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Design Analysis Cover Sheet

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Page: 1 of: 94

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7. Originator	DAVID E. HAMANN RICHARD NOVOTNY CHRIS WEDDLE	[signature] for	9-16-97
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11. Remarks

1. A. Richard Novotny prepared the structural analysis of the two following major items of mobile equipment: WP Transporter (Attachment I) and Emplacement Gantry (Attachment II)
   B. Chris Weddle prepared the mechanical analysis of the Emplacement Gantry (Attachment V), Gantry Carrier (Attachment VI), and Transport Locomotives (Attachment VII)
   C. Dave Hamann prepared the balance of this design analysis.

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2. DESIGN ANALYSIS TITLE
PRELIMINARY WASTE PACKAGE TRANSPORT AND EMPLACEMENT EQUIPMENT DESIGN

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1. PURPOSE

The purpose of this design analysis is the preliminary design of various mobile and stationary equipment necessary for the waste package (WP) Transport and Emplacement in the proposed subsurface nuclear waste repository at Yucca Mountain. The same WP handling concept and equipment may also be used for WP retrieval under normal conditions. The basic WP handling concept and the equipment requirements are based on the findings of a previous design analysis prepared during FY96 (Reference 5.5).

The objective of this analysis is to generate design input for the development of:

1. Preliminary drawings for WP transport and emplacement equipment and component drawings.

2. Preliminary descriptions and data sheets for the WP transport and emplacement mobile and stationary equipment.

2. QUALITY ASSURANCE

A classification of permanent items has not been performed in accordance with QAP-2-3, *Classification of Permanent Items,* for the WP emplacement equipment considered in this analysis. However, the WP emplacement is on the project *Q-List* (Reference 5.3) by direct inclusion; Therefore, the items evaluated in this analysis will be treated as Q items. The activity for performing this work has been evaluated in accordance with QAP-2-0, *Control of Activities*, and has been found to be subject to the requirements of the *Quality Assurance Requirements and Description* (QARD), (Reference 5.2).

Much of the input data used in the formation of this document is preliminary and "unqualified" and therefore the output of this analysis is also unqualified and shall be considered TBV. The output from this analysis will not be used for procurement, fabrication, or construction. Therefore, the TBVs do not need to be tracked in accordance with NLP-3-15.

3. METHOD

This Design Analysis is a refinement of previous work (Ref. 5.5) which was performed to select a system design from various alternatives for handling WPs. This analysis uses that basic information to develop more precise technical data for the design and/or selection of the various components by considering mechanical, structural, electrical, remote handling, instrumentation, and control and operational implications. The methods used in this analysis to accomplish that objective include:

1. Gathering of relevant MGDS design input requirements and CDA assumptions for the WP transport and emplacement equipment.

2. Preparation of layouts and sketches to further refine the equipment arrangements. Due to the preliminary nature of this analysis, the effects of tolerances shall not be considered.

3. Preparation of both hand and computer calculations for selection of individual mechanical components and sizing of the structural systems of selected equipment. Hand calculations were performed for selecting mechanical components and some weight determinations for the emplacement equipment. Computer calculations were used only for the structural analysis of the WP Transporter and the Emplacement Gantry.

4. Selection of standard vendor equipment for incorporating into the mechanical arrangement drawing of the mechanical equipment where appropriate.

5. The WP Emplacement Gantry structure was designed using static load analysis. STAAD-III computer software was used to perform the stress analysis for the Gantry structural frame. STAAD-III facilities for steel design and code checking are based on the AISC-ASD code (4.4.1).

6. The WP Transporter structure was designed using static load analysis. STAAD-III computer software was used to perform the stress analysis for the Transporter structural frame. STAAD-III facilities for steel design are based on the AISC-ASD code (4.4.1).

7. The Reusable Rail Car and the Gantry Carrier were designed only on the basis of restraints of the mechanical arrangement. No structural stress analysis or design was performed on either of these WP emplacement equipment.

8. To accomplish the objective as stated in Section 1., this analysis will:

    1. Gather design input based on MGDS design requirements and CDA assumptions for WP transport and emplacement equipment.

    2. Develop the mechanical aspects of the mobile and stationary equipment. This activity will include development of mechanical arrangements and mechanical system details where required to identify size and describe those systems, and components such as motor drives, actuators, Rail Car couplers, and wheels.

    3. Perform a structural analysis for major equipment where important considerations in the areas of overall size envelopes or total operating weight will have an effect on the outcome and conclusions of this analysis.

    4. Include shielding considerations for selected transport and emplacement equipment.

This analysis identifies the assumptions to be used and those that have been validated. Conclusions presented in Section 8 provide recommendations for a viable emplacement equipment concept, and include dimensional relationships of WP transport and emplacement equipment.

4. DESIGN INPUTS

4.1 DESIGN PARAMETERS

4.1.1. Grade of the North Ramp is -2.1486% (Ref. 5.44, pg. 26).

4.1.2 Grades in the Mains are:

East Main	-1.35% (maximum), (Ref. 5.44, pg. 26)
West Main	-1.394% (maximum), (Ref. 5.44, pg. 26)
North Main	-2.1486% (maximum), (Ref. 5.44, pg. 26)

4.1.3 Radius of curves in North Ramp is 305 m (Ref. 5.44, pg. 16).

4.1.4 Excavated diameter of North Ramp is 7.62 m (Ref. 5.44).

4.1.5 The major equipment to be addressed in this design analysis includes the following (Ref. 5.5):

Mobile Equipment:

- WP Transporter
- Reusable Rail Car
- Emplacement Gantry
- Gantry Carrier
- Transport Locomotives

Supporting Stationary Equipment:

- Rail system for North Ramp, Main Drift, Emplacement Drift Turnout and Emplacement Drift
- Emplacement Drift Transfer Dock
- Emplacement Drift Isolation Doors

Supporting systems that directly interface with the mobile equipment are covered in the following two design analyses in progress:

- Repository Rail Electrification Analysis
- Emplacement System Control and Communications Analysis

4.1.6 Dimension of the WP with the smallest diameter to be handled (Ref. 5.12)

WP Type	Diameter (mm)	Length (mm)	WP Loaded Mass (kg)
12 PWR UCF	1298	5335	32,236

Notes:

1. Dimensions for additional WPs to be considered are shown under Assumptions (Ref. 4.3.11) (TBV).
2. The Outer Diameter has been changed to 1.250 m. (Ref. 5.8, pg. 4-21, EBDRD 3.7.1.J.1).

4.1.7 The 1.28 m dimension - measuring from top of rail in the drift turnout to top of rail for the reusable rail car- is from a previous document (Ref. 5.6), and is a critical dimension used in the Transporter arrangement (Used in Figure 7.2.2) (TBV).

4.2 CRITERIA

The applicable requirements document is the *Repository Design Requirements Document* (RDRD) (Reference 5.7):

4.2.1 Equipment used for waste-handling operations shall be designed so that waste-handling operations can be performed in reverse order to permit retrieval of emplaced WPs (Ref. 5.7, RDRD 3.7.4.1A.4).

4.2.2 Cranes and similar handling equipment shall be capable of meeting the requirements specified by DOE Order 6430.1A, Section 1460 and the CMAA 70 Standard (Ref. 5.7, RDRD 3.7.4.1 A.5).

4.2.3 For the purpose of this design analysis a high surface wind with a speed of 75 mph was used. Rationale: RDRD paragraph 3.3.1 and 3.3.4.B requires the use of applicable codes in DOE Order 6430.1A under General Design Criteria (-0109), Reference Standards and Guides, which cites ANSI 58.1, now ASCE 7-88. The requirements of ASCE 7-88 indicate that the project area is in a special wind region, therefore ESFDR 3.2.1.2.1.1C was used -because it is site specific- until specific repository environments are identified. (Used in Attachment I).

4.3 ASSUMPTIONS

The assumptions used in this analysis are listed below. The rationale for the assumptions is cited in the *Controlled Design Assumptions* (CDA) (Reference 5.1). All controlled CDA assumptions require confirmation as the design proceeds. Other design assumptions either from previous work or generated specifically for this analysis shall be verified and any deviations shall be addressed in Section 8.

4.3.1 Key 010 (Ref. 5.1, pg. 3-21): Rail transport will be used for subsurface transport of WPs. (used throughout the analysis)

4.3.2 a. Key 011 (Ref. 5.1, pg. 3-22): WPs will be emplaced in-drift in a horizontal mode. (Used in Attachment V)

4.3.2 b. Key 066 (Ref. 5.1, pg. 3-55): WPs will be placed center-in-drift, on pedestals, using Gantry emplacement. (Used in Attachment V)

4.3.3 Key 013 (Ref. 5.1, pg. 3-23): No human entry is planned in emplacement drifts while WPs are present. The waste emplacement/retrieval equipment may use robotics and/or remote control features to perform operations and monitoring within the emplacement drifts. (used throughout the analysis)

4.3.4 Key 031 (Ref. 5.1, pg. 3-33): (Used throughout the analysis)

A. WP containment barriers will provide sufficient shielding for protection of WP materials from radiation-enhanced corrosion.
B. Individual WPs will not provide any additional shielding for personnel protection.
C. Additional shielding for personnel protection will be provided on the subsurface Transporter and in surface and subsurface facilities.

4.3.5 The WP, after being sealed, shall be capable of withstanding a 2.4 m drop without breaching. Note, the drop height in this assumption is greater than the drop height of 2-m onto an unyielding surface without breaching, as identified in the CDA (Ref. 5.1, EDBRD 3.7.1.1.F). Two possible approaches to consider a potential drop height of 2.4 m are: Either WPs need to be re-evaluated for the greater drop height, or the use of energy absorbing materials shall be used to limit the net effect of WP drop in excess of 2-m (Used in Section 8.1.2). (TBV)

4.3.6 Not Used.

4.3.7 Key 070 (Ref. 5.1, pg. 3-59): The following diameters are assumed for underground openings. (Used in Attachment V)

Underground Opening	**Diameter (m)**
Ramps	7.62
Access/Service Main	7.62
Emplacement Drift	TBD (for additional details, see assumption 4.3.21)

Note: For mobile equipment design, the inner operating envelops in the ramps and drifts, which are smaller that 7.62 m, have to be considered. The thickness of ground support in the ramps, access drifts/service mains has been assumed to be 300 mm (Ref. 5.51).

4.3.8 DCSS 009, (Ref. 5.1, pg. 7-6):

Maximum grade in emplacement drifts: minimize within 0.25 to 0.75 percent for drainage.

Note: Round 0.75% grade up to 1.0% maximum for Gantry drive equipment sizing purposes (Used in Attachment V).

4.3.9 Emplacement Drift Temperatures

a. DCSS 019, (Ref. 5.1, pg. 7-13):

Maximum allowable air temperature in emplacement drifts during emplacement: 50°C dry-bulb, only in portion requiring access

b. DCSS 023 (Ref. 5.1, pg. 7-16)

Maximum allowable preclosure rock surface temperature in:

Shafts:	35°C	- unventilated
Ramps:	35°C	- unventilated
Mains:	50°C	
Emplacement Drifts:	200°C	

Temporary increases in these temperatures are allowed during initial cooling of emplacement drifts for maintenance, performance confirmation, retrieval, and backfilling (Used in Attachment II, Section 2.0).

4.3.10 Not Used

4.3.11 Dimensions and the weight of the largest WPs to be handled shall not exceed:

A.	Outer Diameter:	1970 mm (Ref. 5.1, pg. 8-6, DCWP 005)
B.	Outer Length:	5850 mm (Ref. 5.1, pg. 4-21, EBDRD 3.7.1.J.1)
C.	Loaded Mass:	69,000 kg (Ref. 5.1, pg. 4-22, EBDRD 3.7.1.J.2)

Notes:

1. A rounded up value of 2.0 meters has been used as the maximum Outer Diameter of the WP to determine clearances in this analysis (Used throughout this analysis).
2. The Outer Length has been changed to 5.900 m, (Ref. 5.8, pg. 4-21, EBDRD 3.7.1.J.1).
3. The Loaded Mass has been changed to 70,000 kg, (Ref. 5.8, pg. 4-22, EBDRD 3.7.1.J.2).

4.3.12 Weight of WP is assumed to be uniformly distributed through-out the volume of the WP for this analysis. It is understood that radioactive waste material will be placed in symmetrically oriented basket assemblies and therefore the weight and center of gravity would also be uniformly distributed. (Used in Attachments I, II, V, and IV)

4.3.13 Rail turnouts that connect each Emplacement Drift with the Main Drifts shall have a radius of 20.0 m when measured from the centerline of a drift (Ref. 5.51, 4.3.23). (Used in Attachments III and VII)

4.3.14 The bottom of the Drift Turnout excavation will be 0.80 m below the bottom of the Emplacement Drift excavation, based on the current thinking that the elevation difference of 0.8 allows for the WP emplacement operation (Ref. 5.44) (Used in Figures 7.9.2 thru 7.9.5 and Section 8.4.2).

4.3.15 WP placement order in the Emplacement Drift shall be from the center of the Emplacement Drift to the drift entrance (Ref. 5.6). (Used in Attachment V)

4.3.16 DCSS 036 (Ref. 5.1, pg. 7-27): Doors are required at entrances to Emplacement Drifts. (Used in Section 7.7)

4.3.17 There will be an end skirt at both ends of the WP that will allow for the WP to be handled during emplacement. Each skirt will have an inner length of approximately 225 mm (Ref. 5.6, Volume III, Appendix B). (Used in Attachment II and V)

4.3.18 Radiation shielding used on the WP Transporter to provide protection from gamma and neutron radiation will consist of materials and thickness as follows (Ref. 5.17): (Used in Attachments I and III)

Material	Material Density (kg/m³)	Reference for Material Type and Density	Material Thickness	Reference for Thickness
Borated Polyethylene (1.5%)	920	Ref. 5.17, pg 21	101.6 mm one layer *76.2 mm one layer	Ref. 5.17, pg 75 Ref. 5.17, pg 80
Carbon Steel	7832	Ref. 5.17, pg 20	152.4 mm one layer *177.8 mm one layer	Ref. 5.17, pg 75 Ref. 5.17, pg 80
Stainless Steel	7949.7	Ref. 5.17, pg 20	5 mm two layers	Ref. 5.17, pg 80

*The shielding dimensions and materials for the WP Transporter are assumed to be 5 mm SS316L + 152.4 mm carbon steel + 101.6 mm borated (1.5% boron) polyethylene + 5 mm SS316L in the radial direction; and 5 mm SS316L + 177.8 mm carbon steel + 76.2 mm borated polyethylene + 5 mm SS316L in the axial direction. Rationale: The radial shielding dimensions are based on the results of a detailed shielding analysis (Ref. 5.17) to limit the Transporter surface dose rate to less than 50 mrem/hr. The axial shielding dimensions are derived from the radial shielding thickness by increasing the carbon steel thickness by 25.4 mm to account for the increased gamma radiation field from the fuel assembly end fitting sources, and by decreasing the reduced neutron field on the ends.

4.3.19 The Gantry rail size will be 44.6 kg/m (90 lb/yd) ASCE rail. This assumption is a conclusion from a previous analysis (Ref. 5.6, Appendix E). (Used in Attachments II and V) (TBV)

4.3.20 Seismic Design Inputs for the Exploratory Studies Facility at Yucca Mountain: Seismic Horizontal and Vertical Acceleration = 0.27g (Ref. 5.9, Table 1) Seismic analysis will consist of a quasi-static analysis applying a horizontal and vertical acceleration to the waste package transporter and to the emplacement gantry. Final design will evaluate equipment under the dynamic loading in accordance with the site-specific response spectra. The seismic force will be the product of the horizontal and vertical accelerations times the weight of the item being analyzed. (Used in Attachments I and II) (TBV)

4.3.21 Preliminary design assumptions for the subsurface repository are based on an excavated Emplacement Drift bore diameter of 5.5 meters. Ground support systems in the Emplacement Drifts are assumed to be up to 200 mm thick. Based on these assumptions, the useful inner drift diameter will then be approximately 5.1 meters (Ref. 5.5, 4.3.14). (Used in Attachment V)

4.3.22 From a preceding analysis (Ref. 5.6, Appendix E), the track gage of 1.44 m (56½ in.) for both the Transporter and the Reusable Rail Car will be used. (Used in Attachments III and IV)

4.3.23 The Gantry rail gage will be 2.58 m and the top of rail shall be 1.0 m above the invert of the Emplacement Drift (Ref. 5.38, Attachment II, Figure 14, pg. II-42). (Used in Sections 7.4 and 7.9, Attachments II and V)

4.3.24 The Transporter and Gantry Carrier rail will be a 57 kg/m (115 lb/yd) AREA rail (Ref. 5.6, Appendix E). (Used in Attachments III and VI) (TBV)

4.3.25 The Reusable Rail Car rail inside the Transporter and in the Emplacement Drift will be 44.6 kg/m (90 lb/yd) ASCE rail from a preceding design report (Ref. 5.6, Appendix E). (Used in Attachment IV) (TBV)

4.3.26 The Reusable Rail Car will operate only on straight, without any curves, track inside the Transporter, in the entrance of the Emplacement Drift and in the Waste Handling Building (WHB) and does not require pivoting trucks (Ref. 5.6, Appendix E). (Used in Attachment IV) (TBV)

4.3.27 Maximum Transport Locomotive speed of 8 km/hr (5 mph) is from a previous design report (Ref. 5.6, pg. E-4). (Used in Attachment VII) (TBV)

4.3.28 The WP Transporter shall be moved from the surface WHB by two trolley-powered locomotives located on each end of the Transporter and operating in tandem (Ref. 5.5, pg. 33). (Used in Attachment VII) (TBV)

4.3.29 The WP Transporter shall be moved from the East Main Drift into the Emplacement Drift Turnout and up to the Emplacement Drift Transfer Dock by one trolley-powered locomotive (Ref. 5.5, pg. 34). (Used in Attachment VII) (TBV)

4.3.30 The gantry design shall allow for a minimum clearance of 1.00 m between emplaced WPs. Rationale: This dimension meets the necessary clearance required for alignment and mis-positioning of the gantry lifting mechanism. (Used in Section 8.4.5) (TBV)

4.3.31 The top of the shadow shield in the Emplacement Drift shall be 75 mm above the height of the largest diameter WP of 2.0 m. From (Ref. 5.17 Section 4.3.2.9) the diameter of the shadow shield is assumed to be 2.15 m to completely cover the projected end of the largest WP diameter. It is inferred that the centerline of the 2.15 m diameter shield is on the same centerline as the 2.0 m WP and that the outside diameter of the shield extends past the outside diameter of the WP by one-half of the difference in diameter (150 mm) or 75 mm. (Used in Sections 7.4.3.2 and 8.1 and Figure 7.9.1) (TBV)

4.3.32 Shielded Electrical Cabinets are constructed of 51 mm (2 in.) thick ASTM A36 steel plate (Ref. 5.36, pg. 77) (Used in Section 7.4.3.9) (TBV).

4.3.33 The centerline of a 2.0 m diameter WP on support pedestals shall be 1.96 m above the invert of the Emplacement Drift (Ref. 5.14). (Used in Section 7.4, Attachment V) (TBV)

4.3.34 WP carry-over capability shall be considered in the development of the emplacement gantry. WP carry-over means the ability to move one WP over another already emplaced WP in an emplacement drift. Rationale: Carry-over is a desirable gantry feature, if WP retrieval becomes necessary at a later date, but it is not a requirement. (Used in Section 7.4.3.2) (TBV).

4.3.35 Administrative operator controls will be in place to limit radiation exposure to workers from the time the WP is removed from the WP transporter until the time it is transferred behind the shadow shield. (Used in Section 7.1.1) (TBV).

4.4 CODES AND STANDARDS

4.4.1 American Institute of Steel Construction (AISC)

AISC M016-89 AISC Manual of Steel Construction, Allowable Stress Design, Ninth Edition

4.4.2 American Society of Mechanical Engineers (ASME)

ASME NOG-1-1995 Rules for Construction of Overhead and Gantry Cranes (Top Running Bridge, Multiple Girder)

4.4.3 American Welding Society (AWS)

AWS D1.1-94 Structural Welding Code - Steel, Thirteenth Edition
AWS D14.1 Specification for Welding Industrial and Mill Cranes

4.4.4 American Society for Testing and Materials (ASTM)

ASTM A36/A36M-94 Standard Specification for Carbon Structural Steel

4.4.5 Crane Manufacturers Association of America Inc (CMAA)

CMAA No. 70 — Specification for Top-Running Bridge and Gantry Type Multigirder Electric Overhead Traveling Cranes, Revised 1994

4.4.6 American Railroad Engineering Association (AREA)

Manual for Railway Engineering, 1994

4.4.7 American Society of Civil Engineers (ASCE)

ASCE 7-88 — Minimum Design Loads for Buildings and Other Structures (Revision of ANSI A58.1-1982), July 1990

4.4.8 Association of American Railroads (AAR)

Manual of Standards and Recommended Practices, Section C-Part II, Specifications for Design, Fabrication, and Construction of Freight Cars, M-1001, Volume I Standard, Issue of 1988.

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5.3 DOE (U.S. Department of Energy), *Q-List,* YMP/90-88Q, Rev. 4, Office of Civilian Radioactive Waste Management

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5.5 CRWMS/M&O, *Evaluation of WP Transport and Emplacement Equipment,* DI:BCAF00000-01717-0200-00002 Rev. 00

5.6 CRWMS/M&O, *Mined Geologic Disposal System Advanced Conceptual Design Report,* DI:B00000000-01717-5705-00027 Rev. 00, Volume II

5.7 DOE (U.S. Department of Energy), *Repository Design Requirements Document.* YMP/CM-0023 Rev. 0, ICN 1

5.8 CRWMS/M&O (Civilian Radioactive Waste Management System Management and Operating Contractor), *Controlled Design Assumptions Document*. DI:B00000000-01717-4600-00032 Rev. 04, ICN 02

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5.14 CRWMS/M&O, *Waste Package Support and Pier Static and Seismic Analysis*, DI: BBAB00000-01717-0200-00006 REV 00.

5.15 CRWMS/M&O, Interoffice Correspondence, *Gantry Lifting Fixture- Preliminary Design Concept*, Zekai Ceylan, LV.WP.ZC.12//96-288

5.16 *Software Requirements Document for Structural Analysis and Design/Integrated Structural Design System* (STAAD-III/ISDS), Version 4-8 MB, Rev. 16.0. Computer Software Document Number: SRD-02, Revision 0, Computer Software Configuration Item (CSCI) number 20.93.3002-AAU4-8MB.

5.17 CRWMS M&O, *MGDS Subsurface Radiation Shielding Analysis*, DI:BCAE00000-01717-0200-00001 REV 00.

5.18 *Whiting Crane Handbook* 1967. Third Edition, Whiting Corporation.

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PCI Design Handbook, 1985. Third Edition.

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5.23 U.S. Motors, *Constant Speed Gear Products*, Catalog No. PB 460, 7/95

5.24 REX Power, *Transmission and Conveying Components*, Bulletin R100, 1992

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5.27 Ferdinand P. Beer and E. Russell Johnson, Jr., *Mechanics for Engineers, Statics*, 1956, McGraw-Hill Book Company, Inc.

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5.39 Kasgro Rail Corp, fax from Gabe M. Kassab to Chris Weddle, 4/29/97, Rail Car Specialty Listing, 125-ton equipment.

5.40 CRWMS M&O, *Repository Rail Electrification Analysis*, DI:BCAC00000-01717-0200-00002 REV 00.

5.41 SKE, *Product Service Guide*, April 1992.

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5.49 Association of Iron and Steel Engineers (AISE), *Guide for the Design and Construction of Mill Buildings*, AISE Technical Report No. 13, August 1991.

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5.51 CRWMS M&O, *Repository Subsurface Layout Configuration Analysis*, DI:BCA000000-01717-0200-00008, REV. 00.

5.52 CRWMS M&O, *Emplacement Drift Air Control System*, DI:DCA000000-01717-0200-00005 REV. 00.

5.53 STAAD-III Analysis portion of Attachment I, Waste Package Transporter Structural Analysis, in electronic media form.

5.54 STAAD-III Analysis portion of Attachment II, Waste Package Transporter Structural Analysis, in electronic media form.

6. USE OF COMPUTER SOFTWARE

6.1 Computer hardware used for this analysis was a Compaq 5120, 120 MHZ Pentium.

6.2 STAAD-III, Release 22W, Revision 22.0W for Windows (Ref. 5.16), is the computer software used to perform structural steel stress analysis. This software has not been verified and validated in accordance with applicable CRWMS M&O procedures, but is satisfactory for use for this preliminary analysis. Results are presented in Attachments I and II. Since the software used has not been validated, all related outputs are designated TBV.

7. DESIGN ANALYSIS

7.1 WASTE PACKAGE TRANSPORT EMPLACEMENT SYSTEM

7.1.1 Introduction

The overall objective of the WP Transport Emplacement System is the safe transport of spent nuclear fuel waste materials in the subsurface portion of the repository, the placement of these materials in underground storage, and, if required, the retrieval of these materials on an as needed basis. The results from this design analysis shall be considered as TBV. The basic components of this system are:

- The rail transport system consisting of rails, Transport Locomotives, shielded WP Transporter with Reusable Rail Car, Gantry Carrier, and DC power system.
- The emplacement system consisting of Emplacement Gantry, Gantry rails, and DC power system.
- Emplacement Drift Isolation Doors and Transfer Dock.

Functions of the rail transport system are:

- To receive WPs from the WHB.
- To safely transport WPs to the underground Emplacement Drifts.
- To provide shielding for workers who must perform duties around the WP Transporter.
- To deliver WPs to the Emplacement Drift Transfer Dock.
- To transport the Emplacement Gantry from the Gantry storage to the underground Emplacement Drifts, from drift to drift, and back to storage.

Functions of the emplacement system are:

- To receive WPs from the transport system at the Emplacement Drift Transfer Dock.
- To transport WPs in the Emplacement Drifts.
- To place WPs on preset pedestals on the Emplacement Drift invert.

For retrieval of WPs under normal conditions, the emplacement operations are repeated in reverse order. Retrieval under abnormal conditions may require additional equipment not covered in this analysis.

Subsystems included in the transport system are required for the following purposes:

- Rails - Provide the access route and vehicle guidance from surface facilities to the Emplacement Drifts. The Rail System also provides the electrical ground for the direct current (DC) power system.
- Locomotives - Provide the means to move the WP Transporter or Gantry Carrier on the rail system. Locomotives will be equipped for manual or remote control. Normal operation of the Locomotives will be manual with human operators. During loading and

unloading of the WPs the Locomotive operation will be remotely controlled to prevent human exposure to radiation when the WP is outside the shielding of the WP Transporter (Ref. 4.3.35).

- WP Transporter - Provides the means to transport the WPs underground and deliver them to the emplacement system. The WP Transporter will have shielding to reduce emitted radiation, it will also be equipped with a rigid chain system for loading, unloading, and securing the Reusable Rail Car with WP during transport operation.
- Reusable Rail Car - Provides a moveable base for supporting and containing the WP during transportation to the Emplacement Drift and is an interface between the WHB and emplacement system. The Reusable Rail Car travels to and from the Emplacement Drift inside the Transporter. The Rail Car in this application is reusable compared to a previous concept of leaving the Rail Car in the Emplacement Drift as permanent support for the WP.
- Gantry Carrier - Provides a means to transport the Gantry to and from the Emplacement Drifts. The Gantry Carrier is similar to a railroad flat car equipped with rails for supporting the Gantry. An electrified third rail system is provided to supply power for the loading and unloading operation of the Gantry.
- DC Overhead Power System - Provides DC power to the Locomotive through an overhead pantograph system. This system is covered in a separate analysis (Ref. 5.40).

Systems included in the emplacement system are required for the following purposes:

- Gantry - Provides the means for lifting, transporting and placing WPs in the Emplacement Drifts (Ref. 4.3.35).
- Gantry Rails - Provide support and guidance for the Gantry in the Emplacement Drift and the electrical ground for the DC power system.
- DC Third Rail Power System - Provides DC power to the Gantry. This system is covered in a separate analysis (Ref. 5.40).
- WP Transporter Unloader System Guides in Emplacement Drift - Provides for extending the travel of the Rail Car unloader into the Emplacement Drift.
- Emplacement Drift Isolation Door - Provides for separation of the Emplacement Drift from the turnout area to restrict entry, to control ventilation, and to provide for a limited amount of radiation protection. This system is covered in a future analysis that addresses the drift ventilation.
- Emplacement Drift Transfer Dock - Provides an elevated platform or dock with rails and unloader guides that line up with the respective rails and guides in the WP Transporter or Gantry Carrier for the delivery of the Reusable Rail Car or Gantry into the Emplacement Drift.
- Shadow Shield - Provides radiation protection to permit personnel access in the main drift and turnout areas (Ref. 5.17).

As indicated in the preceding paragraphs there are many interfaces between equipment items within the WP Transport and Emplacement System. This system also interfaces with facilities that are outside the limits of the system, such as the WHB. All of these interfaces will be identified in the body of this analysis.

7.1.2 Sequence of Operations

Transport of WPs to the underground repository begins with the loading of WPs onto a Reusable Rail Car within the WHB. The Reusable Rail Car is then loaded into a WP Transporter (the transporter) and the transporter doors are closed. At least one Transport Locomotive (the locomotive) is always connected to the transporter and moves the loaded transporter out of the WHB. These operations are remotely controlled. Once the transporter has cleared the WHB, operators board the locomotive and an additional, or secondary locomotive is then coupled to the rear of the transporter. The secondary locomotive, connected in the rear of the transporter, is a requirement for movement of the transporter from the WHB to the access drifts (Section 4.3.28). During WP transport through the North Ramp and the Mains, where gradients are both positive and negative, two locomotives will be coupled to the transporter. One locomotive will be in the front (the primary locomotive) and one in the back of the transporter (the secondary locomotive), facing the shielded doors of the transporter. The dual locomotive arrangement has the advantage that in either travel direction on an incline (upward or downward) there will always be one locomotive ahead of the transporter, this arrangement may prevent a potential run-away situation for the transporter. The train travels to the repository via the North Ramp into either the East or West Main Drift, depending on the location selected for WP disposal. Which of the two locomotives (primary or secondary) will lead the train down the North Ramp depends on the selection in which emplacement drift WP unloading and emplacement will take place. The locomotive operators control the train from the locomotive that leads the train down and back up the North Ramp.

The train proceeds past the turnout area adjacent to an active Emplacement Drift. At this point, the secondary locomotive at the rear of the transporter is decoupled because clearance is required at the rear of the Transporter for WP unloading at the Emplacement Drift Transfer Dock. The operators vacate the Locomotive and control is once again performed remotely. A rail switch is thrown enabling the WP Transporter to be backed into the turnout towards the Transfer Dock and Isolation Doors located at the Emplacement Drift entrance.

As the train approaches the loading dock area the WP Transporter doors and the Emplacement Drift Isolation Doors are opened by remote control. The WP Transporter is then backed up to the Transfer Dock at the Emplacement Drift.

Once the Transporter is aligned with the Emplacement Drift Transfer Dock, the Reusable Rail Car and WP within the Transporter are unloaded using an internal loading/off-loading mechanism.

After the Reusable Rail Car containing the WP is off-loaded into the Emplacement Drift, the rail mounted Gantry moves into place over it. The Gantry grips the WP at both ends and raises it. The Gantry then transfers the WP into the Emplacement Drift for placement on preset pedestals. After placement, the empty Gantry returns to the Emplacement Drift entrance to await the arrival of the next WP. After the WP is removed, the Reusable Rail Car is retrieved back into the WP Transporter. The Locomotive and Transporter containing the empty Reusable Rail Car move away from the dock area. Once the Locomotive and Transporter clear the dock area, Transporter doors and the Emplacement Drift Isolation Doors are closed.

The Locomotive and WP Transporter containing the Reusable Rail Car move towards the Main Drift via the turnout. Once the train reaches the Main Drift it moves past the rail switch and recouples to the second Transport Locomotive. Remote control is discontinued and operators reboard the lead Locomotive. The rail switch is then opened and the train, with both Locomotives, moves through the Main Drift and up the North Ramp to the WHB.

7.2 WASTE PACKAGE TRANSPORTER

7.2.1 Attachment Reference

Attachment I - Structural Analysis
Attachment III - WP Transporter-Mechanical Equipment Selection
Attachment VIII - WP Transporter/Gantry Carrier Truck Arrangement and Weight Analysis
Attachment IX - WP Transporter-Coupler Vendor Information and Weight
Attachment X - Serapid Rigid Chain Information

7.2.2 Functional Requirements

The purpose of the Transporter is to transport the WPs on the Reusable Rail Car to the Emplacement Drift. A basic equipment outline and pertinent features of the WP Transporter are shown in Figures 7.2.1 through 7.2.4. The design contains the necessary flexibility to transport WPs of varying sizes and weights up to and including a 2.0 m diameter, 5850 mm long WP, weighing 69 metric tons (MT). In performing the transporting function the Transporter must complete the following subfunctions:

- Provide shielding for workers during transport and emplacement operations.
- Provide a safe stable platform for transporting the WP and Reusable Rail Car.
- Provide structural integrity for supporting the load, coupling to the Locomotive(s), and braking systems to aid in stopping and speed control.
- Accurately line up Transporter and Reusable Rail Car rails to Emplacement Drift rails and provide solid connection/restraints.
- Load and unload the Reusable Rail Car with WP.
- Open and close shielding doors.
- Provide for manual connections on front for power, control, and air brakes to Locomotive.

7.2.3 Transporter Subsystems

To perform the primary functions and subfunctions as identified in Section 7.2.2, the WP Transporter is composed of several component systems including the shielding, underframe, undercarriage, couplers and connectors, brake system, door operators, Reusable Rail Car restraint, Reusable Rail Car unloading system, wiring, interlocks, and instrumentation.

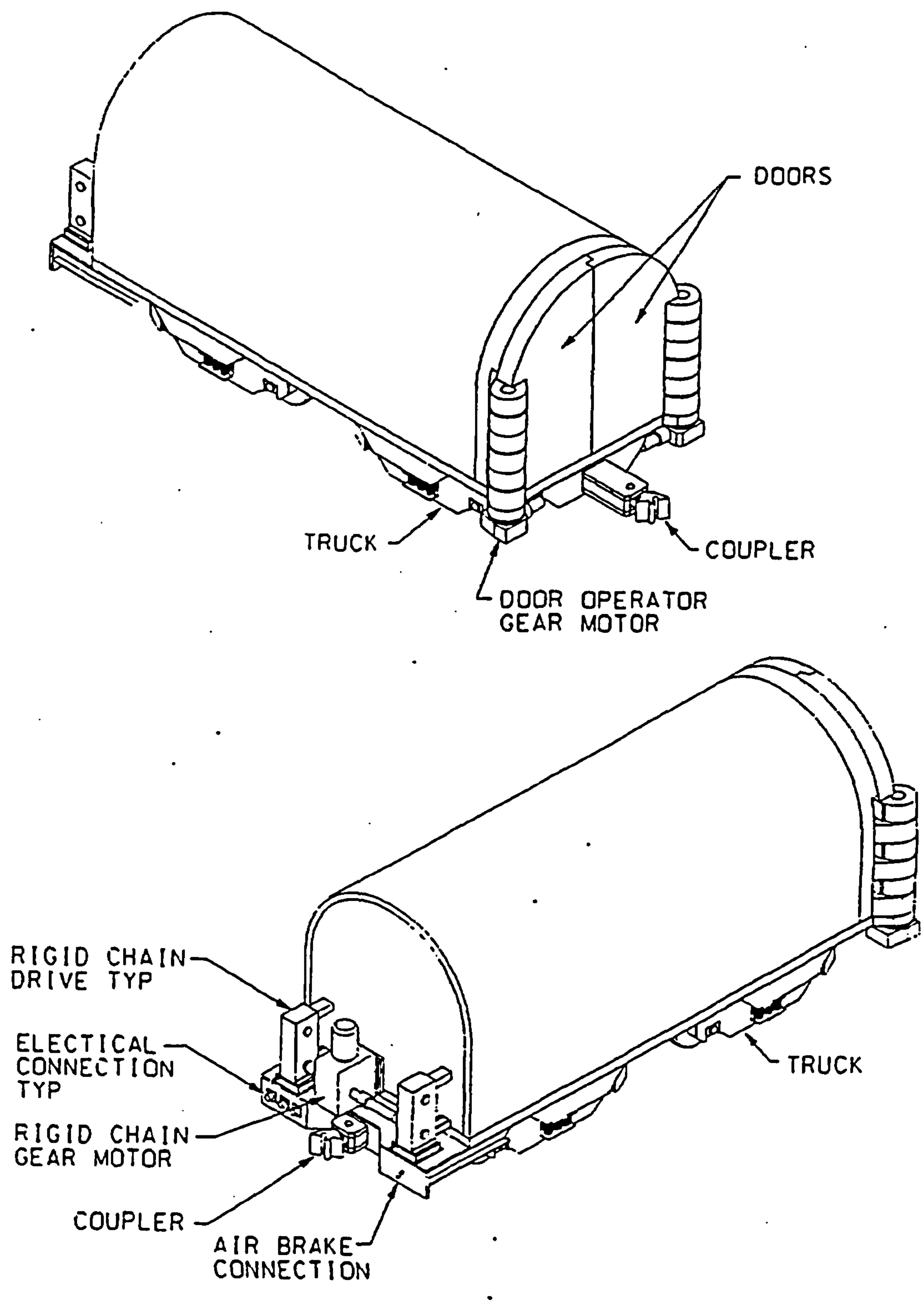

FIG. 7.2.1
WASTE PACKAGE TRANSPORTER ARRANGEMENT

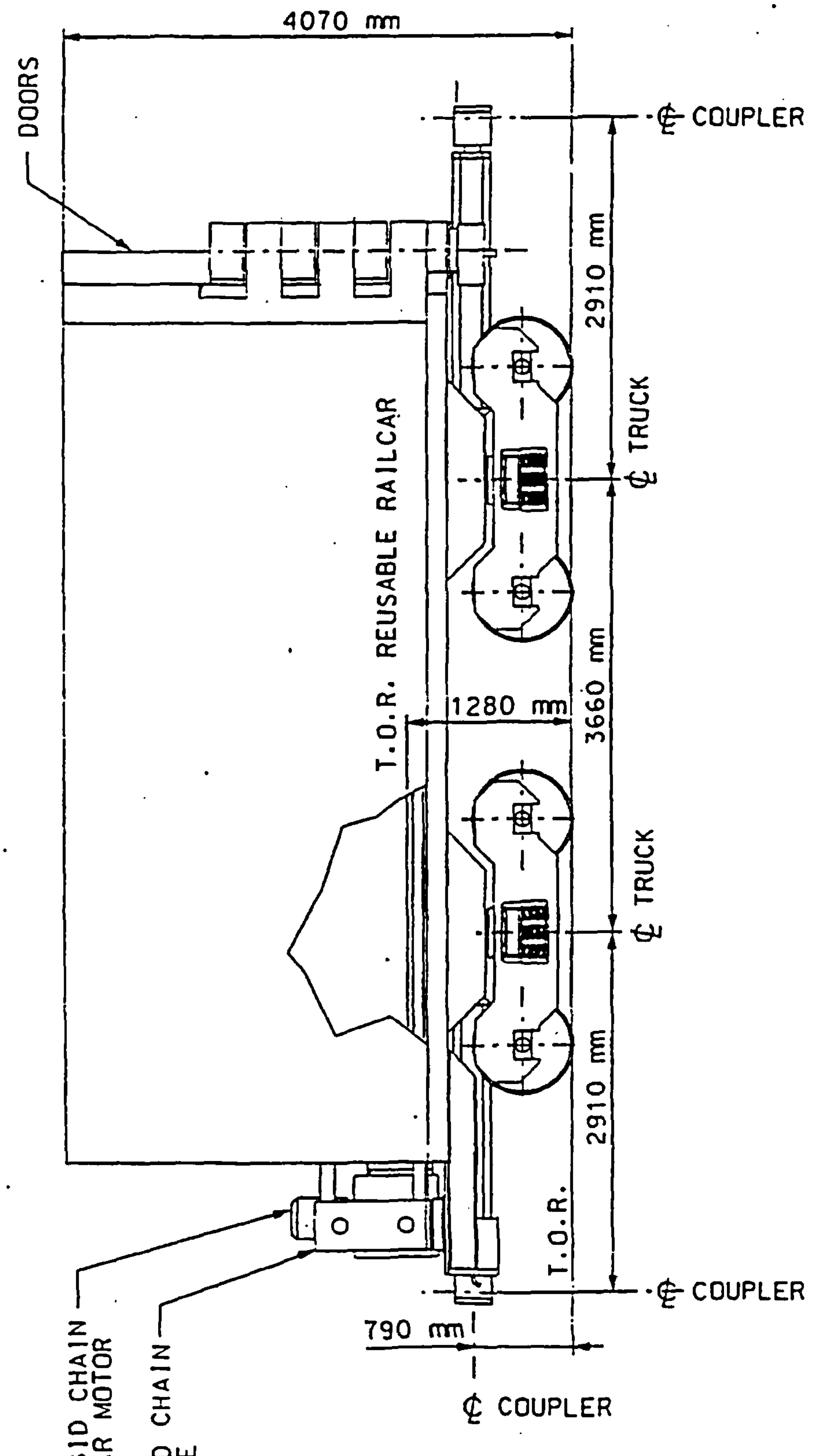

FIGURE 7.2.2
WASTE PACKAGE TRANSPORTER
SIDE ELEVATION

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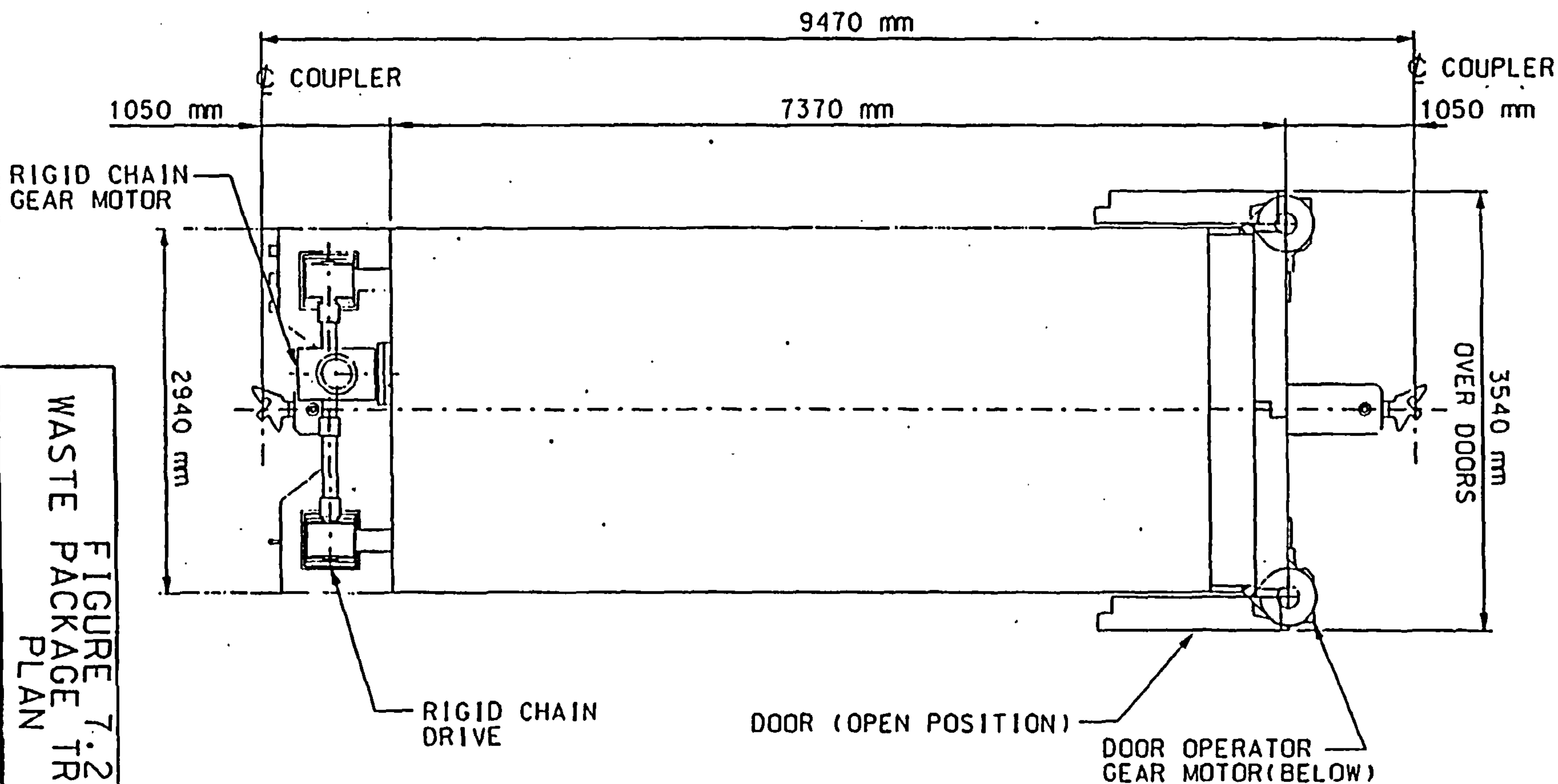

FIGURE 7.2.3
WASTE PACKAGE TRANSPORTER PLAN

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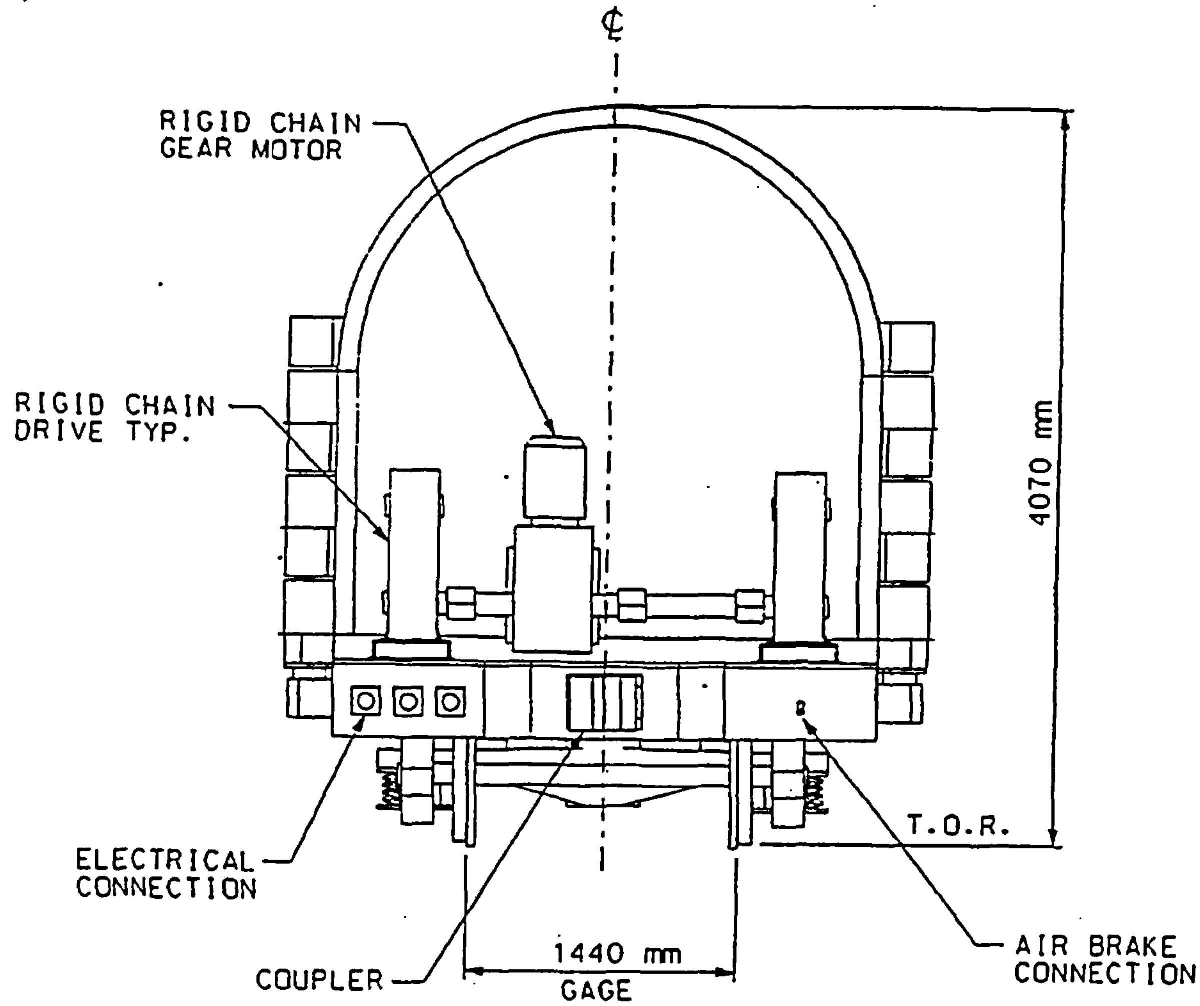


FIGURE 7.2.4
WASTE PACKAGE TRANSPORTER
END ELEVATION

7.2.3.1 Shielding

The shielding is designed to reduce the radiation from the WP inside the Transporter to an acceptable safe level of less than 50 mrem/hr at the surface of the Transporter, which is compatible with the operations within the Main Drifts (Ref. 5.17). The shielding is a composite of stainless steel, carbon steel, and a borated polyethylene material with a total thickness of 264 mm (10.4 in.) (Section 4.3.18). The individual thickness of the carbon steel and the borated polyethylene that make up the total shield thickness vary with the location of the shielding in relation to the WP. Refer to Section 4.3.17 for the respective shielding material thicknesses in the radial or axial direction from the WP and the supporting rationale. The carbon steel shield faces the inside of the Transporter providing the gamma shielding and serving as the shielding structure. The 1.5% B-poly neutron shielding material is attached to the outside surface of the carbon steel. The 1.5% B-poly is covered on the outside with stainless steel (refer to Figure 7.2.5). The Transporter has two swinging doors for Rail Car unloading constructed of the composite material. The doors swing 270 degrees out and around to the side of the Transporter. The carbon steel inner shield material can be fabricated and machined into door hinges and other features as required.

7.2.3.2 Underframe

The underframe is the structure below the floor which connects the Transporter shield and the couplers to the trucks. It also provides location and support for the auxiliary equipment, including the door operator, air brakes, WP Rail Car unloader, and miscellaneous electrical equipment, controls, wiring, and instrumentation. This analysis has developed the shield as a rigid box with doors that is not only self-supporting but also used as a structural component of the underframe. The underframe is fabricated of structural steel sections and plates with welded or bolted connections. There are two sections of the underframes, one front and one rear, which are connected and integrated with each other by the underside of the shielding floor. The front underframe includes the front coupler, front bolster plate, and an equipment platform for the WP Rail Car unloader. The rear underframe section includes the rear coupler, truck bolster plate, and the door operator drive. A structural analysis verifying major underframe members selection is presented in Attachment I.

7.2.3.3 Undercarriage (Trucks and Wheels)

The trucks provide support mobility and guides for the Transporter. They are of standard rail car configuration adapted to this specific application (see Attachment VIII for equipment details). The trucks include wheels, axles, bearings, brakes, and springs, which are incorporated into the truck frame, and the bolster, which transmits the load from the truck to the bolster plate of the underframe. The truck bolster includes a bolster pin, which centers the truck in a corresponding hole in the bolster plate and allows for the trucks to pivot and the Transporter to negotiate curves in track. The truck capacity is based on maximum operating load, which is the maximum operating weight divided over the 8 wheels of the two trucks. The maximum design operating weight is used to select a wheel and rail combination. The maximum operating weight is 233.15 MT (257.1 tons) per Attachment I and the design wheel load is 44,961 kg (99,138 lbs) per Attachment III. A corresponding wheel selection is of 762 mm (30 in.) diameter and a 57 kg/m (115 lb/yard) AREA rail. The wheels used

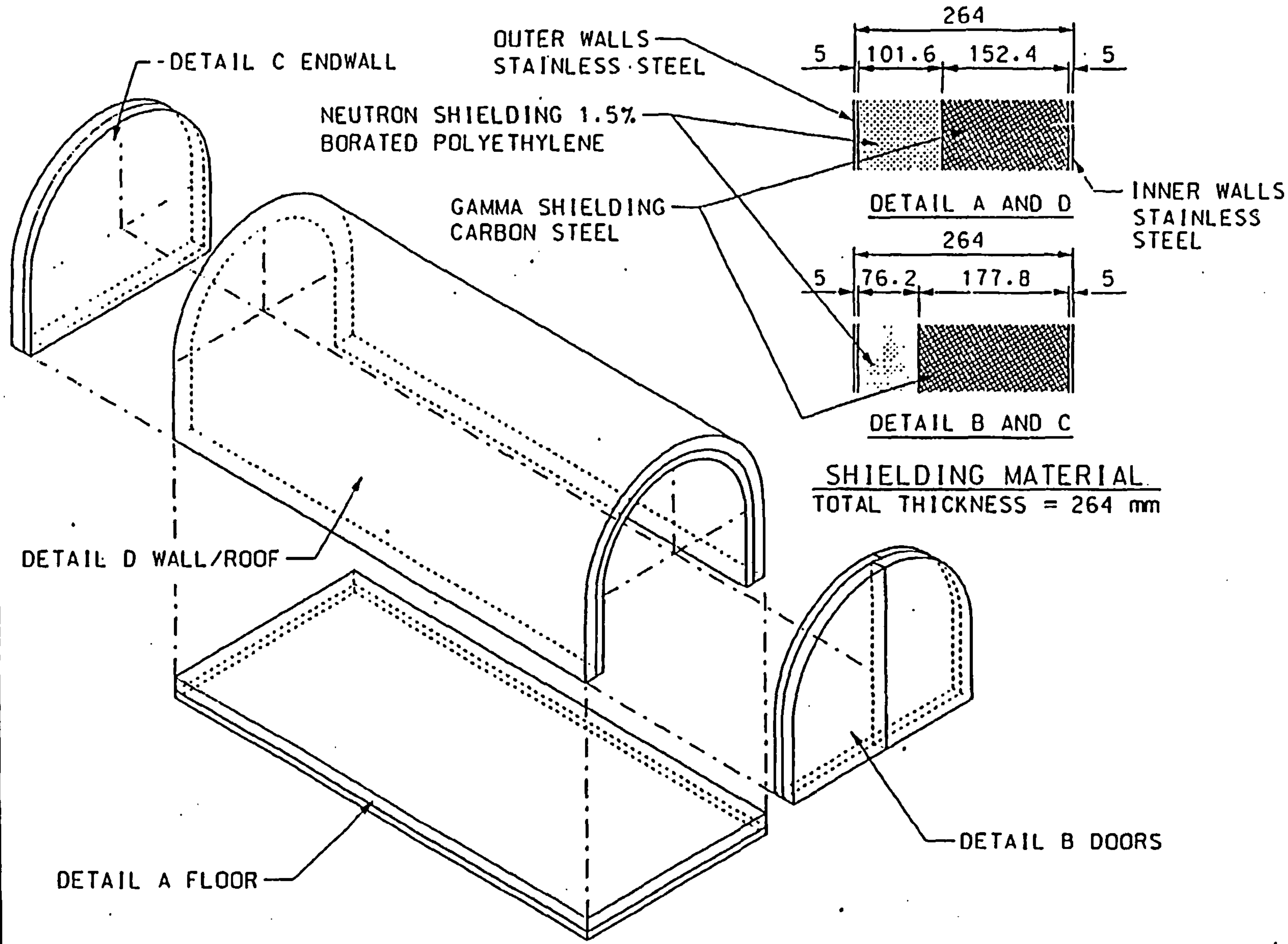


FIGURE 7.2.5
WASTE PACKAGE TRANSPORTER
SHIELDING ARRANGEMENT

CAD FILE: t:\repss\mech\figs\ssme0241.fig in the bridge crane service are heat treated to a hardness of 58 RC (615 BHN) and the rails to a 320 BHN. Refer to Attachment III for wheel selection and rail verification.

7.2.3.4 Couplers and Connectors

The Transporter is equipped with two Willison-type couplers common to underground mining and tunneling rail equipment (Attachment IX). This provides for a Locomotive at the front and the rear during the transport operation from the surface and a more positive braking in the downgrade descent to the Emplacement Drift (Section 4.3.28). The couplers will have automatic release. However, only the rear coupler will require activation during normal operation when the rear, or secondary Locomotive is disconnected for Transporter unloading. The automatic coupler on the secondary Locomotive shall have a remotely controlled pneumatic actuator which disconnects the Locomotive from the Transporter. All power and control connections for the Transporter functions are from the primary Locomotive which remains coupled in normal operation. The secondary Locomotive will be controlled from the primary Locomotive and will not require an electric and air brake connection to the Transporter as with the case for the primary Locomotive, which is not disconnected for normal emplacement operation.

A clearance or "pocket" is required in the face of the Emplacement Drift Transfer Dock for the rear Transporter coupler and is described further in Section 7.9.

7.2.3.5 Brake System

The Transporter is equipped with a fail-safe air brake system that is interconnected and operates in conjunction with the Primary Locomotive, similar to rail industry practice. The system utilizes spring set, air release brakes and includes the brake shoes, air cylinders, and operating linkage installed on the trucks with the air reservoir, piping, and miscellaneous equipment located on the underframe. The air brakes are connected to the Primary Locomotive with rail industry standard manual connections. A separate redundant braking system will be provided for the Transporter and will be addressed in a future analysis.

7.2.3.6 Transporter Door Operators

The Transporter will have an automatic door operator, controlled from the Primary Locomotive through a removable connector between the two. Each door shall have an operator and both, working in unison, will open and close the shielded doors for WP Rail Car loading at the surface WHB and the unloading at the Emplacement Drift, (Figure 7.2.6). Each door weighs approximately 10,011 kg and is fixed to a hinge pin, which rotates in the lubricated sleeve bearings of the door hinge allowing it to swing 270 degrees to a location at the side of the Transporter (Attachment III). The door hinge includes a thrust bearing to support the vertical load of the door. The doors are rotated in either direction for opening or closing by the 178 mm (7 inch dia.) hinge pin that extends through the shield floor and is connected through a spline joint to a low speed (1 rpm) motor gear reducer. Refer to Attachment III for door operator speed selection. The motor gear reducer is a right angle helical-worm unit for flange mounting to the underside of the shield floor. Gearmotor input is 1750 rpm and output is 1 rpm with 268 kg-m (in-lb) of torque and requires a ¾ hp, 1750 rpm

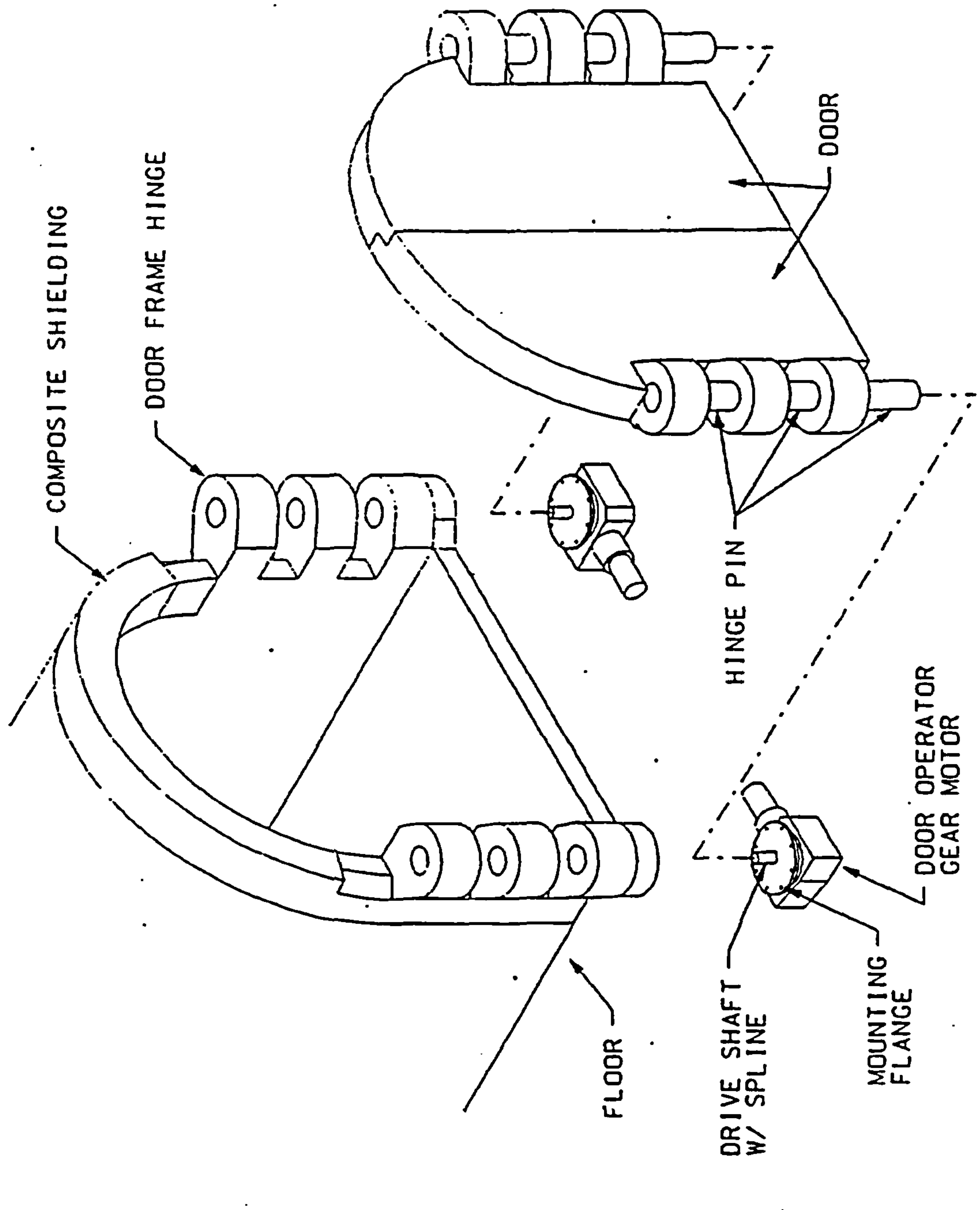


FIGURE 7.2.6
WASTE PACKAGE TRANSPORTER
DOOR OPERATOR

CAD FILE: t:\reposs\mech\fig\ssme024l.fig motor. The doors will also be equipped with a locking device to secure the closed doors against the body of the Transporter, maintaining the radiation seal and preventing accidental opening during transport operations. The door lock will be operated remotely and interlocked with the door operator operation. Details for the door design and locking device will be the subject of further analysis.

7.2.3.7 Reusable Rail Car Restraint

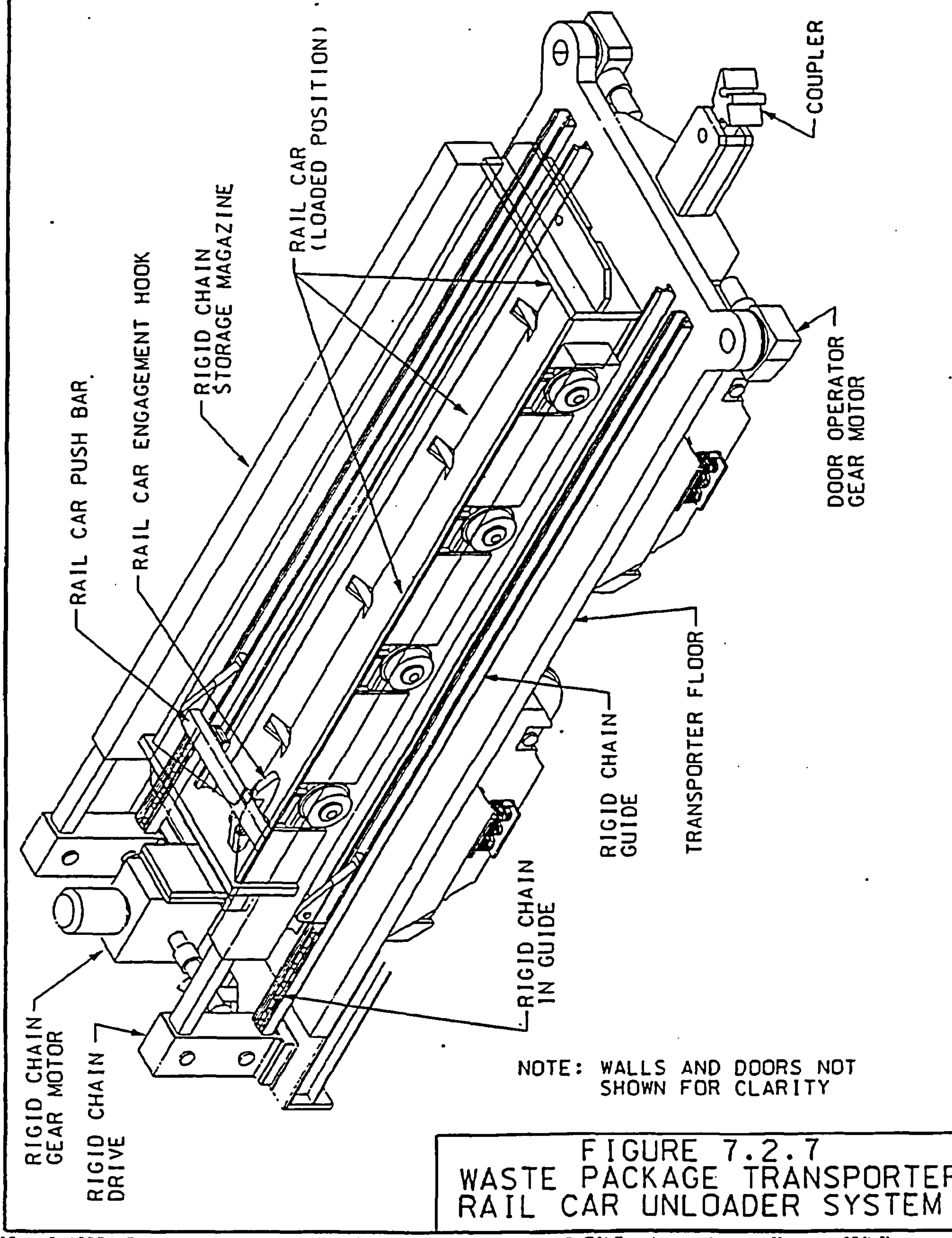
The Reusable Rail Car is attached to the loading/unloading mechanism which positions the Rail Car on the rails both inside and outside the Transporter. A restraint will secure the Rail Car when the Transporter is in the transit mode. The Rail Car unloader engagement hook secures the Rail Car from movement on the rails, and the restraint secures the WP and Rail Car in all degrees of freedom. The restraint will be the subject of further analysis.

7.2.3.8 Reusable Rail Car Unloader System

One of the major operations in the final placement of the WP is the unloading of the WP Reusable Rail Car from the Transporter to the Emplacement Drift using the Rail Car unloader system. The loading/unloading mechanism utilizes a proprietary "rigid chain" design, which can be coiled up like a conventional chain, but assumes the characteristics of a rigid bar when uncoiled and subjected to a force along its uncoiled length (Attachment X). This feature allows for storage of the chain in the coiled state to provide for the 12 m travel required for unloading the Reusable Rail Car. The main chain components are the storage magazines, right-angle chain drives, and guides for the chain in the rigid state. This concept utilizes two separate rigid chain systems, which allows all chain to be stored in magazines inside the Transporter on each side of the WP Rail Car with drives located outside the shielding on an equipment platform for repair and service. The two drives are connected to a common gearmotor for synchronous operation. Each of the rigid chains are drawn from the respective magazine through a penetration in the shielding to the drive and then driven back through another penetration in the shielding. Inside the shielding, the rigid chain runs in a guide installed on the floor where it is connected to a "pusher bar," which engages the Rail Car and moves it through the open doors of the Transporter and out into the Emplacement Drift. The pusher bar is supported off of the two Rail Car rails on equipment rollers with vertical guide rollers to maintain alignment. The rigid chains run in guides in the Transporter floor and continue on into the tunnel in the same guides installed in the tunnel floor. The pusher bar is equipped with a counterweighted self latching engagement hook that serves to maintain contact between the bar and the Rail Car front plate and draw the Rail Car back into the Transporter. The engagement hook may be released to free the Rail Car from the unloader if needed in the WHB (Figure 7.2.7) or for maintenance. The mechanism for the hook release is the subject of future analysis.

7.2.3.9 Transporter and Emplacement Drift Transfer Dock Alignment and Support

The alignment of the Rail Car tracks and the two rigid chain guides at the interface point with the emplacement dock is critical for the proper operation of this concept. This interface is depicted in Figure 7.2.8. Refer to Section 7.9 for functional requirements. The Transporter loading and unloading alignment and support system design is the subject of future analysis.



NOTE: WALLS AND DOORS NOT SHOWN FOR CLARITY

FIGURE 7.2.7
WASTE PACKAGE TRANSPORTER RAIL CAR UNLOADER SYSTEM

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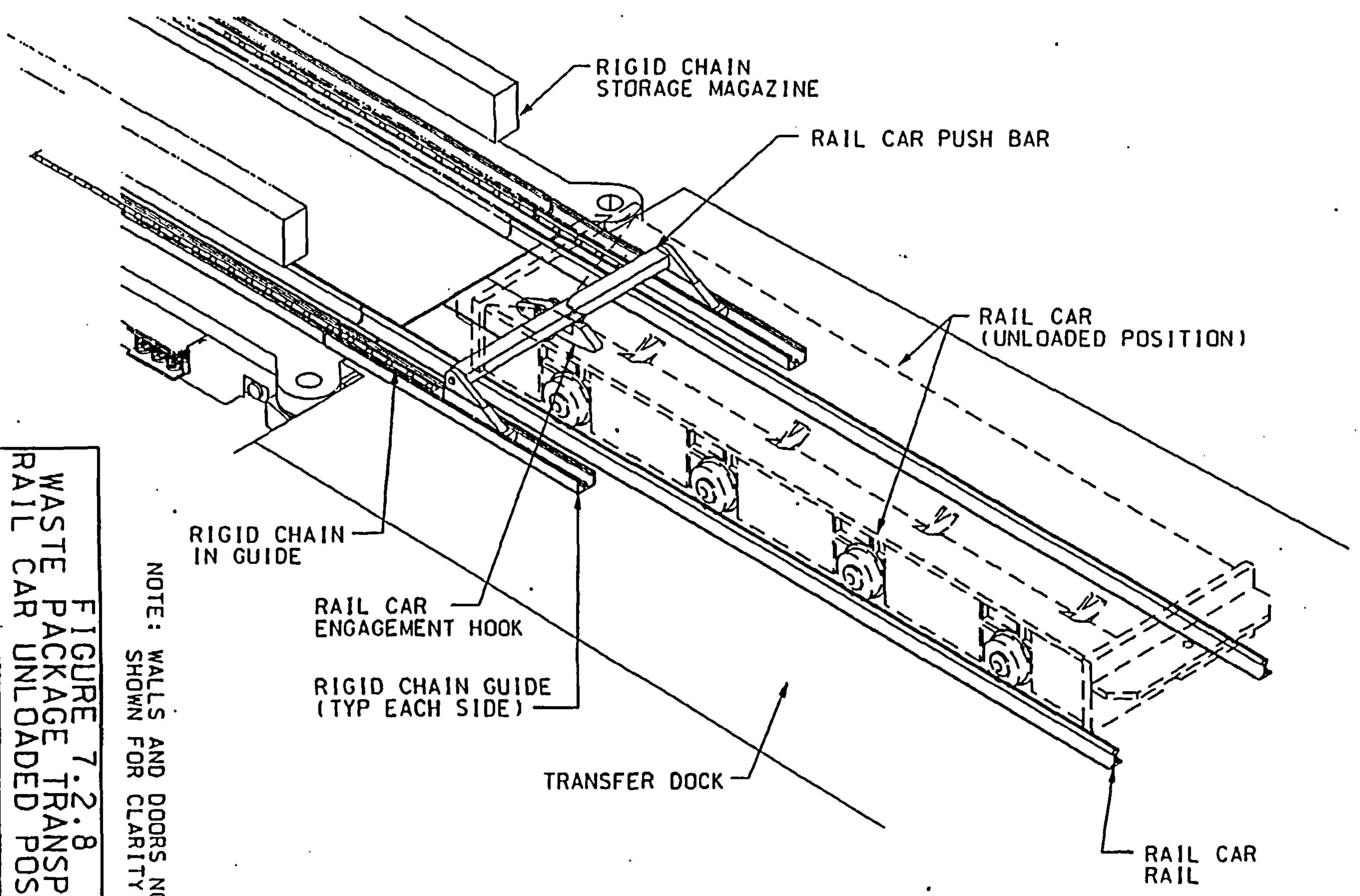

NOTE: WALLS AND DOORS NOT SHOWN FOR CLARITY

FIGURE 7.2.8
WASTE PACKAGE TRANSPORTER
RAIL CAR UNLOADED POSITION

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7.2.3.10 Wiring, Interlocks, and Instrumentation

The Transporter will be prewired in rigid conduit for power and control, located outside the Transporter shielding for ease in maintenance and/or repair. The wiring, required for interconnection to the Primary Locomotive, will terminate in separate cable bundles for power and control to separate terminal blocks in a common enclosure. The connection of the cables between the Transporter and Locomotive will be by quick connect/disconnect connections. The control electronics will be installed inside an enclosure with easy access for connection to the Locomotive cables. The Transporter will be equipped with interlocking devices for safety and proper operation. The Transporter will also be equipped with instrumentation for monitoring internal environmental conditions, such as temperature and status conditions, such as "door locked". This information will be relayed back to the Primary Locomotive for control and monitoring purposes.

Required electrical connections between the Transporter and the Primary Locomotive include:

- Door system power, control and status
- Unloader system power, control and status
- Reusable Rail Car unloader connection power, control and status
- Transfer/loading dock alignment status
- CCTV monitors power and controls
- Environmental data (temperature, etc.) connections
- Lighting

7.2.4 Interfaces

The WP Transporter will interface with the following equipment items or systems:

- WHB
- Reusable Rail Car (which is considered a part of the WP Transporter)
- WPs
- Transport Locomotive
  1. Couplers
  2. Electrical, Control and Instrumentation Systems
  3. Air Brake System
- Rail and Switch Systems Included in North Ramp, Main Drift, and Drift Turnouts
- Emplacement Access Tunnels
- Emplacement Drift Transfer Dock
- Rail Car Unloader Chain Guides in Emplacement Drift
- Rail Car Rails in Emplacement Drift

7.2.5 Transporter Structural Analysis

A structural analysis was performed on the Transporter structure using STAAD-III/IDS software. The analysis performed on the Transporter was less comprehensive than the analysis for the Emplacement Gantry since, in this application, the most important output was the total operating weight rather than overall envelope size. The maximum operating weight was essential to the sizing of the trucks, wheels, rails, and locomotives. The final Transporter arrangement reflects the results of the analysis performed.

The Transporter structural analysis is addressed in Attachment I.

7.3 REUSABLE RAIL CAR

7.3.1 Attachment Reference

Attachment IV - Reusable Rail Car - Mechanical Equipment Selection

7.3.2 Functional Requirements

The purpose of the Reusable Rail Car is to provide a means of supporting and moving the WP into the Transporter, supporting and securing the WP inside the Transporter, functioning with the WP unloader system, and transferring WP from the Transporter to the Emplacement Drift.

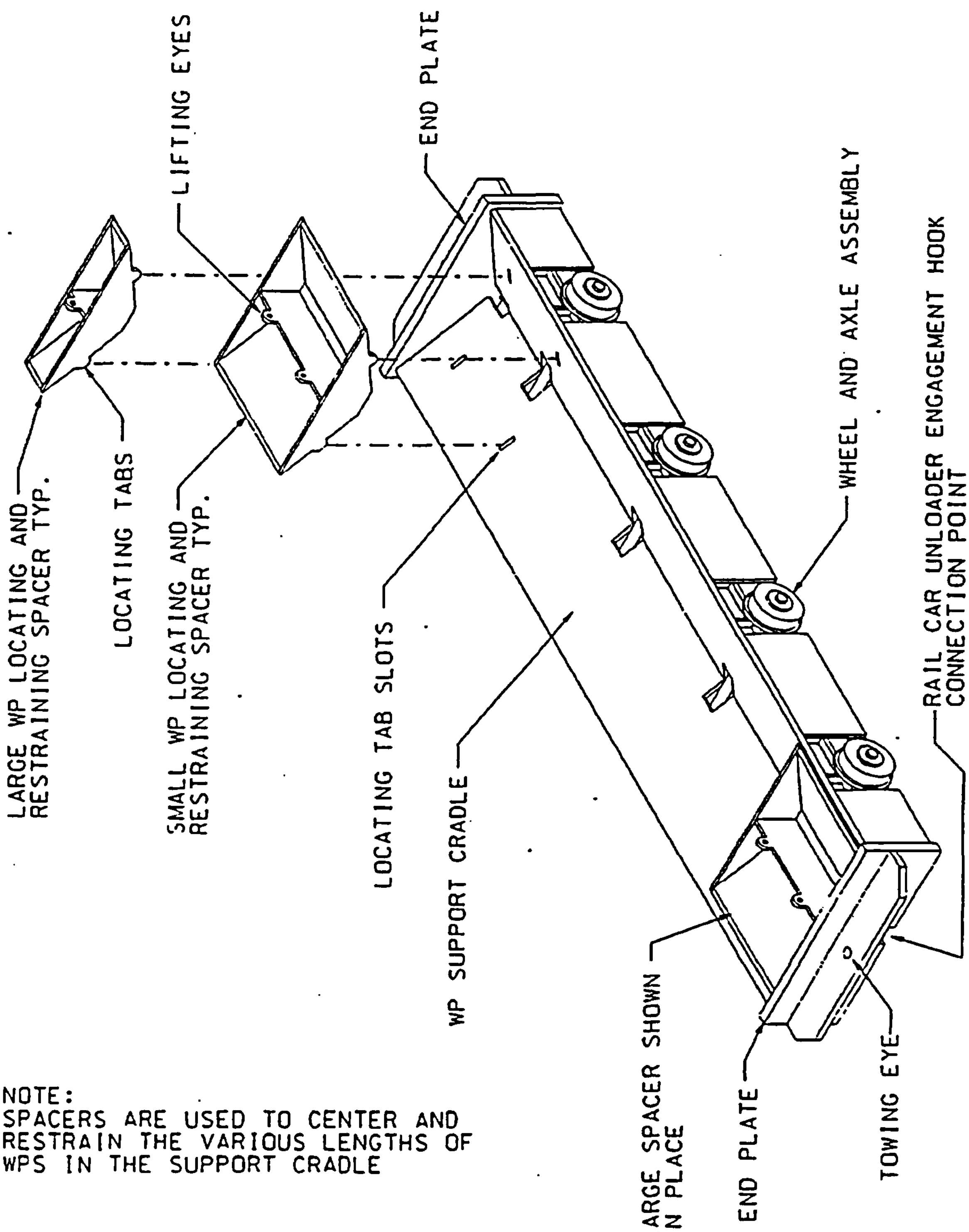
The Rail Car, as depicted in Figures 7.3.1 through 7.3.4, accommodates various sizes of WPs. The physical sizes and weights of the WPs carried by the Rail Car are outlined in (Section 4.1.6 and 4.3.11). The Rail Car design is based on a WP of the maximum dimension, 2.0 m dia. x 5.85 m long (Section 4.3.11), and maximum weight, 69,000 kg (Section 4.3.11). The Rail Car may be detached from the Transporter loader/unloader system as necessary for WP loading operations in the WHB or for maintenance. The Rail Car is equipped with a towing eye on both ends for connection to a prime mover when detached from the loader/unloader.

7.3.3 Rail Car Subsystems

To accomplish these functions the Rail Car is composed of the following subsystems.

7.3.3.1 Underframe and WP Support Structure

The Rail Car is fabricated from ASTM A36 structural steel plate welded per AWS D1.1 (Ref. 4.4.3). The V-shaped WP support, or cradle, is supported on a boxed structural section with provisions for the attachment of four axle assemblies. The length of the cradle accommodates the longest design WP of 5850 mm. WPs of shorter length are centered lengthwise between removable spacers placed at each end of the cradle, which not only prevents longitudinal shifting of the WP in transit, but also ensures equal loading of wheels, and positions the WP in the Rail Car for engagement by the Gantry crane lifting heads.



NOTE:
SPACERS ARE USED TO CENTER AND RESTRAIN THE VARIOUS LENGTHS OF WPS IN THE SUPPORT CRADLE

FIGURE 7.3.1
REUSABLE RAIL CAR ARRANGEMENT

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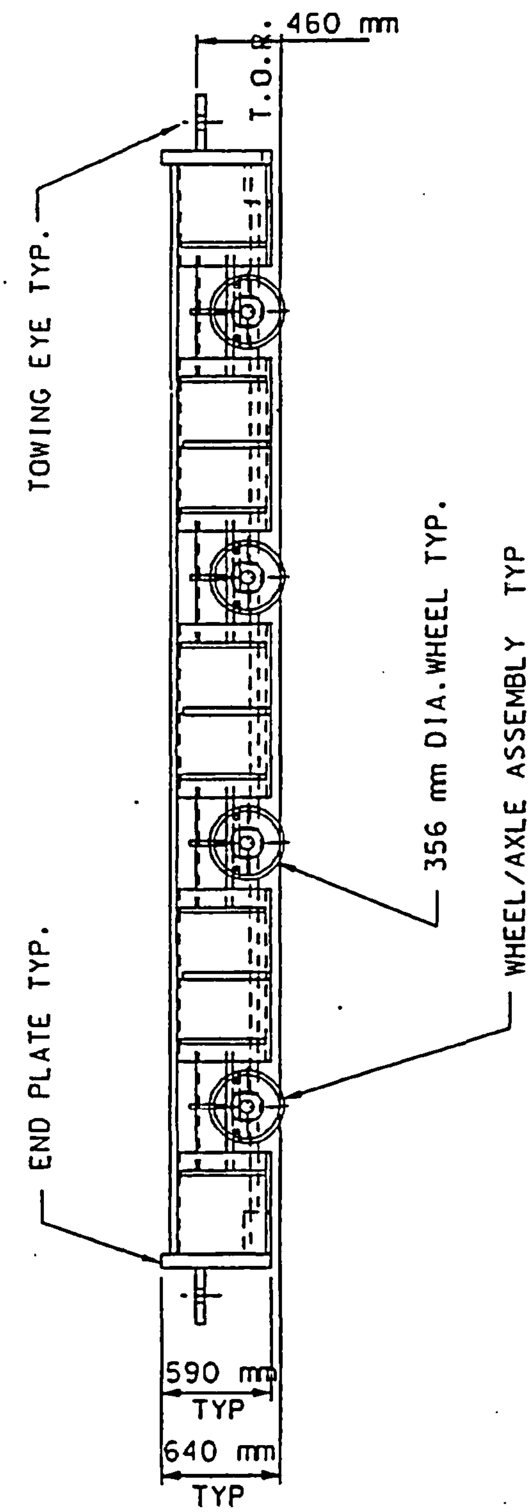

FIGURE 7.3.2
REUSABLE RAIL CAR
ELEVATION

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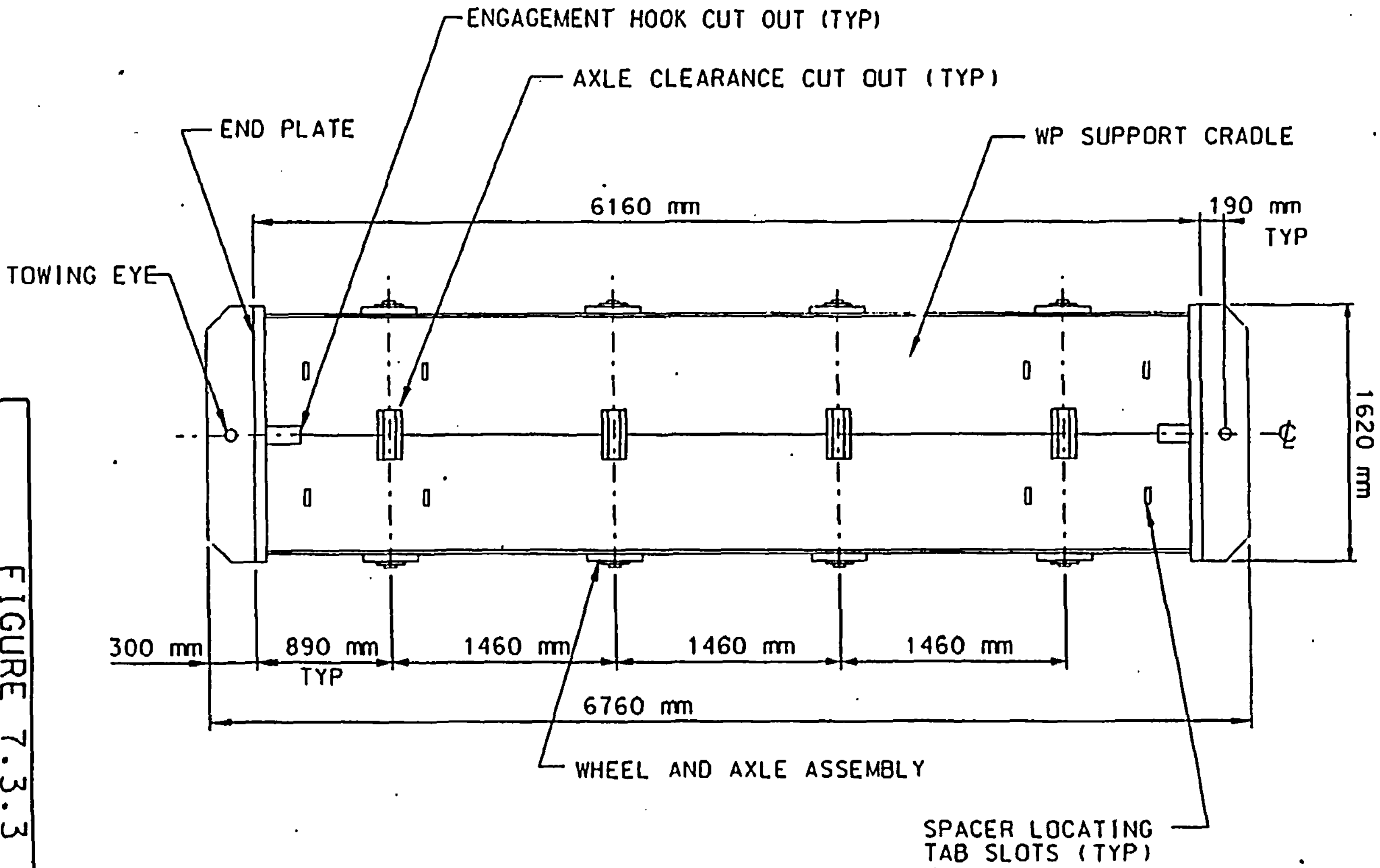

FIGURE 7.3.3
REUSABLE RAIL CAR
PLAN VIEW

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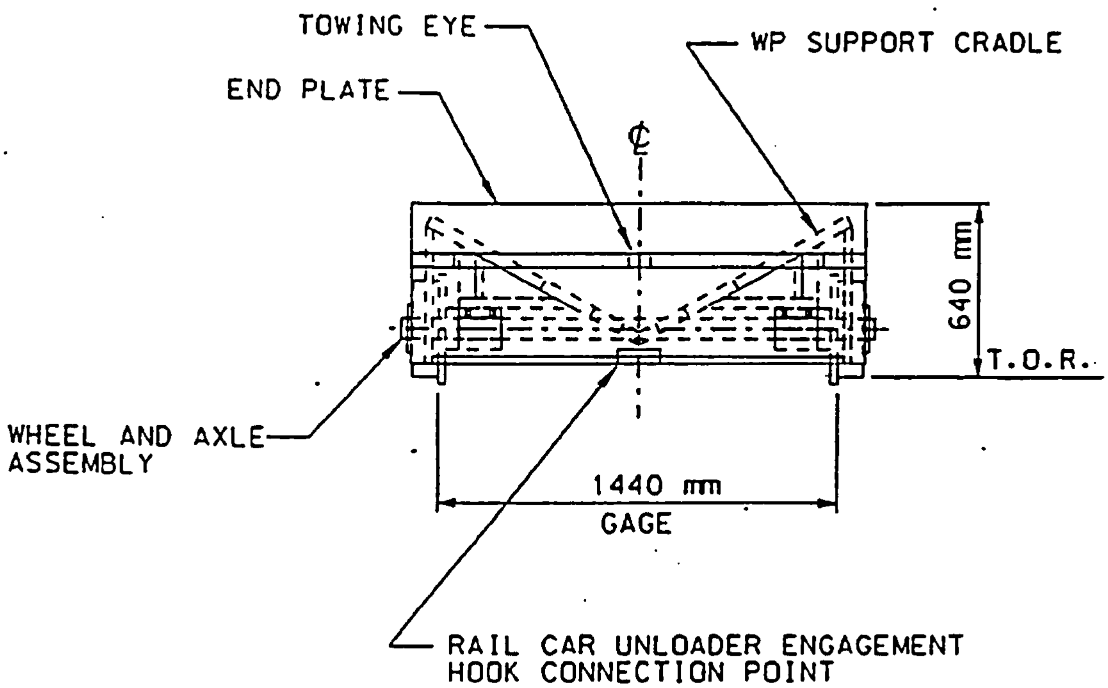

FIGURE 7.3.4
REUSABLE RAIL CAR
END VIEW

7.3.3.2 Wheel/Axle Assembly

The Rail Car and WP are supported on four equally spaced wheel and axle assemblies. Each assembly includes an axle with supporting brackets for attachment to the underframe and wheels with integral bearings attached to each end of the axle. The assembly is the standard manufacturers product used in heavy-duty and severe applications, such as foundries, mills, and mines. The unit selected for this application utilizes 356 mm (14 inch) diameter single-flange rail wheels with a rated capacity of 13.6 MT (15 tons). The Rail Car will travel on 44.6 kg/m (90 lb/yd) ASCE rail in the Transporter and in the Emplacement Drift Transfer Dock. Refer to Attachment IV for Rail Car weight analysis, wheel selection, and rail verification.

7.3.3.3 WP Spacers

The Reusable Rail Car WP support cradle will accommodate the longest WP (5850 mm) between the end plates. For the two other lengths (5335 mm and 3790 mm) (Section 4.1.6), one of two sizes of removable spacers will be placed at each end of the cradle to assure positioning and retraining the WP in the middle of the cradle. The spacers, if required, would be installed at the WHB with the WP. Locating tabs on the spacers fit into respective slots in the cradle, and position the spacers against each end plate.

7.3.4 Interfaces

The Rail Car will interface with the following equipment items or systems:

- WPs
- WHB
  1. Rail
  2. WP Loading System
- WP Transporter
  1. Rail
  2. Unloader
  3. Rail Car Restraint
- Emplacement Drift Rail
- Emplacement Gantry

7.3.5 Structural Analysis

No structural analysis was performed for the Reusable Rail Car. The sizing of materials and the rail car configurations shown are based on a mechanical evaluation of envelope limits and WP sizes.

7.4 EMPLACEMENT GANTRY

7.4.1 Attachment Reference

Attachment V - Gantry Loads and Equipment Selection

7.4.2 Functional Requirements

The purpose of the Emplacement Gantry is to receive and transport WPs from the Reusable Rail Car to the placement position in the Emplacement Drift. The Emplacement Gantry, shown in Figures 7.4.1 through 7.4.4, is a self-propelled remotely operated vehicle. In performing the emplacement function the Gantry must complete several subfunctions, including:

- Accurate positioning over the WP on the Reusable Rail Car to effect engagement of the WP by the lifting heads.
- Lifting of the WP from the Reusable Rail Car.
- Lifting of the WP over the Rail Car end plate and the concrete shadow shielding, if in place.
- Traversing of the WP to its designated placement position.
- Accurate positioning of the WP over the existing support pedestals.
- Lowering of the WP onto the support pedestals.
- Returning to the drift entry to receive the next WP.
- If deemed necessary at a later time, retrieve WPs from the Emplacement Drift by reversing the placement process.

In addition to these functions and in accordance with earlier preliminary analysis (Ref. 4.3.21), the Emplacement Gantry has been designed to operate in a 5.5 m diameter drift with 200 mm radial allowance for ground support. Also, a 100 mm radial clearance for variations in drift diameter has been incorporated into the analysis. Consideration has been given to various Gantry functions to prevent encroachment of this clearance and to provide the Gantry with a maximum lift capability.

7.4.3 Gantry Subsystems

To complete the above functions the Emplacement Gantry is composed of several component systems, including the Gantry frame, traversing system, hoisting frame with adjustment capability to accommodate various WP lengths, and lifting head trolley. In addition, several other systems are required to aid in the operation and control of the Gantry. These systems include TV monitoring, braking, and traversing load support. All operating components of the Gantry are electrically operated from a Gantry power distribution system which receives primary DC power from brush contactors running on an electric conductor bar or third rail in the Emplacement Drift. Each of these components has a specific purpose as described in the following sections:

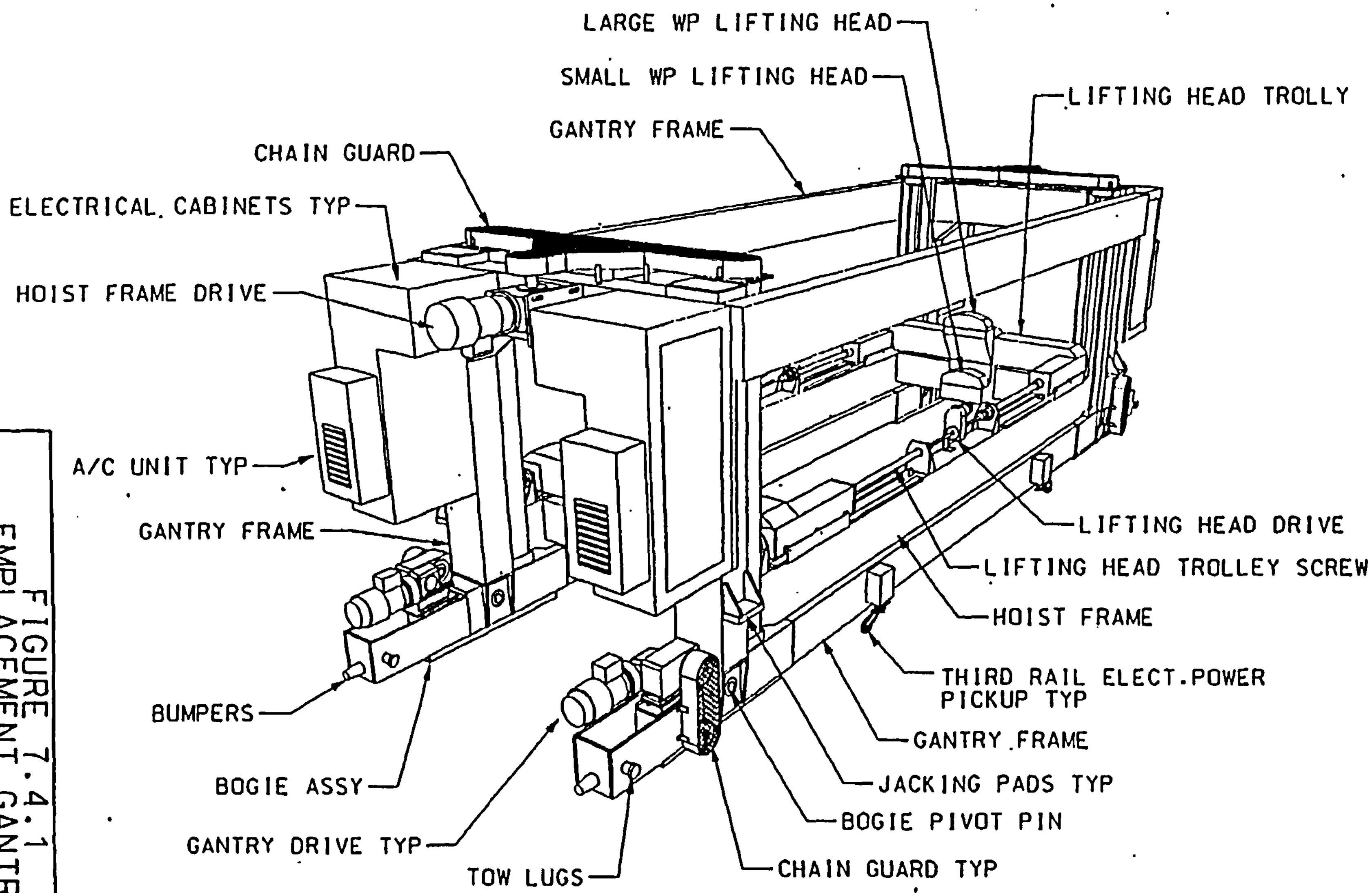

FIGURE 7.4.1
EMPLACEMENT GANTRY
ARRANGEMENT

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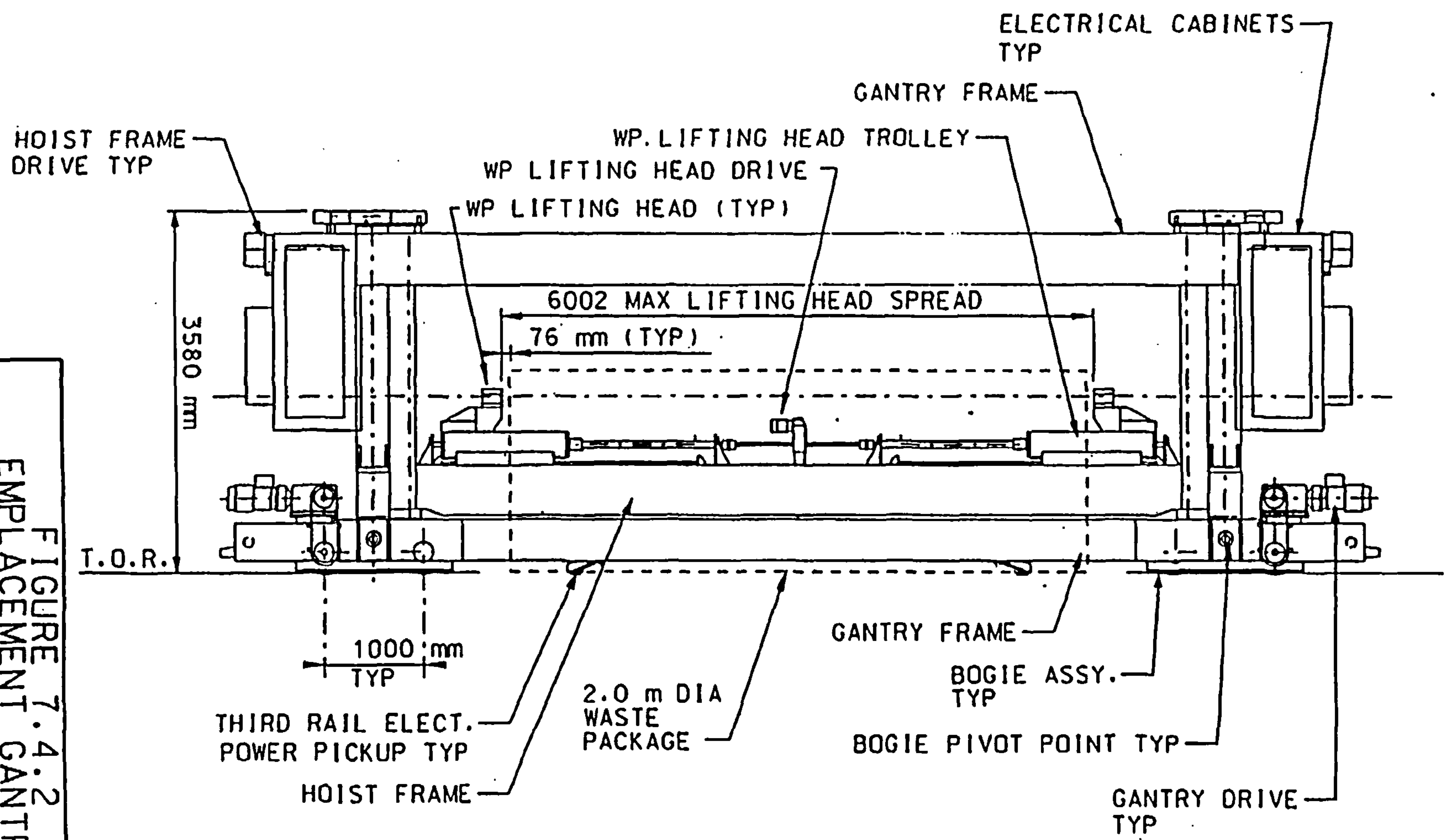

FIGURE 7.4.2
EMPLACEMENT GANTRY
ELEVATION

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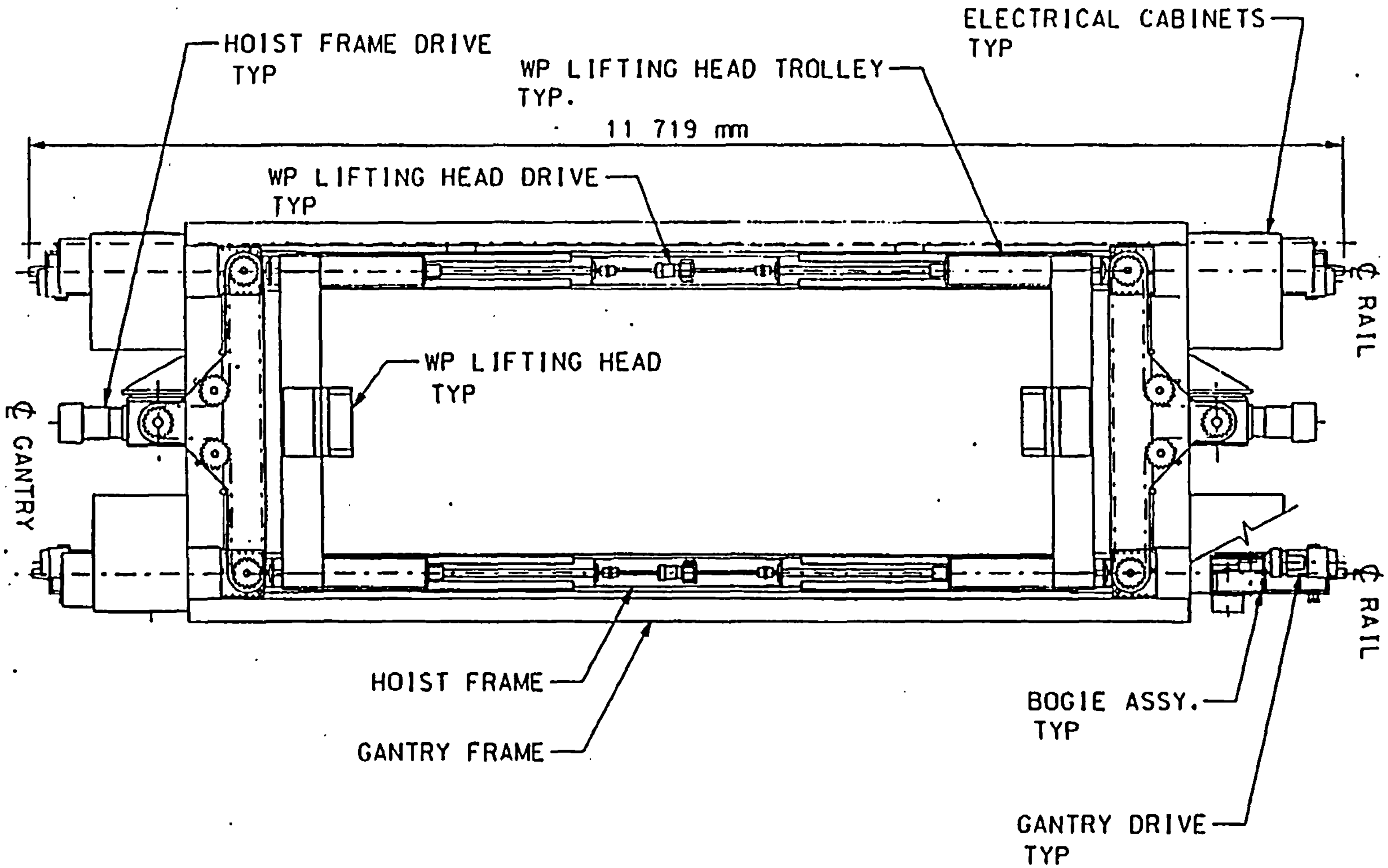

FIGURE 7.4.3
EMPLACEMENT GANTRY
PLAN VIEW

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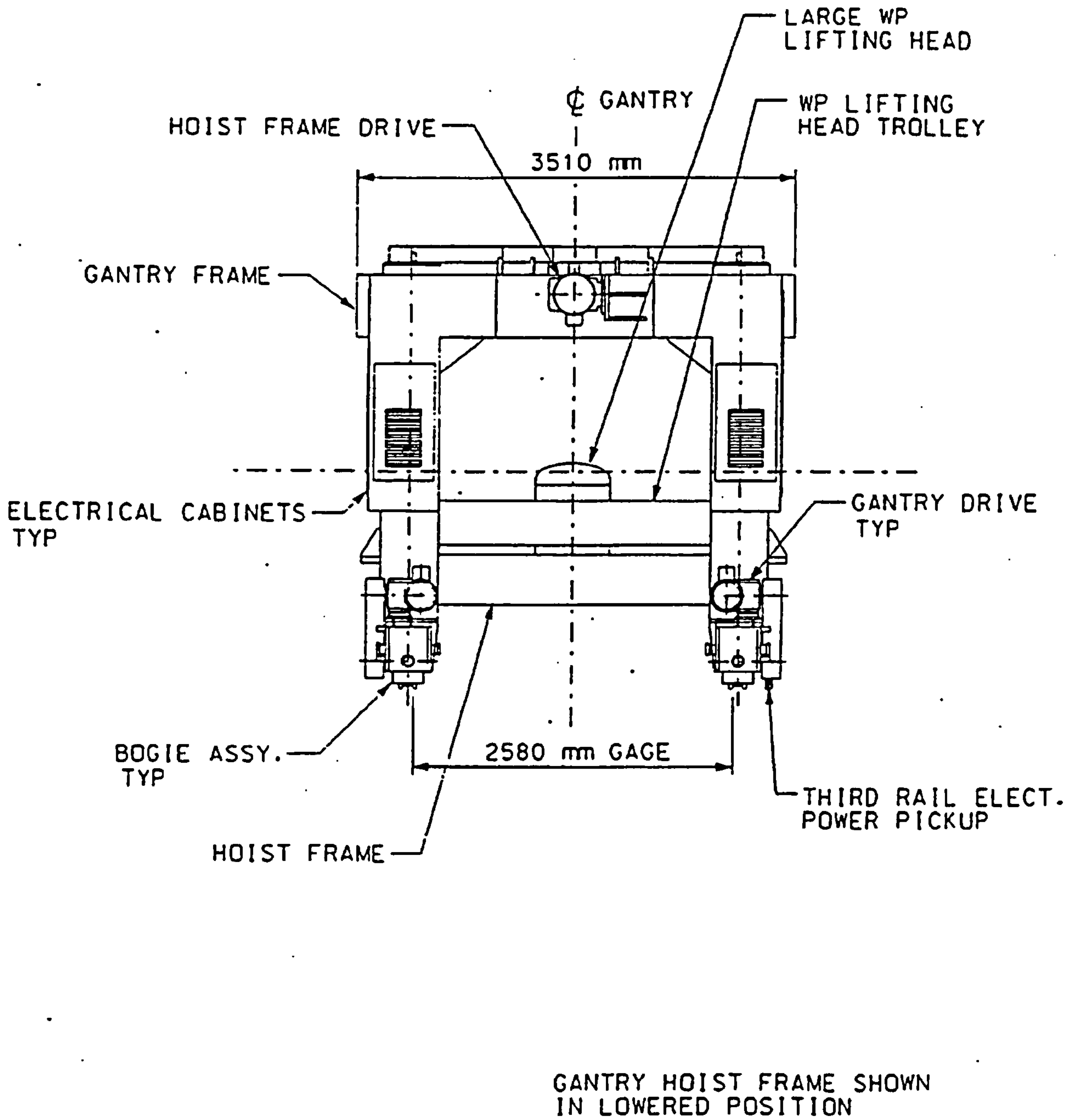

FIGURE 7.4.4
EMPLACEMENT GANTRY
END VIEW

7.4.3.1 Gantry Frame

An important component of the Emplacement Gantry is its structural frame. Structural failure may jeopardize the emplacement process. The Emplacement Gantry structure, shown in Figure 7.4.5, has several purposes. First, the structure must be designed to support its own weight, the WP, and the loads imposed during a seismic event. Secondly, the structure provides the bearing and drive support for the four lifting ball screw mechanisms required to lift the WPs. Third, the structure provides base support for the traversing bogies. The Gantry structural members are composed of ASTM A36 steel shapes and fabrication is in accordance with American Welding Society (AWS) standards. Steel materials, ASTM A36, used for this design reflect the lowest boundary of material strength satisfactory for use in the gantry structure. The material is durable, ductile and resists fatigue better than most of the higher strength steels. ASTM A36 and other high strength steels will be evaluated for appropriate use during the final design process. An analysis for sizing the structural members is presented in Attachment II.

7.4.3.2 Gantry Frame Options

The height to which it is necessary to lift a 2.0 m WP impacts the overall height of the Gantry frame and is a determining factor in selection of drift diameter. For this analysis lifting height is dictated by the following conditions:

1. Carrying a 2.0 m WP over another 2.0 m WP resting on pedestals on the drift invert. This option may be desirable (but is not required) if it becomes necessary to retrieve WPs at a later date. A 150 mm clearance between WPs is provided. This condition is depicted in Figure 7.4.6.

2. Retrieving a 2.0 m WP from the Reusable Rail Car. To complete this function, the controlling factor is the capability of the hoisting frame to clear the WP on the Reusable Rail Car. A 150 mm clearance between the hoisting frame and WP is provided. This condition is depicted in Figure 7.4.7.

3 Lifting a 2.0 m WP over a shadow shield installed on the drift invert. Height of the shadow shield is 75 mm above the top of a 2.0 m WP on a pedestal, and a 150 mm clearance between the shield and WP is provided. This condition is depicted in Figure 7.4.8.

Each one of these conditions has a corresponding set of dimensions, the sum of which will determine the Gantry lifting height. In addition, if all of these conditions are met at the same time, the condition resulting in the largest lifting height would control Gantry frame design. The dimensions associated with each condition are as follows:

1. For carry-over condition lift height is composed of the following dimensions:
   - 76 mm - Clearance between WP skirt (WP on pedestals) and lifting head
   - 1000 mm - one-half diameter of WP resting on pedestals in the Emplacement Drift invert
   - 1000 mm - one-half diameter of the lifted WP

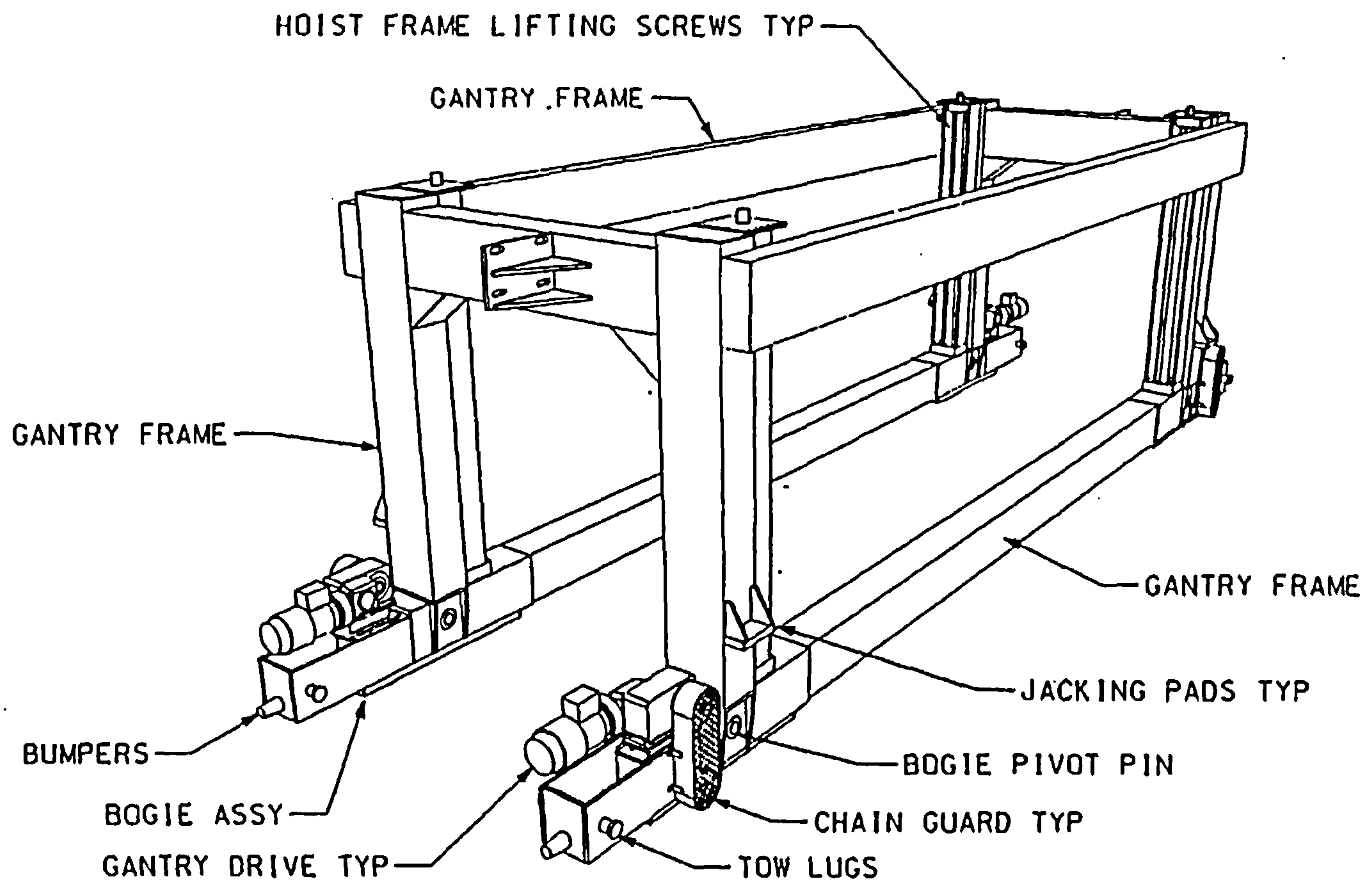

FIGURE 7.4.5
GANTRY FRAME ASSEMBLY

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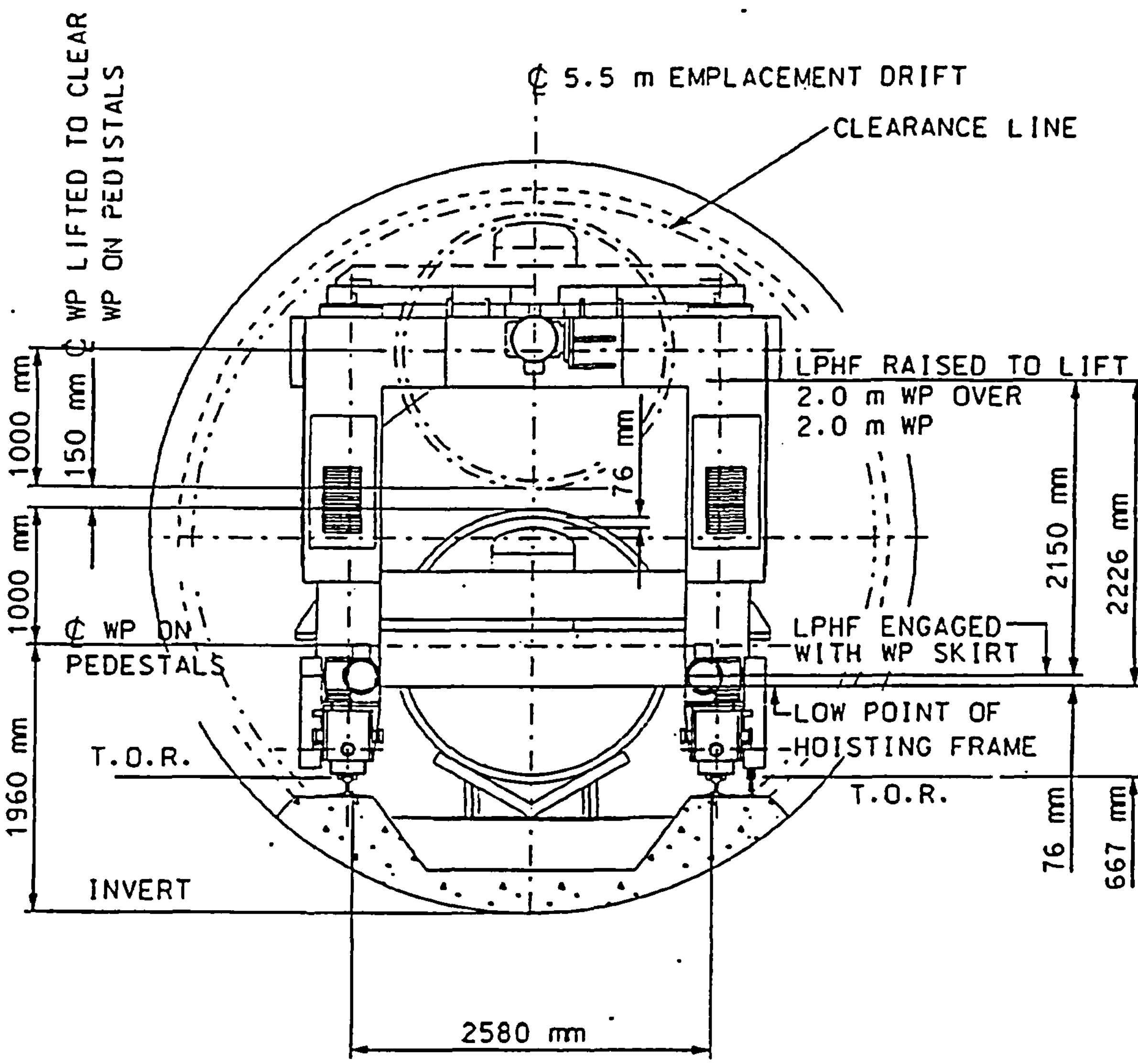
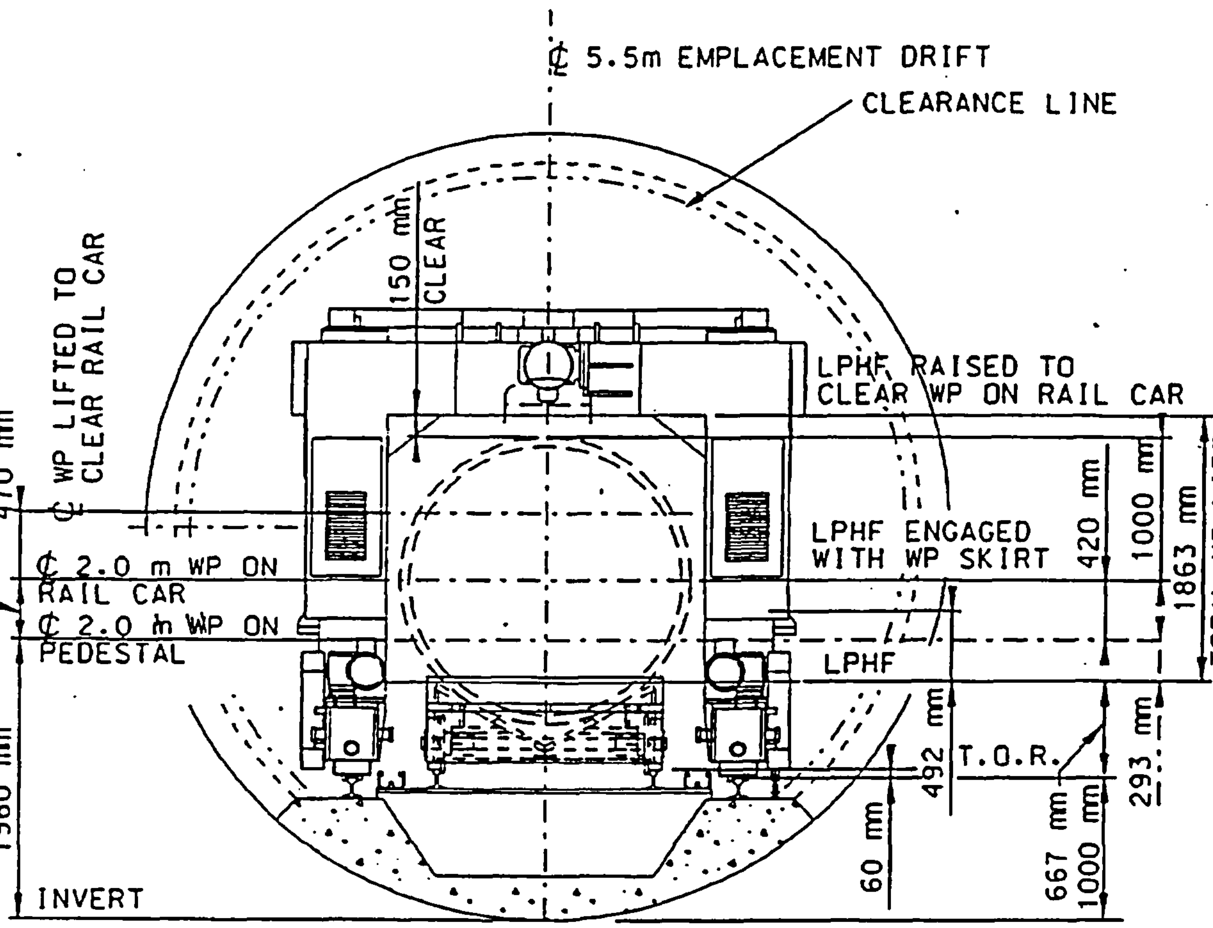


END ELEVATION VIEW

GANTRY CRANE POSITIONED
IN THE 5.5 m EMPLACEMENT DRIFT.
HOIST SHOWN IN RAISED AND LOWERED
POSITION

NOTE:
1. LPHF- LOW POINT OF HOISTING FRAME.
2. WP- WASTE PACKAGE

FIGURE 7.4.6
GANTRY WITH CARRY OVER



END ELEVATION VIEW

GANTRY CRANE POSITIONED IN 5.5 m EMPLACEMENT DRIFT
MAXIMUM LIFT OF HOIST FRAME
TO CLEAR WP ON RAIL CAR

NOTE:

1. LPHF- LOW POINT OF HOISTING FRAME.
2. WP- WASTE PACKAGE

FIGURE 7.4.7
GANTRY TO CLEAR
WP ON RAIL CAR

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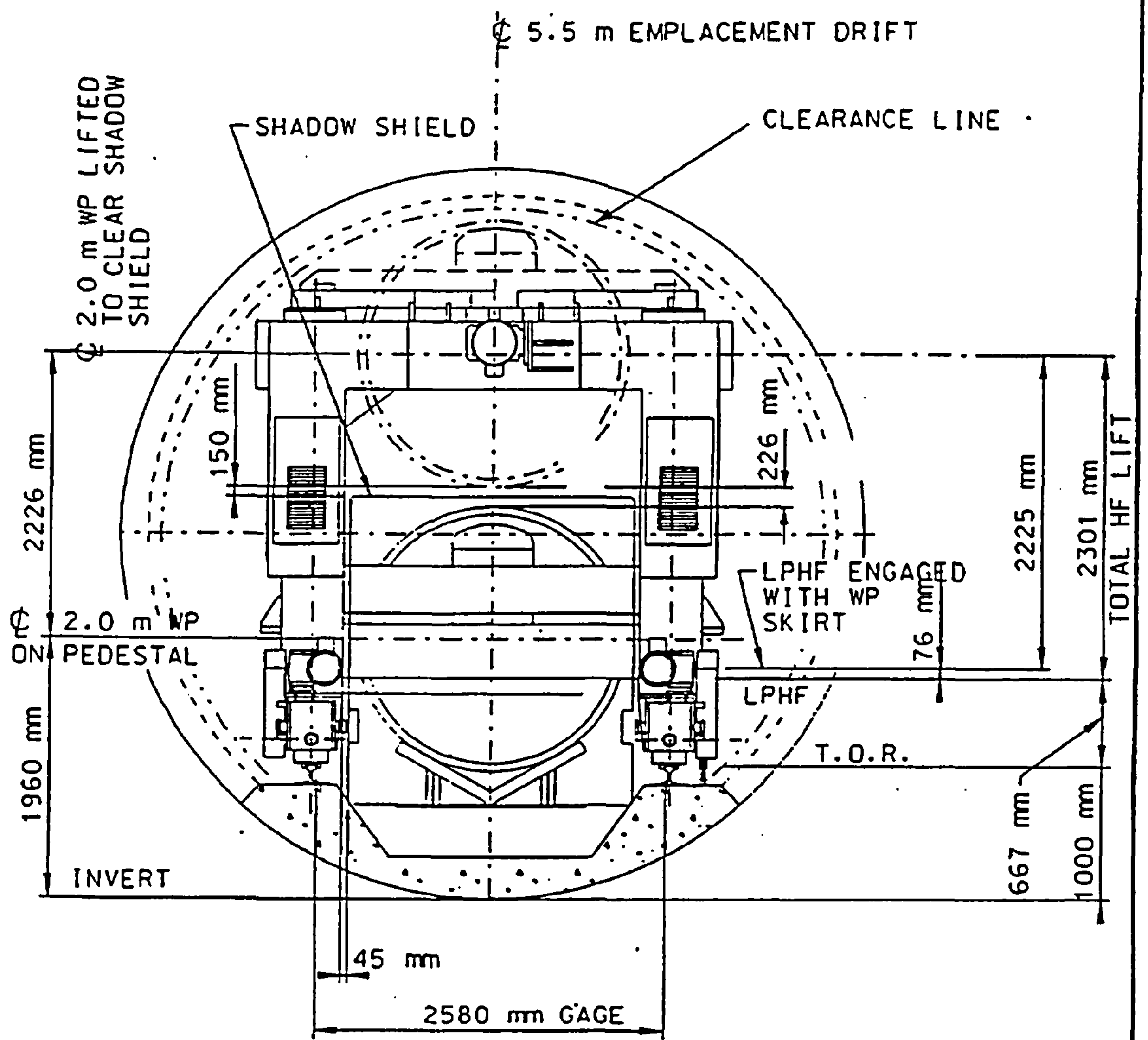

END ELEVATION VIEW

GANTRY CRANE POSITIONED
IN THE 5.5 m EMPLACEMENT DRIFT.
HOIST SHOWN IN RAISED AND LOWERED
POSITION

NOTE:

1. LPHF- LOW POINT OF HOISTING FRAME.
2. WP- WASTE PACKAGE

FIGURE 7.4.8
GANTRY WITH CARRY OVER
WITH SHADOW SHIELD

- 150 mm - Clearance between WPs (Note: The clearance dimension of 150 mm is based on engineering judgement, it has been used throughout this design analysis).

Lift height = 76 mm + 1000 mm + 1000 mm + 150 mm = 2226 mm.
Refer to Figure 7.4.6

2. To retrieve a WP from the Rail Car, the hoisting frame must clear the top of the WP. The lifting height is composed of the following component dimensions:

   - 713 mm - Distance from low point of hoisting frame (LPHF) in its lowest position to centerline of WP on Rail Car (293 mm + 420 mm)
   - 1000 mm - Radius of 2.0 m WP
   - 150 mm - Clearance over WP (see note to item No. 1 above)

   Lift = 713 mm + 1000 mm + 150 mm = 1863 mm
   Refer to Figure 7.4.7 (with lift height of 1863 mm, i.e. without WP carry-over capability the excavated drift diameter could be reduced to approximately 5.3 m.)

3. For a 2.0 m WP to clear an installed shadow shield the lifting height is composed of the following dimensions:

   - 76 mm - Clearance between WP skirt (WP on pedestals) and lifting head
   - 2000 mm - Diameter of WP resting on pedestals on the Emplacement Drift invert
   - 75 mm - Height of shadow shield above 2.0 m WP (Ref. 4.3.31)
   - 150 mm - Clearance between WP and shadow shield (see note to item No. 1 above)

   Lifting Height = 76 mm + 2000 mm + 75 mm + 150 mm = 2301 mm
   Refer to Figure 7.4.8

From these calculated lifting heights it can be concluded that the presence of a shadow shield is the determining factor as to whether the Gantry is designed with or without the carry-over option. In the case that a shadow shield is required and installed prior to the start of the emplacement functions, then the gantry will have to lift each WP above the shadow shield. This requirement will automatially result in the inherent carry-over capability of the gantry, because the shadow shield would have to be as tall or taller than the diameter of the largest WP, to be an effective shield.

The present design, with 2226 mm lifting height, is capable of lifting a 2.0 m WP over an installed shadow shield; However, clearance between the WP and shadow shield would be reduced to approximately 75 mm. Referring to Figure 7.4.8, with the current Gantry design the shadow shield will have to be constructed with notches in the sides to provide clearance to the tow lugs. The minimum side clearance shown as 45 mm will be approximately 60 mm since the largest WP diameter is 1970 mm and 2000 mm was used for the general arrangement. The selection of appropriate clearances was based entirely on engineering judgment. A more precise determination of clearance requirements should be the subject of a future analysis.

7.4.3.3 Traversing System

The Emplacement Gantry is self-propelled using four two-wheel bogies with one bogie placed at each corner. One wheel in each bogie is driven by a 5 hp DC variable-speed electric motor through a right angle gear reducer and roller chain assembly. This drive option is shown in Figure 7.4.1 and provides the largest clearances between drift wall and drive mechanism. The gear motor, having a nominal reduction ratio of 50:1 and a chain drive ratio of 1:1, will provide an output speed of 35 rpm. At this wheel speed the Gantry travel speed will be up to 0.732 m/sec (144 ft/min.), but will be controlled not to exceed 0.711 m/sec (140 ft/min.). Acceleration of the Gantry will be controlled by varying the motor voltage. Speed and deceleration of the Gantry will be controlled by a brake, which is integrally mounted in the motor. A redundant restraining system shall also be provided for emergency braking, constraining Gantry in position over the WP in both lifting and placing operation and on the Gantry Carrier. This will be a subject of future analysis. The use of four motors for travel meets the requirements of ASME NOG-1, which requires that each four-wheel unit utilizes a drive arrangement that provides power to at least 50 percent of the wheels. Bogie wheels are 400 mm in diameter and have a load capacity of 19.8 MT. Since there are 8 wheels per Gantry, the total load capacity is 158.4 MT, which is sufficient to support the weight of the WP and Gantry structure of 114 MT by a factor of 1.39. Calculations for wheel and gearmotor sizing along with verification of rail size are presented in Attachment V. Evaluation of air and hydraulic braking and restrainment systems for stabilizing lifting and lowering operations designed to operate in high radiation environments will be considered in future studies.

7.4.3.4 Traversing System Options

While the base Gantry drive option (shown in Figure 7.4.1) provides the advantage of large clearances, it has the disadvantage of using more parts that can fail. For this reason a direct drive option was considered. The arrangement was considered with the shaft mounted drive located both on the outside (Figure 7.4.9) and inside (Figure 7.4.10) of the bogie. As can be seen in the figures with the outside drive mounting, the drive encroaches upon the 100 mm clearance zone. With the inside mounting the rail gage is increased to avoid interference with the 2.0-m WP, and the rail is located on the outside edge of the invert haunch. From a design standpoint, this is an undesirable location for the rail. For these reasons the base case option is considered the best choice.

7.4.3.5 Hoisting Frame

The hoisting frame directly supports the WP when it is lifted by the Gantry, and it is designed so that WPs can be placed horizontally in the Emplacement Drift with a minimum spacing of 0.92 m (Reference 4.3.30). This frame is depicted in Figure 7.4.11 and is constructed wide enough to accommodate the 2.0 m diameter WP and long enough to accept the 5850 mm WP with a maximum clearance of 6002 mm between lifing heads. Adjustment for the length of the WP is included in the hoisting frame. This is accomplished with ball screws located on the outside of the longitudinal support beams. The screws position the lifting head trollies at each end of the hoisting frame to accommodate the various WP lengths. This adjustment mechanism is depicted in Figure 7.4.12. The ball screws are 57 mm (2¼ in.) in diameter and are made in right hand and left hand

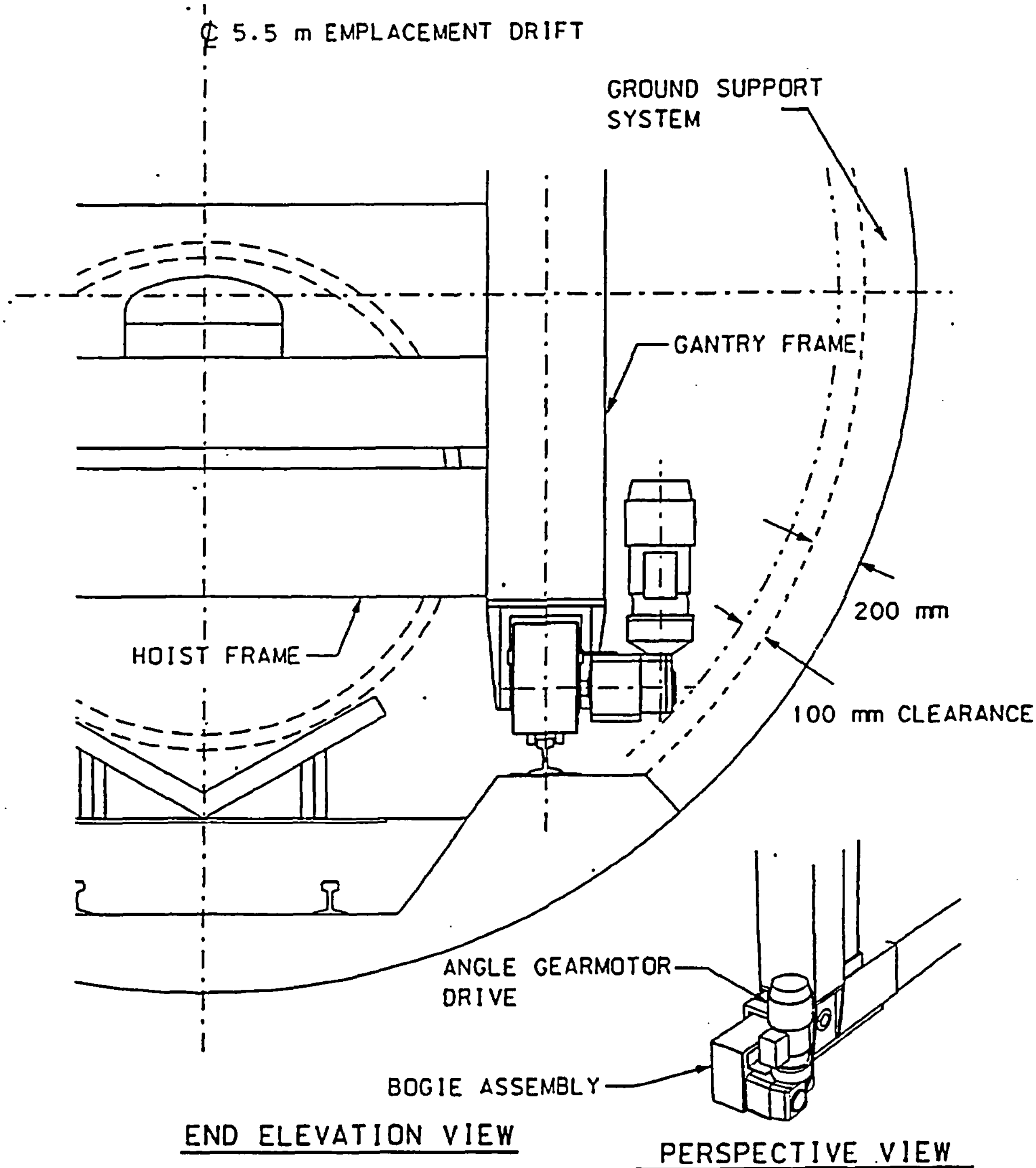

FIGURE 7.4.9
TRAVERSING DRIVE
OUTSIDE MOUNTING

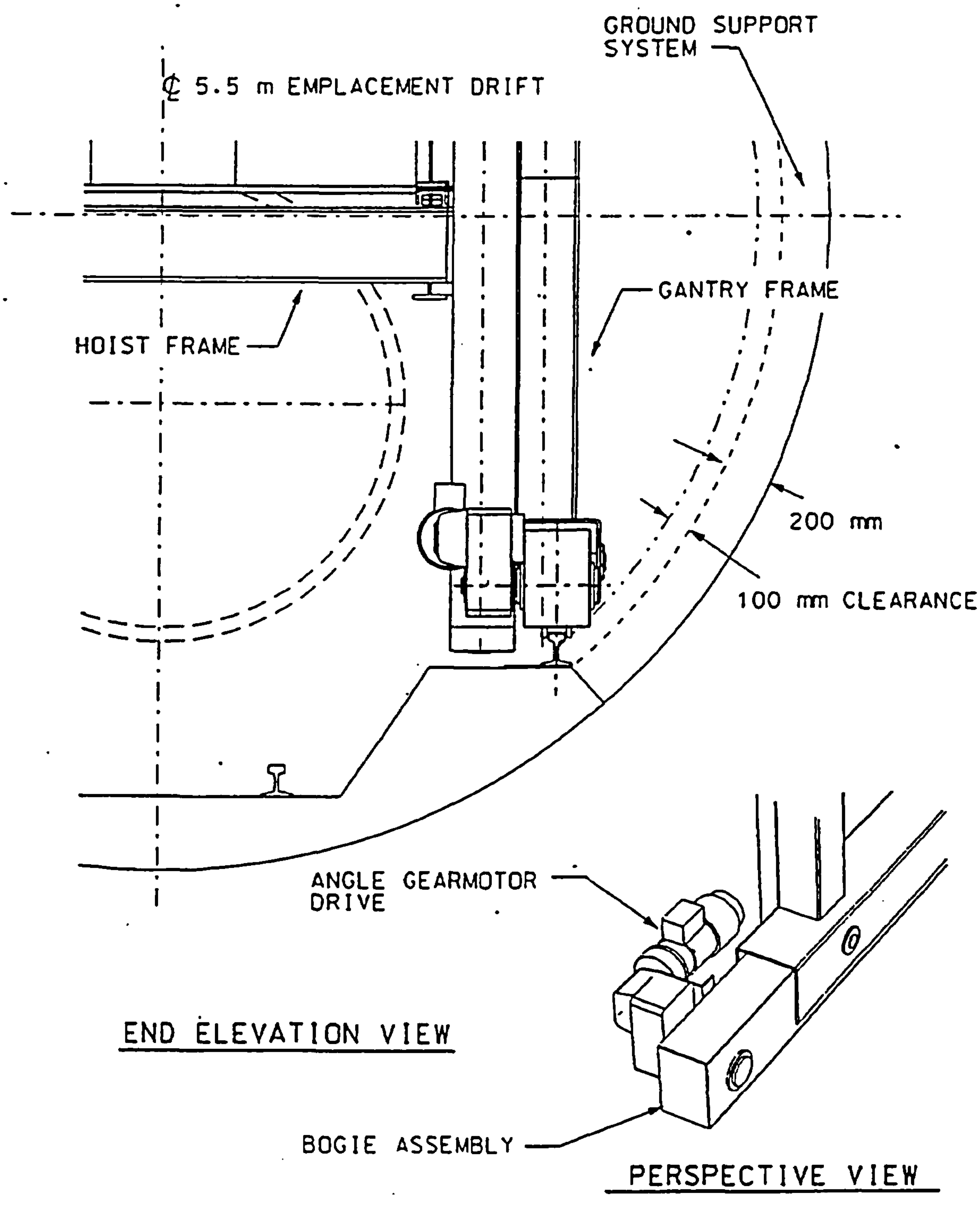

FIGURE 7.4.10
TRAVERSING DRIVE
INSIDE MOUNTING

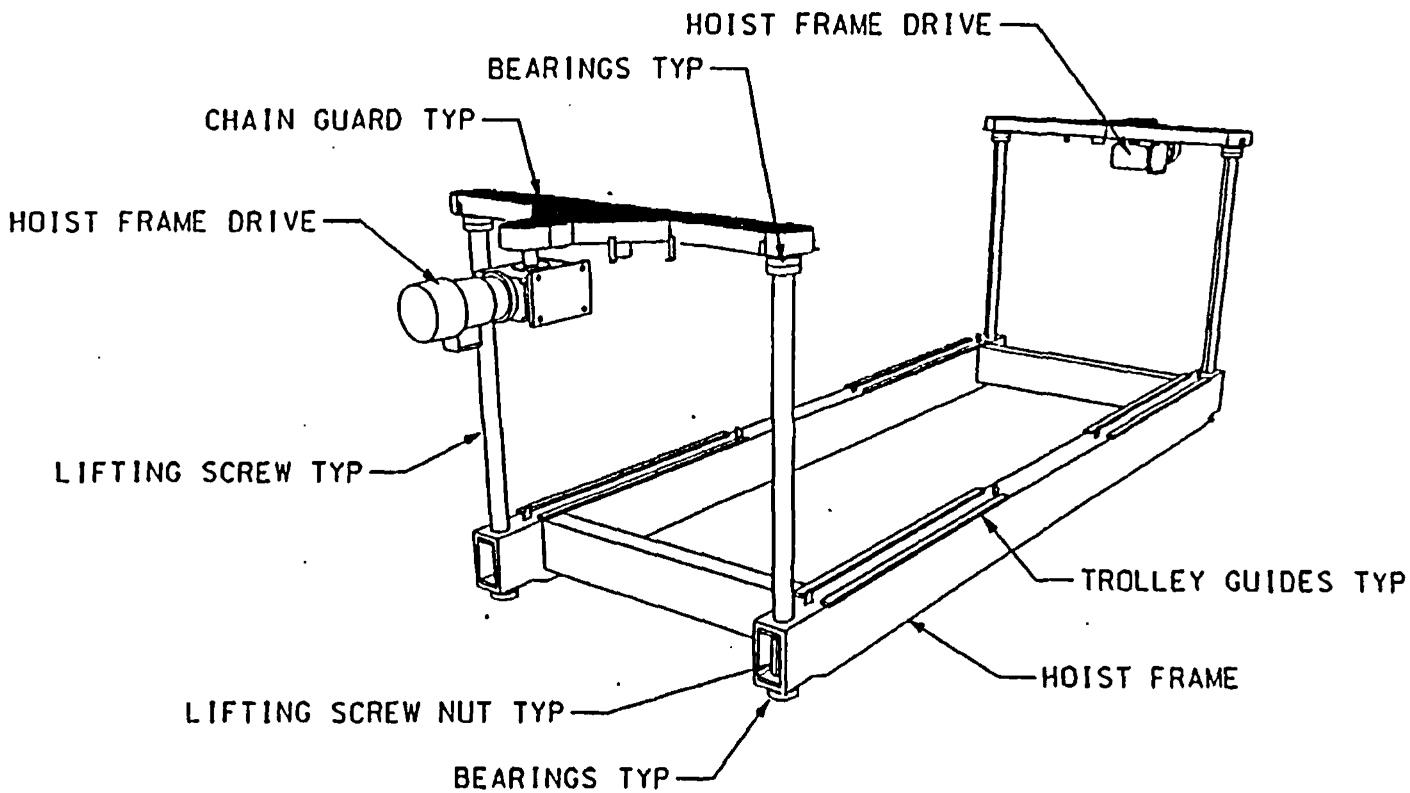

FIGURE 7.4.11
GANTRY HOIST FRAME
ASSEMBLY

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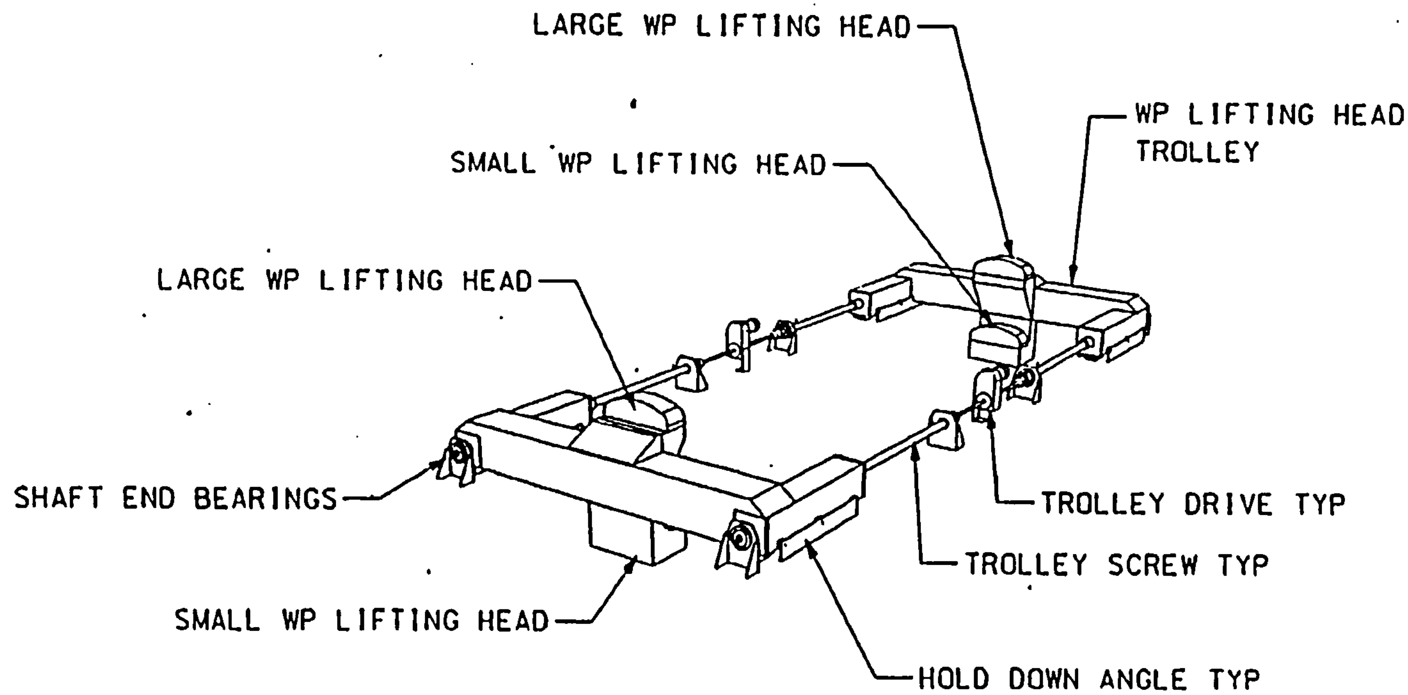

FIGURE 7.4.12
GANTRY TROLLEY
ASSEMBLY

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pairs enabling the lifting head trolleys to move in or out in unison to adjust to the length of the WP. The ball screws are sized to withstand the acceleration load of a seismic event and are powered by one-third hp gear motors having an output speed of 188 rpm. Speed of the lifting head trolleys will be 4.77 m/min. The hoisting frame will be fabricated similar to the Gantry structure with A36 built-up structural shapes welded in accordance with AWS standards. Calculations for the trolley ball screws are presented in Attachment V.

Once the WP is engaged by the lifting head trollies, the WP is lifted off the Reusable Rail Car. Lifting screws, located inside of the vertical C-section columns at each corner of the hoisting frame, provide the vertical lift. The ball screws are supported at both ends by rigidly mounted angular contact bearings and are chain-driven by electric gearmotors mounted on the top cross-member of the Gantry. This base case drive option is shown in Figure 7.4.1 and 7.4.11. The gearmotors are right-angle foot-mounted units, providing an output speed of 28 rpm with a rating of 7.5 hp. The selected lifting ball screws have a 100 mm diameter and a 25.4 mm per revolution lead. A maximum vertical lift of 2226 mm is required to be able to lift a 2.0 meter diameter WP over another 2.0 meter WP. This design will allow the WP to be elevated to its maximum height in approximately 3 minutes and, during lifting of the WP, an integral motor brake included in the traversing motors will ensure that movement of the Gantry cannot occur. During traversing of the Gantry, solenoid operated locks placed in the C-section columns will engage the hoisting frame to support the load and minimize flexing of the ball screws. Calculations for the ball screw selection are presented in Attachment V. Future studies will evaluate and refine the design of other options for supporting the load during traversing of the Gantry.

7.4.3.6 Hoisting Frame Drive Options

Other options for driving the ball screws were considered. They include direct drive (Figure 7.4.13) and individual chain drives (Figure 7.4.14). As indicated in Figure 7.4.13, direct drives infringe into the required clearance envelope for a 5.5 m diameter drift; therefore, this option is not suitable. Individual chain drives are a suitable option; however, these drives would have to be mounted where the electrical boxes are now located as shown in Figure 7.4.1. For these reasons, the design as depicted in Figure 7.4.1, is considered the best choice.

7.4.3.7 Lifting Head and Trolley

A preliminary design evaluation has confirmed that the method to lift each WP by its two end skirts (Ref. 5.13) is acceptable when using two lifting heads/fixtures (Ref. 5.15). The functions of the lifting head trolley are to provide an adjustment for the variations in length of the WPs, to engage the recessed ends of the WP, and to provide a structure for lifting the WP. The trolley is carried on the top flange of the longitudinal beams of the hoisting frame and, as described earlier, is positioned by the trolley screws, which are also carried on the hoisting frame (Figure 7.4.11). Movement of the trolley on the top flange of the longitudinal beams is facilitated by the use of flat rollers. The trolley is secured to the hoisting frame by sliding plates welded or bolted to the hoisting and trolley frames. Each trolley is carried by six rollers each having a capacity of 7.25 MT. Calculations supporting the selection of the trolley rollers are presented in Attachment V.

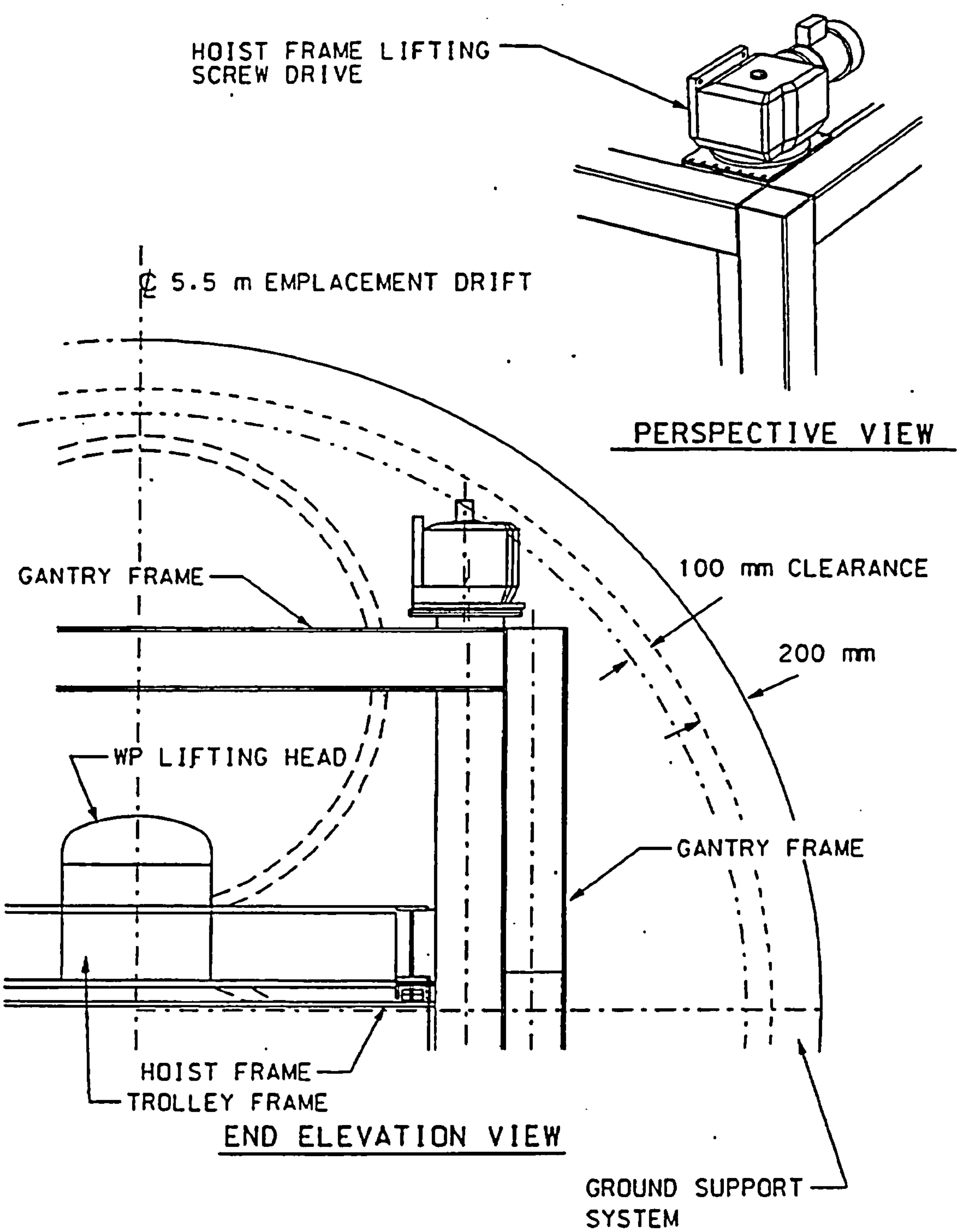

FIGURE 7.4.13
LIFTING SCREW DRIVE
INDIVIDUAL DIRECT DRIVE

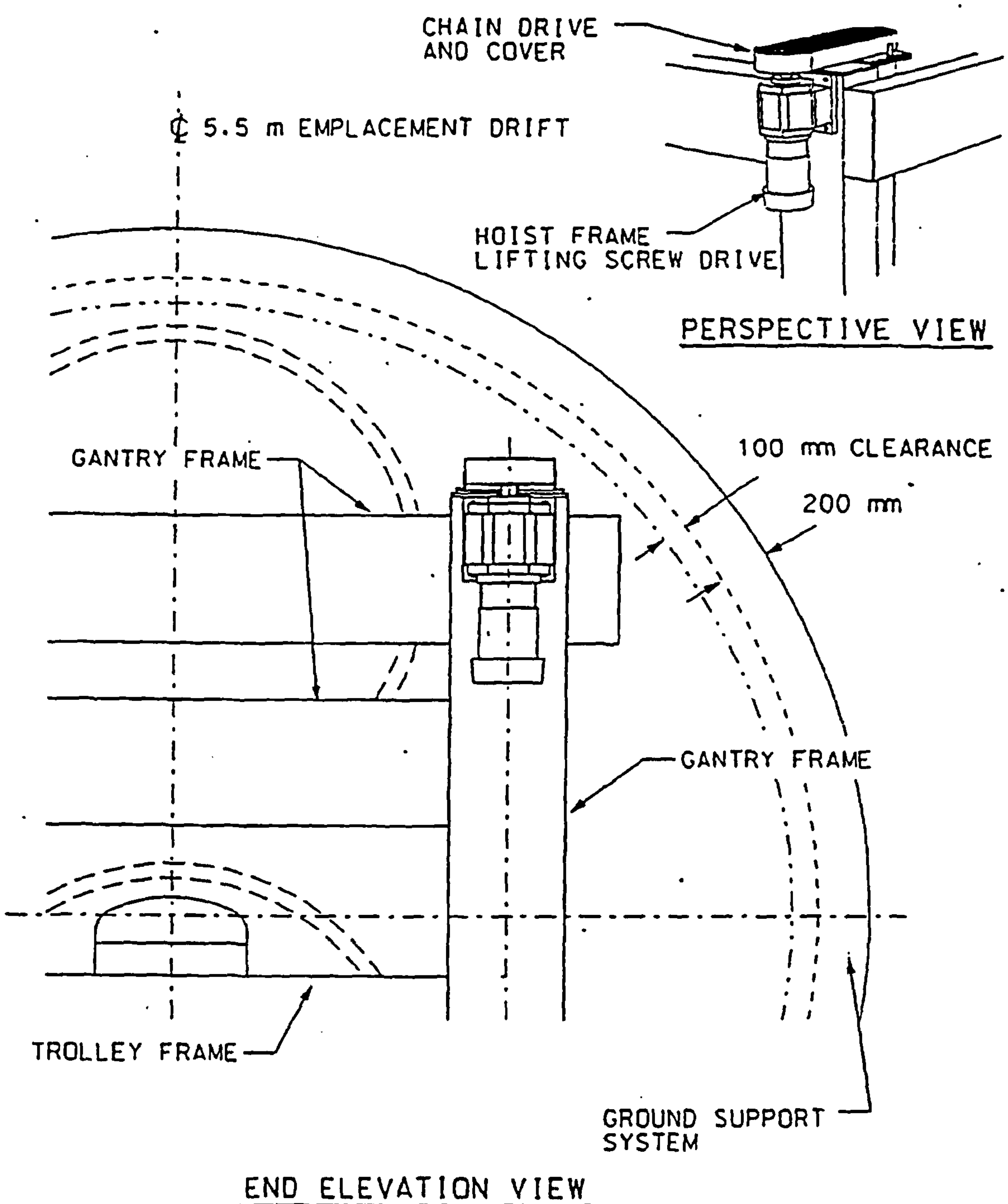

FIGURE 7.4.14
LIFTING SCREW DRIVE
INDIVIDUAL CHAIN DRIVE

7.4.3.8 Gantry Power

Electrical power to operate the Gantry is the subject of a separate design analysis. That analysis calls for the power to be supplied from a third rail installed on the drift invert. Power will be supplied through two spring loaded brush-type contacts which slide along the rail as the Gantry moves within the Emplacement Drift. These contacts will be located approximately 1.0 m inside of each bogie so that if continuity is interrupted on one contact the other will continue to supply the Gantry. Electric power from the contactors will feed into a power distribution panel on the Gantry.

7.4.3.9 Gantry Controls and Communications

The control system for the Emplacement Gantry is the subject of a separate analysis now in progress. The current concept calls for the use of a set of redundant on-board programmable logic controllers (PLCs) and control computers. The control system will control and monitor all vital on-board operations and functions such as vehicle locomotion, speed, acceleration, braking, and positioning. It will also control and monitor WP hoist and lifting head drive motors, limit switches, and operate load locking/latching devices. In addition, the control system will operate and interface to on-board camera and lighting systems, thermal monitoring and control systems, radiation monitoring systems, power supply and distribution systems, and remote communication systems.

The present Gantry design incorporates four radiation shielded electrical enclosures (as shown in Figure 7.4.1). Each enclosure is constructed of a minimum thickness of 51 mm of carbon steel (Ref. 5.36, page 78).

The current design concept calls for the Emplacement Gantry to remotely communicate with operators who are located at a control station on the surface. The operators will be linked to the Gantry by a subsurface communications network. The network will consist of a fiber-optic communication throughout the main and perimeter drifts and will provide for wireless remote control within the Emplacement Drifts. Communication technologies currently being considered for use inside the Emplacement Drifts include direct radio via a distributed antenna system, radio control via a leaky feeder cable system, or the use of slotted microwave guide technology. These will be addressed in a future analysis on the Emplacement System Controls and Communications.

7.4.4 Gantry Interfaces

The Emplacement Gantry must interface with the following equipment items or systems:

- Cross cut drift envelope
- Reusable Rail Car
- WP
- Emplacement Drift Transfer Dock
- Emplacement Drift envelope
- Emplacement Drift Gantry rails
- Emplacement Drift WP pedestals

- Emplacement Drift Gantry third rail electric power
- Remote control system
- Gantry Carrier
- Gantry Carrier third rail

7.4.5 Gantry Structural Analysis

A comprehensive structural analysis was performed on the Gantry structure using STAAD-III/IDS, a computer software with specific application for analyzing and designing steel frames and accessories. The level of this analysis was considered appropriate since the verification of the Gantry structure and resulting envelope is a major factor in determining the minimum size of the Emplacement Drift diameter, Isolation Door, and Gantry rail, along with other supporting interfaces, such as power.

The structural analysis for the Gantry is presented in Attachment II and includes a design basis that was generated for this equipment. Input for the analysis is identified and described in the input sections of this analysis. The results of this analysis are reflected in the structural member size and arrangement of the Gantry design.

7.5 GANTRY CARRIER

7.5.1 Attachment Reference

Attachment VI - Gantry Carrier Weight and Wheel Selection
Attachment VIII - Transporter Truck Arrangement and Weight Analysis

7.5.2 Functional Requirements

The purpose of the Gantry Carrier is to transport the Gantry from the surface storage facility to the underground drifts and from drift to drift during normal operation of the repository. Due to the reduced operating weight of the Carrier, only one of the Transport Locomotives will be required.

7.5.3 Carrier Subsystems

The Carrier will be similar to a railroad flat car with 44.6 kg/m (90-lb/yd) ASCE rails mounted on the Carrier bed to accommodate the Gantry. A third power rail will also be mounted on the Carrier to supply power to the Gantry during loading and unloading operations. The Gantry will utilize the same remote control for loading and unloading as used for the WP placement operation. The Gantry will be secured to the Carrier with a spring set clamp. The track gage for the rails will be 2.58 m, which matches the drift invert rail gage. Dimensions of the Carrier flat bed are as shown in Figure 7.5.3. The Carrier arrangement is shown in Figures 7.5.1 through 7.5.4. Descriptions of the Carrier components are as follows:

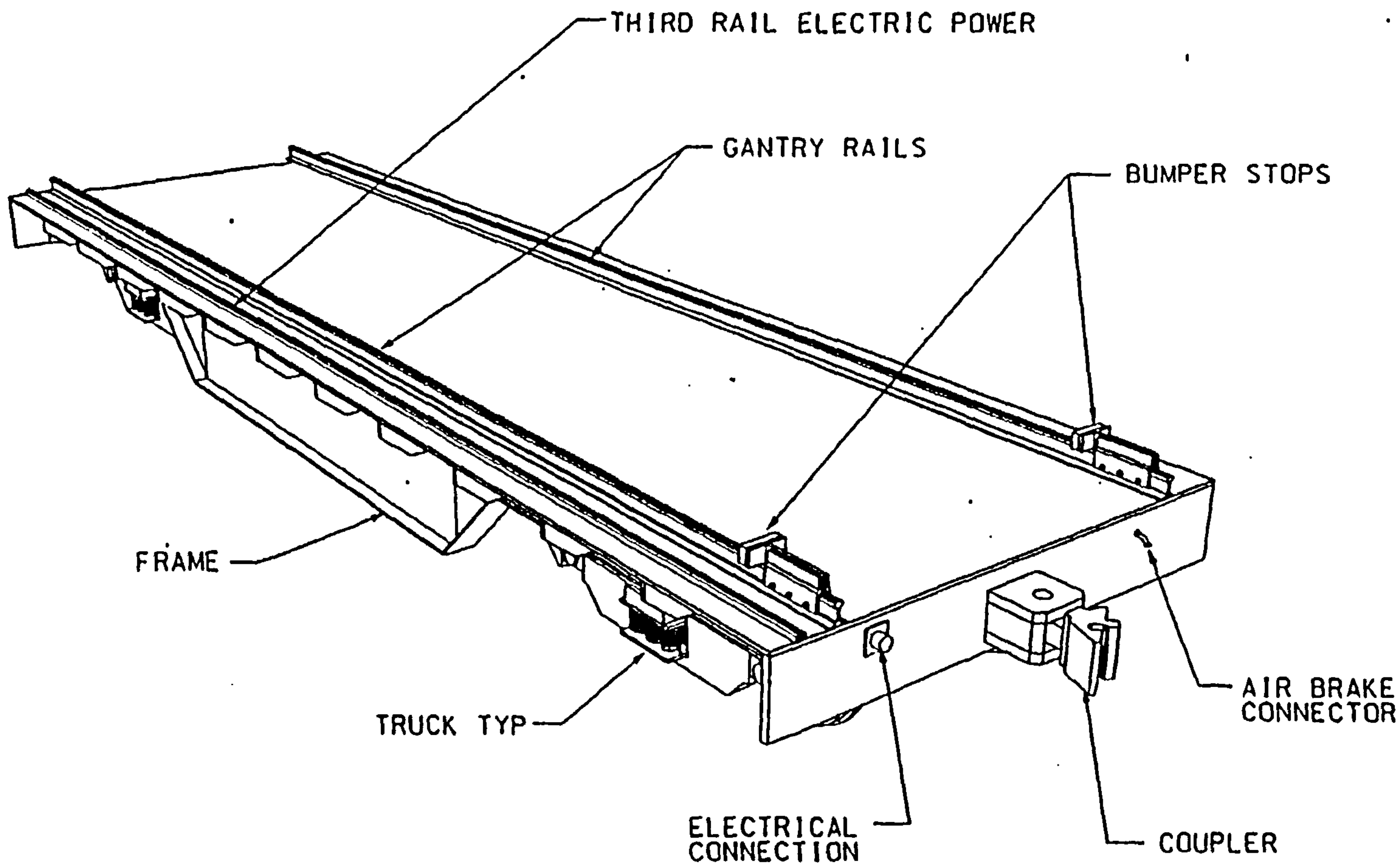

FIGURE 7.5.1
GANTRY CARRIER
ARRANGEMENT

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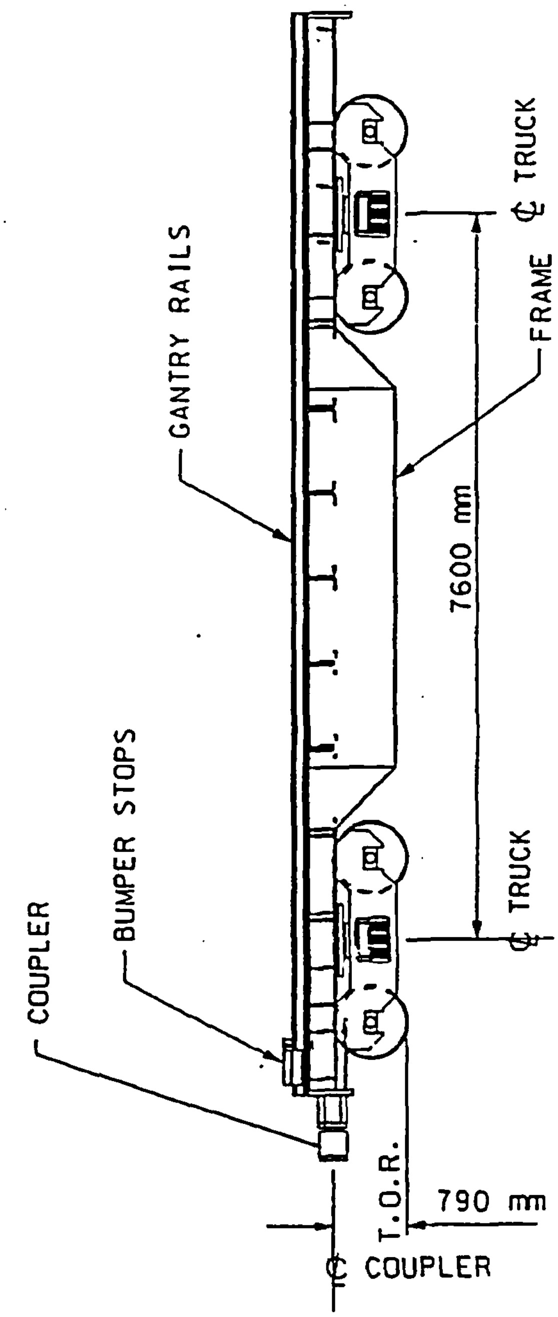

FIGURE 7.5.2
GANTRY CARRIER
ELEVATION

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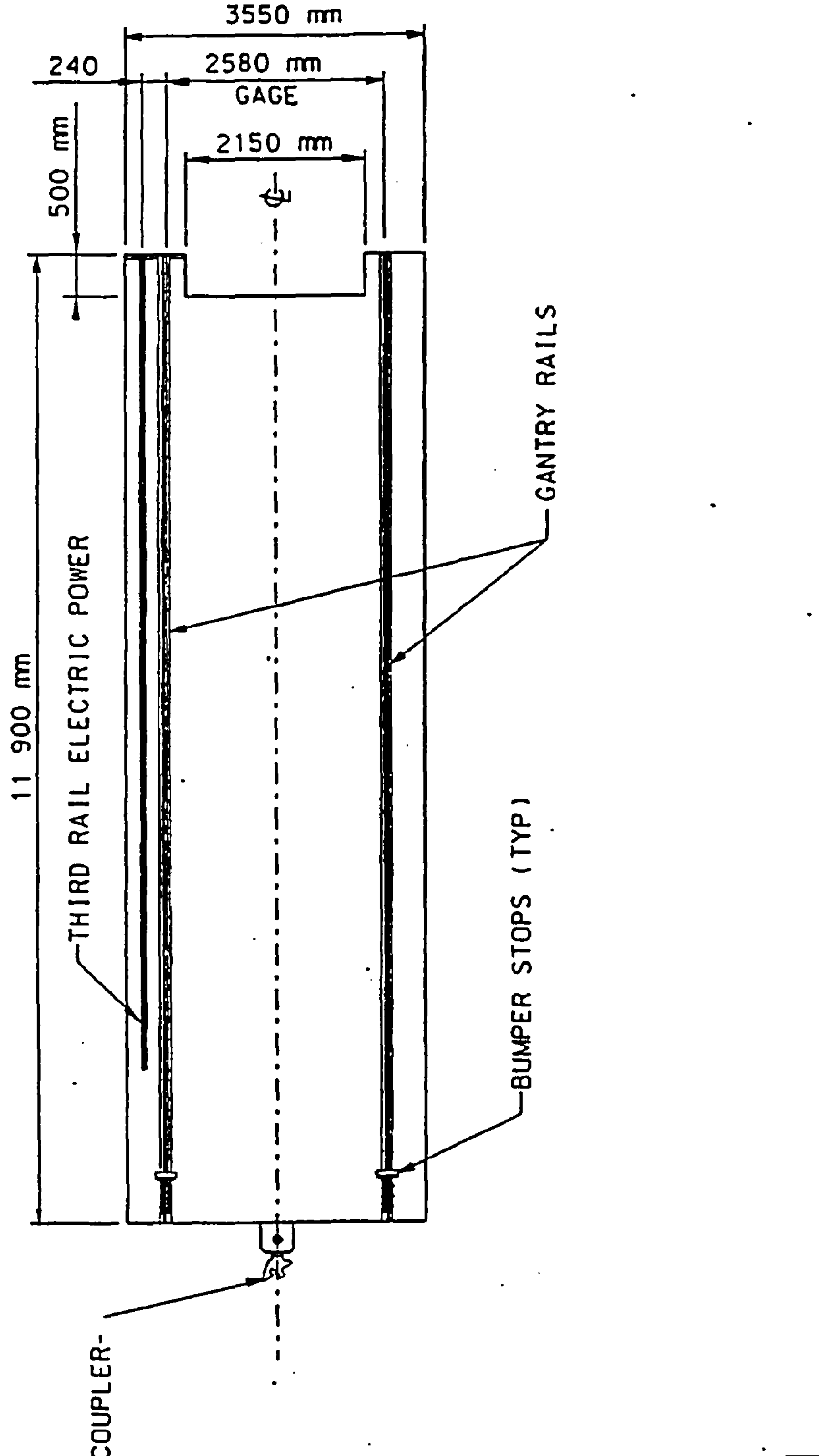

FIGURE 7.5.3
GANTRY CARRIER
PLAN VIEW

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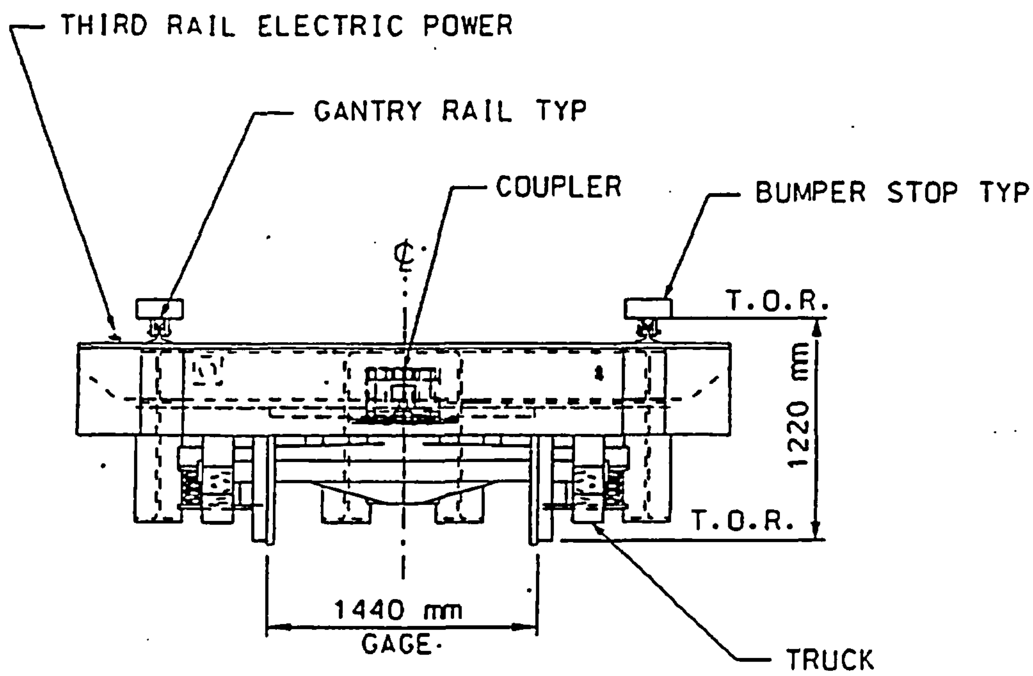


FIGURE 7.5.4
GANTRY CARRIER
END ELEVATION

7.5.3.1 Underframe

The underframe structure will be a standard railroad flat car design with horizontal main stringers running the full length of the car near its midpoint. The stringers will support the bolster plates and bolsters at each end of the car to provide the connections to each undercarriage. The stringer also provides an anchor point for the coupler located at the end of the car. Outside stringers located at the outside edge of the car will be attached to the main stringers with crossbeams, spaced at close intervals and running at 90 degrees to the main stringers. These will be attached to the rails and will support the weight of the Gantry. The underframe is fabricated of structural sections and plate with welded or bolted connections.

7.5.3.2 Undercarriage (Trucks and Wheels)

To provide uniformity between emplacement equipment items the Carrier trucks will be the same as those used on the Transporter. Refer to Attachment VI for wheel selection and rail verification.

7.5.3.3 Couplers

The Gantry Carrier interfaces with one of the Transport Locomotives through a coupler provided at one end of the Carrier. The coupler will be the same as that provided with the Transporter.

7.5.3.4 Brake System

The Carrier braking system will be the same as the brake system installed on the Transporter. Refer to Section 7.2.3 for a description.

7.5.3.5 Gantry Restraint

Once the Gantry is loaded onto the Gantry Carrier, a restraint is required to secure the Gantry when the Carrier is in motion. The restraint secures the Gantry in all degrees of freedom. The restraint to be used is a spring set/electric or air pressure release fail-safe type, which engages the Gantry at several locations. The restraints shall be either part of the Carrier or Gantry and controlled accordingly. Details of the restraint system will be the subject of a future analysis.

7.5.3.6 Third Rail Gantry Power System

A third rail electric power system installed on the Carrier will provide power for the loading and unloading of the Gantry from the Carrier. This rail will be identical to the system installed in the Emplacement Drifts. Power will be supplied to the third rail from the Locomotive through an electrical cable connection to the Carrier. The Gantry has a brush contact installed approximately 1.0 m from each end so that power is continually supplied to the traversing drive controls during loading and unloading operations.

7.5.4 Gantry Options

Figures 7.5.5 and 7.5.6 depict the two Gantry options (carry-over and no carry-over capability) loaded on the Gantry Carrier.

7.5.5 Interfaces

The Gantry Carrier will interface with the following equipment or systems:

- Transport Locomotive
  1. Air brake connection
  2. Gantry third rail power connection
  3. Gantry restraint power and control connection
- Emplacement Drift Transfer Dock
- Emplacement Gantry
- Rail and switch systems
  1. North Ramp
  2. Main Drift
  3. Drift Turnout
  4. Emplacement Drift Gantry rail

7.5.6 Structural Analysis

No structural analysis was performed for the Gantry Carrier. The sizing of materials and the carrier configurations shown are based on a mechanical evaluation of envelope limits and WP sizes.

7.6 TRANSPORT LOCOMOTIVE

7.6.1 Attachment Reference

Attachment VII - Transport Locomotive - Equipment Selection

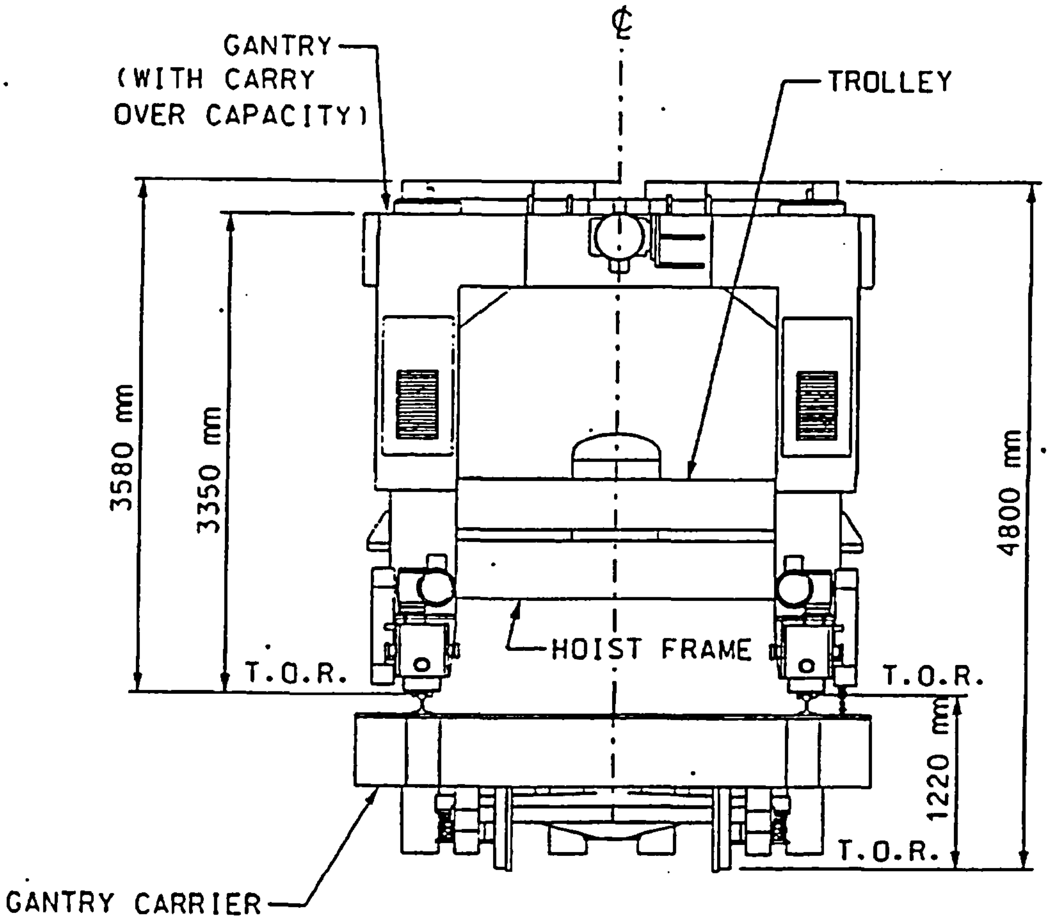
7.6.2 Functional Requirements

The purpose of the Locomotives is to move the WP Transporter from the WHB to the Emplacement Drifts. A Locomotive is also required to transport the Emplacement Gantry to and from the Emplacement Drifts. The arrangement of the Locomotive is shown in Figures 7.6.1 through 7.6.4.

7.6.3 Locomotive Subsystems

7.6.3.1 Overhead Trolley Power Connector System

The Locomotive shall receive 600 VDC single conductor power through a pantograph mounted on the Locomotive (Ref. 5.40).

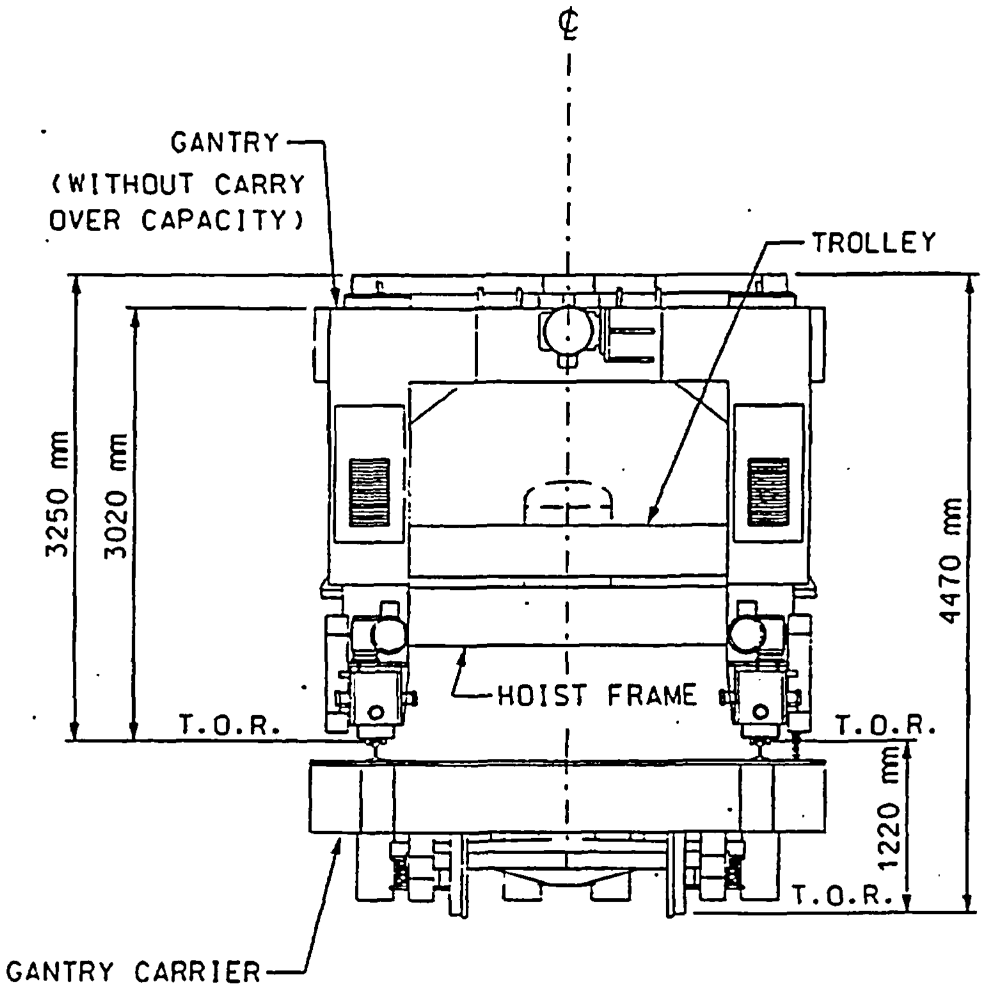


END VIEW

HOIST FRAME SHOWN IN LOWERED POSITION

NOTE: THIS CONCEPT ALSO ASSUMES A SHADOW SHIELD IN USE

FIGURE 7.5.5
GANTRY CARRIER AND
GANTRY WITH CARRYOVER



END VIEW

HOIST FRAME SHOWN IN LOWERED POSITION

NOTE: THIS CONCEPT ASSUMES NO SHADOW SHIELD IN USE

FIGURE 7.5.6
GANTRY CARRIER AND
GANTRY WITHOUT CARRYOVER

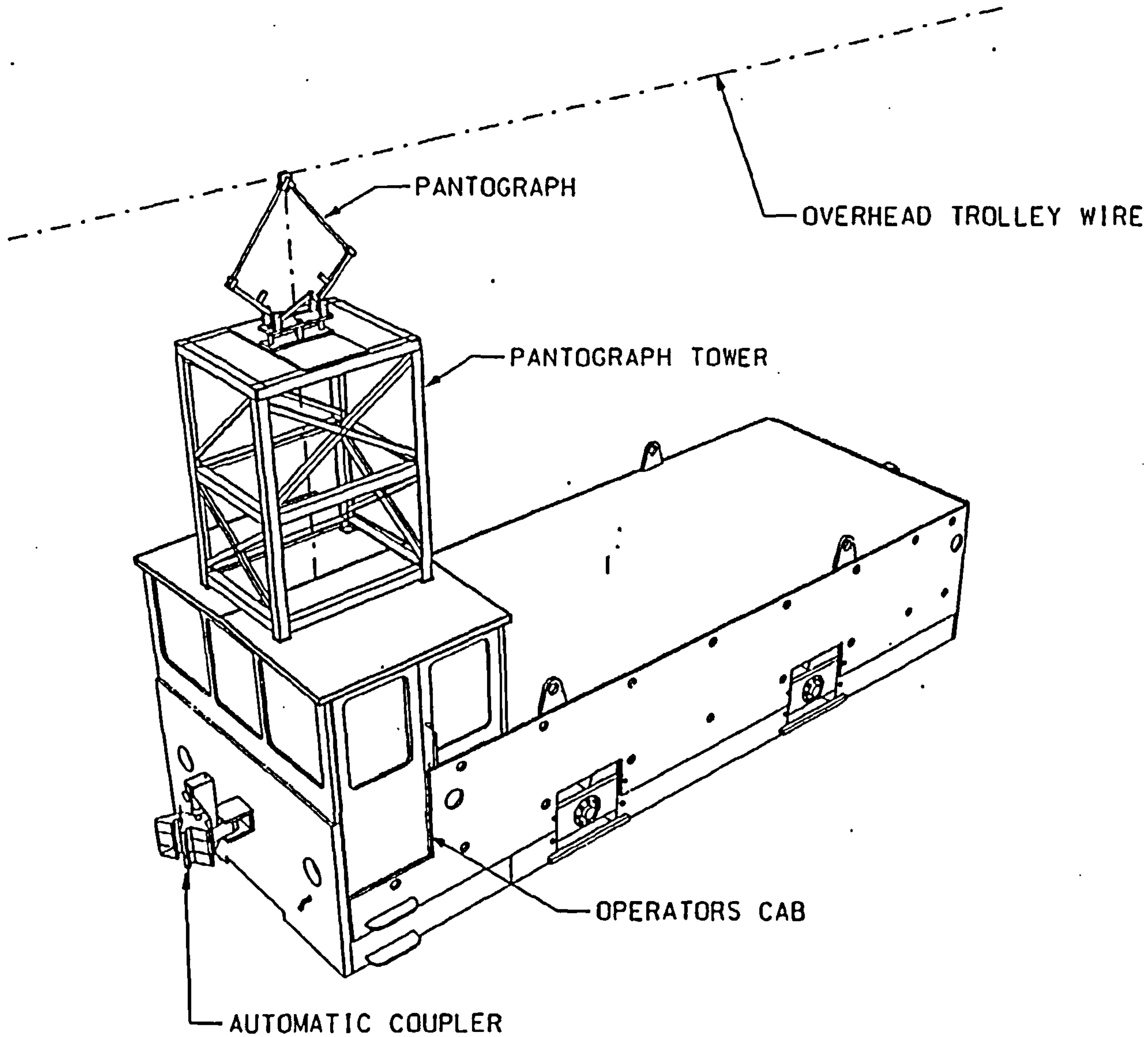


FIGURE 7.6.1
TRANSPORT LOCOMOTIVE ARRANGEMENT

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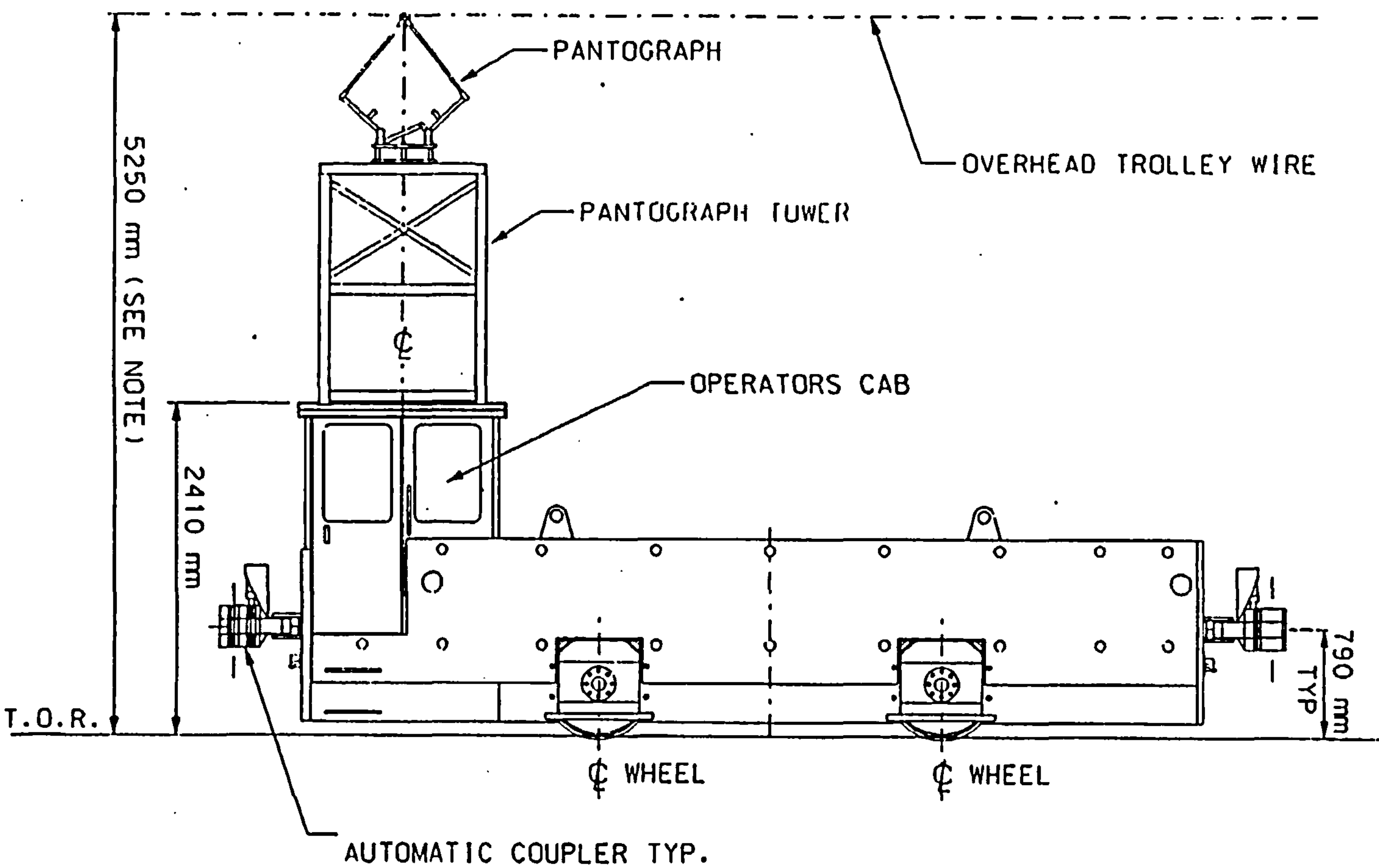

NOTE: TROLLEY WIRE HEIGHT SHOWN IS FOR CLEARANCE ABOVE CARRY OVER GANTRY ON GANTRY CARRIER

FIGURE 7.6.2
TRANSPORT LOCOMOTIVE
ELEVATION

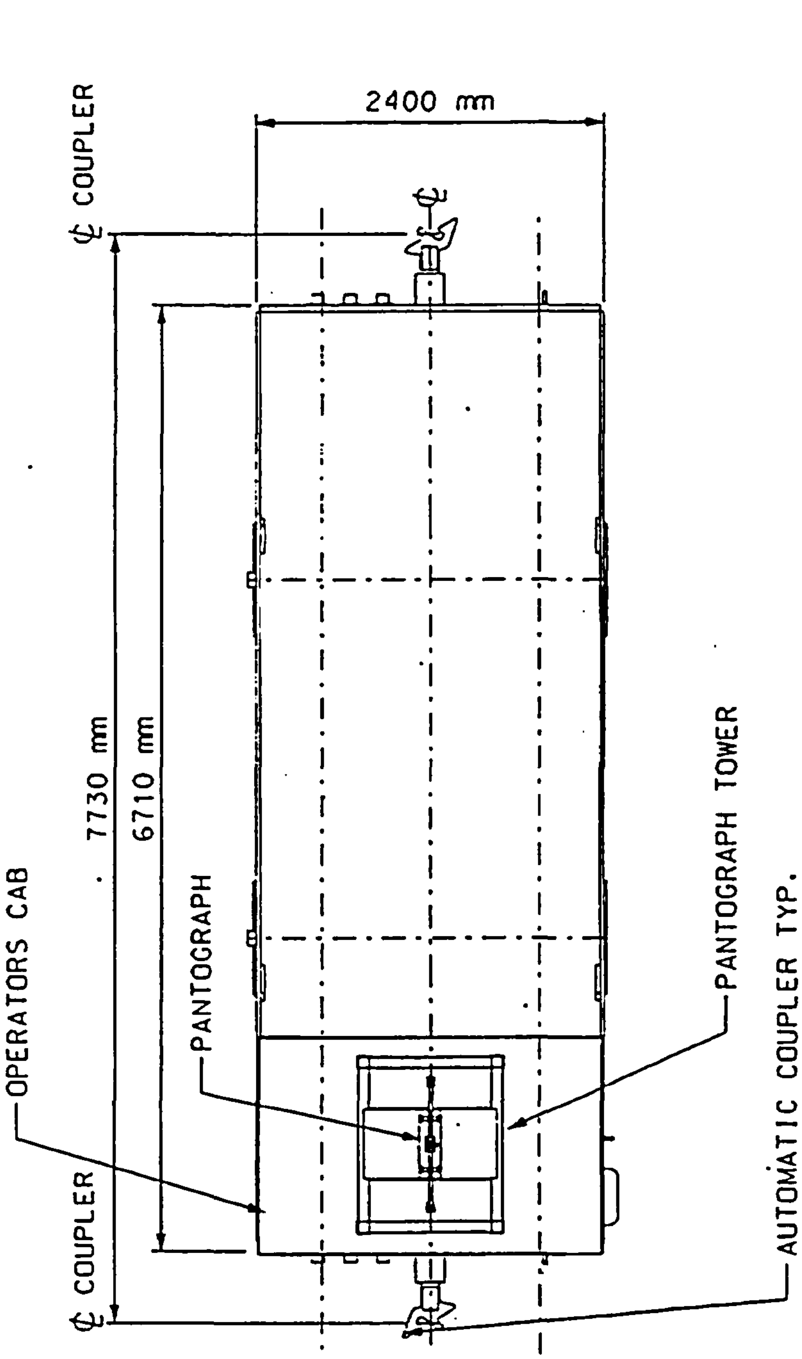

FIGURE 7.6.3
TRANSPORT LOCOMOTIVE
PLAN VIEW

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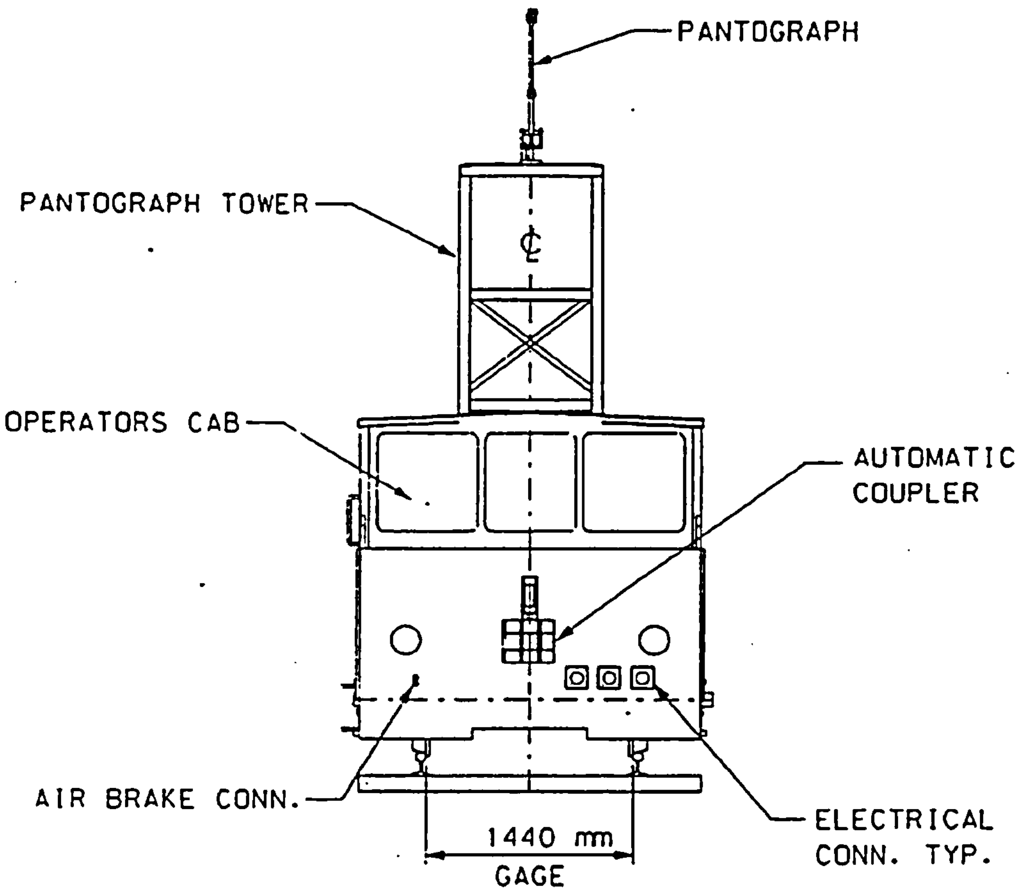


FIGURE 7.6.4
TRANSPORT LOCOMOTIVE
END VIEW

7.6.3.2 Tractive Power System

The tractive power system consists of 600 VDC motors, transmissions, and wheel sets with axles, bearings, and load springs. This system provides the horsepower necessary for the running tractive effort plus the reserve power for adequate acceleration.

7.6.3.3 Locomotive Frame

The Locomotive frame provides the structural integrity required for operator safety, component alignment, and proper transfer of drawbar trailing loads to the wheels.

7.6.3.4 Couplers

Couplers on the Locomotives are Willison type, compatible with couplers on the Transporter and Gantry Carrier. They transfer the drawbar pull required to move the loads. Refer to Section 7.2 and Attachment IX for a description of the coupler.

7.6.3.5 Brake System

The brake system will be an automatic (fail safe) air brake system which applies the brakes by releasing air from the system. The advantage of this type of system is that should the air line break or leak the brakes will be set automatically, stopping the train. The brake system will function as a speed control device during downhill transport of WPs, as well as a stopping device. The Primary Locomotive controls the brakes on the Transporter and the Gantry Carrier through the air brake connection between the two. In the case of Transporter application, the brakes of the Secondary Locomotive shall be remotely controlled from the Primary Locomotive. Details of the Locomotive remote control will be the subject of a subsequent analysis.

7.6.3.6 Remote Control Features

The Locomotives will be either manually controlled or remotely controlled when conditions prevent on-board operators. In addition to the remote Locomotive control equipment, the Primary Locomotive, which is always connected to the Transporter, will also include the controls for the Transporter functions. Control and power for the Transporter functions will be through cable connections between the Transporter and Primary Locomotive. Details of the Locomotive and Transporter controls and instrumentation shall be the subject of a subsequent analysis.

Loading and unloading of the WPs from the Transporter along with the shielded door operation will be included in the operational functions of the Locomotives remote controls.

7.6.4 Locomotive Capacities

Locomotive capacity was analyzed (Attachment VII) with regard to the following operating conditions:

- Two Locomotives retrieving a WP loaded in a WP Transporter up the North Ramp.
- One Locomotive moving a loaded WP Transporter at the Emplacement Transfer Dock
  1. Case A - Operation in Main Drift with 1.4% grade.
  2. Case B - Operation in Drift Turnout with 20 m radius curve and 0.5% grade.
- Locomotives moving a loaded WP Transporter down the North Ramp.
  1. Case A - Two Locomotives moving a loaded WP Transporter.
  2. Case B - One Locomotive moving a loaded WP Transporter.

The result of this analysis indicated that the one Locomotive operation placing the WP Transporter at the Emplacement Drift Transfer Dock requires the maximum capacity. The largest contribution to the 45 ton capacity requirement was the turnout with 20 m radius. A specially designed Goodman Type 201 Locomotive was selected to perform this duty (Refer to Figure 7.6.1). This Locomotive has 30 in. diameter wheels on 100 in. centers and can negotiate the 20 m curve in the turnout. The Type 201 is normally used as a 20 to 25 ton trolley or battery operated Locomotive with a 914 m (36 in.) rail gage. The standard design would be modified with larger motors, additional weight, and an increased width to accommodate the 1441 mm (56½ in.) rail gage. An alternate Locomotive, as shown in Attachment XI, is a standard 35 ton Locomotive utilizing a two truck system and eight driving wheels. It could also be modified to fit this application and will be considered in future studies.

The selection basis for this analysis is a Goodman Model 201 trolley-powered mining or tunneling Locomotive which from Goodman General Assembly Drawing No. 201914-000 has the following dimensions which are used in Figures 7.6.2 through 7.6.4:

Overall length:	6706 mm (264.0 in.)
Coupler centerlines:	7728 mm (304.25 in.)
Top of rail to top of cab:	2410 mm (95.06 in.)

This drawing shows only an overall width to walkways which extend out from each side of the Locomotive. Another source was used for the width of the Locomotive frame. From Attachment XII, *Goodman Data Sheet for Coal Mining and Tunneling Locomotives*, a Type 201 trolley-powered Locomotive has an overall width of 1778 mm (70 in.) for a 914 mm (36 in.) track gage.

To accommodate the greater track gage of 1440 mm (56½ in.) as identified above, the Locomotive width was adjusted as follows:

Increase in track gage = 1440 m - 914 m = +526 mm

Assuming that Locomotive width would increase by the same amount,

Estimated width = 526 mm + 1778 = 2304 mm

Rounding up, the new estimated Locomotive width used in Figures 7.6.2 through 7.6.4 is 2400 mm.

7.6.5 Locomotive Interfaces

Locomotives will interface with the following pieces of equipment or systems:

- WP Transporter
  1. Mechanical coupling
  2. Air brake connection
  3. Transporter functions power and control
  4. Instrumentation and status connections.
- WHB
- Rail and switch systems
  1. North Ramp
  2. Main Drift
  3. Drift Turnouts
- Electrical, control and instrumentation systems
- Gantry Carrier
  1. Mechanical coupling
  2. Third rail power connection
  3. Air brake connection
  4. Gantry restraint control connection
  5. Interlock and status connections

7.6.6 Structural Analysis

No structural analysis was performed for the Transport Locomotive(s) because locomotives for similar applications and operating in related industries have been built, used and have demonstrated reliability for over many years.

7.7 DRIFT ISOLATION DOORS

7.7.1 Attachment Reference

None.

7.7.2 Functional Requirements

Each Emplacement Drift entrance will require Emplacement Drift Isolation Doors (Ref. 4.3.15). Major functions of the Emplacement Drift Isolation Doors should include the following:

- Control access to the Emplacement Drifts
- Provide for easy transfer of remotely controlled mobile equipment for emplacement, retrieval, performance confirmation, and maintenance functions
- Provide a degree of radiation protection (Ref. 5.17)
- Remote control operation
- Control air flow and air leakage into the drifts

- Provide controlled louvers or dampers for airflow when needed
- Resist varying thermal loadings
- Require low maintenance/high reliability

7.7.3 Arrangement

A swing-type door system will be used as an initial selection. It consists of two door panels, each pivoted about a hinged point. The use of two doors rather than one shall minimize the load on the hinged connection between the door and the supporting door frame. The doors are sized to accommodate the largest piece of equipment loaded and unloaded at the Transfer Dock. As discussed further, each of the two swing doors will have a remotely controlled actuator to open and close the doors, as required, for the Emplacement Drift functions.

7.7.4 Construction

A minimum thickness of 25.4 mm (1 in.) 304 stainless steel has been selected for this analysis (Ref. 5.17). The thickness is significant to personnel radiation protection in the main drifts, when the door is closed. This thickness can be obtained using one single thickness of material or the total of several. The door will be constructed of a channel frame with at least a 12.7 mm (½ in.) 304 stainless steel skin on each side. The frame (an undetermined width at this time) will serve to add rigidity to the door structure, similar to a hollow-core door, and aids in retaining the door shape in varying thermal loading conditions. The hollow interior may be filled with insulation if the door is also required to serve as a thermal barrier. The door supporting frame and bulkhead, which complete the isolation between the Drift Turnout and the Emplacement Drift, may also be of the same construction. The doors will pivot on hinges, which could be machined from steel and slipped in between and attached to the door and support frame outer skin. Final size of door and actuator will be based on loading requirements determined in future analysis.

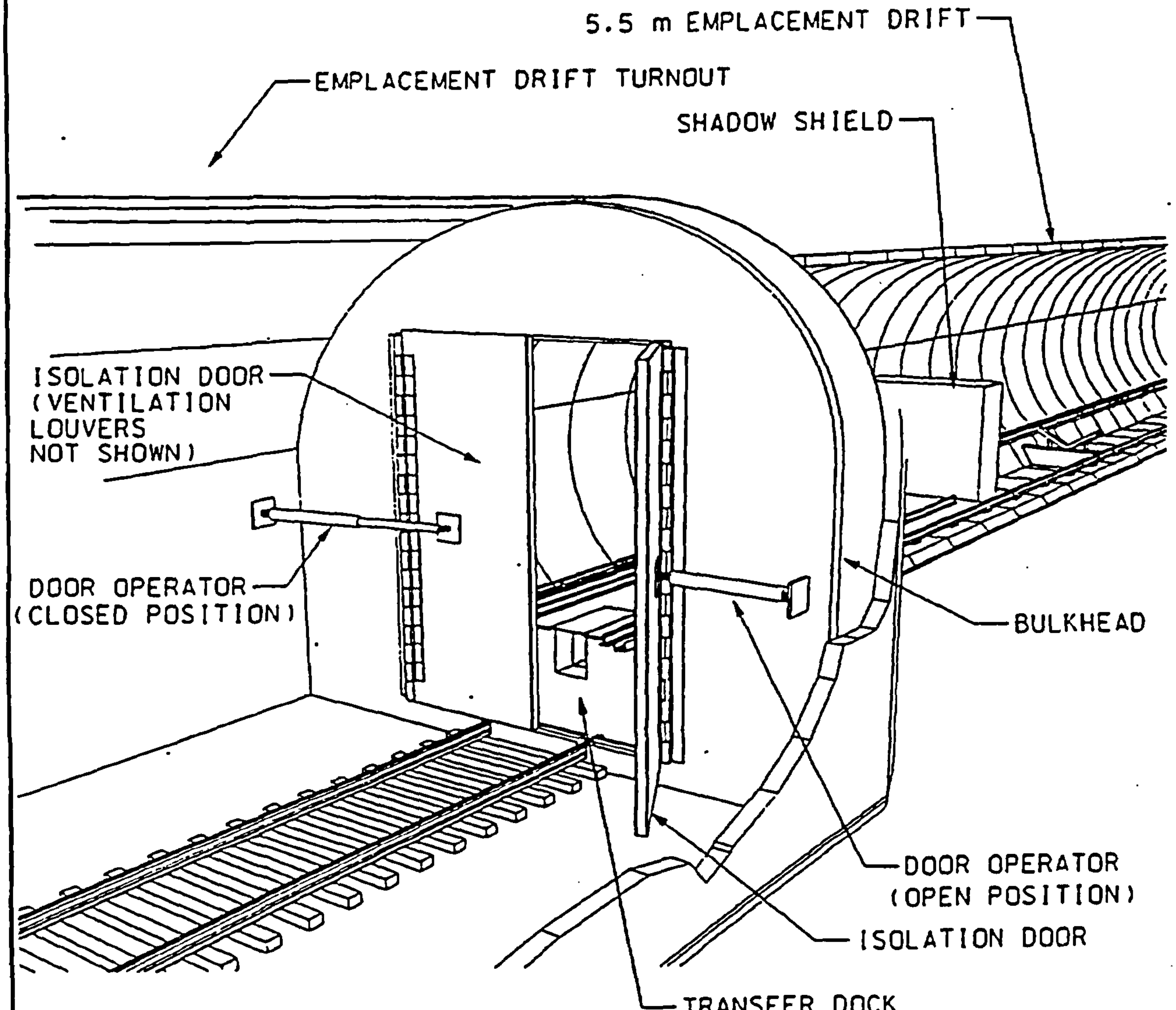
Basic door construction and actuation is addressed in this analysis; however, additional requirements for the door and bulkhead will be defined and addressed in a future analyses (Ref. 5.52).

1. Low leakage air flow seals for doors and door frames that functions under both differential temperature and pressure conditions.
2. Automatically controlled ventilation louvers or dampers in the doors and/or bulkhead.
3. Radiation seals for doors and door frames.
4. Door construction that resists warpage due to differential operating temperatures and pressures.

7.7.5 Door Operators

Three different methods of controlling door movement were considered: hydraulic cylinder, pneumatic cylinder, and a linear actuator or rotary drive, both driven with electric gear motors.

The hydraulic system is a low-pressure design which normally operates at 10.34 mPa. An individual cylinder would be used to activate each door as depicted in Figure 7.7.1. A hydraulic pump driven



NOTE: TUNNEL WALLS SHOWN CUT AWAY

FIGURE 7.7.1
DRIFT ISOLATION DOORS ARRANGEMENT

by an electric motor would provide the fluid pressure. An accumulator would be used to supplement the hydraulic pump. The accumulators could also handle abnormal situations when the pump is out of operation and the doors need one additional movement.

The pneumatic operator would be similar to the hydraulic system with one air cylinder on each door to provide the necessary operating force. Air would be supplied by the facility compressed air system with accumulators used for backup. The accumulators would provide sufficient air pressure for movement in the event that the main supply line failed or the surface-mounted air compressors became inoperative.

An electric linear actuator could be a ball screw type and in the same arrangement as the hydraulic or pneumatic cylinder. The ball screw drive would be a right-angle gear motor and require only electric power and controls.

The rotary drive considered for this application would utilize a right-angle helical worm-gear unit to transfer the rotary motion to the door through the door pin, which is restrained in the door hinge but revolves in the door frame hinge. A gear motor located at the top or bottom of the hinge pin provides the required torque at the door hinge similar to the Transporter door operator (Figure 7.2.6).

The hydraulic design introduces a new system into the overall repository design. Hydraulic pumps and electric power would be required. It is a low-pressure system, which has the potential for leakage through piping or pump seals, which would have to be maintained through a rigorous preventative maintenance. The hydraulic fluid is also a potential source of fire, although flame-resistant fluids are available. Redundant pumps would be required with a backup power system to maintain operation. The system may not be capable of responding to abnormal conditions without additional equipment items. Hence, for this application, the hydraulic system is not recommended.

The rotary drive design, as with the electric linear actuator, is a relatively simple system requiring very little support equipment, such as fluids, pipe, valves, pumps, and accumulators. However, an independent power supply is required to handle emergency or other up-normal conditions. Emergency generators located on the surface could supply backup power, or a small uninterrupted power supply providing power to all door gear motors could be used. The design and maintenance problems associated with locating the gear motors at either end of the pin and the additional provisions that are required in the door hinge construction are reason to reject this design for the Emplacement Drift Isolation Doors. In addition, the final loads due to pressure differentials across the door (determined in a future analysis) may also make this method of operating doors less effective than the use of linear actuators or cylinders acting directly on the door.

The pneumatic system design would meet the requirements for control of the Emplacement Drift Isolation Doors. An air supply of adequate capacity is readily available using redundant air compressors located on the surface. Accumulators also provide backup capability. Operating pressure allows use of standard commercially available pipe and valves. The pneumatic cylinders, compressed air piping, and valves are located on the positive-pressure side of the door, allowing maintenance to be performed without special equipment and procedures. The pressure differential also aids in achieving a good door seal.

In the event that the final design loads on the door exceed the maximum capacity obtained with the largest practical size pneumatic cylinder available, the electric linear actuator would be the alternate consideration. Very high forces may be obtained with that type of actuator, as demonstrated in the use of these types of actuators for the lifting requirements of the Gantry, and they would be as accessible as the pneumatic cylinder for maintenance.

7.7.6 Interfaces

In the scope of this analysis, the Emplacement Drift Isolation Doors interface with the following equipment items or systems:

- Power (depending on type of operation)
- Compressed air (depending on type of operation)
- Instrumentation monitoring and control systems
- Emplacement Drift Transfer Dock
- Emplacement Gantry
- WP Transporter
- Emplacement Drift Air Control

7.8 RAIL SYSTEM

7.8.1 Attachment Reference

Attachment III - WP Transporter - Mechanical Equipment Selection
Attachment IV - Reusable Rail Car - Mechanical Equipment Selections
Attachment V - Gantry Loads and Equipment Selections

7.8.2 Rail Layout

Figure 7.8.1 is a plan view of the track layout in the East Main Drift access, Drift Turnout area, Emplacement Drift Transfer Dock, and Emplacement Drift.

7.8.3 North Ramp Rail

Conventional 57 kg/m (115 lb/yd) AREA rail will be used in the North Ramp. This load capacity is necessary to support the WP Transporter maximum operating weight of 233.15 MT (Attachment III). The load capacity of this rail is based on a 320 BHN hardness and is verified in Attachment III.

7.8.4 Main Access and Drift Turnout Rail

The rail system used in the main access and turnout area, which is subjected to the same loads as the North Ramp, will also use the 57 kg/m (115 lb/yd) AREA rail. The track has both, straight and curved rail sections. Each drift turnout from the main drift to the emplacement drift area will utilize a remotely controlled track switch No. 4.

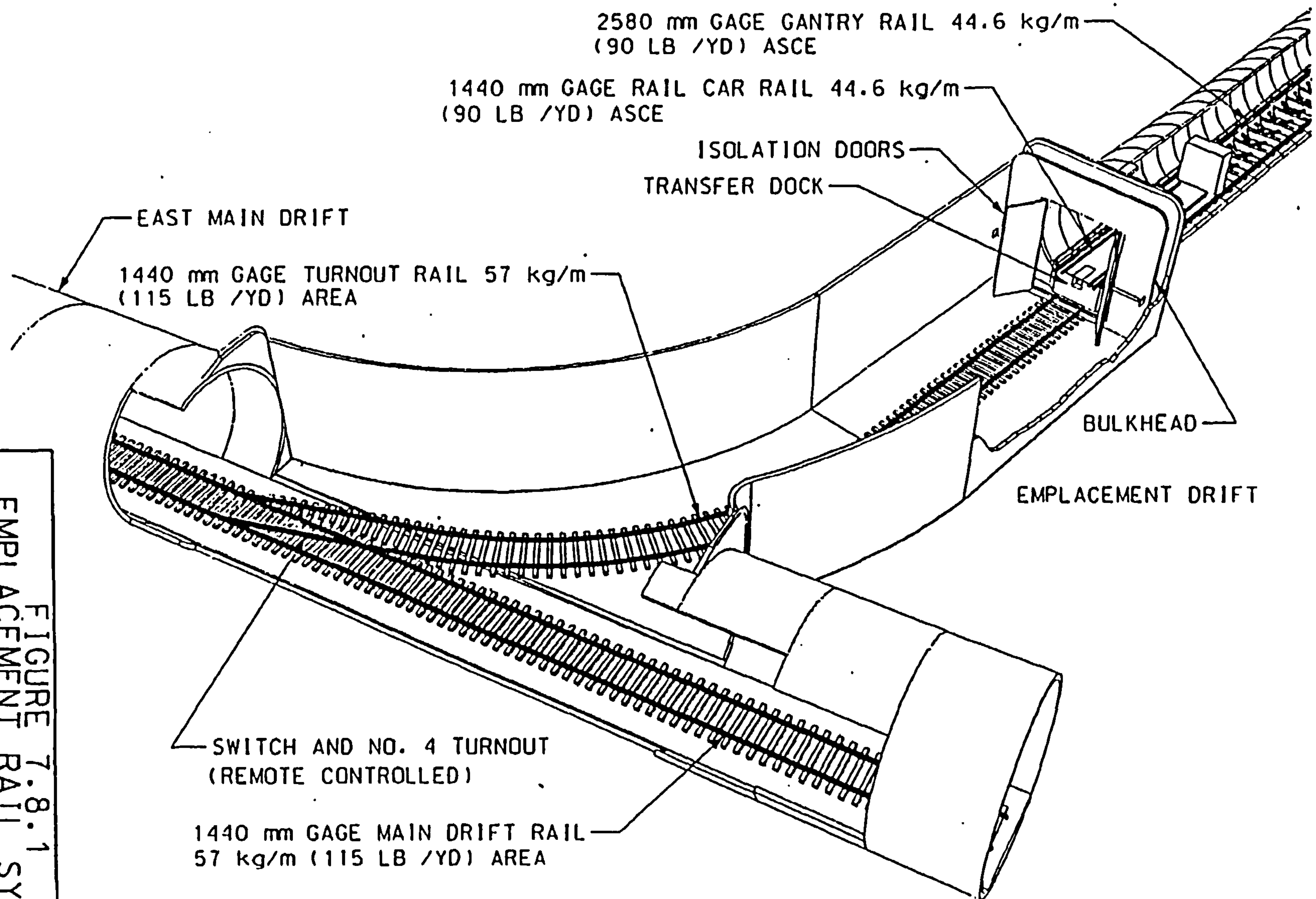


FIGURE 7.8.1
EMPLACEMENT RAIL SYSTEM
TYPICAL TURNOUT DETAIL

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7.8.5 Emplacement Drift Rail

The conventional 44.6 kg/m (90 lb/yd) ASCE rail was selected for the Emplacement Drift because of the lighter weight of the Reusable Rail Car and WP and the Gantry and WP. The capacity of the 44.6 kg/m (90 lb/yd) ASCE rail with a 320 BHN was verified for both the 1.44 m gage application for the Reusable Rail Car (Attachment IV) and the 2.85 m gage application for the Gantry (Attachment V). The Emplacement Drift rail for the Rail Car and Gantry is straight without curves.

7.8.6 Rail System Interfaces

- North Ramp Rail
  - Transporter
  - Locomotives
  - Gantry Carrier
  - Ramp invert
  - Remote rail switch operator
  - Other rolling stock, man trip cars and maintenance equipment
- Main Access and Turnout Rail
  - Transporter
  - Locomotives
  - Gantry Carrier
  - Emplacement Drift invert
  - Remote rail switch operators
  - Emplacement Transfer Dock
  - Emplacement Drift Isolation Door
- Drift Emplacement Rail
  - Reusable Rail Car
  - Gantry
  - Emplacement Drift invert
  - Emplacement Transfer Dock

7.9 EMPLACEMENT DRIFT TRANSFER DOCK

7.9.1 Attachment Reference

None

7.9.2 Functional Requirements

Provide an elevated platform or dock to accommodate the unloading of emplacement rail equipment from Rail Carriers to the Emplacement Drift:

1. Reusable Rail Car with WP from the Transporter

2. Gantry from the Gantry Carrier

In performing the transfer of the emplacement equipment to the Emplacement Drift, the Emplacement Drift Transfer Dock must perform the following subfunctions:

- Provide the required elevation between the top of rail in turnout and the top of rail in Emplacement Drift for the particular transfer (Rail Car or Gantry).
- Edge of Emplacement Drift Transfer Dock to allow for flush contact with the corresponding edge of the Transporter or Gantry Carrier.
- Face of Emplacement Drift Transfer Dock to accommodate any protrusion from Transporter or Gantry Carrier that would prevent a flush contact between the two.
- Provide a means of aligning, supporting, and maintaining the alignment of the Rail Car and Gantry rail and the Rail Car unloader guides both before, during, and after the transfer of the Rail Car or Gantry.
- Provide a means of proving alignment and support before transfer may proceed.

7.9.3 Emplacement Transfer Dock Description

The Emplacement Drift Transfer Dock is shown in Figures 7.9.1 through 7.9.5. The dimensions shown in Figure 7.9.5 are either identified as assumptions in Section 4.3 or as design selections in the attachments. These dimensions have been selected and used for the Gantry analysis and are also shown on the figures in that section. The dimensions and their source is as follows:

- The 0.8 m between the inverts (Section 4.3.13) of the turnout and Emplacement Drifts is shown in the figure. This is a key dimension in the construction of the drifts.

- The 1.0 m dimension from the Emplacement Drift invert to the top of the Gantry rail is from a previous analysis (Section 4.3.23) (Ref 5.38).

- The 1.28 m dimension from the top of the turnout rail to the Rail Car rail is from a previous document (Ref. 5.6), and is a critical dimension used in the Transporter arrangement (see Figure 7.2.2).

- The dimensions 1.22 m (top of Gantry rail to turnout rail) and 1.06 m (top of Rail Car rail to Emplacement Drift invert) are developed from the preceding dimensions.

For Reusable Rail Car unloading (Figure 7.9,3), the rails and the rigid chain drive guide extend over the edge of the dock and line up with the respective rails and guides of the Transporter. At the same time, they are supported on the back edge of the Transporter floor, which is exposed after the Transporter doors are open. The face of the dock has a pocket for the coupler and allows for a flush fit of the Transporter floor. The face of the dock is also relieved to allow clearance from the Transporter door operator motors (Figure 7.9.1).

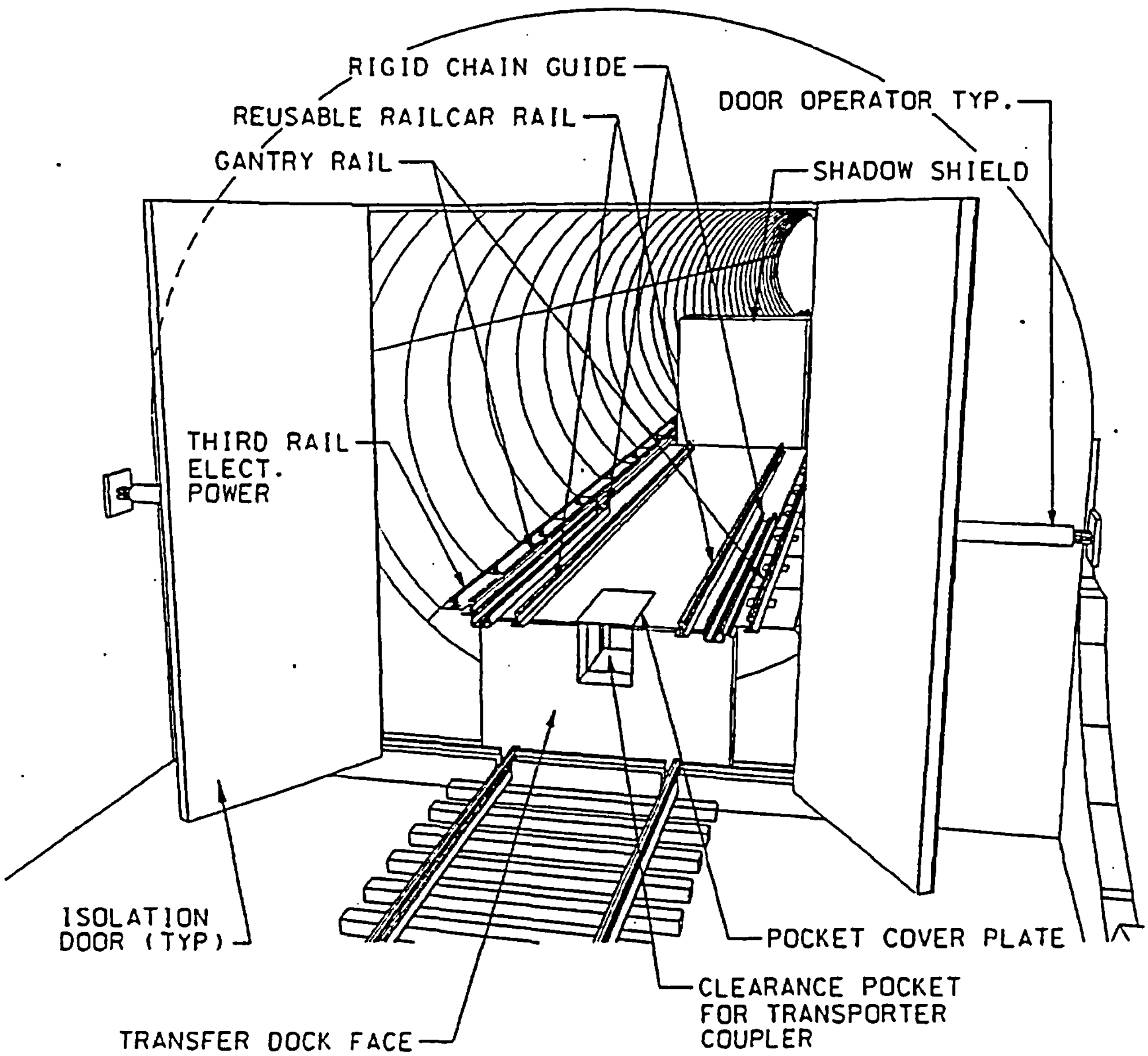

FIGURE 7.9.1
EMPLACEMENT DRIFT
TRANSFER DOCK

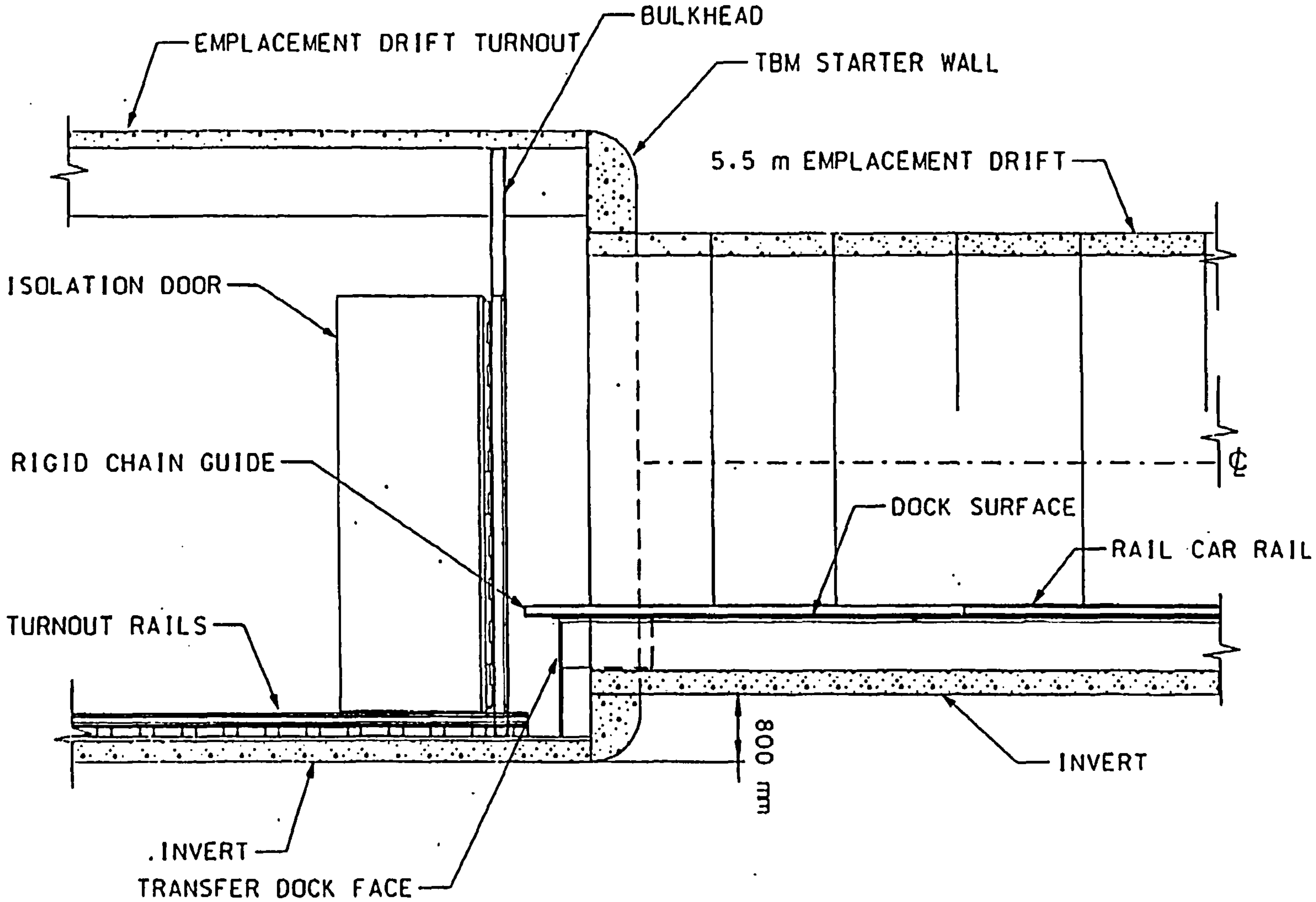

FIGURE 7.9.2
EMPLACEMENT DRIFT
TRANSFER DOCK

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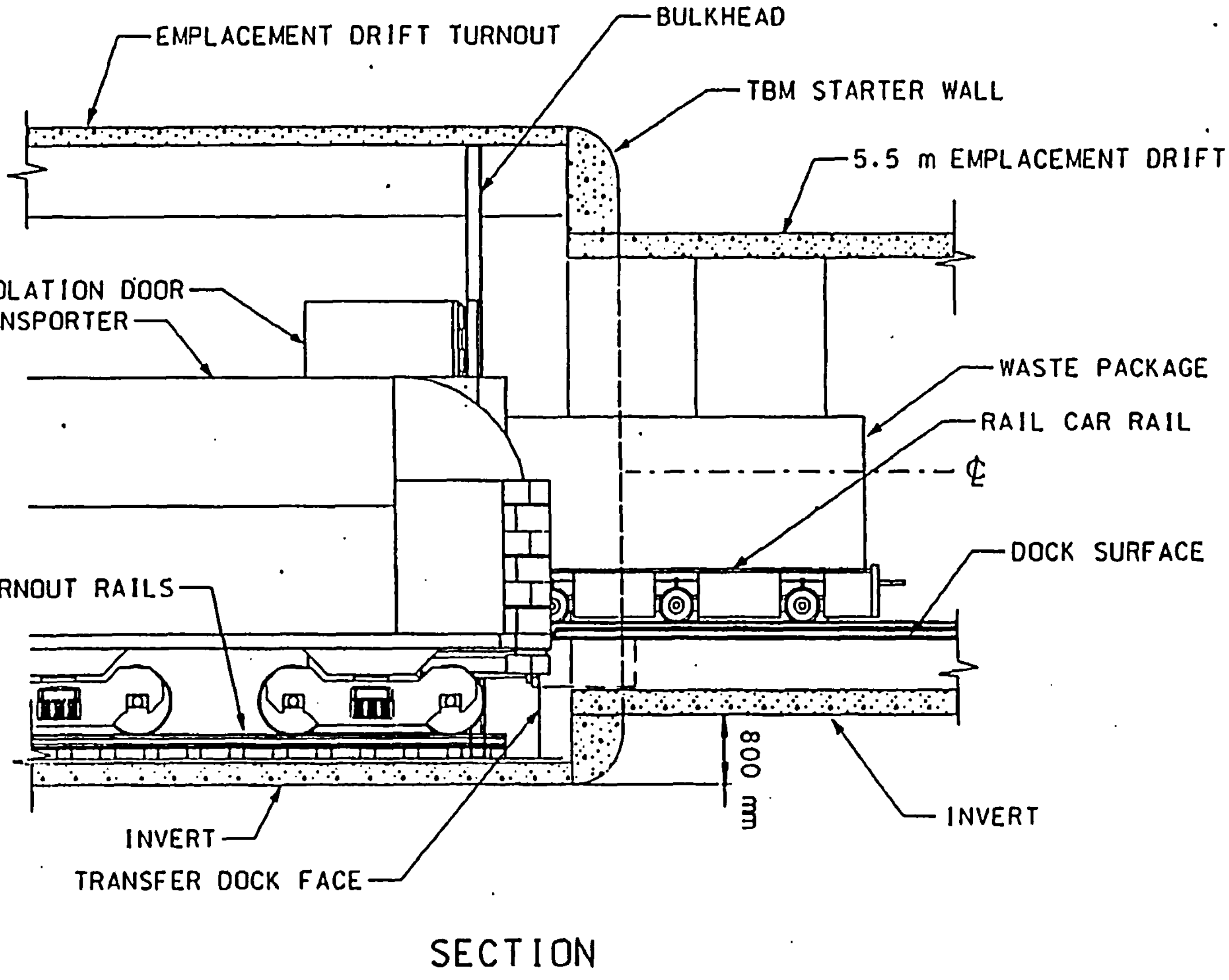

FIGURE 7.9.3
TRANSFER DOCK
RAIL CAR UNLOADING

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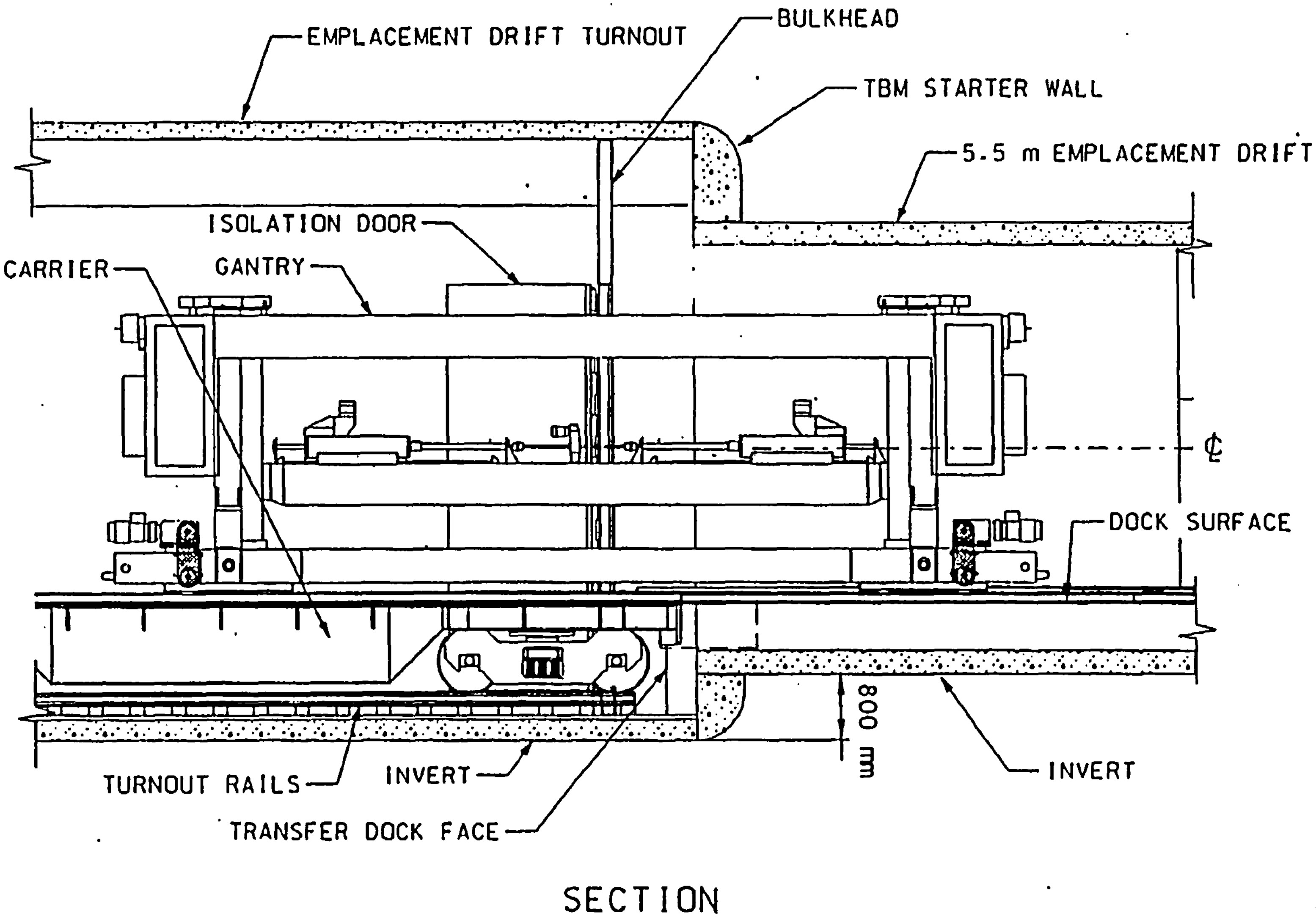

FIGURE 7.9.4
TRANSFER DOCK
GANTRY LOADING

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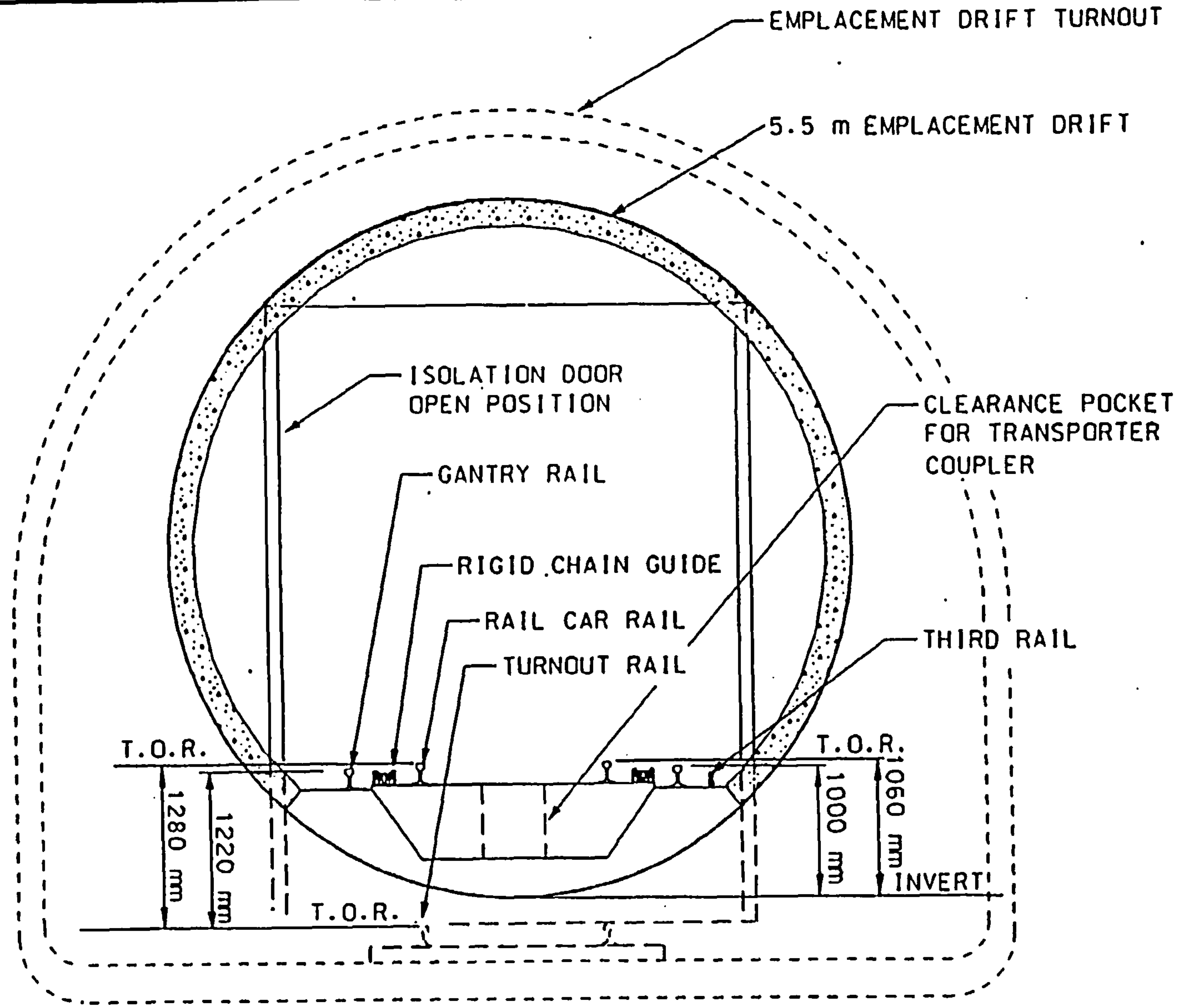

FIGURE 7.9.5
EMPLACEMENT TRANSFER
DOCK SECTION

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The Gantry Carrier is the other piece of mobile equipment that uses the Emplacement Drift Transfer Dock, to load and unload the Emplacement Gantry. As shown in the figure (Figure 7.9.1), there are rails provided for the Gantry, which are flush with the end of the dock and butt up to the Carrier rails for loading and unloading (Figure 7.9.4). The third rail for Gantry electrification is not required to butt up with the respective Carrier third rail since the Gantry has two power contactors, which allow it to always draw power from one or the other of the rails.

Three major functions of the Emplacement Drift Transfer Dock required to successfully effect the transfer operations, listed below, have not been evaluated in this analysis:

1. How the alignment of rails and guides shall be accomplished and verified.
2. How the alignment shall be maintained as the load shifts from the Transporter or Carrier in the loading or unloading operation.
3. How the release of the Transporter or Carrier from the alignment/supporting device may be safely controlled.

The designs of devices or systems to perform these functions will be the subject of further analysis.

7.9.4 Interfaces

- Transporter
- Gantry Carrier
- Drift Emplacement rails
- Reusable Rail Car
- Gantry
- Reusable Rail Car unloader
- Emplacement Drift Isolation Doors

7.10 CONTROL

All control systems for the WP Transport and emplacement equipment need to have remote control capability and interface to a higher area network data acquisition system, such as the main repository control system. The control systems for the mobile emplacement equipment will be PLC based. PLC systems are robust microprocessors specifically designed to process large amounts of discrete input/output (I/O). PLCs were originally designed to replace conventional relay-based logic systems. The advantages of a PLC-based control system are that they are easily programmed, highly reliable, flexible, small, and relatively inexpensive. It is also possible to design redundant processors in a PLC control system to improve overall reliability of the process.

The control processor for the Locomotive will also contain a limited amount of logic and interlocks required by the WP Transporter and Gantry Carrier. Remote I/O card racks will be on the WP Transporter and Gantry Carrier and will be cable-connected to the control processor in the Locomotive.

The control system for the Emplacement Gantry will also be PLC based. The operator interface for the Gantry control system will be located out of the Emplacement Drift. The emplacement operators will monitor the Gantry's control system actions by either a direct radio or slotted microwave communication network installed within the Emplacement Drifts. The PLC will control all of the Gantry's motor drives, as well as receive information from all of the other instrumentation monitoring systems. This information will then be transmitted to the operators at the remote location, allowing for safe, reliable placement of the WP at its final emplacement position.

The control systems for the stationary emplacement equipment, such as Isolation Doors and rail switches, will also be PLC based. The logic for the mobile and stationary control systems will incorporate appropriate interfaces for implementing interlocks and safety function as determined by the Geologic Repository Operation Area (GROA) operating philosophy, personnel, and equipment protection (Ref. 5.36).

8. CONCLUSIONS

8.1 GENERAL

Much of this analysis is based on preliminary information as noted by the TBVs shown in Section 4. If any of these inputs change, the conclusions of this analysis may be impacted.

The conclusions of this analysis confirm that the use of rail based systems for the transportation and emplacement of the WPs, as stated in a previous analysis (Ref. 5.5), is a suitable concept. Transportation of each WP will be in a shielded Transporter. WP emplacement will be by means of an unshielded, remotely controlled Gantry. The conclusions from this design analysis shall be considered as TBV. Based on the results of this analysis, other more specific conclusions can be drawn. These conclusions are as follows:

1. This analysis confirms that a Gantry system, with the capability for emplacement and WP carry-over (Ref. 5.8), will work in an excavated Emplacement Drift of 5.5 m diameter when utilizing a ground support system with 0.2 m lining. The carry-over capability provides for selective emplacement and removal of WPs, if subsequently required.

2. The following table summarizes the various operating conditions, lift requirements, and corresponding drift diameters:

Shadow-Shield In Place	Carry-Over Required	Lift Required (mm)	Drift Diameter (m)
Yes	Yes	2301	5.5
No	Yes	2226	5.5
No	No	1863	5.3
Yes	No	2301	5.5

Note: The lift exceeds the 2.0 m maximum drop capacity as indicated in Ref. 4.3.5.

3. A distance of 0.8 m from the bottom of the turnout excavation to the invert of the Emplacement Drift excavation was assumed in the analysis to set the elevation of the Emplacement Drift Transfer Dock. This analysis confirms that the equipment can be designed to operate with the 0.8 m dimension.

4. The Reusable Rail Car will have a total of 8 wheels, each 356 mm (14 in.) in diameter.

5. One 45-ton Transport Locomotive (340 hp) is required to move the loaded Transporter up the maximum 0.75% grade of the Emplacement Drift Turnout. The major contributor to the Locomotive selection in this operating mode is the 20 m curve radius of the Emplacement Drift Turnout. The Locomotive size could be reduced with an increase in the curve radius.. To move a loaded Transporter back to the WHB requires two (2) Transport Locomotives.

6. The loaded Transporter containing the heaviest WP on a Reusable Rail Car weighs 233 MT. At this weight, the Transporter requires two 4-wheel trucks, each having 762 mm (30 in.) diameter wheels.

7. Gantry Carrier wheels will be 762mm (30 in.) in diameter.

8. Transporter and Gantry Carrier wheels are required to be hardened to 615 Brinell Hardness to support the loads of the WPs and Gantry.

9. The Emplacement Gantry can be supported by four bogies, each having two wheels. These wheels are 400 mm (15.75 in.) in diameter and must be hardened to 320 Brinell Hardness.

10. At least one-half of the Gantry traversing wheels must be powered.

8.2 STRUCTURAL ANALYSIS

8.2.1 WP Transporter

The WP Transporter was analyzed in Attachment I using materials per Assumption 4.3.18. Only the Transporter's roof, side walls, and floor (radial direction elements) were considered as structural elements to the structure model and they were analyzed using ASTM A36 steel materials. The stainless steel and borated polyethylene, in addition to the Transporter's end wall and door sections (axial direction elements), were considered non-structural and treated as dead load to the structure model. The WP Transporter was designed to withstand dead, live, wind, seismic, and anticipated impact forces. Maximum stress, as well as Transporter stability, were evaluated to provide overall structural integrity to the Transporter. The following major structural components of the WP Transporter were analyzed in Attachment I:

A. WP Transporter Side Walls: 152.4 mm (6.0 inches) thick plate.
B. WP Transporter Top (Roof): 152.4 mm (6.0 inches) thick plate.
C. WP Transporter Floor: 152.4 mm (6.0 inches) thick plate.

The analysis showed that the WP Transporter is satisfactory for the operational loading of the Reusable Rail Car with the six different WPs shown in Sections 4.1.6 and 4.3.11.

8.2.2 Emplacement Gantry

The Emplacement Gantry was analyzed in Attachment II. Figure II-1 of Attachment II shows the structural details developed by this analysis for the Gantry. The Emplacement Gantry consists of two major parts:

1. A Gantry hoist frame with horizontal traveling lifting head trolleys to lift different size WPs.

2. A Gantry structural frame to lift and transport the hoist frame with lifting head trolleys and different size WPs.

The Emplacement Gantry was designed to withstand dead, live, anticipated impact, inertia, thrust, out-of-plumb, skewing, racking, collision, and seismic forces. Both maximum stress and fatigue stress range were considered in the analysis. Member deflections and support displacements, as well as Gantry stability, were evaluated to provide overall structural integrity to the Gantry.

The Gantry was designed using ASTM A36 steel materials. The following major components of the Emplacement Gantry were designed in Attachment II:

A. The Lifting Head Trolley Frame: Built-up box and tube sections with 69 MT maximum WP live load (Figure II-1).

B. The Hoist Frame: Built-up box sections (Figure II-1).

C. The Gantry Structural Frame: Built-up box and channel sections (Figure II-1).

The Emplacement Gantry, as analyzed, is satisfactory for the operational loading of six different WPs shown in Sections 4.1.6 and 4.3.11.

8.2.3 Structural Considerations for other Major Equipment

Equipment listed in this section includes the Gantry Carrier, the Reusable Rail Car, the Transport Locomotives and the Drift Isolation Doors. For this equipment no structural analysis has been performed at this time. The sizing of various components and the selection of configurations is based on engineering judgement as it relates to the numerous design input criteria. All of these items will require a structural evaluation during future design activities.

8.3 RECOMMENDATIONS

Based on the preceding conclusions, the recommended rail transportation and Gantry emplacement system will use a *45-ton-capacity Transport Locomotive, a WP Transporter with a Reusable Rail Car having a capacity of 69 MT, an Emplacement Gantry with total lifting height capability of 2226 mm and a load capacity of 69 MT, with a 5.5 m Emplacement Drift diameter (excavated). The mechanical and structural analysis of this system have shown that it is feasible; however, additional future design activities are recommended to evaluate and refine the following component requirements:

- Refine structural design of WP Transporter.
- Refine structural design of Emplacement Gantry.
- Structural design of the Gantry Carrier.
- Structural design of the Reusable Rail Car.
- Transporter door locking device and control device.
- Reusable Rail Car:
  1. Restraint device on WP Transporter.
  2. Mechanism for positive connection and release of engagement.
- Emplacement Gantry:
  1. Auxiliary braking and rail clamping.
  2. WP support during traversing operation.
  3. Handling of a radiologically contaminated Emplacement Gantry.
- Final Locomotive selection and specification.
- Emplacement Drift Isolation Doors design.
- Emplacement Drift Transfer Dock supporting and aligning devices for WP Transporter and Gantry Carrier.
- Evaluate alternative design solutions to avoid movement of a loaded WP transporter with open doors at the emplacement drift entrance or in the WHB.
- Issues that will be addessed during subsequent design analysis work, but prior to VA.:
  - Emplacement Throughput,
  - Service Life,
  - Emplacement Cycles,
  - Emplacement Tolerances,
  - Waste Package Handling and Transportation Loads.

8.4 VERIFICATION OF DESIGN BASIS INITIAL SELECTIONS

Initial design selections identified in Attachments I through IX were verified in those attachments. The following sections summarize the design selections.

---

*Two (2) Transport Locomotives in the North Ramp and Mains and one (1) Transport Locomotive at the Emplacement Drift entrance.

8.4.1 Track and Drift Layout

1. A 20 m radius for the Drift Turnout is compatible with:

   - A Transporter with two trucks, each having two axles and four 762 mm (30 in.) diameter wheels.
   - A Gantry Carrier with two trucks, each having two axles and four 762 mm (30 in.) diameter wheels.
   - A two-axle Locomotive having four 762 mm (30 in.) diameter wheels with axles on 2540 mm (100 in.) centers.

2. A 2.580 m track gage is compatible with the Emplacement Gantry.

3. A 1.44 m track gage is compatible with the WP Transporter, the Gantry Carrier, and the Reusable Rail Car.

4. A 44.6 kg/m (90 lb/yd) ASCE rail in the Emplacement Drift is compatible with the Reusable Rail Car and the Gantry.

5. A 57 kg/m (115 lb/yd) AREA rail is compatible with the Transporter, the Gantry Carrier, and the Transport Locomotive(s).

6. A Brinell Hardness of 320 for all rail is required for this analysis.

8.4.2 Dimensional Relationships

1. A distance of 1.32 m from the center line of the largest diameter (2.0 m) WP on the Reusable Rail Car to the top of the rail is required for this analysis.

2. A distance of 1.28 m from the top of the rail (T.O.R.) in the Transporter and the Transfer Dock to top rail in the Drift Turnout is required for this analysis.

3. A distance of 1.22 m from the top of the Gantry rail on Gantry Carrier to top of the rail in the Drift Turnout is required for this analysis.

4. The assumed distance of 0.8 m (Ref. 4.3.14) from the bottom of the Drift Turnout excavation to the invert of the Emplacement Drift excavation is compatible with this analysis.

8.4.3 Grade Relationships

1. The existing grade of -2.1486% (Design Parameter 4.1.1) in the North Ramp is compatible with the WP transport equipment as sized and selected.

2. The existing grade of -1.35 percent in the East Main, and the designed maximum grade of -1.394% in the West Main, is compatible with the WP transport equipment as sized and selected.

3. A maximum grade of -1.0 percent in the Emplacement Drifts is compatible with Emplacement Gantry operations.

8.4.4 Waste Package Configuration, Dimensions, and Weights

1. A maximum WP weight of 69 MT is compatible with the requirements of this analysis.

2. A maximum WP diameter of 2.0 m is compatible with the requirements of this analysis.

3. A maximum WP length of 5850 mm is compatible with the requirements of this analysis.

4. WP skirts having a length of 225 mm at both ends of the WPs are compatible with the lifting requirements of this analysis.

8.4.5 Waste Package Emplacement

1. An excavated drift diameter of 5.5 m, with the ground support system in place, is compatible with the requirements for a Gantry to lift a 2.0 m diameter WP over another 2.0 m diameter WP, resting on pedestals in the Emplacement Drift, with or without an installed shadow shield.

2. An excavated drift diameter of 5.3 m, with the ground support system in place is required for a Gantry without carry-over capability and without an installed shadow shield, to emplace a 2.0 m diameter WP onto pedestals in the Emplacement Drift.

3. The assumption (4.3.30) that the gantry must be able to place WPs as close as one meter apart (end-to-end) is not currently met with all WP configurations. Further design work is needed to comply with this assumption.

8.4.6 Emplacement Equipment

1. A travel speed of 8 km/hr (5mph) in the North Ramp and Mains is compatible for WP transport..

2. A wheel diameter of 762 mm (30 in.) for the WP Transporter truck wheels is compatible with this analysis.

3. A Reusable Rail Car having 4 axles and 8 wheels is compatible with the loads from the heaviest WP.

4. A Reusable Rail Car wheel diameter of 356 mm (14 in.) is compatible with the loads from the heaviest WP.

5. A Reusable Rail Car unloading speed of 7.6 m/min (25 ft/min) is compatible with the requirements of this analysis.

6. A Gantry length of approximately 12 m is compatible with this analysis.

7. A Gantry speed of 0.73 m/sec (144 ft/min) is compatible with this analysis.

8. A Gantry supported by 8 wheels is compatible with this analysis.

9. A Gantry wheel diameter of 400 mm (15.75 in.) is compatible with the requirements of this analysis.

10. A Brinell Hardness of 320 for all equipment wheels is compatible with the requirements of this analysis except 615 for Transporter wheels.

8.4.7 Intermediate Diameter Waste Packages

The concept depicted in this analysis is valid (i.e.; clears the shadow shield and provides carry-over capability) for the largest diameter WP and the smallest diameter WP considered (see Section 4 Design Inputs). However, additional design effort is needed to be able to accommodate all intermediate WP diameters. In particular, the lifting block configuration will need design revisions so that it can engage the full range of WP diameters and provide sufficient lifting height to clear the shadow shield and provide carry-over capability. Accommodation of intermediate WP diameters can be provided by changing the lifting block configuration to provide more engagement surfaces and/or by providing interchangeable lifting blocks.

Waste packages sizes are not yet well defined, and a recent change in the CDA has led to not only changes in the diameters and lengths, but also to the specification of size ranges as opposed to definitive dimensions for each package type. It is anticipated that this analysis will be re-visited when waste package sizes are better defined.

8.4.8 Impact of most recent CDA changes regarding the Equipment Design Concept.

Just prior to completion of this analysis several design inputs were changed (Ref. 4.1.6 and Ref. 4.3.11), these include the following dimensions:

1. Loaded Mass of the heaviest WP changed from 69,000 kg, to 70,000 kg. (Ref. 4.3.11, Note 3). The increase from 69,000 kg and 70,000 kg equals 1.4 % of loaded mass.
2. Outer Diameter of smallest WP changed from 1.298 m, to 1.250 m. (Ref. 4.1.6, Note 2). The decrease from 1.298 m to 1.250 m equals 3.7 % in diameter.
3. Outer Length of longest WP changed from 5.85 m, to 5.900 m. (Ref. 4.3.11, Note 2). The increase from 5.85 m to 5.900 m equals less than 1 % in length.

The changes as indicated above will not invalidate the basic design concept in the foregoing conclusions.

9. ATTACHMENTS

Attachment:	Pages:	Description:
I.	45	Waste Package Transporter Structural Analysis Note: STAAD-III Analysis and Stability Analysis portions of this attachment, comprising paragraph/pages I-39 through I-235, for REV 00B, are located in Ref. 5.53, in electronic media form. These can be located in RPC under Batch # MOY-970806-05
II.	78	Emplacement Gantry Structural Analysis Note: STAAD-III Analysis and Stability Analysis portions of this attachment, comprising paragraph/pages II-68 through II-804, for REV 00B, are located in Ref. 5.54, in electronic media form. These can be located in RPC under Batch # MOY-970806-05
III.	21	Waste Package Transporter - Mechanical Equipment Selections
IV.	18	Reusable Rail Car - Mechanical Equipment Selections
V.	17	Gantry Loads and Equipment Selections
VI.	17	Gantry Carrier Weight and Wheel Selection
VII.	14	Transport Locomotive - Equipment Selection
VIII.	6	Transporter/Gantry Carrier Truck Arrangement and Weight Analysis
IX.	7	Transporter Coupler Vendor Information and Weight
X.	11	Serapid Rigid Chain Information
XI.	4	Goodman Locomotive Information
XII.	50	Vendor Data, Various

ATTACHMENT I

WASTE PACKAGE TRANSPORTER STRUCTURAL ANALYSIS

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TABLE OF CONTENTS

NOTE: STAAD-III Analysis and Stability Analysis portions of this attachment, comprising pages I-39 through I-45, are located in Reference 5.53, Batch No. MOY-970806-05, in electronic media form.

WASTE PACKAGE TRANSPORTER STRUCTURAL ANALYSIS

1.0 PRINCIPAL LOADS

Since this design analysis is for the purpose of viability assessment only, the structural analysis for the Waste Package Transporter will consider only those major loadings considered to most significantly effect the design.

Estimated Total Loaded Weight of Transporter = 233.15 MT (514.09 kips) (Attachment I, Section 10.0, Part 3.) (includes WP and Reusable Rail Car)

1.1 Dead Load (DL) (CMAA 70, Section 3.3.2.1.1.1, Ref. 4.4.5)

The weights of all effective parts of the transporter structure, the machinery parts and the fixed equipment supported by the structure.

- Rigid Chain Gear Motor = 320 kg (0.71 kips) (Attachment III, Section 3.3.9)
- Rigid Chain Drive (Ea.) = 240 kg (0.53 kips) (Attachment III, Section 3.3.6)
- Rigid Chain Magazine (Ea.) = 630 kg (1.39 kips) (Attachment III, Section 3.3.11)
- Rigid Chain Guide (Ea.) = 43 kg/M (0.03 kips/ft) (Attachment III, Section 3.3.6)
- Truck (Ea.) = 4,989 kg (11.0 kips) (Attachment VIII, Section 2.1)
- Coupler (Ea.) = 79.4 kg (0.175 kips) (Attachment IX, Section 2.0)
- Rail = 44.64 kg/m (90 lb/yd) = 30 lb/ft (Section 4.3.25)
- Door Operator Gear Motor = 200 kg (0.44 kips) (Attachment III, Section 3.2.5)
- Rigid Chain Drive Removable Shield = 6.4 kips (estimated)
- Waste Package Transporter Material Weights (Section 4.3.18)
  - Floor, Sides and Top (Radial Direction)
    - Stainless Steel = 5mm (0.20 inches) = 8.14 psf, round to 8.2 psf
    - Borated Polyethylene = 101.6 mm (4.0 inches) = 19.14 psf, round to 19.2 psf
    - Carbon Steel = 152.4 mm (6.0 inches) = 244.47 psf, round to 244.5 psf
    - Stainless Steel = 5 mm (0.20 inches) = 8.14 psf, round to 8.2 psf
  - End Sections (Axial Direction)
    - Stainless Steel = 5 mm (0.20 inches) = 8.14 psf, round to 8.2 psf
    - Borated Polyethylene = 76.2 mm (3.0 inches) = 14.4 psf
    - Carbon Steel = 177.8 mm (7.0 inches) = 285.2 psf
    - Stainless Steel = 5 mm (0.20 inches) = 8.14 psf, round to 8.2 psf

1.2 Live Load (LL) (CMAA 70, Section 3.3.2.1.1.3, Ref. 4.4.5)

Empty Weight Reusable Rail Car = 11.0 MT (24.3 kips) (Attachment IV, Section 3.1.7)

Waste Packages (WP) to be Transported (Section 4.3.10):

WP Type	Diameter (mm)	Length (mm)	WP Loaded Mass (kg)
4 DHLW	1785	3790	30,511
12 PWR UCF	1298	5335	32,236
44 BWR UCF	1604	5335	46,424
21 PWR UCF	1650	5335	50,423
—	1970 (77.6 in.)	5350 (210.6 in.)	69,000 (152.2 kips)
—	1850	5850	69,000

1.3 Dead and Live Load Impact (Vertical Inertia Forces, CMAA 70, Section 3.3.2.1.1.4, Ref. 4.4.5)
Vertical Inertia Forces include those due to the motion of the transporter, and are included by the application of a separate factor by which the vertical acting loads are multiplied.

1.3.1 Dead Load Factor (DLF) (CMAA 70, Section 3.3.2.1.1.4.1, Ref. 4.4.5)

Transporter travel speed = 8 km/hr = 437 ft/min. (Ref. 5.6)
Use DLF = 1.2

1.3.2 Live Load Impact Factor (HLF) (Whiting Crane Handbook, Ref. 5.18, pg. 57)
Class "E" Crane, Impact = 50%, is highest class indicated.
Transporter of equivalent service Class "F" crane is reasonable because transporter performs critical tasks and must provide highest reliability.
Use HFL = 0.50

2.0 EXTRAORDINARY LOADS

2.1 Stored Wind Load (WLS) (CMAA 70, Section 3.3.2.1.3.1, Ref. 4.4.5 and ASCE 7-88, Ref. 4.4.7) - See Wind Load Analysis, Attachment I, Section 8.0
Use WLS = 23 psf

2.2 Seismic (EQF) (Section 4.3.20)
Horizontal and Vertical Acceleration = 0.27 g (Ref. 5.9, pg. 13)

2.3 Impact Loads (AAR M-1001, Vol. 1, Ref. 4.4.8)

2.3.1 Horizontal Impact Force (HIF) (AAR M-1001 Vol. 1, Section 4.1.10.1)
Coupler Force = 1,250,000 lbs applied to one end of the car
Use HIF = 1,250,000 lbs

2.3.2 Vertical Dynamic Amplification Factor (AAR M-1001 Vol. 1, Section 4.1.11.1)

$$a = 1 + \frac{2hH}{bW}$$

where: a = amplification factor
b = distance between truck centers in feet = 3.66 m (12.0 ft)
h = vertical distance, centerline of coupler to maximum center of gravity height in feet = 1.55 m (5.08 ft)

Note: Distance T.0.R. to car center of gravity = 92" (Attachment I-Section 10.0, Part 3)
Distance T.0.R to centerline coupler = 0.79 m (31") (Fig. 7.2.2)
h = 61" (5.08 ft)

W = rail load limit less weight of trucks in pounds =

$$233.15\text{ MT} - \frac{2(11.0\ kips)}{2.205\ kips/MT} = 223.17\text{ MT } (492{,}090\text{ lb})$$

H = horizontal impact force (HIF) in pounds = 1,250,000 lb

Amplification Factor, a = 3.2

3.0 LOAD COMBINATION

The combined stresses shall be calculated for the following design cases:

3.1 Load Combination 1: Transporter in regular use under principal loading; Stress Level 1 (CMAA 70, Section 3.3.2.4, Ref. 4.4.5)
DL(DLF) + LL(1+HLF)
Note: TL (DLF) and IFD not applicable to this analysis.

3.2 Load Combination 3: Extraordinary Loads; Stress Level 3 (CMAA 70, Section 3.3.2.4, Ref. 4.4.5)

3.2.1 Transporter subjected to stored wind loading
DL + LL + WLS

3.2.2 Transporter subjected to seismic force

DL + LL + EQF

3.3 Load Combination 4: Critical Load Condition; Stress Level 4 (AAR M-1001 Vol. 1, Section 4.2.2.6, Ref. 4.4.8)

Transporter subjected to critical impact loads

a (DL + LL) + 1.0 (HIF)

4.0 ALLOWABLE STRESSES

4.1 Maximum Allowable Stresses in Structural Steel Members

CMAA 70, Section 2.7, Ref. 4.4.5: Consider transporter to be of equivalent crane Service Class "F" because:

- Performs critical tasks
- Must provide highest reliability

CMAA 70, Tables 3.4.7-1 and 3.4.7-2A, Ref. 4.4.5
Service Class "F"

Joint Category B; allowable fatigue stress range	= 17.0 ksi
Note: Joint Category B is selected because transporter is principally a built-up box section (CMAA 70, Table 3.4.7-2A)	
Plus Approximate Transporter Dead Load Stress (to be verified by this analysis)	= <u>1.0 ksi</u>
Use Total Allowable Stress	= 18.0 ksi

AISC, M016-89 (F1-1 & F3-1) pg. 5-45 & 5-48 (Ref. 4.4.1); $F_b = 0.66\ Fy$

For STAAD analysis, limit the allowable stress range for fatigue by reducing Fy for ASTM A-36 steel:

Use reduced $Fy = \frac{F_b}{0.66} = \frac{18.0\ ksi}{0.66} = 27.3$ ksi, ASTM A-36 steel

4.2 Allowable Stress Level

Load Combination 1, Stress Increase $= \frac{0.60}{0.60} = 1.0$; 0% stress increase (CMAA 70, Section 3.4, Ref. 4.4.5)

Load Combination 3, Stress Increase $= \frac{0.75}{0.60} = 1.25$; 25% stress increase (CMAA 70, Section 3.4, Ref. 4.4.5)

Note: The allowable stress increase for Load Combination 3 is a percentage increase above the allowable combined stress level under principal loading, Stress Level 1, Load Combination 1.

Load Combination 4, Stress Increase $= \frac{1}{0.60}\left(\frac{36.0}{27.3}\right) = 2.20$; 220% stress increase (AAR M-1001 Vol. 1, Section 4.2.2.6, Ref. 4.4.8)

Note: The factor of $\left(\frac{1}{0.60}\right)$ increases the allowable stress from normal bending stress of 0.60 Fy to the Reduced Fy for STAAD analysis of 27.3 ksi for ASTM A-36 steel. The factor of $\left(\frac{36.0}{27.3}\right)$ increases the allowable stress from Reduced Fy for STAAD analysis of 27.3 ksi to ultimate load-carrying capacity of 36 ksi for ASTM A-36 steel.

5.0 TRANSPORTER STABILITY

5.1 Stability (ASME NOG-1-1995, Section NOG-4457, Ref. 4.4.2)

The transporter stability safety factors against overturning of NOG-4457 when subjected to the load combinations for normal and extreme environmental loads (ASME NOG1-1995, Section NOG-4140) are by inspection comparable and appropriate for use with the CMAA 70, Section 3.3.2.4, Ref. 4.4.5, load combinations.

Normal Operating Condition
Overturning Safety Factor ≥ 1.5 (Load Combination Case #1)

Extraordinary (Extreme Environmental) Loading Condition
Overturning Safety Factor ≥ 1.1 (Load Combination Case #3)

5.2 Vertical Center of Gravity (AAR M-1001, Vol. 1, Section 2.1.3, Ref. 4.4.8)

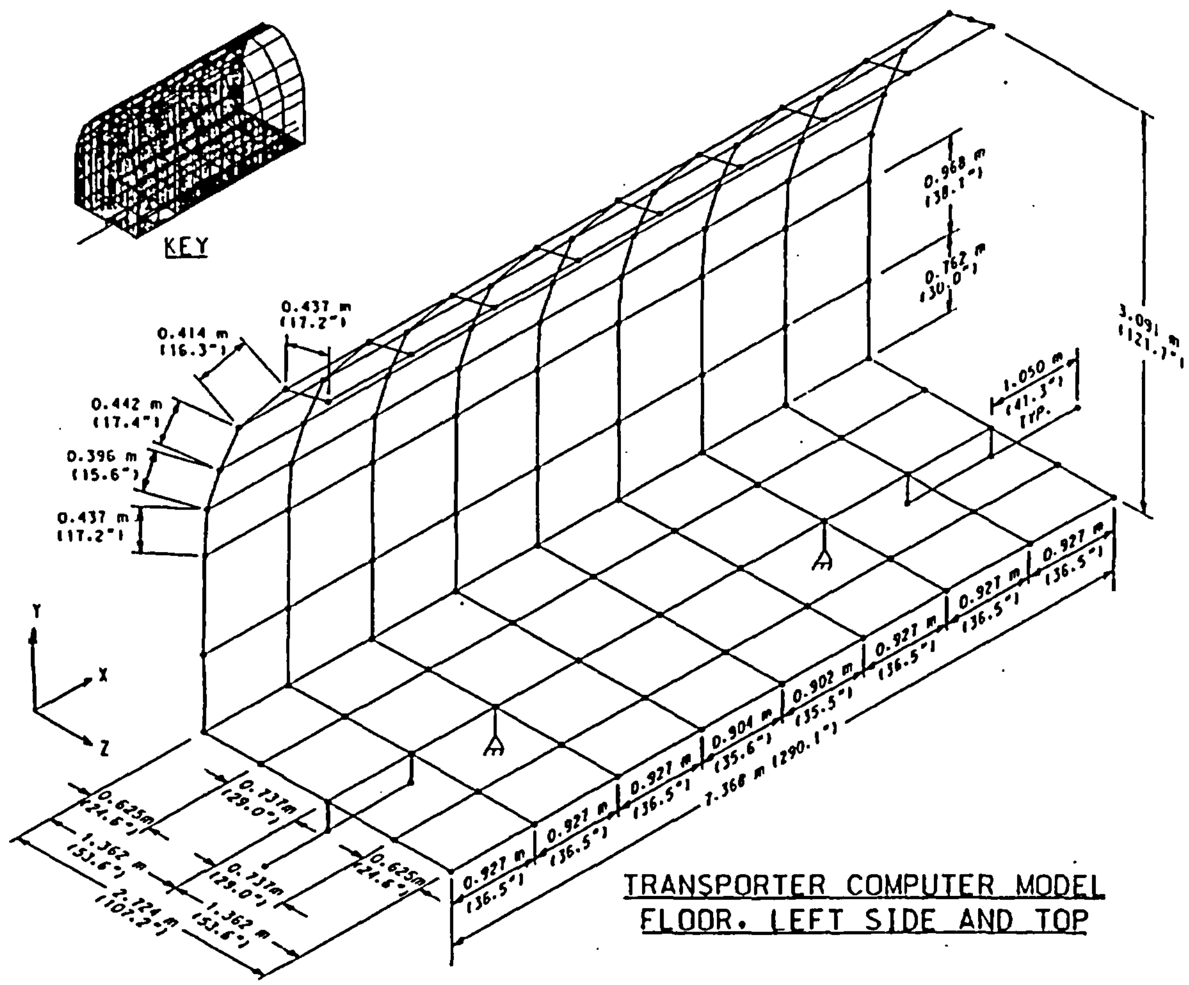
Height of center of gravity of fully loaded car (including weight of trucks) shall not exceed 98 inches above top of rail.

6.0 DESIGN METHODS

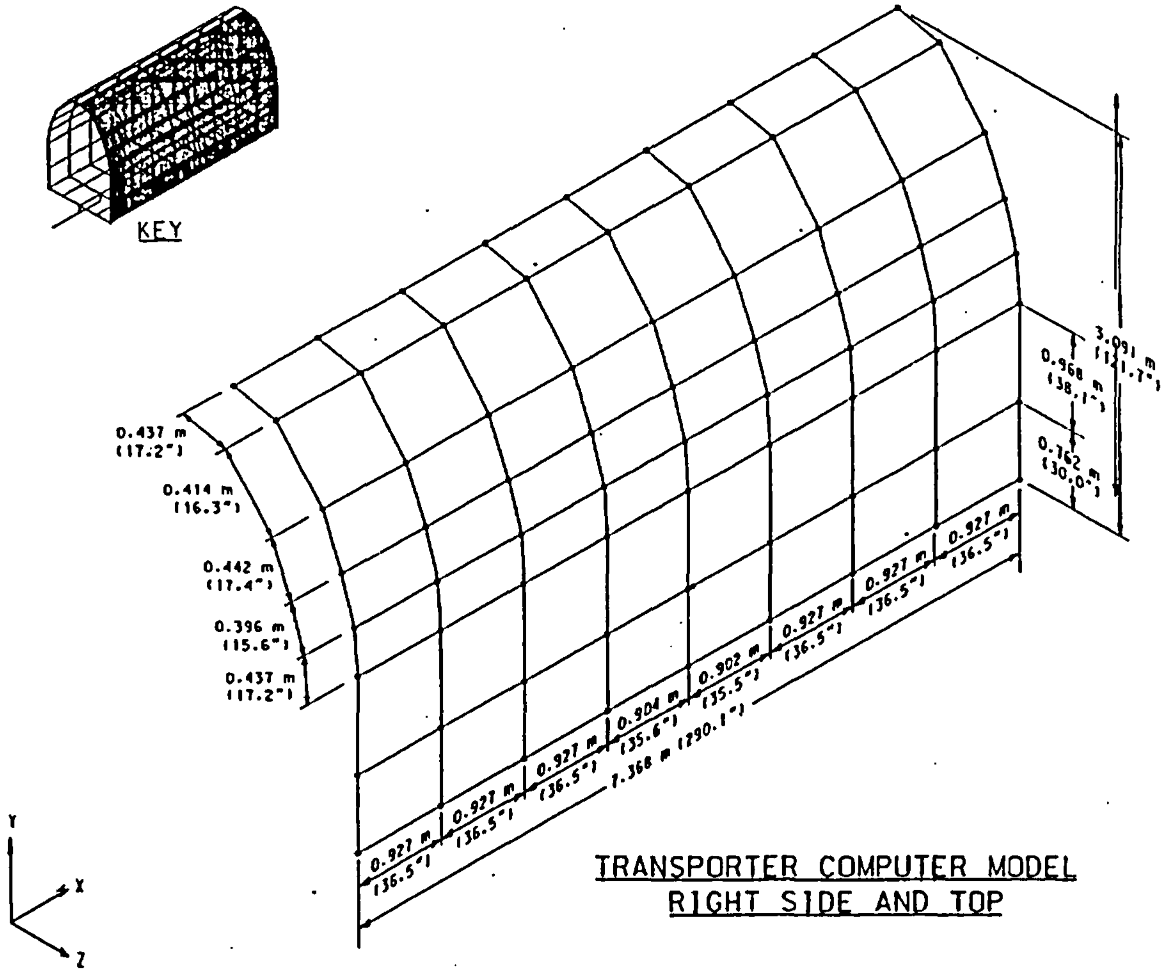
The Waste Package Transporter will be designed using static load analysis. STAAD-III computer software will be used to perform the stress analysis for the transporter structural frame. STAAD-III facilities for steel design will be based on the AISC-ASD code.

7.0 STAAD-III MODEL

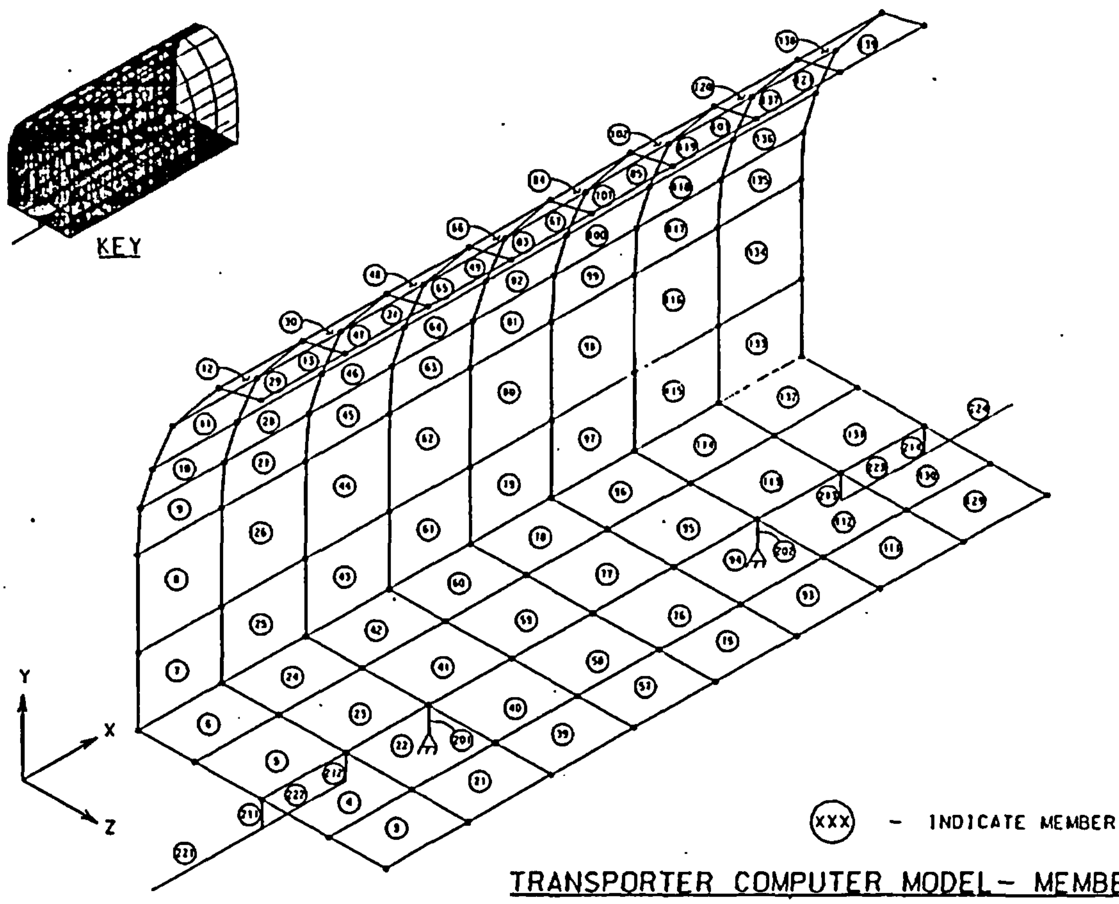
STAAD-III model diagrams indicate model configurations with model members and joints identified. The basis for the computer model is shown in Figures 7.2.1, 7.2.2, 7.2.3, 7.2.4, 7.2.5, 7.2.6, 7.2.7, 7.3.1, 7.3.2, 7.3.3, and 7.3.4.



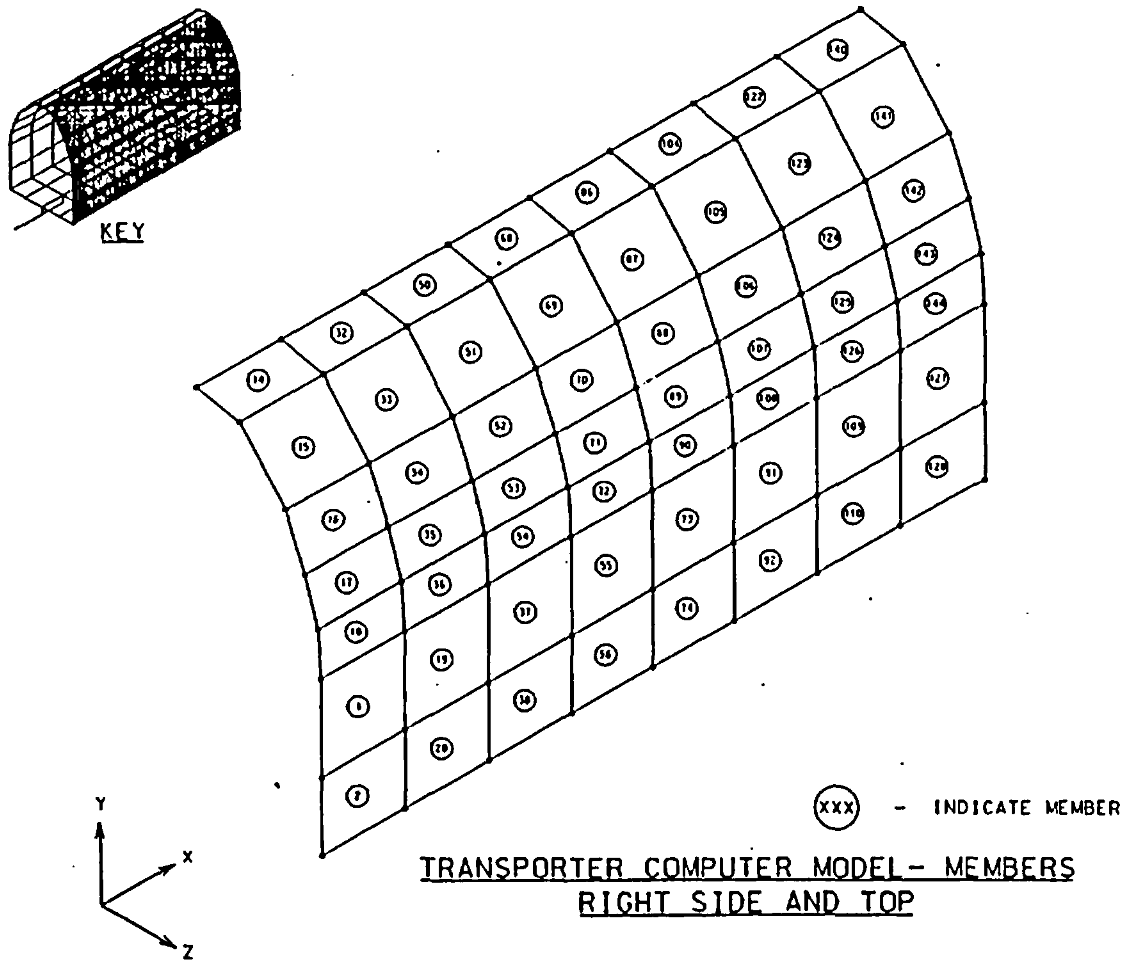
TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP



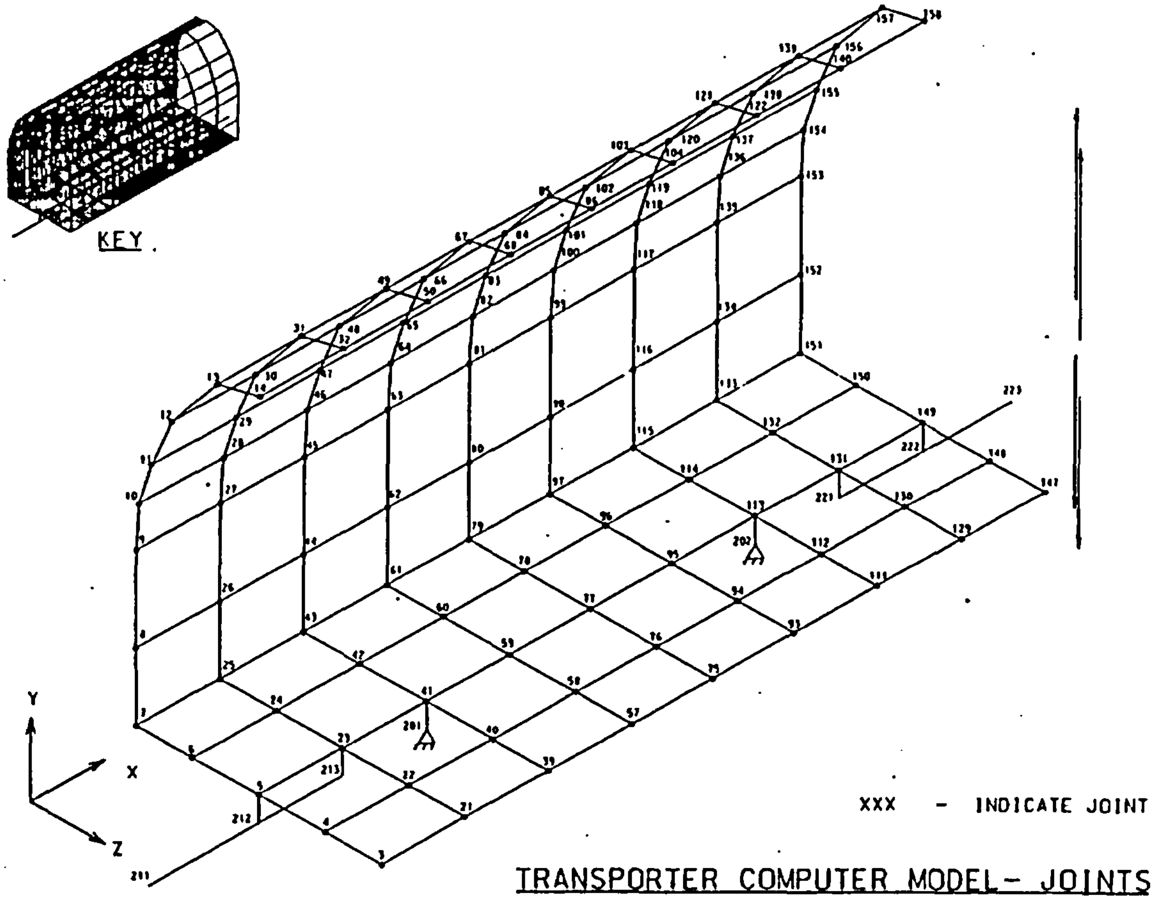
TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP



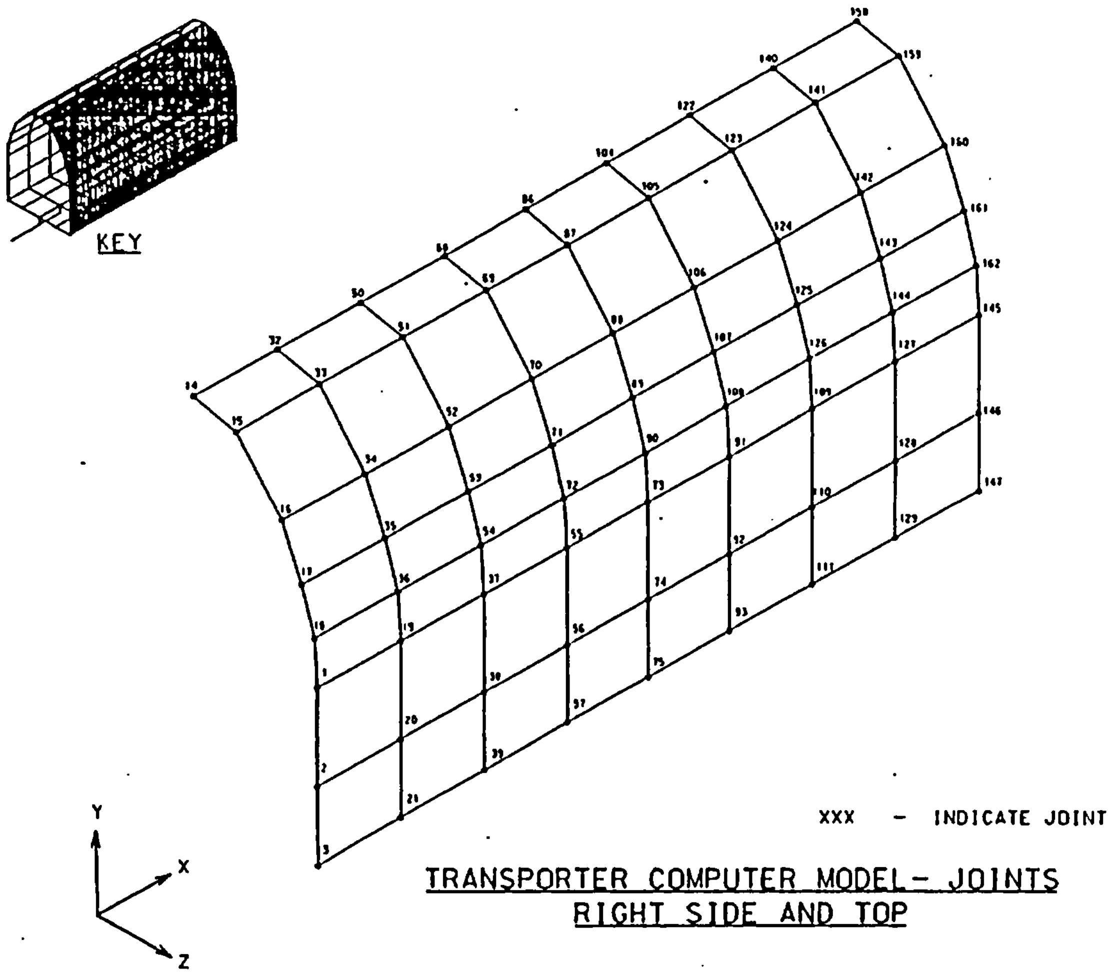
TRANSPORTER COMPUTER MODEL- MEMBERS
FLOOR, LEFT SIDE AND TOP



TRANSPORTER COMPUTER MODEL- MEMBERS
RIGHT SIDE AND TOP



TRANSPORTER COMPUTER MODEL- JOINTS
FLOOR, LEFT SIDE AND TOP



TRANSPORTER COMPUTER MODEL- JOINTS
RIGHT SIDE AND TOP

8.0 WIND LOAD ANALYSIS

WIND LOAD ANALYSIS

ASCE STANDARD ASCE 7-88 (FORMERLY ANSI A58.1)
Minimum Design loads for Buildings and other Structures, Revision of ANSI A58.1-1982

ASCE 7-88 $q_z = 0.00256\,K_z\,(I\,V)^2$ Eq 3
V = basic wind speed
I = Importance factor
K_z = velocity pressure exposure coefficient

V = 75 mph (SECTION 4.2.3, ~~EXPLORATORY STUDIES FACILITY DESIGN REQUIREMENTS SECT 3.2.1.2.1.1.C~~) RLW 8-26-97
> 70 mph ASCE 7-88, FIGURE 1

I = 1.07 CATEGORY III Essential Facility (ASCE 7-88)
K_z = 0.80 ASCE 7-88 TABLE 6
for z, ht of transporter = 4.07 M = 13.3'
for Exposure "C", Ref 5.47 ~~DOE ORDER 6430.1A~~ RLW 8-26-97
Sections 0111-2.4.2 General Design Criteria, WIND LOADS, BUILDINGS AND OTHER STRUCTURES

q_z = velocity pressure at height z above ground

$$q_z = 0.00256(0.80)[1.07(75\,mph)]^2 = 13.19\ psf$$

WIND LOAD ANALYSIS - CONT.

Main Wind Force, $F = q_z\, G_h\, C_f\, A_f$ (ASCE 7-88 TABLE 4)

design wind pressure = $q_z\, G_h\, C_f$

maximum wind gust = 97 mph, Ref 5.47 Rev 8-26-97, ~~DOE 6430.1A~~ SECTION 0111-2.4.2

Gust Response Factor, $G_h = 1.32$ ASCE 7-88 Sect. 6.6 & TABLE 8

$\frac{\text{Wind Gust}}{\text{Basic Wind}} = \frac{97\text{ mph}}{75\text{ mph}} = 1.29 < 1.32$ therefore use $G_h = 1.32$

Force coefficient, $C_f = 1.3$ ASCE 7-88, TABLE 12 (FIGURE I-1) for $\frac{h}{D} = \frac{4.07\text{ M}}{2.94\text{ M}} = 1.38$

A_f = projected area normal to wind

design wind pressure = 13.19 psf (1.32)(1.3) = 22.6 psf

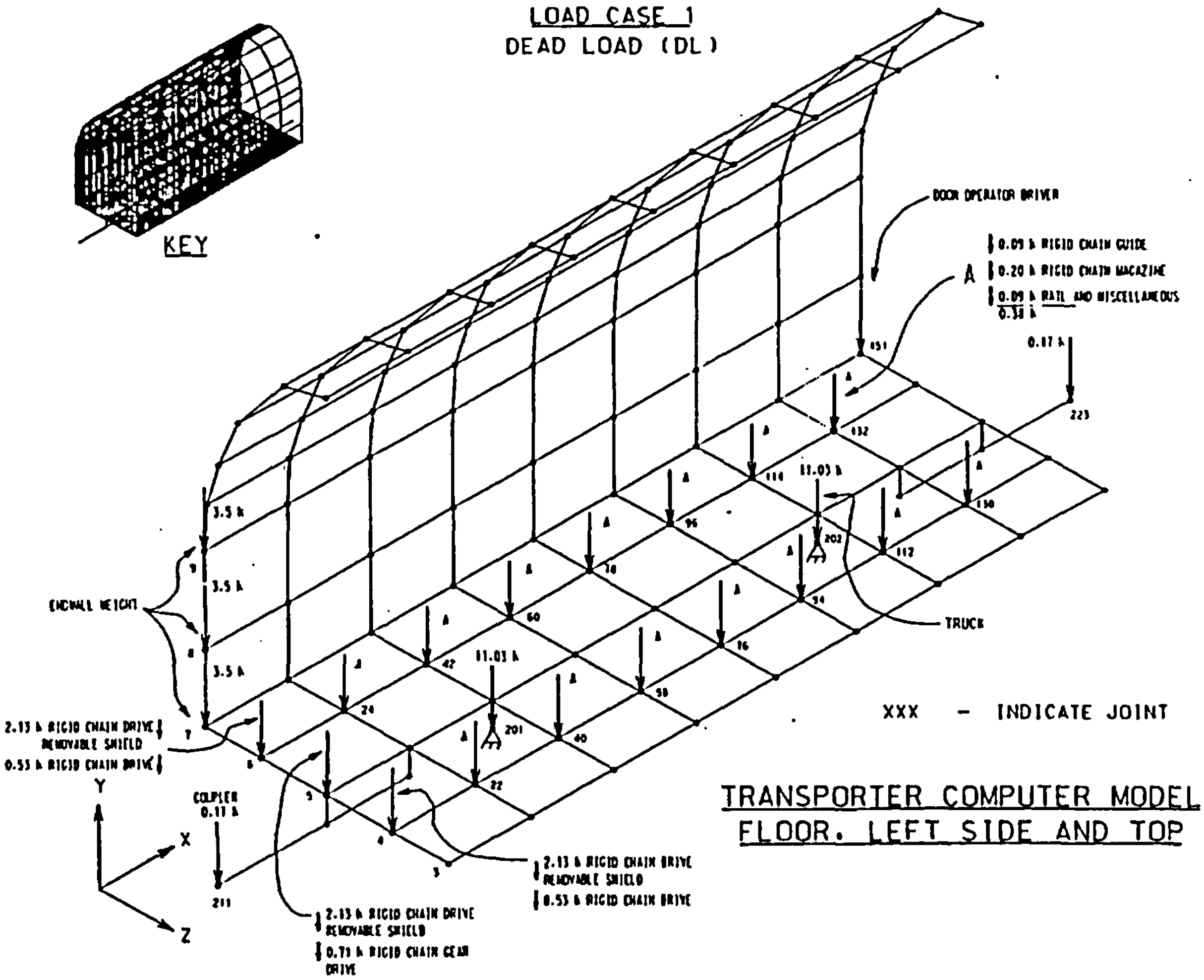
use design wind pressure = 23 psf

9.0 STAAD-III MODEL LOAD DIAGRAMS

STAAD-III model load diagrams indicate model configuration with applicable model members and joints identified.

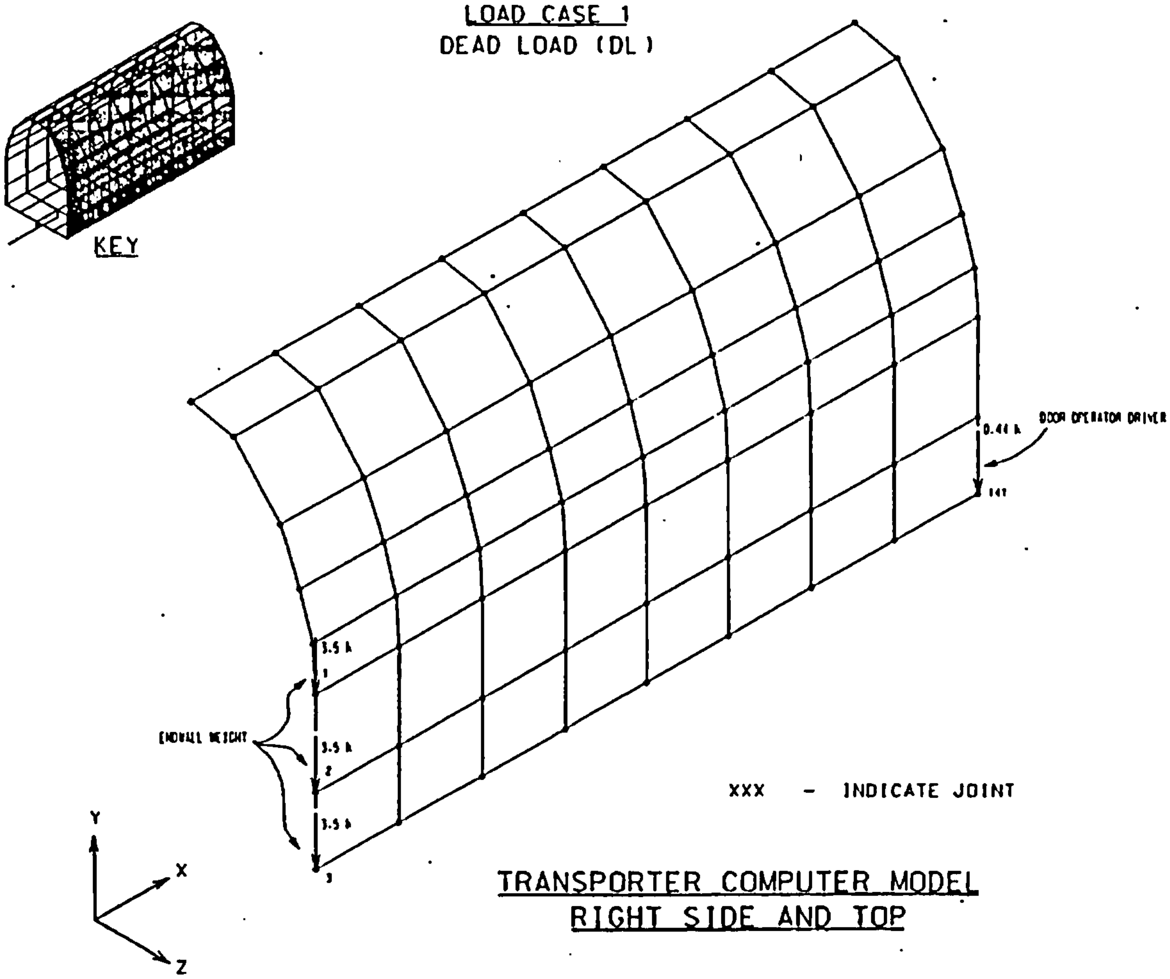
LOAD CASE 1

DEAD LOAD (DL)



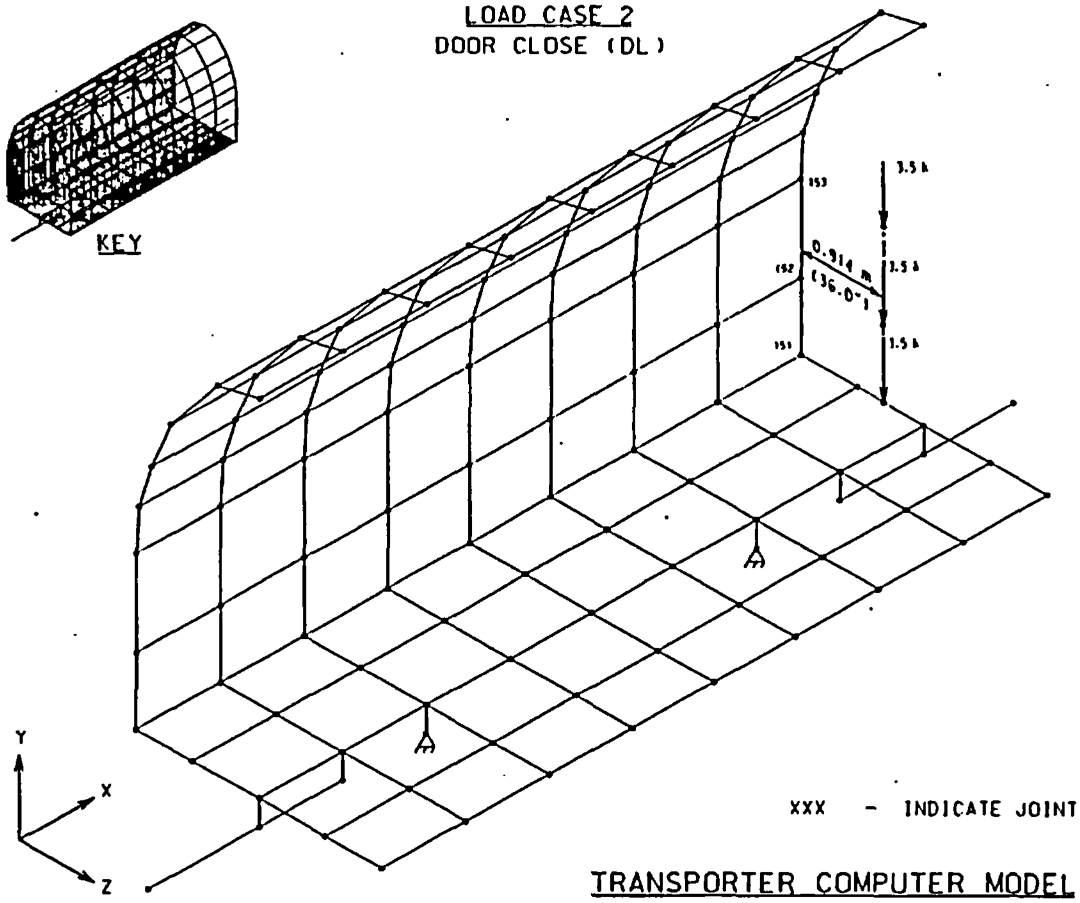
TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

LOAD CASE 1
DEAD LOAD (DL)

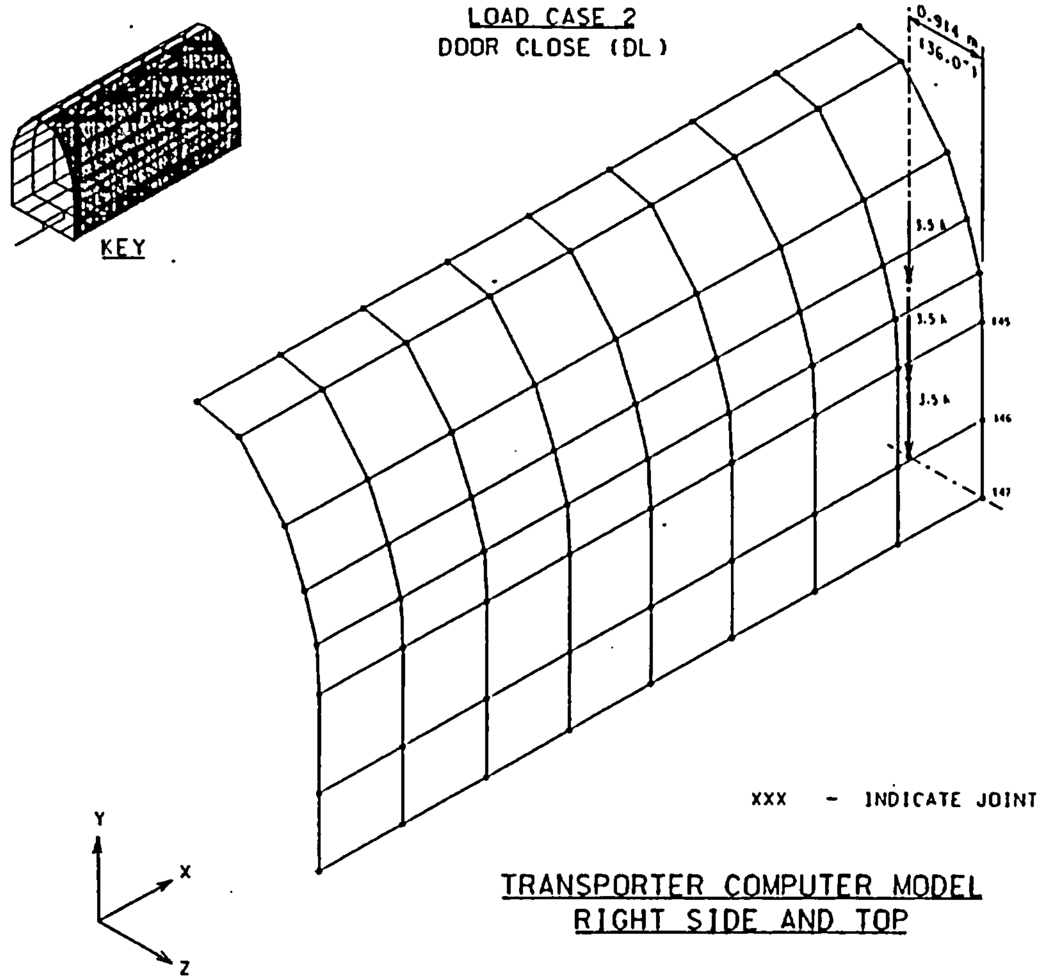


TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP

LOAD CASE 2
DOOR CLOSE (DL)

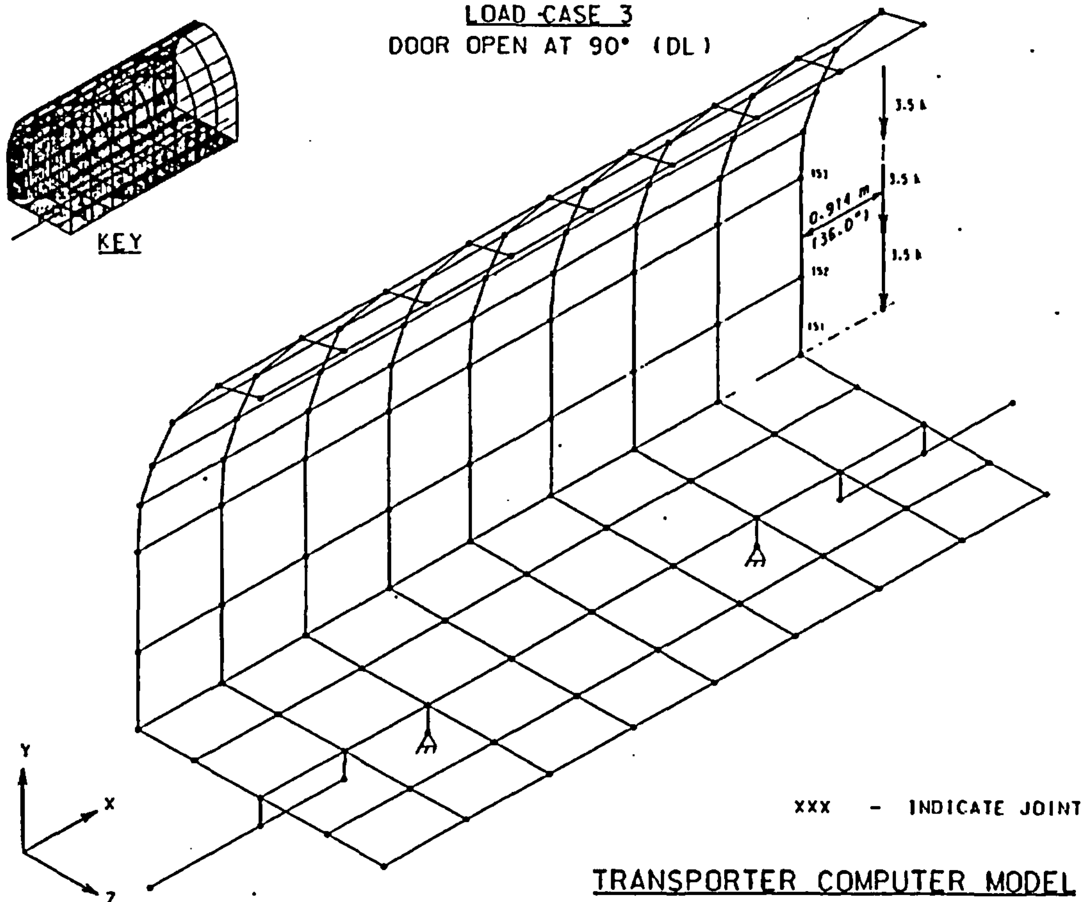


TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP



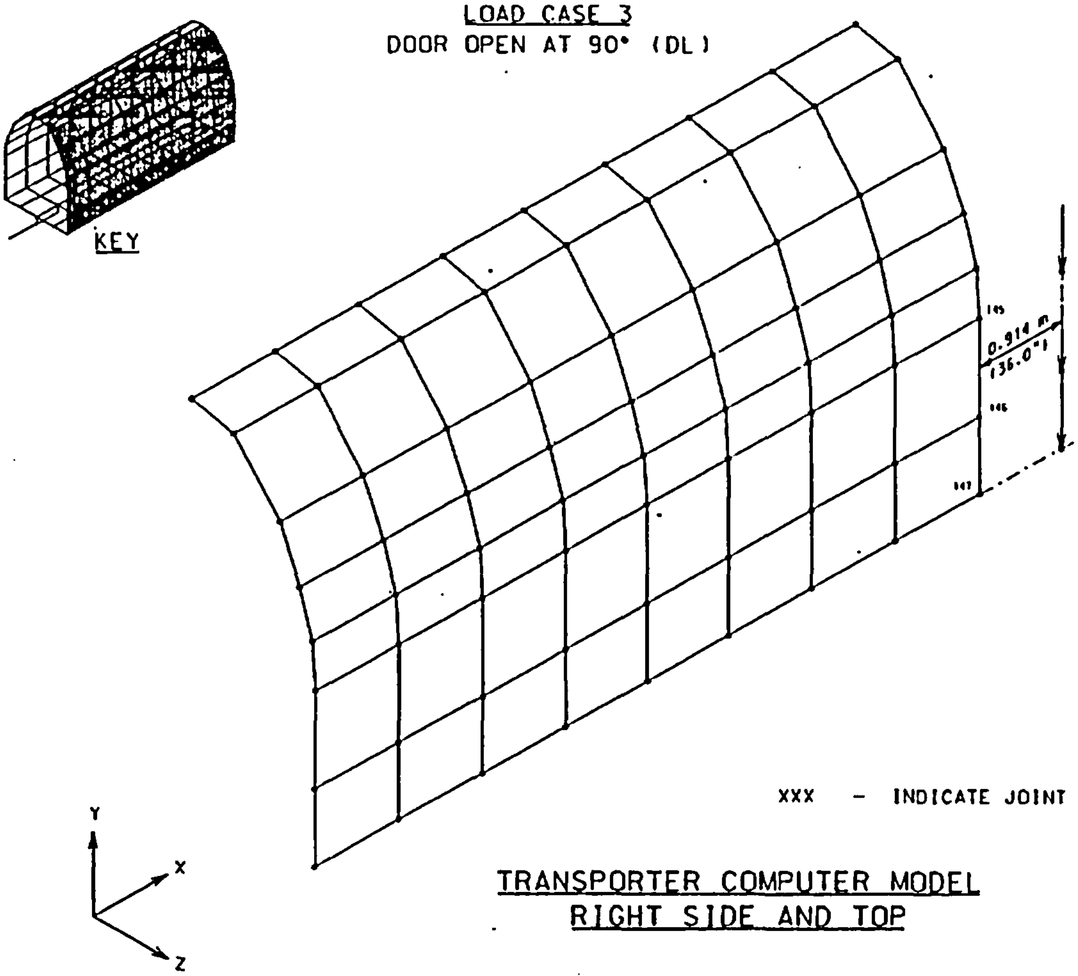
TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP

LOAD CASE 3
DOOR OPEN AT 90° (DL)

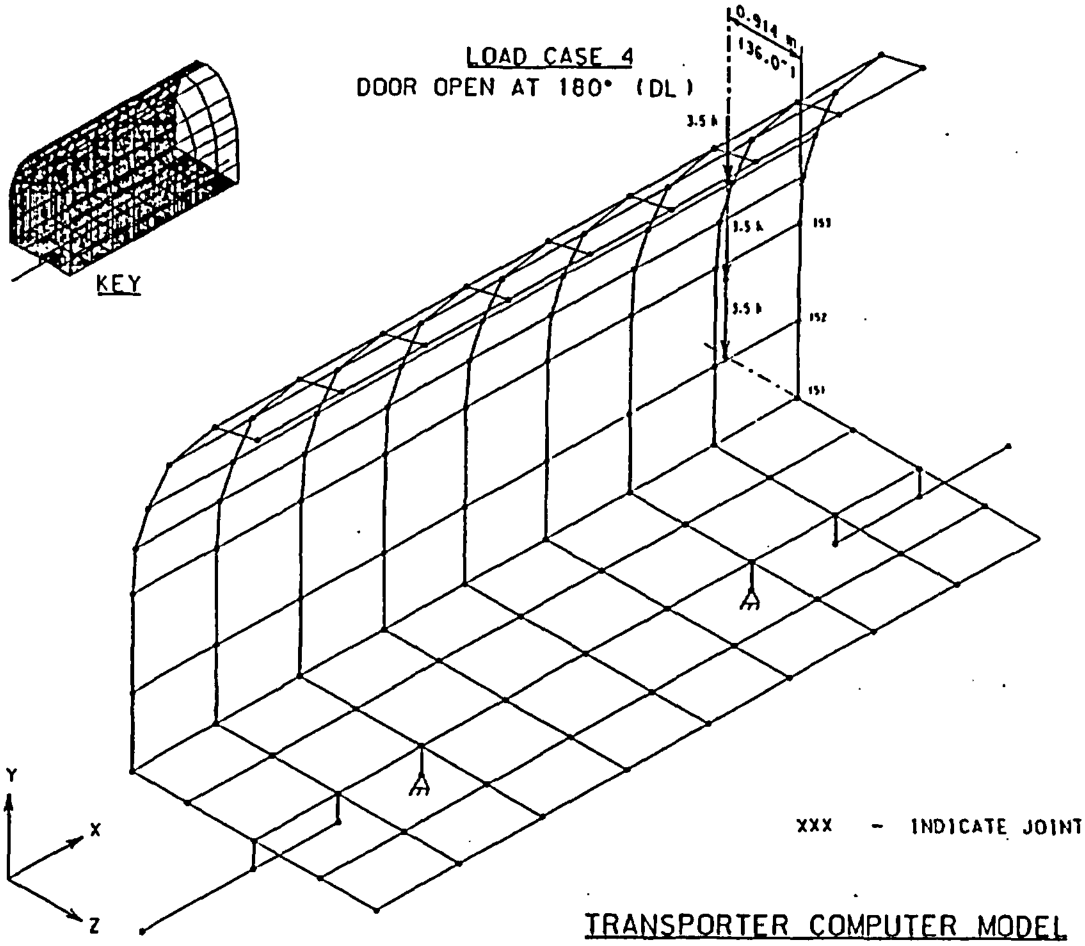


TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

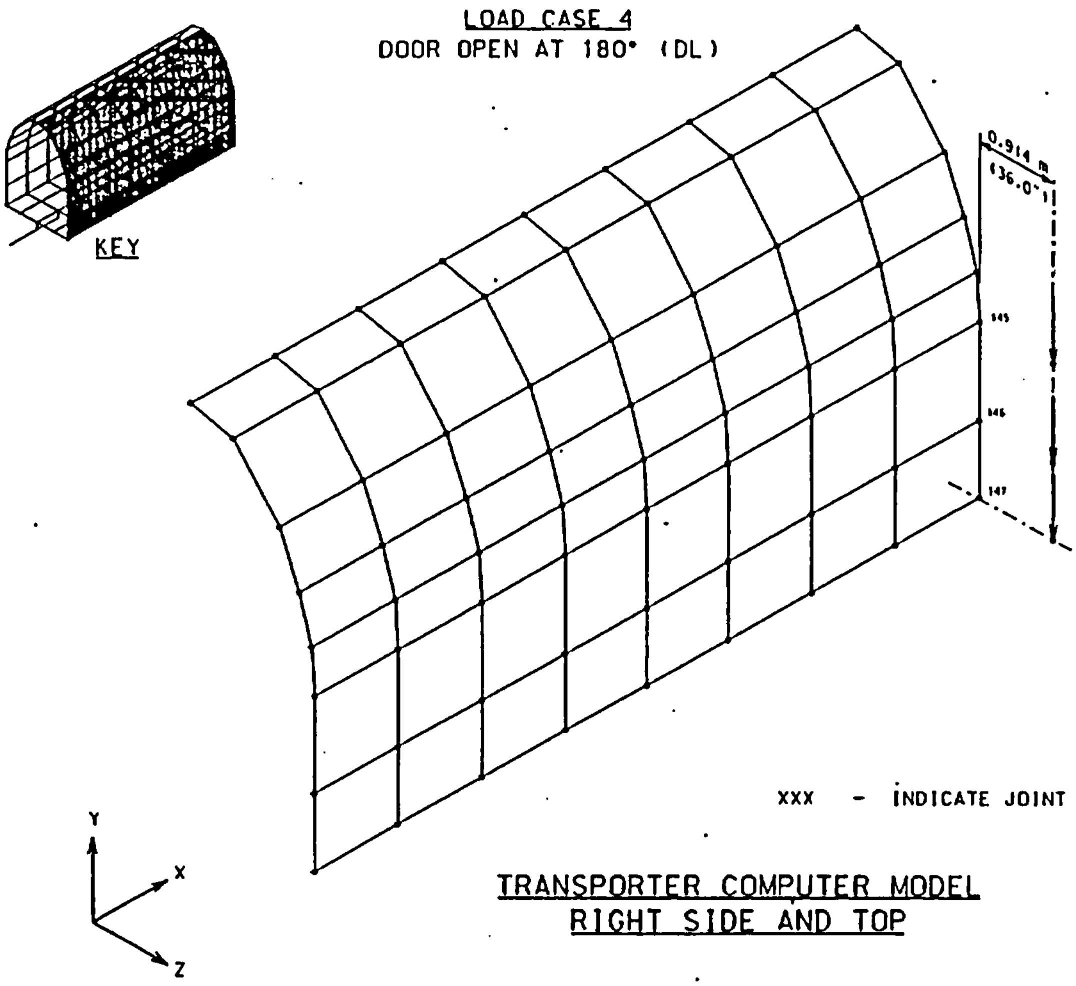
LOAD CASE 3
DOOR OPEN AT 90° (DL)



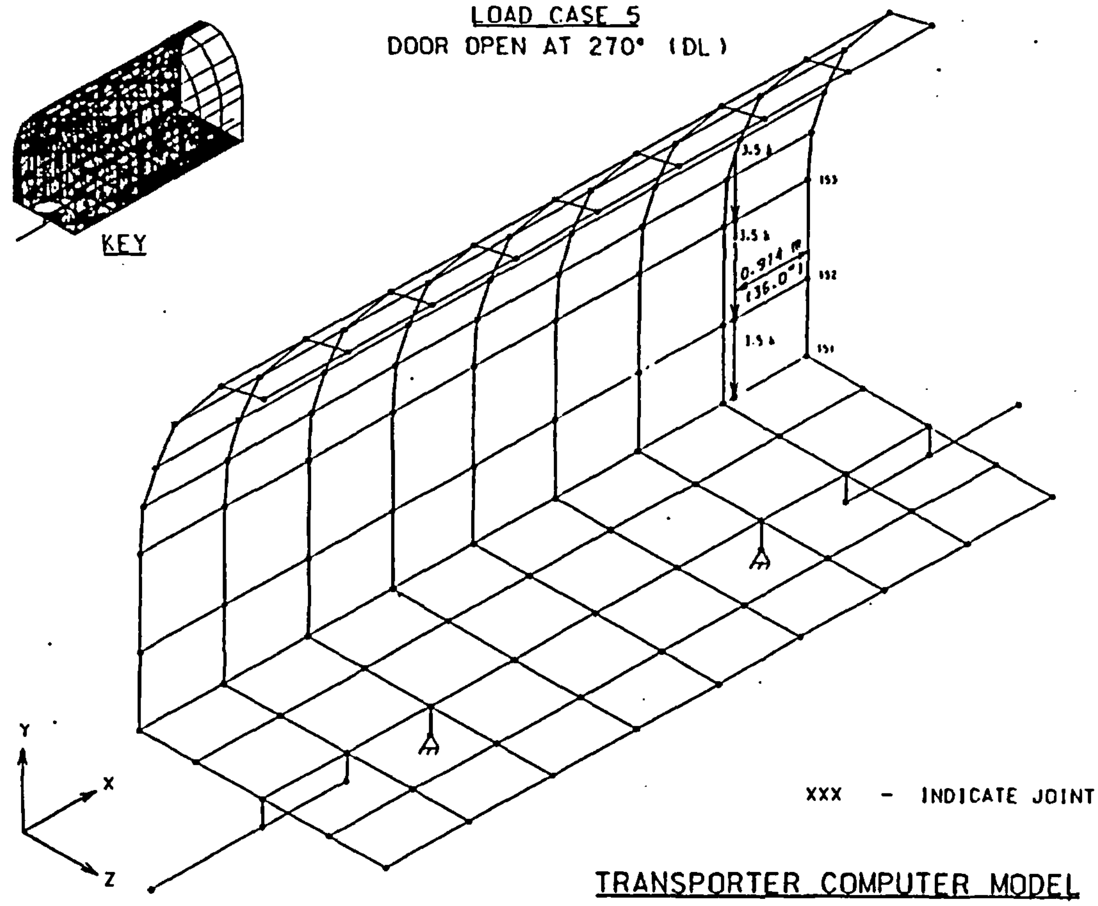
TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP



TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP



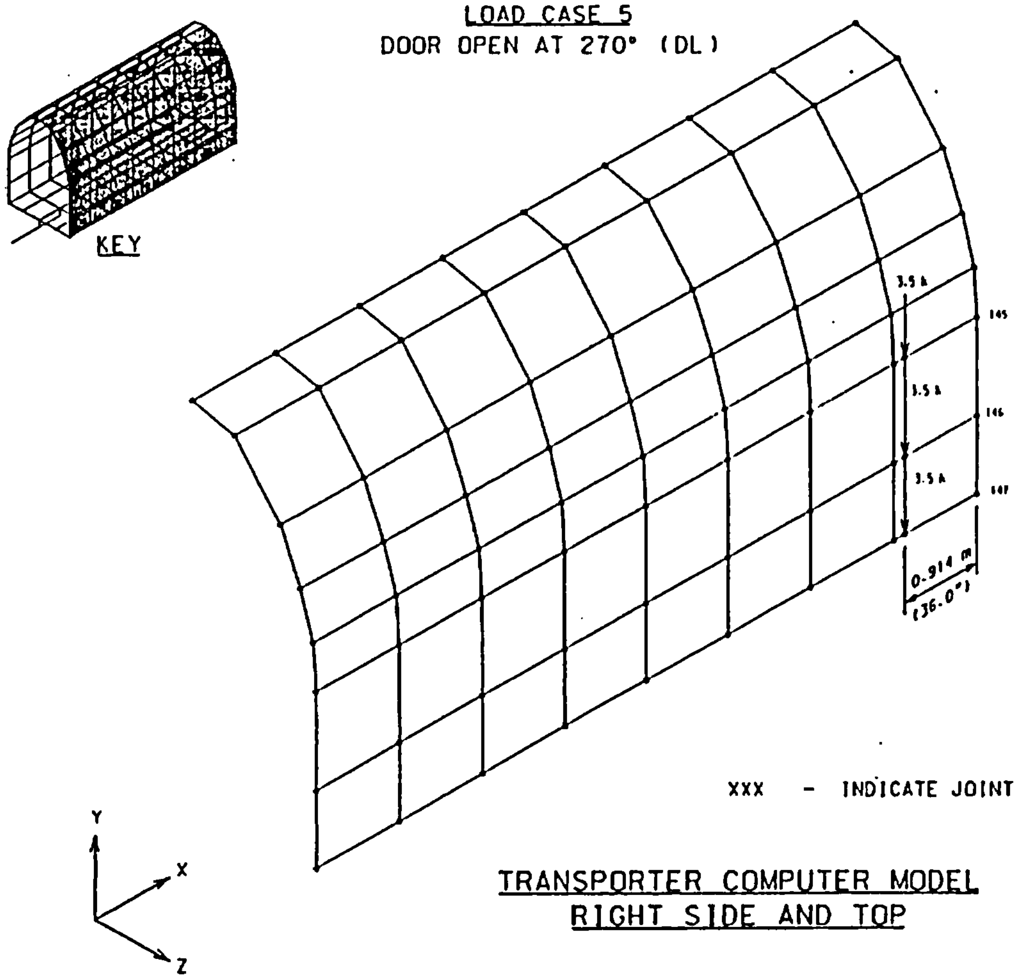
TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP



TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

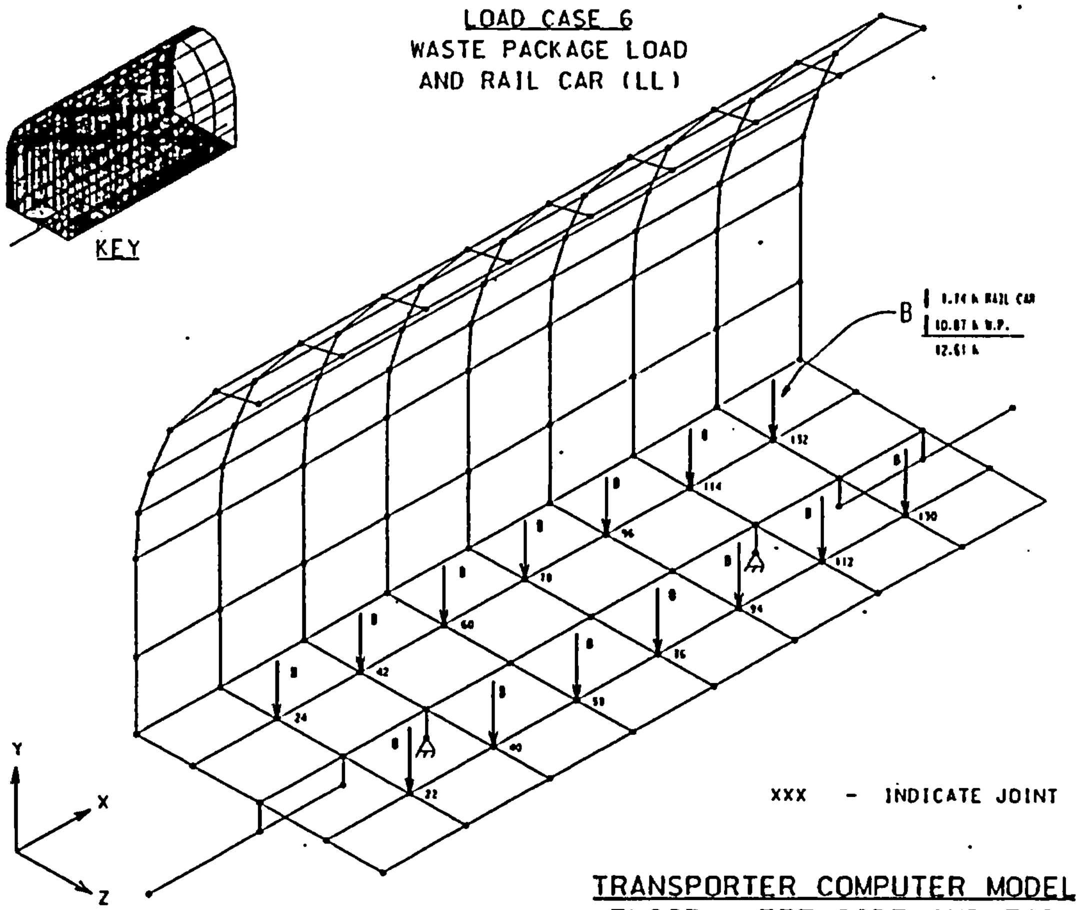
LOAD CASE 5

DOOR OPEN AT 270° (DL)



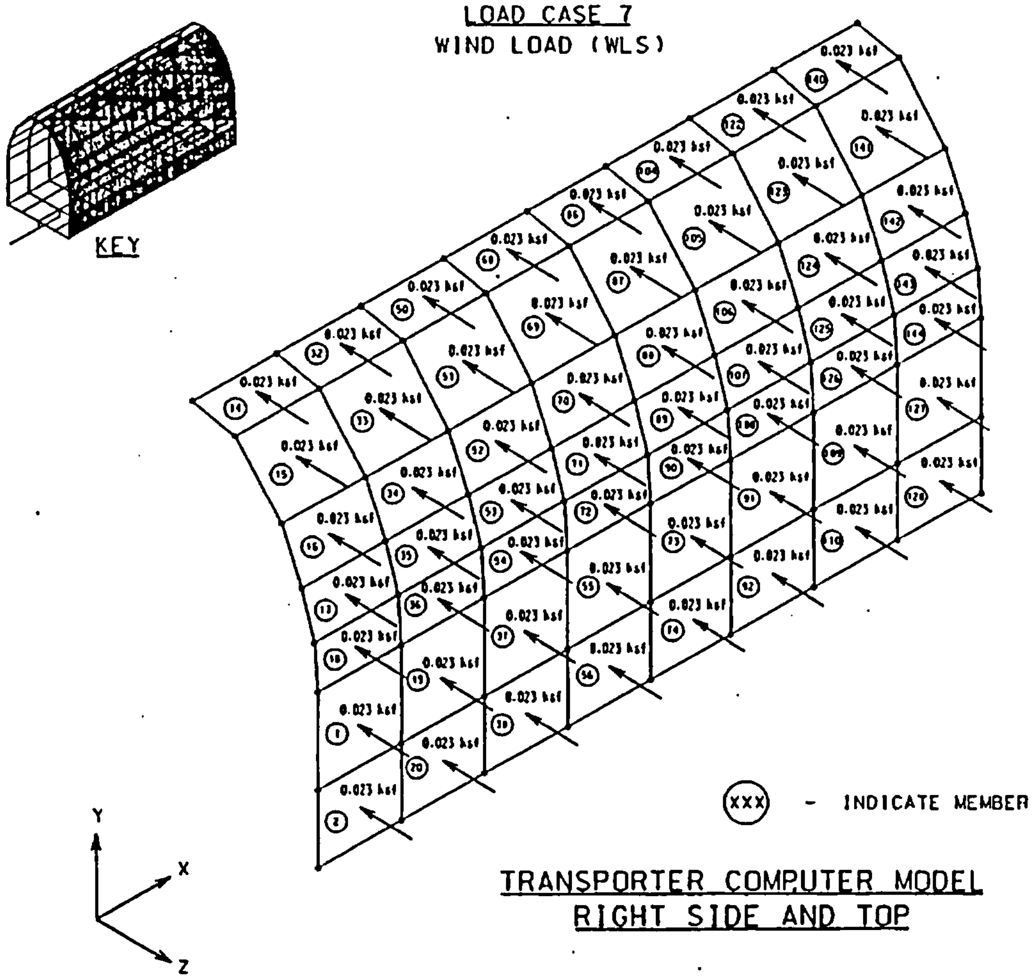
TRANSPORTER COMPUTER MODEL

RIGHT SIDE AND TOP



TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

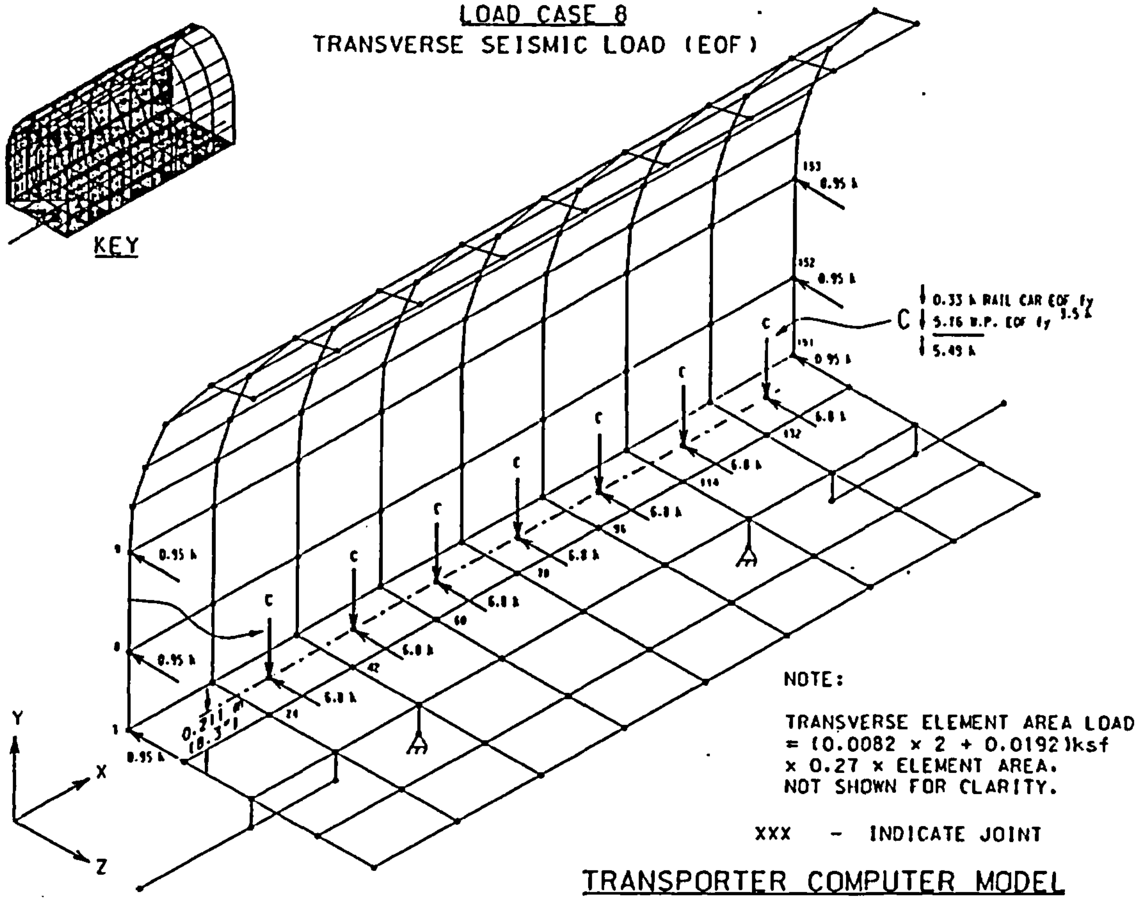
LOAD CASE 7
WIND LOAD (WLS)



TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP

LOAD CASE 8

TRANSVERSE SEISMIC LOAD (EOF)

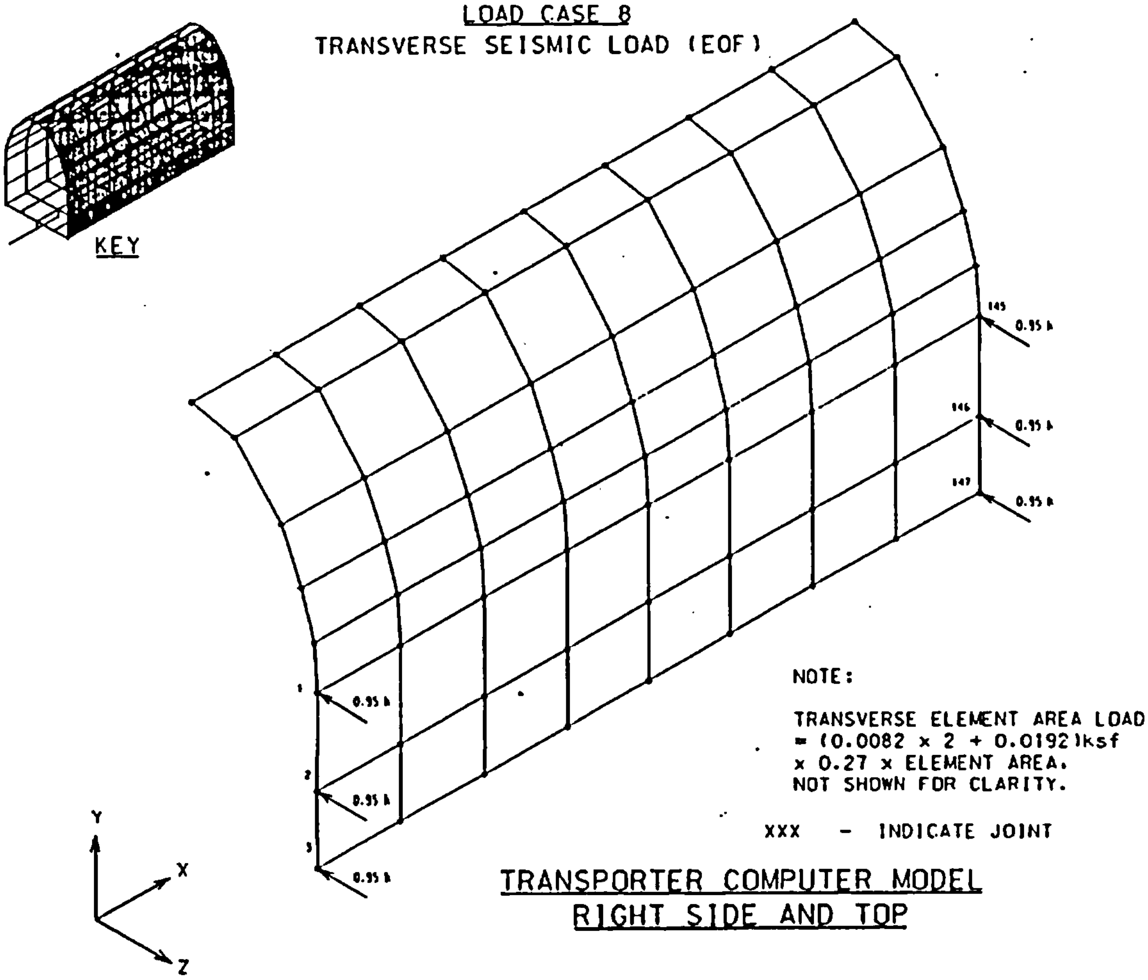


TRANSPORTER COMPUTER MODEL

FLOOR, LEFT SIDE AND TOP

LOAD CASE 8

TRANSVERSE SEISMIC LOAD (EOF)

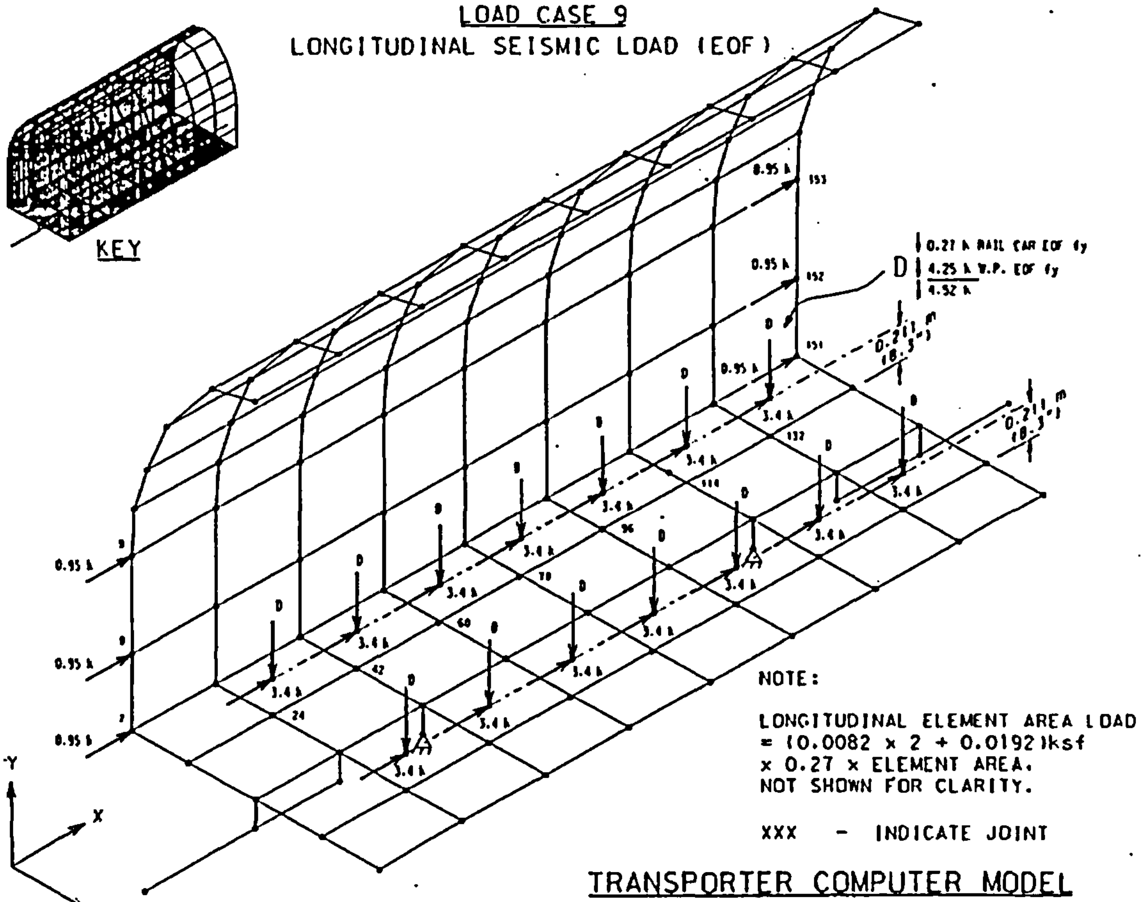


TRANSPORTER COMPUTER MODEL

RIGHT SIDE AND TOP

LOAD CASE 9

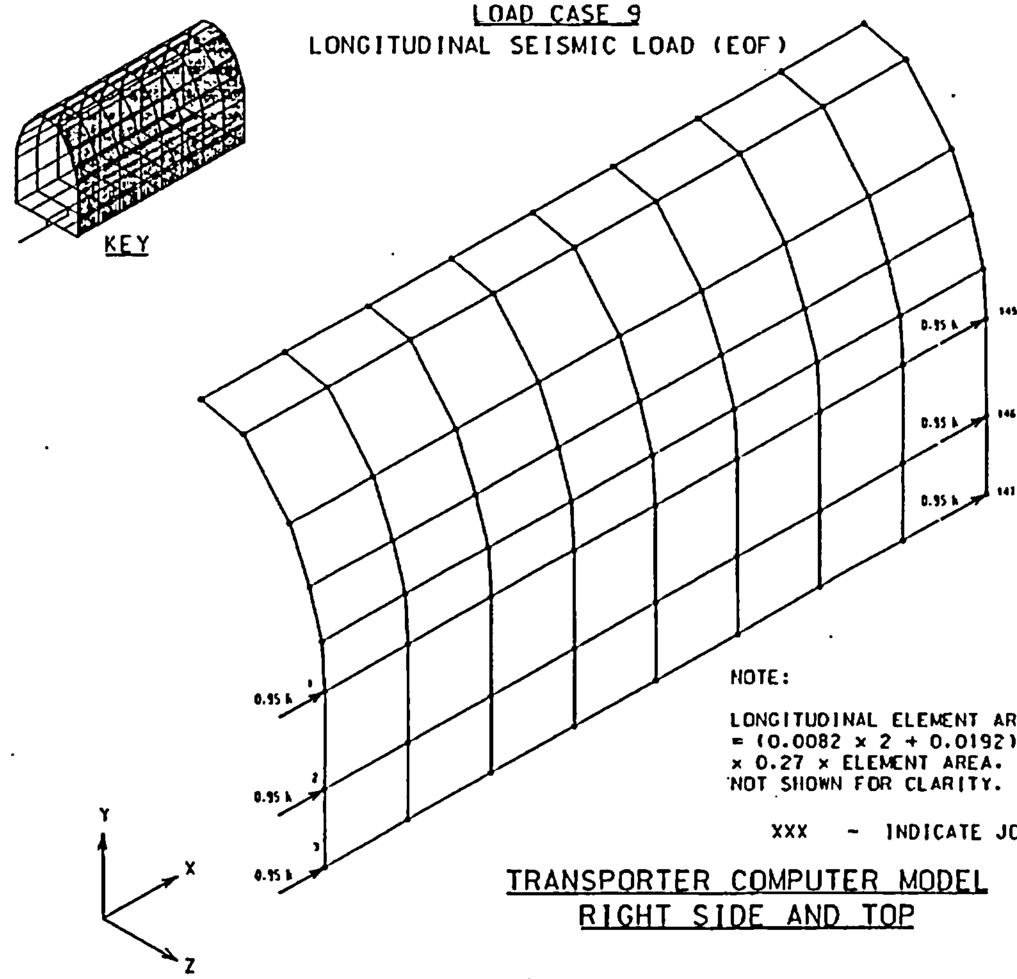
LONGITUDINAL SEISMIC LOAD (EOF)



TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

LOAD CASE 9

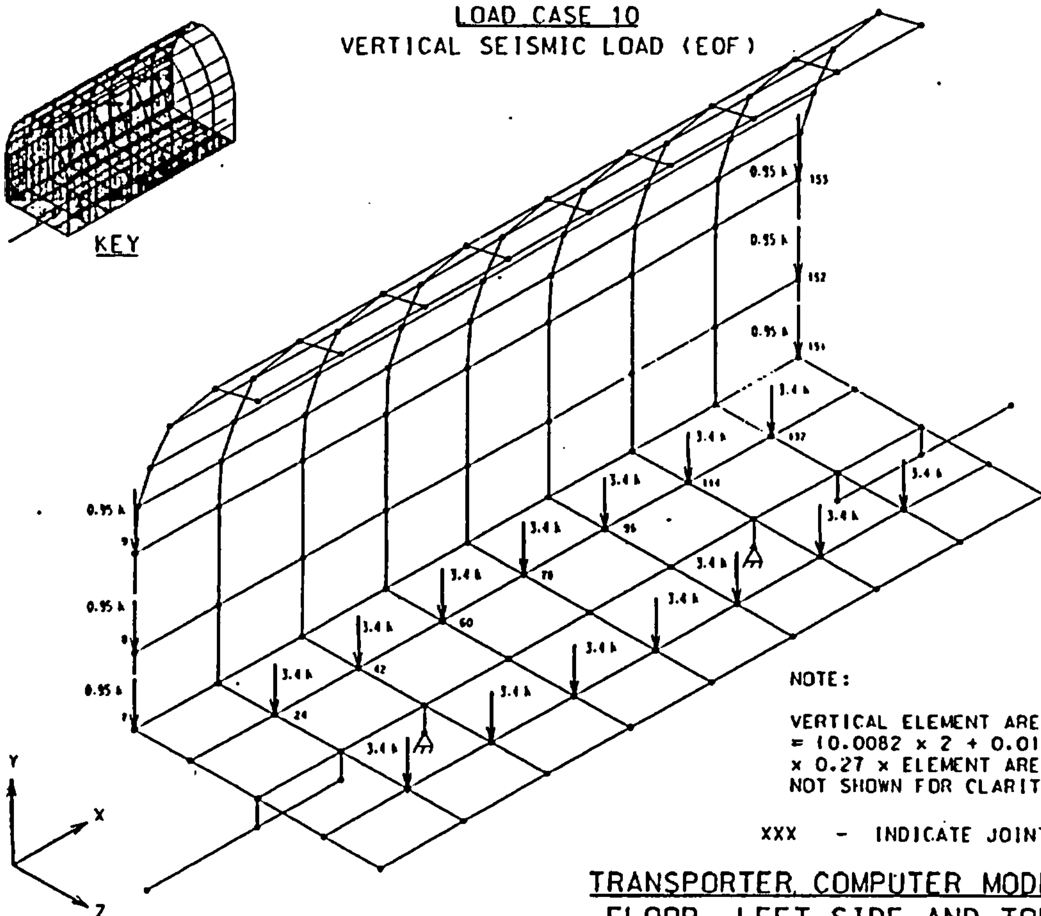
LONGITUDINAL SEISMIC LOAD (EOF)



TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP

LOAD CASE 10

VERTICAL SEISMIC LOAD (EOF)



NOTE:

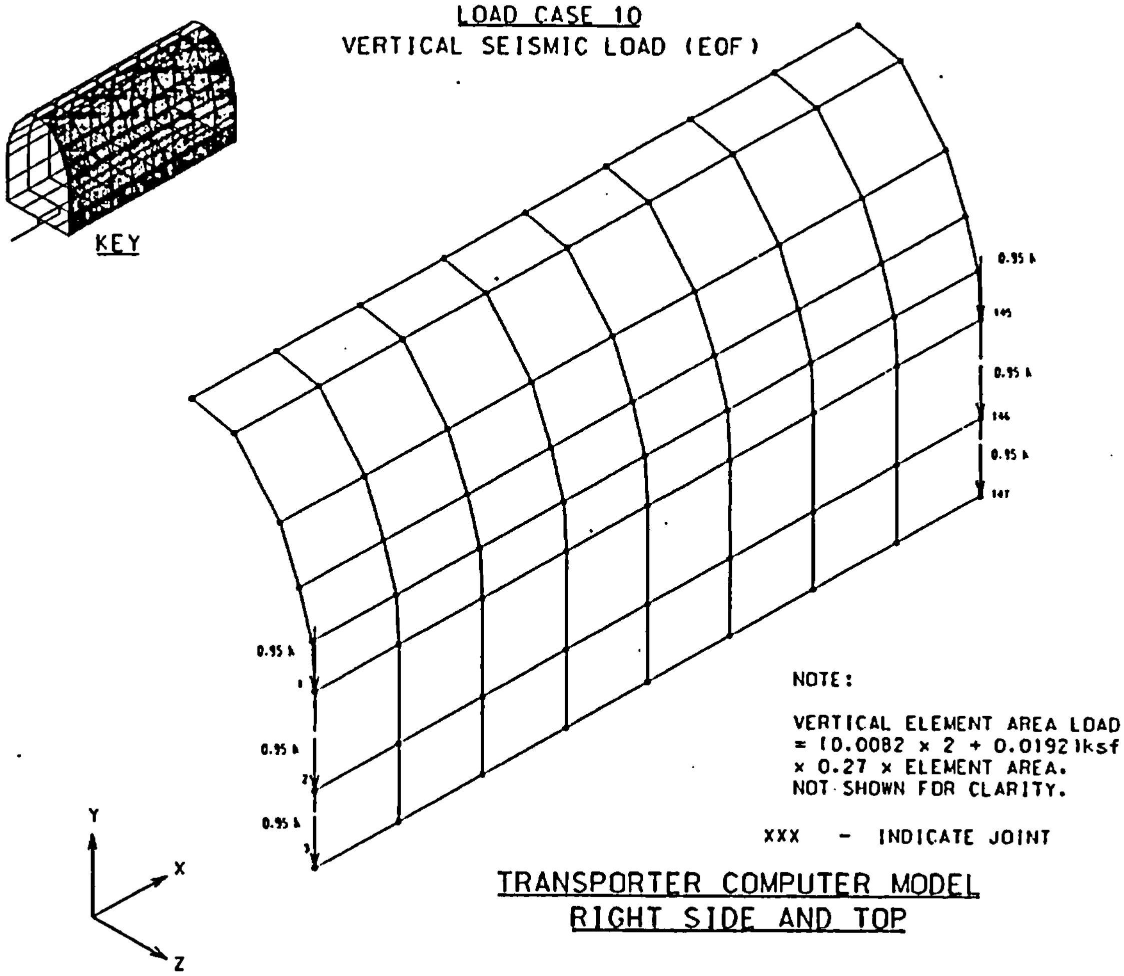
VERTICAL ELEMENT AREA LOAD = (0.0082 x 2 + 0.0192)ksf x 0.27 x ELEMENT AREA. NOT SHOWN FOR CLARITY.

XXX - INDICATE JOINT

TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

LOAD CASE 10

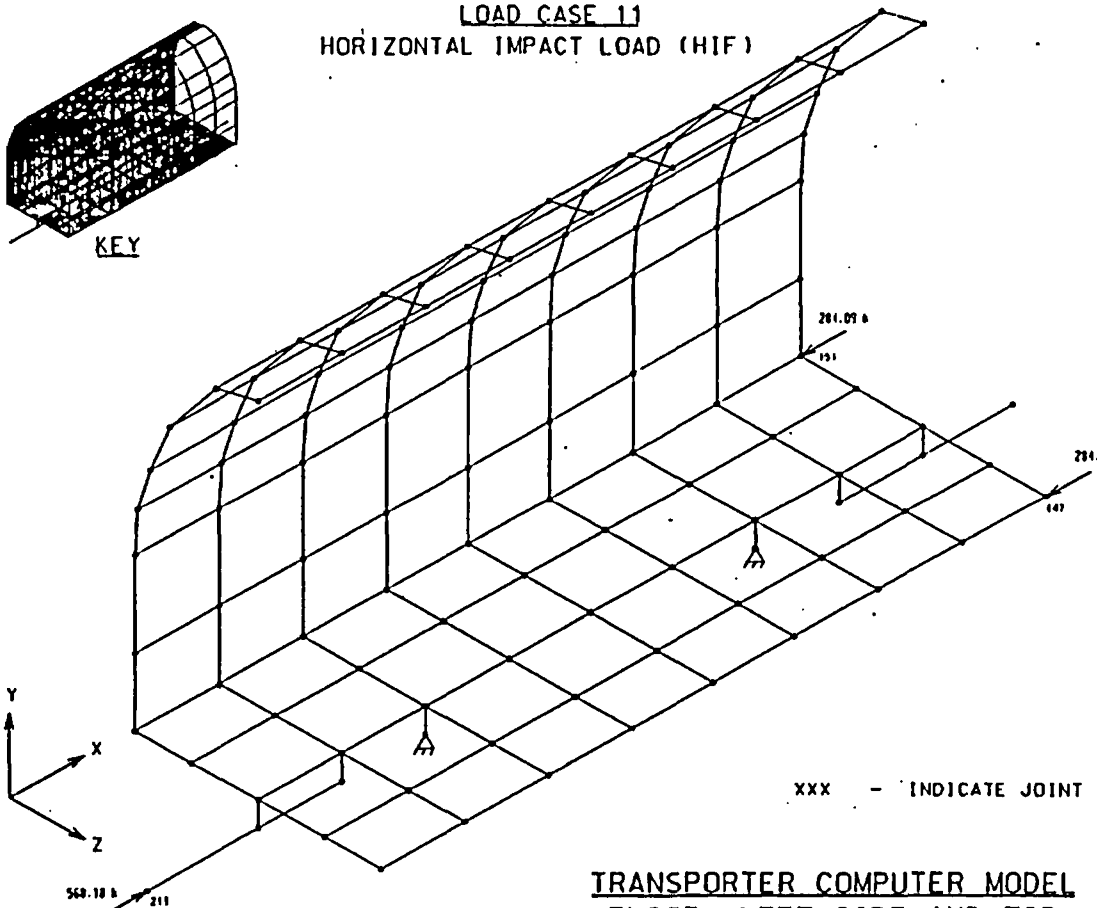
VERTICAL SEISMIC LOAD (EOF)



TRANSPORTER COMPUTER MODEL
RIGHT SIDE AND TOP

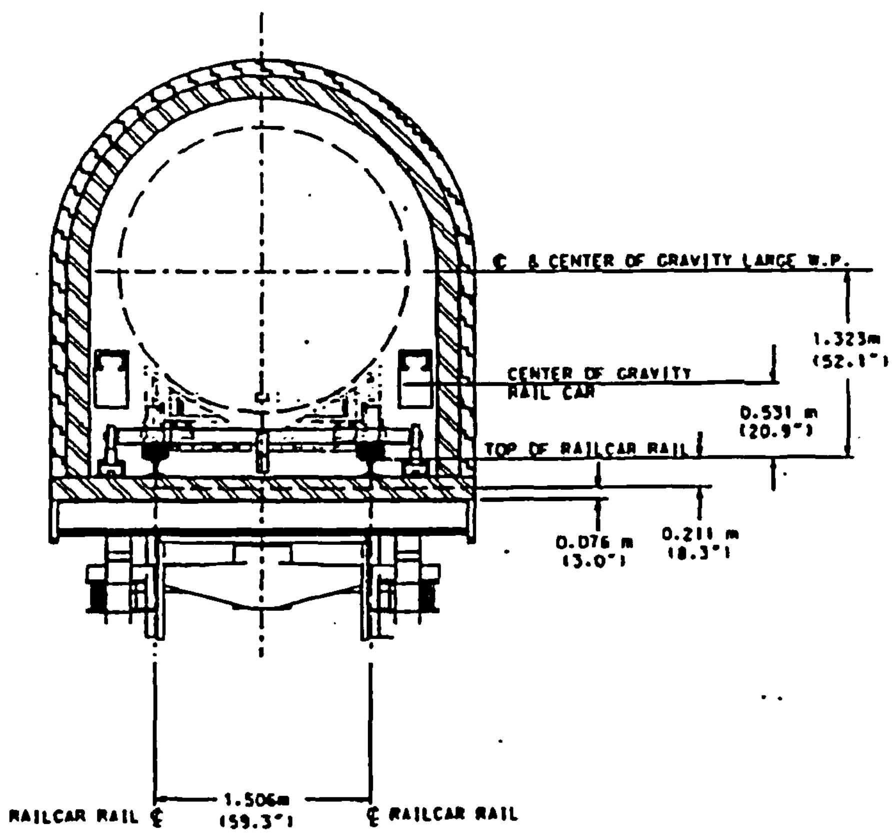
LOAD CASE 11

HORIZONTAL IMPACT LOAD (HIF)



TRANSPORTER COMPUTER MODEL
FLOOR, LEFT SIDE AND TOP

DIMENSION REFERENCE FOR LOAD CASE 8 & LOAD CASE 9



10.0 STABILITY ANALYSIS

STABILITY ANALYSIS:

1. Wind loading (transverse) on Transporter without Waste Package and Rail Car :

A. Overturning safety factor at top of truck:

Dimension of transporter:

Length L = 7.37 M (290.16 Inch), figure 7.2.3

Height H = 3.09 M (121.65 inch), figure I - 1

Wind velocity pressure P =23.0 psf (Attachment I, Section 8.0)

Overturning force F = P X L X H

23.0 psf X 290.16" X 121.65" /144 = 5637.87 LB= 5.64 Kips

Overturn moment at top of truck.

Distance from bottom transporter to top truck = 0.280 M (11.0"), Figure I - 1.

Distance from center of transporter to top of truck = 3.09/2+0.28 =1.825 M (5.99')

Overturn mom. Mot= F X arm = 5.64k X 71.85" /12= 33.77 k-ft

Resisting moment at top of truck

Transport self-weight

Support reaction from STAAD III output file " transpor " (Ref 5.53)

Support joint 201		Support joint 202		
load case 1	Load case 2	load case 1	Load case 2	total
175.76k +	(-10.64)k +	118.83k +	31.64k =	315.59 k

Center to center truck side bearing B = 1.27 M (4.17') Figure I - 1

Resistance mom. Mr = W X B/2 = 315.59k X 50" /12/2= 657.48 k-ft

Safety factor = Mr / M ot = 657.48 / 33.78 = 19.46 > 1.1 OK

B. Overturning safety factor at top of rails:

Distance from center of transporter to top of rails. Figure I - 1.

Moment arm = 3.09 / 2 + 0.98 = 2.525 M (8.28')

Overturn mom. Mot= F X arm = 5.64k X 99.41" /12= 46.72 k-ft

Resisting moment at top of rails

Rail gage B1 = 1.44 m (4.72'), Figure I - 1.

Resistance mom. Mr = W X B1/2 = 315.59k X 56.7" /(2X12) = 745.58 k-ft

Safety factor = Mr / M ot = 745.58 / 46.72 = 15.96 > 1.1 OK

C. Overturning of wind loading in longitudinal direction to transporter by inspection is OK because longitudinal support (spacing of trucks) is larger than transverse support (space between rails) and because longitudinal wind loading is only approximately 40% of transverse wind load (width / length of transporter 2.94 M / 7.37 M).

2. Seismic loading (transverse) on transporter with Waste Package and Rail Car.

A. Overturning safety factor at top of truck:

Distance from transporter gravity center to top of truck = 1.666 M (5.47'),

(Figure I - 1, From STAAD III analysis).

Distance from W P center to top of truck =1.89 M (6.2'), Figure I - 1.

An upper bounding for Rail car center of gravity of 1.098 M (3.60 ft) above top of truck is conservative and reasonable for this analysis. Figure I - 1.

Items	Weight (k)		ground Acc.		Height (ft)		=	M ot (k - ft)
Transporter	315.59k	X	0.27	X	65.6"	/12	=	465.81
WP	152.2k	X	0.27	X	74.4"	/12	=	254.78
Rail Car	24.3k	X	0.27	X	43.2"	/12	=	23.62
					M ot		=	744.21

Resisting moment from top of truck.

B = 1.27 M (4.17')'. Figure I - 1

Transporter	315.59k	X	50"	/(12 X 2) =	657.48	k-ft
WP	152.2k	X	50"	/(12 X 2) =	317.08	k-ft
Rail Car	24.3k	X	50"	/(12 X 2) =	50.63	k-ft
			M_R	=	1025.19	k-ft

Safety factor = Mr / M ot = 1025.19 / 744.21 = 1.38 > 1.1 OK

B. Overturing safety factor at top of rails:

Distance from transporter gravity center to top of rail = 2.366 M (7.67'), Figure I - 1.

Distance from Waste Package center to top of rail = 2.59 M (8.50'), Figure I - 1.

Distance from rail car gravity center to top of rail =1.798 M (5.9'), Figure I - 1.

Items	Weight (k)		ground Acc.	Height (ft)	=	M ot (lb - ft)
Transporter	315.59	X	0.27	X 93.2"	/12 =	661.79
WP	152.2	X	0.27	X 102.0"	/12 =	349.30
Rail Car	24.3	X	0.27	X 70.8"	/12 =	38.71
				M ot	=	1049.80

Resistance moment from top of rail.

Center of center of rail B = 1.44 M (4.72'), Figure I - 1.

Transporter	315.59	X	56.7"	/12/2 =	745.58	
WP	152.2	X	56.7"	/12/2 =	359.57	
Rail Car	24.3	X	56.7"	/12/2 =	57.41	
				M_R =	1162.56	k-ft

Safety factor = Mr / M ot =	1162.56	/ 1049.80	=	1.11	> 1.1	OK

C. Longitudinal seismic load on transporter by inspection is OK because longitudinal supports (truck spacing) is larger than transverse supports (space between rails).

3. Vertical center of gravity above top of rail:

Moment above top of rail:

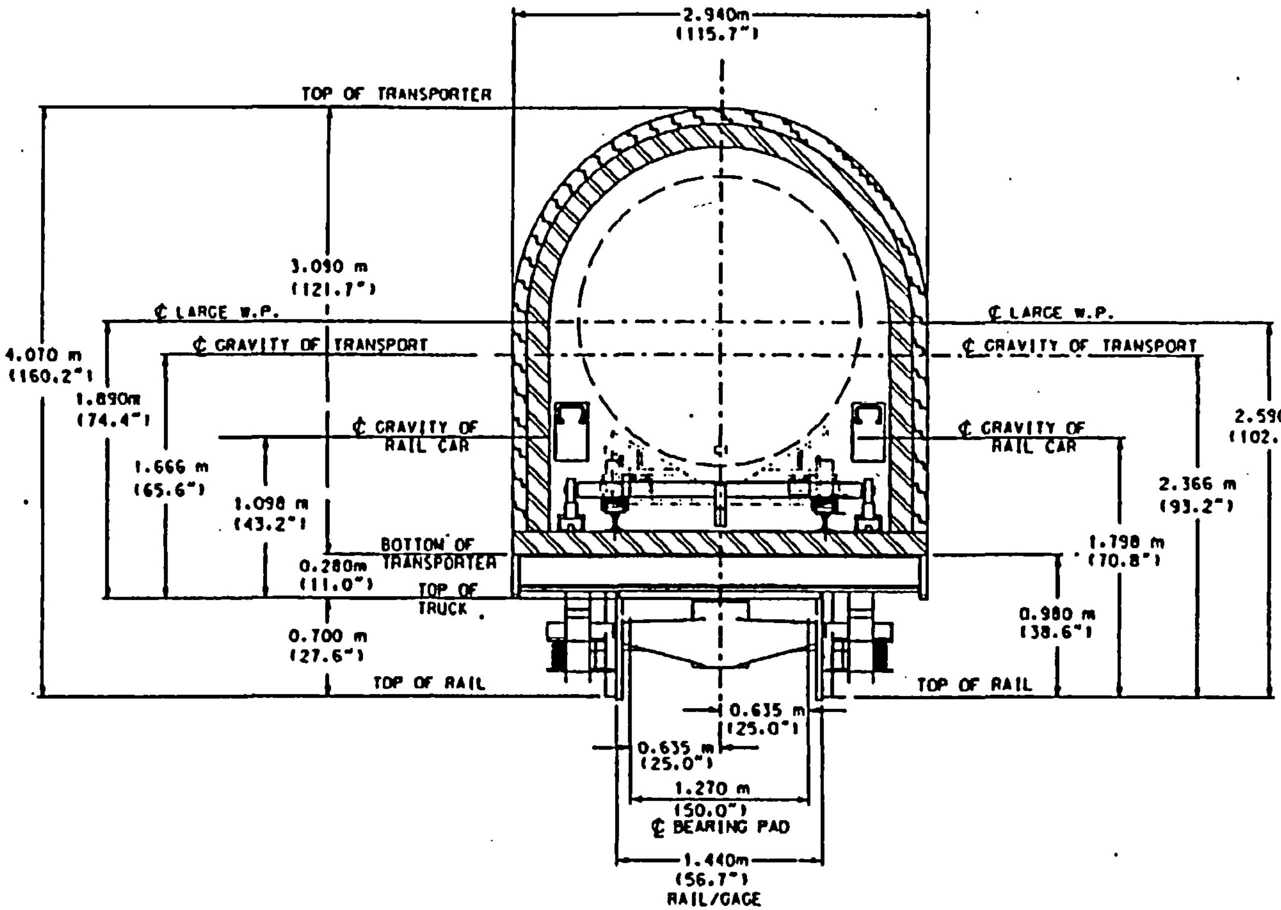
An upper bounding for Truck center of gravity of 0.7 M (2.3') above top of rail is conservative and reasonable for this analysis, Figure I - 1.

Items Weight (k) Height (in) = M (k-ft)

Figure I - 1

Transporter	.315.59	X	93.2"	/12 =	2451.08	
WP	152.2	X	102.0"	/12 =	1293.70	
Rail Car	24.3	X	70.8"	/12 =	143.37	
Truck	22.0	X	27.6"	/12 =	50.60	
total =	514.09 (233.15 MT)				3938.75	
distance =	3938.75	/	514.09	=	7.66	ft

Gravity of center is 7.66 X 12 = 92." from top of rail < 98" (Attachment I, section 5.2) is OK.



TRANSPORTER ARRANGEMENT

CROSS SECTION

NO SCALE

FIGURE I-1
TRANSPORTER ARRANGEMENT
CROSS SECTION

ATTACHMENT II

EMPLACEMENT GANTRY STRUCTURAL ANALYSIS

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TABLE OF CONTENTS

NOTE: STAAD-III Analysis and Stability Analysis portions of this attachment, comprising pages II-68 through II-804, are located in Reference 5.54, Batch No. MOY-970806-05, in electronic media form.

EMPLACEMENT GANTRY STRUCTURAL ANALYSIS

Since this design analysis is for the purpose of viability assessment only, the structural analysis for the Emplacement Gantry will consider only those major loadings considered to most significantly effect the design.

1.0 PRINCIPAL LOADS

1.1 Dead Load (DL) (CMAA 70, Section 3.3.2.1.1.1, Ref. 4.4.5)
The weights of all effective parts of the gantry structure, the machinery parts and the fixed equipment supported by the structure.

Lifting Head Trolley Drive (Ea.) = 100 lb (Attachment V - Section 3.6)
Lifting Head Trolley Screw (Ea.) = 30 PLF (Attachment V - Section 3.5)
Hoist Frame Drive (Ea.) = 800 lb (Attachment V - Section 3.4)
Hoist Frame Screw (Ea.) = 430 lb (Attachment V - Section 3.3)
Gantry Bogie Wheel and Bearing Housing (Ea.) = 340 lb (estimated)
Gantry Bogie Wheel Drive Motor (Ea.) = 400 lb (Attachment V - Section 3.2)
Electrical Cabinet - Shielded (Ea.) = 6.2 kips. Estimated weight based on 2" thick A36 steel (Section 4.3.32)

Estimated Total Self-Weight of Gantry = 45 MT = 99.2 kips (verified by this analysis)

1.2 Trolley Load (TL) (CMAA 70, Section 3.3.2.1.1.2, Ref. 4.4.5)
The weight of the trolley and the equipment attached to the trolley.

Lifting Head (Ea.) = 1000 lb. (estimated for this analysis)
Trolley (Ea.) = as determined by this analysis

1.3 Lifted Load (LL) (CMAA 70, Section 3.3.2.1.1.3, Ref. 4.4.5)
The lifted load consists of the working load (waste packages to be lifted) and the weight of the lifting devices (developed in this analysis) used for handling and holding the working load:

Waste Packages to be Lifted (Section 4.3.10):

WP Type	Diameter (mm)	Length (mm)	WP Loaded Mass (kg)
4 DHLW	1785	3790	30,511
12 PWR UCF	1298	5335	32,236
44 BWR UCF	1604	5335	46,424
21 PWR UCF	1650	5335	50,423
—	1970 (77.6 in.)	5350 (210.6 in.)	69,000 (152.2 kips)
—	1850	5850	69,000

NOTE: Lifted load shall include weights of waste package, hoist frame, and trolley for design of gantry structural frame.

1.4 Dead and Live Load Impact (Vertical Inertia Forces, CMAA 70, Section 3.3.2.1.1.4, Ref. 4.4.5)

Vertical Inertia Forces include those due to the motion of the cranes or crane components and those due to lifting or lowering of the hoist load. These additional loadings are included by the application of a separate factor for the dead load (DLF) and for the hoist load (HLF) by which the vertical acting loads are multiplied.

1.4.1 Dead Load Factor (DLF) (CMAA 70, Section 3.3.2.1.1.4.1, Ref. 4.4.5)

This factor covers only the dead loads of the crane, trolley and its associated equipment.

Gantry travel speed = 144 ft/min. (Attachment V - Section 3.2C)

Use DLF = 1.1

1.4.2 Hoist Live Load Factor (HLF) (CMAA 70, Section 3.3.2.1.1.4.2, Ref. 4.4.5)

This factor applies to the motion of the rated load in the vertical direction, and covers inertia forces, the mass forces due to the sudden lifting of the hoist load, and the uncertainties in allowing for other influences.

HLF = 15% $\leq$ .005 X hoist speed (ft/min) X 100 $\leq$ 50%

Hoisting Speed = 0.74 m/min. = 2.43 ft/min. (V - Section 3.0)

0.005 X 2.43 ft/min. X 100 = 1.2% < Bucket Crane Impact = 50%

In comparison:

Whiting Crane Handbook (Ref. 5.18) pg. 57, Class "E" Crane, Impact = 50%

Use HLF = 0.50

1.5 Tractive Inertia Forces from Drives (IFD) (CMAA 70, Section 3.3.2.1.1.5, Ref. 4.4.5)

The inertia forces occur during acceleration or deceleration of gantry motions. The horizontal load (longitudinal to the gantry runway) due to acceleration or deceleration shall be a percentage of the vertical load and shall be considered as 7.8 times the acceleration or deceleration rate (FT/SEC^2) but not less than 2.5 percent of the vertical (live and dead) load.

Gantry acceleration/deceleration rate = 1.0 ft/sec^2 (V - Section 3.0)

Tractive Force percentage of vertical load (IFD) = 7.8 (1.0 ft/sec^2) = 7.8% > 2.5%

IFD = 7.8% x [full crane dead load and lifted load]

Use IFD = 7.8% (99.2 + 152.2) = 19.6 kips

1.6 Crane Side Thrust Forces

1.6.1 Force Longitudinal to Gantry Runway (LTF)

Horizontal thrust occurs during loading operation when gantry is not positioned directly over the waste package to be lifted. This force is limited to sliding resistance of either the waste package or gantry, whichever is less.

Coefficient of Friction Steel on Steel = 0.25 (PCI, pg. 6-18, Ref. 5.19)

Longitudinal Thrust Force (LTF) = coefficient friction X gantry weight

Use LTF = 0.25 (99.2 kips) = 24.8 kips (maximum)

1.6.2 Force Normal to Gantry Runway (NTF)

AISE Technical Report #13 Section 3.4.1 Table 1, Ref. 5.49. Since the lifted load cannot swing, consider gantry similar to ladle crane:

NTF = 40% of lifted load

Note: Lifted load shall include weight of waste package, hoist frame and trolleys. (The weight of hoist frame and trolleys = approximately 25% gantry weight) (verified by this analysis)

Use NTF = 40% [152.2 + 25% (99.2)] = 70.8 kips

1.7 Out-of-Plumb Forces (OPF) - Longitudinal/Transverse

The static horizontal gravity component due to the out-of-plumb gantry condition shall be applied to all gantry dead and live loads.

1.7.1 Longitudinal to Gantry Runway (OPFL)

Maximum Grade Emplacement Drift = 0.75% (Section 4.3.8)

Rail Installation Tolerance Maximum Vertical Rate of Change = ¼" in 20 ft = 0.10% (CMAA 70 Table 1.4.2-1, Ref. 4.4.5)

Additional Tunnel Construction Tolerance Differential = 1" in 20 ft = 0.42%

Total Longitudinal Out-of-Plumb = 0.75 + 0.10 + 0.42 = 1.27%

Longitudinal Out-of-Plumb (OPF) = 1.27% x [full crane dead load and lifted load]

Use OPF-Longitudinal = 1.27% (99.2 + 152.2) = 3.2 kips

1.7.2 Transverse to Gantry Runway OPFT

Rail Installation Tolerance Rail to Rail = 3/16" (CMAA 70 Table 1.4.2-1, Ref. 4.4.5)

Additional Tunnel Construction Tolerance Differential = 1"

Total Transverse Out-of-Plumb = 1 3/16" in 104.17" (2.646 m) = 1.14%

Transverse OPF = 1.14% X [full crane dead load and lifted load]

Use OPF-Transverse = 1.14% (99.2 + 152.2) = 2.9 kips

2.0 ADDITIONAL LOADS

2.1 Forces Due to Skewing (SK) (CMAA 70, Section 3.3.2.1.2.2, Ref. 4.4.5)
When two wheels (or two bogies) roll along a rail, the horizontal forces normal to the rail, and tending to skew the structure shall be obtained by multiplying the vertical load exerted on each wheel (or bogie) by coefficient S_{sk} which depends upon the ratio of the span to the wheel base.

Gantry Ratio $= \frac{SPAN}{WHEELBASE} = \frac{2646\ mm}{1000\ mm} = 2.65$; $S_{sk} = 5\%$

SK = 5% X [full crane dead and lifted load]

Use SK = 5% (99.2 + 152.2) = 12.6 kips total skewing force

2.2 Horizontal Racking Force to Gantry (HRF)
Consider drive wheels operating one side of gantry or gantry towed from one side of gantry, and on other side gantry drive wheels are locked. Note: One drive wheel per bogie

Coefficient Friction = 0.25 (PCI, Table 6.6.1, pg. 6-18, Ref. 5.19)

Maximum Vertical Load per Drive Wheel = (99.2 + 152.2) ÷ 8 wheels = 31.43 kips

Maximum HRF = 0.25 (31.43 kips) (2 drive wheels) = 15.7 kips

Use HRF = 15.7 kips each side gantry

2.3 Vertical Racking Force to Gantry (VRF)
Rail Installation Tolerance Rail to Rail = ±3/16" (CMAA 70 Table 1.4.2-1, Ref. 4.4.5)

Total Vertical Relative Displacement = 2 (3/16") = 0.375"

The maximum VRF will result from the stress developed during the vertical relative displacement: On same side gantry frame, displace front bogie up 0.2" and rear bogie down 0.2"

2.4 Temperature Forces

2.4.1 Maximum Allowable Air Temperature in Emplacement Drifts During: (Controlled Design Assumption #DCSS-019, Ref. 5.1)

Emplacement = 50°C

Retrieval = 50°C

Use Δ temp = 50°C = 122°F

Consider gantry unrestrained from normal expansion, therefore, neglect thermal stresses in gantry structure

2.4.2 Effect of Heat on Structural Steel (AISC M016-89, Part 6 pg 6-3 & 6-5, Ref. 4.4.1)
Short-time elevated-temperature tensile tests on the constructional steels permitted by the AISC Specification indicate that the ratios of the elevated-

temperature yield and tensile strengths to their respective room-temperature strength values are reasonably similar at any particular temperature for the various steels in the 300°F to 700°F range, except for variations due to strain aging.

The mechanical properties of steels are largely unaffected by heating operations, provided that the maximum temperature does not exceed 1100°F for quenched and tempered alloy steels, and 1300°F for other steels.

3.0 EXTRAORDINARY LOADS

3.1 Collision Forces (CF) (CMAA 70, Section 3.3.2.1.3.2, Ref. 4.4.5)

Special loading of the crane structure resulting from the bumper stops, shall be calculated with the crane at 0.4 times the rated speed assuming the bumper system is capable of absorbing the energy within its design stroke. Load suspended from lifting equipment and free oscillating load need not be taken into consideration.. Where the load cannot swing, the bumper effect shall be calculated in the same manner, taking into account the value of the load.

AISE Tech. Report #13, Section 7.6.2, Ref. 5.49

Maximum Allowable Deceleration Rate = 16 ft per sec²
Gantry + waste package weight = 99.2 + 152.2 = 251.4 kips
Impact gantry weight per side = 0.5 (251.4 kips) = 125.7 kips

$$\text{Maximum Stopping Force per Side} = 125.7 \text{ kips} \left(\frac{16 \text{ ft per sec}^2}{32 \text{ ft per sec}^2}\right) = 62.9 \text{ kips}$$

Use CF = 62.9 kips each side gantry

3.2 Seismic (EQF)

Horizontal and Vertical Acceleration = 0.27 g (Ref. 5.9, Pg. 13)

The application of the vertical inertia force factors of DLF and HLF results in a combined inertia force factor approximately = 1.34

$$\left(\frac{1.1\ (99.2^{k}) + 1.5\ (152.2^{k})}{99.2 + 152.2^{kips}} = 1.34\right),$$

which is greater than the combined gravity and vertical seismic acceleration of 1.27 (1+0.27g = 1.27); therefore, vertical acceleration is not considered in this analysis.

3.3 Wind: Since gantry operations take place underground, wind loading to gantry during waste package handling operations is not applicable. Wind loading can occur to gantry during transport on gantry carrier but is neglected in this analysis.

4.0 LOAD COMBINATION; CMAA 70, Section 3.3.2.4, Ref. 4.4.5

Gantry structural frame is analyzed for the waste package/hoist beam at three positions (II-Section 10.0, Figure II-6):

Low Position = hoist beam and WP in lowest position.

Medium Position = hoist beam at mid-height gantry column.

High Position = hoist beam and WP in highest position.

The combined stresses shall be calculated for the following design load combinations for each hoist beam position:

4.1 Load Combination 1: Crane in regular use under principal loading (Stress Level 1)

DL(DLF) + TL(DLF) + LL(1+HLF) + IFD + OPF

4.2 Load Combination 2: Crane in regular use under principal and additional loading (Stress Level 2)

DL(DLF) + TL(DLF) + LL(1+HLF) + IFD + OPF + LTF + NTF (or SK, whichever is greater) + HRF + VRF

4.3 Load Combination 3: Extraordinary Loads (Stress Level 3)

4.3.1 Crane subjected to stopping collision force

DL + TL + LL + OPF + VRF + CF

4.3.2 Crane subjected to seismic force

DL + TL + LL + IFD + OPF + VRF + EQF

5.0 ALLOWABLE STRESSES

5.1 Maximum Allowable Stresses in Structural Steel Members

CMAA 70, Section 2.7, Ref. 4.4.5: Consider emplacement gantry to be of equivalent Service Class "F" because:

- Performs critical tasks
- Must provide highest reliability

CMAA 70, Tables 3.4.7-1 and 3.4.7-2A, Ref. 4.4.5
Service Class "F"
Joint Category B; allowable fatigue stress range = 17.0 ksi

Note: Joint Category B is selected because gantry is designed principally of built-up box and channel sections. (CMAA 70, Table 3.4.7-2A)

Plus Approximate Gantry Dead Load Stress (by this analysis) = 1.0 ksi
Use Total Allowable Stress = 18.0 ksi

AISC M016-89 (F1-1 & F3-1) pg. 5-45 & 5-48, Ref. 4.4.1; $F_b = 0.66\ Fy$

For STAAD analysis, limit the allowable stress range for fatigue by reducing Fy for ASTM A-36 steel:

Use reduced $Fy = \frac{F_b}{0.66} = \frac{18.0\ ksi}{0.66} = 27.3$ ksi, ASTM A-36 steel

5.2 Allowable Stress Level; CMAA 70, Section 3.4, Ref. 4.4.5
The allowable stress increase is a percentage increase above the allowable combined stress level under principal loading, Stress Level 1, Load Combination 1.

Load Combination 1, Stress Increase $= \frac{0.60}{0.60} = 1.0$; 0% stress increase

Load Combination 2, Stress Increase $= \frac{0.66}{0.60} = 1.1$; 10% stress increase.

Load Combination 3, Stress Increase $= \frac{0.75}{0.60} = 1.25$; 25% stress increase

6.0 ALLOWABLE DISPLACEMENTS

6.1 Gantry Girders

Description	Displacement
Vertical Deflection (ASME-NOG-1-1995, Section NOG-4341, Ref. 4.4.2 and CMAA 70, Section 3.5.5, Ref. 4.4.5) Note: The more stringent allowable deflection is used to ensure highest reliability.	Δ_{D+L} w/o Impact $\leq \frac{Span}{1000}$
Lateral Displacement (CMAA 70, Section 1.4, Ref. 4.4.5)	$\Delta \leq \frac{Span}{400}$

6.2 Horizontal Displacement - Bogie Column (ASME NOG-1-1995, Section NOG-4345, Ref. 4.4.2)
Side thrust at the runway rail due to gantry leg spreading caused by girder span or cantilever deflection or thermal movement shall be held at an acceptable level by

providing adequate clearance between the rail head and the wheel flanges, or by means of other design features incorporated into the gantry structure.

Description	Displacement
Allowable gantry leg spreading	use 0.3"

7.0 GANTRY STABILITY (ASME NOG-1-1995, Section NOG-4457, Ref. 4.4.2)

The gantry stability safety factors against overturning of NOG-4457 when subjected to the load combinations for crane operational loads and extreme environmental loads (ASME NOG-1-1995, Section NOG-4140) are by inspection comparable and appropriate for use with the CMAA 70, Section 3.3.2.4, Ref. 4.4.5 load combinations.

Normal Operating Condition
Overturning Safety Factor ≥ 1.5 (Load Combination #1 & 2)

Extraordinary (Extreme Environmental) Loading Condition
Overturning Safety Factor ≥ 1.1 (Load Combination #3)

8.0 MATERIALS

Following are the structural materials to be considered in the design of the Emplacement Gantry (CMAA 70, Sections 3.1 and 3.2, Ref. 4.4.5)

8.1 Structural Steel:

ASTM A36	Plates, bars and shapes

8.2 Weld Material:

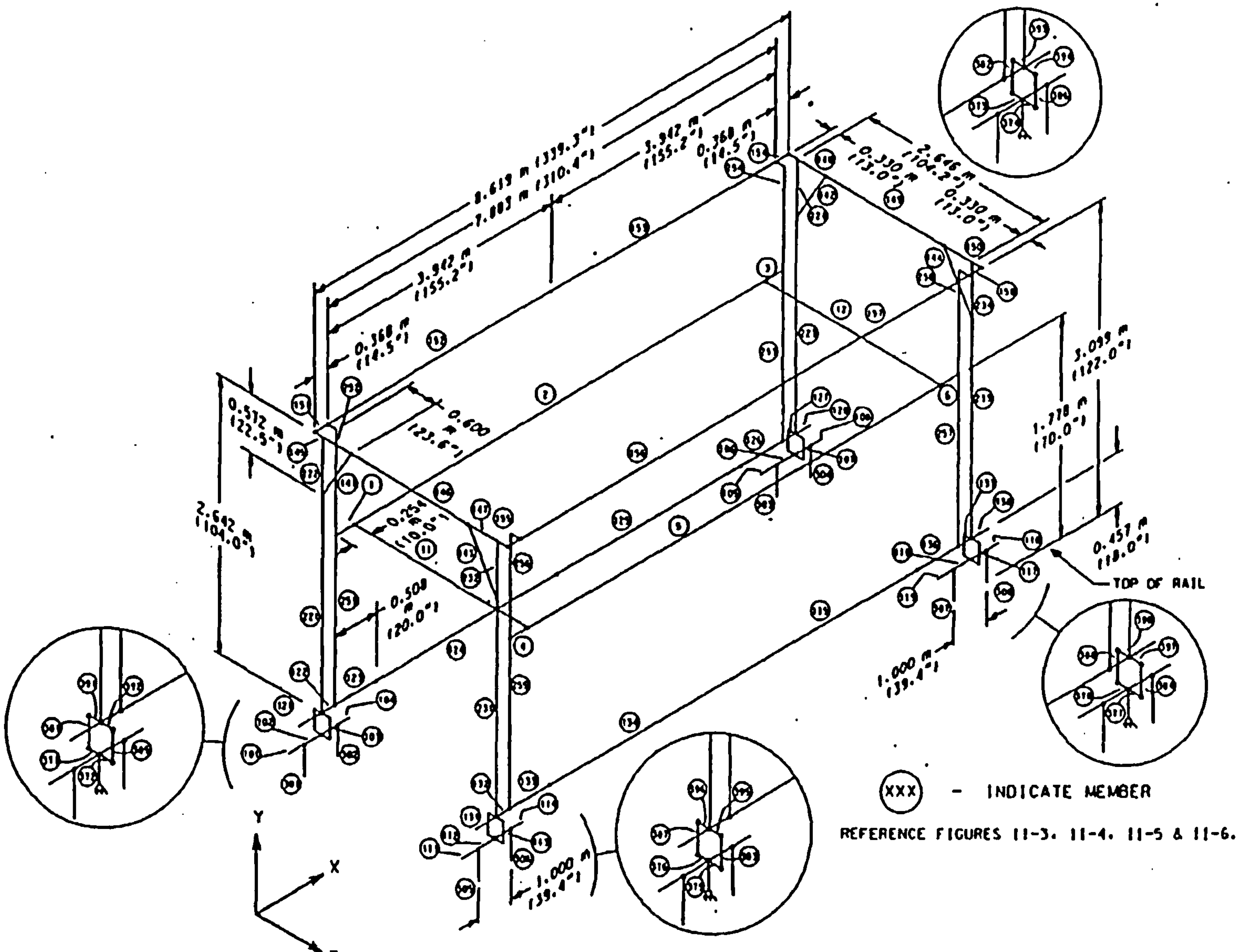
AWS D14.1	E70XX Electrodes

9.0 DESIGN METHODS

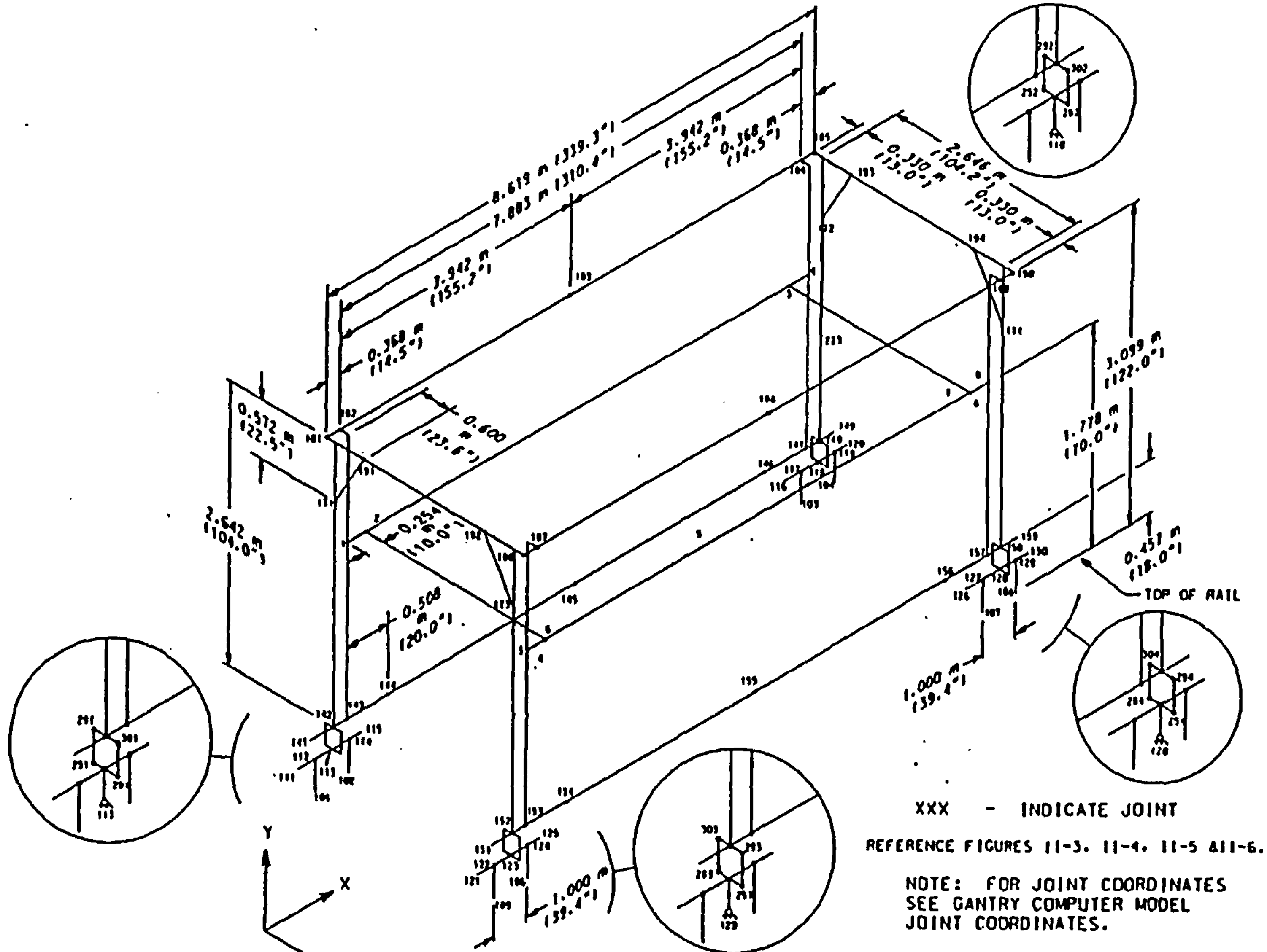
The Waste Package Emplacement Gantry will be designed using static load analysis. STAAD-III computer software will be used to perform the stress analysis for the gantry structural frame. STAAD-III facilities for steel design and code checking will be based on the AISC-ASD code.

10.0 STAAD-III MODEL

STAAD-III model diagrams indicate model configurations with model members and joints identified. The basis for each computer model is shown in Figures II-1, II-2, II-3, II-4, II-5, and II-6. Member centerline offsets are not shown in model diagrams for clarity, but are identified in the STAAD-III input and output files.



GANTRY COMPUTER MODEL- MEMBERS



GANTRY COMPUTER MODEL- JOINTS

GANTRY COMPUTER MODEL JOINT COORDINATES

No. of Joint	X	Y	Z
101	-34.2	0.0	-52.1
102	5.2	0.0	-52.1
103	305.2	0.0	-52.1
104	344.6	0.0	-52.1
105	-34.2	0.0	52.1
106	5.2	0.0	52.1
107	305.2	0.0	52.1
108	344.6	0.0	52.1
111	-50.0	13.0	-52.1
112	-34.2	13.0	-52.1
113	-14.5	13.0	-52.1
114	5.2	13.0	-52.1
115	20.0	13.0	-52.1
116	290.0	13.0	-52.1
117	305.2	13.0	-52.1
118	324.9	13.0	-52.1
119	344.6	13.0	-52.1
120	360.0	13.0	-52.1
121	-50.0	13.0	52.1
122	-34.2	13.0	52.1
123	-14.5	13.0	52.1
124	5.2	13.0	52.1
125	20.0	13.0	52.1
126	290.0	13.0	52.1
127	305.2	13.0	52.1
128	324.9	13.0	52.1
129	344.6	13.0	52.1
130	360.0	13.0	52.1
141	-30.0	18.0	-52.1
142	-14.5	18.0	-52.1
143	0.0	18.0	-52.1
144	20.0	18.0	-52.1
145	155.2	18.0	-52.1
146	290.0	18.0	-52.1
147	310.4	18.0	-52.1
148	324.9	18.0	-52.1
149	340.0	18.0	-52.1

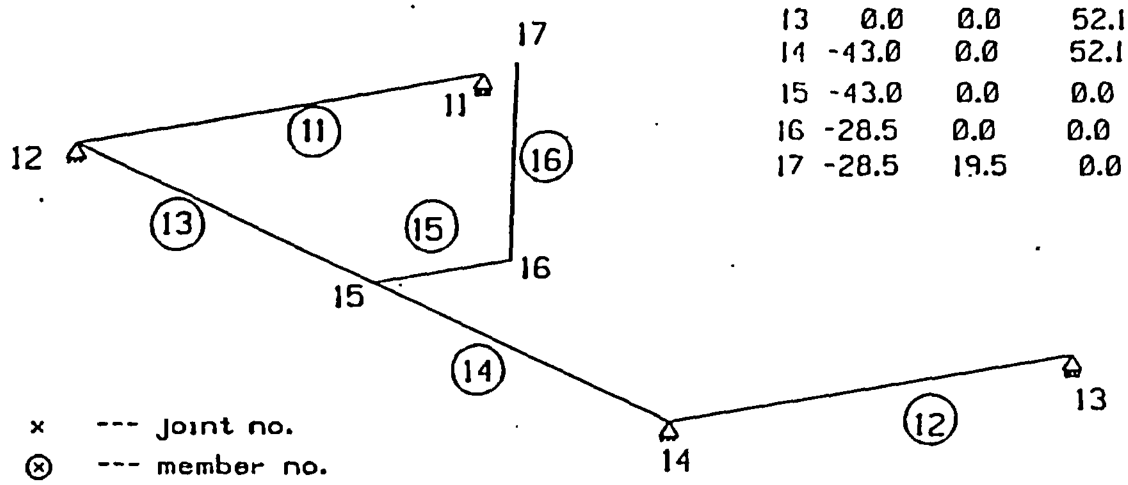
No. of Joint	X	Y	Z
151	-30.0	18.0	52.1
152	-14.5	18.0	52.1
153	0.0	18.0	52.1
154	20.0	18.0	52.1
155	155.2	18.0	52.1
156	290.0	18.0	52.1
157	310.4	18.0	52.1
158	324.9	18.0	52.1
159	340.0	18.0	52.1
171	-14.5	100.5	-52.1
172	324.9	100.5	-52.1
173	-14.5	100.5	52.1
174	324.9	100.5	52.1
181	-14.5	122.0	-52.1
182	0.0	122.0	-52.1
183	155.2	122.0	-52.1
184	310.4	122.0	-52.1
185	324.9	122.0	-52.1
186	-14.5	122.0	52.1
187	0.0	122.0	52.1
188	155.2	122.0	52.1
189	310.4	122.0	52.1
190	324.9	122.0	52.1
191	-14.5	122.0	-28.5
192	-14.5	122.0	28.5
193	324.9	122.0	-28.5
194	324.9	122.0	28.5
251	-14.5	13.0	-58.6
252	324.9	13.0	-58.6
253	-14.5	13.0	58.6
254	324.9	13.0	58.6
281	-14.5	13.0	-45.6
282	324.9	13.0	-45.6
283	-14.5	13.0	45.6
284	324.9	13.0	45.6
291	-14.5	18.0	-58.6
292	324.9	18.0	-58.6
293	-14.5	18.0	58.6
294	324.9	18.0	58.6

No. of joint	X	Y	Z
301	-14.5	18.0	-45.6
302	324.9	18.0	-45.6
303	-14.5	18.0	45.6
304	324.9	18.0	45.6
Hoist in high position			
1	0.0	119.9	-52.1
2	10.0	119.9	-52.1
3	300.4	119.9	-52.1
4	310.4	119.9	-52.1
5	0.0	119.9	52.1
6	10.0	119.9	52.1
7	300.4	119.9	52.1
8	310.4	119.9	52.1
Hoist in mid. position			
1	0.0	70.0	-52.1
2	10.0	70.0	-52.1
3	300.4	70.0	-52.1
4	310.4	70.0	-52.1
5	0.0	70.0	52.1
6	10.0	70.0	52.1
7	300.4	70.0	52.1
8	310.4	70.0	52.1
Hoist in low position			
1	0.0	33.0	-52.1
2	10.0	33.0	-52.1
3	300.4	33.0	-52.1
4	310.4	33.0	-52.1
5	0.0	33.0	52.1
6	10.0	33.0	52.1
7	300.4	33.0	52.1
8	310.4	33.0	52.1

TROLLEY COMPUTER MODEL (TROLY-LB)

Joint coordinates

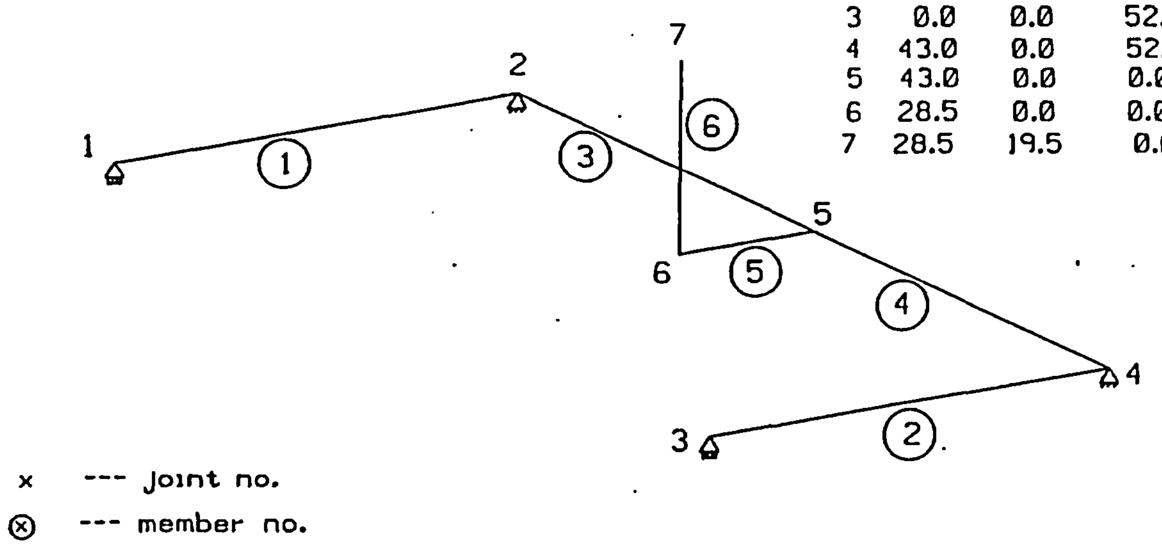
Joint no.	X	Y	Z
11	0.0	0.0	-52.1
12	-43.0	0.0	-52.1
13	0.0	0.0	52.1
14	-43.0	0.0	52.1
15	-43.0	0.0	0.0
16	-28.5	0.0	0.0
17	-28.5	19.5	0.0

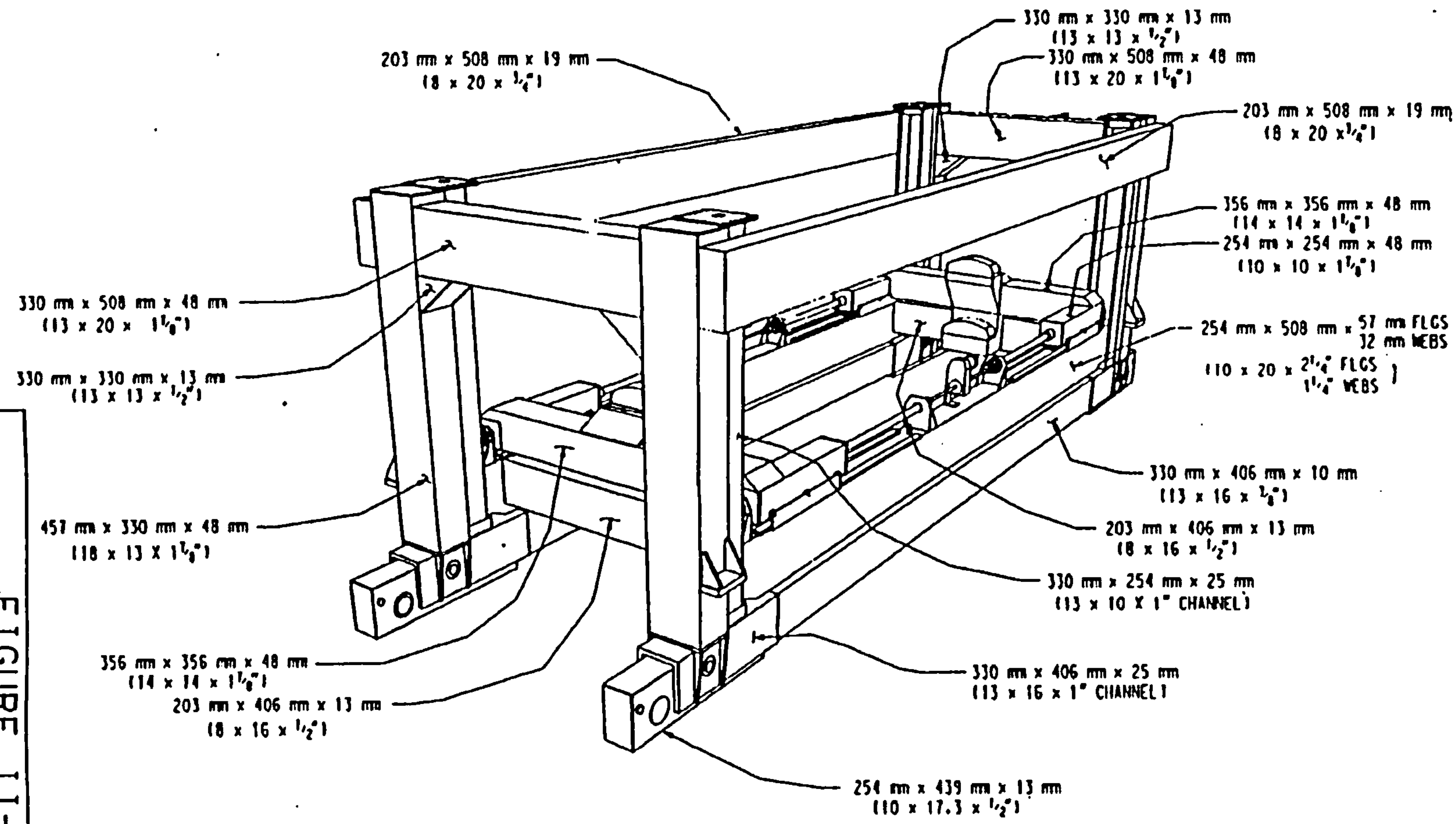


TROLLEY COMPUTER MODEL (TROLY-LF)

Joint coordinates

Joint no.	X	Y	Z
1	0.0	0.0	-52.1
2	43.0	0.0	-52.1
3	0.0	0.0	52.1
4	43.0	0.0	52.1
5	43.0	0.0	0.0
6	28.5	0.0	0.0
7	28.5	19.5	0.0





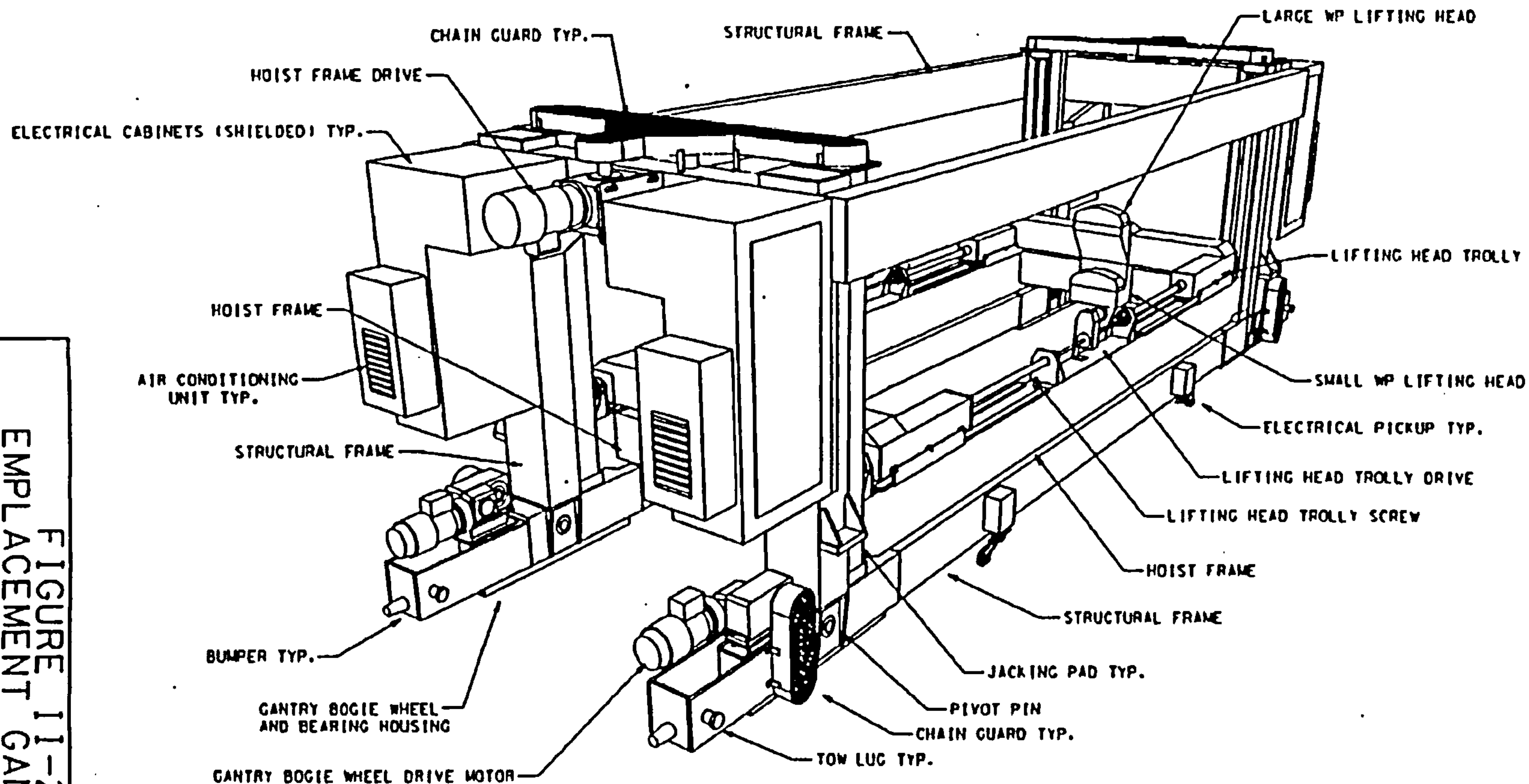
GANTRY ASSEMBLY

PERSPECTIVE VIEW

NO SCALE

STRUCTURAL SECTIONS ARE BOX SECTIONS. UNLESS OTHERWISE NOTED OF WIDTH x DEPTH x THICKNESS INDICATED. ALL DIMENSIONS ARE mm (Inches).

FIGURE II-1
EMPLACEMENT GANTRY FRAME

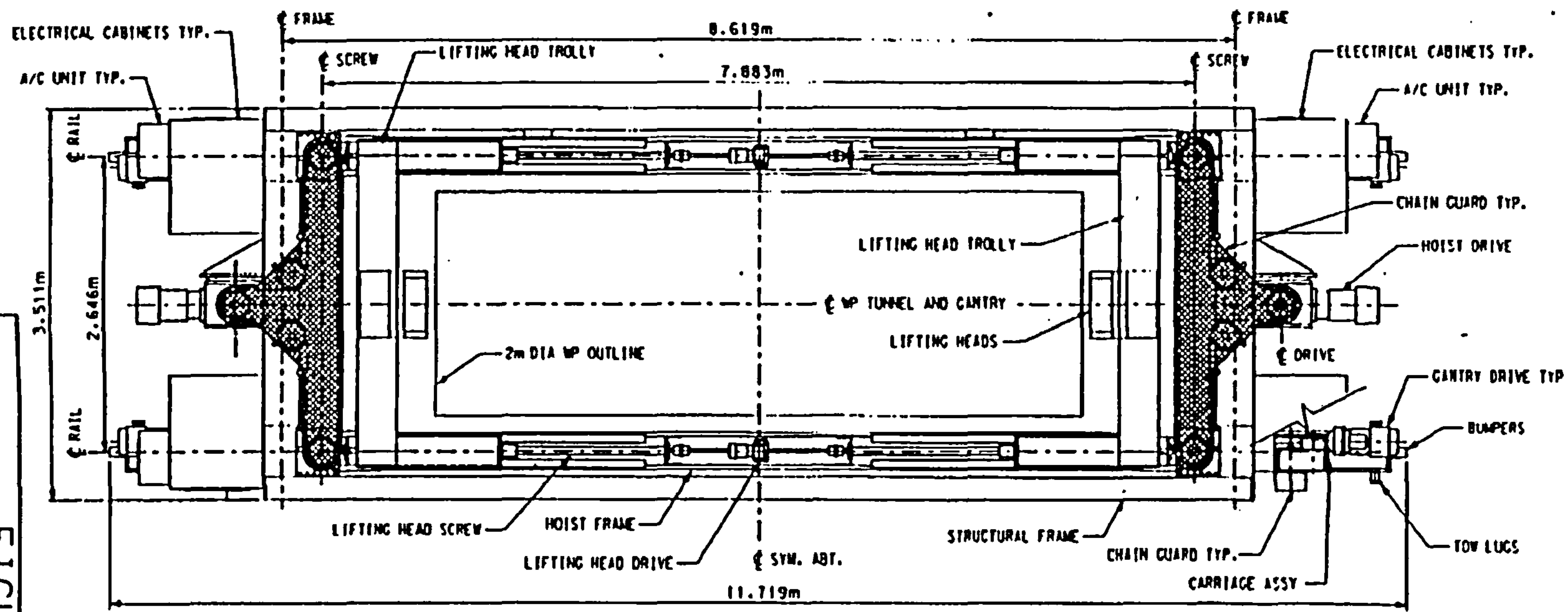


GANTRY ASSEMBLY

PERSPECTIVE VIEW

NO SCALE

FIGURE II-2
EMPLACEMENT GANTRY ARRANGEMENT



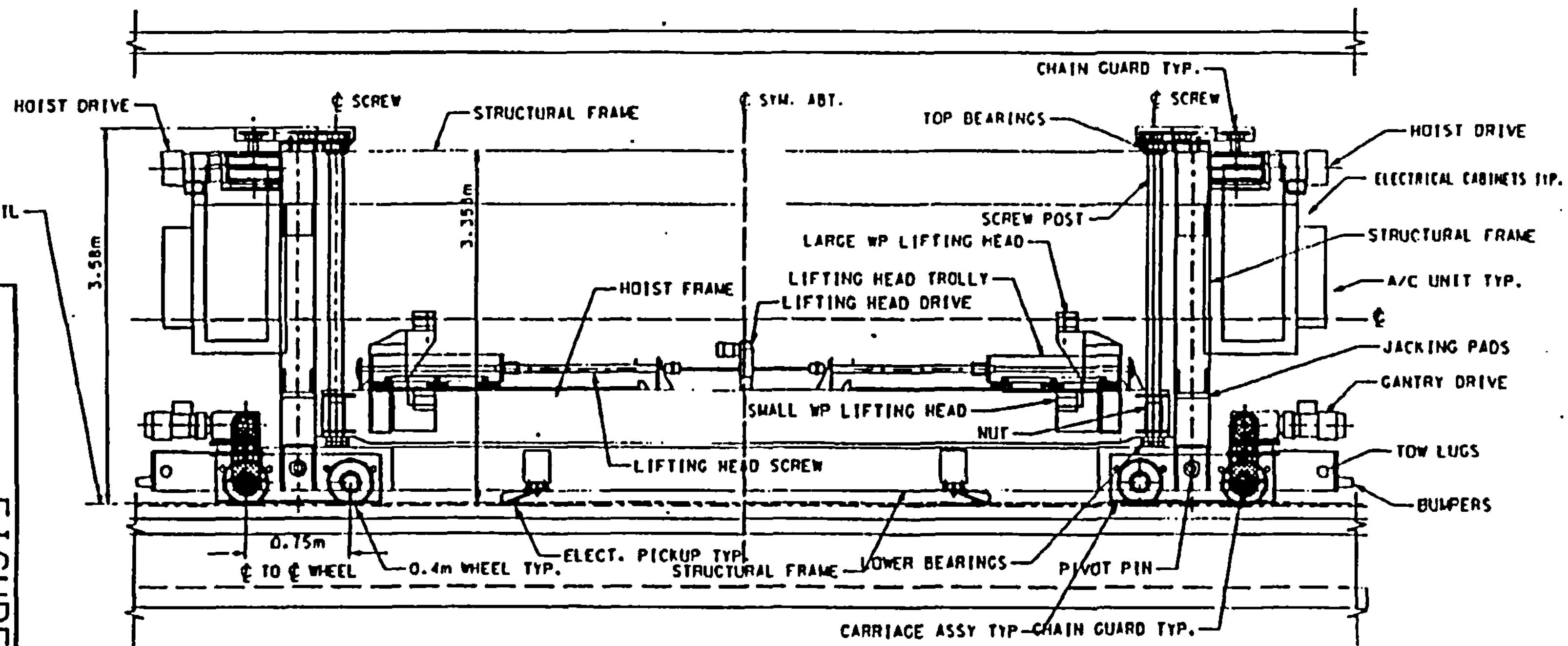
GANTRY ASSEMBLY-PLAN

NO SCALE

NO SCALE

FIGURE II-3
EMPLACEMENT GANTRY
PLAN

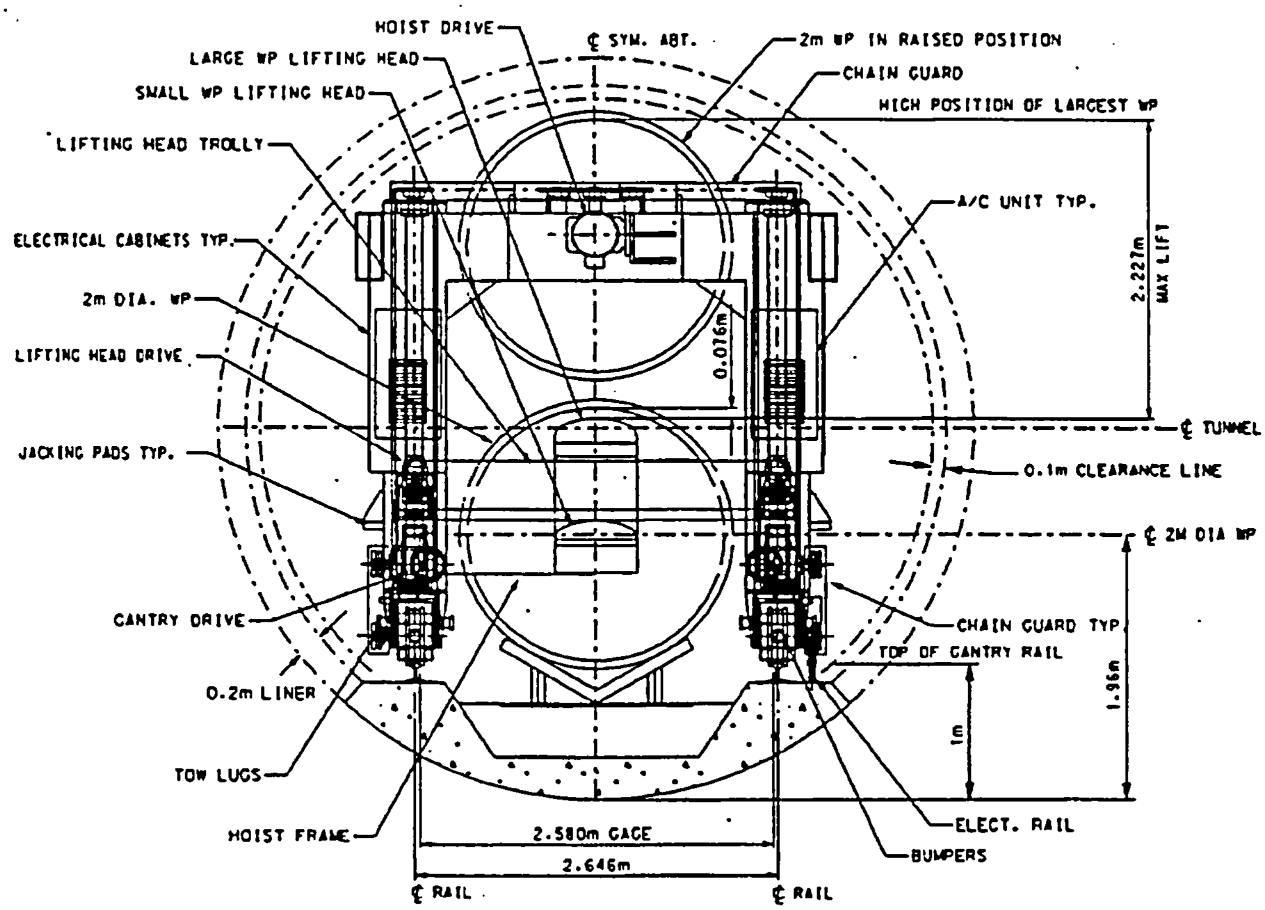
FIGURE II-4
EMPLACEMENT GANTRY
SIDE ELEVATION



GANTRY ASSEMBLY

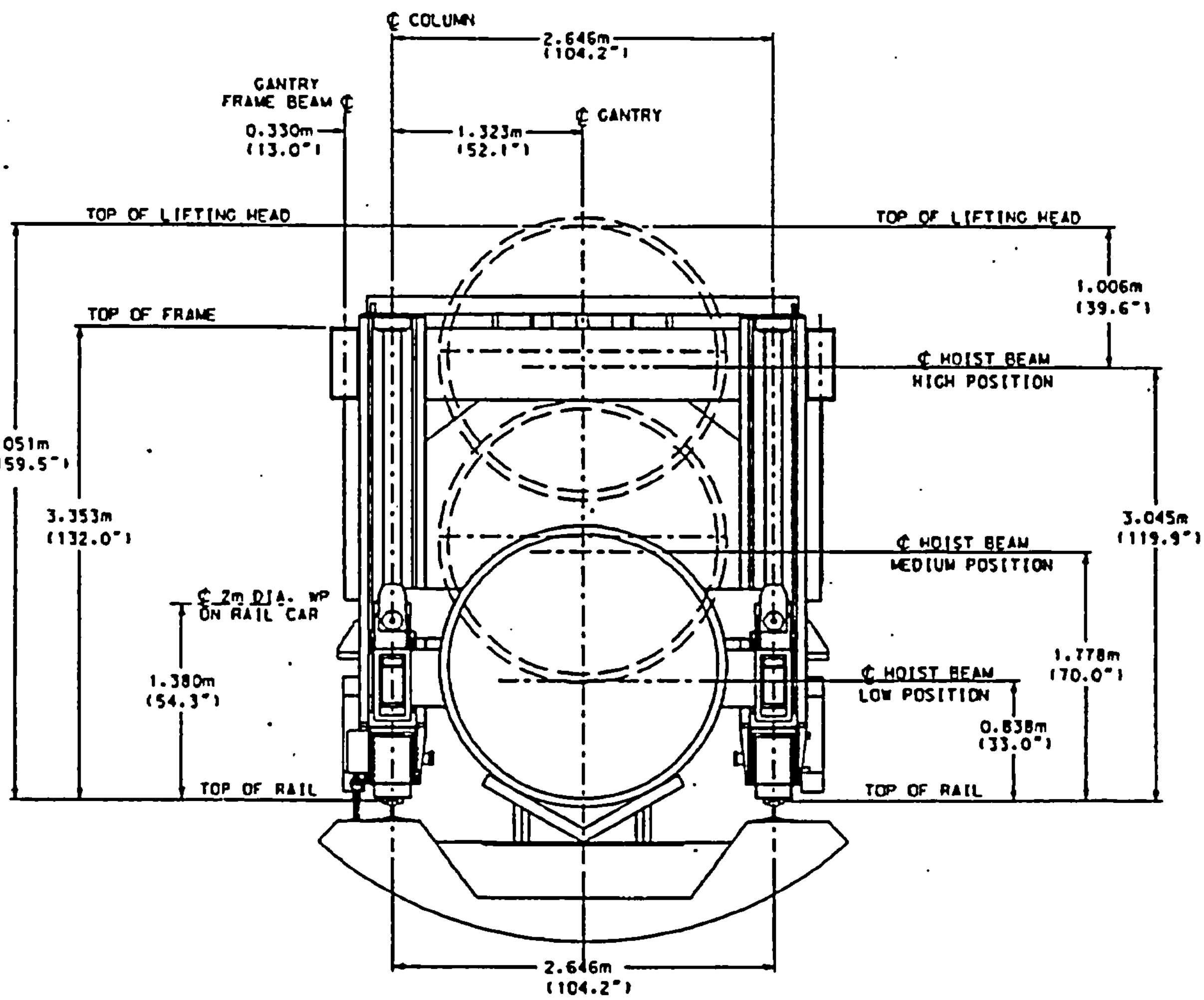
SIDE ELEVATION

NO SCALE



GANTRY ASSEMBLY
END ELEVATION
NO SCALE

FIGURE II-5
EMPLACEMENT GANTRY
END ELEVATION



GANTRY ASSEMBLY

CROSS SECTION

NO SCALE

FIGURE II-6
EMPLACEMENT GANTRY
CROSS SECTION

11.0 WASTE PACKAGE SIZE/LOAD COMPARATIVE ANALYSIS

Waste Package Size / Load Comparative Analysis

STAAD III input file " wp-compa.std " for waste package size/load comparative analysis, provides the input to determine the maximum moment in the hoist frame during vertical and lateral seismic load conditions for the waste package to be lifted.

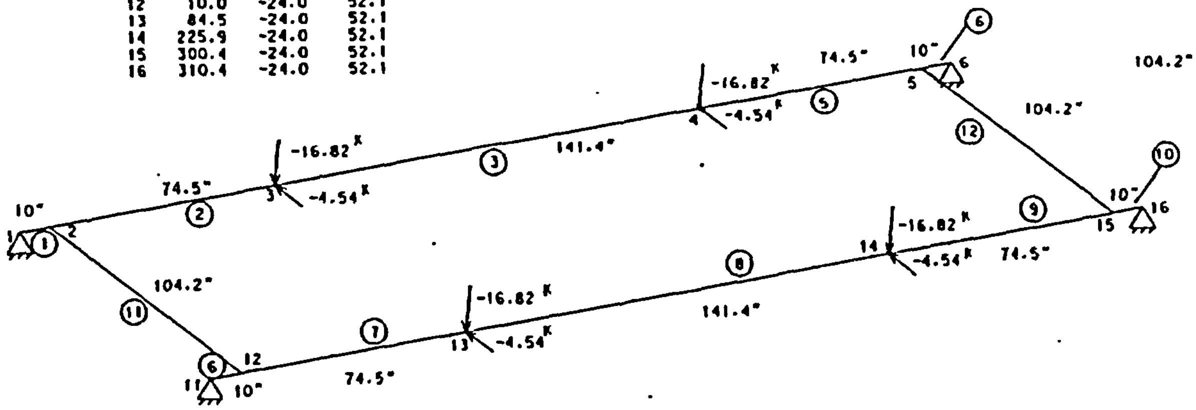
	Diameter	Length	WP Loaded mass		Mz	
WP type	(mm)	(mm)	(Kg)	kips	(kip-ft)	
4 DHLW	1785	3790	30,511	67.265	142.06	
12 PWR UCF	1298	5335	32,236	71.266	103.81	
44 BWR UCF	1604	5335	46,424	102.347	139.06	
21 PWR UCF	1650	5335	50,423	111.164	149.04	
—	1970	5350	69,000	152.114	194.54	* Control *
—	1850	5850	69,000	152.114	163.19	

Waste Package at dia. = 1970 mm, length = 5350 mm, and load mass = 69,000 kg control emplacement gantry design for structural and deflection analysis.

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	84.5	-24.0	-52.1
4	225.9	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	84.5	-24.0	52.1
14	225.9	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(x) Indicates member number
x Indicates joint number



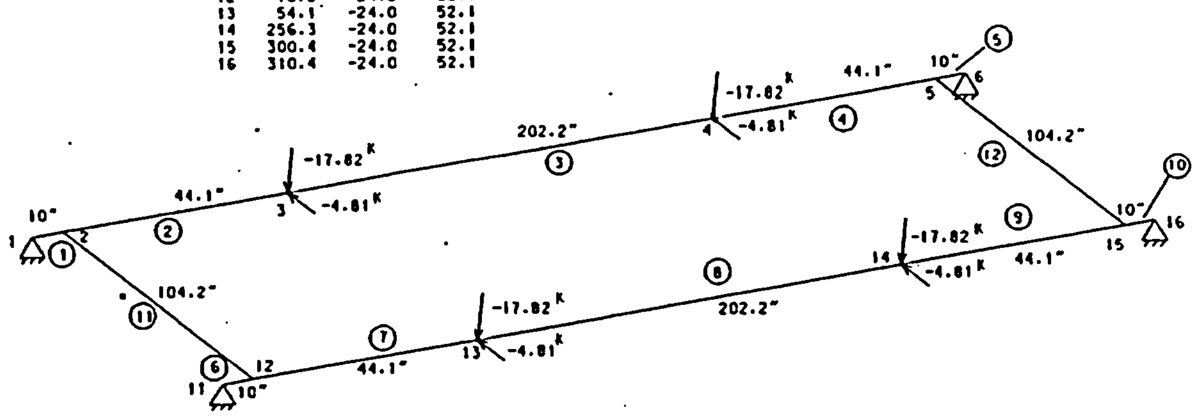
COMPARATIVE ANALYSIS --- WASTE PACKAGE " 4D HLW "

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	54.1	-24.0	-52.1
4	256.3	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	54.1	-24.0	52.1
14	256.3	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(X) Indicates member number

X Indicates joint number

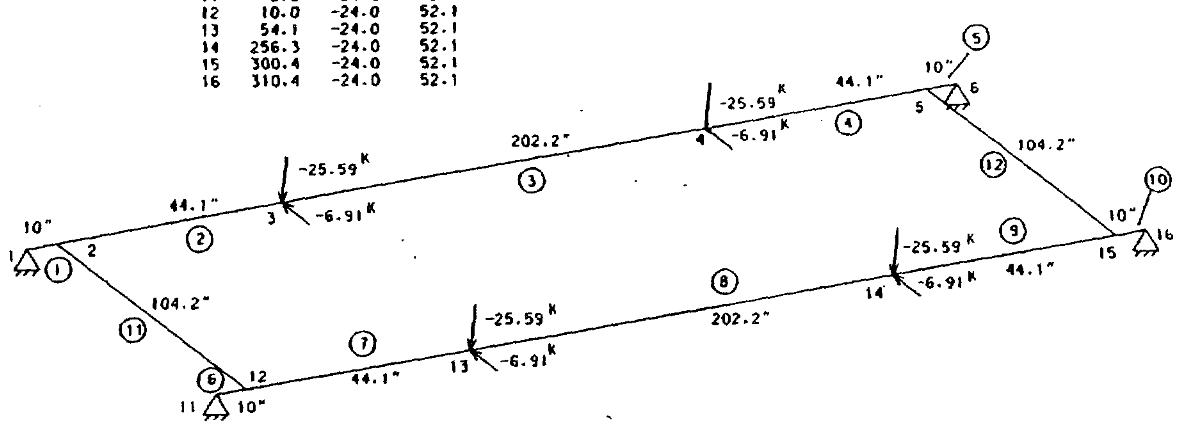


COMPARATIVE ANALYSIS --- WASTE PACKAGE " 12 PWR UCF "

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	54.1	-24.0	-52.1
4	256.3	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	54.1	-24.0	52.1
14	256.3	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(X). Indicates member number
X Indicates joint number

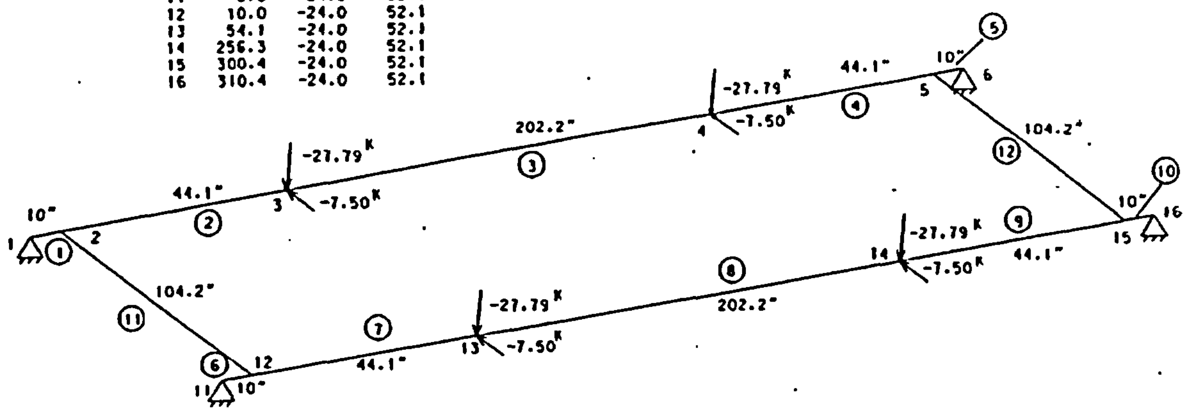


COMPARATIVE ANALYSIS --- WASTE PACKAGE " 44 BWR UCF "

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	54.1	-24.0	-52.1
4	256.3	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	54.1	-24.0	52.1
14	256.3	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(x) Indicates member number
x Indicates joint number

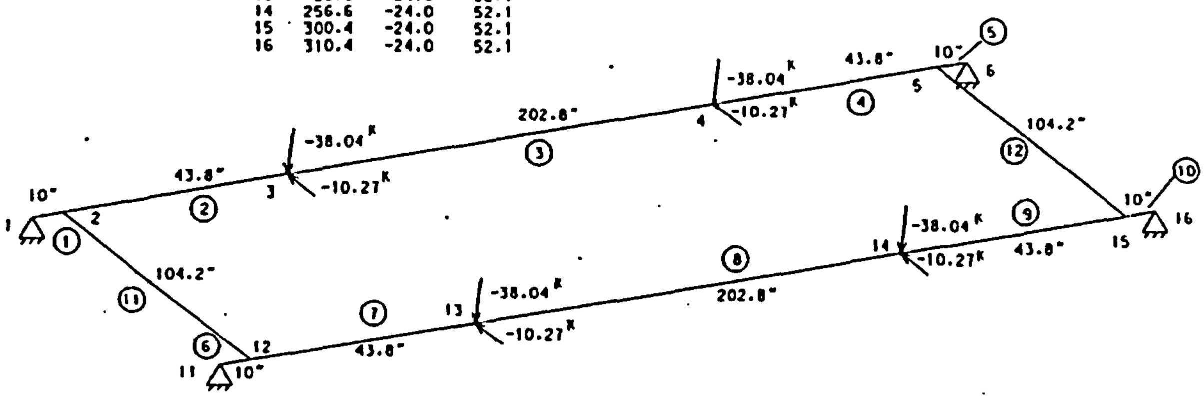


COMPARATIVE ANALYSIS --- WASTE PACKAGE " 21PWR UCF "

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	53.8	-24.0	-52.1
4	256.6	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	53.8	-24.0	52.1
14	256.6	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(X) Indicates member number
X Indicates joint number

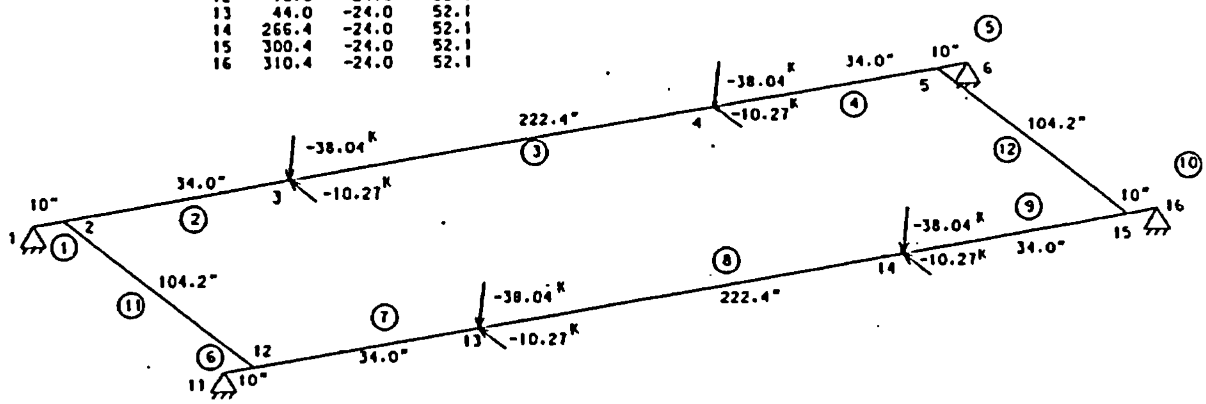


COMPARATIVE ANALYSIS --- WASTE PACKAGE " --- 69 MTON L = 5350mm "

JOINT COORDINATE

JOINT	X	Y	Z
1	0.0	-24.0	-52.1
2	10.0	-24.0	-52.1
3	44.0	-24.0	-52.1
4	266.4	-24.0	-52.1
5	300.4	-24.0	-52.1
6	310.4	-24.0	-52.1
11	0.0	-24.0	52.1
12	10.0	-24.0	52.1
13	44.0	-24.0	52.1
14	266.4	-24.0	52.1
15	300.4	-24.0	52.1
16	310.4	-24.0	52.1

(x) Indicates member number
x Indicates joint number



COMPARATIVE ANALYSIS --- WASTE PACKAGE " --- 69 MTON L = 5850mm "

```
File: WP-COMPA.std      Page: 1 of 2

staad space comparative wp loads
* wo3969/gantry-11/wp-compa     waste packages load comparisons
page length 72
unit kip inch
joint coordinates
1 0. -24. -52.1;2 10.0 -24. -52.1;3 300.4 -24. -52.1;4 310.4 -24.0 -52.1
5 0. -24.  52.1;6 10.0 -24.  52.1;7 300.4 -24.  52.1;8 310.4 -24.0  52.1
member incidences
1 1 2 3;4 5 6 6;11 2 6;12 3 7
member offset
11 12 start 0 3 0
11 12 end   0 3 0
constant
E steel all
density steel all
start user table
table 1
tube
T20x10
79.88 20.0 10.0 2.25 4262 1065 100 90.0 45.0
end
member property
1 to 6 upt 1 t20x10
11  12 ta st tube dt 16 wt 8 th 0.3125
support
1 4 5 8 pinned
loADing 1 vertical and lateral load (LL=30.511mton, L=3.790m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -16.82  74.5
2 5 con gy -16.82 215.9
2 5 con gz  -4.54  74.5
2 5 con gz  -4.54 215.9
loADing 2 vertical and lateral load (LL=32.236 mton, L=5.335m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -17.82  44.1
2 5 con gy -17.82 246.3
2 5 con gz  -4.81  44.1
2 5 con gz  -4.81 246.3
loADing 3 vertical and lateral load (LL=46.424 mton, L=5.335m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -25.59  44.1
2 5 con gy -25.59 246.3
2 5 con gz  -6.91  44.1
2 5 con gz  -6.91 246.3
loADing 4 vertical and lateral load (LL=50.423mton, L=5.335m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -27.79  44.1
2 5 con gy -27.79 246.3
2 5 con gz  -7.50  44.1
2 5 con gz  -7.50 246.3
LOADing 5 vertical and lateral load (LL=69. mton, L=5.350m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -38.03  43.8
2 5 con gy -38.03 246.6
2 5 con gz -10.27  43.8
2 5 con gz -10.27 246.6
LOADing 6 vertical and lateral load (LL=69. mton, L=5.85m)
SELFWEIGHT Y -1.0
selfweight z -0.27
member load
2 5 con gy -38.03  33.96
2 5 con gy -38.03 256.44
2 5 con gz -10.27  33.96
2 5 con gz -10.27 256.44
PERFORM ANALYSIS
unit inch
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File: WP-CCMPA.std Page: 2 of 2

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parameters
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code aisc
check code
unit feet
print support reaction
print joint displacement
load list 1
print maxforce env
load list 2
print maxforce env
load list 3
print maxforce env
load list 4
print maxforce env
load list 5
print maxforce env
load list 6
print maxforce env
steel take off
finish
```

User ID: mk
PAGE NO. 1

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*                                           *
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*          Revision 22.0W                   *
*          Proprietary Program of           *
*          Research Engineers, Inc.         *
*          Date=   MAY 19, 1997             *
*          Time=    16:14:18                *
*                                           *
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 1. STAAD SPACE COMPARATIVE WP LOADS
 2. * WO3969/GANTRY-11/WP-COMPA    WASTE PACKAGES LOAD COMPARISONS
 3. PAGE LENGTH 72
 4. UNIT KIP INCH
 5. JOINT COORDINATES
 6. 1 0. -24. -52.1;2 10.0 -24. -52.1;3 300.4 -24. -52.1;4 310.4 -24.0 -52.1
 7. 5 0. -24.  52.1;6 10.0 -24.  52.1;7 300.4 -24.  52.1;8 310.4 -24.0  52.1
 8. MEMBER INCIDENCES
 9. 1 1 2 3;4 5 6 6;11 2 6;12 3 7
10. MEMBER OFFSET
11. 11 12 START 0 3 0
12. 11 12 END   0 3 0
13. CONSTANT
14. E STEEL ALL
15. DENSITY STEEL ALL
16. START USER TABLE
17. TABLE 1
18. TUBE
19. T20X10
20. 79.88 20.0 10.0 2.25 4262 1065 100 90.0 45.0
21. END
22. MEMBER PROPERTY
23. 1 TO 6 UPT 1 T20X10
24. 11  12 TA ST TUBE DT 16 WT 8 TH 0.3125
25. SUPPORT
26. 1 4 5 8 PINNED
27. LOADING 1 VERTICAL AND LATERAL LOAD (LL=30.511MTON, L=3.790M)
28. SELFWEIGHT Y -1.0
29. SELFWEIGHT Z -0.27
30. MEMBER LOAD
31. 2 5 CON GY -16.82  74.5
32. 2 5 CON GY -16.82 215.9
33. 2 5 CON GZ  -4.54  74.5
34. 2 5 CON GZ  -4.54 215.9
35. LOADING 2 VERTICAL AND LATERAL LOAD (LL=32.236 MTON, L=5.335M)
36. SELFWEIGHT Y -1.0
37. SELFWEIGHT Z -0.27
38. MEMBER LOAD
39. 2 5 CON GY -17.82  44.1
40. 2 5 CON GY -17.82 246.3
41. 2 5 CON GZ  -4.81  44.1
42. 2 5 CON GZ  -4.81 246.3
43. LOADING 3 VERTICAL AND LATERAL LOAD (LL=46.424 MTON, L=5.335M)
44. SELFWEIGHT Y -1.0
45. SELFWEIGHT Z -0.27
46. MEMBER LOAD
47. 2 5 CON GY -25.59  44.1
48. 2 5 CON GY -25.59 246.3
49. 2 5 CON GZ  -6.91  44.1
50. 2 5 CON GZ  -6.91 246.3
51. LOADING 4 VERTICAL AND LATERAL LOAD (LL=50.423MTON, L=5.335M)
52. SELFWEIGHT Y -1.0
53. SELFWEIGHT Z -0.27
54. MEMBER LOAD
55. 2 5 CON GY -27.79  44.1
```

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                                                          User ID: mk
 COMPARATIVE WP LOADS                                     -- PAGE NO.     2

  * WO3969/GANTRY-11/WP-COMPA     WASTE PACKAGE
 56. 2 5 CON GY -27.79 246.3
 57. 2 5 CON GZ  -7.50  44.1
 58. 2 5 CON GZ  -7.50 246.3
 59. LOADING 5 VERTICAL AND LATERAL LOAD (LL=69. MTON, L=5.350M)
 60. SELFWEIGHT Y -1.0
 61. SELFWEIGHT Z -0.27
 62. MEMBER LOAD
 63. 2 5 CON GY -38.03  43.8
 64. 2 5 CON GY -38.03 246.6
 65. 2 5 CON GZ -10.27  43.8
 66. 2 5 CON GZ -10.27 246.6
 67. LOADING 6 VERTICAL AND LATERAL LOAD (LL=69. MTON, L=5.85M)
 68. SELFWEIGHT Y -1.0
 69. SELFWEIGHT Z -0.27
 70. MEMBER LOAD
 71. 2 5 CON GY -38.03  33.96
 72. 2 5 CON GY -38.03 256.44
 73. 2 5 CON GZ -10.27  33.96
 74. 2 5 CON GZ -10.27 256.44
 75. PERFORM ANALYSIS

          P R O B L E M   S T A T I S T I C S
          -----------------------------------

  NUMBER OF JOINTS/MEMBER+ELEMENTS/SUPPORTS =      8/     8/     4
  ORIGINAL/FINAL BAND-WIDTH =      4/     3
  TOTAL PRIMARY LOAD CASES  =      6, TOTAL DEGREES OF FREEDOM =     36
  SIZE OF STIFFNESS MATRIX =      756 DOUBLE PREC. WORDS
  REQRD/AVAIL. DISK SPACE =  12.02/  669.3 MB,  EXMEM =   58.8 MB

++ Processing Element Stiffness Matrix.                  16:14:20
++ Processing Global Stiffness Matrix.                   16:14:20
++ Processing Triangular Factorization.                  16:14:20
++ Calculating Joint Displacements.                      16:14:20
++ Calculating Member  Forces.                           16:14:21

  76. UNIT INCH
  77. PARAMETERS
  78. FYLD 27.3
  79. CODE AISC
  80. CHECK CODE
```

User ID: mk
-- PAGE NO. 3

COMPARATIVE WP LOADS

WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

STAAD-III CODE CHECKING - (AISC)

ALL UNITS ARE - KIP INCH (UNLESS OTHERWISE NOTED)

MEMBER	TABLE	RESULT/ FX	CRITICAL COND/ MY	RATIO/ MZ	LOADING/ LOCATION
1	ST T20X10	PASS	AISC- H1-3	0.096	5
		4.98 C	113.22	-416.33	10.00
2	ST T20X10	PASS	AISC- H2-1	0.105	5
		0.34 T	-164.12	-402.17	0.00
3	ST T20X10	PASS	AISC- H1-3	0.096	5
		4.98 C	113.22	-416.32	0.00
4	ST T20X10	PASS	AISC- H2-1	0.095	5
		4.98 T	111.62	-416.26	10.00
5	ST T20X10	PASS	AISC- H1-3	0.109	5
		0.34 C	-165.28	-430.42	0.00
6	ST T20X10	PASS	AISC- H2-1	0.095	5
		4.98 T	111.62	-416.26	0.00
11	ST TUB E	PASS	AISC- H1-3	0.470	5
		0.13 C	277.34	0.40	0.00
12	ST TUB E	PASS	AISC- H1-3	0.470	5
		0.13 C	-277.34	0.40	0.00

81. UNIT FEET
82. PRINT SUPPORT REACTION

User ID: mk
COMPARATIVE WP LOADS -- PAGE NO. 4

* W03969/GANTRY-11/WP-COMPA WASTE PACKAGE

SUPPORT REACTIONS -UNIT KIP FEET STRUCTURE TYPE = SPACE

JOINT	LOAD	FORCE-X	FORCE-Y	FORCE-Z	MOM-X	MOM-Y	MOM Z
1	1	3.24	20.54	5.62	0.00	0.00	0.00
	2	2.60	21.54	5.89	0.00	0.00	0.00
	3	3.52	29.31	7.99	0.00	0.00	0.00
	4	3.78	31.51	8.58	0.00	0.00	0.00
	5	4.98	41.75	11.35	0.00	0.00	0.00
	6	4.29	41.75	11.35	0.00	0.00	0.00
4	1	-3.24	20.54	5.62	0.00	0.00	0.00
	2	-2.60	21.54	5.89	0.00	0.00	0.00
	3	-3.52	29.31	7.99	0.00	0.00	0.00
	4	-3.78	31.51	8.58	0.00	0.00	0.00
	5	-4.98	41.75	11.35	0.00	0.00	0.00
	6	-4.29	41.75	11.35	0.00	0.00	0.00
5	1	-3.24	20.53	5.46	0.00	0.00	0.00
	2	-2.60	21.53	5.73	0.00	0.00	0.00
	3	-3.52	29.30	7.83	0.00	0.00	0.00
	4	-3.78	31.50	8.42	0.00	0.00	0.00
	5	-4.98	41.74	11.19	0.00	0.00	0.00
	6	-4.29	41.74	11.19	0.00	0.00	0.00
8	1	3.24	20.53	5.46	0.00	0.00	0.00
	2	2.60	21.53	5.73	0.00	0.00	0.00
	3	3.52	29.30	7.83	0.00	0.00	0.00
	4	3.78	31.50	8.42	0.00	0.00	0.00
	5	4.98	41.74	11.19	0.00	0.00	0.00
	6	4.29	41.74	11.19	0.00	0.00	0.00

************** END OF LATEST ANALYSIS RESULT **************

83. PRINT JOINT DISPLACEMENT

COMPARATIVE WP LOADS

User ID: mk
-- PAGE NO. 5

* WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

JOINT DISPLACEMENT (INCH RADIANS) STRUCTURE TYPE = SPACE

JOINT	LOAD	X-TRANS	Y-TRANS	Z-TRANS	X-ROTAN	Y-ROTAN	Z-ROTAN
1	1	0.00000	0.00000	0.00000	0.00002	0.00081	-0.00152
	2	0.00000	0.00000	0.00000	0.00001	0.00065	-0.00122
	3	0.00000	0.00000	0.00000	0.00002	0.00088	-0.00165
	4	0.00000	0.00000	0.00000	0.00002	0.00095	-0.00178
	5	0.00000	0.00000	0.00000	0.00002	0.00124	-0.00234
	6	0.00000	0.00000	0.00000	0.00002	0.00108	-0.00202
2	1	-0.00001	-0.01531	-0.00813	0.00002	0.00080	-0.00151
	2	-0.00001	-0.01235	-0.00657	0.00001	0.00064	-0.00121
	3	-0.00002	-0.01672	-0.00889	0.00002	0.00087	-0.00164
	4	-0.00002	-0.01796	-0.00954	0.00002	0.00093	-0.00176
	5	-0.00002	-0.02364	-0.01255	0.00002	0.00123	-0.00232
	6	-0.00002	-0.02044	-0.01087	0.00002	0.00106	-0.00200
3	1	0.00001	-0.01531	-0.00813	0.00002	-0.00080	0.00151
	2	0.00001	-0.01235	-0.00657	0.00001	-0.00064	0.00121
	3	0.00002	-0.01672	-0.00889	0.00002	-0.00087	0.00164
	4	0.00002	-0.01796	-0.00954	0.00002	-0.00093	0.00176
	5	0.00002	-0.02364	-0.01255	0.00002	-0.00123	0.00232
	6	0.00002	-0.02044	-0.01087	0.00002	-0.00106	0.00200
4	1	0.00000	0.00000	0.00000	0.00002	-0.00081	0.00152
	2	0.00000	0.00000	0.00000	0.00001	-0.00065	0.00122
	3	0.00000	0.00000	0.00000	0.00002	-0.00088	0.00165
	4	0.00000	0.00000	0.00000	0.00002	-0.00095	0.00178
	5	0.00000	0.00000	0.00000	0.00002	-0.00124	0.00234
	6	0.00000	0.00000	0.00000	0.00002	-0.00108	0.00202
5	1	0.00000	0.00000	0.00000	-0.00001	0.00080	-0.00154
	2	0.00000	0.00000	0.00000	-0.00001	0.00065	-0.00124
	3	0.00000	0.00000	0.00000	-0.00001	0.00088	-0.00168
	4	0.00000	0.00000	0.00000	-0.00001	0.00094	-0.00180
	5	0.00000	0.00000	0.00000	-0.00001	0.00124	-0.00237
	6	0.00000	0.00000	0.00000	-0.00001	0.00107	-0.00205
6	1	0.00001	-0.01553	-0.00807	-0.00001	0.00079	-0.00153
	2	0.00001	-0.01252	-0.00651	-0.00001	0.00064	-0.00123
	3	0.00002	-0.01696	-0.00883	-0.00001	0.00086	-0.00167
	4	0.00002	-0.01822	-0.00948	-0.00001	0.00093	-0.00179
	5	0.00002	-0.02397	-0.01250	-0.00001	0.00122	-0.00235
	6	0.00002	-0.02073	-0.01081	-0.00001	0.00105	-0.00203
7	1	-0.00001	-0.01553	-0.00807	-0.00001	-0.00079	0.00153
	2	-0.00001	-0.01252	-0.00651	-0.00001	-0.00064	0.00123
	3	-0.00002	-0.01696	-0.00883	-0.00001	-0.00086	0.00167
	4	-0.00002	-0.01822	-0.00948	-0.00001	-0.00093	0.00179
	5	-0.00002	-0.02397	-0.01250	-0.00001	-0.00122	0.00235
	6	-0.00002	-0.02073	-0.01081	-0.00001	-0.00105	0.00203
8	1	0.00000	0.00000	0.00000	-0.00001	-0.00080	0.00154
	2	0.00000	0.00000	0.00000	-0.00001	-0.00065	0.00124
	3	0.00000	0.00000	0.00000	-0.00001	-0.00088	0.00168
	4	0.00000	0.00000	0.00000	-0.00001	-0.00094	0.00180
	5	0.00000	0.00000	0.00000	-0.00001	-0.00124	0.00237
	6	0.00000	0.00000	0.00000	-0.00001	-0.00107	0.00205

************** END OF LATEST ANALYSIS RESULT **************

84. LOAD LIST 1
85. PRINT MAXFORCE ENV

User ID: mk
-- PAGE NO. 6

COMPARATIVE WP LOADS

• W03969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	20.54	0.00	1	0.00	0.00	1			
		5.62	0.00	1	4.66	0.83	1	3.24 C	0.00	1
	MIN	20.31	0.83	1	-17.02	0.83	1			
		5.56	0.83	1	0.00	0.00	1	3.24 C	0.83	1
2	MAX	20.10	0.00	1	-16.25	0.00	1			
		5.43	0.00	1	23.17	12.10	1	0.22 T	0.00	1
	MIN	-20.10	24.20	1	-140.53	12.10	1			
		-5.43	24.20	1	-10.38	0.00	1	0.22 T	24.20	1
3	MAX	-20.31	0.00	1	0.00	0.83	1			
		-5.56	0.00	1	4.66	0.00	1	3.24 C	0.00	1
	MIN	-20.54	0.83	1	-17.02	0.00	1			
		-5.62	0.83	1	0.00	0.83	1	3.24 C	0.83	1
4	MAX	20.53	0.00	1	0.00	0.00	1			
		5.46	0.00	1	4.53	0.83	1	3.24 T	0.00	1
	MIN	20.30	0.83	1	-17.01	0.83	1			
		5.40	0.83	1	0.00	0.00	1	3.24 T	0.83	1
5	MAX	20.10	0.00	1	-17.78	0.00	1			
		5.43	0.00	1	23.07	12.10	1	0.22 C	0.00	1
	MIN	-20.10	24.20	1	-142.06	12.10	1			
		-5.43	24.20	1	-10.48	24.20	1	0.22 C	24.20	1
6	MAX	-20.30	0.00	1	0.00	0.83	1			
		-5.40	0.00	1	4.53	0.00	1	3.24 T	0.00	1
	MIN	-20.53	0.83	1	-17.01	0.00	1			
		-5.46	0.83	1	0.00	0.83	1	3.24 T	0.83	1
11	MAX	0.21	0.00	1	0.03	0.00	1			
		-3.46	0.00	1	15.04	0.00	1	0.13 C	0.00	1
	MIN	-0.20	8.68	1	-0.42	4.34	1			
		-3.46	8.68	1	-15.00	8.68	1	0.02 C	8.68	1
12	MAX	0.21	0.00	1	0.03	0.00	1			
		3.46	0.00	1	15.00	8.68	1	0.13 C	0.00	1
	MIN	-0.20	8.68	1	-0.42	4.34	1			
		3.46	8.68	1	-15.04	0.00	1	0.02 C	8.68	1

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

86. LOAD LIST 2
87. PRINT MAXFORCE ENV

User ID: mk
-- PAGE NO. 7

-- COMPARATIVE WP LOADS

* WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	21.54	0.00	2	0.00	0.00	2			
		5.89	0.00	2	4.88	0.83	2	2.60 C	0.00	2
	MIN	21.31	0.83	2	-17.85	0.83	2			
		5.83	0.83	2	0.00	0.00	2	2.60 C	0.83	2
2	MAX	21.10	0.00	2	-17.24	0.00	2			
		5.70	0.00	2	15.84	12.10	2	0.18 T	0.00	2
	MIN	-21.10	24.20	2	-102.58	12.10	2			
		-5.70	24.20	2	-7.19	0.00	2	0.18 T	24.20	2
3	MAX	-21.31	0.00	2	0.00	0.83	2			
		-5.83	0.00	2	4.88	0.00	2	2.60 C	0.00	2
	MIN	-21.54	0.83	2	-17.85	0.00	2			
		-5.89	0.83	2	0.00	0.83	2	2.60 C	0.83	2
4	MAX	21.53	0.00	2	0.00	0.00	2			
		5.73	0.00	2	4.75	0.83	2	2.60 T	0.00	2
	MIN	21.30	0.83	2	-17.85	0.83	2			
		5.67	0.83	2	0.00	0.00	2	2.60 T	0.83	2
5	MAX	21.10	0.00	2	-18.46	0.00	2			
		5.70	0.00	2	15.75	12.10	2	0.18 C	0.00	2
	MIN	-21.10	24.20	2	-103.81	12.10	2			
		-5.70	24.20	2	-7.29	24.20	2	0.18 C	24.20	2
6	MAX	-21.30	0.00	2	0.00	0.83	2			
		-5.67	0.00	2	4.75	0.00	2	2.60 T	0.00	2
	MIN	-21.53	0.83	2	-17.85	0.00	2			
		-5.73	0.83	2	0.00	0.83	2	2.60 T	0.83	2
11	MAX	0.21	0.00	2	0.03	0.00	2			
		-2.78	0.00	2	12.08	0.00	2	0.13 C	0.00	2
	MIN	-0.20	8.68	2	-0.42	4.34	2			
		-2.78	8.68	2	-12.04	8.68	2	0.02 C	8.68	2
12	MAX	0.21	0.00	2	0.03	0.00	2			
		2.78	0.00	2	12.04	8.68	2	0.13 C	0.00	2
	MIN	-0.20	8.68	2	-0.42	4.34	2			
		2.78	8.68	2	-12.08	0.00	2	0.02 C	8.68	2

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

88. LOAD LIST 3
89. PRINT MAXFORCE ENV

User ID: mk
-- PAGE NO. 8

COMPARATIVE WP LOADS

W03969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	29.31	0.00	3	0.00	0.00	3			
		7.99	0.00	3	6.63	0.83	3	3.52 C	0.00	3
	MIN	29.08	0.83	3	-24.33	0.83	3			
		7.93	0.83	3	0.00	0.00	3	3.52 C	0.83	3
2	MAX	28.87	0.00	3	-23.49	24.20	3			
		7.80	0.00	3	21.03	12.10	3	0.24 T	0.00	3
	MIN	-28.87	24.20	3	-137.39	12.10	3			
		-7.80	24.20	3	-9.72	0.00	3	0.24 T	24.20	3
3	MAX	-29.08	0.00	3	0.00	0.83	3			
		-7.93	0.00	3	6.63	0.00	3	3.52 C	0.00	3
	MIN	-29.31	0.83	3	-24.33	0.00	3			
		-7.99	0.83	3	0.00	0.83	3	3.52 C	0.83	3
4	MAX	29.30	0.00	3	0.00	0.00	3			
		7.83	0.00	3	6.50	0.83	3	3.52 T	0.00	3
	MIN	29.07	0.83	3	-24.32	0.83	3			
		7.77	0.83	3	0.00	0.00	3	3.52 T	0.83	3
5	MAX	28.87	0.00	3	-25.16	24.20	3			
		7.80	0.00	3	20.94	12.10	3	0.24 C	0.00	3
	MIN	-28.87	24.20	3	-139.06	12.10	3			
		-7.80	24.20	3	-9.82	0.00	3	0.24 C	24.20	3
6	MAX	-29.07	0.00	3	0.00	0.83	3			
		-7.77	0.00	3	6.50	0.00	3	3.52 T	0.00	3
	MIN	-29.30	0.83	3	-24.32	0.00	3			
		-7.83	0.83	3	0.00	0.83	3	3.52 T	0.83	3
11	MAX	0.21	0.00	3	0.03	0.00	3			
		-3.76	0.00	3	16.36	0.00	3	0.13 C	0.00	3
	MIN	-0.20	8.68	3	-0.42	4.34	3			
		-3.76	8.68	3	-16.32	8.68	3	0.02 C	8.68	3
12	MAX	0.21	0.00	3	0.03	0.00	3			
		3.76	0.00	3	16.32	8.68	3	0.13 C	0.00	3
	MIN	-0.20	8.68	3	-0.42	4.34	3			
		3.76	8.68	3	-16.36	0.00	3	0.02 C	8.68	3

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

90. LOAD LIST 4
91. PRINT MAXFORCE ENV

User ID: mk

COMPARATIVE WP LOADS -- PAGE NO. 9

* WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	31.51	0.00	4	0.00	0.00	4			
		8.58	0.00	4	7.13	0.83	4	3.78 C	0.00	4
	MIN	31.28	0.83	4	-26.16	0.83	4			
		8.52	0.83	4	0.00	0.00	4	3.78 C	0.83	4
2	MAX	31.07	0.00	4	-25.26	0.00	4			
		8.39	0.00	4	22.49	12.10	4	0.26 T	0.00	4
	MIN	-31.07	24.20	4	-147.25	12.10	4			
		-8.39	24.20	4	-10.43	0.00	4	0.26 T	24.20	4
3	MAX	-31.28	0.00	4	0.00	0.83	4			
		-8.52	0.00	4	7.13	0.00	4	3.78 C	0.00	4
	MIN	-31.51	0.83	4	-26.16	0.00	4			
		-8.58	0.83	4	0.00	0.83	4	3.78 C	0.83	4
4	MAX	31.50	0.00	4	0.00	0.00	4			
		8.42	0.00	4	6.99	0.83	4	3.78 T	0.00	4
	MIN	31.27	0.83	4	-26.16	0.83	4			
		8.36	0.83	4	0.00	0.00	4	3.78 T	0.83	4
5	MAX	31.07	0.00	4	-27.05	24.20	4			
		8.39	0.00	4	22.39	12.10	4	0.26 C	0.00	4
	MIN	-31.07	24.20	4	-149.04	12.10	4			
		-8.39	24.20	4	-10.53	0.00	4	0.26 C	24.20	4
6	MAX	-31.27	0.00	4	0.00	0.83	4			
		-8.36	0.00	4	6.99	0.00	4	3.78 T	0.00	4
	MIN	-31.50	0.83	4	-26.16	0.00	4			
		-8.42	0.83	4	0.00	0.83	4	3.78 T	0.83	4
11	MAX	0.21	0.00	4	0.03	0.00	4			
		-4.04	0.00	4	17.56	0.00	4	0.13 C	0.00	4
	MIN	-0.20	8.68	4	-0.42	4.34	4			
		-4.04	8.68	4	-17.52	8.68	4	0.02 C	8.68	4
12	MAX	0.21	0.00	4	0.03	0.00	4			
		4.04	0.00	4	17.52	8.68	4	0.13 C	0.00	4
	MIN	-0.20	8.68	4	-0.42	4.34	4			
		4.04	8.68	4	-17.56	0.00	4	0.02 C	8.68	4

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

92. LOAD LIST 5
93. PRINT MAXFORCE ENV

COMPARATIVE WP LOADS

User ID: mk
-- PAGE NO. 10

WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	41.75	0.00	5	0.00	0.00	5			
		11.35	0.00	5	9.43	0.83	5	4.98 C	0.00	5
	MIN	41.52	0.83	5	-34.69	0.83	5			
		11.29	0.83	5	0.00	0.00	5	4.98 C	0.83	5
2	MAX	41.31	0.00	5	-33.51	24.20	5			
		11.16	0.00	5	29.17	12.10	5	0.34 T	0.00	5
	MIN	-41.31	24.20	5	-192.18	12.10	5			
		-11.16	24.20	5	-13.68	0.00	5	0.34 T	24.20	5
3	MAX	-41.52	0.00	5	-0.00	0.83	5			
		-11.29	0.00	5	9.43	0.00	5	4.98 C	0.00	5
	MIN	-41.75	0.83	5	-34.69	0.00	5			
		-11.35	0.83	5	0.00	0.83	5	4.98 C	0.83	5
4	MAX	41.74	0.00	5	0.00	0.00	5			
		11.19	0.00	5	9.30	0.83	5	4.98 T	0.00	5
	MIN	41.51	0.83	5	-34.69	0.83	5			
		11.13	0.83	5	0.00	0.00	5	4.98 T	0.83	5
5	MAX	41.31	0.00	5	-35.87	0.00	5			
		11.16	0.00	5	29.07	12.10	5	0.34 C	0.00	5
	MIN	-41.31	24.20	5	-194.54	12.10	5			
		-11.16	24.20	5	-13.77	0.00	5	0.34 C	24.20	5
6	MAX	-41.51	0.00	5	0.00	0.83	5			
		-11.13	0.00	5	9.30	0.00	5	4.98 T	0.00	5
	MIN	-41.74	0.83	5	-34.69	0.00	5			
		-11.19	0.83	5	0.00	0.83	5	4.98 T	0.83	5
11	MAX	0.21	0.00	5	0.03	0.00	5			
		-5.32	0.00	5	23.11	0.00	5	0.13 C	0.00	5
	MIN	-0.20	8.68	5	-0.42	4.34	5			
		-5.32	8.68	5	-23.07	8.68	5	0.02 C	8.68	5
12	MAX	0.21	0.00	5	0.03	0.00	5			
		5.32	0.00	5	23.07	8.68	5	0.13 C	0.00	5
	MIN	-0.20	8.68	5	-0.42	4.34	5			
		5.32	8.68	5	-23.11	0.00	5	0.02 C	8.68	5

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

94. LOAD LIST 6
95. PRINT MAXFORCE ENV

User ID: mk
-- PAGE NO. 11

COMPARATIVE WP LOADS

* WO3969/GANTRY-11/WP-COMPA WASTE PACKAGE

MEMBER FORCE ENVELOPE

ALL UNITS ARE KIP FEET

MAX AND MIN FORCE VALUES AMONGST ALL SECTION LOCATIONS

MEMB		FY/ FZ	DIST DIST	LD LD	MZ/ MY	DIST DIST	LD LD	FX	DIST	LD
1	MAX	41.75	0.00	6	0.00	0.00	6			
		11.35	0.00	6	9.43	0.83	6	4.29 C	0.00	6
	MIN	41.52	0.83	6	-34.69	0.83	6			
		11.29	0.83	6	0.00	0.00	6	4.29 C	0.83	6
2	MAX	41.31	0.00	6	-33.68	0.00	6			
		11.16	0.00	6	23.93	12.10	6	0.30 T	0.00	6
	MIN	-41.31	24.20	6	-161.16	12.10	6			
		-11.16	24.20	6	-10.50	0.00	6	0.30 T	24.20	6
3	MAX	-41.52	0.00	6	0.00	0.83	6			
		-11.29	0.00	6	9.43	0.00	6	4.29 C	0.00	6
	MIN	-41.75	0.83	6	-34.69	0.00	6			
		-11.35	0.83	6	0.00	0.83	6	4.29 C	0.83	6
4	MAX	41.74	0.00	6	0.00	0.00	6			
		11.19	0.00	6	9.30	0.83	6	4.29 T	0.00	6
	MIN	41.51	0.83	6	-34.69	0.83	6			
		11.13	0.83	6	0.00	0.00	6	4.29 T	0.83	6
5	MAX	41.31	0.00	6	-35.71	0.00	6			
		11.16	0.00	6	23.83	12.10	6	0.30 C	0.00	6
	MIN	-41.31	24.20	6	-163.19	12.10	6			
		-11.16	24.20	6	-10.59	0.00	6	0.30 C	24.20	6
6	MAX	-41.51	0.00	6	0.00	0.83	6			
		-11.13	0.00	6	9.30	0.00	6	4.29 T	0.00	6
	MIN	-41.74	0.83	6	-34.69	0.00	6			
		-11.19	0.83	6	0.00	0.83	6	4.29 T	0.83	6
11	MAX	0.21	0.00	6	0.03	0.00	6			
		-4.59	0.00	6	19.93	0.00	6	0.13 C	0.00	6
	MIN	-0.20	8.68	6	-0.42	4.34	6			
		-4.59	8.68	6	-19.89	8.68	6	0.02 C	8.68	6
12	MAX	0.21	0.00	6	0.03	0.00	6			
		4.59	0.00	6	19.89	8.68	6	0.13 C	0.00	6
	MIN	-0.20	8.68	6	-0.42	4.34	6			
		4.59	8.68	6	-19.93	0.00	6	0.02 C	8.68	6

********** END OF FORCE ENVELOPE FROM INTERNAL STORAGE **********

96. STEEL TAKE OFF

```
                                                                User ID: mk
   COMPARATIVE WP LOADS                                        -- PAGE NO.    12

 * WO3969/GANTRY-11/WP-COMPA    WASTE PACKAGE


STEEL TAKE-OFF
--------------

PROFILE                 LENGTH(FEET)     WEIGHT(KIP )

ST  T20X10                  51.73           14.034
ST  TUB E                   17.37            0.816
                                      ----------------
                           TOTAL =          14.85

************ END OF DATA FROM INTERNAL STORAGE ************

 97. FINISH

          *************** END OF STAAD-III ***************

            **** DATE= MAY 19,1997   TIME= 16:14:24 ****

     *****************************************************
     *        For questions on  STAAD-III,    contact:     *
     *             Research Engineers, Inc at              *
     *  West Coast: Ph- (714) 974-2500  Fax- (714) 921-2543 *
     *  East Coast: Ph- (508) 688-3626  Fax- (508) 685-7230 *
     *****************************************************
```

12.0 STAAD-III MODEL LOAD DIAGRAMS

With the exception of the dead load structure and equipment self weights (DL), skewing forces (SK), horizontal racking forces (HRF), and the vertical racking forces (VRF), all forces applied to the gantry models are applied to the top of the trolley lifting head.

For purpose of this analysis, the longitudinal forces resulting from the Tractive Inertia Forces (IFD), Longitudinal Thrust Force (LTF), Longitudinal Out-of-Plumb Forces (OPFL), Collision Forces (CF), and the Longitudinal Seismic Force (EQF), are considered to originate in the waste package and are distributed through the lifting lug to the trolley, hoist frame, and gantry structural frame. No structural connection of the waste package to the gantry exists in the longitudinal direction. Frictional forces developed by the waste package resulting on the lifting lugs will not be considered effective in uniformly distributing these longitudinal forces to the forward and rear gantry trolleys. Thus, to ensure the structural integrity of the gantry structural frame, hoist frame and trolleys, 100% of the longitudinal forces are applied to the forward lifting lug and trolley in design of the gantry structural frame, hoist frame, and trolleys. Further, to ensure structural integrity of the trolleys, 50% of the longitudinal force is applied to the lifting lug at the rear trolley for design of the trolley only.

STAAD-III model load diagrams indicate model configurations with applicable model members and joints identified. Member centerline offsets are not shown in load diagrams for clarity, but are identified in the STAAD-III input and output files. Applied loads eccentric to joints are shown in the load diagrams. Resulting concentrated moments due to these eccentricities (load x eccentricity) are not shown in load diagrams for clarity, but are identified in the STAAD-III input and output files.

Trolley Structural Analysis

See STAAD III run files

"troly-lf" for front trolley

"troly-lb" for back trolley

In direction of travel: Front trolley will be designed to resist 100% longitudinal loads, 50% vertical, and 50% lateral loads (Section 4.3.119).

Back trolley will be designed to resist 50% longitudinal loads, 50% vertical, and 50% lateral loads (Section 4.3.119).

Gantry Frame Structural Analysis

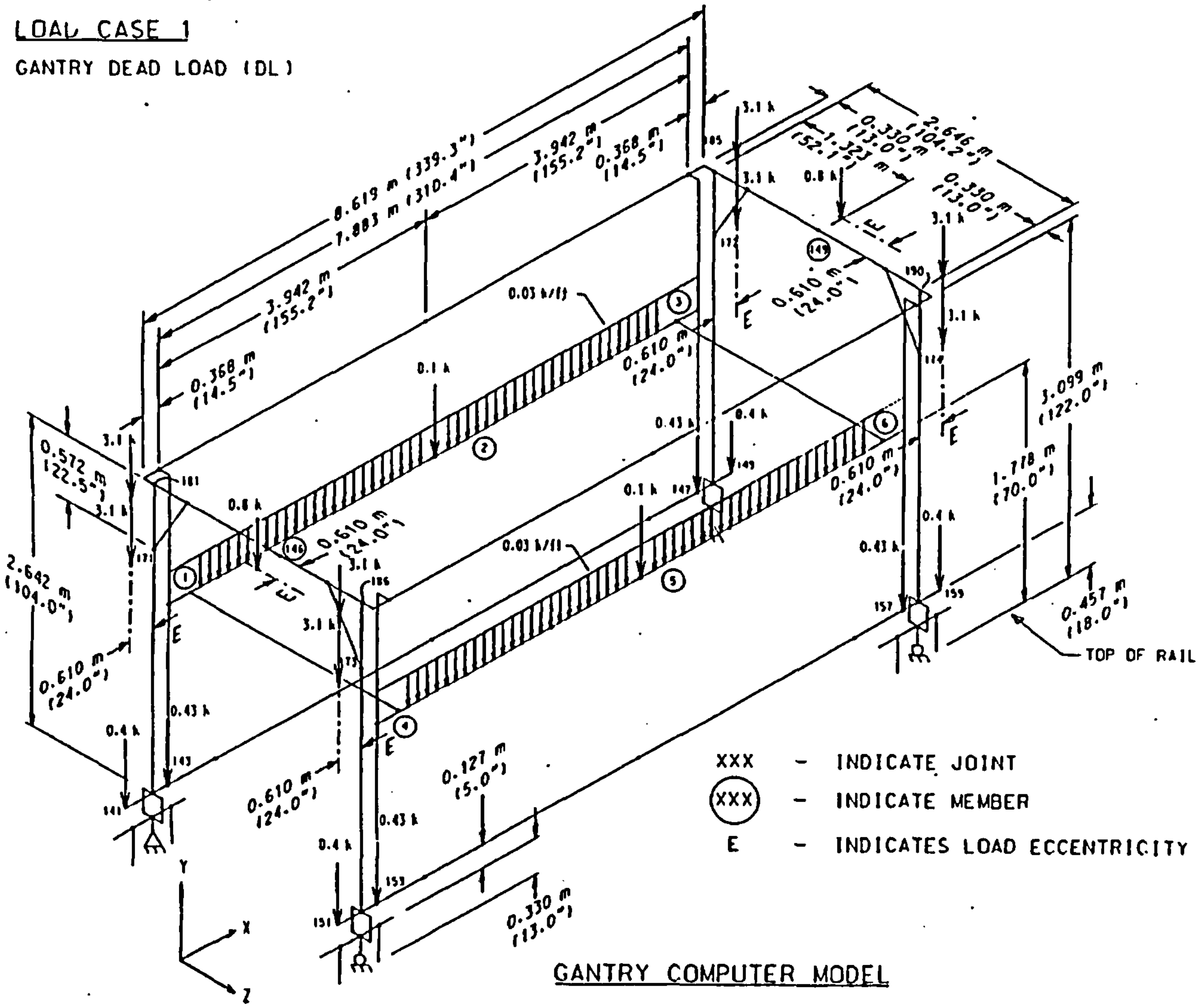
Gantry structural frame will be analyzed with the hoist frame at three heights so as to determine maximum stresses and deflection in the gantry frame structure (Attachment II, Section 10.0, Figure II-6).

Distance from top of rail to centerline of gantry hoist frame:

Hoist Frame Position	Centerline of Gantry Frame Above Top of Rail	STAAD III File Name
High position	119.9 inch	"gantry-h"
Mid-position	70.0 inch	"gantry-m"
Low position	33.0 inch	"gantry-l"

LOAD CASE 1

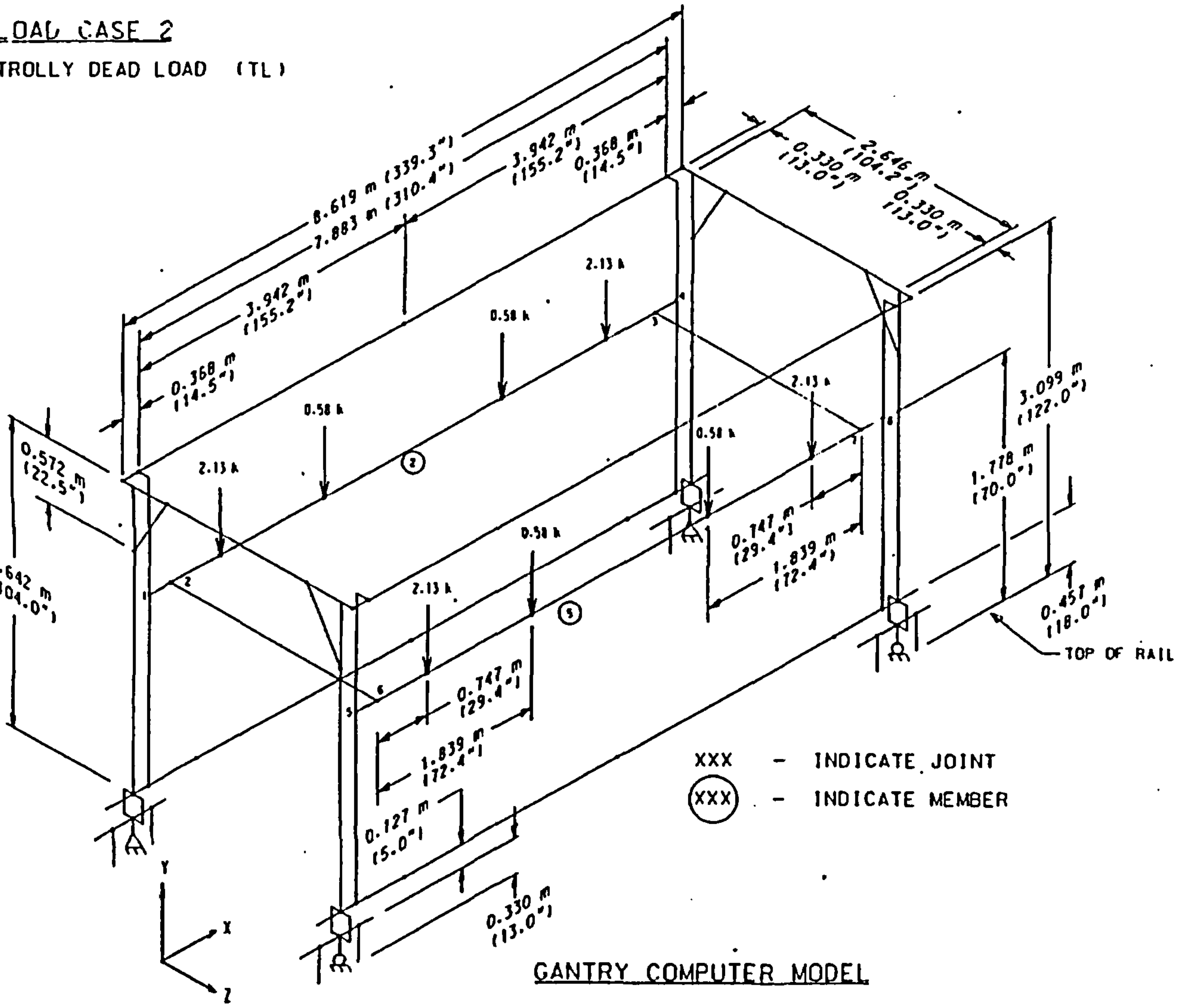
GANTRY DEAD LOAD (DL)



GANTRY COMPUTER MODEL

LOAD CASE 2

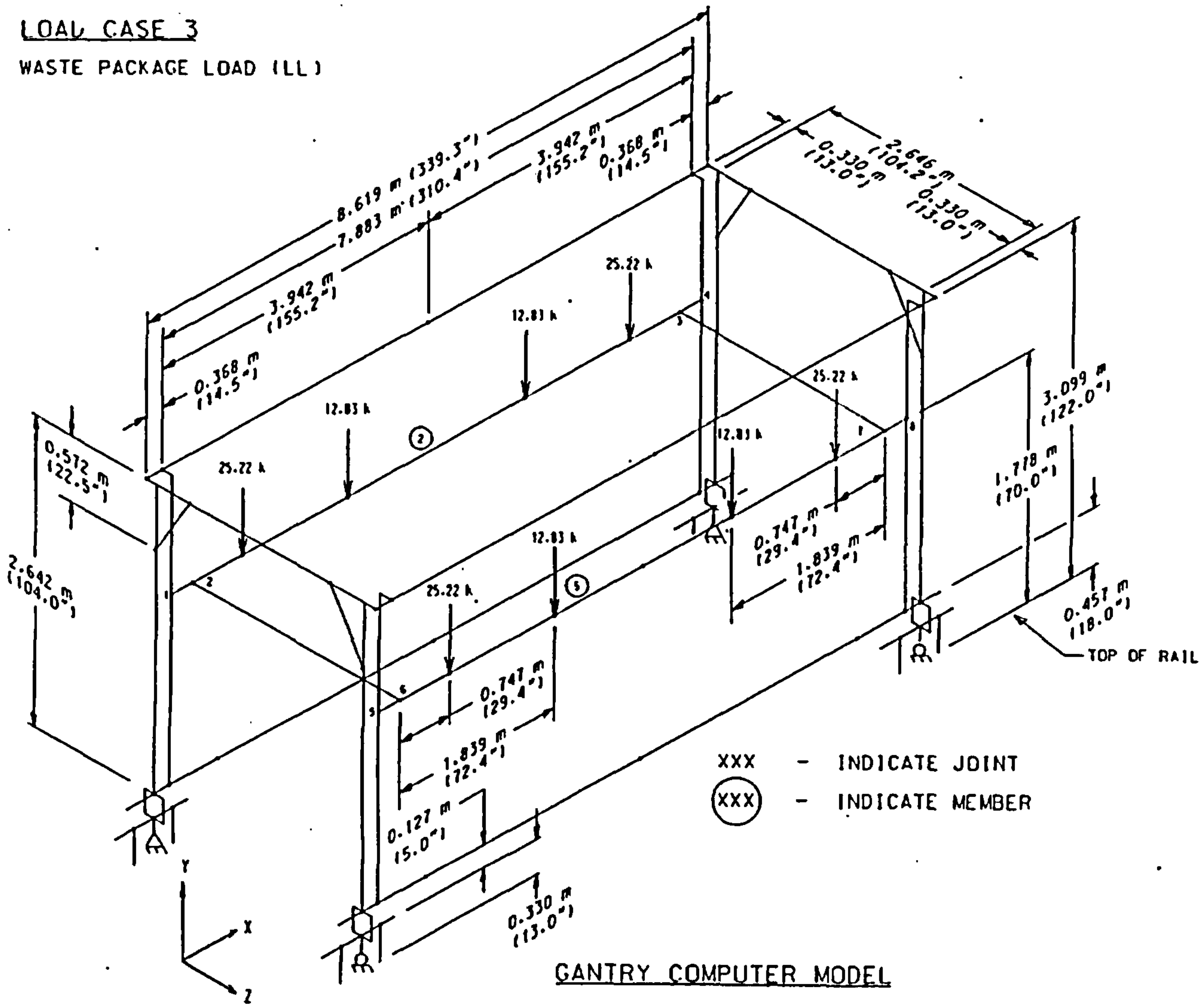
TROLLY DEAD LOAD (TL)



GANTRY COMPUTER MODEL

LOAD CASE 3

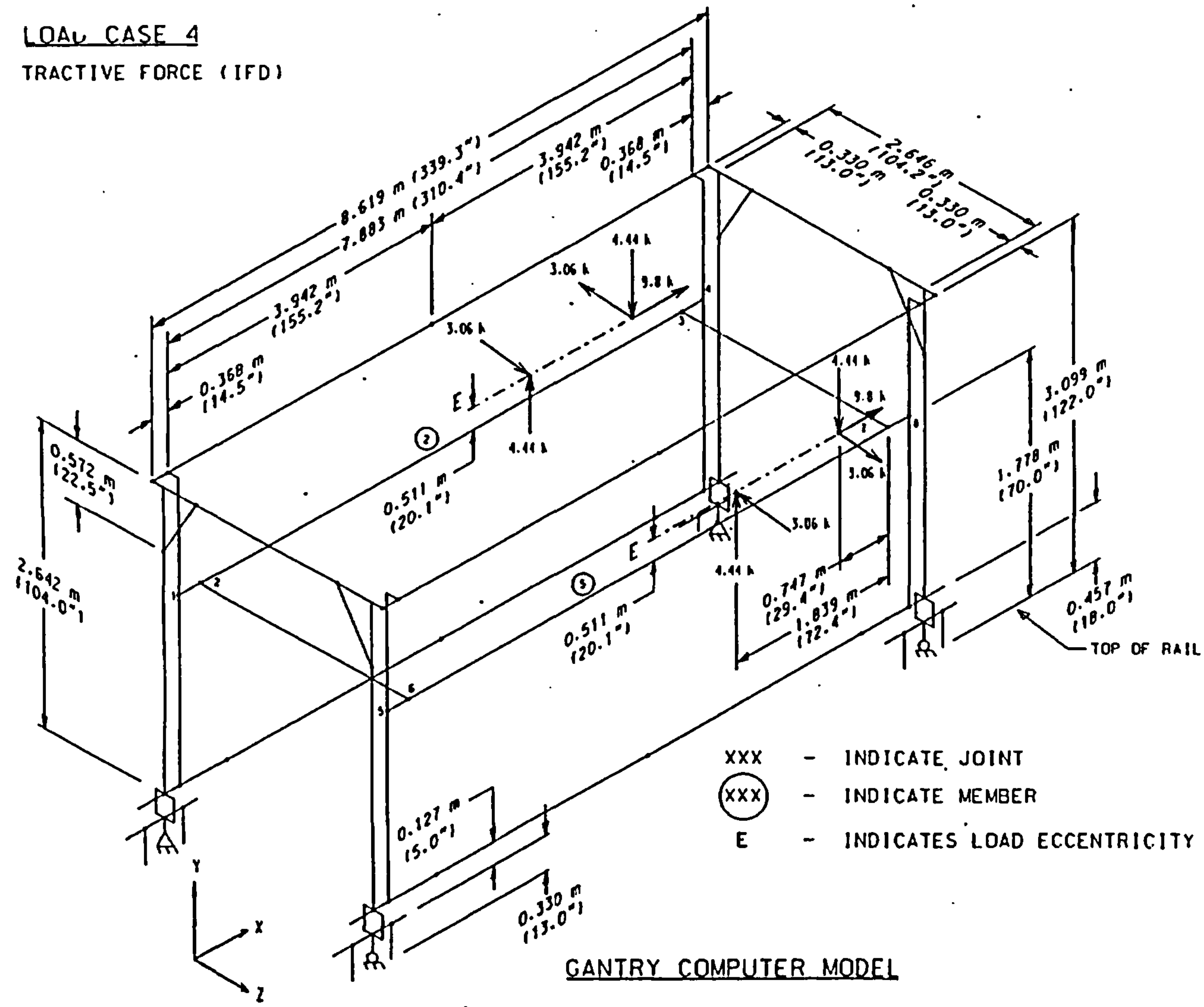
WASTE PACKAGE LOAD (LL)



GANTRY COMPUTER MODEL

LOAD CASE 4

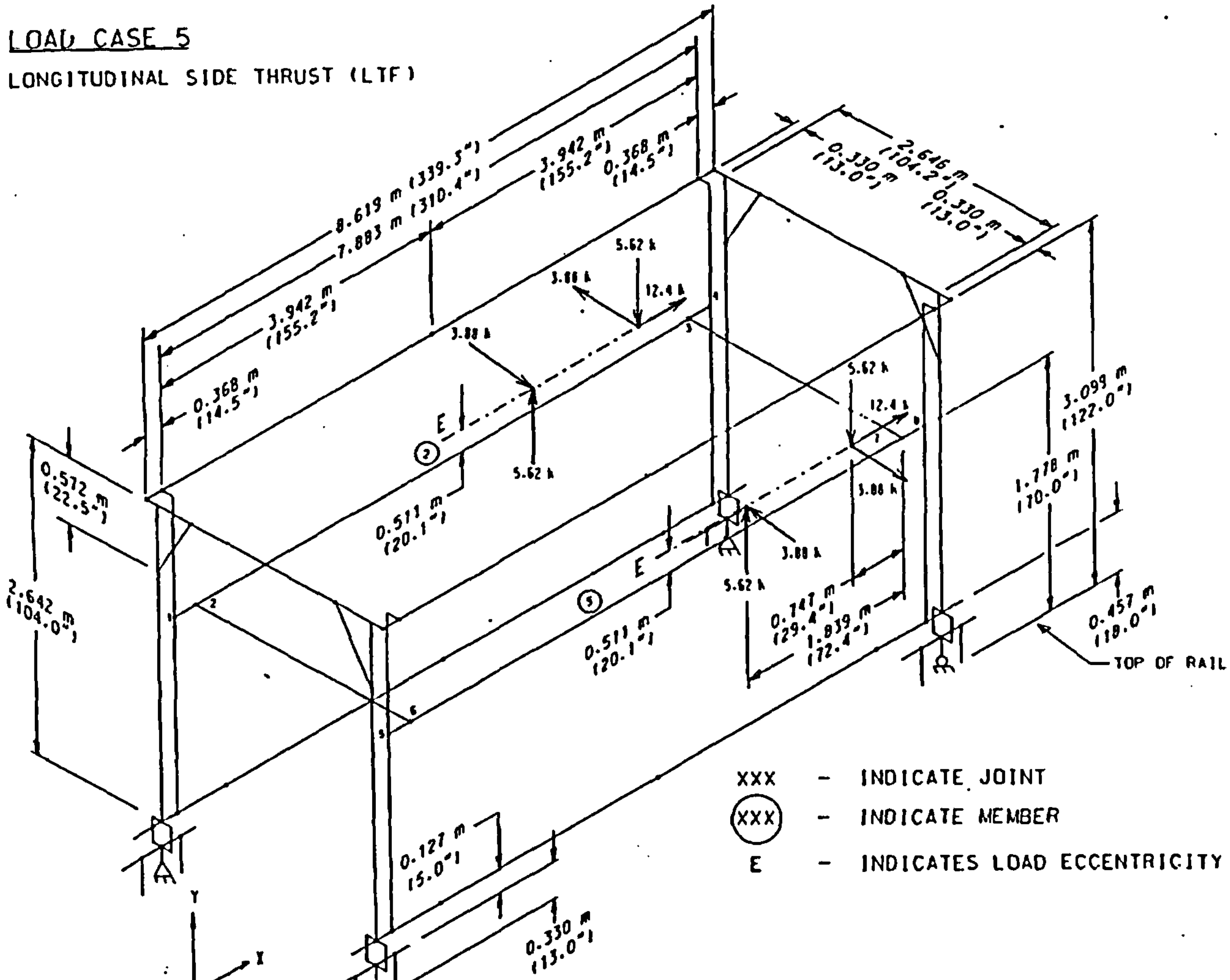
TRACTIVE FORCE (IFD)



GANTRY COMPUTER MODEL

LOAD CASE 5

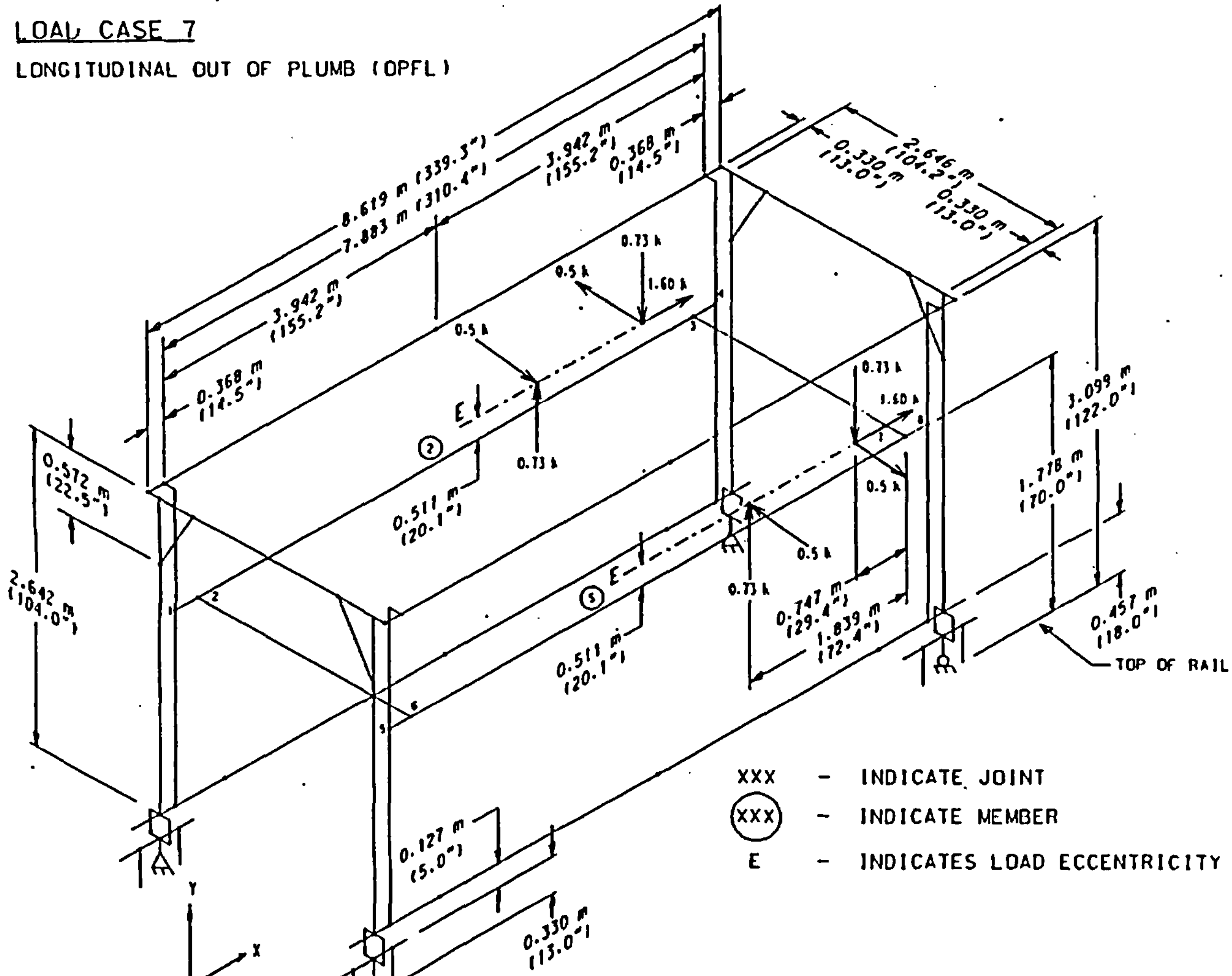
LONGITUDINAL SIDE THRUST (LTF)



GANTRY COMPUTER MODEL

LOAD CASE 7

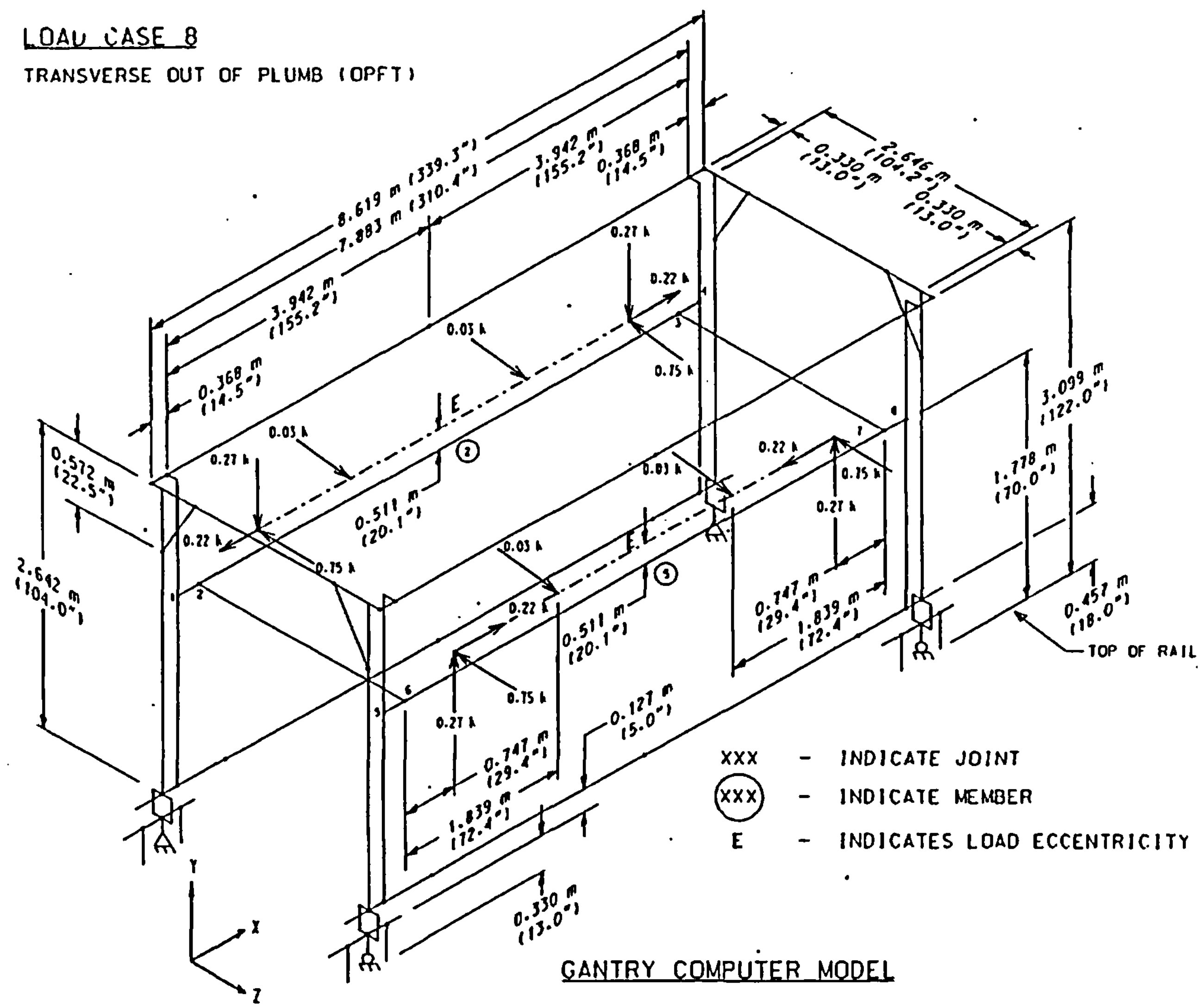
LONGITUDINAL OUT OF PLUMB (OPFL)



GANTRY COMPUTER MODEL

LOAD CASE 8

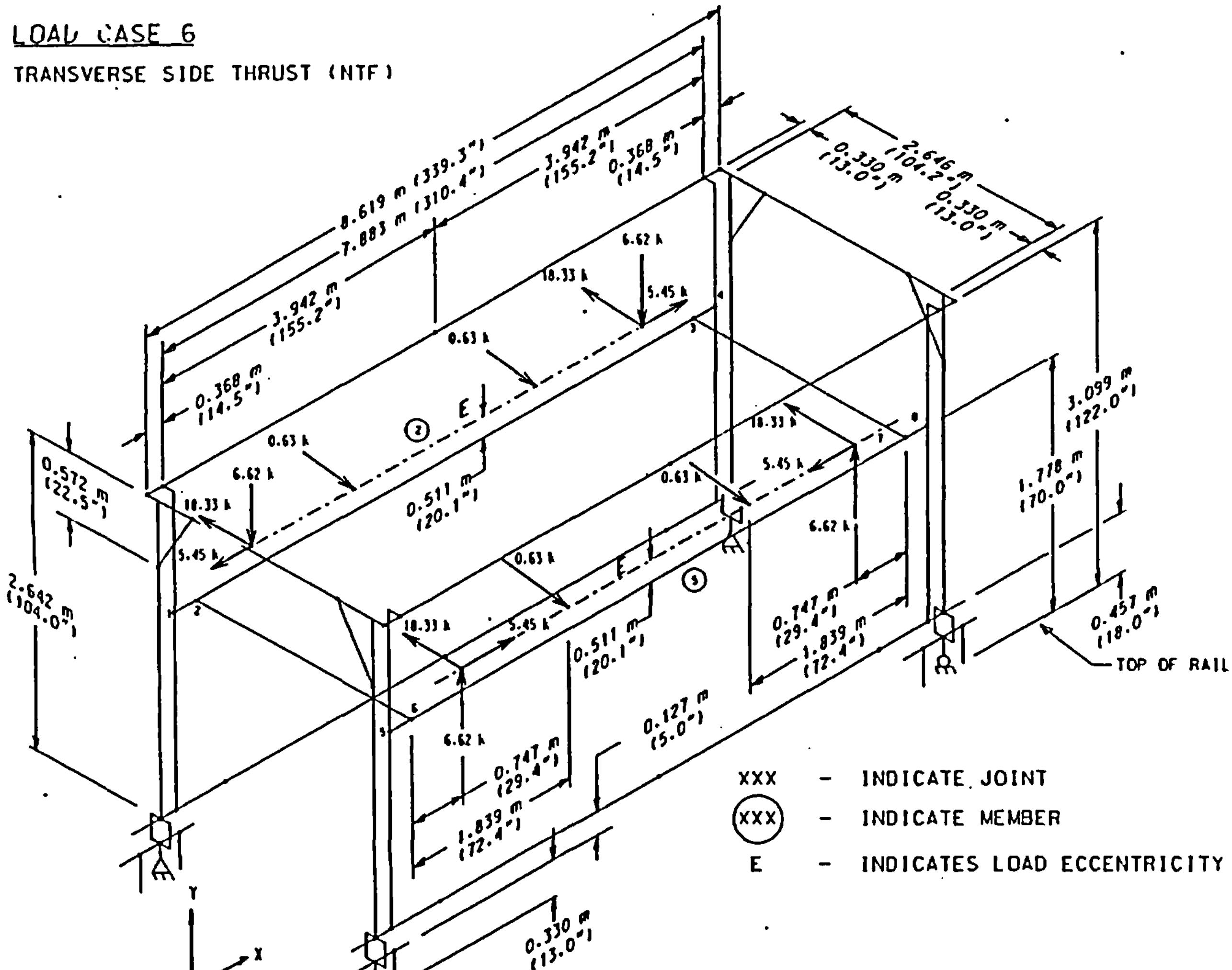
TRANSVERSE OUT OF PLUMB (OPFT)



GANTRY COMPUTER MODEL

LOAD CASE 6

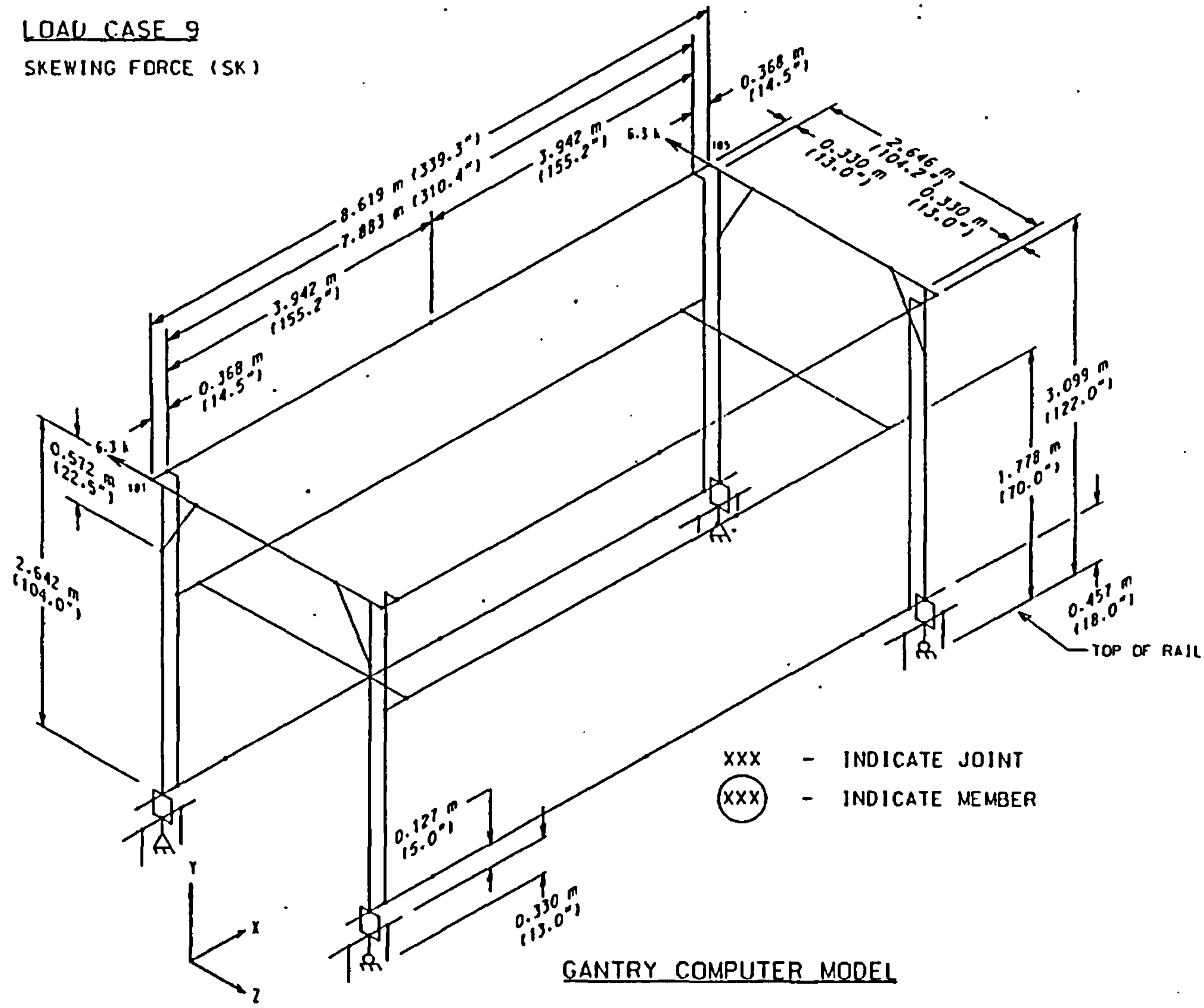
TRANSVERSE SIDE THRUST (NTF)



GANTRY COMPUTER MODEL

LOAD CASE 9

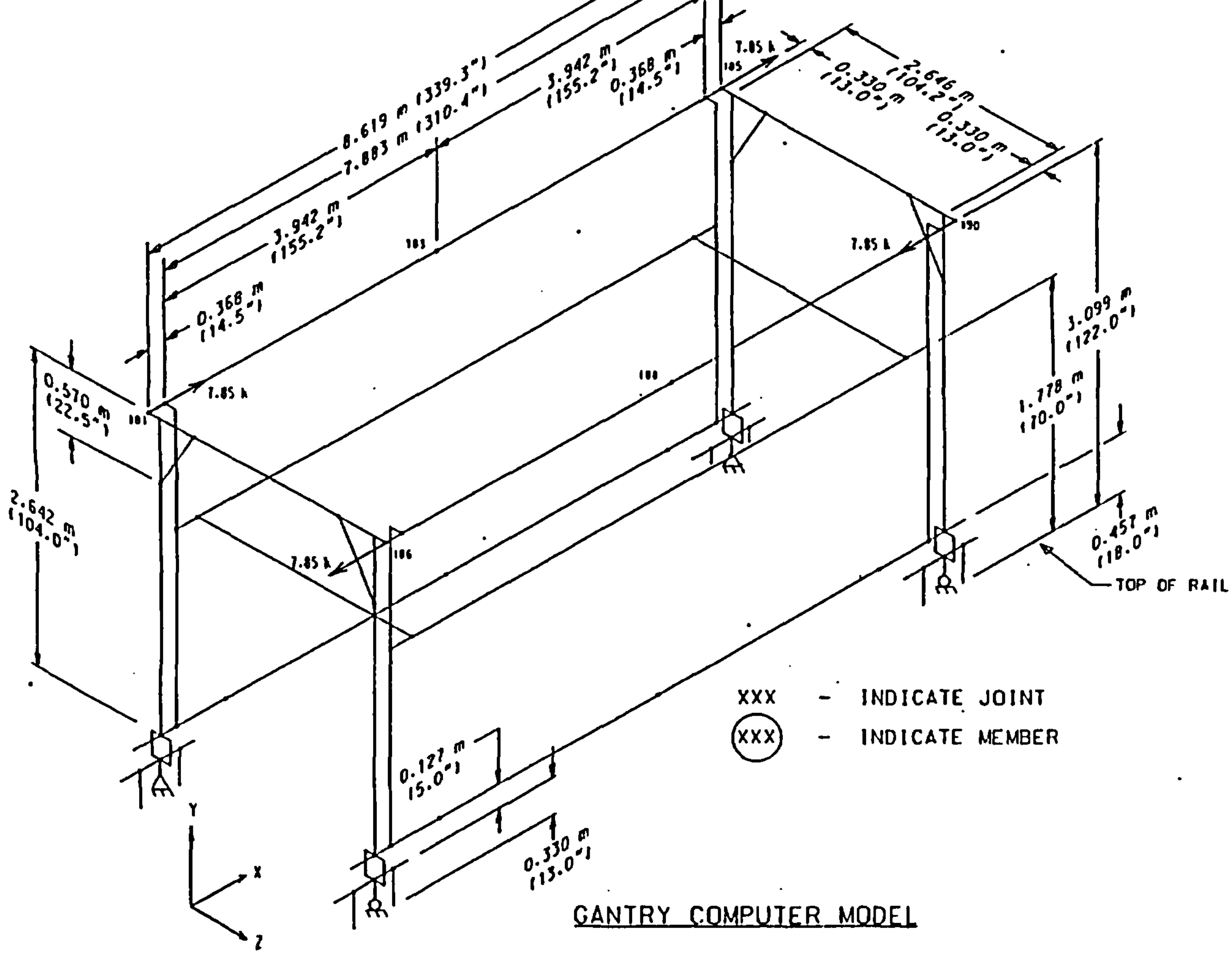
SKEWING FORCE (SK)



GANTRY COMPUTER MODEL

LOAD CASE 10

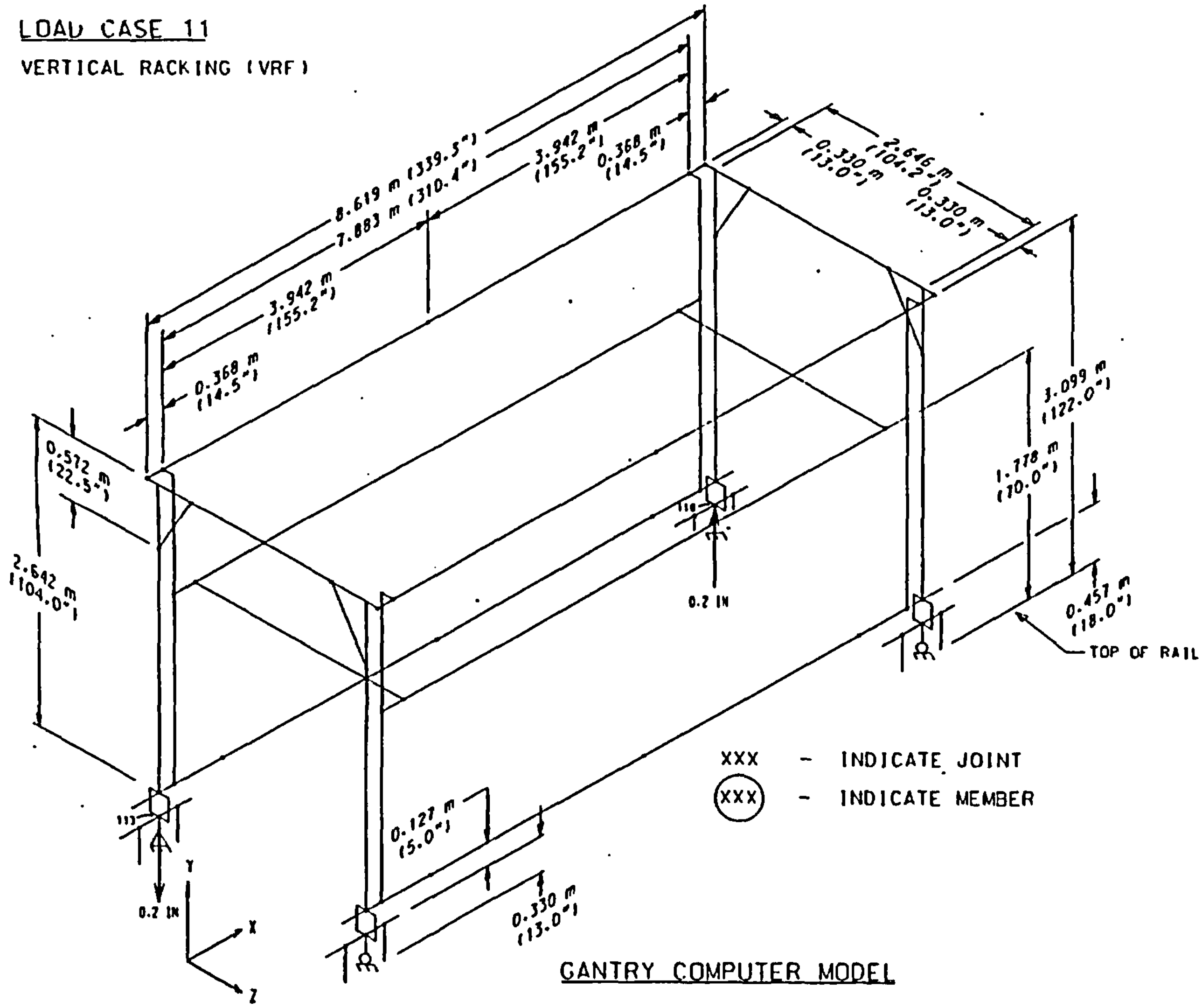
HORIZONTAL RACKING (HRF)



GANTRY COMPUTER MODEL

LOAD CASE 11

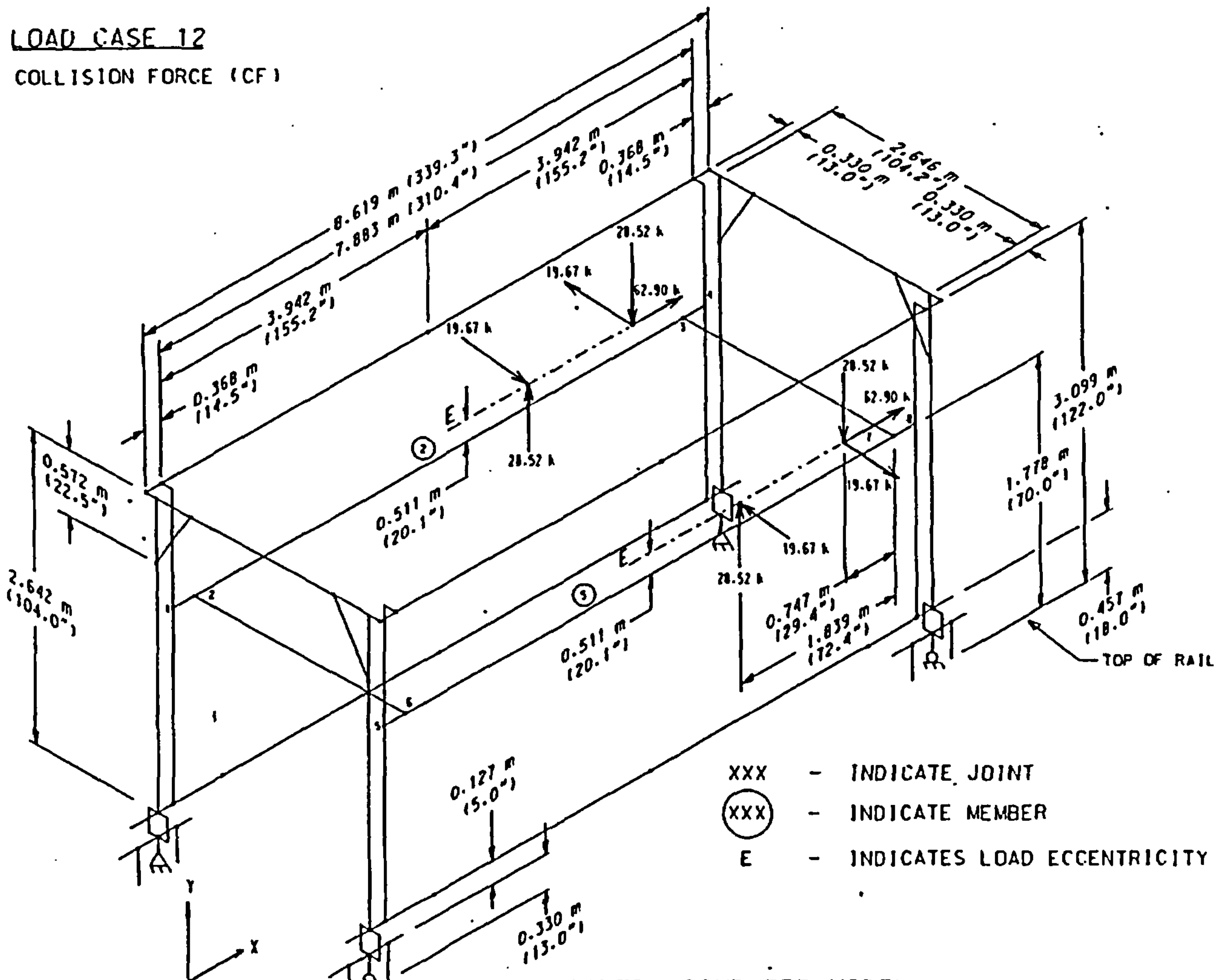
VERTICAL RACKING (VRF)



GANTRY COMPUTER MODEL

LOAD CASE 12

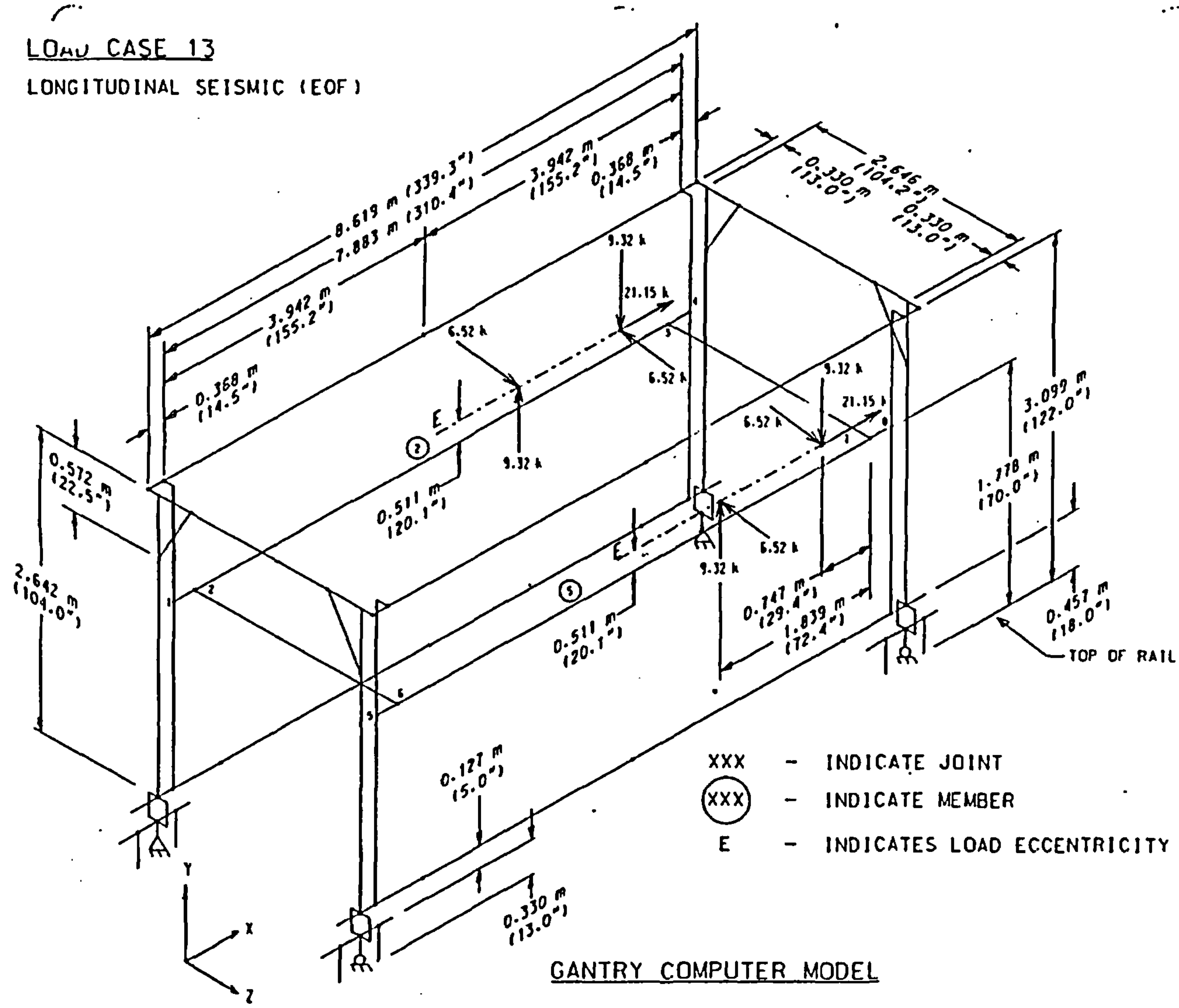
COLLISION FORCE (CF)



GANTRY COMPUTER MODEL

LOAD CASE 13

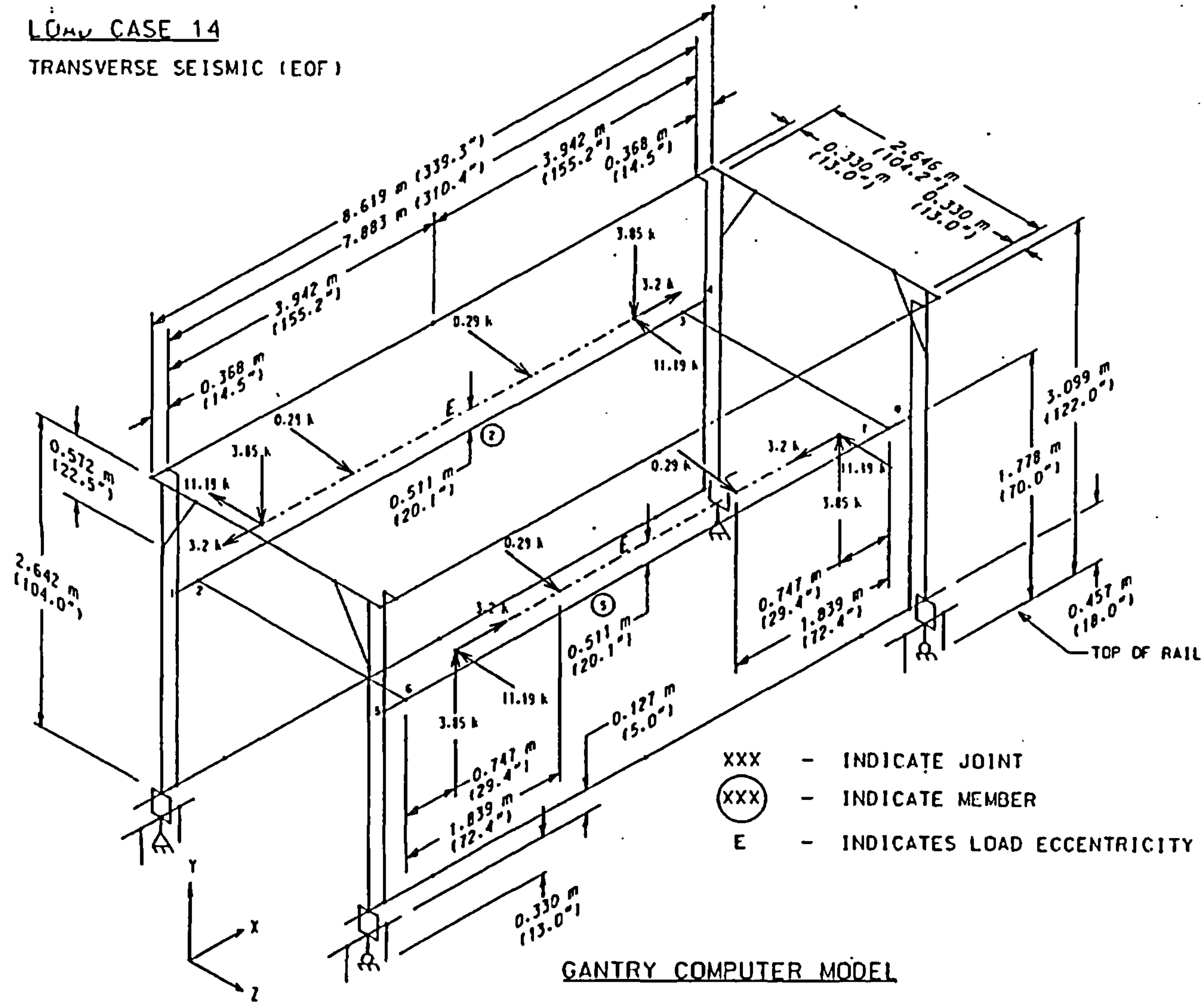
LONGITUDINAL SEISMIC (EOF)



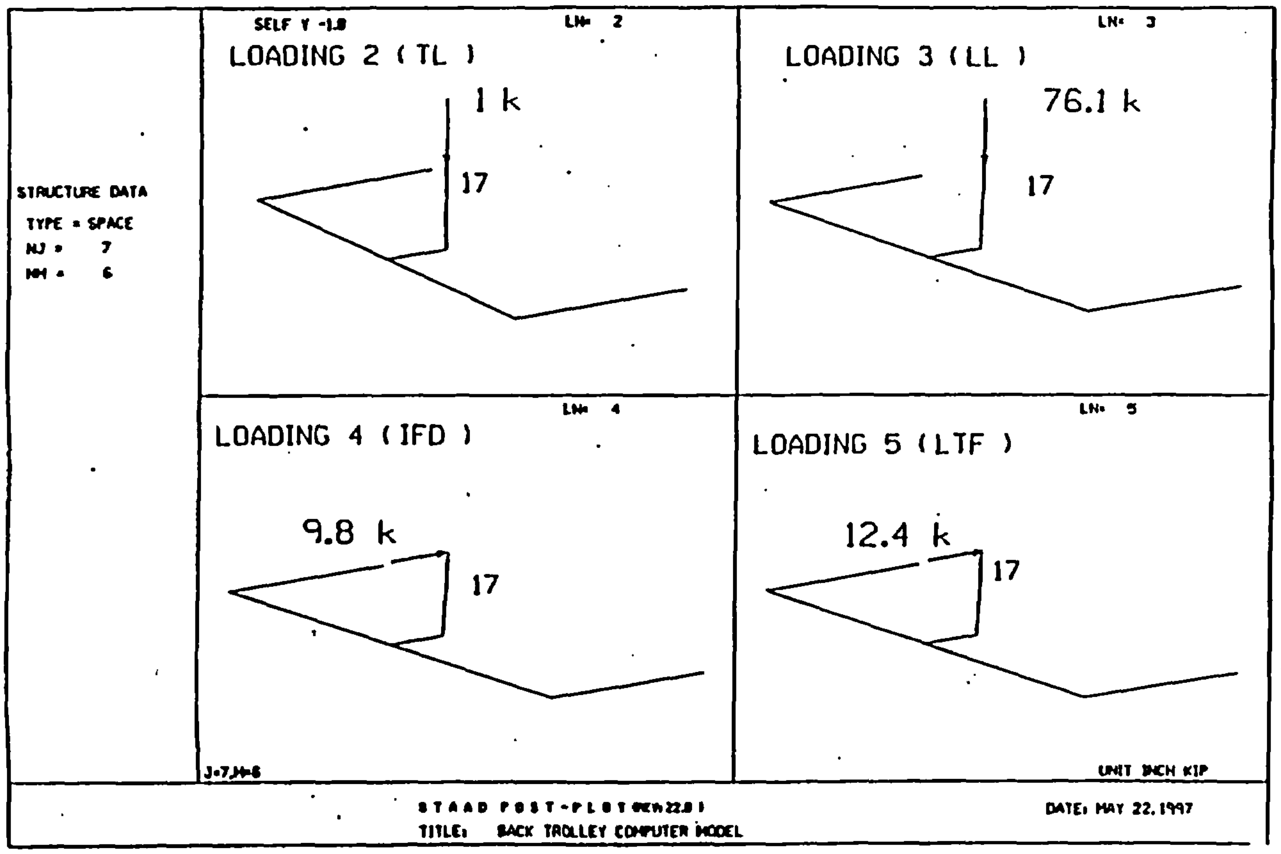
GANTRY COMPUTER MODEL

LOAD CASE 14

TRANSVERSE SEISMIC (EOF)



GANTRY COMPUTER MODEL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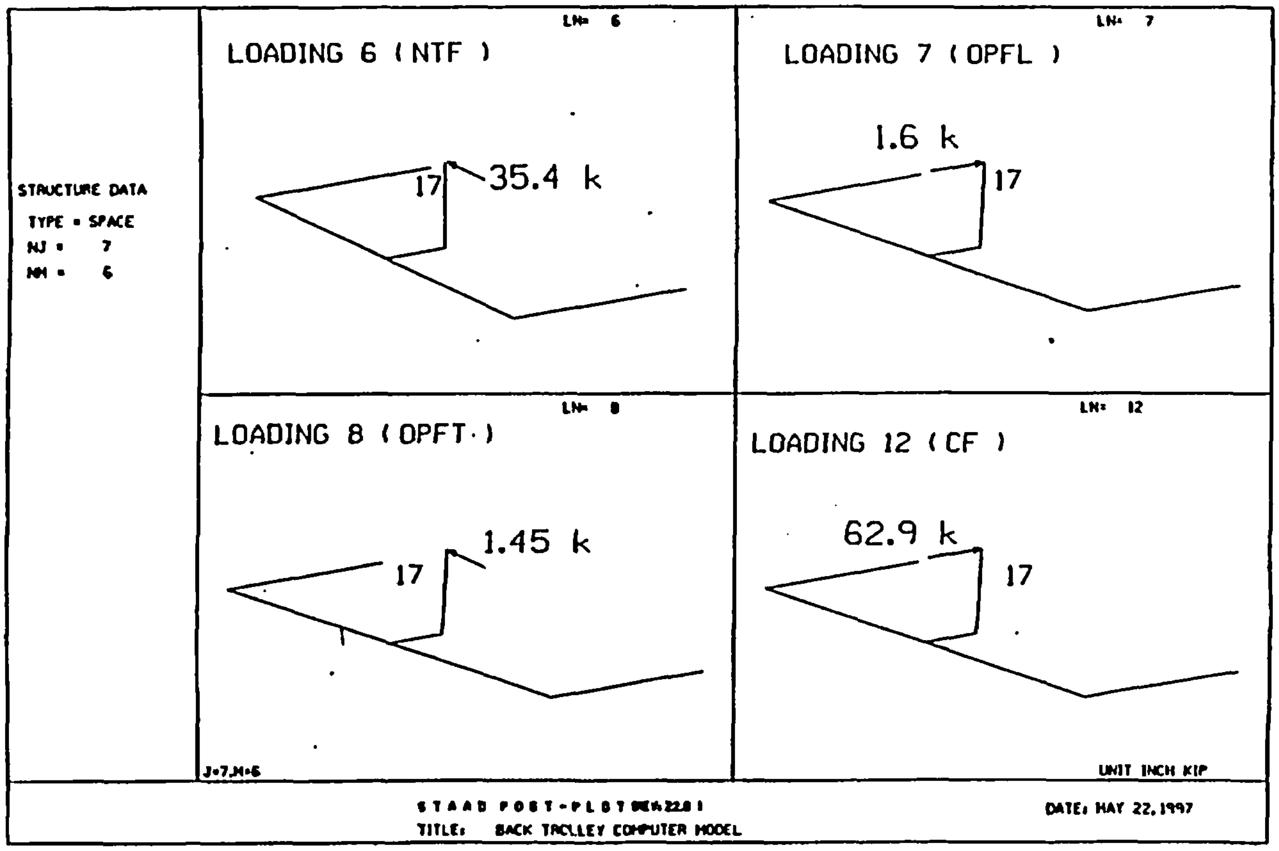
NM = 6
SELF Y -1.0
LN= 2
LOADING 2 (TL)
1 k
17
LN= 3
LOADING 3 (LL)
76.1 k
17
LN= 4
LOADING 4 (IFD)
9.8 k
17
LN= 5
LOADING 5 (LTF)
12.4 k
17
J=7,M=6
UNIT INCH KIP
STAAD POST-PLOT
TITLE: BACK TROLLEY COMPUTER MODEL
DATE: MAY 22,1997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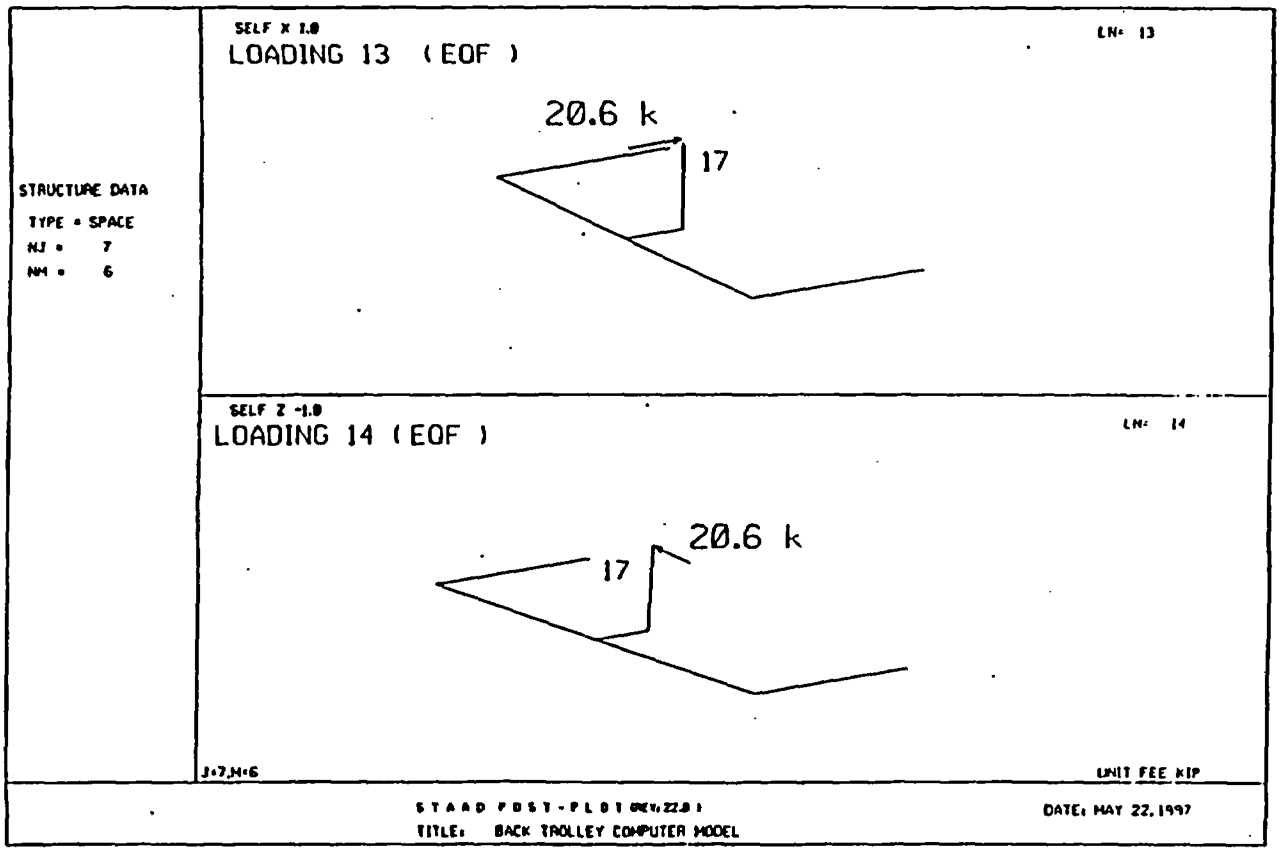
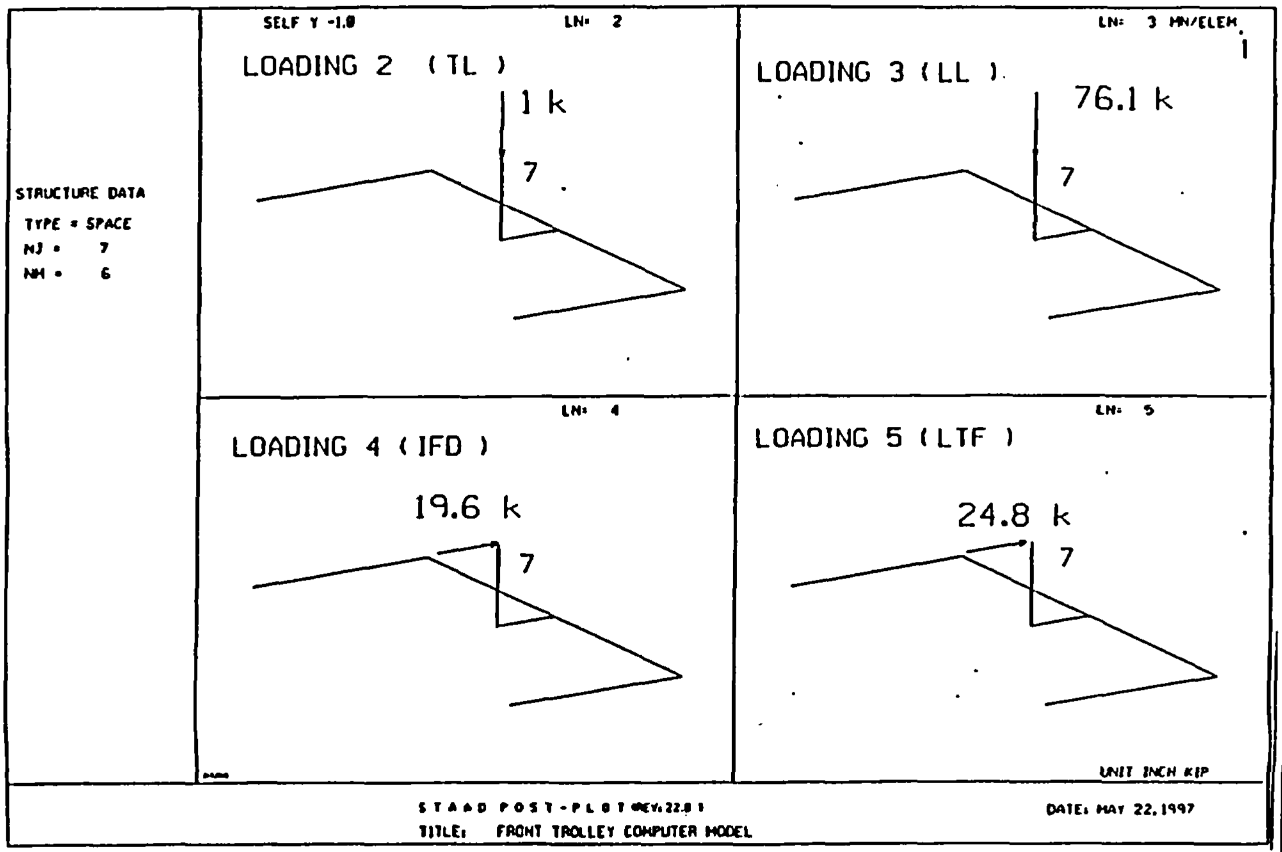
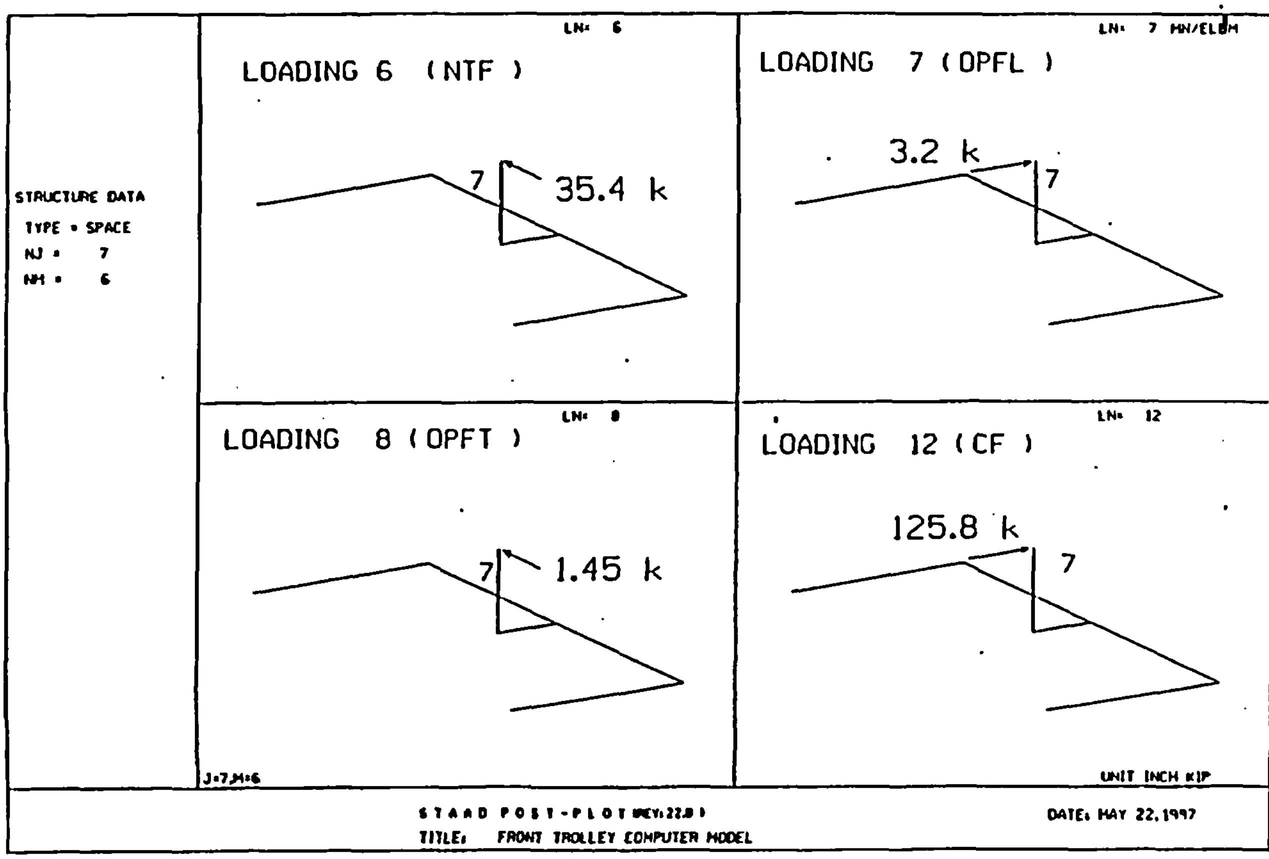
TYPE = SPACE
NJ = 7
NM = 6
LN= 6
LOADING 6 (NTF)
17
35.4 k
LN= 7
LOADING 7 (OPFL)
1.6 k
17
LN= 8
LOADING 8 (OPFT)
1.45 k
17
LN= 12
LOADING 12 (CF)
62.9 k
17
J=7,M=6
UNIT INCH KIP
STAAD POST-PLOT
DATE: MAY 22,1997
TITLE: BACK TROLLEY COMPUTER MODEL


SELF X 1.0
LOADING 13 (EOF)
LN: 13
20.6 k
17
STRUCTURE DATA
TYPE = SPACE
NJ = 7
NM = 6
SELF Z -1.0
LOADING 14 (EOF)
LN: 14
20.6 k
17
J=7,M=6
UNIT FEE KIP
STAAD POST-PLOT (REV.22.0)
DATE: MAY 22,1997
TITLE: BACK TROLLEY COMPUTER MODEL


STRUCTURE DATA
TYPE = SPACE
NJ = 7
NM = 6
SELF Y -1.0
LN: 2
LOADING 2 (TL)
1 k
7
LN: 3 MN/ELEM
1
LOADING 3 (LL)
76.1 k
7
LN: 4
LOADING 4 (IFD)
19.6 k
7
LN: 5
LOADING 5 (LTF)
24.8 k
7
UNIT INCH KIP
STAAD POST-PLOT
DATE: MAY 22, 1997
TITLE: FRONT TROLLEY COMPUTER MODEL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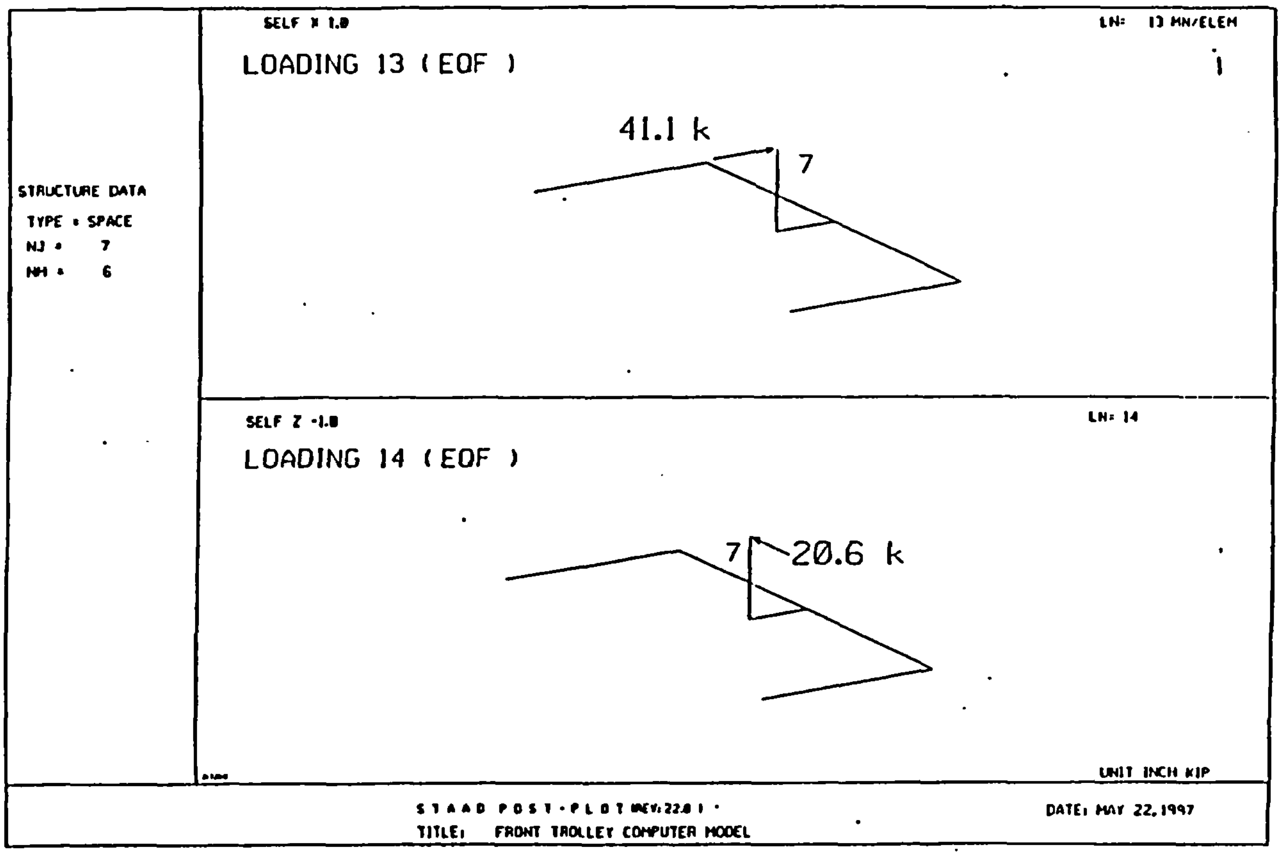
NM = 6
LN= 6
LOADING 6 (NTF)
7
35.4 k
LN= 7 MN/ELEM
LOADING 7 (OPFL)
3.2 k
7
LN= 8
LOADING 8 (OPFT)
7
1.45 k
LN= 12
LOADING 12 (CF)
125.8 k
7
J=7,M=6
UNIT INCH KIP
STAAD POST-PLOT (REV:22.0)
TITLE: FRONT TROLLEY COMPUTER MODEL
DATE: MAY 22,1997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NM = 6
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LN: 13 MN/ELEM
LOADING 13 (EQF)
41.1 k
7
SELF Z -1.0
LN: 14
LOADING 14 (EQF)
7
20.6 k
UNIT INCH KIP
STAAD POST-PLOT
DATE: MAY 22,1997
TITLE: FRONT TROLLEY COMPUTER MODEL

13.0 STABILITY ANALYSIS

GANTRY STABILITY ANALYSIS

gantry self-weight

from support reactions of load case 1 and 2 in STAAD III analysis " Gantry-H "

Gantry weight — load case 1

Support joint	weight (k)
113	21.49
118	21.48
123	21.48
124	21.48

Sub-total = 85.93 k

trolley weight — Load case 2

Support joint	weight (k)
113	2.71
118	2.71
123	2.71
124	2.71

Sub-total = 10.84 K

total gantry self-weight = 10.84 + 85.93 = 96.77 kip

96.77 kips / 2.205 kips per MT = 43.89 M ton

Less than estimated weight 45 M ton, therefore it is OK.

1. Hoist frame at high position by longitudinal direction forces IFD, LTF and OPFL.

Overturning force and moment

arm = distance from top of rail to top of lift lug. 4.051 m (159.5 in), Figure II - 6

loading No.	Description	force (k)	X	arm (in)	= moment (k-in)
4	IFD	19.60	X	159.5	= 3126.20
5	LTF	24.80	X	159.5	= 3955.60
7	OPFL	3.20	X	159.5	= 510.40
				M_{ot}	= 7592.20 k-in

resistance force and moment

arm = distance between center of gantry columns. 8.619 m (339.3 inch), Figure II - 3

support joint 113

loading No.	force	Reaction force	X	arm	= moment
1	gantry	21.49	X	339.3	= 7291.56
2	trolley	2.71	X	339.3	= 919.50
3	WP	38.05	X	339.3	= 12910.4
Support joint 123					
1	gantry	21.48	X	339.3	= 7288.16
2	trolley	2.71	X	339.3	= 919.50
3	WP	38.05	X	339.3	= 12910.4
				M_R	= 42239.46 k-in

safety factor = M_R / M_{ot} = 42239.46 / 7592.20 = 5.56 > 1.5 OK

2 a. Hoist frame at high position by transverse direction forces NTF and OPFT

Overturning force and moment

loading No.	Description	overturn force (k)	X	arm (in)	= moment (k-in)	
6	NTF	70.80	X	159.5	= 11292.60	
8	OPFT	2.90	X	159.5	= 462.55	
				M_{ot}	= 11755.15	k-in

Resistance force and moment

arm = distance between center of gantry columns. 2.646 m (104.2 in), Figure II - 6

support joint 123

loading No.	Description	Resistant force	X	arm	= moment	
1	gantry	21.48	X	104.2	= 2238.22	
2	trolley	2.71	X	104.2	= 282.38	
3	WP	38.05	X	104.2	= 3964.81	
Support joint 128						
1	gantry	21.48	X	104.2	= 2238.216	
2	trolley	2.71	X	104.2	= 282.38	
3	WP	38.05	X	104.2	= 3964.81	
				M_R	= 12970.82	k-in

safety factor = M_R / M_{ot} = 12970.82 / 11755.15 = 1.1 < 1.5 NO GOOD

highest position fails to provide safety factor against overturning of 1.5.

b. Check hoist beam by transverse direction forces NTF and OPFT in mid. position which is higher than lifting position*. Figure II - 6 (70.0"+39.6"=109.6" > 93.94")

6	NTF	70.80	X	109.6"	= 7759.68	
8	OPFT	2.90	X	109.6"	= 317.84	
				M ot	= 8077.52	k-in

safety factor = M_R / M_{ot} = 12970.82 / 8077.52 = 1.61 > 1.5 OK

* lifting position of WP on Reusable Rail Car. Top lug to rail =1.380+1.006=2.386 m (93.94")

3. Hoist frame at high position by transverse direction forces OPFT and SK

Overturning force and moment

loading No.	Description	overturn force	X	arm	= moment	
		(k)		(in)	(k-in)	
8	OPFT	2.90	X	159.5	= 462.55	
9	SK	12.60	X	122.0 *	= 1537.20	
				M_{ot}	= 1999.75	k-in

* distance from center line of top beam of gantry frame to top of rail = 3.099 m (122.0 inch), (Gantry Computer Model-Members Diagram, Attachment II, section 10.0)

Resistance force and moment

support joint 123

loading No.	Description	Resistant force	X	arm	= moment	
1	gantry	21.48	X	104.2	= 2238.22	
2	trolley	2.71	X	104.2	= 282.38	
3	WP	38.05	X	104.2	= 3964.81	
Support joint 128						
1	gantry	21.48	X	104.2	= 2238.216	
2	trolley	2.71	X	104.2	= 282.38	
3	WP	38.05	X	104.2	= 3964.81	
				M_R	= 12970.82	k-in

safety factor = M_R / M_{ot}	=	12970.82	/ 1999.75	=	6.49	> 1.5 OK

4. Hoist frame at high position by longitudinal direction forces IFD, OPFL, WP EQF, and Individual member EQF.

Overturning force and moment

loading No.	Description	overturn force (k)	X	arm (in)	=	moment (k-in)	
4	IFD	19.60	X	159.5	=	3126.20	
7	OPFL	3.20	X	159.5	=	510.40	
EQF	by self-weight						
	WP EQF	41.08	X	159.5	=	6552.26	
	Trolleys	2.93 *	X	159.5	=	467.34	
	Trolley drives	0.05 *	X	148.0 **	=	7.40	
	Trolley screws	0.41*	X	142.0 **	=	58.22	
	Hoist frame drives	0.43 *	X	132.0 **	=	56.76	
	Top beams	2.02 *	X	122.0 **	=	246.44	
	End beams	1.42 *	X	122.0 **	=	173.24	
	Hoist beams	3.90 *	X	119.9 **	=	467.61	
	Hoist beams	0.31 *	X	119.9 **	=	37.17	
	Electric control cabinets	6.70 *	X	114.0 **	=	763.80	
	Columns	3.47 *	X	75.0 **	=	260.25	
	Guiding channels	1.21 *	X	75.0 **	=	90.75	
	Hoist screws	0.43 *	X	75.0 **	=	32.25	
	Bottom beams	1.16 *	X	18.0 **	=	20.88	
	Bottom beams	0.83 *	X	18.0 **	=	14.94	
	Wheel beams	0.38 *	X	18.0 **	=	6.84	
	Bogie drives	0.37 *	X	18.0 **	=	6.66	
	Bogie houses	0.43 *	X	13.0 **	=	5.59	
				M_{ot}	=	12904.99	k-in

* see following spread sheet.
** distance from gravity center line of member to top of rail. (Gantry Computer Model-Members Diagram, Attachment II, section 10.0)

Resistance force and moment

support joint 113

loading No.	Description	Resistant force (k)	X	arm (in)	= moment (k-in)
1	gantry	21.49	X	339.3	= 7291.56
2	trolley	2.71	X	339.3	= 919.50
3	WP	38.05	X	339.3	= 12910.37
Support joint 123					
1	gantry	21.48	X	339.3	= 7288.164
2	trolley	2.71	X	339.3	= 919.50
3	WP	38.05	X	339.3	= 12910.37
				M_{R}	= 42239.46 k-in

safety factor = M_R / M_{ot} = 42239.46 / 12904.99 = 3.27 > 1.1 OK

5. Hoist frame at high position by transverse direction forces OPFT, WP EQF, and individual member EQF.

Overturning force and moment

loading No.		Description	overturn force (k)	X	arm (in)	=	moment (k-in)
8		OPFT	2.90	X	159.5	=	462.55
	EQF	by self-weight					
		WP EQF	41.08	X	159.5	=	6552.26
		Trolleys	2.93 *	X	159.5	=	467.34
		Trolley drives	0.05 *	X	148.0 **	=	7.40
		Trolley screws	0.41*	X	142.0 **	=	58.22
		Hoist frame drives	0.43 *	X	132.0 **	=	56.76
		Top beams	2.02 *	X	122.0 **	=	246.44
		End beams	1.42 *	X	122.0 **	=	173.24
		Hoist beams	3.90 *	X	119.9 **	=	467.61
		Hoist beams	0.31 *	X	119.9 **	=	37.17
		Electric control cabinets	6.70 *	X	114.0 **	=	763.80
		Columns	3.47 *	X	75.0 **	=	260.25
		Guiding channels	1.21 *	X	75.0 **	=	90.75
		Hoist screws	0.46 *	X	75.0 **	=	34.50
		Bottom beams	1.16 *	X	18.0 **	=	20.88
		Bottom beams	0.83 *	X	18.0 **	=	14.94
		Wheel beams	0.38 *	X	18.0 **	=	6.84
		Bogie drives	0.43 *	X	18.0 **	=	7.74
		Bogie houses	0.37 *	X	13.0 **	=	4.81
					M_{ot}	=	9733.49 k-in

* see following spread sheet.

** distance from gravity center line of member to top of rail. (Gantry Computer Model-Members Diagram, Attachment II, section 10.0)

Resistance force and moment

support joint 123

loading No.	Description	Resistant force (k)	X	arm (in)	= moment (k-in)
1	gantry	21.48	X	104.2	= 2238.22
2	trolley	2.71	X	104.2	= 282.38
3	WP	38.05	X	104.2	= 3964.81
Support joint 128					
1	gantry	21.48	X	104.2	= 2238.216
2	trolley	2.71	X	104.2	= 282.38
3	WP	38.05	X	104.2	= 3964.81
				M_R	= 12970.82 k-in

safety factor = M_R / M_{ot} = 12970.82 / 9733.49 = 1.33 > 1.1 OK

6. Hoist frame at high position by longitudinal direction forces CF and OPFL.

Overturn force and moment

loading No.	Description	overturn force	X	arm	= moment	
7	OPFL	3.20	X	159.5	= 510.40	
12	CF	125.80	X	159.5	= 20065.10	
				M_{ot}	= 20575.50	k-in

Resistance force and moment

support joint 113

loading No.	Description	overturn force	X	arm	= moment	
1	gantry	21.49	X	339.3	= 7291.56	
2	trolley	2.71	X	339.3	= 919.50	
3	WP	38.05	X	339.3	= 12910.37	
Support joint 128						
1	gantry	21.48	X	339.3	= 7288.16	
2	trolley	2.71	X	339.3	= 919.50	
3	WP	38.05	X	339.3	= 12910.37	
				M_R	= 42239.46	k-in

safety factor = M_R / M_{ot} = 42239.46 / 20575.5 = 2.05 > 1.1 OK

Individual member EQF

Individual member EQF = weight of member X horizontal seismic ground acceleration (0.27 g).

Weight of member = X-section area of member X length of member X 490 lb/ cf X number of units

description	X-section Area (sf)		length (ft)		Density (kips/cf)		weight (Kips)		number Of units		Acc. (G)		Seismic Force (kips)
trolley							5.42	X	2	X	0.27	=	2.93
hoist beams	0.57	X	25.86	X	0.49	=	7.22	X	2	X	0.27	=	3.9
hoist beams	0.15	X	7.85	X	0.49	=	0.58	X	2	X	0.27	=	0.31
top beams	0.27	X	28.23	X	0.49	=	3.73	X	2	X	0.27	=	2.02
end beams	0.75	X	7.18	X	0.49	=	2.64	X	2	X	0.27	=	1.42
columns	0.69	X	9.50	X	0.49	=	3.21	X	4	X	0.27	=	3.47
guide channels	0.24	X	9.50	X	0.49	=	1.12	X	4	X	0.27	=	1.21
bottom beams	0.14	X	22.30	X	0.49	=	1.53	X	2	X	0.27	=	0.83
bottom beams	0.29	X	7.55	X	0.49	=	1.07	X	4	X	0.27	=	1.16
wheel beams	0.14	X	5.18	X	0.49	=	0.36	X	4	X	0.27	=	0.38
electric control cabinets							6.2	X	4	X	0.27	=	6.7
trolley drives							0.1	X	2	X	0.27	=	0.05
trolley screws	0.03	X	25.5			=	0.77	X	2	X	0.27	=	0.41
hoist frame drives							0.8	X	2	X	0.27	=	0.43
hoist frame screws							0.43	X	4	X	0.27	=	0.46
bogie drives							0.34	X	4	X	0.27	=	0.37
bogie housings							0.40	X	4	X	0.27	=	0.43

ATTACHMENT III

WASTE PACKAGE TRANSPORTER - MECHANICAL EQUIPMENT SELECTIONS

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural-steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

WP TRANSPORTER MECHANICAL EQUIPMENT SELECTIONS

1.0 PURPOSE

This calculation will provide the basis for selecting and/or verifying the following based on the design concept shown in Figures 7.2.1 through 7.2.4:

1. Transporter wheel load.
2. Transporter wheel and rail selection.
3. Door operator drive.
4. Unloader system.
5. Unloader system weight.

2.0 INITIAL DESIGN SELECTIONS

No. Input Description

1. Maximum load on one truck is 55% of the total Transporter operating weight, which is considered good engineering practice to accommodate a minor degree of load shifting.
2. Maximum operating weight of the reusable rail car is 80 MT. (Attachment IV, Section 3.2)
3. Brinell hardness of Transporter wheels is 615 BHN, which is indicated in Ref. 4.4.5 Table 4.13.3-4 as the hardness required for bridge crane wheels with the maximum capacity.
4. The Transporter truck wheel has a maximum diameter of 762 mm (30 in.) to attain the lowest profile and still have the wheel load capacity of a standard bridge crane wheel of that size.
5. Transporter door swing is 270° to provide minimum interference with isolation doors and opening time is approximately 1 minute, which is a reasonable and safe rotational rate for a large mass such as the Transporter door.
6. Transporter rail car unloader shall have a minimum travel of 12.0 m determined as follows:

   A. From Figure 7.2.3, outside dimension of Transporter shield is 7.38 m.

   B. From Figure 7.4.3, outside dimension of Gantry is 11.719 m.

No. Input Description

C. The length of the WP on the Rail Car is centered in the Transporter shield.

D. End of Transporter shield is flush with transfer dock face for the WP unloading operation

The unloader travel is the total of the following distances the center of the WP travels in the unloading operation:

Middle of shield to transfer dock face	$= \frac{7.38\ m}{2}$	= 3.69 m
Transfer dock face to end of Gantry		= 1.0 m
End of Gantry to middle of Gantry	$= \frac{11.719\ m}{2}$	= 5.86 m
	Total Travel	10.55 m
	Rounded up to 12 m	

E. The length of the WP on the Rail Car in the unloader extended position is centered under the Gantry.

F. The end of the Gantry in the lift-off position over the WP shall be 1 m behind face of transfer dock.

7. Reusable Rail Car unloading speed shall not exceed 7.6 m/min (25 ft/min) (Ref. 4.4.5, pg. 80). Suggested slow bridge operating speed for 150-ton bridge crane is 7.6 m/min (25 ft/min).

8. A distance of 1.32 m from centerline of largest diameter WP (2.0 m) mounted on the Rail Car to the top of the rail will be used. This is considered a reasonable dimension for the purpose of WP transport and transfer into the emplacement drift on a reusable Rail Car. This dimension does not address tolerances due to the primary nature of this analysis.

9. The distance from the top of the Transporter rail to the top of the Rail Car rail is 1.28 m (Ref. 5.6, Fig. 8.6.4-1).

10. Maximum transport locomotive speed is 8 km/hr (5 mph) (Ref. 5.6, page E-3).

3.0 SOLUTIONS

3.1 Transporter Wheel Load

From Attachment I, Section 10, Parts:

The maximum Transporter operating weight = 233.15 MT
Round down to 233 MT (257.5 tons)

From 4.3.10:

Maximum waste package weight is 69,000 kg or 69 MT.

Minimum operating weight = maximum Transporter operating - maximum WP weight

= 233 MT - 69 MT = 164 MT (181.2 tons)

From Design Selection No. 1:

Maximum load on one of the two 4-wheel trucks is 55% of total operating weight.

Converting to English units for wheel selection using English unit formula

Maximum truck load = (0.55) (257.5 tons x 2000 lb/ton) = 283,250 lbs

Maximum wheel load $= \frac{283{,}250\ lbs}{4\ wheels}$ = 70,813 lbs

For the same truck:

Minimum truck load = (0.55) (181.2 tons x 2000 lb/ton) = 199,320 lbs

Minimum wheel load $= \frac{199{,}320\ lbs}{4}$ = 49,830 lbs

The above wheel loads are for a static condition and require adjustments for speed, load and service factors as follows:

From Ref. 4.4.5, pg. 49:

The equipment durability wheel load Pe is:

Pe = maximum wheel load x K_{WL}

where: K_{WL} = wheel load service coefficient
$= K_W \times C_S \times S_m$
K_W = mean effective load factor
C_S = speed factor
S_m = wheel service factor

From Ref. 4.4.5, pg. 33:

$$K_W = \frac{2(maximum\ load) + (minimum\ load)}{3(maximum\ load)} = \frac{2(70{,}813\ lbs) + (49{,}830\ lbs)}{3(70{,}813\ lbs)} = 0.90$$

From Section 4.3.27:

The maximum speed for Transporter is 5 mph or:

5 mi/hr x 5280 ft/mi x hr/60 min = 440 ft/min.

From Design Selection No. 5:

The Transporter wheels are 30 in. diameter

$$\text{Wheel speed} = \frac{1\ rev}{(\pi)(30)\,in.} \times 440\ \text{ft/min.} \times 12\ \text{in/ft} = 56.0\ \text{rpm}$$

From Ref. 4.4.5, pg. 49:

For rpm ≥ 31.5

$$C_S = 1 + \left(\frac{rpm - 31.5}{328.5}\right) = 1 + 0.075 = 1.075$$

From Ref. 4.4.5, Section 2.7:

The Transporter application is considered a Class F service which includes specially designed cranes for continuously handling loads approaching maximum capacity under severe conditions with the highest reliability.

From Ref. 4.4.5, pg. 50:

For Class F service crane:

$S_m = 1.45$

The equipment durability wheel load Pe is then:

Pe = maximum wheel load x K_{WL}

$$\begin{aligned} K_{WL} &= K_W \times C_S \times S_m \\ &= 0.9 \times 1.075 \times 1.45 \\ &= 1.40 \end{aligned}$$

Pe = 70,813 lbs x 1.40 = 99,138 lbs

For this application, the basic allowable wheel load (B_{WL}) is indicated in Ref. 4.4.5, pg. 47:

and is the product of:

$$B_{WL} = DWK$$

where: D = wheel diameter (in.)
W = effective rail head width (in.) (head width - 2 x corner radius)

$$K = 1300\left(\frac{BHN}{260}\right)^{0.33}$$ for wheels with a hardness of 260 or greater

From Design Selection No. 3:

Transporter wheel hardness = 615 BHN

$$K = 1300\left(\frac{615}{260}\right)^{0.33} = 1727$$

From Section 4.3.24:

The rail is 57 kg/m (115 lb/yd) AREA

From Attachment XII, Section 1, pg. 26:

$$\begin{aligned} W &= 2\text{-}29/32 \text{ in.} - (2)(3/8") \\ &= 2\text{-}29/32 \text{ in.} - 24/32 \text{ in.} \\ &= 2\text{-}5/32 \text{ in.} = 2.156 \text{ in.} \end{aligned}$$

Basic allowable wheel load is then:

$$\begin{aligned} B_{WL} &= DWK \\ &= (30 \text{ in.})(2.156 \text{ in.})(1727) \\ &= 111{,}712 \text{ lbs} \end{aligned}$$

The durability wheel load (Pe) of 99,138 lbs is less than the B_{WL} of 111,712 lbs.

From Ref. 4.4.5, Table 4.13.3-4:

For crane wheels with 615 BHN hardness, the respective wheel loadings are based on the corresponding rails having a minimum hardness of 320 BHN.

The 762 mm (30 in.) diameter crane wheel with a 58 RC or 615 BHN hardness when used with a 57 kg/m (115 lb/yd) AREA rail hardened to 320 BHN is sufficient for this application.

3.2 Door Operator Drive Selection

3.2.1 Determine weight of door (from Figures 1 and 2):

$$A_1 = \frac{1}{4}\pi R^2 = \frac{\pi}{4}(1.46\text{ m})^2$$

$$= (0.785)(2.13\text{ m}^2) = 1.67\text{ m}^2$$

$$A_2 = (1.460\text{ m})(1.618\text{ m}) = 2.36\text{ m}^2$$

From Section 4.3.18:

Door composition section is:

1 layer	76.2 mm (B-poly) at 920 (kg/m^3)
2 layers	5 mm S/S at 7949.7 (kg/m^3)
1 layer	177.8 carbon steel at 7832 (kg/m^3)

Weight per m^2 = density (kg/m^3) thickness (m)

Weight for one square meter of door:

Materials	Density (kg/m^3)	Thickness (m)	Wt/m^2 (kg/m^2)
B-poly	920	0.762	701.0
2 layers S/S	7949.7	0.010	79.5
Carbon steel	7832	0.178	1394.0
		Total	2174.5 kg/m^2

round up to 2175 kg/m^2

Weight for each of the two area:

$$WT_{A_1} = 1.67\text{ m}^2 \times 2175\text{ kg/m}^2 = 3632\text{ kg}$$
$$WT_{A_2} = 2.36\text{ m}^2 \times 2175\text{ kg/m}^2 = 5133\text{ kg}$$

Weight for three carbon steel door hinges:

$$WT = (3)\frac{\pi}{4}(D_0^2 - D_1^2)(0.260\text{ m})(7949.7\text{ kg/m}^3)$$

where: D_0 = outside diameter = 0.457 m
D_1 = inside diameter = 0.178 m

Figure 1

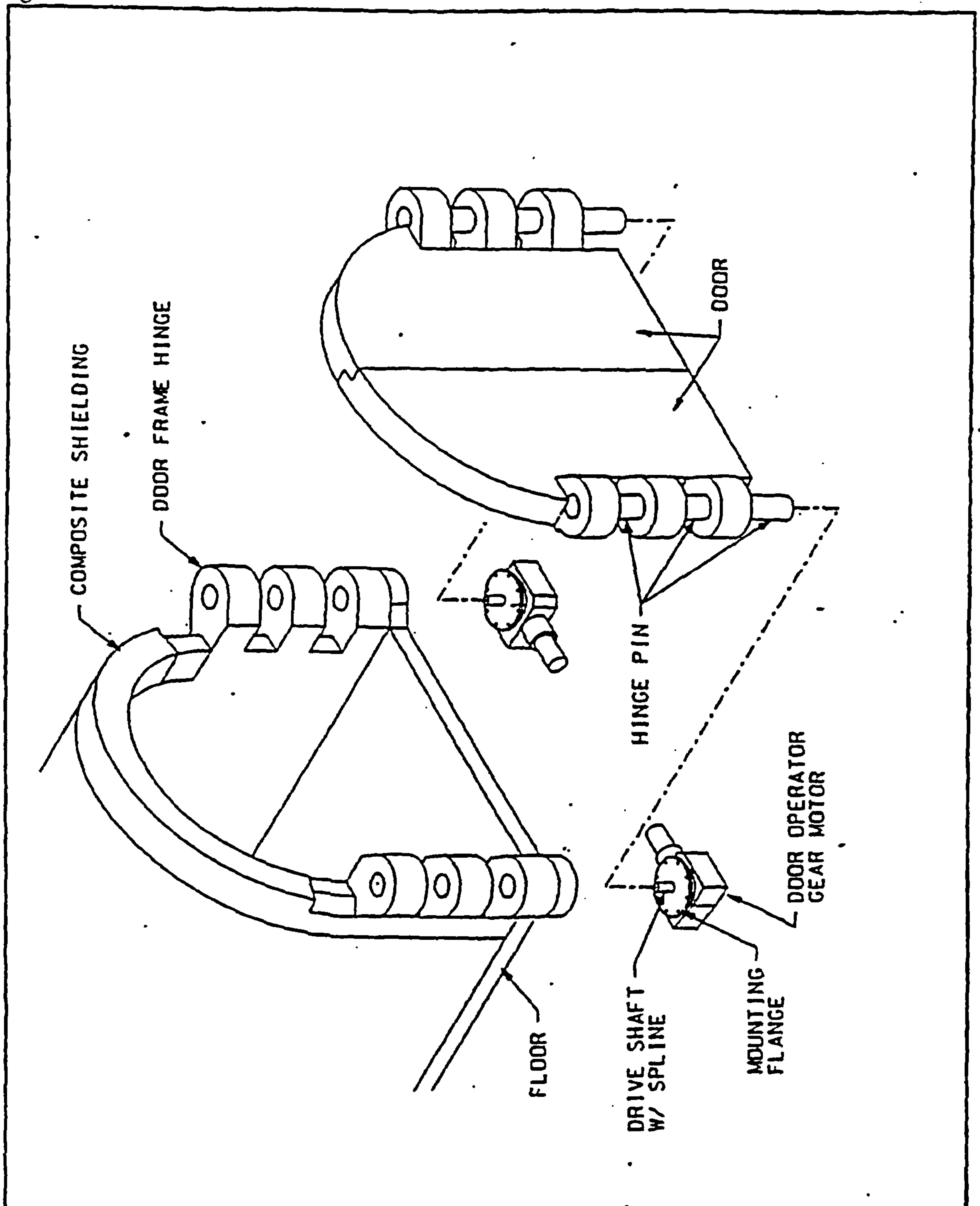

FIGURE 1
WASTE PACKAGE TRANSPORTER
DOOR OPERATOR

Figure 2

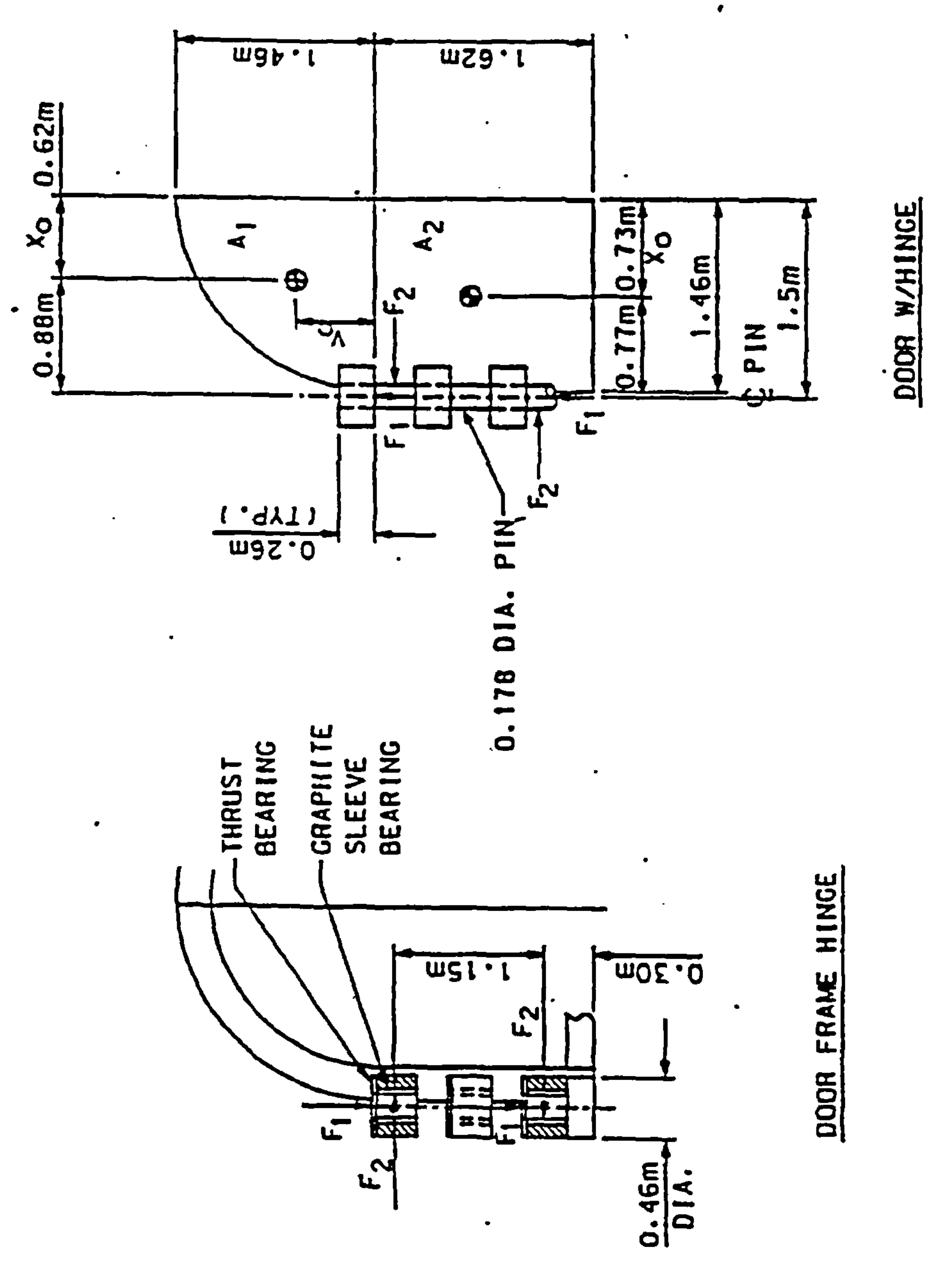

FIGURE 2
WP TRANSPORTER
DOOR AND HINGE

$$WT = (2.36)\ (0.209\ m^2 - 0.032\ m^2)\ (0.260\ m)\ (7949.7\ kg/m^3)$$
$$= 863.4\ kg$$
round down to 863 kg

Weight for door hinge pin:

$$WT = \frac{\pi D^2}{4}\ (1.618 + 0.260)\ (7949.7\ kg/m^3)$$

$$= (0.79)\ (0.178\ m)^2\ (1.88\ m)\ (7949.7\ kg/m^3)$$
$$WT = 374\ kg$$

Total weight for door:

A_1	3641 kg
A_2	5133
Hinges	863
Pin	374
Total	10,011 kg

3.2.2 Axial Load on Hinge

Due to tolerances in fit between the door and the door frame hinges, assume that the axial load from the door is supported at the upper and lower hinge only.

3.2.3 Load on Thrust Barriers

Load on each of the two bearings is:

$$F_1 = ½\ (\text{axial load})$$
$$= ½\ (10{,}011)\ kg = 5006\ kg\ (11{,}036\ lbs)$$

3.2.4 Moment Forces on Door Frame Hinge

The upper and lower door frame hinge sleeve bearing will support the equal and opposite forces from the door hinge pin.

Determine CG of each of the two overhanging areas (A_1 and A_2) from ℄ pin for Area A_1:.

From Ref. 5.29, pg. 3-18:

The center of gravity for a quadrant of a circle:

$$X_o = Y_o = 0.4244\ r$$

In this case, r = 1.46 m

$$X_o = (0.4244)\,(1.46\text{ m}) = 0.62\text{ m}$$

CG of A_1 from ℄ pin = 1.500 m - 0.62 m
= 0.88 m

For Area A_2:

A_2 is a rectangle and $X_o = \frac{1.460\ m}{2} = 0.730$ m

CG of A_2 from ℄ pin = 1.500 m - 0.730 m
= 0.77 m

$\Sigma M = 0$ at ℄ of pin

$$\begin{aligned} F_2\,(1.15\text{ m}) &= (0.88\text{ m})\,(WT_{A_1}) + (0.77\text{ m})\,(WT_{A_2}) \\ &= (0.88\text{ m})\,(3641\text{ kg}) + (0.77\text{ m})\,(5133\text{ kg}) \\ &= 3204\text{ m-kg} + 3952.4\text{ m-kg} \\ &= 7156\text{ m-kg} \\ F_2 &= 6223\text{ kg, round up to } 6225\text{ kg} \end{aligned}$$

Load of sleeve bearings is 6225 kg (13,724 lbs)

3.2.5 Door Hinge Friction

From Ref. 5.29, pg. 3-48 to 3-49:

Torque from single journal bearing:

$$M_J = f\,P\,r \text{ (in-lbs)}$$

where: P = total load of journal (lbs)
r = radius of journal (in.)
f = frictional coefficient

Torque for single thrust bearing:

$$M_T = \tfrac{1}{3} f\,L\,(D^3-d^3)/(D^2-d^2) \text{ (in-lbs)}$$

where: L = total load of bearing (lbs)
D = outside diameter of bearing (in.)
d = inside diameter of bearing (in.)
f = frictional coefficient

From Ref. 5.29, pg. 3-40, Table 1:

For hard steel on lubricated graphite
For static condition: f = 0.09

Converting to English unit for use in formula

$$r = \frac{178\ mm}{2} \times \frac{in.}{25.4\ mm} = 3.5\ \text{in.}$$

$$D = 457\ \text{mm} \times \frac{in.}{25.4\ mm} = 18\ \text{in.}$$

$$d = 178\ \text{mm} \times \frac{in.}{25.4\ mm} = 7.0\ \text{in.}$$

$M_J = f\,p\,r = (0.09)\ (13{,}724\ \text{lb})\ (3.5\ \text{in.})$

$M_J = 4323.1$ in-lb per bearing

$$M_T = \tfrac{1}{3} f L \frac{(D^3 - d^3)}{(D^2 - d^2)} = \tfrac{1}{2} (0.09)\ (11{,}036\ \text{lb})\ \frac{(18^3 - 7^3)\ in^3}{(18^2 - 7^2)\ in^2}$$

$$= \tfrac{1}{3} (0.09)\ (11{,}036\ \text{lb})\ \frac{(5832 - 343)\ in^3}{(324 - 49)\ in^2}$$

$$= (331.1)\ \frac{5489}{275}\ in\text{-}lb$$

$M_T = 6609$ in-lbs per bearing

Total torque = $2M_J + 2M_T$
= (2) (4323.1) in-lb + 2(6609) in-lbs
= 8646.2 in-lb + 13,218 in-lb
Total torque = 21,864.2 in-lbs, round up to 21,870 in-lbs

From Design Selection No. 5:

The 270 degree door swing shall take approximately 1 minute.

A 1-rpm gearmotor speed will satisfy that requirement.

Gearmotor selection criteria:

1. 21,870 in-lbs of torque
2. 1 rpm output shaft speed
3. Flanged mount for mounting to Transporter floor below door pin
4. Spline output shaft for a splined connection to the door pin for ease in installation and removal.

5. Right-angle gear drive for minimum obstruction of clearance envelop around rear of Transporter.
6. Service factor for the application = 1.5 (see following for selection rationale)

From Attachment XII, Section 6, pg. 7:

For right-angle helical-worm drives, select a reducer loading of extreme shock load (loads do not exceed 1.75 of normal input) due to the torque of starting the large inertia of the door from rest.

The associated safety factor (SF) for that condition which will occur less than 3 hours per day (door opening and closing twice a day) is 1.5.

From Attachment XII, Section 6, pg. 346:

A S92R62 helical wormgear motor is selected with output speed = 1 rpm. Output torque = 23,200 and a SF of 1.6. Both torque and SF exceed the selection criteria selection for door gearmotor.

Door operator gearmotor selection

SEW Eurodrive helical-wormgear motor S92R62 w/0.75 hp DT80K4 motor flange mounted with keyed or splined output shaft.

From Attachment XII, Section 6, pgs, 417, 422, 423:

Door operator gearmotor weight is as follows:

S92R62 w/DT80K4 motor	= 407 lbs
Lubricant for V1L mounting position 3.2 gals at 7.5 lb/gal	= 24 lbs
Total weight	431 lbs
Round up to	440 lbs
	or 200 kg

3.3 Rail Car Unloader Equipment Selection

3.3.1 General

Rail Car unloader utilizes two rigid chains and the associated drives to push the Rail Car loaded with the waste package from the inside of the Transporter into the emplacement drift (see Figure 3).

Figure 3

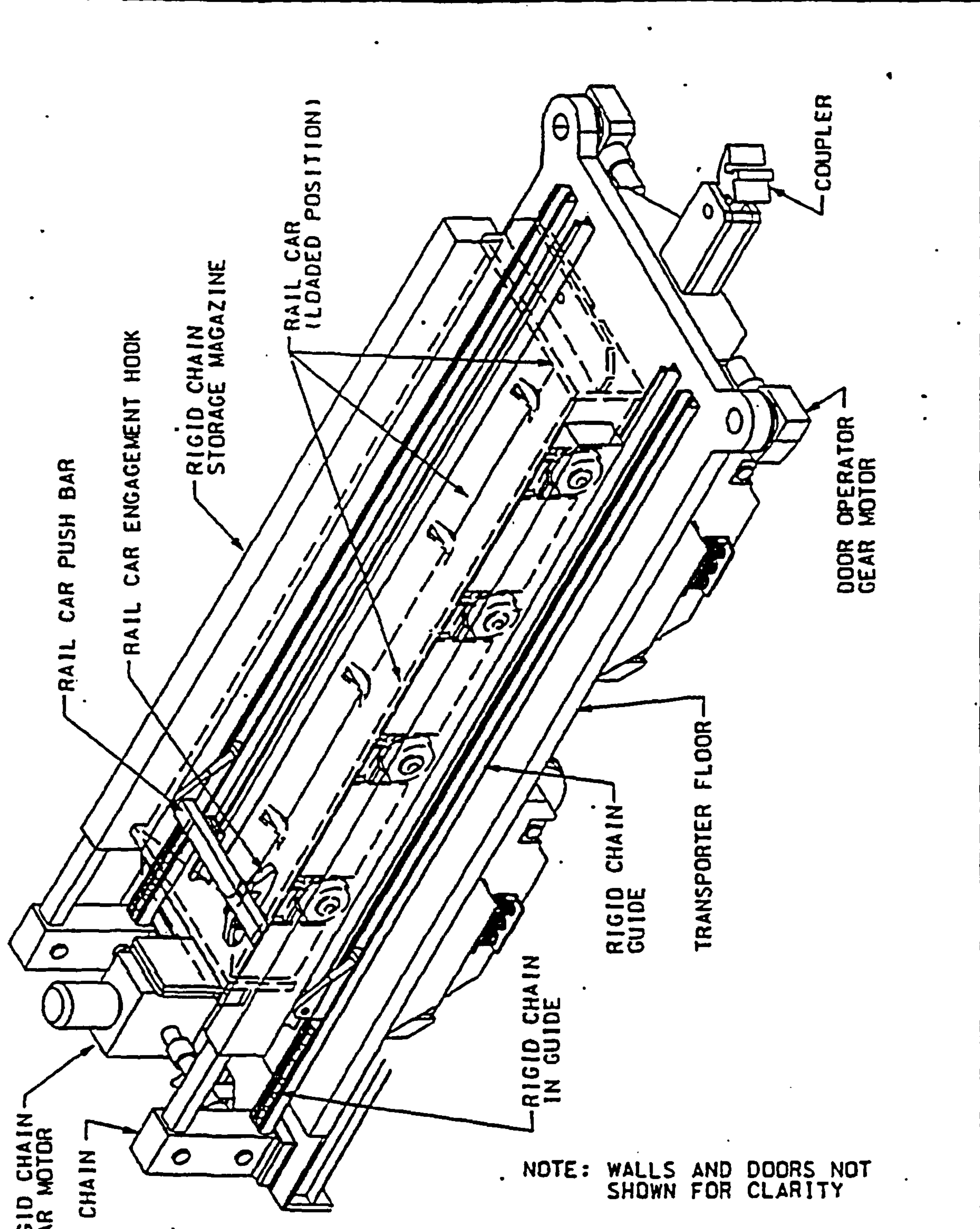

FIGURE 3
WASTE PACKAGE TRANSPORTER RAIL CAR UNLOADER SYSTEM

3.3.2 Rail Car Rolling Resistance

From Attachment XII, Section 4068, pg. 1:

Rolling resistance from Rail Cars with ball or roller bearings is 10.02 kg/MT) (20 lb/ton).

From Attachment IV, Section 3.2:

Maximum Rail Car operating weight = 80 MT

Roll Resistance (Loaded) = $20\ \text{lb/ton} \times 80{,}000\ \text{kg} \times \frac{2.205\ lb}{kg} \times \frac{1\ ton}{2000\ lb} = 1764\ \text{lb}$

Roll Resistance (Unloaded) = $20\ \text{lb/ton} \times 11{,}000\ \text{kg} \times \frac{2.205\ lb}{kg} \times \frac{1\ ton}{2000\ lb} = 243\ \text{lbs}$

3.3.3 Incline Force

From Section 4.3.8:

Maximum grade in emplacement drift = 0.75%

Drift angle = arc tan 0.0075 = 0.43°

Incline Force (Loaded) = $80{,}000\ \text{kg} \times \frac{2.205\ lb}{kg} \times \sin 0.43^{\circ} = 1324\ \text{lb}$

Incline Force (Unloaded) = $11{,}000\ \text{kg} \times \frac{2.205\ lb}{kg} \times \sin 0.43^{\circ} = 182\ \text{lb}$

3.3.4 Push Bar Force for Placing Loaded Rail Car

Initial push bar force against Rail Car will be gradual due to the use of a variable-speed unloader drive and acceleration forces are negligible.

Push bar force = roll resistance + incline force
= 1764 lbs + 1324 lbs
= 3095 lbs

3.3.5 Push Bar Force Summary

Rail Car Condition	Rolling Resistance Loaded (lb)	Rolling Resistance Empty (lb)	Incline Force Loaded (lb)	Incline Force Empty (lb)	Push Bar Chain Force Total (lb)	Push Bar Chain Force EA (lb)
Emplace loaded	1764	N/A	1324	N/A	+3088	+1544
Retrieve loaded	1764	N/A	1324	N/A	-440	-220
Retrieve empty	N/A	243	N/A	182	+61	+31

3.3.6 Rigid Chain Selection

Maximum rigid chain compression force = 1544 lb, round up to 1600 lb

Changing to metric units for catalog selection:

$$\text{Maximum rigid chain compression force} = 1600 \text{ lb} \times \frac{kg}{2.205\ lb} = 726 \text{ kg (force)}$$

From Attachment XII, Section 5, pg. 5 and 6:

Single row, 60-mm pitch, 60 SG chain capacity is 2500 daN or kg (force)
Single row, 90-mm pitch, 90 SG chain capacity is 9000 daN or kg (force)

Both the 60 SG and the 90 SG chain capacities exceed the load requirements. In this analysis, the largest chain will be selected and used in the unloader design and drawings as the worst case for space and power requirements.

Chain selection is then Serapid single chain 90-mm pitch, 90 SG with rollers and maximum horizontal pushing force of 9000 daN used with a 90 SG guide.

From Attachment XII, Section 5, pg. 6, 8, and 9:

90 SG chain weight is 32 kg/meter
90 SG guide weight is 43 kg/meter
90-mm pitch, 90° drive housing weight is 120 kg

For a 180° drive housing, use 2 EA 90° drives for a total weight of 240 kg.

3.3.7 Chain Drive Power Requirement

From Attachment X, pg. X-8:

$$\text{Torque} = \frac{\textit{force x chain pitch}}{\textit{eff}}$$

Maximum chain force = 726 kg (1600 lbs)

Chain pitch = 90 mm (3.54 in.)

Eff = 0.8 (which includes friction of chain in glide)

$$\text{Torque} = \frac{(1600\ lbs)\ (3.54\ in.)}{0.8} = 7080 \text{ in-lbs}$$

$$\text{Horsepower} = \frac{\textit{torque (in-lb) x shaft speed (rpm)}}{63000 \times eff}$$

From Design Selection No. 7:

Rail Car unloading rate is 7.6 m/min. (25 ft/min.)

From Attachment XII, Section 6, pg. 9:

For 90-mm pitch drive, 1 rev is 540 mm (21.3 in.) of chain.

$$\text{Drive speed} = 25 \text{ ft/min.} \times \frac{rev}{21.3\ in.} \times \frac{12\ in.}{ft} = 14 \text{ rpm}$$

Horsepower is now:

$$HP = \frac{(7080\ in\text{-}lb)\ (14\ rpm)}{63{,}000 \times 0.8}$$

Note: eff = 0.8 is within the range of combined motor/reducer efficiencies

HP = 1.96 per chain

Total drive hp for both chains:

HP_T = 3.9 hp, round up to 5 hp

3.3.8 Motor-Reducer Selection

Selection criteria:

1. 1750 rpm input
2. 14 rpm output

3. Foot-mounted motor reducer with two output shafts, located between the two rigid chain drives
4. 5 hp minimum

From Attachment XII, Section 6, pg. 7:

For right-angle helical gear motors, select:

AGMA Class II b

1. moderate shock loads not exceeding 1.25 x rated load torque
2. minimum service factory = 1.4

From Attachment XII, Section 6, pg. 223:

For a 5-hp helical-bevel gearmotor, select a Model K106 w/DT100L4 motor

13 rpm output, 2.2 service factor
24,200 in-lb of torque

Rigid chain drive gearmotor selection:

SEW Eurodrive helical-bevel gearmotor Model K106 with DT100L4 motor, reversible 5 hp, 1750 rpm input, 13 rpm output, 24,200 in-lb of torque, foot mounted.

Refer to Attachment XII, Section 6, pg. 246 for gearmotor dimensions.

3.3.9 Gearmotor Weight

From Attachment XII, Section 6:

Page 324: Mounting position B611
Page 334: 8.5 gallons of lube
Page 332: K106 w/DT100 weight is 624 lbs

Item	Wt
Gear motor	624 lb
Lube (8.5 gal x 7.5 lb/gal)	64 lb
Total =	688 lb (312 kg)
	round up to 320 kg

3.3.10 Rigid Chain Magazine Storage Requirements

From Design Selection No. 6:

Transporter Rail Car unloader shall have a minimum travel and storage capacity for 12.0 m of rigid chain.

Rigid chain storage magazine arrangement is shown on Figure 4.

From Figure 4, total length of retrievable chain stored in magazine:

$$2\ell + \frac{\pi\ 0.2}{2} = 12.0\ \text{m}$$

$$2\ell = 12.0\ \text{m} - 0.314\ \text{m} = 11.69\ \text{m}$$
$$\ell = 5.84\ \text{m}$$

Total length of magazine:

$$L = \ell + 0.050\ \text{m} + 0.300\ \text{m}$$
$$= 5.84\ \text{m} + 0.350\ \text{m} = 6.19\ \text{m}$$

3.3.11 Rigid Chain Magazine Weight

From Figure 4:

Magazine is 6.19 m long x 0.400 m high x 0.230 m wide

Magazine enclosure is constructed of 3.18 m (0.125 in.) carbon steel plate.

Magazine surface area:

$$A = 2\ (0.400\ \text{m} + 0.230\ \text{m})\ (6.19\ \text{m})$$
$$= 7.8\ \text{m}^2.$$

From Ref. 4.4.1, pg. 6-8:

Density of cold-drawn steel = 490 lb/ft³

Converting to metric units:

$$D = 490\ \text{lb/ft}^3 \times \frac{kg}{2.205\ lb} \times \left(\frac{3.28\ ft}{m}\right)^3$$

$$= 7842\ \text{kg/m}^3$$

Figure 4

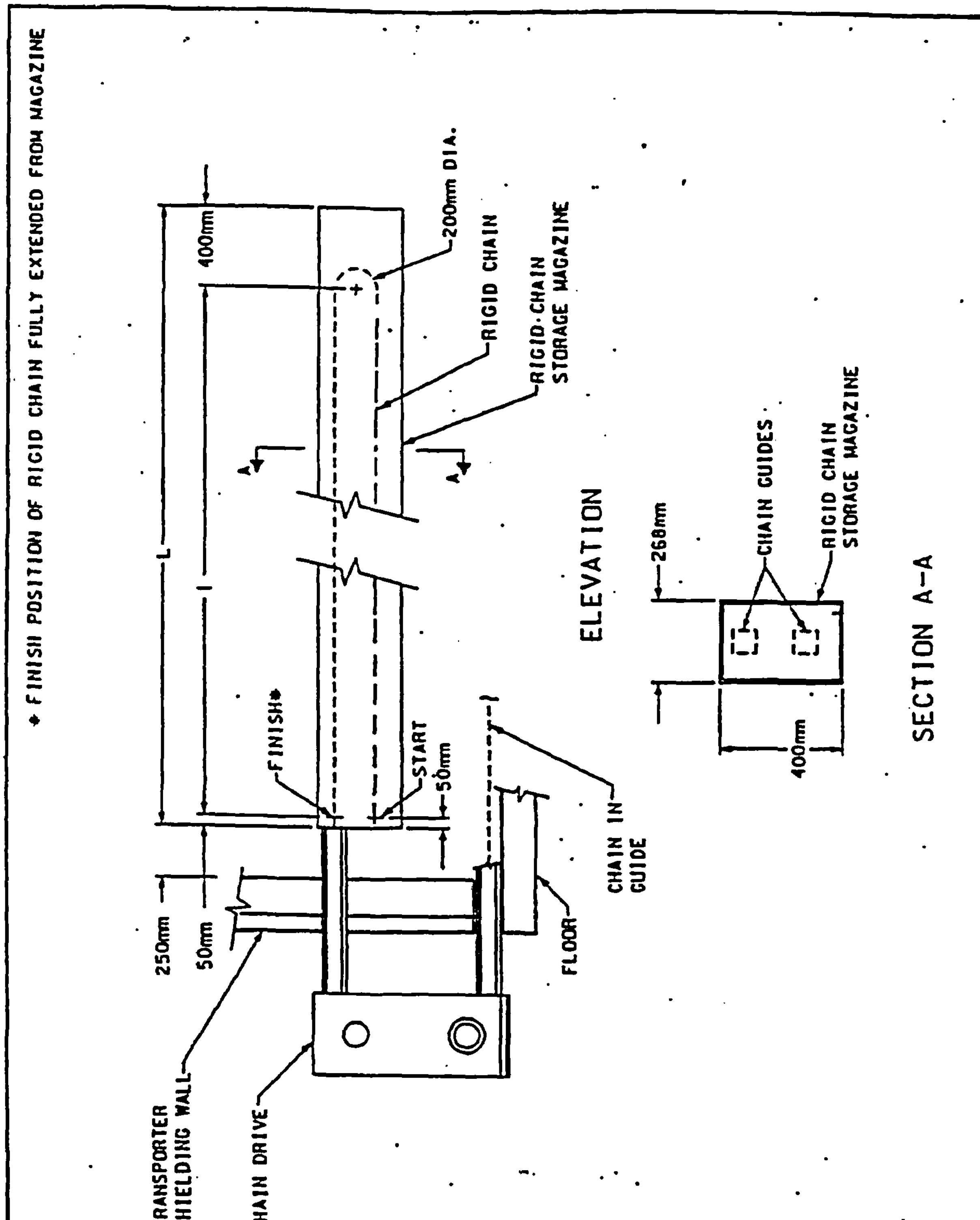

FIGURE 4
RIGID CHAIN MAGAZINE ARRANGEMENT

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For the thickness of 3.18 mm, the density per m^2 is:

$$\text{Density}/m^2 = 7842\ kg/m^3 \times 0.00318\ m = 24.9\ kg/m^2$$

$$\text{Empty magazine weight} = (7.8\ m^2)(24.9\ kg/m^2) = 194\ kg$$

Rigid chain weight:

From Attachment XII, Section 5 spec sheet page 6:

Type 90 SG rigid chain weight is 32 kg/m

Rigid chain weight = (12 m) (32 kg/m) = 384 kg

Total magazine weight for each:

Rigid chain	384 kg
Magazine	194
Total =	578 kg
Misc. support (10% of total)	57
Total =	635 kg
round down to	630 kg

ATTACHMENT IV

REUSABLE RAIL CAR - MECHANICAL EQUIPMENT SELECTION

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

REUSABLE RAIL CAR - MECHANICAL EQUIPMENT SELECTION

1.0 PURPOSE

Based on Rail Car dimensions shown in figures 1 and 2, this calculation provides the following:

1. Rail car weight and operating weight with heaviest WP for use as input in this and other calculations.

2. Rail car wheel and axle selection.

3. Verify the capacity of the 44.6 kg/m (90 lb/yd) ASCE rail for the rail car application.

2.0 INITIAL DESIGN SELECTION

No. Input Description

1. From a preceding analysis (Ref. 5.5), a distance of 1.32 m from centerline of largest diameter WP (2.0 m) mounted on the rail car to the top of the rail will be used. (Attachment III)

2. The rail car will have 4 axles and 8 wheels, with the maximum combined load of the rail car and the heaviest waste package evenly distributed over the wheels. This condition is considered achievable with the proper placement of the wheel/axles on the rail car. (Attachment IV)

3. The rail car wheels will be 356 mm (14 in.) diameter based on manufacturer's standard size for industrial rail cars in the load range of this application. (Attachment IV)

4. Brinell hardness of the gantry transporter, rail car and gantry carrier wheels shall be 320 BHN (Ref. 4.4.5, Table 4.13.3-4). The 320 BHN hardness is a standard hardness of crane wheels used for cranes with a higher range of load conditions. (Attachments V and IV)

3.0 SOLUTIONS

3.1 Rail Car Empty Weight

Determine rail car empty weight based on:

A. Rail car dimensions in Figures 1 and 2.

B. Rail car fabricated from A36 steel with a density of 490 lb/ft^3 (Ref. 4.4.1, pg. 6-8).

C. Standard wheel/axle components as listed in Ref. 5.33, pg. 6.

For the weight determination, the rail car is separated into the assemblies and individual components as shown in Figure 3. The weights for each are determined as follows.

3.1.1 Wheel/Axle Support Assembly A (Figure 4)

Determine volume of individual components (refer to Figure 5).

1. Plate : 25 mm (1") thick

$$\text{Area} = (45\text{ mm})(2)(735\text{ mm}) + 2\,(½)(423\text{ mm})(735\text{ mm})$$
$$= 66{,}150\text{ mm}^2 + 310{,}905\text{ mm}^2$$
$$= 377{,}055\text{ mm}^2$$

$$\text{Volume} = (0.377\text{ m}^2)(0.025\text{ m}) = 9.4 \times 10^{-3}\text{ m}^3$$

2. Plate: 38 mm (1½") thick

$$\text{Area} = \left(405\text{ mm} - \frac{65\text{ mm}}{2}\right)(710\text{ mm}) = 264{,}475\text{ mm}^2 = 0.264\text{ m}^2$$

$$\text{Volume} = (0.264\text{ m}^2)(0.038\text{ m}) = 10.0 \times 10^{-3}\text{ m}^3$$

3. Plate: 25 mm (1") thick

$$\text{Area} = \left(163\text{ mm} - \frac{15\text{ mm}}{2}\right) \times 710\text{ mm} = 110{,}405\text{ mm}^2 = 0.11\text{ m}^2$$

$$\text{Volume} = (0.11\text{ m}^2)(0.025\text{ m}) = 2.75 \times 10^{-3}\text{ m}^3$$

4. Plate: 25 mm (1") thick

$$\text{Area} = (163\text{ mm})(58\text{ mm}) + ½(58\text{ mm TAN }30°)(58\text{ mm})$$

$$\text{Area} = (163\text{ mm})(58\text{ mm}) + ½(33\text{ mm})(58\text{ mm})$$
$$= 9454\text{ mm}^2 + 957\text{ mm}^2$$
$$= 10{,}411\text{ mm}^2$$
$$= 0.0104\text{ m}^2$$

Volume = (0.0104 m^2)(0.025 m) = 0.26 x 10^{-3} m^3

5. Plate: 25 mm (1") thick

Area = (½)(148 mm)(257 mm)
= 19,018 mm^2
= 0.0190 m^2

Volume = (0.0190 m^2)(0.025 m) = 0.48 x 10^{-3} m^3

6. Wheel and Axle Assembly (2 wheels, axle, 2 axle boxes)

From Ref. 5.33, Pg 6:
Irwin Car and Equipment
Model No. TA-104 thru axle wheel assembly
Capacity: 15 ton
Approximate weight: 695 lbs (316 kg)

Determine density for structural plate steel in metric units:
From Ref. 4.4.1, Pg. 6-8:
Density of rolled steel = 490 lb/ft³

Converting to metric: $490\ \frac{lb}{ft^3} \times \frac{kg}{2.205} \times \left(\frac{3.281\ ft}{m}\right)^3 = 7849\ kg/m^3$

Assembly A Volume and Weight Summary

No.	Des.	Qty.	Vol. (m^3 x 10^{-3})	Density (kg/m^3)	Wt. EA. (kg)	Total (kg)
1.	P	2	9.4	7849	73.8	147.6
2.	P	2	10.0	7849	78.5	157.0
3.	P	4	2.75	7849	21.6	86.4
4.	P	2	0.26	7849	2.04	4.1
5.	P	2	0.48	7849	3.77	7.5
6.	Assem.	1	--	--	316.0	316.0

Total Weight Assembly A = 719 kg
Round up to 720 kg

3.1.2 Assembly B (Figure 6)

Determine volume (refer to Figure 7):

1. Plate (25 mm thick)

$$\text{Area} = (952\text{ mm})\left(508\text{ mm} - \frac{16\text{ mm}}{2}\right) = 476{,}000\text{ mm}^2 = 0.476\text{ m}^2$$

$$\text{Volume} = (0.476\text{ m}^2)(0.025\text{ m}) = 11.9 \times 10^{-3}\text{ m}^3$$

2. Plate (25 mm thick)

$$\begin{aligned}\text{Area} &= (45\text{ mm})(2)(708\text{ mm}) + 2\,(½)(407\text{ mm})(708\text{ mm})\\ &= 63{,}720\text{ mm}^2 + 288{,}156\text{ mm}^2 = 351{,}876\text{ mm}^2\\ &= 0.352\text{ m}^2\end{aligned}$$

$$\text{Volume} = (0.352\text{ m}^2)(0.025\text{ m}) = 8.8 \times 10^{-3}\text{ m}^3$$

3. Plate (25 mm thick)

$$\text{Area} = (1470\text{ mm})(952\text{ mm}) = 1{,}399{,}440\text{ mm}^2 = 1.4\text{ m}^2$$

$$\text{Volume} = (1.4\text{ m}^2)(0.025\text{ m}) = 35.0 \times 10^{-3}\text{ m}^3$$

Assembly B Volume and Weight Summary

No.	Des.	Qty.	Vol. ($m^3 \times 10^{-3}$)	Density (kg/m^3)	Wt. EA. (kg)	Total (kg)
1.	Plate	2	11.9	7849	93.4	186.8
2.	Plate	1	8.8	7849	69.1	69.1
3.	Plate	1	35.0	7849	274.7	274.7

Total Assembly B = 530.6 kg
Round down to 530 kg

3.1.3 Assembly C (Figure 6)

Determine volume (refer to Figure 8):

1. Plate (25 mm thick)

$$\text{Area} = (555\text{ mm})\left(508\text{ mm} - \frac{16\text{ mm}}{2}\right) = 277{,}500\text{ mm}^2 = 0.28\text{ m}^2$$

$$\text{Volume} = (0.28\text{ m}^2)(0.025\text{ m}) = 7.0 \times 10^{-3}\text{ m}^3$$

2. Plate (25 mm thick)

$$\text{Area} = (1470\text{ mm})(555\text{ mm}) - (228\text{ mm})(152\text{ mm})$$

$= 815{,}850 \text{ mm}^2 - 34{,}656 \text{ mm}^2$
$= 781{,}194 \text{ mm}^2 = 0.78 \text{ m}^2$

Volume $= (0.78 \text{ m}^2)(0.025 \text{ m}) = 19.5 \times 10^{-3} \text{ m}^3$

Assembly C Volume and Weight Summary

No.	Des.	Qty.	Vol. ($m^3 \times 10^{-3}$)	Density (kg/m^3)	Wt. EA. (kg)	Total (kg)
1.	Plate	2	7.0	7849	54.9	109.8
2.	Plate	1	19.5	7849	153.1	153.1

Total Assembly C = 262.9 kg
Round up to 265 kg

3.1.4 Cradle Plate D (38 mm thick) (Figure 9)

Determine volume and weight:

Note: Cutouts for axles are not considered for a more conservative weight estimate.

Area $= (2)(0.88 \text{ m})(5.998 \text{ m}) = 10.56 \text{ m}^2$

Volume $= (10.56 \text{ m}^2)(0.038 \text{ m}) = 0.40 \text{ m}^3$

Weight $= (0.40 \text{ m}^3)(7849 \text{ kg/m}^3) = 3139.6 \text{ kg}$

Total Weight Plate D = 3139.6 kg
Round up to 3,140 kg

3.1.5 End Plate E (76 mm thick) (Figure 9)

Note: Cutout in end plate is not considered for a more conservative weight estimate.

Determine volume and weight:

Area $= (1.624 \text{ m}) \times (0.590 \text{ m}) = 0.96 \text{ m}^2$

Volume $= (0.96 \text{ m}^2)(0.076 \text{ m}) = 73.0 \times 10^{-3} \text{ m}^3$

Weight $= (73 \times 10^{-3} \text{ m}^3)(7849 \text{ kg/m}^3) = 573.0 \text{ kg}$

Total Weight End Plate E = 573.0 kg, round up to 575 kg

3.1.6 Towing Eye Plate F (50 mm thick) (Figure 9)

Determine volume and weight:

$$\text{Area} = (1624\text{ mm})(304\text{ mm}) - (152\text{ mm})(152\text{ mm}) - \frac{\pi(76\text{ mm})^2}{4}$$
$$= 493{,}696\text{ mm}^2 - 23{,}104\text{ mm}^2 - 4537\text{ mm}^2$$
$$= 466{,}055\text{ mm}^2 = 0.466\text{ m}^2$$

$\text{Volume} = (0.466\text{ m}^2)(0.050\text{ m}) = 23.3 \times 10^{-3}\text{ m}^3$

$\text{Weight} = (23.3 \times 10^{-3}\text{ m}^3)(7849\text{ kg/m}^3) = 182.9\text{ kg}$

Total Weight Plate F = 182.9 kg
Round up to 185 kg

3.1.7 Rail Car Empty Weight Summary

Assembly A Volume and Weight Summary

No.	Des.	Qty.	Wt. EA. (kg)	Total (kg)
1.	Assembly A	4	720	2,880
2.	Assembly B	3	530	1,590
3.	Assembly C	2	265	530
4.	Plate D Cradle	1	3,140	3,140
5.	End Plate E	2	575	1,150
6.	Towing Eye Plate F	2	185	370

Total = 9,660 kg
adding 10% for miscellaneous steel = 966 kg
Total Rail Car Empty Weight = 10,626 kg

Round up to 11,000 kg

3.2 Maximum Rail Car Operating Weight

From Section 4.3.10:

Maximum WP weight = 69.0 MT

Empty rail car weight		11.0 MT
WP		69.0 MT
	Total	80.0 MT

> Maximum Rail Car Operating Weight = 80 MT

3.3 Determine Maximum Wheel Load

Maximum operating weight = 80 MT

Converting to English system for selection from manufacturer's data

Maximum operating weight = 80,000 kg x 2.205 lb/kg
= 176,400 lbs

From Design Selection No. 2:
The Rail Car is designed with 4 axles and 8 wheels.
All 8 wheels are equally loaded

Weight per wheel = $\frac{176{,}400 \text{ lbs}}{8}$ = 22,050 lbs
= 11.03 tons

3.4 Wheel Selection

From Ref. 5.33, page 6:

A Model No. TA-104 thru axle wheel assembly has a rated capacity of 15 tons, well above the maximum wheel/axle load for a rail gauge of 1.44 m.

Selection:

> Irwin Car and Equipment, Model No. TA-104, thru axle assembly, 1.44 m (56½ in.) rail gauge, tapered roller bearings, 356 mm (14 in.) diameter wheels. 695 lbs approximate weight.

3.5 Rail Size Verification

From Section 4.3.25: The Rail Car rail is 44.6 kg/m (90 lb/yd) ASCE rail

From Ref. 4.4.5 (Section 4.13.3, page 47):

The tabulated allowable wheel loads for various diameters, hardness and rail application are determined from the product of:

$$D \times W \times K$$

where: D = diameter of wheel (in.)
W = effective width of rail (in.)

$K = 1300\left(\frac{BHN}{260}\right)^{0.33}$ for wheel hardness, BHN greater or equal to 260

From Ref. 4.4.5 (Table 4.13.3-4, pg. 50):

The effective width of rail is defined as the width of the rail head minus corner radii

From Design Selection No. 4, wheel hardness = 320 BHN

and: $K = 1300\left(\frac{320}{260}\right)^{0.33} = (1300)(1.23)^{0.33}$

$= (1300)(1.07) = 1391$

For the maximum capacity of the selected wheel, the minimum effective rail width is:

$$W = \frac{(15\ ton)\ (2000\ lb/ton)}{(K)\ (D)}$$

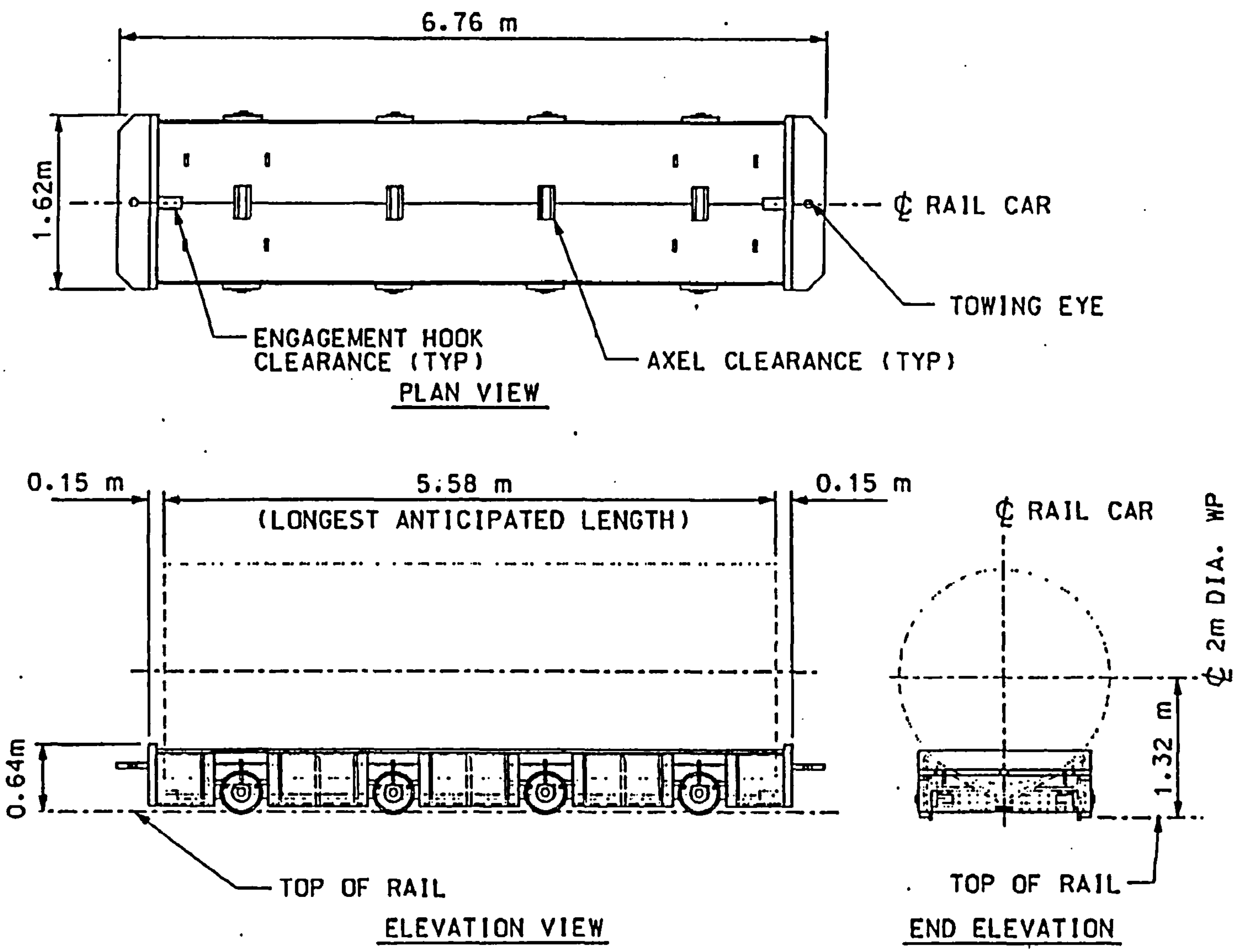
$$= \frac{30{,}000\ lb}{(1391)\ (14\ in.)} = 1.54\ \text{in.}$$

From Attachment XII, Section 1, page 19 for a 44.6 kg/m (90 lb/yd) ASCE rail:

Rail head width = 2⅝ in.
Corner radii = 5/16 in.

W = effective head width = 2⅝ in. - 2(5/16) in.
= 2 in.

The W of 2 in. for the 44.6 kg/m (90 lb/yd) ASCE rail is greater than the minimum required W of 1.54 in. and is adequate for this application.



NOTE: SEE SECTION 7.3 FOR ADDITIONAL INFORMATION ON THE RAIL CAR

FIG. 1
REUSABLE RAIL CAR ARRANGEMENT

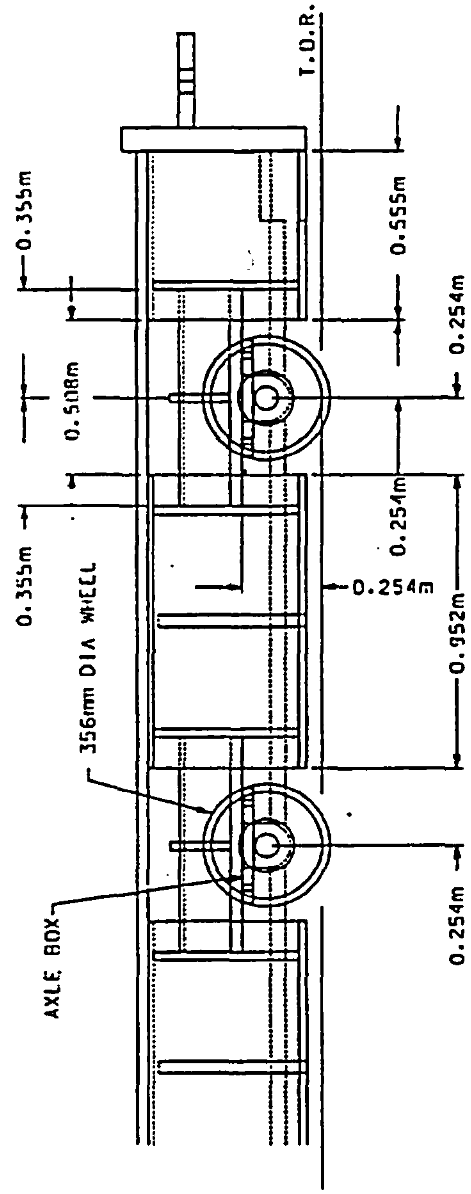


FIG. 2
REUSABLE RAIL CAR
WHEEL ASSEMBLY DETAIL

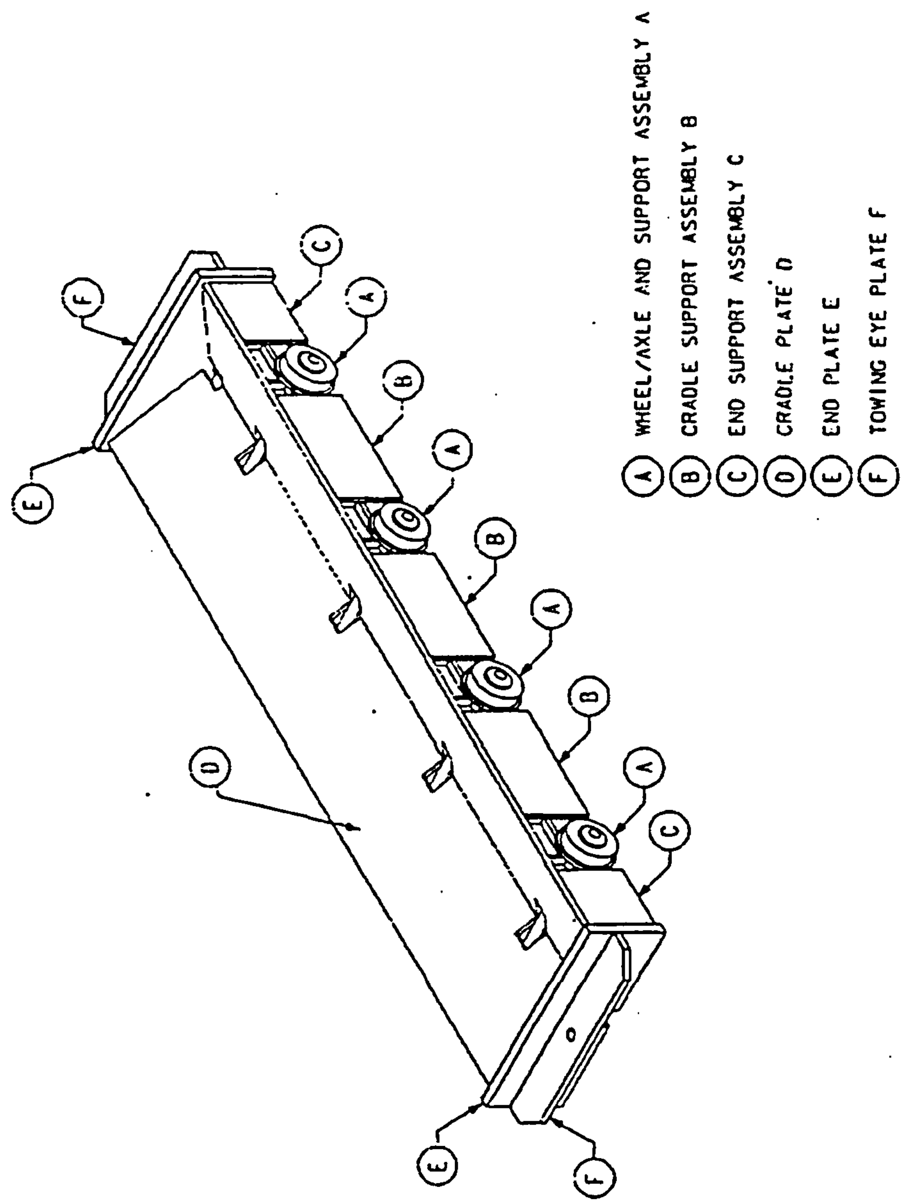


FIG. 3
REUSABLE RAIL CAR
ASSEMBLIES AND COMPONENTS

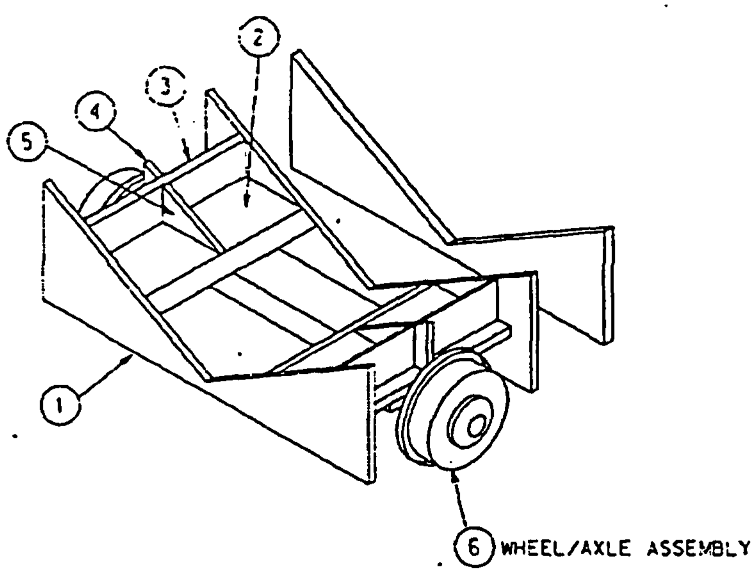
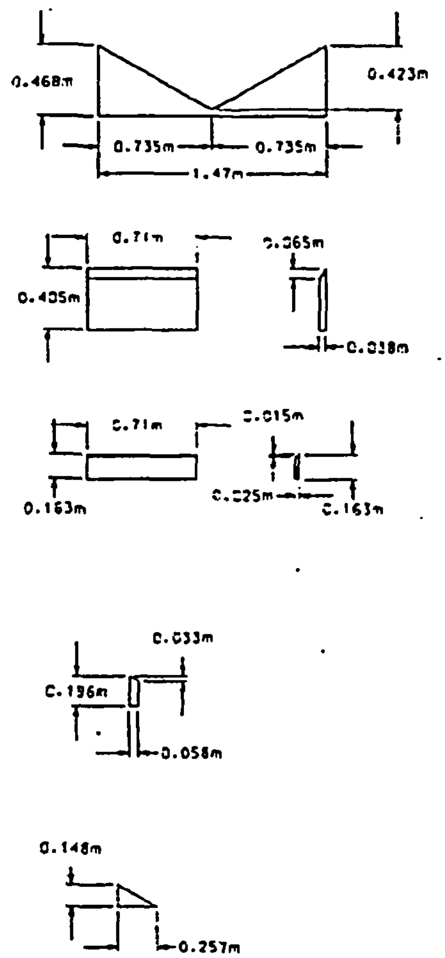


FIG. 4
WHEEL/AXLE SUPPORT
ASSEMBLY A



NOTE: UNLESS NOTED OTHERWISE ALL PLATE THICKNESS IS 25mm

FIG. 5
WHEEL/AXLE SUPPORT ASSEMBLY A

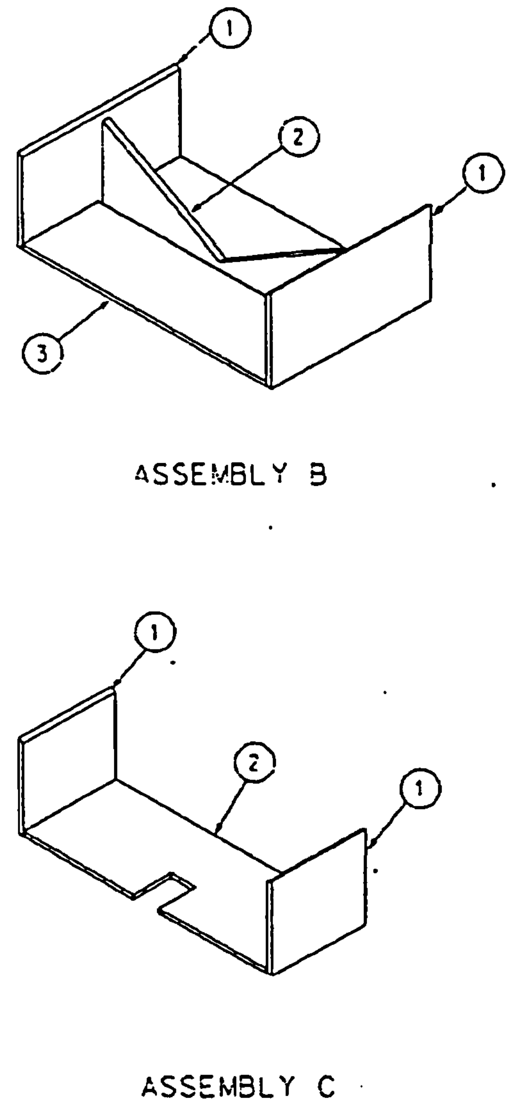
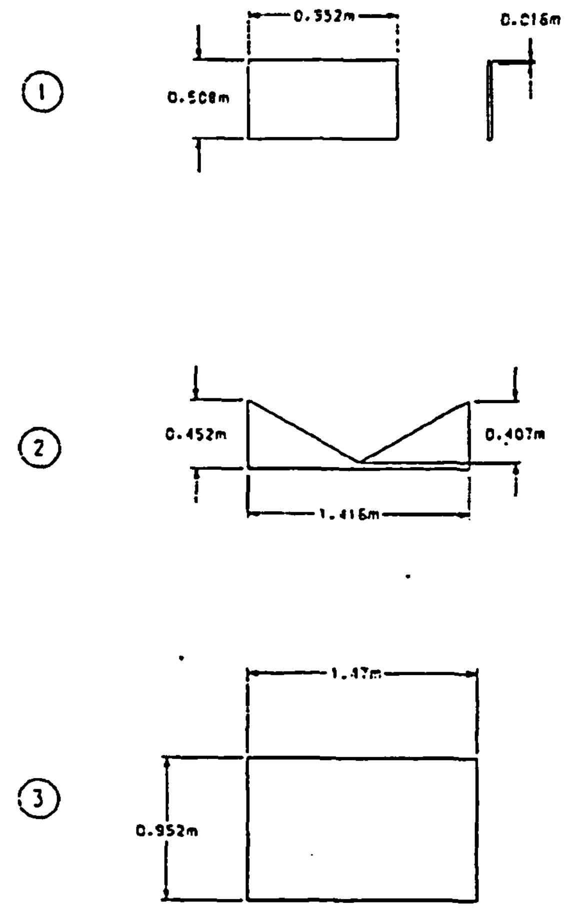
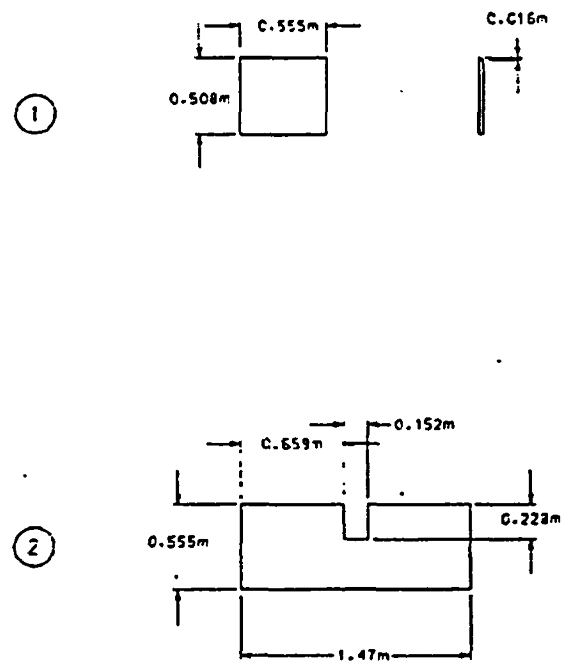


FIG. 6
CRADLE SUPPORT
ASSEMBLIES B&C



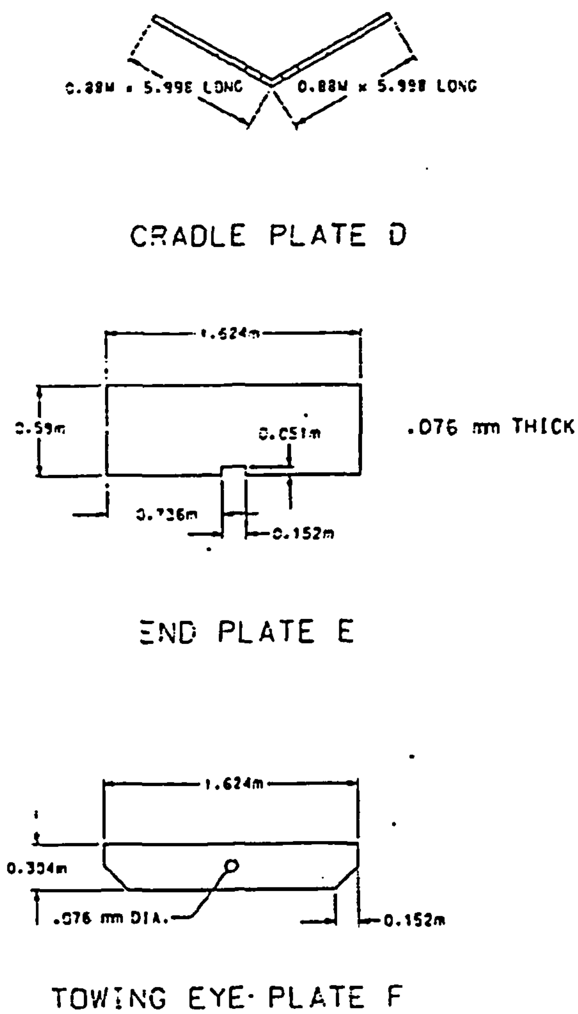
NOTE: UNLESS NOTED OTHERWISE ALL PLATE THICKNESS IS 25mm

FIG. 7
CRADLE SUPPORT
ASSEMBLY B



NOTE: UNLESS NOTED OTHERWISE ALL PLATE THICKNESS IS 25mm

FIG. 8
CRADLE SUPPORT
ASSEMBLY C



NOTE: UNLESS NOTED OTHERWISE ALL PLATE THICKNESS IS 25mm

FIG. 9
CRADLE AND ENDPLATE
D,E & F

ATTACHMENT V

GANTRY LOADS AND EQUIPMENT SELECTIONS

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TITLE: GANTRY LOADS AND MECHANICAL EQUIPMENT SELECTIONS

1.0 PURPOSE

This calculation will provide the basis for selecting and quantifying the following:

1. Gantry wheel size
2. Traversing drive
3. Lifting screw size
4. Lifting screw drive
5. Lifting head trolley screw size selection
6. Lifting head drive selection
7. Lifting head roller selection
8. Gantry racking forces

2.0 INITIAL DESIGN SELECTIONS

The following is a summary of the initial selections used in and verified by this calculation.

No.	Input Description
1.	Weight of the gantry structure shall be 45 MT. (Attachment II, Section 1.1)
2.	The gantry structure shall be supported by 4 bogies each having a minimum of 2 wheels with diameters and face widths sufficient to carry the maximum loads. One wheel of each bogie shall be driven. All 8 gantry wheels will be equally loaded due to the symmetry of the gantry arrangement.
3.	Brinell hardness of the gantry wheels shall be 320 BHN (Ref. 4.4.5, Table 4.13.3-4). The 320 BHN hardness is a standard hardness used for crane wheels with a higher range of load conditions.
4.	Gantry wheel diameter shall be 400 mm, which is a manufacturer's standard size for applications with similar loading (Ref. 5.38 Section 8.1, pg. 20).
5.	The maximum gantry speed shall be 0.76 m/sec (150 ft/min) (Ref. 4.4.2, Section 5333.1). The design tolerance for a design rated load speed shall be −10%.
6.	Wind loads are not applicable to Gantry operation since the Gantry is operated underground.

No.	Input Description
7.	Hoisting frame weight is 25% of total gantry weight. Lifting head trolley weight = 5000 lb (Estimated).
8.	Maximum waste package lift shall be 2226 mm.
9.	Gantry crane mechanical equipment shall be selected in accordance with Class E crane requirements (Ref. 4.4.5, Pg. 11).
10.	Lifting head trolley travel speed shall not exceed 0.07 m/sec (14 ft/min) (Ref. 4.4.5, Figure 6-2, Pg. 80). Lifting head trolley speed shall be slow for the fine control required at the point of engagement of the head and the skirt of the waste package. Figure 6.2 indicates a recommended slow hoist speed of 14 ft/min for the minimum load of 3 tons, which exceeds the weight of the lifting head trolley and suffices for this application.
11.	Lifting head trolley acceleration shall be 0.30 m/sec^2 (1 ft/sec^2) (Ref. 5.18, Pg. 111). A low rate of acceleration will permit better indexing characteristics for gripping WPs.
12.	The static coefficient of friction for locked Gantry wheel on rail of steel on steel will be 0.25 (Ref. 4.4.5, Table 5.2.9.1.2.1-B). The table value is 0.20, which is increased by a safety factor of 1.25, which is considered reasonable to accommodate off-normal conditions.
13.	Kinetic friction shall be 75% of static friction (Ref 5.27, Pg 251).
14.	Waste package lifting speed shall not exceed 6 ft/min (Ref. 4.4.2, Table NOG-5331.1-1).

3.0 SOLUTIONS

3.1 Traversing Wheel Selection

Based on procedure outlined in Ref. 4.4.5.

A. Allowable Wheel Load

From 4.3.10
Max. weight of waste package = 69 MT
Min. weight of waste package = 30.5 MT

From Design Selection No. 1
Weight of gantry structure = 45 MT

Total max. weight = WP weight + gantry weight
= 69 MT + 45 MT = 114 MT
= 114,000 kg

Total min. weight = 30.5 MT + 45 MT = 75.5 MT = 75,500 kg

From Design Selection No. 2, there are 8 wheels

Max. wheel load $= \frac{114{,}000\ kg}{8} = 14{,}250$ kg/wheel

Min. wheel load $= \frac{75{,}500\ kg}{8} = 9437.5$ kg/wheel

From Ref. 4.4.5, Pg. 47: The basic allowable wheel load (BWL) in lbs:

BWL = KDW (lbs)

where: K = hardness coefficient of the wheel when BHN ≥260

$$= 1300\left(\frac{BHN}{260}\right)^{0.33}$$

BHN = Brinell hardness
= 320 from Design Selection No. 3

D = wheel diameter
= 400 mm from Design Selection No. 4

W = effective rail width
= head width - 2 x corner radius
from Ref. 4.4.5, Pg. 50
Dimensions from Ref. 5.20, Pg. 19 for 90 lb ASCE rail

W = 2⅝ in. - 2 x 5/16 in. = 2 in.

$$K = 1300\left(\frac{320}{260}\right)^{0.33} = 1392$$

BWL = KDW = 1392 x 0.4 m x 3.28 ft/m x 12 in./ft x 2 in.
= 43,831 lb

BWL = 43,831 x 0.453 kg/lb = 19,856 kg

B. Durability Wheel Load

Mechanical mean effective load factor, Kw
from Ref. 4.4.5, Pg. 33.

$$Kw = \frac{2(maximum\ load) + (minimum\ load)}{3\ (maximum\ load)}$$

$$Kw = \frac{2\ (14250\ kg) + (9437.5\ kg)}{3\ (14250\ kg)} = 0.887$$

From Design Selections No. 4 and No.5:

$$\text{Wheel speed} = \frac{150\ ft/min}{3.28\ ft/m \times \pi \times 0.4\ m/rev} = 36.4\ \text{rpm}$$

From Ref. 4.4.5, Pg. 49
For speed factor C_S for rpm ≥ 31.5

$$Cs = 1 + \frac{RPM - 31.5}{328.5}$$

$$Cs = 1 + \frac{36.4 - 31.5}{328.5} = 1.015$$

From Ref. 4.4.5, Pg. 50
Wheel service factor, Sm = 1.25 (Class E)

From Ref. 4.4.5, Pg. 49
Wheel load service coefficient, K_{WL}

K_{WL} = Kw x Cs x Sm = 0.887 x 1.015 x 1.25 = 1.13
and:
Equivalent durability wheel load Pe

Pe = maximum wheel load x K_{WL}

= 14,250 kg x 1.13 = 16,103 kg

This wheel load is compared to the BWL load of 19,856 kg. ∴ The 400 mm diameter wheel is sufficient.

3.2 Gantry Traversing Drive Selection

A. Motor Horsepower

The motor horsepower including power components for acceleration and friction is based on procedures from Ref. 4.4.5, Pg. 59:

$HP_A = Ka \times W \times V \times Ks$

where:

Ka = acceleration factor for motor type

Ks = service factor which accounts for the type of drive and duty cycle

Ks = 1.2, Ref. 4.4 5 (Pg. 61 for an adjustable voltage with DC shunt motor for Class E cranes)

W = total weight = 114 MT (125.7 TON)

V = rated drive speed = 150 fpm

Since the equations in Ref. 4.4.5 are empirical, the following calculations are in English units.

$$Ka = \frac{f + \frac{2000\ a \times Cr}{g \times E}}{33{,}000 \times Kt} \times \frac{Nr}{Nf} \quad \text{(Pg. 59)}$$

where:

f = rolling friction
f = 15 lbs/ton (Pg. 61, 15 in. ϕ wheel)
a = average acceleration rate
= 1.0 ft/sec² for dry rails and 50% of wheels driven. Acceleration can be up to 2.4 ft/sec² (Pg. 60)
Cr = rotational inertia factor
= 1.05 + 1/7.5 = 1.18 (Pg. 59)
g = 32.2 ft/sec² (Pg. 59)
E = mech eff. = 0.9 (Pg. 59)
Nr = rated speed of motor at full load
= 1750 rpm
Nf = free wheeling speed = 1750 rpm
Kt = acceleration torque factor

= 1.5 (Pg. 60)

For adjustable voltage dc shunt wound motors

$$Ka = \frac{15 + \frac{2000 \times 1.0 \times 1.18}{32.2 \times 0.9}}{33{,}000 \times 1.5} \times 1 = 0.0019$$

Note: The worst case Nr/Nf = 1 was used.

Motor $HP_A = 0.0019 \times 125.7 \times 150 \times 1.2 = 43.0$
Since there are 4 motors:
Motor $HP_A = \frac{43.0}{4} = 10.7$ HP per motor

B. Incline Horsepower

Horsepower for moving up the drift incline (HP_I)

From Ref. 5.45, Pg. 26:

$$HP_I = \frac{\text{incline force} \times \text{velocity}}{33{,}000}$$

Incline force $F_I = \text{Sin } 0.57 \times 125.7 \times 2000$ lb/TON
$= 2501$ lb

$$HP_I = \frac{2501\ (lbs) \times 150\ (fpm)}{33{,}000} = 11.4$$

From Ref. 4.4.5, Pg. 59:
Efficiency = 90%

$$HP_I = \frac{11.4}{0.9} = 12.7$$

Since there are 4 motors:

$$HP_I = \frac{12.7}{4} = 3.2 \text{ HP per motor}$$

C. Gantry Traversing Drive Horsepower

Gantry drive motor horsepower, HP_g is from Ref. 4.4.5, Pg. 62, combines motor and incline horsepower

$$HP_g = 0.75\ (HP_A + HP_I)\ Ks$$

where:
Ks = 1.2 (Pg. 61)

Since the calculated HP_A already has Ks Factored in, HP_g can be rewritten:

$HP_g = 0.75\ HP_A + 0.75\ HP_t \times Ks$
$HP_g = 0.75 \times 10.7 + 0.75 \times 3.2 \times 1.2$
$HP_g = 10.9$ per motor

From Ref. 5.21, Pg. C-15:

Shunt-wound DC motors provide a relatively flat speed-torque characteristic. The shunt-wound motor also provides the lowest starting torque (275% of rated torque) and will provide the most accurate indexing performance for engaging and placing WPs.

Based on the curve for shunt-wound motors, starting torque (or horsepower) will be 275% of the rated torque (or horsepower)
$\therefore$ Rated HP $= \frac{10.9}{2.75} = 3.96$

Rounding up to the next standard motor size
Motor HP = 5

From Ref. 5.21, Pg. C-41:

For 5 HP, 500 VDC, 1750 rpm, a C1811ATZ motor frame is selected. Dimensions of this motor are given on Pg. C-139. Weight of this motor is 175 lbs. This motor will be fitted with a 180TC C-face bracket, Pg. C-79.

Size Gearmotor for $HP_g = 10.9$ Rounded to 10 HP, which is considered satisfactory for this preliminary analysis.

From Ref. 5.23, Pg. D-34:

For 10 HP, 1750 input rpm, 35 rpm output is the closest speed (compared to 36.4 rpm), AGMA Class 1 (3-10 hours per day operation) S.F. = 1.1

A 25030tN U.S. Gear Reducer is selected. Dimensions of this reducer are given on Pg. D-125. Weight of this reducer is 193 lbs. This reducer will require a C-face flange adaptable to the 180TC C-face motor bracket.

Total drive weight = 175 lbs + 193 lbs = (368 lbs) 166.0 kg

Round to 400 lbs for structural analysis

Due to lower gearmotor output speed, Gantry speed will be less than 150 ft/min.

Gantry Speed = wheel speed x 3.28 ft/m x π x wheel diameter
= 35 rev/min x 3.28 ft/m x π x 0.4 m/rev
= 144 ft/min

This speed satisfies the ±10% requirement of Ref. 4.4.2. A speed of 140 fpm was used for structural analysis. Since the drive is variable speed, the speed can be controlled not to exceed 140 fpm.

From Ref. 5.23, Pg. D-34:

Allowable overhung load (OHL) = 3887 lb

Allowable output torque = 16,880 in-lb

$$\text{Minimum sprocket radius} = \frac{16{,}880\ in\text{-}lb}{3887\ lb} = 4.3\ \text{in.}$$

∴ minimum sprocket pitch diameter (PD) is 4.3 in. x 2 = 8.6 in.

From Ref. 5.24, Pg. A-40:

For 35 rpm at 10 HP, choose a 140-pitch chain with 17-tooth sprocket.

From Attachment XII, Section 3, Pg. E-75:

Choose a 140-pitch, 17-tooth sprocket PD = 9.524 in.

3.3 Lifting Screw Size Selection

From Design Selection No. 1:
Total gantry weight = 45 MT

From Design Selection No. 7:
Hoisting frame weight = 0.25 x 45 MT = 11.25 MT

$$\text{Lifting screw load} = \left[\frac{69\ MT + 11.25\ MT}{4}\right]$$

= 20.1 MT (44218 lb)

From Attachment XII, Section 2, Pg. 19:

For life expectancy of 1 x 10^6 in. of travel, any size ball screw over 3 in. diameter will support this load.

From Ref. 5.41, Pg. 35:

For single row angular contact bearings to support the 44,218-lb load, the bearing inside diameter must be 3.937 in.

∴ select a 4" Thomson Saginaw ball screw with 1" lead. Capacity = 85,000 lb.

Estimate lifting screw weight for structural analysis:

From Design Selection No. 8:

Max. lift = 2226 mm (87.6 in.)

From Attachment XII, Section 2, Pg. 51:

4" ball nut length = 12.593 in.

Screw length = lift + ball nut length
= 87.6 in. + 12.593 in. = 100.2 in.

From Ref.Attachment XII, Section 2, Pg. 26:

Wt for 4" ball screw = 2.87 lb/in.
Total wt = (100.2 in.) (2.87 lb/in.) = 288 lbs
Wt for 4" ball nut = 53.5 lbs

Total wt for ball screw and nut = (288 lbs + 53.5 lbs)
= (341.5 lbs) 154.9 kg

allowing an additional 25% for end bearings

Total lifting screw wt = (1.25) (341.5 lbs) = (426.9 lbs) 193.6 kg
Round off to 195 kg (430 lbs)

3.4 Lifting Screw Drive Selection

A. Screw Drive

From Attachment XII, Section 2, Pg. 26.
Screw lead = 1.0 in. x 25.4 mm/in. = 25.4 mm

From Ref. 4.4.2, Pg. 50:

Slow lifting speed for a 69 MT (76 ton) load gantry is 6 ft/min. or 1.83 m/min.

From Design Selection No. 8:
Maximum lift = 2226 mm

Using a lifting time of 3 min.
Lifting rate = 2226 mm/3 min. = 74 mm/min. (2.43 ft/min.)
2.43 ft/min is well under the slow rate of 6 ft/min.

$$\text{Screw speed} = \frac{0.74\ m/\min}{0.0254\ m/rev} = 29.0\ \text{rpm}$$

From Attachment XII, Section 2, Pg. 14:

Torque = 0.177 x load x screw lead
= 0.177 x 44218 lbs x 1 in. = 7827 in-lb, rounded to 7830 in-lb

$$\text{Screw HP} = \frac{\text{operating load x screw lead x screw speed}}{3.564 \times 10^5}$$

$$= \frac{44218\ lb \times 1\ in. \times 29.0\ rpm}{356{,}400}$$

$$= 3.6\ \text{HP}$$

Using a gear reducer efficiency of 90% and a chain drive efficiency of 90% Screw HP = $\frac{3.6}{0.9 \times 0.9} = 4.4$

Since a shunt-wound motor will provide 150% of rated torque at rated speed, motor rated HP = $\frac{4.4}{1.5}$ = 2.9.

Rounding up to the next standard motor size, use 3.0 HP for one drive per screw shaft or 7.5 HP for one drive per 2 screw shafts.

From Ref. 5.23, Pg. D-28:

For 3 HP, 1750 input rpm, 28 rpm output is the nearest speed (compared to 29 rpm), AGMA Class 1 (3-10 hours per day operation) S.F. = 1.0

A 24030tN U.S. Gear Reducer is selected. Dimensions of this reducer are given on Pg. D-125. This reducer will require a C-face flange adaptable to the 180TC C-face motor bracket.

From Ref. 5.23, Pg. D-125:

Weight of reducer = 53.5 kg (118 lb)

From Ref. 5.21, Pg. C-41:

For 3 HP, 500 VDC, 1750 rpm, a C1811ATZ motor frame is selected. Dimensions of this motor are given on Pg. C-139. This motor will be fitted with a 180TC C-face bracket, Pg. C-79.

From Ref. 5.21, Pg. C-139:

Weight of this motor = 79.4 kg (175 lb)
Weight of this drive = 175 lb + 118 lb = 132.9 kg (293 lb)
Round up to 500 lb for structural analysis

From Ref. 5.23, Pg. D-32:

For 7.5 HP, 1750 input rpm, 28 rpm output is the nearest speed (compared to 29 rpm), AGMA Class 1 (3-10 hours per day operation) S.F. = 1.0

A 25030tN U.S. Gear Reducer is selected. Dimensions of this reducer are given on Pg. D-125. This reducer will require a C-face flange adaptable to the A180TC C-face motor bracket.

Reducer weight = 86.2 kg (190 lb)

From Ref. 5.21, Pg. C-41:

For 7.5 HP, 500 VDC, 1750 rpm, a C1811ATZ motor frame is selected. Dimensions of this motor are given on Pg. C-139. This motor will be fitted with A180TC C-face bracket.

Motor weight = 79.4 kg (175 lb)

Total weight of drive = 190 lb + 175 lb = 164 kg (365 lb), round up to 800 lb for structural analysis

B. Chain Drive for 3 HP Gearmotor

From Ref. 5.23, Pg. D-28:
Overhung load = 3146 lb
Output torque = 6,427 in-lb

Sprocket radius $= \frac{6427\ in\text{-}lb}{3146\ lb} = 2.04$ in.

∴ Any sprocket with PD greater than 4.25 in. is satisfactory

From Ref. 5.24, Pg. A-38:

For crane and hoist, load classification is Class C with a SF = 1.7

Design HP = HP x SF = 3.0 x 1.7 = 5.1

Speed = 28 rpm

Ref. 5.24, Pg. A-40 for 7.5 HP:
Sprockets are 140 pitch, 17 teeth

From Attachment XII, Section 3, Pg. E-75:
Pitch diameter of a 17-tooth, 140-pitch sprocket = 9.524 in.

Max. bore = 3 in. This bore is sufficient for both gearmotor and ball screw shafts.

C. Chain Drive for 7.5 HP Gearmotor

From Ref. 5.23, Pg. D-32:
Overhung load allowed = 4121 lb
Torque = 15,889 in-lb

Sprocket radius $= \frac{15,889\ in\text{-}lb}{4121\ lb} = 3.86$ in.

∴ Any sprocket with PD larger than 7.75 in. is satisfactory

From Ref. 5.24, Pg. A-38:
For crane and hoists, load classification is Class C, with a SF = 1.7
Design HP = HP x SF = 7.5 x 1.7 = 12.75

From Ref. 5.24, Pg. A-41:

For a speed of 28 rpm using 15-HP, sprockets are 160 pitch, 21 teeth.

From Attachment XII, Section 3, Pg. E-79:

Pitch diameter of a 21-tooth, 160 pitch sprocket is 13.419 in., with maximum bore of 3½ in., which is sufficient for the gear motor and the ball screw.

3.5 Lifting Head Trolley Traversing Screw Selection

Maximum horizontal load on the lifting head trolley screws will occur due to seismic acceleration.

From Section 4.3.20:

Horizontal acceleration is 0.27 g rounded off to 0.30 g.

Since two screws will support the load

Screw load $= \frac{69\ MT \times 0.3}{2} = 10.35$ MT

= 10.35 MT x 2205 lb/MT
= 22,821.8 lb; round off to 23,000 lb.

From Attachment XII, Section 2, Pg. 19: Select a 2.25 in. Saginaw ball screw with a 1" (25.4 mm) lead

Determine traversing screw wt:

From Attachment XII, Section 2, Pg. 26:

Weight of 2.25 in. diameter ball screw is 0.906 lb/in. To allow for ball nut, end bearing and miscellaneous supports, increase ball screw wt by a factor of 2.

Wt per foot = (2) (12 in/ft) 0.906 lb/in = 22 lb/in.
Round off to 30 lb/ft for structural analysis

3.6 Lifting Head Trolley Traversing Screw Drive Selection

From Sections 4.1.6 and 4.3.11:

Length of longest WP = 5850 mm
Length of shortest WP = 3790 mm
Δ WP length = 2060 mm

From Section 4.3.17:
WP end skirt is 225 mm at both ends of the WP. Allowing 200 mm for travel at both ends.
Total Travel = 2060 mm + 2 x 200 mm = 2460 mm

Since there is a takeup at each end of the WP

$$\text{Takeup travel} = \frac{2460\ mm}{2} = 1230\ \text{mm}$$

Using a travel time of 15 sec is reasonable considering the short travel distance.

$$\text{Lifting head travel speed} = \frac{1230\ mm \times 60\ \text{sec/min}}{15\ sec}$$

$$= 4920\ \text{mm/min}$$

$$\text{Screw speed} = \frac{4920\ mm/\text{min}}{25.4\ mm/rev} = 193\ \text{rpm}$$

From Attachment II, Section 13.0:

Weight of trolley plus lifting head = 5,420 lb

From Ref. 5.50, Pg. 2:

Coefficient of friction = 2.7%, rounded up to 3.0%

Friction Force = 5420 lb x 0.03 = 163 lb

From Design Selection No. 11:

Acceleration of Trolley = 1 ft/sec^2

$$\text{Acceleration Force} = \frac{5420\ lbm}{32.2\ \frac{lbm}{lbf}\ \frac{ft}{sec^2}} \times 1.0\ \text{ft/sec}^2 = 168\ \text{lb}$$

Total Screw Load = (168 lbs + 163 lbs) = (331 lb) 150.1 kg

From Attachment XII, Section 2, Pg. 14:
Torque = 0.177 x load x screw lead
= 0.177 x 331 lbs x 1 in. = 58.6 in-lb

HP to turn lifting head screw at 193 rpm

$$HP = \frac{force \times screw\ lead \times speed}{3.564 \times 10^5 \times 0.9} = \frac{331\ lbs \times 1\ in. \times 193\ rpm}{356{,}400 \times 0.9} = 0.2, \text{ use } 1/3 \text{ HP}$$

From Ref. 5.23, Pg. E-18, use a U.S. Gear Reducer, MbN gear frame size, 2102, 188 rpm output is the nearest output speed with 1750 rpm input speed and is considered satisfactory for this preliminary analysis, ⅓ HP

From Ref. 5.23, Pg. E-101:
Weight of reducer = (42 lb) 19.0 kg

$$\text{Lifting head trolley travel speed} = 1\ \frac{in.}{rev} \times \frac{188\ rev/min}{12\ in./ft} \times 0.3048\ m/ft = 4.77\ m/min.$$

From Ref. 5.21, Pg. C-4:

For ⅓ HP, 1750 rpm, a SG56HC motor frame is selected. Dimensions of this motor are given on Pg. C-20. This motor is normally a C-face motor. Voltage is 90 V armature, 100 V field.

From Ref. 5.21, Pg. C-17:

Motor weight = 31 lb

Total drive weight = 42 lb + 31 lb = (73 lb) 33 kg
Round to (100 lb) 45.4 kg for structural analysis

3.7 Lifting Head Roller Selection

From Section 4.3.11:
Waste package weight = 69 MT = 152,145 lb

$$\text{Load carried per lifting head} = \frac{152{,}145\ lbs}{2} = 76{,}072\ lb$$

From Attachment V, page 3:
Weight of lifting head trolley = 5000 lb
With 4 rollers per trolley equally loaded

$$\text{Lifting head roller load} = \frac{76{,}072\ lbs + 5000\ lbs}{4} = 20{,}268\ lb$$

From Attachment XII, Section 7, Pg. 2:

Capacity of the 8-OT roller assembly is 16,000 lb.
∴ 6 roller assemblies will be required per trolley.

Lifting head roller load $= \frac{76{,}072\ lbs + 5000\ lbs}{6} = 13{,}512$ lbs

From Attachment XII, Section 7, Pg. 2: Roller selection is Hilman 8-OT.

Future analysis should refine the design of the trolley and lifting head to verify loading on trolley rollers.

ATTACHMENT VI

GANTRY CARRIER WEIGHT AND WHEEL SELECTION

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TITLE: GANTRY CARRIER WEIGHT AND WHEEL SELECTION

1.0 PURPOSE

This calculation will provide the basis for preliminary evaluation of:

A. Gantry Carrier Weight
B. Gantry Carrier Wheel Selection
C. Gantry Carrier Rail Selection

2.0 INITIAL DESIGN SELECTIONS

No.	Input Description
1	Weight of gantry structure shall be 45 MT, Attachment II, Section 1.1
2	Weight of coupler shall be 175 lb, Attachment IX, Section 2.0
3	Weight of truck shall be 11,000 lb, Attachment VIII, Section 2.1
4	Gantry carrier wheel hardness shall be 320 BHN.
5	Gantry carrier wheel diameter shall be 762 mm (30 in.).
6	Gantry catrier rails shall be 57 kg/m (115 lb/yd) AREA rail.

3.0 SOLUTION

3.1 Gantry Carrier Weight (Figures 1 and 2)

A. Wheel Truck Assembly - A (Fig. 3&4)

From Fig. 4:

1. $2.901\text{ m} \times 0.812\text{ m} \times 0.025\text{ m} = \underline{0.059\text{ m}^3}$

2. $(1.422\text{ m} \times 0.812\text{ m}) - (0.198^2\text{ m}^2\,\pi)$
   $1.1547\text{ m}^2 - 0.1232\text{ m}^2 = 1.032\text{ m}^2 \times 0.051\text{ m} = \underline{0.053\text{ m}^3}$

3. Wheel truck - 4950 kg (Attachment VIII)

B. Deck Assembly - B (Fig. 5&6):

From Fig. 6:

1. 11.42 m x 3.546 m x 0.025 m = 1.0124 m^3

2. 90 lb/yd rail = 30 lb/ft x 1.488 $\frac{kg/m}{lbs/ft}$ = 44.6 kg/m

Rail weight = 11.42 m x 44.6 kg/m = 509.33 kg

From Fig. 10:

3. Wheel stops 0.0045 m^3

(1) Item	(2) Des.	(3) Qty.	(4) Vol. m^3	(5) Density kg/m^3*	(6) Wt. Ea. kg (4)x(5)	(7) Total kg (3)x(6)
A-1	PL	2	0.059	7849.8	463.1	926.3
A-2	PL	2	0.053	7849.8	416.0	832.0
A-3	Assm.	2		7849.8	4950.0	9900.0
B-1	PL	1	1.0124	7849.8	7947.1	7947.1
B-2	Rail	2		7849.8	509.3	1018.6
B-3	Assm.	2	0.0045	7849.8	35.3	70.6
						20,694.6 kg

*From Ref. 4.4.1

490 lb/ft³ x 16.02 $\frac{kg/m^3}{lbs/ft^3}$ = 7849.8 kg/m^3

C. **Frame Support Assembly - C (Figures 7 and 8)**

From Fig. 8:

1. Side Frame (2 outside, 2 inside)
2.9 m x 0.0158 m^2 x 2 = 0.0916 m^3
4.172 m x 0.0276 m^2 = 0.1151 m^3

0.672 m x 0.254 m = 0.171 m^2
0.672 m x 0.382 m = 0.257 m^2
$\frac{0.672\ m}{2}$ x 0.652 m = 0.219 m^2
0.254 m x 1.038 m = 0.263 m^2
0.91 m^2 x 0.018 m = 0.0164 m^3 x 2
= 0.0328 m^3
0.0916 m^3
0.1151 m^3
Total volume of side frame = 0.2395 m^3

2. Outside structural member (30 required)
1.001 m x 0.0158 m^2 = 0.0158 m^3

3. Inside structural member (14 required)
$0.592\ m \times 0.0158\ m^2 = \underline{0.0094\ m^3}$

4. Coupler support assembly
$0.592\ m \times 0.0178\ m^2 = 0.0105\ m^3$
$0.089\ m \times 0.642\ m \times 0.025\ m = 0.0014\ m^3$
$0.592\ m \times 0.524\ m \times 0.025\ m = \underline{0.0078\ m^3}$
Total volume $= 0.0197\ m^3$

(1) Item	(2) Des.	(3) Qty.	(4) Vol. m³	(5) Density kg/m³*	(6) Wt. Ea. kg (4)x(5)	(7) Total kg (3)x(6)
C-1	WF	4	0.2395	7849.8	1880.0	7520.0
C-2	WF	30	0.0158	7849.8	124.0	3720.0
C-3	WF	14	0.0094	7849.8	73.8	1033.2
C-4	Assm.	1	0.0197	7849.8	154.6	154.6

Total Weight of Frame Support Assembly = 12,427.8 kg

D. Gusset Plate

From Fig. 9:
$0.1389\ m^2 \times 0.025\ m = 0.0035\ m^3 \times 7849.8\ kg/m^3 \times 22 = \underline{604.4\ kg}$

E. End Plate

From Fig. 9:
$3.546\ m \times 0.559\ m \times 0.051\ m = 0.1011\ m^3 \times 7849.8\ kg/m^3 \times 2 = \underline{1587.2\ kg}$

F. Coupler

From Attachment IX, Section 2.0:
Coupler Weight = 79.0 kg (175 lb) (Ref. 3.4)

Coupler box plates from Fig. 9:
$0.4\ m \times 0.353\ m \times 0.051\ m = 0.0072\ m^3 \times 2 = 0.0144\ m^3$
$0.172\ m \times 0.353\ m \times 0.051\ m = 0.031\ m^3 \times 2 = \underline{0.0062\ m^3}$
$0.0206\ m^3$
$\times\ 7849.8\ kg/m^3 = 161.7\ kg$
$\underline{79.0\ kg}$
$240.7\ kg$

Total car weight

Assembly A&B	20,694.6 kg
C	12,427.8 kg
D	604.4 kg
E	1,587.2 kg
F	240.7 kg
	35,554.7 kg

Weight of gantry carrier = use 36,000 kg

3.2 Gantry Carrier Wheel Selection

From Attachment II, Section 1.1:
Weight of gantry = 45 MT

From Section 3.1:
Weight of gantry carrier = 36.0 MT

The gantry carrier will have 8 wheels, uniformly loaded.

$$\text{Maximum wheel load} = \frac{45\ MT + 36.0\ MT}{8} = 10.1\ \text{MT}$$

$$\text{Minimum wheel load} = \frac{36.0\ MT}{8} = 4.5\ \text{MT}$$

From Ref. 4.4.5, Pg. 33
Mechanical mean effective load factor K_W

$$K_W = \frac{2\ (maximum\ load) + (minimum\ load)}{3\ (maximum\ load)} = \frac{2(10.1\ MT) + 4.5\ MT}{3(10.1\ MT)} = 0.82$$

From Ref. 4.4.5, Pg. 49:
Wheel load service coefficient, K_{WL}

$$K_{WL} = K_W \times C_S \times S_m$$

where: C_S = speed factor; dependent on wheel speed
S_m = wheel service factor = 1.25 Cd
Cd = machine service factor

From Ref. 4.4.5, Pg. 33
Cd = 1.16

From Input 4.3.27:
Carrier speed = 5 mph

$$\text{speed} = \frac{5\ mi}{hr} \times \frac{88\ ft/\min}{mi/hr} = 440\ \text{ft/min}$$

From Design Selection No. 5:
Wheel diameter = 30 in.

$$\text{Wheel speed} = \frac{440\ ft/\min}{\pi \times \frac{30}{12} ft} = 56.0\ \text{rpm}$$

From Ref. 4.4.5, Pg. 49:
Since wheel speed is > 31.5 rpm

$$C_S = 1 + \left(\frac{56 - 31.5}{328.5}\right) = 1.07$$

$$\therefore K_{WL} = 0.82 \times 1.07 \times 1.25 \times 1.16 = 1.27$$

From Ref. 4.4.5, Pg. 47:
Basic allowable wheel load, B_{WL}

$$B_{WL} = K \times D \times W$$

where: K = hardness coefficient of the wheel when BHN ≥260

$$K = 1300\left(\frac{BHN}{260}\right)^{0.33}$$

D = wheel diameter = 30 in.
W = effective rail width

From Ref. 4.4.5, Pg. 50:
W = width of rail - 2 x corner radius

From Design Selection No. 6:
Gantry Carrier rails are 115 lb/yd AREA rails

From Attachment XII, Section 1, Pg. 26:
Width of rail head = 2-23/32 in.

Corner radius - ⅜ in.

W = 2-23/32 - 2 x ⅜ = 1.97 in.

BHN = Brinell hardness

From Design Selection No. 4:
BHN = 320

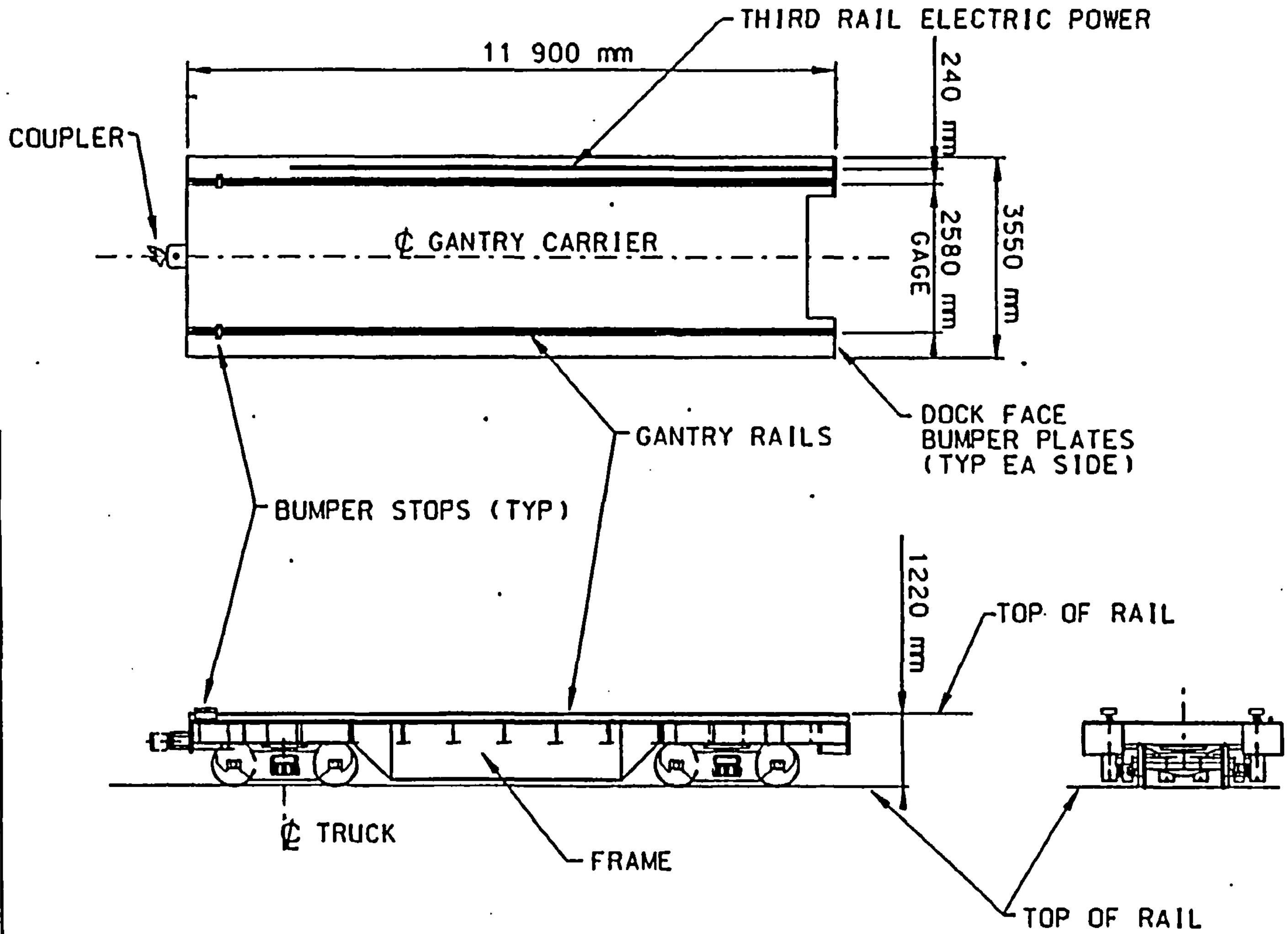
$$K = 1300\left(\frac{320}{260}\right)^{0.33} = 1392$$

B_{WL} = 1392 x 30 in. x 1.97 in.
= (82,270 lb) 37,020.0 kg

From Ref. 4.4.5, Pg. 49:
Equivalent durability wheel load, Pe

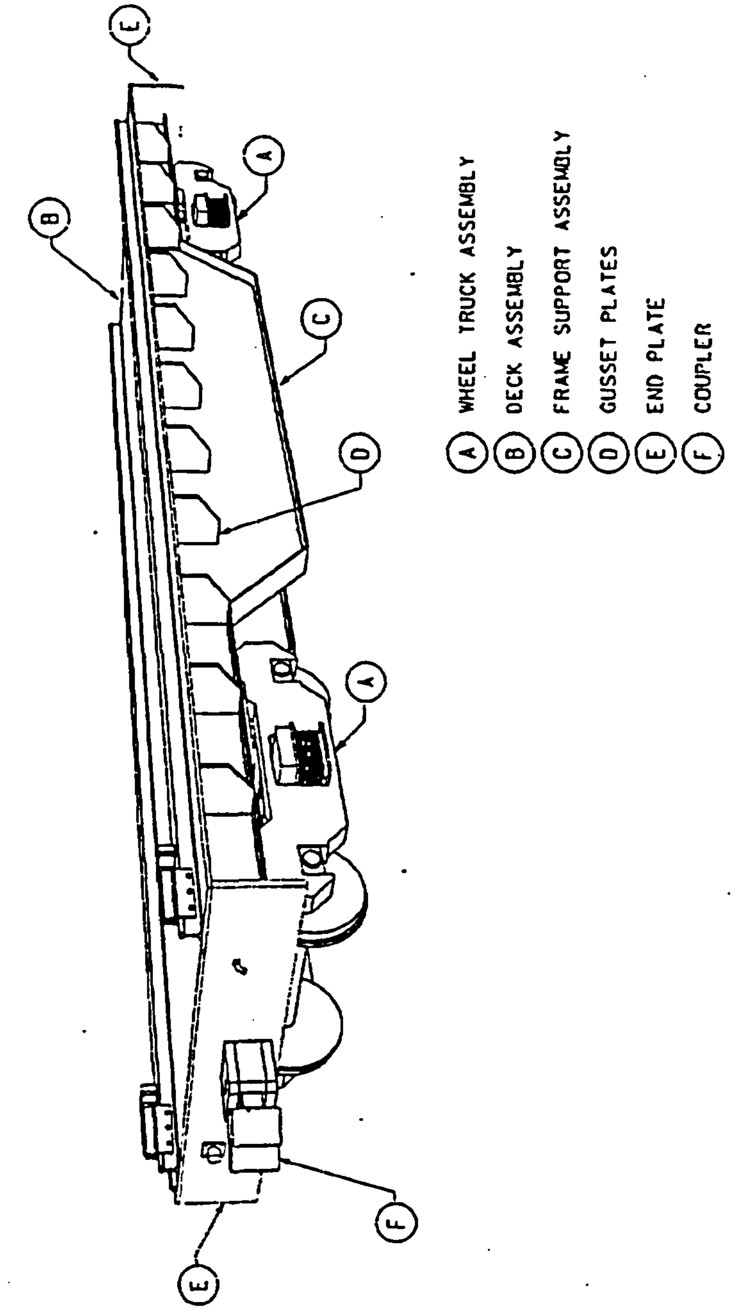
Pe = maximum wheel load x K_{WL}
= 10.1 MT x 1.27 = 12.8 MT (28,311 lb)

This load is compared to the basic allowable wheel load, B_{WL} = 82,270 lb.
∴ the 30 in. diameter wheel and 115 lb AREA rail is satisfactory.

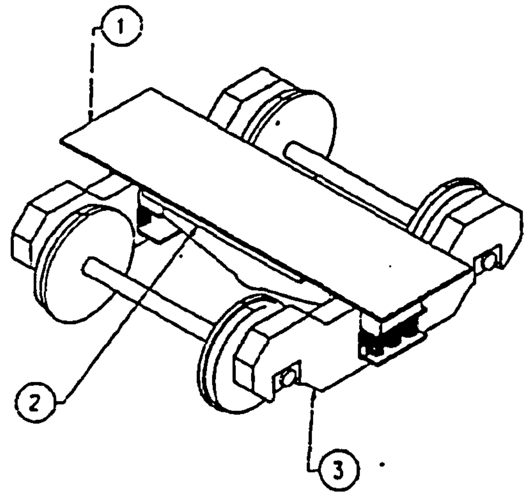


GANTRY CARRIER
ARRANGEMENT
FIG. 1

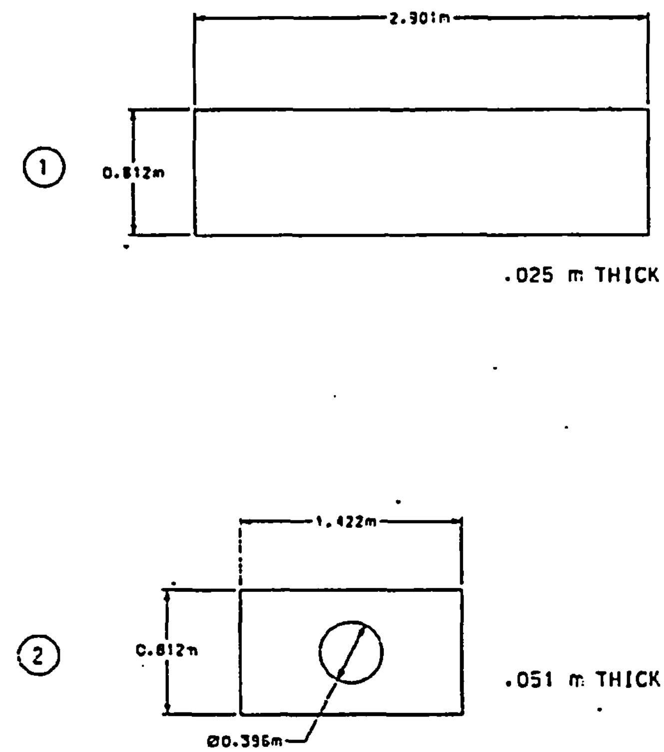
CAD FILE: t:\repss\mech\fig\ssme0244.fig



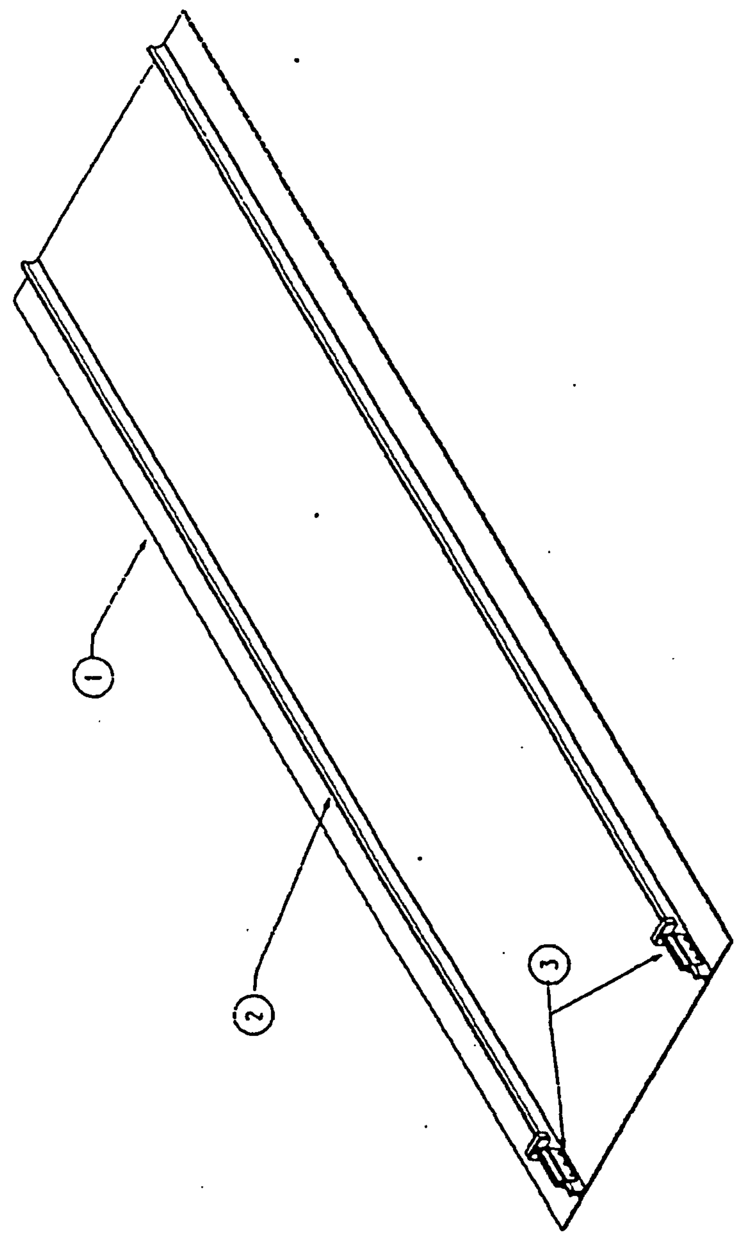
GANTRY CRANE CARRIER ASSEMBLIES AND COMPONENTS
FIG. 2



GANTRY CRANE CARRIER
WHEEL TRUCK ASSEMBLY
FIG. 3

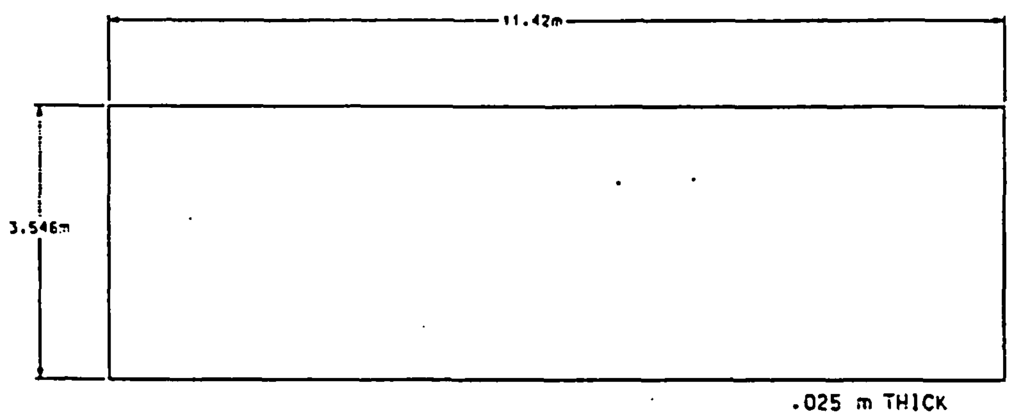


WHEEL TRUCK ASSEMBLY
FIG. 4



DECK
ASSEMBLY
FIG. 5

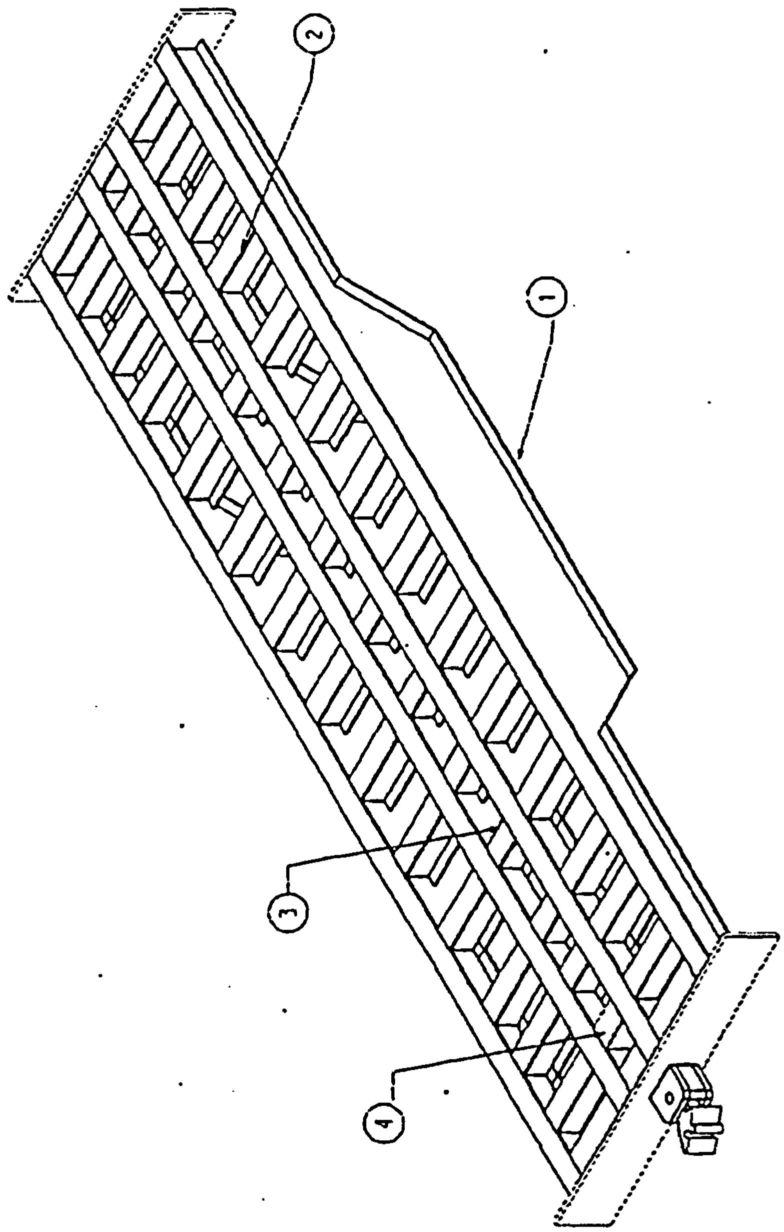
(1)



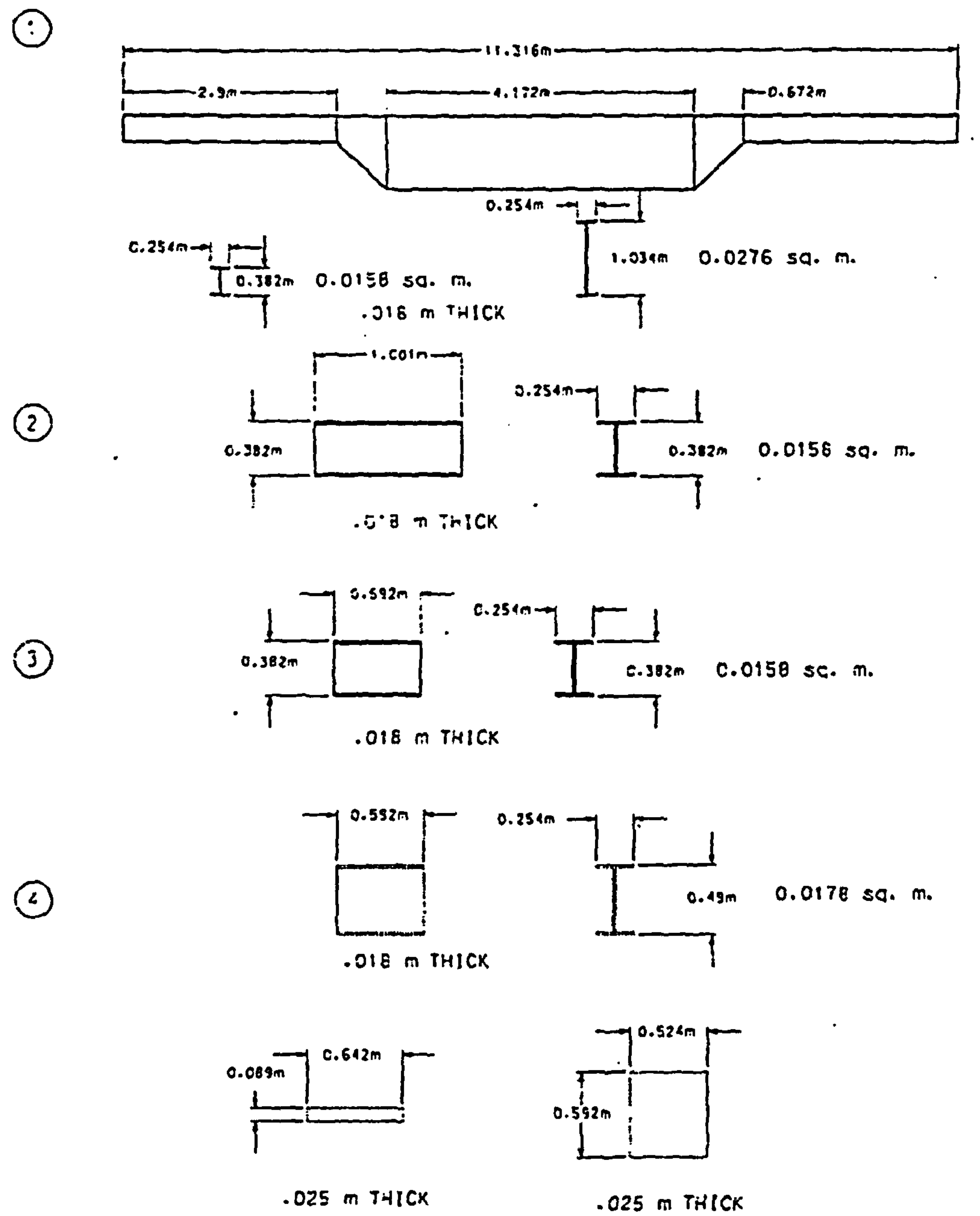
(2) 90 lb. RAIL - 11.418m LONG

(3) WHEEL STOPS - 164.6 Kg. each

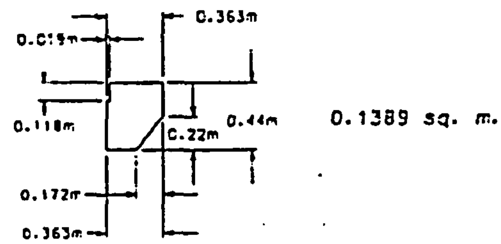
DECK COMPONENTS
FIG. 6



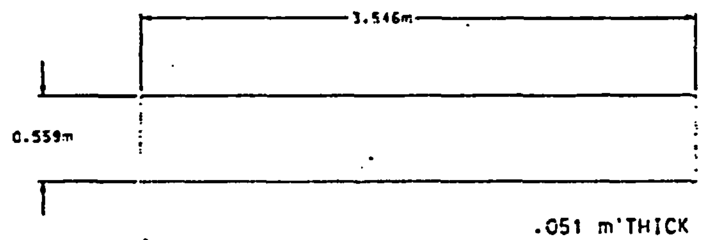
FRAME SUPPORT
ASSEMBLY
FIG. 7



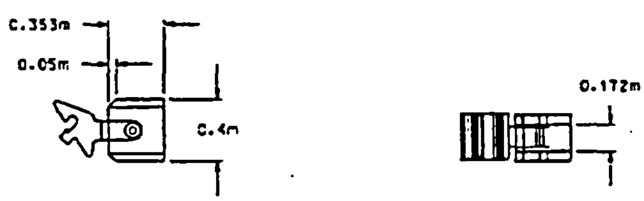
FRAME SUPPORT COMPONENTS
FIG. 8



ASSEMBLY D - GUSSET PLATE

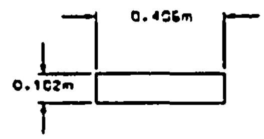


ASSEMBLY E - END PLATE



ASSEMBLY F - COUPLER

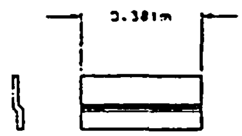
ASSEMBLIES
D.E & F
FIG. 9



.026m THICK

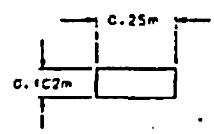
0.102m x 0.406m x 0.026m = 0.0011 cu. m

A=0.0038 sq. m



.018m THICK

.0038 sq. m x .381m = 0.0014 cu. m x 2 = 0.0028 cu. m



.026m THICK

0.102m x 0.25m x 0.026m = 0.0006 cu. m

0.0011
0.0028
0.0006
0.0045 cu. m

WHEEL STOPS
ASSEMBLY
FIG. 10

ATTACHMENT VII

TRANSPORT LOCOMOTIVE - EQUIPMENT SELECTION

NOTE: A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TRANSPORT LOCOMOTIVE - EQUIPMENT SELECTION

1.0 PURPOSE

The purpose of this calculation is to evaluate locomotive capacity requirements under various operating conditions to provide a basis for locomotive selection. The following conditions are evaluated:

A. Tandem locomotives retrieving a loaded WP with the Transporter and transporting up the north tunnel ramp

B. One locomotive placing a WP at the emplacement loading dock
   1. Case A - operation in the main drift
   2. Case B - operation in the emplacement turn out with 20 m radius curve

C. Locomotives transporting a WP with the Transporter down the north tunnel ramp
   1. Case A - two locomotives transporting WP
   2. Case B - one locomotive transporting WP

D. Verify locomotive selection for 20-meter curve in drift access turn out.

2.0 INITIAL DESIGN SELECTIONS

No. Input Description

1. The following rolling resistance and adhesion data is from (Attachment XII, Section 4, Section 4068) and are used in Attachments III through VII:

   Rolling resistance for rail cars with ball or roller bearings is 20 lb/ton (10.02 kg/MT).
   Locomotive running adhesion of driving wheels on clean dry rails and flat grade is:

Steel wheels	25% of wheel load
Cast iron wheels	20% of wheel load

2. Two locomotives operating in tandem shall each provide 50% of the tractive effort to transport the load. (Attachment VIII)

3. The considered worst case of 100% braking of the trailing load by one of the two locomotives operating in tandem shall be included in the determination of the capacity. (Attachment VIII)

No.	Input Description
4.	Maximum grade of North Ramp shall be 2.1486% (From Section 4.1.1) and is rounded up to 2.5% for sizing locomotive capacity.
5.	Loaded Transporter weight shall be 233.15 MT (257 tons) (Attachment I, Section 1.0)
6.	Emplacement turnout grade shall be +0.5%. Section 4.3.8 Grades vary from 0.25% to 0.75%. Use the average of 0.5%.
7.	Main Drift slope shall be 1.35% rounded up to 1.4%. (Section 4.1.2)

3.0 SOLUTION

3.1 Determine Locomotive Capacity for Condition A (Figure 1)

For locomotive sizing with retrieval of loaded transporter transporting up to the north portal:

<u>Operating Criteria</u>:

1. Trolley power
2. Speed = 5 mph (Section 4.3.27)
3. Slope = 2.5% (Design Selection No. 4)
4. Two locomotives with equal loads
5. Curve = 305-m, round down to 300 m (984ft) (Section 4.1.3) in main track
6. K = 0.25
7. Transporter Weight = 233.15 MT = 257 T (Design Selection No. 5)

The following definitions and/or formulas are from Attachment XII, Section 4068, Pg. 1.

Rolling resistance, R_R, is friction from car journals and friction between wheel treads and flanges and rail. Typical values are 20 lb/ton for cars with ball or roller journal bearings.

$$R_R = 20 \text{ lb/ton} \times \text{Weight(ton)}$$
$$R_R = 20 \text{ L lb/ton}$$

Grade resistance, R_I, is resistance due to grades and is 20 lb/ton, for each percent rise in grade

$$R_I = 20 \text{ lb/ton} \times \text{Weight} \times \%\text{ Grade}$$
$$R_I = 20 \times L \times \%$$

Curve resistance, R_C, additional friction between wheels and rail due to track curvature.

$$R_C/\text{ton} = \frac{400 \times Wheelbase\ (ft)}{radius\ of\ curve\ (ft)}$$

$$R_C = \frac{400 \times Wheelbase\ (ft)}{radius\ of\ curve\ (ft)} \times L$$

Train resistance, R_T, is the summation of rolling, grade, and curve resistance.

$$R_T = R_R + R_I + R_C$$
$$R_T = 20\,L + 20\,L \times \%\text{ Grade} + \frac{400 \times Wheelbase\ (ft)}{radius\ of\ curve\ (ft)} \times L$$

Tractive effort, TE, is the force in pounds exerted at the driving wheels.

$$TE = \text{Adhesion} \times L = KL$$

Adhesion, K, is the percentage of weight on the driving wheels. K = 25% for steel wheels on steel.

For the transporter:

$$R_I = 20 \text{ lb/ton} \times \text{Weight (ton)} \times \%\text{ Grade}$$

$$R_I = 20 \text{ lb/ton} \times 257 \text{ ton} \times 2.5 = 12{,}850 \text{ lbs}$$

$$R_R = 20 \text{ lb/ton} \times \text{Weight(ton)}$$

$$R_R = 20 \text{ lb/ton} \times 257 \text{ ton} = 5140 \text{ lbs}$$

$$R_C = \frac{400 \times wheel\ base\ (ft)}{radius\ of\ curve\ (ft)} \times \text{load}$$

Use 6 ft for wheel base (See dimensions in Attachment VIII, Pgs. VIII-5 and VIII-6)

$$R_C = \frac{400 \times 6\ ft}{984\ ft}\ (257 \text{ lbs}) = 627 \text{ lbs}$$

$$R_T = R_R + R_I + R_C$$

R_T = 12,850 lbs + 5140 lbs + 627 lbs
= 18,617 lbs

For the locomotive:

$TE = R_I + R_R + R_C + F_{DB}$
Where F_{DB} = Draw bar pull
Using L(ton) as the locomotive weight

R_I = 20 lb/ton x % x L = 20 lb/ton x 2.5 x L = 50 L lb/ton
R_R = 20 lb/ton x L = 20 L lb/ton
$R_C = \frac{400 \times WB}{R} \times L = \frac{400 \times 8.3\ ft}{984\ ft} \times L = 3.37\ L$ lb/ton

Wheel base = 8.3 ft (Attachment XII, Section 4)
TE (min. required) = KL = 0.25 x L x 2000 lb/ton = 500 L lb/ton

Since the locomotives are sharing the load equally,

F_{DB} (draw bar force) = ½ R_T
or $F_{DB} = \frac{18{,}617\ lbs}{2} = 9308.5$ lbs

Substituting in the equation for TE:

500 L lb/ton= 20 L lb/ton + 50 L lb/ton + 3.37 Llb/ton + 9308.5 lbs

426.6 L lb/ton = 9,308.5 lbs

L = 21.8 tons (minimum required for traction)

∴ For this operating condition select a standard 30 ton locomotive.

3.2 Determine Locomotive Capacity for Condition B (Figure 2)

Single load operation for placement of WP transporter at emplacement tunnel loading dock. For this condition the locomotive is evaluated for two cases:

A. Operation in the Main Drift without curves
B. Operation in the Emplacement Drift Turnout with 20 m radius curve

<u>Operation Criteria</u>:

1. Trolley power
2. Speed = 5 mph (Section 4.3.27)
3. Emplacement Turnout Slope = +0.5% (Design Selection No. 6)
4. Main Drift Slope = + 1.4% (Design Selection No. 7)

5. One locomotive
6. Curve = 20 m (65.7 ft)
7. K = 0.25

Case A
Refer to definitions and equations provided for solution to Condition A.
For the transporter:

$R_T = R_C + R_R + R_I$

$R_C = 0$ There are no curves

$R_R = 20$ lb/ton x Weight (ton) = (20 lb/ton)(257 T) = 5140 lbs

$R_I = 20$ lb/ton x Weight (ton) x % Grade = 20 lb/ton x 257 ton x 1.4 = 7196 lbs

$R_T = 0 + 5140$ lbs + 7196 lbs = 12,336 lbs (compression)

For the locomotive:

$TE = F_{DB} + R_C + R_I + R_R$

Tractive effort at impending slippage is

TE (min required)= 0.25 x 2000 lb/ton x L (Attachment XII, Section 4068, Pg. 1)
= 500 L lb/ton

$F_{DB} = R_T = 12{,}336$ lbs

$R_C = 0$, There are no curves

$R_I = 20$ lb/ton x 1.4 L = 28 L lb/ton

$R_R = 20$ L lb/ton

Substituting in the equation for TE:

500 L lb/ton= 12,336 lbs + 0 + 28 L lb/ton + 20 L lb/ton

452 L lb/ton = 12,336 lbs

L = 27.3 tons (use a 30-ton locomotive)

∴ For the case of a single locomotive operating in the main drift a 30-ton locomotive is required.

Case B:
Refer to definitions and equations provided for solution to Condition A.
For the transporter:

$R_T = R_C + R_R + R_I$

$R_C = \frac{400 \text{ x Wheelbase (ft)}}{\text{Radius (ft)}} \text{ x Weight (ton)} = \frac{400 \text{ x } 6 \text{ ft}}{65.7 \text{ ft}} \text{ x } 257 \text{ ton} = 9388 \text{ lbs}$

R_R = 20 lb/ton x Weight (ton) = (20 lb/ton)(257 T) = 5140 lbs

R_I = 20 lb/ton x Weight (ton) x % Grade = 20 lb/ton x 257 ton x 0.5 = 2570 lbs

R_T = 9388 lbs + 5140 lbs + 2570 lbs = 17,098 lbs (compression)

For the locomotive:

$TE = F_{DB} + R_C + R_I + R_R$

Tractive effort at impending slippage is

TE (min required)= 0.25 x 2000 lb/ton x L (Ref. 5.28, Attachment XII, Section 4068, Pg. 1)
= 500 L lb/ton

$F_{DB} = R_T$ = 17,098 lbs

$R_C = \frac{400 \text{ x } 8.3 \text{ ft}}{65.7 \text{ ft}} \text{ x L} = 50.5 \text{ L lb/ton}$

R_I = 20 lb/ton x 0.5 L = 10 L lb/ton

R_R = 20 L lb/ton

Substituting in the equation for TE:

500 L lb/ton= 17,098 lbs + 50.5 L lb/ton + 10 L lb/ton + 20 L lb/ton

419.5 L lb/ton = 17,098 lbs

L = 40.8 tons (use a 45-ton locomotive)

∴ For the case of a single locomotive operating in the drift turn out a 45-ton locomotive is required.

3.3 Determine Locomotive Capacity for Condition C (Figure 3)

Locomotive with loaded Transporter descending down north ramp from portal. For this condition the locomotive is evaluated for two cases:

A. Draw bar force shared by two locomotives.
B. Draw bar force on one locomotive.

Refer to definitions and equations provided for solution to Condition A.
Operating Criteria:

1. Trolley power
2. Speed = 5 mph (Section 4.3.27)
3. Slope = 2.5% (Design Selection No. 4)
4. Curve = 300 m round down to 300 m (984ft) (Section 4.1.3) in main track
5. K = 0.25

For the transporter:

$$R_C = \frac{400 \text{ x } Wheelbase\ (ft)}{Radius\ (ft)} \text{ x Weight (ton)} = \frac{400 \text{ x } 6\ ft}{984\ ft} \text{ x 257 ton} = 627 \text{ lbs}$$

R_I = 20 lb/ton x Weight (ton) x % Grade = (20 lb/ton)(257 ton)(2.5) = 12,850 lbs

R_R = 20 lb/ton x Weight (ton) = 20 lb/ton x 257 tons = 5,140 lbs

R_T = 12,850 lbs − 5140 lbs − 627 lbs = 7083 lbs (downslope)

Case A

For the locomotive operating in tandem with load shared equally:

Draw bar force on each locomotive $= \frac{7083\ lbs}{2}$ = 3542 lbs/locomotive

$$R_C + R_R + F_B = R_I + 3542 \text{ lbs}$$

$$F_B = R_I + 3542 \text{ lbs} - R_C - R_R$$

L = minimum weight required for braking with 25% adhesion

$F_B = KL = 0.25 \times 2000$ lb/ton $\times L = 500\ L$ lb/ton

$R_I = 20$ lb/ton x Weight(ton) x % Grade = 20 lb/ton x L x 2.5 = 50 L lb/ton

$R_R = 20\ L$ lb/ton

$$R_C = \frac{400 \times 8.3\ ft}{984\ ft} \times L = 3.4\ L\ lb/ton$$

Substituting in equation the equation for F_B:

500 L lb/ton = 50 L lb/ton - 20 L lb/ton - 3.4 L lb/ton + 3542 lbs
(500 - 50 + 20 + 3.4)L lb/ton = 3542 lbs
473.4 L lb/ton = 3542 lbs
L = 7.5 ton

Braking force $F_B = 0.25\ L$ lb/ton
= 0.25 x 7.5 ton x 2000 lb/ton = 3750 lbs

∴ For the case of two locomotives operating in tandem transporting down the north ramp, the locomotive must be 8 tons.

Case B

Draw bar force is all on one locomotive

$R_C + R_R + F_B = R_I + 7083$
$F_B = R_I + 7083 - R_C - R_R$
$R_I = 20$ lb/ton x L x 2.5 = 50 L lb/ton
$R_R = 20\ L$ lb/ton
$F_B = 0.25\ L \times 2000$ lb/ton = 500 L lb/ton

$$R_C = \frac{400 \times 8.3\ ft}{984\ ft} \times L = 3.4\ L\ lb/ton$$

Substituting in equation in the Equation for F_B:

500 L lb/ton = 50 L lb/ton - 20 L lb/ton - 3.4 L lb/ton + 7083 lbs

(500 - 50 + 20 + 3.4) L lb/ton = 7083 lbs

473.4 L lb/ton = 7083 lbs

L = 15 ton

Braking force $F_B = 0.25$ L lb/ton
$= 0.25 \times 15$ ton $\times 2000$ lb/ton $= 7500$ lb

∴ For the case of a single locomotive transporting down the north ramp the locomotive must be 15 tons.

3.4 Verify Locomotive for 20-meter Radius Curve

From Condition B Case B, a 45-ton locomotive is selected.

This locomotive will be a modified Type 201 locomotive with 100-inch wheel base and 30-inch-diameter wheels. See Attachment XII, Section 4, minimum radius for these conditions is 46 ft.

∴ The 20 m (65.6 ft) curve radius is satisfactory.

For the 45-ton locomotive:

$$HP = \frac{TE \times S}{356.25}$$ (Attachment XII, Section 4068, Pg. 2)

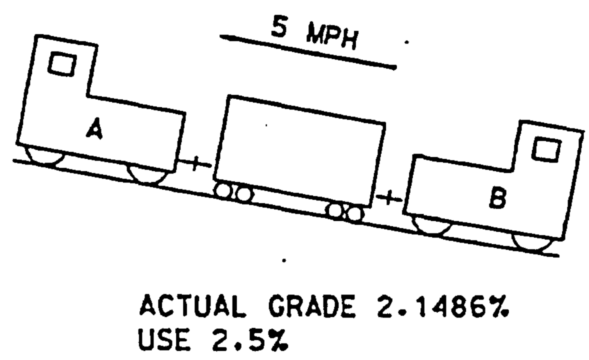
$$HP = \frac{500 \times 45 \times 5}{356.25} = 316$$

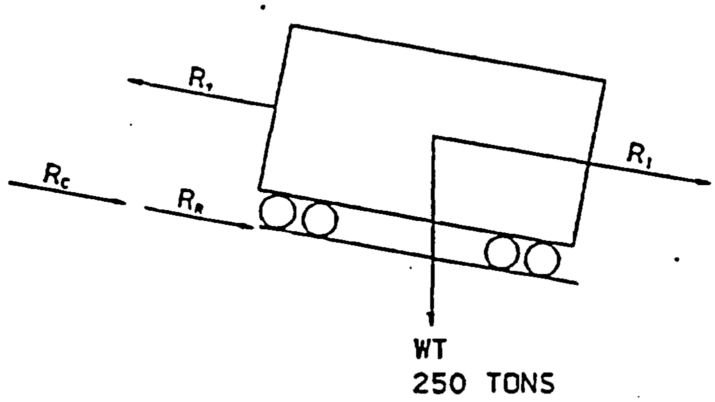
From Attachment XII, Section 4068, Pg. 2
HP = 15/ton for locomotives over 15 tons travelling at 10 miles per hour.

∴ For a locomotive speed of 5 mph, HP = 45 tons x 7.5/ ton = 337.5

3.5 Recommendation

Use a two motor, 45-ton locomotive with motors rated at 170 HP each.
Total rated HP = 340 for each locomotive.





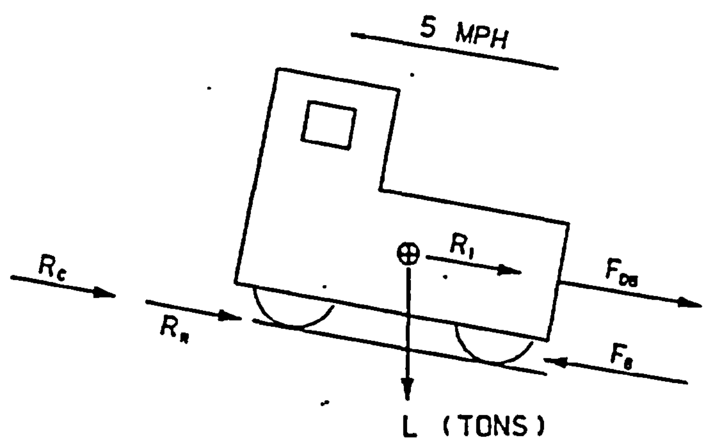
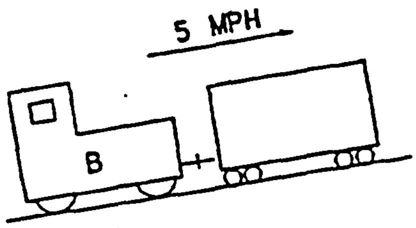
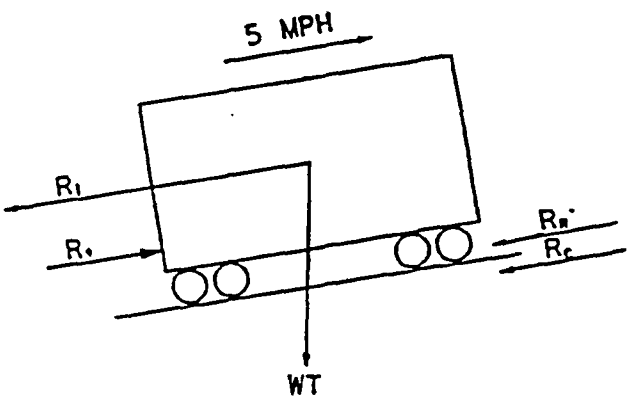


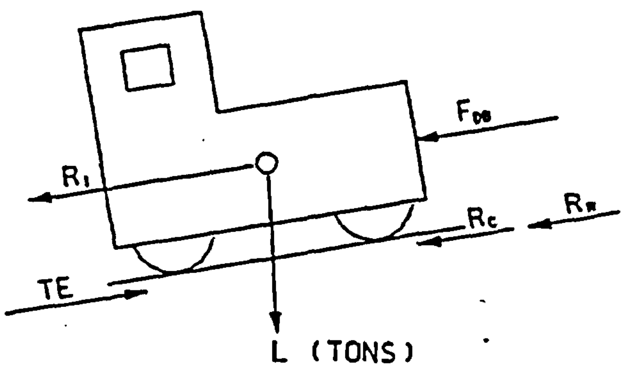
FIGURE 1
TRANSPORT LOCOMOTIVE
CONDITON NO. A



OPERATING CONDITION



TRANSPORTER



LOCO B

FIGURE 2
TRANSPORT LOCOMOTIVE
CONDITION NO.B

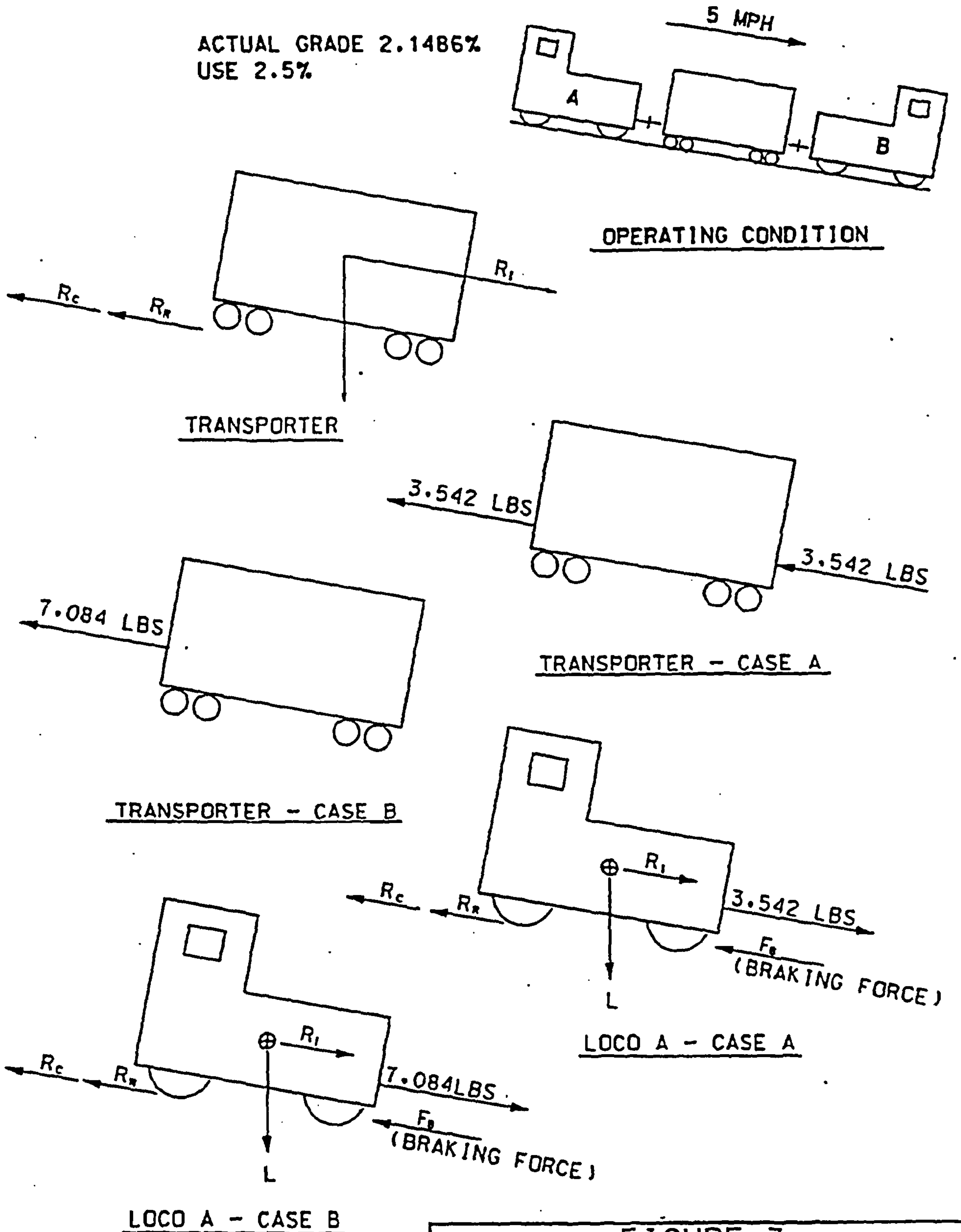


FIGURE 3
TRANSPORT LOCOMOTIVE
CONDITION NO.C

goodman

GOODMAN ELECTRIC LOCOMOTIVES FOR COAL MINING, METAL MINING AND TUNNELING

COAL MINING

	BATTERY POWERED				COMBINATION: BATTERY/TROLLEY	
Operating Weight*	5-ton	10-ton	14-ton	15-ton	10-ton	15-ton
Type	Mule	7508	Trackmaster	154	7588	1738
T.E. at 25% ADH., lbs.	2,500	5,000	7,000	5,000	5,000	7,500
Voltage	96	96	128	128	250	250
Number of Motors	1	2	2	2	2	2
1 Hr. HP Each Motor**	18	25	41	50	30	50
T.E. at 1 Hr. HP, lbs.	1,450	3,080	4,900	5,300	3,450	6,300
Speed at 1 Hr. HP, MPH**	4.7	6.0	5.8	7.0	6.4	5.8
Minimum Gauge, inches	42	36	42	36	36	36
Minimum Curve Radius, ft.	18	35	36	46	35	48
Wheelbase, inches	60	89	90	100	89	100
Wheel Diameter, inches	16	26	28	30	28	30
Height Over Covers, in.	32	32	29	38	32	38
Overall Width, inches	64	70	84	75	70	75
Length w/o Couplers, in.	160	223	231	256	223	256

*Operating weight will vary according to weight of battery used
**Horsepower and speed will vary according to actual voltage being furnished by the battery.

	TROLLEY POWERED						
Operating Weight*	8-ton	11-ton	11-ton	11-ton	15-ton	15-20 ton	20-25 ton
Type	7588	Trimotr	1738	184	Trimotr	281	282
T.E. at 25% ADH., lbs.	4,000	5,500	5,500	5,500	7,500	7,500-10,000	10,000-12,500
Voltage	250	250	250	250	250	250/500	250
Number of Motors	2	2	2	2	2	2	2
1 Hr. HP Each Motor	30	41	50	60	65	100	170
T.E. at 1 Hr. HP, lbs.	3,450	4,800	6,300	6,000	7,550	7,530	11,840
Speed at 1 Hr. HP, MPH**	6.4	5.8	5.8	7.5	5.9	9.7	10.5
Minimum Gauge, inches	36	42	36	42	42	36	42
Minimum Curve Radius, ft.	31	36	31	34	36	48	55
Wheelbase, inches	78	90	66	84	90	100	110
Wheel Diameter, inches	28	26	30	26	26	30	36
Height Over Covers, in.	32	29	36	28	29	42	44
Overall Width, inches	64	84	60 or 66	74	84	68 or 70	67 or 71
Length w/o Couplers, in.	223	231	167	228	231	261	282

*Operating weight is nominal and will vary according to the addition of optional equipment and other customer requests.
**Horsepower and speed will vary according to actual voltage supplied by the trolley system.

TUNNELING

	BATTERY POWERED									
Operating Weight*	2-ton	4-ton	6-ton	8-ton	10-ton	15-ton	15-ton	20-ton	30-ton	30-ton
Type	Trammer	Titan	Mancha	Mancha	750	158	154	1368	221	281
T.E. at 25% ADH., lbs.	1,000	2,000	3,000	4,000	5,000	7,500	7,500	10,000	15,000	15,000
Voltage	48	80	80	80	80	240	128	240	288	240
Number of Motors	1	1	1	2	2	2	2	2	2	2
1 Hr. HP Each Motor**	5.5	14	20	20	19	40	50	75	125***	125***
T.E. at 1 Hr. HP, lbs.	475	1,180	1,300	2,600	2,740	4,890	5,300	8,000	14,000	10,850
Speed at 1 Hr. HP, MPH**	3.7	4.0	4.9	4.9	5.1	6.0	7.0	6.9	6.6	8.5
Minimum Gauge, inches	18	18	18	18	24	24	33	36	36	36
Minimum Curve Radius, ft.	7	9	14	14	22	28	28	31	47	47
Wheelbase, inches	24	30	37.5	37.5	54	60	60	66	100	100
Wheel Diameter, inches	14	16	21	21	26	30	30	33	32	30
Height Over Battery, in.	44	43	55	55	52	60	63	66	65	64
Overall Width, inches	33 to 43	43	41 or 53	41 or 53	42 or 57	54 or 60	59	84	85	85
Length w/o Couplers, in.	71	122	144	144	153	178	172	186	261	261

*Operating weight will vary according to weight of battery used.
**Horsepower and speed will vary according to actual voltage being supplied by the battery.
***Motor rating is with forced ventilation.

Figure 4: Wheelbase Dimensions

ATTACHMENT VIII

TRANSPORTER/GANTRY CARRIER TRUCK ARRANGEMENT AND WEIGHT ANALYSIS

Note:

A requirement for this analysis is that quantities be represented in Metric units. Quantities and values derived in the main body of this analysis are presented in this manner. In the case of values carried into the main body of the analysis from the attachments, however, the information used as source material (such as vendor equipment data or standard structural steel members) are available typically only in English units. Because of this, generally all calculations and derivations are performed in English units within the attachments, with the final results converted to Metric units in the main body of the analysis (Sections 7 and 8). In such cases the value is represented in the main body of the analysis first in Metric units followed in parentheses ( ) by the corresponding English units.

TITLE: TRANSPORTER/GANTRY CARRIER TRUCK ARRANGEMENT AND WEIGHT ANALYSIS

1.0 PURPOSE

This calculation will provide the basis for preliminary evaluation of the weight of the 125-ton-capacity trucks to be used on the WP transporter and gantry carrier.

2.0 SOLUTION

2.1 Truck Weight

From Ref. 5.39:
Various truck components identified in Ref 5.39 are listed along with their weights. Quantities shown are the number required for a complete rail car. Additionally, for several components, different grades are listed in Ref 5.39. Therefore, not all of the components in Ref. 5.39 are required to provide a complete truck. The following components are required:

Item No.*	Description	Quantity	Weight Each (lb)	Total Weight (lb)
1.	Truck bolster	1	1,460	1,460
8.	Frame assembly	2	1,030	2,060
9.	Center plate vertical wear liner	1	15	15
10.	Center plate horizontal wear plate	1	5	5
12.	Brake beam wear plate	2	3	6
14.	Wheels, cast steel	4	830	3,320
23.	Axles	2	1,415	2,830
27.	Bearings	4	135	540
29.	Bearing adaptors	4	55	220
			Total weight	10,456 lbs

*Item numbers refer to item numbers in Ref 5.39.

Round up truck weight = 11,000 lb or 4989 kg.

From Reference 5.39, Freight Car Equipment Weights:

AILCAR SPECIALTY LISTING
125-TON EQUIPMENT

NO.	DESCRIPTION	SUPPLIER	QTY	WT. EACH
	TRUCKS:			
1.	Truck Bolster Grade-B-Steel	Buckeye	4	1460
2.	Center Plate Bowl, 16 In.			
3.	Truck Bolster Grade-C-Steel	Buckeye	4	1460
4.	Center Plate Bowl, 16 In.			
5.	Bolster Cup, 16-1/2 O.D.	Astralloy	24	
6.	Side Frame Grade-B-Steel	Buckeye	8	886
7.	Side Frame Grade-C-Steel	Buckeye	8	886
8.	Frame Assembly	Buckeye	8	1030
9.	Cen.Pl.Vert.Wear Liner	Buckeye	2	15
10.	Cen.Pl.Horz.Wear Pl.	Buckeye	2	5
11.	16" Cup Wearliner, F4638	Astralloy	4	
12.	Brake Beam Wear Plate	Buffalo	16	3
13.	Bolster Assembly	Buckeye	4	1740
14.	Wheels Cast Steel	Grif.Std.Stl.	8	
15.	CB-38 Class C			830
16.	CC-38 Class C	Std. Steel		925
17.				
18.	Wheels Wrought Steel	Grif.Std.Stl.		
19.	B-38, Class C	--		830
20.	C-38, Class C			925
21.				
22.				
23.	Axles 7 x 12	Std. Stl.		
24.	Class G Untreated			1415
25.	Class G Heat Treated		8	1415
26.				
27.	Roller Bearings 7 In. x 12 In.	Timken	16	135
28.				
29.	Bearing Adaptors 7 In. x 12 In.	Timken	16	55
30.	Hardened Thrust Shoulders			55

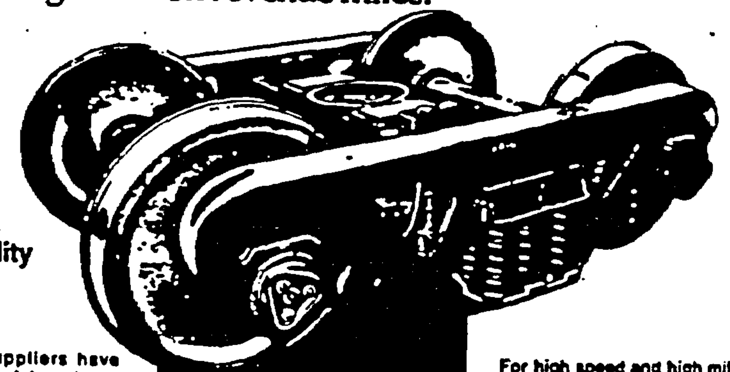
KASGRO
RAIL CORP

PRESIDENT
Gabe M. Kassab

320 East Cherry St.
New Castle, PA 16102
412-656-8061
412-656-7638 FAX

To: Mr. Chris Weddle

The BARBER S-2-HD Heavy Duty Truck
Featuring C-PEP Center Plate Extension Pads
More than 150,000 cars equipped with C-PEP now in service... some exceeding 1 million revenue miles!



Now approved for A.A.R. specification M-965-81 — special devices to control stability of freight cars

The railroads and suppliers have opened new frontiers of knowledge about the American Railroads with highly sophisticated monitoring of track and freight car performance over a wide range of operating conditions.

Among the first to put this new information to work, and to apply it through computer models for practical, workable hardware is the Standard Car Truck Company. The results are solutions to typical train dynamic problems—such as 100-ton loaded cars leaving the rails at low speeds, bouncing at high speeds, and empties hunting.

An improved Barber S-2-C® truck is now available as a "Heavy Duty" model, or S-2-HD. A stronger new generation truck developed from computer simulations and field validated. Engineered with Barber load-sensitive variable damping, the S-2-HD is the contemporary expression of our "Total Design Concept."

Above: The improved heavy-duty S-2-HD Truck with optimized Barber load-sensitive variable damping — computer refined and translated into ton/mile savings for you by the men from Barber.

Below: Typical of the thorough evaluations undertaken is the computer-generated drawing of the finite element model used for stress analysis of the truck bolster. Several new patentable features evolved from this research program.

For high speed and high mileage service, the S-2-HD Truck suspension incorporates substantially increased damping for both light and loaded cars.

Other important design improvements are in the friction casting, bolster end, and side frame construction. These, coupled with a replaceable bolster pocket wear plate and C-PEP®, the center plate extension pads, comprise the finest heavy duty truck suspension system available.

This long spring travel S-2-HD Truck provides a full load-carrying capacity with standard AAR load coils and no supplemental damping devices in the spring group. These changes are typical of Standard Car Truck Company's dynamic modeling, structural finite element analysis and service performance testing. Taken together, they have resulted in the S-2-HD Truck for improved car utilization.

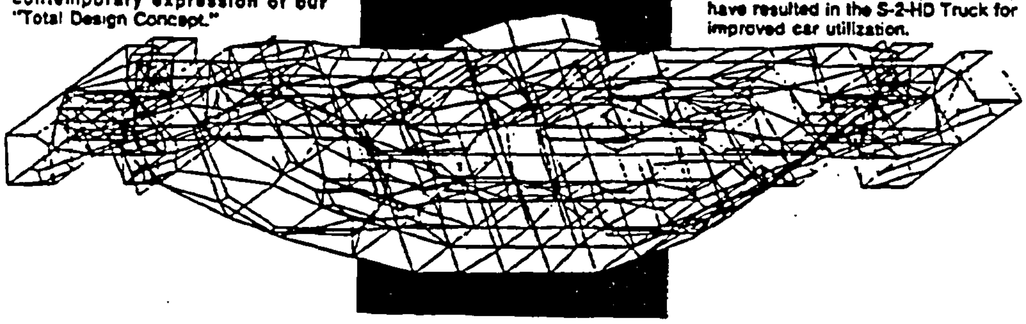
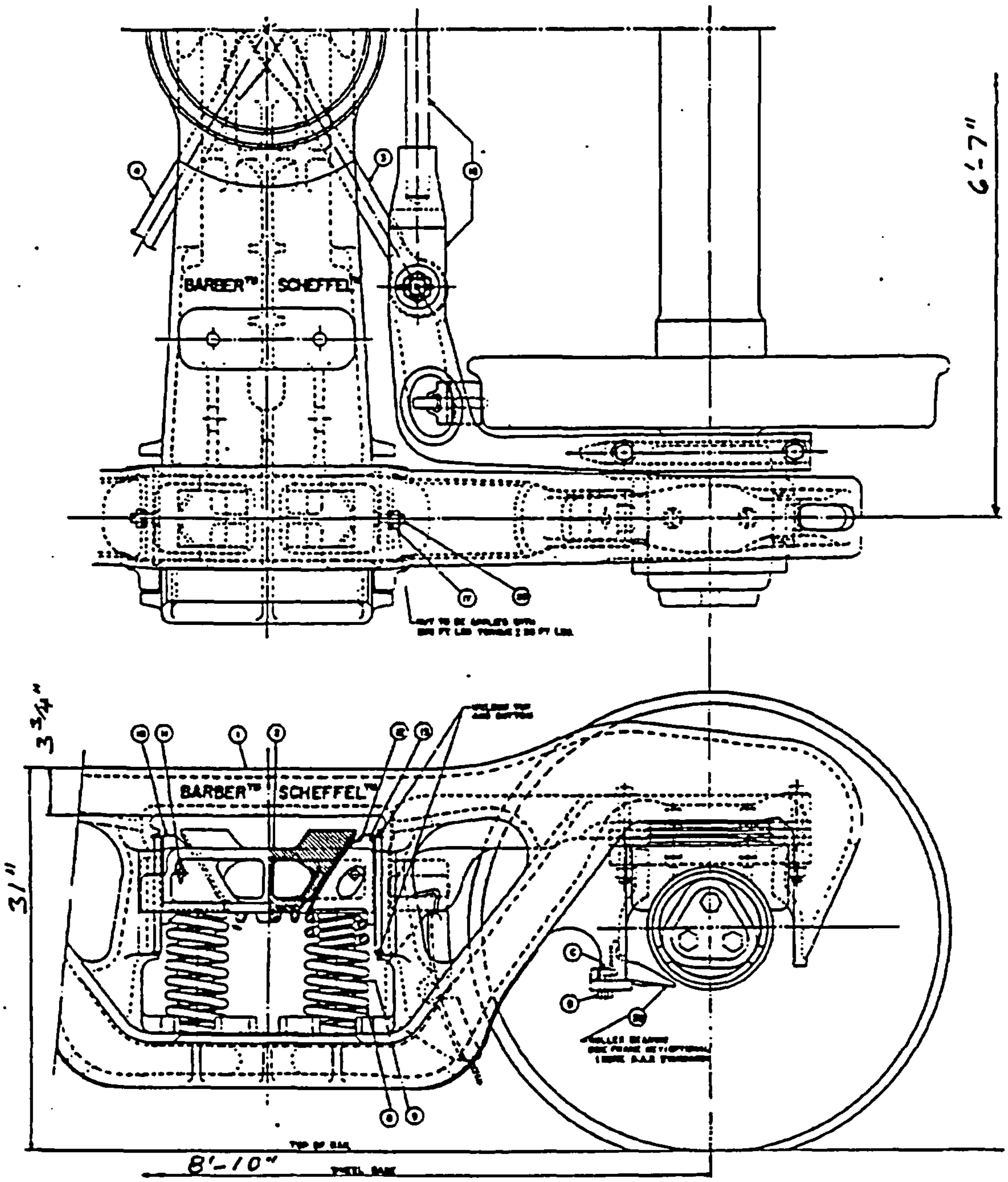


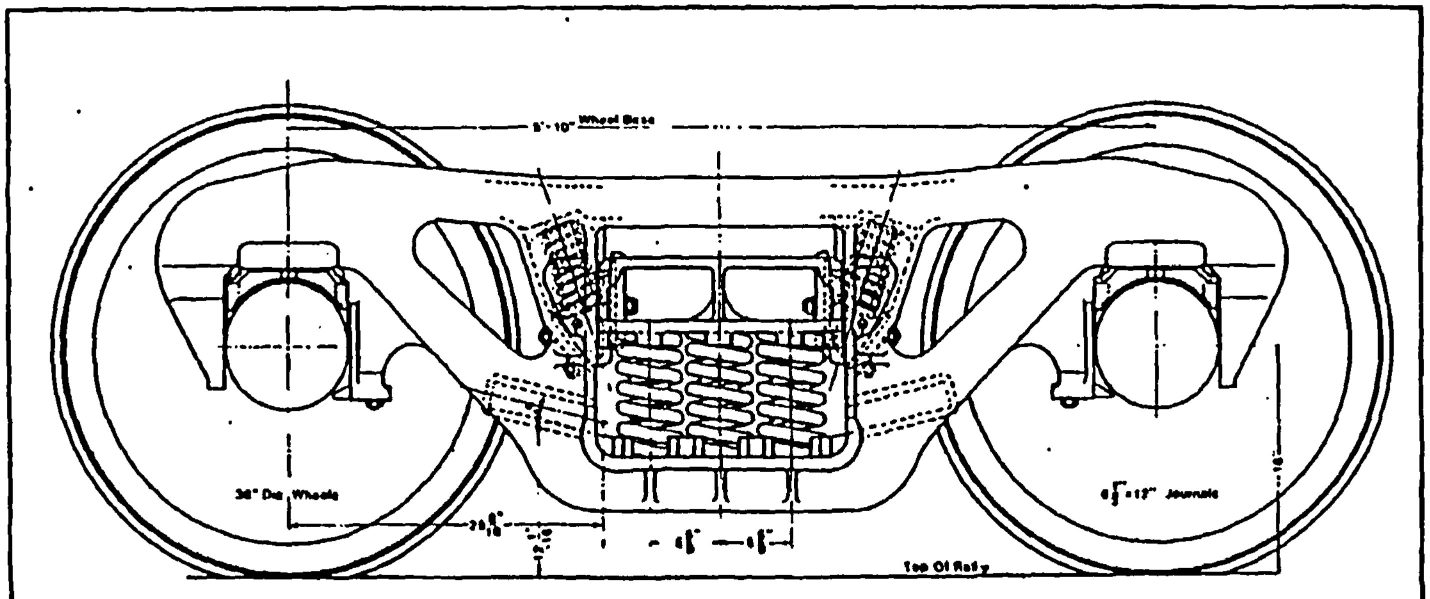
Figure 1: Reference 4.4.8 Page 492



BARBER®-SCHEFFEL® High Speed 100-ton Radial Truck by Standard Car Truck Co.

Figure 2: Reference 4.4.8 Page 495

National Super C-1 Wedgelock Truck for heavy-haul, high-mileage freight cars



NATIONAL SUPER C-1 WEDGELOCK TRUCK

Figure 3: Reference 4.4.8 Page 496

ATTACHMENT IX

TRANSPORTER COUPLER VENDOR INFORMATION AND WEIGHT

TRANSPORTER COUPLER VENDOR INFORMATION AND WEIGHT

1.0 PURPOSE

This attachment provides a vendor drawing of the selected Willison coupler and the estimated weight to be used in the structural analysis of the transporter (Attachment I).

2.0 INFORMATION

2.1 Coupler Weight

From Ref. (5.43), Appendix B:

The rounded-off coupler weight is 79.4 kg (175 lbs).

3.0 APPENDIX

INDEX

Item	Description	Sheets
A.	Fax from NACO Technologies to Dave Hamann from Mark Benigas 5/2/97 with vendor drawing of Willison automatic coupler No. 35958 (Ref. 5.42)	3
B.	Telecon from National Castings Inc. to Dave Hamann from Mark Benigas (5/2/97) with coupler selection and component weights (Ref. 5.43)	2

APPENDIX A

FAX FROM NACO TECHNOLOGIES

MAY 02 '97 16:17 FR NACO TECHNOLOGIES 630 792 2015 TO 912083866658 P.01/02

NACO Technologies 335 Eisenhower Lane So. Lombard, Illinois 60148

FAX

Date: 5/2/97
Number of pages including cover sheet: 2

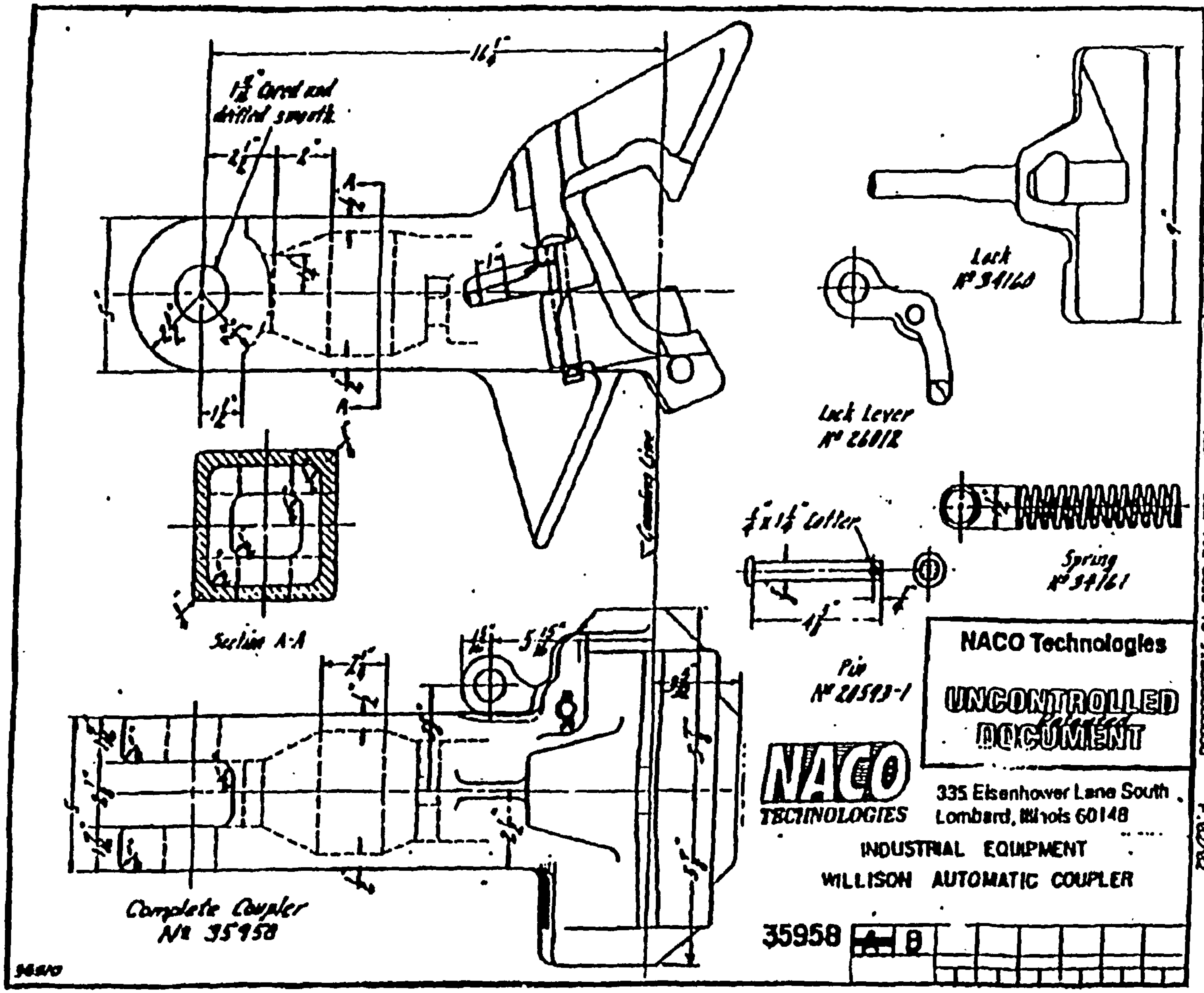
To:	
	Mr. Dave Hamann
	@ MK
Phone:	
Fax phone:	208 / 386-6658
Reference:	Print Request
CC:	

From:	
	Gloria's Desk
FOR:	MARK BENIGAS
	MELROSE PK. SALES
	708 / 344-4675
Phone:	630 / 792-2010 Ext. 231
Fax phone:	630 / 792-2015

REMARKS: ☐ Urgent ☒ For your review ☐ Reply ASAP ☐ Please comment

Print Requested: #15958 - Industrial Equipment Willison Automatic Coupler

[signature]


Complete Coupler
No 35958
Section A-A
Lock
No 34160
Lock Lever
No 26012
Spring
No 34161
Pin
No 28593-1
NACO Technologies
UNCONTROLLED DOCUMENT
NACO TECHNOLOGIES
335 Eisenhower Lane South
Lombard, Illinois 60148
INDUSTRIAL EQUIPMENT
WILLISON AUTOMATIC COUPLER
35958

APPENDIX B

TELECON FROM NATIONAL CASTINGS

MORRISON-KNUDSEN ENGINEERS, INC.
A MORRISON KNUDSEN COMPANY

TELECON REPORT

TO:	Dave Hammm [initials]	DATE:	5/2/97
LOCATION:	BHO	TIME:	
FROM:	Mark Benigas National Casting Inc.	CONTRACT OR W.O. NO:	3969
LOCATION:	Melrose Park, Illinois	DISTRIBUTION:	File
SUBJECT:	Willison Automatic Coupler For WP Transporter		

Per the information I had previously given Mark and Ray Bednak at NACO Technologies on the coupler application for the transporter and transport locomotive, they selected the following coupler:

Willison Automatic Coupler #35958 for Industrial and Mining Application.

Coupler	= 125 lbs.
Tail Bolt	= 31 lbs.
	156 lbs.
10% S.F.	= 15 lbs.
	171 lbs.

Round up to 175 lbs.

DH/kaw

ATTACHMENT X

SERAPID RIGID CHAIN INFORMATION

1409 Allen Drive, Suite E
Troy, MI 48083

SERAPID USA, Inc.
Rigid Chain Engineering

Tel: (810) 588-8530
Fax: (810) 588-8963

November 30tn, 1995

Dave Hamann
M.K.E.S CP-2
720 Park Blvd.
Boise, ID 83729

Dear Dave Hamann:

Thank you for your interest in our product.

The Serapid rigid chain can be used to load translation applications, and can be seen in the enclosed information package.

This very unique push/pull mechanism can:

- save engineering time: 4 clearance holes for mounting, standard product that can be used over;
- save installation time;
- save machining time;
- save on maintainance: fully mechanical system, no lubrication
- save space.

The Serapid rigid chain is especially suited for long stroke and high load applications.

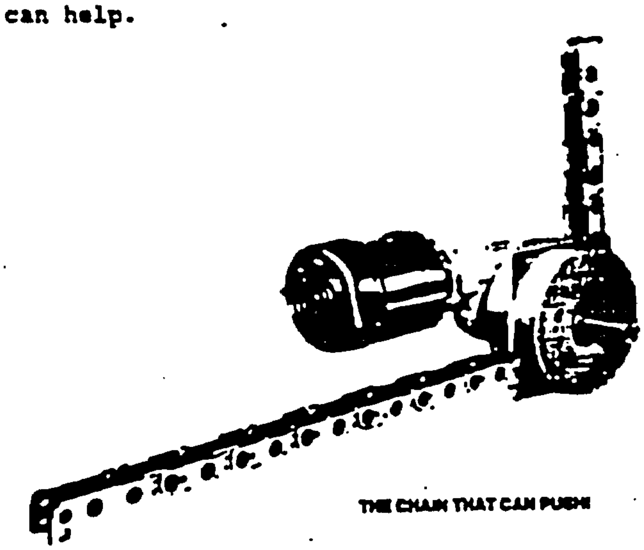
Please let us know how we can help.

Best Regards,

Said Lounis
Sales Manager

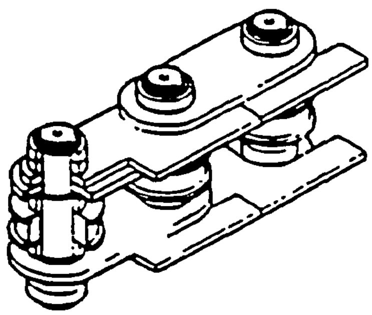
Encl

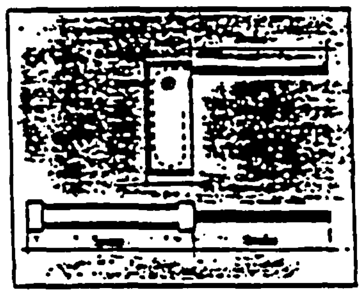
SL:km
8505

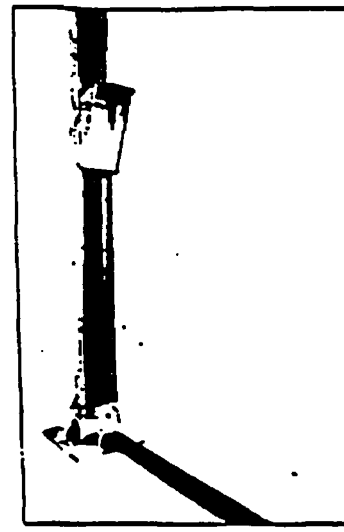


THE CHAIN THAT CAN PUSH!

The product



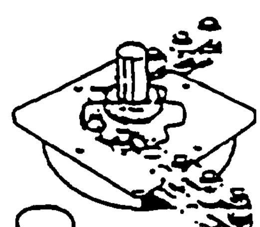


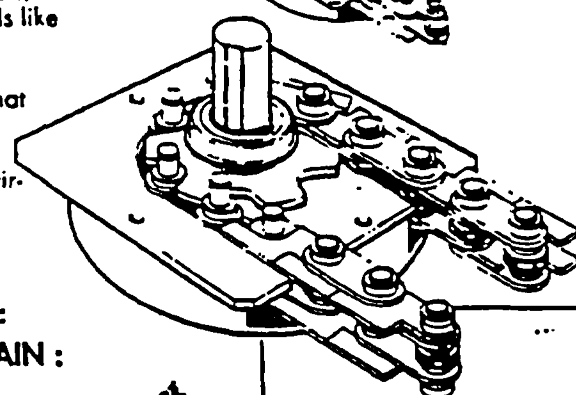


THE SERAPID CHAIN : A JACK WITH A FLEXIBLE ROD

The Serapid system is quite original, in the sense that it stays very compact while allowing to pull or push a load horizontally, vertically or at an angle. Capable of long travel and positioning accuracy, regardless of weight. When retracted, the Serapid chain coils like a regular chain in minimum space.
These features are what makes the Serapid chain exceptional, with applications in virtually all industries.

The Serapid chain pushes and pulls loads weighing from a few pounds to hundreds of tons.
Operating temperatures can vary between 0 and several hundred degrees Fahrenheit.

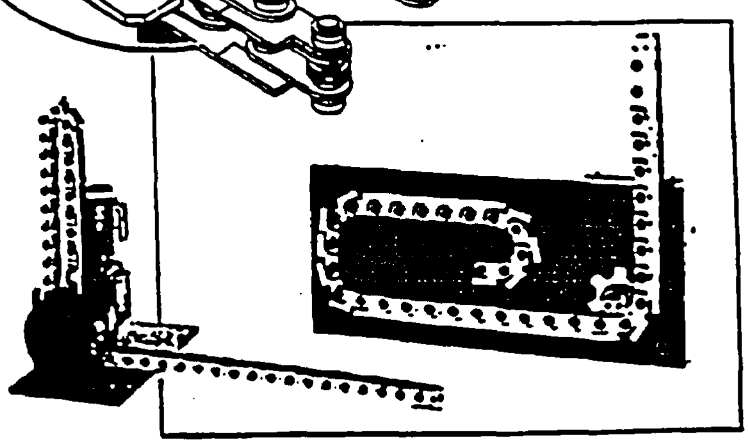




Very stable mechanically

ADVANTAGES OF THE SERAPID CHAIN :

- Power of a jack.
- Flexibility of a chain.
- Limited space use.
- Accuracy and repeatability.
- Long strokes.
- Low maintenance.
- Low cost.
- Easy implementation and/or installation.
- Safety.



An industrial innovation has made its appearance, bringing revolutionary solutions to the many problems of conveying and positioning loads in all cases where limited floor space for handling equipment, maximum manoeuvrability, precise positioning and non-shifting of loads are required.

Basilically, it is a question of a power chain having the original characteristic of becoming as rigid when thrusting as when pulling, because of the special design of its links.

By virtue of their shape, these links have shoulders which rest against each other as and when they emerge from the drive housing.

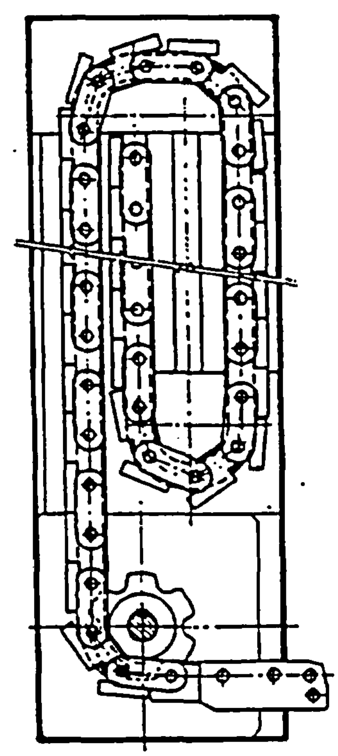
Contrary to ordinary chains and subject to observing certain conditions of use, the chain can then transmit a thrust motion as well as a tractive force.

The chain drive itself is obtained by means of the rotation of twin pinions integrally keyed onto the shaft which the housing supports.

For this purpose, two rollers bearing on each link axis engage the teeth of the two pinions.

The thrust motion is absorbed by the drive housing by means of a simple device. The axis of each link carries, between the two drive rollers, a central roller which revolves onto a steel plate fitted inside the housing. In this way, the links remain set inside the housing, between the teeth of the pinions and the steel plate which absorbs the thrust motion.

We have thus produced a device operating with a thrust motion and a tractive force by preserving, at the same time, the articulation necessary for the chain drive and its changes of direction.



Magazine illustrating double recoil feature

SERAPID
USA, Inc.
Rigid Chain Engineering

MOST FREQUENTLY ASKED QUESTIONS ON SERAPID CHAIN

Please describe the chain:

There are two main components in a Serapid Rigid chain system:
- The chain itself.
- The drive housing.

The chain is composed of:
- Side plates, which are fine blanked for precision tolerance. The side plates interlock with each other and have mating surfaces which are flat, and which provide the push force. These mating surfaces are called heels.
- Shaft pins, and various rollers which are CNC machined, heat treated, then ground.

The drive housing is composed of:
- A drive shaft on sealed bearings.
- Drive sprockets, 1 to 6 depending on chain size.
- Reaction plates made out of high grade steel, precision machined.
- Cast housing pair for assembly of the above.

How does it work ?

Imagine holding a bar of steel in your hand. That is how the Serapid chain feels with the heels (butting surface of links) at the bottom. Its own weight keeps it straight and rigid. The chain stays rigid under its own weight, or the applied load, because of the locking moment that locks the links together.

Now pick up the free end and roll it back. It coils like a conventional chain. This is what happens through the drive housing. The chain is coiled 90 degrees or 180 degrees to store it out of the way.

As the chain pushes, it tends to walk away from the sprockets. This is why the chain is contained by steel reaction plates inside the drive housing. The reaction plates are designed to take up the reaction to the pushing force.

What are the two holes in the front link for ?

A pin is installed in each hole to attach the front of the chain to the load.
The pin which is in line with the rest of the chain pins is used for pulling the load.
The other pin, which is in line with the chain heels, is used for pushing.

SERAPID
USA, Inc.
[illegible] E Allen Drive • Troy, Michigan [illegible]
Tel: (810) [illegible] • Fax: (810) [illegible]

SERAPID
USA, Inc.
Rigid Chain Engineering

The purpose of the pushing pin is to ensure that the chain locks up during pushing, by ensuring that the reaction from the load comes back along the chain heels.

Does it need a guide ?

No. The chain is designed to operate without a guide, as long as the load locks up the chain. The chain is designed to excert a force along its heels when pushing, and along its pins when pulling. No side load on the chain is allowed.

A guide is only required if:

- The chain may be subject to side or excentric loads.
- The load is not centered on the chain, or is difficult to locate.
- The reaction force from the load (when pushing) is not along the chain heels.
- The travel is too long in relation to chain size. For exemple, for 10 feet of travel, using a guide may allow the use of a 40 pitch chain, whereas without a guide, a 60 pitch would be required.

Note: Guides when used do not have to run the length of the stroke. For exemple, for 10 feet of travel, 5 feet only could be guided at the begining of the stroke, or 5 sections of 1 foot long guides could be used at rendom along the stroke, etc...

How much can it push ?

1 daN = 2.2 lbs.

Nominal Push/pull forces range from 110 lbs for the size 25 (plastic) chain, to 10,000 lbs for the 901 chain.

The smallest steel chain is the size 40PS, with a nominal push/pull force of 1650 lbs.

What is meant by nominal is:

- 40" of stroke.
- Chain supported.
- Chain unguided.
- No shocks.
- Low speed (up to 10 feet/minute).
- Intermittent duty.
- Horizontal translation.

Longer strokes, higher speeds, shocks present, heavy use, etc... all reduce the nominal ratings.

All above rating parameters accumulate.

Please call the factory for application engineering assistance.

SERAPID
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1400 E Allen Drive • Troy, Michigan 48083
Tel: (810) [illegible]-8530 • Fax: (810) 589-[illegible]

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Rigid Chain Engineering

Some rules of thumb:
When the chain is guided, nominal capacity remains regardless of stroke.
In lifting applications, cut nominal capacity in half.
When the stroke is doubled, cut the chain capacity in half.
When the chain is unguided and unsupported, cut the capacity by a third.

How fast can I run this chain ?

There is no actual limit to the speed of the chain.
But like any mechanical system, speed means faster wear.
Some precautions can be taken to improve the wearability of the chain and components in high speed applications, like ramping the speed up and down to avoid shocks.
Please consult factory for application engineering assistance.

How do I loop this chain for storage ?

Please look at the first page of our specification sheets. Shown are typical storage configurations, or chain returns.
Simply, the chain must be stored where space is available.
The most common chain returns are # 2, 5, and 8. All these chain returns have a single loop storage, and are free. They offer the advantage of reduced storage space, and yet do not require a guided storage magazine. The end of the chain is simply attached back to the drive housing, and the chain self supports itself.
Please see page 12 of our spec. sheets for dimensions.
If the space available is still too small, then a multiple loop storage magazine is required.

Does the chain need lubrication ?

Usually not. An occasional oil spray helps keep the chain clean and free from rust.
In high speed and/or high cycles applications, it may be necessary to lubricate for heat dissipation and reducing friction between rolling/sliding components.

How do I select a chain ?

The main parameters of chain selection are:
- What is the force required to move your load.
- How far you want to move it.
- How fast you want to move it.

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Please check other factors involved in determining actual push force, as shown in "how much can it push" paragraph.
Please fill out our application questionnaire, or call the factory. Our Application Engineering Dept is always available to assist you.
Following above "rules of thumb" will also work, but please verify your selection with our Application Engineering Dept.

What torque do I need ?
Once the thrust (push/pull) force is determined, and the chain selected,
torque = (in.lbs): force (lbs) x sprocket pitch (in.) : 0.8 (drive efficiency).
In rolling load applications, please remember to include acceleration and decel. loads.

How do I calculate Serapid shaft speed (RPM) ?
Translation speed (feet/mn) x 60 : pitch perimeter (feet) = Shaft RPM.
Pitch perimeters:
Size 25 chain: 0.673 feet.
Size 40 chain: 0.821 feet.
Size 60 chain: 1.236 feet.
Size 90 chain: 1.854 feet.

What motor horsepower do I need ?
HP at the Serapid drive shaft = Torque (in-lbs) x Serapid shaft speed (RPM) : 63000.
To obtain input HP, the above figure must be devided by reducer/motor efficiency.

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RIGID PUSHING CHAINS

SPECIFICATION SHEET
PAGE 1

GENERAL

[illegible]

METHODS OF USING THE CHAIN

[illegible]

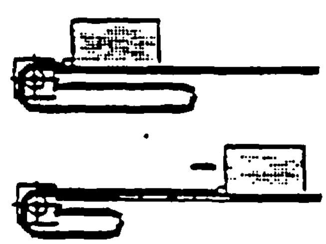
DIMENSIONS AND CAPACITIES OF THE CHAIN

[illegible]

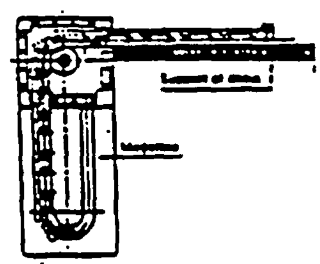
EXAMPLES OF USE

[illegible]

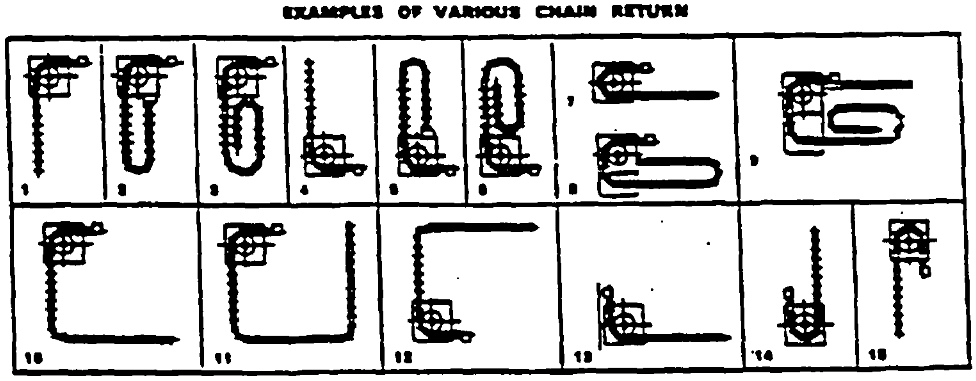
MOVEMENT OF A LOAD



EXAMPLE OF SPACE SAVING



EXAMPLES OF VARIOUS CHAIN RETURN



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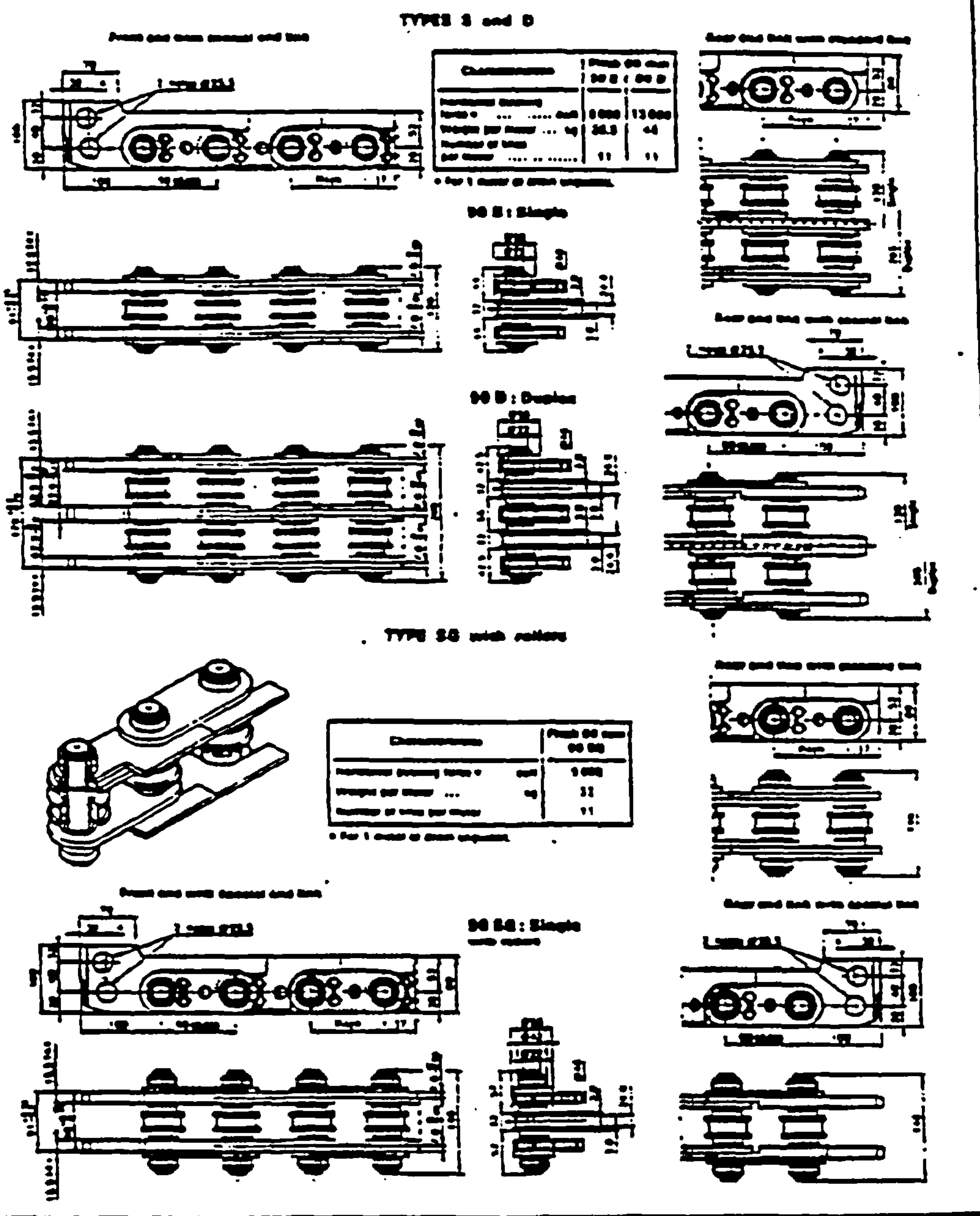
RIGID PUSHING CHAINS
PITCH 90 mm - TYPES S - D and SG

SPECIFICATION SHEET
PAGE 6



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Troy, Michigan 48083

SPECIFICATION SHEET
PAGE 9

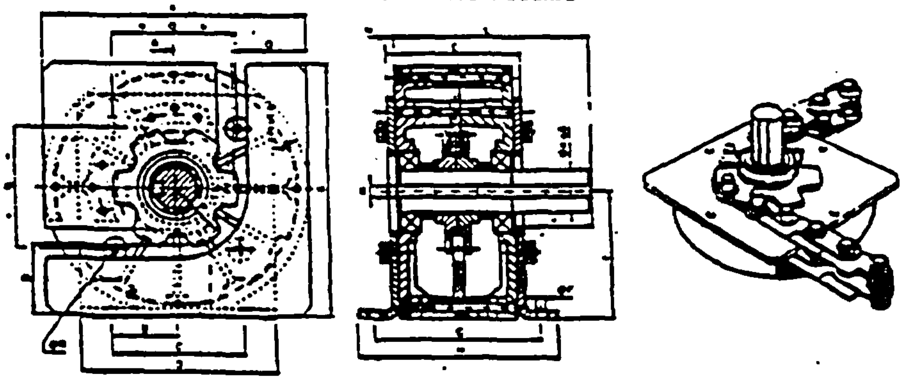
90° and 180° DRIVE HOUSINGS
for SINGLE CHAINS - PITCHES 40 - 60 and 90 mm

GENERAL

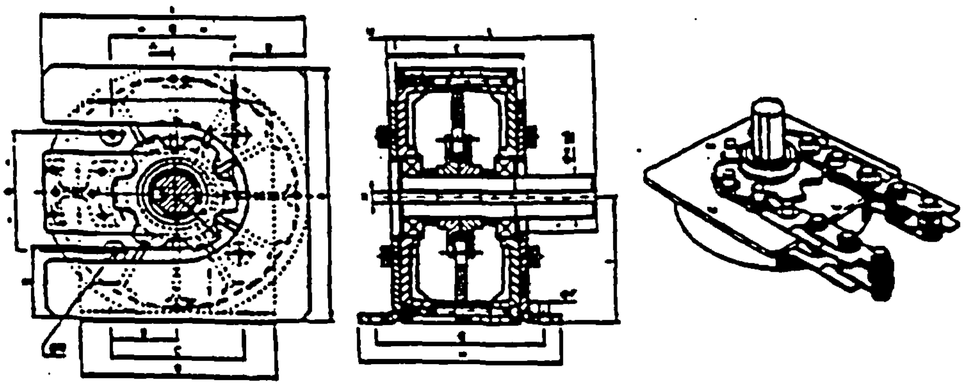
[illegible]		Pitch 40 mm	Pitch 60 mm	Pitch 90 mm
[illegible]	mm	240	360	540
[illegible]				
[illegible]	kg	13	35	120
[illegible]	kg	8.5	15	

[illegible]

90° DRIVE HOUSING



180° DRIVE HOUSING



DIMENSIONS

Type	Pitch	A	B	C	D	E	ØF	G	H	I	ØJ g6	L	M	N	O	P	R	ØS	V
Single	40	200	60	130	160	112	4 x 8.5	140	164	161	25	146	6	8 x 7	60	60	100	M8	64.5
Single	60	270	70	140	200	130	4 x 11	170	200	130	45	200	8	14 x 8	75	80	120	M10	82
Single	90	400	110	200	310	200	6 x 17	200	314	262	70	301	10	20 x 12	110	90	184	M16	147.5

[illegible]

Tel: (810) 588-8530
Fax: (810) 588-8983
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Troy, Michigan 48083

ATTACHMENT XI

GOODMAN LOCOMOTIVE INFORMATION

NOTE: A 45-ton locomotive will be similar to the 35-ton locomotive shown except that weight will be added and higher horsepower motors will be used. Only data sheets for the 35-ton locomotive were available.

SECTION 11D76-A, Page 2
July 1975

goodman MINING MACHINE SALES MANUAL
COAL MINING

35 TON-4 MOTOR TROLLEY LOCOMOTIVE

FOUR 201 MOTORS - 500HP BLOWN AT 250 OR 500 VOLTS - STANDARD ENCLOSURE

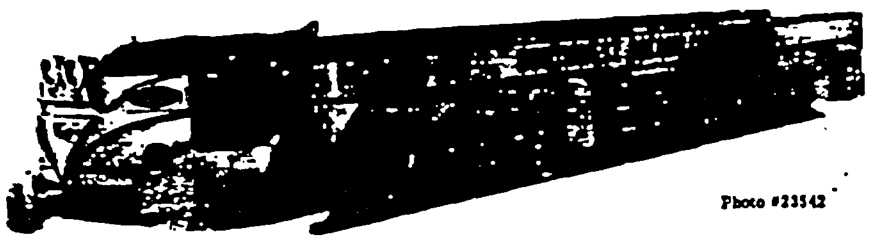


Photo #23542

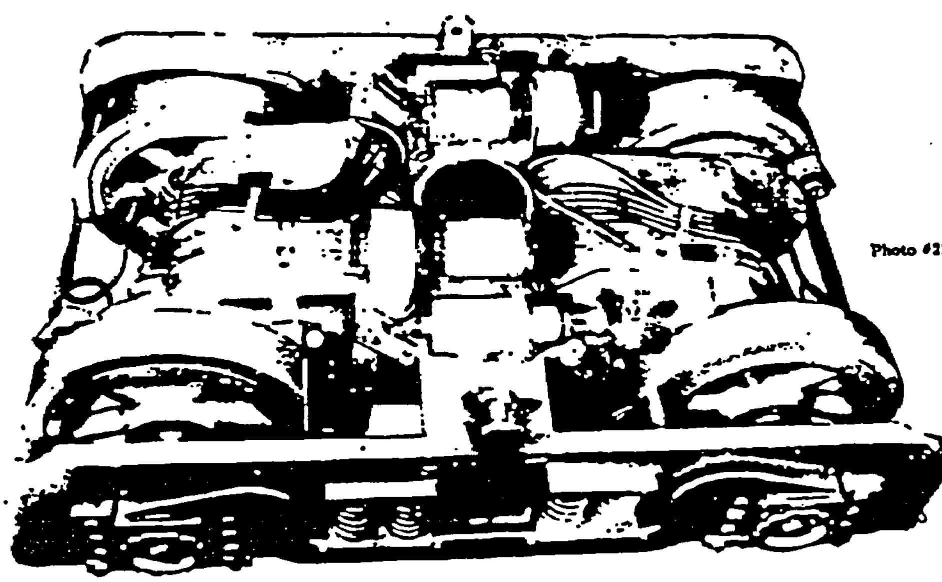


Photo #23024

TRUCK FOR 35 TON LOCOMOTIVE EQUIPPED WITH TWO GOODMAN 201-125HP MOTORS. THE BLOWERS FOR THE MOTORS ARE MOUNTED ON THE TRUCK, MAKING CONNECTIONS TO THE MOTORS MUCH SIMPLER THAN IF MOUNTED ON THE MAIN FRAME. THE CENTER BEARING, UPON WHICH THE MAIN FRAME RESTS IS 11" IN DIAMETER. SIDE BEARINGS PREVENT ROCKING OF THE LOCOMOTIVE AND HOLD THE MAIN FRAME KING PIN IN THE CENTER BEARING.

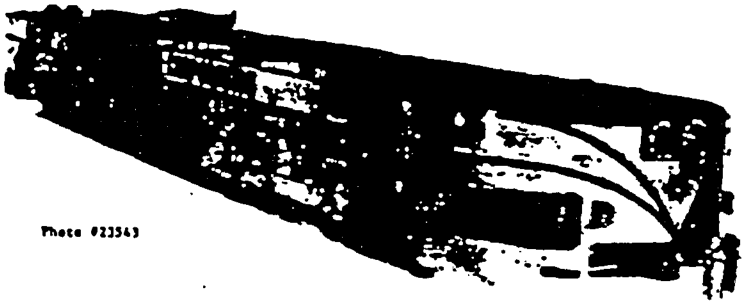


Photo #23343

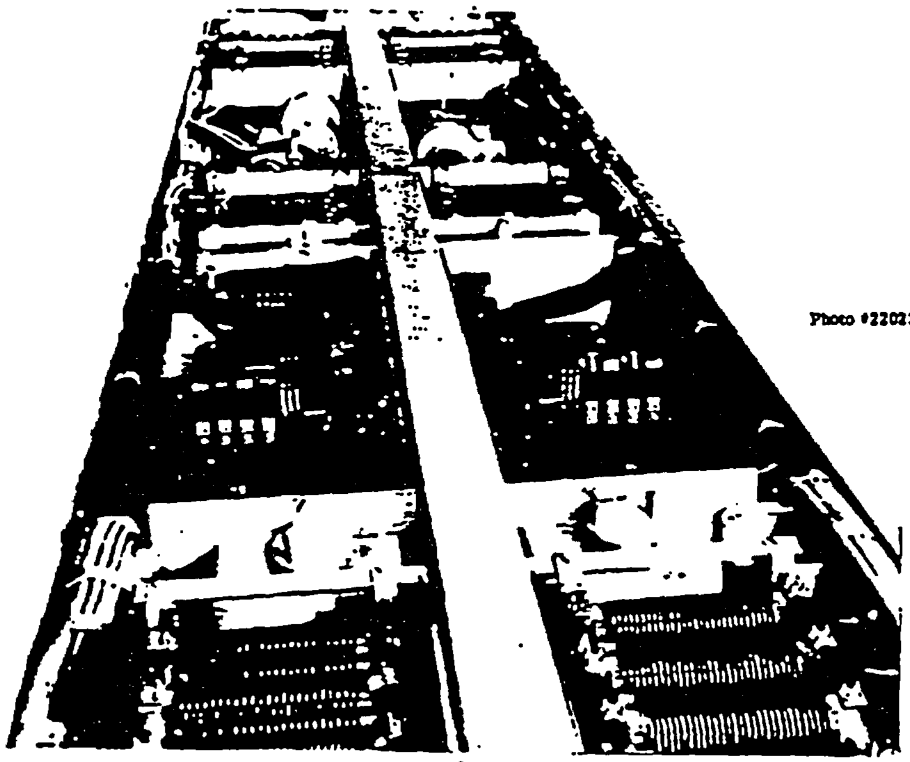
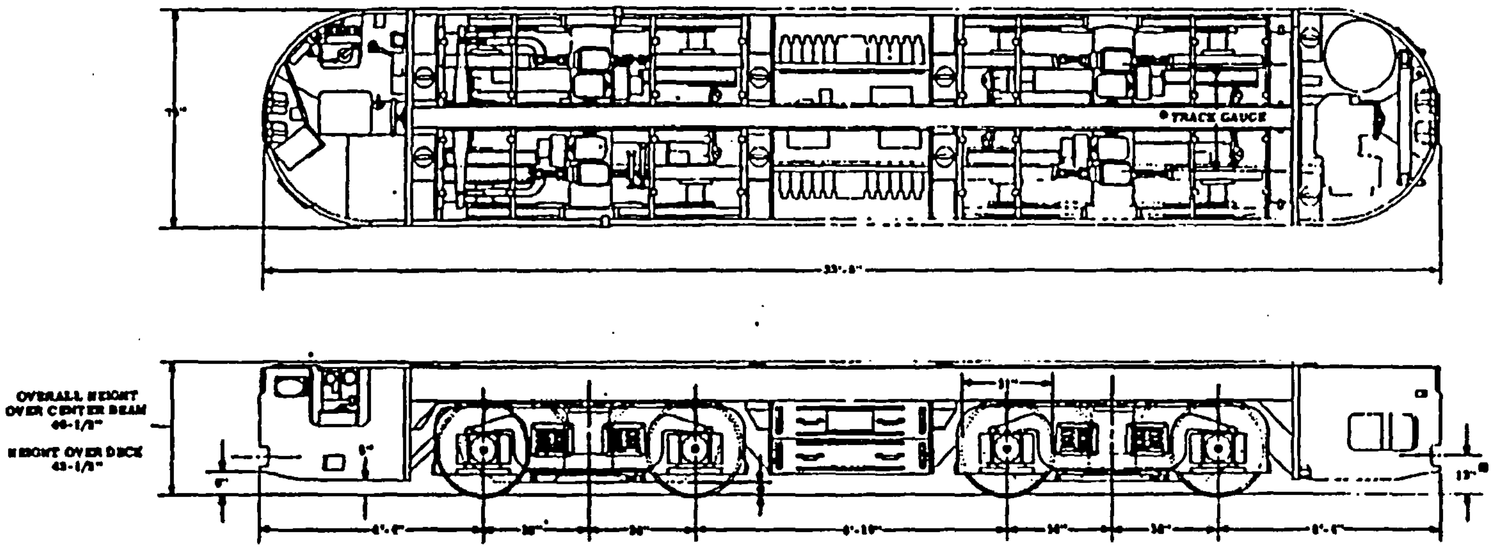


Photo #22023

TOP VIEW OF LOCOMOTIVE ILLUSTRATING DUAL CONTROL CIRCUIT. ONE SET, (REVERSER, CONTACTORS AND RESISTANCE) ON EACH SIDE OF THE LOCOMOTIVE CONTROLS TWO MOTORS. FOUR SAND BOXES ARE CLEARLY VISIBLE, HOWEVER, EIGHT SAND BOXES ARE PROVIDED.

35 TON-4 MOTOR TROLLEY LOCOMOTIVE

FOUR 201 MOTORS - 500HP BLOWN AT 250 OR 500 VOLTS - STANDARD ENCLOSURE



● The above frame is designed to accommodate any gauge from [illegible] to [illegible]. [illegible] gauge is shown.
● [illegible] clearance under motor and 3-1/2" clearance under gear case.
■ Distance from top of rail to center-line of coupler slot may be varied to suit customer requirements.

MINIMUM TURNING RADIUS - 50'

FRAME DESCRIPTION

The main frame is an all welded unit. It consists of a 7" x 12" solid center beam with 1-1/2" cross ties to 3" side plates. The curved end bumpers are of 1-1/2" steel plates. The main frame is equipped with 11" king pin bearings for engaging the trucks.

STANDARD EQUIPMENT

RESISTANCE: Continuous strip steel.
OVERLOADS: Magnetic overload relay (automatic reset).
CONTROL: Pneumatic, straight parallel, 10 running points, [illegible] dynamic braking points.
32 VOLT CONTROL & BATTERY: 32 Volt battery for providing power for the headlights and control circuit. The battery is charged automatically by the use of a solid state charger mounted on the locomotive.
HEADLIGHTS: Four 32 volt sealed beam headlights with fuse and switch, energized from 32 volt battery independent of trolley pole contact.
WHEELS: 31" Diameter steel tired wheels with cast steel wheel centers.
AXLE JOURNAL BOXES: Pedestal type with removable shims and frame guides.
AXLE JOURNAL BEARINGS: Anti-friction axle journal bearings - tapered roller.
MOTOR AXLE BEARINGS: 2 Roller Bearings
Commutator End - Tapered Roller - Double Row
Gear End - Single Row - Cylindrical
AIR BRAKES: 18 Cubic foot straight air brake system.
DYNAMIC BRAKES: Operates regardless of trolley pole position due to 32 volt control circuit.
SANDING EQUIPMENT: Eight sand boxes equipped with air sanders.
BLOWERS FOR MOTORS: Each motor force ventilated by a blower driven by a 1/4HP, 250 volt, compound wound, 3300 RPM, ball bearing motor.
WARNING SIGNAL: Air horn.
GAGE LIGHT: For illuminating instrument panel.
GROUND SHOE: Each motor equipped with a grounding shoe to eliminate currents from passing through the motor support bearings.

APPROVED MODIFICATIONS FOR CHASSIS
(Extra Charge Items)

1. Automatic couplers:
   - M.C.B., 1/2, 3/4 or standard size.
   - Willison, approximate 3/4 size
   - Ohio Brass, approximate 3/4 size.
   - Uncoupling or pin pulling device for above couplers.
2. Rubber under draft gear.
3. Deadman controller handle or foot switch.
4. Extra trolley pole and automatic transfer switch for use with two trolley poles.
5. Air operated trolley pole retriever.
6. Automotive type speedometer.
7. Ships bell, or electric horn.
8. Low air pressure alarm.
9. Ammeter in each pair of motors
10. Headlight dimmers.
11. Time delay to reduce arcing at trolley when reversing.

Supersedes Section 11070, pages 1 & 3 dated February 1973

ATTACHMENT XII

VENDOR DATA, VARIOUS

SECTION I

MIDWEST RAILS, TRACKWORK AND ACCESSORIES CATALOG, MIDWEST CORPORATION

MIDWEST RAILS, TRACKWORK AND ACCESSORIES

MIDWEST CORPORATION

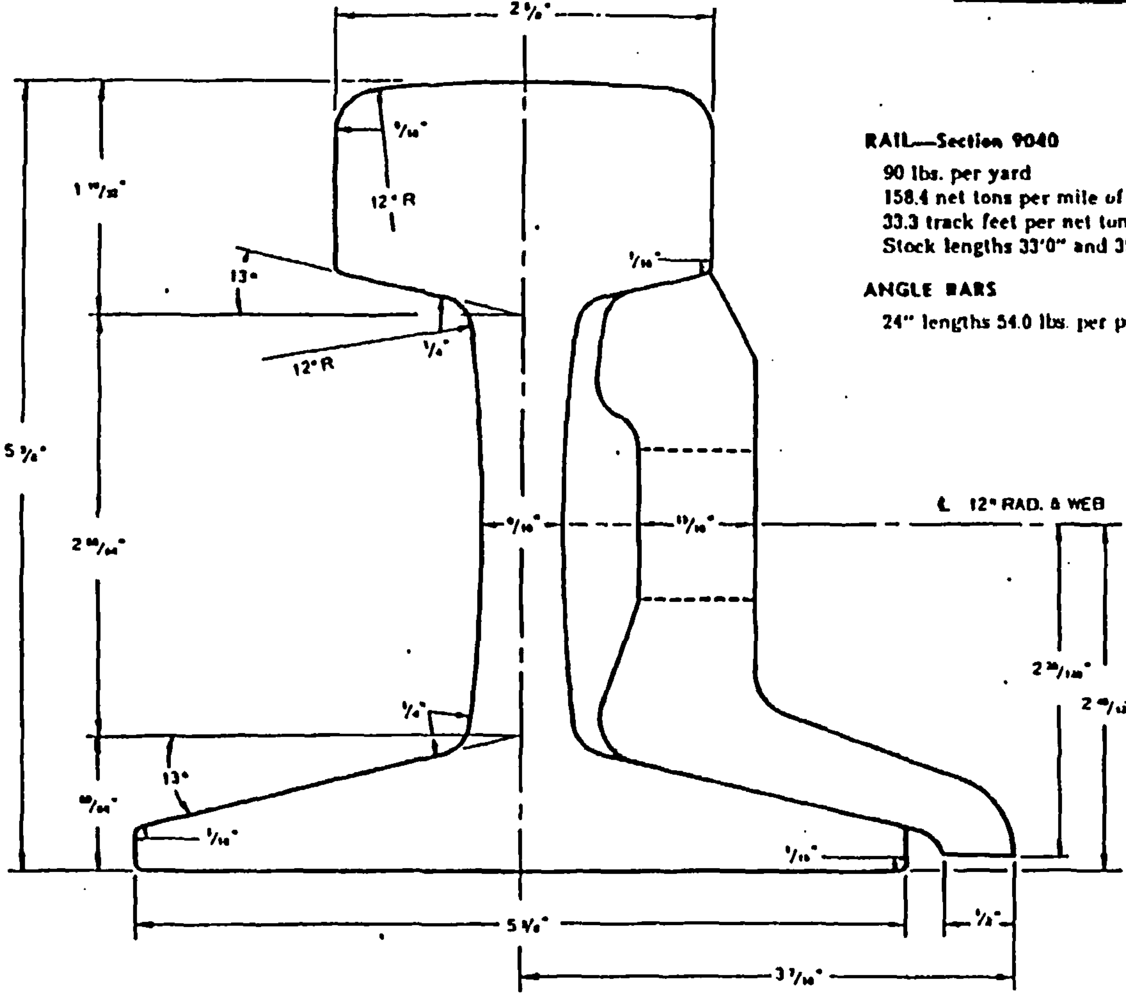
MIDWEST STEEL DIVISION

General Offices
610 Capitol Street
Charleston, W. Va. 25321
(304) 343-8874

Branches
1107 22nd Street
Granite City, Illinois 62040
(314) 241-8061

736 East Main Street
Pomeroy, Ohio 45769
(614) 992-3285

[redacted]



RAIL—Section 9040

90 lbs. per yard
158.4 net tons per mile of track
33.3 track feet per net ton
Stock lengths 33'0" and 39'0"

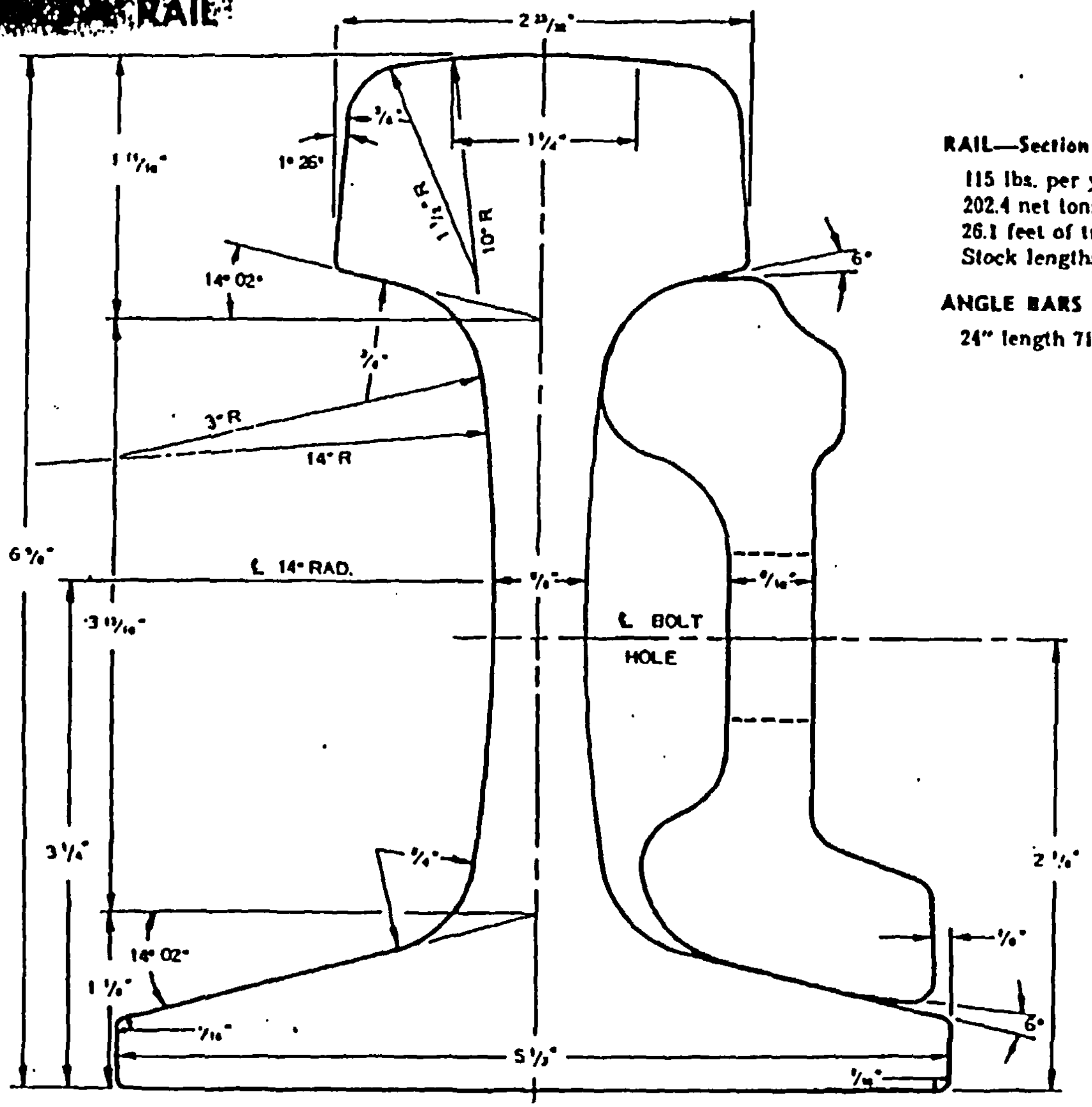
ANGLE BARS

24" lengths 54.0 lbs. per pair

MIDWEST STEEL DIVISION

MIDWEST CORPORATION

[redacted] RAIL



RAIL—Section 11525

115 lbs. per yard
202.4 net tons per mile of track
26.1 feet of track per net ton
Stock lengths 33'0" and 39'0"

ANGLE BARS

24" length 71.6 lbs. per pair

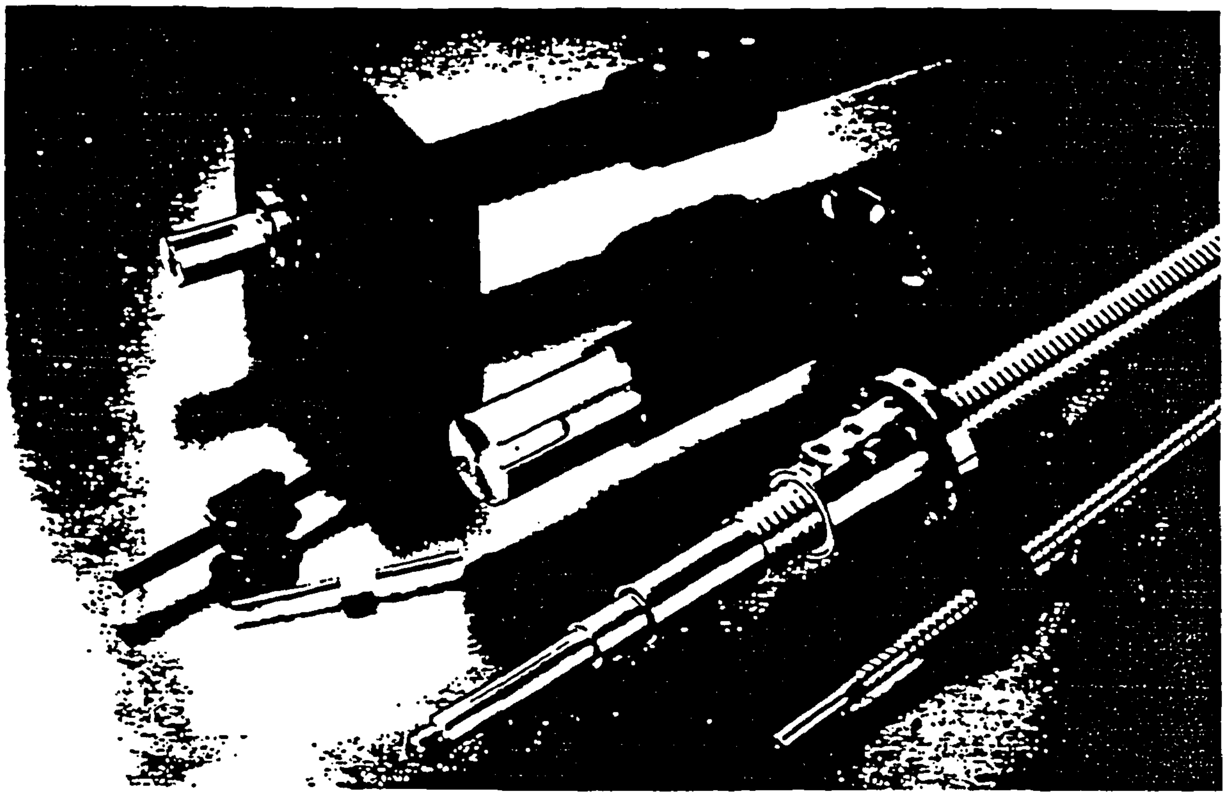
MIDWEST CORPORATION

SECTION 2

THOMPSON SAGINAW, ADVANCED LINEAR ACTUATOR GUIDE, CATALOG

Kaman Industrial
Technologies Corporation
1600 Commerce Avenue
Boise, Idaho 83701
(208) 342-6583 Fax (208) 345-7706
KAMAN

Advanced Linear Actuator Guide



The complete selection guide
for linear actuation components

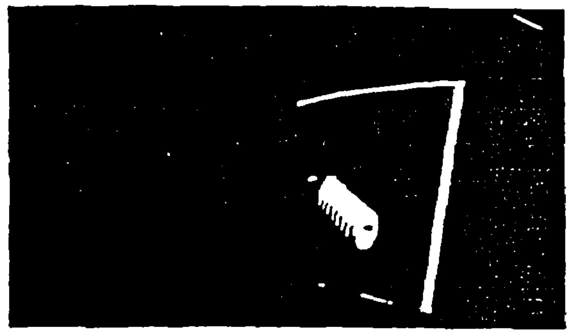
THOMSON SAGINAW
First in Linear Actuator Technology

ES

Engineering Support

Thomson Saginaw products are engineered to world class performance standards, so that you can specify them worldwide. This Engineering Section provides charts, formulas and technical information for:

- Load considerations for horizontal and vertical mounted applications
- Ball screw life
- Operating speeds
- Column stiffness



Design Formulas

These formulas allow you to calculate a number of important factors which govern the application of Thomson Saginaw ball screws.

1. Equivalent Operating Load

$$F = \sqrt[3]{q_1 (F_1)^3 + q_2 (F_2)^3 + q_3 (F_3)^3 + \dots q_n (F_n)^3}$$

q_n = Proportion of Stroke or Cycle at F_n
F_n = Increment of Load (lb_f)
F = Equivalent Operating Load (lb_f)

2. Ball Screw Life (L)

$$L = \left(\frac{C}{F}\right)^3 \times L \text{ in inches or } L_h = \frac{L}{n_m \times 60}$$

C = Rated Dynamic Load Capacity (lb_f)
F = Equivalent Operating Load (lb_f)
L = Life in Inches
L_h = Life (hours)
n_m = Average Speed (in. × min. -1)

3. Rotational Speed Required for a Specific Linear Velocity

$$n = \frac{\text{Travel Rate (in.} \times \text{min.}^{-1})}{\text{Lead (in.)}}$$

n = rpm

4. Machine Service Life

After ball screw life (L) is calculated, apply it to the following formula to determine machine service life.

$$\text{Machine Service Life (in years)} = \frac{L_h}{\text{(machine operating hours)} \cdot \text{(days/year)} \cdot \left(\frac{\text{ball screw operating hours}}{\text{machine operating hours}}\right)}$$

5. Torque

a. Driving torque: $T_d = \frac{F \times P}{2\pi e} = 0.177 \times F \times P$ (lb_f × in.)

b. Backdrive torque: $T_b = \frac{F \times P \times e}{2\pi} = 0.143 \times F \times P$ (lb_f × in.)

(conversion of linear to rotational motion)

F = Equivalent Operating Load (lb_f)
P = Lead (in.)
e = Efficiency = 0.90
T_d = Driving Torque
T_b = Backdrive Torque

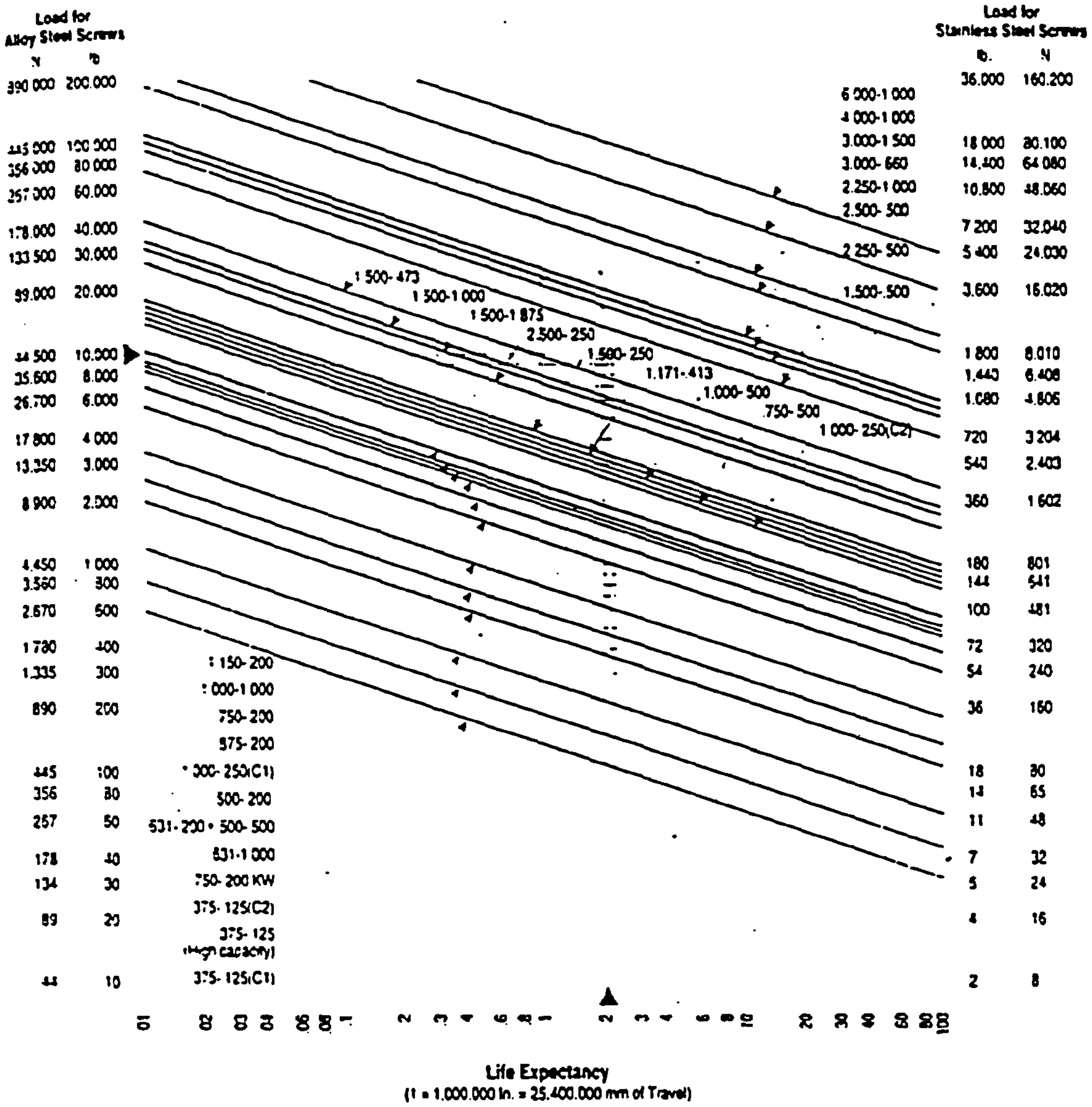
6. Power

$$P_d = \frac{F \times P}{(2\pi)\, e} \times \frac{n}{6.302 \times 10^4} = \frac{F \times P \times n}{3.564 \times 10^5} \text{ (HP)}$$

P_d = Power (HP)
n = rpm

U:\96\WPTRANS RPT\ATT-X...

Life Expectancy for Precision *Non-Preloaded* Ball Screw Assemblies



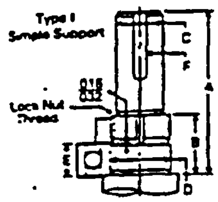
Example: Application life expectancy (total travel) desired is 2 million in. (50.8 million mm).
Normal operating load is 10,000 lb. (44,500 N).
All screws with curves which pass through or are above and to the right of the plotted point are suitable for the example.

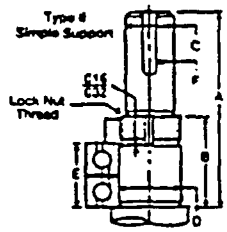
To order, call 517-776-4123

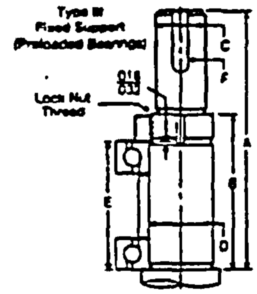
THOMSON SAGINAW

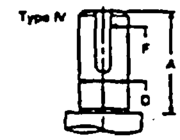
Typical Ends

The typical ends shown below can be machined on all ball screws* and ball splines*. Specific dimensional data is provided in the chart. Shaded area represents the extension required for a keyed drive coupling.









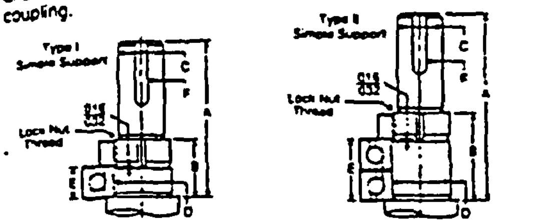
Screw Size*		Root Min. Dia. (Ball Screws)	Type I† Typical Journal for Single Bearings			Type II† Typical Journal for Duplexed Bearings			Type III† Typical Journal for Multiple Sets of Duplexed Bearings or Quick Mount Bearing Blocks				Dimensions Common to End Configurations Types I, II, and III — Standard Hardware				Type IV† Typical Journal for Pillow Block		
			A	B	E	A	B	E	A	B	E	Part No.	C	D	F	Brg	A	D	F
.187	.062	0.1400	1.360	0.380	0.156	1.516	0.516	0.312	1.828	0.828	0.624		.093/.092	.1251/.1248		R2 Radial			
.375	.125	0.3000	1.808	0.858	0.278	2.178	0.828	0.562	2.728	1.678	1.104	.186 DIE 10-13-97	.187/.186	.2757/.2752	1/16 x 1/32	607 Radial	RADIAL DIE 10-13-97		
.500	.200	0.3600	1.970	0.720	0.315	2.285	1.035	0.630	2.915	1.665	1.260		.250/.249	.3544/.3541	3/32 x 3/64	435 Radial			
.500	.500	0.3800	1.970	0.720	0.315	2.285	1.035	0.630	2.915	1.665	1.260		.250/.249	.3544/.3541	3/32 x 3/64	635 Radial	RADIAL DIE 10-13-97		
.631	.200	0.4800	2.110	0.818	0.394	2.504	1.204	0.788	3.292	1.992	1.576	7824154	.406/.405	.4725/.4723	1/8 x 1/16	201	2.625	.5000/.4995	1/8 x 1/16 x 1.500
.631	1.000	0.4800	2.110	0.818	0.394	2.504	1.204	0.788	3.292	1.992	1.576	7824154	.406/.405	.4725/.4723	1/8 x 1/16	201	2.625	.5000/.4995	1/8 x 3/16 x 1.500 (1/16 DIE 10-13-97)
.750	.200	0.5970	1.870	0.870	0.433	2.310	1.310	0.866	3.180	2.180	1.732	7824155	.500/.499	.5908/.5905	1/8 x 1/16	202	2.820	.5625/.5620	1/8 x 1/16 x 1.500
.750	.500	0.5970	1.870	0.870	0.433	2.310	1.310	0.866	3.180	2.180	1.732	7824155	.500/.499	.5908/.5905	1/8 x 1/16	202	2.850	.5625/.5620	1/8 x 1/16 x 1.500
.875	.200	0.7350	2.233	0.918	0.472	2.706	1.390	0.944	3.648	2.334	1.888	7824156	.562/.561	.6885/.6882	1/8 x 1/16	203	2.625	.6250/.6245	3/16 x 3/32 x 1.500
1.000	.200	0.8600	2.375	1.080	0.561	2.926	1.611	1.102	4.028	2.713	2.204		.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000	.250	0.8200	2.375	1.060	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000	.500	0.820	2.375	1.080	0.561	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000	1.000	0.8200	2.375	1.060	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.150	.200	1.0050	2.880	1.120	0.591	3.271	1.711	1.182	4.453	2.893	2.364	7824158	.750/.749	.9848/.9842	3/16 x 3/32	205	2.844	1.0000/.9995	1/4 x 1/8 x 1.500
1.171	.413	0.8700	2.375	1.060	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.844	.8750/.8745	3/16 x 3/32 x 1.500
1.250	.200	1.1130	2.880	1.120	0.591	3.280	1.710	1.181	4.440	2.880	2.362		.750/.749	.9848/.9842	3/16 x 3/32	205	3.150	1.0000/.9995	1/4 x 1/8 x 1.500
1.250	.500	1.0490	2.680	1.120	0.591	3.280	1.710	1.181	4.440	2.880	2.362		.750/.749	.9848/.9842	3/16 x 3/32	205	3.150	1.0000/.9995	1/4 x 1/8 x 1.500
1.500	.200	1.3470	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520		1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.438	1.3125/1.3120	5/16 x 5/32 x 1.750
1.500	.250	1.3200	2.970	1.180	0.630	3.600	1.790	1.260	4.880	3.050	2.520	7824159	1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.438	1.3125/1.3120	5/16 x 5/32 x 1.750
1.500	.500	1.2300	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520	7824159	1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.250	1.1875/1.1870	1/4 x 1/8 x 1.750
1.500 1.500	.473 1.000	1.1400	2.880	1.120	0.591	3.271	1.711	1.182	4.453	2.893	2.364	7824158	.750/.749	.9848/.9842	3/16 x 3/32	305	3.250	1.1250/1.1245	1/4 x 1/8 x 1.750
1.500	1.875	1.1880	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520		1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.250	1.1875/1.1870	1/4 x 1/8 x 1.750
1.750	.200	1.6130	3.850	1.440	0.906	4.556	2.366	1.812	6.368	4.178	3.624		1.375/1.374	1.5752/1.5747	5/16 x 5/32	308	4.438	1.5000/1.4995	3/8 x 3/16 x 2.500
2.000 2.250 2.250	.200 .500 1.000	1.8500	3.730	1.540	0.984	4.714	2.524	1.968	6.682	4.492	3.936	7824160	1.375/1.374	1.7721/1.7716	5/16 x 5/32	309	4.938	1.7500/1.7495	3/8 x 3/16 x 3.000
2.500	.250	2.3200	4.560	1.680	1.063	5.823	2.746	2.126	7.749	4.872	4.252	7824161	1.750/1.749	1.9689/1.9684	3/8 x 3/16	310	5.188	2.0000/1.9995	[illegible] x 3.000
2.500	.500	2.3200	4.560	1.680	1.063	5.823	2.746	2.126	7.749	4.872	4.252	7824161	1.750/1.749	1.9689/1.9684	3/8 x 3/16	310	5.188	2.0000/1.9995	1/2 x 1/4 x 3.000
3.000 3.000	.660 1.500	2.4800	5.560	1.875	1.221	6.781	3.096	2.442	9.223	5.538	4.884		2.250/2.249	2.3627/2.3621	1/2 x 1/4	312	7.250	2.4375/2.4365	5/8 x 5/16 x 4.688
4.000	1.000	3.3380	6.950	2.280	1.535	8.485	3.795	3.070	11.555	6.865	6.140		3.000/2.999	3.1501/3.1496	3/4 x 3/8	316	7.563	2.9370/2.9360	3/4 x 3/8 x 4.000
6.000	1.000	5.1430	8.508	3.348	2.284	10.879	5.629	4.567	15.446	10.196	9.134		4.937/4.936	5.1188/5.1179	1-1/2 x 3/4	326	15.500	4.9207/4.9197	1-1/2 x 3/4 x 7.430

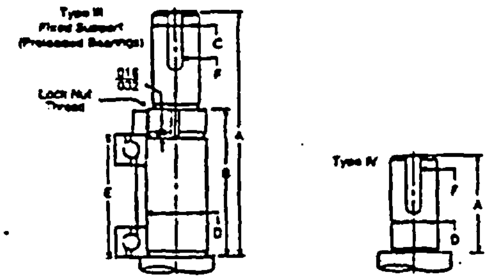
* Screw Size refers to all standard ball screws. For ball splines, root diameter limitations apply.
† To provide for bearings with larger capacity than those available for these typical ends, adaptors can be used to increase the diameter.

See Quick Mount® bearing support blocks on page 75.

Typical Ends

The typical ends shown below can be machined on all ball screws* and ball splines*. Specific dimensional data is provided in the chart. Shaded area represents the extension required for a keyed drive coupling.





Screw Size*	Root Min. Dia. (Ball Screws)	Type I† Typical Journal for Single Bearings A	B	E	Type II† Typical Journal for Duplexed Bearings A	B	E	Type III† Typical Journal for Multiple Sets of Duplexed Bearings or Quick Mount Bearing Blocks A	B	E	Part No.	Dimensions Common to End Configurations Types I, II, and III C	D	F (Standard Hardware)	Avg Can Brg	Type IV† Typical Journal for Pillow Block A	D	F
.187 .062	0.1400	1.380	0.380	0.154	1.516	0.516	0.312	1.826	0.826	0.624		.590/.580	.1251/.1248		R2 Radial	RADIAL		
.375 .125	0.3000	1.900	0.880	0.276	2.176	[illegible]	0.552	2.728	1.478	1.104	.186 BiE 10-13-97	.187/.186	.2757/.2754	1/16 x 1/32	R7 Radial	RADIAL	BiE 10-13-97	
.500 .200	0.3800	1.970	0.720	0.315	2.285	1.035	0.630	2.915	1.665	1.290		.250/.249	.3544/.3541	3/32 x 3/64	R8 Radial			
.500 .500	0.3800	1.970	0.720	0.315	2.285	1.035	0.630	2.915	1.665	1.290		.250/.249	.3544/.3541	3/32 x 3/64	R8 Radial	RADIAL	BiE 10-13-97	
.631 .200	0.4800	2.110	0.810	0.394	2.504	1.204	0.788	3.292	1.992	1.578	7824154	.406/.405	.4728/.4723	1/8 x 1/16	201	2.625	.5000/.4996	1/8 x 1/16 x 1.500
.631 1.000	0.4800	2.110	0.810	0.394	2.504	1.204	0.788	3.292	1.992	1.578	7824154	.406/.405	.4728/.4723	1/8 x 1/16	201	2.625	.5000/.4995	1/8 x 1/16 x 1.500
.750 .200	0.5870	1.870	0.870	0.433	2.310	1.310	0.866	3.180	2.180	1.732	7824155	.500/.499	.5908/.5905	1/8 x 1/16	202	2.620	.5425/.5420	1/8 x 1/16 x 1.500
.750 .500	0.5870	1.870	0.870	0.433	2.310	1.310	0.866	3.180	2.180	1.732	7824156	.500/.499	.5908/.5905	1/8 x 1/16	202	2.800	.5425/.5420	1/8 x 1/16 x 1.500
.875 .200	0.7350	2.233	0.916	0.472	2.705	1.380	0.944	3.649	2.334	1.888	7824156	.562/.561	.6695/.6692	1/8 x 1/16	203	2.625	.6250/.6245	3/16 x 3/32 x 1.500
1.000 .200	0.8200	2.373	1.080	0.561	2.926	1.611	1.102	4.028	2.713	2.204		.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000 .250	0.8200	2.373	1.080	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000 .500	0.820	2.373	1.080	0.561	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.000 1.000	0.8200	2.375	1.060	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.719	.7500/.7495	3/16 x 3/32 x 1.500
1.150 .200	1.0050	2.880	1.120	0.591	3.271	1.711	1.182	4.453	2.880	2.364	7824158	.750/.749	.9846/.9842	3/16 x 3/32	205	2.844	1.0000/.9995	1/4 x 1/8 x 1.500
1.171 .413	0.8700	2.375	1.060	0.551	2.926	1.611	1.102	4.028	2.713	2.204	7824157	.625/.624	.7877/.7873	3/16 x 3/32	204	2.844	.8750/.8745	3/16 x 3/32 x 1.500
1.250 .200	1.1130	2.880	1.120	0.591	3.280	1.710	1.181	4.440	2.880	2.362		.750/.749	.9846/.9842	3/16 x 3/32	205	3.150	1.0000/.9995	1/4 x 1/8 x 1.500
1.250 .500	1.0490	2.580	1.120	0.591	3.280	1.710	1.181	4.440	2.880	2.362		.750/.749	.9846/.9842	3/16 x 3/32	205	3.150	1.0000/.9995	1/4 x 1/8 x 1.500
1.500 .200	1.3470	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520		1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.438	1.3125/1.3120	5/16 x 5/32 x 1.750
1.500 .250	1.3200	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520	7824159	1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.438	1.3125/1.3120	5/16 x 5/32 x 1.750
1.500 .500	1.2300	2.970	1.160	0.630	3.800	1.790	1.260	4.880	3.050	2.520	7824159	1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.250	1.1875/1.1870	1/4 x 1/8 x 1.750
1.500 .473 1.520 1.000	1.1400	2.880	1.120	0.591	3.271	1.711	1.182	4.453	2.880	2.364	7824158	.750/.749	.9846/.9842	3/16 x 3/32	305	3.250	1.1250/1.1245	1/4 x 1/8 x 1.750
1.500 1.875	1.1880	2.970	1.180	0.630	3.800	1.790	1.260	4.880	3.050	2.520		1.000/.999	1.1814/1.1810	1/4 x 1/8	208	3.250	1.1875/1.1870	1/4 x 1/8 x 1.750
1.750 .200	1.6130	3.650	1.400	0.906	4.556	2.364	1.812	6.364	4.178	3.624		1.375/1.374	1.5752/1.5742	5/16 x 5/32	308	4.438	1.5000/1.4995	3/8 x 3/16 x 2.500
2.000 .200 2.250 .500 2.250 1.000	1.8500	3.730	1.540	0.984	4.714	2.524	1.968	6.882	4.492	3.936	7824160	1.375/1.374	1.7720/1.7716	5/16 x 5/32	309	4.938	1.7500/1.7495	3/8 x 3/16 x 3.000
2.500 .250	2.3200	4.560	1.680	1.063	5.623	2.746	2.126	7.749	4.872	4.252	7824161	1.750/1.749	1.9685/1.9680	3/8 x 3/16	310	5.188	2.0000/1.9995	1/2 x 1/4 x 3.000
2.500 .500	2.3200	4.560	1.680	1.063	5.623	2.746	2.126	7.749	4.872	4.252	7824161	1.750/1.749	1.9685/1.9680	3/8 x 3/16	310	5.188	2.0000/1.9995	1/2 x 1/4 x 3.000
3.000 .660 3.000 1.500	2.4800	5.360	1.875	1.221	6.731	3.096	2.442	9.223	5.538	4.884		2.250/2.249	2.3627/2.3621	1/2 x 1/4	312	7.250	2.4375/2.4365	5/8 x 5/16 x 4.688
4.000 1.000	3.3380	6.950	2.280	1.535	8.485	3.796	3.070	11.555	8.086	6.140		3.000/2.999	3.1501/3.1496	3/4 x 3/8	316	7.563	2.9370/2.9360	3/4 x 3/8 x 4.688
6.000 1.000	5.1430	8.596	3.546	2.284	10.879	5.629	4.567	15.446	10.198	9.134		4.937/4.936	5.1189/5.1179	1-1/2 x 3/4	326	15.500	4.9207/4.9197	1-1/2 x 3/4 x 7.430

*Screw Size refers to all standard ball screws. For ball splines, root diameter limitations apply.
†To provide for bearings with larger capacity than those available for these typical ends, adaptors can be used to increase the diameter.

See Quick Mount* bearing support blocks on page 75.

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THOMSON SAGINAW
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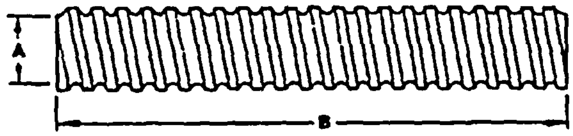
Precision Thread Screws

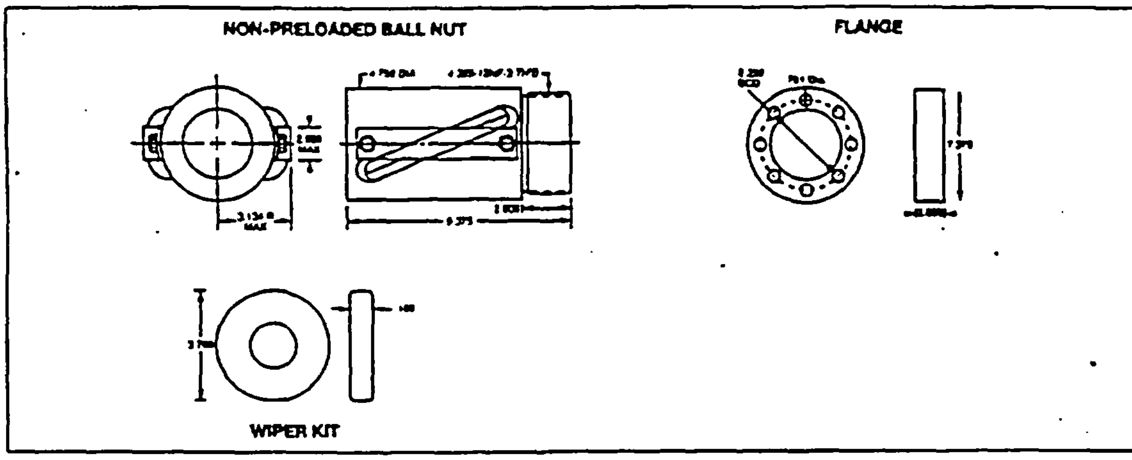
Precision Screw Stock

- Lead Accuracies:
  - .004 in./ft. accuracy standard
  - .002 in./ft. accuracy available
  - .001 in./ft. accuracy available



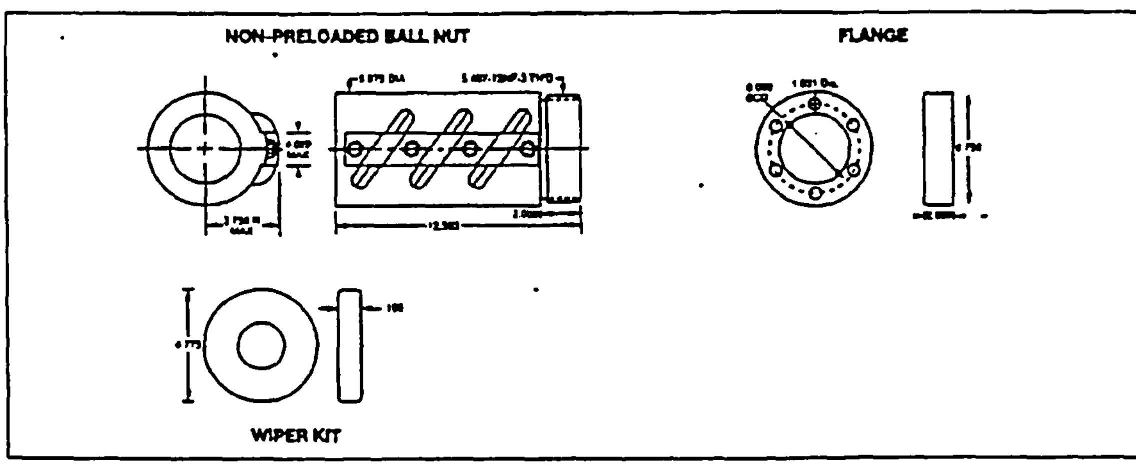
Ball Circle	Lead (in.)	Direction	Part Number	A Root Diameter (in.)	B Maximum Lengths Available (ft.)	Weight (lb./in.)	Use With These Ball Nuts	w/Mounting Blocks
.187	.050[†]	RH	7821634	.140	1	.006	7821632, 7821609	N/A
	.062[†]	RH	7821633	.140	1	.006	7821579, 7831631	N/A
.375	.125	RH	5707538	.300	4	.026	5709574,5707502	N/A
	.125	LH	5708532	.300	4	.026	5708282, 5709576	N/A
	.125[†]	RH	5706540	.300	4	.026	5709578, 5707643	N/A
	.125	RH	7824974	.285	4	.026	7824973	N/A
.500	.200	RH	7826721	.360	6	.040	7826721, 7826763, 7823871	N/A
	.500[†]	RH	5706740	.360	4	.042	5707506, 5709582, 7826767	N/A
	.500[†]	RH	5706846	.360	6	.042	5707644, 5709584	N/A
.631	.200	RH	5707540[*]	.480	6	.069	7820827[*], 7820955, 7823584	7824154
	.200	LH	5707541[*]	.480	6	.069	7820828[*]	7824154
	.200[†]	RH	5705378	.480	6	.069	5707645	7824154
.631	1.000	RH	7826712	.505	6	.069	7826713, 7827531	
.750	.200	RH	7824298	.600	6	.121	7824297	7824155
	.200	RH	7826770	.625	6	.125	7826768, 7823870	7824155
	.500[†]	RH	7824361	.580	6	.131	7824358, 7826991	7824155
.875	.200	RH	5708859	.735	12	.148	5708277, 7823585	7824156
1.000	.250	RH	7820426[*]	.820	16	.183	5707508[*], 5700348[*], 5708278, 5704167, 7823586	7824157
	.250	LH	7820428[*]	.820	16	.183	5707535[*], 5708284, 5704168	7824157
	.500[†]	RH	7824290	.830	16	.183	7824286	7824157
	1.000[†]	RH	7820429	.820	16	.183	5707509	7824157
1.150	.200	RH	7820430	1.005	16	.265	5701566, 5704270, 7823587	7824158
	.200	LH	7820431	1.005	16	.265	7820207, 7820206	7824158
1.171	.413	RH	7820432	.870	16	.231	5707511	7824157
1.500	.250	RH	7820595	1.320	20	.432	5709587, 5704271, 7823588	7824159
	.250	LH	7820596	1.320	20	.432	5701990	7824159
	.473	RH	7820597	1.140	20	.373	5707513, 5708345 (KW)	7824159
	.500	RH	7824253	1.140	20	.399	7824246	7824159
	1.000[†]	RH	7820598	1.140	20	.373	5708280, 5700698	7824159
	1.000[†]	LH	7825925	1.140	20	.373	5701895	7824159
	1.875[†]	RH	7820599	1.188	20	.438	5707654, 5704272	7824159
2.250	.500	RH	7820600	1.850	20	.906	5707516, 5708346 (KW), 7823589	7824160
	.500	LH	7820602	1.850	20	.906	5704000	7824160
	1.000[†]	RH	7820604	1.850	20	.906	5704555	7824160
2.500	.250	RH	7820606	2.320	20	1.288	5703243, 7823590	7824161
	.500	RH	7824262	2.100	20	1.167	7824136	7824161
3.000	.660	RH	7820607	2.480	20	1.635	5707519, 5708347 (KW), 5703045	N/A
	1.500[†]	RH	7820609	2.480	20	1.610	5704986	N/A
4.000	1.000	RH	5703262	3.338	20	2.869	5703258	N/A
6.000	1.000	RH	5704762	5.220	20	6.830	5704738	N/A

Longer lengths available on request. Contact factory.
[*] Stainless Steel
[†] Multiple Start Thread
[°] Value Priced
(KW) Keyway



Ball Circle Diameter (size)	Lead (in.)	General Description	Ball Nut Part Number	Ball Screw Part Number	Dynamic Load (lb.)	Max. Static Load (lb.)	Ball Nut Weight (lb.)	Flange Part Number	Wiper Kit Part Number	End Block Part Number
3.000	1.500	RH, NP	5704986	7820609	53,646	253,617	27.20	5707575	5702661*	N/A

* Brush-type wiper; formed felt available (5700458-RH); wiper dia.—3.640 in., width—.750 in.; retainer dia.—3.740 in. Increases length at each end by .375 in.



Ball Circle Diameter (size)	Lead (in.)	General Description	Ball Nut Part Number	Ball Screw Part Number	Dynamic Load (lb.)	Max. Static Load (lb.)	Ball Nut Weight (lb.)	Flange Part Number	Wiper Kit Part Number	End Block Part Number
4.000	1.000	RH, NP	5703258	5703262	85,758	476,970	53.50	5303307	5303306*	N/A

* Brush-type wiper; formed felt available (5704057-RH); wiper dia.—4.652 in., width—1.000 in.; retainer dia.—4.740 in. Increases length at each end by .500 in.

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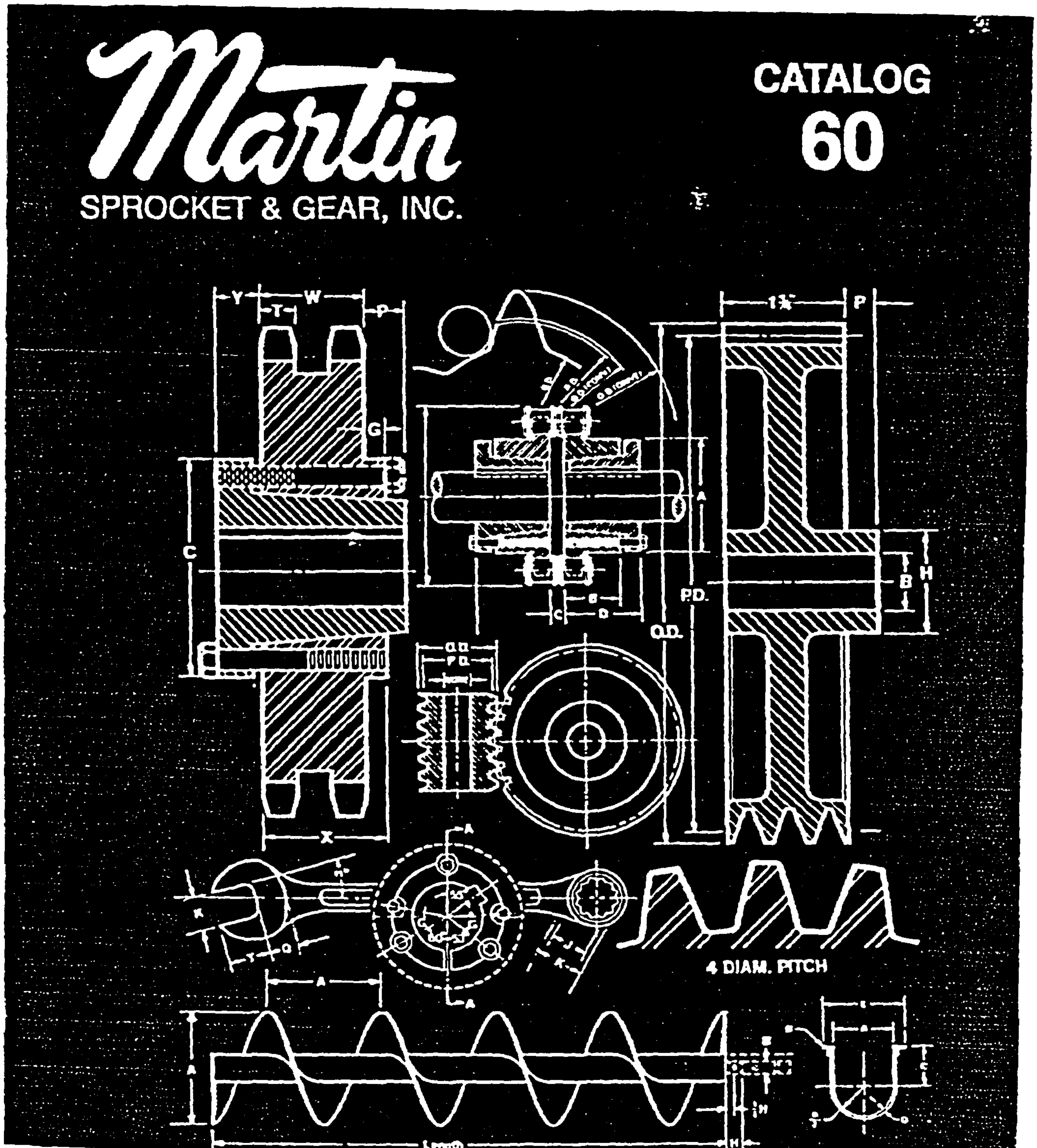
THOMSON SAGINAW

SECTION 3

MARTIN SPROCKET AND GEAR, INC. CATALOG NO. 60


Martin
SPROCKET & GEAR, INC.
CATALOG
60
4 DIAM. PITCH

Martin

All Steel Stock Sprockets No. 140 1¾" Pitch

Single - Taper Bushed With Hardened Teeth

No. TEETH	CATALOG NUMBER
12	140BTB12H
13	140BTB13H
14	140BTB14H
15	140BTB15H
16	140BTB16H
17	140BTB17H
18	140BTB18H
19	140BTB19H
21	140BTB21H
26	140BTB26H

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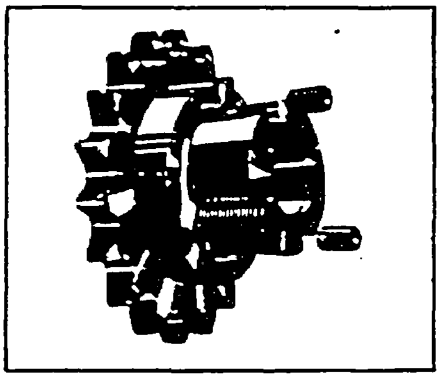
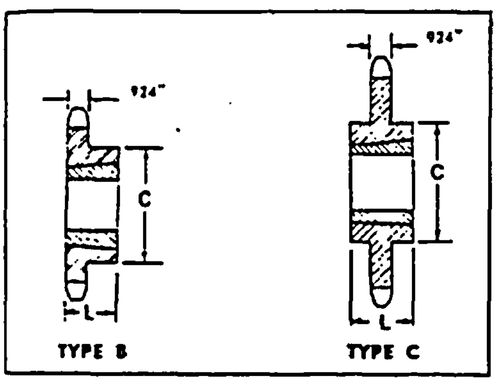




Single - Taper Bushed

No. Teeth	Catalog Number	Bushing	Diameter: Outside Diameter	Diameter: Pitch Diameter	Max. Bore	Dimensions: L	Dimensions: C	Type	Weight (App.): Rim Only	Weight (App.): Bushing Only
12	140BTB12	2517	7.581	6.762	2[illegible]	1[illegible]	4[illegible]	B	7.0	3.5
13	140BTB13	3020	8.150	7.313	3	2	5[illegible]	B	8.0	6.5
14	140BTB14	3020	8.718	7.864	3	2	5[illegible]	B	10.0	6.5
15	140BTB15	3020	9.283	8.417	3	2	5[illegible]	B	12.0	6.5
16	140BTB16	3020	9.848	8.970	3	2	5[illegible]	B	14.0	6.5
17	140BTB17	3020	10.411	9.524	3	2	5[illegible]	B	16.0	6.5
18	140BTB18	3020	10.975	10.078	3	2	5[illegible]	B	18.0	6.5
19	140BTB19	3020	11.537	10.632	3	2	5[illegible]	B	20.0	6.5
21	140BTB21	3020	12.660	11.742	3	2	5[illegible]	B	24.0	6.5
26	140BTB26	3020	15.483	14.518	3	2	5[illegible]	B	40.0	6.5
35	140CTB35	3535	20.494	19.523	3[illegible]	3[illegible]	6[illegible]	C	78.0	14
45	140CTB45	4040	26.078	25.087	4	4	7[illegible]	C	118.0	22
60	140CTB60	4040	34.442	33.438	4	4	7[illegible]	C	188.0	22
70	[illegible]	4040	40.017	39.006	4	4	7[illegible]	C	241.0	22

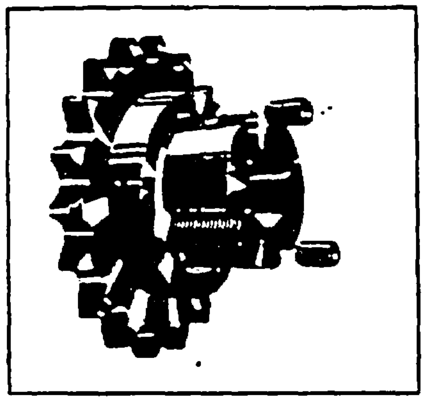
Martin

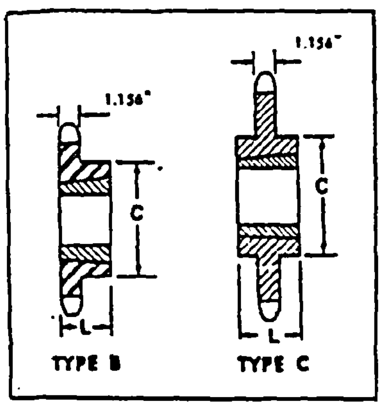
All Steel Stock Sprockets No. 160 2" Pitch

Single · Taper Bushed With Hardened Teeth

No. Teeth	Catalog Number
11	160BTB11H
12	160BTB12H
13	160BTB13H
14	160BTB14H
15	160BTB15H
16	160BTB16H
17	160BTB17H
18	160BTB18H
19	160BTB19H
21	160BTB21H
26	160BTB26H

SABER TOOTH®





Single · Taper Bushed

No. Teeth	Catalog Number	Bushing	Diameter: Outside Diameter	Diameter: Pitch Diameter	Max. Bore	Dimensions: L	Dimensions: C	Type	Weight (App.): Rim Only	Weight (App.): Bushing Only
11	160BTB11	2517	8.011	7.099	2[illegible]	1[illegible]	4[illegible]	B	9.0	3.5
12	160BTB12	3020	8.664	7.727	3	2	5[illegible]	B	11.0	6.5
13	160BTB13	3020	9.314	8.357	3	2	5[illegible]	B	13.0	6.5
14	160BTB14	3020	9.963	8.988	3	2	5[illegible]	B	16.0	6.5
15	160BTB15	3535	10.608	9.620	3[illegible]	3[illegible]	6[illegible]	B	25.0	14.0
16	160BTB16	3535	11.256	10.252	3[illegible]	3[illegible]	6[illegible]	B	28.0	14.0
17	160BTB17	3535	11.896	10.885	3[illegible]	3[illegible]	6[illegible]	B	32.0	14.0
18	160BTB18	3535	12.543	11.518	3[illegible]	3[illegible]	6[illegible]	B	35.0	14.0
19	160BTB19	3535	13.185	12.151	3[illegible]	3[illegible]	6[illegible]	B	39.0	14.0
21	160BTB21	3535	14.470	13.418	3[illegible]	3[illegible]	6[illegible]	B	48.0	14.0
26	160BTB26	3535	17.671	16.583	3[illegible]	3[illegible]	6[illegible]	B	68.0	14.0
35	160CTB35	4040	23.422	22.312	4	4	7[illegible]	C	118	14.0
45	160CTB45	4040	29.802	28.671	4	4	7[illegible]	C	186	22.0
60	160CTB60	4545	39.362	38.215	4[illegible]	4[illegible]	8[illegible]	C	292	30.0

SECTION 4

GOODMAN LOCOMOTIVES, MINING MACHINE SALES MANUAL

LOCOMOTIVE GENERAL DATA

LOCOMOTIVE STANDARDS

Information below was extracted from National Electrical Manufacturers Association Mining and Industrial Electric Locomotive Standards.

TROLLEY LOCOMOTIVES

LOCOMOTIVE SIZE: The locomotive size given in tons shall be the locomotive's rated weight ready to operate, with standard equipment, but the rated weight shall not include the weight of additional equipment or accessories.

TOLERANCES IN RATED WEIGHT: The tolerances in rated weight of locomotives shall be:

4 and 6 tons ________ 10 Percent
8, 11 and 15 tons ________ 8 Percent
Over 15 tons ________ 6 Percent

DRAWBAR PULL: The drawbar pull in pounds on a straight and level track with dry, clean rails shall be determined as follows:

1. The running drawbar pull with steel tread wheels shall be 25 percent of the rated weight of the locomotive.
2. The starting drawbar pull with sand and steel tread wheels shall be 30 percent of the rated weight of the locomotive.

HORSEPOWER PER TON: The standard horsepower per ton of rated weight of 2 motor locomotives shall be:

Up to and including 8 tons ________ 10 HP/ton
11 tons up to and including 15 tons ________ 12 HP/ton
Over 15 tons ________ 15 HP/ton

TOLERANCES IN HORSEPOWER PER TON: The tolerances in standard horsepower per ton of rated weight of 2 motor locomotives shall be:

Up to and including 15 tons ________ 15 percent - plus or minus
Over 15 tons ________ 10 percent - plus or minus

NOTE

A locomotive having more than standard horsepower per ton may be expected to exceed its rated weight, depending upon the amount of excess horsepower over standard horsepower. A locomotive equipped with the standard horsepower of the next larger size may be expected to weigh as much as the next larger size.

LOCOMOTIVE SPEEDS: The standard speed of locomotives at running drawbar pull for steel tread wheels, shall be:

Up to and including 8 tons ________ 6 MPH
11 tons up to and including 15 tons ________ 8 MPH
Over 15 tons ________ 10 MPH

NOTE

1. Slow speed shall be defined as 5 miles per hour or less.
2. While slow-speed locomotives call for less horsepower per ton, yet the reduced horsepower is the result of the reduced speed, leaving the torque required to develop the running drawbar pull of the locomotive the same as at standard speed. Thus, with the same gear reduction and wheel diameter, the size or overall dimensions and weight of the motors at slow speed are essentially the same as at standard speed.

TOLERANCES IN LOCOMOTIVE SPEEDS: The tolerances in standard speed of locomotives shall be:

Up to and including 15 tons ________ 15 percent - plus or minus
Over 15 tons ________ 10 percent - plus or minus

LOCOMOTIVE RATING: Locomotive rating shall be based upon the running drawbar pull and speed in miles per hour. This speed shall be determined at the running drawbar pull from motor performance curves, making an allowance for loss of tractive effort for each spur gear transmission in accordance with the following table, and further allowance for loss in journals, flanges and all other losses of one percent of the rated weight of the locomotive.

INPUT % Of One-hour Rating	GEAR LOSS % Of Input
200	2.5
150	2.5
125	2.5
100	2.5
75	2.5
60	2.7
50	3.2
40	4.4
30	6.7
25	8.5

NOTE

The word "transmission" is defined to mean contact between any two gears.

VOLTAGE RATING: Standard voltages shall be 250 and 500 volts direct current.

The following tabulation is a consolidation of the above information:

Locomotive Rated Weight, Tons		Total Horsepower per Locomotive		Locomotive Speed, Miles per Hour	
Standard	Tolerance	Standard	Tolerance	Standard	Tolerance
4	3.6 to 4.4	40	34 to 46	6	5.1 to 6.9
6	5.4 to 6.6	60	51 to 69	6	5.1 to 6.9
8	7.4 to 8.6	80	68 to 92	6	5.1 to 6.9
11	10.1 to 11.9	132	112 to 152	8	6.8 to 9.2
15	13.8 to 16.2	180	153 to 207	8	6.8 to 9.2
20	18.8 to 21.2	300	270 to 330	10	9 to 11
27	25.4 to 28.6	404	364 to 444	10	9 to 11
37	34.8 to 39.2	556	500 to 612	10	9 to 11
50	47 to 53	752	676 to 828	10	9 to 11

NOTE

1. The standard speed for each weight class reflects the best safe speed, as determined by general operating practice, for the mine conditions under which locomotives of this class are used. The higher speeds for larger locomotives reflect the fact that they are used in larger mines with heavier trackage, and other conditions which permit higher operating speed with reasonable safety.
2. The above tolerances make allowance for manufacturing and design variations. Since a locomotive has a definite rated drawbar pull, horsepower tolerances follow the corresponding speed tolerances.

Supersedes Section 4060-B, pages 1 & 2 dated March 1971

LOCOMOTIVE GENERAL DATA

TROLLEY LOCOMOTIVE CALCULATIONS

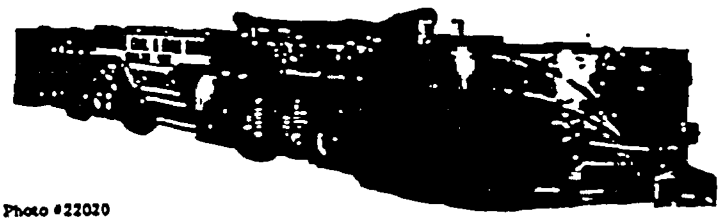


Photo #22020

The load that a locomotive can haul depends on a number of factors; namely, weight on the driving wheels, adhesion between driving wheels and rail, and train resistance which includes rolling resistance, resistance due to curves and resistance due to grades.

DEFINITIONS

ROLLING RESISTANCE
Rolling resistance includes friction of car journals and friction between wheel treads and flanges and rail. Windage is not considered.

Values of rolling resistance vary widely, depending on size of car, whether four wheel or eight wheel type, the kind of journal bearings on cars and weight, alignment and ballast of rail. For accurate calculations, the value of rolling resistance should be estimated from data derived from actual experience on property being studied.

Where such data is not available, values of 30 pounds per ton for cars with bronze journal bearings and 20 pounds per ton for cars with ball or roller journal bearings, are used as representative of average conditions.

GRADE RESISTANCE
The resistance due to grades has an exact value of 20 pounds per ton, for each percent rise in grade.

CURVE RESISTANCE
Curve friction is added friction in car journals and between wheel treads and flanges and rail due to track curvature. It is influenced by the degree of curvature and spread of rails at curves.

$$\text{CR-lbs./ton} = \frac{400 \times \text{Wheelbase in feet}}{\text{Radius of curve in feet}}$$

In mine haulage, curve friction is frequently neglected as a separate and specific part of train resistance, since only a small part of the train is on a particular curve at a time. It is considered as a part of car and track friction and the value selected for car and track friction is high enough to include the friction due to curves.

NOTE
To convert degree of curvature to radius of curve in feet which is easier to measure, use following formula:

$$\text{Degree of curvature} = \frac{5730}{\text{Radius in feet}}$$

TRAIN RESISTANCE (Summation of Rolling, Grade and Curve Resistances)

TRACTIVE EFFORT
Tractive effort is the force in pounds exerted at the driving wheel treads.

LEVEL DRAWBAR PULL
Level Drawbar Pull is the pull in pounds exerted by the locomotive at the drawbar on level dry track. Under various conditions, it is percent of the actual weight of the locomotive, which is called percent of adhesion.

ADHESION
Adhesion, expressed as a percentage of weight on the driving wheels, is a measure of resistance to slipping. Values of adhesion between steel tread (steel tires or rolled steel wheels) and clean dry steel rail well above 30 percent, have been observed. However, in mine haulage there is a wide range in rail weight, alignment and ballast and in rail conditions, varying from clean and dry to greasy and wet. Therefore, values for adhesion must be based on estimates found from actual experience but locomotives are rated as follows:

Rated Running Drawbar Pull with:
- Steel tread wheels (steel tires or rolled steel wheels) — 25%
- Cast Iron, chilled tread wheels — 20%

Rated Starting Drawbar Pull, with sand, with:
- Steel tread wheels (steel tires or rolled steel wheels) — 30%
- Cast Iron, chilled tread wheels — 25%

NOTES
1. Above data is based on level track with dry clean rails.
2. Wet rails have an adhesion value, varying between 5 and 15%.

TRACTIVE EFFORT AND DRAWBAR PULL
The theoretical difference between tractive effort and level drawbar pull is the force required to roll the weighted locomotive, wheel rims and flanges on grades or level track. Since most locomotives are equipped with roller bearings, the rolling resistance of a locomotive is considered to be 20 pounds per ton, or 1% of its weight. Therefore, if the drawbar pull of a locomotive is 25% of its weight, the T. E. will be 26% of its weight.

Example - assume 15 ton locomotive.

Drawbar pull = 15 x 2000 x .25 7500 pounds.
Tractive Effort = 7500 + .01x15 x 2000 = 15 x 2000 x .26
= 7800 pounds.

Supersedes Section 4068, pages 1 thru 4 dated March 1971.

SELECTING THE PROPER SIZE TROLLEY LOCOMOTIVE

The following formulae furnish a convenient method of estimating the required size of locomotive for a given application.

1. Train resistance: Up grade = $R_r + (G \times 20)$
   Down grade = $R_r - (G \times 20)$
   where:
   R_r = Rolling resistance
   G = Grade in percent

2. Weight of locomotive required to haul a given trailing load and itself on level or up a grade.
   $$L = \frac{W \times R_t}{2000\,(K + .01) - (G \times 20 + R_2)}$$
   where:
   L = Weight of locomotive in tons. (Ton = 2000#)
   W = Weight of trailing load in tons. (Ton = 2000#)
   R_t = Train resistance (lbs/ton)
   K = Adhesion (expressed as a decimal)
   G = Grade in percent
   R_2 = Locomotive rolling resistance (20 lbs/ton)

3. Rated Tractive Effort = 2000 L x (K + .01)

4. Rated Drawbar Pull = 2000 L x K

5. Required Running Tractive Effort = $L(R_2 + 20G) + WR_t$

6. Required Running Drawbar Pull = W x R_t

7. Horsepower required -
   $$H.P. = \frac{T.E. \times S}{336.25}$$
   where:
   S = Speed MPH.
   336.25 = conversion factor including gear efficiency

8. K.W. - Required input to locomotive
   K.W. = H.P. x .746 ÷ e
   where:
   e = Motor Efficiency

9. Rate of acceleration (A) miles per hour per sec.
   $$A = \frac{T.E.\ (Max) - T.E.\ (running\ required)}{100\,(L + W)}$$

10. Time to accelerate
    $$Ta = \frac{V_2 - V_1}{A}$$
    where:
    V_1 = Starting velocity (mph)
    V_2 = Attained velocity (mph)
    Ta = Time accelerate in seconds

11. Distance to accelerate = $\frac{V_2 + V_1}{2} \times Ta \times 1.47$ = (ft)

12. Braking force - down grade
    F = 2000 KL + $W(R_t - 20G) - L(20G)$
    where:
    K = Adhesion (decimal)
    G = Grade up, in percent

13. Braking force - up grade
    F = $L(2000K) + W(R_t + 20G) + L(20G)$

14. Distance to Brake (ft)
    $$D_b = \text{Distance (ft)} = \frac{(L + W)\,2000\,(V_1^2 - V_2^2)}{64.32FE}$$
    where:
    V_1 and V_2 Velocity ft., sec.
    (1.47 ft. /sec. = 1 MPH)
    E = Braking efficiency = .8
    F = Braking force
    V_1 = Starting velocity in ft. /sec.
    V_2 = Attained velocity in ft. /sec.

15. Time to brake -
    $$T_b = \frac{D_b}{\text{Average Velocity}}$$
    $$T_b = \frac{D_b \times 2}{(V_1 - V_2)\ 1.47}$$
    where:
    V_1 = Starting velocity in MPH
    V_2 = Attained velocity in MPH
    T_b = Time brake in seconds
    D_b = Distance to brake in feet

16. Continuous current required by root means square methods (see sample calculations).
    Blowers are being used quite extensively to increase the continuous rating. When installed, blowers will supply enough additional ventilation to increase the continuous rating of a motor up to approximately 75% of the one hour rating and the one hour rating will be increased approximately 10 to 15%.

HAULAGE CAPACITY TABLE

Locomotives: Weight in Tons	Locomotives: Rated Draw Bar Pull in Pounds (Dry rails-no sand)	Available Weight of Trailing Load in Tons — Percent of grade - against load: 0	1	2	3	4	5	6
		at 20 lbs. per ton rolling resistance on cars						
4	2000	100	48	31	22	17	13	11
6	3000	150	72	44	33	25	20	16
8	4000	200	96	61	44	34	27	22
10	5000	250	120	77	55	42	33	27
13	6500	325	156	100	72	55	43	35
15	7500	375	180	115	83	63	50	40
20	10000	500	240	153	110	84	67	54
25	13000	650	315	200	138	109	87	70
30	15000	750	360	230	165	126	100	81
40	20000	1000	480	307	220	168	134	108
		at 30 lbs. per ton rolling resistance on cars						
4	2000	67	38	28	20	15	12	10
6	3000	100	58	40	29	23	18	15
8	4000	134	77	53	39	31	25	20
10	5000	167	96	66	49	38	31	25
13	6500	217	125	85	64	50	40	33
15	7500	250	144	99	73	57	46	38
20	10000	334	192	132	98	76	62	51
25	13000	434	250	171	127	99	80	67
30	15000	500	288	197	147	114	92	75
40	20000	668	380	263	196	153	123	102

goodman

GOODMAN ELECTRIC LOCOMOTIVES FOR COAL MINING, METAL MINING AND TUNNELING

COAL MINING

	BATTERY POWERED				COMBINATION: BATTERY/TROLLEY	
Operating Weight*	5-ton	10-ton	14-ton	15-ton	10-ton	15-ton
Type	Mule	7508	Trackmaster	154	7588	1738
T.E. at 25% ADH., lbs	2,500	5,000	7,000	5,000	5,000	7,500
Voltage	96	96	128	128	250	250
Number of Motors	1	2	2	2	2	2
1 Hr. HP Each Motor**	18	25	41	50	30	50
T.E. at 1 Hr. HP, lbs.	1,450	3,080	4,900	5,300	3,450	6,300
Speed at 1 Hr. HP, MPH**	4.7	6.0	5.8	7.0	6.4	5.8
Minimum Gauge, inches	42	36	42	36	36	36
Minimum Curve Radius, ft.	18	35	36	46	35	46
Wheelbase, inches	60	89	90	100	89	100
Wheel Diameter, inches	15	26	28	30	26	30
Height Over Covers, in.	32	32	29	38	32	38
Overall Width, inches	64	70	84	75	70	75
Length w/o Couplers, in.	160	223	231	256	223	256

*Operating weight will vary according to weight of battery used.
**Horsepower and speed will vary according to actual voltage being supplied by the battery.

TROLLEY POWERED

Operating Weight*	8-ton	11-ton	11-ton	17-ton	15-ton	15-25 ton	20-25 ton
Type	7588	[illegible]	1738	184	[illegible]	281	202
T.E. at 25% ADH., lbs.	4,000	5,500	5,500	5,500	7,500	7,500-10,000	10,000-12,500
Voltage	250	250	250	250	250	250/500	250
Number of Motors	2	2	2	2	2	2	2
1 Hr. HP Each Motor	30	41	50	60	65	100	170
T.E. at 1 Hr. HP, lbs.	3,450	4,800	6,300	6,000	7,550	7,530	11,840
Speed at 1 Hr. HP, MPH**	6.4	5.8	5.8	7.5	5.9	9.7	10.5
Minimum Gauge, inches	36	42	36	42	42	36	42
Minimum Curve Radius, ft.	31	36	31	34	36	46	55
Wheelbase, inches	78	90	66	84	90	100	110
Wheel Diameter, inches	26	26	30	26	28	30	36
Height Over Covers, in.	32	29	36	28	29	42	44
Overall Width, inches	64	84	60 or 66	74	84	66 or 70	67 or 71
Length w/o Couplers, in.	223	231	167	228	231	261	282

*Operating weight is nominal and will vary according to the addition of optional equipment and other customer requests.
**Horsepower and speed will vary according to actual voltage supplied by the trolley system.

TUNNELING

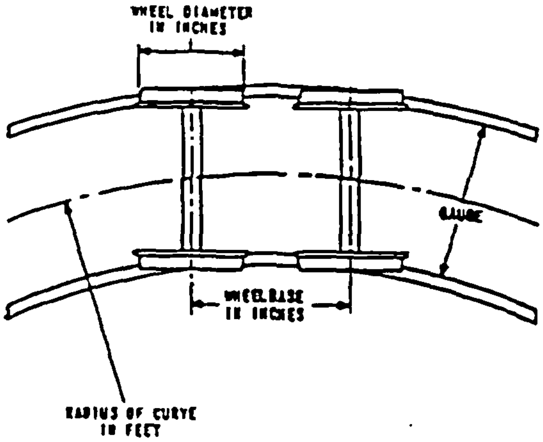
BATTERY POWERED

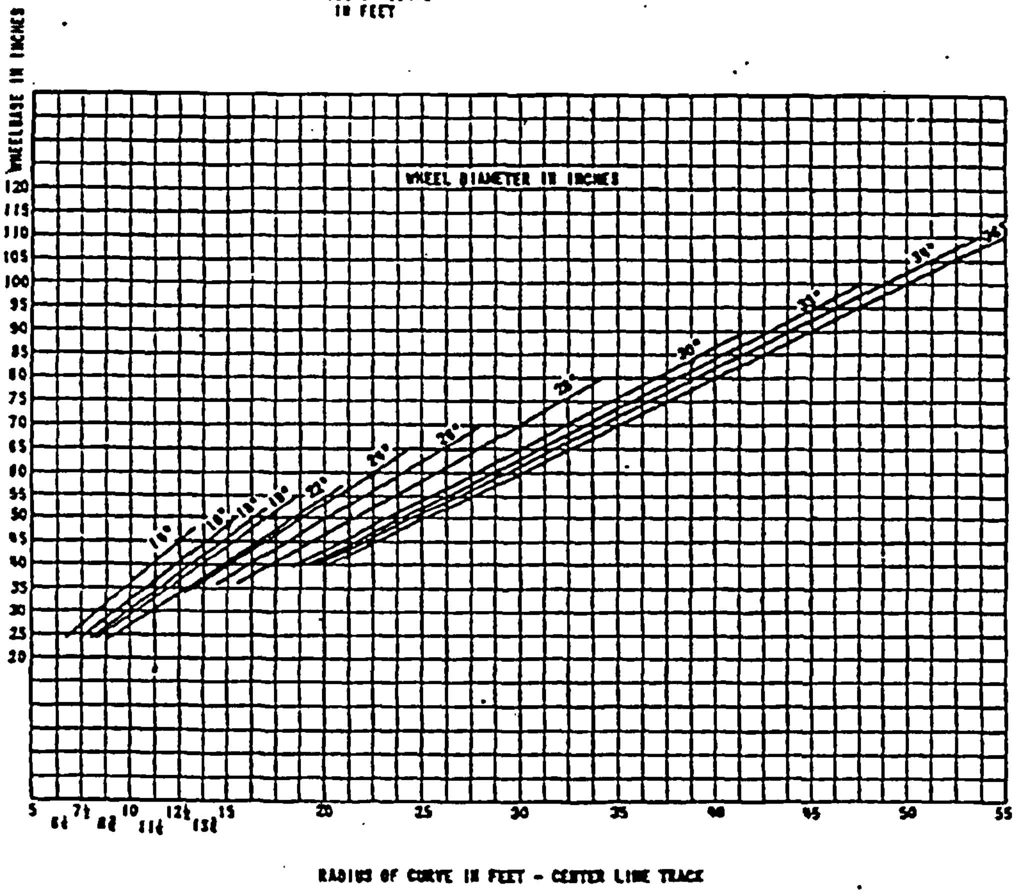
Operating Weight*	2-ton	4-ton	6-ton	8-ton	10-ton	15-ton	15-ton	25-ton	36-ton	36-ton
Type	Trammer	Titan	Mancha	Mancha	750	158	154	1388	221	281
T.E. at 25% ADH., lbs.	1,000	2,000	3,000	4,000	5,000	7,500	7,500	10,000	15,000	15,000
Voltage	48	80	80	80	80	240	128	240	288	240
Number of Motors	1	1	1	2	2	2	2	2	2	2
1 Hr. HP Each Motor**	5.5	14	20	20	19	40	50	75	125***	125***
T.E. at 1 Hr. HP, lbs.	475	1,160	1,300	2,600	2,740	4,890	5,300	8,000	14,000	10,850
Speed at 1 Hr. HP, MPH**	3.7	4.0	4.9	4.9	5.1	6.0	7.0	6.9	6.6	8.5
Minimum Gauge, inches	18	18	18	18	24	24	33	36	36	36
Minimum Curve Radius, ft.	7	9	14	14	22	28	28	31	47	47
Wheelbase, inches	24	30	37.5	37.5	54	60	60	66	100	100
Wheel Diameter, inches	14	16	21	21	26	30	30	33	32	30
Height Over Battery, in.	44	43	55	55	52	60	63	66	65	64
Overall Width, inches	33 to 43	43	41 or 53	41 or 53	42 or 57	54 or 60	59	64	85	85
Length w/o Couplers, in.	71	122	144	144	153	178	172	186	261	261

*Operating weight will vary according to weight of battery used.
**Horsepower and speed will vary according to actual voltage being supplied by the battery.
***Motor rating is with forced ventilation.

MINIMUM RADIUS OF CURVE

For Operation of Locomotives With Given Wheel Diameters And Wheelbases.





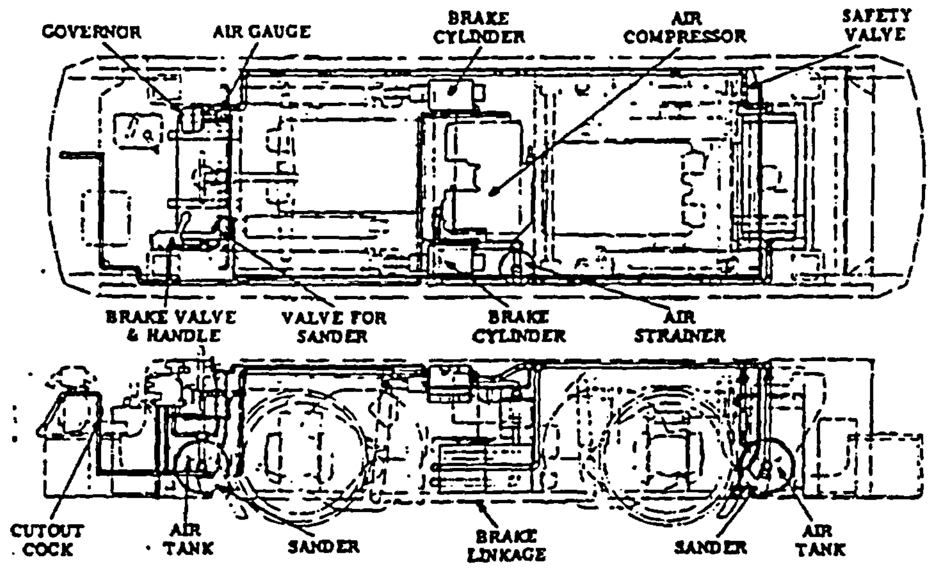
LOCOMOTIVE GENERAL DATA

AIR BRAKES AND AIR SANDERS

Air braking systems may be divided into two types, straight and automatic. A straight air brake system is one which applies the brakes by letting air into the system. An automatic air brake system is one which applies the brakes by letting air out of the system, and has the advantage that should the air line break or leak the brakes will be set automatically and stop the train.

The straight air brake system is the one generally used on mine locomotives. The reason being, that it is a very simple system and too, in mines the cars are not usually equipped with brakes, therefore, the system only applies to the locomotive itself. However, this system does have one disadvantage, and that is, should there be a break or leak in the line the brakes would fail. As a safety measure, all locomotives equipped with this system are also equipped with standard hand brakes.

In this section, only the straight air brake system is covered, as it is the system most widely used on mine locomotives.



TYPICAL AIR BRAKE AND AIR SANDING SYSTEM

Equipment used:

1. An independent motor driven air compressor.
2. A governor.
3. One or two brake cylinders depending on the design.
4. One or two air tanks depending on the size of locomotive and type of service.
5. A brake or throttle valve.
6. An air strainer.
7. A safety valve.
8. An air gauge.
9. A connecting switch.
10. Piping and wiring.

Function of the Equipment:

1. The independent motor driven air compressor serves to keep pressure in the air tanks. The compressor is made in two sizes, 10 cu. ft. per min. and 16 cu. ft. per min. The size depends on the total air devices used (which might consist of air sanders, air whistle, or any other mechanism other than the air brakes which would be operated by air) also the size of the locomotive, grades, weight of train, etc.
2. The governor controls the amount of air that the compressor supplies to the tank. The operating pressure is ordinarily set at about 90 lbs. The governor cuts in and starts the motor which drives the air compressor at about 80 lbs. and stops the motor at 95 lbs. The governor is adjustable for average pressure.
3. The purpose of the brake cylinder is to convert air pressure into mechanical energy which is applied against the brake linkage. In some installations one cylinder is used, in which case this one cylinder operates the entire linkage. In other installations two cylinders are used, one operates the brake linkage on each side of the locomotive.
4. The air, under pressure, is held in air tanks. One or two tanks are used depending on the amount of air required and the design of the locomotive.
5. The control of the brakes is through a control or throttle valve. This valve is so constructed that the amount of air let into the system may be varied, thus enabling the braking pressure to be adjusted to suit requirements. Where two locomotives operate in permanent tandem a control valve is located on the primary locomotive only, which operates the brakes on both locomotives.

   Two locomotives in separable tandem may be arranged so that there is a control valve on each locomotive. In this case the valve on each locomotive will operate the brake on that particular locomotive when the locomotives are operated separately. When operated in tandem the control valve on the primary locomotive controls the brake system for both locomotives.
6. The air strainer cleans the air before it goes into the compressor.

Supersedes Section 4067-B, pages 1 & 2 dated August 1968

SECTION 5

SERAPID USA INC., RIGID CHAIN ENGINEERING, SPECIFICATION SHEET

RIGID PUSHING CHAINS

SPECIFICATION SHEET
PAGE 1

GENERAL

Because of the unique design of the links, the SERAPID rigid chain can provide an answer to load movement problems over long distances, whether pulling or pushing, lifting or lowering, from a few kilogrammes to loads well in excess of 100 tons.

The system allows precise positioning on each stroke and offers space saving benefits compared with conventional systems.

The rigidity of the chain is achieved by the applied moment locking the shouldered links under compressive load.

The chain is driven by twin pinions contained in a rigidly constructed drive housing. These pinions are keyed onto a shaft which extends through the side panels of the drive housing for subsequent coupling to the power pack.

Inside the drive housing are reaction and guiding plates which absorb the thrust reaction and ensure the correct positioning of the chain through the housing.

The pinions drive the chain via rollers positioned on the axle of the chain link.

METHODS OF USING THE CHAIN

Because of the knuckle type action of the links, the chain can be coiled in one articulation and becomes rigid in the other. It is therefore possible to envisage various means and methods of chain return storage (examples of which are shown below).

This basically gives the effect of using a bar which, when retracted, can "roll up". The amount of "roll up" is usually determined by the space availability and the distance of the stroke.

A special rear end link allows a self stacking facility, as shown in examples 2 and 3 below.

In vertical load movement applications, the use of the SERAPID system can mean, in many instances, a reduction in the superstructure requirements, as the thrust reaction is absorbed on the ground.

Wherever possible, the specially designed chain guide systems must be used, thus ensuring guaranteed results from the chain, regardless of the length of stroke.

Applications where the guide system cannot be used should always be discussed with SERAPID design engineers.

DIMENSIONS AND CAPACITIES OF THE CHAIN

There are 6 sizes of chain within the range, four of which are detailed in the following pages.

These are 25 mm, 40 mm, 60 mm and 90 mm. These figures refer to the pitch centres of the chain links.
All the links are fine blanked to achieve close tolerances.

Two other sizes are 130 mm and 200 mm but these should be specified by SERAPID design engineers only.

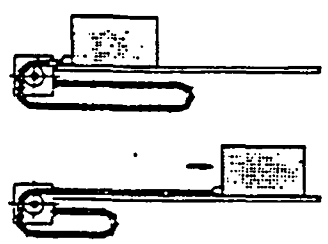
The selection of the grade of materials used depends on the individual application, [illegible] for size 25 mm, carbon steel and stainless steel for sizes 40, 60 and 90 mm being the most commonly used.

EXAMPLES OF USE

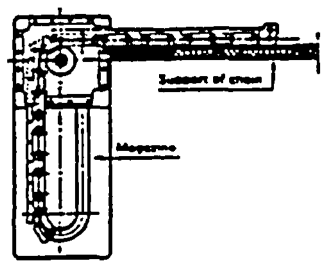
Rapid Bolster and Die Changing, Furnace Charging, Goods Lifts, Elevators, Rolling Mills, Automated Positioning Systems for Conveyors, Storage and Racking.

The chain system is widely used in the following industries: AUTOMOTIVE, CHEMICALS, STEEL MILLS, NUCLEAR, MINING, ELECTRICAL POWER SUPPLY, MACHINE TOOL.

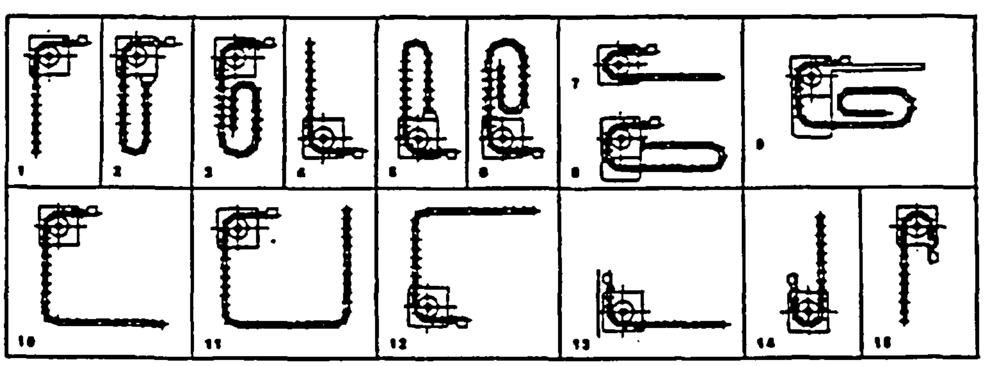
MOVEMENT OF A LOAD



EXAMPLE OF SPACE SAVING



EXAMPLES OF VARIOUS CHAIN RETURN



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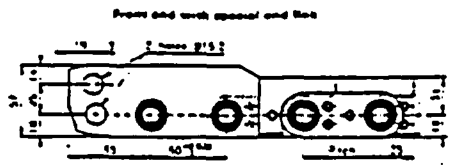
SPECIFICATION SHEET
PAGE 5

RIGID PUSHING CHAINS

PITCH 60 mm - TYPES S and D - SG and DG

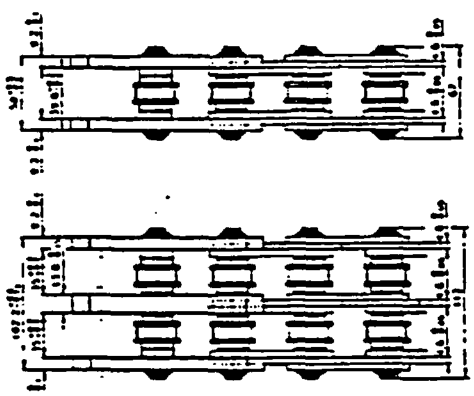
TYPES S and D



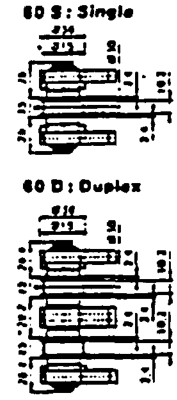
Characteristics	Pitch 60 mm 60 S	60 D
Horizontal pushing force * daN	2500	3750
Weight per meter kg	5.9	[illegible]
Number of links per meter	17	17

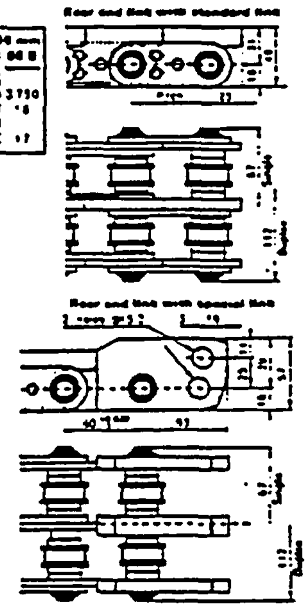
* For 1 meter of chain unguided

60 S: Single

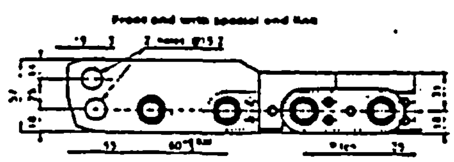


60 D: Duplex





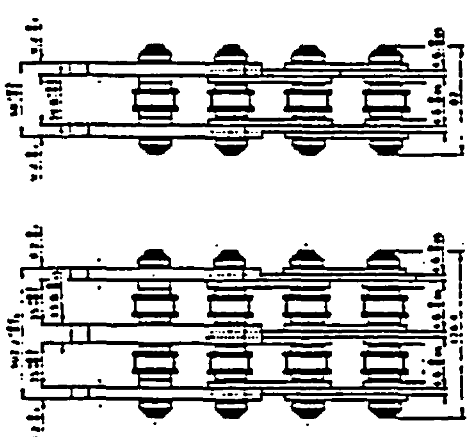
TYPES SG and DG with rollers



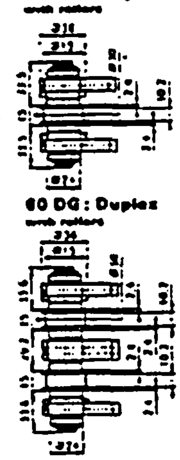
Characteristics	Pitch 60 mm 60 SG	60 DG
Horizontal pushing force * daN	2500	3750
Weight per meter kg	11	16.3
Number of links per meter	17	17

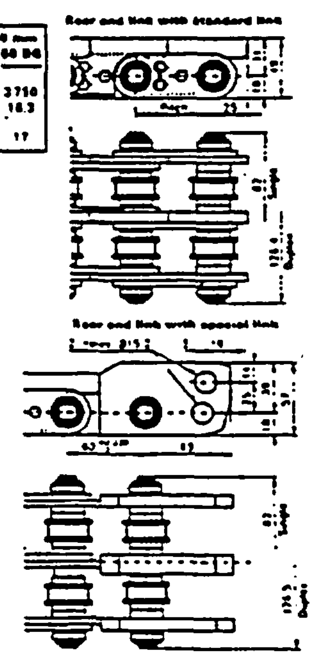
* For 1 meter of chain unguided

60 SG: Single with rollers



60 DG: Duplex with rollers





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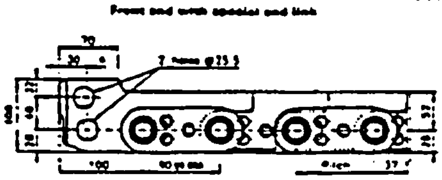
RIGID PUSHING CHAINS

PITCH 90 mm - TYPES S - D and SG

SPECIFICATION SHEET
PAGE 6

TYPES S and D

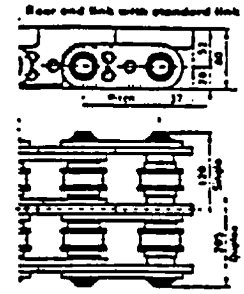
Front end with special end link



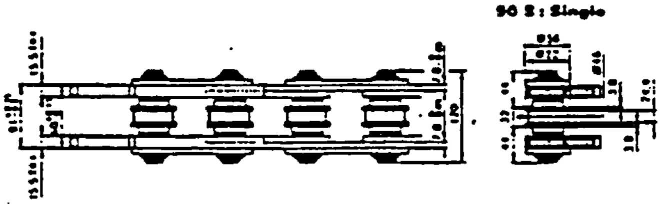
Characteristics		Pitch 90 mm 90 S	Pitch 90 mm 90 D
Horizontal pushing force *	daN	8 000	13 000
Weight per meter	kg	28.5	45
Number of links per meter	..	11	11

* For 1 meter of chain unguided

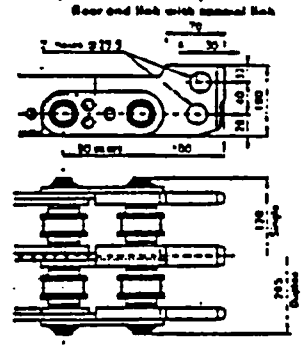
Rear end link with standard link



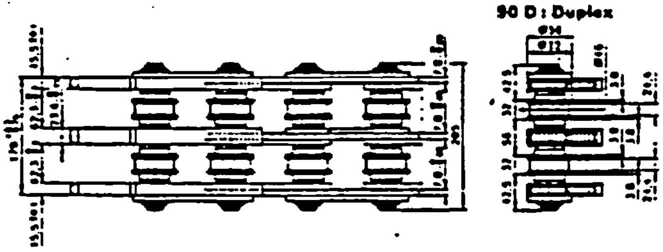
90 S : Single



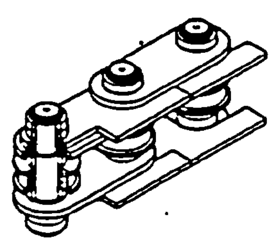
Rear end link with special link



90 D : Duplex



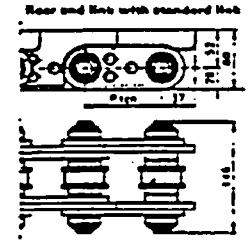
TYPE SG with rollers



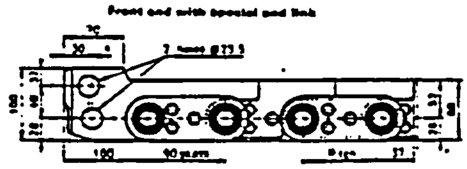
Characteristics		Pitch 90 mm 90 SG
Horizontal pushing force *	daN	9 000
Weight per meter	kg	22
Number of links per meter	..	11

* For 1 meter of chain unguided

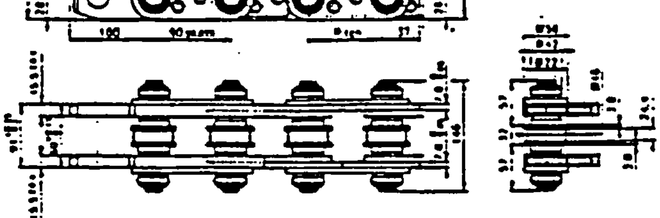
Rear end link with standard link



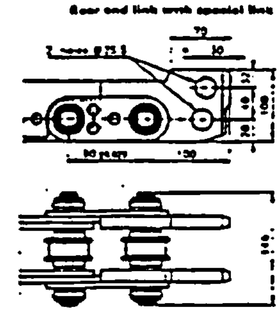
Front end with special end link



90 SG : Single with rollers



Rear end link with special link



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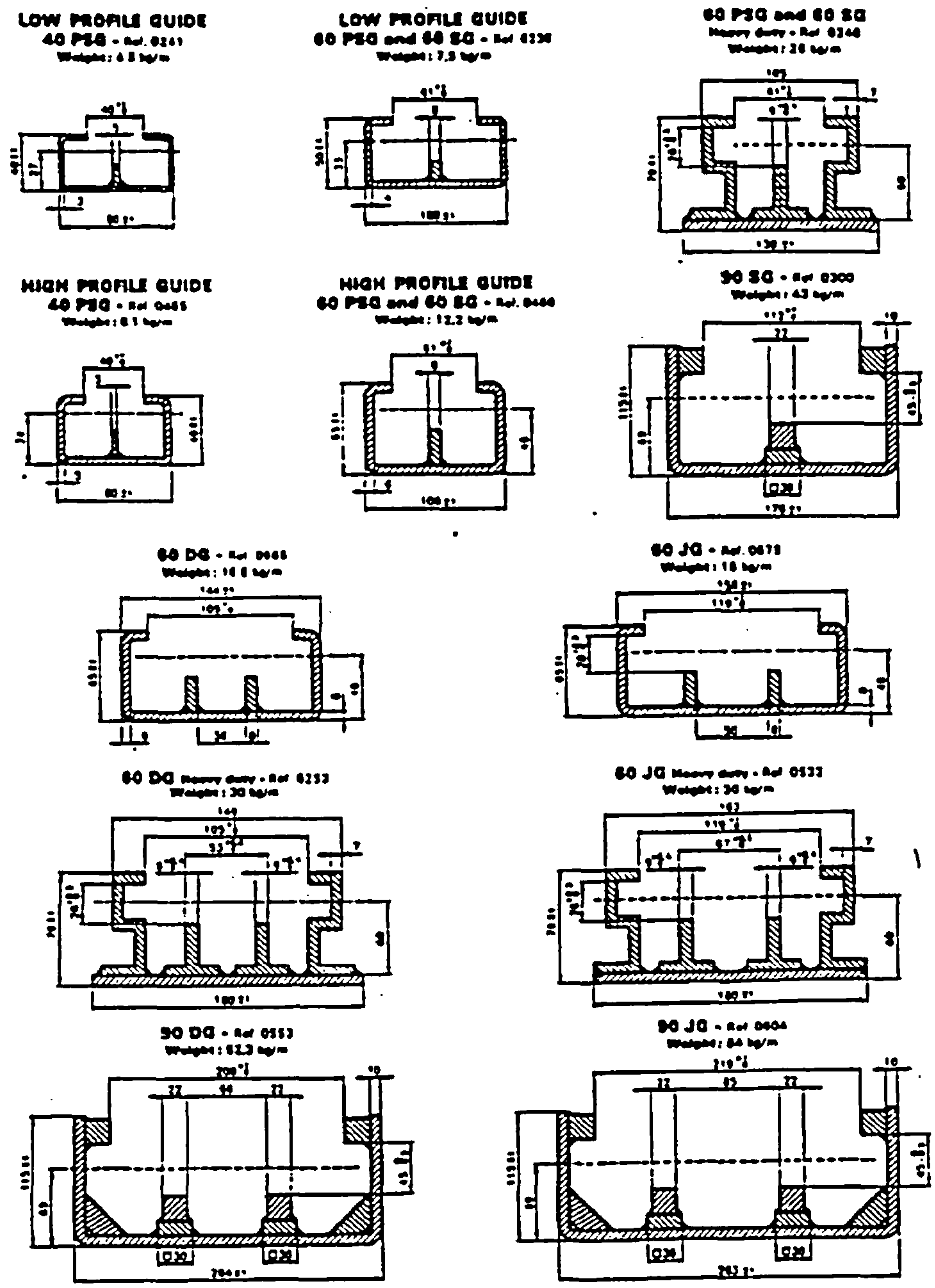
GUIDES for rigid pushing chains with rollers
PITCHES 40 - 60 and 90 mm

SPECIFICATION SHEET
PAGE 8



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SPECIFICATION SHEET PAGE 9	90° and 180° DRIVE HOUSINGS for SINGLE CHAINS - PITCHES 40 - 60 and 90 mm

GENERAL

Characteristics		Pitch 40 mm	Pitch 60 mm	Pitch 90 mm
Length of chain developed for a revolution of the shaft	mm	240	360	540
Weights: Single chain				
Cast iron drive housing	kg	13	35	120
Aluminium drive housing	kg	6.5	25	

A drive shaft supported by 2 bearings is keyed into the twin pinions which engage with the drive rollers on the chain's side.

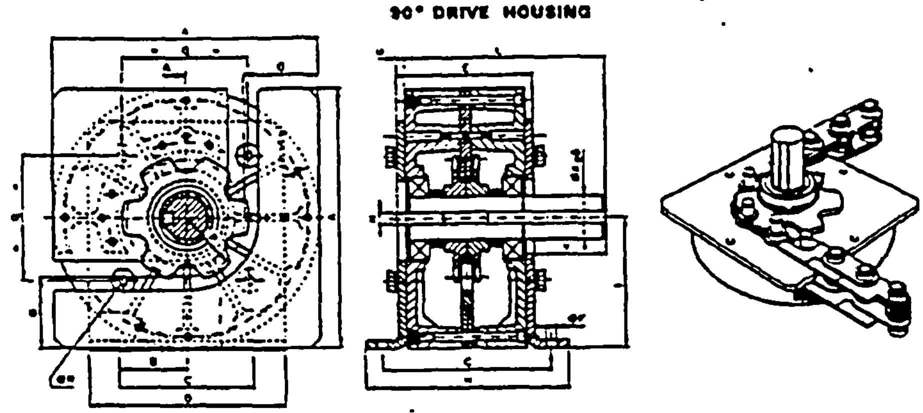
Two profiled plates serve the double function of guiding the chain through the drive housing and absorbing the reaction of the applied force.

The profile of the plates determines the entry and exit angles into and out of the drive housing.

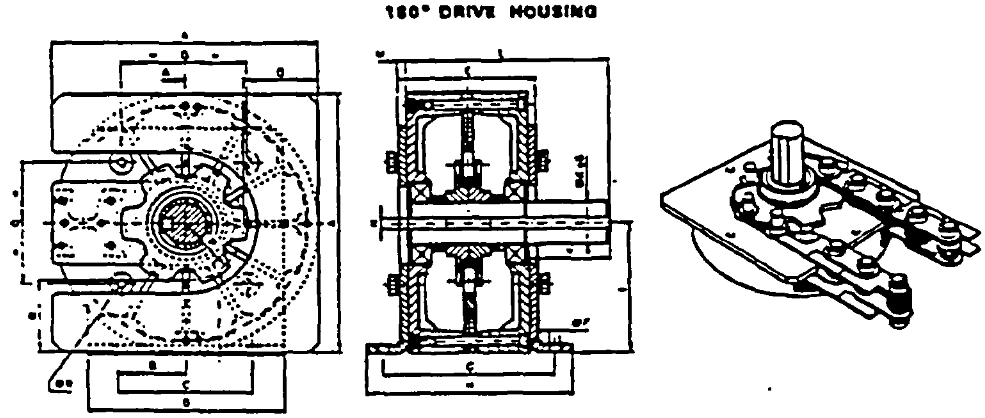
The material specifications are determined by the nature of the application.

The drive housings are supplied with fixing brackets, which can be attached in any one of four positions at 90° (see page C1).

90° DRIVE HOUSING



180° DRIVE HOUSING



DIMENSIONS

Type	Pitch	A	B	C	D	E	ØF	G	H	I	ØK g6	L	M	N	O	P	Q	ØR	V
Single	40	200	90	120	150	112	4 x 6.5	140	164	101	25	146	8	8 x 7	60	40	100	M8	46.8
Single	60	270	70	140	200	138	4 x 11	170	204	138	45	200	8	14 x 9	75	60	130	M10	82
Single	90	400	150	300	250	208	6 x 17	300	250	202	70	301	19	20 x 12	110	90	184	M10	147.5

Shaft end may be left or right hand

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Rigid Chain Engineering

MOST FREQUENTLY ASKED QUESTIONS ON SERAPID CHAIN

Please describe the chain:

There are two main components in a Serapid Rigid chain system:
- The chain itself,
- The drive housing.

The chain is composed of:
- Side plates, which are fine blanked for precision tolerance. The side plates interlock with each other and have mating surfaces which are flat, and which provide the push force. These mating surfaces are called heels.
- Shaft pins, and various rollers which are CNC machined, heat treated, then ground.

The drive housing is composed of:
- A drive shaft on sealed bearings,
- Drive sprockets, 1 to 6 depending on chain size,
- Reaction plates made out of high grade steel, precision machined,
- Cast housing pair for assembly of the above.

How does it work ?

Imagine holding a bar of steel in your hand. That is how the Serapid chain feels with the heels (butting surface of links) at the bottom. Its own weight keeps it straight and rigid. The chain stays rigid under its own weight, or the applied load, because of the locking moment that locks the links together.
Now pick up the free end and roll it back. It coils like a conventional chain. This is what happens through the drive housing. The chain is coiled 90 degrees or 180 degrees to store it out of the way.
As the chain pushes, it tends to walk away from the sprockets. This is why the chain is contained by steel reaction plates inside the drive housing. The reaction plates are designed to take up the reaction to the pushing force.

What are the two holes in the front link for ?

A pin is installed in each hole to attach the front of the chain to the load.
The pin which is in line with the rest of the chain pins is used for pulling the load.
The other pin, which is in line with the chain heels, is used for pushing.

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The purpose of the pushing pin is to ensure that the chain locks up during pushing, by ensuring that the reaction from the load comes back along the chain heels.

Does it need a guide ?

No. The chain is designed to operate without a guide, as long as the load locks up the chain. The chain is designed to excert a force along its heels when pushing, and along its pins when pulling. No side load on the chain is allowed.

A guide is only required if:

- The chain may be subject to side or excentric loads,
- The load is not centered on the chain, or is difficult to locate,
- The reaction force from the load (when pushing) is not along the chain heels,
- The travel is too long in relation to chain size. For exemple, for 10 feet of travel, using a guide may allow the use of a 40 pitch chain, whereas without a guide, a 60 pitch would be required.

Note: Guides when used do not have to run the length of the stroke. For exemple, for 10 feet of travel, 5 feet only could be guided at the begining of the stroke, or 5 sections of 1 foot long guides could be used at rendom along the stroke, etc...

How much can it push ?

1 daN = 2.2 lbs.

Nominal Push/pull forces range from 110 lbs for the size 25 (plastic) chain, to 10,000 lbs for the 90J chain.

The smallest steel chain is the size 40PS, with a nominal push/pull force of 1650 lbs.

What is meant by nominal is:

- 10" of stroke,
- Chain supported,
- Chain unguided,
- No shocks.
- Low speed (up to 10 feet/minute).
- Intermittent duty.
- Horizontal translation.

Longer strokes, higher speeds, shocks present, heavy use, etc... all reduce the nominal ratings.

All above rating parameters accumulate.

Please call the factory for application engineering assistance.

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Some rules of thumb:
When the chain is guided, nominal capacity remains regardless of stroke.
In lifting applications, cut nominal capacity in half.
When the stroke is doubled, cut the chain capacity in half.
When the chain is unguided and unsupported, cut the capacity by a third.

How fast can I run this chain ?

There is no actual limit to the speed of the chain.
But like any mechanical system, speed means faster wear.
Some precautions can be taken to improve the wearability of the chain and components in high speed applications, like ramping the speed up and down to avoid shocks.
Please consult factory for application engineering assistance.

How do I loop this chain for storage ?

Please look at the first page of our specification sheets. Shown are typical storage configurations, or chain returns.
Simply, the chain must be stored where space is available.
The most common chain returns are # 2, 5, and 8. All these chain returns have a single loop storage, and are free. They offer the advantage of reduced storage space, and yet do not require a guided storage magazine. The end of the chain is simply attached back to the drive housing, and the chain self supports itself.
Please see page 12 of our spec. sheets for dimensions.
If the space available is still too small, then a multiple loop storage magazine is required.

Does the chain need lubrication ?

Usually not. An occasional oil spray helps keep the chain clean and free from rust.
In high speed and/or high cycles applications, it may be necessary to lubricate for heat dissipation and reducing friction between rolling/sliding components.

How do I select a chain.?

The main parameters of chain selection are:
- What is the force required to move your load,
- How far you want to move it,
- How fast you want to move it.

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Please check other factors involved in determining actual push force, as shown in "how much can it push" paragraph.
Please fill out our application questionnaire, or call the factory. Our Application Engineering Dept is always available to assist you.
Following above "rules of thumb" will also work, but please verify your selection with our Application Engineering Dept.

What torque do I need ?

Once the thrust (push/pull) force is determined, and the chain selected,
torque is (in.lbs) : force (lbs) x sprocket pitch (in.) : 0.8 (drive efficiency).
In rolling load applications, please remember to include acceleration and decel. loads.

How do I calculate Serapid shaft speed (RPM) ?

Translation speed (feet/mn) x 60 : pitch perimeter (feet) = Shaft RPM.
Pitch perimeters:
Size 25 chain: 0.673 feet.
Size 40 chain: 0.821 feet.
Size 60 chain: 1.236 feet.
Size 90 chain: 1.854 feet.

What motor horsepower do I need ?

HP at the Serapid drive shaft = Torque (in-lbs) x Serapid shaft speed (RPM) : 63000.
To obtain input HP, the above figure must be devided by reducer/motor efficiency.

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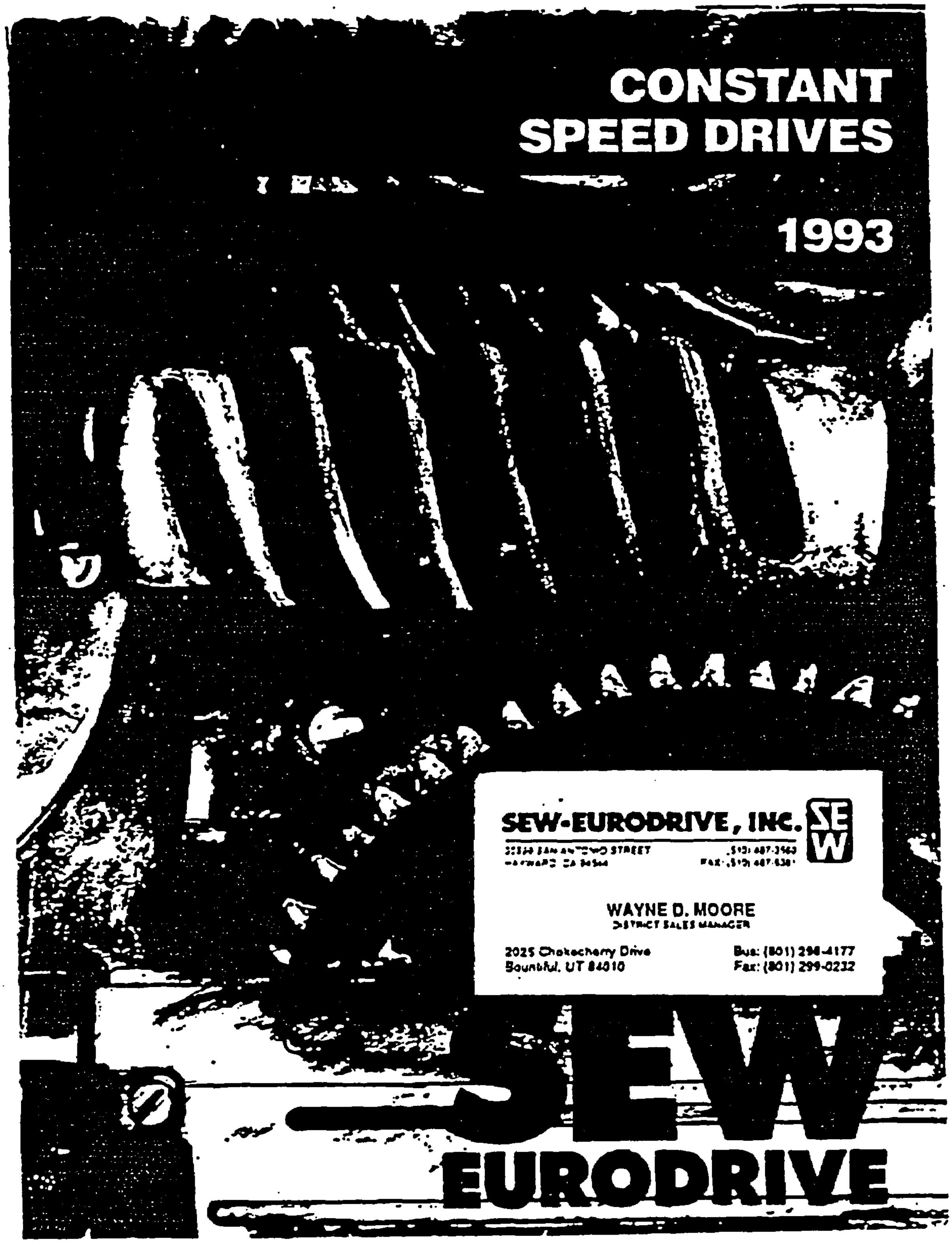
SECTION 6

SEW-EURODRIVE, INC., CONSTANT SPEED DRIVES, CATALOG CS993 USA


CONSTANT SPEED DRIVES
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WAYNE D. MOORE
DISTRICT SALES MANAGER
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Bountiful, UT 84010
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Fax: (801) 299-0232
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Service Factors Using AGMA Criteria

SEW-Eurodrive gear units may be service factored using criteria set forth in the various AGMA Standards.

For a) Parallel helical (Models R and FA) gearmotors
b) Right angle helical-bevel (Model K) gearmotors

AGMA uses service classes I, II, and III, which are based on:

Class I: Steady loads not exceeding normal rating and 8-10 hours running time per day.
Service Factor 1.0 minimum

Class II: a. Steady loads not exceeding normal rating and 24 hours running time per day.
b. Moderate shock loads, not exceeding 1.25 × Rated Load Torque running 8-10 hours per day.
Service Factor 1.4 minimum

Class III: a. Moderate shock loads, 1.25 × Rated Load Torque and 24 hours running time per day.
b. Heavy shock loads, exceeding 1.25 × Rated Load Torque and 8-10 hours running time per day.
Service Factor 2.0 minimum

Reference AGMA Standard 6019-E89 for Service Class listings by application.

motors

For: a) Parallel helical (Model R and FA) reducers
b) Right angle helical-bevel (Model K) reducers
c) Right angle helical-worm (Model S) reducers and gearmotors

AGMA uses service factors for electric motors, turbines, and hydraulic motors as listed by the chart below.

In the chart, the reducer loading may be classified as follows:

(1) Uniform Load. Recurrent shock loads do not exceed the nominal specified input or prime mover power.

(2) Moderate Shock Load. Recurrent shock loads do not exceed 1.25 × the nominal specified input or prime mover power.

(3) Heavy Shock Load. Recurrent shock loads do not exceed 1.50 × the nominal specified input or prime mover power.

(4) Extreme Shock Load. Recurrent shock loads do not exceed 1.75 × the nominal specified input or prime mover power.

NOTE: The magnitude of any recurrent shock loads should be estimated or determined through test by the system designer. Recurrent shock loads can be of such a short duration that they may not be reflected in motor amperage readings. In these cases actual loads are usually determined by strain gaging the driven shaft of the machine.

Duration of Service (Hours per Day)	Uniform Load	Moderate Shock	Heavy Shock	Extreme Shock
Occasional .5 hour	—	—	1.00	1.25
Less than 3 hours	1.00	1.00	1.25	1.50
3-10 hours	1.00	1.25	1.50	1.75
Over 10 hours	1.25	1.50	1.75	2.00

When the prime mover is a single or multi-cylinder engine, the service factors must be modified by the following:

Steam and Gas Turbines, Hydraulic or Electric Motor	Single Cylinder Engines	Multi-Cylinder Engines
1.00	1.50	1.25
1.25	1.75	1.50
1.50	2.00	1.75
1.75	2.25	2.00
2.00	2.50	2.25
2.25	2.75	2.50
2.50	3.00	2.75
2.75	3.25	3.00
3.00	3.50	3.25

Starting conditions where peak loads exceed 200% of rated load and applications with frequent starts and stops require special load analysis.

Service Factor listings by application may be found in:

AGMA 6010-E88 for Models R, FA and K reducers
AGMA 6034-A87 for Model S reducers and gearmotors

SEW EURODRIVE

Helical-Bevel Gearmotors

Output Power P_a HP	Output Speed n_a rpm	Service Factor	Torque T_a lb-in	OHL F_{Ra} lb	Ratio i	Model Gear	Model Motor	Dimension Pages K	KF	KA	KH	KAF
5.0	77	2.0	4090	1290	21.79	K76	DT100L4	245	257	267		280
	68	1.7	4630	1290	24.73	K76	DT100L4	245	257	267		280
	61	1.6	5160	1290	27.66	K76	DT100L4	245	257	267		280
	54	1.4	5830	1270	31.20	K76	DT100L4	245	257	267		280
	47	1.2	6700	1240	35.51	K76	DT100L4	245	257	267		280
	47	2.4	6700	4580	35.65	K86	DT100L4	248	258	268		281
	42	2.1	7500	4670	39.54	K86	DT100L4	248	258	268		281
	41	1.1	7680	1200	41.45	K76	DT100L4	245	257	267		280
	37	1.9	8510	4720	45.37	K86	DT100L4	248	258	268		281
	31	2.8	10200	6070	54.83	K96	DT100L4	248	259	268		282
	30	1.5	10500	4720	55.55	K86	DT100L4	248	258	268		281
	27	2.3	11700	6070	62.18	K96	DT100L4	248	258	268		282
	25	1.3	12600	4720	65.90	K86	DT100L4	248	258	268		281
	23	1.2	13700	4720	74.26	K86	DT100L4	248	258	268		281
	23	2.0	13700	6070	74.06	K96	DT100L4	248	259	268		282
	20	1.0	15800	4720	85.65	K86	DT100L4	248	258	268		281
	19	1.8	16600	6070	87.55	K96	DT100L4	248	259	268		282
	17	1.5	18500	6070	98.00	K96	DT100L4	248	259	268		282
	16	2.7	19700	10600	104.11	K106	DT100L4	248	259	269		282
	15	1.3	21000	6070	111.63	K96	DT100L4	248	259	268		282
	13	1.1	24200	6070	125.01	K96	DT100L4	248	259	268		282
	13	2.2	24200	10600	130.16	K106	DT100L4	248	259	269		282
	13	2.2	24200	10600	91.71	K106	DV132M8	248	259	269		282
	12	2.3	23800	10600	140	K106R82	DT100L4	251	263	274		
	11	1.9	28600	10600	104.11	K106	DV132M8	248	259	269		282
	10	1.9	28600	10600	161	K106R82	DT100L4	251	263	274		
	9.0	1.7	31700	10600	184	K106R82	DT100L4	251	263	274		
	7.5	1.4	38200	10600	226	K106R82	DT100L4	251	263	274		
	6.0	1.1	47600	10600	287	K106R82	DT100L4	251	263	274		
	6.0	1.9	47800	15400	292	K126R82	DT100L4	252	264	275		
	5.5	2.8	51700	20200	298	K156R92	DT100L4	253	265	278		
	4.5	1.4	63700	15400	358	K126R82	DT100L4	252	264	275		
	4.5	2.3	63200	20200	373	K156R92	DT100L4	253	265	278		
	4.0	1.3	72100	15400	400	K126R72	DT100L4	252	264	275		
	4.0	1.3	71800	15400	425	K126R82	DT100L4	252	264	275		
	3.8	1.9	75100	20200	444	K156R92	DT100L4	253	265	278		
	3.2	1.0	90300	15400	524	K126R72	DT100L4	252	264	275		
	3.1	1.8	92000	20200	545	K156R92	DT100L4	253	265	278		
	3.0	2.4	94500	27000	558	K166R92	DT100L4	254			277	
	2.4	1.2	118500	20200	699	K156R92	DT100L4	253	265	278		
	2.2	1.7	129700	27000	755	K166R92	DT100L4	254			277	
	2.0	1.0	142200	20200	858	K156R92	DT100L4	253	265	278		
	1.8	1.4	158500	27000	921	K166R92	DT100L4	254			277	
	1.6	2.0	177300	33700	1066	K186R92	DT100L4	255			278	
	1.5	1.2	190800	27000	1088	K166R92	DT100L4	254			277	

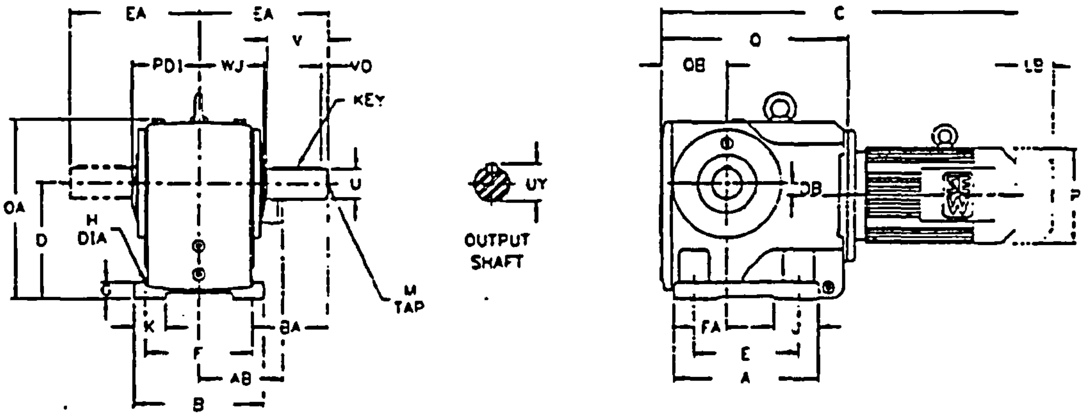
NOTES: Models 46 - 156 also available as: Flange mount, add letter F (e.g. KF86 DT71C4); Shaft Mount, add letter A (e.g. KA86 DT71C4); Shaft/Flange mount, add letters AF (e.g. KAF86 DT71C4). Models 166, 186 with shrink disc connection, add letter H (e.g. KH186 DV180M4). Overhung loads (OHL) apply only for K and KF gearmotors and are at the shaft midpoint.

Consult Assembly Center for Dimension Pages not listed.

See Page 332 for weights.

SEW EURODRIVE

Helical-Bevel Gearmotors
Foot Mounted



Gearcase

Model		A	B	BA	D	DB	E	EA	F	FA	G	H	J	K	OA
K86	inch	10.24	9.06	5.91	$8.35^{+0}_{-.02}$	0.80	7.09	9.45	7.09	2.17	1.26	0.87	3.15	2.17	13.31
	metric	260	230	150	$212^{+0}_{-.5}$	20.4	180	240	180	55	32	22	80	55	338
K96	inch	12.99	11.42	6.73	$10.43^{+0}_{-.04}$	1.11	9.45	11.46	9.45	2.95	1.50	1.02	3.94	2.76	16.26
	metric	330	290	171	$265^{+0}_{-.1}$	28.3	240	291	240	75	38	26	100	70	413
K106	inch	15.35	13.39	8.35	$12.40^{+0}_{-.04}$	1.97	11.02	13.66	10.63	3.74	1.65	1.30	4.33	3.54	19.45
	metric	390	340	212	$315^{+0}_{-.1}$	50	280	347	270	95	42	33	110	90	494

Gearcase

Model		PD1	O	OB	WJ
K86	inch	4.92	13.94	4.88	4.53
	metric	125	354	124	115
K96	inch	5.91	16.89	6.02	5.71
	metric	150	429	153	145
K106	inch	6.89	20.12	7.48	6.69
	metric	175	511	190	170

Output Shaft Inch Series/*Optional Metric Series*

Model		U	UY	V	VO	Key	M
K86	inch	$2.375^{+0}_{-.001}$	2.65	4.72	0.71	5/8 x 5/8 x 3 1/4	—
	metric	*$60^{+.030}_{+.011}$*	*64*	*120*	*10*	*18 x 11 x 100*	*DM20 x 42*
K96	inch	$2.875^{+0}_{-.001}$	3.20	5.51	0.97	3/4 x 3/4 x 3 1/2	—
	metric	*$70^{+.030}_{+.011}$*	*74.5*	*140*	*15*	*20 x 12 x 110*	*DM20 x 42*
K106	inch	$3.625^{+0}_{-.001}$	4.01	6.69	1.03	7/8 x 7/8 x 4 5/8	—
	metric	*$90^{+.035}_{+.013}$*	*95*	*170*	*15*	*25 x 14 x 140*	*DM24 x 50*

Motor

Model		DT 80	DT 90	DT 100	DV 112M	DV 132S	DV 132M	DV 132ML	DV 160M	DV 160L	DV 180	DV 200	DV 225
	AB	5.43	6.73	6.89	7.40	7.40	9.21	9.21	9.21	10.75	10.75	12.01	12.17
		138	171	175	188	188	234	234	234	273	273	305	309
	LB	2.52	3.35	3.35	3.15	3.15	4.41	4.41	4.41	6.14	6.14	6.14	6.14
		64	85	85	80	80	112	112	112	156	156	156	156
	P	5.71	7.76	7.76	8.70	8.70	10.83	10.83	10.83	13.03	13.03	15.51	15.51
		145	197	197	221	221	275	275	275	331	331	394	394
K86	C	23.31	24.06	26.02	27.32	29.09	29.96	32.32	32.32	34.17	—	—	—
		592	611	661	694	739	761	821	821	868	—	—	—
K96	C	—	26.77	28.78	30.08	31.85	32.72	35.08	35.08	36.93	39.76	—	—
		—	680	731	764	809	831	891	891	938	1010	—	—
K106	C	—	—	31.73	33.07	34.84	35.71	38.07	38.07	39.82	42.76	44.85	47.87
		—	—	806	840	885	907	967	967	1014	1086	1134	1216

Dimensions are inch/metric

Dimension AB is to motor conduit box

Dimension LB is for brake option

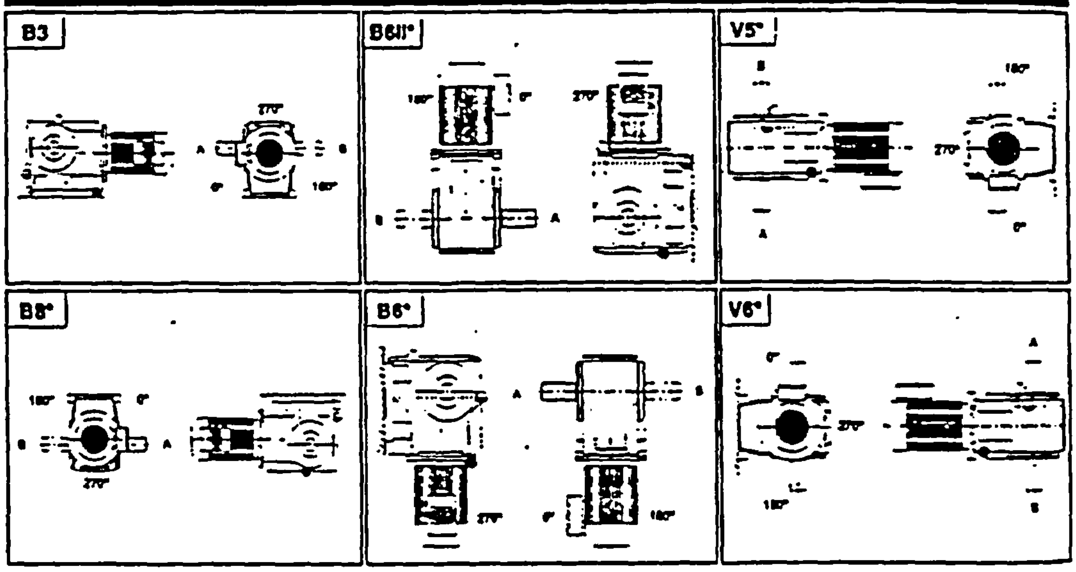
Eyebolts are removable

Dimensions subject to change without notice

SEW EURODRIVE

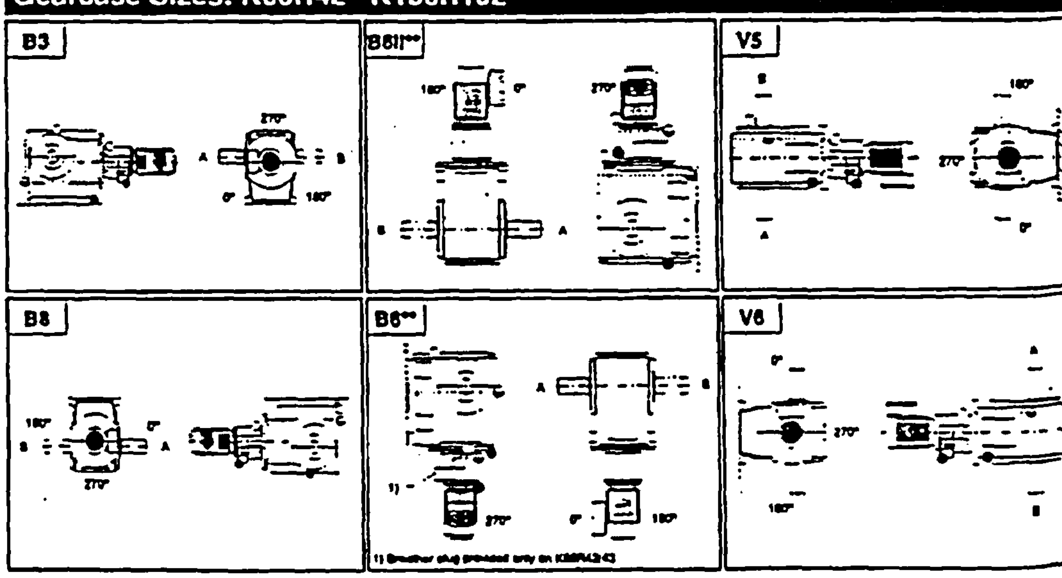
Mounting Positions Helical-Bevel Gear Units Foot Mounted

Gearcase Sizes: K66 - K106



* For gear unit sizes K76 - K106 with input speeds greater than 2700 rpm as well as for sizes K126 - K156 with input speeds greater than 1800 rpm, please refer to our engineering department.

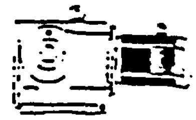
Gearcase Sizes: K66R42 - K156R102



** For gear unit sizes K156R92/83 and K156R102 with input speeds greater than 2700 rpm, please refer to our engineering department.

SEW EURODRIVE

Helical-Bevel Gear Units Weights



Listed below are weights for complete units less oil. Reducer weights are shown in the Gear Unit chart and combined reducer and motor weights are shown in the Gearmotor chart. For flanged and/or hollowshaft reducers as well as gearmotors add the flange and/or hollowshaft weight shown in the Gear Unit chart (a negative value must be subtracted). For brakemotors add the brake weight listed at the bottom of the Gearmotor chart.

Note: Oil weighs approximately 7.5 lbs/gallon (2 lb/liter). Reference Lubrication Sheet for volume of oil required. All weights in lbs

Gear Unit

Model	Reducer	Add for Flange	Add for Hollowshaft
K46	49	8	-3
K66	68	9	-2
K66R42	84	9	-2
K66R43	86	9	-2
K76	115	8	-5
K76R42	128	8	-5
K76R43	130	8	-5
K86	194	22	-9
K86R62	214	22	-9
K86R63	216	22	-9
K96	353	41	-17
K96R62	370	41	-17
K96R63	373	41	-17
K106	604	40	-31
K106R72	624	40	-31
K106R73	626	40	-31
K106R82	686	40	-31
K126	1045	88	-73
K126R72	1054	88	-73
K126R73	1056	88	-73
K126R82	1098	88	-73
K156	1592	132	-97
K156R92	1653	132	-97
K156R93	1664	132	-97
K156R102	1778	132	-97
K166	2350	—	-79
K166R92	2412	—	-79
K166R93	2423	—	-79
K166R102	2546	—	-79
K186	3571	—	-141
K186R92	3633	—	-141
K186R93	3644	—	-141
K186R102	3768	—	-141

Gearmotor

	DT				DV			
Model	71	80	90	100	112M	132S	132M	132M[illegible]
K46	63	70	82	97	—	—	—	—
K66	81	87	101	117	139	152	—	—
K66R42	98	105	117	132	—	—	—	—
K66R43	100	107	119	134	—	—	—	—
K76	126	134	147	163	188	201	240	262
K76R42	142	149	161	176	—	—	—	—
K76R43	144	151	163	178	—	—	—	—
K86	—	204	217	236	258	269	320	339
K86R62	227	233	247	263	285	298	—	—
K86R63	229	235	249	265	287	—	—	—
K96	—	354	369	388	412	425	472	484
K96R62	383	389	403	419	441	454	—	—
K96R63	386	392	406	422	444	—	—	—
K106	—	—	—	624	644	659	705	725
K106R72	635	643	656	672	697	710	758	771
K106R73	637	645	658	674	699	—	—	—
K106R82	—	696	709	728	748	761	812	831
K126	—	—	—	—	—	—	1107	1129
K126R72	1065	1073	1086	1102	1127	1140	1188	1201
K126R73	1067	1075	1088	1104	1129	—	—	—
K126R82	—	1108	1121	1140	1160	1173	1224	1243
K156	—	—	—	—	—	—	—	1818
K156R92	—	1654	1669	1688	1712	1725	1772	1794
K156R93	—	1665	1680	1699	1723	1736	1783	1805
K156R102	—	—	—	1808	1828	1843	1889	1908
K166	—	—	—	—	—	—	—	—
K166R92	—	2413	2428	2447	2471	2484	2531	2553
K166R93	—	2424	2439	2458	2482	2495	2542	2564
K166R102	—	—	—	2566	2586	2601	2647	2647
K186	—	—	—	—	—	—	—	—
K186R92	—	3634	3649	3668	3692	3705	3752	3774
K186R93	—	3645	3660	3679	3703	3716	3763	3785
K186R102	—	—	—	3788	3808	3823	3869	3889
Add for Brake	6.5	6.5	22	22	26.5	33	53	55
Add for Double Disc Brake	—	—	—	—	—	—	—	—

SEW EURODRIVE

Helical-Bevel Gear Units Lubrication

Each gear unit is supplied from the factory with the correct grade and quantity of lubricant for the specified mounting position. The following lubricants are supplied from our North American Facilities. Under special circumstances such as high or low ambient temperatures optional oils should be used.

Standard Oil

Gear Units[1]	Type	Manufacturer	Ambient Temperature °C
K46 - 156 K/KH166 - 186	Mobilgear 630 [M]	Mobil Oil Corp.	0 to +40
K46 - 156 K/KH166 - 186	Omala 220 [M]	Shell Oil Co.	0 to +40

[M] Mineral Oil

Optional Oil

Gear Units[1]	Type	Manufacturer	Ambient Temperature °C
K46 - 156 K/KH166 - 186	Mobilgear 629 [M]	Mobil Oil Corp.	-15 to +25
	Mobil SHC630 [S]		-25 to +80
	Mobil SHC629 [S]		-30 to +60
K46 - 156 K/KH166 - 186	Tivela SD460 [S]	Shell Oil Co.	-30 to +80

[M] Mineral Oil
[S] Synthetic Oil

For ball and roller bearings of gear units the following greases are recommended:

Mineral Grease

Type	Manufacturer	Ambient Temperature °C
Mobilux EP2	Mobil Oil Corp.	-20 to +40
Alvania Grease R3	Shell Oil Co.	-30 to +60

Synthetic Grease

Type	Manufacturer	Ambient Temperature °C
Mobiltemp SHC 32	Mobil Oil Corp.	-45 to +60

The approximate lubricant in US gallons/liters per mounting position is as follows:

Gear Unit[1]		Mounting Position						
		B3, H1, B3I	B8, B8I	B5	B5I	B5II	B6	B7
K	46	0.16/0.60	0.53/2.0	0.32/1.2	0.48/1.4	0.37/1.4	0.32/1.2	0.40/1.5
K	66	0.24/0.90	0.85/3.2	0.63/2.4	0.87/3.3	0.74/2.8	0.61/2.3	0.69/2.6
K	76	0.50/1.9	1.5/5.8	1.1/4.2	1.6/6.2	1.3/5.0	1.1/4.1	1.3/4.8
K	86	0.69/2.6	2.4/9.1	1.9/7.3	2.6/9.8	2.3/8.6	1.9/7.1	2.2/8.3
K	96	1.4/5.4	4.9/19	3.8/14	5.2/20	4.3/16	3.7/4.1	4.2/16
K	106	2.4/8.9	8.5/32	6.2/24	8.9/34	7.4/28	6.1/23	7.1/27
K	126	3.6/14	14/54	10/39	14/55	13/49	11/40	13/48
K	156	7.0/27	24/92	18/67	25/94	22/82	17/64	21/77
K/KH	166	8.2/31	31/118	—	31/118	—	—	—
K/KH	186	15/57	51/194	—	51/194	—	—	—

Gear Unit		Input Shaft Orientation	
		Horizontal	Vertical
R	42/43	0.08/0.30	0.26/1.0
R	62/63	0.16/0.60	0.53/2.0
R	72/73	0.34/1.3	0.96/3.7
R	82	0.74/2.8	2.1/8.0
R	92/93	1.3/4.8	3.4/13
R	102	1.8/6.7	5.4/21

For compound drives the R reducer requires its own oil filling as shown in the above chart.

Gear Unit[1]		Mounting Position						
		V1, V1I	V5	V6	H2	H3	H4	H5, H6
K	46	0.34/1.3	0.40/1.5	0.40/1.5	0.37/1.4	0.48/1.8	0.32/1.2	0.34/1.3
K	66	0.82/3.1	0.79/3.0	0.82/3.1	0.66/2.5	0.79/3.0	0.58/2.2	0.79/3.0
K	76	1.7/6.3	1.6/6.1	1.6/6.2	1.2/4.5	1.5/5.7	1.1/4.1	1.6/6.0
K	86	2.6/10	2.5/9.6	2.5/9.6	2.1/7.9	2.4/9.0	1.9/7.1	2.5/9.3
K	96	5.3/20	5.2/20	5.2/20	4.1/15	4.9/19	3.7/14	5.2/20
K	106	8.7/33	8.5/32	8.5/32	6.9/26	8.3/32	6.1/23	8.5/32
K	126	15/56	15/56	15/58	13/48	14/53	11/42	15/58
K	156	26/100	26/98	26/98	21/79	24/92	18/67	26/100
K/KH	166	25/95	—	—	—	—	—	—
K/KH	186	41/155	—	—	—	—	—	—

[1] Gear Unit size 46-156 also applies for KF, KA and KAF

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SEW EURODRIVE

Helical-Worm Gearmotors

Output Power P_a HP	Output Speed n_a rpm	Service Factor	Torque T_a lb-in	OHL F_{Ra} lb	Ratio i	Model Gear	Model Motor	Dimension Pages S	SF	SA	SAF
0.75	13	1.6	2250	2250	126.00	S62	DT80K4	363	368	376	383
	12	1.5	2400	2250	144.44	S62	DT80K4	363	368	376	383
	11	1.4	2580	2250	158.08	S62	DT80K4	363	368	376	383
	9.5	1.3	2930	2250	178.96	S62	DT80K4	363	368	376	383
	9.0	2.5	3200	2920	186.68	S72	DT80K4	364	369	377	383
	8.5	1.2	3220	2250	204.72	S62	DT80K4	363	368	376	383
	8.5	2.5	3390	2920	203.80	S72	DT80K4	364	369	377	383
	7.5	2.2	3780	2920	229.48	S72	DT80K4	364	369	377	383
	7.0	1.9	4590	2920	238	S72R42	DT80K4	365	372	379	
	6.0	1.6	5360	2920	275	S72R42	DT80K4	365	372	379	
	6.0	1.9	4570	2920	186.68	S72	DT80N6	364	369	377	383
	5.0	1.4	6240	2910	348	S72R42	DT80K4	365	372	379	
	5.0	1.7	5390	2920	229.48	S72	DT80N6	364	369	377	383
	4.5	1.9	7460	6560	366	S82R62	DT80K4	366	372	380	
	4.5	2.6	6510	6590	232.29	S82	DT80N6	364	369	378	384
	4.0	1.7	8270	6520	423	S82R62	DT80K4	366	372	380	
	3.9	2.4	7390	6580	282.38	S82	DT80N6	364	369	378	384
	3.7	1.1	8170	2660	461	S72R42	DT80K4	365	372	379	
	3.1	1.4	10500	6400	549	S82R62	DT80K4	366	372	380	
	3.1	2.7	11000	7870	549	S92R62	DT80K4	366	373	380	
	2.4	1.1	13200	6220	701	S82R62	DT80K4	366	372	380	
	2.4	2.1	13800	7870	698	S92R62	DT80K4	366	373	380	
	2.0	1.8	16500	7870	856	S92R62	DT80K4	366	373	380	
	1.5	1.4	21100	7870	1168	S92R63	DT80K4	366	373	380	
	1.3	1.2	24000	7870	1337	S92R63	DT80K4	366	373	380	
	1.0	1.6	23200	7870	1723	S92R62	DT80K4	366	373	380	
	0.78	1.2	29800	7550	2243	S92R62	DT80K4	366	373	380	
	0.62	1.0	35800	6850	2751	S92R62	DT80K4	366	373	380	
1.0	272	1.6	205	400	6.24	S32	DT80N4	362	367	374	381
	229	1.5	240	415	7.42	S32	DT80N4	362	367	374	381
	215	2.4	260	595	7.89	S42	DT80N4	362	367	374	381
	197	1.3	280	425	8.64	S32	DT80N4	362	367	374	381
	184	2.2	300	615	9.24	S42	DT80N4	362	367	374	381
	178	1.2	310	430	6.24	S32	DT90S6	362	367	374	381
	168	1.2	325	435	10.13	S32	DT80N4	362	367	374	381
	159	2.0	345	635	10.69	S42	DT80N4	362	367	374	381
	149	1.1	365	445	11.43	S32	DT80N4	362	367	374	381
	138	1.8	395	655	12.32	S42	DT80N4	362	367	374	381
	125	2.8	445	920	13.64	S52	DT80N4	363	368	376	382
	119	1.6	455	680	14.31	S42	DT80N4	362	367	374	381
	108	1.5	500	690	15.74	S42	DT80N4	362	367	374	381
	107	2.5	510	960	15.85	S52	DT80N4	363	368	376	382
	98	2.3	560	980	17.42	S52	DT80N4	363	368	376	382
	97	1.0	520	510	17.55	S32	DT80N4	362	367	374	381

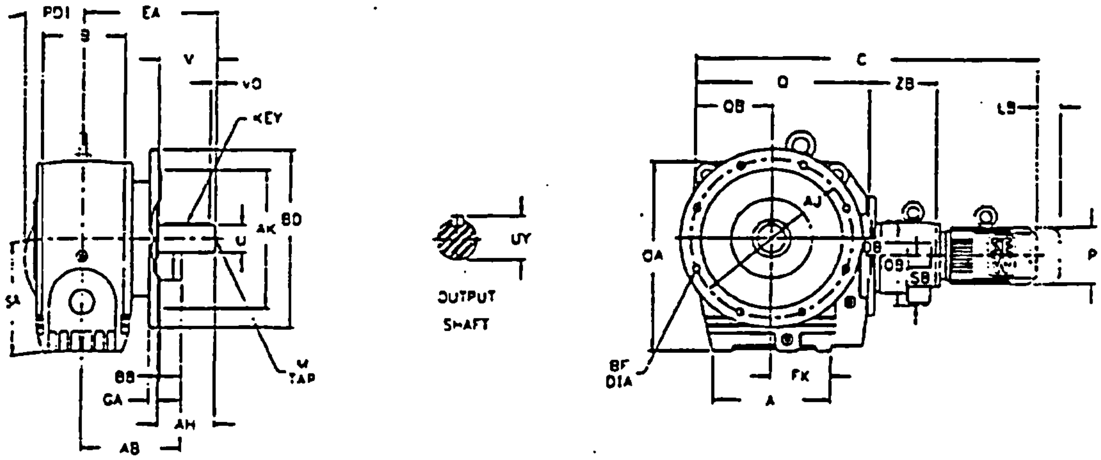
NOTES: All units also available as flange mount. Add letter "F" to model e.g. SF62 DT71C4.
All units also available as shaft mount. Add letter "A" to model e.g. SA62 DT71C4.
All units also available as shaft/flange mount. Add letter "AF" to model e.g. SAF62 DT71C4.
Overhung loads (OHL) apply only for S and SF gearmotors and are at shaft midpoint.

Consult Assembly Center for Dimension Pages not listed.

See Page 422 for weights.

SEW EURODRIVE

Helical-Worm Gearmotors
Flange Mounted



Gearcase

Model	A	B	DB	EA	FK	OA	OB	PD1	Q	QB	SA	SB	ZB
SF82R62 SF82R63	14.57 370	8.15 207	1.73 44	12.95 329	8.86 225	19.29 490	8.23 209	5.91 150	17.40 442	7.95 202	11.22 285	4.65 118	6.85 169

Output Shaft Inch Series/*Optional Metric Series*

Model	U	UY	V	VO	Key	M
SF82R62 SF82R63	2.875 +0/-.001 *70 +.030/+.011*	3.20 *74.5*	5.51 *140*	0.97 *15*	¾ x ¾ x 3½ *20 x 12 x 110*	— *DM20 x 42*

Flange

AH	AJ	AK	BB	BD	BF	GA
5.51 140	15.75 400	13.780 +0/-.001 350 +0/-.036	0.20 5	17.72 450	0.71 18	0.87 22

Motor

Model		DT 71	DT 80	DT 90	DT 100	DV 112M	DV 132S
	AB	5.43 138	5.43 138	6.73 171	6.89 175	7.40 188	7.40 188
	LB	2.52 64	2.52 64	3.35 85	3.35 85	3.15 80	3.15 80
	P	5.71 145	5.71 145	7.76 197	7.76 197	8.70 221	8.70 221
SF82R62 SF82R63	C	31.89 810	33.86 860	34.65 880	36.61 930	37.99 965	39.88 1013

Dimensions are inch/mm

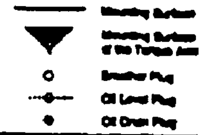
Dimension AB is to motor conduit box

Dimension LB is for brake option

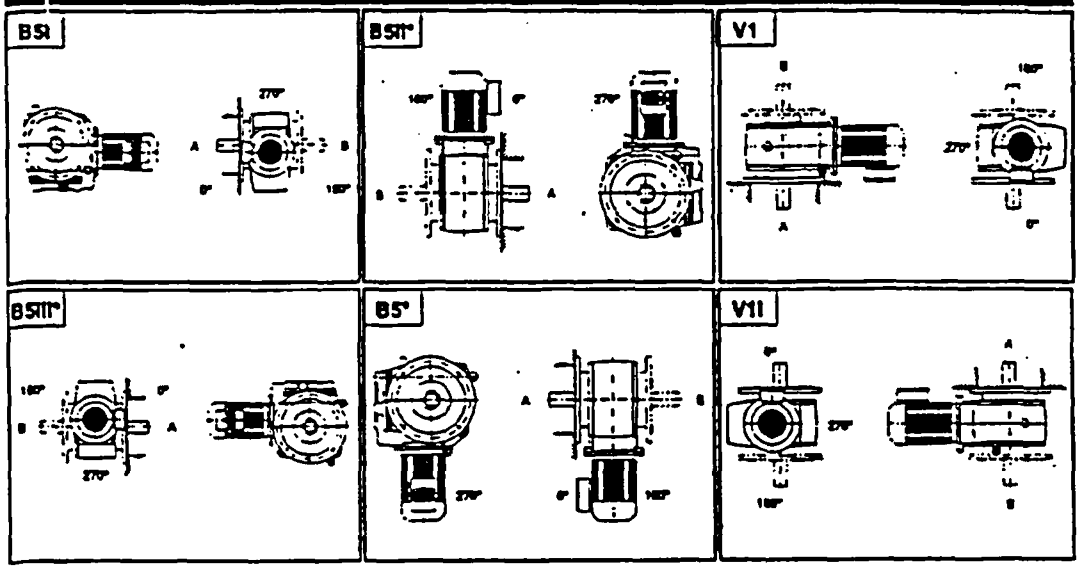
Eye bolts are removable

SEW EURODRIVE

Mounting Positions
Helical-Worm Gear Units
Flange Mounted

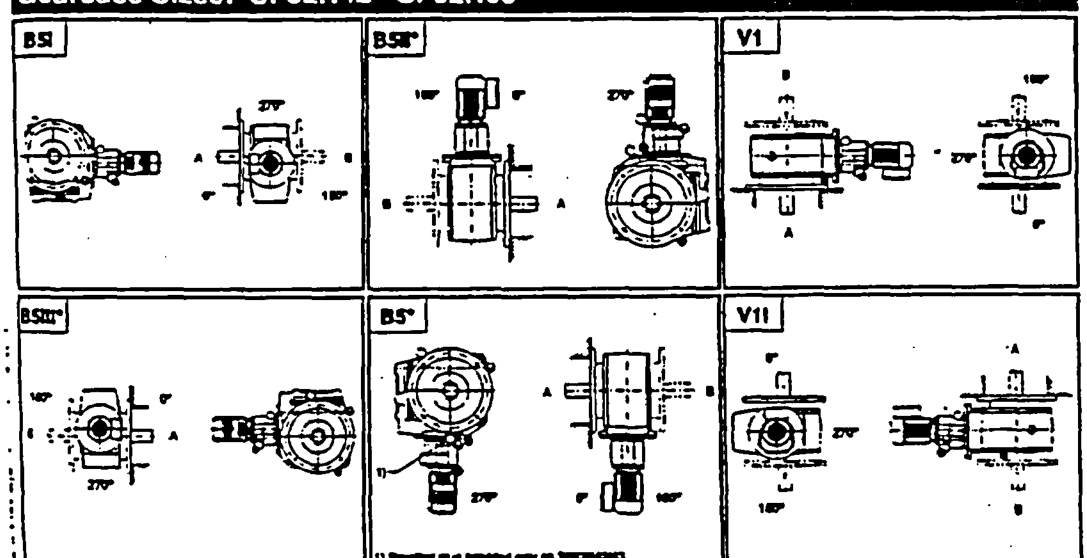


Gearcase Sizes: SF42 - SF92



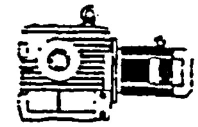
* For gear unit sizes SF72 - SF92 with input speeds greater than 2700 rpm, please refer to our engineering department.

Gearcase Sizes: SF62R42 - SF92R63



SEW EURODRIVE

Helical-Worm Gear Units Weights



Listed below are weights for complete units less oil. Reducer weights are shown in the Gear Unit chart and combined reducer and motor weights are shown in the Gearmotor chart. For flanged and/or hollowshaft reducers as well as gearmotors add the flange and/or hollowshaft weight shown in the Gear Unit chart (a negative value must be subtracted). For brakemotors add the brake weight listed at the bottom of the Gearmotor chart.

Note: Oil weighs approximately 7.5 lbs/gallon (2 lbs/liter). Reference Lubrication Sheet for volume of oil required.

All weights in lbs.

Gear Unit

Model	Reducer	Add for Flange	Add for Hollowshaft
S32	18	0	0
S42	24	8	7
S52	33	8	4
S62	60	15	13
S62R42	77	15	13
S62R43	79	15	13
S72	106	22	16
S72R42	121	22	16
S72R43	123	22	16
S82	209	34	-4
S82R62	236	34	-4
S82R63	238	34	-4
S92	366	35	-4
S92R62	388	35	-4
S92R63	300	35	-4

Gearmotor

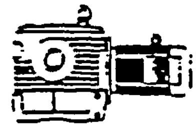
Model	DT 71	80	90	100
S32	29	33	44	—
S42	38	45	57	—
S52	49	55	66	86
S62	73	79	93	109
S62R42	91	96	110	125
S63R43	93	100	112	127
S72	117	125	138	154
S72R42	135	142	154	168
S72R43	137	144	156	171
S82	—	219	232	251
S82R62	246	253	269	285
S82R63	251	257	271	287
S92	—	367	382	401
S92R62	401	407	421	437
S92R63	403	409	423	439
Add for Brake	6.5	6.5	22	22

Gearmotor

Model	DV 112M	132S	132M	132ML	160M	160L	180M	180L	200
S32	—	—	—	—	—	—	—	—	—
S42	—	—	—	—	—	—	—	—	—
S52	—	—	—	—	—	—	—	—	—
S62	131	144	—	—	—	—	—	—	—
S62R42	—	—	—	—	—	—	—	—	—
S63R43	—	—	—	—	—	—	—	—	—
S72	179	192	240	—	—	—	—	—	—
S72R42	—	—	—	—	—	—	—	—	—
S72R43	—	—	—	—	—	—	—	—	—
S82	271	284	335	354	363	502	—	—	—
S82R62	307	320	—	—	—	—	—	—	—
S82R63	309	—	—	—	—	—	—	—	—
S92	425	438	485	507	518	659	716	747	878
S92R62	459	472	—	—	—	—	—	—	—
S92R63	461	—	—	—	—	—	—	—	—
Add for Brake	26.5	33	53	55	55	83	90	93	112
Add for Double Disc Brake	—	—	—	—	—	—	99	102	121

SEW EURODRIVE

Helical-Worm Gear Units Lubrication



Each gear unit is supplied from the factory with the correct grade and quantity of lubricant for the specified mounting position. The following lubricants are supplied from our North American Facilities. Under special circumstances such as high or low ambient temperatures optional oils should be used.

Standard Oil

Gear Units[1]	Type	Manufacturer	Ambient Temperature °C
S32	Mobil SHC 634 [S]	Mobil Oil Corp	0 to +60
S42 - 92	Mobilgear 636 [M]	Mobil Oil Corp	0 to +40
S42 - 92	Omala 680 [M]	Shell Oil Co.	0 to +40
S32 - 92	Tribol 800/680 [S]	Tribol	0 to +60

[M] Mineral Oil
[S] Synthetic Oil

Optional Oil

Gear Units[1]	Type	Manufacturer	Ambient Temperature °C
S42 - 92	Mobilgear 630 [M]	Mobil Oil Corp.	-15 to +25
S42 - 92	Mobil SHC634 [S]	Mobil Oil Corp.	0 to +60
S32 - 92	Mobil Glygoyle HE680 [S]	Mobil Oil Corp.	0 to +60
S42 - 92	Omala 200 [M]	Shell Oil Co.	-15 to +25
S32 - 92	Tivela SD460 [S]	Shell Oil Co.	-25 to +10

[M] Mineral Oil
[S] Synthetic Oil

For ball and roller bearings of gear units the following greases are recommended:

Mineral Grease

Type	Manufacturer	Ambient Temperature °C
Mobilux EP2	Mobil Oil Corp.	-20 to +40
Alvania Grease R3	Shell Oil Co.	-30 to +60

Synthetic Grease

Type	Manufacturer	Ambient Temperature °C
Mobiltemp SHC 32	Mobil Oil Corp.	-45 to +60

The approximate lubricant in US gallons/liters per mounting position is as follows:

Gear Unit[1]	Mounting Position							
	B3, B6I	B3I, B6II	B5	B5I	B5II	B5III	B6, B8I	B8
S 32	0.07/0.25	0.16/0.60	0.11/0.40	0.07/0.25	0.16/0.60	0.14/0.50	0.11/0.40	0.07/0.25
S 42	0.05/0.20	0.28/1.0	0.21/0.80	0.11/0.40	0.32/1.2	0.21/0.80	0.29/1.1	0.16/0.60
S 52	0.08/0.30	0.40/1.5	0.26/1.0	0.12/0.45	0.45/1.7	0.32/1.2	0.42/1.6	0.29/1.1
S 62	0.16/0.60	0.74/2.8	0.61/2.3	0.24/0.90	1.0/4.0	0.61/2.3	0.66/2.5	0.42/1.6
S 72	0.29/1.1	1.3/5.0	1.1/4.0	0.40/1.5	2.0/7.4	1.3/4.8	1.4/5.2	0.87/3.3
S 82	0.55/2.1	2.6/10	1.7/6.3	0.87/3.3	2.9/11	1.6/6.0	2.9/11	1.6/6.0
S 92	1.0/3.8	5.2/20	3.3/13	1.5/5.5	5.9/23	3.8/14	5.4/21	2.9/11

Gear Unit	Input Shaft Orientation	
	Horizontal	Vertical
R 42/43	0.08/0.3	0.26/1.0
R 62/63	0.16/0.6	0.53/2.0

For compound drives, the R reducer requires its own oil filling as shown in the above chart.

Gear Unit[1]	Mounting Position							
	V1A, V1/B	V1B, V1/A	V5, V5I	H1	H2	H3	H4	H5, H6
S 32	0.11/0.40	0.11/0.40	0.11/0.40	0.07/0.25	0.14/0.50	0.16/0.60	0.11/0.40	0.11/0.40
S 42	0.21/0.80	0.16/0.60	0.16/0.60	0.11/0.40	0.21/0.80	0.29/1.1	0.20/.075	0.18/0.70
S 52	0.29/1.1	0.21/0.80	0.24/0.90	0.12/0.45	0.29/1.1	0.40/1.5	0.28/1.0	0.24/0.90
S 62	0.61/2.3	0.55/2.1	0.42/1.6	0.24/0.90	0.61/2.3	0.92/3.5	0.55/2.1	0.53/2.0
S 72	1.2/4.4	1.1/4.0	0.82/3.1	0.40/1.5	1.1/4.0	1.6/6.1	0.92/3.5	0.95/3.6
S 82	1.6/6.1	1.5/5.7	1.5/5.6	0.87/3.3	1.5/5.7	2.7/10	1.6/6.0	1.6/6.1
S 92	3.1/12	2.8/11	2.8/11	1.5/5.5	3.3/13	5.4/20	3.1/12	3.2/12

SECTION 7

HILMAN INC., HILMAN ROLLER CATALOG AP/94

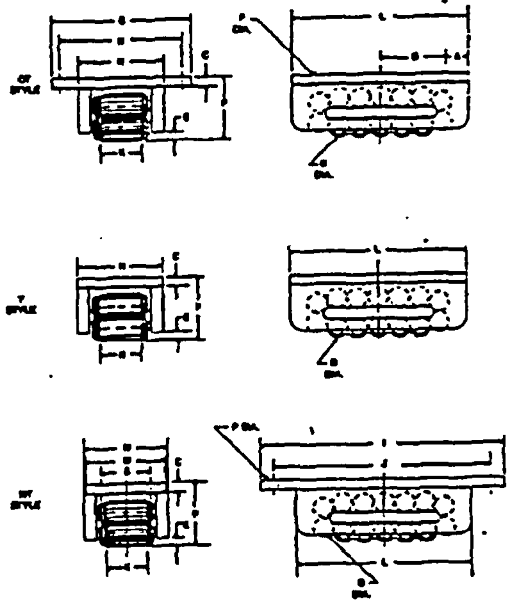
HILMAN ROLLERS

MOVE THE HEAVYWEIGHTS

OT, NT, T SERIES

• INDIVIDUAL ROLLERS .75-75 TONS

T Style NT Style OT Style



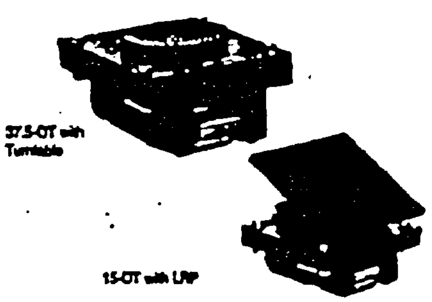
A variety of top plate styles, exceptionally strong materials of construction, and a wide range of modification possibilities make OT, T, and NT Rollers the true OEM leaders of the Hilman line! Rollers in this series are built for long life and a wide range of operating conditions.

For basic linear motion, these rollers are used as bearing slides, conveyors, guides, casters or dollies. Typical applications include the straight motion required for thermal expansion under a furnace or heat exchanger, as slides under nuclear shield doors, upside down as a conveyor in heavy die handling carts, in concrete forming for the casting and curing process as well in the launching of heavy segments, attached to a cradle for the launching of huge ships, or as a heavy duty wheel-like device to mobilize grandstand seats.

The OT type has an oversized top, the T type has a flush top, and the NT type has a long narrow top. The top plates add extra strength to the frame and variety to mounting capabilities, making one type ideal for cavity mount, bolt-on or weld mount. OT and NT Rollers are available with standard top plate sizes and mounting hole patterns, which can be modified to customer specifications. Hilman Elastomeric Preload Pads are available in either fabric incorporated or neoprene form for each roller model.

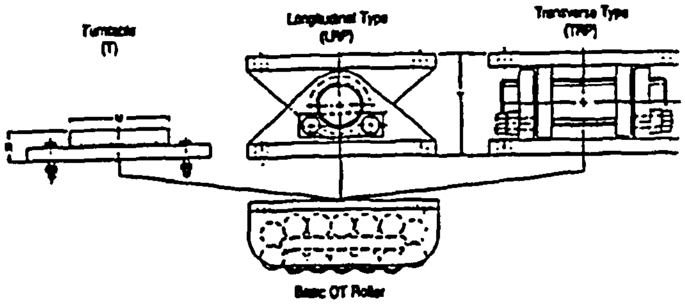
PRODUCT NUMBERS OT STYLE	NT STYLE	T STYLE	CAPACITY (TONS)	DIMENSIONS (INCHES/MM) A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	S	CONTACT ROLLS	LBS./KGS. OT	NT	T
.75-OT	.75-NT	.75-T	.75	3/4 19	2-1/2 64	3/8 10	11/16 17	3/8 10	3-1/8 79	7 178	5-1/2 140	8 203	7-1/4 184	2 51	6-1/2 165	3-1/2 89	3-5/8 92	5/16 8	2 51	6	12 5	12 5	11 5
2.5-OT	2.5-NT	2.5-T	2.5	3/4 19	2-1/2 64	3/8 10	15/16 24	3/8 10	3-5/8 92	7 178	5-1/2 140	9-1/2 241	8 203	2-7/16 62	6-1/2 165	4 102	4-1/8 105	9/16 14	2 51	4	21 10	19 9	17 8
5-OT	5-NT	5-T	5	3/4 19	3-1/4 83	3/8 10	15/16 24	7/16 11	3-1/8 79	8 203	6-3/4 171	11-3/4 298	10-1/8 257	2-7/16 62	8 203	4-1/2 114	4-7/16 113	9/16 14	3 76	5	22 10	21 10	19 9
8-OT	8-NT	8-T	8	3/4 19	3-1/4 83	3/8 10	15/16 24	7/16 11	3-1/8 79	8 203	6-3/4 171	11-3/4 298	10-1/8 257	3-5/16 84	8 203	5 127	5-5/16 135	9/16 14	3 76	5	25 11	23 10	23 10
15-OT	15-NT	15-T	15	1-5/8 41	3-11/16 94	5/8 16	1-3/16 30	1/2 13	3-7/8 98	10 254	8-1/2 216	14-3/4 375	12-11/16 322	2-3/4 70	10-5/8 270	5 127	5-3/16 132	11/16 17	2-5/16 59	5	25 11	23 10	23 10
20-OT	20-NT	20-T	20	1-5/8 41	3-11/16 94	5/8 16	1-3/16 30	1/2 13	3-7/8 98	10 254	8-1/2 216	14-3/4 375	12-11/16 322	4 102	10-5/8 270	6-1/2 165	6-7/16 164	11/16 17	3-1/2 89	5	49 22	49 22	49 22
37.5-OT	37.5-NT	37.5-T	37.5	2 51	5-1/2 140	3/4 19	1-5/8 41	5/8 16	5-1/2 140	12 305	10-1/2 267	21 533	18-3/4 476	3-1/2 89	15 381	7 178	7 178	13/16 21	4-1/4 108	6	121 55	114 52	108 49
50-OT	50-NT	50-T	50	3-3/4 95	5-1/2 140	3/4 19	1-5/8 41	5/8 16	5-1/2 140	12 305	10-1/2 267	21 533	18-3/4 476	3-1/2 89	18-1/2 470	7 178	7 178	13/16 21	4-1/4 108	8	147 67	142 64	138 63
75-OT	75-NT	75-T	75	1-1/4 32	9-1/4 235	1 25	1-15/16 49	1/2 13	6-3/4 171	14 356	11-1/2 292	27 686	34 864	3-5/8 92	21 533	7-1/2 191	7-3/8 187	1-1/16 27	5 127	7	241 109	227 103	213 97

• OT, NT SERIES TURNTABLES
• OT SERIES ROCKER TOPS



37.5-OT with Turntable

15-OT with LRP



Turntables are available for OT and NT Series rollers in capacities from 5 tons to 75 tons. Turntables bolt to the base roller unit and allow the roller to swivel or provide some freedom of lateral motion in Accu-Roll applications where tracks aren't perfectly parallel. Connection hardware is included.

Transverse (TRP) and Longitudinal (LRP) Rocker tops for the OT Series contain two attachment plates connected by a pivot pin assembly. Rockers are designed to be bolted to the base roller unit, then attached to either the load or the surface, enabling the roller to follow an incline, or to allow a load to be tilted, launched, or more accurately positioned. Connection hardware is included.

PRODUCT NUMBERS TURNTABLES / ROCKERS T-OT	T-NT	LRP	TRP	FITS ROLLER MODEL	DIMENSIONS IN (MM) R		U		T		WEIGHTS TURNTABLES / ROCKERS LBS.	KGS.	LBS.	KGS.
T-5-OT	T-5-NT	5-LRP	5-TRP	5-OT/NT	1-1/2	38	6	152	4-3/8	111	15	7	38	17
T-8-OT	T-8-NT	8-LRP	8-TRP	8-OT/NT	1-1/2	38	6	152	4-3/8	111	18	8	42	20
T-15-OT	T-15-NT	15-LRP	15-TRP	15-OT/NT	1-1/2	38	6	152	5	127	22	11	101	46
T-20-OT	T-20-NT	20-LRP	20-TRP	20-OT/NT	1-1/2	38	6	152	5	127	22	11	120	53
T-37.5-OT	T-37.5-NT	37.5-LRP	37.5-TRP	37.5-OT/NT	2-3/8	60	8	203	6-1/2	165	54	25	196	89
T-50-OT	T-50-NT	50-LRP	50-TRP	50-OT/NT	2-3/8	60	8	203	7	178	54	25	195	89
T-75-OT	T-75-NT	75-LRP	75-TRP	75-OT/NT	2-3/8	60	8	203	9-1/4	235	85	39	389	177