ELEVATOR HOISTWAY EQUIPMENT: Mechanical and Structural Design, Part II

by George W. Gibson

Learning Objectives

After reading this article, you should:

- Have developed a basic understanding of the construction, terminology and characteristics of steel wire ropes.
- Have learned that the two most important criteria affecting the life of suspension (hoist) ropes are the sheave diameters and rope lubrication.
- Understand that rope external and internal wear continues over the service life of the ropes, and that the residual strength of the ropes decreases accordingly.
- Have developed a basic understanding of the principles of rope traction; specifically, that the amount of available traction between the ropes and grooves must always exceed the required traction to move the hoistway masses in a controlled and safe manner.
- Have developed a basic understanding that the amount of available traction between the drive sheave grooves and suspension ropes is a function of the actual coefficient of friction between them, the groove shape, and the arc of contact that the ropes make as they wind over the drive sheave.
- Understand that the major inherent safety feature of a traction drive is its ability to lose traction if either the car or counterweight bottoms on its buffer.
- Understand that the number and size of hoist ropes on any elevator is a function of the strength of the ropes and the factor of safety, and the groove pressures developed between the ropes and groove surfaces.
- Have learned that elevator enclosures (cabs) must be strong enough to withstand the forces imposed during normal service, as well as, during safety application and buffer engagement, and that their materials must meet fire protection requirements.
- Have learned that there are several configurations of elevator doors.
- Have learned the functions of the car door operator, and that the resulting door closing speed caused by the operator must be limited, depending on the type of door reopening device used.
- Understand that the riding quality of an elevator is based on the travel path imposed by the guide rails, as well as the position of loads inside the car, rope vibrations and air pressures in the hoistway.
- Have developed a basic understanding about the effects of building sway on the elevator systems, principally the hoist and compensating ropes, and traveling cables.

Continued

Value: 2 contact hours (0.2 CEU)

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10. ROPES AND TRACTION:

10.1 Steel Wire Ropes and Manufacture:

Steel wire ropes used with traction elevators are primarily in two general classifications:

A. 8 X 19 Class, which contains eight metal strands wound around a fiber core

B. 6 X 19 Class, which contains six metal strands wound around a fiber core

Each strand can have 19, 21, 25 or 26 individual wires in its configuration. The fiber core is a tightly wound three-strand rope. It may be made of vegetable or synthetic fibers.

In an elevator wire rope, the arrangement of the wires in the strands which determine its specific label is of four types as shown in Figure 30.

Taken in the order of predominant usage, the most common is the Seale construction, where each strand consists of a comparatively heavy center wire around which is a layer of nine smaller wires. The outer layer is supported in the valleys formed by the first layer; hence, both layers have the same number of wires. By simple wire count, this strand is designated 1-9-9. The large wires in the outer layer provide abrasion and wear resistance to this type.

The filler wire constructions shown are designed to give more strand flexibility. The six small filler wires laid in between the main wires of the inner layer provide a series of valleys into which wires of the outer layer fit nicely. There are twice as many wires in the outside layer as in the inside layer. By wire count, this type is designated: 1-6-6-12.

The Warrington construction is less commonly used today. It is made up of two layers of wire about a center wire. There are six wires in the inner layer and 12 in the outer layer, the outer layer of wires being alternately large and small. The wire count designation here is 1-6-12.

From the foregoing, it will be seen that 8 X 19 Seale simply means a wire rope of eight strands of Seale construction having 19 wires. The number of wires in the rope is equal to eight times 19, which is equal to 152. Generally speaking, the more wires per rope, the more flexible the rope. The large outer wires of the Seale construction strand make it comparatively stiffer than the filler wire strand, but eight-strand rope is more flexible than six-strand. So a 6 X 25 filler wire has more flexible strand construction, but having six strands compromises overall rope flexibility. However, six times 25 equals 150 wires, which attests to substantial rope flexibility by the general rule of total number of wires. The 8 X 25 filler wire combines both flexibility factors and it has eight times 25, or 200 wires.

The elevator wire rope then is six or eight adjacent steel strands in continuous helical formation tightly formed around a fiber rope center which provides internal support. The term *rope lay* is used to designate the distance along the rope's length that one strand requires for one complete helical turn around the rope. It is approximately equal to 6.5 times the nominal rope diameter, as shown in Figure 31.

Several grades of steel are used in the manufacture of elevator wire ropes. The so-called "iron" rope is made from relatively soft, low-carbon steel. It is a low strength rope used mostly for compensation arranged with tie-down. Traction steel contains somewhat more carbon and has a tensile strength of 160,000 psi, minimum. This is the steel used for general elevator rope application. In six-strand traction steel rope, all of the wires are made from traction steel, but eight-strand traction steel rope is made from a combination of traction steel wires in the outer layer and special high strength steel wires for the center wire and the inner layer. The reason for this is that eight-strand rope has a smaller cross-sectional steel area than six-strand rope, but both six- and eight-strand traction steel ropes have the same breaking strength.

In line with the construction of high-rise office buildings in the late 1950s and 1960s, the need for higher strength rope was apparent and the wire rope industry responded with extra high-strength traction steel rope. The rope strength was increased 25% to 40% and wire tensile strength to 220,000 psi, minimum. Whereas 1/2- and 5/8-inch diameter ropes were formerly common sizes, now 11/16-, 3/4-, 13/16- and 7/8-inch diameter hoist ropes are not uncommon in today's mega-high-rise buildings. Former high speed hoist

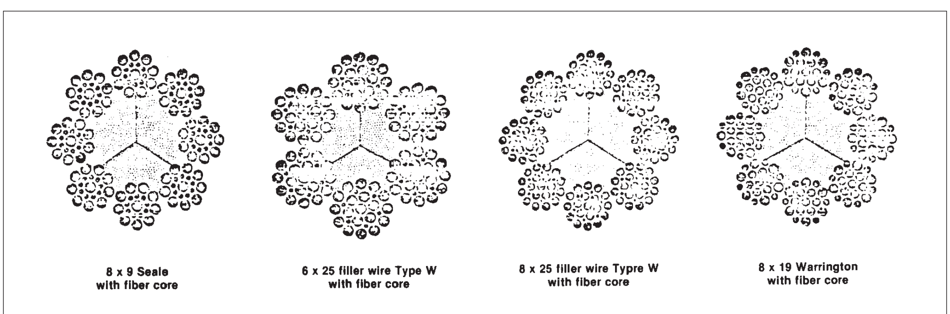

Figure 30

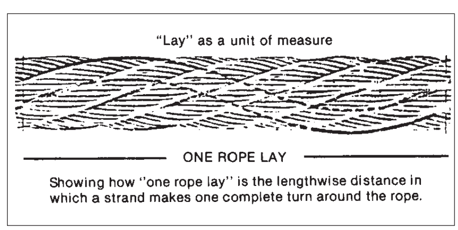

Figure 31

rope safety factors required by ASME A17.1 Code Rule 2.20.3 for passenger service of 11.00 for 700 fpm have receded into the background as current considerations are for 11.90 at 1,600 fpm and higher.

A rope is an element which we know in advance will have to be replaced after a few years of service. In view of this, the so-called factor of safety becomes a judgment factor set by the ASME A17.1 Code of the allowable maximum stress on a rope during its expendable life.

The strength of a rope is determined by taking a specimen of new rope and pulling it apart on a tensile testing machine and the factor of safety is defined as the ratio between the minimum load necessary to break the rope, and the load imposed on it in service. However, after a period of service, changes occur in the rope consisting principally of wear in the outer wires and the formation of microscopic fatigue cracks which develop into broken wires.

As this wear process continues, the residual breaking strength decreases, resulting in a corresponding decrease in the actual remaining factor of safety. When most steel hoist ropes have reached the point where replacement is necessary due to wear, it is not uncommon that the breaking strength is 60% of the original breaking strength of the ropes in their new condition.

The strength of a new rope then is not the permanent measure of safety. Safety in ropes is obtained only at the expense of continued inspection and their prompt removal at the appearance of signs of approaching failure. These signs, at present, are the number of breaks of outer wires on the worst lay length of the rope, as well as the distribution of these breaks over the various strands. Another is minimum rope diameter when it is reduced due to crown wire wear and core deterioration.

The status of a rope is usually determined by physical examination of its outside appearance. Modern-day advances in instrumentation have made possible the monitoring of ropes on a continuous basis by passing the ropes through a magnetic field from which the remaining steel area integrity can be checked. As smaller steel wire rope diameters are deployed, such monitoring devices will gain popularity because of their accuracy.

The *ASME A17.2 Guide for Inspection of Elevators, Escalators, and Moving Walks* gives rules for the inspection and condemnation of wire ropes. The service life of a rope is affected by many things, but there are two in particular that should be mentioned, i.e., rope lubrication and sheave diameter. When a rope runs over a sheave, sliding friction occurs between the different wires of the rope, between the strands and the hemp center, and, usually, between the outside wires and the surface of the sheave. This action causes abrasion of the wires and a breaking down of the hemp center. Considerable difficulty was experienced in this respect many years ago, but improvement in lubrication practice, both in rope manufacture and in elevator maintenance, has acted to reduce these troubles.

In lubricating traction drive elevator ropes, we must avoid a lubricant which will excessively reduce the coefficient of friction between the ropes and sheave. An extreme pressure lubricant or one containing graphite, for instance, would be very harmful in this respect. The ideal lubricant should penetrate the strands of the rope and thicken up enough to remain in place and protect them from corrosion, but should not produce a greasy or slippery surface. In general, lubrication of ropes is recommended when it becomes dry to the touch, but over-lubrication is to be avoided.

The ASME A17.1 Code Rule 2.24.2 specifies that for hoist ropes, no sheave diameter shall be less than 40 times the rope diameter except that a ratio of 30 times the rope diameter is permitted for private residence elevators [ASME A17.1 Rule 5.3.1.16.2(b)(1)(a)]. The most obvious advantage of a large sheave is that it reduces the bending stresses in the wires of the rope. Another advantage, not quite so obvious, is that the larger the sheave the less the radial pressure between rope and sheave.

While a large sheave is thus beneficial from the standpoint of sheave wear and long rope life, it naturally increases the cost of the machine. In a geared machine, it results in more load on the gear and, in a gearless machine, it requires more torque from the motor. Thus, we find that, as in so many phases of machine design, the size of the sheave must be a compromise between these conflicting influences.

Over the last few decades, elevator wire rope application considerations include preforming, synthetic fiber cores and pre-stretching.

Preforming is a wire rope fabrication technique whereby the helical shape of the strands in its position in the finished rope is permanently formed before the strands and fiber core are assembled into rope. The non-preformed wire rope has its wires and strands constrained in their final position in the rope; therefore, the ends must be held with wire bands called seizings. In non-preformed wire rope, severed wires or strands will tend to straighten out. In preformed rope, they do not. The advantages of preformed rope are increased flexibility and longer useful life. The disadvantages are increased rope stretch and greater difficulty in detecting broken wires during maintenance examinations and regulatory inspections.

Continued

Wire ropes with synthetic fiber cores have had successful application in oil fields due to their resistance to moisture and prevalent environmental conditions detrimental to natural fibers. They have been used for hoist and compensating rope applications. Steel wire ropes with synthetic cores stretch more than their fiber core counterparts.

Construction stretch is permanent stretch and primarily due to progressive embedding of the wires into the fiber core in the kneading action as new ropes run over the sheaves. This process is experienced as being fairly rapid in the early stages after installation of the ropes and diminishes with time over a period from a few months to over a year. The amount of this stretch can be fairly accurately estimated and the new rope cut short to allow for this anticipated stretch. The manufacturer's values for estimated stretch of wire rope with fiber core are as follows:

6 X 19: 1/2% to 3/4%
6-9 inches per 100 feet
8 X 19: 3/4% to 1%
9-12 inches per 100 feet

For preformed ropes, these figures may be increased by 50%.

Pre-stretching is an optional finish treatment of rope at manufacture which removes most of the initial stretch (construction stretch). Its additional cost must be weighed against the cost of "shortening ropes" when excessive construction stretch occurs, especially in high-rise installations.

10.2 Traction:

In most mechanical systems, a considerable emphasis is placed upon reducing the friction between parts. The opposite is true for elevator traction drive machines. In this latter case, we strive to utilize this friction that exists between the hoist ropes and the machine drive sheave.

The early roped machines utilized a large drum upon which the hoist ropes were wound. As buildings became taller, it was obvious that the winding drums must become larger and larger. This became the impetus behind the development of the traction drive. The major inherent safety feature of the traction drive machine is its ability to lose traction if either the descending car or counterweight hits its buffer, thereby preventing the machine from pulling the ascending mass into the building overhead structure.

The ASME A17.1 Code defines a *traction machine* as: "a direct drive machine in which the motion of the car is obtained through friction between the suspension ropes and a traction sheave."

ASME A17.1 Rule 2.16.8 stipulates that traction elevators be designed and installed to safely lower, stop and hold the car under rated load and under certain overloads.

ASME A17.1 Rule 2.24.2 further requires that drive sheaves shall be of metal and provided with finished grooves for ropes. The grooves of sheaves used to transmit power may be lined with nonmetallic material provided that, in the event the lining should fail, there will be sufficient traction still available in the groove to safely stop and hold the car with 125% of the rated load. Elastomer-lined grooves are permitted as long as these requirements are met. Otis Elevator Co. introduced its polyurethane groove, known as Cable-Save™, in the 1970s. There are many remaining in service. Future revisions to the ASME A17.1 Code are being considered which will allow for nonmetallic groove contact surfaces beyond elastomeric materials.

One of the best examples to show the advantage of harnessing friction is seen on the waterfront when a ship is docking. A hawser is thrown from the ship to the pier and a longshoreman wraps it a few times around a capstan. With as little as a 30-pound pull by the longshoreman, he can sustain a pull in the rope exerted by the moving ship equal to 10,000 pounds, as seen in Figure 32.

This action is made possible because of the coefficient of friction existing between the rope and capstan, and also to the number of turns that the rope makes as it is wound around the capstan. Therefore, it can be seen that traction depends on two criteria: friction and angle of contact.

Figure 33 shows a rope passing over a driving sheave. For the sake of the example, let us say that the portion of the rope leaving the sheave at point C leads to the car with a weight of S_1 = 1000 pounds. The portion of the rope leaving the sheave at point D leads to the counterweight with a weight of S_2 = 800 pounds. The tension in the ropes varies from its minimum value of S_2

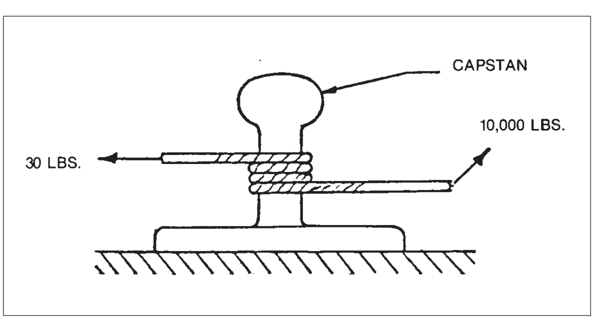

Figure 32

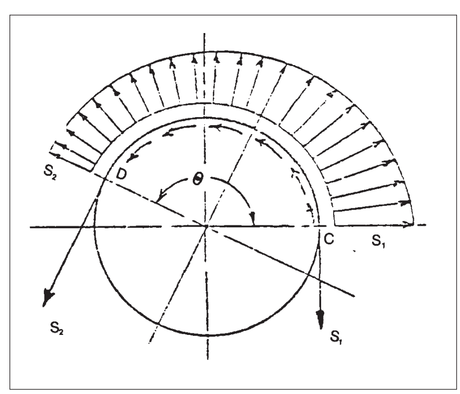

Figure 33

= 800 pounds at *D* to its maximum value of S_1 = 1000 pounds at point *C*. This is shown by the force vector distribution drawn around the sheave.

It is obvious then that something must exist that causes the rope tension to increase from the 800-pound value to 1000 pounds between points *D* and *C*. This something is called the tractive effort, where

Tractive effort = S_1-S_2

This tractive effort is based on the frictional resistance between the sheave and the rope. The amount of frictional resistance has certain limitations, and herein lies the basis for the theory of the traction drive.

Since the rope tensions S_1 and S_2 are unequal, this difference in tensions, referred to above as the tractive effort, produces a phenomenon called creep. Creep is the incremental movement of the hoist ropes over their arc of contact with the driving sheave due to the tractive force, the tensile elasticity of the ropes and the friction work, occurring in the direction of the greater tension, and is independent of the direction of rotation of the driving sheave. Creep exists in all traction systems and is not to be confused with the loss of traction.

The maximum available traction that can be developed in a groove is a function of the actual coefficient of friction between the rope and groove, the shape of the groove and the angle of contact that the rope makes with the circumference of the

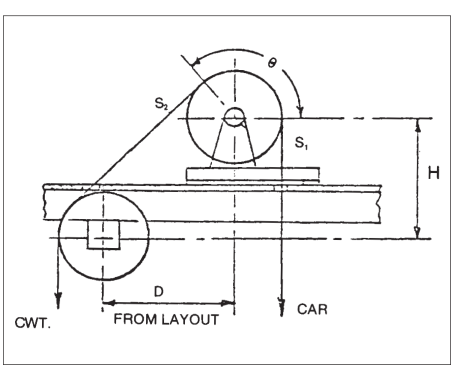

Figure 34

sheave. The required traction for any elevator is expressed as:

$$\text{Required traction} = \frac{S_1}{S_2}$$

where S_1 is greater than S_2. The maximum value of the ratio S_1/S_2 that can occur without the rope slipping over the sheave is called the available traction and is expressed as

$$\text{Available traction, } \alpha = e^{f_a\theta}$$

where e = the base of natural logarithms, f_a = the apparent coefficient of friction which embodies the actual coefficient of friction and the shape of the groove, α = available traction (dimensionless) and θ = the arc of contact that the rope makes with the sheave, expressed in radians.

Obviously then, the maximum traction is developed when the quantity, $f_a\theta$, is maximized as a result of increasing the apparent coefficient or by increasing the arc of contact or both. Elastomer-lined drive sheave grooves have higher coefficients of friction and afford a very efficient method of increasing the available traction.

Figure 34 shows a rope running over a driving sheave and a deflector as in the more or less typical overhead single wrap traction (SWT) installation. It is evident that, if hoistway layout conditions permitted a

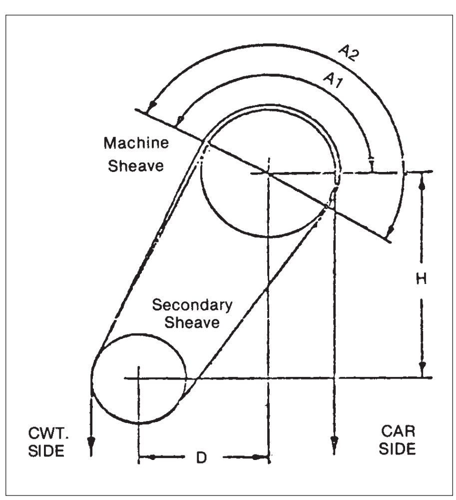

Figure 35

rope drop, i.e., the horizontal distance from the center of the car hitch to the center of the counterweight hitch without deflector, the maximum arc of contact between the rope and sheave would be θ = 180° = 3.14 radians.

With the deflector sheave as shown in Figure 34, it is seen that it is possible to obtain a slight increase in θ by lowering the deflector sheave as much as possible. However, when this method still produces insufficient traction, we must resort to the double wrap traction (DWT) roping arrangement as shown in Figure 35, or alter the groove shape to produce the required traction.

Whether single wrap or double wrap roping is used, it is obvious that there must be sufficient available traction to safely lower, stop and hold the car under all conditions of load without the ropes slipping. As noted above, the available traction is a function of two variables.

The hoist ropes, in passing over the traction machine drive sheave, exert a pressure between the contacting surface of the drive sheave grooves and the ropes. This pressure is directed radically in the plane of the sheave, and its magnitude is a function of the hoist rope tension and the diameter of the sheave. The larger the sheave diameter, the smaller the radial pressure per unit length along the arc of contact that the rope subtends on the sheave circumference.

This radial force causes a pressure distribution over the lower boundary of the rope profile thereby maintaining it in a state of vertical equilibrium. It is physically distributed over the regions of rope-to-groove contact. Such pressure distributions are shown graphically to a relative scale in Figure 36 for the various groove configurations of round-seat and undercut grooves.

Continued

Referring to Figure 36, it is seen that the maximum rope pressure occurs at the bottom of the groove. In the cases of the undercut grooves, the line of contact between the ropes and groove is interrupted where the pressure would ordinarily be the greatest. Therefore, since the vertical summation of pressure vectors must be in equilibrium with the radial force per unit length of rope, the pressure vectors must be exerted over a smaller region of groove contact, thereby inducing groove pressures higher than those of the round-seat groove. The maximum rope pressure will occur at the edge of the undercut.

The most significant increase in traction can be achieved by changing the groove shape, using the undercut groove, which increases the apparent coefficient of friction. The round seat groove is used on DWT installations, whereas the 90° and 105° grooves were used by Otis and other manufacturers on SWT machines. While the magnitude of the groove pressure distributions associated with each of the grooves as shown in Figure 36 varies, it is to be noted that in no case does the "actual" rope-to-groove coefficient of friction change. However, the "apparent" coefficient is a direct function of the groove pressures and the actual coefficient of friction, and it does change as the groove shape changes.

In conventional traction drives, drive sheaves are furnished with round-seat grooves or undercut grooves depending upon the required traction that must be developed to safely lower, stop and hold the elevator. The sheave material is usually a high grade of cast iron. From the standpoint of getting maximum rope life, it would be desirable to furnish round-seat grooves in all cases. However, in order that sufficient traction be developed between the steel rope and iron sheave, undercutting the bottom of the rope groove is required, thereby increasing the groove pressures upon which the higher available traction is based. While this conventional method provides the required traction, the attendant rope life and sheave life is greatly reduced.

The mathematical proof of the theory of traction is beyond the scope of this brief presentation; however, the proof concludes the following:

1. The available traction can be increased by increasing the arc of contact that the rope subtends with the sheave.
2. The available traction can be increased by increasing the angle of undercut in the sheave groove.
3. The available traction can be increased by increasing the actual coefficient of friction between the rope and groove surface.

There is, of course, a limit on the maximum undercut angle since the larger the undercut, the less support the rope receives from the groove, and therefore, the less load we can put on the ropes without causing rapid sheave wear and rapid rope failure. For this reason, the allowable load per rope decreases with a corresponding increase in the undercut angle.

There are two design criteria for the selection of the number of ropes needed:

1. Factor of safety, as prescribed by ASME A17.1 Rule 2.20.3.
2. Limiting rope-to-groove pressure.

Where round seat grooves are used, the rope selection is based on the required factor of safety. However, when undercut grooves are furnished, the rope selection will be based on the allowable rope-to-groove pressure. Each manufacturer decides this latter point. It should be noted that the European elevator code, EN 81-1: 1986, codified the limiting pressure from the following formula:

Metric:
$$p_{max} \leq \left[\frac{12.5 + 4\,V_r}{1 + V_r} \right]$$

where: p_{max} = Specific rope-to-groove pressure (Newtons per square millimeter)

v_r = Rope speed (mps)

Expressed in terms of imperial units, this formula becomes:

$$p_{max} \leq \left[\frac{3.125 + \dfrac{V_r}{196.85}}{1 + \dfrac{V_r}{196.85}} \right]$$

where: p_{max} = Specific rope-to-groove pressure (psi)

v_r = Rope speed (fps)

The Europeans discontinued the rope-to-groove formula in the EN 81-1: 1998 Code. Further research into Hyman's 1926 theory showed that this was not a representative condition for most passenger elevators, and that the pressures should be lower than this formula would yield, leaving it to the manufacturers to set their own limits.

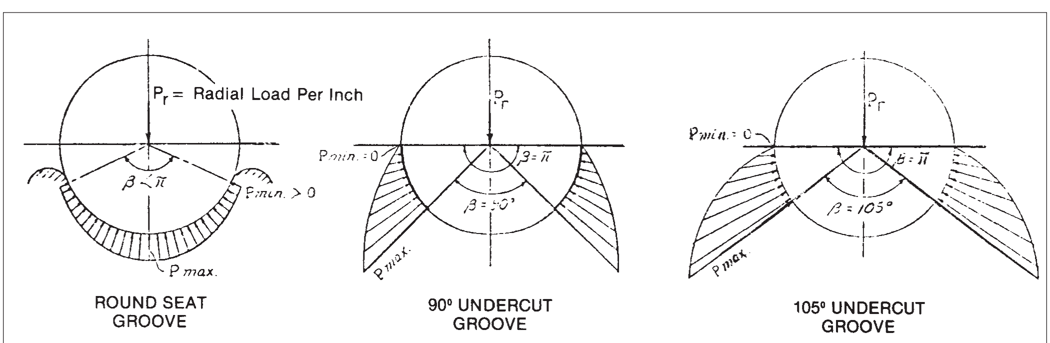

Figure 36

The use of compensating ropes or chains should be covered briefly in concluding this subject since their application directly relates to traction. The purposes of compensation are:

1. To reduce the required traction r-elations
2. To permit best sheave groove for longest rope life (undercut grooves)
3. To reduce the required torque on the elevator driving machine motor
4. To keep the load on the machine constant, irrespective of the position of the car in the hoistway

There are no requirements establishing a maximum permissible amount of rope slippage since there is no way to establish such a limitation that would be suitable for all designs, all capacities and all speeds of elevators. Some rope movement relative to the drive sheaves is normal for all traction elevators and typically manifests itself as creep, which was discussed above.

11. ELEVATOR ENCLOSURES:

ASME A17.1 Code Section 2.14 requires that elevator cars be permanently enclosed on all sides, except the sides used for entrance and exit, and on the top. Further, the enclosure must be securely fastened to the car platform such that it cannot become displaced in ordinary service or upon the application of the car safety or on buffer engagement.

The ASME A17.1 Code Rule 2.141.3 further requires that the enclosure be strong enough to withstand a force of 75 pounds/feet (330 Newtons) applied horizontally at any point on the walls of the enclosure and that the deflection will not reduce the running clearance between the car and counterweight, specified in ASME A17.1 Section 2.5.1, below the minimum, nor to exceed one inch (25 millimeters).

Car enclosure tops must be designed and installed as to be capable of sustaining a load of 300 pounds on any square area two feet on a side and 100 pounds applied at any point.

Materials for car enclosures and car enclosure linings must be metal or glass, or conform to the fire protection requirements that the flame-spread rating be within the range of 0–75, and the smoke-development rating be within the range of 0–450. Samples of the cab materials in their end-use configuration must be tested in accordance with the requirements of ASTM E-84, ANSI/UL 723, NFPA 252 or CAN/ULC-S102.2, whichever is applicable in the jurisdiction in which the elevator is being installed. The reader is referred to ASME A17.1 Rule 2.14.2 (latest edition) for the complete provisions. The two types of enclosures are passenger elevator cabs and freight elevator enclosures, discussed below.

11.1 Passenger Elevator Cabs:

Passenger cabs are generally designed and constructed to meet aesthetic requirements primarily, and structural parameters secondarily. In their most basic form, passenger cabs will consist of a car front which contains the door assembly and car operating fixtures, the car sides and rear sections, ceiling (which may further embody a suspended arrangement), and an emergency top exit. In accordance with the latest ASME A17.1 Code, side emergency exits are prohibited. Depending on the decorative treatments, hang-on panels might be installed on a metal skeleton frame or similar structural arrangement. Handrails would be applied last. A typical cab is shown in Figure 37.

The general door arrangements are center-opening doors, shown in Figure 37, wherein two door panels move in opposite directions and meet in their closed position at the center of the opening. Single-slide doors are frequently used on smaller cars and consist of a single moving door. Two-speed doors are frequently used on hospital elevators where a large door opening is required to accommodate stretchers and beds. In this latter door arrangement, one door moves at twice the speed of the other. Typical door arrangements are shown in Figure 38, taken from *NEII Building Transportation Standards and Guidelines*. These can be seen in more detail at http://www.neii.org.

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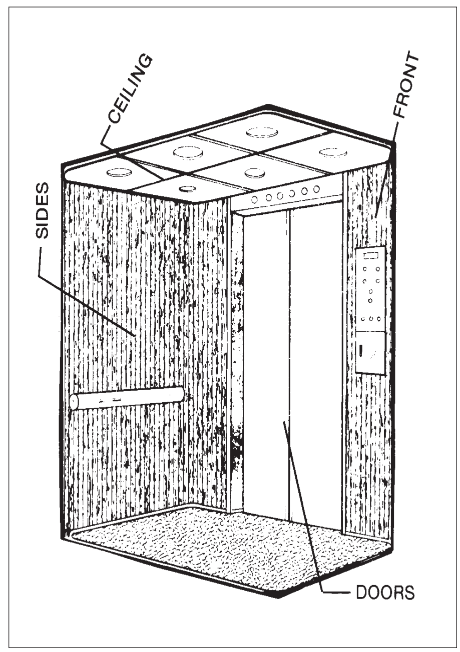

Figure 37

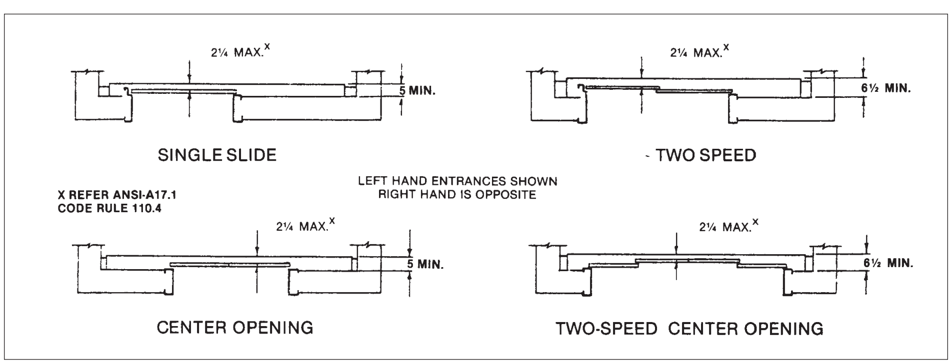

Figure 38

11.2 Freight Elevator Enclosures:

The principal parts of a freight enclosure are:

A. Enclosure panels
B. Extension panels
C. Car top

The specific designs will vary from company to company; however, the primary requirements are that they be structurally sound to meet the loading conditions discussed above. The usual construction embodies sheet steel panels suitably reinforced.

12. DOOR OPERATORS:

Today, all passenger elevator door operators are of the "master" type. This means that the door operator carried on top of the elevator cab operates the cab car door and each of the hoistway doors that the elevator serves.

The only hoistway door that can be opened is the one at the landing at which the elevator stops; thus, you are assured of a very safe situation.

Figure 39 shows an example of an Otis 6970 door operator located in its normal position above the elevator cab. Note the swinging arms that are attached to the door assembly. As the arm swings or is driven by the motor, a horizontal sliding of the door takes place.

It might be well to point out here that a door operator is called upon to open and close doors quickly and quietly. This is very much like an elevator, which is required to start, run, and slow down with imperceptible noise or motion. The acceleration and deceleration follow a modified simple harmonic motion curve. A DC motor is often used since superior control can be achieved. Most elevator companies used harmonic drive door operators until the introduction of linear drive door operators in the early 2000s. Both types serve the market safely and efficiently.

Figure 39 also shows a cam, which is a vertical steel bar used to couple the door to the selected hoistway door. The cam is long enough to permit advance opening of the doors as the elevator levels into the floor.

Figure 40 shows the cam (or "vane," as it is often called).

Now that the principle of the master door operator is established and we know how the safe coupling takes place, we should talk about safety.

The coupled car and hoistway doors represent a weight of 400-600 pounds and, if a person were hit with this moving mass at typical industry door speeds, allegations of door strike incidents would surely ensue.

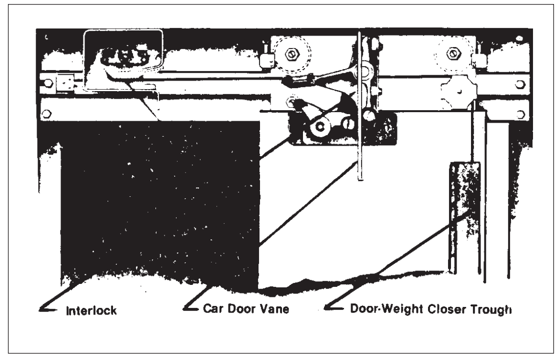

Figure 40

ASME A17.1 Code Rule 2.13.4.1 requires that the kinetic energy, based on an average closing speed, shall not exceed 7 ft-lbf when a reopening device is used, and not more than 2.5 ft-lbf for systems with no reversing device.

ASME A17.1 Code Rule 2.11.3 requires a spring closer to assure that if, for any reason, the car moves away from the floor, the hoistway door will close and protect the area. Since this has been added to the code, many accidents have been avoided.

The author has written more extensively on the subject of kinetic energy, and the reader is referred to these for further reading:

1. "Kinetic Energy of Passenger Elevator Door Systems, Part I – Technical Overview," ELEVATOR WORLD, December 1989.
2. "Kinetic Energy of Passenger Elevator Door Systems, Part II – Mathematical Analysis," EW, January 1990.
3. "Instantaneous Maximum Kinetic Energy of Horizontally Sliding Passenger Elevator Door Systems," EW, April 1997.

13. CAR RIDING QUALITY:

Attendant to the general subject of guides and guide rails is the performance parameter referred to as car riding quality, because this is one of the important criteria by which an elevator's performance is judged from a commercial perspective.

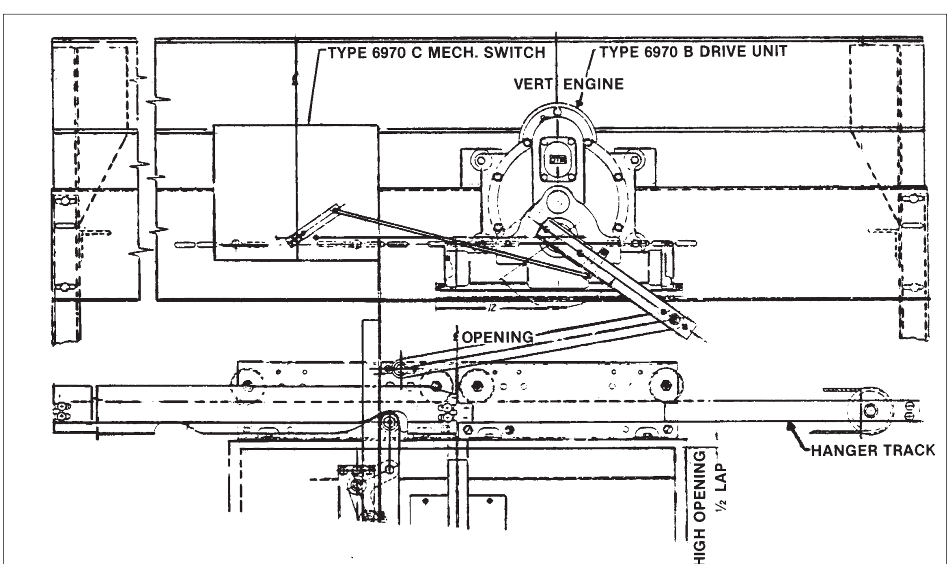

Figure 39

The principal factors which influence car ride are those that produce horizontal excitations (accelerations and retardations) which act upon the car, described as follows:

A. Curvature and twist of the rails caused by misalignment at installation or due to building compression. Long gentle curvatures to the rails occurring over several spans do not represent as big a problem as do distortions occurring in localized spans; however, it should be recognized that, in any case, when the deflection of the guide rail exceeds the available float embodied in the roller guide, an excitation will be imparted to the car.
B. Steps at the rail joints. Being manufactured items, the guide rails are subject to tolerances which may produce a step due to the buildup of tolerances at the tongue-and-grove joint.
C. Deflections of guide rails. Guide shoe forces due to any cause produce rail deflections, the magnitudes of which increase and decrease as a function of the vertical position of the car guide shoes in relation to rail brackets, which, in turn, is a function of the square of the car speed. These forces typically occur as a result of an eccentric live load in the car, which is the most frequent load condition that occurs due to the random manner in which passengers distribute themselves. In addition, unbalanced cars produce guide shoe forces at all positions in the hoistway, and even balanced cars produce guide shoe forces as they move away from the center of the rise, shown below in Figure 41.
D. Transverse rope vibrations included in the hoist ropes and/or compensating ropes caused by hitch points moving sideways, thus causing a transverse excitation to the end of the ropes due to the horizontal deflection of the rails or by hoistway wind disturbances due to stack effects or building sway or due to rope oscillations acting on short rope lengths as the car approaches the upper terminal.

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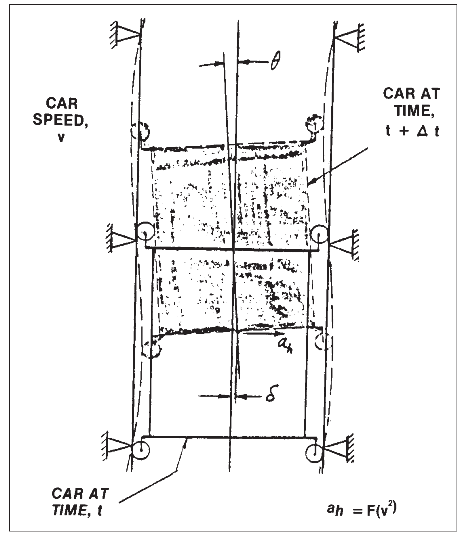


Figure 41

E. Air pressure: The displacement of air around a single car produces forces against the side of the car. In the case of multiple cars, the turbulence produced by adjacent cars passing each other in adjacent common hoistways causes side loads on the cars which, in turn, sets up horizontal accelerations. Further unevenness in air pressure produced by projections extending into the hoistway which cause dimensional variations and configurations in the hoistway produces horizontal accelerations on the car.

Some or all of the above factors will always be present to some degree in every elevator system and, in general, with the same running clearances, same roller guide float, etc., the horizontal accelerations and retardations which influence car riding quality vary as the square of the car speed.

The excitations or disturbances imparted to the car manifest themselves as a sequence of acceleration pulses. The amplitudes of these pulses indicate the severity of the car shaking, the intensity of which is coupled to the frequency of occurrence, rate change of acceleration, etc.

In terms of the passenger's sensitivity to motion, the longer the stroke permitted in the suspension system, which, in this case, comprises the roller guides, the less the acceleration. The limiting values of these strokes are presently dictated by current door vane design, switches, cams and safeties.

A review of some work done in the area of human response to motion suggests that acceptable peak values of acceleration alone are not sufficient to describe car ride unless coupled to additional parameters, namely:

a. Duration, or time interval over which the acceleration occurs
b. Onset, which is the time rate at which an acceleration builds up, commonly known as "jerk"
c. Frequency (Hz)
d. Total excursions (magnitude of displacements)

A Japanese source, Hitachi, Ltd., published an article on "Analysis of Lateral Quaking of High-Speed Elevators" (EW, March 1973), which discussed its own standard for riding comfort in terms of accelerations and frequency ranges. While this published data might not have been completely documented in the article, the results may be viewed qualitatively and support the point that acceleration values are insufficient as sole criteria to judge car ride.

A major difficulty in modeling and evaluating human body response to vibration arises from the nature of the human body itself. There is no "standard" human body, due to variations in size, age, sex, weight, etc. Therefore, any conclusion as to sensitivity levels must consider ranges of values which encompass the parameters noted above.

The establishment of car ride standards involves determining levels of comfort based on subjective opinion, an inexact approach at best. The North American elevator industry has done little work in this area of human response to car vibration, but there has been extensive work done in the field of biomechanics to develop data on human sensitivity levels. In the absence of a rational basis for car ride standards, we should pursue the acceptance of a range of values defining acceptable car ride as opposed to setting a precise value of acceleration which implies an exactness which does not exist.

There has been material published on this subject of lateral accelerations imparted to people and certain criteria published by the International Organization for Standardization (ISO), as well as studies at Dulles International Airport by the Department of Transportation, some 25 years ago.

As a point of information, the Japanese elevator industry met the problem of improving car ride quality on a high-rise, high-speed installation in the late 1970s, leading to the 8 Gal (approximately 8 milli-g) peak-to-peak acceleration, by doing several things:

a. The guide rail finish was upgraded.
b. Special guide rail joint configurations were introduced.
c. Extreme care was exercised in the installation.
d. Elimination of projections into the hoistway insofar as possible.
e. Adding airflow deflectors to the top and bottom of the car.
f. Adding traveling rope carriages to effectively reduce free rope lengths, thus providing vibration nodes.

Several years ago, manufacturers, consultants, architects and users became interested in seeing a uniform methodology for measuring car ride quality adopted on a worldwide basis which could be used as a standard. As a result of this growing interest, the International Standards Organization through ISO Technical Committee 178 established a technical working group, ISO/TC178 WG9, which developed an accepted measurement and reporting standard, now published as *ISO 18738: 2003 Measurement of Lift Ride Quality.* The standard defines and uses performance parameters where they are integral to the evaluation of ride quality, but does not specify acceptable or unacceptable ride quality. Parameters relevant to lift performance include jerk and acceleration. It is available on the ISO website at: www.iso.org/iso/iso_catalogue/catalogue_tc/catalogue_detail.htm?csnumber=33586&commid=53970.

14. BUILDING SWAY:

The modern trend in building construction is ever-increasing height and the design of lightweight buildings has been made possible through the use of sophisticated analytical methods on the computer. Accordingly, more efficient use is made of the structural materials. The resultant tall and flexible buildings may have large wind-induced motions which are not only perceptible within human thresholds, but also impact on the elevator systems, principally the compensating ropes and traveling cables. Notwithstanding, a well-designed elevator system is expected to operate under these conditions without hazard to the passengers.

By way of definitions, the following terms are reviewed:

A. Frequency – the number of times that a body moves up and down (vibrates) or from side-to-side in a unit of time.
B. Natural Frequency – the frequency at which a body vibrates without the action of a forced or disturbing frequency. For example, a weight suspended on a spring. If the weight is pulled down and released, the weight will move up and down at its natural frequency.
C. Forced Frequency – the disturbing frequency or vibration.
D. Resonance – the condition which occurs when the forced frequency is the same as the natural frequency and results in a multiplication of the periodic reactions of the forced vibrations on the supporting structure or floor.
E. Amplitude – the distance the vibrating body moves up or down or from side-to-side in one cycle.
F. Damping – frictional resistance that is present in all practical mounts whether rubber or spring.

Damage to elevator equipment as a consequence of building sway is a function of a number of factors, the most significant of which includes the natural frequency or period of a building, elevator travel (rise) and various cable loadings.

The specific danger occurs when cable and/or ropes, due to their individual lengths and tensions, tend to sway at a frequency at or near this building frequency. This resonance, or near resonance, condition can, in a very short time, induce large cable and/or rope displacements, resulting in a variety of damaging conditions to the elevator equipment.

Building deflections vary considerably as a function of construction. The Empire State Building, in New York City, deflects about 6.5 inches from center in an 80-miles-per-hour wind. On August 9, 1976, instrumentation in the World Trade Center, also in New York City, recorded 18.5 inches of building motion from center at the 110th floor during Hurricane Belle at comparable wind bursts.

Because the newer buildings utilize less concrete, plaster, brick, masonry, etc., for load-bearing structures as is typically found in older buildings, the new buildings possess less damping so that once building motions are induced due to wind, they persist through many cycles.

Since elevators are confined to travel along the path dictated by the guide rails, they sway with the building in a horizontal plane, but in addition, they have a vertical velocity, as well. This creates a wave form in those flexible items which hang from the elevator car frame. The forcing frequency impressed upon the elevator system will be the natural frequency of the building itself. The horizontal motion of the elevator will force the various flexible media, i.e., hoist ropes, compensating ropes, traveling cables, governor ropes, selector tape, etc., to vibrate. See Figure 42.

The effect will be similar to pushing a child on a swing; a small force acting through a small distance will cause an increasing length of arc through which the child swings; provided that the force is repeated and is synchronized with the swing.

If the various flexible members, such as compensating ropes and traveling cables, have a natural frequency which matches the forced frequency imposed on the elevator system, resonance will take place with resultant high magnitudes of amplitude. Since these flexible members have very little damping, they tend to accumulate large amounts of kinetic energy if allowed to swing through many cycles. This will take place even if the elevator is not moving vertically. In fact, the longer the elevator remains at a given location, the more kinetic energy will develop. Repositioning the elevator will change the natural frequency of the various cables to non-critical values in a majority of cases.

Several items contribute to the amplitude of vibration of the various flexible ropes and cables. Principally, the maximum elastic deflections of the building due to wind gusts, the number of successive gusts occurring in a moderate time period, the natural frequency of the building in relation to the forcing frequency and the damping present.

Continued

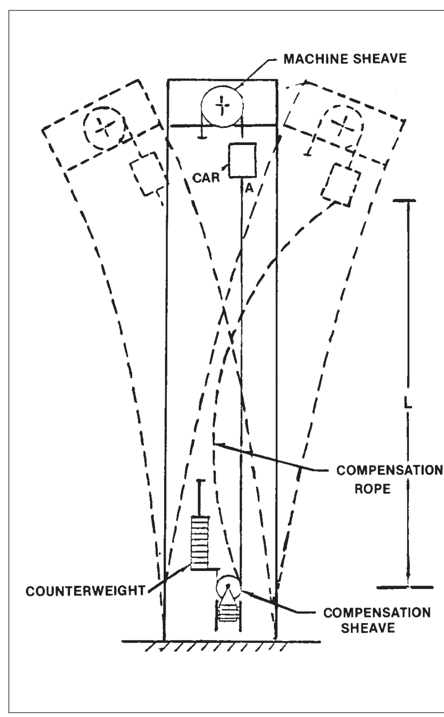


Figure 42

If an elevator is in the upper part of the building when the onset of building sway occurs and excites the compensating ropes so that they sway, once the car starts to descend at full speed, centrifugal forces, which are proportional to the square of the speed, are induced thus causing more violent swaying/whipping of the decreasing rope length. The increased excursions of the compensating ropes may cause them to strike components or brackets in the hoistway.

Because of the high kinetic energy that the ropes possess, once swinging, the high car speeds, especially in the down direction, set up rope excursions of increasing amplitude. Once large building sway-induced rope or cable motions in the hoistway are induced, the cables will swing out into doors, interlocks and hoistway switches, and snag on rail brackets and twist around one another, thus becoming entangled. In the case of compensating ropes, this tangled set of ropes cannot travel over the compensating rope sheave in the pit without causing serious rope or sheave frame damage.

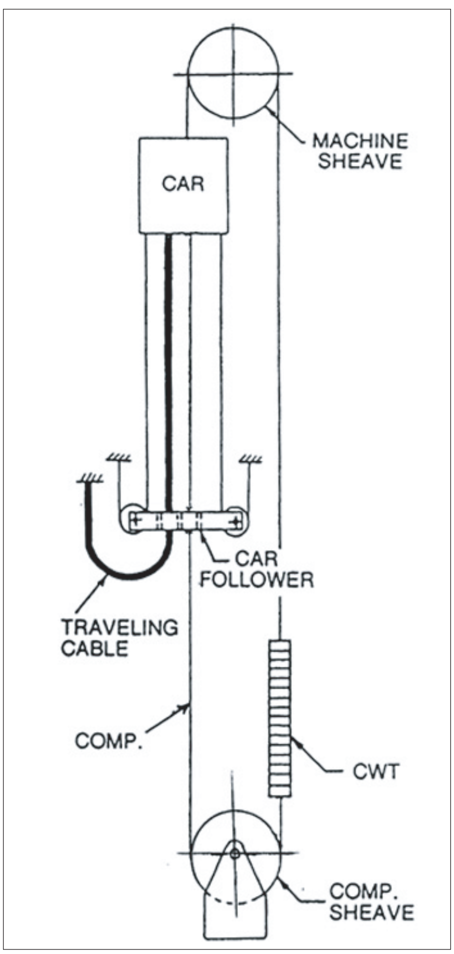


Figure 43 (Reprinted with permission from the ***Vertical Transportation Handbook****)*

Mathematical analysis has shown that the factors affecting rope or cable motion are all related to the mass of the rope, its length and tension. The rope termination, fixed, free, etc., is also an important parameter. Accordingly, there is a number of solutions available to address the building sway problem. These include the detuning techniques of changing the rope tension, which alters the natural frequency of the rope, and modifying the effective length of the rope, which changes its natural frequency. This latter issue of changing rope lengths is easily accomplished by parking the elevators at certain floors, thereby altering the free length of rope or cable hanging below the car. In addition, reducing the car speed at the onset of detected building sway will reduce the severity of swaying ropes and cables, either manually or automatically. There may be several other detuning techniques that each specific manufacturer may employ that will not be further covered here.

In general, the building sway problem seems to become critical as buildings approach the 40th-floor mark. When planning the elevator equipment for such large buildings, it is important that building oscillation-related data be obtained from the building designers so that appropriate elevator design considerations can be made. This data includes the periods of building oscillation (seconds) for the two principal axes, i.e., directions of the building, as well as twist, percentage of critical damping and the expected peak amplitude of building deflection along both building axes and wind speed. This data should be based on full occupancy in the building. The building axes with respect to the north/south direction should be included. For the specific cases of super high-rise buildings, where the above detuning techniques are insufficient, the elevator designer might have to consider the use of rope follower guides.

The follower rope guide, also referred to as the traveling carriage, is a device used to prevent compensation ropes and traveling cables from large-amplitude swing during high-wind weather. By roping it 2:1, the follower travels at half the speed of the elevator car. Its purpose is to divide the free length of the hanging ropes, cables, etc. effectively in half irrespective of the position of the car, thus changing the natural frequency of the ropes, cables, etc. It is hoisted upward by the elevator car and lowered by gravity of its own weight. See Figure 43. This method was first disclosed in a Japanese patent, now expired.

When the car is near the bottom of the hoistway, at which point the follower is very close to the car plank, restraint to lateral motion of the compensating ropes and traveling cables is no longer effective. However, the part of compensating ropes between the counterweight and the rope compensation sheave is close to the hoistway wall. Any lateral motion causing sway of this run of ropes would result in the ropes hitting the hoistway wall, causing them to detune.

An important factor in the design of the follower rope guide is to have sufficient weight to enable it to track properly as it rides on the car guide rails. Heavier follower weight smooths its downward motion especially when rail alignment is not good. Accordingly, heavier structural members are purposely selected for the construction and, as a result, the strength of every component is usually many times greater than actually required for any loads. If the traveling carriage is designed symmetrically about its own geometric center, it will be subjected to no external forces except its own weight. With a symmetrical follower, no appreciable forces can be expected to be impressed on the roller guides.

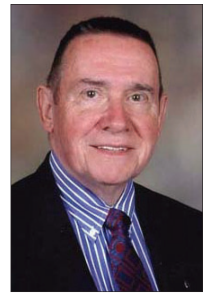
Another important design consideration is to have the proper relationship between the vertical and horizontal distances between the roller guides of the follower. The ratio of the vertical wheel base to the horizontal distance between guide rails must be at least 1, preferably 1.25, in favor of the vertical distance. This is a basic technical requirement to make sure that the carriage rides up and down freely without hanging up, breaking ropes, jumping rails, etc. Rope follower guides have been successfully installed by Otis at the World Trade Center in New York City, and by Westinghouse at the Sears Tower in Chicago.

The structural frame of the follower is suspended by a system of ropes with one end of each rope hitched to the car frame plank and the other near the middle of the hoistway. An important design consideration relates to the proper tensioning of the traveling carriage guiding ropes. On the one hand, it is undesirable to develop slack rope. Therefore, a slack cable switch is important. On the other hand, it is also undesirable to overtighten the ropes. Therefore, an additional switch is necessary. An effective arrangement is obtained by using a bidirectional cam and switch to monitor both conditions.

Larning-Reinforcement Questions

Use the below learning-reinforcement questions to study for the Continuing Education Assessment Exam available online at www.elevatorbooks.com or on page 113 of this issue.

- Why is it important to periodically examine the hoist ropes during their life cycle?
- Why is the type of rope lubricant important?
- What are the advantages and disadvantages of pre-formed ropes?
- What was the incentive leading to the development of the traction drive for tall buildings?
- Why is it important to have a basic understanding of the principles of rope traction?
- Where does the maximum pressure between the drive sheave grooves and the hoist ropes occur in a drive sheave with undercut grooves?
- If the required traction were more than the available traction, what effect might occur if a fully loaded car descended toward the bottom terminal landing, or an empty car ascended into the top floor?
- Why is it necessary to fasten a cab to the platform of a traction elevator?
- Describe the common door arrangements used for passenger elevators. What makes each type popular?
- Why does the ASME A17.1 Safety Code limit the kinetic energy of closing horizontal doors on passenger elevators? Do these same limits apply if there is no door reopening device provided?
- What is the effect of air pressures induced in the hoistway where there are multiple cars?
- What are the effects on the elevator systems due to high winds acting on the outside of a high-rise building?



George W. Gibson has an extensive technical and managerial background in the elevator industry, spanning over 50 years. He is a graduate civil engineer and has taken extensive post-graduate studies in advanced theories of structures, mechanics and mathematics. He is the chairman of the Advisory Board of NAESA International, a past regent of the Elevator Escalator Safety Foundation and a member of the Board of Executives of the International Association of Elevator Engineers. He is a member of the board of directors of Elevator World, Inc. Gibson retired early from Otis Elevator Co. in January 1993. In a career spanning 37 years with Otis, he has held several design engineering, engineering management and corporate management positions, including manager of mechanical R&D, manager of product engineering, manager of mechanical engineering, director of engineering administration, and director of codes and product safety. He is the president of George W. Gibson & Associates, Inc., an elevator consulting firm specializing in elevator technology, strategic technical planning, codes and standards, product safety, and technical support of litigation. He is a member of the American Society of Mechanical Engineers (ASME), ASME Board of Directors on Codes and Standards, ASME A17 Elevator Safety Code Standards Committee, and chairman of the A17 Mechanical Design Committee, A17 International Standards Committee, A17 Earthquake Safety Committee, A17 Ad Hoc Committee on Door Protection, A17 Ad Hoc Committee on Elevator Stopping, and a founding member of the QEI Standards Committee. In his role as chairman of the ASME A17 International Standards Committee, he has been the head of the U.S. delegation to the International Standards Organization Technical Committee 178 on Elevators and Escalators since 1981. In 1997, he was the recipient of the ASME Codes and Standards Medal, and was awarded the ASME grade of fellow, and the ASME Dedicated Service Award in 2007.

ELEVATOR WORLD Continuing Education Assessment Examination Questions

Instructions:

- Read the article **"Elevator Hoistway Equipment: Mechanical and Structural Design, Part II"** (page 91) and study the learning-reinforcement questions.
- To receive **two hours** of continuing-education credit, answer the assessment examination questions found below online at www.elevatorbooks.com or fill out the ELEVATOR WORLD Continuing Education Reporting Form found overleaf and submit by mail with payment.
- Approved for Continuing Education by **NAEC for CET and NAESAI for QEI**.

1. "Rope lay" is a common term in the steel wire rope industry used to designate a length:
   a. Equal to the rope diameter.
   b. Equal to one wrap over the drive sheave.
   c. Required for one strand to complete a helical turn around the rope.
   d. Equal to the distance between two adjacent ropes on the drive sheave.

2. One of the most important factors that affects the life of a set of hoist ropes on a regular passenger elevator is the ratio of the drive sheave diameter to the rope diameter. The ASME A17.1 Safety Code requires that this ratio:
   a. Must equal 40.
   b. Must not be less than 30.
   c. Must not exceed 54.
   d. Must not be less than 40.

3. Construction stretch of hoist ropes is:
   a. Reversible when the car load is removed.
   b. A characteristic of new ropes.
   c. Caused by the outer wires becoming embedded in the drive sheave grooves.
   d. Caused by the rope lubrication being squeezed out.

4. The major inherent safety feature of a traction drive is:
   a. The drive sheave grooves are metallic.
   b. The low maintenance required to sustain it for long periods of time.
   c. Its ability to lose traction if either the car or counterweight bottoms on its buffer.
   d. The available traction is independent of the rope lubrication.

5. In the design of a traction drive, it is necessary to evaluate the hoist rope tensions and the properties of the drive sheave groove in order to ensure that:
   a. The required traction is greater than the available traction.
   b. The creep of the ropes is within A17.1 Code limits.
   c. The available traction is greater than the required traction.
   d. There is no rope creep.

6. In a drive sheave in which an undercut groove has been furnished at the bottom of the grooves, the pressures between the rope and groove surface will be at their maximum:
   a. At or near the tops of the grooves.
   b. At the edges of the groove undercut.
   c. At the bottom of the groove undercut.
   d. At the top of the rope where the strands stretch more than at the bottom.

7. The available traction can be increased in a number of ways. Which of the following is not correct?
   a. By increasing the actual coefficient of friction between the hoist rope and drive sheave groove surface.
   b. By increasing the arc of contact (angle of wrap) that the hoist rope makes with the drive sheave.
   c. By increasing the angle of undercut in the compensating sheave grooves.
   d. By increasing the angle of undercut in the drive sheave grooves.

8. Which of the following is correct? The ASME A17.1 Safety Code requires that there be sufficient traction available between the drive sheave grooves and hoist ropes to:
   a. Safely stop and hold the car with 150% of the rated load.
   b. Safely hold the car with 120% of the rated load.
   c. Safely stop and hold the car with 125% of the rated load.
   d. Safely stop the car with 125% of the rated load.

9. The purpose of securely fastening the elevator cab to the platform is:
   a. To make sure that it doesn't become displaced during normal service, or during safety or buffer tests.
   b. To make the car sound-proof.
   c. To prevent rattling noises.
   d. None of the above.

10. Two-speed car and hoistway doors are frequently used on hospital elevators to accommodate stretchers and beds. The advantages are:
    a. Passenger transfer is enhanced with a wide door opening in a long car.
    b. A large car door opening results with little wasted space in the hoistway adjacent to the car.
    c. A large car door opening results with little wasted space in the pit.
    d. A small door opening results with little wasted space in the building overhead.

11. A master type of door operator functions to:
    a. Only open the sliding car doors.
    b. Only open the sliding car doors and hoistway doors together.
    c. Open the sliding car doors and swing hoistway doors.
    d. Open and close sliding car doors and hoistway doors together.

12. The measure of impact from horizontally closing passenger elevator doors is given by the ASME A17.1 Code as kinetic energy based on the average door closing speed in the code zone distance. Which of the following is correct? Continued

a. The A17.1 Code specifies no limits when "nudging", i.e., no reopening device, is used.
b. The A17.1 Code specifies the limiting kinetic energy when a door reopening device is used, and reduces that allowable value when there is no reopening device.
c. The A17.1 Code allows higher values when an attendant controls the door closing.
d. The A17.1 Code specifies no limits when a door reopening device is used.

13. The principal factors which influence car riding quality are those which produce horizontal forces (accelerations and retardations) which act upon the car. Which of the following is incorrect?
    a. Guide rail deflections in between rail brackets.
    b. Air pressure caused by adjacent cars passing each other in adjacent common hoistways.
    c. Curvature and twist of the guide rails.
    d. Unequal hoist rope tensions.

14. The undesirable effects of building sway on the elevators caused by external winds in a high-rise building can be mitigated by a number of strategies. Which of the following is incorrect?
    a. Changing the hanging lengths of ropes and cables by parking the cars at selected floors.
    b. Employing traveling carriages to reduce the hanging lengths of ropes and cables.
    c. Reducing the car speed.
    d. Parking all the elevators at the upper or lower terminals.

15. When traveling carriages are used to divide the free length of ropes and cables in half, the operational speed of these carriages will be:
    a. The same as the car speed.
    b. Twice the car speed.
    c. Half the car speed.
    d. One-third of the car speed.

ELEVATOR WORLD
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5.	a	b	c	d	13.	a	b	c	d
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