

AVONDALE SHIPYARDS, INC.

SLEW CRANES IN SHIPYARDS

-A STUDY-

by

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Contributing Editor: Dr. James Pinnells

1985

Report Documentation Page				Form Approved OMB No. 0704-0188	
Public reporting burden for the collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing this burden, to Washington Headquarters Services, Directorate for Information Operations and Reports, 1215 Jefferson Davis Highway, Suite 1204, Arlington VA 22202-4302. Respondents should be aware that notwithstanding any other provision of law, no person shall be subject to a penalty for failing to comply with a collection of information if it does not display a currently valid OMB control number.					
1. REPORT DATE 1985		2. REPORT TYPE N/A		3. DATES COVERED -	
4. TITLE AND SUBTITLE Slew Cranes in Shipyards A Study				5a. CONTRACT NUMBER	
				5b. GRANT NUMBER	
				5c. PROGRAM ELEMENT NUMBER	
6. AUTHOR(S)				5d. PROJECT NUMBER	
				5e. TASK NUMBER	
				5f. WORK UNIT NUMBER	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) Naval Surface Warfare Center CD Code 2230 - Design Integration Tools Building 192 Room 128 9500 MacArthur Bldg Bethesda, MD 20817-5700				8. PERFORMING ORGANIZATION REPORT NUMBER	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)				10. SPONSOR/MONITOR'S ACRONYM(S)	
				11. SPONSOR/MONITOR'S REPORT NUMBER(S)	
12. DISTRIBUTION/AVAILABILITY STATEMENT Approved for public release, distribution unlimited					
13. SUPPLEMENTARY NOTES					
14. ABSTRACT					
15. SUBJECT TERMS					
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT SAR	18. NUMBER OF PAGES 71	19a. NAME OF RESPONSIBLE PERSON
a. REPORT unclassified	b. ABSTRACT unclassified	c. THIS PAGE unclassified			

PROGRAM MANAGEMENT

This report is one of the many projects managed and cost shared by Avondale Shipyards, Incorporated, under the auspices of the National Shipbuilding Research Program. The program is a cooperative effort between the Maritime Administration's office of Advanced Ship Development and the U.S. shipbuilding industry.

Executive administration and supervision were provided by Mr. E.L. James, Vice President, Production Planning, Avondale Shipyards, Incorporated.

Project definition was provided by the members of the Society of Naval Architects and Marine Engineers Panel SP-1 Shipyard Facilities and Environmental Effects and Mr. R.W. Schaffran, Maritime Administration, Office of Advanced Ship Development.

The special, advisory group was made up of the following:

Avondale Shipyards-, Inc. - Eugene Aspuru
Avondale Shipyards, Inc. - Richard A. price
Emscor, Inc. - Walter P. Manning, Jr.
National Steel & Shipbuilding Co. - L.P. Haumschilt

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1. INTRODUCTION

1.1. PURPOSE

The purpose of this study is to examine the various types of cranes in current use in shipbuilding yards round the world, and to make a recommendation on which type is most cost effective for installation in U.S. shipyards. In the study particular attention has been paid to the four aspects of crane design that impact most significantly on shipbuilding operations: <1> double-boom as opposed to single-boom design; <2> column mounting as opposed to turntable mounting for the slewing part of the crane -- and for column mountings, the kind of bearings to be used; <3> balanced boom as opposed to unbalanced boom design; and, most important, <4> the provision of level luffing as opposed to traditional luffing. Since these distinctions may not be universally familiar; the purpose of Section 4 of the study is to outline the theory of operation of each system. The short sketch of crane development on both sides of the Atlantic is intended to explain this information gap: why is it that U.S. operators are often so unfamiliar with developments in crane technology outside the United States? Having filled in the background, the purpose of Section 5 is the detailed examination of five actual cranes with a view to establishing first the investment cost of each type -- its "cost-to-build" -- and then its maintenance and running costs. Operational efficiency, i.e. , the speed, accuracy, and downtime of each crane type, are also investigated. Finally the safety and training of the operator are considered as investment factors.

It is intended that the data and conclusions offered in this report would be of value to any U.S. shipyard during the planning and scheduling of replacement cranes.

1.2. PROBLEM

It has become apparent that the performance of the turntable cranes traditionally used in U.S. shipyards has fallen behind the performance offered by crane designs available in Europe and the Far East. Most owners, however, lack sufficient background knowledge to decide exactly what type of crane is best suited to their needs. Without such knowledge, a rational study of cost-effectiveness is, unfortunately, impossible. What is urgently required at this time is a study that spells out the significant differences in design theory and explores the practical implications of various designs in day-to-day shipyard operation.

1.3. SCOPE

This study is not exhaustive. It confines itself to shipbuilding and ship repairing operations in shipyards. Other maritime applications -- fast cargo or container handling, offshore oil-rig installation and so on -- are not evaluated. Further, no attempt has been made to review systematically all the myriad types of cranes available. This would serve no useful purpose. Instead the study has concentrated on the handful of design features that are important for shipyard applications. Another self-imposed limitation is that in discussing investment and running costs, exact dollar figures have not been given, partly because they are not reliably available, and partly because, in a worldwide study, the currency roller-coaster would soon make such figures worthless. More useful are comparative figures, i.e., cost multipliers, that can be attached to cranes of different types, and these have been given. In general, the study has not tried to achieve quasi-scientific completeness; rather, it has highlighted the information a crane owner who was about to make an investment decision might find useful and relevant.

1.4. ACKNOWLEDGEMENTS

A number of organisations have made important contributions to this study, especially in collecting data about operational cranes -- a time-consuming process -- and in making it available for publication. In particular, the author would like to thank the following companies and organisations without whose help this study would not have seen the light of day:

Avondale Shipyard, New Orleans, U.S.A.

Beratungsstelle für Stahlverwendung, Düsseldorf, West Germany

Blohm and Voss AG, Hamburg, West Germany

Howaldtswerke-Deutsche Werft AG, Hamburg, West Germany

Keppel Shipyard, Singapore

National Steel and Shipbuilding Company (NASSCO),
San Diego, U.S.A.

M. A. N. AG, Nürnberg Works, Nürnberg, West Germany

Use has also been made of published literature and catalogs of a number of companies, in particular: American Revolver Cranes; Clyde Ironworks, Inc.; DEMAG, Wetter; The Dravo Corporation; M.A. N. AG; and M. A. N.-Wolffkran.

Other works consulted in the preparation of the study include:

- Ernst, H. Die Hebezeuge, Bemessungsgrundlage, Bauteile, Antriebe. Braunschweig: Vieweg Verlag, 1973.
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2. MANAGEMENT OVERVIEW

PURPOSE OF THE STUDY

The purpose of this study is to evaluate operational cranes in shipbuilding yards, and to report on the most cost-effective design for U.S. shipyards.

CRANE DEVELOPMENT

In Europe, a rapid expansion of shipbuilding took place during the industrial revolution; thus, from the outset, shipbuilding cranes were custom-built for dockyards. This tradition means that European shipbuilding cranes have a mini-technology of their own. In the U.S., on the other hand, the industrial revolution was concentrated on the construction industry; dockyard cranes were simply adaptations of cranes from building sites. With no tradition of independence, the design of dockyard cranes has stagnated.

SLEWING CRANES

In operation, a crane moves in four ways: traveling, slewing, luffing, and hoisting.

Traveling is different from crane to crane only in that a disproportionately heavy crane requires disproportionately heavy track, and that a wide crane requires inconveniently wide track.

In slewing operations, the essential problem is the way in which the slewing part of the crane is mounted. This can be either a column system, as favored in Europe, or a turntable system, as is traditional in the U.S. Various designs based on columns are possible, the latest featuring the centerless roller-race bearing.

Luffing is the movement of the boom allowing the load to travel inwards and outwards. In normal luffing the load travels not only horizontally, but is also raised and lowered with the boom. In level luffing, the load-path is horizontal only; this offers a saving of energy and a significant increase in accuracy. Perfect level luffing is achieved only by a double-boom design, but a very close approximation is offered by a "rope-store" built into the hoisting mechanism.

Hoisting operations vary from crane to crane only in that quasi-level luffing requires a rope-store. However, an important concept in crane specification arises here. A modern crane should achieve a "constant load moment." This means that when the load raised $\times$ is multiplied by the range of the load $\times r$ (i. e., the distance between the load and the center-line of the crane), the end-figure $\times lr$ should remain constant. In inefficient cranes, especially in turntable cranes, the loads raised at extended ranges are far too small to achieve this standard.

COMPARATIVE EVALUATION

Five cranes were evaluated. These were: <1> the double-boom (gooseneck) crane, chosen because it achieves perfect level luffing; <2> the single-boom crane with rope-store level luffing, chosen because it is still the workhorse of shipbuilding outside the U. S.; <3> the balanced boom crane (also with rope-store level luffing) -- an older design but still in use in the Far East; <4> the single-boom crane without level luffing, chosen for its simple, basic design (although it is a state-of-the-art product); and <5> the turntable crane -- the standard American design to date.

In comparing operational efficiency, the first criterion is speed. Although high-speed luffing is a feature of the gooseneck and balanced boom, speed has little practical value. The speed of all other movements depends not on the design itself but on the drive units. Mere speed is not, however, a real advantage in operation: smoothness, i.e., stepless, shock-free operation, is far more important.

As to accuracy, the gooseneck enjoys some advantage; the other four types are equivalent.

In terms of energy consumption, the gooseneck again has a slight edge in luffing operations. This advantage is offset by the additional weight of the crane which creates an energy deficit in nearly all other operations.

In summary, no crane had a decisive advantage in efficiency. In examining costs, however, clear distinctions did arise.

First, analysis of maintenance costs (poorly recorded in most yards) shows simply that complicated cranes (goose-neck and balanced boom) cost more to maintain. Safety and training costs show no interesting divergences. The most significant difference arises in the "cost-to-build" factor. This is, obviously, not the same as the price of the crane. To find relative "cost-to-build," the weight of the crane and the labor applied per ton must be netted, always assuming that cranes of the same load moment are specified. The results here are significant. Taking the single-boom crane with rope-store level luffing as standard (i. e., it has a cost-to-build multiplier of 1.0), then the cost-to-build for a goose-neck crane is a prohibitive 1.5 (one and a half times" as much). The balanced-boom crane is also expensive at 1.3. The turntable crane (not strictly comparable because of its poor performance) has a multiplier of 1.2. Only the single-boom crane with normal luffing is cheaper to build, with a multiplier of 0.9.

RECOMMENDATION

Two cranes seem to be most cost-effective, taking into account their different capabilities. For heavy-duty cranes, where level luffing is seldom of practical significance, the single-boom column crane with normal luffing is recommended; for medium-duty, fitting-out operations, the single-boom column crane with level luffing. For light duty, the exact application of the crane should be studied to see if the benefit of a level luffing rope-store is worth the extra cost; in any case a single-boom crane with top-mounted race bearing is again the most cost-effective choice.

3. THE EVOLUTION OF SLEW CRANES IN SHIPYARDS

The purpose of this section is to contrast traditional shipbuilding cranes in Europe and the United States; the two different evolutionary patterns explain the radically contrasted designs that have emerged on the two sides of the Atlantic.

3.1. THE EUROPEAN PERSPECTIVE

Modern shipbuilding dates back to the First Industrial Revolution, to the dawn of steam, electrical power, and the mass production of rolled steel. The colonial empires of Britain, France, Germany and Holland required enormous merchant fleets as well as powerful navies to protect the sea-lanes. Accordingly, shipyards proliferated along the coasts, usually concentrated in areas where boatbuilding was a tradition dating back to medieval times.

At first, the building technique remained traditional: the steel hull was constructed on an inclined sliding berth (or slip way) as if it were simply a larger and heavier version of the wooden hull. Soon, however, the tremendous demand for new vessels obliged yards to lay down three or even four parallel slip ways. Batteries of cable cranes with 5-, 10-, or 20-ton trolleys worked on the hulls. In more sophisticated yards, an overhead crane system was installed which allowed multiple lifts involving up to ten trolleys — such lifts must have presented formidable problems of load command and placing. Figure 1 shows the liner Europa being launched in Hamburg from under an overhead system of this type in the old Howaldt yard.

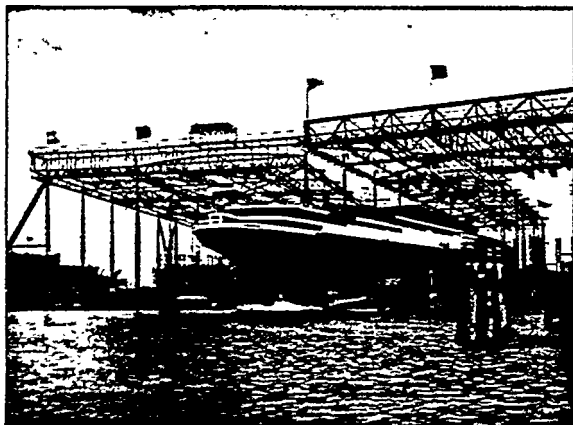


Figure 1:

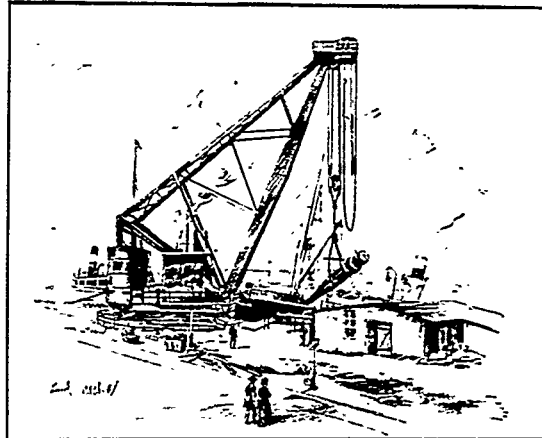
The launch of the Europa, Hamburg, August 15 1928, from under a system of cable crane trolleys.

The significant fact here is that no heavy-lift cranes were required for hull building as such, since the basic construction technique was

rivetting; it was only when the ship was fitted out that heavy-lift cranes were required. The fitting-out of the hull was performed alongside a quay on which a heavy-lift slew crane was erected. The oldest known crane of this type was built in Hamburg in 1887: it was steam operated and had a capacity of 150 tons. (See-Figure 2)

Figure 2:

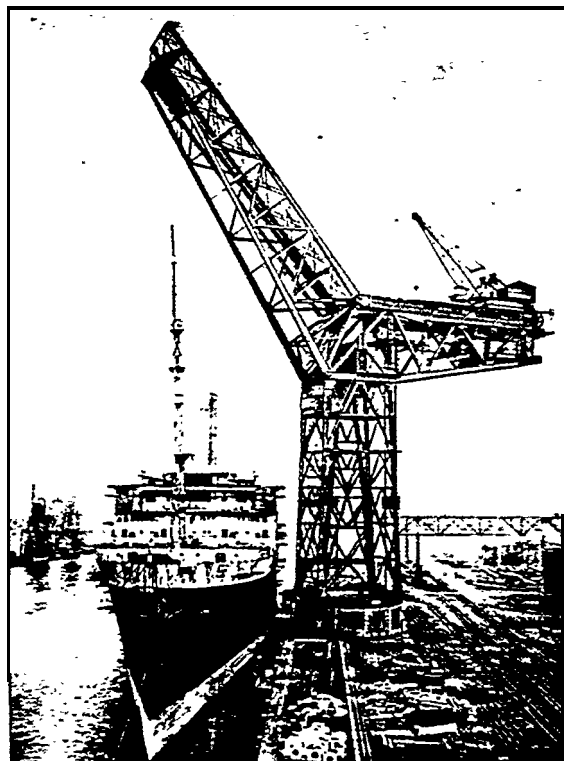
Heavy-lift crane of
150 tons capacity.
Built in Hamburg, 1887.



This was the era of enormous, hammerhead cranes with capacities of 100 tons at 150 feet or more. Due to their vast weight (up to 2000 tons), such hammerheads were often stationary — it was cheaper to move the hull along the quay than to build rails for the crane to travel. One such crane is shown in Figure 3; it was electrically operated, and was installed early in this century in the Blohm and Voss yard in Hamburg.

Figure 3:

Stationary fitting-out
crane, 250 tons
capacity, electrically
operated, installed at
Blohm and Voss, Hamburg,
here shown in 1914
fitting a mast to the SS
Vaterland. The crane
was destroyed in 1944.



The design of such cranes resulted directly from the social and technological conditions of the early twentieth century. Labor was cheap while materials were expensive; therefore, little thought was given to labor costs in building, in maintaining, or in operating the cranes. From the technological viewpoint, steel rolling mills produced essentially angles, flats, bars and beams; plate-makers could produce only heavy boiler-plate manufactured to rather wide tolerances. The primary construction technique was rivetting. For crane design, all this meant that heavy-lift slew cranes were more or less restricted to quayside, fitting-out operations.

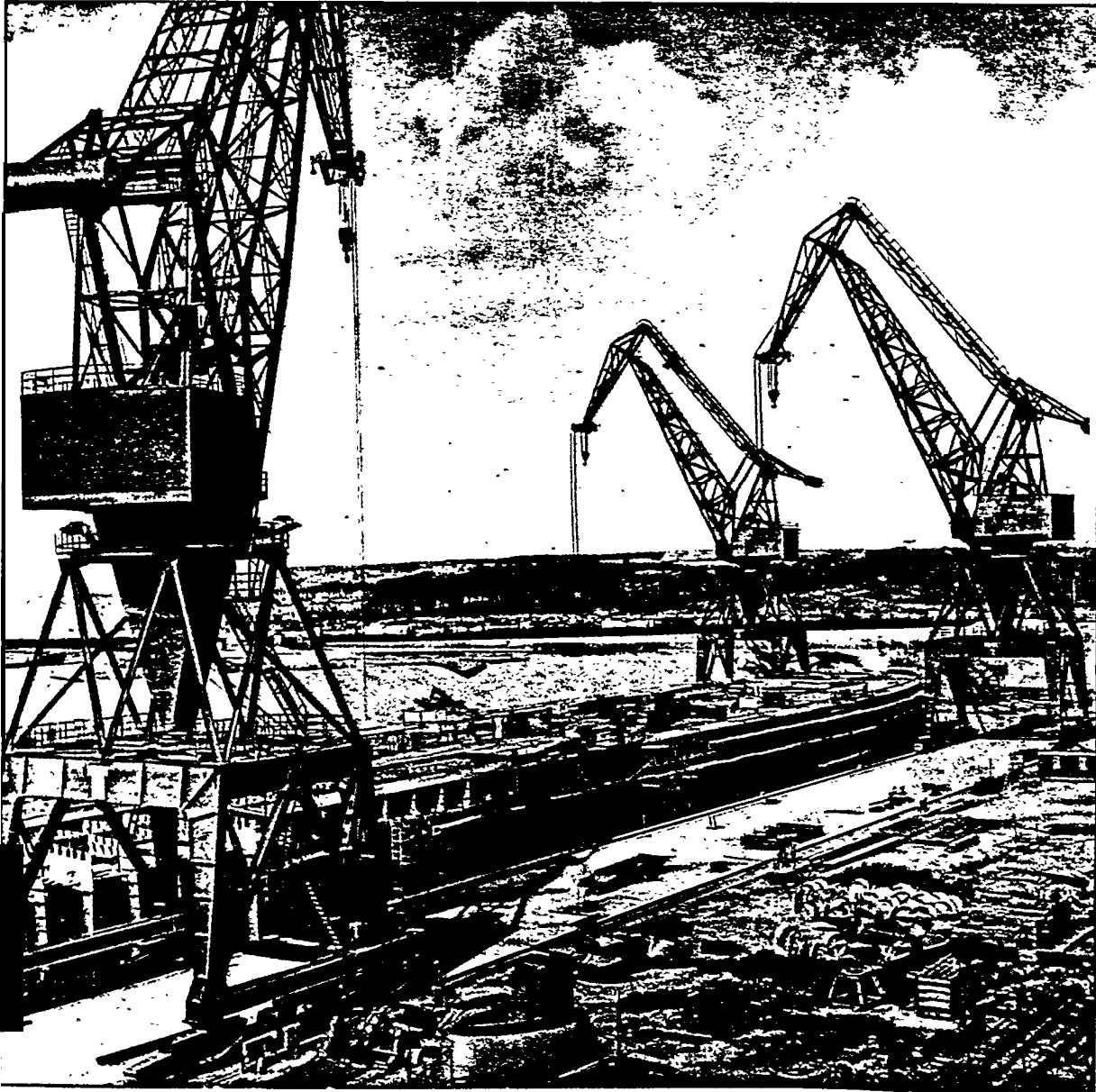


Figure 4: Early 60-ton gooseneck cranes (photographed after the Second World War) operating in Goteborg, Sweden.

The next step in crane design was the introduction of goose-neck cranes. These elegant structures brought with them certain advantages over the old hammerheads: they were lighter for their capacity and were therefore easier to put on tracks; a relatively small gooseneck could reach over high superstructures; and the short unguided rope length allowed accurate load placement. (All these features will be presented in detail in later chapters.) Figure 4 shows a fairly early set of gooseneck cranes with the characteristic lattice design.

After the devastation caused by the Second World War, most shipyard equipment had to be replaced -- this was the golden age of the gooseneck. Figure 5 shows the slipway system of the Blohm and Voss ship-yard in Hamburg with seven post-war gooseneck cranes in operation.

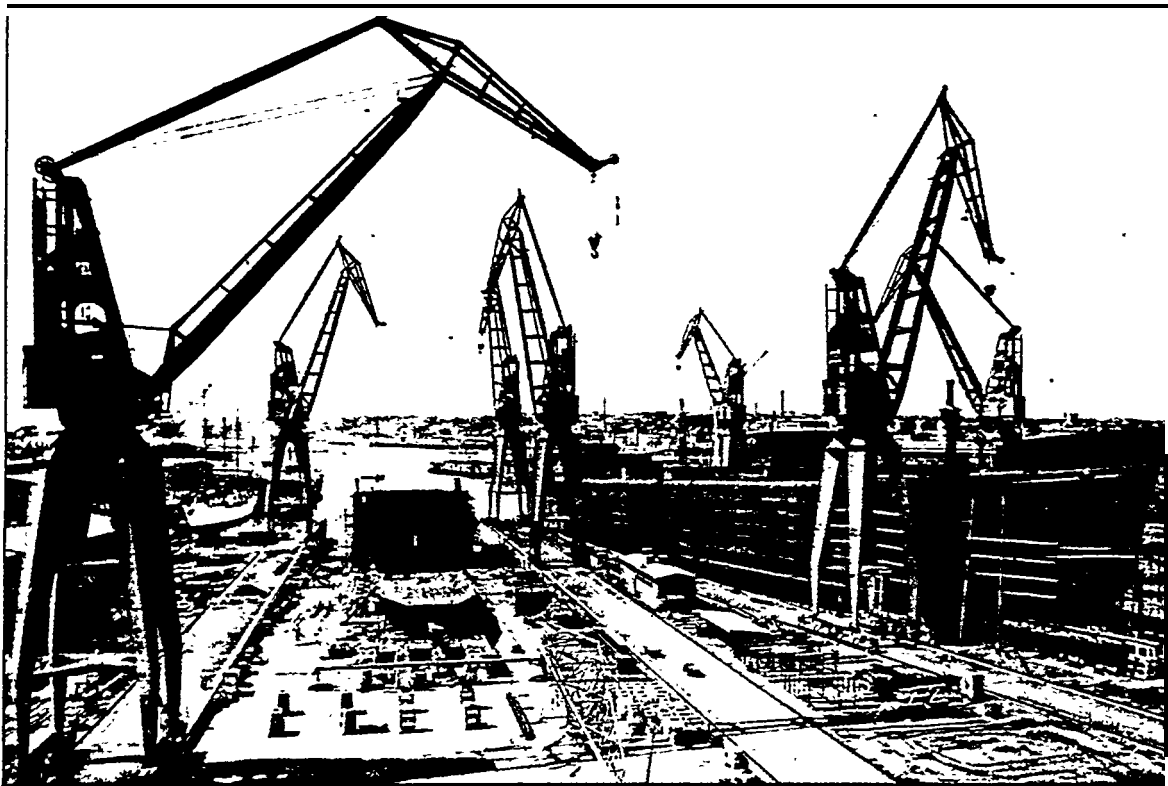


Figure 5: Double-boom (goose-neck) cranes operating on a slipway system in the Blohm and Voss yard, Hamburg, 1955.

3.2. THE AMERICAN ALTERNATIVE

In the United States a different situation emerged. Firstly, without an empire to colonize, to trade with, and to protect, there was a less massive demand for large steel boats. In fact while Europe was building its navies, the United States was experiencing the skyscraper boom. For building these immense steel structures, a different type of material-handling device was required -- the mobile crane. The mobile crane started out as a derrick on a truck, but quickly developed booms, - jibs, and, most important, its own power-source, usually a diesel engine; the key factor, in any case, was mobility. The early mechanical gear-shifting systems, which were jerky and suffered excessive wear and tear, were soon replaced with primitive, but nevertheless much smoother, hydraulic systems. Such cranes were cheaply produced in enormous numbers. As to shipyards, from the outset they adopted the mobile cranes developed for the building industry, set them atop high, steel platforms, and used them for ship-building~ Figure 6 Shows a Dravo Whirler C-17, typical of such cranes; it had a capacity of 60 tons at 35 feet, and 20 tons at 80 feet, figures that are perhaps more appropriate on a building site than in a shipyard.



Figure 6

A Dravo Whirler
C-17 in an East
Coast shipyard.

Capacity:
60 tons x 35 feet;
20 tons x 80 feet.

In summary, two shipyard traditions developed on either side of the Atlantic. In the United States, the cheapness and availability of mobile cranes developed for building sites evidently persuaded ship yards to adapt them for shipbuilding purposes. In Europe, on the other hand, the enormous demand for ships caused shipbuilding cranes to develop as machines in their own right, specially tailored to the needs of the shipyard. They tended to be: electrically powered rather than diesel powered, and to be custom built rather than mass produced.

Once a sufficient body of specialized knowledge was available, standards for engineering crane structures and rope systems were instituted, for example, DIN 120 in Germany and BS 2573 in England. These standards, constantly reviewed, offer the theoretical underpinning of the highly sophisticated European crane industry.

3.3. THE GOLIATH CRANE

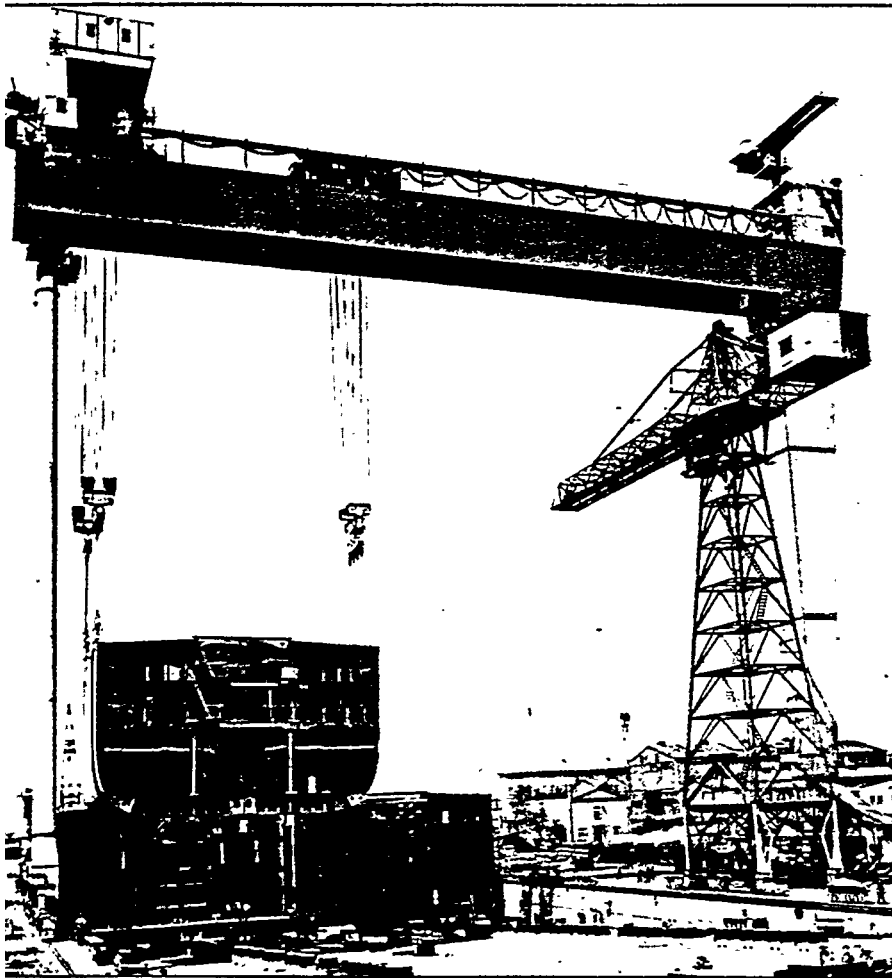


Figure 7: 500-ton Goliath in Bremen in 1965, showing main and turn-around crabs. Height: 177 ft (54 meters).

Figure 7 shows a 500-ton Goliath unit towering 177 feet above a gravity dry dock. Although this study concerns itself with slew cranes, it may be appropriate to mention the Goliath crane, since it has had some influence on the market for slew cranes. The Goliath seen here has two crabs, the main one, and a turn-around crab which enable it to turn 500-ton hull sections for easier welding. Despite the usefulness of this maneuver, the Goliath is costly to install and costly to run. Characteristically it has 2000 kW of installed power; this makes it absurdly expensive in operation since 90% of lifts, even in the construction of vicc vessels, are in the range of 300 lbs - 6000 lbs.

For smaller, lighter loads a smaller, lighter crane is needed. This has meant that the Goliath units have, perhaps unexpectedly, created a market for auxiliary, supporting units. Originally, as seen in Figure 7, a tower-crane was used. Today, as seen in Figure 8, a slew crane of some sort would be the answer.

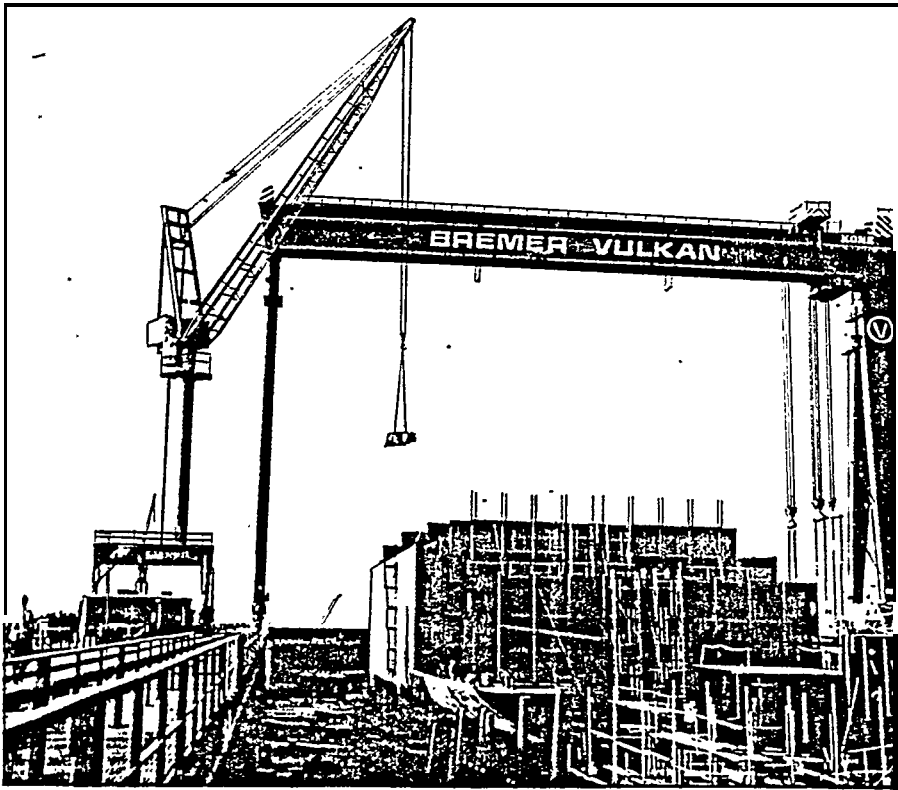


Figure 8: Slew crane in the Vulkan Shipyard, Bremen, 1974, in support of a Goliath.

4. THE SLEW C R A N E :

A SYSTEM OF ACTING AND REACTING FORCES

The purpose of this section is to outline the basic engineering principles involved in the construction and operation of slew cranes.

4.1. SLEW CRANES

4.1.1. Introduction

Crane development has been characterized by the the need for ever bigger cranes with ever faster operation. Along with increased load-handling and speed, the crane-user has demanded economy and reliability. These demands have forced crane-designers to refine their engineering techniques, achieving better results from lighter, more cleverly designed structures into which increasingly powerful machinery can be installed. This has meant applying the principles for the design of lightweight. welded steel structures with ever greater rigor and finesse.

In operation a crane is subjected to four kinds of forces. The least important are the atmospheric forces brought to bear on the structure: wind, ice, snow. The operational mass of the crane itself, as it moves, naturally generates a significant pattern of forces. Less obvious, but no less important, are the forces at work in the support-structure of the crane, the track-wheel system on which it runs. Finally, and most important, are the forces exerted by the load to be moved.

The relationships among these forces are now well understood. Further, materials science has developed to the point where the patterns of stress can be closely tied to the performance of the structure, allowing maximum efficiency at no sacrifice of absolute safety. In these technological developments, computer analysis has played a key role.

4.1.2. The Slew Crane: Definition

AU cranes offering a circular field of operation are called "slew cranes." (See Figure 9.)

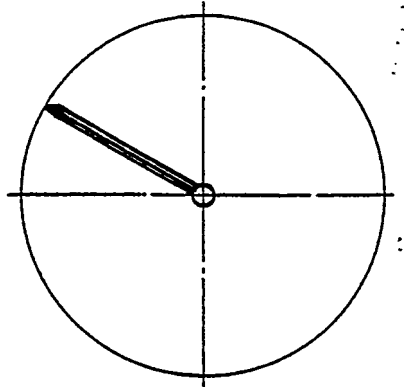


Figure 9

Circular Field of Operation
of Slew Crane

With the addition of a luffing boom, the field of operation may be broadened to a circular ring area, as in Figure 4.2.

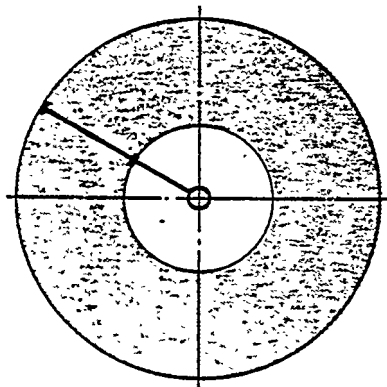


Figure 10

Ring Field of Operation
of Luffing Slew Crane

With the addition of a traveling system, the area of operation can be elongated at will, as in Figure 11.

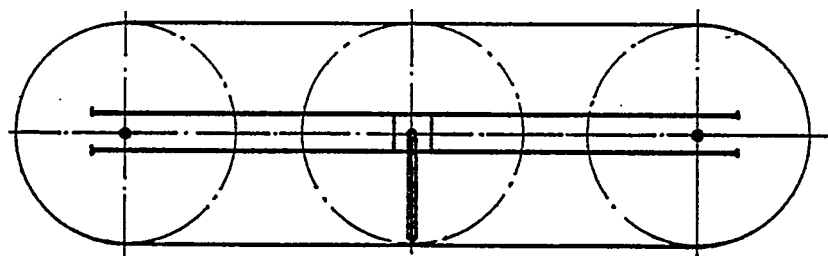


Figure 11: Maximum Field of Operation of Traveling Slew Crane

4.1.3. The Slew Crane: Description

There are many technical solutions to the problem of allowing a crane to slew, but essentially all slew cranes consist of:

- (a) a boom or system of booms that permits a hook or tackle to reach outward from a center. See ① in Figure 12.
- (b) a structure to support the boom. See ②
- (c) housing(s) to allow convenient placement of machinery. See ③
- (d) a substructure which supports all the above. See ④

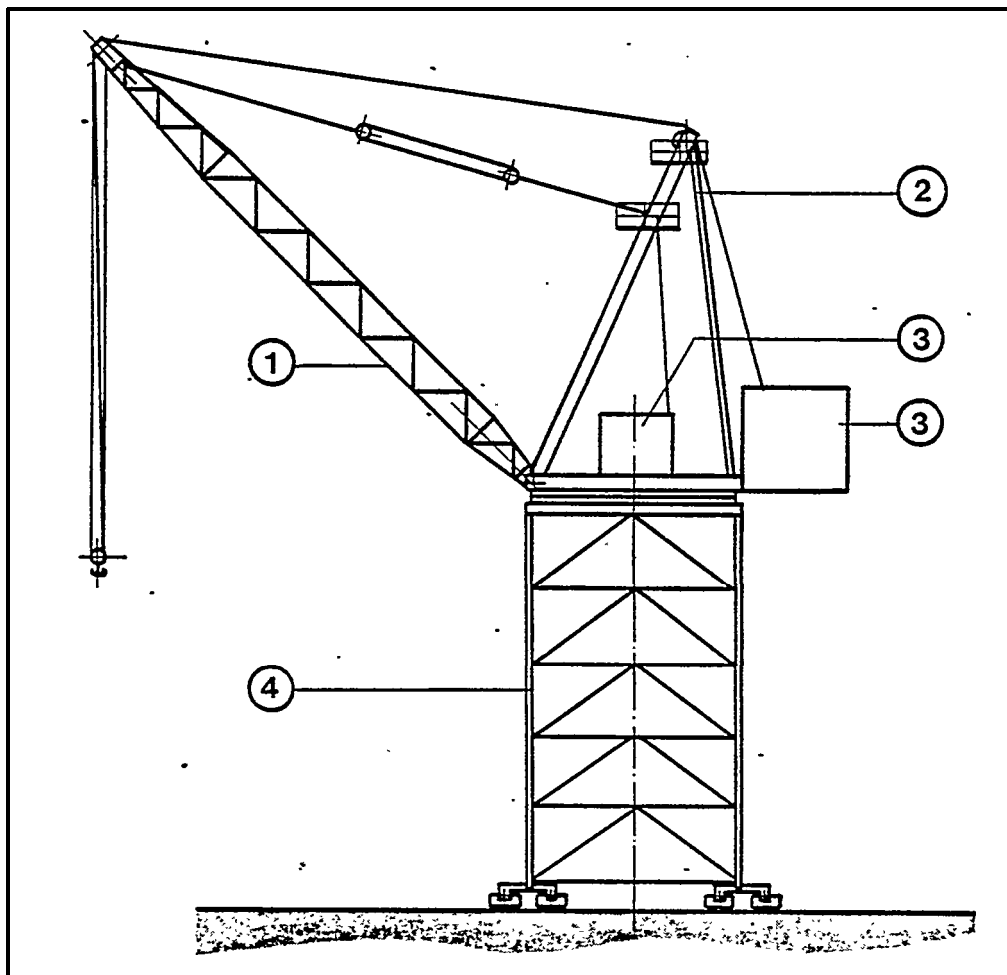


Figure 12: Main Parts of a Slew. Crane

In essence a crane, by using a pulley system, combines two of mankind's most basic inventions -- the wheel for turning forces round corners, and the rope for extending the human arm -- thus allowing the lifting and moving of great weights.

4.1.4. The Slew Crane: Operation

4.1.4.1. Hoisting

Modern cranes operate on the basis of a design load moment. (For details see Section 5.2.1.) In brief: a shipyard might require a crane able to lift 50 tons at 20 meters (66 ft). The designer multiplies these two figures together, and builds a crane of 1000 mt (or 7360 ft kips). Since the figure 1000 mt (7360 ft kips) remains constant, it is clear that the crane will lift 25 tons at 40 meters (131 ft), or 100 tons at 10 meters (33 ft). The "constant load moment crane," i.e., a crane with a capacity that varies with range, is standard in shipbuilding. It is shown schematically in Figure 13.

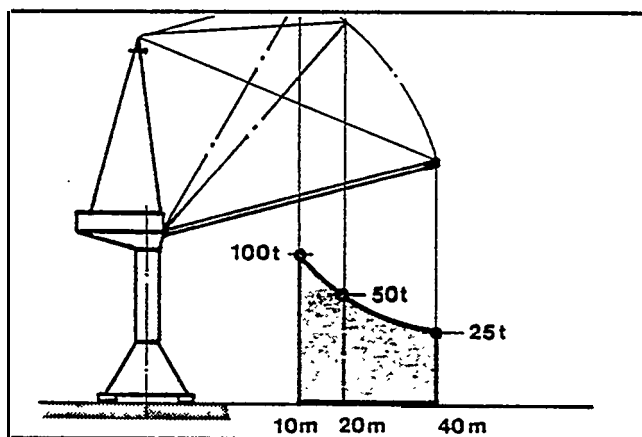


Figure 13:

Constant Load
Moment Crane

There is another possibility: the shipyard might specify the crane with a capacity of 50 tons at 40 meters, but that 50 tons would be the maximum load. In this case, the crane would be a 2000 mt (14700 ft kips) unit. Such cranes are typically used as fitting-out or as wingwall cranes. See Figure 14.

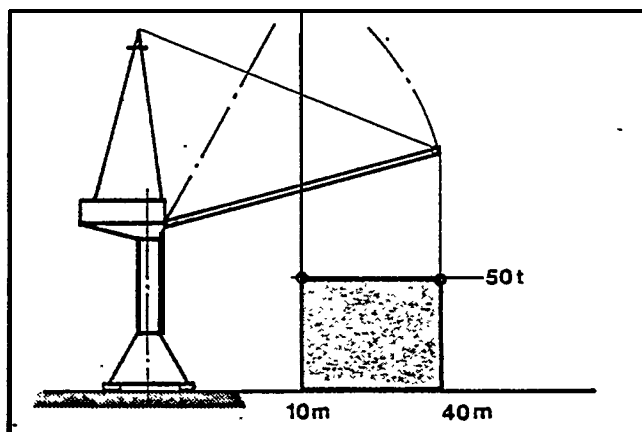


Figure 14:

Constant Load
Crane

The cost comparison here is interesting: the constant load moment crane (at least in European practice) would need hoist machinery sufficient to raise 100 tons; rope pull capacity would be 12.5 tons; two rope lines would be reeled on the drum, each with a 4-fall pulley-block for a total of 8 falls. On the other hand, the constant load crane would require heavier steel construction, but the hoist machinery would be much lighter -- only 4 falls would be needed. For this reason the constant load crane would be, in the end effect, cheaper to install.

Traditional American crane design has produced a further variant of this pattern, a design in which there is an enormous difference between the load capacity at minimum and at maximum extension, and in which the load moment (= load x extension) does not remain constant but falls off drastically as range increases. Perhaps this type could be labeled the "non-constant load moment crane." For example, one Clyde standard model (the 28 E, discussed as Crane 5 in the next chapter) can lift 190 000 lbs (86 metric tons) at 45 feet (13.7 m). This is impressive. The load moment at this minimum extension is 8 550 ft kips (1204 mt). At maximum extension (160 feet), however, the load is a mere 37 000 lbs (17 metric tons). This produces a far lower figure for load moment: 5920 ft kips (833 mt). This collapse in performance is well below the normally accepted international standard found in constant load moment cranes. What accounts for this deficiency? The center of gravity of turntable cranes must be kept within the radius of the turntable: this feature is inherent in the design. The unfortunate results are twofold: first, at maximum extension the crane lifts extremely unfavorable loads; secondly, the heavy machinery installed to hoist enormous loads at minimum extension will be ever more underutilized (by international standards) as range increases. This underutilization not only puts up the purchase price but also increases running costs. Figure 15 shows a crane of this type.

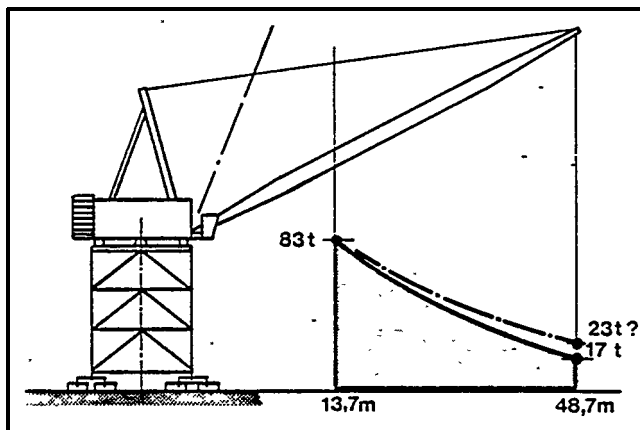


Figure 15:

The "Non-Constant Load Moment" Crane

A word about hoists. Normally a slew crane is equipped with two hoists: <1> the heavy lift or main hoist, and <2> the auxiliary or whip hoist. The main hoist usually operates at a range of load/speed ratios, with a slower speed for heavier loads and a higher speed for lighter loads. The whip hoist takes only light loads, but moves them very quickly.

The main criterion in the art of hoisting is that the load moves through the shortest possible distance. This means that the crane must perform all operations (lifting/lowering, luffing, slewing, traveling) SIMULTANEOUSLY. This requires great skill on the part of the crane-driver. For this reason, the driver should be situated high in the crane to allow the best possible view of the field of operation. Further, the driver's cabin should move with the slewing part of the crane. For the actual maneuvering of the load, the driver will have an easy-to-operate "joystick" in each hand, ergonomically designed so that the movement is "natural." In other words, right-hand pull brings the load up; right-hand push lowers it; right-hand right slews the crane to the right, right-hand left slews it left. The joystick in the left hand controls luffing and traveling operations.

4.1.4.2. Mechanical Operation

The prime movers that raise loads, either the cargo or the boom itself, are electric motors or diesel engines. Between the prime mover and the rope drums are either mechanical reduction gears or a hydraulic reduction system.

The main object of a modern drive is to adjust speeds to actual needs as efficiently as possible. Electrically, this may be achieved by slipringmotor-eddycurrent brakes and / or a stepped variable resistor; stepless adjustment is achieved by the use of a DC-drive. Superior performance is offered by a hydraulic adjustment which allows stepless control from zero to full power.

4.1.2.3. Slewing

A crane must slew while loaded; the engineering problem here is to design a slewing mechanism that will bear all the gravitational forces plus the "overturning" moment set up by the load. Three systems (with minor variations) are in use:

System 1: The turntable with kingpost

A turntable crane is shown in Figure 16. The center of gravity without load is shown as (G-1), while the center of gravity with load is shown as (G). Both centers remain within the radius of the turntable (r). Support rollers (R) running on a circular rail bear the total weight. The kingpost (K) is merely a means of centering the turntable and bears no part of the load.

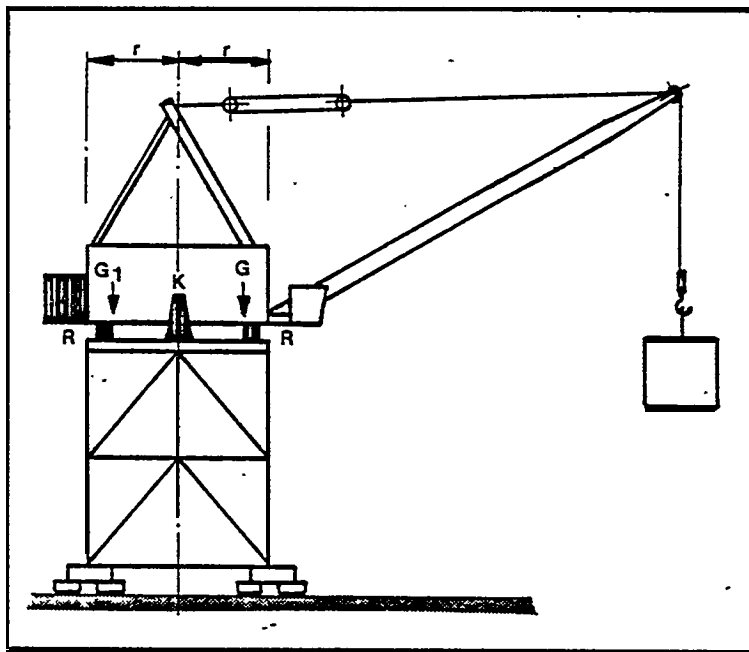


Figure 16: Turntable Crane with Kingpost

This system has a number of disadvantages: first, it requires a very wide turntable if heavy loads are to be lifted -- an excessively wide substructure is then also necessary; secondly, heavy loads require installation of an exorbitantly large number of rollers; finally, as the crane ages, adjustment between rollers and kingpost becomes increasingly difficult.

System 2: Column crane with upper and lower bearing

A two-bearing column crane is shown in Figure 17. In this design the weight of the slewing part of the crane and of the load are borne by the lower bearing, while the crane's "overturning" moment is distributed between the upper bearing and the lower.

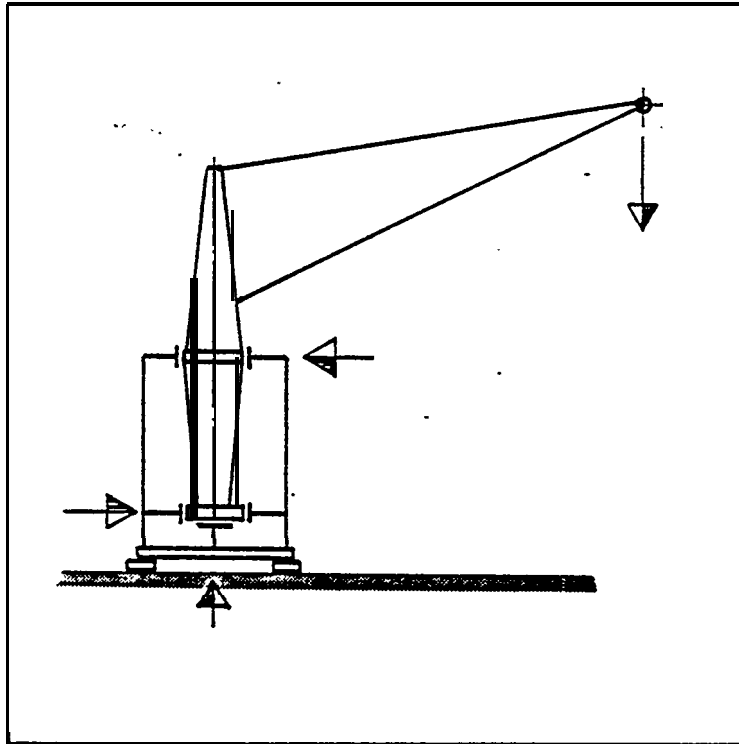


Figure 17: Column Crane with Upper and Lower Bearing

This design is better than the turntable for heavy cargo cranes since the lower bearing requires comparatively little fine adjustment -or other maintenance. The main disadvantage of this design is the extreme weight of the central column: a large amount of energy is required to turn it, and a bulky structure is required to support it.

System 3: Crane with centerless roller-race bearing

A crane with a centerless roller-race bearing is shown in Figure 18. It is clear at a glance that it differs radically in appearance from the two earlier designs.

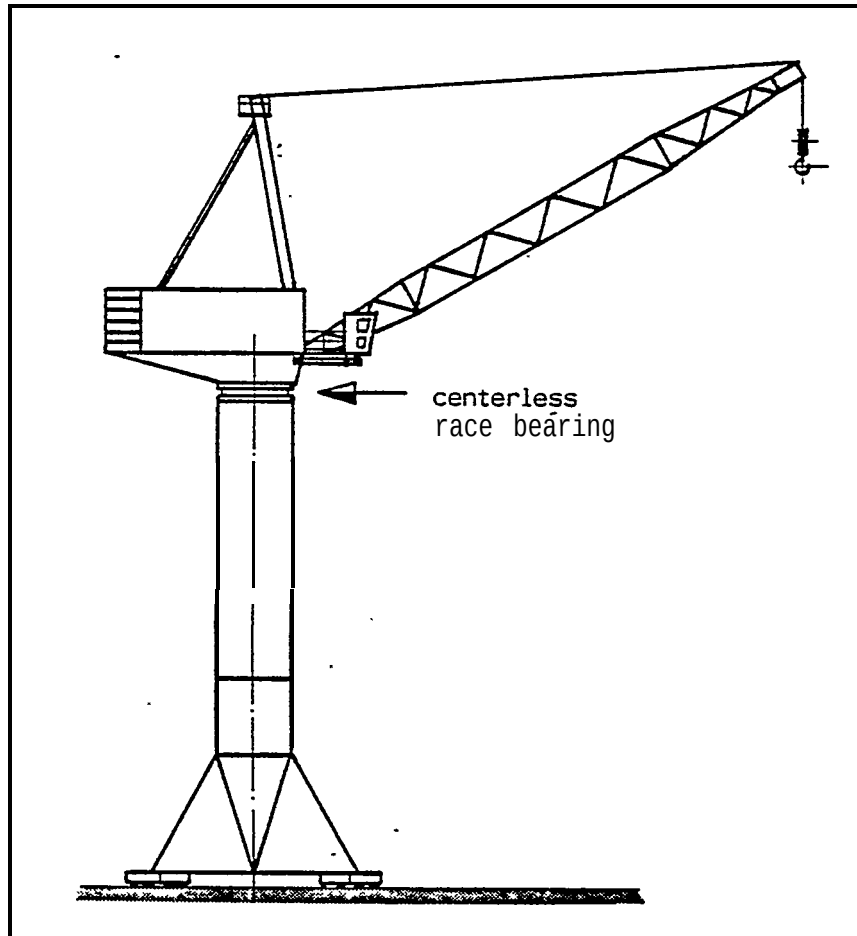


Figure 18: Crane with Centerless Roller-Race Bearing

The secret of the design is the bearing itself. It is shown in cross-section in Figure 19. The bearing normally consists of three heavy steel rings. The lower ring is bolted to the stationary structure; the upper ring is bolted to the slewing structure; the middle ring holds the other two rings together and houses two sets of antifriction roller or ball elements.

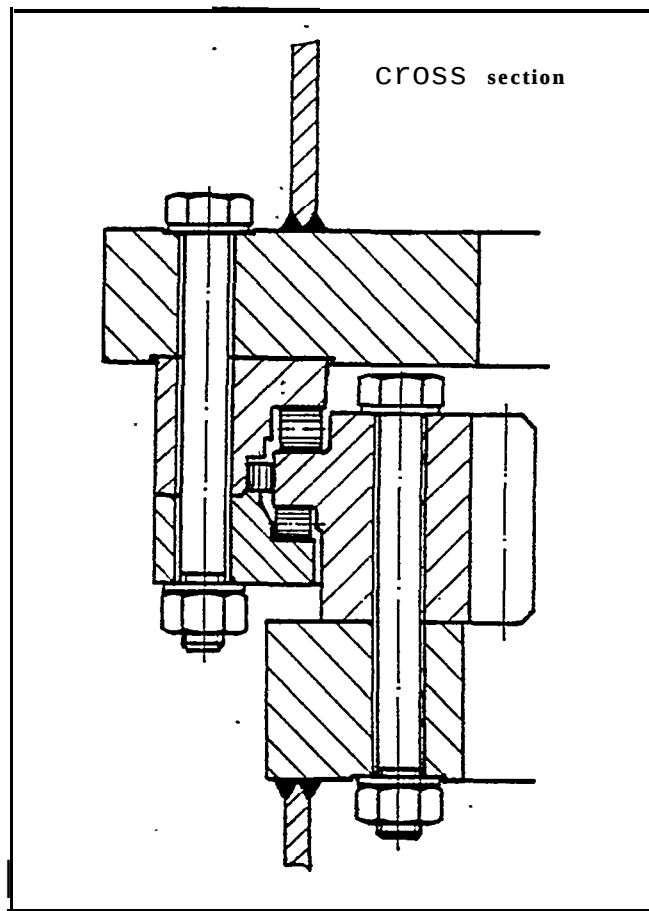


Figure 19: Centerless Roller-Race Bearing

This oversized "antifriction" bearing can handle both vertical forces and the "overturning" moment of the loaded crane. It can be manufactured to meet the heaviest capacities required today. Care must be taken; however, to manufacture the support structure with sufficient rigidity since the bearing itself allows almost no "bending."

The centerless roller-race bearing has a number of very important advantages: first, because it is centerless, it offers a wide aperture allowing convenient access between the stationary and the slewing part of the crane; secondly, the design is extremely safe and requires little more than "routine greasing by way of maintenance. Finally, the elegant narrowness of the design can be extremely useful: if long trucks or heavy wheel-loads can be accommodated, - then rather narrow gauge track can be used; further, such slim luffing cranes may work in close proximity allowing twin or even quadruple lifts. This kind of flexibility is not possible with other, less up-to-date designs.

4.1.4.4. Safety

A slew crane in operation today can be made virtually foolproof. With a state-of-the-art crane, all movements are controlled by limit switches, electronic guards, loadmoment and overload-control devices. Emergency appliances in case of power failure -- cut-outs, manual but controlled brake lifts, and cross-over arrangements -- are fitted. The cranes, as their safety records show, are as foolproof as human ingenuity can make them.

4.1.5. The Supporting Structure

Railroad accidents have become more common in the United States as old roadbeds decay; in fact, there is little point developing modern locomotives and rolling-stock if substandard track obliges them to run at 20 mph. The same principle applies to cranes. The track on which a crane runs is as important as the crane itself.

To make clear why this is so, consider the forces operating on the track. When a traveling crane accelerates or brakes, it generates dynamic forces which act parallel to the track; slewing movements generate crosswise forces (or "horizontal load") which the track must withstand. As well as dynamic forces, the track must also withstand static forces exerted by the structure of the crane; these forces vary in operation as the balance and center of gravity of the crane change. These static forces are exerted parallel and/or crosswise to the track. Figure 20 shows these forces at work.

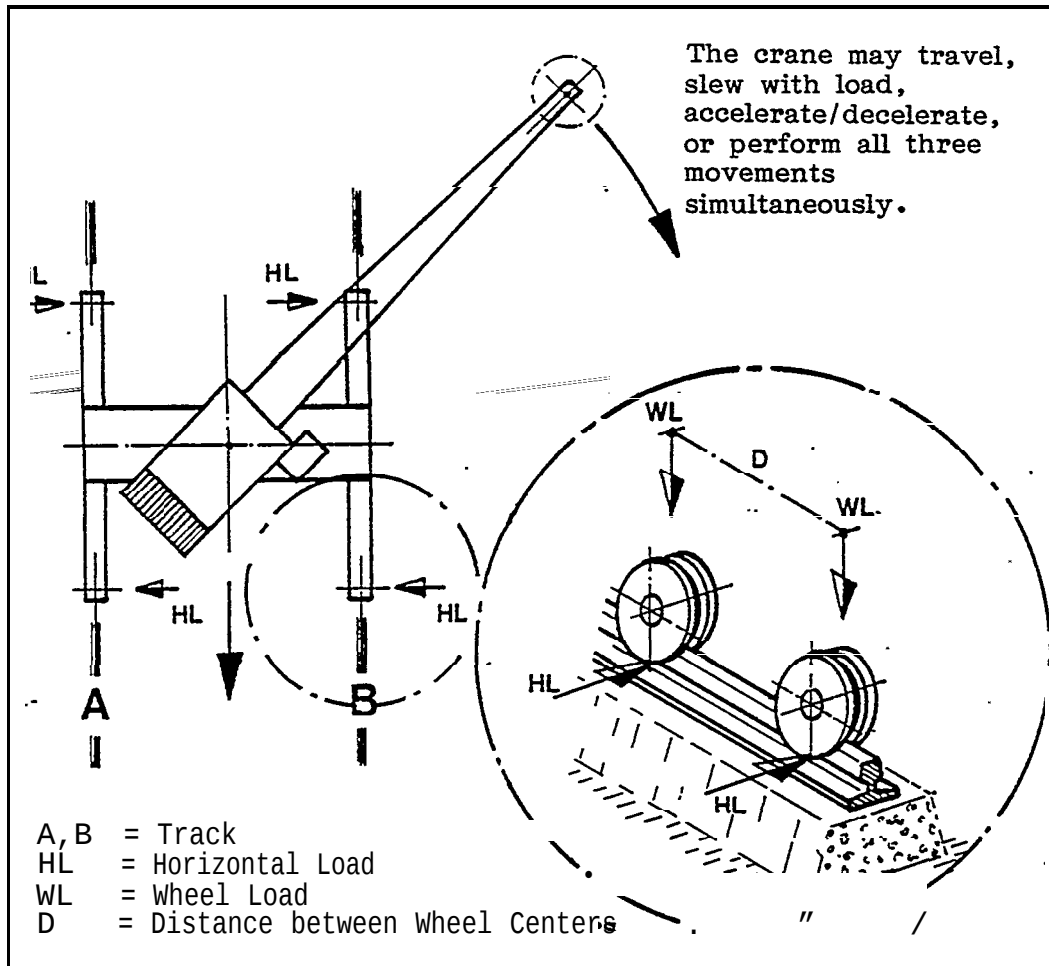


Figure 20: Reacting Forces Operating on a Slew Crane

The wheel-load (WL) bears vertically on the track. The design assumption is that WL is distributed maximally through a 45 degree cone into the supporting substructure (concrete, packed gravel, etc). The horizontal load (HL) is exerted via the flanges of double-flanged wheels against the rail head; its effect is to bend both the rail and the rail-support. The design assumption is that HL will not exceed 1/10 of WL.

During the operational life of a crane, the forces bearing on the track show a marked tendency to increase. The increase occurs because of wear and tear on the guide system and the heads of the rails themselves; the geometry of the track may also change from the settling of the soil, from corrosion, or from other causes. The increased forces caused by all these changes must obviously be allowed for.

The potential for change in track geometry has an important influence on the design of the portal structure: on perfect track, the portal will rest on four points; corner pressure (CP) will change with slewing action; each of the bogey wheels on any- given corner will bear exactly the same load. This situation is shown in Figure 21.

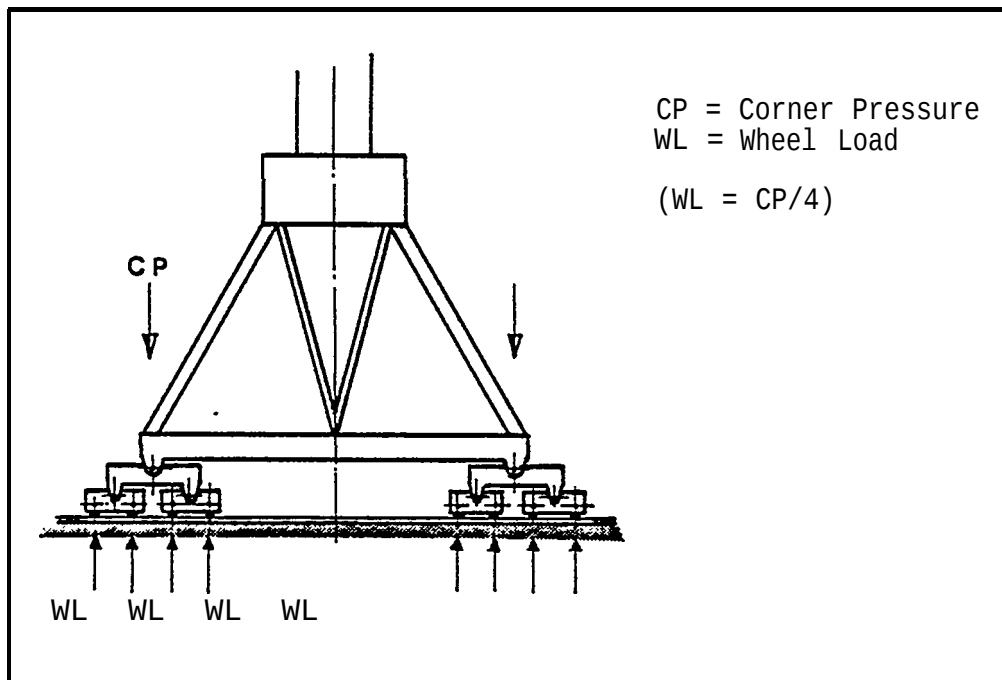


Figure 21: Corner Pressures and Wheel Loads on a Slew Crane

Given wear, or any other change in the track, this support will be reduced to three points. If the center of gravity of the crane now shifts and passes over the so-called "tipping edge" (the diagonal line between two supporting corners), there will be a moment when

the portal rests not on three but effectively on only two (diagonally opposite) supporting points. Changes in wheel pressures for a 2000-mt (114-ton) crane under these various conditions are given in Figure 22 and Table 1 below.

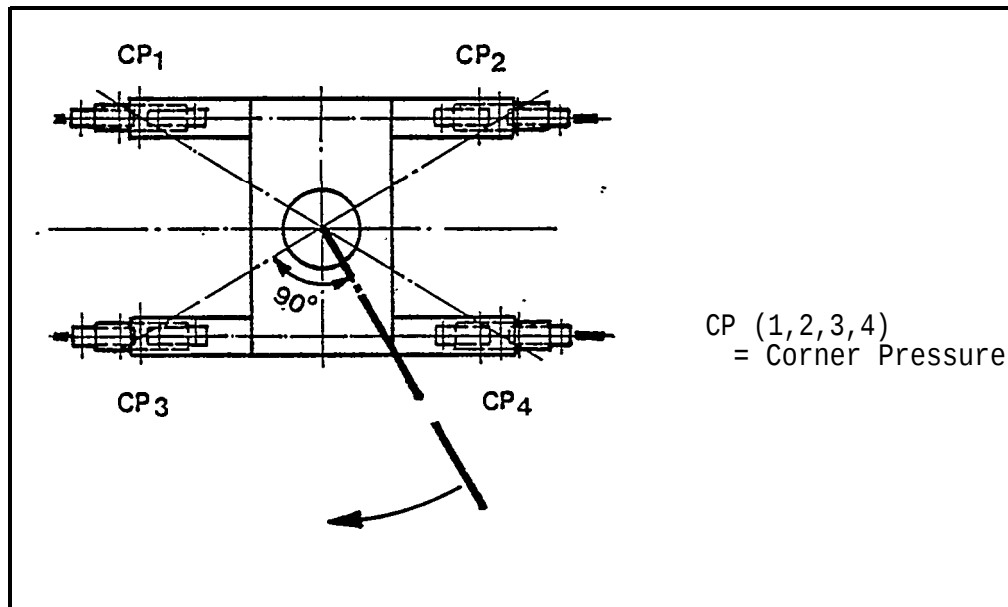


Figure 22: Changes in Wheel Load During Slewing

	Pressure (in kips) on:			
	CP1	CP2	CP3	CP4
4-point support All corners in equal ideal contact	77	268	268	458
3-point support CP1 is off contact because of worn track		345	345	381
2-point support While slewing, crane crosses "tipping edge"; CP 4 is off contact and only CP2 and CP3 bear load until CP1 comes in contact again		345	726	

Table 1: Changes in Corner Pressure for a 2000-mt (114-ton) crane on Worn Track

The significance of these figures is Clear: given poor track, corner pressure can increase from 458 kips (as assumed in the design) to some 726 kips, a figure roughly 60% in excess of what is safe. Modern portal design, i.e. , the design of the crane itself, allows for this shift of forces, but it is obviously important that the track itself be correctly designed to reduce this kind of structural stress. Design rules for the steel track itself were developed in Germany during the fifties and sixties when massive crane replacements had to be made. These rules have meant that derailments, which entail costly down-time and repairs, occur extremely. seldom. Design rules for the concrete substructure of the track have not, unfortunately, been formulated as yet. Accordingly a geological survey as well as the advice of a civil engineer and of a soil expert, is essential in deciding whether existing track can be reused or a new track must be built.

As to the future, experiments are being carried out to free slewing-cranes altogether from track by fitting them with rubber-tire bogies; this would give the crane capabilities similar to those of a rough-terrain vehicle. Regrettably, the cost of such cranes is, for the foreseeable future, prohibitive .-

4.2. LUFFING AND LEVEL LUFFING -- THEORY

4.2.1. Definitions

Luffing :

The movement of the boom of a crane upward or downward.

Level-Luffing:

The procedure of moving a load on a crane's hook (or the empty hook itself) along a horizontal (or nearly horizontal) path when the boom of the crane is luffed upward or downward.

4.2.2. Amplification

Figure 23 shows what happens when the normal luffing movement takes place. As the boom of the crane is raised, the distance between the boom-head sheave and the load remains constant; the load accordingly describes a parabolic path through the air. Figure 24 shows the load path during level luffing: the height of the boom-head sheave and its distance from the load remain constant so that the load travels horizontally.

The advantages of level luffing are that the crane's hoist does not have to make adjustments to allow for the vertical movement of the hook or load; more important, the luffing mechanism must only raise the deadweight of the boom -- no additional "work" (in the mechanical sense of the word) is required, and no unnecessary energy is consumed.

Within the crane industry, level luffing is loosely taken to mean the sum of all the methods whereby the deadweight of the boom is moved; taken into account are friction losses from the rope and pulleys, pressure loss at the boom hinge, and losses resulting from wind pressure, ice load and so on. The load proper is assumed to cause no work and to use no energy.

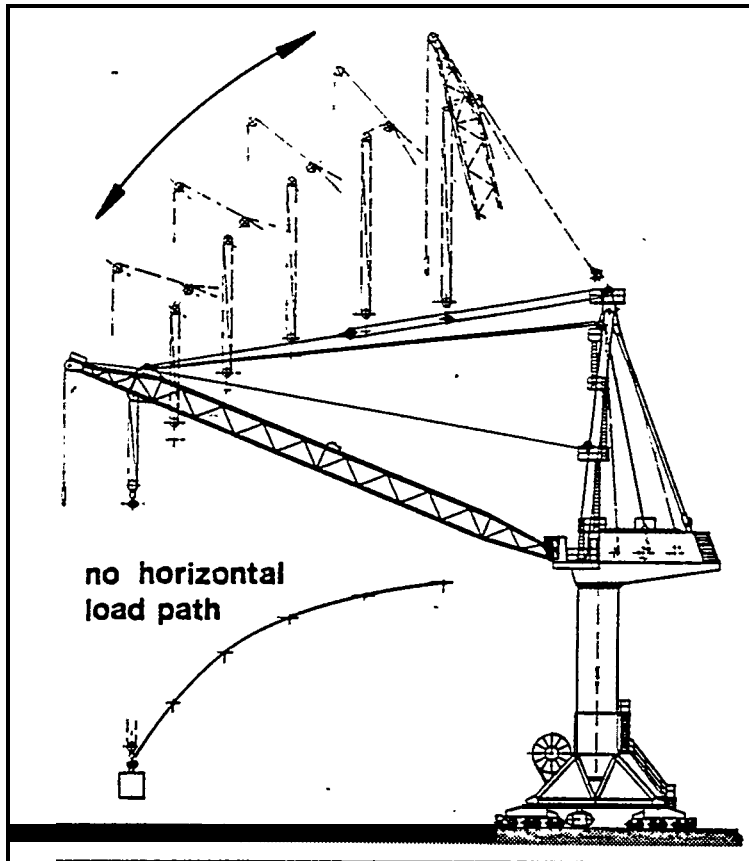


Figure 23:
Luffing Movement
Without Level
Luffing

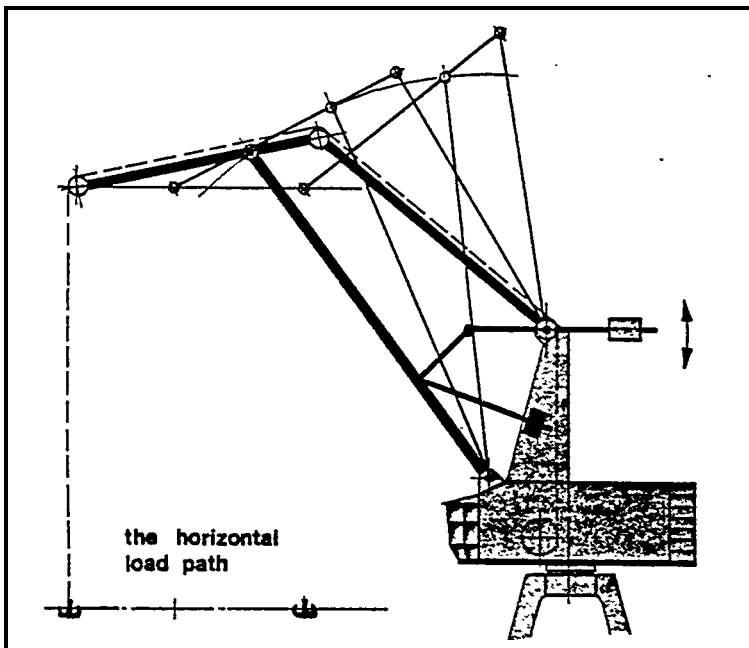


Figure 24:
Luffing Movement
Showing Level
Luffing

4.3. LEVEL LUFFING – PRACTICE

4.3.1. The Double-Boom Level Luffing Crane

Description:

The "gooseneck" crane (more properly known as the double-boom, or double-lever boom crane) is the best known means of achieving level luffing. In principle the gooseneck crane consists of three booms, two of which are hinged to the main slewing element of the crane (see Figure 25). The strut boom <3> (also called the main boom or compression boom) is hinged at its topmost end to the fly boom <1> (also called the head boom) which it supports. The boom tie <2> (also known as the tension boom) is hinged to the inner end of the fly boom and to the highest point of the main crane structure. The main boom is raised and lowered by means of luffing drive <5> which may be of a screw type, rack-and-pinion type, or hydraulic type. A counterweight <4> compensates for the weight of the main boom.

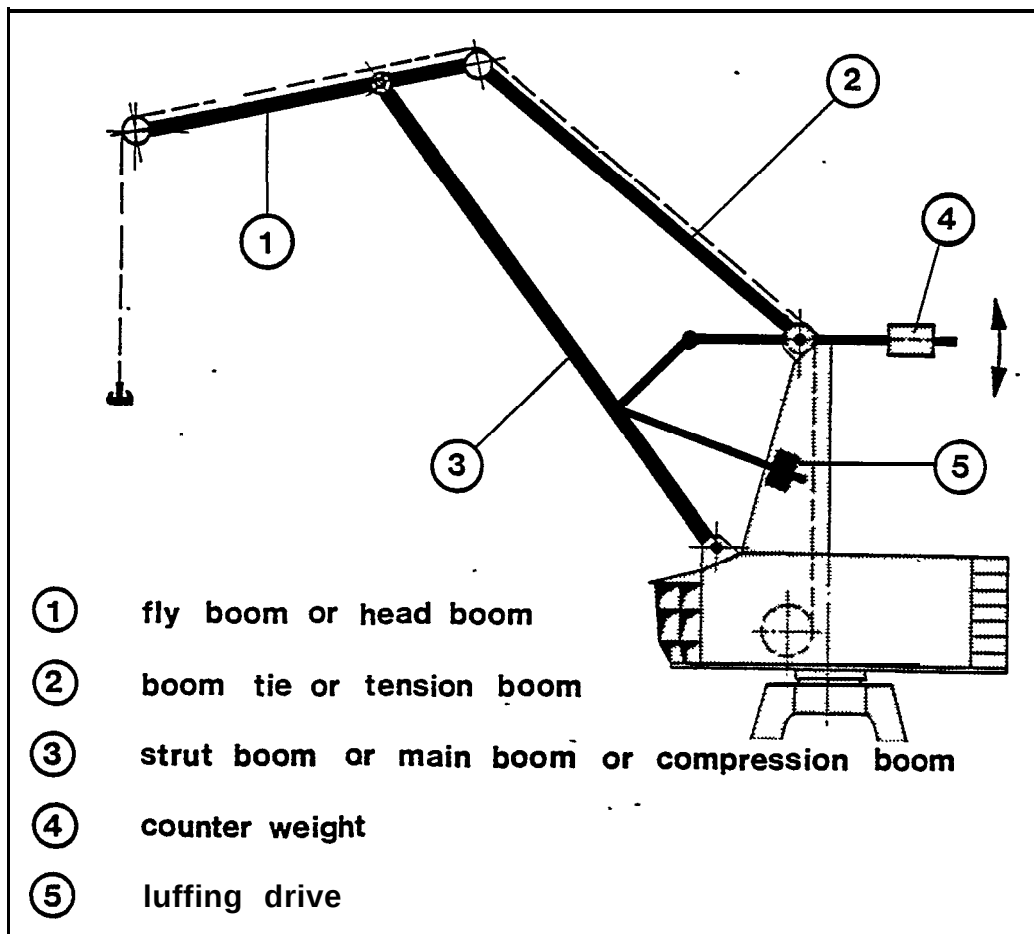


Figure 25: Major Parts of the Double-Boom (Gooseneck) Crane

Operation:

The quadrilateral formed by the three booms and the structure of the crane is so constructed that when the main boom is raised or lowered, the outer end of the fly boom remains at a constant height (see Figure 26). The hoist rope is made to travel over three pulleys located at the lower end of the boom tie, at the hinge between the boom tie and fly boom, and at the outer end of the fly boom; its length is thus unaffected by the raising or lowering of the strut boom. The end effect is that the hook/load remains at a constant height and perfectly horizontal level luffing is achieved.

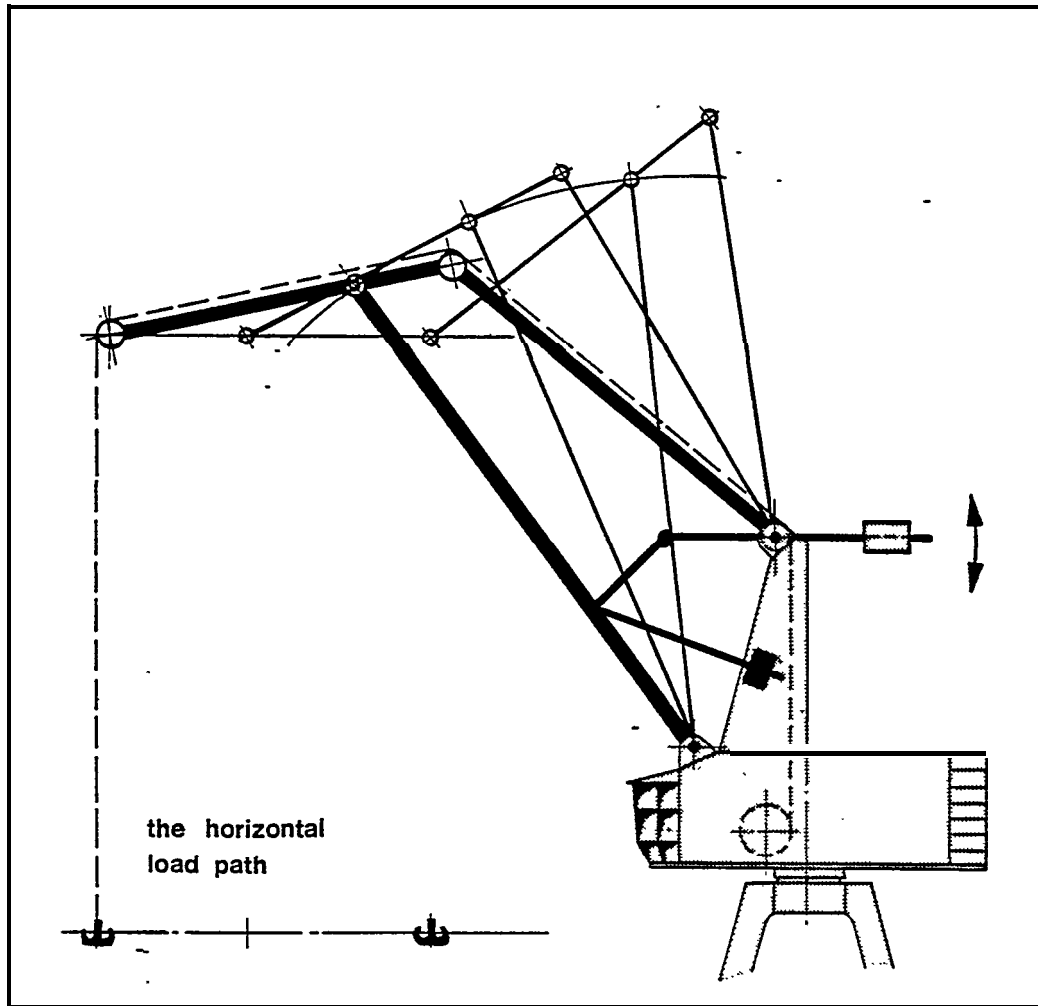


Figure 26: Level Luffing as Achieved by the Gooseneck Crane. (The hook level remains constant, whatever the elevation of the strut boom.)

4.3.2. The Single-Jib Crane with Normal Luffing

Description:

The single jib slew crane consists of a single jib hinged to the main slewing structure of the crane (see Figure 27). The jib is raised and lowered by a simple rope which passes over a pulley at the topmost point of the main structure. The hoist rope uses the 4-fall, simple-reeving system.

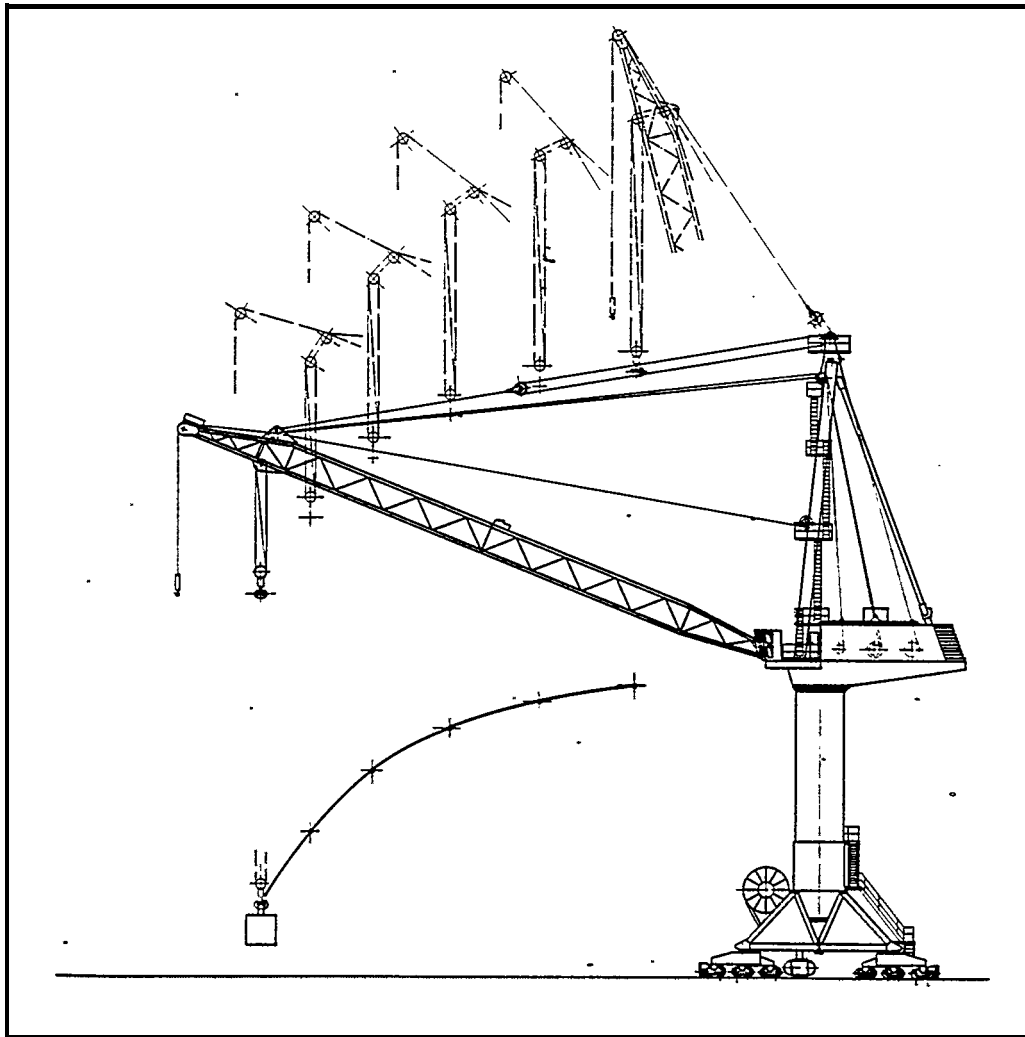


Figure 27: Single-Jib Slew Crane with Simple Hoist Rope: Basic Load Path when Luffing

Operation:

When operating with a simple 4-fall hoist rope (see Figure 28), the single-jib slew crane raises and lowers the load at the same time that the jib is raised and lowered. This imposes considerable additional "work" on the crane and uses additional energy. Using this rope system, level luffing cannot be achieved.

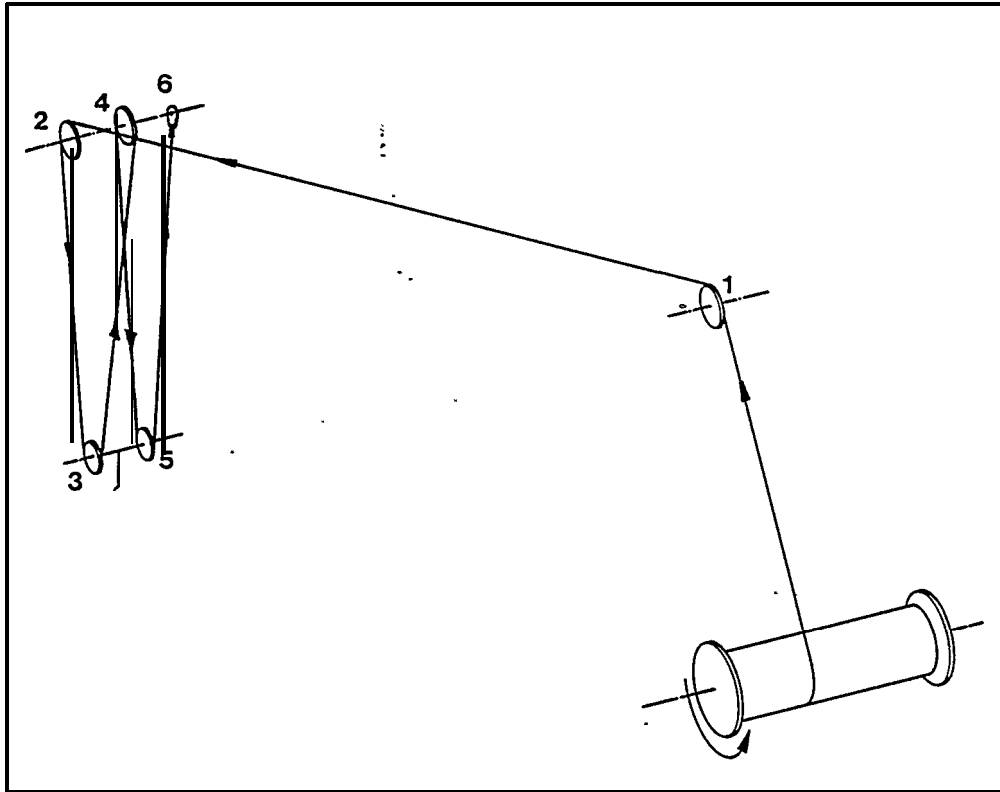


Figure 28: Rope System for a Simple 4-fall Hoist Rope.

4.3.3. Single-Jib Level-Luffing Crane

Description:

This crane is constructed in the same way as a single-jib crane that has normal luffing (see Figure 4.20). The only difference is that a 2-fall, 3-reeving hoist rope system is used.

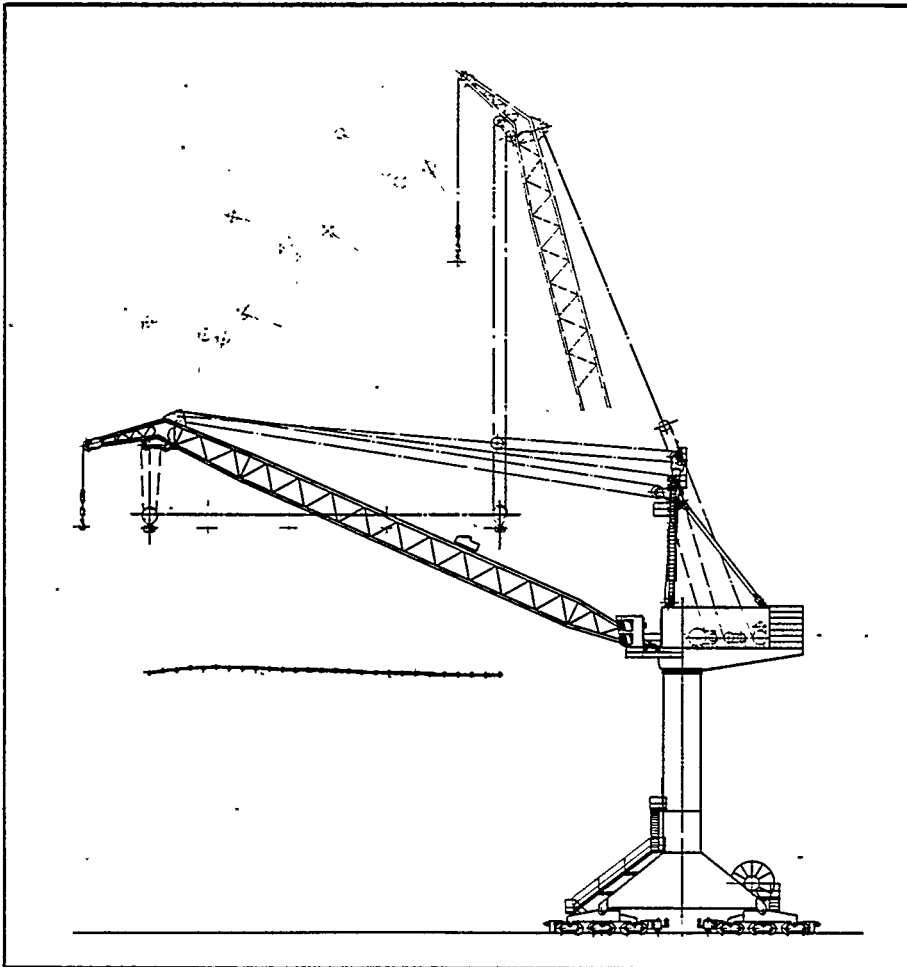


Figure 29: Single-Jib Slew Crane with 3-Reeved Hoist Rope:
Basic Load Path when Luffing

Operation:

The 2-fall, 3-reeving rope system is shown in Figure 30. - In fact, this system acts as a "rope store," taking in and paying out rope as necessary to achieve a close approximation to level luffing.

The outer end of the jib; the hinge of the jib, and the topmost point of the main structure form a triangle. As the jib is raised, the distance between its outer end and the topmost point of the structure is reduced. Rope that has been "stored" between these points is paid out automatically as the jib rises; as the jib is lowered, rope between the hook and the jib is automatically taken up and "stored" again. The overall effect is that the load stays at much the same height whatever the elevation of the jib. This closely approximates the true level luffing of the gooseneck crane.

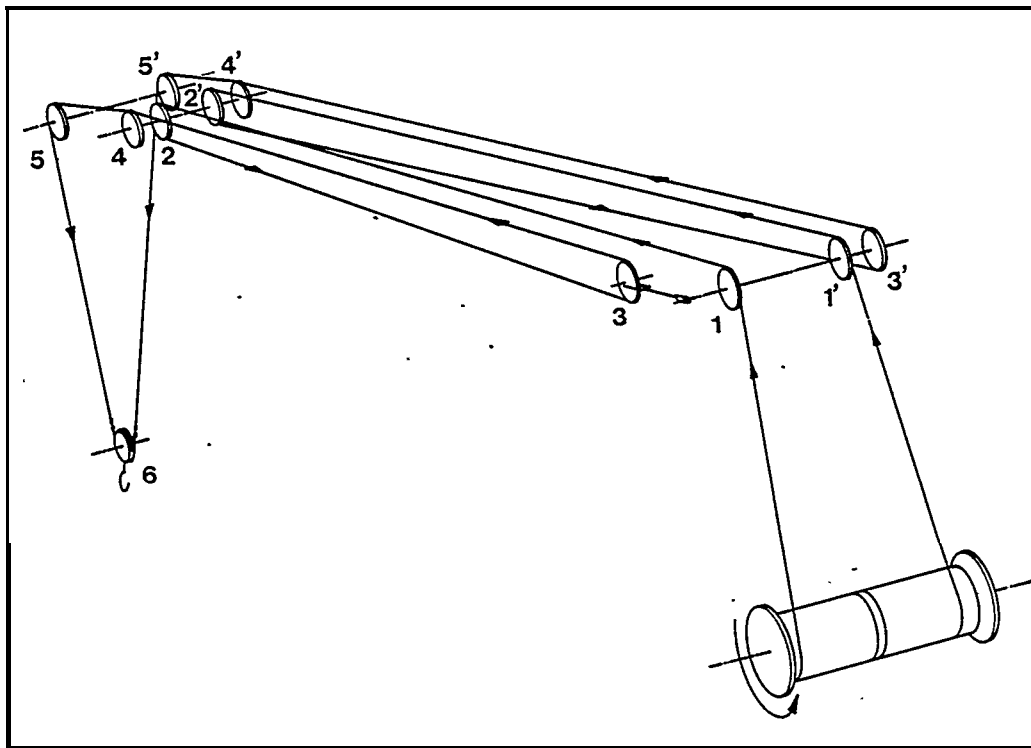


Figure 30: Rope System for a 2-Fall, 3-Reeving Hoist Rope:
Approximates Level Luffing in Use

A further refinement is to introduce a balancing mechanism to compensate for the weight of the boom during luffing. This operates in the same way as the counterweight of the double-boom crane discussed earlier. The advantage of the balanced boom is that luffing requires less power; the disadvantages are the extra weight to be accelerated and decelerated during slewing and traveling operations, the extra moving parts with the attendant maintenance, and the increased wind attack area.

5. COMPARATIVE EVALUATION

The purpose of this section is to present and explain the performance and cost data collected on five cranes. These figures form the basis of the analysis and conclusions offered in Section 6.

5.1. SELECTION OF CRANES

In this study, five cranes have been chosen for comparative evaluation. These are:

- Crane 1: A Double Boom (Goose Neck) Crane
- Crane 2: A Single (Strut) Boom Crane with Rope Luffing (Column Crane with Upper and Lower Bearing)
- Crane 3: A Single Boom Crane with Balanced Boom and Mechanical Luffing (Column Crane with Upper and Lower Bearing)
- Crane 4: A Single Boom Crane with Rope Luffing (with Centerless Roller-Race Bearing)
- Crane 5: Single Crane with Rope Luffing (with turntable and kingpost)

The mechanical differences between these various designs were explained in Section 4 earlier.

These five cranes have been chosen as representing, on a worldwide basis, the most common approaches to the construction of shipyard cranes. Crane 1, the Goose Neck, is the only design that can achieve perfect level luffing and was popular in Europe until the early seventies. Cranes 2 and 3 represent the European/Far Eastern approach to achieving level luffing with cranes of single boom design. The difference between them is that Crane 3 uses a balanced boom which reduces the load when luffing. Crane 4 dispenses with level luffing, which is of dubious cost-effectiveness in shipbuilding, but incorporates the most modern slewing technology. Crane 5 (which also lacks level luffing) illustrates the traditional American approach to crane design. Comparative technical details for each crane are given in Section 5.3. below.

It was decided that cranes in actual operation should be evaluated rather than generic crane-types. This is because, in the first place,

generic data are not always available, and, in the second, because actual performance may not measure up to theoretical levels. On the other hand, by choosing specific operational cranes to study, it is possible that a particularly good or a particularly poor model may distort the figures to some degree. Some allowance for this possible distortion should be made in reading this section of this report, although experience suggests that the expected and actual performance of the cranes in question are reasonably close.

5.2. SELECTION OF EVALUATIVE CRITERIA

5.2.1. Rating

Evaluating several pieces of machinery against each other requires the existence of a standardized system of measurement. For cranes, no internationally accepted standard exists for assessing their capacity. Since, however, (at least outside the United States) most shipbuilding cranes are custom-made, some kind of "sizing" system is clearly necessary for comparison purposes. In Europe, cranes are usually compared by defining their highest possible load moment. The formula is:

$$\text{load moment} = m \times t$$

where (m) is the distance in meters between the load and the crane, and (t) is the maximum load in tons. The use of this figure allows cranes of different types to be compared, while the normal American practice of comparing simply maximum load is only useful in comparing cranes of exactly the same type. To take one example, Crane 1 in the study is, in American terms, a "60-ton crane." The European would look at the load moment: at 33 meters this is 44 tons (33 x 44 = 1452 mt); while at 25 meters the load is 60 tons (25 x 60 = 1500 mt). The maximum load moment is accordingly 1500 mt, and the crane would be considered a "1500 crane."

It is possible to convert the metric value into an (albeit unusual) American value, the ft kip. The conversion formula is:

$$1 \text{ mt} = 10 \text{ KNm} = 7.376 \text{ ft kip}$$

The five-cranes studied are accordingly rated as follows:

	Max. Load	Rating in mt	Rating in ft kip
Crane 1	60-t	1500	11000
Crane 2	100-t	2000	14700
Crane 3	80-t	3300	24300
Crane 4	114-t	2000	14700
Crane 5	90-t	1204-833	8550 -5920..

Table 2: Comparative Rating of Cranes

It is clear at a glance that the European ratings are not directly proportional to the American, since they take into account the efficiency of the crane design as well as the deadweight lifted.

5.2.2. Operational Criteria

Operationally a crane is judged by its speed and by its accuracy. Further, the efficiency of its design can be measured in terms of the energy it consumes in achieving these primary goals. Accordingly, these three standards -- speed, accuracy, and energy consumption -- are used to evaluate each crane.

5.2.3. Investment Criteria

The first factor to be considered here is obviously the actual cost of installing a crane. Training personnel to use the crane is another aspect of start-up costs. Maintenance costs and running costs must also be appraised.

In deciding which crane to buy, absolute primacy must be given to the duties the crane is expected to perform. Although this point may seem obvious, it is not always fully implemented. Surprisingly, " many yards specify a set of cranes simply on the basis of maximum lift, even, though this lift may be performed by only one crane once every few months, or even once every few years. It would be more economical to look at the loads a crane (or set of cranes) must handle three or four times an hour: the lifting of welding sets, containers, cabledrums, and so on. An appropriate mix of cranes would then allow small, fast units to do an economical job most of the time, reserving the clumsy, expensive

unit (or units) for the tasks they do best. Mere size in itself is not a guarantee of efficiency. To take an extreme case: the heaviest unit of its kind in the world is probably the Goliath crane at the old Westinghouse nuclear float plant in Jacksonville. The value of the crane in day-to-day operations was actually negligible.

Another factor that can be overstressed is speed. An analysis of the operations in any given yard may show that its cranes are seldom (or never) used at maximum speed, or that a costly investment in super-fast machinery may in practice save only a few man-hours a day. A case in point is the Nassco yard in San Diego. Here cranes are used to transport part-fabricated units over distances of several hundred yards using a remarkable, though far from new, railroad system. In updating this -system, the hoist-speed of the new cranes would be of little importance; prime emphasis would have to be given to installing a rugged truck system.

The decisive figure in initial crane investment is the estimated job-hour-cost. Based on this figure, an appropriate duty-mix can be developed. To give an example: analysis may show that medium-sized cranes can accomplish nearly all the tasks in a yard, leaving only occasional very heavy loads that can, in fact, be twin-lifted. Although twin-lifting takes extra time, the yard may decide on two medium cranes rather than one large one since in normal operation it will then have two nifty cranes in action instead of one clumsy one, with a considerable saving" of time. This saving may quickly pay back the cost of investing in two cranes rather than in one, especially when such factors as the installation of heavy track and energy costs are also taken into account.

As to maintenance costs, the primary factors here are smoothness and shock-free operation. State-of-the-art hydraulic systems allow shock-free start-up and stop in both slewing and hoisting operation. Old-fashioned slewing technology puts massive strain on the steel structure of the crane, especially on the tower, and could lead to premature metal fatigue; old-fashioned hoisting technology will contribute to -structural strain, but, more important, it will lead to rapid and excessive wear and tear on all the hoist gear.

5.3. THE FIVE CRANES IN DETAIL

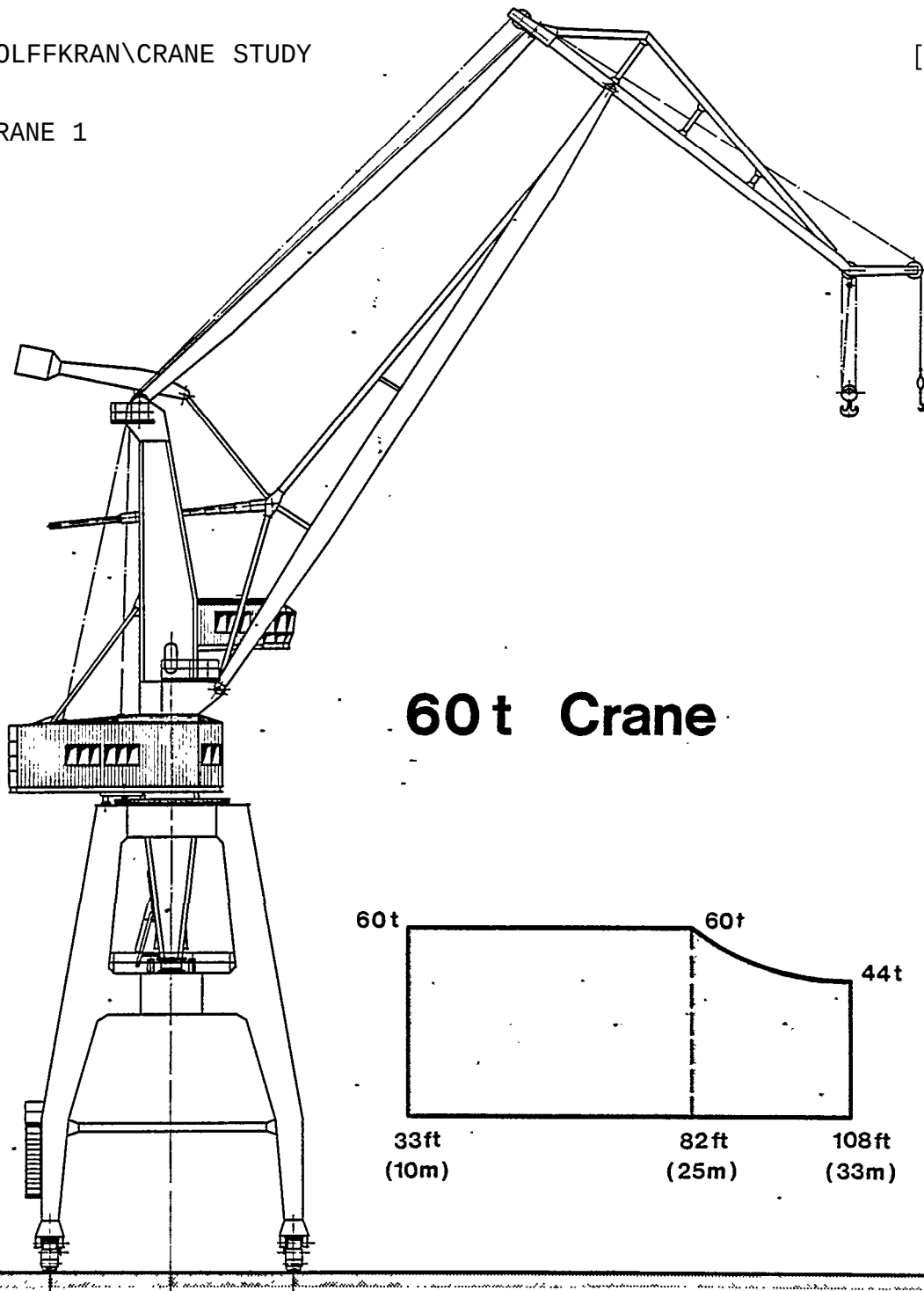
In this section, the vital statistics of each crane are given, followed in each case by a scale drawing.

5.3.1. Crane 1

Type	Double Boom (Goose Neck) Level Luffing Crane
Year	1 9 5 8
Location	Howald-Deutsche Werft (HD W), Hamburg
Built by	MAN-Nuremberg
Max. Load	133 000 lbs at 82 ft (60 tons at 25 meters)
Rating	1500 mt (11 000 ft kip)
Max. Radius	108 ft with 97 000 lbs (33 meters with 44 tons)
Min. Radius	33 ft (10 meters)
Whip Hoist	11 000 lbs at 120 ft (5 tons at 36.5 meters)
Rail Gauge	39 ft (12 meters)
Drives	Traditional electro-mechanical slip ring motors with resistor control and multiple reduction gear boxef
Design Principle	Vertical slew column in box-girder" structure supported by a portal also of box-girder structure. All box-girder parts of mild steel plates, fully welded.

Remark: This crane is one of the biggest double-boom (gooseneck) cranes built for ship yard duty in Europe. Two units of this type are still on site today.

CRANE 1



crane 1	Type	Double Boom (Goose Neck) Level Luffing Crane
	Year	1958
	Location	Howald-Deutsche Werft (HDW), Hamburg
	Built by	MAN-Nuremberg
	Max. Load	133 000 lbs at 82 ft (60 tons at 25 meters)
	Rating	1500 mt (11 000 ft kip)
	Max. Radius	108 ft with 97 000 lbs (33 meters with 44 tons)
	Min. Radius	33 ft (10 meters)
	Whip Hoist	11 000 lbs at 120 ft (5 tons at 36.5 meters)
	Rail Gauge	39 ft (12 meters)

5.3.2. Crane 2

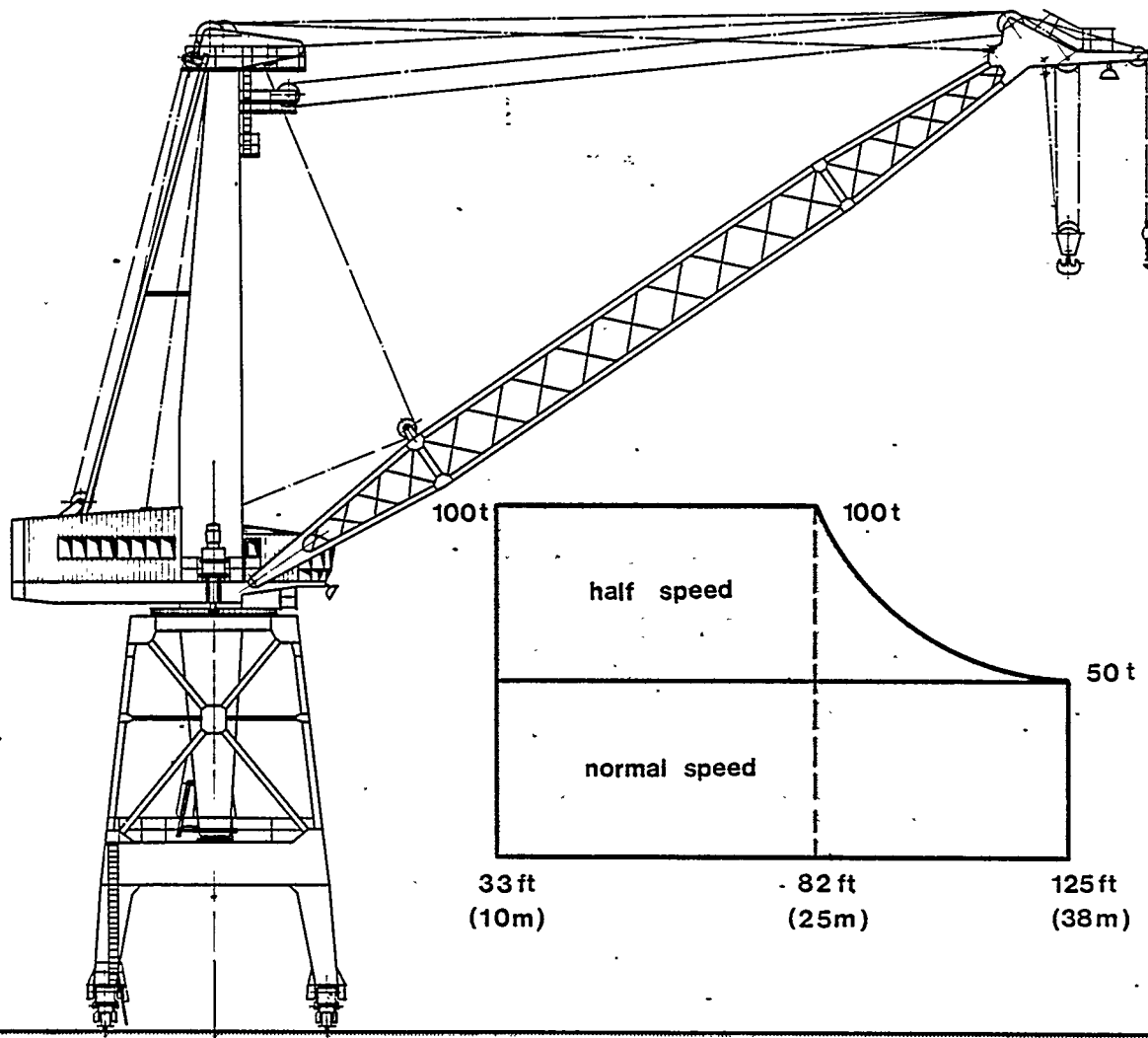
Type	Single (Strut-) Boom Crane with Level (Rope-store) Luffing
Year	1 9 5 9
Location	Ottenser Eisenwerke, Drydock Elbe XVII (Now near Athens, Greece)
Built by	MAN-Nuremberg
Max. Load	220 000 lbs at 82 ft" (100 tons at 25 meters)
Rating	2000 mt (14 700 ft kip)
Max. Radius	125 ft with 110 000 lbs (38 meters with 50 tons)
Min. Radius	33 ft (10 meters)
Whip Hoist	11 000 lbs at 136 ft (5 tons at 41.5 meters)
Rail Gauge	33 ft (10 meters)
Drives	Traditional electro-mechanical slipring motors with resistor control and multiple reduction gear boxes
Design Principle	Vertical slew column in box-girder structure supported by a portal also of box-girder structure. Boom of lattice design; remainder of box-girder. parts of mild steel plates, fully welded.

Remark: This was the first modern single boom crane with (quasi-) level luffing (achieved by the rope storage system) built in Europe for shipyard duty.

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CRANE 2

100 t Crane



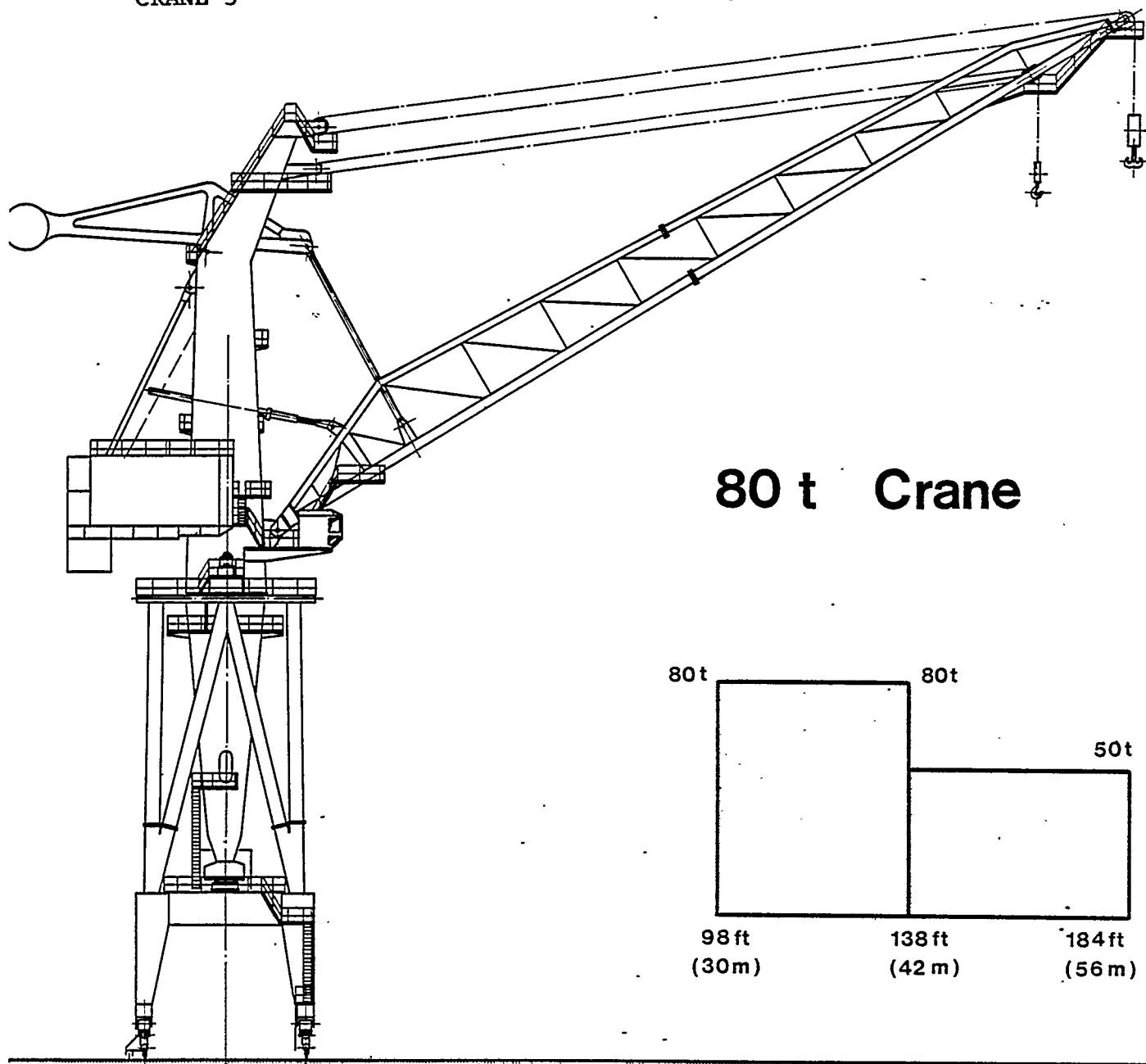
Crane 2	Type	Single Boom Crane with Level (Rope-store) Luffing
	Year	1959
	Location	Ottenser Eisenwerke, Drydock Elbe XVII
	Built by	MAN-Nuremberg
	Max. Load	220 000 lbs at 82 ft (100 tons at 25 meters)
	Rating	2000 mt (14 700 ft kip)
	Max. Radius	125 ft with 110 000 lbs (38 meters with 50 tons)
	Min. Radius	33 ft (10 meters)
	Whip Hoist	11 000 lbs at 136 ft (5 tons at 41.5 meters)
	Rail Gauge	33 ft (10 meters)

5.3.3. Crane 3

Type	Single Boom Crane with Balanced Boom and Mechanical Luffing .
Year	1980
Location	Keppel Tuas Shipyard, Singapore
Built by	Mitsui Engineering, Japan
Max. Load	176 000 lbs at 138 ft (80 tons at 42 meters)
Rating	3300 mt (24 300 ft kip)
Max. Radius	184 ft with 110 000 lbs (56 meters with 50 tons)
Min. Radius	87 ft (27 meters)
Whip Hoist	33 000 lbs at 164 ft (15 tons at 50 meters)
Rail Gauge	33 ft (10 meters)
Drives	Traditional electro-mechanical slipring motors with resistor control and multiple reduction gear boxes
Design Principle	Vertical slew column in box-girder structure supported by a portal also of box-girder structure. Boom of lattice design; remainder of pipes or box-girder parts of mild steel plates, fully welded.

Remark: Many cranes of this type have been constructed. The crane studied here is one of the largest. Four identical units are at present installed in the Keppel Tuas yard in Singapore. The crane achieves (quasi-) level luffing by means of a rope-store. The balanced boom assists in the performance of luffing operations. (The design is originally European.)

CRANE 3



Crane 3	Type	Single (Balanced) Boom Crane with Level Luffing
	Year	1980
	Location	Keppel Tuas Shipyard, Singapore
	Built by	Mitsui Engineering, Japan.
	Max. Load	176 000 lbs at 138 ft (80 tons at 42 meters)
	Rating	3300 mt (24 300 ft kip)
	Max. Radius	184 ft with 110 000 lbs (56 meters with 50 tons)
	Min. Radius	87 ft (27 meters)
	Whip Hoist	33 000 lbs at 164 ft (15 tons at 50 meters)
	Rail Gauge	33 ft (10 meters)

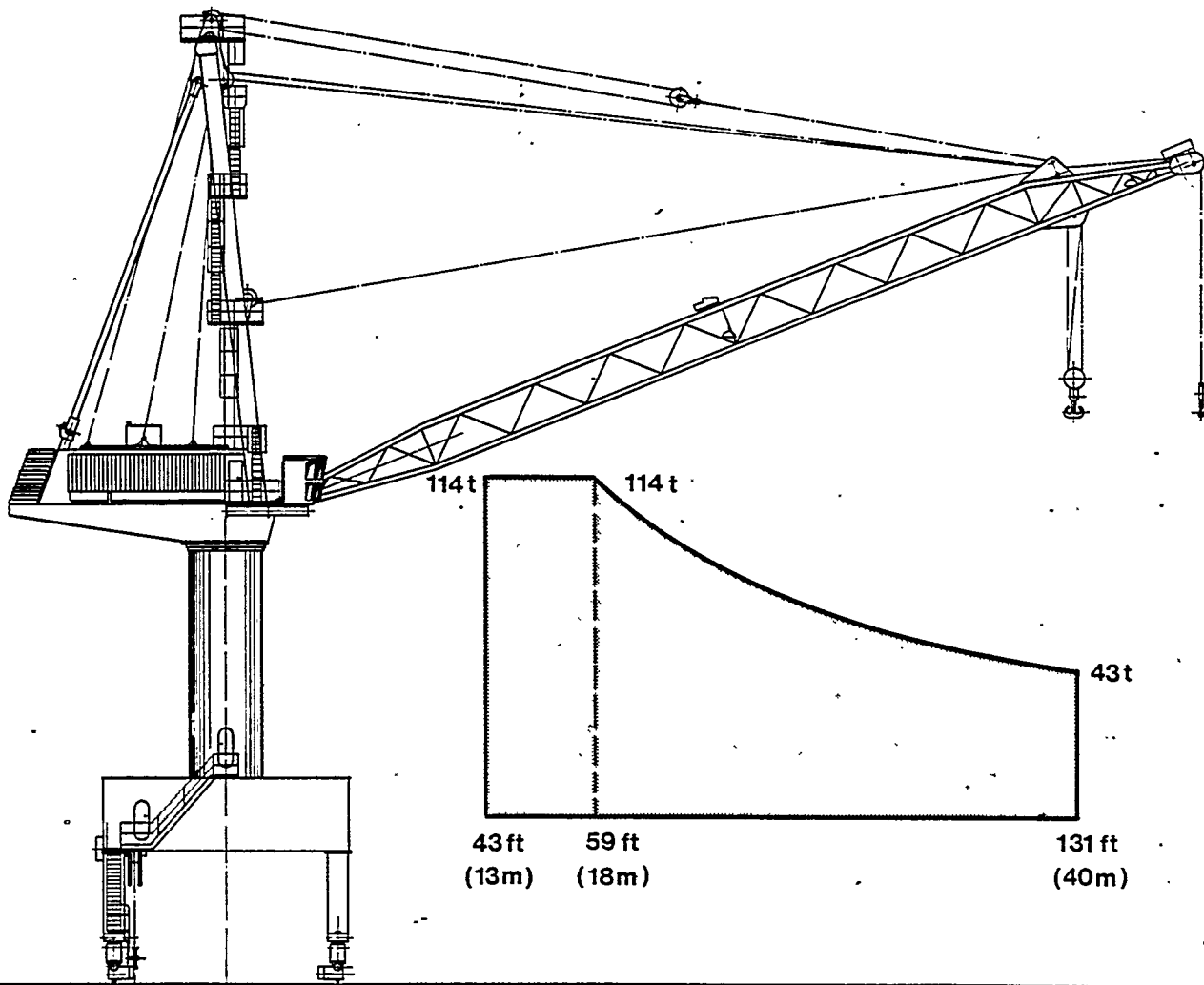
5.3.4. Crane 4

T y p e	Single boom crane with rope luffing; stationary pipe column and centerless roller-race bearing construction
Year	1 9 8 1
Location	Avondale Shipyard, New Orleans
Built by	MAN-Wolffkran
Max. Load	251 000 lbs at 60 ft (114 tons at 18.3 meters)
Rating	2000 mt (14 700 ft kip)
Max. Radius	131 ft with 95 000 lbs (40 meters with 43 tons)
Min. Radius	43 ft (13 meters)
Whip Hoist	40 000 lbs at 151 ft (18 tons at 46 meters)
Rail Gauge	35 ft (10.7 meters)
Drives	All drives are electro-hydraulic, direct driven.' The crane has no mechanical reduction gears.
Design Principle	Stationary pipe column on box-girder portal beam. Boom of pipe lattice construction. Main structure is plate manufactured or rolled section assembled, fully welded.

Remarks: The design incorporates the advantages of the modern centerless roller-race bearing. The pipe column does not move during operation. The crane dispenses with level luffing. since, for shipbuilding applications, the additional cost of installing a rope store does not seem to bring comparable benefits. The use of an electro-hydraulic drive system is particularly valuable in achieving smooth, accurate operation. AU safety and operational aspects are electronically controlled.

CRANE 4

114 t Crane



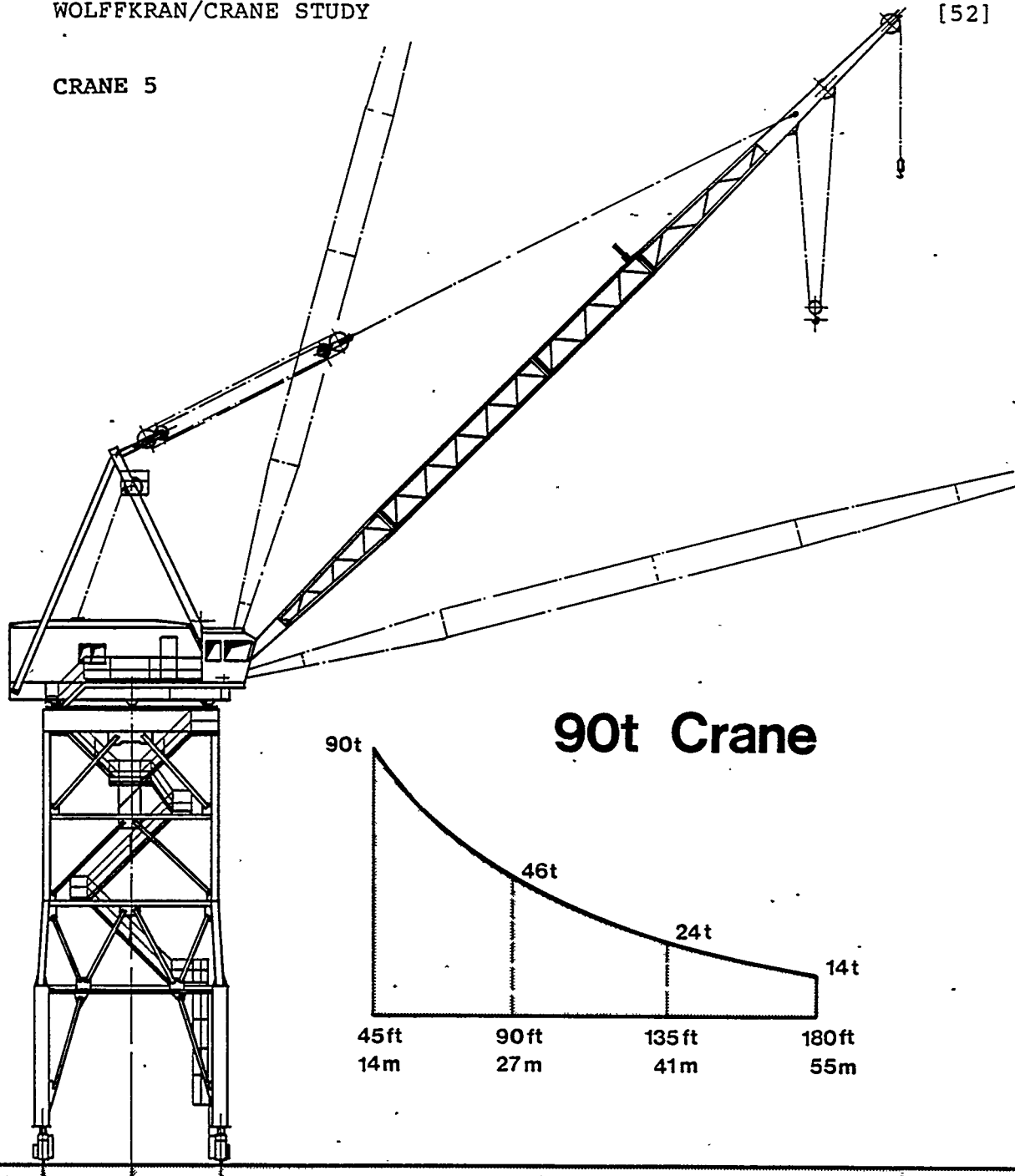
Crane 4	Type	Single Boom Crane with Rope Luffing
	Year	1981
	Location	Avondale Shipyard, New Orleans
	Built by	MAN-Wolffkran
	Max. Load	251 000 lbs at 60 ft (114 tons at 18.3 meters)
	Rating	2000 mt (14 700 ft kip)
	Max. Radius	131 ft with 95 000 lbs (40 meters with 43 tons)
	Min. Radius	43 ft (13 meters)
	Whip Hoist	40 000 lbs at 151 ft (18 tons at 46 meters)
	Rail Gauge	35 ft (10.7 meters)

5.3.5. Crane 5

Type	Single-boom crane with rope luffing, turntable and kingpost. slewing
Year	1971
Location	Nassco Shipyard, San Diego, California
Built by	Clyde-USA (Model 28 E)
Max. Load	190 000 lbs at 45 ft, with 16-part line (86 tons at 14 meters)
Rating''	max. 1204 mt (8550 ft kip) min. 833 mt (5920 ft kip)
Max. Radius	160 ft with 37 000 lbs (49 meters with 17 tons)
Min. Radius	45 ft (14 meters)
Whip Hoist	50 000 lbs at 175 ft, with 4-part line (23 tons at 58 meters)
Rail Gauge	40 ft (12.2 meters)
Drives	Diesel-electric prime mover; dc drives; pneumatic control
Design Principle	Table-top support structure of lattice design with turntable-kingpost slewing. A large,. heavy machine housing mounted on the slewing part of the crane. Boom of angle-lattice design, welded.

Remarks: Cranes of this type are made by a number of American manufacturers in either electro-hydraulic, diesel-hydraulic, or di-electric versions. They are standard equipment in the U. S., and in areas influenced by U.S. traditions. Many are now at the end of their useful life, and a fair number are second-hand units.

CRANE 5



Crane 5	Type	Single Boom Crane: Rope Luffing, Turntable Slew
	Year	1971
	Location	Nassco Shipyard, San Diego, California
	Built by	Clyde-USA (Model 28E)
	Max. Load Rating	190 000 lbs at 45 ft (86 tons at 14 meters)
		max. 1204 mt (8550 ft kip)
		min. 833 mt (5920 ft kip)
	Max. Radius	160 ft with 37 000 lbs (49 meters with 70 tons)
	Min. Radius	45 ft (14 meters)
	Whip Hoist	50 000 lbs at 175 ft (23 tons at 58 meters)
	Rail Gauge	40 ft (12.2 meters)

5.4. OPERATIONAL EVALUATION

5.4.1. Speed

The speed of a crane can be measured during the performance of four distinct operations: hoisting, luffing, slewing and traveling. Table 3 shows the measured performance of the five cranes.

	CRANE 1 (60-t)	CRANE 2 (110-t)	CRANE 3 (80-t)	CRANE 4 (114.-t)	CRANE 5 (90-t)
HOISTING in ft/min					
Speed with max. load	26	11	26	10	26
Max. speed	26	25	53	40	26
Load at which max. speed achieved	max	50-t	40-t	25-t	max
LUFFING in ft/min					
With full load	98	33	66	12	12
Without load	98	33	66	23	12
SLEWING in rpm					
At max. radius	0.75	0.7	0.3	0.5	0.5
At min. radius	0.75	0.7	0.3	1.5	0.5
TRAVELING in ft/min					
	98	98	98	131	125

Table 3: Comparative Evaluation by SPEED

Hoisting

The difference in hoist speeds is more apparent than real. Any of these cranes equipped with the appropriate drive unit and rope system could hoist at any sensible speed. In fact, high-speed hoisting is rarely used with heavy loads. For very light loads, the whip hoist is always available. The main point here is that

achieving high speeds with heavy loads is a disproportionately expensive business: not only do more expensive prime movers have to be installed, but the increased dynamic stresses have to be allowed for in the structure itself.

Luffing

The double-boom (Crane 1) and balanced boom (Crane 3) allow considerably higher luff speeds than the traditional rope-pull system. Over the course of a work-day or shift, the very high luffing speed of the double-boom crane would offer a significant advantage in the number of heavy loads handled.

S l e w i n g

Slew speeds depend on two factors: <a> the structural mass to be accelerated and decelerated; the outreach of the crane and the consequent circumferential load speeds – these could, if extreme, exert considerable tangential stress on the crane structure. Because of the heavy mass of the slewing part of Crane 1 and Crane 3, they are at some theoretical disadvantage when it comes to slewing; this can be overcome only by the installation of powerful and expensive prime movers. The relatively high speed achieved at considerable cost by Crane 1 (0.75 rpm) may not, however, be of much advantage: a 44-ton load at 108 ft radius will be traveling at about 6 mph, which is fairly fast even on a bicycle. The apparently slower speed achieved by Crane 3 (0.3 rpm) will move a 50-ton load at a radius of 184 ft at a speed of some 4 mph, which is quite fast enough.

Speed is not the essence here, but smoothness. The stepless, shockfree acceleration of a modern slew crane (hydraulic or DC) allows much more efficient handling of the load. The old saying is appropriate here: More haste, less speed.

Traveling

The lighter the crane, the better its traveling performance will be. Again, high speed as such is not normally required – an average walking speed of 130 ft/min (or about 1.5 mph) is actually ideal in shipbuilding practice. It is rapid acceleration and deceleration that are most significant in assessing the traveling efficiency of a crane; here lighter mass is of the greatest importance.

The importance of correctly installed track was stressed in Section 4, but it should be repeated here. A crane can travel only as fast as its track allows.

5.4.2. Accuracy

The double-boom or gooseneck crane has a significant advantage here. It can achieve the shortest theoretically possible horizontal load path. Because it has the shortest unguided rope-length between the crane and the load, it allows the least possible free sway. This permits quick, accurate placing of the load. A relatively low crane structure is also able to reach over the masts or other superstructure of a vessel. Figure 31 makes this advantage clear.

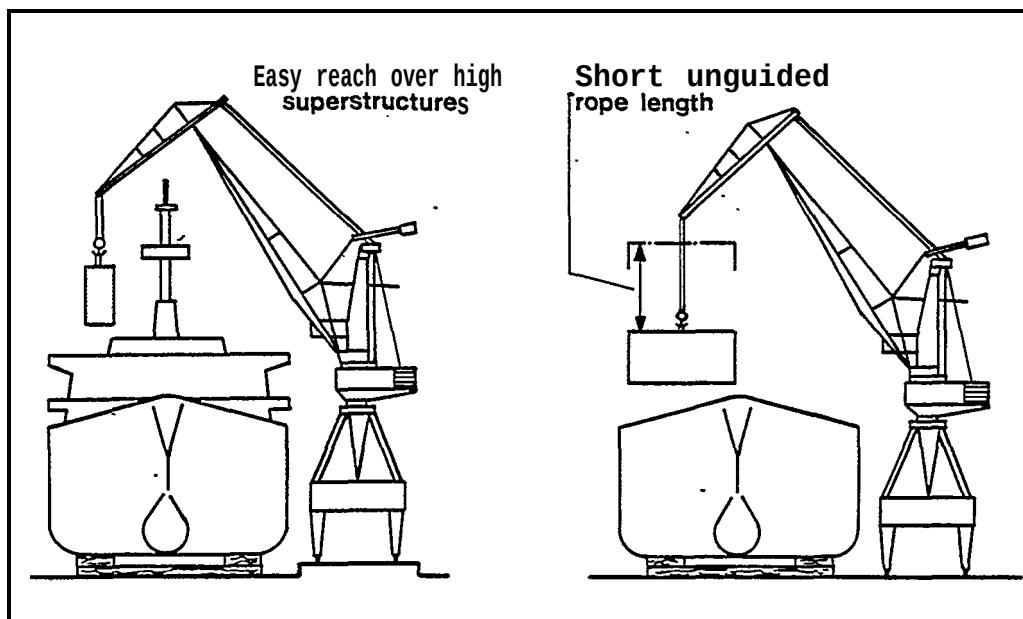


Figure 31: Accuracy Advantage of the Double Boom Crane

With all the other four cranes, accuracy is a matter of three factors: <a> Operator training; the smoothness of the drive system; and <c> the rigidity of the structure, especially the boom. In achieving smooth drive, the use of hydraulic or DC systems (as installed in Crane 4) is of the utmost importance. The standard slipringmotor / AC drive simply cannot achieve the requisite degree of smoothness. As to the rigidity of the structure itself, a plate-fabricated or a single-pipe boom is more likely to bend in operation than a lattice boom, with an obvious deleterious effect on accuracy. Most rigid, and therefore most accurate, is the pipe-lattice boom. The pipe-lattice boom (as installed in crane 4) is not merely the most rigid: it has, in fact, the best weight/stiffness ratio. Further, the resulting lighter boom offers a smaller wind-attack area, an added advantage of this type of construction.

Beyond question, the superior accuracy of the double-boom crane is important for fast cargo-handling in a seaport. Whether this accuracy is cost-effective in shipbuilding applications is, however, open to doubt.

5.4.3. Energy Consumption of Total Installed Capacity

To calculate energy use, a load of 0.75 of maximum has been assumed in each case. Drive units included are: main hoist, whip hoist, luffing drive, slewing drive, travel drive (with min. 50% of wheels driven). An energy "surcharge" of 15% has then been added to take into account lighting, air-conditioning, signals, power-outlets and other auxiliary uses of energy. The results are given in Table 4 below.

	CRANE: 1 (60-t)	CRANE 2 (110-t)	CRANE 3 (80-t)	CRANE 4 (114-t)	CRANE 5 (90-t)
HOI STING in kw					
Main hoist	108	96	132	73	152
Whip hoist	30	30	55	37	*
LUFF ING in kw					
	30	60	55	75	76
SLEWING in kw					
Units	2	2	2	2	1
Total	60	74	44	65	57
TRAVELING in kw					
Units	4	2	8	8	4
Total	100	100	88	90	76

* Same motor used for main and whip hoist

Table 4: Comparative Evaluation by ENERGY CONSUMPTION

The most obvious feature of this table is the energy advantage – about 50% – enjoyed by the double-boom crane- (Crane 1) during luffing operations. This advantage would become significant, however, only when a large number of heavy loads must be moved in such a way that luffing movements occur very frequently. This is not the case in shipbuilding operations.

In fact, no single crane in this study appears to enjoy any inherently significant advantage in energy costs.

5.5. INVESTMENT EVALUATION

5.5.1. Cost of Installation

The cost of a crane derives from two sources: <a> the structure itself, and the machinery and electrical systems. In examining Cranes 1, 2, and 3 (the traditional European cranes), it is clear that the machinery and electrical systems do not significantly differ, and that the costs of cranes of these types will vary, in essence, because of the amount of steel in their structures and the amount of work required to build them. First, then, the weight of the materials used in these structures. This has been calculated in Table 5.

	CRANE 1 (60-t)	CRANE 2 (110-t)	CRANE 3 (80-t)	CRANE 4 (114-t)	CRANE 5 (90-t)
	(All weights in tons)				
Boom or boom system	64	31	65	19	25
Slewing part without boom	74	102	190	69	53
Portal	148	137	209	106	103
Machinery	61	73	63	46	77
TOTAL CRANE LESS BALLASTS	357	353	537	250	258
Slewing Ballast	28	105	263	130	143
Center Ballast	113	139	50	250	
TOTAL OPERATING WEIGHT	498	597	850	400	401

Table 5: Comparative Evaluation by WEIGHT

It is obviously impossible to compare the exact cost-to-build of various crane types, but nevertheless a rule-of-thumb method exists which allows the buyer to calculate a rough "cost-to-build" multiplier for each type of crane. The first step is to equivalence the five types of crane studied, so that each has the same nominal load moment. This calculation assumes that to get the same load moment, an exactly proportional amount of steel would be required. Taking Crane 2, the 100-ton, 2000 mt load moment crane as standard, the following load moment equivalency factors emerge:

	Weight	Rating in mt	Equivalency Factor	Equivalenced Weights
Crane 1	357 t	1500	1.3	464 t
Crane 2	353 t	2000	1.0	353 t
Crane 3	537 t	3300	0.6	322 t
Crane 4	250 t	2000	1.0	250 t
Crane 5	258 t	1200	1.6	412 t

Table 6: Rating Equivalency of Cranes

In other words, to achieve the same load moment as Crane 2, a crane of type 1 would take (roughly) 464 tons of steel and other parts.

The amount of steel in the crane cannot simply be multiplied by a standard figure for labor to find its cost. This is because the work involved in constricting each type of crane is different, and because some cranes require large numbers of moving parts that are expensive to buy and to install. The key figure here is the amount of labor that has to be invested for each ton of steel worked. For each type of crane the pluses and minuses are as follows:

- Crane 1 The extreme weight of this crane requires disproportionately heavy and expensive bearings. The installation of, the many joints required by the gooseneck requires a large amount of drilling and bolting as well as extreme precision in assembly.
- Crane 2 This crane design is standard. It has no unusual features that require special techniques, and is built of basic, inexpensive materials.
- Crane 3 This crane is. expensive to build. It has many moving parts requiring exactly manufactured, heavy bearings. The central column has a "knick" at the top which is costly to make.
- Crane 4 This crane uses expensive steel pipe for its jib, which adds to its cost; however, the lighter construction of the jib means that less welding is required. The centerless roller bearing is an expensive feature.
- Crane 5 The use of angle iron in this crane requires a large amount of welding, but in general manufacturing costs are about the same as for Crane 2.

If these various advantages and problems are netted, a rough "labor cost per ton of steel" factor emerges for each crane. Taking Crane 2 again as the standard ("labor" cost factor = 1), the various ratings are given in Table 7. In the same table, the total weight of steel is multiplied by this "labor" factor, giving a figure for cost-to-build. This end figure, it must be stressed, expresses only a relationship – it is not attached to a unit of any kind.

	Relative "Labor" Cost	Equivalence Weight of Steel	Relative cost-to- Build
Crane 1	1.2	464 t	557
Crane 2	1	353 t	353
Crane 3	1.4	322 t	450
Crane 4	1.3	250 t	325
Crane 5	1.0	412 t	412

Table 7: 'Labor" Cost per Ton of Steel and Relative Cost-to-Build

If Crane 2 is once again taken as the industry standard (relative cost-to-build = 353), then an extremely useful multiplier now emerges: the . cost-to-build multiplier. This multiplier is calculated simply by dividing the various cost-to-build figures by the cost-to-build of crane 2. Table 8 shows the resulting figures.

CRANE TYPE	COST-TO-BUILD MULTIPLIER
Crane 1: Double Boom (Goose neck)	x 1.5
Crane 2: Single Boom, Level Luffing	x 1.0
Crane 3: Balanced Boom, Level Luffing	x 1.3
Crane 4: Single Boom, Normal Luffing (European type)	x 0.9
Crane 5: Single Boom, Normal Luffing (U.S. type)	x 1.2

Table 8: Cost-to-Build Multipliers (The figures assume identical specification)

To take an example, a shipyard specifies a single-boom, level luffing crane (a type-2 crane); the manufacturer quotes a price of \$1 million. Given the same capacity and performance, the other types of crane would then cost \$1 m x the appropriate multiplier: a double-boom crane would cost \$1.5 m; a balanced single-boom crane \$1.3 m; a single boom crane with normal luffing \$0.9 m; and an American-type turntable crane \$1.2 m.

Useful as this rule-of-thumb figure is, it must be stressed that these are manufacturing cost figures, and that they bear no direct relationship to the price of a crane in the market-place. Clearly price depends on such things as exchange rates, the buyer's negotiating skill and the seller's keenness to win the order. For the foreseeable future, there should be a buyer's market for cranes.

Another important consideration arising from weight is the wheel pressure exerted on the track system: the greater the pressure, the more expensive the track. For purposes of comparison, it has first been assumed in Table 9 that each of the five cranes would run on 10 wheels per corner; figures for the actual number of wheels and resulting weights are given in the lower part of the table.

	CRANE 1 (60-t)	CRANE 2 (110-t)	CRANE 3 (80-t)	CRANE 4 (114-t)	CRANE 5 (90-t)
Speed (ft/min)	98	98	98	131	125
Max. corner pressure	209-t	294-t	380-t	204-t	160-t
Design wheel-load (10 wheels per corner)	21-t	30-t	38-t	21-t	16-t
Actual wheels per corner	8	10	10	6	4
Actual wheel-load (max.)	26-t	30-t	38-t	34-t	40-t
Width of railhead (in inches)	3	3	4	4	4
Gauge (actual) (inft)	39	33	33	35	40
Wheel diameter (in inches)	25	25	25	25	24

Table 9: Comparative Evaluation by WHEEL LOADINGS

The double-boom crane (Crane 1) and the turntable brane (Crane 5) enjoy a certain advantage here in actual wheel load; this is because of the very wide gauges they use. The principle is that the wider the gauge, the less leverage the load will be able to exert on the track. This advantage is not, however, of any great significance; in general, the deadweight of the crane can be taken as directly proportional to the wheel load. This means that an increase in weight requires either distribution through a larger number of wheels, or an improved track – both expensive items. Further heavier deadweights naturally require more powerful drives and brakes. This is an important cost consideration when existing track is to be used for a new, and probably more powerful, crane.

5.5.2. Maintenance

Maintenance costs for drive units and other machinery, for electric circuitry; and for the traveling system will be roughly equal for each crane. Most yards do not keep records of oil changes and so on, but all report regular changes of oil in gearboxes on a 6-12 month basis.

Areas of special concern for maintenance teams include:

- a. The use of existing, but inadequate track for new, heavy cranes.
- b. The effect of outside shot-blasting residue on all moving parts.
- c. The wear caused by improperly adjusted king-posts and roller trains.
- d. Wear on slew gear-pinions in older cranes.
- e. Sticking link-systems in older double-boom cranes.
- f. The cutting or damaging of mains electricity supply cable during normal operation.

Since, however, these problems (apart from <e>) are common to all types, they are of no significance in assessing investment costs.

Another procedure common to all yards was the inspection and maintenance of the crane structure itself. In all cases the structure was examined for defective members on a regular (6-12 month) basis. Not a single case of damage from metal fatigue was reported for any type of crane. Disappointingly, only 2 yards, a naval support yard in Germany and one in Malaysia, carried out the corrosion preventive shot blasting and repainting recommended by the manufacturers. The general argument was that the machinery and circuitry wear out in a 20-25 year span, requiring the replacement

of the crane; the structure itself is still good at that time, even with its first coat of paint.

The only area in which a maintenance cost differential arises is from the number of regular maintenance points on each crane. These are: <a> links, hinge-points, and bearing points; pulleys.

Table 10 gives the number of such points on each crane.

	CRANE 1 (60-t)	CRANE 2 (110-t)	CRANE 3 (80-t)	CRANE 4 (114-t)	CRANE 5 (90-t)
Links and hinges	9	3	8	2	2
Pulley stations	4	7	4	6	6
TOTAL	13	10	12	8	8

Table 10: Comparative Evaluation by MAINTENANCE POINTS

In general, the double-boom crane, because of its many links and hinges, will be more costly to maintain than a single boom. The balanced single boom also, however, requires a considerable number of linkages and will also be relatively expensive to maintain.

The only parts of a crane requiring regular replacement are the hoist ropes. No 'exact figures for replacement periods have been obtained, but the general trend is clear: hoist ropes in double-boom cranes last significantly longer than hoist-ropes in single-boom cranes with rope-store level luffing. Normal figures seem to be 2-3 years for double-boom cranes and 2 years for single booms. The exception to this rule is that when a heavy-load, double-boom crane has a multiple hoist tackle of 12 or 16 falls, the rope seems to have about the same life as that of a single-boom crane. The reason for this is that the (otherwise advantageous) short free-swing distance between boom end and lower block, requires constant, multiple bending of the rope.

A final word of warning: crane owners around the world all identify one hoist-rope "eater": the welding of inadequately grounded parts to hull structures. The hoist-rope then becomes the ground - this can shorten its life to a meager 3 or 4 months.

Cash figures for crane maintenance are not available. Crane drivers themselves normally carry out regular daily greasing. "When-needed" maintenance is performed by teams that report, normally, to Plant Engineering. The cost of these personnel and of the repairs is thus allocated to general overhead, either directly or via Plant Engineering. Specific crane records are not kept. However, a rule of thumb emerged from discussions with maintenance

experts: during the first ten years, a busy yard will spend 3-5% of the initial purchase cost on maintenance. In slack times this will drop to 1.5-3%. After ten years, maintenance costs should be taken as percentages of replacement cost.

The question of downtime' was also addressed at each yard. Again, no record-keeping had taken place, and comment is, therefore, somewhat impressionistic: In general, no yard can afford downtime on a crane, and so good stocks of all "wear-and-tear" parts were maintained. The only common cause of serious downtime appeared to be during the running-in of a new crane when the manufacturer had to make adjustments or solve teething troubles. In general, yards feel that a good crane should stand up to weeks or even months of hard, continuous use during peak times without needing any maintenance. The only case reported of a major crane breakdown was with a 10-year-old gooseneck: the double-boom had stuck due to a lack of grease and took two weeks to repair.

The downtime required for a rope-change was generally reported as 20-26 man-hours. A three-man team could thus change rope during a single shift. In fact, most yards thought that their cranes were in operation during only about 70% of the work-year (2000 hours per year on a 40-hour single shift; 4000 hours on double shift). This clearly allows adequate time for preventive maintenance during non-operational periods.

5.5.3. Safety of Equipment and Operator

In specifying a crane, no short cuts should be taken in requiring a full battery of safety devices. In the view of the writer, all cranes, regardless of minimum local or national safety requirements, should have the following safety features:

5.5.3.1. Load Moment Control

This control prevents the crane luffing outwards beyond safe limits, or lifting too heavy a load for a given radius. When invoked, this control should not inactivate the crane completely, but must permit the appropriate corrective action, e.g. , luffing inwards or lowering the load.

Load Moment Control may be mechanical or electronic; it calculates the product of actual load and radius, and automatically prevents incorrect movements.

5.5.3.2. Load Control

This control may be independent, or an integrated part of the load moment control. Load data is evaluated to prevent the lifting of an excessive load.

There are many ways of collecting load data: electronic load cells may be mounted at the free end of the hoist line, or an eccentric pulley may be mounted in the line system, to mention just two alternatives.

5.5.3.3. Emergency-Out

In emergency situations, activation of a red push-button within easy reach of the operator instantly shuts down all crane movements. Two further emergency-out stations should be available on the ground.

Restarting should only be possible when all control levers have been returned to the zero position.

5.5.3.4. Dead-Man Control

The dead-man control should be installed only in cranes run by remote control with a portable control system. The principle is familiar from railroad engines. If the operator removes his hand from the dead-man control handle, the crane is automatically shut down. There is usually a time-lapse relay allowing a few seconds grace in case of accidental release.

If the crane is directly operated by the driver, the spring-controlled, zero-action levers must return to zero whenever released by the operator.

5.5.3.5. Anti Two-Blocking Device

At safety congresses, the possibility of two-blocking (i. e., the accidental collision of upper and lower blocks when the hook is in its uppermost position) is often discussed by American makers and owners as a serious danger, while Europeans are wholly unfamiliar with this type of accident. Why? Probably because an anti two-blocking device is a safety requirement in Europe.

Ideally anti two-blocking devices consist of switches mounted on all ropedrums. The switches are mounted in rugged housings, and require no adjustment after installation. They are preset for various speeds of operation – full, half, in chin (= 1/10 speed) – and operate at both ends of the crane's range, i.e., when the blocks are close together, and when the load is at ground-zero. The switches are installed to control both luffing and hoisting operations.

5.5.3.6. Windspeed. Safeguards and Storm Anchors

Windspeed indicators operate by means of audible and/or visible alarms. The alarms are activated normally by windspeeds of Force 8 on the Beaufort scale (= 45 mph, 40 knots, 71 kph, or 20 meters per second). Under Force 8 conditions, the brakes on the crane's traveling drive should

be adequate to prevent movement; in addition, however, easily operated rail clamps should be available which either grip the railhead or wedge the crane wheels against the railhead.

In winds of Force 11 or stronger, the crane should be parked in a safe position and anchored by means of guy ropes to secure points on the ground. Two factors must then be considered: first, the traveling drive on the crane must be specified as having enough power to move the crane into its safe position against Force 10 winds; secondly, limit switches must be installed so that the crane cannot be inadvertently started while still anchored to the ground.

5.5.3.7. Further Measures

A boom is normally further protected by a rubber or hydraulic-ram back-stop. A centrifugal speed controller is also sometimes installed to prevent over-rapid descent of a boom; this device is especially favored in Belgium and the Netherlands.

Special attention should also be paid to so-called "safety brakes." These are normally double-shoe, heavy disc, or band brakes. They must be rugged enough to hold even maximum test loads without slipping or overheating.

The safety devices built into a modern crane make it practically foolproof, apart from the "human element." Two problems emerge here. The first is the abuse of limit switches. The opening section of the "German Accident Prevention Regulations" reads:

"Limit switches should never be used as Operational switches. A good operator tests that limit switches are operational at the start of each day's work, but never uses them during . operation.

The point cannot be overstressed: good operation depends on the operator's skill not on the functioning of limit switches. The next problem area is the personal safety of the operator. Here the manufacturer's role is limited to providing the safest crane possible, following the guidance of Osha (USA) or VBG-9 (W. Germany) – it is up to the operator, however, to use the crane safely.

In ensuring operator safety, an owner specifying a crane should include specific wording to ensure that the following features are built in:

- ① Ladders should have square rungs not round ones, and one edge of the square rung should be slightly raised. This marine-type ladder is much safer when wet.
- ① Access should be, where possible, covered. Column cranes which use a centerless bearing have a big safety advantage here as against turntable cranes.

° The operator's cab should be to the side of the center-line of the boom. Research shows that the operator's judgement of distance is at its poorest when the cab is centered under the boom.

° A good-size fire-extinguisher, checked at regular intervals, should be provided in the operator's cab.

° On many cranes (particularly on certain turntable crane), the operator can leave the cab only when the turntable reaches one or two of its slewing positions. *This is dangerous.* Ideally the operator should be able to leave the cab whatever the position of the crane. If, for technical reasons, this ideal cannot be achieved, then the cab, must be equipped with a safety rope that has a belt and-automatic reel brake to allow emergency exit.

A final point on safety and convenience: the provision of an elevator, a toilet, or even an elaborate rest-room in a crane makes no contribution to safety. These expensive items require constant maintenance and deteriorate very quickly. The costs do not appear to be justified by the benefits.

5.5.4. Operator Training

There is little significant variation in the cost of operator training among crane types. All yards report similar methods of recruiting drivers: workers in related trades who show an interest in crane-driving are allowed to show their aptitude during a 2-4 week hands-on training session with an instructor (a driver himself). After a satisfactory probation, the new driver is given his own crane, always an older, minor unit, and can then "work his way up" to more important jobs.¹ Skilled drivers are greatly valued, and yards make every effort to keep them on "their" cranes. The classic case is a team of 6 drivers who have worked a set of 4 goose-neck cranes in a German shipyard for over 15 years. Their kind of expertise is not a matter of training but of experience.

Additional training courses for drivers on such subjects as general safety, rigging, basic electrics, and so on, are offered almost everywhere, but nowhere is there an official requirement that crane-drivers be trained in any way whatsoever, though labor unions recommend training and offer programs. Despite this general apathy, the writers of this report would recommend that junior operators during their first 3 years be given regular and repeated instruction in four particular areas:

- a. Safety, particularly the correct checking of limit switches.
- b. Signals and methods of communication with the ground.
- c. What constitutes crane abuse.
- d. Early spotting of irregularity in a crane's functioning.

6. RECOMMENDATION

The five cranes featured in Section 5 allow a fair survey of ship-building cranes in use today .- By comparing these five cranes, and eliminating less suitable designs, a rational investment decision can be made. The process of eliminating unsuitable designs has four steps; in each step the cranes are divided into two groups according to the presence or absence of some particular feature (e. g., double boom vs. single boom). In fact, one group contains only one crane, while the other group contains all the cranes still in contention. In each case, the single crane is eliminated. The remaining cranes are then regrouped so that the next decision can be made. The first grouping pits the double-boom crane (Crane 1) against the group of four single-boom cranes. After the double-boom is eliminated, four cranes remain: three column cranes and one turntable crane. The turntable design can now be eliminated. Among the three remaining 'cranes, two have an unbalanced boom, and one has a balanced boom, thus generating the third elimination. Finally the choice must be made between two designs: the level luffing and the non-level luffing crane.

The grounds for the preferences established below have already been staked out in Sections 4 and 5. In this section, therefore, the arguments are simply summarized. The arguments are derived both from theory, and from the buying practice of shipyards round the world; in every case theory and practice lead to the same conclusion.

6.1. CHOICE 1: DOUBLE-BOOM OR SINGLE-BOOM

IN THEORY

The double-boom crane has had its day. Design techniques *today*, especially safe welding and precise stress control, make it possible to build fast, safe cranes of almost any height at far lower cost both in terms of materials and labor applied. Level luffing, one definite advantage of the double-boom crane, can be closely approximated by rope-store methods. High-speed luffing, again an advantage of the double-boom, is seldom of practical significance in ship-building operations.

IN THE MARKETPLACE

During the 1970's and 1980's in shipyards worldwide nearly all new cranes have been of single-boom design. The remainder, the double-boomers, have been built in Communist Europe at "political prices" (prices aimed at earning foreign currency) or for barter.

RECOMMENDATION :

SINGLE-BOOM CRANES

6.2. CHOICE 2: TURNTABLE OR COLUMN ?

IN THEORY

The turntable crane is rugged--and comparatively simple in design, but it has grave drawbacks. It has a clumsy top structure that requires excessive erection time and causes unwarranted maintenance costs. Its main disadvantage, however, is that its center of gravity must remain within the periphery of the circular runway that supports the turntable. This limitation means that as capacity increases the runway becomes impractically wide; further, heavy loads cannot be hoisted at wide ranges without a disproportionate increase in the size of the crane. In fact, turntable cranes necessarily fall below the constant-load-moment standard that is the internationally expected today; this surely means that the turntable crane has been superseded. The design theory of the column crane, on the other hand, suffers from none these limitations; it offers superior all-round performance in terms of both weight and cost.

IN THE MARKETPLACE

The only areas of the world where new turntable cranes have been installed since the 1950's are areas where, for political or economic reasons, the buyers have had no real choice. Where crane-builders have been approached to design state-of-the-art machines for their clientele, the invariable choice (outside the U.S.) has been the column crane.

RECOMMENDATION

COLUMN CRANES

6.3. CHOICE 3: BALANCED-BOOM OR UNBALANCED-BOOM ?

IN THEORY

The balanced boom was created as a replacement for the double-boom. It attempted to reproduce the best features of the double-boom at a lower cost using more sophisticated engineering. It succeeded in achieving these aims: this crane achieves a fairly high luffing speed for a low application of energy. The attendant disadvantages are, however, weight and maintenance costs. For shipyard use, the benefits offered by this design are not worth the additional investments in the machine and its maintenance. Long after European designers had abandoned this design, it was still favored by the Japanese who invested heavily in these cranes in the late 50's and early 60's. Many of these cranes have been built under licence in shipyards in developing Asian countries where they can be seen in great numbers.

IN THE MARKETPLACE

During the 1970's and 1980's very few cranes indeed of the balanced-boom type have been installed. Designers have universally favored an unbalanced, single-boom design.

RECOMMENDATION

UNBALANCED BOOM CRANES

6.4. CHOICE 4: LEVEL LUFFING OR NORMAL LUFFING. ?

Cranes used in shipyards can be divided into three categories according to capacity.

Type 1. The Heavy-Duty Crane

The heavy-duty crane lifts loads of from 50 000 lbs (23 metric tons) to 440 000 lbs (200 metric tons). In shipyards, such lifts are required only occasionally, and never on a continuous basis – a few lifts a day or just one a month would be the normal spectrum. In general such lifts are pure hoists, with little need for - simultaneous luffing.

Type 2. The Medium-Duty, Fitting-Out Crane

The load range here is from 30 000 lbs (14 metric tons) up to 132 000 lbs (60 metric tons). Lifts in this range are very common in shipbuilding. Since these cranes are characteristically used in a fitting-out berth or in a floating dry dock for repairs, the loads may have to be fairly accurately placed. Movements will tend to be vertical rather than horizontal, though both kinds of movements are needed.

Type 3. The Light-Duty, Auxiliary Crane

The load range of the light-duty crane is between 11 000 lbs (5 metric tons) and 60 000 lbs (30 metric tons). These cranes are highly mobile and work in conjunction with a heavy-duty crane. Lift path is normally vertical, though there may be applications where a light crane must make frequent horizontal movements.

Since heavy lifts are performed very slowly, both in hoisting and luffing, and since heavy moves require very careful, step-by-step performance, speed will never be an essential factor in heavy-duty operations. For this reason, the advantages offered by a level-luffing

crane can be discounted. For the heavy-duty crane, normal-luffing offers perfectly adequate performance at appreciably lower cost.

Another factor here is the rope-storage system: above a certain hook-weight – roughly 220 000 lbs (100 metric tons) – the weight of the rope stored between the the A-frame and the top of the boom can become so heavy that it exercises an excessive pull on the boom tip. For heavy loads, there seems to be no economically feasible way either to avoid the immense weight of a triple-reeving system or to counterbalance this weight. This means that the top limit on level luffing (using a rope-store) is reached at about 220 000 lbs, which is appreciably below the heaviest lifts required.

A fitting-out crane of medium capacity, however, might well benefit from the ability to move loads horizontally with speed and accuracy. This would be especially true where the load path is obstructed and the load must pass round obstacles. At this weight, the rope-storage system encounters no problems.

As to light cranes, a study of a particular work-station might show that a light crane working there must often perform horizontal movements. If this is so, the level-luffing version of the crane would be preferable. In general, however, experience shows, that light cranes typically perform vertical load movements, and that the extra cost of level-luffing would not bring comparable benefits.

RECOMMENDATION

For heavy-duty : NORMAL LUFFING

For fitting out : LEVEL LUFFING

For light-duty : NORMAL/LEVEL LUFFING AS APPROPRIATE

6.5. CONCLUSION

For shipbuilding applications today, the most cost-effective choice is the:

COLUMN-MOUNTED CRANE WITH A SINGLE, UNBALANCED BOOM

For fitting-out and other medium-duty purposes, the addition of rope-store level luffing is a desirable extra feature.