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# THE FUTURE OF QUAYSIDE CONTAINER CRANES

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## INTRODUCTION

Container traffic continues to grow worldwide at about eight percent a year. To keep up with this growth, container ships are getting bigger and bigger. Post-Panamax ships with 16 containers abeam and 4600 TEU are now standard. New ships are operating with 17 containers abeam and 6000 TEU. Ships with close to 7000 TEU are expected to enter service in 1998.

For the largest ships, the efficiency of the terminals that serve them is especially critical. This is today's challenge to terminal operators. The front line of a terminal is its container handling quay cranes. New container cranes must move containers higher, further, faster, and more accurately than ever before. This paper looks at some of the issues this presents to crane designers and terminal operators.

There are four major design concerns with the new cranes:

**Size:** The cranes must be large enough and have the required strength and stiffness to service the ships and provide an effective platform for today's advanced mechanical and electronic systems. These cranes must be designed for greater wind and inertia forces because of their size. The increased outreach increases the overall loading on the crane.

**Speed:** The cranes must be fast enough to handle more containers over greater distances in less time. A minimum rate of 40 moves per hour is the standard requirement.

**Durability:** To justify the cost of their substantial investment, the cranes must be able to work continuously, without downtime. Today's cranes are designed for two million lift cycles or more, equivalent to 100,000 moves per year for 20 years. A properly designed and maintained crane should last longer without major overhaul.

**Wheel Loads and Stability:** The crane wheel loads must be compatible with the quay strength. The cranes should be stable in storm winds to eliminate the need for tie-downs that are cumbersome to install and can interfere with wharf traffic.

## SIZE

The size of container cranes has more than doubled since the first container cranes were built in the late 1950s. The Matson container cranes built by Paceco in 1959 were designed to lift 22.7t boxes 15.6m over the rails with an outreach of 23.8m. The latest container cranes lift 50.8t boxes 36m over the rails with an outreach of 56m. The new cranes are taller and heavier than any before – up to 75m high and weighing 1300t.

Typical size characteristics of state-of-the-art conventional single trolley post-Panamax container cranes are shown in Table 1.

| | Post-Panamax 16 wide | Post-Panamax 18 wide | Post-Panamax 20 wide |
|---|---|---|---|
| Gantry rail gauge (m) | 30.5 | 30.5 | 30.5 |
| Clear between legs (m) | 18.3 | 18.3 | 18.3 |
| Lift over the rails (m) | 34 | 34 | 36 |
| Total main hoist lift (m) | 50 | 52 | 54 |
| Clear under portal (m) | 12 | 12-15 | 15 |
| Out to out bumpers (m) | 27 | 24-27 | 24-27 |
| Outreach from waterside rail with 4m set back (m) | 45-47 | 50-52 | 56 |

Table 1: Size Characteristics of State-of-the-Art Container Cranes

### Wind Load

In many locations, the storm wind load condition controls the design of the crane legs and sill beams. Extra steel is required, though it is not needed for operating conditions. For this reason, it is often worthwhile to do wind tunnel testing of a scale model. A properly conducted wind tunnel test can determine the exact forces for different wind speeds as well as identify potential wind induced vibration problems. Without wind tunnel test data, the designer must rely on standard codes for wind loads which can be conservative because of their general nature.

Wind tunnel testing conducted at Colorado State University for Mitsubishi's new super-cranes allowed design wind loads to be verified and data to be collected for future cranes.

### Boom Length

Given the cost of the cranes and the rapid growth in ships, new cranes should be designed to handle at least one row of containers beyond the expected largest ships. Even if the extra capacity is not needed, the extra outreach allows for improved production because the trolley will not have to work in the slow down zone at the end of the boom.

### Trolley

The selection of a crane's trolley system type is significant for the structure, for wheel loads, and for maintenance considerations. The trolley can be rope towed or machinery type.

With a rope towed trolley system, the trolley drive, main hoist, and boom hoist are located in the machinery house on the frame. Trolley and main hoist ropes run from the machinery house to the end of the trolley girder, through the trolley, and to the tip of the boom. This arrangement allows the trolley to be shallow and lightweight, allowing greater lift height and smaller loads on the crane structure.

A machinery on hoist trolley has the trolley and main hoist drives on board. With most of the machinery on the trolley, the machinery house on the frame is much smaller, containing only the boom hoist. No trolley drive ropes are required, and the main hoist ropes are shorter than for a rope towed trolley.

### Structural

For the structural design, weight is the main difference between the two types of trolleys. The weight of the rope towed trolley is less than a third of a machinery trolley. Most dramatically, the heavier machinery trolley increases fatigue damage on a similar crane with rope towed trolley by a factor of 3 to 5. Table 2 gives an example of the effect on the structure. The wheel loads for a machinery trolley crane are about 15% higher than for rope towed.

| | Rope Towed Trolley | Machinery Trolley |
|---|---|---|
| Trolley weight<br>(electronic anti-sway) | 20.4t | 63.5t |
| Moving load<br>Trolley + spreader + HB + 50 LT container + impact | 104t | 152t |
| Moving load for fatigue damage | 68t | 114t |
| Fatigue damage | 1.0 | 4.63 |
| Total crane weight<br>30.5 m gage, post-Panamax | 1050t | 1200t |
| Wheel loads | 49t/m | 57t/m |

Table 2: Effect of Trolley Type on Structure

The choice of trolley will also affect the boom design. Generally, twin trapezoidal girders are used on cranes with rope towed trolleys because it makes rope reeving relatively simple. For machinery trolleys, with fewer ropes, rectangular or trapezoidal mono-girder booms are generally used.

A properly designed mono-girder boom crane weighs less than a properly designed twin girder boom crane for both rope trolley cranes and machinery trolley cranes. The eccentric lifted load applies additional vertical load on one of the two girders on the twin girder crane, which results in heavier sections. On a mono-girder boom, this same loading results in torsion only, which generally does not control. Therefore a larger section is not required.

### Maintenance

The configuration of the machinery trolley has several advantages from a service standpoint. The maintenance of the boom tip equalizer platform, landside turning sheaves, catenary trolleys, rope tensioners, deflector sheaves, and slap blocks is eliminated. The elimination of the trolley drive ropes and shorter main hoist ropes reduces costs for stocking and replacing wire rope. Pollution from oil spilling from the ropes is also minimized.

Operationally, because hoist ropes are much shorter and there is no stretch of trolley tow ropes, the machinery trolley provides better load control.

### Wheel Loads

The main disadvantage of the machinery trolley is the increase in crane weight and wheel loads on the quay. Some quays are not strong enough to support the latest super-cranes with a machinery on hoist trolley. The Port of Goteborg recently put out a tender for two super-cranes with machinery hoist trolleys. The Port eventually selected rope towed trolley cranes because the lower crane wheel loads were more compatible with the limited strength of the existing quay structure.

An interesting note: Noell Inc. of Germany won the Goteborg contract with a rope driven trolley system, even though a machinery trolley system was specified. They also won a large order from American President Lines for machinery trolley cranes. For the APL tender, a rope towed trolley was specified, but Noell proposed the machinery hoist trolley. The APL cranes were to be located at a new facility designed to accommodate the new and larger cranes. With the higher allowable wheel loads on the wharf, the machinery trolley crane was the preferred system.

The trolley selection should be based on the needs at a particular location, as well as the preference of the owner and operators.

### Design for Automation

The new container cranes have increasing degrees of automation that increase crane productivity. For automation to operate correctly, the location of all system components must be known. For fixed objects, this is an easy task. For moving objects, such as the crane structure flexing with the movement of the trolley, the task becomes more difficult.

One approach is to require a very stiff structure with strict deflection limits. A stiff structure helps with load control and provides an easier ride for the operator. It is also heavier. A detailed structural design process is required to minimize the weight and optimize the geometry and sections.

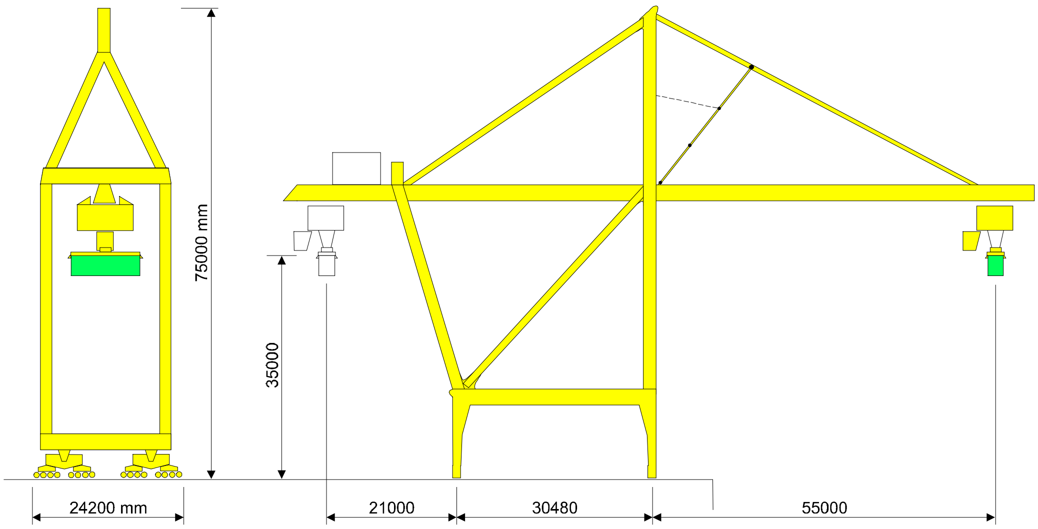
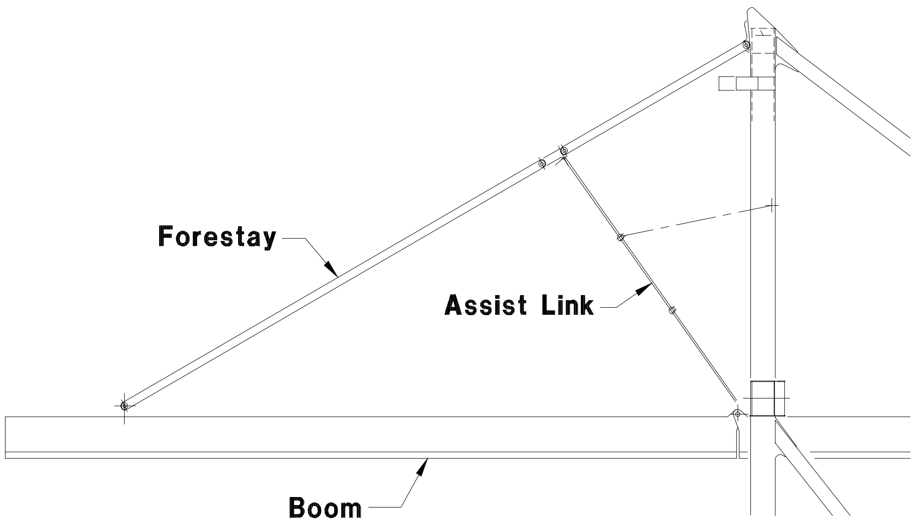



Figure 1: Rigid Structure Megacrane by MHI

The alternative is to account for crane movement in the load control system design and not specify deflection limits. This requires more complex software, but will result in a lighter crane structure.

If the strict deflection limit approach is taken, a significant source of vertical deflection of the boom is due to curvature and the presence of links in the stays. This is because the self-weight of the forestay causes it to sag along its length and rotate at the link. This effect actually raises the boom by a small amount when there is no load on it.

An assist link (patent pending) has been developed for Mitsubishi's latest super-cranes that eliminates the link deflection and reduces the curvature. The link works by holding the forestay in the sagged position. See Figure 2.




MHI patent pending

Figure 2: Assist Link

## SPEED

The challenge of the new cranes is not only an increased throughput, but an increased throughput while lifting boxes from deeper, higher, and over greater lengths in the ships than ever before.

Typical speed characteristics of the state-of-the-art conventional single trolley post-Panamax container cranes are shown in Table 3.

| | Post-Panamax 16 wide | Post-Panamax 18 wide | Post-Panamax 20 wide |
|---|---|---|---|
| Hoisting with rated load (m/min) | 50 | 60 | 75 |
| Hoisting with empty spreader (m/min) | 120 | 130 | 150 |
| Trolley travel (m/min) | 200 | 245 | 245 |
| Lowering with rated load (m/min) | 60 | 70 | 75 |
| Lowering with empty spreader (m/min) | 120 | 130 | 150 |
| Gantry travel (m/min) | 46 | 46 | 46 |
| Boom hoist time to stowed position (min) | 3 | 3 | 3 |

Table 3: Speed Characteristics of State-of-the-Art Container Cranes

### Power

To meet the mega crane challenge, the trolley and main hoist drives are becoming increasingly powerful to provide greater speeds and accelerations. Increasing drive and brake power is not difficult. The problem is to provide a complete and balanced system.

One casualty of the higher operating speeds is the operator. Severe accelerations and decelerations can be uncomfortable and tiring. Many new cranes are specified to provide an additional rail on the boom and trolley girder for a separate operator cab that moves independently from the trolley.

The festoon systems that power the trolleys and provide communications are a problem with the faster trolley drives. With high trolley accelerations and speeds, festoon systems can lose control, get tangled, and snare. Motorized festoon systems have been developed to keep up with the faster trolleys. These systems can be high maintenance items. New systems, such as inductive power and wave guide communications, are being developed and used in the industry.

The baloney cable that communicates with and powers the spreader is also a problem at high hoist speeds; it tends to get out of control. Powered baloney cable reels are now being provided on some cranes.

### Snag

Snag occurs when the spreader gets caught in the ship hold while hoisting at full speed. The tremendous energy this imparts to the crane can cause severe damage to crane systems. As cranes become larger and more powerful, the snag load becomes more critical and requires an engineered solution.

One option is to design the structure, and all components, to withstand the snag load. The alternative is to provide a snag system. A snag system is a mechanical solution that dissipates the snag energy of the ropes, usually by means of hydraulics, to prevent damage to the crane.

### Anti-Sway

Anti-sway load control systems continue to be developed. Electronic anti-sway is now state-of-the-art. If crane operators are unskilled, these systems can improve crane productivity significantly. However, many skilled operators are more productive and prefer to work without anti-sway.

### Automation

Full crane automation, from ship to shore, may be the answer to greater crane speeds and productivity demands. This is difficult to achieve, however, for several reasons.

Accuracy within a few centimeters will be required to automatically pick a container from a ship and set it on a truck on the quay. As discussed above, it is easy to track fixed objects accurately, but difficult to track moving objects.

Not only does a crane flex and stretch, but the ship twists, rises, and sags internally, as well as moving in relation to the wharf. Unless Automatic Guided Vehicles (AGVs) are used, it is very difficult to know exactly where the hustler or truck to receive the container is located on the quay.

Some manufacturers are working on systems that “see” the container when the spreader is close, and guide it automatically to lock on to the box.

### Other Crane Concepts

Advanced crane concepts have been developed to provide increased crane productivity.

Dual hoist cranes have a second hoist located over the quay. This can increase productivity by about 50%, but also increases the initial cost of the crane by 30 to 50 percent, adds an operator, and increases operating costs. The Baltimore Sumitomo cranes at Seagirt Terminal and the ECT Nelcon cranes at Delta Terminal in Rotterdam are the archetypes.

Dual hoist elevating platform cranes are dual hoist single trolley cranes except the shuttle runway elevates to the ideal elevation. See Figure 3. These cranes cost more than dual hoist cranes but produce more. The Virginia International Terminals NIT Kone cranes are the only cranes of this type.

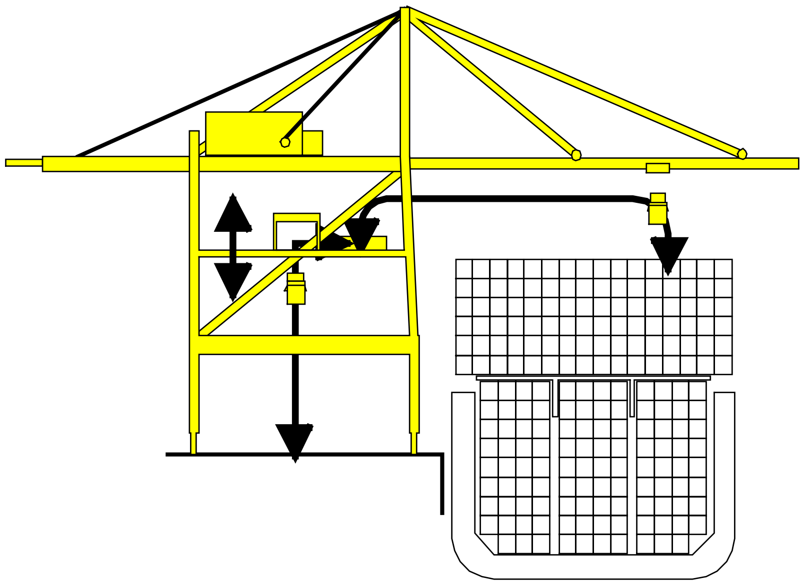


Figure 3: Dual Hoist Elevating Platform Crane

Other advanced concepts are being studied but have not been built or have not taken hold in the industry. Though highly productive concepts have been developed, the basic problems with new crane systems are increased complications and cost, and unknown reliability.

## DURABILITY

The first cranes were designed before steel fatigue was properly understood. Many of these cranes are still in service 30 years later. The key to attaining and surpassing a crane's design life is proper understanding of the effect of fatigue, proper inspection, and proper maintenance.

For the crane purchaser, it is important to have comprehensive specifications that address relevant issues of fatigue and service life. Either a crane manufacturer with a proven track record should be selected, or a comprehensive independent quality analysis program should be instituted during fabrication, or both.

### Fatigue Design

Modern codes, such as *BS 7608 Code of Practice for Fatigue Design and Assessment of Steel Structures*, recommend stress levels for many details. These stress levels are based on the statistical evaluation of thousands of tests. In most cases, the allowable stresses provide a 97.7 percent chance of attaining the design life.

The design criteria may be made more or less stringent to adjust the reliability. More important than design criteria is the use of "good" details.

Reliability depends on the details. A proper design can be, and often is, degraded by carelessly adding attachments for hand rails, walkways, and mechanical and electrical equipment. The degradation is dramatic. The expected fatigue life of a welded box beam is 2.5 times the expected life of the beam with a fillet welded lapped plate.

Just as important as using good fatigue details is maintaining the integrity of the design throughout the life of the crane. This starts with the elimination of thoughtless attachments and ends with a thorough periodic maintenance program.

If cracks are found and repaired, the structure's life can be extended indefinitely. Since the repairs may be made with more care and inspection than the original, the repaired joint is often better. For one series of cranes that have been routinely inspected and repaired, the frequency of crack occurrence has actually decreased.

### Mechanical/Electrical Systems

A new crane may have state-of-the-art systems and controls, but chances are that some of these systems will be out of date before the end of the crane's useful life. Cranes need systems that can grow with the technology, or be easily replaced. Reliable manufacturers will provide these systems.

## WHEEL LOADS AND STABILITY

When determining the needed crane characteristics for new or upgraded cranes, it is necessary to determine any restrictions that an existing quay may have. In 1994, a study was made for the Port of Oakland to determine the applied rail loads for 30 container cranes and the strength of the supporting quays. This study has been updated as three new cranes were purchased. The cranes represent a good sample of existing Panamax and post-Panamax cranes. The three new cranes are state-of-the-art post-Panamax 18-wide cranes.

## Crane Loads - Quay Design Criteria

When designing new quays, it is necessary to consider the possible future cranes. Part of the Port of Oakland study was the development of design criteria for the new quays.

The study is consistent with *Building Code Requirements for Reinforced Concrete, ACI-318*. Most of the crane loads are identifiable within the ACI definitions. Factors for loads that are not defined meet the intent of the code. Load combinations and factors are shown in Table 4. Typically, the ratio of the combined unfactored or working loads to the operating factored loads is 1.45.

| Load Name | Comb Name | Operating OP1 | Overload OL1 | *Stowed S1 |
|---|---|---|---|---|
| Crane Loads: | | | | |
| Dead Load | DL | 1.4 | 1.0 | 1.05 |
| Trolley Load | TL | 1.4 | 1.0 | 1.05 |
| Lifting System | LS | 1.4 | | 1.05 |
| Lifted Load | LL | 1.7 | | |
| Impact | IMP | .85 | | |
| Operating Wind | OWL | 1.3 | 1.0 | |
| Stall Torque | STL | | 1.0 | |
| Stowed Wind | SWLU | | | 1.3 |
| Wharf Loads: | | | | |
| Dead Load | WDL | 1.4 | 1.0 | 1.05 |
| Superimposed Live Load | WLL | 1.7 | 1.0 | 1.28 |
| Soil Load | SOIL | 1.7 | 1.0 | 1.28 |

EXAMPLE: OP1 is 1.4 x WDL + 1.7 x WLL + 1.7 x SOIL + 1.4 x DL + 1.4 x TL + 1.4 x LS + 1.7 x LL + 0.85 x IMP + 1.3 x OWL.

*Stowed: The boom is up, the trolley and lifting system are in the stowed position and tie-downs, if any, are in place.
Wind is acting in the most adverse direction, "angled wind."
Impact is reduced, since the value at the gantry rail is much less than the value at the trolley rail.
If tie-downs are required during stowed wind, the factored tie-down force should be 0.9 times (DL+TL) plus 1.3 times stowed wind or 1.4 times stowed EQ.
All loads causing and combined with overloads have a load factor of 1.0.

Table 4: Load Combinations

Applied factored loads for the existing cranes at the Port of Oakland are tabulated in Table 5.

| Manufacturer and Year | Factored Rail Load tons/m × m spread | | Crane Description |
|---|---|---|---|
| | Land side | Water side | |
| Paceco 1965-69 | 32x9 | 39x9 | Panamax A-frame, Trussed Boom, 15.2m gage, 40 ton. |
| KSEC 1986 | 40x11 | 53x11 | Post-Panamax A-frame, Articulated Twin Plate Girder Booms, 30.5m gage, 50 ton. |
| Paceco 1977 Raised 1990 | 35x9 | 44x9 | Panamax A-frame, Trussed Boom, 30.5 gage 40 ton. "SL-7" |
| Krupp 1980 | 44x5.5 | 40x8 | Panamax A-frame, Mono-Girder, 30.5m gage, 40 ton. |
| Hitachi 1980 Raised 1993 | 31x9 | 49x9 | Panamax A-frame, Articulated Twin Plate Girder Boom, 30.5m gage, 40 ton. |
| Paceco 1968 Raised 1993 | 52x9 | 44x12 | Panamax Low Profile, 30.5 m gage, 40 ton. |
| Paceco 1990 | 28x12 | 54x12 | Post-Panamax A-frame, 30.5m gage, Twin Girder Boom, 50 ton. |
| Kocks 1988 | 44x12 | 70x12 | Post-Panamax Low Profile, 29.3m gage, 50 ton |
| Mitsubishi 1988 | 49x13 | 49x13 | Post-Panamax A-frame, Articulated Plate Girder Boom, 30.5m gage, 40 ton. |
| Paceco 1993 | 36x13 | 53x13 | Post-Panamax A-frame, Articulated Plate Girder Boom, 30.5m gage, 40 ton. |
| ZPMC 1996 | 44x12 | 70x12 | Post-Panamax A-frame, Articulated Plate Girder Boom, 29.3 m gage, 50 ton. |
| ZPMC 1997 | 41x12 | 71x12 | Post-Panamax A-frame, Plate Girder Boom, 30.5m gage, 50 ton. |
| The linear load was calculated by dividing the corner loads by the length over which the wheels spread the load. | | | |

Table 5: Factored Operating Loads from Existing Cranes

### Recommended Design Criteria For New Crane Runways

The girder should be designed using ACI 318 or an equivalent code. Prestressed concrete piles should be designed using Section 4.7.6 of the *PCI Design Handbook, 4th edition,* with modifications for pile slenderness, or equivalent.

For girders on piles, the stiffness of the piles and soil should be included in the strength calculation. For girders on spread footings, the beam on elastic foundation method should be used. The girders may be modeled with cracked section properties, since the girder moments will crack the section.

The effective span length is greater than the pile spacing because the curvature does not reverse between piles. Consequently, the girder does not have to be designed as a deep beam. The applied loads should be multiplied by the load factors tabulated in Table 4. The minimum recommended factored loads on the landside and waterside rails are shown in Table 6.

With the gantry wheels spaced at 1.1 times their diameter, the theoretical working load limit for a 175 pound ASCE rail is 85 t/m base on the wheel-rail interface. The factored load limit, much more than expected, is 124 t/m.

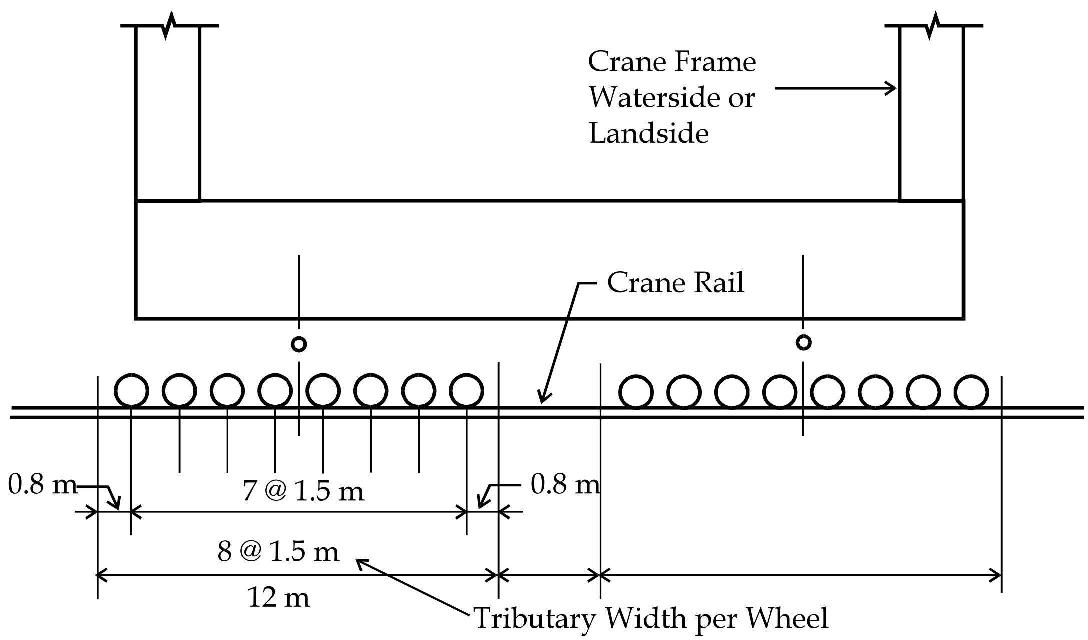



Figure 1: Wheel Load Distribution

| Factored Load | t/m | Comments |
|---|---|---|
| Vertical | 82 | See Figure 2 for wheel load distribution. |
| Horizontal | | |
| ⊥ to Rail | 3.7 | The total load on a rail should be the distributed load times the effective length used for the vertical load. |
| \| \| to Rail | 4.5 | |

Table 6: Recommend Minimum Factored Rail Loads

Occasionally, a ship's bulbous bow will strike the girder piles. An economic study should be made to determine the incremental cost to design the girder with one and two piles missing. In many cases, a three-dimensional analysis of the girders shows that a missing pile is less serious than expected. The appropriate missing pile criteria should be determined after the study. If no provision is made for missing piles, an analysis should be made to determine the allowable factored load with one and two piles missing.

### Stability

Of all the container cranes totally destroyed because of a failure, most, over 30, have been victims of storm wind loads. In every case of storm wind failure, the failure has not been in the crane but in the tie-down system.

Typically there are two errors. The embedded hardware has not been designed properly and the prying effect is not included. To correct the first error, the designer should imagine the failure mode of the embedded hardware. Many designs fail by opening and releasing the pin. To correct the second error, the designer should be sure the tie-down force includes prying.

Prying occurs when the stowage shear pin that transfers the horizontal wind load to the quay is on the equalizer system. The crane manufacturer usually neglects this. Even if the manufacturer reports that no tie-downs are needed, there still may be a problem. Figure 2 shows the loads and load path for the effects of the horizontal force tending to rotate the equalizer system. If the shear pin is on the main equalizer beam, the problem is still there. If the shear pin is on the sill beam, there is no prying.

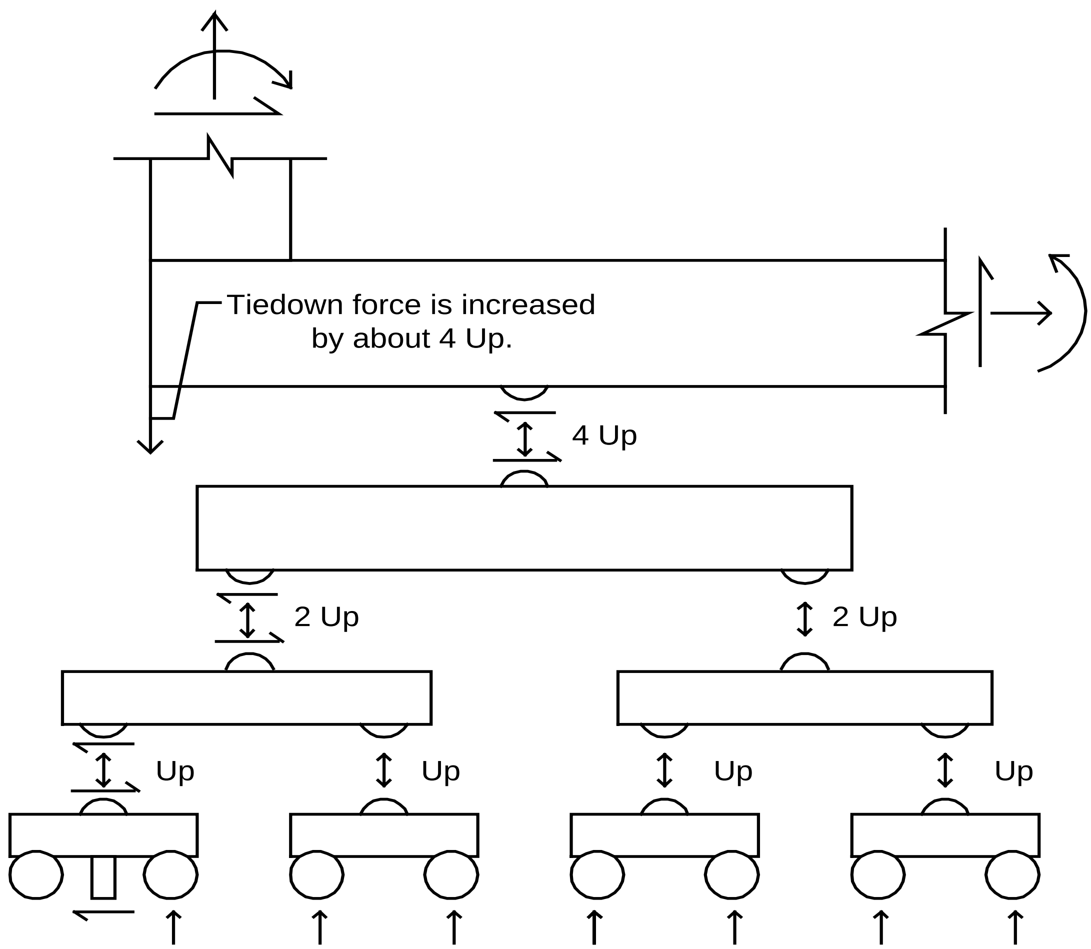



Figure 2: Prying Action for Stowage Pin on Equalizer System

## CONCLUSION

New cranes can be designed that are big enough and fast enough to keep up with the demands of the new larger ships.

The design of these cranes must be the result of cooperation between terminal operators, structural, mechanical and automation engineers, as well as wharf designers and crane operators.

On a big scale, the size of ships, the throughput capacity of the terminal, and the strength of the wharf must be taken into consideration.

On a smaller scale, the crane must be stiff enough to allow electronics to operate or the electronics must take the crane deflections into account.

The selection of a trolley system and trolley and hoist speeds should be the result of careful evaluation of several factors.

When these issues are addressed, an effective crane can be designed that will suit the present and futureneeds of the end user.

## ACKNOWLEDGMENTS

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