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Stability Margins of Heavy-Lifting Machines with a Telescoping Boom and Jib

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***Abstract*— Machines that use a telescoping boom with an attached jib to raise loads to great heights have deadly tip-over hazards. To keep machines safely away from tip-over conditions, machine producers provide a variety of countermeasures, such as outriggers that increase the width of the stability base, counterweights, configuration sensors, control input smoothing, and control computers that stop the machines from moving out of the stable envelope of reachable positions. The computers are programmed to have a stability margin that restricts machine motion well within the envelope of actual stability. These stability margins are set by industry standards that limit the allowable payload weight. However, margins created by payload limits do not provide good margins for other machine parameter variations and configuration errors. This paper calculates stability margins that are not considered by industry standards. The results indicate that these neglected stability margins can be both small and inconsistent throughout the reachable workspace. Therefore, telescoping-boom machines with attached jibs pose safety hazards that are neither well understood, nor adequately addressed by industry standards.**

I. Introduction

All long-reach heavy-lifting machines, such as mobile boom cranes and aerial lifts, possess deadly tip-over hazards. Commercially-available crawler cranes can reach over 500-feet high, and the tallest in the world, the Sarens SGC-250, has a maximum height of over 800 feet [1]. When a tall crane tips over, the operator and nearby workers can be severely injured or killed, while others further away are also at risk from the falling boom and payload.

In order to improve safety, the mechanics of lifting machines and the factors contributing to tip-over events must be better understood. There have been several previous investigations of the tip-over stability of cranes. Neitzel *et al.* [2] reviewed available information on crane-related injuries, and gave recommendations for improving crane injury prevention. Jeng, Yang, and Chieng [3] introduced two indices, a moment-index and a force-index, to quantify the tip-over behavior of mobile cranes. The force-angle stability measure in [4] provides an indication of proximity to tip over. Other research has investigated the development of optimization frameworks for lift-path planning [5] and generating safe lift plans for multiple concurrent heavy lift operations [6]. Towarek [7] investigated the dynamic stability of a boom crane on a flexible soil foundation.

When heavy payloads are moved by the crane, the payload oscillations can have a significant influence on the stability of cranes. In [8], payload swing caused by base excitation was investigated and limited by reeling and unreeling the hoisting cable. Shaping the commands sent to the crane has proven to be an effective method of reducing payload swing [9]–[12]. The tip-over stability of a mobile crane due to payload oscillations was investigated in [13]. The comparison between the static stability and the full-dynamic stability revealed that a simple semi-dynamic analysis provides good approximations for the tip-over stability properties.

Fig. 1. Telescoping Boom Crane with Attached Jib.

A previous analysis of the stability margins of lifting machines with telescoping booms was presented in [14]. This paper uses a similar model and stability analysis approach, but augments the model to include an attached jib and analyzes the effects of the jib angle on the stability margins.

A picture of a large mobile telescoping-boom crane with a jib is shown in Fig. 1. A boom crane uses a boom that pivots up and down in a vertical plane (luffing) and rotates about a vertical axis (slewing). The payload is hoisted up and down by a cable attached to the end of the boom. These motions (luffing, slewing, and hoisting) allow the operator to position the payload anywhere in the workspace.

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A. Tip-Over Mitigation Through Mechanical Means

Tip-over mitigation methods include using outriggers to widen the base of support, installing counterweights, using configuration sensors, providing control input smoothing, and integrating control computers that restrict machine configurations to within the stable envelope of positions for the measured load.

Counterweights provide balancing forces so that the machine does not tip over as it moves heavy payloads. As such machines get taller, the counterweight masses increase significantly, thereby making the machines more expensive, harder to transport, and more difficult to erect. Heavier cranes also require extensive ground preparation [15]. In order to decrease the required counterweight, manufacturers have developed cranes with movable counterweights. Movable counterweights for heavy machinery have existed for nearly a century [16].

B. Tip-Over Mitigation Through Stability Margins

In addition to mechanical means for reducing tip-over risk, computers are used to monitor the machine configuration and load. The computer can restrict machine motion to well within the envelope of the actual tip-over stability. The size of these stability margins is mandated by industrial standards. For example, the rated load capacity of mobile cranes are limited to 85% of the actual tipping load by the ASME B30.5 standard.

The load-percentage limit provided by the ASME B30.5 standard does not provide consistent stability margins in terms of how much machine configuration error can be tolerated. For example, if a boom crane is in a certain configuration, the 85% limit may allow the boom to lower down 8 degrees past the angle limit prescribed by the 85% load limit before tipping actually occurs. However, in a different configuration, the same crane may only be able to travel 5 degrees past the boom angle limit before tipping.

This paper presents a method for calculating the actual stability envelope and the stability margins for a telescoping-boom heavy-lifting machine with an attached jib. Application of the ASME B30.5 industry standard load limit is investigated and shown to give inconsistent stability margins, particularly for longer booms.

C. Paper Contributions

Section II presents a parameterized model of a telescoping-boom lifting machine with an attached jib. Section III describes different types of stability margins. Parameter-based stability margins are obtained by applying the load-percentage limit from the ASME B30.5 standard. The results show that the industry standard yields inconsistent, and small, stability margins when a telescoping boom is greatly extended. Finally, conclusions and future work are discussed in Section IV.

II. Tip-Over Stability Based on Sum of Moments

Heavy-lifting machines are subject to forces from gravity, wind, payload motion, ground undulations, *etc*. Tip-over stability can be investigated by calculating the sum of moments created by the various forces about a corresponding tip-over axis. A baseline model of a boom truck with a telescoping boom, jib, and payload is considered. The outside edges of the wheels, tracks, or outriggers that support the machine form possible tip-over axes. For the machine to be stable, the sum of the moments about any possible tip-over axis must be directed toward the base of the machine.

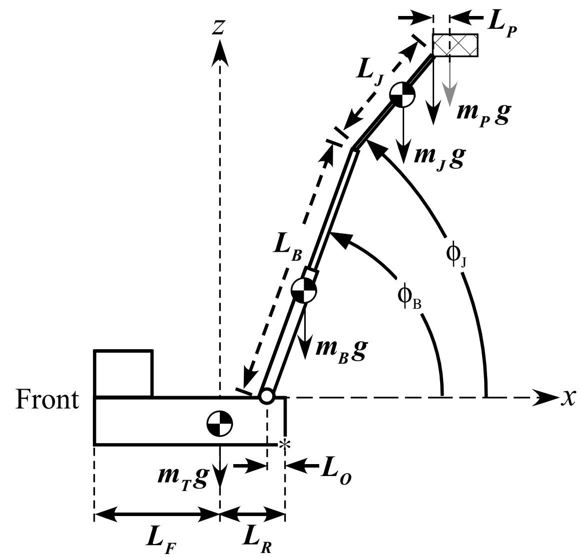

Fig. 2. Schematic Diagram of a Telescoping Boom Truck with Jib

A schematic model of a telescoping-boom truck with an extension jib is illustrated in Fig. 2. The model can represent a payload suspended from the tip of a crane jib or an aerial lift personnel platform. In order to establish baseline stability properties, the machine is assumed to be on level ground with no wind or other external loading such that only the weight forces of all components contribute to the tipping moment. This model is similar to that used in [14], augmented with a jib on the end of the boom.

The base of the model is a chassis supported by outriggers whose total mass is m_T, and the center of mass of the base is located a distance L_F from the front (shown on the left) and a distance L_R from the rear of the machine. A telescoping boom is mounted at an offset distance L_O from the rear. The mass of the boom is m_B and its extended length is L_B. Due to the telescoping nature of the boom, its center of mass is generally located at less than half its extended length. The angle that the boom makes with the horizontal ground plane is denoted as ϕ_B.

The mass of the jib attached to the end of the boom is m_J and its length is L_J. The angle that the jib makes with the horizontal ground plane is denoted as ϕ_J. At the end of the jib there is a payload or work platform whose mass is denoted as m_P. When a platform is attached, the mass center of the platform extends an additional horizontal distance L_P beyond the end of the jib.

The moment generated by the forces acting on the different crane parts about a certain axis is:

$$\vec{M}_{ij} = \vec{a_j} \cdot (\vec{r_i} \times \vec{f_i}) \tag{1}$$

where $i = T, B, J, P$ and $j = 1, 2, 3, 4$. $\vec{M}_{ij}$ is the moment generated by the force $\vec{f_i}$ about a unit vector $\vec{a_j}$ along the j^{th} tip-over axis. $\vec{f_i}$ is the force acting on body i at its gravitational center. $\vec{r_i}$ is a position vector from any point on the tip-over axis to any point on the force's line of action.

The individual moments found using (1) are combined to get the total moment about each tip-over axis:

$$\vec{M_j} = \sum_{i=1}^{4} \vec{M_{ij}} = \sum_{i=1}^{4} \vec{a_j} \cdot (\vec{r_i} \times \vec{f_i}) \quad (2)$$

A. Backward Tipping

Considering only tipping about the rear tip-over axis and assuming that the boom is pointed directly backward, the tipping-moment can be expressed using the variables in Fig. 2. The stabilizing moment provided by the base is:

$$M_T = -L_R m_t g \quad (3)$$

where g is the acceleration due to gravity.

The tipping moment caused by the boom is:

$$M_B = [L_B \cos(\phi_B) Boom_{cg} - L_o]\, m_B g \quad (4)$$

where $Boom_{cg}$ is the location of the boom center of gravity in terms of the percentage of the boom length. For example, $Boom_{cg} = 0.4$ means that the boom center of gravity is located at 40% of the boom length.

The tipping moment caused by the jib is:

$$M_J = [L_B \cos(\phi_B) - L_o + L_J \cos(\phi_J) Jib_{cg}]\, m_J g \quad (5)$$

where Jib_{cg} is the location of the jib center of gravity in terms of the percentage of the jib length.

The tipping moment caused by the platform or payload is:

$$M_P = [L_B \cos(\phi_B) - L_o + L_J \cos(\phi_J) + L_P]\, m_P g \quad (6)$$

The total tipping moment is then given by:

$$M_{rear-tip} = M_T + M_B + M_J + M_P \quad (7)$$

For a stable configuration, the load moment that acts to tip the machine should be less than or equal to the maximum moment of the machine that prevents tipping. Therefore, a given machine configuration is stable when the total moment about the rear tip-over axis, $M_{rear-tip}$, is negative.

III. Parameter-Based Tip-Over Stability Margins

An important safety consideration is a machine's proximity to a tip-over condition. The proximity to tipping can be evaluated in several ways to establish a stability margin. One method for characterizing the stability margin considers the additional load weight that will induce a tip (e.g., the ASME B30.5 standard), the change in boom length that will cause a tip, the change in boom and jib angles that will induce a tip, *etc*. A goal of this paper is to investigate and evaluate the utility of information provided by various types of stability margins used to ensure heavy-lift machines operate safely.

The ASME B30.5 industry standard for cranes creates a stability margin by limiting the rated load capacity to, at most, 85% of the tipping load. Such a limitation allows the actual load to exceed the rated capacity by 17.6% $(1/0.85 = 1.176)$ tip-over occurs. The 17.6% load weight margin is meant to accommodate imperfect conditions, such as not operating on perfectly level ground, operating in the presence of wind loads, excessive payload swing, *etc*. Often, such load-based stability margins do not provide consistent safety margins for many real-world variables and they do not consider any parameters beyond the load weight.

TABLE I
Parameters of Telescoping Boom and Jib Machine

Parameter	Nominal Value	Maximum	Minimum
m_T	40,000 lbs	50,000	30,000
m_B	18,000 lbs	20,000	15,000
m_J	2,000 lbs	2,500	1,500
m_P	1,000 lbs	3,000	500
L_R	15 ft.	20	10
L_B	150 ft.	200	100
L_J	30 ft.	50	25
L_P	2 ft.	5	0
L_O	5 ft.	10	0
$\phi_J - \phi_B$	0 degrees	30	-150
$Boom_{cg}$	40%	50	30
Jib_{cg}	50%	50	30

In order to be useful, stability margins should be based on the kinds of uncertainties, disturbances, and use cases associated with the machines being considered. For example, when evaluating stability of telescoping boom trucks with an attached jib, there is potential for uncertainty, or error, in the boom extension length, the length of the jib, the angle of the boom, the angle of the jib, and the weight of the attached payload. Therefore, this section presents parameter-based stability margins for telescoping boom trucks with an attached jib directly based on such important parameters. Furthermore, these stability margins are compared to the ASME B30.5 standard.

Using the analysis presented above in Section II, telescoping boom machine configurations that result in tip over can be calculated. Such a study provides the stability "envelope" that determines safe and unsafe operating conditions. A parameter-based stability margin can be obtained by calculating the parameter change required for the machine to reach an unstable configuration. For example, assume a machine has a certain boom extension length, boom angle, jib length, jib angle, and payload weight. If the boom angle must be lowered 5 degrees to cause tip-over, then the boom-angle stability margin is 5 degrees for that particular configuration.

Other important configuration parameters, such as jib angle and boom extension length, can be used to calculate parameter-based stability margins. A safe operating envelope with reliable and consistent stability margins can be established by calculating parameter-based margins for a machine throughout its reachable workspace and as a function uncertain parameters.

The calculation of parameter-based stability margins are demonstrated in this paper for a telescoping boom truck with an attached jib that has its boom pointed directly backward, where the tipping moment equations for this system were presented in Section II-A. Table I lists the nominal parameter values and their maximum and minimum values used in this analysis to explore the stability margins.

Similar to the modeling approach in [14], the locations of the centers of gravity of the boom and jib are defined using a percentage distance along the structure where the cg is located, and the nominal values are shown in the bottom two rows of Table I. The nominal value for the jib corresponds to a single-section jib whose cg is near its center, or a fully retracted telescoping jib whose cg would also be near its center. For the telescoping boom, the nominal value represents a case where the inner, smaller sections are extended to the top of the boom, and since they do not have as much mass as the larger outer sections, the cg is closer to the base of the telescoping boom. Also, the relative jib angle, defined as the difference between the absolute boom and jib angles shown in Fig. 2, or $\phi_J - \phi_B$, is listed in the table. Its nominal value of 0 degrees corresponds to the case when the jib is aligned with the boom ($\phi_J = \phi_B$).

A. Boom Angle Margins

The tipping moment was calculated as a function of the boom angle for various boom extension lengths ranging from 40 ft. to 150 ft. with the jib aligned with the boom ($\phi_J = \phi_B$). The resulting tipping moments are shown in Fig. 3. For these results, the nominal values given in Table I were used, except for the boom cg location. For the 40 ft. case, the boom was modeled as fully retracted, and the cg was placed in the middle of the boom with $Boom_{cg} = 50\%$. At the nominal extended length of 150 ft., the value of $Boom_{cg}$ is 40%. The values of $Boom_{cg}$ for intermediate boom lengths were obtained by interpolation.

Fig. 3 shows that the tipping moment increases as the boom angle decreases. When the boom is elevated to high angles close to vertical, the machine is very stable because the boom and jib do not extend much outside the base of the support. However, the destabilizing tipping moments increase as the boom angle is lowered and the boom and jib extend further and further outside the base of support. Also, the tipping moment switches from negative to positive as the boom is lowered for most of the boom lengths, indicating tip over will occur. The tipping angle for the nominal case is 63.76 degrees. Furthermore, the machine becomes more stable and has a much lower tipping angle as the boom length decreases because the cg of the boom and jib move back toward the base of support. The boom angle can be lowered to 0 degrees without causing tipover for boom lengths less than approximately 50 ft.

In order to find the boom angle that will limit the machine to the 85% load limit imposed by the industry standards, the payload mass must be increased from 1,000 lbs. to 1,176 lbs. (85% of 1,176 = 1,000) and then a similar sweep can be performed to find the boom angles at which the machine will tip over. For the nominal case, the tipping angle corresponding to 1,176 lbs. was calculated to be 64.29 deg. Therefore, when the machine has a payload of 1,000 lbs., the boom angle must be limited to values greater than or equal to 64.29 degrees in order to satisfy the industry standard. Note that while this limitation provides for a 17.6% margin in terms of payload overload, it only proves a 0.53 degree margin in boom angle for this configuration. That is, if the

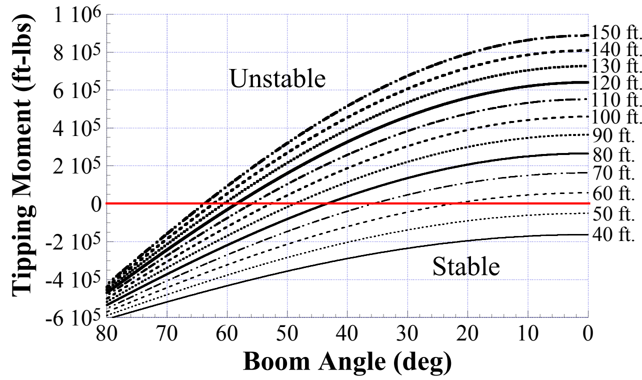

Fig. 3. Tipping Moment as a Function of Boom Angle and Length

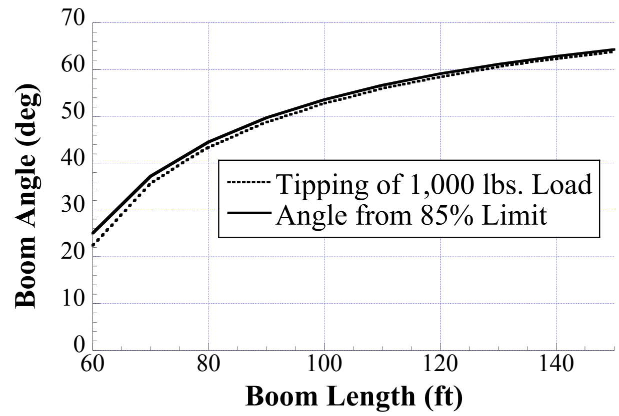

Fig. 4. Tipping Angle as a Function of Boom Extension Length

boom angle is 0.53 degrees lower than measured by the boom sensor, then the machine will tip over. If this boom-angle margin is expressed as a percentage of the 64.29 boom angle limit, then the margin is only 0.8%.

The above result indicates that the industry standard load-based margin provides only a very small boom-angle margin. The 0.53 degree boom-angle margin results from the specific nominal parameters used above that correspond to a fairly long boom. It is of interest to calculate the boom-angle margin as a function of the telescoping boom length, as well as other parameters such as the jib angle and length.

1) Boom Angle Margin vs. Boom Extension Length: The results in Fig. 3 were used to identify the tipping angle for each boom length, and the resulting tipping angles are shown in Fig. 4. The allowable boom angle given the 85% load restriction is also shown in the figure. The nominal case examined above, where the 85% limit only provided a 0.53 degree boom angle margin, is shown on the right side of the figure. As the boom length decreases, the 85% limit does provide more boom angle margin, as shown in Fig. 5. However, even when the boom is retracted all the way to 60 ft., the boom angle margin increases to only 2.6 degrees.

The 85% load limit fails to provide significant boom-angle margin because the payload weight is only a small contributor to the destabilizing moments applied to the machine. The long boom and jib are the main contributors, so limiting the payload weight does not address the primary factors creating tip-over moments. For example, for the nominal case in Table

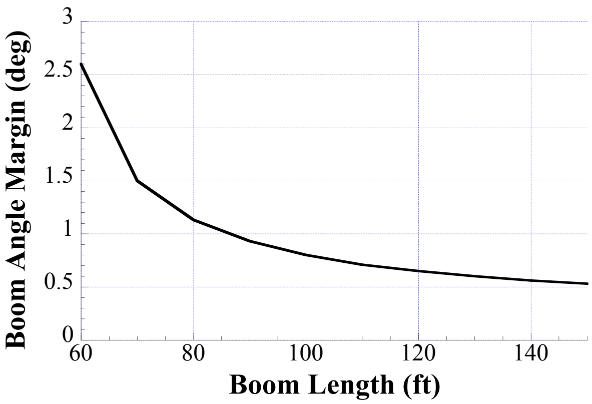

Fig. 5. Boom Angle Margin as a Function of Boom Extension

TABLE II
MOMENT CONTRIBUTIONS FOR NOMINAL CASE NEAR TIP-OVER

Component	Moment Contribution (ft-lbs.)	% of Total
Truck Chassis	-5,886,000	-100
Boom	3,801,400	64.6
Jib	1,333,200	22.7
Payload	751,290	12.8

I, the moment contributions from the four components at the tip-over angle are shown in Table II. The right column shows that the payload contributes less that 13% of the destabilizing tipping moment. Therefore, limiting its weight to 85% of the value that induces tipping only reduces the total tipping moment by a small amount. On the other hand, the boom contributes nearly 65% of the tipping moment, so even small changes in the boom parameters, such as its angle or length, result in large changes in the tipping moment.

The nominal case with a long boom and a light payload is not an unimportant anomaly lying at the edge of the performance envelope. Rather, it is precisely the type of conditions that occur when machines with telescoping booms are performing their most dangerous duty - that of lifting to high heights. When telescoping booms are used as aerial lifts, the payload weight, comprised of the personnel platform, the workers, and their tools and supplies, is often near the 1,000 lbs. value considered in the nominal case. Furthermore, the industry standard limit provides only a very small boom-angle margin when the telescoping boom is very long and the payload is relatively light. Unfortunately, these low-margin configurations correspond to some of the most dangerous operating conditions.

To demonstrate the link between lighter payloads and the poor stability margins caused by the 85% load limit, the boom-angle stability margin for heavier payloads can be examined. The tipping angle as a function of the boom extension length was calculated for payload weights of 2,000 lbs. and 3,000 lbs, as well as for the payloads that yields 2,000 lbs. and 3,000 lbs as their 85% load limits, or payloads of 2,353 lbs. and 3,529 lbs., respectively. The boom-angle margins for the three payloads are shown as a function of the boom extension length in Fig. 6. Because these load margins of 353 lbs. and 529 lbs. are larger than the 176 lbs. margin for the 1,000 lbs. payload, the resulting boom-angle margins are larger.

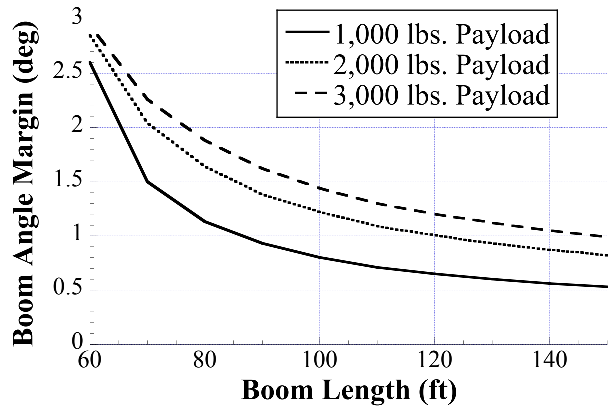

Fig. 6. Boom Angle Margin as a Function of Boom Extension and Payload Mass

The results in Fig. 6 show that the 85% load capacity creates progressively larger boom-angle margins as the payload increases. In other words, the 85% limit imposed by the ASME standard provides increasing safety margins in terms of boom angle errors as the payload weight increases. On the other hand, the boom-angle margin gets progressively smaller as the payload weight decreases, and boom angle errors or effects that decrease boom angle could result in tipping.

The root-cause of this trend is that as payloads get heavier, they comprise a correspondingly larger percentage of the tipping moment. This can be illustrated by comparing the moment contributions of the 1,000 lbs. load given previously in Table II to the corresponding moment contributions for heavier payload masses. Fig. 7 shows the tipping moment contributions for a range of payload masses, and the moment contribution percentages are labeled for the 1,000, 2,000, and 3,000 lbs. payload cases examined above. The payload contribution to the tipping moment increases from only 12.8% to 22.9% when the payload increases from 1,000 lbs. to 2,000 lbs, then further increases to 31.2% for a 3,000 lbs. payload. Overall, Fig. 7 illustrates how the payload tipping moment contribution increases for heavier payloads. Therefore, when the 85% limit is applied to the heavier payloads, it has the effect of limiting a larger percentage of the overall tipping moment and provides a correspondingly larger boom-angle margin.

2) Boom Angle Margin vs. Relative Jib Angle: With other parameter values held constant at the nominal values, the tipping moments as a function of the relative jib angle for a range of boom angles between 60 and 70 degrees are shown in Fig. 8. Boom angles below 62 degrees lead to tipping regardless of the jib angle, whereas boom angles between approximately 62 and 65 degrees can lead to tipping for certain jib angles.

The tipping moments are less sensitive to changes in the relative jib angle for the nominal parameter values than they are to the boom extension length, as was shown in Fig. 3. Therefore, the critical boom angles that induce tipping only vary slightly as the relative jib angle is changed. Fig. 9 shows the critical tipping boom angle as a function of the relative jib

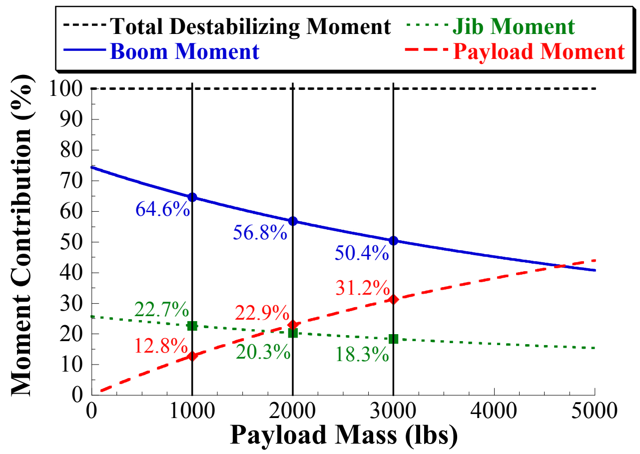

Fig. 7. Boom, Jib, and Payload Contributions to the Destabilizing Moment at Tip-Over vs. Payload Mass

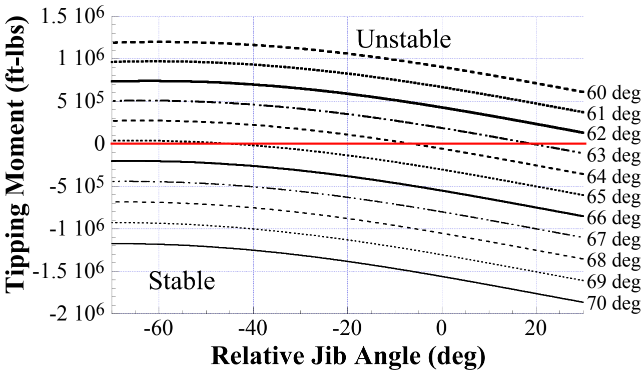

Fig. 8. Tipping Moment as a Function of Jib Angle for a range of Boom Angles with a 1,000 Lbs. Payload

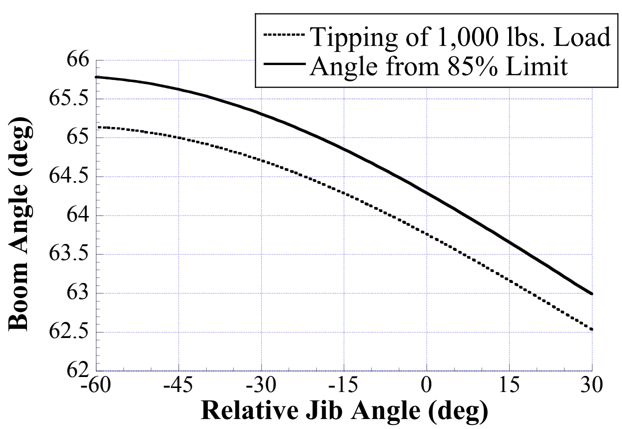

Fig. 9. Tipping Boom Angle as a Function of Relative Jib Angle

angle. The allowable boom angle for the 85% load restriction is also shown in the figure. For smaller and positive relative jib angles, the boom can be lowered slightly further (up to approximately 2.5 degrees lower) without causing tipping.

Fig. 10 shows the boom angle margins for the 85% load limits for three payload masses as a function of the relative jib angle. The boom angle margins change as a function of the relative jib angle and higher payload masses lead to larger boom angle margins, like for the boom length. Also, the boom angle margin decreases for smaller relative jib angles.

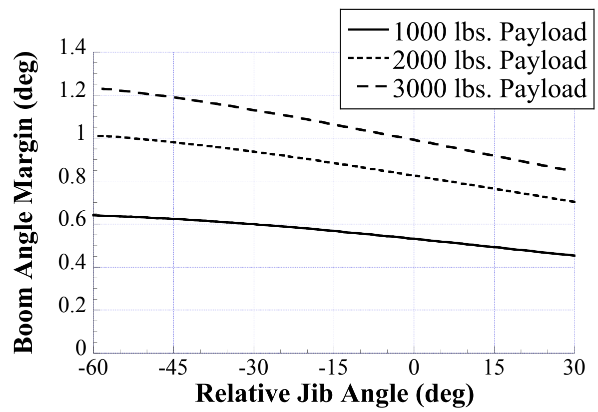

Fig. 10. Boom Angle Margin as a Function of Relative Jib Angle and Payload Mass

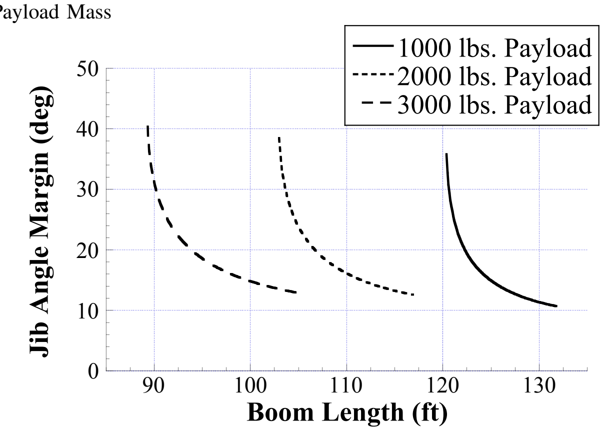

Fig. 11. Jib Angle Margin as a Function of Boom Length and Payload Mass for a Boom Angle of 60 degrees

B. Jib Angle Margins

A jib angle margin can be defined as the difference between the relative jib angle that results in tipping for the 85% limit and that for the rated payload mass. The jib angle margins for the 85% load limits for 3 payload masses and with a boom angle of 60 degrees are shown in Fig. 11 as a function of the boom extension length. For certain boom lengths to the left of the curves for each payload mass, applying the 85% load limit results in predicting tip-over, but using the rated payload mass does not result in tipping at those lengths for any jib angle. Therefore, the jib angle margin is undefined. For certain boom lengths to the right of the curves for each payload mass, applying the 85% load limit always results in predicting tip-over, unless the relative jib angle margin is able to be greater than 30 degrees (the maximum relative jib angle considered in the stability analysis).

Fig. 11 shows that the load-based stability margin defined by the industry standard is much less sensitive in terms of the jib angle than it is for boom angle, at least for the nominal system parameters under consideration. With the nominal system parameters and a 60-degree boom angle, a jib angle sensor would need to have a large error of at least 10 degrees for the machine to unexpectedly tip over with the payload masses shown. However, like the other stability margins described in this paper, the margins change as a function of the system configuration.

IV. Conclusions and Future Work

Heavy-lifting machines with long, telescoping booms and jibs have complex stability properties that change significantly as the machine configuration changes. The industry standards used to design such machines attempt to ensure adequate stability margins by limiting the allowable payload to a fixed percentage of the actual payload weight that would induce a tip-over event. Unfortunately, the industry standards fail to ensure adequate safety margins to other possible variations in the machines, such as boom angle and jib angle. In fact, the stability margins actually decrease in size as the working height of the machines are extended higher. Therefore, the results presented here show that the industry standards provide the least stability margin precisely when the machines are in the most dangerous configurations.

References

[1] L. Barnard, "'Big Carl': How to handle the world's biggest crane at Hinkley Point," https://www.internationalcranes.media/news/-big-carl-how-to-handle-the-world-s-biggest-crane-at-hinkley-point/8015165.article, September 29 2021, Date Accessed: May 9 2023.

[2] R. L. Neitzel, N. S. Seixas, and K. K. Ren, "A review of crane safety in the construction industry," *App. Occupational and Environmental Hygiene*, vol. 16, no. 12, pp. 1106–1117, 2001.

[3] S. Jeng, C. Yang, and W. Chieng, "Outrigger force measure for mobile crane safety based on linear programming optimization." *Mechanics Based Design of Structures and Machines*, vol. 38, no. 2, pp. 145–170, 2010.

[4] E. Papadopoulos and D. Rey, "A new measure of tip-over stability margin for mobile manipulators," in *Int. Conf. on Robotics and Automation*, 1996, pp. 3111–3116.

[5] J. An, M. Wu, J. She, and T. Terano, "Re-optimization strategy for truck crane lift-path planning," *Automation in Construction*, vol. 90, pp. 146–155, 06 2018.

[6] A. Mousaei, H. Taghaddos, S. M. Bagheri, and U. Hermann, "Optimizing heavy lift plans for industrial construction sites using Dijkstra's algorithm," *Journal of Construction Engineering and Management*, vol. 147, no. 11, p. 04021160, 2021.

[7] Z. Towarek, "Dynamic stability of a crane standing on soil during the rotation of the boom," *International Journal of Mechanical Sciences*, vol. 40, no. 6, pp. 557–574, 1998.

[8] E. Abdel-Rahman and A. Nayfeh, "Pendulation reduction in boom cranes using cable length manipulation," *Nonlinear Dynamics*, vol. 27, no. 3, pp. 255–269, 2002.

[9] G. P. Starr, "Swing-free transport of suspended objects with a path-controlled robot manipulator," *ASME Journal of Dynamic Systems, Measurement, and Control*, vol. 107, pp. 97–100, 1985.

[10] A. Khalid, J. Huey, W. Singhose, J. Lawrence, and D. Frakes, "Human operator performance testing using an input-shaped bridge crane," *Journal of Dynamic Systems, Measurement and Control*, vol. 128, pp. 835–841, 2006.

[11] D. Kim and W. Singhose, "Performance studies of human operators driving double-pendulum bridge cranes," *Control Engineering Practice*, vol. 18, no. 6, pp. 567–576, 2010.

[12] S. Rhim, A.-P. Hu, N. Sadegh, and W. J. Book, "Combining a multirate repetitive learning controller with command shaping for improved flexible manipulator control," *ASME Journal of Dynamic Systems, Measurement, and Control*, vol. 123, no. 3, pp. 385–390, 2001.

[13] A. Rauch, W. Singhose, D. Fujioka, and T. Jones, "Tip-over stability analysis of mobile boom cranes with swinging payloads," *J. of Dynamic Systems, Meas., and Control*, vol. 135, no. 3, pp. 031 008–1–6, 2013.

[14] W. Singhose, "Control safety margins for telescoping-boom mechatronic lifts," in *2020 4th International Conference on Automation, Control and Robots (ICACR)*, Rome, Italy, 2020, pp. 127–133.

[15] G. Romanello, "Stability analysis of mobile cranes and determination of outriggers loading," *Journal of Engineering, Design and Technology*, vol. 16, 11 2018.

[16] E. Wigglesworth, "Improvements in or relating to cranes," 1922.