Underwind and overwind protection systems with enhanced self-sufficiency

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Synopsis

Underwind and overwind protection system concepts for mine hoist shafts were developed in conjunction with and for the Safety in Mines Research Advisory Committee (SIMRAC). End-of-wind operation of mine hoists is the most hazardous aspect of mine hoisting, carrying the highest risk of loss of life, injuries to workers and loss of production.

Systems operating mechanically and with local actuation were developed. This was done in order to remove the increased risk associated with dependency on remote or external energy and information supply associated with most existing protection systems.

The proposed underwind protection system concept absorbs the energy of motion of a conveyance overrunning the design lower limit of travel by drawing a metal strip through a set of rollers. This action causes dynamic cyclic plastic bending of the strip material that converts the kinetic and potential energy of the conveyance into strain energy of the metal. In the concept design a pair of steel wire rope slings attached to the strips catch the overrunning conveyance and transfer the retardation force.

The proposed overwind protection system concept absorbs the energy of motion of a conveyance overrunning the design upper limit of travel by early detaching of the conveyance from the hoist rope. This detaching is carried out via an additional detaching hook activation mechanism fitted at a sufficient height below the spectacle plate to allow the conveyance to retard to standstill under gravity before it would crash into the spectacle plate. The conveyance so brought to rest is prevented from falling by means of jack catches on the conveyance interacting with a rack (toothed profile) fitted on the guide rails in the retardation zone.

Scale models (1:10 scale) of both protection systems were designed to conform to established retardation standards. The retardation standards were limited by the requirement of passenger safety in retarding cages. The models were then built and tested in a 1:10 scale shaft model. Retardation performance close to the required levels was achieved. The retardation distance required should allow such systems to be retrofitted in most existing shafts. Given the simplicity and robustness of the designs, further development was recommended because of their enhanced self-sufficiency and reduced risk of malfunction.

Keywords: underwind, overwind, deceleration, mineshafts, hoisting.

Introduction

Mine hoist operation carries particular risk of crash-type accidents at the upper and lower extremities of the hoisting range. Furthermore, a number of factors combine to increase the severity of the consequences that such an accident could have. Among these are the large number of workers carried by cages and the high load of ore carried by skips, large number of hoists per day, long hoisting distance, high hoisting speed, the fact that a number of conveyances are operated in a shaft, and the dependency of mining operations on the shaft.

An underwind incident occurs when the conveyance overruns the design lower operational limit and is at risk of crashing into the shaft bottom. An overwind incident occurs when the conveyance overruns the design upper limit of travel and is at risk of crashing into the headgear. These events may be caused by the failure of the braking or control system and may have catastrophic consequences with regard to the loss of human life and/or mine capital equipment and production.

In a sample of South African mines, predominantly in the North West region, for the period January 1988 to July 1995, 46 incidents were found[1]. In three incidents in 1973, 1991 and 1997 fatalities numbered 16, 7 and 13 respectively[1]. Because of this risk and of the history of incidents of this nature[1], the Safety in Mines Research Advisory Committee (SIMRAC) commissioned a study on additional end-of-wind protection systems.

As workers represent a much more fragile load than material, the design retardation rates were restricted by human tolerance. To prevent injuries to people transported in conveyances, the maximum deceleration for the underwind system is 2.5 g (24.5 m/s^2) and for the overwind system 1 g (9.8 m/s^2)[2]. According to the British Columbia legislation referred to here, this is deemed a 'walking away safe'

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deceleration. This legislation on the deceleration value is not included in the South African Mines and Safety Act (29/1996). There is a directive C2 to all mining operations requiring maximum deceleration values of 2.4 to 2.7 m/s^2 for mine winders and 2.7 to 2.9 m/s^2 for rock winders. This is, however, not enforced.

Mine shafts are already equipped with a number of safety systems of which the Lilly speed regulator is one, but accidents still occur. The cause of such accidents can often be traced back to remote- or event-related impairment of signals and energy supply to the primary safety device. Therefore a primary requirement for the additional protection systems was that they should be independent of the existing systems. They preferably had to be mechanical devices because of the risk that electrical or hydraulic energy and signals could be compromised by the primary cause of failure. The study focused on vertical shafts as these installations present the more stringent requirements. The solutions, however, were required to be adaptable to inclined shafts.

Different concepts were generated for the protection systems. The concept generation process was informed by a literature and product survey conducted to establish the nature and causes of end-of-wind incidents and to review available technology for end-of-wind devices. This study was complemented by a user needs survey and study of existing installations. Specifications were compiled with the aid of a functional analysis study according to standard design procedure[3]. The concepts generated were evaluated against the system requirements.

Scaled-down models of the selected concepts for both protection systems were designed, built and installed for testing in the 1:10 scale shaft model at the Department of Mechanical and Aeronautical Engineering, University of Pretoria.

Underwind

The primary design parameters for the underwind protection device were:

- The maximum deceleration to prevent injuries to people transported in the conveyance was taken as 2.5 *g* (24.5 m/s^2) as explained above.
- The maximum speed of the conveyance is 18 m/s. The system had to provide for underwind situations but was not required to provide for free fall.
- After stopping the conveyance, the conveyance had to be prevented from falling down the shaft.
- After retardation the occupants of the conveyance had to be able to exit with reasonable ease. No permanent distortion of the conveyance should occur.
- The system preferably had to be a mechanical device. External sources of energy or signals supplied via electricity or hydraulics could be affected by the primary cause of failure.
- The system had to have low maintenance and inspection requirements due to difficulty in gaining access to the equipment.

Apart from the concept selected for the underwind system, other concepts generated and evaluated during the design process included spring damper systems, oleo buffers and the Technogrid[4] energy absorption system.

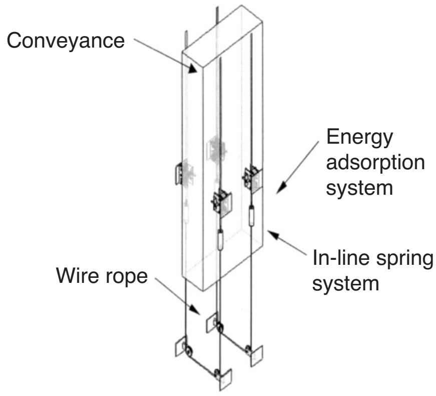

Figure 1—Typical shaft layout of underwind system

In the selected concept the conveyance runs into steel wire ropes strung across the shaft in the section that the conveyance would traverse were it to exceed the lower limits of travel. The steel wire ropes are attached to metal strips that are deformed plastically by being extruded through roller assemblies. Conversion from kinetic and potential energy of the conveyance to strain energy of the strip material is effected by the arrangement. The system is duplicated in the shaft to provide redundancy but also to prevent the cage from tilting in the guide rails, resulting in a jamming action, which will increase the deceleration value to above the allowable 2.5 m/s^2. Two steel wire ropes and four energy absorption mechanisms are therefore used to catch and decelerate the conveyance as shown in Figure 1.

The steel strip of the energy absorption device is pulled through a set of three rollers arranged so that the strip material undergoes cyclic plastic deformation. The force required to do this strain conversion work retards the conveyance. The arrangement is depicted in Figures 2 and 3. This system is similar to the Selda (Strain energy linear ductile arrestor) strip[5,6].

The energy absorption principle can be explained as follows. To induce a certain amount of strain in a metal strip, an applied force is required that will deform the material beyond the yield point of the strip material. This force required to cause plastic deformation of the strip is a function of the material properties i.e. the yield or elastic limit of the material and the rate at which work hardening occurs[7]. The force also depends on the radius of the roller and on the thickness and width of the strip. Assuming that the radius of the roller over which the strip is being pulled and bent is sufficiently small that rigid perfect plasticity occurs, the quasistatic force for bending and pulling the strip over a roller is given by[6]:

$$F = 2\sigma_y w\left[(2R+t) - \sqrt{(2R+t)^2 - t^2}\right] \qquad [1]$$

In this equation σ_y represents the yield limit, w represents the width of the strip, R the diameter of the roller and t the thickness of the strip.

In the design, the strip of material is pulled through a set of three rollers, taking the material through the yield point three times. The arrangement is depicted in Figures 2 and 3. In doing so ,work is done in the form of strain energy every time the strip is bent over a roller, so reducing the mass of strip material that has to be accelerated for retardation of the conveyance. The number of energy dissipation cycles is limited by the amount of work hardening of the material per cycle, which will eventually cause brittleness of the material, resulting in fatigue failure.

The amount of work dissipated per bending cycle can be calculated from the following equation[7]:

$$W = volume \int \sigma d\varepsilon \qquad [2]$$

In this equation W is the amount of work, σ is the stress induced in the strip, and $d\varepsilon$ is the strain rate to which the strip is submitted.

The energy absorption systems as well as the rollers holding the lower span of steel wire rope are fastened to the shaft wall by rock bolting into the shaft wall. Stainless-steel wire ropes are used to limit corrosion. The required length of steel strip is also enclosed inside a length of tubing that is packed with grease to protect the strip against corrosion. Corrosion protection is critical as corrosion will degrade the material properties and thereby reduce the maximum pulling force that can be applied. Since all the components of the system can be protected against corrosion through material selection or packaging, no maintenance should be required and inspection intervals could be extended.

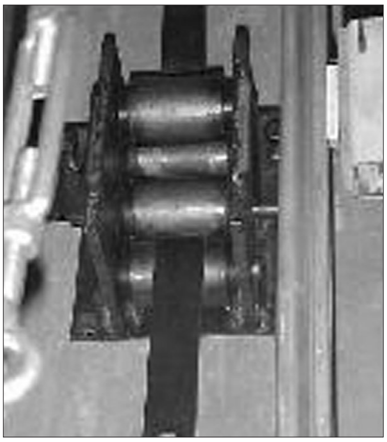

Figure 2—Kinetic to strain energy conversion system

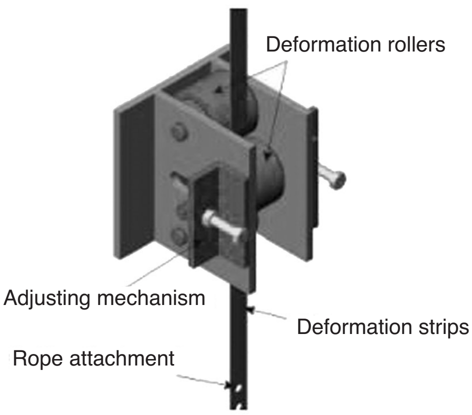

Figure 3—Kinetic to strain energy conversion system

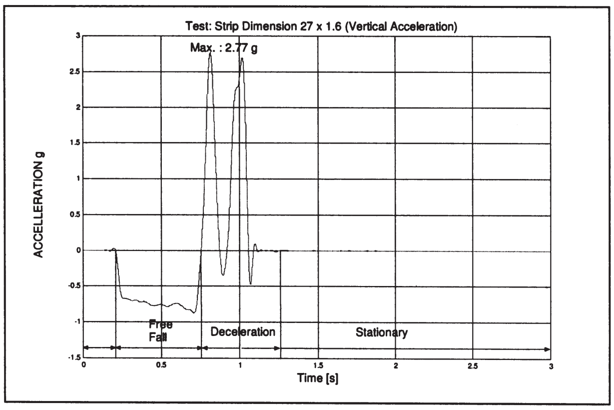

Figure 4—Underwind protection system test (deceleration)

In the selected concept compression spring sets were also introduced in the connection between the steel wire ropes and the steel strips to smooth the transfer of the force required for initial acceleration of the strips. In subsequent work it was found possible to remove these spring sets by tapering the strips[8], which also had the benefit of removing unwanted inertia from the retardation system. Inertia of the moving parts of the retardation system has to be as small as possible to limit the force on the conveyance resulting from acceleration of the wire ropes and their associated retardation elements.

Although the system is designed for a maximum conveyance speed of 18 m/s, placing such energy absorption systems in series can in principle accommodate higher conveyance speeds and even free falling conveyances.

A 1:10 scale model was designed and built. The scale model was installed and tested in the 1:10 scale shaft at the University of Pretoria. The height scaling was distorted, to allow the concept to be tested within the height constraints of the shaft, which was designed with shaft cross-sectional scaling as the primary requirement[9]. The available height was employed to allow an acceleration distance under gravitational force, and a matched deceleration range to employ the energy absorption principle as proposed. The graph depicted in Figure 4 shows the deceleration results of a mass of 80 kg, representing a conveyance, free falling from a distance of 4 m on to the catching wire ropes.

The first part of the graph shows the free fall of the mass. At time 0.75 seconds the mass contacted the stainless steel wire rope and the deceleration process started. A maximum deceleration of 2.77 g was measured, which exceeded the maximum limit of 2.5 g. However, the short duration and small size of the spike is acceptable[10]. After deceleration an upward acceleration (negative deceleration) is experienced as the conveyance tries to rebound from the catching wire rope. This negative deceleration never exceeds 10 m/s^2 and therefore the passengers or load will not experience weightlessness and the resulting risk of injury.

For a conveyance travelling at initial velocity u = 18 m/s being decelerated at a = 2.5 g the stopping distance can be calculated, using the basic kinematics equation:

$$v^2 = u^2 + 2as \qquad [3]$$

where v is the final velocity, which is zero in this case. Under these assumptions it is found that the retardation distance required s = 6.6 m.

This distance is small compared to the height available for this purpose in most mine hoist shafts, so that retrofitting of the system should be feasible. One factor that will increase this distance is the need to accommodate partially loaded conveyances within the deceleration limits required. A scheme of multiple systems in series was proposed to address this need[8], which will require additional height. Alternatively, a system designed to retard a half mass conveyance at the required deceleration would require double the height to retard a fully loaded conveyance. Since even a retardation height of 13 m should still be accommodated in most shafts, the solution would remain feasible for such ranging of conveyance mass capability from a kinematics point of view. Dynamically, as the mass of the strip material would approximately double while the mass of the conveyance would be halved, additional care would be required in designing for the acceleration forces of the retardation device's moving parts.

Overwind

As discussed in the introduction, overwind protection was required that would adhere to the same design philosophy as the underwind protection described above, in order to reduce the risk of the conveyance being hoisted past the upper limit of design operation.

The primary design parameters for the overwind protection system were:

- The maximum deceleration to prevent injuries to people transported in a conveyance was taken as 9.8 m/s^2 (1 g) as explained in the introduction.
- The maximum speed of the conveyance was 18 m/s.
- After stopping, the conveyance had to be restrained from falling down the shaft.
- After retardation the occupants of the conveyance had to be able do disembark via emergency access as applicable. No permanent distortion of the conveyance was allowed.
- The system preferably had to be a mechanical device. External sources of energy or signals supplied via electricity or hydraulics could be compromised by the primary cause of failure.
- The system had to require minimal maintenance and inspection due to the remote positioning of the equipment.

The overwind protection system in a vertical shaft can only use gravity to decelerate the conveyance as any other method would result in a deceleration higher than 1 g. This would be unacceptable because it could result in unrestrained passengers impacting against the ceiling of the conveyance and subsequently against the floor when the conveyance stops.

A standard detaching device (detaching hook)[11,12] is fitted to the top of the conveyance for attaching the hoisting steel wire rope, as is current practice. Opposing striking pins are fitted in such a way that they push the lower scissor arms of the detaching hook inwards when they are impacted on their outer ends. If an overwind situation occurs, the detaching device (detaching hook) is activated earlier than in existing spectacle plate[12] activated installations. This earlier activation is done by an additional detaching hook activation mechanism fitted at a sufficient height below the spectacle plate to allow the conveyance to retard to standstill under gravity before it would crash into the spectacle plate. The conveyance so brought to rest is prevented from falling by means of jack catches[13] on the conveyance interacting with a rack (toothed profile) fitted on the guide rails in the retardation zone. In Figure 5 the layout of the overwind protection system is shown schematically, including the additional detaching device consisting of the opposing striking pins actuated by pivoting arms pinned to the guide rails.

As the stopping distance of the conveyance depends on its speed, the space required in the headgear depends on the maximum speed of the conveyance. This distance for deceleration is governed by the basic equation of motion[3] as above.

In this equation the final velocity v = 0 again, while initial velocity u can vary from very low values up to 18 m/s. The maximum deceleration allowed a = 9.8 m/s^2. The required stopping distance s can then be calculated. This stopping distance depends on hoisting speed as shown in Table I. It can be seen that the stopping distance required and thus the height of the guide rail section that has to be fitted with teeth for the jack catches to engage, increases with the square of the increase of the hoisting speed. Due to the varying possible stopping distances, a sufficiently long rack of jack catch teeth has to be fitted to the guide rails below the spectacle plate so that the maximum stopping distance can be accommodated. The detaching of the conveyance only takes place after the conveyance has entered the section of the shaft equipped with teeth that can engage the jack catches fitted to the conveyance.

As briefly stated above, the mechanism for early detaching does not use the spectacle plate to activate the detaching hook. Horizontal forces, generated by opposing horizontally mounted striking pins actuating the lower ends of the scissors plates, activate the detaching hook. As mentioned before, this occurs at a sufficient height below the spectacle plate to allow the conveyance to retard to standstill under gravity before it would crash into the spectacle plate. The way in which this is achieved is explained below. A schematic layout of the mechanism for early detaching is shown in Figure 5 with photos of the scale model shown in Figures 6 and 7.

Table I

Stopping distance required for different hoisting speeds

Hoisting speed (m/s)	Stopping distance required/ Height of rack for jack catches to engage (m)
6	1.8
8	3.3
10	5.1
12	7.3
14	10.0
16	13.0
18	16.5m

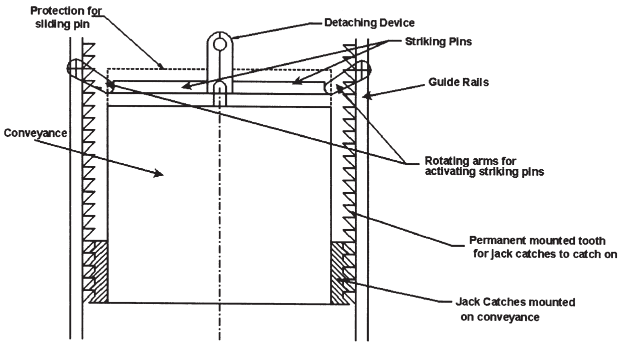

Figure 5—Layout of the overwind protection system with mechanism for early detaching

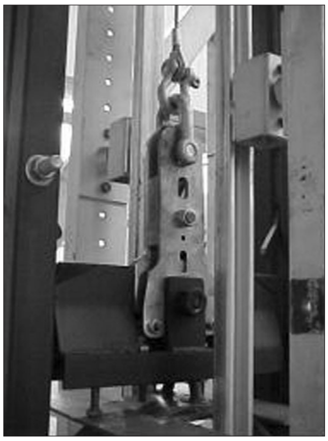

Figure 6—Mechanism for early detaching, with striking pin

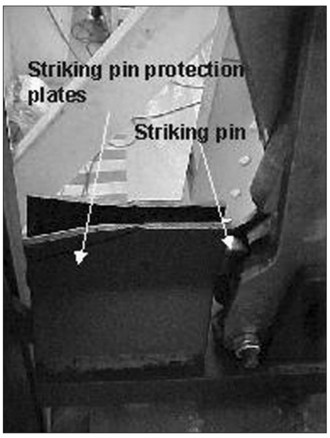

Figure 7—Mechanism for early detaching, with striking pin and protection plates

The early activation of detachment is initiated by two pivoting arms (as shown in Figure 8) that are attached to the structure of the headgear via pivoting pins. They are fitted at a sufficient height below the spectacle plate to allow the conveyance to retard to standstill under gravity before it would crash into the spectacle plate. As the conveyance moves past the pivoting arms, the arms lock into the outer ends of the striking pins of which the action of activating the

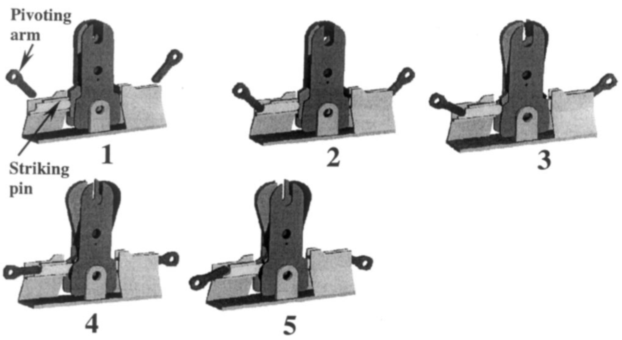

Figure 8—Working of mechanism for early detaching in sequence as the conveyance moves past the pivoting arm

detaching hook was described above. These striking pins are constrained to move only horizontally. When the conveyance moves further upward, the pivoting arms push the striking pins inwards, which in turn force the scissors plates of the detaching hook inwards to activate the detaching hook and detach the steel wire rope. Figure 8 schematically shows a sequence of the functioning of this early detaching mechanism. It is important to notice that the detaching device is activated only after the conveyance has passed the first sets of jack catch teeth on the racks fitted to the guide trails. As the inertia is low, the components are stiff and the interaction between the pivoting arms and striking pins is positive the detachment action will be effective at low and high speeds.

A 1:10 scale model was designed, built and tested in the scale model test shaft at the University of Pretoria also used for the underwind protection system test described above.

The graph depicted in Figure 9 shows the deceleration of the conveyance during a sample test. For the first period of 1.75 seconds the conveyance was accelerated. At time 1.75 seconds the conveyance ran into the pivoting arm, which in turn accelerated the striking pin to open the detaching device. The conveyance model (80 kg) then decelerates at 1*g*. Just before the point of detaching there is also a dip in the graph, evidence of the conveyance entering the first set of jack catch teeth. During the deceleration phase the maximum deceleration actually exceeds 1 *g* to a maximum value of

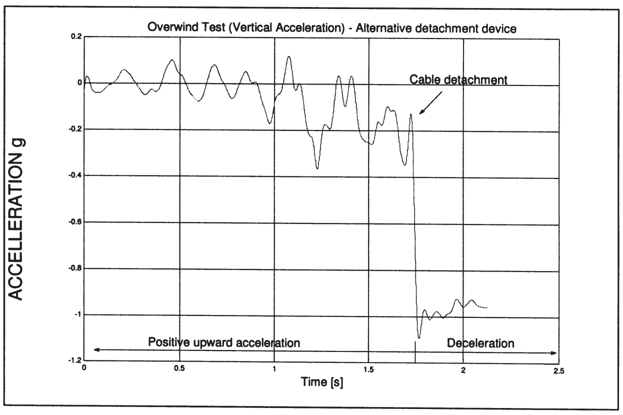

Figure 9—Overwind protection system test (deceleration)

approximately 1.1 g for a very short period of time. This extra deceleration is caused by the friction of the jack catches and the friction between the wheels and the guide rails.

From the test results it may be concluded that the kinematics of the detaching device provide an adequate detaching function. Further calculation is required to establish the dynamics of a full-scale detaching event in order to ensure that all elements are strong enough to provide the functions required.

Furthermore, it is important to establish whether sufficient space would be available in existing headgears to allow retrofit of the system.

From a survey conducted among South African mines it was concluded that it would be viable for the majority of the respondents.

Conclusion and recommendation

The test results proved that it was possible to provide the required deceleration values for both underwind and overwind with simple and robust mechanical systems. These systems complied with the design requirement of being self-sufficient. They were localized mechanical systems in order to remove dependency on remote or external energy and information supply, including electrical and hydraulic supply, that increase the risks associated with most existing protection systems.

The underwind protection system operational height required above the crash bar at the shaft bottom for a conveyance traveling at 18 m/s was calculated to be 6.6 meters. Depending on the risk of partially loaded conveyances, additional height might be required to decelerate such conveyances within the deceleration limits required. Nevertheless, the height remains sufficiently small that retrofitting the system in existing shafts should be feasible.

The height requirement investigation of the overwind protection system showed that the headgear of a significant proportion of shafts has enough space below the spectacle plate to install the rack of jack catch teeth as well as the detaching mechanism of the overwind protection system.

From the test results and design calculations, the feasibility and likely effectiveness in the surveyed incidents of the proposed concepts could be confirmed sufficiently to warrant further development.

Full-scale design should include the following:

Underwind protection system:

- ➤ Rock engineering design of rock bolts to fasten the system onto the rock of the shaft wall.
- ➤ Design of the contact zone between wire ropes and conveyance to prevent the conveyance from undergoing permanent deformation.

Overwind protection system:

- ➤ Analysis of the headgear structure to accommodate the forces generated by the jack catches catching onto the rack fitted to the guide rails.
- ➤ Analysis and testing of the jack catches to ensure safety during operation.
- ➤ Analysis of the headgear structure to accommodate the horizontal force transmitted by the pivoting arms when actuating the striking pins used to open the detaching device.
- ➤ Analysis of the pivot arms and striking pins to establish their dynamics and strength.

Before shaft tests of the protection systems, full-scale tests in a suitable test facility are recommended to further qualify the systems.

Although designed with vertical shafts in mind, in principle the protection systems can be used in incline shafts as well.

The risk of underwind and overwind incidents is sufficient that additional self-sufficient protection devices should be installed to decrease the risk of injury or loss of life of workers being transported in mine conveyances.

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References

1. OTTERMANN, R.W., VON WIELLIGH, A.J., BURGER, N.D.L., DE WET, P.R., BLOM, T.F., and VISSER-UYS, P.E. The Identification, Investigation and Analysis of End-of-wind Protection Devices for Vertical and Incline Shafts. *Safety in Mines Research Advisory Committee (SIMRAC),* GAP 638. Johannesburg. 2000.
2. MINISTRY OF ENERGY AND MINES, British Colombia. Health, Safety and Reclamation Code, Part 7. 1997.
3. BLANCHARD, B.S., and FABRYCKY, W.J., *Systems Engineering and Analysis (3rd edn).* Prentice Hall. 1998.
4. HORNE HYDRAULICS. Technogrid energy absorption device. http://www.horne.co.za. 2003. Johannesburg.
5. SELTRUST ENGINEERING. SELDA Strain Energy Linear Ductile Arrestor. Bellambie Mining ID 9085806, 11 June 1999. London.
6. ROSSLEE, F., COETZEE, J., and PRETORIUS, L. Using cyclic plastic bending as an energy absorption mechanism. *R&D Journal,* 1998, vol. 14, no.1. Johannesburg. pp. 15–21.
7. TIMOSHENKO, S.P., and GERE, J.M., *Mechanics of Materials,* (3rd edn.) Chapman and Hall, UK. 1994.
8. STEYNBERG, A.J.J. Dynamic Cyclic Bending, Kinetic to Strain Energy, Deceleration Systems. M. Eng. dissertation. University of Pretoria. Department of Mechanical and Aeronautical Engineering. Pretoria. 2003.
9. HEYNS, M., and VAN NIEKERK, H.J. Design of a small-scale conveyance dynamics rig. *Laboratory for Advanced Engineering Report. LGI 91/004.* University of Pretoria. Pretoria. 2 January 1991.
10. EIBRAND, A.M. Human Tolerance to Rapidly Applied Accelerations, NASA Design Memo, 5-19-59E. Washington DC. 1959.
11. PEELE, R. and CHURCH, J.A. *Mining Engineers' Handbook. (3rd edn.*) Wiley. New York. 1941.
12. ROUX, J.D., DREYER, E., and WAINWRIGHT, K.A. Safety detaching hook specification. GAP 536. *Safety in Mines Research Advisory Committee (SIMRAC).* Johannesburg. May 1999.
13. JEPPE, B.C. *Gold Mining on the Witwatersrand, Vol II.* Transvaal Chamber of Mines. Johannesburg. 1946. ◆