SAE The Engineering Society For Advancing Mobility Land Sea Air and Space ® INTERNATIONAL
400 Commonwealth Drive, Warrendale, PA 15096-0001

SURFACE VEHICLE INFORMATION REPORT

Submitted for recognition as an American National Standard

SAE J1366

REAF. JUN2000

Issued 1994-10
Reaffirmed 2000-06

Superseding J1366 OCT1994

Rating Lift Cranes Operating on Platforms in the Ocean Environment

***Foreword*—**This document has been changed to comply with the new SAE Technical Standards Board format. Definitions are now Section 3. All other section numbers have changed accordingly.

1. *Scope*

***Scope*—**The scope of this SAE Information Report is limited to a lift crane mounted on a fixed or floating platform, lifting loads from a vessel alongside. The size of the vessel is assumed not to exceed that of a workboat as defined in 3.15.

1.1 Purpose—The purpose of this document is to establish the design dynamic loads, the calculation procedures for rated loads, and a load-rating chart format for lift cranes operating in a variety of sea conditions.

2. *References*

2.1 Applicable Publications—The following publications form a part of this specification to the extent specified herein. Unless otherwise indicated, the latest issue of SAE publications shall apply.

2.1.1 SAE PUBLICATIONS—Available from SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001.

SAE J959—Lifting Crane, Wire-Rope Strength Factors
SAE J987—Rope Supported Lattice-Type Boom Crane Structures—Method of Test
SAE J1093—Latticed Crane Boom Systems—Analytical Procedure

2.1.2 OTHER PUBLICATIONS

1. American Institute of Steel Construction, Inc. (AISC), "Manual of Steel Construction," Eighth Edition (or more current), Chicago, IL, 1980
2. American Petroleum Institute (API) 2C 1988 Standard
3. J. J. Meyers, C. H. Holm, and R. F. McAllister, "Handbook of Ocean and Underwater Engineering," McGraw-Hill Book Company, 1969
4. J. S. Bendat and A. G. Piersol, Random Data: "Analysis and Measurement Procedures," New York, NY, John Wiley and Sons, Inc., 1971
5. M. S. Lonjuet-Higgins, "On the Statistical Distribution of the Heights of Sea Waves," Journal of Marine Research, Vol. II, No. 3, 1952, pp. 245–266
6. W. J. Pierson, Jr., G. Newman, and R. W. James, "Practical Methods for Observing and Forecasting Ocean Waves by Means of Wave Spectra and Statistics," U.S. Navy Hydrographic Office Publication Number 603, Washington, DC, 1955, Reprinted 1960 and 1967
7. J. M. Gere and W. Weaver, Jr., "Analysis of Framed Structure," New York, NY, Van Nostrand Reinhold Company, 1965

2.2 Related Publications—The following publications are provided for information purposes only and are not a required part of this document.

1. Civil Engineering Laboratory, Technical Memorandum M-51-76-11: "Dynamic Vertical Forces on a Crane Loading (Unloading) a Floating Platform," by C. C. Ward, Port Hueneme, CA, Sept. 1976
2. K. V. Johnson, "Theoretical Overload Factor Effect of Sea State on Marine Cranes," Paper presented at Offshore Technology Conference, 1976 (Paper No. OTC 2584)
3. Civil Engineering Laboratory, Technical Note N-1371: "The Motion of Floating Advanced Base Components in Shoal Water—A Comparison Between Theory and Field Test Data," by D. A. Davis and H. S. Zwibel, Port Hueneme, CA, Jan. 1985
4. V. L. Streeter, "Fluid Mechanics," McGraw-Hill Book Co., 1966
5. T. C. Gillmer, "Fundamentals of Construction and Stability of Naval Ships," United States Naval Institute, 1969

3. Definitions

3.1 Significant Value—The average of the highest one-third of given population values. Population values may refer to wave height, roll, pitch, or yaw angles, etc. The significant value is twice the standard deviation, assuming a zero mean time history.

3.2 Sea State—An indicator relating the height of the waves to sea conditions in relative terms.

3.3 Wave Instrument Reading—Used on the load rating chart indicates the value obtained from a wave buoy or a wave staff that relates to the sea conditions. The wave instrument reading can be analyzed to form the ratio of the average wave height to the average period (H/T).

3.4 Wave Height—The vertical distance from wave crest to trough.

3.5 Surge—The single amplitude (SA) fore and aft ship motion along the longitudinal axis through the center of gravity (see Figure 1).

3.6 Sway—The SA athwart ship motion along the transverse axis through the center of gravity (see Figure 1).

3.7 Heave—The SA vertical ship motion along the vertical axis through the center of gravity (see Figure 1).

3.8 Roll—The SA angular ship motion about the longitudinal axis through the center of gravity (see Figure 1).

3.9 Pitch—The SA angular ship motion about the transverse axis through the center of gravity (see Figure 1).

3.10 Yaw—The SA angular ship motion about the vertical axis through the center of gravity (see Figure 1).

3.11 Offlead—The percent slope from the vertical in the vertical plane of the boom that locates the position of the load with respect to the tip of the boom.

3.12 Sidelead—The percent slope from the vertical normal to the vertical plane of the boom that locates the position of the load with respect to the tip of the boom.

3.13 Dynamic Rated Load (W_D)—The maximum load that can be lifted under specified dynamic conditions without exceeding allowable strength limits.

3.14 Static Rated Load (W_S)—75% of the maximum load that can be lifted under normal land conditions without exceeding allowable strength limits.

3.15 A Typical Workboad—A vessel of 180-ft length, 40-ft beam, and 1500-long-ton displacement.

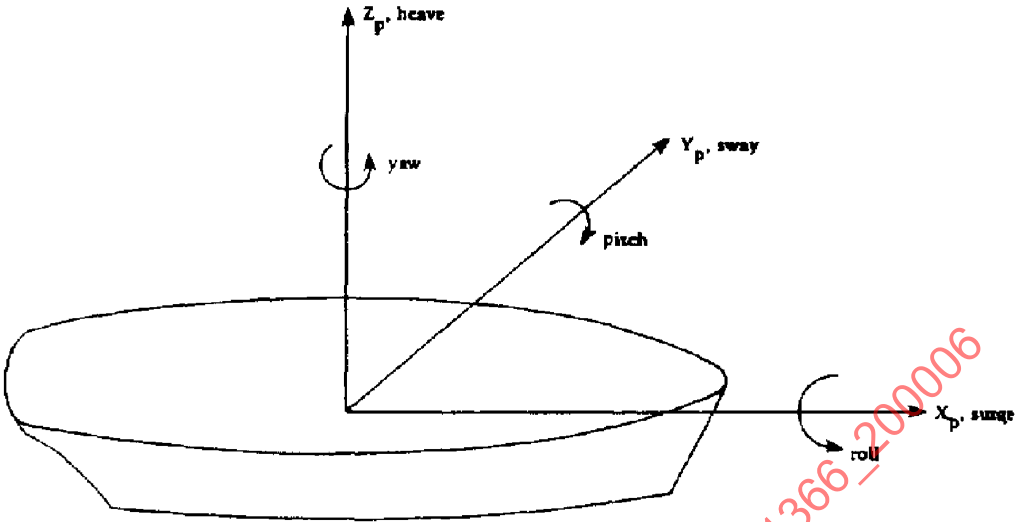

FIGURE 1—PLATFORM MOTION COORDINATE SYSTEM

3.16 List—The angle of inclination of ship about the longitudinal axis through the center of gravity.

3.17 Trim—The angle of inclination of ship about the transverse axis through the center of gravity.

3.18 Load Hoist Line Velocity—The full load hoist speed at the drum based on the maximum rated load for the specified rope.

4. Applications—This document establishes a method of arriving at standard dynamic loads presumed to act on a crane operating in various specified sea conditions. The load factors are used to determine dynamic rated loads for the crane when operating in the specified sea conditions. The dynamic loads specified were arrived at by consultations between crane manufacturers, users, and Navy personnel on the interaction between environmental effects and ship motions.

4.1 Crane Manufacturer's Responsibility—To calculate the static- and dynamic-rated capacity of the crane for the standard specified sea conditions and to publish the rating chart.

4.2 User's Responsibility—To determine sea condition existing when lift work is to be performed so that the proper rated-load column can be selected. The sea condition may be determined by consultation with on-site expert personnel such as ship captain, lift-work superintendent, etc. The appropriate rating thus determined should be relayed to the crane operator and stay in effect until similarly revised.

4.3 Customer's Responsibility—Provide the crane manufacturer with the required response-amplitude operators for the crane supporting vessel. The heave, roll, and pitch response-amplitude operators for a 135 degree incident wave shall be provided for the desired operating water depth.

The response-amplitude operators are complex numbers in polar form for each wave frequency to be considered. They are expressed in terms of the modulus (feet/foot or radius/foot) and amplitude (radians). Additionally, the customer shall specify the location of the crane on the deck of the supporting vessel and the height of the deck above the water. The customer shall also provide the necessary crane parameters such as the desired boom length and number of parts of load hoist line. The customer-supplied information will be used as input for the rating software described in Appendix A.

4.4 Special Considerations Affecting Application

4.4.1 Ballasting systems capable of reducing load-induced platform motion throughout the crane operating procedure(s) are to be considered in determination of list and trim loading on the crane (Appendix B).

4.4.2 A mooring system of anchor winches capable of adequately limiting platform response to wave action is to be considered in determination of platform motion response factors.

4.4.3 A heave-compensating device used in the load-hoist system may allow an increase in the dynamic rated load. The amount of reduction should be based on the crane manufacturer's interpretation of performance test data.

4.4.4 For loads lifted from a platform on which a crane is mounted, V_D, A_D, and V_{BP} in Equation 2 are assumed zero, and the picking peak dynamic load is greatly reduced. A rated load greater than the static rated load as defined in 6.3 shall not be used.

4.4.5 After the crane lifts the load, the load rating should consider boom point acceleration and load pendulation effects (see Section 9).

5. Dynamic Load—The dynamic load being addressed is imposed on the crane at the time of load liftoff from the moving deck of the vessel alongside. Additionally, consideration is directed to the effects of the horizontal displacement in the plane of the boom and normal to the plane of the boom caused by surge and sway of the vessel.

After studying the motions of a workboat, it is assumed that the vertical motion follows wave amplitude and that horizontal motions include sway, surge, and yaw, but exclude drift.

The vertical dynamic load, P (lb), that occurs when the lifted load, W (lb), is directly under the boom tip is given by the equation:

$$\bar{T}p = W\left\{1 + \left[\frac{K}{gW}(V_D + V_H)^2 + \left(\frac{A_D}{g}\right)^2\right]^{1/2}\right\} \qquad \text{(Eq. 1)}$$

where:

g = acceleration due to gravity (ft/s^2)
K = vertical structural stiffness component with the load force at appropriate offlead (lb/ft)
V_D = vertical velocity of the workboat deck at the pick point (ft/s)
V_H = velocity of the load hook which includes the load hoist line velocity, parts of line, and the boom point velocity V_{BP} (ft/s)
A_D = acceleration of the workboat deck at the pick point (ft/s^2)

Since Equation 1 is a quadratic equation in terms of W, Equation 1 may be solved for W and called W_D:

$$\frac{1}{W_D} = \frac{-B \pm \sqrt{B - 4AC}}{2A} \qquad \text{(Eq. 2)}$$

where:

$$A = P_{max}^{\ 2} \qquad \text{(Eq. 3)}$$

$$B = -2P_{max} - \frac{K(V_D + V_H)^2}{g}$$

$$C = 1 - \left(\frac{A_D}{g}\right)^2$$

Wave heights and periods, V_D, A_D, and V_{BP} are calculated considering the sea spectrum, ship motion response to the spectrum sea, and boom point position. The values for V_D, A_D, and V_{BP} selected for rating purposes, are 90% probability of occurrence values (see Appendix A). The crane is assumed to be in a rating position as shown in Figure 2. Implicit in Equation 2 are the dynamic effects of picking peak dynamic load as well as sidelead and offlead effects.

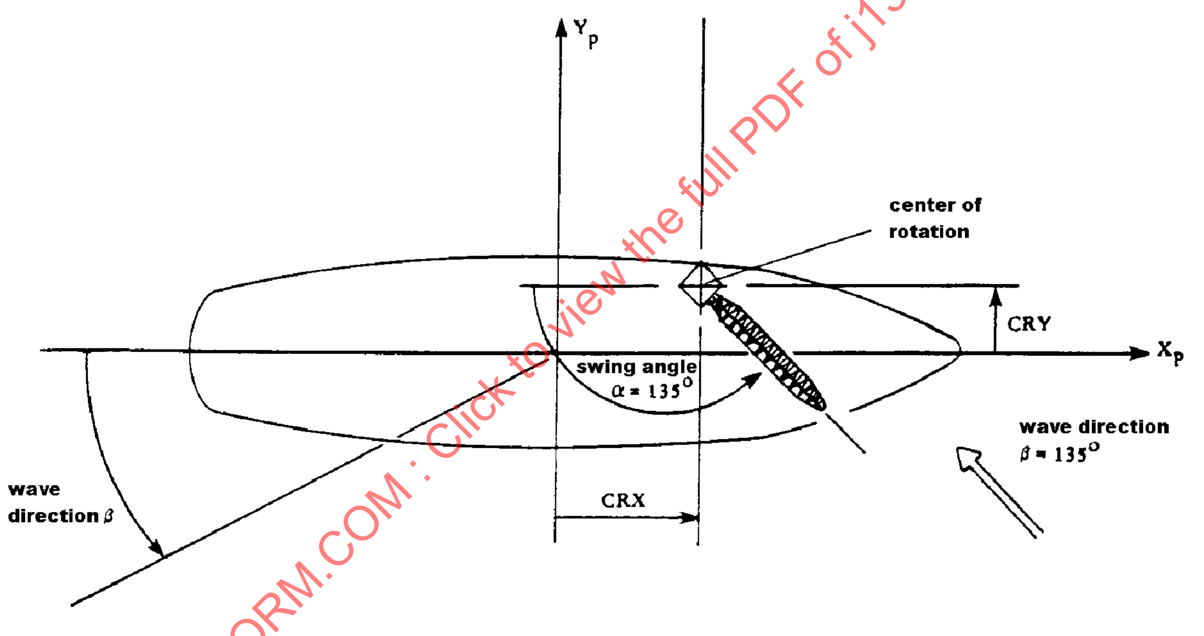

FIGURE 2—STANDARD CRANE RATING CONDITIONS: CRANE ORIENTATION AND SEA DIRECTION

See Table 1 for sidelead and offlead values. Equation 2 is to be used to calculate a series of values for W_D corresponding to values of H/T, such as those listed in Figure 3.

TABLE 1—PERCENT OFFLEAD AND SIDELEAD[1]

	Beaufort Wind Height	Significant Wave Height $H_{1/3}$ (ft)	Wave Average Period T (s)	Average Wave Length λ (ft)	Wave Instrument[2] Reading, H/T (ft/s)	Offlead (%)	Sidelead (%)
Static							
	—	—	—	—	0	0	2.7
Dynamic							
	3	1.0	2.4	20.0	0.26	6	3
	4	2.9	3.9	52.0	0.46	8	4
	5	6.9	5.4	99.0	0.79	12	6
	6	13.0	7.0	164.0	1.15	16	8
	7	23.0	8.7	258.0	1.64	22	11

1. Alternatively,

$$\%\ \text{offlead or sidelead} = \frac{(180\ \text{ft})(\%\ \text{indicated})}{\text{boom tip height above boat deck}}$$

Minimum sidelead not to be less than 2.7% of static load W_S.

2. Readings taken at the center of the Beaufort Wind Force range (table 11-10, Reference 1).

Manufacturer ____________________

Crane Model ____________ Serial Number ____________ Boom Length ____________

Crane Supporting Platform ____________________

Distance From Platform Deck to Sea Level ____________________

Crane Location:____________Distance From Stern to Center of Rotation ____________

Distance From Port Side to Center of Rotation ____________________

Distance From Platform Deck to Boom Foot ____________________

[Hoist Line: 7/8 6x25 IWRC - EIPS]

Radius (ft)	Boom Angle (deg)	Static Rating (lb)	Dynamic Ratings at Following Instrument Readings, $\bar{H}/\bar{T}$ (ft/s) 0.26	Dynamic Ratings at Following Instrument Readings, $\bar{H}/\bar{T}$ (ft/s) 0.46	Dynamic Ratings at Following Instrument Readings, $\bar{H}/\bar{T}$ (ft/s) 0.79	Dynamic Ratings at Following Instrument Readings, $\bar{H}/\bar{T}$ (ft/s) 1.15	Dynamic Ratings at Following Instrument Readings, $\bar{H}/\bar{T}$ (ft/s) 1.64
Four-Part Main Hoist Line							
25	78.5	90 000	72 000	56 000	42 900	30 500	
30	75.6	90 000	69 000	56 000	42 900	30 500	
40	69.6	90 000	67 900	56 000	41 900	30 500	
50	63.3	82 000	56 300	46 900	34 700	25 400	
One-Part Whip Hoist Line							
25	80.1	22 700	18 900	16 900	16 300	13 900	11 500
30	76.1	22 700	18 900	16 900	16 300	13 900	11 500
40	70.3	22 700	18 900	16 900	16 200	13 900	11 500
50	64.0	22 700	18 400	16 500	15 800	13 400	11 000

FIGURE 3—MARINE CRANE RATING CHART FOR FLOATING PLATFORM

6. ***Calculation Procedure***—The dynamic rating calculation procedure outlined as follows is implemented in computer software which is described in Appendix A. The calculated dynamic rated loads (W_D) will not produce a peak dynamic load that will exceed the land rated load (P_L).

6.1 Land Rated Load—Begin the calculation procedure with a land rated load (P_L) for each W_D that will be listed on the dynamic-rating chart. The land rated load, P_L, is determined by referencing strength margins to standard industry test practice for land cranes, without limitations due to the number of parts of load hoist line. This paragraph does not imply that a crane test is required; however, it does imply that the crane has been properly rated for land use, that is, it would pass SAE J987 if tested.

6.1.1 Strength margins for land rated loads in all areas above the top slewing ring mounting face shall be not less than those specified in SAE J987 (structures) and SAE J959 (ropes) with an applied vertical load P_L that has been modified to include an out of plane side load and optionally inplane offlead according to Table 1 for each sea state shown in the dynamic rating chart. The SAE J987 test rated crane may be derated for additional effect of sideleads in excess of 2.0% x P_L using calculation procedures such as those specified in SAE J1093.

6.1.2 Strength margins for land rated loads in all areas below the bottom of the slewing ring mounting face shall not be less than those specified in Section 5 of the AISC Manual of Steel Construction (Reference 1) with 1.5 P_L as the applied load, or 1.5 times the strength margin specified by SAE J987 when calculated or tested with 1.0 P_L as the applied load (see 6.1).

6.1.3 The strength margin within the swing circle assembly including fasteners shall be such that the computed stress with dead load plus 3.75 P_L shall not exceed the engineering ultimate tensile strength of the material.

6.2 Calculate P_{max}—For each W_D to be shown find P_{max}, the smallest of

$$P_{max} = \frac{P_L \cos\beta}{\cos(\beta - \alpha)} \qquad \text{(Eq. 4)}$$

which accounts for offlead when offlead is not considered in 6.1.1 or

$$P_{max} = \frac{N_P F_{BS}}{3.5} \qquad \text{(Eq. 5)}$$

which accounts for the number of parts of line where:

P_L = the land rated load (lb)
β = the boom angle (degrees)
α = the offlead angle (degrees)
N_P = the number of parts of line in the load-hoist system
F_{BS} = the load-hoist line breaking strength (lb)

6.3 Calculate W_S—Calculate the static rated load W_S where W_S is the smallest of

$$W_s = 0.75 P_L = \frac{P_L}{1.33} \qquad \text{(Eq. 6)}$$

which accounts for the crane components or,

$$W_s = \frac{N_P F_{BS}}{(1.33)(3.5)} = \frac{N_P F_{BS}}{4.66} = \frac{N_P F_{BS}}{5.0} \qquad \text{(Eq. 7)}$$

which accounts for the load hoist ropes. The 5.0 factor has been included to be consistent with the American Petroleum Institute (API) 2C 1988 standard (Reference 2).

6.4 Calculate K—For each radius to be shown, calculate the vertical stiffness (spring rate) of the system, effective at the load point. The system stiffness is a function of the load line, suspension line, A-frame, boom, pedestal, and jib under service loads. Fewer components may be used as this will make the resulting ratings more conservative.

6.5 Calculate V_H—From the heave, pitch, and roll response amplitude operators supplied by the customer, calculate the vertical velocity of the boom point. The absolute velocity of the hook (V_H) is determined by adding the vertical velocity of the boom point to the block hoisting speed which is the load-hoist line velocity, defined in 3.18, divided by the number of parts of line.

6.6 Calculate V_D and A_D—Using the assumptions that the workboat motion follows the wave motion calculate the vertical velocity and acceleration of the deck at a point directly under the boom tip. The Bretschneider two parameter spectrum is assumed to describe the motion of the sea.

6.7 Calculate W_D—For each radius and sea state to be shown on the rating chart calculate W_D using Equation 2. Furthermore, no W_D shall be greater than W_S.

6.8 Rated Load Reduction—The calculated load for a given sea state shall be adjusted so that no rated load exceeds that for a smaller operating radius.

6.9 Calculate Swing Torque Requirement—The swing system shall be designed to start and stop any load within rated capacity at any rotational position of the crane, and it shall be designed to hold any load within rated capacity against the maximum calculated or expected sidelead as shown in Table 1, and any deck in inclination. Consideration must also be given to the unloaded configuration. This provision is not in the software described in Appendix A, although the significant pitch and roll are computed in the software and printed.

7. Marine Crane Rating Chart Format—A suggested format for a marine crane rating chart is given in Figure 3. This format is suggested for use at the discretion of each manufacturer; some suggestions and comments regarding the content of this chart follow:

7.1 Each rating chart should be prepared for a fixed number of parts of load-hoist line.

7.2 Each rating chart should also indicate ratings for a whipline if the crane is so equipped.

7.3 The wave instrument reading, $\overline{H}/\overline{T}$, is a statistical description of the sea condition which is intended to be obtained from a wave recorder as described in 3.3 (see Equation 8.). Note that

$$\frac{\overline{H}}{\overline{T}} = 0.62\frac{H_{1/3}}{\overline{T}} \qquad \text{(Eq. 8)}$$

where:

$\overline{H}$, $H_{1/3}$, and $\overline{T}$ are taken at the center of the Beauford wind force range.

8. Rating for List—In many applications the crane is mounted on a platform (e.g., barge) that has a considerable list when a load is lifted even in calm seas. The list may be constant, or may vary as the load is swung to a different orientation on the platform. For these applications in calm seas, structural consideration must be given to sidelead and offlead. Particularly, the swing system and boom must be capable of withstanding the maximum calculated or expected sidelead. The calculations shown in Appendix A do not include rating for list.

8.1 Constant List Rating (Static)—These ratings are for specified list conditions when list is invariant with swing angle and no mounting vessel specifications are known. List (θ_T) is specified in degrees. The calculations in Appendix A do not include constant list rating.

8.1.1 SIDELEAD—The % sidelead to be designed for is shown in Equation 9:

$$SL = 100(\sin\theta_T \cos\psi) + 2.7 \quad \text{(Eq. 9)}$$

where:

θ_T = barge list angle (degrees)
ψ = crane rotation angle (degrees)

8.1.2 OFFLEAD—The % offlead to be designed for is:

$$OL = 100(\sin\theta_T \cos\psi) + 2.7 \quad \text{(Eq. 10)}$$

8.1.3 Stresses throughout the machine are calculated at various rotation angles (ψ) to find worst combinations of OL and SL.

8.2 Variable List Rating (Near Static)—The ratings are for calm sea conditions when barge mounting specifications are known. See Appendix B for derivation and nomenclature. Appendix A calculations do not include variable list rating.

8.2.1 SIDELEAD, VARIABLE—The % sidelead to be designed for is:

$$SL = 100(\cos\psi \sin\theta_T + \sin\psi \sin\theta_L) + 2.7 \quad \text{(Eq. 11)}$$

where:

θ_T, θ_L, and other nomenclature are defined in Appendix B

8.2.2 OFFLEAD, VARIABLE—The % offlead to be designed for is:

$$OL = 100(\sin\psi \sin\theta_T - \cos\psi \sin\theta_{TL}) \quad \text{(Eq. 12)}$$

8.2.3 Stresses throughout the machine are calculated at various rotation angles (φ) to find worst combinations of OL and SL.

9. Pendulation—After the load is picked from the workboat, the continued boom point motion of the floating-platform-mounted crane causes pendulation of the load. Load pendulation can cause considerable damage to surrounding cargo, vessels, or the crane itself. It is recommended that tagline restraint systems be used to restrain pendulation of the load.

PREPARED BY THE SAE OFF-ROAD MACHINERY TECHNICAL COMMITTEE SC31—
CRANES AND LIFTING DEVICES

APPENDIX A

CALCULATION OF THE DYNAMIC LOAD RATING

A.1 Objective—The dynamic load rating will be discussed and an algorithm for its determination will be presented.

A.2 Scope—In the usual practice of lifting a load from the deck of a heaving ship, the crane operator begins the hoist operation at some arbitrary position on a wave. Because of the relationship between hoist speed and wave speed, the pickpoint will be assumed at a random point where the velocity and the acceleration of the pickpoint on the workboat deck can be any value, but correlated with each other. The workboat motion is induced by wave action, which is a function of sea state, hull shape, and the displacement of the boat. The common workboat used around offshore platforms is usually 1500-ton displacement or less. Boats of this size respond strongly to the sea, and it is assumed for these calculations, based upon ship motion studies, that the vertical motion of the workboat rear quarter deck is the same as the vertical wave motion.

A crane mounted on a ship or barge is subject to the motion of its supporting vessel. This motion induces loads on the structure that are not considered in land operations. The peak dynamic load is incurred when the load is lifted from a moving workboat. This load is a function of the relative velocity between the hook and the workboat deck. The velocity of the hook includes the hoist velocity and the velocity of the boom point caused by the motion of the supporting vessel.

The response amplitude operators for the crane supporting vessel are assumed to be the responsibility of the customer. It is assumed that a linear ship motion theory will be employed. Rating calculations are the responsibility of the crane manufacturer. The sea used by the manufacturer shall be described with a Bretschneider spectrum (Reference 3).

A.3 Approach—The dynamic rated load is determined from Equation 2, which requires values for the maximum static load, the workboat deck velocity and acceleration, and the hook velocity. The maximum static load (P_{max}) is a function of the rated load including provision for sidelead for the entire machine (determined outside of this procedure) modified for offlead and the number of parts of load-hoist line. The workboat motion is assumed to follow the wave; therefore the velocity and acceleration amplitudes are a function of $H_{1/3}$ and T and the average wave length. The boom point velocity amplitude is obtained by transforming the ship response-amplitude operators to the boom point and multiplying by the appropriate Bretschneider spectrum. The velocity of the load-hoist line is combined with the boom point velocity in order to get the velocity of the hook. A simulation of the desired motion is performed. A sinusoidal simulation is used to simplify the random description of the sea. The random sea waves are represented by a sinusoidal wave having a frequency related to the wave average period. The amplitudes of the wave displacement, velocity, and acceleration are related to the significant wave height. The motion of the boom point is assumed to follow the supporting platform, therefore, the associated phase angle is zero. The motion of the workboat, under the boom point is out of phase with respect to the supporting platform and the boom point. This phase difference is accounted for by relating the horizontal distance from the center of gravity of the supporting platform to a point under the boom point, to the ocean wave average wave length. The dynamic rated load is calculated using values required by Equation 2 at 101 different locations on the simulated wave. The minimum dynamic rated load is selected for each operating radius shown in the rating chart. The algorithm details for determining the dynamic rated load follow.

A.3.1 Algorithm—The procedure that follows describes the computer software that is used to calculate the dynamic chart.

A.3.1.1 BASIC INPUT PARAMETERS—The input parameters required are defined in the SAE HS-1366 computer software manual in Section III of this publication. They will not be described at this point, suffice it to say that the basic crane geometry, crane member properties, ship system properties, and other geometric and related data are required.

A.3.1.2 LAND RATED LOAD, P_{MAX}, AND STATIC RATED LOAD—The land rated load is the standard SAE J987 rating for the crane with modifications for sidelead and optionally offlead. The standard land rated load with a 2.0% sidelead is modified to include each of the additional sideleads and optional offlead shown in Table 1 of this document. These ratings are not determined as part of this procedure but are part of the input parameters along with the associated boom radius and boom point elevations. The offlead rating, if not previously considered, is determined as shown in Equation A1:

$$P_{OL} = PL\frac{\cos\beta}{\cos(\beta - \alpha)} \qquad \text{(Eq. A1)}$$

which was determined by solving for a constant boom foot moment using the nomenclature in Figure A1.

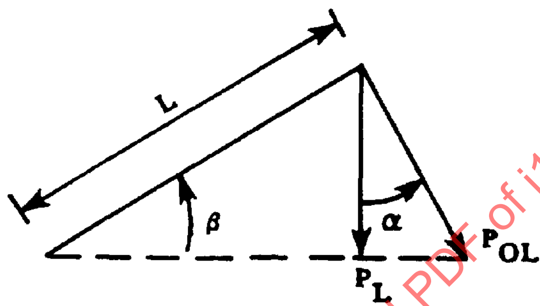

FIGURE A1—BOOM OFFLEAD RATING EQUATION NOMENCLATURE

where:

L = the length of the boom (ft)
β = the boom angle (degrees)
α = the offlead angle (degrees)
P_{OL} = the offlead load that will give the same boom foot moment as the vertical load (lb)
PL = the vertical load (lb)

The load-line rating P_{LL} is determined from the breaking strength of the load line ropes and the number parts of line between the boom point and the hook. This value is reduced by 3.5 as specified in SAE J959: Equation A2.

$$P_{LL} = \frac{N_P F_{BS}}{3.5} \qquad \text{(Eq. A2)}$$

where:

N_p = the number of parts of line in the load-hoist system
F_{BS} = the load-line breaking strength

The P_{max} term used in Equation 2 of the document is the minimum of the offlead rating and the load-line rating.

Finally, the static rated load, W_S, is determined so no dynamic load, W_D, will exceed W_S, where W_S is the smallest of (see Equation A3):

$$W_s = 0.75P_L = \frac{P_L}{1.33} \qquad \text{(Eq. A3)}$$

which accounts for the crane components or (see Equation A4):

$$W_s = \frac{N_P F_{BS}}{(1.33)(3.5)} = \frac{N_P F_{BS}}{4.66} \quad \text{(Eq. A4)}$$

which accounts for the load-hoist ropes. This procedure, including replacing the constant 4.66 by 5.0, is done so the SAE ratings do not exceed those produced by the American Petroleum Institute (API-2C, 1983) ratings.

A.3.1.3 LOCATE THE POINT OF INTEREST UNDER THE BOOM POINT—Considering the general coordinate system is located at the center of gravity of the crane supporting platform. The crane may be located at an arbitrary position on the platform. The vector in the horizontal plane that locates the boom point with respect to the platform center of gravity is calculated as shown in Equation A5:

$$r_x = O_r \cos(\gamma) + X_{cr} \quad \text{(Eq. A5)}$$

$$r_y = O_r \sin(\gamma) + Y_{cr}$$

$$r = \sqrt{r_x^2 + r_y^2}$$

$$\varepsilon = 180.0 - \frac{180.0}{\pi}\tan^{-1}\left(\frac{r_y}{r_x}\right)$$

where:

O_r = the operating radius (ft)
X_{cr}, Y_{cr} = the x and y distances between the ship cg and the center of rotation (ft)
γ = the swing angle with respect to the x axis (degrees)
r_x, r_y = the x and y distance to the point under the boom point in the x-y plane (ft)
ε = location of the r vector with respect to the negative x axis (degrees)

A.3.1.4 CALCULATE THE VELOCITY AND ACCELERATION OF THE PICK POINT ON THE WORKBOAT DECK—The workboat is assumed to follow the wave. The wave is described using the Bretschneider two parameter ($H_{1/3}$, T) spectrum. The area under this spectrum is the variance of the wave motion for a given sea state (Reference 4). The variance (θ^2) for a zero mean record can be converted to $a_{1/3}$ (single amplitude) or $H_{1/3}$ (double amplitude) significant wave heights by appropriate factors (2σ and 4σ, respectively) according to References 5 and 6. In this procedure a constant amplitude sinusoidal wave is used to simulate the motion of the workboat. The wave amplitude is $a_{1/3}$, the frequency is derived from the average period (T) of the random sea, $\omega = 2\pi/T$ in rad/s. The phase angle, ϕ, with respect to the center of gravity of the supporting platform is determined by comparing a projection of the r vector, determined in A.3.1.3, upon the average wave length vector λ as follows in Equation A6:

$$\phi = 2\pi\frac{r\cos(\xi - \varepsilon)}{\lambda} \quad \text{(Eq. A6)}$$

where:

ξ = the direction of the incident waves with respect to the negative x axis (degrees)
ε = the location of the r vector with respect to the negative x axis (degrees)
λ = the average wave length of the impinging waves (ft)

The workboat motion with respect to time t (s) is given by Equation A7:

$$D_D = a_{1/3}\sin(\omega t + \phi) \qquad \text{(Eq. A7)}$$
$$V_D = \omega a_{1/3}\cos(\omega t + \phi)$$
$$A_D = -\omega^2 a_{1/3}\sin(\omega t + \phi)$$

where:

$$\omega = \frac{2\pi}{T}$$

and D_D, V_D, and A_D are the workboat deck displacement, velocity, and acceleration, respectively.

A.3.1.5 VERTICAL STIFFNESS—The vertical deflection of the crane boom tip is computed using the stiffness matrix method of analysis (Reference 7). The origin of the coordinate system used to describe the crane is the boom foot. Various members can be included in the model (e.g., the boom, pendants, and gantry). The method employed uses a two-dimensional model. Provision is made to permit certain crane components to change position as the boom angle changes; the mast, pendants, and boom move as a unit when the boom is raised or lowered, for instance. A pedestal crane, a crawler crane, a truck crane, a ring supported crane, or any general configuration may be modeled.

The vertical deflection of the load-hoist fall is computed separately from the boom tip deflection, then the two quantities are combined. The deflection of the load hook due to the load line is shown in Equation A8:

$$\Delta_{LL} = \frac{12.0P_H[B_L + N_P(D_{BP} + D_W)]}{N_P^2 A_R E_R} \qquad \text{(Eq. A8)}$$

where:

Δ_{LL} = the deflection of the load line (in)
P_H = the load on the hook, a dummy 10 000 lb load is used
B_L = the boom length (ft)
N_P = the number of parts of line
D_{BP} = the distance between the deck and the boom point (ft)
D_W = the distance between the deck and the water surface (ft)
A_R = the area of rope (in^2)
E_R = the modulus of elasticity of the ropes (psi)

The vertical stiffness is computed from the deflection. The vertical boom point deflection is added to the deflection of the load-hoist hook times the cosine of the offlead angle to account for offlead. Then the hook load (10 000 lb) is divided by the total vertical deflection to produce the vertical stiffness. Of course, this procedure is repeated for each boom angle shown on the load-rating chart.

Vertical stiffness values are generally calculated by individual manufacturers using their own design programs. This document uses the previous procedure to make it generally applicable to all manufacturers. Alternate methods such as SAE J1093, for computing the vertical stiffness are acceptable. However, appropriate modification to the rating programs would have to be done by the manufacturer.

A.3.1.6 BOOM POINT VELOCITY AND HOOK VELOCITY—The boom point vertical velocity is determined from the ship response amplitude operators (RAO) and the Bretschneider spectrum. The customer is responsible for providing the heave, roll, and pitch displacement RAOs. These quantities must include the modulus (absolute value) and amplitude (argument) of the function in order to fully describe the complex number RAOs in polar form. The RAO is assumed to be determined with respect to the crane supporting platform center of gravity. The displacement RAO for the point of interest (i.e., the boom point) is computed using the appropriate transformation.

The displacement RAOs are converted to auto spectrum functions by multiplying the Bretschneider two parameter sea spectrum. The crane supporting platform vertical displacement RAOs for a unit wave are transformed to the boom point to represent the boom point vertical displacement. Multiplying the square of the boom point vertical displacement RAO by the Bretschneider spectrum (e.g., $H_{1/3}$ = 8.0 ft and $\bar{T}$ = 5.7 s) produces the auto spectrum (PSD) for the boom point vertical displacement. This displacement auto spectrum is converted to velocity by multiplying each ordinate by the corresponding angular frequency. The area under the spectrum is the variance of the velocity time history (assuming a zero mean velocity time history). The boom point velocity can be selected knowing the variance is related to the standard deviation by the square root function. A standardized variable may be used in conjunction with the standard deviation to give a particular value where one can be assured that there is a small chance of exceeding that value. For example, a standardized variable equal to 1.28 multiplied by the standard deviation of the boom point vertical (V_{BP}) yields approximately a 10% chance of exceeding that product value.

A sinusoidal simulation of the boom point velocity, consistent with the simulation of the workboat can be accomplished. Using the standard deviation and a 1.28 standardized variable for the desired amplitude, the boom point vertical velocity can be simulated by V_{BP} cos (ωt). To this quantity for each t, $\bar{T}$/100 increments, add the constant load line velocity to get the velocity of the hook. Notice that cos (ωt) was used in order to be consistent with the workboat velocity simulation explained in step 4. Since the amplitude of the cos (ωt) term is the boom point vertical velocity, ω cos (ωt) would be redundant.

A.3.1.7 SOLVE FOR THE DYNAMIC RATED LOAD—The dynamic rated load is solved using a quadratic equation for W_D. This equation is as shown in Equation A9:

$$\frac{1}{W_D} = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A} \qquad \text{(Eq. A9)}$$

where:

$$a = P_{max}^2 \qquad \text{(Eq. A10)}$$

$$B = -2.0P_{max} - \frac{K(V_D + V_H)^2}{g}$$

$$C = 1 - \left(\frac{A_P}{g}\right)^2$$

A_D = acceleration of the workboat deck (ft/s^2)
V_D = velocity of the workboat deck (ft/s)
V_H = the vertical velocity of the hook which includes the vertical velocity of the boom point and the load-hoist system (ft/s)
P_{max} = (defined under A.3.1.2) (lb)
g = the acceleration due to gravity (ft/s^2)
K = the vertical stiffness (lb/in)

As with any quadratic equation there are three possibilities for solution:

a. Two equal roots ($B^2 - 4AC = 0$)
b. Two real and distinct roots ($B^2 - 4AC > 0$)
c. Conjugate complex numbers ($B^2 - 4AC < 0$)

The last case is not practical for this application, therefore it is considered erroneous.

The dynamic rated load is defined as the minimum safe load that can be lifted at a given boom angle. Thus, the minimum value is selected for case A.3.1.a or A.3.1.b. Furthermore, a sinusoidal simulation of the workboat velocity (V_D) and acceleration (A_D), and the boom point velocity (V_{BP}) is done at T/100 increments to insure the proper relationship exists between these quantities and to ensure the computed load population contains the minimum dynamic load. Lastly, the computed dynamic rated load is compared against the static load defined in Section 2 to ensure that this later load is not exceeded. A provision has been added to ensure the dynamic rated load is never larger than the dynamic rated load for smaller operating radii.

APPENDIX B

STATIC SIDE LOADING OF BARGE-MOUNTED CRANES

B.1 Introduction—Cranes operating on floating barges induce list and trim that vary with the slew angle of the crane, load being handled, and vertical and horizontal position of the unbalanced load relative to the metacenter. List is the angle between the plane of the barge and the horizon in the transverse direction, and trim is the angle between the plane of the barge and the horizon in the longitudinal direction. Because most barges are not built or loaded symmetrically, the combined list and trim may result in a sidelead as well as an offlead angle. It is the purpose of this appendix to develop a concise mathematical approximation for the estimation of these static side loading components for any arbitrary crane orientation.

B.1.1 Naval Architecture Method—To calculate the list and trim produced by a crane operating on a barge of known characteristics, it is first necessary to determine the location of the barge metacenter and the loads and moments with respect to the metacenter.

B.1.1.1 The following assumptions are made:

a. The crane is mounted on a rectangular, wall-sided barge. The weight of the barge, crane, load, and foundation is W (lb) and the area of the barge at the water plane is A (ft^2).
b. For small angles of list and trim, the changes in A are negligible.

B.1.1.2 The average barge draft, d, is calculated by using Equation B1:

$$d = \frac{W}{\lambda A}(ft) \quad \text{(Eq. B1)}$$

where:

λ = density of sea water, 64 lb/ft^3

B.1.1.3 The transverse metacentric height above the center of gravity, GM_T, is given by Equation B2:

$$GM_T = BM_T - BG \quad \text{(Eq. B2)}$$

where:

BG = distance from the center of buoyancy (approximately d/2) to the center of gravity
$BM_T = b^2/12d$, where b is the width of the barge

B.1.1.4 Similarly, the longitudinal metacentric height, GM_L, is given by:

$$GM_L = BM_L - BG \quad \text{(Eq. B3)}$$

where:

$BM_L = L^2/12d$, where L is the length of the barge

B.1.1.5 The list angle θ_T and the triangle θ_L are given by Equations B4 and B5:

$$\theta_T = \sin^{-1}\left[\frac{FM_T + RM\sin(\psi)}{WGM_T}\right] \quad \text{(Eq. B4)}$$

and

$$\theta_L + \sin^{-1}\left[\frac{FM_L + RM\cos(\psi)}{WGM_L}\right] \quad \text{(Eq. B5)}$$

where:

FM_T = fixed moment in the transverse direction (ft-lb)
RM = rotating moment (ft-lb)
ψ = angle of rotation of the crane with respect to the longitudinal centerline of the barge

B.1.2 Alternate Solution Method—Where only the barge dimensions are known, or the customer requests assistance from the crane manufacturer in selecting a proper barge size, the following method gives list and trim angles sufficiently accurate for final design.

B.1.2.1 The following assumptions are made:

a. An unbalanced load P (lb) is applied to a rectangular wall-sided barge at an original radius R (ft) from barge center and a height H above the center of gravity of the loaded barge. It is standard practice, when using this method to size a barge, to take P and R as the total weight and center of gravity of the loaded crane including any counterweight and to select the values of P and R that produce the maximum moment about the barge centerline.
b. For small angles of list and trim, the changes in barge area A at the water plane are negligible.

B.1.2.2 Considering the barge list θ_T:

a. The average barge draft (and the draft at the longitudinal barge axis) is given by Equation B6:

$$d = \frac{W}{\lambda A}(\text{ft}) \quad \text{(Eq. B6)}$$

where:

λ = density of sea water, 64 lb/ft^3

B.1.2.3 The transverse metacentric height above the center of gravity, GM_T, is given by Equation B7:

$$GM_T = BM_L - BG \quad \text{(Eq. B7)}$$

where:

BG = distance from the center of buoyancy (approximately d/2) to the center of gravity
$BM_T = b^2/12d$, where b is the width of the barge

B.1.2.4 Similarly, the longitudinal metacentric height, GM_L, is given by Equation B8:

$$GM_L = BM_L - BG \quad \text{(Eq. B8)}$$

where:

$BM_L = L^2/12d$, where L is the length of the barge

B.1.2.5 The list angle θ_T and the triangle θ_L are given by Equations B9 and B10:

$$\theta_T = \sin^{-1}\left[\frac{FM_t + RM\sin(\psi)}{WGM_T}\right] \quad \text{(Eq. B9)}$$

and

$$\theta_L = \sin^{-1}\left[\frac{FM_L + RM\sin(\psi)}{WGM_L}\right] \quad \text{(Eq. B10)}$$

where:

FM_T = fixed moment in the transverse direction (ft-lb)
RM = rotating moment (ft-lb)
ψ = angle of rotation of the crane with respect to the longitudinal centerline of the barge

B.1.3 Alternative Solution Method—Where only the barge dimensions are known, or the customer requests assistance from the crane manufacturer in selecting a proper barge size, the following method gives list and trim angles sufficiently accurate for final design.

B.1.3.1 The following assumptions are made:

a. An unbalanced load P (lb) is applied to a rectangular wall-sided barge at an original radius R (ft) from barge center and a height H above the center of gravity of the loaded barge. It is standard practice, when using this method to size a barge, to take P and R as the total weight and center of gravity of the loaded crane including any counterweight and to select the values of P and R that produce the maximum moment about the barge centerline.
b. For small angles of list and trim, the changes in barge area A at the water plane are negligible.

B.1.3.2 Considering the barge list θ_T:

a. The average barge draft (and the draft at the longitudinal barge axis) is given by Equation B11:

$$d = \frac{W}{\gamma A}(ft) \quad \text{(Eq. B11)}$$

where:

W = total weight of balanced and unbalanced loads (lb)
γ = density of sea water, 64 lb/ft^3

b. The actual barge draft at any distance X from the longitudinal barge axis is given by Equation B12:

$$d_x = d \pm x\tan(\theta_t) = \frac{W}{\gamma A} \pm x\tan(\theta_T) \quad \text{(Eq. B12)}$$

and also by Equation B13:

$$d_x = \frac{1}{\gamma}\left(\frac{W}{A} \pm \frac{M_x}{I_T}\right) \quad \text{(Eq. B13)}$$

resulting in the Equation B14:

$$\tan\theta_T = \frac{1M}{\gamma T^I} = 0.156\frac{M}{I_T} \quad \text{(Eq. B14)}$$

where:

I_T = barge transverse moment of inertia = L $b^3/12$ (ft^4)