

# Design constraints on trackless mining equipment in low and extra low stope hard rock mining

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Trackless mining in low profile (LP) and extra low profile (XLP) platinum stopes has forced equipment designers to re-evaluate their equipment profiles to match the undulating reef. Low profile equipment designed to work in conditions with smooth and flat floors will simply not function with undulating floors and roofs and will become wedged between the floor and roof, beached on undulations due to lack of ground clearance, or suffer excessive mechanical damage. Tyres, too, need special evaluation, as conventional tyres for trackless mining do not work in an extra low profile environment.

## Introduction

Most low and extra low profile operations are either room-and-pillar or some form of open stope or breast operation both following the reef. In this paper, 'low profile' stopes are defined as operating with a stope width between 1.6 and 1.8 metres and 'extra low' stopes are defined as operating with a stope width between 1.1 and 1.2 metres. Platinum mining methods, in particular, blast the footwall below the orebody parting to enable extraction of the richest part of the ore which lies at the bottom of the orebody, so both the footwall and hangingwall surfaces are blasted rock faces rather than geological parting planes. Furthermore, the nature of the Bushveld Complex is that the reef has discontinuities, potholes and continuous undulations.

These changes in the reef horizon and the consequent undulations in the footwall and hangingwall mean that the trammed equipment is going to be in frequent mechanical contact with the roof and floor in these low profile stopes. Experience has shown us that this mechanical contact will be in direct proportion to the three-dimensional shape of the machine and not just its height.

This paper will discuss two major design constraints, these being optimization of the machine shape to minimize mechanical damage and tyre selection as this controls tractive effort for tramming. Other major constraints, such as operator visibility and the ergonomic effects of working in a low seam environment, will not be discussed in this paper.

## Design constraints

### Optimizing the machine shape

The first time trackless equipment was placed in these low stope operations it was found that their operation was constrained by the envelope shape of the equipment in all directions and that severe mechanical damage occurred to the top and sides of the machine frames.

Initial low profile machine design only considered the height of the machine in relation to the stope width as well as the arrival and departure angles under the ends of the

frame to accommodate floor surface variations. No consideration was given to the top of the frame, which was largely flat, with no approach or departure angles on top of the frame (Figure 1).

Prior experience with trackless equipment did not have to consider mechanical contact with the roof as the stopes were sufficiently wide to allow machine clearance with the roof.

After introducing the first suite of low profile equipment, excessive operational mechanical damage occurred to the top of the frame and covers. It was then realized that the designer now had to consider how the machine moved in these confined conditions to minimize the mechanical contact with both the roof and floor in order to both optimize machine manoeuvrability and minimize damage.

To illustrate the typical extent of the damage caused to a loader frame, Figure 2 shows the damage to the frame of an extra low profile loader, the EJC 88XLP, after only a short time in service. Extensive damage can be seen on the top rear of the frame as well as on the sides from collisions with the walls in the stopes.

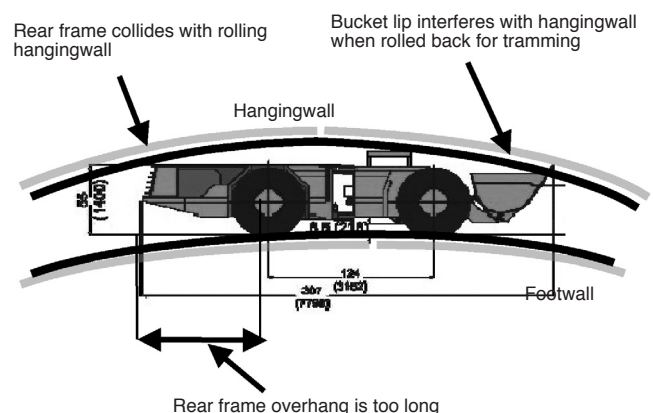
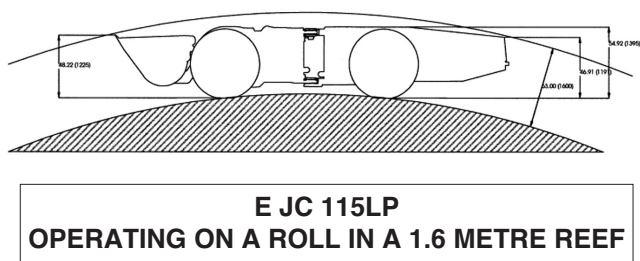


Figure 1. Initial loader profile. Dimensions assume non-undulating floor and no roof contact with frame (EJC 115LP)



**Figure 2. Severe damage to machine extremities from undulating floors and mechanical contact with roof in an extra low profile stope (EJC 88XLP)**



**Figure 3. CAD-simulated ideal machine profile in side view (longitudinal dimensions)**

It was found that, as the machine moves over the uneven floor, the top edges of the machine are the first to come into contact with the roof as the wheels rise over obstacles.

Two new design factors now are apparent. Firstly, the shape of the top of the machine is important to limit mechanical contact with the hangingwall and, secondly, the machine length also determines its effective clearance height because of the undulating motion of the machine as it trams in these stopes. This immediately necessitated that the top of the machine should be rounded (in the radiused profile of an inverted 'saucer') and the overall length and width of the machine should be reduced to minimize the mechanical contacts of the edges of the frame with the roof.

The designer now faced the question of what radius the saucer should approximate. Initially this looked impossible to evaluate as there were too many variables. Further examination of the undulating machine movement highlighted that not only should the roof be considered but, simultaneously, the shape of the floor should also be considered.

To arrive at a redesign starting point, it was assumed that mechanized drilling improves the blast and fragmentation quality and results in approximately parallel floors and roofs. It then became apparent that the worst design situation existed in a parallel stope width with a roof radius on the same centreline as the floor. Figure 3 gives an idealized CAD simulated side view of this design consideration where the approach and departure angles of the top of the machine have been taken into consideration as well as the machine length to minimize contact with the hangingwall. However, in reality, there are no smooth hanging- or footwalls and bumps in these make the situation worse. These bumps or hollows can be of the

order of 300 mm high by 500 mm long.

On further examining Figure 3, the smallest radius of the convex floor that the machine will cope with is one where the centre of the dome just touches the belly of the machine with all wheels still in contact with the ground. The ground clearance here is therefore critical, as the machine will become beached on its belly with the wheels off the ground and unable to tram if there is insufficient clearance.

These constraints apply to both the longitudinal dimensions and the cross-sectional dimensions. As the machine rises and dips as it trams, it also rolls from side to side and the edges of the frame come into contact with the roof.

Conversely, if the floor is concave and the roof convex, the approach and departure angles and the ground clearance of the machine will determine the minimum radius of the floor depression. Refer to Figure 4, which shows the final design of the tapered profile of the loader with approach and departure angles on both the top and bottom of the frames.

Optimizing the machine shape with these stope envelopes gives the following points:

- A short wheelbase, and track width of approximately the same dimension.
- A rounded transverse profile as well as a rounded longitudinal profile that looks like a 'used soap bar'.
- Approach, departure and hump angles all need to be approximately equal and the ideal profile is actually 'used soap bar' in shape (Figure 4).

Although the EJC 115LP has been reviewed in detail in this discussion, the lessons learnt about the machine profile were applied to the EJC 88XLP from the outset as the environment was even more demanding in the extra low stope. Despite this, modifications were necessary to the EJC 88XLP to alter and reinforce the top of the rear frame because of collision damage in service.

#### ***Drill rig and bolter profiles in XLP stopes***

Although the initial exercise looked only at the loader profiles, these design constraints applied to all trackless equipment operating in these confined stopes. Drills and roof bolters also have to manoeuvre around the stopes; generally in worse underfoot conditions as the loader can use its bucket to clear a roadway.

Figures 5 and 6 show the modifications done to a roof bolter to follow the 'used soap bar' shaped profile.

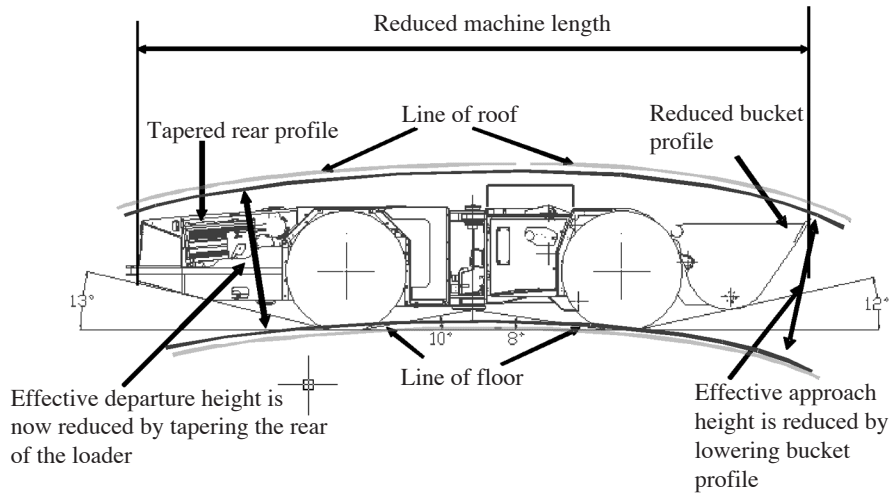
Figures 7a and b show how the machine profile can be altered dynamically using the boom hydraulics to suit the undulations in the stope.

#### **Tyre selection, the major constraint on tractive effort in extra low profile stopes**

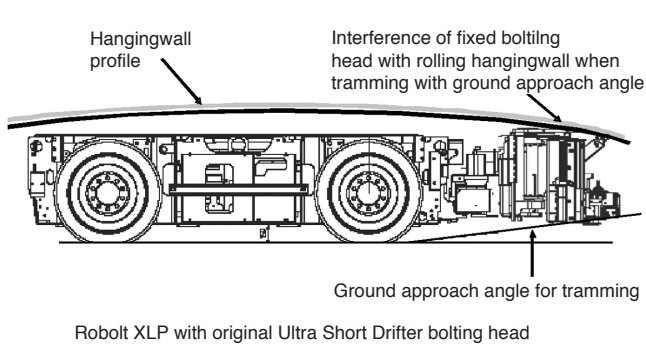
Tyre selection is easier in low profile stope operations as commercially available tyres will operate satisfactorily and the low profile suite used commercially available tyres. However, it was found that in extra low profile stopes the situation changed considerably.

#### ***Tyres for the XLP drill and bolter***

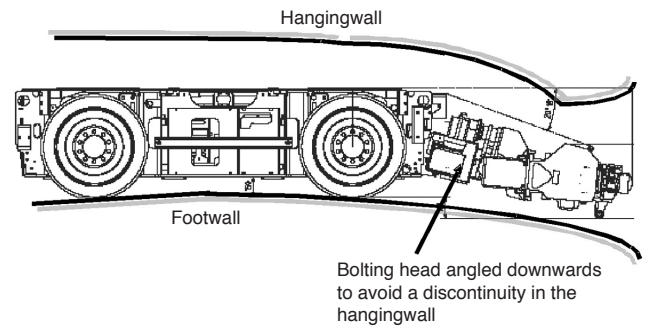
For the drill and bolter the constraint is the gradient traction as the machine trams along the face either for face drilling or for bolting. Both operations, however, require fine manoeuvring of the machine on its wheels to position it correctly and, consequently, consistent traction is important to minimize wheel slip and skidding down the slope of the footwall.



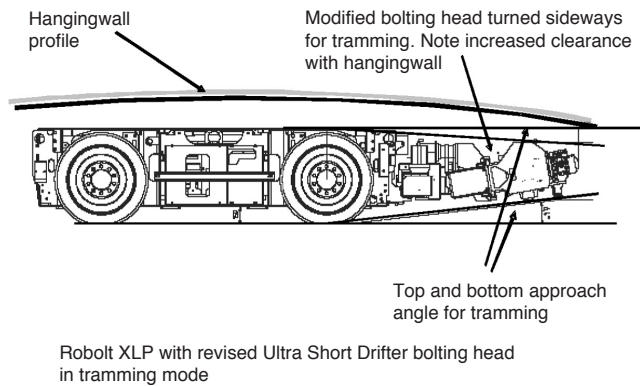
**Fig. 4. Modified EJC 115LP loader profile to accommodate design constraints. Note the tapered rear frame and the lowered bucket profile**



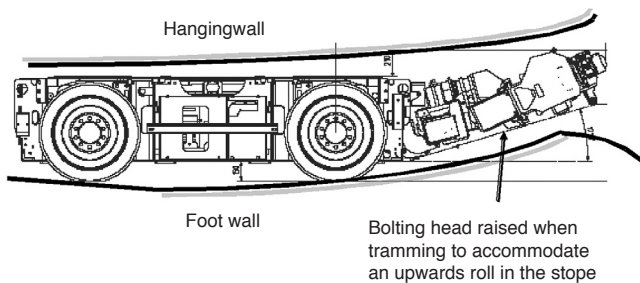
**Figure 5. Initial roof bolter profile. Bolting head is too high for effective tramming**



**Figure 7b. Modified bolter profile showing boom manipulation to accommodate a downwards rolling stope**



**Figure 6. Reduced profile bolter with bolting head in tramming mode shown with top and bottom approach angles to facilitate tramming**



**Figure 7a. Modified bolter profile showing boom manipulation to accommodate an upwards rolling stope**

For the drill and bolter the initial approach was to use segmented solid tyres to achieve both tyre compliance and traction. The secondary benefit was to be able to replace a failed tyre section rather than a complete tyre and reduce the tyre running costs. These segmented tyres did not work as the segments failed prematurely in the harsh conditions and they were expensive and difficult to replace. The attachment bolts rusted into the rim, necessitating removal by grinding. If the broken segments were not replaced timeously then the adjacent segments failed as well and the result was buckling of the wheel rim, from direct contact with the ground. This necessitated a complete wheel replacement. Therefore, as an experiment, solid tyres were fitted to both the drill and bolter.

Initially an imported solid compliant tyre was fitted and this worked well but the lead times were very long so a simpler design solid-wrapped tyre was manufactured locally and fitted. The traction of this solid tyre was as good as the segmented tyre in comparative sites.

On a trial on very steep stopes (approximating 20 degrees) the tyre traction was marginal and, to compensate for this, wider tyres were fitted as an option in this application.

#### **Tyres for the XLP loader**

Tractive effort is the major consideration for the XLP loader. There is no boom lift or bucket roll-back function to assist with weight transfer to the front tyres to increase traction while loading. The loading operation is one of forcing the bucket into the muck pile to fill the bucket and

this action requires both high tractive effort and weight on all the drive wheels, so the loader is heavily built to obtain the necessary weight.

The extra low stope width places a severe constraint on conventional tyres. For the loader to operate in a 1.1 m to 1.2 m stope, the tyres need to be 950 mm diameter or less to cater for frame fenders and still have sufficient clearance between the tyre and the roof when driving over obstacles.

A tyre that is too large in diameter and that becomes wedged between a floor obstruction and the roof will jam. The sudden shock load will severely damage the machine drive train.

As wheel motor hydrostatic drives are, in fact, the only compact and practical layout for XLP loaders and drills, the resultant size of the wheel hub is such that a conventional 350 mm (15 inch) wheel rim will not fit and, accordingly, the wheel rim diameters are approximately 500 mm or larger, requiring special low aspect ratio tyres to comply with the restricted overall diameter.

Conventional section width tyres with these restricted diameters mean small footprints and low carrying loads. As the tyres are relatively narrow, this places a severe restriction on the tractive effort that can be transmitted between tyre and ground.

To get around this constraint, initially dual foam-filled tyres were fitted. These multiple ply specials needed 1.0 MPa air pressure to carry the loads. Pressurized foam filling at 1.0 MPa was then necessary to eliminate these high pneumatic pressures and special reinforced rims with heavy-duty flanges and lock rings were also required.

#### *Experiments with different tyre combinations*

There was no prior tyre history which could be referred to for this application, so the decision was made to try three different tyre options and review the results.

The three options tried were

- dual foam-filled tyres (mentioned above),
- solid industrial-type tyres, and
- wide base single foam-filled tyres.

During the initial trial the ground clearance was found to be too low and the tractive effort from the dual tyres was insufficient. As an experiment, chains were fitted to increase both ground clearance and increase the tyre footprint to increase tractive effort.

Dual tyre life was very poor and is reviewed below. The replacement carcass process was complicated. It was discovered that the used carcasses had to be machined off the rim when replacement was necessary, as the welded intermediate flange between the dual tyres prevented the carcasses and outer loose flanges from being pressed off when the lock rings were removed. This dual design was, therefore, unacceptable and the dual configuration tyre was abandoned early in the trial phase in favour of solid tyres, which were used for most of the subsequent trials.

Initially the chains seemed to provide the ground clearance and tractive effort needed so it was decided pursue the tyre trials with chains also fitted to solid and wide single tyres.

The solid tyre and chain combination gave better tyre life than the duals and acceptable tractive effort but it was nearly impossible to keep the chain on the tyre in service, leading to frequent breakdowns and loss of loader availability. The lack of compliance of the solid tyre meant that the chains did not maintain their correct tension and needed frequent retightening or repair to damaged links from the excessive chain tensions being used to overcome the slippage. This down time from chain failures alone was

estimated to be 15% or more of shift time, reducing the machine availability from an acceptable 85% to an unacceptable 70% or less. Additionally, when the chain slipped off the tyre, it often wrapped itself around the casing of the wheel hub reduction gearbox, causing catastrophic damage to the gearbox casing on several occasions, with further down time. Different tyre side profiles were tried to cure this problem but to no avail (Figure 8).

The third option tried, was an ultra-wide, single pneumatic carcass that was also foam filled and fitted with chains. The improved footprint from the wide single profile and the consequent improved tyre compliance should have provided the best solution. Although this combination seemed to work reasonably well, the initial tyre carcass and wheel rim costs were disproportionately high and tyre life was not better than the solid tyres. So this option was also discarded.

As a quick experiment this ultra wide single tyre was also tried without chains to see the effect of a rubber contact with the rock in place of the chains. The bare rubber traction was surprisingly good but as there were concerns about tyre life, initially this concept was not pursued.

#### *Chains and tyre life*

The increased footprint from the chains improved the tractive effort and, with the better ground clearance, this tyre and chain combination appeared to be the solution even though the machine height was increased by 60 mm and it now stood at over 900 mm high over the framework.

The project team had also been assured that the chains would protect the tyre carcasses as a secondary benefit.

During the trials it was found that the chains do not protect the tyre carcass as these very low aspect ratio tyres continuously slip inside the chains. Any rocks protruding through the chain links work like a flensing knife and peel the tread off the tyre as the tyres slip inside the chains, especially if water is present.

Tyre failures from spin cuts started to appear from about 100 hours on the dual tyre configuration and dual tyre life was only 300 hours per carcass. Initially this was accepted as a rubber compounding problem with the dual foam-filled tyres but similar spin cuts also appeared on the set of solid tyres with chains. This damage in our experience is more severe with the chains than conventional spin cuts on unprotected tyres.



**Figure 8. Typical chain failure. The chain has slipped off the tyre**

As the chain retention problem could not be resolved, the decision was made to go back to review the brief test of the single wide base tyres running with ‘rubber on the rock’.

To maintain the improved ground clearance that had been a benefit of using the chains and which was acknowledged as being critical, it was therefore decided to try larger diameter solid rubber tyres without chains. Initially the solid tyres were wrapped to 900 mm. This approximated the original installed chain diameter over the tyres but, subsequently, the solid tyres were wrapped to 950 mm diameter. Subsequent to that, the section width was increased from 400 to 430 mm because the tyres no longer required buffing to fit the chains.

We were advised that the lack of elastic compliance of solid tyres would give a limited footprint when compared with pneumatic tyres, as the solid tyres tend to ride up and over obstacles, and that this could mitigate against using them—both from the traction aspect and the driver’s comfort.

In practice it was found that the low profile pneumatic tyre is so stiff when foam filled to the high pressure of 1.0 MPa that it is no longer compliant and behaves like a solid tyre.

Sandvik Mining and Construction and their local suppliers experimented with various rubber compounds on solid tyres. Although the tractive effort was marginally less for bare solid tyres than with tyres and chains, the increase in machine availability was dramatic because there were no tyre breakdowns and the tyres were only removed after they had worn to the replacement point.

The most successful type of solid tyre was made from an industrial rubber compound containing short steel wire strands, which increased cut resistance. In addition, the solid tyre had the effect of rolling the footwall fines flat during loading. In contrast, tyres with chains spun during the loading cycle, digging holes in the footwall fines.

Comparative costs of replacement foam-filled pneumatic carcasses and rims against solid tyres being ‘rewrapped’ are not discussed here as there was conflicting information at the time of writing. Table I shows the running costs of chains and the down-time costs.

Not only is the solid tyre and rim fitted to the machine and run without further down time until worn out after a longer service life, but the loader operating costs are at least R49 per hour less than tyre and chain combination costs for all four wheels.

Drivers were asked to evaluate comfort levels between the foam-filled pneumatic tyre with chains and the solid tyre, and reported that there was a small, perceived, increase in shock loading with the solid tyre.

## Conclusion

To make low seam mining mechanization successful, the equipment designer needs to have a clear understanding of how equipment behaves dynamically under these conditions and the machine envelope shape has to be altered accordingly. Modelling with flat parallel stopes is not valid in this more demanding environment. Old rules for trackless mining machine design no longer apply in this more demanding operation with limited operating clearances.

The final selection on tyre design for the XLP suite favours solid rubber industrial tyres without chains, versus special low profile multiple ply foam-filled carcasses protected with wheel chains or solid tyres protected with chains. This selection is currently based on improved machine availability and longer tyre life when using solid tyres without chains.

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**Table I**  
**Comparison of running costs and downtime costs of tyres with and without chains**

Item	Tyres with chains	Solid wrapped rubber tyres without chains
Purchase cost of 4 wheel chains per loader	R60 000	N/A
Service call-out cost for chain failure	R500	N/A
Chain repair costs (average)	R800	N/A
Call-out frequency per loader (single shift operation)	One lost shift per 6-day week	N/A
Estimated chain life	2 000 h	N/A
Chain service costs over 1 000 hours (based on 15 service call-outs per 1 000 loader hrs)	R19 500	NIL
Estimated cost per hour of chains	R49.5	NIL
Down time from chain failures	15 to 20 % of weekly shift	NIL
Average life of tyres	300 to 500 h	> 750 h

