CRANES and DERRICKS

FOURTH EDITION

LAWRENCE K. SHAPIRO

JAY P. SHAPIRO

Cranes and Derricks

The cover photographs of the project at 11 West 51st Street in New York City were taken by Paul Yuskevich.

Cranes and Derricks

Lawrence K. Shapiro, P.E.
Jay P. Shapiro, P.E.
Howard I. Shapiro & Associates
Consulting Engineers, P.C.
Lynbrook, New York

Fourth Edition

McGraw Hill

New York Chicago San Francisco
Lisbon London Madrid Mexico City
Milan New Delhi San Juan
Seoul Singapore Sydney Toronto

ISBN: 978-0-07-162558-6

MHID: 0-07-162558-5

The material in this eBook also appears in the print version of this title: ISBN: 978-0-07-162557-9, MHID: 0-07-162557-7.

Dedicated to Howard I. Shapiro, 1932–2007
Father, mentor, and colleague

This page intentionally left blank

About the Authors

Lawrence K. Shapiro, P.E., is a principal of Howard I. Shapiro & Associates, an engineering consulting firm based in the suburbs of New York City. A participant on national and international committees for writing crane standards, his field of practice includes tower- and mobile-crane installation design, forensic engineering, temporary structures, and construction planning.

Jay P. Shapiro, P.E., is a principal of Howard I. Shapiro & Associates. He has been consulting on means and methods of construction, including design of installations for mobile cranes and derricks, primarily in New York City, for the last 25 years. Mr. Shapiro has been a member of the New York City Crane Advisory Council. His practice includes design of rigging for heavy lifts, proof of competence analysis of mobile cranes, heavy equipment support, and demolition engineering.

This page intentionally left blank

Contents

Preface to the Fourth Edition

With the publication of this fourth edition, the stewardship of *Cranes and Derricks* passes to a new generation. And yet, it is not a complete change of tenure. My coauthor brother Jay and I have been part of this project from its start in the 1970s. As a teenager, I snapped and printed photographs that carry into the present edition. In our college years, we both critiqued portions of the original manuscript and even crafted a few lines. Finally, as practicing engineers and business partners, we shared with our father in the writing of the second and third editions.

My brother and I both developed areas of expertise that gave us the ability to contribute meaningfully to the second and third editions. Despite equal billing with our father as authors, there had never been a question that he was the maestro, the creator of *Cranes and Derricks*, and the only person in the world who was in full command of all of its subject matter. When an occasional someone would refer to it as "Howard's book," there was no point in either of us contradicting.

The passing of Howard Shapiro in 2007 was followed soon after by a request from McGraw-Hill for a new edition. Jay and I were confronted with the choice of abandoning the book or "stepping up to the plate." Despite the concurrent commitment of running a very active professional firm, neither of us had a doubt that we would take it on. At the beginning, we preferred not to dwell on the magnitude of the project. But the actual task of revising the text was eased by the comforting sense that the writing was a continuation of dialog and collaboration with our father.

Every chapter of the book has been revised, some more than others. The overall goal of our effort was to bring the text up to date with contemporary practices and technology, make it more accessible and useful to a broader audience, and better serve the crane world outside the United States. Largely, the intent of each of the eight chapters remains as before, with Chaps. 1 and 2 explaining the fundamentals; Chaps. 3 and 4 addressing engineering theory; Chaps. 5, 6, and 7 covering practice; and Chap. 8 presenting our thoughts about safety and risk control. The glossary has been greatly reworked and expanded, and there is a new appendix listing crane standards from around the world.

Cranes and Derricks has the peculiar role of covering both deep theory and day-to-day practice. There are some portions of the previous edition that are arcane, and useful only to a handful of specialists. In the present edition, we have put some of the more abstruse text in appendixes so that it does not deter the typical reader. Other portions, particularly in the discussion of derricks, describe practices that are close to obsolescence. There was a temptation to delete this material from the present edition, but we kept most of it because it presents a way of problem-solving that draws on artful skills and imaginative engineering.

The authors have been assisted by a coterie of enthusiastic reviewers, advisors, and contributors from the crane and construction industries. We limit our explicit acknowledgment to a few who made particularly noteworthy contributions. It was especially gratifying to have been mentored by our father's longtime crane industry colleagues Norman Hargreaves (Terex) and Daniel Quinn (Link Belt); they were persistent sources of assorted help and advice. Mike Quinn of Morrow Crane was invaluable in advancing the wind engineering portion of Chap. 3. Dominick D'Antonio and John Kelly of Falcon Steel provided astute critiques of several chapters; Stephanie Bass of our own office created the new line art. Michael Zhou of Zoomlion provided the list of Chinese crane standards in English translation. We give our heartfelt thanks to the others who gave us assistance large and small but are not listed here.

Lawrence K. Shapiro, P.E.

Preface to the Third Edition

The crane world is never static—that is not intended as a pun. Ever larger mobile cranes are being produced, but more importantly, innovations have come onto the scene with them. Controls and drives keep pace with microchip technology, rethinking of telescopic extension systems reduce boom weight permitting greater lengths, and new integral rapid self-assembly and disassembly arrangements are introduced. Even the notion of the load chart has been challenged by the computer revolution—separate charts for each of many configuration options result in multi-page compilations in an on-board "black box" as well as on paper.

What has not changed at all is the need for management and supervisory personnel to pay close attention to lifting operations. To do so effectively, they need to have a more complete understanding of cranes and of their strengths and weaknesses. On construction sites, and for many other applications, cranes are the key to meeting schedules and controlling costs. That is why we have written this third edition. Not only do the new technologies need to be addressed, but additional material has been developed to assist in operation planning. Our goal has always been to encourage an intelligent balance between safety and efficiency in crane use.

This third edition includes improvements to each chapter. The more significant additions are highlighted in the following paragraphs.

Chapter 2 contains new material on non-crane lifting devices, jacking towers and hydraulic gantries. We felt this material would be interesting and useful on its own, but these devices also often compete with cranes for lift assignments. Planners should therefore be conversant with them, so that rational comparisons may be made and the best equipment chosen for a project.

The wind material in Chap. 3 now includes the new wind load requirements of ASCE 7-95 which has supplanted ANSI A58.1 as the governing code. The old wind formulas and maps have been entirely superseded by this code revision which measures wind over a three-second averaging period instead of the old fastest-mile wind speed. The effect of this change in speed measurement methodology is that a new 100 mi/h (44.7 m/s) wind is *slower* than the old 100 mi/h wind. Chapter 3 offers instruction in relating the two systems.

At the end of Chap. 4 in the first edition of *Cranes and Derricks,* there was a plea for someone to devise an explicit solution to the problem of crane stability under dynamic loading. By the time the second edition was published, the authors had given up on this hope. But now a Russian engineer-friend, Anatoly Zaretsky, has come through with a solution. In the process, Professor Zaretsky has formulated a new way to conceptually consider overturning stability—a very versatile and useful way. His material has been added to Chap. 4; the chapter also now includes a study of barge-mounted cranes.

Chapter 5 has been fine tuned with more extensive discussions on crane support considerations, pick-and-carry work, and tailing operations in order to broaden planners' insights into these important matters. The authors wish to thank Mr. Ron Kohner, P.E., for his comments and observations which have served to greatly improve this chapter.

In redrafting Chap. 8, we felt an examination of the potential risks in lifting operations could help steer management and planners through their decision processes. Consideration of risk led naturally to the question of who is responsible for what among the parties involved. Significant work on that question has been done by the Specialized Carriers and Rigging Association (SC&RA), and they have graciously permitted us to use their material. Bill Smith, Director of Safety and Training for the International Union of Operating Engineers, has earned the authors' gratitude for reviewing the risk and responsibilities parts of this chapter. With his experience in the seat of a crane and in his present capacity, Bill was the right person for a reality check. The chapter and the book end with our own subjective and judgmental system for using risk assessment in determining how much planning and control is necessary for specific lifting operations.

Howard I. Shapiro, P.E.
Jay P. Shapiro, P.E.
Lawrence K. Shapiro, P.E.

Postscript

In the Preface to the first edition, the original author acknowledged the assistance of his recent-graduate sons, Jay and Lawrence. The second edition's Preface boasted, after a ten-year interval of time, that the sons were then co-authors of the book and partners in the engineering practice. Now, after nearly ten years more, the over-retirement-age original author glows with pride. His sons and coauthors are seasoned engineers with their own well established reputations and loyal client following. The technologies and methodologies they have introduced to the firm have greatly expanded the capabilities of this small office far beyond what I had ever dreamed was possible. Lastly, and with the greatest pleasure, I report that they now own and run the engineering practice, and that I work for them.

Howard I. Shapiro, P.E.

This page intentionally left blank

Preface to the Second Edition

There have been some important changes in the crane world during the ten years since the first edition of this book was published. Larger and more versatile telescopic mobile cranes have become commonplace, affecting both rigging and construction work, and luffing boom tower cranes are now a regular feature of the urban skyline, having replaced guy derricks, for example, at many steel erection projects. Along with those changes, however, society has become even less tolerant of accidents than in the past, and global competition now demands greater efficiency and productivity from all segments of the economy. This new edition reflects the changes in the industry, but both the first edition and this revised one address safety and efficiency in the same way—with an impassioned plea for the adequate preplanning of crane operations as the primary step to achieve both goals.

A recent development in telescopic cranes, the all terrain crane, has captured a significant share of the mobile-crane market; a description of this crane type has been added to Chap. 2.

Chapter 5 has been expanded to include additional means for supporting mobile cranes, especially near building foundation walls. A shortcoming of the first edition has been rectified by inclusion of a full treatment of multiple crane lifts. This new material includes discussion of the effects of the characteristics of the load on lift planning, how load handling affects crane loading, why multiple crane lifts are far more critical than single crane lifts, and what you can do to control the risks.

Much of Chap. 6 has been rewritten to reflect current practices in tower crane installations and to amplify coverage of such subjects as exterior climbing cranes. Additionally, material on tower crane erection, jumping, and dismantling has been revised and enhanced to give far more practical information on how to plan these vital operations.

Chapter 8 has been updated with an expanded discussion of controlling lifted loads as a basic factor in crane safety. This includes consideration of operator and supervisor training as well as load and load-moment devices. Operations on rubber and "pick-and-carry" operations are also more fully discussed, and the section on tower crane accidents has been significantly enlarged. A new section on lifting personnel with cranes has been added.

Perhaps the most significant development of the last ten years is that the original author's sons are his coauthors for this new edition and his partners in a firm practicing crane engineering. Most of the new insights, new methods, and expanded coverages of this edition reflect their input. They have indeed fulfilled the original author's wish, as expressed in the first edition's dedication, by "... carry[ing] the work much further themselves."

Howard I. Shapiro, P.E.
Jay P. Shapiro, P.E.
Lawrence K. Shapiro, P.E.

Preface to the First Edition

Anyone seriously interested in cranes and derricks will be able to find pearls, in a figurative sense, between these covers, but many readers will not feel justified in opening too many oysters in their search for pearls. Although this book can be read through from end to end, readers with previous experience or familiarity with the equipment may wish to read only part of it. Since the book is addressed to an audience with a fairly broad range of interests, those interests will govern how individuals will want to read and use this book.

The first two chapters are introductory and describe the forms of equipment in use, as well as their components and accessories. These chapters are essential for beginners, but more experienced readers will find it worthwhile at least to skim through this material rather than bypass it altogether. Chapter 1 describes the mechanisms and accessories common to hoisting equipment, but—more importantly—it introduces the terminology, jargon, and concepts used in the industry. The basic mathematics of hoisting devices is included. Chapter 2 deals with the equipment itself, showing how different arrangements of the basic mechanisms and components result in changes in function and capability. In addition, size range, capacity, uses, and limitations are examined.

The next two chapters deal with the operating regime, that highly technical set of conditions which affects the practical work produced by cranes and derricks and which is so intimately connected with operational safety. The mathematical treatment enables design engineers to satisfy their need to quantify the happenings in crane and derrick life, while the descriptive material gives the insights needed to appreciate the limitations and dangers associated with various equipment types. Although many readers can pass over the technical portions, at least a general understanding of the concepts in these chapters is essential for all readers.

Chapter 3 discusses both obvious and subtle static and dynamic loading effects associated with crane and derrick functional motions and delves rather deeply into the subject of wind. The effect of wind, and particularly gust action, is too often overlooked or insufficiently appreciated. Chapter 4 explores the physical concepts and the mathematics of stability against overturning, both as a static phenomenon and in a dynamic context. Overturning is the primary failure mode for the most common hoisting equipment.

The chapters that follow are type-specific in that they detail the considerations essential in making practical installations of the main equipment types. The focus of the material is the jobsite and the measures that need to be taken to assure productivity and safety. Here, again, the treatment is both mathematical and descriptive. Readers with an interest in only one equipment type can safely skip the chapters dealing with other types or, better yet, skim through them.

Chapter 5 concerns mobile crane installations and the movement of the machines both on the site and between sites. The mathematics for assuring proper crane support and for determining functional clearance is given. For those who do not need this material, it is important to know that these matters *can* be calculated. There is no need to resort to guesswork or rules of thumb.

Chapter 6 covers tower cranes from erection to dismantling in each of the installation configurations. Tower cranes are perhaps the least understood crane type in the United States, but their growth in physical size and frequency of use makes it necessary for all those concerned with these machines to be familiar with operational and installation concepts.

Chapter 7 is a discussion of installation details and criteria for the various derrick forms, including operational effects on the host structure. Mathematical procedures are introduced as needed.

The closing chapter, which covers safety and liability, encompasses materials applicable to all crane and derrick types. The emphasis is on preventing accidents and establishing practices that will improve productivity while reducing liability in case of accident.

If this book could be said to carry a message, it is that preplanning is a requisite for successful crane and derrick use. Preplanning will not only lead to a reduction in accidents but also measurably increase productivity and therefore directly justify its cost. During preplanning, problems are identified and solved in advance so that field crews are not left standing about waiting for someone to make a decision. With preplanning, the right equipment is sent to the jobsite, because field needs have been predetermined and capacities, sizes, etc., evaluated. And lastly, when there is preplanning, field personnel are not forced to push the machines up to and beyond their capabilities in order to get the job done. Therefore, accident potential is reduced, and the equipment can be utilized to its full productive potential.

Several people have assisted in the writing of this book by reading various portions and offering their criticisms. All but a few of their suggestions have been incorporated. Robert Del Duca, an executive of Gendelman Rigging and Trucking, Inc., who began his career on a rigging crew, has read most of the book, and his help has been invaluable. Additional critical review was provided by George A. Allin, P. E., who until his untimely death was Director of Engineering for the Harnischfeger Corporation; Erik Andersen, President of Tower Cranes of America, Inc.; Dinesh Seksaria, formerly with Lima Division of Clark Equipment Company; and William Chieco, my colleague at Charles M. Shapiro and Sons, P.C., Consulting Engineers.

My sons, Jay P. Shapiro and Lawrence K. Shapiro, have been very much involved with this work since its inception, when they were both still students. They have been militant proponents of the clearly expressed thought.

I happily acknowledge this assistance and remain indebted to the people who were kind enough to give me their help.

Howard I. Shapiro, P.E.

This page intentionally left blank

Cranes and Derricks

This page intentionally left blank

CHAPTER 1

Basic Concepts and Components

Cranes and derricks lift and lower *loads,* by means of ropes and pulleys, and move the loads horizontally. Machines that do not perform those functions, or do them by other means, are neither cranes nor derricks by common definition. A *crane* is a self-contained piece of equipment, whereas the prime mover or power source of a *derrick* is a freestanding unit separate from the hoisting structure. The derrick and its power source are brought together for each particular job.

The speeds at which load-handling functions are performed, the load weights that can be handled, the heights to which loads can be lifted, and the locations of the machines while doing work are attributes that differentiate an extremely diverse range of equipment. The equipment varies from the humblest of devices to state-of-the-art automatic machines; they are built by backyard mechanics and transnational companies. The nature of their activities exposes these lifting machines to dynamic loading arising from their operating motions, but they often also support loads imparted by their operating environment, including the effects of wind, snow and ice, even earthquakes, and temperature extremes.

Cranes and derricks are utilized for lifting service across a wide spectrum of operating conditions. They may be exposed to infrequent duty, as in power plant turbine house service where passive work—maintenance and testing—is the usual use and productive working lifts are occasional, or they may be punished by intense use such as in steel mill service where round-the-clock operations induce millions of loading cycles. The assortment in forms, environments, and operating regimes makes the selection and installation of *hoisting* equipment an artful skill that is critical to a successful project.

An ever-present concern whenever cranes and derricks are at work is safety. By their nature, rigging and lifting loads entail risk, and when accidents occur, they are almost always dramatic and newsworthy. Safety is an abstract notion. The term is easily used in discussion or writings, but the actuality is not always easy to achieve in practice; there are few simple fixes and no panaceas. Devices intended to enhance safety may yield no benefit in actual use, or even have a contrary effect. What may improve safety under one set of circumstances can have negative effects under another. In many situations, the most effective safety enhancements are small and subtle measures tailored to the risks associated with the particular operation. Safety is best served by time spent in planning lifting operations, thereby avoiding the exigencies and uncertainties of procedures devised on the spur of the moment.

One goal of this book is to help sharpen the awareness of lift planners in identifying the sources of risk and direct them toward their mitigation. Another goal is to assist planners in applying cranes and derricks to do useful work, to aid efficiency and productivity, and to reduce costs. There is no conflict between cost reduction and safety. Well-planned safety enhancements will be proportional to the risks; they will not be burdensome to field crews, and in fact, they often enhance productivity.

Good planning will not go far unless there are competent and responsible workers to carry it out using quality equipment suited to the task. Several fine field references and pocket handbooks are available for crane and derrick workers. This book is complementary to such books rather than competitive. The field references offer "how to" and "how not to" guidances for field crews, and the handbooks provide specific information about hardware and practices. *Cranes and Derricks* is a planning guide and tool. Though field personnel and supervisors will find much useful material in this book, its place is in the office, and its role is to support equipment selection, installation design, and prelift planning.

The presentation is addressed to engineers and to crane and derrick users and supervisors with a technical background. Materials of particular interest to engineers are set off in contrasting type. For readers who do not require the engineering material, there will be no loss of continuity if the portions in contrasting type are passed over.

1.1 Introduction

The principles used in designing and installing hoisting equipment cover the breadth of engineering disciplines but most particularly the fields of mechanical and structural design. Take, for example, a telescopic truck crane. A contemporary example of one of these machines typically is a complex of diesel power, hydrostatic and hydrodynamic transmissions, electric and computer controls, and perhaps pneumatics too. Some have myriad sensors wired into an onboard computer that stores the load charts, monitors operations and performs diagnostics. The *boom* with its panoply of attachments, the machinery deck, and the truck frame are subjected to widely varying loads. Some of the components must be designed to criteria that limit deflections, others for stress, and perhaps some with consideration of service life.

Installation design has its complexities too. Loadings and motions are in three-dimensional space and vary with time. Care is required to discover the conditions that most affect machine supports and to assure that adequate clearances or protections are provided to prevent collisions with other objects. The installation designer often must also consider the logistics of getting the equipment in place and the capabilities of the crew doing the work.

Engineering first came into being as a field of practice when scientific principles started to be applied to the problems of designing machines and structures. Until then, things as varied as cathedrals and flour mills were designed and built by tradesmen; journeymen working under the supervision of a master chosen for skill, experience, and proven judgment. The engineer's tools for applying scientific principles—then and now—are the various components of mathematics. Solutions were obtained by tedious calculations, often with the help of graphical constructions, tables, and simplified formulas. But the use of experience did not die out. When science proved inadequate, or mathematics or calculation too difficult or slow, rules based on experience came into play.

Rules of thumb are guides based on experience, and together with tables of data and graphs of experimental results are important tools in the engineering design office. A good rule of thumb may yield reliable results, but such is not always the case. Working with this method alone can lead to wasteful and occasionally inadequate design. Nonetheless, rules of thumb may be a useful means for deriving a trial solution to an engineering problem.

Engineering design passed through a transformation, starting about 1970, emerging from the era of the slide rule and the rule of thumb into the electronic age. Inexpensive user-friendly programs now provide a potent approach to problem solving. Experts and novices alike enjoy access to an ever-expanding magnitude of computational power. It is an alluring notion that much less engineering experience is needed because of the profundity residing behind those desktop screens. But the reams of data and mesmerizing color graphics that these programs output could be nothing more than garble; and even when computer modeling is done correctly, it requires skilled interpretation. Uncritical acceptance of program output is even more likely to lead to profound errors than blithe use of rules of thumb.

Too many of us have been lulled into belief by the convincing appearance of a computer program's output, only to learn later that it is fatally flawed by a modeling error or oversimplified methodology. Verification of computer solutions is well advised. The engineer's toolbox still should include graphs, charts, and formulas to carry out these verifications. And nonengineers should beware of programs intended to replace engineering expertise with quick on-screen answers.

While a rule of thumb is an amalgam of experience with little or no science, a computer program offers a mathematical solution that may have a tenuous attachment to the real world. Each by itself will not always prove satisfactory as a general problem solver. Together they are part of a comprehensive set of tools that give modern engineers capabilities that their predecessors would envy.

Successful crane and derrick engineering practice requires more than analytical tools and rules of thumb. Time spent in the field observing crane setup and operations is an important reality check, which culls out those ideas that actually work from those that only work on paper. An engineer who spends time in the field also learns the crucial role of the human element. A capable crane and rigging installation designer is one who understands the abilities and limitations of contracting firms and their people just as well as the equipment they will use.

1.2 The Basic Hoisting Mechanism

Figure 1.1 shows a lifting device with a load attached to the lower block and the block in turn supported by two ropes, or *parts of line*, suspended from the upper block. Each rope must therefore carry half the weight of the load; this gives the system a mechanical advantage of 2. Had the load been supported by five ropes, the mechanical advantage would have been 5. Mechanical advantage is governed by the number of ropes actually supporting the load. As parts of line are

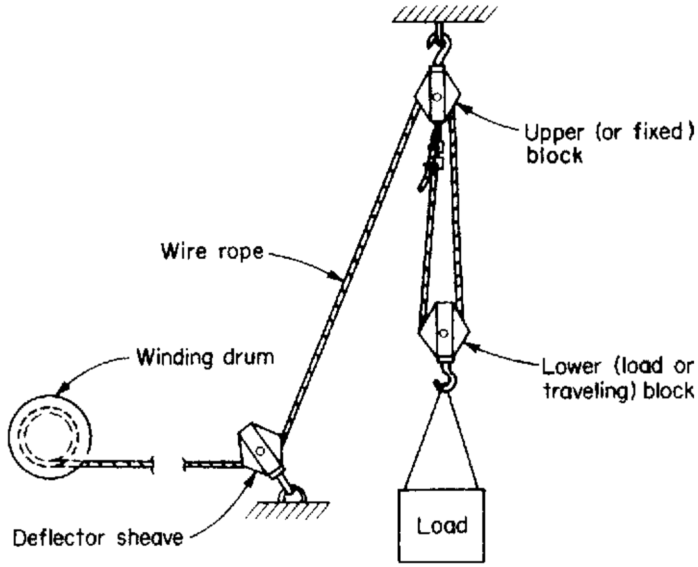


FIGURE 1.1 Basic hoisting mechanism.

added, the force needed to raise or lower the load decreases, and load movement speed decreases as well.

The blocks contain pulleys, or *sheaves*, so that the rope is in one continuous piece from the end attached to the upper block to the winding drum. This makes the force in all parts of the rope uniform in a static system. The value of the rope load is found by dividing the weight of the lifted load by the mechanical advantage; in Figure 1.1 the lifted load would include the lower block, sometimes called the *hook block*. When the distance between the upper and lower blocks is great, it is necessary to include the weight of the parts of line as well.

The load in the rope is also equivalent to the force that must be generated at the winding drum in order to hold the load.

The effects of friction come into play as soon as the system is set into motion. Friction losses occur at the sheave shaft bearings and in the *wire rope* itself, where rope losses result when the individual wires rub together during passage over the sheave. These losses induce small differences in load between each rope segment (i.e., each section of rope from sheave to sheave). The loss coefficient can vary from a high of about 4½% of rope load for a sheave mounted on bronze bushings to a low of as little as 0.9% for a sheave on precision ball or roller bearings. An arbitrary value of 2% is a reasonable approximation for sheaves on common ball or roller bearings when the rope makes a turn of 180°.

The tension in the rope at the winding drum is different when the load is raised and when it is lowered. Friction losses are responsible for this difference. When load-weighing devices that operate by reading the tension in the line to the drum are used, the variation is readily observed.

When an unloaded hook must be lowered, lowering will be resisted by friction, by the weight of the rope between the upper block and the *deflector sheave*, and by the inertia of the winding-drum mass. Mechanical advantage works in reverse in this case, as a mechanical disadvantage so to speak, so that the weight at the hook must exceed the rope weight multiplied by the mechanical advantage plus an allowance to overcome friction and inertia. If the weight at the hook is less than the result of this calculation, the hook will not lower; for that matter, if the weight is significantly less, the hook will rise on its own and will not stop until it strikes the upper block. To prevent this action, it is necessary to have a lower block with adequate weight or to add an *overhauling weight* (overhaul ball) so that the rope will overhaul through the system. Since the overhaul weight becomes part of the dead weight of the mechanism and remains in place throughout operations, it must be taken into account in operating plans. It is part of the lifted load.

Figure 1.2 shows the basic hoisting mechanism on a working crane. In fact it illustrates two separate sets of mechanisms; the *main fall* is a multipart line suspended from the boom tip, and the *auxiliary fall* is a single-part arrangement. The lower block on the main fall is

Figure 1.2 Link-Belt model RTC-8090 Series II rough terrain crane. Note the two wire ropes each leading from a winding drum over the upper surface of the *boom* to a deflector sheave at the *boom* tip. One rope continues to the fixed upper block and thence to the load block, while the other runs over a second deflector sheave to the auxiliary hook and ball. (*Link-Belt Construction Equipment Company.*)

provided with heavy side plates, called *cheek weights*, for *overhauling* while the auxiliary fall is overhauled by a cast weight sometimes called a *headache ball*.

The friction effect on *line pull* can be large on systems with many parts of line or multiple sheaves between the *fall* and the winch. On such systems, friction should not be ignored. For a given number of sheaves and friction loss per sheave, the system loss can be calculated. Referring to Figure 1.1, W is the weight of the load and the lower block, P the force at the winding drum, and μ the loss coefficient. During raising, the rope between the deflector sheave and the upper block carries the force $(1 - \mu)P$, and the ropes supporting the load carry $(1 - \mu)^2P$ and $(1 - \mu)^3P$, respectively. In order to lift but not to accelerate the load, there must be a force

$$P = \frac{W}{(1-\mu)^2 + (1-\mu)^3}$$

When lowering the load, the friction effect is opposite, so that the force in the rope from *deflector sheave* to upper block becomes $P/(1-\mu)$. The ropes supporting the load will then experience forces of $P/(1-\mu)^2$ and $P(1-\mu)^3$, and the holding force at the drum will be

$$P = \frac{W}{(1-\mu)^{-2} + (1-\mu)^{-3}}$$

The preceding equations can be generalized. If n is taken as the number of parts of line supporting the load and m is the number of 180° turns taken by the rope between the upper block and the drum (turning angles for each of the sheaves are added to find the number of 180° multiples), then

$$P = \frac{W}{r} \tag{1.1}$$

For raising the load,

$$r = (1-\mu)^{m+1} + (1-\mu)^{m+2} + \cdots + (1-\mu)^{m+n}$$

which simplifies to[1]

$$r = \frac{(1-\mu)^{m+1} - (1-\mu)^{m+n+1}}{\mu}$$

For lowering the load,

$$r = \frac{1}{(1-\mu)^{m+1}} + \frac{1}{(1-\mu)^{m+2}} + \cdots + \frac{1}{(1-\mu)^{m+n}}$$

which simplifies to

$$r = \frac{1-(1-\mu)^n}{\mu(1-\mu)^{m+n}}$$

Example 1.1

1. A load of 15,700 lb (7121 kg) is to be lifted by a hoisting device with four parts of line and a lower block of 300 lb (136 kg). Neglecting friction, how much force must be developed at the drum?

Solution When friction is neglected, drum force will be the same as rope loading. The mechanical advantage is 4 when four parts of line support the load.

$$P = \frac{15{,}700 + 300}{4} = 4000 \text{ lb } (17.79 \text{ kN})$$

2. When friction is taken into account, how much force must the drum exert to raise the load? How much to lower it? Assume that there are sheaves mounted on ordinary ball bearings and three deflector sheaves taking the rope through two 180° turns.

---

[1]For these simplified expressions, the authors wish to thank Mr. Keith Trommler of Bragg Crane and Rigging, Long Beach, California.

Solution Using $\mu = 0.02$, $m = 2$, and $n = 4$, we get $1 - \mu = 0.98$. From Eq. (1.1), for raising a load,

$$r = \frac{0.98^3 - 0.98^7}{0.02} = 3.65$$

$$P = \frac{15{,}700 + 300}{3.65} = 4384 \text{ lb } (19.50 \text{ kN})$$

and for lowering the load,

$$r = \frac{1 - 0.98^4}{0.02 \times 0.98^6} = 4.38$$

$$P = \frac{15{,}700 + 300}{4.38} = 3653 \text{ lb } (16.25 \text{ kN})$$

3. If the rope from the deflector sheave to the upper block weighs 20 lb (9.1 kg) and 50 lb (22.7 kg) is needed to overcome drum friction, will the system overhaul when unloaded? If not, how much overhauling weight must be added?

Solution Ignoring sheave friction, the weight that must be overcome in order to lower the load block is $20 + 50 = 70$ lb (31.7 kg). With an MA of 4, this requires $4 \times 70 = 280$ lb (127.0 kg) at the load block. But the load block weighs 300 lb (136.1 kg); the system will overhaul.

With sheave friction considered, the weight to be overcome is $50/0.98^2 = 52.1$ lb (23.6 kg) after passing through two 180° turns, plus 20 lb of rope weight, for a total of 72.1 lb (32.7 kg). For lowering, r was found to be 4.38. The weight required for overhauling is then

$$72.1 \times 4.38 = 316 \text{ lb } (143.2 \text{ kg})$$

a result greater than the lower block weight. Thus an additional weight of at least 16 lb (7.2 kg) is needed.

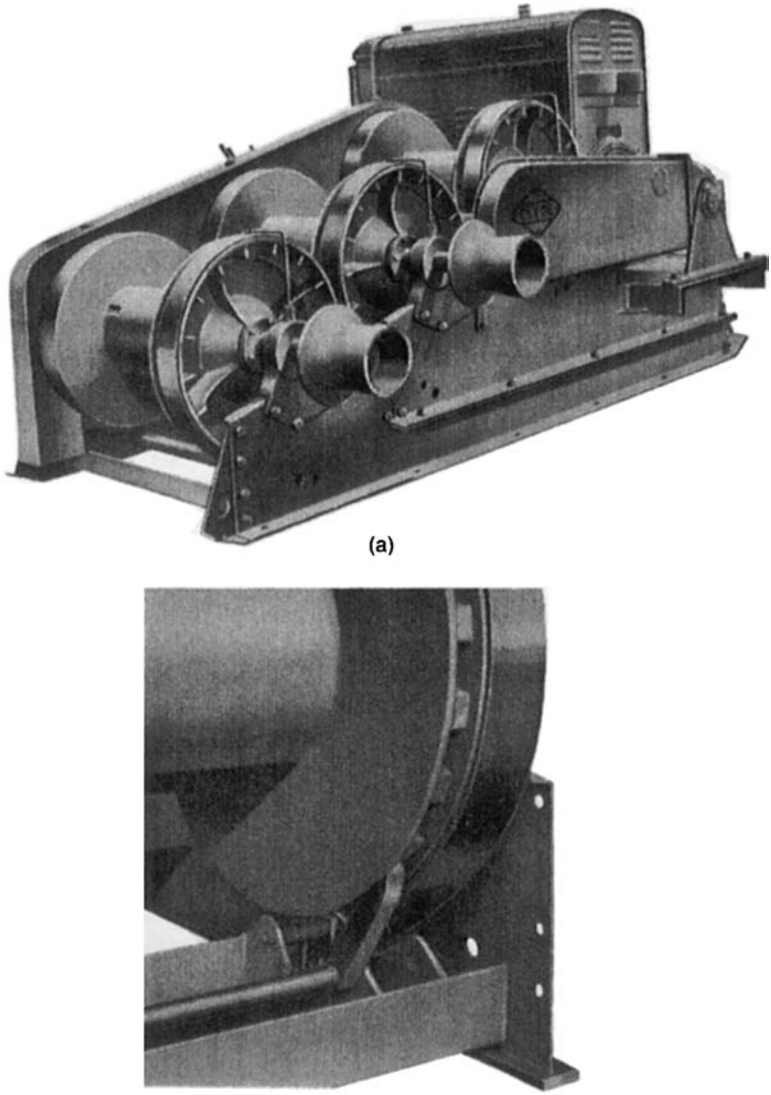
In practice, an overhaul weight somewhat in excess of the calculated value is used. This allows for possible inaccuracy in the loss coefficient and provides the mass needed to overcome inertia and induce acceleration. When friction losses are neglected in calculations, it would be wise to increase the overhauling weight appreciably.

1.3 Drums, Hoists, and Sheaves

An assemblage of one or more winding drums mounted on a frame is called a *hoist* or *winch* (Figure 1.3). A free-standing winch unit with integral controls and power plant is called a *base-mounted drum hoist.* Power sources, drives, and control systems vary widely.

A conventional winch is powered by a gasoline or diesel engine with a mechanical gear train and two or three drums. A disconnect *clutch* may be provided between the engine and gear train with additional clutches mounted at each drum.[2] A brake band is built around the periphery of one flange on each drum, with the flange widened for this purpose. Each drum can therefore be individually controlled. For additional safety, a *ratchet-and-pawl system* can be included at each

[2]Previous editions of this book used the term *friction* for the individual drum clutches. This word usage has become obsolescent.



(a)

(b)

FIGURE 1.3 (a) Three-drum gasoline hoist with torque converter. (b) Detail showing ratchet-and-pawl dogging arrangement. (*Clyde Iron, a unit of AMCA International Corporation.*)

drum, providing a means for positively locking the drum against inadvertent spooling out of rope. When the *pawl* is engaged, the drum is said to be dogged or *dogged off*.

The operator starts the winch by running the engine and engaging the *main clutch*. This action sets the gear train in motion, but the drums will not turn with their individual clutches disengaged. To lift a load, the operator engages the clutch on the appropriate drum. As the drum starts to spool in rope, the pawl will automatically disengage from the ratchet on the drum flange. The engine throttle is usually set to operating speed before lifting so that load acceleration is controlled by gradual engagement of the clutch in conjunction with use of the foot brake. To stop lifting, the clutch is disengaged and the foot brake applied. The drum pawl should be engaged if the load must be suspended for an extended period. To lower a load, the pawl is first released by momentarily engaging the drum clutch. Lowering is controlled by the foot brake.

Before the operator leaves the winch unattended, the load must be lowered to the ground—all drum dogs engaged, clutches disengaged, and the engine stopped. A grounded load will pose no threat of falling, but a machine with the engine running and main clutch engaged can be dangerous and in violation of regulations such as OSHA. With a live winch, an inadvertent engagement of a clutch could raise the hook, or a momentary engagement could release the dog and cause the hook to fall.

Modern hoist machines only superficially resemble their rudimentary predecessors. Some now include pneumatic, electric, or hydraulic controls and hydraulic transmissions. Powered or free fall lowering options are available—power-down now being the more prevalent mode—and dogs may disengage automatically.

Because so many older winches remain in use, derrick operation often still employs mechanical winches with clutches at each drum. On some older winches, lowering is a *free-fall* operation, retarded by the brakes. On others, lowering may be controlled by torque converter slippage or by the brakes.

Derricks installed on the roofs of tall buildings pose special winch problems. Lowering heavy loads over great heights overheats the brakes or the torque converter; this necessitates cooling-off stops. Urban rigging contractors often use modified winches with oversized brake bands and transmission-oil coolers for this severe service. These specialized winches are often capable of disassembly into components that can be transported on freight elevators and up stairwells; they may also have narrow drums to reduce the fleeting lead distance so that they can be set up in snug spaces. Information about derrick winches will be found in Chap. 7.

Winch units for use in a crane are tailored to the specific requirements of the crane and are integrated with the control and mechanical systems. Hydraulically driven winches and some electric winches can be furnished with automatic brakes that are normally engaged. When either power-up or power-down control signals are initiated, the application of power to the *drum* triggers release of the brake. These winches often are not equipped with *dogs*.

Tower cranes and many contemporary mobile cranes have electric or hydraulic winches that do not permit free-fall lowering. Powered lowering—or lowering against the retarding action of a torque converter—provides good load control for precision work such as placing machinery on anchor bolts.

Hoist Drums

A winding drum transmits power to the wire rope, and it also serves as a reservoir by spooling and storing rope that is in excess at the moment. Each turn of the rope around the full circumference of the drum is called a *wrap*. Rope is helically wrapped around the drum, starting at one end flange (see Figure 1.4) and progressing toward the opposite flange. A series of wraps extending from flange to flange is referred to as a *layer*. If spooling continues after the completion of a layer, the wraps proceed back toward the starting flange in a second layer.

In an operating winch, when the drum is spooled full, some flange must be left projecting beyond the surface of the outermost layer of rope. A flange projection of 1 to 1½ rope diameters is recommended.[3] Flange projection above a fully loaded drum is a necessary precaution so that the rope will not slip off. As the rope spools in on the uppermost layer, the rope is being forced in the direction of the flange by the previous wrap. When the end of the drum is reached, friction between rope and flange will cause the rope to attempt to climb the flange. That friction force is quickly overcome—as the rope climbs—by the component of line tension that develops in the opposite direction.

Another condition that requires ample flange height develops when the *hoist line* is relieved of load. If the unloading rate is rapid, the rope releases its strain like a mildly stretched rubber band and could jump over the flange edge.

The end of a rope is attached to the drum barrel by a socket or clamp. As rope is spooled out, at least three full wraps must remain on the drum for safety.[4] (If unspooling continued until all the rope was paid out, the rope attachment would be subjected to a shock loading as the lowered load suddenly stopped and reversed. The rope attachments are neither as strong as the rope itself nor as capable of absorbing shock.)

To assist in making first-layer wraps closely placed and uniform, drum barrels are often grooved. A good first layer is a necessity if succeeding layers are to wrap properly; the rope itself provides a groove effect for subsequent layers. Grooves are cut to suit a particular rope diameter, and grooved drums can be properly used only for that diameter of rope. A helical pattern can be used, but better performance

---

[3]ISO 10972-2: 2009.

[4]ISO 10972-2:2009 requires three wraps minimum when the rope is anchored to the drum with a wedge anchor and a minimum of five wraps when anchored with set screws clamped to the rope.

results from grooves cast parallel to the rope direction in a proprietary arrangement known as *Lebus grooving*.

Removable shells, called *laggings*, can be added to drum barrels that are arranged to receive them. Laggings permit drums to be adapted for rope of another size but can also be used to increase line speed by increasing barrel diameter (reducing rope storage capacity at the same time). Drums operating at very high line speeds exhibit spooling problems that increase as the number of layers increases. Lower layers tend to become loose and sloppy, causing excessive wear and premature failure. This occurs in part from rapid braking inducing greater inertia on the upper layers than on the lower. Subsequent reloading tightens the upper layers, leaving the lower layers loose. Respooling under tension, starting at the first layer, is the recommended remedy. When a hoist continues to operate with loose lower layers, wraps from one layer may fill the gaps created in a lower layer. As rope spools out under load, the wraps that had become pinched in a lower layer can experience significant wear, abrasive damage, and shock load when pulled free. Layers may also loosen from raising an unloaded hook with an overhaul ball that is too small.

Wear and damage can also occur at *flange points* and *crossover points*. A flange point is where the rope contacts the drum flange as the rope starts another layer. A crossover point is where rope on one layer contacts and crosses the rope in a preceding layer.

Single-layer drums are used, where feasible, to eliminate several causes of wear and rope damage, but another feature of a single-layer drum is its ability to deliver constant line speed and constant *line pull*. Winches equipped with single-layer drums have two other notable features. They can be furnished with spooling protection devices which shut down the *hoist drum* should a second layer develop. They can also be arranged to spool both ends of the rope, eliminating the *wire rope dead end* and doubling hoisting speed.

Fleet Angle

To aid proper spooling and to prevent excessive wear on the rope and on drum grooves, the maximum angle at which the rope leads onto the drum, called the *fleet angle*, must be kept within controlled limits. Figure 1.4 illustrates the fleet angle, which is the angle whose trigonometric tangent is one-half of the drum-barrel width divided by the distance between the shaft centerlines of the drum and lead sheave. The lead sheave is a fixed-position deflector sheave aligned with the center of the drum. Fleet angles should be no less than ½° and no more than 1½° for smooth drums or 2° for grooved drums when the lead sheave is centered on the drum. A fleet angle of 1½° requires that the lead distance be 19 ft for each foot of drum width. A 2° fleet angle requires 14⅓ ft. (In the SI, the same distances apply but with meters as the unit.)

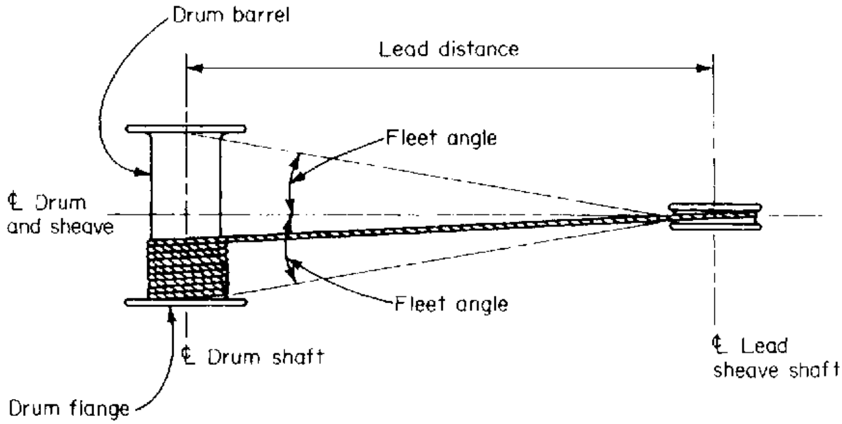


FIGURE 1.4 Fleet angle.

Sometimes it is neither practical nor possible to install the lead sheave at the required distance, and a shorter distance must be accommodated. Proper spooling can still be maintained if a pivoted block (Figure 1.1) is used for leading to the drum. A pivoted block will lie over from side to side following the rope as it spools on the drum. When using a pivoted block, one must be sure that it is set close enough to the drum to provide at least ½° of fleet angle in the maximum layover position. Too small a fleet angle will cause the wraps to pile up at the flange (particularly at low loading levels), whereas an adequate angle will guide the rope away from the flange.

If a still shorter lead distance is needed, the lead sheave can be mounted on a horizontal shaft that lets the sheave move laterally as the rope spools. This is called a *fleeting sheave*. The shaft length must be selected so that the minimum recommended fleet angle is respected.

In all lead sheave arrangements, but especially when the lead is short, it is advisable to mount the sheave preceding the lead sheave so that it aligns as closely as possible with the lead sheave; this will mitigate the wear on rope, sheaves, and bearings.

Drum Capacity

Wire ropes are initially manufactured with oversized diameters, but then they elongate and reduce in diameter because of wear and tear as they are used in service. Also the tightness of the wraps can vary from one spooling to the next. These factors affect the length of rope that spools onto a drum and make *spooling capacity* calculations inexact. Reasonably good values can be determined, nonetheless, using the method that follows. Referring to Figure 1.5, let us consider three cases of drum capacity, each using the equation

$$L = (D + E)EBs \tag{1.2}$$

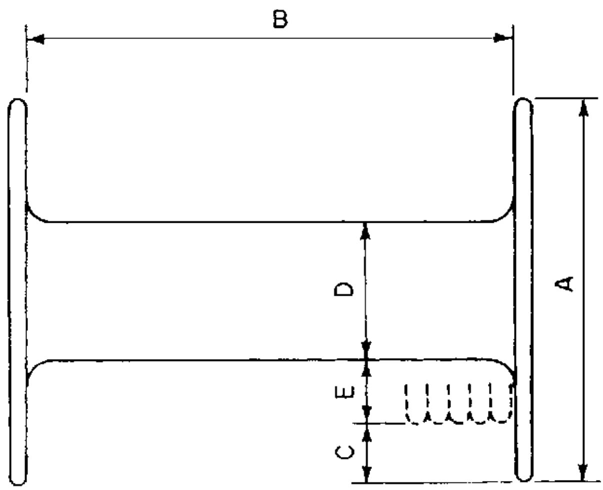


FIGURE 1.5 Hoist-drum dimensions.

U.S. customary units L is given in feet when dimensions D, E and B are in inches, and the spooling factor is taken from Table 1.1 or calculated using Eq. (1.2a). Table 1.1 assumes new rope and includes a rope diameter oversize factor of 5%. Taking d as the actual rope diameter, in inches for U.S. customary units or millimeters for SI units, the following expressions for the spooling factor may be used in lieu of Table 1.1 values:

$$s = \frac{0.2618}{d^2} \quad \text{in U.S. customary units}$$
$$s = \frac{0.00285}{d^2} \quad \text{in SI units} \qquad (1.2a)$$

The first case involves the maximum quantity of rope that can be stored on a drum when the hoist is not in operation. Dimension C is taken as zero and

$$E = \frac{A - D}{2} \qquad (1.3)$$

which is then substituted into Eq. (1.2), giving the maximum stored length L.

The second case is used to determine the maximum quantity of rope that can be spooled onto the drum of an operating winch. C is taken as ½ in (12.7 mm) or preferably as one rope diameter, and

$$E = \frac{A - D - 2C}{2} \qquad (1.4)$$

The third case is used to estimate the quantity of rope found on a drum. The dimension C is measured, and Eq. (1.4) is solved for E. When E is substituted into Eq. (1.2), the stored length is found.

Rope Diameter *d*	Spooling Factor *s*	Rope Diameter *d*	Spooling Factor *s*	Rope Diameter *d*	Spooling Factor *s*	Rope Diameter *d*	Spooling Factor *s*
½	0.925	13/16	0.354	1⅜	0.127	2	0.0597
9/16	0.741	⅞	0.308	1½	0.107	2⅛	0.0532
⅝	0.607	1	0.239	1⅝	0.0886	2¼	0.0476
11/16	0.506	1⅛	0.191	1¾	0.0770	2⅜	0.0419
¾	0.428	1¼	0.152	1⅞	0.0675	2½	0.0380

Table 1.1 Spooling Factors *s* for Drum-Capacity Calculations in U.S. Customary Units

The preceding equations can be manipulated and used to solve any number of practical drum problems, as will be seen later in this chapter.

Line Pull

Hoist drums are rated by *line pull* (the tension the drum is capable of applying to a rope leading onto it) and by line speed. Ratings are generally specified at the first layer but may be given for the top layer as well. As rope spools onto the drum, line pull decreases and line speed increases, but available torque remains constant. If we use the dimensions of Figure 1.5, with a rated line pull of P_r on the first layer, the rated torque T is given by

$$T = \frac{P_r(D+d)}{2}$$

for a system with nominal rope diameter d and with consistent dimensional units. The usable line pull P_u at any other drum layer is found from

$$P_u = \frac{2T}{A-2C-d} = \frac{P_r(D+d)}{A-2C-d} \tag{1.5}$$

If the rated first-layer line speed is given in feet per minute as V_r and drum dimensions are given in inches, the drum speed ω in revolutions per minute is given by

$$\omega = \frac{12V_r}{\pi(D+d)}$$

For SI units, the value 12 is replaced by the power of 10 needed for dimensional consistency. The line speed V_u at any other rope layer is

$$V_u = \frac{\omega\pi(A-2C-d)}{12} = \frac{V_r(A-2C-d)}{D+d} \tag{1.6}$$

Sheaves and Blocks

Sheaves are used to change the direction of travel of wire ropes. Assembled in multiples, in the form of blocks, they are able to provide almost any required mechanical advantage.

Ideally, sheaves should be mounted in exact alignment with each other, but since in practice this rarely occurs, the grooves are shaped to provide some tolerance for misalignment. In the discussion on fleet angle, it was noted that a 2° lead angle can be accommodated without difficulty, but that figure is maximum amplitude of an ever-varying angle; a constant misalignment causes the rope to rub one side of the groove, resulting in wear on the rope and sheave and shortening the useful lives of both.

Sheaves rotate about their mounting shafts on bushings or bearings. A reasonable value for friction loss at bushings can be taken as 4½%, while bearings produce losses of 1% to 2%, depending upon their quality and the conditions of service. These losses are rough average figures for ropes making a bend of 180° over the sheave and can be reduced for smaller turning angles. Actual friction losses are a function of the style of rope, the ratio of sheave to rope diameter, and the bearing type.

There is no minimum sheave or drum diameter that would prevent a hoisting mechanism from operating, but increasing sheave and drum diameters have a direct correlation with improved rope life, as indicated by Figure 1.6. The minimum ratios of sheave or drum to rope diameter for cranes and derricks stipulated in U.S. codes are rigidly fixed and do not vary with a rope-life parameter, as some other codes do. U.S. practice requires that the winding-drum barrel diameter be no less than 18 rope diameters. While the ratio for the upper block is also 18 minimum, the lower-block ratio may not be less than 16. These ratios apply to the load-hoisting systems of construction cranes and derricks. Overhead and industrial crane practice is more conservative.

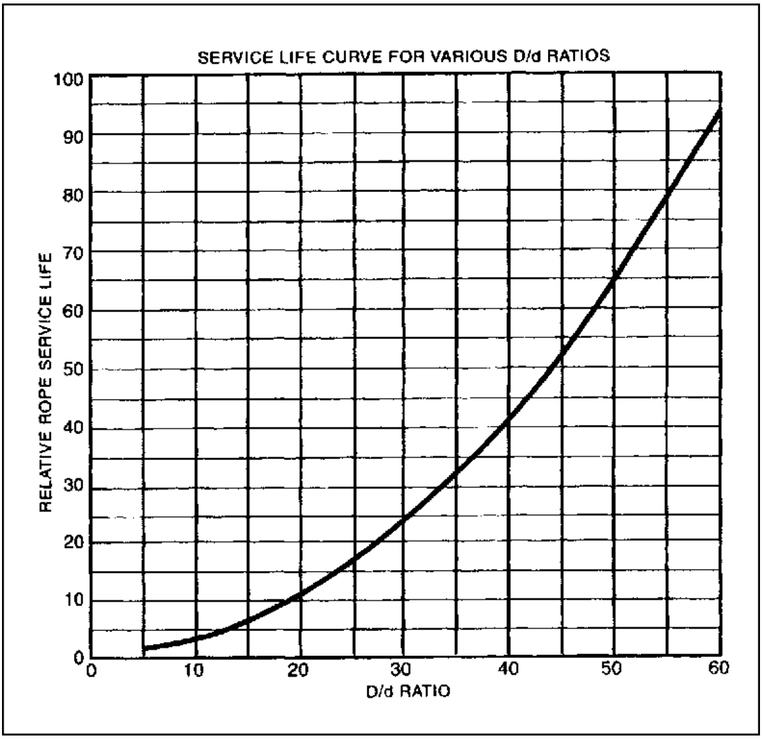


Figure 1.6 This generic curve relates relative rope service life to the ratio of sheave to rope diameter (*D/d*) considering only bending and tensile stresses during use. The scale is nominally the percentage of the life of a straight rope undergoing the same loading without passing over sheaves. The table of relative bending life factors on the next page permits adjustment of the curve values to different styles of rope. (*Wire Rope Technical Board Users Manual*, 3d ed., used with permission.)

A sheave is described by the rope diameter for which it is grooved and by four other diameters: the outside diameter of its flanges; the diameter to the base of the groove, or *tread diameter;* its shaft diameter; and its *pitch diameter*. Ratios of sheave-to-rope diameter are given using the pitch diameter, which is the diameter to the center of the rope on the sheave, in other words, the tread-plus-rope diameter. These sheave-to-rope diameter ratios are often referred to as D/d ratios.

The weight of a sheave increases with its size, as do the weight and size of its mountings. Adding dead weight at the lift point reduces the lifting capacity directly. Crane designers are thus motivated to use the smallest D/d ratios permitted by code, though the compulsion to reduce sheave size is lessened when sheaves are fabricated from lightweight synthetic materials. Because properly conducted regular inspections should prevent actual rope breakages, the designer's choice of D/d ratio should affect costs and performance but not safety. Rapid wear can force ropes to be replaced in the field where labor costs are higher and production is lost, instead of replacing them in the shop.

Rope Construction	Factor	Rope Construction	Factor
6 × 7 or 7 × 7 aircraft	.60	7 × 25 FW	1.15
19 × 7 or 18 × 7 R.R.	.70	6 × 29 FW	
6 × 19 S	.80	6 × 36 WS	
6 × 19 W	.90	6 × 36 SFW	
6 × 21 FW		6 × 43 FWS	
6 × 26 WS		7 × 31 WS	
6 × 25B FS		8 × 25 FW	1.25
6 × 27H FS		6 × 41 WS	
6 × 30G FS		6 × 41 SFW	
6 × 31V FS		6 × 49 SWS	
7 × 21F W	1.00	7 × 36 FW	
6 × 25 FW		6 × 46 SFW	1.35
6 × 31 WS		6 × 46 WS	
8 × 19S		8 × 36 WS	
8 × 21 FW	1.10	6 × 61 FWS	
		6 × 57 SFWS	

*This table, with some modification, is based on outer wire diameter relationships.

Relative Bending Life Factors*

Figure 1.6 shows that for the common range of D/d ratios, from 15 to 30, the plot is nearly linear. This gives rise to an approximate relationship between service lives at any two ratios of

$$L_r = \frac{r_2 - 10}{r_1 - 10} \tag{1.7}$$

where r_1 and r_2 are the D/d ratios being compared and L_r is the ratio of the relative rope lives. As an example, if a machine with a sheave ratio of 18 has given satisfactory service and rope life in single-shift operation, what ratio would provide similar rope life (calendar) for two-shift operation? Solving Eq. (1.7) with $L_r = 2$ and $r_1 = 18$ gives a required ratio of 26. In a similar manner, three-shift operation would suggest a ratio of 34.

Every time a rope bends around a sheave, there is an episode of added stress for the rope. When the bending direction reverses in adjacent sheaves, the stress also reverses and the service life is potentially reduced by fatigue. The smaller the sheave diameter, the more severe is this effect. Since deterioration due to fatigue or abrasive wear determines rope life, they must be factored into any rational scheme of rope and sheave selection. The science involved is almost entirely empirical.

Sheave blocks are manufactured in several styles to satisfy the varying needs of hoisting service. In a general sense, they can be classified as being of oval or diamond pattern or of the *snatch-block* type, also known as a *gate block*, as illustrated in Figure 1.7. Blocks can be provided with fixed or swiveling single or double hooks as well as with fixed or swiveling shackles or with bails. Snatch blocks have the advantage of permitting reeving when the end of the rope is not free.

Blocks or any other sheave-mounting arrangement should be provided with guards to prevent the rope from leaving the sheave groove. The simplest form of guard is a pin or bolt placed just clear of the edge of the sheave flange. Where appropriate, a pipe spacer on the bolt will keep the side plates at constant separation, allowing the bolt to be torqued to prevent unwanted loosening.

A diagram like Figure 1.8 showing the arrangement of ropes in a hoisting system is called a *reeving diagram*. There are two ways to reeve a set of two blocks, so that parts of line are either equal to the number of sheaves or greater by one. This depends on which block the fixed end, or *dead end*, of the rope is fastened to. For blocks with many sheaves, more complicated reeving arrangements are often used in an attempt to balance the friction loads. When more sheaves are present than are needed for the number of parts of line, the reeving pattern should be balanced so that the block will hang plumb and ropes will lead as straight as possible out of the sheave grooves.

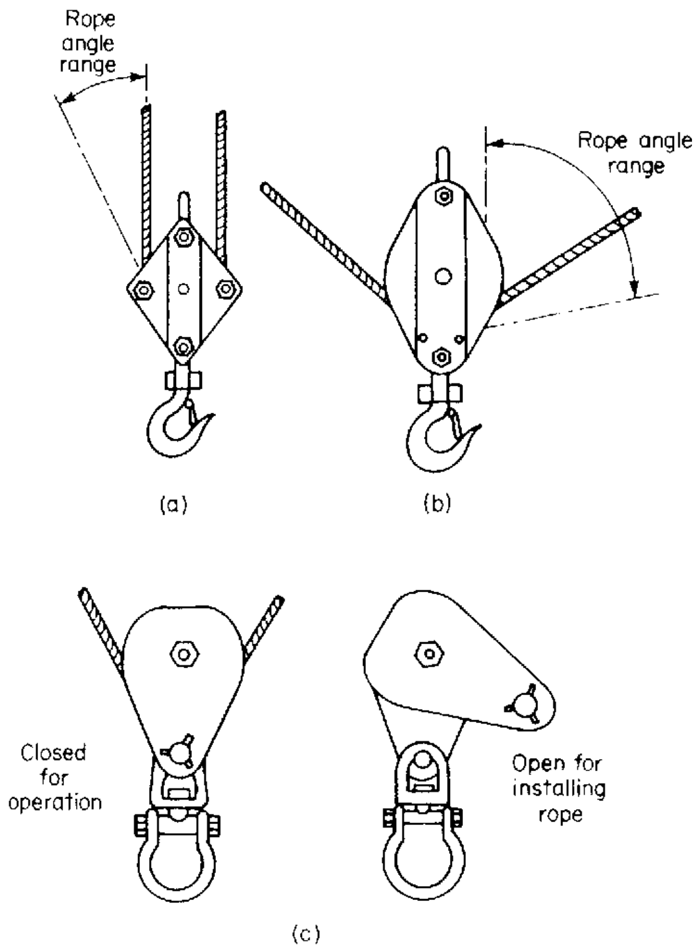


FIGURE 1.7 Principal block styles. (a) Diamond pattern. (b) Oval pattern. (c) Snatch blocks. The rope angle ranges shown reflect the degree of confinement of the rope in the block and do not mean to imply that hoisting should be done under those conditions.

1.4 Wire Rope and Fittings

Without wire rope, there would be few cranes or derricks. Traditional fiber rope is so limited in strength that it is generally used only for unpowered applications; and chains are both awkward and heavy.[5] The cold-drawn wire used for wire-rope construction has a tensile

[5]At the time of writing this, high performance synthetics had not made inroads into the marketplace for crane hoist ropes. The potential advantages of reduced weight and improved service life are outweighed by problems such as elongation as well as degradation from heat and ultraviolet light. Inevitably innovators will overcome these problems and high strength synthetics will find their way into regular use.

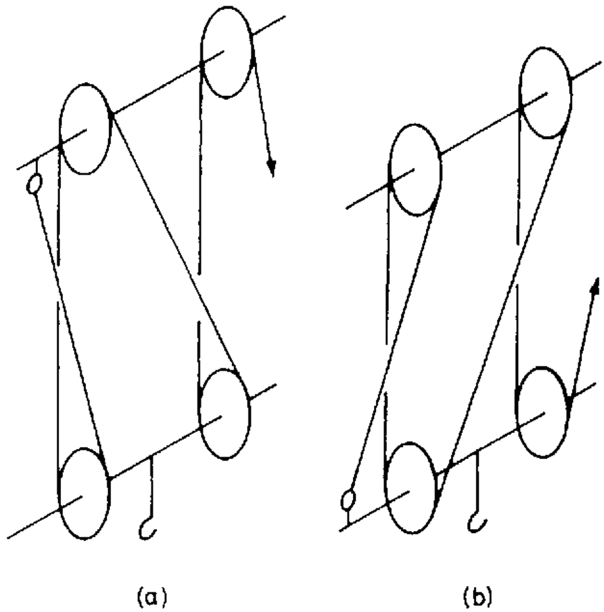


FIGURE 1.8 Reeving diagrams. (a) Four parts of line. (b) Five parts of line.

strength ranging from about 225,000 to 340,000 lb/in^2 (1550 to 2350 MPa); this gives wire ropes outstanding strength-to-weight ratios and makes possible the array of lifting equipment now available.

In constructing wire rope, individual wires are laid, not twisted, together into *strands* and the strands in turn are laid over a core to form the rope. The number of wires in a strand, the number of strands in a rope, and the nature of the core vary. Ropes are categorized by classes, such as 6 × 19, which give the number of strands to the rope and the nominal number of wires in the strand. The class designations have a basis in tradition rather than in absolute fact, and a 6 × 19 class rope may have anywhere from 15 to 26 wires in its strands. A particular 6 × 19 class rope is a 6 × 25 rope, which has 25 wires in each of its 6 strands. Figure 1.9 shows a few of the many styles of rope available.

Rope cores of several types are available, namely, fiber core (FC), wire strand core (WSC), and wire rope (independent wire-rope core, IWRC). The core acts to support the strands, holding them in position, and the wire cores add strength to the rope. Wire cores reduce flexibility; however, they generally increase resistance to crushing and bending fatigue.

Wire ropes are manufactured in several grades: improved plow steel (IPS), extra-improved plow steel (EIPS or XIPS), and now extra-extra-improved plough steel (EEIPS). The older plow steel is rarely used today. Rope strengths are given in catalog listings for the size, construction, and material of the rope; they are listed in terms of

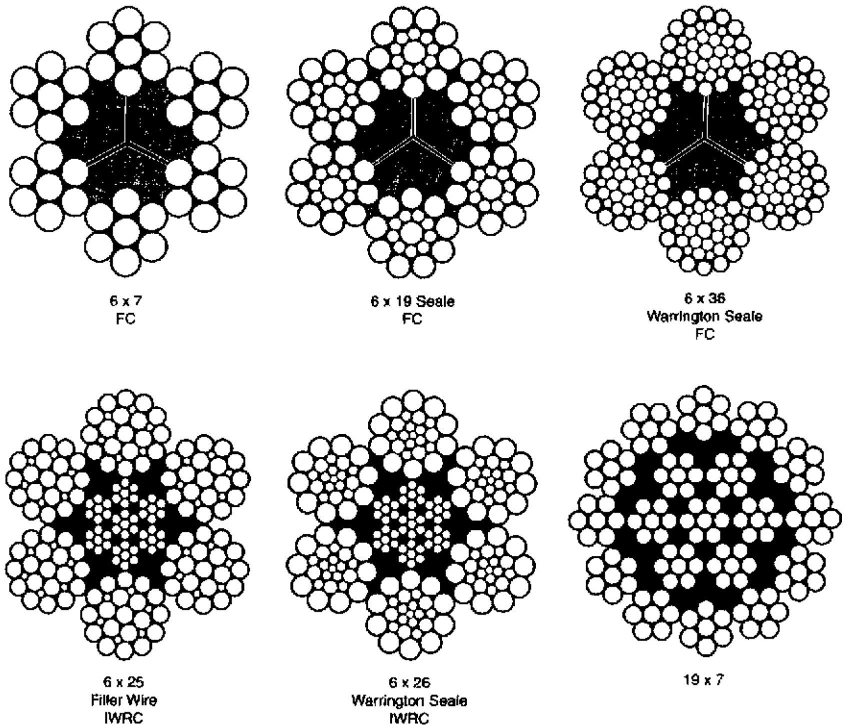


Figure 1.9 A few of the many wire rope constructions. (*Wire Rope Technical Board Users Manual*, 3d ed., used with permission.)

minimum breaking strength. The safety factors, or *strength factors*, used with wire rope vary with rope application and will be covered later in this section.

For ordinary rope design situations, only four rope properties are of importance:

Strength. Controlled by size, grade, construction, and core

Flexibility and fatigue resistance. Improved by strands with a large number of small wires and by preforming

Abrasion resistance. Enhanced by large outer wires or by *Lang lay* construction

Crushing resistance. Improved with IWRC or WSC, large outer wires and regular-lay rope

The *lay* of the rope refers to the direction of rotation of the wires and the strands. Regular-lay (right or left) ropes are made with the wires in the strands laid in one direction and the strands laid in the opposite direction, so that individual wires have the appearance of running parallel to the long axis of the rope. Regular-lay ropes are easy to handle, resist kinking and twisting, and are stable. In ropes with Lang lay, wires and strands are laid in the same direction and individual wires seem to run diagonal to the long axis of the rope, offering greater surface exposure and hence greater abrasion resistance. They are more susceptible to untwisting and kinking than regular-lay ropes and have less crushing resistance; moreover these ropes are unsuited to applications where one end of the rope is free to rotate as would occur with a swivel installed. But ropes with Lang lay are more flexible and fatigue-resistant than regular lay.

In *preformed wire rope*, each strand is shaped to its ultimate helical form just before being laid into the rope and the individual wires are also shaped by the process. This process produces a stable rope which will not only unwind when a cut end is left free but also return to its original form after unwinding under load, and is less likely to kink or foul. Preformed rope also has improved fatigue resistance. Most wire rope is now preformed.

The arrangement of the wires in a strand contributes to the properties of the rope. The common arrangements are

Simple. All wires of the same size

Seale. Large wires on the outside for abrasion resistance and small wires inside to increase flexibility

Warrington. Alternate wires large and small to combine flexibility with abrasion resistance

Filler wire. Very small wires placed in the spaces between the wires in the inner and outer layers of wire for increased fatigue resistance

A complete specification for a particular wire rope construction could read 6 × 25 *filler wire, preformed, IPS IWRC, right-regular-lay rope*; if not designated, right regular lay is assumed.

For derrick or general hoisting use, 6 × 19 class ropes seem to be most popular, as they offer a good balance between abrasion resistance and flexibility. Cranes employ 6 × 19 as well as 6 × 36, 8 × 19, and other rope styles to meet particular requirements.

Drums are called *over-wound* or *under-wound* depending on whether the rope spools onto the drum from the top or the bottom. When an over-wound drum is seen from the rope side, if the rope attachment is at the right side, a right-lay rope is required for proper spooling; left lay is needed when the attachment is at the left. The opposite is true for under-wound drums seen from the rope side.

The natural state of ropes, when loaded, is such that the strands seek to abandon their original helical shape for an elongated helix, and the rope attempts to unwind if the ends are unrestrained. The rope will stretch under initial loading until each wire is in line contact with adjacent wires. This elongation is called *constructional stretch*. Elastic strain will cause further elongation. If the rope ends are unrestrained, some unwinding, or spin, will occur, marked by end rotation and additional elongation.

The free end of the rope can be restrained by clamping against rotation. In resisting end rotation, the clamp must develop a torque reaction. If a load were to be freely supported by a single rope, the clamp would transmit the torque to the load and the load would spin unless the load itself were prevented from rotating. *Taglines* (usually manila ropes connected to the load and controlled by workers on the ground) are used to prevent load spin. (They also orient and steady the load.) A somewhat controversial alternative to restraining rotation is the use of a swivel to allow it. Some authorities maintain that the unwinding of the rope that takes place when a swivel is used leads to significant rope weakening and may hasten the onset of fatigue failure or–worse—may cause a core failure that would be difficult to detect during a routine inspection.

When an 8 × 19 rope is constructed with an IWRC so that the strands and IWRC are laid in opposite directions, the rope has spin-resistant characteristics at some levels of loading and reduced spin at others. Another construction, 19 × 7, has similar characteristics when it is made with the inner and outer strands laid in opposite directions. The tendency of one layer of strands to rotate one way is counteracted by the tendency of the other layer to rotate the other way, and a measure of torque balance is maintained. These *ropes*, and some other styles similar to them, are called *rotation-resistant ropes*. They have become prevalent for single-part crane hoist use.

More recently, *compacted wire ropes* have become a common solution for demanding applications. These have multilayer constructions with good rotation resistance; the wires in some or all of the strands are deformed so that the strands are smoother and more densely packed. Ropes with compacted wires also exhibit enhanced strength compared to conventional ropes of the same diameter; they are also well suited for withstanding crushing in multilayer spooling, abrasion, and fatigue.

A spinning load is likely to be an uncontrolled and dangerous load. The advantage in using rotation-resistant ropes, therefore, is that under many jobsite conditions the load will be stable and a tagline will not be necessary. The time that would otherwise be required to connect, control, and remove taglines is instead applied to placing additional loads.

The disadvantages of rotation-resistant ropes are threefold: (1) they are more sensitive to damage from mishandling than standard ropes, (2) their torque balance generally occurs at low levels of loading, about one-eighth of breaking strength, and (3) in the normal working range of load they can spin, the amount of spin increasing with loading.

As a rotation-resistant rope rotates, its core becomes overpowered by the outer strands and twists tighter; concurrently, the outer strands unwind. The tightening of the core tends to shorten the rope while the unwinding outer strands tend to lengthen it; this causes the load on the rope to redistribute. The core becomes burdened with a disproportionate share of the load, which creates a tendency toward internal damage that is difficult to recognize during ordinary visual inspections.

Special rotation-resistant ropes have been developed with proprietary designs that render them truly torque balanced. Under the full range of loading ratios experienced in practice, these ropes spin little or not at all. Without rotation, there is no load redistribution between the outer and inner portions of the rope; all wires are equally loaded. These ropes, commonly in the 35 × 7 classification, have compacted strands and benefit from all of the attendant advantages.

Working Loads

U.S practice has been simply to divide the rope-breaking strength by a safety factor or design factor and use the result as the working load. American National Standards Institute (ANSI) codes for cranes and derricks generally require a design factor of 3.5 to 5.0 for *running lines*, those which travel over drums or sheaves; 3.0 for *standing lines*; or guys, and 5.0 for rotation-resistant ropes, which must only be used as running lines. Derrick practice, as evidenced by some handbooks, has been to use a factor of 5 or more. High design factors have historical precedent and reflect a time when load weights were not easily determined, material properties were less closely controlled, and rigging design was done mostly by rule of thumb. On the other hand, high design factors on running ropes can improve service life. Contemporary design factors are a compromise balancing service life against the weight and cost penalties engendered by larger ropes. When applying the stipulated design factors, the common U.S. practice is to use dead and live loads without modifying them to include the effects of sheave friction or *dynamic loading* (impact). Needless to say, the common design practice might at times be superseded by engineering judgment, as there are design conditions where the combined friction and impact can be significant and demand consideration.

In European practice, the basis of rope selection is not limited to maximum loading but also includes factors of the intensity of use the rope would be expected to undergo and the number of sheaves the rope will have to pass over. Where the arrangement of the reeving is fixed and reliable loading predictions can be made, this system is rational and can be very effective in optimizing rope selection.

Design factors for *slings* are generally set higher than for running ropes and standing ropes on cranes. Since slings are often used repeatedly under particularly severe and loosely controlled conditions, provision is made for their inevitable deterioration and for potential misuse. National and international regulations such as those from OSHA and the European Union (EU) specify sling capacities in various configurations and conditions of use. Actual load capacity values can be found in rigging handbooks and vendor literature.

Fittings

The fittings used to attach wire ropes to structural connections and to each other are of several kinds and serve different purposes. Of importance here are the general types of fittings and their strength. Sometimes the strength is expressed as a percentage of the rope strength, termed *efficiency*.

Rope-end fittings provide the rope end with a loop, eye, pinhole, or hook for attachment. Permanent fittings, which require cutting the rope for removal, are the most efficient, often matching the strength of the rope. The exception is hand-spliced loops, which have about 70% to 75% of the rope strength in the commonly used rope sizes. The removable fitting types, *wedge sockets*, clip and thimble loops, and variations, also develop about 80% of rope strength. Figure 1.10 illustrates some

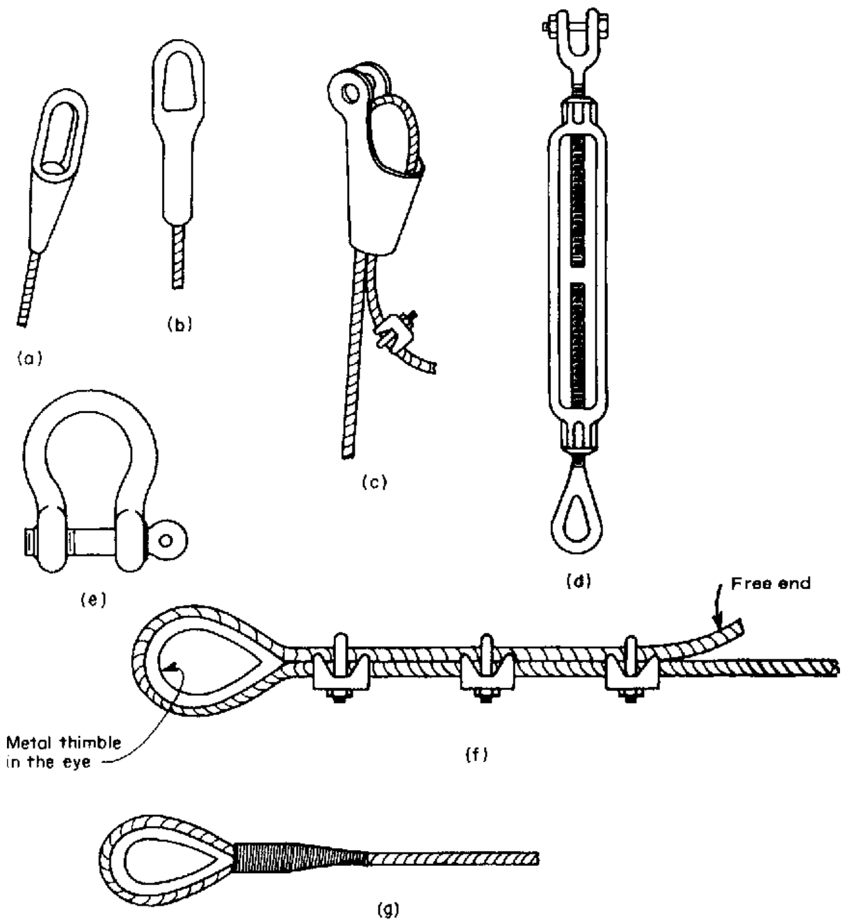


Figure 1.10 Some common wire-rope fittings. (a) Spelter (zinced) socket (efficiency 100%). (b) Swedged socket (efficiency 100%). (c) Wedge socket (efficiency about 80%). (d) Turnbuckle. (e) Shackle. (f) Crosby-type clips and thimble (efficiency about 80%). (g) Hand-spliced eye and thimble (efficiency about 80%).

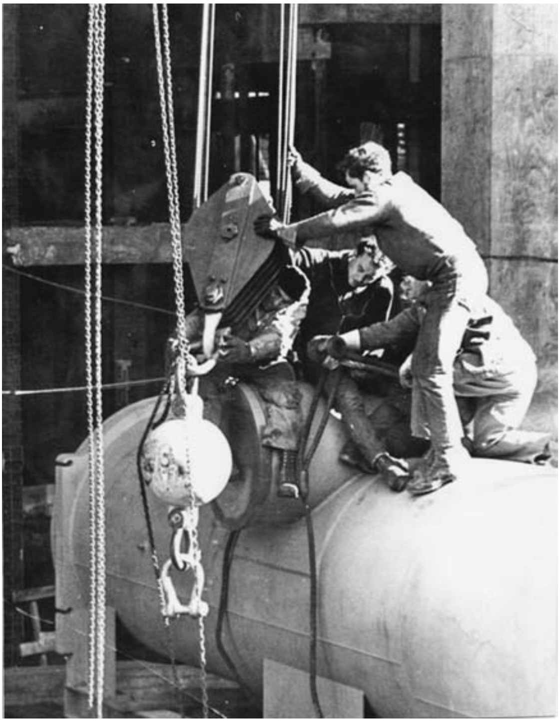


Figure 1.11 Riggers slinging a load in preparation for lifting. Note the overhaul ball and fittings. This photograph was taken before OSHA required the use of hard hats and fall-protection measures. (*Lawrence K. Shapiro.*)

common rope fittings. More detailed information can be found in a wire-rope handbook (see also Figure 1.11).[6]

Permanent rope fittings should be used wherever feasible because of their superior strength and reliability. There are, however, many purposes for which permanent fittings are impractical, the dead end attachment of the hoist rope being an obvious example. In design practice, the efficiency of the rope end fittings is commonly ignored. Common or not, this practice should be used judiciously.

Shackles are used to connect a rope-end loop or eye at a structural pinhole, lug, or *bail*. To accommodate varied conditions of use, the mouth opening is made quite wide compared with the thickness of a typical connecting plate. Often the plate must be built up or the shackle packed out with washers to create a reliable concentric connection.

Turnbuckles are used to remove slack from standing ropes such as *guy lines* or *pendants*. They are available with hook, loop, or jaw (shackle) end fittings.

[6]See pp. 33–34 of *Wire Rope Users Manual,* 3d ed., Wire Rope Technical Board, Woodstock, Md., 1993.

1.5 The Basic Luffing Mechanism

The word *luffing*, like so many crane and derrick terms, has its origin in nautical usage from the days of sailing ships. Booms were rigged to handle cargo as well as sails, and the technologies of hoisting and seamanship developed together.

As used in crane practice, *luffing* means changing the angle that the main load-supporting member makes with the horizontal. Other names used for this same motion include *topping*, *derricking*, and *booming*. Henceforth, these terms will be used interchangeably.

Raising and lowering loads with the hoisting mechanism have been covered previously. Derricking also raises and lowers loads to a limited extent, but mainly this motion serves to move loads horizontally. Together with a *swing*, or horizontal rotating motion, luffing enables a hoisting device to move loads within all or part of a vertical cylindrical zone. The photographs in Figures 1.12 and 1.13 show the two ends of a luffing system that one of the authors designed for a stiffleg derrick. Note the arrangement of pairs of blocks to allow for use of 13 parts of line.

The derricking lines carry a portion of the dead weight of the *boom* strut, and they suffer an increase in loading whenever a hook load is lifted. Nonetheless, as a rule, the luffing-system ropes experience less severe service than do hoisting-system ropes. Luffing is a relatively slow operation. A well-planned crane or derrick installation should be arranged to keep luffing movements at a minimum for the

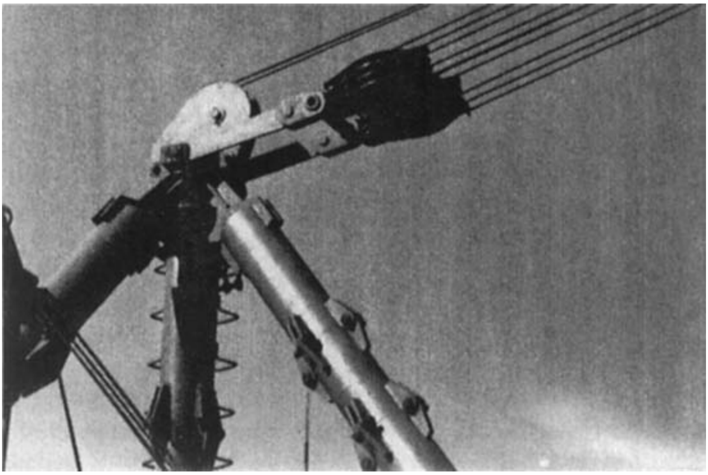


Figure 1.12 The mast top end of a stiffleg derrick luffing system. The two top lines are the load hoist and derricking lead lines; they run down the center of the mast over deflector sheaves to the winch. (*Lawrence K. Shapiro.*)

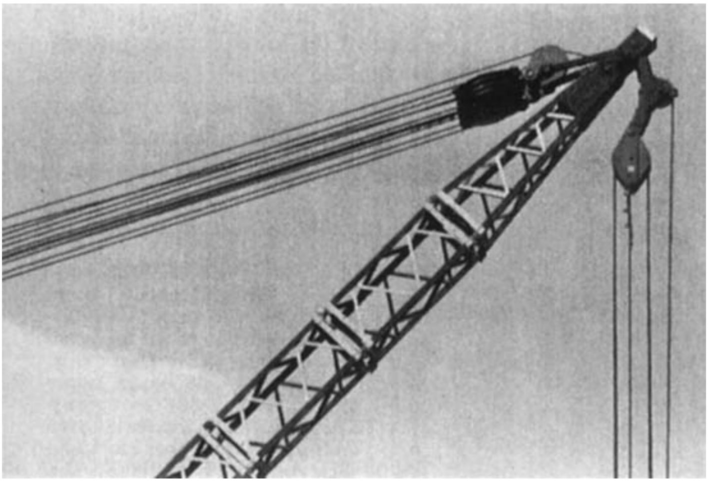


Figure 1.13 The *boom* tip end of the luffing system of Figure 1.14. Note the load lead line coming up at the right. It passes over a deflector sheave and through the *boom* tip. It then passes over the vertical sheave mounted on the derricking block and goes back to the mast. (*Lawrence K. Shapiro.*)

sake of production efficiency. This implies that luffing ropes will undergo most of their loading sequences while static, a condition that ropes are far better able to endure than loaded passage over sheaves. For this reason, the ANSI codes specify minimum D/d ratios for luffing sheaves and drums of only 15.

Force in the luffing ropes is determined by moment equilibrium. Referring to Figure 1.14, if W is taken as the weight of the load and lower load block and if the strut is assumed to have negligible weight, the moment about the strut bottom pivot is

$$M = WL\cos\theta = WR$$

where θ is the angle the strut makes with the horizontal. In order to support that moment, the luffing ropes must develop the horizontal reaction

$$T_h = \frac{M}{h} = \frac{WL\cos\theta}{h} = \frac{WR}{h}$$

so that the load in the luffing ropes is given by

$$T = \frac{T_h}{R}\left[R^2 + (h_t - h)^2\right]^{1/2} = \frac{M}{Rh}\left[R^2 + (h_t - h)^2\right]^{1/2} \tag{1.8}$$

As with the hoisting mechanism, a static *topping lift* loads each part of line equally. When derricking starts, friction is introduced at each sheave and the resulting losses can be taken as having the values given in Sec. 1.2.

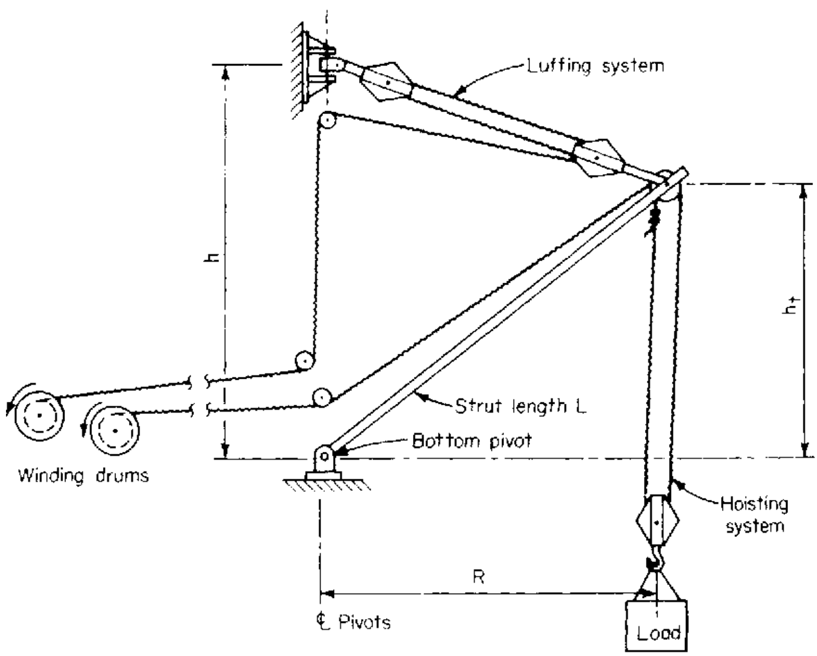


Figure 1.14 Basic derrick arrangement.

Lacking an overhaul weight, an unloaded boom strut might lack sufficient weight to overcome resistance; it may fail to lower at the will of the operator. The overhaul weight added to draw the *load line* down also compels the boom or strut to derrick out. The forces of resistance are similar to those retarding the *load hoist system*, deriving from friction at the sheaves and drum combined with the weight of the vertical portion of the lead rope.

The overhaul weight calculated for the load hoist system will often be adequate for derricking as well. Referring to Figure 1.14, if the height to the top pivot h is less than the strut length L, the weight is likely to be governed by the overhaul demand of the load hoist system. As the topping height h increases, the force in the derricking lines reduces; this enhances the likelihood that derricking will govern the demand for overhaul weight.

Friction must be taken into account for more precise calculations. If the line to the first luffing deflector sheave carries a force P_h, an equilibrium state must be satisfied such that

$$T = P_h + (1-\mu)P_h + (1-\mu)^2 P_h + \cdots + (1-\mu)^{n-1} P_h$$

or conversely

$$P_h = \frac{T}{1 + (1-\mu) + (1-\mu)^2 + \cdots + (1-\mu)^{n-1}}$$

where the angle the line to the deflector sheave makes with the derricking lines has been neglected, sheave friction loss is μ, and the number of parts of line supporting the strut is n. Adding the effect of the losses at the deflector sheaves, we find P, the force at the drum required to initiate luffing in Figure 1.14, to be

$$P = \frac{P_h}{(1-\mu)} = \frac{T}{(1-\mu)^1 + (1-\mu)^2 + \cdots + (1-\mu)^n}$$

For luffing out, equilibrium requirements will show that

$$P = \frac{T}{(1-\mu)^{-1} + (1-\mu)^{-2} + \cdots + (1-\mu)^{-n}}$$

The equations governing these operations can be generalized. If n is taken to include the line to the deflector sheave and m is the number of 180° turns over sheaves between the upper luffing block and the drum, then the luffing line, or the boom hoist line, load at the drum is

$$P = \frac{T}{r} \tag{1.9}$$

where for luffing in

$$\begin{aligned} r &= (1-\mu)^m + (1-\mu)^{m+1} + \cdots + (1-\mu)^{m+n+1} \\ &= \frac{(1-\mu)^m - (1-\mu)^{m+n}}{\mu} \end{aligned}$$

and for luffing out

$$r = \frac{1}{(1-\mu)^m} + \frac{1}{(1-\mu)^{m+1}} + \cdots + \frac{1}{(1-\mu)^{m+n-1}} = \frac{1-(1-\mu)^n}{\mu(1-\mu)^{m+n-1}}$$

Although in the derivation for Eq. (1.8) the weight of the strut was ignored for simplification, in practice strut weight is significant and must be included when determining the moment M.

Example 1.2 Assume that the load-hoist-system design required a block plus overhaul weight of 600 lb (272.2 kg) with the following data:

Strut	**Luffing System**
Weight 240 lb (108.9 kg)	Distance $h = 12$ ft (3.66 m)
Length 20 ft (6.1 m)	Minimum $R = 3.5$ ft (1.07 m) (h_t is therefore 19.7 ft = 6.0 m)
CG located 11 ft (3.35 m) from bottom pivot, or at 0.55L	$n = 5$ parts of line
	$m = 2$
	$\mu = 0.045$ and $1-\mu = 0.955$
	Drum friction 50 lb (222.4 N)
	Weight of vertical part of *boom hoist line* = 8 lb (3.63 kg)

1. What is the static load in the luffing ropes? Will the strut be able to derrick out with no load on the hook?

Solution

Taking moments about the strut bottom pivot at $R = 3.5$ ft (1.07 m), we have

$$M = 600(3.5) + 240(0.55)(3.5) = 2562 \text{ lb} \cdot \text{ft } (3.47 \text{ kN} \cdot \text{m})$$

The load in the luffing system is found using Eq. (1.8).

$$T = \frac{2562}{3.5(12)}[3.5^2 + (19.7 - 12)^2]^{1/2} = 516 \text{ lb } (2.30 \text{ kN})$$

and from Eq. (1.9) for luffing out,

$$r = \frac{1 - 0.955^5}{0.045(0.955)^6} = 6.024$$

$$P = \frac{516}{6.024} = 86 \text{ lb } (381 \text{ N})$$

But this does not reflect the weight of the vertical lead rope, so the net force at the drum is

$$P_{\text{net}} = 86 - \frac{8}{0.955} = 78 \text{lb } (345 \text{ N})$$

which is greater than the friction at the drum. Therefore, the *boom* strut will derrick out.

2. With the topping height h increased to 25 ft (7.62 m), will the *boom* strut still be able to derrick out?

Solution

The moment remains the same, but

$$T = \frac{2562}{3.5(25)}[3.5^2 + (19.7 - 25)^2]^{1/2} = 186 \text{ lb } (827 \text{ N})$$

so that

$$P = \frac{186}{6.024} = 31 \text{ lb } (138 \text{ N})$$

and

$$P_{\text{net}} = 31 - \frac{8}{0.955} = 22.6 \text{ lb } (101 \text{ N})$$

which is less than the 50 lb (222.4 N) of resistance at the drum. The boom strut will not derrick out. More overhaul weight is needed; this demonstrates what happens when the topping height h *is* increased. The required weight can be found by working backward through the same equations.

$$\text{Required } T = \left(50 + \frac{8}{0.955}\right)6.024 = 351.7 \text{ lb } (1.56 \text{ kN})$$

$$\text{Required } M = 351.7(3.5)\frac{25}{6.35} = 4846 \text{ lb} \cdot \text{ft } (6.57 \text{ kN} \cdot \text{m})$$

$$\text{Required } W = \frac{4846 - 240(0.55)(3.5)}{3.5} = 1253 \text{ lb } (568 \text{ kg})$$

An overhaul weight in excess of 1253 lb (568 kg) is needed. Had the 8-lb (3.63-kg) lead rope been neglected, calculations would indicate an overhaul weight of 1054 lb (478 kg). This illustrates the extreme sensitivity of the overhaul problem and implies that an overhaul weight amply in excess of the calculated value be used in practice.

Example 1.3 Given the same data as in the previous example with $h = 25$ ft (7.62 m), if the strut is required to carry a load of 22,000 lb (9979 kg) at the hook in addition to an overhauling weight (including the block) of 1500 lb (680 kg), what will be the load in the derricking ropes and how much line pull must the derricking winch provide for booming in? Take 20 ft (6.10 m) as the R distance.

Solution

The moment about the strut bottom pivot becomes

$$M = (22{,}000 + 1500)(20) + 240(0.55)(20)$$

$$= 472{,}640 \text{ lb}\cdot\text{ft } (640.8 \text{kN}\cdot\text{m})$$

$$h_t = 0$$

and the load in the luffing system is then

$$T = \frac{472{,}640}{20(25)}[20^2 + (0 - 25)^2]^{1/2} = 30{,}265 \text{ lb } (134.6 \text{ kN})$$

The *line pull* needed at the drum is found from Eq. (1.9).

$$r = \frac{0.955^2 - 0.955^7}{0.045} = 4.168$$

$$P = \frac{30{,}265}{4.168} = 7260 \text{ lb } (32.30 \text{ kN})$$

As a running line, the rope used for luffing is stipulated in U.S. practice to have a minimum strength margin of 3.5. The minimum permitted rope-breaking strength is therefore

$$\text{BS}_{\text{min}} = \frac{30{,}265}{5}\left(\frac{3.5}{2000}\right) = 10.6 \text{ tons } (94.23 \text{ kN})$$

which can be satisfied by a ½ -in-diameter (12.7-mm), 6 × 19 class rope of IPS or better, but the winch must have enough available torque to deliver a *line pull* of 7260 lb (32.30 kN) with the quantity of rope that will be on the drum at $R = 20$ ft (6.10 m). Will line pull be adequate for the following data?

From Figure 1.5,

$A = 27$ in (686 mm) $B = 23$ in (584 mm) $D = 11$ in (279 mm)

First-layer *line pull* = 9000 lb (40.0 kN)

Spooling factor for ½ -in rope $s = 0.925$ (SI: $s = 1.77 \times 10^{-5}$)

Initially 2000 ft (610 m) of rope was stored on the drum, and luffing-system dimensions are given in Figure 1.15.

Assuming that the sheaves are of minimum D/d, that is 15, the pitch diameter will be 15(½) = 7½ in (190 mm). With the strut at R = 3.5 ft (1.07 m), the total length of rope spooled out is

$$L_r = 65 + 8 + 4(6) + 4(\tfrac{1}{2})(\pi)\frac{7.5}{12} = 101 \text{ ft } (30.8 \text{ m})$$

The luffing-system lengths at $R = 3.5$ ft (1.07 m) and $R = 20$ ft (6.10 m) are found from the geometry to be

$$L_{3.5} = [3.5^2 + (25 - 19.7)^2]^{1/2} = 6.35 \text{ ft } (1.94 \text{ m})$$

$$L_{20} = [20^2 + (25 - 0)^2]^{1/2} = 32.02 \text{ ft } (9.76 \text{ m})$$

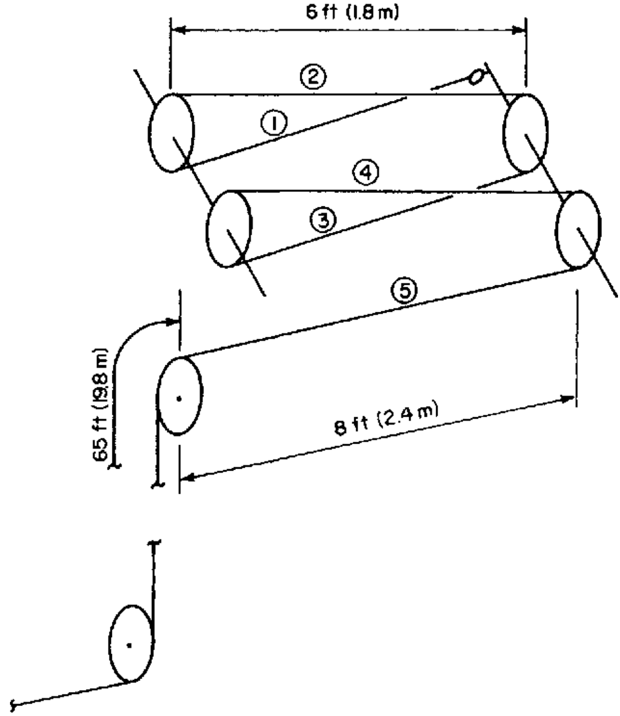


FIGURE 1.15 Topping-lift reeving and dimensions for Example 1.3.

When the strut is derricked out from $R = 3.5$ to $R = 20$, each part of line must increase in length by 32.02 – 6.35, so that the total quantity of rope spooled out at $R = 20$ is

$$L_r = 101 + 5(32.02 - 6.35) = 229 \text{ ft } (69.80 \text{ m})$$

With 2000 ft (610 m) of rope on the drum initially and 229 ft (69.80 m) spooled out, Eq. (1.2) can be used to determine the radius at which the lead line will act at the drum so that usable line pull can be checked.

$$2000 - 229 = (11 + E)E(23)(0.925)$$

$$E^2 + 11E - 83.24 = 0 \quad E = 5.15 \text{ in } (131 \text{ mm})$$

In Eq. (1.5), for usable *line pull*, the C dimension is required.

$$C = \frac{27 - 11 - 2(5.15)}{2} = 2.85 \text{ in } (72.4 \text{ mm})$$

$$P_u = \frac{9000(11 + 0.5)}{27 - 2(2.85) - 0.5}$$

$$= 4976 \text{ lb } (22.13 \text{ kN})$$

but 4976 < 7260 lb (22.13 < 32.30 kN); therefore the winch will not be capable of delivering the needed *line pull* unless excess rope is unloaded.

Check again with 500 ft (152 m) of rope initially on the drum.

$$500 - 229 = (11 + E)(E)(23)(0.925) \quad E = 1.06 \text{ in } (26.9 \text{ mm})$$

$$C = 6.94 \text{ in } (176.3 \text{ mm})$$

$$P_u = \frac{9000(11.5)}{27 - 2(6.94) - 0.5} = 8201 \text{ lb } (36.48 \text{ kN})$$

Since 8201 > 7260(36.48) > 32.30), the winch provides adequate line pull to power the luffing ropes.

Obviously it is advantageous to have a winch with enough available torque that it will not limit the available line pull. If that is not possible, the available line pull can be improved by limiting the layers of rope on the drum; alternatively, the imposed line pull can be reduced by increasing the parts of line or increasing the topping height. Figures 1.12 and 1.13 illustrate a derrick luffing system with 13 parts of line.

1.6 The Basic Derrick

The derrick illustrated in Figure 1.16 is of the most basic form, comprising a hoisting system, a luffing system, and a strut or boom. It is called a *Chicago boom* and is used by rigging contractors to hoist machinery and equipment onto high-rise buildings, by stone contractors to hoist and place exterior stone or precast facing panels, and by other trades as well. Capacities range from less than 1 ton (900 kg) to at least as high as the 35-ton (32-t) model designed by one of the authors. Chicago booms are usually installed by mounting them to

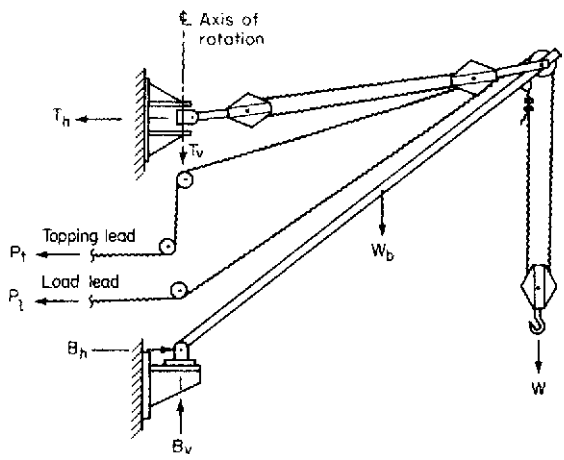


Figure 1.16 A Chicago *boom* derrick.

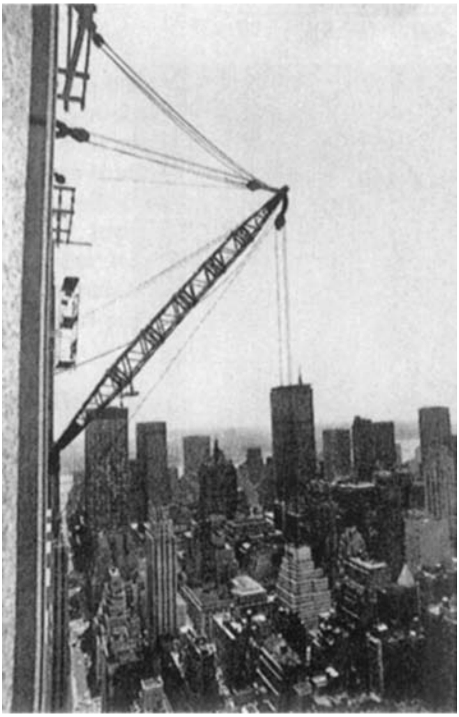


Figure 1.17 Chicago *boom* mounted at the 50th floor during construction of the McGraw-Hill Building in New York City. What appears to be an upper *topping lift* is a side (swing) guy. The load lead line is reeved over a sheave in the upper block and then it runs under the *boom*. The topping lead comes off a sheave at the *boom* tip to a deflector sheave at the building face.

building columns as in Figure 1.17. The same arrangement of derrick becomes a *guy derrick* when it is mounted to a mast and a *stiffleg derrick* when it is fixed to a frame. The various forms of derricks and how they relate to the Chicago boom will be treated more fully in Chap. 2.

Lateral Motions

In derrick practice, the luffing system is generally called a topping lift, and, as shown in Figure 1.16, the topping lift and the bottom of the strut are attached to pivots vertically in line with each other. These pivots permit the derrick to rotate, or *swing*, so that the hook traverses a path along the arc of a horizontal circle. The pivots define a vertical line called the *axis of rotation*. The horizontal distance from the axis of rotation to a plumb line through the CG of the load (hook) is called the *operating radius* (previously referred to as the *R* distance). The topping lift is used to change the operating radius.

The derrick configuration must be stable in both the vertical and horizontal planes. Vertical stability is ensured by the triangular arrangement of the boom, the topping lift, and the structure on which the derrick is mounted. Lateral stability is provided by guy lines attached to each side of the boom (strut) tip and running back to the structure.

The same lines that provide lateral stability are used to swing the boom. This is done by *taking up* on one guy line while *paying out* the other, a procedure usually done manually but sometimes by winches or hydraulic or electric devices.

Force Analysis

In Secs. 1.2 and 1.5, methods were developed for calculating the loads in the hoisting and luffing systems. The loads in the boom and on the supporting structure follow from them by simple statics. In the derrick of Figure 1.16, maximum reactions under a constant hook load follow easily discernable trends: the topping lift load P_t and horizontal reactions B_h and T_h increase with radius, the vertical reaction B_v increases as the boom is raised, and the load *hoist line pull* P_l remains constant. The vertical topping lift component T_v is the only reversible force, acting upward at high boom and downward at low boom. Table 1.2 shows the reactions in the derrick of Figure 1.18 for a constant hook load. Except for the constant *load line pull* and the axial load acting on the boom, extreme values occur at minimum and maximum radii.

Forces in the derrick and in the supporting structure are resolved by simple statics. The three conditions of static equilibrium

$$\Sigma H = 0 \qquad \Sigma V = 0 \qquad \Sigma M = 0$$

must be satisfied. A thumbnail method, suitable for preliminary analysis, ignores the angles made by the lead lines. In this method, using the information in Figure 1.18 and taking moments about the boom foot, we have

$$\Sigma M = WR + W_b(8.25) - T_h h = 0$$

$$(22{,}000 + 1500)15 + 240(8.25) - T_h(10) = 0$$

from which $T_h = 35{,}450$ lb (157.7 kN) and

$$T_v = \frac{T_h(h_t - h)}{R} = \frac{35{,}450(13.23 - 10)}{15} = 7630 \text{ lb (33.9 kN)}$$

Continuing with the static equilibrium requirements, we get

$$\Sigma H = T_h - B_h = 0 \qquad T_h = B_h = 35{,}450 \text{ lb (157.7 kN)}$$

$$\Sigma V = W + T_v - B_v = 0 \qquad B_v = (22{,}000 + 1500) + 7630$$

$$= 31{,}130 \text{ lb (138.5 kN)}$$

Operating Radius, ft*	h_t, ft	T_h, lb	T_v, lb	Topping Load, lb	P_t, lb	B_h, lb	B_v, lb	Boom Axial Load, lb
2.5	19.84	4,975	19,575	20,200	5,050	6,765	54,080	54,510
5.0	19.36	9,915	18,560	21,050	5,260	13,535	52,820	54,520
7.5	18.54	14,800	16,850	22,430	5,600	20,320	50,615	54,540
10.0	17.32	19,615	14,360	24,300	6,080	27,115	47,320	54,540
12.5	15.61	24,370	10,940	26,710	6,680	33,920	42,690	54,520
15.0	13.23	29,075	6,260	29,740	7,435	40,710	36,210	54,480
17.5	9.68	33,735	–620	33,740	8,435	47,485	26,540	54,400
20.0	0	38,290	–19,150	42,810	10,700	54,150	0	54,150

*1 ft = 0.30480 m; 1 lb = 4.4482 N.

TABLE 1.2 Chicago *Boom* Reactions for the Dimensions and Loads Given in Figure 1.18

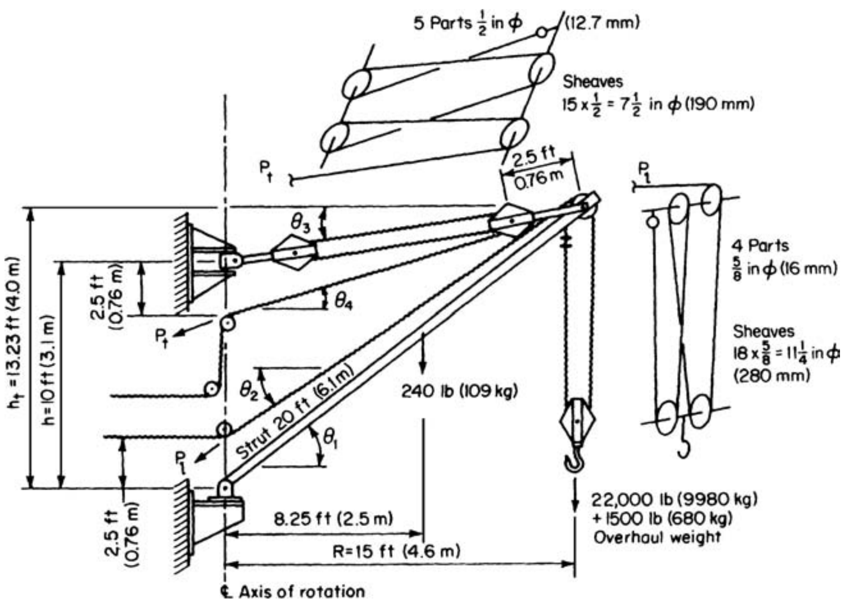


FIGURE 1.18 Derrick of Figure 1.16 dimensioned for sample calculations.

The topping-lift load is given by

$$T = \sqrt{T_h^2 + T_v^2} = \sqrt{35{,}450^2 + 7630^2} = 36{,}260 \text{ lb (161.3 kN)}$$

But four parts are in the *topping lift* and one is the lead line, so that the lead-line force is

$$P_t = \frac{T}{n} = \frac{36{,}260}{5} = 7250 \text{ lb} \quad (32.3 \text{ kN})$$

and the topping lift is left with 36,260 – 7250 = 29,010 lb (129.0 kN). The load lead-line force is found by dividing the lifted load by four parts of line, giving (22,000 + 1500)/4 = 5875 lb (26.1 kN). If the angle the lead makes with the *boom* is ignored, the lead-line force will act as an axial loading on the boom. The boom load will then be

$$B_a = P_l + \sqrt{B_h^2 + B_v^2} = 5875 + \sqrt{35{,}450^2 + 31{,}130^2}$$

$$= 5870 + 47{,}180 = 53{,}050 \text{ lb (236.0 kN)}$$

Note that all horizontal reactions are in the vertical plane defined by the boom and the topping lift. As the boom swings, the horizontal reactions rotate as well (Figures 1.17 and 1.19). Structural members supporting these reactions may then be exposed to axial forces, biaxial bending, and torsion. These effects are covered in Chap. 7.

FIGURE 1.19 A 125-ft (38-m) Chicago *boom* mounted on a steel stack at a Consolidated Edison Company plant in New York City during alterations. Mounting provisions and installation were designed by one of the authors. (*Lawrence K. Shapiro.*)

For more accurate calculations, it is necessary to evaluate the angles made by the derrick members, as indicated in Figure 1.18. The first three angles are readily found from the given dimensions.

$$\cos\theta_1 = \frac{R}{L} = \frac{15}{20} \qquad \theta_1 = 41.41°$$

$$\tan\theta_2 = \frac{h_t + 11.25/(2\times 12) - 2.5}{R} = \frac{11.20}{15} \qquad \theta_2 = 36.74°$$

$$\tan\theta_3 = \frac{h_t - h}{R} = \frac{13.23 - 10}{15} \qquad \theta_3 = 12.15°$$

θ_4 is somewhat more complex but can be expressed with satisfactory accuracy as

$$\tan\theta_4 = \frac{13.23 - 10 + 2.5 - 2.5\sin\theta_3 - 7.5/(2\times 12)}{15 - 2.5\cos\theta_3}$$

$$= \frac{4.89}{12.56}$$

$$\theta_4 = 21.28°$$

Taking moments about the boom pivot under static conditions and noting that four parts of the topping lift go to the anchorage while the fifth is the lead gives

$$(22{,}000+1500)15+240(8.25)-(\tfrac{1}{2})(22{,}000+1500)(2.5\cos 36.74)$$

$$-(10-2.5)P_t\cos 21.28-4(10P_t)\cos 12.15=0$$

from which $P_t = 7435$ lb (33.1 kN) and $T_h = 4P_t \cos 12.15 = 29{,}075$ lb (129.3 kN). From the horizontal equilibrium condition,

$$\Sigma H = B_h - 29{,}075 - 7435 \cos 21.28 - 5875 \cos 36.74 = 0$$

$$B_h = 40{,}710 \text{ lb } (181.1 \text{ kN})$$

The vertical requirement then gives

$$\Sigma V = B_v - 29{,}075 \tan 12.15 - 22{,}000 - 1500 - 240$$

$$-7435 \sin 21.28 - 5875 \sin 36.74 = 0$$

$$B_v = 36{,}210 \text{ lb } (16.1 \text{ kN})$$

and the *boom* axial load is

$$B_a = (40{,}710^2 + 36{,}210^2)^{1/2} = 54{,}480 \text{ lb } (242.3 \text{ kN})$$

In like manner, the topping lift load is

$$T = (29{,}075^2 + 6260^2)^{1/2} = 29{,}740 \text{ lb } (132.3 \text{ kN})$$

All these loadings are about 2½% greater than those found using the thumbnail method. Table 1.2 gives all member loads and reactions at several radii with the hook load remaining constant. Using the same format, the reactions induced during motion of the luffing or the hoisting systems can be determined by inserting friction losses where appropriate.

1.7 A Contemporary Crane

The basic crane described up to this point might be recognizable to a medieval cathedral builder or perhaps even to a construction laborer of ancient Rome. Materials and power systems have advanced through the Industrial Age up to the present, but the essential arrangement of drum, boom strut, and block and tackle have been fairly constant. Manual or animal power has been replaced by electricity and internal combustion, hemp by metallic rope, and wood spars with steel booms. Lifting gear that was once fixed in place is now mobile, and what was once exclusively custom-built has become modular and standardized. Contemporary cranes and derricks have branched out into many forms to perform varied industrial roles. The state of the art, however, is well represented by the example of a *telescoping cantilevered boom crane*, commonly called a *hydraulic crane* (Figure 1.20).

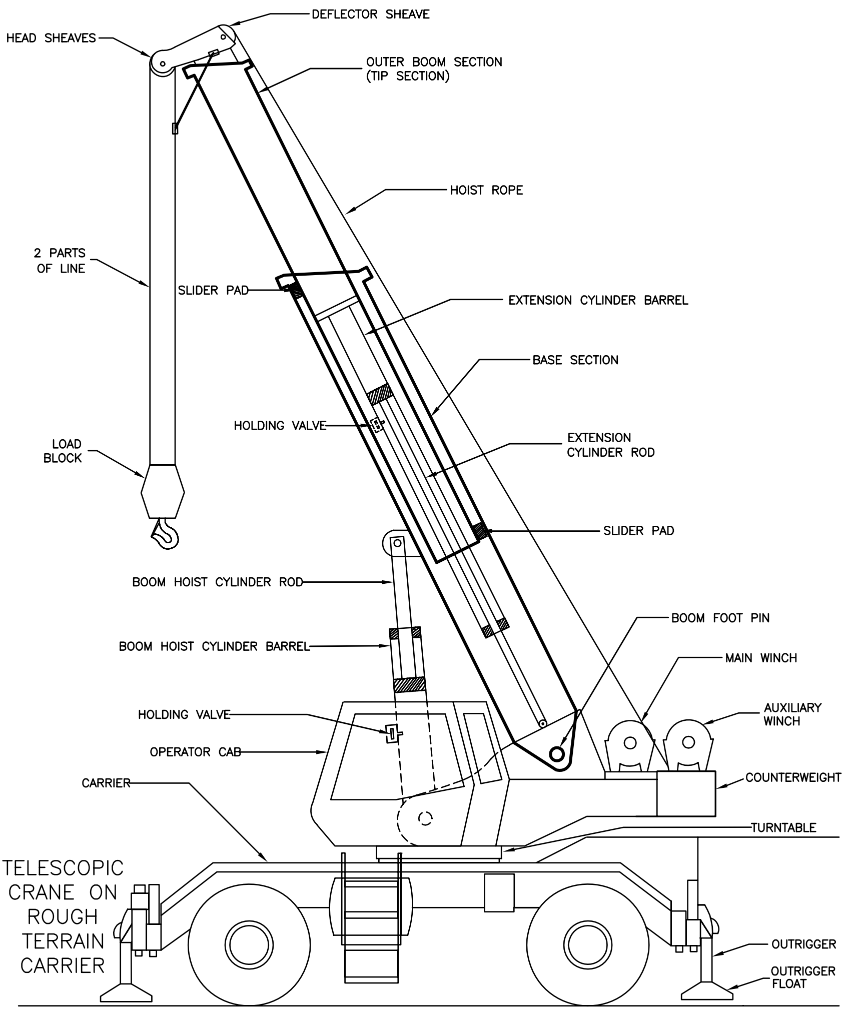


Figure 1.20 Crawler-mounted mobile crane annotated for consideration of stability.

A telescoping boom is almost always mounted on a wheeled chassis with outriggers, thereby embodying two of the key crane adaptations of the twentieth century (Figure 1.2). The boom configuration and its operating principals are themselves fundamental departures from antecedent crane types that have fixed booms and topping lifts. The cantilevered telescopic boom is made possible by advances in steel quality and in fabrication techniques as well as by the development of fluid power transmission systems.

The telescopic crane does not necessarily dispense with all of the old mechanical contrivances; some in fact use ropes and pulleys in novel ways. But its characteristic systems utilize hydraulic (hydrostatic) power transmissions for extending (*telescoping*) and elevating (*luffing*) the boom, and vice versa.[7] The boom is raised and lowered with one or a tandem pair of *hydraulic cylinders*, also called *rams* or *pistons*, acting on its base section. Telescoping of the boom sections is powered by one or more hydraulic cylinders hidden within the closed sections; some models have these pistons assisted by ropes and sheaves.

In both luffing and telescoping hydraulic power circuits, a *holding valve* is mounted on each cylinder to prevent failure in the event of a loss of pressure; it is combined with a *counterbalance valve* that equilibrates the flow of hydraulic fluid from one side of the piston to the other so that the movement of the piston is kept steady.

Though the prime mover of a mobile telescopic crane is usually a diesel engine, power is mediated through a gearbox to hydraulic pumps and motors that drives the various crane motions. A small crane model is equipped with one engine that powers both the crane and the carriage that drives it, with fluid passing through a hydraulic swivel between the crane superstructure and the carrier; a larger mobile crane has separate engines for driving the vehicle and operating the crane.

Chapter 2 describes some mobile crane types that have powerful mechanical drives and free-fall winches. These features can be compatible with a latticed boom supported by a topping lift but not readily so with a cantilevered boom. A cable-suspended latticed boom can absorb severe application of loads that would quickly damage or destroy a cantilevered boom. In keeping with the relatively delicate nature of the equipment, hydrostatic drives used on telescopic cantilevered cranes modulate the application of load and do not allow free-fall. Likewise, users of these machines should take care to avoid *duty cycle* work and other harsh service applications.

1.8 Basis for Load Ratings

A *load-rating chart* is a key document in crane and derrick practice, fraught with practical and legal significance. Though the concept is straightforward, a proper definition of a *load rating* is quite convoluted: it is the maximum allowable load for a specific radius with the crane or derrick in a particular configuration while operating under defined conditions. Parsing the elements of this definition gives a clearer understanding.

[7] *Hydraulic power* and *fluid power* are broad terms that properly include hydrostatic and hydrodynamic transmission as well as open channel flow. *Hydrostatic power* is a technically more accurate term for the transmission of force and motion by pressure and flow of fluid. Essentially a hydrostatic transmission pushes fluid to transmit force; the associated motion can be linear—delivered by a piston—or rotary through a drum or hub motor. However, the vernacular term *hydraulic power* is more commonly understood and thus will be generally used in this volume.

Starting with the last element, defined conditions of operation, consider that if the device had been designed for a particular loading in conjunction with wind at 25 mi/h (40 km/h), operation in winds of 45 mi/h (72.4 km/h) would require a reevaluation of the load-lifting capacity. Wind forces will have increased to about 3 times the design value. In like manner other factors such as speed of operation, out-of-level mounting, and irregular reeving can create loadings or variations in loadings not contemplated in the design.

In Example 1.3 the load, operating radius, and boom length correspond to the last radius in Table 1.2. Note, however, that the topping-lift load in Example 1.3 is 30,265 lb (134.6 kN) including the lead line while in the table it is 42,810 lb (190.4 kN) without the lead line. The difference, aside from minor effects of accuracy of method, is brought about solely by the difference in topping-lift heights h, which are 25 and 10 ft (7.62 and 3.05 m), respectively. The *boom* axial loads vary as well and are 24,560 and 54,150 lb (109.2 and 240.9 kN) for the two cases. Thus, a change in derrick configuration has imposed a completely different set of member loads on the system.

Each element in the opening definition of a load rating can be seen as conveying the concept that load ratings are valid only within the bounds of defined conditions. Although the term *load rating* almost invariably appears by itself without the essential modifiers, one should never fail to understand that specific conditions are implied whenever the term is used. A complete load chart should include notes or some reference to delineate its preconditions.

Limitations

The numerical examples given in earlier sections indicate load-lifting capacity of a derrick can be limited by rope strength of the topping lift or load hoist, line pull availability, strength of the boom, and strength of the mountings. These are limitations of a simple example derrick; among cranes and other derrick forms there are a variety of other components that can also limit capacity. Strength is usually the governing criterion, but deflection, hydraulic pressure, power draw, or service life factors can limit, too.

Cranes and derricks are subjected to cycles of loading in the course of their service lives. When the combination of the number of loading cycles and the range of stress variation causes cumulative damage, there is a limit on probable life, after which fatigue failure is distinctly possible. Most lifting devices do not operate with set loads and motions but are exposed to variable working conditions. The loads and the stress variation they produce are random, and to complicate the matter further, operating radii and even machine configurations can be random variables as well. To evaluate the equipment in order to prepare a design capable of a specified service life requires the application of probabilistic concepts to augment ordinary deterministic

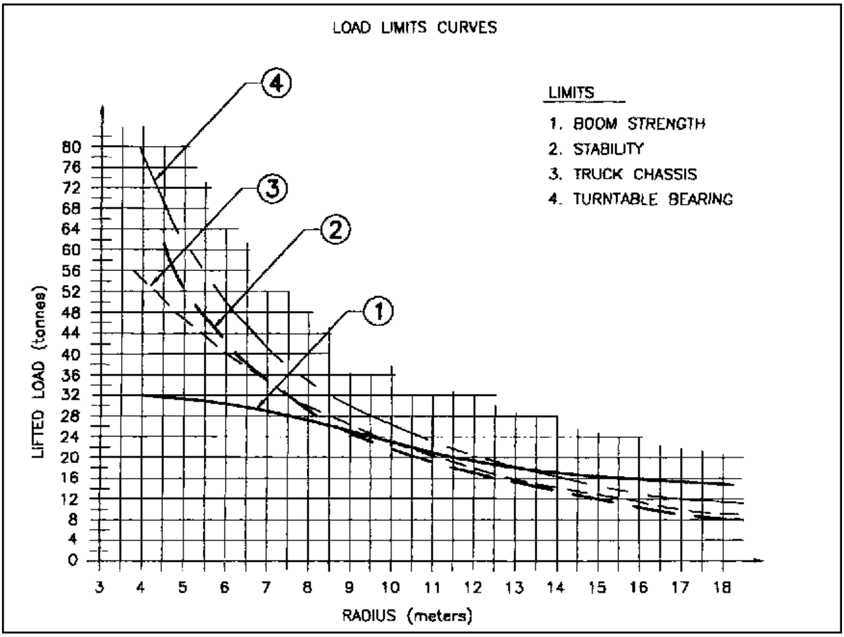


Figure 1.21 P&H model 6250-TC truck crane of 300-ton (272-t) capacity in a state of balance during stability experiments. (*Harnischfeger Corporation.*)

methods of structural analysis. Contemporary fatigue design methodology is far from perfect. Fortunately, however, most construction cranes and derricks have few, if any, components falling in the fatigue design range.

Stability against overturning is the most important factor controlling load ratings for mobile cranes and some other equipment types. Rail-mounted portal and tower cranes, for example, are often similarly limited. For still other equipment, load ratings are based on the assumption that foundations or bases will be so constructed that security against overturning will be provided. Figure 1.21 presents four curves of load limits that are considered in establishing load ratings for a truck crane.

Stability Against Overturning

The crane shown in Figure 1.22 can be made to overturn if a large enough load is boomed out to a great enough radius. Tipping will take place about a definite line called the *tipping fulcrum*. That line can be identified for the various forms of equipment and will be in the more complete stability studies in Chap. 4. At this point, it is sufficient to note that such a line exists and is definable. The CGs of the

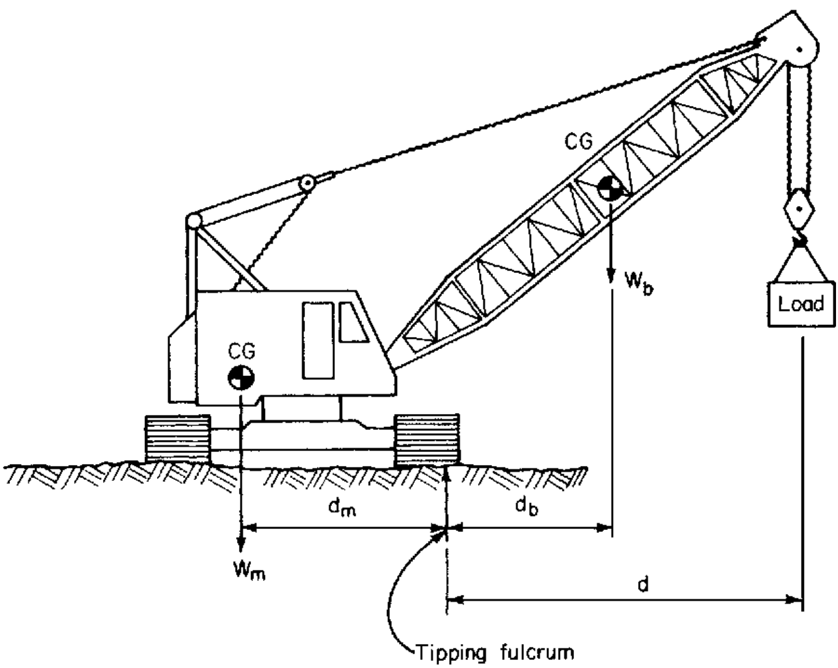


FIGURE 1.22 Barge-mounted crane listing under the weight of a 75-ton (68-t) bridge section during unloading operations. Lifting accessories were designed and operations planned by one of the authors. (*Lawrence K. Shapiro.*)

crane components are well defined. Given these data, the moment of the machine weight about the tipping fulcrum can be expressed as $W_m d_m$; it is called the *machine resisting moment*, the *stabilizing moment*, or simply the *machine moment*. The moment of the load and the boom, given by $W_b d_b + Wd$, is called the *overturning moment*. If the *load radius* is increased to the point where the overturning moment and the machine moment are equal, the point of incipient overturning has been reached. Any increase in the load or *radius* will cause the crane to fall over (Figure 1.23). The load that produces incipient overturning is the *tipping load* at that radius.

In U.S. mobile crane practice, when stability governs, load ratings are limited to a fixed percentage of the tipping load. The percentage used for each type of equipment has been set as the result of cumulative experience over the years rather than from some mathematical determination. With the recent acceptance of very high-strength steels and the increasing needs at construction sites, cranes have experienced a remarkable growth in lift capacity and boom lengths. With this growth have come situations where the old stability margins might not provide adequate reliability.

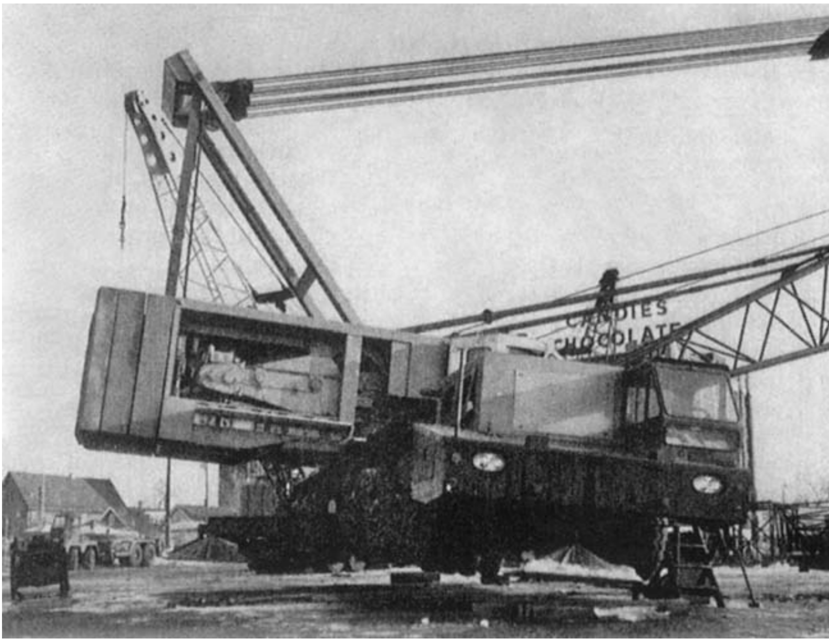


Figure 1.23 General arrangement of a basic telescopic cantilevered boom crane, more commonly known as a hydraulic crane.

Machine moment remains constant if the machine does not swing, but as the radius increases, the boom moment absorbs an increasing proportion of available moment and the tipping load decreases rapidly. As maximum radius is approached, the *rated load* can become rather small and the overturning moment can become 95% or more of the resisting moment. At such high percentages, cranes are quite sensitive, and a small perturbation might induce overturning. A crane weighing 400,000 lb (181 t) with ratings at 85% of the tipping load and a long-radius rated load of 3000 lb (1360 kg) will tip with the addition of only 530 lb (240 kg), or 0.13% of machine weight. That small increment of load can easily be induced by wind or by dynamic effects. An alternative approach toward stability-based ratings that takes into consideration the relative contribution of the boom weight has been adopted in some countries. This method is described fully in Chap. 4.

At close radii, strength is likely to govern versus tipping. For some configurations such as luffing jibs, strength-based ratings may be prevalent.

Figure 1.23 shows a 300-ton-capacity (272-t) truck crane during stability research experiments conducted some years ago by the Harnischfeger Corporation at a test facility. That 360,000-lb (163.3-t)

Figure 1.24 Curves of lifting capacities as limited by *boom* strength, crane stability, truck chassis strength, and turntable bearing strength. For each radius, the crane rated load will be the lowest value from these and other limiting factors.

machine is in a state of balance about its tipping fulcrum, the entire weight of the machine and load being supported on the two *outrigger beams* and pads.

Since floating cranes do not have a firm base for support, they must list under load (Figure 1.24). List angles are therefore also limiting factors for load ratings. The stability of barge-mounted cranes is also covered in Chap. 4.

CHAPTER 2

Crane and Derrick Configurations

In Chap. 1 basic *crane* and *derrick* concepts and mechanisms were introduced; this chapter builds on those concepts to describe various machines. Equipment form, operating motions, capabilities, advantages, and disadvantages are discussed. The reader will become generally familiar with individual machines and with the range and diversity of equipment available. The ability of each device to satisfy construction or industrial needs is viewed in the perspective of the array of crane types marketed.

2.1 Introduction

The commentator who facetiously said that a camel is a horse designed by a committee obviously was not an engineer. Though both animals are beasts of burden, an engineer would certainly have appreciated their operating environments and performance characteristics are in no way alike. He would have recognized that the differences are an inevitable and necessary consequence of disparate places of habitation and work roles.

The same commentator might have offered a similar observation about mechanical beasts of burden, such as cranes. Here too, evolution has led to differentiation to fit a variety of tasks and environments. The major differentiation has been between machines for construction versus those for general industry. The dividing line is rather hazy, however. For the most part, industrial machines find application in long-term permanent installations with consistent operating conditions. Construction cranes, on the other hand, are used for work-site assignments that may last from several hours to several years; these machines are likely to be exposed to a broad variety of work assignments and operating conditions. Despite these differences, a surprising number of machine types find extensive application in both construction and industry.

The variety of cranes and crane-like machines is ever-growing. Lifting equipment such as *articulating boom cranes* (Figure 2.1) and

FIGURE 2.1 Articulating boom, commonly referred to as a knuckle boom, is useful for deliveries and for reaching between floors of an open building. (*Lawrence K. Shapiro.*)

boom trucks (Figure 4.3) are used more for delivery of materials than for construction; so for the sake of brevity they are omitted from broad discussion in this volume. Other types that are tangential to construction work such as *revolver cranes* and gantry cranes are given passing reference. On the other hand, some lifting devices such as *jacking towers* and *telescopic gantries* that are not cranes but are commonly used for heavy rigging are treated in some detail.

2.2 Derricks

Derrick (or Derick) was the name of a London hangman in about 1600. Given the similarity between a gallows and lifting devices in use at that time, it is not hard to see how our present equipment came to bear this name. But, for the sake of clarity, a definition is in order.

The authors define a derrick as a device for raising and lowering loads laterally through use of a hoisting mechanism employing ropes but whose hoisting engine is not an integral part of the machine. In the ANSI Safety Code for Cranes, Derricks, Hoists, Jacks and Slings, a derrick is defined as "an apparatus consisting of a mast or equivalent member held at the end by guys or braces, with or without a boom, for use with a hoisting mechanism and operating ropes." This latter definition is perhaps too restricting, leaving a number of hoisting devices without a generic name. The authors' definition also provides a simple definition for a crane, contrasting it from a derrick by the simple integration of a hoist engine.

Although we tend to think of derricks as comprising struts and perhaps other structural members, the author's definition could include derricks made of nothing more than ropes and fittings. They could rely on the structure on which they are mounted for rigidity and stability.

Since many types of cranes and derricks are little more than elaborations or variations on the Chicago boom discussed in Chap. 1, the *Chicago boom* is the first derrick to be treated in detail.

Chicago Boom Derrick

A Chicago boom can be mounted on a building frame during or after construction, on a tower, or on any frame. Indeed, Figure 1.19 shows one installed on a power plant stack. When a boom or strut is assembled in the form of a Chicago boom, it can range from as little as 10 ft (3 m) to as much as 125 ft (38 m) in length, and capacities can range from a low of, say, ¼ ton (225 kg) to a practical upper limit of perhaps 35 tons (32 t). In the not too distant past, booms were made of wooden poles. Short lightweight booms are easily and inexpensively made of single steel pipes fitted with the necessary attachments, but most booms are trussed, or latticed, structures of angle irons or tubing or a combination of the two. Aluminum and synthetic composite booms are plausible, too, where site conditions favors such unconventional materials.

The *topping lift* usually employs ordinary hoisting *blocks*; one is fitted at the boom tip held off with steel straps while the opposite end is mounted at the pivot fitting on the support structure also with straps. The upper *load block* may consist of sheaves built into the boom head, or it may be a common block suspended on straps. The purpose of straps is to allow clearance between the rope suspension system and adjoining boom elements through the full range of luffing motion. See Figures 1.14, 1.15, 1.17 and 2.2 for examples of various rope suspension arrangements.

A Chicago boom is able to hoist materials to a height above the boom foot, with the horizontal reach limited by the length of the boom and the swing arc by the host structure. Swing guys often are fitted to the boom tip and are run laterally on each side to a point of anchorage. Wind, friction at the pivots, and the resistance of the opposing guy must be overcome when pulling on the line to swing. With a manual arrangement, the guys are fiber ropes arranged with several parts of line. Hand pulling through several parts can take several minutes to swing the boom through 90°. Where production economics justifies the expense to attain greater speed, mechanical swing systems are used.

In the typical installation, a two-drum winch is used to power the hoisting and topping motions, but when the work involves only lifting and swinging, the topping motion will not be needed. A fixed-rope guy line can then be installed, or to make adjustment easier and to provide flexibility, the system can be reeved to a chain-fall of the type shown hanging in Figure 2.2. In this case, only a single drum winch is needed.

Visual control can be established when the winch is located at the floor level of the boom foot, particularly when the loads are to be

Figure 2.2 A 36-ft (11-m) Chicago boom installed at the McGraw-Hill Building in New York City during construction. The curved steel straps on the upper load block permit concentric mounting while allowing the derrick to boom in-to very small radii. The chain-fall in the foreground is for transferring loads from the derrick to the building floor. (*Lawrence K. Shapiro.*)

hoisted to this floor, but the winch can be located at any level. When the winch is too large or too heavy to be lifted in the job-site material hoist or in the elevator of an existing building, it may be necessary to use a small temporary winch to operate the derrick in order to hoist the working winch. Alternatively the winch can be positioned on the ground.

When the winch is on the ground, the operator has direct communication with the ground crew and can have the boom and load in view at all times. When the winch is on the same floor as the boom foot and the loads are to be hoisted to that floor, the operator has direct communication with the swing and load-landing crew and has the boom but not the load in view at all times. Landing the load is often the most difficult part of the operation, so that direct and immediate communication at landing level is a distinct advantage. With less rope weight to overcome, overhauling weight can be reduced when the winch is at the higher elevation.

A Chicago boom is not a high-production machine. Its capital cost is very low and installation costs are moderate, but its crew size is large in relation to productive capacity, and its reach is limited. Typical applications are "on again, off again" rigging work or situations where the load placement is above the reach or capacity of mobile cranes. With quick-setup telescopic cranes reaching ever higher, the demand for these lifting machines has diminished.

Guy Derrick

Guy derricks at one time were the only practical means to erect steelwork on high-rise structures, but that work is mostly done now with tower cranes. Guy derricks may also be used for general rigging, stone quarrying, and construction of refineries and chemical plants. A guy derrick can be described as a Chicago boom with its own integral column, called a *mast*, held vertical by six or more guy ropes, as shown in Figure 2.3. The guys radiate in a horizontal circle about the derrick and are spaced as evenly as site conditions permit. A common configuration would include a mast of about 125 ft (38 m) and a boom of about 100 ft (30 m), although both larger and smaller derricks may be found. Capacities can range to 200 tons (181 t) or more. Both boom and mast are almost invariably latticed.

The boom of a guy derrick is usually made shorter than the mast so that at close-in radii the boom can swing under the guys, enabling the derrick to work through an arc of 360°. Narrow loads such as steel beams are easily handled in this way, but care must be exercised to prevent loads from striking the boom. Some derricks have steel plates on the load side of the boom to prevent this.

When a guy derrick swings, the boom and mast move together; a pivoted fitting is provided at the mast top, and a ball-and-socket joint is mounted at the bottom. Above the top pivot is the fitting to which

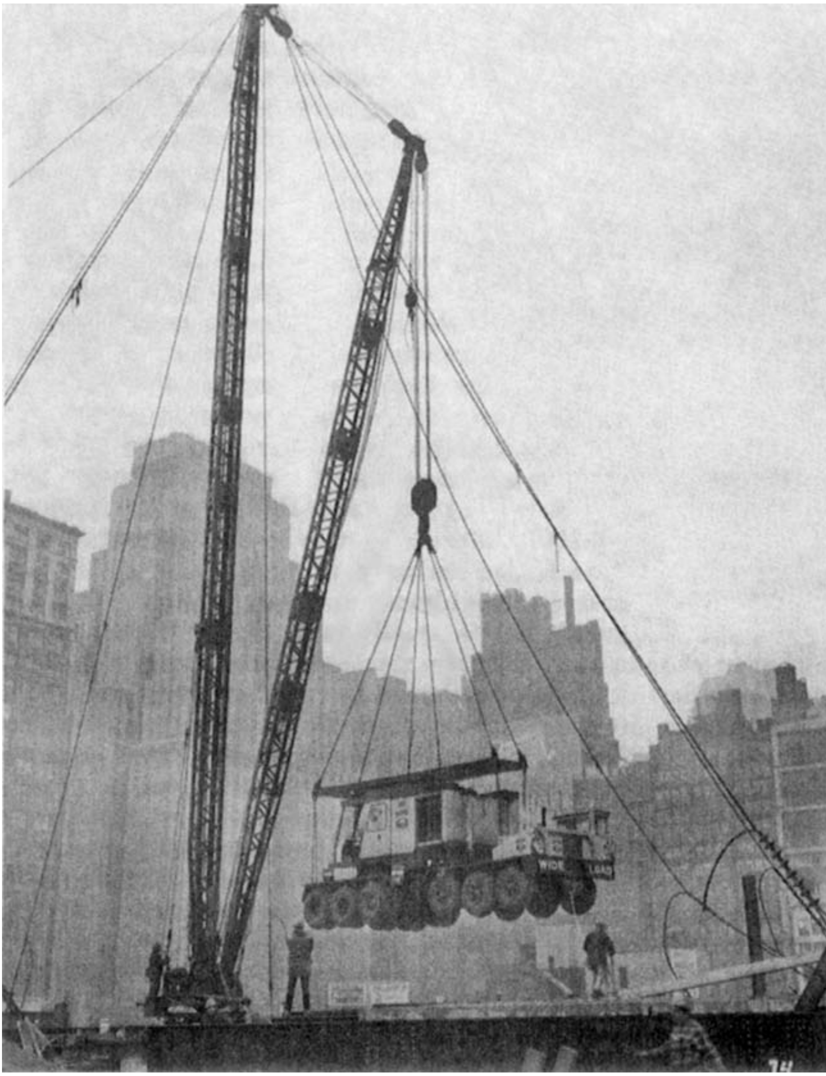


Figure 2.3 A dramatic lift by a guy derrick during steel erection at a New York City high-rise office building. Note the planks inserted to prevent guy turnbuckles from unwinding.

the guys are attached, called a *spider.* The top pivot itself is called a *gudgeon pin*. Swinging, or *slewing*, is accomplished by using a large horizontal wheel, called a *bull wheel*, which is fitted at the bottom of the mast. Wire rope is run around the bull wheel, and winch power is used for swinging. Alternatively, on small derricks, an arm, called a *bull pole*, can be used to provide leverage for pulling the derrick around manually. Another method, sometimes used on steel erection derricks, makes use of a handwheel and gearing so that one person at the base of the derrick can swing the boom.

The topping lift is attached to the mast just below the gudgeon pin. The topping and load lead lines come into the mast at the top and above the boom foot, respectively, and run down the center of the mast. The lead lines run into the derrick base, which is fixed in position, over sheaves and out to the winch. Because the lead lines run through the center of rotation, they do not encumber the swing motion.

The mast must be held close to plumb by the guys; excessive lean will make swinging difficult to control, akin to running up and down a hill. Even when tensioned to remove excess sag, the guys hang in a *catenary* shape. Under load, those opposite to the boom are stressed; this causes rope elastic stretch and a reduction in catenary sag, which make the mast lean toward the boom. In order to control mast lean, the size, construction, and initial tension of the guy rope must be taken into consideration, together with guy configuration and load levels.

Guy derricks are practical for steel erection work because it is possible for the derrick to lift itself, or jump, as the height of the work increases. The winch is left at base level as the derrick jumps; the winch operator works blind, receiving all instructions by signal or voice communication.

A guy derrick must be supported at the mast base, and anchorage points are required for the guys. On a building, a steel grillage is usually placed under the mast to transfer the loads to the host structure. The guys are then anchored to the building frame. Inasmuch as steelwork first proceeds with only partial bolting at the connections, analyses must be made to assure that the partially connected structure is adequate to support the derrick reactions together with structure dead weight, construction loads, and wind. This study must be done for each derrick-mounting level.

The initial erection of a guy derrick requires use of another lifting device such as a crane. The grillage and its supports, the guy anchorages, and the jumping operation are all cost factors that need to be considered. But in addition, in order to swing a load past a guy, the derrick must be boomed-in until the boom can pass under the guy, or alternatively the guy must be lowered and then replaced after the boom passes. This cyclical operation is both awkward and time-consuming; its relative inefficiency should be factored when comparing the guy derrick to alternative erection means.

Though at one time they were stock tools of the lifting trade, guy derricks are now unfamiliar to most ironworkers, and the skill set needed to operate them has been largely lost. For industrial work, they have been replaced by other equipment types. In high-rise construction, they are not easily adapted to the large bays and lighter floor framing of contemporary structures. Thus guy derrick utilization is a rarity nowadays.

Gin Pole Derrick

A *gin pole* is the most elemental form of derrick—it could be described as primitive—essentially a leaning mast incapable of swinging and held in place by guys. With no boom, lifting is done from a load fall at the tip of the mast. Large and small gin poles find roles quite distinct from one another. In the middle range alternative lifting devices are more practical, and so middle-sized poles are a rarity.

Long heavy-duty gin poles operate at fixed radii with a lean of only 5° or 10° from the vertical. They are primarily used for rigging work, raising heavy machinery, vessels, or structural components into place. Poles of 250 ft (76 m) or more are available with capacities ranging to about 300 tons (270 t). The practical limit to the size results from the need to use other lifting equipment for assembly.

At a refinery or chemical plant, large gin poles are sometimes paired and used to erect vessels, as shown in Figure 2.4. The derrick hooks are made fast to the vessel at a point above the center of gravity. As the derricks lift, the bottom end of the vessel rotates and must be moved laterally toward the derrick bases. To accommodate those

FIGURE 2.4 A pair of gin poles prepared for lifting a vessel. The tailing crane is not visible.

movements, a *tailing crane* lifts the vessel bottom and leads it in toward the poles. Tailing cranes are discussed in detail in Chap. 5.

A small gin pole fitted with a topping lift and side guys can luff in and out. A pole of 30 ft (9 m) or so with capacity of about 5 tons (4.5 t) is often used for raising loads to the roof of an existing building or for erecting a larger derrick. Though simple in form, a luffing pole requires a skilled rigger to operate it safely as it can easily become unstable. As other types of lifting equipment have replaced them, the requisite skill needed to operate these rigs is increasingly difficult to find in the labor force.

Stiffleg Derrick

Like a guy derrick, a *stiffleg derrick* has a boom supported from a vertical mast. However, instead of guys it has two rigid inclined legs supporting the mast. Without guys for the boom to clear, the mast can be shorter than the boom, but the operation is restricted by the stifflegs to an arc that is between 2/3 and 3/4 of a full circle. Lightweight and easily reassembled in the field from small components, it is often a good choice for a heavy-duty lifting apparatus to be mounted on a rooftop. A stiffleg derrick is also commonly deployed in such industrial settings where simplicity and ruggedness are an advantage and where limited mobility is not a drawback.

In an industrial application—concrete batching plants, stone yards, and barge unloading facilities are common settings—the stiffleg derrick could be mounted in the ground or set on a gantry or fixed tower. When permanently installed at ground level, the mast bottom and stifflegs are anchored in structural foundations. For aboveground mounting, horizontal members called *sills* are added to connect the leg ends and mast bottom. The sills react to the horizontal thrust from the stifflegs.

Stiffleg derricks are available in a wide range of sizes and capacities. For general light building work, 12-ton (11-t) and lighter models are made with booms from 30 to 50 ft (9 to 15 m) or so. At the other end of the scale are machines with capacities to 700 tons (635 t) and booms to 265 ft (80 m). For special applications, stiffleg derricks can be mounted on towers or on portal frames either fixed or traveling.

The mast, legs, sills, and boom can be of latticed construction with angle-iron or tubular members, or legs and sills can be pipes, rolled shapes, or laced channels. Typically, the mast rotates together with the boom during swinging; it has a top pivot at the joining of the legs and a pivot or ball-and-socket joint at the mast bottom. The mast base is mounted in a fixed position and houses the lead sheaves. Lead lines come down the center of the mast and out at the base sheaves, just as in a guy derrick.

The stiffleg derrick designed by the authors' firm shown in Figures 2.5 and 2.6 has mast and legs of pin-connected pipes in short

Figure 2.5 Stiffleg derrick with a 62-ft (19-m) boom lowering part of an air-conditioning chiller. This derrick of 35-tons (32-t) capacity is made up of short-pinned components so that it can fit into elevators or jobsite material hoists. Both derrick and installation were designed by the authors' firm. (*Lawrence K. Shapiro.*)

lengths so that the parts can be lifted in a jobsite material hoist. The boom is also in short pinned lengths. Sliding anchorage fittings are provided to accommodate variations in building framing dimensions from jobsite to jobsite. The sills themselves are rolled beams in pinned segments. The mast is fixed and does not rotate; instead, the boom is stepped in a fitting that rotates about the mast on a Teflon bushing. Swinging is achieved through use of a small-diameter bull wheel (30 in or 0.76 m) and a hand-operated winch.

Other Derrick Forms

Gallows Frame[1]

When two gin poles are assembled with a horizontal beam across the pole heads, the device is called a *gallows frame* and resembles

[1]Riggers in the New York City area have used this term to refer to another device, a four-legged frame used in plant machinery rigging.

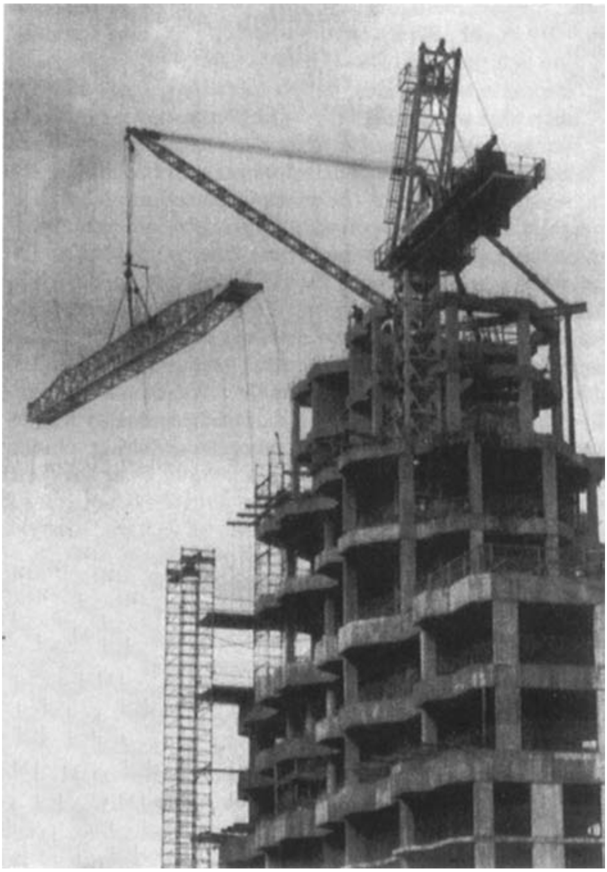


Figure 2.6 The derrick of Figure 2.4 dismantling a tower crane 58 stories above the street. The very small roof area made it necessary to support and tie down both sill ends on a setback two stories below. The crane installation, the derrick installation, and the dismantling procedures were designed by the authors' firm. (*Jay P. Shapiro.*)

the apparatus that the hangman derrick used for his macabre work. The upper load blocks are fitted to the beam, which is arranged so that the poles are loaded concentrically. This gives the gallows frame a greater capacity than is available with a similar double gin pole; the increase can be in the order of 50%.

As with gin poles, the height of a gallows frame is limited by the equipment needed to erect it, but units with hook lifts of 200 ft (61 m) are available with capacities of 600 tons (545 t). Shorter configurations achieve ratings as high as 1200 tons (1090 t).

A double gin pole can be used to erect a vessel taller than the poles, but because of the gallows beam, a gallows frame is limited to shorter vessels.

Column Derrick

A *column derrick* is a modified guy derrick that is useful where loads must be lifted to a building setback or penthouse. Instead of guys, a structural frame is installed at the mast top spider and is attached to building columns or other substantial bracing means (Figure 2.7). The frame acts as a rigid mast support system as opposed to the guys ordinarily used; because guys are somewhat slack, a guyed mast will lean toward the boom as a load is lifted. A column derrick can be installed at locations where space is limited or where there is not sufficient space or clearance for a full set of guys. But there must be an adjacent structure high enough for attachment of the top frame.

A column derrick can swing freely since there are no guys in the way, and it is not necessary to utilize that time-consuming maneuver of booming-in and ducking under the guys.

Shaft Hoist

This device is a derrick only under the authors' definition. It is a simple device without compression members, made by suspending an upper load block in an elevator shaft or other framed shaftway by wire ropes tied to the four corner columns or to other members. The weight of the load together with the four tensioned ropes holds the block fixed in position either at the center or offset in the shaftway. An alternative form would have the block supported from beams spanning the shaft. The lead line can be run from the upper or the lower block to a snatch block and thence to the winch. The capacity of a shaft hoist is limited only by the strength of the shaft framing or the cost of reinforcing the framing. This device is inherently inexpensive for most installations. Figure 2.8 shows a shaft hoist, which the authors' firm designed for lowering equipment to a subcellar machinery space in an office tower.

Cathead

Another derrick under the authors' definition, the *cathead*, is a beam cantilevered out from a support with the upper block attached to its outer end. An alternative form has the outer end of the beam supported by ropes tied back to the structure. A cathead is used for loads ordinarily on the order of 5 tons (4.5 t) maximum, but capacity is limited mainly by the strength of the support framing or the cost of frame reinforcement. The typical cathead is incapable of changing load radius, so loads must be handled by *drifting*. A load is drifted when it is pulled laterally to change its horizontal position; thereby it makes the support ropes depart from the free vertical. The cathead is an inexpensive device, but its capabilities are limited.

When a load is suspended from a long line, drifting a short distance requires very little force and the lateral, or side, loading imposed on the derrick is slight. Both forces increase as suspension length decreases or drift increases. A load cannot be landed onto a building

Figure 2.7 A column derrick used to place heavy precast facing panels on a high-rise hotel under construction. This is a guy derrick without guys, a concept of A. J. McNulty & Co., Inc., designed by the authors' firm. The upper support (which replaces the guys) and the base support are cantilevered from the building frame. (*A. J. McNulty & Co., Inc.; photo by Robert Weiss.*)

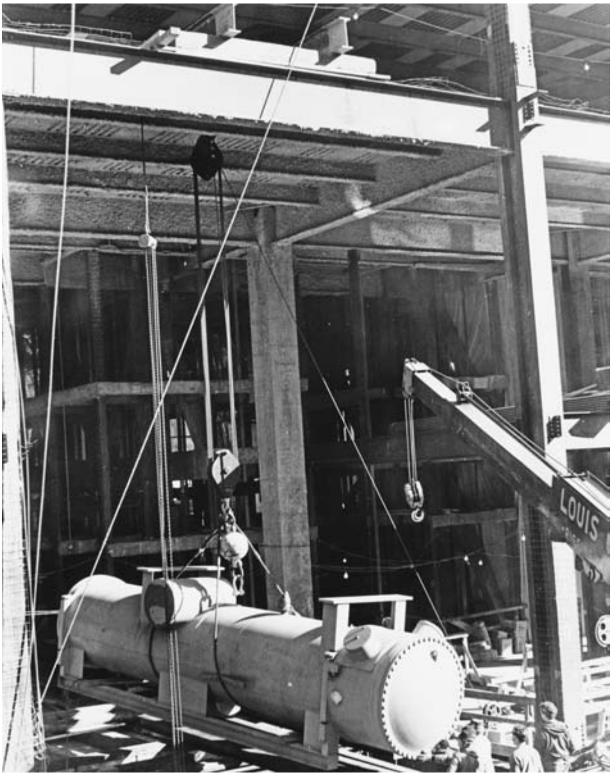


Figure 2.8 The bottom element of a large air-conditioning chiller ready to be lowered into a subcellar by a shaft hoist. The unit was rigged over the opening on steel beams, which will be removed by the small telescopic crane. The 30-ton (27-t) machine was one of several installed at the McGraw-Hill Building. (*Lawrence K. Shapiro*.)

floor until its center of gravity is within the building. This often cannot be accomplished by drifting, and the ordinary rigger's methods of shifting loads may be too time-consuming to be economical for highly repetitive work. In such cases a cathead might be fitted with a trolley that suspends the upper block while permitting radius change and ease of landing. The trolley can be controlled by a small hand winch, a chain-fall, or a come-along.

On one job, a cathead was needed to hoist large building-facing panels weighing some 7½ tons (6.8 t) to a maximum height of about 500 ft (150 m). The panels were to be transferred from the cathead to monorails for lateral distribution and positioning for placement. Because the panels had to be hoisted past panels previously set into place and were large wind-catching surfaces, hoisting had to be done well clear of the building face. The monorails, on the other hand, had to be set close to the face to simplify panel placement. The necessary radius-change capability was provided by a trolley system and hand

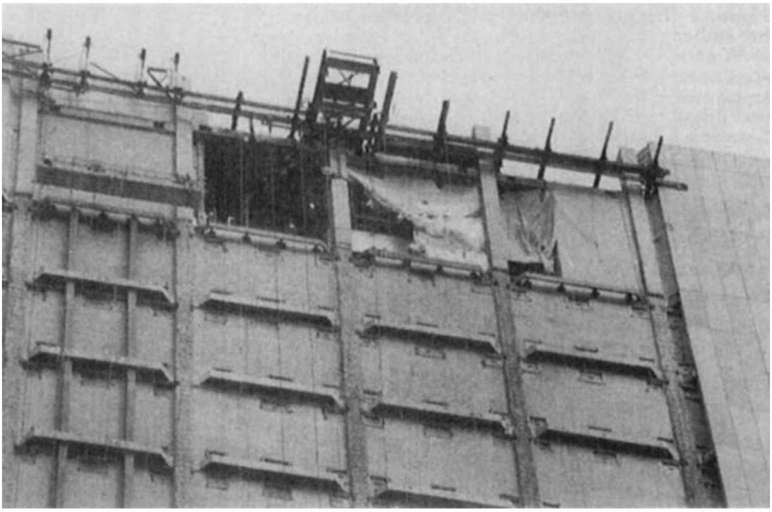


Figure 2.9 Cathead and monorail system for installing stone facing panels on a telecommunications building. The cathead lifts units well clear of previously set panels and then trolleys the load in for transfer to monorails for lateral distribution and placement. (*A. J. McNulty, Inc.; photo by H. Bernstein Assoc., Inc.*)

winch. Capacity and load stability were enhanced by use of a double beam. Figure 2.9 shows the installation.

Flying Strut

The flying strut is an invention of one of the authors that has been used once. It is mentioned here only to demonstrate that seemingly intractable rigging problems can yield to imaginative thinking.

The flying strut was used at a hospital construction site (Figure 7.39) after the framework was in place and the concrete floors had been poured. It was necessary to lower a series of 34-ton (31-t) chiller units and other machinery into a subcellar. Conditions dictated that the lowering be done halfway back into an alley of limited width between an existing building and the new structure. The alley was at subcellar level, and there were beams framed across and a partial deck poured at street level. No existing equipment or procedures could perform the work economically under the given conditions.

The flying strut, a compression member suspended horizontally by ropes perpendicular to the building face, was relatively inexpensive to make and install (Figure 7.38). Half inside and half outside the building, the inboard end is held by four positioning ropes attached to columns on either side of the strut at the building face. Stabilizing ropes, or *preventers,* are run to interior columns as well. At the outboard end of the strut, the upper load block is mounted together with suspension ropes capable of supporting the load and running up several stories to column anchorages. Side preventers are also provided at the outboard end. The load block is not supported by the strut, instead, the strut is a means for positioning the block for the work. By providing take-up devices on the positioning and preventer ropes, the position and attitude of the strut can be controlled.

The strut and its suspension system needed to be designed only for the lateral thrust resulting from positioning the load block. At the hospital job, it proved to be a quick and easy means of maneuvering the loads.

2.3 Mobile Cranes

There was a time when *mobile cranes* were identified with the United States and *tower cranes* with Europe, but the globalized modern market for all cranes now transcends regions and borders, blurring these differences.

The original author of this book liked to recall a yarn told by a Danish engineer, a tale that summed up a common European view of Americans, expressed in terms of crane practices. The Dane offered the view that the modern descendent of the Old West six-shooter is the American mobile crane with the gunslinger spirit living on among its operators. Whether one agrees or not with its underlying derision, this yarn reflected a genuine difference in crane design philosophy that existed at the time it was told, a difference that persists to this day. Muscular mobile cranes, like muscle cars, have their origins in the open spaces of the American heartland. The mobility and unfettered power of these machines—at odds with European sensibilities that favor constraint—fall in line with the American ethos. For better or worse, traditional American mobile cranes put the operator in full control.

The European approach, established first with tower cranes but now applied at large, is to take that discretion away from the operator by modulating power and building in automated limits to the crane controls. In the European Union itself, the attitude is codified in the Machinery Directive, a broad-based safety standard—applied to cranes among a wide variety of machines—that demands manufacturers eliminate risks and take into account foreseeable misuse. Though American attitudes have not much changed, the prominent position of European manufacturers and the need to satisfy the Machinery Directive for sales in the EU gives the more restrained European approach acceptance in many of the world markets.

Aided by technology that integrates sensors, software, and control systems, mobile cranes are following the path of tower cranes by becoming more automated and less reliant on “seat of the pants” skills. Nonetheless, they have more freedom of movement than tower cranes, and thus operator skill, though somewhat changed in character, is far from obsolete. While still keeping a sharp eye out the cab window and giving the signalman rapt attention, today's mobile-crane operator also needs to interface with the rich array of information presented on a computer console. The old skill set that favored guts over brains has been reversed.

Crawler, Truck, All-Terrain, and Rough-Terrain Carriers

A crane that can move freely about the jobsite under its own power is a mobile crane. Movements about the jobsite are called *travel* movements, and movements from site to site are called *transit* movements. All mobile cranes can travel; wheeled mobile cranes are capable of self-powered transit movements, but *crawler cranes* are not. Borrowing a nautical term, the upper framework (above the swing circle) of a mobile crane is called the *superstructure* or *upperstructure.* The term *front-end attachment* refers to the boom or the combination of boom and other struts that support the hook and load. The lower framework is called a *carrier* on a *truck crane* and a *carbody* on a crawler crane.

The unofficial system used by manufacturers for rating mobile cranes is based on a somewhat fanciful notion of *lifting capacity*. The nominal rating—one might hear of a particular crane being a 250-ton machine, for example—reflects the greatest theoretical load that the crane can pick in any configuration. It would typically correspond to the shortest boom at the shortest radius. But a massive load so constrained might be so close to touching the boom and the cab that there is no practical application for such a lift. The nominal rating is thus nothing more than a rough index for classifying a crane; a meaningful assessment of lift capacity requires a study of the load charts.

Mobile-crane load ratings are usually governed by the ability of the machine to resist overturning. The machine weight itself provides considerable overturning resistance, but in order to increase capacity, *counterweights* may be added at the rear of the superstructure. There are exceptions where adding counterweight does not benefit; strength may limit short booms and lifts at close radii whereas deflection can limit very long booms. And there is a limit to the magnitude of the counterweight that can be added; the crane must have adequate resistance against tipping over backward when short booms are in use.

Counterweights are detachable for transporting when road limitations require. Some cranes are rated for different counterweight options so that the user is spared the cost of shipping more than is needed. As mobile cranes have become larger, manufacturers have made more components detachable for transport.

Crane controls function through mechanical linkages, pneumatic or hydraulic pressure, data busses, or electrical cables, depending on the size of the machine, manufacturer's preference, and control function. Power trains are geared from high to very low ranges, and torque converters or hydrostatic transmissions are common (Figure 2.10).

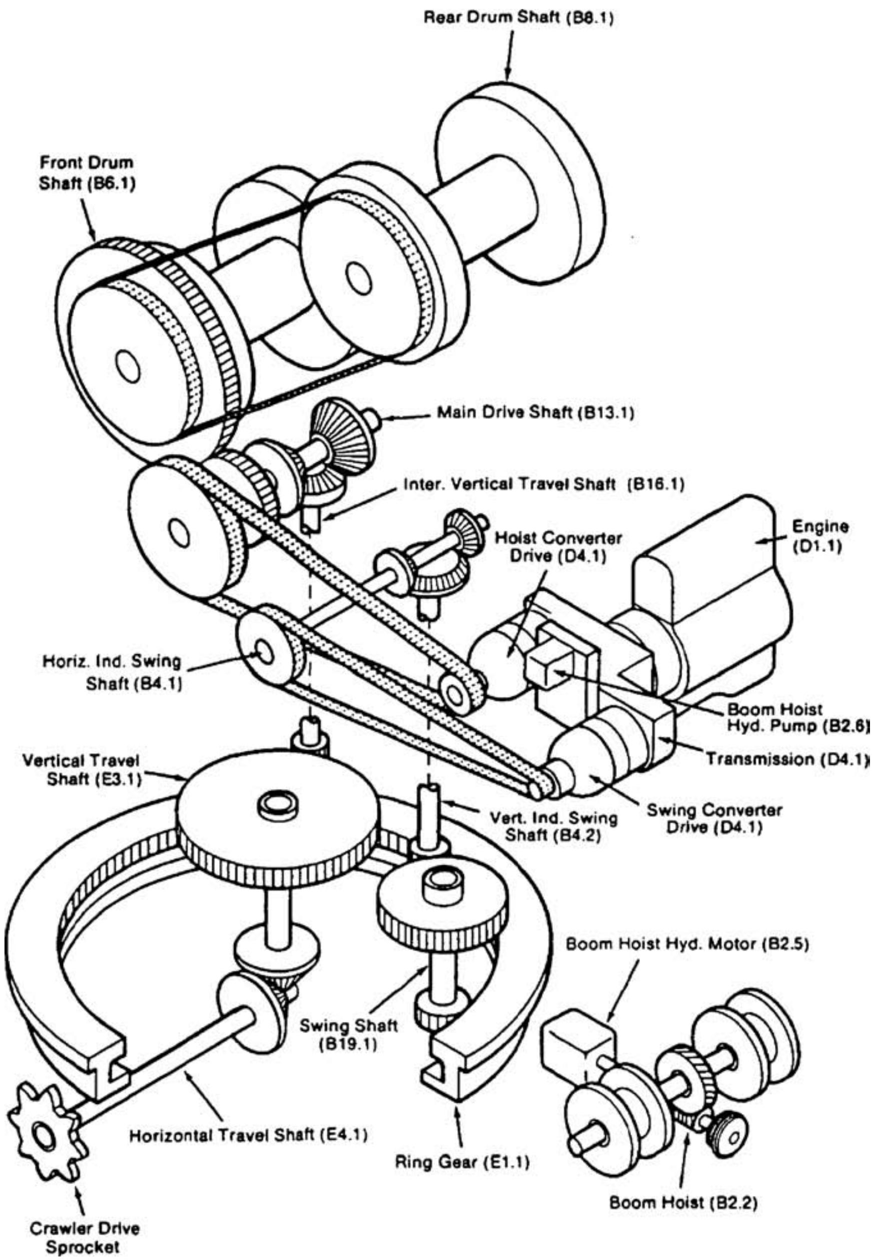


Figure 2.10 Schematic of machinery typically found in a mobile-crane superstructure. (*Manitowoc Company, Inc.*)

The elaborate control and drive systems are intended to enable an operator to maintain high production rates or to crack the shell of an egg with a 100-ton load. But some cranes have the power to produce operating speeds sufficient to destroy themselves. The high lift, luffing, swing, and braking speeds and accelerations attainable are not a reflection of recklessness or irresponsibility on the part of the manufacturers but a consequence of the range of operating configurations and conditions in which the crane must function. A driver need not operate an automobile at top speed, but the power needed to achieve top speed makes for good performance and often enhances safe operation at lower speeds. The same is true with a crane; as with the automobile, operator judgment and responsible behavior are necessary.

Crawler Bases

The first *crawler* crane came into existence around 1920. It might well have been assembled in a small American midwestern shop by an inventive mechanic or perhaps in one of the heavy-equipment factories scattered across that region. Though their essential form has remained constant, the work function of crawler cranes has evolved. They have always been used as all-purpose off-road lifting machines tasked with general construction as well as with quarrying, dredging, pile driving, and a host of other specialized tasks. Most of their *lift crane* workload, at the smaller end of the lifting spectrum, has been taken over by wheel-mounted all-terrain telescopic cranes. But modern crawler cranes are still essential for off-road work, deploying a large panoply of specialized *attachments* for the traditional list of tasks and also for carrying out long-reach and extraordinarily heavy lifting assignments.

The robust construction and generously sized power plant of a crawler crane make it suited to *duty cycle* work. The nature of this type of operation is that it requires steady and repetitive lifting with the rate of production a key factor. Pulling a *dragline* or lifting and releasing a concrete bucket clamshell in a steady cycle can be harsh on the hardware. On less hardy equipment, impact and reversing loads on components can induce fatigue cracking. Wear components such as clutches, brakes, bushings, and slide bearings may wear out quickly. Heat buildup can destroy delicate machinery. Many crawler cranes are designed with this harsh exposure in mind. Some manufacturers divide their crawler crane product lines to separate duty cycle machines from lift cranes, while some others offer the same basic machine in different versions.

Given its work role, a crawler base is designed to be rugged and stiff. (Stiffness is a trait needed to prevent the base from deflecting excessively under load.) The base mounting includes the hublike car body, a robust frame usually built up from heavy plates with the swing circle fixed on top. The bearing or *roller path* of the swing circle must transmit loads—weight, overturning moment, and swing torque—imposed by the superstructure on the base. At the sides of the car body, crawler frames are mounted; these hold the axles, propel drive, idlers, and crawler tracks.

The *tracks* of a crawler crane are intended for two purposes: they distribute the ground bearing loads over a wide area and they give the machine mobility, even while carrying a load on the hook. These advantages do not confer an ability to go anywhere under any conditions. The boom and jib are sensitive to side loading that can come about when the crane goes out of level; when dressed out, but most especially when there is a load on the hook, firm level ground support is crucial. Moreover, counterweights and front-end attachments, not properly managed, can create gross imbalance resulting in severe local *ground bearing pressure* or even a tip over.

Most crawler crane propel systems are powered by the engine in the superstructure through a vertical shaft on the axis of rotation, which in turn drives horizontal shafts to sprockets and drive chains on the *side frames*. An alternative method makes use of hydraulic motors mounted on the side frames (Figure 2.11). Some very large machines have a separate propel engine mounted on the base. Travel speeds from about 0.5 to 1.5 mi/h (0.8 to 2.4 km/h) are usual; smaller cranes can operate at the higher speeds. Parallel parking and other tight maneuvers are possible.

Each track is driven by a *drive sprocket* at the rear end. At the front end is a nonpowered *idler sprocket*. Either of these sprockets is also called a *tumbler*. Track rollers mediate between a fixed side frame and the moving articulating treads. Ground bearing load passes from the supple track to the rigid side frame through these rollers.

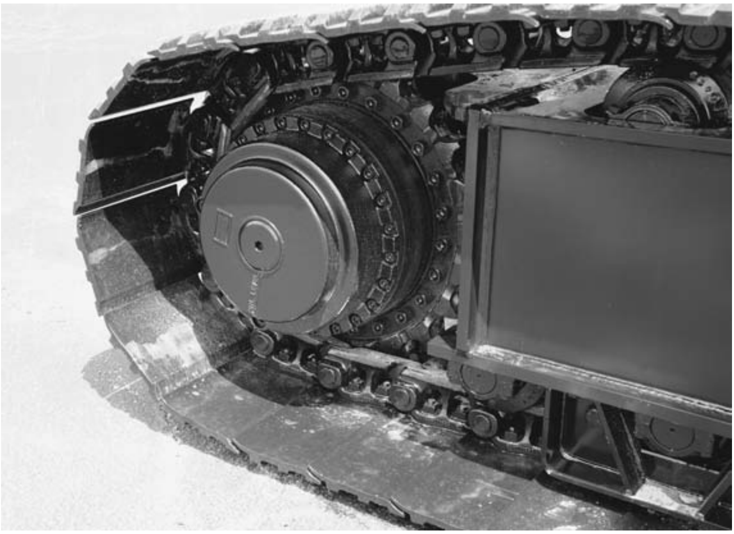


FIGURE 2.11 The crawler is propelled by a hydraulic motor at the rear end of the side frame, mounted on a gearbox and powering the drive sprocket. The idler wheel at the top right maintains tension in the track. Track rollers are on the bottom. (*Link Belt Construction Equipment Company.*)

Each tread propel drive has a separate clutch, brake, and lock for full independent control of travel and turning motions. The locks are engaged during operation and when parked on grades. The propel system is geared down for power, enough to "climb the side of a barn," in the words of one manufacturer's engineer. This great power is needed for maneuvering on soft ground and for traveling on ramps and grades. Crawler cranes may manage a 30% grade when not under load, but travel on steep slopes should be done only under controlled conditions and within limitations imposed by the manufacturer.

The distance between the crawler tracks is directly related to overturning stability, as shown in Figure 2.12. Enlarging the track spread can improve load-lifting capacity twofold: it directly enhances stability under load, and by improving backward stability, it allows more counterweight to be added. Some crawler frames are made to be retractable for transport and for operation in narrow quarters. Most machines with retractable crawlers are equipped to extend or retract without assistance but may require jacks, blocking, and tools.

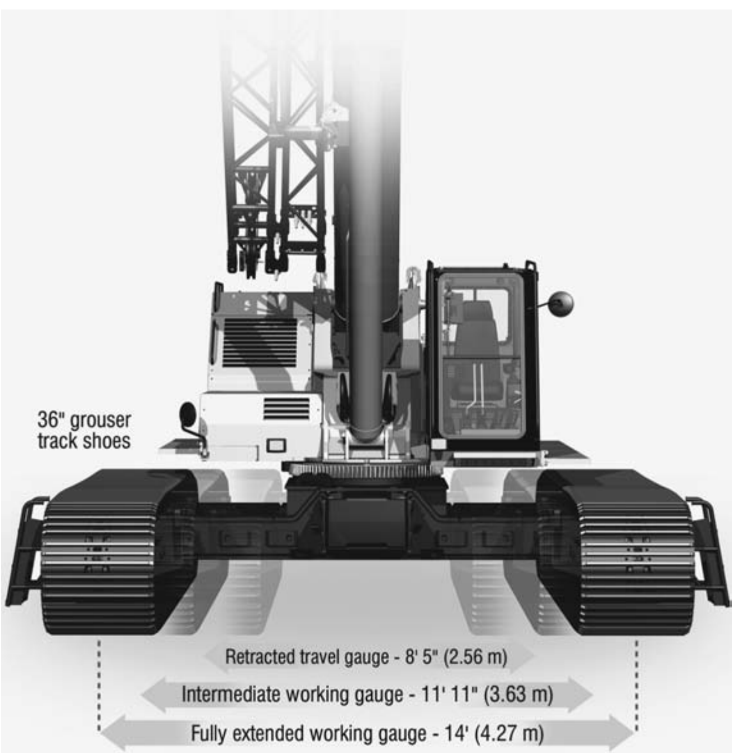


Figure 2.12 Increasing the spread of the crawler tracks adds capacity to any crane, but it makes rail or truck transit more difficult or impossible. Many crawler cranes, therefore, are equipped with extensible-retractable tracks. (*Link Belt Construction Equipment Company.*)

Beam-type low-bed trailers are used to carry small crawler cranes during road transit. For rail transit or for conforming to roadway restrictions, the side frames of large cranes can be removed without disassembly of the drive chains. The base mounting can also be separated from the superstructure, and all or part of the counterweight removed to reduce weight. Some models are able to lift the heavy side frames and counterweights from the *gantry* so that an assist crane is only needed for assembling the lighter boom components.

Crawler cranes offer lower cost and rental rates than do *truck crane*s, but their transit and assembly costs are higher. For that reason, economics favors them for longer-term assignments over *truck crane*s. This seeming disadvantage has also unbounded them from size and weight constraints; whereas *truck crane*s are limited to dimensional and weight limitations for driving on public highways, crawler cranes are broken down into as many pieces as necessary for transit. Designers have leveraged this distinction by developing crawler cranes into some of the largest lifting machines on Earth.

Truck Carriers

Truck-type crane carriers should not be confused with ordinary commercial truck chassis. A crane carrier is designed and built for no other purpose than crane service. Figure 2.13 shows a carrier for a 300-ton model. It is still clear that this chassis is quite different from that of a standard truck. The largest units may have as many as nine axles.

A variant is the so-called *boom truck* which has a telescopic boom and stabilizers mounted on a more-or-less conventional truck chassis. The mounting must meet specifications for frame stiffness, weight,

Figure 2.13 Carrier for a Link Belt HC-278H *truck crane*. (*Link Belt Construction Equipment Company.*)

and axle limitations. A boom truck, rated for up to 50 tons and reaching as high as 180 feet (55 m), has an advantage of being able to transport the loads it lifts.

Carriers must be designed for movement about the jobsite and on the highway. To accommodate those extremes, transmissions with as many as 33 forward gears are provided, and some carriers are equipped with special creeping gears as well. Road speeds vary from crane to crane but a range from 35 to 50 mi/h (56 to 80.5 km/h) is common. Jobsite ramps of 20 to 40% can be traveled without load, depending on the crane model and surface conditions. Brakes are designed to hold on the maximum travel grade.

A good crane carrier must have an extraordinarily stiff, torsionally rigid frame to keep from deflecting excessively under load, but this requires the addition of diaphragms, stiffeners, and reinforcements, which add to weight. For roadability, weight must be held to legal limits. Many important differences in the quality of carriers rest on the compromises made in satisfying those conflicting needs.

Carriers in the United States can be as wide as 13 ft (4 m), and the entire crane, in transit configuration, can weigh as much as 350,000 lb (160,000 kg). In many areas, highway load limitations do not permit such load weights or widths. Machines up to 12 ft (3.7 m) wide can be readily shipped on U.S. railroads, but wider loads require special routing. Often special provisions for transit must be made, as covered more fully in Chap. 5.

All-Terrain Carriers

First appearing in Europe during the late 1970s, the *all-terrain crane* now accounts for a majority of mobile-crane sales throughout the world. The carriers used on these telescoping cantilevered-boom machines combine the high road speeds of truck carriers with some of the off-road capabilities of the rough-terrain crane.

Large-capacity multiaxle models have appeared featuring the characteristics necessary to carry the all-terrain label: high road speed, off-road maneuverability, and drive positions in both the chassis-mounted cab and in the operator's cab mounted on the rotating superstructure (Figure 2.14). To achieve maneuverability, these cranes typically have all-axle drive and steering as well as crab steering (Figure 2.15). Many machines are furnished with sophisticated suspension systems that maintain equalized axle loading on uneven surfaces while the crane is in motion or is static. New models are continuing to appear, and models with more than 800-ton (700-t) capacity are available. Though pick-and-carry operations are usually permitted, *outriggers* are used for most lifting.

Rough-Terrain Carriers

Rough-terrain carriers have two-axle chasses used with telescoping cantilevered boom with up to about 130-ton (118-t) capacity. There is

Figure 2.14 A 90-ton all-terrain hydraulic *truck crane* moving off road. Characteristics of all-terrain cranes include high road speed, off-road maneuverability, pick-and-carry ratings, all-wheel drive and steering, crab steering, and driving controls in both the carrier and the upper cabs. (*Link Belt Construction Equipment Company.*)

Figure 2.15 Crab steering on a 200-ton all-terrain crane. (*Link Belt Construction Equipment Company.*)

FIGURE 2.16 The unusually large wheels on this rough-terrain crane make maneuvering easier on uneven ground. Driving and operating functions are controlled from one station. (*Terex Corporation.*)

one operator-driver station that may swing with the boom (Figure 2.16) or remain fixed. Carriers of this type are characterized by oversized tires that make travel easier under adverse site conditions. As with all-terrain cranes, pick-and-carry operations are usually permitted, but outriggers are used for most lifting.

These carriers perform transit moves at less than highway speeds, about 30 mi/h (48 km/h), and with some discomfort to the operator as a result of the high center of gravity and the high ratio of unsprung weight.[2] For long moves, these cranes are generally transported on low-bed trailers. A typical rough-terrain suspension has hydraulic cylinders that oscillate when the upper structure is centered over the front for travel, and a mechanism to execute an automatic *lockout* to improve lift performance when the crane swings.

Front-End Attachments

The "business end" of the crane mounting to the front end of the superstructure is called the *front-end attachment.* This attachment could be a simple *boom*; it might also be part of an elaborate hardware

[2]Unsprung weight is the part of the machine weight that is below the spring suspension. Vehicles with low unsprung weight have better performance and comfort at roadway speeds.

set that extends the hook reach or a device that expands the functionality of the crane. What is called a boom in the U.S. mobile-crane lexicon is a *jib* in the British.

Latticed Booms

A long slender latticework is the iconic shape usually used to depict a crane boom. For a century it was the dominant form, though today it is perhaps eclipsed by the telescoping box. The lattice form provides excellent strength in relation to weight, an important characteristic because the weight of the boom subtracts from the lifting capacity of the crane. Booms designed for lifting service are usually fabricated using high-strength steels, often tubular, to keep the weight low. Those intended for duty-cycle applications such as dredging or operating a clamshell are likely to be stockier and use mild steels. *Lattice booms* are available with capacities considerably greater than 1000 tons (900 t).

The common latticed boom is built up from sections that are designed for easy trucking and assembly at the site. It is made up of a *base section* or *butt section*, any number of *inserts* of varying lengths in the center, and a *tip section*. A *boom butt* and *boom tip* assembly without inserts is referred to as a *basic boom* and is the arrangement with which the crane's maximum lifting capacity is achieved. A manufacturer might offer a *tapered tip* and several specialized alternatives. One popular option is a light duty tip that improves load ratings when using one or two parts of line, common on a long boom. Another is a *hammerhead tip* that is used (Figure 2.17) for heavy close-up picks

FIGURE 2.17 A two-crane lift with hammerhead-tipped booms. The hammerhead tips hold the load away from the boom by means of offset sheaves to permit the load to be raised clear of the booms. (*Manitowoc Company, Inc.*)

as might be needed for such applications as rigging machinery into a building. These are stubby in shape because the boom tip sheaves are well offset from the centerline, creating more clearance between the load and boom. Yet another is the open throat tip, usually tapered, which providing open space in the front for load clearance.

A boom tip can have a secondary tip called a *rooster* (Figure 2.18) to carry an *auxiliary fall.* The main fall carries heavier loads whereas the lighter faster *whip line* running over the rooster carries smaller ones. In steel erection, for example, the main fall is used to pick bundles and the auxiliary to set small members such as bar joists and purlins.

Figure 2.18 Rooster sheave on a boom tip above Times Square in New York City. Main and auxiliary hoist lines are visible. (*Lawrence K. Shapiro.*)

With various combinations of *boom inserts*, as specified by the manufacturer, any boom length can be assembled in increments that normally are 10 ft (3 m). With few exceptions, mobile cranes are capable of raising the maximum boom lengths assigned to them without assistance from other equipment.

Smaller cranes carry basic booms of 40 ft (9 m) or so, but for large cranes they can be 70 to 100 ft (20 to 30 m). Maximum boom lengths vary with the crane model; the longest booms in use for stock model cranes are about 400 ft (122 m). Most boom cross sections are limited to 8 ft (2.44 m) square so they do not require special road permits in most jurisdictions. Long booms are shipped in sections on flatbed trucks, but some cranes carry short booms of 100 ft (30 m) or so folded for quick jobsite erection. Many cranes of small to midsize can self-assemble their booms; however, larger ones need an assist crane.

In comparison with that of a Chicago boom, the topping lift for a mobile crane is positioned low, particularly on long booms, thus inducing high compressive loads in the boom. To help reduce this load, and thereby increase capacity, a *live mast* is used on some cranes. The mast, shown in Figure 2.19, serves the single purpose of raising the topping lift to reduce boom compression.

The term *topping lift* is not used in the mobile-crane industry. It is replaced by the terms for its two components: the fixed-length ropes fastened to the boom tip are called *pendants,* boom guy lines, or *hog lines,* and the running lines that produce the luffing motion are called *boom-hoist* ropes. The frames in which the boom-hoist sheaves are mounted are called the upper and lower *bail*. Luffing-system components can be seen in Figures 2.18 and 2.19. Brakes on the boom-hoist winches are spring-set so that they automatically engage when the system is not in motion. They are also equipped with dogs to lock the winch against accidental release of the boom.

Telescopic boom cranes have drastically eroded the market share of latticed booms for lift crane service, particularly truck mounts. For short-term work, the assembly time and trucking of boom sections puts latticed booms at a disadvantage. Skills needed to operate the older machines equipped with friction drives are harder to find with each passing year. Nonetheless, crawler cranes, especially duty cycle applications, will continue a demand for latticed booms; they might lose iconic status, but they are not about to disappear.

Fixed Jibs

Jibs are generally used to increase the vertical reach of a crane, although in some instances they increase horizon *reach*. (The U.S. jib is called a *fly jib* in the EU.) A *fixed jib* is a lightweight boomlike structure mounted at the boom tip, as shown in Figure 2.20 There are two fixed jib styles, suspended and stowaway, associated with lattice booms and telescopic booms respectively.

FIGURE 2.19 The live mast on these crawler cranes serves to increase the angle between the boom and the boom guy lines, and thus reduce the compressive force in the boom and increase structural capacity. (*Link Belt Construction Equipment Company.*)

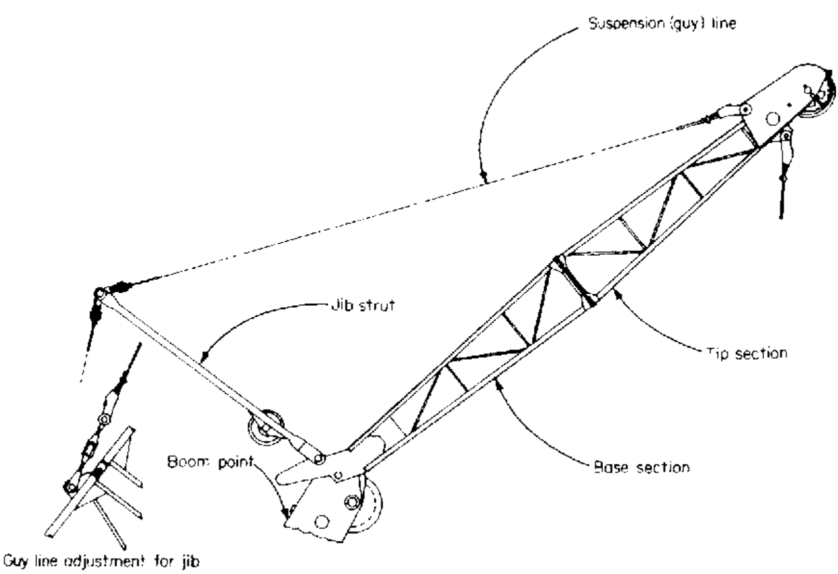


FIGURE 2.20 A typical jib mounting, showing the jib in its shortest configuration. The *jib offset* is 20° from the boom.

A *suspended jib* is supported from a *jib strut* by fixed-length *forestay* ropes and—both terms borrowed from the nautical lexicon—*backstay* ropes secure the strut to attachment points on the boom. Fixed jibs can be mounted to remain in line with the boom centerline or the *jib offset* can be an angle up to 45°. Figure 2.21 shows a long fixed jib at a small offset angle mounted on a long boom, for a lift height some 440 ft (134 m) above ground.

Stowed on brackets located on the side of the base section, a *swingaway jib* travels on a telescopic boom and requires no additional

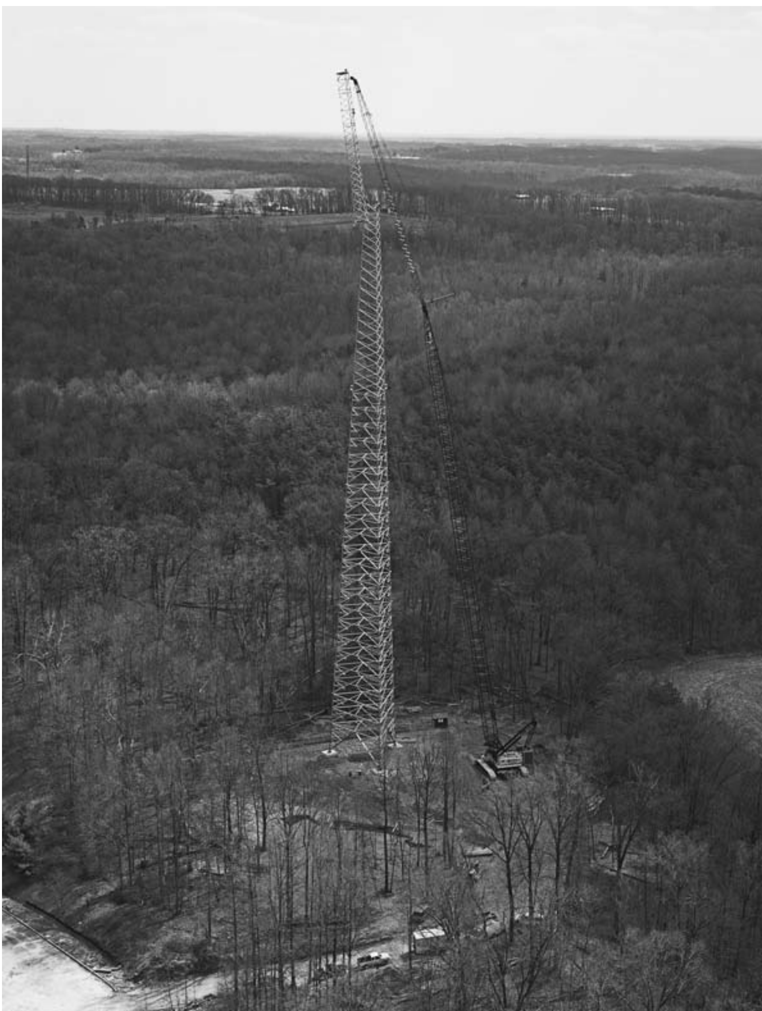


FIGURE 2.21 A 300-ton crawler crane with 435 feet of boom and jib places the final portion of a television tower. (*Link Belt Construction Equipment Company.*)

trucking. Some are hinged at midlength to allow longer overall length than the base section. Sufficient space must be made available at the work site to allow unfolding of this extra section. Another design innovation is the addition of a hydraulic cylinder that allows the offset angle to be varied by the crane operator after the boom is aloft. This feature mimics the luffing jib (see the next section), though unlike with a luffing jib, an angle change under full load is often not possible.

A fixed jib is a relatively low-capacity structure in comparison with the boom on which it is mounted. In particular, jibs are not very strong with respect to side loading. Jib lifts are most often made with a single part of line, although they can be rigged for two, and in some cases more parts of line. A typical crane is arranged so that it can carry the main hook suspended from the boom tip while another hook is suspended from the jib. The jib line is then used for light-duty service and the hoist rope it carries is called a *whip line*.

Luffing Jibs

The most obvious physical difference between a fixed and a *luffing jib* is that the stationary backstays are replaced by a suspension of running cables that allow the jib to articulate under the full control of the operator. The lattice-boom crane will need at least four drums to make full use of the boom and jib: one to operate the main boom, another for luffing the jib, a winch for a boom fall, and another for the jib fall. On some models, the luffing jib allows an additional fixed jib or light-duty extension to carry a *whip line.*

Whether mounted on a conventional or telescopic boom, a luffing jib configuration can dramatically expand the versatility of a mobile crane, creating a large working envelope with robust lifting capacities. On an open site, it can sweep at long radii. In congested urban or industrial sites the main boom can be set at a high angle to clear obstructions such as buildings where as the jib reaches to set loads over those obstructions, as in Figure 2.22.

The luffing jib can be longer than the boom. It is assembled on the ground in line with the boom and raised up like a jackknife (Figure 2.23). The space needed for assembly is slightly longer than the combined length of the crawler or carrier plus the boom and jib. Whether in a city or on an industrial site, space must be allocated to allow assembly and dismantling; the boom-jib combination might also need to be set down for contingencies such as inspection, repair, or high winds.

A mobile crane with a *luffing attachment* might be used for work that would otherwise be performed by a tower crane or derrick. The relatively complicated and expensive initial installation required of the latter machine types generally renders them less economical for assignments of less than a few months, whereas a luffing jib on a mobile crane might be assembled in the morning and complete its lifts in the afternoon. Moreover, mobility confers operational advantages.

Figure 2.22 Liebherr LTM 1500 telescopic *truck crane* at the New York Stock Exchange lifting mechanical equipment on the roof with the luffing jib. With the help of the guy attachment, visible on the back of the boom, this configuration offers high lifting capacity reaching far over the front façade. (*Jay P. Shapiro.*)

One drawback is that the *minimum radius* of the jib fall is relatively large; this makes close-in work difficult. Sometimes the main fall can perform close picks, but another solution on some models is a *midfall* deployed from the midsection of the jib. Work at close quarters should command extra attention during planning, especially at tight urban and industrial sites that lack "wiggle space."

Tower Attachments

Some older mobile-crane models can be equipped with an optional front-end attachment composed of a fixed vertical tower pinned at the position on the superstructure where a boom is ordinarily mounted. A luffing boom is affixed at the tower top, and a jib is sometimes fitted at the end of the boom.

Figure 2.23 Liebherr LTM 1500 telescopic *truck crane* with the luffing jib being raised up after assembly on a city street. The luffing jib is out in front of the retracted boom; its tip has a wheel that allows it to be rolled on the ground as the boom is raised. When the angle between the boom and jib is 90°, the jib tip is lifted off the street. After the jib is fully aloft, the boom is telescoped as needed. (*Jay P. Shapiro.*)

The tower attachment creates a hybrid between a mobile crane and a tower crane, offering the mobility and relative ease of assembly of the former along with the reach over obstructions afforded by the latter. One of these may be advantageous at an urban or dense industrial work site where the crane must be placed close to a structure and still be able to raise loads over and beyond (Figure 2.24). These devices are not suited for the heaviest lifts, but maintain good lifting capacities at long radii as compared to conventional booms.

Figure 2.24 A crawler crane with a tower attachment. The book is folded down in an out-of-service position. The boom is not working because the concrete placing the boom in the foreground is in operation. (*Mathieu Chaudanson.*)

Towers and booms are made up of inserts of various lengths so that a number of tower-height and boom-length combinations can be assembled. The manufacturer's instructions generally include an advice that the tower boom must be lowered and fastened to the mast when the crane is out of service (an exception is when very high winds are expected, in which case the mast must be lowered to the ground or guyed). Therefore, the mast should be longer than the boom. However, there are exceptions, in which case the crane manufacturer issues guidelines for securing a boom that cannot be fully folded down.

Most tower attachments require a small *assist crane*, commonly a telescopic machine, for unloading and placing tower and boom sections, and for assembling the rigging and the counterweights. Tower attachments utilize unique rigging and require many adjustments and the assembly of many special parts. Typically, assembly takes more than a standard workday for the larger models. Some models, the very tallest, may require a large assist crane to help raise the tower and boom off the ground.

A tower attachment is productive and versatile, but its niche in the market has been displaced by luffing jibs mounted on lattice booms. Set at a near-vertical angle, a boom on one of the latter serves the same function as a tower.

Heavy Lift Attachments

Mobile-crane manufacturers have devised various schemes that extend reach and lift capacities beyond the envelope of conventional machines, sometimes dramatically. These arrangements require special attachments that are not part of the everyday equipment set, and time to set up; substantial space and site preparation may be needed. They may introduce working procedures that are too cumbersome for all but the most demanding lifts and that require more training and attentiveness than regular operations. Moreover, as dramatically as the outer reach of the lift envelope is improved by these systems, backward stability considerations may curtail lifting at close quarters.

Extraordinarily heavy lifts demand large numbers of parts of line in the *boom suspension* and the load fall. Operations are correspondingly slow, but slowness can be an advantage when lifting massive loads of high value.

Two general principles are behind all of these arrangements, either singly or jointly: additional overturning resistance is provided to lessen or eliminate the stability limitations of the mobile carriage, and stress on the boom is reduced to allow an increase to its working capacity.

The first principle is illustrated by a guyed boom, an option available on some models. The advantage over ordinary guy derricks is in the mobility of the machinery platforms between setups. Guy derrick attachments for truck- or crawler-mounted cranes have capacities to about 600 tons (544 t) and booms to near 400 ft (122 m).

The second principle is illustrated by the use of backstay struts and cables to stiffen the cantilevers on a telescopic boom. A few telescopic crane models have this option (Figure 2.25). The backstays, requiring a few hours of additional installation time, compensate for the inherent structural limitation and limberness of a cantilever. Load capacities that might otherwise be limited by stress or deflection are enhanced.

FIGURE 2.25 Liebherr LTM 1500 telescopic *truck crane* with a guy attachment dismantling a heavy structure. The guy attachment stiffens the boom, increasing its structural limitations. (*Mathieu Chaudanson.*)

There are a variety of proprietary arrangements that use both principles to transform mobile cranes, principally crawlers, into lifting leviathans. These setups require considerable ground space and logistical costs, but they are capable of lifts to as much as 2000 tons (1800 t) and reaches in excess of 700 ft (213 m). Boom stress reduction is achieved by raising the boom hoist suspension on a high *live mast*. Stability against overturning is enhanced by supplementary counterweights placed behind the superstructure. The tray carrying the supplementary counterweights might roll on the ground or on a circular track. As the crane swings, the counterweight tray and superstructure rotate in together about the center pin. Though there are some cranes designed from the ground up with these characteristics, more commonplace are kits called *heavy-lift attachments* that

Figure 2.26 A Manitowoc 4100 W crawler crane, fitted with a Ringer attachment, at work on Metrorail in Dade County, Florida. Note that the boom and live mast are stepped onto a carrier riding on the large-diameter ring that constitutes the machine's tipping fulcrum. The added concrete counterweight blocks also ride on the ring. The mast gives the boom added structural capacity, while the additional counterweight and the placement of the tipping line well in front of the machine increase its stability. (*Manitowoc Company, Inc.*)

adapt conventional crawler cranes to the same end. The Manitowoc Company's Ringer (Figure 2.26) is one such adaptation; the boom is stepped onto a carrier extending from the superstructure and the carrier rides on a track. Counterweights are placed on a rearward-extending tray that also rides on the track. A high-latticed *live mast* is also used.

A newer variant of the heavy-lift concept uses a tray of *auxiliary counterweights* suspended from the *live mast*. The tray stays aloft while the load is on the hook; if the crane swings, there must be a clear path for the suspended counterweights and when the load is released the

Figure 2.27 A Demag CC2800 crawler crane with live mast and a tray of auxiliary counterweights suspended from it. The enhanced capacity of the arrangement is needed for heavy lifting assembling this offshore oil platform. (*Lawrence K. Shapiro.*)

weights must be set down on the ground (Figure 2.27). For subsequent picks, the tray is repositioned. On some models the counterweight tray rides on a wheeled carriage fitted with a complex steering mechanism; the wheels can be oriented for either *slewing* or travel (Figure 2.28). On other models, the crane turns around to pick the tray to put it at the location corresponding to the next pick. Either way a great increase in lifting capacity as achieved, but the operation of the suspended counterweight is considerably slower and more demanding of attention than a conventional arrangement. It should only be used when needed.

That last observation comports with a broader one: operations associated with heavy-lift attachments need rigorous planning and organization.

Telescoping Cantilevered Boom Cranes

The term *cherry picker* was at one time befitting of a telescoping boom crane, but the modern machine of this type has physically outgrown its nickname; the term is more aptly used now for truck mounted bucket lifts. The term *hydraulic crane* is ambiguous as it might just as well describe some power trains of machines that are not telescopic. For want of something better, *telescopic boom crane* is the term that will

Figure 2.28 A Manitowoc 2250 crawler crane with a proprietary wheeled counterweight tray and a live mast. The wheels are shown oriented for *slewing*, but they can also be repositioned for travel. (*Manitowoc Company, Inc.*)

be used here. All of these cranes use hydraulic rams called *boom hoist cylinders*, sometimes singly but usually in a pair, for luffing the cantilever boom (Figure 2.29).

Telescopic *truck crane*s are produced up to 1320-ton (1200-t) capacity with booms that telescope as long as 328 ft (100 m) in length. Latticed extensions, fixed and luffing jibs are furnished to extend height and horizontal reach.

These booms can be mounted on any type of carrier or base, but truck, rough-terrain, and all-terrain carriers predominate. A crawler

FIGURE 2.29 A 90-ton hydraulic *truck crane* lifting a military vehicle. The swing-away jib is stowed on the side of the boom. Lifting is from a block suspended from the main fall while the whip line is idle. (*Link Belt Construction Equipment Company.*)

base is offered by some manufacturers; though apparently inconsistent with the concept of a telescopic crane, the crawler base is useful for some on-site work and for pick-and-carry operations (Figure 2.30).

The world of telescopic cranes has seen a steady march of design evolution since the first one was introduced in the middle of the twentieth century. Self-weight has always been a limiting factor for cantilevered booms; therefore much of the thrust of innovation has been to reduce weight so as to permit longer booms and greater lift capacities. Boom shapes offering improved structural efficiency have been introduced and clever telescoping systems have been devised. Older designs used one cylinder for supporting each extending

FIGURE 2.30 Once a rarity, a telescopic boom mounted on a crawler base is not uncommon for site work, particularly pick and carry. (*Link Belt Construction Equipment Company.*)

section so that multiple cylinders filled with hydraulic fluid made up a large part of the boom weight, but now on some models some cylinders are replaced with ropes or chains with pulleys, and many use a single hydraulic cylinder for extending and retracting all the boom sections. Advances in control technology have allowed automation of the procedures for extending and locking boom sections to attain the desired length.

Booms are made up of nested segments, starting with a *base section,* ending with a *fly section* and having one or more *midsections* between. At one time the typical cross section was rectangular or trapezoidal but now contours of boom cross sections tend to be more complicated because design and fabrication methodology allow for greater optimization of the shape. The fly is topped by the boom *head section* which carries the upper load and lead sheaves.

Old style booms have each telescoping section coupled to an *extension cylinder,* although fly sections are often extended manually or power assisted by other means. The extended fly is then locked into place by a pin. Extension cylinders support the compression load on the boom. Surprisingly on some older models, long cylinders may buckle under load, but on buckling they come to rest in contact with a wall of the boom section. This “stiffens up” the cylinder, enabling it to carry a greater load. When the load is removed, the cylinders relax and straighten; this explains why in some instances the booms cannot be extended while under load.

A weight-saving variation of the traditional multicylinder arrangement uses light-duty cylinders to extend the boom sections (Figure 5.40a). Once extended, the sections are locked into place with remotely activated pins.

Weight can be reduced even further when one extension cylinder does the telescoping for all boom segments (Figure 5.40b). Sections are sequentially telescoped and locked into set positions by remotely activated pins. When the crane is ready for lifting, load is transmitted from one section to the next through the pins, and the extension cylinder does not contribute to the support of the boom. Unlike traditional models, the boom cannot telescope while under load, and the working boom can only be set at the fixed lengths established by the pin locations.

In some booms, ropes and pulleys inside the boom are set up in combination with an extension cylinder so that one cylinder telescopes multiple sections. The extension cylinder drives one section directly while pulling ropes to extend one or more others, and the sections move together in lockstep (Figure 5.40c).

The boom sections telescope, one within the other, on slider pads or rollers. The pads wear, particularly in dusty environments, allowing increased clearance, or *slop,* to accumulate between sections. Routine inspection and occasional replacement of bearings or pads is required, as slop can impede telescoping or, if uneven from side to side, can put a twist in the boom. Slop can also increase load radius without changing the reading of the boom-angle indicator.

A *latticed extension* is often provided to augment the reach of a telescopic boom. Such an extension is called a *swingaway jib* when it can be folded sideways and latched to the *boom base* section for storage (Figure 2.20). Though mounted in-line with the boom centerline, some latticed extensions can be offset. Because of their light weight, load ratings may be greater on a latticed extension than on the telescopic boom alone while at the same radius. A latticed extension is mounted on the crane in Figure 2.31.

A telescopic crane can be ready for work soon on arrival at the jobsite. Maneuvering into position, cribbing the outriggers, leveling the crane, and perhaps reeving the hook block could be all that is required before commencing with lifts. A fly section or fixed jib, if needed, takes a little more time to deploy. Even a complex attachment such as a luffing jib takes only a few hours to assemble. Thus a great advantage of this crane style is its availability for work with little investment of time or labor. These advantages overcome the high cost in relation to lift capacity giving telescopic cranes an expanding share of the crane market.

Even large telescopic cranes enjoy this advantage. Partial disassembly is usually required for transit. Counterweights are shipped apart and self-installed. On the very largest machines, other components such

Figure 2.31 A 75-ton hydraulic *truck crane* lifting with a latticed extension lifting an air conditioning unit. (*Link Belt Construction Equipment Company.*)

as boom sections and winch packages may be removable for separate shipping, too. Nonetheless shipping costs are moderate for the capacity of the crane and assembly times are reasonable. In comparison with conventional cranes of similar capacity, these machines still get to work quickly after arrival at the site. Their high rental costs, however, favor them for short-term assignments.

Telescopic cranes are often worked in tight quarters. In roadwork, for example, lane restrictions can make full outrigger extension infeasible. One solution is to utilize the "on-tires" lift rating charts provided with most models. However, this leads to a substantial reduction in lift capacity.

In response to industry demand, crane manufacturers offer partial outrigger extension as an enhancement over "on-tire" ratings. Not all cranes can be so used, however. Only those cranes specially arranged and rated by the manufacturer are permitted these capabilities. Although cranes rated for partial outrigger extension can work in less space, the *tail swing* is no less and there are other drawbacks as well. These matters will be more fully discussed in Chap. 5.

Despite the steady march of innovation that has pushed telescopic cranes to displace traditional types out of numerous work assignments, there are some limitations that are unlikely to be overcome any time soon. In particular, their delicate machinery and finely tuned structures are not suited to duty-cycle operations. Though it is not inconceivable that designers could work around present limitations, the exigencies of the marketplace are not pushing in this direction.

Wind and Mobile Cranes

Wind must be mentioned in any discussion of mobile cranes. Among all the factors in operating a mobile crane, wind is perhaps the most elusive because of its capriciousness, compounded by the difficulty in gauging and managing it. Wind can compromise the crane structurally, reduce stability, or rob the operator of the ability to maintain control of the load. Planning for wind can be largely standardized for tower cranes and derricks because these equipment types operate according to circumscribed plans with fixed locations and well-defined motions. Mobile cranes do not fit the same pattern: their configurations and working environments are so varied that generalized guidance with respect to wind is elusive. But that is not to say that planning for wind is out of the question, nor is it to say that sound instructions cannot be given.

Guidance for crane use with respect to the wind fits into two categories: the first describes the limitations of lifting loads in the wind; the second explains when a crane should be taken out of service and provides the procedures for doing so. Additional information about mobile-crane operations with wind is provided in Chaps. 4 and 5.

Preparing a mobile crane for a storm is easy when the boom can be readily telescoped down or lowered to the ground. Long booms and complicated configurations pose a larger challenge. A mobile crane should not be placed on a site without having a means to take it out of harm's way in advance of a storm. In a congested urban area or industrial site, this can be a difficult proposition.

The recommended practice for designing latticed crane booms in the United States[3] makes no direct provision for the wind velocity the boom must endure when the crane is *out of service*. Instead, it

---

[3]SAE Information Report J1093, Crane Boom Systems–Analytical Procedure, Society of Automotive Engineers, Warrendale, Pa., March 1994.

stipulates that the manufacturer shall specify the velocity at which the boom must be lowered to the ground or otherwise secured. Manufacturers' instructions usually call for luffing jibs and similar attachments to be jackknifed down to ride out moderately severe storms. For the most severe storms, however, lowering the boom and jib to a completely prone position may be required.

A few years ago a hurricane threatened New York City. At the time, a crawler crane with a long boom-jib combination was at work on a busy Midtown avenue. There was no way to secure that boom with confidence. The only viable choice for safety was to lay the boom down in coordination with the police and traffic authorities; the boom and jib crossed and completely blocked off two crosstown streets. Users of long-boom cranes must be prepared to take actions such as this at any time at each and every jobsite when high winds are a threat.

2.4 Tower Cranes

Much like a mobile crane, a tower-mounted crane moves loads by executing three motions: the hook is raised and lowered by means of a winch and fall, carried in a circular path by the swing gear, and carried in a radial motion by either luffing the jib or rolling a *trolley* carriage on its underside. The simplest of tower cranes have only these motions, but more complicated arrangements include mechanisms that allow the base to roll on a track, the crane to change elevation by climbing, or the jib to articulate on a hinge point.

In the twentieth century, tower crane and mobile-crane industries developed independently on opposite shores of the Atlantic Ocean in response to needs and cultural proclivities of their respective lands. Simply put, Europe was the realm of tower cranes, North America the realm of mobile cranes, and the rest of the world a patchwork of the two. The globalized marketplace and rapid diffusion of knowledge in the present century have eroded differences in construction practices worldwide so that the selection of a crane type now is more likely to be determined by its suitability to the work than by the country where the work is taking place. At least that is the broad picture; contractors in some localities cling longer to traditional practices than others.

Tower cranes are the lifting machines of choice worldwide for most mid- and high-rise building construction. They are used also on expansive sites where the broad hook sweep and the relative ease of coordinating multiple tower cranes is an advantage. There are niche markets for these as well; cable stay and suspension bridges, offshore oil platforms, and power plants are some examples.

In most of the world outside U.S., small tower cranes are used for modest-size residential and commercial projects. Many of these rigs are self-erecting machines that are pulled to the site by truck.

In North America, similar work would utilize a small telescopic crane. Gradually, however, self-erecting tower cranes are penetrating the U.S. market. There is some doubt whether these machines should be classified as tower cranes; though their operating motions fit the pattern, their deployment—setup time, operation, inspection, maintenance, and demobilization—is more like that of mobile cranes.[4]

Freestanding *hammerhead tower cranes* range up to about 300 feet (91 m) in height; for *luffing tower cranes* the limitation is less. Though most tower cranes free-stand and remain at a fixed height, various self-climbing arrangements permit a tower crane to attach to a building under construction and rise with it. With such supplemental means of support, a tower crane can ascend to any building height. Very high *line speeds* up to 1000 ft/min (5 m/s) available with some models yield good production rates even at extraordinary heights. Some machines can operate in winds up to 45 mi/h (70 km/h), which is far above mobile-crane wind limits.

Lift capability of tower cranes is gauged by a moment rating expressed as tonne-meters. The tonne-meter rating is obtained by multiplying *rated capacity* in metric tons by the working *radius* in meters. This is done according to a method that averages a range of boom lengths and working radii. The smallest machines used for light construction have ratings of about 20 meter-tons and the very largest in production exceed this by a factor of about a hundred. Most used for heavy construction are in the range of 150 to 650 meter-tons.

The cost associated with installing and removing a tower crane is small for the self-erecting type but can be considerable for most others. At minimum, those costs would include trucking, hiring a rigging crew and an assist crane, construction of a foundation, electrical hookup, and the services of a trained technician. More complicated installations such as those which climb have considerable additional expenses. The high costs of installation, as well as the considerable investment of time and planning, make all but the smallest tower cranes a tool for longer-term projects where these expenditures can be amortized.

Excluding self-erectors, tower crane designs follow the "erector set" concept; that is, they are composed of multiple components of common design. Components connected together by pins or bolts are often interchangeable among an array of models.

As a tower is a cantilever subjected to high-bending moments that shift and reverse, each tower leg must be designed to resist alternating compressive and tensile forces. The usual stress levels and the expected number of lifetime loading cycles are not exceptional; thus leg design does not pose a difficult fatigue problem for designers.

---

[4]At the time of writing this, the ASME standard B30.29 for self-erecting tower cranes was in development.

The connections between tower sections are another matter, being a perpetually nettlesome design and maintenance concern. Some designs use bolts and others use pins to join the legs of one section to the next. Bolts must be preloaded to carefully controlled levels. Failure to establish and maintain the preload can result in bolt fatigue failure. The alternative pin-type connection requires tight fabrication tolerance; imperfect towers can be either too sloppy or difficult to fit together. They also must be carefully designed to avoid the pitfall of premature fatigue failure.

With the exception of a few diesel-powered machines with hydrostatic drives, tower cranes are powered by electricity and motors driving all the machinery are almost universally electric, too. The hoist motors on older machines have high- and low-speed ranges with stepped increments in each range. More recent models have variable-frequency drive or other forms of continuously adjustable speed motors with friction or eddy-current brakes and creep speed. Automatic acceleration for all motions is typical on many cranes. Remote controls are sometimes offered.

A tower crane is said to be *top-slewing*—*slewing* being the preferred tower crane term for swing motion—if the swing circle is mounted near the tower top as in Figures 2.32 and 2.33 and it's said

Figure 2.32 A saddle jib tower crane in a small U.S. city. All of the key components are visible including a climbing frame. (*Morrow Equipment Corp.*)

FIGURE 2.33 Pecco SN160 luffing boom tower crane at work as an external climber braced to the structure in an urban canyon of downtown Manhattan. The mast is braced to the building. Just below the crane superstructure is the climbing frame used to raise the crane for insertion of another tower section. (*Lawrence K. Shapiro.*)

to be *bottom-slewing* if the swing circle is near the base as in Figures 2.34 and 2.35. Bottom-slewing machines might have become largely obsolete had the concept not been revived by recent developments with self-erecting machines.

Jib Types

In the U.S. tower crane lexicon the terms *boom* and *jib* are used more-or-less interchangeably, though *jib* is more likely to be applied to a

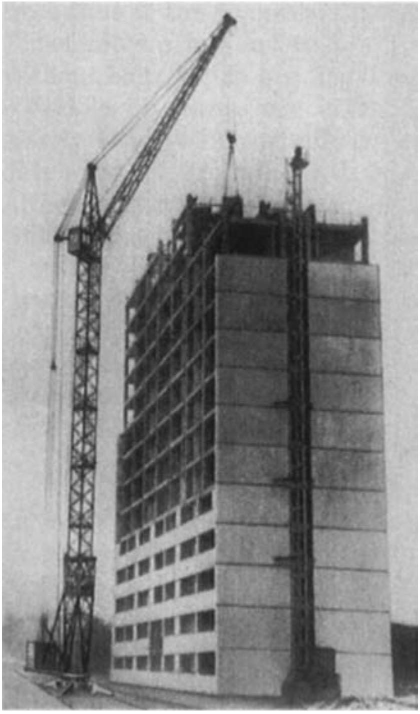


Figure 2.34 Luffing jib tower crane built in the former Soviet Union. This rail-mounted crane slews at the bottom. (*VNIISTROIDORMASH.*)

hammerhead style and *boom* to a luffing crane. Most everywhere else, "jib" is the common term for both.

Three jib types are in general use:

1. Those mounted to the tower in a horizontal or slightly upward inclined position are called *hammerhead jibs*. The traditional hammerhead form is called a *saddle jib*, illustrated in Figure 2.32. The *flattop* style shown in Figure 2.36 (sometimes called *topless*) has become more popular of late.
2. A *luffing jib* is mounted with a pivot and derricking system to a machinery deck on the tower, as shown in Figure 2.33; it is similar in many respects to a mobile-crane superstructure mounted on a tower.
3. *Articulated jib* (Figure 2.37) is usually a hybrid of a saddle jib and a luffing jib, with both a luffing capability and a hook trolley. The articulation occurs in the middle portion of the jib where a pivot allows the segment to be inclined differently.

FIGURE 2.35 Self-erecting tower crane with a telescoping mast and articulating jib. The crane is bottom *slewing* and stabilized by counterweights. (*Morrow Equipment Corp.*)

Hammerhead Jibs

A hammerhead jib carries a trolley that traverses the bottom of the jib to change the hook radius. Its upper load block is integral with the trolley. With the load hoist rope dead end fixed at the jib tip, the hook maintains a constant elevation as the trolley changes radius. Trolley travel (hook radius) is controlled by an independent winch and rope system.

An opposing strut, called a *counterjib*, projects opposite the jib to carry the counterweights while also supporting the load winch, power plant, and control panels. The operator's cabin is typically mounted just below the jib but above the slewing circle, allowing the operator a full view of the load and the trolley.

Figure 2.36 A Cluster of flattop-style tower cranes near an airport flight path. The tower heights are offset so that the hook of one crane can clear over the jib and counterjib of another. All the cranes must be free to weathervane when not in use. (*Terex Comedil.*)

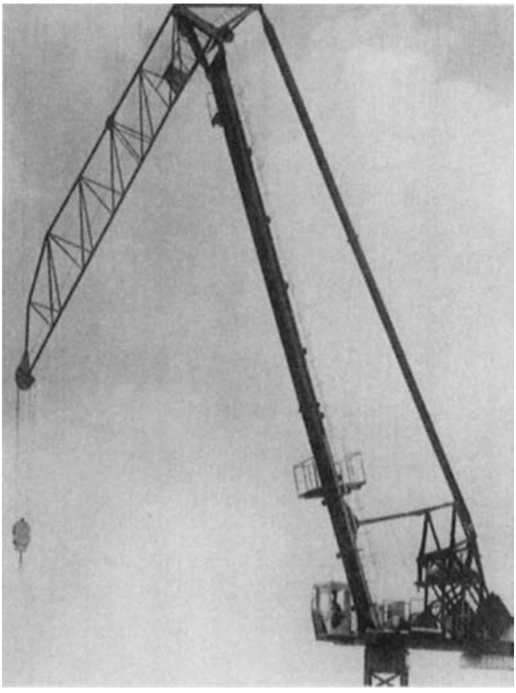


Figure 2.37 The upper portion of an articulated jib tower crane. A counterweight is used to aid in luffing, and the hook can be kept level as radius changes. (*F. B. Kroll A/S.*)

With the saddle-jib style, a small *tower top*—some call it a *rooster*—anchors pendants that suspend both jib and counterjib. These pendants may be wire ropes or steel bars. The top tower also houses the lead sheaves, giving sufficient distance to the winch to maintain proper fleet angle.

The flattop style of hammerhead jib has no tower top. Its lesser height and lack of pendant suspensions make it easier to erect than a saddle jib. The reduced height is advantageous in some other situations such as for installation in height-restricted zones near airports or where one crane swings over another.

With few exceptions, no obstruction can be permitted to prevent a hammerhead crane from slewing through a full 360°. The crane must swing freely with the wind, or allow *weathervaning,* when out of service. It is not designed to accommodate severe storm winds taken from the side of the jib but is arranged to slew freely in order to present its smallest profile to the wind. Thus, at the completion of a work shift the operator must leave the crane with the *slewing* brakes disengaged. Where multiple hammerheads have overlapping swing circles, they must be offset at different elevations to avoid interference (Figure 2.38). For hemmed-in sites, however, some cranes can be furnished with short jibs that can be locked in position to endure storm-wind forces.

A variety of *operational aids* and *limiting devices* are provided with hammerhead tower cranes; some are mandated by codes and standards whereas others are elective. A device that senses line pull and cuts winch power when the permitted level is exceeded, together with another sensor mounted on the jib pendants, is intended to prevent overloads. Limit switches prevent overtravel of the trolley at either end.

FIGURE 2.38 Overlapping hammerhead tower cranes must be offset in elevation to allow overswing so that there is full hook coverage of the site. (*Morrow Equipment Company.*)

Another limit switch prevents the load block from striking the upper block, an occurrence known as *two blocking*. None of these devices should substitute for attentiveness and good judgment by an operator.

Avoidance of collisions between jibs and load suspensions is a concern on large projects that deploy multiple hammerhead cranes. Protocols for communication and coordination among operators are always necessary. Automated devices are available to assist. These devices are of two general types: *zone protection systems* that restrict where the crane hook is permitted to travel and *anticollision systems* that actively sense and prevent conflicts. Though some of the more advanced devices work in three dimensions and communicate with other cranes, no system should be viewed as a replacement for eyes or minds.

Hammerhead cranes are available in a complete range of sizes as illustrated in Figures 2.39 and 2.40. Though self-erectors and luffing jib

Figure 2.39 A hammerhead tower crane of the flattop style, freestanding on a chassis base stabilized with ballast. (*Manitowoc Company, Inc.*)

Figure 2.40 A 2000-tonne-meter tower crane freestanding on an industrial site in Mexico. The tower sections, too large for conventional trucking, are designed to be broken down into small pieces. (*Morrow Equipment Company.*)

models have made strong inroads, hammerheads are still the dominant tower crane type worldwide. Given their simplicity and relatively low cost, this is unlikely to change soon.

Luffing Jibs

The shortcoming of a hammerhead crane is that it requires a wide swath of space for the boom to operate; and when the crane is not operating, the boom must have clearance over a full circle so that it can act like a weathervane. Built-up areas of cities and industrial sites sometimes do not have the requisite clear space. Increasingly, moreover, government authorities enforce *air rights* provisions that limit the right of a crane user to occupy the space even for short periods over adjoining land. These difficulties have provided a market opportunity for tower cranes with luffing jibs and have made them dominant in some highly built-up cities.

A luffing jib has several distinct advantages over a hammerhead. With a higher vertical reach, it can accomplish the same task with a lesser tower height. When arranged for climbing, its climbs do not need to be scheduled to synchronize with other cranes that overlap. But the luffing jib machine is heavier than a comparable hammerhead and unlikely to match either the minimum or the maximum hook radius; moreover, the mechanical works of the luffer are more complex and its jib presents a more formidable task for erection.

The oldest luffing jib tower cranes were bottom *slewing* as can be seen in Figure 2.34. A tower on one of these old machines is mounted on a slewing platform which also carries the power plant and the counterweights. The jib is supported by pendant ropes anchored to a pivoting strut at the tower top. Luffing ropes control the angle of the pivoted strut; these ropes run roughly parallel to the tower and act to relieve the tower structure of most of the bending load; thus the tower on this type of crane is essentially a compression member. Modern self-erecting tower cranes have much in common bottom slewing tower cranes in their general arrangement.

A modern version is top slewing, with a rotating superstructure supported on a slew bearing atop a mast as shown in Figure 2.33. The boom and luffing mechanism is similar to a latticed boom mobile crane. Machinery and counterweights are mounted on the counterjib, and counterweights may shift back and forth in synchrony with the boom. Just as with a hammerhead it is required to be set to act like a weathervane when out of service, though there are a very few that are permitted to have the swing locked. There is a danger that the crane boom will not point with the wind, as it must do to reduce exposure, if it is set at too high an angle. An installation planner must be cognizant of the need to allow adequate space for the boom to clear a full swing circle at a suitable luffing angle; the governing clearance might be between the boom and a nearby tall building, or it might be crane to crane.

Articulated Jibs

Once an oddity, jibs of this type have become commonplace with the growing popularity of self-erecting tower cranes. There are so many possible configurations that a general description is not possible other than to say that the jib has a pivot point somewhere in its middle area. The hook might be suspended from a trolley—resembling a hammerhead jib—or from a *head sheave*.

One type of crane has a hinged jib arranged so that the outer portion remains horizontal; this gives it the ability to pass over obstructions. Some models are *level luffing* (Figure 2.37); that is, the hook elevation remains constant as the radius changes. With self-erecting cranes, the hinge in the jib is an accommodation that allows it to fold up for transport and to alter its attitude while in working mode.

Wind and Tower Cranes

Like most mobile rigs, self-erecting cranes can fold up on short notice. All other tower cranes must ride out severe storms. That requirement has induced engineers to identify two principal classes of load combinations germane to the design of these machines: *in-service* loads act on the crane in operation, and *out-of-service* loads occur under the influence of high winds when the crane is unmanned. Some cranes have an additional load case that applies only when the crane is in the process of climbing. Each load case combines the dead weight of the crane with an appropriate dose of wind and other relevant factors. In-service loading is composed of the dead load, a permissible wind

Figure 2.41 A hammerhead tower crane using a suspended tower section as a balancing weight for climbing. There is another tower section hung from the climbing trolley, poised for insertion below the superstructure. (*Morrow Equipment Company.*)

for operation, *slewing torsion*, the lifted load and impact. Out-of-service loading combines dead load and a storm wind that ideally should be suited to the locale where the crane is operating.

Designed to European codes and standards, *freestanding tower cranes* are generally evaluated for storm winds of 94 mi/h (42 m/s) at jib elevation,[5] but design practices with respect to wind may vary with the manufacturer or country of origin. Some areas are prone to more extreme winds. In the United States, these regions are on the Atlantic, Gulf, and Northwest coasts as well as some river valleys and mountain passes. Installation practices should be adjusted for those regions. A heavier mast and foundation might be used or the freestanding height reduced. Alternately, a contingency plan to guy, brace, or remove the crane might be considered.

When in the midst of climbing, a tower crane is in a *balanced* stance that may be particularly sensitive to disturbance from wind (Figure 2.41). In order to assure that the climbing operation is well-controlled, the wind limitation is often set particularly low, usually 20 to 25 mi/h (9 to 11 m/s).

---

[5]Starting in 2010, a consortium of European tower crane manufacturers has agreed to design tower crane installations in Europe on the basis of local wind conditions under the rules of product standard EN14429.

Base Mountings

Almost every tower crane is freestanding, if not for the duration of a project then at least at the starting phase. A freestanding tower requires a base mounting that is either weighted with *ballast* or anchored to a massive structure that can resist overturning moment. There are three common types of mountings: static, traveling, and climbing.

Static Base

A *fixed-base tower crane* has the bottom of the tower firmly attached to a concrete mass or structural framing that can support the imposed crane loads and can in turn transmit those loads to the Earth or to a host structure. This *static base* might be an *undercarriage* carrying ballast blocks (Figures 2.42 and 6.4) or a frame anchored to another structure (Figure 6.18); it could also be a concrete spread footing, a pile cap, or a concrete block tied down with rock anchors as in Figure 6.16. Attachment of tower legs to a concrete base sometimes utilizes anchor bolts (Figure 6.14) or alternately a set of *expendable legs* (Figure 6.5) cast into the footing mass.

The crane delivers vertical load, lateral load, and moments to the foundation. Vertical force is the weight of the crane plus the hook load. Lateral load on a freestanding crane arises strictly from wind, though slew torsion also acts on the anchorage and foundation in a horizontal plane. Overturning moment on the tower derives from the weight of the jib and the counterjib, ropes, trolley, the suspended

FIGURE 2.42 A travel base of a tower crane, with a knee-braced chassis and ballast weights on bogies and rails. The crane is on a parking section of track anchored down in preparation for a storm. (*Manitowoc Company, Inc.*)

load, and the wind. It may apply in any azimuth angle as the crane slews. While the loads acting on tower legs and anchors are maximal when the overturning moment acts on a diagonal with respect to the tower, the least stable direction on a ballasted base is directly over the side of the base.

An out-of-service crane has no load on the hook; the overturning moment is composed of dead load contributions and wind with the latter usually dominating. Tower deflection is a second-order effect that magnifies the moment. When a freestanding tower is tall, out-of-service loading is likely to govern the design of the base and anchorage with tower deflection being a significant effect. Both strength and stability must be considered in the design of most foundations. There is no slew torque imposed by an out-of-service crane that is weathervaning.

As moderate uniform settlement does not affect crane operation or safety, permissible soil bearing values may be higher than those used in general building construction. Differential settlement is another matter; the entire subject of crane foundations is treated in detail in Chap. 6.

On a high-rise project, the freestanding crane might not have sufficient height to complete the structure. This limitation can be overcome by *climbing*. There are various climbing systems, but they broadly fall into two methods: *top climbing* entails adding sections to the crane tower and *internal climbing* uses the rising host structure to support the crane. With the latter, the static base is abandoned when the host structure reaches sufficient height to support a *climbing base*.

Top climbing is the more common of the two methods. A new section is inserted at the top of the tower and that procedure is repeated until the desired tower height is achieved (Figure 2.41). By means of a *climbing frame* employing hydraulic rams, the upper works of the crane is raised to allow a new section to be inserted. Figure 2.32 and 2.33 show cranes with climbing frames. As the tower rises in height, *collars* are periodically attached to brace it, and each collar is connected by a *mast tie* or guys to a host structure. There is virtually no limit to the crane height that can be erected, though cost or availability of tower sections can be a prohibitive factor for a very tall building; a climbing base might be favored in such a situation.

Models differ considerably with respect to labor and time needed to climb. Moreover, the cost of bracing to the host structure can vary. These factors should be assessed during planning as they can have a significant impact.

Traveling Base

A *travel base* comprises a chassis on which ballast weights are stacked, supported on bogies, usually a *walking beam configuration*, with four sets of wheels guided on steel rails. The tower can be rigidly fixed to the platform or knee braces. The bogies have electric drives.

The rails must be well supported so that they will not bend or settle excessively under operating loads. The rails must also be installed level to keep the tower plumb. A section of track, called a *parking track,* is set up with anchorage arrangements and added strength to resist the loads produced by storm conditions (Figure 2.42). Curved track can be used in the travel path. Travel bases enable one or more cranes to be arranged to provide hook coverage throughout a large construction site, industrial facility, shipyard, or port.

A crane on travel base can be climbed in the same way as a static-mounted crane, but of course bracing cannot be used for increasing height. The crane can be assembled to a height easily erected by whichever mobile crane is available or can be self-erected in some cases and then built to full height by top climbing. Travel bases are seldom used in U.S.

Climbing Base

When a crane is installed within the confines of a building under construction, the crane can raise itself by use of a climbing frame and climbing base as work progresses. When arranged this way, it is called an *internal climbing* crane. The initial installation may be freestanding on a static base, and transfer to the climbing apparatus takes place after the structure has been erected high enough to support a jump. The crane can sometimes be located in an elevator shaft, but tower size or scheduling of the elevator installation usually makes this choice impractical. In most instances, temporary floor openings are made to accommodate the passage of the crane through the building.

A climbing base is a steel frame that transmits the weight of the crane to the host structure while the crane is set up for operation. In some arrangements, the base is integral with the mast (Figure 2.43) whereas in others it is a detached component. If it remains attached, the base typically has some form of retractable ends arranged to clear the floor openings as the crane climbs. A detached climbing base is made to disassemble easily so that the pieces can be passed up for reuse at a higher floor.

In preparation for a *jump* support framing is needed at two different elevations. The lower support is the climbing base and the upper is the climbing frame. In some systems they are composed of identical hardware; the climbing frame for one phase becomes the climbing base in the next. At the completion of a climb, the tower is wedged horizontally to the building structure at two levels at a prescribed minimum distance apart. Wedges transfer horizontal forces and resist overturning moment. Some systems have the tower directly wedged against the edge of the floor opening, but usually the contact is mediated through a *wedge frame*. Quite often, the climbing frame and climbing base also act as wedge frames.

Climbing requires a means to transport the balanced crane vertically from one support level to the next. Sometimes hydraulic rams are provided with sufficient stroke to accomplish this in a single stroke unaided by other devices. More often a *climbing ladder* is used in conjunction with rams or some contrivance that serves the same

Figure 2.43 A luffing jib tower crane on a climbing base sitting within the framework of Freedom Tower at the World Trade Center site. (*Mathieu Chaudanson.*)

purpose. Climbing the ladder is carried out by hydraulic rams and dogs or pawls that enable the crane to advance up the rungs. Alternatives to the climbing ladder are systems that pick up the crane with ropes and winches, or mechanisms that climb steel bars or threaded rods. Another arrangement uses hydraulic rams to push down on a central pole; in principle this is a climbing ladder in reverse.

Crane weights are substantial; they can weigh more than 200 tons (180 t). In most installations the climbing-frame and climbing-base reactions will overload the structure of any one floor. It is then necessary to install temporary shoring to distribute the loads to additional floor levels.

The topmost floor, after a climb, is often a wedging floor. With a fast construction sequence, it is critically important to be certain that the structure can support the wedging loads. On a reinforced concrete building this is a question of having developed sufficient concrete strength, while on a steel structure, the floor diaphragm must be sufficiently developed for these lateral loads. The matter is discussed more fully in Chap. 6.

When the structure is completed, there is a tower crane on the roof and no way for the crane to disassemble or lower itself. If a mobile crane can reach, it is usually the most economical tool for removal. An alternative is to install a derrick, usually a stiffleg derrick, on the roof for dismantling and lowering the crane components piece by piece (Figure 2.6). Helicopters are occasionally used to remove tower cranes. However, components of large cranes are generally too heavy for helicopter lifts. Aside from the rigging problems, a safe flight path and a close-by staging area are needed.

Careful study of the building site and host structure should be carried out to select a tower crane solution suited for operational needs and economy. The best location is usually outside the structure with no ties connecting the mast to the building under construction, but that possibility exists only when constructing to a height of 20 stories or so. For higher structures, the relative merits and shortcomings of top climbing and internal climbing should be weighed. This question is treated in Chap. 6.

2.5 Self-Erecting Cranes

Self-erecting cranes have been around for some decades in Europe, but in the U.S. they have only recently advanced from being an oddity to becoming an important and fast-growing segment of the lifting machine market. Functionally these are tower cranes but deploy like mobile cranes (Figure 2.35). A typical machine of this type is driven to the construction site and folds out like an origami crane, strangely mimicking the paper version of its bird namesake. When the work is done, it folds up as easily and is hauled away. If a storm is coming, the crane is folded up readily and need not ride it out. The larger of these cranes have maximum lifting capacities of 4 metric tons and can reach 54 m in height.

A self-erecting crane is pulled to the site and deployed rapidly with a small erection crew. Its base is stood on outriggers Figure 2.44. There may be ballast blocks to impart stability. The telescoping mast, composed of tubular or latticed sections, gives it the ability to operate close to buildings. A larger model may be mounted to an integral truck carrier and have an operating *cab* that rides up the mast. Nearly all self-erecting cranes are bottom slewing.

A crane of this type fits into tight spaces such as a narrow street, alleyway, or courtyard. The saddle jib with an underslung trolley can operate flat or at an inclined angle. This nimble jib can tilt and fold to clear obstacles.

The self-erecting concept is generally applied to cranes that service light construction, but there are exceptions, and the future is apt to expand production to heavier applications. One that presently

Figure 2.44 Small self-erecting tower crane for light construction. The crane is self-contained on a trailer that is towed to the site. (*Manitowoc Company, Inc.*)

stands apart due to its extraordinary size and capacity is the GTK 1100, a specialty machine developed for wind turbine erection. It can lift 77 tons and reach as high as 459 ft. This model also stands apart from other self-erecting cranes because it requires multiple truck loads to be brought to the site, and it must be assembled (Figure 8.3).

2.6 Pedestal-, Portal-, and Tower-Mounted Cranes

Of the equipment types described earlier in this chapter, nearly all are construction cranes. Lift crane installations for construction are by nature temporary, and the equipment is designed to be taken up and moved from site to site with some regularity. Designers contemplated that these machines would operate under conditions typical for construction activity: intermittent use, variable loading, limited hours of operation, and relatively few episodes of high stress on critical components. But not all cranes are designed with such uses or limitations in mind. Some cranes are created by their designers to be permanently or semipermanently installed in one location or to endure usage that is harsh and unrelenting. They might also be designed for both harsh usage and permanence.

For heavy service, such as duty cycle work, rugged machines are needed. Duty cycle work comprises steady repetitive work at short cycle time with fairly constant loading levels for one or more daily shifts. Ship unloading with a bucket is typical of duty cycle work and is but one of many industrial tasks requiring sturdy lifting equipment.

Severe service implies a higher proportion of lifted loads that are at full rating, more loading cycles, and perhaps greater speeds and accelerations applied to the various motions. Thus, the possibility of fatigue damage may be introduced and the statistical probability of overload is increased, as is the probability that minor defects will cause trouble. On the whole, the structure assumes a lower level of reliability when an ordinary construction crane is subjected to this heavier service; a reduction in ratings may be necessary to compensate. Another solution, more prevalent now than in the past, is to use a machine that is designed from the bottom up for severe service.

Portals, towers, and pedestals are permanent or semipermanent structural crane mountings that raise the boom or jib to clear obstacles or gain better access to the work. Almost any of the crane types previously discussed can be supported on one or more of these mountings. Lattice boom machines with ratings controlled by stability are often rugged enough to endure relatively severe service when so used; construction tower cranes may be capable of this, too, albeit at reduced load ratings.

A portal is a steel support structure on legs. Cranes are portal-mounted to allow equipment passage below the crane and perhaps to allow crane travel. The wide stance also improves stability and reduces reactions on the supporting structure. Most *portal cranes* travel on rails with electric-powered drives, but the cranes themselves may be electric-, diesel-, or diesel-electric-powered (Figure 2.45).

Figure 2.45 Traveling portal-mounted tower crane. The portal mount permits the crane to pass over materials or vehicles between the tracks. (*Manitowoc Company, Inc.*)

A pedestal is a fixed mounting to which the crane's *slewing* ring is attached. Pedestals may be made out of large steel tubes, mass concrete or a concrete tube, steel frameworks, or even a concrete-walled core section of a building. The pedestal serves as a means for raising the crane to the required working height and for anchoring it structurally by resisting all loads induced by the crane and its operation.

When still more working height is needed, a steel-framework tower is mounted on the portal frame. The "tower" in this context differs from that of a construction tower crane in that it need not be made up of slim modular sections; it is typically a stout frame with a wide stance. Capacities of portal- and tower-mounted cranes may be governed by resistance to overturning. In this case, backward stability criteria must also be satisfied. Figure 2.46 illustrates a portal-mounted tower crane set up to travel on rails.

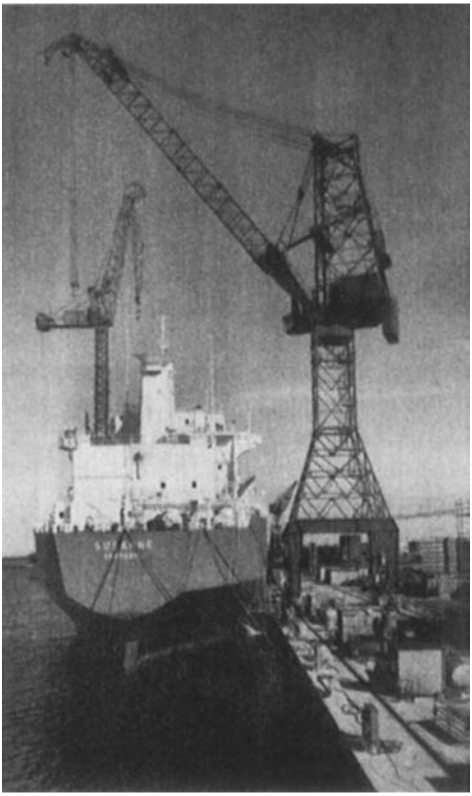


Figure 2.46 Kroll K-3000L portal-mounted crane. This crane with an unusual configuration has a capacity of 138 tons (125 t) at 82 ft (25 m) and 38.6 tons (35 t) at a reach of 197 ft (60 m). The height to its *slewing* circle is 121 ft (37 m). (*F. B. Kroll A/S.*)

Revolver cranes are a series of large latticed boom cranes that resemble mobile-crane superstructures, available in capacities of 4200 tons (3800 t) or more. They are often bargemounted or shipmounted, but when portal, pedestal, or tower mounted, they are used at such places as dam construction sites and shipyards. Many can be seen in fabricating yards preparing deep-sea drilling platforms and other facilities for oil exploration and extraction.

2.7 Overhead and Gantry Cranes

An overhead crane is a traveling machine that rides on a runway structure or pair of tracks above the work floor. The crane includes a bridge that spans between the tracks and a fixed or trolley-mounted hoisting system. Overhead cranes, also called *bridge cranes,* can be manually operated but, when powered, are almost invariably electric. They are sometimes formally referred to as *overhead electric traveling (OET) cranes*. Cranes may be top running on rails or underslung, in which case they are suspended from, and ride along, the bottom flanges of the runway beams. Bridge girders may be either single or double. Cranes of 250-ton (225-t) capacity with 100-ft (30-m) spans are not unusual.

OET cranes are mainly used in industrial service but are also found at stone or concrete precast yards, steel-fabricating shops, and storage facilities.

A *gantry crane* is a crane, usually an OET, mounted on legs that ride on rails. The bridge, which can be a trussed or plate structure, may be cantilevered on either or both ends. When one end of the bridge is on legs and the other on a runway beam, the machine is called a *semigantry crane*.

In the 1970s, the original author of this book was astonished when he first saw the colossal gantry crane in use at the Kockums AB shipyard at Malmo, Sweden.[6] Spanning some 574 feet (175 m) and nearly as tall as it was wide, it had a capacity of 1650 tons (1500 t). By the standards of any epoch of human history, this machine is a world wonder.

2.8 Cableways

Similar in form to aerial tramways that carry skiers and tourists, a *cableway* is a means for transporting material to remote locations. Once an oft-used tool for dam and bridge construction, they are now a rarity. A cableway was used to build the New River Gorge Bridge in West Virginia during the 1970s, and a multitude were deployed for constructing the Hoover Dam in the Black Canyon of the Colorado

[6]In 2002, having been purchased by Hyundai Heavy Industries, the crane was dismantled and later reinstalled in Ulsan, South Korea.

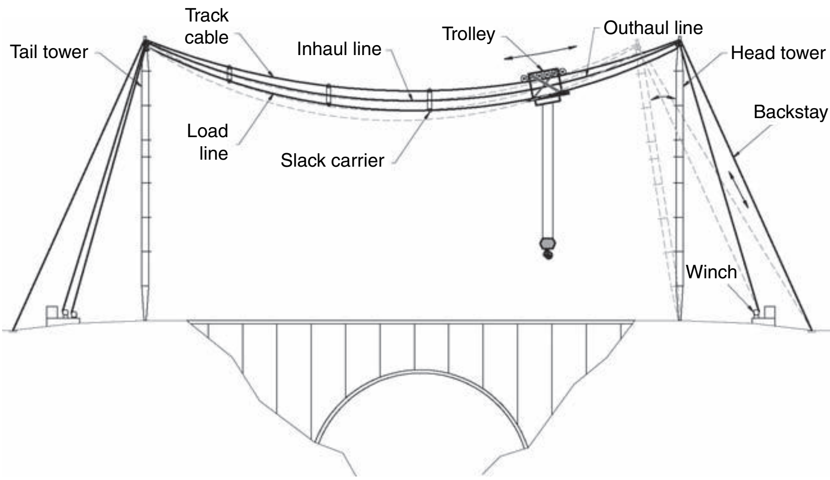


FIGURE 2.47 Basic arrangement of a cableway system; the trolley runs between two masts. Lifting capacity is raised by leaning one or both masts to increase the catenary sag of the *track cable.*

River during the Great Depression. A span of a half-mile and 50-ton lift capacity is attainable.

A basic cableway (Figure 2.47) is composed of a load-carrying trolley riding on a *track cable* suspended between two masts. Numerous other cables, static and moving, are needed to make the cableway a practical lifting apparatus. Each mast is stabilized by a forestay, a backstay, and two opposing side guys. The side guys can be live, permitting the masts to luff sideways so that the load on the trolley can be maneuvered laterally. The backstay is usually live on just one mast, called the *head tower*, allowing the slack in the cable to be adjusted; the opposing *tail tower* would be stationary. Increasing the catenary sag in the track cable by tilting the head tower forward causes an increase in the load carrying capacity.

The trolley is pulled in either direction by *inhaul-outhaul lines.* A load hoist cable raises and lowers the trolley. *Slack carriers* are suspended from the track cable to take out the sag from the haul ropes and the load line. A *messenger line* suspended from the masts carries electrical wires and the like.

Just as with some derrick types, installation and operation of a cableway require specialized skills that are no longer readily found in the labor pool. The cost and time for installing and operating are relatively great. As a result of these drawbacks and the availability of alternatives, cableway applications are limited to a very few circumstances typically involving long-term lift work over a chasm or to a remote location. A helicopter, tower crane, or other work method is likely to prove more practical in nearly all instances.

2.9 Unconventional Lifting Devices

Lifting of heavy objects must often take place in situations where a crane or derrick is impractical. The space may be confined or the overhead room restricted, or perhaps the weight or movement path cannot be handled by conventional equipment. Not uncommonly, these problems have been solved by multi-stage operations, or by onerous, time-consuming "brute force" rigging schemes. Classical rigging entails moving heavy objects by means of jacks, cribbing, rollers, or other such basic devices which provide mechanical advantage or temporary means of support, but at the expense of much time and labor.

Alternate equipment has been developed over the years to provide more efficient solutions to unconventional lifting problems. Some of these have developed niche markets in places like oil refineries and electric power plants.

Jacking Towers

Jacking towers are used primarily for vertical movement of large loads in tight quarters or out-of-the-way locations where a large crane may not be suitable. There are two well-established types: those which push the loads up can be called push-jack systems, and others which pull from above are *strand-jack* systems.

A typical jacking tower is a frame supported on a foundation and perhaps guyed for stability. Towers may be triangular or rectangular in cross section. A tandem pair is usual, but four towers can be utilized if needed. The towers can be braced or guyed to one another.

The push-jack system uses a crossbeam from which the load is suspended; hence towers must always be taller than the load. The crossbeam in turn is suspended from yokes which climb within guide slots built into the near legs of each tower; a pair of hydraulic jacks at each end pushes the beam up until a locking plate, or dog, can be inserted and support the load. With the crossbeam safely seated, the jacks can be retracted in preparation for the next increment of climbing. The jacks are short and remain in place; climbing is accomplished by inserting solid-steel lifting pistons each time the jacks are retracted. Between strokes, previously placed lifting pistons rest on the dog. There are two jacks located at the base of each tower; the lifting pistons are inserted, and the lock plates are located at the base of the towers. Thus, all climbing work is accomplished from the ground.

Advantages of push-jack systems are that they are rudimentary and thus easily understood and repaired, provide positive load guidance, and can be worked from the ground. Disadvantages are that they are slow, lack means for lateral adjustment, and the guys must be assiduously watched and maintained.

A strand-jack system requires suspending loads from framing at the tower tops; the framing can be concentrically supported by the towers or may be supported on the inboard tower legs. Slings or lifting bars are attached to the load and run up to a lift beam. The lift beam is fitted to receive the lifting tendons which are typically a bundle of 18-mm-diameter high-strength wire-rope strands selected to satisfy load weight.

The lifting tendons go up to strand jacks mounted on the overhead cross beam (really a pair of beams or trusses). The center-hole strand jacks pull up the tendons stroke by stroke, gripping them for jack retraction and repeating the cycle as needed. At the start, as the jacks begin to extend, the upper grippers grab the tendons by means of friction and wedging forces. At the end of the stroke, the jacks are reversed and the lower grippers are similarly engaged; after this, the upper grippers are released. After full retraction, the cycle is repeated. Withdrawn tendon material is spooled. The power pack powering the jacks is normally located at top as well.

A pair of towers can support one or two point picks. Four towers can be used for either two or four point lifts. In the four-tower arrangement, a crossbeam is employed to support each pair of load-supporting jacks, and header beams support the crossbeam (or beams). As heavy loads are often supported near the center of the pair of headers, these are commonly robust members. A four-tower setup can be self-bracing; this eliminates the need for external guys. The savings in guys and *deadmen*, however, would probably be exceeded by the costs of shipping and erecting two extra towers, header beams, and rigid bracing members. Therefore, a four-tower arrangement would only be used when dictated by advantage or necessity such as when the load must be shifted laterally or the lift is from four points.

When two towers are used, they may have to be anchored to a substantial moment-resisting footing or would need guys to restrain wind and side loading during the operation as shown in Figure 2.48. If the towers are guyed, the guys should be preloaded because of the base fixity. Preloaded guys imply larger deadmen for anchorages than passive guys.

Strand jacking offers precise control over the load. A system can also be configured to provide for transverse and longitudinal load movement, even rotation, for precise placement, but these features come at added cost. For loads of common magnitude, chain falls can be used to pull the load into final alignment. If sufficient drift height has been provided, pulling forces are not very large and can be easily accommodated, but the designer must allowed for this in advance.

Although jacking equipment is at the top of the towers, controls and instrumentation can be on the ground. Pressure and stroke data for each jack (including guy preload jacks) can be continuously metered and can show up in readouts at an instrumentation station. One control lever operates the jacks and a second lever operates the gripper hydraulics. Jacks can be operated singly or simultaneously;

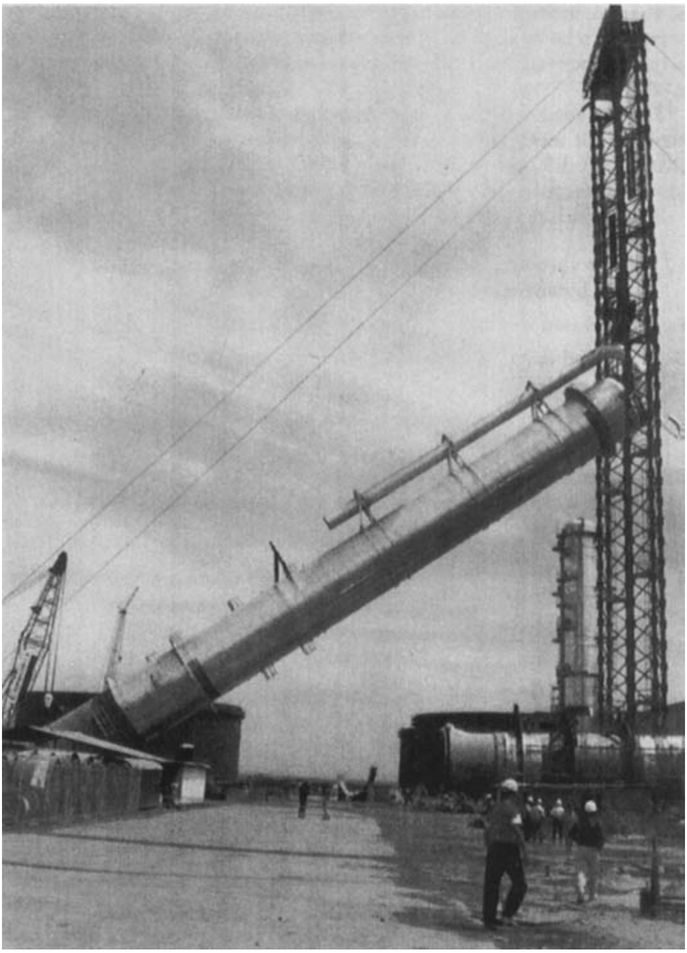


Figure 2.48 A strand-jack system utilizing two 300-ton (272-t) jacks to lift the 450 ton (408 t) vessel at this Venezuelan facility. The 270-ft (83-m) towers are braced to one another and guyed for stability. (*PSC Heavy Life, Inc.*)

speed can be varied as well. However, with the mechanical action taking place at the top, including spooling of excess strand, it would be prudent for someone to be at the top to verify that equipment is functioning properly. TV cameras can also be considered for that purpose (Figure 2.49).

Though good synchronization requires reliable instrumentation, the strand-jack method has a track record of accomplishing many difficult heavy lifts. During the lift, the load is always held in the grippers when static, and on the center-hole jacks when in motion.

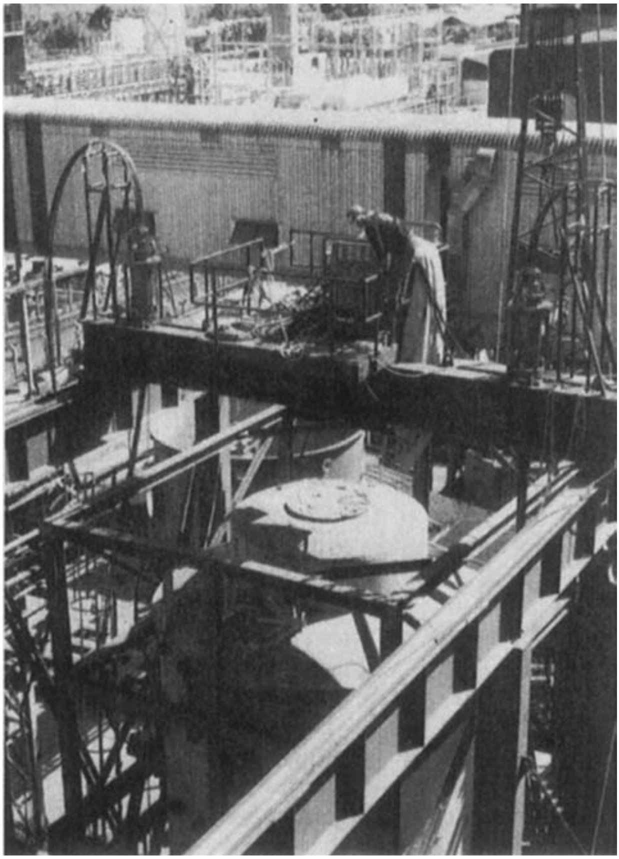


FIGURE 2.49 160-ton (145-t) chemical tanks being lifted with strand jacks. As the load rises, the strands rise out of the top of the jacks and must be spooled or otherwise controlled to avoid damage. Jacking permits precise control of movements for any weight up to jack capacity.

Strand-jack system advantages are

1. Precise vertical load control and the ability to correct for lateral out of plumbness by operating one jack
2. Control and monitoring from the ground
3. High load capacity utilizing simple towers that can break down into small sections
4. Moderately fast operation, with typical loads being able to be lifted within a normal work day
5. Adjustable guys that can readily correct alignment and out of plumbness to within tolerance even as the system changes from the unloaded to loaded condition

Strand-jack system disadvantages are

1. Site-specific tower-bracing design
2. Critical necessity to maintain alignment and plumbness
3. Erection of the towers and guys being a major construction operation (To raise the towers and set the heavy tower top equipment, a large mobile crane is needed.)
4. Dual tower setups that require guys for stability (Proper design and preloading of the guys is critical as is consideration of the additional axial loads in the towers and their foundations induced by the guys.)

Hydraulic Telescoping Gantries

Telescoping gantries are typically used for heavy lifting where mobile cranes or derricks are not feasible, where classic "jack and slide" procedures are awkward, or where other conditions make use of the gantry the most economical means to perform the work required. These systems first came into regular use for rigging during the early 1980s.

Characteristics of hydraulic gantries are different from those of other rigging equipment. Cranes will fail without warning when structurally overloaded to the limit. Gantries, on the other hand, can be made to refuse excessive loads by control of hydraulic pressure. But gantries can be sensitive to inadequacies in the strength, levelness, stiffness, or arrangement of their support structures and require much more deliberate attention to support than cranes do. Close engineering scrutiny of ground support is seldom needed for mobile cranes; for gantries it is the general rule.

There are two basic types of hydraulic telescoping gantries. Both types work in groups of two or four jacks. Each jacking unit is made up of a wheeled structural box on which one or two hydraulic cylinders are mounted. A steel header beam is fitted across each pair of jacking units. In a typical operation, loads to be lifted are suspended from these beams by slings attached to lifting link yokes over the header (Figure 2.50). The assembly can be used in place for lifting, or can be made to travel along steel runway beams or on steel plates over concrete.

The original and most common type of telescopic gantry utilizes large diameter multistage *lift cylinders* with as many as five stages and fully extended height of as much as 45 ft (13.7 m). The capacity for a pair of units can be 1000 tons (900 t) or more, but the capacity reduces with cylinder extension. The stages sequentially extend with the smallest diameter, and hence the weakest stage at the top. The lifting ability, structural strength, and structural stability of gantry legs depend entirely on the cylinders. Because lateral strength is

Figure 2.50 A simple telescopic gantry with two stocky lift cylinders, a header and a pair of lifting yokes, picking a piece of machinery by slinging around a pair of trunnions. (*Budco Rigging Co.*)

limited by these cylinders behaving as a cantilevered column, the system is sensitive to side loading. Load pendulation, out of levelness, and side-pull from rigging operations can cause side loading.

The second type of hydraulic telescoping gantry was developed to reduce sensitivity to side loading. These units have a single cylinder at each jacking unit, but the cylinder is enclosed in and braced by a telescoping steel box structure called a *lift boom* (Figure 2.51).The stout boom all but eliminates the threat of structural failure from side loading, but it does not make the gantry immune to overturning. Exposure to the overturning hazard can occur from side pull while tripping a load, particularly when there is not much vertical load acting to stabilize a unit. Ordinarily two pairs of jacks are used for tripping; the operation must be carefully controlled to mitigate side loading. When a single pair is used, holdback lines are needed to restrain against overturning.

Gantry legs interact with one another. In the direction perpendicular to travel, the header beams dictate that lateral movement at the top of one leg must be exactly matched by movement in its partner. Similarly, the weight and rigidity of the load can cause the tops of the legs in the second pair of jacking units to move in response to movement in the first pair.

Once a load is lifted, a gantry system can be used to travel laterally with the load before lowering. Travel along the track is referred

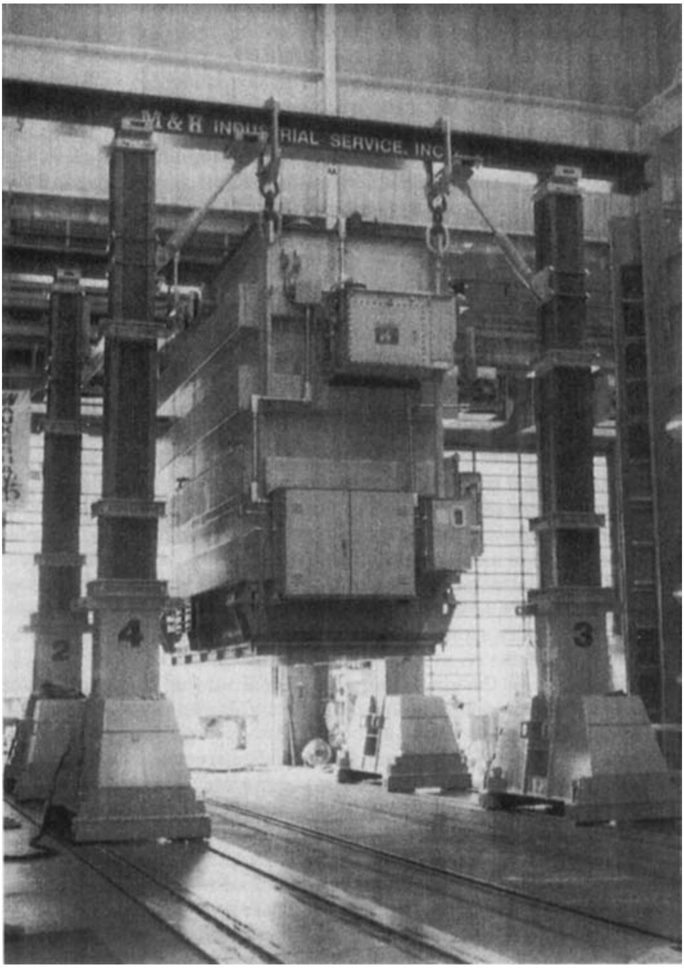


FIGURE 2.51 The lift cylinders are enclosed in telescoping steel boxes that add structural rigidity in this four-leg hydraulic gantry system. Note how the load is suspended from yokes that can be positioned anywhere along the headers. (*J & R Engineering Co., Inc.*)

to as a propel movement. Lateral shifting can also be accomplished by rolling or sliding the lift links along the header beam. When pulling is done from outside, there is side loading that must be counteracted using opposing guys or other means.

Two tandem pairs of telescopic gantries are sometimes used to carry a load placed atop the header beams. The higher the load center of gravity is above its base in comparison to base width, the more care is needed to keep the load level to avoid tipping or uneven distribution of loading among the lift cylinders.

For boom-type gantries, clearance and wear at slider pads is inelegantly referred to as slop and the lateral displacement it causes at the top of the vertical boom is called *drift*. Aside from slop, there are several possible causes for lateral displacement at the tops of gantry jacking units, but displacement from out-of-level supports can be the largest. The entire gantry structure is affected by the levelness at each of the jacking units. Firm and level track support is crucial, and the tracks themselves should be designed to deflect very little under load.

When the top of a jacking unit moves with respect to its dead plumb position, the wheels supporting that jacking unit are no longer uniformly loaded; there are reactions at the wheels, which change to counterbalance the shift at the top. Movement at the top indicates that the load has also moved, and that each unit will no longer be subjected to its original share of load weight. Track beams must be capable of accommodating this reallocation of forces. The gantry structure relies on the track beams for support of its individual wheels and to maintain the entire structure plumb and steady as the gantry travels.

The base of each jacking unit is typically furnished with eight steel wheels, two pair on each side, and two track beams are typically used on each side of a gantry set up when travel is contemplated. Pairs of wheels are centered on the track beams; a steel bar is welded to the beams in the space between paired wheels to keep the wheels centered.

Some support conditions can be referred to as "hard points." A hard point is a support point, that is, for practical purposes, nonyielding, or incapable of meaningful displacement. Examples of hard points would be solid rock, a substantial concrete mass, or steel bolsters which are in turn supported on a hard point. Any other support material will displace either elastically or permanently, the amount of displacement varying with the material. Natural wood or plywood displaces elastically at low loading levels and permanently at high levels. Soil displacement has both elastic and permanent components, and those components can vary with the nature of the soil, degree of compaction, and/or the moisture content of the soil. Soil displacement, or settlement, is time dependent; part of the settlement takes place immediately and the remainder as time progresses.

Displacement at any point along a track beam is the algebraic sum of the support medium displacement, track global displacement, and local wheel displacement. The track beam design process must substantiate that the differential displacements that will inevitably take place along the track beam or pair of beams are within tolerances that will not be detrimental to gantry operation and safety.

At any one point, a track beam which has initially been set level can go out of level when loaded if the shims at one or more support points are looser, thicker, or of different material than the adjacent shims, or if the supporting material is of a yielding nature with respect to the support of the adjacent track beam.

The centering of the load, with respect to the jacking units, affects the distribution of load among the units and thus can affect lean. Likewise, if one header beam is permitted to go out of level, the "table leg effect" would cause a redistribution of loading. This is similar to a four-legged table on a hard floor. The table will rock and will be, in essence, supported on two diagonally opposite legs. In like manner, when one jacking unit is extended more than its partner, it and the diagonally opposite unit would try to carry the entire load—the equivalent to a rocking table. In actuality, those jacking units will tend to carry less than the full load because of the limited hydraulic pressure available and elastic effects, including the stretching of wire-rope slings.

There is no trade association, nor is there a formal industry standard covering, for hydraulic gantry design, manufacture, testing, or use. Therefore, all aspects of these machines are subject to the judgment of manufacturers and riggers.[7]

---

[7]There is, however, a guidebook: *"Recommended Practices for Telescopic Hydraulic Gantry Systems,"* Specialized Carriers & Rigging Association, Fairfax, Va., 2005.

CHAPTER 3

Loads and Forces

On a steamy summer workday morning some years ago, a potent thunderstorm spilled out of Arkansas onto a city along the Mississippi River. Three tower cranes—all clustered over a stadium project—were mauled by the wind with leg connections failing at the bottoms of the masts. Miraculously, all three became propped on the stadium structure and none toppled; the operators got out safely.

One of the authors was called by the general contractor to help with stabilization and removal of the cranes. On arrival at the site, he was stunned to see the devastation wrought by the brief storm: a public building that looked as if a truck had slammed into the front and continued out the rear, trees and telephone poles sheared, and those three leaning tower cranes. The storm brought havoc to the whole city but it was most wrathful along a narrow corridor that ran from the river through the construction site.

Bearing with a force that would have been a designer's nightmare, the storm had struck without warning while the cranes were operating. The wind, in other words, did not behave in accordance with the neat dichotomy that crane engineers have created to distinguish *in-service* working loads from *out-of-service wind* loads. This storm pattern, a squall front known as a bow echo, is of a type that can produce straight-line winds of up to 100 mph (160 km/h) and strong vertical wind shear as well (Figure 3.1).

The lesson taken by the author was not that wind calculations for these tower cranes were deficient. On the contrary, this event, unusual but not unique, was a stark reminder that the engineer's view of the world has limitations. The calculations followed a mathematical model that was formulated to mimic reality, but does not always do so successfully. Atmospheric physics underlying the model is not necessarily a good fit for all storms. But even if the modeling was correct in this case it is built on the assumption of a statistically based storm intensity that can be exceeded. Quite possibly, this storm was one of the rare intensities, outside of statistical norms.

All structural design comes up against issues such as these: limitations of the analytical model and the possibility that loading norms will be exceeded.

Figure 3.1 Havoc wreaked on tower cranes by a squall known as a bow echo front. The cranes were at work when the storm struck without warning. (*Lawrence K. Shapiro.*)

The word *load* means something different to the engineer who works with cranes than to most people who operate or use them. Users and operators usually take this word to mean the weight suspended from the hook. The engineer, on the other hand, considers the suspended object as but one of several loads acting on the crane. Additional loads might include effects imposed by wind and acceleration of the crane motions; they might also include effects of jarring movements, side pulls, out-of-level setup, or even earthquakes.

Engineers quantify loads from these various sources to gage their effects on the crane and to remedy problems they might otherwise cause. Analytical methods of the engineer, such as those furnished in this chapter, can be forbidding to most people. Nonetheless, these other folks often gain a practical understanding of loads. In particular, crane operators and riggers who have experienced load effects in the field may be especially capable of connecting the engineering concepts described in this chapter with their personal observations.

Comprehension of how loads are created and how they affect the equipment is a key to the mastery of crane engineering, crane operation, and the practice of rigging. The mathematics and physics presented in this chapter is for all readers who may appreciate it, but it is primarily intended for engineers. The descriptive text, however, should help all readers relate the concepts described to their actual experiences on the job, and help them foresee, understand, and allow for what they may encounter in the future.

3.1 Introduction

The text of Chap. 1 describes how to distribute the effect of boom dead weight and the hook load to the members of an elemental derrick, and how to calculate the support reactions induced by those loads. Chapter 2 introduced more complicated lifting machines with variations in the arrangement of components resulting in entirely different concepts and capabilities in load handling. These more elaborate contraptions may frighten us into thinking that they are too difficult to understand how they function. But such fear is unfounded. All things in nature must obey the same set of physical laws, laws that hold true whether applied to cranes or billiard balls. When complex machines are broken down into basic components and evaluated step by step, even the most intricate of them can be understood. How they work is nothing more than the application of familiar physical processes.

At this point it is useful to distinguish between loads and the more general category "forces." In the jargon of engineering, loads are distinguished from other forces because they induce reactions at external points of support. As installation designers and field staff are largely concerned with support of the equipment, loads are for them the important consideration. Other forces, those that act on or within crane components rather than at the supports, are mainly the province of crane designers. The discussion that follows is about forces in general. In keeping with the target audience of this book, however, the thrust of the chapter is mainly about loads.

While cranes and derricks operate and go through their motions, they simultaneously generate forces and react to them. Some of these forces and their reactions are steady-state and others are time-varying. Forces are introduced by the operating environment, by the work being done, and by the inertia of the masses that have been put into motion, as well as by the weights of the crane components and the suspended load.

The operating environment imposes loads associated with wind, snow, ice, cold, and heat, and on occasion, earthquakes. But in addition, the working environment includes more-subtle forces induced by imperfections in the machinery and its setup. Out-of-level and misaligned mounting are among the sources of such forces. So is friction when dust, mud, sand or stones interfere with the smooth working of moving parts.

The work being done by crane movements generates forces, even when imperfections and environmental influences are absent. The discrete action of hoisting, swinging, trolley motion, luffing, and travel each create a system of forces within the structure and the machinery works. No motion occurs without friction, and thus friction is an element of each moving system.

There are also those forces that result from acceleration of the suspended load and the moving crane components during luffing, trolley, travel, slew, and hoist motions. Several motions also introduce pendulum or centrifugal action in the lifted load, which is accompanied by radius change and lateral force.

For any defined set of conditions it is possible to calculate with reasonable accuracy the forces developed and their effect on each machine component. But in an abstract sense—which is to say in the world of the crane designer—it is seldom possible to determine aggregate, real-world operating conditions accurately. Many of the forces described are time-variable in magnitude and in direction, as are the agencies introducing those forces. The lifted load itself is a fuzzy variable.

How then do the machine designer and the installation designer deliver adequate reliability while maintaining reasonable cost? What properly defines the force systems for a particular lifting device?

Design Loading Concepts

Two general methods are now in common use for applying design loadings for cranes and for structures in general. One is deterministic, meaning that loads and strength of material values are treated as if they are predictable and invariant; the other is probabilistic, recognizing that there is a range of variation among loads and material properties reflecting a measure of unpredictability. The older deterministic approach, allowable stress design (ASD) is built on generations of ad hoc reasoning and experience. The newer method, limit state design (LSD)[1] is based on statistical modeling augmented by a more systematic reasoning than the older method. For crane engineering, ASD is still prevalent in the U.S. while LSD has taken hold in Europe and to some extent, elsewhere in the world. Though much of the manufacturing of cranes has shifted to Asia, design methodology still flows principally from one or the other of the continents where the crane industry first took shape.

There is nothing inherently incorrect about either method; many pints of beer could be downed as German and American engineers argue through an evening over their relative merits, with no resolution or changes of heart likely. ASD is intuitive and has a reliable history, and LSD has the power of formal reasoning. Underlying both methods is the notion that there is a sound balance to be found between the safety and the economy of an engineering design. But neither method is pure:

---

[1]In the United States, this method is often called Load and Resistance Factor Design, abbreviated as LRFD. However, the limit state concept is broader than implied by this description, as it also includes serviceability limits such as deflection.

shortcomings of ASD have been overcome by grafting onto it methodology that mimics LSD, and when lacking a statistical basis, developers of LSD may simply use intuition or borrow factors from ASD.

Both methods require loads to be applied in sensible combinations. Some combinations are inconsistent and can be ruled out; a load lifted in the midst of a gale force wind is one example. Other loads are so unlikely to act in concert that their combination should not be considered except occasionally in the most risk-sensitive applications. Dynamic forces, particularly, are typically of very short duration and are unlikely to combine with other short-lived events. Simultaneous occurrence of peak values of lifted load, wind, and dynamic force would usually be considered highly improbable, for instance.

Design codes and standards written specifically for cranes endeavor to provide loads and load combinations that are rational for cranes. Without some modifications, loads and load combinations in building codes for permanent building structures are not usually well suited to crane applications.[2] Cranes may be structures like buildings, but they are also machines, obviously unlike buildings in the ways they are designed, constructed, used, and maintained. Fabrication practices are much tighter for cranes than for buildings, control of loads and environmental exposures are different, and duration of use is typically much shorter. The list of differences is extensive and could be differentiated by crane type. The point of raising the issue is that there is a trend, particularly in the United States, to apply building codes to cranes without recognition of these distinctions.

ASD places upper bounds on stresses and deflections using loads that are nominally based on working conditions. When true loadings are unpredictable and difficult to control, such as side loading and impact on cranes, ASD may be empirical in the choice of the applied loads. Though in practice the two methods should generally produce similar results, there are several conditions where LSD can be more powerful than ASD with respect to cranes.

Nonlinearity. The relationship between a load and its effect is nonlinear. An example is a beam column such as a mast or boom. Another example is the bearing pressure under an eccentrically loaded footing such as a tower-crane foundation.

Balanced moments. Resultant forces must resist a finely tuned balance between an overturning moment and a resisting moment, such as counterweight acting against the moment of the boom and the hook load. Load factors should reflect the degree of manufacturing quality control, particularly the fidelity of design weights and CGs to fabrication and installation practices.

---

[2]An example of a load combination that is unsuited to cranes is found in ASCE 7-05: the ASD combination of 0.6 Dead Load + Wind Load.

Varying contribution of dead load. Variable geometry of some crane types causes the relative contribution of dead load and live load toward its stress state to vary widely over a range of motion. (This could be considered a form of nonlinearity.) A prime example is a long luffing boom. At a near-vertical angle, the deadweight of the boom has little effect, but when that boom is near-flat, the stress is nearly all from self-weight. At a long working radius, consequently, the allowable hook load can be very sensitive to dead load deviations.

These shortcomings in ASD can be mitigated by applying sliding safety factors and performing additional analytical checks.

The LSD concept relies on the notion that both loads and strength of materials are imprecise; both statistically vary in a pattern that could be characterized by bell-shaped curves. These variations, established from experimental data and subjected to statistical analysis are addressed by applying individual magnification factors to each load and a reduction factor to the strength of materials.[3] Driving these factors is an intention to assure that the probability of survival is sustained at a level that weighs the costs and consequences of failure. As a prerequisite for application of this concept, a body of statistics is needed that can completely describe each random variable characteristic of lifting-device operation as well as those environmental factors, material properties, and fabrication practices which are random variables. Evaluation and acceptance of permissible levels for probability of failure require economic analysis of the consequences of failure, including placing money values on loss of life and injury.

With an adequate statistical base, there is no question that the probabilistic approach toward design leads to a more economical and certain end product, but the cost of collecting a reliable statistical base could be prohibitive and its outcome perhaps dubious. As a general case, there is too much variety of hoisting equipment types and too great a range of applications, uniformity of practices, and quality control to make the collection of adequate statistical data for crane design entirely practical. It is not reasonable to expect that the full scope of needed data can be made available, and so compromises become necessary. If compromises are made to reduce the required statistical database, conservative deterministic parameters must be inserted to replace the missing data. Then, if too many trade-offs are made, the method can deteriorate into an illusion of probabilistic science and may indeed produce designs no safer or economical than the ASD method.

---

[3]In some design codes, the separate factor applied to each contributing load is called a *partial safety factor*.

Design Codes and Standards

Organized national efforts to codify crane design and practices have a century-long history. The American Society of Mechanical Engineers (ASME) issued the Code of Safety Standards for Cranes in 1916, and the first version of a U.S. national crane standard was published in 1943.[4] In Germany, the DIN 120 crane code was originally published in 1936.[5]

A few countries, Australia and Japan for example, have national design codes for cranes. These national codes are the law of the land. In the European Union, the European Committee for Standardization (CEN) has replaced individual national codes such as the German DIN series. The CEN codes are incorporated into law in the countries of the EU.[6]

But there are no national codes in North America. Engineers in the United States and Canada rely on *consensus standards*, industry standards, proprietary standards or foreign sources. Strictly speaking, consensus standards such as ASME B30 do not have the force of law; but they are sometimes incorporated into regulations and are often enforced by authorities such as OSHA as de facto regulations. Recognizing that they represent an industry view of good practices, the U.S. tort system gives consensus standards a high standing in civil courts. The ASME B30 consensus standards are rich with advice for operation, maintenance, and inspection of cranes but are generally lean in the area of design guidance. For mobile cranes, U. S. design practices are partially codified in consensus standards issued by the Society of Automotive Engineers (SAE).

Internationally, consensus standards for cranes are issued by the International Standards Organization (ISO). The European Federation of Material Handling (FEM)[7] is another source that is used worldwide for some crane types, but in Europe FEM is being eclipsed by the codes issued by CEN.

Classification of Loads

Loads that affect cranes can be organized into three categories[8]: regular, occasional, and exceptional. Each of the three categories includes loads that are static and others that are dynamic.

---

[4]*Safety Code for Cranes, Derricks, and Hoists*, ASA B30.2-1943

[5]*Deutsches Institut für Normung.* In English this is the *German Institute for Standardization.*

[6]The acronym is from the French *Comité Européen de Normalisation.* This organization is the standards writing body of the European Union.

[7]The acronym is from the French name *Federation Europeenne de la Manutention.*

[8]From International Standard ISO 8686/1, Cranes and Lifting Appliances—Design Principles for Loads and Load Combinations—Part 1: General, clause 7, International Organization for Standardization.

Regular loads derive from gravity effects and from acceleration induced by the crane drives and brakes. These are the loads that repeatedly occur during normal crane operation. But the peak values of these loads do not usually occur simultaneously.

The occasional load category includes in-service wind, snow, ice, and temperature effects; *skewing* is added to this group for bridge-, gantry-, and portal-mounted cranes. Wind is almost always present when a crane is working, but it infrequently reaches velocities as high as the value used for design.

Exceptional loads are infrequent, introduced by events such as extreme storm winds, erection and dismantling, testing, and earthquakes. The category of a load should not be taken as an indication of its importance. Exceptional loads often govern the design of crane supports and some crane components.

3.2 Static Loads

Static loads associated with crane and derrick operations occur during a state of rest but also underlie dynamic loadings. They derive from the lifted load, from the deadweight of the machine, and from snow or ice accumulation.

A crane or derrick going through its motions has a continuously changing distribution of loads and forces affecting the supports and various components. Inertial dynamic effects superimposed onto the static loads come into play mainly during acceleration and braking; they often will not be significant for mobile-crane installation designs but can be important for the supports of high-speed cranes in general and for some tower cranes.

Temperature changes produce effects that can be taken as static loading. Parts or components that are constrained in position by other portions of the structure built of a different material undergo loading with temperature change because of dissimilar coefficients of expansion.

For several crane types, wind loading is of crucial significance. An extensive subject in its own right, with both static and dynamic considerations, wind has been treated on its own in Sec. 3.4.

Lifted Loads

The definition of lifted load is, quite simply, that which is given by the manufacturer of the particular machine in question. The definition is not the same for all equipment. For mobile cranes and many other types of lifting devices, it includes the hook block and overhaul weights as well as the live load and all accessories used to attach and hold the load. For hammerhead tower cranes and some large-capacity lifting machines where the hoisting-system *reeving* is unchanging, the hook block and overhaul weights are taken as machine dead weight and not as part of the load. The rating chart or documentation for the device will state which items are to be included as part of the load.

Slewing engenders centrifugal force that causes the load lines to go out of plumb and the load radius to increase. Likewise, inertia will cause the load to lag behind. Both effects introduce horizontal forces that act at the sheaves of the upper load block and at the load itself (for each effect, the two loads introduced are equal but opposite in direction). The magnitude and direction of the forces are such that the moments are balanced; that is, the vertical load on the hoist system multiplied by the horizontal displacement of the load must be equal to the horizontal force multiplied by the vertical distance from the load CG to the center of the upper-block sheave shaft (Figure 3.2). Although these forces accompany motion, their magnitude remains constant under constant velocity, so that they can be applied as static. Determination of the forces will be left to the following section on dynamic loads.

Whenever a load is suspended from a single hook, its center of gravity will place it in direct vertical alignment with the hoist rope regardless of the arrangement of the slings, lifting beam, or other attachments. This is dictated by the need for the sum of all forces and moments to be zero for a body in equilibrium. Failure to understand that principle is a source of jobsite accidents where the load slips or shifts while aligning itself below the hook. Needless to say, slings or lifting attachments must be arranged to place the hook as close as possible to a vertical line through the CG of the load before the lift begins.

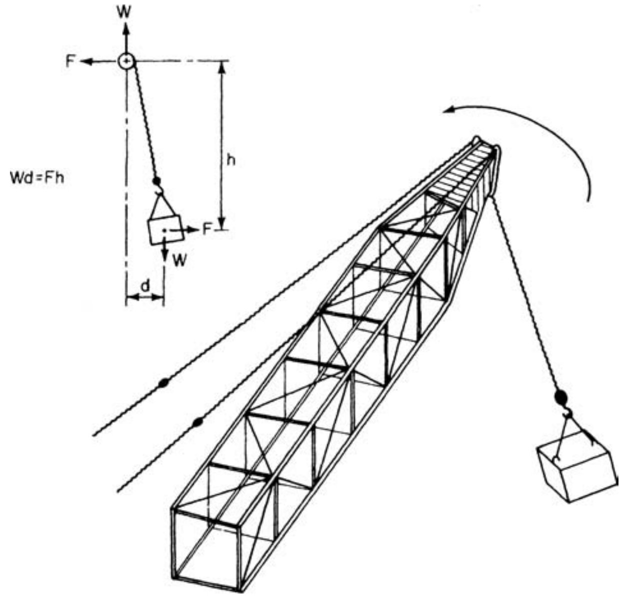


FIGURE 3.2 When a boom swings, the load lags behind and sweeps out to a greater radius as well.

Dead Loads

For critical calculations, deadweights from handbooks or from the nominal dimensions of plate stock should be adjusted to allow for normal variations. These variations in mill products can produce weight differences of a few percent. If the case in point involves a number of different section sizes distributed roughly, evenly in proportion to weight, statistically the variations should balance. But if the design includes little variation in sections, overweight or underweight can result. For custom-built equipment a coefficient of 1.07 should be considered to allow for dimensional variations, fabrication tolerances, weld deposit, and other minor factors. Where weight induces loading or stress, the appropriate coefficient should be used as a multiplier, but where weight acts to enhance stability, it becomes a divisor.

As parts move in relation to each other during machine functions, dead loads of components must be taken in their active positions. Luffing booms, live masts, and boom-hoist spreaders are common components that move with radius changes.

Snow and ice loadings need be considered only for machines destined for use in areas known for considerable accumulations. Even then, only machines that are stability-sensitive or wind-sensitive need be investigated in the out-of-service condition. Strength margins are sufficient to permit the additional initial weight, but significant incrustations need to be broken loose before operations are started.

A machine that is sensitive to wind or instability suffers a double penalty when snow or ice buildup occurs. The added weight on a boom or jib diminishes the machine's stability moment or adds to structural moment, while the enlarged members offer increased wind resistance.

Effects of Load Distribution

As a crane superstructure swings, forces acting on its supports redistribute. For a fixed-base machine, compressive support reactions may change to uplift reactions that require tie-downs and vice versa. The reactions are readily found by static equilibrium with consideration of consistent strains.

Machines supported on wheels, outrigger floats, or tracks are not anchored, and as a consequence the support points cannot resist uplift reactions. The reaction instead goes to zero, any additional uplift being distributed among the remaining supports. Procedures for determining support-point reactions for mobile cranes are fully derived in Chap. 5, but the same method is applicable to other machines that are not anchored.

Zero loading at a support point, such as an outrigger *float* on a mobile truck crane, does not necessarily signal an unsafe or critical condition. Over-the-corner lifting causes carrier-frame twisting deformation. Combined with outrigger beam deflection, that can cause an outrigger float to lift clear of the ground without ever going out of bounds of allowed load and operating conditions.[9]

When a second outrigger float shows signs of lifting, there can be no doubt that excessive load is being lifted for the operating environment and that the machine has reached the tipping point (Figure 1.22).

For wheeled machines that ride on rails, zero loading at a supporting wheel is another matter. When a wheel or bogie lifts clear of the track, there is a good chance that derailment will ensue if a perturbation causes a lateral shift. Wheel flanges should not be permitted to rise above the crown of the track under any condition of operation.

Friction

Earlier discussions have shown how friction is an important factor in computing wire-rope loads, lead-line forces, and overhauling weight requirements. Friction must, of course, be considered in the design of every mechanism. A characteristic of friction that cannot be overlooked is the decrease in frictional resistance from the static to the moving state; coefficients of friction for static parts in contact are higher than those for the same parts once motion has begun. This affects such things as the starting torque required of the drives for each functional motion and the strength of the drive anchorages.

Travel-system storm anchorages or braking systems that rely on friction must be designed with that characteristic in mind. If a storm-wind peak gust would overcome the static frictional resistance, lower-level winds or gusts could keep the machine in motion with disastrous results.

Mobile cranes, whether operating on tires, tracks, or outriggers, are not attached to the ground. Yet during operation swing acceleration and deceleration produce low-magnitude horizontal dynamic forces at the supports. These forces, however small, must be passed into the supporting surface through frictional resistance, which also prevents crane lateral movement. For that reason, some mobile crane operators like to avoid metal-to-metal bearing contact at tracks or outrigger floats. They insert wood between metal bearing surfaces to increase the coefficient of friction.

Out-of-Level Supports

All cranes and derricks must be mounted level to within close tolerances. A common specification for mobile equipment permits a maximum of 1% out of level between supports. Tall and limber machines are more sensitive; an initial deviation from a level base will be amplified by beam-column action so that even more strict tolerances are the norm (see Chap. 6). For machines with high CGs—tower cranes

[9]Excessive lift-off, however, implies the possibility of inadvertent and uncontrolled radius increase and indicates a carrier design that has insufficient torsional stiffness.

are the prime example—a relatively small difference in levelness of the supports can result in a significant horizontal displacement of the CG. This alters support reactions and component loadings and can reduce resistance to overturning (see Chap. 4).

Out of levelness may induce difficulties in swinging as the swing path develops an uphill and downhill aspect. Responding to this phenomenon is a distraction to the operator or, worse, may cause a loss of control. Out-of-level mounting also produces a side pull on the boom over some of the swing arc and a change in the true, as opposed to the apparent, operating radius. That may be detrimental to structural capacity or stability.

Misalignment and Skew

The wheels of track-mounted equipment are flanged on one or both edges or have lateral guide rollers to keep the machine on the rails. Either way, permitted rail alignment and spacing tolerance is determined by the displacement the wheels can accommodate. Excessive displacement causes side forces on the wheels and tracks. At the very least, side forces increase load on travel-train components and drive motors causing accelerated wear at the points of contact; the thermal effect can induce the motors to cut out and interrupt travel. In the extreme, track fastenings could shear or loosen with subsequent derailment and the possibility of overturning.

Similar problems to those arising from poor track alignment can occur when a rail-mounted travel base goes askew. Susceptibility is greatest on a device such as an overhead traveling crane with a short wheelbase in relation to the rail gage. On relatively lightweight cranes, the consequence of jamming due to skew is usually limited to drive-motor stall or triggering of the overload protection.

3.3 Earthquake

Studies of earthquake effects on cranes are few, and code development in this area is in its infancy. Generally, permanent installations such as bridge cranes and port cranes can be subjected to seismic analysis using the same principles as those used for other fixed structures. A decision to analyze a crane seismically should be based on the degree of risk as weighed against potential consequences of a loss. Risk may be assessed by study of earthquake maps. In areas of low or moderate earthquake risk, seismic study may be demanded only for the most-sensitive applications, such as nuclear work.

In adopting a philosophy for earthquake resistance, the crane analyst or designer might consider one or more of three risk mitigation levels, or limit states.

1. The earthquake design does not cause structural damage to the crane. All stresses remain in the elastic range. The crane should remain serviceable.

2. The design earthquake may result in some damage that could be readily repaired and the crane restored. Failure may occur in components that are not part of the main force-resisting system. Component failures cannot put workers or the public at risk, and significant collateral damage to surroundings is not permissible.
3. Controlled ductile yielding may result in the complete functional loss of the crane, which would be replaced, but the avoidance of a catastrophic failure leaves the public, workers, and surroundings protected.

A designer might choose to calibrate the design to only one of these states or, alternatively, consider associating each of them with a different magnitude earthquake event. As all three imply a high level of life protection, this decision would be based only on economic considerations.

Except for those few industrial cranes that are in near-constant use, probability favors the premise that the device will be unloaded if an earthquake should occur. Nearly all cranes used in both general industry and construction will have a substantial load on the hook only a small percentage of the calendar year. With few exceptions, then, earthquakes might reasonably be evaluated only for out-of-service consideration.

Though earthquakes are dynamic events, the simplest methods of seismic analysis make use of equivalent static loads. These methods are suited to areas of low or moderate seismicity or for structures that are relatively simple in their response to excitation. Other methods in the toolboxes of seismic engineers may be applied for more complex situations or where the level of risk warrants the investment.

A freestanding tower crane may respond well to moderate earthquakes because its long period of oscillation will not resonate with the higher-frequency ground motion. However, the crane could be at risk in a severe earthquake due to base shear or from vertical acceleration acting on the counterweights. In some soils, liquefaction could pose a risk. On a tower crane that is secured to a building, tied to the outside or mounted within, the interaction with the building can lead to higher seismic loads compared to those expected for standard freestanding erections.[10]

Generalized assessment of earthquake risks for a mobile crane can be difficult because a typical machine changes location frequently and its boom disposition changes constantly. Overall exposure should not be great. There could be vulnerability under certain conditions, however. For example, ground acceleration might induce an unloaded machine with a short boom at a high angle to tip backward.

[10]Seiji Takanashi and Yutaka Maeda, "Research of Earthquake Resistance of Tower Crane for Construction", National Institute of Occupational Safety and Health, Japan 2001(Abstract).

Perhaps the most studied seismic event with respect to cranes was the Kobe, Japan, earthquake of 1995. Documented failures included

- Total collapse due to foundation failures probably caused by liquifaction
- Overstress of towers from horizontal shear, with resulting diagonal and connection failures
- Leg tension failures due to overturning moment on pillar cranes and tower cranes
- A bridge crane girder lifting off its supports
- Overturning of a rail-mounted gantry crane

In 2002, two tower cranes toppled from the 60th floor of a steel-frame building under construction in Taipei, Taiwan, during a moderately severe earthquake. The failures were not caused directly by the earthquake, but rather by the cranes oscillating in resonance with the building.

3.4 Dynamic Loads

In addition to the centrifugal force previously mentioned, *dynamic loads* are for the most part those associated with masses undergoing changes in motion. Each crane and derrick motion—hoist, trolley, luff, slew, and travel—produces a dynamic force as the motion begins and ends. Each accelerating mass within the crane is subjected to a dynamic force following Newton's Second Law, $F = ma$, or its rotational equivalent. Acceleration of the various crane motions is largely determined by the nature of the mechanical drives. Friction cranes allow the operator considerable influence, for example, retarding downward motion only with the brake during *freefall* lowering.[11] Hydraulic and electric drives modulate acceleration by design, though certain abrupt or inappropriate actions of the operator can defeat the designer's intentions and cause design values to be exceeded.

Another potential source of dynamic load is pendulation of the hook load. In usual crane operations the operator is able to keep the pendulum action small; but the effect could be significant for a crane lifting in rough seas, a lift crane struggling against gusty wind, or a duty cycle machine working in a rapid cycle. Finally, there are the rotating masses in the various drive systems producing local dynamic effects; if a rotating mass such as a drum is significant in relation to the crane mass, it can affect the crane during powering up or braking.

[11] At the time that earlier editions of this book were written, friction crane drives and derrick winches were much more prevalent and freefalling a more common practice. Freefalling has become associated for the most part with duty-cycle operations and old friction cranes.

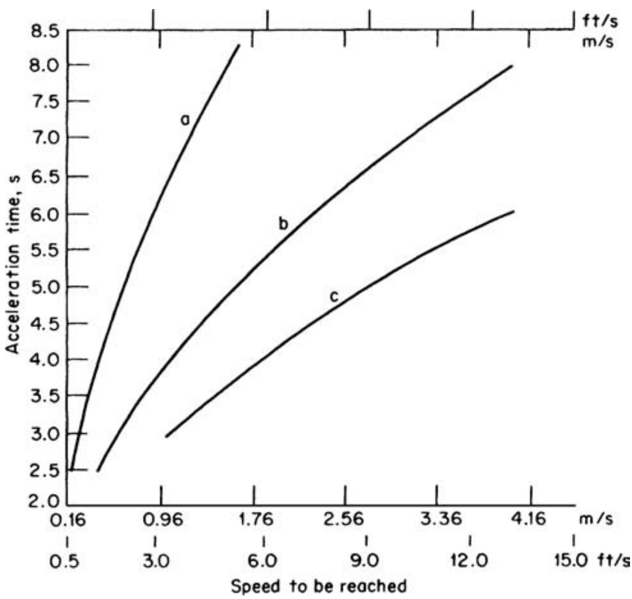


FIGURE 3.3 Nominal acceleration times suggested by the FEM for heavy lift equipment. Curve a: low and moderate speed with long travel. Curve b: moderate and high speed (normal applications). Curve c: high speed with high accelerations.

Linear Motion

The force required to accelerate or decelerate a mass in linear motion can be found simply by applying Newton's law if the acceleration rate is known. For friction cranes acceleration and deceleration is not linear, as drive and braking systems are generally capable of far greater force than needed and are modulated by the operator. The appropriate acceleration rate to be used in design is then a matter for judgment, but the FEM offers some guidance.[12] Curves representing acceleration times suggested by the FEM for heavy-lifting equipment are shown in Figure 3.3. With acceleration time and final (or initial) velocity given, average acceleration is simply the velocity divided by the time. Many machines have drive systems that do not provide linear acceleration, but for most work a linear assumption is satisfactory since system inertia tends to smooth out the acceleration.

For friction machines with brakes that engage automatically, the brake will be rated either for braking force or for torque. When these values are mathematically referred to the point of contact of a wheel

[12]Fédération Européene de la Manutention, "Rules for the Design of Hoisting Appliances," 2d ed., Paris, 1970, sec. I, Heavy Lifting Equipment. In the 3rd Edition (1998) the graph is replaced by Table T.2.2.3.1.1.

with a rail or of a rope to its winch drum, the force inducing the acceleration becomes known and acceleration rates and times can then be calculated.

Three conditions can create large dynamic loads in a hoisting system: picking a load abruptly from a condition of rest, a sudden stop during lowering, or a sudden release, emptying a clamshell bucket or dropping a magnet load of scrap, for instance.

When a load is dropped in free fall, the acceleration is retarded by friction at the sheaves and the inertia of the winch drum. Considering only gravity and friction, resulting acceleration is given by

$$a = g\frac{M_a}{r} \tag{3.1}$$

where g = acceleration due to gravity
M_a = nominal mechanical advantage of system
r = lowering factor given in Eq. (1.1)

The final velocity is found from

$$v = (2ah)^{1/2} \tag{3.2}$$

with h as the height of fall. If the brakes fully and suddenly engage so that deceleration is virtually instantaneous; the load will not stop immediately. The hoist ropes will stretch springlike and provide some additional movement. Taking the kinetic energy at the time of application of the brakes and adding the potential energy of the load W and dynamic rope stretch Δ, we can equate this with the potential energy in the ropes after stretching is complete. If F is the final total force in the ropes,

$$\frac{Wv^2}{2g} + W\Delta = \frac{F\Delta}{2}$$

but the spring rate of the ropes is $k = F/\Delta$. Substituting this and rearranging, we get

$$F^2 - 2WF - \frac{Wkv^2}{g} = 0$$

$$F = W + \left(W^2 + \frac{Wkv^2}{g}\right)^{1/2} \tag{3.3}$$

$$\frac{F}{W} = 1 + \left(1 + \frac{kv^2}{Wg}\right)^{1/2}$$

Equation (3.3) shows that if the initial velocity is zero (that is, if the load is not dropped but is suddenly applied to a slack rope system), the effect on the ropes is to double the load. Theoretically this is the lower bound of dynamic amplification because any initial velocity will result in an effect more than twice the load. But the model used in the derivation of Eq. (3.3) is oversimplified. Winch-drum and sheave inertia will retard acceleration more than indicated by Eq. (3.1), and, of course, neither automatic nor operator-controlled brakes will cause an instantaneous stop.

In actuality, Eq. (3.3) overstates dynamic amplification by a factor of 2 or more depending on the rate of application of the brakes and the elastic properties of the system. In general, the amplification effect will be between 1.0 and 2.0.[13]

Although the term *impact* is generally, and more correctly, applied to the striking of a body by another body, it is also used to describe the increase in load effect due to dynamic causes. The impact force F_i is then given by $F - W$, and impact can be expressed as

$$i = \frac{F - W}{W} = \frac{F_i}{W} \tag{3.4}$$

Operators do not often allow other than minor loads or empty hooks to free-fall. Substitution of even small fall heights into Eq. (3.2) followed by solution of Eq. (3.3) will show that impact quickly rises to high multiples of the load. On machines that permit free fall, it is usual for an operator to ride the brake in order to modulate velocity and be in a position to maintain acceptable limits of deceleration for the final stop; this is done by judgment alone.

The FEM suggests values for impact for heavy-lift equipment that are given by

$$i = \begin{cases} \dfrac{3\xi v}{10} & \text{U.S. customary units} \\ \xi v & \text{SI units} \end{cases} \tag{3.5}$$

The velocity v is in feet per second (meters per second) and

$$\xi = \begin{cases} 0.6 & \text{for overhead traveling and bridge cranes} \\ 0.3 & \text{for boom or jib cranes} \end{cases}$$

When Eq. (3.5) is used, a minimum value for i of 0.15 is contemplated and i is to remain constant for velocities exceeding 3⅓ ft/s (1 m/s).

[13]Ibid, clause 7.1.4; as pointed out by Prof. Dr.-Ing. Rudolf Neugebauer, Technische Hochschule Darmstadt, Germany, in private correspondence. The final result depends directly on the brake force actually applied, which is a difficult parameter to determine.

Some decades ago, a task force of the American Institute of Steel Construction (AISC) conducted tests to determine how best to account for impact in derrick design. In their report they state that "energy applied by motion at the hook is absorbed immediately and simultaneously by all elements of the derrick setup and cannot be separately identified in the ensuing elastic vibrations of the masses of the individual elements. . . ."[14] The task force found that simply increasing the live load by an impact factor did not yield good correlations with test results. Instead, they recommend increasing axial bending stresses, including dead load but not lateral load, by an impact factor. This procedure produced values that closely match measured stresses. For lifting full structurally based rated loads, a factor of 20% is suggested. The tests were deliberately carried out to produce extreme, or upperbound, impact compared with normal, proper production operations.

The AISC task force makes another observation that is particularly pertinent for derricks. A winch properly matched to a derrick will be incapable of stopping rated loads instantaneously. However, when an overcapacity winch is matched to a derrick, brake overcapacity will present a potential for excessive impact loading.

Example 3.1

1. An operator allows a 5000-lb (2268-kg) load to free-fall for 10 ft (3.05 m) before realizing that it is going too fast. In panic, he applies the brake fully, causing a virtually instantaneous stop. What peak force will develop in the rope if three parts of line were in use, friction loss can be taken as 2%, and the rope spring constant is 2300 lb/in per part (402.8 N/mm per part)?

Solution

$$1-\mu=1-0.02=0.98$$

From Eq. (1.1) for lowering load with three parts of line and assuming that $m=0$, we have

$$r=\frac{1-0.98^3}{0.02\times 0.98^3}=3.12$$

The nominal mechanical advantage MA = 3 for three parts of line, so from Eq. (3.1)

$$a=32.2\text{ ft/s}^2\left(\frac{3}{3.12}\right)=30.96\text{ ft/s}^2(9.44\text{ m/s}^2)$$

and from Eq. (3.2),

$$v=[2(30.96\text{ ft/s}^2)(10\text{ ft})]^{1/2}=24.88\text{ ft/s (7.58 m/s)}$$

[14]"Guide for the analysis of Guy and Stiffleg Derricks," American Institute of Steel Construction, Inc., New York, 1974, p. 6.

Equation (3.3) then gives the ratio of rope load to lifted load.

$$\frac{F}{W} = 1 + \left[1 + \frac{3(2300\ \text{lb/in})(12\ \text{in/ft})(24.88\ \text{ft/s})^2}{(5000\ \text{lb})(32.2\ \text{ft/s}^2)}\right]^{1/2}$$

$$= 18.87$$

$$F = (5000\ \text{lb})(18.87) = 94{,}350\ \text{lb}\ (419.7\ \text{kN})$$

Impact has caused a theoretical increase in load of some 1787%!

2. What maximum velocity would have to be maintained to keep impact from exceeding 30% if the stopping distance is not to exceed 5 ft (1.52 m)?

Solution

The stopping-distance requirement indicates deceleration through braking as the load comes to rest. Applying Newton's law gives

$$F_i = ma = 0.30\,W \qquad a = \frac{0.30\,W}{m}$$

But $m = W/g$; therefore

$$a = 0.30g = 9.66\ \text{ft/s}^2\ (2.94\ \text{m/s}^2)$$

Inserting values into Eq. (3.2) gives

$$v = [2(9.66\ \text{ft/s}^2)(5\ \text{ft})]^{1/2} = 9.83\ \text{ft/s}\ (3.00\ \text{m/s})$$

Had the velocity been held to 9.83 ft/s (3.00 m/s), with the stop taking place over 5 ft (1.52 m) of vertical travel, the impact would have been limited to 30%.

3. For the velocity developed in part 1, 24.88 ft/s (7.58 m/s), what stopping distance would be needed to hold impact to 30%?

Solution

Rearranging Eq. (3.2) and noting that for 30% impact we have found that deceleration will be 9.66 ft/s² (2.94 m/s²), we get

$$h = \frac{v^2}{2a} = \frac{(24.88\ \text{ft/s})^2}{2 \times 9.66\ \text{ft/s}^2}$$

$$= 32.04\ \text{ft}\ (9.77\ \text{m})$$

4. With the same initial velocity used above, what is the impact for a stopping distance of 5 ft (1.52 m)?

Solution

Rearranging Eq. (3.2) again, we have

$$a = \frac{v^2}{2h} = \frac{(24.88\ \text{ft/s})^2}{2 \times 5\ \text{ft}}$$

$$= 61.90\ \text{ft/s}^2\ (18.87\ \text{m/s}^2)$$

From Newton's law, $F = ma = Wa/g$,

$$F_i = \frac{W}{g}a = \frac{5000\ \text{lb}}{32.2\ \text{ft/s}^2}(61.90\ \text{ft/s}^2) = 9610\ \text{lb}\ (42.76\ \text{kN})$$

From Eq. (3.4),

$$i = \frac{9610 \text{ lb}}{5000 \text{ lb}} = 1.92 \text{ or } 192\% \text{ impact}$$

5. What impact values does the FEM suggest should apply for heavy-lift boom-type cranes for the velocities, 24.88 and 9.83 ft/s (7.58 and 3.00 m/s)?

Solution

The note under Eq. (3.5) indicates that there should be no increase for velocities above 3⅓ ft/s (1 m/s); therefore

$$i = \frac{3(0.3)(3\tfrac{1}{3} \text{ ft/s})}{10} = 0.30 = 30\%$$

The Equation of Motion

In applying the deterministic approach to dynamic problems, the time variation in loading must be fully known although it may be oscillatory or irregular. When the methods of structural dynamics or vibration theory are used, structural response is given in terms of displacements, and the time-varying moments and stresses can then be obtained.[15]

A static problem has a single solution, but a dynamic problem has a series of solutions represented by an equation with time as an independent variable or by values at each of the times of interest. A more fundamental difference exists between static and dynamic situations, however. Under static loading, the external loads on a structure are in equilibrium with the moments and elastic forces developed within the structure. Under dynamic loading, the time-varying external loads induce accelerations and displacements in the structure and the ensuing time-varying inertial forces also enter into the equilibrium equation.

Cranes and derricks can generally be modeled as single-degree-of-freedom (SDOF) systems; that is, a sufficiently accurate deflection time history of the structure can be formulated in terms of a single displacement or by means of an expression based on a single displacement. SDOF structures can be classified as rigid-body assemblages having springs to represent permitted elastic displacements at discrete points or continuously elastic structures deforming throughout their length. For both idealizations, SDOF behavior is forced on the system by assuming a displacement form or configuration that can be described in terms of a single displacement.

Newton's second law of motion, positing that the rate of change in momentum of a mass is equal to the force acting to produce that change,

---

[15]See R. W. Clough and J. Penzien, "Dynamics of Structures," McGraw-Hill, New York, 1975, and J. E. Shigley, "Mechanical Engineering Design," 2d ed., McGraw-Hill, New York, 1972, chap. 16.

was given earlier as $F = ma$. In differential equation form, with mass held as a constant, this can be expressed as

$$f(t) = m\ddot{z}(t)$$

where $f(t)$ = applied time-varying force vector
m = mass
$z(t)$ = position or displacement vector

The dots superscript represents differentiation with respect to time. When this is rearranged and stated as

$$f(t) - m\ddot{z}(t) = 0 \tag{3.6}$$

the second term can be called the *inertial force* resisting the acceleration; this reflects d'Alembert's principle, which states that the inertia force associated with a mass is proportional to the acceleration of the mass. This principle permits dynamic situations to be expressed in terms of equations of equilibrium.

Equation (3.6) is an *equation of motion* that describes the relationship between displacement and the initiating dynamic load. When the differential equation is solved, the resultant expression will give displacements at any point in time. Since equations of motion reflect only dynamic effects, static loading effects must be added.

But Eq. (3.6) is a simple form of the equation of motion. A fully generalized form would be

$$m^*\ddot{Z}(t) + c^*\dot{Z}(t) + k^*Z(t) = f^*(t) \tag{3.7}$$

$$m^* = \int_0^L m(x)[\psi(x)]^2\,dx + \Sigma m_i\psi_i^2 + \Sigma I_{0i}(\psi_i')^2 \tag{3.8}$$

$$c^* = \int_0^L c(x)[\psi(x)]^2\,dx + \Sigma c_i\psi_i^2 \tag{3.9}$$

$$k^* = \int_0^L k(x)[\psi(x)]^2\,dx + \int_0^L EI(x)[\psi''(x)]^2\,dx + \Sigma k_i\psi_i^2 - \int_0^L N(x)[\psi'(x)]^2\,dx \tag{3.10}$$

$$f^*(t) = \int_0^L f(x,t)\psi(x)\,dx + \Sigma f_i\psi_i \tag{3.11}$$

where the terms are generally defined in Figure 3.4.[16] In addition, the displacement at any point x along the length of the structure is given in terms of the deflection $Z(t)$ at one selected point through use of the nondimensional *shape function* $\psi(x)$, so that $z(x, t) = Z(t)\psi(x)$. For displacement that is collinear with applied load, the shape function has

[16]Clough and Penzien, op. cit., pp. 34–36.

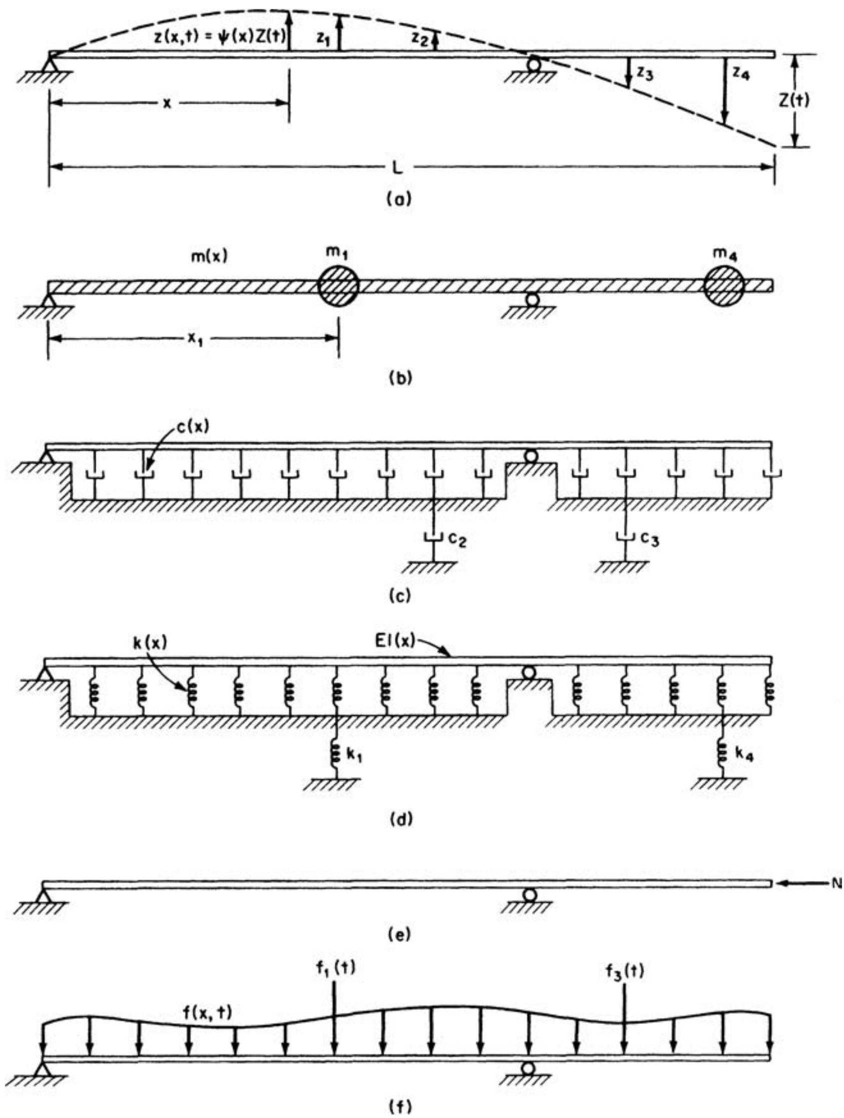


FIGURE 3.4 Properties of a generalized SDOF system. (a) Assumed shape. (b) Mass properties. (c) Damping properties. (d) Elastic properties. (e) Axial loading. (f) Applied loading. (*From R. W. Clough and J. Penzien*, Dynamics of Structures, *McGraw-Hill, New York, 1975, p. 35; used by permission.*)

the value of unity. The dot and double notations over $Z(t)$ represent the first and second derivative respectively. When the generalized properties that vary along the length, $m(x)$, $c(x)$, $k(x)$, and $EI(x)$, and the axial loading $N(x)$ are in fact uniform and continuous or constant rather than varying with x, the constant values can be inserted into

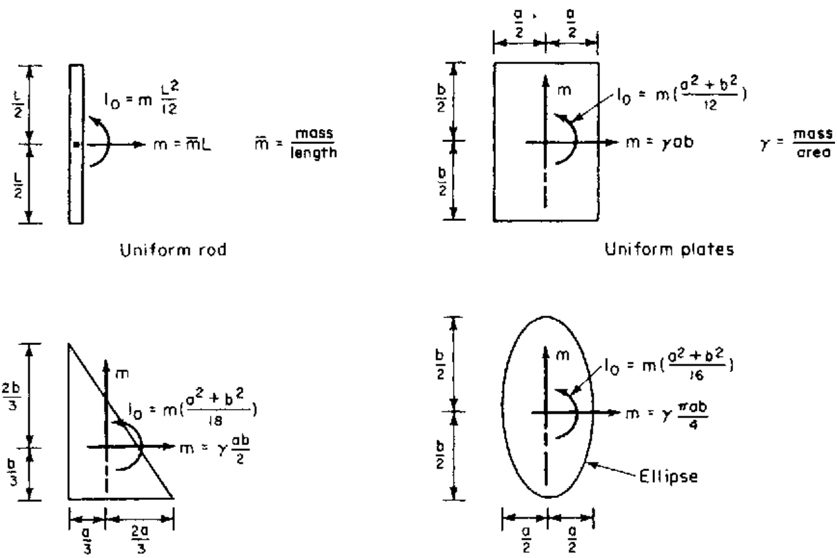


FIGURE 3.5 Rigid-body mass and mass moment of inertia. (*From R. W. Clough and J. Penzien*, Dynamics of Structures, *McGraw-Hill, New York, 1975, p. 24; used by permission.*)

the equations. E refers to the modulus of elasticity of the member material, and I represents the moment of inertia of the cross-sectional area of the member; these values are used only when bending members are present. For mass moment of inertia I_0 see Figure 3.5; note that this parameter applies only when a mass undergoes angular displacement.

The m_i are lumped masses, the c_i are discrete dampers, the k_i are discrete springs, the f_i are discrete time-varying loads, and the ψ_i are the values of the shape function at those discrete points. In the mathematical model used for analysis, *lumped masses* are aggregates of mass concentrated at the point where the inertia forces associated with those masses would act.

The shape function provides a means for describing the displacement at any point in the system in terms of the primary displacement sought; since all displacements are then in terms of one displacement, the SDOF criterion is satisfied. In Figure 3.6*a* the rigid bar AC has two lumped masses, two springs, and a hinge at A. If $Z(t)$ is the displacement at C, the displacement at B will be $\psi(x)Z(t) = (a/L)Z(t)$.

The shape function given in Figure 3.6*b* for a continuously elastic side-loaded cantilevered column is a fairly accurate representation of the deflection curve. Although the relationship is not at all direct, a small error in the deflection curve assumed will yield a smaller error in calculation results. An approximation to the true deflection curve, which is often quite complex, will therefore be satisfactory for most work.

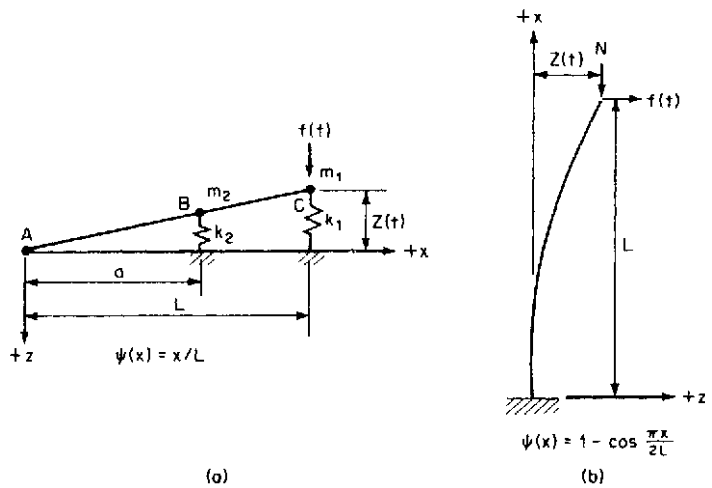


FIGURE 3.6 The two forms of SDOF structure idealization: (a) Rigid-body assemblage. (b) Continuously elastic structure. The shape function $\psi(x)$ is used to force SDOF behavior on the system by relating deflection $Z(t)$ to displacement at any point x.

Undamped Vibrations

Many practical dynamic situations involve loadings of very short duration and, in any case, peak values of dynamic response are often the only values that are pertinent. Damping tends to moderate response with time, but immediate response is virtually free of damping effects. For determination of peak values, damping can generally be ignored. With the damping term dropped from and, after dividing through by mass, Eq. (3.7) becomes

$$\ddot{Z}(t) + \omega^2 Z(t) = 0 \tag{3.12}$$

where $\omega^2 = k^*/m^*$

for undamped free response (i.e., response occurring after application of the dynamic loading has stopped). This common form of differential equation has the solution

$$Z(t) = C_1 \cos \omega t + C_2 \sin \omega t$$

The constants of integration C_1 and C_2 are evaluated by inserting the initial conditions, conditions at $t = 0$. For an initial displacement z_0, $C_1 = z_0$, and for an initial velocity $\dot{z}_0$, $C_2 = \dot{z}_0/\omega$. The equation then becomes

$$Z(t) = z_0 \cos \omega t + \frac{\dot{z}_0}{\omega} \sin \omega t \tag{3.13}$$

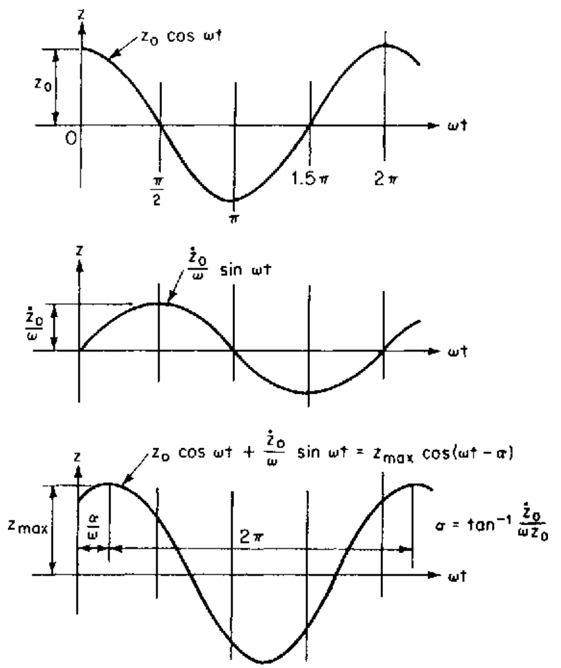


FIGURE 3.7 Dynamic displacement includes two components, one of which may be zero. An increment related to initial displacement is contributed by $z_0 \cos \omega t$, and $\dot{z}_0/\omega \sin \omega t$ is the component due to initial velocity. Adding the components yields system displacement, as shown in the third curve.

This equation clearly shows that displacement is composed of two components, one depending on initial displacement and the other on initial velocity. Figure 3.7 graphs each of the components and the full displacement plotted against time.

Further examination of Eq. (3.13) shows that the two terms are orthogonal vectors, so that the resultant, which is the maximum response amplitude, is

$$z_{\max} = \left[z_0^2 + \left(\frac{\dot{z}_0}{\omega} \right)^2 \right]^{1/2} \tag{3.14}$$

The argument ωt of the periodic cosine and sine functions causes the functions to repeat at angular intervals of 2π or at time intervals of $2\pi/\omega$. The time interval is called the *period of free vibration* and after substitution for ω becomes

$$T = 2\pi \left(\frac{m^*}{k^*} \right)^{1/2} \tag{3.15}$$

The *frequency of free vibrations*, the number of vibration cycles per unit of time, is then $\omega/2\pi$, or

$$f = \frac{1}{2\pi}\left(\frac{k^*}{m^*}\right)^{1/2} \tag{3.16}$$

The parameter ω is often called the *circular frequency* because it has units of radians per unit of time.

The response of a system during loading or under continuing loading becomes more involved and is beyond the scope of our work at this point. For further information, the reader is referred to texts on structural dynamics or vibration theory.

The material just covered can be applied to Example 3.1. In the first part, initial displacement is zero and initial velocity has been calculated as 24.88 ft/s (7.58 m/s). The applied load is not time-varying, so $f(t) = 0$; the system can be thought of as being loaded by the initial velocity. Generalized mass, from Eq. (3.8), is simply W/g because the shape function has a value of unity when the displacement is collinear with the applied load. The generalized spring rate, given in Eq. (3.10), is the summation of the spring rates in each of the three parts of line, or

$$k^* = (3 \times 2300 \text{ lb/in})(12 \text{ in/ft}) = 82{,}800 \text{ lb/ft (1208 kN/m)}$$

From Eq. (3.12),

$$\omega = \left[\frac{82{,}800 \text{ lb/ft}}{(5000 \text{ lb})/32.2}\right]^{1/2} = 23.09 \text{ rad/s}$$

Peak displacement is then given by Eq. (3.14).

$$z_{\text{max}} = \left[0^2 + \left(\frac{24.88}{23.09}\right)^2\right]^{1/2} = 1.08 \text{ ft (328 mm)}$$

This entire displacement reflects elastic rope stretch, which is proportional to the initiating force divided by the spring rate. Therefore,

$$F_i = (82{,}800 \text{ lb/ft})(1.08 \text{ ft}) = 89{,}400 \text{ lb (398 kN)}$$

but this does not include the static load of 5000 lb (2268 kg), so the total force on the ropes is 94,400 lb (420 kN), which is essentially the value previously calculated.

With zero initial displacement, only the initial-velocity term contributes to displacement, and this becomes a maximum when $\omega t = \pi/2$, as can be seen from Eq. (3.13). The total time lapse during which rope stretch takes place and impact force rises to full value is then

$t = \pi/2\omega = 0.068$ s, a very short time indeed. Eq. (3.13) gives us displacement values and therefore rope dynamic force at any time t. For intervals of 0.017 we have

	Displacement		Dynamic Rope Force	
Time, *s*	**ft**	**mm**	**lb**	**kN**
0.017	0.41	125	33,950	151
0.034	0.76	232	62,900	280
0.051	1.00	305	82,800	368
0.068	1.08	328	89,400	398
0.085	1.00	305	82,800	368
0.102	0.76	232	62,900	280
0.119	0.41	125	33,950	151
0.136	0.00	0	0	0

Thus, in about ⅛ s the rope has become free of *dynamic load* and will remain so for about another ⅛ s, at which time loading will start again. Although the magnitude of the rope load is rather high, the very short duration of impulse loadings is such that it is unlikely that the crane will overturn unless the static load constitutes a substantial portion of the tipping load. For the data given, however, the rope would have failed.

In part 2 of the problem, impact was limited to 30%; therefore, rope stretch must be limited to

$$Z_{\text{max}} = \frac{0.30 \times 5000 \text{ lb}}{82{,}800 \text{ lb/ft}} = 0.0181 \text{ ft (5.52 mm)}$$

The value of ω remains unchanged, so using Eq. (3.12) gives

$$\ddot{Z}_{\text{max}} + 23.09^2(0.0181) = 0$$

$$\ddot{Z}_{\text{max}} = -9.65 \text{ ft/s}^2 \; (-2.94 \text{ m/s}^2)$$

where the minus sign indicates deceleration. This is the value previously determined that gave the velocity of 9.83 ft/s (3.0 m/s) asked for.

In part 4, acceleration was found to be –61.90 ft/s^2 (–18.87 m/s^2). From Eq. (3.12),

$$-61.90 + 23.09^2 Z(t) = 0$$

$$Z_{\text{max}} = 0.116 \text{ ft (35.4 mm)}$$

$F_i = 0.116$ ft (82,800 lb/ft) = 9610 lb (42.76 kN) which is the previous result that led to 192% impact.

Rotational Motion

The dynamic effects accompanying rotational motion include inertial torque, centrifugal force, and load pendulum action. For many cranes, centrifugal force produces a twofold action: it creates a small tensile force on the boom that slightly relieves the axial compressive forces, and more importantly, it increases the overturning moment on the crane.

In the overall scheme of things, centrifugal force on the dead-load masses of cranes can usually be neglected, since swing speeds rarely exceed 1 r/min. However, centrifugal force throws the load out to an increased radius, enlarging the overturning moment (Figure 3.2). If n is the number of slewing revolutions per minute, we have

$$F_c = \frac{WR}{g}\left(\frac{\pi n}{30}\right)^2 \tag{3.17}$$

where F_c = centrifugal force
W = weight of load
R = operating radius
g = acceleration due to gravity

F_c will act horizontally at the upper load sheave shaft away from the axis of rotation and parallel to the horizontal projection of the boom centerline (see Chap. 4).

Load pendulation is another matter. This dynamic motion can be induced by the inertia of the load as the crane powers up or brakes during swinging, *trolleying*, or travel, or it can be driven by wind or the motion of a waterborne vessel. The inertial force is horizontal, tangential to the slewing arc. Consider a slewing boom. Where braking occurs and the boom slows to a stop, load inertia will continue its linear motion until a force equilibrium is reached. After the boom stops, the load will swing to and fro as a pendulum.

The period of swing will vary with hoist line length L, and the distance from load CG to the suspension point on the boom, with the relationship

$$T_p = 2\pi\left(\frac{L}{g}\right)^{1/2} \tag{3.18}$$

where T_p is the swing period. Line length is a random variable, which makes T_p a random variable as well, but T_p will increase with L. If L is such that T_p corresponds to the natural period of vibration of the boom, resonance should occur. In a pure theoretical undamped system this will lead to a steady increase in vibration amplitude and eventual failure. In a real-life system with damping always present, amplitude will reach a peak value of some 5 to 10 times the effect of the same force statically applied. Fortunately, in actual systems the structure has a period of vibration that is always shorter than load pendulum period; thus resonance does not take place. The FEM therefore suggests doubling the values obtained when using mean acceleration. The factor 2 is taken to account for the elasticity in the system. For some very rigid devices, smaller values may be appropriate. On a practical level, the problem is lessened because most crane operators learn to minimize load pendulation.

During acceleration, the load will lag behind and the angle θ that the load hoist rope makes with the vertical is given by

$$\theta = \tan^{-1}\frac{a}{g} \tag{3.19}$$

where a is the tangential swing acceleration. Following this geometric representation, if W is the load weight, the inertial force F causing the lag can be expressed as

$$F = W\tan\theta = \frac{Wa}{g} = ma \tag{3.20}$$

which is the familiar basic inertial-force relationship. It should be noted that inertial force will vary inversely with stopping or powering-up time; that is, if the time doubles, the acceleration rate and inertial force will halve.

In terms of rotational motion the basic inertial equation can be stated as

$$T = J\alpha \tag{3.21}$$

where T = inertial torque
J = polar moment of inertia
Σmr^2 = moment of inertia of all rotating masses about axis of rotation
α = angular acceleration

For any point on the rotating structure, its tangential acceleration is $a = \alpha r$, where r is the radius to the point under consideration. The customary units for α are radians per second squared.

In calculating rotational inertial forces a convenient concept is that of *equivalent mass:* the single concentrated mass located at a particular radius and having the same rotational characteristics as the distributed masses it represents. It is a lumped mass. The concept can be expressed by the equation

$$m_e = \frac{md^2}{D^2}$$

where m_e at radius D is the equivalent of mass m whose CG is at radius d. This can be generalized as

$$m_e = \sum \frac{m_i d_i^2}{D^2} \tag{3.22}$$

to find the mass equivalent of a series of individual masses.

The inertial force F produced by angular acceleration α at a point at radius D from the center of rotation is therefore

$$F = m_e \alpha D = \frac{\alpha \Sigma m_i d_i^2}{D}$$

and the inertial torque is $T = m_e D^2 \alpha = \alpha \Sigma m_i d_i^2$.

To calculate bending moments induced in a swinging boom or jib by inertial forces, the *equivalent mass* of only those portions of the boom at greater radius than the moment plane are used. The inertial force acting at the CG of this mass is then found. The moment arm is the difference in radius between the moment plane, or cross section, and the CG of the mass.

Example 3.2

1. What is the magnitude of the centrifugal force when an 11,000-lb (4990-kg) load (including the block) swings at 0.7 r/min at a radius of 100 ft (30.5 m)?

Solution

From Eq. (3.17),

$$F_c = \frac{(11{,}000 \text{ lb})(100 \text{ ft})}{32.2 \text{ ft/s}^2}\left[\frac{(0.7 \text{ r/min})(\pi)}{30}\right]^2 = 184 \text{ lb (817 N)}$$

which is only about 1½% of the load.

2. What load-suspension rope length will permit resonance to occur during slewing speed changes if the natural period of vibration of the boom is 1.2 s?

Solution

From Eq. (3.18),

$$1.2 \text{ s} = 2\pi\left(\frac{L}{32.2 \text{ ft/s}^2}\right)^{1/2}$$

$$L = 32.2\left(\frac{1.2}{2\pi}\right)^2 = 1.17 \text{ ft (358 mm)}$$

3. If the swing brake is applied, causing the boom to come to a stop in 3.6 s from 0.7 r/min, what will be the inertial force induced by the load in part 1 and what angle will the load rope make with the vertical?

Solution

There are 2π rad in 360° (or 1 revolution), so the angular velocity ϕ is

$$\phi = 0.7 \text{ r/min} \frac{2\pi}{60 \text{ s/min}} = 0.073 \text{ rad/s}$$

Then $\phi = \alpha t$, or

$$\alpha = \frac{0.073}{3.6\ \text{s}} = 0.020\ \text{rad/s}^2$$

Since tangential acceleration $a = \alpha r$,

$$a = 0.020(100\ \text{ft}) = 2.0\ \text{ft/s}^2 (0.61\ \text{m/s}^2)$$

From Eq. (3.20),

$$F = \frac{Wa}{g} = (11{,}000\ \text{lb})\frac{2.0}{32.2} = 683\ \text{lb}\ (3.04\ \text{kN})$$

which is the inertial force called for.

From Eq. (3.19),

$$\theta = \tan^{-1}\frac{2.0}{32.2} = 3.6°\ \text{from the vertical}$$

Example 3.3 A tower-crane slewing motor can deliver 100,000 ft · lb (135.6 kN · m) of torque and is set for 1.0 r/min maximum slewing speed. If the entire slewing portion of the crane weighs 150 kips (68.04 t) and the CG is located 30 ft (9.14 m) from the axis of rotation, how many seconds will be required to power up to full swing speed in calm air?[17] How many degrees of swing arc will be needed?

Solution

The mass moment of inertia (polar) of the system is

$$J = \frac{150\ \text{kips}}{32.2\ \text{ft/s}^2}(30\ \text{ft})^2 = 4192.5\ \text{kip} \cdot \text{ft} \cdot \text{s}^2\ (579.6\ \text{t} \cdot \text{m} \cdot \text{s}^2)$$

From Eq. (3.21),

$$\alpha = \frac{T}{J} = \frac{100\ \text{kip} \cdot \text{ft}}{4192.5} = 0.024\ \text{rad/s}^2$$

$$t = \frac{\phi}{\alpha} = \frac{(1.0\ \text{r/min})(2\pi)}{0.024(60\ \text{s/min})} = 4.36\ \text{s to full speed}$$

$$\theta = \frac{1}{2}\alpha t^2 = 0.23 \quad 0.23\frac{360°}{2\pi} = 13.2°\ \text{of arc traversed}$$

Equations (3.7) and (3.12) have rotational-motion counterparts where $Z(t)$ is replaced by angular displacement $\phi(t)$. Carrying this substitution to the solution Eq. (3.13) gives

$$\phi(t) = \phi_0 \cos \omega t + \frac{\dot{\phi}_0}{\omega} \sin \omega t \tag{3.23}$$

and

$$\phi_{\max} = \left[\phi_0^2 + \left(\frac{\dot{\phi}_0}{\omega}\right)^2\right]^{1/2} \tag{3.24}$$

[17] 1 kip = 1000 lb.

where ϕ_0 is the initial angular displacement and ϕ_0 is the initial angular velocity. ω has the same definition as before.

3.5 Wind Loads

All practical calculations involving wind are approximate. This notion is dictated by the nature of wind, but it is not rational to treat wind effects loosely; doing that would only lead to a coarser approximation and uncertainty. More accurate and useful results are obtainable when the mathematical treatment of wind is carefully performed. But mathematical perfection alone is insufficient; lacking a sound understanding of wind behavior, an engineer is prone to commit conceptual errors.

With respect to cranes, a large concern about wind is out-of-service storm loading. Among certain crane types, especially tower cranes, it is a principal design consideration. Yet there is no broad-based agreement on treating cranes with respect to storm wind. The FEM standard is commonly used around the world, but the FEM treatment of winds is most appropriate for typically mild continental European climates. It should be applied judiciously in the more severe wind regions found elsewhere and even in some parts of Europe. With tower-crane manufacturers in the lead, European thinking is changing toward considering local geography as a factor in design loading.[18]

The authors have long advocated their own ideas that emphasize, among other considerations, the geographic variability of wind. Recent developments in consensus standards ASCE 7[19] and ASCE 37[20] have given the authors an opportunity to dovetail these ideas with the mainstream of U.S. structural engineering practices. A growing trend among U.S. building officials, specification writers, and design engineers is to require that tower crane applications comply with one or both of these standards. Methods presented herein should provide crane application engineers some means to do so.

The Nature of Wind

Wind is nothing more than the fluid movement of air. Contact of the moving air with the Earth's surface exerts a drag effect similar to friction. One can readily visualize that the drag induced by open prairie will be less than that caused by woodlands, which in turn will be less

[18]Starting in 2010, a consortium of major European tower crane manufacturers has agreed to begin using EN 14439 in Europe. This CEN standard divides Europe into wind intensity zones and introduces other out-of-service considerations not previously considered, such as recurrence intervals and frontal wind.

[19]ASCE/SEI 7-05 Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers.

[20]SEI/ASCE 37-02 Design Loads on Structures During Construction, American Society of Civil Engineers.

than that produced by a dense urban area. This drag causes wind speed to vary with height and with the coarseness of the landscape. Furthermore, topographical features, such as hills, valleys, and city streets, channel wind to create lateral variations.

At any particular location, actual wind measurements will continuously and randomly vary in both velocity and direction. Within the randomly varying pattern, a trend or average can be discerned for any time interval.[21] Velocity is not reported as a peak, or gust, value but rather as an average over a defined period. A short sampling period implies less gust variation from the reported velocity than a long period.

Weather bureaus furnish historical storm wind speed data, which is the basis for structural design. In Europe, the standard method for reporting maximum storm winds is based on a 10-min average in open country at 10 m above the ground. Until recently, in the United States, the standard method was based on the time it takes 1 mi (1.6 km) of air to pass the monitoring point at an elevation of 10 m (33 ft). This *fastest-mile* wind is an average of the gusts and lulls that occur during a maximum period of 1 min when the wind is blowing at 60 mi/hr (34 m/s). In 1995, a new wind standard was adopted in the United States that uses storm winds averaged over a 3-s period.[22] The new sampling period closely matches the sensitivities of common instruments and also the human perception of wind. U. S. design codes have adopted the 3-s average as the norm, by and large, and it is also used in Australia.

Each methodology for applying statistical wind implies an associated factor for gusts that is calibrated to the sampling period; other factors such as for directionality and for variation with terrain and height may be part and parcel of the method as well. To avoid a drastic error, the analyst relying on this data must know which kind of wind standard and most particularly what averaging time is being used and then apply the correct constellation of associated factors to the basic wind. All else being the same, no matter what sampling period, the end result should be approximately the same wind pressure and forces acting on the crane.

Wind-Velocity Pressure

Air at rest at sea level induces an ambient pressure of about 14.7 lb/in^2 (101.4 kN/m^2) absolute. When air is in motion, this normal pressure changes by a small percentage as it moves around an object. These local low-level variations exert pressure on the surface of the object.

---

[21]For example, see Typical Wind Speed Recording in J. Kogan, "Crane Design: Theory and Calculations of Reliability," Israel Universities Press, Jerusalem, 1976, p. 212.

[22]ASCE 7-95, Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers.

The pressure can be windward or leeward, the latter being a form of suction. To put the phenomenon in scale, a change of only 1% in normal pressure is equivalent to 21 lb/ft^2 (1014 N/m^2)—a mighty wind capable of great destruction.

The equivalent static pressure induced by wind is a function of the density of air, which varies with temperature, elevation, and barometric pressure. These variations are small, however, and are usually ignored in making practical wind calculations. The static-pressure relationship is given by

$$q = \frac{1}{2}\rho v^2 \tag{3.25}$$

where ρ is the density of air. For U.S. customary units, when velocity v is given in miles per hour, the resultant pressure q is in pounds per square foot and Eq. (3.25) takes the form

$$q = \frac{v^2}{391} \approx \frac{v^2}{400} \tag{3.25a}$$

For velocity in meters per second and pressure in Newtons per square meter, the expression is

$$q = 0.613v^2 \approx \frac{5v^2}{8} \tag{3.25b}$$

The Earth's surface and its features have a retarding effect on wind velocity and tend to increase gustiness. With increase in elevation above ground surface, the drag effect becomes less pronounced, until an elevation is reached at which the wind is free of drag. This drag-free elevation is higher than heights that would be germane to most crane installations (700 to 1500 ft, or 213 to 460 m); conversely, in all but the most extreme instances cranes are installed in the drag-effect zone.

Design guidance to account for the drag effect has been available for more than half a century. A 1954 U.S. Navy publication gives a power-law expression using an exponent of 1/7, apparently with a view toward coastal installations.[23] Another early source, given in Table 3.1, differentiates between inland and coastal areas. The late Canadian researcher A.G. Davenport proposed power-law exponents of 1/7 for open country and coastal areas; 1/4.5 for wooded areas, towns, outskirts of cities, and rough coastal areas; and 1/3 for the centers of large cities.[24] Davenport's values, referenced to fastest-mile

---

[23]*Tech. Publ.* NAVDOCKS TP-te-3, U.S. Navy, Bureau of Yards and Docks, Washington, May 15, 1954, p. 9.

[24]A. G. Davenport, A Rationale for the Determination of Design Wind Velocities, proc. pap. 2476, *J. Struct. Div. ASCE*, vol. 86, no. ST5, May 1960, pp. 39–68.

Height Zone, ft†	Basic Wind Velocity, mi/h									
	60	67	75	80	85	90	95	100	115	130
For inland areas										
0–50	60	70	75	80	85	90	95	100		
50–150	70	80	90	95	100	105	110	120		
150–400	80	90	100	110	115	125	130	140		
400–700	90	100	115	120	130	135	145	150		
700–1000	100	110	125	130	140	145	155	160		
1000–1500	105	115	130	135	145	150	160	165		
For coastal areas										
0–50	60	70	75	80	85	90	95	100	115	130
50–150	85	95	100	105	110	115	120	125	140	150
150–400	115	125	130	135	140	145	150	155	170	180
400–600	140	150	160	165	170	175	180	185	190	195
600–1500	150	160	165	170	175	180	185	190	195	200

†1 ft = 0.30480 m; 1 mi/h = 0.44704 m/s = 1.609344 km/h.

Source: William McGuire, *Steel Structures*, Prentice-Hall, Englewood Cliffs, N.J., 1968; used by permission.

Table 3.1 Fastest Mile of Wind for Various Height Zones above Ground†

wind, were adapted into an ANSI standard.[25] A set of curves using these values is given in Figure 3.7.

The variation of velocity with height h is given by

$$v = v_0\left(\frac{h}{h_0}\right)^p \tag{3.26}$$

where v_h = velocity at height h above adjacent ground
v_0 = reference velocity at the standard height h_0, 30 ft or 10 m
p = power-law exponent

Wind Pressure on Objects

The effect that wind has on an object has much to do with the object itself. Shape, orientation to the wind and the juxtaposing of multiple objects all come into play. Equation (3.25*a* and *b*) give the basic static pressure induced by wind on an object without consideration of these factors. Obviously a knife-edged object will cause little disturbance or pressure change in a windstream while a large flat surface will have quite the

[25]ANSI A58.1–1982, Building Code Requirements for Minimum Design Loads in Buildings and Other Structures, American National Standards Institute. This is the antecedent to the current ASCE/SEI 7 standard.

opposite effect. Through research and tests, data have been gathered relating the shape of objects to the resistance they induce. Table 3.2 gives values for these force coefficients for a number of shapes common to crane designs.

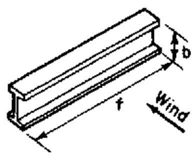
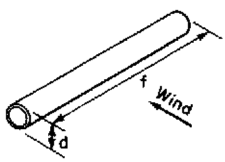
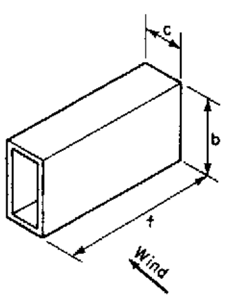
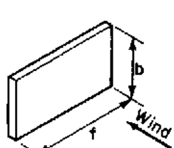
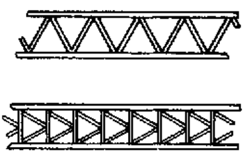
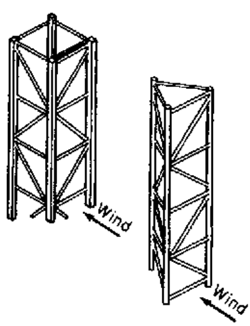
			f/b	C_f
Profiles, angles, box sections (small)			50	1.90
			40	1.70
			30	1.65
			20	1.60
			10	1.35
			5	1.30
			f/d	
Tubes		$dV < 50$ ft²/s where V = wind velocity, ft/s, and d is in feet	50	1.10
			40	1.00
			30	0.95
			20	0.90
			10	0.80
			5	0.75
		$dV \geq 50$ ft²/s	50	0.80
			40	0.75
			30	0.70
			20	0.70
			10	0.65
			5	0.60
			f/b	
Large box sections, over 14 in square and 10 by 18 in rectangular		$\frac{b}{c} \geq 2$	40	2.20
			30	2.10
			20	1.95
			10	1.75
			5	1.55
		$\frac{b}{c} = 1$	40	1.90
			30	1.85
			20	1.75
			10	1.55
			5	1.40
		$\frac{b}{c} = \frac{1}{2}$	40	1.40
			30	1.35
			20	1.30
			10	1.20
			5	1.00
		$\frac{b}{c} = \frac{1}{4}$	40	1.00
			30	1.00
			20	0.90
			10	0.90
			5	0.80
			f/b	C_f
Flat plates above ground level			≥ 80	2.00
			60	1.85
			40	1.75
			20	1.50
			15	1.40
			10	1.30
			≤ 5	1.20

TABLE 3.2 Wind-Pressure Coefficients C_f

Wire rope	Wind normal to rope				1.20
				φ†	
Latticed frames		Profiles, angles, box sections, plates		≤0.05	1.95
				0.10	1.90
				0.20	1.75
				0.30	1.60
				0.40	1.45
				0.50	1.45
	Single frames, wind normal to face (use shielding factor η_m for multiple frames)	Tubular members	$dV < 50\ \text{ft}^2/\text{s}$	≤0.05	1.30
				0.10	1.25
				0.20	1.20
				0.30	1.10
				0.40	1.05
				0.50	1.05
			$dV \geq 50\ \text{ft}^2/\text{s}$		0.80
		Square frames of profiles, angles, box sections, and plates		<0.025	4.0
				0.025–0.45	$4.13 - 5.18\varphi$
				0.45–0.7	1.8
				0.7–1.0	$1.33 + 0.67\varphi$
		Wind on diagonal, multiply normal by $1.0 + 0.75\sigma$			
				φ†	
		Triangular frames of profiles, angles, box sections, and plates		<0.025	3.6
				0.025–0.45	$3.71 - 4.47\varphi$
				0.47–0.7	1.7
				0.7–1.0	$1.0 + \varphi$
	Assembled frames, square or triangular, wind normal to face	Square and triangular frames with tubular members, multipliers for values above		<0.3	$^2/_3$
				0.3–0.8	$0.66\varphi + 0.47$
				0.8–1.0	1.0

†$\varphi = A_f/A_g$ where A_f = sum of face areas of members in frame and A_g = gross area enclosed by borders of frame.

TABLE 3.2 Wind-Pressure Coefficients C_f (*Continued*)

When one object is in front of another identical object as in a latticed boom or mast, the "shadow" effect can be considered. Both the profile shapes and the spacing between the objects have a bearing. Figure 3.9 gives values for the shielding coefficient η, which represents that part of the wind on the first of two bodies which acts on the second body. The coefficient can then be applied successively to additional bodies. However, this consideration of shielding effect is only applicable to matching objects or latticeworks, not to incidental shielding such as from nearby buildings.

Wind-pressure effects diminish if the wind is not head-on to the surface under consideration. The effective pressure in the direction of the wind becomes the head-on value multiplied by the square of the sine of the angle between the wind and the surface.

The force the wind exerts on an object can now be expressed as

$$F = q_h AC \tag{3.27}$$

where F = force on surface which acts in direction of wind
q_h = static pressure at height h of object
A = face area of object on which wind acts
C = configuration coefficient
= C_f for wind acting normal to object
= $C_f \eta_m$ for consecutive identical equidistant objects
= $C_f \sin^2\theta$ for wind at angle to object
C_f = force coefficient as given in Table 3.2
η_m = cumulative effect o shielding on m identical equidistant objects
$= 1 + \eta + \eta^2 + \eta^3 + \cdots + \eta^{m-1} \le (1-\eta)^{-1}$
η = shielding coefficient as given in Figure 3.9
θ = acute angle between wind and surface

When a surface is at an angle to the wind and the effect normal to the surface is required, the force can be resolved in the ordinary manner. For wind blowing on the diagonal to a square tower, however, the value for C may be taken as 1.0 + 0.75φ times the head-on value for a single tower face with shielding of the second face taken into account. φ is the ratio of the solid area to the gross area of the face. The force will act on the diagonal.

A working value for q_h can be found by using Eq. (3.25*a* or *b*) with an adjustment for height using Eq. (3.26) or an equivalent method, but these equations can be combined to form

$$q_h = \begin{cases} \dfrac{V^2}{391}\left(\dfrac{h}{h_0}\right)^{2p} & \text{U.S. Customary units} \quad (3.28a) \\[2ex] 0.613V^2\left(\dfrac{h}{h_0}\right)^{2p} & \text{SI units} \quad (3.28b) \end{cases}$$

As explained in the section Storms and Statistics, the reference velocity V is a variable that should be ascertained on the basis of the local wind climate. In ASCE 7-05, the mathematical model for wind pressure variation over height follows the same general form as above, but terms are changed; the expression $(h/h_0)^{2p}$ is replaced by the variable k_z, where

$$K_z = 2.01(z/z_g)^{2/\alpha}$$

Exposure Category	α	Z_g ft (m)
A*	5.0	1,500 (457)
B	7.0	1,200 (366)
C	9.5	900 (2274)
D	11.5	700 (213)

*Zone A was deleted in ASCE 7-02 and 7-05. It is kept in this table for continuity.

TABLE 3.3 Height and Terrain Regime Adjustment Parameters for ASCE 7-05

The parameter α is the power-law exponent, and z_g the gradient elevation for the terrain regimes. Applicable values of α and z_g are given in Table 3.3. This exponential form is normalized for the 3-s sampling period.

Davenport's definitions for terrain regimes have been modified somewhat. The descriptions that follow are condensed; the Standard or other recognized literature should be consulted for more precise definitions or for doubtful cases.

Exposure A. Centers of large cities. Exposure A has been removed from ASCE 7.

Exposure B. Urban and suburban areas, woods or other terrain with numerous closely spaced obstructions the size of single-family dwellings or larger, prevailing in the upwind direction for a distance of at least 2600 ft (800 m).

Exposure C. Open terrain with scattered obstructions, including surface undulations or other irregularities having heights generally less than 30 ft, prevailing in the upwind direction for a distance of at least 2,600 ft (800 m).

Exposure D. Flat unobstructed areas exposed to wind flowing over flat unobstructed surfaces such as open water for a distance of at least 5000 feet (1500 m).

Consideration of Tall Objects

When analyzing tall vertical objects such as towers, a common practice is to divide the height into discrete bands and to calculate a pressure approximate to each band. A simpler and more conservative approach is to use the pressure at the top for the full height, but for towers greater than 50 m (165 ft.) or so, this simplification may be too conservative.

The question arises: What reference height should be used for determining the design wind pressure q_h? For a body whose top is at height h_2 and lower end at h_1, where $h_1 \geq h_0$, and with width b and a constant shape factor C, the wind moment about ground level will be

$$M = C\int_{h_1}^{h_2} q(y)y\,dA$$

where $q(y) = q_0(y/h_0)^{2p} = q_0 h_0^{-2p} y^{2p}, y = 0$ at ground level, and the differential area is $dA = b\,dy$. Then

$$M = q_0 h_0^{-2p} Cb \int_{h_1}^{h_2} y^{2p+1} dy$$

$$= q_0 h_0^{-2p} Cb \frac{h_2^{2p+2} - h_1^{2p+2}}{2p+2} \tag{3.29}$$

When $h_1 = 0$,

$$M = \frac{q_0 C b h_0^2}{2} + q_0 C b h_0^{-2p} \frac{h_2^{2p+2} - h_0^{2p+2}}{2p+2}$$

$$= q_0 Cb \left(\frac{h_0^2}{2} + h_0^{-2p} \frac{h_2^{2p+2} - h_0^{2p+2}}{2p+2} \right) \tag{3.30}$$

When it is necessary to take moments about a point at height h_3 ($h_3 \geq h_0$), the moment arm y above is replaced by $y - h_3$, this yields

$$M = q_0 h_0^{-2p} Cb \left(\frac{h_2^{2p+2} - h_3^{2p+2}}{2p+2} - h_3 \frac{h_2^{2p+1} - h_3^{2p+1}}{2p+1} \right) \tag{3.31}$$

The horizontal wind shear expression is similarly found. When $h_1 \geq h_0$,

$$H = q_0 h_0^{-2p} Cb \frac{h_2^{2p+1} - h_1^{2p+1}}{2p+1} \tag{3.32a}$$

and when $h_1 = 0$,

$$H = q_0 Cb \left(h_0 + h_0^{-2p} \frac{h_2^{2p+1} - h_2^{2p+1}}{2p+1} \right) \tag{3.32b}$$

Equations (3.29) to (3.32) account for the continuous change in wind pressure with height in the Davenport model. The height to the center of wind pressure is simply M/H.

Example 3.4

1. Calculate the static wind force per unit of tower height for the square tower shown in Figure 3.8 with wind at 60 mi/h (26.8 m/s) at standard height. Take the velocity at midheight as representative of the entire tower, and use Table 3.1 (coastal areas) for velocity variation with height.

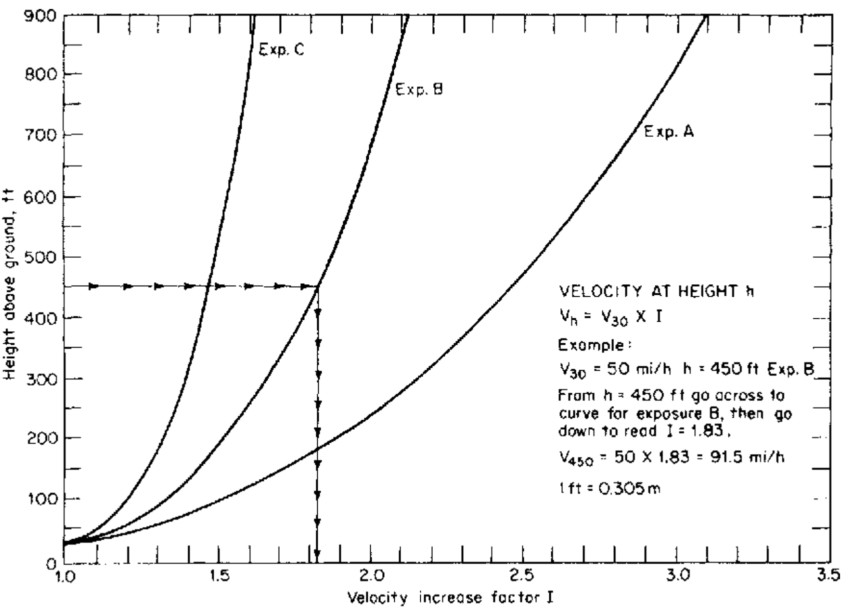


FIGURE 3.8 Tower dimensions for use in Example 3.4.

Solution

First, the force coefficients are found for each of the members using Table 3.2. For the main-chord square tubes, the length-to-width ratio is

$$\frac{f}{b} = 21 \text{ ft} \frac{12 \text{ in/ft}}{3.5 \text{ in}} = 72 > 50$$

and therefore

$$C_f = 1.9$$

For the round battens, the horizontal members, the ratio is

$$\frac{f}{b} = \frac{48 \text{ in} - 2(3.5 \text{ in})}{1.5 \text{ in}} = 27.33$$

The drag on cylindrical objects depends on whether flow is smooth or turbulent; this in turn is a function of wind speed and tube diameter. From Table 3.1 at tower midheight wind velocity is to be taken at 115 mi/h (51.4 m/s) when base-height velocity is 60 mi/h (26.8 m/s).

$$115 \text{ mi/h} = 168.7 \text{ ft/s (51.4 m/s)}$$

$$dV = \frac{1.5 \text{ in}}{12 \text{ in/ft}} 168.7 = 21.1 < 50$$

Therefore by interpolation,

$$C_f = 0.94$$

The length of the round diagonals is

$$f = [(42-1.5)^2 + (48-2\times 3.5)^2]^{1/2} = 57.6 \text{ in}(1.46 \text{ m})$$

$$\frac{f}{b} = \frac{57.6}{1.5} = 38.4 \quad dV < 50$$

From Table 3.2,

$$C_f = 0.99$$

by interpolation.

The shielding effect on the leeward face can be found from Figure 3.9. The gross area of any one-panel section of the tower is

$$A_g = (42 \text{ in})(48 \text{ in}) = 2016 \text{ in}^2 \ (1.30 \text{ m}^2)$$

and the sum of the face areas for all members in a panel is

$$\begin{aligned} A_f &= 2(3.5 \text{ in})(48 \text{ in}) + [48 \text{in} - 2(3.5 \text{ in})](1.5 \text{ in}) \\ &\quad + (57.6 \text{ in})(1.5 \text{ in}) \\ &= 441.9 \text{ in}^2 \ (0.29 \text{ m}^2) \end{aligned}$$

The ratio $A_f/A_g = \varphi = 441.9/2016 = 0.22$, and the depth-to-width ratio a/b is unity for a square tower. From Figure 3.9, the shielding coefficient η is found to be 0.72.

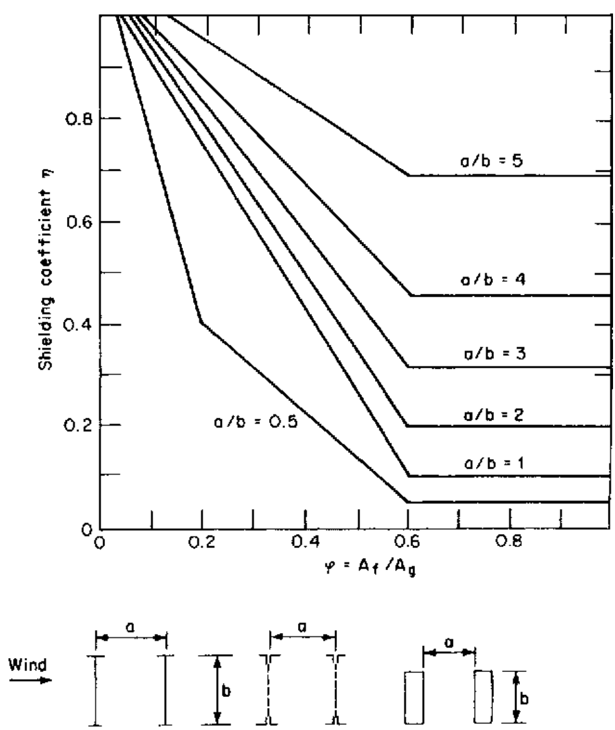


FIGURE 3.9 Shielding coefficient η. (*Reprinted by permission from Fédération Européene de la Manutention, "Rules for the Design of Hoisting Appliances," 2d ed., sec I, p. 23, Paris, 1970.*)

This means that only 72% of the wind force on the windward frame will fall on the leeward frame. For $m = 2$, $\eta_m = 1 + \eta = 1.72$.

The products of the individual surface areas and their force coefficients can now be summed for a panel and the cumulative effect of shielding taken into account; then dividing by the panel length of 42 in = 3.5 ft (1.07 m) gives the unit effective wind area AC.

$$AC = \frac{1}{3.5\text{ ft}}\left[(3.5\text{ ft})(2)(3.5\text{ in})\frac{1.9}{12\text{ in/ft}} + (48\text{ in} - 7\text{ in})(1.5\text{ in})\frac{0.94}{144\text{ in}^2/\text{ft}^2} + (57.6\text{ in})(1.5\text{ in})\frac{0.99}{144\text{ in}^2/\text{ft}^2}\right](1.72)$$

$$= 2.40\text{ ft}^2/\text{ft}\ (0.73\text{ m}^2/\text{m})$$

From Eq. (3.25*a*), the velocity pressure for winds at 115 mi/h (51.4 m/s) is

$$q = \frac{115^2}{391} = 33.82\text{ lb/ft}^2(1.62\text{ kN/m}^2)$$

and then from Eq. (3.27),

$$F = 33.82(2.40) = 81.17\text{ lb/ft}\ (1.18\text{ kN/m})$$

Had Figure 3.10 been used for velocity variation with height, for exposure C at 210.5 ft, we would have had $I = 1.32$. Then $V = 1.32(60\text{ mi/h}) = 79.2\text{ mi/h}$ (35.4 m/s),

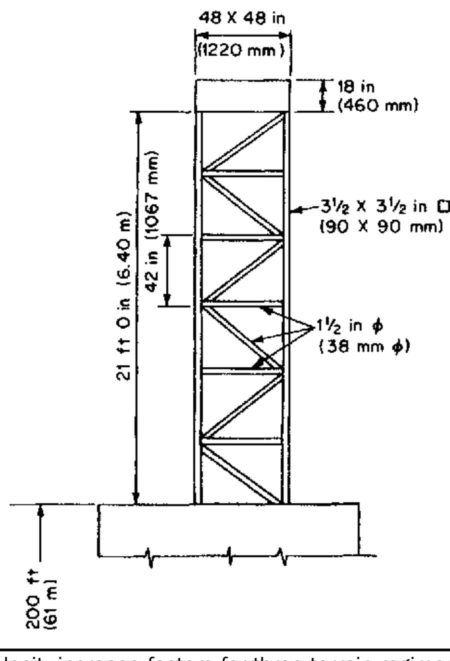


Figure 3.10 Velocity-increase factors for three terrain regimes. *Exposure A*, which is no longer allowed by ASCE 7, is at for centers of large cities. *Exposure B represents* suburban areas, towns, city outskirts, wooded areas, and hilly terrain. *Exposure C* is for flat open country, coastal belts, and grassland.

and Eq. (3.25*a*) gives $q = 79.2^2/391 = 16.04\ \text{lb/ft}^2$ (768 N/m^2). As an alternative, Eq. (3.28) makes the adjustment directly.

$$q = \frac{60^2}{391}\left(\frac{210.5}{30}\right)^{2/7} = 160.06\ \text{lb/ft}^2\ (769\ \text{N/m}^2)$$

The Davenport formulas produce pressures significantly lower than Table 3.1 apparently because the table is inclusive of gusts.

Another interpretation of Table 3.2, applying the values under single-latticed frames and then the shielding factor, the effective wind area comes to 2.32 ft^2/ft (0.72 m^2/m) instead of the previous value of 2.40 ft^2/ft. Table 3.2 distinguishes between frames with tubular members and frames of other shapes. To use these data for the structure in Figure 3.8, a separate value must be found for each type of frame. The final value is determined by applying a ratio relating each member type to the face area of the entire frame. There are 294 in^2 (0.19 m^2) of square tubes in a panel and 147.9 in^2 (0.10 m^2) of round tubes out of the full panel face area of 441.9 in^2 (0.29 m^2). From Table 3.2 with $\phi = 0.22$, for a square-tubed frame, $C_f = 1.72$ and for a round-tubular frame, $C_f = 1.18$. For the combined frame

$$C_f = \frac{(294\ \text{in}^2)(1.72) + (147.9\ \text{in}^2)(1.18)}{441.9\ \text{in}^2} = 1.54$$

$$AC = \frac{441.9\ \text{in}^2}{144\ \text{in}^2/\text{ft}^2}(1.54)\frac{1.72}{3.5} = 2.32$$

We can use Table 3.2 in yet another way and evaluate Figure 3.8 using the data for frames assembled into a square. For an assembled frame of square tubes with $\varphi = 0.22$,

$$C_f = 4.13 - 5.18\varphi = 2.99$$

and for a round-tubular frame, $C_f = 2.99(2/3) = 1.99$. For the actual frame,

$$C_f = \frac{294(2.99) + 147.9(1.99)}{441.9} = 2.66$$

Noting that shielding effects are already included in the assembled frame values, we get

$$AC = \frac{441.9}{144}\frac{2.66}{3.5} = 2.33$$

2. What will be the force on the rectangular plated section at the top of the tower using Davenport's exponent?

Solution

We use values for large box sections in Table 3.2 to get

$$\frac{f}{b} = \frac{18\ \text{in}}{48\ \text{in}} = 0.38 \qquad \text{and} \qquad \frac{b}{c} = 1$$

so $C_f = 1.4$

$$AC = 48\frac{18}{144\ \text{in}^2/\text{ft}^2}1.4 = 8.4\ \text{ft}^2 (0.78\ \text{m}^2)$$

$$h = 221.75\ \text{ft}\ (67.59\ \text{m})$$

$$q = \frac{60^2}{391}\left(\frac{221.75}{30}\right)^{2/7} = 16.31\ \text{lb/ft}^2\ (781\ \text{N/m}^2)$$

$$F = 16.31(8.4) = 137\ \text{lb}\ (609\ \text{N})$$

3. With the base wind at 100 mi/h (44.7 m/s), what force will develop on this part?

Solution

$$q = \frac{100^2}{391}\left(\frac{221.75}{30}\right)^{2/7} = 45.29\ \text{lb/ft}^2\ (2.17\ \text{kN/m}^2)$$

$$F = 45.29(8.4) = 380\ \text{lb}\ (1.69\ \text{kN})$$

Storms and Statistics

A working crane buffeted by wind in combination with other loads is exposed to the *in-service* condition. An unattended crane is exposed to *out-of-service* loads.

The wind intensity associated with in-service loading must be held in check by human intervention. Every crane manufacturer sets limits on the usage of its product in the wind, but wind is monitored and judged by the user. Operational wind should be measured at an appropriate elevation in consideration of the height of the boom and the suspended load. A jobsite anemometer is useful, but rough estimates can be made by observation of common objects in the environment on the basis of the Beaufort scale, a version of which is in Table 8.2.

Though the user has no control over the passing weather, in some circumstances out-of-service loading can be and should be held in check by human intervention. In the face of an impending major storm, some crane types that would regularly be left with boom aloft are required to be taken out of harm's way. Latticed-boom mobile cranes and self-erecting tower cranes are in this category. Before one of these cranes is placed on a site, the user should have a contingency plan for lowering the front-end attachment or executing some other prescribed measures to secure it. The minders of the crane need to keep an eye on local weather and have a crew available to muster if there is an ominous turn overnight or on a holiday.

When a crane cannot be removed from harm's way in the face of a major storm, it had better be able to withstand the worst that storm has to offer. As there is no possibility for the engineer to know in advance what intensity of wind a crane will be subjected to under such circumstances, a presumptive design wind must be substituted.

The magnitude of the presumptive wind may be selected by judgment or by applying a deterministic set of wind values, but preferably it will be derived using a sound statistically based method. Deterministic values are described in some codes such as FEM 1.001, whereas a statistical approach is intrinsic to ASCE7.[26]

People are generally aware that storm winds are more severe in some regions or localities than in others, but statistically based maps of maximum wind intensity are needed to go beyond mere anecdotes and memories of great storms. These maps show *isotach lines* that thread together places that share the same average maximum wind velocity. Wind maps can be used for the design of equipment installations within a given geographic zone when nongeographic factors are also taken into account. However, a few places such as some mountain passes and river valleys have microclimates that do not correspond to the broad pattern of isotach lines; the designer should obtain local information for such places.

The usual statistical approach begins by extracting a basic wind speed V from a wind map. This value is taken to be at a standard height of 10 m in an open-field condition; it must be factored to adjust for the actual height and terrain. V represents the average maximum wind for the given *recurrence period;* most maps use a period of 25, 50, or 100 years. Maximal winds increase with the recurrence period; for example, the value of V for a particular location in any 5-year period may be 60 mi/h (26.8 m/s), while the 100-year value may be 110 mi/h (49.2 m/s). A comparison of Figures 3.11 and 3.12, respectively, 25-year and 50-year recurrence-interval maps for extreme fastest-mile winds in the United States, illustrates this point.

Figure 3.13 is a U.S. map of 50-year winds based on the 3-s gust. A comparison of this map with Figure 3.12 confirms what is intuitive: The wind values for a 3-s gust are greater than the corresponding fastest-mile wind. For example, in New York City, the fastest-mile wind speed (50 year recurrence) is 80 mi/h (36 m/s) from the map, whereas the corresponding 3-s average speed is 98 mi/h (44 m/s). The difference is compensated by disparate valuation of the gust response factor applied with each method. Conversion from a fastest-mile to a 3-s wind or vice versa can be performed using the curve in Figure C6-4 of ASCE 7-05.[27] Use of this curve would show, for instance, that the conversion from the 3-s 98-mi/h wind to an equivalent fastest-mile wind would be $98(1.27/1.54) = 81$ mph—closely matching the map value.

A 50-year recurrence is commonly used for permanent structures; accordingly many published wind maps are for this interval. For a crane,

---

[26]FEM 1.001 is not inimical to the statistically-based approach; it provides such an approach as an alternative method and recommends its use in some circumstances.

[27]This graph is known as a Durst Curve, reported in various sources to be the work of C. S. Durst (1960).

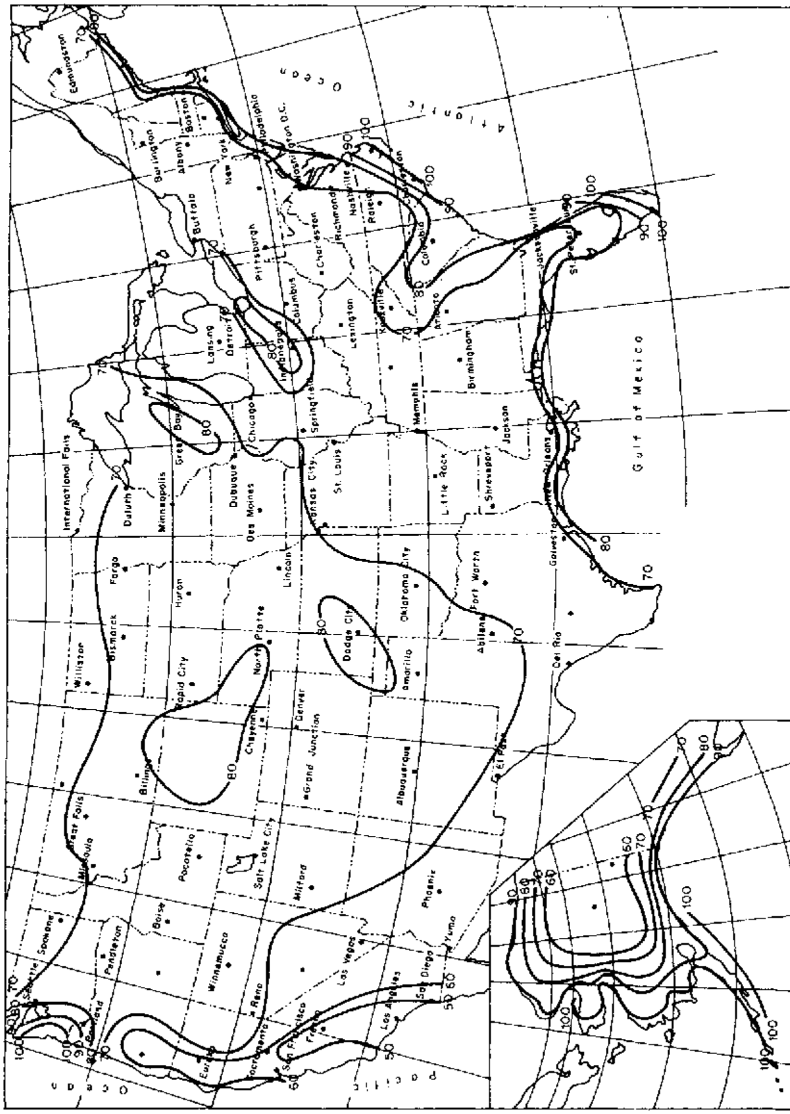


Figure 3.11 Basic wind speed in miles per hour (annual extreme fastest-mile speed 30 ft above ground), 25-year mean recurrence interval. (*From ANSI A58.1-1972; reprinted by permission of the American National Standards Institute, New York.*)

it is sensible to apply a recurrence period that is in line with the duration of the installation. In the absence of a map for a shorter period, adjustment factors are used. ASCE 37-02 Sec. 6.2.1 provides a table to adjust 50-year wind values to lesser recurrence periods germane to construction activity and Table C6-7 in ASCE 7-05 can serve the same end.

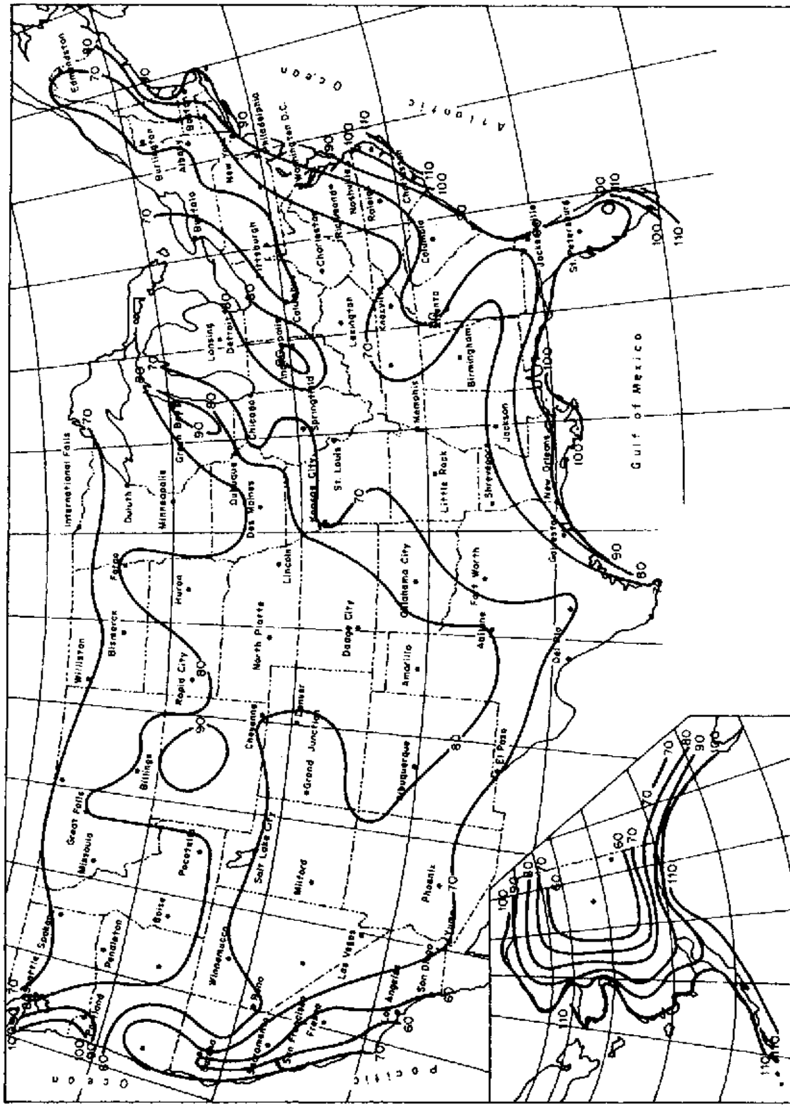


Figure 3.12 Basic wind speed in miles per hour (annual extreme fastest-mile speed 30 ft above ground), 50-year mean recurrence interval. (*From ANSI A58.1-1972; reprinted by permission of the American National Standards Institute, New York.*)

Why should the recurrence period be matched to the duration of the installation rather than the life of the crane? The reason is that each installation is an independent event. A crane in construction is not continuously exposed; it may be out of harm's way for extended stretches between assignments; each time it is erected, it is in a new

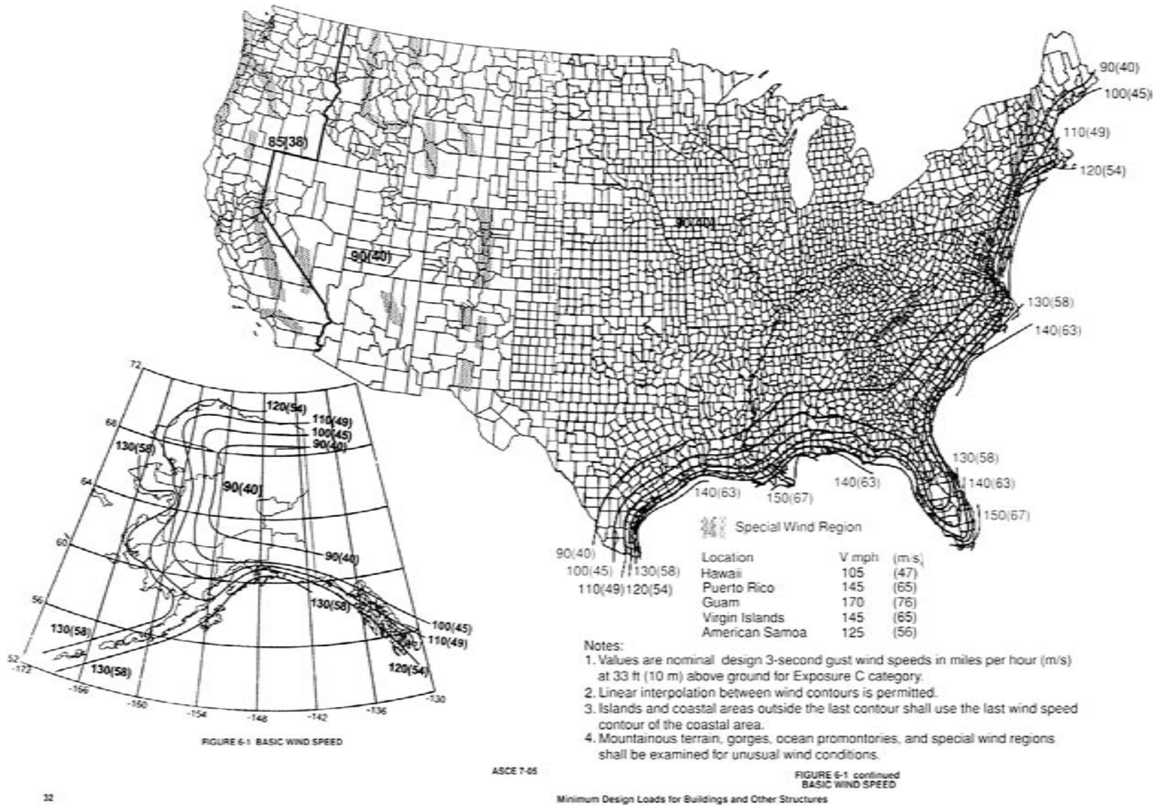


FIGURE 3.13 Basic wind speed in miles per hour (3-s gust 10 m above ground), 50-year mean recurrence interval. (*From ASCE7-05; reprinted by permission of the American Society of Civil Engineers, Reston, Va.)*

environment and likely in a different configuration with different users and beneficiaries. Unlike a building that must bear any wind that nature brings, a crane might be dismantled, braced or lowered, or its erection might be postponed. The risks associated with each installation are unique and discrete from others.

An alternative approach—one that addresses the question of risk more directly—treats the catastrophic failure due to wind as an actuarial exercise. This method requires wind statistics to be mathematically projected to reflect time periods much greater than the length of time that records have been kept. Wind events are taken as a Poisson process, inasmuch as each wind event is statistically independent of any other wind event. The probability that a catastrophic wind will not occur in time T_w is then given by

$$P(T_r > T_w) = \exp(-\gamma T_w)$$

where γ = mean recurrence rate of occasions when the crane may reach incipient failure due to gusting

T_r = time during which the crane can be expected to remain in a wind-prone environment until the catastrophic wind arrives

T_w = time during which the crane will remain installed at work site

T_r, being dependent on a random wind event, is a random variable. The cumulative distribution function is then

$$P(T_r \le T_w) = 1 - \exp(-\gamma T_w) \tag{3.33}$$

If a probability of wind-induced failure gusting of .001 is selected for any single year of operation, the recurrence period $1/\gamma$ derived from Eq. (3.33) will be 999.5 years. This same equation will then show a probability of failure .010 for 10 years of operation and .020 for 20 years. It will also show that the probability of an event occurring within its return period is .632.

But crane bases generally approximate a square. Wind typically poses potential danger for overturning when normal to a side; for structural failure wind must be normal for some crane types and on the diagonal for others. If "normal" and "on the diagonal" are extended to include a zone 12.9° to either side of these positions, the probability that a gust will be oriented to pose a threat can be taken as 25.8°/90° = 29. The extended zones thus defined comprise those arcs within which wind effects are at least 95% of the dead-center values.

When the orientation probability is considered, design gusts can occur more frequently than 999.5-year mean intervals while still maintaining .001 probability of failure, because only 29% of all gusts would be expected to be in the critical orientation.

Rational peak design wind speed selection can be achieved by studying actual design relationships. A common strength factor used in ASD is 1.67. With that factor, the relationship between design and failure wind speed should be

$$V(\text{failure}) = (1.67)^{1/2}\, V(\text{design})$$

$$= 1.29\ V(\text{design})$$

which implies a failure wind recurrence period of about 500 years if the recurrence period of the design wind is 50 years. For design codes that permit allowable stress to be increased by 1/3 when considering wind loading.[28]

$$V(\text{failure}) = (1.67/1.33)^{1/2}\, V(\text{design})$$

$$= 1.12\ V(\text{design})$$

Assuming that dead load stress is 20% of allowable stress, and a 1/3 increase is applicable, the wind induced stress would be 1.13 times allowable, and

$$V(\text{failure}) = (1.67/1.13)^{1/2}\, V(\text{design})$$

$$= 1.22\ V(\text{design})$$

In U. S. practice, stability-based ratings for tower cranes and heavy-duty-type cranes are commonly set at 2/3 of the static load that will induce overturning. It is interesting to note that this rating basis produces the same relationship between V(failure) and V(design) as the structural situation. Thus, for the general case, one design wind speed can be chosen for both structural strength and stability evaluations.

Using information derived from ASCE 7-05, the relationship between wind return periods for hurricane and other areas can be expressed in equation form[29]

$$V_R = V_{50}C \tag{3.34}$$

where V_R is the wind speed associated with recurrence interval R, V_{50} is the pertinent 50-year recurrence wind speed, and the factor C is for hurricane areas,

$$C_h = 0.33 \log R + 0.44$$

---

[28]The practice of increasing allowable stress by a factor of 4/3 for wind has a rationale, as extreme wind loading is both transient and rare, but the acceptance of this practice in American design codes is on the decline. ASCE 7 applies the inverse of this factor (0.75) when combining time-variable loads. Where the allowable stress increase is permitted, the engineer must exercise discretion when contemplating using both the stress increase and the load reduction factors. It might be justifiable only if the possibility of the simultaneous occurrence of the time-variant loads approaches nil.

[29]ASCE/SEI 7-05, Table C6-7.

and for other areas,

$$C_o = 0.23 \log R + 0.61$$

With the directional probability .29 inserted into Eq. (3.33), the joint probability which is the combined probability that a strong-enough gust will occur within the critical directional zone, the probability of structural or stability failure becomes

$$P(T_r \le T_w) = .29[1 - \exp(-\gamma T_w)]$$

Assuming a failure-level wind recurrence interval R of 25 years, $\gamma = 1/25$. For each year, the crane remains on the site $T_w = 1$, Eq. (3.35) reveals a probability of failure of .01, or 1%. For sites where it is unlikely that people would be present during high-level storm winds, the risk would concern damage to property. To maintain a lower probability of failure per year, say .001, the failure-level wind recurrence period R would need to be about 250 years.

It is more comfortable to address the probability of survival rather than failure; the probability of survival is simply 1 minus the probability of failure. The selection of an appropriate probability of survival is subjective and site specific. However, when threat to life is minimal, the decision would rest on the likely extent of property damage associated with a failure.

Example 3.5

1. A crane is to be installed at a site for about 2 years in an area where the 50-year recurrence wind is 80 mi/h (35.8 m/s) and hurricanes do not occur. Dead load stress is about 20% of the allowable stress, and the allowable stress can be increased by 1/3 when wind-induced stress is included. What design wind speed should be used if a survival probability of .999 is desired?

Solution

Using Eq. (3.35),

$$1 - .999 = .29[1 - \exp(-2\gamma)]$$

which by iteration gives $\gamma = 1/500$, or $R = 500$ years.

From Eq. (3.34),

$$V_R = 80(0.23 \log 500 + 0.61) = 80 \times 1.23$$
$$= 98 \text{ mi/h } (43.8 \text{ m/s})$$
$$V(\text{design}) = V(\text{failure})/1.22 = 98/1.22$$
$$= 80 \text{ mi/h } (35.8 \text{ m/s})$$

2. What design speed should be used if the crane remains at the site for 1 year with the same survival probability?

Solution

Using Eq. (3.35),

$$1 - 0.999 = 0.29\,[1 - \exp(-\gamma)]$$

which by iteration gives $\gamma = 1/250$, or $R = 250$ years.

From Eq. (3.34),

$$\begin{aligned} V_R &= 80(0.23 \log 250 + 0.61) = 80 \times 1.16 \\ &= 93 \text{ mi/h } (41.6 \text{ m/s}) \\ V(\text{design}) &= V(\text{failure})/1.22 = 93/1.22 \\ &= 76 \text{ mi/h } (34.0 \text{ m/s}) \end{aligned}$$

3. Suppose the survival probability is reduced to .995 for a two-year installation. How will this affect design wind speed?

Solution

$$1 - 0.995 = 0.29\,[1 - \exp(-2\gamma)]$$

which by iteration gives $\gamma = 1/115$, or $R = 115$ years.

$$\begin{aligned} V_R &= 80(0.23 \log 115 + 0.61) = 80 \times 1.08 \\ &= 93 \text{ mi/h } (38.4 \text{ m/s}) \\ V(\text{design}) &= V(\text{failure})/1.22 = 93/1.22 \\ &= 70 \text{ mi/h } (31.3 \text{ m/s}) \end{aligned}$$

4. A general-purpose construction tower crane has a useful life of perhaps 25 years, but installed duration at a typical construction site is short, say 2 years at most. As a general-purpose machine it may be installed at any geographical location. What value should be used for the design wind speed if the survival probability is .99?

Solution

$$1 - .99 = .29\,[1 - \exp(-2\gamma)]$$

which by iteration gives $\gamma = 1/60$, or $R = 60$ years.

$$\begin{aligned} C_h &= 0.33 \log 60 + 0.44 = 1.027 \\ C_o &= 0.23 \log 60 + 0.61 = 1.019 \\ V(\text{design}) &= C V_{50}/1.22 \\ &= 0.84 V_{50} \end{aligned}$$

V_{50}, the 50-year recurrence peak wind velocity, is a function of the installation geographic location, but a serially manufactured crane design has fixed properties regardless of installation location. The manufacturer, then, could prepare tables giving mast-height and jib-length limitations for a number of 50-year recurrence wind values and survival probabilities. With this information included in the crane documentation, installation designers could apply the proper crane configuration at any work site location with due consideration for local conditions.

Gusting and Gust Factors

In order to include the gust effect into the consideration of wind force acting on an object, the factor G_f is inserted into Eq. (3.27), this gives

$$F = G_f q A C \tag{3.35}$$

The gust effect of wind on an object, and hence the gust factor, depends on several contributing phenomena.

- Length of the sampling period; a shorter sampling period gives a smaller value of G_f.[30]
- Atmospheric turbulence as determined by the roughness of the terrain, the wind speed, and the height above the ground.
- The size of the object, as a gust must completely envelop the object to be fully effective.
- The flexibility and natural frequency of the object, as some oscillating objects may resonate with the wind at low frequencies that impart high energy.

Gusts are random; as a result they can cause a crane to oscillate unevenly. When a gust coincides in direction with crane displacement velocity, it adds energy to the system and increases displacement. Gusts that exceed the natural vibration period of the structure will magnify the effect of the mean wind. If the system energy carries displacement beyond a limiting value, the crane will no longer have a stabilizing moment and can overturn. Wind oscillation can also magnify stress and cause structural damage or failure.

A long gust carries more energy than a short gust; repeating, it can induce oscillation in a structure with a low natural vibration frequency. A limber crane with its mass concentrated at the top—a tower crane being the prime example—can be especially susceptible to this effect. The tendency is exacerbated if the base support is flexible. Gust effects can be an important consideration in the analysis of tower cranes.

For a gust to have the greatest effect in loading a structure such as a crane tower, it must completely envelop the tower. Portions of the gust mass must extend from a position in advance of the windward side to beyond the leeward side. The gust depth must then be from 4 to 8 times tower depth, according to various researchers. Given a 10-ft-square (3.05-m) tower and wind at 60 mi/h (26.8 m/s), a gust of at least 1/2 to 1-s duration is required to cover this depth of 40 to 80 feet. The size of a typical tower crane, not just its natural frequency, makes it particularly susceptible to energetic long gusts.

ASCE 7 provides a practical methodology for determining a gust-effect factor. The natural frequency is first assessed to determine whether the structure is flexible or stiff; if the structure has a natural frequency of more than 1.0 Hz, it is considered stiff. A stiff crane responds only to low-energy short gusts insufficient to induce resonance and can be assigned a simple gust-effect coefficient given as 0.85. The Standard does, however, permit a lower value to be sought by calculation.

---

[30]The Durst curve in Fig F03-14 implies this relationship.

ASCE7 provides a method to calculate the gust effect for a limber structure, one with a natural frequency of less than 1.0 Hz. It also permits an alternative analysis from recognized literature.

The Natural Frequency of a Crane

The natural frequency follows the form of the general expression given in Eq. (3.16). Off-the-shelf programs available to all engineers provide ready capability for determining the natural frequency of complex structures. Following is a practical hand-calculation method that yields an approximate result satisfactory for use in calculating gust-response factors. This method derives from the fact that the natural frequency is a physical property of a system that is independent of system orientation. Taking a horizontal beam of stiffness k loaded with vertical load W (see Eq. [3.12]),

$$\omega = (k/m)^{1/2} = [g/(W/k)]^{1/2} = (g/y)^{1/2}$$
$$f = \frac{1}{2\pi}(g/y)^{1/2} \tag{3.36}$$

where y is the deflection induced by W and g is the acceleration caused by gravity.

For a tower crane subjected to wind forces, for example, we can approximate the natural frequency for use in calculating gust-response factors by using the static vertical loads as though they were acting horizontally. Then

$$y = \frac{h^3}{EI}\left(\frac{Q}{3} + \frac{wh}{8}\right) \tag{3.37}$$

where y = lateral displacement at the slewing ring
Q = weight of that part of the crane above the slewing ring
w = unit weight of the tower structure
I = tower moment of inertia
E = modulus of elasticity
h = height to the slewing ring

Theory of Gust Effect on a Flexible Structure

Within ASCE 7 is a thorough practical algorithm to calculate the gust-effect factor. For those who wish to have an alternative approach or seek a deeper understanding, the following discussion is offered.

A general expression for the gust-effect is given by

$$G_f = \frac{1.0 + \lambda\sigma/\overline{P}}{d^2} \tag{3.38}$$

In this expression, $\sigma/\overline{P}$ is the ratio of the standard deviation of the wind loading to the mean wind loading and λ represents the number of standard deviations of gust-induced displacement. The variable d normalizes the gust factor to the sampling period.

t, s	2	5	10	30	60	100	200	500	1000	3600
d	1.53	1.47	1.42	1.28	1.24	1.18	1.13	1.07	1.03	1.00

TABLE 3.4 Values for Conversion Factor d

f, Hz	≥ 1	0.5	0.33	0.25	0.20	0.167	0.143
T, s	≤ 1	2	3	4	5	6	7
λ	3.00	2.90	2.80	2.70	2.65	2.55	2.50

TABLE 3.5 Suggested Values for λ

Theoretical study of gust effects on elastic structures is best performed using a long sampling period, say 1 h. In this way, the gusting portion of the wind is clearly segregated, and the gusts will contain the full spectrum of frequencies. The value of d, given in Table 3.4, is unity when the sampling period is 1 h.

The value λ is the number of standard deviations, which is related to the effective number of cycles of crane vibration during a wind-speed averaging period. The natural period of oscillation therefore has an influence on this value; with more cycles the probability a large-order response increases. That is the logic behind the relationship between λ and the crane oscillation frequency f exhibited in Table 3.5. A value of $\lambda = 3.0$ was adopted in ANSI A58.1–1972 from the research of Vellozzi and Cohen.[31] Table 3.5 suggests that this value is slightly conservative for cranes that oscillate at the more energetic—and thus problematic—frequencies under 1 Hz.

A still-useful formulation of the gust factor for flexible structures is presented here from ANSI A58.1–1972, an antecedent of ASCE 7. The result is calibrated for a fastest-mile wind and must be converted to a 3-s or any other sampling period using the Durst curve (Figure 3.14).

$$G_f = 0.65 + 1.95\frac{\sigma}{\overline{P}} \tag{3.39}$$

This formula can be derived from Eq. (3-38) for a fastest-mile wind blowing at 60 mi/h. The term $\sigma/\overline{P}$ is given a separate expression for latticed structures,

$$\frac{\sigma}{\overline{P}} = 1.7\left[T\left(\frac{2h}{3}\right)\right]\left(\frac{PF_g}{\beta} + \frac{S}{1+0.001b}\right)^{1/2} \tag{3.40a}$$

[31]J. W. Vellozzi and E. Cohen, Gust Response Factors, proc. pap. 5980, *J. Struct. Div. ASCE*, vol. 94, no. ST6, June 1968, p. 1296.

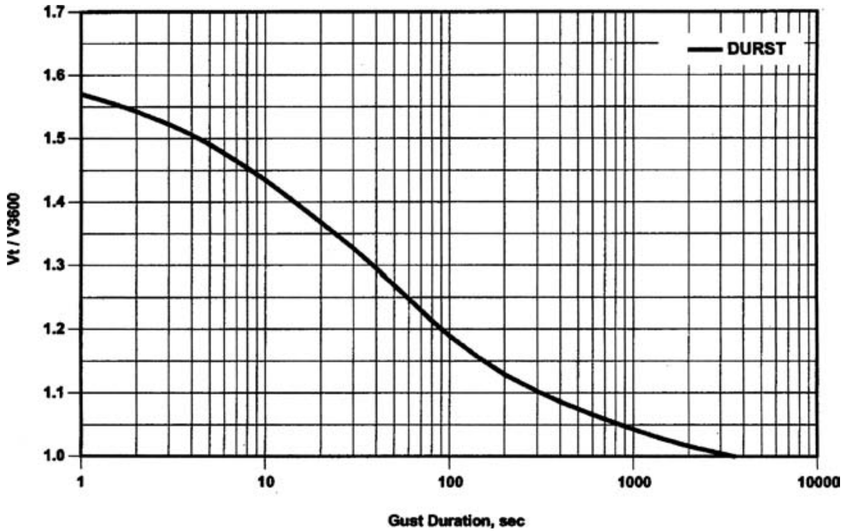


FIGURE 3.14 *Durst curve allows conversion between different recurrence intervals for statistical wind* analysis. (*From ASCE7-05; reprinted by permission of the American Society of Civil Engineers, Reston, Va.)*

and for solid structures,

$$\frac{\sigma}{\overline{P}} = 1.7\left[T\left(\frac{2h}{3}\right)\right]\left(\frac{0.785PF_g}{\beta} + \frac{S}{1+0.002b}\right)^{1/2} \qquad (3.40b)$$

where $\quad T(y)$ = exposure factor at height y, given by Figure 3.15
$T(2h/3)$ = value of exposure factor at $y = 2h/3$

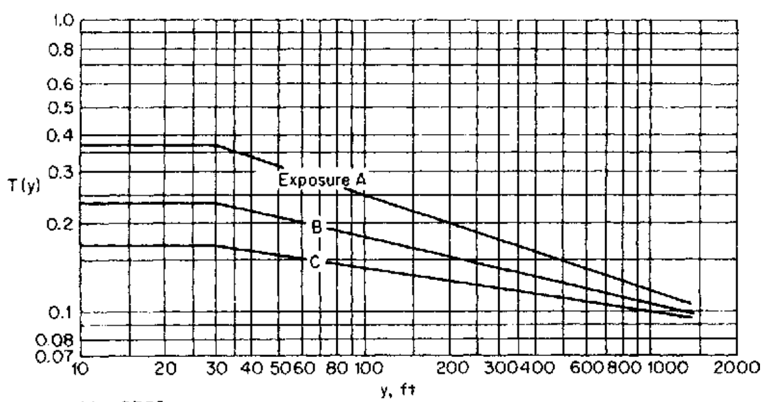


FIGURE 3.15 Exposure factor *T*(*y*). (*From ANSI A58.1-1972; reprinted by permission of the American National Standards Institute, New York.*)

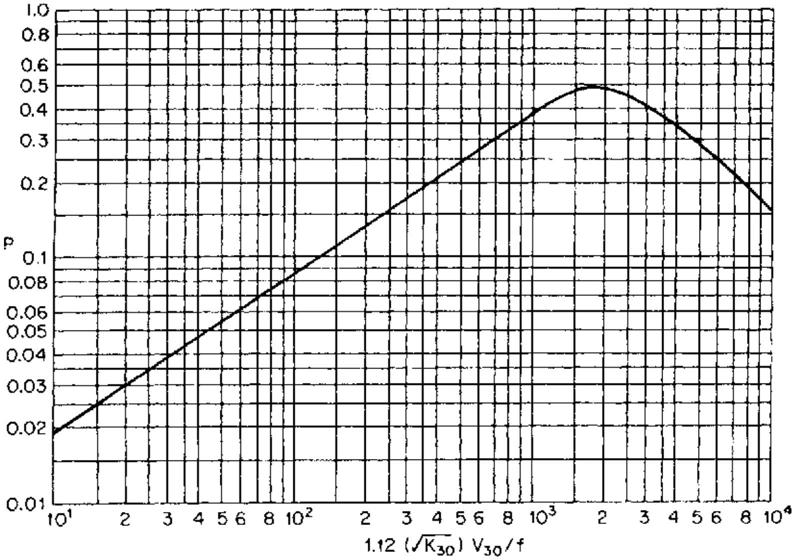


FIGURE 3.16 Gust power factor *P*. (*From ANSI A58.1-1972; reprinted by permission of the American National Standards Institute, New York.*)

P = gust power factor given by Figure 3.16
K_{30} = velocity-pressure coefficient at y = 30 ft (10 m) for exposure under consideration
K_y = velocity-pressure coefficient at height y
V_{30} = wind velocity at standard height (map value), mi/h
f = natural frequency of the structure in direction parallel to the wind, Hz
F_g = gust correlation factor as given by Figure 3.17
β = damping coefficient (percent of critical), taken here as 0.01[32]
S = structure size factor as given by Figure 3.18
b = average horizontal dimension of structure in direction normal to the wind, ft
h = height of structure measured from ground level, ft
h_e = portion of structure height exposed to wind, ft, for use in determining F_g and S

Example 3.5

1. A tower crane is to be installed at an exposure *B* site and the map value for fastest-mile 25-year-recurrence storms is 70 mi/h (31.3 m/s). The tower is 7.5 ft (2.29 m) square and stands 150 ft (45.72 m) tall. What will be the gust factor under ANSI A58.1–1972 if the natural frequency for this crane is 0.30 Hz?

[32]Values of the damping ratio in the literature vary. A typical value for a welded steel structure is 0.012 and 0.020 for bolted structures.

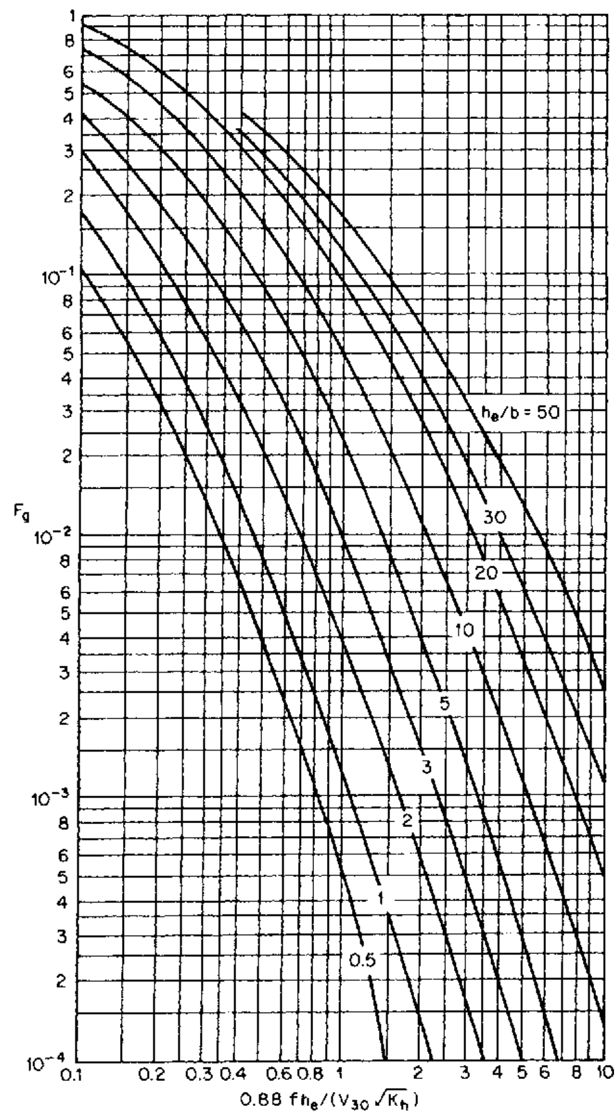


Figure 3.17 Gust correlation factor F_g. (*Adapted from ANSI A58.1-1972; by permission of the American National Standards Institute, New York.*)

Solution

From ASCE 7-05,

$$K_{30} = 0.5 \qquad \text{and} \qquad K_y = 1.02 \text{ for 150 ft (45.72 m)}$$

From Figure 3.15,

$$T(2h/3) = T(100) = 0.18$$

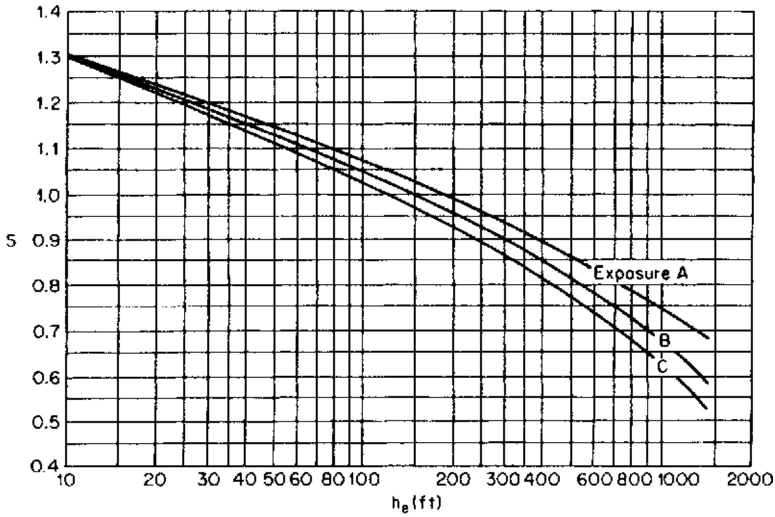


FIGURE 3.18 Structure size factor *S*. (*Adapted from ANSI A58.1-1972; by permission of the American National Standards Institute, New York.*)

From Figure 3.16,

$$1.12\sqrt{K_{30}}\,\frac{V_{30}}{f} = 1.12\sqrt{0.5}\,\frac{70}{0.30} = 185$$

Therefore

$$\text{Gust power factor } P = 0.12$$

From Figure 3.17,

$$\frac{0.88 f h_e}{V_{30}\sqrt{K_y}} = 0.88(0.30)\frac{150}{70\sqrt{1.02}} = 0.56$$

$$\frac{h_e}{b} = \frac{150}{7.5} = 20$$

Therefore

$$\text{Gust correlation factor } F_g = 0.20$$

From Figure 3.18,

$$\text{Structure size factor } S = 1.0$$

Putting these values into Eq. (3.40*a*) and taking $\beta = 0.01$ gives

$$\frac{\sigma}{\overline{P}} = 1.7(0.18)\left(0.12\frac{0.20}{0.01} + \frac{1.0}{1+0.001\times 7.5}\right)^{1/2}$$

$$= 0.564$$

Pulsation Coefficient m_g			Frequency Coefficient ξ	
Height				
ft	m	m_g	f, Hz	ξ
< 65	< 20	0.35	< 0.17	3.3
65–130	20–40	0.32	0.17–0.20	3.2
130–200	40–60	0.28	0.20–0.25	3.15
200–260	60–80	0.25	0.25–0.33	2.95
260–330	80–100	0.23	0.33–0.50	2.65
330–650	100–200	0.21	0.50–1.0	2.25
650–1000	200–300	0.18	1.0–4.0	1.75
			> 4.0	1

TABLE 3.6 Values of Wind-Gust Parameters for Eq. (3.43)

The gust factor is then given by Eq. (3.39)

$$G_f = 0.65 + 1.95(0.564) = 1.75$$

2. What would the gust factor be using Eq. (3.38)?

Solution

To find the conversion parameter d use $t = 3600/70 = 51.4$, which gives d the value of 1.25 by interpolation in Table 3.4. By interpolation, $\lambda = 2.76$ from Table 3.5, and

$$G_f = \frac{1.0 + 2.76(0.564)}{1.25^2} = 1.64$$

A Simplified Gust-Factor Approach

Finally, there is a gust-factor equation suggested by Kogan.[33] His method, based on a probabilistic analysis using wind data from the former Soviet Union, results in a deterministic expression similar to Eq. (3.39). Kogan offers

$$G_f = 1 + m_g \xi \tag{3.41}$$

where m_g is a pulsation coefficient that varies with height and ξ is a coefficient that varies with crane natural frequency. Values for both parameters are given in Table 3.6. Because wind records from the former Soviet Union were based on 2-min averaging, Eq. (3.41) yields overstated gust factors when used a 3-s sampling period. The difference can be bridged by applying the curve in Figure 3.14.

[33]J. Kogan, "Crane Design: Theory and Calculations of Reliability," Israel Universities Press, Jerusalem, 1976, p. 219.

The ASCE 7-05 Wind Load Provisions[34]

The formulation of wind force for "other structures" in ASCE 7-05 (Eq. 6-28) is applicable to cranes.

$$F = q_z G_f C A$$

where $q_z = K_z K_{zt} K_d I V^2/391$ (psf)
$q_z = 0.631 K_z K_{zt} K_d I V^2$ (N/m², m/s)
G_f = gust factor
C = shape factor
A = face area, ft² (m²)
K_z = height and terrain factor
K_{zt} = topographic factor = 1.0 except for extraordinary situations
K_d = directional factor = 0.85 or as given in Table 6-6 of ASCE 7-05
I = importance factor = 1.0 or as given in Table 6-1of ASCE 7-05
V = basic map wind speed, mi/h (m/s)

Development of the factors G_f, C, A, and V has been discussed previously. A topographic factor K_{zt} is applied when there are abrupt changes in the topography such as hills, ridges, or escarpments. The directionality factor K_d should not be used for the *weathervaning* part of a tower crane because it follows the wind.

Example 3.6[35] Using the ASCE 7-05 provisions, determine the gust factor for a tower crane with an 8-ft² mast that is 178 ft. tall. The crane is to be located in a Zone C area with a basic 50-year recurrence wind speed (3-s sampling at 10 m height) of 146 mi/h (65 m/s). The natural frequency of the tower crane has been determined to be 0.598 Hz. Damping ratio is taken as β = 0.01.

Solution

Using the variables of ASCE 7-05,

B = 8 ft (2.43 m), L = 8 ft (2.43 m), h = 178 ft (54 m), V = 146 mi/h (65 m/s), $n_1 = 0.598$

From Table 6-4 of the Standard,

$\bar{b} = 0.65$ z = 0.6 h $\bar{\alpha} = 1/6.5$ c = 0.2 l = 500 ft (152 m) $\bar{\epsilon} = 1/5.0$

Calculating on the basis of the equations in the Standard,

$$I_z = c(33/z)1/6 = 0.164$$

$$L_z = l\left(\frac{z}{33}\right)^{\bar{\epsilon}} = 632.4 \quad \text{or in SI units} \quad L_z = l\left(\frac{z}{10}\right)^{\bar{\epsilon}} = 192.5$$

$$Q = \sqrt{\frac{1}{1 + 0.63\left(\frac{B+h}{L_z}\right)^{0.63}}} = 0.88$$

---

[34]Ibid

[35] The authors are grateful to Michael Quinn and William Seeber of Morrow Equipment for providing this example. To fully follow along, it is necessary to refer to ASCE 7-05.

As the tower crane, having a frequency less than 1.0 Hz, is classified as a flexible structure,

$$g_R = \sqrt{2\,\mathrm{In}(3600n_1)} + \frac{0.577}{\sqrt{2\,\mathrm{In}(3600n_1)}} = 4.065$$

$$\bar{V}_z = \bar{b}\left(\frac{\bar{z}}{33}\right)^{\bar{\alpha}} V\left(\frac{88}{60}\right) = 166.75\text{ fps, or in SI } \bar{V}_z = \bar{b}\left(\frac{\bar{z}}{10}\right)^{\bar{\alpha}} V = 50.8\frac{\text{m}}{\text{s}}$$

$$N_1 = \frac{n_1 L_{\bar{z}}}{\bar{V}_z} = 2.268$$

$$R_n = \frac{7047N_1}{(1+10.3N_1)^{5/2}} = 0.083$$

$$\eta = \frac{4.6n_1 h}{\bar{V}_z} \qquad R_h = \frac{1}{\eta} - \frac{1}{2n^2}(1-e^{-2n}) = 0.283$$

$$\eta = \frac{4.6n_1 B}{\bar{V}_z \quad R_B} = \frac{1}{\eta} - \frac{1}{2n^2}(1-e^{-2n}) = 0.918$$

$$\eta = \frac{15.4n_1 L}{\bar{V}_z \quad R_L} = \frac{1}{\eta} - \frac{1}{2n^2}(1-e^{-2n}) = 0.716$$

$$R = \sqrt{\frac{1}{\beta} R_n R_h R_B (0.53 + 0.47R_L)} = 138$$

With g_Q and g_V both having the value of 3.4 and $\beta = 0.01$, the gust factor is

$$G_f = 0.925\left(\frac{1+1.7I_{\bar{z}}\sqrt{g_{Q^2}Q^2 + g_{R^2}R^2}}{1+1.7g_V I_{\bar{z}}}\right) = 1.317$$

The value would be higher had it been calculated to correspond with the old fastest-mile sampling period.

Summary of Procedures for Calculating Out-of-Service Wind Forces

For a crane that must stay aloft under all conditions, out-of-service wind forces might be determined by a prescriptive code, specification, or statistical method. The latter is preferred because it can be tailored to the conditions at the site and the gust response of the crane. The following is a summary of the steps for performing such an analysis; though presented to comport with ASCE 7, its rationale can be readily adapted outside of this standard.

1. Determine the basic design wind speed V from a map such as Figure 3.13 or some other suitable source. The sampling period, usually, 3-s in the United States, needs to be consistent through all the steps of the analysis.
2. Assess the terrain so that it can be categorized for its effect on wind turbulence. In ASCE 7-05, these categories are classified B, C, and D.

3. Convert V to an appropriate mean recurrence period. Tables in ASCE 7 or ASCE 37 may be used to determine a conversion factor. A period of 10 years corresponds to a factor of 0.84, a reasonable value to use for a construction crane in a project of usual duration.[36]
4. Calculate the base level wind pressure q with Eq. (3.25*a* or *b*).
5. Determine the height exposure coefficient K_h for the given height and terrain; or use a similar factor based on the Davenport model. For a tall crane, K_h might be determined separately for components at several elevations.
6. Obtain wind area A and estimate centers of pressure for each relevant crane component.
7. Determine shape factors for components from Table 3.2 and Figure 3.09, from an alternative source.
8. Calculate the natural frequency to determine whether the crane is classified as rigid or flexible; determine the appropriate gust-effect factor G_f. For a rigid crane structure (frequency not exceeding 1 Hz), G_f can be taken as 0.85.
9. For each crane component, the design wind force is the product of the base-level wind pressure q, the height exposure coefficient k_h, the exposure area A, a shape factor C, and the gust-effect factor G_f. In rare instances, a topographical factor K_{zt} and an importance factor I may also be applied. A directionality factor K_d may also be applied, though not for a weathervaning part of a tower crane.
10. Sum the forces and moments to obtain the wind shear and wind moment of the crane at a suitable reference elevation. Combine these with other forces and moments as appropriate.

To close this discussion on wind, readers are again cautioned not to allow the extensive theoretical presentation and generous supply of mathematical equations, curves, and tables of data to distort their perspective. The study of wind and its effects may be based on scientific principles, but the wind at a particular site at any instant in time has studied neither science nor engineering; it is random and beyond precise prediction. At best, wind calculation results will be approximate. The goal in calculating wind effects should be an ample, not absolute, protection against failure. That goal is often not easy to attain with assurance while avoiding unnecessary cost. It requires the engineer to perform an analytical study appropriate for the conditions and to exercise sound judgment in the process of doing so.

---

[36]In ASCE 7-05, for 10-year recurrence Table C6-7 provides a factor of 0.84 when V = 85-100 Mph (38-45 m/s) and 0.74 when V is greater than 100 Mph (34 m/s).

CHAPTER 4
Stability Against Overturning

For cranes that are not anchored in place, stability is the primary factor controlling load ratings and the ability of machines to perform useful work. Elements and actions that add and detract from overturning stability are important to understand for the designer and the user. This knowledge enhances the ability to evaluate equipment for particular jobsite tasks. It also helps us to establish jobsite policy and procedures. Though the nonengineer might have difficulty with the mathematical treatment of stability in this chapter, the crane user will have little difficulty meshing the general presentation with real-world experience.

4.1 Introduction

One of the authors was once called on to conduct a stability test of a telescopic truck crane at a contractor's operations yard. The crane operator had never done test work before but was experienced with this particular crane. For the test, a carefully weighed load was assembled, lifted just clear of the ground at close radius over the side of the crane, and slowly boomed out keeping the load just clear of the ground. When one outrigger float lifted free of the support surface, booming out was continued at a still slower rate. As the second outrigger float broke free, booming out was stopped, the load was eased to the ground, and the load radius was measured as the tipping radius for that load.

The test team took a lunch break at this point. Because of the slow, gentle procedures used, the operator had not felt the crane approach tipping; without that seat-of-the-pants indication of tipping familiar to many mobile-crane operators, he was convinced that the crane had not yet reached the tipping radius. At the end of the lunch break, the operator decided to check his intuition by trying to take the load out a little further. When the test engineer got to the scene, he started frantically waving his arms and shouting "Stop!" The load was being hoisted, but the load wasn't lifting; instead the counterweight-side outrigger floats had risen to about 2 ft (600 mm) clear of the ground!

That anecdote underscores two points for crane operators: It is imperative to know the load and the radius before making a lift, and it is good to be obsessively attuned to the behavior of the crane.

Newer cranes, particularly cranes developed for use in the European Union, have automated devices that prevent overload and modulate the crane motions, but overreliance on these devices is an unwise practice. There are a number of dangerous operating conditions, caused by the environment, handling of the load, or the support of the crane, that can defeat the cleverest of *operational aids*. Telescopic booms and long latticed booms, which are heavy and take up a large part of the tipping moment, can go from a stable to an unstable equilibrium state with no marked change in the operator's perception of machine condition. Once in motion, a large crane may develop inertia beyond the operator's ability to stop in time to prevent overturning. The small and rugged cranes of old were less capable but more forgiving of seat-of-the pants operation.

General Concept of Stability

Toward the end of Chap. 1, the main ingredients of the tipping equation were presented. It was shown how the moment of the machine superstructure and carrier about the tipping fulcrum resists overturning and hence is the *stabilizing moment*, also called the *machine resisting moment* or simply the machine moment. In opposition, the weight of the boom and the lifted load were shown to induce tipping; they produce the *overturning moment*.

This simplistic representation correctly models the general static tipping problem as a matter concerning the weights of the load and the crane components as well as the location of the various CGs and the tipping fulcrum. The real-life tipping problem also includes the action of wind and to some extent dynamic forces, the elastic deformations of the components under load, the reliability of the supporting surface, and the levelness of the crane.

4.2 Mobile Cranes

Mobile cranes are stability-sensitive, and most load ratings are governed by tipping. For a truck crane, the stability factor—the ratio of the tipping load to the rated load—can be as low as 1.18, which is less than the design factors of the structural components of the crane. The mismatch of these factors indicates that an overloaded mobile crane could tip even if the load rating chart denotes that its lifting capacity is determined by strength of materials.

Stability-governed rated loads are established for a level machine on firm supports in calm air in the absence of dynamic effects. In the presence of any one or more of these detrimental conditions, the rated, or lifted, load must he reduced in order to maintain the stability margin. Jobsite tipping accidents can result from a failure to make reductions for these factors. Some users do not read the fine print on their rating charts and so assume that the ratings hold for all conditions of operation. On the contrary, the ratings are based on ideal conditions, and appropriate reductions, based on judgment, must be made for any variation from the ideal.

With a slim margin against tipping, accuracy of the weights and CG locations of crane components are crucial. These are determined by calculation or by weighing, but the ratings and the accuracy of the weight and CG data are then confirmed by test.[1] For an engineer to perform overturning stability calculations, the location of the tipping fulcrum can easily be identified for each type or physical arrangement of crane.

Location of Tipping Fulcrum on Outriggers

The tipping line is quite clearly defined for cranes on outriggers. Operating instructions usually require that the crane be raised clear of the ground so that the tires are free of crane weight. When this is done, the tipping lines shown in Figure 4.1 apply.

When a portion of crane weight remains on the tires, the crane cannot be considered well stabilized. In the past, when screw jacks were used at the outrigger floats, the crane had to be set level while still on its tires. Although it was then possible to raise the crane off the tires by turning the superstructure first to one side and then to the other, in practice this was rarely done. More often, the screw jacks were tightened manually, using long steel bars for mechanical advantage,

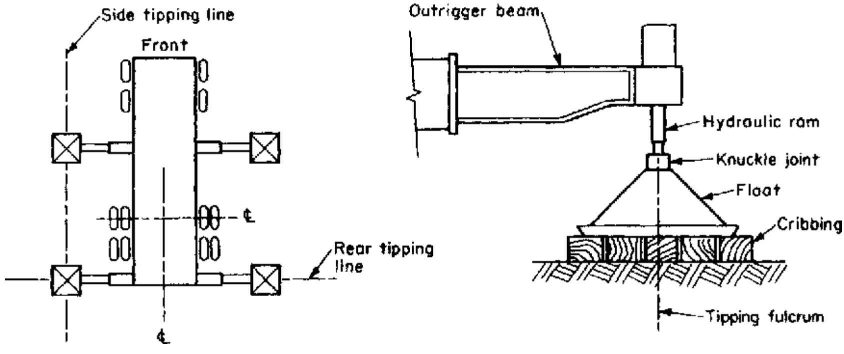


FIGURE 4.1 Tipping lines for a crane on outriggers.

[1]In the United States, mobile crane stability testing is done to SAE J765, Crane Load Stability Test Code, Society of Automotive Engineers, Warrendale, Pa., 1990.

to load the floats without raising the machine off the tires. With the relatively short-boomed cranes of that time, there was little stability difficulty, as the floats controlled the levelness of the crane fairly well. With uneven ground surfaces, however, it was often difficult to level the crane properly in the first place.

With the larger, high-capacity, longer booms of today, all cranes are equipped with hydraulic jacks at the outrigger beams or floats. This is essential, as today's machines must be raised completely off the tires. If the tires are left in hard contact with the ground when the crane is initially set up, both the tires and the outrigger beams share the load. The pressure of the tires against the ground sets up a moment that acts against the resisting moment and effectively reduces it, thereby reducing crane capacity. The crane cannot develop its proper resisting moment without first tilting to unload the tires, but this in turn increases the radius and the overturning moment.

A typical truck crane is raised on its outriggers by means of hydraulic rams on the ends of the outrigger beams (Figure 4.2). The beams are extended horizontally by rams and are controlled individually, while the floats are mounted to a knuckle that permits pivoting in any direction. With the rams on the ends of the beams, in case of hydraulic-line rupture or loss of pressure, holding valves prevent retraction and lowering of the crane.

Figure 4.2 Outrigger on a telescopic crane with a hydraulic ram at the outboard end of its outrigger beam. After the beam is extended by means of a concealed ram, the vertical end ram is activated to raise and level the crane. The wheels of the crane must be completely elevated above the ground for the outriggers to be fully effective. (*Lawrence K. Shapiro.*)

Figure 4.3 A *boom truck* is a telescopic boom joined to a truck chassis, with stabilizers. These machines are frequently used to deliver materials to an upper floor or inside a site. The stabilizers do not raise the wheels off the ground. (*Terex Corporation.*)

Some mobile lifting machines have *stabilizers* that are not true outriggers (Figure 4.3). Stabilizers resist the overturning moment, but they do not pick up the full weight of the crane. Though easier to set up, and thus well suited to smaller and highly mobile devices such as boom trucks, they are not as efficient as outriggers in effectuating resisting moment. Whereas outriggers must be designed in tandem with a carrier—one that is very stiff and strong—stabilizers are often used with a conventional truck chassis.

Extension of Outriggers

Outriggers must be fully extended as stipulated on the rating chart; otherwise, the ratings are not applicable. This can be illustrated by referring to Figure 1.22, which allows us to formulate the stability relationship as

$$W_m d_m = W_b d_b + Wd$$

noting that W_m includes the weight of both the superstructure and the undercarriage. If we take, for example, a machine with

$$W_m = 319{,}000 \text{ lb } (144.7 \text{ t}) \qquad d_m = 18.95 \text{ ft } (5.78 \text{ m})$$

$$W_b = 45{,}200 \text{ lb } (20.5 \text{ t}) \qquad d_b = 115.5 \text{ ft } (35.2 \text{ m})$$

$$d = 250 \text{ ft } (76.2 \text{ m})$$

after rearranging, the tipping load is

$$W = \frac{W_m d_m - W_b d_b}{d} = 3300 \text{ lb (1500 kg)}$$

If the outriggers are 1 ft (300 mm) short of proper extension, or 92% of specified extension for the particular crane that the data represent, d_m decreases and d_b and d increase by that amount. When those values are substituted, W becomes 1830 lb (840 kg), or only 56% of the proper tipping load. With the outriggers short of proper extension by only 3 in (76 mm), the tipping load is reduced by 11%! These figures make it quite clear why operating instructions for truck cranes specify that the crane is to be operated only with the outriggers at specified extension. They also dramatically illustrate the importance of the true fulcrum location in defining calculated stability values.

Full outrigger beam extension often proves to be a problem for operation planners and crane operators at urban construction and highway reconstruction sites. In both situations, it is commonly found that limited space is available for positioning cranes because of the need to keep traffic lanes open. Oftentimes the outrigger beams on the traffic side are not extended at all. But generally it is inadvisable to do this without some forethought and guidance from the load chart and operating manual, for several reasons.

- Some stabilizing moment is lost as a result of the outboard outriggers being retracted.
- The counterweight may still encroach over the traffic lane.
- The crane might not be operating in accordance with the manufacturer's load chart.
- Depending on the counterweight configuration, *backward stability* could be a problem.
- The operator needs to be aware that the crane will not behave the same in every direction.

No engineering reason prevents a crane from being rated for operation when the outrigger beams on one side are not extended and the boom is working on the opposite side or over the end. If the manufacturers provide such ratings or guidance, working that way is practical and legal. Without such guidance, outriggers on both sides of the crane should be extended equally as stipulated on the rating chart.

Mobile crane manufacturers often furnish rating charts for partially extending all of the outriggers. Under this condition, there is a dramatic reduction in stability when a load is swung from over the end to over the side. Only cranes specifically intended for partially extended outriggers may be so used because features of the crane must be specially designed to accommodate this configuration and the load chart must stipulate the working limitations. Outrigger beams or boxes are loaded differently when outriggers are partially extended; it is conceivable that some outrigger designs could collapse if loaded under partial extension.

Partly extended outriggers are stiffer than fully extended ones. Because the operator will feel less flex as loads are lifted and may therefore feel unduly secure, there may be a greater inclination for a seat-of-the-pants operator to lift excessive loads.

Stability calculation for a basic mobile crane with a latticed boom and perhaps a fixed jib is straightforward. It is not difficult for an engineer to assemble the required data from the manufacturer's literature to perform a stability analysis. Accessories such as luffing jibs, live masts, and auxiliary counterweights add complication, but it is still within the capabilities of a resourceful engineer to gather the required data and perform. The weight of towers, masts, and booms may act on either side of the tipping fulcrum, in some instances adding to stability and in other instances diminishing it. A live luffing component such as a mast might shift from having a stabilizing effect to having a destabilizing effect as it goes through its range of motion.

Stability analysis of a telescopic crane does not yield so readily even to a clever and resourceful engineer. The varying boom lengths include moving components hidden from view. Weights and CGs cannot be determined unless the engineer has data from the manufacturer—unlikely—or the boom is taken apart, measured, and weighed—just as unlikely.

The weight of the load ropes suspended from the boom tip is usually considered as crane deadweight, but when lowering a load below grade the rope below grade, is taken as part of the lifted load. However, some manufacturers take all of the rope weight as part of load weight; when that is the case, a note to that effect will appear on the rating chart. Using the notations of Figure 4.4, the machine resisting moment is expressed as

$$M_r = W_m d_m + W_f (d_o - d_f) - W_b d_b - W_r d$$

where W_r is the weight of the suspended hoist ropes when taken as part of crane deadweight. As the crane booms out, d_f will increase until it exceeds d_o, and that term will be self-correcting. The static overturning moment $M_o = Wd$. The limit of stability is $M_r = M_o$; suspended load at the verge of tipping is

$$W = \frac{W_m d_m + W_f (d_o - d_f) - W_b d_b - W_r d}{d} \tag{4.1}$$

Figure 4.5 shows the typical condition wherein the front end of a truck-crane carrier overhangs beyond the front outriggers. The weight

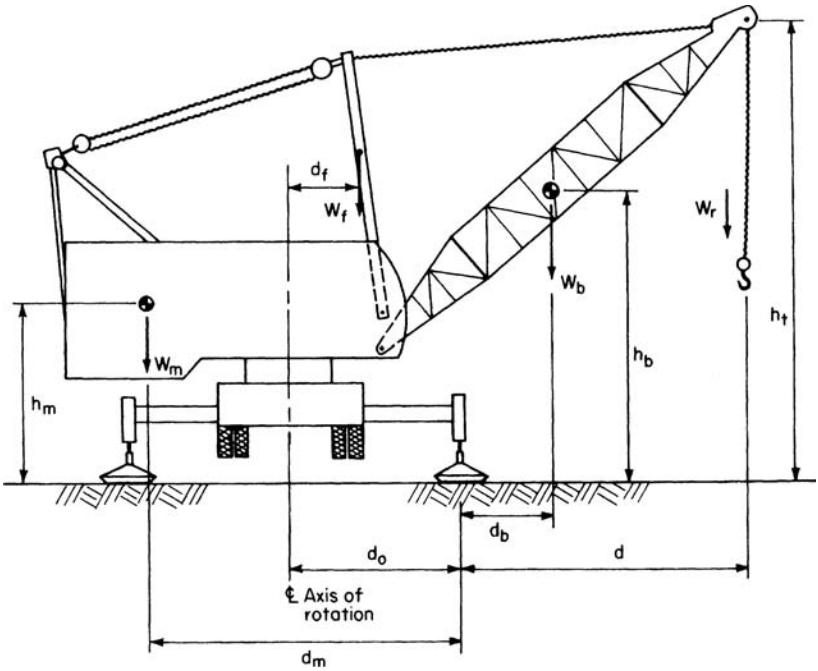


FIGURE 4.4 Crane with a live mast and supported on outriggers, annotated for stability studies.

FIGURE 4.5 A truck crane with the cab overhanging the front outrigger. The crane may be restricted from lifting over the front quadrant unless the front outrigger is set in place, as in Figure 4.6. (*Link Belt Construction Equipment Company.*)

FIGURE 4.6 Telescopic truck crane with a front outriggers that allows the crane to operate in a full circle. (*Link Belt Construction Equipment Company.*)

of that portion of the frame, the driver's cab, and the carrier engine will then materially reduce the machine moment for operation over the front. The reduction is so great that the load charts for latticed boom cranes often prohibit lifting in that *operating sector*. For those machines for which operation is permitted, a jack or outrigger generally is used at the very front of the crane (Figure 4.6).

The same restrictions do not apply to rough-terrain cranes and to two-axle or four-axle all-terrain cranes having outriggers positioned at the four corners of the carrier. Those machines can safely pick loads over a full horizontal circle. But some larger multiaxle all-terrain machines have lifting limitations over the front of the carrier similar to truck cranes.

Location of Crawler-Crane Tipping Fulcrum

A crawler crane must operate on a level surface or on a run of timber or other material laid to level the crane path. The crane itself has no leveling mechanisms. The bulldozer-like tracks or treads the crane uses for traveling are loose cast-steel segmented bands that bear on the ground for the purpose of spreading crane weight and reactions to an enlarged bearing area. If the tracks were unfastened and stretched out on the ground, the crane could still operate and run back and forth on the tracks, albeit with less lifting capacity for loss of the ballast weight imparted by the tracks.

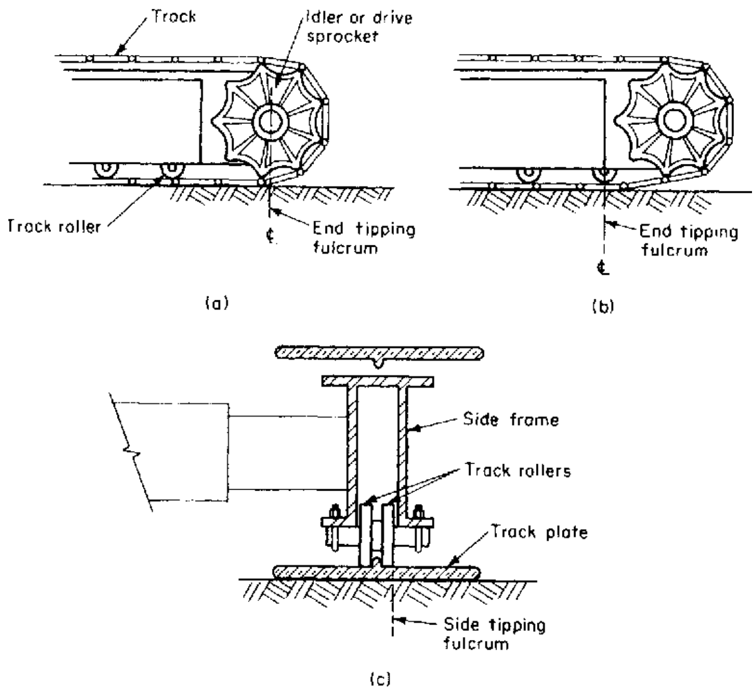


Figure 4.7 Crawler crane tipping lines (a) Apparent over-end tipping fulcrum at sprockets. (b) Actual over-end tipping fulcrum at first track roller. (c) Over-side tipping fulcrum. Any number of configurations of side frame, sprocket, track roller, and track are possible; the ones shown are merely illustrative.

Track rollers, seen in Figures 2.10 and Figures 4.7, ride on the tracks and define the position of the side fulcrum line. The tracks do not alter the location of the tipping fulcrum.

Because the tracks are loosely pinned together and rest on the ground, the track opposite the tipping fulcrum is not fully engaged as a ballast weight in resisting side overturning. The weight of that part draped on the ground cannot be included in the stabilizing moment.

By appearance, the tipping fulcrum at either track end is located vertically below the centerline of the shaft of the *drive sprocket* or the *idler sprocket* (see Figure 4.7*a*). But typically crawler tracks have raised ends as in Figure 4.7(*b*), placing the actual tipping fulcrum further back at the first track roller. A raised end enables the machine to turn and to travel onto ramps and over soft ground more easily. Over either side, the tipping fulcrum is the line formed by the outside of the track rollers, shown in Figure 4.7(*c*).

The stability of a crawler crane can be seriously diminished by improper blocking under the tracks or by uneven ground support. The problem can be subtle and go unnoticed during a casual inspection. In essence, this problem may occur when one or more of the track ends are not firmly supported, in effect shortening the tipping fulcrum. For example, when a few timbers are placed under a track to firm up a soft spot, those timbers may create a new tipping fulcrum short of the normal fulcrum.

Under such circumstances, a crane can tip over at substantially less than rated load. The blame is often mistakenly placed, with soft ground usually attributed as the cause. However, the real cause is that the tracks are not uniformly and firmly supported along their entire length. Another common instance of this inadequate-support situation can occur when the track is partly supported on a hard surface such as concrete and partly on softer ground. The edge of the concrete may then become the effective tipping fulcrum, and insufficient stability may result. A crawler track may be able to accommodate imperfect support conditions near its center, but unless both ends are fully and firmly supported trouble is likely to ensue.

The stability margin can be improved when lifting over the ends by inserting timber blocks against the first track segment that angles up from the ground. Some machines require blocking in order to raise and lower maximum boom lengths unassisted. There can be no harm in operating the crane with ends chocked so. In fact, this procedure helps assure that the tipping fulcrum has not been compromised by the unevenness of the ground support; but the operator must not use it as license to exceed the rated load.

Location of Tipping Fulcrum for Mobile Cranes on Tires

When a crane operates on tires, the location of the tipping fulcrum depends on the physical arrangement of the suspension system or on special provisions employed to create the fulcrum position desired. If an axle is spring mounted, the spring position dictates the location of the fulcrum line. For unsprung axles pivoted to oscillate about the longitudinal centerline of the crane, illustrated in Figure 4.8(*a*), the pivot controls the position of the fulcrum line.

When two axles are mounted on beams parallel to the crane centerline (often called *walking beams*) and each beam is pivoted to the crane frame (the whole assembly is called a *bogie*), the pivot points on the frame control the fulcrum location. See Figure 4.8(*b*).

Some cranes are mounted on axles solidly affixed to the frame and mediated by rubber shock pads. For these machines, the center of the tires or the center of a pair of dual tires marks the location of the tipping line. Short of using outriggers. This arrangement, shown in Figure 4.8(*b*), provides the greatest relative stability and hence the highest rated loads of all the "on-rubber" fulcrum positions discussed.

For machines with sprung or oscillating axles or with bogies, mechanical blocking can be used to increase stability. When the blocking is engaged—typically automatic or executed using a control in the *cab*—the axles become locked to the frame and the tires then

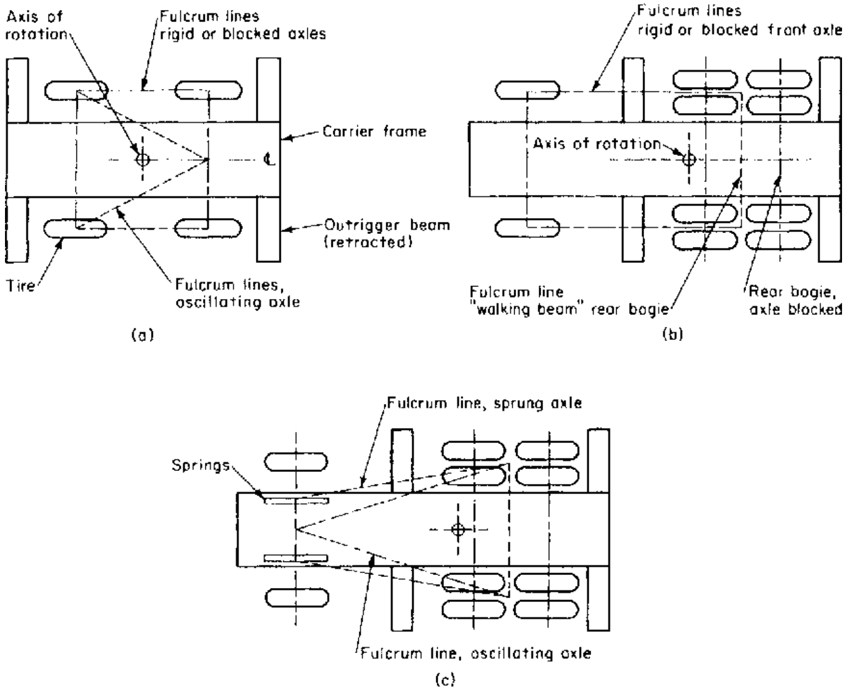


Figure 4.8 Tipping lines for cranes on tires with and without axle blocking. (a) Rough-terrain crane. (b) Truck crane with rigid front axle. (c) Truck crane with oscillating or sprung front axle.

define the fulcrum line. The blocking arrangement is called an axle *lockout.* The fulcrum lines—for axles both blocked and unblocked—are shown in Figure 4.8.

Tires deflect when loaded—a fact that should not be overlooked. Moreover, when the crane is loaded with a moment, deflection will not be uniform and the crane will tilt. This action increases the load radius and also shifts the CG of the crane closer to the fulcrum line. Both effects reduce stability and introduce an element of anxiety and risk to operation on tires.

Finally, stability is not the only limitation for operations on tires. Some ratings may be controlled by the ability of the tires or rims to support load.

Operating Sectors

Looking down at a crane from above, if the longitudinal centerline is marked as 0° when the boom is over the rear, then operation square over the side will be at 90°. For any angle less than 90°, the distance to the CGs of the rotating elements becomes shortened with respect to the side-tipping fulcrum.

The parameters used earlier in calculating the resisting moment were for the undercarriage and superstructure combined. When horizontal angles of operation are introduced, those two components must be segregated, the equation for the machine resisting moment becomes

$$M_r = W_u d_u + W_c d_c + W_f(d_0 - d_f) \tag{4.2}$$

where W_u and d_u refer to the entire superstructure including the boom and hoist ropes and W_c refers to the undercarriage. d_c is the general tipping-fulcrum distance for the undercarriage CG, while d_{cs} specifically refers to the side distance and d_{ce} to the end (rear) distance. When the boom is at a horizontal angle θ to the longitudinal centerline, the equation for resisting moment must reflect this angle; Eq. (4.2) then becomes

$$M_r = W_c d_{cs} + W_u d_u \sin\theta + W_f(d_0 - d_f \sin\theta)$$

To clarify this point, consider a crane without a boom foot mast so that the term in parentheses can be dropped. Equating the overturning moment M_0 with the resisting moment gives

$$M_0 \sin\theta = W_c d_{cs} + W_u d_u \sin\theta$$

$$M_0 = \frac{W_c d_{cs}}{\sin\theta} + W_u d_u \qquad \theta >> 0°$$

For any angle other than 90°, sinθ will be less than unity and the first term on the right becomes amplified. With otherwise constant parameters, as θ changes from 90°, the equality is maintained only if M_0 increases, that is, only if the tipping load increases. This equation shows that the least value for M_0 and the tipping load occurs at $\theta = 90°$.

As θ decreases, a point is reached at which the rear end fulcrum line becomes effective rather than the side. This will occur as the boom passes the intersection of the end and side fulcrum lines. With the boom over the intersection, M_0 achieves its maximum value; that is, the tipping load is at its greatest when the boom is over this corner position.

When stability is contemplated in this way, it becomes quite clear that the tipping load varies with the swing angle θ. But to take advantage of this fact would require a very complex form of rating chart indeed. Furthermore, practical study of crane use and field requirements would show that little is likely to be gained from such a bewildering chart.

Rating charts published by mobile-crane manufacturers impose stability limits in a more simplified format. If one set of ratings is given, the least value for the entire permitted operating arc (horizontal)

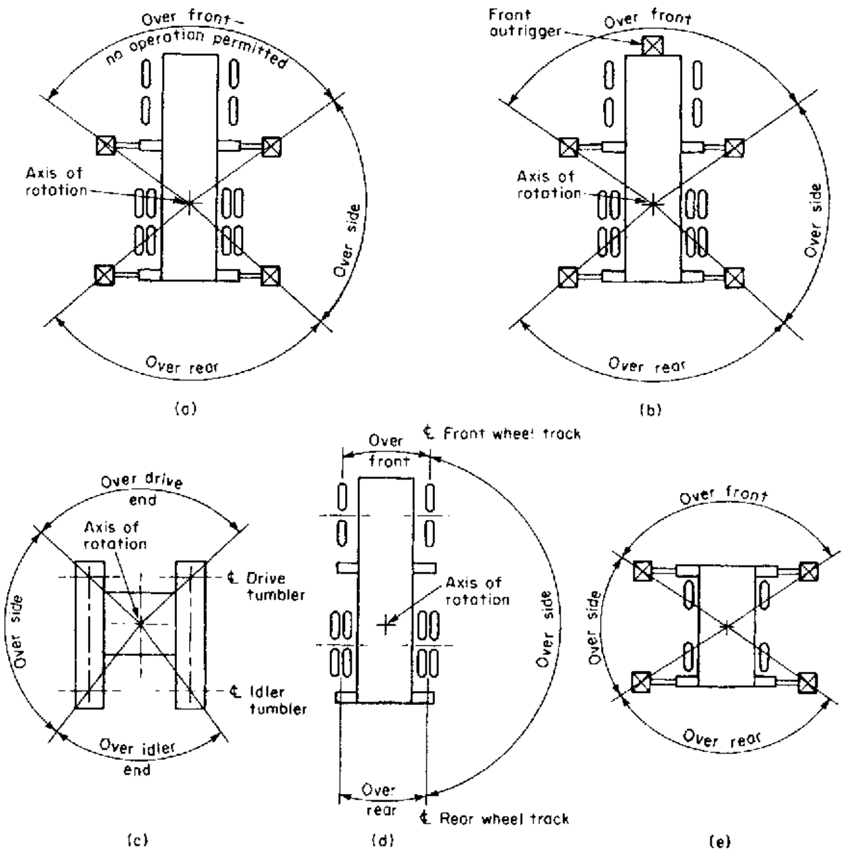


Figure 4.9 Mobile-crane operating sectors for various crane configurations. (a) Truck crane on outriggers, normal configuration. (b) Truck crane on outriggers with front outrigger. (c) Crawler-mounted crane. (d) Truck crane on tires. (e) Rough-terrain crane on outriggers.

is supplied. If two sets of ratings are listed, they will correspond to the swing angles $\theta = 0°$ and $\theta = 90°$. The stability margin at other angles would be somewhat greater.

These considerations together with the preference for simple, readily understood charts have given rise to the definitions of operating arcs shown in Figure 4.9.[2] The sectors indicated are in common use, but a manufacturer may specify any suitable arcs if the crane characteristics justify them.

On a truck crane, the greatest stability margin is obtained when a boom is operated directly over a rear outrigger float. For a crawler

[2]See also SAE J1028, Mobile Crane Working Area Definitions, Society of Automotive Engineers, Warrendale, Pa., 1989.

crane, it is obtained when the boom is over an idler or drive sprocket. This knowledge, intuitively known to most operators and riggers, should not be used as justification to exceed rated loads when the boom is in those positions. Considerations other than stability are contemplated when load ratings are set by the crane designer; exceeding rated loads, aside from the legal implications, can cause damage or failure in one or more components.

Effect of Out-of-Level Operation

When a machine is mounted out of level and operated with the boom over the low side, the resisting moment will be less than the design value because the tilt raises the machine CG and moves it closer to the tipping line; but this effect is small unless the tilt angle is severe.

For a statically positioned crane, the operating radius and boom position are not affected because the location of the load does not change. However, for a crane that picks up a load while level and then swings 90° or so to a low side or end and for a crane traveling with load, variations from level will alter the operating radius and *boom angle*. Therefore, two cases must be examined: the first where the radius remains constant, and the second where the radius increases due to out of level.

For the first case, taking machine CG height as h_m and horizontal distance from the tipping line as d_m (Figure 4.4), the radial distance of the CG from the tipping line r_1 will be

$$r_1 = \left(h_m^2 + d_m^2\right)^{1/2}$$

and the angle γ which the radial line makes with the horizontal is then

$$\gamma = \tan^{-1} \frac{h_m}{d_m}$$

Should the crane be mounted out of level at an angle α to the true horizontal with the boom on the low side, the radial line to the CG will be at angle $\gamma + \alpha$ to the horizontal and the distance to the tipping line will be reduced to

$$\begin{aligned} d'_m &= r_1 \cos(\gamma + \alpha) \\ &= r_1(\cos\gamma\cos\alpha - \sin\gamma\sin\alpha) \end{aligned}$$

But α is a small angle; therefore using small-angle geometry (and angles measured in radians), $\cos\alpha \approx 1$, $\sin\alpha \approx \alpha$, and

$$\begin{aligned} d'_m &= r_1 \cos\gamma - \alpha r_1 \sin\gamma \\ &= d_m - \alpha h_m \end{aligned}$$

For the simple case of a static out-of-level crane without a boom foot mast, the tipping load of Eq. (4.1) can be restated as

$$W = \frac{W_m(d_m - \alpha h_m) - W_b d_b - W_r d}{d} \tag{4.3a}$$

or in other words, the reduction in tipping load is brought about solely by the term containing the out-of level angle α, which diminishes the distance to the tipping line. In percentage terms, the loss in stability is then

$$L_o = \frac{W_m \alpha h_m}{Wd} \times 100$$

Earlier a large crane was used to illustrate what can happen when outrigger beams are not fully extended. Taking the data for this same crane (see the subsection Extension of Outriggers earlier in this section) and adding rope weight $W_r = 500$ lb (227 kg) and CG height $h_m = 6.1$ ft (1.86 m), the tipping load for a level crane can be found. For $\alpha = 0$, Eq. (4.3*a*) will give the tipping load.

$$W = \frac{319{,}000(18.95 - 0 \times 6.1) - 45{,}200 \times 115.5 - 500 \times 250}{250}$$

$$= 2800 \text{ lb (1270 kg)}$$

The usual installation tolerance for levelness is given[3] as 1% maximum out of level; that is, the crane can be mounted to a slope not exceeding 1:100, which is an angle of 0.01 radians. The loss of capacity is then

$$L_o = \frac{319{,}000 \times 0.01 \times 6.1}{2800 \times 250} \times 100 = 2.8\%$$

For a static truck crane, the out-of-level slope causing the crane to be at the brink of tipping while lifting rated load (85% of the tipping load) is

$$15\% = \frac{319{,}000 \times \alpha \times 6.1}{2800 \times 250} \times 100$$

$$\alpha = 0.054 \text{ rad} = 5.4\% = 3.1°$$

which is equivalent to a difference of 1.3 ft (395 mm) across the 24 ft (7.3 m) outrigger spread, an extreme condition indeed!

[3]Clause 5-3.4.6 of ASME B30.5, *Safety Code for Mobile and Locomotive Cranes*, American Society of Mechanical Engineers, New York, 2007.

For the second case, where the radius increases, the radial distance from the tipping line to the boom tip r_2 will be

$$r_2 = (h_t^2 + d^2)^{1/2}$$

and the angle β which the radial line makes with the horizontal is then

$$\beta = \tan^{-1}\left(\frac{h_t}{d}\right)$$

With the crane mounted out of level at angle α to the true horizontal with the boom on the low side, the radial line to the boom tip will be at an angle $\beta - \alpha$ to the horizontal and the distance to the tipping line will increase to

$$\begin{aligned} d' &= r_2 \cos(\beta - \alpha) \\ &= r_2(\cos\beta \ \cos\alpha + \sin\beta \ \sin\alpha) \\ &= r_2(\cos\beta + \alpha \sin\beta) \\ &= d + \alpha h_t \end{aligned}$$

The tipping equation becomes

$$W = \frac{W_m(d_m - \alpha h_m) - W_b(d_b + \alpha h_b)}{d + \alpha h_t} - W_r \tag{4.3b}$$

Using the same example as for the first case, with $\alpha = 0.01$, $h_t = 191.7$ ft (58.4 m), and $h_b = 99.9$ ft (30.4 m), the loss in capacity is

$$\begin{aligned} \text{Loss } &= 2800 - \{[319{,}000(18.95 - 0.01 \times 6.1) \\ &\quad -45{,}200(115.5 + 0.01 \times 99.9)]/(250 + 0.01 \times 191.7)\} - 500 \\ &= 284 \text{ lb } (129 \text{ kg}) \end{aligned}$$

which represents a 10% loss in capacity, much greater than the 2.8% loss of the first case.

Such calculations unequivocally demonstrate the out-of-level sensitivity of the second case, where the radius increases due to the out-of-level condition. They also give insight into the loss of capacity that can be expected as a result of frame twist and outrigger beam deflection, carrier characteristics that can also shift the CG toward the tipping line while causing the radius to increase in nonstatic situations.

Furthermore, cranes poorly supported on soft ground will go out of level over time. Even for well-supported machines, unless the levelness of the crane is monitored and corrected from time to time, the crane may go out of level and lose capacity.

On most mobile cranes the CG is low, and the loss in capacity is small for a static crane kept level within manufacturer's specifications, as has been demonstrated. But there are crane attachments that raise CG height, such as tower attachments for latticed boom cranes and luffing jibs for telescopic cranes. Those machines are more sensitive to out-of-level conditions, and the standard 1% out-of-level tolerance may be insufficiently rigorous.

Crane stability can be examined from another perspective, a view that considers the work required to rotate the crane to the point of tipping rather than only static balance.[4] Using the parameters given in Figure 4.10, the point O is the tipping fulcrum of that generalized abstract model of a crane, and the crane is loaded by lifted load W together with horizontal static forces not shown in the model. The crane is supported on a surface at angle α to the horizontal; it is out of level. A moment function $F(\alpha)$ is induced by all forces acting on the crane and includes out-of-level effects. The coordinates d_m, h_m, h_t,

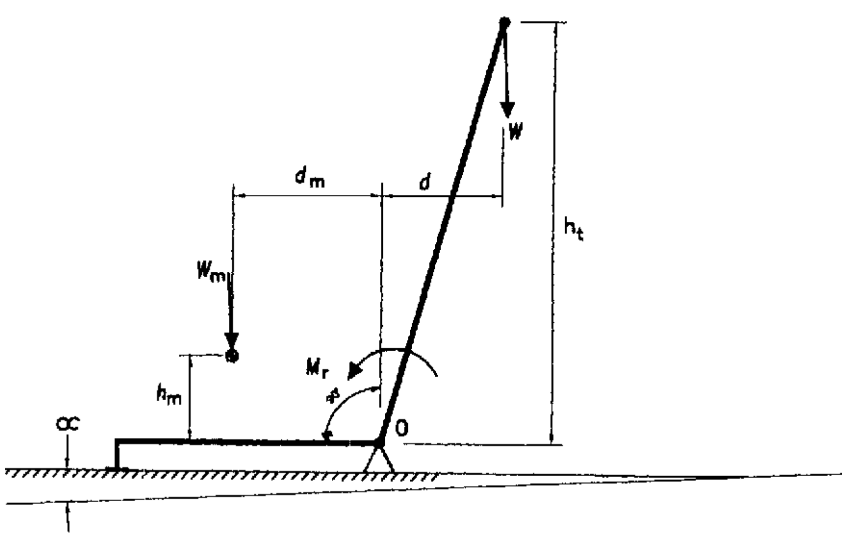


FIGURE 4.10 A generalized abstract model of a freestanding crane. The crane is not precisely level and has load W suspended from the boom tip. M_r is the static moment resisting overturning, including the effect of the out-of-level base, and W_m is the weight of the entire crane including the boom.

[4]This concept of stability was presented in earlier editions of this work, but has been further developed by Prof. Anatoly A. Zaretsky of Moscow, Russia.

and d in Figure 4.10 are determined for $\alpha = 0$; they are properties of the crane and remain constant when the crane tilts. As in the derivation of Eq. (4.3), take the radial distance from O to the point of action of W_m as r_1 at angle γ to the horizontal; likewise, the radial distance to the boom tip take as r_2 at angle β to the horizontal. Note, however, that W_m is now taken as the weight of the entire crane including the boom. Then the deadweight moment $M_D = W_m r_1 \cos\gamma$, the lifted load moment $M_W = Wr_2 \cos\beta$, and M_{hor} will be taken as the moment of horizontal static forces about O including steady-state wind, centrifugal force, and so on.

The moment on the crane now becomes $(\alpha = 0)$

$$F(0) = M_D - M_W - M_{\text{hor}} = M_{\text{net}}$$

but, as the crane is rotated clockwise through the out-of-level angle $\alpha > 0$,

$$\begin{aligned}F(\alpha) &= W_m r_1 \cos(\gamma + \alpha) - Wr_2 \cos(\beta - \alpha) - M_{\text{hor}} \\ &= W_m r_1 \ (\cos\gamma\cos\alpha - \sin\gamma\sin\alpha) \\ &\quad - Wr_2(\cos\beta\cos\alpha \ + \ \sin\beta\sin\alpha) - M_{\text{hor}}\end{aligned}$$

Reintroducing the coordinates given in Figure 4.10 by using their equivalents $d_m = r_1 \cos\gamma$, $h_m = r_1 \sin\gamma$, $d = r_2 \cos\beta$, and $h_t = r_2 \sin\beta$; introducing a new term, $S_h = W_m h_m + Wh_t$; restating the moment terms $M_D = W_m d_m$, $M_W = Wd$; and after simplifying, the equation for the moment becomes

$$F(\alpha) = M_D - M_W - M_{\text{hor}}/\cos\alpha - S_h \tan\alpha$$

But α is a small angle; therefore, using small-angle geometry (and the angle measured in radians), $\cos\alpha \approx 1$, $\tan\alpha \approx \alpha$, and

$$F(\alpha) = M_D - M_W - M_{\text{hor}} - S_h\alpha = M_{\text{net}} - S_h\alpha \tag{4.4}$$

Two stability cases must be considered. First, the situation where the radius increases due to out-of-level effects; tilting of the entire crane makes Eq. (4.4) applicable, and

$$M_r = M_{\text{net}} - S_h\alpha \tag{4.5}$$

where M_r is the static moment resisting overturning including the effect of an out-of-level base.

Second, a static crane where the radius is not changed by the out-of-level condition; distance d of Figure 4.10 holds constant because neither the load nor boom positions are affected by the out-of level angle α. Therefore,

$$M_r = M_D - M_W - M_{\text{hor}} - W^*h^*\alpha \qquad (4.6)$$

where W^* is crane weight less boom weight, and h^* is the height of the point of action of W^*.

$S_h = W_m h_m + W h_t$ is a new parameter for use in stability calculations, a parameter that recognizes the heights of the crane CG and lifted load as playing a direct role in evaluating the stability of a crane when out-of-level conditions cause a change in radius. In like manner, the parameter W^*h^* is used for a crane not subject to radius change.

Using the same data and crane examined in the subsection Extension of Outriggers in Sec. 4.2, the data will now take another form.

$$W_m = 364{,}700 \text{ lb } (165.4 \text{ t}) \qquad d_m = 1.92 \text{ ft } (585 \text{ mm})$$

$$h_m = 17.98 \text{ ft } (5.48 \text{ m})$$

$$W^* = 319{,}000 \text{ lb } (144.7 \text{ t}) \qquad h^* = 6.1 \text{ ft } (1.86 \text{ m})$$

$$d = 250 \text{ ft } (76.2) \qquad h_t = 191.7 \text{ ft } (58.43 \text{ m})$$

For a dead-level crane $(\alpha = 0)$ and $M_{\text{hor}} = 0$, Eq. (4.4) gives

$$M_{\text{net}} = 364{,}700 \times 1.92 - 250W$$

At the limit of stability; $M_{\text{net}} = 0$, therefore the tipping load as before is

$$W = 2800 \text{ lb } (1270 \text{ kg})$$

For a crane that is 1% out of level $(\alpha = 0.01)$, using Eq. (4.6), that is, for the case where radius does not change,

$$M_r = 0 = 364{,}700 \times 1.92 - 250W - 319{,}000 \times 6.1 \times 0.01$$

$$W = 2723 \text{ lb } (1235 \text{ kg})$$

for a loss of 2.8% of capacity, the same as calculated before.

Solving for the out-of-level angle that would cause loss of stability while lifting a rated load (85% of the tipping load), we get, as previously,

$$M_r = 0 = 364{,}700 \times 1.92 - 250 \times 2800 \times 0.85 \times 319{,}000 \times 6.1\alpha$$

$$\alpha = 0.054 \text{ rad} = 5.4\% = 3.1°$$

Considering a crane where conditions induce a change in radius due to out of level, from Eq. (4.5) for 1% out of level, we get

$$S_h = 364{,}700 \times 17.98 + 191.7W$$

$$0 = 364{,}700 \times 1.92 - 250W - (364{,}700 \times 17.98 + 191.7W)\ 0.01$$

$$W = 2519 \text{ lb } (1143 \text{ kg})$$

for a loss of 10% of capacity, as before.

An out-of-level crane operating at constant radius does not necessarily lose much stability, but there are other detrimental effects created by this condition. The leaning crane first swings "uphill" then "downhill," with the swing drive initially struggling to overcome added torsional resistance and then braking to resist freewheeling downhill. With the superstructure stopped in a cross-slope position, the operator may struggle to keep it in place without use of the swing brake because the imbalance introduced by the out-of-level condition may overcome frictional resistance in the swing bearing. The operator may have difficulty holding the load in position for precise load placement—a potentially dangerous circumstance for workers handling the load on the end of the hook. The effect is similar to working against a crosswind. A corollary effect is that the boom and jib are exposed to side loading. Jibs in general, offset and luffing jibs in particular, do not react well to side loading.

A crane traveling with load traverses an out-of-level travel path on a down slope will tilt, and the load radius will increase. Both the load and the boom weight will move farther from the tipping line. Thus the tipping load will decrease more severely than in the static case. More capacity will be lost. If not controlled, the load will swing like a pendulum with further loss of stability and potentially catastrophic results.

The actual loss of capacity caused by the crane being out of level will vary with crane model, configuration, and radius. A representative comparison between figures for the particular crane and configuration in the earlier calculations for static (no radius change) and nonstatic (radius change) conditions follows:

Percent Loss of Capacity (for a particular crane)		
Percent Out of Level	**Static Crane**	**Nonstatic Crane**
0.5	1.4	5.0
1	2.8	10.0
2	5.5	19.9
5	13.9	48.8
10	27.8	94.1

The figures are for one example and will vary markedly for different cranes, configurations, and operating radii. Moreover, out-of-level operation can induce side loading of the boom and jib. In extreme cases, those added side-load stresses may lead to failure.

Deflection

Immediately on lifting the rated load, a mobile crane installed to true level on adequate supports set precisely to maximum permitted radius will exceed that permitted radius and the lift will go over the rated capacity! First, the outrigger beams will deflect and the carrier frame will twist causing the radius to increase. Be it a crawler crane, carbody and side frames will also deflect and twist, albeit less than a truck crane base. Second, the boom pendant and boom hoist lines' ropes will stretch under load, further increasing the radius; if the boom is telescopic, the cantilever deflection will be much larger than a pendant-suspended boom of the same length. All of this deflecting and stretching will cause the radius to change between the unloaded and the loaded state. Operating radii given in the load rating charts are *loaded radii*; to compensate, a lift at full rated load should be made at a shorter radius. An experienced operator can sometimes accurately predict the radius change between the unloaded and loaded states and allow for it in advance, but generally, overcompensation toward a lesser starting radius is a prudent measure.

Effect of Wind

Wind affects a mobile crane differently depending on whether it bears from the front, rear, or side.

A frontal wind might topple the crane backward, sometimes in concert with other factors such as out-of-levelness of the base. Vulnerability to frontal wind can be exacerbated in several ways.

- Raising the *boom angle* to near-vertical reduces its stabilizing moment at the same time that it increases detrimental frontal wind exposure. If the boom or jib is a lattice, the wind might flip it backward until interrupted by boom stops, preventing a catastrophic boom-over-cab failure that might ensue if the stops were not there to intervene.
- A boom pointing over the side of the carrier is facing the least stable, and thus most vulnerable, direction.
- Adding counterweights on the rear of the superstructure increases backward moment.
- When outriggers or tracks opposite the boom are not fully extended, the resistance to tip backward is reduced.

A rear wind works against the forward stability of a mobile crane. When the wind blows from this direction, it applies force to the boom and to the load that adds to the overturning moment; rear wind has the same effect as adding load at the hook. Reduced outrigger or track extension on the boom side and operation over the side of the carrier can increase susceptibility to forward tipping.

Wind from the side of the boom is a strength issue. Jib extensions to the boom, especially luffing jibs and fixed jibs with large offset angles, are particularly susceptible to structural damage from side winds due to the potential for inducing torsion on the boom. Overturning stability is generally a concern only when outriggers or tracks are not fully extended.

The question of wind and its effect on safe lifting capacity is a difficult subject that falls in the chasm between the intentions of the crane designers and the prerogatives of the crane user.

In the U.S. mobile-crane design practice, wind is not part of the load combination considered for stability-based load ratings. For strength-based ratings, full rated loads are taken together with a side wind of 20 mi/h (8.9 m/s) and a side loading at the boom tip of 2% of the rated load. This load combination has been found to be a good representation of typical operating conditions, but with caveats: It may not account for the wind area of a large load suspended from the hook or for lifting in higher-speed winds. A 30-mi/h (13.4-m/s) wind—a speed often considered being at the upper limit for mobile crane operation—has a wind force that is 225% of the force of a 20-mi/h wind. In an operating environment that is considered to be normal and permitted, forces acting on the crane could be exceeding the expectations of the designer. Clearly design practices with respect to wind do not comport with conditions that inevitably occur in the field.

Manufacturers invariably have something to say about wind. They may simply forbid lifting when the wind exceeds a threshold level, or they may stipulate that the load capacity must be adjusted downward without offering the means to do so. Recently some manufacturers have begun to provide formulas or tables that give the user the means to eliminate guesswork. Some mobile cranes now have built-in anemometers wired into the control systems; it is conceivable that, in time, onboard computers could adjust load ratings for the wind.

Overturning stability is influenced both by wind acting on the load and by wind acting on the crane itself. For most loads lifted in winds up to 20 mi/h (9 m/sec), this force will be negligible. A 20-mi/h wind will exert a force of only 1⅛ psf (54 Pa) on a flat-surfaced load; without an appreciable sail area, the wind force on the load will be small and it is unlikely that the crane will be much influenced.

But for loads with large sail areas and for operation in higher winds, the effect on the crane can be great. It behooves supervisors, riggers, and operators to recognize that crane ratings may need to be reduced. But lacking guidance from the manufacturer, this frontline group may be left to rely on judgment alone. The cloud of uncertainty allows for the unsatisfactory possibility that a safe lift will be canceled out of uncertainty and fear or, worse, that an unsafe lift will go forward out of ignorance.

In Chap. 5, an approximate field method is given that can be used to check both strength and stability under wind conditions. It enables jobsite personnel to determine whether a lift can be made.

Stability-Based Ratings

Most crane codes and standards set stability-based ratings as a percentage of the tipping load. Just what percentage to use is subjective, and judgments vary on this point. In the United States and Canada, 85% is commonly used for truck cranes and 75% for crawler cranes, whereas elsewhere ratings from 66.7% to 75% of tipping are prevalent.

When load ratings are established this way, a crane carrying a rated load can, under some circumstances, have a slender margin against tipping. That is especially so when the margin is measured against the *overturning moment*, a quantity that is composed of contributions from both the boom and the load.

A long latticed boom or a fully extended telescopic boom at a large radius can induce a moment approaching tipping without ever picking a load. Indeed many telescopic boom range diagrams include blanked-out areas, radii at which it is not permitted to reach with an unloaded boom because of stability concerns. When a rated load which is either 75% or 85% of the tipping load is then added at the hook, the margin measured against the tipping moment can be razor thin.

An alternative approach, is to use moments instead of loads as the basis for stability ratings. The logic is that the whole machine tips; therefore the ratings should be based on the stability of the entire crane. On the other hand, the logic of load-based ratings is that ratings are established as a protection against lifting excessive loads; therefore it is the load that matters and should be the basis for ratings.

Most crane engineering follows the load approach; nevertheless all must face the fact that for long booms at long radii the tipping load may be considerably less than 1% of machine weight. Under such conditions, the crane is very sensitive to both wind and dynamic effects and can easily be overturned. But it is also a fact that cranes behave very well at short radii with present rated loads.

Wrestling with this problem and the conflicting judgments and theories on the subject, a committee of the International Organization for Standardization (ISO) has agreed on a simple and practical rating system that almost marries the two approaches. It is expressed by the equation

$$\text{Rated load} = \frac{T - 0.1f}{1.25}$$

where T is the tipping load and f is the weight of the boom referred to the boom tip or boom tip weight.

Initially it was proposed that 10% of boom tip weight be deducted from the tipping load to account for the dynamic effect of braking after booming out. But although later tests showed this to be a variable rarely exceeding 5%, the probability of braking to a sharp stop with the rated load exactly at rated radius is rather remote.

The authors' view and many other engineers have come to feel the same way is that the 10% figure, however arbitrary, makes a good practical adjustment to the rating curve. It has little or no effect at short radii, where adjustment is not needed, but as the radius increases, it becomes more prominent. At maximum radius, rated load may be reduced to 2/3 of the tipping load for most cranes and less for very long booms. But the adjustment is a function of boom weight and therefore more pronounced in heavy booms, such as telescopic types, than in lightweight tubular booms. Thus, after a fashion, it is a self-adjusting rating modifier that strikes a good balance between safety and machine utilization.

There is another side to this coin, however. Looking over the data used for stability calculations earlier in this section, the ISO formula would reduce the rated load to just over 500 lb (227 kg), or about 18% of the tipping load at that very long radius. As a practical matter, this rating method will eliminate some lifting at long radii.

The ISO rating formula has been incorporated into some national codes.

4.3 Tower Cranes and Self-Erecting Cranes

Stability characteristics differ considerably between a top-slewing and a bottom-slewing crane. With counterweights and hoist machinery atop the mast, a machine of the first category has a form that is readily identified as a tower crane. A modern machine of the second category is likely to be a self-erecting crane.

A top-slewing crane has a large mass of equipment mounted on a relatively slender mast. Its high CG presents an inherent instability that is resisted by anchoring or ballasting the base, unless the mast is braced or guyed. In the U.S. practice, the rated load of a freestanding crane cannot exceed 63% of the tipping load, but additional stability checks are specified as well[5]. Nearly the same margin is specified in the ISO as well as the widely used FEM *consensus standards*; [6]

[5]Refer to Sections 3-1.2 and 3-1.3 of the American Society of Mechanical Engineers Standard B30.3-2004, *Construction Tower Cranes.*

[6]The figures are from ISO 12485, Stability Requirements for Tower Cranes, International Organization for Standardization, Geneva, Switzerland, 1997, and are matched in Table T.9.15a of the FEM standard (Fédération Européene de la Manutention, "Rules for the Design of Hoisting Appliances," 3d ed., Paris, 1998).

however, the additional stability checks (described below) differ from the U.S. standard.

A characteristic of a top-slewing tower crane, affecting both strength and stability, is the flexibility of the mast. It deflects under the action of unbalanced loads from the deadweight, lifted load, wind, and dynamic forces. The substantially elevated mass acts on the primary elastic deflections causing increased deflection and an amplification of the bending moments. The mast is a beam-column.

The structural behavior is not quite the same for a bottom-slewing machine. Its ballast is at the base with little mass at the top; there may be pendant lines run parallel to the mast relieving the mast of most of the applied moment while adding axial compression. But this mast, too, is a beam-column.

For a tower crane in service, wind can be taken from either behind the operator or broadside on the jib, with the latter case often the more severe. Under the influence of side wind, the wind and primary load produce moments acting at right angles to each other. The overturning moment is then the resultant of these moments. That resultant can have any orientation, with the shortest fulcrum at the base usually being the least-favorable direction for overturning stability.

Stability for a tall tower crane is usually governed by the out-of-service condition under the influence of storm winds. A weathervaning jib exposes a smaller area to the wind and produces wind moments that are opposite in direction to the dead-load moment, which is dominated by the counterweights. The stability of a tower crane on a short mast or a self-erector crane is usually governed by in-service loading. In the latter case, the crane can be folded up in advance of a predicted severe storm.

Stability ratings of tower cranes are determined by calculation, as it is too dangerous to verify tipping loads by test. According to the FEM standard, three primary stability conditions should be verified.

1. *Basic stability*. The crane is under static loading in calm air with the rated load not to exceed 62.5% of the static tipping load.
2. *Dynamic stability*. The crane is in service; wind and dynamic loads are applied as appropriate. Rated load is taken as not more than 74% of the dynamic tipping load.
3. *Stability under extreme loading*. The crane is out of service and subjected to 120% of storm wind effects.

Under the first two conditions, the tipping load—the load that brings the crane to the point of incipient overturning—is found. The lesser of the calculated loads is then used as the load rating. For the third case, the crane must not overturn under the stipulated loads.

An additional criterion that must be satisfied concerns backward stability, that is, stability against overturning toward the counterweights. In mobile-crane practice, the amount of counterweight is limited by controlling the location of the CG of the unloaded crane with respect to the backward-tipping fulcrum. The governing case occurs with the shortest boom is at its highest boom angle. In tower-crane practice, the high CG location engenders a greater sensitivity to stability. In addition to static backward stability, the crane must remain stable under dynamic conditions such as the spring-back effect that accompanies sudden release of load. The U.S. standard has a criterion that accounts for this effect.[7]

Elastic Deflections of Tower Cranes

Mast elastic deflections are caused by load eccentricity and by wind and are amplified by beam-column action, also called the P-Δ effect. If M_c is the net moment about the crane centerline (Figure 4.11) in the absence of wind, the displacement of the mast top from the centerline δ_c is given by

$$\delta_c = \frac{M_c}{Q}\frac{1-\cos kh}{\cos kh} \tag{4.7}$$

where Q is the weight of the slewing portion of the crane (at the mast top) plus the load and 1/3 of the mast weight,

$$k = \left(\frac{Q}{EI}\right)^{1/2}$$

E is the modulus of elasticity of the mast materials, and I is the moment of inertia of the mast cross section. The trigonometric arguments are in radians, and Eq. (4.7) intrinsically includes beam-column effects.

With wind introduced, taking W_w as the wind force on the concentration of exposure area above the slewing circle and w as the wind force per unit of length on the mast, we get

$$\delta_w = \frac{1}{Qk}\left[w_w(\tan kh - kh) + wh\left(\tan kh - \frac{kh}{2}\right) - \frac{w}{k}\frac{1-\cos kh}{\cos kh}\right] \tag{4.8}$$

Taking moments about the crane base gives

$$M_c' = M_c + Q\delta_c = \frac{M_c}{\cos kh} \tag{4.9}$$

[7]Refer to Section 3-1.3 of the American Society of Mechanical Engineers Standard B30.3-2004, *Construction Tower Cranes.*

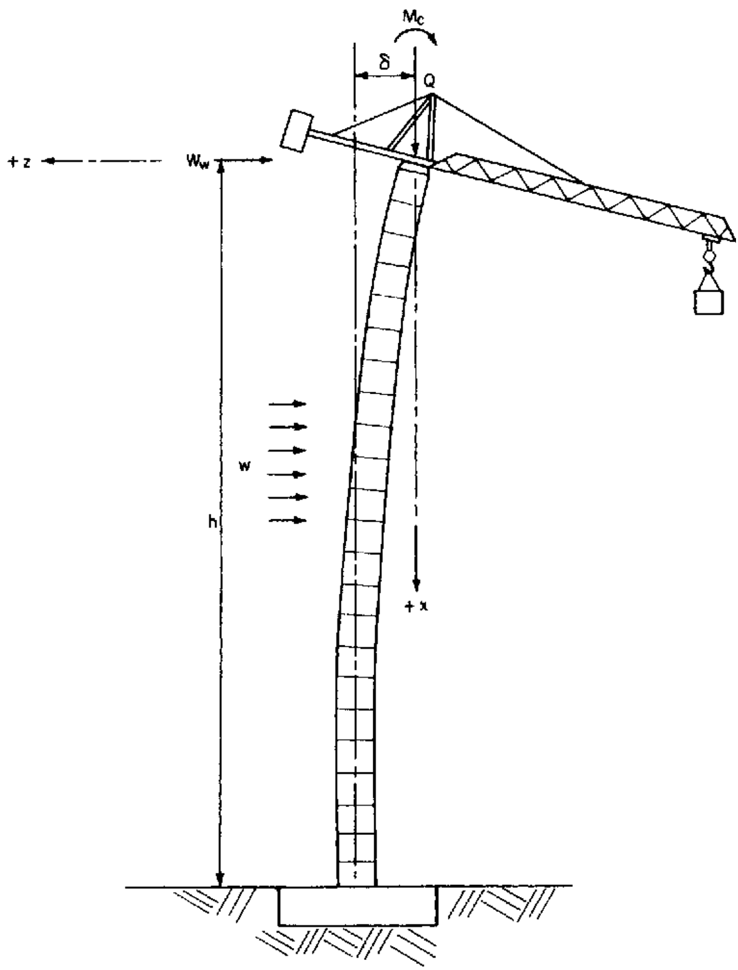


FIGURE 4.11 Model of tower crane annotated for calculation of elastic deflection.

and for the wind moment M_w (noting that $M_w = W_w h + wh^2/2$),

$$M'_w = M_w + Q\delta_w = (W_w + wh)\frac{\tan kh}{k} - \frac{w}{k^2}\frac{1-\cos kh}{\cos kh} \tag{4.10}$$

The augmented moments of Eqs. (4.9) and (4.10) are used in both stability and strength calculations. For total deflection, δ_c and δ_w can be added as vectors. The moments can be similarly added.

Example 4.1

1. A tower crane has an unbalance moment of 1500 kip-ft (2034 kN-m) about the axis of rotation while operating in calm air. The weight of that portion of the crane above the slewing ring plus the load and one-third of the mast weight is 100 kips (45.36 t), and mast height is 150 ft (45.72 m). For $E = 30{,}000$ kips/in^2 (200 GN/m^2) and $I = 45{,}000$ in^4 (1,873,000 cm^4), what will be the mast top deflection and the base moment?

Solution

From Eq. (4.7),

$$k = \left(\frac{100 \text{ kips}}{30{,}000 \text{ kips/in}^2)(45{,}000 \text{ in}^4)}\right)^{1/2}$$

$$= 2.722 \times 10^{-4} \text{ rad/in } (1.072 \times 10^{-2} \text{ rad/m})$$

$$\cos kh = \cos[2.722 \times 10^{-4})(150 \text{ ft})(12 \text{ in/ft})]$$

$$= 0.8824$$

$$\delta_c = \frac{1500 \text{ kip}\cdot\text{ft}}{100 \text{ kips}} \frac{1 - 0.8824}{0.8824} = 2.00 \text{ ft (610 mm)}$$

From Eq. (4.9),

$$M'_c = \frac{1500}{0.8824} = 1699.9 \text{ kip}\cdot\text{ft (2305 kN}\cdot\text{m)}$$

2. If the wind force on the side of the jib is 1200 lb (5.34 kN) and the unit wind force on the mast is 7.5 lb/ft (109.5 N/m), what will be the wind deflection at the mast top and the base moment due to wind?

Solution

From Eq. (4.8),

$$\tan kh = \tan[(2.722 \times 10^{-4})(150 \text{ ft })(12 \text{ in/ft})]$$

$$= 0.5333$$

$$\delta_w = \Big\{(1200 \text{ lb})[0.5333 - (2.722 \times 10^{-4})(150)(12)] + (7.5 \text{ lb/ft})(150 \text{ ft})\left[0.5333 - (2.722 \times 10^{-4})(150)\frac{12}{2}\right] - \frac{7.5(1 - 0.8824)}{(12)(2.722 \times 10^{-4})(0.8824)}\Big\} \Big/ (100{,}000 \text{ lb})(2.722 \times 10^{-4})$$

$$= 2.59 \text{ in (66 mm)} = 0.22 \text{ ft}$$

From Eq. (4.10),

$$M'_w = [1200 \text{ lb} \times 150 \text{ ft} + (7.5 \text{ ft/lb})(150 \text{ ft})^2/2 + 100{,}000 \text{ lb} \times 0.22 \text{ ft}]/1000 \text{ lb/kip}$$

$$= 286.4 \text{ kip}\cdot\text{ft (388.3 kN}\cdot\text{m)}$$

3. How much moment amplification has been introduced by the beam-column effect?

Solution
For the load moment

$$\left(\frac{1699.9}{1500}-1\right)(100)=13.3\%$$

For the wind moment, the unamplified wind moment is

$$W_w h+\frac{wh^2}{2}=\frac{1200(150)+7.5(150^2/2)}{1000\ \text{kips/ft}}$$
$$=264.4\ \text{kip}\cdot\text{ft}\ (358.4\ \text{kN}\cdot\text{m})$$

Then

$$\left(\frac{286.4}{264.4}-1\right)(100)=8.3\%$$

4. The moments for load and wind are at right angles to each other. What are the resultant moment and deflection?

Solution

$$M=(1699.9^2+285.9^2)^{1/2}=1723.8\ \text{kip}\cdot\text{ft}\ (2338\ \text{kN}\cdot\text{m})$$
$$\delta=(2.00^2+0.22^2)^{1/2}=2.01\ \text{ft}\ (613\ \text{mm})$$

To derive Eqs. (4.7) and (4.8), it is necessary to make use of the concepts of elastic stability.[8] For a beam-column, the basic expression is the differential equation

$$EI\frac{d^2z}{dx^2}=-M_x$$

for coordinate axes, as shown in Figure 4.11. When a cantilevered beam-column is loaded with only an axial force and a moment at its upper end,

$$M_x=M_c+Q_z$$
$$EIz''=-M_c-Qz \qquad (4.11a)$$
$$z''+\frac{Q}{EI}z=-\frac{M_c}{EI}$$

where the prime notation refers to differentiation with respect to x. Substituting $k^2=Q/EI$, the general solution to Eq. 4.11(a) becomes

$$z=A\cos kx+B\sin kx-\frac{M_c}{Q} \qquad (4.11b)$$

[8]See S. P. Timoshenko and J. M. Gere, “Theory of Elastic Stability,” McGraw-Hill, New York, 1961.

but the constants of integration A and B are unknown. They can be evaluated by inserting boundary conditions that define the requirements this beam-column must meet. The first, $z(0)=0$, states that deflection must be zero at the top (with respect to the chosen origin for the coordinate axes). Substituting this into Eq. (4.11b), we have

$$0 = A\cos 0 + B\sin 0 - \frac{M_c}{Q} \qquad A = \frac{M_c}{Q}$$

Second, the slope at the fixed end must be 0, or $z'(h)=0$.

$$z' = -\frac{M_c}{Q}k\sin kx + Bk\cos kx$$

$$0 = -\frac{M_c}{Q}\sin kh + B\cos kh \qquad B = \frac{M_c}{Q}\tan kh$$

With the constants evaluated, Eq. (4.11b), for elastic deflection at any point in the beam-column, simplifies to

$$z = \frac{M_c}{Q}(\cos kx + \tan kh \sin kx - 1)$$

Maximum displacement will occur at the base, or fixed end; $z(h)=\delta_c$

$$\delta_c = \frac{M_c}{Q}\left(\cos kh + \frac{\sin^2 kh}{\cos kh} - 1\right)$$

$$= \frac{M_c}{Q}\frac{1-\cos kh}{\cos kh}$$

This is Eq. (4.7).

Under wind loading only, but with the axial force also acting,

$$M_x = W_w x + \frac{wx^2}{2} + Qz$$

$$z'' + k^2 z = -\frac{W_w x + wx^2/2}{EI}$$

$$z = C\cos kx + D\sin kx - \frac{W_w x + wx^2/2}{Q} + \frac{w}{Qk^2}$$

Inserting the boundary condition $z(0)=0$ gives

$$0 = C + \frac{w}{Qk^2} \qquad C = -\frac{w}{Qk^2}$$

From the slope condition $z'(h) = 0$,

$$z' = -Ck\sin kx + Dk\cos kx - \frac{W_w + wx}{Q}$$

$$0 = \frac{w}{Qk}\sin kh + Dk\cos kh - \frac{W_w + wh}{Q}$$

$$D = \frac{W_w + wh}{Qk\cos kh} - \frac{w}{Qk^2}\tan kh$$

The maximum wind-induced deflection δ_w, measured at the fixed end, is then given by $z(h) = \delta_w$,

$$\delta_w = -\frac{w}{Qk^2}\cos kh + \frac{W_w + wh}{Qk}\tan kh - \frac{w}{Qk^2}\tan kh \sin kh$$

$$-\frac{W_w h + wh^2/2}{Q} + \frac{w}{Qk^2}$$

$$= \frac{1}{Qk}\left[W_w(\tan kh - kh) + wh\left(\tan kh - \frac{kh}{2}\right) - \frac{w}{k}\frac{1-\cos kh}{\cos kh}\right]$$

as given in Eq. (4.8). These deflection expressions intrinsically include the secondary effects of the axial load acting on the primary displacement, the P-Δ effect.

Static Mounting

The majority of tower cranes are *static base* mounting relying either on a footing mass or a ballasted *undercarriage* for support. The wider the base, the less the mass needed for ballast and the smaller the support reactions or ground bearing pressure. The benefit of spreading out the base should be weighed against considerations of space. An undercarriage, however, is a stock manufactured assembly that has a fixed spread.

When a single concrete mass is used as a crane foundation, its edges are taken as the tipping lines. For this assumption to be valid, the underlying subgrade must have adequate capacity to support the maximum calculated bearing pressure. It is insufficient, thus, to consider the problem of stability without contemplating the bearing pressure under the crane foundation, too[9]. In designing a pile foundation, the tipping line is routinely taken through the centers of the edge piles.

[9]At the time of writing, this the authors are unaware of any developed Limit State Design procedure for tower crane footings. Such a procedure should compare the bearing pressure resulting from a factored set of tower crane loads to several possible limit states including tipping and soil failure.

Most crane footings are rectangular and least stable toward the edge closest to the crane. The situation is more complicated when the footing is asymmetrical or when the potential overturning moment is not uniform in all directions. Oftentimes, the loading on the foundation must be evaluated for both working conditions and for storm exposure. This is treated more fully in Chap. 6 which covers tower cranes.

Foundation settlement is of minor importance insofar as stability is concerned, but differential settlement can cause plumbness to go out of tolerance and moment on the mast and base to increase. In an extreme case, difficulty in swinging and a diminution in load control may ensue, and the crane may be prevented from weathervaning properly. For a climbing tower crane that will later be braced to the building, the loss of plumbness may spoil the alignment of the crane to the intended attachment points on the building. Differential settlement may be caused by irregular soil conditions or by a prevailing lopsidedness in soil bearing pressure.

Cranes are usually more tolerant of footing settlement than buildings; they may indeed experience greater settlement because of their dynamic back-and-forth loading. Joining a crane footing to building footings should therefore be done with some caution due to the potential undesirable effects on the building structure.

Traveling Bases

The ballast required for a traveling tower crane is mounted right on the travel base itself, generally in the form of blocks of precast concrete. In one direction, the rail centers define the tipping line. In the other direction, the wheel-shaft centers mark the tipping fulcrum for a crane with four wheels. When four bogies with eight or more wheels are used, the bogie pivots control tipping. Bogies equalize the load among the wheels in each set.

Rail lines must be constructed and maintained in such manner that the crane will operate under conditions assumed in the design. A dip in the track will reduce stability by causing the crane to go out of plumb. The dip can be introduced by initial inaccuracy in placing the track, by a short stretch of poor supporting soil, or by local settlement over a period of time. If supports are placed too far apart, the track may deflect between supports, causing the crane to lean. Mechanical rail splices introduce the possibility of rail-top misalignment at the joint; this can induce a horizontal dynamic force when hit by a wheel.

A travel base must be constructed to offer a high degree of resistance to rotation; knee braces are often installed for this purpose. A base that lacks sufficient rigidity will permit the mast to lean with a subsequent loss of stability. Should the mast bottom rotate through an angle φ, the top of the mast will be displaced φh if φ is taken in radians. This displacement will introduce a new moment $Q\varphi h$, which will cause additional displacement. The final displacement δ_b due to base rotation is then

$$\delta_b = \varphi \frac{\tan kh}{k} \tag{4.11}$$

and the added moment is

$$M_b' = \frac{Q\varphi}{k} \tan kh \tag{4.12}$$

where the rotation angle φ is given by

$$\varphi = \frac{\Sigma M}{2a^2 k_0} \tag{4.13}$$

where a is the distance between the tipping lines in the direction of the moment and k_0 is the rotational spring rate of the travel base. Only coplanar moments are included in the summation.

One stability condition peculiar to traveling cranes concerns the application of travel brakes. Braking action decelerates the crane, inducing inertial forces at the crane masses so that overturning moments result. Travel-brake stopping rates must therefore be controlled to prevent the crane from overturning.

Buffers, or stops, are placed at the ends of sections of travel trackage, but the crane is not supposed to strike them. If the crane should suffer the high deceleration accompanying a buffer strike, it would surely overturn. Instead, automatic brake trippers are placed at a distance from the buffers. The buffers serve as a backup only in the event that the brakes fail to stop the crane fully in the allotted distance.

4.4 Barge- and Ship-Mounted Cranes

By nature, a floating crane continually goes out of level. Moment induced by boom, load, and counterweights causes the crane to be imbalanced except in passing; a net moment is imposed on the vessel in addition to crane and load weight. Since most vessels are much longer than they are wide, only operations over the side usually need to be evaluated for stability. Some ship-mounted revolver cranes have higher lifting capacities over the rear than over the sides.

Consider a crane positioned dead center on a barge. With the boom pointed along its longitudinal axis, the barge will be dead level side-to-side because there will be no moment applied, provided other equipment and materials on the barge are also in balance. When a mobile crane swings over the side, it imposes a moment about its tipping line that is in the direction of the counterweight. However, with respect to the barge centerline, the maximum moment will be toward the load, and the barge will tilt, or *list*. A pedestal-mounted crane will most likely produce its governing tilt condition at the maximum value of rated load-times-radius and will tilt the barge downward on the boom side as well.

A crane that moves about the barge is similar to a mobile crane on the centerline. As the crane travels from the barge centerline toward the side of the barge, the weight of the crane causes the barge to tilt down toward that side. When the crane picks up a load, the list angle of the barge increases.

Vessels of usual sizes used for cranes would not be expected to list excessively, but potential listing must always be checked because mobile-crane ratings will have to be adjusted if the tilt is more than 1%. Pedestal-crane ratings may also require adjustment. For small angles of tilt, an approximate method may be used to determine tilt angles for flat-bottomed vessels such as a typical barge. First, taking a pedestal crane mounted on the vessel centerline, consider a vessel of width B, length L, and weight Q; the list angle α in radians is given by

$$\alpha = \frac{12M}{B^3 L w} \tag{4.14}$$

where M is the moment the crane imposes on the vessel, w is the density of water [62.4 lb/ft^3 (1000 kg/m^3)] for fresh water and [64 lb/ft^3(1025 kg/m^3)] for salt water and α is in radians. To convert α to degrees, multiply by 180 and divide by π. Equation (4.14) is applicable, providing the entire barge bottom remains below the surface of the water. This can be checked by calculating barge draft s. Taking W_t as the weight of the crane and load.

$$s = \frac{Q + W_t}{BLw} \tag{4.15}$$

If $\alpha \le 2s/B$, the entire bottom will remain below the surface. The problem becomes more complex if part of the bottom rises above the surface, and Eq. (4.14) is no longer applicable.

If barge weight Q is not known, and as a practical matter it seldom will be known, it can now be easily calculated using Eq. (4.15). The draft s can be measured with or without the crane, and either Q or $Q + W_t$ (less load weight) can then be calculated from barge dimensions. Draft might be measured at each of the four corners of the barge, and the average used in Eq. (4.15).

Equation (4.14) has been derived by balancing the moment applied to the barge with the righting effect of the triangle of water displaced by the tilt; it is sufficiently accurate for angles of tilt of a few degrees. Equation (4.15) equates barge plus crane weight with the weight of the water displaced.

When a mobile crane is either centered on the barge or moves off the centerline toward the side of the barge, the moment M becomes

$$\begin{aligned} M &= (W_m + W)d_t - M_r \\ &= (W_m + W)d_t - M_{\text{net}} + S_h\alpha \\ \alpha &= \frac{(W_m + W)d_t - M_{\text{net}}}{B^3Lw/12 - S_h} \end{aligned} \tag{4.16}$$

with M_r taken from Eq. (4.5) for a level crane and d_t being the offset distance of the crane from centerline of barge to the appropriate crane tipping line.

Barge tilt causes the crane to go out of level with a similar affect on crane lift capacity as discussed earlier for an out-of-level support surface. Once barge tilt angle has been determined, the effect on capacity is given by Eq. (4.5), but only for mobile cranes. Barge tilt angles less than or equal to 1% (100 times α) will be within the mobile crane manufacturer's level limits and will not affect rating chart capacities. Pedestal-crane rating adjustment may require structural analysis.

The equations given are only approximate. They are intended as a quick and simple checking means to assess barge suitability or to gauge relevance of standard rating charts. Further work is required before any crane can be utilized on a barge, such as given in the more refined equations in the technical section that follow. In addition to consideration of stability and basic crane ratings, however, side loading must be evaluated. When the listing of the barge is 1% or more, it is likely that side-loading effects will require rating reductions.

There is an important difference in how a vessel-mounted crane with a lifted load behaves as compared to how a land-based machine behaves. Whenever the crane lifts or places a load that is not on the vessel, the barge list changes. Lifting loads causes the crane to lean toward the load, while landing a load produces the opposite effect. As the crane takes a strain on the load, the vessel and crane commence tilting and the boom tip moves away from the vessel centerline. Assuming the boom tip started out directly above the load CG, on lifting, the load will swing out. An operator or signalman normally tries to compensate for this by judgment, but it would be unusual for that compensation to produce precise results. The load will swing either away from the crane or toward it. This is virtually unavoidable and must be anticipated.

When a mobile crane operates from a single position on a vessel, fixing the crane in that position adds a measure of security against unintended movements. For a crawler crane, the track ends can be chocked with wood; steel stops would then be welded to the barge deck to hold the chocks in place and lashings, usually chains, placed to secure against overturning. Lashings should be left sufficiently

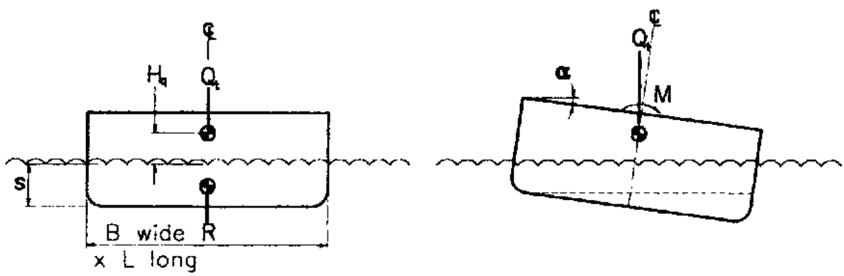


Figure 4.12 Diagrammatic view of a crane barge when level and when caused to list by applied moment *M*. Q_t is the combined weight of the barge, the crane, and the lifted load. *R* is the weight of the displaced water, which must be equal to Q_t.

slack to permit the tracks to lift a few inches so that the undercarriage can flex. When a truck crane is mounted on a barge in a fixed position, similar means can be applied. In this case, the chains would be placed at the outer ends of the outrigger beams and also left slack.

A far more complex situation arises should a crane be operating in rough waters. Here both the crane vessel and the load vessel move vertically due to wave action and may be moving in opposite directions. The crane can suddenly snatch the load off the deck of the load vessel, causing significant dynamic reactions, and then roughly land it on the deck again. Those reactions pose the threat of damage or tipping for the crane and damage to the load. Operation under such conditions is a special situation that is beyond the scope of this book, but some insights into the problem can be gained from an SAE recommended practice document.[10]

A more exact solution will have to account for movement of the CGs of the vessel and crane that takes place as the vessel lists. That movement adds to the moment acting on the system. Figure 4.12 shows a level barge and a barge tilted by a moment. The weight of the displaced water equals the weight of the barge, crane, and lifted load Q_t. Therefore, the weight of the displaced water will remain constant if Q_t is constant, regardless of the magnitude of system moment or how much the barge lists. As the barge tilts to angle α, a triangular wedge of water is created. The offset of the center of mass of that wedge from the barge centerline creates a moment that resists the applied moment. But, for a pedestal crane mounted on the barge centerline, tilting of the barge causes the CG of Q_t to displace and add to the applied moment. The amplified applied moment acting to tilt the barge now becomes

$$M_a = M + Q_t h_q \alpha$$

[10]SAE J1238, *Rating Lift Cranes on Fixed Platforms in the Ocean Environment*, Society of Automotive Engineers, Warrendale, Pa., 1978.

where M is the applied moment introduced by the crane and load and h_q is as shown in Figure 4.12. The buoyancy of the displaced water must support the barge plus crane and load weight and balance the amplified applied moment.

The cross-sectional area of the displaced water will be sB whether the barge is level or listed. Using small-angle geometry, the cross-sectional area of the displaced wedge of water will be $B^2\alpha/2$, and the centroid of that triangular area will be located at distance $B/6$ from the centerline of the barge. The centroid of the remaining portion of sB (the part that is not included in the wedge), will be centered with respect to the barge centerline. Therefore, the centroid of the entire mass of displaced water will be located

$$d_w = \frac{\frac{B}{6} \times \frac{B^2}{2}\alpha}{sB} = \frac{B^2\alpha}{12s}$$

from the centerline of the barge. d_w is the moment arm for the righting force generated by the displaced water to resist the amplified applied moment M_a. Dividing M_a by Q_t yields the eccentricity of the vertical loads, which must be equal to the eccentricity of the righting force because the forces themselves are equal and the entire system is in equilibrium.

$$\frac{B^2\alpha}{12s} = \frac{1}{Q_t} \times (M + Q_t h_q \alpha)$$
$$\alpha = \frac{12M}{B^3Lw - 12Q_t h_q} \tag{4.17}$$

which is more accurate than the angle calculated using Eq. (4.14) by virtue of the adjustment in the denominator.

Mobile cranes, whether or not positioned on the barge longitudinal centerline, will impose moment M_r on the barge, as given by Eq. (4.5), and

$$\frac{B^2\alpha}{12s} = \frac{1}{Q_t} \times [(W_m + W)d_t - M_{\text{net}} + S_h\alpha + Q_t h_q \alpha]$$
$$\alpha = \frac{(W_m + W)d_t - M_{\text{net}}}{B_3Lw/12 - S_h - Q_t h_q} \tag{4.17a}$$

which is similar to Eq. (4.16), but with an adjustment to account for the CG shift.

Once tilt angle α has been determined, Eq. (4.5) can be used to calculate mobile-crane stability ratings that are consistent with barge list and the appropriate ratio of rated to tipping load, but this need only be done should the list exceed 1%.

The preceding analysis has been concerned only with the stability aspects of crane ratings on a listing barge. Another important consideration is side loading, because most crane booms are prone to damage from severe side loads. For a mobile crane on the barge centerline, peak side loading will occur when the boom is oriented 45° to the centerline. Therefore, the moment M in Eq. 4.17 is calculated with respect to the side of the barge when the boom is at a 45° horizontal angle, or in other words 0.71 times its usual value. Now, if 0.71 times α calculated over the side is less than or equal to 1%, side loading will be within permitted limits. If greater than 1%, the rated load must be reduced accordingly, with guidance from the manufacturer. However, conservative reduced ratings may be obtained by dividing the rated load by $(1+35.5\alpha)$.

Mobile cranes that travel about the barge will experience side loading during most lifts, regardless of boom horizontal angle. For those installations, the crane manufacturer should be consulted.

Example 4.2 A mobile crane is mounted on a barge away from the centerline so that it imposes a moment of 10,000,000 lb-ft (13,558 kNm). The barge is 75 ft (22.9 m) wide and 200 ft (61.0 m) long with gross weight of 2300 kips (10,230 kN) including crane weight. The barge is in fresh water. How much will the barge tilt? Will the mobile-crane rating chart be valid?

Solution

Without using a full set of mobile crane data, a rough approximation is available by using Eq. (4.14).

$$\alpha = \frac{12\times 10{,}000{,}000}{75^3\times 200\times 62.4} = 0.023 \text{ rad} = 1.31°$$

Check using Eq. (4.15).

$$s = \frac{2{,}300{,}000}{75\times 200\times 62.4} = 2.46 \text{ ft (749 mm)}$$

$$\alpha < 2\times 2.46/75 = 0.066$$

This means that the entire bottom remains submerged and the solution is valid. The barge will tilt about 2.3%, and the mobile crane-rating chart will not be valid because tilt is more than 1%. Therefore, a more refined study using Eq. (4.17*a*) will be needed.

4.5 Track Mounting and Other Special Considerations

The list of crane types and configurations in Chap. 2 is extensive but far from complete. Treatment of stability issues for all is impractical, but the concepts described here, applied with some thought given to physical characteristics and the operating environment, can be extrapolated to other types. Not all crane types and configurations are prone to overturning.

Scrutiny of an unfamiliar or odd type of lifting devices should consider its subtleties. For example,

- High-Speed operations may induce large dynamic effects.
- Asymmetrical mounting may require evaluation from multiple directions.
- Backward stability may be a controlling factor for a heavily counterweighted machine.
- A revolver crane not attached to the slew ring has a circular tipping line.
- A high flexible structure warrants consideration of second-order effects.
- Operation on a curved or sloped track has static and dynamic implications.

A rail-mounted crane has a distinct set of considerations. Track placement tolerance is not dealt with in U.S. standards; for this FEM provides guidance.[11] The limits specified are intended by FEM for permanently installed cranes however; for temporary installations used in construction, the criteria may be too restrictive.

FEM specifies that the distance between rails must be kept within ±3 mm (1/8 in), while the difference in elevation between any two points perpendicularly across the rails may not exceed 0.15% with a maximum of 10 mm (3/8 in). Any two points along the track within one wheelbase length must be kept to within ±3 mm (1/8 in) in elevation for gages up to 3 m (10 ft), or within ±0.1% of the wheelbase for wider gages. Rail straightness is controlled by the requirement permitting no more than ±1 mm (1/25 in) deviation in any 2 m (6.5 ft) of length. Misalignment of the rail joints is not permitted at all.

For centrifugal force, FEM suggests that the effect be applied to the load only and ignored for the crane components; the experience of the authors indicates that this simplification may be sometimes inappropriate when evaluating stability for long-boomed cranes.

4.6 Dynamic Stability

Dynamic loads were discussed in Chap. 3. Those affecting stability include centrifugal force, inertial forces associated with travel, inertial effects from braking while lowering a load, and to some extent the

[11]Section 8.2, Fédération Européene de la Manutention, "Rules for the Design of Hoisting Appliances," 3d ed., Paris, 1998. See also ISO 8306, Cranes—Overhead travelling cranes and portal bridge cranes—Tolerances for cranes and tracks, International Organization for Standardization, Geneva, 1985.

harmonic loads from hoisting system acceleration abrupt acceleration or deceleration. The latter is not likely to cause overturning except under extraordinary circumstances. Complex situations do develop, however for cranes with heavy loads (with respect to rated load) undergoing for cranes with high CGs, when misalignments are present at track joints, and for other travel path irregularities. For some of those situations, dynamic load factors can be taken from FEM tolerance criteria, which will not be treated in detail here.

Dynamic loading is often considered in the stability equation by statically applying the maximum value and requiring that the crane remain stable when so loaded. That approach has the advantage of ease of calculation and usually proves conservative, but not always, as will be demonstrated. The error produced by this simple method can be quite large, however, when compared with more rigorous techniques.

In this section, a procedure will be presented for evaluating the effect on stability of short duration dynamic forces resulting from acceleration or deceleration of travel and hoist-system drives. This method offers a tool for crane operation planners to avoid overturning and reduce risk in crane operations where dynamic events may pose a threat to stability.

Centrifugal Force

When a crane swings, centrifugal force, as given in Eq. (3.17), will cause the load to move to an increased radius (Figure 4.13). The new radius will be a function of the load-line length, a random parameter that is practically inscrutable. Both the load and the centrifugal force can be taken as acting on the crane at the boom tip, however,

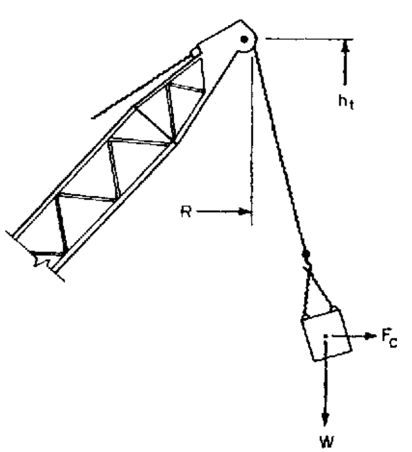


Figure 4.13 When a crane swings, centrifugal force F_c causes the load to move to an increased radius.

as the ropes pass the loads to the crane at this point. Taking moments, we find

$$W' = \frac{WR + F_c h_t}{R} = W + F_c \frac{h_t}{R} \tag{4.18}$$

where W' is the effective vertical load acting at radius R that is equivalent in effect to both W and F_c.

Many mobile cranes are capable of a rapid swing that can induce a strong centrifugal force. When the crane is also lifting a rated load, the addition of centrifugal force can bring the crane to the point of tipping. The following example illustrates:

Using the crane data supplied earlier in treating out-of-level operation, which for that particular crane describes $R = 262$ ft (79.9 m) and $h_t = 191.7$ ft (58.4 m), and taking a load that is 85% of the tipping load while the crane swings at 1 r/min, from Eq. (3.17) we get

$$F_c = \frac{0.85(2800)(262)}{32.2}\left(\frac{\pi}{30}\right)^2 = 212.4 \text{ lb } (994.8 \text{ N})$$

Equation (4.18) yields the effective load at radius R.

$$W' = 0.85(2800) + 212.4\frac{191.7}{262} = 2535 \text{ lb } (11.3 \text{ kN})$$

which shows that centrifugal force has taken up almost 40% of the difference between rated and tipping load. Of course, centrifugal force acts on the boom as well; it can be treated as though acting at the boom CG. At the load radius R, the effective load is then found to be about 3300 lb (1497 kg), which is greater than the tipping load. The indication is that the crane will overturn.

However, other factors may mitigate the opportunity for the crane to tip. For example, the high centrifugal force might not occur when the crane is in its least-stable orientation, or the crane may swing past that least-stable point before the slow tipping process takes hold. Moreover, most crane operations are carried out over too short a swing arc for a high rotational speed to develop.

Centrifugal force is less problematic for a tower crane. Slewing speed is modulated by the drive system and the ratings make allowance for load dynamics. The high CG of tower cranes makes these precautions necessary.

Inertial Forces Affecting Stability

An accepted practice when establishing lift ratings on the basis of crane stability is to simulate short-lived inertial forces acting on a crane by adding peak dynamic values to static loads, by using arbitrary dynamic factors, or by taking a percentage of the tipping load.

Over time, those approaches have proven to be suitable for rating purposes, but they are crude simplifications of dynamic effects. In some instances such as in the investigation of failures, these methods may be valid only as early stage approximations.

The traditional methods are in effect force-based systems in which static and dynamic components are added algebraically. But in order to tilt a crane and cause it to overturn, work is required. A dynamic event converts kinetic energy into work, as when a crane traveling with a load makes a quick stop or a load being lowered is abruptly stopped before reaching the ground. When that energy is sufficient to do the work needed to overturn the crane, it will overturn. If the energy is not sufficient to carry the crane to the critical tilt angle, it may tilt but not overturn. The static loads acting on the crane define the tilt angle at which the crane will overturn and therefore the work required to tilt the crane to the limit. The greater the static forces with respect to the tipping load, the less energy required to do the work and cause failure. Among the static loads, wind from an unfavorable direction produces an effect similar to adding load on the hook; gusting adds further loading as well as causing a dynamic "bounce."

Dynamic (inertial) forces resulting from acceleration or deceleration furnish energy input. The closer the crane is to tipping because of its static loads, the more sensitive it will be to dynamic disturbances; an experienced crane operator knows sensibly that motions should be made less abrupt when operating close to capacity. An analysis of the physics of crane dynamics affirms this intuition.

The ability to determine how dynamic actions affect crane stability can be a useful tool in the hands of crane installation planners, particularly for situations pushing the norms of established practice. As this section will show, each element in the dynamic stability equation offers planners insights into the means available to ameliorate or control dynamic effects and thereby reduce risk and improve the probability of success. On the other hand, dynamic stability analysis can also show when unexpected dynamic episodes will offer no stability threat, permitting risk control assets to be more favorably employed.

Figure 4.10 can be considered a generalized dynamic model of a crane.[12] It will be assumed that the load is suspended close to the boom tip for one or a few parts of hoist line, or further from the tip for multipart lines. This assumption is made to minimize the effect of harmonic spring effects on the hoist system so that the entire crane

[12]The method for evaluating dynamic stability that follows has been taken from SAE Technical Paper 972721, *Overturning Stability of a Free Standing Crane Under Dynamic Loading*, Anatoly A. Zaretsky and Howard I. Shapiro, Society of Automotive Engineers, Warrendale, Pa., Sept. 1997.

can be considered as a *single-degree of freedom (SDOF) system*.[13] The overturning static force from load weight can then be taken to act at the boom tip. Horizontal static forces as from steady-state wind or centrifugal force are not shown on Figure 4.10, but will be included in the moment of static forces acting on the crane, Eq. (4.4), and the equations that follow. Wind gusts are not considered.

Newton's second law offers an equation of motion describing the response of the loaded crane of Figure 4.10 to the action of very short duration dynamic forces, or after the cessation of mechanism acceleration or deceleration. The differential equation is

$$J\ddot{\varphi} = F(\varphi) \tag{4.19}$$

where J is the moment of inertia of crane and load masses about point O of Figure 4.10, the tipping; φ is the crane angle of tilt (positive clockwise); and F(φ) is the nonlinear function reflecting the moment about O of all forces acting on the crane. Angles are in radians, and the dots superscript reflects differentiation with respect to time.

After integration, Eq. (4.19) becomes

$$J\dot{\varphi}^2/2 = \int_{\varphi_o}^{\varphi} F(\varphi)d\varphi \tag{4.19a}$$

where φ_o represents a stable equilibrium condition under static loading; it is the tilt angle of the crane at the instant the dynamic event commences. But noting the general expression for kinetic energy in a rotating system,

$$T = J\dot{\varphi}^2/2$$

Eq. (4.19*a*) can be interpreted in a different way. It can be thought of as expressing a basic relationship where the left side portrays system kinetic energy and the integral on the right side reflects the work that will be done as the crane tilts from φ_o to φ. This interpretation reveals that there is a direct link between input energy and the resulting crane tilt. At its upper limit, φ will be the angle of unstable equilibrium φ_{un}. Therefore, the minimum amount of energy required to tilt the crane to φ_{un} is the energy that will produce overturning failure.

Also, this crane mathematical model can be viewed as being comparable to a nonlinear conservative pendulum, since dissipating forces have not been taken into account in the crane model. It is reasonable to ignore damping, because only part of the first oscillating cycle of the crane is of interest and very little dissipation occurs during

---

[13]Further work by Prof. Zaretsky has demonstrated that these results are valid for multi-degree-of-freedom systems. Therefore, the close-to-the-boom-tip assumption need not be imposed. The mathematical proofs were offered in private correspondence received just before publication of this book.

the first cycle. Pendulum oscillations go on without end once they begin, but as a pendulum passes the point of stable equilibrium, both $T_{\max}$ and $\dot{\varphi}_{\max}$ will occur. Using the pendulum analogy motivates this criterion for crane dynamic stability.

$$T_{\max} = J\dot{\varphi}_{\max}^2/2 \leq \int_{\varphi_o}^{\varphi_a} F(\varphi)\,d\varphi \qquad (4.20)$$

If the input energy is less than or equal to the work required, the crane will not rotate beyond φ_a or overturn if φ_{un} is the limit. A value for φ_a, the upper limit of the integral, may be selected that is less than φ_{un}. This may be done to provide a margin of protection against overturning, or in the case of rail-mounted cranes, to prevent the wheel flanges from rising above the track and in that way avoid possible derailment when the crane returns to the static state.

From an equilibrium expression including all static forces acting on the crane, φ_{un} can be found; the procedure is similar to that used in deriving Eqs. (4.5) and (4.6), but substituting the tilt angle φ for the out-of-level angle α. This will lead to

$$F(\varphi) = M_r - S_h\varphi \qquad (4.20a)$$

where the terms M_r and S_h are as defined for Eqs. (4.5) and (4.6); M_r intrinsically includes out-of-level effects. When the crane has tilted to the point of unstable equilibrium, $F(\varphi) = 0$, the static forces that had induced moments are balanced and φ is at its limit φ_{un},

$$0 = M_r - S_h\varphi_{un}$$

$$\varphi_{un} = M_r / S_h$$

Now assume a spring at the left crane support in Figure 4.10 to reflect elastic effects at that support. The actual support spring response is often so stiff that it can be ignored in practical calculations, but it must be used to develop the theoretical behavior of the crane. The static loading on the crane will therefore cause some elastic subsidence at the spring support point; this will induce a negative tilt angle.

$$\Delta = -\frac{M}{cb}$$

where c is the support spring constant and b is the distance between the spring support and fulcrum point O. The crane will now be at rest at the initial tilt angle φ_o.

$$(\varphi_o < 0) \qquad \varphi_o = \frac{\Delta}{b} = -\frac{M}{c_o} \qquad M = -\varphi_o c_o$$

where $c_o = cb^2$. At any time $\varphi < 0$, the moment of the spring reaction will balance the moment of crane forces [as reflected in Eq. (4.20*a*)] that induced the spring reaction. For the initial state, $\varphi = \varphi_o$, the entire moment must be balanced because the crane is in static equilibrium; therefore

$$M = F(\varphi_o)$$
$$-\varphi_o c_o = M_r - S_h \varphi_o$$
$$\varphi_o = -M_r/(c_o - S_h)$$

The defining points have now been developed for a plot relating moment $F(\varphi)$ and tilt angle; $F(\varphi_o) = 0$, $F(0) = M_r$, and $F(\varphi_{un}) = 0$. Figure 4.14 shows that curve with straight lines connecting the points. As will be explained later, using straight lines when perhaps curved lines would be more exact does not affect the accuracy of this analysis. From Figure 4.14, the sloped legs of the curve yield the following equation:

$$\begin{aligned} F(\varphi) &= M_r + (c_o - S_h)\varphi \qquad \varphi < 0 \\ F(\varphi) &= M_r - S_h\varphi \qquad \varphi \geq 0 \end{aligned} \tag{4.20b}$$

Equation 4.20*b* and Figure 4.14 reveal that the maximum moment experienced by the crane occurs when the spring support becomes fully unloaded.

The area under the curve of Figure 4.14 between φ_o and φ_a is equivalent to the value of the integral on the right side of Eq. (4.20); $I(\varphi)$ includes the areas of triangle abc and trapezoid bcde.

$$\begin{aligned} I(\varphi) &= \mathrm{A}_{\mathrm{abc}} + \mathrm{A}_{\mathrm{bcde}} = 0.5M_r^2/(c_o - S_h) + M_r^2(2-\mu)\mu/S_h \\ &= 0.5M_r^2[S_h/(c_o - S_h) + (2-\mu)\mu]/S_h \end{aligned} \tag{4.20c}$$

where $\varphi_a = \mu\varphi_{\mathrm{un}}$ and $0 < \mu \leq 1$.

With the exception of T_{max}, all of the values of Eq. (4.20) have been developed. Note that the stability criterion depends on the area

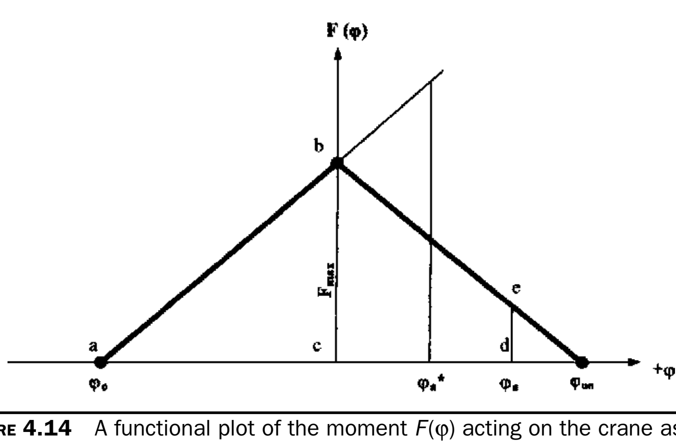


FIGURE 4.14 A functional plot of the moment $F(\varphi)$ acting on the crane as it tilts from its initial condition of static equilibrium at angle φ_0.

bounded by the points defining the curve of the integral between the limits φ_o and φ_a, but does not depend on the shape of the curve because the work done by the system is conservative. For a conservative system, the work done is a function of the endpoints, not the path taken. Whether the lines connecting the points in Figure 4.14 are curved or straight does not affect the value of the integral. For that reason, we can use any function $F^*(\varphi)$ to replace the function $F(\varphi)$, in order to simplify the function, provided the equality of the integrals is maintained. Thus

$$\int_{\varphi_o}^{\varphi_a} F(\varphi)\,d\varphi = \int_{\varphi_o}^{\varphi_a^*} F^*(\varphi)\,d\varphi \tag{4.20d}$$

provides the rule that can be used to simplify the system by linearization.

Figure 4.15 depicts a linear system of loading wherein the dynamic moment $P(t)$ attains full value instantaneously at $t = 0$ and maintains a constant value until $t = t_a$, at which time it diminishes instantaneously to zero. This linear system is examined with the goal of manipulating it through use of Eq. (4.20*d*) to produce a practical solution to Eq. (4.19). The linear system can be described by a form of the equation of motion for an undamped SDOF system common in the literature.

$$J\ddot{z} + (c_o - S_h)z = P(t) \tag{4.20e}$$

where the new variable z, $z = (\varphi - \varphi_o)$, is used to make the solution to Eq. (4.20*e*) independent of the initial conditions since $\dot{z} = \dot{\varphi}$ and $z(0) = 0$.

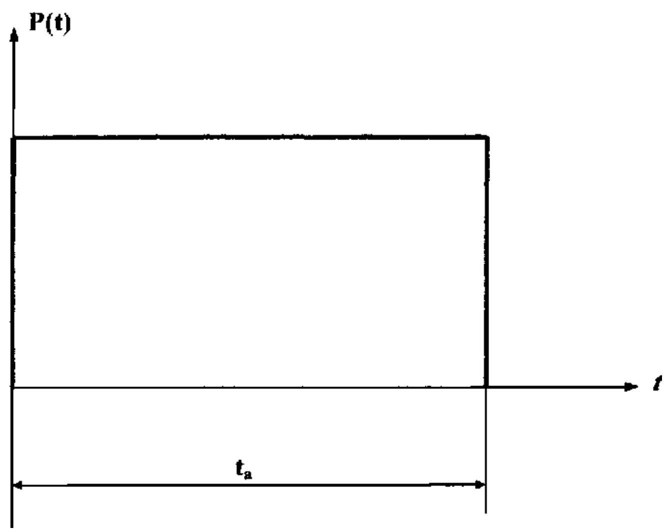


Figure 4.15 Hypothesized moment/time model for linearizing the effects of dynamic events on the crane of Figure 4.10.

With the additional boundary condition $\dot{z}(0) = 0$, the general solution to Eq. (4.20*e*) is

$$z = [P/(c_o - S_h)](1 - \cos \omega t)$$

where $\omega^2 = (c_o - S_h)/J$ and ω is the circular frequency of loaded crane free oscillations. The solution applies for $0 < t \le t_a$, but we will also need displacement z during the interval $t > t_a$. After using a two-step solution and mathematical manipulation, we get

$$\dot{z} = 2P[\omega/(c_o - S_h)](\sin 0.5\omega t_a)\cos \omega (t - 0.5t_a) \tag{4.20f}$$

Maximum velocity $\dot{z}_{\max}$ will be given by Eq. (4.20*f*) when the cosine term equals 1. But, after multiplying the right side of Eq. (4.20*f*) by $0.5\ \omega t_a/0.5\ \omega t_a$, Eq. (4.20*f*) can be further simplified and then equated to $T_{\max}$ in consideration of the energy criteria, Eq. (4.20), to yield

$$T_{\max} = 0.5\ J[(Imp/J)(\sin 0.5\ \omega t_a)/0.5\ \omega t_a]^2 \tag{4.20g}$$

where $Imp = Pt_a$, an impulse of the moment P. Designating a new variable,

$$k_t = (\sin 0.5\ \omega t_a)/0.5\ \omega t_a$$

we can note that the limit of $k_t = 1$ as $t_a \to 0$. By substituting $t_o = 2\pi/\omega$, the natural period of loaded crane oscillations

$$k_t = (\sin \pi t_a/t_o)\pi t_a/t_o \tag{4.21}$$

The remaining portion of Eq. (4.20*g*) can be interpreted as the magnitude of $T_{\max}$ produced by instantaneous acceleration T_o. The k_t term, where $k_t \le 1$, reflects that the event takes place during an actual time interval t_a. For the general case,

$$T_{\max} = T_o k_t^2 \tag{4.22}$$

The natural form of T_o is $0.5\ J(Imp/J)^2$ as in Eq. (4.20*g*), but from the law of conservation of energy we can obtain a more practical equation.

$$T_o = 0.5 \sum m_i v_j^2 \tag{4.23}$$

where m_i is a mass (or part of the crane) moving at linear velocity v_j when the *jth* mechanism is applied.

If M_d is taken to represent the moment induced by all dynamic forces causing tilting, the crane will remain stable provided $M_d \le M_r$. Therefore the stability limit has been reached should $M_d = M_r$.

In order to make the kinetic energy of the nonlinear system of Eq. (4.19) equal to the kinetic energy of the linear system of Eq. (4.20*e*), the linearization rule of Eq. (4.20*d*) must be satisfied. This can be accomplished by setting the value of the integral given by Eq. (4.20*c*) equal to $T_{\max}$ of Eq. (4.22) and noting that at the limit $M_d = M_r$. Manipulation then yields the following condition for dynamic stability:

$$M_d = \{2T_o k_t^2 S_h / [S_h/(c_o - S_h) + (2 - \mu)\mu]\}^{1/2} \leq M_r \tag{4.24}$$

Equation (4.24) defines the maximum value that M_d can take without causing the crane to tilt beyond the limit φ_a, and represents but another way to express the criterion of Eq. (4.20). M_d is an equivalent moment representing the effects of the time-varying dynamic forces that impose kinetic energy $T_{\max}$ on the crane; it is conceptual rather than physical. M_d should not be used for structural calculations because it will not reveal peak values for stress; it is a parameter for use only in stability determinations.

Because $T_{\max}$ has been derived using extreme value considerations [the cosine term of Eq. (4.20*f*) has been set equal to 1], and some energy may in fact be dissipated by spring action in the hoist ropes, M_d represents the maximum value that this static equivalent moment can take. Therefore, if $M_d \leq M_r$, the crane tilt angle will be less than or equal to, but will not exceed, φ_a.

When two or more drive mechanisms simultaneously induce dynamic forces from acceleration or deceleration, the kinetic energy from each mechanism can be added in the following manner:

$$T_C = \sum T_{oi} k_{ti}^2 \tag{4.25}$$

where T_{oi} is the kinetic energy of the mass accelerated by the *i*th mechanism and k_{ti} takes into account the time during which the acceleration or deceleration of that mechanism takes place. T_C is used in place of $T_o k_t^2$ in Eq. (4.24) when more than one mechanism is engaged simultaneously. Therefore, M_d is not the algebraic sum of the outcome of multiple dynamic factors. When mechanisms do not engage simultaneously, their effects will be out of phase with one another, and the sum of nonsimultaneous dynamic actions will never produce more energy than reported by Eq. (4.25) for simultaneous events.

Plotting Eq. (4.21) will show that k_t will be zero when t_a/t_o takes full integer values, but magnitudes of the ratio t_a/t_o in the vicinity of 1 and greater produce such small energy values when inserted into Eq. (4.24) that the ensuing moment M_d will be of little consequence; therefore, only mechanisms where $t_a/t_o < 1$ need be considered.

Every loading event will cause a crane to tilt, however small the tilt angle will be. The rigidity term c_o tends to be very large for most actual support situations when compared with the moment S_h. That results in the first term in the denominator of Eq. (4.24) being small, often falling between 0.005 and 0.025; in those instances, that term can be eliminated to simplify the equation, which then becomes

$$M_d = \{2T_o k_t^2 S_h/[(2-\mu)\mu]\}^{1/2} \tag{4.26}$$

Rearranging Eq. (4.24), and taking the limit value for M_d produces an equation for the maximum energy from a single mechanism that can be tolerated.

$$\begin{aligned}[T_o k_t^2]_{\max} &= 0.5M_r^2[S_h/(c_o - S_h) + (2-\mu)\mu]/S_h \\ &= \varphi_{\text{un}} M_r[S_h/(c_o - S_h) + (2-\mu)u] \end{aligned} \tag{4.27}$$

For several mechanisms,

$$[T_c]_{\max} = \varphi_{\text{un}} M_r[S_h/(c_o - S_h) + (2-\mu)u] \tag{4.27a}$$

The static equivalent moment M_d can be calculated and compared with the static resisting moment M_r. There is no mathematical correlation between those moments to inform designers when adequate protection from failure has been achieved, other than $M_d \le M_r$. Designers and planners are therefore left with their experience and judgment.

The dynamic stability equations, Eqs. (4.24), (4.26), (4.27), and (4.27*a*), include variables that present planners with opportunities to reduce overturning risk than dynamic forces. Each parameter represents an implication that a codition in the field might be varied or limited to control risk. First, there are obvious measures for increasing the static moment M_r, such as minimizing the operating radius; disassembling a portion of the lifted load, when feasible (which will also reduce S_h); establishing more severe limits for levelness, especially for travel; limiting swing speed to reduce centrifugal force and hence M_{hor}; and reducing the ambient wind speed limit at which operations will be permitted to go forward to reduce M_{hor}.

Second are the measures available that relate to the mechanisms. For each mechanism that could be engaged simultaneously, suppressing motion speeds will reduce the energy parameter T_o, and if measures can be taken to be certain that braking stop times can be increased, k_t values will be reduced accordingly. Increasing the number of parts of line and use of power load lowering are viable means for accomplishing reductions in both T_o and k_t for the hoist mechanism, but that mechanism does not often produce strong dynamic effects. The travel drive mechanism usually induces a strong influence.

Third and last using the shortest length of boom consistent with maintaining adequate clearances will increase M_r because the effective dead load moment will increase, and S_h will decrease as well.

Example 4.3 To demonstrate dynamic stability effects, we will consider a crane similar in size to the one used earlier in studying out-of-level effects, but for this evaluation the crane will be a crawler-type machine with a shorter boom [250 ft (76.2 m) versus 320 ft (97.5 m)] at a shorter radius [70 ft (21.3 m) versus 250 ft (76.2 m)], and hence a heavier load. The parameters for this crane are

Solution

Lifted load

$$W = 62{,}800 \text{ lb } (28.5 \text{ t}) \qquad d = 58 \text{ ft } (17.7 \text{ m}) \qquad h_t = 249.4 \text{ ft } (76.0 \text{ m})$$

Crane weight

$$W_m = 351{,}300 \text{ lb } (159.3 \text{ t}) \qquad d_m = 13.83 \text{ ft } (4.2 \text{ m})$$

$$h_m = 19.37 \text{ ft } (5.9 \text{ m})$$

Less boom

$$W^* = 319{,}000 \text{ lb } (144.7 \text{ t}) \qquad h^* = 6.1 \text{ ft } (1.9 \text{ m})$$

$$M_{\text{hor}} = 147{,}500 \text{ lb}\cdot\text{ft } (200 \text{ kNm}) \qquad \text{from wind}$$

The tilt angle limit φ_a is set to φ_{un}, the angle of incipient overturning.

Part of the travel path is 1.5% out of level (which exceeds limits set by most crane manufacturers).

Travel speed $v_1 = 1.5$ ft/s (0.46 m/s) Stopping time $t_1 = 0.5$ s

Hoist lowering speed (two parts) $v_2 = 3.0$ ft/s (0.9 m/s)

Stopping time $t_2 = 0.5$ s

Loaded crane natural period

$$t_o = 2.25 \text{ s}$$

First, determine if the crane will be stable following a sharp travel stop.

From Eq. (4.4) with α set to 0

$$M_{\text{net}} = 351{,}300 \times 13.83 - 62{,}800 \times 58 - 147{,}500$$
$$= 1{,}068{,}579 \text{ lb}\cdot\text{ft } (1449 \text{ kNm})$$

$$S_h = 351{,}300 \times 19.37 + 62{,}800 \times 249.4$$
$$= 22{,}467{,}000 \text{ lb}\cdot\text{ft } (30{,}461 \text{ kNm})$$

Eq. (4.5),

$$M_r = 1{,}068{,}579 - 22{,}467{,}000 \times 0.015$$
$$= 731{,}574 \text{ lb}\cdot\text{ft } (992 \text{ kNm})$$

$$t_1/t_o = 0.5/2.25 = 0.22$$

From Eq. (4.21),

$$k_t = \sin(0.22\pi)/0.22\pi = 0.92 \qquad k_t^2 = 0.85$$

From Eq. (4.23),

$$\begin{aligned} T_o &= 0.5(351{,}300 + 62{,}800)1.5^2/32.17 \\ &= 14{,}481 \text{ lb}\cdot\text{ft (19.6 kNm)} \end{aligned}$$

Eq. (4.26) with φ_{un} as the limit $\mu = 1$ and the equation simplifies to

$$\begin{aligned} M_d &= (2 \times 14{,}481 \times 0.85 \times 22{,}467{,}000)^{1/2} \\ &= 743{,}697 \text{ lb}\cdot\text{ft (1008 kNm)} \end{aligned}$$

$M_d > M_r$. The crane is unstable.

Measures that can be taken to make the crane stable include

- Tightening ambient wind limitations at the site; limiting wind speed to 75% of the previously considered value $M_{\text{hor}} = 0.75^2 \times 147{,}500 = 82{,}969$ lb · ft (112.5 kNm)

$$\begin{aligned} M_r &= 731{,}574 + 147{,}500 - 82{,}969 \\ &= 796{,}105 \text{ lb}\cdot\text{ft (1079 kNm)} \\ &> M_d \quad \text{The crane is stable.} \end{aligned}$$

- Reducing travel speed to 1.0 ft/s (0.3 m/s)

$$\begin{aligned} T_o &= 14{,}481 \times 1^2/1.5^2 \\ &= 6436 \text{ lb}\cdot\text{ft (8.7 kNm)} \\ M_d &= 743{,}697(6436/14{,}481)^{1/2} \\ &= 495{,}798 \text{ lb}\cdot\text{ft (672 kNm)} \\ &< M_r \quad \text{The crane is stable.} \end{aligned}$$

- Levelling the travel path to 1% maximum grade

$$\begin{aligned} M_r &= 731{,}574 + 0.005 \times 22{,}467{,}000 \\ &= 843{,}909 \text{ lb}\cdot\text{ft (1144 kNm)} \\ &> M_d \quad \text{The crane is stable.} \end{aligned}$$

Had this evaluation been performed by traditional methods, i.e., using Newton's second law equation $F = ma$,

$$\begin{aligned} M_d^* &= (351{,}300 \times 19.37 + 62{,}800 \times 249.4)(1.5/0.5)/32.17 \\ &= 2{,}095{,}000 \text{ lb}\cdot\text{ft (2841 kNm)} \end{aligned}$$

which overstates travel stop dynamic effects by about 180% of the value determined by the method just presented previously.

For the hoisting function,

$$t_2/t_o = 0.5/2.25 = 0.22$$

This means that k_t^2 remains 0.85.

$$T_o = 0.5 \times 62{,}800 \times 3^2/32.17 = 8785 \text{ lb}\cdot\text{ft } (11.9 \text{ kNm})$$

$$M_d = (2 \times 8785 \times 0.85 \times 22{,}467{,}000)^{1/2} = 579.252 \text{ lb}\cdot\text{ft } (785 \text{ kNm})$$

$M_d \leq M_r$. The crane is stable. Even with an instantaneous stop ($k_t^2 = 1$), the crane will remain stable. Because crawler cranes are rated at 75% of the tipping load, hoist-system dynamic effects rarely pose a threat to stability. When evaluated using Newton's second law,

$$M_d^* = 62{,}800 \times 58 \times (3.0/0.5)/32.17 = 679{,}341 \text{ lb}\cdot\text{ft } (921 \text{ kNm})$$

This is an overstatement of only about 17% compared with the energy method previously presented.

For illustrative purposes, assume that this crane is rail mounted and tilting will be limited to a rise of 1 in (25 mm) so that the wheel flange does not lift above the rail top. The crane base dimension $B = 21$ ft (6.4 m). With a 1% out-of-level limit

$$\varphi_{un} = 843{,}909/22{,}467{,}000 = 0.04 \text{ rad}$$

$$\varphi_a = 1/(21 \times 12) = 0.004 \qquad \mu = 0.004/0.04 = 0.1$$

For travel dynamic effects,

$$M_d = 743{,}697/[(2 - 0.1)0.1] = 3{,}914{,}195 \text{ lb}\cdot\text{ft } (5307 \text{ kNm})$$

What this result reveals is that M_r must be very high indeed if a crane is to respond to dynamic excitation by tilting through a very small angle. It is interesting to note that when using the traditional method for dynamic analysis, this last problem cannot be solved because the traditional method does not address tilt-angle limitation.

CHAPTER 5

Mobile-Crane Installations

Cranes and airplanes have something in common other than the airspace they share. The early history of aviation belongs to the barnstormers and bush pilots who pushed their rudimentary craft to the limits with hardly a thought for risk. Likewise, crane operation was once a seat-of-the-pants skill governed largely by the notion that lifting equipment could be used right up to the point of tipping over.

Both cranes and aircraft of early vintage were machines with modest capabilities that responded almost like extensions of the operators themselves. Machines were relatively small and inexpensive, and pilots frequently survived crashes while crane operators usually walked away. Collateral damage to persons and property was usually inconsequential. Not so today where property damage claims can be stratospheric and the public is often be in harm's way. Society today is far less tolerant of accidents.

A high-performance contemporary crane cannot be run from intuition; an operator who tries to work that way will not be warned by his gut of an approaching failure and will be blindsided when it inevitably happens. And when it does happen the cost will be intolerable.

More cerebral demands on the operator reflect changes both in construction equipment and construction culture. Risk management—not bravado—is the new order of the day in mobile crane practice. An underlying rule of this practice is that a balance must be struck between safety and economy. Success demands well-structured organization and unrelenting attentiveness to details.

A successful mobile-crane operation is not simply one that has been carried out without mishap. True success has been achieved only when, in addition, the operation has been executed at the lowest possible cost consistent with tolerable risk. The measurement of risk management success is imprecise, however, and may be hard to appreciate by considering only an individual operation. But, over time, repetitive accident-free and productive crane use will show up favorably on the balance sheet.

There are proven risk management measures, such as worker training and rigorous equipment maintenance, to name but two. Another key measure for mobile-crane use is planning. The need will vary widely from operation to operation, to be sure. Sometimes an hour reviewing site conditions, loads, and equipment characteristics is sufficient. At the other extreme, some operations demand hard months of preparation.

This chapter is concerned with the details of planning. It presents steps that would be carried out in selecting a mobile crane for an assignment and putting it to work, in particular,

- Evaluating access for the crane to reach the site and get set up for operation
- Selection and positioning
- Assessing working clearances
- Determining loads imposed by the crane on the supporting surface
- Specifying supports under the crane
- Special arrangements such as pick-and-carry and multiple crane lifts

Knowing these elements of planning can help decide which are necessary for any one situation and how elaborate the planning process must be. Mathematical procedures are introduced and explained which may not always be required, but can be valuable resources at times. However, planning a crane installation is an art requiring extensive field experience and knowledge of cranes. The material given here can be put to good use by those who have that experience and knowledge and can be invaluable to those who wish to gain it.

5.1 Introduction

Two years out of engineering school, one of the authors made his first mobile-crane lift plan. It involved a crawler crane picking a series of massive precast columns from trailers and rotating each one in the air before setting it on anchor bolts. He designed the custom hardware, specified the heavy rigging, and plotted out the elaborate procedure. He prepared fabrication sketches and supervised the manufacturing of the lift brackets. It was not a novice lift plan. How proud of himself he was! On the first pick the crane turned over.

The only injury was to the author's self esteem, though he soon realized that the accident was not his doing. It was caused by poorly prepared ground support: The end of one track was on a piece of plywood concealing a void. Working at full-load chart capacity, the plan engendered what he would recognize today as a series of *critical* *lifts*. He had in fact requested a heavier crane but was told it was not in the budget. Unknown to him at the time—it should not properly have been an expectation of a junior-level engineer to know—without adequate field controls, engineering alone is insufficient preparation for a critical pick. His lift plan was technically sound. But a more comprehensive plan would have included human factors and field controls such as a crew briefing, surveying, preparing the ground, and inspection prior to each pick. In hindsight, the field crew's error merely compounded management-level decisions to use a marginal crane and exercise lax controls.

A mobile crane presents day-by-day challenges for the user to avoid calamity. This is not to say that users of mobile cranes are inherently irresponsible people. Safe operation can be a routine and crews can go home every night without having tales of close encounters to share with buddies or families. But a mobile crane at full working capacity can have a thin margin against tipping, and a crane that must be set up again and again—sometimes daily or more often—exposes repeated opportunities for critical mistakes.

The following sections offer engineering solutions to many installation planning problems, but pure engineering alone is insufficient. A theoretical understanding of constraints should be tempered with an awareness of how these limitations play out at the site—how they ultimately affect safety and productivity. Situations at construction sites can change quickly, particularly those concerning access, obstructions, and ground conditions. The engineer needs to be partnered with field personnel who share the same concerns but have a different perspective.

A planner should take notice that the world of mathematics and graphics is no more than a simulation of the jobsite. A crane will not be positioned accurately to the inch, indeed, often not to the foot. It may be out of plumb and it will deflect; when swinging and hoisting loads, inertial forces cause parts of a crane to displace from the static position. A building is not made in a machine shop; its tolerances may be measured in inches, and temporary items may be in place projecting beyond the clean outlines shown on the design drawings.

5.2 Transit to the Site

Mobile crane planning often starts with a look at the means of getting the machine to the work location. Dimensions, turning radius, gross weight, and axle loads come up against transit route restrictions. Each crane model, whether it is hauled or it drives under its own power, has unique characteristics that affect its transit. To add complications, states, provinces, and other jurisdictions promulgate their own road weight and dimension regulations. Any proposed route, moreover, must be examined for width and height limitations as well as bridge or culvert capacities. Power lines running over the route require close scrutiny and might need to be turned off, relocated, or shielded.

Crane components carried on flat bed or lowboy trailers are usually wide loads and may be high as well, often requiring a check of overhead clearance at underpasses. Truckers need to be aware that latticed booms, jibs, and extensions do not get along well with chains. Chain tightening can bend, nick, or wrinkle sections and seriously reduce their ability to support load. Even nylon strap tie downs can bend boom or jib chords. Tie downs should be located only where the chords are supported by diagonal members, and padding should be inserted between chains and load-supporting members. Some riggers visit local carpet shops to get cutoffs and remnants to use as padding.

Construction site gates and internal access roads are often too constricting for a large crane. Abrupt changes in grade can cause the chassis of a truck crane or the bed of a lowboy haul trailer to hang up. If the site area is limited, space may need to be found to marshal trucks in the neighborhood until their loads are needed for assembly of the crane. Large telescopic cranes with luffing jibs may require ten or more truckloads of components.

A crane on a *rough-terrain carrier* is roadworthy but its low road speed and uncomfortable ride limit self-powered transit moves to distances of only a few miles. For longer runs, the crane should be carried on a trailer. An *all-terrain crane*, on the other hand can comfortably travel at highway speed and over long distances. Gross vehicle weight of either type is low enough to allow passage over most roads and bridges though larger all-terrain cranes may need some stripping down. Width and height do not often prove troublesome. Many models are provided with four-wheel steering, making them very maneuverable and enabling them to be operated in tight quarters, even at times within a building if the floor is able to support the weight.

A crawler crane is capable of self-powered travel about the jobsite, but for transit it must be trucked or carried by rail. Smaller crawler cranes can be driven directly onto drop-bed trailers, often with the basic boom mounted, to make the move in one piece if road restrictions permit. As weight and width increase, it becomes necessary to remove more and more components in order to remain within road or rail limitations. For rail transit, the carrier needs to be consulted, as limitations vary with the routing. To move the largest crawler cranes, dozens of truck trailers or rail cars may be needed. Designers of new crawler models strive to make them easy to assemble. Some can self-assemble, all or in part.

A truck crane makes all but exceedingly long transit moves under its own power. Most are designed to be flexible in this regard, with as many as 15 or 20 individual components removable for axle-load reduction or balance. Manufacturers publish tables of axle-load

Boom Length	A	B	C	D	E	F	G	H	J	K	L
50′ 0″	16′ 10$\frac{3}{16}$″	53′ 4″	12′ 11$\frac{1}{2}$″	71′ 11$\frac{3}{16}$″	50″	6′ 11$\frac{1}{4}$″	42″	19′ 2″	7′ 1$\frac{1}{4}$″	76$\frac{1}{2}$″	32′ 6$\frac{1}{4}$″

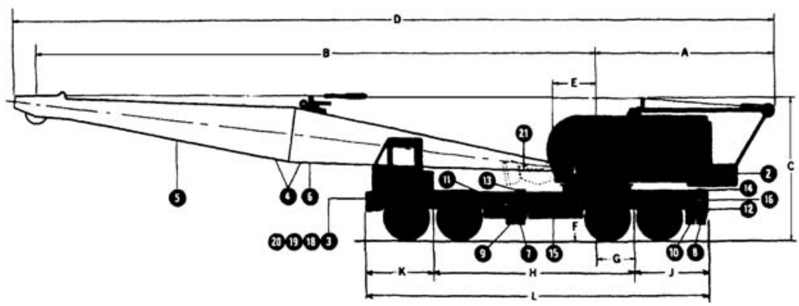


FIGURE 5.1 Typical truck-crane axle-loading adjustment chart and general dimension diagram. (*Harnischfeger Corporation.*)

adjustment data (Figure 5.1). Formats vary but the general scheme of adjustments is the same; the tables are used to find just which, if any, components need to be removed to satisfy axle-loading regulations. They also can be used to give the axle-load information necessary to check on-site travel moves or moves over critical bridges, ramps, or floors. For this work, additional data and some calculations beyond addition and subtraction may be required.

For the crane of Figure 5.1, item 1A gives the gross weight as 137,065 lb (62,172 kg) for a standard machine with a 50-ft (15.2-m) boom mounted and with hydraulic outriggers. Under the heading Effect of Adding, the weights of optional equipment are listed.

Assume that it is necessary to have nearly uniform loading on each axle. This can be accomplished by placing the boom over the rear of the carrier. Referring to Figure 5.1, the following adjustments are made:

	Item No.	Front Bogie, lb	Rear Bogie, lb
Basic machine	1A	48,423	88,642
Remove	4	+10,160	–17,300
	8	+700	–2,940
	10	+625	–2,625
	12	+60	–240
	14	+290	–1,250
	16	+640	–2,720
Final loads		60,898 (27,623 kg)	61,567 (27,926 kg)

The axle loads are balanced to within about 1% by removing the boom and the entire rear outrigger assembly. To produce minimum transit weight, removal of items 2, 7, 9, 11, 13, and 15 will reduce the front bogie load by 23,995 lb (10,884 kg) and the rear bogie load by 7,825 lb (3,549 kg). This will yield a transit weight of 90,645 lb (41,116 kg), or a reduction of about 1/3 from the original weight. Further reduction is possible by detaching (*undecking*) the *superstructure* and moving that component separately. Some of the larger machines are specially designed for quick undecking.

5.3 Traveling Within the Site

Haul roads to remote sites and internal construction site roads are often unpaved, poorly compacted, rutted, and badly graded.[1] A travel path should be defined and surveyed before attempting it; corrective earthwork might be needed. *Rough-terrain cranes* or *all-terrain cranes* are inherently capable of negotiating uneven and soft ground. A crawler crane with a short boom may also manage, but the boom must be positioned at a low angle to avoid bouncing backward into the boom stops. A crane mounted on a truck carrier is not designed for irregular terrain or abrupt grade changes; off-road travel surfaces must be well prepared. Existing ground might be improved by compaction or by firming up with crushed stone, gravel, or even brickbats. Alternatively, the existing ground could be overlaid with timber mats or steel plates.

Travel on steep grades can be a problematic for mobile cranes, especially for truck cranes and crawlers, and it should be studied during planning. The means of negotiating grade changes needs to be assessed, too. Whether going up or down a grade, the CG of the crane moves to either the rear or front and may affect stability fore and aft. But, stability can be controlled by adjusting boom angle and using the weight distribution methods given in this chapter. For some components it may be necessary to estimate CG height, as this information is not furnished in the crane data.

Axle weight data provided by the manufacturer (such as in Figure 5.1) are for a level crane; they will be affected by grades. But from that data, the horizontal location of the CG of the crane (less the boom) can be determined and the CG height estimated component by component and then combined for the whole. Stability on grades can be determined with reasonable accuracy by applying the horizontal and vertical CG positions.

---

[1]The American Institute of Steel Construction (AISC) *Code of Standard Practice for Steel Buildings and Bridges* (3/18/05) stipulates that it is the responsibility of the "Owner's Designated Representative for Construction" to provide "adequate access roads into and through the job site for the safe . . . movement . . . of cranes" for the benefit of the erector.

Crane travel with a load on the hook is a special operation that requires a high standard of ground evenness and levelness. Avoidance of side slope and side wind is of particular importance.

Whenever possible a long-boom truck crane should be assembled in the spot where it is to be operated, but that is not always possible. Driving a crane burdened with a long boom or an extensive *front-end attachment* is similar to traveling with a load under hook, including sensitivity to side slope and side wind. With all components intact, axle loads are optimized by varying the boom angle and orientation. Optimization does not necessarily equate to balancing the bogie loadings; many truck cranes are built with more axles supporting the rear than the front and the optimum loading will be proportioned to axle capacities. Generally front axles are favored so that the steering mechanism is not overburdened. In some instances, counterweights must be removed for travel to protect axles or tires from overload. When in doubt the manufacturer should be consulted.

Boom angle adjustments are used to shift the load between the front and rear bogies as needed. Example 5.1 below has as a basis the assumption that the optimum distribution is equal. An equal balance might not be possible with a short boom; with a long boom it is usually achieved with the boom pointed over the front of the carrier. Travel path preparation needs to take that into account so axle loadings for a fully assembled crane may be quite high (Figure 5.2).

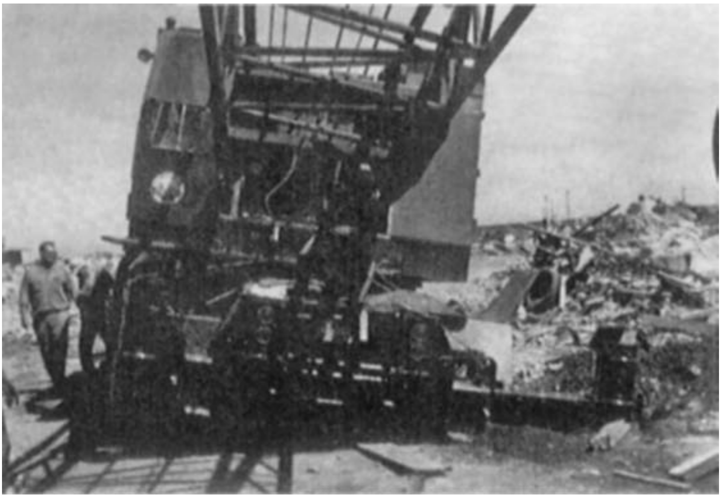


FIGURE 5.2 The soft access road could not support the wheel loads imposed by this crane. An attempt to right the crane by using the outriggers also failed because the load was not spread to a large enough bearing area.

Space is often at a premium, particularly at urban sites, this makes assembly a challenge. Latticed booms and luffing jibs require a generally level stretch of ground somewhat longer than the member to be assembled. Consideration should be given to the space required for the trucks delivering the sections and perhaps also for an assist crane to assemble boom and counterweights. Should power lines or similar dangers be present at or adjacent to the site, proper distances must be allowed for both the assist and main cranes.

Example 5.1

The crane of Figure 5.1 is to be operated with 180 ft (54.9 m) of boom. It will be equipped with hydraulic outriggers, *front bumper counterweight*, and boom backstops. What boom angle must be maintained for the crane to travel with balanced axle loads while fully assembled? What will be the axle loading? The boom (including guy lines and spreader) weighs 17,100 lb (7756 kg), and the boom CG is located 85 ft (25.9 m) from the boom foot pin measured along the boom centerline.

Solution

For the boom over the front of the carrier,

	Item No.	Front Bogie, lb	Rear Bogie, lb
Basic crane	1A	24,178	112,887
Add boom backstop	17	+1,075	+350
Add bumper counterweight	25	+19,000	–5,000
Remove basic boom	4	–12,800	+5,660
Total crane less boom		31,453 (14,267 kg)	113,897 (51,663 kg)

For balanced axle loads, each bogie must carry (including boom weight)

$$(31{,}453 + 113{,}897 + 17{,}000)\,(1/2) = 81{,}225 \text{ lb } (36{,}843 \text{ kg})$$

This requires that the front bogie loading be increased by

$$81{,}225 - 31{,}453 = 49{,}772 \text{ lb } (22{,}576)$$

as its share of the boom effect.

Crane dimensions are given in the diagram in Figure 5.1. The distance from the axis of rotation to the centerline of the rear bogie is 42 in (106.7 cm) and from boom foot pin to axis of rotation is 50 in (127.0 cm); the wheelbase is 19 ft 2 in = 19.17 ft (5.84 m). Taking moments about the rear bogie, we have

$$49{,}772 \text{ lb} = 17{,}100 \text{ lb}\,\frac{(42 \text{ in} + 50 \text{ in})/(12 \text{ in/ft}) + (85 \text{ ft})(\cos\theta)}{19.17 \text{ ft}}$$

$$= 892.02\,(7.67 + 85\cos\theta)$$

$$\cos\theta = 0.5662 \qquad \theta = 55.5^\circ \text{ boom angle}$$

For a two-axle bogie, axle loads will be

$$1/2\ (81{,}225) = 40{,}613 \text{ lb } (18{,}422 \text{ kg})$$

The result can be checked by taking moments about the front bogie.

$$\frac{17{,}100\,(85 \cos 55.5° + 7.67 - 19.17)}{19.17} = 32{,}688 \text{ lb } (14{,}827 \text{ kg})$$

This should equal the required reduction in rear bogie loading of 113,897 − 81,225 = 32,672 lb (14,820 kg), which it very closely does.

With a narrow travel base and considerable give to the tires, a truck crane is sensitive to transverse out of levelness and wind. Moving one equipped with a long boom, an operator often extends the outriggers to add an element of security. The floats are lowered so that they skim about 2 in (50 mm) clear of the ground surface, and the swing is locked to prevent it from inadvertently rotating. The crane must travel at a very slow speed on a firm well-graded travel path with watchers posted at each outrigger float. These precautions, if properly executed, assure that a gust of wind or a soft spot in the travel path will not cause the crane to turn over. But forward motion must be stopped quickly when a float touches the ground.

Impact loading will be a minor effect when traveling is done at a creeping speed (as it should be) especially if a smooth travel-path surface is prepared. Nonetheless, when analyzing a structure supporting the traveling crane, the calculated axle loads should be increased by perhaps 15% to allow for less-than-perfect balance as well as for wind, impact, acceleration, and braking. Needless to say, a move of a long-boom crane should be postponed when the wind is up, not only for the effect on axle loads but more importantly out of concern for sideways stability.

An assembled crawler crane is designed for traveling on-site, but it cannot do so unrestricted. As with a truck crane, out of levelness in the transverse direction can be problematic, though less so. With its wide stance, instability is not a threat, but *side load* from out of levelness can overstress the boom and jib. A greater concern is instability fore and aft caused by deficient ground support. The tracks must always have firm support, particularly at the heavy end, which is usually the rear end under the counterweights. Heavy ground pressure at the rear end is advantageous for travel; when the leading edges of the tracks are lightly loaded, they advance without digging into the soil. Turning, the tracks slide more easily over the surface without pushing along much soil. A machine so powerful that it can manage travel up a 30% grade would not be expected to have difficulty maneuvering a turn on heavily loaded tracks, but it can happen. Steering places greater strain and on crawler components than any other motion and draws more power. Watching a crawler crane move about a site is instructive.

When a crawler crane enters a sharp downgrade or overtops an upgrade, there is potential for sudden uncontrolled movement. As the machine approaches the transition, the leading edge of the crane's rigid base will start to project out over open space. As the crane CG passes the abrupt change, the leading edge will drop to close the gap and the crane will lurch.

To avoid damage, the crane should be stopped before the transition and be crept slowly forward over this critical point. Alternatively, after the crane is stopped just short of the breakpoint, the operator can slowly lower the boom, which will gradually and smoothly bring the CG over the transition. A travel surface with a smooth gradual changeover is preferable to a breakpoint, but not always feasible.

Example 5.2

The truck crane of Figure 5.1, which was the subject of Example 5.1, will be traveling down a 20% grade with a 180 ft (54.9 m) boom over the front. Before entering the downgrade, what angle should the boom be positioned at to obtain equal axle loading while on the grade?

Solution

A 20% downgrade is a slope in which the drop is 20 units for each 100 horizontal units; in other words it is an angle whose tangent is 0.2, or 11.3°. Figure 5.1 furnishes data which can be used to find the horizontal position of the CG of the crane without its boom. Noting that the wheelbase, or distance between bogie centers, is 19.17 ft (5.84 m), measured from the rear bogie center, that horizontal distance C is

$$C = \frac{31{,}453 \text{ lb} \times 19.17 \text{ ft}}{31{,}453 + 113{,}879} = 4.15 \text{ ft } (1.26 \text{ m})$$

Figure 5.1 gives the distance from the rear bogie to the boom foot as E plus G, or 92 in which is 7.67 ft (2.34 m), and that figure must be added to the boom CG horizontal position given in Example 5.1.

The CG height B_h for the boom at angle θ to the horizontal (before entering the slope), taking account of the boom foot height of 6.94 ft (2.12 m), is

$$B_h = 6.94 + 85 \sin\theta$$

The height of the CG of the crane (less the boom) can be calculated with some accuracy by taking component by component and using an estimated CG height for each. However, let us assume that for this crane the figure is 5.0 ft (1.52 m), and note that these calculations are not very sensitive to CG height estimation errors. We now have sufficient information to balance axle loads by taking moments about the rear bogie center with the crane on an 11.3° downslope.

$$\begin{aligned} 0.5 \times 19.17 \cos 11.3 = \{&145{,}350\,(4.15 \cos 11.3 + 5.0 \sin 11.3) \\ &+ 17{,}100\,[(7.67 + 85 \cos\theta) \cos 11.3 \\ &+ (6.94 + 85 \sin\theta) \sin 11.3]\}/(145{,}350 + 17{,}100) \end{aligned}$$

This simplifies to

$$1.0 = 2.22 \cos\theta + 0.44 \sin\theta$$

Solving by iteration gives 75° which is the boom angle before the crane enters the slope that will produce uniform axle loadings on the slope. While on the slope, the boom will be at 75 – 11.3 = 63.7° to the horizontal.

Had the height of the CG for crane less boom been estimated at 4 ft (1.22 m) instead of 5 ft (1.52 m), the boom angle would come to 74°. This result supports the assertion that the calculation is not very sensitive to the CG height estimation for the crane absent the boom.

5.4 Clearances

With each work cycle, a crane picks up a load, swings it and places it where needed. The path of both load and boom must clear obstructions through the full range of movement. The boom must be long enough to raise the load to the required height without danger of collision between the load and the boom, but not so long that it will swing into nearby structures. The tail end of the superstructure requires a clear swing path, too. And, after completing the work, the crane must be able to extricate itself and lower its boom for dismantling. Far from all mobile-crane lifts run against these constraints, but they are common enough to demand attention.

It is embarrassing, to say the least, to send a crane to a job only to find that it is incapable of placing the loads where needed. Embarrassment may be only the first problem in a series, as some accidents come about because field crews try to improvise and work around an unexpected limitation.

There are numerous ways that a crane might come up short. Shortness of *reach* or *lifting capacity* is an obvious example. This is best averted by diligent review of loads, radii, and crane capabilities. But there are less obvious shortcomings just as debilitating to an operation as the straightforward ones that might be ferreted out only by careful study with the benefit of an experienced eye. The more subtle shortcomings often are interferences between the moving crane or load and other objects. These are referred to as *clearance deficiencies*.

Clearance problems can come up in various ways.

- The lifted load is at risk of fouling the boom or jib. A wide load lifted close to the *head machinery* at a high boom angle could be at risk. This is sometimes a dilemma faced by riggers placing unwieldy vessels or machinery. Several styles of boom heads such as the *hammerhead tip* or *offset tip* are configured to lessen the possibility of fouling. The *head sheaves* of telescopic booms are often similarly offset, too.
- The suspended load or the boom tip fouls a nearby building or other obstruction. In tight quarters, the boom might not be capable of being raised high enough to swing past the obstruction. Urban and industrial sites can be places of heightened exposure to this peril.

- The boom head is obstructed from above by an overhanging structure or object.
- Reaching some distance past the leading edge or parapet of a building, the underside of the boom or jib cannot clear it. The authors refer to this as a *swing clearance problem.*
- The hook cannot reach sufficient height to place the load. This is known as a *drift deficiency*. It is sometimes caused by an underestimation of the length slings or height of the rigging supporting the load.
- The aft end of the crane superstructure cannot clear an obstruction. The counterweights, for instance, interfere with a tree or a live mast fouls a building.
- Any part of the crane or the suspended load encroaches within a restricted zone surrounding a power line.

Basic clearance checking can be done using the *range diagram* (Figure 5.3) provided with the crane documentation, a copy of which is often also mounted in the crane cab. For most work, the rough data obtainable with a range diagram are sufficient to verify that the job can be done and that boom and jib lengths and jib offset will offer satisfactory clearance. A clearance problem is shown in Figure 5.4 where crane reach is obstructed by the building in front of it. When conditions appear to be too close to permit reliance on the diagram, calculations or a more comprehensive graphical approach might be used. In many instances, a conventional CAD drawing, accurately dimensioned and detailed, is a suitable tool for clearance checking (Figure 5.5).

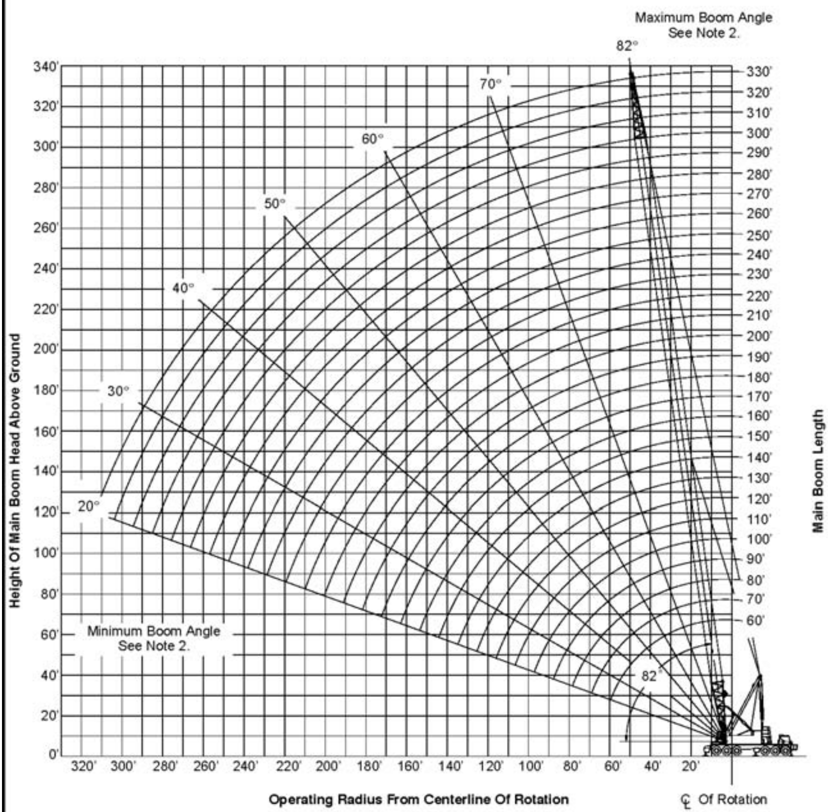
These problems can be difficult to visualize with two-dimensional tools even for experienced planners. As this manuscript is being prepared, technological aids to assist lift planners are in existence but with limited utility. Building Information Modeling (BIM), in particular, is promising, but presently the lead time and costs needed to generate a useful model make sense only for a select few projects. No doubt some of its limitations will disappear over time, and other advanced planning aids will also come into being. Until the improbable arrival of the day when a planner will be able to don virtual reality goggles and walk through a comprehensive 3-D model of a crane set up on the site, it will be necessary to hone visualization skills and make the best possible use of more rudimentary analytical tools such as those that follow. And no matter what mathematical model is used, obviously it is only useful to the extent that it matches the actual field conditions.

Drift Clearance

A load aloft can be only brought as high as the maximum attainable hook elevation minus the height of the load and rigging. In addition, some wiggle space is needed for maneuvering. What is enough wiggle space allowance is a matter of some judgment. Ten ft (3 m) could be

Working Range Diagram

60' To 330' Main Boom On Outriggers



Notes:

1. **Boom geometry shown is for unloaded condition and crane standing level on firm supporting surface. Boom deflection, subsequent radius, and boom angle change must be accounted for when applying load to hook.**
2. **Maximum and minimum boom angles are equal to the values listed in the capacity chart for each boom length.**

Not for crane operations

HC–278HII 13 of 66 M2P0065

Figure 5.3 Typical range diagram for a latticed boom crane. Such diagrams are adequate for most height and swing clearance checks. (*Link Belt Construction Equipment Company.*)

FIGURE 5.4 On a city street, a telescopic crane with an offset jib places mechanical equipment on the roof of a building. Clearance checking is needed if successful placement is not to be left to chance. (*Mathieu Chaudanson.*)

an acceptable minimum for a carefully measured rigging job, but that figure might be doubled or tripled depending on the reliability of the planning information and the need for production speed.

Drift is defined as the vertical clearance between the top of a lifted load and the crane hook when the hook is in its highest possible position; it is a measure of the gap available for rigging and for maneuvering the load. The drift is a useful piece of information for the planner to make decisions about the rigging and handling of the load.

A load block cannot be permitted to contact the head sheaves. When this does happen, it is called *two-blocking*.

Older models, particularly *friction cranes*, rely entirely on the operator to avert two-blocking. A friction winch may be powerful

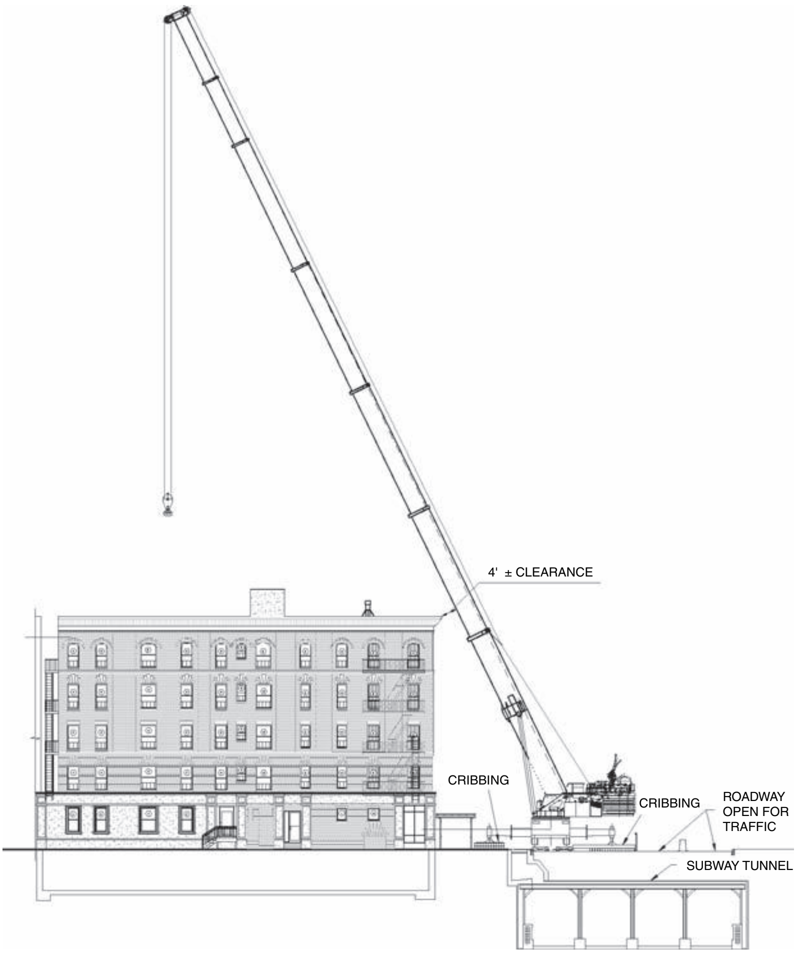


FIGURE 5.5 Elevation view of a crane lifting over a roof cornice. The clearance is verified graphically using a CAD program.

enough to break the hoist rope. This can happen, with resulting loss of the load, if the operator is not able to sense the change in engine tone as it starts to lug following two-blocking. With a light load on the hook, the winch can pull the boom back over the cab.

A contemporary mobile crane is required to have an *anti-two-block device*. However, even when such a device is installed and functioning, sometimes the inertia of a rapidly raised load block can overrun it. An operator should behave as if the device is not there, slowing down the lift speed as the block approaches the upper limit.

Moreover, when the block is raised high, there is more opportunity for it to swing into the boom and cause damage. This possibility is lessened with a *gooseneck boom*, when the boom has an *offset tip* or, on a latticed boom, an *open throat tip*.

The subtraction distance from boom point to hook can substantially reduce the effective lifting height. Sometimes this figure is listed in the crane documentation. When not given, an allowance should be made; anywhere from 4 ft (1.2 m) to 18 ft (4.6 m) is common. Unfortunately it is hard to pin down a suitable allowance without specific knowledge of the crane. Not only does it vary with boom size and style, but it also by the load-block characteristics.

Swing Clearances

A crane is often called on to pick or set a load at some distance beyond the face of a building. If it is a tower crane or a *luffing jib* mobile crane of sufficient height it can readily reach over such an obstruction (Figure 5.6).

Figure 5.6 A 300-ton Link Belt 348 HYLAB 5 with a luffing jib erecting a precast concrete parking garage. The boom tip is higher than the building, so the entire range of motion for the jib is in the clear. (*Link Belt Construction Equipment Company.*)

But a conventional boom, with its boom foot below the top of the building, can only reach in so far before the boom will hit its leading edge. Checking clearances between the bottom of the boom and that edge is a straightforward graphical or geometry problem with the boom is at right angles to the building. More often than not, however, the crane is either placed too close to the wall to permit it to swing perpendicular or a load must be placed at a location that does not put the boom at a right angle. With an acute horizontal angle between the boom and the wall, the clearance analyis becomes a complicated trigonometric problem. A BIM model or 3-D CAD may be helpful, but in the absence of these tools the mathematical derivations that follow offer a solution. With a spreadsheet program these formulas can be applied with relative ease. Visualization skills are most useful, nonetheless.

The formulas can be used to probe for solutions to the following questions, and others:

- How far can the hook reach beyond the face of the building?
- What is the optimum location for the crane to place a load at a particular spot over the building?
- When placing a load at a particular spot, how much clear distance will there be between the boom and a face of the building?
- What is the maximum permitted building height that can be cleared with the crane configuration and location?
- What minimum boom length is needed to place the load at a given spot over the building?
- What is the optimum jib length and *offset angle* to place a load at a particular spot on the building?

Assume a crane with boom of length L, width B, and depth D. In the notation of Figure 5.7, the operating radius will be

$$R = t + L\cos\theta \tag{5.1}$$

when boom angle θ is known. Conversely, when the radius is known

$$\theta = \cos^{-1}\frac{R-t}{L} \tag{5.2}$$

For telescopic cantilevered booms where the boom foot pin is behind the axis of rotation, t is taken as negative.

The horizontal angle φ the boom makes with the wall or other obstruction is given by

$$\varphi = \sin^{-1}\frac{g+e}{R} \tag{5.3}$$

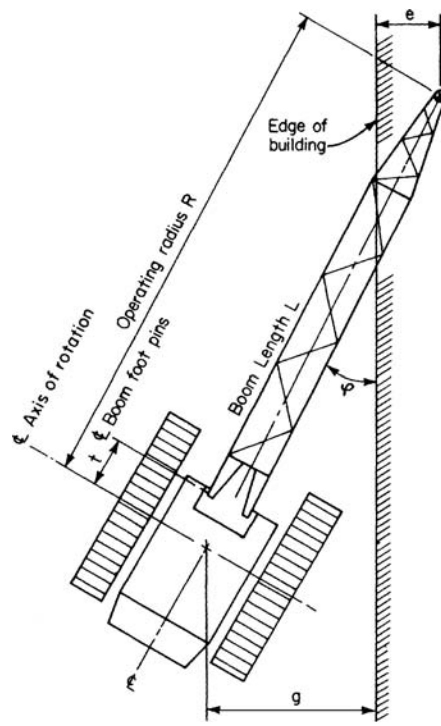


FIGURE 5.7 Plan view of a crawler crane showing the position parameters used for clearance calculations when the boom is at an angle to the wall.

The dimension *e*, it should be noted, represents the distance the load must be placed beyond the face of the wall. With $e = 0$, the load-clearance problem is one of drift, discussed previously. But as the boom passes over the leading edge of the wall, the lower corner of the boom that is closest to the wall must be kept sufficiently clear of it. If the boom is latticed, the chords can be in compression; a small lateral thrust from bumping the wall edge could cause it to collapse.

The angle that the clearance line makes with the horizontal (Figure 5.8) is defined as τ, where

$$\tau = \tan^{-1}\frac{g+e}{R\tan\theta} \tag{5.4}$$

Following the derivation in Appendix D, the clearance is determined to be

$$C = \frac{1}{A}\left[\left(R - t - \frac{H-t}{\tan\theta} - \frac{D}{2\sin\theta}\right)\sin\varphi - \frac{B}{2}\cos\varphi - e\right] \tag{5.5}$$

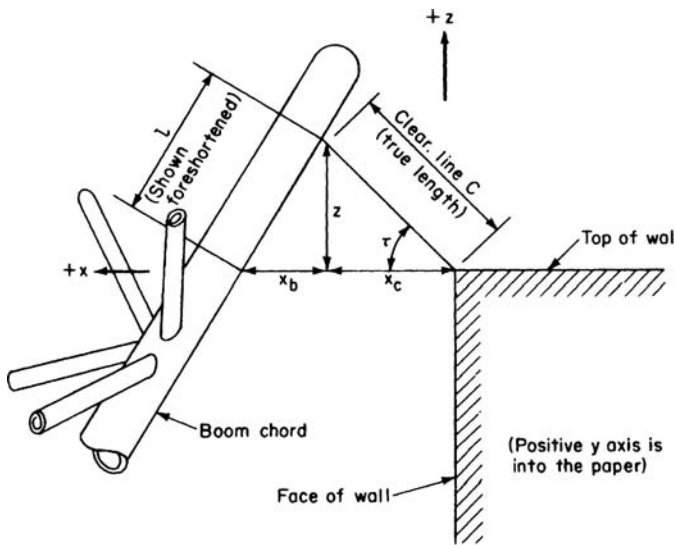


FIGURE 5.8 Elevation view parallel to the wall.

where

$$A = \frac{1+\sin^2 \tau}{\cos \tau} \tag{5.6}$$

Boom clearance C can be found by applying Eq. (5.1) to (5.6) with crane dimensions L, D, B, t, and h known. The site-specific parameters R, e, g, and H must also be given. Once the clearance is calculated, however, its acceptance is a matter of judgment with consideration of the field conditions.

Various factors might be considered before judging what is acceptable as a minimum clearance:

- Accuracy of the field data, as the analytical model is only as good as the geometric input.
- Accuracy of the field layout. A difference of a few feet in the crane position can sometimes make all the difference in the success or failure of a lift.
- Boom type and length affect deflection and movement. Working crane booms are never still, even in calm air. Long booms, especially telescopic booms can deflect and move about considerably.
- Boom deflection as the crane picks up a load. Deflection may reduce clearance.

- Wind speed. Particularly when a load has a large wind area, it may induce correspondingly large boom movements. If clearance is not generous, a lift might wisely be postponed until a calm day.
- Operator and rigging crew experience. Hazard recognition and responsiveness to signaling are essential if clearance is to be close.
- The need for maneuvering room. Setting a piece of machinery, for example, may call for adjustments and manipulation before the hook is set free.
- Visibility of the clearance point to the operator (Figure 5.9) or signalperson. Close clearances should not be contemplated if observation and control cannot be spot on. A signalperson should be close by the clearance point and in continuous contact with the operator.

Figure 5.9 The operator cannot judge clearance well from the cab. A well-placed signalperson with a dedicated radio link to the cab is advisable. (*Link Belt Construction Equipment Company.*)

As a rule of thumb, 2 ft (0.6 m) of clearance should prove adequate for short booms and 4 ft (1.2 m) for long booms. These values might be cut down to as little as half, but only with a signalperson in close proximity for monitoring clearance, little wind, and slow operating speeds.

Equation (5.5) can be manipulated to determine the maximum permissible service height. With all parameters but H given and a value for C preselected, that height is

$$H = h - \frac{D}{2\cos\theta} + \left(R - t - \frac{CA + e}{\sin\varphi} - \frac{B}{2\tan\varphi}\right)\tan\theta \tag{5.7}$$

Equation (5.7) is useful for the construction planning of tall structures. It indicates when a jib should be added or a longer boom used. The value of A is determined from Eq. (5.6).

Alternatively, for a given service height and minimum clearance, the minimum boom length can be ascertained by the formula

$$L = \frac{H - h}{\sin\theta} + \frac{D}{2\sin\theta\cos\theta} + \frac{(CA + e)/\sin\varphi + B/(2\tan\varphi)}{\cos\theta} \tag{5.8}$$

The boom angle θ, however, is unknown. With the preceding equations and a starting assumption for the boom length, an iterative analysis is performed to converge on a compatible pairing of boom length and vertical angle. In the final result, the boom length must be one that is available for the particular crane.

Swing Clearance with a Fixed Jib

Adding a jib may be a simple solution to a clearance problem, particularly if the boom point remains above the top of the obstruction. Jibs are used for increasing lift height. That height combined with a jib offset angle can provide the means to reach over an obstacle.

Taking a jib of length J, width b, and depth d that is mounted at an offset angle μ to the boom gives

$$R = t + L\cos\theta + \cos(\theta - \mu) \tag{5.9}$$

If the radius is given, the boom angle θ can be found by iterating

$$\frac{R - t}{L} = \cos\theta + \frac{J}{L}\cos(\theta - \mu) \tag{5.10}$$

until θ has been determined to a satisfactory accuracy (about 0.1° for long booms at high angles). Once θ is known, clearance calculations can proceed as for a boom alone because boom clearance will govern.

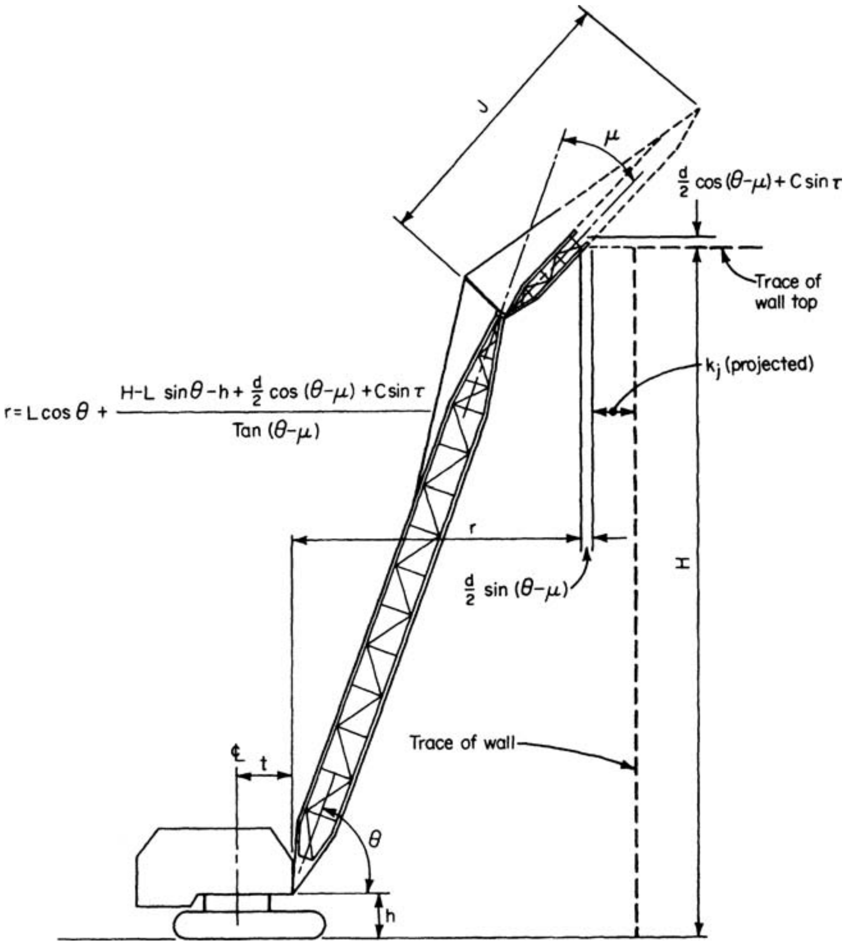


FIGURE 5.10 Elevation of a crane viewed normal to the boom.

When the boom point is below the obstruction height (Figure 5.10), the angle the jib clearance line C_J makes with the horizontal can be expressed as

$$\tau_j = \tan^{-1}\frac{\sin\varphi}{\tan(\theta-\mu)} \tag{5.11}$$

and the jib clearance to the leading building edge is

$$C = \frac{1}{A_j}\left[\left(R - t - L\cos\theta - \frac{H - L\sin\theta - h}{\tan(\theta-\mu)} - \frac{d}{2\sin(\theta-\mu)}\right)\sin\varphi - \frac{b}{2}\cos\varphi - e\right] \tag{5.12}$$

where

$$A_j = \frac{1 + \sin^2 \tau_j}{\cos \tau_j} \tag{5.13}$$

A full derivation of Eq. (5.12) is given in Appendix D.

When the boom tip is below the top of the obstruction, the clearance between the boom head and the building may need scrutiny. There is a treatment of this problem for latticed booms in Appendix D. But for telescopic booms, the geometry of the boom head and jib mounting is usually more complicated and cannot be generalized for all cranes. Where the boom head of a telescopic crane is anticipated to approach closely to a building face, a graphical or 3-D study may be preferred to a mathematical approach.

Acceptability of the jib clearance value is a matter of judgment as it is with the boom. Greater sway and exposure to wind, however, combined with reduced operator perception can create an elevated potential collision hazard that should be compensated by planning a generous clearance. An allowance not less than 4 ft (1.2 m) is advised unless a signalperson provides close monitoring.

Keeping in mind that the formula applies only when the boom point is below the leading edge of the building, Equation (5.12) can be manipulated to determine the maximum permissible service height for the jib:

$$H = h + L \sin \theta - \frac{d}{2 \cos (\theta - \mu)} + \left(R - t - L \cos \theta - \frac{CA_j + e}{\sin \varphi} - \frac{b}{2 \tan \varphi} \right) \tan (\theta - \mu) \tag{5.14}$$

The boom length, jib length, jib offset, clearance, crane position, and reach into the building can be manipulated to seek an optimum lift arrangement.

An optimal jib length can be selected using an iterative procedure with the expression

$$j = \frac{H - h - L \sin \theta}{\sin (\theta - \mu)} + \frac{d}{2 \cos (\theta - \mu) \sin (\theta - \mu)} + \frac{(CA_j + e)/(\sin \varphi) + b/(2 \tan \varphi)}{\cos (\theta - \mu)} \tag{5.15}$$

Starting with a trial jib length and offset angle μ, Eq. (5.10) can be iterated to determine the vertical boom angle θ. Using Eq. (5.15), the calculated value of J can be compared with the trial value. If the trial value is the next longer available jib length, the shortest acceptable jib has been found and the specified clearance will be maintained.

In most installations where clearance is a concern, the crane boom crosses over a single hurdle which is the height of the wall or structure to be cleared (Figure 5.5). More complex situations arise when buildings have multiple setbacks. Each setback has a corresponding value of g and e. Sometimes the multiple setbacks are not all parallel to one another.

With a little imagination, these methods can be applied to mobile cranes with various front-end attachments or to derricks, tower cranes, and other equipment. For instance, they were used to assure adequate clearance during the tower-crane boom erection shown in Figure 5.11.

Figure 5.11 Tower-crane erection and dismantling are rigging projects where many of the most critical operations depend on maintaining clearance between the erection crane boom and the load. Placement of this hydraulic truck crane was controlled by adjacent buildings and by space needed on the street to assemble the tower-crane boom at a reach within the crane's capacity. (*Paul Yuskevich.*)

Figure 5.12 A view looking down from a roof setback on the Empire State Building to a long-boom telescopic crane on the street below. Deflection can substantially affect the geometry on such machines. (*Mathieu Chaudanson.*)

A telescopic boom is a cantilevered beam that will deflect substantially under load (Figure 5.12). There is no reliable way for the user to estimate the deflection. Rating chart radii, however, take into account deflection under rated load. Though the clearance algorithm given here does not take deflection into account, it is conservative because the convex deflected shape of the actual boom will provide more clearance for a given radius than the straight line of the calculation.

Example 5.3

1. A crane is to be used for placing concrete on a high-rise structure. What maximum height H can be serviced while maintaining a 4-ft (1.2-m) clearance between the boom and the edge of the structure? Crane dimensions are $t = 5$ ft (1.52 m), $h = 6.75$ ft (2.06 m), $D = B = 7.83$ ft (2.39 m), and boom length $L = 320$ ft (97.53 m). Installation parameters are $R = 90$ ft (27.43 m) and $g = 25$ ft (7.62 m). The crane will be lifting a concrete bucket that will be discharged into a hopper. The point of discharge will be 3 ft (0.91 m) beyond the building edge.

Solution

From Eq. (5.2),

$$\theta = \cos^{-1}\frac{90-5}{320} = 74.60°$$

From Eq. (5.3),

$$\varphi = \sin^{-1}\frac{25+3}{90} = 18.13°$$

From Eq. (5.4),

$$\tau = \tan^{-1}\frac{\sin 18.13°}{\tan 74.60°} = 4.90°$$

The maximum height will then be given by Eq. (5.7) after finding A with Eq. (5.6).

$$\begin{aligned} A &= \frac{1+\sin^2 4.90°}{\cos 4.90°} \\ &= 1.011 \\ H &= 6.75 - \frac{7.83}{2\cos 74.60°} \\ &\quad + \left[90 - 5 - \frac{4(1.011)+3}{\sin 18.13°} - \frac{7.83}{2\tan 18.13°}\right]\tan 74.6 \\ &= 175.0 \text{ ft } (53.34 \text{ m}) \end{aligned}$$

2. If the crane is moved farther from the building face so that g becomes 40 ft (12.19 m), what will be the effect on H?

Solution

If the radius is held constant, Eq. (5.3) gives the new value of φ as 28.54° and Eq. (5.4) shows that τ changes to 7.50° and A becomes 1.026. With these values inserted into Eq. (5.7), $H = 220.5$ ft (67.20 m). The 15-ft (4.57-m) increase in g has led to an increase in H of 45.5 ft (13.87 m).

3. If g is increased to 87 ft (26.52 m), the boom will be perpendicular to the building face. What maximum work height can be reached before it becomes necessary to add a jib?

Solution

It is now given that $\varphi = 90°$, and from Eq. (5.4) τ has become 15.40° and $A = 1.110$. Using Eq. (5.7) once more, we get $H = 273.6$ ft (83.38 m).

4. With the boom at minimum radius of 60 ft (18.29 m), the boom angle is 80.10°. Keeping the boom perpendicular to the building, what maximum height can now be reached?

Solution

There is no change in φ, which remains 90°, but τ is now 9.90° and A = 1.045. Inserting values into Eq. (5.7) yields H = 258.0 ft (78.63 m), or a reduction in height from the previous trial.

Example 5.3 shows that with the radius held constant, the maximum work height increases with dimension g (Figure 5.5) until the optimum point is reached. The optimum will occur when the boom is perpendicular to the structure. However, for any angle of boom to structure, there is an optimum radius that will give maximum height. Considering all conditions, greatest height is achievable with minimum e, $\varphi = 90°$, and radius optimized, but changing the radius may not offer a significant increase in height. For the example, the optimum radius is 110 ft (33.53 m), but this gives a work height only 1.7 ft (0.52 m) greater than the 90-ft (27.43-m) radius used in parts 1 to 3.

Example 5.4

What boom length would be needed to place a load 26 ft (7.92 m) in from the edge of a roof 72 ft (21.94 m) high? The crane must be placed so that $g = 36$ ft (10.97 m) exactly. Crane dimensions are $t = 4$ ft (1.22 m), $h = 6.5$ ft (1.98 m), $D = 5$ ft (1.52 m), and $B = 5.5$ ft (1.68 m); a clearance of 4 ft (1.2 m) will be used.

Solution

From the data, $H = 72$ ft (21.94 m) and $e = 26$ ft (7.92 m). An educated guess can be made for the boom angle with the assumption that the boom will be perpendicular to the wall. The distance normal to the wall from boom foot to wall is 36 – 4 = 32 ft (9.75 m). If the vertical height from roof edge to boom centerline is estimated to be about 20 ft (6.1 m), the boom centerline above the roof edge will then be approximately 72 – 6.5 + 20 ≈ 85.5 ft (26.1 m) above the boom foot. The boom angle θ will then be

$$\theta \approx \tan^{-1}\frac{85.5}{32} \approx 70°$$

With the boom perpendicular to the wall, $R = 36 + 26 = 62$ ft (18.90 m). Then from Eq. (5.1),

$$62 = 4 + L\cos 70° \qquad L \approx 170 \text{ ft } (51.8 \text{ m})$$

Equation (5.4) will give the clearance-line angle.

$$\tau = \tan^{-1}\frac{1}{\tan 70°} = 20.0°$$

After A has been found with Eq. (5.6), Eq. (5.8) will give the first calculated value for boom length.

$$A = \frac{1 + \sin^2 20.0°}{\cos 20.0°} = 1.189$$

$$\begin{aligned} L &= \frac{72 - 6.5}{\sin 70°} + \frac{5}{2\sin 70° \ \cos 70°} \\ &\quad + \frac{(4 \times 1.189 + 26)(\sin 90°) + 5.5/(2\tan 90°)}{\cos 70°} \\ &= 167.4 \text{ ft } (51.02 \text{ m}) \end{aligned}$$

Therefore, 170 ft (51.8 m) was a good guess and is the boom length needed. To check, find the angles needed:

$$\theta = \cos^{-1}\frac{62 - 4}{170} = 70.1° \qquad \tau = \tan^{-1}\frac{1}{\tan 70.1°} = 19.90°$$

Then from Eq. (5.6),

$$A = \frac{1 + \sin^2 19.90^\circ}{\cos 19.90^\circ} = 1.187$$

This is inserted into Eq. (5.5) to find the clearance.

$$C = \frac{\left(62 - 4 - \dfrac{72 - 6.5}{\tan 70.1^\circ} - \dfrac{5}{2 \sin 70.1^\circ}\right) \sin 90^\circ - (5/2) \cos 90^\circ - 26}{1.187}$$

$$= 4.74 \text{ ft } (1.45 \text{ m})$$

The actual clearance is greater than the minimum specified. To be certain, let us try a 160-ft (48.8-m) boom. The boom angle becomes

$$\theta = \cos^{-1} \frac{62 - 4}{160} = 68.75 \qquad \varphi = 90^\circ \qquad \tau = 21.25^\circ \qquad A = 1.214$$

and C = 3.17 ft (0.97 m), which is less than allowed. The 170-ft (51.8-m) boom is the shortest that can perform the work and maintain the specified clearance.

Example 5.5

1. Given the crane and installation dimensions of part (1) of Example 5.3 with an 80-ft (24.4-m) jib added at a 10° offset angle, what maximum height can be *reach*ed? Jib dimensions are $d = b = 2.5$ ft (0.76 m), and C_j is to be taken as 4 ft (1.2 m). Boom tip dimensions are $b_t = d_t = 2.5$ ft (0.75 m), and the tapered tip section is 40 ft (12.2 m) long.

Solution

The boom angle is found by iterating Eq. (5.10).

$$\frac{90 - 5}{320} = \cos \theta + \frac{80}{320} \cos (\theta - 10^\circ)$$

$$0.266 = \cos \theta + 0.250 \cos (\theta - 10^\circ)$$

$$\theta = 79.7^\circ$$

The horizontal angle of the boom to the building φ remains unchanged from part (1) of Example 5.3 at 18.13°, but the clearance-line angle to the horizontal changes and is given by Eq. (5.11)

$$\tau_j = \tan^{-1} \frac{\sin 18.13^\circ}{\tan (79.7^\circ - 10^\circ)} = 6.57^\circ$$

The clearance parameter A_j, from Eq. (5.13), is

$$A_j = \frac{1 + \sin^2 6.57^\circ}{\cos 6.57^\circ} = 1.020$$

The maximum height achievable with jib mounted is found by using Eq. (5.14).

$$H = 6.75 + 320\sin 79.7^\circ - \frac{2.5}{2\cos 69.7^\circ} + \left[90 - 5 - 320\cos 79.7^\circ - \frac{4(1.020)+3}{\sin 18.13^\circ} - \frac{2.5}{2\tan 18.13^\circ}\right]\tan 69.7^\circ$$
$$= 321.3 \text{ ft } (97.92 \text{ m})$$

on the basis of jib-clearance criteria. Boom tip height = 6.75 + 320 sin 79.7° = 321.6 ft (98.02 m), which is just greater than H. Checking boom clearance, from Eq. (5.4) we find

$$\tau = \tan^{-1}\frac{\sin 18.13^\circ}{\tan 79.7^\circ} = 3.24^\circ$$

The boom tip position parameter e_t is given by Eq. (D.1).

$$e_t = 3 - (80\cos 69.7^\circ)\sin 18.13^\circ = -5.64 \text{ ft } (-1.72 \text{ m})$$

The clearance parameter A is given by Eq. (5.6).

$$A = \frac{1+\sin^2 3.24^\circ}{\cos 3.24^\circ} = 1.005$$

Equation (D.2) then gives the boom tip clearance.

$$C_t = \frac{5.64 - (2.5/2)\sin 79.7^\circ \sin 18.13^\circ - (2.5/2)\cos 18.13^\circ}{1.005}$$
$$= 4.05 \text{ ft } (1.23 \text{ m}) > 4 \text{ ft } (1.2 \text{ m}) \quad \text{OK}$$

The clearance at the start of the taper section is given by Eq. (D.3).

$$C_{st} = \frac{[40\cos 79.7^\circ - (7.83/2)\sin 79.7^\circ]\sin 18.13^\circ + 5.64 - (7.83/2)\cos 18.13^\circ}{1.005}$$
$$= 2.93 \text{ ft } (0.89 \text{ m}) < 4 \text{ ft } (1.2 \text{ m})$$

This is not acceptable. Clearance height will then be given by Eq. (5.7). The boom radius is 5 + 320 cos 79.7° = 62.2 ft (18.96 m) and

$$H = 6.75 - \frac{7.83}{2\cos 79.7^\circ} + \left[62.2 - 5 - \frac{4(1.005) - 5.64}{\sin 18.13^\circ} - \frac{7.83}{2\tan 18.13^\circ}\right]\tan 79.7^\circ$$
$$= 262.5 \text{ ft } (80.01 \text{ m})$$

Boom clearance governs so as to maintain the specified 4 ft (1.2 m) value. Adding the 80-ft (24.4-m) jib increased H by 87.5 ft (26.7 m).

2. Using the same radius, $R = 90$ ft (27.43 m), but with the boom set perpendicular to the building, what height can be achieved?

Solution

This position requires that $g = 87$ ft (26.52 m), and of course, $\varphi = 90°$. The clearance angle τ_j increases to

$$\tau_j = \tan^{-1}\frac{1}{\tan 69.7°} = 20.30°$$

$$A_j = \frac{1+\sin^2 20.30°}{\cos 20.30°} = 1.195$$

$$H = 6.75 + 320\sin 79.7° - \frac{2.5}{2\cos 69.7°} + \left[90 - 5 - 320\cos 79.7° - \frac{4(1.195)+3}{1} - \frac{2.5}{2\tan 20.30°}\right]\tan 69.7°$$

$$= 362.9 \text{ ft } (110.61 \text{ m})$$

The calculated maximum service height H must be approaching the jib point height, making clear the need to check lift clearance. The jib point height will be

$$\text{Point height} = 6.75 + 320\sin 79.7° + 80\sin 69.7°$$

$$= 396.6 \text{ ft } (120.88 \text{ m})$$

This means that only 396.6 – 362.9 = 33.7 ft (10.27 m) is available above the roof to accommodate the load clearance, load, slings and attachments, and the minimum distance from hook to jib point. This figure must be checked against the particular equipment to be used for the work in order to be certain that lifts to that height can actually be made. In a boom-clearance check, $R_t = 62.22$ ft (18.96 m) but $g = 87$ ft (26.52 m); therefore boom clearance is obviously okay.

Clearance in Tight Quarters

It is remarkable how a mobile crane, by its very nature a huge apparatus, is routinely shoehorned into snug urban or industrial settings. The means of getting the crane assembled and into a postage-stamp sized position is a subject unto itself. Once there and operating, clearance for both the front end and the back end is often tight and needs verification.

A large front-end attachment such as a boom with a luffing jib has a correspondingly large minimum working radius. While the boom or jib might have clear space to swing from the pickup location to the load setting zone, the minimum radius prevents loads from being placed close to the crane. A luffing jib might be fitted with a *mid-fall* hoist line to overcome this inadequacy.

The tail-swing radius may encroach over an active traffic lane or face interference from an obstruction. A large crane can have a rear-projecting gantry, strut, backstays, or a live mast capable of fouling trees, buildings, or electric power lines. Placement of the crane is often close to the face of a building that prevents the tail end from swinging through a full circle. Tail-swing of a live mast or guy struts varies with the boom angle; the full working range of boom angles needs to be considered.

The front-end attachment, be it a straight boom, a boom-jib combination, or another type, could also be prevented by buildings or other obstructions from swinging in a full circle. A crane that starts a project with a free swing might not have clear space by the time the erection work is completed. These limitations not only weigh on the operation but also need to be considered when the front-end attachment is to be stowed during nonworking hours or the crane is to be secured in advance of a storm.

Interference is sometimes caused by a secondary element that might easily be missed unless the planner takes a second look. A *cab*, slack cable, or transition piece might easily be overlooked. The clearance dimensions of projecting or overhanging components are usually given in general dimension diagrams provided with the crane documentation.

5.5 Crane Loads to the Supporting Surface

Reliable information for determining mobile crane support reactions has not always been readily available; rough conservative approximations have often been the norm. Now crane manufacturers routinely supply software and tables that can give the user accurate and comprehensive answers when the resources are properly used.

This section provides analytical methods for the user to calculate support reactions from basic crane data. Though tools provided by manufacturers will usually make calculations unnecessary, these methods are useful for the exceptions and also for those users who wish to have a deeper level of comprehension. The full body of data needed to perform the calculations is not ordinarily given in crane manufacturer's literature; it might be estimated or obtained by other means. The required information is

Carrier. Weight and CG horizontal distance from the axis of rotation

Superstructure. Weight, including counterweights, and CG horizontal distance from the axis of rotation

Boom. For each boom length (for both latticed and telescopic booms), weight and CG location coordinates, including the effects of appurtenances such as guy lines, upper spreader and mast

Jib. For each jib length, weight and CG location coordinates, including the effects of guy lines and jib mast

For boom and jib data, the CG locations should be given in terms of a distance along the centerline measured from the foot pin and an offset above and perpendicular to the centerline (Figure 5.13). It is

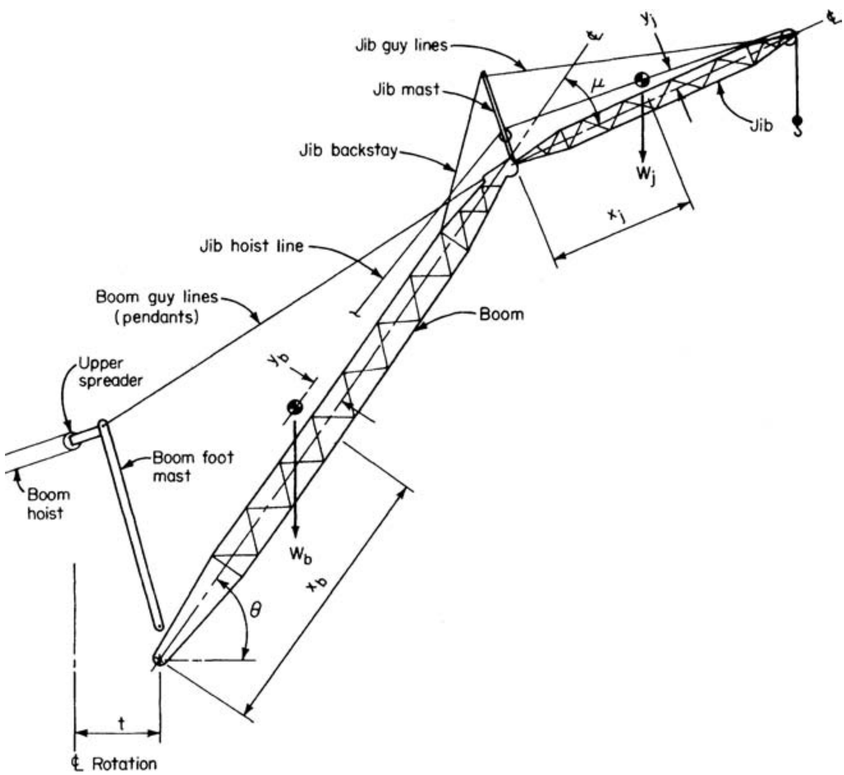


FIGURE 5.13 Elevation of a boom showing the parameters needed for calculating boom moment.

helpful to transform the boom and jib CG location data from cartesian to polar coordinate form.

$$\theta_b = \tan^{-1}\frac{y_b}{x_b} \qquad L_b = \left(y_b^2 + x_b^2\right)^{1/2}$$

$$\mu_j = \tan^{-1}\frac{y_j}{x_j} \quad \left[J_j\left(y_j^2 + x_j^2\right)^{1/2}\right]$$

where θ_b and L_b and μ_j and J_j define the position of the boom and jib CG, respectively. Polar data will enable expression of the moment of the boom about the axis of rotation as

$$M_b = W_b[t + L_b \cos(\theta + \theta_b)] \tag{5.16}$$

With a jib mounted, the moment becomes

$$M_{bj} = M_b + W_j[t + L\cos\theta + J_j \cos(\theta - \mu + \mu_j)] \tag{5.17}$$

The entire crane structure above the swing circle—the superstructure—can be represented mathematically by a moment and a vertical load. If the weight of the superstructure, less boom and jib weights W_b and W_j, is called W_u and its CG is located horizontally from the axis of rotation a distance d_u, then the moment for operating radius R, including the lifted load W and the weight of the suspended hoist ropes W_r, is

$$M_u = M_b + (W + W_r)R - W_u d_u$$

or

$$M_u = M_{bj} + (W + W_{rj})R - W_u d_u \tag{5.18}$$

when a jib is being used and the vertical load is given by

$$V_u = W_b + W + W_r + W_u$$

or

$$V_u = W_b + W_j + W + W_r + W_u \tag{5.19}$$

when a jib is being used. Both moments and vertical loads act at the axis of rotation.

Crane manufacturers prefer to segregate the weight of the counterweights from the balance of the superstructure weight. Correspondingly the $W_u d_u$ term of Eq. (5.18) and the W_u term of Eq. (5.19) can be replaced by two terms.

Truck-Crane Outrigger Loads

The common operating state for a truck crane is on outriggers. Though theoretically it is possible to have a machine with three outriggers, in practice the minimum is four. Some models have additional outriggers at the front or rear of the carrier to enhance the lifting capability in the end *quadrants of operation*.

The centroid of the outriggers should not be assumed to coincide with the projection of the axis of rotation onto the ground, although many cranes are arranged so. In the formulation that follows, the outrigger centroid is taken as being to the rear of the axis-of-rotation projection a distance x_0 along the longitudinal centerline of the crane (Figure 5.14). If the outrigger centroid is forward of the axis projection, x_0 is taken as a negative quantity.

The carrier weight is designated W_c with its CG located a distance d_c forward of the outrigger centroid and along the longitudinal axis. The net moment of all loads about the outrigger centroid for operations over the rear of the crane is

$$M_{nr} = M_u - V_u x_0 - W_c d_c$$

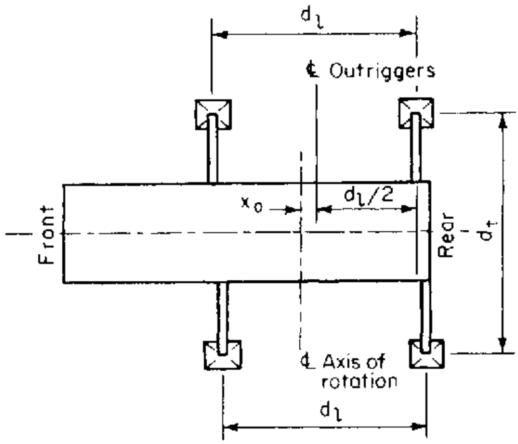


FIGURE 5.14 Plan view of a truck crane. Note how the outrigger beams are offset from one side to the other to permit retraction.

For operations over the side the net moment is

$$M_{ns} = M_u$$

In addition to the moments, the outriggers support the vertical load.

$$V = V_u + W_c \tag{5.20}$$

There are two distinct theories that can be applied for distributing moments and vertical loads to the individual outrigger floats and hence to the ground. Either is built on assumed behaviors to get around the static indeterminacy of a four-point support. The first considers the crane carrier and outriggers as inelastic and the ground support to be linear elastic; points representing outrigger reactions plotted on a three-dimensional graph will appear in a common plane.

The alternative theory follows a different useful fiction. The symmetry of the outrigger pattern is used to get around the static indeterminacy, with the center of force of the superstructure apportioned side to side and front to rear by simple ratio. For this theory to work, the frame must be supple to accommodate the assumptions. Results from both methods satisfy the requirements of equilibrium, but values will usually be different except when the outrigger pattern is a square.

Method One

This method distributes outrigger loads by rigid body theory. If the distances between outriggers (Figure 5.14) longitudinally and transversely are respectively d_l and d_t, when the crane is lifting over the rear, each outrigger will carry

$$P = \frac{V}{4} \pm \frac{M_{ur}}{2d_l}$$

The second term is added for the rear outriggers and subtracted for the front floats.

For operations over the side, the distribution front to rear is not typically equal because the *center of rotation* does not align with the centroid of the outriggers. For identification, the front outrigger loads on the boom side are designated P_{fb} and on the counterweight side as P_{fc}. Similarly, the rear outrigger loads are P_{rb} and P_{rc}. The full set of outrigger loads are then

$$P_{fb} = \frac{V}{4} + \frac{M_{ns}}{2d_t} + \frac{W_c d_c + V_u x_0}{2d_l}$$

$$P_{fc} = \frac{V}{4} - \frac{M_{ns}}{2d_t} + \frac{W_c d_c + V_u x_0}{2d_l}$$

$$P_{rb} = \frac{V}{4} + \frac{M_{ns}}{2d_t} - \frac{W_c d_c + V_u x_0}{2d_l}$$

$$P_{rc} = \frac{V}{4} - \frac{M_{ns}}{2d_t} - \frac{W_c d_c + V_u x_0}{2d_l}$$

A similar distribution pattern occurs during the more complex case of operations with the boom at an angle to the crane longitudinal centerline. With α as the horizontal angle between the boom and the longitudinal axis of the crane measured from the rear,

$$\begin{aligned} M_{nr} &= M_u \cos\alpha - V_u x_0 - W_c d_c \\ M_{ns} &= M_u \sin\alpha \end{aligned} \tag{5.21}$$

M_{nr} is the portion of the moment acting over the rear and M_{ns} is the portion over the side. When these moments are distributed to the individual outriggers together with vertical loads, the loads acting on the outriggers become

$$\begin{aligned} P_{fb} &= \frac{V}{4} + \frac{1}{2}\left(\frac{M_{ns}}{d_t} - \frac{M_{nr}}{d_l}\right) \\ P_{fc} &= \frac{V}{4} - \frac{1}{2}\left(\frac{M_{ns}}{d_t} + \frac{M_{nr}}{d_l}\right) \\ P_{rb} &= \frac{V}{4} + \frac{1}{2}\left(\frac{M_{ns}}{d_t} + \frac{M_{nr}}{d_l}\right) \\ P_{rc} &= \frac{V}{4} - \frac{1}{2}\left(\frac{M_{ns}}{d_t} - \frac{M_{nr}}{d_l}\right) \end{aligned} \tag{5.22}$$

Equations (5.21) and (5.22) make up a general solution that gives the outrigger loads for any boom orientation, including over the rear and over the side.

When handling a load near the full capacity over a corner, it is not unusual for the outrigger opposite the boom to be load-free or even to lift off the ground. The mathematical expression of this occurrence is a zero or negative value for one of the outrigger loads in Eq. (5.22). The physical meaning of the minus sign is that the ground must pull down on the float, which, of course, it cannot do. The reactions must be rebalanced to give zero reaction at the negative float, thus matching the mathematical with the physical condition. If two outrigger floats have negative values, the crane is shown to be unstable and will tip over.

Rebalancing is done by adjusting the reactions without changing the moment or vertical load equilibrium. If a negative reaction has absolute value a, it is zeroed by adding a. Concurrently, the value a is added to the reaction diagonally across from the heretofore negative outrigger float, and the same quantity is subtracted from each of the remaining two outrigger reactions. If P_{rc} had a calculated value of $-a$, set $P_{rc} = 0$, add a to P_{fb}, and subtract a from both P_{rb} and P_{fc}. The result is that there has been no change in the total of the vertical reactions or in outrigger moments about the centroid.

The sum of all outrigger reactions must equal V. Likewise, the sum of all moments about the crane longitudinal centerline and about any transverse line must both be zero if the system is to be in equilibrium. These are the check conditions that must be satisfied by any set of outrigger reactions.

Actual outrigger reactions will not coincide exactly with calculated values based on the rigid body theory. Carriers and outrigger assemblies are not truly rigid; actual behavior will result in part from spring-like elastic response. Neither wind nor dynamic effects have been considered in the calculations. In consideration, cribbing or other supports should never be designed with a thin margin against failure. Cribbing is not only furnished to prevent situations as shown in Figure 5.15, but to keep the crane level and free of rocking motions. Ground support failure under a loaded crane is often catastrophic. The small cost of placing adequate structural materials under the outrigger floats is money well spent. Outrigger floats on soil or fill will almost invariably need cribbing to reduce the ground bearing pressure.

The reader may have noticed that the method presented does not account for the offset of outrigger beams from one side to the other in the fore and aft direction. If dimensions used reflect the centers of the pairs of outrigger beams, the offsets will have an insignificant effect on the calculated values. However, where support under the outriggers requires that the floats be spotted exactly, the offsets can be very important. This situation can arise where the outriggers are to bear directly on structural framing or must clear an obstacle on the ground.

Several loading cases need to be considered in order to discover governing values for the design of supports. Each load should be checked at its maximum radius and at one or more swing angles from

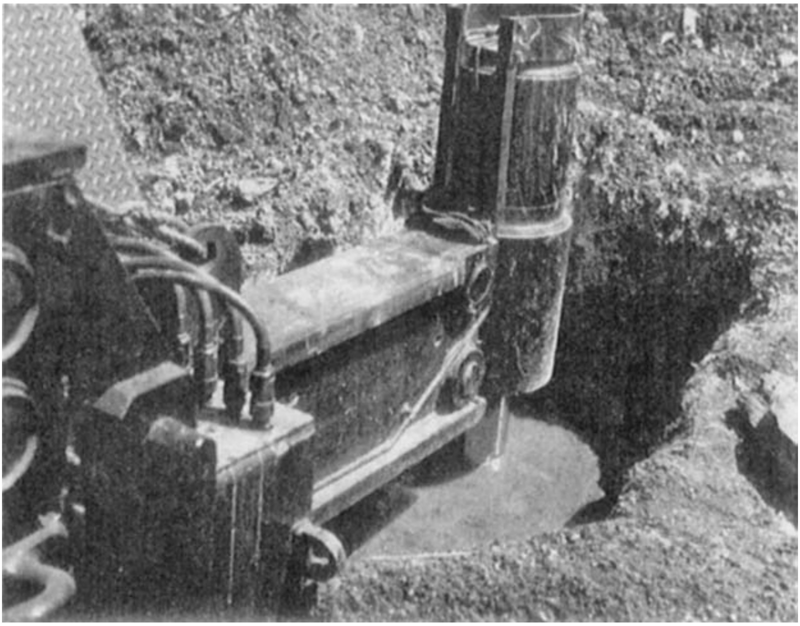


FIGURE 5.15 This photograph demonstrates that on poor ground it becomes essential for outrigger floats to be placed on timber or steel cribbing support that will spread the load to a larger bearing area.

picking to placing the load.[2] When several loads are to be handled, the operation at the maximum working radius and the condition causing the maximum load moment should be evaluated.[3] Other cases to check include the minimum radius with no load on the hook, as counterweights may induce the controlling outrigger reactions. Erection of a long boom flat over the rear could also be the condition that causes maximum loading on the rear outriggers.

Front Outriggers

On some crane models an outrigger is placed at the front (Figure 5.16) to permit operation in that sector. Although a five-support pattern is statically indeterminate, an approximate value for the front outrigger reaction can be found. If it is assumed that the fifth outrigger will remain essentially unloaded when the crane is stable on four outriggers, it follows that the fifth outrigger will act only when the rear outriggers become unloaded. This will be a condition of three-point

[2]Maximum reaction on a single outrigger may occur when the boom is swung to an angle perpendicular to the axis drawn between outrigger centers at opposing corners.

[3]Load moment is hook load times radius.

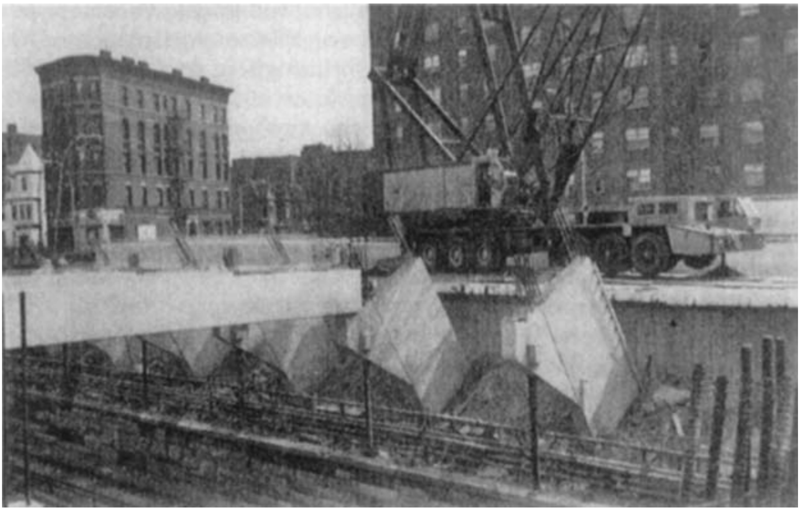


FIGURE 5.16 The front outrigger is needed in order to set the next beam. When used, it extends the over-the-side operating zone for this machine toward the front to the line from the axis of rotation through the front outrigger—the line on which this photograph was taken.

support that can readily be solved by taking moments. Because of frame elasticity, the assumption should yield reasonable results for a front outrigger initially set without excessive preload.

Example 5.6

A truck crane will be lifting 35 kips (15.9 t) (including hook block and slings) over the rear at 50-ft (15.24-m) radius, swinging full over the side and then booming out to 70 ft (21.33 m) to place the load. Find the maximum reaction at each outrigger during this operation. Crane data are as follows:

$$t = 5.0 \text{ ft } (1.52 \text{ m}) \qquad x_0 = 0.58 \text{ ft } (177 \text{ m})$$

$$d_t = 24 \text{ ft } (7.31 \text{ m}) \qquad d_l = 22 \text{ ft } (6.71 \text{ m})$$

$$\theta(50) = 72.5° \qquad \theta(70) = 64.3°$$

$$W_r = 0.35 \text{ kip } (160 \text{ kg}) \qquad W_b = 27 \text{ kips } (12.52 \text{ t})$$

Load block and sling weight = 3 kips (1.36 t)

Boom length = 150 ft (45.7 m)

$L_b = 68.0$ ft (20.73 m) $\qquad \theta_b = 5.0°$

	Weight		CG Location to Axis of Rotation		
Component	**kips**	**t**	**ft**	**m**	
Superstructure	90	40.8	6.0	1.83	toward counterweights
Counterweights	94	42.6	17.5	5.33	toward counterweights
Carrier	126	57.2	3.67	1.12	toward front or from
			4.25	1.30	outrigger centroid

Solution

Since the requirement is to find maximum loading at each outrigger, six combinations of position and condition must be checked.

	$W + W_r$		Radius		
Position	**kips**	**t**	**ft**	**m**	**Horizontal Angle x**
Over rear	3.35	1.52	50	15.24	0°
Over rear	35.35	16.0	50	15.24	0°
Over corner	3.35	1.52	50	15.24	$\tan^{-1}(24/22) = 47.49°$
Over corner	35.35	16.0	50	15.24	$\tan^{-1}(24/22) = 47.49°$
Over side	3.35	1.52	50	15.24	90°
Over side	35.35	16.0	70	21.33	90°

For the unloaded case over side, 50-ft (15.24-m) radius was chosen because 70 ft (21.33 m) will control for the loaded case with the boom over a corner. At 50 ft it is possible that the outrigger under the counterweight will have the maximum loading. As there is no way to tell what radius the operator will choose for the return; the more critical should be assumed unless the operator is to be given specific instructions.

The boom moment, given by Eq. (5.16), is needed for both operating radii.

$$M_b(50) = 27[5 + 68\cos(72.5° + 5.0°)]$$
$$= 532.4 \text{ kip}\cdot\text{ft } (721.8 \text{ kN}\cdot\text{m})$$
$$M_b(70) = 27[5 + 68\cos(64.3° + 5.0°)]$$
$$= 784.0 \text{ kip}\cdot\text{ft } (1062.9 \text{ kN}\cdot\text{m})$$

The superstructure moment is needed for three combinations, both loaded and unloaded at 50-ft (15.24-m) radius and loaded at 70 ft (21.33 m). Using Eq. (5.18),

$$M_u(50, 3.35) = 532.4 + 3.35(50) - 90(6.0) - 94(17.5)$$
$$= -1485.1 \text{ kip} \times \text{ft } (-2013.5 \text{ kN} \times \text{m})$$
$$M_u(50, 35.35) = -1485.1 + (35 - 3)(50)$$
$$= 114.9 \text{ kip} \times \text{ft } (155.8 \text{ kN} \times \text{m})$$
$$M_u(70, 35.35) = 784.0 + 35.35(70) - 90(6.0) - 94(17.5)$$
$$= 1073.5 \text{ kip} \times \text{ft } (1455.5 \text{ kN} \times \text{m})$$

Equation (5.1*a*) gives the superstructure vertical load that is needed for both the loaded and unloaded cases.

$$V_u(3.35) = 27 + 3.0 + 0.35 + 90 + 94 = 214.35 \text{ kips } (953.5 \text{ kN})$$
$$V_u(35.35) = 214.35 + (35 - 3) = 246.35 \text{ kips } (1095.8 \text{ kN})$$

The total vertical loads include carrier weight, Eq. (5.20), and for the unloaded and loaded conditions.

$$V(3.35) = 214.35 + 126 = 340.35 \text{ kips } (1513.9 \text{ kN})$$
$$V(35.35) = 246.35 + 126 = 372.35 \text{ kips } (1656.3 \text{ kN})$$

With these preliminary data prepared, each combination of position and condition can be explored. For the over-the-rear position, from Eq. (5.21) it is seen that $M_{ns} = 0$, while for the unloaded condition.

$$M_{nr}(50, 3.35, \text{rear}) = -1485.1 - 214.35(0.58) - 126(4.25)$$
$$= -2144.9 \text{ kip}\cdot\text{ft } (2908.1 \text{kN}\cdot\text{m})$$

The individual outrigger loads are then found by using Eq. (5.22).

$$\frac{V}{4} = \frac{340.35}{4} = 85.1 \text{ kips } (378.5 \text{ kN})$$
$$\frac{M_{nr}}{d_l} = \frac{-2144.9}{22} = -97.5 \text{ kips } (-433.7 \text{kN})$$
$$P_{fb} = P_{fc} = 85.1 + (1/2)(97.5) = 133.8 \text{ kips } (595.2 \text{ kN})$$
$$P_{rb} = P_{rc} = 85.1 - (1/2)(97.5) = 36.3 \text{ kips } (161.5 \text{ kN})$$

In the same position, for the loaded condition,

$$M_{nr}(50, 35.35, \text{ rear}) = 114.9 - 246.35(0.58) - 126(4.25)$$
$$= -563.5 \text{ kip} \times \text{ft } (-764.0 \text{ kN}\cdot\text{m})$$

$$\frac{V}{4} = \frac{372.35}{4} = 93.1 \text{ kips } (414.1 \text{ kN})$$
$$\frac{M_{nr}}{d_l} = \frac{-563.5}{22} = -25.6 \text{ kips } (-113.9 \text{ kN})$$
$$P_{fb} = P_{fc} = 93.1 + (1/2)(25.6) = 105.9 \text{ kips } (471.1 \text{ kN})$$
$$P_{rb} = P_{rc} = 93.1 - (1/2)(25.6) = 80.3 \text{ kips } (357.2 \text{ kN})$$

For the over corner position, when unloaded, Eq. (5.26) provides

$$M_{nr}(50, 3.35, \text{ corner}) = -1485.1 \cos 47.49° - 214.35(0.58) - 126(4.25)$$
$$= -1663.3 \text{ kip}\cdot\text{ft } (-2255.2 \text{ kN}\cdot\text{m})$$
$$M_{ns}(50, 3.35, \text{ corner}) = -1485.1 \sin 47.49°$$
$$= -1094.8 \text{ kip}\cdot\text{ft } (-1484.3 \text{ kN}\cdot\text{m})$$

$$\frac{V}{4} = \frac{340.35}{4} = 85.1 \text{ kips } (378.5 \text{ kN})$$
$$\frac{M_{nr}}{d_l} = \frac{-1663.3}{22} = -75.6 \text{ kips } (-336.3 \text{ kN})$$
$$\frac{M_{ns}}{d_l} = \frac{-1094.8}{24} = -45.6 \text{ kips } (-202.9 \text{ kN})$$

$$P_{fb} = 85.1 + (1/2)(-45.6 + 75.6) = 100.1 \text{ kips } (445.3 \text{ kN})$$

$$P_{fc} = 85.1 - (1/2)(-45.6 - 75.6) = 145.7 \text{ kips } (648.1 \text{ kN})$$

$$P_{rb} = 85.1 + (1/2)(-45.6 - 75.6) = 24.5 \text{ kips } (109.0 \text{ kN})$$

$$P_{rc} = 85.1 + (1/2)(-45.6 + 75.6) = 70.1 \text{ kips } (311.8 \text{ kN})$$

With the load applied,

$$M_{nr}(50, 35.35, \text{ corner}) = 114.9\cos 47.49^\circ - 246.35(0.58) - 126(4.25)$$

$$= -600.7 \text{ kip} \cdot \text{ft } (-814.4 \text{ kN} \cdot \text{m})$$

$$M_{ns}(50, 35.35, \text{ corner}) = 114.9 \sin 47.49^\circ$$

$$= 84.7 \text{ kip} \cdot \text{ft } (114.8 \text{ kN} \times \text{m})$$

$$\frac{V}{4} = \frac{372.35}{4} = 93.1 \text{ kips } (414.1 \text{ kN})$$

$$\frac{M_{nr}}{d_l} = \frac{-600.7}{22} = -27.3 \text{ kips } (-121.4 \text{ kN})$$

$$\frac{M_{ns}}{d_l} = \frac{84.7}{24} = 3.5 \text{ kips } (15.6 \text{ kN})$$

$$P_{fb} = 93.1 + (1/2)(3.5 + 27.3) = 108.5 \text{ kips } (482.6 \text{ kN})$$

$$P_{fc} = 93.1 - (1/2)(3.5 - 27.3) = 105.0 \text{ kips } (467.1 \text{ kN})$$

$$P_{rb} = 93.1 + (1/2)(3.5 - 27.3) = 81.2 \text{ kips } (361.2 \text{ kN})$$

$$P_{rc} = 93.1 - (1/2)(3.5 + 27.3) = 77.7 \text{ kips } (345.6 \text{ kN})$$

When operating over the side $\alpha = 90°$ and for the unloaded condition at 50-ft (15.24-m) radius.

$$M_{nr}(50, 3.35 \text{ side}) = -214.35(0.58) - 126(4.25)$$

$$= -659.8 \text{ kip} \cdot \text{ft } (-894.6 \text{ kN} \cdot \text{m})$$

$$M_{ns}(50, 3.35 \text{ side}) = -1485.1 \text{ kip} \cdot \text{ft } (-2013.5 \text{ kN} \cdot \text{m})$$

$$\frac{V}{4} = \frac{340.35}{4} = 85.1 \text{ kips } (378.5 \text{ kN})$$

$$\frac{M_{nr}}{d_l} = \frac{-659.8}{22} = -30.0 \text{ kips } (-133.4 \text{ kN})$$

$$\frac{M_{ns}}{d_l} = \frac{-1485.1}{24} = -61.9 \text{ kips } (-275.3 \text{ kN})$$

$$p_{fb} = 85.1 + (1/2)(-61.9 + 30.0) = 69.2 \text{ kips } (-275.3 \text{ kN})$$

$$P_{fc} = 85.1 - (1/2)(-61.9 - 30.0) = 131.1 \text{ kips } (583.2 \text{ kN})$$

$$P_{rb} = 85.1 + (1/2)(-61.9 - 30.0) = 39.2 \text{ kips } (174.4 \text{ kN})$$

$$P_{rc} = 85.1 - (1/2)(-61.9 + 30.0) = 101.1 \text{ kips } (449.7 \text{ kN})$$

The final condition has the machine loaded at 70-ft (21.33-m) radius.

$$M_{nr}(70, 35.35, \text{side}) = -246.35(0.58) - 126(4.25)$$
$$= -678.4 \text{ kip}\cdot\text{ft } (-919.8 \text{ kN} \times \text{m})$$
$$M_{ns}(70, 35.35, \text{side}) = 1073.5 \text{ kip}\cdot\text{ft } (1455.5 \text{ kN} \times \text{m})$$

$$\frac{V}{4} = \frac{372.35}{4} = 93.1 \text{ kips } (414.1 \text{ kN})$$
$$\frac{M_{nr}}{d_l} = \frac{-678.4}{22} = -30.8 \text{ kips } (-137.0 \text{ kN})$$
$$\frac{M_{ns}}{d_t} = \frac{1073.5}{24} = 44.7 \text{ kips } (198.8 \text{ kN})$$

$$P_{fb} = 93.1 + (1/2)(44.7 + 30.8) = 130.9 \text{ kips } (582.3 \text{ kN})$$
$$P_{fc} = 93.1 - (1/2)(44.7 - 30.8) = 86.2 \text{ kips } (383.4 \text{ kN})$$
$$P_{rb} = 93.1 - (1/2)(44.7 - 30.8) = 100.1 \text{ kips } (445.3 \text{ kN})$$
$$P_{rc} = 93.1 - (1/2)(44.7 + 30.8) = 55.4 \text{ kips } (246.4 \text{ kN})$$

Summarizing the outrigger load,

Position	Condition, ft	P_{fb}	P_{fc}	P_{rb}	P_{rc}
Over rear	Unloaded, 50	133.8	133.8	36.3	36.3
Over rear	Loaded, 50	105.9	105.9	80.3	80.3
Over corner	Unloaded, 50	100.1	145.7	24.5	70.1
Over corner	Loaded, 50	108.5	105.0	81.2	77.7
Over side	Unloaded, 50	69.2	131.1	39.2	101.1
Over side	Loaded, 70	130.9	86.2	100.1	55.4
Maxima, kips		133.8	145.7	100.1	101.1

Method Two

This method treats the crane carrier frame and outriggers as independent beams and the imposed concentrated vertical load is projected to the supports accordingly. Using parameters from the first method and Eq. 5.26, the net moment acting on the crane

$$M_{\text{net}} = \left(M_{nr}^2 + M_{ns}^2\right)^{1/2} \tag{5.23}$$

The resulting eccentricity of total vertical load V is

$$e = \frac{M_{\text{net}}}{V}$$

at angle β measured from the rear with respect to the longitudinal centerline.

$$\begin{aligned}
&\beta = \tan^{-1}\frac{M_{ns}}{M_{nr}} && M_{nr} > 0 \\
&\beta = 180^\circ + \tan^{-1}\frac{M_{ns}}{M_{nr}} && M_{nr} < 0 \\
&\beta = \alpha && M_{nr} = 0
\end{aligned} \tag{5.24}$$

The position ratios for the vertical load resultant are then

$$\begin{aligned}
e_l &= \frac{e}{d_l}\cos\beta = \frac{M_{\text{net}}}{V \times d_l}\cos\beta \\
e_l &= \frac{e}{d_l}\sin\beta = \frac{M_{\text{net}}}{V \times d_l}\sin\beta
\end{aligned} \tag{5.25}$$

A general expression can be formulated to reflect the portion of vertical load distributed to each outrigger float.

$$\begin{aligned}
P &= V\left(\frac{d_1/2 \pm e\cos\beta}{d_1}\right)\left(\frac{d_1/2 \pm e\sin\beta}{d_1}\right) \\
&= V\left(\frac{1}{2} + e_l\right)\left(\frac{1}{2} + e_t\right)
\end{aligned}$$

using the same outrigger float notation used for Method One.

$$\begin{aligned}
P_{rb} &= V\left(\frac{1}{2} + e_l\right)\left(\frac{1}{2} + e_t\right) \\
P_{re} &= V\left(\frac{1}{2} + e_l\right)\left(\frac{1}{2} - e_t\right) \\
P_{fb} &= V\left(\frac{1}{2} - e_l\right)\left(\frac{1}{2} + e_t\right) \\
P_{fc} &= V\left(\frac{1}{2} - e_l\right)\left(\frac{1}{2} - e_t\right)
\end{aligned} \tag{5.26}$$

Example 5.7

1. Using the crane and data of Example 5.6, find the outrigger reactions when the unloaded boom is over a rear corner at 50 ft. (15.2 m). Use Method Two.

Solution

Using values already found in Ex. 5.6,

$$V(3.35) = 340.35 \text{ kips } (1513.9 \text{ kN})$$

$$M_{nr}(50{,}3.35, \text{ corner}) = -1663.3 \text{ kip-ft } (-2255.2 \text{ kN-m})$$

$$M_{ns}(50{,}3.35, \text{ corner}) = -1094.8 \text{ kip-ft } (-1484.3 \text{ kN-m})$$

$$d_t = 24 \text{ ft } (7.31 \text{ m}) \qquad d_l = 22 \text{ ft } (6.71 \text{ m})$$

From Eq. (5.23),

$$M_{\text{net}} = (1663.3^2 + 1094.8^2)^{1/2}$$
$$= 1991.3 \text{ kip-ft } (2699.8 \text{ kN-m})$$

From Eq. (5.24),

$$M_{nr} < 0$$
$$\therefore\ \beta = 180° + \tan^{-1}(-1094.8/-1663.3) = 213.35°$$

The vertical load resultant position ratios given by Eq. (5.30) are

$$e_1 = \frac{1991.3}{340.35 \times 22} \cos 213.35 = -0.222$$

$$e_t = \frac{1991.3}{340.35 \times 24} \sin 213.35 = -0.134$$

Now there is sufficient data to calculate outrigger reactions using Eq. (5.26)

$$P_{rb} = 340.35(1/2 - 0.222)(1/2 - 0.134) = 34.6 \text{ kips } (153.9 \text{ kN})$$
$$P_{rc} = 340.35(1/2 - 0.222)(1/2 + 0.134) = 60.0 \text{ kips } (266.9 \text{ kN})$$
$$P_{fb} = 340.35(1/2 + 0.222)(1/2 - 0.134) = 89.9 \text{ kips } (399.9 \text{ kN})$$
$$P_{fc} = 340.35(1/2 + 0.222)(1/2 + 0.134) = 155.8 \text{ kips } (693.0 \text{ kN})$$

Comparing these results with those of Method One,

	Method one	**Method two**
P_{rb}	24.5 kips (109. kN)	34.6 kips (153.9 kN)
P_{rc}	70.1 kips (311.8 kN)	60.0 kips (266.9 kN)
P_{fb}	100.1 kips (445.3 kN)	89.9 kips (399.9 kN)
P_{fc}	145.7 kips (648.1 kN)	155.8 kips (693.0 kN)

2. Using Method Two, find outrigger reactions when the crane lifts a 35-kip (15.9-kg) load over the rear at 50-ft (15.2-m) radius.

Solution

Using some results from Ex. 5.6,

$$M_{nr}(50{,}35.35,\ \text{rear}) = -563.5 \text{ kip} \cdot \text{ft } (-764.0 \text{ kN} \cdot \text{m})$$
$$M_{ns}(50{,}35.35,\ \text{rear}) = 0\ V = 372.35 \text{ kips } (1656.3 \text{ kN})$$

Then

$$M_{\text{net}} = M_{nr}$$
$$\beta = 0$$
$$e = -563.5/372.35 = -1.51 \qquad e_1 = -1.51/22 = -0.069$$
$$P_{rb} = P_{rc} = 372.35(1/2 - 0.069) \times 1/2 = 80.3 \text{ kips } (357.2 \text{ kN})$$
$$P_{fb} = P_{fc} = 372.35(1/2 + 0.069) \times 1/2 = 105.9 \text{ kips } (471.1 \text{ kN})$$

These are exactly the same results found with Method One because there is no load eccentricity with respect to the rear.

3. Using Method Two, find outrigger reactions for the same load as in part 2, but over the side at 70-ft (21.3-m) radius.

Solution
From Eq. 5.6,

$$V(35.35) = 372.35 \text{ kips } (1656.3 \text{ kN})$$
$$M_{nr}(70, 35.35, \text{ side}) = -678.4 \text{ kip}\cdot\text{ft } (-919.8 \text{ kN}\cdot\text{m})$$
$$M_{ns}(70, 35.35, \text{ side}) = 1073.5 \text{ kip}\cdot\text{ft } (1455.5 \text{ kN}\cdot\text{m})$$

From Eq. (5.23),

$$M_{\text{net}} = (1073.5^2 + 678.4^2)^{1/2} = 1269.9 \text{ kip}\cdot\text{ft } (1721.8 \text{ kN}\cdot\text{m})$$

From Eq. (5.24),

$$M_{nr} < 0$$
$$\beta = 180 + \tan^{-1}(1073.5/-678.4) = 122.29^\circ$$

From Eq. (5.25),

$$e_l = (1269.9/372.35 \times 22) \cos 122.29 = -0.083$$
$$e_t = (1269.9/372.35 \times 24) \sin 122.29 = 0.120$$

From Eq. (5.26),

$$P_{rb} = 372.35(1/2 - 0.083)(1/2 + 0.120) = 96.3 \text{ kips } (428.4 \text{ kN})$$
$$P_{rc} = 372.35(1/2 - 0.083)(1/2 - 0.120) = 59.0 \text{ kips } (262.5 \text{ kN})$$
$$P_{fb} = 372.35(1/2 + 0.083)(1/2 + 0.120) = 134.6 \text{ kips } (598.7 \text{ kN})$$
$$P_{fc} = 372.35(1/2 + 0.083)(1/2 - 0.120) = 82.5 \text{ kips } (367.0 \text{ kN})$$

Compare these results with those of Method One.

	Method One	Method Two
P_{rb}	100.1 kips (445.3 kN)	96.3 kips (428.4 kN)
P_{rc}	55.4 kips (246.4 kN)	59.0 kips (262.5 kN)
P_{fb}	130.9 kips (582.3 kN)	134.6 kips (598.7 kN)
P_{fc}	86.2 kips (383.4 kN)	82.5 kips (367.0 kN)

Which method furnishes consistently accurate results? Method One is probably more accurate in most instances, but actual results will depend on the stiffness characteristics of the individual crane model and the support surface. True results will likely lie between the values calculated by the two methods. But, if outrigger supports are properly designed, either method will prove satisfactory.

Crawler-Crane Track Pressures

Crawler tracks are an efficient medium for distributing mobile-crane loads to a wide area of the ground. Just as a working truck crane may safely have extreme loading on one outrigger and zero loading on another, crawler tracks sometimes have highly skewed loading patterns.

The axis of rotation of a crawler crane usually coincides with the centroid of the track bearing surfaces, but on older machines the axis is often to the rear of the centroid.[4] The axis of rotation is at distance x_0 (Figure 5.17) to the rear of the bearing center. Let d_l be the effective bearing length of the tracks, w the width of the tracks, and d_t the center-to-center transverse distance between the tracks.

Between the drive sprocket and the front *idler sprocket* are a series of smaller-diameter wheels called *track rollers*. Crane weight and moment effects pass from the crawler *side frames* to the track rollers, thence to the *track shoes*, and finally to the support surface or the ground. Track shoes are the individual segmented plates that make up the crawler tracks. Both front and rear sprockets are usually raised slightly; therefore it is the track rollers that define the length of bearing. A few models have special load charts for lifting over an end with the end blocked. For some crawler cranes, it is necessary to block the front of the tracks in order to raise a long boom or boom-jib combination. The bearing length in such case should be extended at least to the sprocket center.

When cranes operate on yielding soils, the tracks may press into the ground creating the appearance of a sprocket-to-sprocket bearing length. But in making track pressure calculations, it is not prudent to make use of that expanded length. Supports designed using the area defined by the track rollers will be more certain to keep the crane level.

[4] The rear is defined as the end containing the *drive sprockets*.

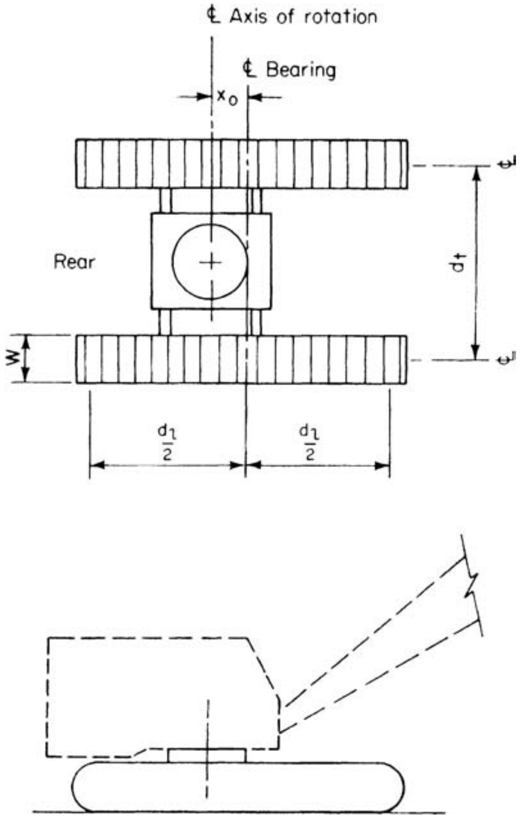


FIGURE 5.17 Annotated views of a crawler crane.

With the boom at a horizontal working angle α from the longitudinal centerline, measured from the front, the net moment applied at the center of bearing will be

$$M_{nf} = M_u \cos\alpha - V_u x_0 - W_c d_c \tag{5.27}$$

over the front, assuming that the undercarriage CG is at distance d_c behind the bearing centroid, and

$$M_{ns} = M_u \sin\alpha \tag{5.28}$$

over the side. The total vertical load is

$$V = V_u + W_c \tag{5.29}$$

If the crane were perfectly balanced with respect to the centroid of the track-bearing surfaces, $M_{nf} = M_{ns} = 0$, the load would be equally shared between the tracks and uniformly distributed along their lengths. Each track would carry $V/2$. But, if $M_{ns} \neq 0$, the distribution of load between the tracks cannot be equal. The difference in track loading must produce a ground reaction moment equal and opposite to M_{ns}.

Taking the reaction under the more heavily loaded track as R_h and under the more lightly loaded track as R_l,

$$V = R_h + R_l$$

The difference between R_h and R_l is caused solely by M_{ns}, this gives rise to the expressions

$$R_h = \frac{V}{2} + \frac{M_{ns}}{d_t}$$
$$R_t = \frac{V}{2} - \frac{M_{ns}}{d_t} \qquad (5.30)$$

The reactions (resultants) of Eq. (5.35) satisfy the equilibrium requirements $\Sigma V = 0$ and $\Sigma M_{\text{side}} = 0$. What remains is to take into account the effects of the moment M_{nf} over the front.

The front moment controls the longitudinal position of the resultants of the track reactions R_h and R_l. When $M_{nf} = 0$, there is no displacement; the resultants of track pressure are at the center of bearing of each track, and each track experiences uniform pressure along its length. For nonzero values of front moment, the reactions are displaced.

$$e = \frac{M_{nf}}{V} \qquad (5.31)$$

Because of eccentricity e, the track pressure diagram will take either a trapezoidal or a triangular shape (Figure 5.18).

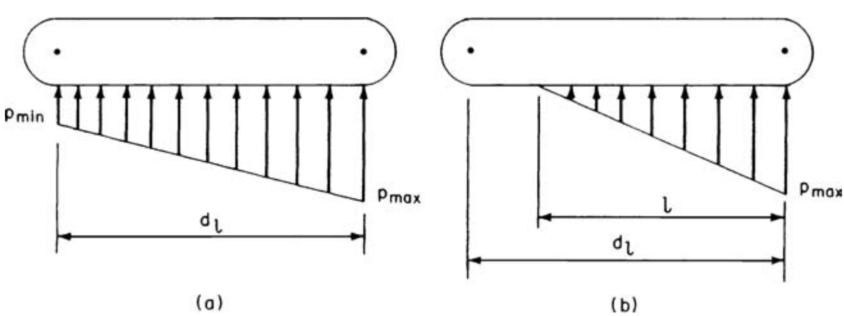


FIGURE 5.18 Crawler-crane track pressure diagram forms; (a) Trapezoidal. (b) Triangular.

The length l of the triangular pressure diagram is found by solving equilibrium equations for vertical load and front moment. For either track,

$$R = \frac{(p_{\text{max}})wl}{2}$$

$$e \cdot R = \frac{(p_{\text{max}})wl}{2}\left(\frac{d_l}{2} - \frac{l}{3}\right)$$

This yields

$$l = 3\left(\frac{d_l}{2 - e}\right) \qquad l \le d_l \tag{5.32}$$

For the triangular pressure diagram, it follows that the maximum pressure is

$$p_{\text{max}} = \frac{2R}{wl} \tag{5.33}$$

where R represents either R_h or R_l, depending on the track being studied.

When $l > d_l$, this indicates that the pressure diagram is trapezoidal (Figure 5.18*a*). The difference between the pressure at the ends of the track comes about because of the front moment. Using the equilibrium condition $\Sigma M_{\text{end}} = 0$, the pressures at the ends of the tracks are

$$p = \frac{R}{wd_l}\left(1 \pm \frac{6e}{d_l}\right) \tag{5.34}$$

This analysis assumes that the crawler frames and *carbody* are absolutely rigid—not an unreasonable assumption for most machines. But, because of the rigidity assumption both tracks will have a common value for e and therefore will always have the same shape of pressure diagram, that is, both triangular or both trapezoidal. Actually, because of elastic effects, e will be the average displacement of the two tracks and small variations from the calculated pressures will be imposed on the ground.

Figure 5.19 shows a large crawler crane placed above subway ventilation gratings. The support structure was designed using the method just described.

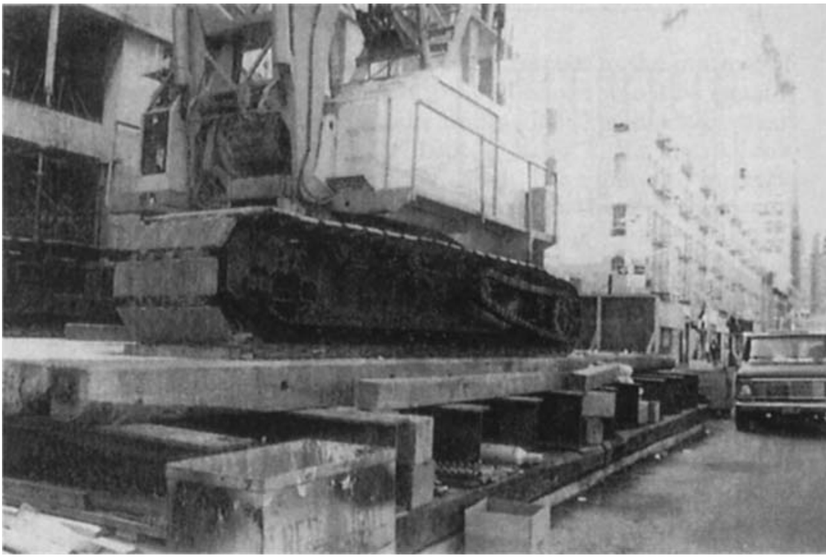


Figure 5.19 Traffic restrictions dictated that this crawler crane be positioned above subway ventilation gratings. The steel beams maintain open areas for the vents and span from the building foundation wall to a spread footing on the street surface well beyond the vents. This footing is composed of heavy timber mats to spread the beam loads and a thin concrete course as a leveling medium. (*XLO Concrete Corp.*)

Example 5.8

1. Find the pressures under the tracks of a crawler crane operating over the side while picking up a load, including hook block and slings, of 227.3 kips (103.1 t). The load is on the main hook at 24 ft (7.3-m) radius, but the 130-ft (39.6-m) boom has a 50-ft (15.2-m) jib mounted at 0° offset. Crane data are

$W_u = 206.88$ kips (93.84 t)	$d_u = 8.46$ ft (2.58 m)
$W_c = 129.31$ kips (58.65 t)	$d_c = 0$
$W_b = 26.46$ kips (12.00 t)	$L_b = 60.04$ ft (18.30 m)
	$\theta_b = 2.15°$
$W_j = 1.50$ kips (680 kg)	$J_j = 19.10$ ft (5.82 m)
	$\mu_j = 6.01°$
$W_r = 1.60$ kips (726 kg)	
$w = 4.0$ ft (1.22 m)	$d_l = 24.0$ ft (7.31 m)
$d_t = 17.08$ ft (5.21 m)	$x_0 = 0$ $\quad t = 4.0$ ft (1.22 m)

Solution

Equation (5.1) is used to find the boom angle.

$$24 = 4 + 130 \cos\theta \qquad \theta = 81.15°$$

The boom moment is given by Eq. (5.21).

$$M_b = 26.46\,[4 + 60.04 \cos(81.15° + 2.15°)]$$
$$= 291.2 \text{ kip}\cdot\text{ft } (394.8 \text{ kN}\cdot\text{m})$$

The moment with jib weight included is then, from Eq. (5.17),

$$M_{bj} = 291.2 + 1.5[4 + 130 \cos 81.15° + 19.10 \cos(81.15° + 6.01°)]$$
$$= 328.6 \text{ kip}\cdot\text{ft } (445.5 \text{ kN}\cdot\text{m})$$

Equation (5.18) is used to calculate superstructure moment.

$$M_u = 328.6 + (227.3 + 1.6)24 - 206.88(8.46)$$
$$= 4072.0 \text{ kip}\cdot\text{ft } (550.9 \text{ kN}\cdot\text{m})$$

The vertical load, from Eq. (5.19), is

$$V_u = 24.46 + 1.5 + 227.3 + 1.6 + 206.88$$
$$= 463.7 \text{ kips } (210.3 \text{ t})$$

The moments over the front and side are given by Eqs. (5.27) and (5.28) with $\alpha = 90°$.

$$M_{nf} = 0$$
$$M_{ns} = 4072.0 \text{ kip}\cdot\text{ft } (5520.9 \text{ kN}\cdot\text{m})$$

From Eq. (5.29), the total machine vertical load is

$$V = 463.7 + 129.3 = 593.0 \text{ kips } (269.0 \text{ t})$$

Equation (5.31) tells us that $e = 0$, and from Eq. (5.32),

$$l = 3(24.0/2 - 0) = 36.0 \text{ ft } (10.97 \text{ m})$$

$l > d_l$; therefore the pressure diagram is trapezoidal. Using Eq. (5.30)

$$R_l = \left(\frac{593.0}{2} - \frac{4072.0}{17.08}\right) = 58.1 \text{ kips } (26.3 \text{ t})$$

$$R_h = \left(\frac{593.0}{2} + \frac{4072.0}{17.08}\right) = 534.9 \text{ kips } (242.6 \text{ t})$$

Equation (5.34) is used to find track pressures. With $e = 0$, pressure will be uniform along each track $(p_{\max} = p_{\min} = p)$. For the counterweight side track,

$$p = \frac{58.1}{4.0 \times 24.0}\left(1 \pm \frac{6 \times 0}{24.0}\right) = 0.61 \text{ kips/ft}^2 \text{ } (29.2 \text{ kN/m}^2)$$

For the boom side track,

$$p = \frac{534.9}{4.0 \times 24.0} = 5.57\ \text{kips/ft}^2\ (266.7\ \text{kN/m}^2)$$

The pressures calculated are for the loaded case only, as asked.

2. Find the track pressures when the horizontal operating angle $\alpha = 30°$.

Solution

Rotation of the superstructure causes a redistribution of upper moments and of track loadings. From Eqs. (5.27) and (5.28),

$$M_{nf} = 4072.0 \cos 30° = 3526.5\ \text{kip}\cdot\text{ft}\ (4781.3\ \text{kN}\cdot\text{m})$$

$$M_{ns} = 4072.0 \sin 30° = 2036.0\ \text{kip}\cdot\text{ft}\ (2760.5\ \text{kN}\cdot\text{m})$$

Using Eq. (5.30),

$$R_t = \frac{593.0}{2} - \frac{2036.0}{17.08}$$
$$= 296.5 - 119.2 = 177.3\ \text{kips}\ (80.4\ \text{t})$$
$$R_h = 296.5 + 119.2 = 415.7\ \text{kips}\ (188.6\ \text{t})$$

From Eqs. (5.31) and (5.32),

$$e = \frac{3526.5}{593.0} = 5.95\ \text{ft}\ (1.81\ \text{m})$$

$$l = 3\left(\frac{24.0}{2} - 5.95\right) = 18.16\ \text{ft}\ (5.53\ \text{m})$$

$l < d_l$; therefore the pressure diagram is triangular. For the more lightly loaded counterweight side track, using Eq. (5.33),

$$p_{\max} = \frac{2 \times 177.3}{18.16 \times 4.0}$$
$$= 4.88\ \text{kips/ft}^2\ (233.9\ \text{kN/m}^2)$$

For the boom side track,

$$p_{\max} = \frac{2 \times 415.7}{18.16 \times 4.0}$$
$$= 11.45\ \text{kips/ft}^2\ (548.3\ \text{kN/m}^2)$$

The pressure diagrams for parts 1 and 2 are illustrated in Figure 5.20.

Rough-Terrain Cranes Outrigger Loads

Outrigger loadings for rough-terrain cranes could be calculated using exactly the same procedures outlined for truck cranes. But the data for telescopic booms is more complicated and difficult to use. For most operations, such elaborate procedures are not called for, however.

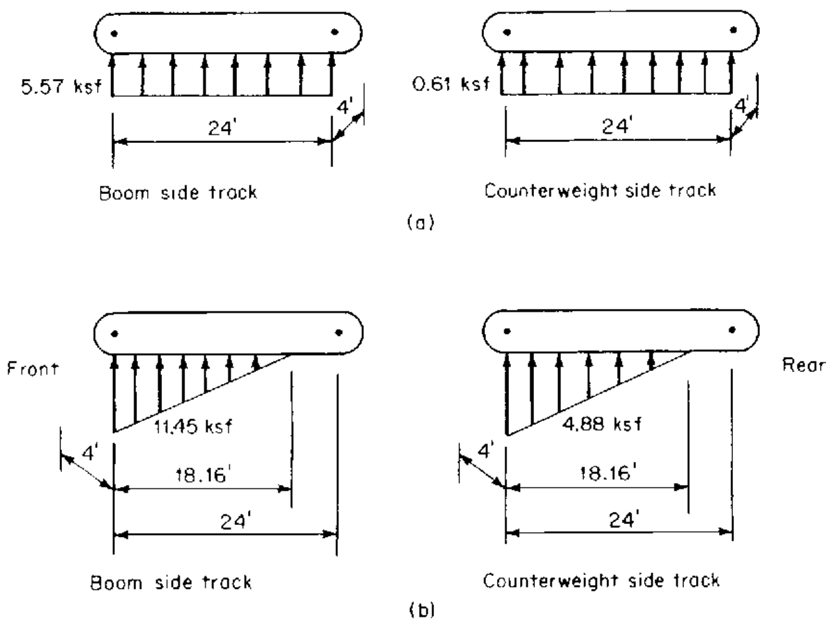


FIGURE 5.20 Solutions to parts (1) and (2) of Example 5.8.

Compared with truck and crawler cranes, rough-terrain cranes are light in weight, although larger and heavier models are continually being introduced. For these machines, a rule of thumb can often be used for approximating maximum outrigger loads. As much as 75% of gross machine plus load weight can be assumed to bear on one outrigger when the boom passes over that corner. For fully over-the-side or over-the-rear work, the percentage will be lower, as the reaction to the boom and load will be shared by two outriggers.

Most manufacturers provide software or tables to aid the user in determining outrigger reactions for telescopic cranes of both truck and rough-terrain types. At minimum, following an industry standard,[5] the upper limit for outrigger loads may be available as part of the technical documentation.

All-Terrain Cranes Outrigger Loads

All-terrain cranes have been introduced with sizes and capacities approaching the largest truck cranes. Outrigger loadings, of course, can be calculated using the procedures given, but the weight and CG data for various boom-length combinations will be much more difficult to acquire. Although the rule of thumb given for rough-terrain

[5]Subcommittee 31 of the Off-Road Machinery Technical Committee in its Recommended Practice SAE J1257, Rating Chart for Cantilevered Boom Cranes.

cranes is also applicable, for the larger machines the pad loads can be very high so that more accurate values are preferable for safety and efficiency. Manufacturers have been providing users with means to obtain these loads.

5.6 Supporting the Crane

A note common to all mobile-crane rating charts states, one way or another, that the "ratings have been derived assuming that the machine is standing on a firm, uniform, level supporting surface." Translated into practical language, this means that the ratings given are appropriate to use only if the outriggers, tracks, or tires are properly supported so that throughout operations, the crane will remain level to within a specified tolerance. Most manufacturers specify ± 1%, but some limit deviations to ±½%. An unstated but important further requirement is that the support arrangement must not permit the crane to rock or sway during operation. A truck, rough terrain, or *all-terrain crane* can rock if three of the outrigger floats, or two diagonally opposite floats, are supported on hard points while the others are on a yielding material. An example would be a truck crane with three outriggers on concrete pavement and the fourth on soil. Similar effects occur with crawler cranes when a portion of one track is on a hard point.

A hard point is a local zone or area that will not settle or displace under load as much as the general support area will settle. Some examples of hard points include bedrock, a massive concrete footing or block, soil cement, or a stiff steel member. A crawler crane with a long boom was discovered to have rocked and subsequently overturned because, during travel with a load, it passed over a buried timber with one track while on a general ground surface that was soft.

Many crane operators develop a sense of the ability of various soils in their home areas to support load and for the nominal cribbing needed to sufficiently spread outrigger float loads to a larger bearing area. But, then again, tipping accidents and near accidents occur too frequently for comfort or complacency. Consideration for crane support should be a part of the planning for just about every crane operation.

As a rule of thumb, for ordinary open-field operations on reasonably good ground, where the load can be grounded at any point during the work, crane supports can be left to the operator. Ground can only be considered reasonably good if there is knowledge or experience with the ground characteristics or if testing has been done. Where those conditions do not exist, crane support should be included in the operational plans.

Ground Support Capacity

Presumptive soil-bearing capacities used for building foundation design are often conservative when applied to crane supports. Buildings present long-term loads to the ground, so that allowable soil capacity must reflect a degree of long-term settlement control not necessary for crane use.

Unless the crane is to remain installed for weeks or months, the initial settlement on first bringing the crane into place will be the only settlement of note; its effect will be overcome by initial leveling of the crane. For long-term installations, periodic releveling may be required. Long-term settlement is of little practical consequence in mobile-crane support design if attention is paid to keeping the crane level at the site.

When evaluated on the basis of strength alone, most soils can support higher crane loads than the building design values would indicate. Although cranes are often operated on the ground on uncribbed floats with apparent success, there is actually great risk involved in doing this on most soils. Standard outrigger floats are limited in size by weight and stowage considerations and therefore can induce very high [15 tons or more per square foot (1.44 MN/m^2)] bearing pressures. Soil shear failure is sudden (Figure 5.15) and often results in tipping. *Cribbing* is needed in most instances.

The soil at any location is a unique blend of constituents that may include very fine particles such as silts or clays, fine- to medium-sized sand particles, coarse sands, gravels, and boulders, or organic materials, as well as water or even ice. Soil makeup and content are important factors in soil bearing strength, as is initial density or degree of compaction.

Organic material in soil makes the mass resilient or spongy and is therefore an unwanted component of support soil. Most surface soils contain some organic plant materials which ordinarily do not have an important effect on crane support properties, but as the organic content and thickness of surface soils increases, bearing capacity drops. A thick organic layer is unsuitable for crane support.

Fine-grained soils tend to shear at relatively low values. As soil particle size and the variation in size within the sample increase, soil strength will increase as well. The highest capacities are obtainable in compact well-graded sand-gravel mixtures made up of rough uneven particles.

A measure of soil capacity is its angle of repose, the maximum angle of a naturally stable slope. Shear strength increases with the angle of repose, and along with it the usable bearing capacity increases. For a given soil mix the angle of repose and strength will improve as compaction, measured by density, increases. Table 5.1 lists presumptive bearing capacities for various common soil types, as well as the authors' suggested bearing capacities for crane support. These are offered to help in installation planning; evaluation of the actual soil by an experienced person, preferably a licensed professional engineer, is certainly advisable if any doubt exists about soil capacity.

Soil Type	Density or State	Approx. Unit Weight, lb/ft³	Presumptive Bearing Capacity, tons/ft²: Typical Building Code	Presumptive Bearing Capacity, tons/ft²: Fore Cranes	Angle of Internal Friction, Degrees
Rock (not shale unless hard)	Bedrock Layered Soft		60 15 8	60 15 8	
Hardpan, cemented sand or gravel			10	10	
Gravel, sand and gravel	Compact Firm Loose	140 120 90	6 4	8 6 4	45 40 34
Sand, coarse to medium	Compact Firm Loose	130 110 90	4 3	6 4.5 3	42 38 34
Sand, fine, silty, or with trace of clay	Compact Firm Loose	130 100 85	3 2	4 3 2	34 30 28
Silt	Compact Firm Loose	135 110 85		3 2.5 2	30 28 26
Clay	Compact Medium Soft	130 120 90	3 2 1	4 2.5 1	30 20 10

†1 lb/ft³ = 16.018 kg/m³; 1 ton/ft² = 95.76 kN/m².

Table 5.1 Soil Bearing Capacity Data†

Ground surfaces can conceal places of potential peril for mobile cranes traveling or working. Pavements may cover vaults, poorly consolidated backfill, or voids where fine-grained soils have washed out. The variety of soil conditions encountered in crane work is as wide as the field of soil mechanics. Where there is complexity or doubt, a geotechnical engineer might be consulted. Here are some general lessons from the authors' experience.

- Recently backfilled areas should be approached with caution, unless the crane user knows that the fill is a select material that has been placed and tamped in layers and then tested for compaction.

- Utility trenches are often hastily backfilled and should be approached by a crane warily unless it is known that a proper backfilling job was done. The surface may look deceptively sound if the backfill is fresh; over time, runoff and pounding of traffic may cause some revealing rutting.
- Backfilling around footings is also frequently loose and uncontrolled, a potential danger for crane support.
- A perilous condition, especially for crawler cranes, is a support that is partly on soft backfill and partly on hard ground leading to differential settlement under the crane. Fresh backfilled areas are prone to this hazard.
- Pavements conceal manhole chambers and utility vaults that are not fully revealed at the surface. Cellars of urban buildings sometimes extend out under the sidewalk or even under the roadway.
- Heavily trafficked areas around a construction site are often well compacted and thus suitable for cranes. If the ground is churned up and muddy, the surface can be improved with crushed stone or brickbats.
- Old urban streets sometimes have sewers and gas mains below that are not in good condition. The problem for cranes may be exacerbated by the disturbance caused by new cellar excavation. Sheeting and bracing may bring about lateral displacement of utility lines or the washout of fine-grained soils around them. Utility lines have been known to rupture from this combination of soil disturbance and crane loading.

A common thread to most of these warnings is that disturbed ground can be problematic. When cranes are operating at a site, excavations and backfilling should be a concern that should ideally be brought under control by sound engineering and construction practices. Some enlightened contractors and owners understand the implications of construction loads, determining in advance where heavy equipment may be placed and preparing a site accordingly.

A second theme of the warnings points is the need to survey a site to evaluate surface and underground conditions before a crane is brought in. In some instances it is not possible to determine observationally whether backfill has been prepared adequately to support a crane. Some load testing might be advisable, say by running trucks or heavy equipment across a backfill to see its effect. The crane itself could be used for a self-check by swinging the counterweight over the backfill with the boom at minimum radius and with no load on the hook.

Supporting Outriggers

Outrigger pads, also called *floats*, of most cranes are small relative to the loads they support. If the soil under a float yields, the float presses into the earth. The resulting abrupt ground settlement might cause the crane to tip. For that reason, most floats have turned-up edges. The suddenly expanded bearing area engendered by the flared edges may inhibit further settlement. Notwithstanding this protective feature, outriggers should not be supported directly on soil. Cribbing ought to be placed under the outriggers to reduce the *ground bearing pressure*.

Large cranes are widely understood to have substantial outrigger loads that need cribbing. Not unusually, plates or timbers for this purpose are sent out with the crane on a truck carrying counterweights or other accessories. But a small rough-terrain crane is not sent out with an accessory truck that can carry cribbing to the site. Though relatively light in weight, a rough-terrain crane usually has small floats that will develop very high unit bearing pressure. The need for sound cribbing is no less than for the large machine. With frequent repositioning, sometimes a few times a day, there is a temptation to skimp. Oftentimes the field crew is left to improvise by using scraps picked up around the site.

Though ground bearing capacity varies from place to place, standard cribbing can be developed for typical conditions. A reliable solution, practicable for smaller cranes moving about a site, is to fabricate steel spreaders such as shown in Figure 5.21. These temporarily attach to the outrigger floats so that they can be carried by the crane from one working position to another.

Cribbing

Installations not requiring cribbing are the exception rather than the rule. Even on paved areas, failure to use cribbing often results in cracked concrete or in a clear impression of the float left in blacktop. A 200-ton (180-t) truck crane can easily have outrigger reactions of 150,000 lb (668 kN) on 30-in-diameter (762-mm) floats, making the pressure on the supporting surface about 15 tons/ft^2 (1465 kN/m^2). Most cranes of this size have yet smaller outriggers, so the pressure would be even higher.

The bearing area needed is obtained by dividing outrigger load P by soil capacity s, keeping units consistent. For timber cribbing, assume a trial timber size, such as 6 × 6 (150 × 150), 6 × 8 (150 × 200), or 12 × 12 (300 × 300). Only as many timbers can be used as can fit under the float and effectively carry its load (Figure 5.22). The dimension $b = ne$, where n is the number of timbers to be used. Timbers can overhang the float in width, but a cross-layer of planks should be added to make up for imperfect centering of the timbers. Timber lengths are chosen so that $bc \geq P/s$, the needed bearing area, and with c usually taken as an even, practical length.

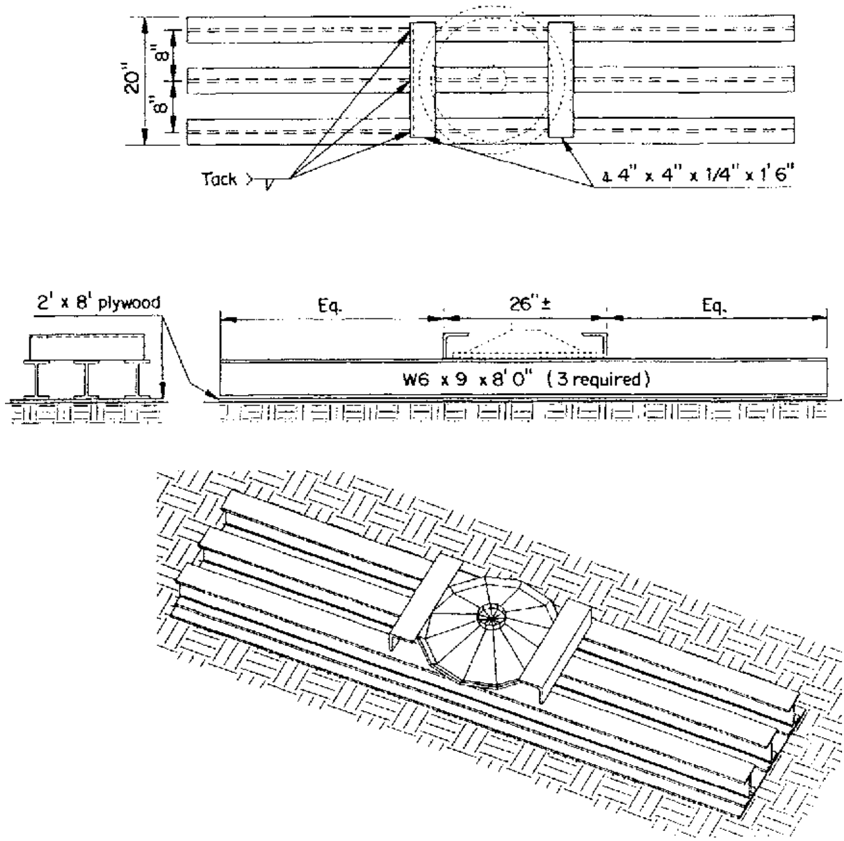


FIGURE 5.21 When rough-terrain or all-terrain cranes are to be moved about a site for a number of work assignments, prefabricated outrigger load spreaders can be used. A spreader of the type indicated here, which has been designed for a 35-ton (32-t) crane, can be lifted by the outrigger jacks and moved with the crane.

Effective width of timber cribbing can be increased by placing a steel plate or cross-timbers below the float. If the overall cribbing height exceeds the stroke of the *outrigger jacks*, the crane can be raised to attain the required height. On many contemporary cranes, the suspension can be raised. If that measure is insufficient or unavailable, wheels can be driven up onto mudsills to raise the whole crane. Concerned about sliding, some users eschew steel-to-steel contact. A simple cure is to insert plywood between float and plate to increase friction.

The applied soil pressure q under the cribbing is then $q = P/bc$ and, of course, q ø s. The timbers must be checked for bending strength. Bending moment is given by

$$M = \frac{qba^2}{2} \tag{5.35}$$

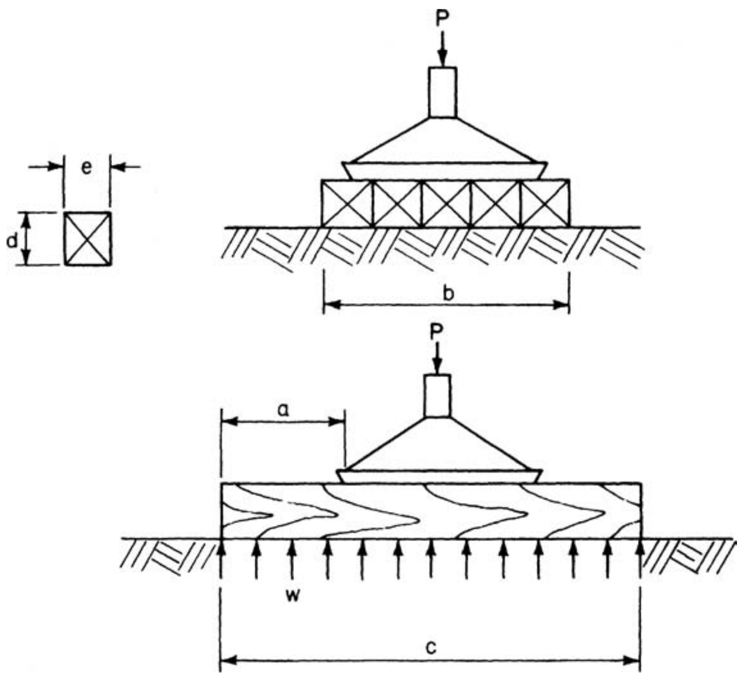


FIGURE 5.22 Timber outrigger float cribbing.

and for n timbers of actual (as opposed to nominal) dimensions d and e, the bending stress will be $f = My/I$ or

$$f = \frac{3qa^2}{d^2} \tag{5.36}$$

For timber typically available at construction sites, the authors use 1200 lb/in^2 (8274 kN/m^2) as an allowable bending stress unless it has been confirmed that material with a higher strength grade will be used. Timber of poorer quality should not be used.

Horizontal shear stress often controls timber cribbing design. If excessive, the timber can crack horizontally at midheight, sharply reducing resistance to bending and inviting failure. The horizontal shear stress v for a rectangular element is given by

$$v = \frac{1.5qa}{d} \tag{5.37}$$

Allowable shear stress for typically available timber can be taken as 100 lb/in^2 (690 kN/m^2) and sometimes higher. Timber design codes allow substantial increases in allowable stress in both bending and shear for short-duration loads. An increase of 25% for a short-duration outrigger load is not unreasonable, though the designer might want to compensate in the opposite direction to allow for degradation of the lumber. The environment and application are, after all, rather harsh.

Steel plates can be used for cribbing outrigger floats. As there is no longer a limit to dimension b, bending stress must be checked in two directions using Eq. (5.35). Equation (5.36) shows that moments in both directions will be equal and that maximum design efficiency will be obtained when $b = c$. Shear will not be important for steel plates, and the maximum permissible bending stress can be taken as 27,000 lb/in^2 (186 MN/m^2) for common mild steel.

That bearing pressure is uniformly distributed under the cribbing is a useful fiction that suffices for most support conditions on soil. In reality, the distribution is bell-shaped with the actual distribution dependent on the stiffness of both the cribbing and the soil. Stiff support members will deflect less than limber ones, causing a more even distribution of pressure beneath the cribbing. No harm can follow if larger or thicker plates or longer or larger timbers are used than specified. Over-length cribbing will deflect, concentrating the load. The soil will then react by compressing and undergoing small settlements that will redistribute loading toward the periphery. A balance will be reached, soil settlements and cribbing deflections combining to produce an equilibrium state.

Both steel plates and beams were used under the outriggers of the crane shown in Figure 5.23. The massive cribbing was needed to spread the substantial outrigger loads on the street adjoining sidewalk vaults and above a subway line. The wheels were driven onto planks to raise the floats high enough to clear the deep supports.

On uneven ground, void space must be filled in below the cribbing. While small voids can be bridged and slight unevenness may be pressed out under the outrigger load, larger voids can cause the crane to rock and to tip over. If in doubt, the cribbing can be tested by rotating the superstructure to place the counterweights over it with the boom at the minimum radius.

When the crane is operating on a structure instead of soil, an assumption that the pressure under the cribbing is uniform is nonconservative. The effect of the cribbing in distributing the load may be assessed either by a simple rule-of-thumb distribution or by a sophisticated analysis such as finite element.

Example 5.9

1. Design timber support cribbing for a 32-by 32-in (813-by 813-mm) float that must carry 150 kips (667.2 kN) on soil with a capacity of 5 tons/ft^2 (478.8 kN/m^2). Assume new lumber, grade 1500f, and allowable shear stress of 125 lb/in^2 (860 kN/m^2).

Figure 5.23 Two-tiered cribbing of plates and beams required to spread the outrigger loads over a wide area due to the presence of a subway line under the street and vaults under the sidewalk in front of the New York Stock Exchange. (*Mathieu Chaudanson.*)

Solution

Try 10 × 10 (250 × 250) timbers; the actual dimensions are $e = d = 9.5$ in (241 mm). Four timbers can be placed under a 32-in float, so that $n = 4$ and $b = 4(9.5) =$ 38 in = 3.17 ft (965 mm). The minimum permitted bearing area is

$$A_{\min} = \frac{150 \text{ kips}}{(5 \text{ tons/ft}^2)(2 \text{ kips/ton})} = 15 \text{ ft}^2 \text{ } (1.39 \text{ m}^2)$$

$$c \approx \frac{15}{3.17} = 4.73 \text{ ft } (1.44 \text{ m})$$

Use $c = 5.0$ ft (1.52 m), which gives $\alpha = (5.0 - 32/12)/2 = 1.17$ ft (357 mm) and $q = 150$ kips/[(3.17 ft)(5.0 ft)] = 9.46 kips/ft^2 (453 kN/m^2). The bending stress in then given by Eq. (5.41).

$$f = \frac{3(9.46 \text{ kips/ft}^2)(1.17 \text{ ft}^2)^2}{(9.5 \text{ in})^2} 1000 \text{ lb/kip}$$

$$= 430 \text{ lb/in}^2 \text{ } (2.96 \text{ MN/m}^2) < 1500 \text{ lb/in}^2 \text{ } (10.3 \text{ MN/m}^2) \qquad \text{Acceptable}$$

Equation (5.42) gives the horizontal shear stress.

$$\nu = \frac{1.5(9.46\ \text{kips/ft}^2)(1.17\text{ft})(1000\ \text{lb/kip})}{(9.5\ \text{in})(12\ \text{in/ft})}$$

$$= 146\ \text{lb/in}^2\ (1.01\ \text{MN/m}^2) > 125\ \text{lb/in}^2\ (860\ \text{kN/m}^2) \qquad \text{Not acceptable}$$

Try 10 × 12 (250 × 300) timbers; e = 9.5 in (241 mm) and d = 11.5 in (292 mm). Rechecking shear, we get

$$\nu = \frac{(146\ \text{lb/in}^2)(9.5\ \text{in})}{11.5\ \text{in}} = 121\ \text{lb/in}^2\ (832\ \text{kN/m}^2) \qquad \text{OK}$$

Use four timbers 10 × 12 × 5 ft.

2. Using the same conditions as in part (1), design a steel-plate float support.

Solution

The minimum bearing area needed is 15 ft² (1.39 m²), so a plate 4 ft (1.22 m) square will be adequate. The actual soil pressure then becomes

$$q = \frac{150\ \text{kips}}{(4\ \text{ft})^2} = 9.38\ \text{kips/ft}^2\ (449\ \text{kN/m}^2)$$

and the dimension

$$\alpha = \frac{4 - 32/12}{2} = 0.67\ \text{ft}\ (203\ \text{mm})$$

When the full allowable bending stress for steel is used, Eq. (5.36) can be rearranged to solve for the unknown plate thickness directly.

$$d = \left(\frac{3qa^2}{f}\right)^{1/2}$$

$$= \left[\frac{3(9.38\ \text{kips/ft}^2)(0.67\ \text{ft})^2}{(27\ \text{kips/in}^2)}\right]^{1/2}$$

$$= 0.68\ \text{in}\ (17.4\ \text{mm})$$

Use one plate 4 ft (1.22 m) square by ¾ in (18 mm) thick.

Supporting Crawler Cranes

Cranes were originally mounted on crawler tracks so that they could move about construction sites with ease. They still can, and often do, when carrying short booms, particularly for clamshell and dragline work. Operating track pressures are relatively low, with even the largest machines exerting maximum pressures of roughly 6 tons/ft² (575 kN/m²) when on soft ground. Wide tracks, low pressures, closely spaced track rollers, and immense power allow these cranes to slog through mud, climb over irregularities, and negotiate grades as great as 30%. It is only when long booms are mounted that special care and very low speeds are needed to restrain excessive boom movements.

Most crawler shoes have a profile that allows for a greater spread of the load on soft soil than on hard ground. On most ground, a crawler crane is not likely to develop enough pressure beneath the tracks to induce soil failure. Excessive settlement is a different question, however, as it can cause the crane to go out of level or lose the firm even support that is necessary for its stability. Often, the solution requires nothing more than placing small-sized scrap lumber beneath the tracks and using crane weight to press the lumber into the soil. It may take a few repetitions before the spreading effect of the scrap will furnish a firm level base. Good support under the ends of the track is essential to the stability of the crane and should be checked frequently. On some soils, from time to time the crane may need to be re-leveled.

When soils are generally good, it is possible that reliable support and a level crane can be attained with the tracks resting directly on the ground (Figure 5.24). For questionable soils, a 1- to 1½-ft-thick (300- to 450-mm) layer of road base or crushed stone will often furnish a satisfactory crane support.

Crawler-crane tracks can arch over minor void spaces or low points if they are situated near the center of a track. But all four track ends need to be well supported. Crawler bases are very rigid;

FIGURE 5.24 On firm level soil, a crawler crane can travel and sometimes operate without cribbing under the tracks. During a turning maneuver, the heavily loaded end of the track may gouge and push the soil. (*Mathieu Chaudanson.*)

any three of the four track ends will define the support plane on which the crane rests. The fourth corner may not be contacting the ground at all, but the loosely draped track will still be in contact with the ground. The only way to detect that track end is not bearing, is to observe that one or more track rollers are not in contact with the shoes. As the crane swings or picks a load it will rock like a table with one leg shortened. The tipping fulcrum will have foreshortened dramatically and the crane might overturn.

A similar situation can exist if the fourth corner is on a soft spot or a void hidden by cribbing. Positioning the crane where there is a hard point in the midarea of one track will produce similar unwanted results because the crane can rock fore and aft about the hard point. The hard point might be rock or concrete, a utility box, highly compacted soil in an area of generally loose soil, or even a piece of timber that the crane has run over and pressed into the ground. Anything that prevents the rigid crane base from setting up the four track ends in uniform contact with the support surface, all in the same plane, can cause rocking.

These support pitfalls seldom occur because the ground is usually sufficiently resilient to provide full support. However, on a number of occasions the authors have found such support subtleties explaining mysterious overturning accidents and close calls.

On poor soils and where a more definite support condition is required, heavy timber mats can be used under the tracks. These mats, colloquially called *pontoons* in some places, are usually made up of four to six 12×12s (300×300s) through-bolted together to form a unit. Design of mat supports can be treated similarly to design of outrigger cribbing by assuming it to be discontinuous in length (Figure 5.25). Alternatively, a timber mat can be analyzed as a continuous beam eccentrically loaded.

During operation, the greatest pressure exerted on the ground occurs when the superstructure swings over a corner of the crawler base. The maximal loading may occur either with a load on the hook or with the boom at minimum radius and nothing on the hook. Under extreme conditions, a double layer of mats may be needed to carry the load, or single pontoons laid parallel to the tracks on top of the mat may satisfy stress limits. Steel plates can also be used to spread track pressures over the mats. Figure 5.26 shows a crawler crane placed on timber mats for leveling and to reduce bearing pressure on the backfill below.

Yet greater pressure may be exerted when raising a long boom off the ground. Performed with the boom facing directly over the track end, the entire weight of the crane can concentrate on a small surface. For this reason, manufacturers sometimes specify *blocking* the ends of the tracks when lifting a long boom. Some models, however, lift the boom over the side of the track using supplemental outriggers.

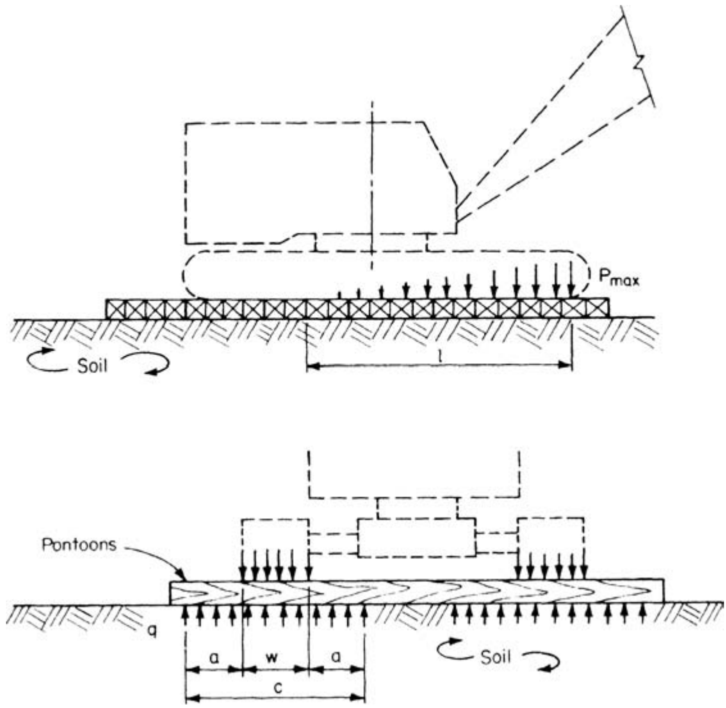


Figure 5.25 Heavy timber mats for crawler crane support.

Figure 5.26 Timber mats were used here to level out the crane working area while also spreading track loads over a wider bearing area and permitting some travel. Plywood is placed over the timber mats for leveling. (*Lawrence K. Shapiro.*)

Example 5.10

Using the crane of Example 5.8 and the track pressures of part (2), design a timber mat for soil of 2 tons/ft^2 (192 kN/m^2) capacity. Assume that timber mats are not new.

Solution

The worst possible crane position on the mat will occur if the high end of the pressure diagram is placed at the edge of a mat. The boom side track has maximum pressure of 11.45 kips/ft^2 (548.3 kN/m^2) diminishing to zero in 18.16 ft (5.53 m) (Figure 5.21b) or at the rate of 0.63 kips/ft^2 per foot (99.2 kN/m^2/m). Each mat is 4 in × 11.5 in = 46 in = 3.83 ft (1.17 m) wide so that the pressure at the lower pressure edge is 11.45 – 3.83 × 0.63 = 9.04 kips/ft^2 (432.7 kN/m^2). The load on this part of the mat is then

$$P = (4\text{ ft})(3.83\text{ ft})\frac{11.45+9.04}{2}$$

$$= 157.0\text{ kips }(698.1\text{ kN})$$

This requires a minimum soil bearing area of 157.0 kips/(2 tons/ft^2 × 2 kips/ton) = 39.25 ft^2 (3.65 m^2), and a minimum value for c of 39.25/3.83 = 10.25 ft (3.12 m). The minimum timber length needed, with the crane centered, is then 10.25 ft + 17.08 ft = 27.33 ft (8.33 m), so that a 28-ft (8.53-m) mats can be used. The final value for c will then be 10.92 ft (3.33 m) and a = (10.92 – 4)/2 = 3.46 ft (1.05 m). Ground pressure will then be 157 kips/(4 ft × 10.92 ft) = 3.59 kips/ft^2 (172.1 kN/m^2) which is lower than the given limit.

Checking bending stress in the timber with Eq. (5.36) gives

$$f = \frac{3(3.59\text{ kips/ft}^2)(3.46\text{ ft})^2(1000\text{ lb/kip})}{(11.5\text{ in})^2}$$

$$= 975\text{ lb/in}^2\ (6722\text{ kN/m}^2) < 1200\text{ lb/in}^2\ (8274\text{ kN/m}^2) \qquad \text{OK}$$

The horizontal shear stress is then given by Eq. (5.32):

$$v = \frac{1.5(3.59\text{ kips/ft}^2)(3.46\text{ ft})(1000\text{ lb/kip})}{(11.5\text{ in})(12\text{ in/ft})}$$

$$= 135\text{ lb/in}^2\ (931\text{ kN/m}^2) > 100\text{ lb/in}^2\ (690\text{ kN/m}^2) \qquad \text{NG}$$

If two layers of mats are used, the shear stress will be halved. Therefore, use two layers of 28-ft (8.53-m) minimum length timbers for the crane support mat.

Operations Near Cellar Walls

When a crane is installed in close proximity to a cellar wall, the installation design is not usually controlled by the soil bearing capacity but by the pressure the wall can safely sustain. Modern walls are built to support the lateral pressure of the earth plus an allowance for adjacent traffic. Crane weight, substantially more than usual traffic, can crack or even collapse a cellar wall.

The best approach is to keep the crane far enough away (Figure 5.27) to preclude a significant increase in lateral pressure. When the crane

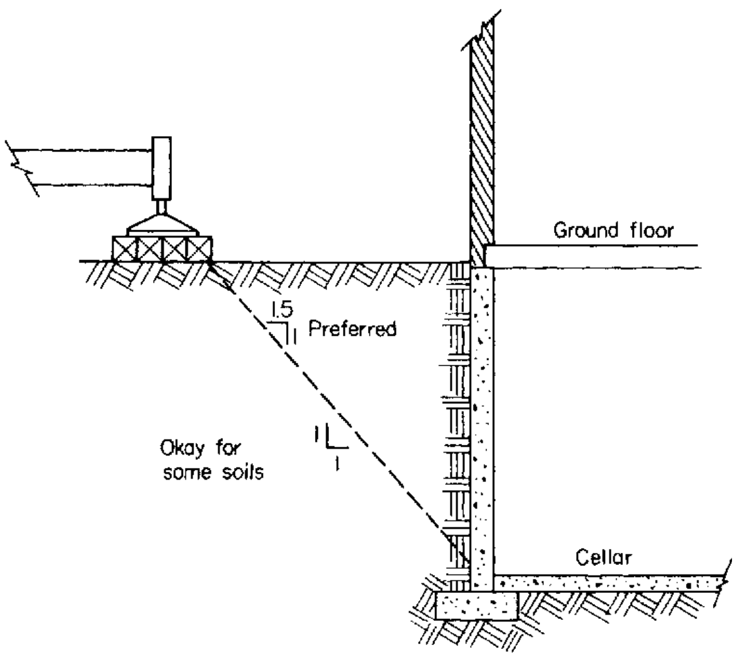


FIGURE 5.27 Positioning a crane next to a cellar wall.

must be positioned closer than the suggested distance, a complex design situation arises that is beyond the scope of this book. In such cases the authors urge that the services of a licensed professional engineer be retained. Not only will the engineer be able to provide the needed design, but his services are required by the laws of most states and provinces.

Site restrictions often push a mobile-crane position up close to a foundation wall. At urban sites, the curb lanes and sidewalk may be the only suitable location for a crane that minimizes obstructions of traffic. The difficulty is lessened if the contractor has had the opportunity and the foresight to reinforce the foundation wall for crane surcharge. With the ground floor in place to brace the top of the wall, backfill properly placed and sound cribbing under the crane, the support problem is solved. Unfortunately the wall is not always designed for the crane surcharge and the backfill or ground floor may be absent.

An existing cellar wall may have unknown properties, and its condition could be cause for concern. If the crane cannot be kept away from such a wall, the unit surcharge might be reduced by introducing a long stiff beam under the inboard outriggers, running parallel to the wall. A wall, new or existing, might also be braced into the cellar (Figure 5.28b and 5.29) or tied back to soil or rock.

Sometimes the best solution for placing a crane alongside a retaining wall or foundation wall is to rest on top of it, taking advantage of

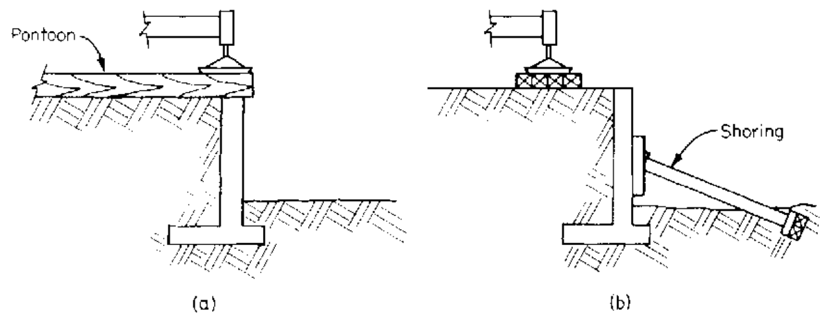


Figure 5.28 Two ways of bringing a crane very close to a retaining wall.

the ample capacity of most structural walls for supporting vertical load. The strength and stability of the wall should be checked by an engineer along with its bearing support. Some walls—those which are designed as grade beams spanning between footing or pile caps are an example—may be incapable of carrying large crane loads. A wall that supports both earth pressure and a vertical crane load will have an interaction of compression and flexure.

A wall might also provide direct bearing support for a crane platform. Composed of steel framing and timber or just timber mats the temporary platform allows the crane to drive into place and operate (Figures 5.31 to 5.32). It must have sufficient strength to accommodate wheel or track loads while the crane is maneuvering into position. The outboard side of the platform should be far enough from the wall to avoid soil instability (Figure 5.31) or excessive lateral pressure (Figure 5.27). Wall braces may be needed (Figure 5.32) for lateral support.

The position of inboard outriggers or tracks in relation to the supporting wall requires attention both in the design office and the field. Support loads acts through the centers of the outrigger jacks or track rollers. If the intention of the designer is to bring these forces to bear directly on the wall, instructions for installation must be clear and the execution precise. The forces cannot be permitted to act on a cantilever portion of a deck (as in Figures 5.30 and 5.31). To do so can cause the crane to tip at less than rated load because the tipping fulcrum will shrink back from the track rollers or outrigger jacks to the wall. On the other hand, the rollers or jacks can be supported outboard of the wall only if the platform can carry their loads without suffering excessive shear, bending or deflection.

Crawler-crane tracks are not precision instruments; runway and platform arrangements must allow for some deviation in position and room for maneuvering. Moving and positioning the crane often involves locking or reversing one track relative to the other. Track motion can shift mats about, destroying the integrity of the deck. Steel plates can be laid down over the surface to improve the

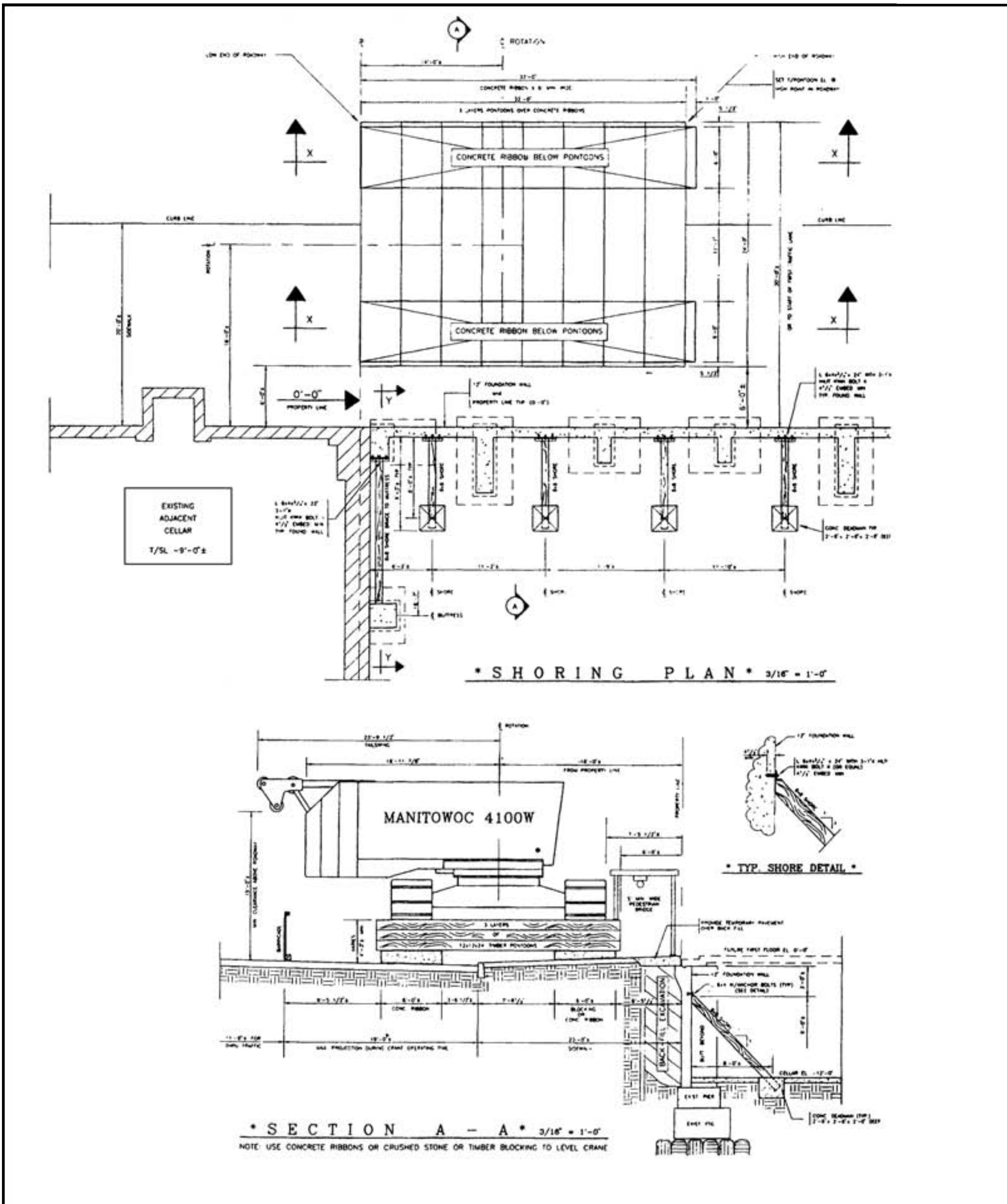


Figure 5.29 An installation drawing for a crawler crane with tower attachment that was placed on a sidewalk adjacent to a back-filled free-standing wall. The wall had to be braced to resist the crane-induced pressure. Note also that several layers of heavy timber mats were used to raise the gantry and provide headroom clearance for traffic at this tight urban site.

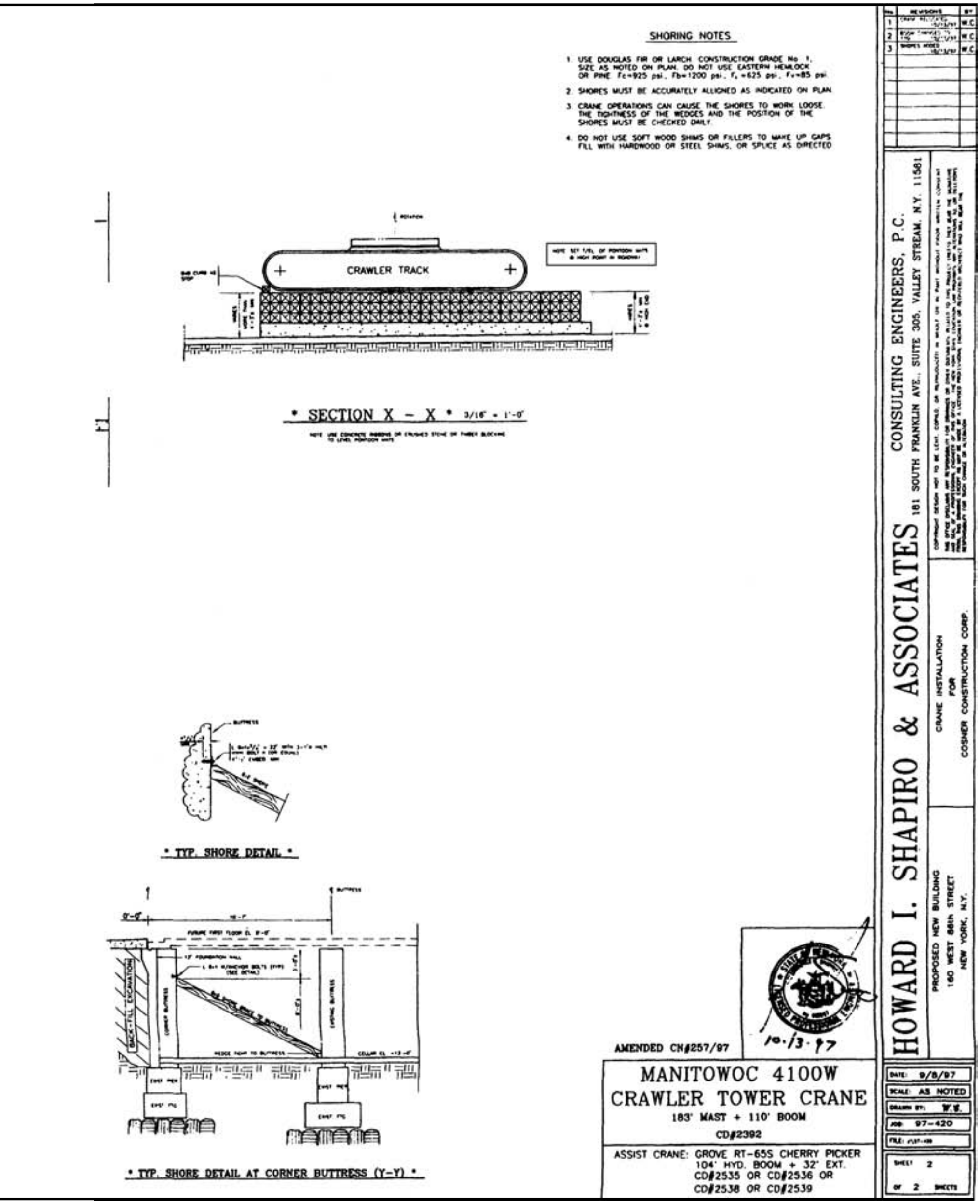

SHORING NOTES
1. USE DOUGLAS FIR OR LARCH CONSTRUCTION GRADE No. 1, SIZE AS NOTED ON PLAN. DO NOT USE EASTERN HEMLOCK OR PINE. Fc=925 psi., Fb=1200 psi., Fc=625 psi., Fv=85 psi
2. SHORES MUST BE ACCURATELY ALLIGNED AS INDICATED ON PLAN
3. CRANE OPERATIONS CAN CAUSE THE SHORES TO WORK LOOSE. THE TIGHTNESS OF THE WEDGES AND THE POSITION OF THE SHORES MUST BE CHECKED DAILY.
4. DO NOT USE SOFT WOOD SHIMS OR FILLERS TO MAKE UP GAPS FILL WITH HARDWOOD OR STEEL SHIMS, OR SPLICE AS DIRECTED
CRAWLER TRACK
* SECTION X – X * 3/16" = 1'-0"
* TYP. SHORE DETAIL *
BACK-FILL EXCAVATION
* TYP. SHORE DETAIL AT CORNER BUTTRESS (Y–Y) *
AMENDED CN#257/97
10.13.97
MANITOWOC 4100W
CRAWLER TOWER CRANE
183' MAST + 110' BOOM
CD#2392
ASSIST CRANE: GROVE RT-65S CHERRY PICKER
104' HYD. BOOM + 32' EXT.
CD#2535 OR CD#2536 OR
CD#2538 OR CD#2539
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
181 SOUTH FRANKLIN AVE., SUITE 305, VALLEY STREAM, N.Y. 11581
CRANE INSTALLATION
FOR
COSNER CONSTRUCTION CORP.
PROPOSED NEW BUILDING
160 WEST 66th STREET
NEW YORK, N.Y.
DATE: 9/8/97
SCALE: AS NOTED
JOB: 97-420
SHEET 2
OF 2 SHEETS

Figure 5.30 By using a heavy timber deck and placing the outriggers on the wall, it was possible to bring this 250-ton (227-t) truck crane close enough to place the precast girders shown. The line of steel plates was needed to spread wheel loads while moving the crane from one working position to the next at this project over the right-of-way of a commuter rail line.

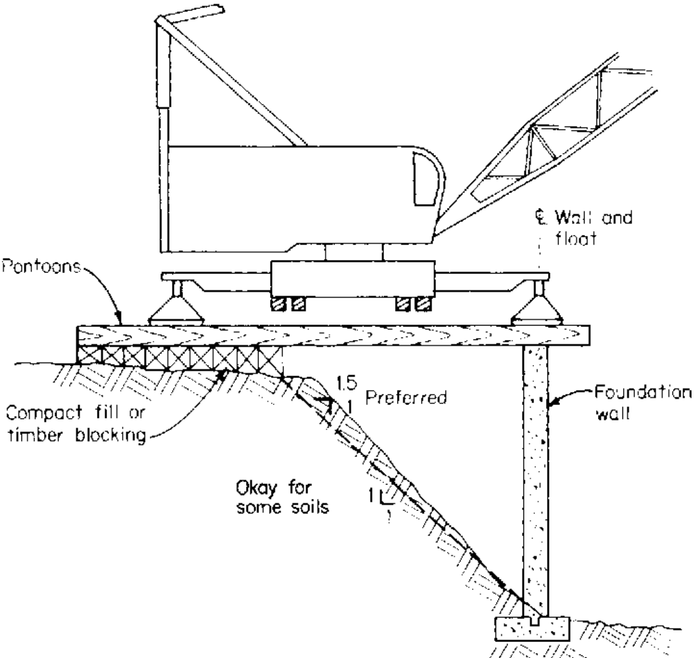


Figure 5.31 Installation of a truck crane on a foundation wall during construction.

Figure 5.32 The timber mats supporting this crawler crane rest on steel beams spanning between the wall and ground supports. The free-standing wall has been backfilled and must also resist ground pressure from the crane, hence the timber braces. Note the steel clip angles bolted to the wall to resist uplift forces at the wall face. (*Jay P.Shapiro.*)

condition, but on occasion it may be necessary to grease the plates to reduce friction. Another alternative is to nail down a "sacrificial" plywood wearing surface.

A platform sufficiently robust to carry a heavy crane may be quite high above the ground. Sometimes the height is deliberately raised so that pedestrians or traffic may pass beneath the tail swing. A temporary access ramp must be created to enable the crane to climb into position. With good fortune, space might be available to make the ramp a straight run like the one shown in Figure 5.33, but sometimes the ramp must be offset so that the crane can make an approach from a safe distance away from the retaining wall. As the crane approaches the top of the ramp, the leading edge of the tracks will be in the air above the platform. The crane will pivot on the head of the ramp, which must be capable of supporting the full crane weight bearing on it.

When the ramp approaches the platform at an offset, the meeting of the two presents some difficulty. An angled approach creates a compound transition that is difficult to reconcile with the rigid nonarticulating carbody and side frames. An easier transition is made by widening the platform temporarily so that the ramp can meet it squarely.

The capability of crawler cranes to negotiate ramps varies from model to model. Specifications for newer models tend to be more

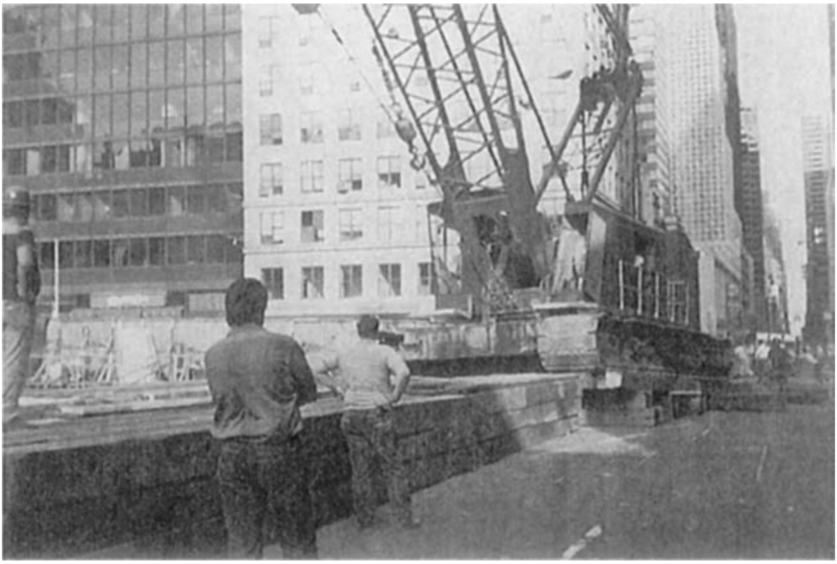


Figure 5.33 A timber ramp had to be constructed for this crawler crane to get up onto its operating runway 3 ft (1 m) above the roadway at the low end. (*Howard I. Shapiro.)*

restrictive than the old ones. The manufacturer's limitation for *gradability* should be respected.

Operations Near Slopes and Retaining Walls

When operating adjacent to sloping ground, the crane should be kept an adequate distance away. If the crane is set too close, its weight may contribute to a shearing failure through the soil mass causing the crane to topple. Stability depends on soil type and local conditions. Figure 5.34 shows a rule-of-thumb minimum distance from the closest part of the crane or cribbing to the toe of the slope. In some instances this rule might be conservative. In some rare circumstances it might be insufficient. When in doubt, local knowledge might be useful, but better, a professional engineer should be consulted.

Figure 5.35 shows suggested safe positioning for work near retaining walls. If it becomes necessary to set the crane closer, an engineering analysis should be performed. Though many retaining walls are designed to sustain surcharge loads for traffic and the like, crane loads are of a higher order. The load might be spread by supporting the crane on a long stiff beam parallel to the wall. Two other possible engineered solutions are shown in Figure 5.28.

Figure 5.29 is an installation design drawing for a large crawler crane reflecting a situation similar to that shown in Figure 5.28b.

A crane operating close to a wall induces a surcharge loading on the wall. The surcharge decreases and spreads over a wider area as the

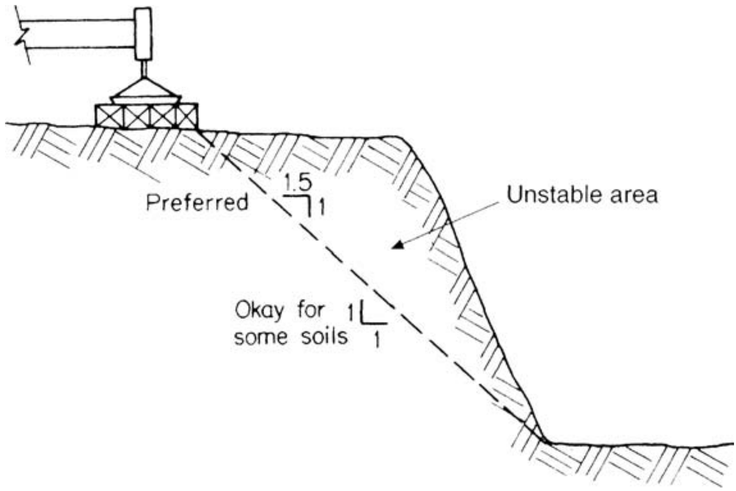


FIGURE 5.34 Placing a crane near the head of a slope.

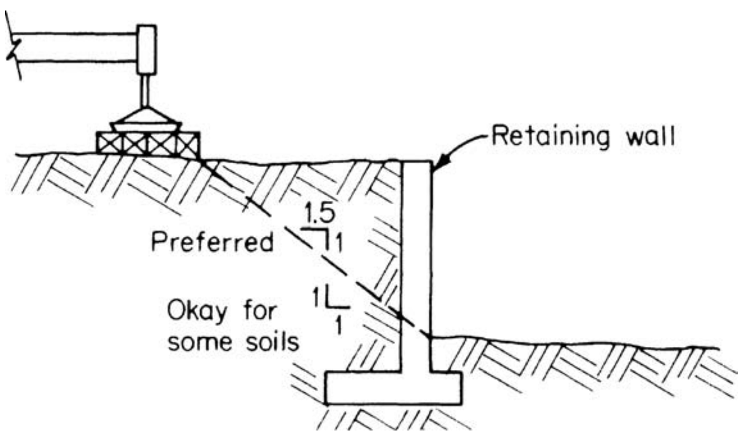


FIGURE 5.35 Placing a crane near a retaining wall.

loading moves farther away. Soil mechanics literature is replete with theories and methods that could be applied to these circumstances. The suitability of any one method depends on several factors; the soil type and resistance of the wall to displacement are usually two of the most important. The selection of a method to determine the load on the wall from the soil burden should also rest on these factors.

The Boussinesq theory, applicable to line loads or point loads in proximity to a nonyielding wall, is often useful for crane applications. For a crane surcharge of uniform ground pressure q starting at distance x from the wall and applied over width b (Figure 5.36), the

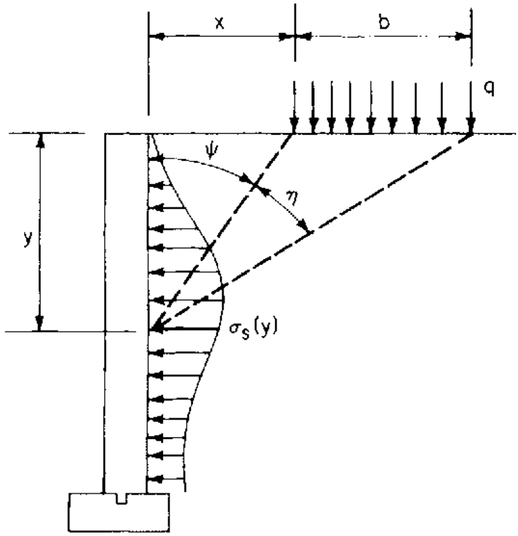


FIGURE 5.36 Pressure induced on a wall by an adjacent crane (surcharge) load.

lateral pressure against the wall at any distance y below the top of the wall for $\alpha = \psi + \eta/2$ is given by

$$\sigma_s(y) = \frac{2q}{\pi}(\eta + \sin\eta \sin^2\alpha - \sin\eta \cos^2\alpha) \tag{5.43}$$

where $\psi = \tan^{-1}(x/y)$ and $\eta = \tan^{-1}[(x+b)/y] - \psi$, both in radians.

Unmodified, the Boussinesq method is conservative for crane support because it is based on an infinitely long strip load whereas the crane cribbing is finite in length. A rational approach is to reduce the surcharge commensurate with the distance to the wall. For example, if the cribbing length is taken as L, the surcharge pressure on the wall can be uniformly reduced by a coefficient equal to $L/(L + x)$, where x is as defined in Figure 5.36.

The total pressure on the wall includes the burden of the soil and any groundwater. The Rankine theory for active earth pressure is useful for analyzing a noncohesive soil acting against a yielding wall such as unbraced sheeting or a retaining wall. It assumes that the wall displaces under lateral loading so that a wedge of soil also displaces and bears against it. Assuming no perched ground water, the pressure induced by the soil is

$$\sigma_\omega(y) = \gamma y K_a \tag{5.39}$$

where γ = soil unit weight (see Table 5.1)

K_a = (1 − sin ϕ) (1 + sin ϕ)

ϕ = angle of internal friction (see Table 5.1)

The total pressure at any depth y is then

$$\sigma_t(y) = \sigma_s(y) + \sigma_\omega(y)$$

Taking pressures at a series of depths y gives the wall pressure diagram (Figure 5.36). The wall must be checked for its ability to sustain the applied pressures; if it is inadequate, shoring or other means must be provided to assure stability.

If the wall does not displace, earth pressure is not active. The active earth pressure coefficient k_a should be replaced in Eq. (5.44) by the coefficient of earth pressure at rest k_0, which is often determined as

$$k_0 = 1 - \sin$$

These methods are useful for some common earth pressure calculation in uniform noncohesive soils; they do not take into account other soil characteristics that may require different approaches. Further information is available by referring to soil mechanics texts.

Operations on Structural Decks and Bridges

Mobile cranes, from rough-terrain models to the largest truck and crawler cranes, are routinely installed on bridges, access ramps, building floors, and plazas with cellars below (Figures 5.37 and 5.38).

Figure 5.37 A 600-ton capacity telescopic crane set up for operation on the upper roadway of the George Washington Bridge. Special grillage frames were designed to carry the substantial outrigger loads to the main bridge girders below. Crane placements and outrigger reactions were carefully controlled to respect the structural limitations stipulated for the bridge. (*James Scheld.*)

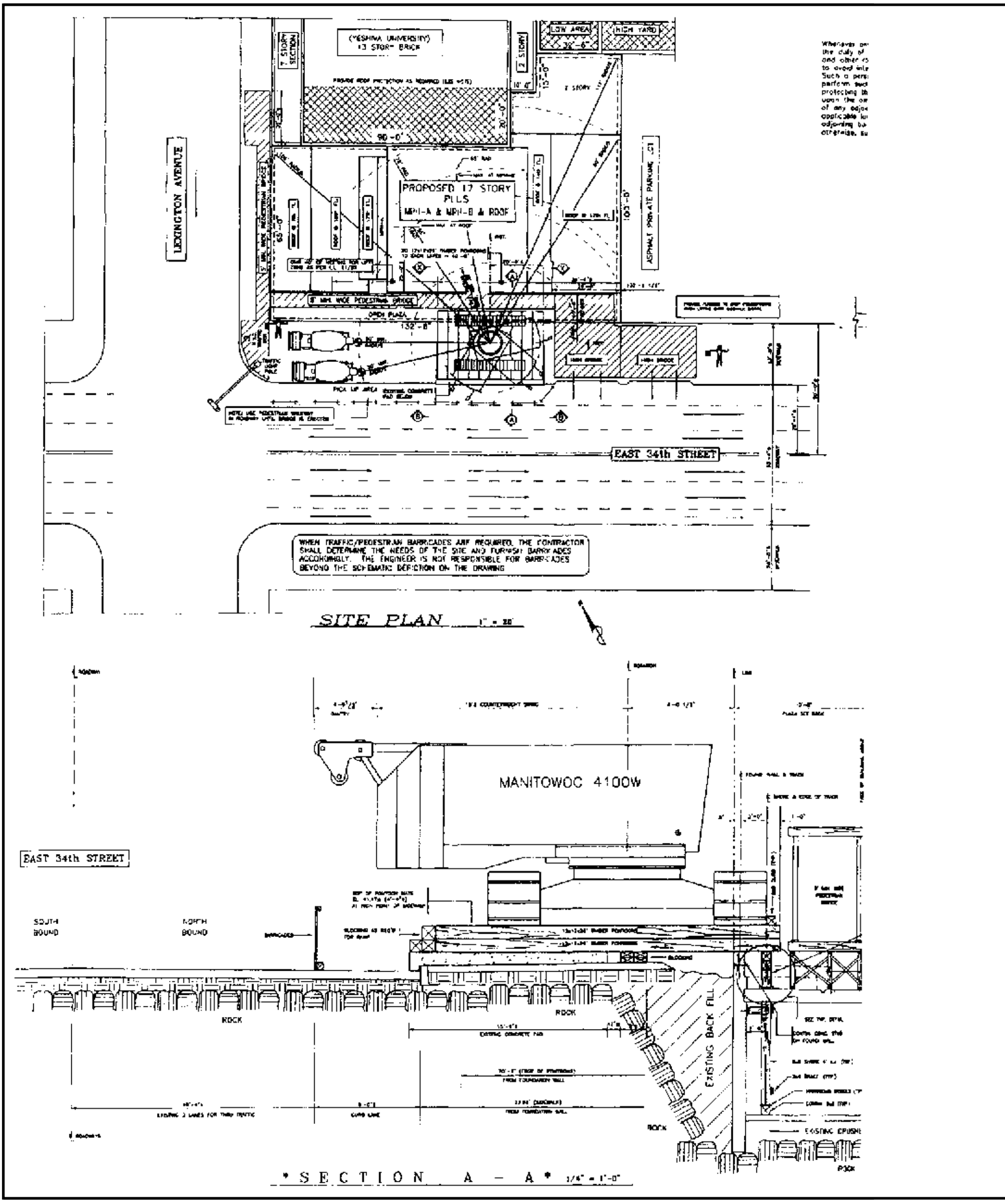
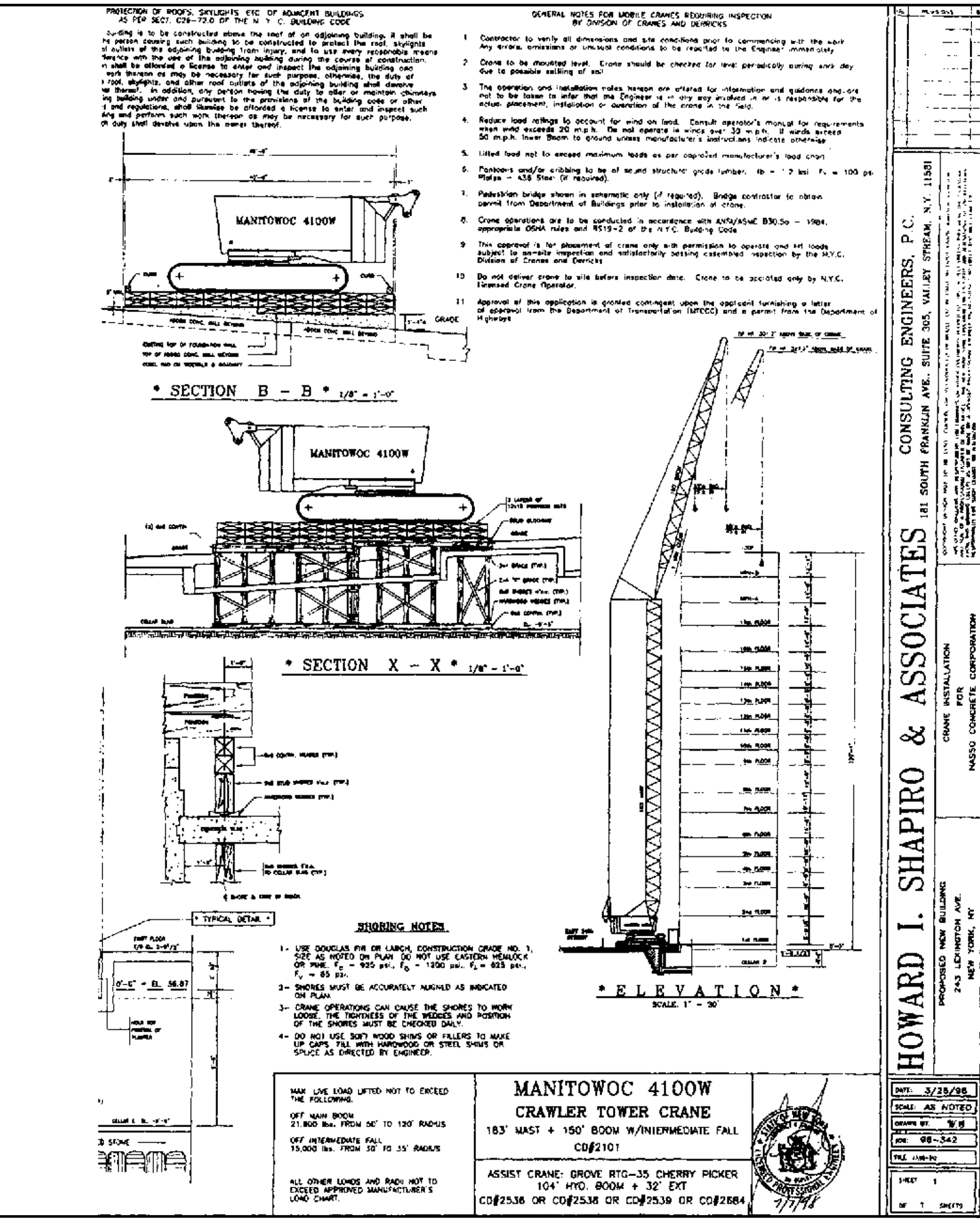


FIGURE 5.38 Installation drawing for a crawler crane with tower attachment at a typically constricted urban site. One track of the crane is partly supported on the ramp down to the parking garage of this reinforced concrete residential building. The ramp is shored, and in turn the shores run up to support the ends of the timber mats.


PROTECTION OF ROOFS, SKYLIGHTS ETC OF ADJACENT BUILDINGS AS PER SECT. C26-72.0 OF THE N. Y. C. BUILDING CODE
GENERAL NOTES FOR MOBILE CRANES REQUIRING INSPECTION BY DIVISION OF CRANES AND DERRICKS
MANITOWOC 4100W
* SECTION B - B * 1/8" = 1'-0"
MANITOWOC 4100W
* SECTION X - X * 1/8" - 1'-0"
* TYPICAL DETAIL *
SHORING NOTES
1- USE DOUGLAS FIR OR LARCH, CONSTRUCTION GRADE NO. 1, SIZE AS NOTED ON PLAN. DO NOT USE EASTERN HEMLOCK OR PINE. Fc = 925 psi., Fb = 1300 psi., FL = 625 psi., Fv = 85 psi.
2- SHORES MUST BE ACCURATELY ALIGNED AS INDICATED ON PLAN.
3- CRANE OPERATIONS CAN CAUSE THE SHORES TO WORK LOOSE. THE TIGHTNESS OF THE WEDGES AND POSITION OF THE SHORES MUST BE CHECKED DAILY.
4- DO NOT USE SOFT WOOD SHIMS OR FILLERS TO MAKE UP GAPS. FILL WITH HARDWOOD OR STEEL SHIMS OR SPLICE AS DIRECTED BY ENGINEER.
* E L E V A T I O N *
SCALE: 1" = 30'
MAX. LIVE LOAD LIFTED NOT TO EXCEED THE FOLLOWING.
OFF MAIN BOOM 21,800 lbs. FROM 50' TO 120' RADIUS
OFF INTERMEDIATE FALL 15,000 lbs. FROM 30' TO 35' RADIUS
ALL OTHER LOADS AND RADII NOT TO EXCEED APPROVED MANUFACTURER'S LOAD CHART.
MANITOWOC 4100W
CRAWLER TOWER CRANE
183' MAST + 150' BOOM W/INTERMEDIATE FALL
CD#2101
ASSIST CRANE: GROVE RTG-35 CHERRY PICKER
104' HYD. BOOM + 32' EXT
CD#2536 OR CD#2538 OR CD#2539 OR CD#2684
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
181 SOUTH FRANKLIN AVE. SUITE 305, VALLEY STREAM, N.Y. 11581
CRANE INSTALLATION FOR NASSO CONCRETE CORPORATION
PROPOSED NEW BUILDING 243 LEXINGTON AVE. NEW YORK, NY
DATE: 3/25/98
SCALE: AS NOTED
JOB: 98-342
SHEET 1
OF 1 SHEETS

The first choice of an operating area should be solid ground. A deck installation requires close attention by planners and field crew. Though a small crane might sometimes be able to travel and operate without restriction on a heavy-duty deck, in most instances each move and operating position must be analyzed and executed with strict controls.

Putting a crane to work on a deck is a two-part problem, the first being the maneuver to get the crane set up and the second the operation of the crane. The first part is seldom a difficult achievement for a telescopic crane on a standard roadway deck, as the traveling weights of these cranes conform to roadway limitations. Additional counterweights are self-installed after the crane is set up on outriggers. The second part—operating the crane on the deck—can be more of a challenge. Even rough-terrain cranes can produce outrigger loads that may be too great for elevated roadways or bridges to sustain safely without additional measures.

Crawlers present a different set of benefits and pitfalls than wheel-mounted cranes. The long rigid side frames are efficient in distributing the machine weight; the boom angle can be set to optimize the pressure distribution under the tracks when the crane is traveling over the deck. On the other hand, the rigid tracks can bear down on a high point, concentrating much of the weight of the crane on a small area and possibly causing a dangerous shift of the tipping fulcrum. As on soils, crawler tracks on a deck must be well supported, especially at the ends of the bearing surfaces.

When a deck does not have strength to support a crane, there are several common approaches for overcoming the inadequacy:

- Shoring under the deck to distribute the loads to other structure or to grade. These elements are simple to design and usually straightforward to install, but shoring is out of the view. Accidental damage or even their removal could go unnoticed, with catastrophic results. Shoring also might interfere with other operations.
- Distributing the loads over a larger area of the deck using plates, beams, or mats. This is a simple solution, if it works.
- Transferring the loads to stronger elements of the supporting structure. A beam or an entire platform may be installed to support an outrigger or the entire crane, depending on the need.
- Reinforce the structure to increase its support capacity. The best time to reinforce a structure is before it is built if the crane user has that opportunity. Temporary braces might be added. Reinforcement of slabs, connections, and beams, on the other hand, are permanent.

The structural aspects of deck operations should be designed by a professional engineer, with both stress and deflections as considerations. It is also important that supports and reinforcements be inspected before the crane is brought onto the deck and that operating personnel be thoroughly briefed on required and permitted procedures before the operation begins. Deck operations demand extra supervisory attention. Structural supports, both permanent and temporary, should be checked periodically for distress.

5.7 Crane Loads

A mobile crane can have a small margin between a hook load that is acceptable and one that will tip the crane. For that reason, the concept of a *critical lift* has particular relevance for mobile-crane operations. A load that approaches the full rated capacity should give pause to the planner and the operator, compelling either of them to give a second look at the load and the radius, and perhaps to look again to confirm that all else is as it should be. Rated loads include the weight of the hook block, slings, and other lifting accessories; a reliable accounting of those weights will be needed for a critical pick.

It stands to reason that a mobile crane should be sized to avoid near-capacity picks whenever possible, most particularly for repetitive work. Though there is no rule that disallows capacity picks, they do by nature engender elevated risk. Tempting fate is not good practice. In most instances the authors prefer to size mobile cranes for about 75% of capacity for work that is repetitive or not tightly controlled.

Job planning sometimes includes the development of an entire scheme for handling the load, requiring the use of special equipment and unique procedures such as tripping a wall panel into a vertical position. The methods needed to handle the loads may impose added loads and specific conditions that will affect the choice of crane and its installation.

Loads with large surface areas act as wind sails. Discussed at length in Chap. 3, the effect can be the equivalent of adding load on the hook. The ratings of available equipment may then dictate that wind limits be placed on the operation, a potential cost for lost time that must be reflected in bids or in job budgets.

Rapidly repeated or *duty cycle* operations impose impact loading on the crane, either by the nature of the load or of the operations themselves. Mass concrete placement by bucketing is one example. Efficient placement requires that cycle time be minimized. Rapid crane movements are needed; the loaded bucket is quickly lifted and swung and then quickly returned and lowered for another load. The severity of this service should compel a prudent user to limit loads to about 75% of rated capacity.

Dangerous loading situations can develop when a crane is used for demolition or debris removal, particularly where the crane is committed to supporting the load once the piece is freed from the structure. Weights can be seriously underestimated especially when pencil and paper are replaced with the eye. Load-indicating devices can serve well in demolition work for loads that can be test-lifted. Where this is not the case, good load estimates are the only means available. This implies that competent supervisory personnel must be present throughout the work.

A common demolition procedure that offers few problems when loads are kept small becomes a significant source of concern with large loads. The operation involves holding the load while it is being cut free from the rest of the structure. The operator takes a strain on the hoist line, trying to match the load to be lifted. If successful, the load will be freely suspended. Judged poorly, the load will suddenly drop or jump on being cut free, depending on whether the strain is too great or too little.

The differential between initial load-line strain and load weight becomes *impact* loading. Its effect on the crane is discussed in Sec. 4.6 under the heading Inertial Forces Affecting Stability, and in Sec. 3.3 under Linear Motion. Use of a load-indicating device permit the operator to preload the hoist line and avoid ill effects but only if the load has been reasonably well estimated in advance. Accuracy of the estimate becomes more important as the lifted load approaches the rated load.

Another danger can arise when a bundle or debris box is picked from the edge of a floor. The potential for peril is twofold: the boom can be side-loaded from pulling the load out from the floor and it can drop if the strain on the load is insufficient to match the weight. Rigging practices should be modified to avoid this type of situation altogether. But if unavoidable, a method such as illustrated in Figure 7.2 can be used.

5.8 Positioning the Crane

In deciding between two or more prospective crane positions, tradeoffs need to be weighed with costs and risks at stake. A cost could be a direct expenditure such as cribbing or indirect stemming from the relative efficiency of operating from one position or another. There can also be collateral costs such as traffic obstruction or a leave-out of construction. Some common risk factors (see Chap. 8 for a broader discussion) are the exposure of the public to the crane operation and the quality of the operator's field of vision. An assessment of alternate positions might also imply a difference of configuration or even a reconsideration of the crane model.

For highly repetitive operations, efficiency—hence cost—is largely a function of cycle time, which depends on the speed of crane motions and the ability to carry them out simultaneously. When a crane is set up to do just a few lifts, installation costs are the predominant cost factor over efficiency of the work cycle. From fastest to slowest, the main crane motions are

- Hoisting
- Swinging
- Booming in or out
- Traveling

Crane manufacturers go through considerable effort to give the operator latitude in controlling hoisting speed. Within a single work cycle, the speed may be varied greatly, creeping when control is needed and sometimes going as fast as the machinery will allow. Most mobile cranes have the facility to run an empty hook at high speeds; friction machines allow freefall. Where control is much more important than speed, the block can be reeved with multiple parts of line to slow down the hook motion.

Swing motion causes pendulation of the load in the direction of swing. Although it has little effect on stability, time may be lost in steadying the load before landing it. Experienced operators have the ability to tame this motion with seeming ease. Small swing arcs can be traversed at low speeds with little load disturbance and minimal elapsed time. Long arcs are time-consuming and may be combined with another machine motion carried out concurrently. On some sites the pick zone might be moved around to limit the arc.

Booming in and out is slow; attempts to speed up the motion will only increase radial pendulum action, decrease control of the load, and reduce stability. For high-production work, radius change should be kept to a minimum. Sometimes the picking zone can be varied to minimize this motion.

Pick-and-carry operation is very slow—when permitted—partly limited by the smoothness of the travel path. But use of a pick-and-carry procedure may enable a very small crane to do work that would otherwise require a large machine. It is within this context that its usefulness should be evaluated. Section 5.10 covers pick-and-carry work.

For large cranes, line of site is a key consideration for positioning. The operator cannot be expected to work from visual cues for the full work cycle; a long boom reaches places where the operator cannot see either the load or hand signals. But there is no point in making the operator blind at both ends of the lift cycle if that can be avoided. Line of site from the cab to the pick point on the street is a clear advantage for both safety and efficiency.

5.9 Crane Selection

Crane selection starts with a determination of the right type of crane for the application followed by short listing suitable models that are capable of performing the intended work. The end stage—picking the crane—will be most influenced by availability, vendor relations, and familiarity. (Better yet, owning the crane could be the most important factor.) Cost is a consideration, too, with assembly and transport being part of that calculation.

Mobile cranes have evolved into a variety of forms to fill industrial niches. Sometimes more than one crane type might be considered for the same assignment with trade-offs of installation costs, rental, and productivity considered in deciding what to use (Figure 5.39).

A rough-terrain or a small all-terrain crane often is used for general service on a work site because of its numerous advantages: low cost, quick setup, maneuverability, broad range of capabilities, and quick movement from task to task at the site with a one-person operating crew. It is useful for a multitude of tasks including unloading trucks, assembling larger cranes, and erecting moderate size buildings. The same characteristics make it useful as a "taxi crane" that can be sent on short notice to a small erection job or to install mechanical equipment on a roof. It is not especially fast moving or robust and thus not well suited for harsh or rapid repetitive applications such as bucketing large volumes of concrete. Duty cycle work is out of the question.

A telescopic all-terrain crane shares many of the same characteristics; it is simply larger and has more accessories to broaden the range of lifting. It is usually given short-term assignments. Some common tasks are short-term erection jobs, tower-crane assembly, and machinery placement.

Latticed boom truck cranes have been almost completely eclipsed by telescopic models (Figure 5.40). The cost of trucking in boom sections along with the time and space needed for assembly make them less alluring to the average user. But for medium-duration erection work they may be economical, with assembly costs outweighed by the faster work cycle.

A crawler crane is expensive to move in and out of a site. That excludes it in most cases from short-term work. It is most useful where the mobility of the tracks is an advantage or where the robust power train is needed for fast and cyclical operations. For a long-term assignment, expensive transportation can be offset by lower monthly cost as compared to a comparable truck crane.

5.10 Pick and Carry

The ability to travel with a load on the hook is a great productivity enhancement. Crawler, rough-terrain, and all-terrain cranes can do pick-and-carry work, though crawler cranes are best matched to the task. Because a wheel-mounted crane is supported on its tires during pick and carry and dynamics are involved, ratings are reduced as compared to those for static work. A crawler crane may also have lower ratings for

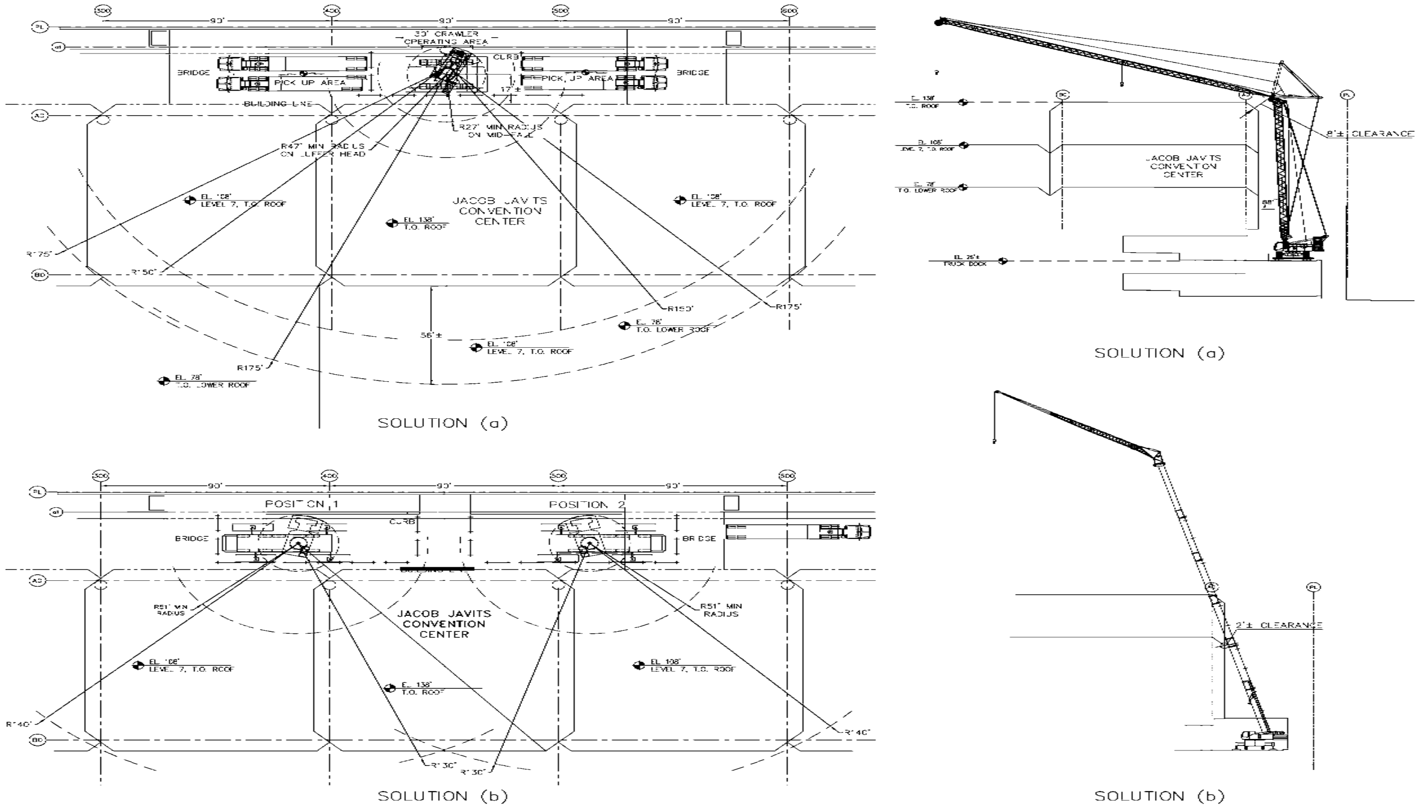


Figure 5.39 Two alternate approaches to the same task of erecting glass panels for a convention center. (a) Crawler crane with a luffing jib sweeping the full work area from one position and offering a productive work cycle. Erecting the jib or laying it down for a storm is complicated by the two flanking bridges; moreover the crane sits on an elaborate platform of steel beams needed to distribute the loads. (b) Two positions are required for a telescopic crane to do the same work, but raising and lowering the jib is readily done in the space between the bridges with the boom retracted. Outriggers only need simple plates for support, but this machine offers a slower work cycle than the crane crawler crane.

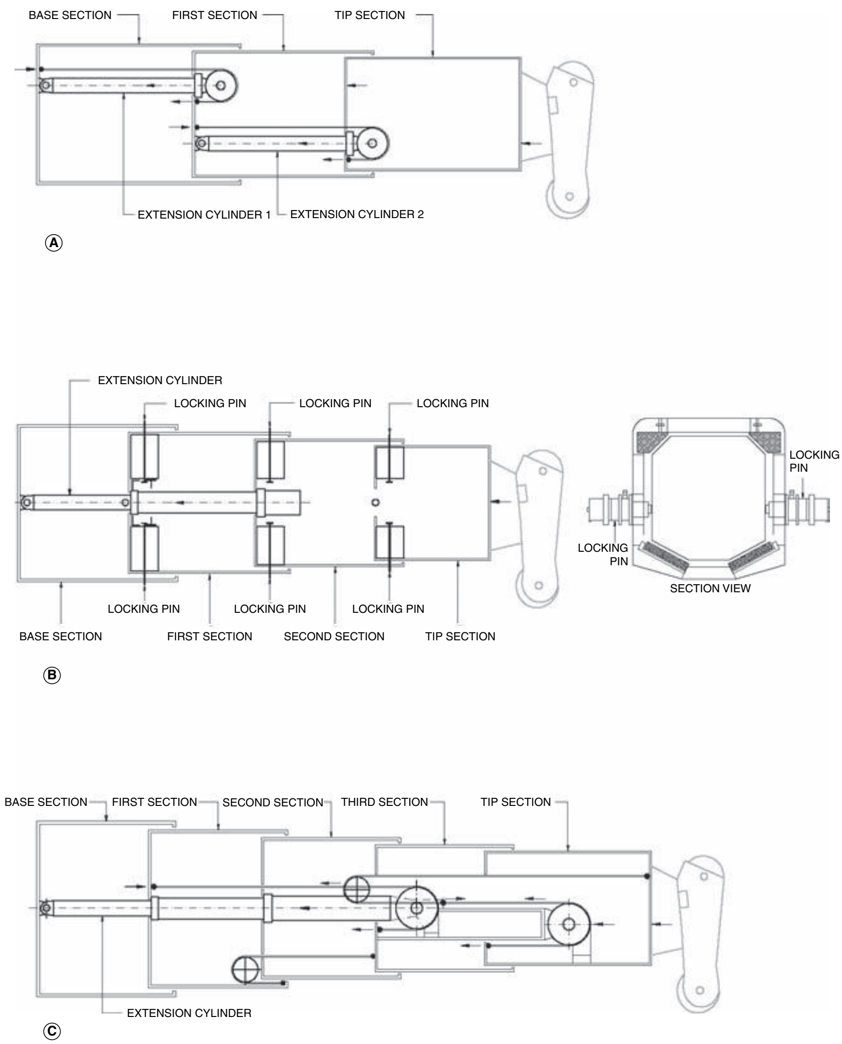


FIGURE 5.40 Three arrangements for telescoping a boom. (a) Light–duty cylinders extend each section. (b) One cylinder telescopes all sections individually. (c) A combination of *boom extension cylinder* and pulleys extend the sections.

travel with load, but its mountings are less sensitive to dynamic effects. To gain maximum advantage from pick-and-carry capability, it is necessary to understand how cranes and loads behave. The operator's manual must be consulted for specific limitations for each particular model.

Wheel-Mounted Cranes

A dual-cab, all-terrain or truck crane, when permitted by the operating instructions to pick and carry, transports lifted loads over the rear.

A rough-terrain 1½ crane carries its loads over the front. All wheel-mounted cranes must travel at very low speeds, generally 1½ mi/h (2.4 km/h) or less. Irregularities in the travel path cause the load to swing and bounce, an effect that is exacerbated by the reaction from the tires. Excessive bouncing, a short-duration dynamic action, would not ordinarily overturn the crane, but could, and therefore should be avoided. This motivates two rules for pick and carry.

1. Modulate travel speed to the condition of the travel path.
2. Keep loads close to the ground, just high enough that they do not strike the ground from the bounce and swing.

The condition of the travel path refers to the uniformity of elevation, or levelness, of all points along the line of travel (high and low spots induce bouncing.) and uniformity of compaction (soft spots induce bouncing, too.). But, if the side-to-side elevations vary, the load will tend to sway sideways pendulum fashion. This could make side stability tenuous and should be controlled by the use of tag lines. Travel path grade must be considered too. Entering onto a grade brings the crane and load combined CG closer to the tipping fulcrum, reducing the stabilizing moment. Downgrades reduce stability for loads carried in front, upgrades for loads over rear. The load, suspended from the boom tip, moves further from the tipping fulcrum, increasing the effective radius and hence the overturning moment. Coupled with bounce, this can prove troublesome.

Fore and aft load movement can be prevented by tying back or snubbing the load to the crane carrier. A chain fall is useful for this purpose because it permits the line to be taken up, pulling the load toward the carrier. Enough take up should be applied to keep the load from swinging and thus avoid shock loading of the snubbing line. Bringing the load closer to the carrier improves stability by reducing the overturning moment, but this should not be done for the purpose of increasing capacity, only for eliminating load swing.

Crawler Cranes

Pick-and-carry limitations are less stringent for a crawler crane than for a wheel-mounted crane. The crawler base offers a more rigid and stable platform that is inherently suited to carrying loads during travel. However, limitations vary from manufacturer to manufacturer and model to model; it is necessary to refer to the operator's manual for explicit data. Some crawler cranes are permitted to travel with full-rated load, loads over front, rear, or side, and up or down grades, but typically there are restrictions on travel speed and travel path levelness, and perhaps boom length as well.

A load should be snubbed to the carbody, just as it is secured to the carrier of a wheel mounted crane. This prevents fore and aft load swing when the boom is over front or rear. But when a load is over the side, there will be a tendency for load swing in the direction of travel that cannot be restrained by snubbing. The swinging load side pulls the boom, this could pose an unsatisfactory structural risk.

When a near-capacity load is carried over the side and the travel path surface is not ideal, snubbing should be done by other means. Front and back tag lines parallel to the travel direction are one possibility. For heavier loads, it may be necessary to snub taglines on front-end loaders or some similar heavy equipment. The anchorages may need to move as travel advances.

A crawler crane is preferred to travel with its load over the rear—the drive sprockets end—and backward with respect to the travel direction. The exceptions are travel with a light load or on a firm support surface. The operator does not see the travel path and must be controlled by a signalperson. With a substantial load on the hook, the rear end of the tracks is more heavily loaded than the front. The front could in fact be free of load. This set of conditions makes travel movement smoother on a soft yielding surface.

Alternatively, if a substantial load is carried in front of the crawler, that end digs into the earth, making for greater resistance to movement and promoting less uniform motion. A turning maneuver will cause the heavily loaded end of the tracks to gouge into the earth (Figure 5.24), sometimes fouling the treads and track rollers. Though carrying over the rear makes for smoother travel, to work that way the operator overcome the instinct to carry the load in front.

Slopes and grade changes along the travel path can be troublesome and should be avoided if possible. First, side-to-side levelness must be maintained within tolerance, which should be set tightly. The crane has no deficit of power to move with the load on a slope. Load control is the concern; that concern grows with the length of the boom. With its long rigid side frames, a crawler crane cresting a hill can tilt forward rapidly. For some moments until it recovers, the crane is unstable. For example, travelling across a 1% change in grade for a crane with 20-ft (6.1-m) bearing length and a 300-ft (91.4-m) boom, will cause the boom tip to move about 3 ft (910 mm) and yet more from elastic displacements. With the load either front or rear, the snubbing line could experience a strong shock and break. With the load over the side, severe *side loading* could induce unwanted swing or, much worse, collapse the boom.

An abrupt crest should be cut down to soften the grade transition. Even a mild crest should be approached slowly and with great caution.

5.11 Multiple-Crane Lifts

When two or more cranes are used to lift a single object, the risk increases exponentially. Consideration of risk makes clear why multiple-crane lifts should be avoided if possible. The cranes interact with one another, and a failure of one is likely to cause all to fail. By nature, a pick involving more than one crane carrying the load simultaneously is a critical lift.

Multicrane lifts require formal planning. The scope of planning and the risk control measures that need to be employed depend on the circumstances of each operation and are matters of judgment and experience. Although most decisions must be made case by case, there are a few rules that should be considered as absolute. The purpose of the discussion that follows is to provide guidance in planning multiple-crane lifts and in making risk control decisions.

Risk control measures, as used herein, refer to all of the actions taken to keep the lifting operation within limits established by the crane manufacturer, the operation planners, and industry or governmental authority standards. Those actions could include such things as crew training; briefing and dry runs, establishment of communication methods and protocols; preparation of lift plans and monitoring of their implementation; site, soil, and level checks; means used to coordinate and control crane movements; load-weight verification and verification of weight sharing between cranes; establishment of wind limits and preoperation weather checks; risk analysis and contingency planning; equipment inspection or certification; and barricading and personnel controls.

The Absolute Rules

1. A single crane lift at full capacity is usually better than a multiple-crane lift at less than full ratings.
2. On multiple-crane lifts, the planners, field supervisors, and crane operators must be experienced and seasoned individuals.
3. If a multiple-crane lift cannot be avoided, use as few cranes as possible. Avoid four-crane lifts on rigid loads unless two of them can be coupled or equalized to create the equivalent of three cranes. Four cranes or more should not be attempted unless full engineering support and rigorous risk control measures are employed.
4. A formal, written lift plan is advisable for multiple-crane lifts. The plan, at a minimum, should set out the position of each crane and the load at the start of the operation, as well as the crane lift radii and applicable lift loads, the movements to be made by each crane including sequence and radius changes, and the operational and risk control measures to be used. The crew should be thoroughly briefed.
5. The weight of the lifted load and its center of gravity (CG) must be known with reasonable accuracy early in the planning process. The distribution of load to each crane, and the resulting effect on their supports, should be determined for

each phase of the operation. The weights of hook blocks and lifting accessories must be considered.

6. Crane load lines must be kept plumb within set tolerances at all times throughout the course of every multiple-crane lift. The selection of crane movements to be made during the operation and the risk control measures chosen to protect the operation must be considered together with that requirement.
7. Normally, complex motions should not be performed. Avoid hoisting and swinging at the same time and hoisting and luffing. No motion should be combined with travel. The cranes can hoist together as needed, and any time that one crane swings, travels, or luffs it will be necessary for the other crane(s) to move synchronously in order to keep the load lines plumb.
8. In most instances, only one crane should have a locked swing with the others freewheeling. This action will limit *side loading*.
9. Cranes can be used to near full capacity on multiple-crane lifts that are fully engineered. Also rigorous operational control and risk control measures must be established. In more typical circumstances, loads should be 75% of rated load or less.
10. A jib, especially one with a substantial offset angle, is not generally suitable to be used in a multiple crane lift because of its heightened vulnerability to side loading failure.

Two-Crane Lifts

The ideal situation to expect when using two cranes would be a lift of uniform weight, such as a bridge girder, by identical cranes which are symmetrically attached, that is, hooked on the same distance from each end of the girder. Each crane would be equally loaded. With their load lines kept plumb, they would remain equally loaded while the load is in the air; they will be safe provided the cranes are both kept within their rated radii, all other normal crane operation measures having been taken (Figure 5.41). The load should be kept nominally level, although rapt attention need not be given to this since the load will not shift from one crane to the other if it goes a little out of level.

When the load is landed, the weight distribution between cranes will not change if both cranes land their loads at the same time. But, if the crane hook-up points and the landing contact points are not the same, the load can shift to the other crane if one end is landed first. Load shifts of that nature are both predictable and quantifiable during the planning process. The point is that for even the most ideal two-crane lifts, planning is necessary to manage the added risks that are not present when a single crane is used alone.

Figure 5.41 A double truss 290 ft (88 m) long and weighing 148 tons (134 t) being lifted by two crawler cranes with 160-ft (49-m) booms prior to traveling several hundred feet with the load. Including lifting attachments; each crane is carrying 84 tons (76 t), 88% of rated load, during this fully engineered and controlled lift. (*Douglas Steel Fabricating Corporation.*)

For most two-crane lifts neither the load nor the attachment points are symmetrical. The cranes often also differ in either model or configuration (Figure 5.42). Distribution of the load between the cranes will conform to a ratio that follows from the relative position of the load CG between the two crane hooks. The crane closest to the

Figure 5.42 Pick of a space frame using a pair of Manitowoc 16000 crawler cranes equipped with the same front-end attachments but different heavy-lift counterweight arrangements. (*Manitowoc Corporation Inc.*)

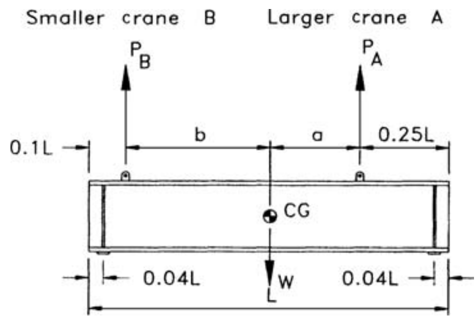


Figure 5.43 A girder dimensioned for determining the load distribution during a two-crane lift.

CG will carry more of the load than the farther crane. In Figure 5.43, the horizontal distance from the CG of the load to the hook of the larger crane A is a, and the distance from the CG to the hook of crane B is b. The horizontal distance between the two hooks is then $a + b$. The load on crane A can be found by taking moments about the hook of crane B.

The moment introduced by the load W must be balanced by the moment of the load P_A on crane A, or

$$WB = P_A(a+b)$$

$$P_A = \frac{Wb}{a+b}$$

Suppose a to be 10 ft (or 3 meters) and b to be 30 ft (or 9 meters). The hooks are then 40 ft (or 13 meters) apart. The load on crane A will be 30/40, or 75 % of the lifted load; the rest of the load, or 1/4 of it, will be on crane B.

If the load is permitted to go out of level, the horizontal distances will become shorter, but the sharing ratio will remain the same. For example, if the load is tilted to the point where a measures 9 ft (or meters), it will be at an angle of about 26° to the horizontal and b will become 27 ft (or meters); the distance between the hooks will then be 36 ft (or meters). The load on crane A will be 27/36, or 3/4 of the lifted load as before. One of the two cranes should have the swing lock disengaged so that the hoist lines stay plumb.

An exception to this behavior occurs when the vertical position of the load CG is much above or below the hook-up point levels with respect to the distance between the hooks or if the hook-up points are considerably offset from one another vertically. For example, if the girder CG is 1.5 ft (or meters) below the hook-up points, crane A will actually carry just over 73% of the lifted load and not 3/4 difference of less than 2% for the extreme case of 26° off level.

Suppose crane A lands its end first. Shifting the right-hand support point from P_A to the end of the girder in Figure 5.37 causes a redistribution of the loads. Taking a as equal to $0.25L$ (which places the CG at the center of the girder), b becomes $0.4L$. When the right end of the girder is landed at its bearing $0.04L$ from the girder end, crane A becomes unloaded. Taking moments about the right end bearing, the moment of the load W must be balanced by the moment of the load on crane B.

$$W(0.5L - 0.04L) = P_B(L - 0.04L - 0.1L)$$

$$0.46WL = 0.86P_BL$$

$$P_B = 0.53W$$

Whereas crane B carried only 38% of the load while the girder was in the air [0.25/(0.4 + 0.25) = 0.38], landing the larger crane's load first increases the load on crane B by almost 40%. Had crane B been near capacity during the lift, this landing sequence could have caused overload and failure. The same danger would occur should crane B try to lift the load first.

The opposite procedure, landing crane B end first, would only increase the load on crane A from 62% of girder weight to about 65%. Clearly, the load lifting and landing procedures must be studied as part of planning, and the lift plan must specify the sequence whenever significant load redistribution can occur.

When the load line goes out of plumb, the effect on the crane could be more severe than the eye would indicate. Strength, stability, or both could be affected depending on the direction of the misalignment. If the hook has been pulled out past the boom tip, which is called *off-lead*, the effect is equivalent to adding load on the hook or booming out to a greater radius; stability is reduced. A multiple-crane lift is usually performed by cranes working at short radii where a small increase in radius corresponds to a large diminution of stability and a corresponding reduction of lift capacity. What may appear as a small off-lead at the start of a lift can become more significant as the load is raised. Stability lessens as the load is raised; a crane can be overloaded with possibly catastrophic results.

As part of planning, out-of-plumb tolerances can be examined. If, for example, the crane shown in Figure 5.44 has an off-lead of magnitude e, the out-of-plumb load line will exert a vertical force component on the boom tip equivalent to load W and a horizontal component of pW where $p = p/h$ and h is the distance from the boom tip to the hook. Taking moments about the boom foot pin F,

$$M_F = W_r + WpL \sin\theta$$

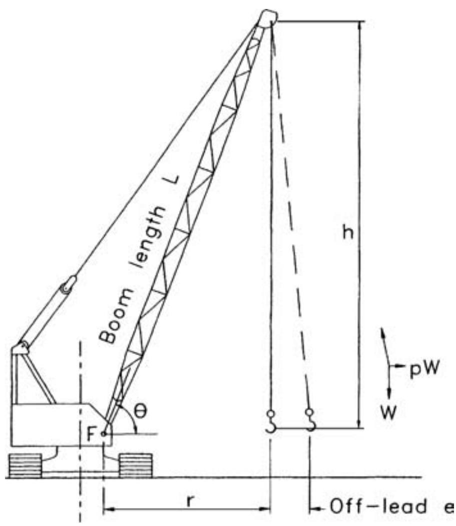


Figure 5.44 A schematic view of a crawler crane annotated for calculating off-lead effects.

where θ is the boom angle and L is the length of the boom. The vertical load effect producing an equivalent moment can be found by dividing through by r, the radius measured from the boom foot.

$$W_{\text{eff}} = W + W\frac{(pL\sin\theta)}{r}$$

Substituting $r = L\cos\theta$,

$$W_{\text{eff}} = W(1 + p\tan\theta)$$

The term in parentheses is the load multiplier that reflects the effects of off-lead. For a 75° boom angle, and several values for p, for example, the term in parentheses becomes

$$1.019 \quad \text{when } p = 0.005$$

$$1.037 \quad \text{when } p = 0.010$$

$$1.075 \quad \text{when } p = 0.020$$

$$1.149 \quad \text{when } p = 0.040$$

If a two-crane lift is being conducted using two identical cranes facing each other with long booms at 75° boom angles, and if the load is initially suspended 200 ft (61 m) below the boom tips with an off-lead of 1 ft (305 mm), $p = 1/200 = 0.005$. The equivalent vertical load will be

$$W_{\text{eff}} = W(1 + 0.005 \times \tan 75°) = 1.019\,W$$

This reflects about a 2% increase in load effect.

Should the load be hoisted to 25 ft (7.6 m) below the boom tips, the off-lead will remain 1 ft, but p becomes $1/25 = 0.040$. The equivalent load becomes $1.149W$, an increase of about 15%; a load that started out as 75% of rated load becomes effectively 86% after hoisting. Had the boom angle been 80° instead, the increase in effective load would be about 23%. That is because the tangent function increases with boom angle; the higher the boom angle, the more pronounced the load increase effect. Also, the plumbness parameter p directly controls the load increase effect. For any off-lead, that parameter will increase as the load is hoisted.

Operating procedures and plumbness checks can be set up to assist the crane operators in adjusting their booms to bring the load lines to within the predetermined plumbness tolerance, but more importantly, the cranes must possess adequate lift capacity to safely support the effective load.

These concepts show the relationship between crane capacity and out-of-plumb tolerance that must be maintained at the jobsite. Unlike a single-crane lift where the load CG will align itself below the hook, in a multiple-crane lift the load line will inevitably go out of plumb. With the load grounded, as the load line is drawn in on the winch drum, the crane starts to become loaded; the boom hoist and pendant ropes start to stretch, the boom shortens under compressive load, the tolerance in the swing bearing is taken up, and the superstructure and the undercarriage frames elastically distort, all of which results in an increase in radius. Operators compensate for this by starting out with the boom at a shorter radius, but they do this by judgment. Unless judgment is perfect the load line will be out of plumb.

An out-of-plumb load line can also cause side loading. This occurs when the deviation from plumb is near right angles to the boom such as when one crane swings and is permitted to pull the second crane with it. Side loading is a structural condition that could cause a boom to collapse without warning. In U.S. practice, a mobile-crane boom is designed to sustain a side load of 2% of rated load (3% for telescopic cranes), an allowance which is intended to account for the minor dynamic effects of a freely suspended load. Experience has shown that this allowance is adequate for single-crane hook work. However, side loading causes appreciable, nonlinear increases in boom stress. Lateral, or sideways, out of plumb in excess of $p = 0.020$ is therefore very hazardous and can result in boom collapse. It would be imprudent to set the out-of-plumbness tolerance too close to $p = 0.020$, since this would leave no allowance for dynamic side loading, wind, or the crane to be out of level.

The reader may have started to think that perhaps undue emphasis has been placed on out-of-plumb load lines. However, the reason for concern is that almost any action during a multiple-crane lift can lead the load lines away from plumb. Therefore, considering the potential dangers out-of-plumb hoist lines engender, risk control measures must rein them in.

The reader can imagine, as an example, two crane hooks set apart at a given distance from one another with the load aloft and the load lines plumb. One crane then lowers or raises the load while the other holds fast; the load goes out of level and the horizontal distance between hooks becomes shorter. This means that the hooks are pulled toward one another, the load lines go out of plumb. But a moderately out-of-level load is really not critical in this regard unless it induces excessive side load. Crude monitoring by eye or other means can keep the load within 5° of level. If that limit is held, the effect will be insubstantial.

The greater the distance between hooks, the more significant levelness becomes. For a distance of 100 ft (30 m), 5° out of level brings the hooks almost 5 in (122 mm) closer, whereas 10° would bring them 18 in (460 mm) closer.

It is generally a good practice to leave the superstructure of the second crane free to follow the other crane. But when one of the cranes swings, the second crane resists swinging and must be pulled around by the first crane. This occurs even with the low-friction swing bearings used on contemporary mobile cranes. The frictional resistance will cause both cranes to be side loaded as will be evidenced by their load lines going out of plumb after the swing commences. When there is a substantial load on the hook, both cranes should power swing together at very slow speed or each crane should swing separately in small increments that will not cause out-of-plumb tolerances in the load lines to be exceeded.

If two cranes with initially plumb load lines face one another and one of the cranes booms up, the distance between boom tips will increase but the load will keep the hooks at their original horizontal separation. The load lines, however, will now be out of plumb with an off-lead that will act on each crane as though the load were at a greater radius than the boom tip. The radius change will only be equal if both cranes were originally sharing the load equally and if the boom tips were at the same height. However, the stationary crane, the one that did not boom in, will now be at an increased effective radius. For any other condition, each crane will experience a radius change of different magnitude, but the stationary crane is usually the one that will suffer an effect greater than its original radius. The crane operators cannot perceive the radius change effect from their *boom angle indicators*; however, these changes can be calculated using principles of static equilibrium, and tolerances can then be established as part of the planning and risk control measures.

If two cranes are going to travel with a common load, there are three ways in which the movement can cause the load lines to go out

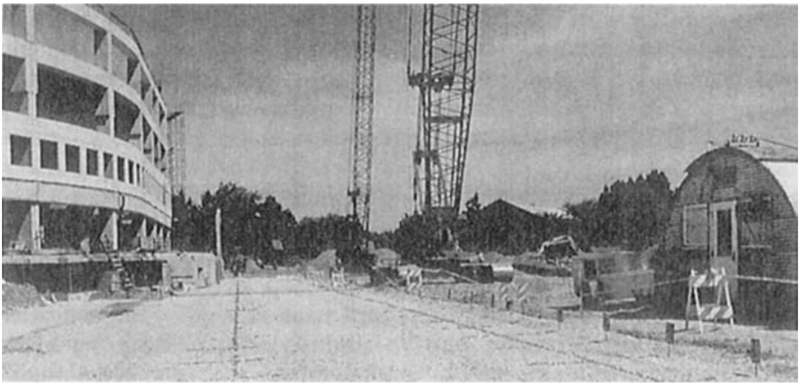


FIGURE 5.45 The travel path for the operation shown in Figure 5.41 has been made of leveled and compacted crushed stone ballast. At right are distance flags used to keep the cranes abreast of one another. A track centerline has been sprayed on the ballast for maintaining alignment, as seen at left center. (*Douglas Steel Fabricating Corporation.*)

of plumb. If the travel path of either crane is not level or if there are "soft spots" that will permit a crawler track or tire to "dig in" and place the crane out of level, the boom tips will change position relative to one another and the load lines will go out of plumb. The same will occur if one crane travels faster than the other and runs too far ahead, or if the cranes are permitted to travel on paths that are not truly parallel (Figure 5.45). If reduced loading levels are specified when travel is required, the more critical effect of these travel irregularities is often side loading of the booms and the danger of structural collapse. Each of the travel conditions may be analyzed, however, and control tolerances established accordingly.

Having presented the numerous ways in which hoist lines go out of plumb, the authors hope that it has become evident to the reader why loads should be held to about 75% of rated lift capacity for most multiple-crane lifts. The 75% figure is an arbitrary number; it is nonetheless practical for the general case though certainly not an absolute limitation. If assumed that at full rated load all of the allowable boom strength is being utilized, little spare strength will remain for the inevitable side loading or off-lead that occurs with multicrane lifts. At 75% of rated load, however, spare strength is available because the reduced hook load will subject the boom to a lower level of direct stress.

A note common to mobile crane rating charts stipulates that the ratings given apply when loads are freely suspended. During multicrane lifts, loads are not freely suspended; because they are attached to a common load, the cranes interact with each other. Lifted loads must be less than rated loads, accordingly, but the amount of reduction is a matter of judgment.

The 75% limit is not absolute; it can be exceeded if lift conditions are sufficiently well controlled or if the operation produces little tendency for the hoist lines to go out of plumb. A suitably controlled operation is one in which meaningful, realistic risk control measures are in place to keep conditions and crane motions within predetermined limits. Predetermined limits apply to the portion of the lifted load going to each crane as the operation proceeds and to the off-lead or side loading to which each crane will be exposed—a function of how much the hoist lines will be permitted to go out of plumb. Risk control measures include establishing a combination of engineering requirements and field monitoring procedures to keep crane loads to known values and to maintain hoist lines plumb to within stipulated tolerances.

Working motions of a telescopic crane do not produce greater side loading than other cranes, but by nature it is more vulnerable to side load effects than a latticed boom. On the latter the pendants act as a restraint. That is why, in U.S. practice, a telescopic boom is designed for a 3% side load rather than the 2% used for a latticed boom. In spite of a greater design allowance, the vulnerability of a telescopic crane should oblige some additional caution in its use for multiple-crane lifts.

In short, the capacity reduction used for a multicrane lift should be a realistic countermeasure enacted to compensate for the inability to achieve the freely suspended load condition, which is a requirement for lifting full rated loads specified on all mobile-crane rating charts.

Tailing Operations

Vessels, or for that matter many loads that are longer than wide, are shipped lying flat and must be raised to a vertical orientation at the final destination. This procedure of setting the piece upright is often called *tripping.* When loads are tripped, much attention is paid to the main crane attached to the top of the vessel, but the smaller crane at the bottom, the *tailing crane*, is too often given short shrift and undersized. The tailing crane is also critical to the success of the lift, and there are subtleties to tailing work that should be better understood by many planners and operators.

Earlier in this section, there was a discussion of how load can shift between a pair of cranes lifting a symmetrical load, and how relatively insensitive the cranes will be to a moderately out-of-level load. Now, however, the reader should consider a load going out of level in the extreme—a full 90° rotation. Minor effects in lifting a nominally horizontal load become important effects during a tripping operation.

In a usual tripping operation, the main crane hoists the vessel as the tailing crane lifts the bottom slightly while shifting it horizontally. The horizontal distance between the hooks must be constantly adjusted to hold off-lead or side loading to within an acceptable limit. As mentioned earlier, the distribution of load between the cranes will remain constant as the load is tripped if the CG and crane attachment points are on a common line. This might be called the ideal case, but it suffers from one shortcoming: for tripping, it doesn't work in practice. As the load approaches vertical, the boom tip of the tailing crane must go inside the load if the hoist line is to remain plumb. There are very few loads indeed for which that is possible. The tailing crane attachment point should therefore normally be at or above the upper edge of the load, while the load is still horizontal, so that the boom tip will remain clear when the load is vertical.

For vessels, the main crane is commonly attached to lift points, usually *trunnions*, mounted on each side of the load along the longitudinal centerline. The tailing crane is attached to a lift point that will be close to the bottom of the load when it is tripped vertical. At the start of the lift the load distribution is determined by the horizontal distance of the load CG from each of the lift points. As the angle of the load centerline to the ground increases, the portion of the load carried by the main crane gradually increases. The shift of the load to the main crane increases at a faster and faster rate as the load approaches vertical.

This load sharing between the cranes can be demonstrated by referring to Figure 5.46. With the vessel horizontal, as in Figure 5.46*a*, the load M on the main crane is found by taking moments about the tailing crane attachment point.

$$M = \frac{Wb}{a+b} \qquad T = W - M$$

where W = weight of the vessel
M = the part of the load on the main crane
T = the part of the load on the tailing crane
a = the horizontal distance from load CG to main hook-up point with the load horizontal
b = the horizontal distance from load CG to tailing hook-up point with the load horizontal.

The main crane hoists the vessel as in Figure 5.46*b* until the centerline makes an angle ϕ with the horizontal. Taking moments again creates an expression for M.

$$M = \frac{W(b\cos\phi + c\sin\phi)}{b\cos\phi + c\sin\phi + a\cos\phi}$$

$$= \frac{W(b + c\tan\phi)}{a + b + c\tan\phi} \tag{5.40}$$

where c is the vertical offset of the tailing hook-up point from the load CG when the load is horizontal.

When $\phi = 0°$, $\tan\phi = 0$ and Eq. (5.40) is identical to the equation derived for a horizontal vessel. When $\phi = 90°$, $\tan\phi = \infty$, making the

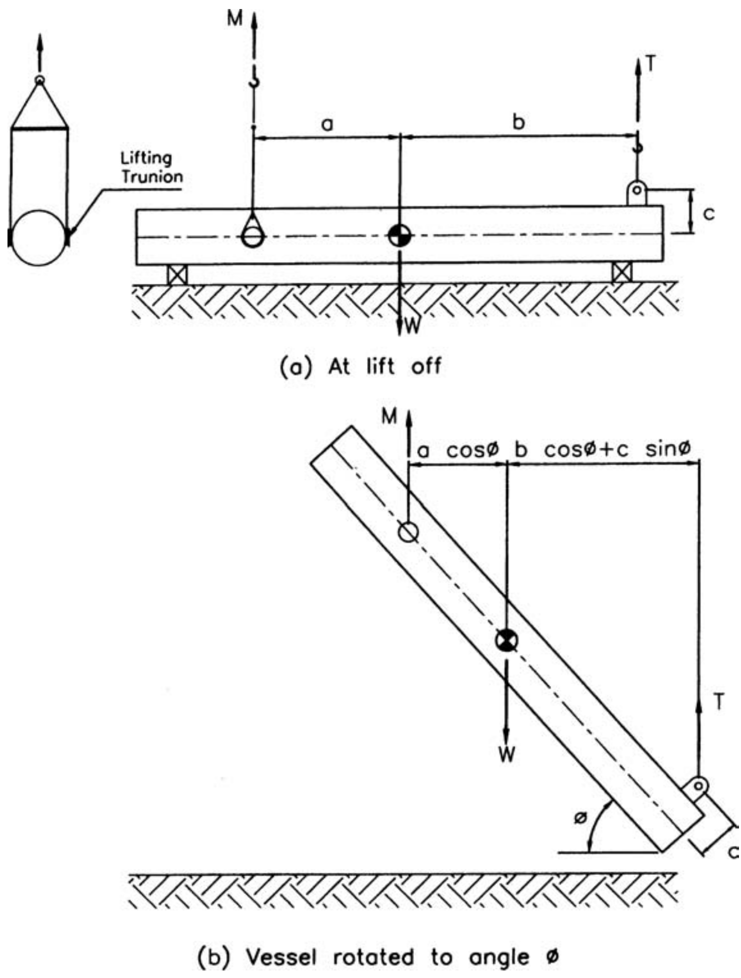


FIGURE 5.46 A vessel annotated to demonstrate load distribution changes that take place between a main and tailing crane as the load is tripped. The main, or upper, crane lifts straight up while the tailing crane must move toward it as the load is lifted. Figure 5.51 shows a lift that follows this configuration.

terms in parentheses in the numerator and denominator both infinite and therefore equal and indicating that $M = W$ or that the main crane takes the entire load. The tangent function increases rapidly approaching 90°. The effect of this is that as the vessel approaches vertical, the load on the main crane will quickly approach vessel weight, while the tailing crane load rapidly drops concurrently. Therefore, for smooth load transfer without causing the load lines to go too much out of plumb, the last 5° or so of rotation should be accomplished by slowly booming out with the tailing crane and slowly lowering the tailing hoist line as needed.

Another possible lift-point arrangement has the lift attachments for both cranes at the upper edge of the load when horizontal as shown in Figure 5.47*a*. The crane loads P_a and P_b can be found by taking moments as done previously.

$$P_a = \frac{W(b\cos\phi + c\sin\phi)}{a\cos\phi - c\sin\phi + b\cos\phi + c\sin\phi}$$

$$= \frac{W(b + c\tan\phi)}{a+b} \qquad (5.41)$$

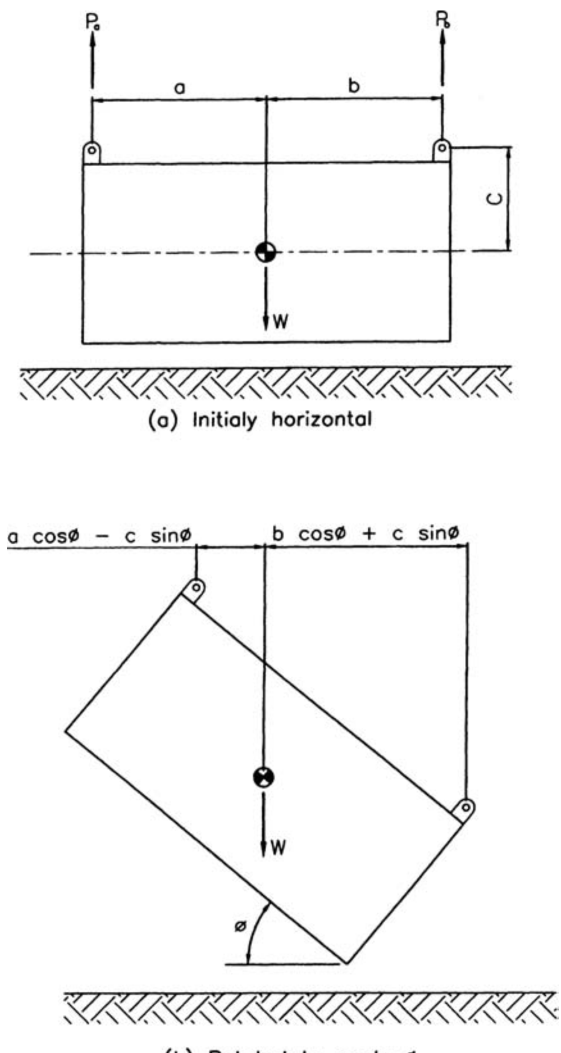


Figure 5.47 An arbitrary load annotated for calculating load distribution changes as the load is tripped by two cranes with lift points at or near the top surface.

From Eq. (5.41) we find that $P_a = W$ when $\tan \phi = a/c$; this tells us that $\phi < 90°$. Although perhaps obvious from examination, that value for ϕ is actually the angle at which the crane attachment at the high end will be directly above the load CG, or in other words, the angle of the load when the tailing crane is released.

This situation, however, poses problems and potential dangers. The tripping operation cannot be easily completed unless the corner of the load is lowered to the ground, which may cause damage to the lifted piece. In order to complete the tripping in the air, it will be necessary to use horizontal (or near-horizontal) lines at the top and bottom of the load as pulling and holdback lines. There are three cases that need to be examined:

1. The CG lies on the diagonal line from the higher lift attachment to the lower corner of the load. When that line is vertical, the CG will be directly above the corner and the load will be in balance. The load can be tripped only if pushed or pulled further, but the second crane must then furnish a balancing force to rotate the load to vertical.
2. The load CG is above the diagonal line. When the CG is directly below the high attachment point, rotation can continue no further unless the corner of the load is placed on the ground or, alternatively, helper lines installed to complete the tripping in the air. But, if placed on the ground, the load will tend to be disposed toward a vertical position unless restrained and controlled by the second crane.
3. The CG of the load lies below the diagonal line. When the load CG has been rotated until directly below the high attachment point, and the corner placed on the ground, the load will tend to fall back to the horizontal position. The second crane cannot prevent this. Therefore, additional equipment is needed: a horizontal pulling line to force rotation to continue to ward vertical and a restraining line to limit side load or off lead. But, when the load CG passes the corner, the load will tend to fall forward to vertical. It must be restrained and controlled by the second crane or the helper lines.

Some lifting arrangements should be avoided unless special precautions are taken. For instance, when the lift points are positioned such that the load CG is higher than both lift points, the load will tend to be unstable and could roll over.

The last potential lift-point arrangement is the best; it is shown in Figure 5.48. When load configuration and other conditions make this arrangement feasible, it has three main advantages to offer.

1. Neither crane will be required to support the entire load weight during any part of the operation.

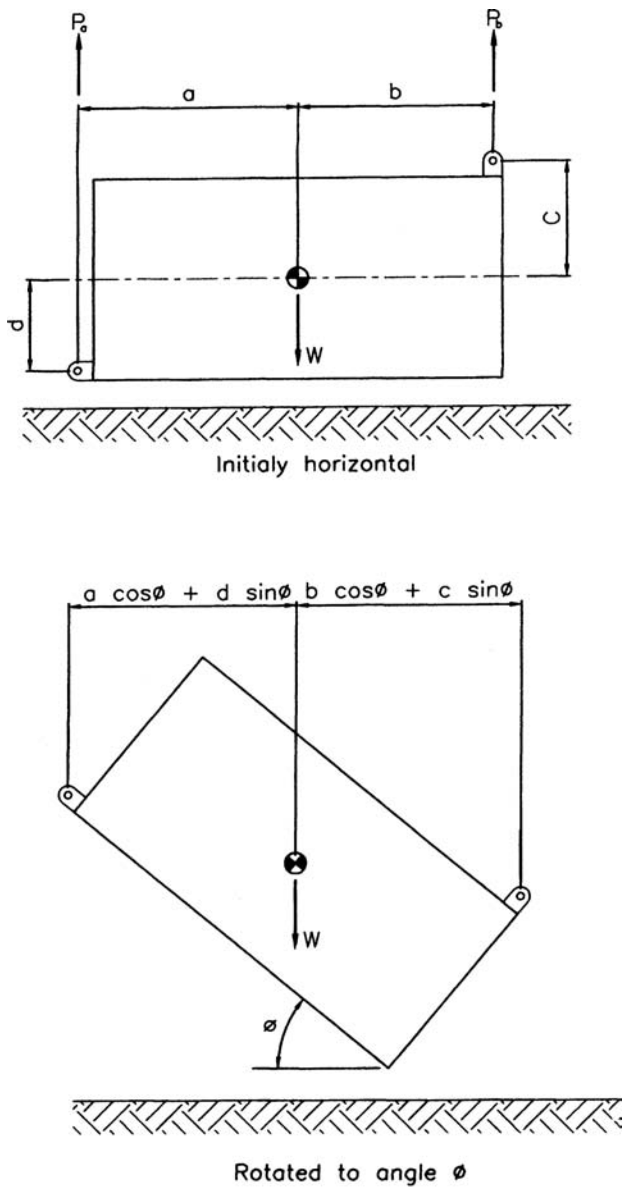


FIGURE 5.48 An arbitrary load annotated for calculating load distribution changes as the load is tripped by two cranes one lifting from at or near the top surface and the other from a low lifting point.

2. The load can be tripped in the air.
3. Both the main and tailing crane will always be well loaded, and therefore the load will be stable and controllable.

Using the same procedures as earlier employed

$$P_a = \frac{W(b\cos\phi + c\sin\phi)}{b\cos\phi + c\sin\phi + a\cos\phi + d\sin\phi}$$

but the only points of interest are for $\phi = 0°$ for a horizontal load and $\phi = 90°$ for a vertical load. Inserting those values,

$$\begin{aligned} P_a &= Wb/(a+b) \quad \text{when} \quad \phi = 0° \\ P_a &= Wc/(c+d) \quad \text{when} \quad \phi = 90° \\ P_b &= W - P_a \end{aligned} \tag{5.42}$$

During tripping, P_a will not exceed the larger of the values from these equations; therefore no other values need to be calculated.

Example 5.11

A vessel weighing 150 tons (136 t) must be tripped to vertical. With the vessel horizontal, the main crane will be attached to the load 90 ft (27.4 m) from the CG, while the tailing crane will be 125 ft (38.1 m) from the CG. The vessel CG is on the longitudinal centerline. It is possible to place attachment points either 6 ft (1.8 m) above the centerline or 4 ft (1.2 m) below. How much load will the main and tailing cranes have to lift for various attachment point arrangements?

Solution

1. Try an arrangement as shown in Figure 5.46, with the main crane attached on the centerline and the tailing crane above it. Using Eq. 5.45, when the vessel is horizontal, the tangent terms zero out and

$$M = \frac{150 \times 125}{125 + 90} = 87.2 \text{ tons } (79.1 \text{ t})$$

$$T = 150 - 87.2 = 62.8 \text{ tons } (57 \text{ t})$$

When the vessel is vertical, the main crane will support the entire load and the tailing crane will be unloaded.

2. Check the set up of Figure 5.49, with both lifting attachments at the top. From Eq. 5.41, with the vessel horizontal, the tangent term will drop out and the results will be as previously. The vessel CG will be directly below the main crane attachment point when

$$\phi = \tan^{-1}\frac{90}{6} = 86.2°$$

At that angle, the base will not be below the CG (125/tan 86.2 = 8.3 > 6), or in other words, the load CG is below the diagonal line from the upper lifting point to the lower corner. The vessel cannot be tripped to vertical without additional help. A holdback will have to be attached at the bottom and a pulling line in the opposite direction near the top. The maximum load on the main crane will be vessel weight; on the tailing crane the maximum will occur while the vessel is horizontal.

3. Try the arrangement of Figure 5.48, where the main crane is attached below the centerline and the tailing crane above. From Eq. 5.42, when the vessel is horizontal, the cranes will share the vessel weight as in the examples. But when the centerline is vertical,

$$M = \frac{150 \times 6}{6+4} = 90 \text{ tons } (81.6 \text{ t})$$

$$T = 150 - 90 = 60 \text{ tons } (54.4 \text{ t})$$

	Main Crane	Tailing Crane
Part (1) Fig. 5.46	150 tons (136 t)	62.8 tons (57 t)
Part (2) Fig. 5.42	150 tons (136 t)	62.8 tons (57 t)
Part (3) Fig. 5.48	90 tons (81.6 t)	62.8 tons (57 t)

Summary of Results: Maximum Crane Loads

Three-Crane Lifts

There are at least three ways in which three cranes can be arranged to lift together. The chief issues that must be addressed are the means for attaching the crane to the load and the positions of the cranes with respect to one another. The characteristics of the load may drive these decisions.

The first arrangement is one already described for two crane picks: tripping a long slender vessel or other load. But in this instance two cranes are used to lift the heavy end, which is usually the top. Commonly the pair of cranes is positioned so that their booms are perpendicular to the long axis of the load. When the hoist lines go out of plumb, cranes arranged this way will be side loaded. In other respects, the loading corresponds to the similar two-crane pick. Side loading of the main cranes is matched by an off-lead on the tailing crane.

Figure 5.49 illustrates the lifting arrangement. Using the dimensions in the figure, if the vessel is assumed to weigh 196 tons (177.8 t), the load on each crane can be calculated. Taking moments about the tailing crane attachment, the upper cranes will each lift

$$1/2 \times 196 \text{ tons} \times 50/70 = 70 \text{ tons } (63.5 \text{ t})$$

so that the tailing crane must carry

$$196 - 2 \times 70 = 56 \text{ tons } (50.8 \text{ t})$$

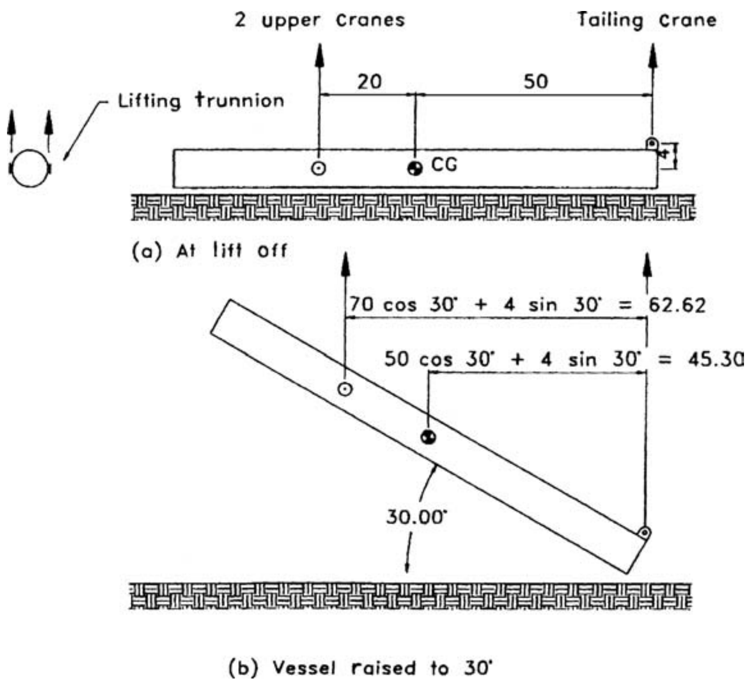


FIGURE 5.49 A vessel annotated for a typical three-crane tripping operation. It is similar to Figure 5.46, except that two cranes are at the top end.

During the course of the lift, when the vessel has been raised until it is 30° to the ground (Figure 5.49*b*), the distribution to the upper pair of cranes becomes

$$1/2 \times 196 \times 45.30/62.62 = 70.9 \text{ tons } (64.3 \text{ t}) \qquad \text{each}$$

and the tailing crane now carries

$$196 - 2 \times 70.9 = 54.2 \text{ tons } (49.2 \text{ t})$$

The shift of load from the tailing to the upper cranes has nothing to do with the load going out of level; it is simply a consequence of the tailing crane attachment point being located above the trunnions. It is easy to verify that with lift points at the same elevation, there will be no shift in load. To do that, the reader can eliminate the $\sin 30°$ terms in the dimensions of Figure 5.49*b*.

A similar arrangement may be used to raise and position a large girder, but the similarities are superficial. For two cranes hoisting from trunnions, only nominal synchronization is needed; if one hook runs a little ahead of the other, the load will turn in the trunnions, but the weight distribution will not change materially. Girder lifts, however, do not have the benefit of trunnions to equalize the two sides. The lift attachments will be at the girder flange, well above the girder CG. If one hook gets ahead of the other, there will be a shifting of load. An equalizer beam should be used to prevent this occurrence. The distribution of the load between the single crane at one lift point and the pair of cranes at the other will then be similar to the two-crane lift.

The second three-crane arrangement is used for loads that are wide with respect to their length. Here, three discrete attachment points are used, in a plan view the three hook-up points make a triangle regardless of the shape of the load. The distribution of load among the three cranes is determined from the position of the load CG with respect to each of the attachment points. During the lift, the load should be kept nominally level, but no meaningful load shift will occur unless the load is seriously out of level, the CG of the load is considerably above or below the attachment points or the attachment points are at different levels.

The distribution of loading using a triangular three-crane lift can be found for the machine shown in Figure 5.50 by using the

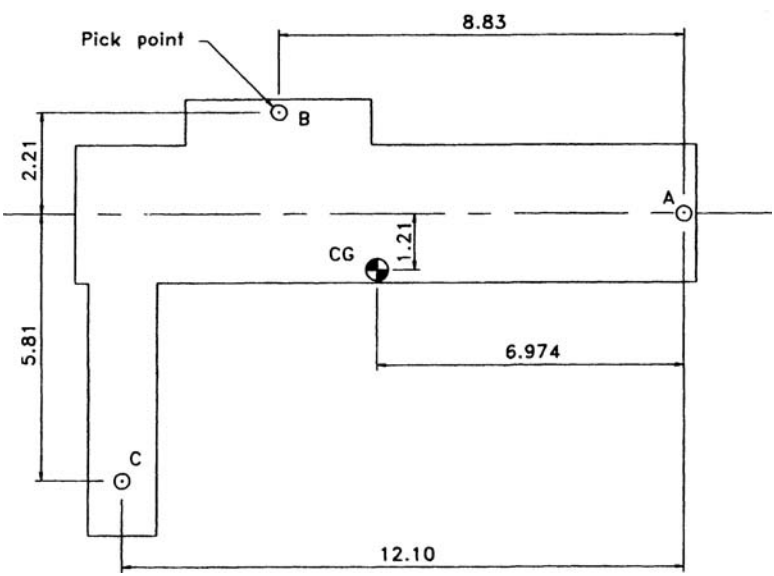


Figure 5.50 Plan view of a machine and its pick points *A*, *B*, and *C*, dimensioned for calculating load distribution.

conditions of equilibrium; the sum of the vertical forces and of the moments must equal zero. Assuming that the machine weighs 141,000 lb (64 t), the vertical force condition gives

$$P_A + P_B + P_C - 141\ \text{kips} = 0$$

Moments to the right of lift point C yield

$$(12.10 - 8.83)P_B + 12.10P_A - 141(12.10 - 6.974) = 0$$
$$3.27P_B + 12.10P_A - 722.77 = 0$$

and moments above point C produce

$$(5.81 + 2.21)P_B + 5.81P_A - 141(5.81 - 1.21) = 0$$
$$8.02P_B + 5.81P_A - 648.60 = 0$$

Those three equations containing three unknowns can be solved simultaneously and will yield equal hook loads of 47,000 lb (21.3 t) at A, B, and C.

The third three-crane arrangement is linear: The cranes are positioned along the length of a long load such as a girder or section of pipe. Unlike the other two arrangements, the distribution of the load between the cranes is indeterminate unless the load is very limber. For the most part, the load on each crane will depend on the actions of the operator, preferably with the help of a load indicator. The load on the middle crane can vary from nothing to almost the entire load weight. When this arrangement must be used, the cranes should be

1. Rated well above the expected load
2. Equipped with load or load-moment indicators in good working order
3. Reeved with multiple parts to permit slow raising and good control
4. Operated very slowly with the indicated load kept within a predetermined range

For three-crane lifts all of the precautions given for two-crane lifts are applicable. As with other arrangements, if a load line goes out of plumb, one or more of the crane booms will be side loaded.

Four-Crane Lifts

The problem with a four-crane lift, where each crane is attached at its own corner of the load, can best be visualized by comparison with a common kitchen table. If a four-legged table is rigidly built, it will almost always rock back and forth because most floors are not dead level and true. That cannot occur with a three-legged table; it always sits firmly.

When four cranes lift a rigid symmetrical load, they cannot be perfectly synchronized and the load will tend to rock like a four-legged table. But the implication for the cranes is that two of the cranes, diagonally opposite to one another, can end up carrying almost the entire load. If two cranes must be capable of sharing the whole load, why use four cranes? If neither pair of diagonal cranes can hold the entire load, sharing can only take place if one or more cranes yield by partially tipping or severely deflecting.

In order to get four cranes to predictably share a rigid load, when each crane is at a corner, two of them must lift by means of an equalizer. In that way the cranes and load will interact as a three-legged table; the load on each crane will become determinable even if the lifted load is unsymmetrical. Without an equalizer, the crane loads are indeterminate. If the four-crane lift cannot be avoided, each crane should be chosen so as to possess excess capacity, equipped with a load indicator, and reeved with extra parts. The lift must then be carried out slowly so that the operators have time to manipulate their hoist controls and keep each crane loaded to within a range of preselected values.

For some loads, pairs of cranes can lift by means of equalizers and the operation then resembles a two-crane lift in many respects. But, when four cranes are arranged in a line for lifting, the same problems prevail as for three cranes similarly arrayed. For this setup, and any time equalizers are not practical, the operating rules given under three-crane lifts should be followed. The same is true for all lifts requiring more than four cranes.

Most arrangements of three cranes and all arrangements of four or more cranes are indeterminate unless equalizers are used. Indeterminacy means that the distribution of load between the cranes cannot be calculated using on statics. Those lifts require strict adherence to planned procedures and controls tailored to the equipment selected for use, the equipment arrangement, and the specifics of the work that needs to be done.

Most of the dangers discussed either do not exist or are of limited importance for single-crane operations. Any of them, and certainly their combinations, compose the envelope of risk that must be addressed for multiple-crane lifts. During planning, the movements and operational steps required to accomplish the particular lift are

Figure 5.51 A two-crane lift of the configuration shown in Figure 5.46. The main crane, a Liebherr LR 1400 with a heavy *lift attachment* is attached to a *trunnion*. The *tailing crane* is a Liebherr LTM 1500. (*Bay Crane*.)

determined and the risks associated with each step identified. Once found, control measures are established to bring each risk, and the set of risks, under management. With defined procedures, rationally set tolerances controlled in the field, and a clear supervisory chain of command and reporting, multiple crane lifts can be viable, productive, and efficient lifting procedures (see Figure 5.51).

CHAPTER 6

Tower-Crane Installations

Tower cranes have become symbols of urban development. A count of tower cranes on the skyline is an instant gauge of a city's economic health. European in origin they are now manufactured in Europe and Asia with perhaps a few made in North America and Australia.

Most tower cranes are installed freestanding on simple above-ground, support bases weighted with *ballast* blocks. These basic installations require minimal site engineering and suffice to erect low-rise buildings.

In past decades, North American contractors were slow to adopt tower cranes, but they now use them readily, though not quite as readily as contractors in other continents. Tower cranes in North America tend to be used for heavy construction and high-rise work. These applications demand expertise for planning and installation design.

This chapter provides a detailed view of common tower-crane installation methods and an overview of some solutions to more difficult installation problems. The intention is not to present cookbook solutions but rather to give sufficient background for resourceful engineers and planners to implement their own solutions. Self-erecting tower cranes are not included in this chapter because the installation issues with these machines are more in line with that for mobile cranes.

There is an emphasis on storm winds in this chapter because they so often are a controlling factor for installation design. The authors have long felt that the geographic variation of wind is an issue that has not been adequately treated by manufacturers and suppliers of tower cranes. Recently this shortcoming has gotten more recognition, and there is some evidence of changed thinking. Tower-crane users in hurricane-prone and typhoon-prone places should take special note. Chapters 3 and 6 combined provide analytical tools for adapting tower cranes to high storm-wind areas.

With the exception of the self-erecting type, tower cranes must ride out storms. During a storm, the unmanned crane is exposed to a much different system of loads than when working. When a tower crane is left unattended, the jib is left free to slew. With the introduction of wind, the jib will undergo *weathervaning*. The counterweights will point into the wind; the back moment acts in opposition to the wind moment. For design, this is described as the *out-of-service* condition.

The *in-service* condition describes the working crane; it includes the effects of lifted load, impact, moderate wind, and *slewing torsion*. In-service loads are determined principally by characteristics of the equipment while out-of-service loads are determined by an interaction of the equipment with the local environment. The design for the installation of a crane with a high mast located on the Gulf Coast obviously will be governed by out-of-service loading, whereas a crane with a stout mast surrounded by tall buildings in Times Square might not be. In general, both load regimes should be investigated for each installation.

6.1 Introduction

Some decades ago, a construction superintendent, a graduate civil engineer, thought he would save his company some money by designing the foundation for a freestanding tower crane by himself. In order to provide hook coverage over the entire job, the foundation was to be shoehorned in among a cluster of building footings. By necessity, the crane was to be mounted considerably off center of the odd-shaped footing block.

Fortunately, before the crane was erected, the superintendent got cold feet and decided to have the design confirmed by an engineer, who was the original author of this book. When stability was checked, it was evident that a wind of relatively low velocity would overturn the crane because of the short fulcrum distance on one side. The author devised a cost-efficient solution that would have augmented the poured footing with some steelwork and mass concrete. When his design arrived at the jobsite, however, the superintendent called frantically with the news that an erection crew and mobile crane were due to start work in 2 days' time while the steelwork specified would take about a week to procure and fabricate. The crane installation could not be delayed, as it was the key to keeping the entire project in motion.

Now cost-effectiveness had to be thrown to the winds and an alternative scheme, previously explored but rejected as too costly, was put to use. The new plan required a crude placement of mass concrete needing no custom materials or processes. After explaining the concept to the superintendent and being informed that there was a backhoe at the jobsite, he was instructed to "set up the backhoe and start digging a trench on the long end of the footing. I'll call you back in about 2 hours and tell you how wide and deep to make it. By then the design will be finished."

And that is exactly how it happened. After 2 hours he was given the details of the ballasting, and the work was then completed in sufficient time for the crane erection to go ahead on schedule. Despite the failed attempt to save money by doing it himself, instead having to plan on-the-fly and dispense with economy, the superintendent and his employer were lucky to have a happy ending. But for the superintendent seeking expert advice at the last minute, this tale might well have ended differently with a delay or a disaster.

By its nature, construction management is a mix of meticulous planning and quick decision making. Preparing for a tower crane is an activity that belongs in the first category. The necessity for careful preparation grows with the size and complexity of the work; projects with multiple cranes and cranes that jump, for example, require a high level of planners' attention. Though the unexpected turns of a job occasionally scuttle the best of schemes, a strong planning discipline is adaptive and can help a contractor out of a jam.

6.2 Planning and Preparation

Picking a suitable model and configuration for a tower crane goes hand in hand with its placement on a site. By nature, a tower crane is nailed down to a spot or, if it is a *rail-mounted crane*, confined to a path. Advantages of a well-selected machine can be undone by a poorly chosen location. There is a great investment in setting up a tower crane: the cost of a foundation, erection, dismantling, electrical hookup, and time in the construction schedule devoted to these tasks. Once installed, there is little choice but to stay put. If the crane is a climber, it will be supported on a host structure, and there are considerations of how the crane will interact with its host.

Where two or more tower cranes are to operate at the same site, the cranes must be arranged so that there is no possibility that the jibs will collide (Figure 6.1). Even when jibs are not in direct conflict, a considerable effort may be required to harmonize the operations of multiple cranes. For *climbing tower cranes*, the interaction changes with the progression of the work.

Crane Selection

Configuration, reach, and lifting capacity are the primary characteristic of a tower crane. The next important traits are the speed of its various motions and the degree of control that the operator has over these motions. Means of adapting to the needs of the site—accessories such as climbing gear or a *traveling base*—round out a tower crane's capabilities.

In the perfect construction world, an optimum crane location is pinpointed and a crane with suitable characteristics is procured to

Figure 6.1 Tower cranes with overlapping operating areas require careful planning, field coordination, and operator attention. The site in the foreground has a concrete placing boom that must not be allowed to interfere with the two nearby tower cranes. (*Morrow Equipment Company.*)

suit the location. In the real world, there are a finite number of cranes available, and there are reasons to choose one or not to choose another; so compromises are made. Owning a tower crane is an advantage to a user that may outweigh all other considerations.

For a user that is not committed by ownership to a particular machine, here are some reasons that might be considered in making a selection.

User preference. Comfort with a vendor and previous experience with a particular crane are subjective criteria, but hardly to be ignored. A tower crane is a key piece of equipment on a project; downtime is costly, and dissatisfaction with a piece of equipment may not be curable. Ideally, a crane user should be able to count on a history of reliable performance from the machine and strong support from the vendor.

Familiarity with a crane is an advantage, too. The reasoning goes beyond "better the devil you know than the devil you don't." A familiar machine will perhaps require less training and less time to erect. Hardware previously procured for rigging or support of this crane might be used once again.

Availability. Fleets of tower cranes are usually geographically dispersed, whereas individual machines are mostly committed to long-term assignments. Thus availability of a preferred crane is hit or miss. A machine coming from another site is a hostage to that site's progress. With a prospect of delay, a contingency plan should be made to bring in another tower crane or to make a short-term mobile crane substitution. Time is needed between installations for the crane to be prepared and for scheduled maintenance. That lag time is variable depending on the condition of the tower crane; a decision to defer needed maintenance for the expedience of meeting a schedule should be avoided.

The question of availability applies as well to accessories such as mast sections, ties, base frames, and climbing gear. During busy times, these items need to be reserved along with the crane.

Power source. Nearly all tower-crane models are electric. Only a few are driven by diesel prime movers; thus there is seldom a choice about an alternate power source. A small electric tower crane might require 50 A at 480 V (three-phase) whereas the very largest machines could need 10 times that amperage. If sufficient power is not available at the site or if the available power is unreliable, a generator is needed.

Reach and capacity. There can be a hidden dimension to reach and capacity requirements. The crane expert and the general superintendent together do not necessarily possess all of the information needed for a satisfactory layout; subcontractors and vendors might need to be queried. For example, oversized gusset plates make some fabricated steel sections heavier than contract drawings shove. The fabricator and steel erection subcontractor should be consulted. In reinforced concrete construction, crane requirements are often governed by the contractor's choice of forms; flying tables, for instance, must be carried free and clear from the perimeter of the building by the crane.

The reach and capacity of the crane needs to include a consideration of pick zones and areas for lay-down, shakeout, or makeup. The end game needs to be looked at, too. The top of a building sometimes carries appurtenances such as cooling towers, chillers, generators, signs, water tanks, antennas, and architectural gingerbread that must be set. Sometimes these picks exceed the demands for hoisting building materials.

Reach and capacity requirements might also include one tower crane dismantling another.

Cost. Selection should consider overall costs and benefits, not just purchase or rental price. Ancillary factors impacting costs could include

- *Trucking*. Consider shipping costs, including consideration of oversized loads.
- *Erection and dismantling*. Height, component weights and complexity affect the type of erection crane selection and the installation labor demand.

- *Climbing*. For cranes that climb, the time and manpower requirements vary from model to model.
- *Rope replacement* is expensive, especially when done on site. Choosing a crane with a rope that is well matched to the application will reduce the opportunity for the rope to wear out prematurely.
- *Power consumption* cost might be the electric bill or, alternatively, diesel fuel and perhaps a generator.
- *Downtime* is predicated not just on the reliability of the equipment but also on the availability of parts and skilled technical support on short notice.
- *Speed and cycle time* are multiplied by thousands of repetitions over the course of a project. A crane executes multiple motions for each work cycle—lift, swing, and trolley (or luff), then the same in reverse. Efficiency can be at least as dependent on operator skill as the characteristics of the crane.
- *Foundation and anchorage* costs are determined by the mass of the crane footing, the price of the anchorage hardware, and the difficulty of placing a template, usually a mast section, for setting the anchorage hardware.
- *Support hardware* such as collars, ties, and jumping beams are fabricated or rented.
- *Loads imposed on the host building* often require supplemental structural elements in the form of shoring, bracing, and reinforcement. Sometimes structural framing is up-sized. Temporary elements such as shoring are removed with the crane after completion.
- *Leave-outs.* Crane openings in decks and wall openings for ties hold up finishing and require comebacks.

Efficiency. Expecting fast work cycle speed from a crane and simultaneously demanding precise control would seem to be mutually exclusive features. Modern drives do permit the same crane machinery to offer both, but not necessarily simultaneously. Advances in variable speed drives come close to allowing the user to have it both ways.

Work cycle speed is a combination of the speeds of all crane motions: hoisting, swinging, and either trolley travel or luffing. Most tower cranes can execute simultaneous motions within the limitations of overall demand on the prime power source. The speeds of most motions are stepped or variable; for instance, a full load would be raised in low speed mode and the empty hook would be lowered to the ground in high speed.

For bulk material handling such as bucketing concrete, cycle speed is a paramount consideration. Swing and boom motions are slower than load hoisting; an efficient crane location is one that keeps these slower motions to a minimum.

Efficiency of steel and precast concrete erection is not much affected by cycle time except in high-rise work. More important for efficiency is an ability to lift optimally sized loads. For example, a project with large trusses is probably better served by a crane big enough to erect these whole than by a smaller one that can only assemble individual truss pieces on falsework. A larger crane might also be capable of lifting heavy bundles of material for rapid unloading of trucks; the bundles might then be shaken out in a location close to where the material is to be used.

Working environment. The commonest of tower cranes has a *hammerhead jib* with a swing circle that circumscribes a large area that might include both the construction site and a piece of the neighborhood around it. But some sites have physical restrictions such as tall adjoining structures or jurisdictional boundaries that would prevent the unrestrained sweep of a jib. *Luffing jibs* are favored for these locations.

These cranes are found often in cities dense with tall buildings and cramped industrial sites. They are also useful in jurisdictions where the law imposes air rights that restrict a jib from passing over adjacent properties and on sites where multiple cranes operate in close proximity (Figure 6.2). Luffing systems are more complicated and thus more expensive to manufacture, maintain, and erect than hammerhead models. For these reasons, the latter are usually preferred where *luffing tower cranes* offer no advantage.

Hammerhead crane limitations, however, can sometimes be overcome by resourceful planning. Multiple cranes can be arranged with overlapping swing arcs (Figure 6.1). Obstructions might be avoided by careful positioning or use of a short jib or a high mast. In some instances, a manufacturer might permit a restricted swing arc with the slewing mechanism locked when not in use.

Lacking a tower top, a *flattop* tower crane has a lesser overall height than an equivalent hammerhead or luffing jib model (Figure 2.36). The reduced height is an advantage where there are aviation restrictions. A lower overall height also reduces the boom tip height required of the erection crane.

Ease of erection and dismantling. Some tower cranes are manufactured with ease of erection paramount. These models have many pin connections instead of bolts. Components come apart in neat packages and erection weights are manageable for medium-size mobile cranes to handle. These cranes typically have thoughtfully placed rigging aids such as lifting lugs, access

Figure 6.2 Luffing boom tower cranes erecting a steel-framed building. The luffing motion helps avoid boom and hook interferences. Coordination is much easier than in Figure 6.1, but communication among operators is still important. (*Morrow Equipment Company.*)

platforms, and alignment holes for spud wrenches at clevis connections. At the opposite extreme, a few tower cranes are complex and awkward to erect.

Jobsite Planning

A jobsite can be a complex environment that changes every working day. Planning for a tower crane must account for the conditions that exist when the foundation is prepared until the day the crane is removed. Rigorous project planning requires visualization in three dimensions plus the dimension of time. This is a difficult skill to master when working from conventional two-dimensional drawings. Renderings and models are helpful; Building Information Modeling (BIM) is now being used, too. As a key piece of equipment that requires substantial preparation and a long procurement lead time, preparation for the tower crane should start early in the construction planning.

Visualization helps identify potential conflicts, coordinate activities, and optimize the flow of materials; but broad-brushed visualization is only the beginning of a more detailed planning regimen. With respect to tower cranes, here are some key aspects that require forethought.

Configuration of a crane is determined largely by the characteristics of the site and the building with consideration of other factors listed here, limited also by the availability of equipment. An ideal tower crane would be simple and freestanding on its own foundation, independent of the structure under construction. The jib would be a wide-sweeping hammerhead type. Unfortunately, tall buildings and congested sites do not comport with these ideals. For these reasons, tower cranes come in varied forms and can be outfitted with accessories such as climbing systems and ties.

Location is governed by myriad considerations, some about optimizing the utilization of the equipment and others about avoidance of hazards and interferences. The crane should be positioned where there will be adequate space to lay out components for erection and to position a mobile crane to do the erection work. When in position, the crane must provide hook coverage and adequate load capacity over its assigned work zone. At the completion of the work there must be access for a mobile crane or other means to disassemble the crane. An optimum location would also take the efficiency of work cycles into consideration as well as sight lines for the operator.

Conflict between the crane and obstacles or hazards needs to be identified and remedied. A conflict might be in the form of an obstruction to the arc circumscribed by the jib. An obstruction could be an adjoining permanent structure, a temporary structure such as formwork or scaffolding, another piece of equipment, or elements of the building under construction. The problem might occur in out-of-service mode—the jib being set free to act like a weathervane—or when a crane is operational and under constant control.

An out-of-service conflict with another tower crane may arise if *weathervaning* circles clash. A mobile crane raising its boom inside the swing arc of a tower crane is another possible cause of a collision, in which case the right-of-way of the latter must be respected by the former. A crane at work might face a different set of obstacles; for example, an extended concrete placing boom could obstruct the jib arc or the load under the hook could foul the jib of another crane.

Though no physical contact would be involved, conflict could be in the form of a boundary violation. A load on the hook might infringe on a zone where lifting is prohibited, such as above an occupied building, a railroad, or a power line. The weathervaning boom might arc over a forbidden space. An aftermarket *zone protection system* can be wired into the control circuitry to prevent the crane hook from lifting over a prohibited area, but it cannot restrict the crane from weathervaning.

Potential conflicts between adjoining tower cranes cannot be fully mitigated by planning. Often overlaps of jib arcs are necessary in order to have full hook coverage of the site. An oft-applied rule is that clearance between cranes, for example between a slewing jib and a tower top, should not be less than 2m (6.7 ft.). Multiple overlapping cranes are choreographed flawlessly on jobsites around the world every day. But for those who want to add a layer of safety, aftermarket *anticollision systems* are available. A device is installed on each crane that tracks the jib orientation, hook radius, and hook height; the devices continuously radio this information to the other cranes, and each operator is able to see the disposition of all the cranes so equipped. If the hook of one crane threatens to foul another, the threatening motion is arrested. When there is a change in the working environment, such as a crane climbing, the change must be programmed into each affected system.

Conflict can also occur at the tower base. The tower and its base mounting could interfere with a roadway or a critical construction access way. Excavation for the footing might impede the placement of buried utilities or clash with existing utilities or foundations.

Reach and capacity would be prime considerations in selecting a crane. But as planning progresses the proposed location and crane configuration should be reviewed against updated information from subcontractors and suppliers. If loads cannot be placed as intended, either the loads or the crane placement must be altered. The crane must also be capable of handling loads in designated pick zones and staging areas.

On a simple steel-framed building, the heaviest loads may be bundles of steel that can be tailored to the capacity of the crane. But on a high-rise building, the core column sections at the bottom of the structure might be the heaviest loads and the crane size needs to be sufficient to set these. Complex structures may have individual massive pieces such as plate girders and transfer trusses that exceed the crane capacity. If such pieces are near the base of the building, a mobile crane might be brought in to assist; otherwise the piece might be divided and spliced in the field or a larger tower crane contemplated.

Reinforced concrete and precast systems tend to present loads that are more uniform and predictable than steel structures. As most poured-in-place concrete nowadays is pumped, the heaviest loads are usually the gang forms or form tables, and sometimes the placing boom is the heaviest pick. Where the crane is used to bucket the concrete, efficient movement is a key to sustaining acceptable pour rates. A floor might require 250 yd^3 (190 m^3) of concrete in one pour. Completion of a pour of this magnitude in one shift is possible only if the rate is in the order of 50 yd^3/h (38 m^3/h) or more. Such a high rate of placement is difficult to sustain unless the bucket carries at least 3 yd^3 (2.3 m^3). The weight of the loaded bucket will be at least 13,000 lbs. (5900 kg).

Penetrations and Interferences with construction are unavoidable in some circumstances. When a tower crane is placed within the perimeter of a new building, it might require deck penetrations. If it is supported by the building, temporary support elements—shores and braces—may be needed as well. Deck openings might prevent the floors from being completed; both the penetrations and shores interfere with the various systems and finishes. A crane that is mounted externally and tied at intervals to the building will have perimeter penetrations that may hold up completion of the wall system and some interior work.

Engineering for a tower crane installation is required at two levels. The crane configuration and the loads it imposes on supporting structures are the impetus for the first; designing the supporting structures is the impetus the second. One could summarize these engineering tasks as determining loads and resistances. The common practice is for loads and configuration to be provided by the crane vendor or manufacturer and resistances to be designed by the contractor's engineer. Both should be accomplished during the planning phase.

Reinforcement, shoring, and bracing are often needed to support tower-crane loads imposed on a structure. Reinforced or up-sized elements are permanent, but shores and some braces are removed with the crane. On steel structures, changes made before shop drawings and fabrication are well underway can be much less expensive than interposing these changes later. In a cast-in-place concrete building, the changes might simply be carried out by adding reinforcement bars to structural elements such as slabs or beams.

Sequencing belongs in the plan for a crane that climbs. There is a mutual dependency between the host structure and the crane. The building cannot progress until the crane is high enough to clear above it, and the crane cannot progress until the building is capable of supporting the extension of the crane. A cast-in-place concrete structure will require forms to be stripped and concrete to reach threshold strength before crane loads can be supported; posttensioning may be required beforehand in some instances. A steel building will have a minimum requirement for completion of bolting, welding, and perhaps decking; sometimes the structural engineer will require concrete floors to be poured and up to a minimum strength, too.

Scheduling is part and parcel of any construction project. All cranes are integral to the project schedule, but none more than the tower cranes that are fixtures of the site. As a tower crane is a finite resource usually in high demand, hook time may be rationed to the various trades. An overbooked schedule is an indication to a planner that an additional crane is needed on the project.

Erection, dismantling, and climbing of a tower crane are events that are disruptive to regular construction activities. They are often scheduled during off-hours to minimize disruption. If not, those activities must be weaved into the regular work flow.

Cranes must be kept sufficiently elevated above the working deck to avoid crimping the operation and to prevent collision hazards. Hammerhead jibs should be maintained at least 10 m (33 ft) above the work to allow clearance for rigging—more if there is a concrete placing boom. Luffing jibs need only about 3 m (10 ft).

Coordination is needed where multiple cranes overlap, especially cranes that climb, or where a tower crane must be synchronized with another piece of equipment such as a concrete placing boom. Harmonizing multiple pieces of equipment is a daily operational necessity, but it should also be part of the advance planning. Coordination is particularly important for hammerhead jibs, as the jib must always have sufficient elevation to clear other equipment and for suspended loads to clear the top of the construction. Planners should parse the tower crane *climbing schedule* to ascertain that the climbs can be steadily kept apace of construction.

Foundation support must fit into the site and be constructed, often long before the crane is brought in. Engineers, inspectors and, surveyors need to be engaged.

Power supply is needed during the erection of an electric tower crane along with a shutoff box, transformer power cable, and the means to suspend the power cable in the mast.

Erection requires hiring a skilled crew with experienced supervision and suitable rigging gear, an appropriately selected and configured assist crane, access, and space to work. The foundation must be ready, and the project accessible for the erection crane and a procession of trucks. Space is needed to dress the erection crane and position it for the task at hand. Arriving trucks must be staged, and the tower crane jib assembled in a position where it can be lifted by the erection crane.

Public agencies in some jurisdictions require permits or plan examination for tower-crane installations. Proximity to a railroad, an airport, or a heliport could also trigger an application and review process. For example, the Federal Aviation Administration (FAA) requires advance notice of potential obstructions to airspace, and they can restrict height or mandate warning lights. Ample time should be allowed for these reviews—they can run sometimes for months—and contingencies considered in case a plan is disapproved.

Storm preparedness is a great concern in some environments, less in others. Manufacturers usually use wind loads obtained from the German DIN standard or the European FEM.[1] What is good and adequate for continental Europe is not necessarily so for all climates. Some regions are prone to periodic extreme winds—coastal areas exposed to tropical storms make the most notable example—and practices should be adjusted for cranes installed in those regions. An adjustment might entail the usage of a heavier mast or reduction of the *freestanding height*. Alternatively, it might require a contingency plan to guy or remove a tower crane in advance of an approaching storm. In some instances where design-level winds derive from hurricane conditions, tower cranes may have to be secured, climbed down, or partially dismantled on the basis of storm warnings. As a measure utilized to avoid the cost penalty of hurricane loading, this must be done only with great forethought and preparedness. When it comes time to implement the preplanned hurricane measures, resources must be available to do so without reluctance and without undue haste.

Dismantling has the same issues as erection and sometimes an additional complication; the building that was erected by the crane may be an obstruction that prevents the same procedure from being worked in reverse (Figure 6.3). Interference with other temporary structures can come about from a lack of coordination or communication. The conflict can be with scaffolding, hoists, concrete formwork, or other cranes.

---

[1]Fédération Européene de la Manutention, "Rules for the Design of Hoisting Appliances," 3d ed., Paris, 1998, Table T.9.15.a.

As the tower crane is normally higher than hoists and scaffolding, the problem does not usually occur until a climbing crane starts to jack down prior to removal.

Removal of a top-climbing crane is usually preceded by jacking down the mast to a lower elevation (Figure 6.3). The tower crane lowers its own sections and loads them out to trucks, an operation that must be considered during planning and could affect the positioning of the crane.

Figure 6.3 A top-climbing tower crane partly jacked down after completion of its work. It cannot swing around, but is able to swing enough for the boom to clear the hoist mast to the left. (*Lawrence K. Shapiro.*)

6.3 Fixed-Base Tower Cranes

A *static base* is the most common tower-crane mounting configuration, supporting what is called a *fixed-base tower crane*. For most units that climb, it is the initial freestanding condition. The bottom of the mast must be firmly anchored into the base, and the base designed to resist weight, *overturning moment*, and slewing torsion from the crane. An insufficiently stiff base will allow excessive mast movement; deflection that adds to the lean of the mast will amplify the overturning moment. An introductory description of the static base was provided in Chap. 2.

The most common static mount is an aboveground steel base frame or *undercarriage* that is an accessory component of the crane mast. Steel or concrete ballast blocks sit on the undercarriage (Figure 6.4) to

Figure 6.4 Tower crane freestanding on a chassis base with concrete ballast. The base must be supported on a competent bearing surface. (*Manitowoc Company Inc.*)

FIGURE 6.5 Tower-crane spread footing under construction with expendable legs visible under the tower section and rebar mat. The tower section is used as a template to assure proper alignment of the expendable legs and to allow the mast to be plumbed before the footing is poured. The footing mass provides ballast for resisting overturning moment. (*Manitowoc Corporation Inc.*)

contribute towards the resisting moment. The frame must be supported on a competent bearing surface that can transmit the imposed loads to the earth without excess settlement. With bogies and wheels running on a track, it becomes a *traveling base* (Figure 2.42).

Another type of static base, used more on larger cranes, is a cast-in-place concrete foundation (Figure 6.5). Foundations of this type are frequently used in North America where large tower cranes dominate the market. It could be above ground or below, an independent spread footing built strictly for the tower crane or integral with the building foundation. In either case, an engineer should be employed to design the foundation and to ascertain that the supporting earth is adequate to carry the loads. The foundation transmits the highly eccentric vertical loads to the ground along with shear forces from wind and *slewing torsion*. These shear forces are generally small and it is only under exceptional conditions that the design of the footing block will be affected by them, but shear may

FIGURE 6.6 A Freestanding saddle-jib tower crane supported on a cantilevered steel base frame. The base frame must be designed to resist *in-service* and *out-of-service* loads and must be stiff enough to keep deflection to a minimum. The high mast and marine location indicate that out-of-service storm wind is likely the controlling design load. (*Manitowoc Corporation Inc.*)

have to be taken into account when the connection between the mast and the block is designed.

An uncommonly used type of static base is a steel frame tied into the building or another structure (Figure 6.6). Engineering this type of installation requires a sound understanding of the mounting requirements and load behavior of the tower crane. Fabrication and installation may be expensive. The supporting structure could be concrete or steel building framing or a foundation. Variations are limited only by economics, physics, and the designer's imagination. With its attendant high-level difficulty and cost, this solution should be used only where site conditions favor it.

Designing for Resistance to Overturning

An out-of-service crane in calm air exerts only dead load on its supports. That includes the vertical load itself and a backward moment induced by the weight of the hoist machinery and counterweights.

When a storm arises, the jib acts as a weathervane with the back moment acting in opposition to the wind. A weathervaning crane swings freely and therefore does not impose torsion on its mast. Weathervaning also reduces the profile of the jib exposed to the wind, reducing in turn the wind shear acting at the top of the mast and the base. In spite of these beneficial effects, in an extreme wind-base moment and shear for an out-of-service crane usually govern the design of the base support.

Though uncommon, some manufacturers allow the swing of an out-of-service jib to be locked. Both side and front wind effects must then be checked. Storm-wind forces acting on the large side area create a moment normal to the deadweight moment, but frontal wind acting on a much smaller area induces a moment that adds directly to the deadweight moment. Both cases must be investigated to verify that the tower can withstand the combined axial and bending effects and to determine reactions on the support structure. A locked swing allows the wind to induce torsion on the mast.

An in-service crane has an *overturning moment* that alternates from rearward to frontward as the load is lifted, maneuvered, and relieved. Wind may come from any direction, and although wind from the rear will directly add to load moment, side wind may be the governing case. In-service wind moment and the load moment are then normal to each other, and the resultant acts on the mast and base in any random orientation. Slewing acceleration, load dynamics, and side wind induce torsion on the mast. *In-service wind* shear is low due to the operation being limited to wind no more than 45 mih (20 m/s) and often less.

The importance of storm winds grows with the mast, and it is storm-wind loading that determines an upper bound to the freestanding tower height. Limitation of *freestanding height*, therefore, may be site-specific, dependent on the design wind for the locality. With a short mast, the design reactions will be governed by the in-service combination of loads; but as *freestanding height* increases, a point will be reached at which the out-of-service design loads become the more important.

Crane manufacturers provide support loads to be used for installation design, although practice varies among them, some giving only maxima and others giving all data for each permissible configuration. Installation designers should insist on being given the data for the configuration they will be working with. A complete set of data is needed for in-service and out-of-service conditions so that the designer can evaluate the particular requirements that will govern in the context of actual site conditions.

Designed to European codes and standards such as FEM, freestanding tower cranes built prior to 2010 are generally evaluated for storm winds of 94 mi/h (42 m/s) at jib elevation, but design practices with respect to wind may vary with the manufacturer or country of origin.[2] Local climate considerations or design standards may dictate that a greater design wind be used.

Consensus standards and codes from different jurisdictions vary somewhat in the requirements for overturning resistance provided by a crane foundation. The widely used FEM standard takes a comprehensive approach . It considers deadweight, live load, wind and inertia as the relevant loads and specifies that five-load combinations be considered: basic stability, dynamic stability, backward stability, extreme wind, and stability during erection or dismantling. This method ignores possible limit states other than tip over, assuming that the supporting ground will remain competent under the influence of the loads that would occur at the onset of overturning. A true limit state approach would consider various failure modes of the supporting surface.

The stability requirement of the American consensus standard is simple. It requires that the resisting moment be at least 150 percent of the maximum overturning moment.[3] Resistance can be provided by ballast or structural anchorage. Like FEM, this standard assumes that the supporting ground remains competent at failure. For some sensitive soils, therefore, the designer might be prudent to consider the safety factor against ground failure rather than tipping.

Load combinations from codes and standards that are for building structures are generally not suitable for tower-crane footings. This notion is discussed more fully in Chap. 3.

A Spread Footing

Where a footing is required for a tower crane, the most common type is a simple spread footing. If soil is inadequate, a pile cap may be an alternative. These are simple solutions suited to sites where space and job conditions permit.

The ratio of footing resisting moment to applied moment is the crucial consideration for crane footing design. With the crane applying a net moment M_o and a vertical load Q to the footing mass, which itself has a weight W, if the footing is a square with sides of length b, then

$$1.5M_o = \frac{(Q+W)b}{2} \qquad b = \frac{3M_o}{Q+W} \tag{6.1}$$

[2]Starting in 2010, major European tower crane manufacturers began following a new CEN standard EN14439 that requires wind to be determined by geographic location.

[3]American National Standard for Construction Tower Cranes, ASME B30.3-2009, American Society of Mechanical Engineers, New York, Section 1.1.1(a).

where 1.5 is the stability factor from the U.S. consensus standard. (Should the crane be supported on a pile cap, the parameter b is taken as the center-to-center distance between the edge piles.) Built into this equation is an assumption that the crane will fail by tipping over the edge before the soil gives way.

The design process is begun by assuming a starting value for W at 1 to 2 times Q. Equation (6.1) can be used to select first trial values for the footing parameters or for rough planning or cost estimates. For final design more accurate figures must be used.

The unit weight of reinforced concrete w is about 150 lb/ft^3 (2400 kg/m^3). In an accurate check of stability, the overturning effect of the horizontal shear force V should not be neglected (Figure 6.7). For a square spread footing, stability will be maintained if

$$1.5(M_o + Vd) \leq \frac{(Q+W)b}{2} \leq \frac{Qb}{2} + \frac{wb^3 d}{2} \tag{6.2}$$

$$d \geq \frac{3M_o - Qb}{wb^3 - 3V}$$

Equation (6.2) can be solved for the footing depth d needed for any trial side dimension b. A rule-of-thumb minimum for the depth is $b/6$. There is also a minimum depth requirement related to the anchorage of the crane to the footing mass, and the local building code may stipulate placement of the footing bottom below the frost line. An expansive footing is generally most advantageous because of

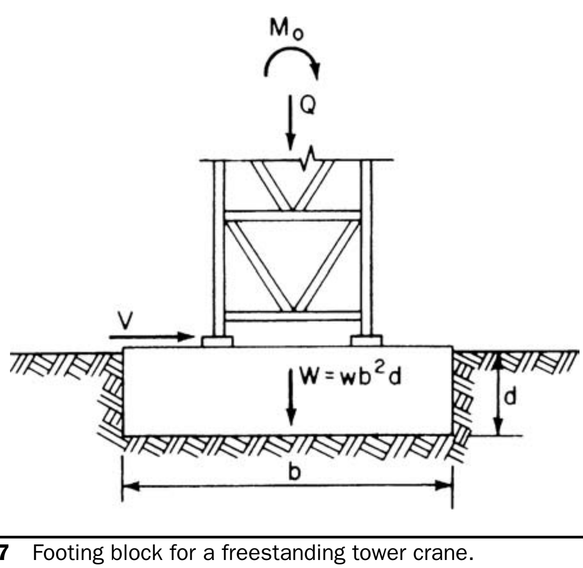


FIGURE 6.7 Footing block for a freestanding tower crane.

its long distance to the tipping fulcrum and attendant low bearing pressure. But the designer must consider all factors including site constraints and then produce the most economical and practical design. Anchorage will be covered later in this section, but first soil pressure must be investigated.

Induced Soil Pressures

Pressure acting under a crane footing acts somewhat differently than a building footing.

- The crane is a temporary structure; it might act on the soil for a couple of months or a couple of years.
- Pressure distributions are highly eccentric due to the large overturning moment imposed by the crane.
- Peak pressures tend to be randomly distributed with respect to direction due to the weathervaning and operational characteristics of the crane.
- There is a dynamic component to the loading, albeit small in comparison to the static loads.
- Loading patterns endure for a short duration, whether induced by wind or operations.
- Uniform settlement of the spread footing is of little consequence; its small magnitude cannot have a significant effect on crane height or lateral ties.

As the crane is insensitive to uniform settlement, values for allowable bearing stress could be permitted to be greater than those used for building design. Because of tower crane footing size and its usual placement with the footing bottom below ground surface, the crane values suggested in Table 5.1 may be conservative. A geotechnical engineer can offer better guidance for the specific soil at the site.

The maximum pressure exerted on the soil by the footing is the combined effect of the vertical loads and the moments. The component of pressure contributed by the vertical loads v for a square footing is given by

$$v = \frac{W + Q}{b^2} = \frac{wb^2 d + Q}{b^2} = wd + \frac{Q}{b^2} \tag{6.3}$$

Using a beam analogy $(\sigma = My/I)$, the moment is found to contribute pressure f, given by

$$f = \pm \frac{(M_o + Vd)b/2}{b^4/12} = \pm \frac{6(M_o + Vd)}{b^3} \tag{6.4}$$

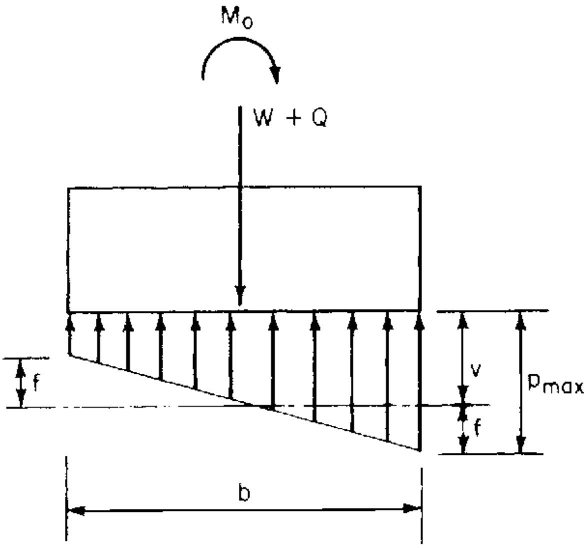


FIGURE 6.8 Trapezoidal pressure pattern under the footing block of a freestanding crane.

For a rectangular footing, the side dimension b is replaced by b_a and b_b. The maximum over-the-side pressure occurs with the moment acting in the short direction. With b_a representing the shorter dimension, the maximum pressure can be obtained by substituting $b_a^2 b_2$ for b^3 in Eq. (6-4).

If $v > |f|$, the resultant pressure under the footing (Figure 6.8) will follow a trapezoidal pattern and the maximum pressure p_{max} is then

$$p_{max} = v + |f| \tag{6.5}$$

But if $v \leq |f|$, the pattern will be triangular (Figure 6.9). Taking the length of the triangle as t and writing the expression for applied and resisting vertical forces, we get

$$W + Q = \frac{p_{max} bt}{2} \tag{a}$$

Then the expressions for applied and resisting moments in equilibrium are

$$(M_o + Vd) = (W + Q)\left[\frac{b}{2} - \frac{t}{3}\right]$$

$$t = 1.5b - \frac{3(M_o + Vd)}{W + Q} = 1.5b - \frac{3(M_o + Vd)}{vb^2} \tag{6.6}$$

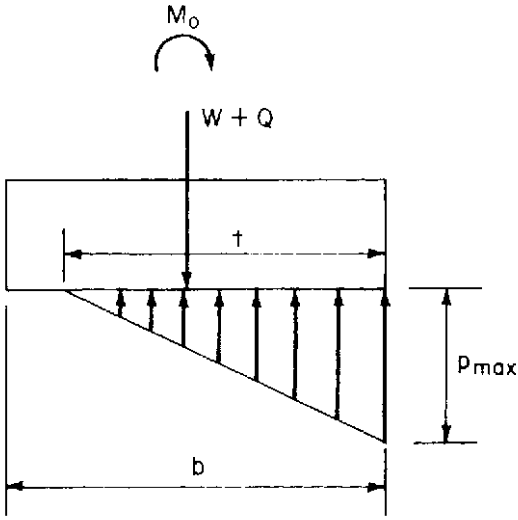


FIGURE 6.9 Triangular pressure pattern under the footing block of a freestanding crane.

The rectangular footing $1.5b$ is $1.5b_a$ and b^2 is $b_a b_b$. Here it is assumed that the analysis is being done with the moment acting in the short direction, but the equation will work with the moment acting either way. Rearranging Eq. (***a***), we get

$$p_{\max} = \frac{2(W+Q)}{bt} = \frac{2vb}{t} \tag{6.7}$$

For the rectangular footing, b is replaced by b_a.

Wind can, of course, blow from any direction, and the magnitude of the wind moment will vary as direction varies with respect to a mast side. There are two conditions of interest, the two extremes of wind blowing normal to a mast face and on the diagonal. The crane documentation should give separate values for the two cases in order to facilitate efficient footing design. Ordinarily, however, only one value is given with the crane data, and it is most likely to be the value for wind on the diagonal.

The maximum soil pressure for the moment acting on the diagonal of a square footing is at the footing corner.

$$p_{\max,\text{diag}} = \frac{(v/2)\left[b/\left(A\sqrt{2}\right)-1\right]}{b/\left(3A\sqrt{2}\right)+(2A/b)\left(1/\sqrt{2}-A/3b\right)-1} \tag{6.8}$$

where

$$A = \frac{M_o + Vd}{W+Q} = \frac{M_o + Vd}{vb^2}$$

The term maximum pressure is used here in a manner consistent with the assumption commonly made by civil engineers in simplifying footing design. Their assumption is that pressure is uniform across the width of the footing, when in fact it drops off near the edges and so must increase toward the central region. The calculated maximum pressures may in fact be exceeded locally, but the values for permitted soil pressures take this simplification into account.

Exact solutions for bearing pressure under odd-shaped footings would be very difficult using traditional methods. Some structural analysis computer programs, however, are capable of modeling these footings and generating solutions for various loadings.

With Figure 6.10 calculus can be used to derive an expression for maximum soil pressure with wind on the diagonal. The differential area across the footing can be expressed as

$$dA = 2x\,dy \quad \text{where } x = \frac{b}{\sqrt{2}} - y$$

If we assume that pressure will diminish to zero at $y = c$, the pressure at any point y is then

$$p(y) = \frac{p_{\max}(y-c)}{b/\sqrt{2}-c}$$

The equilibrium equation for the veritcal forces is

$$\Sigma V = 0 = W + Q - \mathit{f}p(y)dA = W + Q - 2\int p(y)x\,dy$$

$$= W + Q - \frac{2p_{\max}}{b/\sqrt{2}-c}\int_c^{b/\sqrt{2}} (y-c)\left(\frac{b}{\sqrt{2}} - y\right)dy$$

$$\frac{(W+Q)\left(b/\sqrt{2}-c\right)}{2p_{\max}} = -\frac{y^3}{3} + \frac{\left(b/\sqrt{2}+c\right)y^2}{2} - \frac{cby}{\sqrt{2}}\Bigg|_c^{b/\sqrt{2}}$$

$$\frac{b/\sqrt{2}-c}{2p_{\max}} = \frac{1}{W+Q}\left(\frac{b^3}{12\sqrt{2}} - \frac{b^2c}{4} + \frac{bc^2}{2\sqrt{2}} - \frac{c^3}{6}\right) \qquad (b)$$

which is a cubic equation in the unknown c. Likewise, the moment equilibrium equation can be written

$$\Sigma M = 0 = M_o + Vd - \int p(y)y\,dA = M_o + Vd - 2\int p(y)xy\,dy$$

$$= M_o + Vd - \frac{2p_{\max}}{b/\sqrt{2}-c}\int_c^{b/\sqrt{2}} y(y-c)\left(\frac{b}{\sqrt{2}} - y\right)dy$$

$$\frac{b/\sqrt{2}-c}{2p_{\max}} = \frac{1}{M_o + Vd}\left(\frac{b^4}{48} - \frac{b^3c}{12\sqrt{2}} + \frac{bc^3}{6\sqrt{2}} - \frac{c^4}{12}\right) \qquad (c)$$

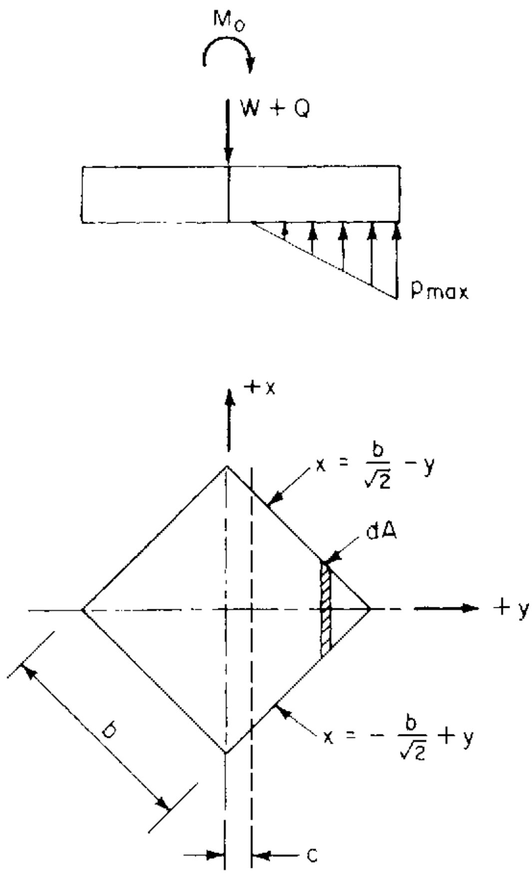


FIGURE 6.10 Tower-crane footing block with moment applied on the diagonal.

Since the left sides of Eqs. (***b***) and (***c***) are identical, the right sides must be equivalent. Setting them equal to each other yields a fourth-order equation in c. Simplified, it reads

$$c^4 - 2\left(A + \frac{b}{\sqrt{2}}\right)c^3 + \frac{6Ab}{\sqrt{2}}c^2$$

$$-\left(3Ab^2 - \frac{b^3}{\sqrt{2}}\right)c + b^3\left(\frac{A}{\sqrt{2}} - \frac{b}{4}\right) = 0 \qquad (\boldsymbol{d})$$

where A is as previously given.

One or more roots will be imaginary and can be ignored. The remaining roots will be identical solutions to the location of the zero-pressure line. Substituting the solution value into either Eq. (*b*) or (*c*) will give the maximum footing corner pressure p_{max}. But through good fortune and the application of pure logic to the physical aspects of the problem, an exact expression for the solution root has been found. When $c = 2A - b/\sqrt{2}$ is substituted into Eq. (***b***) or (***c***), the result is Eq. (6.8).

Example 6.1

1. Design a square spread footing for a tower crane weighing 230 kips (104 t) that will be subjected to a storm-wind net moment of 3600 kip · ft (498 t · m) across the mast faces and 4450 kip · ft (615 t · m) on the diagonal. The wind shear forces are 18.8 and 22.6 kips (8.5 and 10.3 t), respectively. Soil bearing capacity is 4 tons/ft² (383.0 kN/m²).

Solution

Since the component of the diagonal wind that acts across the side is 4450 cos 45° = 3147 < 3600, the side moment will be used to satisfy stability criteria. From Eq. (6.1), try

$$W = 1.1\,(230\text{ kips}) = 253\text{ kips }(115\text{ t}).$$

$$b_1 = \frac{3(3600)}{230+253} = 22.36\text{ ft}\quad\text{try 22 ft (6.71 m)}$$

and from Eq. (6.2) the trial depth is

$$d_1 = \frac{3(3600)-230(22)}{0.15(22^3)-3(18.8)} = 3.73\text{ ft}$$

Check $$\frac{b}{6} = \frac{22}{6} = 3.67\quad\text{try } d = 3.75\text{ ft (1.14 m)}$$

The trial footing is then 22 by 22 by 3.75 ft (6.71 by 6.71 by 1.14 m). Soil pressure must now be checked. The vertical component is given by Eq. (6.3).

$$v = 0.15(3.75)+\frac{230}{22^2} = 1.04\text{ kips/ft}^2\ (49.8\text{ kN/m}^2)$$

The footing side overturning moment $M_o + Vd$ is 3600 + 18.8 × 3.75 = 3670 kip-ft (4976 kN-m). Equation (6.4) then gives the moment contribution to ground pressure.

$$f = \pm\frac{6(3670)}{22^3} = \pm 2.07\text{ kips/ft}^2(99.1\text{ kN/m}^2)$$

Since $|f| > v$, the soil pressure pattern will be a triangle whose length is given by Eq. (6.6).

$$t = 1.5(22)-3\frac{3670}{1.04(22^2)} = 11.13\text{ ft (3.39 m)}$$

The maximum soil pressure from side moment, from Eq. (6.7), is then

$$p_{max,side} = \frac{2(1.04)(22)}{11.13} = 4.11\text{ kips/ft}^2\ (196.8\text{ kN/m}^2)$$

Now the diagonal pressure must be checked. Using diagonal moment and shear, the parameter A takes the value

$$A = \frac{4450 + 22.6(3.75)}{1.04(22^2)} = 9.01 \text{ ft } (2.75 \text{ m})$$

Equation (6.8) gives the maximum diagonal pressure.

$$p_{\max,\text{diag}} = \frac{\dfrac{1.04}{2}\left(\dfrac{22}{9.01\sqrt{2}} - 1\right)}{\dfrac{22}{3(9.01\sqrt{2}} + \dfrac{2(9.01)}{22}\left[\dfrac{1}{\sqrt{2}} - \dfrac{9.01}{3(22)}\right] - 1}$$

$$= 8.81 \text{ kips/ft}^2 (421.8 \text{ kN/m}^2)$$

The diagonal pressure is more than twice the over-the-side pressure. Furthermore, it exceeds the permitted value of 4 tons/ft^2. If we try a larger footing with $b_2 = 22.5$ ft (6.86 m), $d_2 \geq 3.40$ ft (1.04 m) but $b_2/6 = 3.75$ ft (1.14 m). The new trial footing is then 22.5 by 22.5 by 3.75 ft (6.86 by 6.86 by 1.14 m). Only the diagonal pressure need be checked.

$$v = 0.15(3.75) + \frac{230}{22.5^2} = 1.02 \text{ kips/ft}^2 (48.8 \text{ kN/m}^2)$$

$$A = \frac{4450 + 22.6(3.75)}{1.02(22.5^2)} = 8.81 \text{ ft } (2.69 \text{ m})$$

$$p_{\max,\text{diag}} = \frac{\dfrac{1.02}{2}\left(\dfrac{22.5}{8.81\sqrt{2}} - 1\right)}{\dfrac{22.5}{3\left(8.81\sqrt{2}\right)} + \dfrac{2(8.81)}{22.5}\left[\dfrac{1}{\sqrt{2}} - \dfrac{8.81}{3(22.5)}\right] - 1}$$

$$= 7.68 \text{ kips/ft}^2 (367.7 \text{ kN/m}^2) < 8 \text{ kips/ft}^2 \quad \text{OK}$$

2. Using the same data, design a footing with the crane positioned as shown in Figure 6.11.

Solution

The diagonal moment now acts about the side of the footing. For, b try $W = 1.25Q$; $b = 25.80$ ft rounded to 25.75 ft (7.85 m).

$$d \geq \frac{3(4450) - 230(25.75)}{0.15(25.75^3) - 3(22.6)} = 2.98 \text{ ft} \quad \text{try 3 ft (0.91 m)}$$

Although $b/6 = 4.29 > 3$, this orientation will produce smaller bending moments in the footing, and thinner sections may prove to be practical. This trial footing contains about 5% more volume and weight than the footing in part 1.

$$v = 0.15(3) + \frac{230}{25.75^2} = 0.80 \text{ kip/ft}^2 (38.2 \text{ kN/m}^2)$$

$$f = \pm\frac{6[4450 + 22.6(3)]}{25.75^3} = 1.59 \text{ kips/ft}^2 (76.0 \text{ kN/m}^2)$$

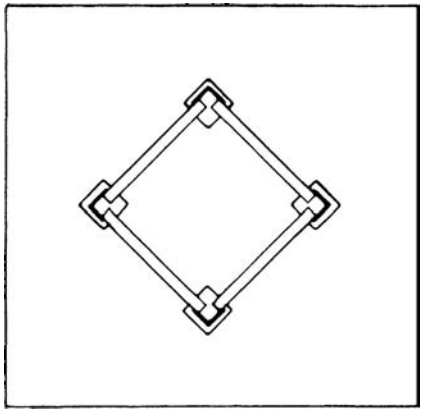


FIGURE 6.11 Tower-crane footing block with the mast rotated 45°.

Since $f > v$, the pressure pattern is triangular with

$$t = 1.5(25.75) - 3\frac{4535}{0.8 \times 25.75^2} = 12.98 \text{ ft } (3.96 \text{ m})$$

$$p_{\text{max,side}} = \frac{2(0.8)(25.75)}{12.98} = 3.17 \text{ kip/ft}^2 (152 \text{ kN/m}^2)$$

Checking the diagonal, we see that the applicable moment was the side moment before.

$$A = \frac{3600 + 18.8(3)}{0.8(25.75^2)} = 6.89 \text{ ft } (2.10 \text{ m})$$

$$p_{\text{max,diag}} = \frac{\frac{0.8}{2}\left(\frac{25.75}{6.89\sqrt{2}} - 1\right)}{\frac{25.75}{3\left(6.89\sqrt{2}\right)} + \frac{2(6.89)}{25.75}\left[\frac{1}{\sqrt{2}} - \frac{6.89}{3(25.75)}\right] - 1}$$

$$= 3.11 \text{ kips/ft}^2 \ (148.9 \text{ kN/m}^2)$$

The side and diagonal ground pressures are almost identical, and both are considerably below those of part 1. Rotating the crane 45° with respect to the footing therefore is a viable arrangement for sites where permissible soil bearing values are low (see Figure 6.11).

Mast Anchorage

The crane mast must be anchored to the footing block so that the vertical load, moment, and shear can be transferred from mast to block. If we assume a square mast measuring s between the centroids of the mast legs on each face, the diagonal distance will be $s\sqrt{2}$. With the moment acting on the diagonal, the force on the legs affected by the moment will be

$$F_{\text{diag}} = -\frac{Q}{4} \pm \frac{M_{\text{o}}}{s\sqrt{2}} \tag{6.9}$$

the negative sign denoting compressive force. For a moment applied parallel to a mast side, the legs would carry

$$F_{\text{par}} = -\frac{Q}{4} \pm \frac{M_{\text{o}}}{2s}$$

The diagonal case can be seen to produce greater leg loads both in tension and compression. For most cranes the portion of leg load contributed by moment is sure to leave net tensile forces in one leg.

The absolute magnitude of the compressive load will be $Q/2$ greater than the tensile load with moment on the diagonal. Therefore, compression will certainly govern the design of the mast legs, but anchor bolts must be designed for a greater load than Eq. (6.9) would suggest.

Good design requires that the ultimate strength of the bolts be no less than that of the mast legs in compression or alternatively that they be capable of transmitting $1.5M_{\text{o}}$ to the footing block without failing. Then in the event of an overload wind, it will be overall crane stability that will control failure. Inasmuch as peak wind effects will be from gusts, the crane will stand a good chance of recovering before overturning completely. If the bolts were the weakest link, their failure might lead to collapse.

The anchor bolts themselves should be heavily greased or sleeved with some material that will prevent bonding to the concrete. Bonding should be prevented because of the fluctuating nature of the leg loading, both in and out of service. The fluctuations would, destroy the bond in any case, but in the process the concrete around the bolt could spall, damaging the compressive support area and loosening the bolts. Unbonded bolts, on the other hand, will stress uniformly over their length and apply loading to the concrete only at the end bearing zones. Consequently, it is necessary to have adequately sized bearing plates anchoring the bottom ends of the bolts.

Common practice is for use of high-strength bolts or threaded rods, and initial pretensioning should be specified by the engineer. Anchor bolts that are preloaded to at least their maximum applied loading will not permit the tower leg baseplate to loosen or to drift.

In steel erection practice, it is not unusual to install leveling nuts on some of the bolts at each anchorage. Then, when the baseplate is set on the nuts, it is at proper elevation and ready for placement of grout. However, this procedure must not be used when the bolts are to be preloaded because the bolts with leveling nuts cannot be preloaded; the entire preload will be confined to that small part of the bolt which lies between the two nuts and will therefore be ineffective.

To assure proper preloading, the base-plates should be leveled on steel shim stock and grouted. When the grout has reached sufficient strength, the bolts can be tightened to the specified preload. Bolt torque should be checked a week or so after the crane has been put into operation, since the dynamic loading could cause further bond breaking or tighter seating of the baseplate on the footing. If the bolts need retightening, they should be checked every week or so until they stay tight.

As already explained, the anchor bolts should terminate at suitably sized plates bearing in the concrete. These should be stiff plates. Though their thickness can be determined by calculation, by rule-of-thumb the thickness should be approximately equal to the diameter of the anchor bolts. Length and width of these embedded plates should usually be at least that of the leg baseplate.

The transfer of tensile load is then through the bolts to the plate mechanically anchored in the concrete mass. The shear, or diagonal-tension, strength of the concrete must lock the plate in place. For a square plate of sides p buried to a depth d_0 in the concrete (Figure 6.12), the imaginary surface that must experience diagonal tension comprises

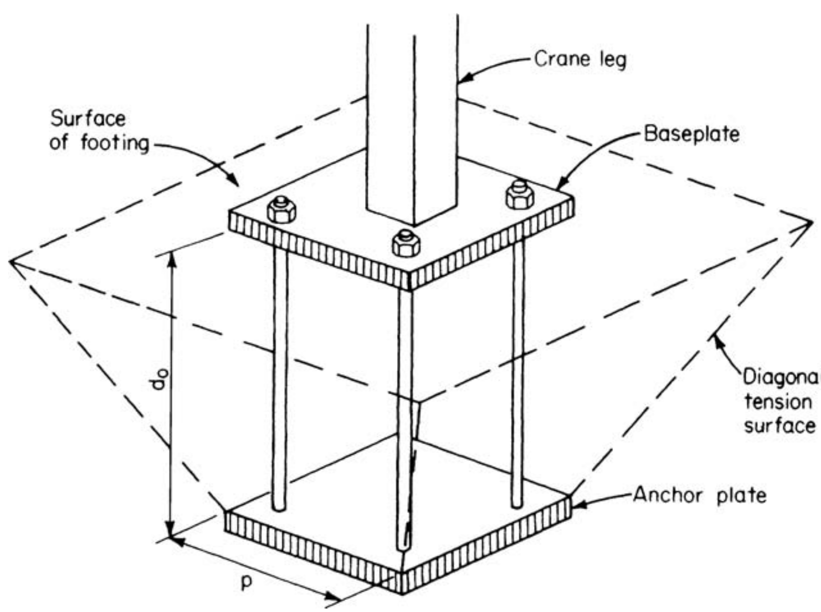


Figure 6.12 The imaginary concrete surface assumed to act to secure the anchor bolt plate against pullout forces.

planes rising from each edge of the buried plate at 45° to the direction of the applied force. In plan view, the projected area of this surface is

$$A = (p + 2d_0)^2 - p^2 = 4d_0(p + d_0) \tag{6.10}$$

and the tensile stress on the area is then (using 1.5 times the applied moment, discussed earlier).

$$\sigma_t = \frac{F^+_{\text{diag}}}{A} = \frac{\frac{1}{2}\left[(-Q/4) + 1.5M_o/\left(s\sqrt{2}\right)\right]}{pd_0 + d_0^2} \tag{6.11}$$

The limiting value (in working stress design) for σ_t is $\sigma_a = 2\sqrt{f'_c}$, where f'_c is the specified 28-day compressive strength for the footing concrete.[4] Sometimes a crane is erected on concrete that is much less than 28 days old, in which case the present strength of the concrete should be used.

Equation (6.11) will give the diagonal tension stress for a square plate with side dimension p at depth d_0 for given loads. If p is taken to match leg baseplate size, then d_0 is the unknown value when full permitted stress is used. Rearranging Eq. (6.11) and solving for d_0 with the quadratic formula gives

$$d_0 = \frac{1}{2}\left[\left(\frac{p^2 + F^+_{\text{diag}}}{\sigma_a}\right)^{1/2} - p\right] \tag{6.12}$$

as the minimum plate depth needed to satisfy both the loads and concrete strength.

A similar effect takes place under the leg in compression, the stressed surfaces spreading downward at 45° from the bearing-plate edges to the bottom of the footing depth d, but d must be greater than d_0, of course. Assuming the square bearing plate with side dimension p, we have

$$\sigma_a \geq \frac{-F^-_{\text{diag}}}{A} \geq \frac{\frac{1}{4}\left[(Q/4) + M_o/\left(s\sqrt{2}\right)\right]}{pd + d^2} \tag{6.13}$$

and to find the minimum acceptable value for d we use

$$d = \frac{1}{2}\left[\left(p^2 - \frac{F^-_{\text{diag}}}{\sigma_a}\right)^{1/2} - p\right] \tag{6.14}$$

When the edge of the bearing (or anchorage) plate is closer to the footing edge than footing depth d (or anchorage plate depth d_0), the

[4]Building Code Requirements for Reinforced Concrete (ACI 318-99), American Concrete Institute, Farmington Hills, Mich., 1999. After this edition, allowable stress design was relinquished by ACI 318 in favor of the limit state (LRFD) method.

full diagonal-tension-resisting surface area will not develop and Eq. (6.13) or (6.11) will no longer be applicable. In this case, Eq. (6.10) must be replaced and the true projected area A determined for the actual footing-edge conditions. The corrected value for A can be used in Eq. (6.13) or (6.11), but neither Eq. (6.12) nor (6.14) can be used.

Another means for anchoring a crane mast to a footing block is by casting a section of mast into the concrete. When this method is used, a special mast bottom section, called an *expendable base,* is employed as the lowermost section of the mast. Once common, expendable bases are now seldom used. At the upper end of the expendable base, standard mast-connection arrangements are provided for erection of ordinary mast sections, but at the bottom end of each leg a stiffened plate is provided. The plate connection to the leg is arranged so that it will be capable of transmitting the maximum tensile and compressive forces developed.

Since the leg baseplates are buried in the concrete, tensile loads are transmitted to the concrete through diagonal-tension-resistance surfaces that rise from the plate edges at 45° to the footing surface in exactly the same way that the anchor-bolt-plate loads were transmitted. Equations (6.10) to (6.12) are therefore applicable for determining the anchorage depth needed, subject to the limitations mentioned for distance to the edge of the footing, of course.

In like manner, the compression loads are resisted on diagonal-tension planes projecting downward from the leg baseplate. Equations (6.13) and (6.14) are therefore applicable, with the limitations mentioned. Now, however, total footing thickness can be no less than $d_0 + d$, and the distance d is the depth of footing remaining below the leg baseplate (Figure 6.13). Thus, when an expendable base is used, overall footing thickness may be controlled by pullout and punching-strength requirements rather than by bending-resistance needs. However, expendable inserts, such as in Figure 6.5, can be designed to minimize footing thickness by providing top plates for punching forces and bottom plates for pullout forces.

More common now than expendable bases, most tower-crane manufacturers furnish individual expendable legs which must be installed with a template to ensure correct alignment (Figure 6.5). These are designed to utilize overlapping tension and compression pullout areas, so on minimize footing thickness.

A *tower section* is commonly the best template for setting expendable legs. Its bulk may pose logistical challenges for getting the piece set in a foundation, but it has several important advantages.

- There is no question of tolerances or fit as the tower section is already mated to the expendable legs.
- A tower section is convenient and reliable for checking plumbness before and after the concrete is poured.

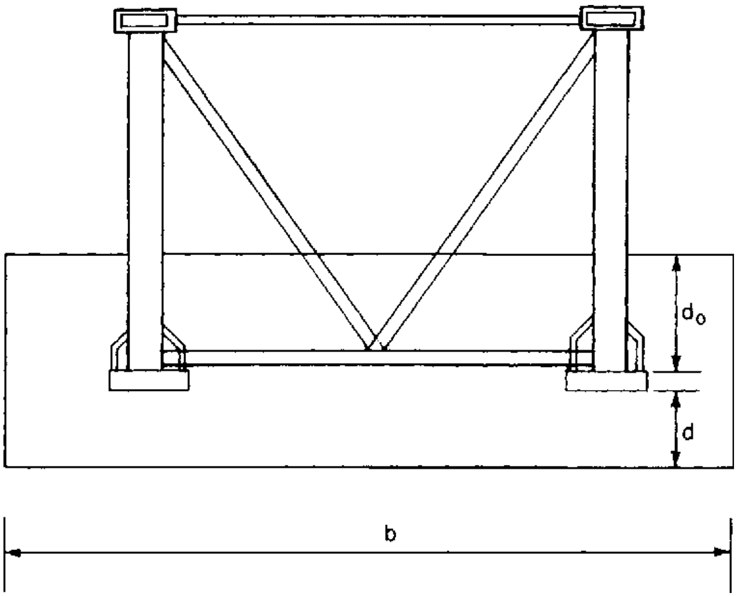


Figure 6.13 Section through a footing block for a crane with an expendable base section.

- Climbing cranes require the mast to be correctly oriented; this is easily checked before the pour when a tower section is in place.
- A template such as that shown in Figure 6.14 can leave a rocking-table effect if it is not perfectly leveled. The factory-perfect stiff mating surface of a tower section will prevent such an occurrence.

Determination of footing depth for anchorage should be the first step in tower-crane footing design. Then with minimum footing depth known, it can be tested with Eq. (6.2) for stability requirements. With footing dimensions set, the footing reinforcement needed to resist the moments and shears can be designed using the ordinary principles of concrete design.

Example 6.2

1. What anchorage depth is needed for a crane with leg centroids 9 ft (2.74 m) apart and with 26-in × 26 in (660 mm × 660 mm) anchor plates? Use the loads of Example 6.1 and concrete with $f_c' = 4000$ lb/in^2 (27.6 MN/m^2).

Solution

Equation (6.9) gives the uplift or tensile leg load that must be accommodated.

$$F_{\text{diag}}^{+} = -\frac{230}{4} + \frac{1.5(4450)}{9\sqrt{2}} = 467.0 \text{ kips } (211.8 \text{ t})$$

FIGURE 6.14 Template for setting tower-crane anchor bolts. Though well suited for anchor bolts, a template like this must be used with great care for anchor stools as there is no satisfactory means of adjustment after the footing is poured. The footing mass will be supplemented by rock anchors. (*Lawrence K. Shapiro.*)

The allowable concrete stress σ_a is

$$\sigma_a = 2\sqrt{4000} = 126.5\ \text{lb/in}^2 (872.1\ \text{kN/m}^2)$$

and the minimum anchorage depth is given by Eq. (6.12).

$$d_0 = \frac{1}{2}\left[\left(26^2 + 467.0\,\frac{1000\ \text{lb/kip}}{126.5}\right)^{1/2} - 26\right]$$

$$= 20.0\ \text{in} = 1.67\ \text{ft}\ (0.51\ \text{m})$$

2. What minimum footing depth will be needed if an expendable base is used?

Solution

The anchorage depth is as calculated in part 1. For resistance to punching through the footing, the compressive load is given by Eq. (6.9)

$$F^-_{\text{diag}} = -\frac{230}{4} - \frac{4450}{9\sqrt{2}} = -407.2\ \text{kips}\ (184.7\ \text{t})$$

and the depth below the bearing plate is given by Eq. (6.14).

$$d = \frac{1}{2}\left[\left(26^2 + \frac{407{,}200}{126.5}\right)^{1/2} - 26\right]$$

$$= 18.2\ \text{in} = 1.52\ \text{ft}\ (0.46\ \text{m})$$

This makes a total depth required of

$$1.67 + 1.52 = 3.19 \text{ ft } (0.97 \text{ m})$$

The footing size chosen in Example 6.1 would satisfy for an anchor bolt base or for an expendable base.

Undercarriage Base with Ballast

A pre-engineered undercarriage has the advantage of needing minimal on-site preparation (Figures 6.4), and often needing no excavation. Some undercarriages are knee braced to the mast. Often two or four footing blocks of relatively small size are used because ballast blocks provide most of the overturning resistance. The principles used for the design of this arrangement are similar to those used with a massive footing block. As given earlier, the resisting moment M_r should be 1.5 times the applied moment M_o. Then

$$M_r = 1.5M_o = (Q + B + W)b/2 \tag{6.15}$$

where B is the weight of the prefabricated ballast blocks, W is the combined weight of the four identical corner footing blocks, and b is the distance between footing block centers each way. For footing blocks measuring $c \times c$ in the plan the an allowable soil pressure of p_{allow}.

$$c = \left(\frac{Q + B + W}{2p_{\text{allow}}}\right)^{1/2} \tag{6.16}$$

The footing blocks of depth d and unit concrete weight w will then weigh

$$W = 4c^2 dw \tag{6.17}$$

For practical problem solving, Eq (6.15) is rearranged to give

$$B + W = 3M_o/b - Q$$

and then Eq. (6.16) is solved for a trial value of c; d is chosen by judgment in consideration of c, permitting W and then B to be calculated. But B represents the weight of available prefabricated blocks and may not match the calculated value. Therefore, blocks are chosen to make two symmetrical piles of ballast blocks weighing not less than the trial calculation value. It is necessary for the CG of the ballast blocks to be on the mast centerline, and this is accomplished with two equal weight stacks on opposite sides of the mast. Now, the footing blocks may need recalculation.

The anchor bolts used to fasten the knee brace legs to the footing blocks need to be of sufficient strength to lift the weight of the block at the allowable tension force for the bolt size. Those bolts can never be exposed to a greater force, but they must also resist horizontal wind shear and slew torsion.

Example 6.3
Using the same data as for Example 6.1 and with b = 24 ft (7.3 m), design the footing blocks and determine the required ballast weight.

Solution
The combined weight of ballast and footing blocks can be calculated from the rearranged form of Eq. (6.15).

$$B+W=3\times 3600/24-230=220\text{ kips (978.6 kN)}$$

Equation (6.16) will then given the footing block plan dimensions

$$c=\left(\frac{230+220}{2x8}\right)^{1/2}=5.3\text{ ft (1615 mm)}$$

but use 5.33 ft or 5′ 4″ (1625 mm).

For a footing of this size, a depth of 3 ft (915 mm) would be appropriate. Equation (6.17) now gives footing weight.

$$W=4\times 5.33^2\times 3\times 0.15=51\text{ kips (227 kN)}$$

which requires ballast weight to be 220 – 51 = 169 kips (752 kN). The ballast selected will therefore weigh somewhat more when available blocks are chosen.

Variations on the Static Mount

The tower-crane foundation solutions described so far are easy to conceptualize and relatively simple to implement, but they are not suited to all conditions. The soil may be inadequate or rock that is expensive to excavate en masse may be present. Sometimes the space available is not sufficient for a spread footing or an uneconomically deep excavation would be required to compensate for a small available plan area. In such cases, the best solution is often to mimic the method used for the building footings or to utilize other foundation methods that are common in local practice. Caissons, piles, mats, rock anchors, and custom fabricated base frames have all been used to support tower cranes (Figure 6.15).

As mobilization of driving or drilling equipment can be expensive, caissons and piles may be economical only when the equipment is already at the site. In crane practice, batter piles are not ordinarily used to resist slew torsion and lateral loads. Instead, the surrounding soil mass is utilized for lateral restraint, but the backfill must then be placed in a controlled manner suitable for this purpose. Uplift capacity of caissons and piles should be exploited, where practicable, to reduce footing mass.

Rock, too, can be exploited for supporting the crane by use of rock anchors. These high-tensile-strength tendons can tie down a base

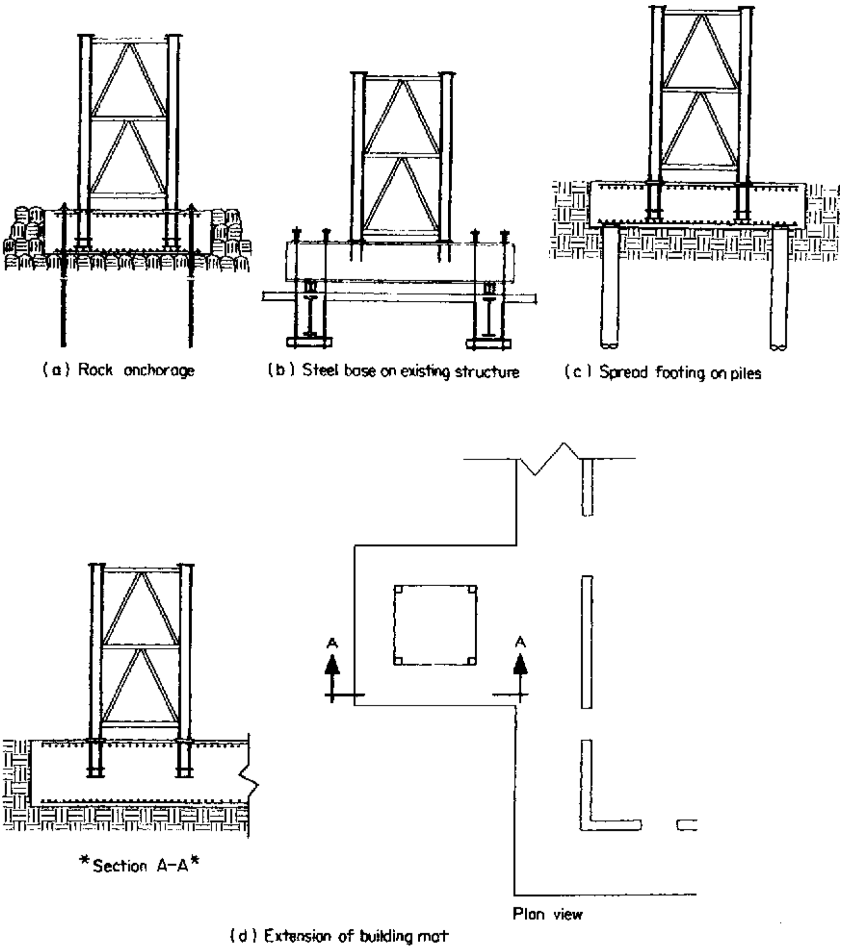


FIGURE 6.15 Several methods of supporting a freestanding tower crane. (a) Rock activation posttensioned anchors may be economical where rock excavation is difficult. (b) A base frame distributes the crane loads to adequate points of support on a building frame. (c) Piles may be suitable where the pile-driving equipment is already on the job. (d) A mat foundation may be well suited to double as a tower-crane base.

frame or individual tower legs (Figure 6.16); they can also replace ballast mass, allowing a footing to be reduced in area (Figure 6.15*a*). Anchors should be installed, tested and preloaded according to Post-Tensioning Institute (PTI) guidelines.[5] An unbonded (free-stressing) length of at least 15 feet (4.6 m) is established, and usually as much length grouted into the rock. Working lock-off and proof-testing loads

[5]Recommendations for Prestressed Rock and Soil Anchors, Post-Tensioning Institute, Farmington Hills, MI, 2004.

Figure 6.16 The foundation of this mast is comprised of an individual footing under each tower leg. The footings, in turn, are anchored to rock with the rock anchors tested and preloaded. Collectively the preload of the rock anchors must exceed the design uplift load of the respective leg. The bearing capacity of the underlying rock must be capable of supporting the combined preload and compression from the crane leg. (*Dominique Singh.*)

are normally set at 60, 70, and 80% of the ultimate tendon strength, respectively. As the application is temporary, the tendons typically do not require corrosion protection. But they should be relieved of tension after the crane is removed; stressed anchor could be live missiles hidden in the ground. Only active anchors need preloading; those which are passive need not be tensioned, nor do they require a free stressing length[6].

A crane base can be improvised by augmenting or interconnecting structural elements of the building. For example, a mat foundation might be extended (Figure 6.15*d*) or several pile caps merged into one mass. Caution must be exercised when a crane footing is to be interconnected with spread footings or friction piles that are part of the building structure. One must assume that the dynamic action of the crane will cause some differential settlement. Control joints might be considered as a remedy to prevent cracks from developing, or the

[6]A passive anchor is one that is unlikely ever to be relied upon to resist load. It might be present to provide a safety margin against overturning or to provide resistance only against out-of-service (design storm wind) loading.

combined footing mass could be made so robust that cracking and excess movement would be precluded.

Sometimes a chosen spot for a freestanding tower crane is impracticable for a conventional foundation. The proposed location might have strong points in near proximity that can support the crane, but intermediary framing is needed; a steel base frame might be fabricated for that purpose (Figure 6.15*b*). The frame should be designed by an engineer who is familiar with the loading patterns, behavior, and connection details of the crane.

Tower-crane leg loads reverse repeatedly from tension to compression. Lateral loads also go through reversals from shifting wind and alternating swing motion. Details of base support structures should be designed with cyclical dynamic loading in mind, and bolt connections should be designed as fully pretensioned slip-critical.

As the plan area of the base increases, corner loads imposed on the building go down but framing size goes up. At some point, a base can be large enough that there are no corner uplift reactions. Base mounting does not necessarily require a mass of ballast.

A base frame designer needs also to consider the shifting orientation of the crane loads and the sensitivity of the crane to deflection. As the crane swings through its full circle, loads acting on individual members and connections also shift. The cumulative effect of individual members deflecting is to make the bottom plane of the mast rotate and cause the mast to lean. Design overturning moment should be adjusted for this second-order effect.

In both design and fabrication, close attention should be paid to the interface between the bottom of the mast and the base frame.

- The designer should be cognizant of the means for bolt installation to assure that there is sufficient installation space and wrench clearance.
- Distances between bolt centers must be held to very tight tolerances.
- Mating surfaces should be square and milled.
- The four points of tower support must be coplanar to avoid a rocking-table effect.
- The mounting plates that receive the feet of the crane should be stiff, at least matching the stiffness of the mast feet.
- Some tower connections have an alignment stud that should be replicated on the base connection.

Both the designer and the detailer need to study the connection. A template—an actual tower section is often used—should be used to lay out the holes and assure proper mating.

Figure 6.17 depicts a crane positioned to erect steel on a building superstructure. In an installation such as this, loads can be shared by

FIGURE 6.17 This crane is supported on a steel base built into existing roof framing. The crane is erecting new steel to add floors. The building requires shoring and reinforcement to carry the crane loads. (*Lawrence K. Shapiro.*)

several tiers of framing or transferred to walls and columns. Shores, tie rods, temporary steel framing, or some combination of these might be used to distribute vertical loads. Horizontal loads from wind and torsion might be distributed through a decking system or alternatively carried by plan braces installed for this purpose (Figure 6.18).

Mast Plumbness

Though the necessity to set a mast reasonably plumb seems intuitively obvious, it is useful to parse the reasons why.

- A leaning crane will have additional moment applied to the mast and base by the second-order effect.
- Out-of-plane loading forces the operator to work against side loading and slew torque.
- Clearances between the crane and other structures will not be according to plan.
- When a crane climbs and is tied to the building, the drifting tower will not align with the building as planned.
- A tower forced into alignment induces locked-in stresses.

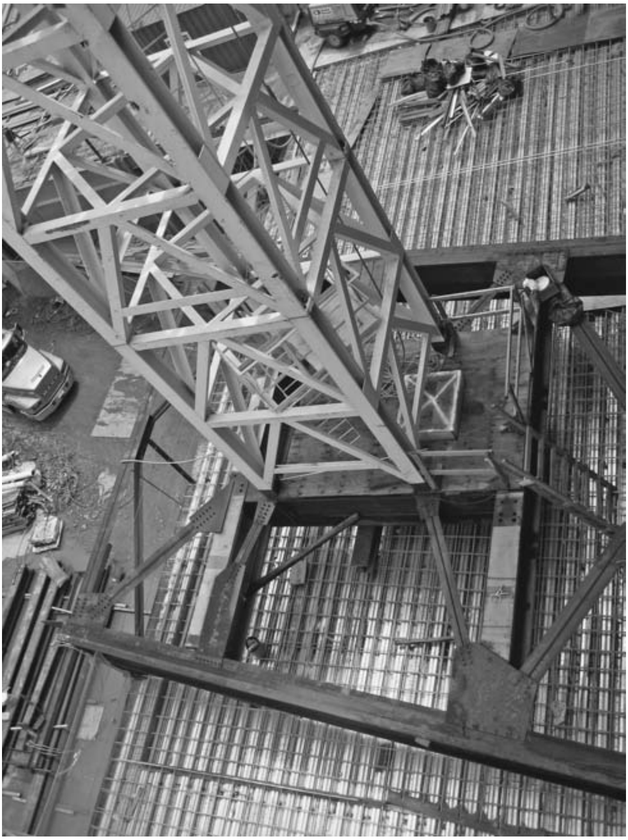


FIGURE 6.18 A steel base frame carrying the crane loads to building columns (not seen) and the foundation wall. The base frame has plan braces to resist slew torsion and lateral loads. (*Dominique Singh*.)

In the course of normal operations, a tall freestanding tower-crane mast can sway 2 ft (0.6 m) or more as a result of elastic deflection. Any degree of out-of-plumbness at the base would directly add to this lean. The operator should not feel any deleterious effect on his ability to control the crane and load unless the total lean (initial out of plumbness plus elastic deflection) is in excess of 1% of the mast height. If there is also a stiff wind, the operator will be working against that at the same time.

When the crane mast is set to within stringent limitations, the effect of tower plumbness on crane operations and base moment is minor and can be ignored. For general purposes, a plumbness tolerance of 1:500 is satisfactory. Some manufacturers specify a tighter tolerance.

However, a crane that is set up to top-climb or that has tight clearances should be set to a much tighter tolerance of perhaps 1:1000.

Ties for a top-climbing crane can either be forced into position or aligned where they fall. Allowing the mast to drift might be acceptable for a midrise building with perhaps one tie, but a tall mast that is out-of-plumb by 1:500 will drift a foot (0.3 m) on a 500-ft. (150 m) building, enough to cause tie alignment problems. The alternative, forcing the mast and ties into alignment, might be preferred, but it is no panacea. A mast that is forced into alignment will bow back and forth with each tie, tower and ties will lock in stresses that have not been considered by the designer. Much aggravation and wasted time can often be avoided by initially taking the time to install the mast to a tight plumbness tolerance.

In some instances, a spread footing might suffer differential settlement after the crane is placed on it. The movement could be caused by the imbalanced moment of the crane or by a lack of uniformity of the ground support. A footing can be monitored using nothing more than a surveyor's level. Determining the intervals between is a question of expectation and experience. Baseline readings should be taken before the upper works of the crane are erected. On relatively soft soils follow-up readings to monitor movement should be taken weekly for 4 to 6 weeks. After that, reading intervals can be expanded to 2 weeks if little movement is noted. Later, monthly intervals can be used if there is a pattern of differential settlement.

The initial and monitoring readings are taken at the four corners of the footing block (Figure 6.19). The points chosen should be marked

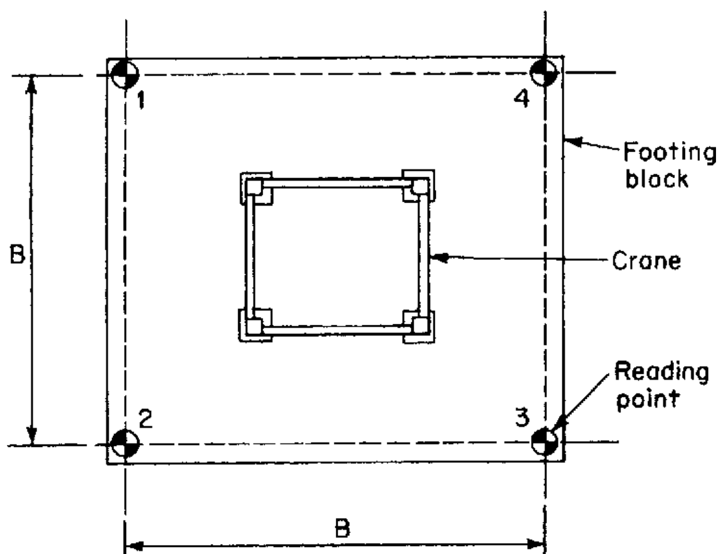


FIGURE 6.19 Locating control points for monitoring footing-block differential settlements.

with a masonry nail or other permanent means and numbered to assure that all readings will be taken at exactly the same points and properly identified.

Insofar as the effect on the crane goes, mast plumbness is what matters; the footing should be monitored to isolate its effect and plot its trend. Differential settlements can sometimes be corrected by parking overnight in such a way that the jib will point in the direction of the lean. The slewing brakes can be locked only if there is assurance that low-level winds will prevail during the unattended period. When this is done, the counterweight moment will act opposite to the lean and tend to correct it. As it is unlikely that loaded moment will exceed the counterweight moment, loading the jib overnight is not a suitable alternative. If no clear corrective progress is evident, more positive measures must be taken such as ballasting the footing block. Monitoring readings should be taken frequently during corrective actions.

Plumbness of a tower crane mast should be measured in the two principal axes. When the upper works are in place the crane is generally imbalanced; accurate readings require that either the measurements be taken with the crane perfectly balanced—this may be difficult to gauge—or with the boom pointed in the same direction as the observer.

Example 6.4

Check the net lean of a tower crane having the base-elevation monitoring marks 25 ft (7.6 m) apart. Initial and follow-up readings R_1, R_2, R_3 and R_4 at the top of the crane footing (clockwise around the footing) were made as follows:

n	R_1	R_2	R_3	R_4
0	132.42	132.38	132.41	132.41
3	132.41	132.37	132.38	132.39

Solution

Initial mast was measured, and the following values, in units per thousand, were calculated:

$$\text{Mast lean} = \begin{cases} 0.25 \text{ from side 1-2 toward side 3-4} \\ 0.33 \text{ from side 2-3 toward side 1-4} \end{cases}$$

The diagonal mast-lean values can be immediately determined from the measured data. The lean from 2 toward 4 is $(0.25^2 + 0.33^2)^{1/2} = 0.41$ and from 3 toward 1 it is $(0.33^2 - 0.25^2)^{1/2} = 0.22$.

From Eq. (6.18), the corner elevation changes are

$$\Delta_1(3) = 132.42 - 132.41 = 0.01$$

$$\Delta_2(3) = 132.38 - 132.37 = 0.01$$

$$\Delta_3(3) = 132.41 - 132.38 = 0.03$$

$$\Delta_4(3) = 132.41 - 132.39 = 0.02$$

The settlement lean of each side is given by Eq. (6.19***a***).

$$C_{21} = 0 \qquad C_{23} = \frac{(0.03 - 0.01)1000}{25} = 0.80$$

$$C_{43} = \frac{(0.03 - 0.02)1000}{25} = 0.40$$

$$C_{14} = \frac{(0.02 - 0.01)1000}{25} = 0.40$$

The diagonal leans are given by Eq. (6.19***b***).

$$C_{24} = \frac{(0.02 - 0.01)1000}{25\sqrt{2}} = 0.27$$

$$C_{13} = \frac{(0.03 - 0.01)1000}{25\sqrt{2}} = 0.57$$

The net leans are determined by adding the initial and settlement leans, giving appropriate consideration to direction.

$$\text{2-1} = 0.33 + 0 = 0.33 \qquad \text{OK}$$

$$\text{2-3} = 0.25 + 0.80 = 1.05 \qquad \text{OK}$$

$$\text{4-3} = -0.33 + 0.40 = 0.07 \qquad \text{OK}$$

$$\text{1-4} = 0.25 + 0.40 = 0.65 \qquad \text{OK}$$

$$\text{2-4} = 0.41 + 0.27 = 0.68 \qquad \text{OK}$$

$$\text{1-3} = -0.22 + 0.57 = 0.35 \qquad \text{OK}$$

The leans are all less than 2 units (1:500), the general-purpose plumbness tolerance. It should be noted that the calculated net leans are not geometrically consistent. This is not a cause for alarm, as the differences result from reading the elevations to the nearest 0.01 ft (3 mm).

6.4 Climbing Cranes

When a static mounted crane cannot reach sufficient height for completion of the work, it must be brought up in elevation by climbing. A *bottom-climbing* or *internal-climbing* tower crane would achieve this by raising itself within the building. A *top-climbing* tower crane would do so by adding sections to the top of the mast and bracing or guying for lateral support.

Climbing requires skill and attentiveness since improper procedures can lead to disaster. The manufacturer's manual should be read and understood. An experienced and knowledgeable crew is highly recommended; there should always be at least one key person on the crane with expert knowledge. The actual jacking should proceed slowly and deliberately with eyes watching all potential snag and distress points.

Before a climb, the climbing apparatus should be visually inspected by someone familiar with the system. The hydraulic mechanism needs to be in good working order; the climb should not proceed otherwise. Only a trained hydraulics mechanic can be allowed to make valve adjustments.

The crane must be *balanced* before the climb; this usually requires that a specified hook load be held at a specified radius, although some cranes are balanced by setting counterweight or boom positions. If a weight is suspended from the hook for balancing, the space below it must be cleared of pedestrians, workers, and traffic, as with any load on the hook. Field adjustment of the radius is almost always needed to achieve a fine balance. It is important that the crew recognizes the appearance and behavior of a balanced crane.

Climbing should not be done in high winds. If the manufacturer does not recommend an upper limit for wind, 25 mi/h (11 m/s) should be used. Wind affects the balance of the crane and puts pressure on side guides or rollers of the climbing apparatus.

6.5 Braced and Guyed Towers

When a tower crane outside a structure needs lateral support, that support can be provided by lateral braces or guys. A laterally braced mast utilizes the strength of an adjoining structure; the forces imparted by the crane must have an adequate load path down through that structure to earth. Where there is no suitable structure alongside, guying to anchorage points on the ground might be considered instead. Braces and guys are applied to support the mast laterally at vertical intervals as it increases height by top climbing. These measures might also be used short term to secure a fixed-height freestanding crane in preparation for extreme winds.

Guying and bracing systems give rise to large forces that must be resisted by both the framework of the tower and the supporting structure. Guys must be anchored either to the earth or to suitably massive elements. While brace loads can readily be sustained by the wind-resisting systems of most high-rise buildings, the local area of brace attachment may have to be reinforced to accommodate these loads. On steel-framed structures that rely on fully made connections and concrete floors to complete the structural system, temporary bracing schemes may be more elaborate.

Guying or bracing operations are commonly coordinated with the addition of tower sections to increase crane height. The combined work of raising and securing the crane is outlined in a climbing schedule which also links the construction schedule to the operation (Figure 6.20). If the building is permitted to move ahead of the crane, the crane will be obstructed by the rising structure. If the crane moves too far ahead of the building, there will not be structure high enough to receive the bracing. Devising a *climbing schedule* requires knowledge of crane limitations and familiarity with the pace and practices of the construction operations.

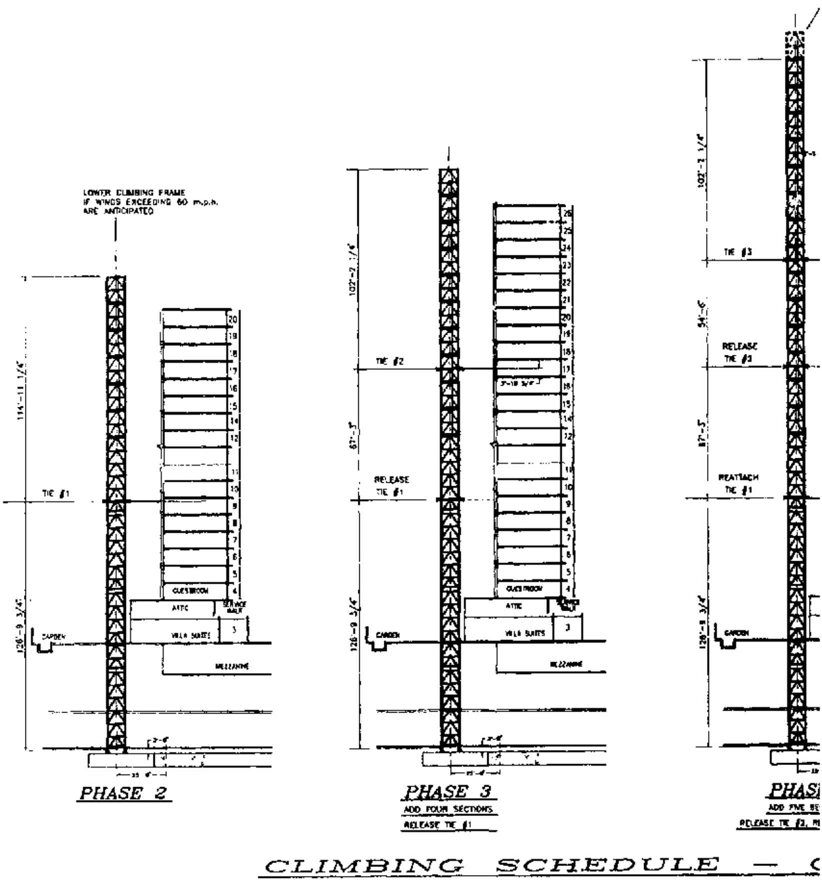
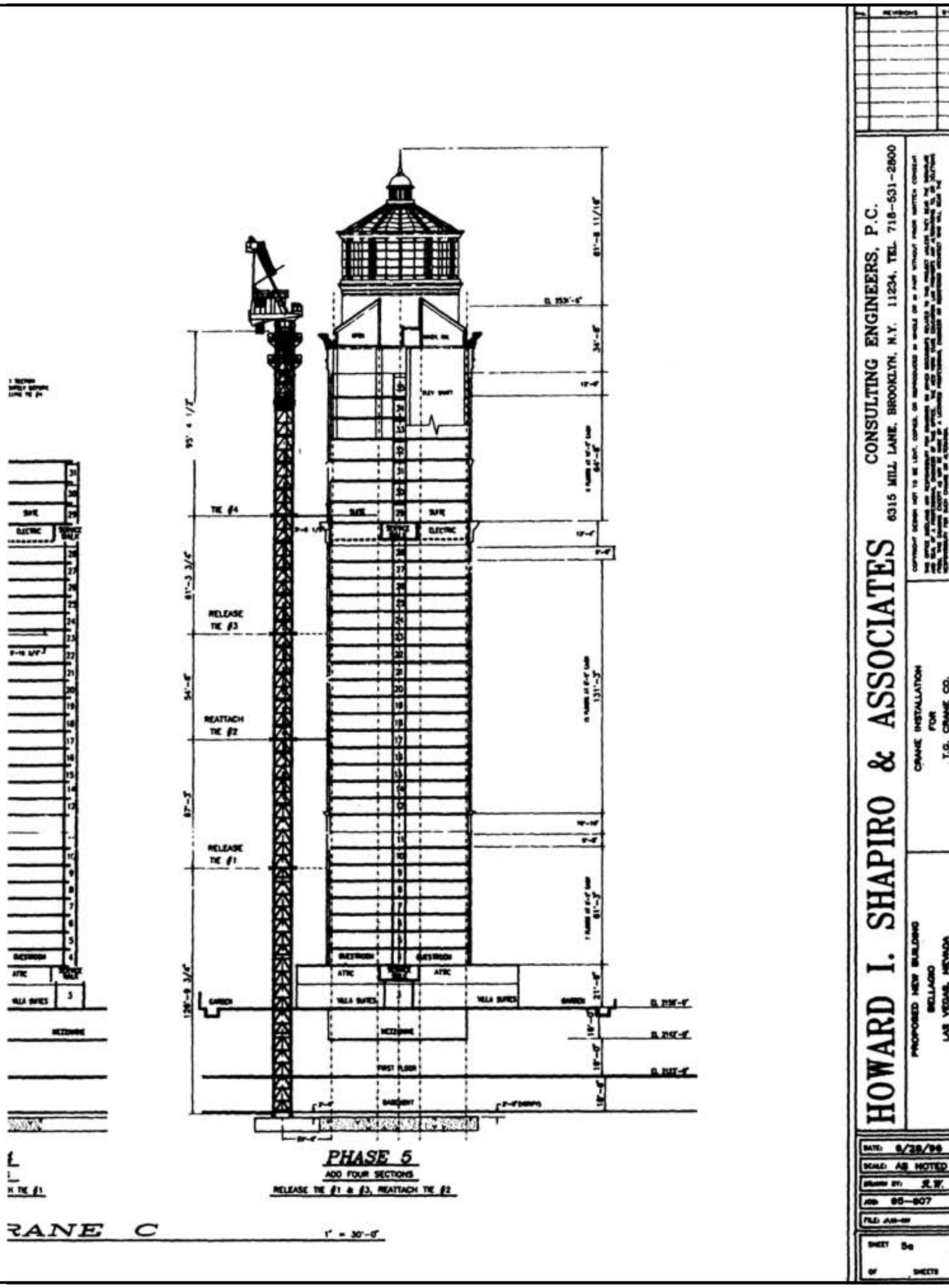


Figure 6.20 Climbing schedule for an externally mounted tower crane building a high-rise Las Vegas Hotel. Wind controlled how high the crane could rise above the top tie.


PHASE 5
ADD FOUR SECTIONS
RELEASE TIE #1 & #3, REATTACH TIE #2
1" = 30'-0"
RANE C
TIE #4
RELEASE
TIE #3
REATTACH
TIE #2
RELEASE
TIE #1
GUESTROOM
ATTIC
VILLA SUITES
GARDEN
MEZZANINE
FIRST FLOOR
BASEMENT
SAFE
ELECTRIC
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
6315 MILL LANE, BROOKLYN, N.Y. 11234, TEL 718-531-2800
PROPOSED NEW BUILDING
BELLAGIO
LAS VEGAS, NEVADA
CRANE INSTALLATION
FOR
T.G. CRANE CO.
SCALE: AS NOTED
SHEET 5a

Top Climbing

Top climbing, described briefly in Chap. 2, is a process of adding height to a crane mast by jacking the superstructure and inserting a section. The sections are relatively short, between 10 ft. (3 m) and 20 ft. (6 m)—and the operation is repeated until the required height is reached. Though the process varies, all methods require the crane to be in a balanced condition. Generally, hydraulic rams raise the upper portion of the crane in conjunction with the placement of a new section of tower above the existing tower.

Each manufacturer has a proprietary top climbing system. In the most common setup, a *climbing frame* forms a sleeve around the tower (Figures 6.21 and 2.41). There is an opening on one face of the climbing frame to permit the insertion of a tower section (and conversely its removal). A trolley beam, platform, or other means is furnished on the open side to temporarily hold a tower section that has been hoisted on the crane hook. Hydraulic rams then raise the climbing section until it engages the *turntable* and picks up the weight of the superstructure of the crane. Extension is continued until the height is sufficient to slide the new section of tower into place. Last, the climbing frame is lowered to bring the weight of the superstructure onto the new tower section, and it is secured.

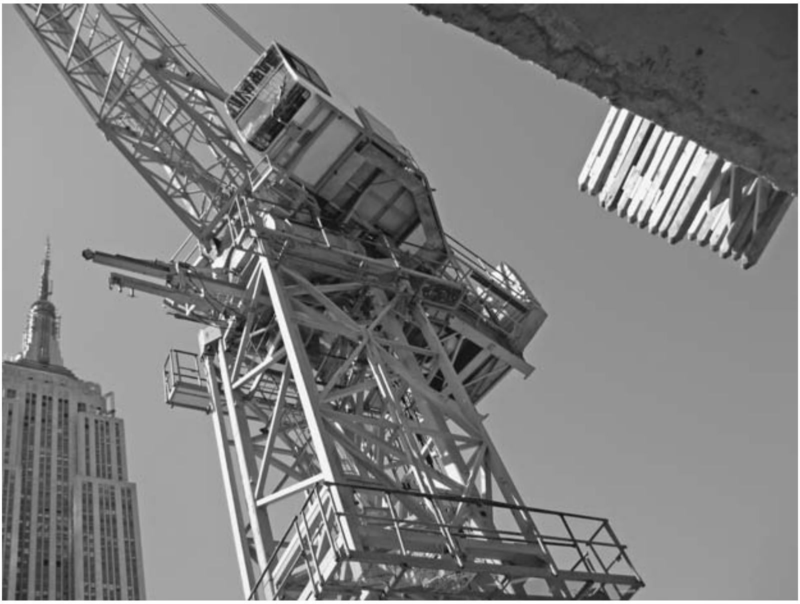


Figure 6.21 Climbing frame for a Liebherr 550 HC tower crane in the retracted position. The projecting trolley beam is used to carry a tower section in or out from the mast just under the turntable. Platforms are located where access is needed for operation of the equipment and making connections. (*Dominique Singh.*)

Often bracing or guying cannot be installed until after the crane has been top climbed. Thus the tower may temporarily exceed the unbraced height limitation. This does not present a problem as long as there will be no high winds while the crane is in this vulnerable condition. Limitations should be noted on the *climbing schedule*. Weather forecasts must be checked before climbing to excess height.

Braced Towers

A *braced tower crane* requires *mast tie* attachments to a structure much more massive than the crane itself (Figure 6.22). The mathematical assumption is that the building is rigid and that the tower flexes. Although this simplification is suitable for calculation of bracing loads, it is contradicted by the motion one feels standing on a slender

FIGURE 6.22 A crane braced to the outside of a high-rise reinforced-concrete building. Curtain wall is erected around the tie struts. During the jacking-down operation, the struts must be removed without causing damage to the delicate wall system. (*Dominique Singh.*)

building to which a working tower crane is attached. All buildings will flex and twist in varying degrees under the influence of a tower crane, particularly one that is mounted outside. The magnitude of side sway will depend on the flexural rigidity of the structure; the extent of twisting will be affected by both torsional rigidity and by the location of the crane with respect to the lateral bracing system of the building.

Expected motions should be springlike; large or sharp motions should be investigated. The sway of the building can be measured with surveyors' instruments during crane operation and the results checked by the structural engineer.

When planning tie spacing, consideration should be given to the preparatory work that must be completed before it is installed and loaded. Cast-in-place concrete structures should have test cylinder breaks or in situ testing to assure that the material anchoring the mast tie has reached sufficient strength. Added local reinforcing or post-tensioning may be required to distribute tie loads into the structure. Steel framed buildings will necessitate completion of connections in the vicinity of the crane tie and perhaps stiffening or reinforcement of the area and temporary braces to levels below. All of these measures must be clearly defined in advance, subjected to the scrutiny of the structural engineer, and coordinated with the *climbing schedule*.

On a steel-framed building, the extent of temporary bracing needed to support crane lateral loads is frequently determined by the progress of construction. Braces might be required to reinforce the steel structure where connections have not been completed or metal decking has not been installed. The subcontractor using the crane typically controls this follow-up work and therefore has a measure of control over the extent of temporary bracing. On the other hand, the extent of bracing needed might be governed by the progress of shear walls or concrete decking, work performed by another trade.

Two factors control tie spacing: the tower alignment to the building and the permissible tower height above the top mast tie. A point on the tower suitable for attachment of a tie must be found that aligns with a suitable elevation for attaching a tie to the building. On most towers, ties can be attached only where horizontal cross-members (panel points) come into the tower legs. On some, reinforcement or internal braces can be installed where ties do not align with panel points. Coincidence of suitable tower and building elevations rarely happen at every floor; therefore ties are hardly ever located at optimum spacing (Figure 6.23). Maximum spacing is limited by the height of the building already erected, which in turn is limited by the tower-crane height. The spacing usually works out to be from 40 to 90 ft (12 to 27 m). A single mast tie may suffice to raise the height of a crane for completion of a building of 15 stories or so, whereas 60 stories may require a half dozen or more (Figure 6.22).

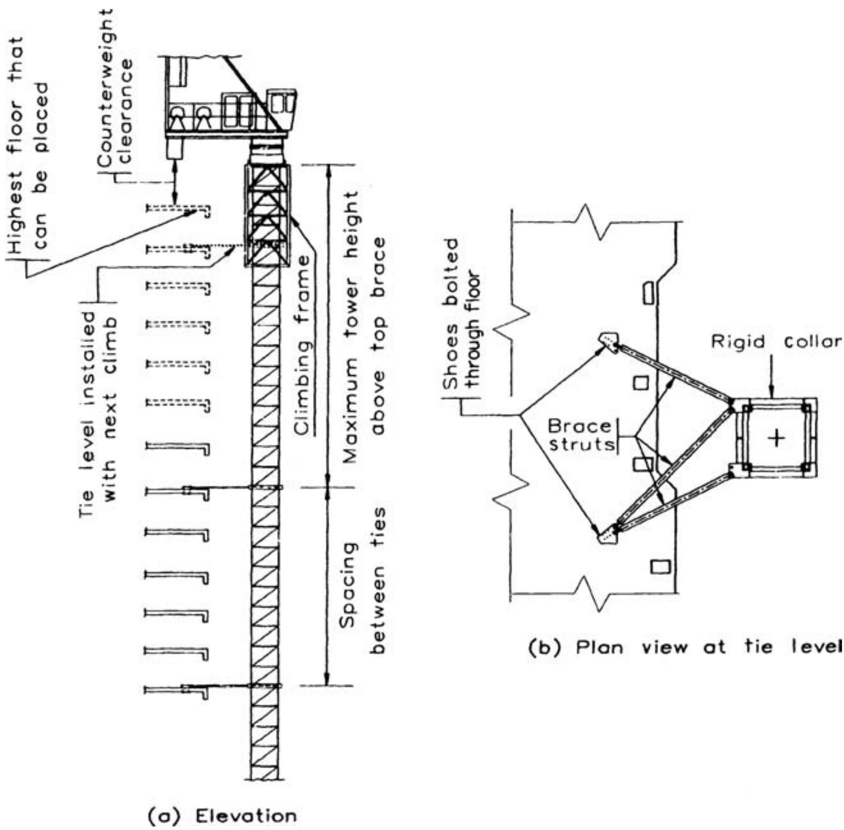


FIGURE 6.23 Top climbing crane mounted outside a building. (a) Tie spacing is controlled by the crane clearance above the building and by the requirement that tie elements must align to suitable elevations on both the mast and the building. (b) Schematic view of a brace such as in Figure 6.22. The reinforced concrete floor acts as a diaphragm to distribute crane forces into the building.

Ties should be spaced apart as generously as conditions allow. Motivation for this is based both on economics and engineering. First, ties are relatively expensive to install. Second, close spacing of ties tends to magnify counterflexure forces causing one brace to "fight" against the next.

The principal elements of a mast tie are a *collar* and struts. Most tie strut arrangements have at least three struts because this is statically determinate. Being the minimum, three is most economical (Figure 6.24). Struts are not connected directly to the crane mast, as the mast framing is not robust enough to sustain the strut loads. Instead they are pinned into the collar that has been fitted tightly around the mast. The collar is a stiff "picture frame" that distributes brace forces concentrically to the legs and inhibits racking distortion

Figure 6.24 A collar and brace support an externally mounted tower crane. The collar, suspended from the mast with slings and turnbuckles, is very rigid to prevent the tower from distorting. (*Dominique Singh*.)

of the tower. The collar is wedged with hardwood or steel to assure that the substantial forces pass squarely to the legs.

The crane vendor usually provides struts, collar, and associated hardware, but sometimes it is the responsibility of the user. Vendor-supplied hardware is generally modular and adjustable. A user is more apt to furnish "rude and crude" tie struts that will be discarded after one use.

A collar may weigh several thousand pounds, and the weight of the struts beasrs on it as well. The collar must be adequately supported on the mast; often it is hung by cables with turnbuckles (Figure 6.24) or by chain blocks. Though it hangs on four or more suspenders, any two opposing suspenders should be capable of carrying the weight. The manufacturer's specifications for installing and supporting the collar should be followed.

Vendors often supply a set of "shoes" that allow the tie struts to be secured to a floor of a concrete building. The floor is normally the place of choice for securing tie struts because of its diaphragm strength and ease of access. Loading is out of the plane of the floor diaphragm and induces slab bending, though the designer tries to minimize this effect. Shoes are secured with high-strength through-bolts. Friction-grip connections may be preferred to minimize strut movement, but if not, then lining the holes with pipe sleeves reduces the otherwise high bearing stress on the concrete. Shear walls and columns are also used sometimes for tie attachment points.

On a steel building, tie struts are typically connected into the framing system with details and fabrication having been modified for that purpose. Naturally, the cost to accommodate the crane ties is lowest when introduced early in the detailing and procurement schedule.

Loads acting on tie struts should be evaluated for both the in-service and out-of-service load regimes. In each, the line of force acts through the center of a mast in any random orientation. An analyst needs to determine individually the force orientation that induces the highest load on each strut and on each connection point to the building. Additionally, the reaction to slew torsion is added to the in-service loading at the top tie.

The top mast tie experiences the highest level of loading with the load on the second tie somewhat less and acting in the opposite direction. Below the top two ties, the loads are relatively small. As the crane climbs and new ties are added, the exposure of each tie to loading changes accordingly.

The permissible tower height above the top tie is controlled by out-of-service wind loading. Consequently, this height is affected by the intensity of wind and by the effective exposure area. It follows that, all other things being equal, a hammerhead crane can be raised higher than a luffing boom crane and a crane midcontinent can be raised higher than one in a coastal hurricane zone.

Under carefully controlled conditions, the installation engineer might permit the tower height to be increased over a short period of time. Installation of a tie is time consuming, typically an all-day operation, and causes an interruption of the work cycle. A temporary mast height increase might buy time for the *tie assembly* to be installed in synchrony with that cycle. In such a case, the crane installation designer must specify the limitations, and the user must be prepared to install the tie on short notice if high winds are forecast.

Guyed Towers

A *guyed tower crane* is a rarity nowadays and the skills required for installation difficult to come by. Guys can be used to raise a tower crane to great heights in a location where there is no other structure available to support the mast.

In some instances, guying is only as a contingency for extreme winds and cannot be left in place while the crane is in service. In that event, the guys might be installed on the mast and left slack or disconnected at the anchorage ends. The anchorages, or dead-men, must be in place ready to receive the guys, and definite procedures are needed to make certain that work crews will be available to complete the guying should high winds threaten. A *dead-man* is a temporary anchorage point, either existing or built for the purpose.

Suspended guys will hang in a *catenary* shape, the amount of sag depending on the tension in the rope. If loaded by wind, the tower will lean with the wind and load one or more guys, causing a reduction in catenary sag. This has the same effect on the mast as lengthening the rope. But loading the rope will also induce constructional and elastic stretch, resulting in further elongation. The combined increases in rope length cannot be permitted to let the mast deflect excessively.

Rope elongation can be controlled. Constructional stretch can be minimized by using prestretched rope, elastic stretch can be limited by using oversized rope, and catenary stretch can be reduced by preloading the guys with turnbuckles or other devices. Preloading must be balanced so that the crane remains plumb.

After setting up a guying system, preloading the ropes and plumbing the mast, it is necessary to monitor both the preload forces and mast plumbness. This should be done every few days initially and after any significant high wind, but the intervals can be extended as preloads stabilize (i.e., when the constructional stretch has been fully removed from the rope). Fittings such as rope clips should be checked, too.

During the 1970s, at a time when nuclear power plants were being constructed all over the United States, the original author designed a number of guyed tower crane installations for construction of hyperbolic cooling towers up to 400 ft (122 m) in base diameter and 530 ft (162 m) high. In each installation, a tower crane was mounted at the center of the cooling tower and climbed as the work progressed. Because the thin concrete-shell sides of the cooling towers did not have sufficient strength for bracing these cranes, sets of eight tower guys were installed at each of three levels as crane height increased and were anchored at the strong reinforced-concrete ring at the structure base. Storm winds proved to be the governing loading case for each level of guying. Because of the steep guy angles, the catenary shape of the guy ropes made the rope spring rate far from linear, and complex procedures were needed to solve for the guy loads. The guys, of course, would interact with the tower, so that tower elastic deflections also had to be determined. The problem was not only that of designing the guys; the structural competence of the tower and its connections had to be verified for the unusual installation height and local wind conditions as well. One of these installations, guyed at three levels, is shown in Figure 6.25; the four corners at each level are guyed in two directions with two-part guys.

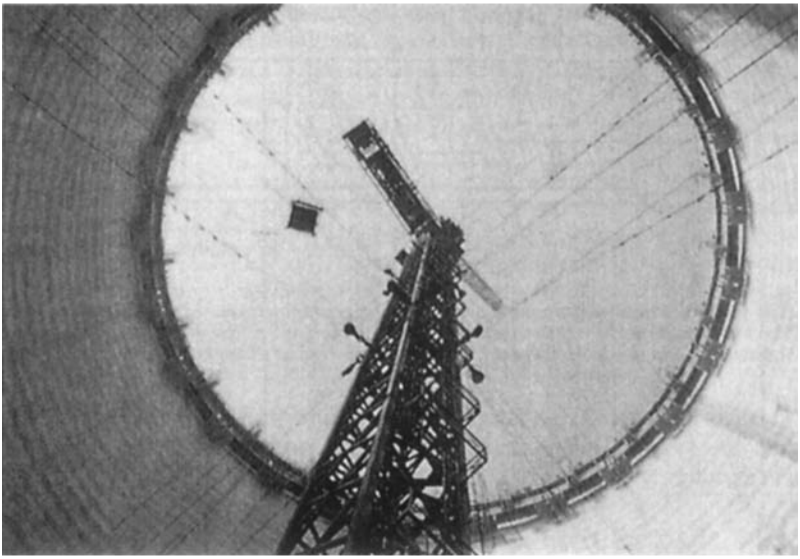


FIGURE 6.25 A guyed tower crane mast at the center of a partly completed cooling tower shell. This installation as dictated by the 400-ft (122-m) mast height. Two-part guys in two directions at each corner of each level were necessary to control mast-top movements because of coastal winds. The same installation inland might have required one less guy level.

An approximate analysis of a guyed mast is carried out to rough out the design for planning or estimating. A more rigorous analysis is done for the final design. For guys intended to carry excess wind loads, the excess wind moment M_e can be approximated from the design wind moment M_d and the wind velocities.

$$M_e = \left[\left(\frac{V_s}{V_d}\right)^2 - 1\right] M_d \tag{6.18}$$

where V_s is the site wind velocity and V_d the design velocity.

A simple guying arrangement (Figure 6.26) would have four guys radiating from the mast corners. The guys are affixed to the mast at a guying frame supplied by the crane manufacturer or in accordance with details supplied by the manufacturer (Figure 6.27). If neither frame nor details are available, the guys should not be attached before a rigorous stress analysis of the mast under storm-wind conditions has been made. Unless they are held within acceptable limits and transmitted to the mast properly, guying loads can collapse a mast leg.

The guying connection should be made high on the mast, between half and two-thirds of the distance up to the slewing circle. The guy angle θ should be kept somewhat flat, preferably about 45°, while angles in excess of 60° should be avoided.

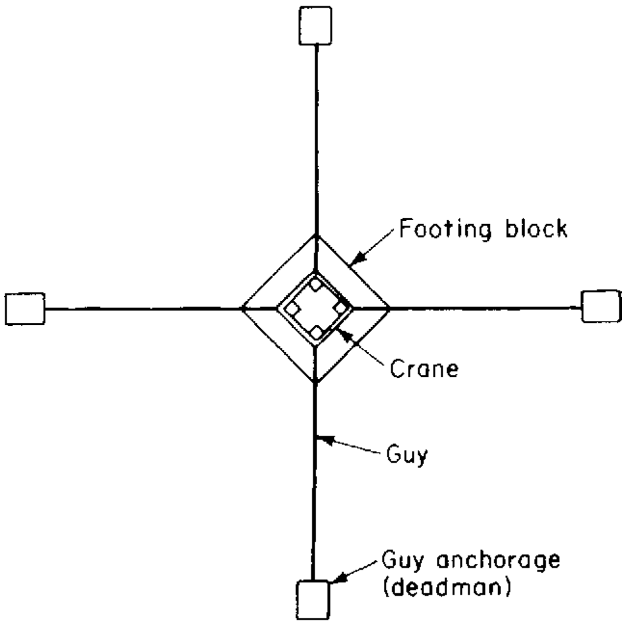


Figure 6.26 Plan view of a typical guying arrangement.

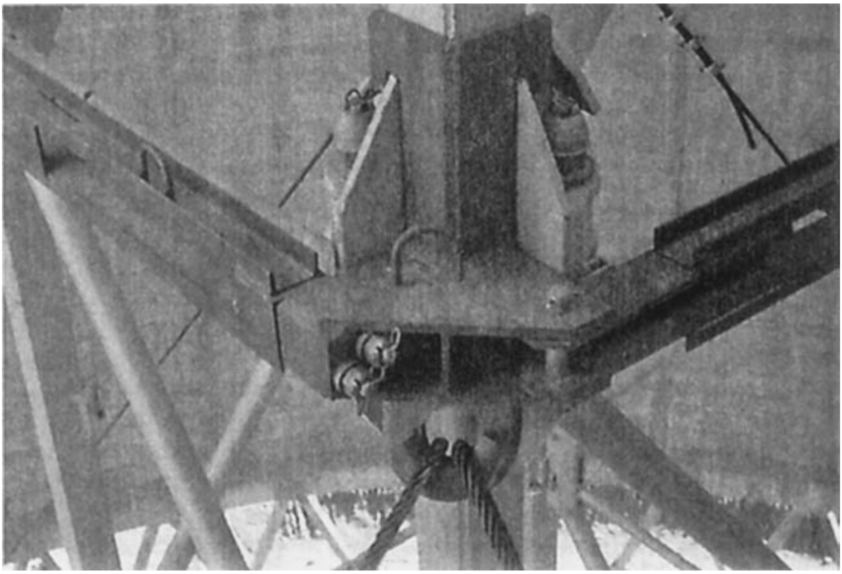


Figure 6.27 Guying frame is similar to a collar for bracing the crane, except that it must transmit both vertical and horizontal components of the guy loads into the mast without distorting the mast cross section. (*J. D. Telenko.*)

To restrain the excess wind given in Eq. (6.18), each guy must be capable of supporting a horizontal force of (Figures 6.28 and C1)

$$F_h = \frac{M_e}{h}$$

or a load in the rope of

$$F = \frac{M_e}{h \cos \theta}$$

Although this appears to be mathematically correct, nevertheless in practice the difference between the elastic properties of the rope and the mast will cause the rope to take on additional load. Approximately, then, take

$$F = \frac{(V_s/V_d)^2 M_e}{h \cos \theta} \tag{6.19}$$

At this stage of approximation, which tends to underestimate guying loads, a rope should be selected with a safety factor of at least 5.

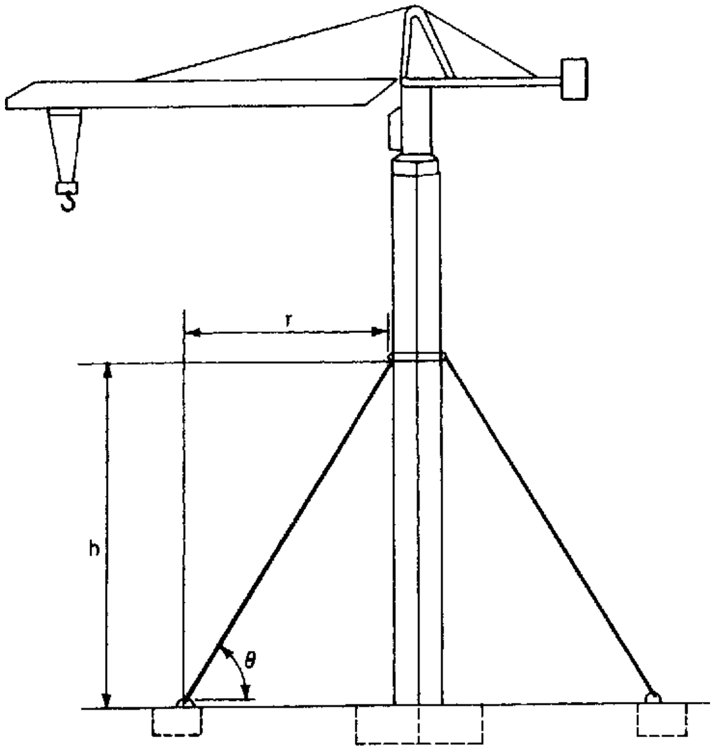


Figure 6.28 Elevation of a guyed tower crane.

Two or more parts per guy can be used to avoid use of large ropes. With multiple parts of line, sheaves or blocks must be placed to assure equalization.

Rigorous analysis of a guyed mast, performed for a final design, should take into account the catenary shape and the elastic stretch of the rope. A mathematically exact method that takes these factors into consideration is presented in Appendix C.

A complete procedure for designing the guying system for a tower-crane mast, utilizing the analysis tools presented in Appendix C, is as follows:

1. Calculate a trial value for F_h. A reasonable first estimate for $\delta - d$ is ½ to 1% of h, and θ can be used as an approximation for α. Use Eq. (C.5).
2. Choose a preload value F_{h1}; a suggestion is $F_{h1} = F_h/4$; then $F_{h2} = F_h$.
3. Select a trial guy rope using F_{h2}, Eq. (C.2), and strength factor (SF) = 3.
4. Determine Δr from Eqs. (C.3) and (C.4), noting that S changes with rope load ($S = y_0/r$).
5. Compare Δ_r with $\delta - d$; if they are satisfactorily close, F_{h2} has been found; if not, repeat the procedure using the mean of Δr and the initial value for $\delta - d$, or Δr alone, in Eq. (C.5) along with a calculated value of α.
6. Verify rope SF using F_{h2} and Eq. (C.2).

Preloading guys stiffens the mast against lateral movement. The change in length under guying loads results partly from taking up the slack from the unloaded state. Reducing initial sag requires that the guys be preloaded; but to be effective the preloading force must be measured and controlled. A tension-measuring devise, together with a turnbuckle to adjust load, is used to set the preload.

Preloading should be done with the crane balanced in calm air; alternatively to balancing, the boom can be pointed perpendicular to the guys being tensioned. The plumbness of the crane should be checked before and after preloading. The preloading from one side to the other should be balanced as closely as possible, with load balance taking precedence over perfecting the plumbness of the crane.

Adequate preloading is well advised. If insufficient, slack guys will whip under wind-gust action throwing shock loadings into the rope and the mast and perhaps permitting excessive mast lean. A highly preloaded system will feel steady and quiescent to the operator in the cab, and its components will not suffer severe load reversals that might lead to fatigue.

Rope characteristics need to be considered in order to assure a well-functioning mast guy system. The change in length that accompanies rope loading is a function of *constructional stretch* and the elastic properties of the rope. Both are reflected to a degree in the modulus of elasticity of the rope; the constructional stretch is caused by looseness in the placement of individual wires and diminishes with loading. As the constructional stretch works out of the rope, the modulus increases.

The constructional stretch can be minimized, and the modulus of elasticity thereby increased, by using *prestretched wire rope*. Prestretching must be specifically ordered when purchasing rope, and a delay in delivery must therefore be anticipated. Ropes are ordinarily prestretched at the factory by loading them to some 55% of the break strength, but on order any level of loading can be provided.

Approximate values of modulus of elasticity for several rope types can be taken as in the following table:

Type	kips/in^2	GN/m^2
6 × 19, fiber core	12,000	82.7
IWRC	15,000	103.4
Prestretched	17,500	120.7
6 × 37, fiber core	11,000	75.8
IWRC	14,000	96.5
Prestretched	16,500	113.8
Bridge rope, prestretched	20,000	137.9
Bridge strand, prestretched	24,000	165.5

Except for the bridge rope and strand, the manufacturer's literature ordinarily does not include data on modulus of elasticity or listings of metallic cross-sectional areas for the various styles and rope sizes. This is largely because the data are seldom requested. Approximate values for metallic area are given in Table 6.1.

Example 6.5

1. A tower crane is designed for freestanding installation to a height of 215 ft (65.5 m) to the hook. Out-of-service wind at 50 mi/h (22.4 m/s) at a base height of 30 ft (10 m) induces a wind moment about the base of 6415 kip · ft (8698 kN · m). Using the approximate method, estimate the guy break strength needed for 60 mi/h (26.8 m/s) winds if the guys are fastened to the mast 121 ft (36.9 m) above the base. Assume that the guying distance r = 125 ft (38.1 m) (Figure 6.28).

Solution

The increased wind moment will be given by Eq. (6.18)

$$M_e = \left[\left(\frac{60}{50}\right)^2 - 1\right](6415) = 2823 \text{ kip} \cdot \text{ft } (3827 \text{ kN} \cdot \text{m})$$

Geometry gives us the guying angle θ as

$$\theta = \tan^{-1}\frac{121}{125} = 44.07°$$

Rope Diameter, in	6 × 7 Fiber Core	6 × 19 and 6 × 37 Fiber Core	IWRC	8 × 19 Fiber Core
¼	0.024	0.25	0.029	0.022
5⁄16	0.037	0.039	0.045	0.034
⅜	0.054	0.056	0.065	0.049
½	0.095	0.10	0.12	0.088
⅝	0.15	0.16	0.18	0.14
¾	0.21	0.23	0.26	0.20
⅞	0.29	0.31	0.35	0.27
1	0.38	0.40	0.46	0.35
1⅛	0.48	0.51	0.58	0.44
1¼	0.60	0.63	0.72	0.55
1⅜	0.72	0.76	0.87	0.66
1½	0.86	0.90	1.0	0.79
1¾		1.2	1.4	
2		1.6	1.8	
2½		2.5	2.9	
3		3.6	4.1	

†1 in^2 = 6.4516 mm^2

Table 6.1 Approximate Metallic Area of Ropes, in^2†

Equation (6.19) then gives the force in the guy rope.

$$F = \left(\frac{60}{50}\right)^2 \frac{2823}{121\cos 44.07°} = 46.75 \text{ kips (208 kN)}$$

For a strength factor of 5, the required break strength (BS) is

$$\text{BS} = 46.75\left(\frac{5}{2}\right) = 117 \text{ tons (1041 kN)}$$

2. Design the guying system using the exact method in Appendix C and the following additional data (Figure C.3):

$$\begin{aligned} M_c &= 1000 \text{ kip}\cdot\text{ft (1356 kN}\cdot\text{m)} & Q &= 207.5 \text{ kips (94.1 t)} \\ W_w &= 16.2 \text{ kips (72.1 kN)} & w &= 0.23 \text{ kip/ft (3.4 kN/m)} \\ e &= 4.25 \text{ ft (1.30 m)} & H &= 221.5 \text{ ft (67.5 m)} \end{aligned}$$

Solution

The elastic parameter $k = 0.000215$ rad/in (0.00848 rad/m).

Equation (C.5) can be used to find a trial value for the horizontal component of the fully loaded guy force, but first the N parameters need to be evaluated. Using radians for all trigonometric arguments, we have

$$h_1 = 221.5 - 121 = 100.5 \text{ ft (30.63 m)} \qquad \theta = 44.07° = 0.769 \text{ rad}$$

$$\sin kH = \sin[0.000215 \text{ rad/in } (221.5 \text{ ft})(12 \text{ in/ft})] = 0.5409$$

$$\sin kh_1 = \sin[0.000215(100.5)(12)] = 0.2564$$

$$\cos kH = \cos[\sin^{-1}(0.5409)] = 0.8411$$

$$\cos kh = \cos[0.000215(121)(12)] = 0.9517$$

$$\tan kh = \tan[\cos^{-1}(0.9517)] = 0.3227$$

$$\tan kh_1 = \tan[\sin^{-1}(0.2564)] = 0.2653$$

$$N_1 = \frac{1}{207.5}\left\{\frac{16.2 + 0.23(221.5)}{0.000215}\left(\frac{0.5409 - 0.2564}{0.8411}\right) + 12 \text{ in/ft}(0.23)\right.$$

$$\times\left(\frac{100.5^2}{2}\right) - 12(0.23)\left(\frac{221.5^2}{2}\right) - 16.2(121)(12)$$

$$\left.-\left[1000(12) + \frac{0.23/12}{0.000215^2}\right]\left(\frac{1 - 0.9517}{0.8411}\right)\right\}$$

$$= 18.53 \text{ in } (471 \text{ mm})$$

$$N_2 = \frac{0.3227 - 2(0.2653)(1/0.9517 - 1)}{0.000215[1 - 0.3227(0.2653)]} - 121(12 \text{ in/ft})$$

$$= 52.48 \text{ in } (1.333 \text{ m})$$

$$N_3 = 4.25(12 \text{ in/ft})\left[\frac{1/0.9517 - 1}{1 - 0.3227(0.2653)}\right]$$

$$= 2.83 \text{ in } (72 \text{ mm})$$

For an initial value for $\delta - d$, try 1/2% of h, which is 7.26 in (184 mm); then

$$F_h = \frac{207.5(18.53 - 7.26)}{52.48 + 2.83 \tan 0.769}$$

$$= 42.35 \text{ kips } (188.4 \text{ kN})$$

when θ is taken to approximate α. Using a preload of $F_{h1} = 42.35/4 = 10.59$ kips (47.11 kN), $F_{h2} = F_h = 42.35$ kips (188.4 kN)

To select a trial guy rope, we must convert the horizontal component of load to maximum rope load using Eq. (C.2) and θ for α.

$$F_t = \frac{42.35}{\cos 0.769} = 58.93 \text{ kips } (262.1 \text{ kN})$$

$$\text{BS required} = \frac{3}{2}(58.93) = 88.4 \text{ tons } (786.4 \text{ kN})$$

Try two parts of 1-in-diameter (25 mm) 6 × 19 IWRC IPS PRF prestretched rope which has a BS of 44.9 tons (399.5 kN) per part, a metallic cross-sectional area $A_r = 0.46$ in²/part (2.97 cm²/part), and a deadweight of 1.76 lb/ft (2.62 kg/m); the modulus of elasticity is $E_r = 17{,}500$ kips/in² (120.7 GN/m²).

The chord length of the guys L_c is $(121^2 + 125^2)^{1/2} = 173.97$ ft (53.02 m), and the rope deadweight projected onto the horizontal is then

$$u' = 1.76\left(\frac{173.97}{125}\right) = 2.45 \text{ lb/ft } (3.65 \text{ kg/m})$$

Note that the guy loads for each part are now $f_{h2} = 42.35/2 = 21.18$ kips (94.21 kN) and $f_{h1} = 10.59/2 = 5.30$ kips (23.58 kN). The sag ratios are then

$$S_2 = \frac{y_o}{r} = \frac{u'r}{8f_{h2}} = \frac{2.45(125)}{8000(21.18)} = \frac{0.0383}{21.18} = 0.00181$$

$$S_1 = \frac{0.0383}{5.30} = 0.00723$$

Equation (C.3) gives the change in guy rope length induced by the loading.

$$\Delta L = 125\left(\frac{8}{3}(0.00723^2 - 0.00181^2)\cos^3 0.769 + \frac{1}{0.46(17{,}500)}\right.$$
$$\times\left\{(21.18-5.30)(1+\tan^2 0.769)+\frac{16}{3}[21.18(0.00181^2)\right.$$
$$\left.\left.-5.30(0.00723^2)]\right\}\right) = 0.48 \text{ ft } (146.3 \text{ mm})$$

From Eq. (C.4),

$$\Delta r = [(173.97+0.48)^2 - 121^2]^{1/2} - 125 = 0.67 \text{ ft}$$
$$= 7.99 \text{ in } (203 \text{ mm}) \neq \delta - d = 7.26 \text{ in } (184 \text{ mm})$$

Going back to Eq. (C.5) with $\delta - d = (7.99+7.26)/2 = 7.63$ in (194 mm) and

$$\alpha = \cos^{-1}\{[1+16(0.00181^2)+\tan^2 0.769+8(0.00181)(\tan 0.769)]^{-1/2}\}$$
$$= 0.773 \text{ rad}$$

$$F_h = \frac{207.5(18.53-7.63)}{52.48+2.83\tan 0.773}$$
$$= 40.94 \text{ kips } (182.1 \text{ kN})$$

$$f_{h2} = \frac{40.94}{2} = 20.47 \text{ kips } (91.1 \text{ kN})$$

$$f_{h1} = \frac{20.47}{4} = 5.12 \text{ kips } (22.8 \text{ kN})$$

from which $S_1 = 0.00748$ and $S_2 = 0.00187$. Continuing, $\Delta L = 0.47$ ft (143 mm) and $\Delta r = 0.65$ ft (198 mm), making the difference from $\delta - d$ only about 3⁄16 in (4.3 mm). This is satisfactory convergence, and the guy loading has been found with sufficient accuracy. Total rope load will be Eq. (C.2)

$$F_t = \frac{40.94}{\cos 0.773} = 57.2 \text{ kips } (254.4 \text{ kN})$$

and the actual rope strength factor (SF) will be

$$\text{SF} = 2(44.9)\left(\frac{2}{57.2}\right) = 3.14 > 3.0 \text{ OK}$$

Assuming that the preloading mechanism will be located near the base of the guy, the preload force needed is

$$F_b = 5.12[1+16(0.00748^2) + \tan^2 0.769 - 8(0.00748)(\tan 0.769)]^{1/2}$$
$$= 7.0 \text{ kips/part } (31.1 \text{ kN/part})$$

3. Calculate mast top deflection, guy horizontal spring rate, and the wind moment remaining at the mast base after guying.

Solution

Before Eq. (C.7) can be used, additional N values are needed.

$$\tan kH = \tan[\sin^{-1}(0.5409)] = 0.6431$$

$$\cos kh_1 = \cos[\sin^{-1}(0.2564)] = 0.9666$$

$$N_4 = \frac{1}{207.5}\left\{[16.2 + 0.23(221.5)]\frac{0.6431}{0.000215} - 16.2(221.5)(12\text{ in/ft})\right.$$

$$-0.23(12)\left(\frac{221.5^2}{2}\right) - \left[1000(12) + \frac{0.23/12}{0.000215^2}\right]$$

$$\left.\times\left(\frac{1}{0.8411} - 1\right)\right\} = 45.67\text{ in }(1.160\text{ m})$$

$$N_5 = \frac{1}{207.5}\left[\frac{0.5409 - 0.2564}{0.000215(0.8411)} + 4.25(12\text{ in/ft})\tan 0.773\left(\frac{0.9666}{0.8411} - 1\right) - 121(12)\right]$$

$$= 0.621\text{ in }(15.8\text{ mm})$$

It is interesting to note that without guys, $\delta = N_4 = 45.67$ in (1.16 m), or about 1.7% or H. With the guys,

$$\delta = 45.67 - 40.94(0.621) = 20.25\text{ in }(0.51\text{ m})$$

or about ¾% of H.

From Eq. (C.6),

$$K_r = \frac{40.94 - 2(5.12)}{0.65} = 47.2\text{ kips/ft }(689\text{ kN/m})$$

The wind moment on the crane is

$$M_w = 16.2(221.5) + 0.23\left(\frac{221.5^2}{2}\right) = 9230\text{ kip}\cdot\text{ft}(12{,}510\text{ kN}\cdot\text{m})$$

so the moment at the base of the crane will be

$$M_{\text{net}} = 9230 + 207.5\left(\frac{20.25}{12}\right) - 1000 - 40.94(121)$$

$$-40.94\tan 0.773(4.25 - 0.65) = 3483\text{ kip}\cdot\text{ft }(4722\text{ kN}\cdot\text{m})$$

4. Calculate the effect of reducing the preload horizontal reaction to 2000 lb (8.9 kN).

$$f_{h1} = 1.0\text{ kip }(4.45\text{ kN}) \qquad S_1 = 0.0383 \qquad \Delta L = 0.79\text{ ft }(241\text{ mm})$$

$$\Delta_r = 1.10\text{ ft }(335\text{ mm}) = 13.2\text{ in}$$

Try $\delta - d = (7.63 + 13.2)/2 = 10.42$ in (265 mm) and $\alpha = 0.773$ rad; from this $F_h = 30.46$ kips (135.5 kN) and $f_{h2} = 15.23$ kips (67.75 kN). This gives $S_2 = 0.00251$, $\Delta L = 0.61$ ft (186 mm), and $\Delta r = 0.85$ ft $= 10.20$ in (258 mm) for reasonable convergence (¼ in or 5.5 mm). Mast lean at guy level has increased by a small amount, but the increase is about 1/3 nonetheless. At mast top,

$$\delta = 45.67 - 30.46(0.621) = 26.75\text{ in }(0.68\text{ m})$$

an increase of some 32%. Spring rate decreases to 33.5 kips/ft (489 kN/m), while the net base moment increases by some 40% to 4906 kip · ft (6651 kN · m).

Some 10 kips (44.5 kN) of force that would have been taken by the guy ropes are instead resisted by the mast itself because of the added lean permitted by reducing the preload.

6.6 Internal-Climbing Cranes

Whereas the top climber stands apart from the building it is erecting, the internal-climbing crane mast is enveloped and supported by it (Figure 6.29). The building under construction forms the base for the crane mast. As new floors are created, the crane raises itself and is supported on them.

Commonly the internal-climbing crane is identical to the static version except for the addition of climbing gear. The crane may initially

FIGURE 6.29 This internal-climbing crane is supported in the building. It must now be disassembled and lowered to the ground by the other roof-mounted crane. (*Morrow Equipment Company.*)

be set up freestanding and later converted to internal climbing after the building height approaches the top of the mast.

To accommodate the mast and to permit climbing, a series of floor openings must be provided through the height of the building. In some instances, an elevator shaft will suit this purpose, but as a rule, on high-rise construction, this may cause an unacceptable delay of elevator installation. If existing openings are not to be used, temporary openings must be made in each floor. These are filled in after the crane passes, except for the last few that allow the crane to be jacked down a few floors at the end of the job to permit dismantling.

The crane imparts two distinct sets of loads to the building: vertical forces derive from the weight of the crane and the hoisted load, and lateral forces are reactions to overturning moment, wind shear, and swing torsion. Each of these sets of loads requires an intermediary structure to transmit the forces from the crane to the building; respectively they are a *climbing base* and a *wedge frame,* usually combined into the same assembly. That same assembly might also serve as a *climbing frame* that supports the weight of the crane as it jumps from one working elevation to the next.

A frame for all three functions, referred to still as a climbing base, is often deployed in a set of three (Figure 6.30) The bottom frame supports the crane weight and the lower wedges; a few floors above a frame is set up to support the upper wedges. And a few floors above

FIGURE 6.30 Climbing base for an internal tower crane provides support for the crane weight during operation. A similar frame at a higher floor supports the *climbing ladder*. These frames are also used for wedging. There is a subframe below the climbing base that adapts it to the particular building. (*Manitowoc Corporation Inc.*)

that a frame is set up to support the next set of upper wedges and perhaps to hang a *climbing ladder*. The crane jumps from one climbing base to the next, following a prepared climbing schedule.

A tower crane might be 200 kips (91 t) or more, a weight that cannot be supported on typical floor framing. This limitation is often overcome by distributing the weight to a number of floors through shoring. An alternative is to support the crane weight directly on shear walls or columns.

Fortunately, lateral forces are of smaller magnitude and dispersed with comparative ease, as they and are applied in the plane of the floor structure which acts as a diaphragm; the floors are strong in that direction. The crane is wedged at two floor levels, creating a couple to resist overturning moment. The wedge floors must be far enough apart to reduce the resultant forces against the crane legs to an acceptable magnitude. The upper wedge level also sustains wind shear and slewing torsion.

Positioning a crane in a building is an art of balancing several concerns.

- A location should comport with the needs of construction such as reach and jib clearance.
- The initial setup should allow for a foundation.
- In a concrete building, temporary openings must not be too close to the building edge or to other large floor openings. These openings must not materially intrude on major building structural members.
- In a steel building, temporary openings should fit between framing elements or be compatible with reworking of framing plans for affected floors.
- Temporary openings should not compromise the structural framing system.
- Crane penetrations and support hardware must repeat one directly above the other throughout the height of the structure regardless of changes in framing arrangements.
- At the end of construction the crane must be in a position where its components can be reached by a suitable crane or derrick for dismantling.

Finishing trades may be disrupted by the shoring and temporary openings, especially when the crane is located in areas of plumbing, HVAC, or electrical chases. Owners and project managers are less tolerant of these disruptions than in the past and often demand that the crane be placed in less active space or outside of the building altogether.

Figure 6.31 is extracted from an installation drawing for a typical internal-climbing crane on a high-rise structure. It shows a climbing base, wedging, climbing ladders, and shoring. The climbing ladder support becomes a wedge frame after the crane climbs and a base support for a subsequent climb. The vertical spacing of the frames should be optimized so that they are correctly placed for all three functions.

Vertical Loads

Vertical loads do not vary greatly during the operating cycle, as by far the greatest part is deadweight. In nearly all cases, less than 25% of the maximum attainable load can be attributed to live load; the effect of impact is very small in comparison to the overall weight.

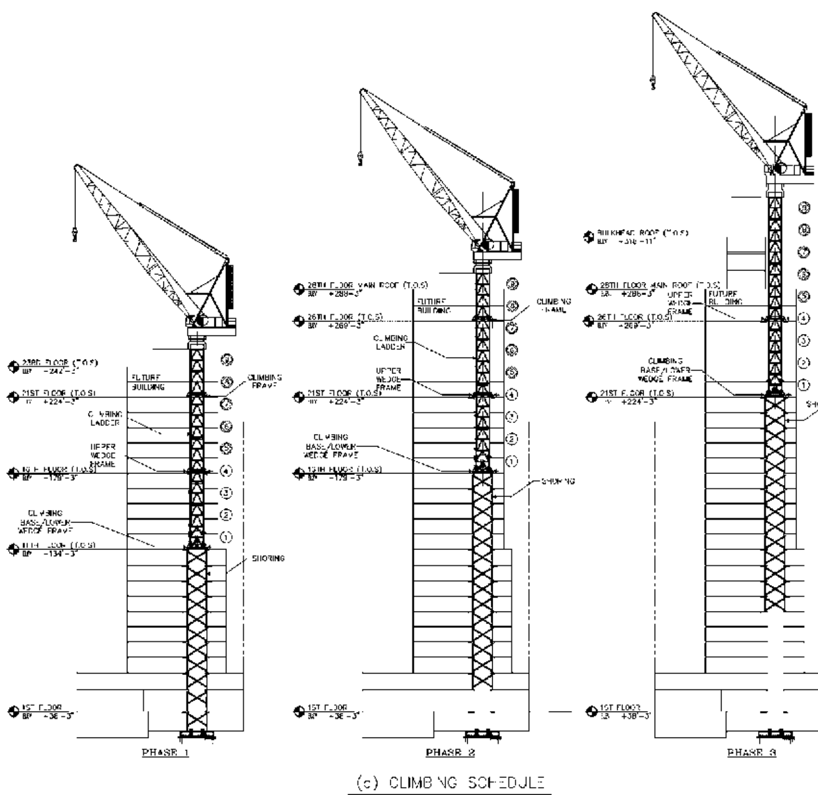


FIGURE 6.31 Extracts from an installation drawing for internal climbing. (a) The climbing, schedule provides guidance by indicating sequence for working support, climbing support, shoring, and wedging. (b) Partial elevation showing the support of the crane on the climbing base, which also carries the lower wedges.

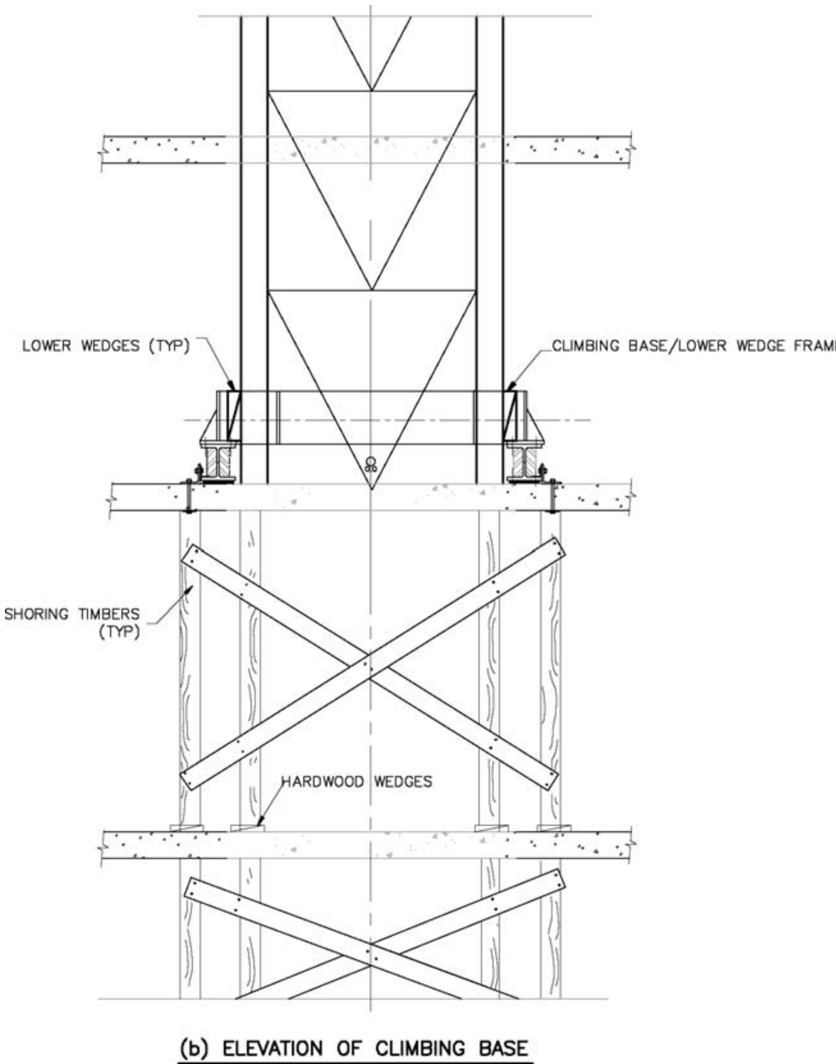


FIGURE 6.31 (*Continued*)

A climbing base is usually provided as a crane accessory from the manufacturer, but it may need a contractor-supplied subframe to fit up with the building (Figure 6.30). The base is composed of a parallel pair of load-bearing beams joined together to make a frame. Usually the crane has two bearing dogs on each side, four in total (Figure 6.32). Level support is important; without it the crane can be thrown out of plumb. If the crane is then plumbed by adjustments at the wedges two undesirable consequences ensue.

Figure 6.32 Working dog on a Potain internal-climbing tower crane, resting on the beams of the climbing base. The dogs retract when the crane is jacked up. Wedges for lateral support are visible in the near corner. (*Manitowoc Corporation Inc.*)

First, the vertical-load bearing points will become unevenly loaded; instead of experiencing 1/4 of the vertical load, any one support could theoretically receive double or more of its share. As a practical matter, the redistribution of loading will cause added elastic deflection at the more heavily loaded parts, with the likelihood that only a portion of the excess load will actually occur. It is not unrealistic, however, for any one support to be burdened by as much as 1/3 of the vertical load, or 33% more than its share.

Second, the upper horizontal reactions could be increased by a force on the order of 5% or more of the vertical load. Under certain circumstances, this can prove critically important.

It is unrealistic to assume that the concrete floor surface surrounding the crane opening will be level enough to receive the climbing base properly. A difference in elevation of ½ in (13 mm) across the opening diagonal might be considered an upper bound; ¼ in (6 mm) difference between any two supports is common. The support frame should be made level to about 1:1000, or roughly to within ⅛ in (3 mm) between any two adjacent supports and 3⁄16 in (5 mm) on the diagonals.

A concrete floor in high-rise construction, being not especially flat, level, or stiff, is hardly an ideal medium for supporting a climbing base. Steel stock is not a very satisfactory leveling means unless it is grouted after the leveling has been completed. The grout will increase the bearing area and reduce surface stresses. But a surprisingly satisfactory material for leveling can be found in abundance on most construction sites: exterior-grade structural form plywood. This material has characteristics that are perfectly suited for supporting a climbing base on a concrete floor. It will not mar the concrete surface or disintegrate, and it will adapt to small irregularities in the concrete surface. The leveling pads should be made up from several thicknesses.

Plywood functions best in this role when it is stressed in the crushing range of 500 to 1500 lb/in^2 (3.4 to 10.3 MN/m^2). Increasing the thickness of the plywood leveling packs improves their capability to equalize the crane load among the support points. Tests performed by the original author demonstrated that 5-in (127-mm) shim packs loaded to 1250 lb/in^2 (8.6 MN/m^2) permit a self-adjustment of about ¼ in (6 mm) between support points. The plywood must be replaced each time the crane is relocated to another floor because the cells will have crushed.

Shoring is used to distribute crane weight to multiple floors or to pass the weight down to an unyielding substructure. The actual concentrated load a single floor can carry should be calculated by a structural engineer familiar with the building. For planning purposes, an assumption that a 5000-lb (2-t) concentrated load is permitted at each corner on each floor is a reasonable start for determining shoring needs. Applying this rule of thumb, a 200,000 lb (91 t) crane would need nine levels of shores to deliver the weight of the hypothetical crane to ten floor slabs.

Timber shoring posts (Figures 6.31 and 6.33) are often a good choice for distributing loading to a series of floors. They have the advantage of being easily fitted and wedged into place and are readily removed when the crane climbs to another level. Addition of braces and barricades require only common lumber and very little labor to fit and place.

Extra shoring material is kept at hand to be installed before climbing to another level. When a new climbing base is set at a higher floor in preparation for the climb, it must be shored in advance, but the previously installed shores are still loaded and must be left in place until after the climb. When the climb is completed, the excess shores can be removed and kept in reserve for the next climb.

The maximum loading on a single shore will occur immediately after the climb and before the crane is plumbed. It is prudent to assume that 1/3 of the weight of the entire crane will be on any one support point for the short time that it takes to plumb the crane. For timber shores, the duration factor for the transient increase in load may also increase the allowable load in a shore by the same proportion.

The load going into each topmost shore is the support reaction less the support value of one floor slab. Each successive level of shoring will carry the load from the shore above minus the support value

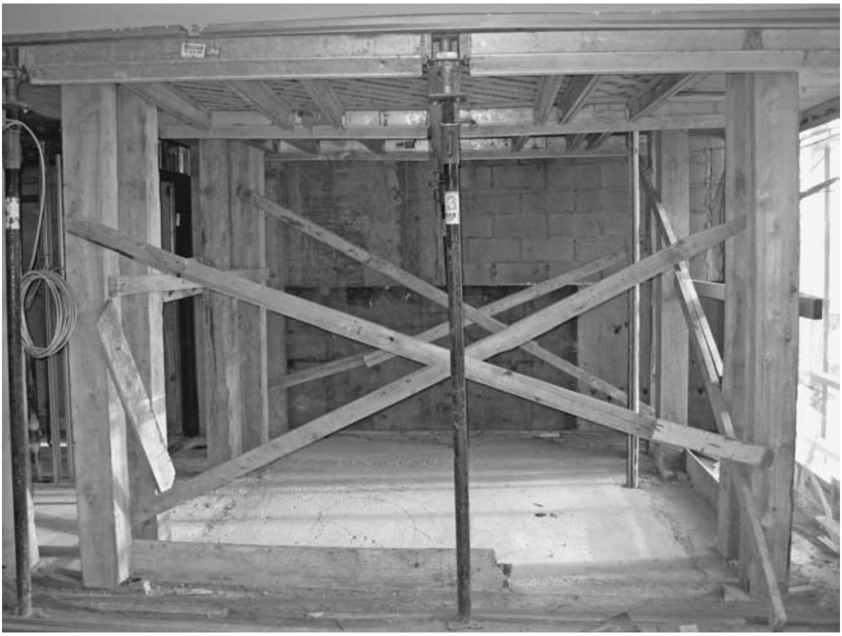


Figure 6.33 Timber shoring for a tower crane mounted inside a concrete frame building. The cross braces are used to prevent accidental dislodgement of the shores. The formwork and steel posts are for filling in floor opening created for the crane to pass through. (*Dominique Singh*.)

of the intervening floor. In theory, smaller shores can be used at successive floors below the support frame. But this is impractical in most cases, as it complicates the work in the field.

Timber shores are designed using ordinary timber-column design formulas. Only sound structural-grade lumber should be used. Wedges must be inserted to assure contact and transmission of loads between the floor slabs and shores. They must be of hardwood firmly driven into place and checked periodically for tightness.

Timber has great end-bearing strength, about double what it can support against the side grain. Consequently, headers and sills cause a reduction in the load-bearing capacity of the shoring system. Material used to fill out or extend the length of a shore must be suitably matched to it in strength, with consideration given to directional properties if the filler is of wood. The same characteristics that make plywood ideal for shimming a climbing base make it entirely unsuitable for shimming shores. Fully loaded shores may impose stresses that will crush the plywood, inducing the slab to deflect excessively and perhaps distress the floor structure. The distribution of the crane weight among load points is thrown out of balance, as some points receive loads that are much greater than the designer had intended. Cracks and large deflections in the slab are the telltale signs of such distress.

A common assumption is that the floors are equally loaded by the shores. This notion is usually reasonable but requires some caution. Its corollary assumption is that all floors deflect equally. Each shore in the vertical progression of shores, however, shortens under its compressive load. With some load dropping off at each floor level, each shore will carry less load than the one above and hence foreshorten less. The shortening in the shores must be taken up by deflection in the floor slabs so that the upper floors will deflect the most and carry a disproportional share of the load.

On a concrete building the effect of the shortening of the shores is counteracted somewhat by the aging of the concrete. With age, concrete gains in strength and stiffness; this added stiffness on the lower floors causes them to pick up a larger share of the load. Changes in floor framing bring about changes in floor stiffness as well. These occurrences can affect the overall distribution of load among floors or local distributions on a floor. Simplified assumptions can usually be used in place of complex analyses, but not without careful consideration of the structural consequences.

Moments and Horizontal Loads

Wedges used to hold climbing cranes upright react to the horizontal loads that are induced by overturning moment, wind, and swing torsion. In some arrangement, the wedges fill in the space directly between the mast and the slab. A more common arrangement is to wedge against a frame that is combined with the climbing base (Figures 6.30 and 6.32). When the crane is exposed to working, or in-service, conditions, the wind effect is small but the torsion reactions can be large. On the other hand, out-of-service loading is heavily influenced by wind exposure, but there is no torsion unless the jib is locked against weathervaning.

Any torsional loading will be resisted at the upper wedge level by two couples, as shown in Figure 6.34*a*. The torsion reaction at any one wedge point is then

$$R_s = M_s/2d \tag{6.20}$$

where d is the horizontal distance between mast leg centroids. No part of the torsion can bypass the upper wedging level unless there is a large elastic deformation permitting the crane to rotate through the upper wedge level. Such deformations are unlikely.

Overturning moment and wind reactions are best evaluated by using a free body diagram to represent the crane mast (Figure 6.35*b*). The terms M_u and V_u are the overturning moment and wind shear at

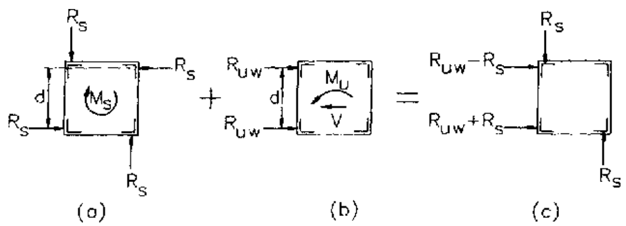


FIGURE 6.34 Distribution of upper wedging-level reactions.

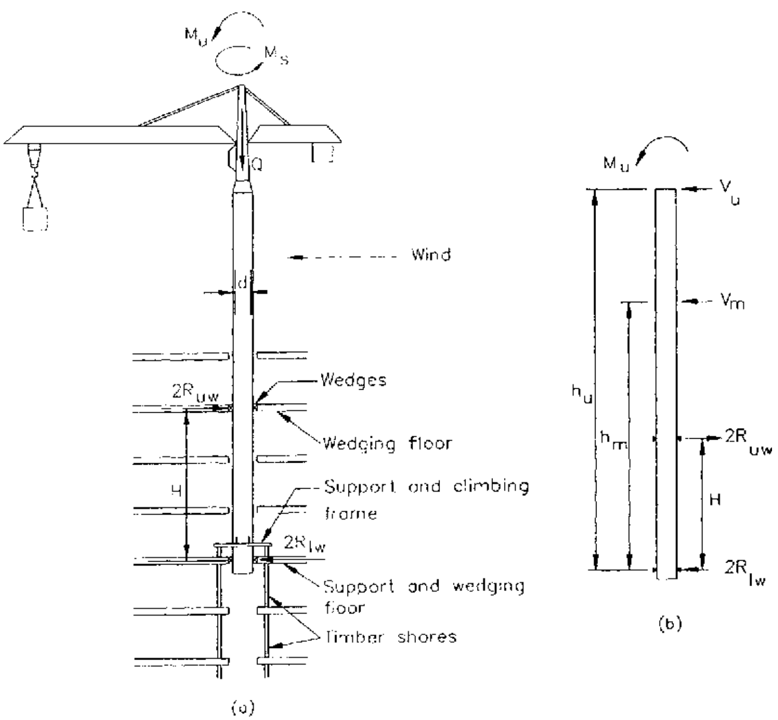


FIGURE 6.35 (a) Elevation of climbing crane installation. (b) Free body diagram for moments and wind forces.

the top of the mast, and V_m is the wind force acting on the exposed portion of the mast. Reactions can be determined by taking moments about the lower wedges. The upper wedge reaction per corner (Figure 6.34*b*) is then

$$R_{uw} = \frac{M_u + V_u h_u + V_m h_m}{2H} \tag{6.21}$$

The maximum wedge reaction at any one corner of the upper wedge floor (Figure 6.34*c*) includes torsion and is given by

$$R_{max} = R_s + R_{uw} \tag{6.22}$$

whereas the reaction at each corner at the lower wedges is

$$R_{lw} = \frac{R_{uw} - V_u - V_m}{2} \tag{6.23}$$

and is opposite to the direction of R_{uw}.

The crane manufacturer specifies the minimum wedging distance *H*. Some masts are furnished with wedge-point reinforcing while others must be positioned so that the wedges line up closely to mast cross-framing.

Example 6.6

Determine the wedge reactions for a crane with an in-service overturning moment of 2100 kip · ft (2847 kN · m) at the top of the mast, a torsional moment of 220 kip · ft (298 kN · m), wind shear at the top of the mast of 3.1 kips (13.8 kN) acting 125 ft (38.1 m) above the lower wedges, and a resultant wind force on the exposed part of the mast of 1.5 kips (6.7 kN) acting 75 ft (22.9 m) above the lower wedges. The mast leg centers are at 7.56 ft (2.30 m) and the distance between wedging levels will be three full 8-ft 9-in (2.67-m) stories.

Solution

From Eq. (6.20),

$$R_s = \frac{220}{2(7.56)} = 14.55 \text{ kips } (64.72 \text{ kN})$$

Since H is 3(8.75) = 26.25 ft (8.00 m), h_u = 125 ft (38.1 m), and h_m = 75 ft (22.9 m), using Eq. (6.21), we get

$$R_{uw} = \frac{2100 + 3.1(125) + 1.5(75)}{2(26.25)} = 49.52 \text{ kips } (220.3 \text{ kN})$$

The maximum wedge reaction is at the upper wedge level and is given by Eq. (6.22).

$$R_{max} = 14.55 + 49.52 = 64.07 \text{ kips } (285.0 \text{ kN})$$

Equation (6.23) gives the lower wedge reactions,

$$\begin{aligned} R_{lw} &= [2(49.52) - 3.1 - 1.5]/2 \\ &= 47.22 \text{ kips } (210.0 \text{ kN}) \end{aligned}$$

Horizontal loads may be transmitted to a reinforced concrete building structure through direct contact with the edge of the floor slab at the side of a crane opening. A small wedge frame might be installed at each corner to reduce the bearing stress on the concrete (Figure 6.36). For small cranes, which impose relatively low-level

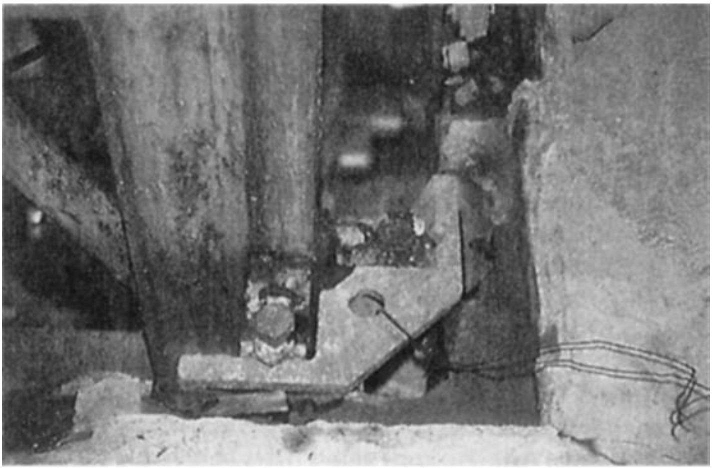


Figure 6.36 Wedging arrangement has small steel wedges that bear against the mast leg but transfer load to a casting which bears on the concrete edge with a larger area. The bolts visible on top of the wedges are used for adjusting the wedges and locking them into final position.

forces, it is sufficient to drive hardwood wedges between the slab and the mast legs. Large cranes need more elaborate wedging systems, usually provided by a multi-purpose climbing base (Figure 6.37). A wedge frame or climbing base can also be adapted to transmit wedge loads to the framework or floor diaphragm of a steel-framed building.

Figure 6.37 Wedging arrangement that is part of a climbing base. (*Manitowoc Corporation Inc.*)

A tower crane progressing upward to keep pace with building construction has its upper wedges placed at a floor that is newly constructed. The floor must have the strength developed to support the lateral forces imposed against it through the wedges or wedge frame. Minimum requirements, a specified minimum concrete strength being the most common criterion, need to be determined and stipulated for the crane to be permitted to climb.

Tall concrete buildings progress in fast cycles; 2 to 4 days per typical floor is usual. If the floor below the last-placed floor is to become the upper wedging floor the concrete might be only 2 days old at the time of climbing. When the climb is done at the end of the work day, as is often the case, the concrete will be less than 3 days old when the crane is next put into service. Modern concrete with finely ground cement can come up to design strength rapidly, with test cylinders perhaps reaching 75% of the standard 28-day strength in three days.[7] Accelerants might be used to achieve high early strength, if needed. But not all concrete ramps up in strength so rapidly. Planners must allow for plausible strength values of the concrete at the upper wedge floor; it is helpful to have local knowledge about the concrete coming from the batch plant and specific knowledge of the design mix.

Field samples should be taken and tested before the climb to make certain that the concrete has reached adequate strength for wedging. If the cylinders are kept at the jobsite and taken to the testing laboratory a few hours before test time, cylinder strength can be assumed to approximate slab strength. If the cylinders test at no less than the stipulated strength, the climb may be executed. If not, the climb should be delayed.

In order to avoid climbing delays, it is best to keep that stipulated minimum strength as low as possible. A workable value for a typical tower crane might be 1500 psi (10 MN/m^2); that would put the permissible bearing stresses at about 500 lb/in^2 (3.4 MN/m^2), which is manageable when compared to the forces to be accommodated. If the mast corners are wedged directly against the edge of the slab, the wedging detail will likely need to provide for some spreading of the load to reduce the unit stess.

It is the duty of the installation designer to adapt the crane to the particular jobsite conditions; the crane was designed in the sterile condition of a free body in space. The structure must be made suitable to support the crane loads. Field personnel must be made aware that tower cranes can impose intense dynamic loads that do not tolerate sloppy field practices. Floor openings that will take wedge loads must be accurately placed with square edges and wedge areas

[7]Cores drilled from the slab itself can be expected to show about 90% of cylinder strength.

well formed and solid. Shores must be correctly aligned and made of approved materials.

Climbing Procedures

A variety of systems have been devised by manufacturers to raise their tower cranes from one operating level to the next. In addition to the methods described below, there are numerous variations and proprietary systems. Nearly all climbers use one or more hydraulic rams; the difference among these hydraulic climbers is in the means employed to overcome the shortness of typical ram strokes.

Long-stroke rams. Extend sufficiently to raise the crane one entire typical floor height. When overly high floors are encountered, an intermediate step must be arranged.

Climbing ladders. Hang on opposite sides of the crane mast (Figure 6.38), supported from a climbing frame set above the new operating level. Pawls (dogs) on the ends of a floating crossbeam engage the ladders. The ram raises the crane by pushing down on this crossbeam. At the end of the stroke, the crane is brought to rest by engaging another set of pawls on the ladders. Retraction and re-engagement of the floating crossbeam pawls complete a cycle that is repeated until the crane has been raised to the new operating level.

Rod climbers. Two varieties: One uses smooth square rods of finite length, and the other uses threaded high-strength reinforcing bars that can be coupled to any length. Cranes that lift themselves on square rods operate similarly to ladder climbers. Rods are

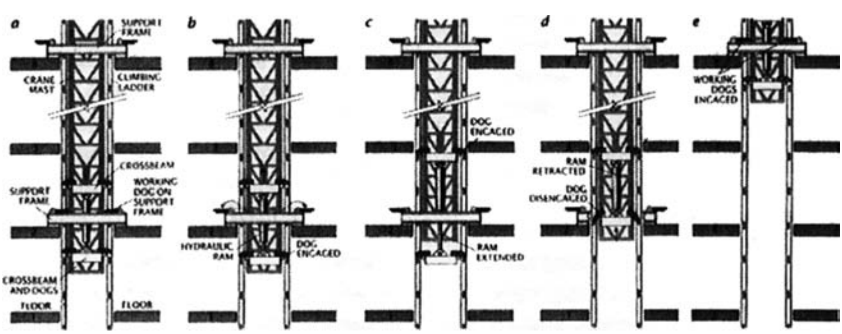


FIGURE 6.38 Sequence of an internal climbing operation. (a) Working dogs impose the crane weight on the lower support frame. (b) Climbing dogs linked to the hydraulic ram engage the climbing ladder and lift the crane. (c) A second set of climbing dogs is engaged after the crane has been raised by one rung. (d) The hydraulic ram is retracted to reengage its dogs on the next rung. (e) After a number of repetitions, the crane reaches a sufficient elevation to come to rest on the next support frame. (*Reprinted by permission of Scientific American.*)

suspended from a climbing frame; the crane climbs the rods in strokes of the hydraulic rams while secured to the rods using jaw-type grippers. On a threaded-rod system, the threads are used to transfer the weight of the crane to the rods.

Tube climber systems. Work like a climbing ladder in reverse, pushing the crane up the rungs of a stiff center post instead of climbing the suspended rungs. A square or round tube column rests atop a crossbeam that engages the crane support frame at the old support level. This column is furnished with holes that are spaced vertically to match the ram stroke, for the purpose of receiving pins. A long pin is inserted through the lowest hole and projects through both walls of the column. The rams, connected to a floating yoke, push down on this pin to raise the crane. At the end of a stroke, a second long pin is inserted, and this engages a second yoke that is fixed to the crane. Retracting the rams ends one step in a cycle that is completed when the crane reaches its next resting level. The tube climbing system is very easy to operate and quick to complete a climb, but the climb is limited to the height of the column, usually 25 to 30 ft (7.6 to 9.1 m).

Winch climber systems. Each has a climbing frame with two or four winches from which the climbing base is pulled up to a new operating level. The winches must be synchronized to keep the crane plumb and hoist ropes equally loaded.

Before and After the Climb

Except for cranes that climb one floor at a time, each climbing operation must be preplanned and a *climbing schedule* prepared (Figure 6.31). The schedule will indicate each floor level that will become a vertical-load support level and each level for horizontal support. In addition, it shows the maximum floor level that can be built before the next climb becomes necessary. In this way, work crews will know which levels will require such things as cast-in-place inserts, climbing and support frames, and shores.

Before a climb can be started, climbing and wedging apparatus are installed at the next level. The future upper wedge floor must be capable of supporting the lateral loads. For a reinforced concrete structure, it is a question of the concrete floor reaching a specified strength. On a steel-framed building lateral strength may depend on the completion of connections and possibly decking or bracing. A preclimb inspection of climbing gear, support hardware, and associated building framing should be performed. Climbing should be postponed if the wind is excessive: the limit should be no more than 25 mih (11 m/s) if not specified by the manufacturer. A crane in the midst of climbing is particularly sensitive to wind disturbance.

After balancing, wedges are loosened and climbing begun. Movements must be watched closely to assure that the motion is uniform without lateral drift, the crane is balanced and unobstructed, and dogs engage properly at each intermediate step.

On completion of the climb, the mast is brought into plumb and firmly set by adjusting and tightening the wedges. At least daily, wedges and shores should be checked for tightness.

6.7 Traveling Cranes

A *traveling tower crane* is erected freestanding on a base frame and ballasted by the user to accommodate in-service loads. When out of service, it may need to be parked and anchored down to prearranged storm ballasting blocks or guyed to resist storm winds. An installation might be on the ground or mounted on another structure such as a building roof.

The base travels on railroad-type rails set to a very wide gauge. At each corner of the base one or more wheels is provided; when more than one wheel is used, they are mounted in a *bogie* that will equalize the load on all wheels at any one corner (Figure 6.39).

Figure 6.39 Bogie, wheels, and rail support for a tower crane on a chassis base. (*Manitowoc Corporation Inc.*)

Some crane manufacturers offer options on the number of wheels to be placed at each corner (Figure 6.40) of any one crane model. As the number of wheels increases, the weight of track and the number of track supports need to decrease. This can have significant ramifications for installation cost, particularly if soil conditions are poor.

Crane rails can be supported in a number of ways, including wooden ties on stone ballast (in this case the term ballast is used to

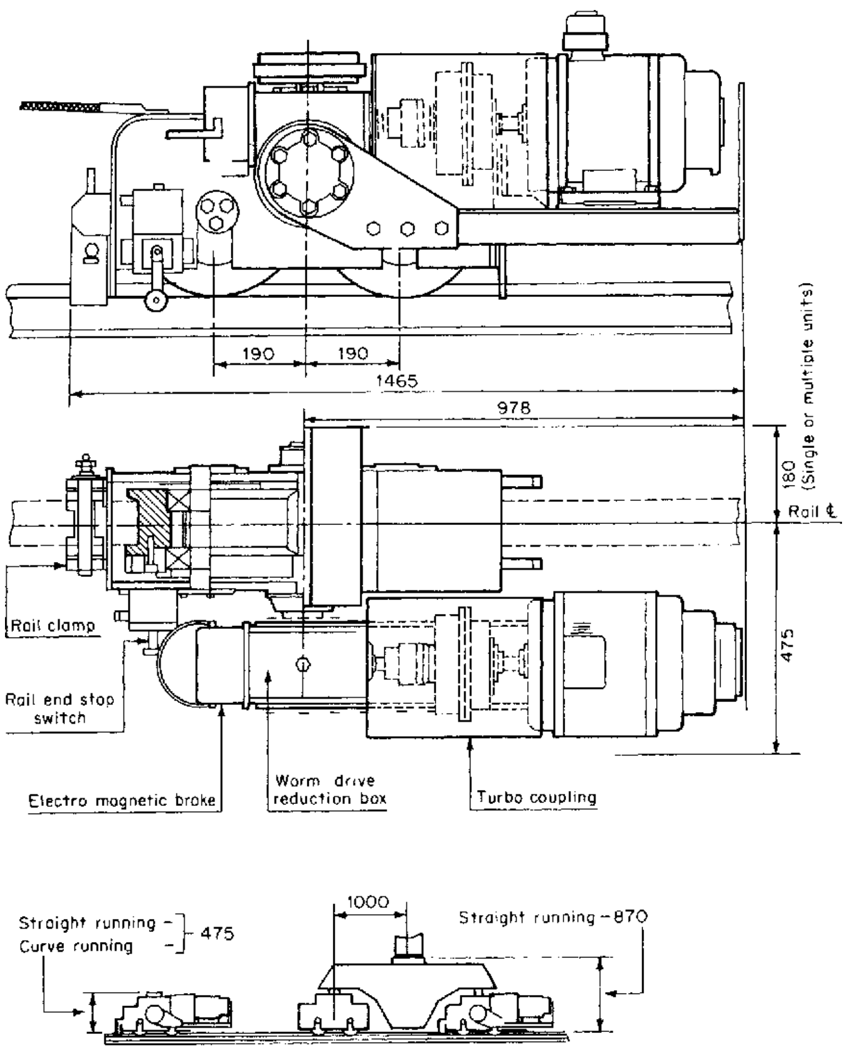


FIGURE 6.40 Optional arrangements of wheels showing one- and two-bogie sets; the dimensions are in millimeters. (*F. B. Kroll A/S.*)

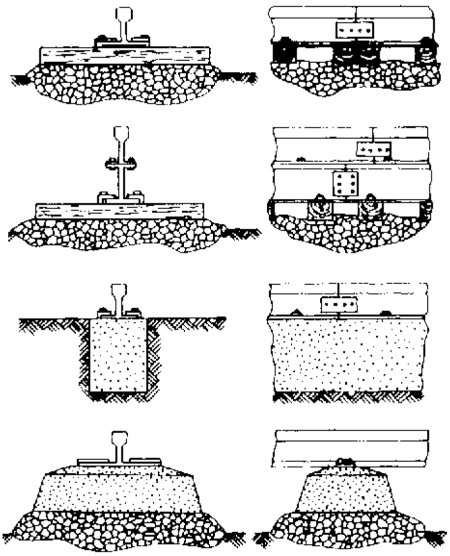


FIGURE 6.41 Several methods for supporting traveling crane rails. (*F. B. Kroll A/S.*)

refer to the bed of material placed between the tie and the native soil or sand base), a continuous steel beam on wooden ties and stone ballast or on concrete footings, or a continuous concrete footing or concrete sleepers on stone ballast (Figure 6.41). The best system is that which will support the crane properly at least cost; this will be a function of crane wheel loads, soil conditions, and availability and cost of the materials at the jobsite. The crane manufacturer provides the wheel-load data, but the installation designer must make the decisions from that point on.

The spacing of sleepers or ties can be determined from rail strength and the wheel loads. For multiple-wheel arrangements, some continuity can be taken into account, but we suggest that supports outside of the bogie should be taken as simple. Deflection should also be checked to avoid lifting the ties off their beds.

Track splices are designed to carry only shear loads, so that splices must be centered between close-spaced supports or placed directly over a support. The spacing must be set so that the two rail ends do not differ in elevation (as a result of deflection or any other cause) as the wheel passes over the splice. This will prevent horizontal impact forces from occurring, forces that can be quite significant given the inertia of the tall crane above.

Rails must be laid to comply with the tolerances given by the manufacturer or specified by code (see Sec. 4.5). There are strict limits to variations permitted in gauge, in elevation along the tracks and between the tracks, in straightness, and in slope.

Crane rails can be laid to curves but only if the bogies are designed to permit it. Centrifugal forces which develop as the crane travels a curve can have an important effect on stability. The manufacturer must supply data for minimum radius of curvature that will permit safe travel at the speed the crane is capable of attaining.

Curved track as well as slewing forces, wind, and rail misalignment induce lateral forces on the rails. Rail strength and anchorage must be sufficient to restrain these forces. Magnitudes, however, are not easily determined; the crane manufacturer's recommendations should be sought and followed.

On poor soils, track differential settlements can be a problem as they may cause track elevations to deviate from permitted tolerances and endanger operations. It would be wise to monitor elevations at marked points. This will show whether settlement is ongoing or stabilizing. With wooden ties, settlements can be corrected by jacking the rail and tie and resetting the stone ballast. For concrete supports it may be necessary to install steel shim plates with sufficient contact area to prevent the concrete from being crushed.

The parking area must be designed and constructed in advance of crane erection. It will consist, for most cranes, of an area with close support spacing for the rails that will be capable of resisting the storm-wind compressive wheel loads. In addition, there must be four buried ballast blocks to which the crane can be tied down by means of cast-in fittings. In U.S. practice, the buried ballast together with the traveling ballast must be capable of counterbalancing 1.5 times the maximum overturning moment (see Sec. 6.2).[8]

At the ends of the tracks, trippers are set that will automatically cause the crane travel brakes to engage. At a distance somewhat beyond the crane stopping distance, end stops, or bumpers, are installed as a last means to prevent the crane from running off the rails.

6.8 Erection and Dismantling

For both real and symbolic reasons, erection or dismantling of a tower crane is a major event in the progression of a construction project. The actual event consumes considerable resources of manpower, time and space. Placement of the crane signals to the world at large that major construction is about to start in earnest; its removal suggests that major structural work is winding down. Time allocated for erecting or dismantling may be slim, perhaps it is within a time slot allocated for a street closing or a weekend when other trade workers are absent from the site, and may lead to a temptation to rush. Thorough planning is the order of the day. Difficult site conditions such as

[8]See ASME B30-3-2009.

shown in Figure 6.3 or 6.17 need to be confronted during planning so that they can be managed and risky or costly complications avoided.

Tower-crane manuals are seldom written from a rigger's perspective. They typically present idealized erection conditions and schemes that cannot always be replicated under jobsite conditions. Often the manuals are literal translations from a foreign original; the intention can become cloudy or obscure particularly where specialized terms or jargon are used.

Erection weights are not always as clear as they could be. All too commonly, riggers work with inaccurate information. The manual may give an extensive list of subassembly weights that in practice need to be compiled and added. Other times, weights of standard assemblies are listed, but in the field additions and subtractions must be made for subassemblies added or deleted to suit erection crane capacity or other jobsite restrictions. If reliable information is not available, assemblies should be put on a scale before erection and the results recorded with appropriate detail of what is included.

Centers of gravity are asymmetrically located on some components. As a consequence, the placement radius of the erection crane may easily be miscalculated. During disassembly, picking eccentrically with respect to the CG is almost inevitable. Recording the connection points and rigging arrangements for these components during assembly will help during disassembly.

Some components need to be picked level to align mating surfaces, pins, or bolts. For example, the machined upper mating surface of a *turntable* bolt circle might be damaged if not set in place dead level.

Making do with whatever slings and shackles happen to be in the gang box can be dangerous. Dedicated rigging should be considered for critical components to avoid mismatching and assure correct specification of the slings. Loads acting on the rigging need to be given ample attention. Four point lifts do not necessarily have equal loads on all legs. On an asymmetrical pick, two legs may take a disproportionate share of load. On a symmetrical pick the slings on two opposing corners of a four-part spreader can theoretically carry the entire load. Slings and rigging hardware should be sized to allow for these contingencies. Some riggers take 1/3 of the picked load per leg as a rule of thumb.

Rigging attachment points should be planned in advance and installed with care. The boom, in particular, is delicate and should be choked with softeners or synthetic slings. Several problems can arise from poorly secured rigging.

- Slippage can cause the load to shift while it is being handled.
- Contact forces from slings or shackles can dish or distort thin elements such as housings or delicate parts such as booms lacings and machinery.

- Awkwardly placed attachments can cause overstress and even permanent deformation of crane components. Side-loaded lifting lugs and boom sections choked between panel points are two common examples.
- Chokers bent over sharp corners with no softeners may be damaged or even fail.

The planner on the ground should think about the weight of rigging and tools to be carried around by the riggers; they should contemplate how the riggers in the air are to be poised as they secure slings, torque bolts, or knock pins. Some preparation might be done on the ground to reduce the airborne work. Fall protection measures must be developed.

The manufacturer's sequence must be carefully studied in planning jib and counterweight placement or removal. Moments acting on the crane mast are naturally unbalanced towards either the jib or the counterjib; a measure of front-to-rear balance must be maintained. Some counterweights may need to be set on the counterjib before the jib can be installed, or some counterweights removed before the jib is removed. A cavalier regard for sequence could result in structural damage or failure, or the crane might overturn.

Mast bolts should be at least partly torqued before the counterjib or jib is placed and, vice versa and kept tight until these imbalanced components are removed. Failure to do so may cause bolts in leg connections to carry loads unevenly with possibly dire consequences.

Wind and darkness are both enemies of the crane rigger. Critical work, such as boom erection, must be scheduled with this in mind and ample time allowed to secure everything before dark unless suitable lighting is provided. Night work is not ideal, as there are numerous aspects of the operation that should be watched and sunset undoubtedly some will be in darkness or shadow. Wind cannot always be avoided, but it is prudent to check the most reliable local forecasts before proceeding with a critical operation.

Inadequate clearance is a common impediment for the mobile crane setup to perform erection or dismantling. Interferences can occur several ways.

- The mobile crane boom or jib bottom fouls a building parapet or some other projecting corner element when lowering the boom or swinging into the building. A method for evaluating this problem is given in Chap. 5. The boom bottom could also foul a high projecting element of the tower crane such as a pendant or the tower top. Temporary construction such as scaffolding or formwork could obstruct, too.
- The head sheave or the suspended load fouls the façade of a building when swinging at minimum radius. The fouling might also be caused by a temporary structure such as a scaffold.

- A back-end element of the mobile crane such as a counterweight or live mast does not clear a tree, canopy, flagpole, or some other projection.
- The hook cannot achieve sufficient height to install the component. This vertical distance from the top of the lifted load to the hook is called *drift*.
- A tower-crane jib is an unwieldy pick that can get in a jam in a number of ways: hanging up on the tower top or a nearby structure, clashing with the mobile crane, or exceeding the lift capacity of the mobile crane. A planner must assure that the jib has a clear spot on the ground that is not beyond the ability of the mobile crane to reach it; if the jib heel is to be boomed out past the mobile crane there must be sufficient lift capacity for the maneuver. Jib erection or dismantling needs to be studied step-by-step through the full lifting procedure.

The amount of clearance taken as acceptable for any specific situation is a matter for judgment but should be influenced by the experience of the erection crew, the height at which the work is taking place, the size and weight of the load, and the means to be used to control the load such as tag lines. Clearance of 4 ft (1.2 m) is sufficient under most circumstances; with less clearance a spotter should be positioned who is capable of immediately signaling the crane operator.

When evaluating hook height and drift requirements, allowance must be made for slings, for the trip settings of the anti-two-block device and for wiggle space to allow the piece to be maneuvered into place (Figure 6.42). Nearly every neophyte planner makes the rookie error of underestimating the need for adequate drift. A rule of thumb to keep peace with the riggers is to add 10 ft (3 m) to whatever looks adequate on paper and preferably more.

Electrical power lines present a special clearance issue covered in Chap. 8.

Erection

The erection supervisor should not expect to see the site for the first time on the day the work is to begin. At the very least, the site should be visited a month or so beforehand for planning, a week before for coordination, and a day before for final details. Close coordination with the tower-crane technician should be maintained through this period to verify that the many important details are attended to.

While it is on the ground between assignments, a tower crane should be thoroughly inspected and maintenance performed. All tower cranes require some degree of care between assignments. At worst, a machine that has not been properly maintained could be hazardous; a lesser but nonetheless troublesome risk is that it might break down and require a difficult airborne repair. No crane is immune from breakdown

Figure 6.42 Erecting a tower-crane boom can entail tight clearances and drift even in open areas. (*Morrow Equipment Company.*)

but the odds should be made as favorable as practical even at the expense of a delay. It is good to know where the crane is coming from as this knowledge may confer an ability to determine its condition.

The crane might leave the yard in pristine condition but be damaged during shipping, unloading, or even while on the ground; inspection on the ground at the installation site immediately prior to erection is prudent. Some critical parts will be exposed to substantial loads during operation so that nicks and dents in its members may not be acceptable. If suspected damage is found, a qualified person should determine whether a repair is needed; repairs must be done in accordance with the manufacturer's specifications and with the knowledge of the crane owner.

Sufficient space must be allocated to allow placement of the erection crane and layout of the tower-crane components. Space should be available for ground storage with consideration given to truck positioning and offloading activities as well as preassembly work.

A top-climbing crane can be erected to a low height and then jacked up as the work progresses. The optimum initial height will depend on availability and costs of the erection crane compared with the cost of climbing.

Decisions about breakdown or assembly of components to be hoisted by the erection crane should be made before the crews are sent to the jobsite and each individual lift identified and planned.

Preassembling components on the ground reduces the number of loads to be set and the work to be done in the air, but these advantages must be balanced against the cost of a larger mobile crane that may be needed. Accurate erection weights must be known in advance. If there is doubt, components should be weighed. A smaller assist crane might be utilized for ground assembly at the site so that utilization of the main erection crane is more efficient.

An erection plan should be plotted out with both pick and setting locations predetermined. Erection weights and radii should be tabulated with adequate allowance made for rigging block, and tackle.

Counterweights and ballast blocks are not always supplied by the tower-crane manufacturer although the distributor or rental agency may do so. Whether fabricated on site or not, they must be made to strict specifications and have sufficient strength to be ready for use when erection begins. For many models, the load jib cannot be erected without some of the counterweights in place.

Each crane model has a particular style of jib support pendants, and the erection procedure will depend on the style in use. The common *saddle jib* has the jib suspended from the *top tower* with fixed-length wire ropes or steel bars. The jib is lifted from the ground at 20° to 30° to the horizontal and the jib foot connection pins inserted. Using chain-falls or similar pulling devices the pendants are pulled into place one by one and the pins set. With all pendants attached and the puller released, the jib can be lowered to its operating position.

A luffing jib is more difficult to erect. After being lifted at an angle to the horizontal, the foot pins are installed. The jib is then hung on temporary pendants or the boom hoist is reeved while the jib remains suspended on the erection crane hook. The latter operation should not be attempted unless there is power to the luffing hoist drum, little wind, and sufficient light.

After the crane is erected and before it is put into service, inspection and testing are required. The manufacturer or governing authority might have specific requirements, but at minimum testing should verify that load *limiting devices* and *limit switches* are correctly set. An inspection minimally should verify that the mast is plumb, components are correctly assembled and undamaged, counterweights are correct, and connections have all been installed in accordance with the manufacturer's requirements.

Erection Checklist

A checklist for erection of a tower crane is provided here as a general outline for near-term planning and executing erection. With the understanding that planning described in Sec. 6.2 has already occurred, this list describes activities that should begin in the weeks and days preceding the actual work execution:

1. Confirm that the tower crane and all its accessories will be available to be brought to the site on the planned date. Ideally

the final marshalling will be done from a local staging location where a full inventory, preparation, and inspection can be performed.

2. Verify that the tower crane has been or is in the process of being inspected and maintained with repairs made as needed. Once erected, these measures will be immensely more difficult and some will be impossible. Inspection and maintenance should not be performed perfunctorily or willy-nilly; at some point shortchanging will exact a price on performance or safe operation.
3. Determine whether *turntable* bolts have been properly maintained or installed prior to arrival at the site; if this work is to be done on-site, assure that it is to be done with the specified hardware and correct tools according to the manufacturer's instructions. The torque wrench should have an up-to-date calibration certificate.
4. Ascertain the condition of the ropes including, if possible, the portion of the rope on the first layer of the drum. Hoist rope length, grade, and construction should be suited to the work assignment. Verify that there is sufficient rope, with at least three wraps remaining, for the crane to operate at maximum height with the intended number of parts of line.
5. Inventory and inspect connection hardware.
6. Confirm that the equipment and personnel intended for erection of the crane will be available:
   a. Key people such as an expert technician and a rigging foreman need to be scheduled along with a crew.
   b. As erection is often done off-hours through long shifts, a rested crew is advisable.
   c. Inspect and inventory rigging hardware and tools.
   d. Arrange lighting for night work.
   e. Verify availability of the planned erection crane in the configuration required including appropriate load block, parts of line, and counterweights.
7. Assess the overall condition of the site for the proposed assembly plan with consideration of accessibility and potential interferences.
   a. Lay out the erection crane and lift zones on the ground to ascertain that there is no interference from surface features such as trees, vaults, manholes, or street furniture. Verify that space is available to assemble the erection crane and that ground conditions are suitable for its support. Oftentimes, the tower-crane assembly operation blocks critical access to a site, a neighboring property or even an emergency route; identify and resolve any such conflict.

   b. Verify key working radii of the erection crane in the field.
   c. Study operational clearances through the various motions of the erection crane and the tower-crane components on the hook. Unless a tower crane is being erected in an open field, there is a strong possibility that the mobile crane performing the erection or some of the pieces it erects will closely skirt buildings, power lines, or other cranes. Tail-swing and the suspended tower-crane boom are common culprits.
   d. Verify that the lifted components will not pass over occupied buildings, pedestrians, workers, or traffic. Prepare control measures to keep unauthorized people out of the work zone and to insulate the operation from the public and other trade workers. Arrangements might be needed to vacate buildings, close thoroughfares, and redirect workers.
   e. A day or so before the operation, ascertain that site conditions have not changed, and if they have, discuss the ramifications with key people.

8. Inspect the foundation and anchorages. Perform an as-built survey to verify that it is constructed according to plan. Survey as-built location, elevation, and orientation of anchor points to flag discrepancies. If the first tower section has been installed, verify its orientation and plumbness.

9. Prepare for electrification.
   a. Check the readiness of the electric supply, cutoff box, and the power cable with its suspension.
   b. Verify that arrangements have been made for grounding the mast.
   c. Confirm that suitable precautions are taken to guard against electrical shocks or electrocution such as lock-out procedures while working on the electrical equipment. Workers exposed to electric hazards should have suitable protective gear and clothing.
   d. In some jurisdictions, electric inspections and permits are required.

10. Review and refine the erection plan, resolving discrepancies and factoring late changes. Validate that erection weights and CGs have been obtained from a reliable source and that the weight of rigging has been considered.

11. Review the crane manual including up-to-date revisions and service bulletins.

12. For a bolt-connected mast, ascertain how bolts are to be tensioned. What is the tensioning method? If tensioning to a prescribed torque are wrenches calibrated? Verify that prescribed lubrication will be done.

13. Verify that appropriate approvals and permits have been obtained or are in process. Approvals may include shop-drawing reviews. If the crane is near an airport or a low-level flight corridor, notification and approval of aviation authorities will be required.
14. Check wind and weather forecast. Prepare for contingencies such as adverse weather.
15. Communicate the lift plan thoroughly to the work crew and other key participants.
16. Work out load test procedure in accordance with the manufacturer's requirements and the applicable code or standard. Procure test weights.

Mast connections are either pinned or bolted, with the first type in double shear and the second in tension. In a perfect world, all masts would be pinned because pins take less installation time and are easier to inspect and maintain when the crane is up. But pin connections are more difficult to design, fabricate, and maintain; between assignments they require careful inspection.

During operation and under wind loading, the mast is exposed to bending moments that cause the legs to undergo cycles of tension and compression, hundreds of cycles in a working day and hundreds of thousands over the working life of the crane. This is a classical situation of a detail exposed to potential fatigue damage, the severity of which will depend on the range of the stress variation, the number of cycles of loading and the details of the connection. The joint design, the quality of the fabrication, and the condition of the hardware all can affect proneness to fatigue failure.

As tower sections are sent out randomly to various job assignments, a typical tower section is not exposed to maximum loads on a regular basis. For that reason, tower sections suffer fatigue infrequently. When they do, there is usually an underlying cause other than a high number of load cycles.

Mast Bolts

Mast bolts are high-strength steel and require preloading. The preload must be maintained at all times otherwise the joints will not perform as designed, and indeed it may fail over the short term from overstress or over the long term from fatigue. A bolt that has lost much of its preload might be identified by tapping it with a small ball-peen hammer and listening for a dulled tone.

Sometimes connections will settle in after a crane is worked for a short time. For that reason, mast bolts should be checked and retightened a week or so after the crane has been put into service; this should be performed with the leg put in compression by swinging the counterweight over it. After one retightening bolts are unlikely to loosen again.

There are various approaches for preloading bolts that fall broadly into tension methods and torque methods. U.S. structural steel codes promulgate the former, but tower-crane manufacturers, following Europeans practices, generally promote the latter. Both are reliable when performed correctly.

A tension method is based on the premise that the relationship between bolt torque and tension is fickle, reproducible in the laboratory but unreliable in the field. U.S. procedures require calibration of torque wrenches on the job to a tension-measuring device that will show the torque needed to secure the tension required under the ambient conditions. Frequent recalibration is required. There is an alternate, highly reliable, U.S. method called *turn-of-the-nut* that requires no calibration equipment whatsoever. The nut is turned from one-half to three-quarters turn after a "snug tight" condition has been reached. The turning range depends on bolt size and the slope of the elements joined, but there is a tolerance of ± 30°. *Snug tight* is defined as the point where an impact wrench starts impacting or as the tightest possible setting made by hand with a spud wrench.

A torque method requires control of conditions in the field so that the fidelity of the torque-tension relationship is maintained. The type of thread, the finish of the bolts and the condition of lubrication is specified. It must be emphasized that the goal is to achieve an adequate bolt preload; if the calibration of the wrench or the field conditions are incorrect, the results will be unreliable. Use of a hydraulic torque wrench will increase reliability.

Most European bolts and nuts are made, rated, and marked to International Organization for Standardization (ISO) specifications or to national standards such as the German DIN series, which are in substantial conformity with ISO. The bolts are graded for strength by numbers, such as 5, 6, 8, or 10, which represent 1/10 of the minimum tensile strength in kilograms force per square millimeter (1 kip/in^2 = 0.70 kgf/mm^2);[9] a second number following indicates the ratio of the minimum yield point to the minimum tensile strength. A decimal point may be inserted between the numbers, producing designations that take the form 5.6, 8.8, 10.9, and so forth (Figure 6.43). Nuts are similarly graded but by tensile-strength number only. The tensile-strength number of properly matched nuts and bolts must agree.

For comparison, ASTM A325 (SAE grade 5) bolts are about equivalent to ISO 8.8 while ASTM A490 (SAE grade 8) are about the same as ISO 10.9.

High-strength bolts are installed by applying torque to secure a high level of pretension. U.S. practice is to preload to 70% or more of

[9]The kilogram force is not an SI unit, but this standard was written before the SI was widely adopted. The kilogram force is called a *kilopond* in some European literature.

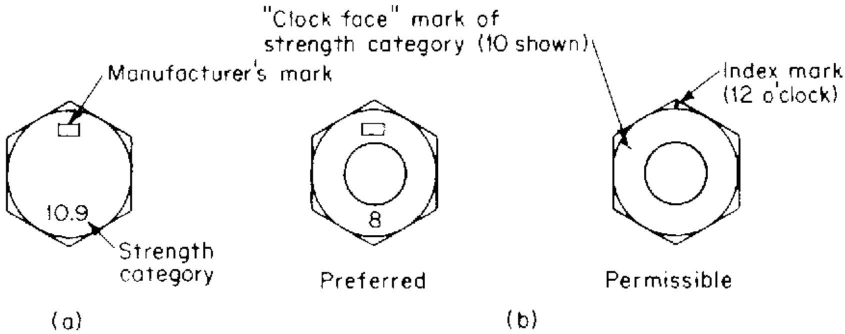


FIGURE 6.43 International Organization for Standardization (ISO) strength markings common to European-made nuts and bolts. (a) Bolt-head marking. (b) Nut marking.

bolt tensile strength with the general understanding that excess preload will do no harm. As Black and Adams have put it,[10]

> It can be reasoned that, if a well-designed bolt does not fail in hard tightening up, it will not fail in service. The reasoning is that, when the bolt is being tightened, the stress will be due to the tensile-load stress combined with the torsional stress due to the tightening torque. The latter stress disappears on removal of the tightening torque; hence the remaining stress is less than the maximum stress during tightening.

Several series of tests have proved that excessive preload does not diminish the ability of the bolt to perform and may even assure better performance. Therefore, when high-strength bolts of any manufacture, domestic or foreign, are installed, it is safer to err in the direction of overtensioning rather than to chance undertensioning.

The joining surfaces of the mast legs are constructed of large, thick steel blocks drilled to receive the bolts. The bolts are inserted and torqued to produce high bolt tensile stress that can extend beyond the bolt yield stress. This will compress the steel leg blocks.

If the bolt stress area is A and the block area is n times A, with bolt prestress load P the blocks will undergo compressive strain (Figure 6.44)

$$\Delta t_c = \frac{Pt}{nAE}$$

That is, the blocks will be under compressive stress while the bolt is in tension. E is the modulus of elasticity and t is the block thickness.

[10]P. H. Black and O. E. Adams, Jr., "Machine Design," 3d ed., McGraw-Hill, New York, 1968, p. 192; used by permission.

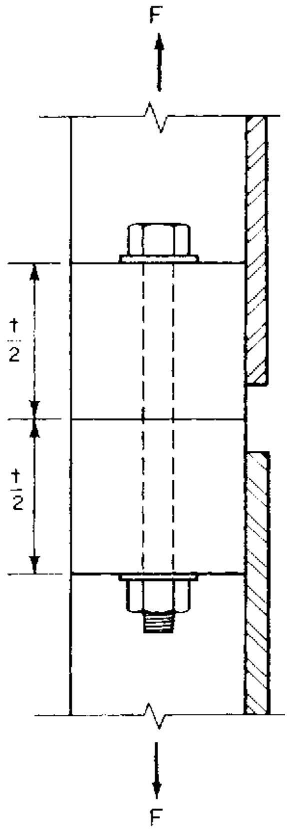


Figure 6.44 Schematic of mast-leg prestressed joint.

Should the mast leg become exposed to tensile load F, block compression will be relieved to that extent, and for $P > F$ the compressive strain will be reduced by

$$\Delta t_t = \frac{Ft}{nAE}$$

but Δt_t is a tensile strain that will be imposed on the bolt and will add to bolt tensile stress. Calling the additional bolt tension F_b, we have

$$\Delta t_t = \frac{F_b t}{AE} = \frac{Ft}{nAE} \qquad F_b = \frac{F}{n}$$

The additional stress on the bolt resulting from leg loading is therefore a function of the ratio of bolt-to-block area. This is true even if bolt tensile stress is above the proportional limit, but in that case the bolt stress will increase less than the block strain would seem to imply.

Large block areas can be effective only if thickness increases as well, so that local strain effects in the vicinity of the bolt head and nut, which act on a smaller effective area, become but a small part of total strain. Thick blocks are also needed so that they will not bend and cause prying action at the bolt head and the additional tensile stress in the bolt this action implies.

If $F \approx P/2$ and $n = 10$, the increase in bolt stress under tensile load will be only 5% if the stress remains below yield and less if above yield. But leg loads will vary from tension to compression. With the introduction of compressive force to the already compressed leg blocks, the blocks will undergo further compressive strain. This strain will be reflected in the bolt as a reduction in tensile load with the bolt now experiencing stress fluctuations that include components of both leg tension and leg compression.

Equation (6.9) was used to describe the design leg forces at the mast-footing interface. If M_o is redefined as the moment at any particular point along the mast height, Eq. (6.9) will give mast joint loads as well. This equation shows us that $Q/4$ is a compressive increment common to each mast leg.

If mast sections are preassembled on the ground with their bolts fully torqued, after final assembly each bolt prestress will be reduced by $Q/4$ because of the compressive strain that will be introduced into the leg block by crane deadweight. The same result will occur if the bolts are fully torqued as the mast sections are erected. A reduction in effective bolt prestress will introduce greater variation in tensile stress in the completed crane, and a heightened susceptibility to fatigue failure will ensue. The approximate bolt stress range will be $2M_o/\left(nAs\sqrt{2}\right)$, including moment effects.

If the crane was fully assembled and the bolts torqued with the crane *balanced*, the stress range in the bolts would be reduced by $Q/4nA$ from the range induced by preassembly. A further reduction can be achieved by applying torque to each joint with the leg in compression. By aligning the counterweights over a mast leg and then torquing the bolts in that leg, the bolt stress range will be reduced to approximately $\left[M_o/\left(s\sqrt{2}\right)-Q/4\right]/nA$, the smallest range of the bolt prestressing conditions.

For this reason, assembly instructions often specify that mast bolts should receive final torque application only after assembly has been completed and with the counterweights above the mast leg on which the bolts are being torqued. This procedure should be followed even if the instructions fail to mention it, as it helps reduce susceptibility to bolt fatigue failure during crane operation.

Dismantling

In the best of circumstances, dismantling a tower crane is exactly the reverse of erecting it. But in most actual cases dismantling is more difficult. Conditions change during the time of the crane's use—pins and bolts corrode, the crane might be raised to a higher elevation, and the new building is now around or alongside it. In spite of these changes, it is helpful to review notes from the erection of the tower crane to verify lift points, rigging, and weights.

A top-climbing crane is usually jacked down before removal. Once below the roof line it will no longer be capable of free swinging because the building will be in the way. During planning, the details of this operation should be carefully considered as it is possible that the crane can be jammed into an awkward or even an unmanageable position. At best an unexpectedly tight spot will slow down the work, but unfortunately such circumstances sometimes compel workers to take excessive risks. Some considerations of the jack-down operation are

- Sufficient clearance between the crane superstructure and the building is necessary; a minimum clearance of 1 ft (0.3 m) or so is recommended to allow for mast sway.
- The climbing frame and its platforms must clear the face of the building as it jacks down.
- A key element of dismantling efficiency is the ability of the crane to lower its tower sections to trucks on the ground; if the crane must swing to reach the trucks, then it must be positioned far enough from the building face to allow that movement.
- Orientation of the climbing frame determined which direction the jib faces while climbing.
- The lowest possible jack-down elevation may be governed by the climbing frame bottoming out against the foundation, a slab opening, or some other interfering element.

During erection, the boom and counterjib can be swung to an advantageous position; in dismantling, the hook must reach the jib and counterweights where they are.

Riggers performing the dismantling have an obligation to try to ship the crane in the condition it was in before dismantling. Connection hardware should be preserved and pieces loaded out onto trucks properly blocked and secured. Known damage and items of concern should be reported to the next custodians of the equipment.

The removal of a pendant suspended jib or counterjib is more difficult than its assembly for several reasons.

- The CG of the pick is hardly ever accurately known. In the best of circumstances the pick points and rigging arrangement will have been recorded during erection and repeated for dismantling.
- On a luffing jib the relationship of the CG to the rigging changes with the boom angle. Unless the jib is dismantled at the same angle that it was erected, an identical rigging setup will result in the boom being picked out-of-balance.
- Heel pins are under load unless the suspended load is picked at the precise balance point and the hook is precisely centered over the pick. Connections that are not floating free must be released with well-considered restraint to avoid sudden uncontrolled movement.
- When an outer segment of a *saddle jib* is removed the jib strut can snap back if not properly restrained.

Before the rigging to be used for removing the jib or *counterjib* is selected, the steps that need to be taken to release the pendants should be reviewed. As shown by the following example, forces in the sling legs will change from their values in the level position; the entire weight could be carried by one side.

Example 6.7

Determine the loads carried by a pair of slings attached to remove a jib (Figure 6.45*a*). The angle between the legs is 60°, the crane hook is positioned directly above the CG and the load weight P = 22 kips (29.8 kN). The distance from the foot pin to the jib CG is d = 50 ft (15 m) and the offset of the slings is e = 5 ft (1.5 m). Assume the piece is lifted to make an angle of 20° with the horizontal in order to unload the pendants.

Solution

For equilibrium of the horizontal components of the sling leg forces (Figure 6.45*b*),

$$F_1 \cos 80° - F_2 \cos 40° = 0$$

and for the vertical forces including foot reaction R.

$$F_1 \sin 80° + F_2 \sin 40° - P + R = 0$$

Taking moments about the foot pin.

$$F_1(d-e) \sin 80° + F_2(d+e) \sin 40° - Pd = 0$$

Solving simultaneously gives $F_1 = 21$ kips (93.5 kN) and $F_2 = 5$ kips (21 kN).

In the initial position, before tilting the jib or counterjib, each sling leg carried a load equal to $0.58P$, so that by tilting the piece 20° the maximum sling leg force has increased by some 66%. Note that tilting the piece 30° would place the entire load weight on one sling leg while the other leg would be unloaded.

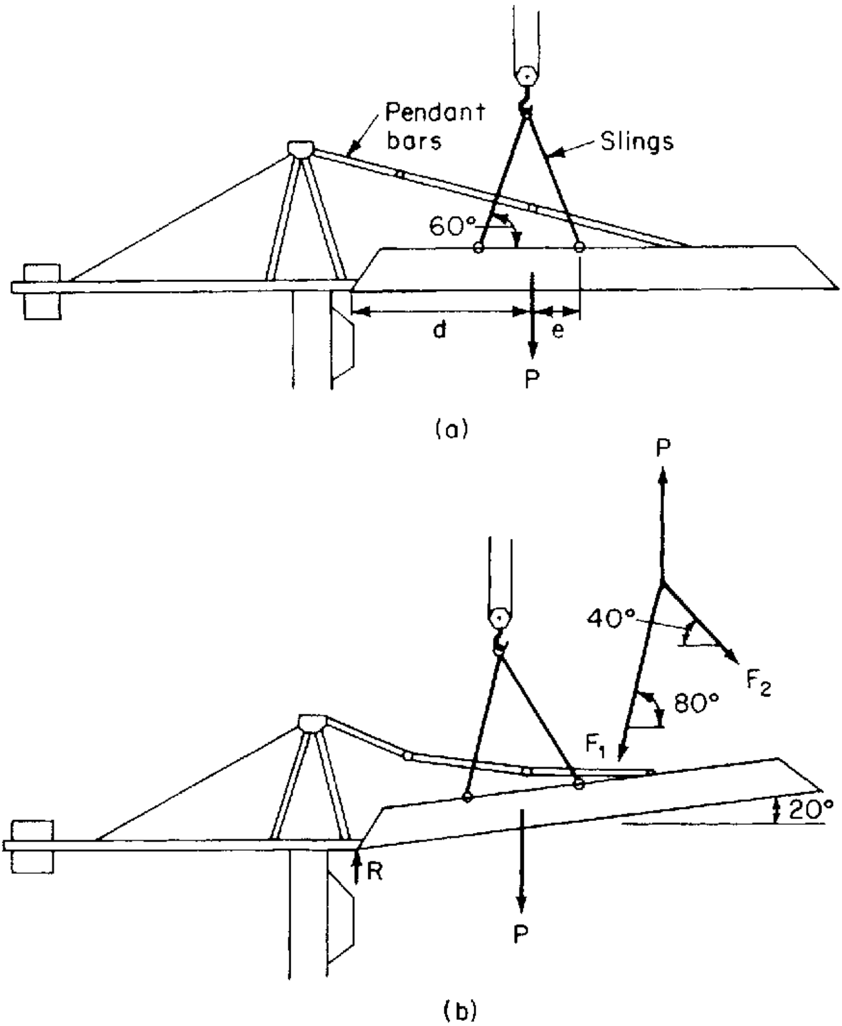


Figure 6.45 Sling geometry during tilting of the jib to slacken the load on the pendants.

6.8 Tower-Crane Operation

For the operator who has migrated from the cab of a mobile crane, a bird's eye vantage point of a tower crane is a new experience. Some enjoy the solitude and the commanding height. Others fear the altitude and dread feeling the mast sway underfoot. Arguably the tower crane is a safer place for the migrating operator than his former perch a few steps above the ground; some of the common hazards of mobile cranes—boom-over-backward, overturning, and ground failure—are rarities. But with the prestige of sitting in that seat high above all the other tradesmen comes the responsibility of scrutinizing and maintaining a crane that cannot be brought back to the yard for others to tend.

Operating a friction mobile machine bears little resemblance to the experience on a tower crane. The friction crane draws power on demand from an internal combustion engine that directly responds to the foot on the throttle. In contrast, the electric power train and control systems of a tower crane modulate motions and limit loads; the operator has only indirect command of the machinery. Those few tower cranes that are diesel-hydraulic behave similarly; the engine acts only as the source of the power—measured power that flows to the machinery through hydraulic hoses.

Tower-crane control systems are made up of various sensors and processors that prevent overloading, restrain acceleration, and set travel limits. Line pull and load moment are monitored; an indicator warns of a pending overload and the control system intervenes to prevent it. When malfunctions occur, the control system returns the crane to a stable or static condition. All tower cranes are furnished with *dead-man controls* that return to neutral and stop the motion when the hand is taken off the lever or button. In addition, circuitry is designed to apply brakes to hoist, trolley, and boom motions when fluid pressure or power is lost. Though it is not a substitute for responsible behavior, the tower crane and its operator are protected by the control system if it is functioning properly.

Motors, brakes, and gears are sized to limit acceleration; hence the forces acting on the crane during operation are also limited. While the structure is protected from excessive stress, the operator is also restricted from having complete freedom of action. He cannot cause the load to freefall or quickly stop during swinging. He might be prevented, on occasion, from swinging against a moderate wind, especially when the load has a large surface area and is at a long radius.

The benevolent guidance of the control system should not be permitted to lull anyone into feeling that the operator can do no harm. The intelligent operator treats the various limits as though they are safety nets; they are there when needed but not intended for jumping into. After all, devices do malfunction; the operator who is careless enough to depend on the vigilance of these devices may also be too cavalier to bother checking them. An operator would not be doing anyone a favor by tampering with the equipment to boost performance or gain an extra foot of reach; any small benefit would hardly be worth the risk of a serious mishap. Some limiting devices can be defeated even without malfunction. Full throttle thrust backed by inertia or wind may overrun a motion limiter. Shock load, a sudden application or release of tension on the load line, cannot always be stopped by a load limit device.

A tower crane is not designed for shock loading. Its effects are insidious because any one episode may impart no visible damage but cumulatively these episodes have a direct link to fatigue failure. A mere few months of abusive service may be enough to initiate failure. Construction tower cranes are generally not intended for duty cycle work. Some examples of work that must not be performed with ordinary tower cranes are pulling formwork free from freshly poured concrete surfaces, dropping a demolition ball, and pulling piles or sheet piling from the ground.

The operator has a different set of responsibilities on a tower crane as compared to on a mobile crane. His role in assembling and setup are secondary and passive; others are in charge. But from the point when the operator takes the seat it is largely his job, assisted by an oiler or relief operator, to inspect from top to bottom. Though an occasional outside inspector or mechanic may come on board, the everyday chores are his.

Specific requirements for inspection can be found in the operating manual and general guidance in standards such as ASME B30.3. Generally, the structural, mechanical, and control systems should be under unrelenting systematic scrutiny. A daily log of inspection and maintenance should be maintained by the operator in the cab.

Limiting devices need to be checked, reset, and recalibrated from time to time. Motion-limiting devices, such as those that prevent two-blocking and trolley overrun, should be checked daily as should the controls in the cab.

Though a full daily inspection of the load line is impracticable, the portion of the rope that is most likely to degrade is also the portion that is visible as it runs off the drum and passes the cab; it should be checked daily along with the boom hoist cable on a luffing jib. Trolley lines and static ropes can be inspected less frequently.

The structure of the crane, especially the *turntable* and mast, should be often surveyed for distortions or evidence of cracks. Connections should be inspected; bolts in the *turntable* and mast should be tapped with a hammer to ascertain that preload is not lost. Base anchorages tend to collect water and debris; they should be kept clean and dry for inspection.

The tower crane is a remote place difficult to access, but the operator should not be shy about sharing concerns about problems with others. An occasional visit by a mechanic or a special inspector will do more than relieve loneliness.

This page intentionally left blank

CHAPTER 7
Derrick Installations

The present authors are old enough to have heard tales of the good old days when derrick riggers were real men. No engineers or planning staff stood behind these real men; the foreman took care of what passed then for design and planning. The equipment was a few steel-pipe and wood-pole masts, or perhaps a latticed angle-iron boom, and whatever was in the gang box. The job was done by guts, manly strength, and wits. But the riggers' hearts were in their mouths from the time they arrived at the job until the piece was finally in place. Hidden behind the bravado were tales of injuries and deaths, flattened booms sprawled on pavement, and pieces dropped. Swift and silent cleanups facilitated willful forgetting that went hand-in-hand with an attitude that progress is possible only with these occasionally ghastly costs.

Engineering analysis is now part and parcel of derrick work but the importance of experience should, nonetheless, not be dismissed. In this chapter, the authors offer insights to aid the installation designer and describe working characteristics of the major derrick types. Much of the following material pertains to the practical details that need to be addressed after the initial installation decisions are settled.

Once dominant as a rigging tool and the choice means to erect high-rise steel buildings, the derrick is now often a lifting means of last resort. With mobile cranes and tower cranes ever more capable of reaching higher, lifting more and overcoming logistical woes, the derrick is not an endangered species of lifting gear but its habitat is shrinking, pushed to ever more remote perches. A derrick is a crude implement compared to modern cranes; the skills needed to assemble and operate it are ever harder to come by as old-timers retire. But it can still be the preferred lifting solution where height or lack of space precludes other solutions. For practitioners who take pride in exercising artful rigging skills, derricks are the purest media on which to work.

7.1 Introduction

Derricks come in several configuration types, as well as in varying dimensions and capacities. They can be placed on foundations on the ground, on raised supports, or anchored to the framework of a building, during construction or long after completion, on a static mounting or movable either horizontally or vertically.

Faced with a derrick job, an installation design engineer starts out with many questions, among which should be

- What type of derrick is best suited to the task?
- How is the derrick and winch to be brought to the working location?
- How is the derrick to be assembled and disassembled?
- What type of loads will the derrick be handling and how will they be rigged?
- Where must the derrick reach to pick loads?
- Where must the derrick reach to set loads?
- Is there sufficient *drift* and clearance between the load and the boom?
- What is the best location for the winch?
- Does the winch power source comport with its proposed location?
- How will the forces imposed by the derrick and winch be supported?

With these questions answered, the designer can rest assured that the hardest part of the engineering job has been done.

There are two general types of derrick installation, permanent and temporary. Permanently installed derricks are not nearly as common today as they used to be. They have largely been replaced by mobile equipment that is far more versatile and easier to operate. Temporary installations can be used for construction or maintenance. The latter includes permanent installation of occasional-use devices that would be used, for example, to replace a transformer or generator in the event of failure. A derrick of this type might be kept in storage with only the mountings permanently fitted to the building.

7.2 Chicago Boom Derricks

Chapter 1 introduced the calculations for the force system of a simple derrick. The *Chicago boom* was used in this illustration because it is the most basic derrick configuration except for the primitive form of the *gin pole*. Unfortunately, its simplicity is superficial, concealing underlying engineering, operating, and rigging complications.

Setting up a lifting apparatus outside the building face engenders some fall protection problems, but the most difficult part of working with a Chicago boom is the handling of the loads after they have been hoisted and swung towards the building. Paramount to the degree of rigging difficulty is the distance that the boom foot and *topping lift* pivots, aligned together in one vertical axis, are mounted from the building face. Increasing the length of these pivots induces greater eccentric loading on the building, a vexing structural problem in itself. The pivot distance can be as little as half the boom width, but preferably it should be greater so that the boom can swing enough of an arc to set the loads close to the building face. Foreshortening the swing arc can make the rigging operations more difficult, and it can expose riggers to fall hazards. Various measures can be carried out for fall protection such as a load-shifting procedure (Figures 7.1 and 7.2) or setting a cantilevered landing platform out from the building face.

Column splice locations need to be considered when establishing boom foot and topping lift anchorage points on steel buildings. Fittings should be placed close to floor level to minimize the biaxial bending moments introduced into the building column. Though any

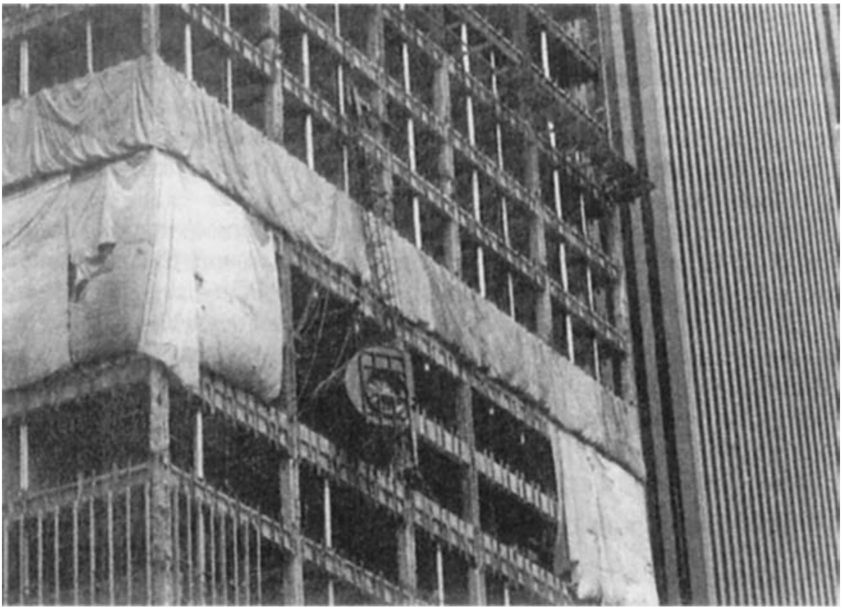


FIGURE 7.1 Loads cannot be landed until the CG is inside the structure. For heavy or bulky loads; this means that a load-shifting procedure, such as shown in Figure 7.2, must be used. (*Gendelman Rigging and Trucking, Inc.*)

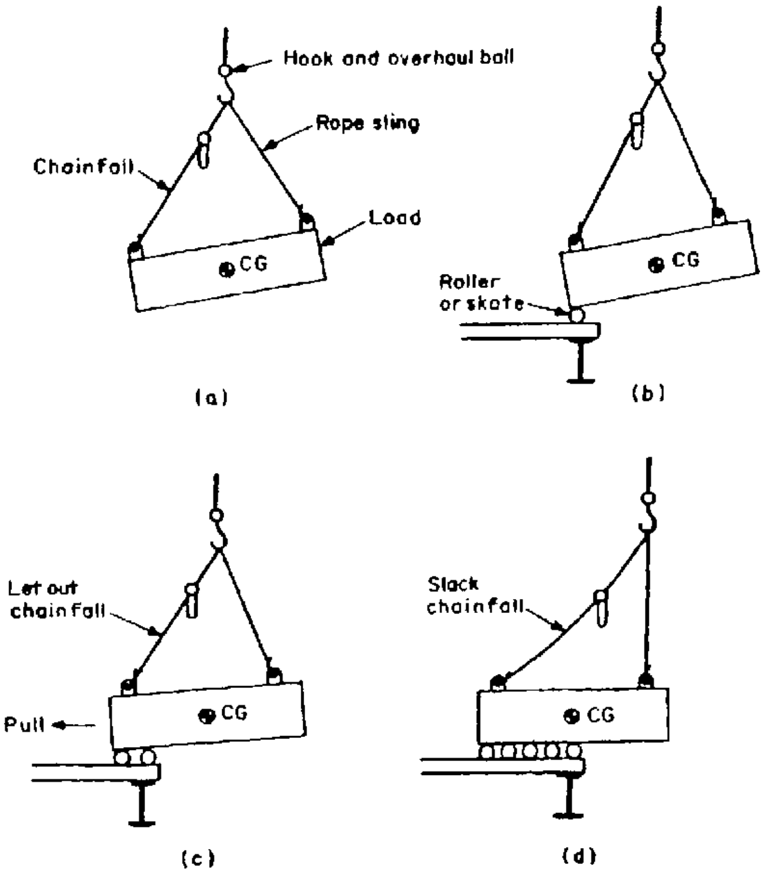


FIGURE 7.2 Procedure used for shifting loads suspended from crane or derrick hooks in order to move the CG into the building. (a) Load is in hoisting position. (b) End of load is landed. (c) Load starts in. (d) CG is inside building.

structurally adequate column can be used, corner columns raise formidable installation problems and should generally be avoided; arrangements to permit swinging through 270° can be complex as can be the fittings needed to allow the lead lines to follow the boom through the swing.

Figure 7.3 shows a boom foot fitting with the boom swung to 45° to the building face, and the boom vertical and horizontal reactions as B_v and B_h respectively. With the load P representing the dead load plus construction live load acting on the column from the floors above, column axial load below the fitting is then $P + B_v$. For strong-axis bending (Figure 7.3*b*), the bending moment is $0.7B_h d(1-d/h)$ by examination. Weak-axis bending comprises two terms, one from the

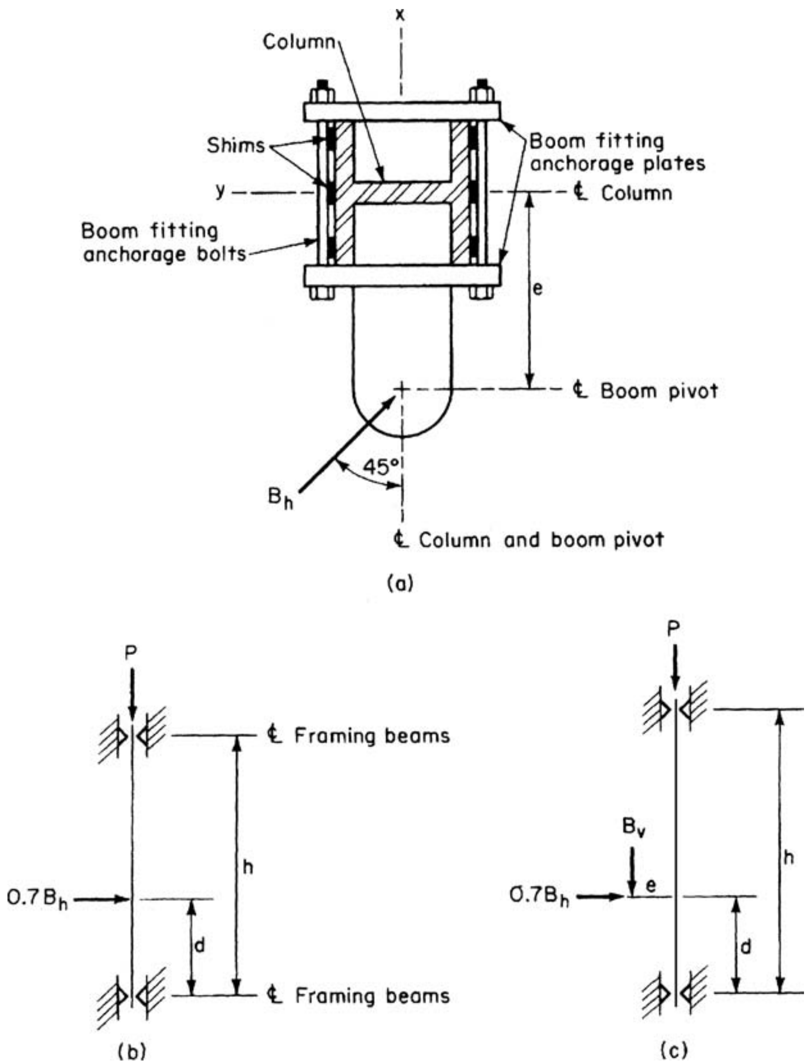


FIGURE 7.3 (a) Boom foot fitting and bending loads. (b) Column strong axis. (c) Column weak axis.

boom vertical reaction and the other from the horizontal. By statics, the reaction at the column foot is

$$R = \frac{B_v e}{h} + \frac{0.7B_h(h-d)}{h}$$

giving the maximum moment in that plane of

$$M = \frac{[B_v e + 0.7B_h(h-d)]d}{h}$$

In this example, the force $0.7B_h$ acting on eccentricity e imposes a torsional moment on the column which can be quite severe. Evaluation and possible strengthening to resist torsion is one of the key problems in designing a Chicago boom installation. The column orientation in this example is arbitrary, and so is the selection of 45° for the calculation; loads imposed by the boom when it is at another angle to the building may be more severe than for the position discussed.

The installation designer must evaluate the structure for the actual loading conditions defined by the operations or those which may be reasonably inferred from expected field procedures. Where necessary, warnings against actions that will cause excessive loadings should be included in the lift plan given to the riggers.

The Chicago booms shown in Figures 7.4 and 7.5 are clamped on the flange tips of the mounting columns, as illustrated diagrammatically in Figure 7.3.

Figure 7.4 Chicago boom clamped to the weak axis of a column and arranged for landing loads to the left. Several parts of line are used in the right swing line to develop enough force to pull the boom away after unloading. (*Gendelman Rigging and Trucking, Inc.*)

Column Torsion

If not adequately treated, column torsion induced by the Chicago boom can be nettlesome, or worse, causing fire-proofing to become loosened, damage to beams and connections, and even column buckling. The added stress can cause yielding at the flange tips and a considerable loss of column strength (Figure 7.6). For this reason, particularly on lighter weight columns, lateral ties are often installed to reduce or eliminate the torsional moment by transferring the force that produces it to adjacent members (Figure 7.7). Torsion reducers are just barely visible in both Figures 7.1 and 7.4.

Wire rope might seem to be an obvious choice for lateral torsion ties, but connection details to the structural elements are difficult and its physical properties are much less than ideal. Even when the rope ties are taken up very tightly, they will stretch considerably; the column would still be exposed to most of the torsional loading. For example, an IWRC rope will stretch some 3½ times more than a steel rod of equivalent diameter. If a 1¼-in-diameter (32-mm) mild-steel rod is needed to absorb the torsional moment, the equivalent in stiffness would be seven parts of ⅞-in-diameter (22-mm) rope. Rope, therefore, will often prove to be impractical for this purpose.

The effect of torsion in a column can be analyzed by using elastic theory. For the model in Figure 7.8, where the column ends are restrained against warping or twisting, the applied torsional moment T will be distributed to the column ends as reactions T_0 and T_h. In order to maintain consistency in angular twist, the greater part of T will be restrained by the shorter and torsionally stiffer segment of the column.

A column unrestrained at its ends will twist or warp evenly (Figure 7.8*b*), causing shear stresses in the cross section. This is known as *St. Venant torsion* and is show in Figure 7.6. If the column ends are restrained, warping becomes nonuniform as resistance to torsion induces bending moments in each of the flanges (also shown in Figure 7.6). The applied moment will then be resisted by the combination of the two effects

$$T = T_v + T_w \qquad (\boldsymbol{a})$$

where the St. Venant torsional resistance is

$$T_v = GJ\frac{d\phi}{dz} \qquad (\boldsymbol{b})$$

where G is the shearing modulus of elasticity and J is a torsional constant to be described later. The nonuniform warping resistance is

$$T_w = -Vs \qquad V = \frac{dM_f}{dz}$$

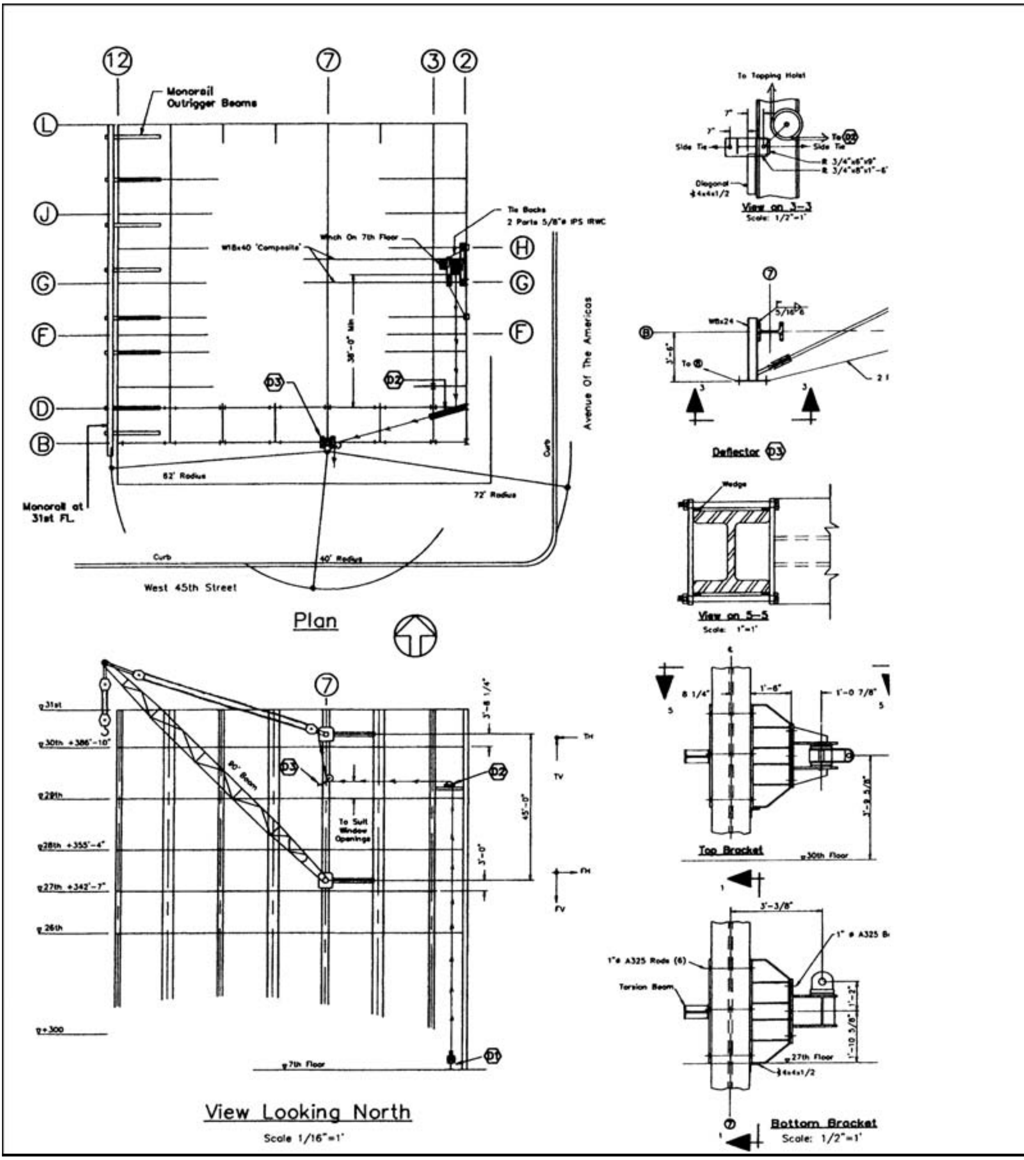
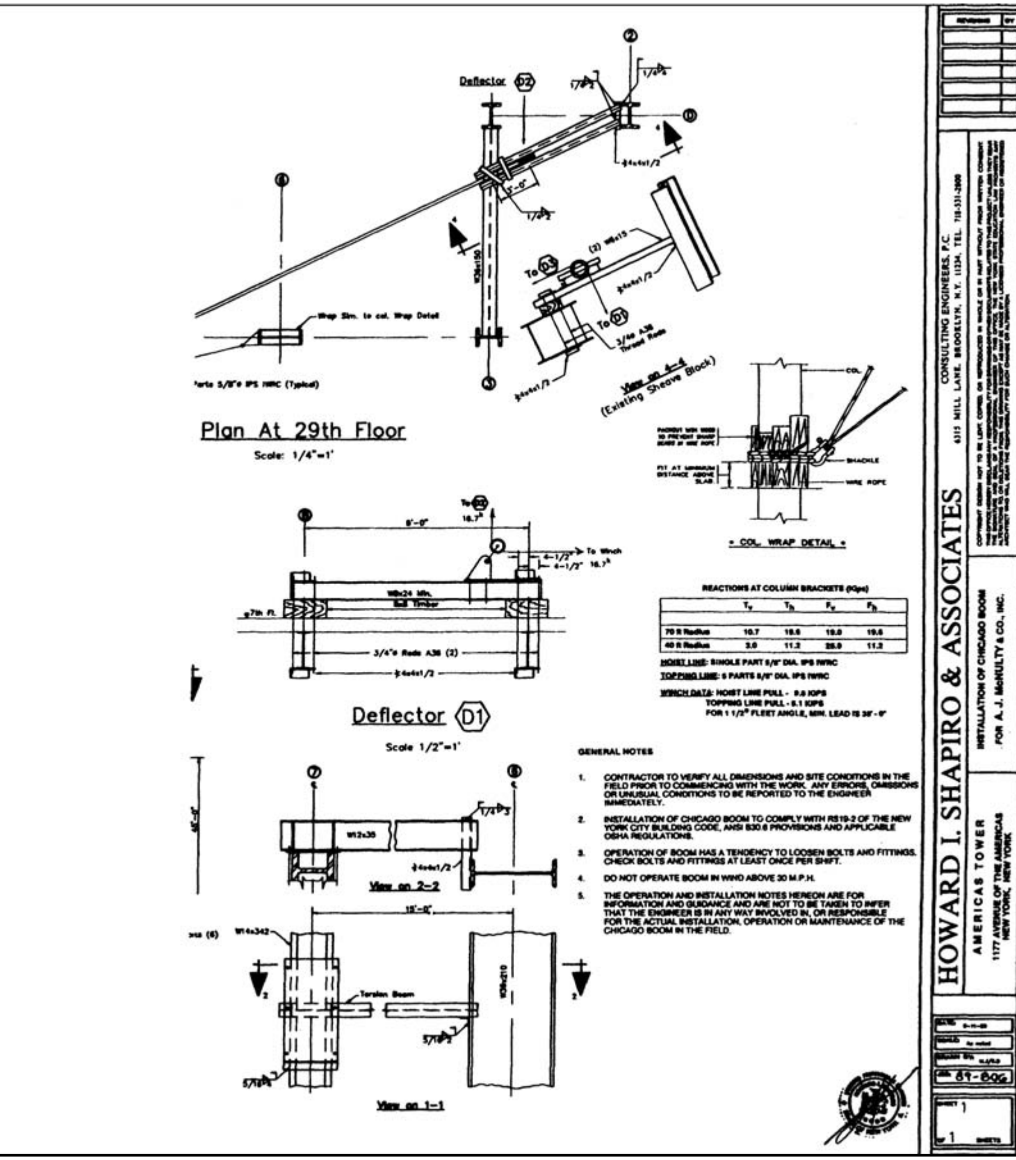


FIGURE 7.5 Installation drawing for a Chicago boom mounted on the steel frame of an office tower under construction. At both the top and the bottom brackets, beams have been used to relieve the columns of torsion loading. Although the boom foot is just above the 27th floor, the winch has been set on the 7th floor and provisions to carry the lead lines are shown. The derrick was used to erect precast facing panels and to feed panels to a monorail placing system.


Deflector D2
Plan At 29th Floor
Scale: 1/4"=1'
View on 4-4
(Existing Sheave Block)
COL. WRAP DETAIL
Deflector D1
Scale 1/2"=1'
3/4"ø Rods A36 (2)
W8x24 Min.
8x8 Timber
To Winch
View on 2-2
View on 1-1
Torsion Boom
REACTIONS AT COLUMN BRACKETS (Kips)
	T_v	T_h	F_v	F_h
70 ft Radius	10.7	19.6	19.0	19.6
40 ft Radius	3.0	11.2	25.0	11.2
HOIST LINE: SINGLE PART 5/8" DIA. IPS IWRC
TOPPING LINE: 6 PARTS 5/8" DIA. IPS IWRC
WINCH DATA: HOIST LINE PULL - 9.0 KIPS
TOPPING LINE PULL - 5.1 KIPS
FOR 1 1/2° FLEET ANGLE, MIN. LEAD IS 38' - 0"
GENERAL NOTES
1. CONTRACTOR TO VERIFY ALL DIMENSIONS AND SITE CONDITIONS IN THE FIELD PRIOR TO COMMENCING WITH THE WORK. ANY ERRORS, OMISSIONS OR UNUSUAL CONDITIONS TO BE REPORTED TO THE ENGINEER IMMEDIATELY.
2. INSTALLATION OF CHICAGO BOOM TO COMPLY WITH RS19-2 OF THE NEW YORK CITY BUILDING CODE, ANSI B30.6 PROVISIONS AND APPLICABLE OSHA REGULATIONS.
3. OPERATION OF BOOM HAS A TENDENCY TO LOOSEN BOLTS AND FITTINGS. CHECK BOLTS AND FITTINGS AT LEAST ONCE PER SHIFT.
4. DO NOT OPERATE BOOM IN WIND ABOVE 30 M.P.H.
5. THE OPERATION AND INSTALLATION NOTES HEREON ARE FOR INFORMATION AND GUIDANCE AND ARE NOT TO BE TAKEN TO INFER THAT THE ENGINEER IS IN ANY WAY INVOLVED IN, OR RESPONSIBLE FOR THE ACTUAL INSTALLATION, OPERATION OR MAINTENANCE OF THE CHICAGO BOOM IN THE FIELD.
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
AMERICAS TOWER
1177 AVENUE OF THE AMERICAS NEW YORK, NEW YORK
INSTALLATION OF CHICAGO BOOM
FOR A. J. McNULTY & CO., INC.
89-806

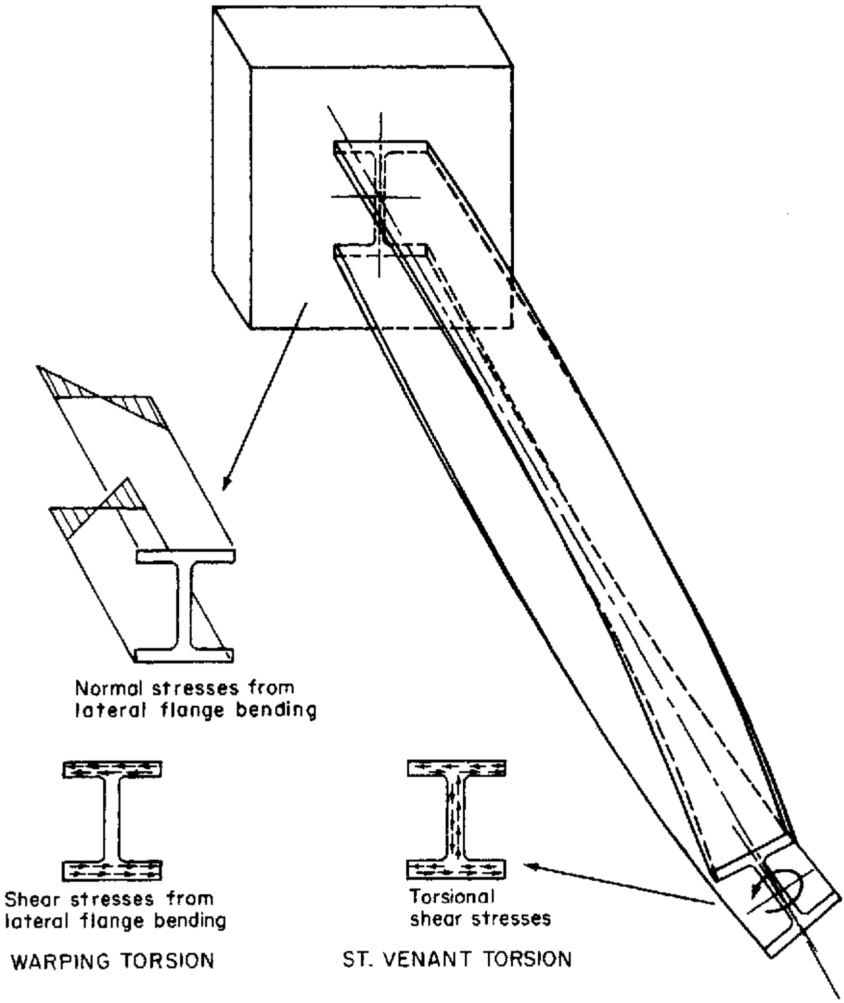


Figure 7.6 Torsional behavior of a column or beam. (*From USS Steel Design Manual ADUSS-27-3400-03, United States Steel Corp., May 1974.*)

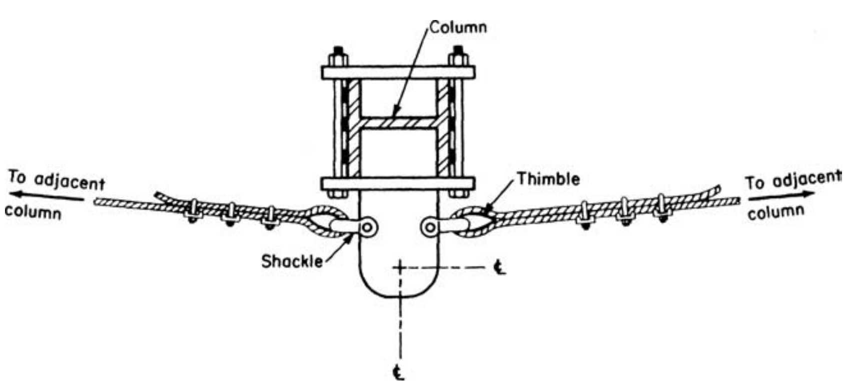


Figure 7.7 Torsion reducers on a Chicago boom column fitting.

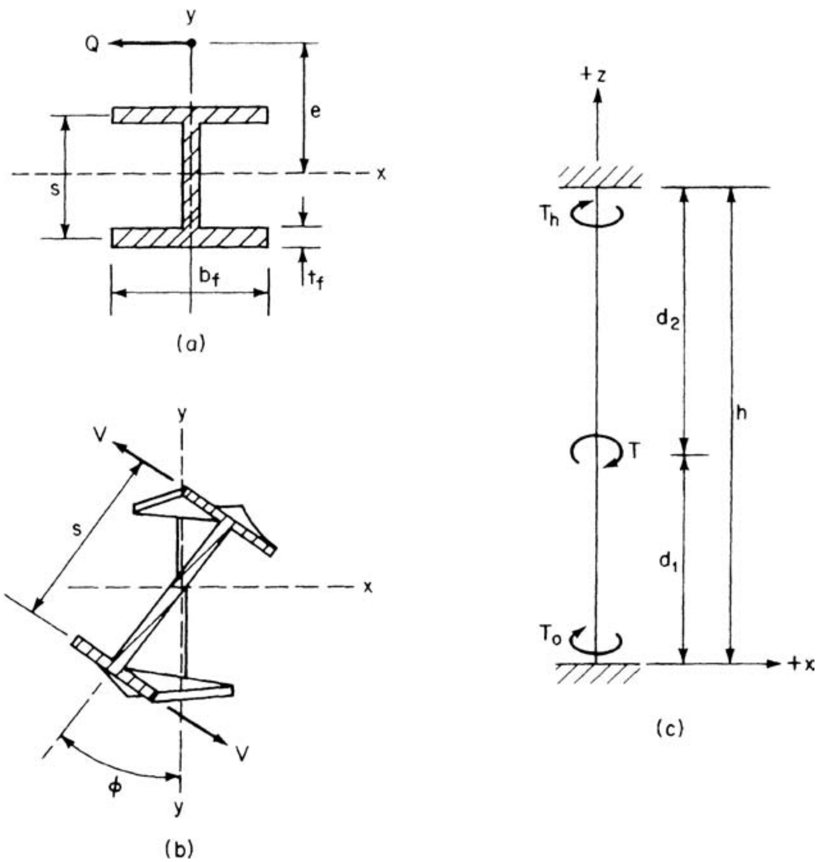


FIGURE 7.8 Column torsion model.

where the flange bending moment is

$$M_f = EI_f \frac{s}{2}\frac{d^2\phi}{dz^2}$$

giving

$$T_w = -EI_f \frac{s^2 d^3\phi}{2dz^3} \qquad (c)$$

when I_f is taken as the moment of inertia of the cross section of one flange about the y axis and E is the modulus of elasticity. Substituting (***b***) and (***c***) into (***a***) gives

$$T = GJ\phi' - EI_f \frac{s^2}{2}\phi'''$$

with the prime notation referring to differentiation with respect to z. Simplifying by inserting $c = GJ$ and $c_1 = EI_f s^2/2$ and rearranging leads to

$$\phi''' - k^2\phi' = -\frac{T}{c_1} \qquad k^2 = \frac{c}{c_1} \tag{d}$$

The general solution to Eq. (***d***) is

$$\phi = \frac{Tz}{c} + A_1 + A_2 \sinh kz + A_3 \cosh kz \tag{e}$$

For the segment $0 \le z \le d_1$, taking values of Eq. (***e***) and its derivatives at $z = 0$ gives

$$\phi(0) = A_1 + A_3 \qquad \phi'(0) = \frac{T_0}{c} + kA_2 \qquad \phi''(0) = k^2 A_3$$

with T_0 used because this segment restrains only that part of applied torsion T (Figure 7.8*c*). The equation for the first segment is then obtained by making substitutions for the constants of integration, A_1, A_2, A_3, in Eq. (***e***) in terms of the function at $z = 0$

$$\phi_1 = \frac{T_0 z}{c} + \phi(0) - \frac{\phi''(0)}{k^2} + \left[\frac{\phi'(0)}{k} - \frac{T_0}{ck}\right]\sinh kz + \frac{\phi''(0)}{k^2}\cosh kz$$

$$= \frac{T_0}{ck}(kz - \sinh kz) + \phi(0) + \phi'(0)\frac{\sinh kz}{k} + \frac{\phi''(0)}{k^2}(\cosh kz - 1)$$

Assuming full restraint at the ends of the column dictates that $\phi(0) = \phi'(0) = 0$, so that this expression becomes

$$\phi_1 = \frac{T_0}{ck}(kz - \sinh kz) + \frac{\phi''(0)}{k^2}(\cosh kz - 1) \tag{f}$$

For the second segment, $d_1 \le z \le h$, the effect of applied moment T must be added, giving

$$\phi_2 = \phi_1 + \frac{T}{ck}[k(z - d_1) - \sinh k(z - d_1)] \tag{g}$$

Introducing the boundary condition $\phi_2(h) = 0$ at the upper end and rearranging leads to

$$\phi''(0) = [T_0(\sinh kh - kh) + T(\sinh kd_2 - kd_2)]\frac{k/c}{\cosh kh - 1} \tag{h}$$

The derivative of Eq. (***f***) is then inserted into the derivative of Eq. (***g***), and the boundary condition $\phi_2'(h) = 0$ is introduced. After rearranging, this gives

$$\phi''(0) = [T_0(\cosh kh - 1) + T(\cosh kd_2 - 1)]\frac{k/c}{\sinh kh} \tag{i}$$

Equating the two expressions for $\phi''(0)$ and solving for T_0, we have

$$T_0 = -uT$$

where
$$u = \frac{\cosh kd_2 - 1 + v(kd_2 - \sinh kd_2)}{\cosh kh - 1 + v(kh - \sinh kh)} \tag{7.1}$$

and
$$v = \frac{\sinh kh}{\cosh kh - 1}$$

When substituted back into either Eq. (***h***) or (***i***), this yields

$$\phi''(0) = \frac{Twk}{c} \tag{7.2}$$

with
$$w = \frac{\cosh kd_2 - 1}{\sinh kh} - \frac{u}{v}$$

The entire torsional action has now been defined for a column with fixed ends, and the following equations are readily drawn from Eqs. (***f***) and (***g***):
For $0 \le z \le d_1$

$$\phi_1 = \frac{T[w(\cosh kz - 1) - u(kz - \sinh kz)]}{ck}$$

$$\phi_1' = \frac{T[w \sinh kz - u(1 - \cosh kz)]}{c}$$

$$\phi_1'' = \frac{Tk(w \cosh kz + u \sinh kz)}{c}$$

$$\phi_1''' = \frac{Tk^2(w \sinh kz + u \cosh kz)}{c}$$

For $d_1 \le z \le h$:

$$\phi_2 = \frac{T[w(\cosh kz - 1) - u(kz - \sinh kz) + k(z - d_1) - \sinh k(z - d_1)]}{ck}$$

$$\phi_2' = \frac{T[w \sinh kz - u(1 - \cosh kz) + 1 - \cosh k(z - d_1)]}{c} \tag{7.3}$$

$$\phi_2'' = \frac{Tk[w \cosh kz + u \sinh kz - \sinh k(z - d_1)]}{c}$$

$$\phi_2''' = \frac{Tk^2[w \sinh kz + u \cosh kz - \cosh k(z - d_1)]}{c}$$

Equations (7.3) are used to give values for each of the moments at any point within the column by referring back to the equations given earlier.

$$T_v = GJ\phi' \qquad M_f = \frac{c_1\phi''}{s} \qquad T_w = -c_1\phi'''$$

and, of course, ϕ is the angle of twist at any point. The stresses are then given by

$$\text{St. Venant shear stress} = f_v = \frac{T_v \delta t_f}{J} = G\phi' \delta t_f \tag{7.4}$$

$$\text{Flange bending stress} = f_t = \frac{M_f b_f}{2I_f} = \frac{Esb_f\phi''}{4} \tag{7.5}$$

$$\text{Nonuniform warping shear} = f_w = \frac{Vq}{I_f t_f} = \frac{\text{Esb}_f^2\phi'''}{16} \tag{7.6}$$

where δ is a stress coefficient given in either Figure 7.9 or 7.10 and the shear modulus G can be taken as 11,000 kips/in^2 (75.8 GN/m^2).

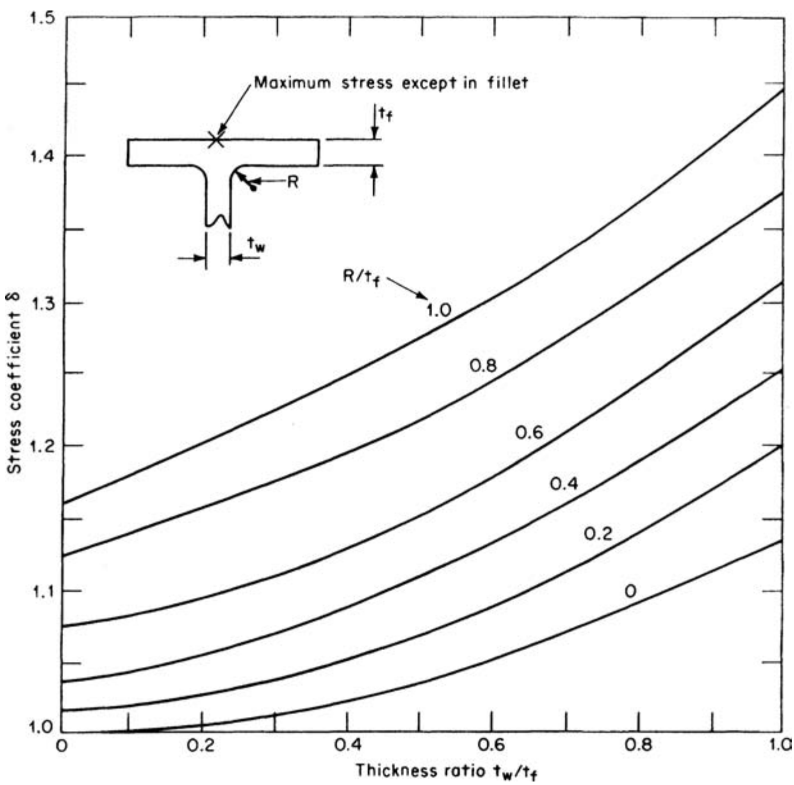


Figure 7.9 Stress coefficients for T or W shapes with parallel flanges. (*From USS Steel Design Manual ADUSS-27-3400-03, United States Steel Corp., May 1974.*)

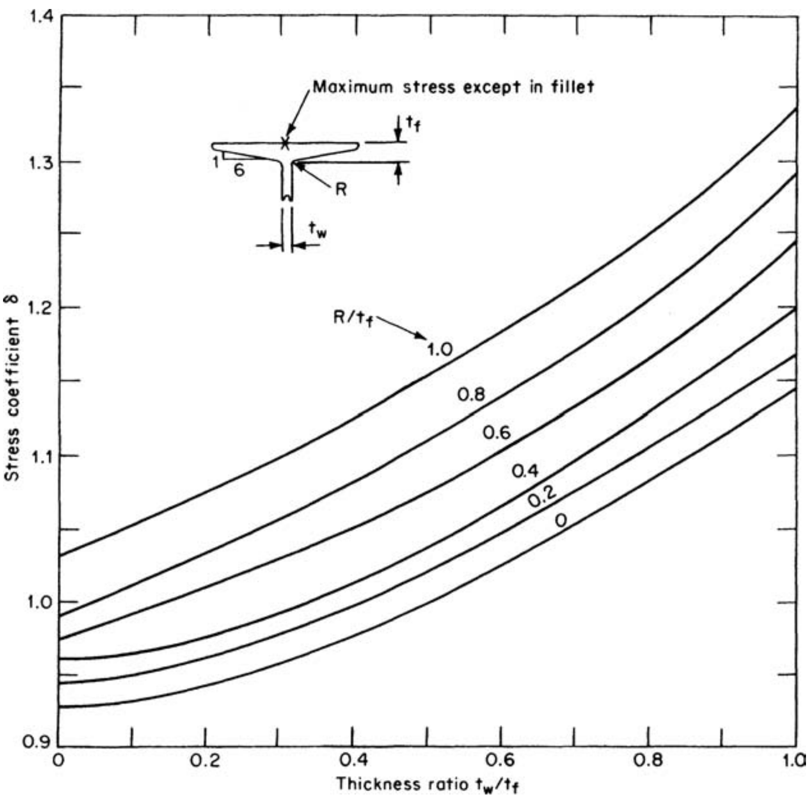


FIGURE 7.10 Stress coefficients for T and S shapes with slopping inner flanges. (*USS Steel Design Manual ADUSS-27-3400-03, United States Steel Corp., May 1974.*)

The notation q refers to the static moment of half of one flange area about the flange neutral axis, the torsional constant and J, is approximately equal to the sum of the $bt^3/3$ values for each of the rectangular elements in an open section such as an H, I, channel, or angle member (when $b/t \geq 6$); more accurate values for this parameter are given in the AISC manual together with the warping constant $c_w = c_1/E$.[1]

When the column ends are unrestrained, there will be no nonuniform warping and therefore no flange bending. The unrestrained case is idealized; actual column ends will always be at least partially restrained. In addition to evaluating the ordinary beam-column stress interaction, flange tip stresses should be summed at the column segments that carry boom foot and topping lift fittings.

[1]"Manual of Steel Construction," 9th ed., American Institute of Steel Construction, Inc., Chicago, 1989.

At a restrained end, the St. Venant shear stress f_v is zero while the nonuniform warping shear stress f_w is a maximum. As distance from the support increases, f_w decreases and f_v rises.

If rod or rope torsion reducers are used, they will carry only a portion of the torsional load, the column taking the balance. The distribution will be such as to create consistent strains. Let Q_c be that portion of the torsion-inducing load Q that is carried by the column. Then Q_r will be the load induced in the reducer located at eccentricity r from the column center. The change in length of the loaded reducer is then

$$\Delta L = \frac{Q_r L_r}{A_r E_r}$$

where L_r = length of reducer
A_r = **metallic area** of rope cross section
E_r = modulus of elasticity

To make the strains consistent,

$$\Delta L = r\phi$$

but ϕ is the angle of rotation induced by $Q_c e$. For clarity, let $\phi_1(d_1)/T = m$; then

$$\Delta L = rQ_c em$$

The torsional moments will be in equilibrium if

$$Qe = Q_c e + Q_r r \qquad Q_c = Q - \frac{Q_r r}{e}$$

Substituting this expression for Q_c into the ΔL relationship gives

$$\Delta L = \left(Q - \frac{Q_r r}{e}\right) emr = \frac{Q_r L_r}{A_r E_r}$$

which when rearranged becomes

$$Q_r = \frac{Q}{L_r/emrA_r E_r + r/e} \tag{7.7}$$

The absolute value of parameter m is to be used in this equation.

With Q_r known, the rope or rod reducer can be checked for stress and Q_c can be found so that the column stresses can be evaluated. If column stresses are still excessive, A_r can be increased and the process repeated.

Example 7.1

1. A W14 × 95[2] column section loaded in torsion has the following parameters (Figure 7.8):

$$b_f = 14.545 \text{ in (369 mm)} \qquad t_f = 0.748 \text{ in (19.0 mm)}$$

$$s = 13.37 \text{ in (340 mm)} \qquad e = 22.06 \text{ in (560 mm)}$$

$$Q = 20 \text{ kips (89.0 kN)}$$

$$d_1 = 24 \text{ in (610 mm)} \qquad h = 150 \text{ in (3.81 m)}$$

$$J = 4.74 \text{ in}^4 (197.3 \text{ cm}^4) \qquad c_w = 17{,}200 \text{ in}^6 (4.62 \text{ dm}^6)$$

$$E = 30{,}000 \text{ kips/in}^2 (206.8 \text{ GN/m}^2) \qquad G = 11{,}000 \text{ kips/in}^2 (75.8 \text{ GN/m}^2)$$

Assuming fully restrained ends, calculate the maximum values for St. Venant and warping shear stresses and the flange-bending stress induced by non-uniform warping.

Solution

First, a series of intermediate values is needed.

$$T = Qe = 20(22.06) = 441.2 \text{ kip}\cdot\text{in (49.85 kN}\cdot\text{m)}$$

$$c_1 = c_w E = 17{,}200(30{,}000) = 5.16\times10^8 \text{kip}\cdot\text{in}^4 (955.4 \text{ kN}\cdot\text{m}^4)$$

$$c = GJ = 11{,}000(4.74) = 52{,}140 \text{ kip}\cdot\text{in}^2 (149.6 \text{ kN}\cdot\text{m}^2)$$

$$k = \sqrt{\frac{c}{c_1}} = 0.01005 \text{ rad/in (0.3958 rad/m)}$$

$$\sinh kd_1 = 0.2436 \qquad \cosh kd_1 = 1.0292$$

$$\sinh kd_2 = 1.6334 \qquad \cosh kd_2 = 1.9152$$

$$\sinh kh = 2.1478 \qquad \cosh kh = 2.3692$$

From Eq. (7.1),

$$\nu = 1.5687 \qquad u = 0.9302$$

From Eq. (7.2),

$$w = -0.1669$$

The St. Venant shear will be greatest at $z = d_1$ and is given by Eq. (7.4), but a value for ϕ' is needed first and this is given by Eq. (7.3),

$$\phi_1'(d_1) = \frac{441.2[-0.1669(0.2436) - 0.9302(1 - 1.0292)]}{52{,}140}$$

$$= -\frac{5.9540}{52{,}140}\text{rad/in}\left(-\frac{0.673}{149.6}\text{rad/m}\right)$$

The stress coefficient δ is found from Figure 7.9 after reference to additional section properties. For this W14 column section, $R = 0.60$ in (15.2 mm), so $R/t_f = 0.80$

[2]This wide flange section is no longer in production.

and $t_w = 0.465$ in (11.8 mm) for a t_w/t_f ratio of 0.62. δ is then read from the curves as 1.25. Now

$$f_v = 11{,}000\left(-\frac{5.954}{52{,}140}\right)(1.25)(0.748)$$

$$= -2.07 \text{ kips/in}^2(-143 \text{ MN/m}^2)$$

For the maximum flange-bending stress, given by Eq. (7.5), ϕ'' must be evaluated at $z = 0$, or, by examination of Eq. (7.3).

$$\phi_1''(0) = \frac{Tkw}{c}$$

$$f_t = \frac{Esb_f Tkw}{4c} = \frac{30{,}000(13.37)(14.545)(441.2)(0.01005)(-0.1669)}{4(52{,}140)}$$

$$= -20.7 \text{ kips/in}^2(-143 \text{ MN/m}^2)$$

Likewise, for warping shear ϕ''' needs to be evaluated at $z = 0$ before Eq. (7.6) can be solved. Again examining Eq. (7.3), we see that

$$\phi_1'''(0) = \frac{Tk^2u}{c}$$

$$f_w = \frac{Esb_f^2 Tk^2 u}{16c} = \frac{30{,}000(13.37)(14.545^2)(441.2)(0.01005^2)(0.9302)}{16(52{,}140)}$$

$$= 4.2 \text{ kips/in}^2(29.1 \text{ MN/m}^2)$$

2. In order to reduce the somewhat high flange-bending stress induced by the torsional load, torsion reducers are installed consisting of angle irons 6 by 6 by ½ in (152 by 152 by 12.7 mm) with the eccentricity of the torsion reducer from column centerline $r = 8.24$ in (209 mm), $L_r = 240$ in (6.1 m), $A_r = 5.75$ in^2 (37.10 cm^2), and of course, $E_r = E$. What will be the tensile stress in the reducer and the new value of flange bending stress?

Solution

The parameter m in Eq. (7.7) is the absolute value of $\phi_1(d_1)/T$, or

$$m = \left|\frac{w(\cosh kd_1 - 1) - u(kd_1 - \sinh kd_1)}{ck}\right|$$

$$= \frac{0.2628}{52{,}140}\text{kip}\cdot\text{in}\left(\frac{29.69}{149.6}\text{N}\cdot\text{m}\right)$$

$$Q_r = \frac{20}{\dfrac{240(52{,}140)}{22.06(0.2628)(8.24)(5.75)(30{,}000)} + \dfrac{8.24}{22.06}}$$

$$= 10.57 \text{ kips (47.0 kN)}$$

The tensile stress in the reducer is then

$$f = \frac{10.57}{5.75} = 1.8 \text{ kips/in}^2(12.7 \text{ MN/m}^2)$$

$$Q_c = 20 - \frac{10.57(8.24)}{22.06} = 16.05 \text{ kips (71.4 kN)}$$

The remaining torsional flange-bending stress is then

$$f_t = -\frac{20.7(16.05)}{20} = -16.6 \text{ kips/in}^2(-114.5 \text{ MN/m}^2)$$

3. If r were increased to match the load eccentricity e using the same angle-iron reducer, what will be the effect?

Solution

$$Q_r = \frac{20}{\dfrac{240(52,140)}{22.06(0.2628)(22.06)(5.75)(30,000)} + 1}$$

$$= 12.76 \text{ kips } (56.8 \text{ kN})$$

$$f_t = \frac{20.7(20-12.76)}{20} = -7.5 \text{ kips/in}^2(-51.7 \text{ MN/m}^2)$$

Column Strengthening

An alternative to installing torsion reducers is to box the columns with plates, taking advantage of the effectiveness of a tube at resisting torsion. Depending on architectural or other needs, they may have to be removed later. Yet another method entails installation of a horizontal beam to transfer the torsional moment into a horizontal reaction at an adjacent column (Figure 7.5). The beam is positioned on the side of the derrick opposite to the load landing side so that it will not interfere with load handling.

Relief of excessive column bending in either or both axes can be achieved with tension or compression struts. A well-placed strut can double as a torsion reducer.

The Chicago boom puts opposing horizontal reactions at the topping lift and boom foot. A diagonal bracing system such as that shown in Figure 7.11 could be installed as a remedy for a frame with insufficient strength and rigidity to restrain this couple. If wire ropes or rods are used, turnbuckles are needed, and with rope the effects of stretch on the distortion of the building frame should be considered. Rigid braces might be a better choice.

When load-carrying ropes are secured to steel framing members, sharp corners must be blunted to prevent shearing of the ropes. Wood blocking or pipe segments are commonly used. In a wrap-and-shackle arrangement such as that shown in Figure 7.12, the angle that the rope wraps make with the shackle indicates a need to specify ropes for at least twice the design load of the attached line.

Fitting Attachment Bolts

Fasteners used to attach boom mounting fittings to a column should generally be preloaded high strength. The preloading prevents loosening; its clamping force resists sliding. But the connection

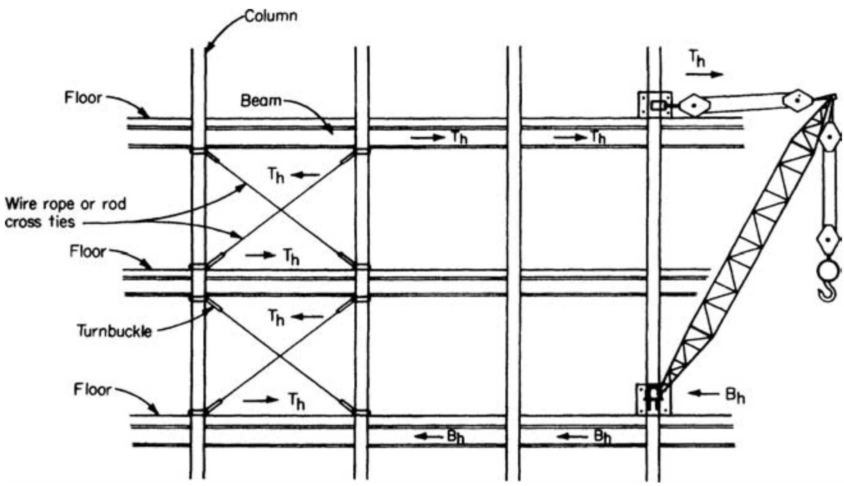


Figure 7.11 Wire-rope or steel-rod crossties can be used o stabilize a frame and prevent distortions from Chicago boom operations. The ties will carry all or a portion of the horizontal force component parallel to the building face, depending on the moment-transfer capabilities of the steel connections.

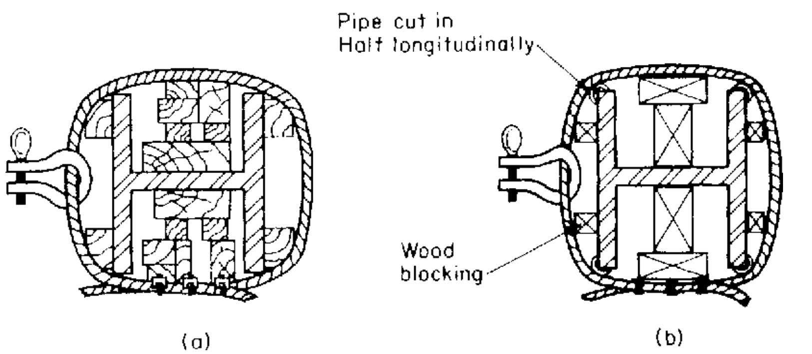


Figure 7.12 Beam or column rope ties can be used for rope anchorages. (a) Wood blocking is driven between steel and rope until the rope is tight and rope bends have been softened. (b) An alternative means for corner softening. When the first method is used, the spaces must be packed out fully both to tighten the rope and to soften the bends. With the second method, the packing is used only for tightening the rope.

must be properly detailed so that the column and mounting plate have the requisite strength and stiffness to resist the high preload forces. A bearing seat (Figure 7.13*a*) is another remedy to prevent the skewing and bending of the bolts that would occur if the mounting slipped.

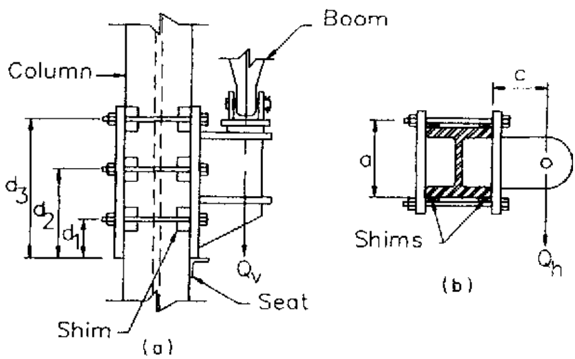


FIGURE 7.13 Foot mounting of a Chicago boom to a seat on a building column. (a) Elevation. (b) Plan.

To illustrate a derrick mounting to a column using high-strength bolts, the following values are assigned to the parameters in Figure 7.13:

$$Q_v = Q_h = 40 \text{ kips } (18.1\ t) \qquad a = c = 15 \text{ in } (381 \text{ mm})$$

$$d_1 = 6 \text{ in } (152.4 \text{ mm})\ d_2 = 12 \text{ in } (304.8 \text{ mm})\ d_3 = 18 \text{ in } (457.2 \text{ mm})$$

From the vertical load moment, each bolt in the vertical lines of bolts can be assumed to be loaded in tension in proportion to bolt distance above the seat. This will make their strains consistent. If F_v is the load thus induced in each of the topmost bolts, then

$$2\left(F_v d_3 + \frac{F_v d_2^2}{d_3} + \frac{F_v d_1^2}{d_3}\right) = Q_v c$$

$$F_v = \frac{1}{2}\left(\frac{40(15)}{18 + 12^2/18 + 6^2/18}\right) = 10.71 \text{ kips } (47.6 \text{ kN})$$

The horizontal moment $Q_h c$ induces tension in each of the left side bolts of

$$F_h = \frac{Q_h c}{3a} = 40\frac{15}{3 \times 15} = 13.33 \text{ kips } (59.3 \text{ kN})$$

The upper left-hand bolt will then be the most heavily loaded, carrying tension of

$$13.33 + 10.71 = 24.04 \text{ kips } (106.9 \text{ kN})$$

The allowable working tensile stress on A325 bolts is 40,000 lb/in^2 (275 MN/m^2) on the nominal diameter of the bolt, while the permitted shear stress is 22,000 lb/in^2 (150 MN/m^2). The shear load will be the resultant of Q_v and Q_h, $40\sqrt{2}/6 = 9.43$ kips (41.9 kN) for each of the six bolts.

The bolt area A can now be found by using a stress-interaction relationship

$$\frac{9.43}{22A} + \frac{24.04}{40A} \leq 1 \qquad A \geq 1.03 \text{ in}^2 \text{ (664 mm}^2\text{)}$$

Thus, 1¼-in-diameter (32-mm) bolts are needed.

The proper preload for an A325 bolt of this size will induce initial bolt tension of at least 71,000 lb (316 kN) for each of the bolts. The net clamping force on the fitting is obtained by summing the bolt preloads and subtracting out all the applied loads; the latter varies with each bolt according to its position. In this example, the net clamping force is 343 kips (1526 kN). Assuming that the steel-to-steel coefficient of static friction will be 0.15—a conservative figure for structural steel—the friction force developed will be about 51.5 kips (229 kN), which is just less than the vertical shear of $40\sqrt{2} = 56.6$ kips (252 kN). The seat must be capable of supporting the small excess. If common bolts were used, the seat would be required to carry the full vertical shear.

7.3 Guy Derricks

Guy derricks once abounded on urban skylines as the preeminent means to erect high-rise steel. By the 1990s, they had been rendered all but extinct by tower cranes and long-boom mobile cranes.

To an installation designer, the two most important aspects of a guy derrick are support of the *mast* loads and selection and anchorage of the guys. The guys must be designed first, as their loadings contribute to the mast load. The positioning and anchoring of the guys are an important consideration in selecting guy-derrick location in buildings under construction.

A guy derrick mast is seated on a base, or *footblock*, with a pivot arrangement that permits the *mast* to swing with the boom. The pivot is usually of the ball-and-socket type, which is necessary in order to allow the mast to lean freely in the direction of loading. It is the freedom of the mast to lean that makes the guying system for this type of derrick far less critical, in a design sense, than the mast guys of tower cranes. The mast of a guy derrick does not support any portion of the overturning forces by means of bending resistance. It is therefore not necessary to employ guy-force-measuring devices, but erection crews should still try to plumb the mast as closely as possible.

Guying Systems

For most practical guy derrick installations, the number of guys used and their placement is based solely on site and operating conditions

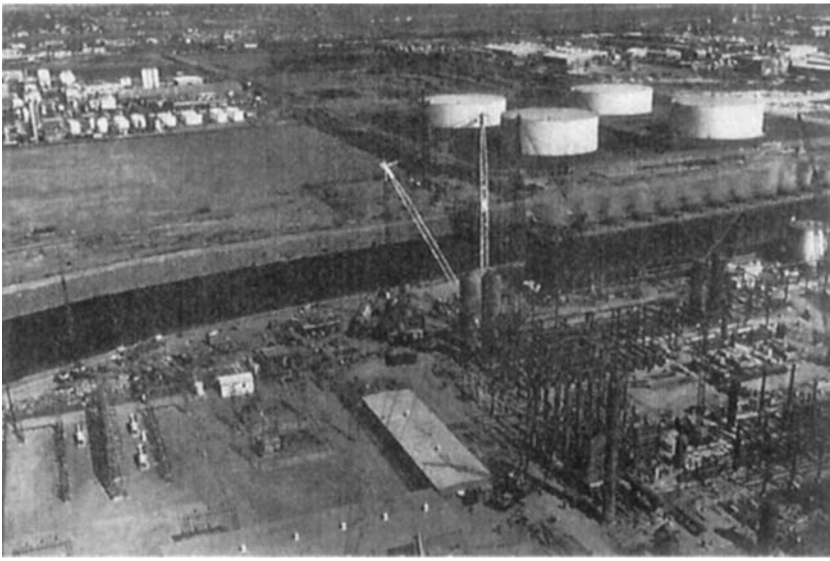


FIGURE 7.14 One of the guy derricks shown is mounted on a mobile-crane base, and the other is a standard derrick form. The congested site has made it necessary to place the guy anchorages far from the bases, some across the river. Flat guy angles, however, permit swinging at a large radius without hitting the guys.

(Figure 7.14). Although six or eight guys are usual, fewer can be used if the design adequately reflects the actual conditions and if the field crew understand and implement operating limitations.

For any operating position of the boom, guys that lie within 90° of either side of the boom in a horizontal circle are called *dead guys*, as they do not contribute to the support of applied loads. Guys within 75° of either side of the boom are taken, however, as imposing dead-load moment to the system.

Guys in the remaining 180° of arc are called *live guys*, and those lying within the 150° zone opposite the boom are considered to be effective, that is, participating, in the support of the boom and load (Figure 7.15).

When the derrick is under load, a horizontal component of force occurs at the mast top at the topping lift connection. The live guys will react to this force. Distribution of the reaction forces among the live guys will be such that the transverse forces will be in equilibrium and the geometric displacements will be consistent with rope strains and changes in the catenary sags. There is no assurance that the mast top will displace directly toward the boom. On the contrary, it can be expected that some transverse displacement will also occur.

A transverse displacement of the mast top will be accompanied by a mast dead-load moment in that direction and some live-load

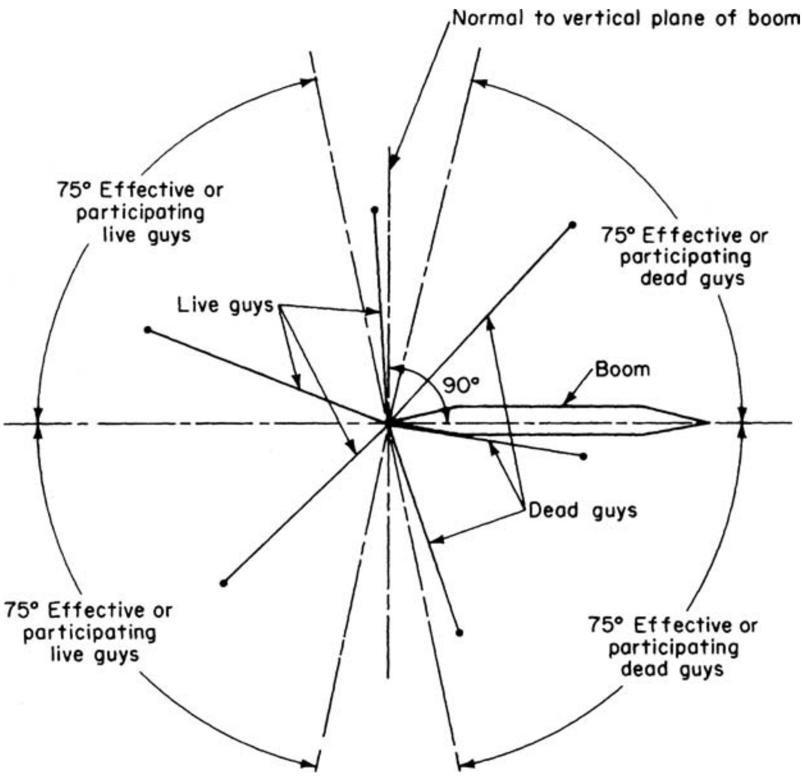


FIGURE 7.15 A guy derrick in plan view showing live guys and dead guys. Guys within 75° of either side of the boom are taken as imposing deadload moment to the system. Guys in the 150° zone opposite the boom are considered to be live guys that participate in the support of the boom and load.

effects as well. With small displacement—actual displacements in practice will be small—the induced moment will not attain an important magnitude, and consequently the loads added to the opposite guys will not be great.

As this type of guying system is not very sensitive, the practice in the United States is to use the approximate AISC method, which has proved to be adequate.[3] In this method, the guys must have a minimum safety factor of 3 with consideration of static loading only. But loading determined for assessing the support structure should include impact in conjunction with wind at 30 mi/h (13.4 m/s).

The load applied to the participating live guys is a horizontal force H at the guy connection level of the mast (Figure 7.16). The force is taken in the direction of the boom and comprises two parts.

[3]"Guide for the Analysis of Guy and Stiffleg Derricks," American Institute of Steel Construction, Inc., New York, 1974, p. 15.

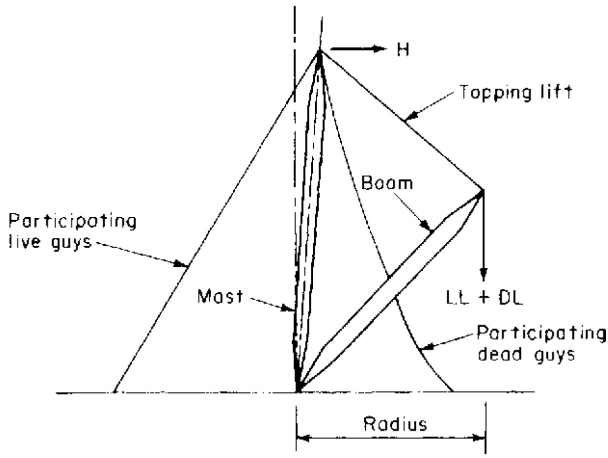


Figure 7.16 The load applied to the participating live guys is a horizontal force H at the guy connection level of the mast.

The first is induced by the dead loads of the boom and topping lift, along with the lifted-load and hoist-rope forces; the second is an empirical quantity reflecting the effect of the participating dead guys. This deadload component of guy force can be taken as

$$H_g = \begin{cases} 0.006N(L_a - 12) & \text{rope diam} \le 1 \text{ in} \\ 0.087N(L_a - 3.66) & \text{rope diam } \le \text{ 25 mm} \\ 0.012N(L_a - 12) & \text{rope diam} > 1 \text{ in} \\ 0.0175N(L_a - 3.66) & \text{rope diam} > 25 \text{ mm} \end{cases} \tag{7.8}$$

where N is the number of effective dead guys and L_a is the average horizontal projection of the effective dead-guy lengths in a direction parallel to the boom in feet or meters. The resulting value for H_g will be in kips or kilonewtons in the respective formulas for inch- and millimeter-size ropes.

The load H must then be amplified to account for the small moment that could be added by a transverse lean of the mast, as mentioned earlier. The AISC suggests an increase of 5%, or a multiplier of 1.05, for derricks used in steel-erection service. That same factor can be realistically taken for most installations.

If each of the live-guy loads, taken in the direction of the guy-rope centerlines, is denoted G_1, G_2, G_3, and G_4 and the guy slope lengths (actual length) as L_1, L_2, L_3, and L_4 with nominal rope cross-sectional areas A_1, A_2, A_3, and A_4, the following expressions can be applied when from one to four guys are effective:
With only one guy in effective live-guy area.

$$G_1 = \frac{HL_1}{x_1}$$

With two guys in effective live-guy area.

$$G_1 = \frac{1.05H}{x_1/L_1 + (L_1^2/A_1x_1)(A_2x_2^2/L_2^3)}$$

$$G_2 = \frac{1.05H}{x_2/L_2 + (L_2^2/A_2x_2)(A_1x_1^2/L_1^3)}$$

With three guys in effective live-guy area.

$$G_1 = \frac{1.05H}{x_1/L_1 + (L_1^2/A_1x_1)(A_2x_2^2/L_2^3 + A_3x_3^2/L_3^3)}$$

$$G_2 = \frac{1.05H}{x_2/L_2 + (L_2^2/A_2x_2)(A_1x_1^2/L_1^3 + A_3x_3^2/L_3^3)}$$

$$G_3 = \frac{1.05H}{x_3/L_3 + (L_3^2/A_3x_3)(A_1x_1^2/L_1^3 + A_2x_2^2/L_2^3)}$$

With four guys in effective live-guy area.

$$G_1 = \frac{1.05H}{x_1/L_1 + (L_1^2/A_1x_1)(A_2x_2^2/L_2^3 + A_3x_3^2/L_3^3) + A_4x_4^2/L_4^3}$$

$$G_2 = \frac{1.05H}{x_2/L_2 + (L_2^2/A_2x_2)(A_1x_1^2/L_1^3 + A_3x_3^2/L_3^3 + A_4x_4^2/L_4^3)}$$

$$G_3 = \frac{1.05H}{x_3/L_3 + (L_3^2/A_3x_3)(A_1x_1^2/L_1^3 + A_2x_2^2/L_2^3 + A_4x_4^2/L_4^3)} \qquad (7.9)$$

$$G_4 = \frac{1.05H}{x_4/L_4 + (L_4^2/A_4x_4)(A_1x_1^2/L_1^3 + A_2x_2^2/L_2^3 + A_3x_3^2/L_3^3)}$$

where x_1, x_2, x_3, and x_4 are the horizontal projections of the guys in a direction parallel to the boom (Figure 7.17). Inasmuch as Eq. (7.9) contains only ratios of areas, nominal as opposed to actual rope metallic cross-sectional areas can be used.

It will generally be necessary to evaluate guy loadings under several conditions. Normally, as is usual in steel erection work, the most severe loading will take place during initial lifting of loads from ground-level truck unloading points to the working level. Each active guy must be evaluated for this case. Next, after booming in, the load will be swung to a landing position. The swing radius during operation must be small to permit the boom to pass under the guys, so it is unlikely that critical or governing loads will develop here, but if they do, each guy is checked when the boom is opposite to the guy in question. Finally, maximum conditions during load placement may well induce maximum loading on particular guys. Each guy is then designed for its greatest calculated load.

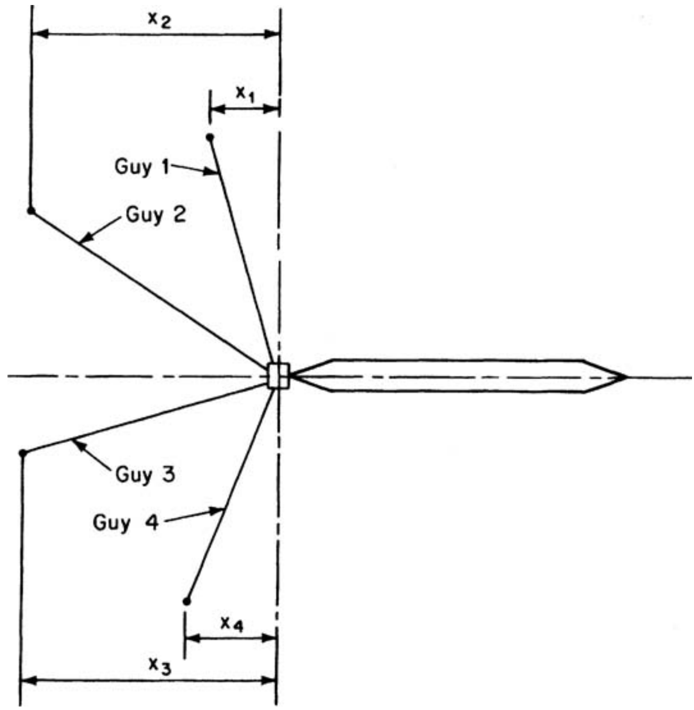


Figure 7.17 Horizontal projections of the guys in a direction parallel to the boom, used in Eq. (7.9).

As a start, it is convenient to assume that each guy in the set will be of the same cross-sectional area; this permits the area parameter to drop out in Eq. (7.9). Once loads have been determined on that basis, guy ropes can be tentatively selected and the calculations repeated to check the ropes chosen.

A value chosen for impact loading will be a matter for the designer's judgment after considering the type of work to be performed and the working conditions. For steel erection work, the AISC suggests 20% with a properly matched winch. Higher impact forces can occur when the winch is capable of much greater line pull than needed. Impact is used in the design of the guy anchorages and of the mast support but not of the guy ropes, unless unusually high impact forces are expected.

Example 7.2

With the boom in the position shown in the guying system of Figure 7.18, guys 2 and 3 receive their heaviest loading. In this position the horizontal reaction at the mast top induced by the boom and load is 55 kips (244.7 kN). Assuming that all dead guys will be greater than 1 in (25 mm) in diameter, design guys 2 and 3 using 6 × 19 class IWRC IPS wire rope. Also assume that guy 1 will be 1⅛ in (29 mm) in diameter.

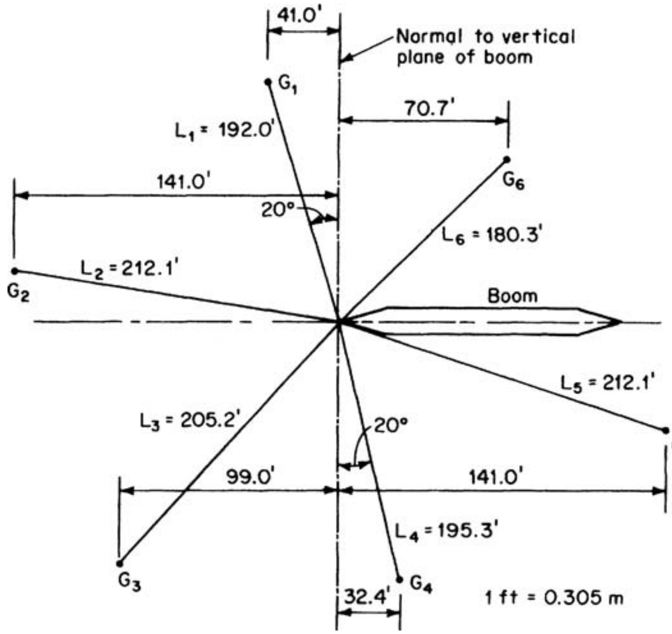


FIGURE 7.18 Illustration of Example 7.2.

Solution

Guys 4, 5, and 6 are each participating dead guys, so parameter L_a of Eq. (7.8) is

$$L_a = \frac{32.4+141.0+70.7}{3} = 81.4 \text{ ft } (24.8 \text{ m})$$

The dead-load effect of the dead guys is then

$$H_g = 0.012[81.4-12)] = 2.5 \text{ kips } (11.1 \text{ kN})$$

making the total horizontal reaction at mast top

$$H = 55.0+2.5 = 57.5 \text{ kips } (225.8 \text{ kN})$$

For a first trial it will be assumed that all participating live guys will be 1⅛ in (29 mm) in diameter in order to simplify Eq. (7.9). Using only the needed expressions for three effective guys, we have

$$G_2 = \frac{1.05(57.5)}{\dfrac{141.0}{121.1}+\dfrac{212.1^2}{141.0}\left(\dfrac{41.0^2}{192.0^3}+\dfrac{99.0^2}{205.2^3}\right)} = 54.8 \text{ kips } (244 \text{ kN})$$

$$G_3 = \frac{1.05(57.5)}{\dfrac{99.0}{205.2}+\dfrac{205.2^2}{99.0}\left(\dfrac{41.0^2}{192.0^3}+\dfrac{141.0^2}{212.1^3}\right)} = 41.1 \text{ kips } (183 \text{ kN})$$

The break strengths required for ropes 2 and 3 are then

$$\mathbf{BS}(2) = 54.8(3/2) = 82.2 \text{ tons } (74.6 \text{ t})$$

$$\mathbf{BS}(3) = 41.1(3/2) = 61.7 \text{ tons } (56.0 \text{ t})$$

For rope 2 we tentatively choose a rope of 1⅜-in (35 mm) diameter and break strength of 87.1 tons (79.0 t), while for number 3 a 1¼-in-diameter (32-mm) rope of 70.4 tons (63.9 t) break strength will be tried. Therefore, $A_2 = 1.49 \text{ in}^2 (961 \text{ mm}^2)$, $A_3 = 1.23 \text{ in}^2 (794 \text{ mm}^2)$, and guy 1 was given as 1⅛-in (29-mm) diameter, or $A_1 = 0.99 \text{ in}^2$ (639 mm²) and break strength 57.0 tons (51.7 t). Inserting these values into Eq. (7.9) for the two ropes to be designed gives

$$G_2 = \frac{60.4}{\dfrac{141.0}{212.1} + \dfrac{212.1^2}{1.49(141.0)}\left[\dfrac{0.99(41.0^2)}{192.0^3} + \dfrac{1.23(99.0^2)}{205.2^3}\right]}$$

$$= 59.6 \text{ kips } (265 \text{ kN})$$

$$G_3 = \frac{60.4}{\dfrac{99.0}{205.2} + \dfrac{205.2^2}{1.23(99.0)}\left[\dfrac{0.99(41.0^2)}{192.0^3} + \dfrac{1.49(141.0^2)}{212.1^3}\right]}$$

$$= 36.9 \text{ kips } (164 \text{ kN})$$

For the revised loads, the required break strengths are

$$\mathbf{BS}(2) = 59.6(3/2) = 89.4 \text{ tons } (81.1 \text{ t})$$

$$\mathbf{BS}(3) = 36.9(3/2) = 55.4 \text{ tons } (50.3 \text{ t})$$

Guy 2 must be changed, as the rope previously selected lacks sufficient strength. When we substitute a rope of 1½-in (38-mm) diameter with break strength of 103 tons (93.4 t), A_2 becomes 1.77 in² (1140 mm²). But guy 3 has excessive strength, and a smaller rope, 1⅛ in (29 mm) diameter, can be tried. Rechecking in Eq. (7.9), we get

$$G_2 = 66.4 \text{ kips } (295 \text{ kN}) \qquad \text{BS}(2) = 99.6 \text{ tons } (90.4 \text{ t})$$
$$G_3 = 27.9 \text{ kips } (124 \text{ kN}) \qquad \text{BS}(3) = 41.9 \text{ tons } (38.0 \text{ t})$$

Guy 2 will be 1½ in (38 mm) in diameter, and guy 3 will be 1⅛ in (29 mm) in diameter.

Figure 7.19*b* shows that selecting guy positions can be quite difficult in the real world of the jobsite.

Footblock Supports

The dominant load acting on the *footblock* is the vertical reaction coming from the boom and the guys, but it is also subjected to horizontal forces and moments of a lesser magnitude due to eccentricities and lateral pulls.

Eccentric loads acting on the mast induce a net horizontal reaction at the ball-and-socket connection between the mast and the footblock. The sources of these eccentric loads are the guy connections, the topping lift, and the boom foot. As the reaction is above the

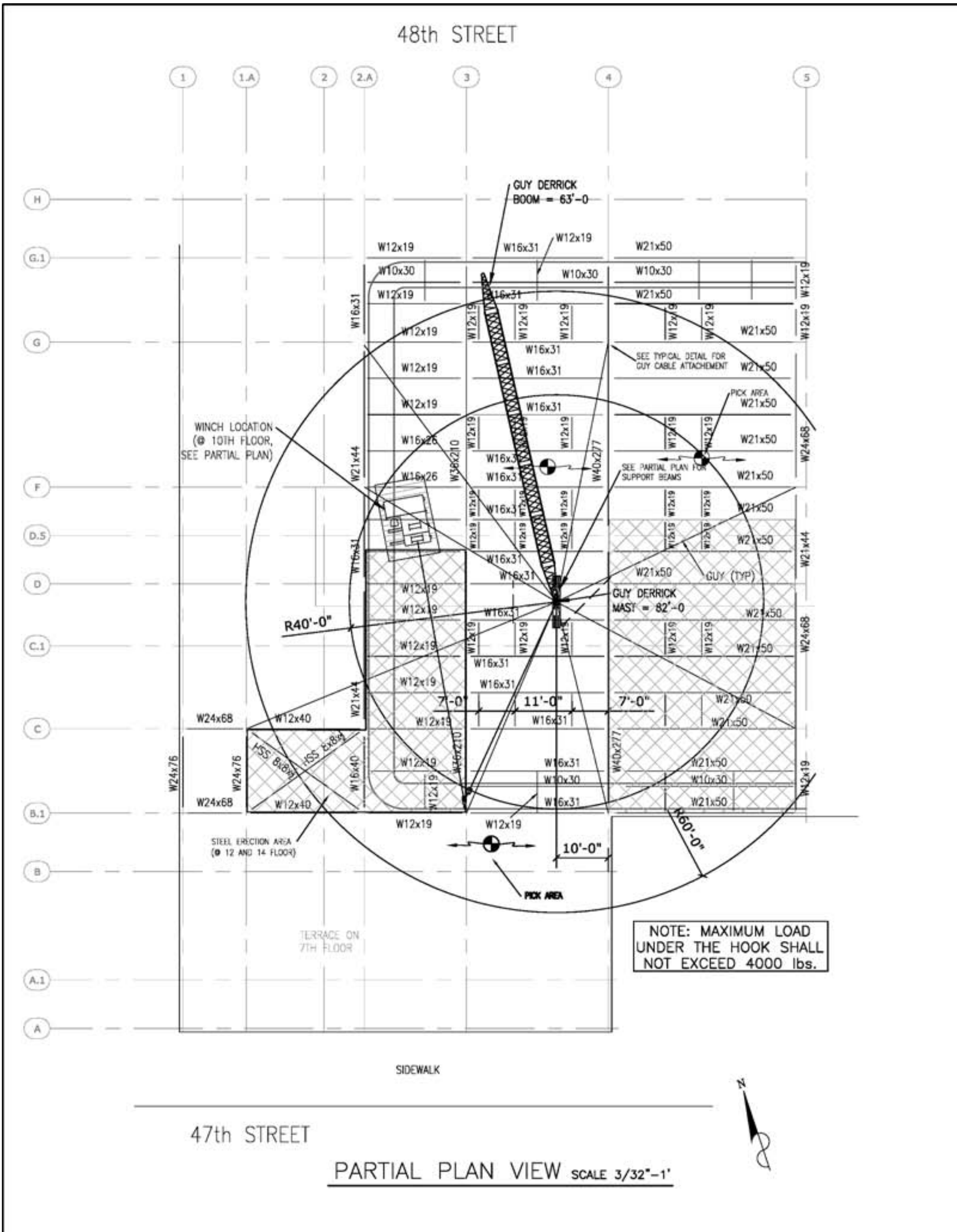
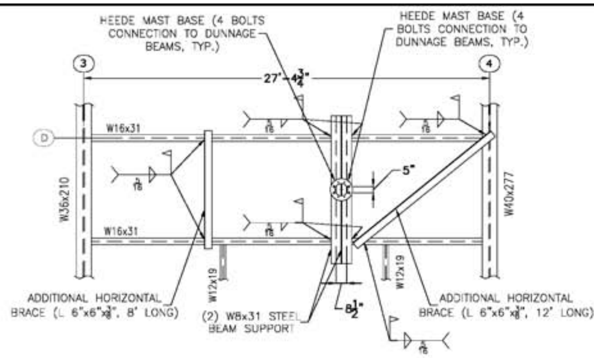
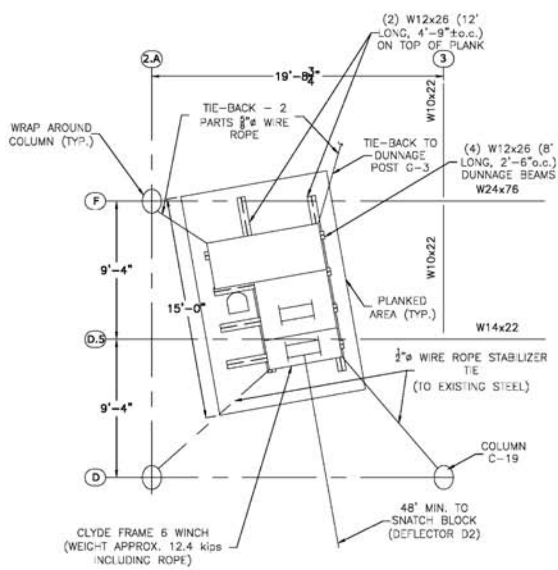


FIGURE 7.19 Although the tower crane has displaced the guy derrick as the choice for lifting machine of choice for high rise steel erection, an occasional application is found for the latter . (a) This light guy derrick was set up on the 12th floor of an existing building to erect steel that has been landed on the building with a mobile crane.



PARTIAL PLAN SCALE ¼"-1'
(SHOWING MAST SUPPORT)



PARTIAL PLAN SCALE ¼"-1'
(SHOWING WINCH SUPPORT)

STEELWORK NOTES:

1. All new steel support members except beams and tubes to be ASTM A36 or better weldable material. New beams to be ASTM A-992, 50 ksi yield. Tubes to be ASTM A-53 Type E or S, Grade B, or ASTM A-500 Grade B, or ASTM A-501, or better.
2. Use E 70xx welding electrodes. Welding to be per AWS D1.1, latest edition, by NYC certified welders.

Maximum Hook Load :	4.0 k	Winch Data :		Reactions under the Derrick [in Kips] – Including 20% Impact Factor			HEEDE GUY DERRICK 8½ t.
Ball and Block etc.:	1.0	Hoist Line Pull :	12.0				C.D.# 2624 C.N.#
Hoist Rope :	0.5	Luff Line Pull :	12.0		Mast	Guys (1 to 8, MAX.)	Mast: 82'-0" Sills: –
20% Impact Factor :	1.0	Total Line Pull =	24	Downthrust ↓	26.2	0.5	Boom: 63'-0" Guys:(8) 3/4"ø
Half Boom and Luff :	1.5	Max Fleet Angle :	1.3	Uplift ↑	0	5.6 (MAX.)	Max. Hook Load: 4000 lbs
TOTAL =	8.0	Weight of Winch :	12.0k				at 60 ft Radius Max.

No. REVISIONS BY

HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
303 MERRICK ROAD, SUITE 400, LYNBROOK, N.Y. 11563
TEL: (516) 791-2600 FAX: (516) 791-5425

DERRICK INSTALLATION FOR DCM ERECTORS

PROPOSED BUILDING ALTERATION AT 1211 Ave. of AMERICAS, NYC

DATE: 12/27/07
SCALE: AS NOTED
SHEET 1 OF 2 SHEETS

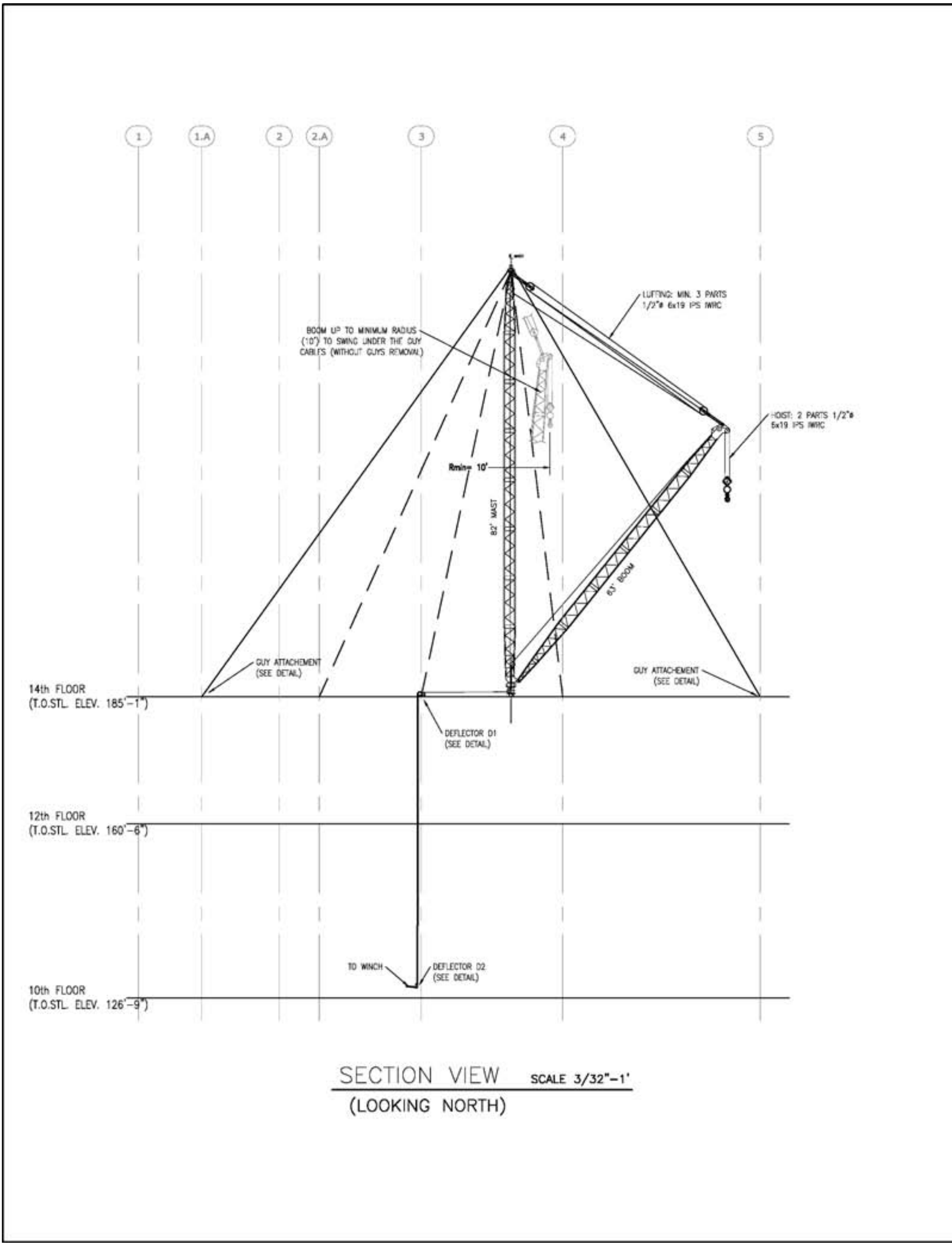
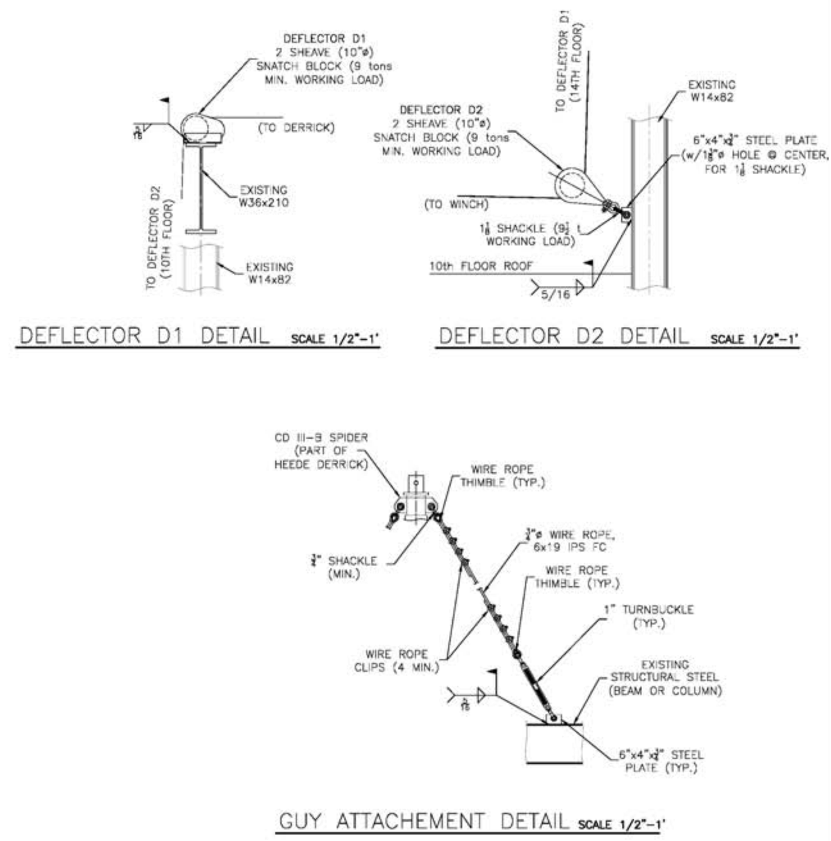
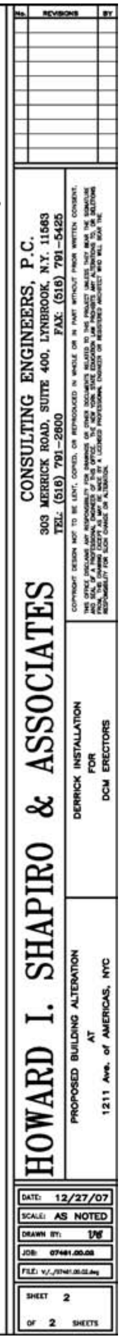


Figure 7.19 (b) By first booming up under the guys, the derrick can swing and place loads over a full 360 degrees.



GENERAL NOTES:

1. Contractor to verify all dimensions and site conditions in the field before commencing work or the fabrication of steel components. Any errors, omissions or unusual field conditions to be reported to the engineer immediately.
2. Thoroughly inspect all derrick components before assembly. Bends, nicks and gouges in chord members are reason for rejection. Nicks and gouges in lacing members are reason for rejection or replacement of the member. Smooth bends in lacings may be straightened. Repair cracked welds per manufacturer's instructions.
3. The base steel for the derrick is to be set at Elev. 185'-9" (top of additional W8X31 steel beams).
4. Install the mast base fitting first. Assemble the full mast horizontal and attach all guys at the top (at spider component). Next raise the mast in the vertical position; adjust all guy cables so they are snug tight and the mast is vertical. DO NOT over-tighten the guy cables. Continue with erection of the boom.
5. After completion of derrick erection and reeving, inspect the entire installation. It is recommended that a load test be conducted before placing the derrick in operation. Test with 5,000 lbs. load at rated 60 ft. radius, raising the load only a few inches and holding for about 10 minutes. After lowering the load, inspect the derrick and support steel for signs of distress. If all is ok, proceed to work.
6. Do not operate the derrick in winds of 30 MPH or more. Raise and lower loads using tag lines to control the load.
7. Operation of the derrick may loosen bolts and fittings. Check bolts and fittings regularly for tightness.
8. The installation, operation and maintenance of the derrick is to be in accordance with ASME/ANSI B30.6 latest edition and OSHA regulations.



base, it induces a moment that acts on the host structure along with a net lateral force. Hoist and topping leads lines running out of the footblock to the winch will introduce additional horizontal force and moment at the footblock.

Slewing moment, albeit small, is resisted by a reaction couple at the base mounting. The derrick can be slewed by a swinger winch and bull wheel, a manually operated crank and gear set, or other means. In calm air only friction forces and the effects of mast lean under load need to be overcome. But when the wind blows, swing torque can materially increase. When a winch is used for slewing, its line pull is additional horizontal force acting on the footblock.

With footblock loads determined, design of the anchorage and supports follows ordinary structural procedures. As shown in Figure 7.19*a*, shoring down several floors may be necessary to distribute mast loads to within framing member capacity. Deflections in the supports may need to be checked, as excessive deflections can cause guys to slacken.

Guy derricks that are used for steel erection are raised, or *jumped,* as the framework of the structure rises. The derrick can be used to raise itself unassisted, taking advantage of the fact that the boom length is limited so that it can pass under the guys and permit full-circle slewing.

First, the boom is brought to a vertical position and held by the topping lift while the boom foot pin is removed. The boom foot is then swung clear of the mast base, rotated 180°, and lowered onto a footplate that has been anchored to the structure in advance. A few temporary guys are made fast to the boom head, and the boom is leaned in toward the mast. By rotating the boom, the load fall has been placed on the mast side and the boom has been converted into a gin pole.

A lashing is placed about the mast at a point above its CG, which that includes the weight of the footblock, and the hook of the gin pole is then attached. When the mast guys have been released the mast is raised together with its footblock to the next level, and support beams are slid into place. The mast guys are then reset at the new level.

Finally, using the topping lift to support the boom head, the boom guys are released and the boom is rotated and raised. With the boom foot pin reinstalled, the guy derrick is ready for service at the new top of steel elevation.

Guy Derrick Adaptations

For added capacity, two guy derricks can be mounted side by side. The derricks can then be operated in tandem with two active hooks or with the load blocks made fast to a lifting beam or similar load-handling device. This setup can perform only lifting and luffing functions, and it is essential that procedures be established to assure that the motions are reasonably well synchronized.

Some older model crawler-mounted mobile cranes are equipped with optional guy derrick front-end attachments, but when so outfitted, they are no longer mobile. The boom hoist is used to erect a mast to vertical, and the guys are then made fast. The derrick boom is raised by using a line from one of the hoist drums, which then serves to provide the luffing motion. A pair of guy derrick-equipped cranes can also be used in tandem or a fixed derrick can be paired with a crane-mounted rig (Figure 7.20).

Another adaptation utilizes the mast and boom of a guy derrick but does not require the use of guys. This variant, called a *column derrick* (Figure 7.21), can be used on building setbacks or at any location where an adjacent structure is taller than the derrick mast; it has even been mounted on a frame cantilevered from the side of a building. The guys are replaced by structural member ties to the adjacent structure that are made fast to the derrick spider or to the gudgeon pin. When used this way, the mast must be held plumb throughout its limited swing arc. A column derrick can often be installed in lieu of a Chicago boom when long reaches and high-capacity lifts must be accommodated.

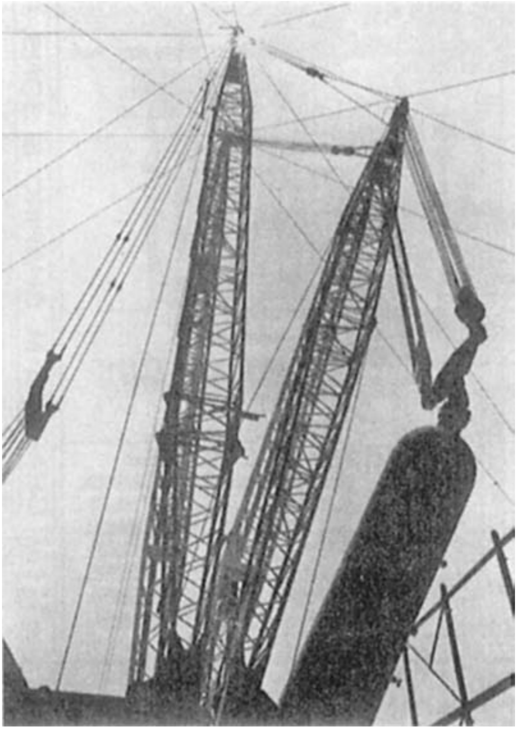


FIGURE 7.20 A mobile-crane-mounted guy derrick is paired with a fixed guy derrick for this tandem lift.

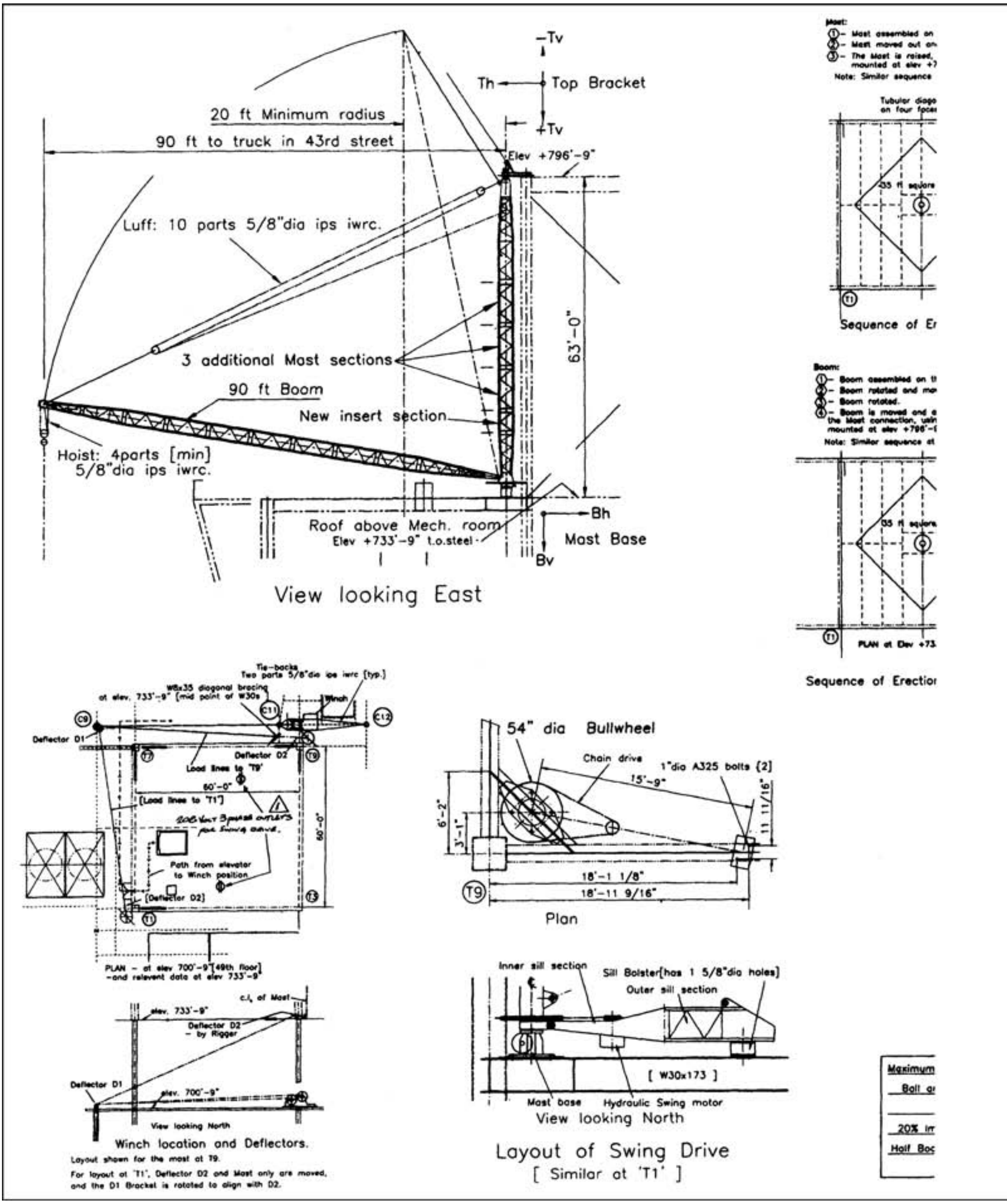
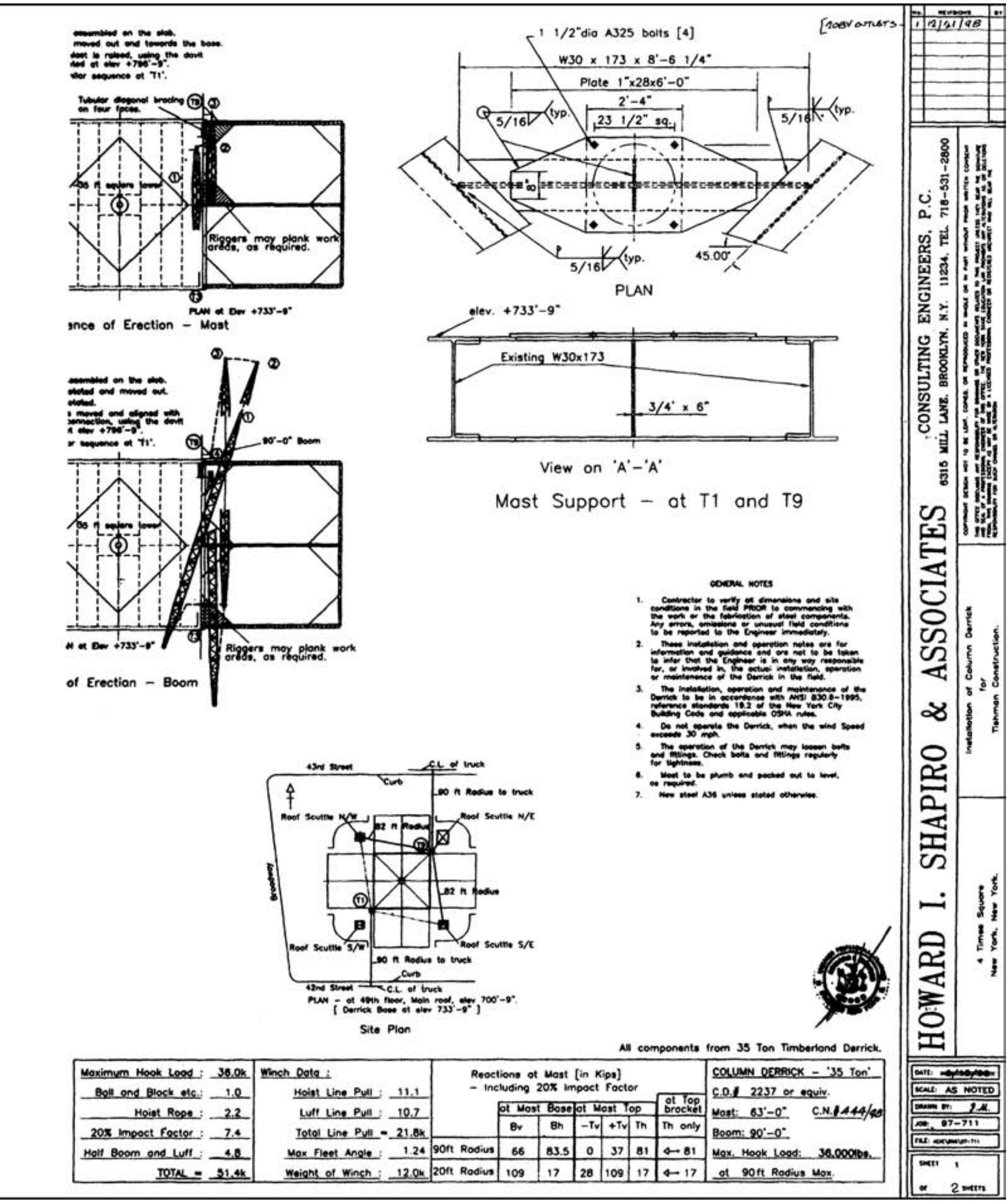


FIGURE 7.21 Installation drawing for a column derrick near the top of a high-rise office tower. The derrick is intended for future use in changing out mechanical equipment. The derrick itself will not be left in place, but all of the arrangements for it will be permanent fixtures. In the center of the drawing are instructions for erecting the mast and the boom. Further details are given on a second sheet (not shown).


1 1/2"dia A325 bolts [4]
W30 x 173 x 8'-6 1/4"
Plate 1"x28x6'-0"
2'-4"
23 1/2" sq.
5/16 typ.
45.00°
PLAN
elev. +733'-9"
Existing W30x173
3/4' x 6"
View on 'A'-'A'
Mast Support – at T1 and T9
Tubular diagonal bracing on four faces.
Riggers may plank work areas, as required.
PLAN at Elev +733'-9"
of Erection – Mast
90'-0" Boom
of Erection – Boom
GENERAL NOTES
1. Contractor to verify all dimensions and site conditions in the field PRIOR to commencing with the work or the fabrication of steel components. Any errors, omissions or unusual field conditions to be reported to the Engineer immediately.
2. These installation and operation notes are for information and guidance and are not to be taken to infer that the Engineer is in any way responsible for, or involved in, the actual installation, operation or maintenance of the Derrick in the field.
3. The installation, operation and maintenance of the Derrick to be in accordance with ANSI B30.6-1995, reference standards 19.2 of the New York City Building Code and applicable OSHA rules.
4. Do not operate the Derrick, when the wind Speed exceeds 30 mph.
5. The operation of the Derrick may loosen bolts and fittings. Check bolts and fittings regularly for tightness.
6. Mast to be plumb and packed out to level, as required.
7. New steel A36 unless stated otherwise.
43rd Street
C.L. of truck
Curb
90 ft Radius to truck
Roof Scuttle N/W
Roof Scuttle N/E
82 ft Radius
Broadway
Roof Scuttle S/W
Roof Scuttle S/E
42nd Street
PLAN – at 49th floor, Main roof, elev 700'-9".
[Derrick Base at elev 733'-9"]
Site Plan
All components from 35 Ton Timberland Derrick.
Maximum Hook Load : 36.0k
Ball and Block etc.: 1.0
Hoist Rope : 2.2
20% Impact Factor : 7.4
Half Boom and Luff : 4.8
TOTAL = 51.4k
Winch Data :
Hoist Line Pull : 11.1
Luff Line Pull : 10.7
Total Line Pull = 21.8k
Max Fleet Angle : 1.24
Weight of Winch : 12.0k
Reactions at Mast [in Kips] – Including 20% Impact Factor
at Mast Base
at Mast Top
at Top bracket
Bv Bh –Tv +Tv Th Th only
90ft Radius 66 83.5 0 37 81 ← 81
20ft Radius 109 17 28 109 17 ← 17
COLUMN DERRICK – '35 Ton'
C.D.# 2237 or equiv.
Mast: 63'-0"
C.N.#444/98
Boom: 90'-0"
Max. Hook Load: 36,000lbs.
at 90ft Radius Max.
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
6315 MILL LANE, BROOKLYN, N.Y. 11234, TEL 718-531-2800
Installation of Column Derrick for Tishman Construction.
4 Times Square New York, New York.
SCALE: AS NOTED
JOB: 97-711
SHEET 1 of 2 SHEETS

7.4 Gin Pole Derricks

A heavy-duty gin pole is nothing more than a guyed mast or boom rigged with a load fall. A single pole is set near vertical and usually not more than 10° from plumb, and is used for lifting without radius change. To assure adequate clearance between the load and the mast, the derrick height must often be considerably greater than lift height.

A large, heavy lift gin pole lends itself to difficult jobsite conditions where other equipment types might be at a disadvantage. As a boy, one of the authors tagged along with his father to the site of a 250-ft (76-m) pole in the center of a power plant building frame. Unusual and dramatic, he recalls the sight of this pole clearly forty years later. The base was anchored to a concrete foundation at ground level and the pole head projected above the roof framing. Four massive plate girders were erected by the gin pole raising and placing the first girder, decreasing the lean for the second, rotating the pole 180°, and repeating for the next two girders. The unusual choice of a pole within a robust high steel framework led to the idea of using only four guys, one to each building corner at roof level. Each guy was fitted at the anchorage end with a multiple-part reeving and a hand-operated winch. This arrangement enabled rapid and accurate changes in the lean between lifts as well as precise placement of the loads. The high elevation of the guy anchorages resulted in small guy loads considering the magnitude of the lifted loads. Nowadays a huge mobile crane would likely have been selected for this operation, but back then such machines existed only in the minds of futurists.

Six or more guys are arranged about the pole exactly as for a guy-derrick mast, and guy design uses Eq. (7.9) as well as the procedures previously described.

Guys on gin poles should be more snugly preloaded than on guy derricks because gin poles are more prone to radius increase when the load is picked. The greater the preloading, the less the guys will permit the radius to drift; but caution is necessary because excessive guy tensioning can cause the mast to be overloaded in compression. Preload and radius increase can be calculated with reasonable accuracy using the concepts given under Exact Analysis of a Guyed Mast in Sec. 6.4.

Alternatively, field procedures can be followed that will preclude the need for exact mathematical analysis and permit the guys to be designed using the AISC method given earlier. The calculated value of H should be increased by about 15%, instead of the 5% suggested for guy derricks, to allow for expected increases in guy forces. Also, the preloading in the dead guys must be allowed for as an amplification of the dead load reaction H_g; 2 times H_g can be used.

The operation starts with adjustment of the guys to support the mast at a radius somewhat less than the desired value. The load is positioned at the planned radius, however. With the load falls made fast to the load, a strain is taken on the load line while the radius of the upper load block is monitored with a surveyor's transit. If the mast leans out too far, the strain is released and the participating live guys are given more preload. But if the load is about to lift off and the mast has not yet come to the required radius, the load line must be released and some slack given to the live guys. After successive adjustments have brought the boom to the proper loaded radius, the derrick is ready to lift.

Tandem Gin Poles

A pair of gin poles erected plumb has often been used for erecting tall vessels at refineries or other such facilities; the vessel can be considerably taller than the poles (Figure 7.22), but the guys must be positioned to allow clear space for the top end of the vessel as it rises.

The poles are initially erected with a slight lean away from each other. Guy preloading is then adjusted so that the poles come into plumb when loaded. The guys should be arranged to provide a clear area for a tailing crane to handle the bottom of the vessel.

With tandem gin poles, heavy awkwardly proportioned loads can be erected in confined working spaces. A tandem installation

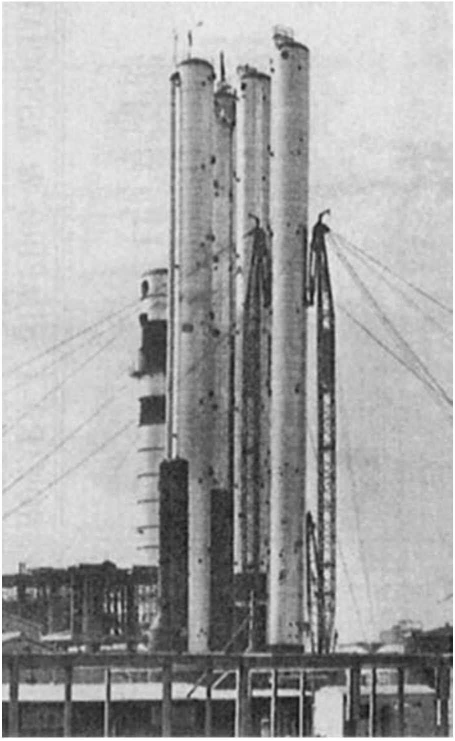


Figure 7.22 A pair of heavy gin poles has been used to erect vessels considerably taller than the poles themselves in this incredibly tight work area. (*American Hoist and Derrick Co.*)

implies a three-hook lift where the third hook is the tailing device. Side loading on the poles from the tailing operation should be considered when designing the guys. As in any three-hook operation, tailing must be closely coordinated with hoisting, and the winches powering each pole must be coordinated with each other.

The guys supporting tandem gin poles fall into two categories: *back guys* react to load weight and *fore-and-aft* guys stabilize the poles while resisting side loading. Back-guy forces can be calculated using the method described for guy derricks, but the side loading of the fore-and-aft guys depends on control procedures used in the field, as described under Tailing Operations in Chap. 5. The adequacy of each guy, however, is only as good as its anchorage, or *deadman*. Design of these anchorage points, with both uplift and lateral loads to consider, is a complicated exercise with numerous possibilities of field conditions and construction solutions. As a geotechnical problem, it is outside of the scope of this text.

Heavy gin poles are simple rugged hoisting devices, practical for a limited realm of lifting assignments, but their limitations are leading to obsolescence. Very few rigging firms now own them, and few riggers are experienced in their use. They are being replaced mostly by ever-more-capable mobile cranes.

A variation of a tandem gin pole was spotted at water's edge by one of the authors in a Gulf Coast facility that fabricates deep-sea oil platforms Figure 7.23.

Figure 7.23 A mammoth-sized tandem gin pole for lifting vessels at a fabrication yard on the Gulf Coast. (*Lawrence K. Shapiro.*)

Light-Duty Gin Poles

Light-duty gin poles are markedly different than the heavy variety in their uses and means of operation. A small pole might be a practical solution for lifting in a confined place where it can be assembled manually and perhaps operated without power, too. Its small loadings allow guy anchorages and foot-block supports to be relatively simple (Figure 7.24). Due to the relatively light loads, lean angle need not be restricted to near-vertical and a live topping lift is possible. But even a small gin pole can engender some rigging subtleties that could make it hazardous in the hands of novices.

At large angles of lean the entire load should be taken by no more than two participating live guys, but one single live guy is also acceptable. As guys are only nominally preloaded, reliance on more than two guys can result in uncertain load distribution and excessive lateral displacement under load. The dead guys, two or more, are then placed on either side and serve only to steady the pole.

When two effective live guys are used, they must be arranged symmetrically with respect to pole lean. Then in the two-guy equations of Eq. (7.9) $L_1 = L_2$, $A_1 = A_2$, and $x_1 = x_2$, give

$$G_1 = G_2 = \frac{1.05HL}{2x} = \frac{0.525HL}{x}$$

assuming that a 5% misalignment factor is appropriate for the installation conditions at hand.

Figure 7.24 The very light duty gin pole in the foreground is used to erect and dismantle the stiffleg derrick seen in part. For the small loads involved, the pole is seated on a piece of planking located above a roof beam, and wire-rope ties are used to stabilize the pole foot. (*Lawrence K. Shapiro.*)

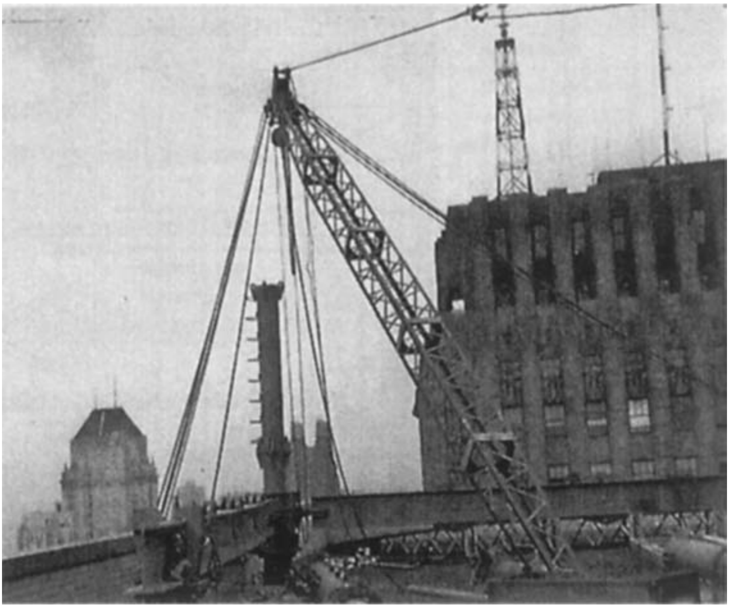


FIGURE 7.25 The same gin pole shown in Figure 7.24. The manila rope side guys are taken up tight to stabilize the pole, and a chain-fall is used for handling the loads. (*Lawrence K. Shapiro.*)

Side guys for light-duty gin poles are often made up of several parts of manila rope which permit easy pole alignment by hand (Figure 7.25). The side guys should be mildly preloaded to improve stability; these guys must also withstand the forces of wind and inadvertent load swing. In designing manila-rope side guys, rope stretch is more important than strength, so a few more parts might be used than dictated by strength needs.

For a gin pole that changes radius while under load, the locations of the side-guy anchorages are critical. The anchor points must form an axis parallel to the pole foot pivot axis; otherwise they will respond asymmetrically to the luffing motion. Without adequate intervention the pole may bend and fail sideways.[4] Ideally the anchorages should be placed exactly on the same horizontal axis as the pole foot pivot, as the guy length will then remain constant throughout the luffing motion. If the side-guy lengths change with

[4]A slackened side guy will allow the pole to lean to one side, shifting the topping load from the topping lift to the erstwhile slackened guy; the guy effectively becomes the topping lift. This action is likely to be a prelude to collapse. However, this sensitivity to side guy balance and the side-collapse hazard disappear when the topping lift is above the gin pole head as in Figure 7-25.

the radius, the derrick can be luffed safely only if the guys are deftly kept adjusted by the riggers.

The footblock must be anchored to prevent lateral movement from the horizontal component of pole axial load and from the lead-line forces. Holdback ropes (Figure 7.26) are sometimes used for this purpose.

7.5 Stiffleg Derricks

Going against the general trend for derricks, stiffleg derricks are still in common use. Perhaps that is because operating one does not demand the deftness needed for other derrick types, and they cannot always be outperformed by more technologically advanced devices.

The first step in the design of a stiffleg derrick installation is, of necessity, determining its mounting location. This is rarely difficult for a derrick situated on the ground. When the derrick is to be installed on a movable portal frame or tower, the additional feature of travel must be evaluated, but still the placement problem is usually not difficult.

An important consideration in orienting a stiffleg derrick is that the boom can operate only in the arc defined by its sills. That arc is usually a right angle (270°) but could be as much as 300°; the boom swing range would be limited to about 30° less. On most derricks the sill angles can be varied to some extent. When the derrick is to be shoehorned into a congested area, the ability to vary the sill angles can be a great advantage.

Mounting a stiffleg derrick atop a building, whether existing or under construction, is another matter altogether. Here the designer does not have complete freedom to place the derrick in the position most favorable for load handling, as other constraints can have an overriding influence. The installation cost for one derrick position can be markedly different from that for another. A derrick must be assembled and disassembled with the assistance of another lifting machine; the logistics of this operation, possibly performed with a mobile crane or a small gin pole, is a necessary part of the planning.

A stiffleg derrick passes its loadings to the host structure through vertical reactions at the mast base and sill ends (Figure 7.27). As it swings during operation, mast and leg end reactions change and actually reverse. Tie-downs are required at the three mounting points to prevent uplift. Compressive reactions, particularly under the mast, are of considerably greater magnitude than the lifted load. The designer must find a position where the necessary derrick operations can be performed efficiently while at the same time the structure can accommodate the loads. Often the mast is positioned directly above a

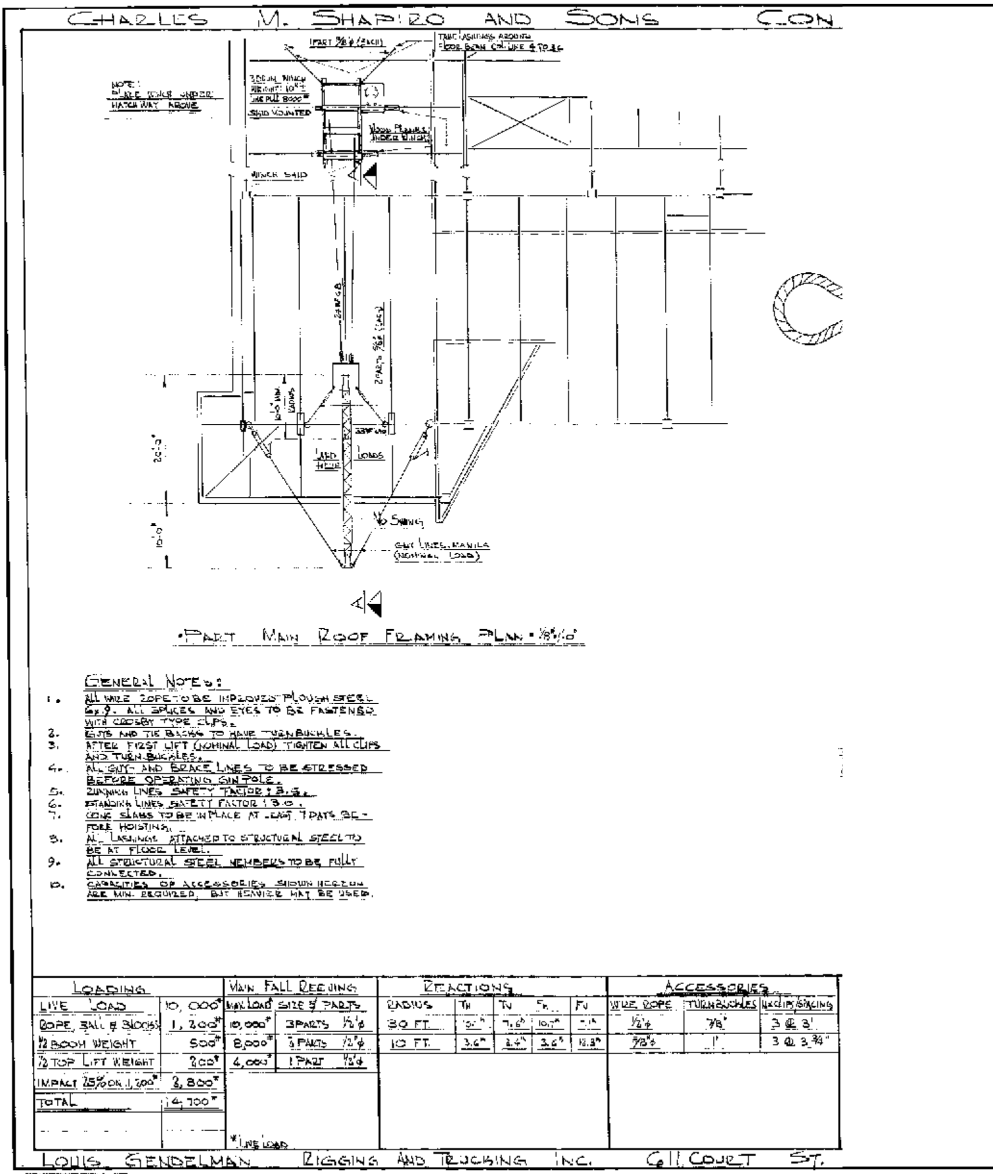
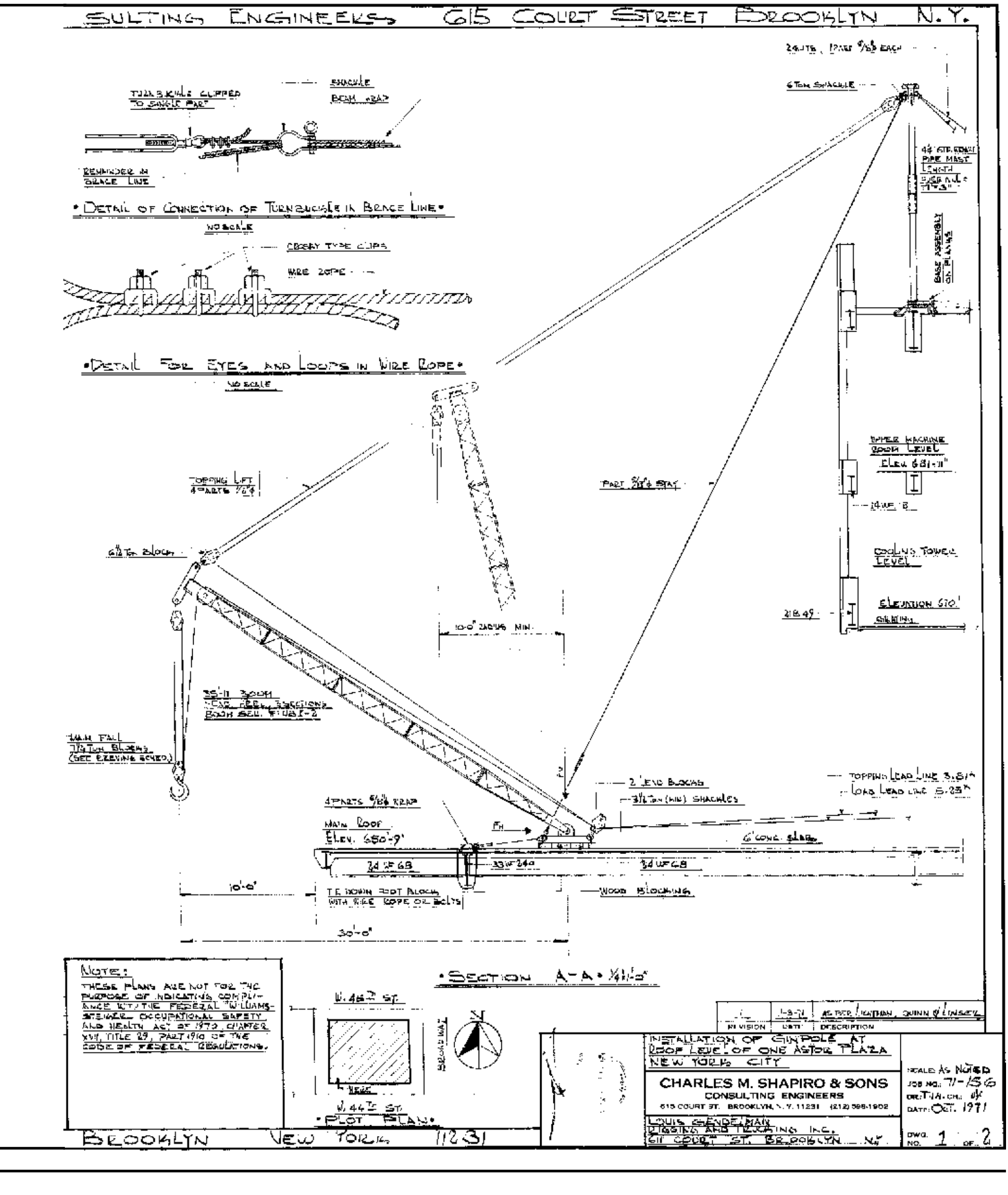


FIGURE 7.26 Installation drawing for a 5-ton (4.5-t) luffing gin pole to be placed on the roof of an office tower. The high topping lift gives the pole stability and permits anchorage of the side guys forward of the foot pivot pin.


SULTING ENGINEERS 615 COURT STREET BROOKLYN N.Y.
• DETAIL OF CONNECTION OF TURNBUCKLE IN BRACE LINE •
NO SCALE
•DETAIL FOR EYES AND LOOPS IN WIRE ROPE•
NO SCALE
CROSBY TYPE CLIPS
WIRE ROPE
6 TON SHACKLE
TOPPING LIFT
6 1/2 TON BLOCK
MAIN FALL
MAIN ROOF
ELEV. 650'-9"
10'-0" RADIUS MIN.
2 LEAD BLOCKS
TOPPING LEAD LINE
LOAD LEAD LINE
6" CONC. SLAB
24 WF 68
33 WF 240
24 WF 68
WOOD BLOCKING
TIE DOWN FOOT BLOCK WITH WIRE ROPE OR BOLTS
10'-0"
30'-0"
UPPER MACHINE ROOM LEVEL
COOLING TOWER LEVEL
ELEVATION 670'
•SECTION A-A• 1/4"=1'-0"
NOTE:
THESE PLANS ARE NOT FOR THE PURPOSE OF INDICATING COMPLIANCE WITH THE FEDERAL "WILLIAMS-STEIGER OCCUPATIONAL SAFETY AND HEALTH ACT OF 1970, CHAPTER XVII, TITLE 29, PART 1910 OF THE CODE OF FEDERAL REGULATIONS.
W. 46TH ST.
W. 44TH ST.
•PLOT PLAN•
REVISION
DATE
DESCRIPTION
INSTALLATION OF GINPOLE AT ROOF LEVEL OF ONE ASTOR PLAZA
NEW YORK CITY
CHARLES M. SHAPIRO & SONS
CONSULTING ENGINEERS
615 COURT ST. BROOKLYN, N.Y. 11231 (212) 596-1902
SCALE: AS NOTED
JOB NO.: 71-156
DATE: OCT. 1971
DWG. NO. 1 OF 2
BROOKLYN NEW YORK 11231

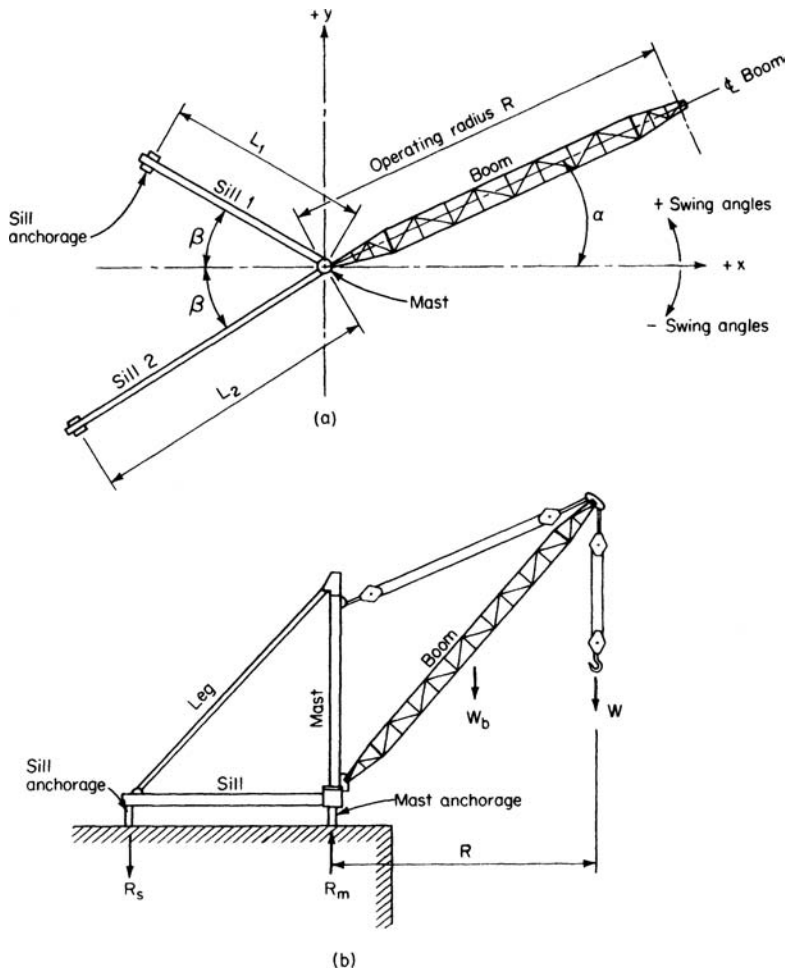


FIGURE 7.27 Installation of a stiffleg derrick: (a) Plan. (b) Elevation.

structure column as in Figure 7.28. Sometimes a column supporting the roof is too light to carry the derrick loads and it must be reinforced for some number of floors until the load path reaches a column of sufficient strength.

Sill dimensions seldom match building framing dimensions; even when the mast is placed on a building column framing, reinforcement or transfer beams are needed to support the leg ends. Just as many buildings have different column spacing in each direction, some derricks are made with sills that can adjust to different lengths or means can be devised for simple load transfer (Figure 7.29). Procuring a derrick that is well matched to the host structure is an

Figure 7.28 A rooftop stiffleg derrick. The mast is set directly above a column, but the two sills require large transfer beams. (*Budco Rigging Co.*)

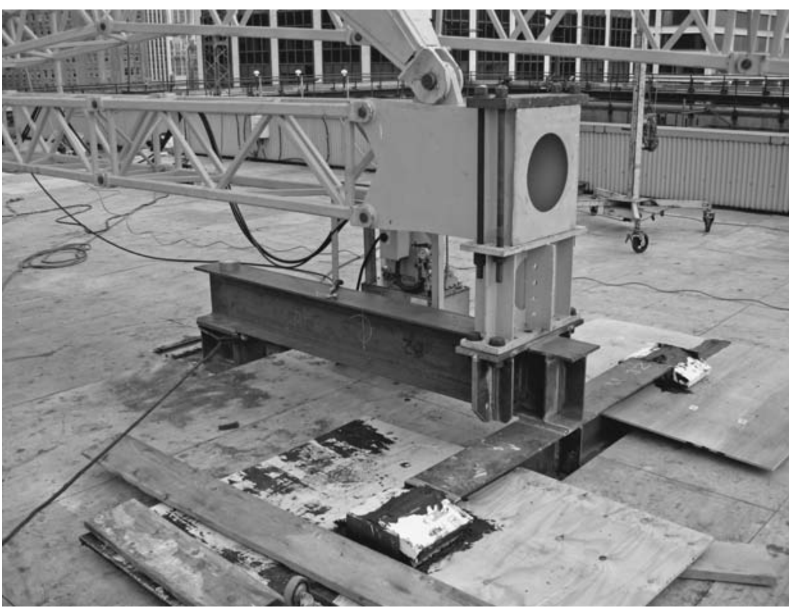


Figure 7.29 Sill end is connected to a relatively simple transfer that carries the load down to suitable building framing. (*Budco Rigging, Inc.*)

advantage. Field modifying the support structure to suit the derrick—installing heavy transfer framing or elaborate reinforcements of existing framing—can be costly. For new construction, planning and building anchorage points into the structure is an excellent strategy.

Horizontal reactions are also imposed on the host structure, but these are relatively small and manageable, especially in comparison to the vertical reactions. Lateral forces arise from hoist and topping lead lines, slewing, and wind forces.

From Figure 7.27, derrick reactions can be determined for any one boom position from the three equations of equilibrium, $\Sigma V = \Sigma M_x = \Sigma M_y = 0$. For the vertical forces, with W_b representing boom deadweight and W for the weight of the load.

$$R_{s1} + R_{s2} + R_m + W + W_b = 0$$

The reaction is positive when the anchorage point experiences tension. Negative values for the solved reactions will therefore indicate compression at the anchorage. If the load moment is taken as positive, for moments about the y axis.

$$M\cos\alpha - (R_{s1}L_1 + R_{s2}L_2)\cos\beta = 0$$

where $M = WR + M_b$ when M_b is the boom dead-load moment about the center of the mast. For the x axis.

$$M\sin\alpha + (R_{s1}L_1 - R_{s2}L_2)\sin\beta = 0$$

The three equations define discrete solutions for the three unknown reactions. In matrix form, after rearranging, these equations become

$$\begin{bmatrix} 1 & 1 & 1 \\ L_1 & L_2 & 0 \\ -L_1 & L_2 & 0 \end{bmatrix} \begin{bmatrix} R_{s1} \\ R_{s2} \\ R_m \end{bmatrix} = \begin{bmatrix} -W - W_b \\ \dfrac{M\cos\alpha}{\cos\beta} \\ \dfrac{M\sin\alpha}{\sin\beta} \end{bmatrix}$$

After solving by Kramer's rule and simplifying, this gives

$$R_{s1} = \frac{M}{L_1}\frac{\sin(\beta - \alpha)}{\sin 2\beta}$$

$$R_{s2} = \frac{M}{L_2}\frac{\sin(\beta + \alpha)}{\sin 2\beta} \tag{7.10}$$

$$R_m = \frac{-M}{\sin 2\beta}\left[\frac{\sin(\beta - \alpha)}{L_1} + \frac{\sin(\beta + \alpha)}{L_2}\right] - W - W_b$$

or

$$R_m = -(R_{s1} + R_{s2} + W + W_b)$$

Dead load from the derrick frame must be subtracted from each of the reactions given by Eqs. (7.10).

Most stiffleg derricks have sills set at a right angle; this simplifies the reaction equations because sin $2\beta = 1$. When $|\alpha| < \beta$, both sill supports will experience uplift and the mast support will carry compressive loading. When these angles are equal in magnitude, the reaction at the sill closest to the boom will be zero.

Figure 7.30 is a set of two drawings for the installation of a stiffleg derrick for removal of a tower crane and for completion of steel erection. The general layout is shown on Figure 7.30*a*, along with details of the arrangements needed to lead the hoisting and derricking lines from the derrick to the winch. The winch is mounted some 35 ft (10.7 m) below the derrick base. Details of the derrick mast and sill accommodations, support steelwork bracing, and building steel reinforcements are given on Figure 7.30*b*. Site conditions made it necessary for the sills of the derrick to be spread to 94°.

For any trial derrick position, the path of the major loads should be plotted through the work cycle to ascertain maximum reactions. These are some key conditions.

- Sill uplift is at a maximum with the boom set normal to the other sill.
- Sill bearing is at a maximum when the boom has reached its closest swing position to that sill.
- The maximum compressive reaction on the mast occurs when the boom is centered opposite the sills with $\alpha = 0°$, but only if the sills are of equal length.

If the sill lengths are not equal, the position inducing extreme loading can be found by differentiating the mast-reaction equation with respect to α. This new expression gives the rate of change in mast reaction as α changes, so equating it to zero and solving will give the maximum-value equation.

$$\frac{dR_m}{d\alpha} = 0$$

$$= \frac{M}{\sin 2\beta}\left[\frac{1}{L_1}(\sin\beta\sin\alpha + \cos\beta\cos\alpha) - \frac{1}{L_2}(\cos\beta\cos\alpha - \sin\beta\sin\alpha)\right]$$

This simplifies to

$$\alpha = \tan^{-1}\left(\frac{L_1 - L_2}{L_1 + L_2}\cot\beta\right) \tag{7.11}$$

The mast base will not always experience an uplift reaction, but when it does, it will be produced when $\alpha > 90°$.

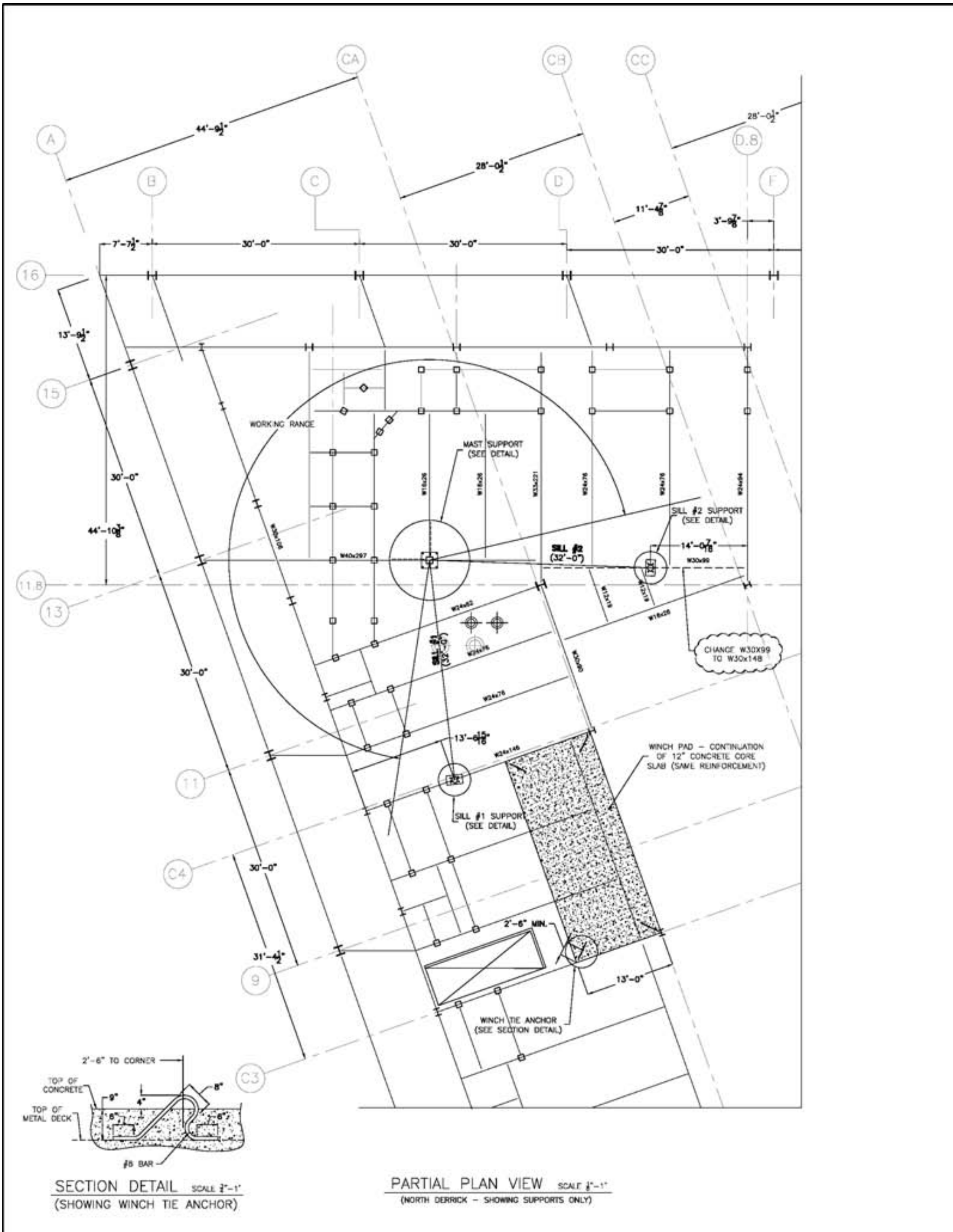
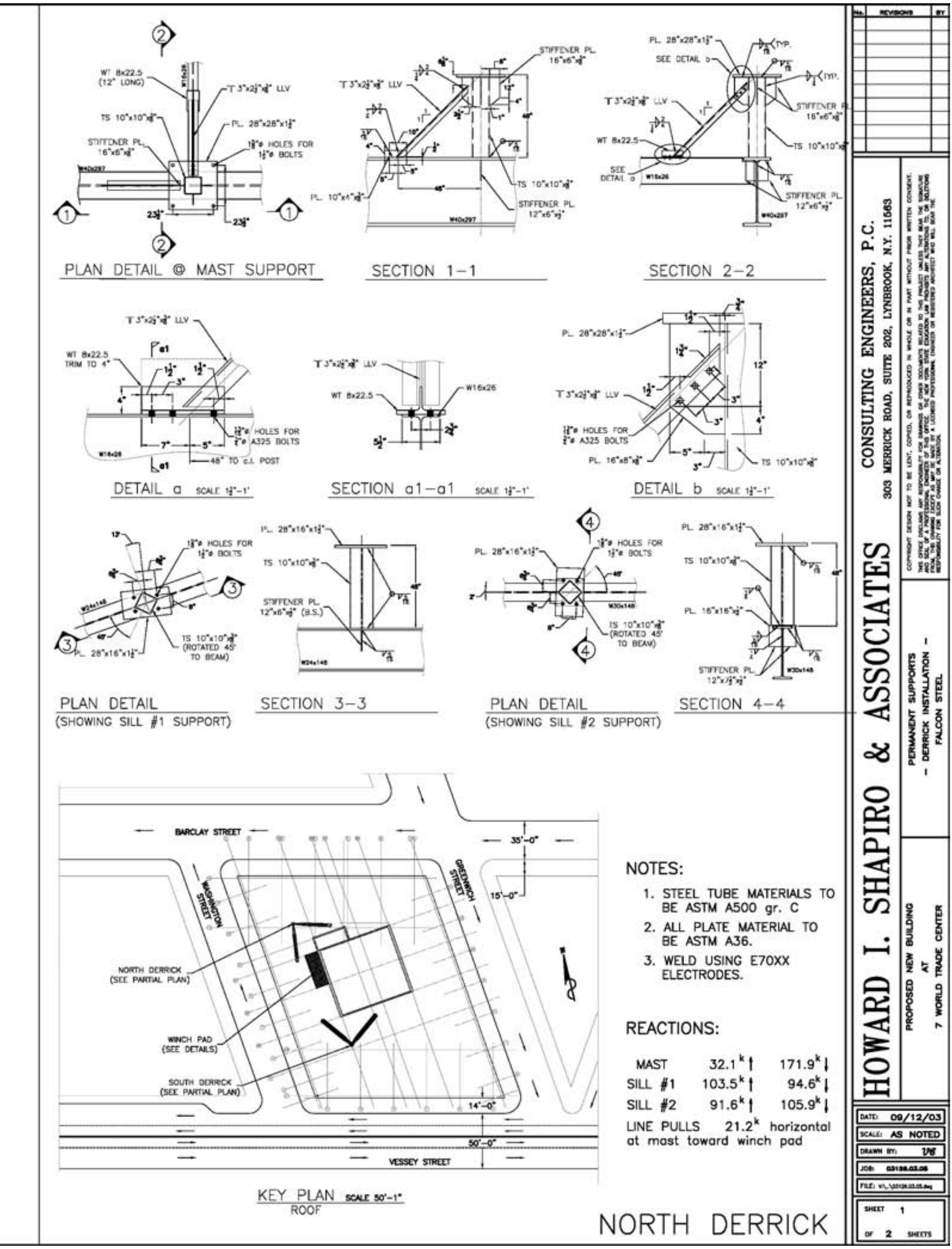


FIGURE 7.30 (a) The first sheet of the installation drawing for a stiffleg derrick placed to remove a tower crane from an office building roof. Pedestal details for mounting the derrick sills and mast are shown.


PLAN DETAIL @ MAST SUPPORT
SECTION 1–1
SECTION 2–2
DETAIL a SCALE 1½"–1'
SECTION a1–a1 SCALE 1½"–1'
DETAIL b SCALE 1½"–1'
PLAN DETAIL (SHOWING SILL #1 SUPPORT)
SECTION 3–3
PLAN DETAIL (SHOWING SILL #2 SUPPORT)
SECTION 4–4
BARCLAY STREET
WASHINGTON STREET
GREENWICH STREET
VESSEY STREET
NORTH DERRICK (SEE PARTIAL PLAN)
WINCH PAD (SEE DETAILS)
SOUTH DERRICK (SEE PARTIAL PLAN)
KEY PLAN SCALE 50'–1"
ROOF
NOTES:
1. STEEL TUBE MATERIALS TO BE ASTM A500 gr. C
2. ALL PLATE MATERIAL TO BE ASTM A36.
3. WELD USING E70XX ELECTRODES.
REACTIONS:
MAST 32.1k↑ 171.9k↓
SILL #1 103.5k↑ 94.6k↓
SILL #2 91.6k↑ 105.9k↓
LINE PULLS 21.2k horizontal at mast toward winch pad
NORTH DERRICK
HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
303 MERRICK ROAD, SUITE 202, LYNBROOK, N.Y. 11563
PROPOSED NEW BUILDING AT 7 WORLD TRADE CENTER
PERMANENT SUPPORTS – DERRICK INSTALLATION – FALCON STEEL
DATE: 09/12/03
SCALE: AS NOTED
SHEET 1 OF 2 SHEETS

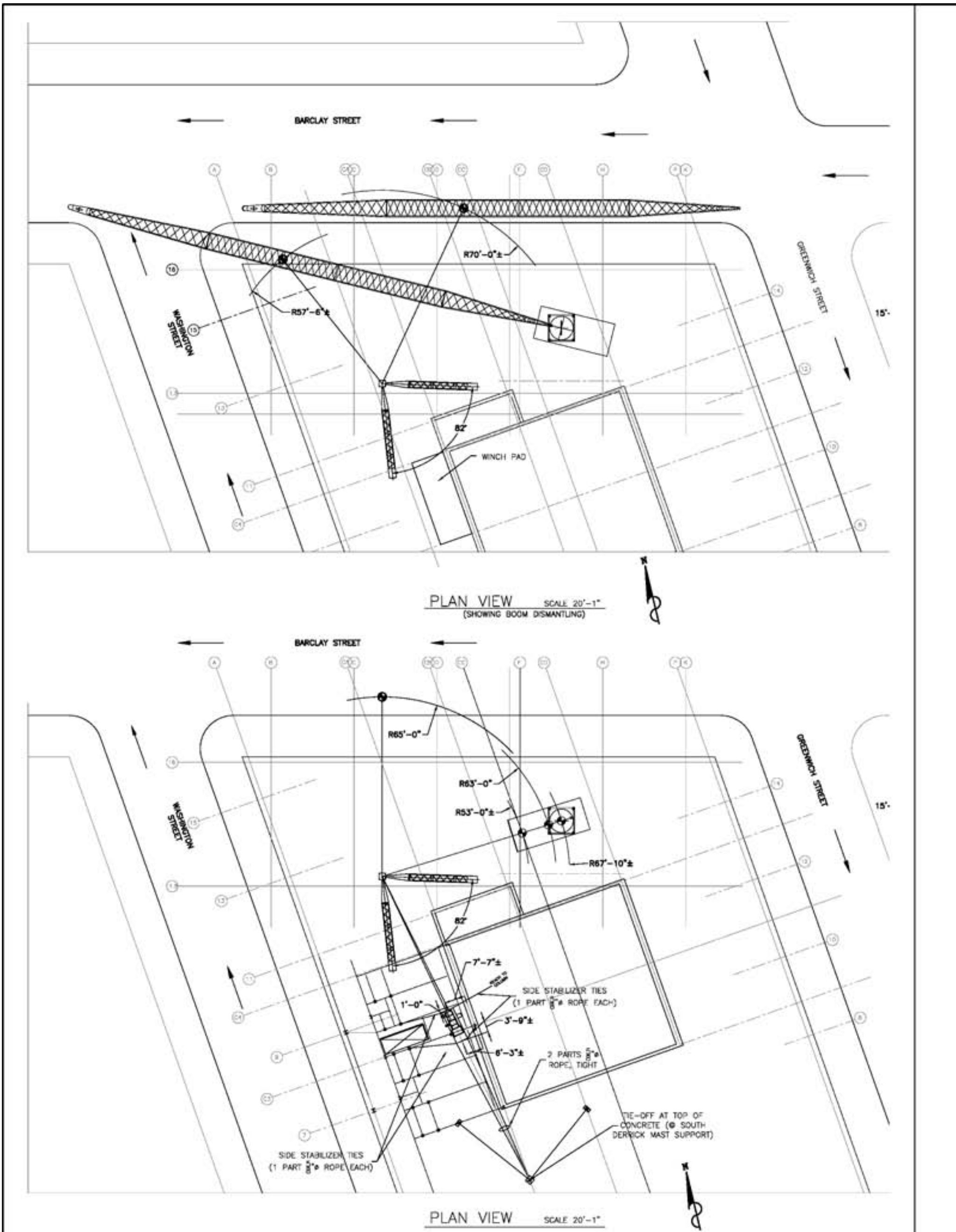
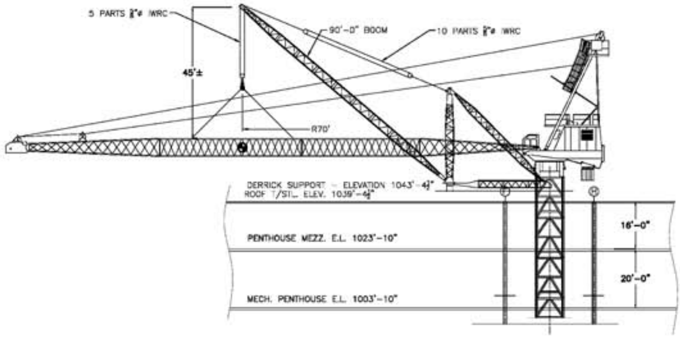
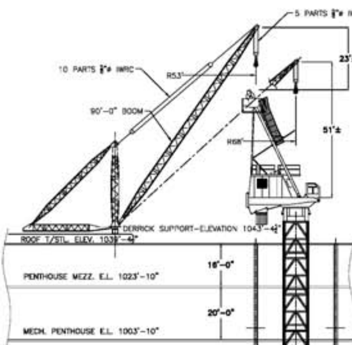


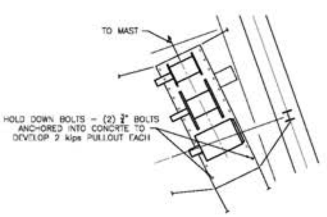
Figure 7.30 (b) This sheet shows the winch mounting and the procedure for dismantling the tower crane.



ELEVATION VIEW SCALE 20'-1"
(SCHEMATIC – SHOWING BOOM DISMANTLING)



ELEVATION VIEW SCALE 20'-1"
(SCHEMATIC – SHOWING OTHER COMPONENTS DISMANTLING)



PARTIAL PLAN VIEW SCALE 3/16"-1'
(SHOWING WINCH TIE–DOWN)

GENERAL NOTES

1. Contractor to verify all dimensions and site conditions in the field PRIOR to commencing with the work or the fabrication of steel components. Any errors, omissions or unusual field conditions to be reported to the Engineer immediately.
2. These installation and operation notes are for information and guidance and are not to be taken to infer that the Engineer is in any way responsible for, or involved in, the actual installation, operation or maintenance of the Derrick in the field.
3. The installation, operation and maintenance of the Derrick to be in accordance with ASME B30.6–latest edition, and applicable OSHA rules.
4. Do not operate the Derrick, when wind Speed at the roof exceeds 30 mph.
5. The operation of the Derrick may loosen bolts and fittings. Check bolts and fittings regularly for tightness.
6. Mast to be plumb and packed out to level, as required.
7. See drawing 1 (job# 03126.03.05 dated 09/12/03) for derrick support structures.

DISMANTLING OF NORTH FMC TG-1900T TOWER CRANE
USING 35 TON TIMBERLAND STIFFLEG DERRICK
AT
7 WORLD TRADE CENTER, NEW YORK, NY
FOR
FALCON STEEL COMPANY

SEQ.	ITEM	WEIGHT	PICK RADIUS (FT.)	LANDING RADIUS (FT.)	DERRICK RATED LOAD[1]
1.	180' boom	26,620 lbs.	58	70	38,000lbs.
2.	"A" Frame	24,140 lbs.	55	65	42,000lbs.
3.	(3) Moving Counterweight	23,600 lbs. (each)	55	65	42,000lbs.
4.	Fixed Counterweight	23,700 lbs	55	65	42,000lbs
5.	Machinery house complete	4,335 lbs.	63	65	42,000lbs
6.	Power package	11,070 lbs.	60	65	42,000lbs
7.	Winch Package (with main drum, 1000 ft. wire rope, ½ fuel tank)	33,660 lbs.	60	65	42,000lbs
8.	Operator's cab	1,650 lbs.	60	65	42,000lbs
9.	Machinery Deck (with turntable bearing, catwalks, counterweight trolley & link)	35,640 lbs.	63	65	42,000lbs
10.	Swing drives	2,320 lbs.	68	68	39,600lbs.
11.	Roballo Mount	16,500 lbs.	68	68	39,600lbs.
12.	40' Tower section	31,000 lbs.	68	68	39,600lbs
13.	13'-7 ½" Tower Sec.	11,670 lbs.	68	68	39,600lbs.
14.	Support Frame (in pcs.) (2) center beams, each (2) end beams, each	18,900 lbs. 23,500 lbs.	68 75	68 68	39,600lbs. 35,500lbs.
15.	Climbing Frame 2 parts	23,400lbs. Ea.	68	68	39,600lbs.
16.	(2) Joined 13'-7 ½" Tower Sec.	23,340lbs.	68	68	39,600lbs.
17.	(1) 13'-7 ½" Tower Sec. + (1) 6'-9 ¾" Tower sec	5,835+ 11,670 = 17,505 lbs.	68	68	39,600lbs.
18.	Support Frame (in pcs.) (2) center beams, each (2) end beams, each	18,900 lbs. 23,500 lbs.	68 75	68 68	39,600lbs. 35,500lbs.

NOTES:
1. Rated load is at the greater of the "pick" or "landing" radius.

Maximum Hook Load :	35.6k
Ball and Block etc.:	1.5
Hoist Rope :	2.8
20% Impact Factor :	7.7
Half Boom and Luff :	5.6
TOTAL =	52.9k

Winch Data :	
Hoist Line Pull :	10.2k
Luff Line Pull :	12.8k
Total Line Pull =	23.0k
Max Fleet Angle :	1.5
Weight of Winch :	13.9k

Load Reactions (Including 20% Impact Factor)

	Under the Derrick		
	Mast	Sill #1	Sill #2
Downthrust	179.9	94.2	105.9
Uplift	4.3	103.5	87.6

35 Ton Stiffleg Derrick
C.D.# C.N.#
Mast: 34'-0" Sills: 32'-0"
Boom: 90'-0"
Max. Hook Load: 35,640lbs. at 66ft Radius Max.

No.	REVISIONS	BY
1.	REV.1 02-11-05	[illegible]

HOWARD I. SHAPIRO & ASSOCIATES
CONSULTING ENGINEERS, P.C.
303 MERRICK ROAD, SUITE 202, LYNBROOK, N.Y. 11563

PROPOSED NEW BUILDING
AT
7 WORLD TRADE CENTER

DERRICK INSTALLATION
TOWER CRANE REMOVAL
FALCON STEEL

DATE: 09/12/03
SCALE: AS NOTED
DRAWN BY: [illegible]
JOB: 03126.03.06
FILE: [illegible]
SHEET D-1
OF 1 SHEETS

FIGURE 7.31 *Bull wheel* used to swing a stiffleg derrick. It is driven by a chain connecting the bullwheel to a smaller sprocket driven by a hydraulic motor. The footblock, mounted directly to the top of a building column, has a pair of sheaves to lead the load hoist and boom hoist lines. (*Budco Rigging Co.*)

When swing provisions are not built into the derrick itself (Figure 7.31), the installation designer must give consideration to the means for providing the swing functions and supporting the associated horizontal reactions.

Example 7.3

A stiffleg derrick will be used to lift loads of 25 kips (11.3 t) at a radius of 55 ft (16.8 m) with the boom at $\alpha = 90°$. After raising the load and booming in to 25-ft (7.6-m) radius, the loads will be swung to $\alpha = -120°$ and set in place. The sills are each 20 ft (6.1 m) long and have dead-load reactions of 2 kips (8.9 kN) each, while the mast dead-load reaction is 3.5 kips (15.6 kN). The boom deadweight can be taken as 3.2 kips (1.5 t) acting at 1/2 the radius. Ignoring wind effects and impact for this preliminary study, find the maximum anchorage reactions when $\beta = 30°$.

Solution

The boom will not swing from the 55-ft -radius (16.8-m) position, so each support reaction must be calculated for this condition ($\alpha = 90°$). Moment from load and boom will be

$$M = WR + M_b = (25\text{kips})(55\text{ ft}) + (3.2\text{ kips})(1/2)(55\text{ ft})$$

$$= 1463\text{ kip}\cdot\text{ft } (1984\text{ kN}\cdot\text{m})$$

Using Eq. (7.10) and subtracting the dead-load reactions gives

$$R_{s1} = \frac{1463}{20}\frac{\sin(30^\circ - 90^\circ)}{\sin[2(30^\circ)]} - 2.0 = -75.15 \text{ kips } (-334.3 \text{ kN})$$

$$R_{s2} = \frac{1463}{20}\frac{\sin(30^\circ + 90^\circ)}{\sin 60^\circ} - 2.0 = 71.15 \text{ kips } (316.5 \text{ kN})$$

$$R_m = (-75.15 + 71.15 + 25 + 3.2) - 3.5 - 2.0 - 2.0$$
$$= -31.70 \text{ kips } (-141.0 \text{ kN})$$

After booming in to 25 ft (7.6 m) radius, the maximum sill uplift will occur with the boom in two positions, $\alpha = 60^\circ, -60^\circ$, so that both sill supports must be designed for this force. The moment now becomes

$$M = 25(25) + 3.2(1/2)(25) = 665 \text{ kip}\cdot\text{ft } (902 \text{ kN}\cdot\text{m})$$

$$R_{s1} = R_{s2} = \frac{665}{20}\frac{\sin(30^\circ + 60^\circ)}{\sin 60^\circ} - 2.0 = 36.39 \text{ kips} (161.9 \text{ kN})$$

The maximum compressive force at sill 2 will develop when the boom comes closest to that support, or when $\alpha = -120^\circ$.

$$R_{s2} = \frac{665}{20}\frac{\sin(30^\circ - 120^\circ)}{\sin 60^\circ} - 2.0 = -40.39 \text{ kips} (-179.7 \text{ kN})$$

The first position checked indicated the maximum compressive condition for sill 1.

Using Eq. (7.11), we find the horizontal boom angle at which the mast receives maximum compressive loading.

$$\alpha = \tan^{-1}\left(\frac{20-20}{20+20}\cot 30^\circ\right) = 0^\circ$$

This will always be the case with sills of identical length. Then, from Eq. (7.10) we have

$$R_m = \frac{-665}{\sin 60^\circ}\left[\frac{\sin(30^\circ - 0^\circ)}{20} + \frac{\sin(30^\circ + 0^\circ)}{20}\right] - 25 - 3.2 - 7.5$$
$$= -74.09 \text{ kips } (-329.6 \text{ kN})$$

After swinging to $\alpha = -120^\circ$, the boom will be at its maximum incursion on the sill side of the y axis. The mast base may then be in tension; we find

$$R_m = -\left[\frac{665}{20}\frac{\sin(30^\circ + 120^\circ)}{\sin 60^\circ} - 40.39 + 25 + 3.2\right] - 7.5$$
$$= -14.51 \text{ kips } (-64.5 \text{ kN})$$

which is a compressive force after all.

The design reactions are then

Unit	R_{s1}	R_{s2}	R_m
kips	–75.15, + 36.39	–40.39, + 71.15	–74.09
kN	–334.3, + 161.9	–179.7, + 316.5	–329.6

7.6 Other Derrick Forms and Details

The most basic form of lifting device, commonly used in manufacturing plants, service shops, and construction sites, is a self-contained hoisting unit which is available in capacities of from about 0.50 to 40 tons (0.45 to 36 t) or more. These hoists are powered electrically or pneumatically or made for manual operation by means of a hand chain; the manual units are often referred to as chain-falls (Figure 2.2).

When used for vertical lifting, the units are merely suspended from any structurally adequate overhead support. The load on the support will include only rated load and the weight of the hoist for manual units because lifting motion is too slow to make dynamic effects significant. On the other hand, powered hoist supports should include an impact allowance of 25% or more if the manufacturer so advises.

Similar lifting functions can be performed by a set of blocks reeved to a winch (Figure 7.32), a job-made device that can be

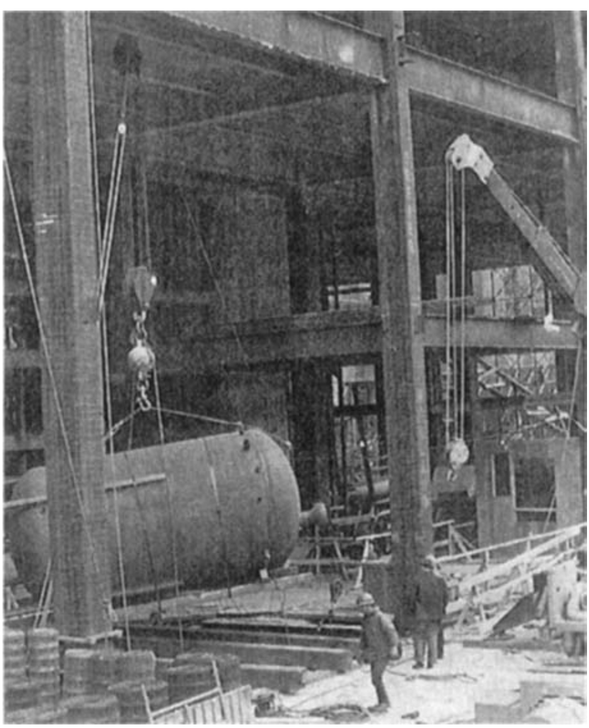


Figure 7.32 One of the most basic lifting arrangements, a shaft hoist. The load was rigged under the hook by rolling it over the timbers. With the load raised, the timbers will be moved aside and the load lowered to a subcellar several stories below the street. (*Lawrence K. Shapiro.*)

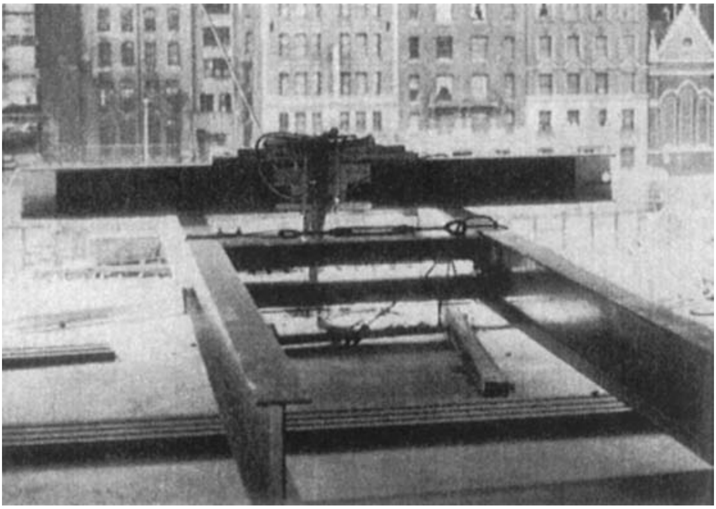


Figure 7.33 Support arrangement for the shaft hoist in Figure 7.32. The long beams are blocked up at their ends so that they can deflect freely under load. At the center, they are tied in against timber separators for lateral torsional stability. The load block is suspended from the crossbeam. (*Lawrence K. Shapiro.*)

arranged for any capacity. The overhead support will carry the same loads as for a powered hoist, but when the lead line runs out of the upper block, its effect must be added.

Figure 7.33 shows the support arrangement used for the hoist in Figure 7.32. Note that the long hoist-support beams will elastically deflect when loaded and for this reason were blocked up clear of the concrete floor. Without this precaution, the lightly framed office building floor probably would have cracked.

When a lead line runs off at an angle to the block, it introduces a lateral force component at the block that could cause lateral load displacement when a few parts of line are being used. Likewise, the line directions at a snatch block define the alignment of the anchorage reaction (Figure 7.34) which is needed for design. With the lead-line directions defined at the snatch block, the lead-line-angle effect at the block can be determined with sufficient accuracy.

Using the notation of Figure 7.35, the vertical load on the pivot point will be $W + P \cos \theta$ if W is taken to include the weight of the rigging. The upper block will be displaced laterally through distance Δ by the lead-line effect, and this will introduce a clockwise moment about the pivot point of

$$M = \Delta(W + P\cos\theta)$$

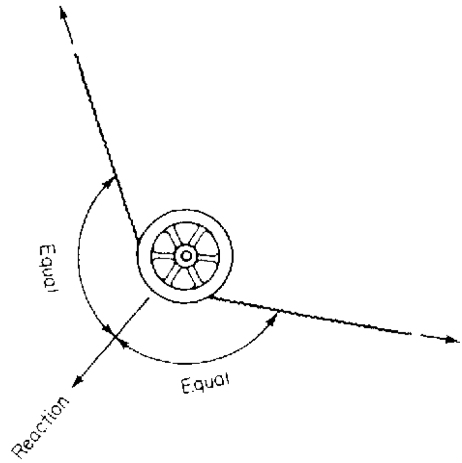
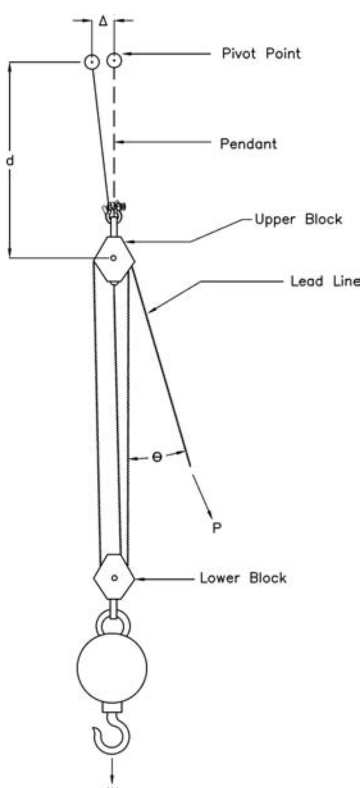


FIGURE 7.34 The line of action of the reaction at a block anchorage is a function of the loaded-rope directions.

FIGURE 7.35 Factors needed to calculate the lateral displacement of the upper block caused by the offset angle of the lead line.



Equilibrium will be maintained by the counterclockwise moment

$$M = dP\sin\theta$$

Solving for the displacement, we get

$$\Delta = \frac{dP\sin\theta}{W + P\cos\theta} \tag{7.12}$$

The same effect takes place when the lead line runs up from the lower block, but the condition is exacerbated as the load is lowered and the distance from the pivot point increases. Absent many parts of line, the load will severely displace and could cause clearance problems unless the lead angle is kept small.

Example 7.4

1. A load of 45 kips (20 t), including the weight of the rigging, is to be hoisted on six parts of line. The sheave shaft of the upper load block is 82 in (2.08 m) below the pivot at the suspension point, and the lead line runs from the upper block at 30° from the vertical. What distance will the load be displaced laterally from the pivot position?

Solution

If friction is neglected, the line pull will be 45/6 = 7.5 kips (33.4 kN). Using Eq. (7.12), we have

$$\Delta = \frac{82(7.5\sin 30°\ )}{45 + 7.5\cos 30°} = 6.0 \text{ in (152 mm)}$$

2. Given the same load and rigging but with the lead running from the lower block at 30° (and included as one of the parts), what offset will have developed when the lower block has reached 30 ft (9.1 m) below the pivot?

Solution

Again neglecting friction, with the lead as one of the parts the lead-line load will be given by

$$5P + P\cos\theta = W$$

$$P = \frac{45}{5 + \cos 30°} = 7.67 \text{ kips}(34.1 \text{ kN})$$

which when inserted into Eq. (7.12), gives the displacement.

$$\Delta = \frac{30(12)(7.67\sin 30°)}{45 + 7.67\cos 30°} = 26.7 \text{ in}(679 \text{ mm})$$

The lateral displacement will reduce the lead-line angle, making the calculated value an overstatement; the actual displacement will be about 2½ in (65 mm) less.

Further, the lead-line angle will change as the lower block is raised or lowered, making it necessary to do the calculation, using applicable lead angles, at each critical clearance point.

To reduce load displacement when the lead is from the lower block, it is necessary to mount the snatch block close to the pivot, in other words, to make the lead line near-vertical. But when the lead must come from the top block, an ample lead angle must be established for the lead line to clear the load. If displacement is critical or if the block is suspended well below the pivot, the problem can be overcome by fixing the block in place with stabilizing lines run laterally from the block bail.

Catheads

Another common basic form of lifting device is the cathead, which is nothing more than a cantilevered beam supporting an upper block at its outer end. When catheads are installed at construction sites, the beam is made to project well into the building to make erection and anchorage easier. The beam is locked into place either by clamping onto floor beams below or by wedging posts between the cathead and floor beams above. Another but less satisfactory method uses counterweights. This makes the capacity depend on stability imparted by the weights, with a great deal of uncertainty introduced because of the unknown value of the impact accompanying lifting and unloading.

Anchorage of the cathead is a two-point support, one at or near the building edge and the other inside (Figure 7.36). Each support must carry on increment of vertical loading from the lifted load and

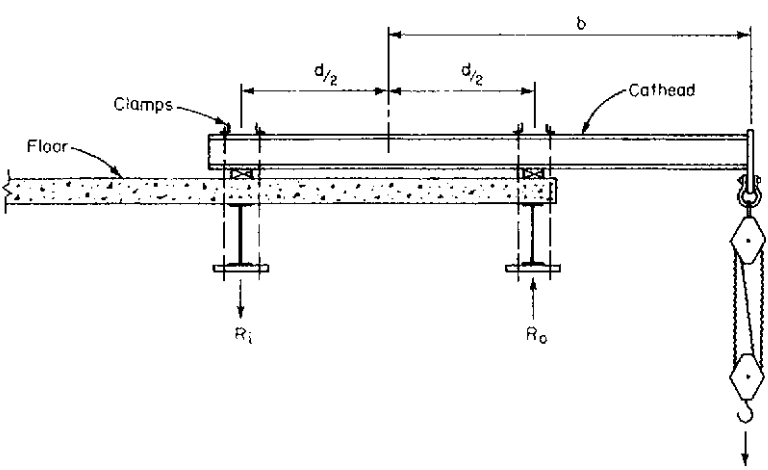


Figure 7.36 Basic cathead arrangement.

cathead weight plus the effect of load moment. Taking support uplift as positive, we find the reactions R form.

$$R = W\left(-\frac{1}{2} \pm \frac{b}{d}\right) \tag{7.13}$$

where the positive value is used for R_i and the negative for R_o.

Bending moment and floor reactions can be partly relieved by suspending the end of the beam from a pendant (Figure 7.37), but the arrangement introduces horizontal thrust into the cathead.

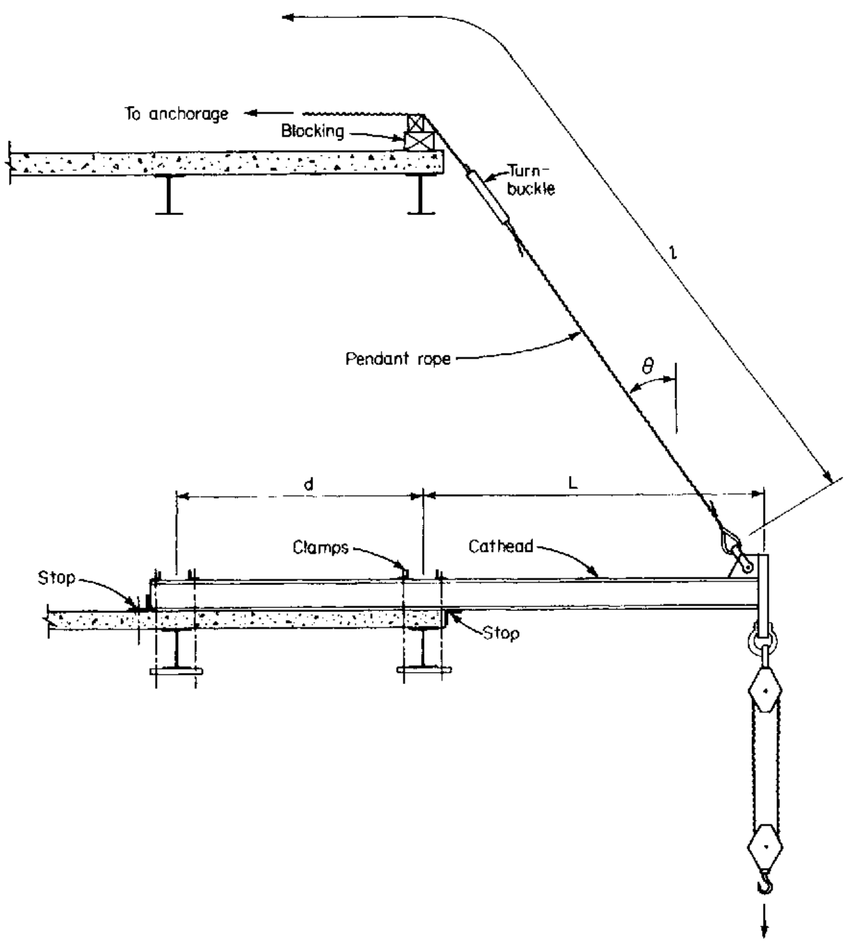


Figure 7.37 Pendant-supported cathead.

The pendant rope acts as a spring. For rope load P_r, the vertical component of rope stretch will be

$$\Delta = \frac{P_r l \cos\theta}{A_r E_r} \qquad (\boldsymbol{a})$$

and for cantilever-beam load P_c, the tip vertical deflection will be

$$\Delta = \frac{P_c L^2}{3EI}(d+L) \qquad (\boldsymbol{b})$$

where I = moment of inertia of beam cross section
E = modulus of elasticity of beam material
A_r = metallic area cross section of rope
E_r = rope modulus of elasticity

The vertical loads must be in equilibrium, as expressed by

$$P_r \cos\theta + P_c = W$$

Equating the deflection relationships (***a***) and (***b***), we have a second equation in the two unknowns. In matrix form.

$$\begin{bmatrix} \cos\theta & 1 \\ \dfrac{l\cos\theta}{A_r E_r} & \dfrac{L^2(d+L)}{-3EI} \end{bmatrix} \begin{bmatrix} P_r \\ P_c \end{bmatrix} = \begin{bmatrix} W \\ 0 \end{bmatrix}$$

After solving by Kramer's rule and simplifying, we get

$$P_r = \frac{W}{(1+1/e)\cos\theta} \qquad P_c = \frac{W}{1+e} \qquad (7.14)$$

where

$$e = \frac{A_r E_r L^2 (d+L)}{3EIl}$$

As the beam moment of inertia is reduced, the parameter e approaches infinity in the limit and the rope load becomes $W/\cos\theta$. This indicates that the cathead can be designed as a beam-column loaded with the thrust induced by the inclined pendant and a secondary moment that will be limited to the product of the thrust and the vertical displacement due to rope stretch. This moment will be

$$M = \Delta_{\text{rope}} P_r \sin\theta = \frac{P_r^2 l \sin\theta\cos\theta}{A_r E_r} = \frac{W^2 l \tan\theta}{A_r E_r} \qquad (7.15)$$

It is assumed, of course, that the design has included the provision that the centerline of the pendant intersect the line of action of the vertical load at the centerline of the cathead. If this is not the case, an additional end moment will be induced by the eccentricity.

Flying Strut

The next logical evolutionary step to follow the pendent-supported cathead is the flying strut. In the previous form of cathead the clamp attachments to the floor served only to stabilize the device and hold it in place. With the load-transfer function of the clamps done away with, the floor is no longer an essential element in the installation.

The flying strut is a pendant-supported cathead held suspended in space (Figure 7.38). It is held in position by the load fall, pendants, and several wire-rope thrust-restraining, and preventer lines. The number and positions of the lines are tailored to the individual installation and are arranged to transfer the strut thrust, derived from the inclined pendant, to the building frame and to assure that the strut will remain stable through any expected maneuvering of the load. It can easily be arranged to accommodate drifting of the load (out-of-plumb load falls) in any direction.

The flying strut is a pure pin-ended column subjected to axial loading plus some end moment introduced by imbalance in thrust-line loadings or tip eccentricity. In addition, side drifting of loads will impose torsional loading. Tube or pipe sections therefore make the most efficient strut members.

When adjustment capabilities are built into the pendant and other wire-rope lines, the strut can be floated laterally, vertically, or forward and backward to secure the most advantageous position (Figure 7.39). With suitable arrangements, this can even be done under load.

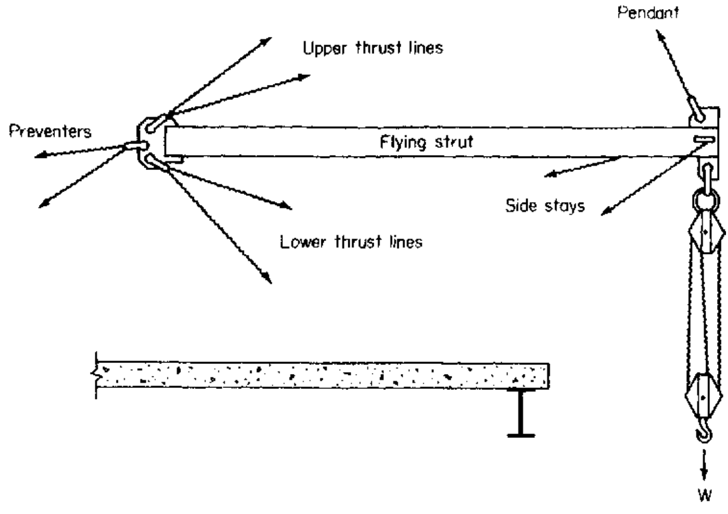


FIGURE 7.38 Flying strut.

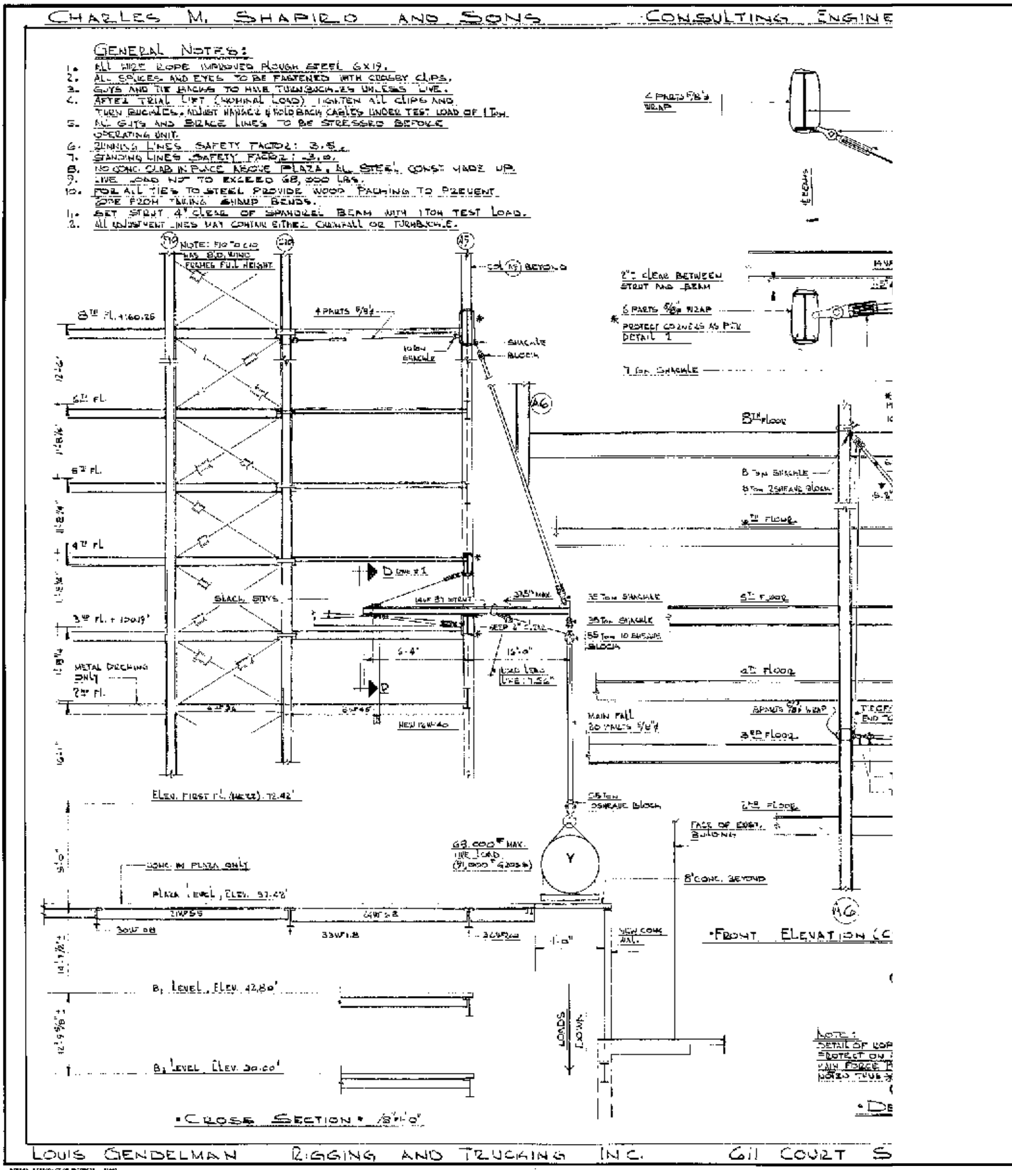
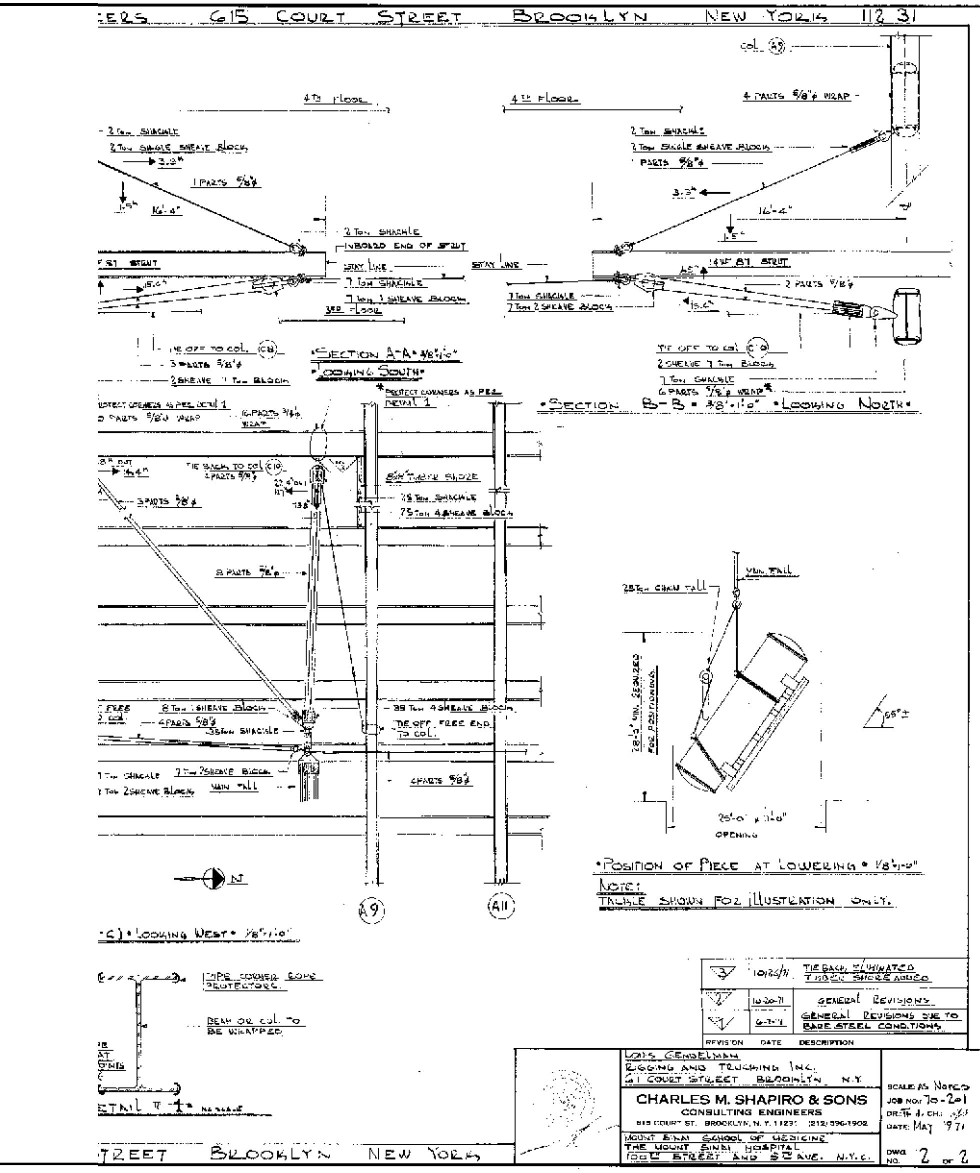


FIGURE 7.39 Installation drawing for a flying strut used for drifting a load some 18 ft (5.5 m) from the pickup point to its freely suspended lowering positions.


615 COURT STREET BROOKLYN NEW YORK 11231
•SECTION A-A• 3/8"=1'-0" •LOOKING SOUTH•
•SECTION B-B• 3/8"=1'-0" •LOOKING NORTH•
•POSITION OF PIECE AT LOWERING• 1/8"=1'-0"
NOTE:
TACKLE SHOWN FOR ILLUSTRATION ONLY.
•LOOKING WEST• 1/8"=1'-0"
CHARLES M. SHAPIRO & SONS
CONSULTING ENGINEERS
LOUIS GENDELMAN RIGGING AND TRUCKING INC.
MOUNT SINAI SCHOOL OF MEDICINE
THE MOUNT SINAI HOSPITAL
REVISION DATE DESCRIPTION
GENERAL REVISIONS
SCALE AS NOTED
JOB NO. 70-201
DATE MAY 1971
DWG NO. 2 OF 2

7.7 Loads Acting on Derricks

Loads act on derricks much the same as they act on other lifting appliances. But just as operating environments and characteristics of mobile cranes are distinct from tower cranes in ways that affect loads imposed, derricks have a different set of circumstances.

In derrick applications, overhaul height and the weight of rope suspended from the boom tip vary over a wide range. A derrick mounted on the ground has a small lift height with a correspondingly small amount of rope to pay out on the load fall, but on a high-rise roof top there could be thousands of feet of rope suspended from the boom tip making up a significant part of the lifted weight. Controlling factors for overhaul weight—the number of parts-of-line and the relative elevation of the winch—also differ from installation to installation. In derrick practice, therefore, both the overhaul weight and the hoist ropes below the upper block are considered as part of the suspended load and must be deducted from the load chart to get a net lifting capacity.

Blanket guidance for assessing wind effects on derricks is elusive. In most situations the effect of wind on derrick anchorages is not significant and the design would be adequate had wind been ignored. For some installations, tall gin poles and guy derricks, for example, wind can be an important factor both during operation and out of service. For some other types, the boom or mast might readily be taken out of harm's way in advance of a storm. The easiest advice to give, of course, is that wind effects should always be included. Though an experienced derrick-installation designer could apply some discretion, for others it is advisable at least to perform a rough evaluation of wind before excluding its effects.

Wind on the lifted load may be another matter, however, and operational planning had best include a check on wind action. This is particularly true when selecting boom tip side guys and swing or tagline arrangements. It is of crucial importance in planning procedures for loads that must be accurately spotted; otherwise, in many instances, work may have to be suspended at wind velocities well below the 30 mi/h (13.4 m/s) maximum normally permitted for derrick operations.

Taglines impart loads on derricks, too, adding both the weight of the lines and holdback forces. While lifting loads that present large sail surfaces to the wind, the taglines might be restrained by snubbing on a capstan or winch head, or perhaps even on the bumper of a truck. The relative effect of these loads is most significant for high lifts on low-capacity derricks. If a pair of taglines is not connected far enough to each side of center of the load, the load may twist. If the taglines are not angled sufficiently outboard and flanking the load, it may pendulate. These deficiencies can cause dynamic oscillation and loss of control.

Impact loading affects all installations; how much so can be a matter for deliberation. For general lifting service, the FEM impact recommendations are incorporated in the expressions of Eq. (3.5). The AISC suggests 20% for steel erection, a figure that is a reasonable albeit simplistic allowance for most applications. Particular maneuvers such as initial liftoff or transfer of the load, if not performed gingerly, can induce unusually high impact. These instances might elude the designer if the planning does not include a step-by-step operational review. Impact as a percentage of the lifted load can be greater, also, if the derrick or winch is oversized.

7.8 Winch Installation

In locating a winch for a derrick, several considerations come into play

- Access to be able to place the winch and continued access again for its removal. While a trivial problem for an open site, it can be a daunting concern for a location within a building.
- Availability of power for an electric winch or, alternatively, a suitable environment for dissipating the fumes and noise produced by internal combustion.
- Space to develop a suitable fleet angle.
- Isolation from unauthorized access; rotating drums and moving ropes can be hazardous.
- Good sight lines, though blind operation is permissible with adequate signaling.
- Means to support the loads imposed by the winch; these loads include both weight and lateral effects caused by the line pull of the ropes coming off the drums.

Winches commonly used with construction derricks weigh about 10,000 to 20,000 lb (4.5 to 9 t). On a building floor, spreader beams or timber mats are often needed to distribute the weight (Figure 7.40). For ground mounting, a ballast pad may be poured to spread the weight and resist lateral forces.

As the loads on the lead lines are of the same order as the winch weight, friction between the base and the ground is insufficient to resist the lead forces of the ropes coming off the drums. In traversing back and forth across the drums, by chance two or more falls can wind off the same side at the same time, inducing twisting moment that could move the winch into misalignment. Furthermore, the vertical offsets of the line loads above the anchorage induce moment that could lift the rear end. To counter all these effects, the winch skid may be bolted to a concrete floor or welded to steel framing. Very often, however, it is more practical to tie the winch at its rear and sides with wire ropes. The ropes are made fast to framing members or other anchor points with

FIGURE 7.40 Dual drum winch designed for derrick work on existing buildings. Its weight is spread out on the roof by steel dunnage. Narrow drums allow the winch to be positioned close to the footblock. This winch breaks down into pieces that are small enough for most stairwells, doorways and freight elevators. (*Budco Rigging Co.*)

turnbuckles used to snug up the lines. When ropes are used for this purpose, they must be monitored to assure that they remain snug.

How the lead lines exit the base varies among derrick types. On a steel erector's guy derrick the lines run straight down through the bottom of the base. A stiffleg derrick usually has them running out horizontally, bisecting the angle between the derrick legs. But the path between the exit point and the winch is not always a straight line; obstacles or winch placement may dictate otherwise. Deflector sheaves are then installed to direct the lead lines on the circuitous route. The sheaves should be positioned so that when the ropes are loaded, the ropes lead squarely into the grooves. When the alignment of the lead rope changes, as when a Chicago boom swings, the deflectors must be mounted to follow the swing.

Anchorages for deflector sheaves must accommodate lead-line forces with consideration for the incoming and outgoing rope directions. The mountings also must be designed to keep the rope on the sheaves and the sheaves in position when unloaded. Figure 7.30*a* and *b* shows two sets of deflector sheaves and their mountings that were used to carry lead lines from a stiffleg derrick to a winch placed more than 35 ft (10.7 m) below. Figure 7.5 includes a deflector arrangement used with a Chicago boom.

CHAPTER 8
Controlling Risk

Lean and mean is a description that implies a nimble and aggressive business entity devoid of excess management. A company of this character moves resolutely always in search of opportunities, seemingly stopped by nothing. There was a time when such an organization could accept an occasional accident—even fatalities of workers—as a cost of doing business to be dealt with primarily by insurance. These times are past. A lean and mean warrior must act differently than in the past or stand little chance of enduring over the long term.

When a serious crane or rigging accident takes place, tort liability may be only the beginning of woes. Other attendant effects of an accident now may tie down a company so severely that being nimble is no longer possible. In the age of the Internet and search engines, a reputation can be instantly sullied, fairly or unfairly, worldwide and for all time. A government entity may react to an accident by shutting down an operation until investigation is complete and perceived risks are removed. Violations and fines, at one time a minor nuisance, can now be a big drain on resources. A record of accidents and violations can hamper a company's ability to obtain insurance, hold licensing essential for operations, or to be short-listed by blue chip customers for new work. Prosecution is even possible. Avoiding such an ending may induce a company to go to great lengths putting up a defense against both litigants and government enforcers, hiring the best, also the most expensive, attorneys and in the process diverting the attention of top managers from the work they should be doing.

Though the consequences of an accident can be severely damaging to companies and the people who run them, staying clear of risks by practicing extreme measures of avoidance is a sure strategy of self-extinction. There must be an acceptance that risk is always present, reducible but never to be extinguished. Risk is to be recognized, quantified, contained, and managed.

An effective risk management strategy should have a compound set of benefits.

- Reducing the direct cost of accidents in lives, injury, and property damage.

- Reducing the attendant costs such as project delays, damaged reputations, and diversion of company resources.
- Reducing insurance premiums.
- When an accident does occur, an organization that is demonstrably diligent in working to prevent accidents will have a better defense in court, in the public forum, and in the eyes of government enforcers.

The authors would like to believe that the preceding seven chapters, providing insights and methods for planning and design, are themselves useful for risk mitigation. This chapter figuratively steps back to look at the subject in a different context. The purpose of this chapter is to help those involved in planning to understand operations in the context of risk management, focus on where attention is needed most, and develop a systematic approach to mitigation. As approached here, risk management is treated both mathematically and subjectively. Though using mathematics to create rankings, statistics, and actuarial models is broadly useful, it does not by itself engender safety. It helps prioritize, but does not solve particular problems.

8.1 Introduction

The original author once stopped at a jobsite to check on a mobile-crane installation he had designed. The large truck crane was to be picking loads from the rear and setting them over the side at nearly full *rated capacity*. The crane was placed on a heavy timber mat deck with the load-side outrigger floats centered on a retaining wall. The operating position corresponding to each pick load was specified.

As he approached the crane, a load was being swung from rear to side, then boomed out by the operator to the final placement radius. He watched with interest as one outrigger beam broke free in its housing and lifted, then with alarm as the other started to show signs of lifting too.

Recognizing that he was on the verge of tipping over, the operator stopped booming out, brought the load back to a lesser radius, and swung it back to where it had been brought aloft. He came out of the cab to join an impromptu inquiry to see what had gone wrong. The crane was set up in the correct configuration and location. Weight and radius were correct. There was no error in reading the load chart. The only respect in which the installation differed from the design was that an off-the-rear screw-jack supplementary outrigger had been put to use. This optional accessory was not needed for this work and should not have affected over-the-side stability in any event. But when the operator was queried about it, the cause of his woe was revealed.

The operator had thought that perhaps some of the trucks delivering the loads would not be able to approach as close as planned and that greater radius picks over the rear might be needed. He had noted in the past that the screw-jack mechanism of the off-the-rear outrigger did not permit the float to be preloaded as the hydraulic side outriggers did. After setting the jack as hard as could be done manually, he loaded it more by easing off pressure on the two adjacent hydraulic jacks.

That action made the crane feel firmer to the operator when he was lifting loads from trucks over the rear, but when he swung the load over to the side, the crane not fully supported by the main outriggers. Preloading the rear float had made the crane less stable over the side. Full resisting moment, and therefore the ability to handle rated loads, could not be restored until the machine tilted sufficiently to engage the relieved side outrigger.

Crane and rigging work involves numerous rules of practice. A large body of unwritten knowledge is learned and passed down from journeyman to apprentice. There is also guidance that is published; it can be found in rigging and training books, operating manuals, and regulations such as OSHA. On a particular site, like this one, there may be also a written set of site-specific plans to provide authoritative guidance. Although a published or an authoritative source is not always correct and may run up against field conditions, an operator or rigging supervisor should think hard and more than once about departing from it, not merely tossing away its authoritative provenance. Departing from the script puts the improviser and his employer at risk for blame, or worse, if something goes wrong.

The physical principle underlying the incident described here is that the equilibrium of the crane had been altered by the action of the operator. He thought he was clever enough to improvise a more stable arrangement over the rear without realizing that he was concurrently diminishing the stability over the side. Had he read the crane manual and load chart, he would have found that his action went against the manufacturer's instructions. Had he followed the engineer's plan, the rear float would never have been out. For either of these reasons, the action that resulted in the instability of the crane should never have occured.

8.2 Sources of Risk in Lifting Operations

Crane and rigging managers are responsible for getting the job done safely and on time while attempting to make some profit in the process. Whether consciously applied or not, risk control is embedded in the methods employed by successful managers. However, even with mindful managers and employees, who are diligent at eliminating risk, some accidents are inevitable; that is reality.

Risk is a consequence of uncertainty associated with events.[1] Effective risk management is thus a process of dealing with uncertainties, specifically trying to determine

- What might go wrong?
- What is the likelihood?
- What are the possible consequences?

A magazine article examining a particular aircraft disaster in extensive detail describes three classes of errors that underlie accidents involving a mix of people and technology.[2] The article's author lists those classes as procedural, design, and system. Consider how they apply to crane and rigging work.

Procedural errors are errors of implementation; avoidance strategies would incorporate practical plans clearly communicated to field supervisors who are competent to diligently guide their execution. The crews executing the work must therefore be experienced, trained to their tasks, and supervised during performance.

Design errors are usually thought of as the responsibility of manufacturers—out of the control of equipment users—but the class also includes equipment and methods originating with the user. A defect in the design of the lifted load would be in this category too. Diligence and competency of designers and planners are paramount, but another element of risk avoidance is feedback from field to planning office and from planners to vendors of material and equipment. Without feedback, there is no opportunity to discover mistakes and learn from them. White-collar managers and credentialed engineers should be encouraged to overcome aloofness in order to learn from their blue-collar colleagues. Too often modifications are made to head office planning by field supervisors. Their companies had not sought feedback and reaped delay, unnecessary cost, and heightened risk.

System errors are an outgrowth of complexity itself; they are often characterized by the left hand not knowing what the right hand is doing. Modern crane and rigging operations might involve input and oversight from numerous parties, and too often one group has no comprehension, or even awareness, of the work of the other individual participants. The system, as used here, is the big picture. It includes everyone and everything involved, from the crane manufacturer to the laborer smoothing the ground.

Unavoidably, the preparations for crane and rigging operations must rely on all of the parties. Those involved could include engineers; equipment designers, building or facility designers, and installation

---

[1]Risk—Informed Decision Making. Bilal M. Ayyub, Peter G. Prassinos, John Etherton, *Mechanical Engineering*, January 2010.

[2]The Lessons of ValueJet 592. William Langewiesche, *Atlantic Monthly*, March 1998.

designers; and managers of personnel, equipment, maintenance, traffic, and operations; and planners; schedulers; and vendors. The preparations may encompass activities at more than one location. Errors are likely to be made during the course of executing these disparate events and activities. Although each individual error may by itself be minor and not likely to cause an accident, several small errors combined could interact to produce disaster.[3] The greater the complexity, the greater the opportunity for such interactions, and the less likely errors will be discovered and corrected. Therefore, whenever possible, managers should employ strategies to simplify and separate activities. There are a number of ways to accomplished simplification including.

- Give forethought to the selection of equipment. When crews are not experienced with particular machines, their use adds complexity. Some heavy-lift rigs require elaborate assembly procedures that depend fully on the competence of one or two technicians.
- Establish control procedures. Surveying instruments, electronic sensors, TV monitors, and human observers can be employed to monitor alignments, forces, and movements, but those multiple inputs to supervisors add complexity to control and decision making.
- Separate portions of the operation into different distinct independent procedures. When tasks can be isolated, performed, and checked independently of the rest of the procedures, the system becomes less complex.
- Require project-specific training. Individuals trained for the tasks they must perform in a planned operation will be less likely to react in unexpected ways; this will reduce the likelihood that wayward behavior will complicate the actual event.
- Weigh risks and benefits associated with particular simplification schemes.
- Allow independent outside scrutiny. This may include third-party review of engineering, planning, or job hazard analysis; it might also include outside inspection of equipment and installation work or auditing of equipment maintenance.
- Clearly assign responsibilities and a decision-making hierarchy.

---

[3] In complex systems, a simple failure rarely leads to harm. Human beings are impressively good at adjusting when an error becomes apparent, and systems often have built-in defenses. . . . When things go wrong, it is usually because a series of failures conspire to produce disaster." From *When Doctors Make Mistakes* by Atul Gawande. The New Yorker, Feb. 1, 1999. The authors' work in accident investigations have revealed the same paradigm.

Dry runs can prove to be a very effective way to reduce systemic risk during complex operations because they can uncover contingencies that were overlooked during planning, identify tasks that crew members do not fully understand, and highlight coordination problems. These practice runs are typically conducted without load, and are best followed by a full-crew debriefing. That gives each team member the opportunity to contribute feedback.

Among crane and rigging accidents, a small percentage arises from one cause or another that could not have been anticipated, nor reasonably have been prevented. Irrational acts, improbable convergences of events, and acts of God are within this percentage. No meaningful risk reduction would be achieved by focusing on such uncontrollable events.

Control of some categories of risks is within the purview of safety professionals and general management rather than in the hands of those who plan and execute lifts. Having a trained, disciplined work force free of drugs and alcohol should be a given. Being furnished with equipment that is well maintained should be as well. Unfortunately, shortcomings in these areas are sometimes passed along and must be dealt with by lift professionals.

Training of a general character is commonplace, close to universal, but more specialized training for crane and rigging work is often lacking. One of the authors, for example, has heard an old-time crane operator complain that he is frequently put at risk by signalmen lacking proper skills.

It has already been mentioned that effective risk management begins with an identification of what might go wrong. In crane and rigging, most failures arise out of one or more of these categories.

- Deficient equipment
- Pressure from cost or time constraints
- Inexperienced management
- Lack of training, knowledge, or skill
- Inadequate planning
- Unreasonable demands of owner or management
- Environmental conditions
- Unclear instructions
- Operator errors
- Changed circumstances

Some aspects of the last four categories, however, defy prediction. To those, adding misjudgments and irrational acts completes the list of risk categories for which preventive actions are not likely to be fruitful.

An analysis of accidents reveals that the majority are the result of more than one contributing cause. The design standards and practices governing hoisting equipment usually enable them to overcome one error or failing, but adding a second or third mistake can overcome their resistance to mishap. Most errors posing lifting equipment risk can be anticipated, making protective actions feasible. Each of the risk categories therefore warrants discussion.

Deficient Equipment

A piece of equipment might either be deficient in the sense that it is not suited to the assigned task or could be the right choice of hardware in a deficient state. A crane that is not correct for the work invites rash acts to compensate for the shortcomings; an ensuing accident might be mischaracterized as an operator or crew error when it is really a planning failure.

A crane that is in deficient condition can pose a risk at several levels

- The operator is distracted or must compensate for the deficiency in some way that deviates from normal operation.
- The crane breaks down at a critical juncture.
- The crane cannot perform according to its specifications.
- Catastrophic failure occurs in a critical component.

Crane accidents can occur through all of these avenues. Maintaining a crane in safe condition, therefore, requires more than keeping key structural components in good shape. Anything that degrades the performance of the equipment or the operator or the machine is a potential hazard.

Pressure from Cost or Time Constraints

Most people strive to perform according to expectations, but some aspire to set the world on fire. The latter is an example of what could be called internal cost/time pressure. Supervision and project schedules, and a job behind schedule, can create an atmosphere of tension, tending to force people to proceed faster than their comfort level or circumstances allow. The too-low bid can result in an example of this. Those situations, usually trickling down from managers, could be called external cost/time pressures. But whether external or internal, the effects of the pressure can be similar.

Lifting operations and rigging work do not take well to efforts to move them more quickly than their proper pace, which should be measured to the task. To an outside observer, some operations appear to advance at a glacial pace. Fast crane lifts, swings, and landings introduce dynamic, or inertial, forces which act on the massive springlike characteristics of crane components, and on the load to make load movements irregular and harder to control. This gives rise to risk to the crane, to nearby objects, to the rigging crew, and to workers in the vicinity. With the exception of duty cycle work, crane operations are intended by the equipment designers to be gradual and deliberate. Generally more work can be produced in a given period of time when a crane operates at a moderate speed because the operator can better control the load in space and while landing. Also the crew will be more at ease, less apprehensive about being slammed or pinned by a wayward load.

Field managers should bear in mind that schedules and budgets have been created by fallible human beings, in an office setting, making estimates and using judgment. The data they produce do not necessarily reflect conditions on the ground; indeed an estimate may be off by a long shot. The executives they report to should recognize that budgets and schedules are goals, not absolutes, and should not encourage their underlings to think otherwise. A forced atmosphere of pressure and a culture of getting it done in spite of all odds are apt to ultimately bring trouble.

Some companies offer rewards or bonuses for work brought in on time or below budget or for meeting specified production goals. This may appear to be forward looking and enlightened means to motivate staff, and often is, but unless backed up with effective discipline, could lead to cutting corners, slipshod work, and added risk.

An example of an overturning accident, which caused one fatality and a serious injury, provides an instance of time pressure affecting risk. A construction contractor had its own crane and long-serving crane operator on site. As bad luck would have it, that crane broke down during the morning before a large and important concrete pour. The job was on a 10-hour, 4-days per week schedule, and the pour was set for a Thursday to allow the concrete to cure over the long weekend. To permit work to go forward, a local rental crane was called in, but it did not arrive until midafternoon. The rental crane operator, under pressure due to the lateness of the start, did not take the time to install the latticed extension, which would have materially increased lift capacity at the pertinent radii. Other circumstances led to the use of an oversized concrete bucket and eventually to an overfilled bucket and tragic results. The company crane operator later stated that he would have objected to the oversized concrete bucket and refused the overfilled load and that it was well-understood company policy to respect a crane operator's objections. The rental operator was not aware of this policy; he was aware only of the pressure of time.

Inexperienced Management

Experience cannot be learned; it can only be accumulated. When an individual attempts to work out of the bounds of accumulated experience, risk is introduced because that person may not fathom the limits of personal knowledge. The result can be either deficient or faulty planning or shortcomings in execution. In crane and rigging work the outcome can be a solution that is impracticable, overly conservative, or dangerous. An impracticable or overly conservative approach may tempt others to ignore or work around the inexperienced manager, creating another avenue for hazard. With equipment and methods in a steady flux, even a veteran runs into situations where his current knowledge is insufficient. Humility in this regard is an asset. There should never be a hesitation to consult or collaborate with others if it makes the job safer.

Lack of Training, Knowledge, or Skill

Not all members of a particular work crew will be equally trained, skilled, or experienced, nor is this often necessary. But when crew assignments are made, consideration should be given both to the background of each individual and to the aggregate qualifications of the group. Training and experience overlap and are complementary. A work gang needs a balance of both.

Crew members rely on one another to take needed actions at appropriate times, and one member's failing to do so could cause injury to another. Working with suspended loads is not a fully predictable exercise, because load behavior relies on a number of factors, including the actions and interactions between the crane or derrick operator, crew members, and the wind; and the reactions of the slings, hoist ropes, and other crane components, and of the load itself. Crew members well placed and alert to respond to, and check, an unexpected load movement could protect other crew members from harm. It is experience that conditions people to react appropriately to the unexpected and to prevent an incident from becoming an accident.

Nuances of crane and rigging work are best learned by mentoring rather than in the classroom, hence the time-honored practice of the tradesman passing on experience to the apprentice. Sustaining skill levels over time is accomplished with an effective apprenticeship program and an appropriate mix of journeymen and apprentices at work sites.

In bygone times training and apprenticeship for construction tradesmen were one and the same. With the growing complexity of equipment, work methods, and regulations—coupled with a lessened tolerance for mishaps—a more formal training regimen is now a necessity as a supplement to apprenticeship. Three tiers of training programs for crane and rigging field workers are available.

- Orientation instruction given to all workers. This includes general safety classes and topical treatment of subjects such as first aid and fall protection.

- Specialized instruction for topics that have wide applicability in the field. Examples are rigging, crane signaling, and reading load charts.
- Training targeted for particular products or systems. These classes are usually offered by a product manufacturer or its agent.

A crane operator often is hired out of a union hall or just arrives with a crane from a rental house. Managers and site supervisors do not have the opportunity to assess the operator's competence other than by observing the hire at work. There is a broad movement in the United States toward requiring operator certification according to a uniform standard. Several organizations, both for profit and nonprofit, conduct written and practical tests for crane operators, and issue certificates to those who pass. In the nonprofit sector, the NCCCO offers certifications for various classes of cranes.[4]

By requiring a certified operator with credentials from NCCCO or an equivalent, a manager can be reasonably assured that a hire will have basic knowledge and skills to operate equipment safely. It does not guarantee a high-level of proficiency or that is the certificate holder is specially trained for every piece of equipment.

Inadequate Planning

All crane operations require at least a small measure of planning. Some need much more.

At one extreme is a crane at an open field site handling a load that is well-below rated load, lifting from a truck bed and landing the load within close visual range. This simple task requires a small crew, slings and fittings, and perhaps some dunnage material. Planning can be informal, involving nothing more than quick conversation. On some operations, planning may need to go a little further and could require a written work order listing the personnel and equipment needed and the means to procure them, and perhaps requiring accounting codes for the various items.

At the other extreme is the major lift utilizing one or several primary lifting cranes, assist cranes, load transport arrangements, custom-made rigging accessories, an extensive crew including specialists, and coordination of activities among several companies. Here planning starts well before equipment and people arrive at the jobsite, include a job hazard analysis, detailed drawings, calculations, materials lists and orders, written procedures, specifications, checklists, scheduling, monitoring, and control equipment and procedures, and a myriad of other details.

[4]National Commission for the Certification of Crane Operators, Fairfax, VA.

A large percentage of lift assignments come between these extremes. Logically these decisions should be based on some assessment of potential risk. The appropriate extent of planning may not be clear or who should do the planning. A company might have a policy to delineate how these decisions are to be made. Consideration might be given to the crane size and type, percentage of rated load, and site-specific concerns.

When planning has been performed well, the appropriate crane will be sent to the site and the accessory equipment and personnel needed for the work will be on hand. The site will have been scouted to verify, among other considerations, that the crane has enough room for assembly, travel, and work and that there are no obstructions or danger from overhead power lines. When planning is lax some needed material, equipment, or person may be missing or inappropriate; a field supervisor may feel pressured to make do with what is there. Experienced field foremen and their coworkers have been known to demonstrate remarkable ingenuity in overcoming such shortcomings and successfully completing a complex task. But the other side to that coin has been avoidable accidents with deaths, injuries, and property damage.

Unreasonable Demands of Owner or Management

Managers with deficient understanding of crane and rigging work can impose policies that bring unnecessary risk to the jobsite. Unrealistic or unachievable cost allowances and schedules have already been mentioned. Some will argue that management shortcomings can be imputed as underlying most of the risks mentioned in this discussion, because it is management's duty to create and maintain a safe work environment. That argument is left for the lawyers. The authors' point of view is that a manager responsible for crane and rigging work needs to develop a keen visceral sense of how policies may play out in the field.

Nearly any company of significant size now has a safety program, a safety manual, and at least one safety manager. Unfortunately these measures are often mostly boilerplate about hard hats and safety harnesses but do not suit the complexity and specialized knowledge inherent in crane and rigging work.

Lifting and rigging activities, particularly the more complex ones, possess a degree of unpredictability, often requiring more time to complete than even a generous estimate allows. The nature of an operation may call for continuous work until completion, forcing crews to be kept on duty for 10, 12, or even 16 hours straight. By its nature, some work must be completed during downtime or off-hours. Needless to say, an individual's reaction time becomes longer and judgment less sure and less dependable as the day stretches far beyond normal working hours, more errors are made and improper conditions go uncorrected. Having investigated many accidents, the authors have noted a repeated theme of failures taking place after long hours of work. Seldom is there a direct link, but the inference that fatigue is a contributing factor is strong. Managers should neither require nor demand excessive hours. Workers should not allow pride, bravado, or greed for overtime pay to take precedence over sensibility. When it becomes necessary to work long hours, consideration should be given to allowing rest for those individuals who, when fatigued, pose a materially heightened risk. For the longest days, bringing in a relief crew might be well advised.

Environmental Conditions

Elements in the environment that can adversely affect lifting operation risk include wind, precipitation, light, noise, site conditions, and adjacent activities.

Chapter 3 covers action of wind, and offers extensive means for calculating wind loads, while Chap. 5 deals with wind on mobile cranes, and Chap. 6 on tower cranes. Those chapters provide information for job preplanning, what might be called wind on paper. However, no level of planning will immunize the jobsite to the hazards of wind. Low-level gusty winds can make loads move excessively and unpredictably, endangering the load-landing crew and raising the possibility of hitting other objects. Thunderstorms can strike with unforeseen speed and intensity. A field supervisor is often faced with decisions respecting measures to deal with severe wind after it has arrived unexpectedly. Sometimes events can overtake even the best judgment; damage can occur before any defensive steps can be taken. The vagaries of storms can create risks without remedies and demand difficult decisions on short notice. For example, a supervisor in the middle of a concrete pour when a storm strikes is in no enviable position, perhaps having to choose between a cold joint in the concrete or a perilous continuation of the crane operation. This is the type of occasion when the question of balancing risks against costs has a degree of poignancy that leaves no room for prolonged debate or contemplation.

Rain and snow restrict visibility. When work proceeds during precipitation, hand signals may need to be modified or abandoned in favor of other means of communication. Slip and fall dangers may inhibit, or at least slow down, some work.

On occasion, it becomes necessary to carry out crane or derrick operations at night. Planners and managers must recognize that artificial lighting can never be equivalent to daylight. Lighting should therefore be planned with the same care as the rest of the operation. Obstructions in the path of the load and boom should be lighted and the positions of lights selected to avoid glare. Operations must be anticipated to be slower than in daylight. These same considerations might also apply to indoor lifting work.

Job supervisors should be prepared for changing conditions of visibility. Sun glare may suddenly inhibit the operator's ability to see signals. A delay may prolong a lift operation into dusk.

One of the authors, as a young engineer, was in charge of dismantling a tower crane inside of a large, hyperbolic cooling tower. Aware of how early darkness engulfed the interior of the tower, he instructed the dismantling crew to wait until morning to make the difficult boom pick. He was overruled, and although the pick was started during daylight, it was not concluded until the interior was shrouded in deep shadow. Neither operator nor riggers could see the boom tip of the truck crane as it lowered the load or see it bend like a fishing pole because the swing was locked during boom lay down. Finally noticed by the operator, the swing was released producing an abrupt and violent reaction. Disaster was narrowly averted.

Noise can bring about risk when it distracts crew members or startles them, or when it interferes with radio or voice communication. High ambient noise levels limit verbal communication, but are not generally unanticipated. Fortunately, hand signals are universally recognized so that special measures need only be planned when more than hand signal communication is needed.

The environment also includes the site. Physical aspects of site conditions that could pose risk include the suitability of the area selected for crane assembly and setup and of load pickup and laydown zones.

The condition of the ground and soil under the crane is another concern, as are pits, trenches, cellars, and underground utilities. Above-ground obstructions to the path of the moving boom, load, counterweight, and gantry may pose hazards, particularly power lines. Preplanning should address these items, but for short-term assignments, such as taxi-crane operations or short-term rentals, planning is usually done at the site after the equipment has arrived, and it is the crane operator who is encumbered with the task. Risk arises because of the limited time, resources, capabilities, and authority available to the typical operator to resolve some of the issues.

Activities adjoining crane and derrick operations can impose risk on the lifting operations too. Noise has been discussed, but excessive dust can also affect visibility, or even divert the attention of an operator by making breathing difficult. A plumbing contractor may turn up unannounced with a backhoe to dig a trench at the edge of a crane, undermining support. In urban areas this work might be done at night or on weekends, with the crane crew unpleasantly surprised to find trench alongside the crane in the morning. Lastly, the presence and movement of people and machines in the immediate vicinity of lifting work can distract both the crane operator and crew from close attention to their own work.

Unclear Instructions

Instructions include both the written and the verbal variety, such as rating charts; operator's manuals; notes on drawings; project specifications; written procedures and checklists; face-to-face, telephone, email, and radio orders and explanations; and even crane or derrick hand signals. When any of them are ambiguous or misunderstood, risk ensues. Hand signals, of course, are affected by ambient light and by distance; when the operator cannot clearly discern intent, operations should not be permitted until signal clarity has been assured. An intermediate signalman or radios may be wanted.

Written and spoken communications need to be plain, direct, concise, and explicit using basic language appropriate to the recipient. One way to verify understanding of verbal instructions is by asking the other party to repeat the instructions. Teams that have worked together for years sometimes need only sparse communications, often little more than simple gestures. For those in less familiar circumstances, critical communications should be confirmed more explicitly and regularly. Confusion and coordination failures can also result from verbal changes to written procedures.

Using a radio to transmit signals to crane or derrick operators is not at all as simple as first impressions may imply. Radios can be interrupted by interference or by battery failure at either end. How then does the operator know when to stop moving the load? How can the operator be sure that the load will be slowed in time to avoid sudden brake application and bouncing as the load destination approaches?

Whenever radios are used, a signal communication protocol needs to be established, explained to the appropriate parties, and enforced. A common approach is to maintain continuous chatter during signaling, by repeating motion commands over and over, and then advising distances as the end point approaches. In this way the operator will know to stop when the chatter stops, because this implies loss of contact.

Operator Errors

Many accidents attributed to operator error are nothing of the kind. Insufficient planning can put crane operators in the unenviable position of having to make judgments that are beyond what should reasonably be expected of them. An example of this occurs when an operator is required to evaluate the ground beneath the crane and to select support cribbing when conditions are unusual or are not typical for the vicinity. A civil engineer devotes class time to studying soil characteristics and behavior, and will usually require tests before rendering a judgment. A crane operator studies the practical side of soil mechanics, but only by virtue of experience on the job. When conditions do not match experience, the operator is being unfairly burdened with a task for which he has not been prepared and which he may not perform successfully. To make matters worse, it is unlikely that adequate cribbing materials will be found on site to deal with the substandard conditions. An unsure operator using deficient materials is not apt to produce suitable crane supports.

Wind presents another series of situations for which crane operators are often burdened with responsibilities beyond their calling. Most mobile-crane rating charts include a note requiring the operator to reduce rated loads to account for ambient wind. Chapter 3 demonstrates how complex and mathematical such a task can be for an engineer. An operator is not blessed with the analytical tools of the engineer, but must make those decisions nonetheless, on the basis of only experience and judgment. Is this a reasonable burden to place on the operator?

It is not unusual for an operator to arrive at a jobsite from the union hall to operate a particular crane model never seen before. Requiring that operator to go right to work, without some warm up or familiarization time, raises the risk of mishap, because each crane model has its own peculiarities, feel, and response. Moreover, a contemporary crane is likely to be far more automated than an older model, with the load chart programmed and the onboard computer assessing inputs from various sensors. The operator cannot reasonably be expected to hop in the cab and intuitively understand all this along with the complex load chart. Instruction time is needed both in and out of the cab.

Even after all of that, operator errors do occur. They occur because of human frailties or because of misjudgments when an operator has to react instinctively to an unexpected event. One-day-rental operators often leave the yard at a very early hour and have near a full day in by lunchtime. When work runs late, the operator will be more fatigued than the crew, and therefore more prone to making errors.

Changed Circumstances

Here is where even the best planning can be foiled. Some party at the site does the unexpected contrary to agreements or understandings. It includes the plumber making a surprise trench, the excavator or teamster piling something where it should not be, the hot wires that were supposed to be deenergized and the promised delivery that broke down in transit. These changes often introduce risk because now equipment may not be well matched to the task or procedures well thought out may require on-the-spot changes. Meticulously planning may go to waste in view of new realities.

Changed circumstances are less likely to be calamitous when planners do not push the limits of equipment utilization; this might be termed as conservative planning. A little excess lift capacity, boom length, or allotted time might be considered.

A good insurance measure is keeping the planners close at hand or on call for consultation. These are the people who have already explored the options and are most likely to know what parts of the plan are most malleable. They also have the analytical tools—the same ones used to develop the plan in the first place.

Conclusion

Crane safety and productivity are intimately associated, but not in the sense often assumed; to gain productivity it is necessary to sacrifice safety. The truth is quite the contrary. Measures taken to ameliorate risks will enhance safety for sure, but those same measures will also promote maximum equipment utilization and productivity. When people have confidence in the equipment, their coworkers, and the plan of action, they will perform at their best.

However, given the most thorough planning, the best equipment, and the most highly skilled personnel, some accidents will occur nevertheless. There will be instructions that are misunderstood; errors made by well-trained and experienced crane operators; changes in circumstances that cannot be dealt with, such as sudden, unexpected high winds; misjudgments by supervisors and crew members who are properly qualified and experienced; and irrational acts of individuals, acts that offer no explanation and that could not have been anticipated.

A Convergence of Errors

A steel-erection subcontractor undertook to construct the framing for a large, long-span, tall single-story building. Their bid, low by half a million dollars, anticipated operating heavy crawler cranes on the ground floor concrete deck built some 20 ft (6m) above the cellar floor. The deck was robust, having been designed for heavy loadings.

Their lifting plan was accepted subject to the deck being shored. Shoring would cost a quarter of a million dollars, which was neither expected nor included in the bid estimate. The subcontractor protested, showing their numbers, asking the owner to pay for that work, in which case they would still be low bidder. The request was refused and erection ordered to proceed. In the view of the subcontractor, the owner *was being unreasonable* in rejecting a proven equitable claim.

Now *circumstances had changed* and a carefully worked out plan was scrapped to be replaced with an entirely new crane operational setup that did not include the expense of shoring. The bid was also based on preassembling full-bay segments of wall framing on the ground to permit faster, lower-cost erection. That feature of the plan had to be retained.

The new lift plan required that the cranes erect from outside the building with long travel runs where the crane would be carrying the load. It was a quickly put together plan with little consideration for travel paths or crane support. In that respect *the planning was inadequate.* The soil in the travel areas was of low capacity, unevenly compacted, and in need of grading.

The first load, of about 150 tons (136 t) had to be traveled about 700 ft (213 m) from assembly area to placement. Two cranes tripped the panel and then travel commenced with the load under the hook of one. The crane bogged down in the soil after a short run, one track digging in deeply and exposing the boom to a high side loading. With assistance from the second crane, recovery was accomplished and the boom lowered and inspected for damage. Now steel plates were sought to put under the tracks, but only a small number were available.

Travel continued without further mishap until the travel path came to a jog where the crane had to cut sharply around a projection on an adjacent building. Before cutting the tracks, because of the width of the load, the crane had to swing the load full over the side. So far, except perhaps for questionable ground support, the operation was in full compliance with the crane manufacturer's instructions.

Cutting first left, then right, the crane started up a short grade of 2% just a little distance from the setting location. After breaking over the crest onto level ground, the crane was stopped so that measurements could be taken for final crane positioning. But when stopped, the rear third of the track jutted over the crest, not in ground contact. What is more, the track closest to the load, the more heavily loaded track, had been stopped with a hard point, the top of a concrete foundation wall corner, at the track center. Most of the weight of the crane and load was concentrated on only a portion of one track, supported on a hard point with soft soil alongside.

The crane operator reported that he had been sitting idly with his hands off the controls for about 2 min when, with no warning, the load took off to his left and tore the boom off. Although sounding like a tall tale, this recounting was verified to be an accurate assessment of what happened. The CG of the crane and load was situated to the rear of the hard point, over soft soil, and a few inches from the edge of the slope. The hard point would not give, but the soil did, ever so slowly, inducing the crane to lean backward. With the lean, the CG moved to the rear until it passed a tipping point at the break of the slope. The crane suddenly tilted back until stopped by the rear of the tracks making ground contact. The load moved with the crane with the disastrous results described. Although there was substantial property damage, remarkably no one was injured.

The *pressure of time and cost constraints* led to inadequate travel path preparation, an unacceptable grade, and too few plates being used for crane support. But, the final straw was the *misjudgment* of stopping the crane where the tracks were not fully and firmly supported and in the presence of a hard point.

Five of the listed risk categories played a part in the denouement of this accident.

- The hard-nosed owner unreasonably demanded that the subcontractor bear the unanticipated cost of shoring, compelling a change of plan.

- Circumstances changed when the cranes were moved from the deck to outside.
- Pressure and time constraints induced corner cutting in preparing the crane runway.
- The new procedure was inadequately planned.
- The foreman lacked knowledge and training to execute the operation, which was outside of his experience.

8.3 Responsibilities

There may be a number of independent parties taking part in one lifting operation. Who those parties are, and the parts they play, will vary with the nature and complexity of the operations. Again, there are extremes, from the taxi crane making a single lift for a local business to a major building or petrochemical project. For that reason, there can be no single set of guidelines to cover all situations. There are, however, some principles that may assist the parties in sorting out responsibilities.

To clarify the role of each party, an SC&RA[5] task force developed a protocol of roles for clarifying work site responsibilities with respect to cranes and rigging. This effort has been further developed within the committee that devises the ASME B30 standards. The material that follows has been drawn from these sources but with some minor alterations by the authors.

The parties that may be involved in lifting operations are

Crane owner. The party furnishing the crane or derrick, who may not have involvement in the use of the equipment at the site.

Crane user. The party who arranges the crane presence on a worksite and controls its use.

Site supervisor. The party supervising the people at the site, who may include people not involved with the lifting work; may be the same party as the lift director on some sites.

Lift director. The person directing the lifts and the personnel involved in the lifts.

Crane operator. The person at the controls of the crane or derrick.

The following is a general list of responsibilities that could be adapted to most crane and rigging operations:

Crane Owner

1. Furnish a crane or derrick in proper condition and properly maintained.

---

[5]Specialized Carriers & Rigging Association, Fairfax, VA

2. Furnish a crane or derrick including necessary parts and components to satisfy the configuration and capacity requirements given by the crane user.
3. Furnish a crane or derrick complete with applicable rating charts, operator's manual, control identification labels, hand-signal placard, electrical hazard warning placard, and warning labels as originally furnished by the manufacturer.
4. Have maintenance and repair records available, if needed.
5. On request, assist the other parties by furnishing crane or derrick data and technical information.
6. Know which local, state, and federal rules and regulations and which industry standards are applicable to lifting operations.
7. Furnish competent personnel to assist in assembly, disassembly, maintenance, repair, transport, and climbing if requested.
8. Establish an inspection, testing, and maintenance program and inform the crane user of the requirements of this program.

Crane User

1. Comply with the requirements of this volume, the manufacturer's requirements, and those regulations applicable at the work site:
2. Use qualified persons as supervisors for crane activities.
3. Ensure that the crane is in proper operating condition prior to initial use at the work site.
4. Verify that the crane has the necessary lifting capacity to perform the proposed lifting operations in the planned configuration:
5. Use qualified operators.
6. Ensure the assigned operator has been notified of adjustments or repairs that have not yet been completed, prior to commencing crane operation.
7. Use qualified personnel for inspections, maintenance, repair, transport, assembly, and climbing.
8. Ensure that personnel involved in maintenance, repair, transport, erection, climbing, dismantling, and inspection are aware of their responsibilities, assigned duties, and associated hazards.
9. Ensure that the inspection, testing, and maintenance programs specified by the crane owner are followed.

Site Supervisor

1. Visit the site to obtain first-hand knowledge of conditions; particularly note power lines, obstructions, limitations on access and operating areas, and if personnel may have to be lifted with the crane or derrick. Ensure that the area for the crane is adequately prepared, including access roads and work areas.
2. Evaluate the operation to the extent necessary to select a crane or derrick with sufficient capacity, height, reach, clearance, and other characteristics to do the work safely; arrange for the special provisions that may be needed for work in proximity to power lines or lifting of personnel; when appropriate, outside assistance should be sought.
3. Review operations for the purpose of identifying elements of risk and reducing or controlling them; when appropriate, outside assistance should be sought. Allow critical lifts to proceed only when procedures are properly executed.
4. Verify that crane or derrick operations have been coordinated with other activities at the site.
5. Check for the existence of underground utilities and subsurface construction.
6. Verify that the crane or derrick provided is in proper condition and has been maintained as required; ropes should be verified for condition, length, and suitability for the work. Ensure that crane maintenance is performed by a designated person.
7. Know which local, state, and federal rules and regulations and which industry standards are applicable to lifting operations.
8. Verify that the lift director is aware of and informed concerning lifting equipment safety laws, rules, and standards.
9. Brief the lift director as applicable on the data and rationale used for selecting the lifting equipment; review the parameters of the work, limitations of equipment, and potential hazards and their remedies, particularly for operations in the vicinity of power lines and the lifting of personnel.
10. Monitor the lift director concerning training and competence of personnel and for adherence to safety laws, rules, and standards. Ensure that work performed by the rigging crew is supervised by a qualified person.
11. Brief the lift director on the plan of lifting operations.
12. Before the lifting equipment arrives at the site,
    a. Verify that access is suitable with respect to width, length, overhead clearances, turns, grading, and surface condition.

   b. Verify that there is sufficient space to erect or assemble the boom and its attachments, and that blocking is available, if needed, for boom assembly.
   c. Verify that the crane operating area is suitable with respect to levelness, surface condition (compaction), proximity to slopes, power lines, underground utilities, subsurface construction, and obstructions to crane operation.
13. Assess the ability of the ground to support the crane in its working condition.
14. Verify that blocking or mats are or will be available as may be needed for crane or outrigger float support.
15. Verify that barricade material is or will be available as may be needed to keep personnel or the public away from counterweight swing or away from crane operations.
16. Verify that appropriate maintenance and inspection procedures are taking place and that appropriate records are being kept for equipment that will be at the site for long periods of time.
17. Be aware of the unique differences in lifting equipment operations when working under the following specific conditions:
   a. Multicrane lifts
   b. Hoisting personnel platforms
   c. Clamshell operations
   d. Pile driving and pulling sheeting
   e. Concrete operations
   f. Demolition operations
   g. Magnet operations
   h. Operation of barge-mounted lift equipment
18. Verify that all concerned parties are instructed in OSHA[6] and ASME/ANSI B30.5[7] requirements for operational safety for operations in the vicinity of power lines; inform personnel of the dangers and limitations of protective measures against electrical hazards.
19. Verify that a proper personnel basket is or will be available if personnel are to be lifted and that personnel involved are informed of OSHA and ASME/ANSI B30.5 requirements; monitor the operation for conformity with those requirements.

---

[6]U.S. Department of Labor Occupational Safety and Health Act, 29 CFR 1926, Construction Industry Standards, or 29 CFR 1910, General Industry Standards, as applicable.

[7]Safety Standard for Mobile and Locomotive Cranes, An American National Standard, American Society of Mechanical Engineers, New York, NY.

20. Monitor lifting operations for conformity with lift plans and general compliance with safety laws, rules, and regulations.
21. Verify that the crane or derrick operator is knowledgeable or experienced with the lift equipment and attachments to be used; if in doubt, arrange for an operator test, or for practice time in a noncritical location. Operators should refuse to handle a load until satisfied it can be handled safely.

Lift Director

1. Prepare for lifting operations, includeing
   a. Review planned operations with site supervisor including working height, boom length, working radius, quadrant of operation, load weight, load dimensions, center of gravity, and required blocking.
   b. Locate and identify site hazards such as electric power lines and piping (above and below ground) together with site supervision.
   c. Know which local, state, and federal rules and regulations and which industry standards are applicable to lifting operations.
   d. Know limitations of protective measures against electrical hazards.
   e. Be aware of the unique differences in lifting equipment operations.
   f. Verify that the lifting equipment has been maintained and inspected and is in suitable condition for the work.
2. Prepare personnel
   a. Verify that the crane or derrick operator is knowledgeable or experienced with the lifting equipment and attachments to be used.
   b. Review operations with the crane or derrick operator and lift crew; discuss limitations of the equipment, potential hazards, and measures that are to be taken to minimize risk. To assist crane or derrick operators in meeting their responsibilities, it is recommended that jobsite policy permit operators refuse to handle a load until satisfied it can be handled safely and that personnel be so informed.
   c. Verify that the load riggers are experienced in slinging loads and in selecting suitable slings and fittings, when appropriate.
   d. See to it that all concerned parties are instructed in OSHA and ASME/ANSI B30.5 requirements for operational safety when operations will be in the vicinity of power lines.
   e. Verify that a proper personnel basket is or will be available when personnel are to be lifted by crane or derrick, and

see to it that personnel are informed of OSHA and ASME/ANSI B30.5 requirements.

f. Be aware, and make sure that the crew is aware, as applicable, of the unique differences in operations when working under the following specific conditions:
   a. Multicrane lifts
   b. Hoisting personnel platforms
   c. Clamshell operations
   d. Pile driving and pulling sheeting
   e. Concrete operations
   f. Demolition operations
   g. Magnet operations
   h. Operation of barge mounted lift equipment

3. Supervise the work including
   a. Review all work concerning lifting equipment.
   b. Check that personnel are present and fit for work, with required protective clothing and tools.
   c. Verify that the crane or derrick operator is performing all required daily maintenance and inspection tasks.
   d. Verify the crane or derrick operator is aware of his responsibilities.
   e. Establish proper communications between the operator, crew, and signalperson, such as the use of hand signals, radios, and so on.
   f. Assign competent signal person(s), experienced in crane or derrick operations and in hand signals if needed, and verify that they are positioned where most effective.
   g. Check that proper slings, fittings, and lift accessories are at hand and that they are in acceptable condition.
   h. Supervise the installation of the rigging on the load and hook.
   i. Verify that the crane is level and blocked, if appropriate, and positioned in the proper location.
   j. Determine or verifying the weights of the loads to be lifted and make certain that this information is given to the load riggers and the crane or derrick operator.
   k. Check that loads are well secured and balanced in the slings.
   l. Check that the lift and swing path is clear of obstructions.
   m. Check that loads are free to be lifted, for example, that loads are not restrained by mud suction, frozen to the ground, or caught on adjacent materials.
   n. Avoid carrying loads over people.
   o. Restrict access to work areas by unauthorized personnel; verifying that barricade material is or will be available as needed to keep people away from counterweight swing and from the potential impact area if an accident were to occur.

p. Check for wind and take needed precautions to maintain stability of the load, and to avoid wind bringing the crane to a tipping condition, exerting excessive side load on the boom, or overpowering personnel on tag lines.
q. Take note of environmental conditions, such as temperature, lighting, and adjacent activities, and adjust operations accordingly so as not to endanger workers, the public, or property.
r. Verify that personnel are following OSHA and ASME/ANSI B30.5 requirements when they are lifted by crane and derrick.
s. Address safety concerns raised by the operator or other personnel and be responsible if he decides to overrule those concerns and directs crane operations to continue. In all cases, the manufacturer's criteria for safe operation and the requirements of laws and applicable standards must be met.

Crane Operator

1. Review the requirements for the crane with the lift director before operations.
2. Verify barriers and/or warning signs are present to prevent inadvertent access to the hazard area of the rotating superstructure.
3. Know what types of site conditions could adversely affect the operation of the crane and consulting with the lift director concerning the possible presence of these conditions.
4. Understand and apply the information contained in the crane manufacturer's operating manual.
5. Understand the crane's functions and limitations as well as its particular operating characteristics.
6. Use the crane's load rating chart and diagrams and apply all notes and warnings related to the charts to confirm the correct crane configuration to suit the load, site, and lift conditions.
7. Refuse to operate the crane when any portion of the load or crane would enter the danger zone of energized power lines.
8. Perform frequent inspections.
9. Reporting the need for any adjustments or repairs to a designated person promptly.
10. Do not operate the crane when physically or mentally unfit.
11. Ensure that all controls are in the off or neutral position and that all personnel on the crane are in the clear before energizing the crane or starting the engine.

12. Do not engage in any practice that will divert his attention while actually operating the crane controls;
13. Test the crane function controls that will be used and then operate the crane only if those function controls respond properly.
14. Operate the crane's functions, under normal operating conditions, in a smooth and controlled manner.
15. Know and follow the procedures specified by the manufacturer or approved by a qualified person for assembly, erection, climbing, and dismantling.
16. Ensure that the load and rigging weight have been provided by the lift director.
17. Calculate or determine the net capacity for all configurations that will be used and verify, using the load-rating chart, that the crane has sufficient net capacity for the proposed lift.
18. Consider all factors known that might affect the crane capacity and inform the lift director if there is a need to make appropriate adjustments.
19. Know standard and special signals and respond to such signals from the appointed signalperson. When the signalperson is not required as part of the lift operation, the operator is then responsible for the movement of the crane. However, the operator shall obey a stop signal at all times, no matter who gives it.
20. Understand basic load rigging procedures.
21. If power fails during operation of a tower crane,
    a. Sett all brakes and locking devices.
    b. Move all power controls and function levers to the off or neutral position.
    c. Land any load suspended below the hook in a safe and controlled manner if practical.
    d. Before leaving the crane unattended,
        (1) Land any load suspended below the hook.
        (2) Bring the load block to the highest position.
        (3) Move controls to the off or neutral position.
        (4) Sett brakes and other locking devices.
        (5) Release of the slewing brake, unless provisions for nonweathervaning have been specified by the manufacturer or a qualified person.
        (6) Restrain the crane from travel if applicable with rail clamps or other means provided.
        (7) Disengage the main control circuit or stop the engine.

Assigning Responsibilities

The preceding lists apportion responsibilities for a major project with extensive crane and rigging work. The minor project with incidental crane use must also be considered. It is not unusual or unexpected on small projects for roles to be combined.

The scope of such work makes it usual for management tasks to become abbreviated and less formal. A site visit might be replaced by telephoned questions and answers, and the site supervisor role will be covered by the crane operator. On arrival, the operator most often becomes the person on site with the greatest crane and lifting expertise. In effect, the crane operator will wear three hats, including that of the lift director. Needless to say, these situations place a heavy burden of responsibility on the crane operator.

Typically, cranes on a construction project are peripheral to the principal project activities. More often than not, full responsibility is given to the subcontractors who hire and use the equipment. Responsibilities may be delegated to specialists or to vendor or subcontractor employees wearing multiple hats. Contractors trying to keep costs down or that are stretched over too many projects may tend to abuse the multiple-hat approach. It is crucial that key positions be adequately staffed so that checks and balances are maintained.

Nonetheless, on small projects, project management, crane operation management, site supervision, and even the lift director can be one person. Real planning may begin when the crane arrives, and it is exceptional for personnel on small sites to have extensive crane and rigging knowledge. The crane operator is counted on by the renter to fill that void. Without being informed of this, the operator is expected to perform an entire range of additional responsibilities. The wide gap between the user's and the operator's perceptions of the latter's responsibilities are only revealed in a courtroom after an accident has occurred.

The lists furnished cover responsibilities for all parties potentially involved in lifting operations. The lists need to be calibrated to the scale of the operation and adjusted for organizational differences. Responsibilities should be discussed, incorporated in contract language and rental agreements, and become part of discussions about crane hire. All responsibilities listed must be assigned to an appropriate party in the work site organization.

In reworking the lists, the work site is the only focus. The goal is to provide

1. A competent operator
2. The right lifting equipment for the task
3. A sufficient number of personnel on site
4. The needed tools, rigging equipment, blocking, dunnage, and hardware

5. Adequate space for assembly, operation, and load handling
6. Proper crane support
7. Coordination to avoid interferences and hazards
8. Clear lines of responsibility and communication

These basic requirements are applicable to every lifting operation.

8.4 Accident Statistics and Avoidance

Various crane accident statistics have been compiled since the first editions of this book. Although some conclusions are readily drawn from these studies, there are few consistent patterns. Methods of reporting and categories vary. Some statistics are based on the nature or description of the accident whereas others are based on the cause. Some report accidents whereas others compile fatalities. Compilations also differ in the time periods they cover and in the geographical area. Thus, New York City data list no power line contact events, whereas reports from other U.S. and Canadian authorities have shown as many as half of all crane-related fatalities derive from that cause.[8]

A British study of mobile-crane accidents in the 1970s shows overturning to be by far the most prevalent cause.[9] However, because the data are from Great Britain where electric lines are below ground, no reports of electrocution from power line contact appear. Furthermore, these data are more than 30 years old, a time when almost every mobile crane would have been a friction machine and the *load moment indicator* (LMI) was a crude ineffectual device. The statistics have little relevance to today's mobile-crane mix especially with the advances of LMI devices and other control technology.

Contemporary statistics of mobile-crane accidents do not offer much guidance for risk avoidance other than to highlight the prime importance of power line contacts. Overturning, erection, and dismantling incidents; wire-rope breakage; rigging failures; and support deficiencies also are significant hazards in North American mobile-crane use. One study reports that mobile cranes represent over 84% of the fatalities in the use of cranes and derricks.[10]

---

[8]*Crane Safety on Construction Sites*, Construction Division Task Committee on Crane Safety, American Society of Civil Engineers, Reston, VA, 1998, p. 4.

[9]A. J. Butler, Crane Accidents: Their Causes and Repair Costs, *Cranes Today*, Edgeware, Middlesex, England, no. 62, March 1978, p. 24; no. 63, April 1978, p. 28; and no. 65, June 1978, p. 27.

[10]J. E. Beavers, J. R. Moore , R. Rinehart, and W. R. Sehriver, Crane-Related Fatalities in the Construction Industry, *Journal of Construction Engineering. and Management*. Volume 132, Issue 9, pp. 901–910, September 2006.

Another study finds that crane accidents are responsible for 16% of construction fatalities in the United States from 1980 to 1992.[11] Although this is not a devastatingly high percentage, it is substantial nonetheless.

Contact with Power Lines

Judged by deaths and severe injuries, in the United States there is no crane hazard greater than power line contact. In Europe and some U.S. cities, the majority of lines are underground and the problem thus of much less concern. Misjudgment of the hazard is comprehensible but not forgivable because most lines appear so innocuous and it is tempting to believe that contact is easy to avoid. Neither could be further from true.

The most reliable way to avoid power line contact accidents is to have the lines deenergized. Unfortunately, that option is not always available. When cranes must be operated within a boom's length of an energized line, defensive measures must be taken; management has an obligation to see to it that adequate procedures are in place. An entire section within the ASME/ANSI B30.5 volume is devoted to setting out minimum standards of good practice in this regard.

A prerequisite to developing adequate procedures is an understanding of the problem. What few people realize is that all overhead power transmission lines are bare wires; there is no insulation to offer protection in case of contact or to prevent arcing as a wire is approached. A boom, load, or pendant line, slings, or the load itself are all conductors that, in effect, make a crane a mobile short-circuit device. Thus, contact or near-contact could allow the electric current to take an unintended path from the power line through the crane and into the ground. But no harm is done unless the current flows, as evidenced by birds sitting unharmed on lines of all voltages. The greatest danger, in case of contact or near-contact, is to the ground crew. When tires interrupt the current, a person touching the crane, load, slings, or even a tag line will complete the circuit and could be electrocuted. Although crane tires act as insulation, steel belted tires have been known to catch fire when exposed to high voltages. The crane operator who remains inside the cab, like the bird, will not likely be harmed.

Power lines are generally furnished with circuit protection devices, either fuses or reclosure devices that make a few attempts to restore power, within seconds of the short circuit, before shutting down the line. Those devices are intended to provide a reliable electric supply,

---

[11] R. L. Neitzel, N. S. Seixas, and K. K. Ren., A Review of Crane Safety in The Construction Industry, *Applied Occupational and Environmental Hygiene.* Volume 16(12): 1106–1117, 2001.

one that is not constantly interrupted by rain or morning dew. The intent is to protect the power grid; none of them work quickly enough or at a low enough level of energy to offset the danger to people.

Electric current can only harm crane or derrick crews if a part of the equipment either contacts or comes close to contacting an energized line. The minimum safe distance specified by ASME/ANSI B30.5 is 10 ft (3 m) for lines carrying 50,000 V (50 kV) or less, with greater distances given for lines of higher voltage. Common experience informs the engineer that current will not arc over such great distances. Why then are such distances required?

The distance limits are arbitrary numbers, but very practical nonetheless. They not only take into account electric behavior, but they allow for how cranes and people behave as well. These distances have been mandated by OSHA for more than 25 years and included in ASME/ANSI B30.5 even longer, and yet crane operation electrocutions remain the greatest leading cause of crane-related fatalities in the United States. Based on that, one might argue that the 10-ft (3-m) minimum is not enough.

A number of elements must be added together to explain the minimum distances specified. Wind, even light winds, will swing power lines and cause booms to move about as well. Boom movements, albeit small, come from the dynamics of hoisting and swinging loads. A crane operator is required to focus his attention on either the load or a signalman. He cannot simultaneously be judging distance from a power line. Should the operator's attention be diverted to the line instead of the handling of the load, people may be endangered; the operator may fail to see and respond to a signal.

Assume that a special signalman is on hand whose only duty is to observe proximity to the power line; this is a good practice. The special signalman will not be the main focus of the operator's attention; the regular signalman will be. There will be a time lapse between the power line warning signal being given and when it is perceived by the operator. Reaction time and proper stopping time add to the delay. Potential movements during those time delays comprise most of the minimum clearance distances mentioned. That is why the distances specified are practical minimums and actually offer only a small margin of safety.

A number of devices are on the market to assist in maintaining safe distances from power lines or preventing current from passing through the crane. These include boom cages, insulating barriers, proximity warning devices, and insulated links. They each offer a somewhat different but limited type of protection. None is fail-safe and thus none can substitute for maintaining a safe distance. The more distance the better, but the *absolute limit of approach* must always be respected.

A barrier, an insulting cage, can be placed around the boom or between the boom and the power line. An insulated boom cage is compresed of nonconductive material mounted on the tip section. Being an open lattice, it is not a complete barrier. Moreover, there is no guarantee that a fault will not render it ineffective or that some unprotected component such as a load line will not contact the line. Nor is an insulating barrier impervious; a sleeve wrapped around the line, for instance, can tear and expose the line.

An insulating link prevents current from being transmitted between the hoist line and the load. It protects the rigger and perhaps anyone holding a tagline, but anyone touching the crane could be electrocuted. The operator is likely to be safe if he remains in the cab. These devices are generally reliable, but a fault caused by mud or a breakdown of the insulation is possible.

A proximity warning device detects electric fields. As helpful as one of these devices might be, it should be used as an additional measure of protection, not as a substitute for distance and human observers. In addition to the possibility that a device might malfunction, rapid or uncontrolled crane motion can defeat its purpose if the distance is too close.

When lines cannot be deenergized, the best protection is to have posted observers who are undistracted by other tasks and who are aided by physical guides such as tape, string lines, or cones.

Overturning

Overturning is a prevalent cause of failure for mobile cranes. It can occur while the crane is lifting, in an operating stance without a load on the hook, or during travel.

Among tower cranes, overturning is unusual but not unheard of. Most tower cranes will fail structurally before overturning; an exception is a tower mounted on a chassis base. In an extreme wind, one that exceeds the out-of-service design criteria, the governing mode of failure for a ballasted base is likely to be stability. When the crane is working, its limiting devices prevent an overload severe enough to cause overturning.

Most overturning accidents are not caused by overloading, and overloading accidents are less common than in the past. Only a small proportion of lifted loads are at or near rated loads, and there is heightened recognition that near-capacity picks on mobile cranes are *critical lifts*. Contemporary mobile cranes, moreover, are almost all equipped with reliable load-moment indicators (LMIs); on some models these devices are wired to stop the crane from being overloaded. The potential for overturning is often close enough—mobile cranes in the United States and Canada are rated to lift between 75% and 85% of the tipping load under optimal operating conditions. Those ideal conditions include a firm supporting surface, leveling to within 1%, nearly absent wind and dynamic effects, and a freely suspended load. Limiting devices notwithstanding, an unskilled or cavalier operator can overturn a crane while lifting.

Cranes furnished with load or load-moment indicators can be overloaded if (1) the device has been turned off or is down due to malfunction, (2) the device is out of calibration, or (3) operating conditions (wind or operating speeds or out of level) are so far from ideal that the published ratings can lead to failure. The mounting of a device is itself no assurance that operations will be safe.

The European practice is to rely on LMIs as a primary defense against overloading. The U.S. practice is to rely first on the discretion of the operator and secondarily on these devices. There is a tendency for some operators to become overly reliant on LMIs and to use them in place of judgment. Nonetheless, an operator would be remiss in ignoring the high-quality information coming from contemporary LMIs. The best operating practice is to use this information, but use it with discretion and as an aid to sound judgment.

Perhaps a larger problem is in misreading of load charts, especially as they have become highly complex on some machines. Operators who do not fully understand the meaning of the values on the rating chart and who do not understand the limitations of the crane and its ratings may operate imprudently or permit supervisors also lacking in knowledge to pressure them into doing so. A contemporary mobile crane typically has the load charts programmed into the LMI, but if the configuration is not dialed in correctly, the values will be wrong and the operation may be unsafe.

Typically, managers and supervisors assume that an operator is qualified because of a license, certificate, or years of service. The truth is that many operators have difficulty interpreting rating charts and most do not understand the strength or stability implications that underlie the ratings. To determine the useful load that can be lifted, rating chart values must be adjusted. In addition to reductions to account for the load block and for slings or other lifting attachments, reductions must be made for such items as jibs (either mounted or stored at the base of the boom), auxiliary or whip lines, optional head sheave arrangements, boom tip lights or pile lead adaptors, and the effects of operating conditions that differ from ideal. Each such item not taken into account reduces the margin of stability and makes the crane more prone to accident. All too often, these required capacity reductions are ignored.

If loads that are close to rated loads at any planned working or operating radius are to be lifted, to achieve a safe operation, it is necessary that the weight of the load be known with reasonable accuracy. To proceed otherwise requires seat-of-the-pants operation, bravado, or culpable negligence. There are many ways to determine load weight before the load is taken to a critical outreach.

Preventing Overloads

What constitutes an overload is a relative matter. Technically and legally, any load in excess of that given in the rating chart for the crane configuration in use is an overload. As a practical matter, any load that causes the crane to overturn or to collapse is an overload; site, operating, or environmental conditions may override the rating chart and make it necessary to limit lifted load to less than chart values. Technical and legal overloads can be prevented by use of load or load-moment indicators or by adequate preplanning. These measures also help in preventing practical overloads; well-trained and experienced staff are the main line of defense.

For many construction operations load-indicating devices are not needed to control lift weights, although they can be effective as a means of checking. When the rigging crew is aware of crane capacity and capable of determining load weights, lifting cycle time is minimized and efficient operations are maintained. The crew will make up and sling the loads, and the indicator will allow the operator to check on them. If an indicator is used as the primary means for determining load weight, loads will be slung, tested, set back down, and adjusted until a proper load is ready for lifting; that procedure can be intolerably slow and inefficient.

As part of the preplanning process, the procedures to be used to control load weights should be established. If manufactured items are to be lifted, weights can be secured in advance from the producer and made available to the jobsite. One person in the crane-operations crew should be assigned the task of load determination. This person must be competent in arithmetic, be comfortable with weights and measures, and understand the rating chart.

Before each lift, the load controller should determine load weight, including the weights of lifting accessories; the operator can then be informed either verbally or by means of a sign placed on the load. For miscellaneous loads, the weights must be calculated by this designated person, and aids such as Table 8.1 or handbooks of steel-pipe, beam, or reinforcing-rod weights must be made available for this purpose.

Once when an operator who had been involved in an accident was asked why he had not referred to the rating chart before making the lift, he replied that he did not have the time to sort out the chart numbers while a crew was standing by waiting for him. A conscientious operator is sometimes torn between the pressure to proceed and the responsibility to operate safely. Planners, supervisors, and construction executives should be aware of this very human problem and take the steps necessary to promote safety while maintaining productivity. Even when load weights are known, overloading and accidents can occur unless the known weights are related to the rating chart and to prevailing conditions.

Material	Weight	
	lb/ft³	t/m³
Aluminum	165	2.64
Brick, hard	140	2.24
Common	120	1.92
Cement	90	1.44
Cereals, corn, rye, etc.	50	0.80
Cinders	45	0.72
Clay, dry	65	1.04
Clay, damp	110	1.76
Coal, anthracite, loose	60	0.96
Coal, bituminous, loose	55	0.88
Concrete, reinforced	150	2.40
Earth, dry, loose	80	1.28
Earth, wet	110	1.76
Gasoline	42	0.67
Glass	160	2.56
Iron, cast	480	7.69
Paper	60	0.96
Petroleum, crude	55	0.88
Sand, gravel, dry	100	1.60
Sand, gravel, wet	120	1.92
Steel	490	7.85
Stone, building	175	2.80
Stone, crushed, loose	95	1.52
Timber, softwood	30	0.48
Timber, hardwood	35	0.88

TABLE 8.1 Weights of Common Materials

One simple measure that can be taken is to provide each crane with a roll of masking tape and a marking pen. When the crane is being assembled, the operator can then run a vertical line of tape on either side of the rating section for the boom length to be used and for the jib length and offset as well. In this way the pertinent set of numbers becomes segregated from all the numbers that do not bear on the work at hand. Furthermore, the operator can mark radius numbers or any other needed data on the tape margins thus created.

When machinery or fabrications are to be trucked into a site and the planners were only able to obtain approximate weights, the loads can be checked at certified scales or even at the site using portable wheel scales. Another option is to have a portable hook scale available to weigh loads during unloading.

Perhaps the strongest measure that can be taken to reduce overloading accidents is forethought in the office before the crane is sent to the site. Once the crane is on the site with a particular boom length mounted and minimum radius limited by site conditions, field crews will usually make every effort to place the loads given to them even if this means knowingly overloading the machine or bypassing indicator warnings. The pressures of time and money make this almost inevitable. Overloading accidents can therefore be prevented in advance by thorough planning. A small investment at this time will be more than reclaimed by improved productivity, not to mention accident prevention.

Slewing-Related Overturning

Slewing-related overturning incidents are difficult to discern from statistics and may be alternatively characterized as having a different causation. The description implies that excess slewing speed is the cause, but included in this category may well be other types of incidents. The hook might swing to a less stable *quadrant of operation*, or there may be a failure of support as the crane weight shifts from one support point to another.

All other conditions being ideal, lifting full-rated loads should not be attempted unless slewing speeds will be kept very low, and it is the least rating in the quadrant of operation that will govern. Whenever an operation requires repeated crane cycles or a set production rate, higher than normal swing speeds can be expected. Load rating reductions are necessary for high swing speeds. Some rating charts specify reductions of 20% to 25% for such *duty cycle* operations. Though this reduction is intended primarily to prevent fatigue failure, it is helpful also to attenuate the effect on stability of fast slewing.

Most crane operators know that cranes can lift greater loads over the corner than they can at other positions, but rating charts provide no such information. This has resulted in accidents from attempts to stretch crane capacity by lifting over the corner. It is foolhardy to attempt this, and supervisors should never permit it.

Few latticed boom truck cranes are rated for lifting over the front. There is greatly reduced resistance to tipping when the boom is in the front quadrant, unless a front outrigger float or other stabilizing means has been provided. If there are no values in the rating chart labeled as applying over 360°, operation over the front is not permitted.

Travel-Related Overturning

Although many truck cranes and nearly all rough terrain and all terrain cranes are equipped with off-road construction-type tires, this feature does not come with a carte blanche license to roam freely about the countryside. On the contrary, cranes of all types, crawlers included, are too stability-sensitive to allow complete freedom to roam, especially with long booms. While travelling assembled, with boom and counterweight, cranes must be kept near level in the side-to-side direction. When traveling without load, speed must be a function of travel-path smoothness. When traveling with load, speed must be held to rates recommended by the manufacturer and rated-load reductions must be followed. See Operations on Rubber later in this section and Sec. 5.10 Pick and Carry for more information on traveling with load.

Travel movements should be anticipated during the planning, and arrangements can then be made in advance to assure that travel paths will be prepared to give adequate support (see Table 5.1) and be suitably smooth. Wherever grades must be traversed, and indeed for all travel movements, instructions should be prepared stating the direction the superstructure must face and the boom angle to be maintained.

Several precautionary measures can be taken to avoid danger during travel moves. When the ability of the ground surface to support the axle loads is in doubt, the heaviest available truck should be driven over the path first. Examination of the tire tracks may reveal soft spots and side-to-side irregularities that need be corrected before crane travel.

Moves should not be made with long booms in the air when windy or gusty conditions prevail. However, when truck cranes with long booms are moved in light winds, they can be protected from overturning using the outriggers. The outriggers should be fully extended with the floats held about 2 in (50 mm) clear of the ground surface on a prepared path. The move is then made at very slow speed, about 1 mi/h (1.6 km/h), with watchers posted at each outrigger to warn of hang-ups. Manufacturer's instructions must be followed for these moves.

Trapped or Caught Loads

When a load is frozen to the ground or embedded in mud, considerable extra force may be required to pull it free. As this force is applied, the hoist lines and boom suspension system stretch elastically. As the load comes free, the elastic stretch induces an immediate rebound. Regardless of the care taken, the load will lift while the boom shifts radius slightly, launching the load into a pendulum swing. Unless damped or returned to ground, the load could overturn the crane.

During investigation of an accident that occurred while lifting a furnace section in a building with a crane placed outside, it was discovered that a small projection on the load had become caught on a steel beam. As the lift continued, the load was made to tilt a little and then it broke free. The tilt caused the load to start swinging and in the end carried the crane over as well.

These descriptions underscore the meaning of a phrase common to all domestic mobile-crane rating-chart notes: the *rated loads apply to freely suspended loads.* Any other condition of the load implies side loading of the boom, the introduction of dynamic effects, load pendulum action, or a combination of these.

The only measures that can be taken to guard against this sort of accident are ground crew and operator training and having alert and knowledgeable supervisors. In addition, when a load is being lifted and there is a possibility that it may become caught on an obstruction, watchers must be posted and there must be adequate light for clear observation of the load throughout its travel; these steps should be discussed during planning.

Wind-Induced Overturning

Jobsite anemometers are low in cost. Fixed types are easy to set up and portable devices will fit in a pocket. The fact remains that few construction executives or supervisors are aware just how sensitive cranes are to wind; fewer realize that mobile-crane rated loads (where stability governs) are established without taking wind into account at all.

Incidents of wind-induced overturning could be nearly eliminated if anemometers were used regularly on jobsites. Other accidents, not easily recognized as wind related, could be reduced in number if supervisors and operators suitably adjusted crane ratings for ambient wind. However, without benefit of an anemometer, wind speed can be approximated either by checking with the local airfield weather station or by using the Beaufort scale for judging wind (Figure 8.1), a version of which is given in Table 8.2. In any case, when wind exceeds 10 mi/h (4.3 m/s), it is advisable that rating reductions be considered. For winds of 20 mi/h (8.9 m/s) or more, reductions in the strength-governed ratings must be made as well as in stability-controlled ratings. When winds exceed 30 mi/h (13.4 m/s), it is usually prudent to cease operations altogether. This guidance is general; wind limitations given in an operator's manual supersede advice given here.

Mobile cranes with tower attachments are especially vulnerable to intermediate-level winds, that is, roughly in the range of 30 to 60 mi/h (13.4 to 26.8 m/s). Some localities are prone to squalls of this intensity that arrive quickly with little warning. Mobile-crane users in those areas must be prepared at all times; those with tower attachments must be ready to fold down the boom or, if winds of 50 mi/h (22.4 m/s) or so are expected, to lower the tower to the ground. The actual measures to be taken depend on the particular manufacturer's instructions for securing the crane against wind.

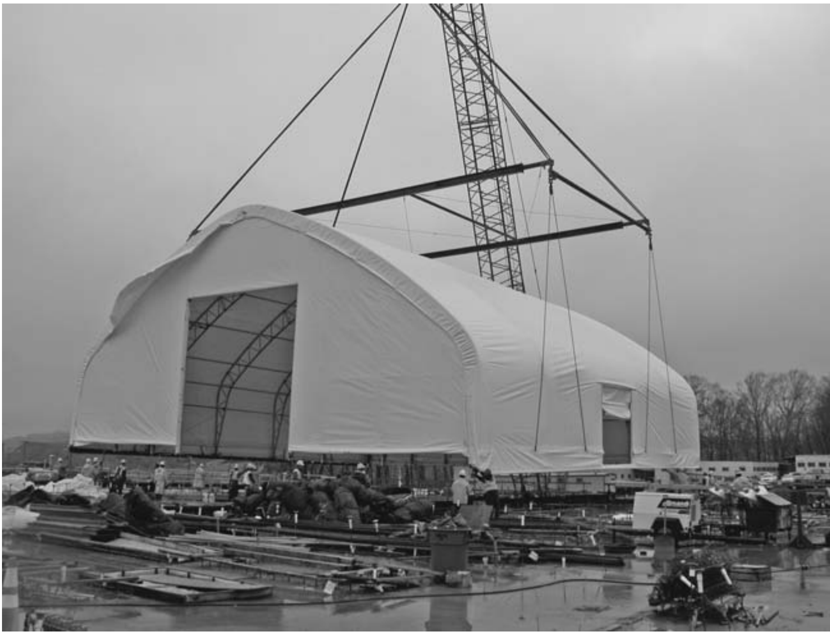


FIGURE 8.1 Temporary building is being lifted into place at a water treatment facility north of New York City. The load is relatively light in comparison to the immense wind exposure area. This load can only be picked in calm air corresponding to the first two categories of the Beaufort scale.

Approximate Wind Speed			
mi/h	**m/s**	**Description**	**Effects**
0–1	0–0.5	Calm	Smoke rises straight up.
1–3	0.5–1.5	Light air	Smoke drifts.
4–7	2–3	Slight breeze	Leaves rustle.
8–12	3.5–5.5	Gentle breeze	Leaves and small twigs move.
13–18	6–8	Moderate breeze	Dust and papers fly, small branches move.
19–24	8.5–10.5	Fresh breeze	Small trees sway.
25–31	11–14	Strong breeze	Large branches move.
32–38	14.5–17	High wind	Walking difficult, trunks of trees hend.
39–46	17.5–20.5	Gale	Twigs break off.
47–54	21–24	Strong gale	Shingles can be carried away.
55–63	24.5–28	Whole gale	Trees may be uprooted.

TABLE 8.2 Estimating Wind Speed by the Beaufort Scale

The principal vulnerability of these machines is to frontal winds, especially when the boom is raised to a high angle. Long-boom mobile cranes are also vulnerable to squalls. With either a tower attachment or a long boom, contingency plans should be made for laying the equipment on the ground. The area for doing this must be chosen in advance and steps taken to keep crane access to that area clear. At urban sites it will probably be necessary to coordinate with the local police or government, as streets will no doubt have to be closed to traffic during lowering and perhaps afterward as well.

A *derate* of the lifting capacities for wind can be approximated after a value for wind velocity has been determined and with a few basic dimensions at hand. For wind on a lifted load with wind-exposure area A and force coefficient C_f related to the shape of the object (see Sec. 3.4), the wind force will be

$$F\begin{cases} \dfrac{v^2}{400}C_f A & \text{U.S. customary units} \\ \dfrac{3v^2}{5}C_f A & \text{SI units} \end{cases} \qquad (\boldsymbol{a})$$

For the U.S. customary units, force will be in pounds when velocity is in miles per hour and area A is in square feet, while for the SI units the force will be given in newtons when velocity is in meters per second and the area is in square meters.

When wind blows from behind the operator, the wind force on the load acts horizontally and takes effect at the boom tip. The hook-load equivalent D, to this force is then

$$D_t = F\frac{H}{R} = F\frac{\sqrt{L^2 - R^2}}{R} = F\sqrt{\frac{L^2}{R^2} - 1} \qquad (\boldsymbol{b})$$

where L is the boom length and R is the operating radius in feet (meters). D_t is the rated-load deduct to account for wind on the lifted load.

Similarly, wind from the same direction acts on the boom itself, creating an overturning moment that can be represented as an equivalent hook load. Here, in determining the wind force on the boom, we will use a configuration coefficient C instead of the force coefficient C_f used before (see Chap. 3). This will make the boom derate D_b take the simplified form

$$D_b = \frac{FL^2}{2R} \qquad (\boldsymbol{c})$$

The force coefficient for wind blowing normal to a latticed boom can be conservatively approximated as 2/3 when the total width of boom is used rather than the actual wind area. For each unit of boom length, $C_f A \approx 2b/3$ if b is the boom overall cross-sectional width in feet (meters). For telescopic cranes $C_f A$ can be taken as 3 (0.9 in SI units).

But Eq. (3.27) tells us that $C = C_f \sin^2\theta$ for wind blowing at angle θ to an object and F acting in the direction of the wind. Then

$$CA = \frac{2b}{3}\left(\frac{\sqrt{L^2 - R^2}}{L}\right) = \frac{2b}{3}\left(1 - \frac{R^2}{L^2}\right) \qquad \text{for latticed booms}$$

$$= 3\left(1 - \frac{R^2}{L^2}\right) \qquad \text{for telescopic booms} \qquad (d)$$

The full wind derate can now be stated as

$$\text{Derate} = D_t + D_b \qquad (8.1)$$

D_t is given, after combining Eqs. (*a*) and (*b*) and taking $C_f = 1.33$, by the equation

$$D_r = \begin{cases} \dfrac{v^2}{300} A\sqrt{\dfrac{L^2}{R^2} - 1} & \text{U.S. customary units} \\[2ex] \dfrac{4v^2}{5} A\sqrt{\dfrac{L^2}{R^2} - 1} & \text{SI units} \end{cases} \qquad (8.2)$$

After Eqs. (*a*), (*c*), and (*d*) are combined, D_b is stated in U.S. customary units by the formula

$$D_b = \begin{cases} \dfrac{v^2 b}{1200}\left(\dfrac{L^2 - R^2}{R}\right) & \text{for latticed booms} \\[2ex] \dfrac{3v^2}{800}\left(\dfrac{L^2 - R^2}{R}\right) & \text{for telescopic booms} \end{cases}$$

For SI units the formula is

$$D_b = \begin{cases} \dfrac{v^2 b}{5}\left(\dfrac{L^2 - R^2}{R}\right) & \text{for latticed booms} \\[2ex] \dfrac{9v^2}{10}\left(\dfrac{L^2 - R^2}{R}\right) & \text{for telescopic booms} \end{cases} \qquad (8.3)$$

An additional check must be made to assure that when the wind comes from the side, boom strength will not be overtaxed. Mobile-crane booms are designed to withstand a 20-mi/h (8.9-m/s) side wind in conjunction with boom tip side loading at 2% of rated load and the rated load itself. The side loading is intended to allow for dynamic effects of slewing with some attendant load swing. We can use this side-load allowance as a measure of boom resistance to winds above

20 mi/h together with wind on the lifted load provided we do not fail to realize that in so doing the dynamic allowance no longer remains.

With the above in mind, boom strength is adequate for latticed boom cranes if

$$\left[\left(\frac{v^2}{400}-1\right)\frac{d}{3}L+\frac{v^2}{400}A\right]50 \le \text{rated load} \qquad \text{for US customary units and } v \ge 20 \text{ mi/h}$$

$$\left[\left(\frac{3v^2}{5}-48\right)\frac{d}{3}L+\frac{3v^2}{5}A\right]50 \le \text{rated load} \qquad \text{for SI units and } v \ge 8.9 \text{ m/s} \tag{8.4}$$

For telescopic booms, replace $d/3$ with 3/2 in U.S. customary units and with 0.45 in SI units in the equations. The rated load in Eq. (8.4) is the printed chart value without deductions while d is the depth of the boom cross section. If the left side of Eq. (8.4) is considerably less than rated load, some allowance for dynamic effects remains, but if the two sides of the equation are nearly equal, a reduction for dynamic effects is essential under any set of operating conditions.

The procedure to be used at the jobsite in preparation for lifts in the presence of wind is as follows:

1. Determine or estimate wind velocity or select a value above which work will be stopped.
2. Determine by measurement the wind-exposure areas of the loads to be lifted.
3. Using Eq. (8.4), evaluate boom strength. For winds below 20 mi/h (8.9 m/s) only the load-area part of the equation need be used. If the left side of Eq. (8.4) is greater than rated load, reduce the operating radius and/or boom length or postpone the lift until wind subsides. If the left side is more than half of rated load, reduce the rating by not less than 10% to allow for slewing effects.
4. Using Eq. (8.1), calculate the value to be deducted from rated load to account for the overturning effects of the wind. If the reduced rating is less than the load to be lifted, reduce the operating radius and recheck or postpone the lift until the wind subsides.

Example 8.1

A crane having 150 ft (45.7 m) of latticed boom mounted is to be used for placing precast concrete panels that measure 12.5 by 8 ft (3.81 by 2.44 m) and weigh 7500 lb (3400 kg). The boom is 72 in (1.83 m) square in cross section, and the lower block and lifting accessories weigh 1200 lb (545 kg). Table 8.5 is the rating chart for the boom length mounted. What maximum operating radii can be utilized when ambient wind is at 15 and 30 mi/h (6.7 and 13.4 m/s)?

Solution

Load area A is 12.5(8) = 100 ft² (9.29 m²). When Eq. (8.4) is used, for wind at 15 mi/h (6.7 m/s), only the load-area part of the equation applies.

$$\frac{15^2}{400}(100)(50) = 2800 \text{ lb } (1275 \text{ kg}) \le \text{rated load}$$

But rated load must be at least equal to 7500 + 1200 = 8700 lb (3950 kg); therefore, no strength limitation will govern, and there is certain to be a considerable side-load allowance remaining to accommodate dynamic effects associated with reasonable swing speeds.

For stability, the derate applying to wind on the load is given by Eq. (8.2).

$$D_t = \frac{15^2}{300}100\sqrt{\frac{150^2}{R^2}-1} = 75\sqrt{\frac{22{,}500}{R^2}-1}$$

The derate for wind on the boom is given by Eq. (8.3).

$$D_b = \frac{15^2(72 \text{ in}/12)}{1200}\frac{150^2 - R^2}{R} = 1.125\frac{22{,}500 - R^2}{R}$$

Then rated load must be greater than the lifted load plus the derate values, or

$$RL = 8700 + 75\sqrt{\frac{22{,}500}{R^2}-1} + 1.125\frac{22{,}500 - R^2}{R}$$

From this equation, the following table has been generated:

Radius		Minimum Acceptable Rated Load		Radius		Minimum Acceptable Rated Load	
ft	m	lb	kg	ft	m	lb	kg
40	12.2	9600	4350	70	21.3	9150	4150
45	13.7	9450	4300	80	24.4	9050	4100
50	15.2	9400	4250	90	27.4	9000	4075
60	18.3	9250	4200	100	30.5	8950	4050

From comparison with Table 8.3, it can be seen that an operating radius of 90 ft (27.4 m) cannot be safely exceeded when ambient wind is at 15 mi/h (6.7 m/s).

Likewise, for wind at 30 mi/h (13.4 m/s), Eq. (8.4) yields

$$\left[\left(\frac{30^2}{400}-1\right)\frac{72/12}{3}150 + \frac{30^2}{400}100\right](50) = 30{,}000 \text{ lb } (13{,}600 \text{ kg}) \le RL$$

Equation (8.2) gives

$$D_t = \frac{30^2}{300}100\sqrt{\frac{22{,}500}{R^2}-1} = 300\sqrt{\frac{22{,}500}{R^2}-1}$$

and Eq. (8.3) becomes

$$D_b = \frac{30^2(72/12)}{1200}\frac{22{,}500 - R^2}{R} = 4.50\frac{22{,}500 - R^2}{R}$$

Operating Radius		Rated Load		Operating Radius		Rated Load	
ft	m	lb	kg	ft	m	lb	kg
40	12.2	36,400	16,500	100	30.5	8,400	3,800
45	13.7	30,400	13,800	110	33.5	7,050	3,200
50	15.2	25,900	11,750	120	36.6	5,950	2,700
60	18.3	19,500	8,850	130	39.6	5,050	2,300
70	21.3	15,400	7,000	140	42.7	4,250	1,900
80	24.4	12,400	5,600	150	45.7	3,650	1,650
90	27.4	10,100	4,600				

TABLE 8.3 Load Ratings for 150-ft (45.7-m) Boom, Side and Rear (for Example 8.1)

Therefore the rated load must be at least

$$RL = 8700 + 300\sqrt{\frac{22{,}500}{R^2} - 1} + 4.50\frac{22{,}500 - R^2}{R}$$

but the minimum acceptable rated load must also satisfy the strength-rating limitation as well as stability.

Radius		Minimum Stability-Rated Load	
ft	m	lb	kg
40	12.2	12,150	5500
45	13.7	11,700	5300
50	15.2	11,350	5150
60	18.3	10,800	4900
70	21.3	10,400	4750

Boom strength now clearly governs the selection of operating radius in the presence of an ambient wind of 30 mi/h (13.4 m/s). Thus operations can be conducted at radii not exceeding 45 ft (13.7 m) where rated load given by Table 8.5 is 30,400 lb (13,800 kg), which exceeds the minimum of 30,000 lb (13,600 kg) needed for lateral boom strength.

As a practical procedure for use at the jobsite, when loads of large wind area are to be handled, maximum operating radii can be calculated in advance for a series of wind speeds in increments between the limits used in the example problem. This will preclude the necessity for field calculations at the last minute while an operating crew stands by. It will also permit work to be safely performed with assurance when wind conditions might otherwise leave field crews uneasy about the crane or unwilling to proceed.

When the wind is gusty, it may be prudent to stop work even if the calculations show that adequate strength and stability are available at peak wind levels. Here the question is the stability of the load and whether it can be safely handled while being tossed about by the wind. The judgment of field supervisors and the operator must be relied on, and their decisions will have to be made at the time and place of work. It is not often that work can be done safely when wind velocity is 30 mi/h (13.4 m/s) or higher.

Stability When Operating on Rubber

Many truck, all-terrain, and rough-terrain cranes are rated for operations on rubber (tires), but those involved in such operations should be made aware of the greater sensitivity to overturning inherent in cranes in this configuration. Also, the limitations applicable to on-rubber ratings may vary from manufacturer to manufacturer. For example, ratings may apply within the entire rear quadrant, only for loads within the zone defined by extending imaginary lines through the centers of the tires, or within a defined angle, say ± 10°, from dead over the rear. Boom lengths may also be limited.

The cranes are capable of lifting their rated loads (of this there should be no doubt) since the ratings can be mathematically proved or verified by test. But a crane on rubber is on an elastic support; therefore any perturbation can cause the crane to lean out to a greater radius while the CG of the machine shifts closer to the tipping fulcrum at the same time. Few people realize that on-rubber ratings depend on a specific tire pressure that may be much higher than normal travel pressure. At the lower pressure, the crane cannot manage rated loads.

The danger from operations on tires derives from the heightened sensitivity of the machine to all things that are a danger for crane operations in general; in particular, they are susceptible to dynamic effects. If rated load reductions are recommended for a crane on crawlers or outriggers, they are imperative for a crane on rubber. On-rubber ratings are for static conditions.

Some cranes are rated to *pick and carry* (see Sec. 5.10). Those ratings apply for very slow travel speeds, as specified by the manufacturer, often on the order of 1 mi/h (1.6 km/h). And the ratings are typically for loads carried over front or rear, as specified.

Pick-and-carry operations require a firm level travel path. Small irregularities can start movements that amplify into serious side loads and radius changes if unchecked. Before executing a pick-and-carry movement with a meaningful load, the crane should be observed during an unloaded dry run. Unwanted load movements are minimized by minimizing travel speed. Snubbing the load to the crane undercarriage can prevent it from swinging to greater radius, but the inbound swing can cause shock loading.

On urban and highway reconstruction projects, space for crane operation is often constrained. Sometimes this leads to cranes improperly set up with outrigger beams extended on one side only or partially extended on each side. Some cranes are rated for specific partial extension positions that permit partial extension, but it must be done in accordance with the manufacturer's instructions. Unless sanctioned by the manufacturer, the crane should operate using on-rubber ratings with the outriggers considered as stabilizers that only reduce crane sensitivity. To do otherwise violates OSHA requirements. (See Extension of Outriggers in Chap. 4.)

Boom-over-Cab Accidents

The incidence of boom-over-cab accidents has declined because of the growing dominance of telescopic booms, which are not susceptible to this hazard. Boom-hoist cylinders provide a firm upper limit to the boom angle with no possibility of going further. On the other hand, luffing tower cranes have grown in popularity, and that the number of incidents among tower cranes.

A latticed boom crane is most susceptible to a boom-over-cab accident when the boom angle is close to the upper limit or the boom is in an upward motion. There are a number of events that could then trigger it, singly or in combination.

- Wind from the front
- Sudden release of the load
- Inertia of the boom in motion
- Improperly set boom-hoist cutout (*boom stop*) or failure of the cutout
- Accidental engagement of the boom-hoist in upward motion
- Two-blocking of the load hoist or whip line
- Uneven ground or bouncing if the crane is traveling
- On a crane with a live mast, the backward pull of the mast weight

Many people mistakenly believe that *boom backstops* will prevent boom-over-cab occurrences. They can when the contact of the boom against the stops is a low-impact event. Backstops are typically designed to prevent the boom from going over backward from a gust of wind or sudden release of a load at the high-boom angle. When a two-blocking situation continues or when the boom hoist continues in engagement (overluffing), the backstops or the boom may structurally fail.

Modern cranes are furnished with a boom-hoist cutout (*boom stop*) that arrests the luffing motion as the boom contacts the backstops. A functioning device of this type would preclude overluffing occurrences leading to boom-over-cab incidents. The operator's daily machine functional checkout should therefore include verification that the boom-hoist cutout is in operating condition; every maintenance inspection should also include examination of the elements making up this device.

A boom stop will do nothing to prevent boom-over-cab accidents caused by two-blocking, trapped or caught loads, rough travel paths, strong wind gusts, or excessive dynamic bounce. Cranes should never be traveled with the boom at maximum angle, since uneven ground, wind gusts, ground slopes, a lurching start forward, or a combination of these can then launch the boom over the cab.

Crawler-crane travel is ideally performed with the boom at that angle which will cause the track pressure to be uniform through the length of the track; this implies that the moment of the crane about the axis of rotation will be zero. Setting Eq. (5.23) equal to zero and inserting Eq. (5.21), the equation for the optimum boom angle is

$$\theta = \cos^{-1}\left[\frac{W_u d_u - (W + W_r)R}{W_b L_b} - \frac{t}{L_b}\right] - \theta_b \tag{8.5a}$$

This equation can be simplified to give the approximate equation

$$\theta \approx \cos^{-1}\frac{W_u d_u}{W_b L_b} \tag{8.5b}$$

resulting in satisfactory values for travel. When the arc cosine argument in either Eq. (8.5*a*) or (8.5*b*) is unity or greater, this means that the boom does not have sufficient weight to produce uniform track pressure even when extended out flat. The boom should then be carried at any reasonably low angle.

For truck cranes, the ideal travel mode will cause the axle loadings to be uniform (see Sec. 5.2), but this ideal should be relaxed if carrying the boom at or near the maximum angle is called for.

A related hazard is the possibility of a *suspended jib* going over backward. This can occur with both fixed jibs and luffing jibs. Some of the same causes of boom-over-cab accidents can cause a jib to flip backwards as well; the cautions and remedies are similar. Limiting devices and load chart limitations must take this hazard into account. A pendant-supported jib is typically fitted with a stopping device that serves the equivalent purpose as a boom stop.

Rope Failures

Strength factors of 3.0 and 3.5 are used for standing and running lines, respectively, in mobile-crane and derrick practice (however, for running lines of rotation-resistant rope 5 is used). The factors apply against rope break strength, while most engineers are accustomed to thinking in terms of factors against yielding, as used in structural-steel design. Yield strength for wire rope is about 60% of the break strength, so that a factor of 3.0 implies working loads that are equal to the yield strength divided by 1.80, or only about 8% more than the 1.67 used for structural steel in tension. For running lines, the factor against yield 2.10 is about 26% greater than 1.67.

Rope-strength factors stated in relation to break strength have misled many people into believing that rope is conservatively rated compared with ordinary structural steel. This is not so, and as the matter is looked into more deeply, it can be argued that rope is less conservatively rated.

All crane service rope is exposed to the weather, to alternate wetting and drying, changes in humidity, and variations in temperature. The rope itself is composed of a great many wires, giving it a large surface area compared with its *metallic area*. The corrosion susceptibility which weather exposure gives it, coupled with the large relative surface area, makes rope a corrosion-critical item; a small amount of corrosion causes a disproportionately large loss of strength. What is more, surface pitting accompanying corrosion and the resulting increased stress (pitting is a stress raiser) is exacerbated by the exaggerated relationship between pit and wire size. Corrosion considerations alone should take up an important part of the strength factor and is reason to be attentive to proper lubrication of ropes.

The nature of loading in crane service is such that the rope is continuously exposed to varying levels of tensile force. Rope is sized by considering static load only, so that in use stresses may exceed the nominal design values and after many cycles of loading ultimate failure due to fatigue becomes most probable. In addition higher levels of tensile stress will be found in ropes that wind over sheaves, and the point of attachment of fittings on standing lines usually includes stress-raising situations.

But these are all fatigue-related conditions, and fatigue is a useful-life type of phenomenon rather than strictly a safety consideration. Is this an issue of economy or safety?

Fatigue life becomes strictly an economic factor only if the rope is removed from service before fatigue failure or better yet, with a reasonable margin of safety still remaining before fatigue failure. Without timely removal, the rope may fail without warning and with dire consequences. The criteria that have been evolved to warn of the time for removal are based on rope selected with the strength factors given here. This results in sufficient remaining strength at the time of removal to ensure that the needs of safety are adequately served. To use lower strength factors together with the same discard criteria would imply less than satisfactory strength at removal and a higher risk of failure just before removal.

Experience indicates that for latticed boom cranes the likelihood of rope failure in the derricking system is substantially greater than for the hoist lines. This goes against the intuition that the boom suspension is a more sedentary system than the load hoist, as anyone familiar with crane operations is aware that derricking is performed far less often than hoisting. For this reason, and more particularly because the load-hoist drums of latticed boom cranes are within sight of the operator, load-hoist ropes are more closely observed. Signs of wear, broken wires, corrosion stains, kinks, bird caging, and other damage can be noted by the operator from the work station as the rope spools in.

The boom-hoist rope can be observed only after climbing to the roof of the crane superstructure, and the pendants cannot be examined unless the boom has been lowered to the ground. The inaccessibility of the suspension ropes makes it less likely that they will be inspected often enough and closely enough. Moreover, fatigue breaks may not be visible unless the inspector has the opportunity to watch the rope as it turns over a sheave.

It is only through inspection that ropes can be evaluated and the need for replacement determined. Telltale signs (discard criteria) reveal when fatigue damage has progressed to the farthest acceptable point, when wear has reached the limit that precludes further use and when other abnormalities require rope replacement. Some rope inspection is required daily before operations begin, and a formal program of rope inspection, evaluation, and record keeping compose an important part of all risk control programs.

Rope Inspection and Discard

Going back to the early days of mining and mine-shaft hoists, many groups have given consideration to the design of rope systems and to the criteria that define the point at which a rope must be removed from service. Usually it is not reasonable or economical to design for unlimited life, and for some aerial tramways the life of the rope was actually found to be increased by reducing the design strength factor.[12] Given the premise that rope will not last for the life of the structure or machine, it follows that timely rope replacement is a requisite for safe operation.

For machines that continuously perform repetitious service, rope replacement records will provide the data necessary to predict when future rope replacements will be required, even though rope life is a statistically random variable. But for general-purpose machines, there is no way whatsoever to predict life reliably; rope inspection provides the only viable means for determining when replacement must be made.

To this end, the American Society of Mechanical Engineers (ASME) B30 Committee, Safety Standards for Cableways, Cranes, Derricks, Hoists, Hooks, Jacks and Slings, has offered recommendations for rope inspection and discard programs for various types of equipment. The following is a compendium including materials extracted from various works produced by that committee but is edited and includes our own thoughts as well. For committee consensus,

[12]J. Kogan, On the Calculation of Cables for Lifting Appliances, *Neve Int.,* Turin, Italy, Spring 1977, p. 50.

the reader is referred to the ASME publications for each equipment type.

Inspection

1. Frequent inspection
   a. For machines that are in service, all running ropes should be visually inspected each working day, the inspection to include observation of all rope that is expected to be in use during the work to be performed that day. This procedure is for the purpose of discovering gross damage which may be an immediate hazard, such as
      (1) Distortion of the rope, that is, kinking, crushing, unwinding of strands, bird caging, main-strand displacement, core protrusion, loss of rope diameter in a short length, or development of uneven outer strands, any of which provides evidence that the rope may need to be replaced.
      (2) General corrosion
      (3) Broken or cut strands
      (4) Number, distribution, and type of visible broken wires
      (5) Core failure in rotation-resistant rope, indicating either that the rope should be removed from service immediately or that a more detailed inspection and evaluation must be made
   b. Care must be taken when inspecting portions of the rope subjected to rapid deterioration such as *flange points, crossover points,* and that part of the rope which first contacts the drum during repetitive lifts.
   c. Care must be taken when inspecting certain ropes such as
      (1) Rotation-resistant ropes because of their sensitivity during handling and their susceptibility to damage
      (2) Boom-hoist ropes because of the difficulty of inspection and the important nature of these ropes
2. Periodic inspection
   a. Inspection frequency should be determined by a person experienced with rope in practice on the basis of expected rope life, as indicated by past observations of this or similar installations, severity of the operating environment, relative frequency of capacity lifts, intensity of service, and exposure to impact loads. Inspections need not be at uniform calendar intervals but should occur more frequently as the rope nears the end of its useful life; the inspection should be made at least annually, however.
   b. The person performing periodic inspections should be particularly qualified for this task. The inspection must cover the entire length of the rope, but only the surface wires of the rope should be checked, and no attempt

should be made to open the rope. A determination must be made whether further use of the rope is permissible when deterioration resulting in appreciable loss of original strength is noted:

(1) Those points listed under frequent inspection
(2) Reduction of rope diameter below the nominal diameter due to loss of core support, internal or external corrosion, or wear of the outside wires
(3) Wires at end connections severely corroded or broken
(4) Severely corroded, cracked, bent, worn, or improperly applied end connections

c. Care must be taken in the inspection of portions of the rope subject to rapid deterioration such as the following, which are in addition to those mentioned under frequent inspection:

(1) Portions in contact with saddles, equalizer sheaves, or other sheaves where rope travel is limited
(2) Portions of the rope at or near terminal ends, where corroded or broken wires may protrude

Rope Replacement

1. Continued use of rope that has been in service depends entirely on the strength remaining in the rope, which in turn implies that this is largely a matter that relies on the good judgment of the person inspecting and evaluating the rope. Since many variable factors are involved in such a determination, there are no precise rules that can be applied to fix the exact time for replacement.
2. Conditions such as the following are reason for questioning the continued use of the rope or for increasing the frequency of inspection:
   a. In running ropes, six randomly distributed broken wires in any one lay or three broken wires in one strand of one lay.
   b. One outer wire that has broken at its contact point with the core of the rope and protrudes or loops out from the rope body. Additional and closer inspection of this portion is required.
   c. Wear of 1/3 of the original diameter of outside individual wires.
   d. Kinking, crushing, bird caging, or any other damage resulting in distortion of the rope structure.
   e. Evidence of heat damage from any cause.
   f. Reductions from the nominal rope diameter of more than shown in the table on the next page.

Reduction		Nominal Diameters			
		From		To	
in	mm	in	mm	in	mm
1/32	0.8	3/8	9.5	1/2	13
3/64	1.2	9/16	14.5	3/4	19
1/16	1.6	7/8	22	1 1/8	29
3/32	2.4	1 1/4	32	1 1/2	38

g. In standing ropes, more than two broken wires in one lay in portions away from end connections or more than one broken wire at an end connection.

3. Replacement rope must have a strength rating at least as great as the original rope and any deviation from the original size, grade, or construction should only be made on the advice of the equipment manufacturer, a rope manufacturer, or a licensed professional engineer practicing in this field.
4. All rope installed on a crane that has been shut down or in storage for a month or more should be given an inspection equivalent to a periodic inspection before it is placed in service.
5. Rope that has been removed from service on a crane should not be used to make slings.

Rope Safety Program

The information presented here should make it clear to supervisors and executives that rope safety rests almost entirely on timely performance of inspections by qualified people. The daily (frequent) inspection made by the operator does not require record keeping other than a notation on the daily crane-inspection check sheet. Periodic inspections should be recorded on dated signed reports. The preparation of a report formalizes the procedure and enhances the importance of the inspections in the minds of those who would otherwise think of them as a routine thing that must be done. This does not refer to the inspector, for any inspector who does not appreciate the importance of the task should be assigned to other duties.

Inspection reports also serve another purpose in our litigious society. They are prima facie evidence that a firm has behaved responsibly in respect to the wire rope used on its cranes, assuming that rope maintenance and use practices are proper as well. Excellent material for field rope practices as well as photographs and drawings of the rope defects discussed, can be found in the "Wire Rope Users Manual."[13]

[13] "Wire Rope User Manual Third Edition," Wire Rope Technical Board, Woodstock, MD, 1993.

Tower Crane Accidents

Overturning accidents, dominating the mobile-crane accident statistics, are almost unknown with tower cranes in North America.

Except during erection, dismantling, and climbing, the vulnerability of tower cranes to accidents is relatively small. But serious accidents do occur from operator misjudgment, inadequate maintenance, tampering with controls and limits, wind, and initial defects.

Without reference to any statistics, the experience of the authors suggests that the leading category would be mishaps related to erection, dismantling, and climbing. Although not necessarily the most numerous, they tend to be the most destructive, exceeding the other categories in injuries, fatalities, and property damage.

Erection, Dismantling, and Climbing Accidents

Erecting and dismantling tower cranes is a serious business, often difficult (Figure 8.2). By nature, it entails a series of *critical lifts*. As discussed in Chap. 6, these activities must proceed according to a complete operational plan that includes sequencing, weights, erection and dismantling crane positions, component staging locations, and erection and dismantling clearances. The crew must be experienced in rigging and equipped with the necessary hardware and tools. At least one person must be present who knows how to assemble and disassemble the mechanical, electrical, and structural systems.

Erection-related accidents can occur at any time, even after the riggers have left the site. Mishandling or incorrect installation procedures can create conditions that lead to later failure. Poor rigging

Figure 8.2 Telescopic mobile crane dismantles a luffing jib tower crane in an urban setting. The tower crane has a limited ability to swing because the newly erected building is in the way, and the position of the mobile crane is constrained by the limited space of the narrow city street. As can be seen, the operation must be carefully planned and communicated to the field crew. (*Mathieu Chaudanson.*)

practice can cause damage, but damage initiated during storage or transport is more common. One can conclude from this that a final visual inspection of structural components before the crane is put to use should be part of the erection process.

If damage is found, the manufacturer may need to be consulted to decide on the course of action because many components of tower cranes are highly stressed and some are exposed to fatigue-inducing conditions. If field repairs are feasible, they must be made in accordance with the manufacturer's or an engineer's procedures; nondestructive testing may be advisable. Repair decisions are critical and should be neither hastily nor lightly made.

Bolting is a common source of installation errors. High-strength designations and markings of nuts and bolts are not widely understood, particularly those with ISO grading. Few workers are aware that much greater harm can be done by undertightening bolts than by overtightening. Not surprisingly, the most common errors are undertightening and mismatching of nuts and bolts. There are several ways that mismatches can happen; common grade nuts are used on high-strength bolts, U.S. and metric components are mixed, or the threads on nuts and bolts are not matched.

Dismantling presents other hazards. The greatest difficulties arise because a building is now present where earlier there was vacant space. Movements and clearances are often hampered by the new construction. Bolts and pins, after having been in place for months in all weathers, can be locked in place and hard to remove, requiring more to undo time than anticipated and perhaps special tools. Chapter 6 discusses problems inherent in dismantling and particularly procedures required for jib removal. As with erection, planning is a prerequisite to a successful dismantling operation, but dismantling planning requires a higher level of practical expertise than erection.

During climbing operations, tower cranes have a heightened vulnerability to mishap. The vulnerability comes about because the equipment is relatively unsecured to allow for climbing movement and the means of support are continually shifting. The crane itself moves in close proximity to fixed objects that might conflict with the movement and the power of the climbing system can do harm if its force is not properly directed. Conditions conducive to safe climbing are good lighting, ample working time, little or no wind, and an experienced, well-placed, and alert crew. The entire climbing system and the support elements, including parts of the host structure that supports the crane, should be inspected in advance of the climb.

A subtle but important hazard associated with tower-crane erection and dismantling (and sometimes maintenance) is electric shock. For the great majority of tower cranes that are electrically powered, high-voltage power is brought up to the top of the mast with a flexible cable suspended on a steel cable. The electricity passes through a slip ring at the turntable and goes to a power supply box on the superstructure. Unless power is turned off and locked out on the ground, a person working on an electrical component of the crane is in potentially grave danger. Any entity that has technicians working on hookup or maintenance of electric tower cranes should ensure that procedures, training, and protective gear conform to applicable standards and best practices. Otherwise a serious mishap, perhaps fatal, is just a matter of time.

Other Causes of Tower Crane Accidents

Causes of tower-crane accidents are often the result of a combination of undesirable conditions. But, if an accident requires more than one condition to occur, correction of only one of the undesirable conditions should be enough to prevent the mishap. This principal is both a reason for optimism and a key to accident prevention.

For example, all tower-cranes are equipped with various automatic controls and limit switches to prevent overloading, overspeeding, and overtravel. A lazy and irresponsible operator will use these devices to control the crane; therefore if a device fails, an accident may ensue. The accident could be attributed to mechanical failure, but it is equally due to the failure of the operator to exercise control over the crane. Had the operator been performing properly, the accident would not have happened even though the mechanical failure occurred.

Another example concerns the prevention of fatigue failures. Some tower-crane components are normally subjected to a high number of stress reversals, conditions conducive to fatigue. Crane makers take heed of this and do the best they can to avoid fatigue through careful detailing and fabrication. Fatigue potential, however, cannot be eradicated, and fatigue itself is a process where incremental damage accumulates over time. Thorough inspection is therefore important, especially as the crane ages. But even abused cranes can be saved from acute damage if the incipient cracks are discovered during inspection between jobs.

There is no better time to avoid fatigue and other ills than the period between jobs when the crane is idle. This is the opportunity to make a thorough visual inspection of the entire crane structure. Should indications of possible damage be found, more stringent examination procedures can then be used.

All tower cranes, when out of service, must be left free to weathervane unless the crane and installation have been specifically designed otherwise. This means that swing brakes and drives must be disengaged whenever the operator leaves the cab and shuts down the prime mover. For luffing cranes, the boom should be left at the lowest boom angle consistent with an unobstructed weathervaning path unless the site installation instructions specify a particular boom angle. As the boom angle decreases, wind effects on the crane decrease accordingly; hence the boom should be lowered when not working.

A wind-related accident can occur while the crane is working or when it is not in service. The magnitude of wind force that might occur while a crane is working is not sufficient by itself to cause collapse, but it can cause the operator to lose control. If the load has a large wind-catching area, damage can be done by either the boom or the load because the wind force may overcome the swing brakes and the plugging resistance of the swing motors. The remedy is to take the crane out of service when the wind starts to make load control tenuous. The difficulty here is that supervisors, from their groundlevel vantage point, will not feel the same wind intensity.

Out-of-service wind failures can be reduced to a small probability, but never eliminated, because the design wind is statistically determined and therefore can always be exceeded. The installation designer must choose a rational wind level that balances risks and economy of design (see Chap. 3).

If a hurricane-level storm is known to be approaching, defensive measures can be taken. The crane supplier or installation engineer should be consulted; the crane configuration at that particular time may not be storm-wind limited, or climbing cranes may perhaps be made safe by climbing down one or two increments. Needless to say, tower wedges, braces, or other attachments should be checked before the storm arrives.

Tower-Crane Maintenance and Inspection

A strong inspection and maintenance program is a key to having safe tower cranes. This is especially true for the time between work assignments, with the crane on the ground where it can be carefully scrutinized and taken care of in the controlled environment of a shop. In boom times, there is a temptation to send cranes out quickly to satisfy demanding customers and to produce income. When work is slow, there is a reluctance to throw money at a machine that is sitting on the ground.

The particulars of crane maintenance and inspection are beyond the scope of this book. The primary reference should be the technical manuals from the manufacturer. Nonetheless, a sound preventative maintenance program might include the following:

- Review observations and complaints from the last crew that operated the crane.
- Visually inspect structural components with closer inspection (including paint removal and NDT at times) for suspect areas.
- Inspect and lubricate running and stationary ropes.
- Check sheave grooves and check bearings for play.
- Check turntable bolt for proper tightening torque.

- Review slew bearing maintenance history.
- Check brake shoes and adjustment.
- Check wires, connections, electrical boxes.
- Check hydraulic hoses and fittings.
- Inspect paint condition.
- Check electric motor–bearing, and brushes.
- Inspect gearboxes and gears.
- Inspect hook and wedge socket.
- Inspect tower bolts.
- Check counterweights for chips or spalls.
- Inspect slip ring.
- Inspect guardrails, ladders, cabin support, and platforms.
- Check manufacturer's service bulletins.
- Control console, instrumentation, and limit switches.

Some inspection and maintenance is best performed on an erected crane. For example, play in the slew bearing is readily checked with a dial gage while slewing the crane.

8.5 Lifting Personnel with Cranes

It is still possible to find an old retired ironworker who can boast of riding up to the top of the steel while standing on a headache ball. Needless to say, times have changed.

More recently, workers would ride on a platform or a bosun's chair suspended from a crane hook. Though clearly better than the earlier practice, many serious accidents demonstrated that considerable risk was involved in using a crane as a personnel lift.

The federal government amended the OSHA personnel lifting standards in 1988. The regulations are aimed at mitigating the dangers of personnel lifting without banning the practice outright. The ASME/ANSI B30.23, Personnel Lifting Systems, covers the same subject in detail. In Canada, the current practices dictated by industry standards and provincial enforcement are essentially the same as in the United States.

Casual lifting of personnel is not permitted. There must be a demonstrable reason for using a crane or derrick to lift people; it must be shown either that it is the least hazardous means for access or that there is no reasonable alternative. A proposed operation satisfying one of those criteria will be permitted only with strict controls. Those controls are designed to directly improve safety, but they are also sufficiently onerous to dissuade people from lifting personnel with a crane unless it is really necessary.

Cranes and derricks are subjected to stricter limitations when used for lifting persons as compared to material handling. Rope design factors are doubled and lift capacity is halved. Freefalling is not permitted in either the boom hoist or the load hoist. The lift platform cannot be an ordinary skip box but must be a specially designed platform used just for people and the tools and materials for the specific job.

Before the platform can be put to use at a jobsite, it is required to be proof loaded, followed by an inspection; the crane must be inspected as well. Each intended use has to be prefaced by a trial run through the anticipated motions using weights instead of people. Procedures are to be previewed in a meeting attended by the operator, lift supervisor, signalman, and people who are to be lifted.

Requirements for personnel lifting with cranes and derricks, comporting with OSHA, are summarized as follows:

1) General limitations
   a. The practice is prohibited except when no safe alternative is possible.
   b. Weight of the loaded platform and rigging cannot exceed 50% of *rated load*.
   c. When personnel on the platform are exposed to falling objects, overhead protection and hard hats are required.
   d. Only persons briefed on the personnel-lifting requirements and the task to be performed are allowed on the platform along with tools, equipment, and materials needed for the job. Materials and tools must be secured and evenly distributed while the platform is in motion.
   e. When operation is over water, apply 29 CFR 1926.106 or other applicable regulations.
   f. Lifting on another load line is not permitted when the crane is being used to hoist personnel.
2) Procedures
   a. When initially brought to the jobsite, after any repair or modification, and prior to hoisting personnel, the platform and rigging must be proof tested to 125% of the platform's rated capacity. Afterward, a competent person must inspect the platform and rigging for defects. Any problem must be corrected and another proof conducted before the platform is put into service.
   b. A trial lift of the unoccupied platform is made before any employees are allowed to be hoisted. During the trial lift, the personnel platform is loaded at least to the proposed lift weight. The lift starts at the location where workers will enter the platform and proceeds to each location where the platform is to be positioned. The trial lift must be performed immediately prior to placing personnel on the platform.

c. A trial lift is repeated when the crane is moved to a new location or returned to a previously one.
d. After the trial lift, the personnel platform is raised a few inches and inspected by a competent person. Any defect shall be corrected before the platform is put to use.
e. Before hoisting personnel, the hoist rope should be checked to be sure that it is free of kinks and twists and that the rope is spooling properly on the drum and through the reeving.
f. A prelift meeting shall be held with all workers involved in personnel-hoisting operations to review the requirements and procedures. This meeting must be held before the trial lift at each new work site and repeated for any persons newly assigned to the operation.

3) Operating practices

a. The crane operator shall be at the controls when the crane is running and the platform is occupied.
b. The operator must check that systems, controls, and safety devices are functioning properly.
c. Movement of the platform must be slow and cautious.
d. Brakes and dogs shall be engaged when the occupied platform is not in motion.

4) Crane requirements

a. The hoist rope is required to have a minimum safety factor of 7, or 10 for rotation-resistant rope.
b. The crane shall have a functioning anti-two-blocking device.
c. *Power lowering* is required. Freefall is prohibited. Boom lowering also must be powered.
d. A boom angle indicator is required. Cranes with telescoping booms shall have a device to indicate the boom extension.
e. Hooks must be closed and locked.
f. The crane shall be firmly supported and level.

5) Basket and rigging requirements

a. A personnel platform shall have permanent markings that indicate their weight and rated load capacity or maximum intended load.
b. The platform must be designed by a qualified person to have a minimum safety factor of 5. The suspension must be designed to resist tipping.
c. The platform shall have guardrails and be enclosed from the toeboard to the midrail. It also must have an inside grab rail, adequate headroom, and a plate or other permanent marking that clearly indicates its weight and rated-load capacity.

d. An access gate, if provided, cannot swing outward during hoisting and must have a restraint to prevent accidental opening.
e. Rough edges are to be ground smooth. Welding on the platform and its components must be performed by a qualified welder familiar with the platform design specifications.
f. A bridle used to connect the platform to the load line shall have its legs connected to a master link or shackle so that the load is evenly distributed to the bridle legs. Bridles and associated rigging for attaching the personnel platform to the hoist line must not be used for any other purpose.
g. Use tag lines unless their use is less safe.

6) Work practices while in the basket
   a. Occupants must stay entirely inside the platform when it is in motion.
   b. A body belt or body harness with a lanyard must be worn, with the lanyard attached to the load block or to a structural part of the platform.
   c. Before workers enter or exit to a work location, the platform should be secured to it unless that creates an unsafe condition.
   d. The person in control of the platform shall stay in direct communication with the operator or signal person.

7) Traveling with the load
   a. Personnel hoisting is prohibited while the crane is traveling except when it is demonstrated to be the least hazardous method to accomplish the task.
   b. Travel is restricted to a fixed track or runway.
   c. The boom must be parallel to the direction of travel.
   d. A complete trial run is required.

The comments given here are a summary. Personnel-lifting operations should not he planned or implemented without first reviewing the full text of the regulations.

8.6 Codes and Standards

In some countries, structural design is guided by *consensus standards* that are used in conjunction with local building codes, and in some other countries the national code is applied uniformly. The United States and Canada follow the first model, and Europe follows the latter. Consensus standards of the American Institute for Steel Construction (AISC) and the American Concrete Institute (ACI) provide practice guidelines for installation of cranes on structures (but not for the cranes themselves) in the United States. The authors endeavor to comport with those widely recognized standards in furnishing practical information and procedures in this book. In a similar vein, the Canadian Institute for Steel Construction (CISC) is a consensus standard used for steel design, with each province applying its own building code. In Europe, the Eurocode series of technical standards provides rules that are adopted by each member country of the European Union.

Much of the world uses Fédération Européene de la Manutention (FEM) standards for determining loading on cranes. In the United States, guidance for wind loading is to be found in ASCE7 from the American Society of Civil Engineers, but this document must be applied with care for other loads and load combinations, as it was not written with cranes in mind.

The International Standards Organization (ISO) through its Technical Committee 96 (TC96) has produced approximately a hundred model standards pertaining to design and use of cranes.

Several countries such as Japan, Canada, the United States, and Australia have national standards governing cranes. In U.S. practice, basic guidance on inspection, maintenance, and operation is to be found in ASME/ANSI Standards, which are nongovernmental consensus standards that have been adopted in large measure into OSHA regulations and enforcement.[14] Crane standards of individual European countries have been replaced by those of the European Committee for Standardization (CEN).

Appendix E lists consensus standards and mandatory standards from both international organizations and individual countries.

The common introduction to the U.S. standards includes a statement so basic to the use of cranes and derricks and so pertinent to the subject at hand that it should be quoted in full.

The use of cableways, cranes, derricks, hoists, hooks, jacks and slings is subject to certain hazards that cannot be met by mechanical means, but only by the exercise of intelligence, care and common sense. It is therefore essential to have competent and careful personnel involved in the use and operation of the equipment, physically and mentally qualified, trained in the safe operation of the equipment and the handling of the loads.[15]

8.7 Rational Methods of Risk Control

Attacking risk willy-nilly is a counterproductive exercise. In the extreme it amounts to a tangible risk being given the same consideration as one that is akin to the possibility of being struck by a watermelon falling out of an airplane. Risk assessment should be rational. This statement of the obvious could be restated with a premise introduced

[14]Safety Standard for Cableways, Cranes, Derricks, Hoists, Hooks, Jacks and Slings, American Society of Mechanical Engineers, New York.

[15]Ibid., with permission of the American Society of Mechanical Engineers.

at the beginning of the chapter. Effective risk management is a process of dealing with what might go wrong, the likelihood of it happening, and the possible consequences.

A common tool for addressing these questions is a *job hazard analysis (JHA)*, a general purpose approach to hazard mitigation that works by breaking down jobs into tasks, identifying hazards associated with each task, and then finding ways to mitigate them. The common JHA approach is suited to task-specific hazards but not so well to systemic ones. On a crane and rigging project, it is a useful technique for rooting out risks to individual workers—falls, crushing, falling objects and the like—but on a complex operation it does not adequately address operational risks, which are often deeply rooted and multifaceted and thus not identifiable by parsing into tasks.

This chapter concludes with an alternative form of JHA suited for major lifting operations (Figure 8.3). Rather than trying to identify particular hazards, it works by categorizing the overall level of risk, thus allowing the operation to be prioritized and assigned a suitable level of scrutiny. The method requires subjective judgments, and therefore is not foolproof. It should be tailored to the particular circumstances of each firm or jobsite. Used effectively, it can produce the dual benefits of risk reduction and directing management assets to where they can be most gainfully used.

Figure 8.3 A wind turbine being erected in Spain by a unique self-erecting tower crane designed for this purpose. Planning and coordination must be extensive to set up such a massive crane in a remote mountainous site. By nature, this is a major lifting operation. (*Manitowoc Corporation Inc.*)

Quantification of Risk Elements

The system requires that each of the items below be assigned a value from 1 to 10, on the basis largely of judgment of a qualified person. A value of 1 signifies that the item is of little significance or poses only minor problems. At the other extreme, a value of 10 signifies the highest worth or negative consequences of a failure.

Items to be assigned values:

1. *The money value of the load*. The replacement value if the load is destroyed including replacement shipping costs.
2. *The potential for injury and property damage* (other than to the load). Should the load be dropped or the crane or rigging equipment overturn, collapse, or otherwise fail.
3. *The potential for consequential damages*. Includes the costs of project delays due to loss or damage to the load or while waiting for repairs or replacements, loss of business, and damage to the property of others.
4. *The nature of the load*. Whether it is easy or difficult to handle, with well-defined lift points or difficult attachment conditions; or is irregular or unsymmetrical, a large wind catcher, flexible and sloppy; or are there a number of lift points requiring lift beams, special accessories, or equalizers; and how load weight relates to the rated capacity of the lift equipment.
5. *The nature of site conditions*. Whether it is a clear, level, open site or there are access problems, restrictions, obstructions, a constricted work area, or poor support conditions; this will be the only operation going on or there will be coordination with other activities needed; there are environmental restrictions or dangers, such as flammable gasses or liquids in adjacent pipes or vessels, or power lines; this is an open windy site or a protected one.

When assigning values, if the operation requires a multicrane lift, the nature-of-load value should be closer to 10; the lower end applies if it is a straightforward unloading operation using two cranes and the higher end as complexity increases.

This evaluation procedure assumes that the equipment to be used will be in proper operating condition, and that government regulations, manufacturer's instructions, and industry standards will be adhered to. If these basic requirements for all lifting are ignored, there is little point in trying to assess other risks. After values have been assigned to each of the five listed items, the values are totaled for use in one of the management action tables that follow, either the lifting contractor table or the project management table. The tables are set up using value ranges that separate projects into four groups on the basis of level of management participation or attention required.

Lifting Contractors

Range 0 to 15 and no single value greater than 6. Implies that the job is routine and that minimal management oversight is necessary; planning can be done by field personnel, but managers should nonetheless discuss plans and monitor the operation.

Range 16 to 25 but not more than one value of 6 or more. Implies that management involvement is needed to review the operation, together with the lift director, to identify risks and to plan measures that can be used to control or minimize them. Consideration should be given to the preparation of a formal plan of operations (lift plan) and/or having engineering participation in planning, otherwise engineering assistance may be advisable for reviewing plans, particularly when values are at the higher end of the range. Field operations should be monitored for compliance with plans and safety rules.

Range 26 to 35. Implies the need for preparation of a formal written operations plan (lift plan), made with the assistance or under the supervision of an engineer. The plan should cover step-by-step procedures and details required to control risk. Depending on circumstances and on the nature of the work, managers may elect to have the plan prepared by in-house staff and reviewed by an outside licensed engineer, but the extent of assistance needed depends on the level and kinds of risk. Field operations should be monitored for compliance with the operations plan and safety rules.

Range 36 to 50. Requires full, complete, detailed, in-depth planning and review. If the project or company does not have access to a qualified in-house engineering staff, outside engineering help should be secured early in the process. The analysis and review should include an effort to identify alternative schemes for doing the job and to explore available means to control and to minimize risk. Field operations should be monitored for compliance with the operations plan and safety rules.

Project Management

Range 0 to 15 and no single value more than 6. Implies that the job is routine and that only minimal oversight is necessary; for the most part, it can be left to field personnel or to the lifting contractor. Management should monitor field operations for safety in general.

Range 16 to 25 but not more than one value of 6 or more. Implies that management involvement is needed to review the operation, together with the lifting contractor, to identify risks and to discuss measures that will be employed to control or minimize them. Consideration should be given to requiring the lifting contractor to submit a formal plan of operations (lift plan) or have an engineer participate in planning. Engineering participation may be advisable to assist in management review, particularly when values are at the higher end of the range. Field operations should be monitored for compliance with the operations plan and safety rules.

Range 26 to 35. Implies that the lifting contractor should be required to prepare and submit a formal written operations plan (lift plan) with engineering participation. The plan should cover step-by-step procedures and details required to control risk. Depending on circumstances, managers may accept the plan without comment or review the plan with the lifting contractor. If the plan is reviewed, engineering assistance should be sought, but the extent of assistance needed depends on the nature of the work. Field operations should be monitored for compliance with the operations plan and safety rules.

Range 36 to 50. Requires full, complete, detailed, in-depth planning and review. Qualified in-house engineering staff or outside consulting engineers should be brought in early in the process. The analysis and review should include an effort to identify alternative schemes for doing the job and to explore available means to control and to minimize risk. Field operations should be monitored for compliance with the operations plan and safety rules.

This method was used to advantage on a project at the San Francisco-Oakland Bay Bridge where 16 separate crane setups were needed for erecting steel girders on a viaduct. The lifts varied from 46 tons (41.7 t) by 91 ft (27.7 m) long to 174 tons (158 t) by 231 ft (70.4 m) long. Some of the girders were curved, five of the positions required two cranes, and 13 involved erection over bridge access ramps or roadways.

Within the four value ranges, 0 to 15, 16 to 25, 26 to 35, and 36 to 50, seven operating positions were found to be in the 26 to 35 range while the remainder were between 16 and 25; the highest rating was 29 and the lowest 16. This quantification scheme identified the most critical parts of the work, focusing the attention of the planners and leading to some crane changes, positioning, and procedures. Better yet, the risk evaluation work convinced state highway authorities to relax traffic control restrictions they had imposed. Those restrictions would have materially added to risk and the likelihood of an accident and a massive traffic tie-up.

Managers and safety professionals often try to identify critical lifts in order to assign resources to where they are most needed to reduce risks. Unfortunately, critical lifts are difficult to identify by generalization. This evaluation system or one like it identifies them by degree. If one wishes to establish a dichotomy between ordinary and critical lifts, any value above 15 could be placed in the latter category.

This page intentionally left blank

APPENDIX A
Conversions

Conversions Between U.S. Customary, SI, and Other Metric Units

To Convert from	To	Multiply by
ACCELERATION		
Feet per second^2 (ft/s^2)	Meters per second^2 (m/s^2)	0.30480
Free fall, standard	m/s^2	32.174
		9.80665
AREA		
Square feet (ft^2)	Square meters (m^2)	0.092903
Square inches (in^2)	Square centimeters (cm^2)	6.4516
BENDING MOMENT, OR TORQUE		
Pound force–inches (lb · in)	Newton-meters (Nm)	0.11298
Pound force–feet (lb · ft)	Nm	1.3558
Kip force–inches (kip · in)	Nm	112.985
Kip force–feet (kip · ft)	Nm	1355.818
FORCE		
Kilograms force (kgf)	Newtons (N)	9.80665
Kips	N	4448.222
Pounds force (lb)	N	4.4482
LENGTH		
Feet (ft)	Meters (m)	0.30480
Inches (in)	m	0.02540
Miles, U.S. statute (mi)	m	1609.334
MASS		
Kips	Pounds	1000
Pounds mass (lb)	Kilograms (kg)	0.45359
Tons, long (= 2240 lb)	kg	1016.047

To Convert from	To	Multiply by
MASS		
Tons or tonnes, metric (t)	kg	1000.00
Tons, short (= 2000 lb)	kg	907.185
POWER		
Foot-pounds force per second (ft · lb/s)	Watts (W)	1.3558
Horsepower (= 550 ft · lb/s) (hp)	W	745.70
Horsepower (metric)	W	735.499
PRESSURE OR STRESS		
Atmospheres, technical (at)	Kilopascals (kPa)	98.0665
Bars	kPa	100.000
Kilograms force per square meter (kgf/m^2)	Pascals (Pa)	9.80665
Pascals (Pa)	Newtons per square meter (N/m^2)	1.000
Pounds force per square foot (lb/ft^2)	Pa	47.8803
Pounds force per square inch (lb/in^2)	kPa	6.8947
UNIT WEIGHT		
Pounds per foot (lb/ft)	kg/m	1.4881
VELOCITY		
Feet per second (ft/s)	Meters per second (m/s)	0.30480
Kilometers per hour (km/h)	m/s	0.27778
Miles per hour (U.S.) (mi/h)	m/s	0.44704
	Kilometers per hour (km/h)	1.609344
VOLUME		
Cubic feet (ft^3)	Cubic meters (m^3)	0.028317
Gallons, U.S., liquid (gal)	m^3	0.0037854
Gallons, U.K. and Canada, liquid (gal)	m^3	0.0045461

APPENDIX B

Glossary

A frame A triangular structural frame that supports the *topping lift* on cranes and derricks; in some instances, it can act as a *jib strut*.

Absolute limit of approach The area surrounding every live power line into which it is strictly forbidden to move any crane boom, load line, or load unless the line has been deenergized.

All-terrain crane A mobile crane with a truck chassis that is suitable both for highway travel and driving on rough terrain.

Alternate lay A type of wire-rope construction in which the strands are alternately regular lay and *lang lay*.

Anticollision system A set of devices that assist in preventing tower-crane booms and suspended loads on two or more cranes from fouling each other.

Anti-two-block device A device that, when activated, disengages all crane functions that can cause the hook block to strike the upper block or head sheaves.

Articulating boom crane A crane with a boom composed of a series of folding pin connected elements, manipulated by hydraulic cylinders.

Articulated jib A tower-crane jib that has a pivot point somewhere in its middle area; also called a pivoted luffing jib.

Assist crane A secondary crane used to help assemble a larger crane or to perform lesser tasks such as unloading trucks and shaking out hardware.

Attachment An accessory that can be installed or removed from a crane to augment the crane's capability to perform specialized tasks. Examples include jibs, towers, and heavy-lift components.

Auxiliary counterweights Supplemental counterweights, sometimes placed on a separate wagon or tray, that enable a mobile crane in a nonstandard configuration to increase its lifting capacity.

Auxiliary fall A second hoist rope connected to a secondary winch, also called a *Whip line.*

Axis of rotation The vertical line about which a crane or derrick swings. This is the centerline around which the crane's *superstructure* revolves. Also called Center of rotation and Swing axis.

Backstay A fixed length, nonrunning cable used to secure a mast or *jib strut* by tying it back to an anchoring structure such as a boom; when part of a jib suspension it is opposed by a *forestay*.

Backward stability Resistance to overturning of the crane in rearward direction.

Bail (1) A structural frame equipped with sheaves that is part of a crane boom suspension. (2) A U-shaped component of a bucket, socket, or other fitting.

Balance The proper state of a tower-crane *superstructure* necessary for climbing; the hook load, boom, and sometimes the counterweights are positioned to cause the superstructure to be in equilibrium with respect to the climbing mechanism.

Ballast (1) Weight added to a crane base to create additional stability; it does not rotate when the crane swings. (2) Material placed under rails, such as crushed stone, to support the rail ties or sleepers, to provide drainage, and to distribute the applied loads.

Base-mounted drum hoist A freestanding *winch* unit with integral controls and power plant.

Base section (1) The lowermost section of a telescopic boom; it does not telescope but contains the boom foot pin mountings and the *boom-hoist-cylinder* upper end mountings. (2) The lowermost section of a *latticed boom*, or *jib*, also called a heel section or butt section.

Basic boom The minimum length of sectional *latticed boom* that can be mounted and operated, usually consisting of a *base section* and *tip section* only.

Basic jib The minimum length of sectional latticed jib that can be mounted and operated, usually consisting of only a jib base and tip section.

Becket See Wedge socket.

Block One or more sheaves or pulleys mounted on a common shaft, used for lifting or pulling.

Blocking Wood or other material used to distribute loads to the ground. The loads could be from crane components (also called dunnage); the blocking could be infill to create even support under the *outriggers* or *tracks* of a mobile crane.

Bogie An assembly of two or more axles arranged to permit differential vertical wheel displacement while equalizing loading on the wheels.

Boom (1) On a *luffing* crane or derrick, a strut or spar used to project the upper end of the hoisting tackle; also called *jib* (European). (2) On a hammerhead tower crane, the horizontal structural member attached to the rotating *superstructure* on which the load trolleys travel when changing the load *radius.*

Boom angle The angle between the horizontal plane and the longitudinal centerline of the boom, boom base, or base section.

Boom angle indicator A device that provides a reading of the vertical boom angle; it can be a simple pendulum or an electronic device.

Boom backstop A device that physically limits the high boom angle of a crane in the event of a sudden release of the load or a strong frontal wind.

Boom base The lowermost section of a *latticed boom* with boom foot pins mounted at its lower end to connect it to the *superstructure*; also called *boom butt*, *butt section*, or heel.

Boom butt See Boom base.

Boom extension cylinder Hydraulic cylinder that powers the telescoping motion of a boom. See Extension cylinder.

Boom foot mast See Live mast.

Boom hoist Mechanism that luffs a crane boom with a running rope, drum, and mechanical drive or, alternatively, with hydraulic cylinders. Also called topping lift.

Boom hoist cylinder Hydraulic cylinder that luffs a crane boom by pushing on the upper end of the boom base section.

Booming See Derricking or Luffing.

Boom inserts See Insert.

Boom point The outer extremity of the crane boom containing the *head machinery.*

Boom stay See Pendant.

Boom stop A device designed to restrict the boom from moving above a certain maximum angle and toppling over backward. It may combine a *boom backstop* with a device that disengages the boom hoist power.

Boom suspension A system of ropes and fittings, either fixed or variable in length, that supports the boom and controls the boom angle.

Boom tip The uppermost section of a sectional latticed boom, which usually includes the weldment mounting the upper load sheaves as an integral part; also called boom point, head section, or tapered tip.

Boom truck A telescopic boom joined to a regular truck chassis.

Bottom climbing A jumping method for a tower crane utilizing a mechanism that pushes or pulls the entire crane up from the lower portion of the *mast*.

Bottom slewing A style of tower crane that has the *slew ring* under the *mast*.

Braced tower crane A tower crane secured by tie struts connecting the *mast* to an adjoining structure at regular vertical intervals; this arrangement allows the crane to exceed the maximum freestanding height.

Bridge crane A vernacular name for an overhead electric traveling crane (OET or EOT); it is composed of a girder ("bridge") riding on rails, a transverse *trolley*, and a load *fall*.

Bridle See Floating harness.

Bull Pole A pole, generally of steel pipe, mounted to project laterally from the base of a derrick mast. It is used to swing the derrick manually.

Bull wheel A horizontally mounted circular frame fixed to the base of a derrick mast to receive and guide the ropes used for swinging.

Butt section See Boom base.

Cab Housing, typically constructed of sheet metal or fiberglass, that protects the crane operator from the elements.

Cableway A lifting device that employs a trolley to transport the load on a *catenary* cable suspended between two *masts.*

Carbody That part of a crawler-crane base mounting which carries the *superstructure* and to which the crawler *side frames* are attached.

Carrier The undercarriage of a *truck crane, rough-terrain crane,* or *all-terrain crane* specifically designed for transporting the rotating crane structure.

Catenary The shape a hanging chain or cable assumes when supported at its ends and acted on only by its own weight.

Cathead An upper *block* suspended from a cantilever or guyed beam, usually projecting from a building or suspended over a hoistway.

Center of rotation See Axis of rotation.

Cheek weights (1) Supplemental weights, often steel plates, attached to the sides of a *load block* so that the block will overhaul. (2) Side-mounted counterweights on a mobile crane.

Cherry picker (1) A truck-mounted bucket lift on a telescopic or articulating boom. (2) A colloquial term for a small *telescopic boom crane.*

Chicago boom A basic derrick form with a boom and *topping lift* mounted directly to the side of a building.

Chord A main longitudinal structural member of *latticed boom* or jib.

Clamshell bucket A bulk extraction vessel for lifting loose material with a crane by closing together two shell halves, hinged at the top. Buckets are typically rated for size in cubic yards and are suspended from the crane boom by two lines, one for holding and one for closing the bucket.

Clearance The distance between the load or a moving part of the crane and another object.

Climbing A process whereby a tower crane is increased in height either by adding sections on the top of the *mast* or by raising the entire crane on the host structure.

Climbing base A moveable frame to support the weight of an inside climbing tower crane operating within the building structure. It might also function simultaneously as a *wedge frame*, *climbing frame,* or both.

Climbing frame (1) For freestanding, braced, or guyed tower cranes, a supplemental structure that envelopes the tower (mast) and is used to raise the *superstructure* of the crane for insertion of an additional tower section. (2) For internal climbing tower cranes, a frame used to raise the crane ; it might also double as a *climbing base* or *wedge frame*.

Climbing ladder A steel member with crossbars (used in pairs) suspended from a *climbing frame* and used as jacking support points for climbing some tower-crane models.

Climbing schedule A diagram or chart giving information for coordinating the periodic raising (climbing or jumping) of a tower crane with the increasing height of the building structure as the work progresses.

Climbing tower crane A tower-crane arrangement that includes a means to raise the crane's *superstructure* either by increasing the tower height (*top climbing*) or by raising the entire crane (*bottom climbing*).

Clutch A mechanism for engagement or disengagement of power.

Collar A rigid frame assembled around the *mast* of a tower crane; it is connected by guys or struts to a bracing structure.

Column derrick An adaptation of a *guy derrick* in which the guys are replaced by rigid ties to an adjoining structure.

Compacted wire rope A wire rope that has the wires of the strands (sometimes just the outer strands) deformed so that they are packed more tightly than a conventional wire rope; a rope of this type has enhanced strength and improved resistance to abrasion, fatigue, and crushing.

Consensus standard A document developed by volunteers in an organization to promulgate industrial rules; a consensus standard is not legally binding unless adopted by a government body.

Constructional stretch Elongation of wire rope when put under load, caused by constriction of the bundled wires and *strands* as the helical shape deforms. It excludes elastic deformation.

Counterbalance valve A pilot-controlled check valve that equalizes the fluid between the two chambers of a hydraulic cylinder.

Counterjib A horizontal member of a tower crane on which the counterweights and usually the hoisting machinery are mounted; also called Counterweight jib.

Counterweight Weight added to a crane *superstructure* to create additional stability. It rotates with the crane as it swings.

Counterweight jib See Counterjib.

Crane A machine designed to lift heavy loads and reposition them by traveling, trolleying, booming, or swinging.

Crawler crane A crane with a base mounting which incorporates a rigid central *carbody* supporting *side frames*. Each side frame is an armature carrying a continuous belt of track pads to travel the crane forward or rearward.

Cribbing Timber mats, steel plates, or structural members placed under mobile-crane tracks or outrigger floats to reduce the unit bearing pressure on the supporting surface below or for leveling.

Critical lift A lifting operation judged to carry a potentially high level of risk due to factors such as load weight, complex procedures, high value or presence of hazards. Though no uniform standard exists for identifying a critical lift, there is a consensus that risk can be mitigated by engineering, planning, and field controls.

Crossover points Locations on a wire rope that is spooled on a drum where one *layer* of rope climbs up on and crosses over the previous layer. This takes place at each flange of the drum as the rope is spooled into the drum, reaches the flange, and begins to wrap back in the opposite direction.

Dead end The point of fastening of one rope end in a running rope system, the other (live) end being fastened at the rope drum.

Dead guys On a *guy derrick*, guy cables on the same side as the boom; they are not actively supporting the boom and load.

Deadman An object or structure, either existing or built for the purpose, used as anchorage for a *guy line*.

Deadman control A spring-loaded mechanism in a control lever that automatically disengages power by returning the lever to the neutral position when the lever is released.

Deflector sheave A sheave that is placed to offset the direction of a running line.

Derate A reduction of *load rating chart* lifting capacities to account for environmental conditions such as wind, operating conditions such as *duty cycle*, or some other aggravating circumstance.

Derrick A lifting device that uses ropes for hoisting, with or without a boom, consisting of a mast or equivalent member held at the head by guys or braces. Unlike a crane, a derrick does not have an integral *winch*.

Derricking Changing the boom angle by activating the boom hoist winch; also called Luffing, Booming, and Topping.

Dog A *pawl* used in conjunction with a ratchet built into one flange of a rope drum to lock the drum from rotation in the spooling-out direction; also, one of a set of projecting lugs that support the weight of a tower crane during climbing or operation.

Dogged off The condition of a rope drum when its *dog* is engaged.

Dragline A material-handling crane equipped with a bucket that is suspended from the boom and pulled over the ground toward the crane to fill it. The filled bucket is then lifted clear of the ground, swung to the side, and dumped at a new location.

Drift (1) The vertical *clearance* between the top of a lifted load and the crane hook when the hook is in its highest possible position; a measure of the gap available for slings and other rigging or for manipulating the load. (2) In *telescoping gantries*, the lateral displacement at the top of a lift boom caused by *slop* between the sections.

Drifting Pulling a suspended load laterally to change its horizontal position.

Drive sprocket As applied to crawler cranes, the gear at the end of the track that engages the track shoes, guiding its change of direction while providing motive force. Also called a drive *tumbler*.

Drum The cylindrical member around which rope is wound for lifting or lowering a load or boom, or swinging the boom supporting structure. Also called Hoist drum.

Drum hoist A hoisting mechanism incorporating a motor and one or more rope drums; also called hoist, winch, or hoisting engine.

Drum rotation indicator An operator aid device on a crane or hoist that indicates in which direction and at what relative speed a particular hoist drum is turning.

Duty cycle Prolonged and highly repetitive work that imposes recurrent stress reversals on machine components from loading and unloading or from repeated patterns of movement; the extreme repetition of loading and unloading can place severe heat and wear demands on mechanical components; concerns about fatigue failure may compel the machine manufacturer to reduce load ratings.

Dynamic loads Those loads introduced in a crane or lifting system as a result of acceleration motions as opposed to simply static weight.

Efficiency The relative strength of a wire-rope termination or splice expressed as a percentage of the full rope strength.

EOT Crane Electric overhead traveling crane. Also called OET crane and Bridge crane.

Equalizer A device that equilibrates loads on tandem wheels or support ropes to compensate for unequal conditions such as length or stretch. On a *latticed boom* crane, this device is a frame with sheaves that mediates between the running *boom hoist* ropes and the fixed *pendants*. Also called a Bridle, Floating harness, or Spreader.

Equivalent mass In engineering, a simplified lump entity that is assigned the requisite physical properties to perform an analysis.

Expendable legs For a static-mounted tower crane, a set of leg segments cast into the concrete footing block. The legs are typically lost to future installations.

Extension cylinder Hydraulic ram used to extend a section of a telescopic boom; the most common but not the only means for power-extending boom sections. Also called a *Boom extension cylinder*.

Fairlead A device to guide wire rope for proper spooling onto drums, typically used in dragline work.

Fall The portion of a *hoist line* freely suspended from the *head machinery* that carries the *load block* and the load. It can have one or multiple *parts of line*.

Fixed-base tower crane A freestanding, braced, or *guyed tower crane* that is rigidly mounted on a foundation or structural frame and does not travel. See Static base.

Fixed harness A frame, forming part of the boom suspension, supporting sheaves for the live suspension ropes and moored to a fixed mounting such as a *gantry* or *mast*. See Lower bail.

Fixed jib A jib configuration pinned at the base to the tip of the main boom set at a fixed operating angle.

Fixed tower The mast of a tower crane that is not set up for climbing.

Flange point The point of contact between the rope and drum flange where the rope changes layers on a rope drum.

Flattop A *hammerhead tower crane* that lacks a *tower top*; the boom and *counterjib* are the highest parts of the crane.

Fleet angle The angle the rope leading onto a rope drum makes with the line perpendicular to the drum-rotating axis when the lead rope is making a wrap against a flange.

Fleeting sheave *Sheave* mounted on a shaft and arranged so that it can slide laterally as the rope spools, thereby reducing fleet angle.

Float An *outrigger pan* that distributes the load from the outrigger to the supporting surface or to *cribbing* placed beneath it; part of the crane's outrigger support system. Also called *Outrigger pad*.

Floating harness A frame, forming part of the boom suspension, supporting *sheaves* for the live suspension ropes and attached to the fixed suspension ropes (pendants); also called Bridle, Equalizer, Spreader, Upper spreader, Upper bail, or Live spreader.

Fly section (1) A boom tip extension, pin supported at its base. (2) On a telescope boom, the outmost powered telescoping section.

Forestay A fixed length, nonrunning cable used to support a jib by suspending the tip to a *jib strut*; the forestay is opposed by a *backstay*.

Footblock A steel weldment or assembly serving as the base mounting for a *guy derrick*, *gin pole*, or *Chicago boom* derrick.

Freefall Lowering the hook (load) or boom by gravity; the lowering speed is controlled only by a retarding friction device such as a brake.

Freestanding height The maximum height to which a tower crane can stand on a static base without mast support from braces or guys.

Freestanding tower crane A tower crane that is supported on a *static base* without benefit of lateral support of the mast such as braces or guys.

Frequency of free vibration Number of vibration cycles that will occur in a unit of time, usually expressed in hertz (cycles per second).

Friction Vernacular term for a mechanical device which utilizes high friction material for engaging and disengaging the drive in the transmission of power from a crane's engine to its operating functions; Also called a Main clutch.

Friction crane A mobile crane that relies on mechanical power transmission; the operator engages clutches and brakes to engage power and stop hoist motion.

Front-bumper counterweights Counterweights attached to the front bumper of the crane carrier.

Front-end attachments Optional load-supporting elements for use on the working end of mobile cranes, e.g., pile driver, *hammerhead boom*, guy derrick attachment, and *tower attachment*.

Gallows frame Two *gin poles* arranged in tandem with a *load block* mounted on a beam spanning between the heads.

Gantry (1) A structure fixed or adjustable in height, forming part of the *superstructure* of a crane, to which the *lower spreader* (carrying live boom-suspension ropes) is anchored; also called an A-frame gantry, A-frame, or back-hitch gantry. (2) The base support structure used to elevate a revolving or luffing boom crane. The gantry may either be fixed in one location or can travel on a rail truck system. (3) A bridge crane that is supported on two A-frames that usually roll on wheels or rail tracks.

Gantry crane A crane that is mounted on legs that in turn ride on rails.

Gate block See Snatch block.

Gin pole The most elemental form of derrick, a leaning mast incapable of swinging and held in place by guys.

Gooseneck boom A boom design with an upper section that projects forward at some offset angle from the normal longitudinal centerline of the boom. This type of design provides increased clearance between the boom structure and lifted loads.

Gradability The slope that a machine has the capability to climb. The slope is typically expressed as a percentage value.

Ground bearing pressure Unit pressure under track, wheel, or outrigger pad that is transmitted to the supporting surface.

Gudgeon pin The pin at the top of a derrick *mast* forming a pivot for the *spider* or for the mast of a *stiffleg derrick*.

Guy derrick A derrick consisting of a central *mast* mounted on a rotating base and held in a vertical position by a system of guy lines. A *luffing* boom, suspended from the top of the mast, is also pin connected to the rotating base to swing with the mast.

Guyed tower crane A tower crane secured by fixed-length cables connecting the mast to structural elements, *deadmen*, or anchors in the ground; this arrangement allows the crane to exceed the maximum *freestanding height*.

Guy line A fixed-length supporting rope intended to maintain a nominally fixed distance between the two points of attachment; also called Stay rope or Pendant.

Hammerhead jib (1) A tower-crane jib (boom) fixed at a horizontal or near-horizontal angle, equipped with a *trolley* to support the hook block and traverse loads radially from the tower. (2) See Hammerhead tip.

Hammerhead tip A *latticed boom* tip section used on mobile cranes where the load *sheaves* are mounted in front and the *pendants* are attached at the rear of the boom longitudinal centerline giving a hammer-like appearance.

Hammerhead tower crane A tower crane with horizontal boom and a load trolley that traverses the boom to change load radius.

Head machinery The *sheaves*, shafts, bearings, and other mechanical components mounted at the tip or head of a boom to reeve hoist ropes for lifting loads.

Head section See Boom tip section.

Head sheave A pulley from which the load line is directly suspended, carrying the hook or *load block*.

Head tower In a *cableway*, the mast that adjoins the winches for the load hoist and *inhaul/outhaul lines*; it may also lean to change the *catenary* sag of the *track cable*.

Headache ball A heavy spherical weight above the load hook on a single part hoist line that lowers the hook when there is no load on it. Also called Overhaul ball.

Heavy-lift attachment A system of accessories that, when fitted to a mobile crane, change its configuration and thereby enhances its lifting capability.

Heel section See Boom base.

Hog line See Pendant.

Hoist (1) To lift an object. (2) The mechanism used for lifting. (3) A term used, sometimes synonymously, for Winch.

Hoist drum See Drum.

Hoist line A rope used to lift loads. On a crane or derrick, the hoist line runs from the hoist drum, over the boom tip sheave(s), and down to the *overhaul ball* or into a reeving with the hook block.

Hoisting The act of raising or lowering a load in a vertical direction through the use of some type of rope and sheave system.

Holding brake A static brake intended to hold a drum or motor once it has been stopped by electric or hydraulic power.

Holding valve A valve that maintains hydraulic or pneumatic pressure to prevent unintended motion in the event of sudden pressure loss.

Hook block A device for suspending a load from a *fall*; it is composed of one or more *sheaves* mounted on a common shaft with a load hook at the bottom. Sometimes *cheek weights* are attached to assist *overhauling*. The hook block is reeved with wire rope that suspends it from the *head sheave*. Also called Load block and Lower block.

Hook rollers Steel rollers that run on the underside of some type of lip or rail to resist uplift forces on various areas of a crane structure.

House assembly See Superstructure.

House rollers A set of circumferential steel rollers mounted to the underside of the *superstructure* and bearing on the *carrier* or *carbody*.

Hydraulic crane The vernacular name for a crane for a telescoping cantilevered boom crane; the telescoping of the boom and the various other motions of the crane are typically powered by hydrostatic (hydraulic) drives.

Hydraulic cylinder A fluid power device that applies linear force and linear motion.

Idler sprocket As applied to crawler cranes, the gear at the opposite end of the track from the *drive sprocket* that engages the track shoes, guiding its change of direction but providing no motive force. Also called an idler *tumbler*.

Impact Increase in load effect from dynamic causes.

Inhaul-outhaul line On a *cableway*, a rope that pulls the trolley and load toward one of the masts.

Inner tower In one style of top-climbing tower crane, a mast that supports the *superstructure* and that climbs on (and is supported by) an *outer tower*.

Insert An intermediate section of a *latticed boom*; transported separately, it is assembled on-site with the *basic boom* and other inserts to attain the full boom length.

In-service The condition of a crane ready for or engaged in work with an operator at the controls.

In-service wind Wind encountered while a crane is working; usually used to define the maximum permissible level of wind pressure or velocity at the site before the crane must be taken out of service.

Intermediate fall A second hoist line that descends from a *sheave* of sheaves located at a middle location along a crane's boom length; its purpose is to overcome minimum radius limitations of the main fall. Also called a *mid-fall*.

Intermediate suspension An additional set of boom-suspension lines attached to the boom at some point between the main suspension attachment and the boom foot. On mobile cranes it is used to reduce boom elastic deflection during erection; on horizontal jib tower cranes it is used as part of the primary support; also called Midpoint suspension, Midpoint hitch, Intermediate hitch, or Intermediate hog line.

Internal climbing A process of climbing by a tower crane from floor to floor in a building as construction advances; once begun the *mast* is supported on the building and no longer bears on the ground.

Isotach lines On a map of wind velocities, the lines that connect locations of equal velocity.

Jacking tower An arrangement of towers, header beams, and a jacking mechanism used for vertical hoisting of a load.

Jib (1) In U.S. practice, an extension to the boom mounted at the boom tip, in line with the boom longitudinal axis or offset to it. (2) On tower cranes, the structural member extending forward from the mast to support the lifting trolley, sheaves, hook block, and load. (3) In Europe, the term used for a crane boom. Correspondingly, the extension to the main boom is known as a fly jib.

Jib angle See Offset angle.

Jib backstay pendant The wire-rope *pendant* that secures the jib mast tip back to the boom of a mobile crane. It acts in conjunction with the jib forestay pendants to support the jib tip and lifted load. Also called Jib Backstay Ropes.

Jib forestay pendant The wire-rope *pendant* that secures the jib tip to the *jib mast* tip on a pendant supported mobile crane jib. It acts in conjunction with the jib backstay pendants to support the jib tip and lifted load.

Jib mast See Jib strut.

Jib offset See Offset angle.

Jib strut A short strut or frame mounted on the boom head to provide a means for attachment of the jib support ropes; also called Jib mast, Rooster, Horse, or A-frame.

Jump The action of raising a *tower crane* or *guy derrick* from one working elevation to the next in concert with the completion of additional floors of the building.

King pin The center pin that holds a *turntable* in alignment. A crane with a *slewing ring* does not have a king pin.

Knuckle boom The vernacular name for an articulating boom that extends and retracts its reach by changing the angle between segments by mechanical or hydraulic means.

Lacing Transverse or diagonal structural members which connect the *chord* members to form a latticed boom section.

Lagging Removable shell in the form of a spool on which rope is wound on a hoist drum. Laggings have either smooth or grooved barrel surfaces.

Lang lay A rope construction in which the lay of wires in each *strand* runs is in the same direction as the strands.

Latticed boom A boom constructed of four longitudinal corner members, called *chords*, assembled with transverse and/or diagonal members, called *lacings*, to form a truss work in two directions. The *chords* carry the axial boom forces and bending moments, while the lacings resist the shear forces.

Latticed extension A trussed cantilevered jib used to extend the reach of a telescopic boom; oftentimes it stows on the side of the boom and swings into place. See Swingaway jib.

Lay The pattern in which the strands of a wire rope are arranged.

Layer A series of wraps of wire rope around a rope drum barrel, extending full from flange to flange.

Lead A latticed structure used in pile driving to guide the hammer as it drives a pile into place, typically supported from the boom tip.

Lebus grooving A proprietary hoist drum groove pattern that has parallel grooves with a crossover area that shifts the winding rope to the next groove.

Level luffing An operating mode of an articulating jib that allows it to change radius while maintaining the hook at a constant elevation.

Lift boom In some *telescopic gantries*, a telescoping box structure that houses a lift cylinder while providing lateral strength and stability.

Lifting capacity The maximum gross load weight that a crane manufacturer has determined a crane can safely handle under specified conditions as stipulated in the load chart.

Lift crane A crane configured for lifting, booming, swinging, or traveling with loads attached to the crane hook block as contrasted to a crane configured for lifting material with a bucket or grapple.

Lift cylinder In a *telescoping gantry*, a vertical hydraulic cylinder that provides the impetus to raise load; lift cylinders are generally arranged in pairs or fours.

Limiting device A mechanical device that is operated by some part of a power-driven machine or equipment to control loads, or motions of the machine or equipment.

Limit switch An electromechanical device that senses when an extreme of some operation is reached and sends a control signal to restrict that operation.

Line pull The pulling force attainable in a rope leading off a rope drum at a particular pitch diameter or layer.

Line speed The speed attainable in a rope leading off a rope drum or lagging at a particular layer on the hoist drum.

Live guys On a *guy derrick*, guy lines opposite the boom that actively support the boom and load.

Live mast A component of some mobile-crane boom suspensions. It consists of an articulating frame or strut hinged near the boom foot that serves to raise the boom suspension ropes, thereby increasing the angle those ropes make with the boom. Its purpose is to decrease the axial compressive force on the boom. Also called a *Boom foot mast*.

Live spreader See Bridle and Floating harness.

Load (1) The suspended weight applied to the crane, including the weight of lifting hardware such as hook block, shackles, and slings. (2) An external agency that induces force in members of a structure. It may be in the form of a direct physical entity (e.g., weight superimposed on the structure or pressure applied by wind) or in the form of an abstract condition (e.g., inertial effects associated with motion).

Load block See Hook block.

Load hoist A hoist drum and rope reeving system used for hoisting and lowering loads.

Load line Another term for hoist rope. In lifting crane service it refers to the main hoist. The secondary hoist is referred to as a *whip line*.

Load moment indicator A system that aids the crane operator by sensing the overturning moment on a crane, that is, load and radius. It compares the lifting condition to the crane's *rated capacity* and indicates to the operator the percentage of capacity at which the crane is working.

Load radius See Radius.

Load rating The maximum permissible lifted load under prescribed conditions and limitations.

Load rating chart A tabulation of load ratings provided in a formal document.

Lockout When lifting from a crane with sprung axles or bogies, the action of mechanically blocking the axles, thereby increasing stability.

Lower bail See Lower spreader.

Lower spreader A frame, forming part of the boom suspension, supporting *sheaves* for the live suspension ropes and attached to the gantry or *superstructure*. Also called Spreader, Lower bail, or Fixed harness.

Luffing See Derricking.

Luffing attachment A front-end attachment for a mobile crane that uses an upper working boom or jib that is capable of changing angle during operation and is mounted on top of a lower main boom.

Luffing jib A tower crane boom or mobile crane jib that is raised and lowered about a pivot to move the hook radially, that is, to change its working *radius*.

Luffing tower crane A crane with a boom pinned to the *superstructure* at its inner end and containing load hoisting tackle at its outer end and with a hoist mechanism to raise or lower the boom in a vertical plane to change load *radius*.

Machine resisting moment See Stabilizing moment.

Main clutch A clutch that engages the engine to the power train.

Main fall The hoist rope spooling off the main drum; the primary hoisting rope.

Main hoist The primary hoist drum system of a crane used for lifting the heaviest loads typically on the main boom.

Manual extension A nonpowered outermost section in a telescopic boom or jib which extended or retracted manually to either a fully extended or a fully retracted operating position.

Mast (1) An upright load-bearing component of a crane or derrick. (2) The tower of a tower crane. (3) A frame or strut hinged at or near the boom foot that raises the topping lift while supporting a boom.

Mast cap See Spider.

Mast tie Struts that secure a tower crane mast laterally to a supporting structure, usually a building.

Mat Individual timbers fastened together into units, steel plates, or woven wires placed under crawler tracks, wheels, or *outrigger pads* to increase the bearing area contacting the ground.

Messenger line On a *cableway*, a cable running between the *masts* that carries electrical and signal lines.

Metallic area Sum of the cross-sectional areas of individual wires in a wire rope or strand.

Mid-fall A load fall positioned in the middle of a luffing jib to overcome minimum radius limitations of the main fall. Also called an *intermediate fall*.

Mid-point suspension See Intermediate suspension.

Mid section On a telescopic boom, the intermediate powered telescoping section(s) mounted between the base and fly sections.

Minimum radius The shortest distance from the crane centerline of rotation at which a crane with a given boom length can lift a load.

Mobile crane A crane capable of either travel or transit under its own power. In Europe, a crane mounted on a truck carrier.

OET crane Overhead electric traveling crane. Also called EOT crane and Bridge crane.

Offset angle The angle between the longitudinal centerline of a jib and the longitudinal centerline of the boom on which it is mounted. Also called a *Jib angle.*

Off lead A condition whereby the hook and hoist line are out at a greater radius than the *head sheave*.

Offset tip A style of boom head that has the *head sheaves* mounted forward of the center axis; its purpose is to lessen the possibility that loads raised close to the tip will strike the boom.

On rubber The disposition of a truck crane set up for lift service while supported on tires.

Open throat tip A style of boom head that is open in the front below the *head machinery*; its purpose is to lessen the possibility that a raised load block will strike the boom.

Operating radius See *Radius*.

Operational aid An accessory that provides operational information to a crane operator or automatically prevents the crane from exceeding a preset physical limit.

Operating sector Portion of a horizontal circle about the *axis of rotation* of a mobile crane providing the limit of a zone where over-the-side, over-the-rear, or over-the-front ratings are applicable.

Outer tower The *mast* of a style of *top-climbing* tower crane that has the superstructure mounted on a climbing *inner tower.* The inner tower in turn is supported on the outer tower; the mast height is increased by adding to the latter.

Out of service The condition of a crane when unloaded, without power and with the controls unattended, and prepared to endure winds above the *in-service* level.

Out-of-service wind The wind speed or pressure that an upright inoperative crane is exposed to; usually used to define the maximum level of wind that the inoperative crane is designed to safely sustain.

Outriggers Extensible arms attached to a crane base mounting, that include means for relieving the wheels (crawlers) of crane weight; used to increase stability.

Outrigger beam The part of an outrigger assembly that extends horizontally from the crane body to the vertical outrigger jack.

Outrigger jack The part of an outrigger assembly that extends vertically to engage the pad on the ground.

Outrigger pad See Float.

Overhauling The application of force or weight on a reeved system to overcome resistance of weight and friction.

Overhauling weight Weight added to a load fall to overcome resistance and permit unspooling at the rope drum when no live load is being supported; also called headache ball; see Cheek weights, Load block, and Headache ball.

Overturning moment The summation of the individual moments that add to the tipping tendency of a crane about its fulcrum, opposed by a *resisting moment*. Load weight, boom weight, wind, dynamic effects and deflection are potential contributing factors.

Over wound a spooling arrangement of rope whereby the rope winds out from the top of the drum.

Parking track For rail-mounted traveling cranes, a section of track supported so that it is capable of sustaining storm-induced *bogie* loads; it is provided with storm anchorages when required.

Parts of line A number of *running lines* (ropes) supporting a load, generally running through two opposing blocks.

Pawl A ratchet device used for positively holding a machine element or structure against undesired motion. It is often used to keep a rope drum from *paying out*.

Paying out Adding slack to a line or relieving load on a line by letting (spooling) out rope.

Pedestal crane A crane with the slewing ring and *superstructure* elevated on a fixed-height support structure.

Pendant A fixed-length rope forming part of the boom-suspension system; also called Boom guy line, Hog line, Boom stay, Standing line, or Stay rope.

Pick and carry A descriptive term for mobile crane operations that involve suspending a load on the crane hook while the carrier or crawler is in motion.

Pitch diameter The diameter of a *sheave* or rope drum measured at the centerline of the rope; tread diameter plus rope diameter.

Pontoon (1) A pan which attaches to the bottom of a vertical *outrigger jack* to distribute loads over the supporting surface. See Float. (2) Steel or timber members arranged for distributing crane tire, track, or outrigger loads to reduce the *ground bearing pressure*.

Portal crane A crane type having of a rotating *superstructure* and boom mounted on top of a gantry frame, either fixed in one location or traveling. The space between the *gantry* legs is usually open to allow passage of vehicles.

Power lowering A system for driving a load down under mechanical power and thereby controlling the speed of descent using engine power to retard speed instead of brake friction.

Preformed wire rope Wire rope that has been shaped during its manufacturing into the helical form of the finished rope.

Prestretched wire rope Wire rope that has gone through a process in manufacturing that takes out a portion of the *constructional stretch*.

Preventer In rigging practice, a means, usually a wire rope, for preventing an unwanted movement or occurrence or acting as a saving device when an anchorage or attachment fails.

Quadrant of operation Four azimuthal segments of the circle that define the working areas through which a crane rotates during the operation such as over the rear, over the side, and over the front. Some cranes have load ratings differentiated by quadrant.

Radius The horizontal distance from the *axis of rotation* to the center of gravity of a freely suspended load. In mobile-crane practice, this is defined as the horizontal distance from a projection of the centerline of rotation to the ground, before loading, to the center of a vertical hoist line while under load. Also called a Load radius and Operating radius.

Rail clamp A tonglike metal device mounted on a locomotive crane car, which can be connected to the track.

Rail-mounted crane Freestanding crane erected on a power-driven rail-mounted undercarriage.

Range diagram A diagram showing an elevation view of a crane with circular arcs marked off to show the *luffing* path of the tip for all boom and jib lengths and radial lines marking boom angles. A vertical scale indicates height above ground, while a horizontal scale is marked with operating radii. The diagram can be used to determine lift heights, clearance of the load from the boom, and clearances for lifts over obstructions.

Ratchet A toothed member attached to or part of the drum, for engagement with the pawl.

Rated capacity See Rated load.

Rated load The maximum allowable working load on a crane or rigging component designated by the manufacturer under specified working conditions; expressed in pounds, kilograms, short tons, or metric tons.

Reach Distance from the centerline of rotation of a crane or derrick; sometimes used synonymously with Radius.

Recurrence period The interval of time between occurrences of repeating events; the statistically expected average interval of time between occurrences, such as storms, of a given magnitude.

Reeve/reeving (1) The pattern followed by a rope over pulleys or *sheaves* and drums. (2) The act of pulling a rope over pulleys or sheaves and drums.

Reeving diagram A drawing that depicts the path of a rope threaded through a system of drums and sheaves.

Regular lay An arrangement of the wire rope *strands* in which the individual wires appear on the surface to be parallel to the long axis of the rope.

Resisting moment The summation of the individual moments that add to the stabilizing tendency of a crane about its tipping fulcrum, acting against the *overturning moment*. Machinery, structure, *ballast*, and *counterweights* are potential contributing factors.

Revolver crane A crane with a latticed boom and a large *slewing ring* mounted on a gantry, pedestal, or barge and used, generally, for marine or industrial applications.

Revolving superstructure See Superstructure.

Roller path Circular steel rail-type surface on which the *house rollers* and *hook rollers* of a crane's rotating upper structure travel.

Rooster Vernacular term for (1) one or more struts at the top of a boom or *mast*, such as a *jib strut*, a tower-crane top tower, or the struts at the top of the mast of a mobile-crane *tower attachment.* (2) A short tip extension to a boom with a sheave to suspend the *whip line*; an extended upper boom point.

Root diameter See Tread diameter.

Rope drum That part of a drum hoist that consists of a rotating cylinder with side flanges on which hoisting rope is spooled in or out (wrapped).

Rotation-resistant rope A wire rope consisting of an inner layer of *strands* laid in one direction covered by a layer of strands laid in the opposite direction. This has the effect of counteracting torque by reducing the tendency of the finished rope to rotate.

Rough-terrain crane A mobile crane with a chassis that has oversized tires for driving over uneven terrain. The carrier is incapable of driving at full highway speed.

Running line A cable that runs over *sheaves* or spools on and off a drum.

Saddle jib A type of jib on a tower crane that is supported by *pendants*. The jib is horizontal or nearly horizontal, nonluffing, and the load hook is suspended from a *trolley* that moves along the jib.

Self-erecting crane A tower crane that assembles and raises itself on the site, usually requiring no *assist crane*.

Shackle A component of rigging hardware consisting of a U-shaped steel clevis with a pin passing through the open ends of the U. A shackle is used to temporarily attach other rigging components such as wire rope or chain slings to fixed lugs or fittings.

Sheave A wheel or pulley with a circumferential groove designed for a particular size of wire rope; used to change direction of a running rope.

Side frame Part of the base mounting of a crawler crane attached to the *carbody* and supporting the *track shoes*, the *track rollers*, and the drive *sprockets* and *idler sprockets*. Crawler frames transmit crane weight and operational loadings to the ground.

Side guys Ropes supporting the flanks of a boom or mast to prevent lateral motion or lateral instability.

Side loading A loading applied at any angle to the vertical plane of the boom.

Sill One of the horizontal stationary members of a *stiffleg derrick*, it forms the bottom leg of a triangle composed also of the vertical *mast* and the diagonal stiffleg.

Skewing Deviation of a traveling crane from its ideal alignment on the track.

Slack carrier On a *cableway*, a hanger that suspends the operating cables from the carrier line.

Slew bearing The large diameter bearing used to transmit axial force and overturning moment in a *slewing ring*.

Slewing see Swing.

Slewing assembly Those structural and mechanical components of a crane or derrick that enables it to be rotated about its *center of rotation*.

Slewing ring A large-diameter roller or ball bearing incorporating ring gear to form the rotating connection between a crane's revolving upper structure and a nonrotating base.

Slewing torsion Moment acting in a horizontal plane induced by the slewing mechanism of the crane. Wind, side loading, and inertia can be contributing factors.

Sling A rope or strap with integral loops at both ends used for supporting a load.

Slop Clearance and wear at slider pads of telescopic booms; it causes deflection at the boom tip.

Snatch block A single- or double-sheave *block* arranged so that one or both cheek plates can be opened, permitting the block to be reeved without having to use a free rope end; also called a Gate block.

Spider A fitting mounted over a pivot (*gudgeon pin*) at the top of a derrick *mast* to allow rotation of the mast underneath and to provide an attachment point for guy lines; also called a Mast cap.

Split drum Two independently controlled hoist drums mounted side by side on the same shaft.

Spooling capacity The maximum length of rope that can be spooled onto a hoist drum without exceeding the capacity of the drum.

Spreader See Floating harness, Lower spreader.

Spreader bar A strut-type structural member used in compression to separate two static ropes.

Stability Ability of a crane to resist overturning.

Stabilizers (1) Auxiliary devices for increasing *stability* of a tower crane, such as guys. (2) Extendable or fixed members attached to the mounting base to increase the stability of a crane, but that may not have the capability of relieving all of the weight from wheels or tracks.

Stabilizing moment The effect of a crane's weight, including *ballast*, which prevents it from overturning when lifting loads.

Standing line A fixed-length line that supports loads without being spooled on or off a drum; a line of which both ends are dead; also called a guy line, *stay rope*, or *pendant*.

Static base Tower-crane bottom support where the crane mast is rigidly set into a foundation or structural frame with or without knee braces.

Stay rope A fixed-length rope forming part of a suspension system; also called a Hog line or Stay rope.

Stiffleg derrick A derrick configuration where the boom is suspended by a *topping lift* supported on a *mast*; the mast is held in place by two diagonal struts at right angle to each other or nearly so.

Strand A group of wires helically wound together forming all or part of a wire rope.

Strand jack A vertical lifting apparatus that uses hydraulic jacks to pull cables or strands in repetitive strokes.

Strength factor Failure load (or stress) divided by allowable working load (or stress).

Structural competence The ability of the machine and its components to withstand the stresses imposed by applied rated and dynamic loads.

Superstructure The portion of the crane that rotates including the operating machinery mounted thereon. Also called Upper, Upperworks, Upperstructure, and House assembly.

Suspended jib A jib that is *pendant* supported.

Swing A crane or derrick function where the boom or load-supporting member rotates about a vertical axis (*axis of rotation*); also called Slewing.

Swingaway jib A jib that travels on the side of a telescopic boom and pins into place as a cantilever extension of the boom tip. See Latticed extension.

Swing axis See Axis of rotation.

Swing lock A mechanical device that can be engaged to resist slewing moment, thereby preventing the superstructure from swinging.

Tackle An assembly of ropes and sheaves arranged for lifting, lowering, or pulling.

Tagline (1) A rope (usually fiber) attached to a lifted load for controlling load spin and pendulation. (2) A wire rope used to stabilize a bucket or magnet during material handling operations.

Tailing crane In a multicrane operation in which a long object is erected from a horizontal starting position to a vertical final position, the crane controlling the base end of the object.

Tailswing The horizontal distance from the *center of rotation* of a crane to the extreme rear swing arc of its *counterweight*.

Tail tower In a cableway, the tower that does not adjoin the winches of the hoist and inhaul/outhaul.

Taking up The process of removing slack from a line or drawing (spooling) in on a line; loading a line by drawing in on it.

Tapered tip A boom tip configuration in which the sheaves are mounted on the longitudinal center axis and the boom chords taper toward a point on the center axis.

Telescopic boom crane A crane with a variable length boom consisting of a base section and one or more nested sections that are extended from the base for attainment of additional length. The telescoping of the boom and the various other motions of the crane are typically powered by hydrostatic (hydraulic) drives. Also called Hydraulic crane.

Telescoping The process of axially increasing or decreasing the length or height of a crane boom or *mast*; it is most commonly applied to the operation of hydraulically extending or retracting a cantilevered-type crane boom.

Telescoping gantry A lifting device that utilizes pairs of vertical hydraulic cylinders joined by header beams to raise and lower loads. These devices usually roll on tracks.

Third drum A third hoist drum, in addition to main and auxiliary hoist drums, often used in pile driving.

Tie assembly Collar and struts used to anchor the mast of a tower crane to the supporting structure, usually a building.

Tip section The outer most segment of a boom that contains the *head machinery* and attaches to the other lower sections.

Tipping fulcrum The support point of line about which a crane will tip if it should overturn. At the moment of overturning, the entire weight of the crane and load will be exerted on this point or line.

Tipping load The load for a particular operating radius that brings the crane to the verge of overturning.

Top climbing A method of raising a tower crane by adding tower sections; with the crane balanced, a climbing frame lifts the crane *superstructure* to permit insertion of a new tower section beneath the turntable. The process is repeated as required.

Top slewing A tower-crane configuration that has the *slewing ring* above the *mast*.

Top tower See Tower top.

Topping See Derricking.

Topping lift See Boom hoist.

Tower Tower-crane *mast*; a vertical structural frame consisting of columns and bracing capable of supporting a crane *superstructure*.

Tower attachment A combination of vertical *mast* and *luffing* boom mounted to the front end of a mobile crane in place of a conventional boom.

Tower crane A lift crane with the working boom mounted on a vertical mast; the working boom slews about a rotation center that is usually fixed at the center of the *mast*.

Tower head See Tower top.

Tower section A modular segment of the *mast* on a tower crane.

Tower top A structural section of a luffing or saddle jib tower crane, projecting vertically above the *turntable*, from which the jib is suspended. The counterjib may also be suspended from the Tower Top. Also called a Top tower or Tower head.

Track Assembled crawler shoes and connecting pins forming a belt around idler rollers and drive sprockets; the part of the crawler that contacts the ground.

Track cable On a *cableway*, the static cable strung between the masts that support the trolley, load, and *slack carriers*.

Track roller On a crawler-crane track, the rollers that mediate between a fixed side frame and the moving articulating treads.

Track shoe An articulating crawler tread segment that pins to adjacent segments forming a continuous band. A track is composed of assembled shoes that ride on *track rollers* mounted to a *side frame*.

Transit Movement or transport of a crane from one jobsite to another.

Travel Movement of a mobile or wheel-mounted tower crane about a jobsite under its own power.

Travel Base The base mounting for a wheel-mounted (traveling) tower crane.

Traveling tower crane A freestanding tower crane mounted on a ballasted platform furnished with trucks that ride along rails.

Tread diameter The diameter of a *sheave* or grooved rope drum measured at the base of the groove; the diameter of a smooth barrel on a rope drum. Also called Root diameter.

Tripping A rigging operation undertaken to shift a load from a flat position to an upright stance; it may be done with the bottom of the piece on the ground or truck bed, or with both top and bottom ends supported on rigging.

Trolley (1) A carriage carrying the hook block for radial movement along the lower *chords* of a horizontally mounted tower-crane jib; in some tower cranes, a counterweight trolley allows the *counterweights* to be moved radially to modulate their backward moment in proportion to load hook *radius*. (2) The unit that travels on the bridge girder of an overhead crane and that contains the load hoisting machinery.

Trolleying Movement of the load horizontally toward or away from the crane tower by moving the *trolley* along the jib or bridge.

Truck crane A crane superstructure and boom mounted on a rubber-tired carrier that can be driven over the road from a driver's station remote from the crane cab.

Trunnion Cylindrical projections on a vessel from which it can be suspended for lifting and forming an axis on which it can be pivoted in a *tripping* operation.

Tumbler A sprocket mounted at the ends of crawler *side frames* that guides the crawler treads as they pass around the end of the frame.

Turntable The *slewing* component of a crane, a term most commonly applied to tower cranes.

Two blocking The condition in which the *load block* comes in contact with the upper load block or *head machinery*.

Undercarriage (1) The self-powered bottom assembly of a crane by which it moves on a jobsite. A crawler *carbody* with side *frame track* assembly is referred to as an undercarriage as is the wheeled support frame for a traveling tower crane. (2) A tower-crane base that resists overturning by carrying *ballast*; it may rest on *bogies*, wheels, and rails to become a *travel base*.

Underhung crane A crane that is suspended from the bottom flange of a runway track or single-track monorail system. Also called Underhung hoist.

Under wound An arrangement of rope with respect to the drum whereby the rope spools out from the bottom.

Upper/upperstructure See Superstructure.

Upper boom point An auxiliary *sheave* assembly mounted above the main boom point sheave assembly in a boom, usually with fewer or smaller sheaves, for running an *auxiliary fall* on a crane.

Upper spreader See Floating harness.

Vangs Side lines reeved to a derrick boom and used to swing the boom.

Walking beam A *bogie* member whose long axis is nominally horizontal and parallel to the direction of travel; it is pivoted at its center and mounts a wheel, wheel pair, axle, or the center pivot of another walking beam at each end. Its purpose is to permit wheel oscillation, thus equalizing wheel loading during passage over travel-path irregularities.

Weathervaning Allowing an *out-of-service* crane to rotate freely in response to wind forces so as to expose a minimal area to the wind.

Wedge frame A steel frame that transmits lateral loads from the mast of an internal tower crane to the host structure. Wedge frames are required at two levels to resist overturning moment; one or both can double as a *climbing frame* or climbing base.

Wedge socket A fitting for securing the free end of a wire rope by looping it around a wedge-shaped insert that fits inside a mating hollow body. The fitting incorporates a pin connection by which it can be fastened to an anchor point. Also called a Becket.

Whip line A secondary or *auxiliary fall* usually with lesser load capacity than the main hoist rope; also called Whip, Runner, or Auxiliary line.

Winch A power-driven drum capable of winding rope for pulling or for lifting and lowering loads.

Wire rope A flexible rope composed of multiple-wire steel strands helically wound around a core.

Wire-rope dead end The end of a running rope attached to a fixed anchorage on the opposite end from the winding drum.

Wrap One circumferential turn of wire rope around a rope-drum barrel.

Zone protection system A device wired into the control system of a tower crane that prevents it from lifting loads over a prohibited area.

APPENDIX C

Exact Analysis of a Guyed Tower Crane

An exact analysis of a guyed mast takes into account the catenary shape and the elastic stretch of the rope. The method presented here is mathematically exact, but inaccuracies are introduced by the variations in the properties of ropes and the approximations made to adapt the real crane to a practical mathematical model—including the mathematical modeling of wind.

The shape of the rope changes with loads. A catenary rope loaded by a horizontal end force F_h and by its own weight (Figure C.1) will transmit that horizontal force to the anchorage at its other end undiminished. A perfectly flexible rope cannot have bending moment. The only applied load within the rope length, the rope weight, is vertical; the applied end load F_h constitutes the only horizontal force and therefore must be present throughout the rope. Taking moments about b (Figure C.1), we get

$$\sum M_b = F_h h - R_{ay} r - \frac{w' r^2}{2} = 0$$

$$R_{ay} = F_h \tan\theta - \frac{w' r}{2} \qquad (\boldsymbol{a})$$

where $w' = w/\cos\theta$ is the unit rope weight projected onto a horizontal plane. Taking moments about c of the loads to the left of c gives

$$F_h(x\tan\theta - y_m) - R_{ay}x - \frac{w' x^2}{2} = 0$$

where y_m is the difference in height between the chord and the rope. Substituting R_{ay} from Eq. ($\boldsymbol{a}$), we get

$$F_h y_m = \frac{w' r x}{2} - \frac{w' x^2}{2} \qquad (\boldsymbol{b})$$

The right side of Eq. ($\boldsymbol{b}$) is equivalent to the bending moment in a simple beam with uniform load w' about a point a distance x from one end. If the same procedure were to be used with a series of

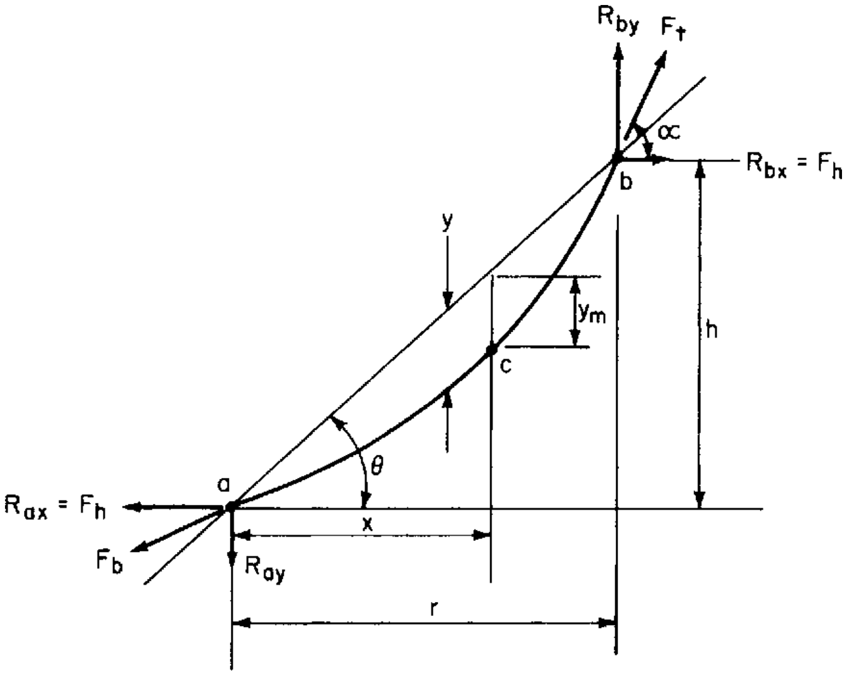


FIGURE C.1 Annotated guy-rope free-body diagram (elevation).

concentrated vertical loads along the rope, the result would give rise to a general theorem: at any point on a catenary rope, the horizontal force multiplied by the vertical distance from the rope to its chord is equal to the moment that would be produced by those same vertical loads about the same point on a simple beam of the same horizontal span.

Going to the center of a span carrying only rope dead load and calling the difference in height between chord and rope at this point y_0, we get

$$F_h y_0 = \frac{w' r^2}{8} \qquad F_h = \frac{w' r^2}{8 y_0} \tag{C.1}$$

Proceeding, we can derive expressions for tension at each end of the rope.[1] For an inclined rope, tension will be greatest at the top, and

$$\begin{aligned} F_t &= F_h(1 + 16S^2 + \tan^2\theta + 8S\tan\theta)^{1/2} \\ F_b &= F_h(1 + 16S^2 + \tan^2\theta - 8S\tan\theta)^{1/2} \end{aligned} \tag{c}$$

[1]See J. B. Wilbur and C. H. Norris, "Elementary Structural Analysis," New York, McGraw-Hill, 1948. p. 250.

where $S = y_0/r$ = sag ratio
F_t = tension in rope at upper end
F_b = tension at lower end

The maximum rope force can be restated as

$$F_t = \frac{F_h}{\cos\alpha} \tag{C.2}$$

where $\alpha = \cos^{-1}[(1+16S^2+\tan^2\theta+8S\tan\theta)^{-1/2}]$.

The true length of the rope can be represented, for practical purposes, by the approximate equation

$$L_0 = r\left(\frac{1}{\cos\theta}+\frac{8}{3}S^2\cos^3\theta\right) \tag{d}$$

which takes only the curvature of the rope into account.

In addition, the rope will stretch under load, but as shown by Eq. (*c*), the force in the rope varies through its length. The average force can be taken as

$$F_{\text{av}} = \frac{F_h r}{L_0}\left(1+\frac{16}{3}S^2+\tan^2\theta\right) \tag{e}$$

From Hooke's law, the elastic increase in rope length can be expressed in terms of the modulus of elasticity E_r as

$$\Delta L_0 = \frac{FL_0}{A_r E_r} \tag{f}$$

where A_r is the sum of the cross-sectional areas of the wires in the rope. Combining (*e*) and (*f*), we have

$$\Delta L_0 = \frac{F_h r}{A_r E_r}\left(1+\frac{16}{3}S^2+\tan^2\theta\right)$$

which when added to the change in length indicated by (*d*) gives the true length change of the rope, including curvature and elastic effects.

$$\Delta L = r\left\{8/3\left(S_1^2-S_2^2\right)\cos^3\theta+\frac{1}{A_r E_r}\left[(F_{h2}-F_{h1})(1+\tan^2\theta)+16/3\left(F_{h2}S_2^2-F_{h1}S_1^2\right)\right]\right\} \tag{C.3}$$

where S_1 is the sag ratio at the initial horizontal load F_{h1} and S_2 is the sag ratio at the final load F_{h2}. It has been assumed that the rope had been preloaded to F_{h1} to reduce initial sag and subsequently loaded by wind, for example, to F_{h2}. The rope-length change given by Eq. (C.3) takes place between those two loading levels. Of course, the equation provides change in length between any two loading levels, a negative change indicating a shortening of the rope.

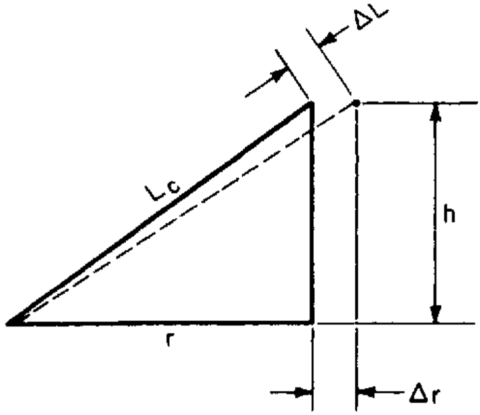


FIGURE C.2 Horizontal displacement of a guy rope due to the deflection of the crane mast.

Change in rope length, as loading increases, must be reflected in elastic lean of the crane mast. In turn, the elastic lean is resisted by the springlike action of the guy. The horizontal displacement, in terms of rope parameters, is expressed by Figure C.2.

$$\Delta_r = [(L_c + \Delta L)^2 - h^2]^{1/2} - r \qquad \text{(C.4)}$$

where L_c, the initial chord length, is $(r^2 + h^2)^{1/2}$.

Using the method of elastic stability (see Sec. 4.3) and the model in Figure C.3, we can develop expressions relating mast elastic deflection and guying restraint.

In the upper section of the mast, $0 \le x \le h_1$, the moment in the mast is given by

$$M_x = W_w x + \frac{wx^2}{2} - M_c + Qz$$

where w is now used to denote wind force on the mast per unit of mast length. The differential equation for this section then becomes

$$z'' + k^2 z = \frac{M_c - W_w x - wx^2/2}{EI}$$

where $k^2 = \dfrac{Q}{EI}$

$$Q = Q_0 + \frac{uH}{3}$$

$u =$ dead load per unit of mast length

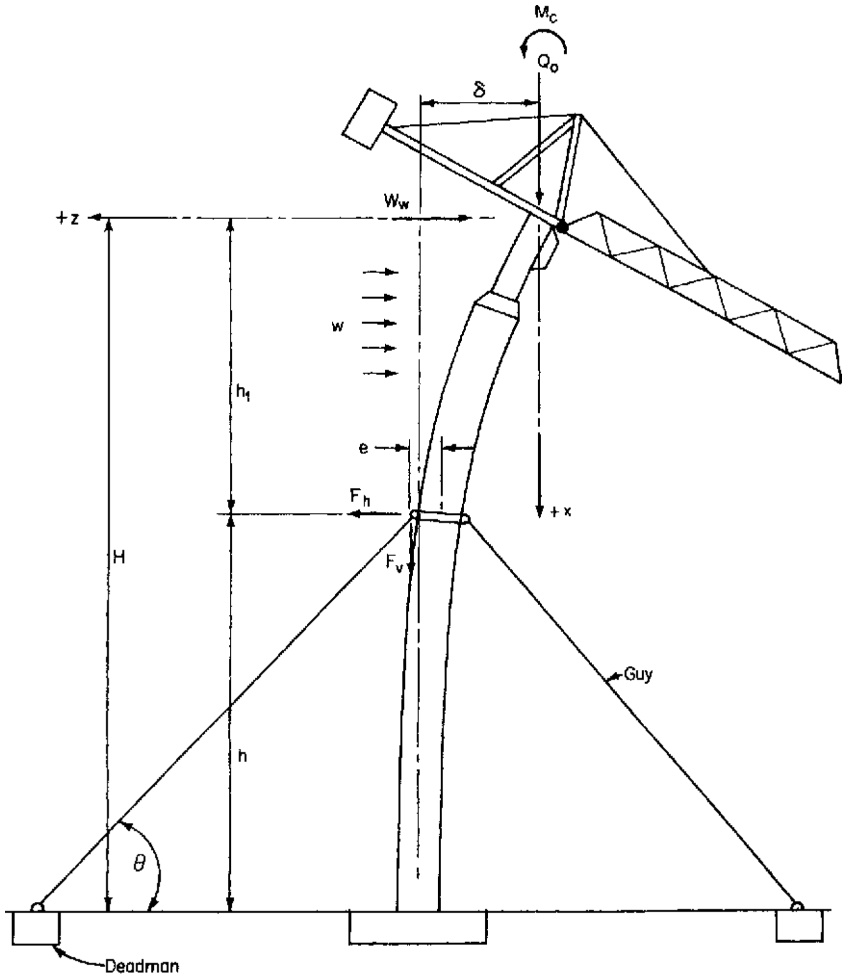


FIGURE C.3 Analytical model of a guyed crane exposed to out-of-service wind loading (bending deflection exaggerated).

The solution to this equation is

$$z = A\cos kx + B\sin kx + \frac{M_c - W_w x - wx^2/2}{Q} + \frac{w}{Qk^2}$$

where A and B are unknown.

From the boundary condition z(0) = 0,

$$0 = A + \frac{M_c}{Q} + \frac{w}{Qk^2} \qquad A = -\frac{M_c}{Q} - \frac{w}{Qk^2}$$

and at the lower end of the section $z(h_1) = d$, which is an arbitrary name for displacement at this level.

$$d = -\left(M_c + \frac{w}{k^2}\right)\frac{\cos kh_1}{Q} + B \sin kh_1$$

$$+ \frac{M_c - W_w h_1 - wh_1^2/2}{Q} + \frac{w}{Qk^2}$$

$$B \sin kh_1 - d = \frac{1}{Q}\left(M_c + \frac{w}{k^2}\right)(\cos kh_1 - 1) + \frac{W_w h_1 + wh_1^2/2}{Q} \qquad (g)$$

The slope at the base of the section will be given an arbitrary value s; $z'(h_1) = s$.

$$z' = \frac{k}{Q}\left(M_c + \frac{w}{k^2}\right)\sin kx + Bk \cos kx - \frac{w_w + wx}{Q}$$

$$s = \frac{k}{Q}\left(M_c + \frac{w}{k^2}\right)\sin kh_1 + Bk \cos kh_1 - \frac{W_w + wh_1}{Q} \qquad (h)$$

For the second section of the mast, $h_1 \leq x \leq H$, the moment is given by

$$Mx = W_w x + \frac{wx^2}{2} - M_c - F_h(x - h_1) - F_v e + Qz$$

motivating the differential equation

$$z'' + k^2 z = \frac{M_c + F_h(x - h_1) + F_v e - W_w x - wx^2/2}{EI}$$

for which the solution is

$$z = C \cos kx + D \sin kx + \frac{M_c + F_h(x - h_1) + F_v e - W_w x - wx^2/2}{Q} + \frac{w}{Qk^2}$$

For continuity with the previous section, $z'(h_1) = s$ must be satisfied.

$$z' = -Ck \sin kx + Dk \cos kx + \frac{Fh}{Q} - \frac{W_w + wx}{Q}$$

$$s = -Ck \sin kh_1 + Dk \cos kh_1 + \frac{F_h}{Q} - \frac{W_w + wh_1}{Q}$$

The arbitrary s can now be eliminated by equating this expression with (h), giving, after simplification,

$$B + C\tan kh_1 - D = \frac{F_h}{Qk\cos kh_1} - \left(M + \frac{w}{k^2}\right)\frac{\tan kh_1}{Q} \qquad (\boldsymbol{i})$$

As a further condition of continuity, $z(h_1) = d$

$$d = C\cos kh_1 + D\sin kh_1 + \frac{M_c + F_v e - W_w h_1 - wh_1^2/2}{Q} + \frac{w}{Qk^2}$$

$$C\cos kh_1 + D\sin kh_1 - d = -\frac{F_v e}{Q} - \frac{M_c + w/k^2}{Q} + \frac{W_w h_1 + wh_1^2/2}{Q} \qquad (\boldsymbol{j})$$

The boundary condition at the base of the crane, $x(H) = \delta$, gives

$$\delta = C\cos kH + D\sin kH + \frac{M_c + F_h h + F_v e - W_w H - wH^2/2}{Q} + \frac{w}{Qk^2}$$

$$C\cos kH + D\sin kh - \delta = -\frac{F_h h + F_v e}{Q} - \frac{M_c + w/k^2}{Q} + \frac{W_w H + wH^2/2}{Q} \qquad (\boldsymbol{k})$$

and finally, the slope at the mast base must be zero, $z'(H) = 0$

$$0 = -Ck\sin kH + Dk\cos kH + \frac{F_h}{Q} - \frac{W_w + wH}{Q}$$

$$-C\sin kH + D\cos kH = \frac{W_w + wH}{Qk} - \frac{F_h}{Qk} \qquad (\boldsymbol{l})$$

Since we now have five equations g, i, j, k, and l, for the five unknowns B, C, D, d, and δ, the system of equations can be uniquely solved. In matrix form,

$$\begin{bmatrix} \sin kh_1 & 0 & 0 & -1 & 0 \\ 1 & \tan kh_1 & -1 & 0 & 0 \\ 0 & \cos kh_1 & \sin kh_1 & -1 & 0 \\ 0 & \cos kH & \sin kH & 0 & -1 \\ 0 & -\sin kH & \cos kH & 0 & 0 \end{bmatrix} \begin{bmatrix} B \\ C \\ D \\ d \\ \delta \end{bmatrix} = \begin{bmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \\ C_4 \end{bmatrix}$$

$$\text{where } C_1 = \frac{1}{Q}\left(M_c + \frac{w}{k^2}\right)(\cos kh_1 - 1) + \frac{W_w h_1 + wh_1^2/2}{Q}$$

$$C_2 = -\left(M_c + \frac{w}{k^2}\right)\frac{\tan kh_1}{Q} + \frac{F_h}{Qk\cos kh_1}$$

$$C_3 = \frac{1}{Q}\left(W_w h_1 + \frac{wh_1^2}{2}\right) - \frac{M_c + w/k^2}{Q} - \frac{F_h}{Q} e \tan\alpha$$

$$C_4 = \frac{1}{Q}\left(W_w H + \frac{wH^2}{2}\right) - \frac{M_c + w/k^2}{Q} - \frac{F_h}{Q}(h + e\tan\alpha)$$

$$C_5 = \frac{W_w + wH}{Qk} - \frac{F_h}{Qk}$$

where $F_h \tan\alpha$ is substituted for F_v.

The matrix expression is $U = A^{-1}C$, where U is the column vector of unknowns, C is the column vector of constants, and A^{-1} is the inverse of the matrix of coefficients. The solution yields values for each of the unknowns and hence the expressions for deflections and moment at any point in the mast for any value of the rope reaction F_h; but a unique value for F_h has not yet been found.

Using Kramer's Rule, the values for displacement d at guy height and δ at mast top are found for any F_h; the difference $\delta - d$ is then the mast lean at guy height. This must correspond to rope stretch under load F_h.

$$\delta - d = N_1 - \frac{F_h}{Q}(N_2 + N_3 \tan\alpha)$$

$$\text{where} \quad N_1 = \frac{1}{Q}\left[\frac{W_w + wH}{k}\left(\frac{\sin kH - \sin kh_1}{\cos kH}\right) + \frac{wh_1^2}{2} - \frac{wH^2}{2} - W_w h - \frac{(M_c + w/k^2)(1 - \cos kh)}{\cos kH}\right]$$

$$N_2 = \frac{\tan kh - 2\tan kh_1(1/\cos kh - 1)}{k(1 - \tan kh \tan kh_1)} - h$$

$$N_3 = e\left(\frac{1/\cos kh - 1}{1 - \tan kh \tan kh_1}\right)$$

By rearranging, an expression for final guy force is obtained as a function of guy level displacement.

$$F_h = \frac{Q[N_1 - (\delta - d)]}{N_2 + N_3 \tan\alpha} \tag{C.5}$$

with trigonometric arguments in radians. Final mast lean is dependent on the net horizontal load, F_h. As wind load is applied, the mast will lean, causing an increase in rope loading on the windward side and a decrease from the preload on the leeward side. Approximately, if preload $F_{h1} \le F_h/2$, the leeward guy will go to nearly slack when F_h is applied and $F_{h2} = F_h$ on the windward side. But if $F_{h1} > F_h/2$, the leeward guy will remain loaded and the final guy loads on both sides will be consistent with mast lean. In that case, the difference between final guy loads must be F_h.

Guy loading will contribute to mast stiffness and natural frequency of vibration; therefore the gust factor can be affected. The horizontal spring rate of the loaded guy will be

$$K_r = \frac{F_{h2} - F_{h1}}{\Delta r} \qquad \Delta r = \delta - d \tag{C.6}$$

Equation (3.10) will reflect the effect of the guys, for any K_r, on the effective mast spring rate used to calculate natural frequency. A reasonable initial approximation for K_r should yield suitable results.

Mast top displacement should also be determined, as it may be necessary to control this parameter in some instances. Its value is given by

$$\delta - N_4 - F_h N_5 \tag{C.7}$$

where

$$N_4 = \frac{1}{Q}\left[(W_w + wH)\frac{\tan kH}{k} - W_w H - \frac{wH^2}{2} - \left(M_c + \frac{w}{k^2}\right)\left(\frac{1}{\cos kH} - 1\right)\right]$$

$$N_5 = \frac{1}{Q}\left[\frac{\sin kH - \sin kh_1}{k \cos kH} + e \tan \alpha \left(\frac{\cos kh_1}{\cos kH} - 1\right) - h\right]$$

If this deflection is excessive, it may be reduced by using more parts of line, rope of larger diameter, or rope with higher modulus of elasticity—or by increasing preload. Each of these measures, or combinations of them, will reduce Δr and therefore $\delta - d$ with an attendant increase in F_h and decrease in δ.

This page intentionally left blank

APPENDIX D

Boom and Jib Clearances

Full derivations of boom and jib clearance equations are given below, along with clearance equations for some unusual cases.

D.1 Boom Clearance Derivation

Following is the full derivation of Eq. (5.5) for determining the clearance distance C between a boom and the leading edge of a building.

The line forming the shortest distance between the approaching boom and the leading wall edge is perpendicular to both, but the three lines are not in one plane. C is defined as the length of that clearance line. The direction cosines (Figure 5.8) for the clearance line are

$$\alpha_c = \frac{x_c}{C} \beta_c = \frac{0}{C} = 0 \qquad \gamma_c = \frac{z}{C}$$

referring to the respective X, Y, and Z Cartesian axes. For the boom (Figure 5.8), let l be the length of the line scribed on the surface running from a point at the height of the leading wall edge to the point of intersection with the clearance line. The direction cosines of this line are

$$\alpha_c = \frac{-x_b}{l} \beta_b = \frac{y_b}{l} \gamma_b = \frac{z}{l}$$

Expressing the projections of the line formed by the boom corner onto the Cartesian axes in spherical coordinate form,

$$x_b = -l \sin \varphi \, (90° - \theta) = -l \sin \varphi \cos \theta$$

$$y_b = l \cos \varphi \sin (90° - \theta) = l \cos \varphi \cos \theta$$

$$z = l \cos (90° - \theta) = l \sin \theta$$

permits the direction cosines to be restated as

$$\alpha_b = -\sin \varphi \cos \varphi \qquad \beta_b = \cos \varphi \cos \qquad \theta \, \gamma_b = \sin \theta$$

But since lines l and C are normal, they must conform to the requirement that

$$\alpha_b \alpha_c + \beta_b \beta_c + \gamma_b \gamma_c = 0$$

$$-\frac{x_c}{C} \sin\varphi \cos\theta + \frac{z}{C} \sin\theta = 0$$

$$\frac{z}{x_c} = \frac{\sin\varphi}{\tan\theta} = \frac{g+e}{R\tan\theta}$$

Let the angle the clearance line makes with the horizontal (Figure 5.8) be τ. Then

$$\tau = \tan^{-1}\frac{z}{x_c} = \tan^{-1}\frac{\sin\varphi}{\tan\theta} = \tan^{-1}\frac{g+e}{R\tan\theta} \tag{5.4}$$

The horizontal line between the top of the wall and the closest boom corner (lying in the same vertical plane as C) has length k (Figure D.1), comprising the segments (Figure 5.8).

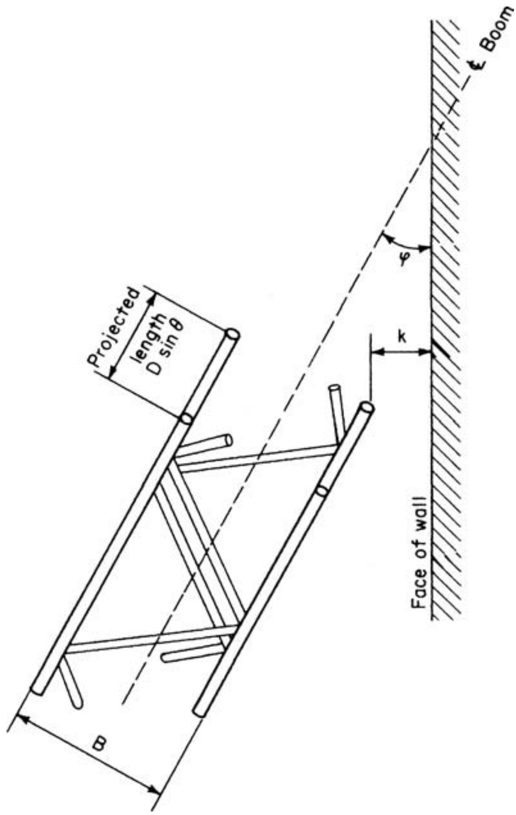


Figure D.1 Plan view of a portion of a boom in proximity to a wall.

$$k = -x_b + x_c = l \sin\varphi \cos\theta + C\cos\tau$$

$$= \frac{l^\circ \sin\varphi \sin\theta}{\tan\theta} + C\cos\tau$$

but $l \sin\theta = z = (x_c \sin\varphi)/(\tan\theta) = (C\cos\tau \sin\varphi)/(\tan\theta)$. After substitution and manipulation this becomes

$$k = \frac{C}{\cos\tau} \tag{a}$$

From Figures 5.7, 5.8, D.1, and D.2 it can be seen that the length k can also be described in terms of readily observable dimensions and angles as

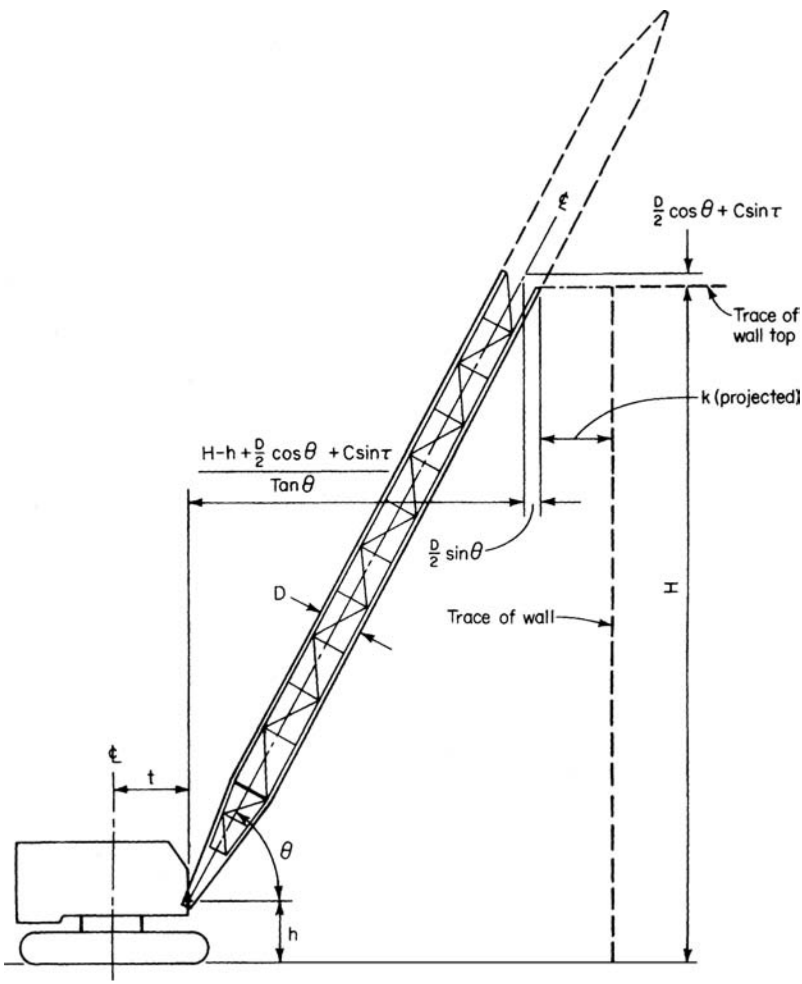


Figure D.2 Elevation of a mobile crane viewed normal to the latticed boom.

$$k = \left[R - t\frac{H - h + C\sin\tau + (D/2)\cos\theta}{\tan\theta} - \frac{D}{2}\sin\theta\right]\sin\varphi$$

$$-\frac{B}{2}\cos\varphi - e$$

$$= \left(R - t - \frac{H - h}{\tan\theta} - \frac{D}{2\sin\theta}\right)\sin\varphi - \frac{B}{2}\cos\varphi - e - \frac{C\sin\tau\sin\varphi}{\tan\theta} \qquad (\boldsymbol{b})$$

By taking $\boldsymbol{k}$ as given in Eq. (***b***), substituting into Eq. (***a***), and rearranging we find that the clearance C is given by

$$C = \frac{1}{A}\left[\left(R - t - \frac{H - t}{\tan\theta} - \frac{D}{2\sin\theta}\right)\sin\varphi - \frac{B}{2}\cos\varphi - e\right] \qquad (5.5)$$

$$\text{where } A = \frac{1 + \sin^2\tau}{\cos\tau} \qquad (5.6)$$

D.2 Jib Clearance Derivation

The full derivation of Eq. (5.12) follows. When the boom point is below the leading edge, this formula determines the clearance C between the edge and the nearest part of the jib. The horizontal distance between the leading edge of the building and the closest jib chord is

$$k_j = \frac{C_j}{\cos\tau_j} \qquad (\boldsymbol{c})$$

From Figure 5.10, the length k_j can also be given by

$$k_j = \left[R - t - L\cos\theta - \frac{H - L\sin\theta - h + (d/2)\cos(\theta - \mu) + C\sin\tau_j}{\tan(\theta - \mu)}\right.$$

$$\left.-\frac{d}{2}\sin(\theta - \mu)\right]\sin\varphi - \frac{b}{2}\cos\varphi - e$$

$$= \left[R - t - L\cos\theta - \frac{H - L\sin\theta - h}{\tan(\theta - \mu)} - \frac{d}{2\sin(\theta - \mu)}\right]\sin\varphi$$

$$-\frac{B}{2}\cos\varphi - e - \frac{C\sin\tau_j\sin\varphi}{\tan(\theta - \mu)} \qquad (\boldsymbol{d})$$

Using k_j as given by Eq. (*d*), substituting into Eq. (*c*), and rearranging, we find the jib clearance

$$C = \frac{1}{A_j}\left[\left(R - t - L\cos\theta - \frac{H - L\sin\theta - h}{\tan(\theta-\mu)} - \frac{d}{2\sin(\theta-\mu)}\right)\sin\varphi - \frac{b}{2}\cos\varphi - e\right] \tag{5.12}$$

where $$A_j = \frac{1+\sin^2\tau_j}{\cos\tau_j} \tag{5.13}$$

D.3 Boom Tip Clearance with Jib Mounted

The formulas that follow are useful for a long latticed boom crane with a jib lifting to a high roof and the boom tip below the roof line. In previous editions these formulas were in the main text, but the work assignments once done this way are now performed principally by tower cranes and large telescopic cranes.

Under some circumstances the boom head might be close enough to the building face that its clearance should be verified, as the head is usually wider than the jib. A boom operating at a high angle and a jib with a small offset angle is particularly prone. For these calculations, new values of some parameters are needed to correspond to the boom head dimensions. For the clearance line to the boom.

$$\tau = \tan^{-1}\frac{\sin\varphi}{\tan\theta}$$

where both φ and θ are the values calculated for the combined boom and jib. The radius of the boom tip will be

$$R_t = t + L\cos\theta$$

and the position parameter e_t of the boom tip will be

$$e_t = e - [J\cos(\theta-\mu)]\sin\varphi \tag{D.1}$$

The parameter e_t locates the centerline of the boom, at the tip, with respect to the face of the structure. Therefore, the actual tip clearance C_t can be derived from the horizontal dimensions. Working from this point, we get

$$C_t = \frac{-e_t - (d_t/2)\sin\theta\sin\varphi - (b_t/2)\cos\varphi}{A} \tag{D.2}$$

where d_t is the boom tip dimension analogous to D, b_t is similarly defined, and A is given by Eq. (5.6). If this clearance value is not acceptable, jib clearance does not control, the calculated maximum service height is not correct, and a new height must be calculated for clearance with respect to the boom [using Eq. (5.7)].

A second point on the boom must also be checked, the point at which the tapered tip section begins. The clearance at this point is similarly derived and is given by

$$C_{st} \frac{[L_t \cos\theta - (D/2)\sin\theta]\sin\varphi - e_t - (B/2)\cos\varphi}{A} \tag{D.3}$$

where L_t is the length of the tapered tip section. If both C_t and C_{st} have been found deficient or if C_{st} alone is too small, the service height with adequate clearance will be given by Eq. (5.7) using e_t for e and R_t for R; R_t is the radius of the boom tip.

With both C_t and C_{st} adequate, the jib controls and the previously calculated service height is correct.

When the tip clearance C_t is not acceptable but the start of taper clearance C_{st} is, the maximum service height is controlled by clearance measured at some point between these positions on the boom. The boom dimensions at the critical height will fall between D and d_t for boom section height and B and b_t for width, say d' and b'. These values will be a function of maximum service height, which in turn will be affected by the values. The parameter d' is given by

$$d' = d_t + \frac{(D - d_t)[L - (H - h + c\sin\tau)/(\sin\theta)]}{L_t} \tag{D.4}$$

b' is given by a similar euuation containing the unknown height H.

Using an estimated value for H, we first find approximate values for d' and b' and insert them into the service-height equation.

$$H = h - \frac{d'}{2\cos\theta} + \left(R_t - t - \frac{b'}{2\tan\varphi} - \frac{CA + e_t}{\sin\varphi}\right)\tan\theta \tag{D.5}$$

With this height put into Eq. (D.4) more refined values will result. Several trials may be necessary until H, d', and d' come into balance, as the convergence may be slow. The value for H thus determined, however, will be the maximum height of structure for which the specified clearance will be maintained.

APPENDIX E

Codes and Standards Applicable to Cranes and Derricks

Consensus standards are voluntary while mandatory standards are codified into law.

E.1 American Society of Mechanical Engineers (ASME)

These are the principle U.S. consensus standards.

B30.1	Jacks
B30.2	Overhead and Gantry Cranes
B30.3	Construction Tower Cranes
B30.4	Portal, Tower, and Pedestal Cranes
B30.5	Mobile and Locomotive Cranes
B30.6	Derricks
B30.7	Base-Mounted Drum Hoists
B30.8	Floating Cranes and Floating Derricks
B30.9	Slings
B30.10	Hooks
B30.11	Monorails and Underhung Cranes
B30.12	Handling Loads Suspended from Rotorcraft
B30.13	Storage/Retrieval (S/R) Machines and Associated Equipment
B30.14	Side Boom Tractors

B30.15	Mobile Hydraulic Cranes (withdrawn—now included in the latest edition of B30.5)
B30.16	Overhead Hoists (Underhung)
B30.17	Overhead and Gantry Cranes
B30.18	Stacker Cranes
B30.19	Cableways
B30.20	Below-the-Hook Lifting Devices
B30.21	Manually Lever-Operated Hoists
B30.22	Articulating Boom Cranes
B30.23	Personnel Lifting Systems
B30.24	Container Cranes
B30.25	Scrap and Material Handlers
B30.26	Rigging Hardware
B30.27	Material Placement Systems
B30.28	Self-Erect Tower Cranes

E.2 Society of Automotive Engineers (SAE)

These are industrial consensus standards.

J220	Crane Boom Stops
J376	Load-Indicating Devices in Lifting Crane Service
J765	Crane Load Stability Test Code
J881	Lifting Crane Drum and Sheave Sizes
J959	Lifting Crane, Wire-Rope Strength Factors
J983	Crane and Cable Excavator Basic Operating Control Arrangements
J987	Crane Structures Method of Test
J1028	Mobile-Crane Working Area Definitions
J1078	A Recommended Method of Analytically Determining the Competence of Hydraulic Telescopic Cantilevered Crane Booms
J1289	Mobile-Crane Stability Ratings
J1305	Two-Block Warning and Limit Systems in Lifting Crane Service
J1939	Latticed Crane Boom Systems—Analytical Procedure

E.3 Australian Standard

AS1418	Cranes, hoists and winches

E.4 Canadian Standards Association (CSA)

These are consensus standards.

Z150	Mobile Cranes
Z248	Tower Cranes

E.5 Fédération Européene de la Manutention (FEM)

These are European industrial consensus standards.

FEM 1.001	Rules for the Design of Hoisting Appliances
FEM 1.004	Recommendation for the Calculation of Wind Loads on Crane Structures
FEM 1.005	Recommendation for the Calculation of Tower Cranes Structures in Out-of-Service Conditions
FEM 1.007	Recommendation to maintain tower cranes in safe conditions

E.6 International Standards Organization (ISO)

ISO 2374	Lifting Appliances—Range of Maximum Capacities for Basic Models
ISO 4301	Cranes and Lifting Appliances—Classification
ISO 4302	Wind Load Assessment
ISO 4304	Cranes Other than Mobile and Floating Cranes—General Requirements for Stability
ISO 4305	Mobile Cranes—Determination of Stability
ISO 4306	Vocabulary
ISO 4308	Cranes and Lifting Appliances—Selection of Wire Ropes
ISO 4309	Wire Ropes—Care, Maintenance, Installation, Examination and Discard
ISO 4310	Test Code and Procedures
ISO 7296	Graphic Symbols
ISO 7363	Cranes and Lifting Appliances—Technical Characteristics and Acceptance Documents
ISO 7752	Lifting Appliances—Controls—Layout and Characteristics
ISO 8087	Mobile Cranes—Drum and Sheave Sizes
ISO 8566	Cabins and Control Stations
ISO 8686	Design Principles for Loads and Load Combinations

ISO 9373	Cranes and Related Equipment—Accuracy Requirements for Measuring Parameters During Testing
ISO 9374	Information to Be Provided for Enquiries, Orders, Offers and Supply
ISO 9926	Training of Drivers (Operators)
ISO 9927	Inspections
ISO 9928	Crane Driving Manual
ISO 9942	Information Labels
ISO 10245	Limiting and Indicating Devices
ISO 10972	Requirements for Mechanisms
ISO 10973	Spare Parts Manual
ISO 11629	Measurement of the Mass of a Crane and Its Components
ISO 11630	Measurement of Wheel Alignment
ISO 11660	Access, Guards and Restraints
ISO 11661	Mobile Cranes—Presentation of Rated Capacity Charts
ISO 11662	Mobile Cranes—Experimental Determination of Crane Performance
ISO 11994	Availability–Vocabulary
ISO 12210	Anchoring Devices for In-Service and Out-of-Service Conditions
ISO 12478	Maintenance Manual
ISO 12480	Safe Use
ISO 12482	Condition Monitoring
ISO 12485	Tower Cranes—Stability Requirements
ISO 12488	Tolerances for Wheels and Travel and Traversing Tracks
ISO 13200	Safety Signs and Hazard Pictorials—General Principles
ISO 13202	Measurement of Velocity and Time Parameters
ISO 14518	Requirements for Test Loads
ISO 15442	Safety Requirements for Loader Cranes
ISO 15513	Competency Requirements for Crane Drivers (Operators), Slingers, Signalers and Assessors
ISO 15696	List of Equivalent Terms
ISO 16880	Bridge and Gantry Cranes—International Standards for Design and Manufacturing Requirements and Recommendations
ISO 16881	Design Calculation for Rail Wheels and Associated Trolley Track Supporting Structure

ISO 19961	Safety Code on Mobile Cranes
ISO 20332	Proof of Competence of Steel Structures
ISO 22986	Stiffness—Bridge and Gantry Cranes
ISO 23813	Training of Appointed Persons
ISO 23814	Competency Requirements for Crane Inspectors
ISO 23815	Maintenance
ISO 23853	Training of Slingers and Signalers
ISO 25599	Jib Cranes—International Standards for Design, Manufacturing, Use and Maintenance Requirements and Recommendations
ISO 27245	Tower Cranes—International Standards for Design, Manufacture, Use and Maintenance Requirements and Recommendations

E.7 European Committee for Standardization (CEN)

These are mandatory standards that must be adopted by all countries in the EU.

EN 13000	Mobile Cranes
EN13001	General Design
EN13852	Offshore Cranes
EN 14439	Tower Cranes
EN14492	Power-Driven Hoists and Winches
EN14985	Slewing Jib Cranes

E.8 Japanese Industrial Standards (JIS)

JIS D6301	Safety Regulations for Cranes and Related Machines

E.9 Chinese Standards

The "GB" series are mandatory national standards. The "JB" series are industrial consensus standards. The list here is partial.

GB/T 3811	Design Rules for Cranes
GB/T 5905	Test Codes and Procedures
GB/T 6067	Safety Rules for Lifting Appliances
GB/T 6974	Lifting Appliances—Vocabulary
GB/T 14405	General Purpose Overhead Cranes
GB 9462	Specification for Tower Cranes
GB 10051	Lifting hooks—Inspection of shank hook in service

GB 12602	Lifting Appliances—Safety Devices to Prevent Overloading
GB 14734	Safety Rules for Port-Floating Cranes
GB 15052	Lifting appliances—Warning Labels and Signs
GB 5082	Signals for Lifting and Moving
GB 5144	Safety Code for Tower Cranes
GB 5226.2	Safety of Machinery—Electrical Equipment
GB 7950	Jib cranes—Load moment limiters
GB/T 10051	Lifting hooks
GB/T 10183	Overhead-Traveling Cranes and Portal-Bridge Cranes—Tolerances
GB/T 12932	Shipbuilding-Jib Cranes
GB/T 13330	Test Methods for Crawler Crane with Lifting Capacities up to 150 Tons
GB/T 13752	Design Rules for Tower cranes
GB/T 14406	General-Purpose Gantry Cranes
GB/T 14560	Technical Requirement for Cranes with Lifting Capacities up to 150 Tons
GB/T 14743	Port Wheeled Cranes Specifications
GB/T 14744	Test Methods for Port Wheeled Crane
GB/T 14783	Rubber-Tired Gantry Cranes
GB/T 15360	Test Methods for Dockside Container Cranes
GB/T 15361	Specifications for Dockside Container Cranes
GB/T 15362	Test Methods for Rubber-Tired Gantry Cranes
GB/T 16562	Specifications for Port Rail-Mounted High Mast Type Cranes
GB/T 17495	Specifications for Harbor Portal Cranes
GB/T 17496	Repair Requirements for Harbor-Portal Cranes
GB/T 17806	Reliability Testing for Tower Cranes
GB/T 17807	Test Methods for Tower-Crane Structures
GB/T 17908	Cranes and Lifting Appliances—Technical Characteristics and Acceptance Documents
GB/T 17909	Crane Operating Manuals
GB/T 18438	Harbor cranes—Acceptance Testing
GB/T 18439	Harbor cranes—Stability Requirements
GB/T 18440	Harbor cranes—Technical Characteristics and Acceptance Documents
GB/T 18441	Harbor Cranes—Information to Be Provided
GB/T 18453	Maintenance Manuals
GB/T 18874	Information to Be Provided
GB/T 18875	Spare Parts Manuals

GB/T 19683	Rail-Mounted Gantry Cranes
GB/T 19912	Safety Code for Rubber-Tired Gantry Cranes
GB/T 19924	Mobile Cranes—Determination of Stability
GB/T 20303	Cabins
GB/T 20304	Tower Cranes—Stability Requirements
GB/T 20776	Classification for Lifting Appliances
GB/T 20863	Lifting Machinery—Classification
GB/T 4307	Lifting Hooks—Nomenclature
GB/T 5031	Performance Testing for Tower Cranes
GB/T 5972	Wire Ropes for Cranes—Inspection and Discard
GB/T 6068	Test Code for Truck Cranes and Mobile Cranes
JB/T 6748	Consoles
JB 8716	Mobile- and Truck-Crane Safety Requirements
JB/T 10559	Ultrasonic Testing of Welds on Lifting Machinery
JB/T 1375	Classification of Truck Cranes and Mobile Cranes
JB/T 4030	Truck Cranes and Mobile Cranes—Operating Reliability Testing
JB/T 6042	Truck-Crane Chassis
JB/T 8906	Jib Cranes
JB/T 9005	Cast Sheaves
JB/T 9738	Truck Crane and Mobile Crane—Technical requirements

Index

This page intentionally left blank

C

D

E