Innovative System for Off-the-Ground Rotation of Long Objects Using Mobile Cranes

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Abstract: Managing heavy-pressure vessel lifts on construction sites requires planning, arranging adequate crane support, and preparing collision-free rotation (from a horizontal position to a vertical position) of the vessel. Generally, selecting mobile cranes and developing engineered lift studies for vessels are done using two cranes and analyzing the lift for each crane individually on the basis of the selected cranes' lift-capacity specifications provided by crane manufactures. This practice is relatively costly and time-consuming. Optimizing the mobile cranes' use and location is also difficult. To assist in the field operation of mobile cranes and to provide engineers with a planning tool, this paper presents a methodology to carry out such a lift utilizing only one crane. Using the developed methodology and mechanism, heavy vessels can be rotated off the ground (in the air) with one crane. The proposed mechanism is supported with a mathematical model that has been developed into a computer system and has been integrated with a previously developed crane selection and ground pressure calculation system and crane database. The developed system provides users with a lift study analysis for a given configuration as well as simulation results with interactive graphics to assist in the selection of an optimum configuration. This research is important as projects involving heavy lifts need to reduce the cost and time associated with construction operations. **DOI: 10.1061/(ASCE)CO.1943-7862.0000309.** © *2011 American Society of Civil Engineers.*

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Introduction

Industrial projects, including oil refineries, usually involve a lengthy and complex process; efficient crane operation can have a significant positive impact on the overall scheduling, cost, and safety of these projects. Prefabrication, preassembly, modularization, and off-site fabrication are strategies employed in industrial projects to facilitate the relocation of portions of work to off-site fabrication centers. However, this relocation of work to off-site destinations implies a much more frequent use of cranes. Currently, planning for mobile crane operations is often performed intuitively and informally (Shapira and Glascock 1996). The use of machinery in the construction industry has always been a major cost element, and mobile cranes are among the most expensive type of construction equipment. Because of the competitiveness of the construction industry, contractors need to optimize the utilization of cranes to reduce the cost and time of the construction process. Many researchers have developed approaches to assist practitioners in optimizing site layout (Lim et al. 2005; Sivakumar et al. 2003; Tam et al. 2001; Chung 1999). In these approaches, constraints such as safety, time, and cost are taken into account to determine the best possible locations for the mobile cranes. Current research on construction cranes focuses primarily on developing tools to assist practitioners in the crane-selection process (Al-Hussein et al. 2001, 2005; Sawhney and Mund 2002; Hanna and Lotfallah 1999; Zhang et al. 1999). Usually, crane selections for lifting heavy-pressure vessels are chosen on the basis of the heaviest lift and/or the largest lift radius, and the potential crane and pick position is identified by an experienced lift engineer. For 2-crane lift operations, there are few written guidelines and published literature. Experienced lift engineers select cranes on the basis of an individual crane's capacity to lift the heavy vessel.

The current industry practice for lifting large, heavy vessels has typically been carried out utilizing two cranes. With each crane hooked to one end of the vessel, the lifting process starts simultaneously, maintaining the vessel at a horizontal position. Then, the main lift crane raises the top of the vessel while the tail crane holds the bottom of the vessel close to the ground. After being rotated into a vertical position, the vessel is placed into position, as shown in Fig. 1. However, 2-crane lifts have certain difficulties, such as the following:

1. Side loading from an out-of-plumb load line can affect crane operation. This may happen when the deviation from plumb is near right angle to the boom, such as when one crane swings and is permitted to pull the second crane with it (Shapiro et al. 1999).
2. Bouncing movement (sometimes called jerking) of the heavy vessel can occur when the lifting operation is not properly coordinated and when the speeds of the cranes are different. As such, the tail crane receives additional undesired stresses, which can cause accidents.

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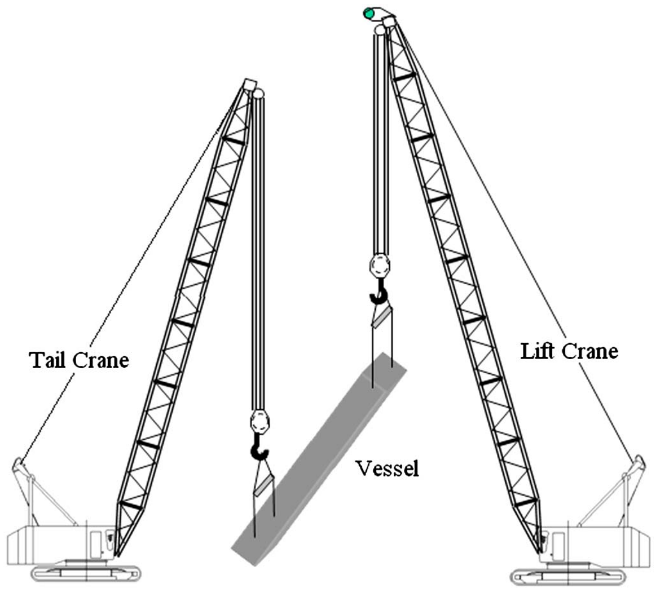

Fig. 1. Heavy-pressure vessel lift using two cranes (current practice)

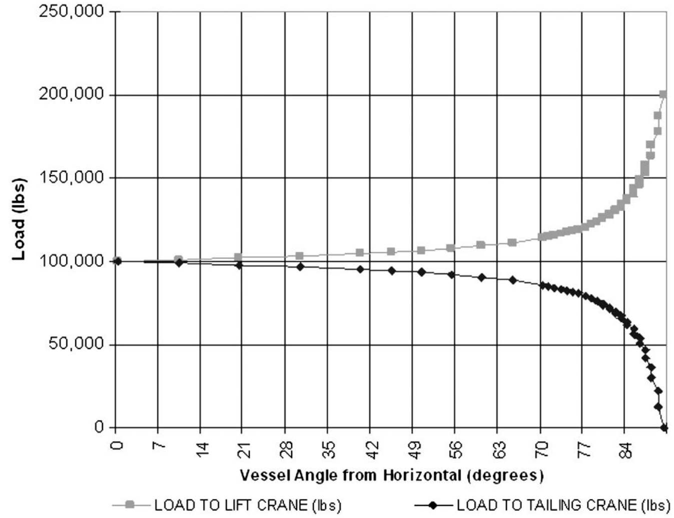

Fig. 2. Typical load distributions of 2-crane operation (1 lb = 0.4536 kg)

3. Swinging movement of the vessel can take place toward the tail crane's travel direction. Lifted objects with high surface areas are sensitive to wind, and at certain elevations, wind forces can displace the object in the air and create swinging movements.
4. Fleeting effect happens because of multicrane operations where the load periodically moves off-center from the lifted hook. Lattice boom cranes are designed to carry the load perpendicular to the ground, and any additional force acting on the structure in a different direction may create significant safety concerns. Fleeting effect can arise at the lift stage when the lift crane lifts and the tail crane walks, which causes forward and backward motions.
5. Booms tip collision can occur during a tandem lift of an object. Such an incident can take place at the end of the 2-crane operation when the load carried by the tail crane is transferred to the lift crane. Any open-air load transfer from one crane to another during a lifting operation could be critical.
6. The load transfer from the tail crane to the lift crane happens within the last few degrees of the rotation, which can lead to impact loading if released too quickly. If not handled properly, the load transfer can momentarily cause the tail crane to take nearly the full load of the vessel.

The typical load transfer from the tail crane to lift crane is shown in Fig. 2. However, during the operation, the tail crane may take full load because of the fault of either crane operator, and an accident may occur. Therefore, 2-crane lifting requires a detailed and costly analysis of each lift to ensure that the objectives of the lift are being met, and the safety of the crew is maintained. As cranes are a major cost item in the construction process, the industry is always seeking to optimize the utilization of the resource. The limitations of current approaches suggest the need for a new methodology to select one crane for performing heavy-vessel lifts at construction sites.

To overcome the limitations of current approaches and to reduce the cost and time of large-scale construction projects (such as oil sands projects), a methodology has been developed to utilize one crane rather than two for the lifting of heavy-pressure vessels. The proposed methodology has been built on the basis of the mathematical model (described in the following section) and developed using Microsoft Visual Basic to ensure a user-friendly interface and to control data integrity. The developed system has also been integrated with a previously developed crane selection and support design module and crane database developed by PCL Industrial Management Inc. (Hasan et al. 2009).

Proposed Methodology

The proposed methodology has been designed to carry out a heavy-vessel lift utilizing only one crane. This method utilizes a lifting mechanism where the lift operator needs to shorten one of the side slings to rotate the vessel into a vertical position before placement. Having independent control over both slings from the lifting point of the crane, the sling connected to the vessel top will start to pull (shorten) using a secondary load line suspended from the boom tip and running through a sheave mechanism to rotate the vessel until it reaches its final vertical position, as shown in Fig. 3. The body of the vessel is not designed to withstand any pressure from the cables while lifting. Therefore, to protect it from any damage, a custom spreader bar is utilized to provide a minimal clearing distance between the body of the vessel and the cables during the rotation of the vessel. In Fig. 4, the ends of the spreader bar (J) are connected to the vessel's lift lugs through fixed length slings (E and F). The spreader bar is also connected to the lifting point through two slings (S_f and S_v). The sling running to the tail lug (S_f) is of fixed length, and the sling to the top lift lug (S_v) is of variable length.

There are physical constraints related to the structure of the spreader bar that have to be observed while finding the proper configuration of sling lengths. One consideration includes the angle between the spreader bar (J) and the lifting slings (S_f and S_v), which needs a minimum of 25° to function. Any configuration of a known set of lengths for the slings and the spreader bar that will satisfy the lifting constraints is considered a solution. However, the objective of this methodology is to find the optimal configuration that will minimize the pulled length (shortening) of the variable sling (S_v) that is needed to complete the full rotation of the vessel from horizontal ($\alpha = 0°$) to vertical ($\alpha = 90^{\circ}$) and satisfy the following conditions:

$$\delta > 25^{\circ} \tag{1}$$

$$\gamma > 25^{\circ} \tag{2}$$

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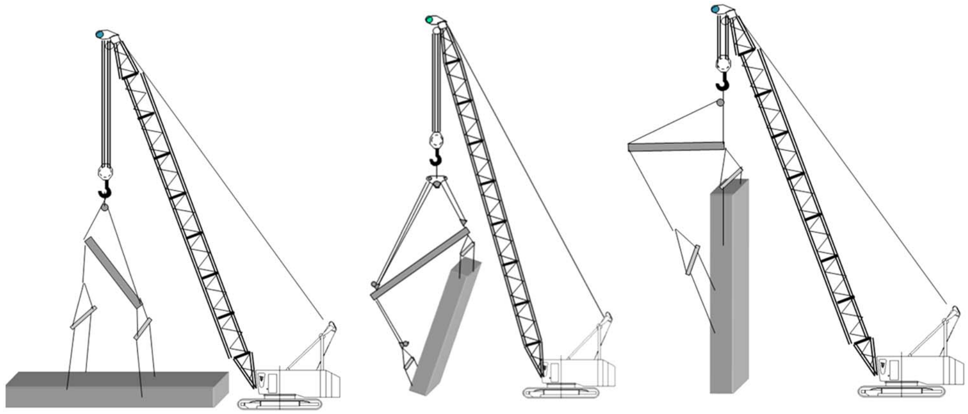

Fig. 3. Rotation processes of a long vessel using single crane

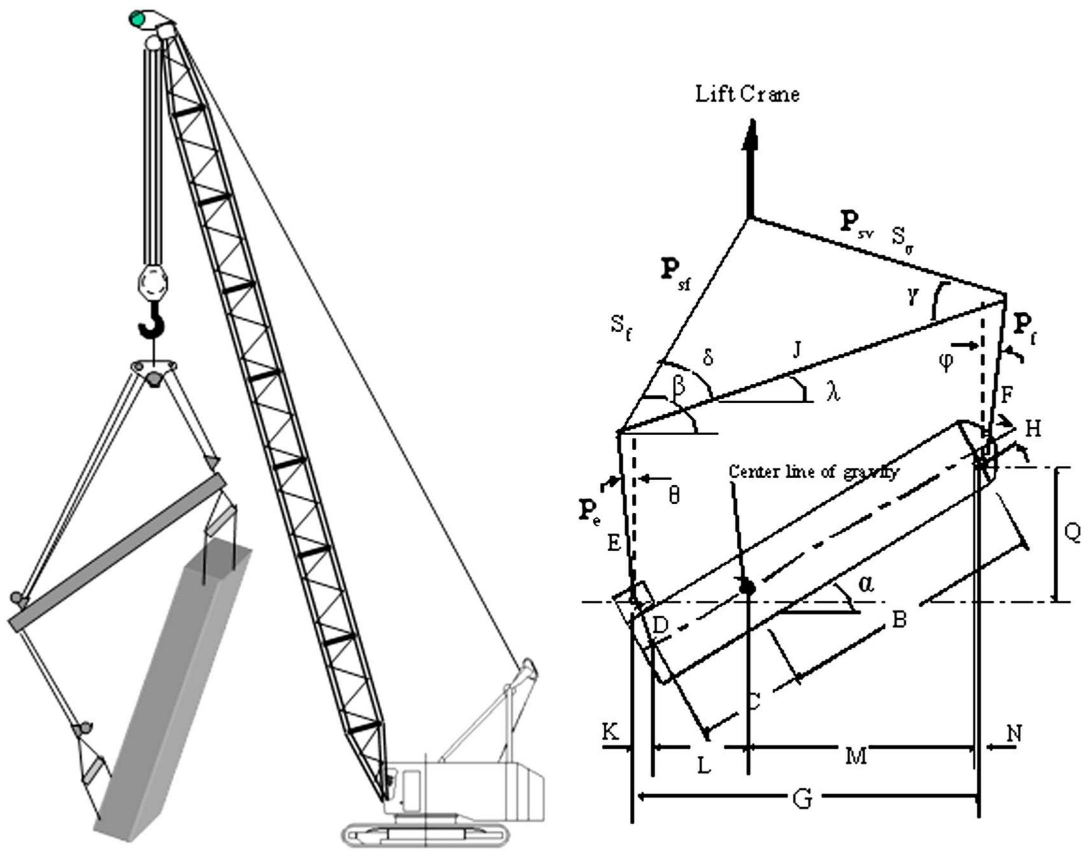

Fig. 4. Lifting mechanism configuration

$$15' < F < 25' \quad (3)$$

$$20' < S_v < 30' \qquad \text{at } \alpha = 90^\circ \qquad (\text{i.e., final vertical position}) \quad (4)$$

$$P_j > 0 \qquad (\text{i.e., always under compression}) \quad (5)$$

$$\text{Clearance between spreader bar and vessel} > 3' \quad (6)$$

where δ = angle between the spreader bar (J) and the fixed lifting sling (S_f); γ = angle between the spreader bar (J) and the variable lifting sling (S_v); F = length of the side sling F; S_v = length of the variable lifting sling; and P_j = force on the spreader bar (J).

The analysis handles three issues:

1. The outcome of the proposed solution: how close is it to the objective and what is needed to achieve it;
2. The lifting mechanism: how the variables are affected by its geometry; and
3. The approach of the proposed solution: how it is undertaken the solution and whether there are any alternate solutions.

The design analysis provides a solution for a given configuration. The calculated loads are static loads, and the rigging and

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spreader bars are designed per ASME B30.20 (ASME 2008), which requires a factor of safety of 5. Cranes are designed with a much lower factor of safety; thus, impact loading must be minimized. The goal is to find the optimal configuration of sling lengths that will satisfy the required set of constraints. The geometry of the lifting mechanism is a focal point in understanding the factors affecting its variables. The geometry can be described as a triangle on top of a 4-sided polygon. The spreader bar (J) and the two lifting slings (S_f and S_v) form the triangle, and the vessel with the two side slings (E and F) together with the spreader bar (J) forms a polygon. To perform force analysis and calculate stresses on the lifting elements, this geometry needs to have a unique, identifiable position in the x, y-coordinate system. The spreader bar is the common element for both the triangle and the polygon. The length of sling S_v in the triangle and the angles of the polygon sides θ and φ govern the coordinates of the spreader bar. The following analogy describes how the unknown variables are affected by the geometry:

1. The spreader bar (J) together with the side slings have a range of movement; therefore, the angles θ and φ are not unique but have a range of possible values limited by the lengths of the spreader bar and slings (E and F), as shown in Fig. 5.
2. The center of gravity of the vessel, while lifting, has to be aligned vertically with the hookup point. Therefore, the two lifting slings (S_f and S_v) connecting the spreader bar (J) with the hookup point will narrow down the range of movement. Since the variable lifting sling (S_v) has an unknown initial length, the range of θ and φ values is reduced, but not to a unique value.

This analogy leads to the belief that there might not be a single position for a given configuration; instead, there is a range of possible solutions whose boundaries have to be identified before choosing one element of that range to carry forward the rest of the calculations.

The main approach is to find out the values for θ and φ for each given position of the vessel and the maximum and minimum length of sling S_v. The position of the vessel is defined by its angle α with the horizontal axis. Angle α is given values from 0 to 90°, representing the full rotation of the vessel from its initial horizontal position to its final vertical position. The rotation of the vessel can be represented in an alternate way, starting with an initial length for the variable lifting sling (S_v) at the initial horizontal position of the vessel. Then, at incremental reductions of the length of sling S_v, the angle of the vessel is calculated, and the full rotation is represented.

The user needs to input an initial value of angle θ. Angle φ can be calculated using Eqs. (7) to (13):

$$K = D \sin\alpha \tag{7}$$

$$L = C \cos\alpha \tag{8}$$

$$M = B \cos\alpha \tag{9}$$

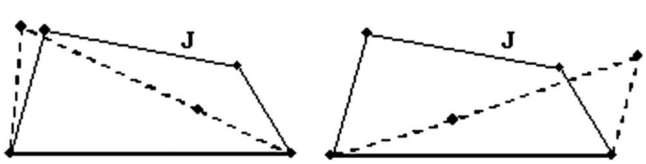

Fig. 5. Spreader bar movement limitation

$$N = H \sin\alpha \tag{10}$$

$$G = K + L + M + N \tag{11}$$

$$Q = (B + C)\sin\alpha - (H + D)\cos\alpha \tag{12}$$

$$\varphi = a\tan\{[(M + N)\tan(\theta - \alpha + \rho) + Q]/(K + L)\} - \alpha + \rho \tag{13}$$

where $\rho = a\tan[(D + H)/(C + B)]$

Angle α is the angle between the vessel and the horizontal axes, an input with any defined increment from 0 to 90°. All these variables have been displayed in Fig. 4.

Once the length of sling S_v has been determined, all angles could be calculated. Rotating the vessel from a horizontal position ($\alpha = 0^\circ$) to a vertical position ($\alpha = 90^\circ$) could be carried out mathematically by increment changes of angle α from 0 to 90° or by changing the length of sling S_v in increments from the minimum length ($S_{v\min}$) to the maximum length ($S_{v\max}$). Instead of representing the rotation of the vessel by incremental changes in angle α, it would be more convenient, from a calculations viewpoint, to do so by incremental changes in the variable length of sling S_v and then calculate angle α accordingly. The analysis concluded that there is a range of possible values for sling S_v while the vessel is at its initial horizontal position. Therefore, the first step would be to find the maximum and minimum limits of sling S_v, then choose an initial length from this range. The length of sling S_v can be calculated using Eqs. (14) to (17):

$$\beta = a\cos(K + L + E\sin\theta)/S_f \tag{14}$$

$$\lambda = a\sin(Q + F\cos\varphi - E\cos\theta)/J \tag{15}$$

$$\delta = \beta - \lambda \tag{16}$$

$$S_v = \sqrt{J^2 + S_f^2 - 2JS_f\cos\delta} \tag{17}$$

The acting force in each sling and in the spreader bar can be calculated using Eqs. (18) to (24):

$$\gamma = a\cos\left(\frac{J^2 + S_v^2 - S_f^2}{2JS_v}\right) \tag{18}$$

$$\tau = \gamma - \lambda \tag{19}$$

$$P_e = P_{\text{tot}}(M + N)/\text{Dist}/\cos(\theta - \alpha + \rho) \tag{20}$$

where $\text{Dist} = \sqrt{(D + H)^2 + (C + B)^2}$

$$P_f = P_{\text{tot}}(K + L)/\text{Dist}/\cos(\varphi + \alpha - \rho) \tag{21}$$

$$P_{Sv} = P_{\text{tot}}/(\cos\tau\tan\beta + \sin\tau) \tag{22}$$

$$P_{Sf} = P_{\text{tot}}\cos\tau/\cos\beta \tag{23}$$

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$$P_j = P_{Sf} \cos R + P_e \cos(\lambda + 90 - \theta) \qquad (24)$$

where P_e = force in side sling E; P_f = force in side sling F; P_{sf} = force in the fixed lifting sling (S_f); P_{sv} = force in the variable lifting sling (S_v); and P_j = force in the spreader bar (J).

All variables have been displayed in Fig. 4.

To determine the optimal configuration that will minimize the pulled length (shortening) of the variable sling (S_v) utilizing given equations is complicated. Thus, a system has been developed using Microsoft Visual Basic utilizing the proposed methodology. The purpose of the developed system is to ensure a user-friendly interface and to control data integrity for the users. The input parameters are the properties of the vessel, spreader bar length (trial), and all the slings' lengths (trial), except the length of S_v. The developed system will calculate the minimum and maximum lengths of S_v for optimal pulling length. The output results are the final length of all the slings and spreader bar and force distribution in the lifting elements. The output results of the developed system have been validated with the proposed methodology using Microsoft Excel. The system has been incorporated into another computer module, utilizing an algorithm developed previously for crane selection and ground pressure calculation (see Hasan et al. 2009) to select a best possible crane.

System Development

The system has been developed to calculate the maximum and minimum lengths of sling S_v (see Fig. 4) to complete the full rotation of the vessel from 0 to 90°. The design process considers the following criteria: (1) the angles δ and γ of the slings (S_f and S_v) with the spreader bar (J) must be greater than what the user has defined, e.g., 25°; and (2) the maximum and minimum forces on the slings should be an acceptable range defined by the user on the basis of the materials' properties. The user needs to input all the slings' lengths (except S_v), vessel dimensions, and the spreader bar's length to design the rotation, as shown in Fig. 6. All output should maintain the constraints; otherwise, the system will warn the user by highlighting the unsafe output in red. Also, the crane-selection module will not be activated unless the 90° rotation of the vessel is safe to do.

The user also has the option to observe the summary of design results by varying the individual lengths of slings E, F, and S_f or of the spreader bar (J), as shown in Fig. 6, by simply clicking on the buttons on each sling or spreader bar. For example, for the varying length of sling E, the user can view the design results as well as observe the graphics of the lifting position on that configuration by clicking on the corresponding length of sling E on the table (see Fig. 7). The user can observe all the design results and can select a safe configuration simply by double-clicking on any row on that table. The main form will display the design results for that configuration. The developed system provides default allowable limits for the forces acting on the spreader bar and each sling. The user can modify the allowable limits on the basis of manufacturers' specifications. When the output forces are unsafe, the user will be prompted to select a different configuration. If all the design results are under the allowable limit, the crane-selection option will be enabled. Changing the input configuration, such as increasing the size or weight of the vessel, will lead to new design results, and as long as all the design results are acceptable, the user can select a feasible crane. Obviously, increasing the weight of the vessel requires a higher-capacity crane.

The results provide the pulling length of the sling S_v and the forces acting on each sling and spreader bar. The user needs to select the type of spreader bar and slings that will be used, which

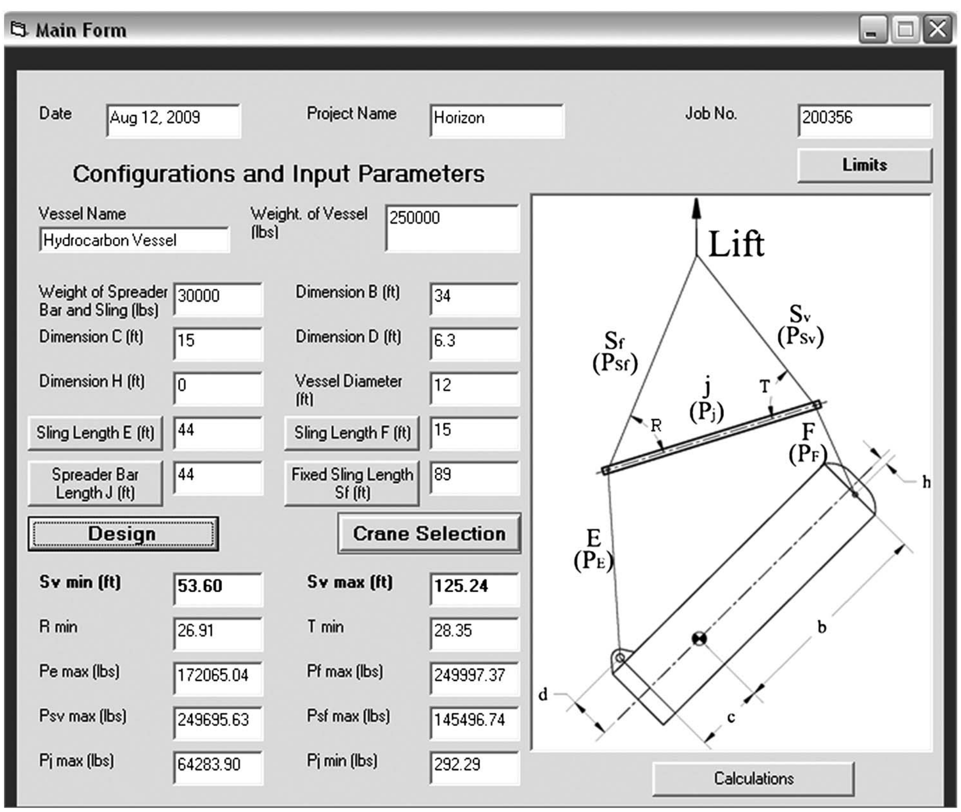
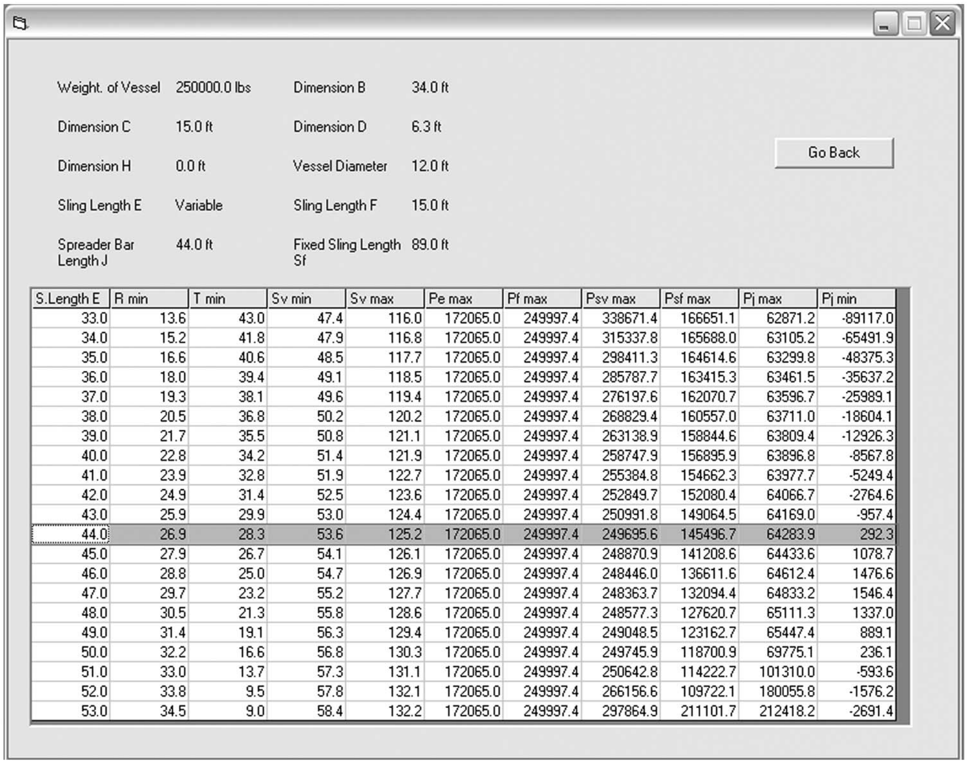


Fig. 6. Vessel lift configuration and design form



Weight. of Vessel 250000.0 lbs | Dimension B 34.0 ft
Dimension C 15.0 ft | Dimension D 6.3 ft
Dimension H 0.0 ft | Vessel Diameter 12.0 ft
Sling Length E Variable | Sling Length F 15.0 ft
Spreader Bar Length J 44.0 ft | Fixed Sling Length Sf 89.0 ft

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S.Length E	R min	T min	Sv min	Sv max	Pe max	Pf max	Psv max	Psf max	Pj max	Pj min
33.0	13.6	43.0	47.4	116.0	172065.0	249997.4	338671.4	166651.1	62871.2	-89117.0
34.0	15.2	41.8	47.9	116.8	172065.0	249997.4	315337.8	165688.0	63105.2	-65491.9
35.0	16.6	40.6	48.5	117.7	172065.0	249997.4	298411.3	164614.6	63299.8	-48375.3
36.0	18.0	39.4	49.1	118.5	172065.0	249997.4	285787.7	163415.3	63461.5	-35637.2
37.0	19.3	38.1	49.6	119.4	172065.0	249997.4	276197.6	162070.7	63596.7	-25989.1
38.0	20.5	36.8	50.2	120.2	172065.0	249997.4	268829.4	160557.0	63711.0	-18604.1
39.0	21.7	35.5	50.8	121.1	172065.0	249997.4	263138.9	158844.6	63809.4	-12926.3
40.0	22.8	34.2	51.4	121.9	172065.0	249997.4	258747.9	156895.9	63896.8	-8567.8
41.0	23.9	32.8	51.9	122.7	172065.0	249997.4	255384.8	154662.3	63977.7	-5249.4
42.0	24.9	31.4	52.5	123.6	172065.0	249997.4	252849.7	152080.4	64066.7	-2764.6
43.0	25.9	29.9	53.0	124.4	172065.0	249997.4	250991.8	149064.5	64169.0	-957.4
44.0	26.9	28.3	53.6	125.2	172065.0	249997.4	249695.6	145496.7	64283.9	292.3
45.0	27.9	26.7	54.1	126.1	172065.0	249997.4	248870.9	141208.6	64433.6	1078.7
46.0	28.8	25.0	54.7	126.9	172065.0	249997.4	248446.0	136611.6	64612.4	1476.6
47.0	29.7	23.2	55.2	127.7	172065.0	249997.4	248363.7	132094.4	64833.2	1546.4
48.0	30.5	21.3	55.8	128.6	172065.0	249997.4	248577.3	127620.7	65111.3	1337.0
49.0	31.4	19.1	56.3	129.4	172065.0	249997.4	249048.5	123162.7	65447.4	889.1
50.0	32.2	16.6	56.8	130.3	172065.0	249997.4	249745.9	118700.9	69775.1	236.1
51.0	33.0	13.7	57.3	131.1	172065.0	249997.4	250642.8	114222.7	101310.0	-593.6
52.0	33.8	9.5	57.8	132.1	172065.0	249997.4	266156.6	109722.1	180055.8	-1576.2
53.0	34.5	9.0	58.4	132.2	172065.0	249997.4	297864.9	211101.7	212418.2	-2691.4

Fig. 7. Design results for varying sling length *E*

depends on the availability in the construction market. Manufacturers usually provide the maximum allowable stresses for their spreader bars and slings. Utilizing the ASME B30.20 design method with a factor of safety of 5, the user can determine the width and thickness for the spreader bar and diameter for the slings. One limitation of the system is to avoid overloading the secondary load line used to modify the variable length sling S_v as the vessel approaches the vertical position. For very long and heavy vessels, the design and capacity of the spreader bar becomes impractical and the added weight reduces the crane capacity.

To select the best possible crane to perform the given lift, the developed system has been incorporated into an existing computer

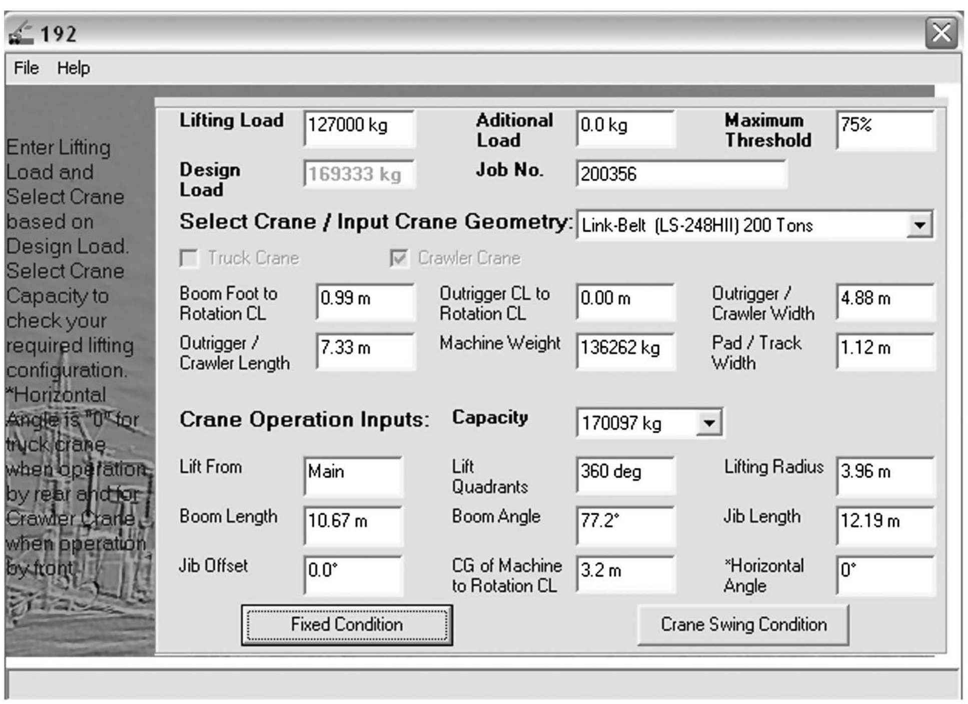

Fig. 8. Crane-selection module

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module (see Hasan et al. 2009). This module has been designed to assist practitioners in the selection of feasible cranes for particular construction site operations from a list of cranes available in a crane database developed by PCL Industrial Management, Inc. Mobile crane geometric specifications, load information, and other data supplied by the manufacturer for 195 different cranes have been stored in this database. The developed system assists the user in the selection of a technically feasible crane to lift and rotate the vessel in the calculation. By simply clicking on the "Crane Selection" button, the crane-selection module will be displayed, as shown in Fig. 8, and will retrieve all the required components from this newly developed "off-the-ground rotation" system. The user can select the best possible crane and calculate the ground pressure using the crane-selection module (see Hasan et al. 2010).

Fig. 9. Traditional 2-crane lift of long vessel

Table 1. Output Results from the Proposed System

Parameter	Length (m)
Spreader bar length	14
Sling length E	12.2
Sling length F	4.6
Sling length S_f	26.8
Sling length S_v min	15.3
Sling length S_v max	37.8
Pulling length	22.5

Case Example

The case considered involves the proposed installation of 30, 100-ton long vessels in one month at an oil sands project in Fort McMurray, Alberta, Canada. Because of the horizontal shipment of the vessel, two cranes would be needed traditionally to lift, rotate, and place the vessel in the desired location (see Fig. 9). An experienced lift engineer has chosen a 300-ton lift crane and a 230-ton tail crane that satisfy all the constraints, e.g., capacity, height, clearance.

The output results of applying the proposed methodology to lift the long objects using the developed system are given in Table 1.

The cost comparison of the traditional method with the proposed method is presented in Table 2.

Implementing the proposed methodology can save 40% compared with the 2-crane lift. Traditionally, complete installation of a long vessel from its horizontal position to a vertical position requires almost a day because of the extra care required for the side-loading effect. Applying the proposed methodology, the majority of time will be required to pull the cable; in this case example the pulling length is 22.5 m (see Table 1). Considering other activities, such as hooking-up, swinging, positioning, and unhooking, the total lifting operation of a long vessel can be expected to complete in less time than traditional 2-crane lift method. Also, there will be savings of space, since a tail crane is not required.

Conclusion

This paper presents the implementation of a new method for lifting heavy, large vessels that is uncommon to current industry practices, with an objective to optimize the lifting configuration. The developed methodology provides not only for the geometry of the lifting mechanism, but also for the force distribution in the lifting elements. The use of the developed tool has led to a belief that a better configuration is achieved when the spreader bar is allowed to carry tension. It is suggested that the spreader bar could be constructed from pipes and cables to sustain tension, since cables will be designed to carry the full load. The calculations of forces carried out in this methodology are all done on the basis of static analysis and equilibrium; the dynamic effect attributable to the movement during the lifting process has not been taken into consideration, as it is dealt with when developing the engineered lift study. The calculations of angles have assumed constant lengths for all members. The proposed system provides the geometric configuration of and the forces acting on the lifting elements. On the basis of these forces, the user needs to select the material type and thickness of the spreader bar and slings following standard industry practice.

The system has been developed using Microsoft Visual Basic and incorporated with a previously developed crane-selection and ground pressure calculation module to assist the user in the

Table 2. Comparison between the Traditional and Proposed Method

Type	Traditional 2-crane lift			Proposed single crane lift		
	Lift crane	Tail crane	Crane operator	Lift crane	Spreader bar	Crane operator
Required number	1—Demag CC1800	1—Manitowoc 4100	2	1—Demag CC1800	1 (14-m length)	1
Rental cost	$100,000	$65,000	included	$100,000	$5,000	included
Mobilization/demonstration	$80,000	$60,000	—	$80,000	—	—
Ground preparation	$7,500	$7,500	—	$7,500	—	—
Total cost		$320,000			$192,500	
Total savings			$127,500 (40%)			

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selection of the best possible crane for lifting heavy and long objects. The key success of this proposed methodology is to eliminate the requirement of a second crane for tailing operation to lift a heavy vessel. Through automating and optimizing heavy-lift crane configurations following this proposed methodology, the construction industry can enhance the planning process and reduce costs and time related to vessel lifting operations.

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