Friction Hoists

The friction (or Koepe) hoist is a machine where one or more ropes pass over the drum from one conveyance to another or from a conveyance to a counterweight.

In either case, separate tail ropes are looped in the shaft and connected to the bottom of each conveyance or counterweight.

The use of tail ropes lessens the out-of-balance load and hence the peak horsepower required of the hoist drive.

When compared with a drum hoist for the same service, the tail ropes reduce the required motor HP rating by about 30%, but the power consumption remains virtually the same.

Tail ropes have been used for a few doubledrum hoist installations to the same effect, but this practice has not gained acceptance by the mining industry.

Because they normally use several hoisting ropes, the largest friction hoists can handle heavier payloads than the largest drum hoists. The drum hoists are normally limited to the capacity of a single rope.

By statute, friction hoists usually require a higher SF on the hoist (head) ropes and most experts consider them impractical for very deep shafts.

For historical reasons, friction hoists (unlike drum hoists) are usually thought of in terms of metric rather than imperial (British) units.

To describe the size of a friction hoist people will say “a 3m wheel diameter” rather than “a 10-foot hoist.” For this reason, the explanations and design calculations that follow are mainly performed in metric units of measure.

A friction hoist with two skips in balance is normally suitable for hoisting from only one loading pocket horizon and for a hoisting distance exceeding 600m (2,000 feet).

Otherwise, a counter-balanced friction hoist (conveyance and counterweight) is usually employed (for multi-level, shallow lifts, or cage hoisting).
A friction hoist with two skips in balance may be suitable for a hoisting distance as shallow as 400m (1,300 feet).

The practical operating depth limit for a friction hoist is 1,700m (5,600 feet) for balanced hoisting and 2,000m (6,600 feet) for counterweight hoisting.

Beyond these depths, rope life may be an expensive problem.

For a tower-mounted skip hoist, the calculated static tension ratio \( T_1/T_2 \) should not exceed 1:1.42, but 1:1.40 is preferable.

For a ground mounted skip hoist, the calculated static tension ratio should not exceed 1:1.44 but 1:1.42 is preferable.

For a cage hoist installation, these values may be exceeded for occasional heavy payloads of material or equipment transported at reduced speed.

Tread pressure should not exceed 17.5 kg/cm\(^2\) (250 psi) for stranded ropes and 28 kg/cm\(^2\) (400 psi) for locked coil ropes.

For stranded hoist ropes, the tread pressure calculated for skip hoists should not exceed 1,700 kPa (250 psi) or 2,000 kPa (275 psi) for a cage hoist when considering occasional heavy payloads of material or equipment.

The total number of friction hoist wheel revolutions for one trip should be less than 100 for skip hoists, but may be as high as 140 for cage hoists.

The hoist wheel rotation at full speed should not exceed 75 RPM for a geared drive, or 100-RPM for a direct drive.

The (ratio of) hoist wheel diameter to rope diameter should not be less than 100:1

The hoist wheel diameter to rope (lock coil) diameter should not be less than 100:1 for ropes up to 1-inch diameter, 110:1 for ropes to 1½ inches diameter, and 120:1 for ropes to 2 inches diameter.

The maximum desirable speed for a friction hoist is 18m/s (3,600 fpm).

In North America, the desirable speed for cage service is approximately 2/3 of the optimum speed calculated for a skip hoist for the same hoisting distance.
Rope tread liners on the hoist wheel should be grooved to a depth equal to one-third (1/3) of the rope diameter when originally installed or replaced.

On most fiction hoist installations, the maximum tolerable groove discrepancy is 0.004 inches, as measured from collar to collar.

A friction hoist is available for production for 108 hours per week. This assumes the hoist is manned 24 hours per day, seven days per week, and that muck is available for hoisting.

With proper maintenance planning, a friction hoist should be available 126 hours per week (18 hours per day).

The easy way to design a friction hoist is to first determine the required hoisting speed and payload then determine the ropes that are needed to meet the required SF.

The hoist parameters can then all be determined only considering the hoist ropes and line speed.

The problem of short rope life in very deep shafts may be alleviated by a high factor of safety on the headropes and/or reducing the tailrope mass by 10-15%.

To avoid stress concentrations, it is desirable to manufacture a friction hoist wheel in one piece. Wheels up to about 3m (10 feet) in diameter can be shipped complete with shaft to most locations.

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(10 feet) in diameter can be shipped complete with shaft to most locations.

The tail rope weight is normally designed equal to the head ropes; however, tail ropes slightly heavier than the head ropes will assist acceleration from the loading pocket.

Slightly lighter tail ropes will provide a greater SF for the head rope section above the conveyance as it approaches the highest point of travel (the point at which uneven rope tension is most severe).

The distance between head ropes (spacing) varies between 8 inches and 12 inches.

At 8 inches, some installations experience rope slap but this is not considered a serious problem, since the ropes are running at the same speed.

The tail ropes should be oriented to overcome the Coriolis effect. If placed in the East-West direction, the tail ropes will freely open and close.

The Coriolis effect can be neglected, as it is much smaller than the movement at acceleration/deceleration and due to rope torque of the tail ropes.

High-speed friction hoists [over 12m/s (2,400 fpm)] are oriented with the wheel diameter East-West to minimize the effect of Coriolis acceleration on the tail ropes.

The effect of Coriolis acceleration on the tail ropes is diminished when a fixed guide system is employed, as opposed to using rope guides.

Listed below are the steps in designing and selecting a friction hoist.

1. Determine the SF required for the given hoist distance
2. Determine hoisting speed, V
3. Calculate the hoist cycle
4. Define (cage) or calculate (skip) the payload
5. Determine the weights of the conveyances required
6. Select hoist (head) ropes
7. Determine the wheel diameter of the hoist
8. Select balance (tail) ropes
9. Calculate the RMS power requirement

Example

Design and select two friction hoists at the same time. One is required for production hoisting and the other for cage service.

Facts:
1. Both hoists will be tower mounted in the same headframe
2. The skip hoist requires a capacity of 500 tonnes/hour
3. The cage hoist requires a payload of 26 tonnes
4. Each has a hoisting distance, H of 1,000m
5. The statutory SFs of Ontario, Canada are to apply
   • SF = 8 - 0.00164D, but not less than 5.5
   • D= approximately H + 50m

\[ V_{\text{skip}} = 0.44 H^{\frac{3}{2}} \text{(rounded)} \quad V_{\text{cage}} = \frac{2}{3} V_{\text{skip}} \text{(rounded up)} \]
\[ T_{\text{skip}} = \frac{H}{V} + 45 \quad T_{\text{cage}} = \frac{2H}{V} + 130 \]

The weight of the cage 20 tonnes
The weight of the skip \( P\{2.5 - (H \times SF/4500)\} \) (> 13 t)
Ropes\text{cage} : 6 × 32mm, 5.58 kg/m, breaking strength (BS) 890 kN.
Ropes\text{skip} : 4 × 32mm, 5.58 kg/m, breaking strength (BS) 890 kN.
SF= Number ropes × BS/maximum suspended load
diameter of the hoist= 100d
Tread pressure= total suspended load / n.d.D
Tread pressure \( \text{max} \) = 2,400 Kpa (skip) or 2,750 kPa (cage)

Tail ropes weight \(_{\text{cage}}\): twice the head rope weight.

Tail ropes weight \(_{\text{skip}}\): 3 ropes of weight \( \frac{4}{3} \times \) (head rope weight)

**Step 1:**

Determine the SF required for the given hoist distance.

Following is the SF required by statute for the hoist ropes.

- SF = 8 - 0.00164D, but not less than 5.5, in which \( D \) = length of suspended rope.
- \( D = \) approximately \( H + 50 \)m to account for rope suspended above the dump and beneath the loading pocket (in the case of a skip hoist) and similar extra rope length in the case of a cage hoist.
- SF = 8 - (.00164 \times 1,050) = 6.3

**Step 2:**

Determine hoisting speed, \( V \).

- The optimum skip hoisting speed, \( V = 0.44 H^{1/2} \) = 14m/s (rounded)
- A suitable cage hoisting speed will be about \( \frac{2}{3} V = 10 \)m/s (rounded up)

**Step 3:**

Calculate the hoist cycle (refer to previous chapter for cycle formulae used).

Calculate the skip hoist cycle time, \( T \) and the number of trips per hour. (Since the hoisting distance exceeds 600m, balanced hoisting with two skips is determined.)

- \( T = H/V + 45 = 1,000/14 + 45 = 116 \) seconds
- Trips per hour = 3,600/116 = 31.0

Calculate the cage hoist cycle time, only (a cage and counterweight is assumed).

- \( T = 2H/V + 130 = 330 \) seconds

**Step 4:**
Define (cage) and calculate (skip) payload.

The cage payload is given at 26 tonnes and the skip payload is calculated by dividing the capacity per hour by the trips per hour.

- Skip payload, \( P = \frac{500}{31} = 16 \) tonnes (rounded)

**Step 5 Determine the weights of the conveyances required**

- **cage:**

  The weight of the cage for 26-tonne capacity will be approximately 20 tonnes if it is steel (steel is typical for friction hoists).

  The weight of the counterweight will be made equal to the empty cage weight plus half the payload = 33 tonnes.

  (In this case, the counterweight will be designed to readily remove a portion of its weight for regular cage service with lighter payloads.)

**Step 5 Determine the weights of the conveyances required**

- **skip:**

  The weight of the skip, \( S \) will be approximately 13 tonnes (refer to next chapter) for a steel bottom dump skip that would normally be used for this application.

  However, this weight might not be enough to maintain the required tension ratio (in this case, the skip would be “ballasted” with extra weight).

  The empty weight of skip required to maintain a tension ratio of 1.40:1 follows.

  - \( S_t = P \{2.5 - (H \times SF/4500)\} = 16(2.5 - 1.4) = 17.6 \) tonnes

  - The skips will be ballasted to weigh 17.6 tonnes

**Step 6.a Select hoist (head) ropes -cage:**

Try 6 lock coil ropes of 32mm diameter weighing 5.58 kg/m and having a breaking strength (BS) of 890 kN.

- SF obtained = Number ropes \( \times \) BS/maximum suspended load
= 6 \times \text{BS/weight of ropes, payload and cage} \\
= 6 \times 890/g (35.5 + 26 + 20) = 6.7 (6.3 \text{ required}) \\

• \frac{T_1}{T_2} \text{ obtained} = \frac{(35.5 + 26 + 20)}{(35.5 + 33)} = 1.19 \\
• \frac{T_1}{T_3} \text{ obtained} = \frac{(35.5 + 33)}{(35.5 + 20)} = 1.23 \\

Maximum total suspended load = (35.5 + 26 + 20) + (35.5 + 33) = 150 \text{ tonnes} \\

**Step 6.b Select hoist (head) ropes -skip:**

Try four of the same lock coil ropes – 32mm diameter, 5.58 kg/m, and BS 863 kN. \\

• \frac{T_1}{T_2} \text{ obtained} = \frac{(23.7 + 16 + 17.6)}{(23.7 + 17.6)} = 1.39 (1.40, or less, desired) \\

Maximum total suspended load = (23.7 + 16 + 17.6) + (23.7 + 17.6) = 98.6 \text{ tonnes} \\

• SF obtained = \text{Number ropes} \times \text{BS/maximum suspended load} \\
= 4 \times \text{BS/weight of ropes, payload and skip} \\
= (4 \times 890/g(23.6 + 16 + 17.6) = 6.35 (6.3 \text{ required}) \\

**Step 7.a Determine the wheel diameter of the hoist, D:**

The statutory requirement for lock coil ropes is 100 times the diameter of the hoist rope at this location. \\

*Cage and Skip Hoist* \\

(Statutory) \( D = 100d = 100 \times 32 = 3,200\text{mm} = 3.2\text{m} \)

The statutory diameter may not be sufficient. 

For example, the diameter should be increased if the permitted tread pressure is exceeded, the number of wheel revolutions per trip is too high, or the ropes are greater than 35-mm diameter. 

**Step 7.b Determine the wheel diameter of the hoist, D:**

The tread pressure is calculated by dividing the total suspended load by the projected contact area of the ropes on the hoist wheel.
Tread pressure should not exceed 2,400 Kpa for a skip hoist or 2,750 kPa for a cage hoist with maximum payload.

**Step 7.c Determine the wheel diameter of the hoist, D:**

- Cage hoist tread pressure = \(150,000 \times g / (6 \times 3.2 \times 32) = 2394\) kPa (ok)
- Cage hoist revolutions = \(H/\pi D = 1,000/3.2\pi = 99.5\) (ok)
- Skip hoist tread pressure = \(98,600 \times g / (4 \times 3.2 \times 32) = 2360\) kPa (ok)
- Skip hoist revolutions = \(H/\pi D = 1,000/3.2\pi = 99.5\) (ok)

A wheel diameter of 3.2m should be satisfactory for both hoists. (On detailed investigation, it may be increased to 3.5m to increase rope life).

**Step 8.a Select balance (tail) ropes:**

Select non-rotating balance (tail) ropes matching the head rope weight and with a natural loop diameter equal to the compartment spacing.

**Step 8.b Select balance (tail) ropes- Note:**

Tail ropes can be custom manufactured to meet precise weight requirements (i.e. kg/m).

For the cage hoist (assuming no deflection sheave), select three non-rotating ropes weighing twice the head rope weight.

The head ropes weigh 5.58 kg/m; therefore, the tail ropes will weigh 11.16 kg/m with a 53mm diameter.

If the ropes are 34 by 7, the natural loop diameter will be \(46 \times 53 = 2,438\)mm (unsatisfactory).

If the ropes are 18 by 7, the natural loop diameter will be \(60 \times 53 = 3,180\)mm (satisfactory).

**Step 8.c Select balance (tail) ropes- Note:**

For the skip hoist (assuming a deflection sheave is required to bring the conveyances closer together in the shaft, say 2m between compartment centers),
select three non-rotating ropes of weight $= \frac{4}{3} \times 5.58 = 7.44 \text{ kg/m}$ with a 43mm diameter.

If they are 34 by 7, the natural loop diameter will be $46 \times 43 = 1,978\text{mm}$ (satisfactory).

**Step 9.a Calculate the RMS power requirement:**

Assume there is a force-ventilated DC or cyclo-converter drive.

The skip hoist RMS power $= a$ constant $(k) \times$ unit weight of the ropes $\times$ (speed)$^{5/4}$

(SF obtained)

- $k=24$ for a standard DC (FV) or cyclo-converter drive (FV)
- The skip hoist RMS power $= (24/6.35) \times 4 \times 5.58 \times 14^{1.25} = 2,284 \text{ kW (3,064 HP)}$

**Step 9.b Calculate the RMS power requirement - Check:**

Skip hoist RMS power $\approx$ full speed power plus 5% $= 16 \times 9.81 \times 14 \times 1.05 = 2,307 \text{ kW}$

- The cage hoist RMS power $\approx$ skip hoist factored for speed, cycle time and out-of-balance loads $= 2,284 \times (10/14)^{1.25} \times (2 \times 116/330)^{0.5} \times 13/16$

$= 1,021 \text{ kW (1,370 HP)}$

This sample problem produces a result satisfactory for a basic engineering study.

Where accurate design is required for the power train, it is recommended that a computerized hoist program be employed.

The tension ratio is the ratio between the suspended weight on the hoisting side and the descending side.

It is referred to as the $T_1/T_2$ ratio, except in the case where a counterweight is hoisted while the empty conveyance descends ($T_2/T_3$).

The determination of a safe ratio is an elegantly simple mathematical exercise; therefore, it is surprising that a controversy rages between European and North American hoisting experts as to the correct design value for this parameter.
50 years ago, when these hoists were first introduced to North America, it was universally agreed that a value of 1.45:1 was a safe, practical value.

In the interim, Europeans maintained this value and even increased it (up to 1.50:1).

At the same time, problems encountered in North America led to the adamant conclusion that the Tension Ratio ought not to exceed 1.40:1 to help ensure trouble-free operation of friction hoists (which at the time were all tower-mounted).

Without going into details, it should be noted that there are a variety of related factors that affect the choice of tension ratios.

For purposes of this course, it is proposed that the conservative value (1.40:1) be employed as a design guideline for any friction hoist installation in North America.

The increased cost (if any) normally only amounts to extra steel in the construction of the conveyance or counterweight (to make them heavier).

In addition, the lower ratio is inherently safer since it permits a greater braking effort to be employed without the danger of incurring rope slip.

**Ground Mount Friction Hoist**

Ground mount friction hoist advantages.

- Shorter headframe.
- Steel headframe
- An elevator is not required in the headframe.
- An overhead bridge crane may not be required.
- Easier access for maintenance.
- A water supply to the top of the headframe is not required.
- Shorter runs of power cables.
• Less susceptible to damage from overwinds, mine explosions, lightning, and earthquakes.

**Tower Mount Friction Hoist**

Tower mount friction hoist advantages.

• Zero or one deflection sheave is required. Two are required for a ground mount – one is subject to reverse bending of the hoist ropes.

• Installing and changing head ropes is less complicated.

• Less real estate is occupied.

• The hoist ropes are not subject to the elements – icing is less of a concern.

• Rope vibration (whip) is less of a concern.

• The headframe tower may be more aesthetically pleasing.