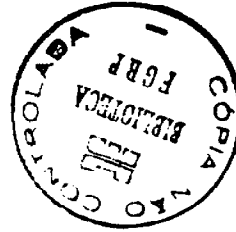


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SPECIFICATION FOR ELECTRIC OVERHEAD TRAVELING CRANES FOR STEEL MILL SERVICE



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ERAL

sity of operation of each individual mechanism of the crane.

- (3) To establish a meaningful mathematical relationship between the measure of intensity and the conventional engineering fatigue analysis concepts which are used to determine the adequacy of structural and mechanical components.

1.2.3 Crane Operation Data (Load Spectra).

The crane user is concerned with the loading conditions, the operating frequency and the operating cycle efficiency with respect to their functions within the production facility which the crane has to serve. Conveying this knowledge or estimate of the anticipated loading and duty cycles as realistically as possible to the crane manufacturer is the responsibility of the purchaser. This transfer of information is necessary so that the designer can properly match the fatigue strength of the crane components to the requirements of its intended operation. The numerical documentation of the crane operating requirements must be in such a form that it will provide the basic information by which the fatigue strength and the fatigue life design criteria of every component and structure of the crane can be derived. The basic operating data for each individual mechanism of the crane may be recorded in the Crane Operating Data Record, Form 1.00, and shall become part of the contract between the manufacturer and the user [see Appendix C and the Owner's Information Sheets (OIS) for procedure and form].

If the crane function is dedicated to a specific production routine, as is typical for ladle cranes, stripper cranes, soaking pit cranes, bucket cranes etc., the operating conditions can be determined from the number of operating cycles/hr, the magnitude of applied loads, the distance of motion and the number of operating hours per given time period. In other cases, such as a general service crane, the duty may become more difficult to determine because the crane is being used for a variety of services. Suitable values can be established based on the user's general crane experience, supported by an evaluation of required performance and by AISE crane duty guidelines (Appendix A) listing typical crane duty classifications for various categories of cranes. The user will also have to establish the service life requirements for the crane, considering technical, economic, safety and environmental (e.g. temperature, humidity, corrosive atmosphere, conductive or abrasive dust, etc.) factors, as well as the probability of obsolescence. The designer will then be able to evaluate the operating data furnished by the user and to determine the effects it will have on the design.

In order to determine if there was, in fact, a wide variation in the intensity of operation of steel mill cranes, a survey of the major North American steel companies was conducted in 1978. The information obtained (included in Appendix A) verified that, for all types of crane service, there is a wide range of actual usage from plant to plant, and emphasized the economy of designing a crane for the operating intensity to which it will be exposed.

1.2.4 Tests and Acceptance. Tests shall be made as specified on the information sheets; otherwise, the manufacturer's standard tests shall be made. In any case, the owner shall be notified sufficiently in advance, so that his representative may witness all tests.

Acceptance shall be subject to compliance with these specifications and information sheets to be determined by inspection after delivery, by results of tests required above, and upon the approval of the owner or his representative. Load tests shall be conducted on the owner's runway with the crane loaded to 125% of rated load, unless otherwise stated on the information sheets.

In case of disagreement between contractor and owner or his representative in regard to the interpretation of any specification or the compliance of the apparatus furnished with the requirements of this specification, the question shall be submitted to the engineering department of the owner's company for interpretation.

1.2.5 Workmanship, Material and Inspection. Workmanship and material shall be subject to the inspection of the owner or his representative at all times.

Weldments of carbon steel (except bridge girders) shall be stress relieved by uniformly heating in a furnace. Field welds shall likewise be stress relieved unless other means agreeable to the owner are specified on the information sheet. The temperature of the furnace when the weldment is placed in it shall not be over 300°F at the start and increased to 1200°F at a rate not exceeding 200°F/hr, then held at the temperature for 1 hr/in. of thickness of material. It shall then be cooled in the furnace at a rate (not exceeding 200°F/hr) to 500°F, before being removed from the furnace.

Weldments of alloy material shall be welded and stress relieved using a procedure specified by the owner.

1.2.6 Painting. All work shall be thoroughly cleaned of all loose mill scale, rust and foreign matter and then given two shop coats of specified or approved paint. All parts inaccessible after assembling, unless otherwise specified by the owner, shall be well painted before assembling, except that high tension friction-type bolted connections or welded work whose surfaces come into contact are not to be painted. The interior of all gear housings shall be painted with one coat of oil-resisting enamel.

The color and quality of the paint shall be as specified on the information sheets.

1.2.7 Safety Devices. All machinery or equipment furnished must be equipped by manufacturer or contractor with all proper safety devices and clearances to comply with the laws of the state and municipality in which it will be installed, the owner's safety requirements pertinent thereto, and, if stated on the owner's information sheets, any safety requirements peculiar to the owner's plant involved.

1.2.8 Clearances. Clearance between any part of the crane, building column, roof chord or other stationary structure shall be not less than that shown on the sketch

accompanying the information sheets. Accuracy of these sketches shall be the responsibility of the owner.

CODES, SPECIFICATIONS AND STANDARDS

The following shall be considered a part of this Report when information is not provided herein; where dual coverage exists, AISE Technical Report No. 6 shall govern but in no case shall the final action conflict with federal, state or governmental regulations.

- Association of Iron and Steel Engineers, Three Gateway Center, Suite 2350, Pittsburgh, PA 15222-1097
 - AISE Technical Report No. 1, D-C Mill Motor Standard, 1991
 - AISE Technical Report No. 8, Insulated Conductors for Crane and Mill Auxiliary Motors, May 1974
 - AISE Standard No. 11, Brake Standards for Mill Motors, September 1972
 - AISE Technical Report No. 13, Guide for the Design and Construction of Mill Buildings, 1991
- American Association of State Highway and Transportation Officials — "Specifications for Highway Bridges," twelfth edition, 1977, as amended 1978, 1979, 1980 and 1981
- American Gear Manufacturers Association — AGMA 218.01, December 1982
- American Welding Society — AWS D1.1, "Structural Welding Code" 1984 edition (except Section 8) and AWS D14.1, "Specification for Welding of Industrial and Mill Cranes and Other Overhead Material Handling Equipment" 1970 edition.
- American Society for Testing and Materials — Standards referred to herein by ASTM numbers such as A 36, A 441, etc., latest edition of each material specification.
- American Institute of Steel Construction — "Manual of Steel Construction" eighth edition, November 1, 1978, and Structural Steel Detailing, second edition, 1971.
- American National Standards Institute — ANSI Z210.0 (for converting units used in this Report to the Standard International System of Units)
- National Electrical Manufacturers Association — "NEMA Industrial Controls and Systems Standard."
- National Fire Protection Association — "National Electrical Code."

REFERENCES

- Cadile, J. V., "Classification of Cranes," Association of Iron and Steel Engineers, 1976
- Cadile, J. V., "Allowable Fatigue and Design Stress," 1976.
- Federation Europeenne de la Manutention, "Rules for the Design of Hoisting Appliances, Section I, Heavy Lifting Equipment," Second Edition

2 STRUCTURAL

2.1 General.

2.1.1 Scope. This Section applies to the structural design of welded bridge girders, outrigger trusses, trolley frames, end carriages, end ties, equalizers, end trucks, fabricated rope drums, gear housings, platforms and all other elements necessary to the strength and rigidity of the load carrying and auxiliary structural function of the crane.

2.1.2 Structural Details. In matters pertaining to the design and detailing of welded joints and related structural parts, reference should be made to Appendix C and to the applicable Sections of the AWS D-1.1 and D-14.1 as prepared by the American Welding Society.

2.1.3 Materials. Structural steel shall conform with the latest revision of ASTM Standard Specification A 36. Other steels may be used provided that the required properties, special welding procedures or other pertinent information is specified on the OIS.

2.2 Loads, Forces and Allowable Stresses.

2.2.1 Vertical Loads on Crane Bridge.

W_A = Weight of column, ram or other material handling device which is rigidly guided in a vertical direction during hoisting action, kips

W_B = Dead weight of bridge structure including all machinery and equipment permanently attached thereto or planned for future installation excluding track wheels, end trucks, saddles or end ties, kips

W_{BE} = Total dead weight of bridge structure including track wheels, end trucks, equalizers, saddles and end ties, kips

W_L = Lifted load, which is the total weight lifted by the hoist mechanism, including working load, all hooks, lifting beams, magnets or other appurtenances required by the service, excepting W_A , as defined above, kips

W_T = Weight of trolley including all machinery and equipment attached thereto but excluding hook block, kips

2.2.2 Impact. Vertical loads due to impact are to be added to the lifted load by the application of impact factors as follows:

- (1) Ladle cranes — $0.2 (W_L)$
- (2) Mill cranes — $0.3 (W_L)$
- (3) Clamshell bucket, magnet cranes, slab yard and billet yard cranes — $0.5 (W_L)$
- (4) Stripper, soaking pit and stacker cranes — the greater of $0.5 (W_L)$ or $0.3 (W_L + W_A)$

Additionally, bridge or trolley truck structures shall be locally designed for an impact factor of 25% of the wheel load, applied to any one wheel.

2.2.3 Horizontal Forces.

2.2.3.1 Horizontal Pushing Forces. All cranes with vertically guided loads shall be designed for horizontal forces applied at the bottom of the arm. This force shall be determined as shown in Sections 2.2.3.1.1 or 2.2.3.1.2:

2.2.3.1.1 When the Force is Applied in the Direction of Bridge Travel.

$$(1) 0.2 (W_A + W_T + W_{BE}) \times \frac{(\text{Number of Driven Bridge Wheels})}{(\text{Total Number of Bridge Wheels})} \quad (\text{Eq 1})$$

- (2) The horizontal force that will tilt the trolley in the direction of bridge travel when the length of the force lever arm is in the minimum position at which a force may be applied.

The lesser of the forces computed using (1) or (2) shall be used as the force applied in the direction of bridge travel.

2.2.3.1.2 When the Force is Applied in the Direction of Trolley Travel.

$$(1) 0.2 (W_T + W_A) \times \frac{(\text{Number of Driven Trolley Wheels})}{(\text{Total Number of Trolley Wheels})} \quad (\text{Eq 2})$$

- (2) The horizontal force that will tilt the trolley in the direction of trolley travel when the length of the force lever arm is in the minimum position at which a force can be applied.

The lesser of the forces computed using (1) or (2) shall be used as the force applied in the direction of trolley travel.

Horizontal forces on vertically guided columns are not to be considered as applied concurrently with horizontal inertia forces due to acceleration.

2.2.3.2 Horizontal Inertia Forces. All cranes shall be designed for horizontal longitudinal forces from the acceleration or deceleration during the movement of the crane along the runway as follows:

- Uniformly distributed load of 20% of the total weight of the crane bridge (less all structural and mechanical weight distributed in the vertical plane of the bridge runway such as track wheels, trucks, equalizers, saddles and end ties).
- Concentrated loads of 20% of the weight of the motor, cab, etc., are not to be assumed as distributed loads.
- 20% of the concentrated loads apportioned to the trolley wheel contact points (i.e., the trolley, hoist, hooks and maximum lifted load). These concentrated loads are to be positioned so as to produce maximum stress due to moment or shear in the girders.

All such longitudinal inertial forces shall be multiplied by the ratio:

$$\frac{\text{Number of Driven Bridge Wheels}}{\text{Total Number of Bridge Wheels}}$$

The moment of inertia of the entire box girder section about the vertical axis shall be used in calculating the stresses due to these horizontal longitudinal forces. In two-girder cranes, the total horizontal longitudinal load shall be proportionately divided between both girders, except in the case of stripper, pit or other fixed arm cranes for which both the horizontal and vertical force induced by tilting must be considered as acting on either girder alone. Consideration must also be given to the effects of stabilized reeving when it is used.

2.2.3.3 Skewing Forces. The bridge girders and end ties shall be designed as a continuous frame in the horizontal plane. The recommended procedure for evaluating skewing is given in the commentary. Frame analysis shall be used to determine the maximum moments and shears at critical locations in the frame due to horizontal inertia and skewing forces.

Note: This paragraph does not apply to cranes with pinned end connections such as ore bridges, semi-gantry and full gantry cranes.

2.2.3.4 Wind Loads. Wind loads on cranes that operate in exposed locations shall be calculated with consideration of geographic location, height above ground and shape of the individual components that make up the structure. In the calculation of these loads, the information in ANSI A58.1 should be followed. In-service wind shall be calculated as required under Section 2.2.9 of this Report, and shall have a magnitude equal to 25% of full wind load.

2.2.4 Bending Moment and Shear. Under vertical load, the bridge girder shall be considered as a simple beam with a span equal to the centerline-to-centerline distance between the runway rails. Inequalities in the distribution of vertical load to the trolley rails shall be considered.

In girders with less than two axes of symmetry, the shear center must be determined to apportion shears due to vertical or lateral load, or both, as well as to determine torsional moments. When the asymmetry is small the shear center may be assumed to be at the centroid of the cross-section.

2.2.5 Torsional Moment. The loads and forces creating torsional stress in the girders are:

- (1) Starting and stopping of the bridge drive motor. The twisting moment at each gear box base is the algebraic difference in input and output torques. Assume that the bridge motor generates a starting torque of 200% of the rated torque.
- (2) Overhanging loads on the side of a girder, such as footwalks, bridge drives, collector bars, cabs and controls. These moments shall be taken as the respective forces due to weight (for impact caused

by loose rail joints see Section 2.2.2) multiplied by the horizontal distances between the respective centers of gravity (or action line of force) and the shear center of the girder section

- (3) Horizontal forces acting eccentrically to the shear center of the girder. The twisting moments shall be considered as these forces multiplied by the vertical distance between the centerline of force and the shear center of the girder. For box girders with an area of the compression flange no more than 50% greater than that of the tension flange and with no greater difference between the area of the two webs, the shear center may be assumed to be at the centroidal axis of the cross-section.

The total twisting moment shall be the algebraic sum of the moments resulting from these loads. Secondary torsional stresses caused by eccentricity as a result of load deflection need not be considered.

2.2.6 Shear Stress. The maximum shear stress in the web of a box girder is the sum of the maximum shear stress due to the resultant shear force through the shear center plus the shear stress due to the torsional moment.

$$f_{v(\max)} = f_{vb} + f_{vt}, \text{ ksi} \quad (\text{Eq 3})$$

For a box girder symmetrical about the vertical axis, with webs each of thickness, t , the shear stress in the web plates due to a vertical resultant shear force, V , may be determined by the following equation:

$$f_{vb} = \frac{VQ}{(2I_x t)}, \text{ ksi} \quad (\text{Eq 4})$$

For unsymmetrical box girder sections the shear stress must be determined by a shear flow analysis.

The shear stress due to torsional moment in a box girder may be computed by the following equation:

$$f_{vt} = \frac{M_t}{(2At)}, \text{ ksi} \quad (\text{Eq 5})$$

An external torque applied to a box girder of uniform cross-section will be resisted by the two adjacent portions of the girder in the same proportion as shears due to a vertical concentrated load applied at the point of torque application. If a box girder has a nonuniform cross-section the distribution of applied torque will be in proportion to the torsional stiffness of the two segments as determined by a torsional analysis.

2.2.7 Stress Sheets. If the purchaser specifies on the OIS or specifications, stress sheets showing the loads, forces and stress calculations be provided, they shall be included with the prints submitted by the contractor to the purchaser for approval of design.

2.2.8 Platform Loads. In addition to the specified loads, unless otherwise designated by the owner, all platforms on traveling cranes shall be designed for 50 lb/ft² live load plus a concentrated load of 500 lb. The concentrated load

to any location on the platform and shall be at locations where it will cause the greatest stress. Port structures for heavy items such as panels, and air conditioners must be analyzed individually. Allowable stress reduction need be made for live loads. Platform live loads are not to be superimposed on bridge and trolley design live loads.

Design Load Combinations and Stress Factors.*

Load Combinations:	Stress Factors:*
Dead load	1.00 × allowable base stress (Table 1) not reduced for repeated loads
Impact	
Wind (to be considered if crane is not exposed to wind)	
Crane forces	
Dead load	1.00 × allowable base stress or fatigue stress range, whichever governs
Impact	
Horizontal inertia forces and centrifugal forces	
Dead load	1.50 × allowable base stress
Impact	
Crane forces	
Dead load	1.50 × allowable base stress
Hoist motor torque	
Dead load	1.33 × allowable base stress
Loaded trolley stored at end of track (if exposed)	
Dead load	1.00 × allowable base stress
Load (not including crane load)	
Reverse wind (if exposed)	
Crane forces	

In no case shall the allowable stress exceed $0.9 F_y$.

3.8 Collision Effects. The crane bridge structure shall be designed to absorb the collision forces as calculated in Section 3.8.

1 Basic Allowable Stresses.

2.2.11.1 Stress in Members. Basic allowable stresses in members made of ASTM A 36 steel are listed in Table 1. Other ASTM certified materials may be used. In addition, members and connections subjected to repeated loading must be designed for fatigue in accordance with provisions of Section 2.2.12.

Table 1 — Allowable Stresses (for A36 steel), ksi

(1) Minimum tensile strength F_u	58.0
(2) Minimum yield strength F_y	36.0
(3) Axial tension	
Except for pin-connected members, the lesser of	
0.60 F_y on the gross area	22.0
0.50 F_u on the effective net area ¹	29.0
For pin-connected members	
0.45 F_y on the net area ¹	16.2
(4) Axial compression	
As limited by the buckling provisions of Section 2.2.13	
(5) Bending	
Extreme fiber tension	
0.60 F_y on the net cross section	22.0
Extreme fiber compression	
As limited by the buckling provisions of Section 2.2.13	
Tension or compression on extreme fibers of solid round or square bars and solid rectangular sections bent about their weaker axis	0.75 F_y
(6) Shear	
0.40 F_y on pins and on the gross section of girder webs, except as limited by the buckling provisions of Section 2.2.13	14.4
(7) Bearing	
On diaphragms and other steel surfaces in contact	27.0
= 0.75 F_y	
On pins not subject to rotation	0.22 F_y
	8.0
On pins subject to relative motion	0.14 F_y
	5.0
1. For determination of the effective net area, see Section 1.4 of the eighth edition of the AISC Specification	
2. For minimum spacing and edge distances, see Section 1.16.4 and 1.16.5 of the eighth edition of the AISC Specification	

2.2.11.2. Basic allowable stress in weld metal on the effective weld area shall be as follows:

- (1) Full Penetration Groove (Butt) Welds. — Same as for the base metal joined. All flange plate splices on crane bridge girders shall be full penetration welds and shall be ground smooth in the direction of stress. These welds shall be inspected by radiography and shall be accepted or rejected on the basis of Section 9.25.2 of the AWS Structural Welding Code D1.1.

- (2) Fillet Welds — Stress on the effective throat of fillet welds is considered as shear stress regardless of the direction of application. Basic allowable stress in the weld metal is as follows:

E70XX electrodes = $.27(70) = 18.9$ ksi

E60XX electrodes = $.27(60) = 16.2$ ksi

Shear Stress A36 Steel = $.4(36) = 14.4$ ksi

For fatigue the stress in the metal of continuous or intermittent fillet welds is Stress Category "F" and the permissible stress is based on the loading condition. See Table 3 in Section 2.2.12.

E70XX (low hydrogen) are the preferred electrodes for A36 steel because they can be used on plates thickness up to 2 in. thick without preheating. E60XX electrodes may be used for thinner plates but require preheating to prescribed temperatures on plates greater than 2 in. in thickness.

2.2.11.3.1 Allowable unit stresses for fasteners shall be as listed in Table 2.

In proportioning fasteners, the nominal diameter shall be used. The effective bearing area of a fastener shall be its diameter multiplied by the thickness of metal on which it bears.

Joints required to resist shear between their connected parts are designated as either friction-type or bearing-type connections. Shear connections subjected to stress reversal, or where slippage would be undesirable, shall be friction-type.

Bolts in bearing-type connections with threads in the shear planes of the contact surfaces between the connected parts must use the allowable stresses shown in Table 2. In determining whether the bolt threads are excluded from the shear planes of the contact surfaces, thread length of bolts shall be calculated as two thread lengths greater than the specified thread length as an allowance for thread runoff.

When high strength bearing bolts are used in tension members, the net section of the connected part shall be checked for fatigue. Bolt holes shall be subpunched and reamed, or drilled. All joint surfaces in friction-type connections shall be free of loose scale, burrs, dirt, oil, paint, lacquer or galvanizing.

2.2.11.3.2 Applied Tension, Combined Tension and Shear. High-strength bolts shall be used for fasteners subject to tension or combined shear and tension.

Bolts required to support applied load by means of direct tension shall be proportioned so that their average tensile stress computed on the basis of nominal bolt area will not exceed the appropriate stress in Table 2. The applied load shall be the sum of the external load and any tension resulting from prying action.

The tension due to the prying action shall be computed in accordance with the method for hanger type connections as provided in the AISC Manual of Steel Construction.

For combined shear and tension in joints using high-tensile bolts the allowable stresses in Table 2 shall be modified in accordance with the AISC Manual of Steel Construction.

2.2.11.3.3 Fatigue. High-tensile bolts subjected to the combined effect of external and prying loads in fatigue shall be designed in accordance with the AISC Manual of Steel Construction.

Table 2 — Allowable Working Stresses for Bolts¹⁻²

Load Condition	ASTM A325 Bolts ⁵	ASTM A490 Bolts
Applied tension, F_t	44.0	54.0
Shear, F_v : Friction-type connection ³		
Standard size holes	17.5	22.0
Oversized and short-slotted holes ⁷	15.0	19.0
Long slotted holes ⁷	12.5	16.0
Shear, F_v : Bearing-type connection ⁴		
Threads in any shear plane	21.0	28.0
No threads in shear plane	30.0	40.0
Bearing ⁵ F_p	$\frac{L F_u}{2.0 d}$ or $1.5 F_y$ (whichever is smaller)	

(1) Low carbon bolts (ASTM A 307) shall not be used in structural connections.

(2) All high tensile bolts (friction or bearing) shall be tightened in accordance with the requirements for a high tension friction bolt as per the AISC publication specification for Structural Joints using ASTM A 325 or A 490 bolts.

(3) Applicable for contact surfaces with clean mill scale.

(4) In bearing-type connections, whose length between extreme fasteners in each of the spliced parts measured parallel to the line of an axial force exceeds 50 in., the tabulated value shall be reduced by 20%.

(5) L is the distance in inches measured in the line of force from the center line of a bolt to the nearest edge of an adjacent bolt or to the end of the connected part toward which the force is directed; d is the diameter of the bolts; and F_u is the lowest specified minimum tensile strength of connected parts.

(6) ASTM A 325 high-strength bolts are available in three types, designated as types 1, 2 or 3. Type 3 shall be required on the plans when using unpainted ASTM A 588 steel.

(7) For definition of the terms oversized, short slotted and long slotted holes refer to the AISC publication Specifications for Structural Joints using ASTM A 325 or A 490 bolts.

2.2.12 Allowable Stress Range Under Repeated Load. Members subject to repeated load shall be designed for maximum stress in accordance with Section 2.2.11 and for the maximum stress ranges in Table 3 for the detail category, given in Table 4 and shown in Fig. 1.

2.2.12.1 Structural Fatigue Service Classes. The equivalent constant amplitude cycles can be determined from the expected crane duty cycle using the following equation:

$$N_{eq} = \sum \left(\frac{LL_i}{LL_{max}} \right)^3 n_i$$

Where:

N_{eq} = Equivalent number of constant amplitude cycles

LL_i = Lifted load for i th portion of variable amplitude loading spectrum

n_i = Number of cycles for i th portion of variable amplitude spectrum

LL_{max} = Maximum lifted load

The crane duty service class can then be determined from the following table:

Equivalent Constant Amplitude Cycles	Service Class
Less than 100,000	1
100,000 to 500,000	2
500,000 to 2,000,000	3
over 2,000,000	4

Information is given in Appendix B for those cases where a more refined service classification is desired.

In establishing the allowable stresses for an actual calculated stress the determination of cycles can be made by various approaches. The simplest and most conservative is to accumulate the total load cycles on the component or main structure with no consideration given to the effects of stress magnitude. An illustration of this would be a crane making 5 lifts/hr, 1 shift/day, 360 days/year, for 50 years, resulting in 720,000 lifts in its estimated life, would have a service class of 3. A more thorough approach to determine the Service Classes involves the evaluation of accumulated cycles and the stress magnitude.

2.2.12.2 Shear Stress. For design calculations pertaining to repeated load under Category F of Table 3 insofar as they apply to fillet welds, the term shear stress refers to the resultant stress of all stress components acting on the throat area of the weld.

2.2.12.3 Attachments and Temporary Welds.

Temporary welds shall be subject to the same welding procedure requirements as the final welds. They shall be removed unless otherwise permitted by the design engineer. When they are removed, the surface shall be made flush with the original surface.

No supplemental welding (tacks, braces or stiffeners not shown on the design drawings or incorporated into the final welds) is permitted without approval of the responsible design engineer. Supplemental welding not removed or incorporated into final welds shall be added to the record drawings as a revision.

2.2.13 Compressive Stress.

2.2.13.1 Columns. The average allowable compressive stress on the gross section area of axially loaded

Table 3 — Allowable Fatigue Stress Ranges, ksi^a

Detail Category (from Table 4)	Service Class 1	Service Class 2	Service Class 3	Service Class 4
A	60	36	24	24
B	45	27.5	18	16
C	32	19	13	10, 12 ^b
D	27	16	10	7
E	21	12.5	8	5
E'	16	9.4	5.8	2.5
F	15	12	9	8

a. The stress range is defined as the algebraic difference between the maximum stress and the minimum stress. Tension is considered to have the opposite algebraic sign from compression stress.

b. For base material adjacent to transverse stiffener or diaphragm welds on webs or flanges.

columns or struts, F_a , when Kl/r , the largest effective slenderness ratio of any unbraced segment, is less than C_c is:

$$F_a = \frac{\left[1 - \frac{\left(K \frac{l}{r} \right)^2}{2C_c^2} \right] F_y}{N}, \text{ ksi} \quad (\text{Eq 6})$$

$$N = \text{Design factor} = \frac{5}{3} + \frac{3 \left(\frac{Kl}{r} \right)}{8 C_c} - \frac{\left(\frac{Kl}{r} \right)^3}{8 C_c^3} \quad (\text{Eq 7})$$

$$C_c = \sqrt{\frac{2\pi^2 E}{F_y}} \quad (\text{Eq 8})$$

The average allowable compressive stress on the gross section area of axially loaded columns or struts when Kl/r exceeds C_c shall be:

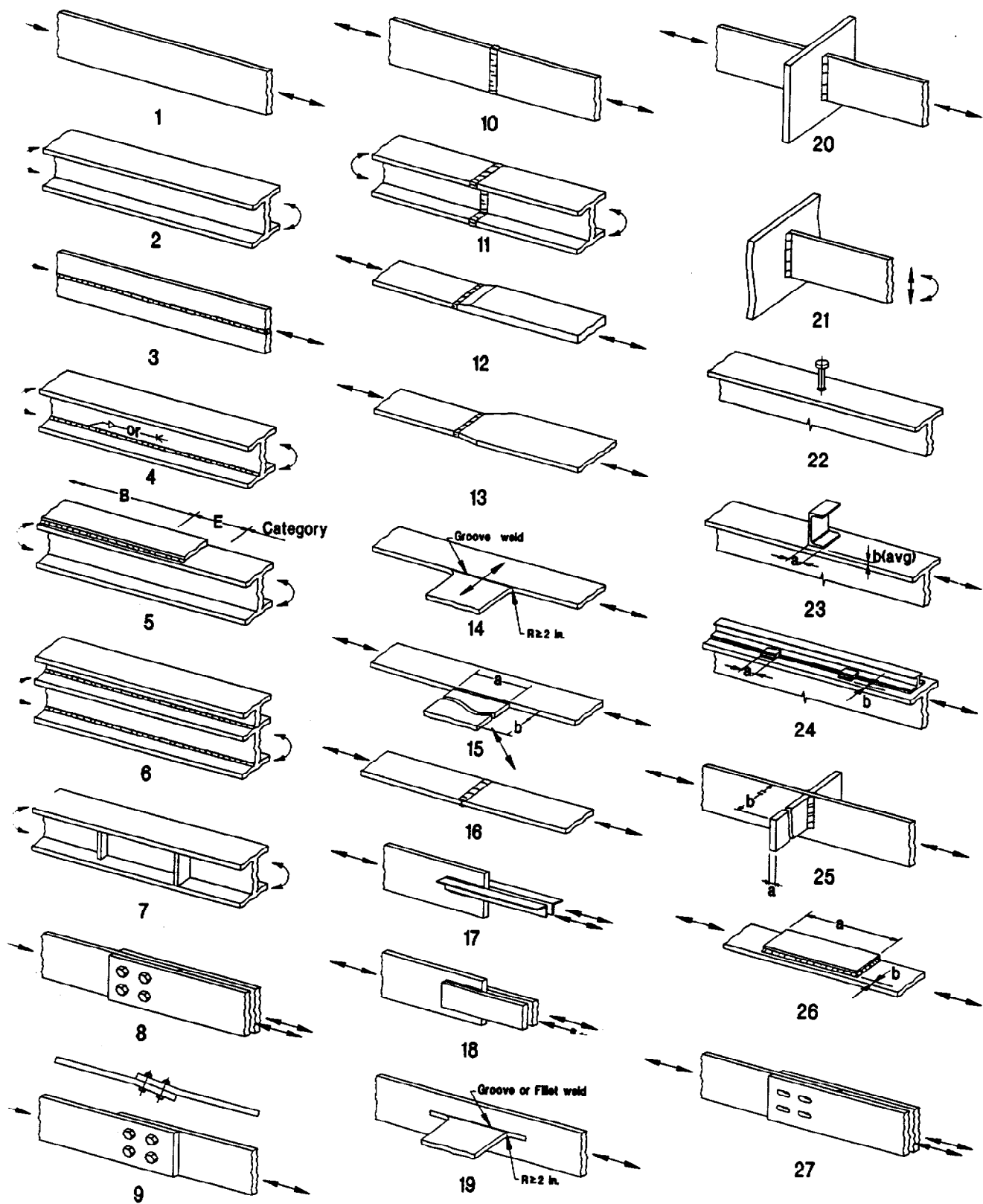
$$F_a = \frac{12 \pi^2 E}{23 \left(\frac{Kl}{r} \right)^2}, \text{ ksi} \quad (\text{Eq 9})$$

2.2.13.2 Beam and Girder Flanges.

(1) Open Sections. For W or S shapes, or for sections having a single web and flange symmetrical about

Table 4 — Joint Fatigue Stress Categories

General Condition	Situation	Kind of Stress ¹	Stress Category (See Table 3)	Illustrative Example Nos. (See Fig. 1)
Plain material	Base metal with rolled or cleaned surfaces.			
Built-up members	Base metal and weld metal in members, without attachments, built up of plates or shapes connected by continuous full or partial penetration groove welds or continuous fillet welds parallel to the direction of applied stress.	T or Rev	A	1,2
	Calculated flexural stress, f_b , in base metal at toe of welds on girder webs or flanges adjacent to welded transverse stiffeners.	T or Rev	B	3,4,5,6
	Base metal at end of partial length welded cover plates having square or tapered ends, with or without welds across the ends.	T or Rev	C	7
	(a) Flange Thickness	T or Rev	E	5
	(b) Flange Thickness	< 0.8 in. > 0.8 in.	E E'	
Fillet welded connections	Base metal at intermittent fillet welds.	T or Rev	E	
Plug or slot weld	Base metal at junction of axially loaded members with fillet welded end connections. Welds shall be disposed about the axis of the member so as to balance weld stresses.	T or Rev	E	17,18,20
	Weld metal of continuous or intermittent longitudinal or transverse fillet welds.	S	F	5,17,18,21
	Base metal at plug or slot welds.	T or Rev	E	27
	Shear on plug or slot welds.			
Groove welds	Base metal and weld metal at full penetration groove welded splices of parts of similar cross-section ground flush, with grinding in the direction of applied stress and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of Table 9.25.3 of AWS D1.1	S	F	27
	Base metal and weld metal at full penetration groove welded splices at transitions in width or thickness, with welds ground to provide slopes no steeper than 1 to 2½, with grinding in the direction of applied stress, and with weld soundness established by radiographic or ultrasonic inspection in accordance with the requirements of Table 9.25.3 of AWS D1.1.	T or Rev	B	10,11
	Base metal and weld metal at full penetration groove welded splices with or without transitions having slopes no greater than 1 to 2½, when reinforcement is not removed and/or weld soundness is not established by radiographic or ultrasonic inspection in accordance with the requirements of Table 9.25.3 of AWS D1.1.	T or Rev	C	10,11,12,13
Mechanically fastened connections	Weld metal of partial penetration transverse groove welds, based on effective throat area of the weld or welds.			
	Base metal at gross section of high-strength bolted friction-type connections except axially loaded joints which induce out-of-plane bending in connected material.	T or Rev	F	16
		T or Rev	B	8



the vertical (web) axis, the allowable compressive stress shall be the largest of the values computed by Eqs 10, 11 and 12, as applicable, (only Eq 12 is applicable to channels) but no greater than $0.60 F_y$

When:

$$\sqrt{\frac{102 \times 10^3 C_b}{F_y}} \leq \frac{l}{r_T} \leq \sqrt{\frac{510 \times 10^3 C_b}{F_y}}$$

$$F_b = \left[\frac{2}{3} - \frac{F_y \left(\frac{l}{r_T} \right)^2}{1530 \times 10^3 C_b} \right] F_y \quad (\text{Eq 10})$$

When:

$$\frac{l}{r_T} \geq \sqrt{\frac{510 \times 10^3 C_b}{F_y}}$$

$$F_b = \frac{170 \times 10^3 C_b}{\left(\frac{l}{r_T} \right)^2} \quad (\text{Eq 11})$$

Or, when the compression flange is solid and approximately rectangular in cross-section and its area is not less than that of the tension flange:

$$F_b = \frac{12 \times 10^3 C_b}{\frac{ld}{A_f}} \quad (\text{Eq 12})$$

Where:

A_f = Area of the compression flange, sq in.

$C_b = 1.75 + 1.05 \left(\frac{M_1}{M_2} \right) + \left(0.3 \frac{M_1}{M_2} \right)^2$, but not more

than 2.3.

(Note: C_b can be conservatively taken as unity. For smaller values see AISC Specification for the Design, Fabrication and Erection of Structural Steel for Buildings.) Where M_1 is the numerically smaller and M_2 the numerically larger bending moment at the ends of the

unbraced length, taken about the strong axis of the member and where M_1/M_2 , the ratio of end moments, is positive when M_1 and M_2 have the same sign (reverse curvature bending) and negative when they are of opposite signs (single curvature bending). When the bending moment at any point within an unbraced length is larger than that at both ends of this length, the value of C_b shall be taken as unity.

l = Distance between cross-sections braced against twist or lateral displacement of the compression flange, in. For cantilevers braced against twist only at the support, l may conservatively be taken as the actual length.

r_T = Radius of gyration of a section comprising the compression flange plus one-third of the compression web area, taken about an axis in the plane of the web, in.

For members meeting the maximum width-to-thickness ratio requirements of Table 5, but not included in the preceding:

$$F_b = 0.60 F_y$$

provided that sections bent about their major axis are braced laterally in the region of compressive stress at intervals not exceeding $76 \frac{b_f}{\sqrt{F_y}}$.

(2) Closed Box Sections. The permissible normal compressive stress due to the bending moment about the horizontal axis, F_{bx} , may be less than the basic allowable stress because of a lack of lateral support against lateral-torsional buckling, or when the width-to-thickness ratio of the compression flange exceeds the permissible value for no stress reduction.

The permissible stress, F_{bx} , for a laterally unsupported box girder may be determined by deriving an equivalent column slenderness ratio using Eq 13 and obtaining directly the permissible stress ($F_{bx} = F_a$) by (Eq 6). K is taken at unity.

$$\left(\frac{l}{r} \right) = \sqrt{\frac{5.1 LS_x}{\sqrt{J I_y}}} \quad (\text{Eq 13})$$

The l/r ratio of the box section about the vertical neutral axis shall not exceed C_c as listed in Table 5.

When the unsupported width-to-thickness ratio, w/t , of a box section compression flange, b , exceeds the limit, w_c/t , tabulated in Table 5, the design will be acceptable if the average stress is less than the basic allowable multi-

Table 5 — Limiting Ratios In Compression Flange

F_y , ksi	$0.60 F_y$, ksi	$C_c = \sqrt{\frac{2 \pi^2 E}{F_y}}$	$\frac{b}{t}$ Ratio for Open Sections* $\frac{95}{\sqrt{F_y}}$	$\frac{w_c}{t}$ Ratio for Box Sections $\frac{238}{\sqrt{F_y}}$
36	22	126.1	15.8	39.7

* b is one-half the width of the flanges of open sections and tees or the full width of stiffeners and other projecting compression elements.

plied by the ratio w_c/w where w is the actual clear unsupported width, and w_c the limiting value listed in Table 5.

2.2.13.3 Combined Stress in Bending. The reduced permissible stresses in the compression flange determined by Eq 13 (with regard to lateral-torsional buckling) or determined by the suggested procedure when w is greater than w_c (with regard to local buckling) are only for a vertical load.

The maximum direct stress under all loads, lateral and vertical combined, should be checked by the following interaction formula:

$$\frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1 \quad (\text{Eq 14})$$

In the case of open sections, f_{by} shall be calculated on the basis of the section modulus of the compression flange alone, including one-sixth of the area of the web, about the vertical (Y-Y) axis. For box sections with diaphragms or adequate cross-frames, f_{by} may be calculated using the section modulus of the complete box section about the vertical (Y-Y) axis. F_{bx} is the reduced allowable stress for vertical loads alone, and is equal to F_a for the equivalent column slenderness ratio. $F_{by} = 0.6 F_y$ (bending need not be considered for lateral loads).

2.2.13.4 Beam and Girder Web Stiffeners.

2.2.13.4.1 Web Plates and Vertical Stiffeners. Unless vertical (transverse) diaphragms or stiffeners are used, the h/t ratio of the web plates shall not exceed:

$$\frac{240}{\sqrt{f_v}} \quad (\text{Eq 15})$$

The spacing of transverse stiffeners, full depth diaphragms or frames in box sections, when required, shall not exceed:

$$\frac{320t}{\sqrt{f_v}} \quad (\text{Eq 16})$$

nor shall it exceed the unsupported depth, h , of the web plate.

If the maximum shear stress in ksi due to bending and torsion combined is less than $\frac{57,600}{\left(\frac{h}{t}\right)^2}$, ksi, the spacing of

full depth diaphragms need be determined only by torsional requirements, i.e., to maintain the shape of cross-section and to distribute the concentrated forces eccentric to the shear center.

The moment of inertia of a pair of intermediate stiffeners, or a single intermediate stiffener, with reference to an axis in the plane of the web, shall not be less than:

$$I = \left(\frac{h}{50}\right)^4 \quad (\text{Eq 17})$$

Intermediate stiffeners may be stopped short of the tension flange, provided bearing is not needed to transmit

a concentrated load or reaction. The weld by which intermediate stiffeners are attached to the web shall be terminated not closer than 4 times nor more than 6 times the web thickness from the near toe of the web-to-flange weld.

The moment of inertia shall be calculated in accordance with Section 2.2.13.6.

2.2.13.4.2 Web Plates and Horizontal (Longitudinal) Stiffeners. Unless horizontal (longitudinal) stiffeners are used the h/t of the web plate shall not exceed:

$$\frac{760}{\sqrt{f_b}} \quad (\text{Eq 18})$$

If horizontal (longitudinal) stiffeners are used the h/t ratio of the web plate shall not exceed:

$$\frac{1520}{\sqrt{f_b}} \quad (\text{Eq 19})$$

The center line of a horizontal (longitudinal) stiffener bar or the gage line of a horizontal (longitudinal) stiffener angle shall be $h/5$ from the inner surface or leg of the compression flange component.

The horizontal (longitudinal) stiffener shall be proportioned so that:

$$I = ht_w^3 \left(2.4 \frac{a^2}{h^2} - 0.13 \right) \quad (\text{Eq 20})$$

2.2.13.4.3 Web Crippling. Webs of beams and welded plate girders shall be so proportioned that the compressive stress at the web toe of the fillets, resulting from concentrated loads not supported by bearing stiffeners, shall not exceed $0.75F_y$; otherwise, bearing stiffeners shall be provided. The governing formulas shall be:

For interior loads:

$$\frac{R}{t(N + 2k)} \leq 0.75F_y \quad (\text{Eq 21})$$

For end-reactions:

$$\frac{R}{t(N + k)} \leq 0.75F_y \quad (\text{Eq 22})$$

Where

R = Concentrated load or reaction, kips

t = Thickness of web, inches

N = Length of bearing (not less than k for end reactions), in.

k = Distance from outer face of flange to web toe of fillet, in.

2.2.13.5 Stiffened Plates in Compression. When two or three longitudinal stiffeners are added to a plate under uniform compression, dividing it into segments

ual unsupported widths, full edge support will be by the longitudinal and transverse stiffeners or ns, and the provisions of Table 5 may be applied sign of the plate material. To provide edge sup- stiffeners must meet minimum requirements as fol-

ulating the moment of inertia of a single stiffener on one side of a plate, it may be assumed that the neutral axis of the stiffener is at the interface of the stiffener and the stiffened plate. The asterisked footnote to Table 5 applies to stiffener b/t ratio.

ne longitudinal stiffener at the center of the compression flange, with $b = 2w$, where w is the unsupported half width between web and stiffener the moment of inertia of the stiffener shall be no less than

$$I_{st} = \left(\frac{1}{b} \right) + 0.2 \left(\frac{a}{b} \right)^2 + 3.0 \left(\frac{A_s a}{b^2 t} \right) b t^3, \text{ in.}^4 \quad (\text{Eq 23})$$

The longitudinal distance between diaphragms or transverse stiffeners, in.

= The area of the stiffener, sq. in.

The thickness of the stiffened plate, in.

ne moment of inertia need not be greater in any case as given by Eq 24 as follows:

$$I_{st} = \left\{ 2.2 + 10.3 \left(\frac{A_s}{b t} \right) \left[1 + \left(\frac{A_s}{b t} \right) \right] \right\} b t^3, \text{ in.}^4 \quad (\text{Eq 24})$$

For two longitudinal stiffeners at the third points of the compression flange, with $b = 3w$, and A_s the area of one stiffener, the moment of inertia of each of the two stiffeners shall be no less than:

$$I_{st} = \left[0.4 \left(\frac{a}{b} \right) + 0.8 \left(\frac{a}{b} \right)^2 + 8.0 \left(\frac{A_s a}{b^2 t} \right) \right] b t^3 \quad (\text{Eq 25})$$

The moment of inertia need not be greater in any case than:

$$I_{st} = \left[9 + 56 \left(\frac{A_s}{b t} \right) + 90 \left(\frac{A_s}{b t} \right)^2 \right] b t^3, \text{ in.}^4 \quad (\text{Eq 26})$$

For three longitudinal stiffeners, spaced equidistant at one fourth width locations ($b = 4w$), and limited to a/b less than three, the moment of inertia may be determined by the following:

$$I_{st} = \left[0.35 \left(\frac{a}{b} \right) + 1.10 \left(\frac{a}{b} \right)^2 + 12.0 \left(\frac{A_s a}{b^2 t} \right) \right] b t^3 \quad (\text{Eq 27})$$

2.2.13.6 Longitudinal Stiffeners.

- (1) The moment of inertia of a single stiffener shall be computed from the combined section consisting of the stiffener plate and a width of the stiffened plate equal to 30 times the plate thickness. For a plate or bar stiffener, this may be approximated by the moment of inertia of the stiffener taken about an axis located at the interface of the stiffener and the stiffened plate.
- (2) Stiffeners shall not be included in the section properties for the computation of stresses; however, they shall be connected to develop the shear transfer which would result when included in the section properties.
- (3) When longitudinal stiffeners are uninterrupted at diaphragms or transverse stiffeners they may be included in the gross moment of inertia for the purpose of computing deflections and camber.

2.2.13.7 Alternate Design. Other arrangements of stiffened panels and diaphragm spacing may be used providing that:

- (1) Special analyses are made to insure that each unsupported region of web plate under combined direct stress (tensile or compressive and shear) shall have a stress factor not less than the design factors listed below for load combination in Section 2.2.9.

For combinations (1), (2), and (6)

$$N = 1.70 + 0.175 (\psi - 1) \text{ not less than } 1.3$$

For combination (5)

$$N = 1.50 + 0.125 (\psi - 1) \text{ not less than } 1.25$$

For combinations (3) and (4)

$$N = 1.35 + 0.750 (\psi - 1) \text{ not less than } 1.20$$

Where:

N is the design factor

ψ is the ratio of normal stresses at edge of plate panel, and is positive when all stresses are compressive and negative when stresses vary from compression to tension ($-1.0 < \psi < 1.0$)

- (2) The stiffeners shall have adequate stiffness and rigidity to prevent buckling at loads less than those which will cause plate buckling
- (3) Substantiating design calculations shall be provided for the purchaser

2.2.14 Concentrated Wheel Effects. The local bearing stress on diaphragms that support the crane rail shall be assumed to be distributed transversely over a distance equal to the base width of the rail plus twice the thickness of the flange plates and wear plates.

In torsion girders, where the rail is centered directly over one of the girder web plates, concentrated wheel loads shall be transferred to the web plate of the girder by a

continuous full penetration weld designed for the resultant of the vertical stress due to local wheel load and the shear stress due to combined bending and torsion. In calculating the vertical normal stress per unit length, the wheel load shall be assumed to be distributed over a distance equal to twice the combined depth of the crane rail, wear plate and girder flange thickness.

When the rail is centered directly over a girder web plate, local wheel load causes additional stress due to local bending of the top flange as a plate. The effective portion of the top flange that resists local bending moment may be assumed to include all material between the free edge of the flange and a location midway between the web plate and the nearest longitudinal compression flange stiffener (or other web if there are no longitudinal stiffeners). The local bending stress in the flange, f_{bw} , shall be computed by Eq 28.

$$f_{bw} = \frac{P t_f}{8(I_R + I_F)} \times \sqrt{2(I_R + I_F)} \frac{h}{t} \quad (\text{Eq 28})$$

Where:

- h = Web depth, in.
- I_F = Moment of inertia of effective portion of top flange, in.⁴
- I_R = Moment of inertia of rail, in.⁴
- P = Maximum local wheel load, kips
- t = Thickness of web plate, in.
- t_f = Thickness of flange plate, in.

2.3 BRIDGE STRUCTURES AND END TIES.

2.3.1 Bridge Structural Details. Intermittent welding parallel to the direction of stress is a Category E fatigue detail and should not be permitted on girder web-to-flange connections nor wear plate-to-flange connections. Intermittent welds, if used in other areas, shall be designed to meet the stress limitations defined in Section 2.2.12.

High-strength bolts shall be spaced not more than 12 times the thickness of the lighter outside section in compression elements.

Welded splices in the webs or flanges of girders shall be full penetration welds.

Bolted splices shall be designed for the average of the calculated stress and the allowable stress of the member but not less than 75% of the allowable stress of the member.

The span-to-depth ratio, l/d , of the girder shall not be more than 18. The span-to-width ratio, l/b , of the girder shall not be more than 60, nor shall it be more than:

$$\frac{l}{d} \times \frac{\text{Maximum Flange Stress Due to Vertical Load (Without Impact)}}{\text{Maximum Stress Due to Horizontal Load}}$$

The total vertical deflection of the girder for the live load ($W_L + W_A + W_T$) and not including impact or dead load of the girder itself shall not exceed 0.001 in./in. of span.

Girders shall be cambered an amount equal to the dead load deflection plus one-half of the deflection caused by live load ($W_L + W_A + W_T$).

Full-depth diaphragms are required at walkway supports, bridge drive supports and line shaft bearing pedestals on the girders. Supplemental external stiffeners adjacent to diaphragms may be required to transmit local forces into the bottom flange. Vertical stiffeners or full-depth diaphragms may be used interchangeably when required by Section 2.2.13.4.

In addition to the required full-depth diaphragms for box section girders, short diaphragms shall be inserted where required to transmit the trolley wheel load including impact to the web plates and to limit the maximum trolley rail stress to 20 ksi based on:

Stress, f_{br} =

$$\frac{(\text{Impacted Wheel Load, kips}) \times (\text{Distance Between Supports, in.})}{(6) \times (\text{Section Modulus of Rail})}$$

Top cover plates shall not be considered as giving support to the rail in computing the diaphragm spacing or rail size.

All diaphragms must bear against the top cover plate. The thickness of the diaphragm must be sufficient to resist the trolley wheel load in bearing on the assumption that the wheel load is distributed over a distance equal to the width of the rail base plus twice the thickness of the cover plate and wearing plate, if provided. The diaphragm plate shall not be welded in this area.

Cranes shall be provided with a full-length wearing plate under the trolley runway rails, if specified on the OIS. This plate is to be at least 3/8 in. thick, at least as wide as the rail base, continuous, and welded in place to the top flange. This wear plate shall not be considered as a part of the girder cross section for stress or deflection calculations.

If an auxiliary truss is used it shall be designed to carry the appropriate portion of the platform load and shall be framed so as to minimize participation in bridge girder loads.

Provision shall be made in box girders to eliminate any accumulation of water, oil or other liquids. If specified on the OIS, welded girders shall be provided with breathing holes to allow for the expansion or contraction of the air inside due to temperature changes. Special care shall be taken with cranes for outdoor use to eliminate crevices or openings where water may accumulate and cause corrosion.

Capacity of each hoist (lbs or tons as specified on the OIS) shall be shown on each side of crane in such a manner as to be easily legible from the floor.

An adequate number of fitted bolts for drilled and reamed holes shall be provided in the end tie or end carriage connections to accurately align the girders with the end trucks during field erection. Connection between girders and end trucks shall be as specified on the OIS.

Squaring marks shall be provided on each girder to facilitate erection and squaring of the bridge.

2.3.2 End Carriages and End Trucks. The wheel base shall not be less than one-sixth of the span. On cranes having eight or more wheels, the wheel base is the distance between centerlines of the two outside wheels.

There shall be a heavy safety lug or strap across the bottom of each carriage near the track wheel and 1 in. above the top of the rail to prevent excessive drop in case of breakage of the track wheel, axle or carriage.

If any part of carriage or track wheel gears projects below the flange of the track wheel, it shall be specifically indicated on the drawings, crane manufacturer's proposal or both.

End carriages and trucks are to be designed to permit easy changing of the wheels.

Pads shall be provided for the use of jacks or wedges when changing track wheels.

Rail sweeps which will prevent any object from being trapped between the advancing wheel and the rail shall be attached to the four corners of the bridge.

2.3.3 Trolley Frames. The trolley frame shall be of welded steel construction. The requirements for the bridge and carriages regarding safety lugs, guards, and clearances shall also apply to the trolley.

Drum bearing brackets shall be integral with the frames. Machinery assemblies shall be mounted on machined surfaces. Shims may not be used except under brakes, motors and drum end bearings.

The trolley shall be of the floored-over type without openings except for the ropes and magnet cable. Deck plates shall be not less than 1/4 in. thick and shall be provided with toe guards around all openings or edges. Load girts are to be designed to carry the load to the side frames.

2.3.4 Footwalks. Level steel footwalks made of either antislip-type floor plate or expanded metal or subway-type grating (if allowed by applicable state code) shall be provided on the outside of the girders on which the bridge drive is mounted for the full length of the girder, and for double the length of the trolley on the idler girder unless full length walkway is specified on the OIS.

The footwalks are to be equipped on both sides with toe plates at least 6 in. in height. It is not necessary to fill in between the inside toe angle and the web of the girder unless called for by the applicable state code. Footwalks shall be of sufficient width to give at least 18 in. clear passage at all points, except between railing and bridge drive where the clearance may be reduced to not less than 15 in. The clearance between railing on the bridge walk and the nearest part of the trolley shall not be less than 18 in. The footwalk along girders should have at least 7 ft 0 in. clearance below roof chords. Width of trolley walks, if provided, shall not be less than 15 in.

2.3.5 Railings. Railings on footwalks shall be made of steel, to purchaser's specification, 42 in. high, and with an intermediate member 21 in. high. Toe plates not less than 6 in. high are required except on stairs.

Railings shall be provided on girder footwalks, ends of bridges, trolleys, landings on cabs and on stairs leading to the bridge girder from the landings on cabs as specified. The distance between rails on stairs shall not be less than 24 in.

2.3.6 Stairs and Ladders. Stairs or ladders shall be provided to give access from the cab to the bridge footwalk as specified on the OIS.

When other stairs or ladders are required they should be listed under special features on the OIS. Wherever possible, the location and the design of stairs shall be such so as not to obstruct the crane operator's vision during operation.

Stair treads shall be of material designed to prevent slipping and shall not be less than 21 in. wide. Where stairs are not constructed with risers, a plate or wire mesh shield shall be attached to the underside and extend the entire length.

The maximum slope of stair shall not exceed an angle of 50 degrees from the horizontal.

Where ladders are provided they shall be of steel construction with rungs welded to the ladder rails to prevent the rungs from turning. The rails shall be extended 42 in. above the landing place at the top to assist in getting on or off the ladder and shall start on a landing platform.

All footwalks, railings, ladders or stairs shall be made so as not to interfere with the removal of any part of the crane.

2.3.7 Operator's Cab. The operator's cab shall be built of steel and fire-resistant material with a clear height with equipment installed of not less than 7 ft 0 in. The cab shall be adequately braced to prevent swaying or vibration; such bracing shall not interfere with access to the cab or with the vision of the operator. All bolts for supporting member connections should be in double shear.

Enclosed cabs shall have a watertight plate roof which slopes to the rear and shall be provided with sliding, hinged or drop windows on three sides, and with a sliding or hinged door. All window sashes shall be equipped with clear, safety glass or as specified on the OIS.

Open cabs shall have the rear side enclosed with steel plate. The other three sides shall be enclosed with standard railing 42 in. high, with the space between the floor and the intermediate member enclosed with steel plate. Where the top rail (if placed 42 in. above the floorline), seriously interferes with the operator's vision, it may be lowered if approved by the owner.

The floor of the cab, which shall be steel plate, shall be extended to form a landing platform which is to be provided with hand railing similar in design to that

specified for footwalks. The floor of the cab is to be covered with thermal insulating material.

Cranes subjected to heat from below must have a shield 6 in. below the bottom of the floor to insulate the floor from heat.

The cab shall be provided with a warning signal device as specified in the OIS and shall be installed so as to be accessible for maintenance and arranged so that parts working loose cannot fall from the crane.

Cabs shall be designed for maximum operator visibility. A visibility diagram shall be furnished to the owner for approval.

The arrangement of equipment in the cab shall be as designated on the OIS.

Other detailed cab specifications, location and arrangements shall be as specified on the OIS.

2.3.8 Other Considerations. Depending on the specific crane application, the Owner may wish to address the following considerations and note his requirements on the OIS accordingly.

- (1) Full length walkways on each side of the bridge.
- (2) Access to the crane at operator's cab level.
- (3) Access between the operator's cab and bridge walkway.
- (4) Access between trolley and top of bridge girder.
- (5) Emergency evacuation provisions for the crane operator.
- (6) Gravity self-closing gates at handrail openings in lieu of chains.
- (7) Service cage and platforms for access to collectors.

- (8) Access platforms for wheels and bearings on cranes with equalizer trucks.
- (9) Ergonomic and environmental considerations affecting the crane operator, including:
 - (a) Range of vision (Sightlines).
 - (b) Seat type and position.
 - (c) Location and type of master switches, controls, and instruments.
 - (d) Noise abatement.
 - (e) Temperature, ventilation, and air quality.
 - (f) Mitigation of vibrations transmitted structurally to the cab.
 - (g) Window cleaning provisions.
- (10) Protective pipe guards along the inside edges of the bottom flange plate of the bridge girder to eliminate damage to or from the wire ropes and/or lifting beams. Note: Attachments to the girders shall observe the requirements of Section 2.2.12 of this report.
- (11) Lifting lugs on large components such as trolley and girders, to facilitate crane erection. Note: Attachments to the girders shall observe the requirements of Section 2.2.12 of this report.
- (12) An outrigger truss for support of walkways, motors, gear boxes, and electric panel boxes positioned along the bridge walkway. Note: Extra design investigation is recommended when cantilevered construction is used.
- (13) Mounting of trolley rails on elastomeric pads with compatible rail clips.
- (14) Welded trolley rails.
- (15) If used, the wear plate may be widened to permit welded attachment of trolley rail clips.

SYMBOLS — STRUCTURAL

A	Membrane area of box section bounded by centerlines of webs and flanges, sq in.	F_S	Skewing force parallel to bridge runway, kips
A_f	Area of compression flange, sq in.	F_{sr}	Allowable stress range, ksi
u	Clear distance between transverse web stiffeners, full depth diaphragms or both, in.	F_t	Allowable tensile stress, ksi
a	Distance from centerline of bridge runway to center of gravity of live load with trolley at nearest approach, ft	F_u	Specified minimum tensile strength, ksi
a	Distance from center of bolt to edge of plate, in.	F_v	Allowable shear stress, ksi
a	Distance between transverse stiffeners, in.	F_y	Specified minimum yield stress, ksi
b	Centerline-to-centerline of web plates, in.	f_b	Computed average bending stress, ksi
b	One-half the width of flanges of open section and tees or the full width of stiffeners and other projecting compression elements, in.	f_{br}	Computed bending stress in rails, ksi
b	Distance from center of bolt to toe of fillet of connected part, in.	f_{bw}	Computed flange stress due to local bending, ksi
b_f	Width of compression flange for an I shape, in.	f_{bx}	Computed bending stress about the X-X axis, ksi
C_b	Bending coefficient dependent upon moment gradient	f_{by}	Computed bending stress about the Y-Y axis, ksi
C_c	Column slenderness ratio, separating elastic and inelastic buckling	f_c	Computed compressive stress, ksi
c	Distance from center of web plate to edge of flange plate, in.	f_f	Force factor due to tractive effort
c_x	Distance from Y-Y axis to extreme fiber, in.	f_{max}	Maximum stress, ksi
d	Depth of girder, center-to-center of flange plates, in.	f_t	Computed tensile stress, ksi
d	Diameter of bolt, in.	f_v	Computed shear stress, ksi
E	Elastic modulus, ksi (29×10^3 for steel)	f_{vb}	Computed vertical shear stress, ksi
e	Location of shear center, in.	$f_{v(max)}$	Maximum computed shear stress, ksi
F	Horizontal equivalent concentrated force, kips	f_{vt}	Computed torsional shear stress, ksi
F_a	Allowable compressive stress, ksi	g	Acceleration due to gravity, 32.2 fps ²
F_b	Allowable bending stress, ksi	H_S	Skewing force normal to bridge runway, kips
F_{bx}	Allowable bending stress about the X-X axis, ksi	I	Moment of inertia, in. ⁴
F_{by}	Allowable bending stress about the Y-Y axis, ksi	I	Minimum permissible moment of inertia of any type of transverse intermediate stiffener, in. ⁴
F_{eq}	Allowable equivalent stress, ksi	I_F	Moment of inertia of effective segment of top flange, in. ⁴
F_L	Force at left end of bridge due to tractive effort, kips	I_o	Required moment of inertia of a stiffener, in. ⁴
F_p	Allowable bearing stress, ksi	I_R	Moment of inertia of a crane rail, in. ⁴
F_R	Force at right end of bridge due to tractive effort, kips	I_x	Moment of inertia about the X-X axis, in. ⁴
		I_y	Moment of inertia about the Y-Y axis, in. ⁴
		J	Polar moment of inertia, in. ⁴
		K	Effective column length factor
		k	Distance from outer face of flange to web toe of fillet, in.
		LL_i	Lifted load for i th portion of variable amplitude loading spectrum
		L	Bridge span, ft or in.
		LL_{max}	Maximum lifted load

L	Distance from centerline of bolt to nearest edge of adjacent bolt or to end of connected part, in.		web area, taken about an axis in the plane of the web, in.
l	Distance between cross-sections braced against twist or lateral displacement of the compression flange, in.	r_y	Radius of gyration about the Y-Y axis, in.
l	Span of girder, in.	S_x	Section modulus about the X-X axis, in. ³
M	Computed bending moment, kip-ft	S_y	Section modulus about the Y-Y axis, in. ³
M_D	Computed bending moment due to dead load, kip-ft	s	Trolley span, in.
M_I	Vertical impact moment, kip-ft	T	Direct tension/bolt due to external load, kips
M_L	Computed bending moment due to live load, kip-ft	t	Thickness, in.
M_{max}	Maximum bending moment, kip-in.	t_f	Thickness of flange plate, in.
M_t	Computed torsional moment, kip-in.	t_w	Thickness of web plate, in.
M_1	The smaller of two end moments, kip-in.	V	Shear force, kips
M_2	The larger of two end moments, kip-in.	V_x	Horizontal shear force, kips
N	Length of bearing, in.	V_y	Vertical shear force, kips
N	Minimum design factor	v	Velocity, fps
N_{eq}	Equivalent number of constant amplitude cycles	W	Uniform load, kip/ft
n_i	Number of cycles for the i th portion of variable amplitude spectrum	W_A	Weight of column, kips
P	Maximum wheel load, kips	W_B	Dead weight of bridge structure excluding track wheels, end trucks, equalizers, saddles and end ties, kips
P	Concentrated load, kips	W_{BE}	Total dead weight of bridge structure including track wheels, end trucks, equalizers, saddles and end ties, kips
Q	Static moment of area, in. ³	W_E	Equivalent concentrated load, kips
Q	Prying tension per bolt, kips	W_L	Weight of lifted load including hook block, kips
R	Transition radius, in.	W_T	Weight of trolley excluding hook block, kips
R	Concentrated load or reaction	w	Unsupported clear width of top flange between longitudinal stiffeners, webs or both, in.
R_L	Reaction at left end of beam, kips	w_c	Limiting clear width of top flange with no reduction in allowable compressive stress, in.
R_R	Reaction at right end of beam, kips	$\bar{y}$	Location of center of gravity, in.
R_1	Ratio of top flange thickness to bottom flange thickness for box girder	Δ	Lateral displacement of girder at centerline during impact, including both girder deflection and bumper travel, ft
R_2	Ratio of top flange thickness to web thickness for box girder	δ	Lateral displacement of girder, in.
r	Radius of gyration, in.	ψ	Ratio of normal stresses at edge of plate panel
r_T	Radius of gyration of section comprising the compression flange plus one-third of the compression		

COMMENTARY — STRUCTURAL

It is the purpose of this commentary to amplify, supplement and explain the basis and application of portions of this Report not covered elsewhere. The comments herein are not part of the Report but are added as supplementary information. Numerals in parentheses refer to the Section number in the text of the Report.

IMPACT (2.2.2). This Report applies only to properly installed runway rails with either continuously welded rails or tight bolted rail joints. Poorly made or worn joints increase the impact effect both on the crane runway girders and on the crane and create an increasing tendency toward fatigue failure and other maintenance problems.

HORIZONTAL FORCES (2.2.3). In the case of stripper and pit cranes or other cranes with vertical arms that are attached to the crane structure, the horizontal forces can be no larger than that force which will tilt the trolley when the length of the lever arm is at a minimum at which a horizontal force can be applied.

However, if the friction slipping force is exceeded prior to tilt, it should be used in place of the tilting force. In this case it is assumed that the coefficient of friction is 0.2. The longitudinal force resulting from friction between the driven wheels and the crane runway rails can be no greater than the coefficient of friction times the load on the driven wheels. This can be determined by multiplying the total load of the entire crane structure (including the full lifted load but not including impact) by the ratio of the number of driven wheels to the total number of wheels.

SKEWING EFFECT (2.2.3.3). The skewing effect occurs most severely when the trolley and lifted load are near one

end of the bridge travel. It is induced in part by one end of the bridge trying to move ahead of the other. For design purposes it will be assumed that the skewing effect is caused by the difference in the bridge tractive effort from one end of the bridge to the other. This difference is split into equal and opposite reactions acting parallel to the bridge runway. The assumed distribution of forces is shown in Figs. 3 and 4 along with necessary equilibrium reactions assumed normal with the runway rails. Reactions at each end of bridge due to tractive effort (Fig. 2):

$$R_L = f_f \left[(W_T + W_A + W_L) \left(1.0 - \frac{a}{l} \right) + 0.5 W_{BE} \right]$$

$$R_R = f_f \left[(W_T + W_A + W_L) \frac{a}{l} + 0.5 W_{BE} \right]$$

Where:

$$f_f = \text{force factor} = 0.2 \times \frac{\text{Number of Driven Wheels}}{\text{Total Number of Wheels}}$$

Skewing forces:

$$F_S = \frac{R_L - R_R}{2}$$

$$= f_f (W_T + W_A + W_L) \left(0.5 - \frac{a}{l} \right)$$

$$H_S = F_S \frac{l}{s}$$

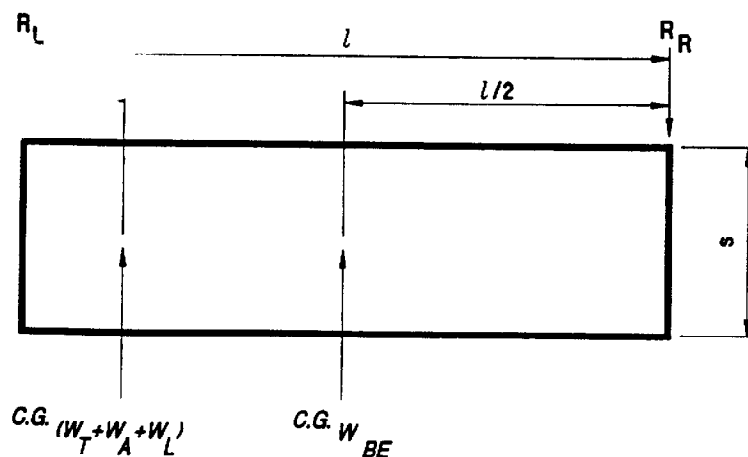


Fig 2 — Bridge frame — Forces at each end of bridge due to tractive effort

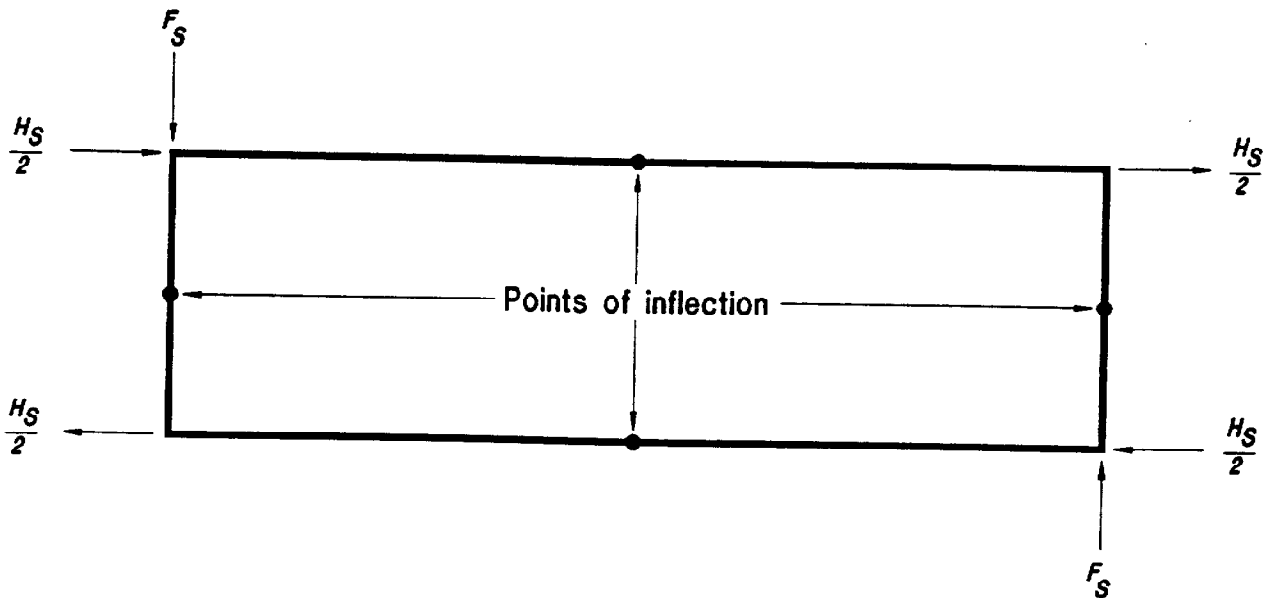


Fig. 3 — Bridge frame — Distribution of skewing forces

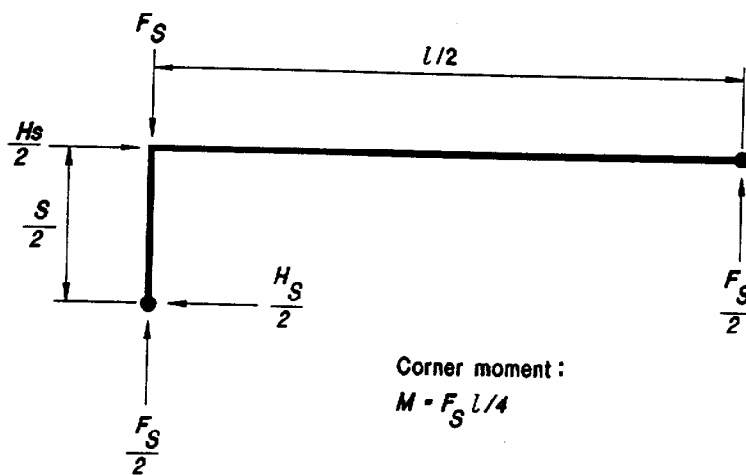


Fig. 4 — Bridge frame — Skewing forces at corners and points of inflection

INEQUALITIES IN THE DISTRIBUTION OF VERTICAL LOAD (2.2.4). Inequalities in the distribution of vertical load to the trolley rails may be due either to the tilting of the trolley in stripper or pit cranes or it may be due to a lack of symmetry in the distribution of the vertical loads such as the cab and controls.

LOCATION OF SHEAR CENTER (2.2.4). As noted, when there is only a small lack of symmetry in a box section the shear center may be assumed to be at the centroid of the cross-section. However, if it is desired to locate the shear center exactly the following equation may be used:

SHEAR CENTER (S.C.) LOCATION (Fig. 5)

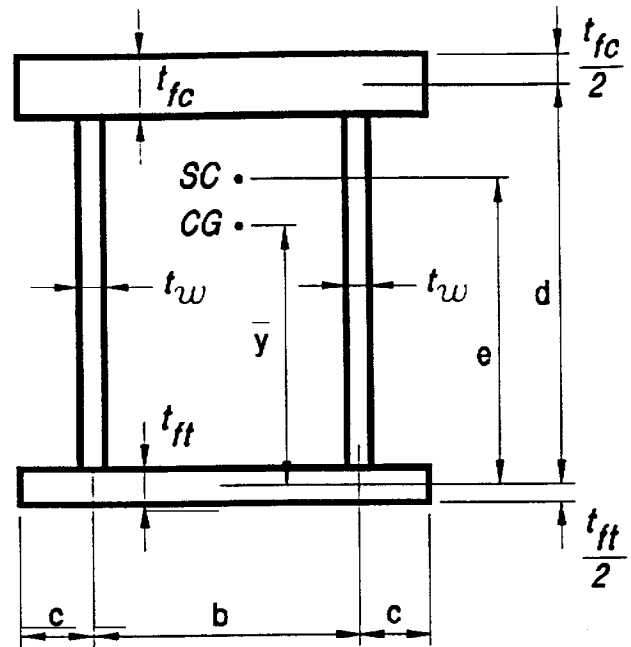
$$e = \left(\frac{A}{B} \right) d$$

$$A = R_2 \left(0.5 + \frac{c}{b} \right)^2 + \frac{\left[R_1 \left(\frac{d}{b} \right) + R_2 \left(\frac{d}{b} \right)^2 \right]}{\left[\left(1 + R_1 \right) + 2 R_2 \left(\frac{d}{b} \right) \right]}$$

$$+ \left(\frac{R_2}{6} \right) \left[-1 - 3 \left(\frac{c}{b} \right) + 4 \left(\frac{c}{b} \right)^3 \right] - \frac{1}{4} \left(\frac{d}{b} \right)$$

$$B = \left(\frac{1}{12} \right) \left(R_2 + \frac{R_2}{R_1} \right) \left[1 + 2 \left(\frac{c}{b} \right) \right]^3 + \frac{1}{2} \left(\frac{d}{b} \right)$$

MAXIMUM SHEAR STRESS IN BOX GIRDER WEB (2.2.5). At any section in a box girder the shear stress is calculated by determining the components of vertical and horizontal shear due to bending about the X and Y axes respectively, which act through the shear center of the section. This shear center may be assumed at the center of gravity under conditions of Section 2.2.5 (3). The torsional moment at any section is the summation of contributions due to both horizontal and vertical forces that are not applied through the shear center. An applied torque at any point in the girder is in equilibrium with resisting torsional moments on either side of the point of application that are in proportion to the torsional stiffness of each segment. In a straight girder of constant cross-section held torsionally at each end, the torsional stiffness to either side of a point where a torque is introduced is inversely proportional to the relative distance to the end of the girder. Thus if a torsional moment is introduced at the quarter point of a straight girder of constant cross-section, three-quarters of the torsional moment will go to the short segment and one-quarter to the long segment. If the girder is not of constant cross-section or is held in a different manner at the two ends, the calculation of torsional stiffness is com-



Where:

$$R_1 = \frac{t_{fc}}{t_{ft}}$$

$$R_2 = \frac{t_{fc}}{t_w}$$

Fig. 5 — Bridge girder — Cross-section showing location of shear center and center of gravity

plex and requires the determination of the summation of incremental torsional stiffnesses of successive segments in either direction from the point considered.

WEB SHEAR STRESS IN SYMMETRIC BOX GIRDERS (2.2.6). Fig. 6 shows the two shear components and the torsional moment acting at the shear center which in this case is the centroid of the cross-section.

In checking the maximum stress at Point A (Fig. 6), the shear stress due to bending moment may be calculated by Eq 3. The shear stress due to torsion, which is calculated according to Bredt's Theory by Eq 5, is additive at A to the shear stress caused by the vertical shear V_y . The shear stresses due to the horizontal component of shear forces, V_x , should also be calculated if the lateral forces are appreciable. At Point A the shear stress due to V_x is zero in the symmetrical case and therefore it may be neglected with little error. If, however, V_x is large in comparison with V_y , the maximum stress due to all three components may be greater at some point above Point A in the figure.

Diaphragms play a very important role in box girders. Local applications of vertical force can be introduced as shears into both webs only if there is a diaphragm at the point of load application. Otherwise, cross bending of the flanges would ensue and one side of the box girder would be stressed more than the other. It would deflect more than the other side with a resultant loss in the shape of the box

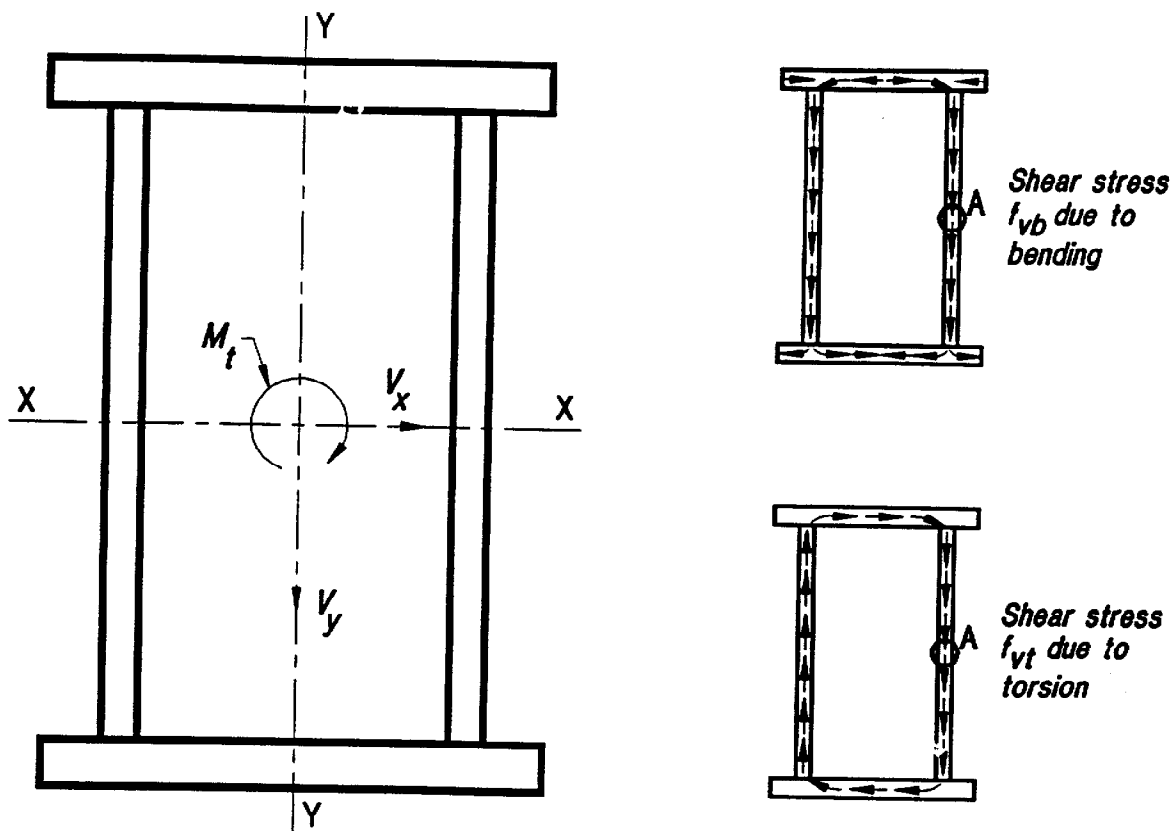


Fig. 6 — Bridge girder — Torsion and shear loads and stresses

cross-section. The same is true of horizontal forces delivered at the top flange or at the top of the rail. The distribution of these forces to the lower flange is made somewhat complex, especially if the diaphragm cannot be welded to the lower flange as is usually the case. The result is a localized bending of the web near the bottom of the girder. The determination of the stress distribution is a complex analytical problem which probably requires consideration only in unusual situations. The diaphragms have an additional function of distributing applied torques into both webs and both flanges, all of which work together in resisting the torsional moment. In short, the diaphragms maintain the shape of the box and permit the assumption that the entire cross-section participates in resisting both vertical and horizontal forces and that Bredt's Theory for the torsional behavior of closed (box) sections may be applied.

COLLISION EFFECTS (2.2.10). This Section is designed to provide adequate safety with regard to the accidental stoppage of a crane at the end of a runway or by impact with another crane. In such a situation the kinetic energy of crane motion is equal to the total work done or energy absorbed during stoppage. This kinetic energy is based on

the assumption of an equivalent spring and mass system, the mass being due to an equivalent concentration of weight at the center of the crane which is equal to the concentrated weight plus one-half of the distribution weight of the crane system. It is presumed that the suspended load will swing and that its mass need not be added to the total. The equivalent force that is calculated is the maximum force at the end of travel, in coming to a stop and the deflection through which the force is assumed to be the combined deflection of the bumper stops and the deflection at the center of the crane itself (all assumed to act as a linear spring system). If the stops at the end of the runway are hydraulic, with a more nearly constant resisting force during travel, the equivalent force at midspan will be somewhat less. If the stops consist of rubber bumpers or if the springs 'bottom' the force may be much greater. During design if the forces produced, using AISE guidelines as given in Section 3.8 for deceleration or bumper energy capacity, are excessive, these can be reduced by increasing the stroke of the hydraulic or spring bumper. Timely and efficient information transfer between crane manufacturer and building designer will result in significant economies. For cranes with vertically guided columns, W_A should be added and if the live load W_L cannot swing freely, it too should be added to W_E .

DESIGN EXAMPLE — STRUCTURAL

Crane Bridge Girder

Crane Specifications:

Crane Type:	2 Girder EOT; Mill Duty, Indoor Service
Capacity:	50 Tons
Span:	100 ft 0 in. (1,200 in.)
Wheel Base:	17 ft 0 in. Bridge (approximately one-sixth of span)
	10 ft 0 in. Trolley
Trolley Span:	12 ft 0 in.
Bridge Wheels:	(8) 24 in. diameter
	(4 wheels driven)
Drive Type:	Double A-5; 5:1 ratio each gear box
	End gear boxes positioned at 6 ft 0 in. from runway.
Hoist Speed:	80 fpm
Bridge Speed:	250 fpm
Bridge Motors:	50 HP @ 460 rpm
Weights:	
Trolley Weight (W_T)	= 64,000 lb (excluding hook block)
Hook Block Weight	= 6,000 lb
Lifted Load (W_L)	= 100,000 + 6,000 = 106,000 lb
Bridge Gear Box	= 3,000 lb each
Bridge Motors	= 7,000 lb each
Trolley Conductors	= 4,000 lb each girder
Bridge Line Shaft	= 12,000 lb each girder
Bridge Footwalks	= 15,000 lb each girder
Bridge Girder	= 60,000 lb each (design assumption)

Notes:

1. Only one of the bridge girders will be designed in this example. This will be that girder with no auxiliary truss and without loads due to cab and controls.
2. For simplicity, the weight of the trolley and lifted load will be assumed to be evenly distributed on the four trolley wheels.
3. For the illustration purposes of this example, only Design Load Combinations No. 1 and No. 2 will be considered.
4. Design calculations presented in this example will pertain only to the point along the girder where the trolley produces the maximum live load moment. It must be understood by the reader that similar design calculations are required at other points along the girder with the trolley positioned at or near those points for maximum loading conditions (i.e.: maximum vertical shear will occur with the trolley positioned at the close end approach; maximum lateral moment at the girder ends due to horizontal inertia will occur with the trolley positioned at one-third span from the end, etc.).

Vertical Forces

$$\begin{aligned}
 \text{Trolley wheel load, without impact} &= \frac{(100,000 + 6,000 + 64,000)}{4} \\
 &= 42,500 \text{ lb} \\
 &= 42.5 \text{ kips}
 \end{aligned}$$

$$\begin{aligned}
 \text{Vertical Impact} &= 0.3W_L = 0.3 \times 106,000 \\
 &= 31,800 \text{ lb} \\
 &= 31.8 \text{ kips}
 \end{aligned}$$

$$\begin{aligned}
 \text{Vertical Impact per Trolley Wheel} &= \frac{31.8}{4} = 7.95 \text{ kips}
 \end{aligned}$$

Total uniform dead load on girder is as follows:

Bridge girder and rail	= 60,000 lb
Trolley conductors	= 4,000 lb
Line shaft	= 12,000 lb
Footwalk	= <u>15,000</u> lb
Total	= 91,000 lb
	= 91 kips

Note: For moving concentrated loads, the maximum bending moment will occur when the centerline of the span is midway between the center of gravity of loads and the nearest concentrated load.

Based on the above rule, the distance from the runway support to the nearest trolley wheel for maximum live load moment will be as follows for equal wheel loads:

$$\frac{1200}{2} - \frac{120}{4} = 570 \text{ in.}$$

From Fig. 7:

$$R_L = \frac{(42.5 \times 510 + 42.5 \times 630)}{1200} = 40.375 \text{ kips}$$

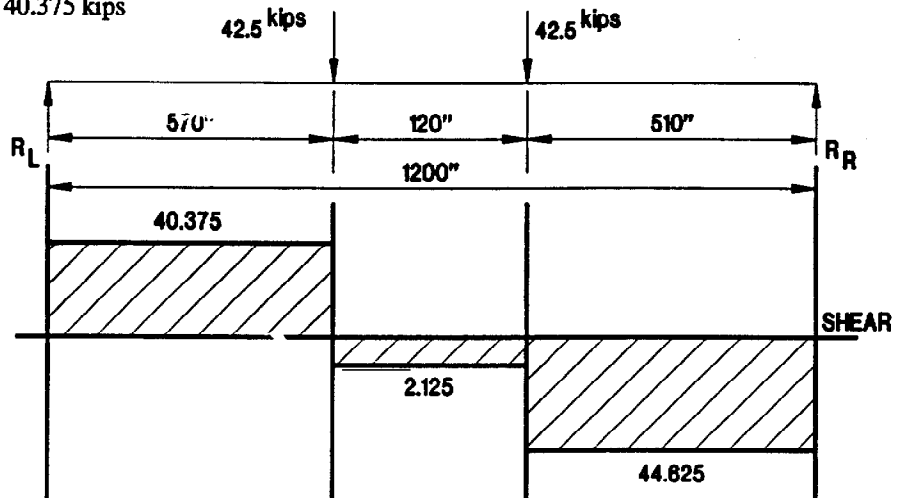


Fig. 7 — Bridge girder loading and shear diagram for maximum vertical live load moment without impact

From Fig. 8:

$$R_L = \frac{(7.95 \times 510 + 7.95 \times 630)}{1200} = 7.55 \text{ kips}$$

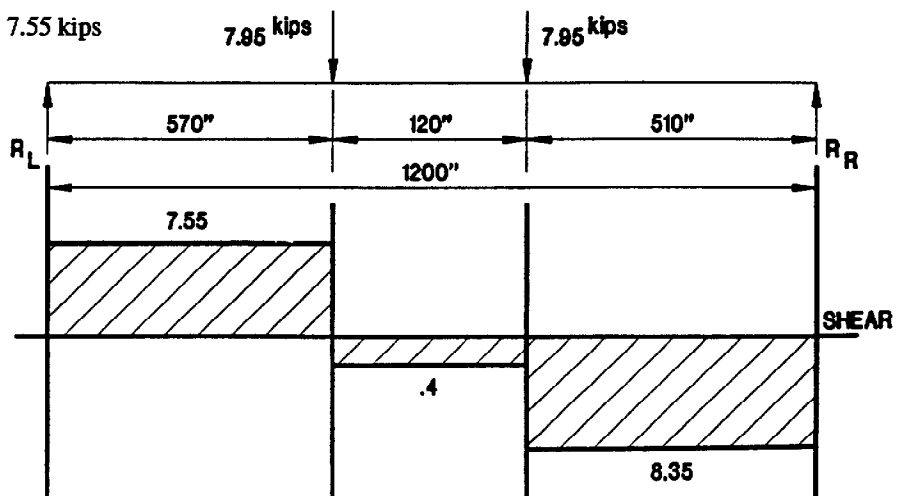
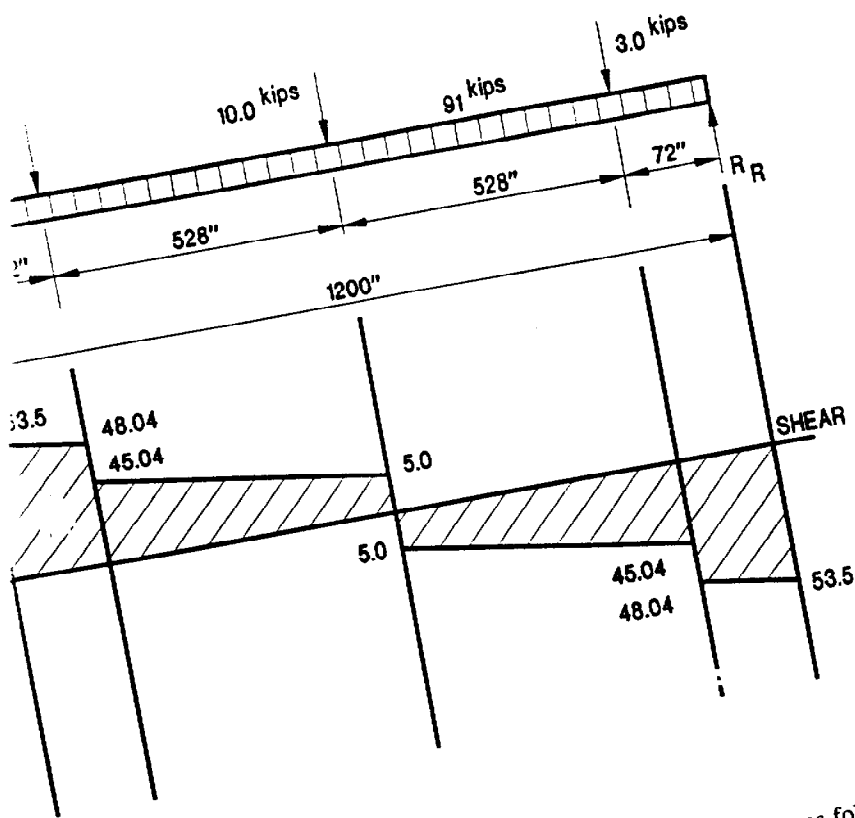


Fig. 8 — Bridge girder loading and shear diagram for maximum vertical impact loading



At the point of maximum live load moment, the vertical moments on the girder are as follows (see

$$= 40.375 \times 570$$

$$= 23,014 \text{ in.-kips}$$

$$= 7.55 \times 570$$

$$= 4,304 \text{ in.-kips}$$

$$= 3.0 \times 72 + 10.0 \times 0.5 \times 570 + \frac{91.0 \times 570 \times (1200 - 570)}{1200 \times 2}$$

$$= 16,682 \text{ in.-kips}$$

acting on the bridge girder are calculated by multiplying the vertical loads by the force factor, f_f ,

$$\frac{\text{wheels}}{\text{wheels}} = 0.10$$

loads are as follows:

$$0.0 \times 0.10 = 9.1 \text{ kips}$$

$$< 0.10 = 1.0 \text{ kips}$$

$$\times 0.10 = 4.25 \text{ kips each wheel}$$

use of this example, the inertia force of the 3 kip loads at the girder ends (end gearbox weight) will be

The bending moments resulting from the horizontal inertia forces are found by rigid frame analysis. For the purpose of this example, it will be assumed that the end ties have the same lateral moment of inertia (I_y) as the main box girder and that both girders are loaded equally in the horizontal direction.

Also, for simplicity, the horizontal inertia forces due to live load will be added together and allied at a single point on the girder (i.e.: $4.25 + 4.25 = 8.5$ kips).

The resulting moment diagram is shown in Fig. 10. There are, of course, several different methods of rigid frame analysis that may be used in obtaining these values. Since the calculations involved are somewhat long and tedious, they have been omitted from this example.

Skewing Forces

Skewing force F_s (Fig. 11)

$$= f_f (W_T + W_A + W_L) \left(0.5 - \frac{a}{l} \right)$$

Where

$$f_f = 0.10 \text{ (as previously calculated)}$$

$$W_T = 64 \text{ kips}$$

$$W_A = 0 \text{ (no column)}$$

$$W_L = 106 \text{ kips}$$

$$a = 570 \text{ in.}$$

$$l = 1200''$$

$$F_s = 0.10 (64 + 106) \left(\frac{0.5 - 570}{1200} \right) = 0.425 \text{ kips}$$

$$H_s = F_s \times \frac{l}{s} = 0.425 \times \frac{1200}{144} = 3.54 \text{ kips}$$

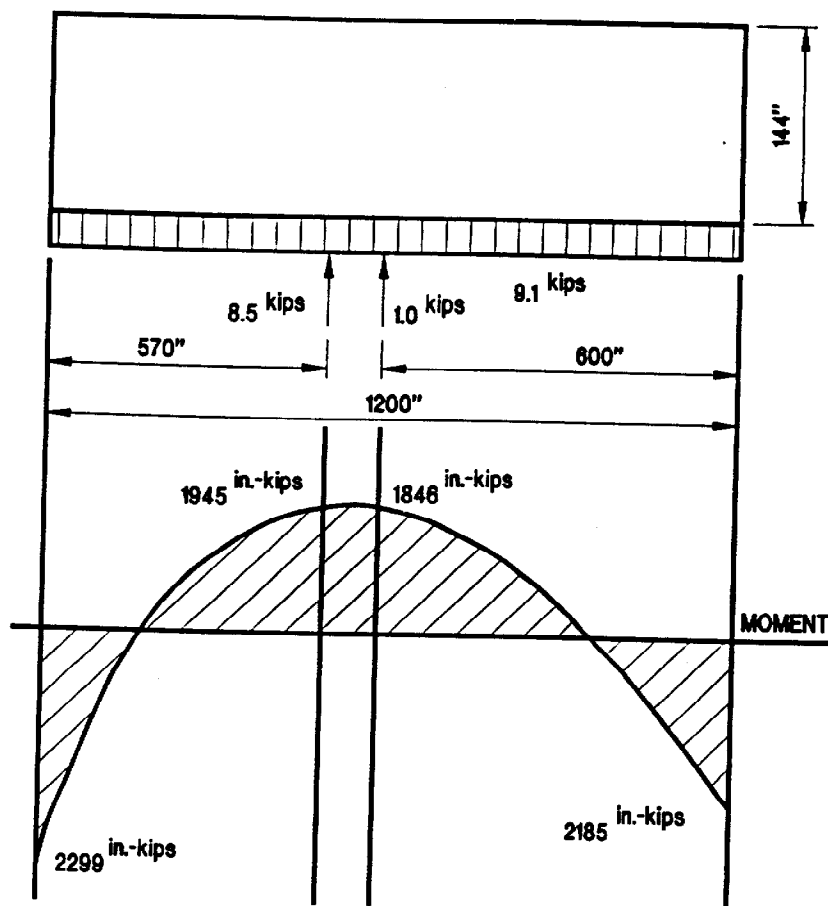
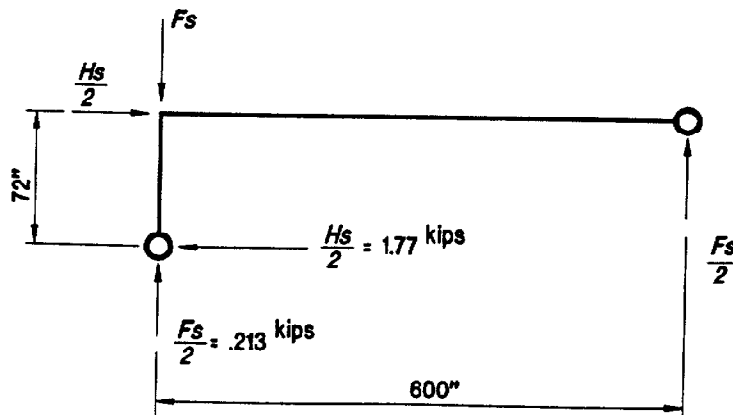


Fig. 10 — Bridge girder horizontal loading and moment diagram (trolley positioned at point of maximum vertical L.L. moment)

Fig. 11 — Bridge skewing forces (trolley at point of maximum vertical L.L. moment)



Skewing moment at 570 in. from runway = $0.213 \times (600 - 570) = 6.39$ in.-kips

Section Properties: (Fig. 12)

$$Area = 106.3 \text{ in.}^2$$

$$I_x = 132,189 \text{ in.}^4$$

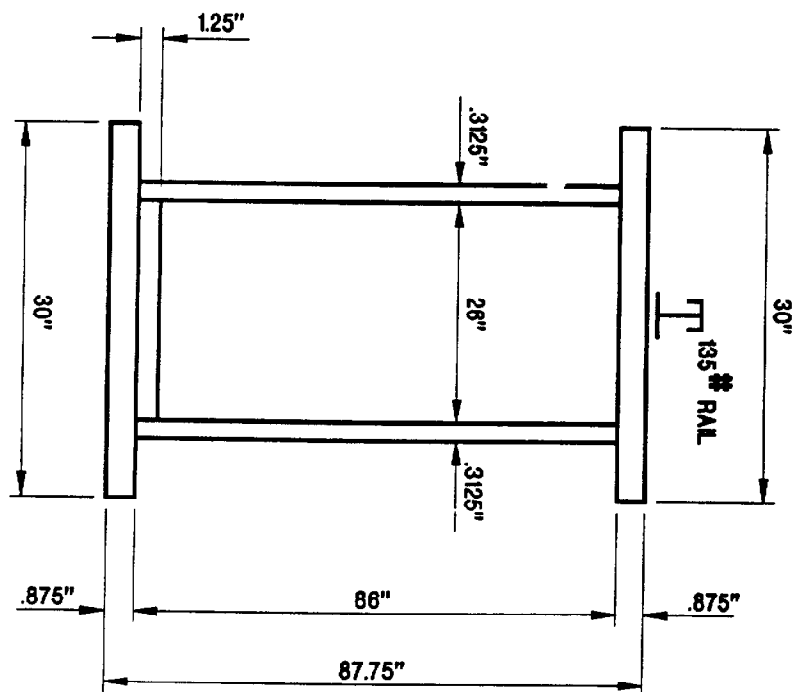
$$I_y = 13,241 \text{ in.}^4$$

$$r_x = 35.3 \text{ in.}$$

$$r_y = 11.2 \text{ in.}$$

Note: Due to symmetry, the shear center is located at the centroid of the section.

Fig. 12 — Trial girder section



Check girder proportions:

$$\frac{l}{b} = \frac{1200}{26.3125} = 45.6 < 60 \text{ OK}$$

$$\frac{l}{d} = \frac{1200}{86.875} = 13.8 < 18 \text{ OK}$$

$$\frac{w}{t} = \frac{26}{0.875} = 29.7 < 39.7 \text{ (Table 5) OK}$$

$$\frac{Kl}{r_y} = 1.0 \times \frac{1200}{11.2} \text{ (K assumed as 1.0)} = 107 < 126.1 \text{ (Table 5) OK}$$

$$\frac{D}{t} = \frac{86}{0.3125} = 275.2 > 170 \text{ Longitudinal stiffeners are required.}$$

$$\frac{D}{t} = 275.2 < 340 \text{ OK}$$

Required distance from tension flange to end of vertical stiffener (AISC, "Specification for the Design, Fabrication and Erection of Structural Steel for Buildings")

$$= 4t = 4 \times 0.3125 = 1.25 \text{ in. min}$$

$$= 6t = 6 \times 0.3125 = 1.875 \text{ in. max}$$

Torsional constant of box section

$$J = \frac{4A^2}{\sum \frac{w}{t}} = \frac{4 \times (26.3125 \times 86.875)^2}{2 \left(\frac{26.3125}{0.875} + \frac{86.875}{0.3125} \right)} = 33,923 \text{ in.}^4$$

Equivalent column slenderness ratio (assume $K = 1.0$)

$$= K \times \sqrt{\frac{5.1 L S_x}{\sqrt{J I_y}}} = 1.0 \times \sqrt{\frac{5.1 \times 1200 \times \left(\frac{132,189}{43.875} \right)}{\sqrt{33,923 \times 13,241}}} = 29.5$$

Allowable base stress for compression, using above column slenderness ratio and AISC Manual of Steel Construction

$$F_{bx} = 19.98 \text{ ksi}$$

Allowable base stress for lateral and vertical tension

$$F_{by} = F_{bx} = 22.0 \text{ ksi (Table 1)}$$

Torsional Moments

Applied torque due to center loads (motor + gear box weight at 45 in. from center of girder)

$$= 10 \times 45 = 450 \text{ in.-kips (225 in.-kips each way)}$$

Applied torque due to end gearbox (also 45 in. arm)

$$= 3 \times 45 = 135 \text{ in.-kips (each end)}$$

Applied torque due to line shaft (45 in. arm)

$$= 12 \times 45 = 540 \text{ in.-kips (uniform)}$$

Applied torque due to footwalk (44 in. arm)

$$= 15 \times 44 = 660 \text{ in.-kips (uniform)}$$

Applied torque due to trolley conductors (24 in. arm)

$$= 4 \times 24 = 96 \text{ in.-kips (uniform)}$$

Total applied torque due to uniform load

$$= 540 + 660 + 96 = 1,296 \text{ in.-kips (648 in.-kips each way)}$$

Torque (each side) due to center drive

$$= \frac{50 \times 5250}{460} \times 5.0 \times 2 \times 0.667 \times \frac{12}{1000} = 45.68 \text{ in.-kips}$$

Torque (at ends) due to end gear boxes

$$= 45.68 \times 5.0 = 228.4 \text{ in.-kips}$$

The moment arm for torque due to live load inertia can be defined as the distance from the top of the trolley rail to the shear center of the girder. For this example this arm is

$$87.75 \times 0.5 + 5.75 = 49.625 \text{ in.}$$

The torque due to live load inertia can be found by multiplying the live load shear (Fig. 7) by the product of the force factor, f_p , and the moment arm.

$$40.375 \times 0.10 \times 49.625 = 200.4 \text{ in.-kips}$$

$$2.125 \times 0.10 \times 49.625 = 10.5 \text{ in.-kips}$$

$$44.625 \times 0.10 \times 49.625 = 221.5 \text{ in.-kips}$$

Torque due to eccentricity of trolley rail = 0

(Trolley wheel loads pass thru shear center of girder about axis Y-Y, thus producing no torque from vertical loading.)

Total torsional moment at 570 in. (Figs. No. 13 through 16):

Motor and gearbox weight	= 225 in.-kips
Uniform weight	= 32 in.-kips
Drive torque	= 46 in.-kips
Live load inertia	= 200 in.-kips
Total	= 503 in.-kips

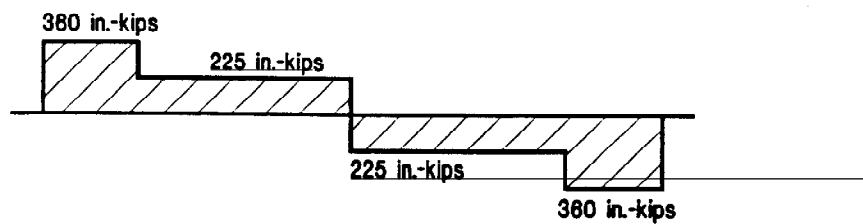


Fig. 13 — Bridge girder torsional moment diagram due to weight of motor gearboxes

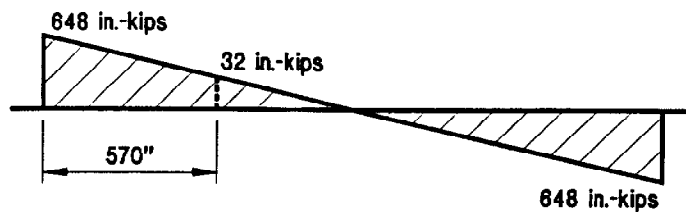


Fig. 14 — Bridge girder torsional moment diagram due to uniform load

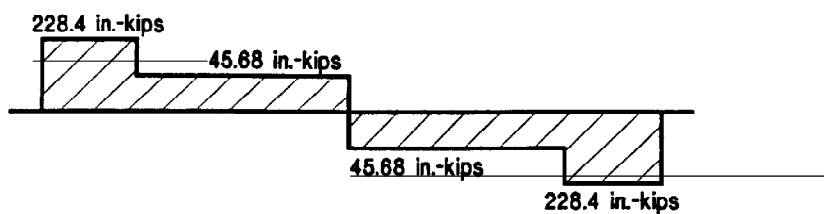


Fig. 15 — Bridge girder torsional moment diagram due to drive torque

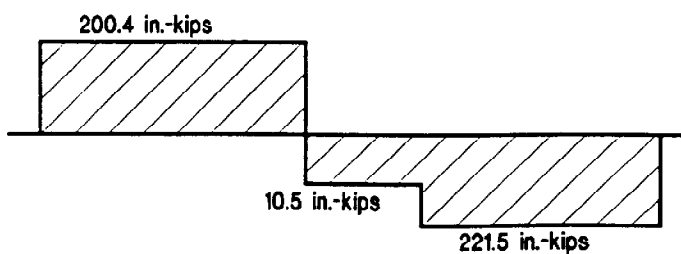


Fig. 16 — Bridge girder torsional moment diagram due to live load inertia

Stress Calculations

Design Load Combination No. 1 (Trolley positioned for maximum vertical moment)

Vertical bending stress on extreme fiber (tension or compression):

$$\text{Dead load} = 16,682 \times 43.875/132,189 = 5.54 \text{ ksi}$$

$$\text{Live load} = 23,014 \times 43.875/132,189 = 7.64 \text{ ksi}$$

$$\text{Impact} = 4,304 \times 43.875/132,189 = \underline{1.43} \text{ ksi}$$

$$\text{Total} = 14.61 \text{ ksi}$$

Horizontal bending stress on extreme fiber (tension or compression):

$$\text{Inertia} = 1,945 \times 15/13,241 = 2.20 \text{ ksi}$$

$$\text{Skewing} = 6.39 \times 15/13,241 = \underline{0.01} \text{ ksi}$$

$$\text{Total} = 2.21 \text{ ksi}$$

$$\text{Interaction formula value (compression)} = \frac{14.61}{19.98} + \frac{2.21}{22.0} = 0.83 < 1.0 \text{ OK}$$

$$\text{Interaction formula value (tension)} = \frac{14.61}{22.0} + \frac{2.21}{22.0} = .76 < 1.0 \text{ OK}$$

Vertical shear force at 570 in.:

$$\text{Dead load} = 7.28 \text{ kips (Fig. 7)}$$

$$\text{Live load} = 40.38 \text{ kips (Fig. 5)}$$

$$\text{Impact} = \underline{7.55} \text{ kips (Fig. 6)}$$

$$\text{Total} = 55.21 \text{ kips}$$

$$\text{Vertical shear stress (webs)} = \frac{VQ}{2I_x t} = \frac{55.21 \times (26.875 \times 21.5 + 26.25 \times 43.4375)}{132,189 \times 0.3125 \times 2} = 1.15 \text{ ksi}$$

$$\text{Torsional shear stress (webs)} = \frac{M}{2At} = \frac{503}{2 \times (26.3125 \times 86.875) \times 0.3125} = 0.35 \text{ ksi}$$

$$\text{Total shear stress (webs)} = s = 1.15 + 0.35 = 1.50 \text{ ksi} < 14.5 \text{ OK}$$

Note: The above shear stress represents the maximum shear stress that will occur on the girder at 570 in. from the runway (point of maximum live load moment). The maximum shear stress that will occur on the girder is not, however, located at this point.

Check l/b ratio not to exceed the following:

$$(l/d) \times (f_{bx}/f_{by}) = \frac{1200 \times (5.54 + 7.64)}{86.875 \times (2.2 + 0.01)} = 82.4 > 45.6 \text{ OK}$$

Stress Calculations

Design Load combination No. 2

(Trolley positioned for maximum vertical moment)

1. Check stress in extreme fiber (tension):

Allowable stress range = 16 ksi (Stress Category B, Service Class 4)

Actual stress range (live load + impact) = 7.64 + 1.43 = 9.07 ksi < 16 ksi OK

Actual stress range (live load + horizontal) = 7.64 + 2.21 = 9.85 ksi < 16 ksi OK

2. Check stress at bottom of fillet weld connecting internal diaphragm to web plate (tension).

Allowable stress range = 12 ksi (Stress Category C, Service Class 4)

$$\text{Actual stress range (live load + impact)} = \frac{(23,014 + 4,304) \times 41.75}{132,189} = 8.63 \text{ ksi} < 12 \text{ ksi OK}$$

$$\text{Actual stress range (live load + horizontal)} = \frac{23,014 \times 41.75}{132,189} + \frac{(1945 + 6.39) \times 13.3125}{13,241} = 9.23 \text{ ksi} < 12 \text{ ksi OK}$$

Girder Diaphragm And Stiffener Requirements

1. Determine the maximum permissible spacing of the internal girder diaphragms for proper support of the trolley rail:

$$\text{Rail stress, } f_{br} = \frac{\text{Wheel load} \times \text{Span}}{6 S_x} \leq 16 \text{ ksi}$$

Where

Wheel load = 42.5 kips

Span = Unknown

$S_x = 17.2 \text{ in.}^3$ (135 lb rail)

Solving for the span yields

$$\text{Span} = \frac{16 \times 17.2 \times 6}{42.5} = 38.9 \text{ in.; say 39 in.}$$

2. Determine the required thickness, t , of the internal diaphragms:

Bearing pressure, f_p

$$= \frac{\text{wheel load}}{\text{Bearing width} \times t} \leq 29 \text{ ksi}$$

Bearing width = $5.1875 + 2 \times 0.875$

= 6.9375 in. (refer to Fig. 17)

Solving for the thickness, t , yields

$$t = \frac{42.5}{6.9375 \times 29} = 0.211 \text{ in.}$$

For fabrication convenience, use $t = 0.3125 \text{ in.}$
(same thickness as web plates)

3. Determine the minimum depth of short (intermediate) diaphragms:

Referring to Fig. 17, the maximum bending moment on the internal diaphragms

$$= 21.25 \times 9.53 + 21.25 \times 3.47 \times 0.5$$

$$= 239.4 \text{ in.-kips}$$

$$\text{Diaphragm bending stress} = \frac{239.4}{S_x} \leq 22 \text{ ksi}$$

Solving for S_x yields

$$S_x = \frac{239.4}{22} = 10.9 \text{ in.}^3$$

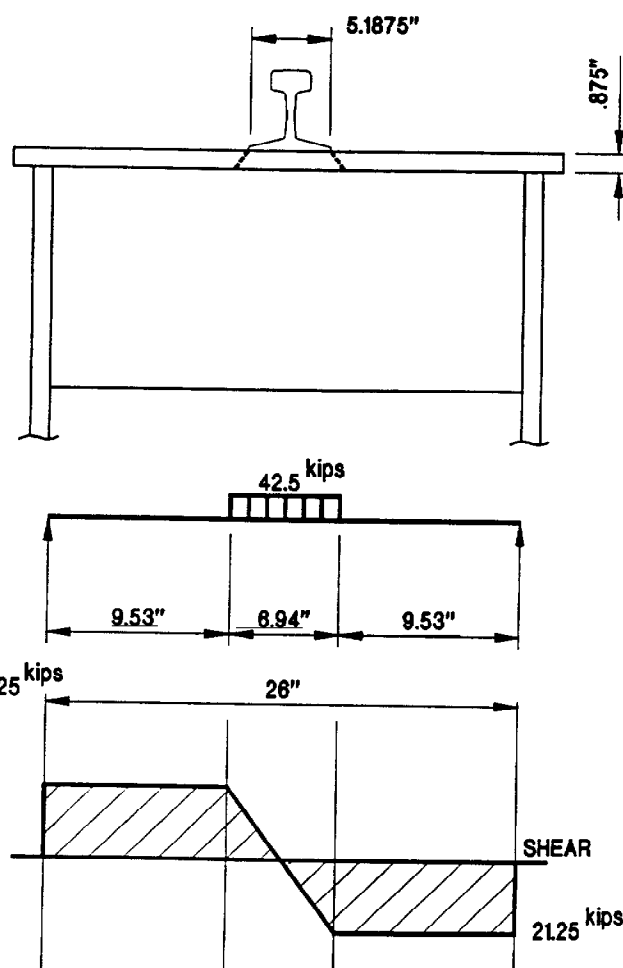


Fig. 17 — Load and shear diagram for internal girder diaphragms

For a thickness $t = 0.3125$

$$S_x = 10.9 = \frac{0.3125 \times d^2}{6}$$

Solving for depth, d , yields

$$d = 14.5 \text{ in. (use 16 in. deep plate)}$$

$$\text{Shear stress on short diaphragms} = \frac{21.25}{(16 \times 0.3125)} = 4.25 \text{ ksi} < 12.0 \text{ OK}$$

4. Determine the vertical diaphragm (stiffener) requirements at center:

$$\frac{237}{\sqrt{f_v}} = \frac{237}{\sqrt{1.5}} = 193.5$$

$$\frac{D}{t} = \frac{86}{0.3125} = 275.2 > 193.5$$

Full depth internal diaphragms or web stiffeners are required.

Required spacing of diaphragms at center

$$= \frac{350t}{\sqrt{f_v}} = \frac{350 \times 0.3125}{\sqrt{1.5}} = 89.3 \text{ in.}$$

or $D = 86 \text{ in.}$ spacing required at center.

Note: Since the maximum spacing of internal diaphragms is 39 in. for proper rail support, full depth diaphragms could be spaced at 78 in. at center of span, with a short (intermediate) diaphragm located midway between the full depth diaphragms. The spacing of full depth diaphragms will most likely have to be decreased, however, toward the ends of the girder due to an increase in the shear stress, f_v .

5. Determine the horizontal (longitudinal) stiffener requirements:

As noted previously, longitudinal stiffeners are required due to the web depth/thickness ratio.

$$\text{i.e.: } D/t = 86/0.3125 = 275.2 > 170$$

These stiffeners are also required because of the following relationship:

$$\frac{727}{\sqrt{f_b}} = \frac{727}{\sqrt{14.61 + 2.21}} = 177 < 275.2$$

Since the longitudinal stiffeners will be used, D/t of web shall not exceed the following:

$$\frac{1454}{\sqrt{f_b}} = \frac{1454}{\sqrt{14.61 + 2.21}} = 354.5 > 275.2 \text{ OK}$$

Required distance from bottom of top flange plate to center of stiffeners:

$$= D/5 = 86/5 = 17.2 \text{ in. (say 17 in.)}$$

$$\text{Required moment of inertia of stiffener about the face of the web} = 86 \times 0.3125^3 \left[\frac{2.4 \times 78^2}{86^2} - 0.13 \right] = 4.84 \text{ in.}^4$$

Suggested stiffener section : 4 in. $\times$ 3/8 in. bar

$$I_w = \frac{0.375 \times 4^3}{3} = 8 \text{ in.}^4 > 4.84 \text{ in.}^4 \text{ OK}$$

Check Vertical Live Load Deflection Of Girder

Note: Due to the rather large ratio of girder span to trolley wheel base, both trolley wheel loads will be added together and applied at a single point for determining the maximum vertical deflection. This approach will be conservative for design purposes.

$$\text{Girder deflection (live load)} = \frac{(42.5 + 42.5) \times 1200^3}{48 \times 30,000 \times 132,189} = 0.772 \text{ in.}$$

$$\text{Allowable deflection (live load)} = 1200 \times 0.001 = 1.2 \text{ in.} > 0.772 \text{ in. OK}$$

ine Required Girder Camber

$$\text{load deflection due to uniform weight} = \frac{5 \times 91 \times 1200^3}{384 \times 30,000 \times 132,189} = 0.516 \text{ in.}$$

$$\text{load deflection due to center weight} = \frac{10 \times 1200^3}{48 \times 30,000 \times 132,189} = 0.091 \text{ in.}$$

$$\text{load deflection due to end weight} = \frac{3 \times 72 \left(3 \times 1200^2 - 4 \times 72^2 \right)}{24 \times 30,000 \times 132,189} = 0.010 \text{ in.}$$

$$\text{al dead load deflection (maximum at center)} = 0.516 \text{ in.} + 0.091 \text{ in.} + 0.010 \text{ in.} = 0.617 \text{ in.}$$

$$\text{amber required} = 0.617 + 0.5(0.772) = 1.003 \text{ in.; say 1 in. camber.}$$

3 MECHANICAL

3.1 Allowable Stresses.

The allowable stresses have been divided into two Sections. Section 3.1.1 deals with allowable design stresses based on the endurance cycles (infinite life) while Section 3.1.2 deals with allowable design stresses based on the actual load application applied to the machinery (finite life).

In either case maximum working stresses in steel mill crane machinery components shall not exceed the maximum allowable stresses, σ_{BA} , σ_{NA} , σ_{XA} , σ_{YA} , τ_A , τ_{TA} , unless otherwise specified. The working stresses, σ_B , σ_N , σ_{BN} , σ_{EB} , σ_{EN} , σ_X ,

σ_Y , σ_{EXY} , σ_{EYT} , τ_S , τ_T , τ_{ET} , τ_{XY} , and τ_{EYT} , are either uniaxial, biaxial, shear, combined or equivalent stresses which are induced in a mechanical component by the working (operational) loads. The maximum working loads shall include dead loads, maximum live loads and acceleration and deceleration forces which result from normal operation of the crane. The maximum calculated working stresses shall include both service and stress concentration factors.

The allowable design stresses in Figs. 18, 19 and 20 are based on normal design conditions, such as machined sur-

faces, increased material size, ambient temperatures and reliability for steel mill service. If the component has a cast, hot rolled, welded or forged surface, or is subjected to surface corrosion, fretting, wear, elevated temperature or other deterrent effects, a reduction of the allowable stresses should be made in accordance with the severity of the existing or anticipated damaging effects.

3.1.1 Allowable Design Stresses (Infinite Life).

The allowable stresses σ_{BA} , σ_{NA} , τ_A and τ_{TA} which shall be obtained from Fig. 18, 19 and 20 vary with the minimum ultimate tensile strength, σ_{UT} , of the material in use, as well as with the fluctuation ratios, R_B , R_N , R_S , R_T , of the working stresses. σ_{XA} and σ_{YA} shall be selected from Fig. 18 or 19 depending on whether σ_X or σ_Y are basically bending or tensile-compressive stresses. τ_{TA} shall be selected directly from Fig. 20. The minimum ultimate tensile strength, σ_{UT} , shall be based on the tensile strength at mid-radius for the raw material size used.

Note: For allowable bearing stresses for pins see Section 2.2.11, Table 1.

3.1.1.1 Stress Ratio. For determining stress fluctuation ratio, it should be observed that the stress having the maximum absolute magnitude (regardless of whether it is a tension, compression, shear, combined or equivalent stress) is to be considered positive in all cases. The minimum stress is to be considered positive if it is of the same sense as the maximum stress, otherwise it is to be taken as negative, in which case R_B , R_N , R_S and R_T also become negative. Referring to Figs. 18, 19 and 20, in regions of combined stress σ_{max} or τ_{max} should be taken as the maximum combined or equivalent stress having the maximum absolute magnitude. σ_{min} or τ_{min} shall be taken as the absolute minimum stresses which do occur at the same location as the maximum stress.

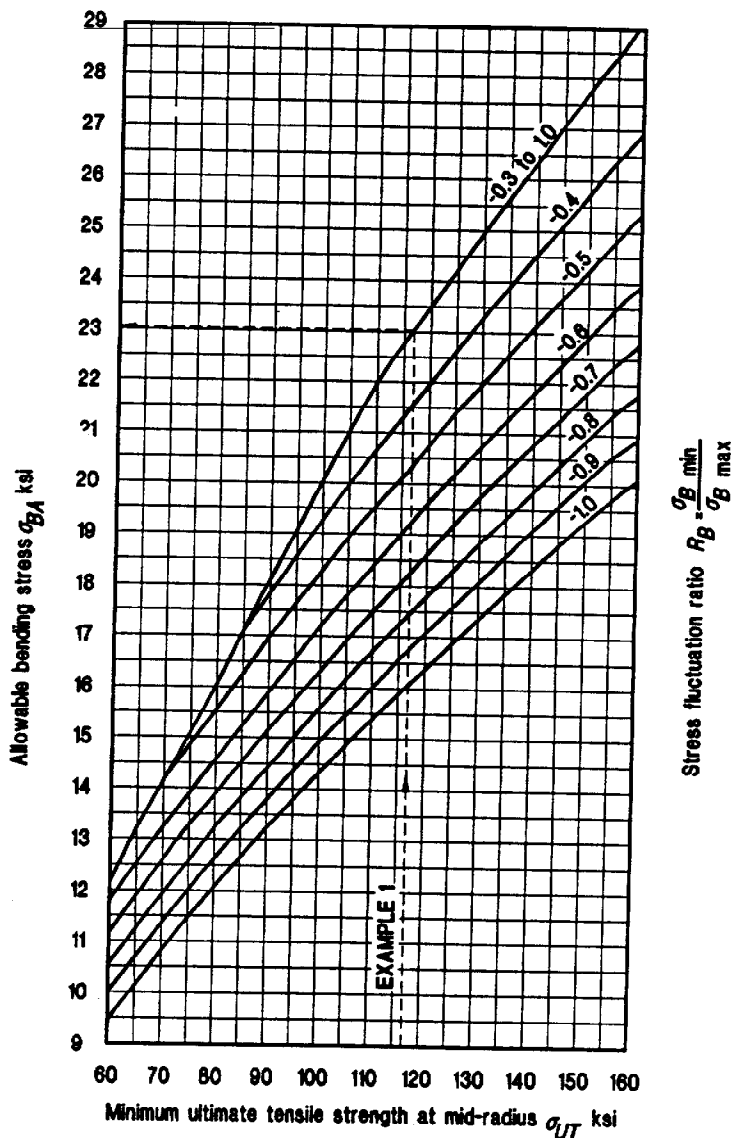


Fig. 18 — Plain pin in bending

3.1.2 Allowable Design Stresses (Finite Life).

The allowable stresses in this Section are determined in conjunction with the evaluation of the applied loading condition. If the loading condition is not supplied by the user or if the designer elects not to evaluate the loading condition, the allowable design stresses of Section 3.1.1 shall apply.

In determining the allowable design stress finite, the load spectra of the component is evaluated and an allowable stress modification factor (K_1) is determined. This factor divided into the allowable stresses from Figs. 18, 19 and 20 determines the new allowable stress. The K_1 stress factor can increase the allowable stress until it reaches the quasi-static ultimate strength of the material. In design application this stress is beyond the range of static limit stress ($\sigma_{UT}/5.0$). Therefore, in all applications the static limit stress shall be calculated and the lower of these two stresses shall be used as the determining factor.

In determining the allowable design stress for a machinery system it is recommended that the lowest cycle component be evaluated, then move progressively to the higher cycle component. If the stress modification factor K_1 is equal to or greater than 1 the allowable stress given in Figs. 18, 19 and 20 is already at its highest value. In these cases the cumulative cycle loading condition is equal to or greater than the endurance cycles of the component condition in question.

3.1.2.1 Allowable Stress Modification Factor (K_1). This factor is used to determine if any increase in allowable stresses is possible from evaluating the actual loading condition of a machinery component. In no case can the evaluated value of K_1 exceed 1; if this occurs the value of allowable stress equals Figs. 18, 19 and 20.

To determine K_1 , the KDS value (Form 2.00, Appendix A and OIS) stress class reduction factor, K , is taken from Section 3.1.2.2.

$$K_1 = K\sqrt{KDS} \quad (\text{Eq 29})$$

3.1.2.2 Stress Class Reduction Factor, K . The stress slope factor adjusts the allowable stress for the influence of cycles, material ultimate strengths, stress concentration and stress fluctuations. In determining K , consideration must be given to the effects of stress concentration in relation to their effects on cycles. The following cycle relation may be used.

- 2×10^6 — Fillet radius, keyways, drilled holes, etc.
- 10×10^6 — Gear strength and durability
- 20×10^6 — Press fits, fretting

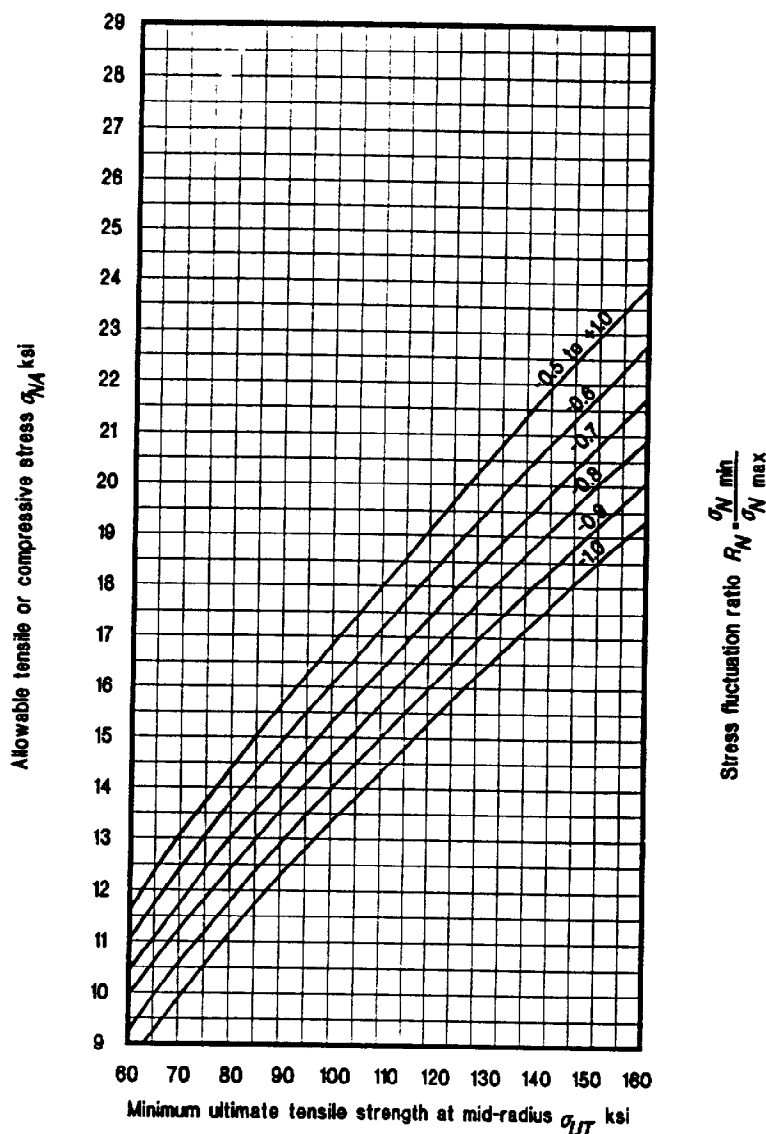


Fig. 19 — Allowable tensile or compressive stress

Table 6 gives the values of K for the above cycle relation for various values of K_{FT} .

K_{FT} = strength reduction factor

$$= \sigma_{UT} \times \frac{K_{NB}}{\sigma_{BA}} \quad (\text{Eq 30})$$

The stress concentration factor K_{NB} and the allowable stress σ_{BA} are entered depending on the mode of stress to be evaluated; bending, torsion, shear, etc.

3.1.3 Stress Concentration Factors.

Stress concentration factors, K_{NB} and K_{NT} , for shafting in bending and torsion may be obtained from Figs. 21, 22 and 23. These factors shall give consideration to the effects on the

Table 6 — Stress Class Reduction Factor, K

KFT	NE — Cycles			KFT	NE — Cycles		
	2×10^6	10×10^6	20×10^6		2×10^6	10×10^6	20×10^6
3.00	9.3731	11.3577	12.2125	27.00	2.5268	3.0618	3.2922
3.50	7.8759	9.5436	10.2619	27.50	2.5114	3.0432	3.2723
4.00	6.9186	8.3836	9.0145	28.00	2.4966	3.0252	3.2529
4.50	6.2487	7.5718	8.1417	28.50	2.4822	3.0077	3.2341
5.00	5.7506	6.9683	7.4927	29.00	2.4681	2.9907	3.2158
5.50	5.3638	6.4996	6.9887	29.50	2.4545	2.9742	3.1981
6.00	5.0535	6.1236	6.5844	30.00	2.4413	2.9582	3.1808
6.50	4.4982	5.8142	6.2517	30.50	2.4284	2.9426	3.1640
7.00	4.5837	5.5543	5.9723	31.00	2.4158	2.9274	3.1477
7.50	4.4007	5.3325	5.9338	31.50	2.4036	2.9125	3.1317
8.00	4.2421	5.1404	5.5272	32.00	2.3917	2.8981	3.1162
8.50	4.1033	4.9722	5.3463	32.50	2.3801	2.8840	3.1011
9.00	3.9505	4.8233	5.1863	33.00	2.3687	2.8703	3.0863
9.50	3.8709	4.6905	5.0435	33.50	2.3577	2.8569	3.0719
10.00	3.7723	4.5711	4.9151	34.00	2.3469	2.8439	3.0579
10.50	3.6832	4.4630	4.7989	34.50	2.3364	2.8311	3.0442
11.00	3.6020	4.3646	4.6931	35.00	2.3261	2.8186	3.0308
11.50	3.5276	4.2746	4.5963	35.50	2.3160	2.8064	3.0177
12.00	3.4593	4.1918	4.5073	36.00	2.3062	2.7945	3.0048
12.50	3.3962	4.1153	4.4251	36.50	2.2966	2.7829	2.9923
13.00	3.3377	4.0445	4.3488	37.00	2.2872	2.7715	2.9801
13.50	3.2833	3.9785	4.2780	37.50	2.2780	2.7603	2.9681
14.00	3.2325	3.9170	4.2188	38.00	2.2690	2.7494	2.9564
14.50	3.1850	3.8594	4.1499	38.50	2.2602	2.7387	2.9442
15.00	3.1404	3.8053	4.0917	39.00	2.2515	2.7283	2.9336
15.50	3.0984	3.7545	4.0370	39.50	2.2431	2.7180	2.9226
16.00	3.0588	3.7065	3.9855	40.00	2.2348	2.7080	2.9118
16.50	3.0214	3.6612	3.9367	40.50	2.2266	2.6981	2.9012
17.00	2.9960	3.6182	3.8905	41.00	2.2187	2.8885	2.8908
17.50	2.9524	3.5775	3.8467	41.50	2.2108	2.6790	2.8806
18.00	2.9204	3.5388	3.8051	42.00	2.2032	2.6697	2.8706
18.50	2.8900	3.5019	3.7655	42.50	2.1956	2.6605	2.8608
19.00	2.8610	3.4668	3.7277	43.00	2.1882	2.6516	2.8511
19.50	2.8333	3.4332	3.6916	43.50	2.1810	2.6428	2.8417
20.00	2.8068	3.4011	3.6571	44.00	2.1739	2.6342	2.8324
20.50	2.7814	3.3704	3.6240	44.50	2.1669	2.6257	2.8233
21.00	2.7571	3.3409	3.5923	45.00	2.1600	2.6173	2.8143
21.50	2.7338	3.3126	3.5619	45.50	2.1532	2.6091	2.8055
22.00	2.7114	3.2855	3.5327	46.00	2.1466	2.6011	2.7968
22.50	2.6898	3.2593	3.5045	46.50	2.1400	2.5932	2.7883
23.00	2.6690	3.2342	3.4776	47.00	2.1336	2.5854	2.7800
23.50	2.6490	3.2099	3.4515	47.50	2.1273	2.5777	2.7717
24.00	2.6297	3.1866	3.4264	48.00	2.1211	2.5702	2.7636
24.50	2.6111	3.1640	3.4021	48.50	2.1150	2.5628	2.7557
25.00	2.5931	3.1422	3.3787	49.00	2.1089	2.5555	2.7478
25.50	2.5757	3.1211	3.3560	49.50	2.1030	2.5483	2.7401
26.00	2.5589	3.1007	3.3340	50.00	2.0972	2.5412	2.7325
26.50	2.5426	3.0809	3.3128				

Figure 1 is a design chart for determining the allowable shear stress (τ_A) in ksi based on the minimum ultimate tensile strength at mid-radius (σ_{UT}) in ksi. The chart includes curves for stress fluctuation ratios ($R_S = \frac{\tau_{min}}{\tau_{max}}$) of 0.0, 0.2, 0.4, 0.6, 0.8, and 1.0. A dashed line labeled "Example 1" shows a path from $\sigma_{UT} = 100$ ksi to $R_S = 0.6$ and then to $\tau_A = 11.5$ ksi.

reaction, such as trol-
frame supported on four

shall not be less than 1.0 for cranes used in such applications. The values of K_{SB} , K_{SN} , K_{SS} and K_{ST} may be determined by the following formulae, but must be entered into calculation even if equal to 1.0.

Stresses. The following basic stress formulas apply when determining working stresses.

$$\sigma_B = \frac{M_B}{S_B} \times K_{SB} \times K_{NB} \leq \sigma_{BA}$$

$$\sigma_N = \frac{P}{A} \times K_{SN} \times K_{NN} \leq \sigma_{NA}$$

$$\sigma_{EBN} = \sigma_B + \frac{\sigma_{BA}}{\sigma_{NA}} \times \sigma_N \leq \sigma_{BA}$$

$$\tau_S = \frac{Q \times P}{I \times t} \times K_{SS} \times K_{NS} \leq \tau_A \quad (\text{Eq 34})$$

$$\tau_S = \frac{1.33 \times t'}{A} \times K_{SS} \times K_{NS} \leq \tau_A \quad (\text{Eq 35})$$

(For maximum shear stress of a circular section.)

$$\tau_T = \frac{M_T}{S_T} \times K_{ST} \times K_{NT} \leq \tau_{TA} \quad (\text{Eq 36})$$

$$\tau_{ET} = \tau_T + \frac{\tau_{TA}}{\tau_A} \times \tau_S \leq \tau_{TA} \quad (\text{Eq 37})$$

$$\tau_{EXYT} = \tau_T + \frac{\tau_{TA}}{\tau_{XYA}} \times \tau_{XY} \leq \tau_{TA} \quad (\text{Eq 38})$$

$$\sigma_{EB} = \sqrt{\sigma_B^2 + \left(\frac{\sigma_{BA}}{\tau_{TA}} \right)^2 \times \tau_{ET}^2} \leq \sigma_{BA} \quad (\text{Eq 39})$$

$$\sigma_{EB} = \sigma_B \times K_{EB} \leq \sigma_{BA} \quad (\text{Eq 40})$$

$$\sigma_{EN} = \sqrt{\sigma_N^2 + \left(\frac{\sigma_{NA}}{\tau_{TA}} \right)^2 \times \tau_{ET}^2} \leq \sigma_{NA} \quad (\text{Eq 41})$$

$$\sigma_{EN} = \sigma_N \times K_{EN} \leq \sigma_{NA} \quad (\text{Eq 42})$$

$$\sigma_{EXY} = \sqrt{\sigma_X^2 + \sigma_Y^2 \left(\frac{\sigma_{XA}}{\sigma_{YA}} \right)^2} - \sigma_X \left(\frac{\sigma_{XA}}{\sigma_{YA}} \right) \sigma_Y \leq \sigma_{XA} \quad (\text{Eq 43})$$

$$\sigma_{EXY} = \sigma_X \times K_{EXY} \leq \sigma_{XA} \quad (\text{Eq 44})$$

$$\sigma_{EXYT} = \sqrt{\sigma_X^2 + \sigma_Y^2 \left(\frac{\sigma_{XA}}{\sigma_{YA}} \right)^2} - \sigma_X \times \sigma_Y \left(\frac{\sigma_{XA}}{\sigma_{YA}} \right) + \left(\frac{\sigma_{XA}}{\sigma_{YA}} \right)^2 \times \tau_{EXYT}^2 \leq \sigma_{XA} \quad (\text{Eq 45})$$

Where applicable, these equations must be used in determining basic stresses in crane machinery components. For determining size of machinery components, the maximum working (operational) moments and shear loads as well as critical section moduli must be entered into the formulas.

Sign convention must be observed when entering σ_X and σ_Y in Eqs 43, 44 and 45. (Tension is positive, compression is negative).

Only stresses which do occur simultaneously at the location where stress is being calculated should be combined. In Eqs 37 through 45 anisotropy and stress fluctuation have been given consideration in a simplified manner for easier use in the design engineering process.

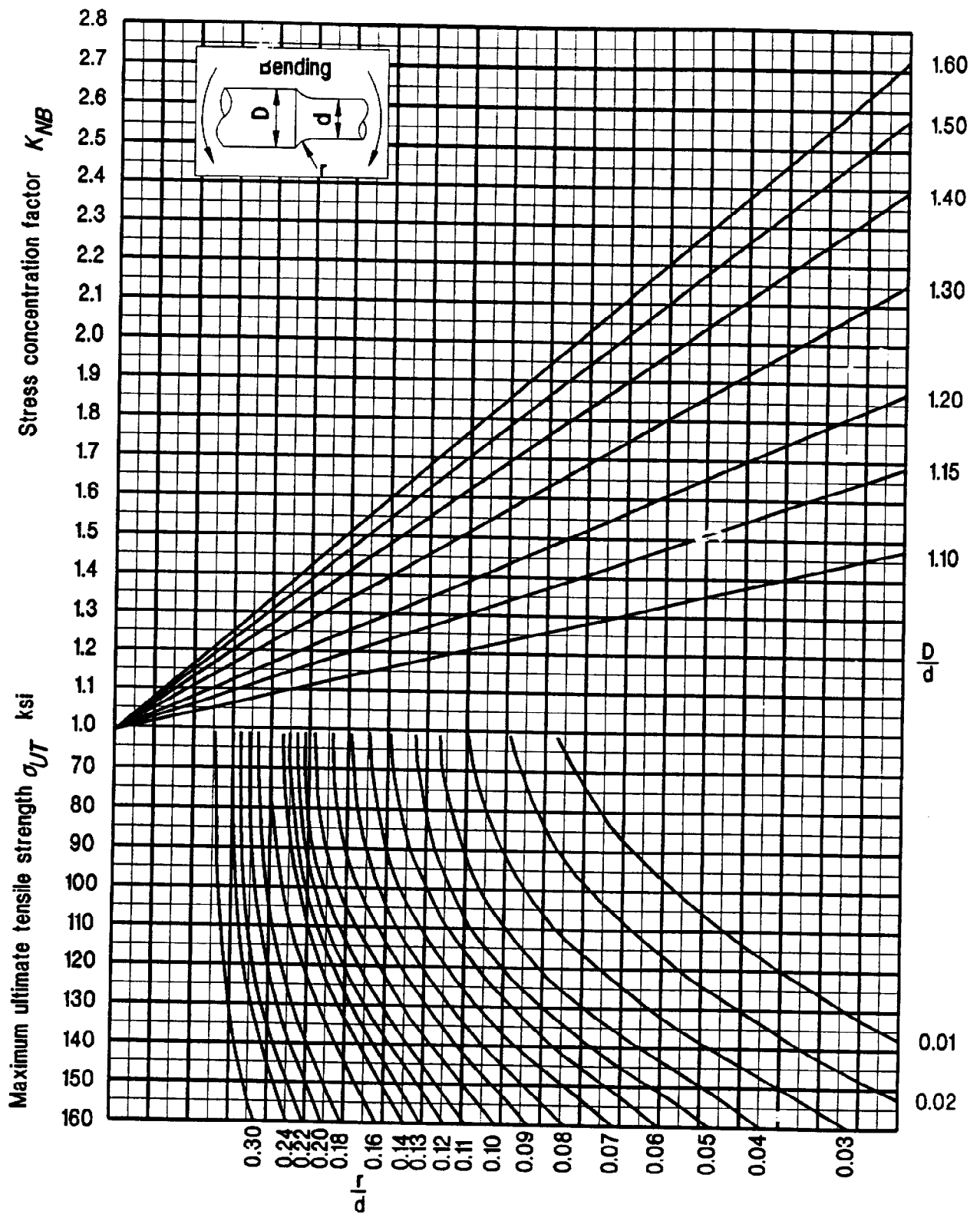


Fig. 21 — Stepped shaft in bending—stress concentration factors

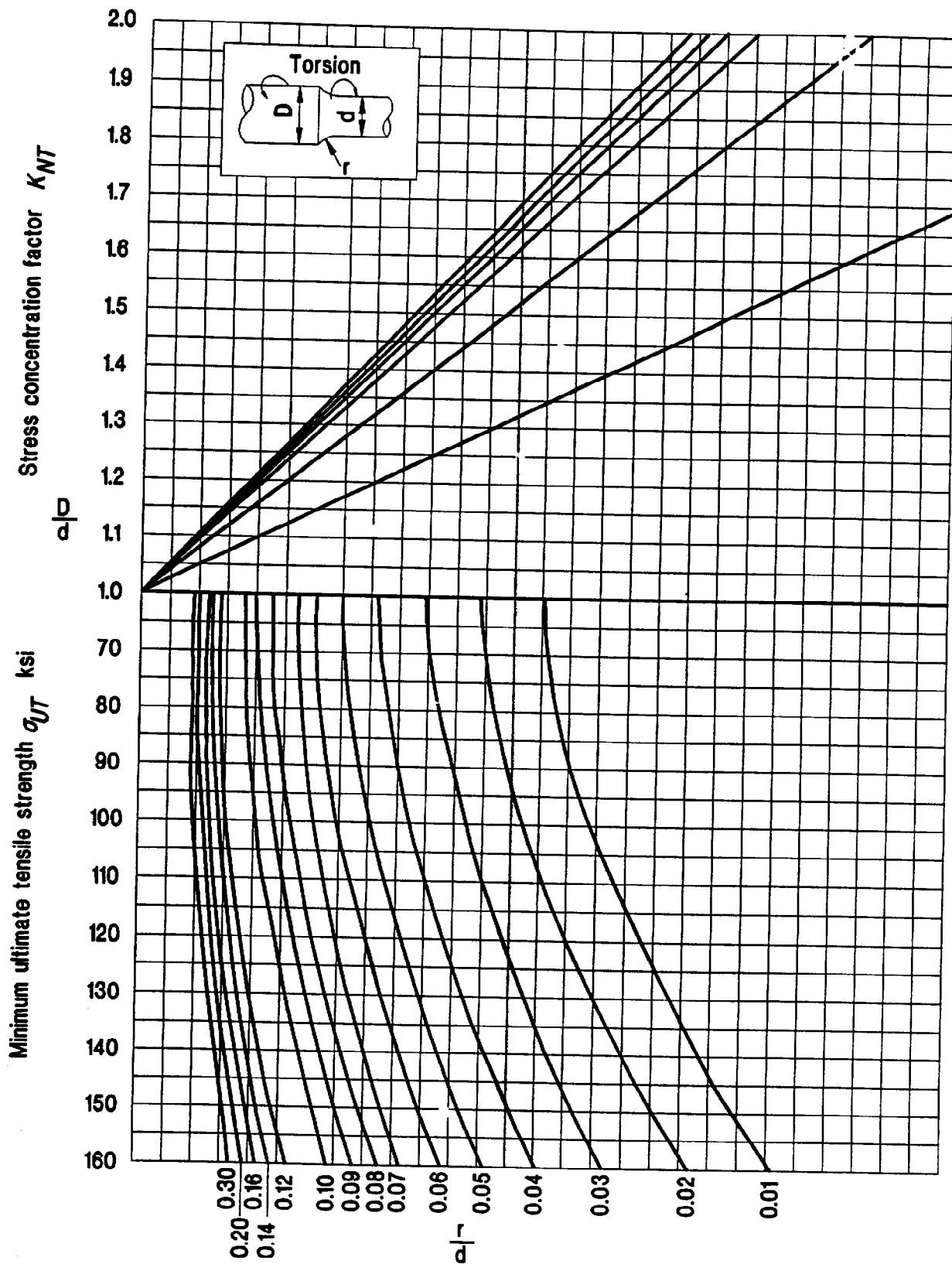
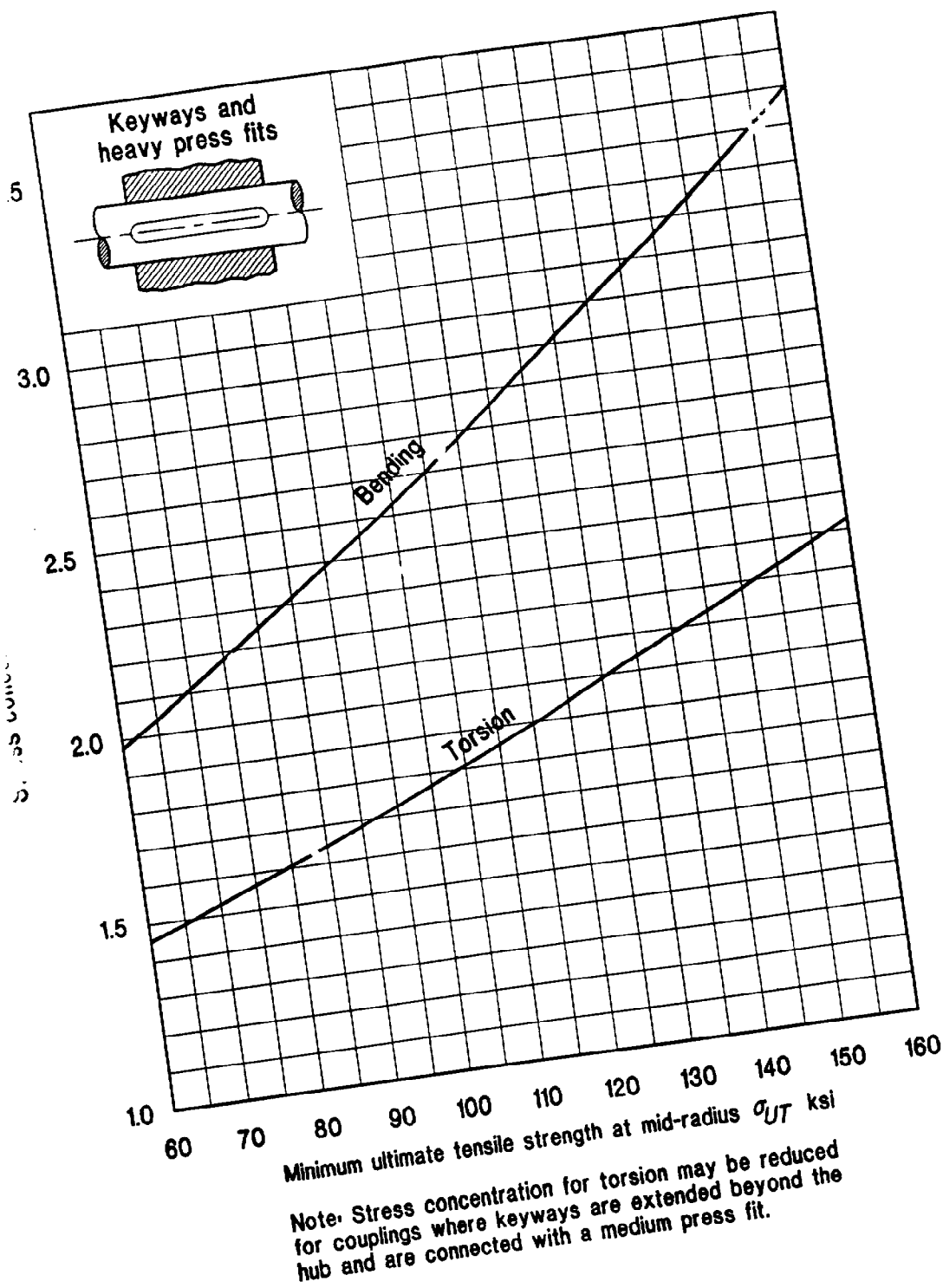


Fig. 22 — Stepped shaft in torsion-stress concentration factors



Note: Stress concentration for torsion may be reduced for couplings where keys are extended beyond the hub and are connected with a medium press fit.

Fig. 23 — Shaft: Stress concentration factors for keyways and heavy press fits—bending and torsion

3.2 Hooks.

3.2.1 General. Hooks shall be designed for infinite life based on the rated load except where the owner specifies finite life design. The design shall be established by analysis or testing.

3.2.2 Hooks shall be forged from fine grain material. Any welding on the hook shall be with the approval of a qualified welding engineer and performed prior to initial heat treatment. The capacity of the hook may be stamped on the hook nose. The hook shall not be painted.

3.2.3 Hook Shank. The calculated maximum stress at the root of the thread of the shank section, including a fatigue stress concentration factor for the type of thread used, shall not exceed $0.33 \sigma_{UT}$.

3.2.3.1 Due consideration shall be given to impact, service and to the possibility of bending forces on the hook shank. These bending forces will be partially dependent upon the geometry of the hook saddle and the coefficient of friction between the hook saddle and the loading element.

The shank shall be undercut below the last threads for a length of at least two pitches to allow for a uniform stress flow. The undercut shall have a radius at each change in diameter.

3.2.4 Hook Body. Hook bodies shall be of standard design where the line of the resultant load on the hook passes through the center of curvature of the inside edge of the hook and coincides with the centerline of the shank.

The maximum combined stress at the inner surface of curvature of the critical section 90 degrees from the vertical load shall not exceed $0.33 \sigma_{UT}$. This applies to hook bodies of trapezoidal section. Where square or rectangular sections are used, these stresses shall be reduced by at least 10%.

A stress analysis procedure for trapezoidal hook sections is provided in the commentary, together with a typical derivation of allowable stresses.

3.2.5 Testing. Where hook capacity has been established by testing, the static load required to straighten out the hook body shall not be less than 5.0 times the rated load.

A certificate of compliance showing both fatigue and static load testing covering both the configuration of the hook body and the hook shank must be provided.

Approval by the owner must be obtained for hooks selected on this basis.

3.2.6 Proportions of hook sections other than the critical section shall be such that the stress does not exceed the stress in the critical sections.

3.2.7 The hook nut and shank threads shall provide adequate strength for the hook capacity. Due consideration shall be given to the weakening effect of the nut locking arrangement.

3.2.8 Hook latches and swivel lock plates shall be provided when specified.

3.2.9 References.

ANSI/ASME B30.10 1982 — Hooks. Safety standard for cableways, cranes, derricks, hoists, hooks, jacks and slings.

AISE Standard No. 4 on Alloy Steel Chains and Alloy Steel Chain Slings for Overhead Lifting

AISE Technical Report No. 7 on Ladle Hooks

3.3 Drums.

Drums shall be rolled or centrifugally cast steel or as specified on the OIS. Flanged ends, if required, shall not be less than 1 in. in thickness and project not less than 2 1/2 in. beyond the pitch diameter of the drum.

Drums shall have wrapped grooves of a depth equal to 1/16 of the diameter of the hoisting rope and a pitch of not less than 1.2 times this diameter. The groove radius shall be 1/32 in. larger than the radius of the rope. Drums shall be designed so that not less than two complete wraps of hoisting rope will remain in the grooves ahead of the first rope clamp when the hook is at the lowest position. In addition, it shall be possible to lay the hook block on the floor for maintenance with one full wrap remaining on the drum.

One empty groove for each rope shall be left on the hoist drum when the hook is in the highest position. This provision is to insure that overlapping of the rope will not occur when the hook is in the highest drifted position.

If provisions for regrooving are to be made, it should be stated on the OIS.

The pitch diameter of the drum for 6 × 19 wire rope shall not be less than 30 times the diameter of the hoisting rope used. The pitch diameter of the drum for 6 × 37 wire rope shall not be less than 24 times the diameter of the hoisting rope used for Classes I and II cranes, and shall not be less than 30 times the diameter of the hoisting rope used for Classes III and IV cranes.

The drum gear shall be provided with a single key and shall be pressed on to the periphery of the hub or shell of the drum, or shall be bolted with fitted bolts to a flange on the drum or by other attachment means as approved by the owner.

3.4 Ropes.

The hoisting ropes shall be of the grade and type specified on the OIS. Based on the static breaking strength, a design factor of 8 shall be used for hot metal handling hoists and 5 for hoists other than hot metal handling.

Where main conductors are located below the runway rail, a guard shall be provided on the crane to prevent the hoist ropes, the lower sheave block or both from coming in contact with conductors.

The sheave arrangement should be reeved so as to eliminate reverse bends except at the drum.

The maximum allowable fleet angle for frequent working positions shall be $3\frac{1}{2}$ degrees for Classes I and II cranes, and $2\frac{1}{2}$ degrees for Classes III and IV cranes. The maximum allowable fleet angle for seldom reached positions shall be $4\frac{1}{2}$ degrees for Classes I and II cranes and $3\frac{1}{2}$ degrees for Classes III and IV cranes.

When special reeving, such as a stabilized reeving arrangement is used, consideration must be given to geometry and dynamics to maintain the appropriate safety factors.

Where high lifts occur, (100 ft or over), provisions should be made to prevent the twisting of the hook block.

Where load swinging can occur due to the crane service, rope lead angles should be set, or other provisions made, to minimize or eliminate the possibility of the rope skipping grooves on the hoist drums. When designing hoist drums the following should be taken into consideration. On high duty cycle cranes, drum grooves should be flame hardened to a minimum of 400 BHN.

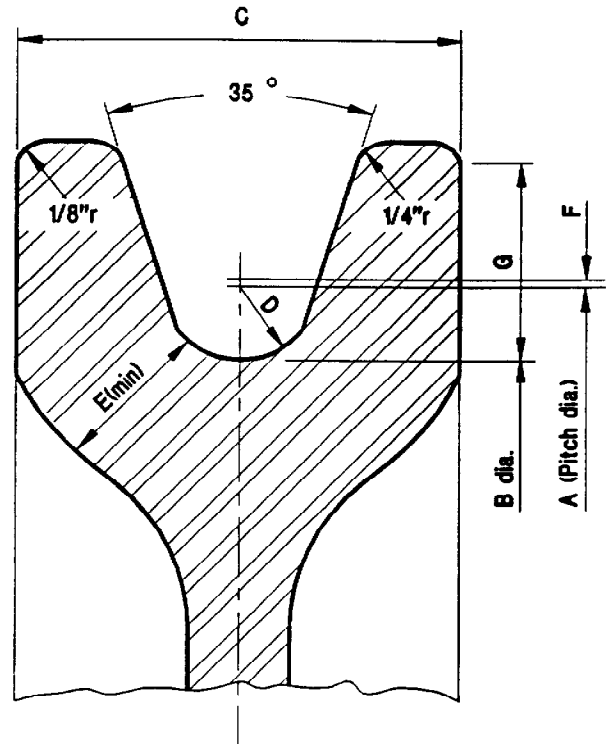


Fig. 24 — Sheave wheel contours

3.5 Sheaves and Hook Blocks.

Running sheaves shall be provided with antifriction bearings. Provision to take care of thrust shall be made. The sheaves shall be used in standard sizes in accordance with the tables below.

Sheave Wheel Contours — 24:1 Sheave-to-Rope Ratio

Rope Dia	A	B	C	D	E	F	G
1/2	12	11 1/2	1 3/4	9/32	1/2	1/32	3/4
5/8	15	14 3/8	2	11/32	5/8	1/32	15/16
3/4	18	17 1/4	2 1/4	13/32	1/4	1/32	1 1/8
7/8	21	20 1/8	2 1/2	31/64	7/8	3/64	1 5/16
1	24	23	2 3/4	35/64	1	3/64	1 1/2
1 1/8	27	25 7/8	3	39/64	1 1/8	3/64	1 11/16
1 1/4	30	28 3/4	3 1/4	11/16	1 1/4	1/16	1 7/8
1 3/8	33	31 5/8	3 1/2	3/4	1 7/8	1/16	2 1/16
1 1/2	36	34 1/2	3 3/4	11/16	1 1/2	1/16	2 1/4

Sheave Wheel Contours — 30:1 Sheave-to-Rope Ratio

Rope Dia	A	B	C	D	E	F	G
1/2	15	14 1/2	1 3/4	9/32	1/2	1/32	3/4
5/8	18 3/4	18 1/8	2	11/32	5/8	1/32	15/16
3/4	22 1/2	21 3/4	2 1/4	13/32	3/4	1/32	1 1/8
7/8	26 1/4	25 3/8	2 1/2	31/64	7/8	3/64	1 5/16
1	30	29	2 3/4	35/64	1	3/64	1 1/2
1 1/8	33 3/4	32 5/8	3	39/64	1 1/8	3/64	1 11/16
1 1/4	37 1/2	36 1/4	3 1/4	11/16	1 1/4	1/16	1 7/8
1 3/8	41 1/4	39 7/8	3 1/2	3/4	1 3/8	1/16	2 1/16
1 1/2	45	43 1/2	3 3/4	13/16	1 1/2	1/16	2 1/4

The pitch diameter of all sheaves, except equalizer sheaves, for 6×19 wire rope shall not be less than 30 times the diameter of the hoisting rope used. The pitch diameter of all sheaves, except equalizer sheaves, for 6×37 wire rope shall not be less than 24 times the diameter of the hoisting rope used for Classes I and II cranes and shall not be less than 30 times the diameter of the hoisting rope used for Classes III and IV cranes. Use the next larger size diameter for lead sheaves. Sheaves shall be enclosed by guards which fit close to the flanges to prevent the ropes from coming out of the grooves.

Sheaves and lower sheave blocks shall be constructed of steel and be entirely enclosed except for the rope openings. The hook shall be free to swivel and shall rotate on an antifriction bearing constructed so as to exclude dirt. The antifriction bearing shall be provided with a means for lubrication.

The bearing assembly in each sheave shall be individually lubricated. The fittings and grease lines shall be located so that they will be protected from damage.

Where possible, upper sheave block mountings shall be above the trolley deck. The upper sheave block should be removable as a unit, from above.

3.6 Equalizer Bars or Sheaves.

Where required, either an equalizer bar or sheave will be acceptable. In either case the bar or sheave shall be positioned to be accessible from the floor of the trolley and made in such manner that it can turn or swivel to align itself with the pull of the ropes. Equalizer sheaves shall have a pitch diameter not less than 18 times the diameter of the rope.

Cranes having hoists which handle hot metal or critical loads should utilize equalizer bars to provide two independent rope systems, not equalizer sheaves.

For increased rope life, consideration should be given to using equalizer sheaves with the same diameter as the running sheave.

3.7 Track Wheels and Rails.

Installation of crane bridge wheel flange/rail lubricators are essential for long wheel and rail service life.

To facilitate the checking of bridge wheel alignment, provision for machine registers on bridge end trucks, machined true to the wheel mounting seats, should be considered.

3.7.1 Track Wheels. All track wheels shall be double flanged. The blanks shall be made by roll forming, forging or casting from grades of steel appropriate to the forming process.

Bridge track wheels shall have either straight or tapered treads which shall not be less than $1 \frac{7}{16}$ in. wider than the rail head as shown in Table 7 for the different rail sections. Unless otherwise specified on the OIS, straight tread wheels shall be furnished. Tapered treads should not be used on 171 lb/yd rail.

Rail		Wheel	Clearances
Weight, lb/yd	Head Width, in.	Tread Width, E, in.	in.
104	2 $\frac{1}{2}$	4	1 $\frac{1}{2}$
105	2 $\frac{9}{16}$	4	1 $\frac{7}{16}$
135	3	4 $\frac{5}{8}$	1 $\frac{5}{8}$
175	4 $\frac{1}{32}$	5 $\frac{1}{2}$	1 $\frac{15}{32}$
171	4	5 $\frac{1}{2}$	1 $\frac{1}{2}$

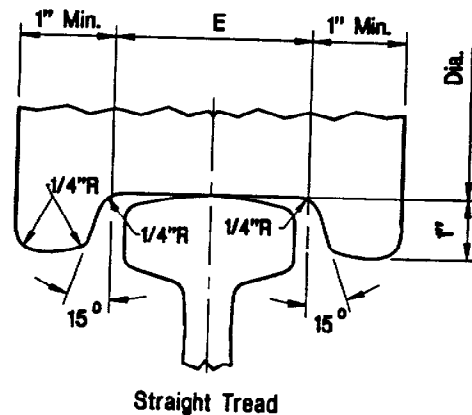


Fig. 25 Typical straight tread wheel/rail arrangement

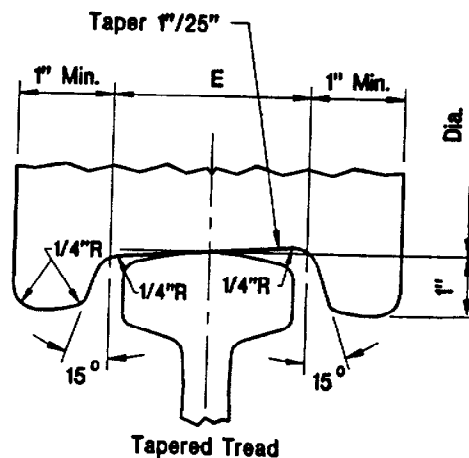


Fig. 26 Typical tapered tread wheel/rail arrangement

Table 8 — Trolley Track Wheel Clearances

Rail		Wheel	Clearances
Weight lb/yd	Head Width, in.	Tread Width, E, in.	in.
60	2 3/8	3	5/8
104	2 1/2	3	1/2
105	2 9/16	3	7/16
135	3	3 9/16	9/16
175	4 1/32	4 5/8	19/32
171	4	4 1/2	1/2

Trolley track wheels shall be as specified on the OIS and shall have straight treads which shall be 7/16 to 5/8 in. wider than the rail head as shown in Table 8 for the different rail sections.

Straight tread bridge and trolley driver wheels shall have matched tread diameters within 0.005 in. maximum of each other when drive wheels are mechanically connected. All

other applications including tapered tread wheels shall have bridge and trolley driver wheels matched within 0.030 in. maximum of each other.

Bridge wheel loads shall be determined with the maximum lifted load on the trolley, which shall be positioned at the closest working approach that produces the maximum wheel load.

Trolley wheel loads shall be determined with the maximum lifted loads.

$$(W_t + W_L + W_A) \quad (\text{Eq 46})$$

The recommended maximum trolley and bridge wheel loads for wheel-to-rail combinations shall not exceed the values given in Table 9 for rim-toughened wheels and Table 10 for case-hardened wheels, modified by the appropriate speed factor given in Table 11 and the service factor given in Table 12.

Recommended Maximum Wheel Load =

$$\left\{ \frac{\text{Allowable Wheel Load}}{\text{Speed Factor} \times \text{Service Factor}} \right\} \quad (\text{Eq 47})$$

**Table 9 — Allowable Wheel Loads Guide for Heat Treated (320 Bhn Minimum) Crane Wheels, lb,
for Speed Modification Factor = 1.0**

Wheel Dia., in.	Rail Section								> 286 BHN at Max Shear Depth, in.	>267 BHN at Max Shear Depth × 1.5 in.
	ASCE			Bethlehem and USS						
	30	40	60	104-105	135	175	171			
8	11840	13930						0.057	0.086	
9	13220	15670	21940					0.064	0.096	
10	14800	17410	24370					0.071	0.107	
12	17770	20890	29250	31340				0.086	0.129	
15	22210	26110	36560	39170				0.107	0.161	
	26650	31340	43880	47010	56410	78350	87760	0.128	0.192	
21		36560	51190	54850	65820	91410	102380	0.150	0.225	
24			58500	62680	75220	104470	117010	0.171	0.257	
27				70520	84620	117530	131640	0.193	0.290	
30				78350	94020	130590	146260	0.214	0.321	
36					112830	156710	175520	0.257	0.386	
	1.063	1.250	1.750	1.875	2.250	3.125	3.500			

Effective rail width, in.

Notes:

Wheel diameters and shear depths are both in inches.

The loads are based on 70% minimum contact of the effective rail width. If the expected rail contact is less than 70% then the above loads should be adjusted downward proportionally.

Since the 171 lb/yd rail is not crowned, special attention must be given to the wheel/rail alignment in order to justify the above loads. The 171 lb/yd rail load values are predicated on a method being provided to insure alignment between the wheel tread and the rail head surfaces (e.g. the use of an elastomeric pad).

The last two columns help insure that the hardness is deep enough to cover the shear stresses that vary with depth down from the surface. At the tabulated depths, the actual hardness must be greater than the hardness shown in the column headings. For example, a 24-in. diameter wheel on the 175 lb/yd rail loaded to 104,470 lbs must have a surface hardness of 320 BHN minimum, a hardness of 286 BHN minimum at a depth of 0.171 in. from the surface and a hardness of 267 BHN minimum at a depth of 0.257 in. from the surface.

**Table 10 — Allowable Wheel Loads Guide for Case-Hardened (58Rc Minimum) Crane Wheels, lb,
for Speed Modification Factor = 1.0**

Wheel Dia, In.	Rail Section							>338 BHN at Max Shear Depth, In.	>315 BHN at Max Shear Depth, x 1.5 In.
	ASCE			Bethlehem and USS					
	30	40	60	104-105	135	175	171		
8	16580	19500						0.068	0.102
9	18650	21930	30710					0.076	0.114
10	20730	24370	34120					0.084	0.126
12	24870	29250	40950	43870				0.101	0.152
15	31090	36560	51190	54840				0.127	0.191
18	37310	43870	61430	65810	78980	109690	122860	0.152	0.228
21		51190	71660	76780	92140	127980	143330	0.177	0.266
24			81900	87750	105300	146260	163810	0.203	0.305
27				98720	118470	164540	184290	0.228	0.342
30				109690	131630	182830	204770	0.253	0.380
36					157960	219390	245720	0.304	0.456
	1.063	1.250	1.750	1.875	2.250	3.125	3.500		
	Effective rail width, in.								

Notes:

Wheel diameters and shear depths are both in inches.

The loads are based on 70% minimum contact of the effective rail width. If the expected rail contact is less than 70% then the above loads should be adjusted downward proportionally.

Since the 171 lb/yd rail is not crowned, special attention must be given to the wheel/rail alignment in order to justify the above loads. The 171 lb/yd rail load values are predicated on a method being provided to insure alignment between the wheel tread and the rail head surfaces (e.g. the use of an elastomeric pad)

The last two columns help insure that the hardness is deep enough to cover the shear stresses that vary with depth down from the surface. At the tabulated depths, the actual hardness must be greater than the hardness shown in the column headings. For example, a 24-in. diameter wheel on the 175 lb/yd rail loaded to 146,260 lbs must have a surface hardness of 58 Rc minimum at the surface and a hardness of 338 BHN minimum at a depth of 0.203 in. from the surface and a hardness of 315 BHN minimum at a depth of 0.305 in. from the surface.

The above loads are based on the wheels running on heat-treated rail (320 BHN minimum). If the wheels are running on untreated rail, the above loads may cause decreased rail life.

This table should not be used for cases with crowned rail harder than 400 BHN because the wheel or rail may spall before 70% rail contact is obtained through rail crown flattening.

Table 11 — Speed Modification Factor for Determining Recommended Maximum Wheel Load

Wheel Dia, In.	Travel Speed, fpm												
	50	75	100	125	150	175	200	250	300	350	400	450	500
8	0.958	1.013	1.049	1.085	1.122	1.158	1.195	1.267	1.340	1.412	1.485	1.558	1.630
9	0.945	1.00	1.033	1.066	1.098	1.130	1.162	1.227	1.292	1.356	1.421	1.485	1.550
10	0.933	0.984	1.020	1.049	1.078	1.107	1.136	1.195	1.253	1.311	1.369	1.427	1.485
12	0.916	0.958	1.00	1.025	1.049	1.074	1.098	1.146	1.195	1.243	1.292	1.340	1.389
15	0.899	0.933	0.966	1.00	1.020	1.040	1.059	1.098	1.136	1.175	1.214	1.253	1.292
18	0.887	0.916	0.945	0.972	1.00	1.017	1.033	1.066	1.098	1.130	1.162	1.195	1.227
21	0.880	0.902	0.927	0.952	0.976	1.00	1.015	1.042	1.070	1.098	1.126	1.153	1.181
24	0.874	0.894	0.916	0.937	0.958	0.980	1.00	1.025	1.049	1.074	1.098	1.122	1.146
27	0.868	0.887	0.906	0.925	0.945	0.962	0.982	1.018	1.033	1.054	1.076	1.098	1.119
30	0.864	0.881	0.899	0.916	0.933	0.950	0.966	1.00	1.020	1.040	1.059	1.078	1.098
36	0.859	0.874	0.887	0.900	0.916	0.929	0.945	0.972	1.00	1.017	1.033	1.049	1.066

For wheel rpm ≤ 31.5

Speed factor = $\left[1 + \left(\frac{\text{rpm} - 31.5}{360} \right) \right]^2$ (Eq 48)

For rpm > 31.5

Speed factor = $1 + \left(\frac{\text{rpm} - 31.5}{328.5} \right)$ (Eq 49)

Table 12 — Proposed Service Factor for Determining Recommended Maximum Wheel Loads

Types of Crane Loading	Load Cycles			
	< 100,000	100,000 to 500,00	500,00 to 2 million	over 2 million
Occasional lifts at rated capacity, normal lifts are very light loads	0.75	0.80	0.85	1.00
Lifts are 1/3 light loads, 1/3 medium loads and 1/3 at rated capacity	0.80	0.85	0.90	1.00
Lifts are regularly at rated capacity	0.85	0.90	0.95	1.00

3.7.2 Rails. Joints on trolley travel rails shall be welded or made by using standard joint bars. There shall be no bolt holes adjacent to the welded joint. Where joint bars are used, the joined ends of the rails shall be laid without openings between the ends.

Provision shall be made to prevent creeping of rails on girders by means of shear lugs welded to the cover or wear plate at each end of the rail, with sufficient clearance to allow thermal movement.

For conventional box girders, rails shall be fastened in place by suitable clamps, held either by direct welding to the cover or wear plate or with studs welded to the cover plate. Rails may also be held by bolts having heads in slotted clamps welded to the cover or wear plate. Rails shall be fastened in place by steel clamps held by throughbolts for single web girders unless otherwise specified on the OIS. Clamps shall be spaced at not more than 36-in. centers.

Heat treated rails may be used for increased rail life.

3.8 Bumpers.

Provisions in the design of the runway and the design of the runway stops shall consider the energy absorbing or storage device used in the crane bumper. The device may be nonlinear (e.g. hydraulic bumpers) or a linear device such as a coil spring.

The maximum deceleration rate for both bridge and trolley shall not exceed 16 fps^2 at 50% of the full load rated speed (full load rated speed shall be used unless adequate information is supplied by owner to determine the actual attainable maximum speed). Additionally, bumpers shall be capable of absorbing the total energy at 100% full load rated speed. See the sample problem calculations for hydraulic and spring bumpers.

Between cranes or trolleys (if two trolleys are located on one bridge) bumpers shall be capable of absorbing the energy from 70% of full load rated speed of both cranes or trolleys traveling in opposite directions, or the energy from 100% of full load rated speed of either crane or trolley, whichever is the greatest.

CRANE BUMPER END FORCE EXAMPLES

Bridge weight, W_B — 200 kips

Bridge full load rated speed, V_B — 360 ft/min (6 ft/second)

Trolley weight, W_T — 40 kips

Trolley speed, V_T — 180 ft/min (3 ft/second)

Impact weight per side,

$$W_E = (0.5 \times W_B) + (0.9^{\dagger} \times W_T) = 136 \text{ kips} \quad (\text{Eq 50})$$

Kinetic energy to be absorbed at 100% full load rated speed,

$$K_E = \frac{W_E \times V_B^2}{2g} = 76.03 \text{ kip-ft} \quad (\text{Eq 51})$$

Maximum allowable end force to decelerate the crane at 16 ft/sec^2 ,

$$F_A = W_E \times \frac{16}{32} = 68 \text{ kips} \quad (\text{Eq 52})$$

Kinetic energy to be absorbed at 50% full load rated speed,

$$K_H = \frac{W_E \times \left(\frac{V_b}{2}\right)^2}{2g} = 19 \text{ kip-ft} \quad (\text{Eq 53})$$

Bumper Selection:

1. Kinetic energy absorption or storage capacity
76.03 kip-ft

2. Bumper stroke required:

$$K_H = \frac{F_A \times S \times \eta}{12} \quad (\text{Eq 54})$$

Where:

S = Bumper stroke, in.

η = Bumper efficiency

(a) Hydraulic bumper ($\eta = 0.8$ for this example)

$$S = \frac{12 \times K_H}{F_A \times \eta} = 4.19 \text{ in.} \quad (\text{Eq 55})$$

(b) Spring bumper ($\eta = 0.5$ for helical coil springs by definition)

$$S = 6.71 \text{ in.}$$

Note: Values of η vary for hydraulic bumpers from manufacturer to manufacturer. Bumper efficiency is defined, for a given set of conditions, as:

$$\frac{\text{theoretical minimum end force}}{\text{actual end force}} \quad (\text{Eq 56})$$

$\dagger$ 0.9 value represents a convenient proportion of the maximum approach of trolley to one side and can vary with design of crane.

The design of all bumpers shall include safety cables to prevent parts from dropping to the floor.

The height of bumpers above the top of the rail shall be as specified on the OIS or as determined by the crane builder.

For computing bridge bumper energy, the trolley shall be placed in the end approach which will produce the maximum end reaction from both bridge and trolley. This end reaction shall be used as the maximum weight portion of the crane that can act on each bridge bumper. The energy absorbing capacity of the bumper shall be based on power-off and shall not include the lifted load if free to swing. Bridge bumpers shall have a contact surface of not less than 5 in. in diameter, be located on the rail centerline and mounted to provide proper clearance when bumpers of two cranes come together and both are fully compressed. Where practical, they shall be mounted to provide for easy removal of bridge track wheels.

The building and end stops shall be designed to withstand those forces of the fully loaded crane at 100% rated speed (power off). The recommended increase in allowable stresses for this case is 50%.

It should be noted that these forces may be reduced by increasing bumper stroke. In the example, increasing the bumper stroke(s) from 4.19 in. to 10 in. reduces end force (F_A) from 68 kips to 28.5 kips.

3.9 Bridge and Trolley Drives.

3.9.1 Bridge and Trolley Drive Arrangements.

Bridge and trolley drives may consist of one of the following arrangements, as specified on the OIS and as shown in Fig. 27. These arrangements cover most types of crane drives regardless of the number of wheels. On four-wheel cranes, half-flexible couplings may be substituted for floating shafts if so specified on the OIS. Other types of drives may be used if approved by the owner.

3.9.1.1 A-1 Drive. The motor is located near the center of the bridge and connected by means of a flexible coupling to a self-contained gear reduction unit also located near the center of the bridge, which shall be connected to the line shaft having solid couplings. The line shaft is connected to the bridge track wheel axles by means of floating shafts with half-flexible couplings.

3.9.1.2 A-2 Drive. The motor is connected by means of a flexible coupling to a self-contained gear reduction unit located near the center of the bridge. The track wheels shall be driven through gears pressed and keyed on their axles and pinions mounted on the end sections of the line shaft. The end sections of the line shaft shall be connected by means of floating shafts with half-flexible couplings. All other couplings shall be of the solid type.

3.9.1.3 A-3 Drive. The motor is located at the center of the bridge and is connected directly to the line shaft. Self-contained gear reduction units located near each end of the bridge shall be connected to wheel axles by means of floating shafts with half-flexible couplings. All other couplings shall be of the solid type unless the high speed shafts of the gear reduction units have no end play due to the type of bearing used. In such cases, they will be connected to the line shaft by means of half-flexible couplings.

3.9.1.4 A-4 Drive. The motors are located near each end of the bridge without torque shafts. The motors shall be connected to self-contained gear reduction units by means of flexible couplings. The gear reduction units shall be connected to the track wheel axles by means of floating shafts with half-flexible couplings.

3.9.1.5 A-5 Drive. The motor is located near the center of the bridge and is connected by means of a flexible coupling to a self-contained gear reduction unit located near the center of the bridge. This reduction unit shall be connected by sections of line shaft having solid couplings to self-contained gear reduction units located near each end of the crane. These reduction units are connected to bridge track wheel axles by means of floating shafts with half-flexible couplings.

3.9.1.6 A-6 Drive. The motors are located near each end of bridge and connected with a torque shaft. On the drive end the motors shall be connected to self-contained gear reduction units by means of flexible couplings. Gear reduction units are to be connected to track wheel axles by means of floating shafts with half-flexible couplings. All other couplings shall be of the solid type.

3.9.2 Bridge and Trolley Drive Design.

3.9.2.1 Torsional Deflection and Vibration. A-1, A-2 and A-5 drives can result in a torsionally very soft drive system if center gear ratios, bridge spans or both are of large magnitude. Natural frequency and amplitude of total torsional deflection of the drive system should be determined. Low frequencies and large total torsional deflections are undesirable for crane operation.

3.9.2.2 Line Shafting and Couplings. Floating shaft — Wherever possible, the flexible halves of half-flexible couplings shall be mounted on the floating shaft.

Line shaft couplings other than the flexible type are to be made from rolled or forged steel. Couplings shall be located close to the bearings and be provided with substantial removable guards which shall extend beyond the ends of the hubs and overlap with the coupling hub OD. Where half-flexible couplings are used, the couplings shall be located close to the bearing on the end truck and the adjacent line shaft bearing shall not be closer than 4 ft 6 in. The flexible coupling

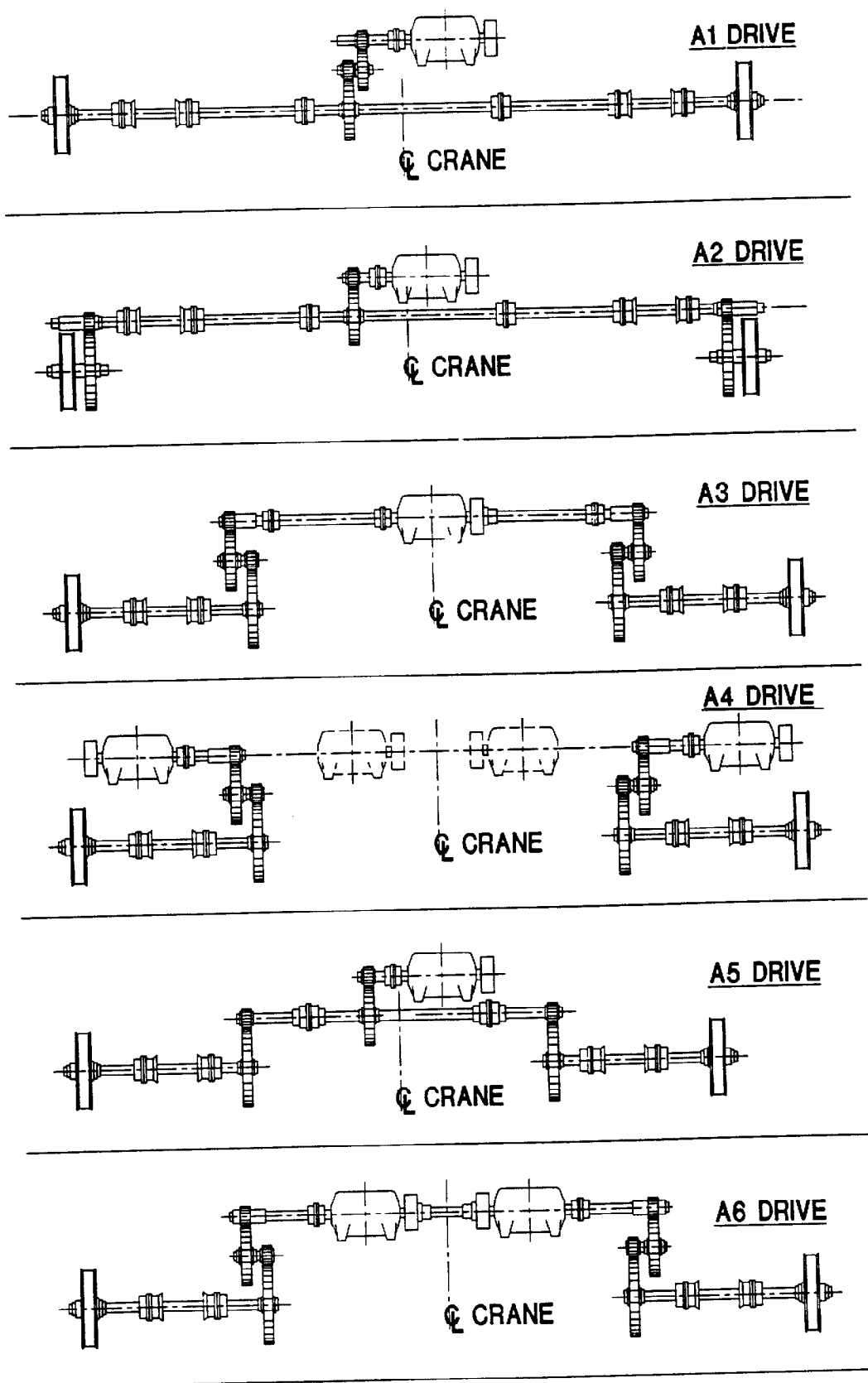


Fig. 27 — Arrangements of crane bridge drives

manufacturer's standards for solid half-couplings shall be used for solid couplings unless otherwise specified on the OIS.

The load shall be transmitted between coupling halves by means of fitted bolts.

For shaft speed below 400 rpm, the following maximum bearing spacing shall be permitted:

- (1) 12 ft for 3 in. diameter
- (2) 14 ft for 3½ in. diameter
- (3) 15 ft for 4 in. diameter
- (4) 16 ft for 4½ in. diameter

For shaft speed in excess of 400 rpm, the above spacing shall be reduced as necessary to avoid harmonic vibrations.

Supports for motor and gear reduction units shall be welded structural steel, rigidly connected to the crane girder (see Section 2).

Bolts for fastening bearing brackets, motors and gear reduction units shall be accessible from above the footwalk.

Angular deflection of the line shaft at torque corresponding to 2 times full load motor torque (60-minute rating), shall not exceed 0.09 degrees/ft of shaft length. (In computing deflections when the gear reduction unit is located at the center of the bridge span, one-half of the torque is to be applied to each half of the line shaft). If the gear reduction unit is not located at the center of the torsional shear, in-line shafting must be proportioned in relation to the shaft length of each section. Limit of length or length-to-diameter ratio shall be defined for application of deflection as the critical design criteria.

3.9.2.3 Motor Selection. Bridge and trolley speed, gear ratios and bridge drive motor sizes shall be calculated according to methods set forth in Section 4 of this Report.

For A-4 drives wheel slippage and minimum operating wheel load (0.20 friction factor) should be considered.

3.10 Shafting.

Design torque for all travel drives shall be based on twice the 60-minute motor rating or wheel slippage at maximum wheel load (0.20 friction factor), whichever is less. Hoist shafting design torque shall be based on 100% of maximum lifted load.

Axles or shafts which are provided with sleeve bearings are to be surface or case-hardened and ground.

Radii for keyseat shall be according to Section 3.14.3.3.

3.11 Press Fits and Keys.

Keys shall be provided for all connections subject to torsion unless otherwise specified on the OIS. Key sizes shall be in accordance with Section 3.14.3.3. All gears, pinions and couplings shall be pressed or shrunk onto shafts in addition to

being keyed unless specified on the OIS. All press fits shall be made in accordance with ANSI B4.1, Preferred Limits and Fits for Cylindrical Parts.

All keys and keyways shall be radiused and/or chamfered according to ANSI B17.1, latest edition.

3.12 Bearings.

Antifriction bearings shall be spherical, tapered, straight or a combination thereof as specified on the OIS.

Antifriction bearings shall be selected on the basis of B-10 life, to give a minimum life expectancy of ten years or 5,000 to 40,000 hr under the service conditions for which the crane is intended. Bearing selection in this specification is based on the total number of cycles which it is expected the bearing will undergo during the number of hours service the crane will be used in a 10-year period. Where other data is not available, the number of hours for the various motions can be estimated from Table 13. The required hours of service/year are given for the various motions concerned (bridge, trolley or hoist) in this Table and may be used for determining total service hours if not otherwise specified.

All bearings selected must meet the required life at 75% of the maximum bearing load (at rated speed) based on the published catalog rating of the bearing manufacturer. Bearings are selected for 75% of the maximum load (at rated speed) on the assumption that this gives a practical average value for fatigue life purposes. If the load on the bearing is essentially constant, the bearings must meet a required life of 100% of the maximum load at rated speed. In some cases axle sizes establish bearing sizes.

With wheel bearings of the antifriction type, one bearing on each wheel axle shall be of the fixed type. The other bearing shall be arranged to allow for expansion or float of the axle. Other arrangements shall be as specified on the OIS.

Where sleeve bearings are applied to track wheel axles, the bearing pressure shall not exceed 750 psi on projected area, except where aluminum-bronze bearings are used, in which case the bearing pressure shall not exceed 1000 psi. Bearings and housings are to be designed to exclude dirt, prevent leakage of oil or grease and eliminate the necessity for frequent oiling or replacement of oil. The bearing design must meet the approval of the owner.

Antifriction line shaft bearings shall have inner races and self-alignment should be provided at each bearing.

Gear housings shall be split or designed to permit easy removal of the shaft.

Gear reduction units should be designed so that gears, shafts and bearings, as well as bearing cartridges and end pieces, can be preassembled as a spare.

Drum bearings and supports for the upper sheave block shall be located so as to equalize the load on track wheels as near as possible.

3.13 Bearing Brackets and Housing.

Bearing brackets, if not integral with the frame, shall be mounted on a machined surface and be kept in alignment by fitted bolts or other equally effective methods.

When shafting is geared together the support structure for all bearing cartridges should, where practical, be integral and located as close as possible to the gears and pinions.

Heavy caps shall be provided with a means for lifting.

3.14 Gearing.

3.14.1 Gearing Types. Gearing shall be spur, herringbone, helical, bevel, or worm as specified on the OIS. No split gears or overhung gears shall be used without specific approval of the owner.

3.14.2 Gearing Design. Horsepower ratings for all spur and helical involute gearing shall be based upon American Gear Manufacturers Association (AGMA) Standard 218.01, latest edition, for Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth.

The pitting resistance power rating is:

$$P_{ac} = \frac{n_p F}{126,000} \frac{I C_v}{C_s C_m C_f C_a} \left(\frac{d S_{ac}}{C_p} \frac{C_L C_H}{C_T C_R} \right)^2 \frac{1}{n_c} \quad (\text{Eq 57})$$

The bending strength power rating is:

$$P_{at} = \frac{n_p d K_v}{126,000 K_a} \frac{F}{P_d} \frac{J}{K_s K_m} \frac{S_{at} K_L}{K_R K_T} \frac{1}{n_t} \quad (\text{Eq 58})$$

The life factor for pitting resistance C_L and the life factor for bending strength K_L may be utilized for life rating purposes. Gearing design shall be based on the total number of cycles which the gear will experience during the number of hours service for the particular crane motion during a 15-year period. Where such data is not available, the number of hours for the various motions can be estimated from Table 13. The required hours of service/year are given for bridge, trolley or hoist motions in this table and may be used for determining total service hours if not otherwise specified.

3.14.2.1 Safety Factors. The safety factors n_c and n_t may be used to provide an additional margin of safety.

Applications involving unusual or severe loading conditions or requiring a high degree of dependability due to the importance of the load handled or to the risk to human life will require special consideration to establish an appropriate safety factor.

3.14.2.2 Typical Service Hours

Table 13 — Service Hours for 20-Year Life				
	AISE Crane Class			
	1	2	3	4
Main Hoist	4,000	11,000	30,000	80,000
Auxiliary Hoist	4,000	9,000	21,000	49,000
Trolley Drive	4,000	9,000	21,000	49,000
Bridge Drive	4,000	10,500	28,000	73,000

3.14.2.3 Gear Ratings. Hoist gear ratings for bending strength and pitting resistance shall be based on the torque required to lift the rated load plus hook block and/or lifting beam and shall take account of mechanical efficiencies listed in Table 20.

Travel drive gear ratings for bending strength shall be based on the 60-minute motor rating, and pitting resistance shall be based on the average horsepower transmitted. Due consideration shall be given to the maximum brake torque which can be applied to the drive.

3.14.2.4 Dynamic Response. Where unusual drive arrangements are employed the dynamic response of the system should be analyzed to insure that any additional loadings are identified.

3.14.2.5 Drum Gear Alignment. The effects of trolley frame and rope drum deflections on the alignment of the hoist drum gear and pinion shall be considered.

3.14.3 Machining Specifications.

3.14.3.1 Machining Standards. All machining on gears shall be done in accordance with AGMA Standards and Recommended Practices for the Manufacturing of Industrial Gearing. Quality shall be as specified in the OIS. Minimum quality shall be Q6. Tolerances to which finished gears and pinions must conform shall be Runout, Pitch, Profile, Lead, Tooth to Tooth Composite and Total Composite, as defined in Vol. 1 AGMA 390.03 March 1980, Section 8. All gears shall have a full root radius unless this compromises other design considerations. The wall thickness over the keyway shall be at least equal to the tooth depth. Gears shall not have a shoulder left in the fillet area.

3.14.3.2 Bores. Bores for gears requiring heat treatment shall be finish-machined or ground to size after heat treatment and shall be no harder than 269 BHN for Classes G-3 and G-4; 345 BHN for Class G-1; and 300 BHN for Class G-2.

3.14.3.3 Keyway Tolerances. Keyway tolerances to be in accordance with ANSI B17.1-Class 2, latest edition.

3.14.4 Metallurgical Specifications.

3.14.4.1 Effective Case Depth. The effective case depth for carburized and induction hardened gears is defined as the depth below the surface at which the Rockwell "C" hardness has dropped to HRC 50 or to 5 points below the surface hardness, whichever is the lower hardness.

Any hardness specified on one scale can be measured on another scale by using ASTM Standard Conversion Tables.

3.14.4.2 Classifications.

Class G-1 — Through Hardened Gears. AISI 4140 or 4340 steel. Treatment — heat treat to a minimum of 280 BHN at the pitch line. Pinions shall be approximately 50 points harder than gears. For close tolerances and/or coarse pitches, rough machine then finish after heat treatment. This class of gear is appropriate for high impact, low wear applications.

Class G-2 — Induction Hardened Gears. AISI 1040/1050, 4140 or 4340 steel. Treatment — Normalize and temper or quench and temper to 243/300BHN. The gear teeth shall then be induction hardened on the tooth profile and root surfaces to 48-53 Rockwell "C" except $\frac{1}{4}$ in. to $\frac{1}{2}$ in. on the ends of the teeth which must be at least 44 Rockwell "C." The minimum effective case depth at the pitch line shall be as

shown on the dotted lines in Fig. 28. Gears shall be tempered at 300°F minimum immediately following induction hardening. This class of gear has better wear characteristics than G-1 but less than G-3 or G-4.

Class G-3 — Carburized and Hardened Plain Low-Carbon Steel (0.15 to 0.25% carbon). Treatment — Gears shall be manufactured to have an effective case depth as shown in Fig. 28. Hardness must be 58 Rockwell "C" minimum at the surface of the teeth at the pitch line. Gears shall be tempered at 300°F minimum immediately after hardening. This class of gear is appropriate for high wear, low impact applications.

Class G-4 — Carburized and Hardened Low-Carbon Alloy Steels (0.15 to 0.25% carbon). Recognized AISI grades include 3300, 4100, 4300 and 8600 and 9300 series. Treatment — Gears shall be manufactured to have an effective case depth as shown in Fig. 28. Hardness must be 58 Rockwell "C" minimum at the surface of the teeth at the pitch line. Gears shall be tempered at 300°F minimum immediately after hardening. This class of gear is appropriate for high wear applications with some impact.

3.14.6 Identification. In the selection of gears and pinions for the crane (based on the service required), it should be noted that the OIS may specify the surface and core hardness

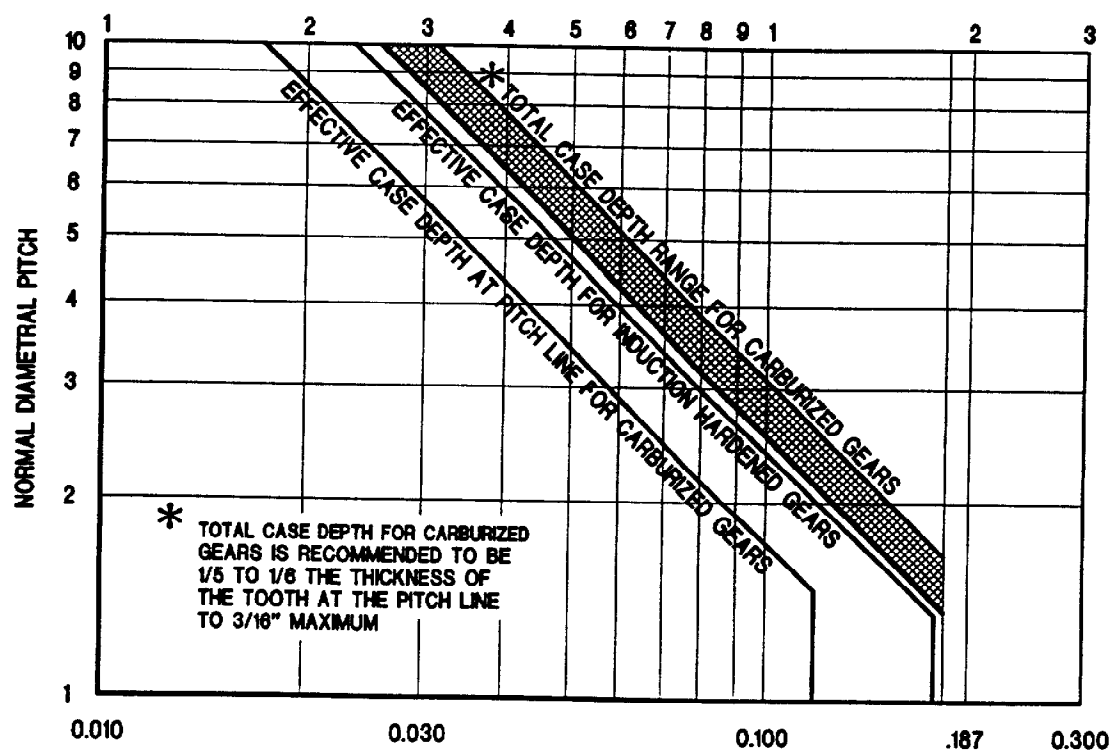


Fig. 28 — Depth of effective case at pitch line

Through bolts should be used to hold the gear cases together and to mount the gear box to its base.

All gear cases shall be mounted on machined surfaces. Shims shall not be used.

Gear box seats shall be of sufficient size to allow the installation of shear blocks to locate the gear box positively.

Gear cases shall be provided with lifting lugs that are suitable for use in lifting the whole gear case assembly without tilting.

3.16 Lubrication.

Provisions shall be made by the contractor for proper lubrication of all parts. The size and type of fittings shall be as selected by the crane manufacturer unless specified on the OIS.

All lines shall be located so as to provide the maximum natural protection, and at the same time lines should be positioned so that ordinary repairs can be made without complete removal of the lines. All lines shall be fastened to the crane's structure.

Centralized automatic grease systems should be considered on trolleys. Manual pumps should be considered for bridge drive assemblies. Flexible hoses should be utilized at component end for easy removal.

3.17 Bolts, Nuts and Welded Connections.

Where high strength bolts are used they shall be in accordance with ANSI Standard B18.2.1, ASTM A-325, ASTM A-490 or ANSI B18.2.2. The heads of bolts in inaccessible places shall be held from turning by recesses or projecting lugs.

For all structural work such as gusset plates and brackets supporting footwalk and bridge line shaft, see Section 2.

SYMBOLS — MECHANICAL

A	Effective cross-sectional area of critical section, sq in.	M_T	Torsional moment, kip-in.
b	Effective width of rail head, in.	m_i	Average length of motion units, ft
C_M	Load distribution factor for gear durability	N_e	Number of design endurance cycles
C_S	Size factor for gear durability	N_i	Total number of design load cycles of stress cycles per load level
C_{SF}	Service factor for gear durability	n_i	Total number of stress cycles per stress level
C_V	Dynamic factor for gear durability	n_l	Total number of lifts per load level during specified life of crane
D	Large diameter of a stepped shaft or round bar, in.	P	Load (weight, force or transverse shear load reaction), kips
D	Diameter of crane wheel, in.	P	Allowable wheel load, lb
d	Small diameter of a stepped shaft or round bar, in.	P_D	Diametral pitch
d	Pitch diameter of gear, in.	Q	Static moment about the neutral axis of the area of that portion of the component cross-section beyond the place where the shear stress is being calculated, in. ³
F	Gear face width, in.	R	Gear factor, $R = \frac{F - 6}{30}$
F_y	Specified minimum yield stress	R_B	Fluctuation ratio for bending, $R_B = \frac{\sigma_{B \text{ min}}}{\sigma_{B \text{ max}}}$
I	Geometry factor for gear durability	R_C	Stress ratio R_S to the K power, $R_N = \left[\frac{S_i}{S_i \text{ max}} \right]^K$
I	Moment of inertia, in. ⁴	R_N	Fluctuation ratio for tension-compression, $R_N = \frac{\sigma_{N \text{ min}}}{\sigma_{N \text{ max}}}$
J	Geometry factor for gear strength	R_n	Cycle ratio, $R_n = \frac{N_i}{\sum N_i}$
K	Wheel load factor	R_S	Fluctuation ratio for shear, $R_S = \frac{\tau_{S \text{ min}}}{\tau_{S \text{ max}}}$
K	Stress class reduction factor	R_s	Stress ratio, $R_s = \frac{S_i}{S_i \text{ max}}$
K_{EB}	Service factor for combining bending and shear stresses	R_T	Fluctuation ratio for torsion, $R_T = \frac{\tau_{T \text{ min}}}{\tau_{T \text{ max}}}$
K_{EN}	Service factor for combining tension-compression and shear stresses	R_U	Cycles per unit of motion
K_{EXY}	Service factor for combining biaxial stresses	r	Fillet radius, in.
K_M	Load distribution factor for gear strength	S_{AC}	Allowable contact stress, psi
K_{NB}	Stress concentration factor for bending	S_{AT}	Allowable bending stress, psi
K_{NN}	Stress concentration factor for tension-compression	S_B	Section modulus, in. ³
K_{NS}	Stress concentration factor for shear	S_T	Polar section modulus, in. ³
K_{NT}	Stress concentration factor for torsion	S_i	Stress amplitude per stress level, ksi
K_P	Cumulative stress effect per stress level, $K_P = R_n \times R_c$	t	Thickness of component where stress is being calculated, in.
K_S	Size factor for gear strength		
K_{SB}	Service factor for bending		
K_{SBA}	Service factor for allowable fatigue bending		
K_{SF}	Service factor for gear strength		
K_{SN}	Service factor for tension-compression		
K_{SS}	Service factor for shear		
K_{SSA}	Service factor for allowable shear		
K_{ST}	Service factor for torsion		
K_V	Dynamic factor for gear strength		
KDS	Stress class factor		
K_{FT}	Strength reduction factor		
K_I	Allowable stress modification factor		
M_B	Bending moment, kip-in.		

V_t	Pitch line velocity, fpm	σ_{UT}	Minimum ultimate tensile strength at mid-radius, ksi
W_A	Weight of column, kips	σ_X	Normal stress about X axis, ksi
W_L	Weight of lifted load, including hook block, kips	σ_{XA}	Allowable normal stress about X axis, ksi
W_T	Weight of trolley, excluding hook block, kips	σ_Y	Normal stress about Y axis, ksi
W_T	Allowable tangential tooth load, lb	σ_{YA}	Allowable normal stress about Y axis, ksi
σ_B	Bending stress, ksi	τ_A	Allowable combined (Equivalent) shear stress, ksi
σ_{BA}	Allowable bending stress, ksi	τ_{ET}	Equivalent torsional shear stress, ksi
σ_{EB}	Equivalent bending (bending and shear) stress, ksi	τ_{EXYT}	Equivalent shear stress in X to Y plane including torsion, ksi
σ_{EBN}	Equivalent bending (bending and tension-compression) stress, ksi	τ_S	Shear stress, ksi
σ_{EN}	Equivalent tension compression (tension-compression and shear) stress, ksi	τ_T	Torsional shear stress, ksi
σ_{EXY}	Equivalent biaxial stress, ksi	τ_{TA}	Allowable torsional shear (Equivalent torsional shear) stress, ksi
σ_{EXYT}	Equivalent stress (biaxial and shear), ksi	τ_{XY}	Shear stress in X to Y plane, ksi
σ_N	Tension-compression stress, ksi	τ_{XYA}	Allowable shear stress in X to Y plane, ksi
σ_{NA}	Allowable tension-compression stress, ksi		

COMMENTARY — MECHANICAL

It is the purpose of this Commentary to amplify, supplement and explain the basis and application of portions of this Report not covered elsewhere. The comments herein are not part of the Report but are added as supplementary information. Numerals in parentheses refer to the Section number in the text of the Report.

ALLOWABLE STRESSES (3.1). Progressive fatigue failures represent the most common mode of failure in steel mill crane machinery. The design criteria in Section 3.1 are, therefore, directed mainly to the prevention of cumulative damage to the material of mechanical crane components.

Material strength properties have been treated on the basis of ultimate strength because of the good relationship of the ultimate strength to the fatigue strength.

Because every component of a crane is subjected to dynamic loading (stress fluctuations), a material's fatigue strength is of prime importance. It should be noted that the yield strength of alloy materials can increase drastically at higher hardnesses, but the fatigue strength will be 50% or less of the ultimate strength. When alloy materials are used, these properties should be certified.

INFINITE LIFE — (3.1.1). Individual consideration shall be given only to the fatigue effects indicated in Section 3.1. Variation in material properties and manufacturing processes have been given consideration in the magnitude of the maximum allowable stress values.

To achieve economical and light-weight crane components while maintaining a high degree of reliability relative to progressive fatigue failure, it is necessary that all detrimental effects on the fatigue strength be reduced to a practical minimum. This may be accomplished by allowing maximum possible fillet radii at all changes of sections, by avoiding abrupt changes of stress flow, improvement of surface finish, etc.

If conventional design and manufacturing methods cannot sufficiently improve an existing critical fatigue condition, special methods of improvement such as grinding and polishing, cross finishing, case or induction hardening of critical component surfaces, shaft shoulder reliefs, compound or elliptical fillet radii may be applied.

Nondestructive testing of the raw material or finished components may further decrease the probability of failure.

The following is a summary of conditions which will affect the fatigue strength of machinery components:

- (1) Hardness or ultimate strength of material

- (2) Notch sensitivity and notch ductility of material
- (3) Material composition
- (4) Material size
- (5) Process used for making raw material for component (cast, forged, hot rolled, cold rolled, etc.)
- (6) Direction of material grain flow relative to direction of principal stress flow
- (7) Type of heat treatment of material
- (8) Local material imperfections
- (9) Material temperature during operation
- (10) Surface conditions (ground, machined, hot rolled, cold rolled, forged, cast, welded, etc.)
- (11) Surface treatment (coating, plating, surface hardening, etc.)
- (12) Direction of surface finish relative to direction of stress flow
- (13) Surface damage prior to cyclic stressing
- (14) Type and magnitude of stress concentration (For stress concentration factors other than those shown in the Report refer to R. E. Peterson "Stress Concentration Factors," John Wiley & Sons, Inc., (1974), or other published documents)
- (15) Type and magnitude of residual stresses
- (16) Stress distribution within component
- (17) Stress spectrum (resulting from all stress cycles during component life including stresses caused by impacts, unintended overloads, as well as natural and resonant vibrations during operation of the crane)
- (18) Stress fluctuation
- (19) Stress combination
- (20) Surface damage simultaneous with cyclic stressing (fretting corrosion wear, etc.)

Fretting corrosion is caused by repeated relative movement (rubbing) of mating component surfaces under pressure. It has, generally, a very damaging effect on the fatigue strength of machinery components and must be given consideration by selecting proper material combinations and application of stress concentration factors. Fretting corrosion exists usually at press fits of track wheels, gears, spacers, antifriction bearings, etc., and at component surfaces where bearing pressures are applied. Relative motions as small as 10^{-6} in. combined with moderate pressures will reduce the fatigue strength of the machinery component. Where heavy fretting actions exist, an increase of material strength usually does not improve the fatigue strength of the component.

FINITE LIFE — (3.1.2). The total spectrum of stresses which a crane component might experience during its expected life should be carefully evaluated to insure maximum

loads and impact
a number of stress
which define the
adequate service fac-
tor for these uninten-
tional factors may be increased
desired, for example,
es.

and

$$S_e = 0.73 \times 0.75 \times 0.814 \times 1.0 \times 0.5 \times \sigma_{UT} = 0.2228 \sigma_{UT} \text{ for } 100,000 \text{ psi } \sigma_{UT} \text{ material}$$

The fatigue diagram shows that the endurance limit in normal
fluctuating stress is:

$$2 \times 0.1963 = 0.3926 \sigma_{UT} \text{ for } 65,000 \text{ psi } \sigma_{UT} \text{ material}$$

and

$$2 \times 0.1822 = 0.3644 \sigma_{UT} \text{ for } 100,000 \text{ psi } \sigma_{UT} \text{ material}$$

The value $0.33 \sigma_{UT}$ for all steels is adopted as suitable for
the majority of crane applications.

Unusual hook applications will merit special considera-
tion. Applications may arise where lower stresses would be
appropriate to ensure compatibility with other hoist com-
ponents, to reduce wear, or to provide an increased margin of
safety.

). The basic stress for-
munity in the recording
throughout the industry.
mbols used in design cal-
ssible to the methods used

he value of $0.33 \sigma_{UT}$ is a
establishing the endurance
n by using this value to plot a
ne mean allowable stress. For
le stress is 2 times the mean

* "Mechanical Engineering Design" by Joseph E.
Shigley, Third Edition

HOOK BODY — (3.2.4).

S_e , the endurance limit is established in the same manner as
for the hook shank where:

$$S_e = K_a K_b K_c K_d K_e K_f S'_e \quad (\text{Eq 60})$$

$$S_e = 0.80 \times 0.75 \times 0.814 \times 1.0 \times 0.5 \times \sigma_{UT} = 0.2442 \sigma_{UT}$$

Where K_a = Surface finish factor for machined or cold drawn
steel of 65,000 psi σ_{UT}

Surface finish factors for machined or cold drawn steel
have been adopted. This allows for a roughness of 250 μ in.
As hook bodies, at the point of maximum stress are usually
cleaned up by rough grinding, this is considered to be in good
agreement with manufacturing practice.

Note that the stress calculated by curved beam theory is
the maximum at what is, in effect, an inherent stress con-
centration. Thus further geometric stress concentration fac-
tors are unnecessary.

The extent of possible readjustment of stress due to plastic
strain is less in the square or rectangular than in the circular
section: hence a lower nominal stress is appropriate. When
designing components of lifting gear having square or rectan-
gular cross-sections, therefore, the design stresses appropriate
to the circular or trapezoidal cross-sections should be reduced
by 10%.*

* Ref: Editorial notes, "British Standard Handbook No.
4", 1959 Edition: Lifting Tackle, Part 2

of mechanical element
of rotating beam specimen

or
actor
factor for stress concentration
is effects factor
materials of 65,000 psi σ_{UT} and 100,000

mitted at this stage, but must be included in the
final stress calculation.

$$0.30 \times 0.75 \times 0.814 \times 1.0 \times 0.5 \times \sigma_{UT} = 0.2442 \sigma_{UT} \text{ for } 65,000 \text{ psi } \sigma_{UT} \text{ material}$$

TESTING — (3.2.5). This clause permits the use of commercially marketed hooks.

The factor of safety from one manufacturer is:

4 : 1 on alloy steel hooks, and

4.5 : 1 on carbon steel hooks

Failure is reported to be by opening of the hook body. Fatigue testing has been carried out.

A factor of safety of 5.0 for all steels is adopted for this standard. It is anticipated that hooks selected on this basis will be more highly stressed than hooks designed to a maximum stress of $0.33 \sigma_{UT}$

METHOD OF ANALYSIS FOR A HOOK OF APPROXIMATE TRAPEZOIDAL SHAPE — (3.2.4).

The analytical method described in this section is intended to apply to hooks with cross-sections having a shape as indicated by the solid line in Fig. 29. This shape does not deviate significantly from a trapezoidal form, and is seen in many crane-hook sections (Fig. 30). This method, while approximate, is faster than the numerical integration method, and in comparative applications, it has been in close agreement.

Essentially the analytical method assumes an equivalent trapezoidal section having an area equal to that of the actual section. The stress thus computed is then corrected for the stress increase in the neutral section (the fibers nearest the center of curvature are farther from the neutral axis than in the case of the equivalent trapezoidal section). It is assumed that the resultant load on the hook passes through the center of curvature of the curved part and that the critical section is at 90 degrees to the resultant load.

In Fig. 29, the solid lines represent the actual hook section, and the broken lines represent the equivalent trapezoidal section. The equivalent section is so chosen that the shaded area A_1 is equal to the areas $A_2 + A_3$. Likewise, A_1' is equal to $A_2' + A_3'$. In Fig. 31, the distribution of stress over the section due to bending alone is indicated. It should be noted that the stress S_o calculated from Eq. 61 yields the bending stress at point A at the inside of the equivalent trapezoid. Because of its greater distance from the neutral axis, the bending stress at point B in the actual hook will be appreciably larger than at point A by an amount S_o as shown. If L_1 is the distance between points B and A, the stress augment S_o will be given approximately by:

$$S_o = (L_1) \left(\frac{ds}{dy} \right)_o \quad (\text{Eq 61})$$

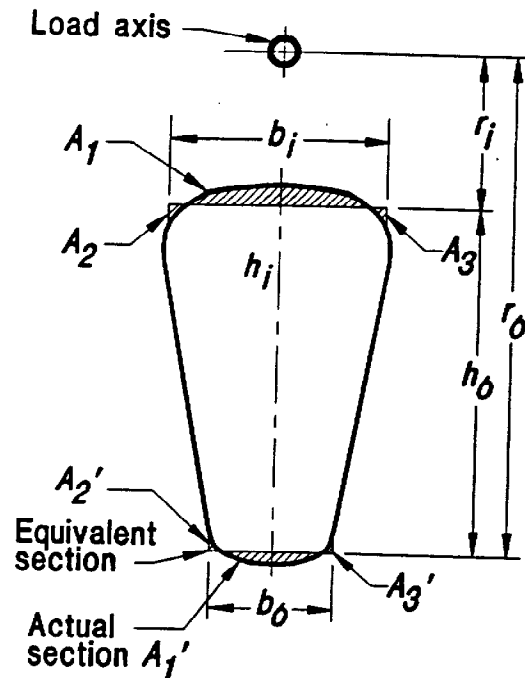


Fig. 29 — Typical hook cross section

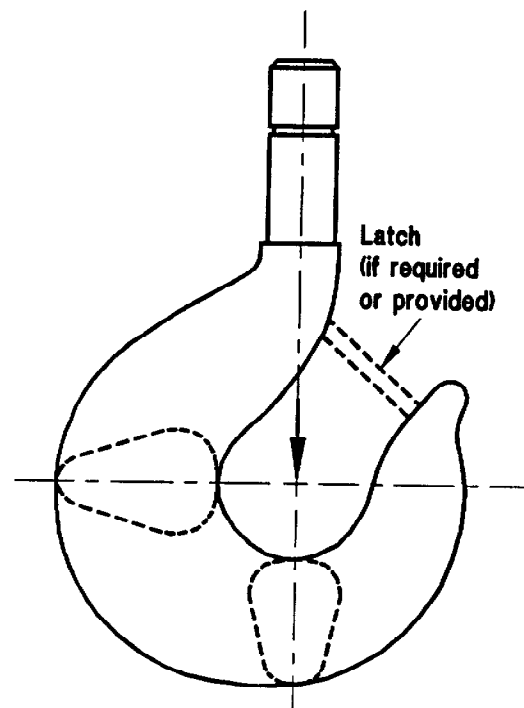


Fig. 30 — Fish hook configuration

From the equations of curved-bar theory, the derivative (ds/dy) may be obtained, and by substitution in Eq. 61, using the previous notations, the stress augment S_o becomes:

$$S_o = \frac{S_b K_2 L_1}{(r_i) \left(K_2 - \frac{1}{\beta - 1} \right)} \quad (\text{Eq 62})$$

where K_2 is given by Eq. 70

The stress due to direct tension is:

$$S_1 = \frac{P}{A} \quad (\text{Eq 63})$$

where A is the area of the cross section.

$$A = \frac{h_o (b_o + b_i)}{2} \quad (\text{Eq 64})$$

where h_o = depth of equivalent trapezoid (Fig. 29).

The maximum stress $S_{\max}$ in the hook at the critical section will be the sum of the bending stress S_b (Eq 68), the stress augment S_o (Eq 62), and the direct tension stress S_1 (Eq. 63). This gives:

$$S_{\max} = S_b + S_o + S_1 \quad (\text{Eq 65})$$

For Fig. 29, let

$b_i b_o$ = inside and outside widths of equivalent trapezoid, respectively

$r_i r_o$ = inside and outside radii of equivalent trapezoid, respectively

$$\alpha = \frac{b_o}{b_i} \quad (\text{Eq 66})$$

$$\beta = \frac{r_o}{r_i} \quad (\text{Eq 67})$$

With these notations, the formulas for bending stress S_b at point A, Fig. 31, at the inside of the trapezoidal section as derived from curved-bar theory becomes

$$S_b = \frac{2PK_1 \left[K_2 - \frac{1}{\beta - 1} \right]}{b_i r_i (1 + \alpha) (K_1 - K_2)} \quad (\text{Eq 68})$$

where

$$K_1 = \frac{1}{\beta - 1} + \frac{2\alpha + 1}{3(\alpha + 1)} \quad (\text{Eq 69})$$

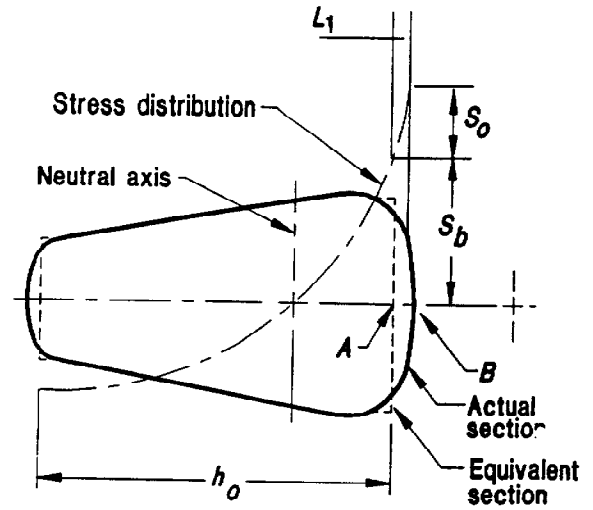


Fig. 31 — Equivalent section

$$K_2 = \frac{\left(\frac{1}{2} \right) (1 + \alpha)}{\left[\frac{(\beta - \alpha)}{(\beta - 1)} \ln \beta \right] - (1 - \alpha)} \quad (\text{Eq 70})$$

In general, the factors K_1 and K_2 should be calculated to at least four significant figures.

The method of analysis and design of a sister hook should be made using the straight beam configuration for the hook.

Fig. 32 shows the general outline and a shape of a sister hook without a pin hole.

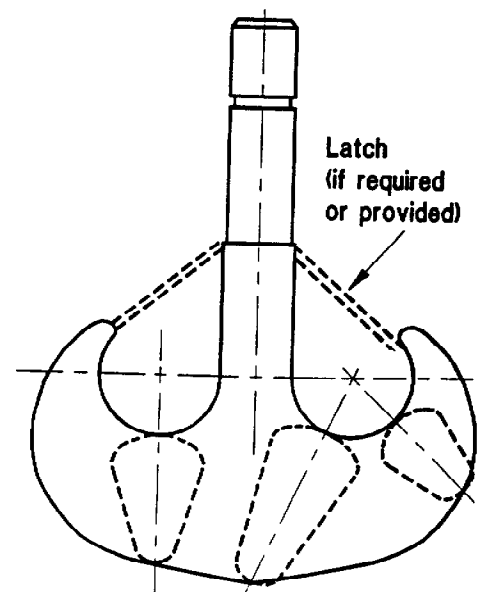


Fig. 32 — Sister hook without a pin hole

Fig. 33 shows the general outline and shape of a sister hook with a pin hole.

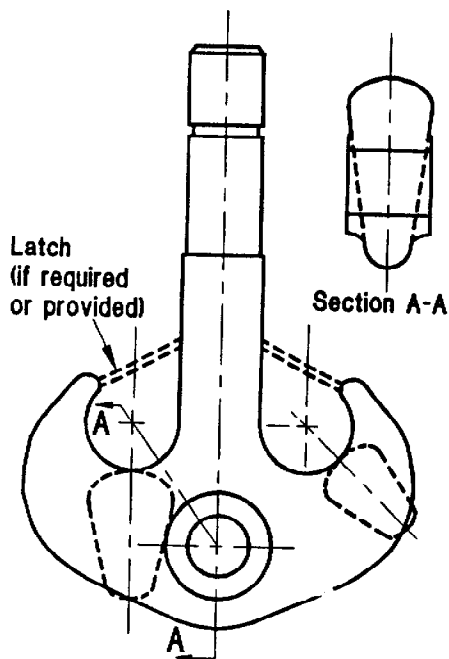


Fig. 33 — Sister hook with a pin hole

CRANE WHEEL HEAT TREATMENTS — (3.7). The purpose of this section is to provide an overview of the currently available technologies for the treatment of steel crane wheels. These technologies all involve heating the wheels either in whole or on the entire tread area, to the steels austenitizing temperature. This is followed by a rapid cooling to change the grain structure and thereby modify the hardness of the wheel. The methods discussed in this report are: rim toughening (spin quench and temper), carburizing, induction hardening and flame hardening.

RIM TOUGHENING — (3.7.1). This procedure involves the uniform heating of the entire wheel section above the austenitizing temperature and then spin quenching the rim section of the wheel for sufficient time for this area to cool below the lower critical range (i.e. M_s temperature — refer to isothermal transformation diagrams for selected steels). The entire wheel is then heated to an intermediate “tempering” temperature to reduce surface hardness to the 321 BHN to 388 BHN range. This is the method of heat treatment specified in ASTM A-504.

This treatment is the most popular method heat treatment for crane wheels because it generally yields the deepest hardness of any of the methods described. The hardness

obtained is above forged wheels in their untreated state (175 BHN - 250 BHN) but below case hardened (600 BHN - 700 BHN) and therefore has load bearing capabilities and machinability characteristics intermediate to those types of wheels.

CARBURIZING — (3.7.2). This procedure involves selectively imparting a high hardness to the periphery of a wheel by exposing the wheel to a carburizing atmosphere such as high purity natural gas or propane. Other atmospheres may include pack carburizing with graphite or cyaniding which is liquid carbonitriding. In most cases the entire wheel is heated and quenched.

The result of this treatment is a high surface hardness due to carbon and/or nitrogen addition at the surface. The depth of hardness is a function of material, time temperature, atmosphere and quench severity.

INDUCTION HARDENING — (3.7.3). In this procedure, the wheel tread is subjected to localized heating generated by highly concentrated and rapidly alternating magnetic field which flows through the inductor or work coil. The pattern of heating obtained by induction is determined by the shape of the coil, the number of turns in the coil, operating frequency and the current power output. The part is selectively heated to the austenitizing temperature and locally quenched to obtain regional high hardness. Depth and hardness level can be controlled by the previously mentioned variables.

FLAME HARDENING — (3.7.4). This procedure involves uniformly heating the rim of the wheel above the austenitizing temperature by exposing it to a high temperature flame generated by the combustion of a fuel gas mixed with oxygen or air, then quenching rapidly, forming martensite to the desired depth. The wheel is then tempered to yield the desired hardness level. Flame hardening is performed by either the Progressive Method or the Spinning Method. In the Progressive Method, the piece is rotated very slowly with the flame head heating one section at a time. A quench spray follows the flame head very closely or is integral with it and subsequently hardens the piece one section at a time until the entire piece is hardened. In the Spinning Method, the piece is spun more rapidly while multiple flame heads uniformly heat the surface. When the piece reaches the desired surface temperature, the entire piece is submerged into an agitated quench tank. The piece is then tempered to the required hardness. Progressive hardening is more typical for the hardening of gears where shallow hardness is required in order to preserve tooth core properties, and spin hardening is more appropriate for crane wheels where depth of hardness is more

DESIGN EXAMPLE — MECHANICAL

Plain pin (Fig. 34) in which consideration is given only to infinite life of the piece:

Given: Material with $\sigma_{UT} = 117$ ksi

$A = 12.57$ sq in.

$K_{SB} = K_{SS} = 1.10$ (example value)

$M_B = 130$ in.kips

$P = 50$ kips

$R_B = R_S = +0.14$ (example value)

$S_B = 6.28$ in.³

Solution: From Fig. 18 $\sigma_{BA} = 23.05$ ksi

From Fig. 20 $\tau_A = 11.45$ ksi

K_{NB} and $K_{NS} = 1.0$

$$\sigma_B = \frac{M_B}{S_B} \times K_{SB} \times K_{NB} = \frac{130}{6.28} \times 1.10 \times 1.0 = 22.8 \text{ ksi} < 23.05 \text{ ksi; OK (Eq 31)}$$

$$\tau_S = \frac{1.33 P}{A} \times K_{SS} \times K_{NS} = \frac{1.33 \times 50}{12.57} \times 1.10 \times 1.0 = 5.82 \text{ ksi} < 11.45 \text{ ksi; OK (Eq 35)}$$

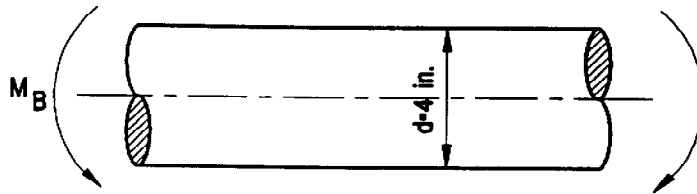


Fig 34 — Plain pin in bending



4 ELECTRICAL

4.1 Brakes — Hoist, Trolley and Bridge.

Magnetic brakes shall conform to AISE Technical Report No. 11.

On a-c cranes, magnetic brakes are to be operated through a d-c magnet. The d-c shunt coil excitation system shall provide quick response similar to a d-c series wound coil. Direct current power shall be provided by static means.

Brakes shall have ample thermal capacity for the frequency of operation required by the service to prevent impairment of functions from overheating.

Brake coil time rating shall be ample for the duration and frequency of operation required by the service. Any traverse drive brake used only for emergency stop on power loss or setting by operator choice shall have a coil, or a coil and excitation system, rated for continuous duty.

Service brakes are defined as the braking means, other than motor braking, used for normal slowing or stopping of a bridge, trolley or cab.

Parking brakes are defined as a mechanical braking means used for holding a bridge, trolley or cab for indefinite periods of time. External wind loads must be considered.

4.1.1 Hoist Brakes. Each hoist on a crane shall be equipped with at least one spring-set magnetic brake. Where a single brake is used it shall be mounted on the outboard end of the motor speed pinion shaft, the end of which shall have a taper fit for the brake wheel of the same dimensions as that on the motor shaft.

All hoists handling hot metal shall be equipped with more than one brake. Other hoists shall be equipped with multiple brakes if specified on the OIS. Unless otherwise specified, these brakes shall be mounted on the outboard ends of additional motor speed pinion shafts if available on multimotor drives. If these additional shafts are not available, additional brakes shall be mounted on motor shafts opposite the drive ends. When all motor speed pinion shafts and motor shafts have been supplied with one brake each, additional brakes may be mounted to other drive train shafts as required.

Brake sizes shall be as recommended by the brake manufacturer for the service, but in no case shall the summation of all brake ratings in percent of hoist full load hoisting torque at the points of brake application be less than the following:

- (1) 150% when only one brake is used
- (2) 150% when multiple brakes are used and the hoist is not used to handle hot metal; failure of any one brake shall not reduce total braking torque below 100%

- (3) 175% for hoists handling hot metal; failure of any one brake shall not reduce total braking torque below 125%.

For example, if two brakes are used, each must be rated 100% of the total full load hoisting torque (125% each for hot metal). If three brakes are used, each must be rated 50% (62.5% each for hot metal). If four brakes are used, each must be rated 37.5% (43.75% each for hot metal). In each of these cases, the failure of one brake does not cause the remaining braking torque to fall below the required minimum.

On multiple motor hoists that are arranged for operation under emergency conditions with one or more motors bypassed, brakes in operation during emergency bypass operation shall provide braking torque in accordance with this Section.

4.1.2 Trolley Brakes.

4.1.2.1 Operator's Cab on Bridge (Fixed or Movable). Trolleys with anti-friction bearings shall be provided with a mechanical drag brake, a spring-set magnetic brake or a remote controlled service brake, as specified on the OIS.

The drag brake shall be installed on the trolley motor shaft and shall be of sufficient capacity to prevent the trolley from drifting. The magnetic brake shall have a torque rating of not less than 50% of the trolley motor 60-minute rated torque and be adjustable so that its torque can be decreased by 50%. The remote controlled service brake shall have a capacity as outlined in Section 4.1.3.1. The brake shall be arranged to set whenever power is removed from the motor unless otherwise specified on the OIS.

4.1.2.2 Operator's Cab on Trolley. A trolley brake shall be provided as described for bridge brakes in Section 4.1.3.2 or as otherwise specified.

4.1.2.3 Floor, Pulpit or Remote Operated Cranes. The requirements for trolley brakes shall be the same as specified in Section 4.1.2.1.

4.1.3 Bridge Brakes.

4.1.3.1 General. Service brakes shall have sufficient thermal capacity and torque range to stop the bridge within a distance not to exceed a length in feet equal to 10% of the full load speed in fpm when traveling at full speed with full load, or to stop the bridge from full load top running speed to zero speed at a deceleration rate for the drive as specified on the OIS. In either case, the deceleration rate should be selected so that wheel slippage does not occur under minimum wheel

load conditions. The thermal capacity shall be adequate for the number of stops/hr specified on the OIS.

When foot-operated, the stroke of the brake foot pedal shall not be more than 8 in. nor require an applied force of more than 70 lbs to stop the bridge as described. The lever shall be designed and positioned so that it will not interfere with necessary movements of the operator's legs or feet while operating the crane.

Brakes on all outdoor cranes, and others if specified, shall be provided with a spring-set parking feature and also be arranged to set on loss of power. The torque capability of the brakes shall be sufficient to statically hold the bridge against the external loads specified.

4.1.3.2 Operator's Cab on Bridge. Each bridge drive shall be equipped with a foot-operated hydraulic or electrical adjustable torque service brake or brakes sized in accordance with Section 4.1.3.1.

4.1.3.3 Operator's Cab on Trolley. Each bridge drive shall be equipped with a brake or brakes having a spring-set parking feature, and also be arranged to set on loss of power. The brake shall be sized in accordance with Section 4.1.3.1. This type of brake system is usable on drives where motor braking is used for routine stopping.

In addition, when motor braking is not used by the operator for routine stops, one of the available remote controlled brake systems which will provide service braking similar to cab-on-bridge cranes should be specified on the OIS.

The several functions may be combined in a single brake.

4.1.3.4 Floor, Pulpit or Remote Controlled Cranes. The requirements for bridge brakes shall be the same as specified in Section 4.1.3.3.

4.1.4 Independently Movable Cab. This drive shall be supplied with a type and rating of brake as specified on the OIS.

Trolley brakes shall be as specified in Section 4.1.2.1 and bridge brakes shall be as specified in Section 4.1.3.3.

4.2 Conductors.

4.2.1 Runway Conductors. The main conductors for the crane bridge travel shall be furnished and erected by the owner unless otherwise specified on the OIS. The location, size and type of these conductors shall also be specified by the owner.

4.2.2 Bridge Conductors. Bridge conductors shall be accessible for service. The conductors may consist of insulated multiconductor (or several single conductor) cables with permanent termination on the bridge and on the trolley together with suitable means for supporting, extending and retracting the cable to allow relative movement of the bridge and trolley

without undue stress or wear on the cable (festooned cable, cable conveyors or cable reels). The conductors may also take the form of rigid structural shapes. Where low contact resistance is required for low current or voltage pilot devices, suitable combinations of conductor and collector material shall be used.

Continuous insulated cable systems are preferred on a-c systems where momentary interruption of current due to collector action can cause a control malfunction, or where low-voltage, low-power signals must be transmitted. Where such systems are used, special attention shall be given to the wear resistance and thermal adequacy of the insulation and the flexibility of the conductor. Cable supports shall not unduly stress nor wear the conductor, and the movable supports shall move freely. Suitable strain relief devices shall be incorporated where stress could otherwise occur in cables. Wire sizes shall be in accordance with AISE Technical Report No. 8, and shall be selected so that the overall system voltage drop does not exceed that acceptable to the equipment involved when the maximum current is imposed. Consideration should be given to the inclusion of spare conductors or provision for the later addition of additional conductors.

Where rigid conductors are specified on the OIS, they shall be located or guarded so that persons cannot normally come into contact with them. They shall be mounted on insulated supports spaced not more than 6 ft apart for flat bars and 8 ft apart for angles, or according to manufacturer's recommendations for other special types. Conductors and supports shall be spaced so as to give a clear electrical separation of conductors or adjacent collectors of not less than 1 1/4 in. for systems up to 600 V.

Provisions shall be made for expansion and contraction of rigid conductors due to temperature changes.

The design and construction of the supports shall be sufficiently strong and rigid to maintain proper alignment.

In some locations, special attention shall be given to dusty and otherwise unfavorable environments. Here, conductors should be mounted to accept siderunning or underrunning collectors and insulators located to prevent excessive dust accumulation. Where sections of conductors are joined together, either welded joints or bolted splices may be used. In either case, the joint must be electrically and mechanically sound, without excessive gaps or misalignment. On cranes where auxiliary cable reels are not specified, provision shall be made for conductor supports and collector staffs to have two additional bars and shoes (or more as specified on the OIS) that could be used for magnet control or other purposes.

4.3 Collector Shoes

4.3.1 D-C Systems. The main bridge collector shoes (a minimum of two for positive and two for negative collectors) and

and the trolley collector shoes are to be furnished by the contractor unless otherwise specified on the OIS. The collectors shall be designed to suit the type of conductors used and shall be proportioned to provide adequate current-carrying capacity.

Double trolley collector shoes shall be furnished in a dynamic lowering loop and on magnet circuits when furnished.

4.3.2 A-C Systems. Most crane drive systems using a-c power are more sensitive to the continuity of their circuits; therefore, special attention shall be given to the design of the collectors. All collectors used with these systems shall be of the double-shoe spring-loaded type. The design shall minimize the chance of binding at hinge points due to dust or corrosion. The spring pressure shall be adequate to keep the shoe in continuous contact with the conductor under all conditions of operation and to provide low voltage drop at the contact junction. Shoe material shall have a low wear rate and adequate current carrying capacity.

4.3.3 Collector Shunts. Current-carrying shunts on all collectors shall be designed so that there is no danger of contact with adjacent collectors. Separate shunts shall be used from each shoe to the cable terminal. The shunt shall be designed so that the movement of the shoes in normal operation does not produce localized stress in the shunt itself which will lead to early failure. The shunts shall be easily replaceable.

4.3.4 Mounting. All bridge collector shoes shall be mounted on rigid, suitably insulated steel staffs and located or guarded so that persons cannot normally come into contact with them. Collectors shall be designed for ease of maintenance and mounted so that they are readily accessible for this purpose. Electrical clearance between live parts of adjacent shoes shall be at least 1 inch. Flexible shunts in their least favorable position shall not reduce this clearance.

4.4 Motors — Hoist, Bridge and Trolley.

The following motor selection procedure is based on the use of AISE Technical Report No. 1. If a motor other than an AISE Standard 1 motor is used, the crane supplier shall provide evidence of mechanical and electrical adequacy (including peak torque and thermal capacity) for the operating conditions and duty cycle specified by the owner.

For multiple motor drives arranged for operation under emergency conditions with one or more motors bypassed, the supplier shall state in the proposal the changes in lifting capacity, speed, acceleration, and duty cycle due to the bypassed condition.

4.4.1 D-C Motors. All d-c motors shall be the totally enclosed mill type in accordance with AISE Standard No. 1 or alternate as specified on the OIS.

4.4.2 A-C Motors. All a-c motors shall be the totally enclosed wound rotor mill type in accordance with AISE Standard No. 1A or alternate as specified on the OIS.

4.4.3 Motor Size Selection, a-c or d-c.

4.4.3.1 General. Because of the large variety of crane drives available and the difference in the effects of those drives on the thermal adequacy of the motors under consideration, a procedure for selecting motor ratings is relatively complex. Therefore, whenever possible the owner should specify the most severe repetitive duty cycle for each motor including intervals of slow speed operation. The supplier shall be responsible for selecting the ratings that will meet the specified duty with the type of control specified. In the absence of duty cycle requirements, the OIS must clearly identify the service class to be used for each motion in the procedure described herein. Table 14 may be used by the purchaser as a guide in the selection of service class; however, the data in that table is only typical and may be modified to meet the specific requirements of any installation.

If the OIS specifies that the motors are to be used for prolonged time intervals in an ambient temperature above 40°C, and if the Owner's Information Sheets also specify that the same margin between allowable temperature of the motor insulation and the rated motor rise at 40°C ambient is to be maintained during those intervals, correction factors from Table 15 shall be used to multiply the horsepower value determined in Sections 4.4.3.2 and 4.4.3.3 before selecting the motor 60-minute rating.

A-c motors and controls shall be suitable for infrequent momentary voltage dips (not to exceed 1 minute duration out of 60 operating minutes) to not less than 85% of name plate voltage. A voltage correction factor, K_v , for a-c hoist drives is to be included in the motor selection if the OIS specifies

Table 15 — Ambient Temperature Correction Factor for a-c and d-c Mill Motors

Ambient Temperature		Ambient Correction Factor, K_t
°C	°F	
40	104	1.00
45	113	1.05
50	122	1.11
55	131	1.18
60	140	1.25
65	149	1.33

NOTE: If the temperature requirement is not specified on the OIS, AISE-type mill motors with Class F or H insulation may be selected for an ambient temperature of 65°C or less without using these ambient correction factors, since AISE Technical Report No. 1 requires ratings based on Class B temperature rise.

Table 14 — Typical Crane Service Data

	Crane			Bridge			Trolley			Main hoist			Auxiliary hoist						
	Average number of hours operation in one year	Normal ambient temperature during operation °F	High ambient temperature during operation °F	Average number of hours operation in one year	Percent time on in operational hour	Number of moves per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of moves per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of lifts per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of lifts per hour	Service Class — Electrical
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Coke Plant and Blast Furnace Cranes:																			
Drawing machine (coke pusher)	7200	80	135	3240	45	17	4	2088	29	9	2	1296	18	7	1	1440	20	6	1
Bucket handling	3240		110	1458	45	19	4	1458	45	25	4	2009	62	25	—	2106	65	27	—
Stock yard	6840			3078	47	14	4	3762	55	48	—	2941	43	38	4	2736	40	100	—
Slag handling	5400		100	1870	31	21	3	1836	34	30	3	2052	38	21	3	3240	60	35	—
Scrap yard	7200			3240	45	30	4	1440	20	27	2	4104	57	80	—	—	—	—	—
Cast house	1080			648	60	32	—	400	37	32	3	616	57	75	—	432	40	20	3
Pig machine						15	1		90	15	—		90	4	—		5	2	1
Ladle house	1800			360	20	22	1	630	35	25	3	576	32	15	3	846	47	17	4
Skull cracker	720			36	5		1	576	80	30	—	648	90	30	—	14	2	1	1
Sand house (bucket)	360			324	90	21	—	342	95	21	—	54	15	21	1	324	90	20	—
Setting basin	720			360	50	30	4	108	15	60	3	180	25	60	3	216	30	60	3
Car repair shop	1440			216	15	20	1	288	20	35	2	432	30	40	2	288	20	20	1
Ore bridge	7560			756	10	10	1	3780	50	35	4	454	6	35	2	3024	40	35	3
Coke ovens (coal bridge)	5760			1440	25	15	2	2880	50	78	4	634	11	78	4	2419	42	78	4
Open Hearth, Electric Furnace, BOP Cranes																			
Charging machines	6840	91	111	2120	31	98	—	3146	46	89	4	1847	27	80	4	1368	20	88	4
Hot metal crane (charging)	6480	118	129	2850	44	27	4	1426	22	32	2	1750	27	13	2	2592	40	31	3
Ladle	6480	101	108	2916	45	33	4	3110	48	23	4	1690	26	9	2	1361	21	18	2
Metal mixer crane	3600	90	120	648	18	12	1	1008	28	18	2	1980	55	20	—	1080	30	33	2
Electric furnace charging cranes	2880			864	30	45	2	864	30	60	3	1728	60	100	—	—	—	—	—
Stock yard	6480	70	80	1940	30	76	4	2851	44	89	4	3110	48	93	4	—	—	—	—
Scrap preparation					10	64	3		50	64	4		40	32	3	—	—	—	—
Scrap baler crane	7560			2040	27	40	2	2268	30	40	2	3780	50	80	4	—	—	—	—
Scrap shear crane	7560			4990	66	40	—	1275	17	40	2	4838	64	80	—	—	—	—	—
Hot top	5400		130	1350	25	55	3	810	15	57	3	1890	35	30	3	2430	45	90	4
Bucket	5040			2420	48	52	4	2218	44	49	4	2066	41	43	4	1915	38	20	3
Slag handling	6480			1820	28	25	3	1944	30	8	2	2268	35	10	3	—	—	—	—
Cinder yard	5040	70	100	2120	42	55	4	2621	52	80	—	3276	65	82	—	4032	80	25	—
Skull cracker	6480	70	80	1620	25	70	3	2592	40	71	3	3110	48	23	4	3564	55	37	—
General service	3960		100	1310	33	22	3	911	23	26	2	555	14	23	1	1584	40	56	3
Ingot Handling Cranes:																			
Ingot handling	6840			1500	22	30	2	1163	17	33	2	3762	55	17	—	342	5	28	1
Soaking pit	3960	122	145	1660	42	42	4	1703	43	42	4	2020	51	60	4	832	21	50	2
Stripper	4680	70	113	1070	23	38	2	1638	35	38	3	2059	44	64	4	1123	24	31	2
Mold yard	6120	72	101	2570	42	44	4	1714	28	63	3	2815	46	55	4	612	10	20	1
Rolling Mill Cranes:																			
Slab yard	6480	60	100	3629	56	26	—	1944	30	34	2	2786	43	30	4	778	12	16	1
Slab furnace charging	4680		105	1400	30	40	2	1640	35	45	3	2800	60	61	—	468	10	10	1
Plate and strip handling	6480	110	112	3694	57	27	—	1426	22	26	2	2203	34	41	3	—	—	—	—
Billet mill	4320		108	1690	39	23	3	1166	27	23	2	2030	47	31	4	—	—	—	—
Billet shipping	7200	100	100	935	13	10	1	864	12	10	1	1800	25	35	2	—	—	—	—
Rail mill	3600		100	1440	40	10	3	720	20	20	1	1800	25	35	2	—	—	—	—
Rail loading dock	3240	100		2430	75	20	—	1620	50	20	4	1620	50	20	4	—	—	—	—
Rail shipping	7920			3240	41	30	4	4356	55	30	—	3722	47	80	4	—	—	—	—
Hot mill	5400		115	1400	26	18	2	1296	24	22	2	1188	22	13	2	1782	33	15	3
Cold strip mill	6840	75	100	2530	37	20	3	1984	29	27	2	1505	22	26	2	2120	31	24	3
Coil storage	7920	75	120	5227	66	73	—	3247	41	72	4	3485	44	66	4	—	—	—	—
Roll shop	4320	75	90	1730	40	28	3	1166	27	24	2	1037	24	19	2	1080	25	26	2
Mill service	5040	90	110	1810	36	30	3	1260	25	30	2	1663	33	19	3	1865	37	40	3
Machine shop	5040			1810	36	31	3	1410	28	41	2	1560	31	33	3	2020	40	30	3
Cooling building	5760		132	1840	32	26	3	2189	38	27	3	2822	49	35	4	—	—	—	—
Inspection and conditioning	3600			1800	50	50	4	1080	30	45	2	2520	70	75	—	180	5	10	1
Pit cover	5400		100	2700	50	25	4					1350	25	20	2	1350	25	25	2

Table 14—Cont'd.

	Crane			Bridge			Trolley			Main hoist			Auxiliary hoist						
	Average number of hours operation in one year	Normal ambient temperature during operation °F	High ambient temperature during operation °F	Average number of hours operation in one year	Percent time on in operational hour	Number of moves per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of moves per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of lifts per hour	Service Class — Electrical	Average number of hours operation in one year	Percent time on in operational hour	Number of lifts per hour	Service Class — Electrical
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Finishing Mill Cranes:																			
Slab storage	6480	80	100	1944	30	40	2	1944	30	40	2	1944	30	20	2	324	5	2	1
Billet yard	3960			1660	42	32	4	475	12	32	2	1069	27	45	2				
Furnace room	4320			2592	60	30	—	648	15	30	1	1080	25	30	2				
Mill service	5040	80	105	1870	37	19	3	1411	26	23	2	1562	31	17	3	1109	22	25	2
Shipping	5760	70	100	2995	52	35	—	1382	24	36	2	2304	40	52	3				
Warehouse	5040	75	100	2320	46	31	4	1260	25	32	2	2419	48	35	4				
Sorting room	7200			3600	50	60	4	1800	25	60	3	2160	30	60	3				
Scale pit	360			126	35	25	3	72	20	25	1	324	90	25	—	324	90	25	—
Hot bed	1440			216	15	5	1	144	10	17	1	288	20	15	1				
Pickling	6840	80	110	2800	41	30	4	1710	25	39	2	2462	36	47	3				
Quenching			100		5	2	1		5	2	1		5	2	1				
Tin mill	6840	75	100	2530	37	36	3	1710	25	41	2	2394	35	42	3	684	10	20	1
Annealing	7200	67	116	2890	40	22	3	2232	31	24	3	2016	28	16	2	2808	39	25	3
Battery shop	5400			540	10	22	1	432	8	22	1	540	10	11	1				
Rod and Wire Mill Cranes:																			
Billet yard	7800	67	87	4300	55	27	—	2340	30	27	2	2730	35	17	3				
Rod mill	4300	80	115	3000	70	25	—	1300	30	25	2	1700	40	20	3				
Rod dock	8000	60	110	6400	80	70	—	4800	60	70	—	3200	40	50	3				
Cleaning house	8000	90	100	4800	60	100	—	1800	20	60	3	3200	40	90	4				
Pot annealing	6900	87	110	3800	55	85	—	2400	35	65	3	1700	25	30	2	5500	80	100	—
Rod storage	7300	94	96	4400	60	34	—	2200	30	34	2	700	10	34	2				
Patenting department	2900	90	94	1900	65	32	—	450	15	32	2	580	20	32	2				
Bar bending and storage	8000	60	110	6400	80	100	—	2400	30	50	2	3200	40	40	3				
Tube Mill Cranes:																			
Hot mill	8000	90	140	4800	60	60	—	1800	20	60	3	2640	33	30	3				
Finishing mill	8400	80	115	5050	60	70	—	1700	20	70	3	2800	33	35	3				
Galvanizing	8400	75	115	5050	60	70	—	1700	20	70	3	2800	33	35	3				
Miscellaneous Cranes:																			
Forging manipulators	5400	80	115	2480	46	33	4	2430	45	30	4	1134	21	18	2	108	2	1	1
Hydraulic forging cranes	8640	80	110	3630	42	17	4	3197	37	40	3	3024	35	22	3				
Warehouse	3960			1700	43	16	4	713	18	17	1	990	25	16	2				
Gantry	5040	70	90	2020	40	34	3	2268	45	81	4	2520	50	88	4				
Repair	1800	80	102	504	28	15	2	360	20	14	1	378	21	12	2	396	22	15	2
Machine shop	5040	80	90	2068	41	21	4	1310	26	21	2	1008	20	12	1	1512	30	18	2
Service	1800			610	34	11	3	288	16	12	1	450	25	14	2				
Power house	1080	90	115	270	27	8	2	270	27	13	2	260	24	7	2	230	21	13	2
Motor room	1080		110	240	22	4	2	162	15	6	1	194	18	4	1	162	15	4	1

1. — A cycle for a bridge or trolley consists of two 'moves', one loaded and one unloaded.

2. — For hoist drives, one cycle consists of two 'lifts', one loaded and one unloaded, plus two lowering operations. Unless otherwise specified, it will be assumed that full load will be raised and lowered, and that 'no-load' will be raised and lowered through the same distance, using the maximum speeds provided by the selected type of drive, with reasonable rates of acceleration and deceleration.

Wherever a dash mark appears in the service class column, the listed data indicates that the particular motion is required to operate more than 50% time on or more than 45 cycles/hr; therefore a value of service factor cannot be assigned and the requirements must be submitted to the supplier for the selection of adequate ratings.

that the motor thermal capacity and acceleration capability be based on a normal condition of the a-c voltage at the control panel which is less than rated voltage (not below 85%). The horsepower values determined by the following procedure should be multiplied by:

$$K_V = \left\{ \frac{\text{Motor Nameplate } V}{\text{Minimum Specified } V} \right\}^2 \quad (\text{Eq 71})$$

Values of K_V at voltages between 85 and 100% of the motor nameplate voltage are given in Table 16

The service factors to be used for each service class, motion and type of drive when no duty cycle has been specified are listed in Tables 17 and 18. These factors are based on past practice and may be conservative in some cases.

Table 16 — Voltage Correction Factor for a-c Mill Motors

Percent Voltage	Voltage Correction Factor, K_V	Percent Voltage	Voltage Correction Factor, K_V
100	1.00	92	1.18
99	1.02	91	1.21
98	1.04	90	1.23
97	1.06	89	1.26
96	1.09	88	1.29
95	1.11	87	1.32
94	1.13	86	1.35
93	1.16	85	1.38

Consideration should be given to the fact that the relationship between the dissipating capability and the internal heating of motors may vary considerably with size, type and manufacturer. In addition, the heat developed in travel motors is influenced by the relative portion of the service class percent time-on devoted to accelerating and braking.

Because over-sized series motors on hoist or travel drives can introduce problems of overspeeding or wheel slippage

and unreasonable gear ratios (Section 4.12.4), consideration can be given to using service factors lower than those in Table 17 (especially for AISE-type frames 804 and smaller).

However, the suitability of any reduced service factors must be verified by duty cycle analysis; a typical example is given in Section 4.4.3.4.

4.4.3.2 Hoists. The hoist motor shall be selected so that its 60-minute rating will not be less than that given by the following formula:

(1) Constant potential or adjustable voltage d-c drives

$$\text{hp} = \frac{(K_s W_L v)}{33,000 E_c} \quad (\text{Eq 72})$$

Where:

K_s = Service factor from Table 17

v = Specified hoisting speed, fpm

W_L = Weight of the lifted load including weight of hook block, lb

E_c = Combined efficiency of gears and sheaves for hoist drives

= $0.93^n \times 0.98^m$ for sleeve bearings

= $0.97^n \times 0.99^m$ for antifriction bearings

Where:

n is the number of gear reductions (sets of gears and pinions) and m is the total number of rotating sheaves between drum and equalizer passed over by each part of the moving rope attached to the drum.

Table 19 shows combined mechanical efficiency for various combinations of ropes and gearing with antifriction bearings.

(2) Constant potential or adjustable voltage a-c drives

$$\text{hp} = \frac{(K_s K_V W_L v)}{33,000 E_c} \quad (\text{Eq 73})$$

Table 17 — Service Factors for Series Motors Operated at 230V, or Shunt Motors on Adjustable Voltage Drives

Maximum Percent Time-on of Motion	20	30	40	50
Maximum Cycles/hr *	15	25	35	45
Service Class Electrical, from Table 15	1	2	3	4
Service Factor, K_s				
Hoist	0.75	0.75	0.82	0.96
Bridge and Trolley	1.1	1.2	1.3	1.4

* See footnote on Table 15 for definitions of cycle

Table 18 — Service Factors for a-c Motors

Maximum Percent Time-on of Motion	20	30	40	50
Maximum Cycles/hr *	15	25	35	45
Service Class Electrical, from Table 16	1	2	3	4
Service Factor, K_s				
Resistance Increased for Slow Speed and Plugging				
Hoist	1.0	1.0	1.1	1.2
Bridge and Trolley	1.1	1.2	1.3	1.4
Fixed Resistance				
Hoist	1.1	1.2	1.3	1.4
Bridge and Trolley	1.3	1.4	1.5	1.6

* See footnotes on Table 14 for definitions of cycle

Where:

K_s = Service factor from Table 18

K_v = Voltage correction factor from Table 16

Note: For an a-c hoist, the specified full load hoist speed should be obtained at not more than rated motor torque. To meet this requirement for an a-c hoist that has some permanent secondary resistance during full speed hoisting, and to include the selected service factor in a way that allows for the reduction in per unit slip when the service factor increases the motor rating, use Eq 74 instead of Eq 73.

The motor rating shall not be less than

$$hp = \left\{ K_s - 1 + \frac{0.97}{1 - Res_{pu}} \right\} \left\{ \frac{K_v W_L v}{33,000 E_c} \right\} \quad (\text{Eq 74})$$

Obviously, if $K_s = 1$, $K_v = 1$, and the slip rings are shorted on a motor with the 0.03 per unit internal resistance

that is assumed in these calculations, $hp = \frac{W_L v}{33,000 E_c}$ that

is, the minimum motor rating is the mechanical hp required for steady-state hoisting of rated load at rated speed.

As an illustration of the effect of permanent resistance and service factor, assume that hoisting the full load at rated speed with shorted slip rings requires 70 hp. If the type of control has 0.2 per unit total secondary resistance at full speed, $K_s = 1$, and $K_v = 1$ the minimum motor rating is $\frac{0.97}{0.80} \times 70 = 84.9$ hp, requiring an a-c 18, 90-hp motor.

Table 19 — Combined Mechanical Efficiency for Hoist Drives with Antifriction Bearings

Total Number of Parts Double Reeved	Rope Reduction, R_r	Number of Sheaves, m	Efficiency of Ropes Only, $(0.99)m$	E_c With Two Gear Reductions, $n = 2$	E_c With Three Gear Reductions, $n = 3$
4	2	1	9.990	0.931	0.904
6	3	2	0.980	0.922	0.895
8	4	3	0.970	0.913	0.886
10	5	4	0.961	0.904	0.877
12	6	5	0.951	0.895	0.868
14	7	6	0.941	0.886	0.859
16	8	7	0.932	0.877	0.851
18	9	8	0.923	0.868	0.842
20	10	9	0.914	0.860	0.834
22	11	10	0.904	0.851	0.825
24	12	11	0.895	0.842	0.817
26	13	12	0.886	0.834	0.809
28	14	13	0.878	0.826	0.801
30	15	14	0.869	0.817	0.793
32	16	15	0.860	0.809	0.785

However, if $K_s = 1.4$ in this example, the minimum is $\left\{ 1.4 - 1 + \frac{0.97}{0.80} \right\} \times 70 = 112.9$ hp, requiring an a-c 25, 125-hp motor. Fig. 40 will show that with a 90-hp motor $\left(\frac{70}{90} = 0.78 \text{ per unit hp} \right)$, 0.2 per unit resistance will result in approximately 0.82 per unit synchronous speed or 18% slip. With a 125-hp motor $\left(\frac{70}{125} = 0.56 \text{ per unit hp} \right)$, 0.2 per unit resistance will result in approximately 0.88 per unit speed or 12% slip.

Fixed resistance in Table 18 indicates that there are no secondary contactors or other means to change secondary resistance, although there may be controlled reactance.

4.4.3.3 Bridge and Trolley. The force required to drive the bridge or trolley consists of the forces necessary to overcome rolling friction, and to accelerate or decelerate the crane. The rolling friction is proportional to the total weight of the crane and is assumed to be constant at all speeds. Unless otherwise specified on the OIS, an overall friction factor, f , from Table 20 shall be used for cranes with antifriction bearings and 24 lb/ton for cranes with sleeve bearings. If the ratio of track wheel diameter to journal diameter is not 4:1, calculate the sleeve bearing friction factor by:

$$f = \frac{0.043}{0.90} \times \frac{\text{journal diameter}}{\text{wheel diameter}} \times 2000 \text{ lb/ton} \quad (\text{Eq 75})$$

The size of the bridge and trolley motor (60-minute mill rating at the selected voltage) shall not be less than that computed from the following formula:

$$\text{hp} = K_s K_a W_t v \quad (\text{Eq 76})$$

Where:

- K_a = Acceleration factor
- K_s = Service factor from Table 17 or Table 18
- v = Specified full load speed after 10 seconds
- W_t = The total weight of the crane or trolley plus load, tons

The factor K_a includes power for both overcoming friction and accelerating the crane or trolley. The derivation of K_a acceleration factors is explained in Section 4.12.3. Based on the assumptions listed in that Section, typical values of K_a for series motor drives are given in Fig. 35 and values for either

a-c drives or adjustable voltage d-c drives with constant motor field strength are given in Fig. 36. The required acceleration for selection of the K_a factor is to be as specified on the OIS. For series motor drives, the specified rate of acceleration applies on the resistor. For a-c or adjustable voltage d-c drives with constant motor field strength, the rate of acceleration applies up to rated speed.

The gear ratio for bridge and trolley motors will be determined as shown in Section 4.4.4, computing the free-running hp from the following formula:

$$\text{hp} = \frac{(f W_t v)}{33,000} \quad (\text{Eq 77})$$

Where:

- f = Rolling friction from Table 20 or Eq 75.

EXAMPLE 1

Series motor for double A 5 bridge drive 230 V constant potential:

$$\text{hp} = K_s K_a W_t v \quad (\text{Eq 76})$$

For 20% time-on and 15 cycles/hr, service factor K_s is 1.1 from Table 18

For 12 lb/ton and 1.0 fps², $K_a = 0.00085$ from Fig. 35

$$v = 500 \text{ fpm}$$

$$W_t = 208 \text{ ton}$$

$$\text{hp} = 1.1 \times 0.00085 \times 208 \times 500 = 97.24 \text{ hp total or } 48.62 \text{ hp per motor}$$

Use two 808, 50-hp, 60-minute motors.

Determine gear ratio by obtaining speed from motor curve at

$$\text{hp} = \frac{(f W_t v)}{33,000} = \frac{(12 \times 208 \times 500)}{33,000} = 37.8 \text{ hp total or } 18.9 \text{ hp per motor} \quad (\text{Eq 77})$$

Fig. 37 shows that an 18.9 hp gear-in. speed should be approximately 1025 rpm.

Table 20 — Overall Friction Factors (Antifriction Bearings)*

Wheel Diameter, in.	12	15	18	21	24	27	30	36
f , lb/ton	15	15	15	12	12	12	10	10

*For cranes equipped with sleeve bearings of normal proportions, $f = 24$ lb/ton. Mechanical efficiencies are included in these factors. As a guide to the selection of acceleration rates, see Section 4.12.

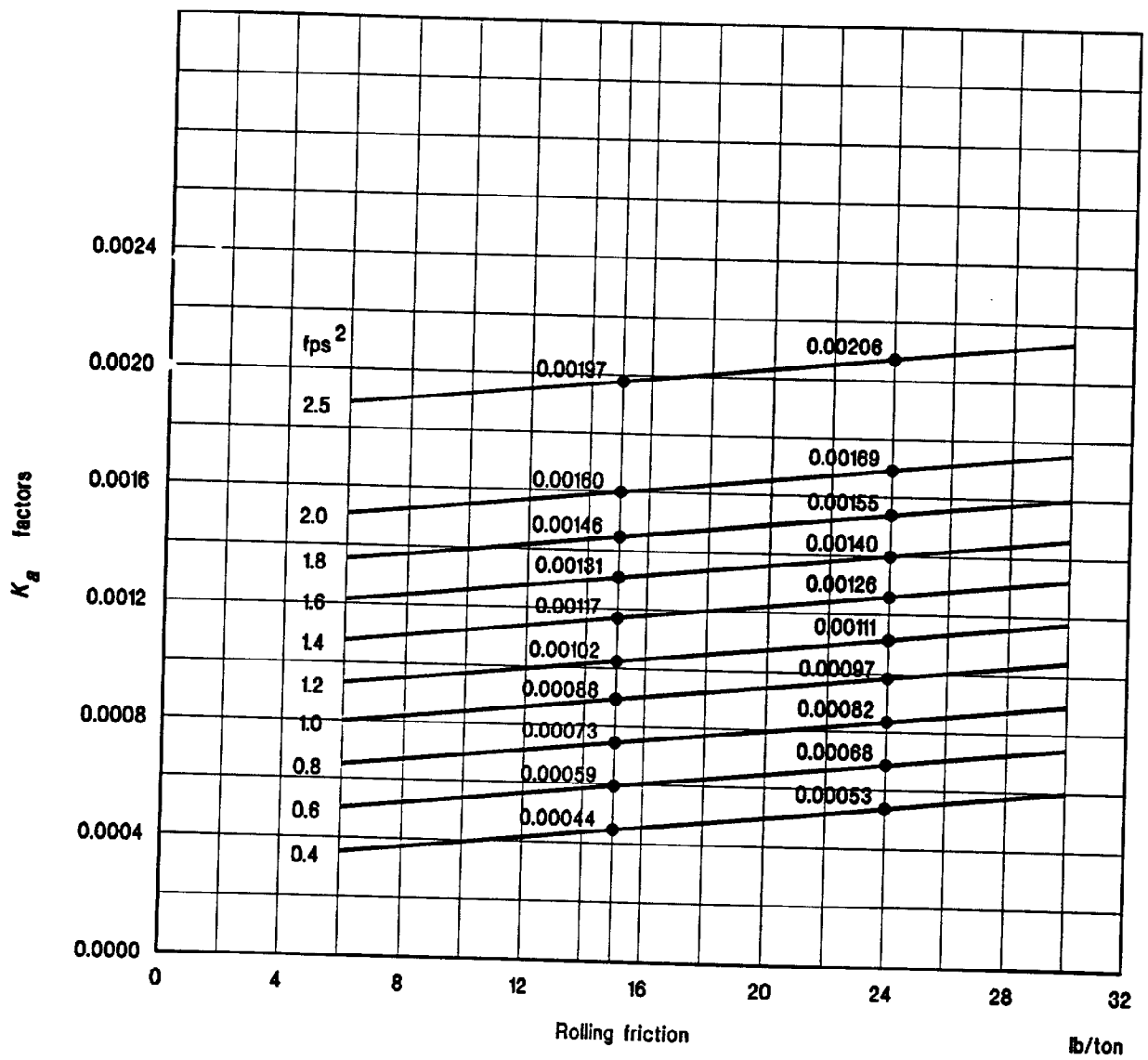


Fig. 35 — K_a factors for series motor drives

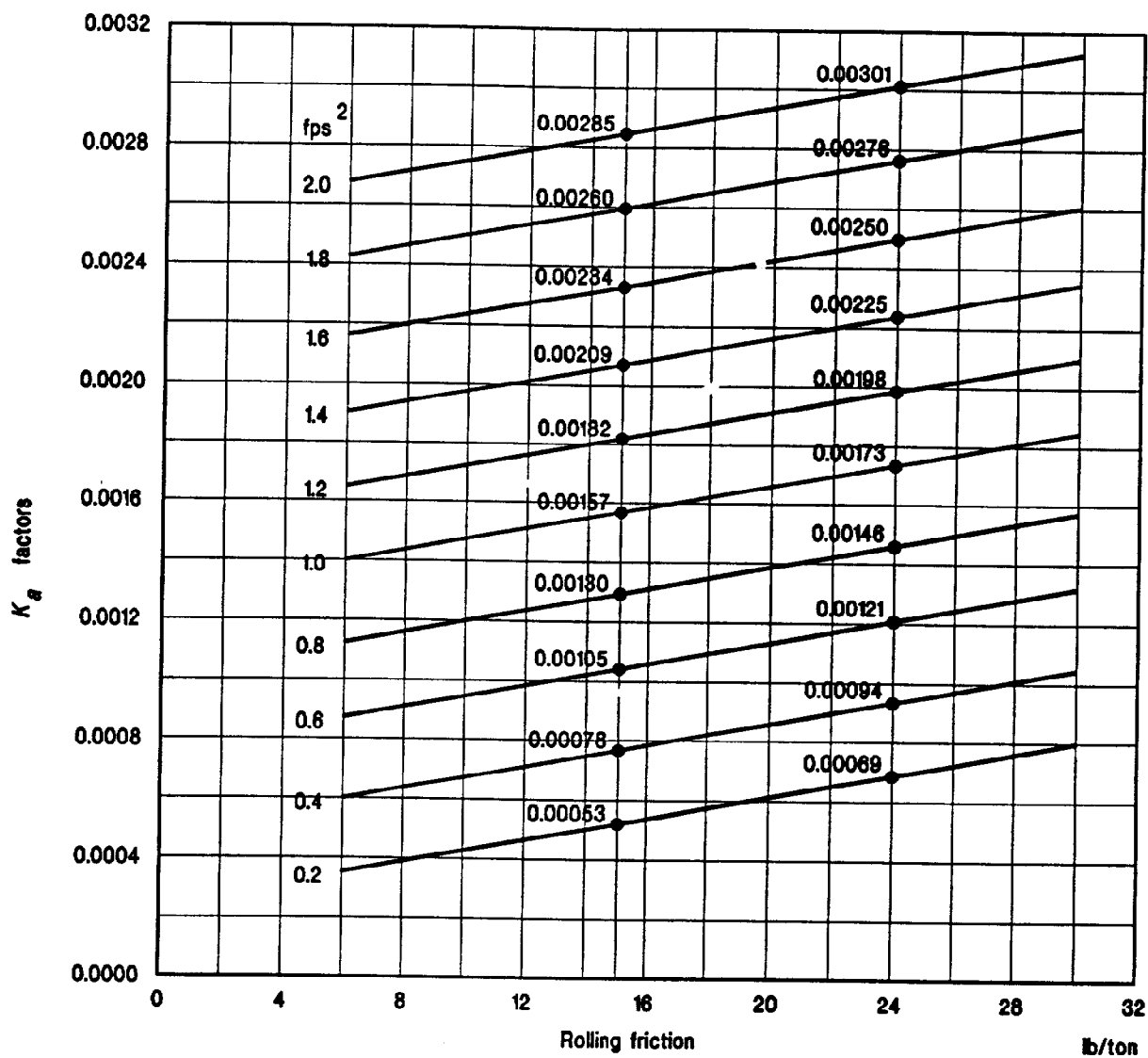


Fig. 36 — K_a factors for a-c and adjustable voltage d-c motors (without field weakening)

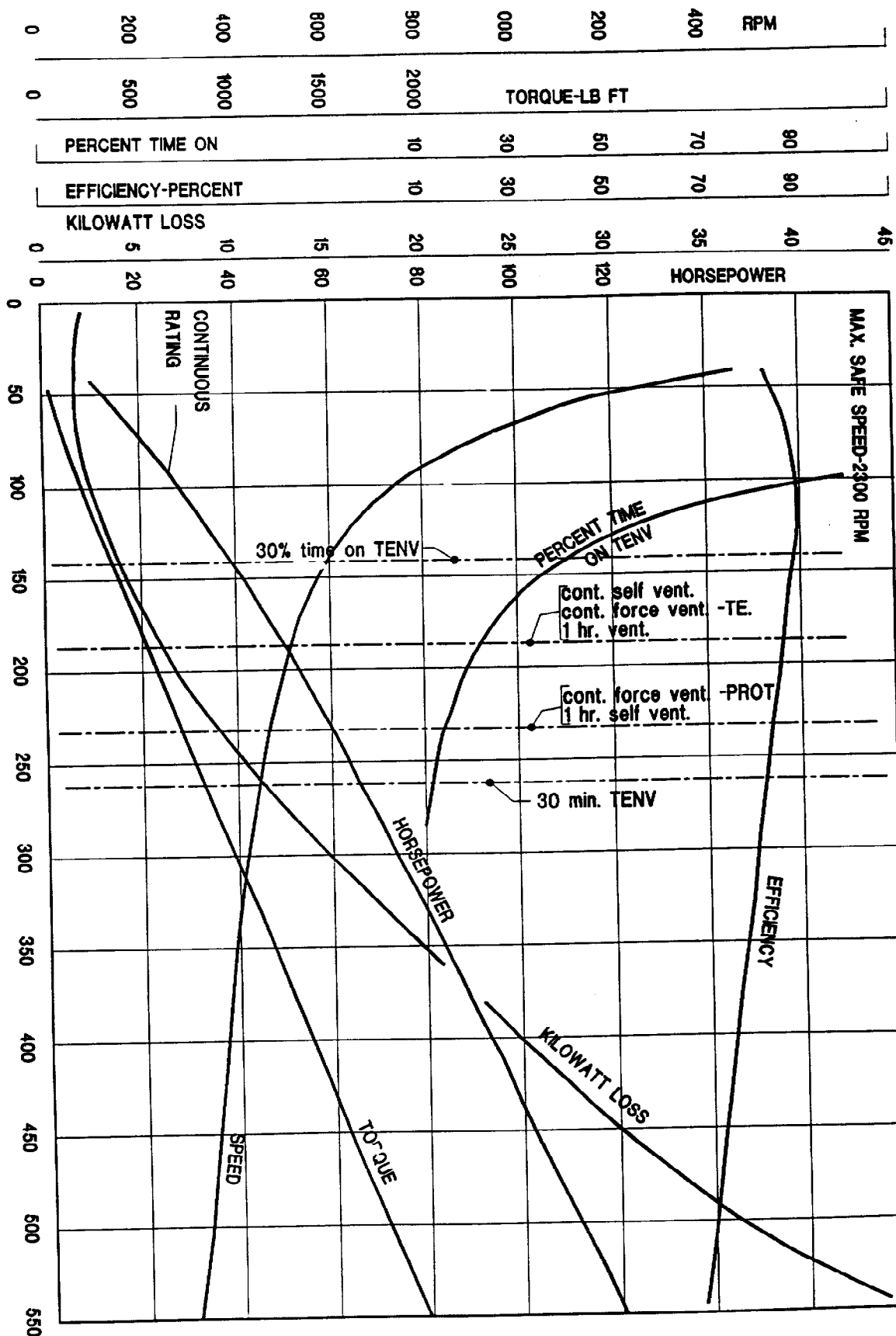


Fig. 37 — Characteristic curves for 808 series-wound, 230 V motors

4.4.3.4 Selecting Motors Based on Duty Cycle (Less than 50% time-on). For selecting motors based on duty cycle (up to and including 50% time-on), use the specified percent time-on and cycles/hr to arrive at the kw loss compared to kw dissipating capability of the selected motor in the specified ambient.

For series motors operated at 230 V, the kw loss and allowable percent time-on curves are to be obtained for the selected motor. Fig. 37, which applies to an 808 motor made by one manufacturer, is typical of the published curves.

Start with a motor having a 60-minute rating obtained by using the service factors in Table 17 with Eq 72 for hoists or Eq 76 for bridges and trolleys.

Establish the critical duty cycle as shown in Table 24, using as many steps as necessary. Calculate the time and motor torque for each step in the cycle. From the motor characteristic curves, tabulate the kw loss corresponding to each torque step, and multiply each kw loss by the corresponding time. Add the kw $\times$ second values to obtain total kw loss.

Divide by the total time the motor is energized, resulting in average kw loss while on. At the current corresponding to that kw loss, read allowable percent time-on from the motor curves. If that value is above the percent time-on in the cycle, the selected motor has adequate thermal capacity. If it is necessary to try a different motor (larger or smaller), be certain to change the gearing as necessary to meet the specified speed, then calculate the inertias and torques resulting from the revised gearing.

EXAMPLE 2

This example shows the kw loss procedure for a bridge drive similar to that in Example 1 (Electrical), by the service factor method.

Wt = 325 ton (650,000 lb)
 v = 150 fpm (2.5 fps)
 a = 0.9 fps²
 f = 15 lb/ton
 n = Motor rpm
 n_f = Motor rpm at the free-running hp (see Eq 77 and Section 4.4.4)

Time-on = 30%

Cycles = 25/hr = 144 second/cycle

Since only the loaded weight is given, consider only the 72-second loaded portion of the cycle.

On time = 0.3 $\times$ 72 = 21.6 seconds

Rest time = 72.0 - 21.6 = 50.4 seconds

Check AISE frame 808 motor, 50-hp, 525 rpm, 60 minutes, 500 ft-lb: For 22.2 hp (free-running), n_f = 920 rpm for 150 fpm (2.5 fps)

$$T = \frac{\text{hp} \times 5250}{n_f} = \frac{22.2 \times 5250}{920} = 127 \text{ ft-lb} \quad (\text{Eq 78})$$

$$808 \text{ motor } WK^2 = 61.0 \text{ lb-ft}^2$$

$$13 \text{ in. brake wheel } WK^2 = 12.8 \text{ lb-ft}^2$$

$$\text{Estimated mechanical } WK^2 = 14.8 \text{ lb-ft}^2$$

(20% of motor and brake for the example)

$$\text{Total } WK^2 = 88.6 \text{ lb-ft}^2$$

$$\text{Equivalent Load } WK^2 =$$

$$650,000 \text{ lb} \times \left(\frac{150}{2 \times \pi \times 920} \right)^2 = 437.7 \text{ lb-ft}^2$$

Total equivalent WK^2 (assume 90% efficiency):

$$\text{For acceleration loaded} = 88.6 + \frac{437.7}{0.9} = 574.9 \text{ lb-ft}^2$$

$$\text{For deceleration loaded} = 88.6 + 437.7 \times 0.9 = 482.5 \text{ lb-ft}^2$$

For acceleration loaded: Assume acceleration on resistors to the 60-minute rated speed, which is 525 rpm. (Note: The type and adjustment of the accelerating relays may result in attaining more than rated speed on the resistor).

$$\text{Speed} = 2.5 \times \frac{525}{920} = 1.43 \text{ fps}$$

$$\text{Time (at } 0.9 \text{ fps}^2) = \frac{1.43}{0.9} = 1.6 \text{ seconds}$$

$$\text{Distance} = \frac{(1.43 \times 1.6)}{2} = 1.14 \text{ ft}$$

$$\text{Acceleration torque} = \frac{574.9 \times 525}{308 \times 1.6} = 613 \text{ lb-ft}$$

$$\text{Motor } T = 613 + 127 = 740 \text{ lb-ft}$$

$$\text{From 525 to 725 rpm, } \Delta n = 200 \text{ rpm}$$

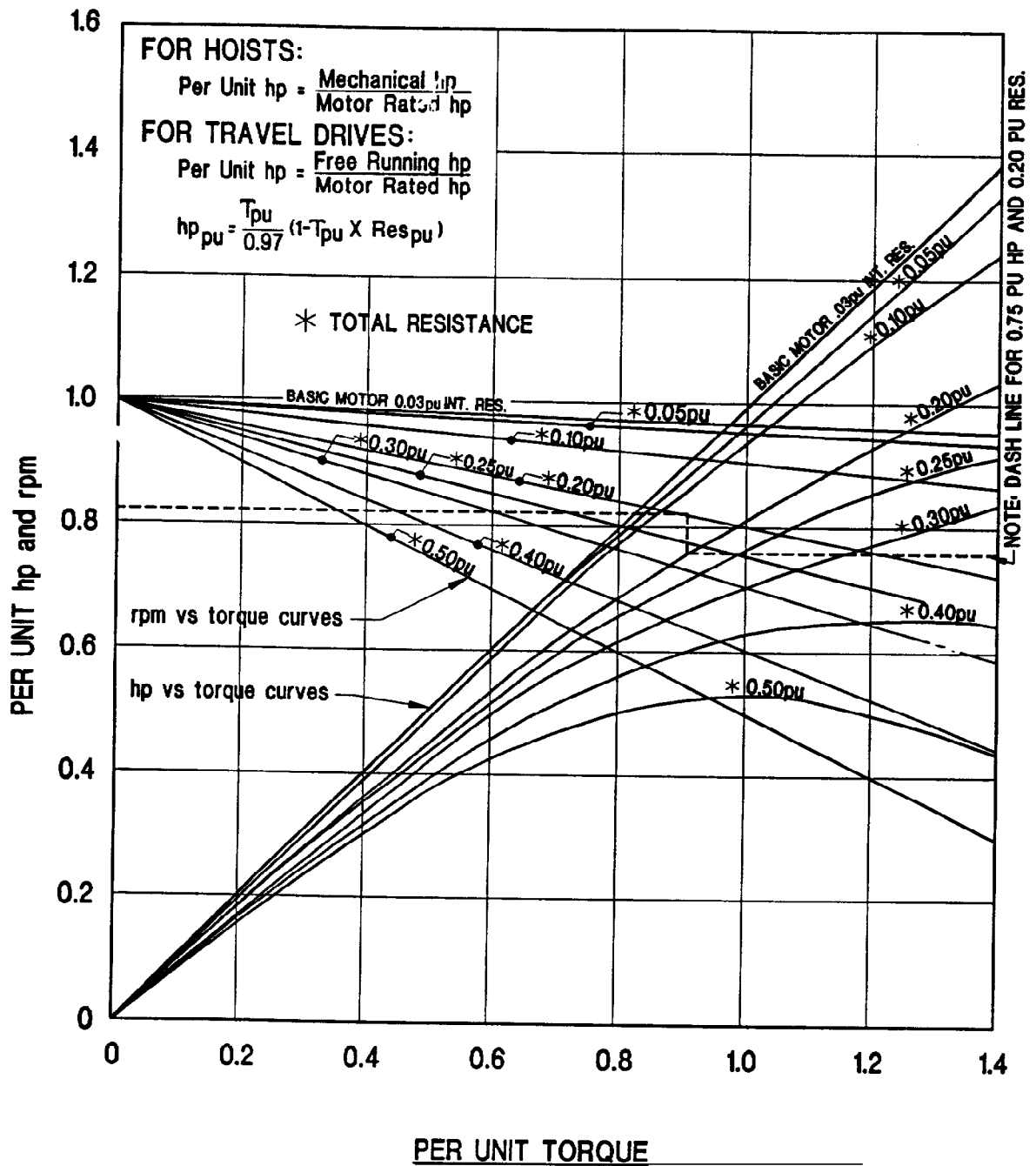
$$\text{At 525 rpm, } T = \frac{50 \text{ hp} \times 5250}{525} \text{ rpm} = 500 \text{ lb-ft}$$

$$\text{At 725 rpm, } T = 240 \text{ lb-ft from graph (Fig. 38)}$$

$$\text{Average motor } T = \frac{500 + 240}{2} = 370 \text{ lb-ft}$$

$$\text{Acceleration } T = 370 - 127 = 243 \text{ lb-ft}$$

$$t = \frac{574.9 \times 200}{308 \times 243} = 1.54 \text{ seconds}$$



One per unit secondary resistance is the total resistance per phase in the motor secondary circuit that will result in rated motor torque at zero speed with rated voltage applied to the motor primary. The values of rated secondary current voltage and one per unit resistance are to be obtained from the motor manufacturer.

One per unit secondary resistance is the total resistance per phase in the motor secondary circuit that will result in rated motor torque at zero speed with rated voltage applied to the motor primary. The values of rated secondary current voltage and one per unit resistance are to be obtained from the motor manufacturer.

Fig. 39 — Characteristics of a-c mill motors

given value of secondary resistance the approximate secondary current can be calculated by:

$$I_{pu} = \sqrt{\frac{T_{pu} \times S_{pu}}{Res_{pu}}} \quad (\text{Eq 79})$$

Where:

I_{pu} = Secondary current, per unit

S_{pu} = Slip, per unit

T_{pu} = Torque, per unit

Res_{pu} = Total per unit resistance, in motor secondary (including internal)

If the calculated I_{pu} is less than the corresponding T_{pu} , use the T_{pu} value.

Also, in order to take into consideration the primary copper losses at very low values of torque, the value of I_{pu} must not be less than 0.4. Start with a motor having a 60-minute rating obtained by using the service factors in Table 18, with Eq 74 for hoists and Eq 76 for bridge and trolleys.

Establish a duty cycle with the time and torque for each step calculated as in Example 2. Convert torque to per unit current by Eq 79 or by the torque-current-speed characteristics of the type of control to be used. Add (the square of the per unit current) $\times$ (time in seconds) $\times$ (per unit variable losses) to (the operating time in seconds) $\times$ (per unit fixed losses). If the total is less than the sum of the seconds times the dissipation factors for each step in the cycle, the motor has adequate thermal capacity. The variable losses, fixed losses and dissipation factors are to be obtained from the selected motor manufacturer, or the cycle summary is to be submitted to the drive manufacturer.

EXAMPLE 3

Assume the motor being considered has variable losses of 0.663 and fixed losses of 0.337, with a dissipation capability of 0.39 at 100% speed, 0.34 at 50% speed and 0.29 at zero speed. (These values are based on 1.0 per unit losses cor-

responding to the motor 60-minute rated hp and speed, with rated voltage on the primary and rings shorted). Compare the losses developed in the motor by a control system having 0.2 per unit total secondary resistance to the losses developed by a control system that increases resistance during acceleration and plugging to result in $I_{pu} = T_{pu}$. Assume that the control limits the average accelerating torque to 150% and the average decelerating torque to 100%.

In the tabulation below, per unit amps for the control with 20% total fixed secondary resistance has been calculated as follows:

For acceleration, average slip = 0.5

$$I_{pu} = \sqrt{\frac{T_{pu} \times S_{pu}}{Res_{pu}}} = \sqrt{\frac{(1.5)(0.5)}{(0.2)}} = 1.94 \text{ per unit amps}$$

Variable losses $\times$ time = $I_{pu}^2 (0.663)t = 12.4$ per unit kw seconds

For run, use minimum I_{pu} of 0.4

Variable losses $\times$ time = $I_{pu}^2 (0.663)t = 1.3$ per unit kw seconds

For deceleration, average slip when plugging = 1.5

$$I_{pu} = \sqrt{\frac{(1.0)(1.5)}{(0.2)}} = 2.74 \text{ per unit amps}$$

Variable losses = $I_{pu}^2 (0.663)t = 23.9$ per unit kw seconds

For either type of control, fixed losses equals $21.6 \times 0.337 = 7.3$ per unit kw seconds. The total losses with the fixed resistance is $37.6 + 7.3 = 44.9$ per unit kw seconds, which is considerably above the 22.5 per unit kw second dissipation; therefore the motor would overheat. In comparison, the total losses in the control designed to make $I_{pu} = T_{pu}$ during acceleration and plugging = $12 + 7.3 = 19.3$ per unit kw seconds, which is below the 22.5 per unit kw second dissipation, therefore the motor would be satisfactory.

	Time, seconds	Per Unit Torque	Per Unit Amps 0.2 pu ohms	Per Unit kw-seconds Variable Losses 0.2 pu ohms or $I_{pu} = T_{pu}$		Per Unit Average Speed	Dissipation $\times$ time, seconds
Accelerate	5.0	1.50	1.94	12.4	7.5	0.5	1.7
Run	11.8	0.25	0.40	1.3	1.3	1.0	4.6
Plug	4.8	1.00	2.74	23.9	3.2	0.5	1.6
Time On	21.6						
Rest	50.4						
Total	72.0			37.6	12.0		22.5

4.4.3.5 Selecting Motors Based on Duty Cycle (Above 50% time-on). Above 50% time-on or more than 45 cycles/hr, the required duty cycle capability must be specified on the OIS. The possible advantages of self-ventilated, forced-ventilated or air-over-frame motor construction should be considered, depending on the atmospheric conditions at each installation and the motor construction specified.

If prolonged or repetitive operation at reduced speed is required, it must be specified on the OIS. If it is of a repetitive nature but not more than 30 seconds or less than 5% speed, the calculations can be included as in Example 2 or 3.

Because the variations in motors and controls can be appreciable, it is essential that ratings selected by any duty cycle calculations be checked by the electrical drive manufacturer after an order has been placed.

4.4.4 Drive Gear Ratios. Drive gear ratios shall be determined as follows:

$$GR = \left\{ \frac{\pi D}{12 R_a} \right\} \left\{ \frac{n_f}{v} \right\} = \left\{ \frac{0.262 D}{R_a} \right\} \left\{ \frac{n_f}{v} \right\} \quad (\text{Eq 80})$$

Where:

- D = Pitch diameter of drum for hoists or wheel tread diameter for traverse drives, in.
- GR = Gear reduction ratio
- R_a = Mechanical advantage of the rope system for hoists ($R_a = 1$ for traverse drives)
- v = Specified speed, fpm
- n_f = Motor rpm corresponding to the steady-state hp of a hoist drive or free-running hp of a travel drive (not including acceleration hp) adjusted for the voltage and control used as follows:

For 230 V d-c series motors, the manufacturer's characteristic curves for 230 V shall be used. At constant-potential voltage other than 230 V, obtain an equivalent 230 V hp by multiplying the free-running hp by 230 divided by the applied voltage. From the curves, use this equivalent hp to obtain the motor speed at 230 V. Calculate the approximate n_f by multiplying the rpm obtained by the applied voltage divided by 230. (This approximation is within acceptable tolerances if the equivalent 230 V hp is not over the motor 60-minute AISE rating. Above that hp, obtain n_f at the desired voltage from the motor manufacturer.)

For a-c wound rotor motors, the typical characteristic curves for wound rotor motors, Fig. 39 shall be used, taking into consideration the total secondary resistance at full speed. The curves are based on motors providing 3% slip at

rated torque with rings shorted and with rated voltage applied to the primary,

$$hp_{pu} = \frac{T_{pu}(1 - T_{pu} \times Res_{pu})}{0.97} \quad (\text{Eq 81})$$

Per unit hp for use of these curves =

$$\frac{\text{Steady State or Free running hp}^*}{\text{hp Rating of Motor}} \quad (\text{Eq 82})$$

*(not including acceleration)

(The steady-state hp for a hoist is calculated by Eq 73 with $K_s = 1$, and the free-running hp for a bridge or trolley by Eq 77.)

At the calculated per unit hp, read per unit torque from appropriate hp-resistance curve and then read per unit synchronous speed at that torque on the speed curve for the same resistance. The dash line is an example at 0.75 per unit hp and 0.2 per unit total resistance, resulting in approximately 0.88 per unit torque and 0.82 per unit synchronous speed.

For d-c adjustable voltage shunt motors, obtain manufacturer's rated speed for armature voltage and field strength used.

4.5 Control — Hoist, Bridge and Trolley

4.5.1 General. Control shall conform to the NEMA Industrial Control and Systems Standard Part ICS 3-442 Class I for Overhead Traveling Cranes, except as modified by these specifications, or the OIS.

Manual control shall not be furnished for any motion unless specifically permitted by the OIS.

The control shall be operable in a 40°C ambient at +10 to -10% variation in the nominal voltage of an a-c power supply unless otherwise stated on the OIS. The voltage variation shall apply at the incoming power terminals at the control panels under minimum-maximum current conditions.

The voltages in push button, master switches and similar remote control circuit devices shall not exceed 150 V a-c or 300 V d-c. If the control circuit is grounded on the crane, these limits apply from either side of the control circuit to ground; if ungrounded, the limits apply from line to line.

Contactors, relays and all other panel components shall be mounted on suitable switchboard materials or steel panels, of ample thickness, on suitable supports, and with the bottom of the lowest panel-mounted device not less than 6 in. from the floor. Power terminal lugs shall have at least a 6-in. clearance from top, sides and bottom of enclosures.

Contactors shall be equipped with means of confining and extinguishing an arc.

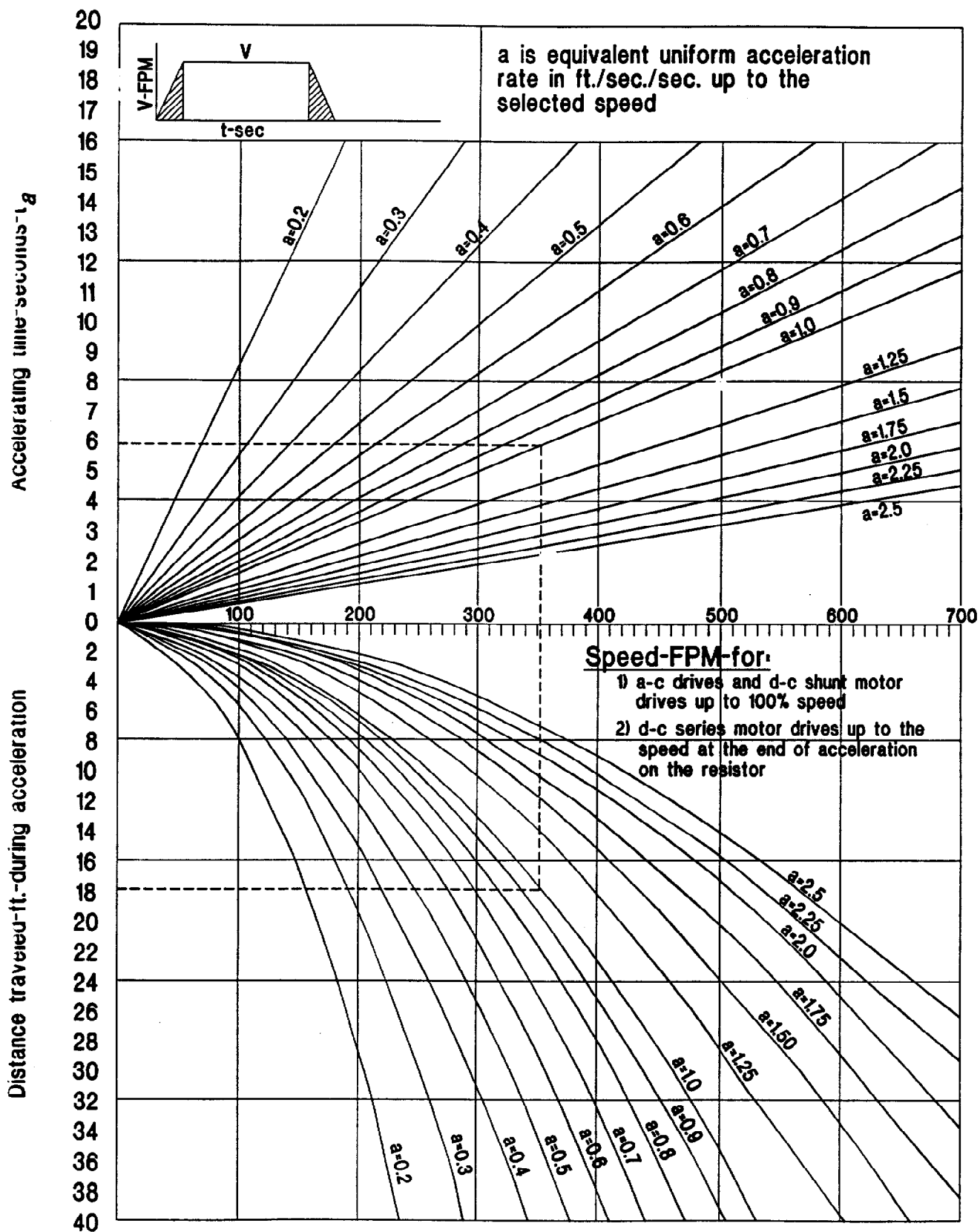


Fig. 40 — Speed - acceleration - time - distance curves

If enclosures are required for the control panels, the type of enclosure shall be in accordance with the classifications as listed in the NEMA Industrial Controls and Systems Standard, Part ICS 1-110 and shall be so specified on the OIS. The doors of NEMA 1, 3 and 12 enclosures shall be hinged to open at least 170 degrees, shall not project more than 20 in. in front of the enclosure when open 90 degrees, and shall be equipped with captive hinge pins that will allow the doors to be removed.

Control panels, either open or enclosed, shall be braced to the crane structure.

All control panels shall be positively pressurized (≤ 0.2 lbs/sq ft) and air conditioned when component and ambient conditions warrant. When air conditioning is applied, air conditioning unit failure shall be annunciated at panel and operator's cab. On cabinets housing critical or multiple drive controllers, redundant air conditioning units should be applied with failure annunciation at panel and operator's cab and automatic primary-to-secondary unit switchover. Further, after completion of all wiring, testing, start-up, and commissioning, all penetrations, escutcheons and the like shall be sealed with a pliable, easily removable, yet fire resistant sealant (foam, paste or caulk is acceptable) to redeem integrity of control cabinets.

Unless otherwise specified, resistors shall be of NEMA Industrial Controls and Systems Standard, Part ICS 2-213 Class 160 or greater, except that the stalled torque may be modified to meet the performance requirements of the application, and resistors for motors rated 30 hp or below may be either edgewise-wound or other nonbreakable type. Above 30 hp, resistors shall be punched grids or continuous non-welded stainless steel on nonbreakable supports in standard mill-type boxes. The boxes shall be mounted in racks that permit independent removal of any selected box and provide spacing recommended by the resistor manufacturer.

Controls for all motions of the crane shall be equipped with acceleration devices or regulators with means for adjustment. Plugging protection shall be provided for all bridge and trolley drives. The crane manufacturer shall furnish complete data including motor thermal service factor, K_s , from which the control supplier is to design resistors, acceleration devices and plugging protection, so as to obtain the specified average acceleration rate and to avoid wheel slippage. (Note: The purchaser shall be notified if wheel slippage limitations make it impossible to meet the acceleration rate with the type of control specified.)

For hoist motions, controlled lowering shall be provided by an electrical braking system without the use of a mechanical load brake. Hoisting shall take place only when the master switch is in a hoisting position. For all loads up to rated load, lowering shall take place only when the master switch is in the lowering position. Exception 1: If a class 152, 162, 172 or 92 resistor is specified, the rated load may descend with the

master switch in a slow speed hoisting position. Exception 2: a-c countertorque control may be provided for bucket or scrap handling magnet service, if specified.

On pendant push button operated cranes, the bridge and trolley speed without load shall not exceed 200 fpm unless otherwise specified. Each pendant station shall be equipped with an emergency trip circuit that will remove power to all motors by opening the main line contactor(s), with a means for resetting.

Radio control for cranes must be designed so that if the control signal for any crane motion becomes ineffective, that crane motion shall stop.

Signals received from any source other than the transmitter assigned to the crane shall not result in operation of any motion of the crane. All motions, except trolleys with drag brakes, shall be equipped with brakes that will set on loss of power to the brake. Continuous reception of a signal from the transmitter to the receiver shall be required to keep closed either a main power contactor or an electrically operated circuit breaker on the crane. Provisions shall be made to limit the distance from which control can be effective.

For a-c wound rotor motors, control for hoist, bridge and trolley drives shall be specified by a complete description on the OIS or by a functional specification using the description in the following Sections. The effect of primary and secondary impedance on motor torque and heating should be considered where the crane duty cycle is critical in motor selection. The types of a-c control are divided into two general categories, static and contactor types.

Static control uses static devices (thyristors, saturable reactors, magnetic amplifiers) to regulate the primary voltage or secondary impedance to develop the required speed and motoring or braking torque characteristics. The desired general equipment requirements should be specified from the following:

- (1) Contactor or static reversing devices
- (2) Primary voltage or secondary impedance control by static devices
- (3) Speed regulated or open loop control
- (4) Stepped or stepless speed control (Note: When specified for stepless control, the master switch may be provided with operating position detents)
- (5) With or without an eddy current load brake.

Contactor-type control refers to conventional magnetic contactor resistor controls.

The type (or types) of control required for each motion, either by complete description or by reference to Sections 4.5.2 through 4.5.4, shall be specified on the OIS.

4.5.2 Constant Potential d-c Control (From Either a d-c Power Supply or an a-c to d-c Converter on the Crane).

4.5.2.1 Hoist. The control shall be of the reversing, dynamic braking lowering, contactor-resistor type for use with a series wound motor and series brake(s), and shall include a spring-closed emergency dynamic braking contactor providing self-excitation of the motor field in the lowering direction. (Exception: for peel elevate and similar applications with limited travel, a reversing, plugging type of control with permanent armature shunt in the lowering direction may be appropriate depending on machinery design.)

Contactor and power limit switch sizes shall be based on the 30-minute TENV motor rating.

The power limit switch required by Section 4.6 shall be directly connected in the motor and brake circuit. When tripped, this switch shall establish a self-excited dynamic braking circuit for the motor in the hoist direction.

A back-out circuit shall be established by simply placing the master switch in a lowering position, and the control shall prevent excessive lowering speed if a tripped power limit switch fails to reset.

On hoists powered by two motors, no provision need be made for single motor operation, except on hot metal cranes or if specifically requested on the OIS. On all hot metal cranes, and when single motor operation is specified, devices on the control panel shall make it possible to electrically isolate either motor, transfer all the series brake(s) to the power circuit of the other motor and continue operation for temporary emergency service.

4.5.2.2 Bridge and Trolley. The control shall be of the reversing contactor-resistor type with at least one step of plugging, unless two steps are specified on the OIS.

Contactor sizes shall be based on the 60-minute TENV motor rating, unless the OIS states that the 30-minute motor rating shall be used.

If there is a limit on the maximum acceptable no-load speed, that limit must be stated on the OIS.

If specified when two or more paralleled motors are used, provision shall be made at the control panel to permit isolating any motor to allow continued operation for temporary emergency service.

If specified, emergency dynamic braking on loss of power shall be provided with armature excitation of the motor fields and with spring-closed contactors connecting dynamic braking resistors to the motor armatures.

4.5.3 Adjustable Voltage d-c Control (From Either Motor-Generator Set or Static Power Supply).

4.5.3.1 Hoist. The control shall provide regulated hoisting and lowering, and may be either reversing-regenerative

or reversing-dynamic braking lowering, unless a definite preference is indicated on the OIS.

The desired no-load hoisting speed shall be specified as a percentage of rated full load hoist speed.

If the full load lowering speed is 150% or more of rated full load hoist speed, or if specified on the OIS, emergency dynamic braking to aid in brake stop shall be provided by a spring-closed contactor and self-excitation of the motor field.

The control shall include a protection circuit to ensure current flow in the motor armature circuit before the brake can be energized.

The power limit switch required by Section 4.6 shall be directly connected in the motor armature and brake coil circuits. Unless otherwise specified, the power limit switch shall establish a dynamic braking circuit when tripped. Placing the master switch in a lowering position shall establish a back-out circuit after ensuring that the polarity of the voltage applied to the motor armature is in the proper direction to obtain rotation in the lowering direction.

Motor field loss protection shall be provided.

The current rating of contactors and power limit switches shall be selected based on consideration of both normal and emergency operating conditions and shall not be less than the 60-minute motor rating.

4.5.3.2 Bridge and Trolley. The control may be either reversing-regenerative or reversing-nonregenerative, unless a definite preference is indicated on the OIS.

Either coasting or electrical braking shall be provided when the master switch is moved from a fast speed point to a slow speed point in the same direction of travel (or into the off position), as specified on the OIS. If coasting is provided, stopping shall be accomplished by moving the master switch into the reverse direction or by operation of brakes, as specified on the OIS. Unless otherwise specified, motor field loss protection shall be provided.

If specified when two or more paralleled motors are used, provision shall be made at the control panel to permit isolating any motor to allow continued operation for temporary emergency service. The current rating of contactors shall be selected based on consideration of both normal and emergency operating conditions and shall not be less than the 60-minute motor rating.

4.5.4 A-C Control.

4.5.4.1 Hoist.

4.5.4.1.1 General. All of the hoist controls in this Section shall include the following features:

- (1) The power circuit limit switch required by Section 4.6 shall directly interrupt two lines to the motor and one line of the brake power circuit.

If the fastest possible setting of the brake is specified, the power limit switch shall open the d-c circuit to the brake coil. A means for lowering out of the upper limit switch shall provide controlled lowering without high motor current and without permitting a speed in excess of the maximum permissible speed for the motor being used.

- (2) Control shall be designed so that during an unplanned single-phase condition it will not be possible to release the hoist brake, or a controlled lowering speed shall be provided (whichever is specified); however, the lowering speed shall not exceed 150% of the rated hoist speed.
- (3) If the full load lowering speed is over 150% of rated full load hoist speed (or on special application) it may be specified that emergency braking be provided to prevent free falling loads under condition of simultaneous power failure and holding brake failure. The emergency braking requirements may be met by providing two brakes when only one is required, or by increasing each brake rating to 150% of full load hoisting torque when two brakes are required.
- (4) Contactor and power limit switch sizes shall not be less than the motor rating (60-minute TENV unless otherwise required by the application or specification).

In addition to the general equipment requirements, the following should be considered:

4.5.4.1.2 Static Control. For static control one or more steps of increased secondary resistance may be required to reduce motor heating if prolonged operation at slow speed is required. Contactors or other means may be used to achieve this and may also be used to change secondary resistance when near full speed. Reduced speed control at light loads, electrical braking to slow hoisting and other special requirements, e.g., load floating, shall be specified.

4.5.4.1.3 Contactor-Type Control. Contactor-type control shall be nonregulated contactor-reversing, with stepped control by means of secondary contactors and resistance, and with Type 1 — d-c dynamic braking lowering; Type 2 — Eddy current load brake; Type 3 — counter-torque lowering.

Types 1 and 3 provide reduced speed lowering control for overhauling loads only. Type 3 is not recommended where slow speed control is required for more than one specified load. Type 2 is suggested where speed control is required for a wide range of loads.

4.5.4.2 Bridge and Trolley.

4.5.4.2.1 General. All of the bridge and trolley controls in this Section must include the following features:

- (1) Contactor sizes shall not be less than motor rating (60-minute TENV unless otherwise required by the application or specification)
- (2) Coasting is to be provided when the master switch is moved from a fast speed point to a slow speed or to the off position, unless otherwise specified on the OIS
- (3) If specified, when two or more motors are used, provision shall be made at the control panel to permit isolating any motor and to continue operation for temporary emergency service.

In addition to the general equipment requirements, the following should be considered:

4.5.4.2.2 Static Control. Special requirements for low or high breakaway torque or special inching requirements shall be specified; plugging may require one or more steps of increased secondary resistance to limit motor heating and to develop the desired plugging torque when the specified control is either the primary or secondary impedance type; some secondary impedance regulating controls do not produce braking torque at speeds above synchronous speed without plugging the motor.

4.5.4.2.3 Contactor Control. Contactor-type control shall be nonregulated, contactor-reversing, with stepped control and plugging protection by means of secondary contactors and resistance.

4.6 Hoist Power Limit Switch.

Each hoist motor shall be equipped with a motor circuit power limit switch sized in accordance with Sections 4.5.2.1, 4.5.3.1 and 4.5.4.1.1 and connected directly in the motor and brake coil circuits as described in Section 4.5 for the type of control being used. The limit switch shall be located above the trolley deck so as to be easily accessible for inspection and, if possible, so it will be operated by the hook block or load beam in such a manner that no sheave wheels are necessary. If sheaves must be used, the pitch diameter shall not be less than 18 times the rope diameter. Cables should be guided through a hole in both ends of the sheave guard. Sheave bearings shall be the antifriction type and designed to exclude dirt (see Section 3.12).

A weight directly connected to the limit switch, with suitable guides acting on the idler cables, shall be used so that twisting cannot occur.

Cable guides shall have replaceable guide blocks of suitable materials to minimize wear on the cable.

If specified on the OIS, an arrangement using a free-swinging weighted beam hinged on one end and having the other end attached to the limit switch operating cable shall be used to operate the limit switch. The trip bar shall be designed so that the cables cannot jump out around the end of the trip bar,

which permits the hook to raise outside the trip bar. The trip bar shall also be designed so that no movement of the hoist and trolley can cause the trip bar to be jammed against any part of the crane structure.

The actuating mechanism of the limit switch shall be located so that it will trip the limit switch (under all conditions of hoist load and hoist speed) in sufficient time to prevent contact of upper and lower blocks.

4.7 Disconnecting Devices.

Each crane shall be provided with a main disconnecting device of the enclosed type in accordance with the National Electrical Code. Provisions shall be made for locking in the open position, with space for three safety locks. The 8-hr rating of the device shall be no less than 50% of the combined short time ampere rating of the motors, nor less than 75% of the short time ampere rating of the motors applied for any single crane motion. For this summation, in no case shall the motor ampere ratings used be less than 133% of the 60-minute rating for constant potential d-c hoist and 100% of the 60-minute rating for all other motors.

Devices of ampacity greater than 600 amps shall be of the bolted lock-type switch, a circuit breaker or a manual magnetic disconnect. Fuses, when specified on the OIS, shall be sized to provide short circuit protection for the cables and equipment on the load side of the device. This device shall be located on the bridge footwalk at a point as near as possible to the main collectors.

A second disconnecting device (or means for operating the disconnecting device on the footwalk) shall be provided in the cab as specified on the OIS.

Individual fused safety switches (or when specified, circuit breakers) shall be provided for auxiliary electrical equipment such as:

- Crane lights
- Electric heaters and ventilating units
- Plug outlets
- Signal lights
- The primary of the transformer supplying power to auxiliary circuits on a-c cranes
- Special devices, when applicable, such as sanders, motor operated buckets, turning devices, etc., as specified.

A magnet power disconnect shall be provided as specified in Section 4.8.2.

Branch circuit protection shall be in accordance with Article 610 of the National Electrical Code.

4.8 Wiring.

4.8.1 General. Wiring shall conform to AISE Technical Report No. 8, except as modified by the following:

- (1) Wires shall be installed in raceways which shall be continuous to switch boxes, junction boxes or connection terminals. Conduits smaller than $\frac{3}{4}$ in. shall not be used. Short lengths of open insulated conductors are permitted at contact conductors, AISE Technical Report No. 1 D-C Motors, power limit switches, resistors, reactors, and similar equipment, unless prohibited by the OIS.
- (2) Short lengths of flexible steel conduit with protective jacketing may be used to make connections to control devices, such as master switches and control limit switches or equipment subject to vibration, and where specifically approved by the purchaser. All flexible conduit fittings shall be inside threaded cone-type or equal.
- (3) Cable trays may be used in place of raceways in desirable locations when specifically approved by the purchaser. Installation and materials must comply with the National Electric Code and other pertinent regulations.

Wiring requirements shall be specified on the OIS, either by a complete description or by reference to Sections 4.8.1 through 4.8.6.

4.8.2 Conduits. All conduits shall be rigidly attached to the crane to withstand vibration and shall have suitable insulated bushings at all conduit ends. Welding of conduit to structural members shall not be permitted. Conduit supports however, may be welded to structural members except the critical tension members.

When conduits are used, the following shall apply:

- (1) Each motor shall be wired independently in separate conduits without common returns
- (2) Except as otherwise allowed by AISE Technical Report No. 8, a-c wound rotor motor circuits shall have primary leads in one conduit and secondary leads in another conduit.
- (3) Except as otherwise allowed by AISE Technical Report No. 8, power, control and shunt field leads shall be in separate conduits.

4.8.3 Standard Cab on Bridge Crane. The following standard method of wiring shall be used:

- (1) From main collector shoes, the wiring shall extend directly to the main disconnecting device mounted on the footwalk
- (2) When a second disconnecting device is used, wiring shall extend directly from the first device to the second
- (3) From the second disconnecting device (or the main disconnecting device when only one is used) branch circuits shall extend to control panels for hoist, bridge and trolley motions

- (4) The disconnecting devices for magnets and auxiliary functions such as lights and heaters, shall be connected between the main and second disconnecting devices when two are specified. When only one main disconnecting device is specified, the magnet and auxiliary function disconnecting devices shall be connected to the line side of the main disconnecting device.

4.8.4 Outlets. When specified, outlets of type and quantity approved by the purchaser for plug receptacles are to be furnished.

4.8.5 Raceways. A complete shop-assembled raceway system shall be furnished for the crane. Where disassembly is necessary to permit shipment, the components of the system shall be properly matchmarked to permit ease of field erection. Where any portion of a raceway run must be disconnected or dismantled to permit shipment, the wire shall not be pulled through during shop assembly. Such wire shall be cut to approximate length and bound in coils marked for the circuit for which it applies.

4.8.6 Pendants. Pendant stations shall be grounded to the crane structure and shall be supported in a manner that protects the electrical conductors from strain.

4.9 Magnet Cable Reel.

On cranes where a magnet cable reel or space for mounting a reel is specified, it shall be located so that the magnet cable will not foul the hoisting cable. Use of sheaves should be avoided, if possible.

If the cable reel is of the type driven by gears from hoist shafting or from extension of the drum shaft, the surface speed of the reel shall be the same as the hook speed. A loop shall be provided in the magnet cable to allow for slack. If specified, this type of reel shall be provided with a disconnect clutch when the magnet is not in use.

Weather protection shall be provided for magnet cable collector rings on cranes for outdoor service.

4.10 Lighting.

All crane cabs, control cabinets and control houses shall be provided with adequate lighting for maintenance and serviceability.

On the bridge structure of each crane, lighting fixtures shall be provided as specified on the OIS. Lighting fixtures

shall be mounted on shock absorbers and installed so they can be serviced from bridge footwalks.

4.11 Signal Lights.

Each crane shall be equipped with signal lights.

The number, location, color and connections shall be as specified on the OIS.

4.12 Acceleration Rates — Bridge and Trolley.

4.12.1 Maximum Rates vs Percent Driven Wheels. Since the wheels must transmit all acceleration forces to the crane or trolley, consideration of the percent driven wheels should be given in selecting the acceleration rate to prevent wheel skidding. Nominal practical limits are as listed in Table 21. The maximum acceleration rates are for full-load conditions. If wheel skidding cannot be avoided for no-load conditions, it shall be brought to the attention of the purchaser for resolution and the maximum full load acceleration rates reduced accordingly.

Similarly, the maximum allowable acceleration for type A-4 bridge drives should be reduced from that shown above due to the effect of the trolley position on wheel loads.

Note: These maximum acceleration rates are based on 20% adhesion between wheel and rail and on a ratio of peak torque to average torque during acceleration of 1.33. For control having a ratio other than 1.33, the maximum acceleration rate should be adjusted accordingly.

4.12.2 Acceleration Rate vs Acceleration Time. The specified acceleration rate for d-c constant potential series motor drive occurs while on resistors. The average acceleration rate for a-c drives and for adjustable voltage d-c drives remains near its specified value up to 100% of rated speed and therefore may be less than for a comparable constant potential d-c series wound motor drive.

To gain quantitative perspective for acceleration, Fig. 40 is given as an aid in relating speeds and acceleration rate into terms of time and distance. Note that the acceleration rate is the average (equivalent) rate to 100% speed for a-c and adjustable voltage d-c drives, but for series motors, the speed in fpm must be determined at the motor rpm attained at the end of acceleration on the resistor.

For example, an a-c or adjustable voltage bridge rated at 360 fpm with a loaded acceleration rate of 1 fps^2 will ac-

Table 21 — Maximum Acceleration Rates

Percent driven wheels	100	50	33 1/3	25	16 2/3
Maximum average acceleration (full load), fps^2	4.8	2.4	1.6	1.2	0.8

celerate from zero to 360 fpm in 6 seconds, during which time it will travel 18 ft. If a series motor bridge drive rated at 600 fpm has a speed of 360 fpm at the end of acceleration on the resistor, an acceleration rate of 1 fps² will still result in traveling 18 ft in the first 6 seconds; however, the bridge will continue to accelerate up to 600 fpm if space permits.

Since decelerating capability is related to acceleration rate, consideration should be given to specifying an acceleration rate and percent driven wheels for high speed cranes or trolleys which will insure adequate stopping capabilities.

4.12.3 Acceleration Factors

4.12.3.1 General. Typical acceleration rates on the resistor for series motors with constant-potential d-c control are listed in Table 22 and typical acceleration rates for either a-c or adjustable voltage d-c shunt motor drives are listed in Table 23.

The K_a factors in Figs. 36 and 37 of this Report are similar to those in the 1949 edition of AISE Specifications for Electric Overhead Traveling Cranes for Steel Mill Service with the values shown for the commonly used 15 and 24 lb/ton friction to eliminate the necessity of interpolating between curves.

Since these factors are based on several approximations, the derivation of the factors is described below, thereby allowing the user of this Report to calculate the factors more

accurately for those drives where the approximations are not reasonably accurate.

The motor hp required during acceleration consists of two components to:

- (1) Overcome the steady-state running friction
- (2) Provide power for acceleration.

The following equation applies except for the hp required to accelerate the motor and other rotating parts:

$$hp = \left\{ \frac{2000 W_t v}{33,000} \right\} \left\{ \frac{n_r}{n_f} \right\} \left\{ \frac{f}{2000} + \frac{a}{32.2 E} \right\} \quad (\text{Eq 83})$$

Where:

a = Acceleration (up to V_{res} in Table 22 or up to the free-running speed in Table 23), fps²

E = Mechanical efficiency of gears for travel drive

f = Rolling friction (draw bar pull), lb/ton

v = Velocity (V_{res} in Table 22 is the fps corresponding to n_r), fpm

W_t = Weight of trolley plus rated load, ton, for trolley acceleration

and

W_b = Weight of bridge plus trolley plus rated load, ton, for bridge acceleration

Table 22 — Typical Acceleration Rates for Series Motors with CP, d-c Control

Free-running Speed		Slow $\frac{n_r}{t_n} = 0.7$			Medium $\frac{n_r}{t_n} = 0.6$			Fast $\frac{n_r}{t_n} = 0.5$		
fpm	fps	a fps ²	V_{res} fps	t_a sec	a fps ²	V_{res} fps	t_a sec	a fps ²	V_{res} fps	t_a sec
60	1	0.4	0.7	1.75	0.6	0.6	1.00	0.8	0.5	0.63
120	2	0.5	1.4	2.80	0.7	1.2	1.71	0.9	1.0	1.11
180	3	0.6	2.1	3.50	0.8	1.8	2.25	1.0	1.5	1.50
240	4	0.7	2.8	4.00	0.9	2.4	2.67	1.1	2.0	1.82
300	5	0.8	3.5	4.38	1.0	3.0	3.00	1.2	2.5	2.08
360	6	0.9	4.2	4.67	1.1	3.6	3.27	1.3	3.0	2.31
420	7	1.0	4.9	4.90	1.2	4.2	3.50	1.4	3.5	2.50
480	8	1.1	5.6	5.09	1.3	4.8	3.69	1.6	4.0	2.50
540	9	1.2	6.3	5.25	1.4	5.4	3.86	1.8	4.5	2.50
600	10	1.3	7.0	5.38	1.5	6.0	4.00	2.0	5.0	2.50

Where:

a = Acceleration rate, fps² (up to V_{res} in Table 22 or up to the free-running speed in Table 23)

V_{res} = Velocity, fps, attained on the resistor; corresponds to n_r in Table 22

t_a = Time, seconds to accelerate from 0 speed to V_{res} in Table 22 or up to the free-running speed in Table 23)

To determine the motor rating necessary to provide the above horsepower during acceleration up to rated motor speed, the general case is covered by

$$\text{Motor rated hp} \geq \frac{2,000 W_t v n_r}{33,000 T_a n_f} \times \left\{ \frac{f}{2000} + \frac{a}{32.2 E} + \frac{a W k_r^2}{32.2 W k_L^2} \right\} \quad (\text{Eq 84})$$

Where:

n_f = Motor rpm corresponding to v , see Section 4.4.4

n_r = Motor rated rpm

T_a = Average per unit motor torque during acceleration

v = Velocity at free-running hp, fpm

$W k_L^2$ = Equivalent inertia of crane (trolley) and load at

$$\text{the motor shaft, lb-ft}^2 = 2000 W_t \left\{ \frac{v}{2 \pi n_f} \right\}^2$$

$W k_r^2$ = Inertia of rotating parts, including motor, brake, coupling, gears, shafts and wheels. (To be exact, efficiency should be taken into consideration for acceleration of the wheels but the difference is usually insignificant; so the equivalent inertia of the wheels is combined with the other rotating parts), lb-ft²

4.12.3.2 Series Wound Constant Potential d-c Drives.

The following approximations are made:

- (1) During acceleration on resistors, the control will cause the motor to deliver an average of 200% of rated torque and $T_a = 2.0$
- (2) At the end of acceleration on the resistor the velocity is 66% of the rated velocity and $n_r/n_f = 0.66$

- (3) The power required to accelerate the motor, brake, gears, shaft and wheels can be approximated if the first acceleration term, a , in Eq 84 is divided by 0.90 and the second a term is dropped

- (4) The mechanical efficiency is 95%

With these approximations the minimum 60-minute motor hp equation becomes:

$$\text{hp} = \left\{ \frac{(2000 - W_t)(0.66 - v)}{33,000 \times 2} \right\} \times \left\{ \frac{f}{2000} + \frac{a}{32.2 \times 0.95 \times 0.90} \right\} \quad (\text{Eq 85})$$

For selected values of f and a , $\text{hp} = K_a W_t v$, which permits calculating the curves of K_a vs lb/ton friction at typical values of a as shown in Fig. 35.

4.12.3.3 A-c Motors and Adjustable Voltage d-c Drives Without Motor Field Weakening. The following approximations are made:

- (1) During acceleration the control will cause the motor to deliver an average of 170% of rated torque and $T_a = 1.7$
- (2) By definition, the acceleration applies to rated motor rpm, n_r . (Note: In the full speed position, the speed/torque curve for drives of this type are relatively flat in the no-load to full-load range, as compared to the steep speed/torque curve of a series wound d-c motor; therefore the ratio of $n_r/n_f = 1.0$)
- (3) The power required to accelerate the motor, brake, gears, shaft and wheels can be approximated if the first acceleration term, a , in Eq 84 is divided by 0.9 and the second a term is dropped
- (4) The mechanical efficiency is 95%.

Table 23 — Typical Acceleration Rates for a-c or Adjustable Voltage d-c Shunt Motor Drives

Free-running Speed		$n_r/n_f = 1.0$							
		Slow		Medium		Fast			
fpm	fps	$\frac{a}{\text{fps}^2}$	$\frac{t_a}{\text{sec}}$	$\frac{a}{\text{fps}^2}$	$\frac{t_a}{\text{sec}}$	$\frac{a}{\text{fps}^2}$	$\frac{t_a}{\text{sec}}$		
60	1.0	0.3	3.33	0.4	2.50	0.6	1.67		
120	2.0	0.4	5.00	0.5	4.00	0.7	2.86		
180	3.0	0.5	6.00	0.6	5.00	0.8	3.75		
240	4.0	0.6	6.67	0.7	5.71	0.9	4.44		
300	5.0	0.7	7.14	0.8	6.25	1.0	5.00		
360	6.0	0.8	7.50	0.9	6.67	1.1	5.45		
420	7.0	0.9	7.78	1.0	7.00	1.2	5.83		
480	8.0	1.0	8.00	1.1	7.27	1.3	6.15		
540	9.0	1.1	8.18	1.2	7.50	1.4	6.43		
600	10.0	1.2	8.33	1.3	7.69	1.5	6.67		

On the basis of these approximations the minimum 60-minute motor hp equation becomes:

$$hp = \frac{(2000 W_f)(v)}{33,000 \times 1.7} \times \left\{ \frac{f}{2000} + \frac{a}{32.2 \times 0.95 \times 0.90} \right\} \quad (\text{Eq 86})$$

or minimum 60-minute motor hp = $K_a W_f v$, with the value of K_a for the selected lb/ton friction and fps² acceleration rate being obtained from Fig. 36.

4.12.4 D-c Travel Drive Gear Ratios, Series Motors.

Because of the differential between available motor ratings, frequently the motor used is larger than needed for the acceleration rate and service factors required.

When this happens, the motor rpm associated with the free-running hp is high (on the steep portion of the motor curve) and the required gear ratio becomes numerically large.

In this area, small variations in friction, voltage or motor characteristics affect the motor rpm appreciably so that the specified speed may not be realized.

It is practical under these conditions to increase the drive speed to ensure realization of the specified speed and to avoid large gear ratios when they are not really necessary to give

the motor adequate leverage for acceleration. To avoid indiscriminate adjustment from the theoretically correct gear ratio as defined in Section 4.4.4, the following procedure shall be used:

- (1) Determine the actual service factor, K_s

$$K_s = \frac{hp(60\text{-minute})}{K_a W_f v K_t} \quad (\text{Eq 87})$$

- (2) Determine the maximum allowable speed, $v_{\max}$

$$v_{\max} = \frac{\text{actual } K_s}{\text{required } K_s} \times \text{specified speed} \quad (\text{Eq 88})$$

- (3) Recalculate the free-running hp based on the new maximum speed and recalculate the gear ratio based on the new free-running hp. This smaller gear ratio should be considered the minimum value permissible to obtain the required service factor.

Should this new higher full load speed be objectionable for the crane's operations, it can be effectively reduced by using a permanent armature shunt resistor.

Consult the drive manufacturer to determine how much speed reduction is practical without causing excessive motor heating.

Table 24 — Duty Cycle Form

Duty Cycle Step No. ¹	Type of Motion ²	Direction ³	Average Acceleration/Deceleration Rate ⁴ , fps ²	Speed v, fpm or fps ⁵	Rated Load on Hook, %	Distance, ft	Time seconds
1							
2							
3							
4							
5							
6							
7							
8							
9							
10							
11							
12							
13							
14							
15							

- 1 — Show a complete critical duty cycle. Include every step to completion
- 2 — Specify 'acceleration', 'run', 'deceleration' or 'off' for each step
- 3 — Up or down for hoist; forward or reverse for travels
- 4 — Average rate of speed change, fps²; zero for run or off
- 5 — Speed of motion (or speed at end of step if acceleration or deceleration)

SYMBOLS — ELECTRICAL

a	Acceleration, fps^2	R_a	Mechanical advantage of reeving system
D	Pitch diameter of drum for hoist or wheel tread diameter for traverse drives, in.	R_{pu}	Resistance, per unit
E	Mechanical efficiency of gears for travel drives	Res_{pu}	Total resistance in a-c motor secondary (including internal), per unit
E_c	Combined efficiency of gears and sheaves for hoist drives	R_r	Rope reduction
f	Rolling friction factor (draw bar pull) for travel drives, lb/ton	S_{pu}	a-c motor slip, per unit
GR	Gear reduction ratio	T	Torque, lb-ft
I_{pu}	Motor current, per unit	T_a	Average motor torque during acceleration, per unit
K_a	Acceleration factor	T_{pu}	Motor torque, per unit
K_s	Service factor	t	Time, seconds
K_t	Temperature factor	t_a	Time to accelerate, seconds
K_v	Voltage correction factor	v	Velocity, fps or fpm
m	Number of rotating sheaves	v_{res}	Velocity attained on resistor, fps
n	Number of gear reductions	v_{max}	Maximum allowable speed, fps or fpm
n	Motor rpm	Wk^2	Rotary inertia, lb-ft^2
n_f	Motor rpm corresponding to the steady-state hp of a hoist drive or free-running hp of a travel drive (not including acceleration hp). See Section 4.4.4 for complete definition	Wk_r^2	Inertia of rotating parts, lb-ft^2
n_r	Motor rated rpm	Wk_L^2	Equivalent inertia of crane (trolley) and load at the motor shaft, lb-ft^2
pu	Per unit = $\%/100$	W_L	Weight of lifted load including weight of hook block, lb
		W_t	Weight of crane or trolley and rated load, ton; and Weight of bridge plus trolley plus rated load, ton, for bridge acceleration

COMMENTARY — ELECTRICAL

It is the purpose of this commentary to amplify, supplement and explain the basis and application of portions of this Report not covered elsewhere. The comments herein are not part of the Report but are added as supplementary information. Numerals in parentheses refer to the section number in the text of the Report.

The basic principles of the Electrical Section have not been changed from the 1969 Tentative AISE Standard No. 6. There are still four service classes to assist in the selection of motor ratings for each motion when the purchaser cannot establish a definite duty cycle. In general, this procedure has helped to avoid errors in selecting a motor larger than necessary for light duty or a motor that does not have sufficient torque and thermal capacity for severe duty. As before, the OIS must identify the service class to be used for each motion.

The mechanical and structural sections of this report divide crane designs into four crane service classes based on total life load cycles.

Electrical equipment, on the other hand, must be selected based on the worst duty encountered for any one-hour period. Therefore, electrical equipment must be selected independently of mechanical or structural classes.

When considering performance and maintenance trade-offs in selection of electrical equipment, the following additional points should be considered.

Entire cranes or certain crane motions, although operating infrequently, experience failures in their electrical systems primarily from or caused by disuse, not use. Insulating materials are subject to aging, reduction of their dielectric and mechanical properties and ultimate failure. Aging occurs with the passing of time, idle or not.

Metallic components of the electrical system, contacts, connections, etc. are subject to galvanic corrosion and attack by airborne chemical and corrosive particles. Often, infrequently used equipment is affected more severely.

This suggests that the same, if not greater care, must be exercised when selecting the electrical system for mill cranes used infrequently, which appear to be candidates for downsizing.

The owner should determine realistic performance requirements, capacities that could actually be used, speeds and accelerations that are no higher than the actual task requires, in addition to a life expectancy which recognizes today's frequent operational changes and anticipates the continuation of technological progress.

This commentary is not intended to cover every change; however, persons familiar with the 1969 Tentative Standard

may find the following comments concerning specific parts of the 1991 Technical Report to be helpful.

HOIST BRAKES (4.1.1). Treatment of calculation of required torque of brakes on hot metal hoists driven by two motors has been expanded.

D-C AND A-C MOTORS (4.4.1 and 4.4.2). If an alternate motor(s), other than AISE Technical Report 1 (d-c) is specified, the user is advised that many sections of this document (AISE Technical Report No. 6) and other referenced documents (eg. AISE Technical Report No. 8) are based on the use of mill motors (AISE Technical Report 1) and that due care must be exercised in such related matters as torque characteristics, wire sizing, control characteristics, etc.

Non-mill motors (or d-c mill motors of the '600' series) may be specified for various reasons such as lack of commercial availability, very light duty requirements, or very mild environmental conditions. Although the selection of an alternate motor(s) may be desirable and such motors may be suitable for use on steel mill cranes, their selection should be verified through prudent design methods and, at minimum, such motors should be examined for torque sufficiency and thermal (duty) adequacy. As a general rule, any such motor(s) ought to conform to NEMA Standard MG 1-18.5 for a-c motors or, in the case of d-c motors, have similar construction features and adequate commutation capability.

MOTORS, GENERAL (4.4.3.1). A second paragraph and note have been added to cover the selection of motors for use in a prolonged ambient temperature above 40°C; the omission of the ambient correction factor up to 65°C under certain conditions is discussed. Table 15 differs from the 1969 Table E4.C.II by making the ambient factors for a-c motors the same as for d-c motors and by adding the factors to be used at 45°C and 55°C.

Table 16 and the paragraph preceding that table provide a voltage correction factor to be used in the selection of a-c hoist motors when specified on the OIS.

Following Table 16 is an introduction to service factors (Tables 17 and 18) which points out that the factors may be conservative due to the many variables in different motors. Problems of wheel slippage and unreasonably high gear ratios on travel drives might be eliminated by assuming a smaller service factor than given in the tables, if the smaller motor resulting from that assumption is confirmed by a duty cycle analysis using the maximum percent time-on and maximum cycles/hr associated with the specified service class.

The service factors summarized in Tables 17 and 18 have been consolidated from pages ED-14 through ED-29 of the 1969 Tentative Standard, making them easier to locate and evaluate.

MOTORS, HOIST (4.4.3.2). The efficiency of gears with antifriction bearings has been increased from 0.95 to 0.97 per reduction and Table 19 differs from the 1969 Table E.4.C.1.a.I as a result.

The selection of an a-c motor for a drive that has permanent secondary resistance during full speed hoisting is covered in more detail with examples at different service factors.

MOTORS, BRIDGE AND TROLLEY (4.4.3.3). The acceleration factors for series motor travel drives shown in Fig. 35 were calculated by Eq 85.

Although the assumptions used in the development of that equation are the same as discussed in the Appendix of the 1949 Standard (Fig. E.4.C.2.1 in the 1969 Tentative Standard was a direct copy of Fig. 43.1 in the 1949 Standard), the calculated values are different. At 15 lb/ton, the new acceleration factors are lower than those in the previous Standards if the acceleration rate is slower than 1.5 fps^2 and the new factors are higher than those in the previous Standards above 1.5 fps^2 . At 24 lb/ton, the new factors are also lower than the previous Standards below 2 fps^2 and higher above 2 fps^2 . This would encourage the use of slower acceleration rates which introduces the possibility of using a smaller motor with a lower gear ratio and less likelihood of a load swing and wheel problems. Of course, the acceleration rates must be high enough to permit the crane to perform its duty satisfactorily.

Fig. 36 is the same as Fig. E.4.C.2.2 in the 1969 Tentative Standard except for a minor correction. The acceleration factors for a-c motors and for adjustable voltage d-c drives without field weakening have been calculated using Eq 86. The resulting values are approximately the same.

MOTOR SELECTION BY DUTY CYCLE (4.4.3.4).

Example 2 is essentially the same as Example 2 in the 1969 Tentative Standard except that the acceleration rate was increased from 0.7 to 0.9 fps^2 in order to maintain the same acceleration factor. The resulting summary changes slightly due to the acceleration rate change. The note added to 'Acceleration loaded' points out that the type and adjustment of the accelerating relays may result in attaining more than rated motor speed on the resistor.

In Example 3 more steps have been added to provide a better illustration of the reduction in motor heating achieved by increasing the wound rotor motor secondary resistance during acceleration and plugging.

A-C WOUND ROTOR MOTOR CONTROL, GENERAL REQUIREMENTS (Included in 4.5.1). The a-c wound rotor motor control section has been rearranged, but there are no major changes from Sections E.5.N.5, .6, .7 and

.8 in the 1969 Tentative Standard. It is still required to provide either a complete description of the required control or a functional specification on the OIS.

DISCONNECTING DEVICES (4.7). The last sentence calls attention to the branch circuit protection requirements in Article 610 of the National Electrical Code.

WIRING (4.8). The word 'raceways' has been substituted for 'rigid metal conduits' based on the definition of raceways in the National Electrical Code. In addition, a sentence has been added to cover the normal practice of using short lengths of open insulated conductors for connections to specific pieces of equipment.

Cable trays used on cranes as conductor raceways have become a popular trend to facilitate ease of failure diagnosis and electrical maintenance on cranes. However, the dynamics of crane operation pose several interesting obstacles not to be overlooked. Induced vibration should be carefully considered when designing support systems and selection of tray material and conductor insulation. Overlapping or crossing of conductors in tray systems should be minimized. Dividing barriers should be used where necessary to separate and isolate conductors. Access should be provided, by walkway or platform, along the entire tray system to accommodate system maintainability.

Specific attention should be given to items such as conductor abrasion and fire barrier system.

ACCELERATION RATES (4.12). The supplier has been assigned the responsibility to check the possibility of wheel skidding under no-load conditions. If that possibility exists, it is to be brought to the attention of the purchaser to see if the specified full load acceleration rate can be reduced.

Two new tables, Table 22 and Table 23, have been added which, in combination with Fig. 40 (formerly Fig. A.E.1), should help the specification writer to select a practical acceleration rate for the work to be performed. It is hoped that this information will help to avoid the problems of wheel slippage and unreasonably high gear ratios for series motor travel drives.

A detailed discussion has been presented to show the equations and assumptions used in calculating the acceleration factors shown in Figs. 35 and 36. The general equation, Eq 84 makes it possible to calculate the minimum motor rating (series wound d-c, adjustable voltage d-c, or a-c) when the conditions differ from the assumptions used in calculating the acceleration factors.

APPENDIX A

CRANE OPERATING INTENSITY GUIDELINES and EXAMPLE OF CRANE OPERATING INTENSITY DATA AND CALCULATIONS

A1 Operating Intensity Guidelines.

As a means of determining the potential benefits to be gained from a crane duty classification system, the AISE conducted a survey in 1978 of major steel companies in the United States and Canada. These companies compiled a list of the loads lifted by 352 cranes. The data obtained showed that there is a wide range of actual crane usage on all of the different types of cranes used in the industry. By mathematically evaluating the load handling intensities, it was then possible to calculate the material handling duty of the cranes.

This method of crane duty classification evaluates the load carrying requirements for the overall crane service. This service applies to the main structural components of the crane, such as the trolley frame and the bridge girders.

Table A1 — Typical Steel Mill Crane Operating Intensities

Type of Crane	Duty Class														
	10 Years			20 Years			30 Years			40 Years			50 Years		
	MIN	MAX	AVE	MIN	MAX	AVE	MIN	MAX	AVE	MIN	MAX	AVE	MIN	MAX	AVE
Skull Cracker	D1	D2	D2	D1	D3	D2	D2	D3	D3	D2	D3	D3	D2	D4	D3
Charging															
BOF	D1	D2	D1	D1	D2	D2	D2	D3	D2	D2	D3	D2	D2	D3	D3
Electric Furnace	D1	D1	D1	D1	D2	D2	D2	D2	D2	D2	D2	D2	D2	D3	D2
Teeming	D1	D2	D1	D1	D2	D1	D1	D2	D2	D1	D3	D2	D1	D3	D2
Scrap Loading	D1	D4	D3	D1	D4	D3	D2	D4	D3	D2	D4	D4	D2	D4	D4
Mold Yard	D1	D3	D1	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2
Stripper	D1	D3	D2	D1	D3	D2	D2	D4	D3	D2	D4	D3	D2	D4	D3
Ingot Handling (Soaking Pit)	D1	D4	D2	D1	D4	D3	D1	D4	D3	D1	D4	D3	D1	D4	D3
Slab Handling	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D4
Mill Service															
Billet	D1	D4	D3	D2	D4	D3	D2	D4	D3	D2	D4	D3	D2	D4	D3
Hot Strip	D1	D3	D1	D1	D4	D1	D1	D4	D2	D1	D4	D2	D1	D4	D2
Cold Strip	D1	D3	D1	D1	D3	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2
Bar or Rod	D1	D3	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2
Coil Handling	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D2	D1	D4	D3
Roll Shop	D1	D3	D1	D1	D4	D1	D1	D4	D1	D1	D4	D1	D1	D4	D1
Product Handling or Shipping	D1	D3	D2	D1	D4	D2	D1	D4	D2	D1	D4	D3	D1	D4	D3

DUTY CLASS

D1
D2
D3
D4

CYCLE RANGE

Less than 100,000
100,000 to 500,000
500,000 to 2,000,000
Over 2,000,000

MATERIAL HANDLING DUTY

Light
Medium
Heavy
Severe

A2 Crane Operating Intensity Data and Calculations — Example

For this Example the record of a typical mill service crane in a modern hot strip mill was selected. This type of crane has a wider variety of loads and duties to handle than most cranes in steel mill service. Discussions with operating and maintenance supervisors produced the following basic data:

HOT STRIP MILL ROUGHING MILL AREA CRANE Capacity 50/35 tons, 100 ft span

ROLL CHANGING

Work rolls	50 tons/set 5 sets/week 4 lifts/set	= 20 lifts/week at 50 tons
Back-up rolls	30 tons/roll + 15-ton chucks + 5-ton spreader 4 rolls/week 4 lifts/roll	= 16 lifts/week at 50 tons
Back-up spacer rig	30 tons/lift 4 lifts/week	= 4 lifts/week at 30 tons
2-high mills		= 8 lifts/week at 40 tons

OFFALL SCRAP BARS (60% handled by this crane)

3 bars/turn	10 lifts/bar = 540 lifts/week × 0.6 = 324
Magnet weight	5,000 lb
Average bar weight	<u>5,000 lb</u>
Total lift	10,000 lb = 5 tons = 324 lifts/week at 5 tons

CARRY-AROUND SLABS

12/week	
Average slab weight	20 tons
Lifting device weight	15 tons
	= 12 lifts/week at 35 tons

STRAIGHTENING WEIGHT

= 15 lifts/week at 20 tons

CROP BUCKET 2/turn for this crane

Bucket weight	20 tons
Crop weight	5 tons
	25 ton
	= 36 lifts/week at 25 tons

MAINTENANCE LIFTS

= 800 lifts/week at 5 tons

The service class can then be determined from the following calculation which is given in tabular 50-year life desired.

$$N_{eq} = \sum \left(\frac{LL_i}{LL_{max}} \right)^3 n_i$$

i	LL_i	$\frac{LL_i}{LL_{max}}$	N_i	$N_i \left(\frac{LL_i}{LL_{max}} \right)^3$
1	50	1.0	93600	93600
2	40	0.8	20800	10650
3	35	0.7	31200	10702
4	30	0.6	10400	2246
5	25	0.5	93600	11700
6	20	0.4	39000	2496
7	5	0.1	2922400	2922

$$N_{eq} = \sum = 134316$$

$100,000 < N_{eq} < 500,000$ therefore Service Class 2

APPENDIX B

If a more refined component service classification than the four classes provided in Section 2.2.12.1 is desired, the following equation may be used to obtain the allowable stress range for any number of load cycles for the detail category given in Table 4 and shown in Figure 1.

$$F_{sr} = K_1 \left\{ \frac{K_2 \times 10^8}{N} \right\}^{\frac{1}{K_3}} \text{ but not less than } K_4$$

where:

- F_{sr} = The allowable stress range for the detail under consideration (ksi). Stress range is the algebraic difference between the maximum stress and the minimum stress.
- K_1 = A reduction factor to account for uncertainties in the assumed duty cycle. A value of 0.75 is suggested. Other values of K_1 may be used at the option of the owner. This extra conservatism is prudent because of the uncertainty in future activity and/or rated capacity. The economic ramification of his conservatism are minimal at the time of the design of a new crane. Note that if K_1 were chosen as 1.0, the allowable stress range values (F_{sr}) would be the same as tabulated in AASHTO, 13th Edition, Table 10.3.1A for redundant structures.
- K_2, K_3 = Constants given in Table B-5 which are dependent upon the category of the detail being considered.
- N = The desired design fatigue life in cycles of the detail being evaluated. N is the expected number of constant amplitude stress range excursions and is to be provided by the owner. If no desired fatigue life is specified, the designer should use the threshold values (K_4) as the allowable stress range (F_{sr}). For cumulative damage analysis of a varying amplitude load spectrum, an equivalent number of constant amplitude cycles can be calculated using the equation in Section 2.2.12.1.
- K_4 = The threshold value of F_{sr} given in Table B-5 for each stress category. When performing a cumulative damage evaluation, let $K_4 = 0$ unless the actual stress range for every load cycle is below the threshold (K_4). Cumulative damage tests have shown that if any cycles exceed the constant amplitude fatigue limit, then all cycles contribute to fatigue damage.

Table B-5

Stress Category	K_2	K_3	K_4
A	651	3.27	24
B	254	3.27	16
C	101	3.33	10, 12*
D	20.8	3.02	7
E	12.7	3.10	5
F	7980	5.86	8
E'	3.57	2.95	2.6

* For transverse stiffener welds on girder webs or flanges.

APPENDIX C

Sample Contract Paragraphs

Specification. The latest revision to AISE Technical Report No. 6, Electric Overhead Traveling Cranes for Steel Mill Service, shall be referred to in all contracts as such and shall form part of such contracts when so stated, whether attached to same or not. Data furnished by the bidder showing specific information with regard to the equipment to be furnished shall be known as the bidder's specification. Unless otherwise agreed to in writing, this Report shall take precedence over bidder's equipment specifications and shall be strictly adhered to, and no other specification or understandings will be considered. All design, fabrication and construction shall comply with this Report and with applicable local, state and federal regulations and codes.

Data issued by the owner stating sizes, special features and requirements shall be known as Owner's Information Sheets (OIS). Where the requirements of the OIS differ from this Report the former shall take precedence.

Manufacturers. The bidder shall submit the number of proposals called for on the OIS, stating therein the price (as specified on the questionnaire) for which the bidder will agree to furnish the work. The bidder shall furnish with each copy of the proposal:

- (1) Complete equipment specifications covering the work proposed
- (2) The data called for on the owner's questionnaire.

Warranty. No warranty of work by the bidder is implied nor shall be inferred from this Report. All applicable warranties shall be separately specified by the owner or the owner's agent.

Patents. All suits, actions, legal proceedings, claims, demands, damage costs, expenses or fees incident to any infringement or claimed infringement of any patent or patents in any way relating to the equipment specification are not covered in this Report and shall be separately covered by the owner or the owner's agent.

Changes or Additions. The owner reserves the right to make such changes in the specification or specific information as may be necessary after the contract is signed. All changes become the responsibility of the manufacturer as far as feasibility and workability are concerned. Any difference in cost resulting from such changes shall be agreed upon before the work is begun.

Erection (Including Assembly, Installation and Starting). All disassembled parts of the crane shall be match-marked by the crane manufacturer and shipped in sections as required for erection. The crane shall be erected by the crane manufacturer, the owner or by another party as specified on the OIS. The crane manufacturer shall pay the cost of all fitting or changes necessary for proper functioning of the work for which the manufacturer is responsible. Erection drawings shall be furnished by the crane manufacturer.

Responsibility for the unloading, storing and protection of all equipment and materials and for demurrage shall be as specified in the contract between the owner and crane erector.

When erection is to be done with supervision of the crane manufacturer, the manufacturer shall furnish the owner, on terms of agreed payment, a qualified person to administer the complete erection of the crane. The crane erector will furnish all other required labor and tools. The crane manufacturer's supervisor shall have duties, responsibilities and reporting procedures as outlined on the OIS.

When the crane is to be started under the crane manufacturer's supervision, the manufacturer shall furnish the owner, on terms of agreed payment, a qualified person to prescribe start-up procedures.

After completion of the work, the erection contractor shall promptly remove from the premises all material and tools brought in for the work, not pertaining to the completed job and leave the premises in an acceptable clean condition.

Tests and Acceptance. Tests shall be made as specified on the OIS; otherwise, the manufacturer's standard tests shall be made. In any case, the owner shall be notified sufficiently in advance, so that a representative may witness all tests.

Acceptance shall be subject to compliance with this Report and Owner's Information Sheets, to be determined by inspection after delivery, by results of tests required above and upon the approval of the owner's representative.

Safety Devices. All machinery or equipment to be provided under this Report must be furnished by the crane manufacturer with all safety devices and clearances to comply with the laws of the state and municipality in which it will be installed, and if stated on the OIS, the owner's safety requirements.

Clearances. Clearance between any part of the crane, building column, roof chord or other stationary structure shall be not less than that on the sketch accompanying the OIS. Accuracy of these sketches shall be responsibility of the owner. Minimum recommended design clearances are 3 in. overhead and 2 in. laterally, with the crane centered on the runway and with no load on the trolley.

AISE TECHNICAL REPORT NO. 6						OPERATING INTENSITY EVALUATION FORM FATIGUE COMPONENT CLASSIFICATION							Form Number 2.00	
Evaluated By		Inq. No.		Prop. No.		Type of Crane		Capacities and Span		No. o' Yrs. Expected Service Life for:		Sheel Number		
Company:		Alt. No.		Rev. No.				Number of Working Days/Year		Support Structures		Gearing		
Plant:		Order No.		Job No.		Component				Bearings		Other Machinery		
Name:		Date:								Years		Years		
										Years		Years		
	1	2	3	4	5	6	7	8	9					
Load Level Ref.	Stress Amplitude Per Stress Level Units In []	Total Number of Stress Cycles Per Stress Level Units In []	Average Length of Motion Units In []	Total Number of Design Load Cycles of Stress Per Load Level ni x mi x Ru	Stress Ratio $\frac{S_i}{S_{i \max}}$	Cycle Ratio $\frac{n_i}{\sum n_i}$	$\left(\frac{S_i}{S_{i \max}} \right)^K$ K =	Cumulative Stress Effect Per Stress Level (Rn x Rc)	Percent of Total Stress Effect					
No.	Si	ni	mi	Rn	Ra	Rn	Rc	Kp	%					
Max.														
1														
2														
3														
4														
5														
6														
7														
8														
9														
10														
11														
12														
13														
14														
15														
16														
17														
18														
19														
20														
Σ						1.000								
Utilization Factor = Ku = $\Sigma N_i/N_e$ =										Stress Level Class =				
Stress Factor = KDS = Ku x ΣK_p =										x =				

OWNER'S INFORMATION SHEETS (OIS)

Company _____

Works _____

Located at _____

Specification No. _____

Dated _____

for

_____ - TON ELECTRIC OVERHEAD TRAVELING CRANE

(This information is to be furnished to the bidder by the purchaser)

The following specific information, together with AISE Technical Report No. 6 for Electric Overhead Traveling Cranes for Steel Mill Service, dated _____, shall form the complete specification of the number noted above.

Contractor shall furnish _____

(1.2)

as covered by these specifications.

Crane to be delivered FOB _____

Complete wiring of crane, including furnishing of switches, panels, lighting fixtures, etc., shall be done by _____

All motors, controls for motors, hoist limit switches and magnetic brakes shall be furnished by _____

(If furnished by the purchaser, they will be delivered FOB contractor's plant for erection by the contractor on the crane.)

Number of sets of prints, etc., to be furnished by contractor

1. Specifications _____

2. Bills of material _____

3. Are prints or tracings required? _____

4. General arrangements, and

(a) Details of such parts as are subject to wear and will require replacement _____

(b) All details _____

Cranes covered by these specifications will be used for _____

Number of proposals to be submitted by the contractor _____

(1.3)

GENERAL DETAILS

1. Building clearance, location of cage and bridge runway conductors are shown on accompanying drawing No. _____
2. Speeds (with maximum working loads)
Main hoist _____ fpm
Auxiliary hoist _____ fmp
Bridge travel _____ fpm
Trolley travel _____ fpm
3. Distance top of runway rail to floor line _____ ft _____ in.
4. Total lift of hook above the floor line (exclusive of travel required to operate the limit switch)
Main hoist _____ ft _____ in.
Auxiliary hoist _____ ft _____ in.
5. Travel of hook below the floor line
Main hoist _____ ft _____ in.
Auxiliary hoist _____ ft _____ in.
6. Span of crane, centerline-to-centerline of bridge runway rails
_____ ft _____ in.
7. Minimum distance, centerline of main hook to centerline of bridge runway rails
Cab end _____ ft _____ in.
End opposite cab _____ ft _____ in.
8. Minimum distance, centerline of main to centerline of auxiliary hook
_____ ft _____ in.
9. Is repair structure over trolley to be furnished? Yes _____ No _____
10. Are track sanders to be furnished? Yes _____ No _____
11. Type of antifriction bearings to be furnished on motors _____

12. Power for operating the crane will be _____ volts _____ phase _____ cycles.

SPECIFICATION DETAILS

The section numbers with each of the following items indicate the corresponding section numbers of the AISE technical report and is made a part of this specification. If this item is not applicable to the crane under construction, the item should be marked "not applicable."

GENERAL

Section 1

Is latticed construction desired? Yes _____ No _____

Section 1.7

Crane to be erected by _____

Contractor to furnish supervision for erection by others. Yes _____ No _____

If yes, contractor's supervisor shall have the following specific duties, responsibilities and reporting procedures. _____

Section 1.8

Special tests required _____

Load tests required _____

Section 1.9

Stress relieving of weldments can be done by the following alternate method _____

Section 1.12

All parts inaccessible after assembling shall be painted before assembling. Yes _____ No _____

Color and quality of paint

First coat _____

Second coat _____

Section 1.13

The following special safety requirement must be met _____

STRUCTURAL

Section 2.1.3

Alternative high-strength steels required _____

Special welding procedures required _____

Maximum working loads

Main hoist _____ 1st trolley _____ lb

2nd trolley _____ lb

Auxiliary hoist _____ lb

Condition of runway is _____

Section 2.2.7

Are stress sheets required? Yes _____ No _____

Section 2.2.8

Design platform loads (other than 75 lb/ft^2) _____

Section 2.3.1

Minimum thickness of metal shall not be less than _____

Are wearing plates required under trolley runway rails? Yes _____ No _____

Are breathing holes required in welded box girders? Yes _____ No _____

Hoist capacity shall be shown on each side of crane in lb or ton _____

Connection between girders and end trucks shall be _____

Section 2.3.2

The method of attaching girders to end carriages during field erection shall be _____

Access shall be provided to the crane bridge from the crane runway by _____

Are cranes with equalizer trucks to be provided with steel platforms? Yes _____ No _____

Section 2.3.3

Trolley frame construction _____

MECHANICAL

Section 3.2

Other root contour threads acceptable for hook shanks _____

Is a safety handle on crane hook to be furnished? Yes _____ No _____

Is a safety latch on crane hook to be furnished? Yes _____ No _____

Is a lock to prevent hook from swiveling to be furnished? Yes _____ No _____

Section 3.3

Drum material _____

Are provisions required for regrooving drum? Yes _____ No _____

Section 3.4

Hoisting rope, grade and type _____

Section 3.5

Furnish equalizer bars or sheaves? Equalizer bars _____ Sheaves _____

Section 3.7.1

Material for track wheels

Bridge _____

Trolley _____

Heat treatment, if required, for track wheels

Bridge _____

Trolley _____

Idler track wheel shall be mounted as follows:

Bridge _____

Trolley _____

Bridge wheel track profile. Straight _____ Tapered _____

Section 3.7.2

Crane runway rails are to be section No. _____

Trolley rails shall be fastened to girders as follows: _____

Trolley rails are to be section No. _____

Section 3.8

Height of centerline on bumpers above top of crane runway _____ ft _____ in.

Type of bumpers to be furnished _____

Section 3.9

Bridge drive shall be of the following type _____

Type of solid couplings other than flexible coupling manufacturers' standard to be used _____

Section 3.10

The length of any section of the line shaft shall not exceed _____ ft _____ in.

Line shaft coupling shall be of the following type _____

Section 3.11

Type of fits for gears, pinions, wheels, couplings etc. shall be _____

Section 3.12

Items which shall have antifriction bearings: _____

Items which shall have sleeve bearings: _____

Specific service data other than Table 15 by which bearings are to be selected are: _____

Type and manufacturer of antifriction bearing _____

Section 3.13

Other methods of wheel axle bearing arrangements _____

Section 3.14

Gearing shall be of the following type _____

Gearing shall be designed and manufactured to comply with AGMA gear standards. Yes _____ No _____

If no, specify design and method of manufacture _____

Is allowance to be made in gearing design to allow 15% increase or decrease in total gear ratio? Yes _____ No _____

Material class _____

Heat treatment class _____

Section 3.14.6

Brinell surface and core hardness of gears shall be as specified in its class _____

or as follows. _____

Is the Brinell surface hardness to be stamped on the rim of the pinion and gear?

Yes _____ No _____

Section 3.14.7

Tolerances and inspection of gearing required other than AGMA standards _____

Section 3.15

Is allowance to be made in gearing housings to allow 15% change in total gear ratio of drives? Yes _____ No _____

May splash oil lubrication of bearings be used? Yes _____ No _____

Are provisions to be made for split-type oil seals to be used as replacements?

Yes _____ No _____

Section 3.16

All lubrication fittings, seals and equipment shall be furnished by the contractor.

Yes _____ No _____

If no, specify:

Size _____

Type _____

Type of fittings to be fitted with grease or oil seals _____

Is a centralized lubrication system to be installed? Yes _____ No _____

Section 3.17

Are regular hex sized bolts, nuts and cap screws to be used in accordance with ANSI Standard B18.2.1, 1972?

Yes _____ No _____

ELECTRICAL

Section 4.1

Type of brakes to be used. Air _____ Hydraulic _____

Is second hoisting brake required? Yes _____ No _____

If yes, is the second brake to be mounted on the motor shaft opposite the drive end? Yes _____ No _____

Section 4.1.2

Trolley to be furnished with:

Mechanical drag brake. Yes _____ No _____

Spring-set mechanical brake. Yes _____ No _____

Remote controlled service brake. Yes _____ No _____

Is remote controlled service brake to be arranged to set whenever power is removed from motor? Yes _____ No _____

Other specific brake requirements _____

Section 4.1.3

Required deceleration rate to stop bridge _____

Number of bridge stops/hr _____

Is bridge to be furnished with spring-set parking feature? Yes _____ No _____

Other specific requirements _____

Section 4.2

Main runway conductors for bridge travel will be furnished and erected by purchaser. Yes _____ No _____

The location, size and type of conductors _____

Specific requirements for bridge conductors _____

Section 4.3

Type of collector shoes to be furnished _____

Number _____

Section 4.4

Duty cycle requirements (including temperature) or electrical service class (Table 15):

Crane _____

Bridge _____

Trolley _____

Main hoist _____

Auxiliary hoist _____

If motors are to be operated under normal conditions which are less than rated voltage, specify percent voltage _____

Are friction factors shown in Table 20 acceptable for crane equipped with anti-friction bearings? Yes _____ No _____

If no, specify _____

Is a friction factor of 24 lb/ton acceptable for crane equipped with sleeve bearings? Yes _____ No _____

If no, specify _____

Required crane accelerations: Bridge _____ Trolley _____

If motor duty cycles are prolonged or repetitive, (greater than 50% time-on or 45 cycles/hr), specify condition. _____

Specify motor construction required:

Self-ventilated Yes _____ No _____ ;

Force-ventilated Yes _____ No _____ ;

Air-over-frame Yes _____ No _____ ;

Other _____

Section 4.5

Specify if manual control is required and function _____

Specify if control is required to operate in excess of $\pm 10\%$ of nominal AC and DC voltage and range _____

Control panel shall be located as follows _____

Relative position of master control switches _____

The magnetic control is to be used on trolley _____

Is control panel enclosure required? Yes _____ No _____

Type _____

When required, control panel enclosure shall be in accordance with NEMA classification _____

Type of resistors _____

NEMA Classification numbers _____

Specify speed if bridge or trolley speed on pendant-operated cranes is to exceed 200 fpm _____

Other control features required for the motion specified _____

Section 4.6

Limit switches shall be of the following type _____

Is a free-swinging weighted beam arrangement acceptable for activating the limit switch? Yes _____ No _____

If no, specify _____

Section 4.7

Are fuses required to provide short circuit protection on load side of main disconnecting device? Yes _____ No _____

What means for operating the safety switch on the foot-walk are to be provided in the cab? _____

Specify type of safety devices required on auxiliary electrical equipment

Fused safety switches. Yes _____ No _____

Circuit breakers. Yes _____ No _____

Section 4.8

Wiring requirements _____

Section 4.9

Are magnet cable reels required? Yes _____ No _____

If yes: Magnet size: _____ Weight _____ Cold amps _____

Type of magnet cable reel _____

Magnet reel to be furnished by purchaser _____ seller _____

Extra flexible magnet cable to be furnished by purchaser _____ seller _____

Magnet control will be furnished by _____

Magnet control will be installed by _____

When magnet is not specified, is space to be provided on trolley for future mounting of magnet cable reel? Yes _____ No _____

Is disconnect clutch for magnet cable reel required? Yes _____ No _____

Section 4.10

Lighting fixtures shall be furnished as follows:

Size _____

Number _____

Section 4.11

Signal lights shall be furnished as follows: size _____ number _____ location _____ connections _____

SPECIAL FEATURES:

Attach sketch illustrating clearance between any part of crane, building column, roof chord or other stationary structure.