NRC FORM 195 (2-76)

U.S. NUCLEAR REGULATORY COMMISSION

DOCKET NUMBER 50-263

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TO: Nr. Dennis L. Ziemann

FROM: Northern States Power Company
Minneapolis, Minnesota
L. O. Mayer

DATE OF DOCUMENT 2/28/77

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DESCRIPTION

Ltr. re our 2/11/76 ltr. and their 11/22/77 ltr....trans the following:

PLANT NAME:
Monticello (1-P)

ENCLOSURE

Encl. 1 - Consists of response to request for additional information regarding Redundant Reactor Building Crane....

Encl. 2 - Changes and Corrections to Design Report for Redundant Reactor Building Crane....

(14-P)

ACKNOWLEDGED
DO NOT REMOVE

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CONTROL NUMBER 2204

NSP

NORTHERN STATES POWER COMPANY

MINNEAPOLIS, MINNESOTA 55401

Regulatory Docket File

DOCKETED USNRC Mar 3 1977 U.S. NUCLEAR REGULATORY COMMISSION Mail Section

RECEIVED MAR 3 1977 U.S. NUCLEAR REGULATORY COMMISSION Mail Section

February 28, 1977

Mr Dennis L Ziemann, Chief
Operating Reactors Branch #2
Division of Operating Reactors
U S Nuclear Regulatory Commission
Washington, DC 20555

Dear Mr Ziemann:

MONTICELLO NUCLEAR GENERATING PLANT
Docket No. 263 License No. DPR-22

Redundant Reactor Building Crane

The information contained in Enclosure (1) of this letter is in response to the request for additional information attached to your letter dated February 11, 1976.

Enclosure (2) contains a number of changes and corrections to our November 22, 1976 license submittal which we have found necessary during the final design stages of the redundant trolley.

Yours very truly,

L.O. Mayer

L O Mayer, PE
Manager of Nuclear Support Services
(Chairman-Safety Audit Committee)

LOM/LLT/ak

cc: J G Keppler
G Charnoff
MPCA
Attn: J W Ferman

2204

Enclosures

Enclosure to NSP Letter
dated February 28, 1977

Responses to Request for Additional Information

QUESTION 1: Your submittal states "The entire crane will be evaluated for the additional weight, load requirements and operating conditions imposed by the new trolley design". Considering the new trolley weight of 128,000 pounds compared with the old trolley weight of 62,000 pounds describe and discuss how this increased trolley weight has been accommodated in the unmodified portions of the system without reducing the 85 ton load rating of the crane. The discussion should include the changes in the factor of safety as well as physical modifications that have been made to retain the same load rating.

RESPONSE 1: An analysis has been conducted to determine the effects of the increased trolley weight on the unmodified portions of the system. The critical load bearing components selected for this review are:

(a) Bridge Girder
(b) Crane Girder
(c) Building Column

Analysis Assumptions

1. The weight of the new trolley is 99,000 pounds. This revised number is based on the latest evaluation of the new trolley weight.

Analysis Method

The analysis methods used in this evaluation are in accordance with the applicable governing codes delineated in Table 1.

The methods used in the evaluation are in accordance with the original design criteria for Monticello, which is keeping with the statements on page 3-8 of our November 22 submittal. The design codes and loading conditions applicable at the time of the original installation did not include the lifted load in the seismic analysis because of the extremely low probability of both events occurring simultaneously.

Results

Table 1 presents a summary of the analysis results. Included in this table are the governing load combinations, applicable governing codes, and a comparison of the original and new factors of safety. These results indicate that the factor of safety of all critical components of the crane system with the increased trolley weight are in excess of one.

Conclusion

It can be stated in conclusion that the unmodified structural system can retain the additional trolley weight together with the 85 ton rated load without exceeding the allowable stress limits.

TABLE 1

SUMMARY OF FACTORS OF SAFETY

ITEM	GOVERNING LOAD COMBINATION	ORIGINAL FACTOR OF SAFETY*	NEW FACTOR OF SAFETY*	GOVERNING CODE/ALLOWABLE
Bridge Girder	DL + LL + I	1.24	1.10	C.M.A.A. Specification #70
	DL + E_s	2.66	2.16	1.6 X C.M.A.A. Specification #70
Crane Girder	DL + LL + I	1.19	1.07	AISC Sixth Edition
	DL + E_s	1.26	1.16	0.9 f_y
Building Column	DL + SL + I	1.67	1.54	AISC Sixth Edition
	DL + SL + E_s	1.24	1.19	1.6 X AISC Sixth Edition

*Factor of Safety = $\frac{\text{Allowable Stress}}{\text{Actual Stress}}$ (Factor of Safety Against Failure Would be Greater)

DL - Deal Load
LL - Live Load
I - Impact
SL - Snow Load
E_s - Safe Shutdown Earthquake

QUESTION 2: In Section 3.3, item C.4.b, you state that subjecting the hoisting machinery and reeving to either the "two block" or "load hangup" test would be in violation of the ANSI B 30.10 standard on hooks. Justify the above statement by indicating how either of these tests violates ANSI B 30.10.

RESPONSE 2: Section 3.3, Item C.4.b incorrectly referenced A.N.S.I. B30.10. O.S.H.A. 1910.179, Paragraph (k) (2) should have been referenced instead. That paragraph states in part that: "Test loads shall not be more than 125% of the rated load unless otherwise recommended by the manufacturer." Both the "two-block" and "load hangup" tests would exceed 125% of the rated load. Industry practice is to not perform the load tests on cranes in excess of 125% of rated load.

QUESTION 3: Describe, discuss and compare the peak loads experienced in the event of a "load hangup" by the presently proposed hoist overload protection system relative to that which would be experienced if compliance with items C.3.j and C.3.1 of Regulatory Guide 1.104 were attained. The discussion should include consideration of the elapsed time before the hoist motor was tripped, the kinetic energy stored in the system, and the load change as a function of time during a "load hangup" event, as well as the assumed distance between load blocks when the hangup occurs.

In addition, describe the tests and time intervals between the tests which will verify the calibration and functional capability of the proposed hoist overload protection system.

RESPONSE 3: The analysis requested would be nearly impossible to perform without imposing highly conservative and, therefore, unrealistic assumptions on the analysis.

To eliminate the possibility of a "load hangup" occurring, power to the trolley and bridge motors will be locked out during the hoisting or lowering of any critical load in the equipment hatch. This is the only area in the Reactor Building where the potential for "load hangup" exists.

Since the potential for "load hangup" does not exist, the overload detection system will not be called on to perform any protection functions. Therefore, there is no need to verify the calibration and functional capability of the system.

QUESTION 4: Item 35 of Section 3.0 of your submittal indicates that in the event of a rope failure, a velocity actuated valve is actuated to create a large pressure drop across the hydraulic cylinder, causing it to act as a dashpot to reduce the shock on the intact reeving and structure. In this regard, provide the following:

(1) A description of the velocity actuated valve, and how the system generates the appropriate signal causing it to be actuated;

(2) The test methods that will be employed to verify its functional capability; and

(3) The time interval between the tests that verify its functional capability.

RESPONSE 4:

(1) The velocity actuated valve operates on the principal that the pressure drop across the device is proportional to the flow rate or velocity through it. At a preset velocity, the pressure drop is high enough to cause a piston to move, blocking the flow. In the event of a rope failure, flow of hydraulic fluid at any rate greater than or equal to the preset value of the velocity actuated valve, will be blocked from the low resistance part of the circuit. The fluid will only have the path afforded by the sequence valves which will offer a high resistance to flow.

(2) The system will be tested at the manufacturer's site. It will be mounted in a suitable fixture and the cylinder rod will be activated at a velocity above the specified velocity required for actuation. This test will verify that the proper sized velocity actuated valve has been used and that all connections have been properly made.

(3) The velocity actuated valves are composed of a spring and a plug in line with the flow of hydraulic fluid. The design is extremely simple and, therefore, the likelihood of failure is extremely remote. In addition we have placed two valves in series to provide protection in the event a valve does fail. Therefore, there are no plans to periodically test the velocity actuated valves.

QUESTION 5: With regard to the two hydraulic cylinders which act as load equalizers, provide the following information:

(1) The means provided to detect the loss of hydraulic fluid and alert the operator; and

(2) The measures taken to preclude the loss of hydraulic fluid.

RESPONSE 5:

(1) The load equalizer cylinders are pressurized in a closed system, therefore loss of hydraulic fluid will result in a decrease in the closed system pressure. An electric pressure switch, included in the system, will send a signal at a specified low pressure level.

(2) Loss of hydraulic fluid is precluded by the manifolding of all valving in blocks at each end of the cylinders, with only a single tube between manifold blocks.

QUESTION 6: Item C.3.p, Section 3.3 of your submittal cites information on pages ED-19 and 20 of AISE Standard No. 6, Specification for Electric Overhead Traveling Cranes for Steel Mill Service, to support the statement that the 110 percent horsepower limitation is not compatible with the established drive motor requirements. The factor K_a on page ED-19 appears to be applicable only to AC and Adjustable Voltage Motors (Without Field Weakening). Your submittal indicates that the existing General Electric Company Maxspeed drive systems utilize direct current motors in which both the field and armature currents are varied.

Provide further clarification on how the information on pages ED-19 and 20 of AISE Standard No. 6 is applicable to the Maxspeed drive systems and hence that the 110 percent horsepower limitation is not compatible with the drive requirements.

Further, from the information in Table E.4.C.2.I of AISE Standard No. 6, it appears that the overall friction factor for the trolley should be 12 pounds per ton rather than the 15 pounds per ton used in your item C.3.p. This value would result in a reduction in the full load running horsepower requirements and a corresponding reduction in the 110 percent horsepower limitation. With regard to the above, provide the following additional information:

(1) Explain why the 12 pounds per ton would not be the more appropriate value to use in this calculation; and

(2) Assuming the 12 pounds per ton is a more appropriate value, describe how it alters your conclusions.

<u>RESPONSE 6:</u> The references to Pages ED 19 and 20 of the A.I.S.E. Manual were to show a typical example of the difference between the full load running horsepower and the connected horsepower of a trolley or bridge. The trolley is equipped with a General Electric Maxspeed control utilizing a D.C. motor for which a table is not available. Page ED 28 of the AI.S.E. Manual states, "These applications should be referred to the selected manufacturer." In this case, a duplicate of the original two horsepower motor was selected so that it would be compatible with the existing control system.

The trolley wheels are 24" in diameter and twelve pounds per ton rolling resistance could be used according to Table E.4.c.2.1 of the A.I.S.E. Standard.

The lower rolling resistance factor would simply reduce the accelerating power to 1.2 pounds per ton. This would result in a theoretical acceleration of 0.01932 feet per second square by the connected horsepower which is much lower than that normally used.

Further, the 12 pound per ton figure in this case would indicate a full load running horsepower requirement of 1.26 and a maximum allowable connected horsepower of 1.386. The nearest available motor size would have been 1½ horsepower which is 119% of the full load running horsepower requirement (more than allowed by Regulatory Guide 1.104) providing a theoretical acceleration rate of 0.03567 feet per second square. This accelerating rate is still much lower than that normally used.

Limiting the connected horsepower of traverse drives to 110% of the full load running torque is not practical. The increments of available motor horsepowers would not in most cases match the requirements. The slow acceleration rates would be inconvenient for the operator and could also cause problems due to motor overheating in most duty cycles.

<u>QUESTION 7:</u> It is stated in your report that the hoist will be provided with three holding brakes, each sized "to hold 125 percent of rated full load hoist motor torque at base speed" that will automatically set whenever electrical power is removed. Considering the changed reeving system and rope size, for each of the spent fuel shipping casks that will be handled, demonstrate that the crane hoist will not subject the various cask trunnions and handling yokes considered in your evaluation to excessive deceleration loads under the following assumptions: (1) the cask is near its upper limit of travel; (2) the cask is being lowered at its maximum speed as defined by the hoist controls; and (3) the hoist experiences a loss of power. Accordingly, in tabular form for each cask, provide the following information:

(1) The static factors of safety of the cask handling yoke, the cask trunnions and the weight of cask;

(2) The maximum lowering speed as defined by the hoist controls; and

(3) The results of dynamic analyses which demonstrate that the cask trunnions and handling yoke have sufficient design margin to preclude their failure due to the deceleration loads created by the hoist brakes.

RESPONSE 7:

Analysis Assumptions

1. The casks are at their upper limit of travel. The length of rope available for stretch during the impulse loading is 58" for NFS-4 cask and 52" for IF-300 cask.

2. The hoist experiences a loss of electrical power while it is lowering the casks at its maximum speed of 5 fpm.

3. Weights of the casks are 52,000 lbs for the NFS-4 cask and 140,000 lbs for the IF-300 cask.

4. Each of the 24 rope parts for the reeving is equally stressed.

5. Only the deformation of ropes is considered to absorb the kinetic energy from the suddenly stopped casks; the strain energy absorbed by the bridge girder and trolley components is neglected in the analysis. Therefore, the analytical results are conservative.

Analysis Method

Using an Energy Balance approach, the kinetic energy of the cask during lowering will be coverted into strain energy of the ropes when braking occurs, thus

$$U_k = U_s \qquad (1)$$

where

$U_k = \frac{WV^2}{2_g}$ the cask kinetic energy

$U_s = \frac{WX^2}{2}$ The strain energy stored in the ropes

Terms are defined by

W = Cask Weight

U = Energy

V = Cask Velocity

K = Spring Constant of the Ropes

X = Incremental Rope Stretch

g = Gravitational Constant

Solving energy equation (1) for incremental stretch of the ropes gives

$$X = V\left(\frac{W}{gk}\right)^{\frac{1}{2}} \qquad (2)$$

The incremental rope force due to dynamic effect is then obtained by considering the force displacement relationship of the ropes.

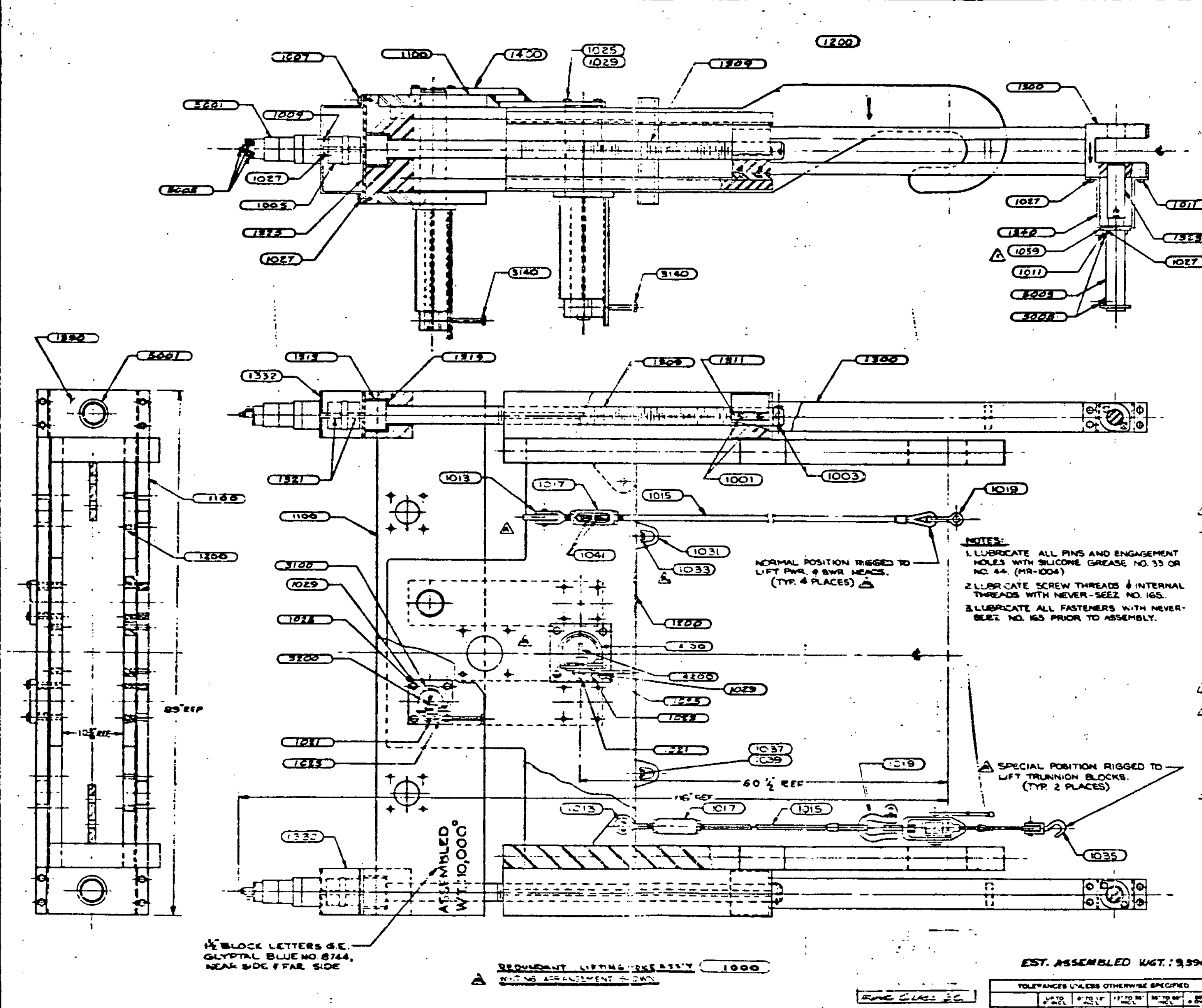
Results

The analysis results are summarized in the following table:

ITEMS	NFS-4	IF-300
Total Force due to Dynamic Effect	136,338 lbs	286,088 lbs
Static Factor of Safety of Yoke	3.0	3.0
Static Factor of Safety of Trunnion	5.7	3.0
Dynamic Factor of Safety of Yoke	1.14	1.47
Dynamic Factor of Safety of Trunnion	2.17	1.47

QUESTION 8: Indicate which of the two IF-300 shipping cask handling yokes will be utilized in the Monticello Nuclear Generating Plant. Discuss and compare the relative merits and disadvantages of the two handling yokes as their requirements relate to the limitations at your facility.

RESPONSE 8: Northern States Power Company will use the "redundant IF-300 cask yoke". The non-redundant yoke will not be used. Drawings of the redundant yoke are shown in Figures 1 and 2.



PART NO	REQ'D	DESCRIPTION	SOURCE	MATERIAL	QTY
1041	8	STD. 1/8" DIA. x 1 1/8 LG. SPIROL PIN	MR-1304	18-8 S ST	8
1039	4	1/2" STD. SPRING LOCKWASHER	MR-1303	18-8 S ST	4
1037	4	1/2-13 UNC-2A x 1 1/4" LG. SOC. HD. CAP SCREW	MR-1302	18-8 S ST	4
1035	2	PULL-A-WAY ~ 'WRIGHT' TYPE 'R'	MR-1307		2
1033	4	BAR	29316-44	18-8 S.ST	4
1029	28	3/4" DIA. LOCK WASHER, (HELICAL SPRING)	G-1005	STEEL	28
1027	26	1/2" DIA LOCK WASHER, (HELICAL SPRING)	MR-1005	18-8 S.ST.	26
1400	2	TIE BAR ASS'Y.	[illegible]		2
1031	4	BUMPER BAR	[illegible]	[illegible]	4
[illegible]	2	[illegible] CYLINDER PLATE	[illegible]		2
1023	28	5/8"-10 UNC x 2 1/2" LG. HEX HEAD BOLT	MR-503	[illegible]	20
1025	3	COLLAR - BOSTON - 33C15	MR-502	18-8 S.ST	3
1021	3	SPROCKET - 11 T ASA 40 - TYPE B - 3/4" Ø BORE	MR-501	18-8 S ST	3
3140	3	SHAFT SUB-ASS'Y	22525-8		3
3200	2	PIN SUB-ASS'Y	22525-10		2
3100	2	PIN SUPPORT SUB-ASS'Y	22525-10		2
4200	1	PIN SUB-ASS'Y	22525-11		1
4100	1	PIN SUPPORT SUB-ASS'Y	22525-11		1
1019	4	3/4" ANCHOR SHACKLE - CROSBY-LAUGHLIN # G-213	MR-408	GALV STEEL	4
1017	4	TURNBUCKLE - 7/8" x 6" CROSBY-LAUGHLIN # G 227	MR-407	GALV STEEL	4
1015	4	CABLE	[illegible]-45	GALV STEEL	4
1013	4	TURNBUCKLE JAW	[illegible]-44	GALV STEEL	4
1332	1	MOTOR SUPPORT ASS'Y (BLUE)	22525-9	~	
1325	2	COVER PLATE	22525-9		2
5003	2	AIR CYLINDER - BIMBA D-13286-A	NOTE 2		2
1323	2	LOCK PIN	22525-8		2
1340	2	AIR CYLINDER SUPPORT ASS'Y	22525-9		2
1011	12	1/2"-13 UNC x 1 1/4" LG. HEX HEAD CAP SCREW	MR-404	SA193 B8	12
1009	6	1/2"-13 UNC x 3/4" LG. HEX HEAD CAP SCREW	MR-405	SA193 B8	6
1007	8	1/2"-13 UNC x 1 1/2" LG. HEX HEAD CAP SCREW	MR-402	SA193 B8	8
5001	2	AIR MOTOR - GARDNER-DENVER # TOB4-B - NOTE 1	MR-803		2
1330	1	MOTOR SUPPORT ASS'Y (WHITE)	22525-9		1
1005	2	COUPLING ~ LOVEJOY TYPE L-190 x .875" BORE	MR-401		2
1321	4	KEY ~ 3/16" SQ.	22525-9		4
1319	2	WASHER	22525-9		2
1313	2	BUSHING	22525-9		2
1003	2	COTTER PIN ~ 1/8" x 3 1/2" LG	MR-206	18-8 SS.	2
1001	8	1/2"-13 UNC x 1 1/4" LG. FLAT HEAD SCREW	MR-104	18-8 S.S.	8
1811	4	KEY ~ 1" SQ.	22525-9		4
1309	2	LIFT SCREW	22525-8		2
1300	2	UPPER ARM ASS'Y	22525-8		2
1100	1	SECONDARY X-MEMBER ASS'Y	13642/44		1
1200	1	PRIMARY HOOK & GUIDE ASS'Y	22525-3		1
1000	1	REDUNDANT LIFTING YOKE ASS'Y	~	~	1

NE65481

Stearns-Roger

SCALE 3/4" = 1'-0" · PART NO 1000 · NEXT ASSEMBLY 1-0000 · 56646

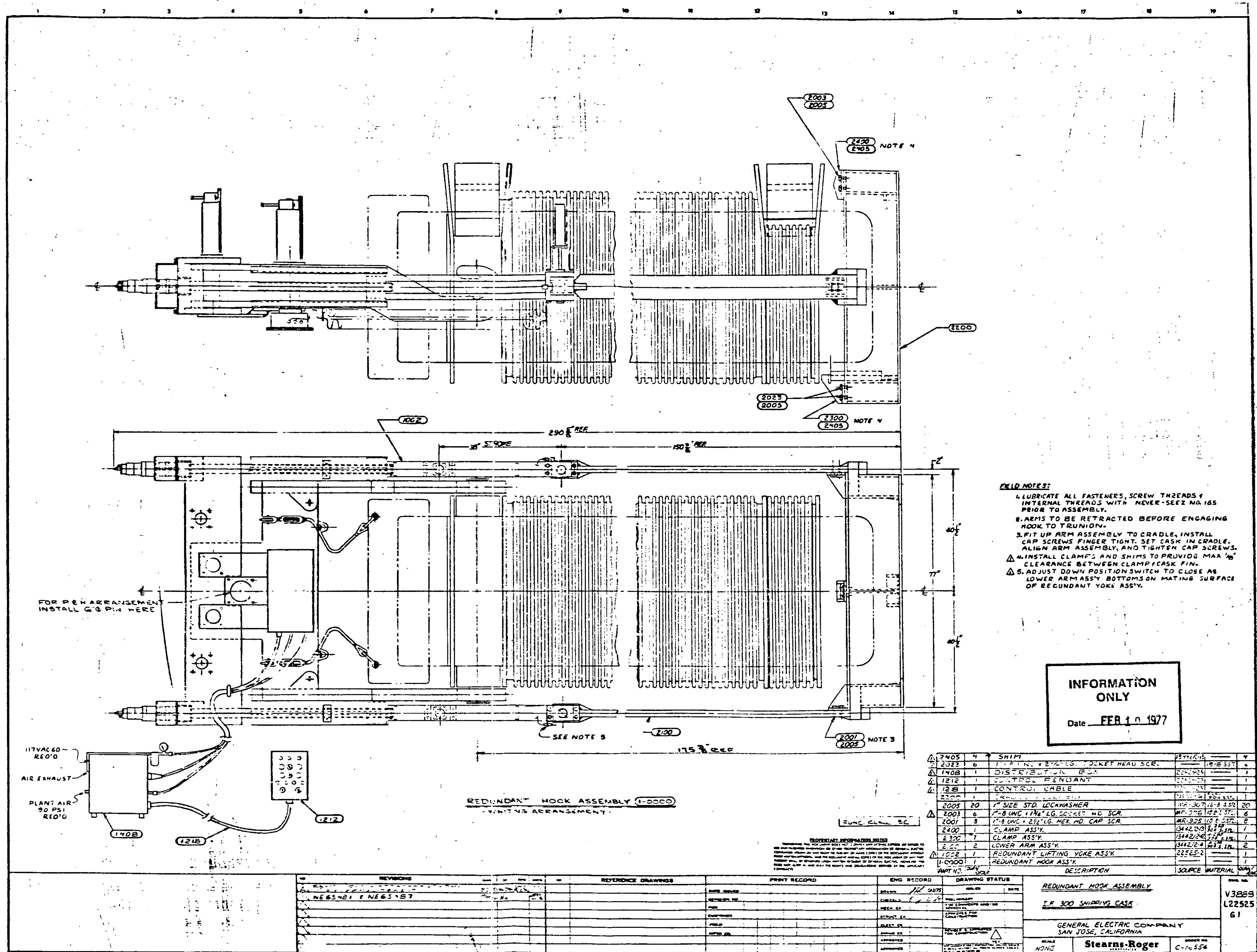
PROJECT: REDUNDANT YOKE - I.F. 300

CUSTOMER: GENERAL ELECTRIC

C-16554 V3888-22525-1

INFORMATION ONLY
FEB 10 1977
Date ______

FIGURE 1



FIELD NOTES:
1. LUBRICATE ALL FASTENERS, SCREW THREADS & INTERNAL THREADS WITH NEVER-SEEZ NO. 165 PRIOR TO ASSEMBLY.
2. ARMS TO BE RETRACTED BEFORE ENGAGING HOOK TO TRUNION.
3. FIT UP ARM ASSEMBLY TO CRADLE, INSTALL CAP SCREWS FINGER TIGHT. SET CASK IN CRADLE. ALIGN ARM ASSEMBLY, AND TIGHTEN CAP SCREWS.
4. INSTALL CLAMPS AND SHIMS TO PROVIDE MAX 1/8" CLEARANCE BETWEEN CLAMP & CASK FIN.
5. ADJUST DOWN POSITION SWITCH TO CLOSE AS LOWER ARM ASS'Y BOTTOMS ON MATING SURFACE OF REDUNDANT YOKE ASS'Y.
INFORMATION ONLY
Date FEB 10 1977
NOTE 4
NOTE 3
SEE NOTE 5
FOR P.E.H ARRANGEMENT INSTALL G.B PIN HERE
117 VAC 60~ REQ'D
AIR EXHAUST
PLANT AIR 90 PSI REQ'D
REDUNDANT HOOK ASSEMBLY (1-0000)
2405 4 SHIM
2005 20 1" SIZE STD. LOCKWASHER
2003 6 1"-8 UNC x 1 3/4" LG. SOCKET HD SCR.
2001 8 1"-8 UNC x 2 1/2" LG. HEX. HD. CAP SCR.
2400 1 CLAMP ASS'Y.
2300 1 CLAMP ASS'Y.
1408 1 DISTRIBUTION BOX
1212 1 CONTROL PENDANT
1218 1 CONTROL CABLE
1002 1 REDUNDANT LIFTING YOKE ASS'Y. 22525-2
1-0000 1 REDUNDANT HOOK ASS'Y.
PART NO. DESCRIPTION SOURCE MATERIAL
REDUNDANT HOOK ASSEMBLY
I.F. 300 SHIPPING CASK
GENERAL ELECTRIC COMPANY
SAN JOSE, CALIFORNIA
Stearns-Roger
SCALE NONE
V3889
L22525
61

FIGURE 2

Enclosure (2) to
NSP Letter Dated February 28, 1977

Changes and Corrections to Design
Report for Redundant Reactor Building Crane

Pages 2-3 -- Items 4 and 6 should be revised to reflect the information contained in our Response 1 in Enclosure (1).

Pages 2-8 -- Section 2.3.2 gives the new trolley weight as 128,000 lbs. This figure should be revised to 99,000 lbs based on the final design information.

Pages 2-10 -- Change new trolley weight shown in Table 2-1 from 128,000 lbs to 99,000 lbs.

Pages 2-11 -- Change the interior fleet angle shown in Table 2-1 from 1.5° to 2°20'. The maximum fleet angle had to be increased to obtain the necessary maximum hook height.

AISE Standard No. 6, Page MD-16, Paragraph M.4.E states, "The maximum allowable fleet angle shall be 2.5° or approximately 1/2 inch per foot in frequently worked positions." The AISE specification is recognized as the most conservative of standards.

The rope inspection, replacement, and maintenance criteria of ANSI B30.2.0-1967 are used at Monticello; and, therefore, any additional rope wear due to the increased fleet angle would be detected well in advance of any rope failure.

Pages 2-12 -- Delete the sentence concerning dynamic braking under the "Motor" section of 2.3.2.3. This was included in the report because it was thought this feature was incorporated in the existing crane control system. Further investigations of the control system indicated that it was not present.

Paragraph C.3.M of Regulatory Guide 1.104 requires one power control braking system and two mechanical holding brakes. The regenerative braking system provides the power control braking system and there are three mechanical holding brakes on the new trolley.

Pages 2-13 -- The discussion on holding brakes at the top of this page indicates that all three holding brakes will set simultaneously. One of the brakes will set immediately and the other two will be sequenced by the addition of a diode and resister in the solenoid circuit which retards the decay of the solenoid magnetic field. The time interval in between each brake application will be 0.5 seconds.

Pages 2-16 and 2-17 -- Additions and deletions to Table 2-2 have been made as shown in the attached revised Table 2-2.

Pages 3-6 -- Paragraph 3.2 states "load combinations for normal operation are dead load (including the new trolley) plus rated load (85 tons) plus 15% vertical impact and 5% coincident lateral load."

Impact loading is limited to girder calculations. This was the intent of the above as evidenced by the statement "including the trolley." The wording, however, indicates that impact loading was included in the factors of safety listed in Table 3-1.

The C.M.A.A.-70 Specification allows a maximum working stress in girders of 17,600 pounds per square inch for A.S.T.M. A-36 steel in order to reduce the dead weight of cranes. This stress level is slightly less than one-half of the yield strength of the material. The same specification limits all other components to a maximum normal working stress level of less than one-fifth of the ultimate strength of the material which is 12,000 pounds per square inch in the case of A-36 steel, and impact is not added to these components because of the more conservative stress levels.

The factors of safety listed in Table 3-1 do not include impact. Impact is, however, included in the structural analysis for the girder shown in Response 1 of Enclosure (1).

Table 2-2
Component Inspection, Testing, and Certification Requirements

ACTIVITY	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
1. Certification of material chemical properties.	X	X	X		X	X		X		X	X		X	X		X	X		X	X	
2. Certification of material physical properties.	X	X	X		X	X		X		X	X		X	X		X	X		X	X	
3. Certification of material nil ductility transition temperature.	X	X	X		X																
4. Ultrasonic test of material.	X	X	X		X						X										
5. 200% overload test.	X	X	X																		
6. Magnetic particle inspection and dimensional check after overload test.	X	X																			
7. Visual weld inspection.					X												X				
8. Weld size gauge inspection.					X												X				
9. Critical weld magnetic particle inspection.					X					X											
10. Shop functional test.				X	X	X	X	X			X	X	X	X	X	X	X	X	X	X	
11. Field functional test.				X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
12. Field full load test.	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
13. Field overload test.	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
14. Certification of compliance or certified data sheet.				X			X		X			X			X			X			X
15. Inspection for conformance to detail drawings.	X	X	X	X	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X	X
16. Sample pulled to failure.									X												
17. MAGNETIC PARTICLE INSPECTION											X										

Table 2-2.
Component Inspection, Testing, and
Certification Requirements

ACTIVITY	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41
1. Certification of material chemical properties.		X					X	X	X				X		X		X			X
2. Certification of material physical properties.		X					X	X	X				X		X		X			X
3. Certification of material nil ductility transition temperature.									X											
4. Ultrasonic test of material.									X											
5. 200% overload test.																				
6. Magnetic particle inspection and dimensional check after overload test.																				
7. Visual weld inspection.																				
8. Weld size gauge inspection.																				
9. Critical weld magnetic particle inspection.								X	X											
10. Shop functional test.	X	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X		X	X
11. Field functional test.	X	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X		X	X
12. Field full load test.	X	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X		X	X
13. Field overload test.	X	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X		X	X
14. Certification of compliance or certified data sheet.	X		X	X	X					X	X	X		X	X	X			X	
15. Inspection for conformance to detail drawings.	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X		X	X
16. Sample pulled to failure.																				

Table 2-2
Component Inspection, Testing, and Certification Requirements

ACTIVITY	AUX. HOOK	AUX. HOOK TRUNNION	AUX. DRUM SHAFT	AUX. HOIST SHEAVE FRAMES		AUX. DRUM HEADS & HUBS	AUX. HOIST DRUM		M.H. BRAKE WHEEL												
1. Certification of material chemical properties.	X	X	X	X		X	X														
2. Certification of material physical properties.	X																				
3. Certification of material nil ductility transition temperature.																					
4. Ultrasonic test of material.	X	X	X																		
5. 200% overload test.	X																				
6. Magnetic particle inspection and dimensional check after overload test (or dye penetrant for stainless stl.)	X																				
7. Visual weld inspection.																					
8. Weld size gauge inspection.																					
9. Critical weld magnetic particle inspection (or dye penetrant for stainless stl.)				X		X	X														
10. Shop functional test.																					
11. Field functional test.																					
12. Field full load test.																					
13. Field overload test.																					
14. Certification of compliance or certified data sheet.									X												
15. Inspection for conformance to detail drawings.																					
16. Magnetic particle exam. (or dye penetrant for stainless stl.)		X	X																		