Office of the Organising Secretary, IMHC
P O Box 2963, FOURWAYS 2055, Rep. of South Africa
Telephone and Fax:+27 (0) 11 888-7163
web site : http://www.cmasa.co.za
e-mail : cma@pixie.co.za

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Controlling Belt Slip

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Authors: Mr. Paul Nel, Mr. Graham Shortt
AATS, RSA
Presenter: Mr. Ernst Dreyer
AATS, RSA

The SA Institution of Mechanical Engineering

SA Institute of Materials Handling

Beltcon 10 – Controlling belt slip

Authors:- Paul Nel , Graham Shortt – AATS
Presenter:- Ernst Dreyer – AATS

1. INTRODUCTION.

Slippage of conveyor belts at the drive is not an uncommon problem. It may become even a larger problem where there is natural water, clays or any other factor that may influence the friction factor between the belt and its drive pulleys. The purpose of this paper is to highlight possible causes of such friction losses. Possible solutions to the problem will be considered and also an attempt to highlight legal guidelines governing conveyor installations in terms of safety will be made.

2. BELT SLIP.

Overall productivity is likely to be negatively affected by unscheduled conveyor stoppages. In particular, belt slip in collieries is considered in a very serious light. When combined with coal dust or duff build-up, which may easily occur in most conveyor installations, belt slip may lead to the duff igniting and leading to severe fires. In one instance some years ago, the duff at the tail end of the conveyor ignited, and, since the conveyor was installed in a totally enclosed gantry, the chimney effect rapidly drew the fire all the way up the gantry, totally destroying the conveyor. Many of the reported stoppages of plant conveyors can be directly attributed to tension related problems.

Some installations can better absorb these tension related problems than others. The causes of slippage are often not clearly identified and the proposed solution is seldom the correct action to take. It may be clear that the authors are referring to the manipulation of the take-up mass. Over the recent past, in every single instance where a client has complained about belt slippage, the request from the client was to check the correctness of the calculated counterweight mass. To increase tension by simply adding take-up mass, as a short-term solution, is probably not entirely an inappropriate action to take. Unfortunately, counterweight mass is regularly added without considering the possible affect it may have on pulley shafts, the efficiency of the belt splices and the transition distances, not to mention the conveyor belting itself.

To cater for rapid advancement in underground systems, multiple storage loop systems are becoming a more common occurrence. In this instance it is important to maintain the correct loop tension in order to prevent or reduce problems caused by slip, belt misalignment, failed joints or spillage.

3. THE BASICS.

For the purpose of this paper one should reconsider the basics. The term T_1 refers to the tension existing in the belt immediately ahead of the drive mechanism. T_2 denotes the tension in the return strand of the conveyor belt between the drive mechanism and the take-up. The following well known formula for the static tension applies.

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

Where
- T_1 = tension in the belt immediately ahead of the drive mechanism (kN).
- T_2 = tension between the drive and the take-up (kN).
- e = the base of Naperian logarithms.
- μ = Coefficient of friction between drive drum and belt. (to be discussed later).
- θ = angle of wrap at the drive pulley (radians).

When tension is applied to a conveyor belt, it will stretch and this stretch will reach a maximum when the belt is under full-load acceleration. To absorb the stretched length of belting and to maintain the ratio of the equation mentioned above, it is important to have a take-up mass and counterweight carriage that can easily comply with the existing requirement, be it a start-up or running condition. This is so that the minimum T_2 tension will be maintained for the loading condition and the conveyor will move without the drive pulleys slipping underneath the belt. The worst possible condition would be if the conveyor is loaded to capacity, stopped and restarted. Under this condition the most stretch will be generated and the counterweight mass will fall to its lowest position. Starting a loaded belt will cause the belt to stretch, possibly accelerating it above the normal operating speed of the drive unit until, due to falling drive tension requirements, it again slows down. The sequence of acceleration and retardation is often repeated until the belt and its load is moving at its constant design speed. Belt surges as a result of dynamics occur to some extent on all conveyors but is more noticeable on long belts. Constant changes in belt speed and stretch during the start-up sequence may cause high shock loads to the belt, splices and the support structure due to the counterweight mass continuously attempting to catch up with the changing conditions. The above is often the basic cause of dynamic misbehaviour.

Continuing then, with this well-known relationship, it is clear that the maintenance of this tension relationship at the drive station, throughout the transient states of the conveyor, whether accelerating, decelerating or subject to a varying load, is of paramount importance. The establishment of the relationship between the tensions T_1 and T_2 implies that these tensions are known. The basic tension that is known, or at least that can be determined with some degree of accuracy in a conveyor system will be the *effective* tension T_e, established by standard methods, such as ISO 5048, DIN 22112, CEMA or any one of the industry accepted methods. The basic relationships are then established by the knowledge of the angle of wrap and the coefficient of friction between the belt and the driving pulleys.

This is where the system becomes subjective. The coefficient of friction between the belt and the driving pulley is found from field testing over many installations. The value of the coefficient can therefore vary considerably, and may even vary on the same installation, depending on environmental factors, such as the presence of spillage, water or slime, or even the state of repair (or disrepair!) of the pulley lagging itself. The quality of the belting covers also plays a major role here. The harder compounds used for the more abrasive resistant belting could lead to a greater tendency to slip, while a softer compound on the pulley lagging could easily lead to rapid lagging pattern wear and the attendant change in characteristics.

The standard values of the coefficient of friction have been listed by most conveyor manuals and belting manufacturers. Common values used are as follows.

Condition	Plain rubber Lagging	Grooved Rubber Lagging	Ceramic Lagging
Dry	0,35 to 0,40	0,40 to 0,45	0,74 to 0,83
Wet	0,35	0,35	0,48 to 0,78
Wet with wet material	0,25 to 0,30	0,18 to 0,22	0,42 to 0,51

The drive arrangement has a marked effect on the overall tension relationship. Of course, the given relationship $\frac{T_1}{T_2} = e^{\mu\theta}$ is based on a single pulley drive. For the case where the drive is dual independent, the basic relationship between the tensions T_1 and T_2 is dependent on the power, or tension, split between the two drive pulleys. The common practice is to arrange the power split in accordance with standardised power packs. In the past, and still in some instances today, the power split was based on different motor powers. For example, a system with a 110 kW power pack on the primary and a 55 kW motor on the secondary drive is listed as a drive with a ratio of 2:1. However, the actual speed of the motors differs (sometimes quite markedly), which puts an additional strain on the fluid couplings (where these are fitted), with one of the units having to cater for a greater degree of slip than the other. On the other hand, even with electronic control, there will always be that small differential in pulley rotational speed which leads to premature lagging wear and considerable motor imbalance in the system.

The contention is, then, that the power packs should always be identical and interchangeable, in order to obviate this potential imbalance. Purely from a maintenance point of view, it is far better to provide the conveyor with the best possible environment, to achieve the best possible results.

To simplify the determination of the tension ratios, it is common practice to use the so-called wrap factor, which brings the relationship between T_1 and T_2 into account. The wrap factor is well-documented in publications such as CEMA and a simple derivation follows.

From $\frac{T_1}{T_2} = e^{\mu\theta}$ and knowing that $T_1 = T_2 + T_e$, it follows that $T_2 + T_e = T_2 \cdot e^{\mu\theta}$

and $T_e = T_2\left(e^{\mu\theta} - 1\right)$, from which $\frac{T_2}{T_e} = \frac{1}{\left(e^{\mu\theta} - 1\right)} = K$, the so-called wrap factor. The determination of T_2 for a single pulley drive is then the simple multiplication of T_e by this wrap factor. Tables of wrap factors are also available and it is a simple expedient to create your own, by substituting values for μ and θ across the range of coefficients and wraps likely to be encountered. Of course, the increase in T_e during starting of the system must be taken into account. The most likely time for a belt to slip is during start-up of a loaded belt and the increase in T_e can be estimated from the starting factor of the accelerating equipment.

Typical starting factors for fluid couplings may be listed as follows.

Coupling Type	Starting Factor
Traction	1,6 to 1,8
Delay Fill Chamber	1,5
Double Delay Fill chamber	1,4
Turbo Soft Start	1,2 to 1,3

The starting factor for Variable Frequency Drives (VFD's) can be as low as 1,05, if the system is designed carefully. The lower the starting factor, the longer the acceleration period will be and the less likelihood there will be for the belt to slip around the driving pulleys. However, limitations of the motors should be taken into account, since the motor will be under accelerating conditions for an extended period and this may have an adverse effect on the temperature of the motor.

It is good practice to allow a small safety margin when specifying the wrap factor. For example, if the wrap on the drive pulley is determined at 195°, use the wrap factor for 190°, to give some leeway for indeterminate coefficients of friction.

When we are faced with dual independent drives, the overall wrap factor changes as a result of the effects of the dual drive. Dual independent drives can have a ratio of 1:1 or

2:1, based on the premise that the power packs should all be the same size and have the same characteristics. The drive ratio refers to the number of power packs per pulley. When the primary drive is equipped with two power packs and the secondary drive with one power pack, that refers to a 2:1 drive. It is important to note that the secondary drive refers to the unit that feeds the take-up. This is important when starting sequences are determined.

Thus, for dual independent drives, the wrap factor K is directly proportional to the drive ratio. For example, if we consider a dual independent drive with two drives having 200° of wrap each. The wrap factor for a single pulley drive, with μ = 0,35 would be $K_n = \frac{1}{\left(e^{(0,35 \times 3,491)} - 1\right)} = 0,418$. For a dual independent drive with 1:1 ratio, the value of the wrap factor would become $K_{1:1} = \frac{K_n}{(1+1)} = 0,209$ Similarly, for a 2:1 drive, the value of the wrap factor would become $K_{2:1} = \frac{K_n}{(2+1)} = 0,139$. If the system has an effective tension T_e of 100 kN for our example, that would give us a value of T_2 as follows

Drive ratio	Wrap Factor	T_2
Single	0,418	41,8 kN
Dual 1:1	0,209	20,9 kN
Dual 2:1	0,138	13,9 kN
Dual 1:2	0,278	27,8 kN

From this the benefit to the system overall tensions, when higher drive ratios are utilised, can be seen. The effect that incorrectly specifying the location of the primary and secondary drive pulleys can have on the system is also demonstrated. Where we have a 2:1 drive ratio and the power packs are installed with 2 drives on the secondary and one on the primary drive, instead of vice-versa, it can be seen that the effect is to *double* the outbye tension requirement, when compared to a 2:1 arrangement. When the drive has been designed as 2:1 and the take-up arranged accordingly, we can see that the 1:2 arrangement (literally 0,5:1) is most likely to slip severely. in this case, the wrap factor would be determined as $K_{1:2} = \frac{K_n}{(0,5+1)} = 0,278$, as indicated in the table above.

4. GRAVITY TYPE TAKE-UP.

Under these conditions it is, however, still beneficial to use a gravity type take-up since it responds rapidly and is simple to use and maintain. The disadvantage of a gravity type take-up in a multi-lap system is that the take-up tension applied must compensate for the belt tension at several pulleys and therefore it is likely that the counterweight mass will be high. Whenever belt splicing is done, the counterweight mass must be made secure to allow a safe working environment. This could be a difficult task if the mass is around 15t. Finally, a large counterweight mass will require more than an average structure and may therefore prove to be more expensive than a fixed tension alternative.

5. FIXED TENSION SYSTEMS.

Especially in underground conveyors it has become popular to use an electric winch in the place of a gravity type take-up. This is not new technology and has been used for some time. The manner in which tension is applied however is still a subject of discussion. When using a winch instead of gravity, it is important to note that the tension settings must be such that they compensate for the start-up condition. This is so the winch rope will not slacken and cause the T_2 value to drop until the drive ratio can no longer be maintained, which implies that the belt is slipping. A common method of operation is to apply tension at the winch to satisfy both starting and running conditions and to lock the winch at this setting. While the belt will more than likely operate without problems, satisfying in particular the client, when the belt eventually stops, at the end of a shift, say, standing tensions will equalise, causing excessive tension in the take-up and drive areas. One has therefore to keep in mind that if the conveyor is not running, there will be no load and therefore little stretch in the belt resulting in unnecessarily high standing tensions. This is not a desirable situation since even though the conveyor is at rest, the belt will still be under its operating tension. Tension in the winch should be reduced but not below the original T_2 value. This is in order to accommodate the tension required to ensure that the holdback will still be effective.

The recommended method of operating a winch type take-up is therefore to apply sufficient T_2 tension to allow for some slack belt to come into the take-up system. At this point adequate T_2 tension must remain at the drive to provide drive traction and for the belt not to slip at the drive pulleys. The belt should be pre-tensioned to the required start-up tension and locked in position, where after the belt is started. The tension then drops naturally to an operating tension. Therefore, ideally, an optimum tension band should be set ranging from a minimum tension to prevent slip, to a maximum level to prevent high tensions when running empty. If the system was controlled via a PLC and load cells, a tension window could be developed to suit the criteria mentioned above. A common mode of operation is to allow the winch to take up slack belt into the take-up when T_2 is dropping, thereby providing drive traction. Having explained about belt surge earlier however this may cause the winch to hunt, releasing and applying tension continuously. The mentioned winch operating window will prevent this.

Commissioning a system such as this often requires some patience since the tension settings provided by the design office can only serve as an indication and can be used as starting points only. Actual conditions on site are often somewhat different to that which the design engineer had in mind. Careful commissioning of such a system is however essential since the efficient operation of the conveyor depends on it. In this instance it is also important to note that the load cell should be situated such that up to date information is relayed to the PLC which in turn will tell the winch to apply or release tension. Needless to say, the tensions must be carefully calculated to suit the selected take-up arrangement, the number of ropes, the number of falls of rope and the number of sheaves. A method of protecting the system against over-tension is recommended here. The protection override should trip the winch and therefore the conveyor, thereby protecting the belt.

Modern advances in winch technology, especially in the field of eddy-current controlled winches has created almost a form of artificial gravity. These winches apply a controlled constant tension to the take-up system and, as a result of the nature of the eddy current coupling, will allow over-tension payout and under-tension winching in an automatic manner. This allows the designer to specify the winches which will act similarly to gravity towers, without the space requirements. Of course, the winch response time is a function of the size of the unit, the amount of pretension specified and the number of falls of rope. For high tension units, multiple falls of rope will imply several rope sheaves and the efficiency (or drag) of these must be taken into account when the units are specified. For these reasons, the specification of eddy-current type winches ought to be delegated to the specialist suppliers.

Like all mechanical equipment, the care and regular maintenance of the machine will affect the performance and reliability of the system. The major advantage of these winches (apart from the size) is the facility to provide a controlled payout of rope in over-tension conditions. In many instances, then, the eddy-current type take-up winch may be preferable to the fixed tension system. Disadvantages of the constant tension type winch installations include the continuous draw of current, since the majority of these units operate at about 55 kW. This must be seen as a continuous operating cost and could influence their selection when compared to fixed tension systems and gravity, which is, of course, free and extremely reliable.

6. SOFT-START DEVICES.

With the arrival of variable frequency drives one need hardly concern oneself with slip at the drive due to an inefficient or direct on line start. However, it is not every conveyor which qualifies to have a VFD device and often it is far less expensive to simply use a fixed-fill fluid traction coupling or a delay fill coupling, particularly with conveyors of lower power. Depending on design parameters, conveyor start-up times vary and fluid coupling fill will vary accordingly. Therefore, if there is too much oil in the coupling the start-up factor will be increased. If torque is applied too quickly at the drive pulley, the belt will slip. This can be compared to pulling away in a motor vehicle. If the clutch is released too quickly and torque applied too rapidly, the driving wheels will spin. However, if good tyres are used and careful driving habits are employed, the likelihood of the wheels spinning will be reduced. So too with belt conveyors.

7. LAGGING.

Like good tyres, the lagging on a pulley can be designed to optimise traction between the pulley and the belt. Originally the quest for better lagging was not to improve the friction factor but to extend the life of the lagging. Minimising slippage is however of obvious importance since this is one of the major causes of reduced belt life and lagging wear. Whenever the lagging from the pulley slips against the rubber belt cover, material is removed from both surfaces. The lagging will, however, wear sooner because of its shore hardness and also because of its comparatively small surface area, when compared to the belt surface. The friction factor is often reduced by the addition of water or fines between the surface of the belt and the pulley. It is therefore important that the belt is safeguarded against these influences in the drive area, implying that belt cleaners should be accurately adjusted and maintained. The practise of adding some abrasive material to the surface of the drive pulley in order to assist the belt in empty belt start-up is also not recommended since this accelerates lagging and belt wear even further.

If traction is a problem due to constantly wet or muddy conditions at the drive, it may be worth considering an investment in some form of ceramic lagging. As can be seen in the table in paragraph 3, this product increases the friction factor μ between the belt and the pulley substantially. We are aware that at this time ceramic type lagging is more expensive than regular rubber lagging, but the advantages of this product and the reduction in possible downtime has to be seen in perspective. The wear on ceramic lagging has not been accurately documented yet, since it has not been around long enough. What is known, is that ceramic lagging does not require replacement as frequently as rubber lagging. However, the effect of the lagging on the wear of the belting is not yet sufficiently documented. Ceramic lagging is not a miracle product which will immediately solve all traction problems. It is however a viable alternative to increasing belt tensions to reduce slippage, which can lead to damaged pulleys, shortened bearing life, failed splices and broken belts, caused by over-stressing the system. Each and every traction problem should be analysed and the most suitable solution applied.

8. POWERPACKS.

If the normal sequence of events during the start-up phase of a conveyor are considered, it will be seen what often happens is that the drive may begin to over-speed and trip the system. This can be described as an aborted start. However, shortly after, another start-up will be attempted. Several start-up attempts in short succession accelerate wear on the power pack components since every start-up attempt increases the system temperature, sometimes substantially and there are seldom sufficient cooling time delays built into the control circuitry. The reason for the system drive over-speeding may be due to the lack of the correct drive tension configuration as shown above and associated loss of tension. Thankfully some larger systems will actually shut down if a certain temperature is reached, allowing the equipment to cool down prior to damage being caused, adding to greater production downtime in the short term but protecting the equipment in the long term.

9. ANGLE OF WRAP AND START-UP SEQUENCE.

As shown in the example in paragraph 3 above, with single drive pulleys, the angle of wrap could be considered of lesser importance, provided that the belt and the drive arrangement is designed according to the selected angle of wrap. Obviously it is still important from the point of view of selecting a belt class. In dual drive arrangements however this figure is of far greater importance. If a dual drive pulley arrangement is used it is important to note that the angle of wrap at the primary drive and that at the secondary drive should be similar. It is important to ensure that the primary drive does not feed slack belt into the secondary drive. On the other hand it is important so that the secondary drive does not attempt to pull the entire load while the belt is slipping at the primary drive. Though not a hard and fast rule, the drive nearest to the take-up is normally considered to be the secondary drive. This is the power pack that should be started first, followed by the power packs on the primary drive pulley. From a power pack load sharing point of view it is therefore important that angles of wrap of primary and secondary drives are similar. There are also other factors that may affect load sharing, such as the incorrect or uneven oil fill in the various fluid couplings, variations in pulley diameter or the state of the pulley lagging.

10. HOLDBACKS.

For a holdback to work it is imperative that T_2 tension be maintained at the pulley where the holdback is installed. If a conveyor with a dual drive arrangement should run back and for some reason belt slip occurs over the pulley containing a low-speed holdback, it may end up driving the other drives in reverse. Consider that inertia from the motor is still turning the input wheel of the fluid coupling in a forward direction and runback is turning the output wheel in the opposite direction, through a reducer which has now become an accelerator. It is likely that the coupling will be accelerated beyond its critical speed. Considering that the fluid coupling normally has an aluminium casing, centrifugal forces induced by the oil inside the coupling can cause the casing to disintegrate, at great danger to anybody and anything in the vicinity. From this point of view, it is imperative that the person responsible for maintenance ensures that the take-up carriage is moving freely and that the take-up rope and counterweight mass movement is not in any way restricted. In the case of high-speed holdbacks fitted inside the reducer, this danger is largely eliminated, but, depending on the holdback torque required, this might cause damage to the reducer. For this reason, if high-speed holdbacks are used inside reducers then the holdback in each reducer should be capable of the total calculated runback torque. This, of course impacts on the ability of the reducer itself to withstand the reverse torque applied in a holdback situation.

11. CURRENT LEGAL REQUIREMENTS.

In the search for legal guidelines or stipulations, it is surprising to see how little is said about conveyors in general, and the issue of possible belt slip in particular, in the various statutory requirements governing mining and safety. If we should recall some of the more common reasons for possible slip at the drive:-

a. Insufficient belt tension at the take-up.
b. Insufficient angle of wrap at the drive pulleys.
c. Worn lagging or a decreased friction factor.
d. Incorrect start-up sequence.

Not a single of the mentioned factors is addressed in any of the well known legal guidelines. This raises certain questions. Consider the following:-

a. If there is not sufficient belt tension at the drive and the belt slips, causing damage to equipment or, even worse, an accident causing injury or loss of life, who would be held responsible?

   (i) The design engineer. The system would probably have been designed in accordance with given design parameters as laid out by the client. Operational personnel would have been informed by means of drawing or procedures (or both) and physical signs of how the equipment is to be used or not used.

   (ii) The contractor that installed the system. Perhaps, if the system was not installed in accordance with the mechanical design.

   (iii) It must be established who would be responsible for ensuring that the system is correctly installed in the first place. Naturally, some of the issues would depend on the contract agreement and this could lead to extended legalities.

   (iv) The Client. Is the client liable for not operating the system in accordance with design parameters and therefore nullifying the original mechanical design? If we bear in mind that a design engineer will only accept responsibility for the design if that machine was operated within its original design parameters. Certain safety factors are built into the designs to protect the machine and its operators but it is impractical and expensive to continuously design for worst case operational practices. Operators should therefore make an effort to be educated in the inherent dangers and hazards of operating conveyors outside of their stipulated design parameters.

b. Slip at the drives due to insufficient wrap at the drive pulleys could easily be blamed on the design engineer. However, what angle of wrap can be regarded as insufficient? If the system was designed for a drive angle of wrap of 180°, and the balance of the drive system, including the take-up, designed in accordance, the conveyor will run without problems. The question as to whether installation was done correctly can then be asked.

c. Worn lagging or a decreased friction factor can only be blamed on lack of maintenance or inappropriate operation by the mine. If the design engineer has designed the system for a factor of say 0,35, which as discussed earlier is fairly common, he does not expect a wet and muddy drive pulley, reducing the actual friction factor to 0,25.

d. The start-up sequence is without a doubt a very important factor to consider. It would be the responsibility of the contractor and the engineer in charge of commissioning to set the start-up sequence in such a way that it is in accordance with the requirements as set out in the mechanical designs. A common problem is that in a dual pulley drive system, the drive nearest to the take-up, which is designated the "secondary" drive, may be arranged on site so that it does not start first. Depending on the installation, this could cause various kinds of problems such as dynamic misbehaviour or slip under start-up at the drive nearest to the take-up since the other pulley (the primary drive) will be feeding slack belt towards it. In this instance the question that must be asked is related to the adequacy of the original design for the required application. If the system is adequately designed, then the responsibility of the failure moves to the contractor responsible for the installation of the system. If the contractor can show that the installation was in accordance with the original designs, then the responsibility moves to the client and the method in which the system was operated.

We can therefore see that there could be considerable speculation with regard to the culpability or liability if the belt slips at the drive. Would it not be to the benefit of the industry as a whole if guidelines were laid down governing the design, the installation, commissioning and operation of the drive and take-up system of a conveyor? It may well be that these guidelines already exist in various forms and we therefore wish to refer to the first page of ISO5048. "Certain conveyors present more complicated problems, for example those with multiple drives, or with undulating profile in vertical elevation. For these calculations which are not covered in this International Standard, it is advisable to consult a competent expert." Anybody that designs in accordance with ISO 5048 will more than likely produce more than adequate designs, but should those designs fail, it is too easy to bring the question of 'a competent expert' into the argument.

12. CONCLUSION.

It must be noted that the existing guidelines are not being criticised. The manner in which engineers in this country can design materials handling systems, sometimes to their own standards, and not be held legally accountable when things go wrong is being questioned. The Mine Health And Safety Act, No 29 of 1996, governs many aspects but is one step removed from the technical guidelines required to ensure that designers of conveyors across the country design machines to a common minimum standard.

From the above it is clear that in the unfortunate event of an accident caused by belt slip at the drives, attempting to point out a responsible party could easily turn into a witch-hunt. In terms of professional liability, loopholes regarding accountability should be plugged. It would therefore be recommended that together, the industry should establish guidelines or a general Code of Practice covering design and operation of conveyors and in conjunction with the Inspectorate of Machinery, make compliance with such guidelines or code compulsory. Design codes are generally available, as mentioned previously, but Codes of Practice are not.

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