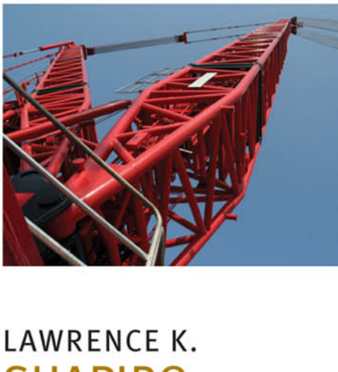
CRANES and DERRICKS

FOURTH EDITION



LAWRENCE K.
SHAPIRO

JAY P.
SHAPIRO

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.

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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

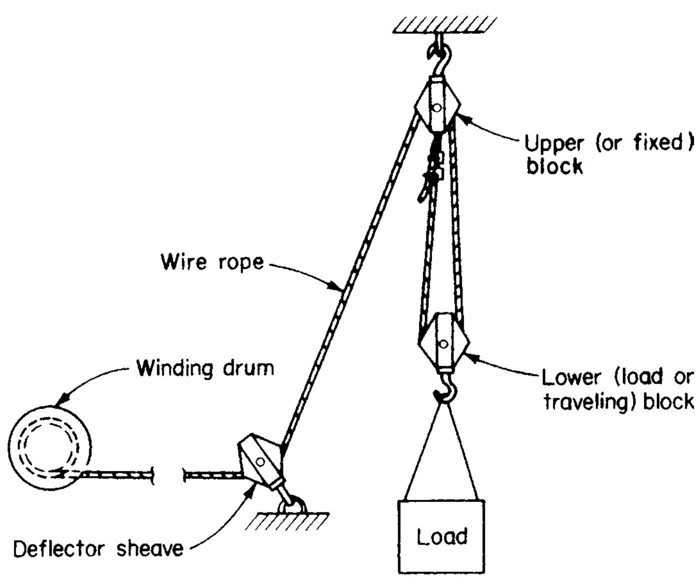


Figure 1.1 Basic hoisting mechanism.

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.

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

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.

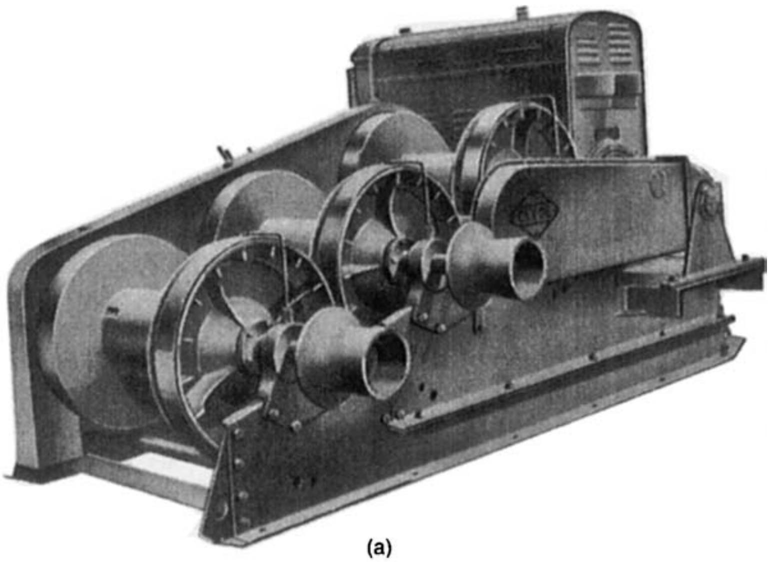
In practice, an overhaul weight somewhat in excess of the calculated value is used. This allows for possible inaccuracy in the loss coefficient and provides the mass needed to overcome inertia and induce acceleration. When friction losses are neglected in calculations, it would be wise to increase the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

1.3 Drums, Hoists, and Sheaves

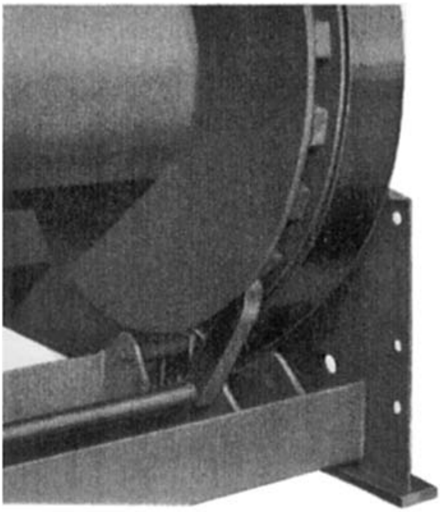
An assemblage of one or more winding drums mounted on a frame is called a *hoist* or *winch* (Figure 1.3). A free-standing winch unit with integral controls and power plant is called a *base-mounted drum hoist*. Power sources, drives, and control systems vary widely.

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

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[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.)

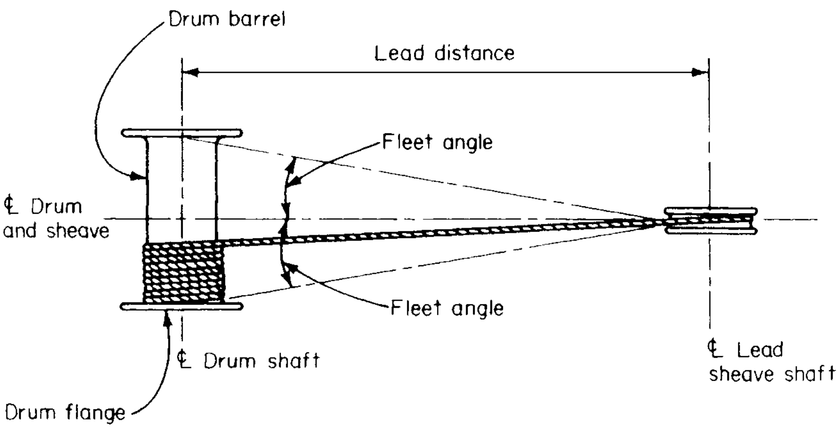

FIGURE 1.4 Fleet angle.

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

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

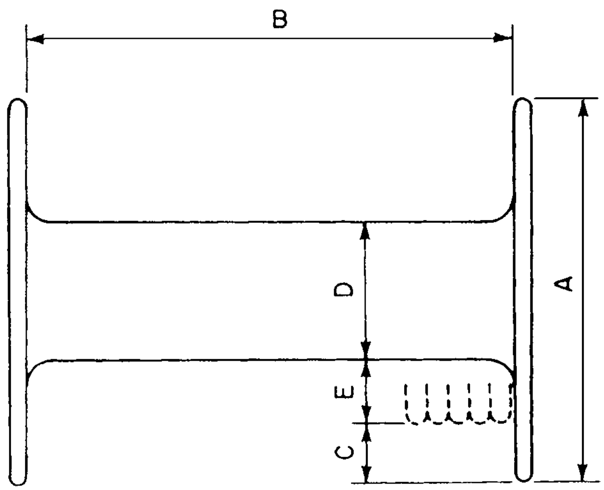

FIGURE 1.5 Hoist-drum dimensions.

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:

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

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

$$E = \frac{A - D}{2} \tag{1.3}$$

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

$$E = \frac{A - D - 2C}{2} \tag{1.4}$$

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*
1/2	0.925	13/16	0.354	1 3/8	0.127	2	0.0597
9/16	0.741	7/8	0.308	1 1/2	0.107	2 1/8	0.0532
5/8	0.607	1	0.239	1 5/8	0.0886	2 1/4	0.0476
11/16	0.506	1 1/8	0.191	1 3/4	0.0770	2 3/8	0.0419
3/4	0.428	1 1/4	0.152	1 7/8	0.0675	2 1/2	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.

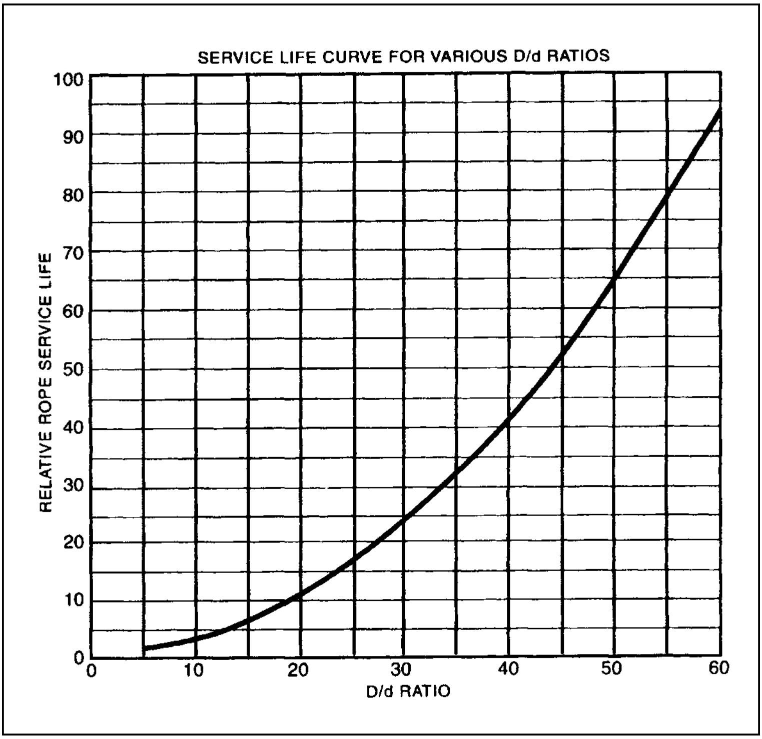

FIGURE 1.6 This generic curve relates relative rope service life to the ratio of sheave to rope diameter (*D/d*) considering only bending and tensile stresses during use. The scale is nominally the percentage of the life of a straight rope undergoing the same loading without passing over sheaves. The 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	
19 × 7 or 18 × 7 R.R.	.70	6 × 29 FW	
6 × 19 S	.80	6 × 36 WS	1.15
6 × 19 W		6 × 36 SFW	
6 × 21 FW		6 × 43 FWS	
6 × 26 WS		7 × 31 WS	
6 × 25B FS	.90	8 × 25 FW	
6 × 27H FS		6 × 41 WS	
6 × 30G FS		6 × 41 SFW	1.25
6 × 31V FS		6 × 49 SWS	
7 × 21F W		7 × 36 FW	
6 × 25 FW	1.00	6 × 46 SFW	
6 × 31 WS		6 × 46 WS	
8 × 19S		8 × 36 WS	1.35
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.

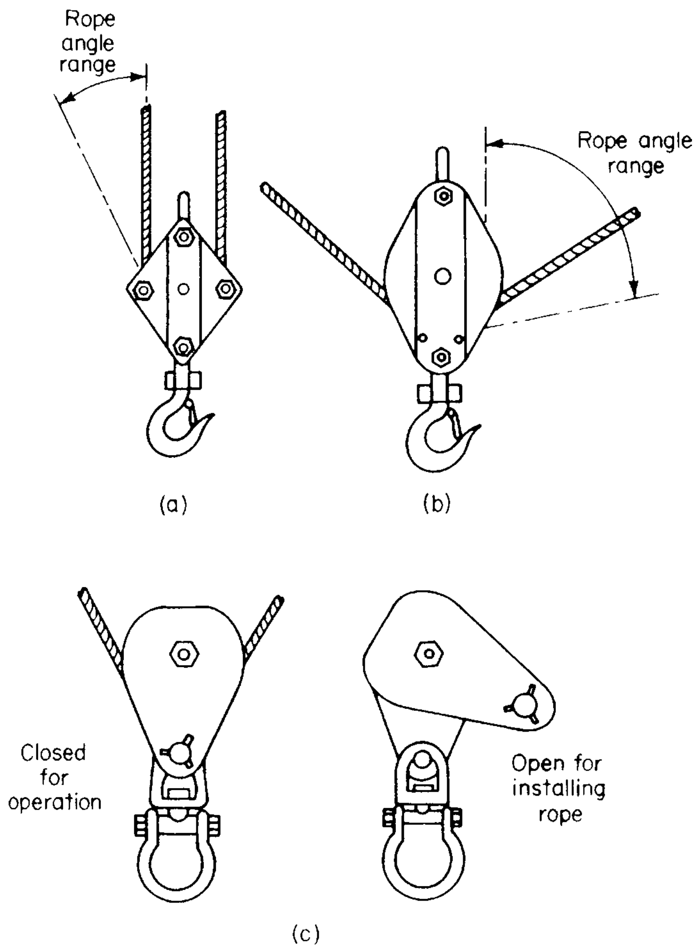

FIGURE 1.7 Principal block styles. (a) Diamond pattern. (b) Oval pattern. (c) Snatch blocks. The rope angle ranges shown reflect the degree of confinement of the rope in the block and do not mean to imply that hoisting should be done under those conditions.

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.

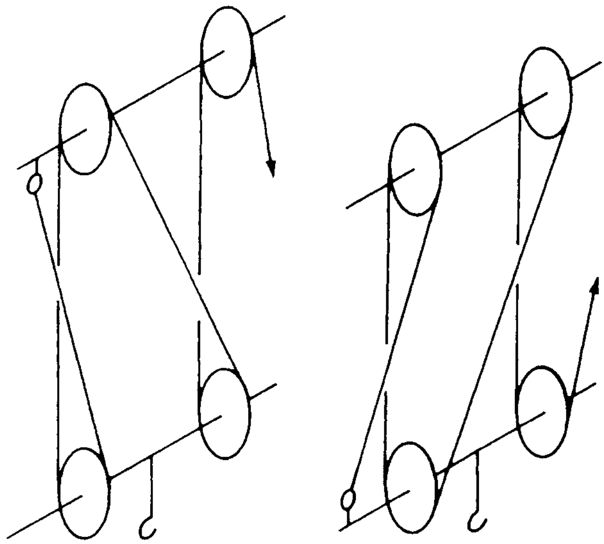

FIGURE 1.8 Reeving diagrams. (a) Four parts of line. (b) Five parts of line.

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

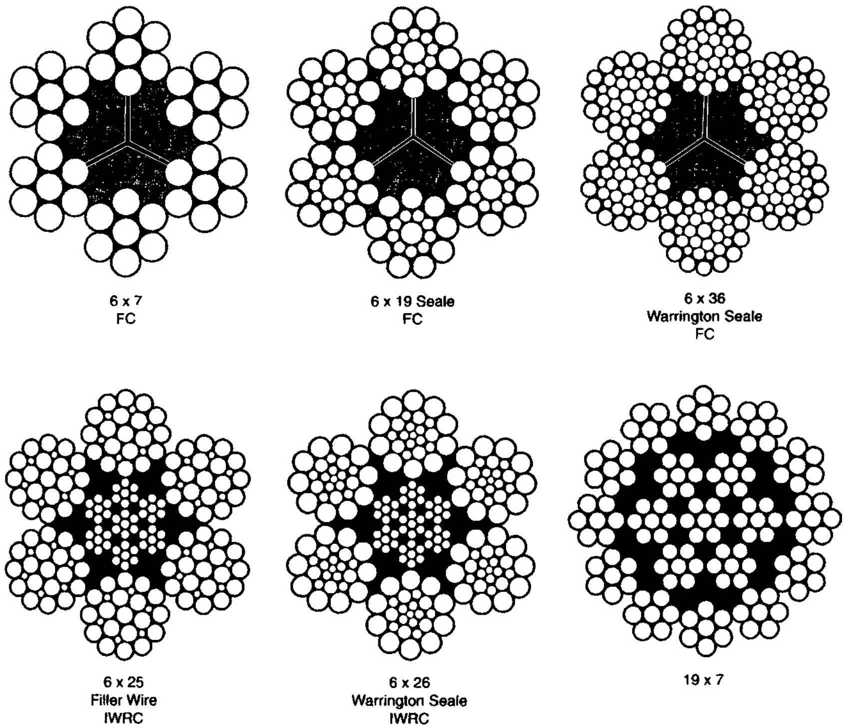

Figure 1.9 A few of the many wire rope constructions. (*Wire Rope Technical Board Users Manual*, 3d ed., used with permission.)

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

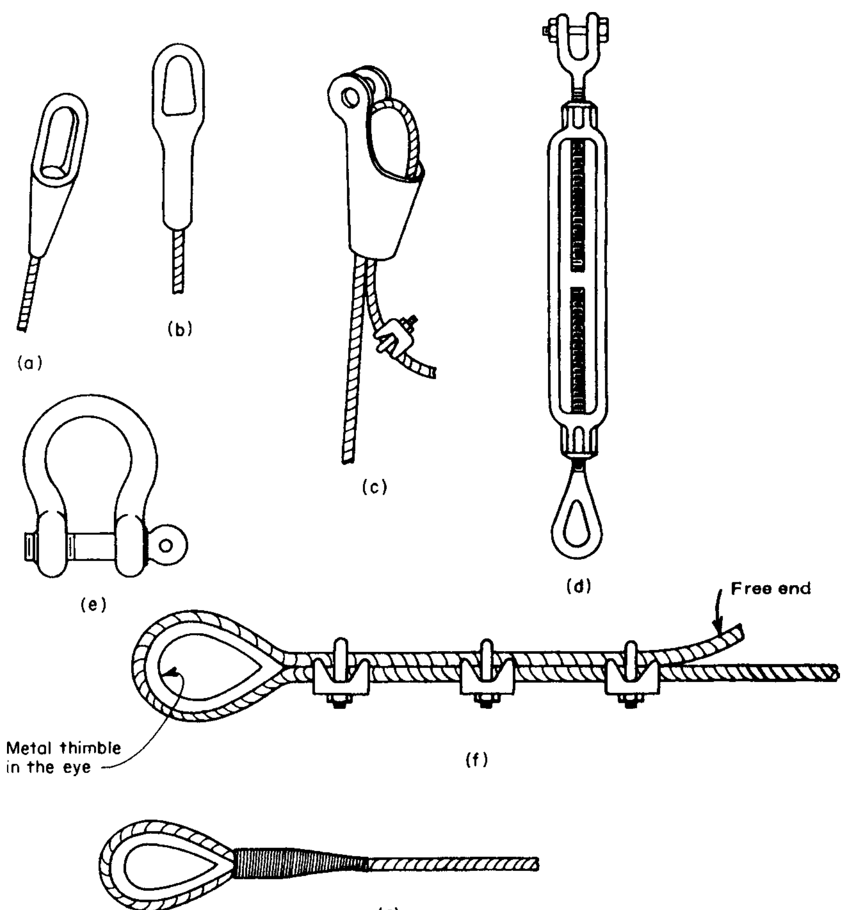

Figure 1.10 Some common wire-rope fittings. (a) Spelter (zinced) socket (efficiency 100%). (b) Swedged socket (efficiency 100%). (c) Wedge socket (efficiency about 80%). (d) Turnbuckle. (e) Shackle. (f) Crosby-type clips and thimble (efficiency about 80%). (g) Hand-spliced eye and thimble (efficiency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

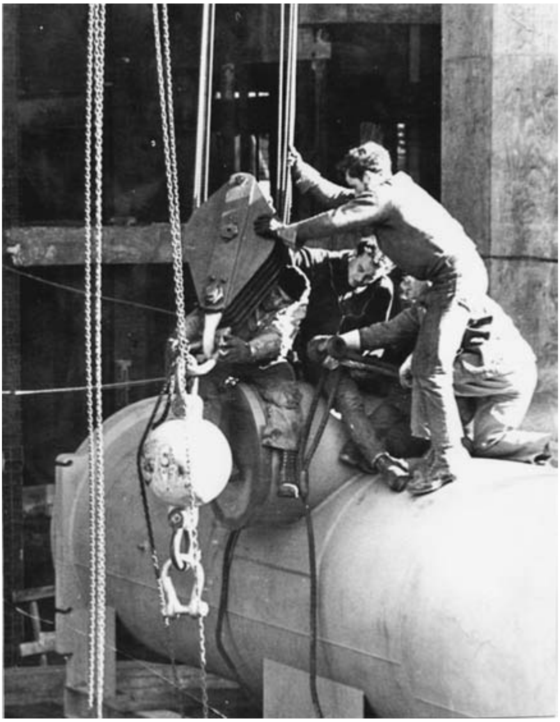

Figure 1.11 Riggers slinging a load in preparation for lifting. Note the overhaul ball and fittings. This photograph was taken before OSHA required the use of hard hats and fall-protection measures. (*Lawrence K. Shapiro.*)

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

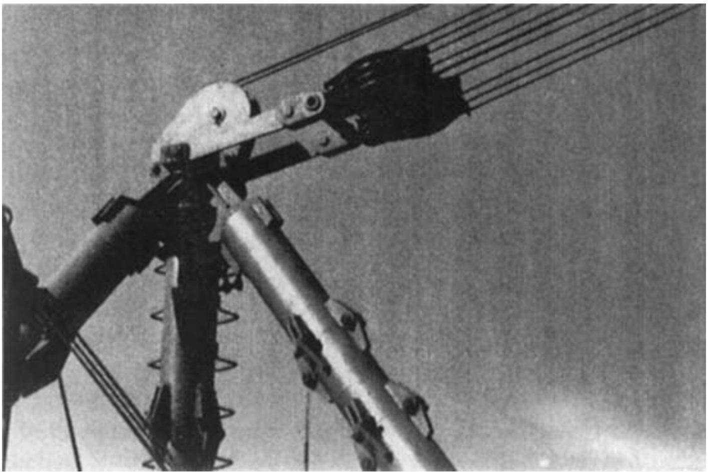

Figure 1.12 The mast top end of a stiffleg derrick luffing system. The two top lines are the load hoist and derricking lead lines; they run down the center of the mast over deflector sheaves to the winch. (*Lawrence K. Shapiro*.)

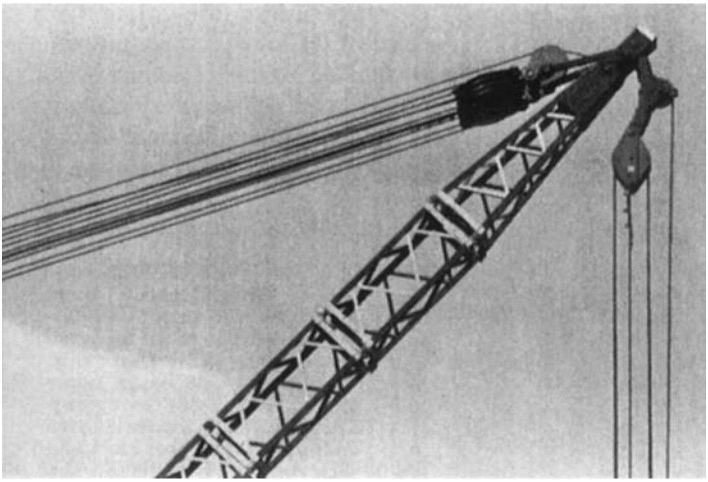

Figure 1.13 The *boom* tip end of the luffing system of Figure 1.14. Note the load lead line coming up at the right. It passes over a deflector sheave and through the *boom* tip. It then passes over the vertical sheave mounted on the derricking block and goes back to the mast. (*Lawrence K. Shapiro.*)

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.

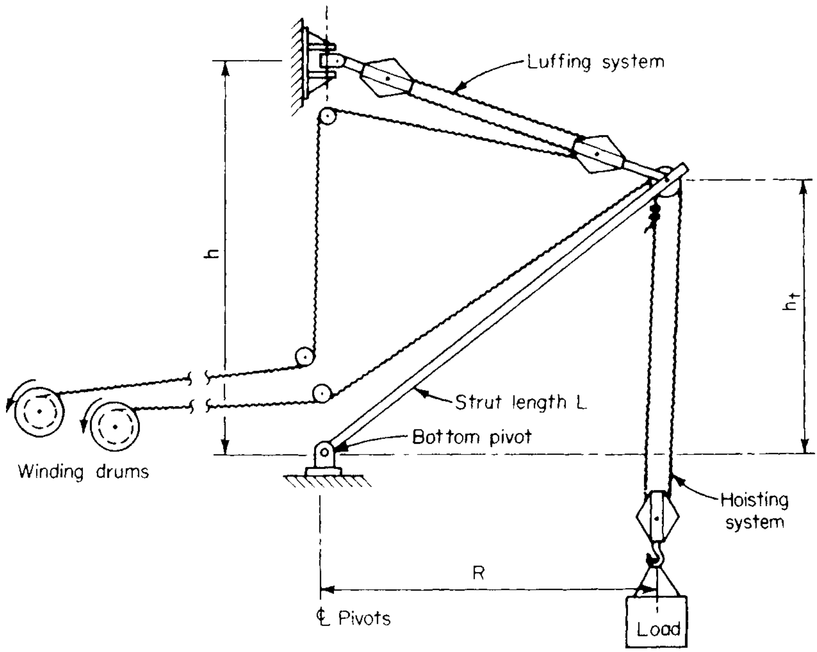

FIGURE 1.14 Basic derrick arrangement.

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

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

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

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

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

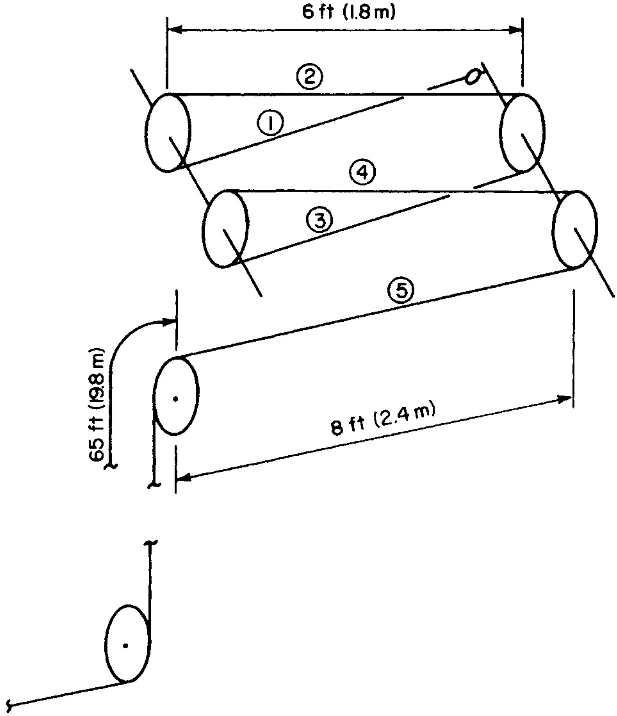

Figure 1.15 Topping-lift reeving and dimensions for Example 1.3.

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

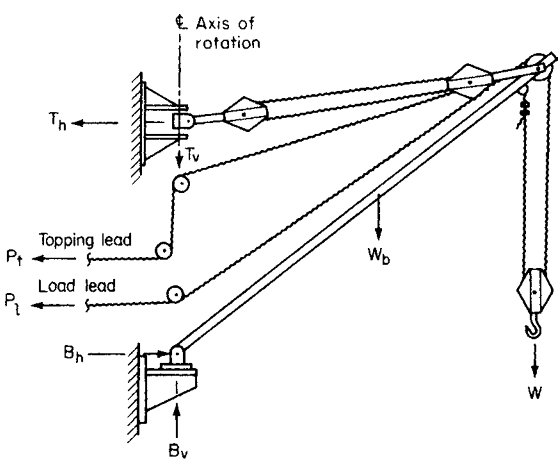

FIGURE 1.16 A Chicago *boom* derrick.

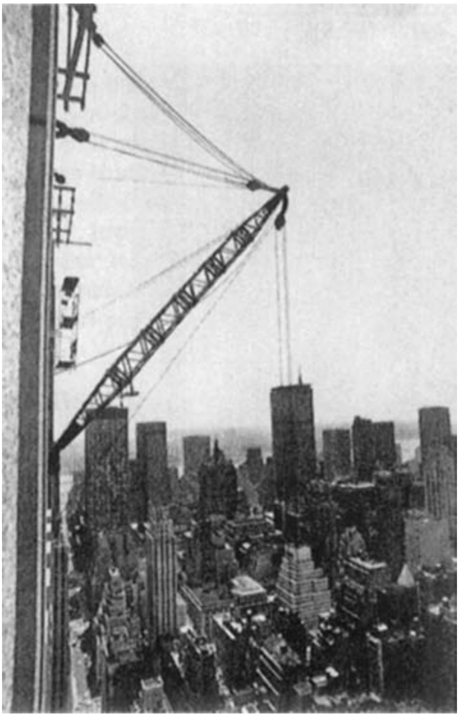

Figure 1.17 Chicago *boom* mounted at the 50th floor during construction of the McGraw-Hill Building in New York City. What appears to be an upper *topping lift* is a side (swing) guy. The load lead line is reeved over a sheave in the upper block and then it runs under the *boom*. The topping lead comes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

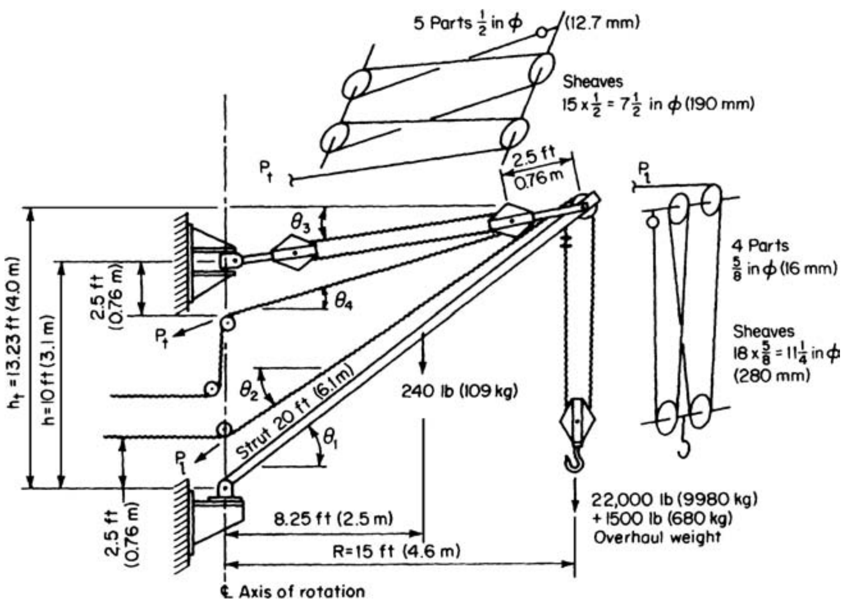

FIGURE 1.18 Derrick of Figure 1.16 dimensioned for sample calculations.

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

$$(22{,}000+1500)15+240(8.25)-(½)(22{,}000+1500)(2.5\cos 36.74)$$

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

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).

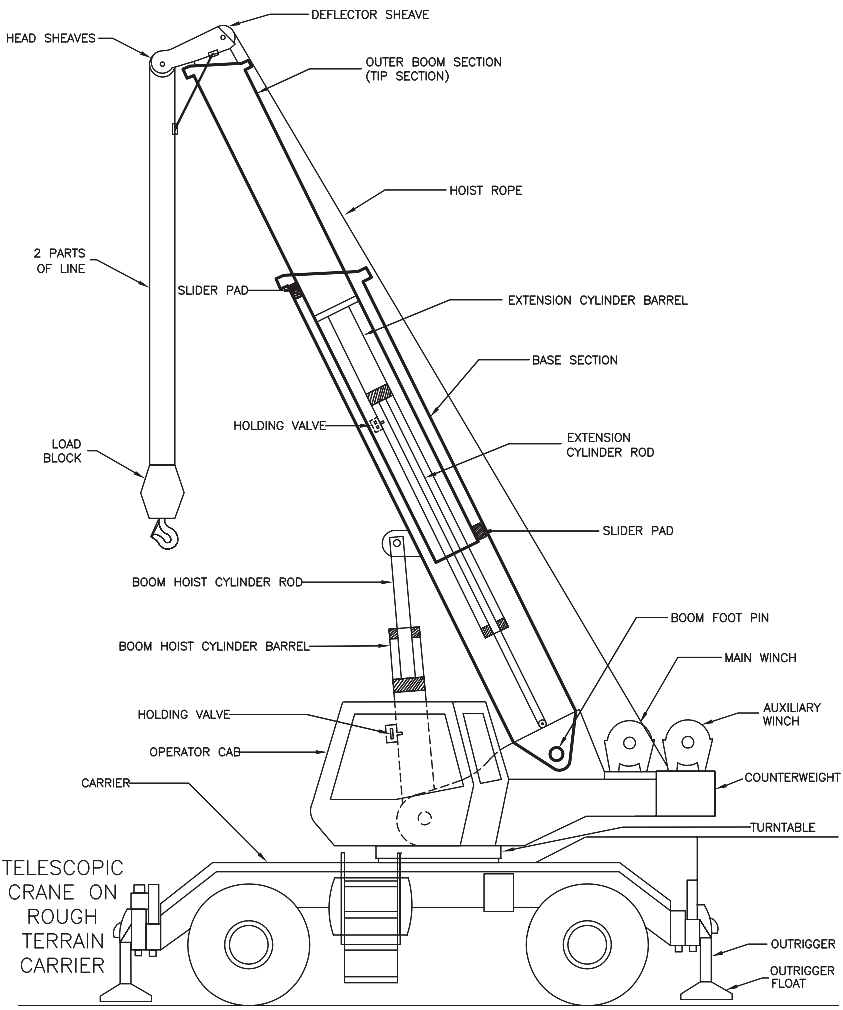

Figure 1.20 Crawler-mounted mobile crane annotated for consideration of stability.

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

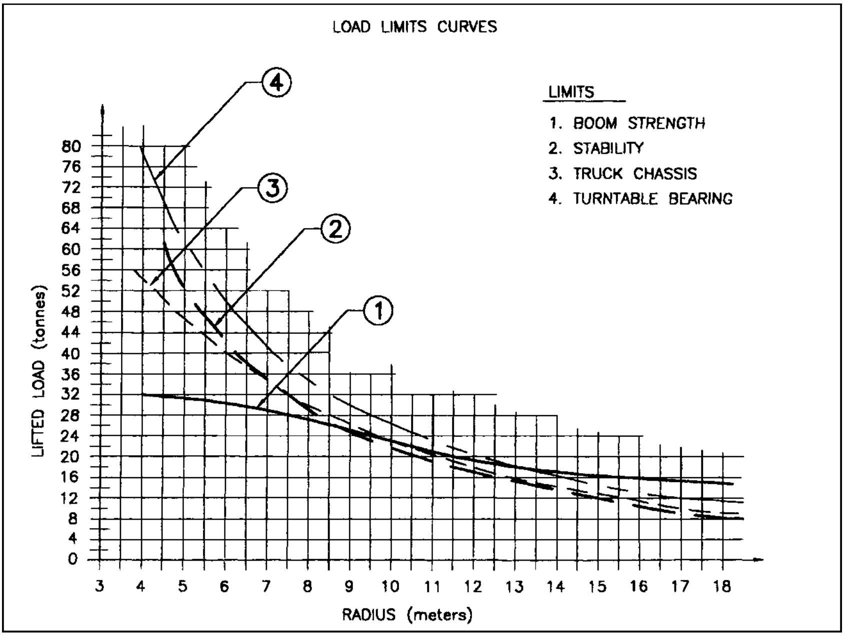

FIGURE 1.21 P&H model 6250-TC truck crane of 300-ton (272-t) capacity in a state of balance during stability experiments. (*Harnischfeger Corporation.*)

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

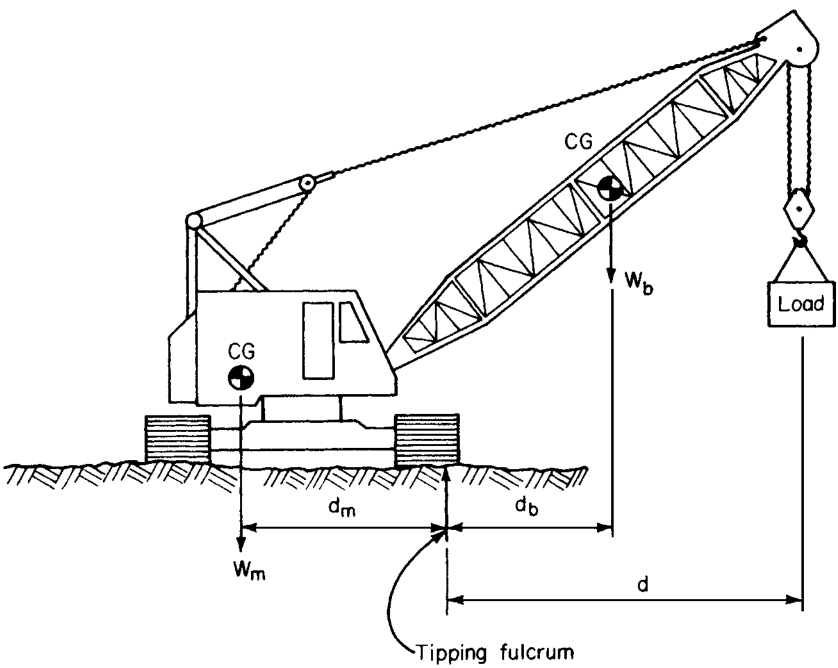

FIGURE 1.22 Barge-mounted crane listing under the weight of a 75-ton (68-t) bridge section during unloading operations. Lifting accessories were designed and operations planned by one of the authors. (*Lawrence K. Shapiro.*)

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.

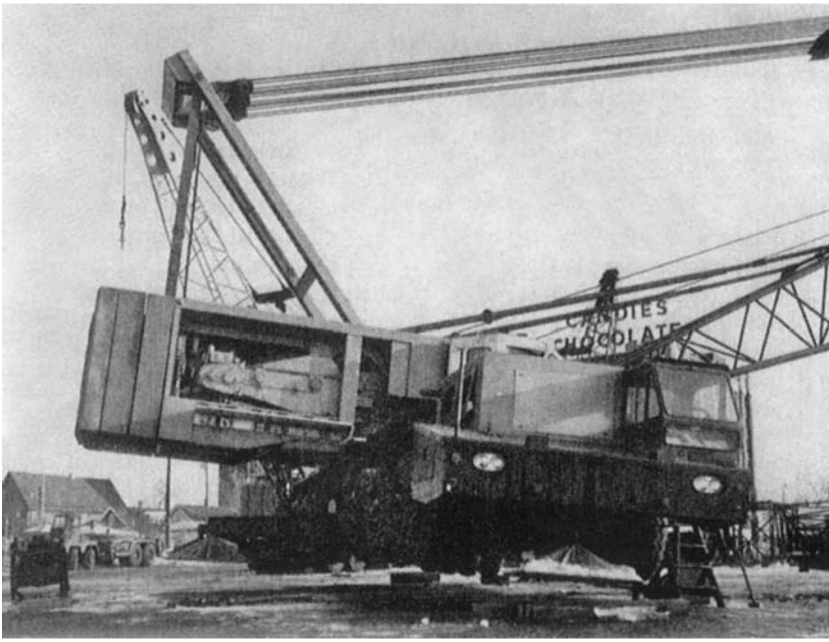

Figure 1.23 General arrangement of a basic telescopic cantilevered boom crane, more commonly known as a hydraulic crane.

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.