CECO 50-237 Dresden #2

Dresden Special Rpt #41, Quad-Cities Special Rpt #16, Suppl A "Reactor Bldg Crane & Cask Yoke Assembly Modifications"........

(Rec'd w/ltr 6-10-75..............#6810-A)

DRESDEN STATION SPECIAL REPORT NO. 41

QUAD-CITIES STATION SPECIAL REPORT NO. 16

SUPPLEMENT A

"Reactor Building Crane and
Cask Yoke Assembly Modifications"

JUNE 1975

We have reviewed the Branch Position on overhead crane handling systems, dated January 10, 1975, in light of the system proposed for installation by Commonwealth Edison at Dresden and Quad Cities Stations (Reference Dresden Special Report No. 41 and Quad Cities Special Report No. 16). The system that has been purchased for the stations cited is a Whiting Redundant Hoist System, consisting of a dual load path through the hoist gear train, reeving system, and hoist load block; along with restraints at critical points to provide for load retention and to minimize uncontrolled motions of the load upon failure of any single hoist component.

This system is judged to be in substantial conformance with the position taken in the draft guideline, as is demonstrated by the itemized summary that follows. To facilitate your review of this summary, the numbering system employed in the Branch Position has been used.

1. Performance Specification and Design Criteria
   a. Not applicable. The proposed Commonwealth systems are replacements for existing equipment in an operating plant and will not be employed for construction.
   b. The Dresden and Quad Cities reactor building cranes are located indoors in a heated environment with a minimum ambient temperature of 70°F. Therefore, Commonwealth Edison considers it impracticable to require that nil ductility data be developed for our cranes. The

1. b. (Cont'd)

suggested requirement of NDTT +60°F is more appropriate to reactor materials and in our judgement is an unreasonable crane design criteria. Moreover, the change to our existing system involves only the replacement of the installed nonredundant trolley; the existing bridge will not be replaced. It would be both impracticable and technically inadvisable to perform the required testing to comply with ASTM Spec. E-208 in order to characterize the NDT temperature of the bridge. In summary, it is judged that the controlled environment in which our proposed system will operate, obviates the necessity of considering NDT.

c. The Dresden and Quad Cities cranes are identified as Safety Class II equipment in the plant operating license. It is not practicable to consider reclassifying the hoist system as Seismic Class I, because this would most probably require a new bridge and extensive modifications to the bridge trackway. The bridge and trolley will be analyzed in a manner consistent with the design codes applicable at the time of the original installation, that is, the allowable stress will be limited to 90% of yield, with only static lifted loads considered.

d. All load bearing welds on the replacement trolley will undergo a magnetic particle inspection. Non-destructive testing of the base material is judged to be impracticable do to the contraints of the joint geometry. However, the

1. d. (Cont'd)

possibility of "Lamellar Tearing" is minimized by the use of small diameter weld rods which cause less shrinkage in the weld, and thereby, minimize the possibility of parent metal damage.

e. Although a comprehensive fatigue analysis has not been performed, it is judged that the design is sufficiently conservative and the actual usage so small that no fatigue problem exists. Factors of safety to yield on all critical components exceed 3.0 and to ultimate exceed 4.5 (See Attachment 1).

f. Preheat and postheat treatment temperatures for all weldments will be specified in the appropriate weld procedures. These procedures will conform to AWS D1.1 and, as such, satisfy the recommendations of Regulatory Guide 1.50.

2. Safety Features

a. The automatic controls and limiting devices are designed in such a manner that operating disorders will not prevent the handling system from being maintained at a safe neutral holding position (See the Referenced Special Reports, Section 2.3).

b. Sufficient redundancy exists both in limit controls on bridge, trolley and load motion, as well as braking systems for those motions that in the case of a subsystem or component failure, the load will be retained and held in a stable or immobile safe position. The braking systems

2. b. (Cont'd)

are redundant and fail safe (electric shoe brakes that fail closed on loss of power). (See the Referenced Special Reports, Section 2.3).

c. Appropriate procedures will be implemented to provide for the safe handling of the load in the event that the main handling system is immobilized.

d. Adequate provision has been made in the plant design to allow for corrective repairs without impairing the safe operation of the nuclear units.

3. Equipment Selection

a. The requirement that a static load of 3W be carried by the load block assembly without permanent deformation is met (See Attachment 1).

b. All lifting rigs, that is yokes, trunnions, etc., used during spent fuel cask handling will be rated at a minimum of 3W; where the rated capacity is the gross dead weight of the loaded cask (NLI 10/24, 200,000lb.). Design of these components will conform to 10CFR71.21c.

c. The proposed trolley system provides for dual hoisting means, with a maximum hoisting speed of 5.75 fpm at full design load.

d. The trolley design incorporates a load equalizing system which maintains alignment and stability under full design load. The dual reeving system allows either load path to support 3W and maintain load stability and alignment through the center of the head block, through all hoisting components, through the center of gravity of the load (See the Referenced

3. d. (Cont'd)

Special Reports, Section 2.2).

e. The wire rope employed is 1¼" diameter IPS-1WC steel center rope (6 X 37 wire). This is lubricated wire rope with a demonstrated history of performance. The rope reeving system is redundant and is described in Sections 2.2 and 4.1 and Figure 2 of the referenced Special Reports.

f. The maximum fleet angle occurring at the drum is 3.58° and diminishes rapidly as the block is lowered. It will go to zero in the first few revolutions of the drum and increase to a maximum 2.29° at the low position. The fleet angles between individual sheaves does not exceed 2.87°, and the pitch diameter for all sheaves is a minimum of 24 times the rope diameter. These values satisfy the requirements of Section 70-4 of CMAA Specification #70.

In support of the reeving system design, we are submitting as Attachment 2 a generic report prepared by the Whiting Corporation entitled, "Whiting Corporation's Redundant Trolley Design".

g. As is clearly identified in Attachment 1, the static factors of safety on all major components are significantly above the 200% overload criteria suggested in this section. In addition, non-destructive testing will be performed on all load bearing components to insure their integrity. However, given the extensive load test program identified

3. g. (Cont'd)

in Section 3.0 of the referenced Special Reports, we consider the manufacturer shop test to be unnecessary. It would require a costly modification to the manufacturer's facility that is unjustified, since it could not be accepted as a conclusive test without a full load field test. It is judged that the extensive field testing previously specified is sufficient.

h. The requirements of this section are satisfied by our crane design with the exception of provision for over temperature devices.

i. The control system to be installed will satisfy the requirements of this section.

j. As is described in the referenced special reports, positive, reliable and capable means to stop and hold the hoisting drum have been provided. However, it is our opinion that no design feature need be installed to absorb the energy of "two-blocking". Inasmuch as the crane control system provides for a minimum of two vertical main hoist limiting devices, it is unreasonable and impracticable to design for an event that should not occur.

k. As is shown in the attached component safety analysis, the main hoist system is purposely designed to provide redundancy sufficient to preclude the possibility of a catastrophic load loss. Where redundancy could not be provided by design, the non-redundant features such as the drum

3. k. (Cont'd)

itself were designed with high safety factors.

l. The breakdown torque rating of the electrical motor drive will be 120% of the design requirement. It is judged that this is reasonable in light of the fact that the design rated load will be lifted at a maximum speed significantly below the rated speed of the hoist.

m. The primary main hoist drive is a 5Hp AC squirrel cage gearhead motor electrically clutched to drive the main drive motor shaft for a continuous duty rated speed of six inches per minute which provides effective power control load handling; in addition, two GE magnetic operated electric shoe brakes are also provided on the main hoist. Overspeed protection will be provided on the main hoist drive, consisting of a phase loss relay on the DC side to prevent operation of the MG set at greater than its AC synchronous speed and to monitor the field circuit of the DC side to prevent motor overspeeding beyond its normal operating characteristics.

Each holding brake is designed to a minimum of 125% of the full load motor torque at the point of application. Administrative procedures will insure that emergency lowering of the design load will not be allowed unless both mechanical brakes are operational. Manual holding brakes are judged to be unnecessary since the incorporation of the inching drive system provides a more than satisfactory alternative.

3. m. (Cont'd)

Procedure will indicate that emergency lowering will not be allowed at greater than 3.5 FPM.

n. Dynamic and static alignment of all hoist components will be maintained throughout the range of loads to be lifted. All components are position anchored on the trolley machinery platform.

o. As has been stated, an inching drive will be provided. Jogging the main hoist motor will be disallowed and will be noted in the crane operating procedure as improper.

p. Although the possibility of overtorque must be addressed, it is judged that the 110% limit suggested is inappropriate in that such a design criteria could lead to accelerated motor wear. As has been stated previously the motors have been designed to 120% of breakdown torque.

The trolley will also have a redundant drive brake on the gearcase pinion shaft extension, and will include a timer to prevent excessive deceleration and consequent load swing.

As a part of the critical load path system which will be installed, the bridge will have an electric parking brake which will set on motion beyond established setpoints.

Opposite wheels on the trolley will be matched for diameter, but their exists no documentation on the existing crane, which will not be modified.

3. p. (Cont'd)

The maximum trolley speed is 33 fpm and the maximum bridge speed is 50 fpm. Since these speeds exceed those recommended in the draft guide they are not recommended as operating speeds; procedure will define the limits within which trolley and bridge motion will be constrained.

q. The requirements of this section will be satisfied.

r. Trolley and bridge bumper stops will be provided to supplement the safety device installed as a part of the critical load path system.

s. The requirements of this section will be satisfied.

t. The requirements of this section do not apply since the hoist system will not have been used for construction.

u. The requirements of this section will be satisfied.

4. Mechanical Check, Testing and Preventive Maintenance

The test program contemplated has been described fully in Section 3.3 of the referenced special reports.

ATTACHMENT 1

COMPONENT FAILURE ANALYSIS
125/9 TON CAPACITY REACTOR ROOM CRANE

for the

COMMONWEALTH EDISON COMPANY

ITEM NO.	DESCRIPTION	FACTOR OF SAFETY (ULT.)	FACTOR OF SAFETY (YIELD)	YES REDUNDANT NO	FAILURE PROTECTION IF NO REDUNDANCY IS PROVIDED	RESULT OF FAILURE IF NO REDUNDANCY IS PROVIDED
1	Sister Hook	8.5	4.3	Yes		
2	Lifting Eye	7.4	3.7	Yes		
3	Hook Swivel	8.5	3.8	No	Block Housing	Sheaves Will be Displaced Vertically by ABT. $\frac{1}{4}$"
4	Hoist Rope	8.0	5.3	Yes		
5	1st. Red Pinion	62.8	43.6	Yes		
6	1st. Red Gear	31.0	13.0	Yes		
7	1st. Red Shaft	35.0	23.3	Yes		
8	2nd. Red Pinion	26.5	19.0	Yes		
9	2nd. Red Gear	12.7	5.3	Yes		
10	2nd. Red Shaft	47.3	36.4	Yes		
11	Drum Pin Shaft	9.7	6.4	Yes		
12	Pinion Key	11.7	6.3	Yes		
13	Gear Key	13.5	7.3	Yes		
14	Extension Shaft	39.3	26.2	Yes		

ITEM NO.	DESCRIPTION	FACTOR OF SAFETY (ULT.)	FACTOR OF SAFETY (YIELD)	YES REDUNDANT NO	FAILURE PROTECTION IF NO REDUNDANCY IS PROVIDED	RESULT OF FAILURE IF NO REDUNDANCY IS PROVIDED
15	Coupling Key	15.9	8.9	Yes		
16	Extra Red Pinion	14.8	10.7	Yes		
17	Extra Red Gear	12.0	6.2	Yes		
18	Main Drum	7.6	3.8	No	Increased Safety Factor	Uncontrolled Descent of Load
19	Drum Shaft	12.3	9.7	No	Safety Lugs to Constrain Drum Hubs	Drum will be displaced vertically by about 1/8"
20	Drum Brg. Supp't	16.9	7.7	No	Increased Safety Factor	Uncontrolled Descent of Load
21	Drum Pin. Brg. Supp't.	31.7	14.5	No	Increased Safety Factor	Uncontrolled Descent of Load
22	Rope Anchor	11.3	8.4	Yes		
23	Equalizer Bar	24.2	12.6	Yes		
24	Equalizer Bushing	22.6	17.0	No	Equalizer Bar Will Rest on Pin	Equalizer Bar will be displaced vertically by 3/8"
25	Equalizer Pin	12.9	8.6	No	Equalizer Bar Will Rest on Support Frame	Equalizer Bar will be displaced vertically by $1\frac{1}{4}$"
26	Equalizer Bar Supp't Frame	9.3	4.2	No	Load Girt Will Catch Equalizer Bar	Equalizer Bar will be displaced vertically by $1\frac{1}{4}$"
27	Load Sensing Sheave Frame	24.4	14.7	No	Increased Safety Factor	Uncontrolled Descent of Load

ITEM NO.	DESCRIPTION	FACTOR OF SAFETY (ULT.)	FACTOR OF SAFETY (YIELD)	YES REDUNDANT NO	FAILURE PROTECTION IF NO REDUNDANCY IS PROVIDED	RESULT OF FAILURE IF NO REDUNDANCY IS PROVIDED
28	Sheave Pins	40.5	26.9	No	Sheave Frame will Catch Sheaves	Sheaves will be displaced vertically by ½"
29	Load Cell Pins	7.5	5.9	No	Load Girt will Catch Sheave Frame	Sheave Frame will be displaced vertically by ½"
30	Rod Eye	6.5	3.3	No	"	"
31	Load Cell Brkt.	6.7	3.1	No	"	"
32	Main Load Girt I	11.9	6.2	No	Increased Safety Factor	Uncontrolled Descent of Load
33	Main Load Girt II	6.5	3.4	No	"	"
34	Trolley Truck	7.5	3.9	No	"	Load Girts will be displaced vertically abt. 1"
35	Trolley Wheel Axle	11.9	7.9	No	"	Truck will be displaced vertically by 1"
36	Bridge Truck	8.6	4.5	No	"	"
37	Bridge Wheel Axle	10.9	6.9	No	"	"
38	Bridge Girders	4.7	2.5	No	"	Uncontrolled Descent of Load

*Redundant implies "Back-up" System provided.

WHITING CORPORATION
PRODUCTION ENGINEERING DEPT.
HARVEY, ILLINOIS 60426 U.S.A.
AREA CODE 312 331-4000

CUSTOMER

C.O. NO. REQN.

DATE 2-28-75 BY RBB

WHITING CORPORATION'S

REDUNDANT TROLLEY DESIGN

(ATTACHMENT 2)

It is the purpose of this report to show that the ATOMIC ENERGY COMMISSION'S proposed problems with respect to reverse bends, excessive fleet angles, and inadequate sheave diameters do not exist or have little or no effect on the design of WHITING CORPORATION'S REDUNDANT TROLLEYS.

TABLE OF CONTENTS

I REVERSE BENDS

II FLEET ANGLES

III SHEAVE DIAMETERS

IV SUMMARY

V REFERENCE DRAWINGS

VI REFERENCES

VII APPENDIX

SECTION I

The objective of this section is to show that reverse bending does not exist in the rope except for the lead lines which pass up to the drum, as shown in drawings U-67052, U-67053, T-53326, and T-53327.

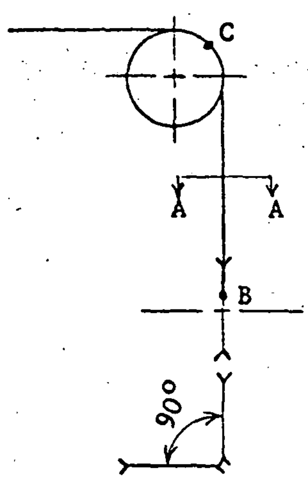
In a typical case the distance from the centerline of the main drum to the centerline of the sheaves in the block (point of tangency) is 9'-4" (112") with the block in the extreme upper position as shown in drawing U-67053. This represents the most severe condition as far as reverse bends are concerned.

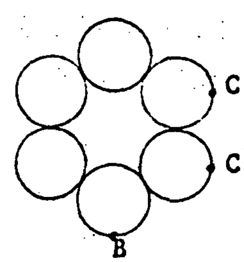
The distance from the point of tangency on one sheave to the point of tangency on the next sheave must be something like one or two lays of rope for the rope to be subjected to reverse bending. (see Reference 1) If the rope is assumed to be 1-1/8" diameter I.W.R.C. 6 x 37, the lay is approximately 7". If the tangent to tangent distance were between 7" and 14", there would be reason for some concern, however in the above-mentioned design the tangent to tangent is 16 rope lays.

Section 1 - continued

Furthermore, as can be seen in U-67053, T-53326, and T-53327 reverse bends are not present since the rope is not changing direction 180°, but only 90° except for the lead lines which pass up to the drum, these lead lines do change direction 180° but the distance is so great that the effect will be negligible (Reference 1 and 2).

Reverse bends relate to fatigue design considerations. The term itself implies cyclic stresses varying from tension to compression at the same point on the periphery of the rope. Keeping this thought in mind, there are no reverse bends when the rope changes direction 90° because the same wire on the periphery of the rope is not being elongated as it passes over each sheave as shown below:





SECTION A-A

We have used the above-mentioned reeving (from drum to block sheave) for seventy years without any problems with respect to rope failure. There are several drawings

Section 1 - continued

(U-55537, T-44785, and T-45012), which are included, showing reevings which are similar to that of a Redundant Trolley reeving with no apparent problem after several years of service.

Finally, consider the relative movement of the block with respect to the drum. A section of rope which is just tangent to the drum will have to move considerably more than 112" in order to become tangent to the block because as the rope starts to move, the block is moving away from the drum and this further increases the effective distance between tangent points.

SECTION II

The objective of this section is to show that a fleet angle of 1 to 12 (4^{o} 45') is not excessive. A maximum fleet angle of 1 to 12 was first published as a Whiting Corporation standard in 1955 in Whiting Crane Handbook reflecting years of previous experience. The above publication was revised in 1967 with the maximum fleet angle (1 to 12) unchanged.

A number of trolleys (T-5012, T-44785, U-55537, U-37901, U-25100, U-58044, and U-58032) have been built with a maximum fleet angle of 1 to 12 and have been in operation for a number of years with no apparent trouble.

A detail analysis (Reference 3) of the change in fleet angle of the lead lines on a typical Redundant Trolley (see drawing U-67053) was made. The original fleet angle based on a 1 to 12 ratio is 4.76^{o}. By the time the drum has turned one revolution, the point of contact on the drum has moved over 1-1/4" and the reach has increased from 9'-4" to 10.56'. This reduces the effective fleet angle to 3.64^{o}. The next revolution decreases the fleet angle to 2.75^{o} (Reference 3).

Section II - continued

Therefore the maximum fleet angle (1 to 12) exists when the block is at its maximum upper position and decreases quickly when the block is lowered.

Finally, there are two more areas of concern with respect to proper design of fleet angles. The fleet angle influence on the life of the rope is reflected by abrasion which may occur from rope to rope on the drum or at the entry of the rope into a sheave. Abrasive considerations in a powerhouse crane are of very little importance because of the limited use of the crane relative to an industrial or a duty cycle crane which is designed on this same basis. A planned crane inspection and maintenance procedure can be considered an additional safety measure to provide for the detection of abrasive wear. The second area in which fleet angle influence is seen is the tendency to properly spool the rope onto the drum or to remain seated in the rope sheave throat. The argument of spooling and setting in the sheave throat are demonstrated by experience and enhanced by the fact that the Redundant reeving is less severe than can be found in conventionally reeved cranes. In any event, the crane is equipped with an alarm system, i.e., the rotating equalizer bar, which will recognize improper spooling and deactivate the crane.

Consequently, we must conclude that the Atomic Energy Commission's concern for the fleet angles in our redundant reeving has no foundation and that in point of fact we can conclude that the design provided should be of less concern than a conventionally reeved crane.

SECTION III

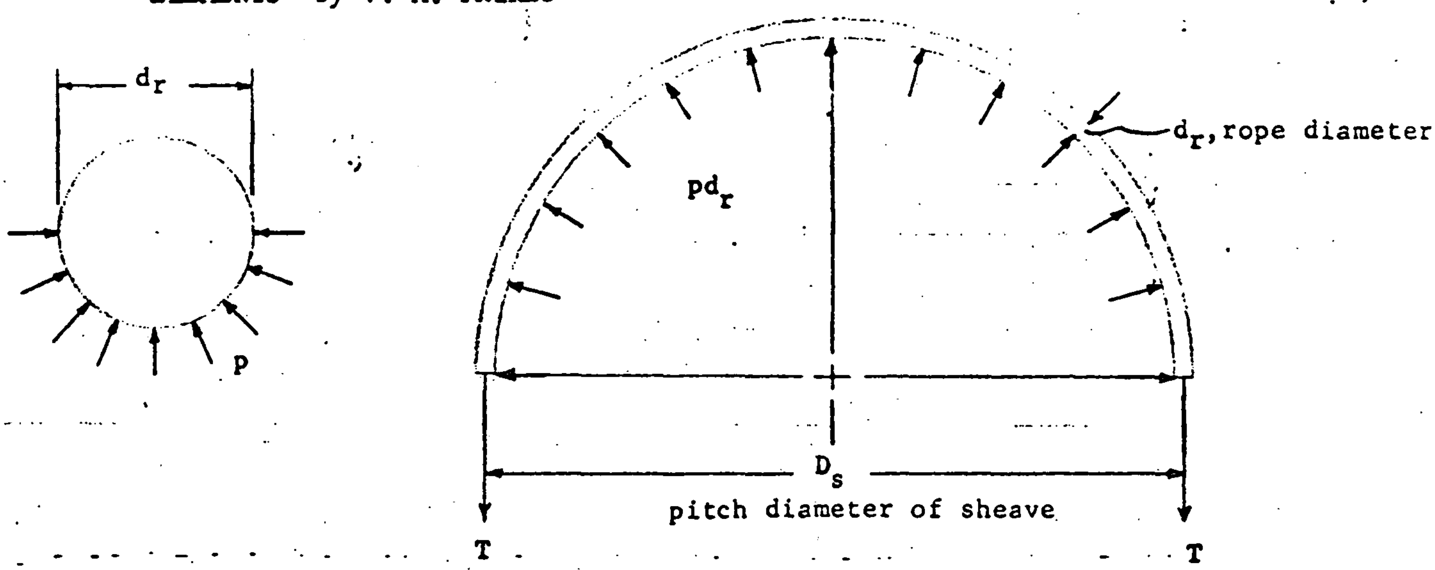
The objective of this section is to show that when the sheave diameters are equal to 26 or 30 times the rope diameter (26 d_r or 30 d_r) verses 24 times the rope diameter ($24d_r$) there is little difference in fatigue life.

A sixteen (16) part rope reeving system was chosen over a twelve (12) part rope reeving system for evaluation because there are more bends of the rope due to a large number of sheaves in the sixteen (16) part rope reeving system.

A typical cycle is shown in Figure 1 while a typical reeving diagram is shown in drawings T-53327 and T-53326 (it was assumed all sheaves were the same diameter). Twenty-four (24) sheaves were determined to be the maximum number of sheaves that a point passes over during one cask handling cycle. The cask handling cycle was assumed to be as shown in Figure 1.

Section III - continued

The following method is from "DESIGN OF MACHINE ELEMENTS" by V. M. FAIRES



T = Maximum capacity/parts of rope

A_r = Area of rope

d_w = Diameter of one wire = $1/22 \times d_r$

STATIC STRESSES

Direct stress

$\sigma = T/A_r = T/.471 d_r^2$

Bending stress

$\sigma_b = E\, d_w/D_s$

Section III - continued

FATIGUE LIFE

$p = 2T/(D_s\ d_r)$

p = bearing pressure per square inch of projected area of the rope on the sheave

Pressure Ratio vs. Cycles to Failure diagram - Figure #2

SAMPLE CALCULATIONS:

Assumed - d_r = 1-1/8" and load (maximum capacity) = 125 tons

$T = [125 \text{ tons } (2000\#/\text{Ton})] / 16 = 15,625.\ \#$

$\sigma = T/A_r = T/.471\ d_r^{\,2} = 15,625\#/.471(1\text{-}1/8")^2 = 26,212.\text{psi}$

for $24d_r$ sheaves and type 304 super tensile material

$E = 28 \times 10^6$ psi

$\sigma_b = E\ d_w/D_s = [E(1/22\ d_r)] / 24d_r = [28 \times 10^6\text{psi } (1/22 \times 1\text{-}1/8")]/(24\text{x}1\text{-}1/8")$

$\sigma_b = 53,030.\text{psi}$

Section III - continued

$p = 2T/(D_s\ d_r) = \left[2(15,625.\#)\right] /(24d_r \times d_r) = 1028.8$ psi

$p/S_{ult.} = 1028.8$ psi/200,000 psi $= 0.005144$

From Figure #2

Number of bends to failure - 75,000

24 Bends/cask handling - page 6

75,000/24 = 3,125 cask handlings

The results are tabulated on the next page using the preceding equations as the sheave diameters and the materials change.

Section III - continued

Sheave Diameters	Material	Number of Cask Handlings
$24d_r$	Type 304 Super Tensile	3,125
	Bright Super Tensile Monitor AAA	4,166
$26d_r$	Type 304 Super Tensile	3,333
	Bright Super Tensile Monitor AAA	5,000
$30d_r$	Type 304 Super Tensile	4,583
	Bright Super Tensile Monitor AAA	5,833

Section III - continued

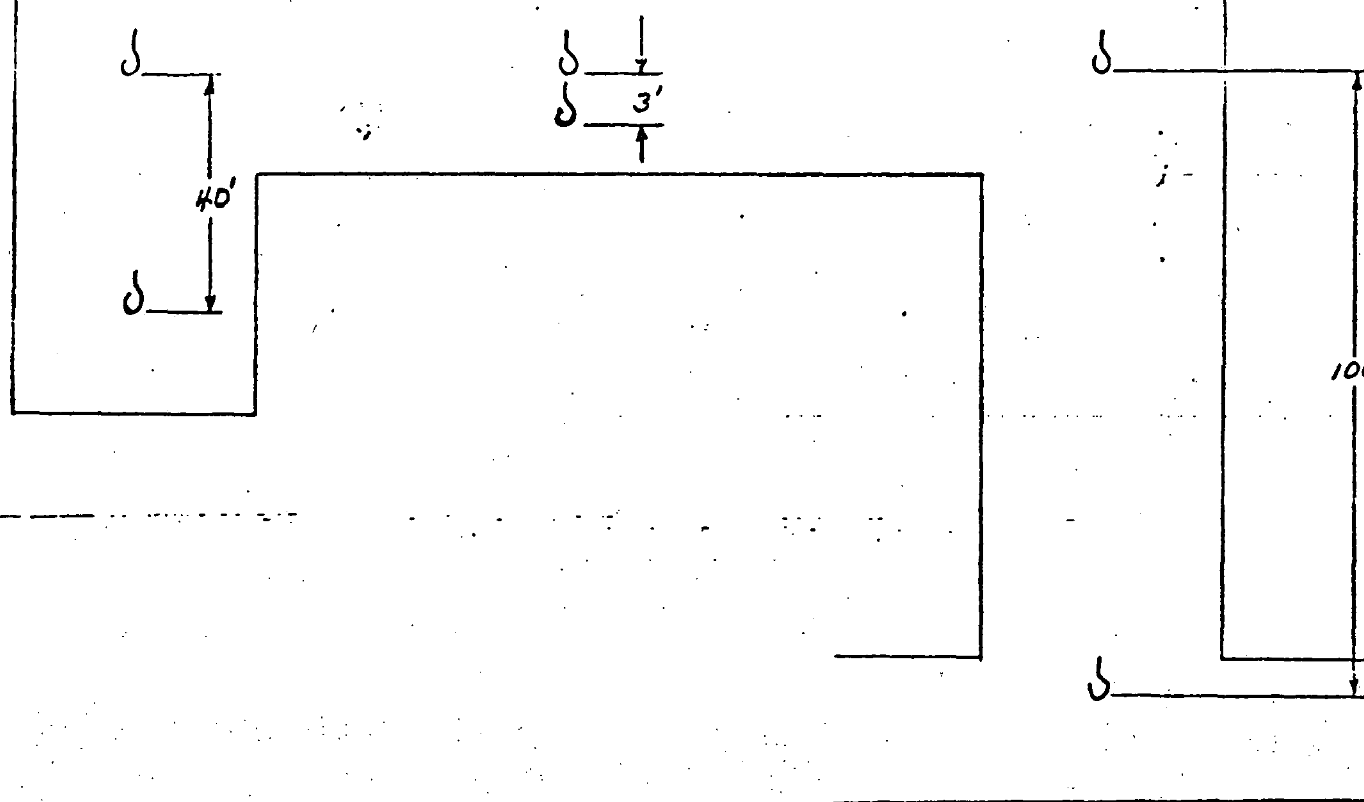
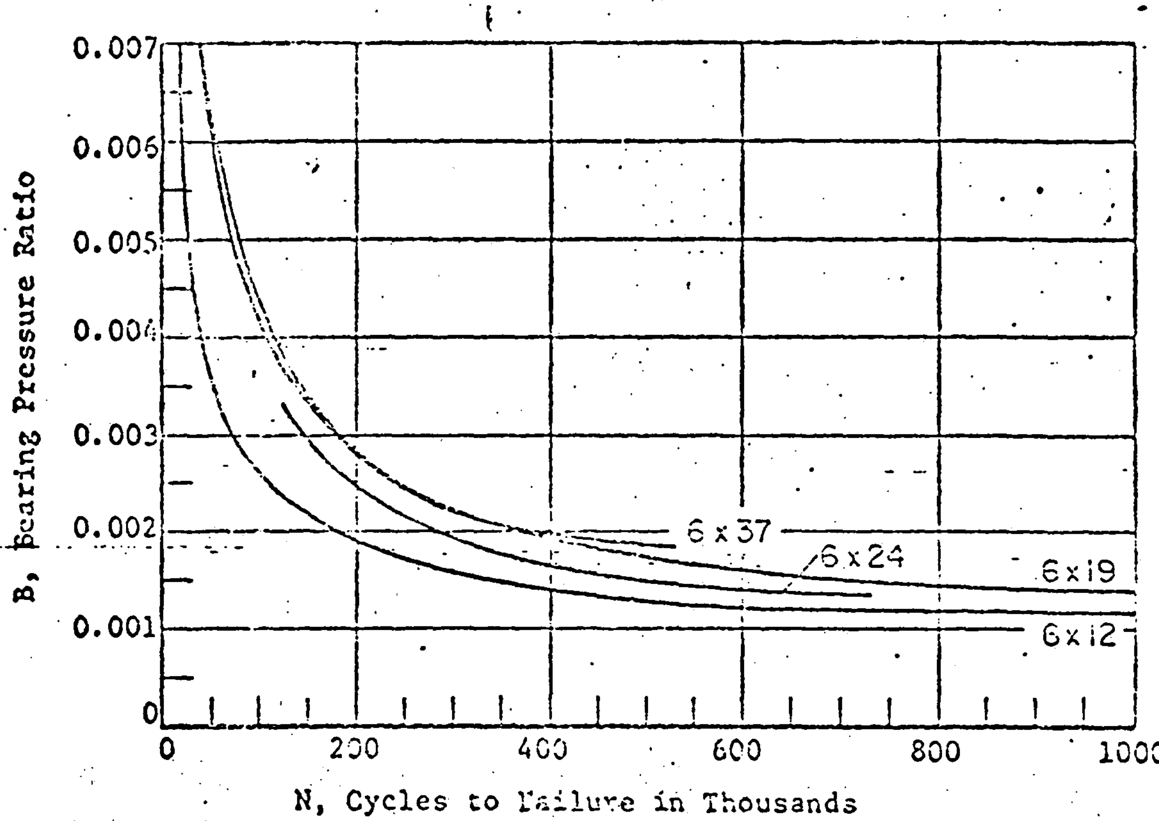


Figure No. 1
Typical Cask Handling Cycle

ORIGINAL DRUCKER AND TACHAU CURVES OF AVERAGE BEARING PRESSURE RATIO VERSUS FATIGUE LIFE



$$B = \frac{2T}{UDd},$$

where B = bearing pressure ratio
T = load on rope, units of force
U = ultimate strength of the wire material, force per unit area
d = rope diameter, length
D = sheave pitch diameter, length.

Figure #2

SUMMARY

It is apparent from this study, that there is no problem with respect to reverse bending since there are no reverse bends other than the lead lines which pass up to the drum which is common in many reeving systems. This is no problem due to fact that the distance from the point of tangency on the drum to the point of tangency on the sheave must be something like one or two lays of rope to be considered a reverse bend application. (see Reference 1). Assuming a 1-1/8" diameter I.W.R.C. 6 x 37 rope the lay is approximately 7". If the tangent to tangent distance was between 7" and 14", there would be reason for concern however, the tangent to tangent distance of a typical Redundant trolley is 16 rope lays.

Also there is no problem with respect to our fleet angles since the fleet angle decreases as the block is lowered. Within the seventy years WHITING CORPORATION has been building cranes there has never been a problem with our fleet angles.

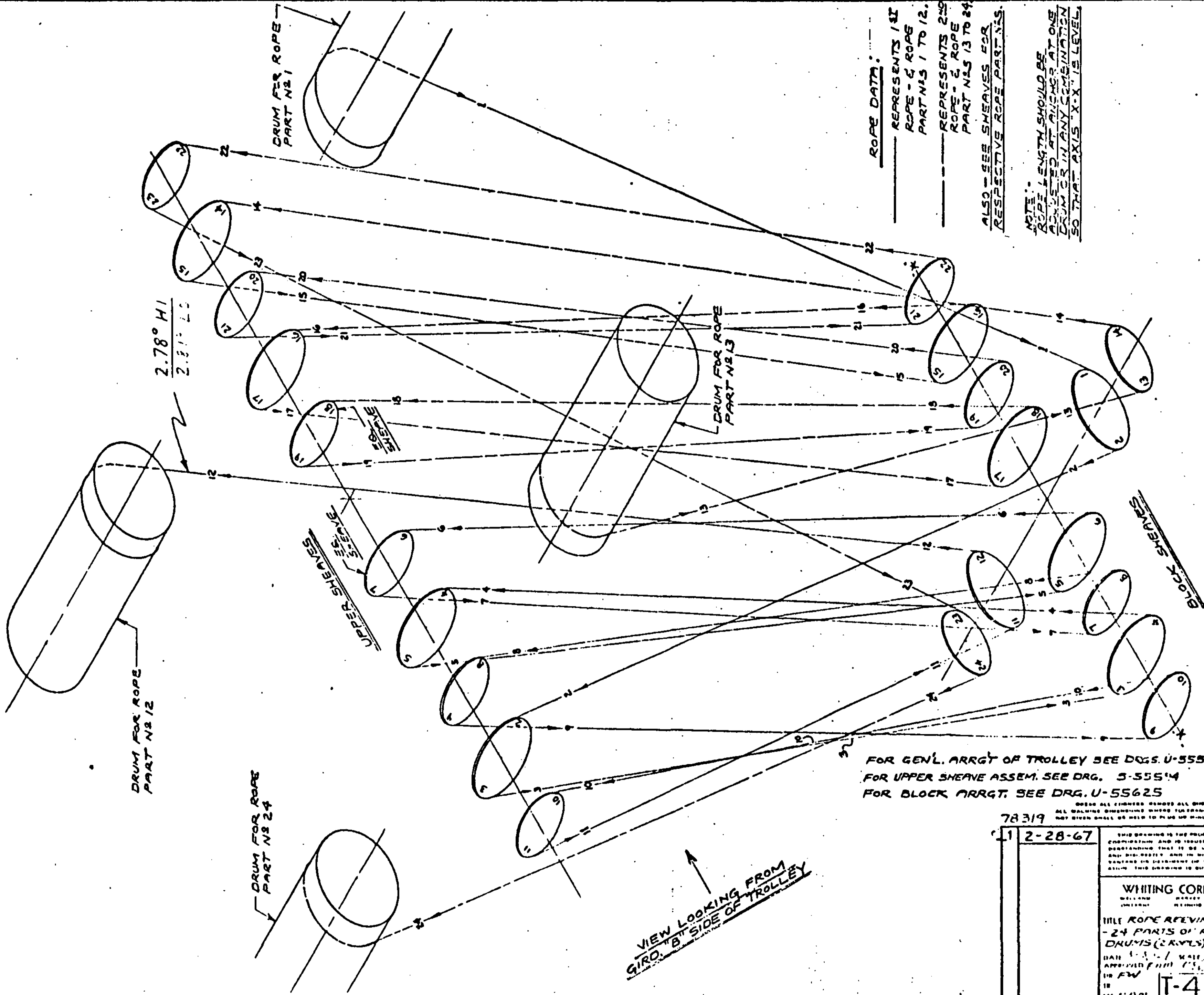
It is shown in Section III that with a minimum of 3,125 cask handlings ($24d_s$, Type 304 Super Tensile Material) the rope would last 260 years (assuming 12 cask handling cycles per year) based on fatigue life.

Summary - continued

Therefore, there is no need to increase the sheave diameters from $24d_r$ to $26d_r$ or $30d_r$, to increase the life of the rope.

REFERENCE

DRAWINGS



ROPE DATA:

—— REPRESENTS 1st ROPE - ℄ ROPE PART NOS 1 TO 12.

— — REPRESENTS 2nd ROPE - ℄ ROPE PART NOS 13 TO 24.

ALSO - SEE SHEAVES FOR RESPECTIVE ROPE PART NOS.

NOTE: - ROPE LENGTH SHOULD BE ADJUSTED AT ANCHOR AT ONE DRUM OR IN ANY COMBINATION SO THAT AXIS X-X IS LEVEL.

FOR GEN'L. ARRG'T OF TROLLEY SEE DWGS. U-55537 & T-44785

FOR UPPER SHEAVE ASSEM. SEE DRG. S-55514

FOR BLOCK ARRGT. SEE DRG. U-55625

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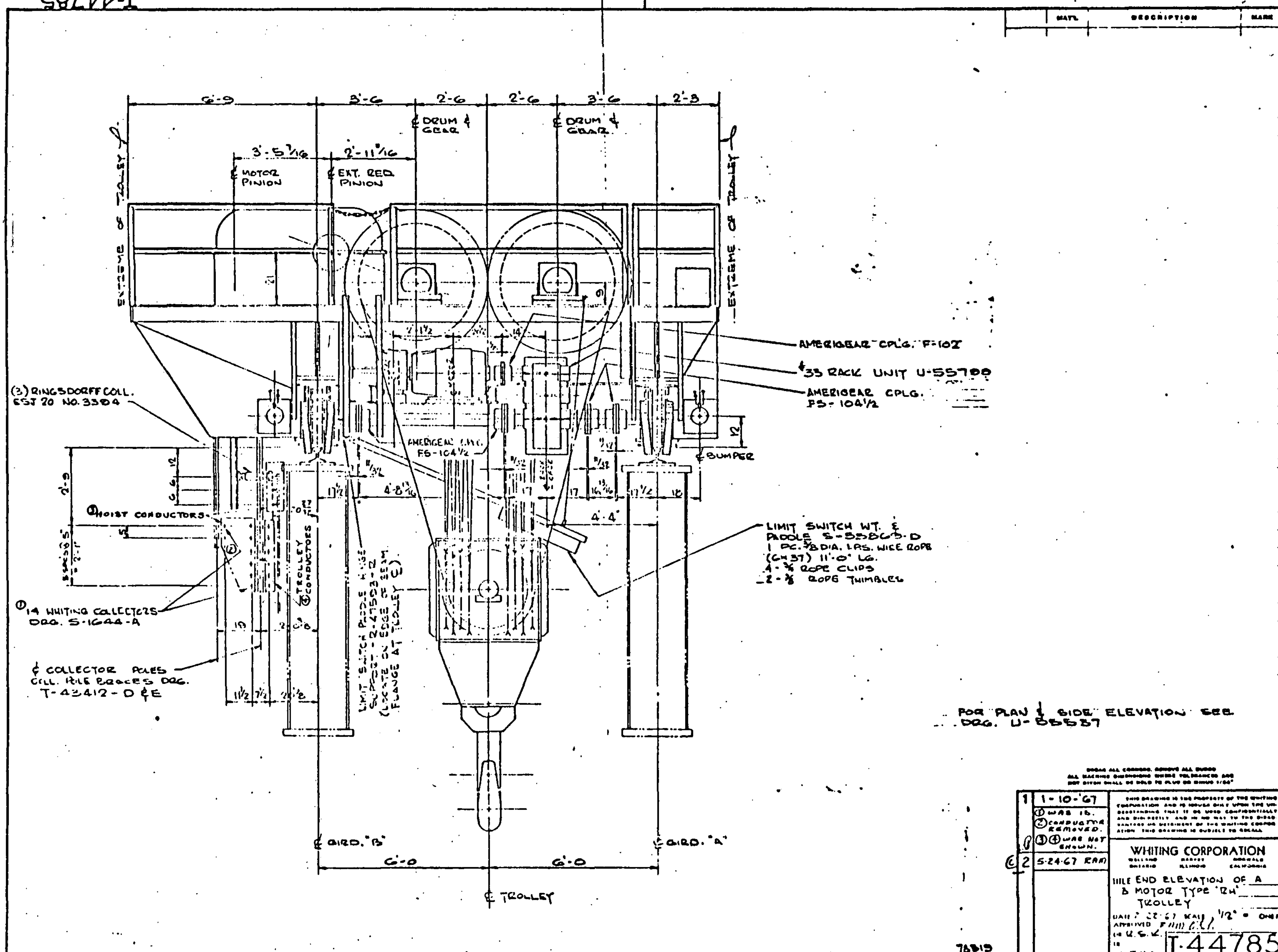
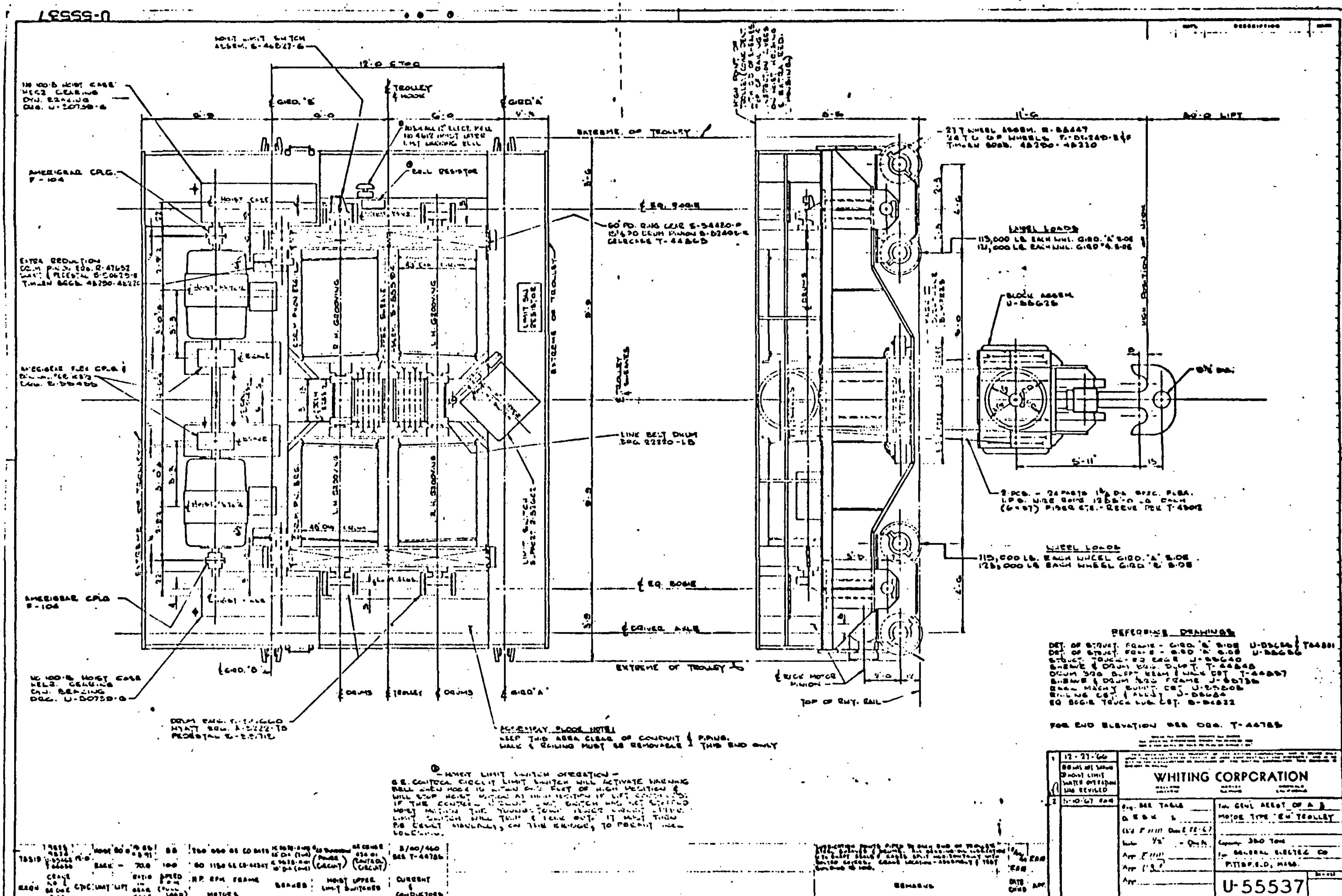
WHITING CORPORATION

TITLE ROPE REEVING DIAGRAM - 24 PARTS OF ROPE & 4 DRUMS (2 RAMPS) FOR RH TROLLEY

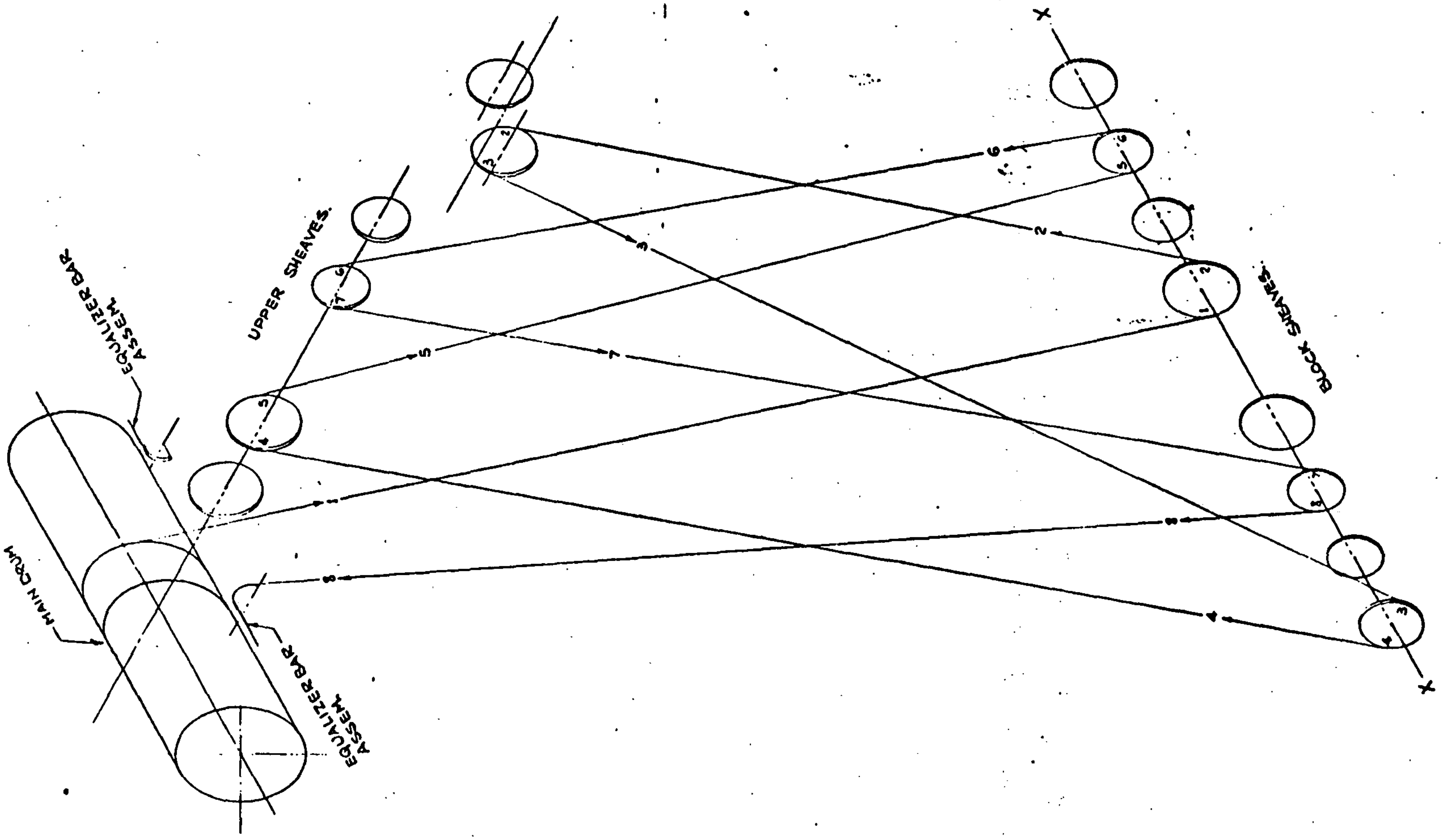
DR FW

T-45012


T-44785
MAT'L
DESCRIPTION
MARK
6'-9
3'-6
2'-6
2'-6
3'-6
2'-3
℄ DRUM & GEAR
℄ DRUM & GEAR
3'-5 7/16
2'-11 9/16
℄ MOTOR PINION
℄ EXT. RED PINION
EXTREME OF TROLLEY
AMERIGEAR CPLG. F-102
℄ 33 RACK UNIT U-55700
AMERIGEAR CPLG. FS-104 1/2
(3) RINGSDORFF COLL. ESJ 20 NO. 3304
AMERIGEAR CPLG. FS-104 1/2
℄ BUMPER
② HOIST CONDUCTORS
4'-8 13/16
4'-4"
LIMIT SWITCH WT. & PADDLE S-55863-D
1 PC. 3/8 DIA. LRS. WIRE ROPE (GH 37) 11'-0" LG.
4 - 3/8 ROPE CLIPS
2 - 3/8 ROPE THIMBLES
④ TROLLEY CONDUCTORS
① 14 WHITING COLLECTORS DRG. S-1644-A
℄ COLLECTOR POLES COLL. POLE BRACES DRG. T-43412-D & E
FOR PLAN & SIDE ELEVATION SEE DRG. U-55537
℄ GIRD. "B"
℄ GIRD. "A"
6'-0
6'-0
℄ TROLLEY
1 1-10-'67 ① WAS 16. ② CONDUCTOR REMOVED. ③④ WAS NOT SHOWN.
2 5-24-67 RAM
WHITING CORPORATION
TITLE END ELEVATION OF A 3 MOTOR TYPE 'RH' TROLLEY
SCALE 1/2" = ONE FT
T-44785
78815



U-55537
TROLLEY
EXTREME OF TROLLEY
LINE BELT DRUM
BLOCK ASSEM.
WHEEL LOADS
REFERENCE DRAWINGS
TOP OF RWY. RAIL
WHITING CORPORATION
REMARKS
MOTORS
BRAKES



MATL | DESCRIPTION | MARK

BREAK ALL CORNERS, REMOVE ALL BURRS. ALL MACHINE DIMENSIONS WHERE TOLERANCES ARE NOT GIVEN SHALL BE HELD TO PLUS OR MINUS 1/64"

L	1	2-8-72

WHITING CORPORATION

TITLE ROPE REEVING DIAGRAM
8 PARTS OF 16 - ROPE No. 1

DATE 2-17-72 SCALE ~ ONE

APPROVED H.W.F. VIG

DR A.V.

TR

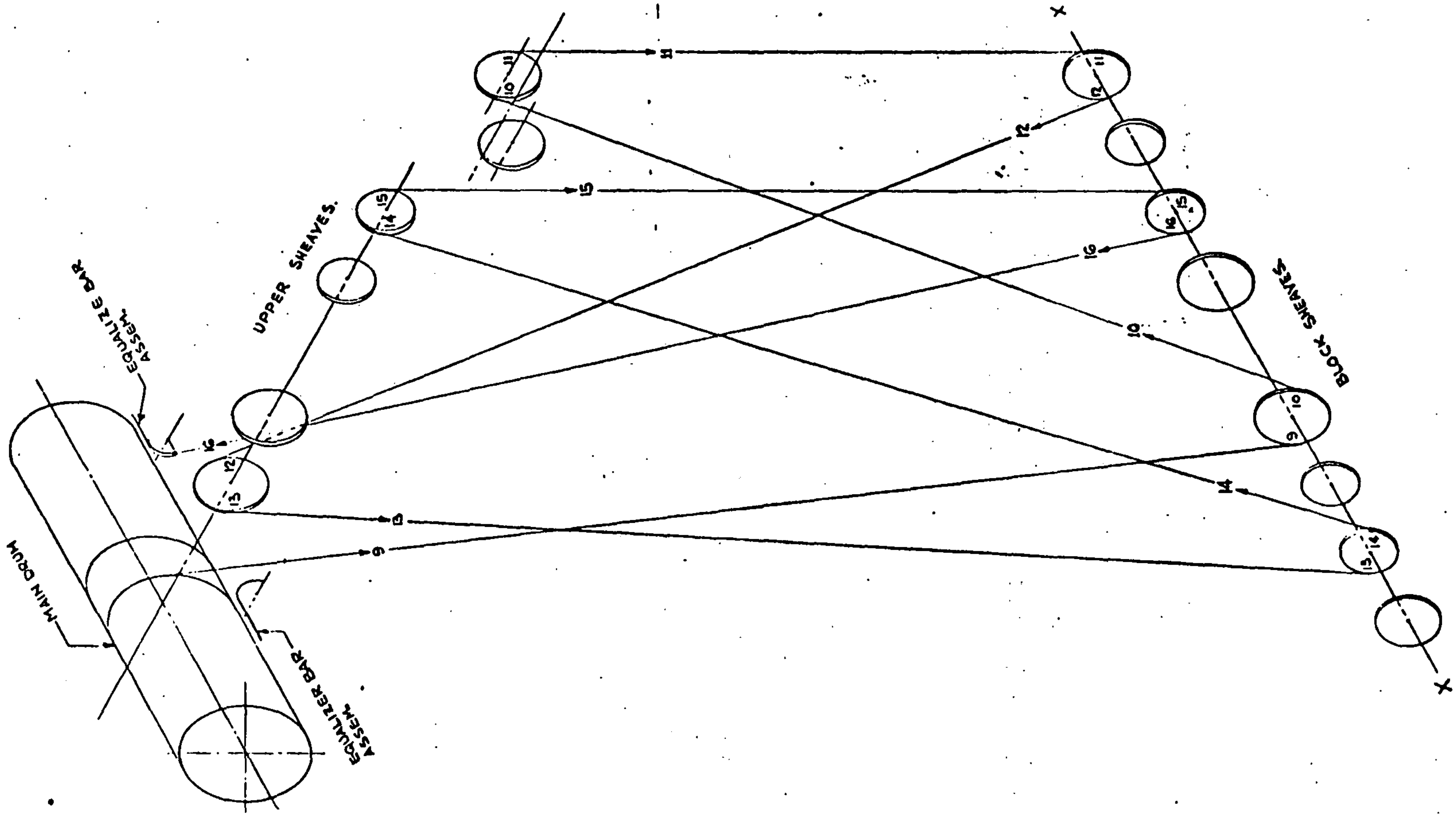
CH H.W.F.

T-53326

REQN 89115

T-53327

MAT'L	DESCRIPTION	MARK



REQ'N 89115.

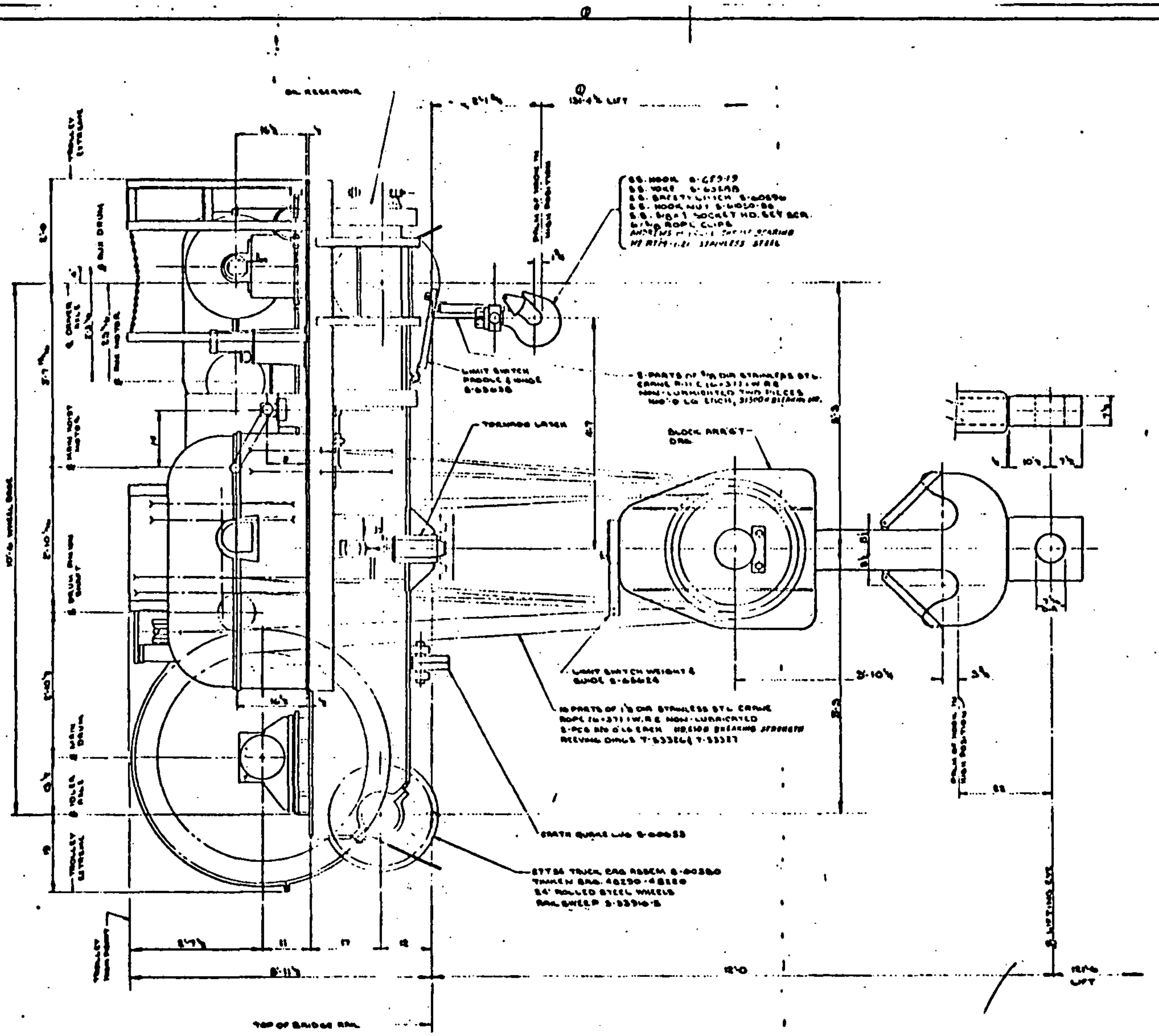
1	2-8-72

WHITING CORPORATION

TITLE ROPE REEVING DIAGRAM
8 PARTS OF 16 ROPE No. 2

DATE 2-17-72 SCALE
APPROVED N.W.F.
BY A.V.
CH N.W.F.

T-53327



MAIN HOIST REDUNDANCY

REDUNDANCY OF ROPE IS PROVIDED THROUGH THE USE OF TWO HOIST ROPES. EACH ROPE WILL BE ANCHORED TO THE DRUM AND THE EQUALIZER BAR ASSEMBLY AND WILL BE REEVED THROUGH BLOCK AND UPPER SHEAVE ASSEMBLIES SO THAT ITS EIGHT PARTS PROVIDE TWO PARTS IN EACH QUADRANT OF THE LOAD BLOCK ABOUT THE VERTICAL AXIS OF THE HOOK.
WITH BOTH ROPES EFFECTIVE, THE LOAD IS SHARED EQUALLY BY ALL SIXTEEN PARTS OF ROPE. IF ONE ROPE LOSES ITS EFFECTIVENESS, THE LOAD WILL BE SUPPORTED BY THE EIGHT PARTS OF THE REMAINING ROPE.

REDUNDANCY OF MACHINERY COMPONENTS IS PROVIDED BY TWO SEPARATE GEAR TRAINS BETWEEN THE HOIST MOTOR AND THE DRUM [illegible] OF THE MINIMUM ULTIMATE STRENGTH OF THE MATERIAL.

IN CASE OF A FAILURE IN THE ABOVE MENTIONED SYSTEMS, THE HOIST WILL REMAIN SUFFICIENTLY OPERABLE TO PERMIT POSITIONING OF THE LOAD UNDER [illegible] POWER OF [illegible] OR BY MANUAL OPERATION OF THE HOIST BRAKES FOR GRAVITY LOWERING.

FURTHER PROTECTION IS PROVIDED BY LUGS TO CATCH THE DRUM IN CASE OF SHAFT FAILURE [illegible] IN CASE OF PIN OR SWIVEL FAILURE.

THE FAILURE OF ANY OF THESE PARTS WILL NOT ALLOW FURTHER OPERATION OF THE HOIST, BUT WILL SUSTAIN THE LOAD.

THE BLOCK ASSEMBLY CONTAINS TWO LOAD CARRYING DEVICES: A LIFTING EYE AND A SISTER HOOK. A FAILURE OF EITHER THE LIFTING EYE OR SISTER HOOK WILL RESULT IN A TRANSFER OF LOAD TO THE SISTER HOOK OR LIFTING EYE RESPECTIVELY.

THE ATTACHMENTS OF THE LOAD TO THE LIFTING EYE & THE SISTER HOOK MUST BE SUCH THAT A FAILURE OF ONE ATTACHMENT WILL NOT PERMIT LOAD TO FREE FALL MORE THAN 6".

LIMIT SWITCH SYSTEM FOR CRITICAL PATH COVERAGE

THIS SYSTEM INCLUDES LOAD SENSING LIMIT SWITCH MOUNTED [illegible] THE WEIGHT OF THE LOWER BLOCK. [illegible]

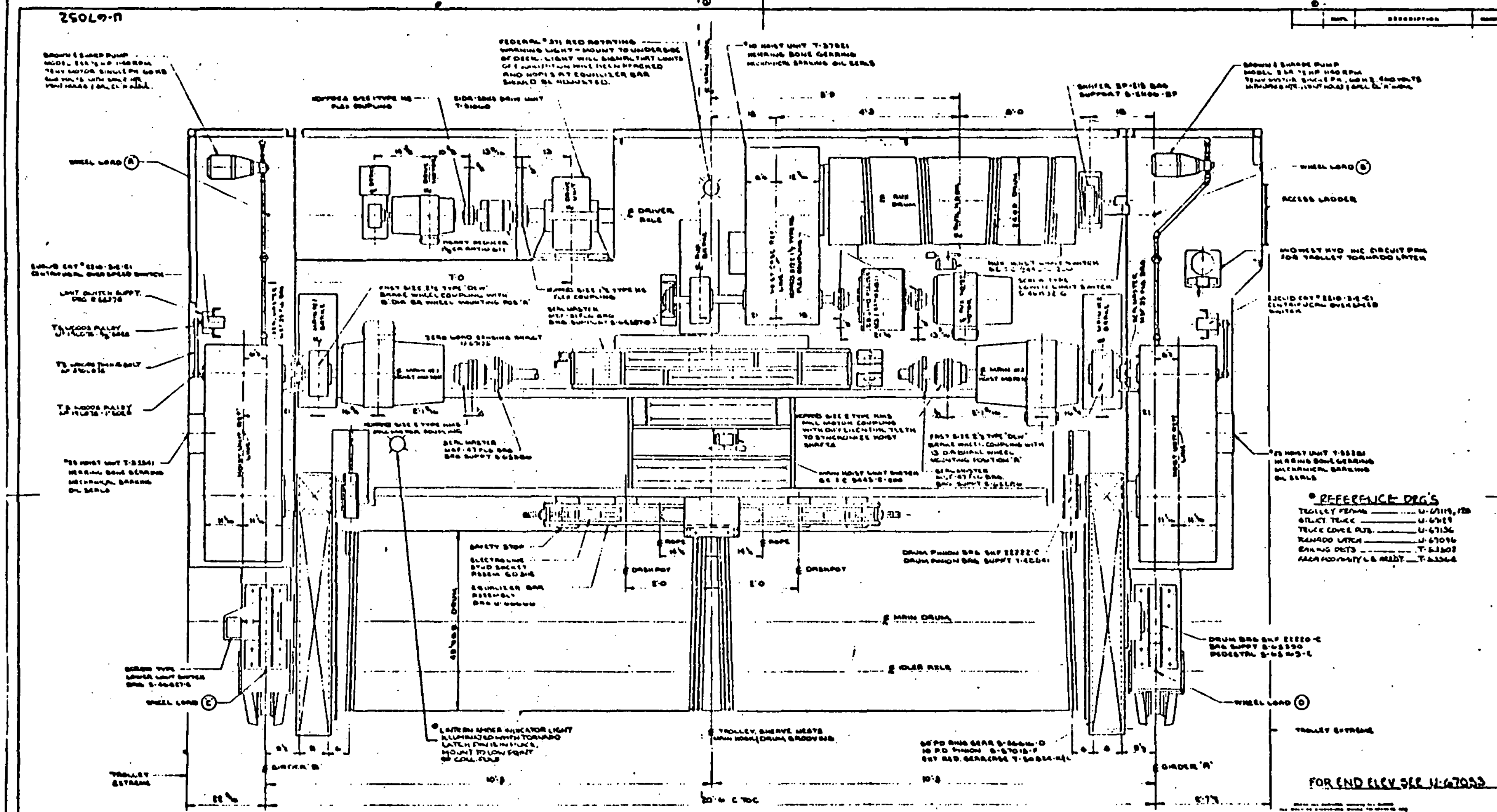
THIS TROLLEY IS DESIGNED TO WITHSTAND A POSTULATED TORNADO [illegible]

FOR PLAN VIEW SEE U-67052

WHITING CORPORATION

GEN'L ARR'G'T OF A THREE MOTOR TROLLEY
—END ELEVATION—

U-67053



REQ'N	CRANE No. AND BRIDGE No.	C. TO C.	UNIT	CAP'Y. TONS	LIFT	GEAR RATIOS	SPEED F.P.M.	H.P.	R.P.M.	FRAME	SOLENOID BRAKES	LIMIT SWITCHES UPPER	LIMIT SWITCHES LOWER	CONDUCTORS	CURRENT	WHEELLOADS A	B	C	D	REMARKS	CUSTOMER
89231 / 89224	10389 U-66720 U-66721 / 10388 U-66720 U-66721	20'-6	MAIN	125	[illegible]	[illegible]	3	2040	500	365TX	WHITING 13" [illegible]	GEN. ELECTRIC [illegible]	GEN. ELECTRIC [illegible]	[illegible] 18-17-0 COLL. SHOE [illegible]	3-60-460 460 MOTORS	[illegible]	[illegible]	[illegible]	[illegible]	ALL LOADBEARING STRUCTURAL STEEL TO CONFORM TO ASTM A588 GRADE B. SPRING BUMPERS MOUNTED ON BRIDGE. TROLLEY TORNADO LATCH WITH TAIL BELT. ZERO LOAD SENSING (AREA LIMIT SWITCHES). FAST FLEX COUPLINGS. OIL COOLERS ON MAIN HOIST CASES.	UNITED ENGINEERS & CONSTR. INC FOR CAROLINA POWER & LIGHT COMPANY. BRUNSWICK STEAM ELECTRIC PLANT. UNITS No. 1 & 2. SOUTHPORT NORTH CAROLINA
			AUX.	5	[illegible]	[illegible]	25	10	900	284TX	WHITING 13" [illegible]	GEN. ELECTRIC [illegible]	GEN. ELECTRIC [illegible]								
			DRIVE	—	—	[illegible]	40	5	1200	215T	WHITING [illegible]	—	—								

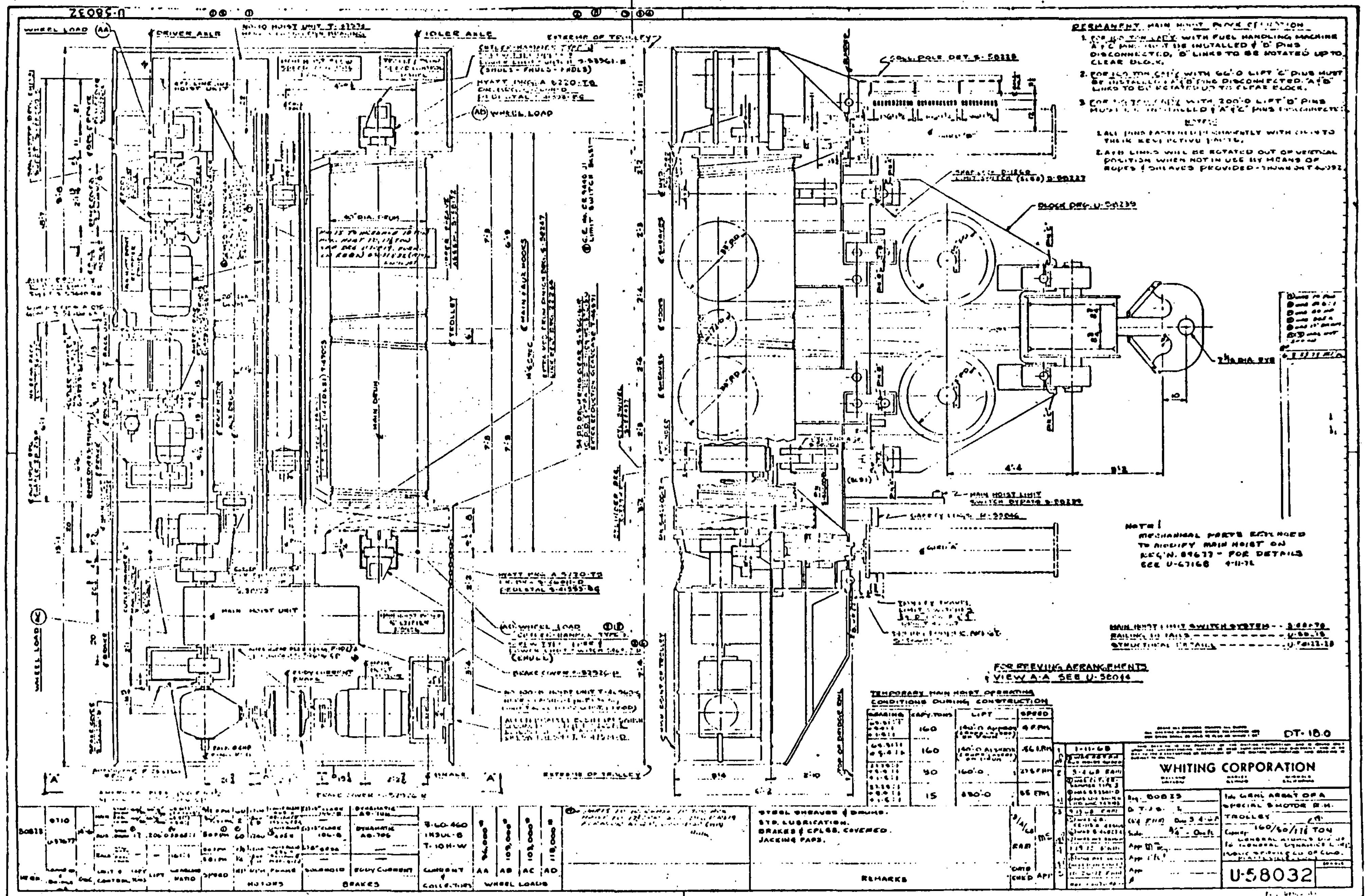
WESTINGHOUSE MOTORS 460 VOLTS

WHITING CORPORATION

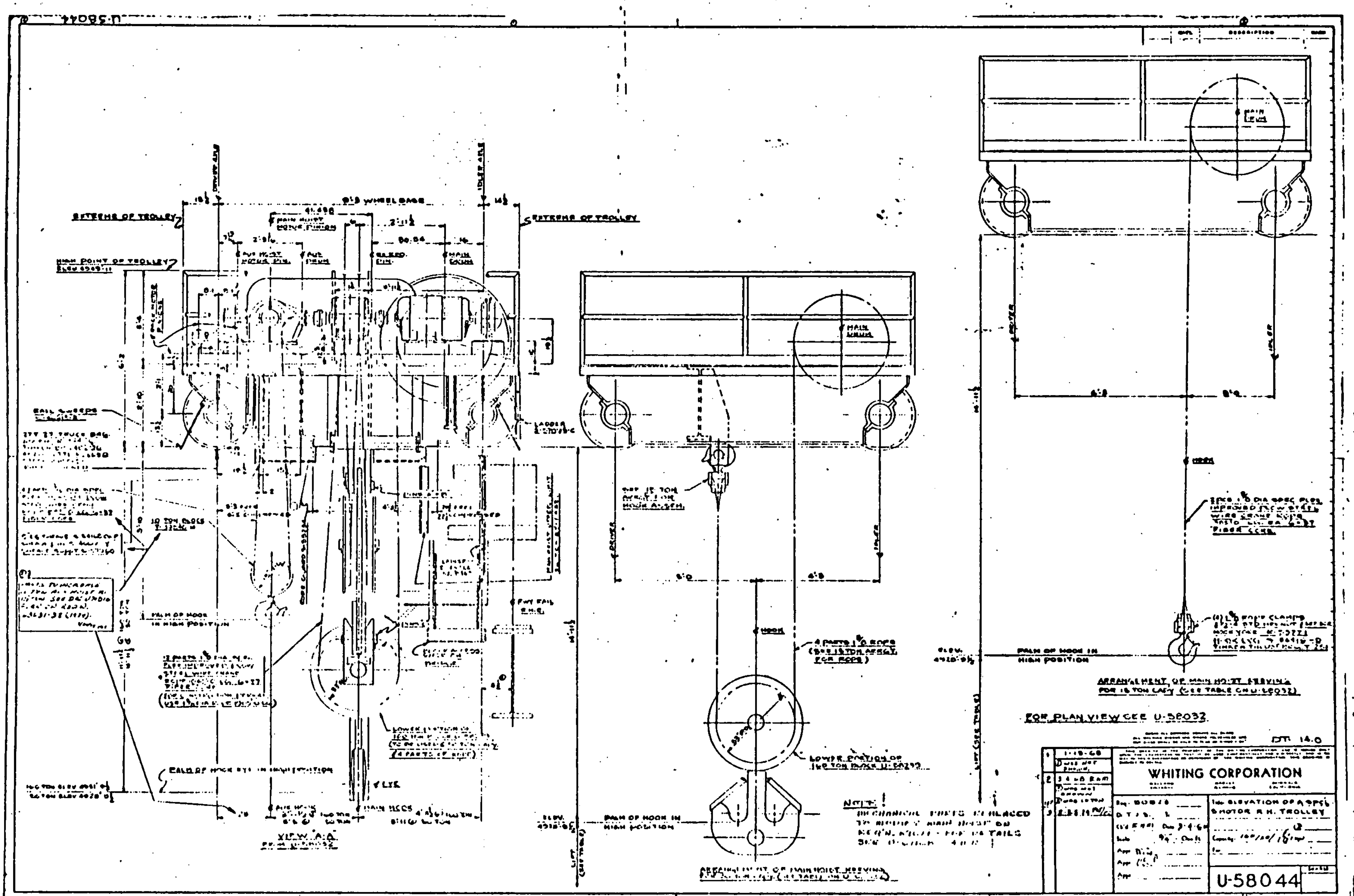
1	3-14-72
2	5-1-72
3	[illegible]

Req. SEE TABLE
Title GEN'L ARR'G'T OF A THREE MOTOR TROLLEY — PLAN VIEW —
Cust. SEE TABLE
Dr. SEE TABLE

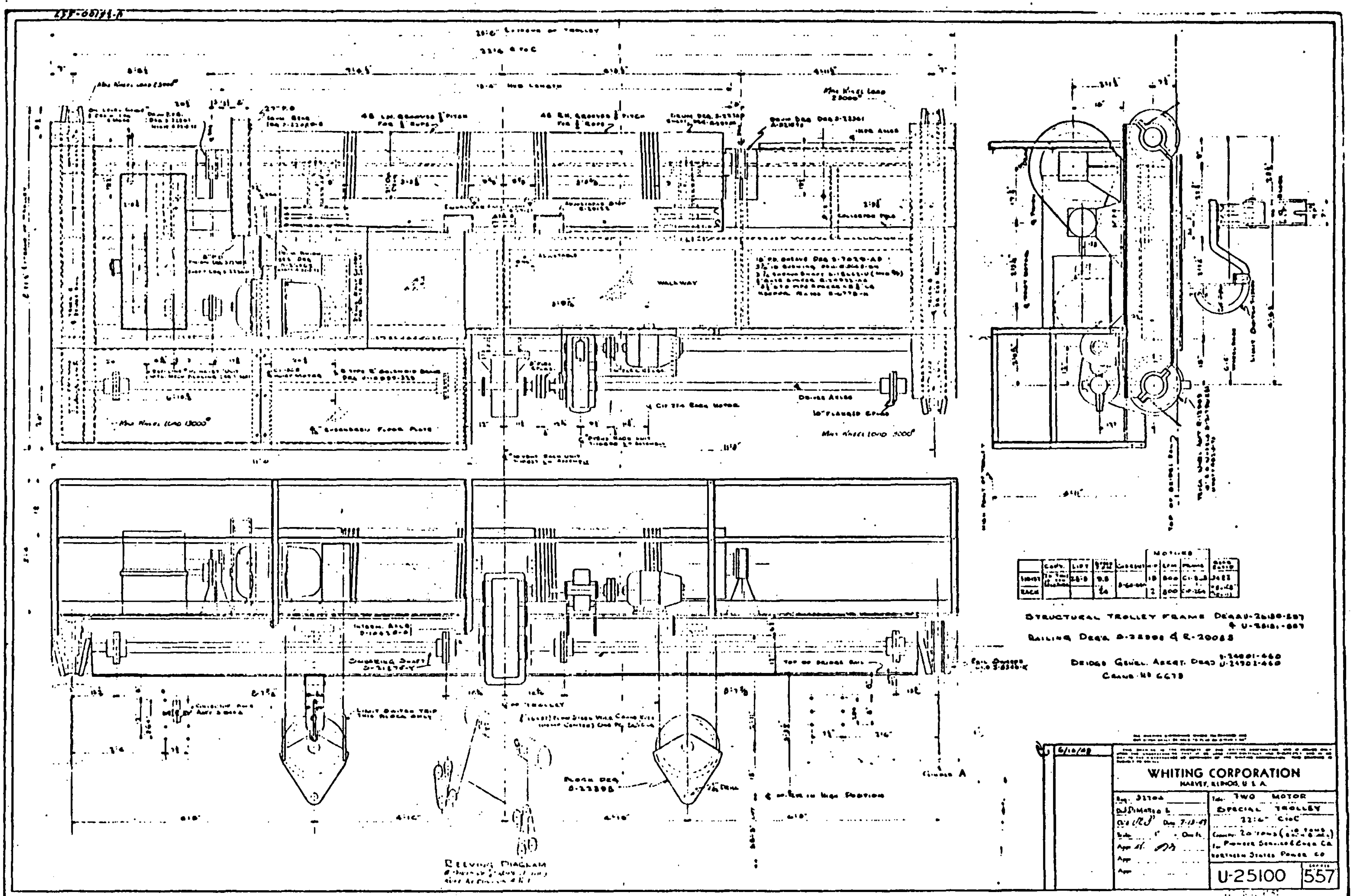
U-67052

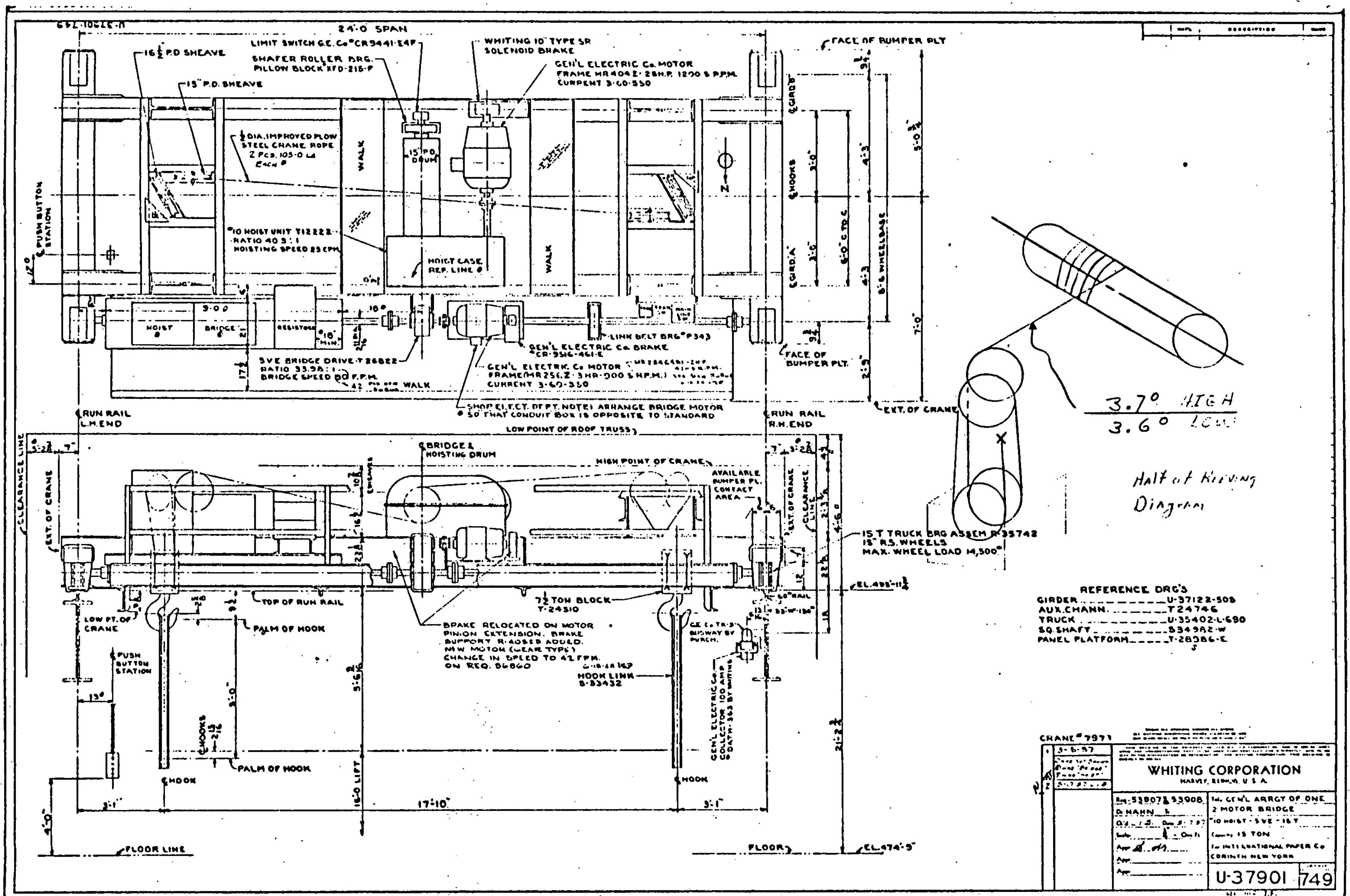


WHEEL LOAD
DRIVER AXLE
IDLER AXLE
EXTREME OF TROLLEY
MAIN HOIST UNIT
WHEEL LOAD
PERMANENT MAIN HOIST BLOCK CONNECTION
FOR REEVING ARRANGEMENTS
VIEW A-A SEE U-58044
TEMPORARY MAIN HOIST OPERATING
CONDITIONS DURING CONSTRUCTION
CAP'Y. TONS
LIFT
SPEED
160
160
90
15
160'-0
830'-0
DT-18.0
WHITING CORPORATION
TROLLEY
160/50/15 TON
U-58032
REMARKS
MOTORS
BRAKES
WHEEL LOADS



EXTREME OF TROLLEY
WHEELBASE
HIGH POINT OF TROLLEY
PALM OF HOOK IN HIGH POSITION
10 TON BLOCK
VIEW A-A
MAIN DRUM
HOOK
DRIVER
IDLER
4 PARTS 1 1/8 ROPE
LOWER PORTION OF 150 TON BLOCK
ARRANGEMENT OF MAIN HOIST REEVING
ARRANGEMENT OF MAIN HOIST REEVING FOR 15 TON CAPY
NOTE:
FOR PLAN VIEW SEE U-58032
DT 14.0
WHITING CORPORATION
ELEVATION OF A 9PSL 5 MOTOR R.H. TROLLEY
U-58044


WALKWAY
STRUCTURAL TROLLEY FRAME
RAILING
BRIDGE GENL. ASSY.
REEVING DIAGRAM
WHITING CORPORATION
HARVEY, ILLINOIS, U. S. A.
TWO MOTOR
SPECIAL TROLLEY
22'-6" C to C
Northern States Power Co
U-25100
557


24'-0 SPAN
LIMIT SWITCH G.E. Co. CR9441-E4F
WHITING 10" TYPE SR SOLENOID BRAKE
16½" P.D. SHEAVE
SHAFER ROLLER BRG. PILLOW BLOCK XFD-215-F
GEN'L ELECTRIC Co. MOTOR FRAME MR404Z · 25H.P. 1200 R.P.M. CURRENT 3-60-550
15" P.D. SHEAVE
FACE OF BUMPER PLT
½ DIA. IMPROVED PLOW STEEL CRANE ROPE 2 Pcs. 105-0 Lg. Each
WALK
15" P.D. DRUM
PUSH BUTTON STATION
10 HOIST UNIT T12222 RATIO 40.3:1 HOISTING SPEED 25 FPM
HOIST CASE REF. LINE
HOIST
BRIDGE
RESISTORS
LINK BELT BRG P343
GEN'L ELECTRIC Co. BRAKE CR-9516-461-E
5VE BRIDGE DRIVE T26822 RATIO 33.98:1 BRIDGE SPEED FPM
GEN'L ELECTRIC Co. MOTOR FRAME MR256Z 3HP 900 RPM CURRENT 3-60-550
FACE OF BUMPER PLT.
8'-6 WHEELBASE
EXT. OF CRANE
SHOP ELECT. DEPT. NOTE: ARRANGE BRIDGE MOTOR SO THAT CONDUIT BOX IS OPPOSITE TO STANDARD
RUN RAIL L.H. END
RUN RAIL R.H. END
LOW POINT OF ROOF TRUSS
CLEARANCE LINE
BRIDGE & HOISTING DRUM
HIGH POINT OF CRANE
AVAILABLE BUMPER PL. CONTACT AREA
15 T TRUCK BRG ASSEM R-35742 15" R.S. WHEELS MAX. WHEEL LOAD 14,500#
EL. 495'-11½"
TOP OF RUN RAIL
7½ TON BLOCK T-24310
LOW PT. OF CRANE
PALM OF HOOK
BRAKE RELOCATED ON MOTOR PINION EXTENSION. BRAKE SUPPORT R-40585 ADDED. NEW MOTOR (GEAR TYPE) CHANGE IN SPEED TO 42 FPM. ON REQ. 56860
HOOK LINK S-33432
PUSH BUTTON STATION
GEN'L ELECTRIC Co. COLLECTOR 100 AMP
HOOK
16'-0 LIFT
17'-10"
3'-1"
FLOOR LINE
FLOOR
EL. 474'-5"
3.7° HIGH
3.6° LOW
Half of Reeving Diagram
REFERENCE DRG'S
GIRDER U-37122-508
AUX. CHANN. T24746
TRUCK U-35402-L-690
SQ. SHAFT S34982-W
PANEL PLATFORM T-28986-E
CRANE # 7971
WHITING CORPORATION
HARVEY, ILLINOIS, U.S.A.
GEN'L ARRGT OF ONE 2 MOTOR BRIDGE
10 HOIST - 5VE - 15T
15 TON
INTERNATIONAL PAPER Co. CORINTH NEW YORK
U-37901
749

REFERENCES

Copy - [illegible]

UNIVERSAL
WIRE PRODUCTS, INC.
222 Universal Drive • North Haven, Connecticut 06473 • Area Code 203 • Tel. 865-4123

September 26, 1973

Mr. P. J Marchese, Manager
Crane Division
Whiting Corporation
Harvey, Illinois 60425

RECEIVED
OCT 1 1973
CRANE DEPT.

Dear Mr. Marchese:

This letter is in response to your letter of September 17 in which you specifically asked for comment on the "reverse bends" imposed on a 1-1/8" rope on the redundant hoist system designed and built for an atomic energy plant.

The most important drawing of the set which you sent is U-67053. This drawing shows the traveling block in the extreme up position and thus represents the most severe condition so far as fleet angle and reverse bends are concerned.

The distance from the center line of the main drum to the center line of the sheaves in the movable block (point of tangency) is 9'4" (112'). The nest of sheaves are on a center line slightly above the center of the drum although I was not confident of picking this dimension off the chart. It appears to be about 5" above the center line. The 5" difference is immaterial, however, because the 112" dimension is more than adequate to preclude any damaging effects from so-called reverse bends.

Reverse bends can be damaging to a piece of wire rope but it is our belief that for this to be of any significance whatsoever, the reverse bends must be close-coupled. That is, the distance from the point of tangency on one sheave to the point of tangency on the next sheave must be something like one or two lays of the rope. The lay of this rope would be approximately 7". If you had a tangent to tangent distance of between 7" and 14", there would be reason for some concern.

In your installation, however, you have 16 rope lays between points of tangency! This gives the rope more than ample time to recover from the last bend before being required to bend in the other direction.

Furthermore, this is not a reverse bend since the rope is not changing direction 180° but only 90° except for the lead lines which pass up to the drum; those do

4001 TORRESDALE AVE., PHILADELPHIA, PA. 19124 • 310 FAY ST., ADDISON (CHICAGO), ILL. 60101 • 1313 W. 11th ST., CLEVELAND, OHIO 44113 • 629 CONGAREE RD., GREENVILLE, S. C. 29607 • 6840 N. W. 37th COURT, MIAMI, FLA. 33147 • 1116 SILVERLAKE DR., SHREVEPORT, LA. 71129 • 22 MISSISSIPPI ST., SAN FRANCISCO, CAL. 94107 • 13 JAY ST., NEW YORK, N. Y. 10013

change direction 180^{O} but the distance is so great that the effect would be negligible.

The Federal Register, Volume 37 #202 dated October 18, 1972 in Section 1910.179 and Paragraph 65 (2) states that all overhead and gantry cranes constructed and installed after August 1971 shall meet ANSI SafetyCode B-30.2. The referenced code was written through the collaboration of a lot of good people. Among those involved was a representative from the Wire Rope Technical Board who represented the entire wire rope industry. There is nothing in that specification regarding reverse bends.

We have referred to some fatigue data in which 1/2" rope was cycled over 12" sheaves (D/d ratio = 24) at a load of approximately 1/6 of its breaking strength. A deliberate close-coupled reverse bend situation was used in order to accelerate the fatigue test. The rope ran between 16,000 and ~~an~~ 18,000 cycle before termination of the test. Termination was either complete failure of the rope or failure of one strand in the rope (don't confuse one strand with one wire). We cite these data not to condone reverse bends per se but to show that reverse bends are damaging only when close-coupled and even then after a considerable number of cycles.

It should also be noted that the above reference fatigue test was run at approximately 4 cycles per minute at which speed sufficient heat was generated in the rope to make it inadvisable to touch the rope with the bare hands. This drives the lubricant out and acts against good fatigue results. In other words, had this rope been operating under the same conditions, but at a slow enough rate to avoid such heat rise, it undoubtedly would have gone considerably longer. Furthermore, had the spacing of the sheaves been opened up to two or three lays, the rope probably would have gone two or three times as long.

It is our opinion that the reverse bend which is involved in your design would produce negligible results so far as rope deterioration was concerned. Most certainly, the inspection requirements that now exist for all such equipment under OSHA rules would be more than adequate to detect broken wires which might develop at such imagined stress concentration points.

Finally, consider the relative movement of the block with respect to the drum. A section of rope which is just tangent to the drum will have to move considerably more than 112" in order to become tangent to the block because as the rope starts to move, the block is moving away from it and this further increases the effective distance between tangent points.

We just can't agree that there is a problem.

Very truly yours,
Universal Wire Products, Inc.

R. P. Ramsey

Richard P. Ramsey, Chief Engineer

RPR/p
cc: D.A. Tuckerman
Ron Space

Reference #1

FEDERATION EUROPEENNE DE LA MANUTENTION

SECTION I

HEAVY LIFTING EQUIPMENT

RULES FOR THE DESIGN OF HOISTING APPLIANCES

Information from :
Secretariat de la Section I
M. SICOT - 10, avenue Hoche
75 - Paris (8e) - Tel. 622-38-00

Reference #2

2nd EDITION - DECEMBER 19

Table T - 2,5311
Values of H_1

Group of mechanism	Drums		Pulleys		Compensating pulleys	
	Normal rope	non-rotating rope	Normal rope	non-rotating rope	Normal rope	non-rotating rope
1_{Bm}	16	16	16	18	14	16
1_{Am}	16	18	18	20	14	16
2_m	18	20	20	22,4	14	16
3_m	20	22,4	22,4	25	16	18
4_m	22,4	25	25	28	16	18
5_m	25	28	28	31,5	18	20

2,5312 Values of H_2

For drums and compensating pulleys, $H_2 = 1$ with any type of reeving.

For pulleys, the values of the coefficient H_2 depend upon the number of pulleys in the reeving and upon the number of reverse bends (S bends), the compensating pulleys not being counted in the number of bends.

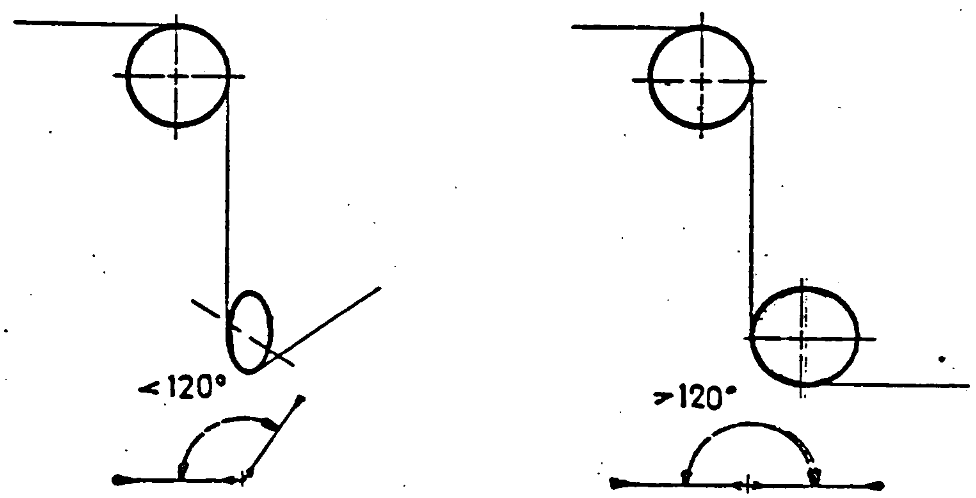
Taking the value of $W = 1$ for a drum
$W = 2$ for a pulley carrying the rope in the same direction of wrap (no reverse bend)
$W = 4$ for a pulley carrying the rope in the reverse direction of wrap (reverse bend).
$W = 0$ for a compensating pulley.

Then the total W_T is the sum of these values of W for a given rope reeving and the corresponding values of the coefficient H_2 are indicated in the table hereunder :

Table T - 2,5312
Values of H_2

W_T	$\leqslant 5$	6 to 9	$\geqslant 10$
H_2	1	1,12	1,25

If 2 adjacent planes of bending make between them an angle less than 120° it is agreed that there is no reverse bend. (see sketch below).



Copy E.C. [illegible]

UNIVERSAL
WIRE PRODUCTS, INC.
222 Universal Drive • North Haven, Conn. 06473 • 203/865-4123 • Telex 96-3464

October 25, 1973

RECEIVED
OCT 29 1973
CRANE DEPT.

Mr. P. J. Marchese
Whiting Corporation
Harvey, Illinois 60426

Dear Mr. Marchese:

This letter is in further response to your letter of September 17 and subsequent telephone conversations, and concerns the fleet angle on the redundant crane design for the Atomic Energy Commission.

I have made a graphic analysis of the change in fleet angle of the lead lines on this crane.

The original fleet angle based on a 1/12 ratio is 4.76°. By the time the drum has turned one revolution, the point of contact on the drum has moved over 1.25" and the reach has increased from 9'4" to 10.56'. This reduces the effective fleet angle to 3.64°. The next revolution decreases the angle to 2.75°.

According to my calculations, the angle at which the lead line just contacts the adjacent wrap is 2.88°. I would not consider this slight convergence serious. After extended use, you might observe somewhat greater abrasive wear at this point on the rope than elsewhere on the line but it certainly would not create any hazard nor seriously impair the serviceability of the rope.

Whiting cranes have been designed to these parameters for many, many years with no adverse results. Based on this actual experience plus the calculations, I really don't feel the problem exists. Certainly, the area in question is a part of the rope which is usually examined and should any problem develop, it certainly would be caught by routine crane inspection.

Very truly yours,

Universal Wire Products, Inc.

R. P. Ramsey

Richard P. Ramsey
Chief Engineer

RPR/p
cc:D. A. Tuckerman
R. Space

Reference #3

APPENDIX

9'-4" ≈ Distance between drum ℄ and block sheaves ℄ with block in full up position

10'-1½" ≈ Distance between upper sheaves (4-5 & 6-1) ℄ and block sheaves ℄ with block in full up position

8'-2" ≈ Distance between upper sheave (2-3) ℄ and block sheaves ℄ with block in full up position

10'-11¼" ≈ Distance between equalizer bar rad of cur. and block sheaves ℄ with block in full up position

		½ circ.
27" P.D.	sheave	42.39"

It was assumed that the rope is 1⅛ dia. 6X37 IWRC with all sheaves being 24d = 27" P.D. sheaves
d = rope dia

∴ 100' Lift

with block lowered 100' — Rope #8 ≈ 110'-11¼"
≈ 1331.25"

Total Length of Rope not on Drum with block in full up position ≈ 7(42.39") + 9'-4" + 2(8'-2") + 4(10'-1½") + 10'-11¼"
≈ 1,221.98"

∴ For a lift of 100', a point which pass over the max. no. of sheave is located on rope #8 and it pass over every sheave and is now on the drum with the block in the full up position.

The "point" which is refered to thruout this report is the location at which rope (#8) comes in contact with sheave (7-8) when the block is in it's lowered position (100' from it's full up position

∴ 3' Lift

The "point" which was on rope # 8 with the block lowered 100' now (full up position) is wrapped around the drum 109.27"

Total Length of Rope not on the Drum with block lowered 3' from full up position of block. $\approx 7(42.39")+ 12'-4"+2(11'-2")+4(13'-1\frac{1}{2})+13'-11\frac{1}{4}" \approx 1509.9$

from p #1 Total Length of Rope not on Drum with block in full up position $\approx$ 1221.98

288" is unwrapped from the drum

288" − 109.27" = 178.73" is the distance the "point" travel from the ℄ of the drum.

The distance which the "point" must travel to pass over one sheave is 12'-4" + ½ circ. of 27" P.D. sheave $\approx$ 190.37"

∴ The "point" will cover 30.73" of the 42.39" surface of one sheave.

40' Lift

The "point" which was on rope #8 with the block lowered 100' now (full up position) is wrapped around the drum 109.27"

Total Length of the Rope not on the Drum with the block lowered 40' from full up position of the block is $\approx 7(42.39") + 49'-4" + 2(48'-2") + 4(50'-1\tfrac{1}{2}")$ $+ 50'-11\tfrac{1}{4}" \approx 5{,}061.98"$

from p#1 Total Length of Rope not on Drum with block in full up position $\approx 1{,}221.98"$

3840" is unwrapped from the drum

3840" − 109.27" = 3730.73" is the distance which the "point" travel from the ℄ of the drum

Distances with block lowered 40'

Equalizer bar ℄ to the block sheaves ℄ + ½ circ. of (1-2) sheave ≈ 653.6
Block sheave (1-2) ℄ to sheave (2-3) ℄ + ½ circ of (2-3) sheave ≈ 620.3
sheave (2-3) ℄ to Block sheave (3-4) ℄ + ½ circ of (3-4) sheave ≈ 620.3
Block sheave (3-4) ℄ to sheave (4-5) ℄ + ½ circ. of (4-5) sheave ≈ 643.8
sheave (4-5) ℄ to Block sheave (5-6) ℄ + ½ circ. of (5-6) sheave ≈ 643.89
Block sheave (5-6) ℄ to sheave (6-7) ℄ + ½ circ. of (6-7) sheave ≈ 643.89
sheave (6-7) ℄ to Block sheave (7-8) ℄ + ½ circ of (7-8) sheave ≈ 643.89

∴ The "point" passes over 5 sheaves

The "point" on each rope passes over 24 sheaves (27" P.D.) plus part of 4 more (see p#2) sheaves (27" P.D.) per cask handing.

Direct stress

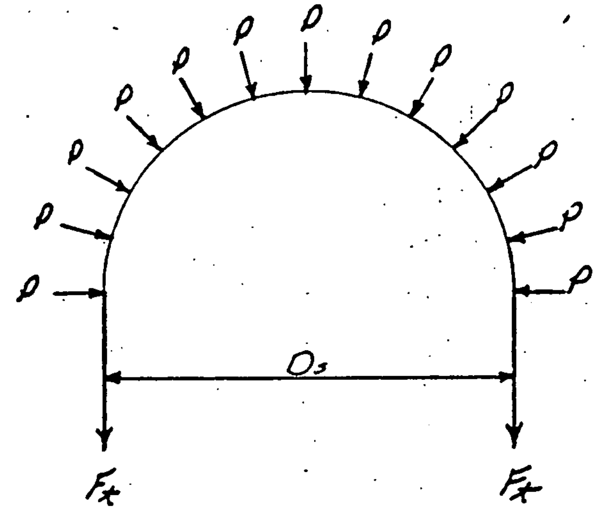
$\sigma = F_t / A_r$

Bending stress

$\sigma_b = E D_w / D_s$

Pressure

$p = 2F_t / (D_r D_s)$



27" P.D. SHEAVES = 24 D_r

1⅛ Dia. rope 6X37 IWRC

Type 304 Super Tensile

Bright Super Tensile Monitor AAA

$A = .471\,(1.125)^2 = .596\,in^2$

$E_w = 28{,}000{,}000.$ psi (Type 304 Super Tensile)

$E_w = 29{,}000{,}000.$ psi (Bright Super Tensile Monitor AAA)

$F_t = (125 \times 2000)/16 = 15{,}625\ \#$

$\sigma = F_t / A_r = 15{,}625\#/.596\,in^2 = 26{,}216.$ psi

$\sigma_b = E D_w / D_s = [28.\times10^6\,psi\,(1/22 \cdot 1.125")]/27" = 53{,}030.$ psi

(Type 304 Super Tensile)

$\sigma_b = E D_w / D_s = [29.\times10^6\,psi\,(1/22 \cdot 1.125)]/27" = 54{,}924.$ psi

(Bright Super Tensile Monitor AAA)

$P = 2F_t/(D_r D_s) = 2(15,625\#)/(1.125" \cdot 27") = 1028.8 \#/in^2$

$S_{ULT.}$ = 200,000 psi (Type 304 Super Tensile)
240,000 psi (Bright Super Tensile Monitor AAA)

From figure # 12-23, p. 484 "DESIGN OF MACHINE ELEMENTS", M.F. SPOTTS

75,000 Bends to Failure (Type 304 Super Tensile)

100,000 Bends to Failure (Bright Super Tensile Monitor AAA)

24. bends / cask handing from p. 5
3,125. cask handings (Type 304 Super Tensile)
4,166. cask handings (Bright Super Tensile Monitor AAA)

29 1/4" P.D. SHEAVES = 26 D_r

σ = 26,216. psi from p. 6

$\sigma_b = E D_w / D_s = [28. \times 10^6 psi (1/22 \cdot 1.125")] / 29.25" = 48,951. psi$
(Type 304 Super Tensile)

$\sigma_b = E D_w / D_s = [29. \times 10^6 psi (1/22 \cdot 1.125")] / 29.25" = 50,699. psi$
(Bright Super Tensile Monitor AA

$P = 2F_t / (D_r D_s) = 2(15,625\#) / (1.125" \cdot 29.25") = 950. psi$

$S_{ULT.}$ = 200,000 psi (Type 304 Super Tensile)
240,000 psi (Bright Super Tensile Monitor AAA)

From figure # 12-23, p. 484 "Design of Machine Elements",
M.F. Spotts

80,000 Bends to Failure (Type 304 Super Tensile)

120,000 Bends to Failure (Bright Super Tensile Moniton AAA)

24 bends/cask loading from p#5
3,333. cask Loading (Type 304 Super Tensile)

5,000. cask Loading (Bright Super Tensile Moniton AAA)

33 3/4" P.D. SHEAVES = 30 D_r

σ = 26,216 psi from p. 6

$\sigma_b = E D_w/D_s = [28. \times 10^6 psi (1/22 \cdot 1.125")] / 33.75" = 43,424. psi$
(Type 304 Super Tensile)

$\sigma_b = E D_w/D_s = [29. \times 10^6 psi (1/22 \cdot 1.125)] / 33.75" = 43,939. psi$
(Bright Super Tensile Moniton AAA)

$P = 2F_t/(D_r D_s) = 2(15,625\#)/(1.125" \cdot 33.75") = 823.$

$S_{ULT.}$ = 200,000 psi (Type 304 Super Tensile)
240,000 psi (Bright Super Tensile Moniton AAA)

from figure # 12-23, p. 484 "Design of Machine Elements"
M.F Spotts

110,000 Bends to Failure (Type 304 Super Tensile)

140,000 Bends to Failure (Bright Super Tensile Moniton AAA)

24 Bends/cask loading from p#5
4,583. cask Loading (Type 304 Super Tensile)

5,833. cask Loading (Bright Super Tensile Moniton AAA)