Liftech
LIFTECH CONSULTANTS INC.

Reprinted from PORTS '98
Proceedings of the Conference
American Society of Civil Engineers
Held March 8-11, 1998, Long Beach, CA

PURCHASING CRANES IN A CHANGING WORLD

Michael A. Jordan, F. ASCE[1]
Liftech Consultants Inc.

HISTORY

Pacific Coast Engineering Company designed and built the world's first dockside container crane for Matson Terminals in 1958. The challenge was to produce a crane that was reliable. Any workable solution would be better than pallets and cargo nets.

The geometry suited the containership that was on the drawing boards. The structure complied with railway bridge standards. The electrical and mechanical components complied with overhead bridge crane standards. The engineers, the manufacturer, and the buyer were all within twenty minutes of each other. Working relationships had already been established on earlier projects. The project progressed smoothly with few misunderstandings.

The growth of containerized cargo changed the shipping industry. One result of cheaper, faster, and more reliable transportation of cargo was the development of the global factory. Now, a product could contain components from anywhere in the world. Manufacturers once had a local advantage. Now they must compete worldwide.

Ironically, the container crane manufacturers were the most affected by improved transportation. Cranes are purchased and produced with little

---

[1] Structural Engineer, Chief Executive Officer, Liftech Consultants Inc., 3666 Grand Ave., Oakland, CA 94610, Phone 510-832-5606, Fax 510-832-2436

regard to geography. We can communicate faster across the globe in 1998 than we could across town in 1958. Electronic communications allow instantaneous transmission of data, drawings, and photographs. The design-build team is worldwide. Components are made worldwide and shipped to an assembly site. The nearly completed crane is shipped to the terminal fully assembled and operationally tested.

The buyer's choices are difficult to make. The characteristics of the crane must suit current needs and technology and be adaptable to changing conditions.

Selecting a manufacturer is often more difficult than making the technical choices. Should the buyer adapt the specifications to suit the manufacturer or should the manufacturer adapt his processes to meet the buyer's specifications? Is cost the only issue? Does the design-build team make a difference?

This paper addresses both the technical and intangible issues. Choices are discussed, but only a few answers are suggested. There are many correct answers. No two situations are the same.

The discussions below are based on the author's forty years of container crane practice and recent experiences with the procurement of over sixty post- Panamax cranes. Many of the post-Panamax cranes are in the Los Angeles and Long Beach ports. The owners and manufacturers are not identified. The manufacturers are all capable and conscientious.

SPECIFICATIONS

The technical specifications describe the size, the design criteria, and the operational characteristics of the crane. The salient items are discussed below.

CONFIGURATION

The conventional Panamax crane has two configurations: one, the operating mode, with the boom horizontal, and another, the stowed mode, with the boom fully raised.

Post-Panamax cranes may have three configurations: the operating mode with the boom horizontal, the stowed mode with the boom partially raised to about 45 degrees, and the maintenance mode with the boom fully raised.

The partially raised mode is possible, since the cranes are so tall that the boom does not need to be fully raised to clear the ship. This saves time, improves stability, and reduces the risk of the boom latch malfunctioning and causing an accident. If the latch malfunctions, the boom hoist may pay out slack rope. If the latch then releases the boom with slack rope, the boom falls until the rope is tight. This happens.

Container cranes are usually capable of full operation only with the boom horizontal. At one time, operators planned to use dockside cranes to handle cargo on the quay with the boom raised. This was not worthwhile. Generally, the crane should be designed to lift the rated load and travel the trolley only when the boom is in the operating position. The trolley should be able to raise and lower the empty spreader with the boom stowed.

The operator's cab is usually on the trolley. However, as automation becomes popular, it may be effective to take the operator's cab off the trolley and put it on an independent runway parallel to the trolley. This has been successfully done at Virginia International Terminals. The new APL, Evergreen, Hanjin, and Oakland cranes have provisions built in for future independent trolleys.

CRANE GEOMETRY

The typical post-Panamax crane dimensions are:

100' gantry rail gage.

50' backreach from the landside rail.

155' to 172′ outreach from waterside rail.

110' Lift above rails.

170' total lift.

50' clear at portal.

60' clear between legs.

88'-6" out-to-out bumpers.

12′ setback of waterside rail from the ship.

No height limit.

In the United States, the rail gage is typically 50′ for the older Panamax cranes and 100′ for the newer Panamax and post-Panamax cranes. Gage selection is controlled by operational requirements. Currently, some operators are considering 150′ rail gages. The increased gage provides more room for traffic under the crane. The backreach will be eliminated, resulting in no increase in yard area occupied by the crane. For large outreach cranes, the increased gage reduces the rail loads and increases the stability. This is discussed in detail in *Super Productive Cranes* (Reference 13).

The common 155′ outreach, with 12′ setback to the waterside rail, is suitable for cranes servicing ships with 16 rows of containers on deck, the current maximum. The new APL cranes in the Port of Los Angeles and several cranes in the Port of Oakland are capable of 172′ outreach. The increased outreach is needed to service ships with 16 containers on deck and with the waterside rail setback 25′-8″ from the ship to provide a traffic lane next to the ship. This will be even more important when the container traffic is automated. Since ships may eventually have 22 rows on deck, some buyers are specifying 200′ outreach with 25′ set back. This outreach is feasible. Even if the full outreach is not required to reach the last container, it is desirable for operations, since the trolley speed needs to be reduced near the end of the boom.

The 110′ lift above the rails and the 170′ total lift are adequate for planned vessels. If more height is needed later, the cranes can be economically modified.

The 50′ clear height under portal should allows for the use of straddle carriers. The increased clear height does not add significantly to cost, but it can cause sway problems due to dynamic response of the frame to the load control forces. The dynamic problem is present with both automatic and manual sway control. This is discussed later.

The distance between the bumpers is limited, so cranes can service alternate hatches. Surprisingly, this dimension is not universal. Some operators limit the dimension to 85′-6″. The reduced dimension increases the wheel loads and reduces the stability. On some cranes, stop blocks are used between the bottom of the sill beam and the main equalizers to provide the desired safety margin without adding ballast.

The selection of the setback of the waterside rail from the ship is problematic. Superficially, the setback seems to reduce the useable yard area. But some space is required to service the ship. The setback distance is selected to suit operational requirements. If the terminal will be automated or the quay will be very busy, for example, the APL Los Angeles terminal, the waterside traffic lane is justified. The cost due to the increase in crane weight and rail loads is significant relative to the crane and rail but not relative to the overall terminal. The increased setback also has the advantage of reducing the chance of a ship colliding with the stowed crane.

CAPACITY

The usual rated load under the spreader is 50 long tons, based on handling 25 long ton twin 20′ containers. Currently some 20′ containers have a maximum gross weight of 30 long tons. This, however, is rare. Since the load can be measured, and the chance of handling two maximum weight containers simultaneously is slight, many cranes can operate within the rated load of 50 long tons, even if one of the twin twenties weighs 30 long tons. A few cranes have a rated load of 60 or more long tons. The cost increase depends on the defined operating cycle and fatigue load spectrum. If these are carefully defined, the cost increase the rated load the 60 long tons may be small.

The cargo beam can usually be rated for the load under the spreader plus 10 long tons, since the cargo beam is not as heavy as the spreader. The capacity of the cargo beam should be determined after the design is completed based on the rated load under the spreader.

SPEEDS

Typical speeds for post-Panamax cranes are:

Main Hoist:

> Hoisting and lowering with rated load 200 to 250 fpm.
>
> Hoisting and lowering with spreader only (no load) 400 to 500 fpm.

Rope and machinery trolleys:

> 800 fpm.

Gantry travel:

> 150 fpm

Boom Hoist

> Time to raise or lower to partially raised position: 3 min.

ROPE TROLLEY OR MACHINERY TROLLEY

Once, nearly all container cranes had rope trolleys. For a rope trolley, the main hoist ropes feed from the main hoist machinery on frame through the trolley to the spreader. The rope trolley is usually towed by ropes from the trolley drive on the frame. The rope towed trolley is the lightest possible trolley. Some rope trolleys are driven by motors on the trolley. These trolleys are called semi-rope trolleys.

Now, some cranes have machinery trolleys. For a machinery trolley, the main hoist and trolley drives are on the trolley. The machinery trolley is the heaviest trolley. But the disadvantage of weight is offset by a much simpler overall mechanical system.

Although most cranes have a rope trolley, APL and others have selected a machinery trolley, primarily for ease of maintenance. APL's new Port of Los Angles machinery trolley cranes are performing well. The heavier wheel loads are not a problem for new wharves.

Either style of trolley will work. Neither has a distinct advantage. The choice depends on the terminal operator's preference.

DESIGN CRITERIA

There is no national standard suitable for dockside container cranes. Buyers usually rely on expert advice or the advice of manufacturers. Proper design criteria produces an economic, reliable crane that fills the need.

Container cranes are significantly different from other dockside cranes and from bridge cranes. Container cranes usually experience millions of cycles of significant loads. The load spectrum can be defined with reasonable accuracy

Building codes, like the AISC *Specification for Structural Steel Buildings* (Reference 1), include some fatigue criteria, but the provisions are superficial and many important requirements are not mentioned. Bridge crane codes, like *AISE Technical Report No. 6* (Reference 2), are suitable for bridge cranes, but again, many important design considerations are not mentioned. Some foreign specifications can be adapted to container cranes.

The proper design criteria will vary from crane to crane depending on the expected use, performance, and environment.

The mechanical criteria in the FEM specifications (Reference 10) and the static provisions of the AISC *Specification for Structural Steel Buildings* (Reference 1) are suitable for all but cyclical stresses, provided some additions and modifications are made.

The British Welding Institute has performed significant and extensive research on the behavior of welded joints subjected to cyclical loading. BS 7608 (Reference 5) is based on this research and is the best guide for fatigue design available. The AISC plate buckling provisions are unsafe. The problems from local buckling of thin, inadequately stiffened web plates on bridges are well known. The DIN (Reference 9) and FEM specification provisions for plate buckling are reasonably good. The DIN and FEM specifications can be modified in accordance with the concepts in the *Guide to Stability Design Criteria for Metal Structures* (Reference 6) to produce reliable economic plate buckling criteria.

Electrical systems are usually designed by the supplier. GE and ABB are dominant suppliers today. Others are available and more will come, but currently these two companies are the leaders. In some jurisdictions, like Los Angeles, the local electrical code must be met. This can cause serious and costly problems. The supplier should be warned if local code approval is needed.

LOAD CONTROL

To be super productive, sway and yaw need to be controlled. Sway is swinging in the direction of trolley travel. Yaw is rotation about the vertical axis. Sway and yaw may be controlled by using rigid reeving, like on the Matson Los Angeles terminal large gantry yard cranes, or by using flexible reeving and controlled trolley motions, as on the APL terminal 300 cranes. Trolley motions may be controlled automatically by an electronic anti-sway system, manually by the crane operator, or by a combination of both. Operators do not agree whether or not automatic control is needed or even beneficial.

Operators generally agree that a good automatic sway control will help the less skilled operator. Some think automatic sway control reduces the productivity of highly skilled operators. This could be. But automatic load control has improved dramatically in the last few years and will soon be so good that all operators will be helped. When semi and fully automatic crane operation is achieved, which will happen in the next ten years, automatic sway control will be required. It is important to recognize the need for automatic control provisions now, since the characteristics of the crane selected now may make later adoption of automatic sway control expensive.

If sway control is manual, then rigid reeving is desirable. For rigid reeving, the main falls should be inclined. But if sway control is automatic, the falls should be nearly vertical.

For rigid reeving the swing period depends on the spring stiffness of the reeving and tributary mass. When the load is eccentric, the effective mass is not the same at both ends of the container. But the stiffness of the falls is the same, so the swinging period is different at each end, and the load tends to yaw.

For vertical falls, the period depends only on the length of the falls. The difference in mass at each end does not affect the period. The load does not yaw and trolley motion alone can eliminate sway.

Containers tend to yaw on rigid reeving and swing on flexible falls. So the reeving that is best for manual control is not suitable for automatic control. And visa versa.

A new problem has developed for load control on post-Panamax cranes with vertical falls. The dynamic interaction between the frame, the trolley, and the load can be problematic. If the natural period of the frame is one half the period of the hanging load, the motion of the trolley that is expected to control sway will instead excite the frame. The frame period depends on the mass and stiffness of the frame.

With vertical falls, the natural period of the hanging load is about 5 or 6 seconds. For inclined falls the period is less. For typical Panamax cranes, designed for moderate stowed winds, the natural period of the frame in the trolley travel direction is about 1.5 seconds, so dynamic resonance is not a problem.

For post-Panamax cranes the mass is significantly increased, especially at the trolley level, and stiffness is reduced because the portal height is increased. The frame period is increased to about 2.5 or 3 seconds. So, the ratio of frame period to the load period is about one-half, the worst case. When the operator or the computer attempts to control swing, the trolley acceleration forces excite the frame instead of controlling the load. This effect can be reduced by changing the controls. But the best solution is to design the crane to avoid the undesirable ratio. The natural period of the frame in the trolley travel direction should be about 1.5 seconds or less. This can be economically achieved by increasing the depth of the legs and the portal tie.

Snag protection, which is not related to sway and not related to stall, is always desirable. Snag loads develop when the high speed empty spreader jams in the ship's cells and the kinetic energy of the rotating machinery must be absorbed by the ropes and structure. This is especially critical for machinery trolley cranes. The snag control device should reset automatically.

POWER SUPPLY

The primary power source is either shore electrical supply or diesel motor generator sets on the crane. Shore power is most desirable. Shore power is typically 4160 volts, but it can be as high as 12,000 volts.

The shore power is transmitted through a cable using cable reels or through buss bars and collectors. Most cable reels are monospiral reels. The cable feeds from a vault through a horn to a trough in the wharf. The trough may be covered, usually with a Panzer belt, or not covered. Cable reels can hold as much as 2000 feet of cable, allowing the crane to travel 4000 feet. Cables contain a fiber optic strand for communications. Buss bar systems are usually below grade in a power trench with hinged covers. Overhead conductors may be used on the landside of the crane. Overhead systems, however, interfere with operations and are not widely used now. Communications cannot be carried in the buss. At the Hanjin terminal in Long Beach, the collector trench includes a parallel micro wave communications system.

Cable reels have the advantage of being less likely to have excessive voltage drop, and it is easier to isolate one crane. The Port of Oakland has converted some collector systems to cable reels in order to transmit more power through existing power trenches.

Both cable reels and buss bars perform well. The choice depends on the conditions at the site and the operator's opinion. Operators don't agree on the choice.

OPERATING AND STORM WIND LOAD

Most operators stop operations when the anemometer on the apex beam indicates a wind speed of 35 mph. In order to allow the crane to be stowed after termination of operations, the design wind speed is 55 mph. As a precaution against sudden gusts, the gantry system should be able to resist a 70 mph gust without moving. The effect of the eccentricity between the rail and the main equalizer pin should be included in the analysis of the gantry drive system. This is discussed in the Ports 95 paper, *Dockside Container Cranes* (Reference 12).

Storm wind load criteria should follow the ASCE Standard 7-95 *Minimum Design Wind Loads for Buildings and Other Structures* (Reference 3). Shape coefficients may be taken from a standard such as BS 2573 (Reference 4) or determined from wind tunnel test. If wind tunnel tests are used they should follow the detailed requirements for boundary-layer wind tunnels specified in ASCE 7-95. Manufacturer's tests are often invalid because of test deficiencies and a misunderstanding of the effects of wind.

If tiedowns are required, the tiedowns and the load path to the tiedowns should be designed using load and resistance factor design principles. The tiedown should be strong enough to resist the factored dead load plus the factored wind. Some manufacturers design for unfactored loads, resulting in undersized tiedowns. At least 30 cranes have been destroyed due to tiedown failures.

WHEEL LOADS

The calculated wheel loads should be calculated using load factors. New wharves for post-Panamax cranes should be designed for a factored operating load of 55 kips per foot on the waterside rail and 45 kips per foot on the landside rail and, for Panamax cranes, 50 kips per foot on the waterside rail and 40 kips per foot on the landside rail. Wheel loads are discussed in reference 12.

SELECTING A MANUFACTURER

Since the advent of the global factory, the selection of the manufacturer has become extremely difficult. The rules have changed, the teams have changed, and there are new players. But to make matters more difficult, there are new players on old teams. It is very difficult to predict performance.

If the manufacturer has a good track record, and if he performs the job with the *same* engineer, fabricator, and erector team, then the chance of success is high. So is the price. The global factory can reduce the price by as much as 20 percent. This is persuasive and sells cranes. But it's not the same team. The labor may be low skilled and cheap. Often the team members have not worked together before. In spite of the problems, cranes have been purchased and successfully delivered from global factories. So success is possible, but it takes a team effort and an understanding of the potential difficulties.

Some designs are patterned after a proven design but the designer unknowingly makes a few changes which seriously downgrade the crane. It is difficult to detect these small but significant changes.

Welding symbols may look like AWS symbols but mean something quite different. One manufacturer uses a complete joint penetration AWS symbol for a partial penetration weld.

Field problems are not handled directly by the design engineers, since the design engineers may no longer be involved after fabrication starts. The engineer no longer walks through the plant. The fabricator may not understand the design. One foreign shop working half a world away from the prime contractor's designers didn't realize where the trolley rail was located and did not meet the strict dimensional tolerances for rail alignment. The erectors often don't know the special requirements for trolley runway alignment. Sometimes the inspectors don't understand the specifications.

Often the voyage bracing is designed by an engineer at a remote fabrication site. Often, the site engineer doesn't understand the fatigue design criteria and designs temporary attachments that seriously downgrade some fracture critical members. The members are then repaired in the field. This is costly.

Erecting and testing the crane at the erection site saves time and money. But what happens if the vessel arrives before the crane is ready? The crane must leave unfinished or be delayed until the vessel finds a new window. So the crane is either finished in a hurry or shipped incomplete. When the crane arrives, a great deal of finishing work or rework is required. This is extremely expensive and always delays delivery. It's difficult to keep the team spirit alive when the contractor is loosing millions because of rework, and the terminal cannot operate because the crane is not ready.

Sometimes the voyage bracing is not proper and the crane is damaged or destroyed at sea. It is common for vessels to lay over en route for repairs.

CONCLUSION

Buying cranes today is a challenge. The global factory has the potential to reduce costs significantly. But the buyer must help form a team and monitor and control the work carefully. Excellent cranes are available at excellent prices.

ACKNOWLEDGEMENT

Mr. Lawrence Wright, mechanical engineer with McKay International Engineers, assisted with the preparation of this paper and the referenced dockside container crane specifications.

APPENDIX: REFERENCES

1. AISC, *Specification for Structural Steel Buildings, Allowable Stress Design and Plastic Design, June 1, 1989, with Commentary,* American Institute of Steel Construction, 1989.
2. AISE Technical Report No. 6, *Specification for Electric Overhead Traveling Cranes for Steel Mill Service,* Association of Iron and Steel Engineers, June 1996.
3. ASCE 7-95, *Minimum Design Loads for Buildings and Other Structures,* American Society of Civil Engineers, New York, December 1995.
4. BS 2573: Part 1: 1983,*Rules for the Design of Cranes,* British Standards Institution, London, 1983.
5. BS 7608: 1993, *Fatigue Design and Assessment of Steel Structures,* British Standards Institution, London, 1993
6. Bhimani, Arun K. and Hoite, Simo, *Machinery Trolley Cranes,* ASCE/PIANC Ports 98 Conference, Long Beach, CA, March 1998.
7. Bhimani, Arun K and Kerenyi, Julius, *Rope-Towed Trolley or Machinery Trolley, Which is Better?* Facilities Engineering Seminar, AAPA, San Pedro, CA, November 1995.
8. Bhimani, Arun K., Morris, Catherine A., and Karasuda, Shuji, *Dockside Container Crane Design for the 21st Century,* Container Efficiency Conference, World Trade Centre, Singapore, March 1996.
9. DIN 15 018, Part 1, DIN, *Cranes, Steel Structures Verification and Analyses,* Deutsche Institut fur Normung e V., Berlin, November 1984.
10. FEM, *Rules for the Design of Hoisting Appliances, Booklet 3, Calculating the Stresses in Structures*, Federation Europeenne de la Manutention, 1.001, 3rd Edition, 1987.
11. Galambos, Theodore, ed., *Guide to Stability Design Criteria for Metal Structures*, 4th ed., formerly CRC Guide, John Wiley & Sons, 1988.
12. Jordan, Michael A., *Dockside Container Cranes,* ASCE/PIANC Ports 95, Vol. 2, Tampa, FL, March 1995.
13. Jordan, Michael A., *Super Productive Cranes*, TOC '97, Barcelona, June 1997.