HERITAGE RAILWAY ASSOCIATION

GUIDANCE NOTE

MECHANICAL BRAKING SYSTEMS
for Rail Mounted Cableways

Purpose
This document describes good practice in relation to its subject to be followed by Heritage Railways, Tramways and similar bodies to whom this document applies.

Endorsement
This document is endorsed by the Rail Mounted Cableways Liaison Committee which includes both Operators and the HSE.

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A. Introduction

1) This Guidance Note deals with mechanical braking systems used in the control of rail mounted cableways and are fundamental to their safe operation. Their condition and maintenance must be of the highest integrity to ensure safe operation.

2) The Guidance Note is primarily aimed at owners and operators of existing cableways and gives information and recommendations based on that premise. It is not intended as a design standard for new installations nor does it provide sufficient information on the skills and experience that may be necessary above and beyond that which can be delivered in a document of this type.

3) Whilst intended for cableways, much of the information given herein is applicable to similar brakes such as those found on railway steam and diesel cranes; those using cranes may find the guidance useful.

4) This document has been provided to assist the duty holders of heritage rail-mounted cableways (R-MC's) and similar bodies in complying with their legal obligations.

5) Duty holders will be aware that they have a legal obligation to passengers, contractors, other visitors and staff under the Health and Safety at Work etc Act, 1974.

6) The term 'man' or 'men' in this Guidance Note should be read as applying equally to men and women and 'he', 'him' and 'his' should be similarly interpreted.

7) The term 'staff' in this Guidance Note should be taken to include unpaid volunteer workers as well as paid staff.

B. Recommendations

1) This Guidance Note is issued as a recommendation to duty holders.

2) Many R-MC's are already operating systems which, in some cases, are to a higher standard than those set out in this Guidance Note. This highlights the fact that it is the responsibility of the duty holder, having undertaken the necessary risk assessments, to implement controls that are necessary relative to the operating conditions of their R-MC.

3) Where R-MC's decide to take actions that are not in conformity with these recommendations, following appropriate risk assessments or for other reasons, it is recommended that these decisions are reviewed by the senior management body of the organisation and a formal minute is recorded of both the decision reached and the reasons for reaching it.

C. Function of the Brakes

1) A brake is a system for
   a) absorbing energy and
   b) resisting force

2) Brakes may take a number of forms; for example, they may drive a pump or generator to absorb the energy. However, most braking systems employ friction to transform the energy into heat. This heat is then transferred to the surrounding environment. This Guidance Note is generally only concerned with friction brakes.

3) The braking systems have to perform several functions:
   a) Retard the system at a rate that does not present danger, preferably under all conditions.
   b) Act automatically in the event of a detected abnormal or emergency situation
   c) Hold the system stationary under all conditions, including inadvertent application of power

4) Service Brake
   a) The term service brake is usually used to describe the brake that is used for the normal control of the system. It usually has to be capable of:
      i) Holding the maximum normal out of balance force plus the full power of the motor applied in the same direction
5) Emergency Brake

a) The term emergency brake is usually used to describe the brake that is brought into operation if there is an unplanned event. Such events include, but are not limited to:
   i) overspeed
   ii) overwind
   iii) slack rope
   iv) emergency stop
   v) operation of deadman’s device
   vi) broken rope or drawgear

b) The emergency brake is generally a fixed level of braking and is not controllable. The level of braking is carefully ascertained at design and commissioning so as to bring the system to rest with the minimum of risk to passengers. It is controlled by the safety system and, other than by operation of an emergency stop, is outside the control of the engineman.

c) The service and emergency brakes may be separate, unconnected systems or they may be combined within one system.

D. Electrical Braking

1) Many electrically powered systems are provided with systems which provide electrical braking in various forms. Whilst such systems are outside the scope of this Guidance Note it is worth mentioning a few important points:

   a) Although a good electrical control system can bring the haulage rope virtually to a stand it cannot hold it entirely stationary and a mechanical brake must always be applied when stationary. (The service brake.)

   b) No electrical braking system is fail safe and the ultimate control of the cableway must be by means of a mechanical brake (The emergency brake.)

   c) Where the normal means of control is by an electrical braking system it is important that the mechanical brakes are kept in good condition and routinely tested to ensure that they perform satisfactorily.

E. Self-Servo Factor and Stability

1) The torque output of any braking system depends on the operating force providing the load on the brake lining. The brake force can be applied radially or axially. In addition, for radial braking systems, it depends on the geometry of the brake and the shoe arrangement. Radial braking systems generally exhibit what is known as a self-servo factor. This self-servo factor is a function of the direction of rotation relative to the anchor point of the shoe, the angle of wrap of the shoe and the applied braking force and can be positive or negative. Shoes that have a positive self-servo action are said to be leading shoes and those with a negative servo action are said to be trailing shoes. With any system that operates in both directions, the status of the brake shoe will change with change in direction.

2) A brake which gives a very high torque capacity for a low operating force is said to have a high self-servo factor. The greater the self-servo effect, the poorer the stability of the brake in its operation. This is because any slight change in the operating force has an exaggerated effect on the braking torque and can thus make it very difficult to control satisfactorily.
F. Types of Mechanical Brake

1) The following is not meant to be a design guide for mechanical braking systems but is provided to give users some idea of the characteristics of these braking systems.

2) Band Brakes
   
a) The band brake is probably the simplest form of brake in use today and was much in favour in the past. It requires no accurate machining and can be easily reproduced in a reasonably equipped workshop. The band brake consists of a flexible steel band which is lined with a friction material. Band brakes typically have an angle of wrap of 270-300°. In its simplest form the band brake provides a high self-servo action in one direction of rotation and, conversely, a negative servo effect in the opposite direction so it is frequently used where braking is usually in one direction only. (i.e. on the hoist rope of a crane.)
b) Band brakes can be prone to grabbing and judder and usually have to be physically held away from the brake path by springs. The steel band is susceptible to distortion, resulting in uneven contact with the brake path and uneven wear of the brake lining.

c) In order to minimise the self-servo effect when band brakes are applied on reversing drives, a split band arrangement is frequently used, with one part providing positive self-servo and the other negative self-servo.

d) The band brake does, however, have one further disadvantage, notably its propensity to fatigue failure, especially at the point where the band is attached to the end fastenings, and it is nowadays frowned upon where it is required to perform a critical task, which most braking systems are. Where band brakes remain in use they should be inspected daily and critically examined at least monthly.

3) Rigid Shoe Brakes

a) The rigid shoe brake is one of the most common types of brake and generally gives a consistent performance. It usually only has a small degree of self-servo effect. Normally this type of brake is formed of a unit having two shoes operated simultaneously by a linkage.

b) There are various arrangements of rigid shoe brakes and these arrangements give rise to differing characteristics. It would be a significant task to detail all the differing variations but the following will provide some information on the more common arrangements.

c) A common arrangement is shown in fig.4 and is generally referred to as a post brake. The two brake posts are firmly anchored to the foundations and are connected together by a tie rod, which is usually provided with a means of adjustment, such as a turnbuckle. This is provided to allow take-up of wear in the linings. Although the anchor brackets are firmly attached to the foundations, there is often some means of adjustment provided to allow the brakes to be set up when first installed. The top tie rod is usually connected to a bellcrank which is in turn attached to another tie rod and then to the operating mechanism, which may be by manual, deadweights, springs, air or hydraulic means.
d) With this type of brake it can be shown that the pressure on the lining varies considerably with very little pressure at the lower end of the lining (See Fig. 5). This results in the lining wearing significantly at the top whilst largely being unworn at the bottom of the shoe. The anchor brackets should not be moved to even out the wear on the linings as this will significantly alter the characteristics of the brake and is likely to result in brake judder.
4) Another common form of rigid shoe brake is one where the shoes are curved. Such brakes are generally referred to as caliper brakes. Such an arrangement is shown in Fig. 6.

Fig 6 A Caliper Brake

Fig 6 B Caliper Brake

5) The arrangement shown in Fig 6A gives very similar characteristics to the post brake with the majority of the brake pressure (and hence lining wear) taking place at the top of the brake lining. It will be seen that, by moving the shoe pivot point round towards the vertical centreline of the brake, as shown in Fig 6B, that there is a significant shift in pressure distribution on the lining and better wear characteristics as a consequence. Fig 6B also shows both brake shoes pivoting at the same pin. This is not good practice as it makes it difficult to set up the brakes in the first instance, especially with larger brake shoes where it is difficult to arrange true concentricity of the brake arrangement. It is far better to keep the two pivot points separate to allow individual adjustment.

6) The geometry of all these brakes is not perfect and there is an increasing gap between the brake lining and the brake path from the bottom to the top of the brake shoe. This gap can be made more or less equal by placing the shoe pivot point at a position √2 x Radius of the brake path from the centreline of the brake path. It is usual to rotate the shoe geometry by about 10° to allow independent brake shoe pedestals.

Fig 7 Root 2 Brake Arrangement
There are limitations to the relative length of a rigid brake shoe. As the lining wears, it becomes asymmetrical in thickness due to the geometry of the shoe, as explained above. In addition, the means of attachment will become insecure as the lining thickness reduces. Once one part of the lining reaches this point it is necessary to replace it as failure to do so may result in the lining being ripped away from the shoe with consequential results. For these reasons and for those of economy, it has been found that the length of lining per shoe should not significantly exceed 90° of arc.

G. Articulated Shoe Brakes

1) Such brakes are commonly found on high speed shafts where the size of the brake is relatively small.

2) Such brakes apply a true radial force to the brake shoe and this gives a more even wear characteristic. However, it is difficult to prevent the end of the brake lining from rubbing on the brake path when in the off position unless springs or other means are provided to help centralise the brake shoe. This becomes less practical as the size of the brake increases.

H. Disc Brakes

1) Disc brakes have been around for over eighty years but it was not until the 1950’s that their use started to become more widespread. Operationally, the disc brake has several advantages over the rigid post brake.

   a) Because the brake acts axially and is always self-opposing, it does not put any load onto the bearings of the rotating shaft.

   b) Lining wear is generally even as the intensity of pressure is effectively constant over the full surface of the brake pad.

   c) Disc brakes can be applied in multiple. This both keeps units to manageable sizes and builds in redundancy, eliminating single line components and reducing the risk of catastrophic failure

   d) They can operate to close clearances, reducing application time.

   e) Disc brake calipers are relatively cheap and hence holding spare calipers is realistic.
2) The disc brakes’ main disadvantage is that the disc needs to be confined to close axial tolerances, meaning their use with floating bearings can be problematical, unless the end float is closely controlled to within the limits specified by the brake manufacturer. Ideally, rotating contact bearings should be used where disc brakes are provided.

I. Brake Linings

1) The earliest brake material was timber, which whilst effective, does not provide a consistent coefficient of friction and, as a consequence, does not provide for consistent and repeatable braking. Woven linings, generally having a cotton base, came to the fore in the 19th century and, in more recent times, asbestos has been the prime brake lining material, generally impregnated with resin. The use of asbestos containing materials is now prohibited but it could quite possibly still be found on brakes that see little dynamic use. If you have woven brake linings or, indeed, any lining of indeterminate material and of an age dating back earlier than the year 2000 you should consider that the material has an asbestos content and act accordingly. Manufacturers would commonly identify the base materials in the lining designations. Thus, Ferodo, who had a significant market share in woven linings, designated their most popular woven lining material ‘ZA1’, signifying that it was asbestos based around a zinc wire. Similarly Ferodo BA is an asbestos based fibre material spun around a brass wire. The type of lining was frequently stencilled onto the back of brake linings. The omission of an ‘A’ in the designation should not, however, be taken to mean that asbestos is not present.

2) In more recent times, cotton has come back into favour as the base material for woven linings and, along with Kevlar and other synthetic fibres is found in common use.

3) Most brake lining materials have a coefficient of friction of the order of 0.35-0.4 but linings with both higher and lower values have been used. The braking system will have been designed about a particular coefficient of friction and, if it is found necessary to change the brake lining material, it is an important consideration in making any choice.

1) Woven linings

1) Woven brake linings are still very much in use on caliper braking systems and are relatively flexible. The lining is generally attached to the brake shoe by means of brass or copper rivets but on larger brake shoes the use of countersunk bolts of brass or copper is quite common. The use of aluminium rivets is not recommended as this material has undesirable scoring propensities when in contact with ferrous brake paths. The riveting or bolting method of fixing is relatively straightforward and can be accomplished in the field without recourse to special tools. If the lining is not pre-drilled it should be carefully and adequately clamped to the brake shoe and drilled through from the shoe side. The hole in both the lining and the brake shoe should be a close tolerance fit for the rivet. The lining should then be counter-bored to a depth of approximately half its thickness with a suitable sized counter-boring drill. If required, the lining can be removed from the shoe for this to be done. The lining should then be carefully re-clamped to the brake shoe ensuring that all the holes align.

The rivets should then be inserted from the face side and clench together with a suitable anvil and tool. In riveting, you should start at the centre of the lining and work alternately towards the ends. Either flat headed or 150° countersunk headed rivets should be used. As a guide, the rivet diameter should be about half of the lining thickness. i.e. use a 3/8” diameter rivet for a 3/4” thick lining. If bolts are used, a similar process should be followed. Care should be exercised to ensure that the bolt head is not pulled through the lining by over tightening.

2) Woven linings held onto a brake shoe in this way should be replaced when the likelihood of the rivet touching the brake path is imminent.
2) Moulded linings

1) Disc brakes and small caliper brakes generally use moulded linings bonded to a backing plate. Moulded linings are available in any number of different materials. As with woven linings, asbestos was once commonly used. Modern linings can be of many different materials from resin bonded synthetic fibres to Kevlar and ceramics and may often have metallic particles embedded, both to provide heat dissipation and give strength. The backing plates of the brake linings are usually specifically designed for the particular brake and are usually only available from original suppliers. However, it is possible to have new linings bonded to old backing plates and various suppliers will undertake this work. The characteristics of different moulded brake linings can vary significantly and users should not change the type of lining without considering the potential impact of such action.

2) Moulded linings bonded to back plates provide several advantages. The usable thickness and thus life of the lining is increased as it is possible to wear the friction material almost down to the back plate. Elimination of rivet holes also reduces the likelihood of scoring of the braking surface. Braking systems designed for use with bonded linings generally work at a higher surface pressure and are less likely to suffer from squeal and judder.

J. Braking Surfaces

1) Brake paths

1) The term brake path is usually used to refer to the braking surface on which a radial brake acts. They are generally either of cast iron or steel. For applications where the brake path is attached to or even part of the haulage drum, cast iron is probably the better material and should be a fine grained pearlitic iron. If steel is used, it should have a Brinell hardness of at least 200. Cast steel is not considered to be a satisfactory material.

2) It is important that the circumferential face of the brake path is concentric with the axis of the drum on which it operates. Over time, it is possible for the brake path to wear oval or even eccentric. A brake path in this condition will still work but the clearance between the brake shoe and the path has to be excessive to prevent the brake shoe from rubbing when in the off position. It will also cause uneven braking and may lead to an uncomfortable ride where the brake is used to control the descent of a conveyance. Brake paths may also become scored in service. This leads to increased wear of brake linings. It is generally possible to machine brake paths in situ and there are several firms which will undertake this work. However, brake paths should not be machined to such an extent that they cannot adequately support the loads imposed by the brake shoes or dissipate the heat induced by brake applications.

3) Some brake paths of cast iron have an outer lip. Unless the brake shoe pivots are exceedingly worn, this lip provides no useful purpose and actually makes it difficult to set up the brakes as the gap between lining and path cannot be observed. Consideration should be given to removing this lip if brake paths are ever machined. Alternatively, small sections of the lip can be ground back to enable the interface between the lining and path to be measured.

4) The brake path should be slightly wider than the brake lining. It is important that the brake lining does not start to overhang the brake path as, once this starts to happen, it will create side loadings on the shoe and rapidly aggravate the situation. The usual cause of this happening is wear or possibly misalignment in the brake shoe pivots. If this does start to happen, a short term remedy is to trim back the brake lining to the same level as the general body of the lining but the cause should be investigated and rectified.

2) Brake Discs

1) Brake discs are also usually made of cast iron or steel and again should be of either a fine grained pearlitic iron or steel having a Brinell hardness of at least 200. Cast steel should not be used.

2) It is important that the radial faces of the brake disc remain true and parallel and that the thickness of the disc does not vary. Brake discs can become warped in service, particularly if they have become overheated.
3) For satisfactory operation it is important that the clearance between the brake disc and the brake shoes is kept to a minimum and, for this reason, it is important that the disc does not have any axial float. This requires close attention to the bearings of the shaft or drum on which the disc is attached to eliminate lift and side play if they are not of the rolling contact type.

4) As with brake paths, it is generally possible to machine brake discs in situ but they are usually much smaller and it is often easier to remove them for this purpose.

K. Back Stops

1) Radial braking systems are frequently provided with back stops. These are small adjustable screws situated behind the brake shoes.

![Fig.10 Back Stop](image)

2) Back stops are provided for two purposes:
   a) As a means of brake adjustment. Due to their design and the force of gravity, radial brakes will tend to fall in a particular direction when the brake is taken off, causing the brake shoe to rub on the brake path. This is generally (but not always) towards the brake operating mechanism. The back stop, as its name implies, provides a stop to limit this. When the brakes come off, this brake shoe will move back until it contacts the back stop and can move no further. The brake shoe on the opposite side will then move away from the brake path.
   b) As an aid to maintenance. The back stops can be used to hold the brake shoe against the brake path and enable the various components of the braking system to be dismantled for maintenance purposes. There is generally one back stop for each brake shoe. Provision of back stops for this purpose is usually confined to larger brake posts.

3) It is important that, if back stops are provided on all brake shoes for maintenance purposes, those on the opposite side to the one required for brake adjustment is wound clear of the brake shoe in normal operation. If it comes into contact with the brake shoe it will prevent the brake from coming properly off and will introduce unwanted forces into the braking system.

4) On some small braking systems back stops are not provided and the brake shoes are held clear of the brake path by means of tension springs.
L. Conveyance Brakes

1) Where possible conveyance brakes should be provided for protection against the haulage rope breaking, failure of a rope attachment or when the maximum permissible speed of the carriages is exceeded. They should be capable of safely stopping the system under the most onerous conditions of speed and load. These brakes can take various forms and, again, what follows can only cover the broad principles.

2) The simplest form of conveyance brake is one that acts on the running surface of the rails. The brake shoe is normally held clear of the rail and is applied either by a collapsing suspension or a downward thrusting spring. The brakes are usually held off by hydraulic pressure which is produced by a manually operated hydraulic pump. The brakes can be applied by means of a manually operated valve but more usually they are applied by a centrifugal governor which is driven by the car wheels. If the speed of the car exceeds a pre-determined level, usually of the order of 20% above normal maximum speed, the brakes are automatically applied. A figure of 20% is used to allow for any overspeed device fitted to the cableway drive engine to operate first as this is the preferred approach. Such a system is well proven and reliable but it is only of any practical use on gradients shallower than about 1 in 8 as a coefficient of friction greater than 0.16 cannot generally be achieved under all conditions. Steeper gradients can be accommodated if tungsten carbide tipped brake shoes are used as these effectively dig into the rail head and are largely independent of railhead condition. Such brake shoes are considered to have a friction coefficient of 0.4 and 1 in 4 is the practical limit for such brake shoes. It is recommended that the shoes are replaced if they have been used in a dynamic situation. At a minimum they should be carefully examined after each occasion when they are deployed to ensure that the tungsten carbide tips are fit for further service.

3) Steeper gradients can be accommodated if spring applied brakes are used in conjunction with what is essentially a linear brake path, which may be a rail or even timber. Any such reaction member must be properly anchored to the track or foundations and capable of taking the full force of the applied brake at all points throughout the length of the system. The brakes can be held off by hydraulic pressure and applied by an overspeed device. As with track brakes this is usually set to be about 20% in excess of the maximum operational speed. As with any friction braking system the applied retardation force is going to be dependent on the friction coefficient and this may vary considerably with the environmental conditions, both natural and man-made.

4) These brakes can also be arranged to be held off by means of the haulage rope or fly rope. The principle here is that, if the rope breaks the loss of tension causes the brakes to be applied, generally by means of springs. This method is only effective with a broken rope and would not operate in the event of a failure of the cableway drive engine brake. Practical testing of such a system is also much harder to achieve.

5) Timber has been mentioned as a reaction member for conveyance brakes and can provide good results. However, it does have a disadvantage in that its physical dimensions can alter significantly with change in atmospheric condition and this variation of dimensions can be present throughout the length of the system. Any braking system that utilises timber must be capable of successful operation at the extremes of dimensions that may be encountered.

M. Rope Brakes

1) Proprietary Rope brakes have been fitted to some Cableways. These devices provide a means of emergency braking if the system goes into an uncontrolled acceleration or unexpected movement is detected. There are several suppliers of these brakes but they generally follow the same principle in that they apply a braking force directly onto the steel wire ropes to stop the conveyance(s). The haulage rope(s) pass through the brake, which has two brake pads, one on each side of the rope. There is generally one fixed pad and one moving pad. The moving pad is held in the off position clear of the ropes and allowing them free passage through the brake. In operation the moving pad squeezes the ropes between it and the fixed pad, the brake force being determined by the load of the applying force; usually springs.
2) Whilst most rope brakes utilise a spring to provide the clamping force the Atwell brake employs a wedging action as its primary application force. The free or drop jaw is held clear of the ropes by an electro-magnet. When the system detects an overspeed or other abnormal condition the electro-magnet is de-energised and the jaw is released, trapping the rope between the moving and static jaws. Movement of the rope through the jaws increases the wedging action and thus increases the braking force. Because of this principle, the Atwell brake is a uni-directional brake and would require installation of a double brake to give emergency braking in both directions.

![Fig 11 Atwell Rope Brake](image)

3) In mounting a rope brake care must be taken to ensure that the ropes do not deviate from a fixed path when in motion. This usually requires the brake to be mounted close to a sheave. If this cannot be achieved some other system of rope control will have to be provided.

4) Rope brakes in themselves only provide a back-up for the brakes on the cableway drive engine and do not cater for a broken haulage rope situation. Some types may not be fail-safe in operation.

5) Where rope brakes are retro-fitted adequate consideration needs to be given to their method of mounting and how the loads induced are transmitted to the structure and foundations. Such will undoubtedly require design and appraisal by a competent person other than the brake manufacturer.

N. Maintenance

It is critically important that Cableways are regularly maintained and this particularly applies to any braking system, even those that do not see regular use. They should be inspected daily and more critically examined periodically. It is recommended that braking systems are dismantled from time to time as part of the scheme of maintenance. At such examinations they should also be subject to appropriate non-destructive examination to detect any flaws, especially the onset of fatigue cracks. It should be noted that where a brake is fitted to a motor or similar shaft the brake forces are transmitted to the drum via shafts and gears and these should be considered to be part of the braking system as their failure will render the brake ineffective.

1) Frequency of examination

It must be up to the operator to determine the frequency at which parts of the braking system are examined, taking account of the manufacturer’s recommendations. This should be based on a critical appraisal which should take account of the consequence of failure. If failure of the component would lead to the total loss of braking at the cableway drive engine it is considered to be
a single line component. Such items should be thoroughly examined annually, including being subject to appropriate non-destructive testing. Other components should be examined at less frequent intervals determined by their history, stress levels, accessibility for routine inspection and not least, impact on the business as a consequence of failure. It is suggested that no part of a braking system should go longer than five years between being dismantled and critically examined.

2) Lubrication

Lubrication should be carried out at regular intervals. Where manufacturer’s guidance is available this should be adhered to. Lubrication of brakes should be done carefully and any excess wiped off as it may easily get onto a brake path and lead to contamination of the lining, which can severely reduce braking performance.

3) Cleaning

Braking systems should not be allowed to become covered in oil, dirt and other detritus as this may well prevent those carrying out inspections from observing defects. Whilst greasing of components is important for both operation and protection from corrosion this should not be to an extent that nuts, bolts, split pins, etc are covered and cannot be seen. If it is deemed necessary to do this to protect them they should be cleaned at each inspection and fresh grease applied.

4) Brake adjustment

It is important that all brakes are maintained in proper adjustment. The frequency of adjustment will largely depend on the amount of use that the brake gets in normal service. Adjustments need to be done for several reasons:

a) If the brakes are not adjusted, there is a danger of them ‘bottoming’ and becoming ineffective.

b) The time for the brake to be applied increases. This can become quite significant, especially with slow acting systems and brake application time is critical in emergency brake applications.

c) With spring applied systems, the springs get weaker as the required travel increases. This, again, can become a critical factor as the brake force is reduced.

Brakes should never be allowed to wear such that the movement of the piston, foot tramp, thruster, etc, is greater than 75% of its travel and should preferably be maintained at less than 50%. If this cannot be achieved it is likely that cumulative wear of linkages is the main cause and this should be investigated and rectified. It is recommended that some form of indication such as a limit switch or marker should be provided to indicate when 75% of brake travel has been reached.

Braking systems vary significantly and what follows can only be taken as a general guide to the adjustment of brakes. Where proprietary equipment is in use, the manufacturer’s instructions should be adhered to. When this is not the case, operators should provide specific detailed written procedures for their own individual systems.

Before adjusting any brakes, the system must be made such that it is safe to take the brake off for an extended period. Reliance must not be made on any electrical braking or control system for this purpose.

5) Radial Brakes

Place the brake in the fully off position. If a backstop is fitted, use the one in use to adjust the clearance between the adjacent brake shoe and the brake path such that it is at the minimum acceptable amount. Bear in mind that the brake lining may not have worn equally and will probably be closer at the bottom of the shoe. Once the first shoe has been adjusted the turnbuckle or adjusting screw on the top tie rod can be used to adjust the second brake shoe.

Whilst brake shoe clearance should be kept to a minimum it should also be borne in mind that, when in use, brakes can heat up significantly and, if this occurs, the effective diameter of the brake path will increase and take up the clearance between the shoe and the brake path. In such circumstances the brake will start to drag and aggravate the situation. If this should occur, it is safer to wait until the brakes have cooled down before making any adjustments as to do so whilst the brake path is hot will mean that there may be excessive clearance once it has cooled down and this may render the brakes ineffective.
6) Disc Brakes

As with radial brakes, it is important to keep disc brakes well-adjusted and as close to the braking surface as the system will allow. Unlike road vehicle disc brakes the majority of disc brakes likely to be used will be of the spring on/pressure off type, whether air or hydraulically operated. If they are left to wear unadjusted, the same problems can arise. Different manufacturers of disc brakes adopt different means of adjustment and the manufacturer's maintenance literature should always be followed. Some disc brakes have self-adjusting mechanisms but they should be checked regularly to ensure that they continue to adjust themselves as intended.

O. Materials and Design

1) Wherever possible, any replacement parts should be made from the same material as the original component. Where this is not known, the materials specified in Appendix A have been proven to give satisfactory service.

2) The use of non-metallic bushes as an alternative to metallic ones needs to be carefully considered. Such bushes are invariably intended to work with steel pins or shafts. If these pins or shafts are not lubricated, they will almost certainly corrode over time, causing them to seize in the bush. This is more likely than normal with brakegear as the rotational movement is generally minimal. However, most manufacturers of non-metallic bushes do not advocate the use of oils and greases as they can adversely affect the material. Where such bushes cannot be used with oils and greases they should not be used.

3) Where components are replaced for whatever reason it is a good idea to look at the design to see if it can be improved without affecting its functionality. For example, screwed tie rods can be much improved with regard to likelihood of fatigue failure by relieving the end of the screw thread. Pins simply held in place by split pins may be a simple solution but there is always the possibility of failure of the split pin. It is preferred that pins are headed, shouldered and have screwed nuts secured by split pins where possible.

<table>
<thead>
<tr>
<th>Poor Design</th>
<th>Good Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tie Rod End</td>
<td>Screw thread machine cut with profiled tipped form tools</td>
</tr>
<tr>
<td>No preparation of rod end and screw thread simply ends</td>
<td>Plain portion of rod tapered to just below thread root diameter and well rounded at the thread</td>
</tr>
<tr>
<td>Pin Joint</td>
<td>Pin headed with nut screwing tight up to a shoulder and secured with a split pin</td>
</tr>
<tr>
<td>Pin only held in place by split pins</td>
<td>No sharp corners on clevis</td>
</tr>
</tbody>
</table>

Fig 12 Examples of poor and good design
P. Contamination

1) No matter how well designed and maintained a braking system is, it relies on a fairly constant coefficient of friction between the brake and the braking surface for safe operation. This coefficient of friction can be quickly and catastrophically modified if the brake linings become contaminated. Causes of contamination are many and varied and what follows is not an exhaustive list.

a) Condensation. This is a real possibility on any braking system operating in an unheated building and is likely to happen when there is a sudden rise in ambient temperature. The cold metallic surfaces take time to heat up to the air temperature and this causes water vapour to condense on them. Condensation will significantly reduce the braking levels. If condensation monitors are not fitted and the building is not heated, operators must be well aware of this possibility.

b) Rain. On many installations there is a possibility of rain being blown onto braking surfaces if the weather conditions are right. Again, operators should be aware of this possibility.

c) Burst pipes. Water or any liquid carrying pipe should not be carried above any braking system as the danger of a leaking or even burst pipe is always there, however slight is the possibility. If a brake is situated in a pit there is also the possibility of water from bursts or leaks filling this.

d) Maintenance. Routine and non-routine maintenance can lead to contamination if care is not taken. Examples are over-lubricating of components, use of paraffin as a cleaning agent and oily hands being placed on braking surfaces. If steel wire ropes are over lubricated the oil can find its way onto the brake paths. Any oil contamination should be immediately removed using clean rags and a suitable solvent degreaser. If oil gets onto a woven brake lining it may become necessary to renew that lining.

2) If contamination is suspected, the cableway should not be operated until satisfactory levels of braking have been established.

Q. Brake Testing

It is fundamental that braking systems are tested, see Appendix B. The brakes need to be tested to:

a) ensure that they will do what they are intended to do, These are generally referred to as commissioning tests.

b) ensure that the brakes continue to do what they are meant to do. These tests are routine and are generally less onerous and easier to carry out.

Commissioning Tests

The Cableways Minimum Requirements guidance states that the brakes should be capable of safely stopping the system under the most onerous conditions of speed and load. Safe stopping means safety to persons and should take account of the operation of the vehicles, ropes and other parts of the installation and the combined effect of multiple braking systems and where modifications are carried out the system should be re-commissioned prior to use for the first time.

There are no hard and fast rules with regard to this. Each system will be different and the commissioning tests will have to be specified for each particular system. With a modern braking system the designer will have done all the necessary calculations and specifications so it will be known what is expected of the brakes. The commissioning tests should be designed to prove that they do produce the intended levels of braking. However, with many heritage cableways any original design parameters are unlikely to still exist so it is not possible to commission against these. Nevertheless, it is fundamental that the braking systems are proved to be satisfactory in normal operation. Where these do not exist, it is recommended that such calculations are done. Appendix C gives an example of how the levels of braking and retardation rates may be calculated.

Every cableway is different so what follows is only provided for guidance. There are no hard and fast rules as to what is an acceptable level of retardation but the retardation rates measured should not generally exceed an average retardation of 2.1m/s² (approx. 7ft/s²) when a full brake application is made. This worst case situation is usually encountered when a full conveyance is ascending the steepest gradient and balanced by an almost empty conveyance descending. Higher rates of retardation will generally result in the potential for persons being thrown out of their seats. (As the ropes are elastic the maximum retardation at the cars and felt by persons will be approximately twice
The retardation rates with a full load descending are less critical with regard to persons being thrown out of their seats as they will be lower than the maximum given above. However, the system has to be able to stop with a reasonable retardation and minimum of 0.9 m/s\(^2\) (3 ft/s\(^2\)) is a sensible criteria to adopt under a full brake application. The effect of the actual retardation rates achieved can best be ascertained by a person actually riding in a conveyance but this should only be done when it is considered safe to do so.

In measuring the retardation rate the time taken from the start of retardation to standstill should be used and not the point at which brake initiation occurs as there will be a time lag between the two. (see Fig 13)

### 1) Service Brakes

The service brake should be capable of holding the haulage rope stationary when the full power of the prime mover is applied in either direction. It should also bring the system to rest safely. In this respect, it is important to consider the effect on persons being carried in the conveyances.

### 2) Emergency Brakes

The emergency brakes (in whatever form) are generally a fixed level of braking and may or may not provide the same brake force as the service brake. However, similar minimum and maximum levels of retardation as given above should be applied. Higher levels of braking may be acceptable if the gradients are not excessive or the seating arrangement is such that persons will not be thrown out of their seats. If standing passengers are allowed the effect on them also needs to be considered.

### 3) Compounding

Compounding occurs when two or more independent braking systems are applied at the same time. As each braking system has to be capable of safely stopping the system when operated their combined effect is going to produce significantly higher levels of retardation. Conditions under which compounding of brakes may occur are:

a) Power failure   As the emergency brakes are designed to be fail safe loss of power will cause them to be applied. If, say, a drum mounted emergency brake and a rope brake are fitted both will come into operation on power failure.

\[
\text{Retardation rate} = \frac{3}{4.25} = 0.71 \text{ m/s}^2
\]
b) Operator intervention. If the cableway safety system detects an abnormality it will apply the emergency brake. The operator may also notice the abnormality and simultaneously apply the service brake.

c) Safety System intervention. The operator may be applying the service brake as normal when an emergency brake is applied because an abnormal situation has been detected or there is a fault. Compounding of braking systems should be included in commissioning tests. Higher rates of retardation are inevitable in such circumstances but the levels attained should not present danger or damage equipment.

4) Degraded braking

Where a braking system has multiple units to provide the brake force, such as with a disc brake having several calipers or a drum with two brake paths operated upon simultaneously, consideration needs to be given to the failure of a critical component. In the case of a disc brake, it is usually regarded as the failure of a single caliper and in the case of a twin path drum brake, failure of one complete side. Degraded braking should be included in commissioning tests to establish the effect that this might produce. Lower rates of retardation are inevitable in such circumstances.

5) Friction drive systems

Where the rope drive is by means of friction a further very necessary consideration is the likelihood of the rope slipping on the drive sheave if the drive engine braking is excessive. The friction between the rope(s) and the drum can be affected by oils, grease, water, condensation and other contaminants. This may well be the critical factor in determining the braking levels to be employed.

R. Failures and the consequences of failure

1) All Cableway operators should critically assess their braking systems to ascertain the consequences of failure of a component. Such an assessment should be recorded. Where any component is considered to be a single line component serious consideration should be given to what can be done to eliminate this situation. A Single Line component is defined as one which, if it fails, will render the braking of the system ineffective. If a single line component is to remain in service it should be subject to routine critical examination (see section L).

2) Where a failure of a component would reduce the available braking but still give a reasonable level of braking this should be considered having regard to what would happen in such circumstances. If such a failure has the potential to cause a catastrophic event, it should be treated as if it were a single line component.

3) Most people do not expect their braking systems to fail but history shows that such events do happen. A very common type is a fatigue failure in a screw thread, as illustrated in Fig.14. Cracks that are a prelude to such failures are usually difficult to see and need non-destructive testing techniques to find them.

Fig 14 Fatigue fracture in a screw thread
4) Worm Gearboxes have several common failure modes. Fatigue failures of both input and output shafts are not unknown and, as the contact between worm and gearwheel is a sliding one, wear of both the worm and worm wheel is inevitable and can eventually lead to the stripping of the gears. Bearing failures can also impact on the efficiency of the braking provided. As mentioned in Section L, where braking is on a high speed shaft connected to the haulage drum by such a gearbox, it should be treated as part of the braking system.

5) Where components are keyed to shafts, there is a possibility that the key might work loose and either drop out or allow the component to come off the shaft. This is especially the case with gib head keys. If there is a possibility of a key dropping out then it should be provided with a means to prevent this.
Appendix A: Recommended Materials for Brakes

Note. BS 970 and BS 4360 have been superseded by European standards. However, BS 970 & BS 4360 are still in universal use and widely understood. These standards are used in the table below.

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
</tr>
</thead>
</table>
| Brake Paths                      | Brake paths can be made either by casting or forming from steel plate. Where possible, new brake paths should be made from castings.  
  Castings: Fine grained pearlitic grey cast iron from either BS 1561 EN-GJL-200 (old BS 1452 grade 14) or BS 1561 EN-GJL-250 (old BS 1452 Grade 17)  
  Formed Plate: BS 970 080M40 (EN.8) |
| Brake shoes and posts            | BS 4360 grade 43A                                                        |
| Anchor brackets                  | BS 4360 grades 43A, 43C, 50C or 50D  
  Cast anchor brackets are not recommended and any replacements should be fabricated from the above. |
| Brake shaft pedestals            | BS 4360 grades 43A, 43C, 50C or 50D  
  Cast brake shaft pedestals are not recommended and any replacements should be fabricated from the above. |
| Brake shafts                     | BS 970 grades 150M19 (En.14A) or 080M40 (En.8) in normalised or P condition |
| Brake levers, bell cranks and triangular levers | BS 4360 grades 43A, 43C, 50C or 50D |
| Spring rods, tie rods, turnbuckles and rod eye ends | BS 4360 grades 43A, 43C, 50C or 50D or BS 970 grade 150M19 (En.14A) in normalised or P condition |
| Pins                             | BS 4360 grades 43A, 43C, 50C or 50D or BS 970 grades 150M19 (En.14A) in normalised condition or 817M40 (En.24) in the T condition |
| General fabrications             | BS 4360 grades 43A, 43C, 50C or 50D                                      |
| Bushes (metallic)                | BS EN 1982 GC CC491K (BS 1400 LG2) or BS EN 1982 GC CC481K (BS1400 PB1) or SAE 660 |

end of appendix
Appendix B: Brake Testing

Commissioning

It is important that any commissioning testing is well thought through before any testing is done and the consequences of any failure considered. Destructive testing is not intended to be part of the process! Whenever a system is being tested it is necessary to build up confidence in the ability of the brakes and associated safety system to do their intended duty. Initial tests will be static to ensure that the brakes properly apply when required. A brake holding test is a fundamental first test and will establish that the brakes will (or will not) hold the full power of the motor or other prime mover. This test can even be carried out without any ropes or conveyances on the system.

Once the function of the brakes has been proved it is time to move to dynamic testing and some acceptable form of instrumentation is necessary for this. At a minimum, this instrumentation must be able to accurately measure speed against time but it is preferable that it can also measure distance. The instrumentation may be attached to the system at either the cableway drive engine or in a conveyance. It's calibration should always be checked.

Initial dynamic tests will be carried out at low speed, without any payload and well away from the stations. As confidence in the operation is gained the speeds can be increased up to the maximum operational speed. The results obtained need to be compared with the expected (design) figures at all stages. Any significant discrepancies should be investigated.

In carrying out dynamic tests the retardation rate recorded should be the average from the point at which the retardation commences to the point at which the cableway drive engine comes to rest. The moment when the brakes are tripped should not be taken as the commencement of retardation as there will inevitably be a time lag from initiation to application and this can be significant.

Once the empty system tests have been carried out and satisfactory results obtained it is the time to add a payload to the cars. It is usual for the payload to be based on a weight of 75 kg (12 stone) per person carried unless a specific weight is stated for the system. The load should be placed in one car. For convenience, it is usual to use a requisite number of 25 kg bags of sand or similar to simulate the payload but anything can be used, provided that the weight is accurately known.

Again, initial dynamic tests will be carried out at low speed to build up the necessary confidence and prove that the brakes are performing satisfactorily. The retardation tests should generally conclude with one from the maximum achievable speed under normal conditions. This is usually the overspeed situation. Where the system is a balanced one, with one car ascending whilst the other descends, the loaded tests should be carried out on both cars as a braking system may produce different results for different directions.

Degraded brake tests should also be carried out. Great care needs to be exercised when carrying out such tests and if (say) a brake disc caliper is isolated to allow this test to be done, a safe means of re-deploying it or another proven brake must be available if required.

In carrying out a series of dynamic tests such as these it is important to remember that the brakes will start to heat up and this rise in temperature should be regularly monitored. Testing should be put on hold if the brakes start to get too hot. It is also important to monitor the whole system. Ropes may be subject to excessive oscillation and possibly be displaced from their normal operational path. Return sheaves may also be subject to additional loads not seen in routine service. Testing should generally be kept to the minimum amount commensurate with proving their ability and suitability under the various conditions.

The tests should be designed to test each element of the braking system on its own as well as collectively, this will help determine the consequences of one part of a system falling as they may not provide equal contribution to the overall braking effect.

Routine Testing

It is only expected that commissioning tests are carried out before the system is first put to use or when significant system changes have been implemented. However, it is still necessary to routinely test the brakes to ensure their continued operational integrity. Such testing should be devised to show that this is the case and carried out at appropriate intervals.
Brake holding test
This is usually the simplest of all tests to carry out and, unless there is good reason, should be carried out on a daily basis, preferably at the start of the day's operations. It can also be done if there is any doubt as to the power of the brakes. It is usual to carry out such a test with the system in balance. To carry out such a test the service brake is fully applied and full power is then applied. The brakes should hold the full power of the motor but this may not always be the case on every system. The motor current that the brakes will hold should be recorded and any significant variation in this should be investigated.

Where the test described above cannot be achieved because of the control system, consideration should be given to the provision of a test switch to allow this to be done.

Dynamic testing
Dynamic testing should be carried out periodically. Whilst instrumented testing is to be preferred it is usually possible to carry out simple dynamic tests without recourse to instrumentation. For example, the empty system can be run at full speed and the emergency brakes tripped at a known point in the system. The position at which the system comes to rest can be recorded and any significant variation should be investigated. In its simplest form this test does rely on the engineman tripping the brakes at exactly the same point but the accuracy of this test can be improved by the provision of (say) a car activated switch which will ensure the brakes are always applied at the same point of travel.

Such a test as given above is simple to achieve and it is recommended that this is carried out at weekly intervals. On systems where the emergency brake does not routinely operate under dynamic conditions such testing helps keep the braking surfaces conditioned.

It is recommended that after any alteration to the brakes or following any adverse event, the brake hold and simple stopping distance test are carried out before resuming passenger carrying operations.

Periodic testing
The above tests are simple and can usually be done without any significant effort or organisation. However, it is still important to carry out more rigorous testing of the various brakes periodically. For engine brakes it should include loaded testing under normal operating conditions and where there is more than one braking system each system should be tested individually. It is not necessary to carry out a full suite of tests, as done when commissioning but it should be sufficient to prove that the brakes (and safety system) are functioning as intended. Usually, it is sufficient to carry out a couple of retardation tests (in both directions, load ascending and load descending), an initial one at slow speed to prove the level of braking and one at overspeed. This should be done with the test load and both cars should be tested. These tests should be done well away from the stations. To ensure that the results are comparable with previous tests they should be done under similar conditions. If the system is provided with a means for governing the speed at the stations the ability of the brakes to safely arrest the system in conjunction with the speed reduction should be tested. For this purpose it is necessary to introduce what is referred to as a false landing some reasonable distance from the stations. How this is achieved will be individual to each system.

Periodic testing should include dynamic testing of any car or other brakes to prove their continued integrity. Where, for example, a car brake is linked to the haulage rope and cannot practically be tested dynamically, consideration should be given to the carrying out of a simple pull test using a pul-lift or similar in conjunction with a load cell.

It is recommended that periodic testing is carried out at approximately six monthly intervals. Dynamic testing should be instrumented, preferably with an instrument that produces a print out of the results obtained.
Appendix C Sample brake calculations

Introduction
The following calculations are from an actual cableway and were carried out to determine the braking required so that the existing braking system could be replaced with a disc brake system. The system had an electric motor drive and consisted of two cars in balance and fitted with a balance rope. (see fig below) Prior to these tests no design information for the original braking system was available. To enable the calculations to be carried out various tests were done with the original braking system. The tests were carried out using a calibrated instrument capable of measuring speed against time.

System Inertia and system friction
These are two fundamental pieces of information necessary for carrying out any braking calculations. Various components go into making up the system inertia, including the mass of the cars and ropes. These figures are usually known but the rotating inertias of the motor, gears, drums, pulleys and rope rollers are much harder figures to establish, as is the system friction. However, it is generally possible to carry out some simple tests to establish these. These tests require a known load to be put in one of the cars and the cars placed at the point of balance, which is usually at the meetings. For the first test, the brakes were taken off and the rate of acceleration was measured. For the second test, the loaded car was driven up the gradient and power removed, the system slowing down under the influence of gravity and friction alone. The rate of deceleration was measured. This gave the following results:

Load in cars: 2.13 tonne
Gradient at meetings: 1 in 9
Acceleration under gravity 0.027 m/s²
Retardation under gravity 0.415 m/s²

Load causing accel’n/decel’n: 2.13 ÷ 9 = 0.237 tonnef ( = 2.324 kN)
Using force = mass × accel’n
2.324 kN – F = M × 0.027 m/s²
And:
2.324 kN + F = M × 0.415 m/s²

Where F = system friction
M = system inertia

Using simultaneous equations we get:

M = \frac{2 \times 2.324 \text{ kN}}{(0.027 + 0.415) \text{ m/s}²}

= 10.52 \text{ tonne (loaded)}

System inertia (empty) = 8.39 \text{ tonne}

And F = (10.52 \text{ tonne} \times 0.415 \text{ m/s²}) - 2.324 \text{ kN} = 2.042 \text{ kN}

System friction = 2.042 \text{ kN (} = 0.208 \text{ tonnef)}

Having established the fundamental parameters of the system, the next step is to establish a basic level of braking required. This value can then be used to choose the most suitable brake calipers for the duty.

Calculation to establish brake force required:

Assume a retardation requirement of 1 m/s² for a fully loaded car descending against an empty car ascending. This figure is chosen to give a comfortable and safe level of retardation.

Load in car: 2.13 \text{ tonne}
Weight of each car: 2.0 \text{ tonne}

With loaded car on steepest gradient (1 in 7) the empty car is on a 1 in 9 gradient

Let BF = brake force at the rope.

Out of balance force (OOB) = \frac{(2.13 + 2.0 - 2.0)}{7} \times \frac{9}{7} = 0.368 \text{ tonnef} ( = 3.608 \text{ kN})

Using force = mass × accel’n.
BF + Friction - OOB = System mass × retardation
BF + 2.042 \text{ kN} - 3.608 \text{ kN} = (8.39 + 2.13) \text{ tonne} \times 1.0 \text{ m/s}²
BF = ((8.39 + 2.13) \times 1.0) - 2.037 + 3.608

Brake force required at rope = 12.091 \text{ kN} (1.233 \text{ tonnef})

Check that the retardation rate for a loaded car ascending is acceptable.
\[ BF + \text{Friction} + \text{OOB} = \text{System mass} \times \text{retardation} \]

\[
\text{Retardation} = \frac{BF + \text{Friction} + \text{OOB}}{\text{System mass}}
\]

\[
= \frac{12.091 \text{ kN} + 2.037 \text{ kN} + 3.609 \text{ kN}}{8.39 \text{ tonne} + 2.13 \text{ tonne}} = 1.686 \text{ m/s}^2
\]

\[
\text{Retardation with load ascending} = 1.686 \text{ m/s}^2
\]

Therefore the limits of retardation under emergency braking with a brake force at the rope of 12.091 kN would be:

- 1.0 m/s²
- 1.686 m/s²

Which is satisfactory.

We next have to consider the effects of any compounding that might occur. Tests carried out to measure the retardation achieved under similar conditions with a full brake application of the motor shaft brake gave results that showed by similar calculation that a brake force at the rope of 12.852 kN was being achieved. With compounding we have to assume that both brakes are simultaneously applied. The worst case scenario is with a loaded car ascending and this is now considered.

Total brake force applied = 12.091 kN + 12.852 kN = 24.943 kN

\[
\text{BF + Friction + OOB} = \text{System mass} \times \text{retardation}
\]

\[
\text{Retardation} = \frac{BF + \text{Friction} + \text{OOB}}{\text{System mass}}
\]

\[
= \frac{24.943 \text{ kN} + 2.037 \text{ kN} + 3.609 \text{ kN}}{8.39 \text{ tonne} + 2.13 \text{ tonne}} = 2.907 \text{ m/s}^2
\]

This level of braking is considered to be acceptable.

Having established that the proposed braking levels will provide safe levels of braking the brake calipers can now be selected. The brake disc has an effective diameter of 2.480 m and the surge wheel an effective diameter of 2.44 m. Two brake calipers were selected from the manufacturer's range, each providing an effective brake force of 6.82 kN at the disc. This equated to a total brake force of 13.82 kN at the rope.

13.82 kN is slightly more than we have so far used in our calculations so it is necessary to re-calculate the theoretical retardations using this figure.

Again, using force = mass x acceleration, we get:
Load descending:
Retardation = BF + Friction - OOB

\[ = 13.82 \text{ kN} + 2.037 \text{ kN} - 3.608 \text{ kN} \]
\[ = 8.39 \text{ tonne} + 2.13 \text{ tonne} \]
\[ = 1.164 \text{ m/s}^2 \]

Load ascending:
Retardation = BF + Friction + OOB

\[ = 13.82 \text{ kN} + 2.037 \text{ kN} + 3.608 \text{ kN} \]
\[ = 8.39 \text{ tonne} + 2.13 \text{ tonne} \]
\[ = 1.850 \text{ m/s}^2 \]

Therefore the limits of retardation under emergency braking with a brake force at the rope of 13.82 kN would be:

1.164 m/s\(^2\) and 1.850 m/s\(^2\)

Both these results are satisfactory

Considering compounding:
Load descending:
Retardation = BF + Friction - OOB

\[ = 13.82 \text{ kN} + 12.852 \text{ kN} + 2.037 \text{ kN} - 3.608 \text{ kN} \]
\[ = 8.39 \text{ tonne} + 2.13 \text{ tonne} \]
\[ = 2.386 \text{ m/s}^2 \]

Load ascending:
Retardation = BF + Friction + OOB

\[ = 13.82 \text{ kN} + 12.852 \text{ kN} + 2.037 \text{ kN} + 3.608 \text{ kN} \]
\[ = 8.39 \text{ tonne} + 2.13 \text{ tonne} \]
\[ = 3.072 \text{ m/s}^2 \]

Although the retardation with load ascending is marginally higher than recommended both these results are satisfactory.
Finally, we should consider the situation of the complete failure of a brake caliper as it is the disc brake that is operated by the safety circuit. One caliper acting on its own will produce 6.91 kN of braking at the rope. The worst case scenario is that of a full load descending and in these circumstance we will have:

Load descending:

\[
\text{Retardation} = BF + \text{Friction} - \text{OOB} \\
\text{System mass}
\]

\[
= \frac{6.91 \text{ kN} + 2.037 \text{ kN} - 3.608 \text{ kN}}{8.39 \text{ tonne} + 2.13 \text{ tonne}} \\
= 0.501 \text{ m/s}^2
\]

Whilst this does produce a reasonable retardation and will stop the system it needs to be considered in conjunction with the terminal conditions, such as enforced speed reduction (slow banking), overrun clearance and retarders, which are outside the scope of this Guidance Note.

**Friction Drive Systems**

A slightly different approach needs to be considered for friction drive systems as the braking forces should not be high enough to induce rope slip.

The point at which slip will occur is governed by the relationship between the different tensions in the two ropes, the coefficient of friction between the rope and drive wheel and the total angle of lap. This relationship comes down to:

\[
T_1 = e^{\mu \phi} \quad \text{and, as long as} \quad \frac{T_1}{T_2} \leq e^{\mu \phi} \quad \text{the ropes should not slip.}
\]

Where:

- \( T_1 \) is the higher rope tension
- \( T_2 \) is the lower rope tension
- \( e \) is 2.7183
- \( \mu \) is the coefficient of friction between the rope and drive wheel
- \( \phi \) is the angle of lap of the rope measured in radians

In the case of the system being used as an example the rope goes round the drive wheel 2½ times, which is 900° or 15.7 radians (1 radian is 57.3°) and we will take the coefficient of friction as 0.16

\[
e^{\mu \phi} = 2.7183^{0.16 \times 15.7} = 12.33
\]

It is usual to ignore friction in this calculation and the additional load due to the weight of rope is negligible so this, too, will be ignored. (On longer systems it may be necessary to consider this.)

\( T_1 \) is the load in the rope on the high tension side which is the sum of

- the gross vehicle load divided by the gradient
To which must be added the force to retard the load \( = (2.13 + 2.0) \times R \text{kN} \)

T2 is the load on the rope on the low tension side which is the sum of

- half of the 1.0 tonne return wheel tension weight
  \( = 0.5 \text{tonne} \)
  \( = 4.904 \text{kN} \)

10.69 kN

To which must be subtracted the force to retard the load \( = (2.0) \times R \text{kN} \)

As long as \( T1/T2 \) is less than \( e^{\mu\varnothing} \) the rope(s) will not slip

At the limiting retardation rate

\[
T1 = e^{\mu\varnothing} \frac{T2}{T2}
\]

From the above we can rearrange the equations to get:

\[
R = \frac{(e^{\mu\varnothing} \times 7.082) - 10.69}{(e^{\mu\varnothing} \times 2.0) + (2.13 + 2.0)}
\]

\[
R = \frac{(12.33 \times 7.082) - 10.59}{(12.33 \times 2.0) + 4.13} = 76.73 \div 28.79
\]

\( = 2.665 \text{ m/s}^2 \)

This is higher than the calculated braking rate achieved under brake compounding (2.386 m/s²) so rope slip should not occur under any likely conditions of operation.

end of appendices