Chapter 6

AMERICAN RAILWAY ENGINEERING AND MAINTENANCE OF WAY ASSOCIATION
Practical Guide to Railway Engineering

Railway Track Design
Railway Track Design

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Railway Track Design

Basic considerations and guidelines to be used in the establishment of railway horizontal and vertical alignments.

The route upon which a train travels and the track is constructed is defined as an alignment. An alignment is defined in two fashions. First, the horizontal alignment defines physically where the route or track goes (mathematically the XY plane). The second component is a vertical alignment, which defines the elevation, rise and fall (the Z component).

Alignment considerations weigh more heavily on railway design versus highway design for several reasons. First, unlike most other transportation modes, the operator of a train has no control over horizontal movements (i.e. steering). The guidance mechanism for railway vehicles is defined almost exclusively by track location and thus the track alignment. The operator only has direct control over longitudinal aspects of train movement over an alignment defined by the track, such as speed and forward/reverse direction. Secondly, the relative power available for locomotion relative to the mass to be moved is significantly less than for other forms of transportation, such as air or highway vehicles. (See Table 6-1) Finally, the physical dimension of the vehicular unit (the train) is extremely long and thin, sometimes approaching two miles in length. This compares, for example, with a barge tow, which may encompass 2-3 full trains, but may only be 1200 feet in length.

These factors result in much more limited constraints to the designer when considering alignments of small terminal and yard facilities as well as new routes between distant locations.

The designer MUST take into account the type of train traffic (freight, passenger, light rail, length, etc.), volume of traffic (number of vehicles per day, week, year, life cycle) and speed when establishing alignments. The design criteria for a new coal route across the prairie handling 15,000 ton coal trains a mile and a half long ten times per day will be significantly different than the extension of a light rail (trolley) line in downtown San Francisco.
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<table>
<thead>
<tr>
<th>Carrier</th>
<th>Horsepower per Net Ton</th>
<th>Horsepower per Passenger</th>
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<td>Conveyors</td>
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Table 6-1 Typical Horsepower-per-net-ton Ratios

6.1 Stationing

Points along an alignment are usually defined by miles, stationing or both. The latter is customary with railway routes throughout North America. Within yards, terminals, and sidings, the miles (termed ‘mileposts’ or ‘mile boards’) are dropped due to the relative close proximity of the tracks to a common point. Stationing (also termed ‘chaining’) is merely the sequential numbering of feet from a beginning point to an ending point. A single station is 100 feet long in US units or 1000 meters in metric units. A point one mile from a beginning station of 0+00 would then be denoted station 52+80 (or 52.8). In metric, that same point would be 1+600.

At the time of construction, all alignments had stationing. Most items along an alignment can be located by stations. This is the primary system used for locations within many engineering records. However, if an alignment has been in place for any long period of time, such as most North American railways, it likely has been changed or relocated since its original construction. These changes usually introduce what is termed a station equation, which is required because the relative length of the alignment has been changed with the alteration. Other causes for a station equation (but certainly not all grounds) include the combination of two separate routes, lost records, or an extended period of time between the stages of construction for the overall alignment.

Mileposts are more commonly used by operating departments for location identification. Though less precise, they are more easily identified and they are referenced along the right-of-way with signs. Bridges are normally identified by...
mileposts, though they also have stationing associated with them. Likewise, it is not uncommon for mileposts to have stationing shown in railway records.

Both the use of mileposts and stationing for the reference of existing railway features are not without pitfalls. This is of concern to the designer when contemplating work along an existing track. The direction of increasing stationing and increasing mileposts may not be the same. There is no guarantee that the records maintained by a railway are correct, or have the most current information (this is more often the case). It is not unheard of for a railway to have re-stationed a line, or even given new mileposts. There are lines on which this has occurred at least two or three times since construction. Though the stationing and mileposts may have changed on the alignment records, many old right-of-way instruments, bridge plans and other information may still reference what was there and not what is there today.

The use of milepost information is particularly hazardous for several reasons. First, the initial stationing over 100 years ago to establish mileposts was not always significantly accurate. The actual length between mileposts may vary by thousands of feet, though most are reasonably close (less than 100-200 feet). Adding further variance to the length was the common railway practice to place the milepost marker on the nearest telegraph pole rather than on a dedicated signpost. As the poles were moved, replaced and changed, the sign moved with them. Signs were lost and replaced, but probably not relocated with any great precision.

Stationing to the mileposts, along with other items which have a tendency to be somewhat transient over the long term, including grade crossings, turnouts, rail rests, etc., should always be subject to much scrutiny before being used as a basis for design. The designer should always establish existing stationing from some item, which has not moved in some time, preferable the abutment of an older structure or culvert, or best of all, a defined right-of-way corner or marker. Though the milepost location and terminology will not generally change as a result of re-establishing its true location, it will provide a frame of reference for the location of new facilities.

### 6.2 Horizontal Alignments

Nearly any alignment can be physically defined with variances of two components: tangents and curves. Horizontal alignments of existing and proposed railway tracks generally are given the highest interest as their location seem to be the easiest to grasp when reviewing the location of facilities relative to one another.

A tangent is simply a straight line between two points. Tangents are usually denoted with bearings (N 3°23'59" E for instance). However, it must be noted that without an accompanying starting point and length associated with that bearing (and thus establishing the location of the second point), there is no way to definitively establish
the tangent’s location in space. Other points along a given tangent can be defined in this manner. Tangents, because they are the most defining parts of alignments and are usually the components used in the establishment of such, should be considered the highest order component. Curves as discussed below, which effectively connect these tangents, are second order as they are fundamentally defined by the location of tangents and can be easily changed without relative wholesale shifts in physical alignment location.

Where an existing tangent must be established and where two points are not easily defined or known, obtain at least three points, which are believed to be along this line. Because a tangent can be defined by only two points, two points located along a curve can define a tangent. It is only through working from at least three points and comparing the bearings established relative to each other, that a true tangent can be established. Though the difference in bearing between three points on a tangent should be zero, the precision afforded by surveying equipment and construction methods is generally less than that calculated from data obtained, particularly when the person performing the calculation has no appreciation for significant digits. Most means for performing linear regression on a set of data points for the purpose of establishing tangents have no allowance for this situation.

Therefore, it must be understood when reviewing the data collected between points, there is a margin within which any three points can be assumed to be tangent. This margin is based upon the judgment of the designer and takes into consideration the relative condition of the existing item upon which the tangent is to be defined, the level of accuracy required, and the overall margin of error, which limits the functionality of the facility.

An alignment comprised of more than one tangent will generally include a set of points known as Points of Intersection, or ‘PI’s.’ The defining points of each tangent are shared with those two tangents to which are immediately adjacent to it. As these points define the tangents, as well as any points, which may have defined the location of the connecting tangents, they should be considered the cardinal points of the alignment. Though second order points, such as Points of Curve (PC’s) and points along curves, can be defining, it is the existence of the PI, which must exist for a curve to exist. It is the PI that will remain constant between two tangents despite what changes are made to the curvature itself.

Curves are alignment elements allowing for easy transition between two tangents. Horizontal curves are considered circular though they are actually arcs, which represent only a portion of a complete circle. All curves can be defined by two aspects. The angle of deflection (I) is defined at the Point of Intersection (PI) by the difference in bearing between the two tangents. This aspect is fixed by the tangents. With I, the curve may be defined by any of the other following aspects (See Figure 6-2).
Curves are generally specified in one of two ways, by Degree of Curve or by Radius R. Degree of curve can be defined in two ways. The chord definition ($D_c$) is defined as the angle subtended per 100-foot chord. The arc definition ($D_a$) is defined as the angle subtended per 100-foot arc. (See Figure 6-3) In either case, the severity or sharpness of the curve is specified as the degree of curve, with larger numbers representing tighter (smaller radius) curves. Though the differences between the chord definition and arc definition are slight at smaller degrees of curvature, the difference gets progressively larger as the curves get tighter (See Figure 6-4). Furthermore, chord defined curves are stationed about the chords subtended, while arc defined curves are stationed about the actual path of the curve (or arc). Again, the differences are slight at small degrees of curvature, but increase, as the curves get sharper. The stationing difference is further magnified by the length of curve.
North American freight railways use the chord-defined curve exclusively. This is in contrast to highway design, some light rail systems and nearly all other alignments historically and currently being designed with arc defined curves. Though the individual differences between chord and arc defined curves may be considered slight for specific curves, this difference can be magnified considerably on longer alignments with moderate amounts of curvature.

Though a curve denoted by a degree of curve is easily recognized and accepted by most engineers as establishing a certain severity of curvature, the relationship between two curves with different degrees of curvature is not as widely comprehended. It must be understood, that the radius of a six-degree curve is not exactly half of that of a three-degree curve. Due to the sinusoidal nature of the formulae, which produce the degrees of curve nomenclature, the relative differences in radii are more logarithmic. For example, the radius for a two-degree curve is 2864.93 feet and 2292.01 feet for a two-and-a-half-degree curve. This compares with 478.34 feet and 459.28 feet for twelve and twelve-and-a-half-degree curves respectively.

There have been some alignments established about the turn of the 20th century in mountainous areas along the west coast, which used curves defined by the angle subtended by a 50-foot chord. It is not known if or how many of these alignments and records may still exist today. There has been some reference made to defining metric

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Figure 6-4 Chord Length vs. Arc Length for Degree of Curve

Figure 6-5 Degree of Curve to Radius Relationship
curves as \( D \) (degrees per 20 meter arc). However, there does not seem to be any widespread incorporation of this practice. When working with light rail or in metric units, current practice employs curves defined by radius.

As a vehicle traverses a curve, the vehicle transmits a centrifugal force to the rail at the point of wheel contact. This force is a function of the severity of the curve, speed of the vehicle and the mass (weight) of the vehicle. This force acts at the center of gravity of the rail vehicle. This force is resisted by the track. If the vehicle is traveling fast enough, it may derail due to rail rollover, the car rolling over or simply derailing from the combined transverse force exceeding the limit allowed by rail-flange contact.

This centrifugal force can be counteracted by the application of superelevation (or banking), which effectively raises the outside rail in the curve by rotating the track structure about the inside rail. (See Figure 6-6) The point, at which this elevation of the outer rail relative to the inner rail is such that the weight is again equally distributed on both rails, is considered the equilibrium elevation. Track is rarely superelevated to the equilibrium elevation. The difference between the equilibrium elevation and the actual superelevation is termed underbalance.

Though trains rarely overturn strictly from centrifugal force from speed (they usually derail first). This same logic can be used to derive the overturning speed. Conventional wisdom dictates that the rail vehicle is generally considered stable if the resultant of forces falls within the middle third of the track. This equates to the middle 20 inches for standard gauge track assuming that the wheel load upon the rail head is approximately 60-inches apart. As this resultant force begins to fall outside the two rails, the vehicle will begin to tip and eventually overturn. It should be noted that this overturning speed would vary depending upon where the center of gravity of the vehicle is assumed to be.

There are several factors, which are considered in establishing the elevation for a curve. The limit established by many railways is between five and six-inches for freight operation and most passenger tracks. There is also a limit imposed by the Federal Railroad Administration (FRA) in the amount of underbalance employed, which is generally three inches for freight equipment and most passenger equipment.
Underbalance limits above three to four inches (to as much as five or six inches upon FRA approval of a waiver request) for specific passenger equipment may be granted after testing is conducted.

Track is rarely elevated to equilibrium elevation because not all trains will be moving at equilibrium speed through the curve. Furthermore, to reduce both the maximum allowable superelevation along with a reduction of underbalance provides a margin for maintenance. Superelevation should be applied in 1/4-inch increments in most situations. In some situations, increments may be reduced to 1/8 inch if it can be determined that construction and maintenance equipment can establish and maintain such a tolerance. Even if it is determined that no superelevation is required for a curve, it is generally accepted practice to superelevate all curves a minimum amount (1/2 to 3/4 of an inch). Each railway will have its own standards for superelevation and underbalance, which should be used unless directed otherwise.

The transition from level track on tangents to curves can be accomplished in two ways. For low speed tracks with minimum superelevation, which is commonly found in yards and industry tracks, the superelevation is run-out before and after the curve, or through the beginning of the curve if space prevents the latter. A commonly used value for this run-out is 31-feet per half inch of superelevation.

On main tracks, it is preferred to establish the transition from tangent level track and curved superelevated track by the use of a spiral or easement curve. A spiral is a curve whose degree of curve varies exponentially from infinity (tangent) to the degree of the body curve. The spiral completes two functions, including the gradual introduction of superelevation as well as guiding the railway vehicle from tangent track to curved track. Without it, there would be very high lateral dynamic load acting on the first portion of the curve and the first portion of tangent past the curve due to the sudden introduction and removal of centrifugal forces associated with the body curve.

There are several different types of mathematical spirals available for use, including the clothoid, the cubic parabola and the lemniscate. Of more common use on railways are the Searles, the Talbot and the AREMA 10-Chord spirals, which are empirical approximations of true spirals. Though all have been applied to railway applications to

\[
V_{\text{max}} = \frac{E_a + 3}{0.0007D}
\]

\(V_{\text{max}}\) = Maximum allowable operating speed (mph).
\(E_a\) = Average elevation of the outside rail (inches).
\(D\) = Degree of curvature (degrees).

Figure 6-7 Overbalance, Equilibrium and Underbalanced

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some degree over the past 200 years, it is the AREMA 10-Chord spiral, which gained acceptance in the early part of the 20th century. The difference in results between the AREMA 10-chord spiral and a cubic parabola upon which it was based are negligible for $\Delta$’s less than 15°, which is sufficient for all situations except some tight light rail curves.

Spirals are defined by length in increments of ten-feet. There are two criteria generally used for the establishment of spiral length. The first is the rotational acceleration of the railway vehicle about its longitudinal axis. The second is the limiting value of twist along the car body. The rotational acceleration criteria will generally only apply at higher speeds. In the event that the rotational acceleration dictates a spiral, which is too long for the location desired, the shorter car body twist value can be used. Though AREMA has long established values for spiral lengths based upon these criteria, many railways use other criteria.

Referencing Section 3.1.1 of the AREMA Manual for Railway Engineering, the recommended formula for the minimum length of the spiral is:

$$L_{(\text{min})} = 1.63(E_u)V$$

Where

- $L_{(\text{min})}$ = desirable length of the spiral in feet
- $E_u$ = unbalanced superelevation in inches
- $V$ = maximum train speed in miles per hour

For specialty passenger equipment equipped with car-body roll mechanics with respect to the track, AREMA recommends the following formula for length of spiral:

$$L_{(\text{min})} = 62 E_a$$

Where

- $L_{(\text{min})}$ = desirable length of the spiral in feet
- $E_a$ = actual elevation in inches

In locations where obstructions make it impossible to provide a spiral of desired length or where the cost of realignment would be prohibitive, the short spiral as defined by:

$$L_{(\text{min})} = 1.22 E_u V$$

may be used.

Where

- $L_{(\text{min})}$ = desirable length of the spiral in feet
- $E_u$ = unbalanced elevation in inches
- $V$ = maximum train speed in miles per hour

The Transportation Research Board (TRB) recommends an additional formula for spiral length for light rail vehicles:

$$L_{(\text{min})} = 1.13 E_v V$$

in addition to the AREMA formulae.
Where \[ L_{(\text{min})} = \text{desirable length of the spiral in feet} \]
\[ E_u = \text{unbalanced elevation in inches} \]
\[ V = \text{maximum vehicle speed in miles per hour} \]

Spiral curves can be staked out by using either the deflection angle method or by using offsets from the tangent. The following procedures are provided from the AREMA Manual for Railway Engineering, Sections 3.1.3 through 3.1.7.

### Notations for Spiral Curve

- \( D \) = degree of circular curve
- \( d \) = degree of curvature of the spiral at any point
- \( l \) = length from the T.S. or S.T., to any point on the spiral having coordinates \( x \) and \( y \)
- \( s \) = length \( l \) in 100-foot stations
- \( L \) = total length of spiral
- \( S \) = length \( L \) in 100-foot stations
-\( \delta \) = central angle of the spiral from the T.S. or S.T. to any point on the spiral
-\( \Delta \) = central angle of the whole spiral
-\( a \) = deflection angle from the tangent at the T.S. or S.T. to any point on the spiral
-\( b \) = orientation angle from the tangent at any point on the spiral to the T.S. or S.T.
-\( \Lambda \) = total spiral deflection angle
-\( B \) = total orientation angle of the spiral
-\( X_o \) = coordinate of designated location of tangent offset \( o \)
-\( X, Y \) = coordinates of S.C. or C.S. from T.S. or S.T.
-\( k \) = increase in degree of curvature per 100-foot station along the spiral
-\( o \) = tangent offset distance from tangent to simple curve extended
-\( t \) = tangent distance from T.S. to S.C. or C.S. to S.T.
**Formulae for Spiral Elements**

\[ d = ks = k1/100; \quad D = kS = kL/100 \]

\[ \delta = \left(\frac{1}{2}\right)ks^2 = dl/200; \quad \Delta = \left(\frac{1}{2}\right)kS^2 = DL/200 \]

\[ a = (1/3)\delta = (1/6)ks^2; \quad A = (1/3)\Delta = (1/6)kS^2 \]

\[ b = (2/3)\delta; \quad B = (2/3)\Delta \]

\[ y = 0.582\delta - 0.00001264\delta^3 \]

\[ x = 1 - 0.003048\delta^2 \]

\[ o = 0.1454\Delta S \]

\[ X_o = \left(\frac{1}{2}\right)L - 0.000508\Delta^2 S \]

\[ X = 100S - 0.000762K^2S^5 \]

\[ Y = 0.291KS^3 - 0.00000158K^2S^5 \]

\[ t = 100S/2 - 0.000127k2S5 \]

\[ T_s = (R + o) \tan (I/2) + Xo \]

\[ E_s = (R + o) \csc (I/2) + o \]

**Staking Spirals by Deflections**

From \((o - X_o)\), \(T_s\), the T.S. and S.T. may be located from the PI of the curve shown above. Determining the \(E_s\) is useful in adjusting the degree \(D\) of the circular curve if it is desired to limit the throw of the center of the curve, or balance the throw of the existing track.

The entire spiral may then be run from the T.S. or S.T., after determining the deflection angle "a" from the tangent to any point on the spiral.
Deflection angles with the transit at any point on the spiral other than the T.S. may be determined from the principle that the spiral at the transit point deflects from a circular curve having the same degree as the spiral at that point at the same rate as it does from the tangent at the T.S. To continue the spiral from any intermediate transit point, the transit is backsighted on the T.S. with an angle set off equal to twice the deflection angle from the T.S. to the transit point. The transit will then read zero along the tangent to the spiral at that point. For any succeeding spiral point, the deflection angle for a circular curve, having the same degree as the spiral at the transit point and a length equal to the distance from the transit to the spiral point, is then calculated. To this, the deflection angle is added for the same length of spiral, but calculated, as it would be from the T.S.

To locate the spiral with the transit at the S.C. or C.S., the deflection angles, to set points on the spiral, are equal to the deflection angles for the corresponding points on the circular curve (extended), less the deflection angles of the spiral from the circular curve. The deflection angles of the spiral from the circular curve are the same as for the corresponding lengths of the spiral from the T.S.

In staking by deflection, it is sometimes convenient to divide the spiral into a number of equal chords. The first or initial deflection \( a_1 \) may be calculated for the first chord point. The deflections for the following chord points are \( a_1 \times \text{chord number squared} \). Examples of the method of staking spirals by the deflection method may be found in the Appendix.

**Staking Spirals by Offsets**

The spiral may be staked to the midpoint by right-angle offsets from the tangent and from there to the normal offsets from the circular curve (between the offset T.C. and the S.C.). The offset at midpoint \( 1/2 \ o \) and the other offsets vary as the cubes of the distances from the T.S. or the S.C. The method of staking a spiral by offsets is illustrated in the Appendix.

**Applying the Spiral to Compound Curves (AREMA 1965)**

In applying a spiral between two circular curves of a compound curve, the length of spiral is determined from the speed of operation and the difference in elevation of the two circular curves. The spiral offset “o” may be found from the formula given, using a value of D equal to the difference in the degrees of curvature of the two circular curves. The spiral extends for one-half its length on each side of the offset point of compound curvature. The spiral deflects from the inside of the flatter curve and from the outside of the sharper curve at the same rate as it would from the tangent. The spiral may be staked by deflection angles from either end. If the transit is located at the spiral point on the flatter curve, reading zero when sighting along the tangent to the circular curve, the deflection angles to set points on the spiral are equal to the deflection angles for corresponding points on the circular curve (extended), plus the
deflection angles of the spiral. If the transit is set at the spiral point on the sharper curve, the deflection angles are equal to the deflection angles for that circular curve (extended) minus the deflection angles for the spiral.

As an alternative, the spiral can be staked out by offsets from the two circular curves. The offset at the middle point of the spiral equals $1/2 \theta$, and the other offsets vary as the cubes of the distances from the ends of the spiral. Examples of applying a spiral to compound curves may be found in the Appendix.

### 6.3 Vertical Alignments

Vertical alignments are generally less complex than horizontal alignments. As such, it would seem that they are often overlooked during the early part of many design processes resulting in unnecessary re-design of horizontal alignments late in the design phase or settling for less than optimal vertical designs. The grades, which must be traversed by rail vehicles, are generally much more limiting than highway vehicles, due to both the limited amount of friction available at the interface of the steel wheel and the steel rail, as well as the substantially smaller power to weight ratio of rail vehicles. Vertical alignments are comprised of the same two components as horizontal alignments (tangents and curves), but with some differences in composition and terminology.

Vertical tangents, commonly referred to as grades, are straight lines effectively plotted in the Z-plane or vertically. These tangents are classified by the grade or incline. The grade is measured in the amount of rise or fall over a distance and is expressed in terms of percent. For example, a grade, which rises 1.5 feet in 100 feet traveled, is referred to as 1.5%. If the grade drops 1 foot over 200 feet, the grade is termed –0.5%. Note that the relative positive or negative is determined by the net gain or loss of elevation in the direction of increasing station. The concepts pertaining to two points defining a line, three points for establishing an existing tangent and two tangents meeting at a PI are identical in concept. Only the terminology is different.

Like horizontal tangents, vertical tangents are generally connected via curves. Unlike horizontal alignments, vertical curves are almost always parabolic in nature rather than circular.

Vertical curves are specified in length and denoted as the total grade change divided by the length of vertical curve. This ratio is denoted as ‘R’. This is effectively the inverse of ‘K’, which is employed by highway designers for which the values are length of curve per percent grade change.

The application of vertical curves through the specification of R is highly misunderstood. AREMA had long specified in the Manual for Railway Engineering
that acceptable values of $R$ should be 0.05 units for sags (valleys) and 0.10 for summits (hills) for main lines, and twice the preceding values for secondary and branch lines. These criteria actually date back some 140 years earlier and were apparently established around the Civil War by Wellington, a very respected engineer whose guidance on railway route design is still held in high regard today. Modern freight railways still use criteria based upon this guidance with the exception of yard and industry tracks where the values are much higher.

$$L = \text{Length of vertical curve in 100-ft stations}$$

$$R = \text{Rate of change of grade per station}$$

$$D = \text{Algebraic difference of rates of grade}$$

$$M = \text{Offset in elevation at B}$$

$$L = \frac{D}{R}$$

When vertical curve is concave downwards

$$M = \frac{(\text{Elev B} \times 2) - (\text{Elev A} + \text{Elev B})}{4}$$

When vertical curve is concave upwards

$$M = \frac{(\text{Elev A} + \text{Elev C}) - (\text{Elev B} \times 2)}{4}$$

The offset for any other point on a vertical curve is proportional to the square of its distance from A or C to B.

Offsets are - when the vertical curve is concave downwards and + when the vertical curve is concave upwards.

The criticism of this criteria is two-fold. First, the result is generally long vertical curves, which are disproportionate to others used in similar applications such as highways. The second is the opinion of many who believe that the establishment of vertical curve length should be partially based upon vehicular speed and thus vertical acceleration, alas again similar to highway design.

Recently, AREMA adopted new criteria similar to that being employed by light rail designers, and some other passenger rail companies have departed from the prior AREMA guidance. The new procedure solves for the length by:

$$L = \frac{D \times V^2 \times K}{A}$$

where: $A = \text{vertical acceleration in ft/sec}^2$
D = absolute value of the difference in rates of grades expressed as a decimal 
K = 2.15 conversion factor to give L in feet 
L = length of vertical curve in feet 
V = Speed of the train in miles per hour

AREMA recommends a value of 0.10 and 0.60 for freight and passenger operations respectively for both sag and summit curves. Specific railways or passenger rail agencies may use different values for A which should be established prior to design.

The new criteria will produce shorter vertical curves for most freight situations. The designer should be cautioned that where the older non-speed-based criteria has been applied, there are not any general restrictions for the locations of any single vertical curve relative to others or horizontal geometry. However, AREMA specifically recommends against placing vertical curves designed with the new criteria within the limits of horizontal spiral elements or within 100 feet of adjacent vertical curves.

Some passenger rail organizations incorporate a formula for the calculation of vertical curve lengths, which is similar to the highway definition using ‘K.’ An example of this formulation is as follows:

Crests  \[ L_{VC} = 250 \times (A) \]

Sags  \[ L_{VC} = 500 \times (A) \]

Where  \( L_{VC} = \) length of vertical curve in feet 
\( A = |(G_2 - G_1)| = \) algebraic difference in gradients connected by the vertical curve in percent 
\( G_1 = \) percent grade of approaching tangent 
\( G_2 = \) percent grade of departing tangent

Generally these values may or may not yield similar results to the new AREMA methodology.

In applying either vertical curve criteria, the designer can generate a calculation for the required vertical curve length to the decimals of a foot. In practice, the designer should round the calculated value up to at least the nearest ten feet (e.g., 537.51 ft becomes 540 ft). Likewise, some railways have limits on the minimum length of vertical curves. For example, the designer must be cognizant of 50 or 100 feet.

### 6.4 Alignment Design

In a perfect world, all railway alignments would be tangent and flat, thus providing for the most economical operations and the least amount of maintenance. Though this is
never the set of circumstances from which the designer will work, it is that ideal that he/she must be cognizant to optimize any design.

From the macro perspective, there has been for over 150 years, the classic railway location problem where a route between two points must be constructed. One option is to construct a shorter route with steep grades. The second option is to build a longer route with greater curvature along gentle sloping topography. The challenge is for the designer to choose the better route based upon overall construction, operational and maintenance criteria. Such an example is shown below.

Suffice it to say that in today’s environment, the designer must also add to the decision model environmental concerns, politics, land use issues, economies, long-term traffic levels and other economic criteria far beyond what has traditionally been considered. These added considerations are well beyond what is normally the designer’s task of alignment design, but they all affect it. The designer will have to work with these issues occasionally, dependent upon the size and scope of the project.

On a more discrete level, the designer must take the basic components of alignments, tangents, grades, horizontal and vertical curves, spirals and superelevation and construct an alignment, which is cost effective to construct, easy to maintain, efficient and safe to operate. There have been a number of guidelines, which have been developed over the past 175 years, which take the foregoing into account. The application of these guidelines will suffice for approximately 75% of most design situations. For the remaining situations, the designer must take into account how the
track is going to be used (train type, speed, frequency, length, etc.) and drawing upon experience and judgment, must make an educated decision. The decision must be in concurrence with that of the eventual owner or operator of the track as to how to produce the alignment with the release of at least one of the restraining guidelines.

Though AREMA has some general guidance for alignment design, each railway usually has its own design guidelines, which complement and expand the AREMA recommendations. Sometimes, a less restrictive guideline from another entity can be employed to solve the design problem. Other times, a specific project constraint can be changed to allow for the exception. Other times, it’s more complicated, and the designer must understand how a train is going to perform to be able to make an educated decision. The following are brief discussions of some of the concepts which must be considered when evaluating how the most common guidelines were established.

A freight train is most commonly comprised of power and cars. The power may be one or several locomotives located at the front of a train. The cars are then located in a line behind the power. Occasionally, additional power is placed at the rear, or even in the center of the train and may be operated remotely from the head-end. The train can be effectively visualized for this discussion as a chain lying on a table. We will assume for the sake of simplicity that the power is all at one end of the chain.

Trains, and in this example the chain, will always have longitudinal forces acting along their length as the train speeds up or down, as well as reacting to changes in grade and curvature. It is not unusual for a train to be in compression over part of its length (negative longitudinal force) and in tension (positive) on another portion. These forces are often termed ‘buff’ (negative) and ‘draft’ (positive) forces. Trains are most often connected together with couplers (Figure 6-10). The mechanical connections of most couplers in North America have several inches (up to six or eight in some cases) of play between pulling and pushing. This is termed slack.
If one considers that a long train of 100 cars may be 6000' long, and that each car might account for six inches of slack, it becomes mathematically possible for a locomotive and the front end of a train to move fifty feet before the rear end moves at all. As a result, the dynamic portion of the buff and draft forces can become quite large if the operation of the train, or more importantly to the designer, the geometry of the alignment contribute significantly to the longitudinal forces.

As the train moves or accelerates, the chain is pulled from one end. The force at any point in the chain (Figure 6-11) is simply the force being applied to the front end of the chain minus the frictional resistance of the chain sliding on the table from the head end to the point under consideration.

As the chain is pulled in a straight line, the remainder of the chain follows an identical path. However, as the chain is pulled around a corner, the middle portion of the chain wants to deviate from the initial path of the front-end. On a train, there are three things preventing this from occurring. First, the centrifugal force, as the rail car moves about the curve, tends to push the car away from the inside of the curve. When this fails, the wheel treads are both canted inward to encourage the vehicle to maintain the course of the track. The last resort is the action of the wheel flange striking the rail and guiding the wheel back on course.

Attempting to push the chain causes a different situation. A gentle nudge on a short chain will generally allow for some movement along a line. However, as more force is applied and the chain becomes longer, the chain wants to buckle in much the same way an overloaded, un-braced column would buckle (See Figure 6-12). The same theories that Euler applied to column buckling theory can be conceptually applied to a train under heavy buff forces. Again, the only resistance to the buckling force becomes the wheel/rail interface.
With this chain example, it becomes apparent that the greater number of curves which must be traversed by a single train, the more the train wants to deviate from its proposed route. It is thus important to conclude that one long curve is better than several smaller curves with collectively the same total deflection. The physical act of bending the train, straightening the train, bending the train and straightening the train exerts more force (i.e. wear and maintenance) on the track structure trying to guide the rail vehicle. If the rail structure happens to vary from a perfectly maintained condition and/or a car of similar maintenance condition happens to pass over the same point, the likelihood for a derailment is increased. It is also less comfortable for passengers.

This reduction in the individual number of curves should be applied to vertical curves as well. (See Figure 6-13) Draft and buff forces can vary greatly over the length of a train as a result of grades. As a train travels the length of an alignment, the forces produced by a given length of train on a given severity of grade is constantly changing. It is far easier for an engineer to compensate for long steady grades than to constantly have to adjust brakes and throttle positions to keep consistent speeds over a rolling terrain.

Though compound curves are not uncommon with railway alignments, reversing curves should be avoided at all costs. With reverse curves, there are two dynamic components acting on a single car or rail vehicle causing a yawing effect, which is of concern. The first uses the chain example. Each railway car represents one link in the chain. One end of the chain has lateral forces applied to it in one direction from the draft or buff forces in addition to the centrifugal forces. The other end of the car has similar forces applied, but in the opposite direction (See Figure 6-14). The net effect is a couple about the center of the car. This compares to a car on a single curve where the forces at either end of the car are acting in the same direction and thus counter-acting one another. This couple effect greatly increases the likelihood of the train buckling and thus a derailment.
Secondly, as the rail vehicle leaves the first curve, the guiding effect of the track is acting to counter the centrifugal force until such time that the first truck exits the curve. The rotational momentum about the vertical axis of the car will generally force the restraining effect from the outside rail to the inside rail immediately after exiting the curve. To have a second reversing curve will cause a sudden and abrupt force acting to change the rotation of the car the other direction. This sudden reversing of direction causes excessive horizontal forces across the rail at the wheel/rail interface, which can be a derailment hazard.

There is also a practical limit between the translations of coupler faces of adjoining cars. A railway car traversing one curve will have the coupler faces at the extreme end of the car translate to the outside of the first curve. This outside of the first curve translates to the inside of the second curve. The adjoining car translating the second curve will shift the coupler face to the outside of the second curve, which is opposite the first. There is a practical limit, based upon the individual car design of each car (which may not be the same for each car), which may result in the forcing of one of the two cars off the track if the curvature is too sharp.

To alleviate this yawing effect, all reversing curves should be separated by a tangent between the curves, though the exact length required will depend upon a number of factors. The AREMA Manual for Railway Engineering provides recommendations for yard tracks only, strongly recommending at least one car length worth of distance between reversing curves. However, there are some provisions for much smaller tangents, or even none between lesser curves in tight, light-use yard tracks. Railways themselves generally have their own criteria, most insisting on at least one car length regardless of the constraints. In the event that reversing curves cannot be avoided, there should be no superelevation applied to the track (0° cross-level) for at least one car length on either side of the point of reverse curvature (PRC).

Tangent length between reversing curves on lines outside of yards and terminals is generally much longer. For freight, each railway has its own requirements, generally being 150 to 300 feet depending upon track speed and conditions. This allows for the subtle instabilities of a railway car exiting a curve onto a tangent to stabilize before introducing forces to cause it to move in the opposite direction. For passenger traffic, the generally accepted criteria is a tangent in length representing two seconds of travel time (some agencies prefer three seconds). This criterion is generally based on
passenger comfort, and may be extended for locations where two seconds of travel would equate to less than a single car length.

Light rail limitations for reversing curves are similar to heavy rail, with the resulting constraints having the same root causes. Because the trains are shorter, and the track can be designed for specific rail vehicles, specific criteria for tangent lengths between curves can be more generally defined.

The Transportation Research Board (TRB) Track Design Handbook for Light Rail Transit recommends a desired tangent length between curves of 300 feet, with an absolute minimum of 100 feet. For lead tracks and industrial spurs, a minimum tangent distance of 50 feet should be provided between curve points. All turnouts should be located on tangents.

Maximum allowable curvature can be defined by several factors. For mainlines, the practical maximum train speed is generally limited by curvature. However, just because a proposed route may be planned for train speeds of only 30 mph, does not necessarily mean that all curves should be made as sharp as possible. Sharp curves result in more maintenance and more operating expense. The designer must weigh the operating conditions and physical conditions, such that both the amount of curvature and severity do not present undue maintenance costs or operating restriction.

Other considerations must be given due thought during alignment development. The use of large amounts of superelevation to allow for high speeds over moderate curves where trains may be frequently stopped will also have an adverse effect. Consider a main line with a timetable speed of 70 mph. However, there is a control point only 1,500 feet past a 1-degree, 45-minute curve. This curve placement causes two concerns. First, there could be sight distance issues where the absolute signal is not visible far enough in advance to be able to stop a train short of a red signal at timetable speed. Second, assuming the railway incorporates one-inch of underbalance, the curve would be superelevated five inches. This extreme crosslevel of the track is a significant derailment risk if trains frequently stop at the absolute signal.

The location of grade crossings or railway crossing diamonds may also limit train speed. A proposed grade crossing of a street with high superelevation may require significant or unacceptable modifications to the vertical profile of the road. Crossing diamonds frequently are restricted to trains speeds of 40 mph or less due to both safety and maintenance considerations.

For standard gage track, the cant of the rail and the conical profile of the wheel tread will generally guide the rail vehicle on curves up to three degrees before flange/rail contact begins to regularly occur (thus significant curve wear of rail head begins). Heavy haul North American freight railways frequently have curvature well in excess of three degrees, and may be as much as ten to twelve degrees or more. In these tight curvature situations, the physical obstacles to the alignment were apparently so costly to remove, that the significant increase in maintenance costs and reduction in operating
efficiency was accepted despite the application of high curvature. Generally speaking, most North American freight railways prefer new lines constructed for moderate to heavy use to incorporate curvature of six to seven-and-one-half degrees or less.

Curvature within yards, terminals and industry tracks is based more upon practical limits and maintenance considerations. For freight railways, most equipment can physically traverse curves of seventeen degrees or more, but there are frequently restrictions upon train make-up that cannot always be accounted for and such extreme curvatures are to be avoided.

The problem of extreme curvature can be manifested in one of three ways. There is a practical limit to how much the railway trucks under the car body can swivel before being restricted by physical features of car design (striker openings of the coupler housing that keep the couplers in line to facilitate coupling). Often times, the car may turn sharper, but critical components such as brake rigging are damaged even though the car is not physically derailed.

The second limit has to do with the relative position of coupler faces at the extreme ends of cars. As a car traverses a curve, the center of the car between the trucks translates to the inside of the curve, while the extreme ends of the car and the couplers translate to the outside of the curve. If the two cars coupled together are of the same design, the limiting factor can be if the inside corners of the ends of the cars may meet, or the coupler is twisted to the point of failure. A greater problem is a longer car coupled to a shorter car (Figure 6-15). The coupler face of the longer car translates farther off the centerline of track than the shorter one, physically pulling the shorter car off the track. The shorter car is usually the one that derails first due to the mechanical advantage of the longer distance from the end of the longer car to the truck center.

Figure 6-15 Long Car Coupled to a Short Car - Railroad Technical Manual - Courtesy of BNSF

Extreme curvature is also a problem resulting from the longitudinal forces in a train. The buff and draft forces acting through the coupler faces of the individual cars on curves will naturally result in a horizontal force component. As the curve radius decreases, the horizontal component of this force becomes larger. As curves become sharper and train forces become greater due to geometric, operational or train size
factors, the likelihood of a derailment, at least partially resulting from increased longitudinal forces, increases.

During the design process for yards, terminals or industrial tracks, the designer should first consider the guidelines provided by the serving railway. These should be followed closely for terminal and yard design, with design exceptions being identified early and the variances minor.

For industrial track design, the designer must weigh all the constraints and the serving railways track standards before exceeding any guidelines. In most cases, the serving railway will have published guidelines limiting curvature on industrial tracks from $9^\circ30'$ to $12^\circ30'$. Depending upon the situation, the servicing railway may or may not approve curvature beyond these limits, but there will be an ultimate limit that they will not serve despite the assertion that the railway equipment can physically traverse the proposed alignment. Any exceptions will have to be approved prior to service by the railway, so this approval should be granted prior to construction. It should be further noted, that in those facilities which handle unit trains or other long cuts of cars, the guidelines provided by the servicing railway are generally much more stringent.

Curvature limits for light rail traffic are much higher than for traditional heavy rail equipment. Like heavy rail, main line curvature is generally limited by a combination of superelevation practices and vehicular speed. However, on most light rail systems, all the vehicles are the same or very similar. Sometimes, rail equipment is specifically designed for the existing system’s track geometry. Because of this homogeneity of equipment on independent systems, curvature limitations can be approached more readily as the same limit applies to all equipment rather than a range of values found with the vastly different equipment handled by heavy rail systems.

The generally accepted minimum radius is 500 feet for general main routes. This can be reduced to as low as 115 feet for track embedded in pavement. Absolute minimums are established by the equipment used and could be as low as 82 feet or less.

Because severe gradients along an alignment can affect the ultimate speed, fuel usage, and power requirements, the gradients on new alignments are usually scrutinized more closely than horizontal alignments. Railway gradients are generally much less severe than roadways. Where a highway in mountainous areas may have grades of six or eight percent, a railway grade may only be 1.5% or up to around two percent. For main line route design, the concept of ruling grade must be defined. The ruling grade along an alignment is the grade whose curvature severity and length combined is the defining criteria for matching locomotive power to train tonnage. (See Chapter 2 – Industry Overview.) This grade may not be the steepest, or the longest. Shorter, steeper grades are termed momentum grades. The severity of these grades are short enough that the momentum of the train moving at track speed combined with the train’s maximum power is able to ascend the grade at an acceptable speed.
If the ruling grade becomes too severe, the railway may have what is termed a helper district. This is a section of the alignment where additional locomotives are added to the train (usually at the end, but occasionally at the front or in the middle) to assist the train up (and sometimes down) the grade. These grades are usually in excess of two percent and should be avoided due to the inefficiencies afforded the operation. In most cases, North American freight railways will limit all new main line grades to under one percent.

Grades for passenger equipment can generally be more severe because the equipment is lighter relative to the power available to overcome the grade. However, the reason that the power to weight ratio is less for passenger equipment is because the desired speed is higher than with freight. The relationship between horsepower, which provides speed, and tractive effort, which is needed for overcoming grades, is such that a slight increase in gradient can result in a substantial loss in speed above 50 mph. Steep grades on passenger routes should be limited to very short segments or momentum grades, or in areas where train speed is already restricted due to curvature or other constraints.

Light rail gradients are even more flexible, with main line grades of four percent commonly ascended without loss in velocity. Short grades may be as high as seven percent or more. However, the vehicle manufacturer and the light rail system criteria define the maximum allowable gradient on any particular system.

Many designers have been taught to attempt to balance grading work during construction. In practice, this has lead to railways constructed with undulating grades. If these grades become significantly different, train handling becomes very difficult because different portions of the train are constantly changing from a draft to buff and back to draft condition. With extreme grade fluctuations, the train can actually break in two from these undulations. Good design practice should allow for a single train to never be on more than one increasing and decreasing grade at one time.

The actual operation of the track needs to be considered when establishing grades. Control points on severe gradients are particularly problematic (Figure 6-16). A heavy train descending a grade will often have difficulty stopping for an unexpected stop signal. Likewise, ascending grades should be limited to 0.50% where heavy trains are to start from a standing stop.
Grades in yards and industrial tracks can occasionally be steeper than those found on main lines due to the limited speed and train length. Serving railways will provide what they consider to be their maximum gradients for industrial and yard tracks (typically the limits are between 1.5% and 3.0%, depending upon the carrier). Though it is occasionally exceeded, three percent seems to be a practical maximum for most freight terminals.

Design of railway grades must take into consideration what is happening with the corresponding horizontal alignment. Train braking systems function through the use of pressurized air. It is not the existence of pressurized air in the reservoir of a railway car’s braking system, but the differential pressure created by the release of air from the reservoir that actually produces the braking action. Without the hand brake set and without air in the braking system, rail cars are free to roll.

This situation lends itself well to the switching and sorting of cars, such that cars can be released (or kicked) and allowed to roll down particular tracks. At major classification facilities, gravity or hump yards are used for the classification of cars. Freight cars are sent over a hump and allowed to roll freely under the influence of gravity into predetermined tracks shaped like a bowl, where they come to rest or are stopped by cars already there. The rollability of cars allows for the classification of grades within
yards as decelerating less than 0.10%), rolling (between 0.10% and 0.25%), and accelerating (greater than 0.25%). See Figure 6-17.

These concepts become important for not only rail yards, but also industrial tracks. In spite of the fact that good operating practice dictates that cars left at an industry or siding should have at least one hand brake set to prevent movement, the number of cars which somehow wind up with the hand brake not set is of some concern. Tracks, which are to have cars at rest for any length of time, should be relatively flat (<0.10%). Yard tracks are also usually dished, such that the lowest point is on the yard tracks for holding cars. Where the gradient for cars is such that cars will be sitting on a grade or the siding is at an elevation greater than the lead track serving it, there must be some other form of mechanical means, such as a derail, to prevent cars from inadvertently rolling back into harms way.
Segment A – Switching Lead or Drill Track: The gradient should be relatively flat with 0.00% preferred.

Segment B & C – Ladder and Switch to the Clearance Point: Preferred gradients range from -.20% to -.30%.

Segment D – Clearance Point to Clearance Point: Preferred gradient is slightly descending from −0.10% to 0.00%.

Segment E, F, G and C – Leaving End of the Yard: Need 300 feet of 0.3% gradient to prevent rollouts.

Grades and grade changes can have a significant effect on horizontal geometries. If the grades over which the train is traveling are acting to complement the action of the train by reducing longitudinal forces, then this situation may allow for more extreme horizontal geometries. For example, consider two railway yards of similar construction consisting of a yard lead and several tracks for the classification of cars. Against the objections of the designer, the first yard was constructed on flat topography and was configured such that the drilling operation took place on a long curve of 10°. At roughly the same time, a yard of similar design was constructed ten miles away, and the main lead was also constructed on nearly a 10° curve. However, the topography was such that the lead and the first portion of the ladder track were constructed on a −0.50% grade to prevent pull-aparts.

At the first yard, the constant daily drill operation of pulling cars out of the yard and shoving cars back into the yard resulted in extreme rail wear on the 10° curve. The rail was transposed at 10 months and replaced at 18 months with head hardened rail. The operator hopes that a three-year life of the head-hardened rail might be realized. The second yard, with a slightly larger amount of traffic, shows no significant rail section loss.

The only significant difference between the two designs is the addition of a significant gradient on the lead. The cars on the flat lead must be shoved with much greater force, causing greater buff forces, which results in significant wear between the wheel flange and the outside rail. At the second yard, once the cars are in motion, they stay in motion not requiring the constant shoving and heavy draft forces. The lack of additional lateral forces has resulted in rail wear that is considered normal.

Until the advent of the new AREMA vertical curve length recommendations, there was no guidance, which specifically prevented the use of vertical curves in horizontal curves or spiral. They also did not and do not have any criteria preventing the use of reversing vertical curves. Nearly all freight railway companies are silent on this topic as well. It can only be assumed that the silence on these issues is the result of not having
any need here-to-for. The increasing belief that the established railway vertical curve criteria resulted in curves which were too long may be correct, because the curves produced are apparently gentle enough that the combination of the vertical acceleration from the vertical curve combined with the horizontal accelerations resulting from curves, and particularly spirals, has not yet produced a condition requiring industry address.

One of the most dangerous conditions can occur at the bottom of a long grade or sag, which is accompanied by a horizontal curve. As the train passes this point, the train has the slack bunched towards the front of the train as a result of braking down the hill. As the engineer applies power to ascend the next grade, the body of the train shifts from bunched to a stretched condition. As this occurs along the horizontal curve, the buff forces compound the centrifugal force, often resulting in derailment from train buckling.

6.5 Turnouts

A railway turnout (sometimes referred to as a switch) is simply a device that splits one track into two. Geometrically, a turnout is comprised of several key parts including a switch, closure curve and a frog (See Chapter 3 – Basic Track). Though there have been a number of turnout designs recently whose names describe the advanced geometry incorporated throughout the turnout (e.g., tangential and secant geometry), they all are comprised geometrically of these three basic components.

Regardless of design, basic turnouts can be classified as lateral, equilateral or curved. The lateral turnout (Figure 6-19) is by far the most common and is generally preferred for most installations. It consists of a divergent track being split from a tangent piece of track. An equilateral turnout comprises two curves of equal radii diverting in opposite directions from a single tangent. In some situations, the divergent
angles of the two routes through the turnout may not be equal. This particular case is termed a split-angle turnout. A turnout comprising a curve diverting from an existing curve is known as a curved turnout.

A switch is a pair of rails (points), which pivot about the heel. When one or the other point rails is placed against the stock rails, the train will take one route over another. Some switch points are curved, but most are tangent. The location where the switch point meets the stock rails is known as the point of switch. The angle at this point is the switch angle. For straight points, this angle is fixed along the length of the points and is defined by the heel block distance, switch point thickness and switch length.

![Figure 6-20 Switch Angle by Definition](image)

For curved points, the point of switch (PS) represents the PC of a curve. With the exception of some advanced designs recently developed, there remains a slight angle deflected between the stock rail and the PC or PS of the curved point.

The actual point of switch does not taper to a knifepoint without a measurable thickness. Switch points of AREMA designs have a thickness of 1/4 inch, a machined thickness of 1/8 inch, or are housed. The later switch design, commonly called ‘Samson points’ uses a switch point milled to 1/16-inch thick, which fits into a specially milled stock rail. Each of these designs has a theoretical point located in advance of the point of switch, which represents the point at which the switch point gauge line projected would meet the gauge line of the stock rail. This theoretical point is of little concern in most design considerations, but is occasionally referred to.

The frog is a track component that allows for the flange of a train wheel to cross over another rail. Frogs are defined by a number, which represents the spread of the two sides relative to length (Figure 6-21).

Though this frog number defines the turnout size, it is the frog angle, which is used by designers in the establishment of divergent alignments. A common error for designers is to assume that the divergent route of lateral turnouts diverts from the main
alignment (the straight side) at a rate equal to the frog number. This is incorrect as this number is based upon the axial distance with equal divergence on either side. This is different than the relative divergence between the two sides.

Like switches, theoretically the two routes through a frog can be extended to a knifepoint. Such a thin point would have a relatively short life due to the limited strength of the resultant thin piece of metal. Frog points are usually at least ½-inch thick (some railways specify 5/8-inch) at the tip of the actual point. This ½-inch point, known as the ‘actual point,’ is set back from the theoretical point a distance equal to the length required to establish the ½-inch point from the theoretical point.

Projecting the centerline of the routes through each side of the frog to a point of intersection is the PI of the turnout. This point, though not especially important in the field construction of the turnout, is of primary importance to the designer as it defines the original deflection point of the divergent route from the primary route.

The length between the point of switch and the actual point of frog is termed the actual lead (lead length). Occasionally, the theoretical lead is designated; the difference being the theoretical point of frog replaces the actual point. This lead, though fixed through standards designs, can be extended or shortened to a limited degree to accommodate convenient placement of joints, etc. (See Appendix B)

Connecting the alignment representing the divergent switch point and the divergent movement through the frog is the closure curve. This curve generally starts at or just past the heel blocks and ends just prior to the toe joint of the frog. The actual radius of the closure curve can be changed outright or by adjusting the lead length.

Though turnouts are designated by their respective frog numbers (e.g., install a No. 10 right-hand turnout), there are any number of different specific designs for any specific turnout size and weight of rail. The two primary points, which are universally the same amount for any number of turnouts with the same frog designation, are the PF and the PI. The remaining geometric differences are primarily related to the incorporation of curved or straight points, point designs, frog joint locations and minor differences to lead lengths and closure curve lengths. Yet with the exception of the PS, which is defined by the actual lead length, these minor variations between different turnout
designs are typically of little concern to overall alignment design as they represent fixed properties isolated to a relatively small area at the beginning or end of a specific alignment.

The true geometries associated within the length of a divergent path of a turnout are complicated enough that several methods for their incorporation into alignment design have been developed to aid the designer. The first method, occasionally known as the ‘PI method,’ requires the designer to locate the PI of the turnout, thus defining the PS. The alignment chaining begins at the PS and follows the through portion of the turnout to the PI, then deflects the frog angle to the divergent alignment.

The second method is the ‘equivalent curve’ and starts with the calculation of the equivalent curve of the turnout. The equivalent curve of a turnout is simply a curve defined with the deflection (I) equal to the frog angle and the tangent distance between the PI and the actual point of frog. The equation is then defined as:

\[ R_{\text{equivalent}} = 2GN^2 \]

Where \( G \) = Gauge & \( N \) = Frog Number

This equivalent curve is then usually rounded up or down to the nearest even 15 minutes. The equivalent curve will remain constant for all turnout designs of a fixed frog number and gauge, unless the gauge changes, then the tangent distance also changes. Thus, for those designers working with non-standard gauge designs, the equivalent curves for such turnouts will also change. With this equivalent curve, the turnout is placed in the proposed alignment with the beginning/end at the PC/PT of the equivalent curve.

Though preferred by some railways, this method results in some peculiarities, which must be understood by both designer and constructor. First, the PC of the curve, and thus the beginning of the alignment, will almost always start some distance (as much as 25 feet) before the actual PS. The stationing of the alignment is usually adjusted such that the PS will fall at an even stationing (e.g., the PC to PS distance of a turnout is 9.44 feet, the stationing for the PC might be –0+09.44 to establish the PS point at 0+00). This distance between the PS and the PC is a function of the lead length and will vary slightly between turnout designs. Therefore, it is critical for both the designer and the constructor to know specifically which turnout design is to be used because it effectively makes the PI the defining point of the turnout location.

It must be noted from a pure geometric perspective, both methods represent approximations of the true geometry of a turnout. This results in minor differences in stationing through turnouts when both methods are compared. Eastern railways generally dictate the use of the PI method. Many western railways have incorporated the equivalent curve method. Some designers, through the use of computers, have begun to incorporate the actual geometry of turnouts into their alignment designs. Regardless of the method used, it should be noted before construction to avoid confusion.
The most defining point of any turnout location is the PF and the frog designation (i.e., the number). This defines the PI and PS location. A given population of identical turnouts installed in track of identical design will have some variation in lead lengths due to construction tolerances, but the frog defines the turnout and the two diverging alignments. Though construction differences in turnout installations are rarely noticeable to the experienced eye, even minor errors in frog locations are easily noticed.

It is important that the designer provide both the PS and PF location for construction, particularly if the proposed turnout construction method is not known at the time of design. The location of the PI might also be given, as it generally helps in the initial field staking of alignments. Turnouts are installed in one of three ways. The oldest and still most prevalent method for industrial work is to construct the turnout in place. The PF location is of primary importance as the frog will be installed first. All other components are generally installed by measurements off the frog.

The second method, panelization, is commonly used on main lines. Turnouts are actually constructed at a remote location being either the original manufacturer or a panelization plant. Turnouts are then shipped in two to four large pieces to the point of construction and ‘cut-in’ by removing a section of track and sliding the new turnout in from one side or off the end of a specially designed car. The third method is a hybridization of the other two, where a turnout is constructed adjacent to the track where it will be installed and cut-in in the same fashion as a panelized turnout. These latter two methods are used because the required window for turnout installation, which shuts down train operations, is significantly reduced. There can also be some labor savings by taking advantage of natural production efficiencies associated with the volume production of finished products. The primary point of reference in panelized turnout construction is the PS. The PS is used over the PF, as it is easier to distinguish and it is close to one end of an entire section of track to be replaced, rather than the PF, which is closer to the center.

The specific turnout to be selected depends upon several factors including design speed, usage and the practices of the servicing railway. Turnout speed is defined by one of two factors: the points and the closure curve. The maximum speed of the closure curve can be easily calculated using the limiting underbalance for the railway. The limiting point speed is calculated in the same fashion, but using the switch angle as the angle of deflection “I” and the length of the points as the tangent distance T to calculate the radius of the curve. For curved points, a third factor to consider is the limiting speed of the point radius based upon underbalance. The most restrictive of these calculations generally represents the limiting speed of the turnout. The tables below are from the AREMA Manual for Railway Engineering and are based upon AREMA designs and 3 inches of underbalance.
### Turnouts with Straight Switch Points

(AREMA)

<table>
<thead>
<tr>
<th>Turnout Number</th>
<th>Length of Switch Points</th>
<th>Speed in Miles Per Hour</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Lateral Turnouts</td>
</tr>
<tr>
<td>5</td>
<td>11'-0&quot;</td>
<td>12</td>
</tr>
<tr>
<td>6</td>
<td>11'-0&quot;</td>
<td>13</td>
</tr>
<tr>
<td>7</td>
<td>16'-6&quot;</td>
<td>17</td>
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<tr>
<td>8</td>
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<tr>
<td>16</td>
<td>30'-0&quot;</td>
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<tr>
<td>18</td>
<td>30'-0&quot;</td>
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<tr>
<td>20</td>
<td>30'-0&quot;</td>
<td>36</td>
</tr>
</tbody>
</table>

### Turnouts with Curved Switch Points

(AREMA)

<table>
<thead>
<tr>
<th>Turnout Number</th>
<th>Length of Switch Points</th>
<th>Speed in Miles Per Hour</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
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<td>Lateral Turnouts</td>
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<tr>
<td>5</td>
<td>13'-0&quot;</td>
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<td>13'-0&quot;</td>
<td>15</td>
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<td>44</td>
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<tr>
<td>20</td>
<td>39'-0&quot;</td>
<td>50</td>
</tr>
</tbody>
</table>

There are exceptions to this. First, the limiting speed through the tangent portion of a turnout is effectively unrestricted. There are upper speed limits for specialty turnout components, which may include spring switches and special frogs. Self-guarded frogs are usually limited to 15-mph regardless of straight or divergent moves. Additionally, the railway may place speed restrictions on certain turnouts due to maintenance considerations.

Though turnouts are generally available from No. 6 to No. 24 and more, most railways have limited the general use of turnouts to four to six designs for ease of standardization and part supply. Mathematically, as the numbers of turnouts increases, the relative difference in angles from one number to the next decreases (See Figure 6-22). Historically, turnouts under the number ten were available in ½ increments. The use of ½-sized turnouts is not generally done any longer as these turnouts were generally dropped from major railway standards 20 years ago. Turnout increments
above No.16 change from 1 to 2 as the angle differences become smaller, and drop to four to six above the No. 20.

Turnouts No. 10 and less are generally restricted for yard use, and few railways will allow sizes under 8 or 9 except for special situations. Turnouts for industries and sidings from main tracks are generally restricted to no less than 10 to 12 (one is now requesting 14’s). This restriction is generally a maintenance concern. The wheel traveling through a frog must transfer from one rail across the flangeway provided from the second route to cross and onto the point or second rail. The desired transfer results in the weight of the wheel being distributed between both rails for a short distance. As the frog angle increases, the effective distance of this load transfer is reduced resulting in higher impact loads and thus, maintenance.

Turnout sizes 14-16 are used for medium-speed divergent moves and some heavy traffic unit train facilities. Turnout Numbers 20 and 24 are used for main line moves and crossovers, allowing for higher speeds through the divergent leads. Some railways are using No. 30’s and higher. These are used only in special situations.

Turnout placement must take several factors into consideration. Generally speaking, conventional turnouts should be used if possible. They usually require less maintenance for all situations except where the movement of traffic through both routes is roughly equal. Turnouts should not be placed in vertical curves due to maintenance and operational concerns. Equilateral turnouts should only be used in yard situations and at the end of double-track territory. Turnout location within existing horizontal curves should be staunchly avoided when possible, due to maintenance concerns. Furthermore, curved turnouts generally cannot be panelized.

Turnout speed through the divergent lead will be restricted to the limiting speed for the turnout. The predominant movement through the turnout (e.g., the main line) should move through the straight side with the less-travelled route representing the divergent movement. On moderate grades, where one line splits into two or more, the up-hill move should be the straight side of the turnout as the tractive force of the locomotives, (produced from ascending the grade) generally creates a greater concentration of longitudinal forces than braking (descending the grade).

The location of turnouts relative to curves should be considered. The divergent lead of a turnout is effectively a curve (consider the equivalent curve concept). Thus, turnout location must consider curve limitations, such as reversing movements.
For example, a facing-point left-hand turnout immediately past a right hand curve (Figure 6-23) will result in a reversing curve movement. Though the derailment risk from reversing curves could be avoided if the turnout was changed from a left-hand to a right-hand, the maintenance concern remains. Cars are still in the process of attempting to assume the straight line of the tangent, resulting in higher maintenance.

Turnout locations relative to each other must be considered. It is possible to locate the PS of trailing point turnout under three feet from the PS of a facing point turnout. However, if the diverging movement of one turnout represents a reversing move through the second, it should be avoided if there are to be train movements through both divergent leads. This situation is also a problem in signalised territory where turnouts must incorporate insulated joints. Turnout placement in signalised main lines may be restricted when attempting to place turnouts in and around control points or absolute signals.

A track, which has several turnouts one after another for a series of parallel tracks or a yard, is referred to as a ladder. Ladder tracks should generally be straight with the diverging parallel tracks maintaining a bearing equal to the frog angle of the turnouts used in the ladder. Where this is not possible, curves are used, but this practice should be avoided. The maximum ladder angle is defined by:

\[ \phi = \sin^{-1} \left( \frac{S}{L} \right) \]

Where \( \phi \) = the deflection angle between the ladder and yard tracks

\( S \) = yard track spacing

\( L \) = length of turnout from original joint to heel of frog

There is a short curve just beyond the heel of frog with a \( \Delta = \phi - F \), where \( F \) equals the frog angle. The length of the curve is determined by the degree of curve desired for use. Be aware that the switch lengths (L), and thus the maximum ladder angles, vary widely among the different types and weights of turnouts. Care should be taken to design for the actual turnouts that are to be used in the field. The designer must be

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1 Railroad Track Design Manual – Prepared by Parsons Transportation Group
aware that along a ladder track with this maximum angle arrangement, adjacent yard tracks must share some switch ties and some switch ties will be eliminated. Also, the first yard track will have to diverge from the second yard track, as there is not enough room for the first track turnout at the beginning of the ladder track.

Where the required ladder angle is especially steep due to space constraints, tandem leads are often used. (Figure 6-24) A tandem lead is a turnout with a second turnout immediately past the first. This easily allows for the doubling of ladder angles. However, the head blocks for the inside turnouts should be extended under the ladder tracks with longer ties and switch rods to avoid the dangerous practice of switchmen moving back and forth across the ladder during switching operations.

Oftentimes, a track or a ladder will end by curving to make the final parallel track. The desired practice in this situation is to incorporate the equivalent curve to accomplish this. By placing an equivalent curve at the end of a ladder, the addition of future tracks is easily afforded by replacing the curve with a turnout at a future date. This is also done at the end of main tracks such as the end of a double track section, where one track ends with a single lateral turnout. (Figure 6-25)

When placing a turnout requiring additional curvature beyond the turnout, such as an industrial lead, the curve should not start until after the last long tie of the turnout. Track today is brought to its final alignment through use of mechanized surfacing and lining equipment, which physically moves the track in the ballast section and tamps them into place. This becomes difficult through turnouts, as the adjustment of one alignment will have a direct effect on the second (the through movement is used to establish the alignment with the divergent alignment being established through the correct use of offsets during construction). To start a curve immediately off, the heel of frog requires the curve to be established during turnout construction by physically spiking the new curve into the long ties (so-called spike-lining). This is not an easy practice to do, if at all, for panelized turnouts, and generally leaves a line swing as the tamper attempts to reconcile the curve through the long ties, which it cannot effectively change with the rest of the curve beyond the long ties.
6.6 Design of Yards

In spite of the possible alternatives to building yards, they are still required in many instances to originate, terminate and store freight cars. In order to reduce the delays generated, yards should be carefully located and well designed so as to be as efficient as possible for the purposes intended.

Freight car yards vary greatly in size and purpose. One of the smallest types can be a storage yard. These may be used for the holding of empty cars awaiting eventual use by a shipper. They can also hold loaded cars awaiting shipping orders that are often dependent on the season of the year or market conditions. Some industries may lease tracks in such a yard, or even own it entirely, to avoid demurrage (detention) payments. If individual cars must be selected (or even have their contents sampled) from a large number of similar cars, then a road between every pair of two tracks for mechanized inspections and inventoring can be very useful. Track centers of 24 ft. or less should suffice for such a road.

Some small yards called local yards are often constructed in industrial and warehouse areas, where cars are brought in and sorted for spotting to receivers. Cars are also gathered here for shipping and/or storage. Empty cars may also be held awaiting the needs of shippers. Tracks in such a yard are usually parallel to a main or drill track, and may even lie on both sides of that track.

Some industries do their own in-plant switching with their own locomotives or car-movers. In these cases, some kind of "interchange track" is built to facilitate the exchange of cars between the railway and the industry.

The next larger yards are designated as flat switching classification yards. These may be all in one body, where trains are received, cars are classified (sorted by destination) and trains departed, all on parallel tracks. Flat switching yards, located in a large terminal, may consist of three main parts, which can be constructed end-to-end or parallel:

1. Receiving yard where arriving trains are taken in,
2. Classification yard where individual cars are sorted by destination, and
3. Departure yard where newly made-up trains leave for other destinations.

Such classification yards are sometimes used for geometric switching, where numerous re-sorting of cars is accomplished. Each re-sort adds different classifications (destinations) of cars equal to the number of classification tracks. The largest of the yards are designated as gravity switching yards. Here, cuts (strings) of cars from trains and switching assignments that have arrived in a receiving yard, are moved over a rise.

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2 Railroad Track Design Manual – Prepared by Parsons Transportation Group
called a hump. They may pass over a weigh-in-motion scale and then are allowed to drift by gravity into one of several classification tracks in a depression called the bowl.

On the downgrade into the bowl, the speeds of the cars being classified are arrested by retarders, which may be pneumatically, hydraulically or electrically operated. The retarders are activated in response to the velocity of a car to keep it separated from other descending cars and to maintain the desired speed into the bowl. Coupling with cars already standing in the bowl should not occur at more than 3 – 4 mph. The switches entering (and sometimes leaving) the bowl are remote and often computer controlled to lead cars into the proper classification tracks. In order to minimize the distances from the hump to the initial clearance point on each track, switches are often lapped (superimposed) over each other.

Cars are trimmed from the bowl classification tracks, grouped by destination and taken to the departure yard from which trains and switching assignments leave. Care must be taken not to hump cars into a classification track while it is being trimmed on its other end. The design of such yards is very complex and depends upon such varied things as grades, curvature, car weights, rolling resistance, temperature and wind speeds. Therefore, the aid of computer programs and retarder manufacturer’s input is strongly recommended. (Prior to automation, riders, who manually applied the hand brakes on cars to control their speeds, controlled the speed of gravity switched cars. This arrangement is now quite rare.)

All yards should be designed to easily capture bad order cars and shuttle them to a (rip) track where repairs can be made. Also, yards of any size may have many ancillary facilities, such as:

- Buildings for administration
- Lockers and toilets
- Equipment and material storage
- Fueling and servicing
- Minor repair facilities for locomotives
- Turntables or wye tracks for turning equipment
- Air compressors
- Floodlighting
- Waste water and oil recovery and treatment facilities
- Various roadways

Good drainage is a must for a yard site; otherwise one can expect to have the yard out of service during and after any storms involving significant precipitation.

The receiving, classification and departure tracks of most yards consist of several parallel or nearly parallel tracks. If yard tracks are constructed at the minimum allowable center-to-center distance (14 ft. in some states), then there is no margin for future shifting of tracks under use. Obviously, impairment will be created. Also, if
there are curves in the body of a yard, wider track centers may be required. Therefore, it is best to add a foot or more to yard track center distances if the available room allows it.

Yard tracks are connected at one end in the case of a stub-ended yard, or on both ends in the case of a double-ended yard to the ladder tracks. Turnouts to each yard track emanate from these ladder tracks and their switch stands or machines are along the ladder track on the side away from the yard tracks. If the ladder tracks at both ends of a double-ended yard are constructed parallel to each other, then almost all of the tracks in that yard will be of the same length. If the ladder tracks are constructed to converge, then the yard tracks are progressively shorter the farther they are from the track where the yard originates.

If there are more than ten tracks in a switching yard and it is likely that more than one switch engine per any shift will be assigned to the yard, then a double ladder on one or both ends should be considered. The yard tracks should be divided equally between the two ladders, but the inside ladder may also connect with the first yard track connected to the outside ladder. With two ladder tracks at an end of a yard, each of two locomotives can be simultaneously switching the tracks attached to each ladder. Means must be provided to prevent locomotives from occupying the same yard track from opposite ends when switching. This can be done with switch lists, radio or speaker communication, signals or switch locks.

If at all possible, tail room should be provided on a track clear of any main tracks for the back and forth shunting movements of locomotives and cars involved in switching at a yard. Ideally these tail tracks should be at least as long as the longest track to be switched in the yard, plus room for two locomotives (200 ft. +/-). During periods of little or no switching activity, such tail tracks can also be utilized to receive and depart trains.

Passenger car storage yards should be located as close as possible to passenger terminals to avoid excessive movements of empty equipment. Sometimes, however, that is not possible in crowded urban settings. Occasionally, passenger terminal tracks may double as storage and light repair tracks during off peak periods.

Car cleaning and washing are often done on storage tracks and a mechanical car washer may be part of the layout. Compressed air, water and electrical outlets are needed, along with adequate lighting for night work. Some stored passenger cars may require standby electrical outlets to operate lights, heating and cooling equipment.

It is useful to have some storage tracks spaced widely enough (20+ ft.) to permit light mechanical and electrical work on cars, including brake work and changing of wheel sets. Jacking pads should be provided along such tracks in order to raise cars off of their trucks using portable electric or pneumatic jacks. Tracks must be locked out of service when work is being performed on cars stored on them. As in freight yards,
buildings will have to be provided for office, locker rooms, and material and equipment storage.

The design and placement of yards have marked effects on the efficiency of an entire railway plant. They are also very expensive to construct and maintain. Often, by prior agreements entered into between railways and operating unions, only certain unions may be allowed to operate certain yards. This can be an important consideration when field siting and planning a yard. Very careful planning and investigations are strongly warranted in the construction of new yards and the reconstruction or elimination of old yards.

### 6.7 Clearances

In order for trains to move safely and efficiently over a track, an envelope of clear space must be provided. The size of this envelope is determined by:

1. The sizes and types of locomotives to be operated. (Electric locomotives drawing power from overhead wires require significantly greater overhead clearances.)
2. The sizes and types of cars to be operated. (Rail transit vehicles with sealed windows can have greatly reduced side clearances.)
3. The dimensions of large loads to be handled (such as double-stacked containers).

Clearances, or the space which is required between the track and other fixed obstructions including other tracks, though defined by the actual size of railway equipment in conjunction with track geometry, are further mandated by both state statute and railway policy. The Association of American Railroads (AAR) has established ‘plates’ or envelopes within which all railway equipment must remain for free interchange amongst North American Railways (Plate C). (Figure 6-26) There are additional plates established for cars that are in excess of these dimensions, which include Plates E & F.

Within Chapter 28 of the AREMA Manual for Railway Engineering, resides the recommended procedure for calculating the specific clearance of fixed objects for cars of different AAR plates and other cars. These procedures should not be used for design purposes unless the facility is designed for the frequent use of non-standard freight or passenger equipment. The more prevalent application of the AREMA procedures is for the checking of the routing of specialty cars for unusually large loads. Railways generally prepare and maintain clearance charts for each section of the railway, showing the maximum dimensions of rail equipment and loads that can pass over that line.
On an existing railway line that anticipates hauling oversize loads, it is absolutely necessary to know from field measurements all of the locations where the least clearances exist. Such a listing must be kept up-to-date to reflect recent changes and should cover such things as overhead wire crossings and other non-railway improvements. Sometimes, when moving extremely large loads by rail, it may be necessary to move an impeding object to allow passage.

Each railway or agency maintains minimum clearances for the location of fixed objects above, below and beside railway tracks. Sometimes these minimum clearances will vary, based upon the level of service or type of track or the specific type of object to be cleared. The acceptable clearance for some facilities, such as passenger platforms, is generally less than others, such as buildings, due to the function of the fixed object.

States also maintain minimum clearance criteria for railway tracks. Where the state criteria are more conservative or comprehensive than the railway’s policy, the state laws generally supercede the railway’s rules. AREMA maintains a fairly comprehensive table of state railway clearance limits in Chapter 28 of the Manual for Railway Engineering. Individual railway clearance standards can be obtained directly from the railways.

In general, minimum clearance standards have been increasing with the introduction of larger track equipment, longer trains, and safer work practices. For most railways having tracks originally constructed over 100-years ago, the cost to increase the clearances to current practice is cost prohibitive. Generally speaking, unless an existing clearance condition represents a specific safety risk, existing tracks with clearances, which do not meet current standards or legal requirements, remain effectively grandfathered. In the event that the designer is contemplating a project that would substantially upgrade a facility or route, he or she might want to address sub-standard clearance issues if they are not already in the initial program.

Side clearances are usually measured from the centerline of tracks. Overhead clearances are measured above top of the (running) rail (ATR). Nothing in the track cross-section can protrude above the top of rail elevation. Track spacing is measured between centerlines of adjacent tracks.

On tangent track, the minimum side clearance from the centerline of track to a fixed object is generally held constant based upon the type of track and the object to be passed. When on curved tracks, additional side clearance must be added due to the end swing of the car overhang beyond the center of the trucks, and the middle ordinate between the truck centers of locomotives and cars. When curved tracks have superelevation, additional side clearance must also be added to provide room for the cant of the rail-mounted equipment.

Railway practice varies slightly from one company to the next as to the specific increased clearance required for curvature and superelevation and generally considers operating practice, underbalance and train speed. Likewise, some state statutes for minimum clearances also address this issue. The designers should verify with the
railway or agency before employing a specific methodology for increasing clearances on curves.

The maximum swingout (Figure 6-27) for most railcars occurs at the mid-point between the two trucks. It is the lateral shift of the railcar to the inside of the curve between the two trucks. The value of the swingout is equal to the midordinate "m" of
the curve subtended by a chord equivalent to the truck centers of the railcar and is calculated as follows:

\[ m = R - (2R - (t/2)^2)^{1/2} \]

The overhang distance “s” of the railcar from the centerline of the track is the sum of the swingout and half of the railcar width “w” as follows:

\[ s = m + w/2 \]

For railcars with extra long overhang beyond the trucks, the swingout at ends of cars must also be checked.

Tilting of the railcar to the inside of a super-elevated curve introduces further clearance requirements at the top edge of the vehicle. This additional clearance requirement “e” due to super-elevation “E” for a vehicle height of “t” can be calculated as follows:

\[ e = E \times t / G \]

Where: \( G \) = the gauge distance of the track

The total additional clearance requirement is the sum of the swingout “m” and the tilt “e” due to superelevation. These values are added to the normal clearance requirement for tangent track.

![Diagram showing rail car swingout](Figure 6-27 Rail Car Swingout – Courtesy of UMA)

Track centers are established to provide adequate room between parallel tracks for movement of equipment and the safety of train crews riding on the sides of cars, taking
into account any curvature, superelevation and swaying caused by rough track. Standard for track centers are maintained by railways and are subject to state statutes.

When considering side clearances, two areas must commonly be addressed. First, despite the need for only nine or ten feet from centerline to a fixed object for a train to clear, such objects may represent a significant obstacle in the maintenance of the track. Likewise, access may be required for other reasons, such as fire protection that may not be directly related to the operation or maintenance of the railway.

The other primary horizontal clearance issue that is a common concern, is the ‘clearance point,’ or that point on two connecting tracks passed which the progression of a train on one track would foul or impede a second train moving through the other route. The specific clearance between the two centerlines again varies from one railway to the next, but is generally between 13 and 15 feet.

In practice, it would be convenient to be able to locate the actual clearance point coordinates on the turnout side on an auxiliary track for a given frog angle and required clearance distance utilized. The following diagram (Figure 6-28) provides the solution to this problem.

![Figure 6-28 Calculating Coordinates for the Clearance Point - Miodrag Budisa](image)

\[ SK = d = X_m \]
\[ SM = m \]
\[ KM = \frac{p}{2} = Y_m \]
\[ p_{\text{min}} = 10Ft - 8In. (AREMA) \]
\[ SM = m = \frac{p}{\frac{\alpha}{2}} = \frac{p}{2 \cdot \sin \frac{\alpha}{2}} \]
Finally, there remain a number of other fixed obstructions, which the designer must be cognisant of during the design process, which may limit the location of turnouts. These include the location of trackside obstructions, which may require additional clearance for railway equipment moving through the curved side of a turnout. Most railways prefer to keep turnouts a fixed distance from the edge of bridges due to track dynamic concerns. In some cases, there may not be room for the placement of the switch stand on one side or the other due to the relative location of other tracks.

Adequate overhead clearances above the rails must be provided. In the past, when trainmen were required to walk and stand on the tops of cars, the required overhead clearances were based on their presence. Under overhead bridges and through tunnels, where such high clearances were not practical, trainmen were warned of approaching limited clearances by an overhead rope fingers called a telltale. In most places, the clearances based upon men riding on top of cars no longer apply.

Though typical freight railway equipment in operation today stands slightly less than 22 feet above the top of rail, minimum vertical clearances for fixed objects are generally established above this threshold to provide for future track maintenance, track alignment variations and future equipment dimension changes.

State statutory clearances usually only apply to so-called "common carrier" railways and are not enforced on military bases or within manufacturing plants, where an industry does its own car switching. This may result in close clearances in such facilities, which are expensive to correct if a common carrier is to take over the operation therein.
References:


