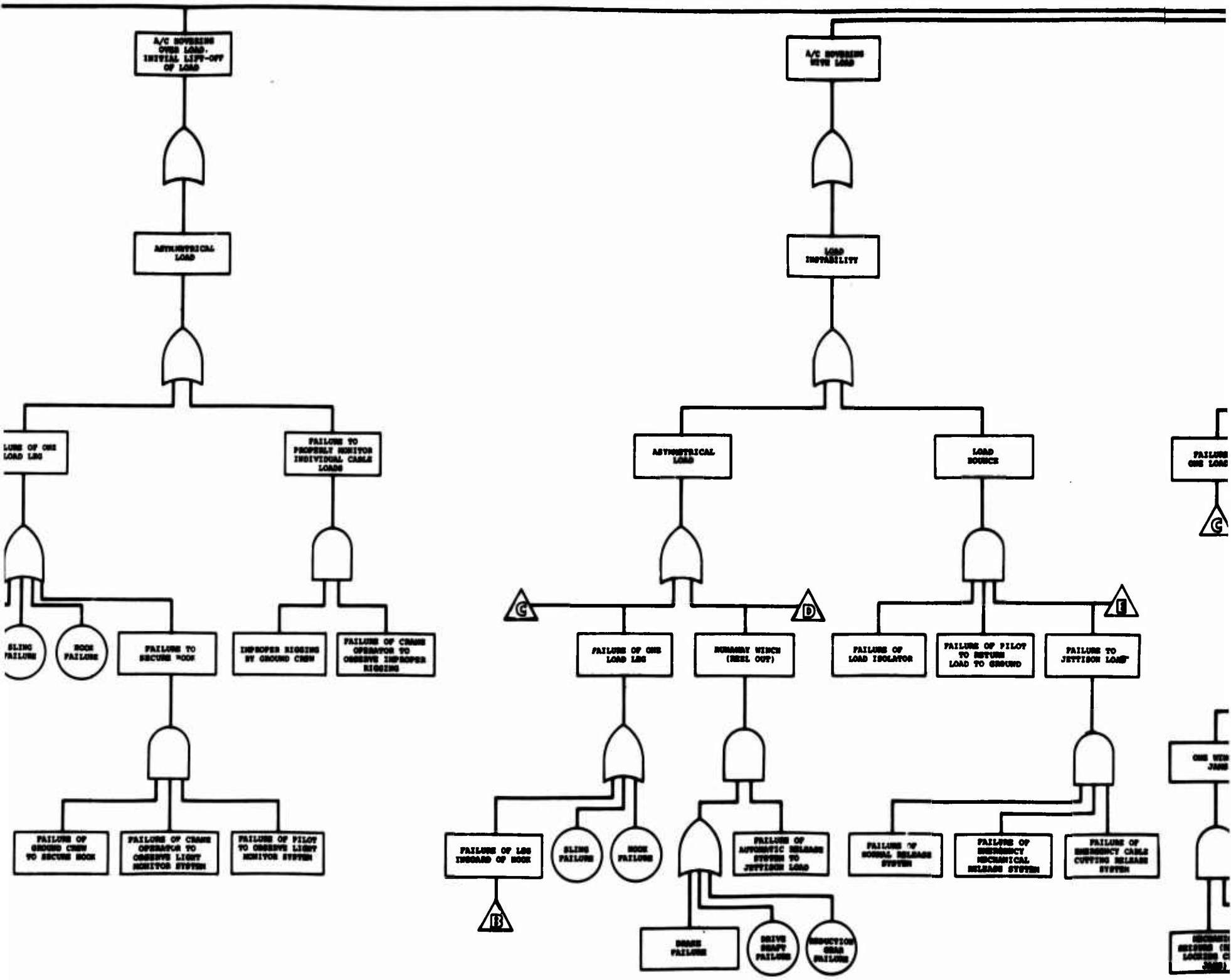
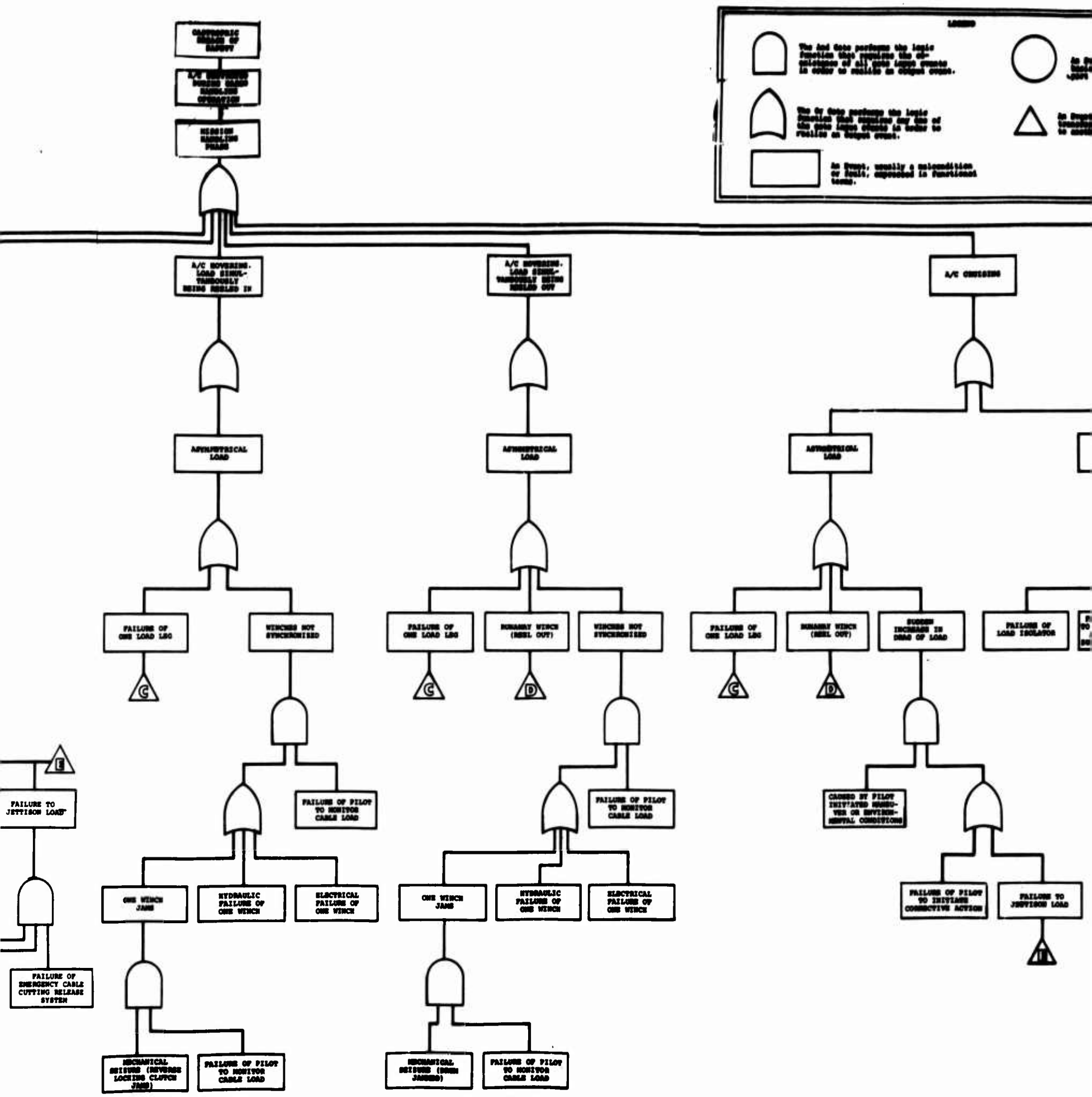
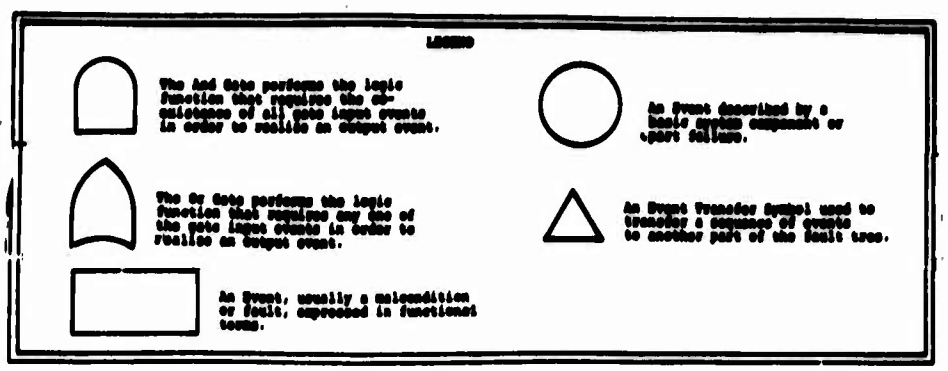


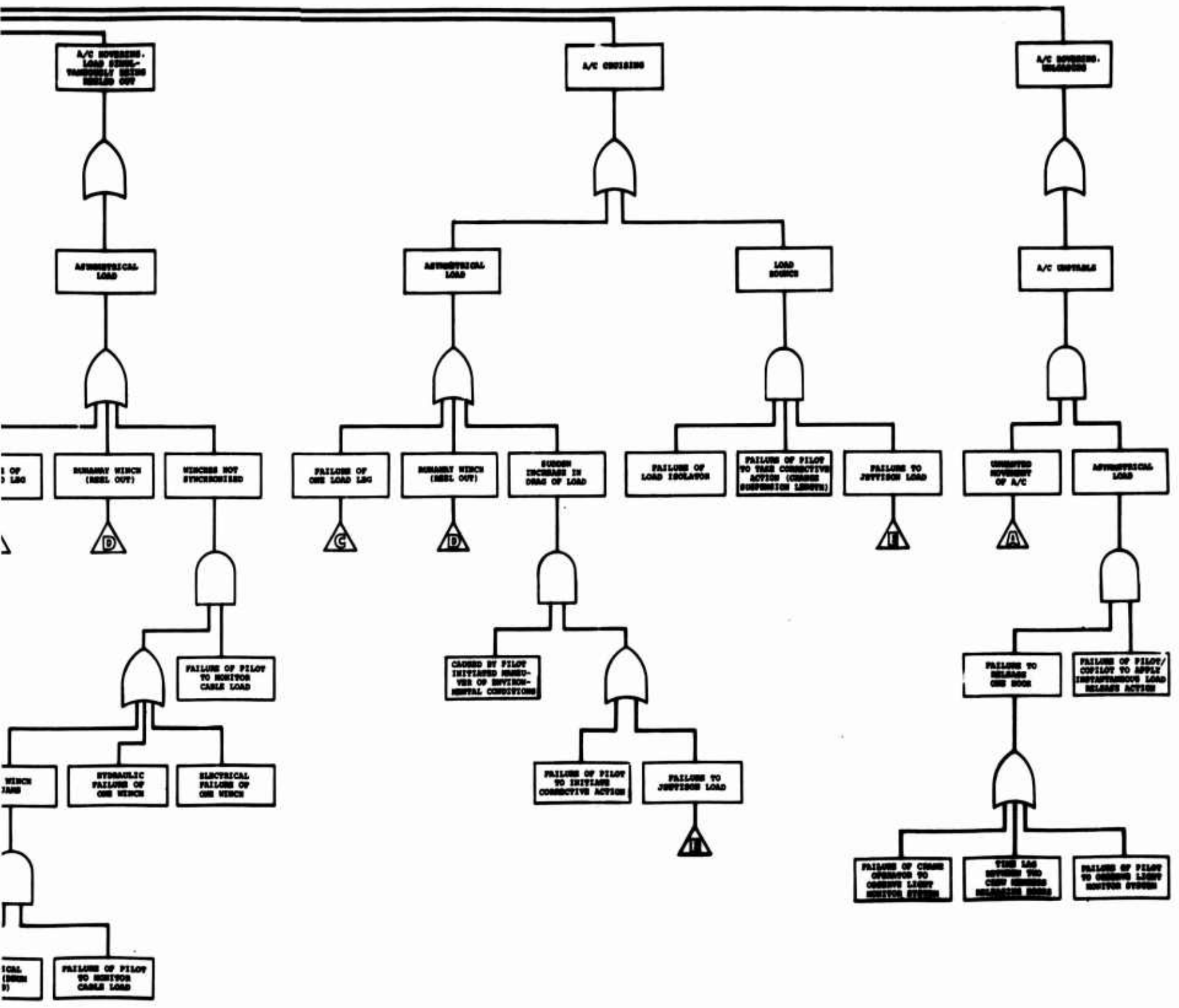
Figure 126. Fault Tree.


A/C HOVERING OVER LOAD. INITIAL LIFT-OFF OF LOAD
ASYMMETRICAL LOAD
LOAD INSTABILITY
A/C HOVERING WITH LOAD
FAILURE TO PROPERLY MONITOR INDIVIDUAL CABLE LOADS
IMPROPER RIGGING BY GROUND CREW
FAILURE OF CRANE OPERATOR TO OBSERVE IMPROPER RIGGING
FAILURE TO SECURE HOOK
FAILURE OF GROUND CREW TO SECURE HOOK
FAILURE OF CRANE OPERATOR TO OBSERVE LIGHT MONITOR SYSTEM
FAILURE OF PILOT TO OBSERVE LIGHT MONITOR SYSTEM
ASYMMETRICAL LOAD
LOAD BOUNCE
FAILURE OF ONE LOAD LEG
RUNAWAY WINCH (REEL OUT)
FAILURE OF LOAD ISOLATOR
FAILURE OF PILOT TO RETURN LOAD TO GROUND
FAILURE TO JETTISON LOAD
FAILURE OF LEG INBOARD OF HOOK
SLING FAILURE
HOOK FAILURE
FAILURE OF AUTOMATIC RELEASE SYSTEM TO JETTISON LOAD
BRAKE FAILURE
DRIVE SHAFT FAILURE
C
D
E
B


LEGEND
The And Gate performs the logic function that requires the co-existence of all gate input events in order to realize an output event.
The Or Gate performs the logic function that requires any one of the gate input events to occur to realize an output event.
An Event, usually a malcondition or fault; expressed in functional terms.
A/C HOVERING. LOAD SIMUL-TANEOUSLY BEING REELED IN
A/C HOVERING. LOAD SIMUL-TANEOUSLY BEING REELED OUT
A/C CRUISING
ASYMMETRICAL LOAD
ASYMMETRICAL LOAD
ASYMMETRICAL LOAD
FAILURE OF ONE LOAD LSO
WINCHES NOT SYNCHRONISED
FAILURE OF ONE LOAD LSO
RUNAWAY WINCH (REEL OUT)
WINCHES NOT SYNCHRONISED
FAILURE OF ONE LOAD LSO
RUNAWAY WINCH (REEL OUT)
SUDDEN INCREASE IN DRAG OF LOAD
FAILURE OF LOAD ISOLATOR
C
C
D
C
D
E
FAILURE TO JETTISON LOAD
FAILURE OF PILOT TO MONITOR CABLE LOAD
FAILURE OF PILOT TO MONITOR CABLE LOAD
CAUSED BY PILOT INITIATED MANOEU-VER OR ENVIRON-MENTAL CONDITIONS
ONE WINCH JAMS
HYDRAULIC FAILURE OF ONE WINCH
ELECTRICAL FAILURE OF ONE WINCH
ONE WINCH JAMS
HYDRAULIC FAILURE OF ONE WINCH
ELECTRICAL FAILURE OF ONE WINCH
FAILURE OF PILOT TO INITIATE CORRECTIVE ACTION
FAILURE TO JETTISON LOAD
FAILURE OF EMERGENCY CABLE CUTTING RELEASE SYSTEM
MECHANICAL SEIZURE (REVERSE LOCKING CLUTCH JAMS)
FAILURE OF PILOT TO MONITOR CABLE LOAD
MECHANICAL SEIZURE (DRUM JAMMED)
FAILURE OF PILOT TO MONITOR CABLE LOAD
I

3


LEGEND
The And Gate performs the logic function that requires the coexistence of all gate input events in order to realize an output event.
An Event described by a basic system component or part failure.
The Or Gate performs the logic function that requires any one of the gate input events in order to realize an output event.
An Event Transfer Symbol used to transfer a sequence of events to another part of the fault tree.
An Event, usually a malcondition or fault, expressed in functional terms.


A/C HOVERING. LOAD STORE. TANDEMLY SLUNG REELED OUT
A/C CRUISING
A/C HOVERING. UNLOADING
ASYMMETRICAL LOAD
LOAD SOURCE
A/C UNSTABLE
RUNAWAY WINCH (REEL OUT)
WINCHES NOT SYNCHRONISED
FAILURE OF ONE LOAD LEG
SUDDEN INCREASE IN DRAG OF LOAD
FAILURE OF LOAD ISOLATION
FAILURE OF PILOT TO TAKE CORRECTIVE ACTION (CHANGE SUSPENSION LENGTH)
FAILURE TO JETTISON LOAD
UNWANTED MOVEMENT OF A/C
FAILURE OF PILOT TO MONITOR CABLE LOAD
HYDRAULIC FAILURE OF ONE WINCH
ELECTRICAL FAILURE OF ONE WINCH
CAUSED BY PILOT INITIATED MANOEUVRE OR ENVIRONMENTAL CONDITIONS
FAILURE OF PILOT TO INITIATE CORRECTIVE ACTION
FAILURE TO RELEASE ONE HOOK
FAILURE OF PILOT/ COPILOT TO APPLY IMMEDIATE LOAD RELEASE ACTION
FAILURE OF CRANE OPERATOR TO OBSERVE LIGHT MONITOR SYSTEM
FAILURE OF PILOT TO OBSERVE LIGHT MONITOR SYSTEM
A
D
C
D
J
J

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5. For the output event "unwanted movement of A/C" to occur, either of input events "sudden gust" or "pilot initiated" must exist (this is indicated by the OR logic gate).

6. For the output event "pilot initiated" to occur, the coexistence of input events "failure of pilot to obtain 'go' signal from crane operator" and "failure of pilot to observe light monitor system" must exist.

7. Likewise, the same logic can be applied in order to determine the various events that could make the output event "asymmetrical load" occur.

Once having satisfied ourselves that this branch of the fault tree has been developed in sufficient depth, we can make a qualitative assessment of the overall safety aspects. A review of the branch shows that the redundancy of the events and the very nature of the events themselves are such that the likelihood of their occurrence in order to cause the top undesired event "destroyed A/C" is remote. On this basis, the safety considerations employed in the design of this phase of the system are considered to be adequate, and the system design is considered to be safe.

Safety Analysis Summary

A system safety review was made of the two-winch, two-point external cargo handling system by using the fault tree analysis technique. The top-most undesired event, i.e., the event which must be kept from happening, was defined as "Aircraft destroyed during cargo handling operation".

A review of the completed fault tree showed that the redundancy of the events and the very nature of the events (with the exception noted below) were such that the likelihood of their occurrence in order to cause the top undesired event, "destroyed A/C", was remote. The system was judged to be safe on this basis.

The exception noted is the failure of a hook and/or a sling. These are the only single component hardware items depicted on the fault tree whose failure could lead to the top undesired event. (The "hook failure" and "sling failure" events follow OR gates only in leading to the top undesired event, indicating that failure of either of these items alone could cause the top undesired event.) Alternatives that can be

followed to ensure safety features in this area are:

1. Follow conservative design practices and employ rigid quality control practices in the fabrication of the hook. The same applies to the sling, but adherence to more rigid quality control standards must be maintained in the field, on the basis that the sling is probably more susceptible to damage from field use.

2. Extend the automatic emergency release system from the beam to the load. This provides the redundant feature of automatically jettisoning the load in the event of a hook or sling failure.

DEVELOPMENT PLAN WITH RESPECT TO TIME FOR THE TWO-POINT HOIST SYSTEM DESCRIBED IN PHASE II

The following development plan estimates the time required to design, fabricate and bench-test the major components of the two-point hoist system. Additionally, the plan covers the assembly of the components into a complete hoist subsystem and the subsystem ground qualification. Further, it encompasses the modification of and installation into a prime mover for flight qualification, operational evaluation and service acceptance trials. For the purpose of this plan, it has been assumed that the prime mover will be a CH-47 Model C helicopter because of its availability within the time frame of this plan.

The development plan further assumes that development work required to resolve any design problems has either been completed or is running parallel with the detail design phase of this program. See Phase II-Development Schedule, page 284.

The development schedule does not indicate any parallel development programs which may be required to substantiate design features defined as problem areas (see Design and Development Problem Areas on page 216 of this report), nor does it show the development time required to design, fabricate and build a facility to ground qualify the hoist system.

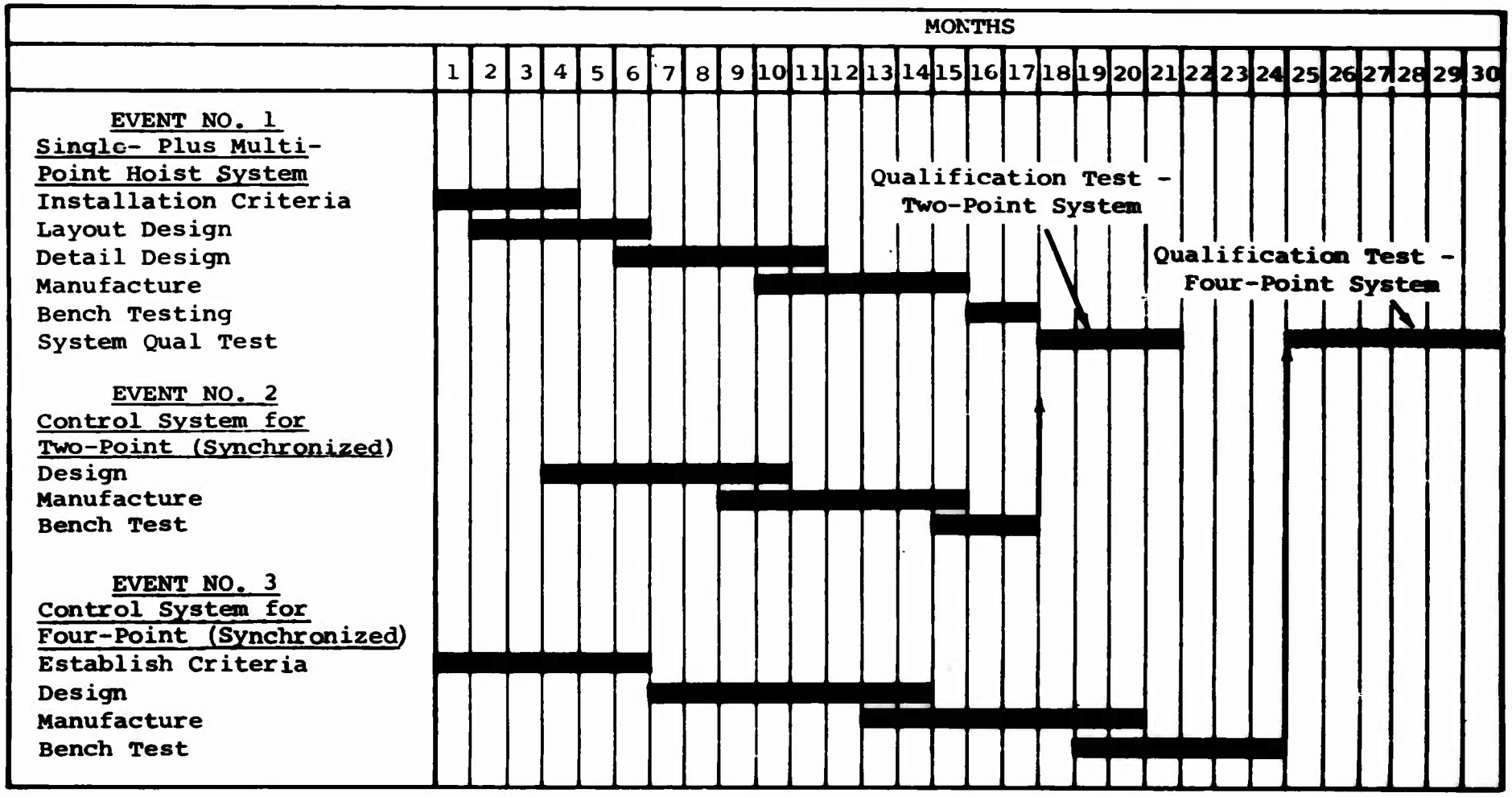


Figure 127. Phase II - Development Schedule With Respect to Time.

PHASE II - PRELIMINARY STRESS ANALYSIS

INTRODUCTION AND SUMMARY

This section presents the loads and stress analysis for the heavy lift helicopter external load handling systems.

The primary function of the external cargo system and its various structural components, such as the capstan drive system, capstan winches, upper suspension ropes, horizontal beam, and lower suspension ropes is to deliver a maximum 40,000-pound external cargo. Basically, the capstan drive system is the mechanism which synchronizes the operation, while the lower ropes, horizontal beam, cargo hooks, and upper ropes provide the means of sustaining and distributing the external loads to the drive system.

The hoisting system uses two capstan-type winches which are located in the fuselage of the heavy-lift helicopter. Each winch provides the mechanical power to hoist a 25,000-pound load through a series of drums and a drive system. The winch scheme incorporates secondary and loaded drums whose functions are to absorb the loading while the storage drum remains unloaded at all stages. A minimum preloading in the ropes wrapped around the storage drum does exist to eliminate rope slack in the system.

SUMMARY OF ULTIMATE MARGINS OF SAFETY

A tabular summary of the minimum margin of safety for each system component is presented below in Table XV . The minimum life of winch drive system gears is also presented.

TABLE XV

MINIMUM MARGIN OF SAFETY FOR EACH SYSTEM COMPONENT

Item	Type of Failure	Condi-tion	Page	Ult. M S
Rope				
Upper Rope	Tension	3B	294	.04
Lower Rope	Tension	3B	293	.12

TABLE XV Continued

Horizontal Beam Longitudinal Member	Bending	3A	306	.01
Capstan Winch Side P_L(Plate)	Bending	3B	313	.57
Secondary Drum	Compression	3B	326	1.80
Loaded Drum	Compression	3B	327	.35
Bellmouth Mountings			330	
Bellmouth D-D	Bending	3B	336	.26
1/2" Diameter Bolts	Tension	3B	340	.06
Winch Drive System First Stage Sun Gear (C)	Tension	1B	346	.33

Minimum Life of Winch Drive System

The second-stage planet gear "E" has a minimum life of 3880 operational duty cycles. The design objective is 3600 operational duty cycles.

STRESS RECOMMENDATIONS

Horizontal Beam

If the single-point mode is eliminated, the horizontal beam can be reduced to be a housing for the release mechanism. Where the lower rope is concentric with the upper rope the horizontal beam is unloaded.

Loading Spectrum

To realistically define the service life of the winch drive system and the winch beam support bearings, any follow-on effort must include a complete loading spectrum. For example, if the life of the system is 3600 duty cycles, what is the distribution of:

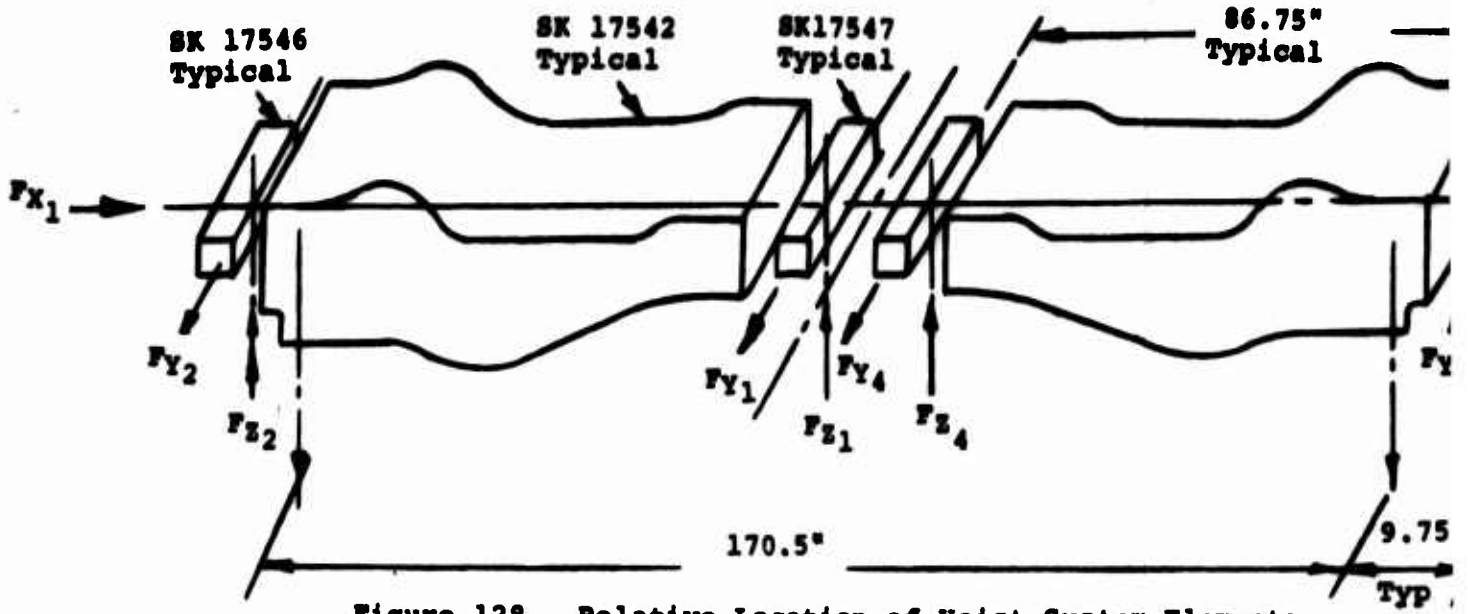
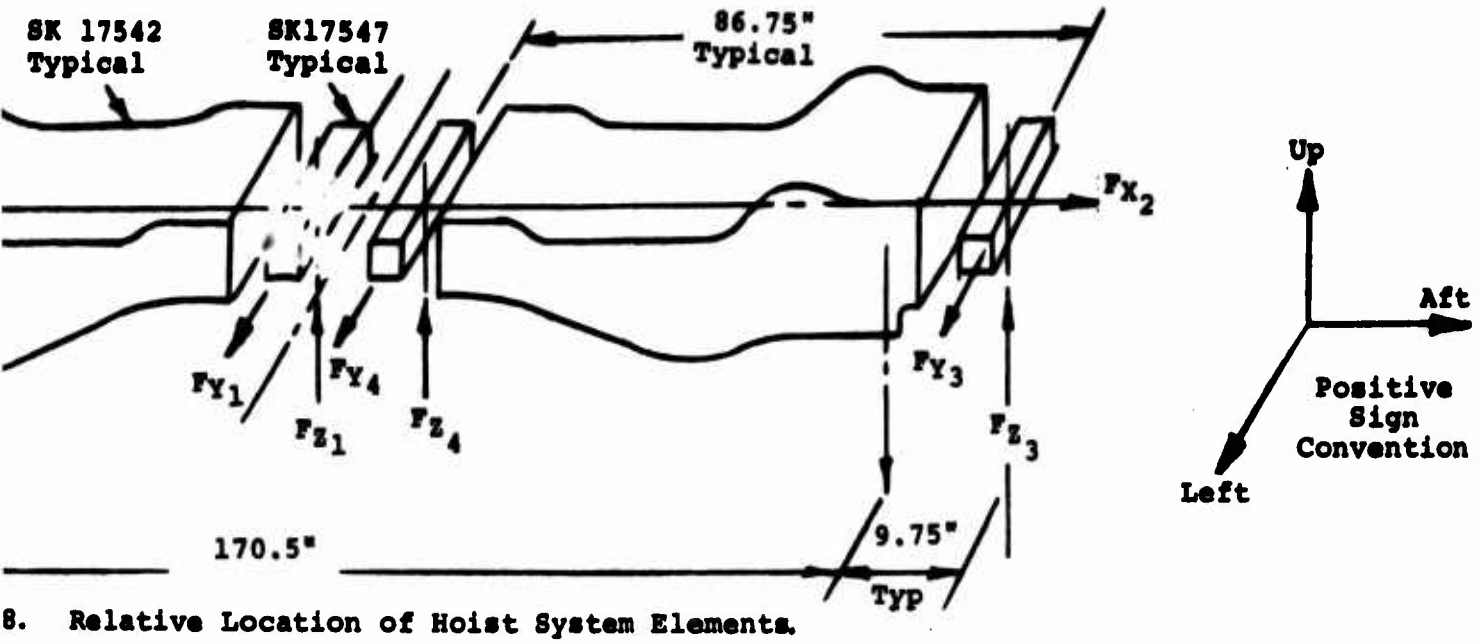


Figure 128. Relative Location of Hoist System Elements.

TABLE XVI
SUMMARY OF CRITICAL REACTIONS ON THE HELICOPTER

LOAD CONDITION	CARGO CG					CARGO SYSTEM R	
(For Load Definition 1A, 1B, etc., Figure 125.)		F_{z_1}	F_{z_2}	F_{z_3}	F_{z_4}	F_{y_1}	F_{y_2}
1A or 1B	Neutral	10.3	72.9	72.9	10.3	0	0
1B	18 in. Fwd	12.1	87.3	58.5	8.5	0	0
1B	18 in. Aft	8.5	58.5	87.3	12.1	0	0
3B	18 in. Fwd	10.7	76.2	51.2	7.5	0	0
(30° Aft Sway)	18 in. Aft	7.5	51.2	76.2	10.7	0	0
3B	18 in. Fwd	10.7	76.2	51.2	7.5	0	0
(30° Fwd Sway)	18 in. Aft	7.5	51.2	76.2	10.7	0	0
2B	18 in. Fwd	10.7	76.2	51.2	7.5	5.2	41
(30° Right Sway)	18 in. Aft	7.5	51.2	76.2	10.7	3.4	27
2B	18 in. Fwd	10.7	76.2	51.2	7.5	-5.2	-41
(30° Left Sway)	18 in. Aft	7.5	51.2	76.2	10.7	-3.4	-27



8. Relative Location of Hoist System Elements.

TABLE XVI
SUMMARY OF CRITICAL REACTIONS ON THE HELICOPTER

CARGO SYSTEM REACTIONS (KIPS)									
F_{z_1}	F_{z_2}	F_{z_3}	F_{z_4}	F_{y_1}	F_{y_2}	F_{y_3}	F_{y_4}	F_{x_1}	F_{x_2}
10.3	72.9	72.9	10.3	0	0	0	0	0	0
12.1	87.3	58.5	8.5	0	0	0	0	0	0
8.5	58.5	87.3	12.1	0	0	0	0	0	0
10.7	76.2	51.2	7.5	0	0	0	0	-46.8	-30.6
7.5	51.2	76.2	10.7	0	0	0	0	-30.6	-46.8
10.7	76.2	51.2	7.5	0	0	0	0	46.8	30.6
7.5	51.2	76.2	10.7	0	0	0	0	30.6	46.8
10.7	76.2	51.2	7.5	5.2	41.6	27.2	3.4	0	0
7.5	51.2	76.2	10.7	3.4	27.2	41.6	5.2	0	0
10.7	76.2	51.2	7.5	-5.2	-41.6	-27.2	-3.4	0	0
7.5	51.2	76.2	10.7	-3.4	-27.2	-41.6	-5.2	0	0

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ULTIMATE LOADING SPECTRUM

1. Loads are in kips. 2. Reactio
system weight at indicated load

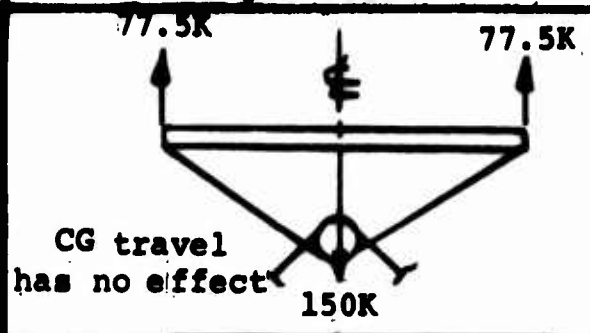
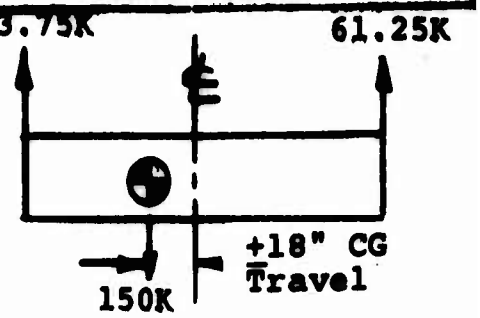
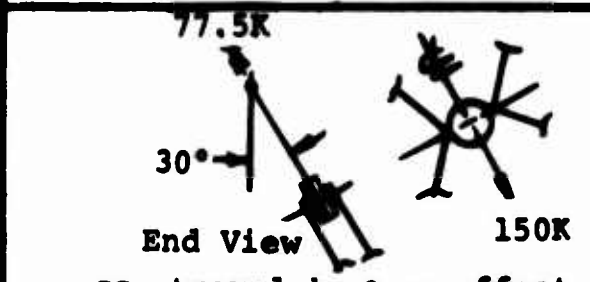
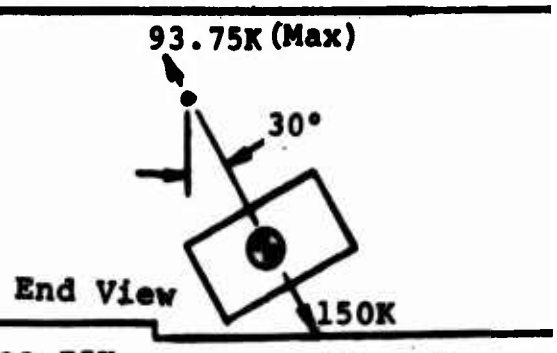
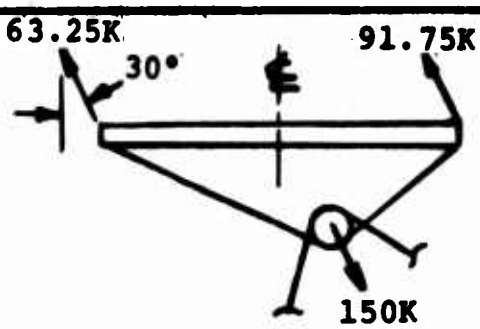
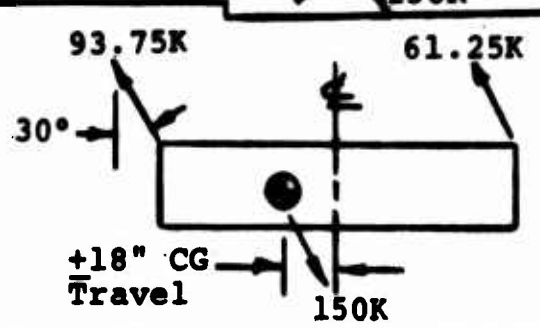
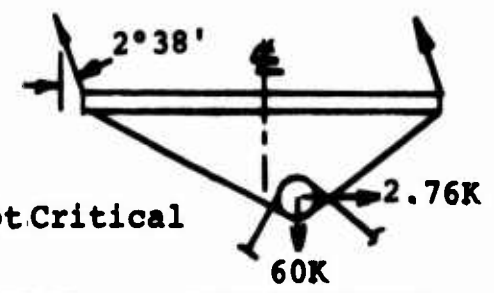
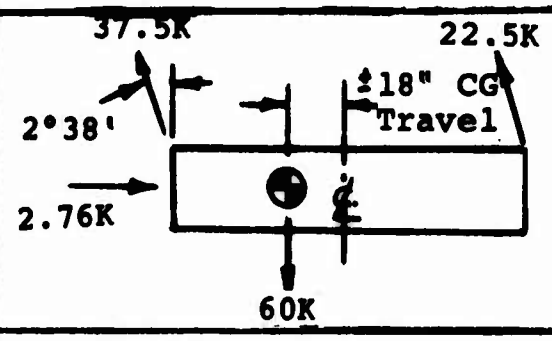
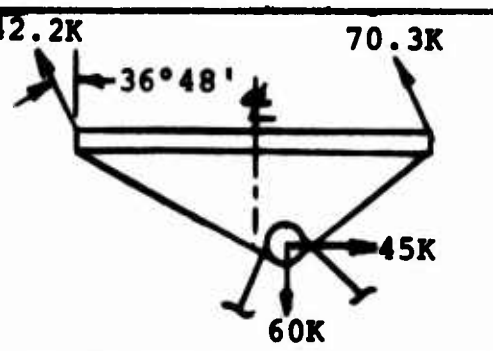
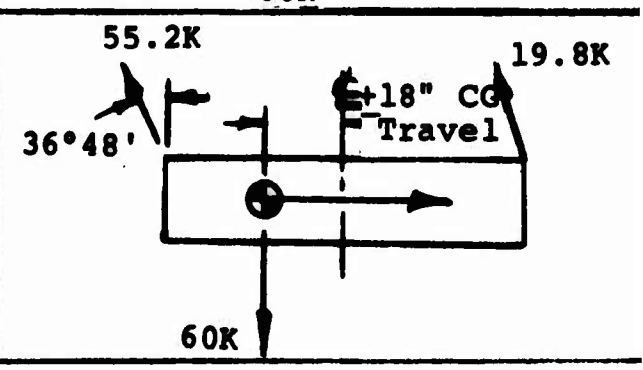
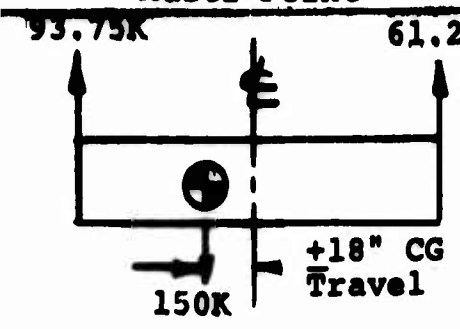
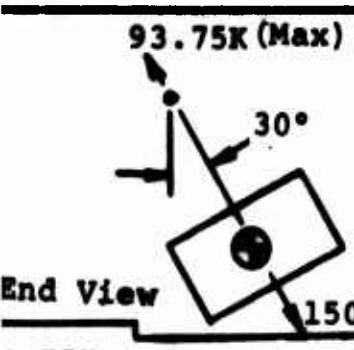
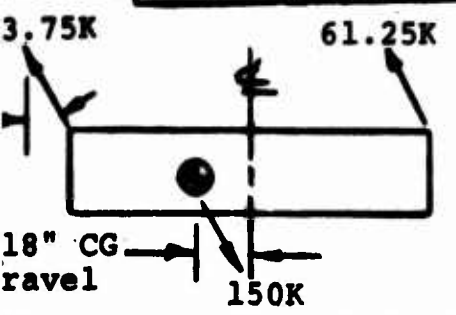
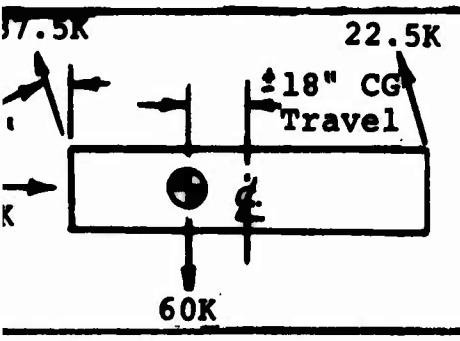
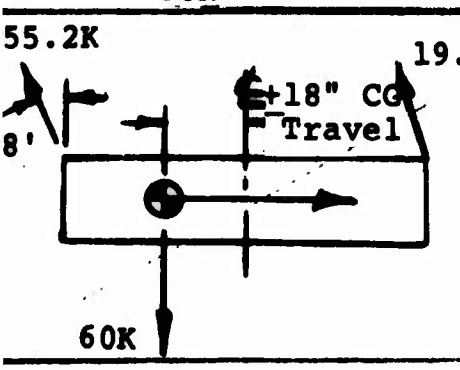
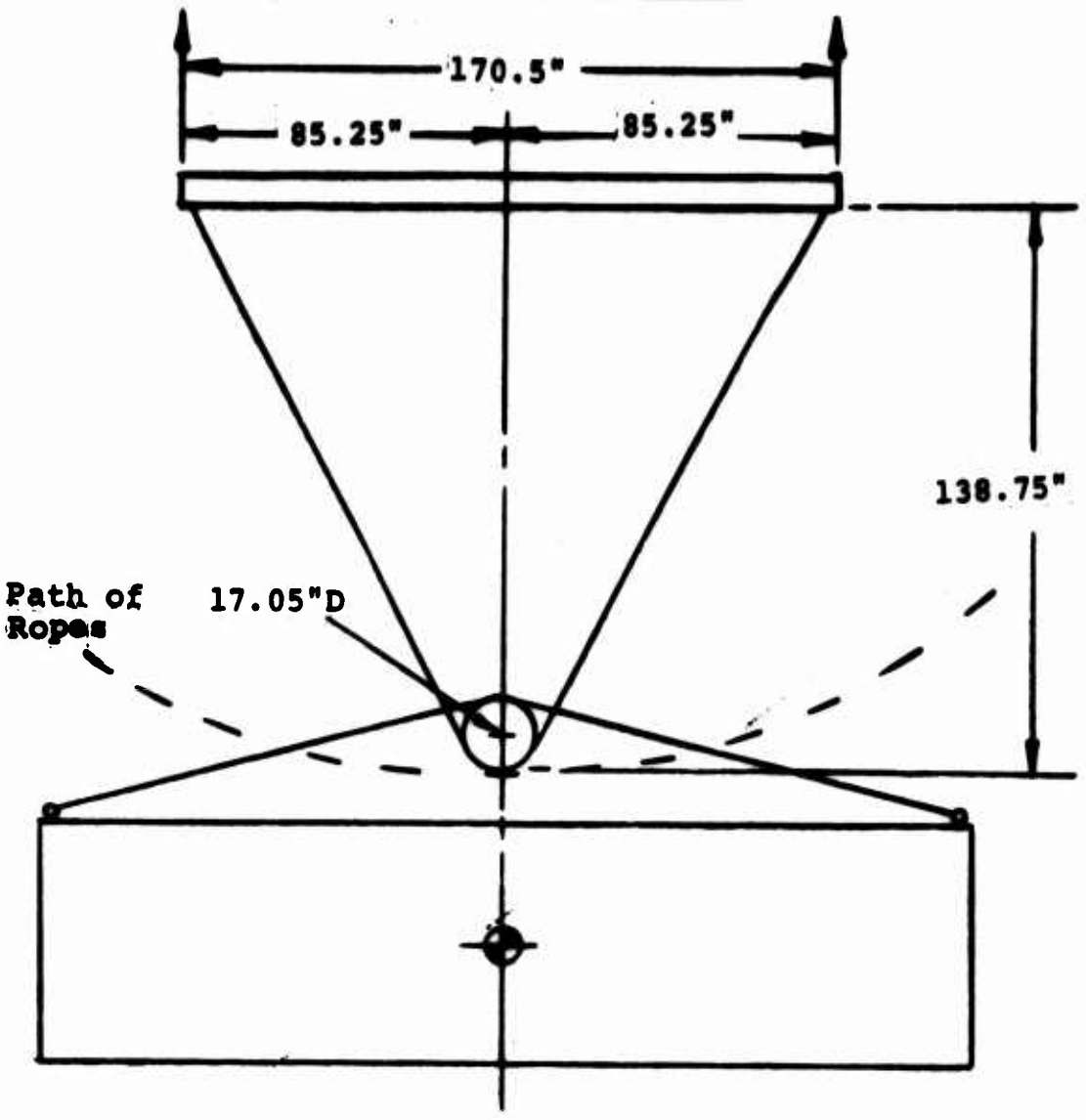
Loading Condition	Suspension Mode			
	Single Point	Cond	Multi-Point	
Vertical load 3.75 ult design factor		1A		
30° Lateral sway 3.75 ult design factor		2A		
30° Longitudinal sway 3.75 ult design factor		3A		
1.5g plus aerodynamic drag (ult) 8' x 8' container at 100 Knots		4A		
Max. ult long. acceleration 1.0g down 0.75g aft (F.S. 1.5)		5A		

Figure 129. Ultimate Loading Sp

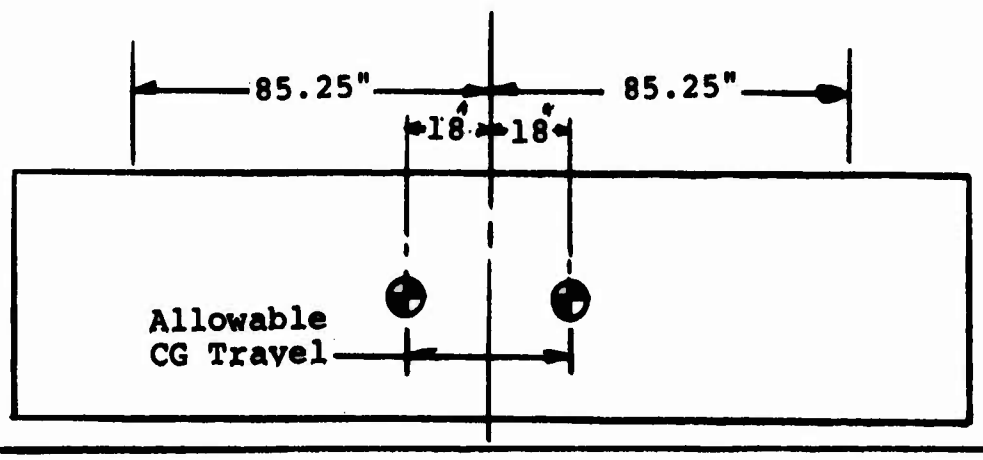
s are in kips. 2. Reactions include payload and
sm weight at indicated load factor at capstan bellmouth.

Mode	Cond
Multi-Point	
	1B
	2B
	3B
	4B
	5B

Single-point lift system



Multi-point lift system



129. Ultimate Loading Spectrum.

1. payload gross weight,

2. cg position,

3. rope extension and retraction.

(For loading computations, reference Figure 128 and Table XVI.)

DISCUSSION OF ULTIMATE LOADING CONDITIONS (LOAD ANALYSIS)

The ultimate loading condition is based on the lift capacity of 40,000 pounds with a limit load factor of 2.5 and an ultimate load factor of 1.5 (ultimate design load factor of 3.75). This ultimate design load factor is applied to the weight of the cargo and is applied 30 degrees from the vertical in any direction. Ten loading conditions are investigated as shown in Figure 129.

The external cargo handling system carries the cargo by either the single-point suspension system or the two-point suspension system.

The single-point suspension system features a floating pulley to align the center of gravity of the cargo with the center line of the horizontal beam (for down loads only). This alignment of the cargo cg will produce equal winch rope loads. For longitudinal loads, the floating pulley will move in an elliptical path such as to align the resultant pulley loads with the resultant cargo loads (vector sum of the vertical and longitudinal loads). The elliptical pulley path is defined by the length of the pulley rope . Pendulum action of the suspension system will react lateral sway loads.

The two-point suspension system utilizes the direct attachment of the winch ropes to the cargo. In this system, pendulum action will support both longitudinal and lateral sway loads. To account for a possible asymmetrical location of the cargo cg, a 18.00-inch allowable cg travel in either direction (from center line of the two winch ropes) is featured in this system. The multi-pointsuspension conditions 1B, 2B, and 3B (refer to load conditions, page 287) and single-point condition 5A yield the maximum winch rope load of 25,000 pounds. The multi-point suspension has the advantage of providing the desired pitch and yaw restraint and has the simplest configuration.

Winch Drive Endurance Loading Condition

The endurance loading condition consists of hoisting a 40,000-pound cargo (at 1.0g) with extreme cg travel in the multi-point suspension mode. The winch rope endurance load is 25,000 pounds; this winch cable load is used in the definition of the operational duty cycle. A duty cycle consists of lifting and lowering a 40,000-pound item 60 feet (60 ft/min rope speed). The life of the gears in the winch drive system is calculated using this endurance load. See Figure 130 below.

Aerodynamic Load Analysis for Cargo

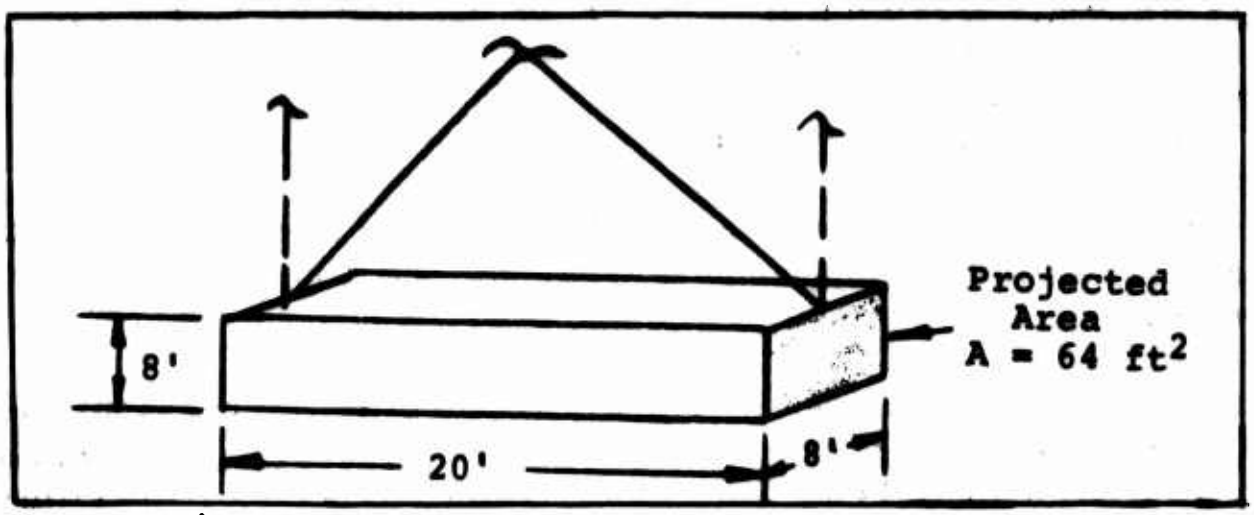


Figure 130. Load Analysis for Cargo.

$$\text{Drag} = q \times S \times C_d$$

$q = 1/2\,\rho V^2$ (dynamic pressure)

V = 100 knots or 115 miles per hour = 169 feet per second

$\rho = 0.002378$ lb-sec^2/ft^4

$C_d = 0.80$ to 1.25

$$q = 0.001189 \frac{(\text{lb-sec}^2)}{\text{ft}^4} \times (169 \text{ ft/sec})^2 = 28.6 \text{ lb/ft}^2$$

$$\text{Drag} = (28.6 \text{ lb/ft}^2)\ (64 \text{ ft}^2)$$

$D = 1840$ lb

$D_{ULT} = 2760$ lb

CG LIMIT: 2-Point Lift

Vertical Condition

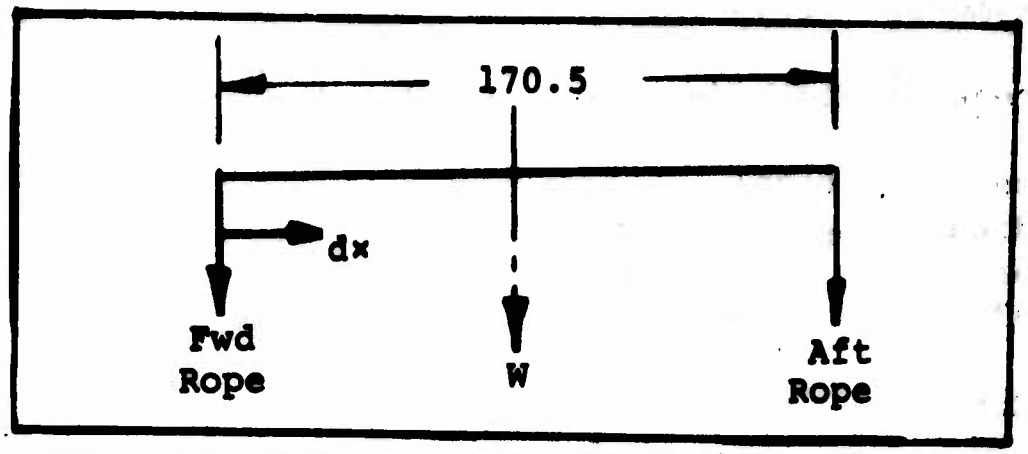


Figure 131. Two-Point System, Two-Point Lift (CG Restrictions).

Compute permissible cg shift:
Boundary conditions at aft rope 91.25^K

$$W = 150^K$$

$$X = \frac{91.25(170.5)}{150} = 103.72$$

cg shift = 103.72 - 170.5/2 = 18.47 in.

Use 18.0 in. cg travel.

STRESS ANALYSIS OF COMPONENTS

Rope

Nonstandard rope must be procured if a maximum diameter of 1 inch is to be maintained. The following analysis shows that a rope procured to a minimum breaking strength of 103,000 pounds will have a margin of safety of +0.04.

Two-Point System With Multi-Point Lift

Lower Rope

Load = 91.080 Kips
Rope weight (13.3 x 3.75 x 1.87) = 0.093

Hook weight (20 x 3.75) = 0.075

Total design load at beam = 91.25^{K}

Rope diameter = 1.00 in.

Rope break strength (special rope) = 103^{K}

Condition 3B critical:
Longitudinal sway = -300
Rope Load = 91.25^{K}
MS = (103/91.25) -1 = 0.12

Upper Rope

Load/capstan at critical cg	=	91.25^{K}
6% bending factor	=	5.48
Rope weight (150 x 1.87 x 3.75)	=	1.05
Hook weight (40 x 3.75)	=	0.15
Half beam assembly weight	=	1.20
Body 2 cable (13.3 x 3.75 x 2.4)	=	0.093
Total design load	=	99.23^{K}

Lift height = 150 ft
Rope diameter = 1.00 in.
Rope break strength (special rope) = 103^{KIP}

Condition 3B critical for ropes :
Longitudinal sway = -30°
Maximum load without 6% factor = 93.75^{K}
MS = (103/99.23) -1 = 0.04

Rope Analysis (Condition 1B)

DEFLECTION ANALYSIS

If special rope (1-in.-dia, 6 x 37 IWRS) is obtained, the breaking strength of the rope will be rated at 103^{K}; thus, both ropes (upper and lower) will be constructed from 1-inch-diameter stock.

$A = 0.404\ in.^{2}$ (1 rope)

$$\delta = PL/AE = \frac{99,230 \times 150 \times 12}{11 \times 10^{6}\ (0.404)}$$

δ = 40.0 in. (3.33 ft) upper rope

$K = P/\delta$ = 2480 lb/in.

$$= \frac{91,250 \times 160}{11 \times 10^6 \ (0.404)}$$

= 3.26 in. lower rope

K = 91,250/3.26 = 27,800 lb/in.

Total possible vertical displacement:

δ = 0.866 x 3.26 + 40 = 42.82 in. (At ultimate load)

<u>TENSION IN UNIFORMLY LOADED ROPE DUE TO DRAG</u>

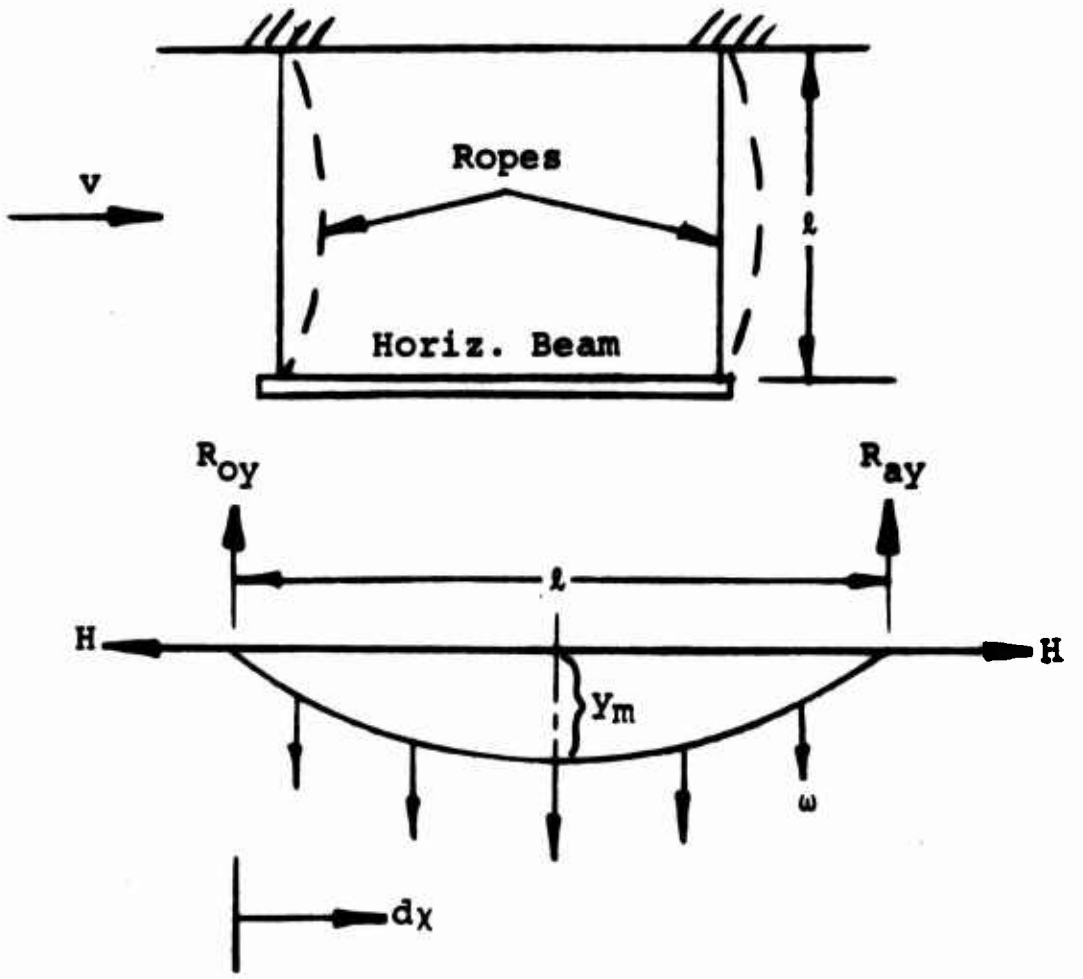


Figure 132. Tension in Uniformly Loaded Rope.

Assume that the cables are pinned at both ends and determine tension in the system:

When

ℓ = 30 ft
V = 100 kt Case 1
C_d = 1.20
Factor = 3.75
ℓ = 150 ft
V = 40 kt Case 2
C_d = 1.20
Factor = 2.5

Cable Tension

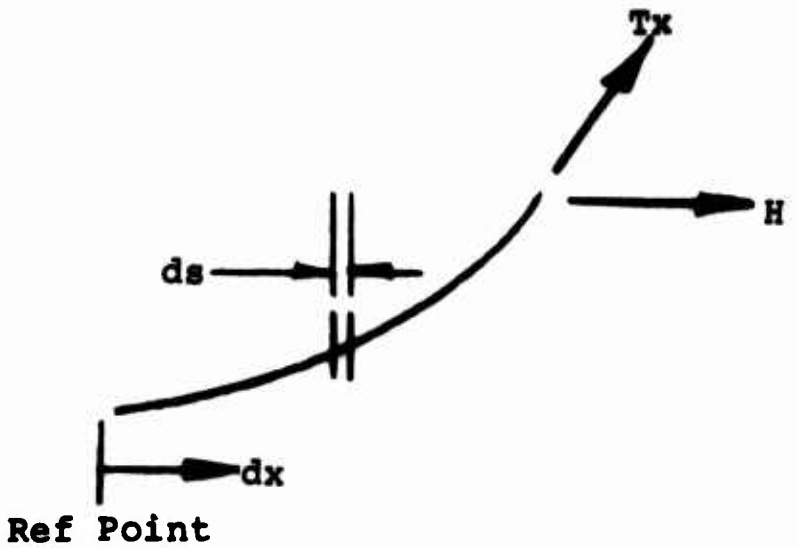


Figure 133. Sketch Showing Tension.

Basic Equations (see Reference 18)

$$HY_m = w\,x/2 - wx^2/2$$

$$T_{MAX} = H(1 + 16\theta^2)^{1/2}$$

$$\theta = h/\ell$$

$$T_x = H(ds/dx)$$

When

$x = L/2$
$Y_m = h$
$H = w\ell^2/8h$
$h = w\ell^2/8H$
$\theta = h/\ell = w\ell/8H$ sag ratio

$$T_{MAX} = w\ell^2/8h\left[1 + 16\left(\frac{w\ell^2}{8H}\right)\right]^{1/2}$$

ROPE AERODYNAMIC LOADS

Case 1

$q = 1/2\rho V^2$
$V = 100 \times 1.69 = 169$ ft/sec
$\rho = 0.002378$ lb-sec^2/ft^4
$q = 0.00119(169)^2$
$q = 34.20$ lb/ft^2
$q = 0.238$ lb/in.^{2}
$q = 0.238 \times 1.2 \times 3.75 = 1.06$ lb/in.^{2}
projected area $= 1$ in.
$w = 1.06$ lb/in.

Case 2

$V = 40 \times 1.69 = 67.5$ ft/sec
$q = 0.00119\ (67.5)$
$q = 5.4$ lb/ft^2
$q = 0.038$ lb/in.^{2}
$w = 0.038 \times 1 \times 1.2 \times 2.5$
$w = 0.115$ lb/in.

Rope Tension (Condition 1B)

Case 1

$\ell = 30$ ft (360 in.)
$H = 93.8^K$
$w = 1.06$ lb/in.
$V = 100$ kt

$$h = \frac{w\ell^2}{8H} \quad \frac{1.06\ (360)}{8\ (78,200)} = 0.222 \text{ in.}$$

$$\theta = \frac{0.222}{360} = 620 \times 10^{-6}$$

$$T_{MAX} = 93.8 \left[1 + 16 \times (620 \times 10^{-6})^2\right]^{\frac{1}{2}}$$

$T_{MAX} = 93.8^K$ (no change).

Case 2

$\ell = 150$ ft (1800 in.)
$H = 93.8^K$
$w = 0.115$ lb/in.
$V = 40$ kt

$$h = \frac{0.115\ (1800)}{8\ (78,200)} = 0.590 \text{ in.}$$

$$T_{MAX} = 93.8 \left[1 + 16 \times (328 \times 10^{-6})^2\right]^{\frac{1}{2}}$$

$T_{MAX} = 93.8^K$

The preceding analysis indicates that the aerodynamic drag on the ropes is negligible and can therefore be disregarded.

HORIZONTAL BEAM ASSEMBLY (Reference Figures 134 and 135)

The conventional method of stress analysis was utilized for this component. Condition 3A (Figure 128, page 287) was critical for the horizontal beam and dictated the use of 4 extruded angles (1.75 in. x 2 in. x 0.156 in.) and the 0.072-inch thickness for the skin frame. All parts are constructed from 7075-T6 aluminum.

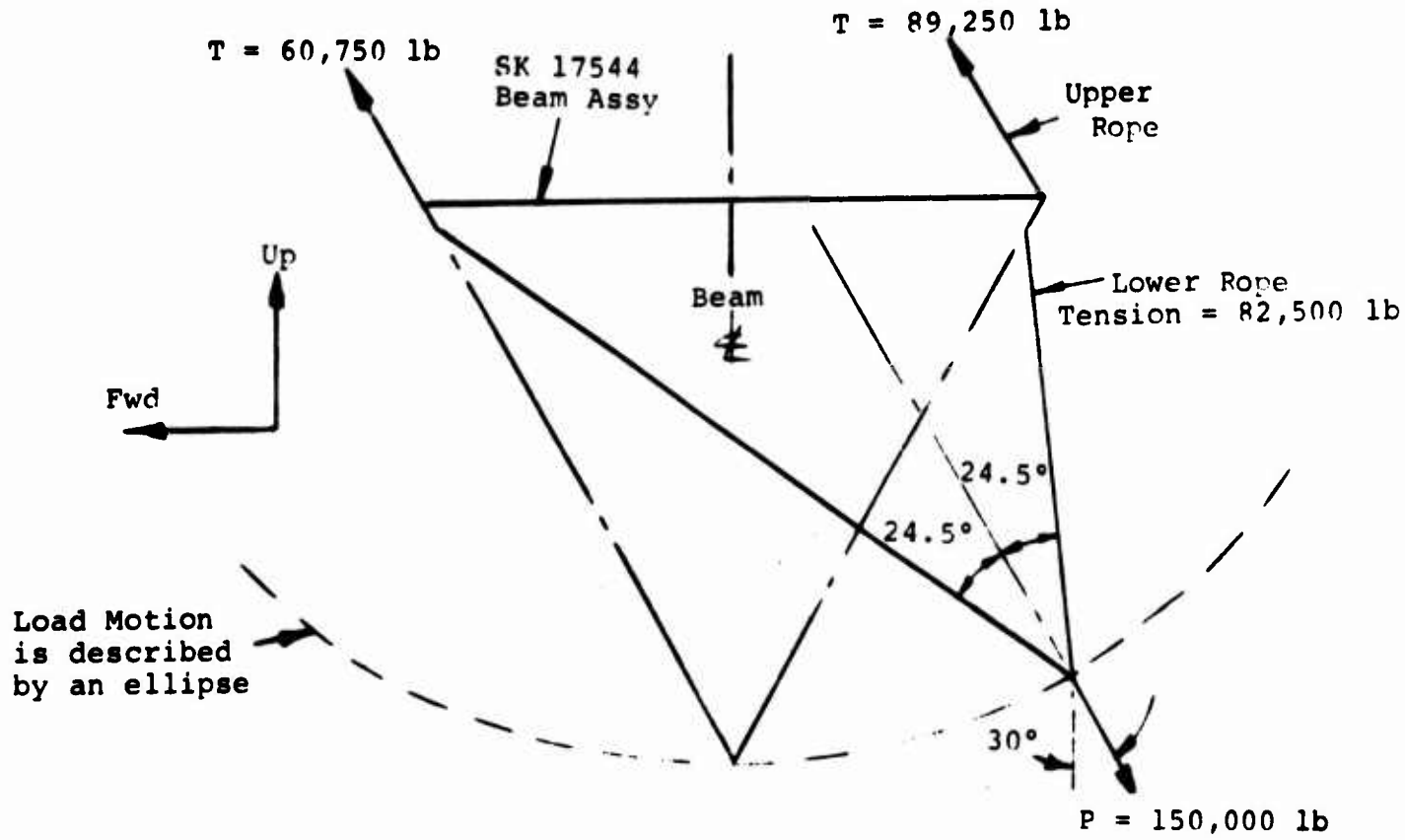


Figure 134. Horizontal Beam, Schematic Diagram.

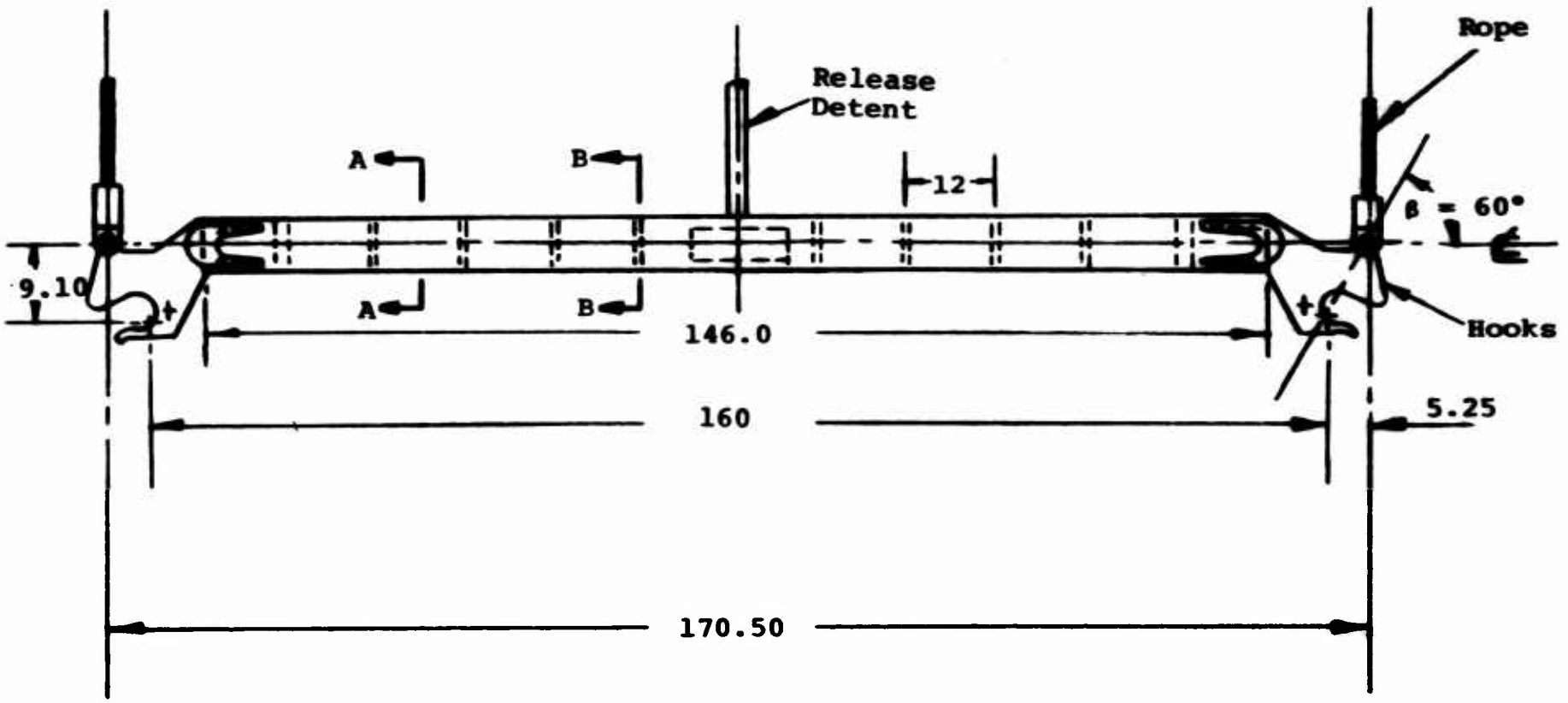
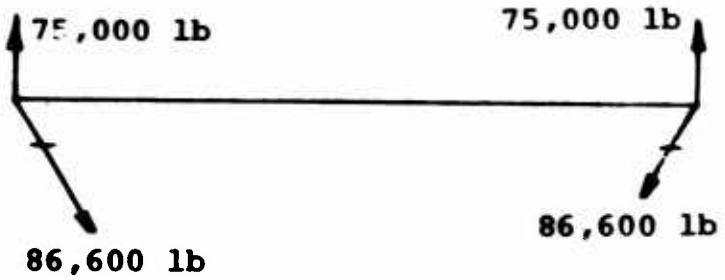


Figure 135. Single-Point Loading - 30 Degrees Longitudinal Sway.

CRITICAL LOADS - SK 17544 BEAM ASSEMBLY

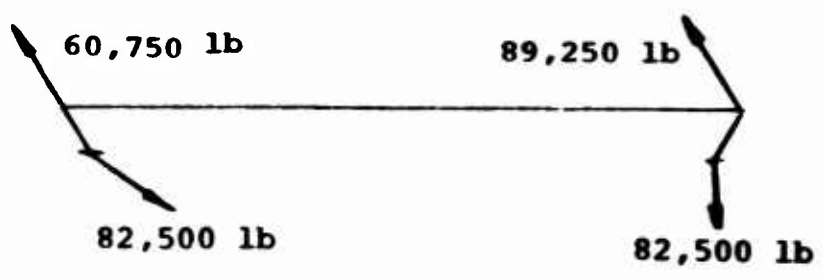
Condition 1A Single-Point loading symmetrical p ck-up at 3.75q (ult)



This case imposes maximum axial load on the beam where:

F_x = -43,300 lb (Comp)
F_z = M = 0

Condition 3A Single-Point loading - 30° Longitudinal sway at 3.75g (ult)



Eccentric loads are introduced to the beam due to the offset of the hook attachment of the lower cables.

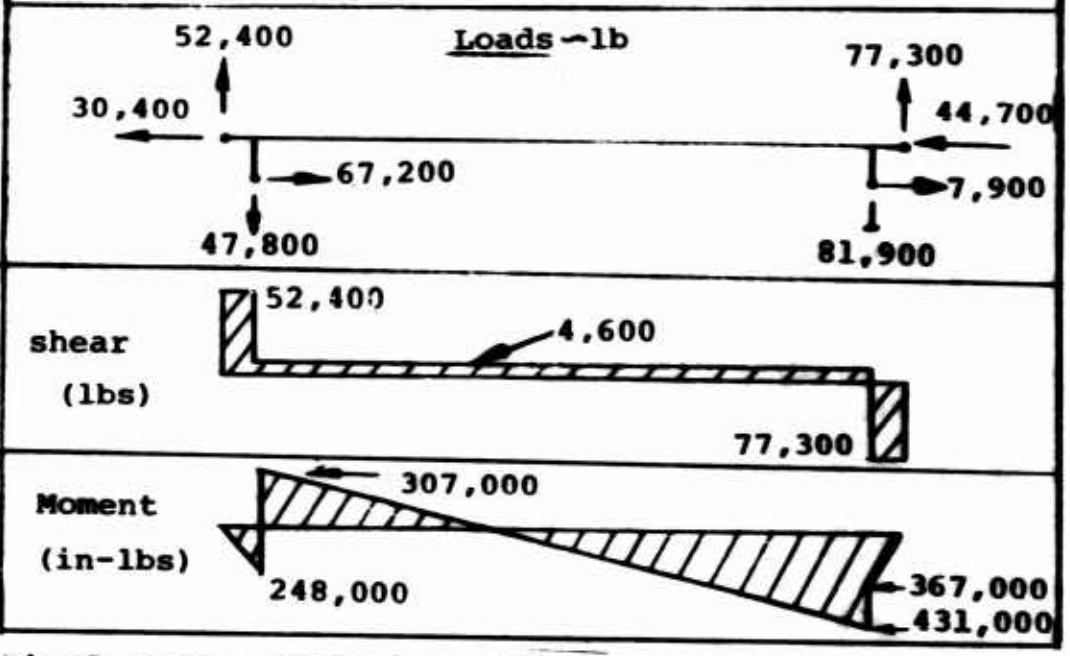


Figure 136. Critical Loads, Beam Assembly.

The beam assembly is composed of 3 major elements: 2 hook fittings and the center or box element joining the hooks.

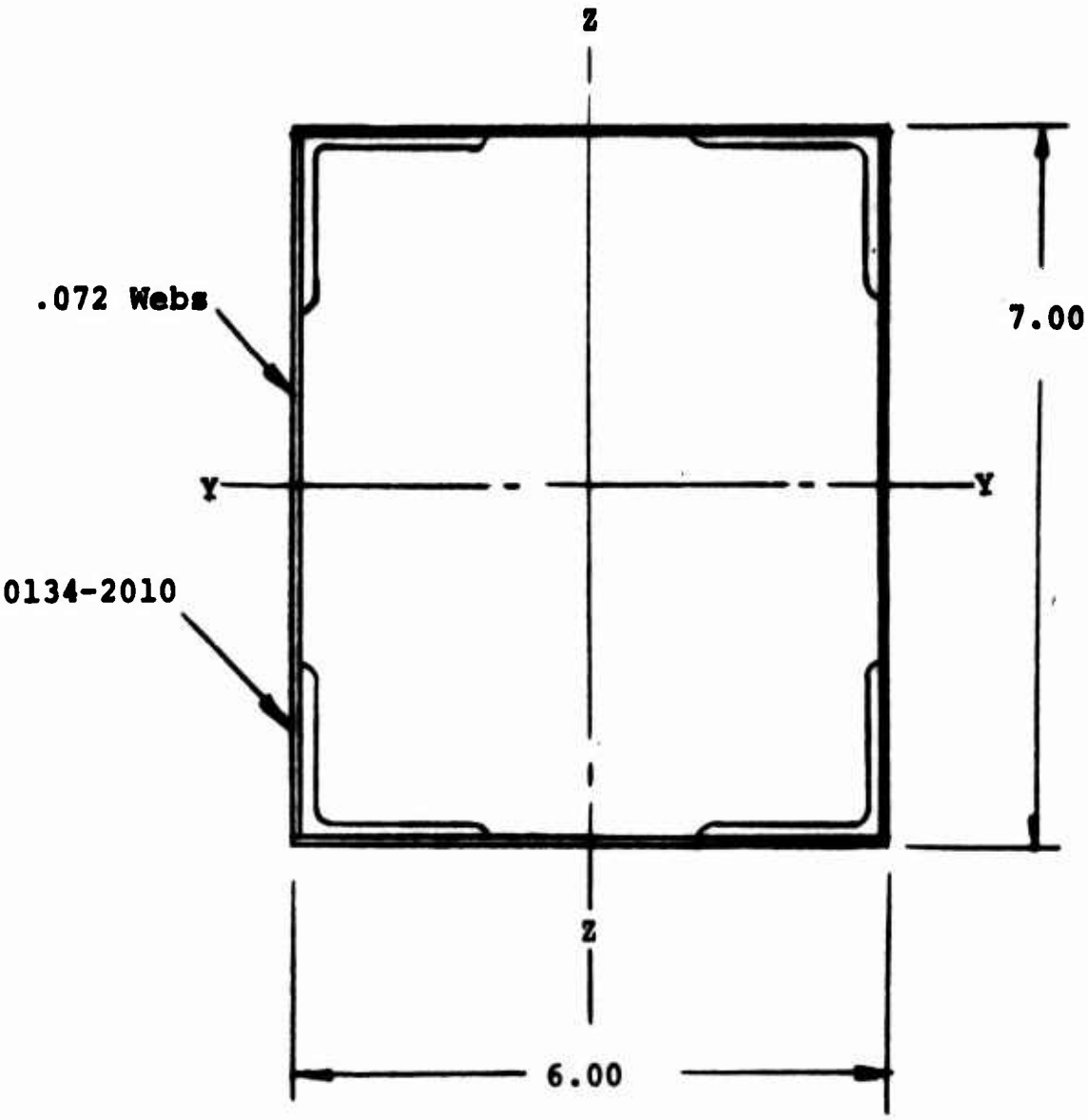


Figure 137. Cross-Sectional Schematic of A-A Section of Figure 135.

Total Area	= 4.11 in.^{2}
Effective area in compression (corner angles + 2W skin)	= 3.128 in.^{2}
I_{yy}	= 29.06 in.^{4}
I_{zz}	= 19.35 in.^{4}
ρ_y	= 3.05 in.
ρ_x	= 2.49 in.

Beam Assembly Section Properties

Since the beam is critical
for combined axial load and
bending on the compression side,
only effective skin is included
in section properties. See Reference 17.

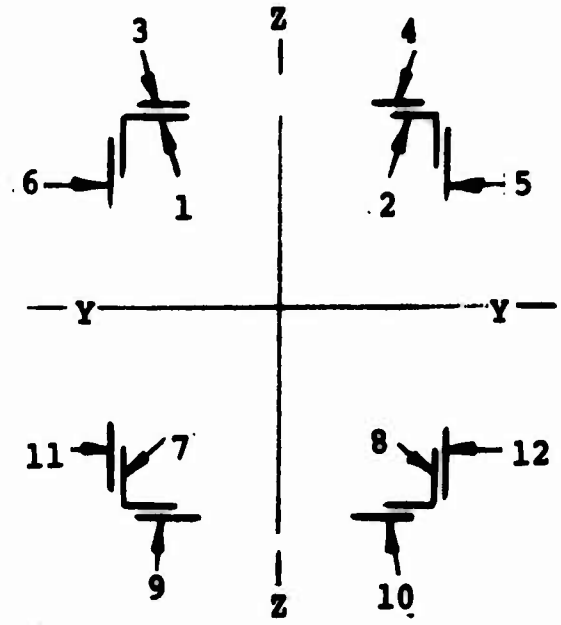


Figure 138. Beam Assembly Section Properties.

TABLE XVII
BEAM SECTION PROPERTIES

Element	Area	y	A_y^2	I_{o_y}	z	A_z^2	I_{o_z}
1	0.558	2.974	4.94	0.151	2.351	3.08	0.212
2	0.558	2.974	4.94	0.151	2.351	3.08	0.212
3	0.112	3.46	1.34	0	2.20	0.54	0.025
4	0.112	3.46	1.34	0	2.20	0.54	0.025
5	0.112	2.70	0.81	0.025	2.964	0.98	0
6	0.112	2.70	0.81	0.025	2.964	0.98	0
7	0.558	2.974	4.94	0.151	2.351	3.08	0.212
8	0.558	2.974	4.94	0.151	2.351	3.08	0.212
9	0.112	3.46	1.34	0	2.20	0.54	0.025
10	0.112	3.46	1.34	0	2.20	0.54	0.025
11	0.112	2.70	0.81	0.025	2.964	0.98	0
12	0.112	2.70	0.81	0.025	2.964	0.98	0
Totals	3.128	---	28.36	0.704	---	18.40	0.948

$I_{yy} = 28.36 + 0.704 = 29.06$ in.^{4}
$I_{zz} = 18.40 + 0.948 = 19.35$ in.^{4}

The box assembly or center section is critical for:

1. Condition 1A: Maximum axial compression
2. Condition 3A: Combined bending and compression

Condition 1A

$F_x = 43,300$ lb (comp)

$P/A_{EFF} = \frac{43,300}{3.128} = 13,800$ psi

$\ell/\rho_{MIN} = \frac{170.5}{2.49} = 68.5$

$F_c = 25,000$ psi

$MS = \frac{25,000}{16,000} - 1 = 0.56$

Condition 3A

$F_x = 36,800$ lb (comp)

$$M_{MAX} = 431{,}000 \text{ in.-lb}$$

The fork legs of the beam end fitting are effective in carrying moment. The box, then, is critical at the end of the fitting, where:

$$M = 0.82\ M_{MAX} = 353{,}000 \text{ in.-lb}$$

$$\frac{P}{A} + \frac{M_y}{I_y} = \frac{36{,}800}{3.128} + \frac{353{,}000\ (3.5)}{29.06} = 54{,}200 \text{ psi}$$

$$F_{cc} = 55{,}000 \text{ psi}$$

$$MS = \frac{55{,}000}{54{,}200} - 1 = 0.01$$

<u>End Beam Connection (Condition 3A)</u> (Reference Figure 139)

Load/Corner

$$M = 0.80 \times 431 = 344^{K} \text{ in.}$$
$$P = 36.8^{K}$$

$$\text{Load/in.} = q = \frac{1}{8}\ \frac{344}{5} + \frac{36.8}{4}\quad q = 9.75 \text{ K/in.}$$

Pitch at 1-1/8

$$\text{Load/Bolt} = 1.125 \times 9.75 = 12.20^{K}$$

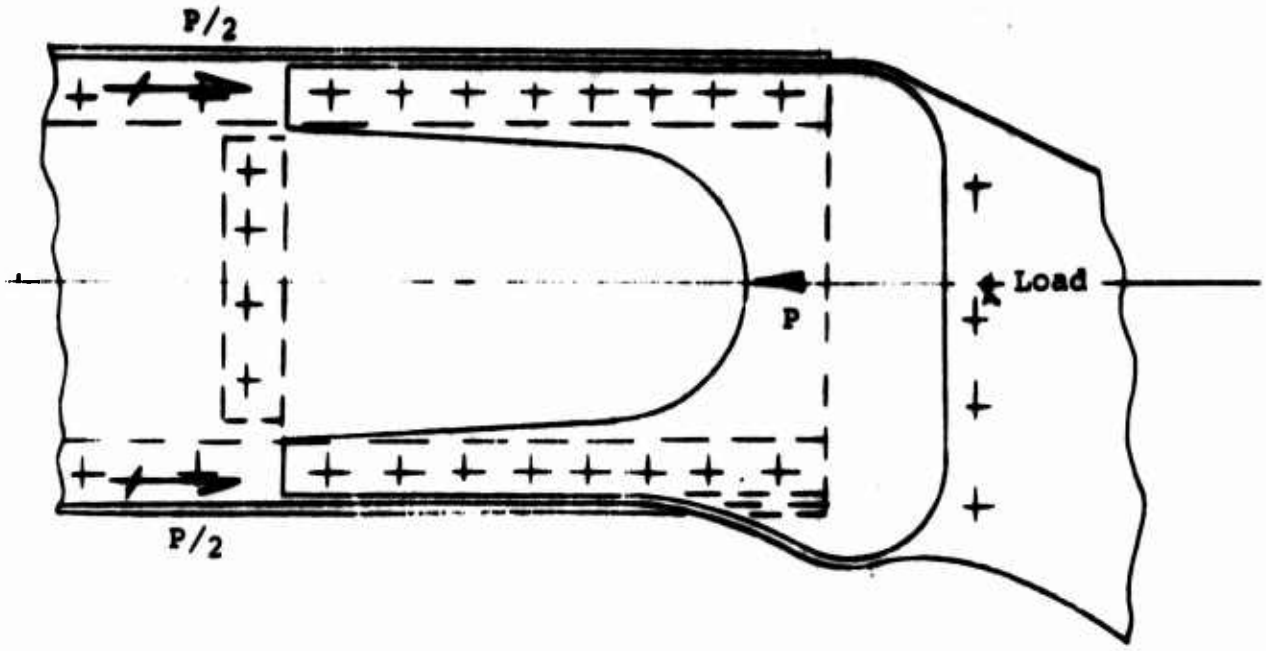
Rivet range out - use bolts.

$$F_{tu} = 180 \text{ Ksi} = 7/16\text{-}20 \text{ bolts}$$

$$P_T = 19.6^{K}$$

$$P_S = 15.4^{K}$$

$$MS = \frac{15.4}{12.2} - 1 = 0.26$$



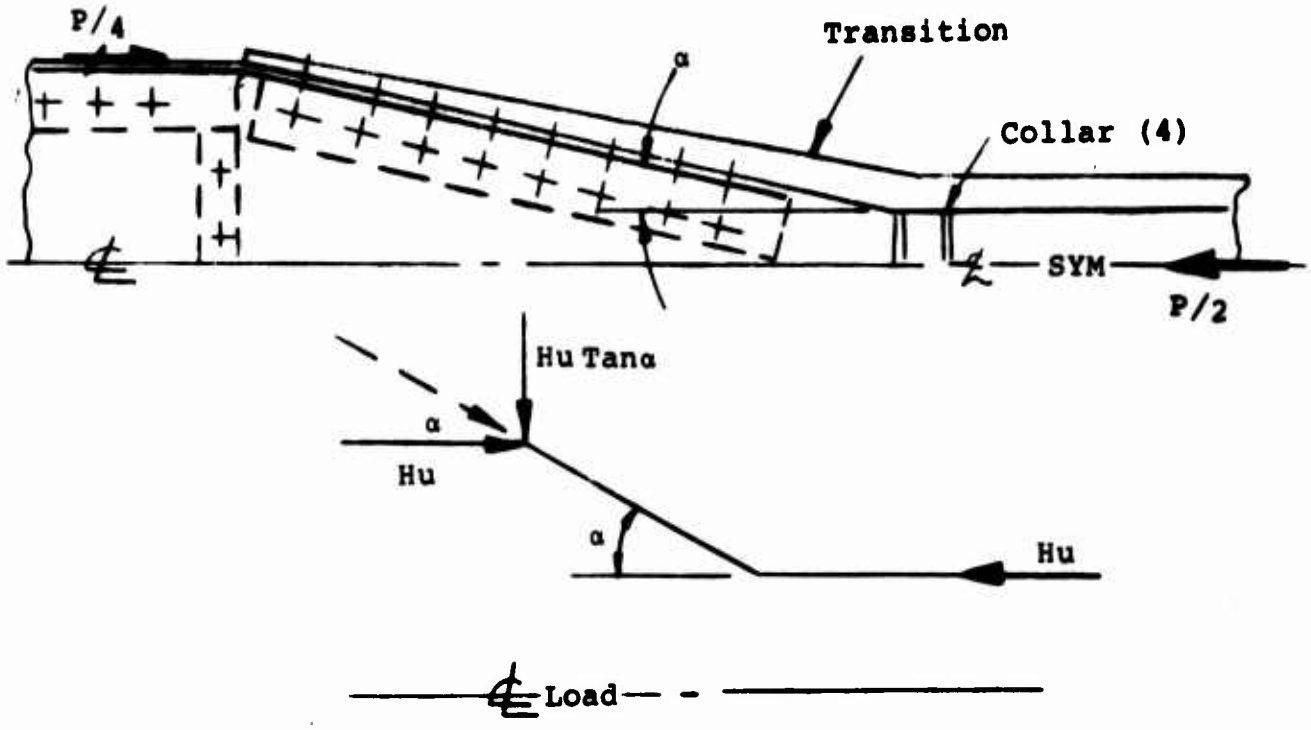


Figure 139. End Beam Connection, Schematic Diagram.

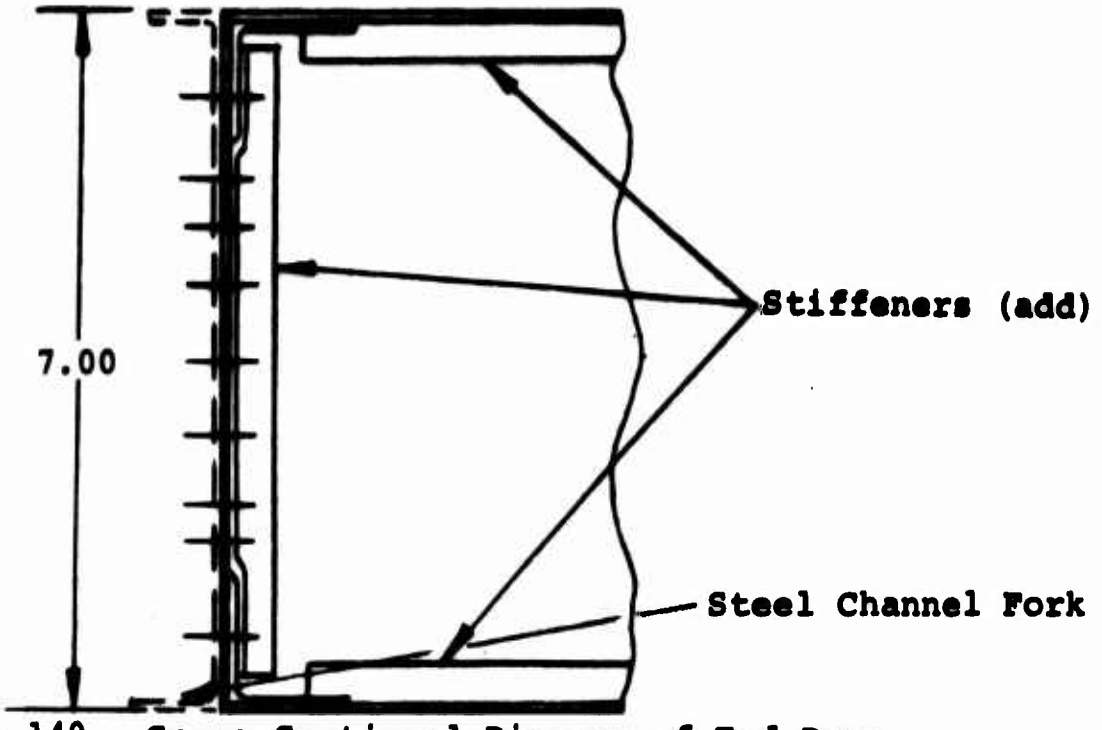


Figure 140. Cross-Sectional Diagram of End Beam Connection Showing Stiffeners.

To compensate for redundancy at transition, it is recommended that stiffeners (0.062 in.) be added to both sides of the frame. These would appear on the sides of the fork connection. In conjunction, the cutout of the fork should be made larger in order to stabilize and diminish the load path on the skin and stringer.

Since bolts rather than rivets are to be used, backup stiffeners will be required in the vicinity of the cutout.

Collars (4) located on hooks will stabilize some of the compressive load at transition, but additional stiffeners will be added.

CAPSTAN BEAM ASSEMBLY

Capstan Beam (Part No. SK17542) (Reference Figure 141)

Each capstan beam with its varying cross sections is supported by two bearings located at the ends of the beam. Both bearings can cope with longitudinal, lateral, and vertical loads. An exception to this is the forward bearing which cannot accommodate longitudinal loads (or thrust). See Figures 128 and 129.

The capstan beam channel sections are constructed from 2024-ST aluminum extrusions. Bending stresses attributed to Condition 3B provided an ample MS for the channel sections.

Beam Assy
SK 17542

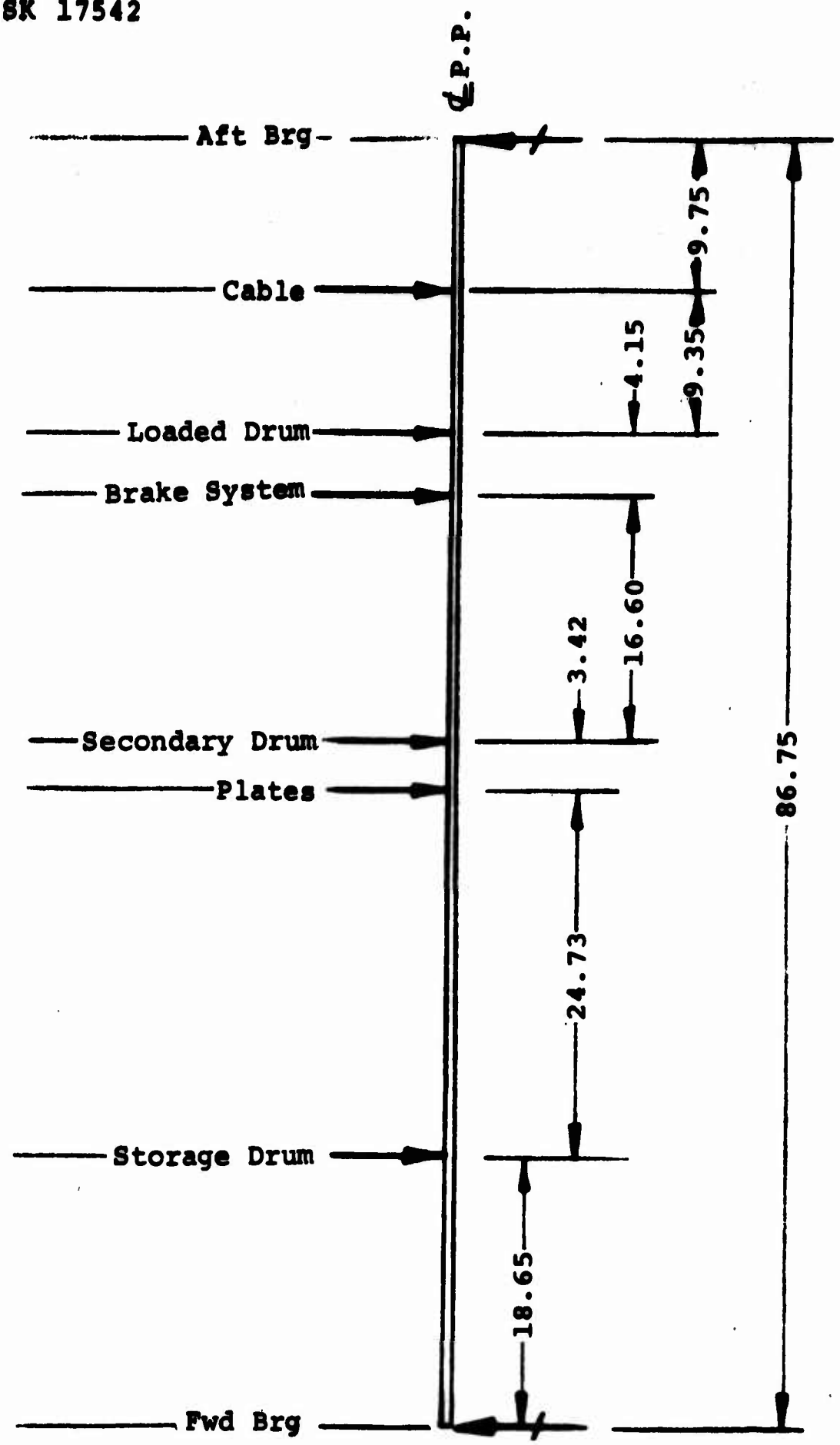
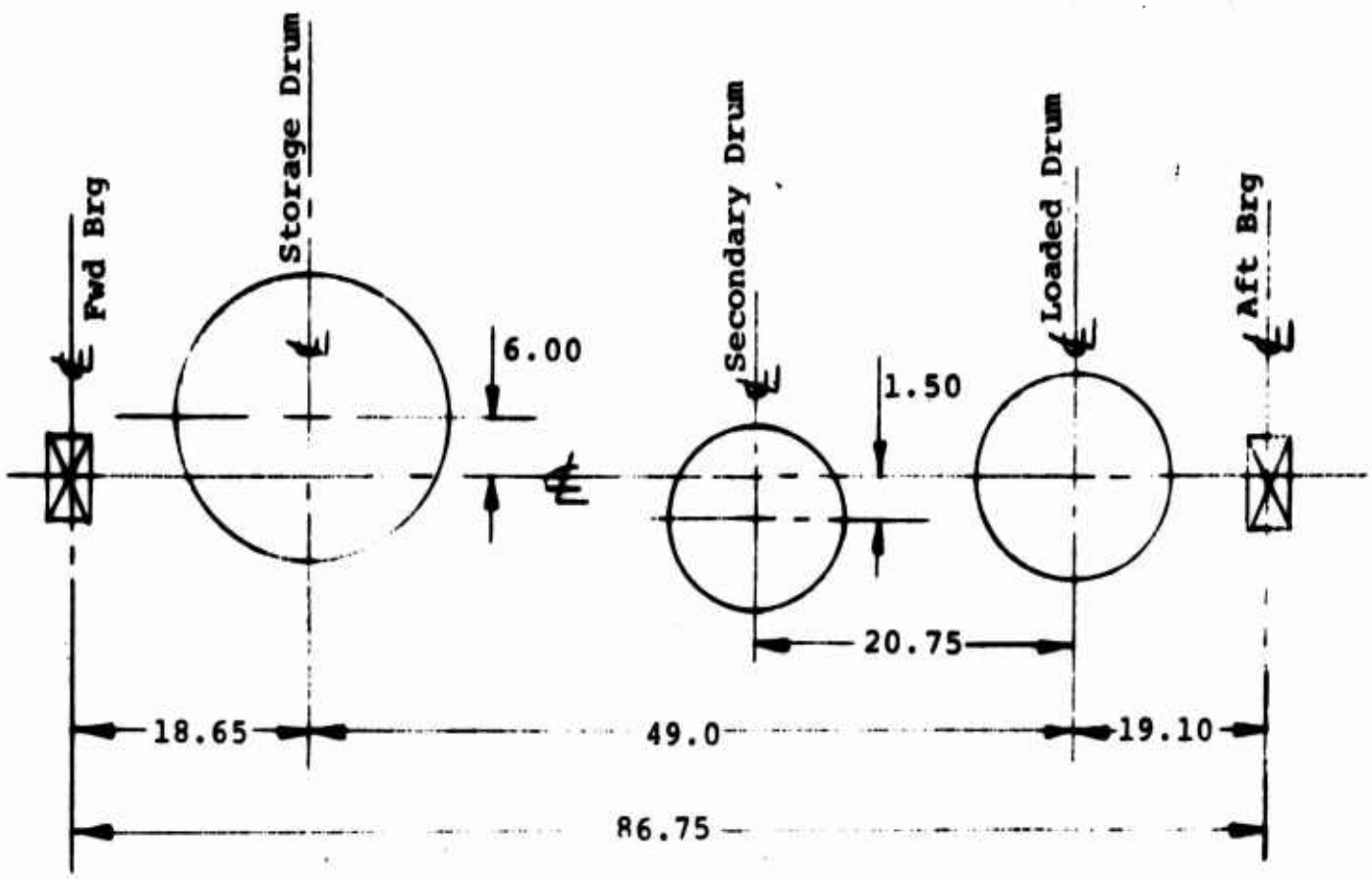
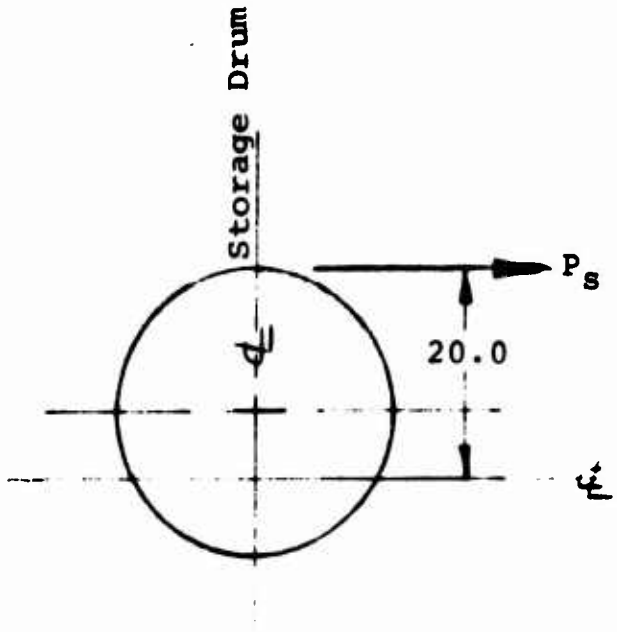
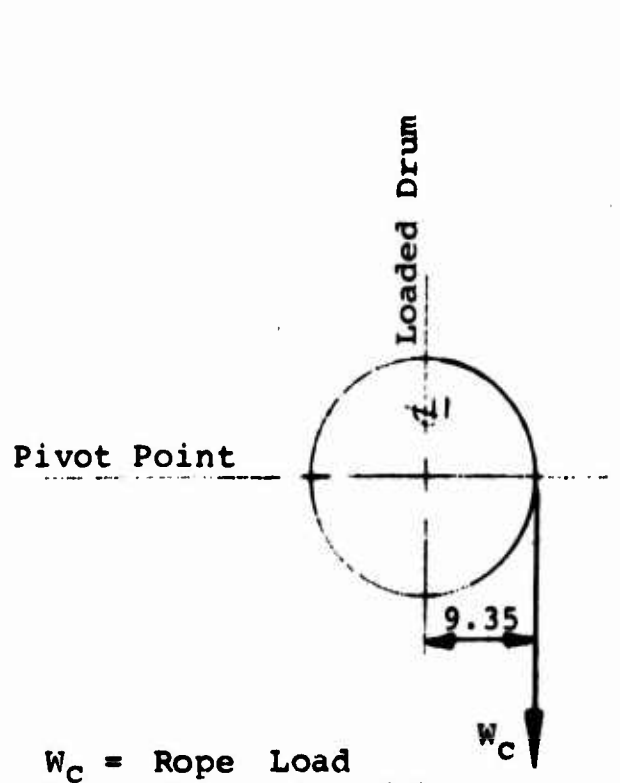


Figure 141. Capstan Load Location, Schematic Diagram.



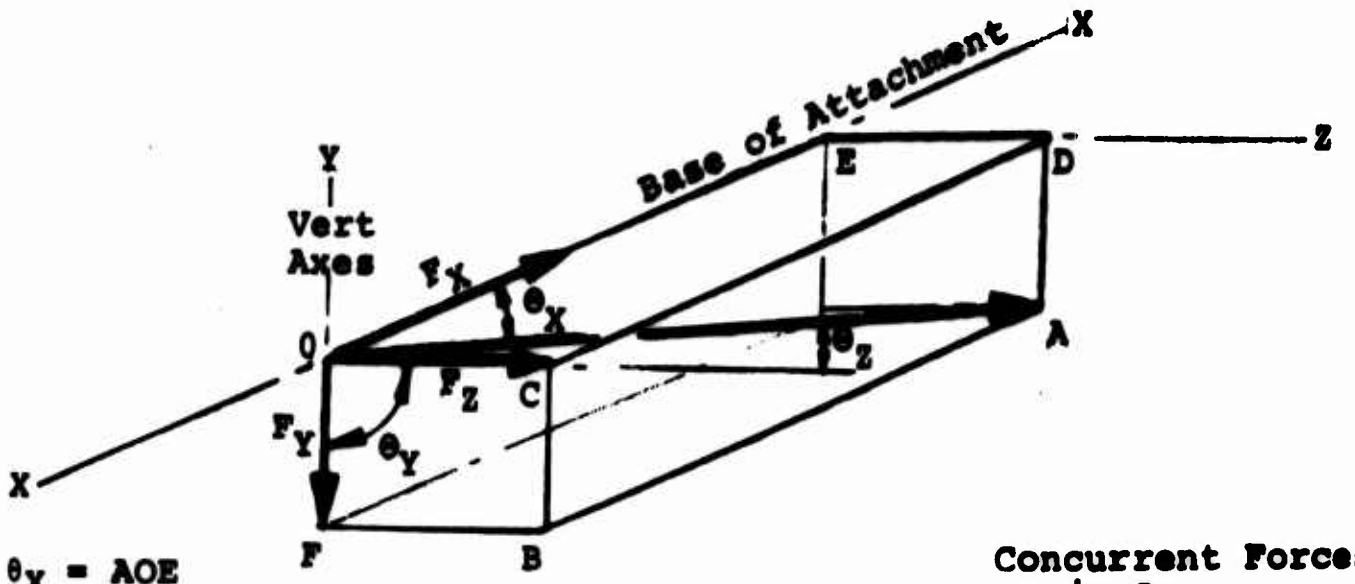




P_s = Preload in Rope 200 lb
W = 12.75 Rad/Min

W_c = Rope Load
Velocity = 60 ft/min

Figure 142. Drum Locations, Schematic Diagram.



θ_X = AOE
θ_Y = AOF
θ_Z = AOC

Concurrent Forces in Space

$$\cos^2\theta_X + \cos^2\theta_Y + \cos^2\theta_Z = 1$$

$$F = (F_X^2 + F_Y^2 + F_Z^2)^{\frac{1}{2}}$$

Let $\theta_X = 60°$

$$\theta_Y = 30° \quad (0.50)^2 + (0.866)^2 + \cos^2\theta_Z = 1$$

$$1 + \cos^2\theta_Z = 1$$

$$\cos\theta_Z = 0$$

$$\theta_Z = 90°$$

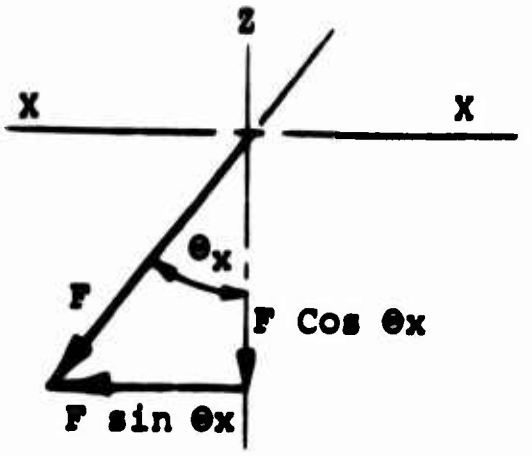
$$F_x = F\cos\theta_X = 0.50\ F$$

$$F_y = F\cos\theta_Y = 0.866\ F$$

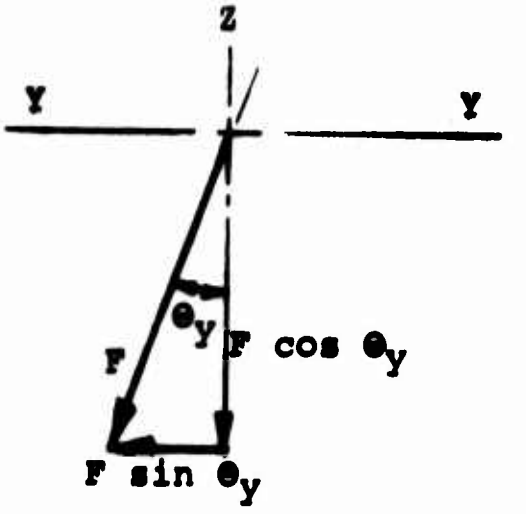
$$F_z = F\cos\theta_Z = 0$$

Figure 143. Capstan Loads.

Side View



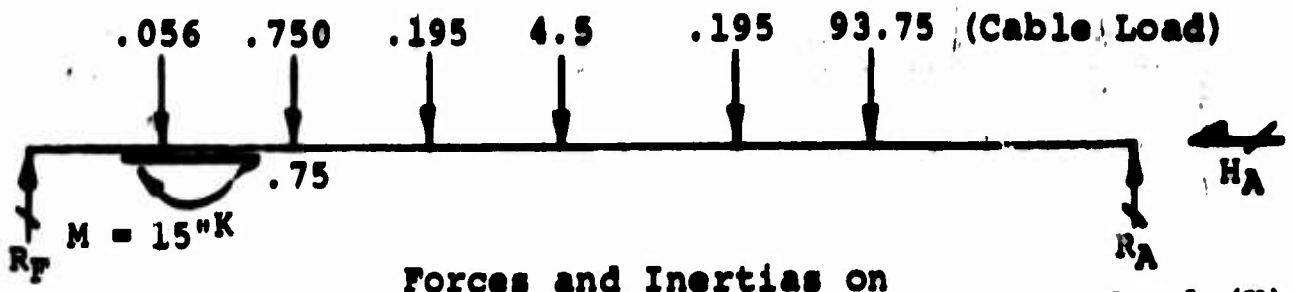
Front View

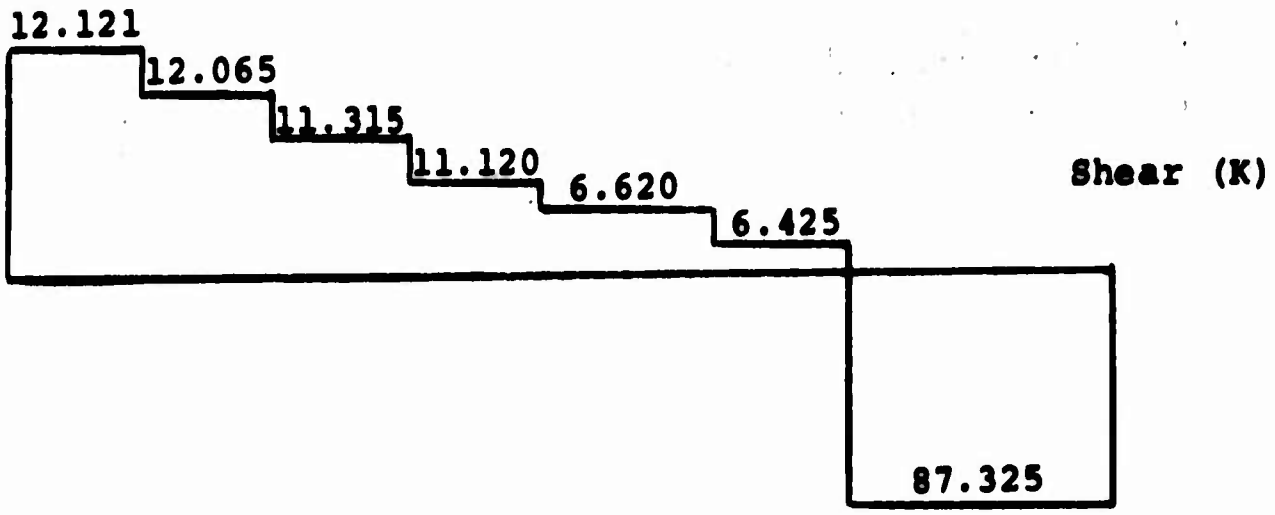


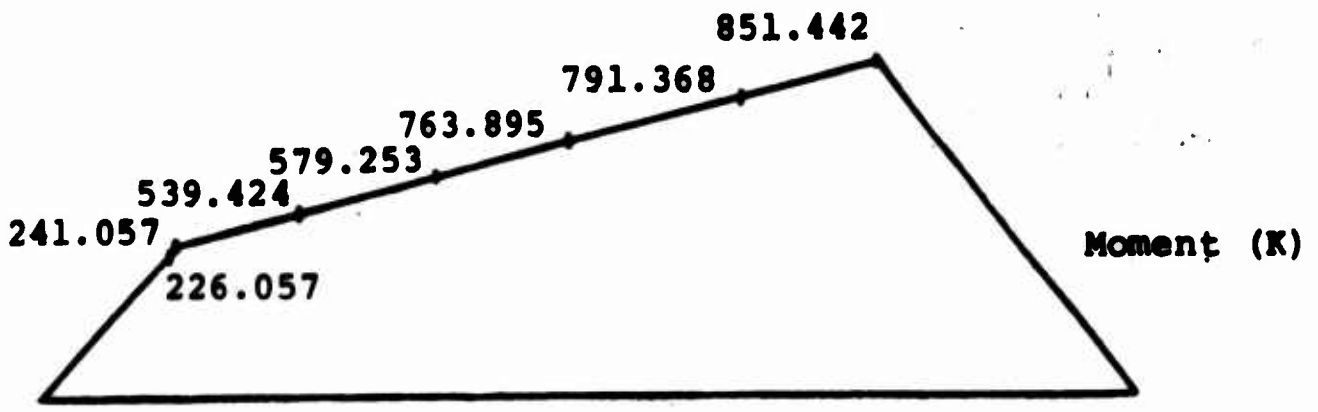
$\theta_x = \pm 30°$
Vertical = 0.866 F
Horizontal = 0.500 F

$\theta_y = \pm 30°$
Vertical = 0.866 F
Horizontal = 0.500 F

Figure 144. Capstan Loads - 30 Degrees.







$$\Sigma MR_F = 0$$

$$18.65\ (0.056) + 46.9\ (0.195) + 43.38\ (0.75) + 63.5\ (4.5) + 67.65\ (0.195) + 77\ (93.75) + 15 = 86.75\ R_A$$

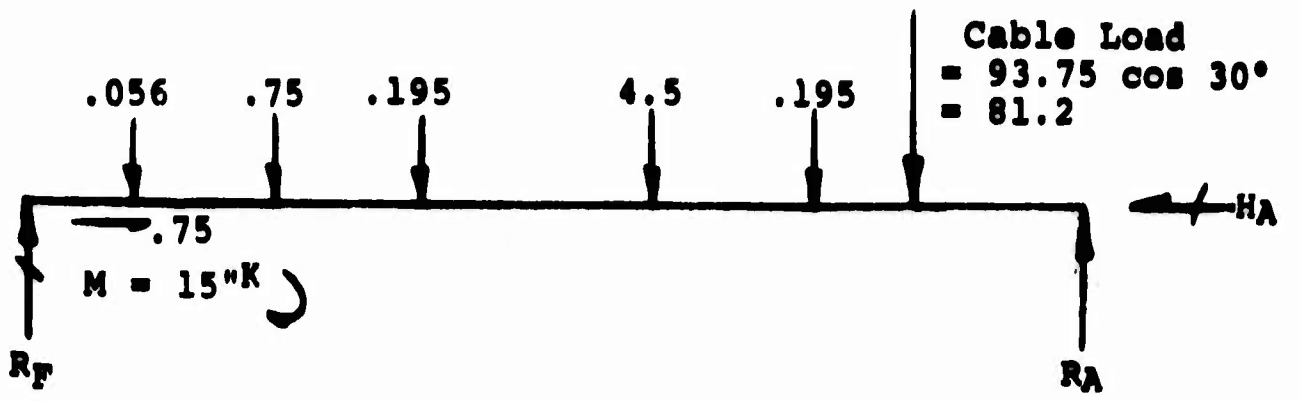
$$R_A = 87.325$$

$$R_F = 12.121$$

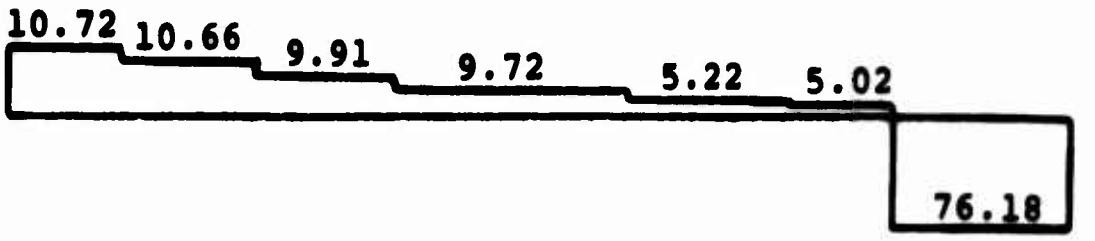
$$H_A = 0.75$$

$$\Sigma W = 99.446$$

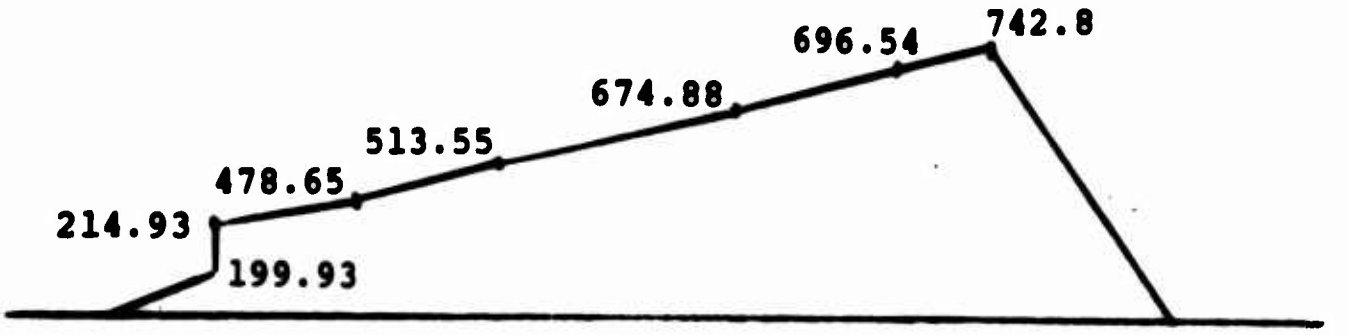
Figure 145. Symmetrical Loading with Maximum CG Travel - Condition 1B.



Load (K)



Shear (K)



Moment (In-K)

$$\Sigma M R_F = 0$$

$$18.65\ (0.056) + 46.9\ (0.195) + 43.38\ (0.75) + 63.5\ (4.5) + 67.65\ (0.195) + 77\ (81.19) + 15 = 86.75\ R_A$$

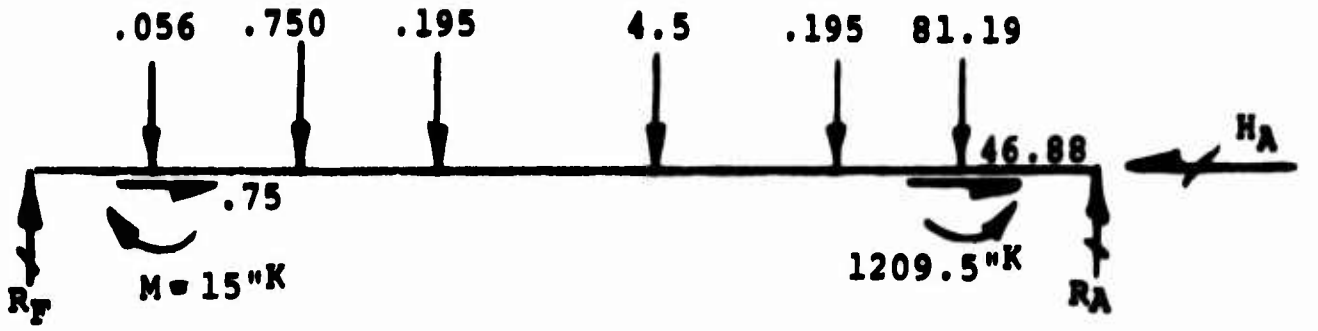
$$R_A = 76.18^K$$

$$R_F = 10.72^K$$

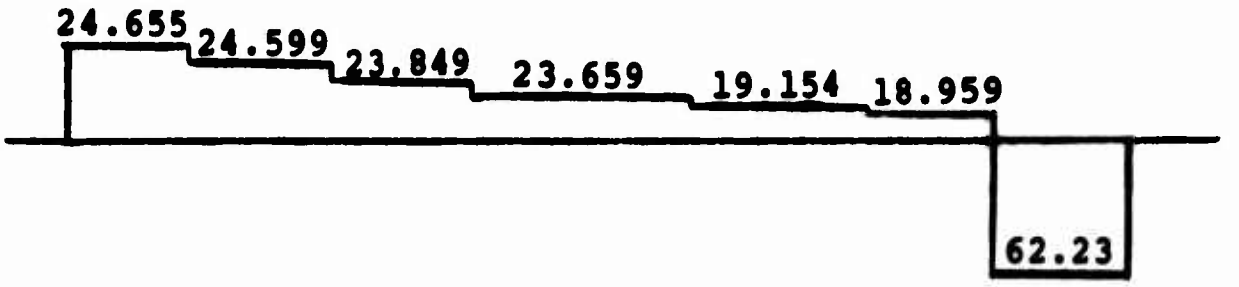
$$H_A = 0.750^K$$

$$\Sigma W = 86.896^K$$

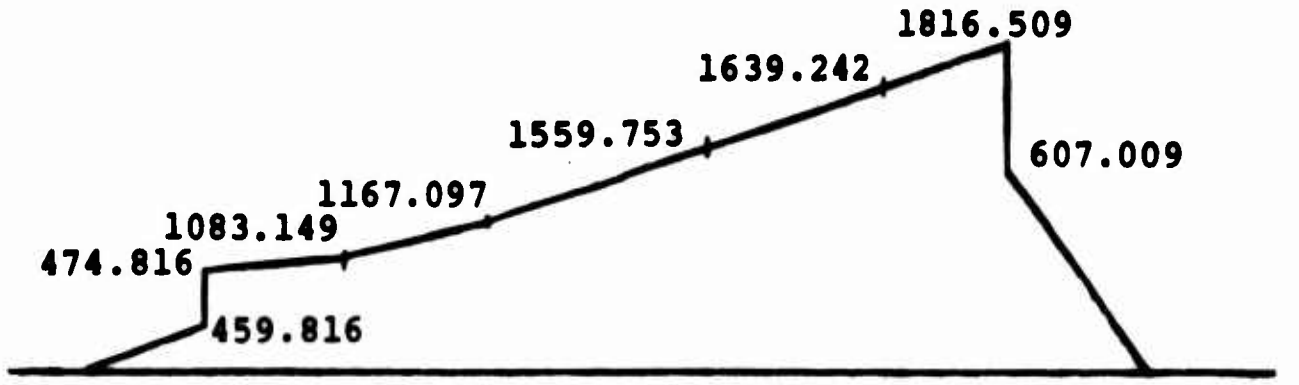
Figure 146. Lateral Sway - 30 Degrees - Condition 2B.



Load (K)



Shear (K)



Moment (In-K)

$$\Sigma MR_F = 0$$

$$18.65\ (0.056) + 46.9\ (0.195) + 43.38\ (0.75) + 63.5\ (4.5) + 67.65\ (0.195) + 77\ (81.19) - 1209.5 + 15 = 86.75\ R_A$$

$$R_A = 62.23^K$$

$$R_F = 24.655^K$$

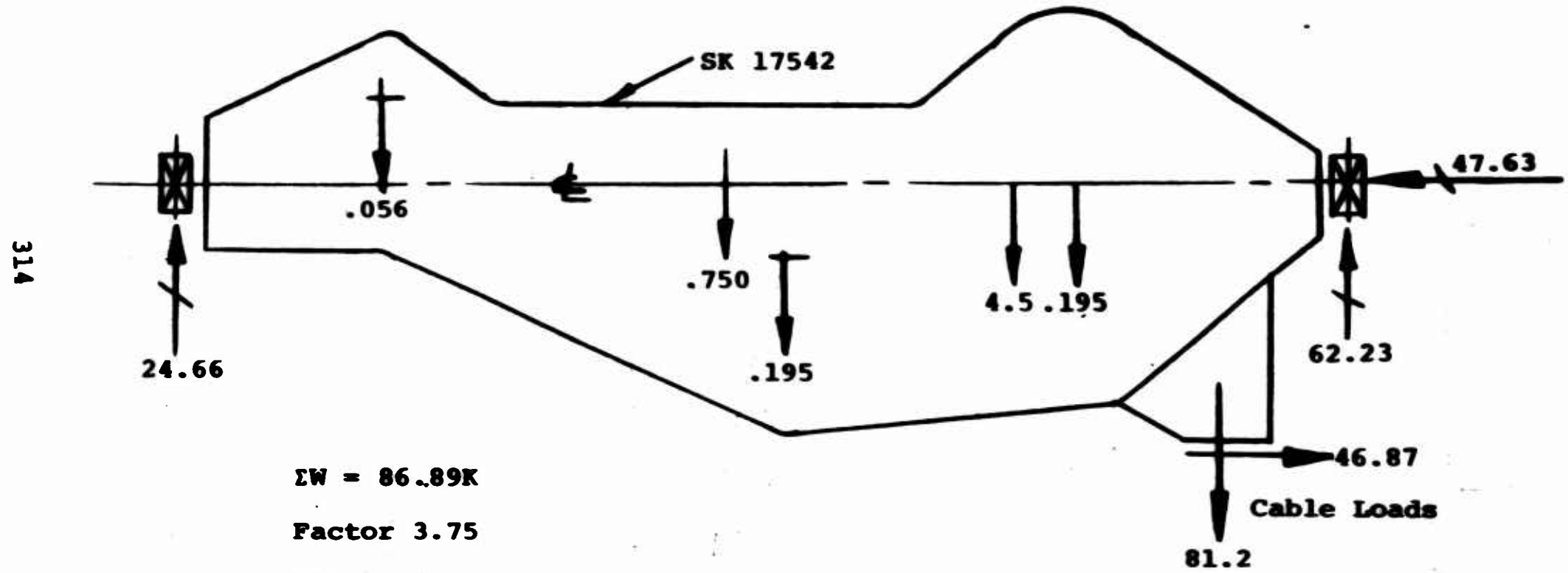
$$H_A = 47.63^K$$

$$\Sigma W = 86.886^K$$

Figure 147. Longitudinal Sway - 30 Degrees, Forward - Condition 3B.

Long. Sway 30° Aft ⟶ .750 (Preload in cable)

Load in Kips



ΣW = 86.89K

Factor 3.75

Ult loads

Figure 148. Capstan Winch, Condition 3B, Schematic Diagram.

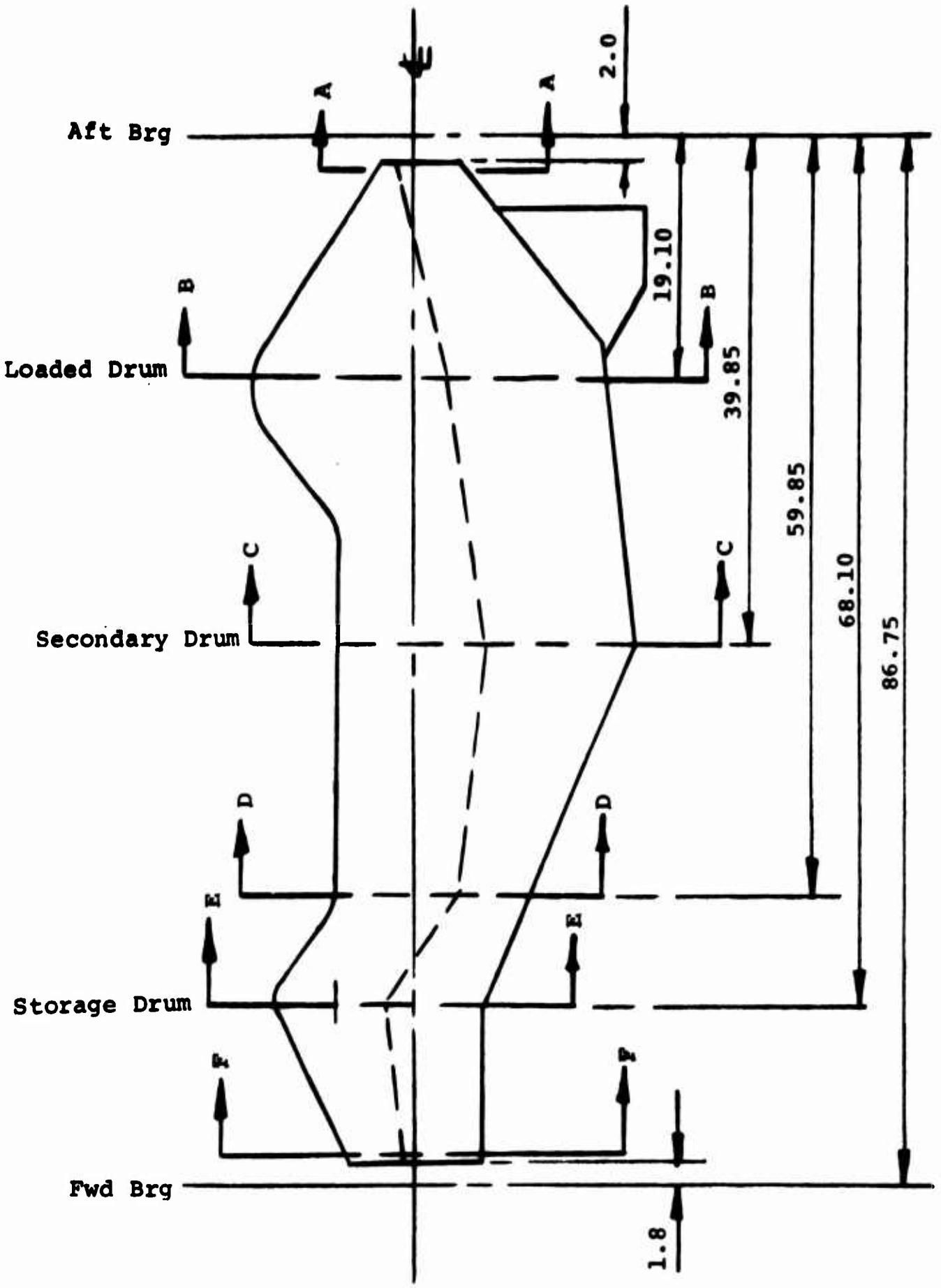


Figure 149. Winch Side Frame.

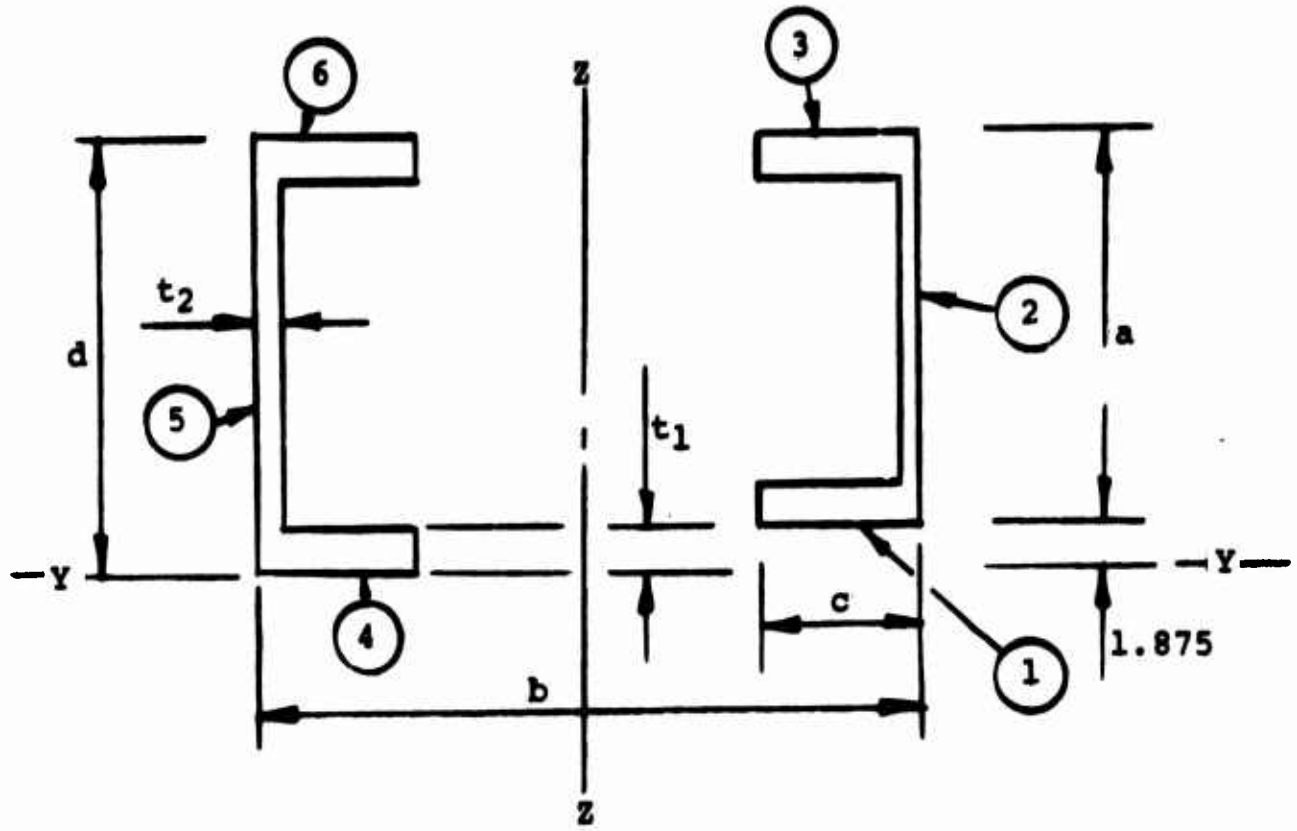


Figure 150. Capstan Winch, Condition 3B, Schematic Diagram.

Values for Section A-A (Figure 149)

a = 9.625 t_1 = 0.40
b = 12.50 t_2 = 0.25
c = 2.00 ℓ^2 = 2.00
d = 11.50

TABLE XVIII
FACTORS AND VALUES OF SECTION A-A OF DRAWING SK 17542

Item	b	h	A	z	A_z	A_z^2	I_o	I_y
1	2.00	0.400	0.800	2.075	1.6600	3.4445	0.011	3.456
2	0.25	8.825	2.206	6.488	14.3125	92.8595	14.310	107.169
3	2.00	0.400	0.800	11.30	9.0400	102.1521	0.011	102.163
4	2.00	0.400	0.800	0.200	0.1600	0.032	0.011	0.043
5	0.25	10.70	2.675	5.750	15.3813	88.4424	25.512	113.954
6	2.00	0.400	0.800	11.30	9.0400	102.154	0.011	102.163
Σ	--	--	8.081	--	49.5938	--	--	428.948

$\bar{y} = A_z/\Sigma A = 6.137$ in.

$I_{yy} = I_y - A\bar{y}^2 = 428.948 - (6.137)^{-2}\ (8.081) = 124.596$

Values for Section B-B (Figure 149)

$a/z = (29.4) + 0.5 = 14.70$ $\quad t_2 = 0.25$
$c = 2.00$ $\quad b = 12.50$
$d/z = 14.70$ $\quad \ell = 19.10$
$t_1 = 0.40$

TABLE XIX
FACTORS AND VALUES OF SECTION B-B OF DRAWING SK17542

Item	b	h	*A	z	A_z	A_z^2	I_o	I_y
3&6	2.00	0.400	1.600	14.50	23.200	336.400	0.022	336.422
2&5	0.25	14.30	7.150	7.15	51.123	365.529	121.794	487.32
Σ	--	--	8.750	--	--	--	--	823.742

*Doubled

$\bar{y} = 14.70$ $\quad I_{yy} = 2(823.742) = 1647.484$
$A = 17.50$

Values for Section C-C (Figure 149)

$a/z = d/z = 0.5\ (24.40) = 12.20$ $\quad t_1 = 0.40$
$b = 12.50$ $\quad t_2 = 0.25$
$c = 2.00$ $\quad \ell = 39.85$

TABLE XX
FACTORS AND VALUES OF SECTION C-C OF DRAWING SK17542

Item	b	h	*A	z	A_z	A_z^2	I_o	I_y
3&6	2.00	0.400	1.600	12.00	19.200	230.400	0.022	230.422
2&5	0.25	11.80	5.900	5.90	34.810	205.379	68.432	273.811
Σ	--	--	7.50	--	--	--	--	504.233

*Doubled

$\bar{y} = 12.20$ $\quad I_{yy} = 1008.466$
$A = 15.00$

Values for Section D-D (Figure 149)

$a/z = d/z = 0.5\ (15.25) = 7.625$ $\quad t_1 = 0.40$
$c = 2.00$ $\quad t_2 = 0.25$
$d = 13.625$ $\quad \ell = 59.85$

TABLE XXI

FACTORS AND VALUES OF SECTION D-D OF DRAWING SK17542

Item	b	h	A	z	A_z	A_z^2	I_o	I_y
3&6	2.00	.400	1.600	7.425	11.880	88.209	.022	88.231
2&5	0.25	7.225	3.613	3.613	13.054	47.164	15.708	62.872
Σ	--	--	5.213	--	--	--	--	151.103

$\bar{y}$ = 7.625 I = 302.206
A = 10.416

Values for Section E-E (Figure 149)

a/z = d/z = 0.5 (16.40 = 8.20 t_1 = 0.40
c = 2.00 t_2 = 0.25
b = 13.625 ℓ = 68.10

TABLE XXII

FACTORS AND VALUES OF SECTION E-E OF DRAWING SK17542

Item	b	h	A	z	A_z	A_z^2	I_o	I_y
3&6	2.00	0.400	1.600	8.00	12.800	102.400	0.022	102.422
2&5	0.25	7.80	3.900	3.90	15.210	59.319	19.766	79.085
Σ	--	--	5.5	--	--	--	--	181.507

$\bar{y}$ = 8.20 I = 363.014
A = 11.00

Values for Section F-F (Figure 149)

a/z = d/z = 0.5 (10.25) = 5.125 t_1 = 0.40
b = 12.625 t_2 = 0.25
c = 2.00 ℓ = 84.95

TABLE XXIII

FACTORS AND VALUES OF SECTION F-F OF DRAWING SK17542

Item	b	h	A	z	A_z	A_z^2	I_o	I_y
3&6	2.00	0.400	1.600	4.925	7.880	38.809	0.022	38.831
2&5	0.25	4.725	2.362	2.363	5.496	12.987	4.394	17.381
Σ	--	--	3.962	--	--	--	--	56.212

$\bar{y}$ = 5.125 I = 112.424
A = 7.925

Capstan Frame Stress (Ultimate)

Conditions:

Multipoint lift (Condition 3B)
Longitudinal sway - 30° aft
Material - 2024-T4 extrusion
Ftu = 57 Ksi
Fty = 42 Ksi

TABLE XXIV
FACTORS AND VALUES OF CAPSTAN BEAM

Section	x (in.)	z (in.)	I ($in.^4$)	I/c ($in.^3$)	M (K in.)	2 σ Ksi
A-A	2.0	6.14	124.6	20.3	124.5	11.9*
℄ Cable	9.75	10.0	600.0	60.0	1816.5	36.2*
B-B	19.10	14.70	1647.5	112.0	1639.2	14.6
C-C	39.85	12.20	1008.0	81.7	1167.1	14.3
D-D	59.85	7.625	302.2	39.6	677.9	17.1
E-E	68.10	8.20	363.0	44.2	474.8	10.8
F-F	84.95	5.13	112.4	21.8	44.4	2.0

$MS = Mc/I + P/A$

$MS = (57/36.2) - 1 = 0.57$

Capstan Beam Optimization

TABLE XXV
OPTIMIZED FACTORS AND VALUES OF CAPSTAN BEAM

Section	x (in.)	M (K in.)	I/c_3 ($in.^3$)	c (in.)	I ($in.^4$)
A-A	2.00	124.5	2.30	6.14	14
℄ Cable	9.75	1816.5	3.30	10.0	330
B-B	19.10	1639.2	29.7	14.7	435
C-C	39.85	1167.1	21.2	12.2	260
D-D	59.85	677.9	12.3	7.63	94
E-E	68.10	474.8	8.7	8.20	72
F-F	84.95	44.4	0.8	5.13	4.0

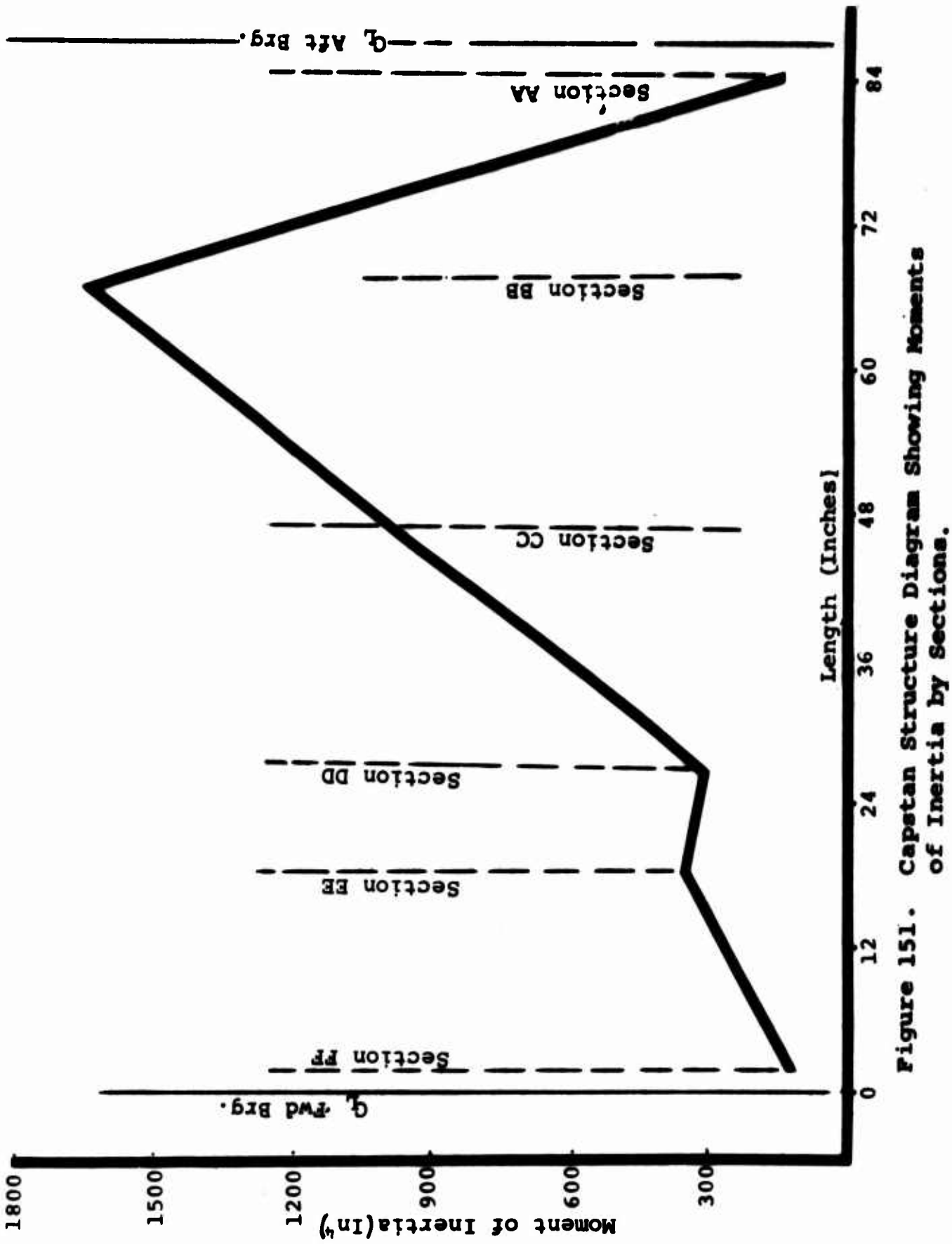


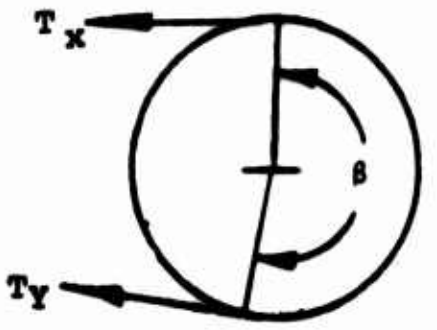
Figure 151. Capstan Structure Diagram Showing Moments of Inertia by Sections.

The above values indicate the desired physical properties of SK17542 which produce a stress level at the indicated section that is equal to Ftu for the critical load case (Condition 3B).

CABLE LOADS AND STRESSES ON LOADED AND SECONDARY DRUMS

The vertical load on the rope is transmitted to the loaded and secondary drums by friction between the cable and the drum. The rope has 7 wraps on the loaded drum and 6 wraps on the secondary drum. As the rope winds around the drum, the rope load is reduced by frictional reaction forces. This relationship can be analytically described by the following equation and sketch:

$$\frac{T_x}{T_y} = e^{f\beta}$$



(See Reference 19.)

where

T_x, T_y are rope loads ($T_x > T_y$).

f is the coefficient of friction, and

β is the wrap angle.

The torque values on the loaded and secondary drums are reacted by the winch drive system. For this particular mode of load transfer, the value of the coefficient of friction is very critical. If the coefficient of friction is below 0.125, the rope load to the storage drum will exceed the allowable 200 pounds. It is therefore recommended that a test be conducted using oily rope to substantiate the drum system.

Drum rope loads for 4 different coefficients of friction are calculated in this section. Both drums are checked for hoop compression due to rope loads (f = 0.125 assumed). The bearings, located inside the drum, will prevent the drum from buckling.

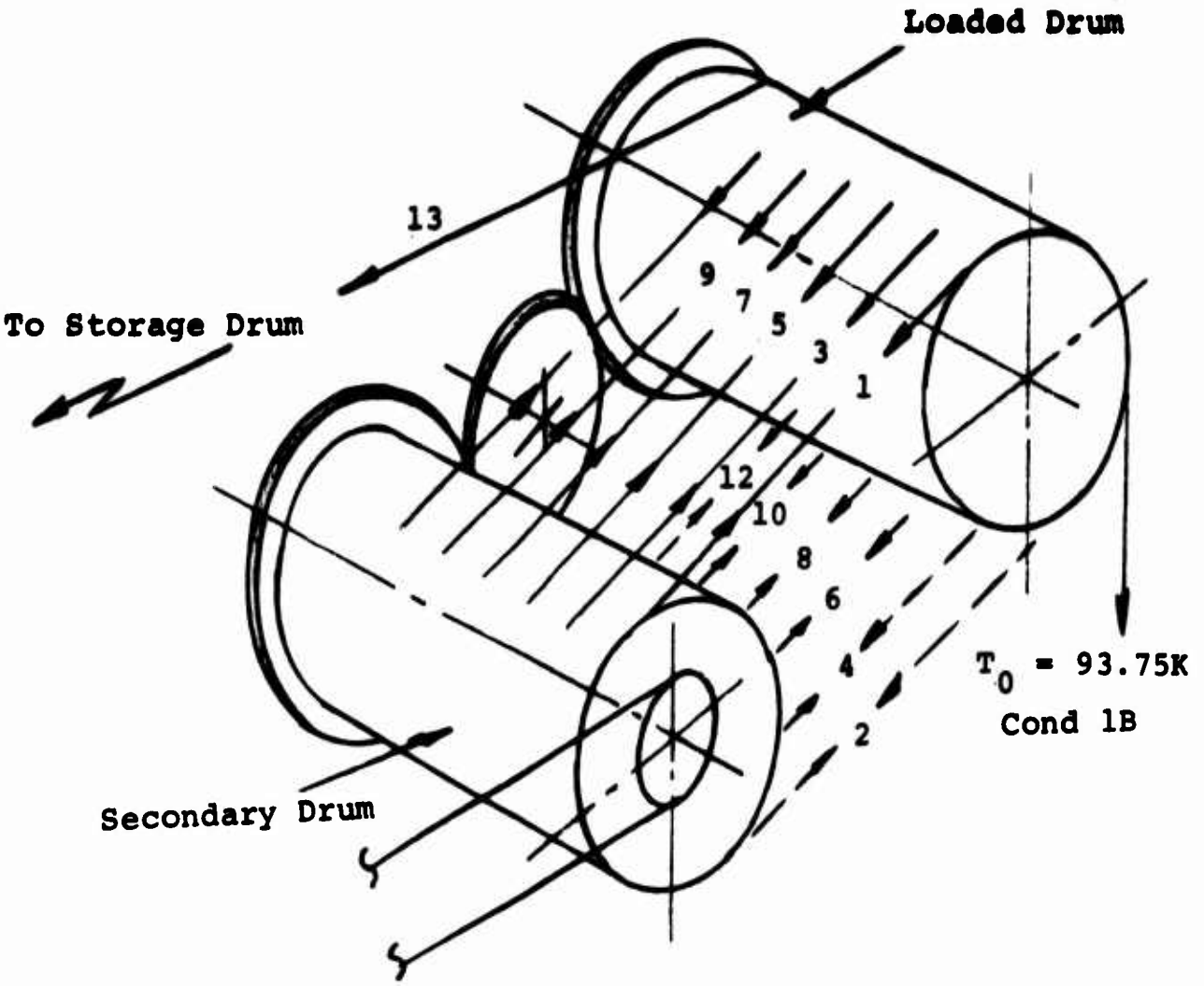


Figure 152. Rope Loads on Secondary and Loaded Drums.

Rope loads are as follows for a coefficient of friction equal to 0.125:

T_0	= 93.75 K	T_7	= 7.37 K
T_1	= 77.2 K	T_8	= 4.98 K
T_2	= 55.2 K	T_9	= 3.36 K
T_3	= 35.2 K	T_{10}	= 2.27 K
T_4	= 23.8 K	T_{11}	= 1.55 K
T_5	= 16.1 K	T_{12}	= 1.05 K
T_6	= 10.9 K	T_{13}	= 0.75 K

The rope load will be calculated for 4 values of the coefficient of friction as follows:

First wrap = $\frac{\pi}{2}$

Subsequent wraps = π

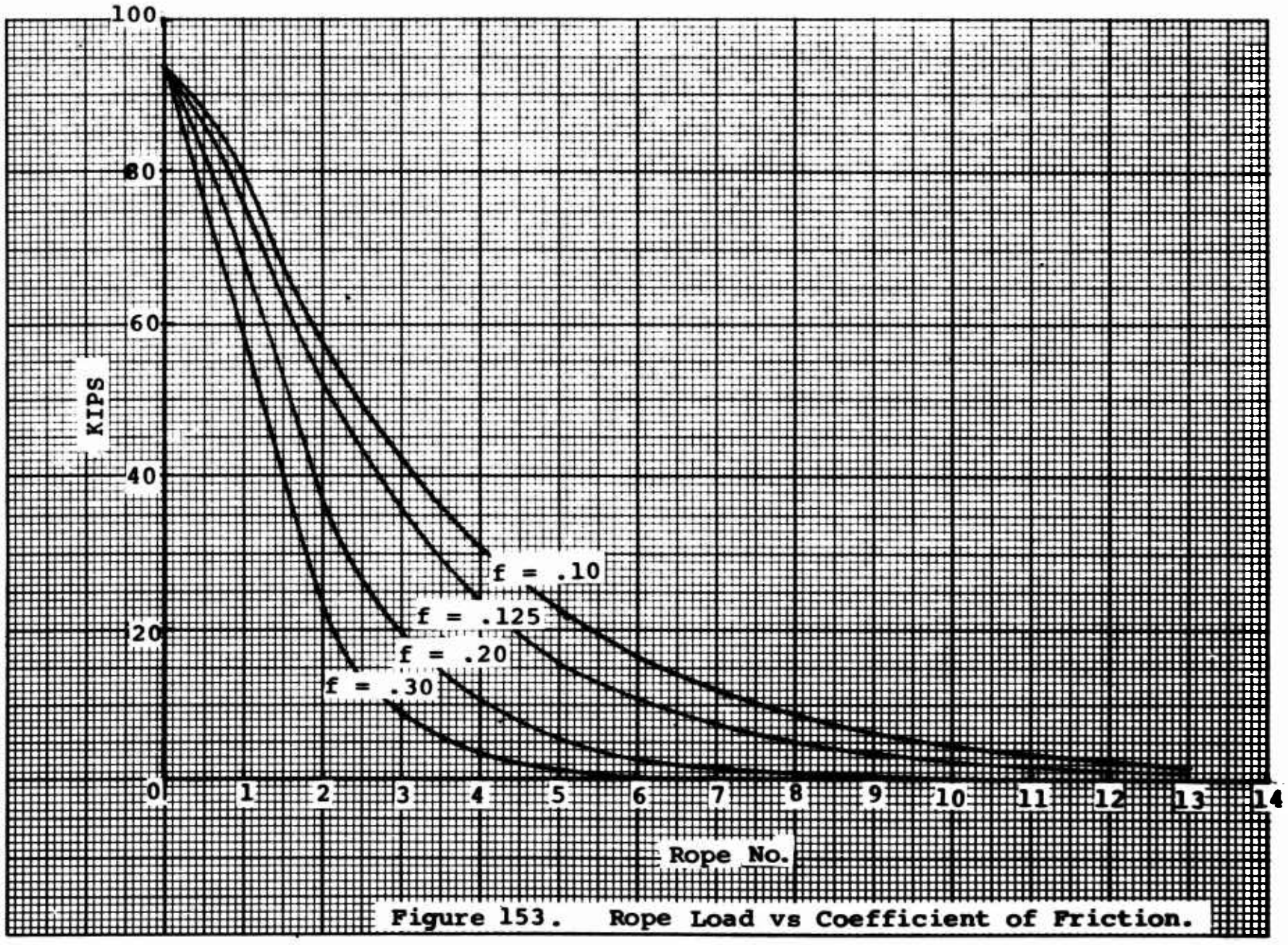


Figure 153. Rope Load vs Coefficient of Friction.

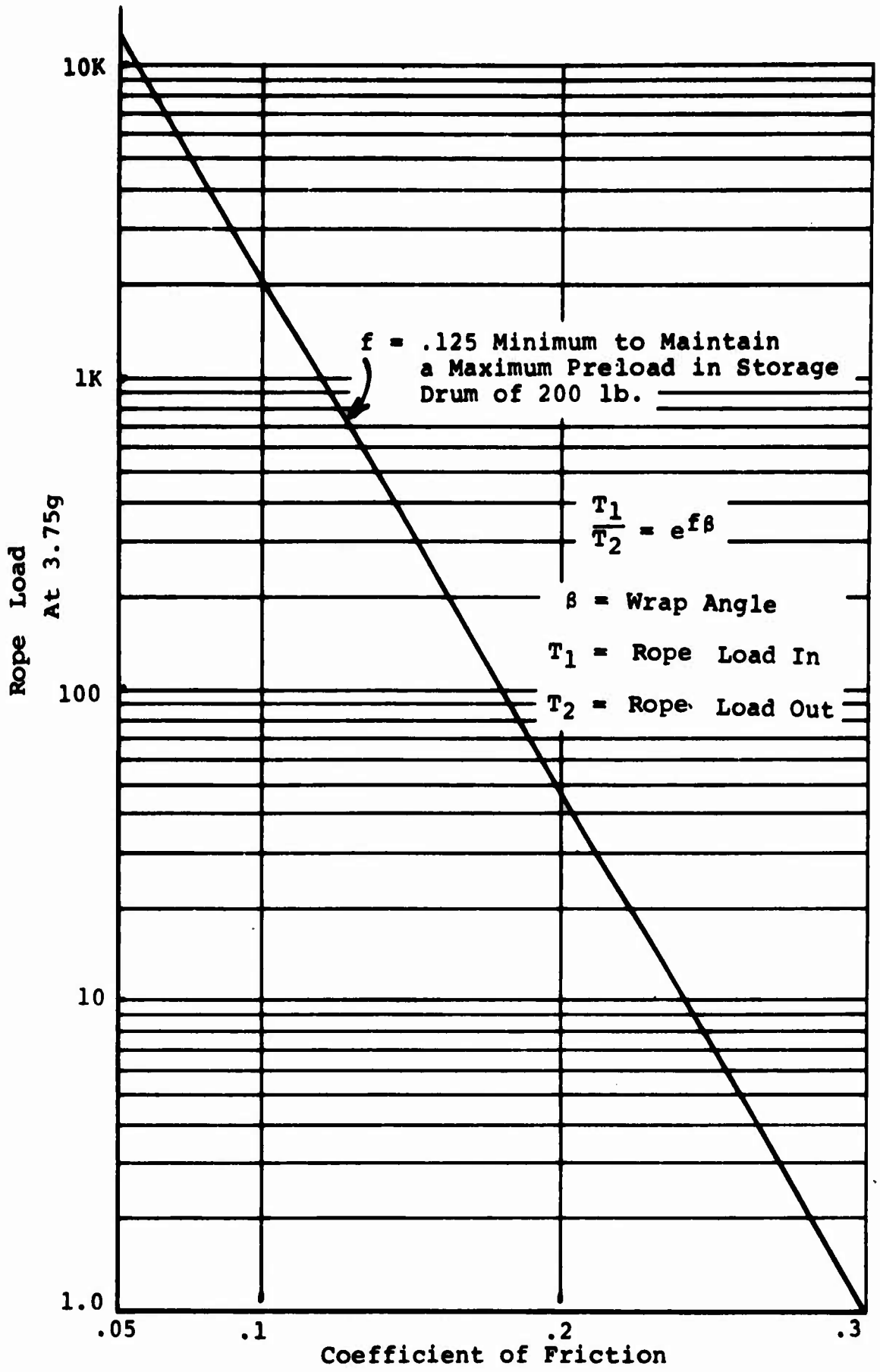


Figure 154. Coefficient of Friction vs Rope Load.

TABLE XXVI
ROPE LOADS FOR FOUR COEFFICIENTS OF FRICTION

Rope Identification	Rope Load (K Pounds) f = 0.10	f = 0.125	f = 0.20	f = 0.30
T_0	93.75	93.75	93.75	93.75
T_1	89.0	77.2	68.4	58.5
T_2	57.3	55.2	36.5	22.8
T_3	42.6	35.2	19.5	8.88
T_4	31.0	23.8	10.4	3.46
T_5	22.6	16.1	5.54	1.345
T_6	16.5	10.9	2.96	0.525
T_7	12.0	7.37	1.58	0.204
T_8	8.75	4.98	0.89	0.0795
T_9	6.38	3.36	0.474	0.031
T_{10}	4.65	2.27	0.253	0.0121
T_{11}	3.40	1.55	0.135	0.0047
T_{12}	2.48	1.05	0.072	0.00183
T_{13}	1.81	0.71	0.0384	0.000715

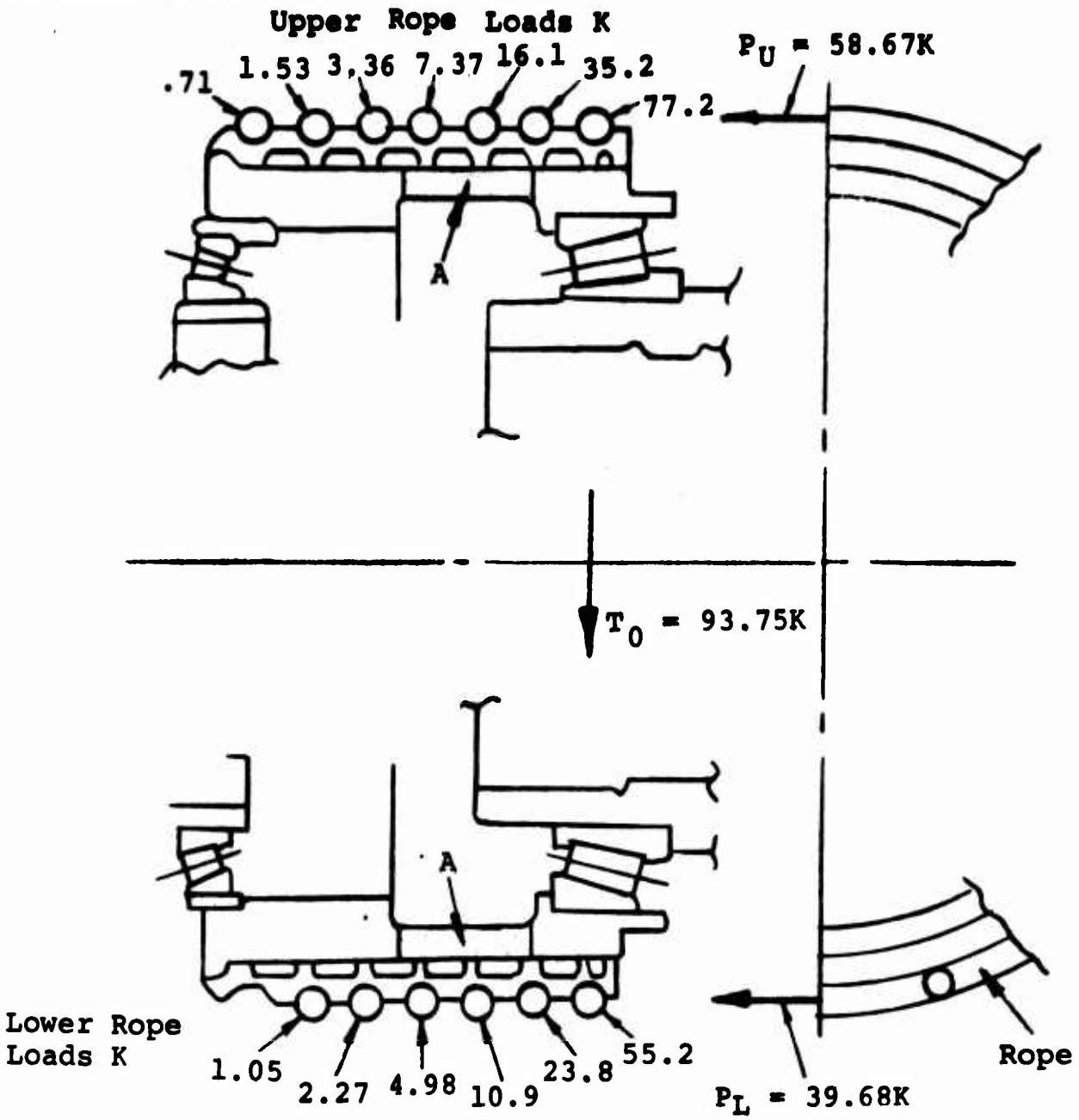


Figure 155. Loaded Drum Section.

Check of Thinness at Section A-A:

$P_U = 7.37 + 16.1 + 35.2 = 58.67$ K

$P_L = 4.98 + 10.9 + 23.8 = 39.68$ K

Hoop compression (area = 0.5 x 3.0)

$$f_c = \frac{58.67}{0.5 \times 3.0} = 39.100 \text{ psi}$$

For 2014 -T6 forging F_{CY} = 53 000 psi

$$MS = \frac{53.000}{39.100} - 1 = + 0.35$$

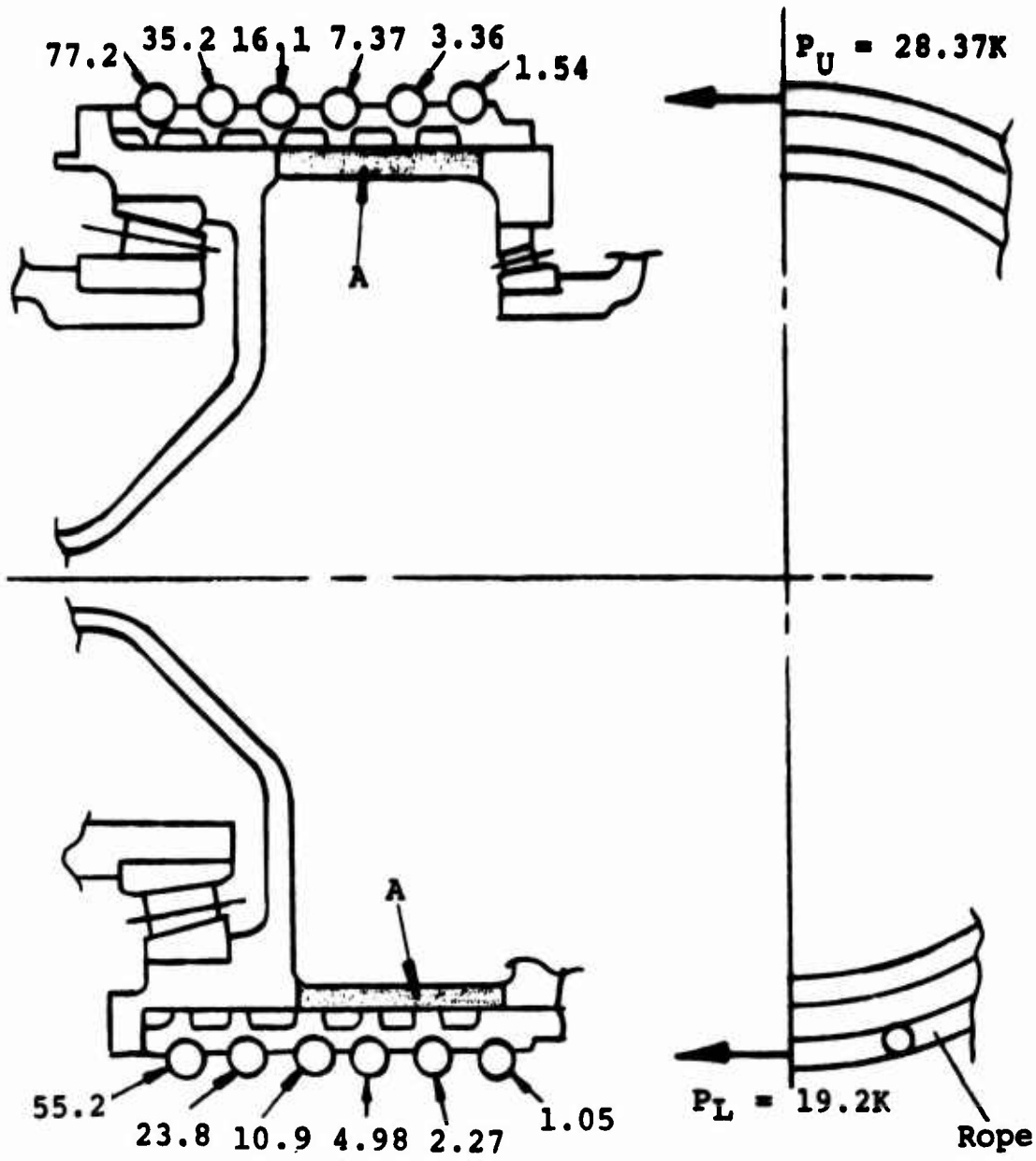


Figure 156. Secondary Drum Section A-A.

Check of Thinness at Section A-A:

$P_U = 16.1 + 7.37 + 3.36 + 1.54 = 28.37$

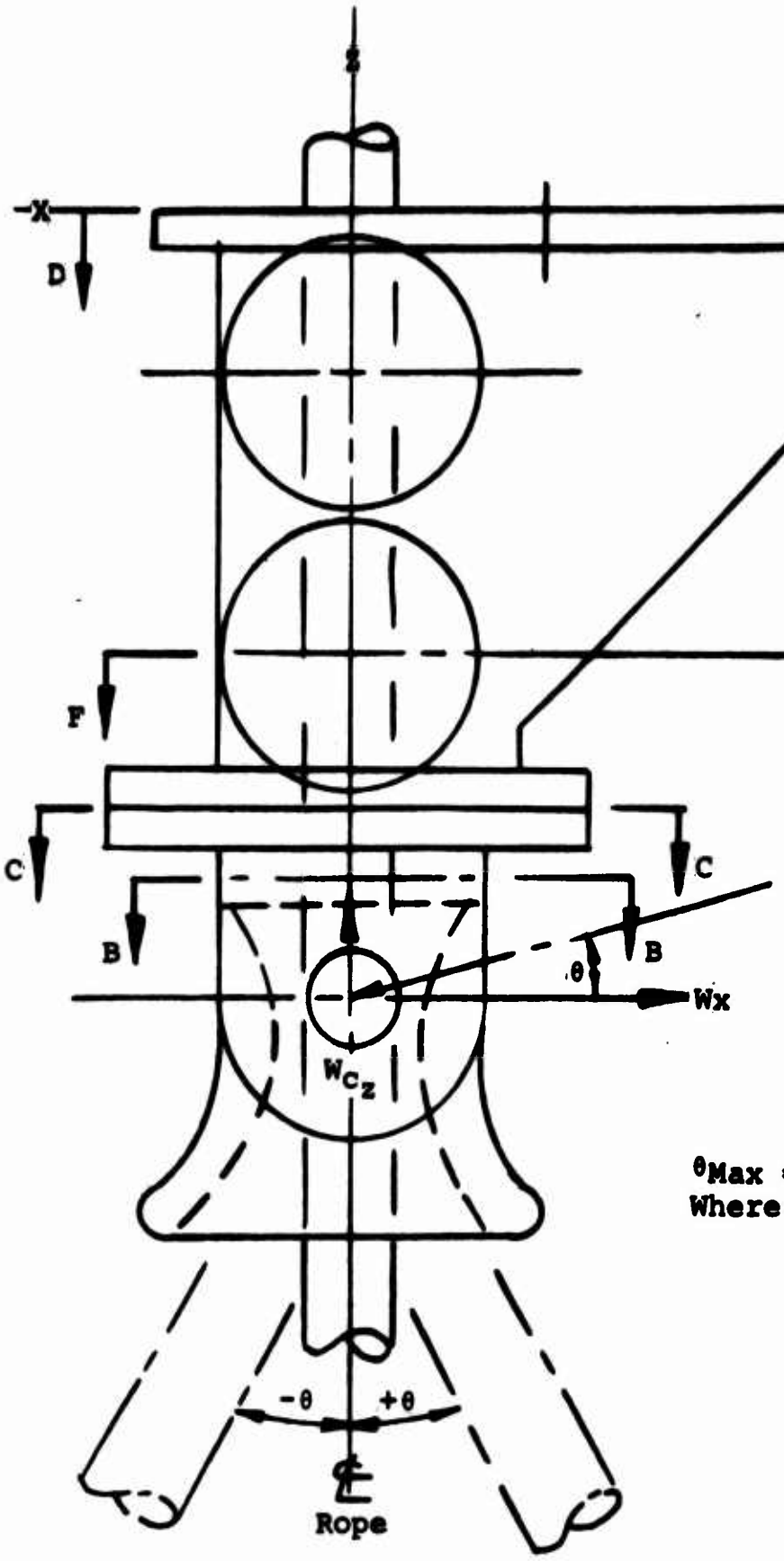
$P_L = 10.9 + 4.98 + 2.27 + 1.05 = 19.20$

Hoop compression (area = 0.5 x 3.0)

$$f_c = \frac{28.37}{0.5 \times 3.0} = 18.900 \text{ psi}$$

For 2014-T6 forging $F_{cy} = 53.000$ psi

$$MS = \frac{53.000}{18.900} - 1 = +1.80$$



Side View

$\theta_{Max} = 30^{\bullet}$ (Cond 3B)
Where: $W_x = 46.8^K$
$W_{C_Z} = 12.55^K$

Figure 157. Bellmouth and Rope Cutter.

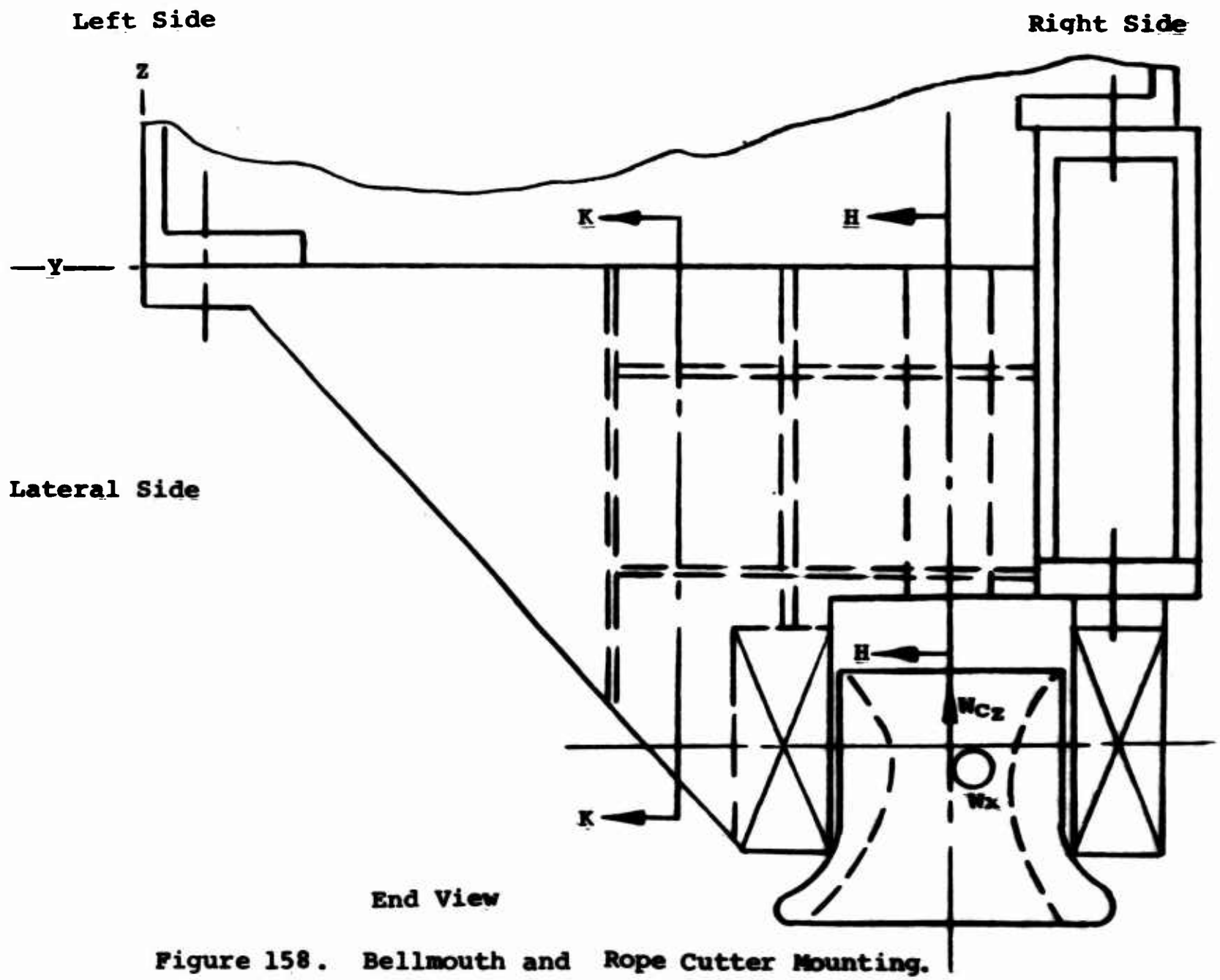


Figure 158. Bellmouth and Rope Cutter Mounting.

BELLMOUTH MOUNTING (PART NO. SK17542)

The bellmouth features a swivel joint which allows the rope to sway in a longitudinal direction without excessive chafing. A swaying load, however, exerts a longitudinal force at the bellmouth joint. The bellmouth analysis examines those elements up to the interface with the capstan beam. Lateral loads at the joint are not considered, as the capstan beam is free to roll upon its supports.

The critical condition in the design of this mounting is the longitudinal sway. Contingent on the vertical and longitudinal loadings from Condition 3B, sections B-B and D-D dominate the overall characteristics of the attachment.

Preliminary analysis of the mounting indicates that ribbing is essential in the y-z plane. The rib thicknesses can vary from 0.250 at the box section to 0.125 at the tapered end of the mounting beam. The fitting is manufactured from an aluminum alloy high-strength casting, A21180-C355.

Bellmouth Analysis

Two classifications of moment distribution exist in the analysis of the bellmouth mounting:

1. Overall bending moments which are a function of beaming loads where moments exist only at segments along the local span length of the beam.

2. Constant bending moments that result also in beaming loads, but in certain planes are constant along the beam span and therefore, in these arbitrary planes, are classified as not being localized.

Loads existing in the y-z plane (M_x) and in the x-y plane (M_z) provide localized moments only and are categorized as overall bending moments or as local moments.

Loads generated in the x-z plane provide constant bending moments in the x-y plane or in the y-z plane (M_y).

Characteristics of the bellmouth are as follows:

MIL-A-21180C high-strength castings
Class 1B
F.S. = 1.5
Casting factor = 1.5
Fitting or bearing factor = 1.25
ULT load = L.L. x F.S. x casting factor x fitting factor

F_{t_u}	41 Ksi
F_{t_y}	31 Ksi
F_{c_y}	31 Ksi
F_{s_u}	29 Ksi
$F_{BRG\ ULT}$	
e/D = 1.5	57 Ksi
e/D = 2.0	74 Ksi
$F_{BRG\ YLD}$	50 Ksi
E	10,100 Ksi

Longitudinal Sway (Condition 3B)

$$\sigma_{ULT} = \pm Mc/I + P/A$$

where $F_{t_u} = 41$ Ksi

$$MS_{ULT} = \frac{F_{t_u}}{\sigma_{ULT}} - 1 > 1.00 \text{ bending or tension}$$

$$1 = \frac{F_{t_u} - \sigma_{ULT}}{\sigma_{ULT}}$$

$$F_{t_u} = 2\ \sigma_{ULT}$$

$$\sigma_{ULT_{W.S.}} = F_{t_u}/2 = 20.5 \text{ Ksi bending or tension}$$

$$MS_{ULT} = \tau_{t_u}/\tau_{ULT} - 1 = 0.50$$

$\tau T_u = 1.5 \quad \tau_{ULT}$

or

$\tau_{ULT_{W.S.}} = 29/1.5 = 19.3$ Ksi shear

$\sigma_{ULT_{W.S.}} = 41/1.5 = 27.3$ Ksi compression

$\sigma_{ULT_{W.S.}} = 57/1.5 = 38$ Ksi bearing

MS (tension and bending) $\geq$ 1.0

MS (compression, shear, or bearing) $\geq$ 0.50

$F_{ULT}/F_{YLD} = 41/31 = 1.33$

$F_{ULT}/F_{YLD} \leq$ ultimate criteria

ULT = 1.5 x 1.5 x L.L. = 2.25 L.L.

ULT = 2.25 x 1.25 x L.L. = 2.82 L.L.

Longitudinal Sway in the x-z Plane (Condition 3B)

Bending and tension:

$W_{cz} = 12.55 \times 2.25/3.75 = 7.6K$

$W_x = 46.80 \times 0.60 = 28.1K$

Analysis of Section B-B (Figure 157)

$M_y = 2.5 \times 28.1 = 70.25K$ in.

$P_z = 7.6K$

Analysis of one lug only,

$M_y = 35.13K$ in., $P_z = 3.8K$

$$\sigma = \pm \frac{6(35.13)}{1.65(2.9)^2} + \frac{3.8}{2.9 \times 1.65}$$

$\sigma =$ 15.8 Ksi

$$MS = \frac{20.5}{15.8} - 1 = 0.30$$

Section must measure 2.90 x 1.65.

Analysis of Section F-F (Figure 157)

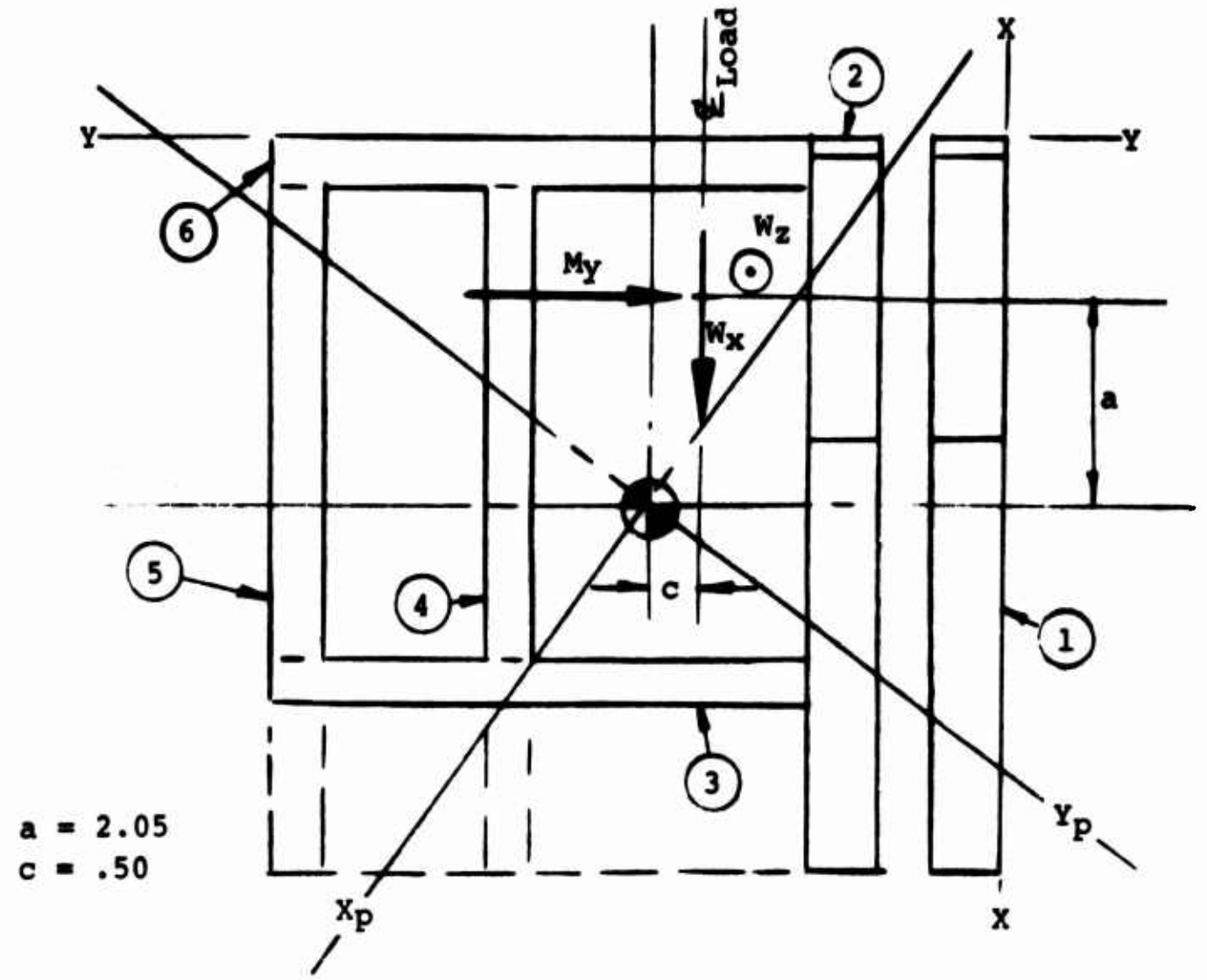


Figure 159. Section F-F of Drawing SK17542.

TABLE XXVII
FACTORS AND VALUES OF SECTION F-F IN x AXIS(Fig. 157)

Item	b	h	A	x	Ax	Ax^2	I_o	I_y
1	0.750	4.30	3.225	5.10	16.448	83.885	4.968	88.853
2	0.750	4.30	3.225	5.10	16.448	83.885	4.968	88.853
3	5.50	0.50	2.750	5.35	14.713	78.715	0.057	78.772
4	0.50	4.60	2.300	2.80	6.440	18.032	4.056	22.088
5	0.50	4.60	2.300	2.80	6.440	18.032	4.056	22.088
6	5.50	0.50	2.750	0.25	0.688	0.172	0.057	0.229
Σ	-	-	16.550	-	61.177	-	-	300.883

$\bar{x}$ = 3.696 in.

I_y = 300.883 - (16.55)(3.696)^2 - 74.804 in.^{4}

TABLE XXVIII
FACTORS AND VALUES OF SECTION F-F IN y AXIS (Fig.157)

Item	b	h	A	y	Ay	Ay^2	I_o	I_x
1	4.30	0.75	3.225	.375	1.2094	0.454	0.151	0.605
2	4.30	0.75	3.225	1.625	5.2406	8.5160	0.151	8.667
3	0.50	5.50	2.750	4.75	13.0625	62.0469	6.932	68.979
4	4.60	0.50	2.300	5.10	11.7300	59.8230	0.048	59.871
5	4.60	0.50	2.300	7.35	16.9050	124.2518	0.048	124.299
6	0.50	5.50	2.750	4.75	13.0625	62.0468	6.932	68.979
Σ	-	-	16.550	-	61.2100	-	-	331.400

$\bar{y}$ = 3.698 in.

I_x = 331.40 - 16.55(3.698)^2 = 105.075

$\Sigma A\bar{x}\bar{y}$ = 186.223

I_{x-1} = 186.223 - 16.55 (3.698 x 3.696)

I_{xy} = -39.979

$$\text{Tan } 2\phi = \frac{2\ I_{xy}}{I_y - I_x} = \frac{2\ (-39.979)}{105.075 - 74.804}$$

Tan 2ϕ = -2.6414

2ϕ = -69° 16'

ϕ = -34° 38'

$I_{yP} = Iy \cos^2\phi + Ix \sin^2\phi - 2\ I_{xy} \sin\phi \cos\phi$

$I_{xP} = Iy \sin^2\phi + Ix \cos^2\phi + 2\ Ixy \sin\phi \cos\phi$

$I_{yP} = 74.804\ (0.8228)^2 + 105.075\ (0.5683)^2 -$

$2(-39.979)\ (0.8228 \times 0.5683)$

I_{yP} = 121.952

$I_{xP} = 74.804\ (0.5683)^2 + 105.075\ (0.8228)^2 + (-37.388)$

I_{xP} = 57.891

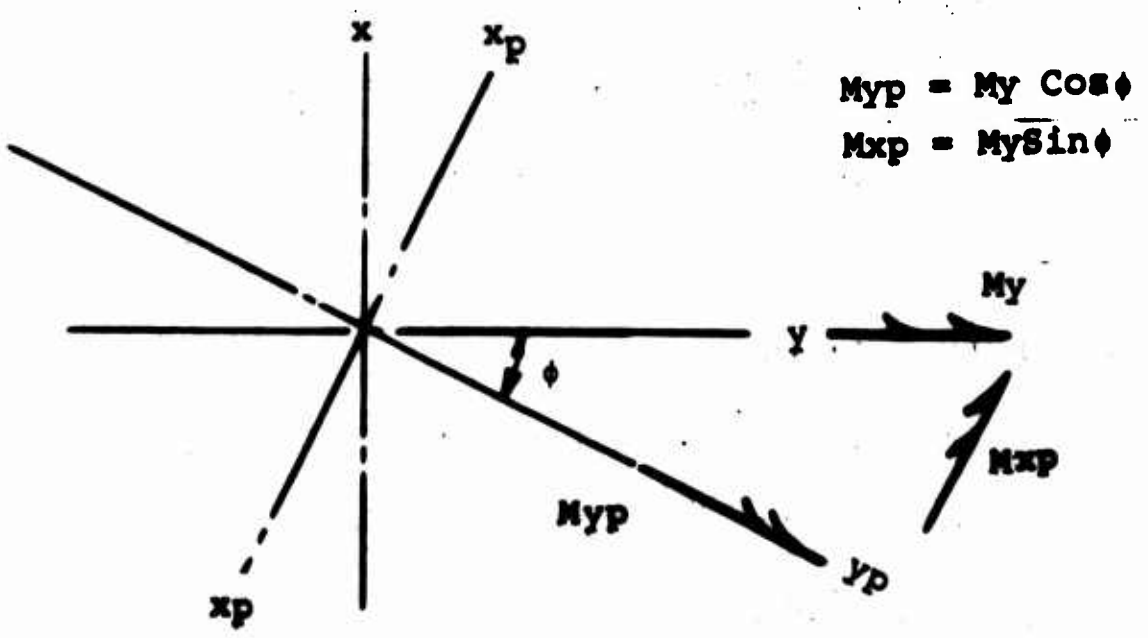


Figure 160. Loading Coordinate System for Section F-F.

W_{cz} = 7.6 K

W_x = 28.1 K

M = 28.1 x 4.5 + 7.6 (2.05) = 131.4 K in.

M_{yp} = 121.4 (0.823) = 108 K in.

M_{xp} = 131.4 (0.568) = 75 K in.

$$\sigma = \frac{108\ (0.60)}{121.952} + \frac{75\ (5.3)}{57.891}$$

σ = 7.54 Ksi

σ = 7.6/16.55 = 0.46 Ksi

$\Sigma\sigma$ = 8.00 Ksi

$$MS = \frac{20.5}{8} - 1 = 1.56$$

The section can be reduced; use ribs at 0.25 inch rather than 0.50 inch.

Analysis of Section D-D (Figure 157)

Overall length = 11 in.

My = 28.1 x 9 + 7.6 x 5 = 291 K in.

$$\sigma = \frac{6\ (291)}{2\ (0.75)\ (8)^2} + \frac{7.6}{2\ (0.75)\ x\ 8}$$

σ = 18.2 + 0.63 = 18.83 Ksi

or

Overall length = 12 in.

M = 28.1 x 9 + 7.6 x 5.8 = 296 K in.

$$\sigma = \frac{6\ (296)}{2\ (0.60)\ (9.5)^2} = 16.2 \text{ Ksi}$$

$$MS = \frac{20.5}{16.2} - 1 = 0.26$$

Use 12-inch overall length and 0.60-inch gage.

Analysis of Section H-H (Figure 157)

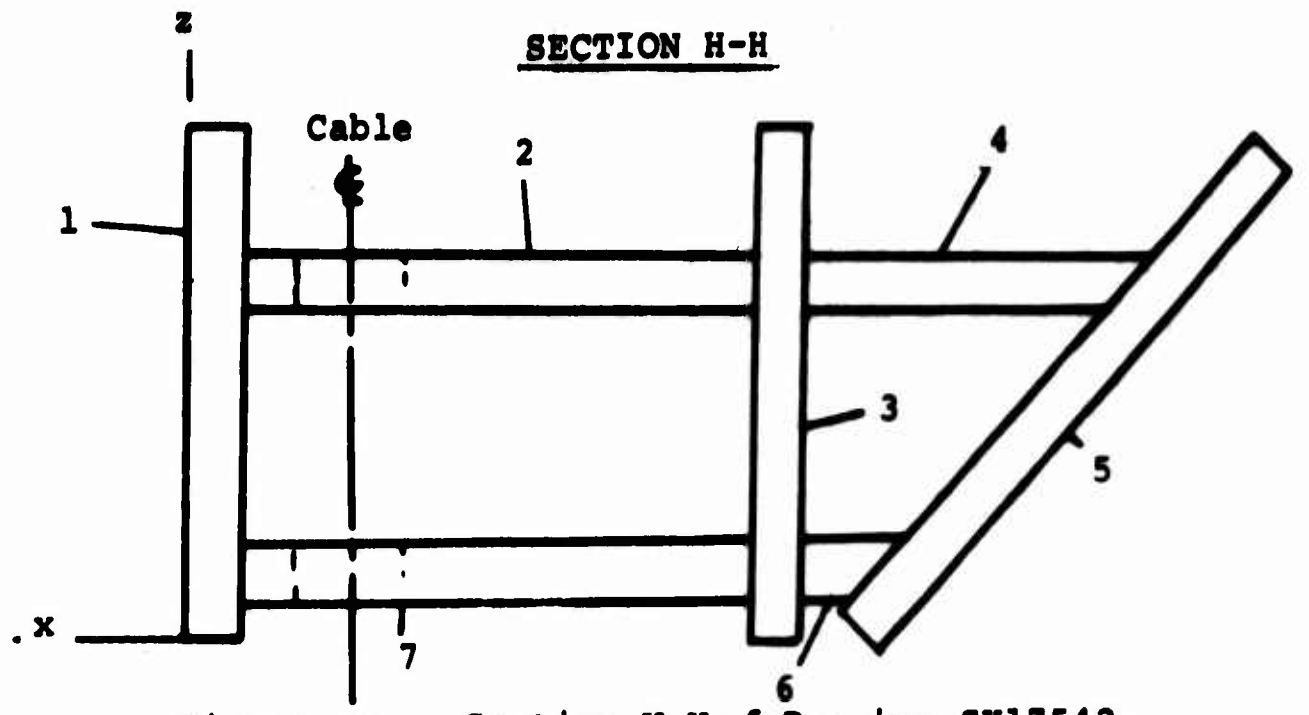


Figure 161. Section H-H of Drawing SK17542.

TABLE XXIX

FACTORS AND VALUES OF SECTION H-H IN z AXIS

Item	b	h	A	z	Az	Az^2	I_o	I_x
1	0.50	4.40	2.20	2.20	4.84	10.648	3.550	14.198
2	4.60	0.50	2.30	3.10	7.13	22.103	-	22.103
3	0.50	4.40	2.20	2.20	4.84	10.648	3.550	14.198
4	2.90	0.50	1.45	3.10	4.495	13.935	-	13.935
5	0.50	5.50	2.75	2.20	6.05	13.310	6.932	20.242
6	0.60	0.50	0.30	0.60	0.18	0.108	-	0.108
7	4.60	0.50	2.30	0.60	1.38	0.828	-	0.828
Σ	-	-	13.50	-	28.915	-	-	85.612

$\bar{z} = 2.143$ in.

$I_x = 85.612 - (13.50)\ (2.142)^2 = 23.674$ in. 4

TABLE XXX
FACTORS AND VALUES OF SECTION H-H IN x AXIS

Item	b	h	A	x	Ax	Ax^2	I_o	I_z
1	4.40	0.50	2.20	0.250	0.550	0.138	0.55	0.688
2	0.50	4.60	2.30	2.90	6.670	19.343	4.056	23.399
3	4.40	0.50	2.20	5.40	11.880	64.152	0.55	64.702
4	0.50	2.90	1.45	7.10	10.295	73.095	1.016	74.111
5	5.50	0.50	2.75	8.05	22.138	178.211	-	178.211
6	0.50	0.60	0.30	5.90	1.770	10.443	-	10.443
7	0.50	4.60	2.30	2.90	6.670	19.343	4.056	23.399
Σ	-	-	13.50	-	59.973	-	-	374.953

$\bar{x} = 4.442$ in.

$I_z = 374.953 - (13.50)\ (4.442)^2 = 108.58$ in.^{4}

$\Sigma Axy = 133.705$

$I_{xy} = Axy - A\bar{x}\bar{z} = 133.705 - 128.449$

$I_{xy} = 5.256$

$$\text{Tan } 2\phi = \frac{2\ I_{xz}}{I_z - I_x} = \frac{2\ (5.256)}{84.91}$$

$\text{Tan } 2\phi = 0.124$

$2\phi = 7°\ 4'$

$\phi = 3°\ 32'$

$I_{xP} = \bar{I}x \cos^2\phi + \bar{I}z \sin^2\phi - 2\ \bar{I}xz \sin\phi \cos\phi$

$I_{xP} = 23.674\ (0.998)^2 + 108.58\ (0.0616)^2 - 2\ (5.256)$
$(0.998)\ (0.0616)$

$I_{xP} = 23.345$

$I_{zP} = \bar{I}x \sin^2\phi + \bar{I}z \cos^2\phi + 2\ \bar{I}xz \sin\phi \cos\phi$

$I_{zP} = 23.674\ (0.0616) + 108.58\ (0.998)^2 + 2\ (5.256)$
$(0.998)\ (0.0616)$

I_{zp} = 108.88

Section ribs could be reduced to 0.25 inch from 0.50 inch. Ribs are required in the area for stability reasons.

Analysis of Section K-K (Figure 158)

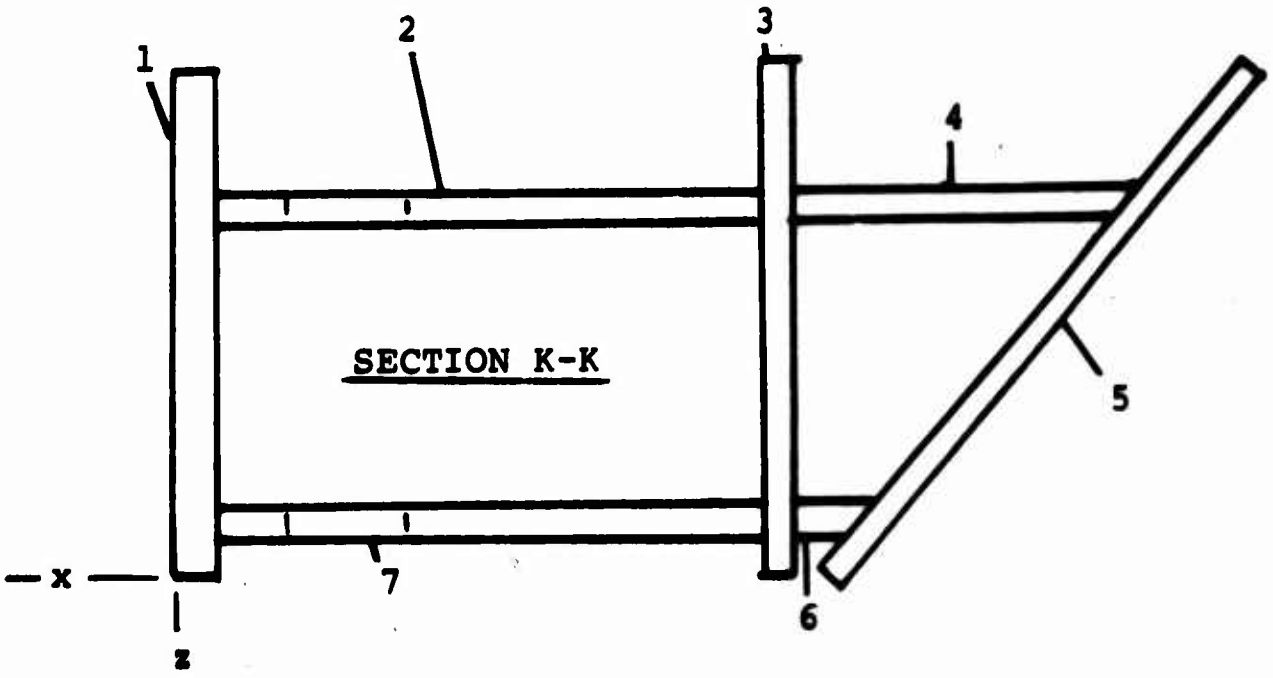


Figure 162. Possible Configuration of Section K-K of Drawing SK17542.

TABLE XXXI
FACTORS AND VALUES OF SECTION K-K

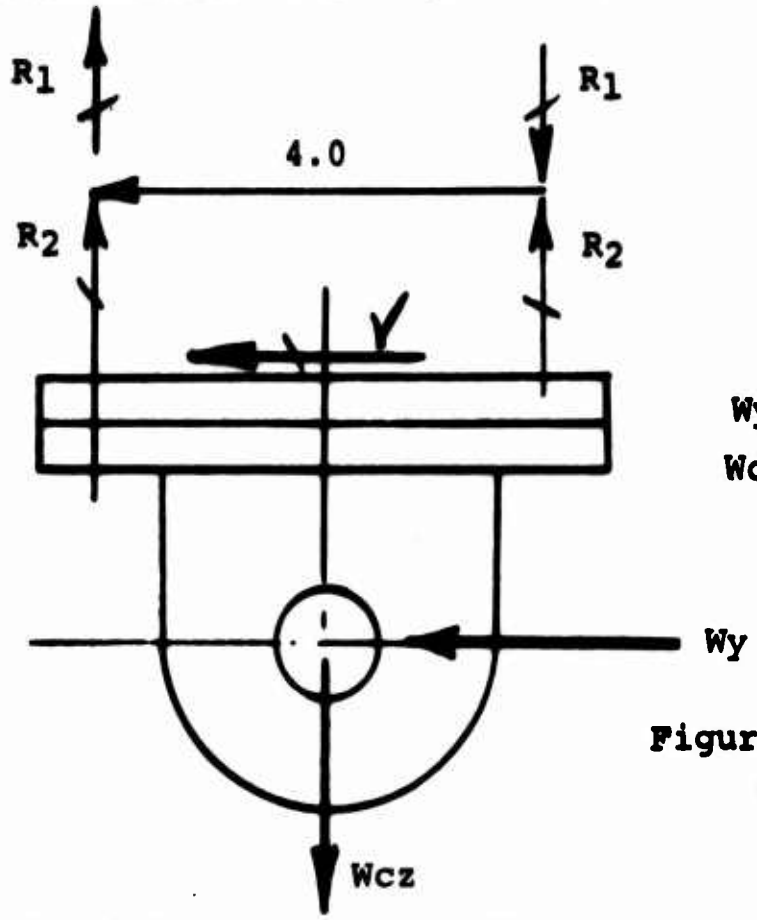
Item	b	h	A	Z	Az	Az^2	I_o	I_x
1	0.40	4.40	1.760	2.20	3.872	8.518	2.839	11.357
2	5.00	0.30	1.500	3.20	4.800	15.360	-	15.360
3	0.30	4.40	1.320	2.20	2.900	6.380	2.129	8.509
4	2.90	0.30	0.870	3.20	2.784	8.909	-	8.909
5	0.30	5.80	1.740	2.20	3.828	8.422	4.878	13.30
6	0.60	0.30	0.180	0.50	0.09	0.045	-	0.045
7	5.00	0.30	1.500	0.50	0.750	0.375	-	0.375
Σ	-	-	8.870	-	19.020	-	-	57.855

$\bar{z}$ = 2.144 in.

$I = 57.855 - (8.87)\ (2.144)^2$

$I_x = 17.082\ in.^4$

Analysis of Section C-C (Figure 153)



Wy = 46.8 KIPS
Wcz = 12.55 KIPS
(Cond 3B)

Figure 163. Section C-C of Drawing SK17542.

Vertical load Wcz is reacted through the bearings into the bearing webs and box bolts and finally into the side frames R_1 and R_2.

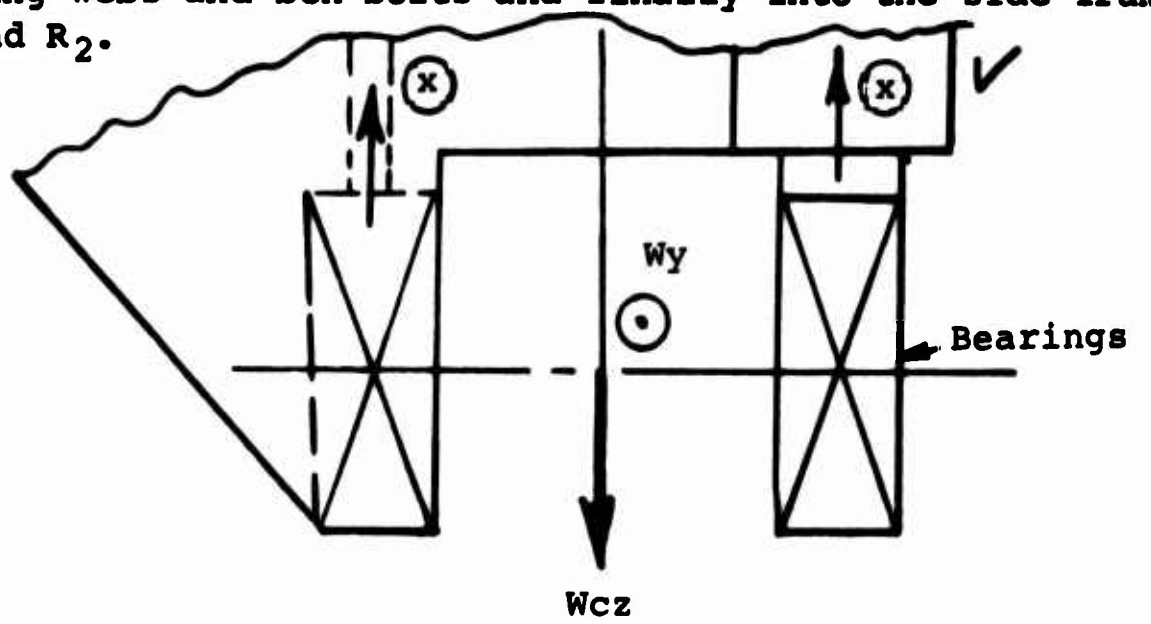


Figure 164. Bearing Webs and Box Bolts of Section C-C.

M = 2.9 x 46.8 - 135 K in.

R_2 = 12.55/2 x 2 = 3 K

R_1 = 135/2 x 4 = 16.8 K

$$V = 46.8/2 \times 2 = 11.7$$

$$P_{TMAX} = 8.4 + (8.4^{-2} + 11.7^{-2})$$

$$P_{TMAX} = 22.8 \text{ K}$$

$$P_{SMAX} = 14.4 \text{ K}$$

F_{t_u} = 180 Ksi ½ - 20 bolts

P_T = 26.8 K

P_S = 20.2 K

$$MS = \frac{26.8}{26.8} - 1 = 0.17$$

Use four ½ - 20 bolts.

Analysis of Section D-D (Figure 153)

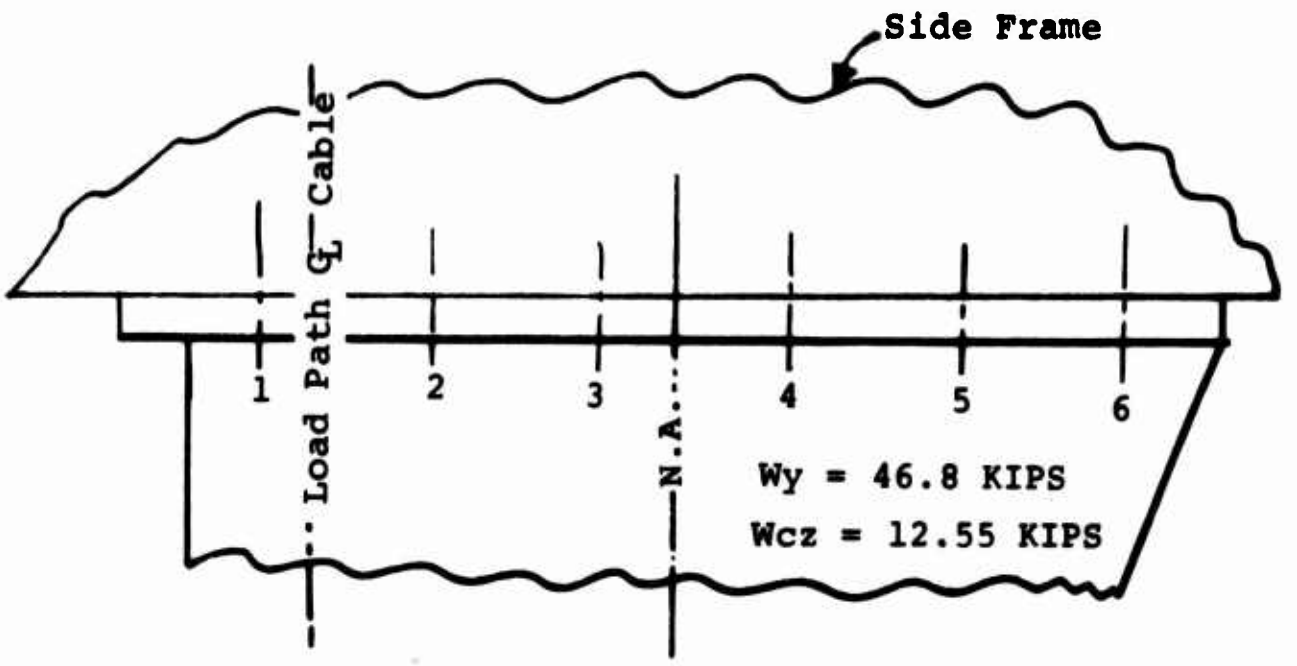
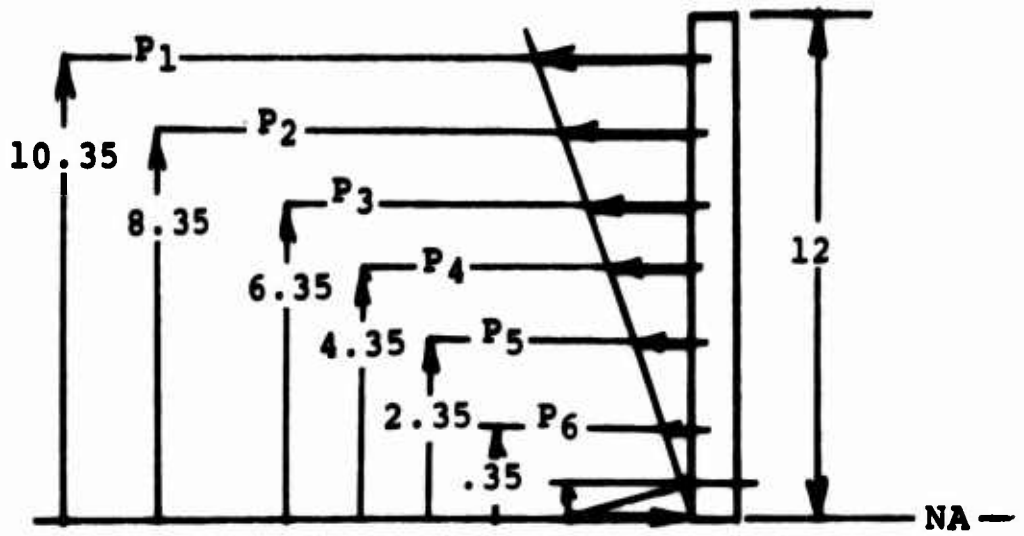


Figure 165. Section D-D of Drawing SK17542.

Assume that 6 bolts attach the bellmouth fitting to the capstan winch side frame.



The above sketch depicts bolt shear at stress loading.

$I = \Sigma Ad^2$

$I = \ell_1^2 + \ell_2^2 + \ell_3^2 + \ell_4^2 + \ell_5^2 + \ell_6^2 = 241.74$

$M_y = 9.60\ (12.55) + 46.80\ (9) = 537$ K in.

$P_n = M_y\ \ell_n / \Sigma d^2_n$

$P_n = 537\ \ell_n / 241.74 = 2.22\ \ell_n$

$P_1 = 2.22 \times 10.35 = 22.98$ K

$P_2 = 0.807\ (22.98) = 18.54$ K

$P_3 = 0.614\ (22.98) = 14.10$ K

$P_4 = 0.418\ (22.98) = 9.01$ K

$P_5 = 0.226\ (22.98) = 5.19$ K

$P_6 = 0.032\ (22.98) = 0.74$ K

71.16 K load in flange

Shear load per bolt = 46.8/6 = 7.45 K

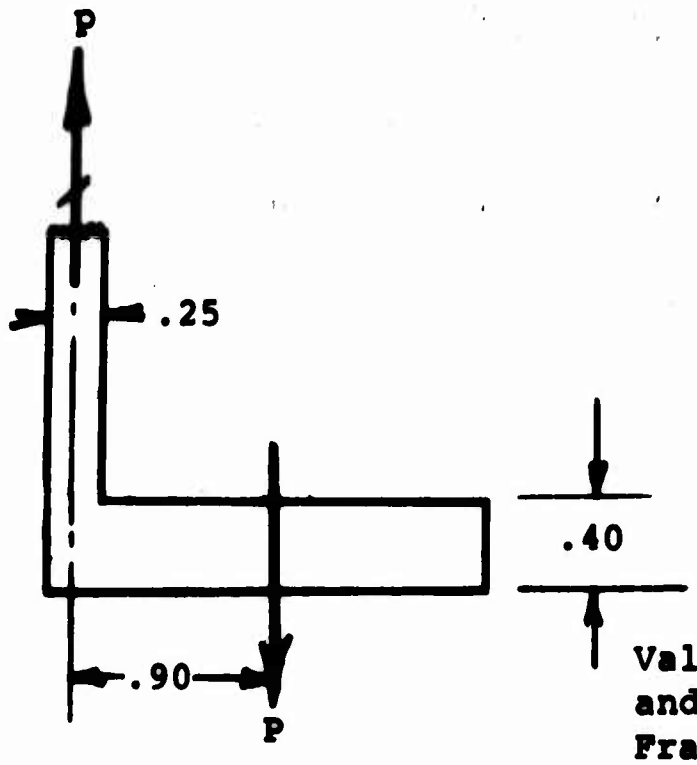
$P_T = 11.49 + (11.49^{-2} + 7.45^{-2})^{\frac{1}{2}}$

$P_T = 25.1$ K

$P_S = 13.6$ K

$$MS = \frac{26.8}{25.1} - 1 = 0.06$$

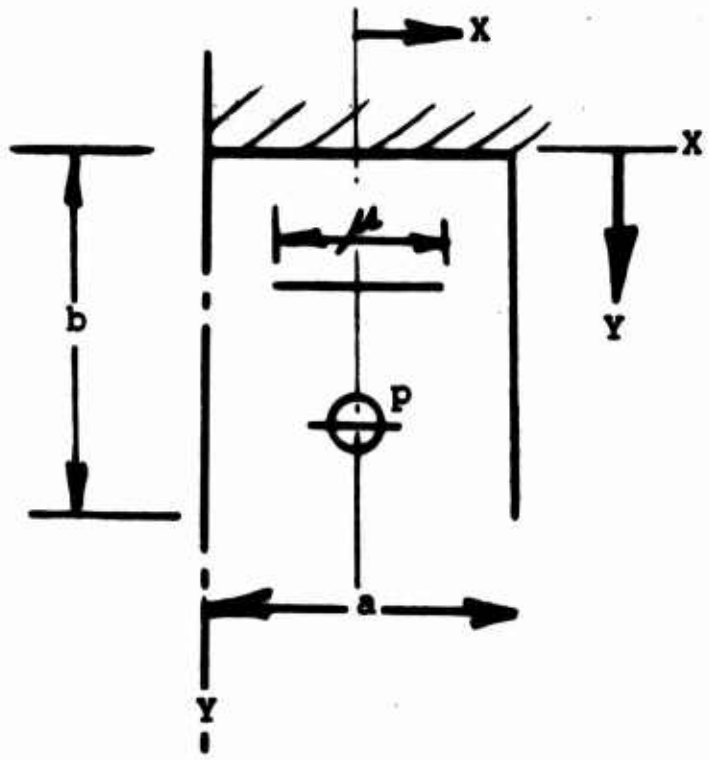
½ - 20 180 Ksi bolts are critical in tension where T_{ALLOW} = 26.8 K.



Material: Aluminum Extrusion
F_{t_u} = 57 Ksi
F_{t_y} = 42 Ksi

Reference Figure 165

Valid for Right and Left Side of Frame



Semi-infinite Plate
Edge y = 0 Built in

$\mu/a = 0$
X = a/z
Y = 0
My = .318 P

$b/a \approx 1.25$
or
$b/a \approx 1.50$

Figure 166. Side Frame Flange (Longitudinal Sway, Condition 3B).

Analysis of Side Frame Flange (continued)

$a = 1.00 \qquad b = 1.50 \qquad b/z = 1.5$

$q = 25.1/1.5 = 16.7 \text{ K/in.}^2$

$\beta_3 = 0.0842 \quad \beta_5 = 0.042$

Assume $q = 0.94 \times 16.7 = 15.6 \text{ K/in.}^2$

$Mx = 0.0842 \ (15.6) \ (1)^2 = 1.32 \text{ in.-K/in.}$

$My = 0.042 \ (15.6) \ (1)^2 = 0.66 \text{ in.-K/in.}$

$t^2 = 6(1.32)/57 = 0.140$

$t = 0.375$ free end, horizontal leg, $x = a/2$, $y = b$

$t^2 = 6 \ (0.66)/57 = 0.069$

$t = 0.265$ vertical leg, $x = 0$, $y = 0$

$I_{ft} = 0.70$

$\sigma = 6 \ (0.66)/(1)^2 \ (0.700) \times 31.7/1 \times 0.700$

$\sigma_{V.L.} = 51.2$ Ksi vertical leg

$\sigma_{H.L.} = 6 \ (1.32)/ \ (0.40)^2 = 49.5$ Ksi horizontal leg

$\sigma t_{V.L.} = 0.700$

$\sigma t_{H.L.} = 0.40$

$MS_{V.L.} = \frac{57}{51.2} - 1 = 0.12$

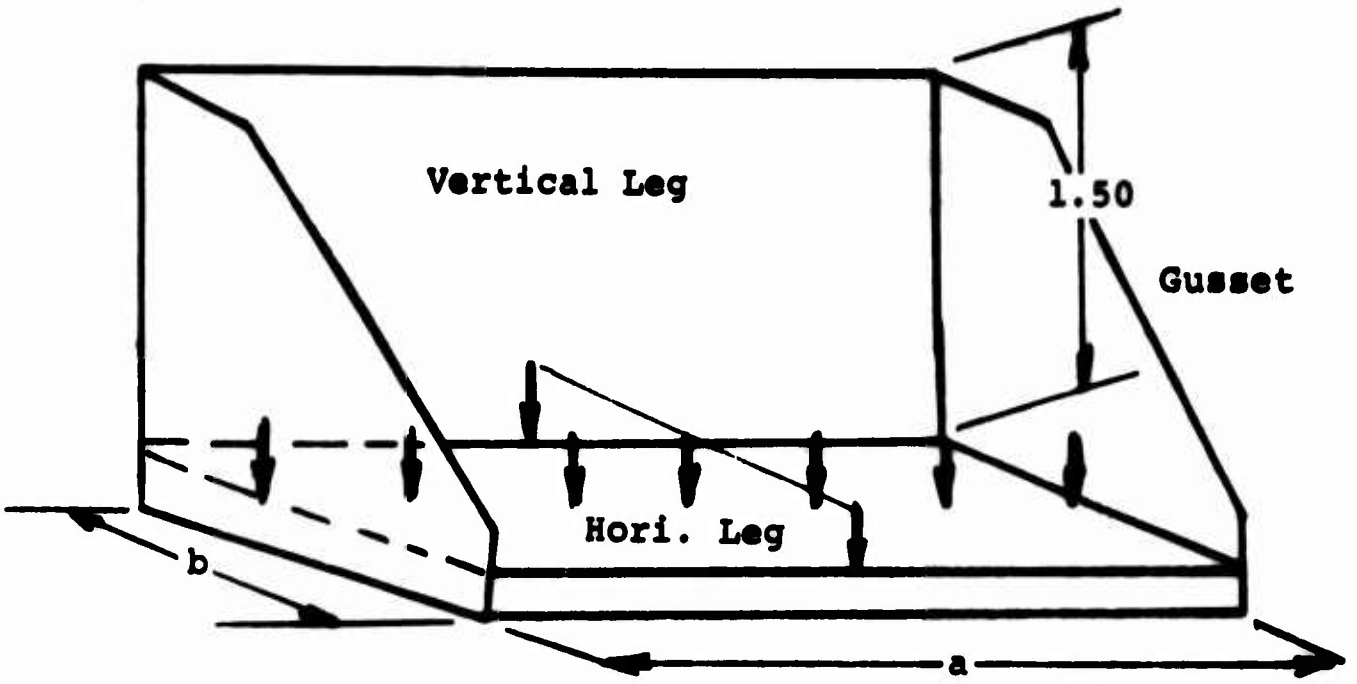


Figure 167. Side Frame Flange.

a = 1.00
b = 1.500
Horizontal leg = 0.400
Vertical leg = 0.700
Gusset = 0.275
Gusset = 1.50

Gusset size:

x = 0
y = b/2
q = 15.6 K/in.^{2}
β_2 = 0.012
My = 0.012 (15.6) (1)^2
My = 0.187 in.-K/in.
Use t = 0.275.

$$\sigma = \frac{6(0.187)}{1.5\ (0.275)^2} + \frac{31.7}{1.5 \times 2 \times 0.275}$$

σ = 48.10 Ksi

$$MS = \frac{57}{48.2} - 1 = 0.180$$

WINCH DRIVE SYSTEM GEAR ANALYSIS

The spur stresses were computed by the Boeing-Vertol Program No. 503. The bevel gear stress analysis is presented in this report. All gears (spurs and bevels) are made from SAE 9310 steel, carburized and ground.

The calculated spur gear endurance stresses, shown in Table XXXII, were compared to the S-N curve, Figure 168, to establish the number of allowable operational cycles (Table XXXIII). The minimum design value curves were used.

The planet gears B, E, and G are loaded in both directions (reversed tooth bending) during the hoisting operation, while the remaining gears are loaded in only one direction. For the gears with reversed tooth bending, the allowable is reduced as shown in Figure 168.

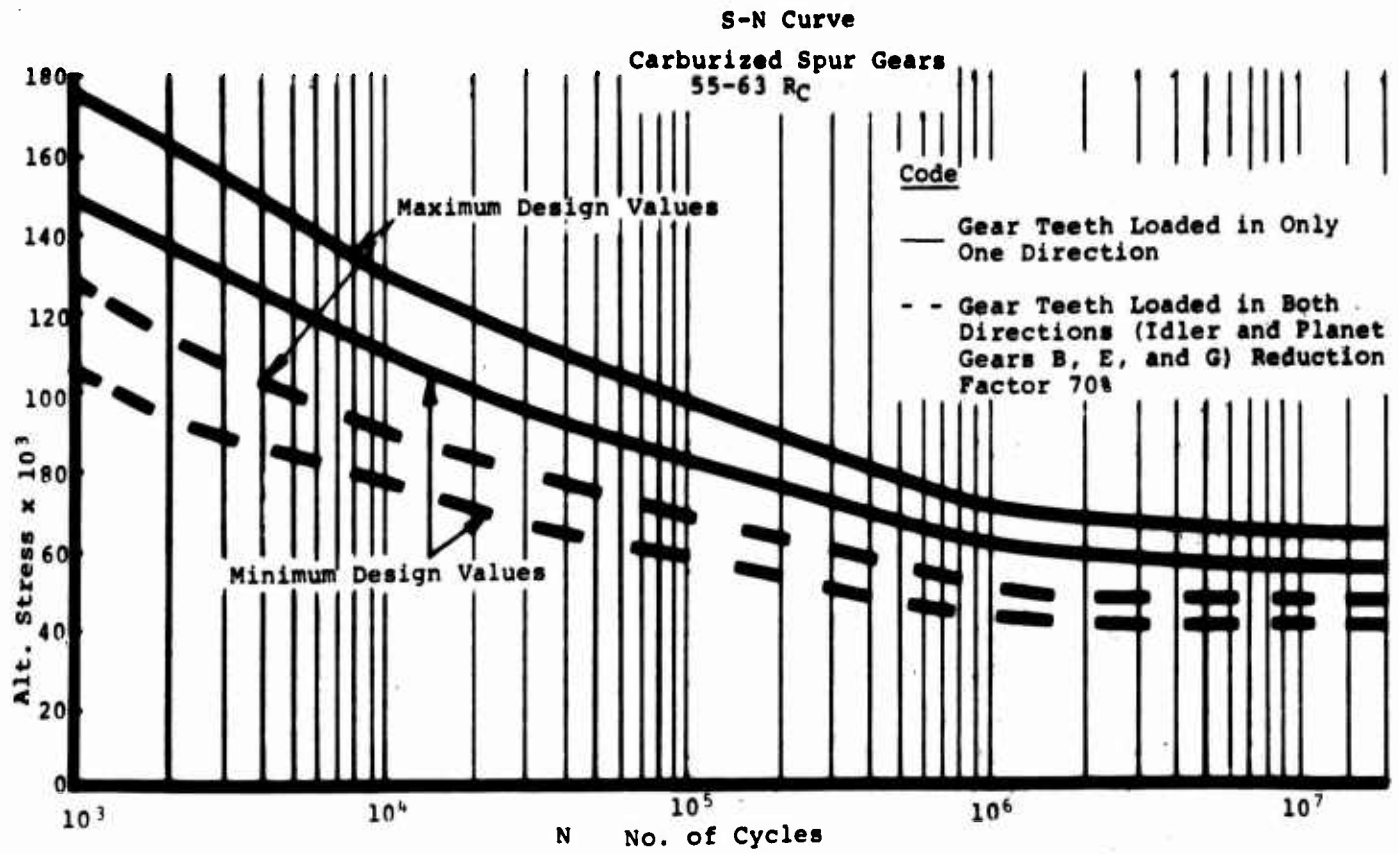


Figure 168. Tooth Bending Stress Case.

TABLE XXXII
DRIVE SYSTEM GEAR SUMMARY

Gear Geometry								
Gear Name	Letter	D_P Pitch dia-in.	P_D Diametral Pitch	F Face in.	Pressure Angle degrees	N No. of Teeth	Y_K	RP
First-stage ring gear	A	13.25	8	2.25	25	106	-	12.0
First-stage planet gear (idler)	B	4.50	8	2.25	25	36	0.5275	35.5
First-stage sun gear	C	4.25	8	2.25	25	34	0.5248	37.6
Second-stage ring gear	D	9.00	11	1.625	25	99	-	0
Second-stage planet gear	E	2.910	11	1.625	25	32	0.5062	78.8
Second-stage sun gear	F	3.180	11	1.625	25	35	0.5143	144.0
Third-stage ring gear	D	9.00	11	0.875	25	99	-	0
Third-stage planet gear	G	3.64	11	0.875	25	40	0.5162	212.0
Third-stage sun gear	H	1.73	11	0.875	25	19	0.4518	894.0
Clutch gears	I	6.00	11	0.625	25	66	0.5713	894.0
	J	6.00	11	0.625	25	66	0.5713	894.0
Bevel gear	K	5.500	6	0.900	pressure angle = 20 spiral angle = 35	64	J=0.210 from Gleason Manual	894.0
Bevel pinion	L	2.500	6	0.900		28		1965.0

TABLE XXXII
DRIVE SYSTEM GEAR SUMMARY

y			Endurance Condition					
ure s	N No. of Teeth	Y_K	RPM	Pitch Line Velocity ft/min	Torque in.-lb	W_t lb	S_t Tension Stress psi	S_c Hertz Stress
	106	-	12.05	41.8	238,000	8970	planet	critical
	36	0.5275	35.5	41.8	-	8970	60,400	222,000
	34	0.5248	37.6	41.8	76,280	8970	60,700	222,000
	99	-	0	0	56,200	3130	planet	critical
	32	0.5062	78.8	120.0	-	3130	41,800	183,000
	35	0.5143	144.0	120.0	19,900	3130	41,200	183,000
	99	-	0	0	22,400	1240	planet	critical
	40	0.5162	212.0	405.0	-	1240	30,200	178,000
	19	0.4518	894.0	405.0	3210	1240	34,500	178,000
	66	0.5713	894.0	1400.0	3210	1070	32,960	128,000
	66	0.5713	894.0	1400.0	3210	1070	32,960	128,000
re	64	J=0.210 from Gleason Manual	894.0	1290.0	3210	1170	pinion critical	210,000
	28		1965.0	1290.0	1460	1170	26,100	210,000

2

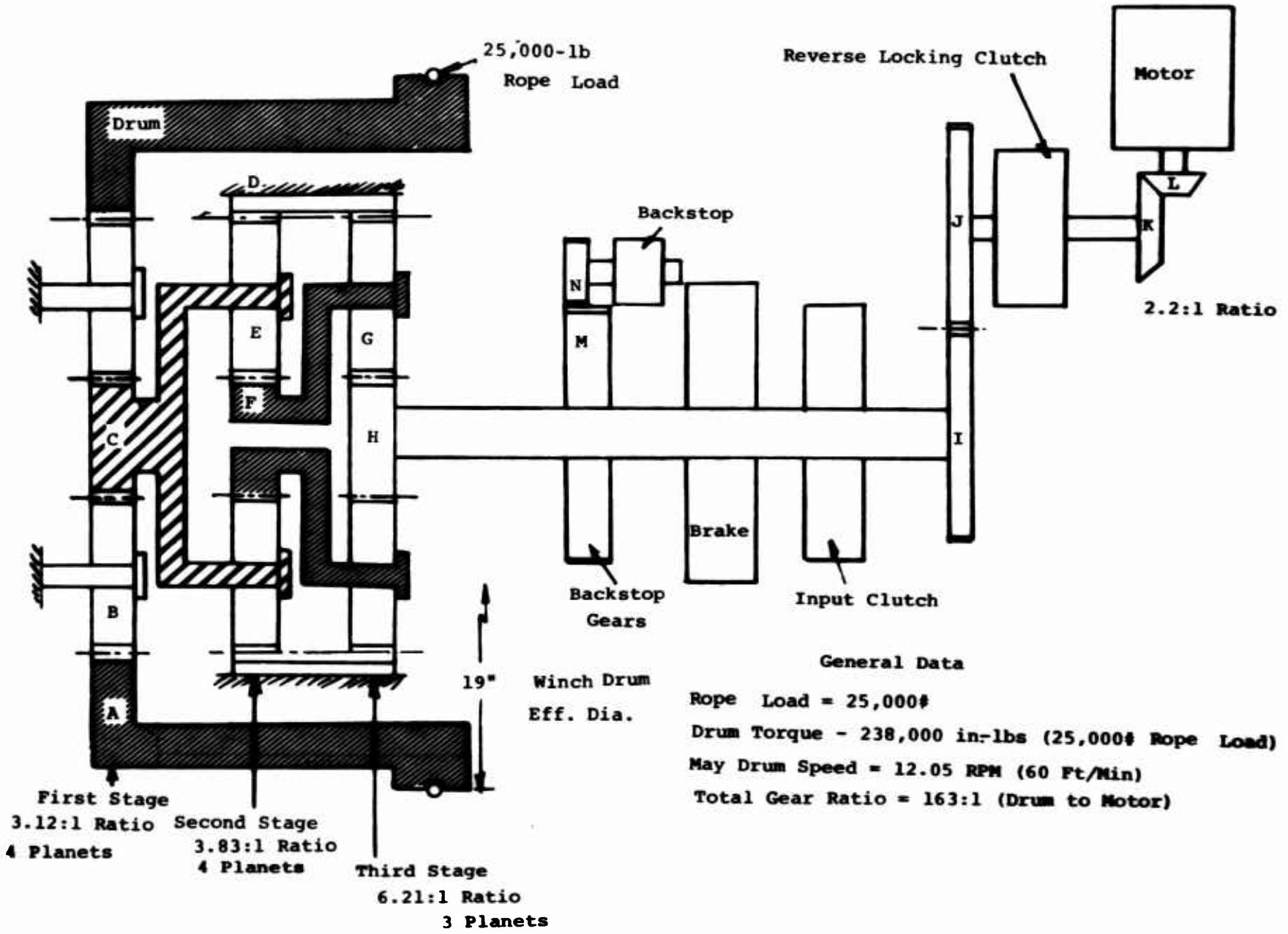


Figure 169. Winch System Drive Schematic.

TABLE XXXIII

DRIVE SYSTEM GEAR OPERATIONAL CYCLE DATA

	'n' Number of Tooth Loads		Number of allowable operational cycles based on,			
Gear Letter	'n' Per One Drum Rev Cycles	'n' Per 3600 Operation Cycles x 10^{-6}	Tension Stress Cycles	Hertz Stress Cycles	Ultimate Tension Stress psi	M.S. Ultimate R_c=55 Min F_{tu}=304,000 psi
A	1.0	0.087	-	planet	critical	
B	5.55	0.0482	5230	22,400	226,000	0.35
C	12.5	0.109	33,000	22,400	228,000	0.33
D	12.5	1.09	-	planet	critical	
E	19.2	1.67	3880*	22,400	157,000	0.94
F	35.2	3.06	infinite	22,400	154,000	0.97
D	35.8	3.12	-	planet	critical	
G	59.2	5.15	infinite	22,200	113,000	1.69
H	187.0	16.30	infinite	8840	129,000	1.36
I	74.2	6.45	infinite	22,400	124,000	1.45
J	74.2	6.45	infinite	22,400	124,000	1.45
K	74.2	6.45	infinite	8930	pinion critical	
L	163.0	14.20	infinite	4060*	98,000	2.10

*Minimum values (3600 cycles is the design objective)

TABLE XXXIV
DRIVE SYSTEM BACKSTOP GEAR DATA

	Gear Name	Backstop Gear	Backstop Pinion
	Letter	M	N
Gear Geometry	D_P Pitch Dia, in.	12.25	5.125
	P_D Diametral Pitch	8	8
	F Face	1.00	1.00
	ϕ Pressure Angle, degrees	25	25
	N No. of Teeth	98	41
	Y_K	0.5908	0.5550
1.0g Load	Torque, in.-lb	3210	1343
	W_t, lb	523	523
	S_t Tension Stress, psi	7096	7555
	S_c Hertz Stress, psi	61,930	61,930
Ultimate	Ultimate Tension Stress, f_{tu} psi	26,600	26,600
	M.S. Ultimate	10.4	10.4

NOTES:
1. These gears are not part of the drive system; therefore, they are not considered in the endurance analysis.
2. F_{tu} = 304,000 psi

Ref: 4

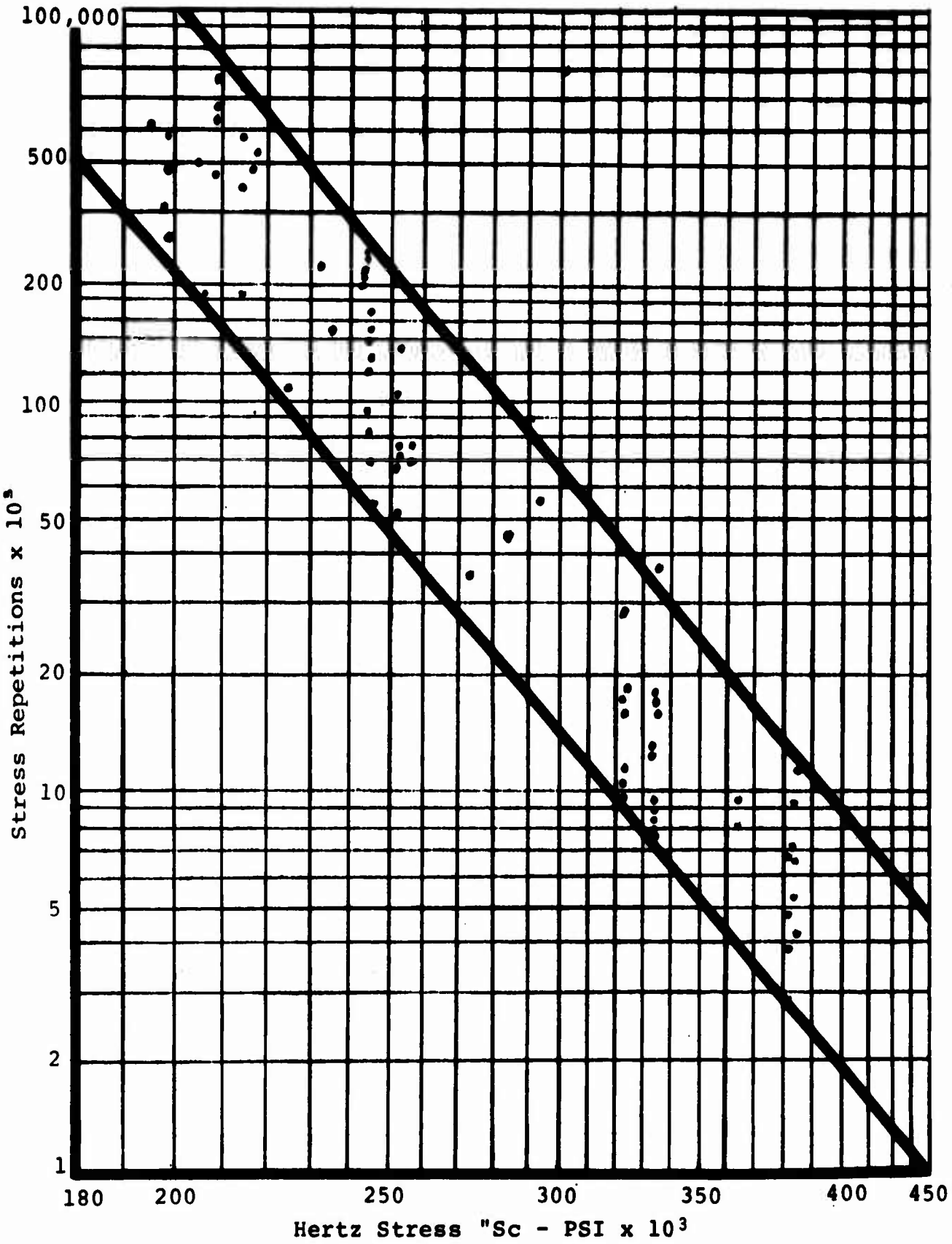


Figure 170. Transmission Gear Life - Surface Fatigue.

The endurance limit for a bevel gear is 20,000 psi (55-63 R_C case carburized).

The calculated spur and hertz stresses, shown in Table XXXII, are compared to Figure 166 to obtain the number of allowable operational cycles. The bottom of the scatter curve is used. The following equations were used to calculate gear tooth stress repetitions for the planetary gear systems:

sun gear load cycles =
(no. of planets) (sun rpm relative to planet carrier rpm)

ring gear load cycles =
(no. of planets) (planet carrier rpm)

planet gear load cycles =
2 x (planet rpm relative to the planet carrier)

Spiral Bevel Gear Data (Gear Letters K and L)

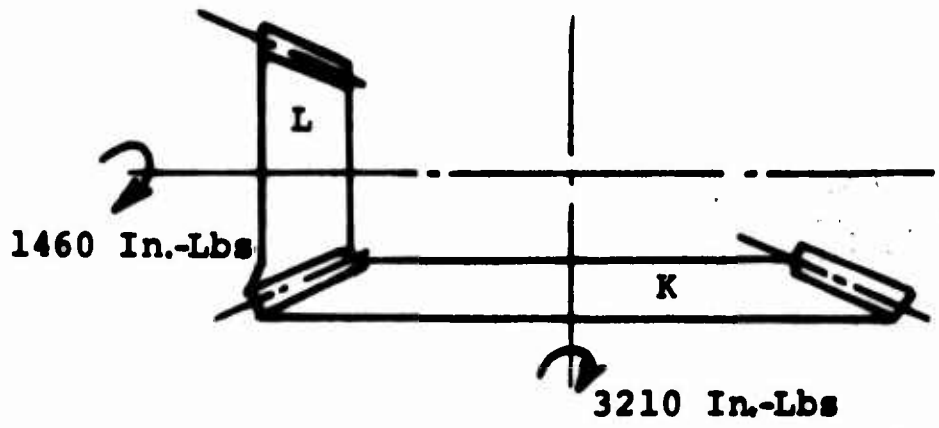


Figure 171. Spiral Bevel Gears (K + L).

Ratio = 2.2:1
Gear pitch diameter = 5.500
Pinion pitch diameter = 2.500
N_P = 15 teeth
N_G = 33 teeth
$P_D = \frac{15}{2.5} = 6.0$
Face = 0.900 in.
W_T = 1460/1.25 = 1170 lb
Shaft angle = 90°
Pressure angle = 20°
Spiral angle = 35°

$J_P = 0.210$
$J_G = 0.210$ geometry factor from Gleason Design Manual

Tension stress - pinion

$$S_t = \frac{W_y \, P_D}{F_d \, J_p} \times K_s \, K_M$$

where

$$K_s = \frac{1}{(P_D)^{0.23}} = \frac{1}{(6)^{0.25}}$$

$$K_s = 0.640$$

$$S_t = \frac{1170 \times 6.0}{0.90 \times 0.210} \times 0.640 \times 1.10 = 26{,}100 \text{ psi}$$

The endurance limit is 30,000 psi.

Hertz stress

$$S_c = 2800 \sqrt{\frac{W_t}{F_d} \times \frac{C_M}{I_G}}$$

where

$C_M = 1.10$

$I_G = 0.11$

$$S_c = 2800 \sqrt{\frac{1170}{0.9 \times 2.5} \times \frac{1.10}{0.102}}$$

$$S_c = 2800 \sqrt{5600} = 210{,}000 \text{ psi}$$

The allowable number of stress repetitions at $S_c = 210{,}000$ psi equals $16 \times 10^{+6}$.

Number of stress repetitions for 3600 operational cycles is $14.2 \times 10^{+6}$.

Allowable number of operational cycles

$$= \frac{16.0}{14.2} \quad (3600) = 4060$$

PROCUREMENT SPECIFICATIONS

The following sample procurement specifications D8-0691 and D8-0692 have been included as guides, and it is not intended that they be construed to be specifications for the final system. Based on specifications used for similar airborne equipment, the guide specifications encompass the minimum elements considered necessary to guarantee required system performance. Structural design criteria (see paragraph 5.2 in D8-0691 and D8-0692 in this report) are not considered complete without including the flight cases covered in the Stress Analysis section of this report. The performance objective, of the contractual work statement, specifies only a vertical lift capacity; it does not refer to load swinging cases or drag acting on load. The load swinging criterion used in this report is a 30-degree cone which is specified in F.A.R. Document 133 (Federal Airworthiness Requirements).

The winch duty cycle has been based on a 60-foot lift height, whereas the design lift height is 150 feet. Thirty-six hundred duty cycles at 150-foot lift height are considered to be unrealistic, since in a given system life the number of missions utilizing the 150-foot capability are estimated to be less than 10 percent and the number of lifts at maximum capacity are estimated to be less than 60 percent of the total missions flown. A duty cycle using 60 feet at a load of 24,500 pounds ± 500 pounds per winch is considered to be conservative.

D8-0691

PROCUREMENT SPECIFICATION,
BEAM ASSEMBLY AND RELEASE UNIT,
20-TON HOIST SYSTEM

THE BOEING COMPANY
VERTOL DIVISION
MORTON, PA.

1.0 PURPOSE

1.1 This specification defines the requirements for the design, manufacture and testing of a beam assembly and release unit to be used in a two-winch, two-suspension rope hoist system which will be fitted to a helicopter for the purpose of lifting, lowering and transporting materiel externally.

2.0 REFERENCES

2.1 The references listed below constitute a part of this specification. In case of discrepancies between specific requirements of the specification and these references, the requirements of this specification shall govern.

SPECIFICATIONS

Military

MIL-S-5002	Surface Treatments (Except Priming and Painting) for Metal and Metal Parts in Aircraft
MIL-B-5087	Bonding, Electrical (For Aircraft)
MIL-C-6021	Castings, Aircraft Structure, General Specification
MIL-D-70327	Drawings, Engineering and Associated Lists
MIL-L-6880	Lubrication of Aircraft, General Specification for
MIL-P-6906	Plates, Information and Identification
MIL-E-7080	Electric Equipment, Piloted Aircraft Installation and Selection of, General Specification for
MIL-S-7742	Screw Threads, Standard, Aeronautical
MIL-I-8500	Interchangeability and Replaceability, Component Parts for Aircraft and Missiles
MIL-D-8513	Drawings and Data Lists, Preparation of Special Support Equipment for Aeronautical and Associated Equipment
MIL-T-9107	Test Reports, Preparation of
MIL-C-5424	Cable, Steel (Corrosion Resisting), Flexible, Preformed (For Aeronautical Use)
MIL-Q-9858	Quality Control System Requirement

MIL-M-9868	Microfilming of Engineering Documents, 35mm, Requirements for
MIL-E-5272	Environmental Testing, Aeronautical and Associated Equipment, General Specification for

Standards

Military

MIL-STD-3	Format for Production Drawings
MIL-STD-130	Identification Marking of U.S. Military Property
MIL-STD-704	Electric Power, Aircraft, Characteristics and Utilization of
MS 33540	Safety Wiring, General Practices for
MS 33586	Metals, Definition of Dissimilar
MS 33588	Nuts and Plate Nuts, Self-Locking, Functional Limitations of

Publications

Military

MIL-HDBK-5	Strength of Metal Aircraft Elements
AF Bulletin No. 23	Material and Process Specification

Air Force-Navy Aeronautical Bulletin

ANA 143	Specifications and Standards - Use of
ANA 147	Specification and Standards of Non-Government Organizations

Drawings

Boeing, Vertol Division

SK 17541	Component Relationship, 20 Ton Hoist System

SK 17544	Beam and Hook Assembly
SK 17543	Release Unit

3.0 DEFINITIONS

3.1 The beam assembly and release unit referred to shall consist of two mechanically actuated cargo hooks secured to the ends of a hollow beam. The beam shall provide the housing for an electromechanical release system and all necessary components to fulfill the detail requirements specified in paragraph 4.0 of this specification.

The release unit referred to shall consist of a drum for electromechanical cable storage and all the necessary components to fulfill the detail requirements specified in paragraph 4.0 of this specification.

3.2 Life - The beam assembly and release unit shall have a minimum service life of 1200 cycles before removal or overhaul and 3600 cycles before retirement.

3.2.1 Duty Cycle - One complete cycle shall consist of

a. Reel out 60 feet, release hooks electrically with each hook loaded to 24,500 ±500 pounds, recock hooks, reload each hook to 24,500 ±500 pounds, reel in 60 feet.

Note: The testing of the beam assembly and release unit shall be integrated with the testing of winches procured under Procurement Specification PS D8-0692.

4.0 DESCRIPTION

4.1 Components - The beam assembly and release unit design shall be based upon the preliminary design shown on Drawing SK 17544 consisting of the following components:

a. Beam (SK 17544)
b. Hooks (SK 17544)
c. Hook Release System (SK 17544)
d. Release Unit (SK 17543)
e. Electrically Actuated Components
f. Electromechanical Conductor Cable

4.1.1 Beam - The beam shall consist of a hollow structure capable of housing, internally, the hook release mechanisms and shall provide a suitable base at each end for the mounting of the cargo hooks. Adequate access shall be provided to ensure inspection or replacement of internal components of the hook release system.

4.1.2 Hooks - The hooks shall be of a mechanical release type. They shall be oriented as shown on Drawing SK 17544. The hooks shall be self cocking after load release and shall have a keeper capable of retaining the load slings with the sling oriented at an angle of 60° from the vertical center line of the hook throat, without imposing any damage to the sling through abrasive action. Independent manual release shall be provided on each hook. The exposed means of achieving the release function shall be plainly marked so as to indicate that the hook release has been reset. Each hook shall have a switch so integrated with the internal operating linkage that it shall provide a signal to indicate to the pilot and crane operator the status of the hook, i.e., safe or unsafe. The hook operating mechanism shall be completely housed and shall be lubricated during assembly. The hooks shall require no lubrication during their service life (i.e., 1200 cycles, ref. paragraph 3.2). Any components which, if they should fail structurally, would result in the dropping of the load shall be subjected to a 100-per-cent X-ray examination before assembly into the hook. Any flaws, inclusions or any other defects considered to impair the strength or fatigue properties of the

material shall be rejected.

4.1.3 Hook Release System - The hook release system design shall be based on that shown in Drawing SK 17544, except that the levered arrangement may be changed provided the design requirements of paragraph 5 are fulfilled. Adequate adjustment of the system shall be provided.

4.1.4 Release Unit - The release unit shall consist of the following components:

4.1.4.1 Storage Reel - The storage reel shall have sufficient capacity to store 160 feet of electromechanical cable having a diameter not less than 1/2 inch.

4.1.4.2 Torque Motor - The torque motor shall be electrically energized and shall be capable of maintaining a torque on the storage reel which shall maintain a tension load at the drum in the electromechanical cable of 110 ±10 pounds when the cable is deployed out 150 feet and shall maintain a tension load in the cable of 50 ±5 pounds at the drum when the cable is fully reeled in. The torque motor shall be provided with a brake which shall be automatically applied when the torque motor is de-energized. The brake shall be designed to slip at a torque which is equivalent to a cable tension of 50 ±5 pounds. Adjustment for the brake shall be provided to permit adjustment during qualification. The torque motor unit shall be a sealed unit requiring no relubrication during its life as defined in paragraph 3.2.

4.1.4.3 Emergency Release Motor - The emergency release motor shall be a mechanical device initiated by a mechanical input from the pilot and/or crane operator stations. Upon initiation, it shall be capable of automatic positive engagement with the storage reel; after engagement, it shall be capable of inducing sufficient torque to the electromechanical cable so as to induce a tensile load into the cable of 300 ±10 pounds under all lengths of cable deployment, i.e., with cable reel empty or cable reel full. The emergency release motor shall impart a minimum of 130 degrees of rotation to the storage reel. The

emergency release motor shall have sufficient energy so as to overcome any resistance created by the torque motor brake, 4.1.4.2 above, and the inertia device, 4.1.4.4 below. The emergency release motor shall be a self-contained, prelubricated, sealed unit requiring no relubrication during its life as defined in paragraph 3.2. Re-energizing of the motor after use shall be accomplished mechanically using standard mechanics tools (i.e., square-ended ratchet socket wrench).

4.1.4.4 Inertia Device - The inertia device shall be mechanical in operation. The inertia device shall lock out automatically when subjected to a velocity, transmitted from the electromechanical cable, of 10 ±2 ft per sec. The lock-out shall be positive and be capable of resisting a force equivalent to a tension of 300 ± 10 pounds in the electromechanical cable under all lengths of cable deployment, i.e., with cable reel empty or cable reel full. The inertia device shall be a self-contained, prelubricated, sealed unit requiring no relubrication during its life as defined in paragraph 3.2.

4.1.4.5 Electromechanical Control Cable - The electromechanical control cable shall comprise a stainless steel center cable, 3/32 inch dia, conforming to MIL-C-5424; fifteen electrical wires, AN20 gauge, conforming to MIL-W-5086A, type II, spirally laid around the center cable; and an outer jacket of braided fiber glass with a Teflon (FEP) filler. The construction of the cable shall be similar to MIL-C-27500 (USAF). The complete electromechanical conductor cable shall not exceed 0.5 inch in dia. An electromechanical disconnect shall be provided as near as is practical to the beam to facilitate removal of the beam from the hoist system. The disconnect shall be capable of withstanding the cable tension forces defined in paragraph 4.1.4.4.

4.1.4.6 Cable Guide and Mounting Provisions - Cable guide and mounting provisions shall be based on the design shown on Drawing SK 17543, unless otherwise specified by the procuring agency.

4.1.4.7 General Provisions - All exposed moving parts of the complete assembly shall be suitably encased to exclude foreign objects. The electrical slip-ring assembly shall be encased separately so as to facilitate inspection of the slip-ring assembly. The encasement of the slip-ring assembly shall be such that exclusion of foreign objects including liquids is guaranteed. Provision for single-point-mode electrical connection shall be provided (reference SK 17545).

4.1.4.8 Electrically Actuated Components - Electrical components of the beam assembly and release unit shall be capable of being operated by utilizing 28-V DC, excepting the torque motor, which shall be 200/115-V AC, 3-phase, 400-cps. Both power supplies shall conform to MIL-STD-704.

5.0 DESIGN REQUIREMENTS

5.1 Mounting - The beam assembly and release unit, unless otherwise specified, shall be mounted as shown on Drawing SK 17541.

5.2 Structural Design Criteria - The beam assembly shall support a vertical load of 25,000 pounds on each hook. The beam assembly shall be also designed to take into consideration the conditions resulting from the integration of the two-point system into a single-point mode. The geometry shown on Drawing SK 17545 shall be used to determine the system single-point-mode loading using a vertical load of 40,000 pounds on the single hook. The above loading shall be subject to a design load factor of 2.5g and a safety factor of 1.5 ultimate unless otherwise specified in this specification; other safety factors and allowable loads and stresses shall be in accordance with MIL-A-8629 (AER), para. 3.2.1. Additionally, the loading cases in the stress analysis shown in this report shall be provided for to the extent specified by the procuring agency.

5.3 Performance - The beam assembly and release unit shall be capable of operating after exposure to the following conditions:

a. Temperature ranging from -65° to 150°F.
b. Relative humidity up to 100 percent
c. Sand and dust encountered in desert areas

5.4 Vibration - The beam assembly and release unit shall be capable of operating after being subjected to vibrations specified in paragraph 6.3.5.

5.5 Capacity and Capability - The capacity and capability shall be as given in paragraph 4.14.

5.6 Life - The beam assembly and release unit shall be designed for a service life and overhaul period as defined in paragraph 3.2.

5.7 Weight and Dimensions - Weight of one complete beam assembly, release unit and 160 feet of electromechanical cable shall not exceed 240 pounds. Physical dimensions

shall not exceed those shown on Drawings SK 17544 and SK 17543 unless otherwise specified.

5.8 Electric Wiring and Components - All wiring and components shall be selected and installed in accordance with MIL-E-7080.

5.9 Lubrication - The beam assembly and release unit shall incorporate self-contained lubrication which shall be in accordance with the requirements of Specification MIL-L-6880 and shall be suitable for use under all operating environments specified herein.

5.10 Screw Threads - Screw threads shall conform to requirements of Specification MIL-S-7742.

5.11 Interchangeability - All parts having the same part number shall be dimensionally and functionally interchangeable.

5.12 Marking

5.12.1 Identification of Product - Parts shall be marked in accordance with Standard MIL-STD-130. Nameplates in accordance with Specification MIL-P-6906 shall be affixed to the components with the following information etched, engraved or otherwise permanently printed:

5.12.1.1 Beam Assembly

Serial No.
Vertol Specification No. P.S.
Stock Number
Manufacturer's Identification
Manufacturer's Part Number
Contract No.
U.S.

5.12.1.2 Release Unit

Serial No.
Vertol Specification No. P.S.
Stock Number

Manufacturer's Identification
Manufacturer's Part Number
Contract No.
U.S.

5.13 Instructions - Brief operating instructions, switch identification, and a wiring diagram shall be permanently mounted on the beam assembly and release unit by using metal foil or metal plate type.

5.14 Corrosion Resistance - Metals shall be chosen for their corrosion resistance; otherwise, they should be finished to resist corrosion.

5.15 Finishes

5.15.1 Surface Finish - Surface finishes of detail parts shall be in accordance with the requirements of MIL-S-5002.

5.15.2 Exterior Finish - The exterior finish shall be medium gray, #36231 per Fed Std 595, unless otherwise specified.

5.16 Dissimilar Metals - Unless suitably protected against electrolytic corrosion, dissimilar metals shall not be used in intimate contact with each other. Dissimilar metals are defined in Standard MS 33586.

5.17 Standard Parts - MS and AN standard parts shall be used where they suit the purpose. Commercial parts may be used where no suitable standard part exists.

5.18 Locking - Self-locking fasteners shall be used in accordance with MS 33588 wherever feasible; otherwise, screws, bolts, nuts and other screw parts shall be prevented from loosening by safety wiring in accordance with Standard MS 33540.

5.19 Fungus Resistance - Materials selected should be fungus resistant or should be protected against attack by fungi. This requirement is for design only and need not be demonstrated.

5.20 Workmanship

5.20.1 General - The beam assembly and release unit, including all parts and accessories, shall be fabricated and finished in a workmanlike manner. The parts shall be free from blemishes, burrs, and sharp edges. All brazing, soldering, welding and riveting should be done in such a manner to ensure sound joints. Sharp protrusions are to be enclosed.

5.20.2 Cleaning - The beam assembly and release unit shall be thoroughly cleaned; loose, spattered, or excess solder, metal chips and other foreign matter shall be removed during and after final assembly.

5.20.3 Gear Assemblies - Gear assemblies shall be properly aligned, shall mesh smoothly, and shall operate without binding.

5.21 Data Requirements

5.21.1 General - Data as required by the procuring agency shall be furnished by the vendor.

6.0 QUALIFICATION TESTING

6.1 Test Procedures - The release unit shall be supplied by the vendor and tested with 160 feet of usable cable to the extent specified herein. The tests shall be performed by a vendor and witnessed by a procuring agency representative unless waived by the procuring agency. The use of two winches in the performance of these tests shall be permissible. Two weeks prior to start of qualification testing, the vendor will furnish procuring agency with two sets of the following:

a. Test procedures
b. Photograph and/or drawings of the test equipment arrangement
c. Outline of data collected

6.2 Test Reports - One month after completion of qualification testing, the vendor will submit to the procuring agency reports of the qualification tests which shall include the following:

a. Description of each test including arrangement
b. Results of each test
c. Explanation of deviations from test requirements
d. Data collected

6.3 Test Methods - Unless otherwise stated, all tests will be conducted at ambient room temperature of plus 68 to plus 86 degrees Fahrenheit. Tolerances on test conditions shall be as specified in Specification MIL-E-5272. Detailed procedures specified by Roman numerals refer to test procedures in Specification MIL-E-5272.

6.3.1 Examination of Product - The beam assembly and release unit shall be inspected to determine compliance with the requirements specified herein with respect to materials, markings and workmanship. Also, the beam assembly and release unit will be inspected to the drawings supplied by the vendor. Any deviation from these requirements will be brought to the attention of the procuring agency.

6.3.2 Operation Load Test - The beam assembly and release unit will be subjected to five duty cycles specified in paragraph 3.2.1. Any stoppage, binding or other malfunction shall be cause for rejection.

6.3.3 Structural Load Test - Release Unit - With power off, the release unit shall be subjected to a yield torque of 750 in./lbs. Any yielding of the drum or release unit structure shall be cause for rejection.

Note: This test shall not be performed with the electromechanical cable which will be ultimately assembled into the unit. An alternate piece of cable shall be used to apply the yield torque. During this test, the drum shall be locked out (i.e., prevented from rotating).

6.3.4 Structural Load Test - Beam Assembly - With beam supported as for single-point mode (SK 17545), the beam shall be subjected to a yield load of 100,000 pounds applied at the single-point-mode hook location. Any yielding of the beam structure, including hooks, shall be cause for rejection.

6.3.5 Vibration

6.3.5.1 The beam assembly and release unit shall be subjected to a vibration test specified in Procedure XII. During the test, the electromechanical cable shall be wound on its storage drum under no load. After exposure to the vibration requirements, the winch shall be subjected to the operational load test specified in paragraph 6.3.2.

6.3.5.2 Upon completion of the vibration test above, the beam assembly and the release unit shall be vibrated in each of their three mutually perpendicular axes at a frequency of 13 cps with an amplitude of $\pm$.04 vertically, $\pm$.05 laterally and $\pm$.04 longitudinally for a period of 50 hours, of for 50 hours in each direction separately. The electromechanical cable shall be fully wound on its drum and a tension of 100 $\pm$10 pounds applied to its end. During the test with the torque motor energized, the cable shall

exhibit no tendency to unwind. After completion of the test cycle, the winch shall be subjected to the operational load test specified in paragraph 6.3.2.

6.3.6 Low Temperature - The beam assembly and release unit shall be subjected to the test of Procedure I. At the conclusion of the test and at the specified temperature, the operational load test of paragraph 6.3.2 shall be run. In addition, the time to start shall be recorded. Failure to function in 3 seconds shall be cause for rejection.

6.3.7 High Temperature - The beam assembly and release unit shall be subjected to the test specified in Procedure II. At the completion of the test, if the chamber is too small to perform the operational load test, the beam assembly and release unit may be removed from the chamber for the test, provided the test is run within 3 minutes after removal from the chamber.

6.3.8 Temperature Shock Test - The test of Procedure I shall be employed. At the conclusion of this test, the operational load test of paragraph 6.3.2 shall be run.

6.3.9 Humidity - The beam assembly and release unit shall be subjected to the humidity test of Procedure I. At the conclusion of the test, the operational load tests specified in paragraph 6.3.2. shall be run. In addition, the release unit shall be examined for corrosion. Any evidence of this shall be cause for rejection.

6.3.10 Salt Spray - The beam assembly and release unit shall be subjected to the tests of Procedure I.

6.3.11 Sand and Dust - After completion of the salt spray tests, the beam assembly and release unit shall be subjected to the tests specified in Procedure I. Upon completion of these tests, the operational load test specified in paragraph 6.3.2 shall be run.

6.3.12 Life Test - After completion of the environmental tests, the beam assembly and release unit shall be subjected to the following life tests:

a. Run 3600 duty cycles specified in paragraph 3.2.1.

b. After 3000 cycles of (a) above, a simulated wire rope failure is to be performed. This test shall be repeated five times. The remaining 595 cycles shall be performed as in (a) above.

Note: During the life test, the electromechanical cable must be inspected and electrical continuity checked after the service life of 1200 cycles. It shall be acceptable to replace the cable if damage or discontinuity is found. Similarly, the cable shall be inspected and electrical continuity checked after 2400 cycles.

Any malfunction, with the exception of those noted above, shall be cause for rejection.

7.0 INSPECTION TESTS

7.1 Operation Test - Each beam assembly and release unit shall be subjected to the operation test specified in paragraph 6.3.2.

7.2 Examination of Product - Each beam assembly and release unit shall be inspected in accordance with the requirements of paragraph 6.3.1.

7.3 Quality Assurance Provision - The vendor shall submit to the procuring agency for approval a log showing these tests and any other tests deemed necessary by him to ensure that the performance of subsequent beam assembly and release unit is essentially that of the qualification unit. This log will then accompany each beam assembly.

D8-0692

PROCUREMENT SPECIFICATION, WINCH ASSEMBLY, 25,000-POUND-CAPACITY, 20-TON HOIST SYSTEM

THE BOEING COMPANY
VERTOL DIVISION
MORTON, PA.

1.0 PURPOSE

1.1 This specification defines the requirements for the design, manufacture and testing of a hydraulically driven winch to be used in a two-winch, two-suspension rope hoist system which will be installed in a helicopter for the purpose of lifting, lowering and transporting materiel externally.

2.0 REFERENCES

2.1 The references listed below constitute a part of this specification. In case of discrepancies between specific requirements of this specification and these references, the requirements of this specification shall govern.

SPECIFICATIONS

Military

MIL-S-5002	Surface Treatments (Except Priming and Painting) for Metal and Metal Parts in Aircraft
MIL-B-5087	Bonding, Electrical (For Aircraft)
MIL-E-5272	Environmental Testing, Aeronautical and Associated Equipment, General Specification for
MIL-H-5440	Hydraulic Systems Design, Installation and Tests of Aircraft
MIL-C-6021	Castings, Aircraft Structure, General Specification
MIL-L-6880	Lubrication of Aircraft, General Specification for
MIL-P-6906	Plates, Information and Identification
MIL-E-7080	Electrical Equipment, Piloted Aircraft Installation and Selection of, General Specification for
MIL-S-7742	Screw Threads, Standard, Aeronautical
MIL-I-8500	Interchangeability and Replaceability of Component Parts for Aircraft and Missiles
MIL-D-8513	Drawings and Data Lists, Preparation of Special Support Equipment for Aeronautical and Associated Equipment

MIL-A-8629	Airplane Strength and Rigidity
MIL-D-26125	Design and Evaluation of Cartridges for Cartridge Activated Devices
MIL-T-9107	Test Reports, Preparation of
MIL-Q-9858	Quality Control System Requirement
MIL-M-9868	Microfilming of Engineering Documents, 35mm, Requirements for
MIL-D-70377	Drawings, Engineering and Associated Lists

Standards

Military

MIL-STD-3	Format for Production Drawings
MIL-STD-130	Identification Marking of U.S. Military Property
MIL-STD-704	Electric Power, Aircraft, Characteristics and Utilization of
MS 33540	Safety Wiring, General Practices for
MS 33586	Metals, Definition of Dissimilar
MS 33588	Nuts and Plate Nuts, Self-Locking, Functional Limitations of

Publications

Military

MIL-HDBK-5	Strength of Metal Aircraft Elements
AF Bulletin No. 23	Material and Process Specification

Air Force-Navy Aeronautical Bulletin

ANA 143	Specifications and Standards - Use of

ANA 147 Specifications and Standards of Non-Government Organizations

Drawings

Boeing, Vertol Division

SK 17541 Component Relationship, 20 Ton Hoist System

SK 17542 General Assembly, Winch (8 sheets)

SK 17546 Main Winch Mount

SK 17547 Winch Mount

3.0 DEFINITIONS

3.1 The winch as hereinafter referred to shall consist of the wire rope, its attachment to the beam and storage drum, capstan-driven drums, drive, brake unit, cable cutters, and all necessary components to fulfill the detail requirements specified in paragraph 4.0 of this specification.

3.2 Life - The winch shall have a minimum service life of 1200 cycles before removal or overhaul and 3600 cycles before retirement.

3.2.1 Winch Duty Cycle - One complete cycle shall consist of:

a. Reel out 60 feet loaded to 24,500 pounds ±500 pounds.
b. Reel in 60 feet loaded to 24,500 pounds ±500 pounds.

Note: This cycle may be run continuously provided that the oil temperature in the transmission and brake housing does not exceed 160° F, it being permissible to recirculate the oil externally through a suitable oil cooler pump arrangement. The winch lift test, para. 6.3.12, and winch brake test, para. 6.3.13, shall be integrated with the lift test of beam assembly and release unit procured under Procurement Spec D8-0691.

4.0 DESCRIPTION

4.1 Components - The winch shall be a hydraulically driven capstan type, the design of which shall be based upon the preliminary design shown on Drawing SK 17542 (8 sheets) consisting of the following components:

a. Hydraulic Motor and Drive (SK 17542 Sheet 2)

b. Brake (SK 17542 Sheet 2)

c. Capstan Drive Drums (SK 17542 Sheet 3)

d. Wire Rope

e. Wire Rope Storage Drum with Level Wind (SK 17542 Sheet 4)

f. Wire Rope Reel-Out Drive (SK 17542 Sheet 7)

g. Dual Rope Cutters (SK 17542 Sheet 8)

h. Rope Footage Indicator - Electrical (SK 17542 Sheet 3)

i. Stops - Rope Out/Rope In - Electrical Limit Switches (SK 17542 Sheet 3)

j. Winch Supports (SK 17546, SK 17547)

k. Winch Exit Bellmouth (SK 17542, Sheet 8)

l. Electrically Actuated Components

4.1.1 Motor and Drive - Power input shall be by means of a hydraulic fixed-displacement motor which, when operated at 3000 psi in combination with a reduction drive, shall produce a minimum wire rope pull of 25,000 pounds at a speed of 60 feet per minute. Motor horsepower shall be based on a maximum motor operating speed of 2500 rpm $\pm$200 rpm.

4.1.2 Brake - The brake system shall be automatic in function and shall follow the design schematic illustrated on

Drawing SK 17542. The brake torque capacity shall be such as to permit holding a 25,000-pound load under the following conditions.

a. 2.0g load factor in vertical direction
b. No power input to drive drum

The winch drive and brake system shall have such self-contained energy-absorption capability as to be able to absorb heat energy generated on lowering the load of 25,000 pounds from 150 feet height, at a rate of 60 ft/min. The capability of the system to dissipate the energy shall be such as to permit the repetition of the work cycle after one 1/2-hour interval. No reliance shall be placed on dynamic or regenerating braking using aircraft hydraulic or electric system, in order to have the self-contained energy-absorption capability of the drive and brake system not dependent on any particular aircraft installation. The entire mechanism, except for the hydraulic motor, shall be enclosed in a sealed housing and submerged in an oil bath.

4.1.3 Capstan Drive Drums - The capstan drive drums shall be of light alloy construction having separate hardened alloy steel treads (grooves). The treads shall be so designed that they may be replaced without damage to the drums. The treads shall be installed with clearances such that stresses generated by temperature change shall not cause any excessive loading of the tread or drum. Unless a qualified superior form of groove can be substantiated, the groove form shown on SK 17542, Sheet 2, shall be used. The main capstan drum shall house a three-stage planetary reduction gear drive unit. The secondary capstan drum shall be driven from the main capstan drum through an idler gear. Provision for a chain sprocket shall be made on the secondary capstan for driving the wire rope storage drum. The ratio of drum tread diameter to the wire rope diameter shall be 18:1. The wire rope fleet angle between capstan drums shall not be greater than 2°.

4.1.4 Wire Rope - The wire rope shall be of 6x37 Warrington Searle construction. It shall have a nominal diameter

of 1 inch and have a minimum breaking strength of 103,000 pounds. The manufacture shall be such that all wires used in its construction are continuous; i.e., no welded wires shall be allowed. The wire rope shall be lubricated during and after manufacture with lubricant conforming to MIL-L-6880 or equivalent. Wire rope attachment to the beam shall be by a pin-jointed end fitting. The end fittings shall be free to rotate about the prime mover's lateral axis.

4.1.5 Wire Rope Storage Drum - The wire rope storage drum shall accommodate 150 feet of rope plus the required wraps for rope retention. The drum anchor rope clip shall have a pull-out rating of 1000 to 1500 pounds. The storage drum shall be driven from the secondary capstan drum by permanently lubricated type chain. The drive to the drum shall be through a slipping clutch device which shall maintain a minimum preload in the wire rope of 300 pounds and shall be capable of adjustment without dismantling the winch to provide a load in the wire rope of 600 pounds. (Subsequent qualification tests shall be used to establish the optimum preload in the wire rope from storage reel to capstan drum.) A rope level wind shall be provided on the storage drum. The level wind shall be driven from the storage drum by a permanently lubricated chain drive. All chain drives shall be suitably enclosed to exclude foreign objects.

4.1.6 Wire Rope Reel-Out - Provision shall be made to power reel-out the rope as it leaves the capstan drum. Treads of reel-out pulleys shall be of hardened alloy steel and shall be of semicircular profile. Adequate allowance for high tolerance of rope diameter shall be provided in establishing groove profile.

4.1.7 Dual Rope Cutters - Dual rope cutters shall be provided. They shall be located immediately outboard of the wire rope reel-out pulleys. The dual rope cutters shall be electrically initiated and wired independently. The cartridge devices used shall have twin bridges in parallel. The rope cutters shall be so designed that the severed rope end shall be retained by the cutter after firings so as to prevent a wild rope end. The

rope cutter assemblies shall be installed so that they can be easily checked or replaced without removing any other component of the winch system. All electrical connections and explosive charges must be waterproof and shall be fail-safe in the event of accidental malfunction. The explosive cartridge used in the cable cutter shall meet the requirements of MIL-D-21625A.

4.1.8 Rope Footage Indicator - Provision for - Provision shall be made for the installation of a rope footage indicator. The rope footage indicator shall provide a signal output which will subsequently be used to provide a display of rope footage out to the pilot and/or crane operator.

4.1.9 Winch Rate Monitor - Provision shall be made for the installation of a rate sensing device; i.e., to provide a rate feedback signal to the system hydraulic control servo to obtain a closed-loop contra-circuit. This requirement may be integrated with the rope footage indicator specified in paragraph 4.1.8.

4.1.10 Stops - Rope Out/Rope In - Electrical Limit Switches - Suitable stops shall be provided to ensure that the rope is not inadvertently played out beyond its 150-foot working length and to ensure that the cargo hook system shall not run into the winch exit bellmouth during system raising. The stops shall operate electrical limit switches. The limit switches may be integrated with the rope footage indicator (paragraph 4.1.8).

4.1.11 Winch Supports - The winch supports shall be compatible with the prime mover into which the winch will be installed. In the absence of any prime mover being specified, it shall be assumed that the winch shall be installed in a transport-configured prime mover. The winch supports shall be so designed to allow the winch to pivot about its longitudinal axis. The major winch support, i.e., that which supports the major portion of the load, shall have provision to accept a load isolator and a load cell system. The load isolator shall support the winch and do so as a compressive device. The load

cell shall be integrated with the load isolator. The major winch support shall be so designed that it is capable of reacting all end loads transmitted to the winch from the system suspension wire rope. The minor winch support shall be so designed that it shall be incapable of reacting any end loads from the winch; i.e., it shall be pin jointed. Both supports shall be capable of reacting any side loads from the winch. The structural design criteria specified in paragraph 5.2 shall apply.

4.1.12 Winch Exit Bellmouth - The winch exit bellmouth shall be so designed that it is capable of swivelling in a fore and aft direction equivalent to a rope swing angle of ±30°. The bellmouth shall be made from hardened steel and be so designed that it neither fouls nor pinches the wire rope over its full travel of ±30°.

4.1.13 Electrically Actuated Components - Electrical components of the winch shall be capable of being operated by utilizing 28VDC. The power supplies shall conform to MIL-STD-704.

5.0 DESIGN REQUIREMENTS

5.1 Mounting - The winch, unless otherwise specified, shall be mounted as shown on Drawing SK 17541.

5.2 Structural Design Criteria - The winch shall have a vertical lift capacity of not less than 25,000 pounds with a design load factor of 2.5g and a safety factor of 1.5 ultimate unless otherwise specified in this specification; other safety factors and allowable loads and stresses shall be in accordance with MIL-A-8629 (AER), paragraph 3.21. Additionally, the loading cases in the stress analysis shown in this report shall be provided for to the extent specified by the procuring agency.

5.3 Performance - The winch shall be capable of operating after exposure to the following conditions:

a. Temperature ranging from -65° to +150° F.
b. Relative humidity up to 100 percent
c. Sand and dust encountered in desert areas

5.4 Vibration - The winch shall be capable of operating after being subjected to vibrations specified in paragraph 6.3.5.

5.5 Capacity - The capacity shall be as given in paragraph 4.1.1.

5.6 Life - The winch shall be designed for a service life and overhaul period as defined in paragraph 3.2.

5.7 Weight and Dimensions - Weight of one winch with hydraulic motor and 190 feet of wire rope shall not exceed 1200 pounds. Physical dimensions shall not exceed those shown on Drawing SK 17542.

5.8 Hydraulic Systems - The winch shall be powered by a 3,000-psi system which shall be in accordance with the requirements of Specification MIL-H-5440 for Type II systems. The flow requirements shall not exceed 40 gpm per winch.

5.9 Electric Wiring and Circuitry - All wiring and components

shall be in accordance with the requirements of Specification MIL-E-7080.

5.10 Lubrication - The winch shall incorporate self-contained lubrication which shall be in accordance with the requirements of Specification MIL-L-6880 and shall be suitable for use under all operating environments specified herein.

5.11 Screw Threads - Screw threads shall conform to requirements of Specification MIL-S-7742.

5.12 Interchangeability - All parts having the same part number shall be dimensionally and functionally interchangeable. The requirements of Specification MIL-I-8500 shall apply to the motor and mounting dimensions.

5.13 Marking

5.13.1 Identification of Product - Parts shall be marked in accordance with Standard MIL-STD-130. A nameplate in accordance with Specification MIL-P-6906 shall be affixed to the winch with the following information etched, engraved or otherwise permanently printed:

Winch Assembly 25,000 lb capacity
Serial No.
Stock Number
Manufacturer's Identification
Manufacturer's Part Number
Contract No.
U.S.

5.13.2 Instructions - Brief operating instructions, hydraulic port identification, limit switch identification, and a wiring diagram shall be permanently mounted on the winch by using metal foil or metal plate type.

5.14 Cooling - The winch shall be capable of operating throughout its duty cycle specified in paragraph 3.2.1 without the need for external cooling.

5.15 Corrosion Resistance - Metals shall be chosen for their corrosion resistance; otherwise, they should be finished to resist corrosion.

5.16 Finishes

5.16.1 Surface Finish - Surface finishes of detail parts shall be in accordance with the requirements of MIL-S-5002.

5.16.2 Exterior Finish - The exterior finish shall be medium gray, #36231 per Fed Std 595.

5.17 Dissimilar Metals - Unless suitably protected against electrolytic corrosion, dissimilar metals shall not be used in intimate contact with each other. Dissimilar metals are defined in Standard MS 33586.

5.18 Standard Parts - MS and AN standard parts shall be used where they suit the purpose. Commercial parts may be used where no suitable part exists.

5.19 Locking - Self-locking fasteners shall be used in accordance with MS 33588 wherever feasible; otherwise, screws, bolts, nuts and other screw parts shall be prevented from loosening by safety wiring in accordance with Standard MS 33540.

5.20 Fungus Resistance - Materials selected should be fungus resistant or should be protected against attack by fungi. This requirement is for design only and need not be demonstrated.

5.21 Workmanship

5.21.1 General - The winch, including all parts and accessories, shall be fabricated and finished in a workmanlike manner. The parts shall be free from blemishes, burrs and sharp edges. All brazing, soldering, welding and riveting should be done in such a manner to ensure sound joints. Sharp protrusions to be enclosed.

5.21.2 Cleaning - The winch shall be thoroughly cleaned; loose, spattered, or excess solder, metal chips and other foreign matter shall be removed during and after final assembly.

5.21.3 Gear Assemblies - Gear assemblies shall be properly aligned, shall mesh smoothly, and shall operate without binding.

5.22 Data Requirements

5.22.1 General - Data as required by procuring agency shall be furnished by the vendor.

6.0 QUALIFICATION TESTING

6.1 Test Procedures - The winch shall be supplied by the vendor and tested with 190 feet of usable cable to the extent specified herein. The tests shall be performed by a vendor and witnessed by a procuring agency representative unless waived by the procuring agency. Two weeks prior to start of qualification testing, the vendor will furnish the procuring agency with two sets of the following:

a. Test procedures
b. Photograph and/or drawings of the test equipment arrangement
c. Outline of data collected

6.2 Test Reports - One month after completion of qualification testing, the vendor will submit to the procuring agency reports of the qualification tests which shall include the following:

a. Description of each test including arrangement
b. Results of each test
c. Explanation of deviations from test requirements
d. Data collected

6.3 Test Methods - Unless otherwise stated, all tests will be conducted at ambient room temperature of plus 68 to plus 86 degrees Fahrenheit. Tolerances on test conditions shall be as specified in Specification MIL-E-5272. Detailed procedures specified by Roman numerals refer to test procedures in Specification MIL-E-5272.

6.3.1 Examination of Product - The winch shall be inspected to determine compliance with the requirements specified herein with respect to materials, markings and workmanship. Also, the winch will be inspected to the drawings supplied by the vendor. Any deviation from these requirements will be brought to the attention of the procuring agency.

6.3.2 Operation Load Test - The winch will be subjected to five duty cycles specified in paragraph 3.2.1. Any stoppage, binding or other malfunction shall be cause for rejection.

6.3.3 Structural Load Test - With power off, the winch wire rope shall be subjected to a yield load of 62,500 pounds. Any yielding of the drums or winch structure shall be cause for rejection.

6.3.4 Vibration

6.3.4.1 The winch shall be subjected to a vibration test specified in Procedure XII. During the test, the wire rope shall be wound on its storage drum with no load on the rope. After exposure to the vibration test, the winch shall be subjected to the operational load test specified in paragraph 6.3.2.

6.3.4.2 Upon completion of the vibration test above, the winch shall be vibrated in each of its three mutually perpendicular axes at a frequency of 13 cps with an amplitude of ±.04 vertically, ±.05 laterally and ±.04 longitudinally for a period of 50 hours, or for 50 hours in each direction separately. The rope shall be fully wound on its drum and a tension of 25,000 pounds applied to the end. During the test, the drive shall exhibit no slippage. After completion of the test cycle, the winch shall be subjected to the operational load test specified in paragraph 6.3.2.

6.3.5 Low Temperature - The winch shall be subjected to the test of Procedure I. At the conclusion of the test and at the specified temperature, the operational load test of paragraph 6.3.2 shall be run. In addition, the time to start shall be recorded. Failure to start in 3 seconds shall be cause for rejection.

6.3.6 High Temperature - The winch shall be subjected to the test specified in Procedure II. At the completion of the test, the operational load test specified in paragraph 6.3.2 shall be run. If the chamber is too small to perform the operational load test, the winch may be removed from the chamber for the test, provided the test is run within 3 minutes after removal from the chamber.

6.3.7 Temperature Shock Test - The test of Procedure I shall be

employed. At the conclusion of this test, the operational load test of paragraph 6.3.2 shall be run.

6.3.8 Humidity - The winch shall be subjected to the humidity test of Procedure I. At the conclusion of the test, the operational load tests specified in paragraph 6.3.2 shall be run. In addition, the winch shall be examined for corrosion. Any evidence of this shall be cause for rejection.

6.3.9 Sand and Dust - After completion of the humidity tests, the winch shall be subjected to the tests specified in Procedure I. Upon completion of these tests, the operational load test specified in paragraph 6.3.2 shall be run.

6.3.10 Life Test - After completion of the environmental tests, the winch shall be subjected to the following life test:

Run 3600 duty cycles specified in paragraph 3.2.1.

Any malfunction, with the exception of rope, rope tread, brake pads or clutch pads, shall be cause for rejection.

6.3.11 Brake Test - With a load of 25,000 pounds applied to the wire rope, the winch shall be reeled out approximately 10 feet, the power switched off, and the drums allowed to come to rest without the 25,000-pound load being relieved. The load shall be held without slipping or rotation of the drums. Failure to do so will constitute reason for rejection.

Note: During the life test, the wire rope shall be replaced after a service life of 1200 duty cycles. The wire rope treads are to be inspected for excessive wear and replaced if necessary after a service life of 1200 duty cycles. The foregoing procedure shall be repeated after 2400 duty cycles.

7.0 INSPECTION TESTS

7.1 Operation Test - Each winch shall be subjected to the operation test specified in paragraph 6.3.2.

7.2 Examination of Product - Each winch shall be inspected in accordance with the requirements of paragraph 6.3.1.

7.3 Quality Assurance Provision - The vendor shall submit to the procuring agency for approval a log showing these tests and any other tests deemed necessary by him to ensure that the performance of subsequent winches is essentially that of the qualification unit. This log will then accompany each winch.

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13. ABSTRACT

The purpose of this two-phase investigation is to define the optimum design of a heavy-lift helicopter external cargo handling system of 40,000-pound capacity. The report covers the results of an interface subsystem configuration analysis and defines the load handling winches and hoist systems required for a feasible, overall cargo handling system (including those aircraft-related systems necessary for control--actuation and monitorship, load attachment, suspension, hoisting, and shock dampening).

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