VERTICAL TRANSPORTATION OF MEN, MATERIAL AND MINERAL FROM ULTRADEEP MINING OPERATIONS – AN ANALYSIS OF OPTIONS

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A project report submitted to the Faculty of Engineering, University of the Witwatersrand, Johannesburg, in partial fulfilment of the requirements for the degree of Master of Science in Engineering.

Johannesburg, 2000

DECLARATION

I declare that this project report is my own unaided work. It is being submitted for the degree of Master of Science in Engineering in the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

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(Signature of candidate)

18th day of August 2000

Abstract

Single stage hoisting from ultra-deep level mining operations is vital for sustainable exploitation of mineral deposits beyond current depths(<3000m). Analysis of options available to hoist men, material and ore takes the form of industry visits, study of cases, proposed and new ideas. Operational feasibility is done by analysis of the methods available. Underground mineral processing is discussed with a view to combining it with other available hoisting methods and taking advantage of likely spin-offs. New materials for ropes, conveyances and pumping accessories and other options are discussed.

Power consumption and consequently unit cost of the same is used extensively in evaluating the possible transportation scenarios.

Skip hoisting with revised factor of safety regulations and new rope materials is still an attractive option so is the concept of hydraulic hoisting of fine material with a centrifugal pump. Rope-less technology is still futuristic for all practical intents.

For you

Nony and Rue

Thank you for the support

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diameter	d
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CHAPTER 1 INTRODUCTION

As shallow and near surface ore reserves run out, the trend has been to explore and develop deeper deposits or mine lower grade by open cast methods. In the context of this project report and based on the Industry guide by the Chamber of Mines, shallow depth is anything down to about 1000m while anything from 1000m to 2250 is moderate. Depths between 2250m and 3500m are deep while those beyond 3500m are ultra deep. This classification is relative and tends to depend much on the prevailing situation in the mining area under discussion. This classification is true in the South African mining industry especially in precious metals like gold and platinum. A large percentage of the production comes from depths around 2000m and below. Figure1.1 is an expression of the trend in mining production and respective depths.

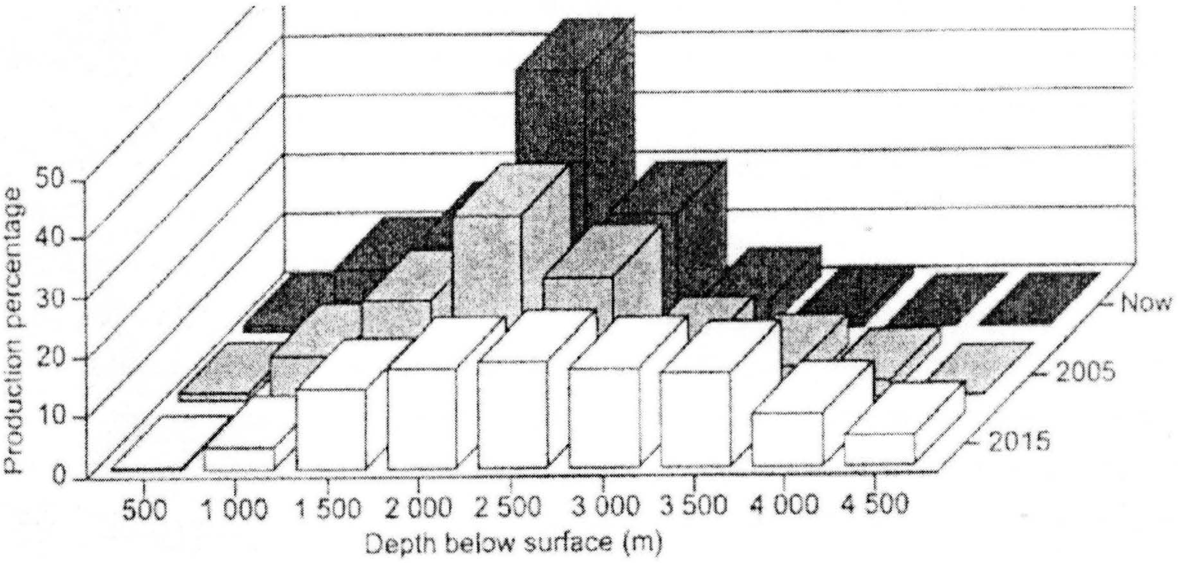


Figure.1.1- **Current and predicted mining depths.**[24]

As the depth of mining increases, so does the need to find suitable technology to enable mining profitably. One of the critical problems at these new depths being the safe, rapid and cost effective access to and from working areas and the movement of man, ore, waste and material.

The main mode of vertical transportation in underground mining is rope winding. Over time the rope configuration has undergone some changes from the uniform strand rope to the tapered rope. Current ropes in use have to satisfy the primary function of providing the transport between surface and underground under safe conditions. The need to carry out the hoisting under safe conditions necessitates the need for a rope factor of safety (static and dynamic). The strength of the rope is of importance as the load to be elevated depends on the rope strength, which in turn depends on the rope construction and the grade of the steel used.

With increase in winding depth, the self-weight of the rope naturally increases. The implication is that, depending on the strength parameter, there must be a critical rope length that will break under its own weight without supporting any load. At such levels there is a trade-off between rope strength and weight. The higher the strength to weight ratio of the rope the deeper mining can be undertaken. This implies that the rope has a limiting effect on single stage winding. For mining to continue beyond such depths a number of alternatives have either been implemented or proposed for further investigation. The reason that hoisting from 4000m is not yet undertaken is because of the traditional winding limitations imposed by the required rope factor of safety. This can be reduced with rope stress management shown in the concept of closed loop brake control by Cegelec at Moab Khotsong (11 shaft in Vaal Reefs, South Africa). A factor of safety is defined as the ratio between the breaking strength of the rope and the maximum static pull in the rope due to its own weight and suspended load. This is sometimes referred to as the "apparent" factor of safety; there is also reference to the "actual" factor of safety, which in addition to the foregoing takes into account the effects of bending, acceleration and kinetic shock waves. The capacity of a rope to resist kinetic shocks depends upon, and is probably roughly proportional to, the weight of the rope and, since weight and length are proportional, it is evident that a long rope is better able to resist such shocks than a short one. With the prevailing factors of safety of 9 at the conveyance (i.e., capacity factor) and 4.5 at the sheave in the South African mining regulations, it is practically impossible to hoist economic quantity of ore from say 4000m.

Another reason is the high cost associated with the vertical and, to some extent, lateral transportation. Vertical transportation costs are about 20% of the operating costs of an average mine (2000m depth) being third after ventilation and labour.

Hydraulic transportation has been used in a few instances for the transportation of rock either as slurry or just crushed rock in suspension in some old coal mines in Europe like Hansa Grube (Germany), Debiensko Mine (Poland) and other base metal and precious metal mines like Lengede (W. Germany) iron ore. This has been over very short vertical distances and the results obtained are not extrapolated in the case of ultra deep transportation.
So far the main application of hydraulic transportation has been horizontal and over very long distances. The advantages of hydraulic transportation in the vertical have already been identified but still such hoisting has not been widely used at present.
In some mining operations like deep-sea mining of manganese and diamonds, other variations of hydraulic transportation have been the sole mode of mineral conveyance from the seabed over depths of more than 5000m. These methods include jet pumping, airlifting or reverse circulation. Underground ore preparation and subsequent vertical hydraulic transportation is used in the uranium industry to minimise workers' exposure to radiation.

Other rope-less technologies are still in their infancy and are still to `bear fruit'.
It is against this background therefore that this paper attempts to make an analysis or evaluation of the possible alternatives to raising economic quantities of ore or rock from deep levels. One of the unanswered questions is: Will it be correct to single out the factor of safety as the only or major area needing re-analysis so as to enable mining at levels beyond 3000m? It could be argued that the factor of safety expresses our level of ignorance of the risk involved when using a rope for hoisting. The less knowledge we have the higher is the factor of safety and vice versa. Hence with the improvement in technology and our understanding of the forces and stresses to which the rope is subjected, consideration should be given to revise the safety factors downwards. Whether this is the best solution or to what extent this solution could solve the impending problem has still to be established.
While the main problem might be the cost and limitations of the winding system of lifting men, ore and material from ever increasing depths at a minimum cost without compromising the safety. It must be realised that, the solution does not necessarily mean considering only a rope to raise the product from deep levels.

Instead widening the options horizon and even considering the different production arrangements and systems could remove the focus from rope hoisting. Figure1.2 is a graphical illustration of the current situation compared with the desired rope hoisting set-up.

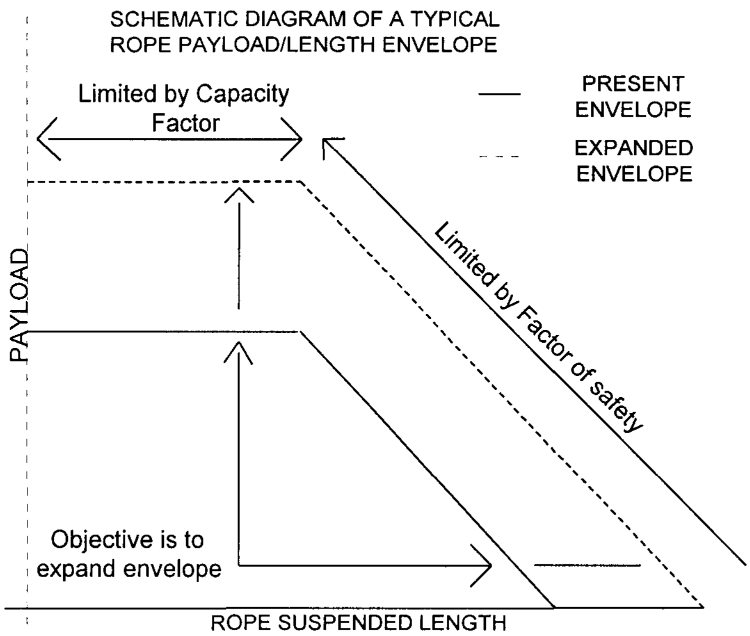


Figure 1.2

CHAPTER 2 CURRENT STATUS OF ROPE HOISTING

This chapter aims to provide an overview of the past and present situation in the field of vertical transport by way of rope hoisting. There is also a brief review of the developments currently in progress or just completed. Extensive use is made of the past and present records and research results in rope manufacture and winder control technology. The winding rope has evolved from the fibre rope in small shallow one-man operations to the present steel ropes and in between different configurations of rope have been in use including tapered ropes. There are basically four types of ropes under review:

1) Uniform section wire/steel rope.
2) Tapered profile steel/wire rope.
3) Tapered mass rope
4) Synthetic uniform section rope.

However uniform stranded wire/steel ropes have been the main type to be used in almost all the situations and they have been in production in Southern Africa since 1921. Taper ropes are not currently in production and have been limited to central and east Europe lead, zinc and silver mines in the first half of the 20th century. Research on Tapered mass rope is still vibrant and some interesting results have been produced. The synthetic ropes are still being investigated even though they have been used on trial basis in some mines in the United States of America under the supervision of the Bureau Of Mines.

2.1 Rope Factor of Safety

The safe use of any rope mentioned above is ensured by the provisions of a very important factor called the Factor of Safety which is the ratio of the breaking load of the rope to the total supported load-as it includes conveyance and attachments. This ratio helps to limit the amount of loading to which the rope is subjected, within a safe range. Below are some of the reasons why the factor of safety is a very important criterion in any form of rope hoisting.

- Rope stretch suddenly imposed during skip loading
- Stresses applied during normal acceleration and deceleration.
- Abnormal conditions such as emergency braking
- Rope deterioration during service life.

- Self-weight of the rope.
- Rope elongation with depth due to the steep temperature gradient.

There has been a different view altogether to the issues raised above as the main reasons for the need for factors of safety. While it is accepted that the reasons are sound, it is argued that most of these dynamic stresses can be minimised to manageable levels so that it should be possible to reduce the factor of safety by between 12 and 25 % [5,14]. The reduction in these factors entails a corresponding increase of between 4 to 8 times in payload.

2.2 Uniform Section Ropes In Winding

Rope winding being currently used for the hoisting of material to and from underground makes use of a rope with a uniform section. The trend in most if not all the mines, beyond 2000m depth, have been winding in stages. The shaft from surface is sunk as far as practically possible and then later during the life of the mine a sub-vertical shaft is sunk to reach the lower portion of the ore body. The main drawback with this approach is the obvious transfer of men and material re-handling at changeover. This wastes shift time and it is generally inefficient. Other disadvantages with the method include the need to provide an extra hoist for the sub vertical shaft plus extra staff and other facilities at the interchange.

A number of hoisting alternatives have been proposed to counter the drawbacks of hoisting in stages. It is worth noting however, that increased hoisting depth with economic payloads is possible with the use of Blair multi-rope winding system but these drum winders have some peculiar disadvantages such as:

- Higher initial mechanical component cost.
- More complex headgear and sheave arrangements.
- Higher maintenance time and cost.
- Require angling or offset of drums to achieve correct rope angles leading to complex mechanical variations and hence the use of a very large universal joint between the drum pairs so as to allow for angling.

Twin ropes have also been recommended (necessitating pairing) as another way to doubling allowable end-load however the consequent power demand goes up as well.

Depending on the depth of wind, the rope to be coiled is usually limited to a maximum number of 4 layers around the drum. The rope tread pressure at maximum static tension must be less than 3.2 MPa and the drum to rope diameter ratio (D/d) has to be greater than 127 to allow for winding speeds of 20m/s and more. The foregoing requirements are critical in determining the drum size plus other dimensions such as the rope coiling compartments. The overall axial length of the drum is critical and this also presents another limitation to the drum size.For example in the experiments being done by Cegelec PLC (UK)[22] the following dimensions were chosen: a rope-coiling compartment of 8.5m diameter by 2.8m wide was chosen. Five layers of rope could have been used but the detrimental effect on rope life would have been too high.

The drum diameter could also have been reduced to values compatible with the rope tread pressure and D/d ratio, however this would have resulted in increased drum width and axial overall length of the winder.[4,5] Use of twin rope drum hoists imposes a new problem of fleeting angle of the rope to the head sheave. Fleet angle refers to the maximum angle included between the rope plane and the rope that is being coiled on the drum.

The axial length of the twin rope compartment drums requires wide centres for the headgear sheaves and conveyances in the shaft. To limit the diameter of the shaft, a universal coupling or Hooke's joint is placed between the two drums to allow the drums to be inclined towards the shaft centre. This alleviates rope-angling problems, even with sheave wheels at close centres. Results have indicated that the loading on the two rope coiling compartments is neither equal nor symmetrical about the drum centre line.

This asymmetrical loading also occurs on drums with single coiling compartments but is more pronounced on drums with twin rope coiling compartments. [3,4,5].

Simplification of the drum loading by assuming a uniform external pressure can lead to inaccurate predictions of the deflections, particularly in the brake disc area. Finite element analysis can be used to determine final drum deflections and stresses. Often where machinery is scaled up in size such as in the case of large drums, like the ones under discussion, there is always a danger of natural frequencies setting in during rotation. With scaling up, the stiffness of the structure will seldom remain proportional to the mass supported. With the hoist of the size under discussion with two rope coiling compartments filled to capacity there is a total mass of 740 tons supported by two spiders, designed with some flexibility to deflect with the drum and the drum barrel.

The natural frequencies of oscillations should be investigated in the axial direction for the drum construction chosen. Furthermore the mass and inertia of the two drums and drive motors on the shaft must be checked for transverse (perpendicular to the shaft) and torsional oscillations on the shaft.

2.2.1 Blair Multi Rope (BMR) Winder

Due to the adaptability of the Blair Multi-Rope winder to deep level applications, permanent BMR shaft winders can be used for sinking duty and in this instance are operated as double drum kibble winders. In that situation the design caters for the D/d and tread pressure requirements for the sinking rope, and use can be made of an additional set of grooved sleeves, if the sinking (kibble) rope diameter differs from the permanent BMR ropes[13]. The statistics are for shaft sinking with the option to convert the hoisting system into a full time production hoist. The bottom line is that the effect of the anticipated changes in the hoisting cycle produces a noticeable improvement in the production capacity of the winder.

It is note-worthy that for a typical wet shaft the effective monthly tonnage is given by: (1+w) x tonnage elevated to surface (where w is the ratio of water present by mass usually 0.1) Therefore the best achievement of such a winder is 90% of the rated capacity.

2.2.2 Calculation of Rope Sizes

Skip Factors

After determination of the payload, the total end load on each rope of the winder is calculated. If *s* is defined as the skip factor where

s= (Mass of skip (including attachments),m_s)/Mass of payload, P

$$s = \frac{m_S}{P}$$

Then end-load: $m_E = (1+S)\cdot P$ (1)

The skip factor can be as low as 0.52 or as high as 1.2. When low factors are used, the light construction of the skip results in maintenance problems. A factor of about 0.65 is considered stable.

Concepts of Rope Design

For a given strength of steel and almost completely independent of rope construction, the following rule holds: *The length of free rope that can be suspended is independent of the diameter.* This length is defined as the ultimate length of rope, i.e.; if the ultimate length of the rope is suspended from one end, the rope tension at the top is equal to the breaking tension. If the steel strength is taken into account;

then $L_u = c.\ \sigma_t$ ...(2)

Where L_u = Ultimate length in m, σ_t = Steel strength in mega pascals (MPa), c is a constant= $9.61 \frac{m^3}{N}$. For example with a steel strength of 1800MPa, the ultimate length is 17.3 km. In all cases of uniform rope design a statutory factor of safety is required. The safe length of rope that can be freely suspended is: $L_s = \frac{L_u}{f} = \frac{9.61\sigma_t}{f}$ (3)

Where L_s is the safe length in metres and f is the factor of safety.

If a certain length of rope is cut from the lower end of this safe length of the rope and an end load of exactly the same mass as the detached rope is substituted, loading conditions along the remaining portion of the rope do not change. Consequently as the detached length of the rope is the same as the end load and if this length is known the, mass per unit length of the rope can be determined and a reference to rope tables shows which diameter most closely fits these requirements [13,19]

These concepts are expressed analytically in Appendix 1.

2.2.3 Developments in Hoist Rope

1.Background

The dependence of the mining industry on the use of wire ropes for deep level mining has exerted a continual pressure on the wire rope industry for improvements and innovations. Organisations like Hagie Rand in South Africa and Bridon in the United Kingdom are among those in the forefront of the drive for new technology and innovation in wire rope manufacture.

2.Criteria for the choice of winding ropes

a) Rope construction

Experience plus experimentation has standardised the choice of rope construction for different types of winders. There are many exceptions to be found in practice and experience on these installations tends to influence current recommendations.

b) Grade of Steel

At the start of the century most ropes were made from wire having tensile strength of 1540 MPa. Higher tensile strengths were not favoured due to problems with brittleness. However current tensile strengths of 1 750 MPa and 1 850 MPa are now standard and winding ropes made from wire of a tensile strength of as high as 2 150 MPa are in use on a developmental basis. Due to the minimum factors of safety which are presently allowed, tensile strengths of 1 750 or 1 850 MPa are generally chosen for winding depths less than 1 800m. The use of ultra high tensile strength steel becomes attractive at depths greater than this. There is a definite increase in the end load, which can be obtained from higher tensile steel.

c) Factors of Safety

For drum winders, ropes chosen in accordance with statutory factors of safety give satisfactory performance. However, where balance ropes are used such as in a Koepe winding system, adherence to statutory factors of safety does not always give acceptable head rope performance and higher factors are frequently indicated.

3. Calculation of Breaking Force

This is achieved by the use of an efficiency factor. The formula below is used to determine the force:

$$F_b = \frac{m_{cl}}{(f.g)^{-1} - L\eta^{-1}} = \frac{m_{cl}(f.g)}{1 - \frac{L}{\eta}} = \frac{m_{cl}.f.g.\eta}{\eta - L} \qquad (4)$$

Where

F_b = Required breaking force in N

m_{cl}= Mass of loaded conveyance in kg

f= Factor of safety

L= Length of suspended rope in m

= Efficiency factor for rope, Nm/kg

And g = acceleration due to gravity (~ 9.81 m/s^2)

The efficiency figure given for the rope is an average figure; the actual varies slightly from rope to rope.

4. Economic And Other Factors Governing Choice of Winding Ropes

The factors affecting rope performance may be listed as follows:

Required output from shaft

Type of winder

Load between rope and drum

Rope construction

Drum to rope diameter ratio

Tensile strength of steel rope

Sheave to rope diameter ratio

Quality of manufacture of rope

Number of layers of rope in the case of drum winders

Type and pattern of drum grooves in the case of drum winders

Payload

Tread material in the case Koepe winders

Treatment of wire in the case of Koepe winders

Type and condition of guides

Rope installation procedure and maintenance

Maintenance of winder and sheaves

Factor of safety

Dynamic load range

Depth of wind

Use of keps

Speed of winding

Acceleration and deceleration pattern

This list, which might not be complete, gives some idea of the complexity of the variations, many of which are interdependent- for example the normal or tangential distance between the Koepe drum and its deflection sheave (if used) and the difficulty in assessing the effect of varying any one of them. Some of these variations become basic data around which other variations depend.

For example, depth of wind, output from shaft and the type of winder form the basis on which the requirements for the rope result, in addition other parameters must be varied to achieve optimum rope performance. Of these factors the most important ones are the drum to rope diameter ratio, force between rope and drum and the effect of speed on rope performance for a given drum to rope diameter ratio.

The drum to rope diameter ratio is a convenient design factor for controlling the bending stresses to which a rope is subjected, when being wound on a drum, as load or force between rope and drum[18,19]. The difficulty in assessing the stresses caused by bending is the reason these stresses are neglected when deciding on a rope to be used for a given application. The statutory factors of safety specified for various types of winders take this absence of knowledge into account and are therefore higher than would be the case if these stresses were accurately known. An added complication is the effect of rope speed on the magnitude of the bending stress and possibilities of fatigue due to variation in tension or compression or a combination of both.

For an example when the 180° wrap over a Koepe drum is increased by introducing a deflecting sheave wheel, the tangential distance of the rope(s) between drum and sheave(s) should be such that no damage is done because of counter bending.

The other factor which is dependent on the drum to rope diameter ratio is the load or force between rope and drum. Multi-layer coiling increases this load.

2.2.4 Developments in Shaped Wires for Locked Coil Hoist Ropes

While the first response to the demand for stronger ropes for deep- level mining was the ultra high strength triangular strand ropes, the continued increase in mining depths has introduced another view. Improvements in wire drawing and die design have resulted in increasing the available tensile strengths of shaped wire, together with improved wire toughness and property consistency. Currently in the United Kingdom nearly 100% of mine hoist ropes are of locked coil construction. The locked coil construction has a number of advantages over the stranded type ropes [20]:

- Size for size they are of greater strength than conventional stranded ropes in the same tensile strength range.
- The smoother exterior surface gives greater resistance to wear by abrasion.
- They are virtually non-rotating.
- The elastic and permanent stretch is less than that of stranded type ropes.
- The modern centre of the locked coil hoist ropes reduces internal cross cutting.
- The layers on interlocking wires offer improved retention of manufacturing lubricant and prevent ingress of corrosive elements.
- They are dimensionally more stable than stranded ropes and particularly advantageous in multi-layer winding.
- They can operate under much higher radial loads/forces than other types of rope.

These factors make locked coil ropes suitable for drum winders in rope guided shafts and Koepe winders operating in deep shafts where torsional stresses in six strand ropes and internal cross cutting in multi-strand ropes reduce fatigue life. However there are two major drawbacks to the use of these ropes:

- Because of their compact cross section, it is desirable that the ratio of the drum or sheave diameter to rope diameter should be in the region of 100-120:1 to obtain satisfactory service life on main shaft winders.
- Reduced ratios have been tolerated with smaller ropes but generally a reduction in service life results. Stranded ropes depending upon hoisting speed have the ability to operate at D:d ratios down to 80:1.
- The use of locked coil ropes therefore necessitates higher capital expenditure on winding equipment.
- Owing to restrictions in wire tensile strength in both full lock and half lock wires the strength to weight ratio of locked coil ropes tend to be inferior to that of stranded ropes where round wires of higher tensile strength can be used. Therefore a higher percentage of the ropes' breaking load is used in supporting itself, thereby reducing the available capacity for payload.

Current Availability

Table2.1 Typical values of the wire tensile strength employed :

Wire Shape	*Tensile Grade*
Full Lock wires	1270 or 1370 N/mm^2
Half Lock wires	1470 or 1570 N/mm^2
Round wires	1770 or 1860 N/mm^2

(Minimum of grades).

Combining the above wire grades in Table2.1 and shapes offers nominal breaking forces as shown in Figure2.1. Figure 2.2 compares the nominal breaking forces with the weights per metre. The gradient of the graph (Figure2.2) is the strength to weight ratio, the average being 151kN/kg.

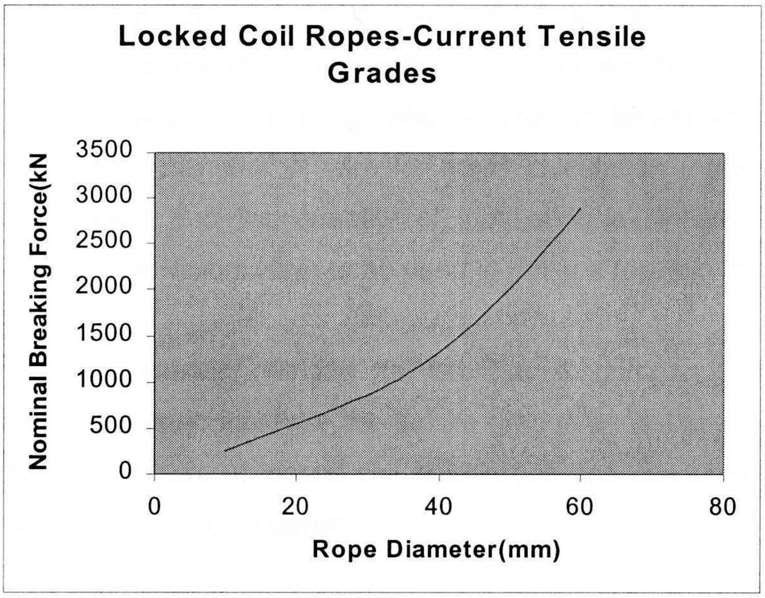


Figure2.1

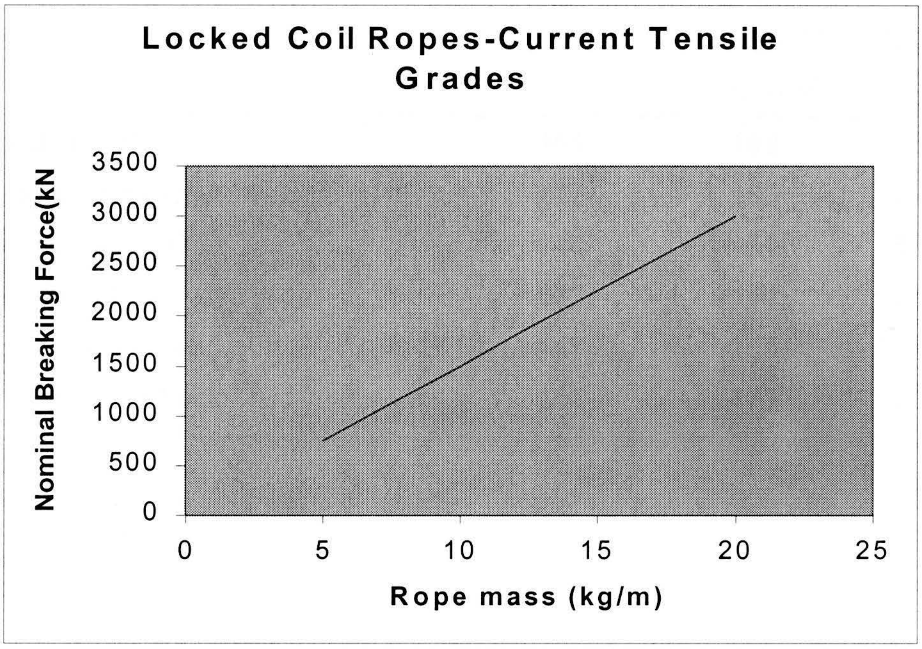


Figure 2.2

From Table 2.2, locked coil construction has lower values than a number of constructions particularly in the higher tensile grades of 1860 and 1960 N/mm^2. Table 2.2 gives comparable figures for other constructions common in mine hoisting. Factors such as spinning loss, position in tensile grade (minimum or minimum average), diameter of wire for mass calculation (nominal or maximum allowable diameter) and the quantity of lubrication all influence the tabulated values. So equitable factors have to be used to make a realistic comparison.

To address this situation Bridon (a UK based company) embarked on a development program to increase the tensile grades available for the shaped wire used in the manufacture of locked coil hoist ropes and concurrently improve wire toughness and property consistency.

Table 2.2 Locked Coil Construction Strength versus Other Common Constructions

Construction	TENSILE GRADE 1770	1860 N/mm^2	1960
Round strand	161	169	178
Triangular strand	157	165	174
Multi Strand (17)	144	151	159
Multi Strand (34)	142	149	157
3 Layer Oval Strand	147	155	163
18 Strand Fish-back	164	173	182

2.3. Tapered rope

It will be realised that since 1912 no work has been recorded except that by Professor Holtzgolh (Przibram mines Austria).

He proposed the use of galvanised tapered rope but was unable to design proper and realistic dimensions of the drum. The limitations then bordered more on the technical capability than the financial implications. There were mainly two approaches to the construction of tapered ropes.

2.3.1 Construction of the Tapered Rope

The following is a translation on taper rope construction from Professor J Hrabak's [6] work on wire ropes entitled *Die Drahtseile*. It can be seen that the manufacture of such ropes was not as complicated as previously claimed: -

" When hoisting is done from very deep shafts it is necessary to use tapered ropes in order to avoid excessive weight in the rope itself. Long suspended prisms of equal strength throughout follow in profile the curves of a cubic parabola. It would be a vain attempt to reproduce the cubic parabola profile in constructing a taper rope. Taper ropes are made in this way: - The lowest section is made in uniform and very long wire strand section(e.g. 300 to 400m), with a certain number of wires of proper thickness.

The upper portion of the tapered rope is made of uniform portion of greater diameter, all portions being equal in lengths, say 200m each.

In practice the section of the taper ropes is varied in two ways: -

- By decreasing the number of wires while retaining the same diameter of wire.
- By decreasing the diameter of the wires while retaining the same number of wires.

Such changes are effected in the strands making up the finished rope.

It was also established that the second method is better to adopt because when a strand is laid, it is impossible to alter the machine to lay, for example, an 8 wire, a 7 wire, and 6 wire strand, equally well; the same spinning head which handles 8 wires would then have to handle 7 wires and then 6 wires and the spinning would not be uniform. Furthermore there would be irregularities when the strands are spun to form the rope.

However the manufacture of tapered rope in this manner depends on unchanged conditions. Only certain systems of rope construction, (and not necessarily of the best), are adapted to this method of producing taper ropes.

The second method produces perfectly reliable ropes if the rope laying machine is properly designed and the installation in which the ropes are to be used are provided with drums and sheaves of correct diameters.

In the lead and silver mines at Przibram in Austria, taper ropes constructed as in the second manner have been in use for a quarter of a century until the mine was closed. They were hoisting originally from two shafts each 1000m deep and later from 5 shafts, which considerably exceeded 1000m in depth. The ropes were constructed of rolled and drawn cast steel wire with ultimate strength of 180kg per sq. mm.

A large and varied experience with these taper ropes has proved that this method of manufacture is satisfactory. The manufacture of the taper rope in this way does not differ in practice from the manufacture of ropes of uniform section. In the ordinary process for the uniform strands, the machine must still be stopped whenever a spool of wire runs out and another spool is substituted. In such cases flattening their ends, lying on top of each other and binding them together with thin copper or brass wire. The joint is then brazed and finished round. The wire is then stretched to the proper tension and the machine is started again. In a similar manner the smaller wires are brazed on to the larger ones when making strands for the taper ropes, only these operations are made to occur at specific regular intervals.

The size is greater than the length of the section of rope to be made from it by design. When the brazing of the smaller wires begins, the brazed joints in the wire where the diameter changes are distributed over a length of say 10 to 20 m of the strand depending upon the number of wires.

2.3.2 Use of the Taper Rope in Mine Winding

The laying up of the strands into the rope is exactly the same for the taper as for the uniform section ropes. Reservations were expressed on the manufacturing process of such a rope as stated earlier in this section. The strength of the brazed or welded joints at all the points where the diameter changed posed many problems, which remain to be solved. This was the view generally expressed by the follow-up analysis undertaken by mining engineers from the Witwatersrand mines. As implied in one of the resolutions during a seminar in 1911 [6,23]: -

"With the acid water of Witwatersrand mines, the electric couple formed at each brazed point would soon lower the factor of safety below 7, and the rope would be just as any other spliced rope." This meant that there was scepticism and these practitioners felt that these were merely spliced ropes. However records are available which prove that brazed taper ropes lasted for a long time in mines known for their acidity possibly due to routine oiling or tarring. This is because the wires forming the full cross section, required for the factor of safety adopted, at the point where wires are 'dropped', ran clear through from the upper end of the rope. On the other hand in a spliced rope, there is a break in a wire where it should be continuous.

Records indicate that the ropes have been used successfully up to depths of about 2000 metres. [23] It is recorded that the use of taper ropes was discontinued primarily because the depths encountered during the period in question could still be managed by using the usual ultra high strength steel with no significant effect on the payload/dead-weight relationship. The use of these ropes with sheave winders is not possible for great depth. In the case of great depth, the weight of a taper rope, which is only possible to use with a conical or cylindro-conical drum, would certainly be much less than that of a rope of uniform section, the reduction in many reported cases being as much as 50%.

2.4 Synthetic Rope Design In Hoisting

The USA Bureau of Mines and the Colorado School Of Mines have suggested a design of composite wire intended to drastically reduce the specific weight of the rope and thus, without compromising on the factor of safety, achieve higher payloads. This is achieved using synthetic material like nylon and other related material but no results have been obtained on carbon fibre. Carbon Fibre's strength properties must provide some improvement in the strength of any rope constructed from using it.

2.4.1 Advantages of Synthetic rope over Wire rope

1) Synthetic rope does not corrode in mine atmosphere
2) Low weight of the synthetic rope vs. steel allows greater payloads especially in deep mines.

2.4.2 Disadvantages vs. Wire ropes

1) Synthetic ropes are soft and easily abraded by sharp objects.
2) No method of determining the time to retire the rope, i.e. failure mode and prediction of its onset.
3) Greater dimensions are necessary

2.4.3 Construction Methods Applied.

Construction of synthetic four-strand high strength polythene rope was undertaken but later found faulty because of its nylon core. With continued use, the rope creeps but the nylon core returns to its original shape and size. This leads to imbalance between the rope strand length and its core thus causing `bird caging'. A design with a core of polythene only as the outer strands performs better because the creeping effect is uniform.

Another construction had alternating long and short strands on the outside of the rope like the Warrington Construction Steel Strand[25]. Small strands are forced out and hence wear quickly. Placing additional strands on the outside of the rope produce a construction similar to the Warrington Seale Construction steel strand.

This leads to 2 advantages:

1 Rope wears out more evenly and lasts longer

2 Design makes it easy to recognise when to retire the rope.

A major limitation in the use of polyutherane ropes has been its low resistance to abrasion.

2.5 Flat Ropes

These are made up of a number small ropes which may be considered as strands, which are laid up side by side and then stitched together. 'Single stitching' in which single wires are employed results in a compact rope of minimum width. 'Ordinary stitching' is effected, using a strand in place of the single wire, but results in a wider rope.

The use of flat ropes has been practically eliminated in Europe even though some mines still use them for support services for example the `Mary Anne' in Southern Africa.

The weight of a flat taper rope will be about more than 20% heavier than that of a cylindrical taper rope for the same load. This increases the weight and the moment of inertia and another disadvantage is the high wear rate of the stitching and the consequent necessity for re-stitching at frequent intervals.

The weight and wear rate, discount any advantages the flat rope might have on other rope constructions which is high torque, low speed at beginning of winding and lower torque and increasing speed as each layer increases the diameter within the reel.

CHAPTER 3

HYDRAULIC HOISTING OF MINERAL CONCENTRATE

This chapter dwells on the general overview of hydraulic transportation, horizontal, incline and more intensively vertical. It aims to make an evaluation of possible hydraulic based vertical transportation options. An analysis of current and past applications of hydro-hoisting is done with a view to evaluate the operational capabilities and feasibility of these methods in real mining situations.

While the principle of hydraulic ore hoisting is well known, it is rarely adopted in most hard rock mining operations. Practical experience in the application is very limited regardless of the expected benefits to be derived from such a reduction in shaft dimensions and a more flexible transport layout.

3.1 Parameters controlling economic hydraulic transport

The initial capital investment of a hydraulic transport system is mainly influenced by the price of pumps and pipeline steel. The major operating costs are power and maintenance supplies like pump parts. Three different duties are to be considered: horizontal transport; vertical hoisting; and transport combined with hoisting in an inclined pipeline.

3.1.1 Horizontal : Ample information is available on horizontal transport and a number of mining operations use horizontal pumping to varying degrees either for concentrate or tailings. Rules of thumb have been proposed which allow a preliminary estimate of optimum slurry feed conditions to minimise total costs.

Low velocities can give rise to undesirable settling of solids which will lead to pipeline wear and /or blockage. For large particle sizes, high transport velocities are necessary to prevent settling of solids. Large velocities and large particles imply high wear rates and excessive power requirements. These latter two disadvantages more than offset the gain which might be expected from having an inexpensive small diameter pipeline carrying solids at high velocity. At high solids concentrations or very fine particle sizes the transported slurry can be very viscous and considerable power is necessary to overcome friction heads. At low solids concentrations an expensive large diameter pipeline is necessary to cope with the required tonnage of rock to be transported.

The wear rate of pipeline steel due to abrasion is related to solids concentration, particle size, transport velocity and pipe diameter as shown below:

$$W \alpha C_v^2 d_p^{3/4} v^{3/2} d_{pd}$$

where W= change in pipe thickness per unit time

C_v= volume concentration of solids

d_p = average particle size

V = transport velocity

d_{pd} =Pipe diameter

From published data it appears that for 150μm particles in a pipeline delivering 400 000t of rock per year the linear wear would be about 200μm per year. If the above relationship is applied to an average particle size of about 1mm the wear would increase to about 1cm per year. In applying this relationship it needs to be realised that the velocity must increase with particle size to prevent settling of solids, and that this high wear rate for 1mm solids results from the large particle size and the increased velocity necessary. It is thus clear that particle size has a very marked effect on the cost of horizontal hydraulic transport.

3.1.2 Vertical Hydraulic Hoisting

The primary consideration is the nature of the system employed. There are two basic systems and both have been tried in the South African mining industry. The first being the open system, which uses high lift pumps at the lower end of the pipeline system to move the slurry. The second is the closed system or recirculating system, in which advantage is taken of the static pressure of a return leg of water to allow a high hydraulic efficiency. The availability of high lift pumps has made the open system practical. These pumps are expensive and the maintenance costs are also high. The recirculating system needs a discontinuously operated lock hopper for efficient operation and ensuring the structural integrity of this vessel under the high static pressures involved can prove difficult.

Comparison of the two systems shows that the open system has the advantage of :

- requiring only a single high pressure pipe line and associated valves;
- being more readily suitable to be made fully continuous; and
- being more efficient in its use of water (the hoisting pipeline in the second system must effectively be pumped clear of solids during each cycle).

The second system requires a lower pumping pressure operating on clear water because it;

- can tolerate the presence of quite large particles (pump dimensions limit the size of particles which can be handled in the first system);
- can operate over larger total lifts than the first which is limited to a total lift of about 900m per stage.

The slurry to be fed to a vertical pipeline system should be as high density as possible. For a given hoisting duty, the increase in total pressure caused by increasing the slurry density is more than offset by the reduced diameter of the high- pressure pipeline.

Particle size is not particularly critical in its influence on hydraulic efficiencies in a vertical system because the hindered settling velocity of particles as large as 1cm is far lower than the transport velocity. There is, however, a limitation on particle size in the open system, which was noted earlier in the previous section. In addition, the high- pressure pipe column is a relatively costly item and abrasive wear becomes very important for large particle sizes.

There appears to be a considerable incentive to reduce the size of the solids to be transported to about 100μm to ensure that the abrasive wear is minimal.

3.1.3 Incline System hydraulic hoisting:

There is relatively little information or data on any work or research being undertaken, however, preliminary studies indicate that slurry feed conditions are probably similar to those needed for horizontal transport.

3.2 Complete or partial underground mineral processing

It is important to note the need for some degree of underground size reduction, or better still, some degree of mineral processing underground so that a concentrated product can be transported to surface thus saving on transportation of waste. With the improvement in mineral processing technology and pumping machinery, hydraulic ore hoisting could be a worthwhile proposition, particularly if the current system of stage winding in deep level mining is taken into consideration.

Moving the mineral processing stage to underground implies that:.

- An adequate and efficient ventilation system will be necessary for the heat and effluent and any gasses arising from the processing stage. The water reticulation system must be well planned to ensure that groundwater is not polluted by any effluent discharged from the processing stage.

- The design and use of intensive and high capacity reciprocating or centrifugal machinery which is preferably corrosion resistant to decrease maintenance and/or replacement due to the possible high temperatures and acidic mine water underground. This saves space and also reduces wear in highly reactive environments.

This is in contrast to the extensive surface processing facilities. The volume of material hoisted to the surface is reduced to the minimum.

- This allows an increase in the ore tonnage's being mined and handled without increasing the hoisting capacity of the shaft. For green-field installations, the hoisting installations could be smaller.

- The rate of excavation could therefore be increased significantly bearing in mind that shorter distances will be involved in the transportation of the ore from the producing face to the processing and concentration stage and translates to a higher productivity.

- Waste and tailings can be handled underground and as such there is no need to haul material to surface and then return it underground for back filling and other support services.

It follows that there will be a need for a transport system underground to raise the tails or fill material to a higher level so as to achieve effective hydraulic back-filling assuming wet fill is used. It is important to consider that excavated material has a swell factor that will influence the amount and volume of fill and as such the percentage of the excavated material that could be accommodated in the mined-out stopes.

The importance and link between underground mineral processing and back filling must be emphasised.

Back filling is normally achieved hydraulically especially at deep levels and the fill material must have good compressive strength and be easily drained. The fill must also be capable of being:

a /. Placed hydraulically without the addition of cement,

b /. Be sufficiently stiff to minimise closure in stopes where the virgin stress may be as high as 100 MPa and

c /. Is stable under most of the dynamic stresses encountered.

This is a benefit for strata control in worked out areas. For extensive back filling carried to within 5m of the face and using a low void-ratio material, the energy release rate at a depth of about 4km could be reduced to about 75%. This in turn will reduce the rock-burst hazards by over 80% in every unit surface area exposed. As the fill is generated underground, as tailings, there is a strong incentive to backfill as extensively as possible to save hoisting waste.

The other benefits drawn from back filling, other than good strata control, are:

-Better control of ventilation air.

-Reduced surface area exposed for heat pickup.

-Marked decline in the timber demand resulting in a lower fire hazard.

There is also better environmental protection, due to controlled metallurgical mineral processing. The entropy does not increase as fast as it would on surface and this produces less impact on global warming.

As the temperature underground is below 100 ^{0}C and the pressure is reasonably high (several hundred bars), consequently high pressure and low temperature reactions are favoured such as, hydro metallurgy with pressure leaching, pressure grinding, pressure flotation and filtration.

Hydropower could be used direct or be transformed into electrical energy, such energy could be used for cooling and other mining excavation and processing as well as hydro transport. JCI have been and are researching on this technology and there is scope for further research in this field.

3.2.1 Implications of Underground Comminution

This implies relocation of some or all of the milling operations. However this will also mean a new design of the equipment and processes involved because of possible space restrictions underground compared with the surface installations.

However this might also be an advantage to those processes underground which are dependent on subsequent mining processes as stated in the previous section such as reduced ore tonnage hoisted and backfill etc.

3.2.2 Current Status in Hydraulic hoisting.

By 1972 Vaal Reefs in South Africa was transporting about 25 000 tons per month of gold ore at 16.5% by volume with a slurry particle size of 2mm from a 2 200m depth using a pipe diameter of 152mm diameter[1,2].

The following are some of the expected advantages of hydraulic hoisting: -

1. Smaller shafts are required because the pipes to hoist the same quantity of ore hydraulically require less space in the shaft than skip hoisting equipment.

2. The hoisting capacity of a mine can easily be increased without the necessity of sinking a new shaft. Large savings in capital costs can thus be achieved.

3. Easy to fully automatise, hence less manpower is required.

One of the factors militating against the implementation of hydraulic hoisting has been the difficulty in designing a satisfactory solids feeder for introducing the solids into the pipeline. A number of feeder systems were experimented with.

An ideal solids feeder should be capable of introducing the solids at high rates and concentration into the pipeline. In addition it should be reliable and have few moving parts in order to ensure minimum wear.

These feeders can be classified as: -

3.2.2.1 Feeders for fine solids-

With particle size smaller than 4mm. These feeders consist of the container in which the fine solids are mixed with water to the required concentration. The mixture is then pumped either through the pump or by some indirect means, thus avoiding it having to pass via the pump. To combat wear, different types of pumps and pumping systems have been developed. -

Centrifugal pumps modified for pumping slurry.

1. The mars pump-developed in Japan by Mitsubishi installed at Vaal reefs Gold mine in South Africa.
2. The Jupiter Pumping system- similar to the Mars pump, it was developed by Hagie Rand.
3. The Hitachi slurry Hydro-hoist-developed Hitachi Ltd. in Japan and similar to the Jupiter pumping system.

3.2.2.2 Feeders for Course Solids: -

There are two basic types of solid feeders by which course solids are introduced into a pipe, namely:

Mechanical feeders

Lock Hopper feeders.

Both systems introduce the solids down stream of the pump so that the solids do not come into contact with the pump.

Mechanical Feeders- the following are three better-known types of mechanical feeders:

1. Plunger type: pump draws water through the hopper and at the same time solids enters the pocket in the plunger. A screen at the bottom of the pocket prevents solids from entering the suction line. Figure 3.1 illustrates this.

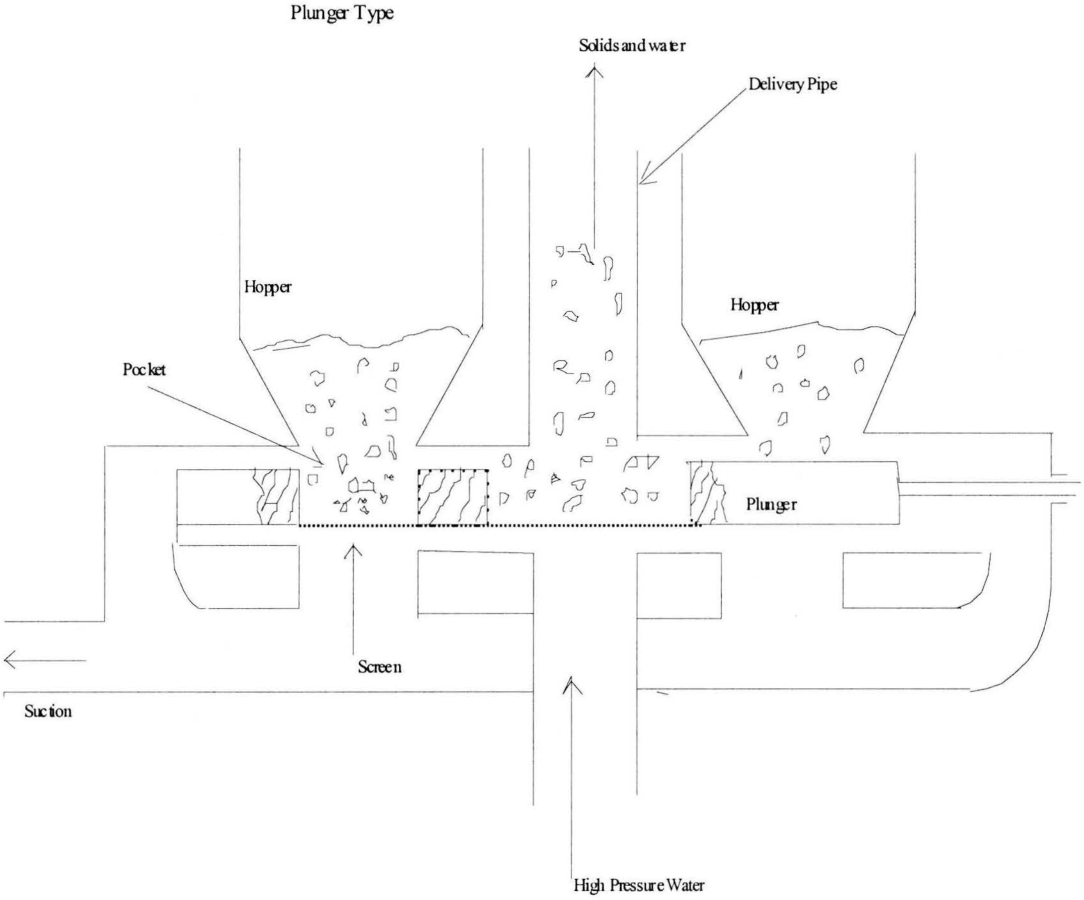


Figure 3.1 - The Plunger Mechanical Feeder

High-pressure water from the pump is simultaneously pumped up through a second pocket and solids are passed into the delivery pipe. Due to the reciprocating movement, the filled pockets are brought into the high-pressure stream while the emptied pockets are brought into the suction line.

2. Rotary Type: The operational principle is almost similar to the plunger type except that the pockets to be filled with solids are not situated in a reciprocating plunger but a rotating cylinder. Figure 3.2 illustrates this.

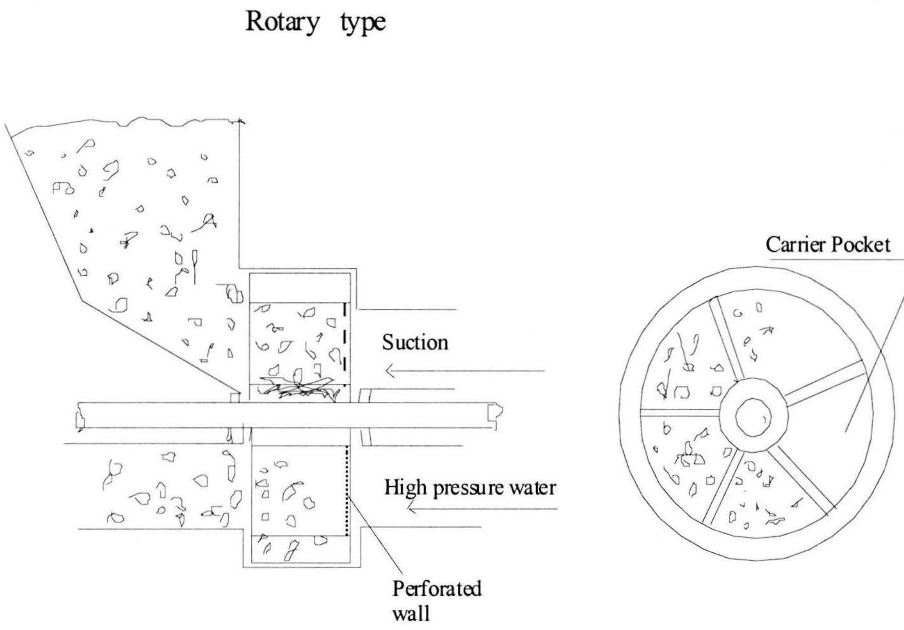


Figure 3.2- The Rotary Mechanical Feeder

3 Hitachi Hydro hoist:

Comprises of three pipe feeders which alternate by way of a system controller. Siemag Transplan, in Netphen, Germany, has developed a more modern and robust version which is now in use. The 3 chamber pipe feeder (RKA), pressure lock with constant inside diameter. The mode of operation is comparable to a Triplex pump without a piston.

The transporting procedure is controlled by a high-pressure water hydraulic system which acts directly onto the mixture to be transported. This mixture is fed into the low-pressure side of the 3- chamber system.

Therefore the high-pressure pump uses clear water only to displace the slurry mixture. This uses pressure lock systems in which solids of up to 100mm and high concentration utilizing pressure of up to 250 bar may be transported through pipe system.

This system is subjected to a low wear rate even when highly abrasive medium is in use. Continuous flow has been proved for shaft depths up to 2000m and horizontal distances of 20km without the use of intermediate pump stations.

This same principle is being used to transport refrigeration water underground using the pressure lock mechanism. This has also been used to feed ice to underground heat exchangers. In this method it has been recorded that energy recovery is possible from the downward limb of the pumping circuit.

The possibility of combining deep level cooling and transportation is therefore a very realistic proposition considering the above developments.

4. Lock- hopper feeders:

Solids are fed into a large drum or hopper that is closed and pressurised. The solids then enter the high-pressure pipeline from the pressurised hopper. Figure 3.3 over-leaf is an illustration of the Lock Hopper mechanism in coarse particle hydraulic transportation.

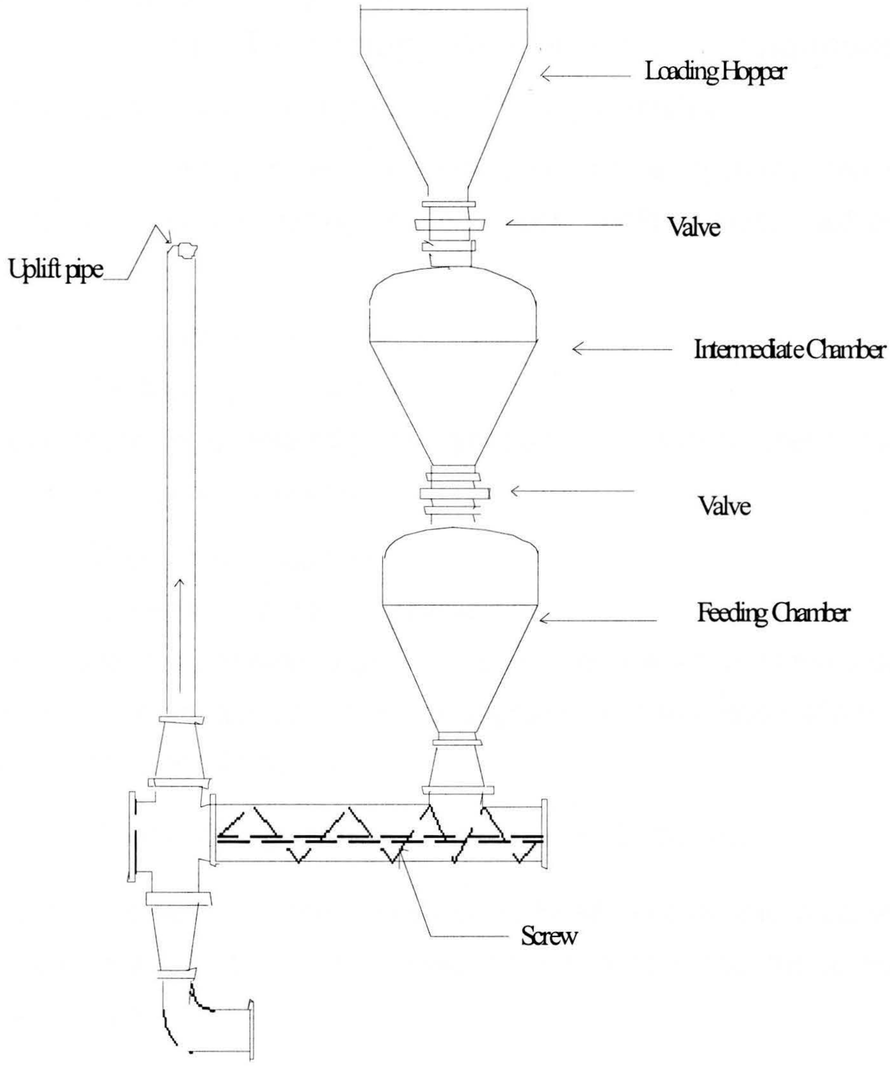


Figure 3.3- Lock Hopper Feeder

Mechanical versus Lock-hopper feeders:

From a mining point of view it is desirable to pump as large a particle size as practically possible. Mechanical feeders often suffer from serious sealing problems and high wear rate. Lock-hopper systems are not normally capable of high throughput. (ie. high concentration). With the hydro-lift system solids can be introduced at high and accurate controllable rates into the pipeline.

3.3 Airlift or Jet Pump System

3.3.1 Adapting the Technology of Deep Sea manganese mining to deep level hard rock mining vertical transportation:

The airlift and jet lifts are two possibilities of vertical hydraulic conveyance whose applicability in deep-sea mining is very good. Both systems exhibit two distinct disadvantages:

1) Low efficiencies
2) High energy consumption

However these are countered by two great advantages over other possible systems of deep level vertical transportation:

1) Continuous operation
2) low wear and extreme simplicity

These two factors combine to give high degree of availability of the system versus the low efficiency cited. It is against this background that adaptation of the method to hard rock operations should be tried.

3.3.2 Mode of operation of Jet pump System:

The system is based on the conversion of kinetic energy into pressure energy. The transport system consists of a clear water pump, a jet device, the suction pipe and the discharge pipe.

In order to lift solids the water velocity in the suction pipe and in the discharge pipe must be greater than the settling velocity of the solids. Only then transportation of solids starts which can be raised further by an increase of the primary flow.

The optimum operating point for the plant would be the intercept of the jet pump and discharge characteristics. The calculation of the jet pump and the conveying pipe is done by a corresponding force balance. For practical application of the Jet pump, two arrangements are possible[7]:

1) The clear water pump can be positioned below water level as close to the jet device as possible.

2) The pump can also be placed above the water level on surface. The first proposition, submersible pumps can be used but their maintenance at such levels will be costly.

On the second arrangement a second pipeline is required which will raise the capital cost of the installation plus higher power consumption due to losses in the long supply line. A submersible pump is found to be more desirable than a surface pump.

Experience in the use of airlift and jet lift techniques has been confined to the marine industry and the adaptation to deep level hard rock mining has not been tried anywhere. The task or question would be how to simulate the deep level marine conditions in hard rock operations. The advantage with marine mining conditions is that there is adequate hydrostatic pressure, outside the pipe provided by the water, to balance the pressure generated inside the pipe. The other obstacle would be the particle size of the mineral being transported and the control of the discharge due the high velocity as the material moves up the pipe to surface.

A great deal of research has been given to the hydraulic transportation of ore and follow-up testing has been undertaken. Through the application of Bernoulli's equation, experiments results, like the equations for the Total Developed Head (TDH) over a vertical pipe, required pumping pressure to overcome head losses over the pipe due to friction can be predicted. Solids in slurry tend to move towards the centre of the pipeline during vertical flow due to the hydraulic lift. The friction caused by the slurry near the pipe wall therefore approximates to that of clear fluid flow. Based on this observation an equation was derived which provided the expected relationship between the pressure loss ***p*** over the vertical pipe, the required production rate and pipeline diameter d_{pd}, for a given maximum particle size. The diameter , d_{pd} determines the slurry density ρ_m and the mixture velocity V_m. The pressure loss ***p***(in pascals) is given by:

$$p = \rho_w \cdot g \cdot \rho_m \cdot z \cdot \left(1 + \frac{\mu_w \cdot V_m^2}{2 \cdot g \cdot d_{pd}}\right) \text{ (in Pascal)..............(5)}$$

Where ρ_w is the density of the carrier fluid, ***g*** is the acceleration due to gravity and ***z*** is the vertical distance and μ_w is the friction factor of the carrier medium. Work done on pilot plants verified equation 5. The equation does not take the effects of irregular shape nor the particle size distribution into account.

During vertical transportation there is settling of solids at the terminal settling velocity v_t.. As a result hold ups are inevitable and this reduces the delivered slurry at the discharge compared to the inlet concentration. Equation 5 was used in pilot test plants to determine the required pumping performance but in order to find the delivered concentration, the effect of the settling behaviour of the solids has to be calculated. The upward velocity of the particles is approximated as the flow velocity of the fluid minus the terminal settling velocity of the solids. This is a rough estimate because it assumes the settling of one spherical particle yet there is a collection of the particles with a great deal of interaction. The effect of a settling collection of particles results in the hindered settling velocity v_t', i.e. the settling velocity for the slurry resulting from the interaction between the particles in the slurry. Hindered settling velocity is defined as :

$$v_t' = v_t * (1 - C_v)^n \quad \text{.......................................}(6)$$

where C_v is the volumetric concentration of the solids in the slurry and n is a factor depending on the Particle 's flow Reynolds Number Re_p. While this relation will not correct the effect of the different size and shape , it allows a more accurate approach for determining the delivered concentration. With the hindered settling velocity v_t' , a prediction can be made for the delivered slurry concentration C_{vd}, using the in-situ concentration C_{vi}. The ratio of these concentrations is equal to the ratio of the mixture velocity V_m to the velocity at which the solids move(V_m- v_t'). This leads to:

$$C_{vd} = \left(1 - \frac{v_t'}{V_m}\right) \cdot C_{vi} \quad \text{...............................}(7)$$

CHAPTER 4 AN ALTERNATIVE HOISTING METHOD

This chapter briefly attempts to review the current development in this latest technology which is very much at a futuristic stage. This section considers the technical aspects of the technology and feasibility of the method.

4.1 Linear Motor Application:

This is a very recent technology still to be tried. It emanates from electrical induction motor theory and involves the following: The conveyance runs up and down the shaft and motive power is no longer supplied via the rope, as in traditional rope hoisting, but supplied by an electrical field induced between the guides and conveyance shoes. There is a very narrow gap between the shaft guide and the conveyance shoes and a magnetic flux is generated in this gap by alternating current induced through the guides.

In principle the guides act as an unwrapped (linear) stator while the conveyance shoes form the rotor that moves relative to the shaft guides. This whole operation requires a fine tolerance and precision due to the small size of the air gap required (a millimetres or so).

A very important feature is the need for cleanliness throughout the shaft. For that reason any spillage of solids from the conveyance is unacceptable. The method demands a great deal of electrical engineering input much as it requires the same from mining engineering. However concerns of power failure and braking systems and ground movement still are to be adequately tackled.

The definite advantage with this method is the limitless range of depth in which it is applicable. Furthermore it does not require a large array of engines and accessories as is essential with conventional winders. Also, it is more amenable to automation than the rope application.

Research is underway in the area of practical application of this technology at various mining research centres, not least of which is in South Africa where the weight of the rope for deep level winding has a deleterious effect on factors of safety. High costs involved in the design of this system hinder any current linear motor application research in deep level mining.

CHAPTER 5.0

SYSTEMS ANALYSIS

This Chapter is directed towards the evaluation and analysis of the reviews and overview of the foregoing options from preceding chapters. The analysis is done on a method by method basis and data including some results are included. A scenario study is used in one case where working examples of the option, to provide performance data, could not be found.

5.1 Rope Hoisting

In order to maintain or increase the payload to depth ratio as the winding depth increases a number of options have been considered. Below are some of the proposed methods aimed at achieving this:

a) Revised safety factors;

b) Enhanced winder system designs, including larger diameter drums, improved control systems, etc.;

c) Lightweight conveyances;

d) High capacity ropes

Strength to mass ratio of a rope determines its load carrying capacity in deep-level mining winding. This capacity can be increased by either increasing the rope's breaking strength or by reducing its mass, or by a combination of both strategies. So far most attention has been targeted towards creating ropes with higher breaking force and very little has been done to reduce rope mass and increase and or retain equivalent or higher breaking strengths.

5.1.1 Rope Winding- Wire Developments- Background

Combined tensile strength of the component wires is the principle contributor to a rope's breaking force. There is a limited scope for increasing the rope's breaking force by changing such design features as lay lengths, since these changes tend to impact adversely on other important rope properties, such as fatigue resistance, flexibility, etc. Furthermore the rope constructions traditionally used for deep-level hoisting- triangular strand and multi-strand non-spin ropes- exhibit efficiency factors (strength-to-mass ratios) among the highest available.

Limits of wire tensile strength

The strength of a wire tends to be a function of its size so that in ungalvanized plain carbon steel wire, the limits within standard wire drawing practices are shown below even though sizes and tensile grades are available outside these limits for special applications.

Tensile strength (Mpa)	Maximum wire size(mm)
1 900	5,00
2 000	3,60
2 100	3,15
2 200	1,50

Table 5.1- Tensile strength vs Maximum wire size

From the above table it shows that generally, the smaller the wire, the higher the tensile strength that can be achieved. However finer wires are less robust and experience has shown that the optimum range of diameters for the outer wires of triangular strand ropes being set at from 2,80mm to 3,60mm with 3,20 mm being considered the optimum. From this, follows the concept that refers to 'Ultra High Tensile wire'. Wires in this size range include tensile strengths of 1 950 MPa and higher.

5.1.2 Deep Level Hoisting Rope Types and Developments

Triangular strand ropes

Applications

This rope type is typically used as hoist ropes on multi-layer drums such as double drum winders, Blair multi-rope winders and similar. Triangular strand ropes provide the following features of relevance in these applications:

a) relatively easy to manufacture;

b) robust and stable;

c) high strength-to-mass ratio;

d) resistant to crushing(on winder drums);

e) good wear properties due to the shape of the strands;

f) acceptable fatigue performance.

Lay length change

In deep hoisting with triangular strand ropes at suspended length exceeding 2000m, the length changes are due to differential loading along the length of the rope caused by the rope's weight. The total force in any element of the rope between the headgear sheave and the conveyance is the sum of the total end load and the mass of rope suspended below that element.

This differential loading creates a differential torque along the rope length due to the helical form of the rope. Each element of the rope must twist about its axis in the appropriate direction until the torque is balanced, for equilibrium. The rope, typically, twists at the sheave end to lengthen the rope lay and at the conveyance end to shorten the rope lay. In shafts which are deeper than 2500 m, lay lengths at the sheave end as high as 80% longer than the manufactured lay length have been observed, while those at the conveyance end have been up to 30% shorter. Even higher unlaying percentages are to be expected when greater depths are encountered

Rope Distortion

Experience has shown that there has been occurrences of rope distortion where the rope develops a corkscrew at the part of the rope just entering the headgear sheave when the conveyance is at the lowest loading point, i.e. at the position of maximum twist and lay length extension. This has been proved true for winding depths greater than 2000m. However the exact reason for this problem has not yet been established.

Cores for triangular strand ropes

When some of the distorted ropes are repaired, it has been established that fibre cores in the distorted section are badly damaged. It has not been determined whether this is a cause or effect but it is possible that the twist in the rope at this point may be causing the damage in the fibre core. This effect casts doubt on the suitability of natural fibres to be used as cores in triangular strand ropes operating at great depth. Synthetic composites can be a viable alternative, for example polypropylene centre enclosed in an extruded polypropylene jacket.

Manufacturing Tolerances

Differences in actual dimensions between the various rope components may result in imbalances in the force distribution around the rope section, a further possible reason for the corkscrewing that cause rope distortion. The dimensional differences are related to the current practical manufacturing tolerances which can be achieved in triangular strand ropes. While such tolerances have proved adequate for the shallower depths, they may be unsuitable for ropes used in deeper shafts.

Traditional non-spin ropes

Applications

These are larger multi-strand ropes comprising of flat strands over an inner triangular strand rope. It has mostly been used as a stage and kibble rope for shaft sinking and has had limited use as permanent rope on drum winders

Features

The attractive features on this rope that makes it ideal for deep level hoisting are:

a) high strength-to-mass ratio;

b) acceptable wear properties;

c) very low torque characteristics, so that lay length changes will be minimal.

Caution will be required, however, to minimise the effects of the less attractive features of the non-spin ropes, compared with triangular strand ropes, viz.:

a) less robust, as a result of the relatively smaller wire sizes in non-spin ropes;

b) less crush resistant, although still better than many other rope types;

c) more complex to manufacture;

d) Unstable, e.g. corkscrews, birdcaging, etc., if not handled and maintained carefully.

Planned Developments

Further tests with grades of steel have been planned and some work is already in progress. In addition to this, further experimentation with this rope type in deep-level hoisting systems is planned to establish service life parameters and operational and maintenance procedures, especially inspection criteria.

5.1.3 Alternative Rope Constructions

In deep shafts, a large proportion of the hoisting load is generated by the mass of rope suspended in the shaft, yet much of the steel in the suspended length is superfluous. Generally, it is only necessary to provide enough rope strength at the conveyance to satisfy the safety factor requirements based on the total attached mass, i.e. the 'Capacity Factor'. However, at the sheave end, the rope must have sufficient additional strength to satisfy the required safety factor based on the attached mass plus the suspended rope mass, i.e. the 'Factor of Safety'.

Proposed amendments summary

Static Factor (SF)=4,5 Man./Material and Mineral (includes rope mass)

Capacity Factor(CF)= 8 Man./Material and Mineral (excludes rope mass)

5% reduction in SF for BMR winders.

If the mine complies with approved safety standards, the rope selection criteria for the rope SF will be given as: SF= 25/(4+L), L= max suspended length of rope in kilometres.

These are part of the new regulations that have been promulgated already, hopefully they will enable single lift hoisting from depths currently tackled by a multiple shaft system.

Tapered Ropes

Proposals have been made in the past to taper the rope in order to reduce the mass of the rope suspended in the shaft. The steel mass of the rope is reduced from the exterior, either by reducing the number of wires or strands progressively from the outside or by reducing the wire size or by adopting both techniques. This approach has, however, proved to be unsatisfactory because:

a) the change in rope diameter with length results in poor coiling on the winder drum;

b) a torque imbalance is set up along the length of the rope due to its varying cross section; this effect is enhanced if the fundamental rope design has an inherently high torque characteristic;

c) change in rope diameter can present difficulties in manufacture.

Tapered Mass Rope

A rope construction under development is intended to reduce or eliminate these problems. Based on a conventional multi-strand, non-spin rope, the rope mass is reduced by varying the cross sectional area of the steel within the rope while maintaining a substantially constant external rope diameter and torque characteristics.

The change of rope mass with length will be achieved by replacing internal steel wires with a suitable material which is less dense than the steel wires and which has adequate integrity to provide support for the remaining steel, so that they remain in the same relative positions in the rope cross-section. Material being considered for the lower mass section of the rope include natural or synthetic fibre, an extruded plastic material or similar. Carbon fibre is one alternative that could be investigated further.

An important consideration is the design of the transition zones at the junction of each pair of different mass section. In these zones, the lower mass material must be introduced into the rope in such a way that the axial symmetry of the rope components, the integrity of the structure and the dimensional characteristics of the rope are all maintained. A method to consider will be splicing in the second material to the respective wires or strand components. Alternatively, rovings of suitable material may be included in the interstices of the rope construction, with the rovings extending between the high and low mass sections of the rope for a sufficient length to maintain the structural integrity of the rope. The graphical representations* help to underline the possible benefits to be realised.: These are based on real data collected from operating shafts and extrapolated to produce the likely behaviour at the anticipated depths .

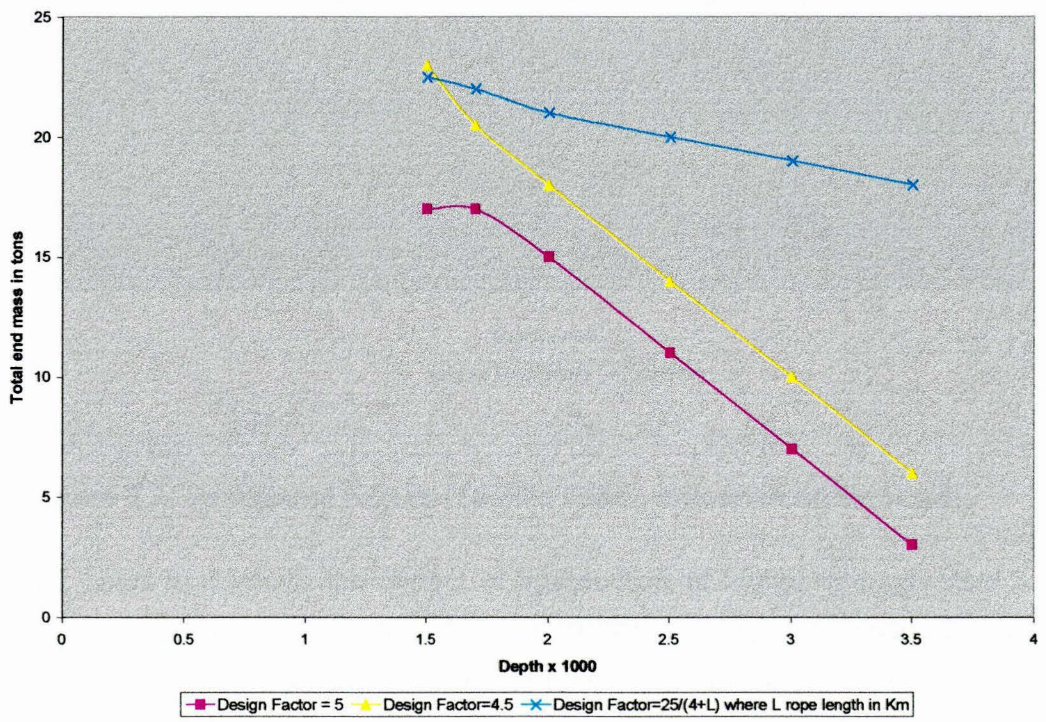


Figure 5.1- Design Requirements Influence on Endload

The factor, 25/(4+L), is determined from the work detailed in the SIMRAC contract document GAP 418. It seeks to introduce the effect of depth on the factor of safety and thus the depth has an indirect proportion to the factor. This also implies that the use of this

* generated from field data and extrapolation from Haggie steel ropes

convention allows greater loads to be tackled with the same rope that currently would be impossible due to present regulations. It is on the strength of these proposals that the hoisting in a single lift from 3000m and beyond should be possible especially if the new developments in rope manufacture are taken into consideration.

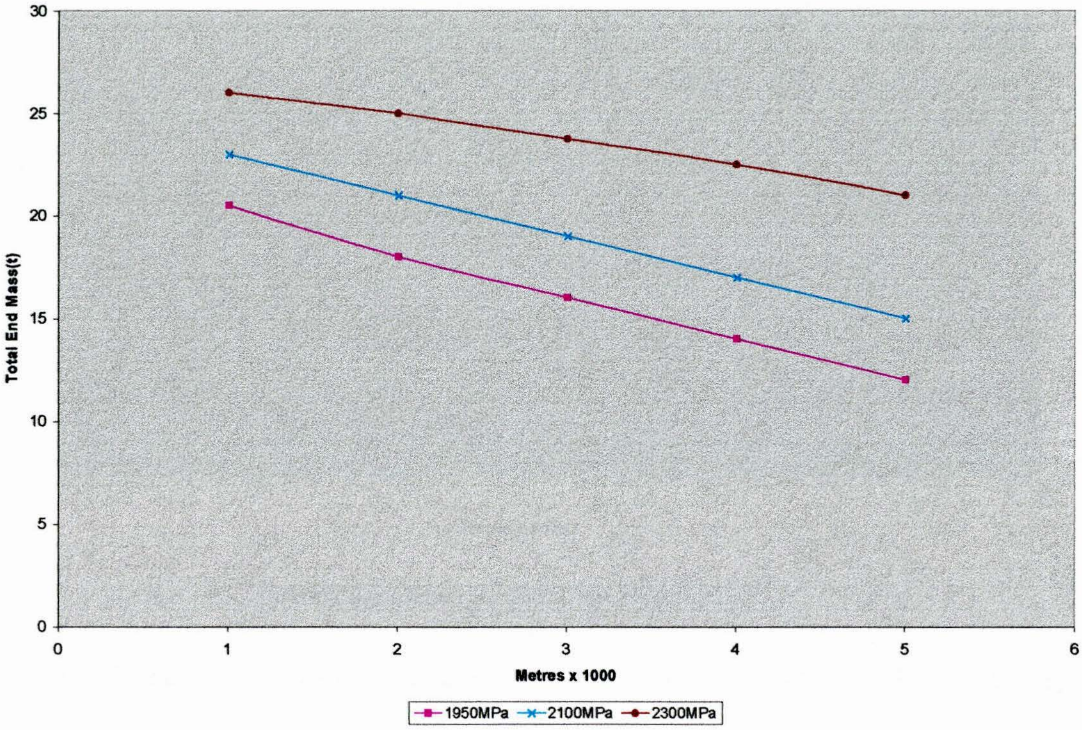


Figure 5.2- Permanent Ropes-(Tensile Grade Influence on Endload)

Figure 5.2 helps to illustrate the impact of introduction of higher strength steel in rope manufacture. In use now is mostly 1800 and 1900Mpa steel, the introduction of higher strengths like 2100mPa and 2200 mPa should result in an average of 15 to 30 % increase in payload.

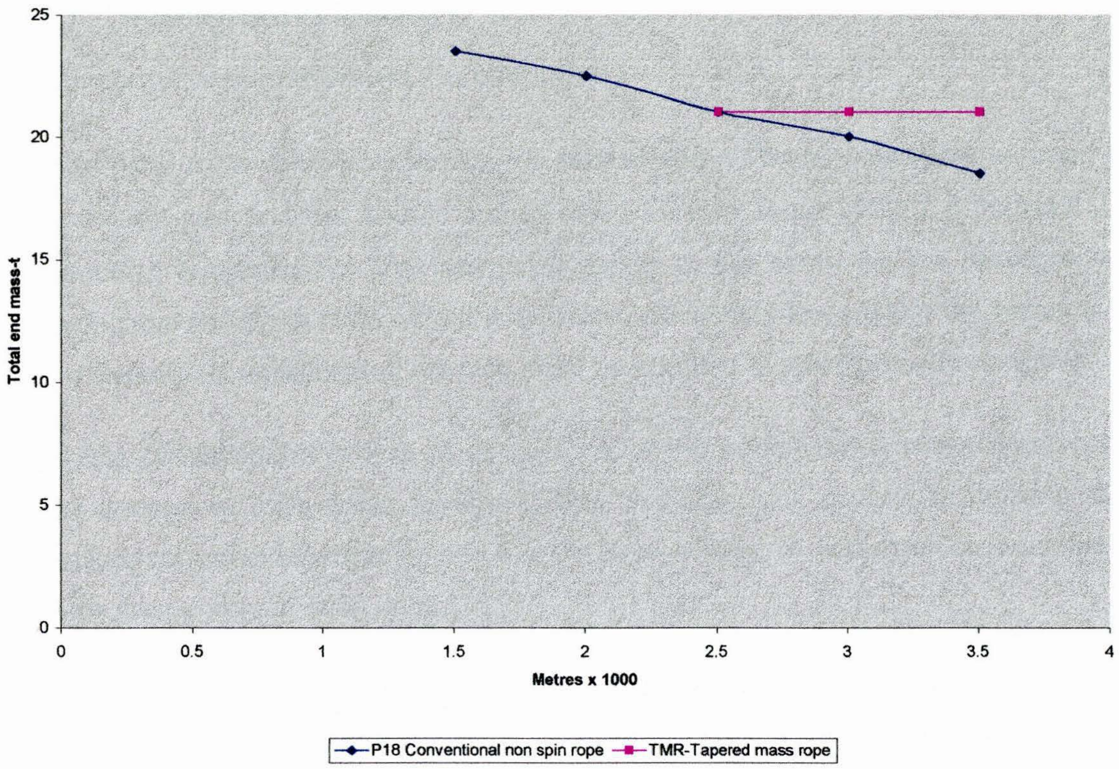


Figure 5.3- Tapered Mass Rope vs Conventional Rope(43mm 18 strand ropes)

Analysis of these graphical representations shows that there is a future for rope hoisting even up to depths in excess of the current proposals. It is also interesting to note that in Figure 5.3, this can still be attained without the introduction of a revised factor of safety as opposed to the first two representations. Tapered mass rope technology should give a very positive impact on many deep-level mining projects. It further adds more impetus to the prospect of single stage hoisting from ultra-deep level mining. However there is still more research required for the rope to be achieved structurally.

The tapered mass rope is no longer being pursued in South Africa due to manufacturing difficulties and also there is not enough benefit in terms of end load increase offered.

5.2 Hydraulic Hoisting

5.2.1 Underground Milling

Implications of Underground Comminution

Run of mine rock which may have a top size of about 12 cm, should be reduced in size until the top size is about 400μm and the medium size about 100μm, if the rock is to be transported hydraulically. The latter size range is the same as the reduction size in conventional reduction plant, and it is therefore necessary to consider the extent to which the conventional metallurgical practice must be adapted to underground conditions.

The first consideration is that of dimensions of the equipment and machinery must be small enough to pass down the shaft. This limitation runs counter to current trends in metallurgical practice which favour the increase in size of comminution equipment and reduce the number of such units.

Therefore the implication of this limitation is to use smaller equipment than is usual in today's surface plants. For instance, the largest jaw crusher which could be readily broken into parts which could fit this limitation would have about a 30x50cm opening. The largest cone crusher would have a head of about 1.2m in diameter. Fine grinding equipment such as rod and ball mills would be tiny by today's standards of surface installation and would probably require redesign for underground applications to enable it to be disassembled for transportation underground.

The prime difficulty arises, from the fine grinding side of the mill circuit. Rod and ball mills are large and have very low output per unit volume. As a result of this deficiency the CSIR mining technology (then COMRO) investigated the performance of centrifugal mills during the past twenty years. These mills are promising in as far as reducing the size and weight of machinery for performing a fine grinding duty. For example, a conventional mill capable of reducing the –2cm output of a cone crusher to a median size of 100μm at a rate of about 1000t per day would be about 3m diameter by 5m and have a mass of 50t. The equivalent centrifugal mill would be about 1m diameter by 2m and have a mass of 10t.

A prototype mill consisting of 18x30cm tubes mounted about a central axis parallel to the long axes of the tubes, and driven by a planetary-type gearbox was tested on a mine for a month and preliminary performance data is shown in the graph in Figure 5.4:

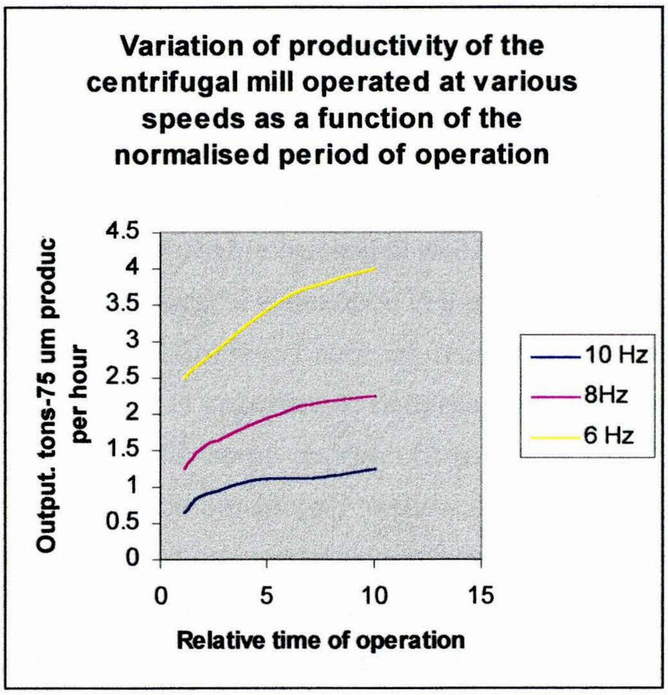


Figure 5.4-Centrifugal Mill Productivity Variation with Time of Operation

The above results obtained are in good agreement with initial predictions based on short-term tests and give confidence that these mills can be designed and scaled-up. The net power consumption in these tests was 25kWh/t - 75μm material operating with an equilibrium-sized grinding medium, which compares satisfactorily with the gross 29kWh found in full scale practice at the mine concerned.

The high wear rate in terms of kg of metal per ton of product is quite high and comparable to that in conventional mills. However, the mass of the wear pieces is naturally far lower in the centrifugal mill, which implies relatively short runs before replacement of the wear pieces becomes necessary, if the wear pieces are made of conventional materials. Attempts to use pneumatic rubber liners system to reduce wear by impacting and hard metal inserts to reduce wear by sliding abrasion have also been reported. It is important to note that in underground mineral processing, space for the ore being processed and machinery is a constraint. The machinery and plant has to be centrifugal high intensity machines as compared with the current extensive surface equipment and plant.

From the foregoing discussion it appears that consideration of what might be involved in crushing and milling will favour a number of sections each about 1000t per day capacity would be needed and that for milling section, some redesign of equipment would be necessary if the equipment is to be lowered underground and of size currently recognized as economic like the centrifugal mill.

5.2.2 Hydraulic Ore Hoisting-Current developments

While the principle of hydraulic ore hoisting is well known, it is rarely accepted in most hard rock mining operations. Practical experience in the application is very limited regardless of the expected benefits to be derived such as reduction in shaft dimensions and a more flexible transport layout. It is important to note the need for some degree of underground size reduction or better still, some degree of mineral processing underground so that a concentrated product can be transported to surface thus saving on transportation of waste. With the improvement in mineral processing technology and pumping machinery, hydraulic ore hoisting should be a worthwhile proposition, especially if the current system of stage winding in deep level mining is taken into consideration.

5.2.3 Power Requirements

Equations 5, 6 and 7 in chapter3 were used in establishing the expected production figures which in turn would determine the optimum operating regimes. From the operating requirements, The Total Hydraulic Power requirement Pr_v(in watts) is calculated.:

$$THPr_V = \rho.g.Q.i.L \quad \text{...}(8)$$

where ***i*** is the hydraulic gradient (unit m head/m pipe) and ***L*** is the length of the pipe. The values for factors ρ and ***f*** can be considered as those for the average slurry and the carrier fluid (generally water) respectively, due to the hydraulic lift near the pipe wall.

When ore is to be transported in a vertical direction, similar horizontal transport is likely to occur as well. The flexibility of a hydraulic ore hoisting installation allows it to be installed away from the shaft, decreasing the underground haulage distance. Slurry transportation of coarse particles in a horizontal pipeline requires considerable pumping pressure as the natural forces working on the particles now act perpendicular to the direction of the flow. [22] Horizontal slurry transport is divided into three separate areas (fully stratified, partially stratified and fully suspended). Horizontal transport of solids in a fluid requires fully-stratified flow. Under such an assumption using the same size of pipe line diameter for both horizontal and vertical slurry transport, the horizontal flow can be assumed to be fully stratified as the mixture velocity V_m is relatively low and the particles are relatively coarse.

For a fully stratified flow, most of the particles move as contact load, i.e. sliding, rolling or by saltation. The fluid pressure has to be sufficient to overcome the mechanical friction between solids and the bottom of the pipe wall. The extra pressure requirement increases the hydraulic gradient i over the horizontal section compared to the vertical section hydraulic gradient. The new gradient is the mixture hydraulic gradient i_m, and can be established using:

$$\frac{i_m - i_w}{\rho_m - 1} = B \quad \text{hence } i_m = i_w + B.(S_m - 1) \quad \text{.................................(9)}$$

where $B \approx 0.8$ for fully-stratified flow, when $V_m \leq 17v_t$. It is suggested to use $V_m \geq 4v_t$ to guarantee vertical conveyance of all solid particles, which verifies the assumption for B. With the calculated i_m the Total Developed Head (TDH_h) over the horizontal section of the installation can be calculated, as well as the Total Hydraulic Power requirement ($THPr_h$). Adding these figures to the ones found for the vertical section results in the total operating requirements for the hydraulic ore hoisting installation.

The Detour Lake Mine Project

Detour Lake Mine is an underground gold mining operation in the north-east corner of Ontario, Canada, owned and operated by Placer Dome Inc. At a horizontal distance of roughly 3 kilometre from the shaft a potential new orebody was found, named the QK-Zone. An independent design completed at Queen's University[8] suggested the QK-Zone to be developed as a separate mine -with access via the current workings-, including the construction of a new small-diameter production shaft. The remote position of the orebody from the processing plant and the moderate depth of the orebody made the QK-Zone project an interesting case for determining the feasibility of installing a hydraulic ore hoisting system, in order to reduce the hoisting costs.

Calculating Procedure

Equations and assumptions mentioned earlier are programmed in a spreadsheet and applied to find TDH and Pr for a hydraulic ore hoisting system at the mine.

The program verifies the flow conditions and requirements for usage of certain relations. The calculated actual production (due to lower delivered slurry concentration) is projected, together with the total production hours that are needed for a planned daily production. Table 5.2 shows the input data for the spreadsheet program:

Description	Symbol	Unit
Specific gravity of the ore	ρ_s	tonnes/m^3
Maximum particle size	d_p	mm
Specific gravity of the carrier fluid	ρ_f	tonnes/m^3
Kinematic viscosity of the carrier fluid	ν	m^2/s
Friction factor	μ	-
Selected pipe diameter	d_{pd}	m
Vertical pipe length	z	m
Horizontal pipe length	L	m
Daily production		tonnes
Daily number of production hours		hours
(Estimation of terminal settling velocity of particle)	(v_t)	m/s

Table5.2 Input data for spreadsheet program

Summation of $Pr_h + Pr_v$ produces the total operating requirements for the hydraulic ore hoisting installation. The summation of $TDH_h + TDH_v$ gives the total head developed which also gives an indication of the pressure to be developed so as to deliver a desired output.

With this input, there is not much flexibility since the factors are normally constant for a particular mining set-up. With the assumption that the maximum particle size is determined by the discharge of the underground crusher, then only two factors allow any adjustment: i.e. pipeline diameter D, and daily production hours.

The plotting of TDH and Pr versus D and daily production hours is used to find the optimal operating conditions at which the pump pressure and consequently the energy consumption is minimal.

This is done for a wide range of values of D to find the operating requirements for different possible equipment set-ups. The pipe diameter is selected from a list of standard pipe dimensions for Extra Strong steel pipe lines, and a minimum diameter is suggested after entering the maximum particle size. The suggested diameter -equal to 5d- prevents slugging of the pipe. The estimation of the terminal settling velocity will only be requested by the program if no valid value for v_t can be calculated.

When all data is entered, the program calculates the TDH and THPr using the equations 5 to 9 from the previous section.
The effect of changing pipe diameters and daily production hours can be found by plotting the TDH and the THPr. These plots can then be used to find the optimal operating conditions at which the daily energy consumption is minimal. By doing so for different maximum particle sizes, the operating requirements will be shown for different possible equipment set-ups.

As mentioned before, most data required in the spreadsheet program is constant at the QK-Zone project, except for the pipe diameter D and daily production.
Two values for the maximal particle size d, were considered, depending on the potential type of hoisting equipment: when using the Siemag 3-chamber pipe feeder, the maximal particle size is 5" (= 127 mm.), set by the discharge of the underground primary jaw crusher.
In case a GeHo Piston Diaphragm Pump is selected, the ore size has to be reduced to 100% < ¼" (= 6.35 mm.), requiring further underground particle size reduction.

The other data used in determining the optimal operating conditions at Detour Lake Mine is shown in Table 5.3.

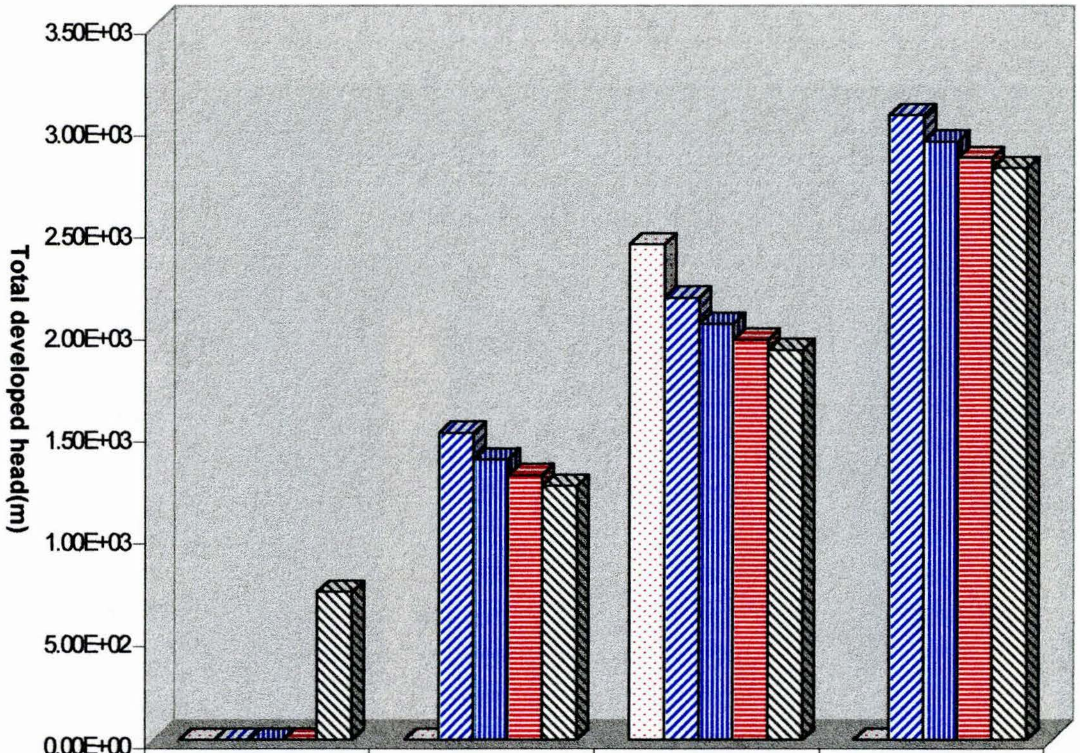
Table 5.3 Input data from Detour Lake Mine Project

Description		Unit
Specific gravity of the ore ρ_s	2.9	tonnes/m^3
Maximum particle size d	6.35/127	mm
Specific gravity of the carrier fluid ρ_f	1	tonnes/m^3
Kinematic viscosity of the carrier fluid ν	1.217e-6	m^2/s
Friction factor f	0.014	-
Vertical pipe length z	898	m
Horizontal pipe length L	3260	m
Daily production	1000	tonnes

Extrapolation Of Results

The TDH and the THPr are determined for pipeline diameters greater than the minimal required diameter and for the entire range of daily production hours for which the actual required hours to reach projected production are less than 24 hours.
A ceiling value of about 3000 m vertical depth or height. TDH is determined by the maximum performance of the GeHo Piston Diaphragm pump.
The planned production for the Detour Lake project turned out to be realisable with one pump due to the relatively shallow depth (about 900m). The result of extrapolation of the total head developed (TDH) and pump pressure required (THPr) has been presented in graphical form for ease of interpretation.

Figure 5.5 shows the TDH for a depth of about 3000m hoisting fine material and with same production capacity as in Table 5.2



	4	6	8	10
6hrs/day	0.00E+00	0.00E+00	2.43E+03	0.00E+00
9hrs/day	0.00E+00	1.50E+03	2.17E+03	3.06E+03
12 hrs/day	0.00E+00	1.37E+03	2.04E+03	2.93E+03
15hrs/day	0.00E+00	1.30E+03	1.96E+03	2.85E+03
18hrs/day	7.24E+02	1.24E+03	1.91E+03	2.80E+03

Pipe size(inches)

Figure 5.5-Hydraulic Pump Pressure Requirement(m)

As illustrated, the general trend is that for any given pipe size, the pump pressure required falls as the number of operating hours per day increases. This is so because the production capacity is held constant and this means less pump pressure is required to do the same work.

Figure 5.6 illustrates the expected result in the event that particle size being carried is course (i.e. =>4inches). The result also shows the effect of different production times as above. The trend confirms the diminishing pressure requirement as the number of operating hours increases.

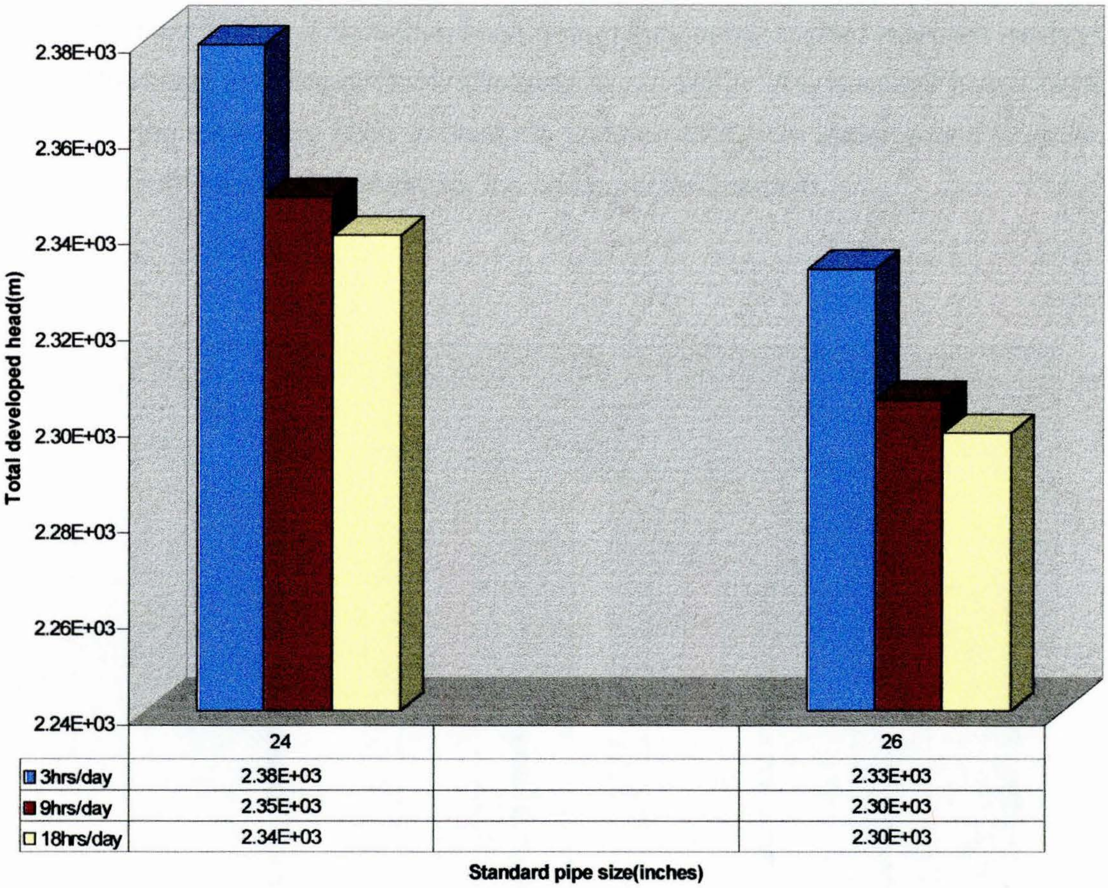
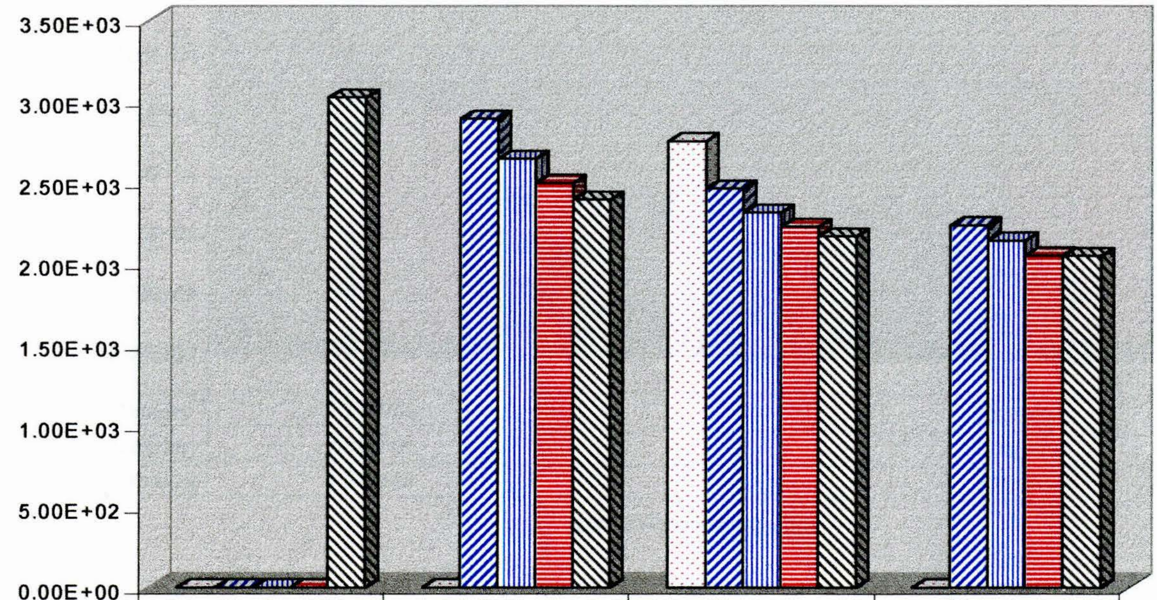


Figure 5.6- Pump Pressure Requirement-coarse size option

Transportation of solids by slurry pipelines has a wide variety of industrial applications, these include conveyance of coal, ores, mineral concentrates and tailings. Depending on the properties of the solid, properties of the liquid, pipe size and flow velocity, different regimes, or modes, of flow can develop. The major modes are homogeneous flow, heterogeneous flow, and flow with stationary bed. The solid-liquid interactive mechanisms are different in each of the flow regimes. If there is more than one size of solid, smaller particles may be transported in homogeneous flow and larger particles in heterogeneous flow. This mode of flow is referred to as compound flow.

In the homogeneous flow regime, the solid particles are homogeneously distributed in the liquid. Homogeneous flow, or a close approximation to it , is encountered in slurries of high solids concentration and fine particle size. Such slurries often exhibit non-Newtonian rheology. The particle settling velocities are insignificant compared with the turbulent motion of the liquid. Appendix 2 gives further explanation of the design parameters and flow properties of slurry transportation.

The factors listed in this section have been determined using applied research methods at Queen's University and Placer Dome (Bekkers et al., 1997). The resultant power demand to generate the necessary head to hoist the crushed rock from underground is obtained. Figure5.7 shows the power demand for the fine particles-scenario.



	4inches	6 inches	8inches	10 inches
6hrs/day	0.00E+00	0.00E+00	2.75E+03	0.00E+00
9hrs/day	0.00E+00	2.89E+03	2.46E+03	2.24E+03
12hrs/day	0.00E+00	2.64E+03	2.31E+03	2.14E+03
15hrs/day	0.00E+00	2.49E+03	2.23E+03	2.05E+03
18hrs/day	3.02E+03	2.39E+03	2.17E+03	2.05E+03

pipe size (inches)

Figure5.7- Hydraulic Power Required (kW)-fines option

The same pumping routine produces a set of characteristics for the pumping of course fractions and the graphical representation shown in Figure 5.8 shows the power requirements for the development of the necessary head to lift this course fraction over the depth in question. Comparison of Figures 5.7 and 5.8 will show that the power demand for the same amount of material over the same depth differs and this will later be discussed on in Chapter 6.

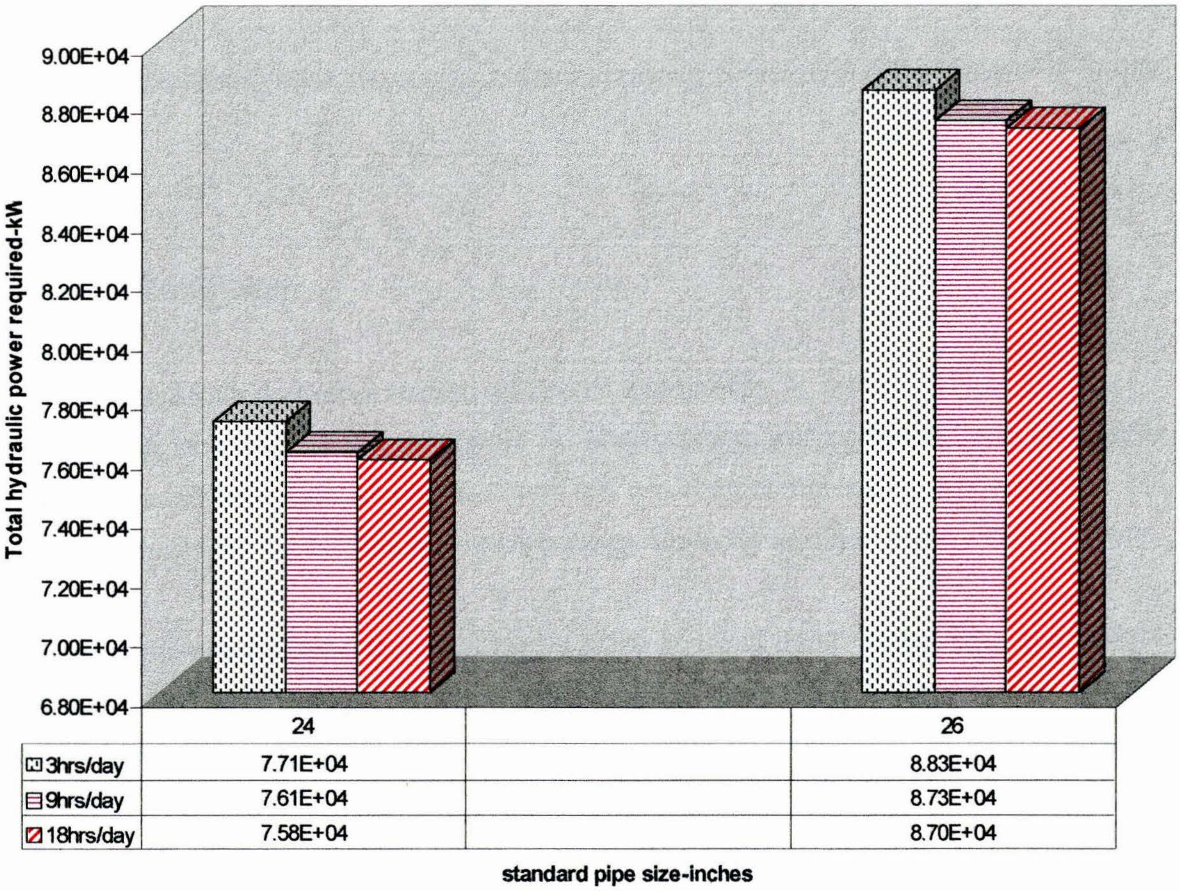


Figure 5.8- Hydraulic power requirement- coarse option

Each option has got its benefits and drawbacks but it must be possible to draw on the strengths of each option resulting in a working system able to deal with a range of material sizes. Under the fine material option, Figure 5.8, the energy efficient approach would be to operate the hoisting system on as long a day as possible with a large diameter pipe (10 inches-254mm).

As will be discussed in Chapter 6.2, it will be realised that there is a lower limit to the particle size that can be pumped much as there is an upper limit to the same. The energy required for pumping smaller size fractions using a 4inch pipe between 3 to9 hours of operation/day is too high when compared to power required for similar duration using larger diameter pipes like 6-10inches. This is mainly due to the high pump pressure required to force the fine material through such a small diameter over short time period. The need to get same daily production target over varying periods of operation per day using an array of different pipe diameter affects the volumetric concentration and consequently the rate of discharge of the material. So the smaller the particle size coupled with a small diameter pipe attracts high power requirement if the production target is held constant.

The power requirement for the coarse material hoisting is however not comparable to that of the fines.

5.3 Rope-less Technology

5.3.1 Linear Motor and other related electronic alternatives

The prospect of rope-less technology as an alternative to the traditional rope hoisting is still being investigated extensively. Currently this is a subject that is gaining momentum but there are no results for any tests or pilot projects done as in other areas such as hydro-hoisting.

The strong point about rope-less technology is the limitless depth range. This is a definite advantage and will present great energy savings compared to any other hoisting mechanism. It however remains as an alternative still to be proven and as such belongs to the future.

5.3.2 Kiruna Method[26]

In partnership with the Henry Krumb School of Mines at Columbia University, ABB 's Research and Development have done a feasibility study on an alternative to skip hoisting at Kiruna in Sweden. This study took the form of two scenarios:

1. Sinking a shaft and then mining. The life of the project is calculated and the corresponding development and operating cost calculated. All the project financials such as NPV, Payback period ,IRR and others are calculated,
2. Ramping down to the ore body and using ramp haulage to a depth of 500m below surface and then sink a shaft from 500m to greater depth. The ramp hauling is done by way of D.C overhead trolley-assisted trucks between surface and 500metres below

surface,and from there on, skip hoisting is preferred. Again like in the first scenario, life of the project, costs and the financials are all calculated.

On comparing the results for the two cases in all the many cases considered, the scenario for 'Ramp first and sink shaft later' always proved more economic and profitable. The major reason being more to do with postponing huge expenditures well into the life of the mine. This is because of the delay in sinking the shaft as opposed to the scenario one where the huge investment in sinking the shaft can not be delayed and also as a result the life of the project is longer in case 1 than it is in case two.

The depths investigated in this situation all fall far too short of what is conceived as deep-level mining. Therefore these results do not make this means of transportation a candidate for application to depths greater than 2000 metres.

CHAPTER 6

SYSTEM COMPARISON AND DISCUSSION

The comparisons are based on the following aspects:

- Payload /depth characteristics
- Power and Energy characteristics
- Cost comparisons

6.1 Payload / depth characteristics

Common to most transportation systems, hoisting tends to be efficient and cost effective with increase in payload and reduction in speed. Thus payload capacity at any depth is a primary concern in evaluating a hoisting system. This is equally true in hydraulic transportation where high volumetric concentrations and low speeds result in a high output and long pipeline life.

Rope hoisting

The payload which can be supported by any system of ropes is determined by the characteristics of those ropes and any statutory or voluntary limitation (so called factor of safety) which the user may accept with regard to the specific loads imposed upon them. Figure 6.1 illustrates the maximum payload capacity of various hoisting systems. The systems under consideration are : BMR, Double Drum and Koepe System. The comparison is based on the following:

1. Statutory Regulation

As laid down by the South African Government Mining Engineer, comparisons are made between drum winders using one rope per conveyance, the Blair winder, either as coupled or gear-less, using two ropes per conveyance and the Koepe winder using four or more head ropes.

2. Winding Ropes for drum and Blair systems

Performances for drum and Blair winders are based on the characteristics of South African-made triangular strand winding ropes constructed in steel of a tensile grade of 1800Mpa (and now ribbon over triangular strand non- spin). These ropes are proven and there is sufficient experience with them. Ropes of a higher steel tensile grade are available and figures indicate possible extension of performance when using ropes constructed of 1950 Mpa steel or more. However there is less experience on the use of such UHT steel ropes as the

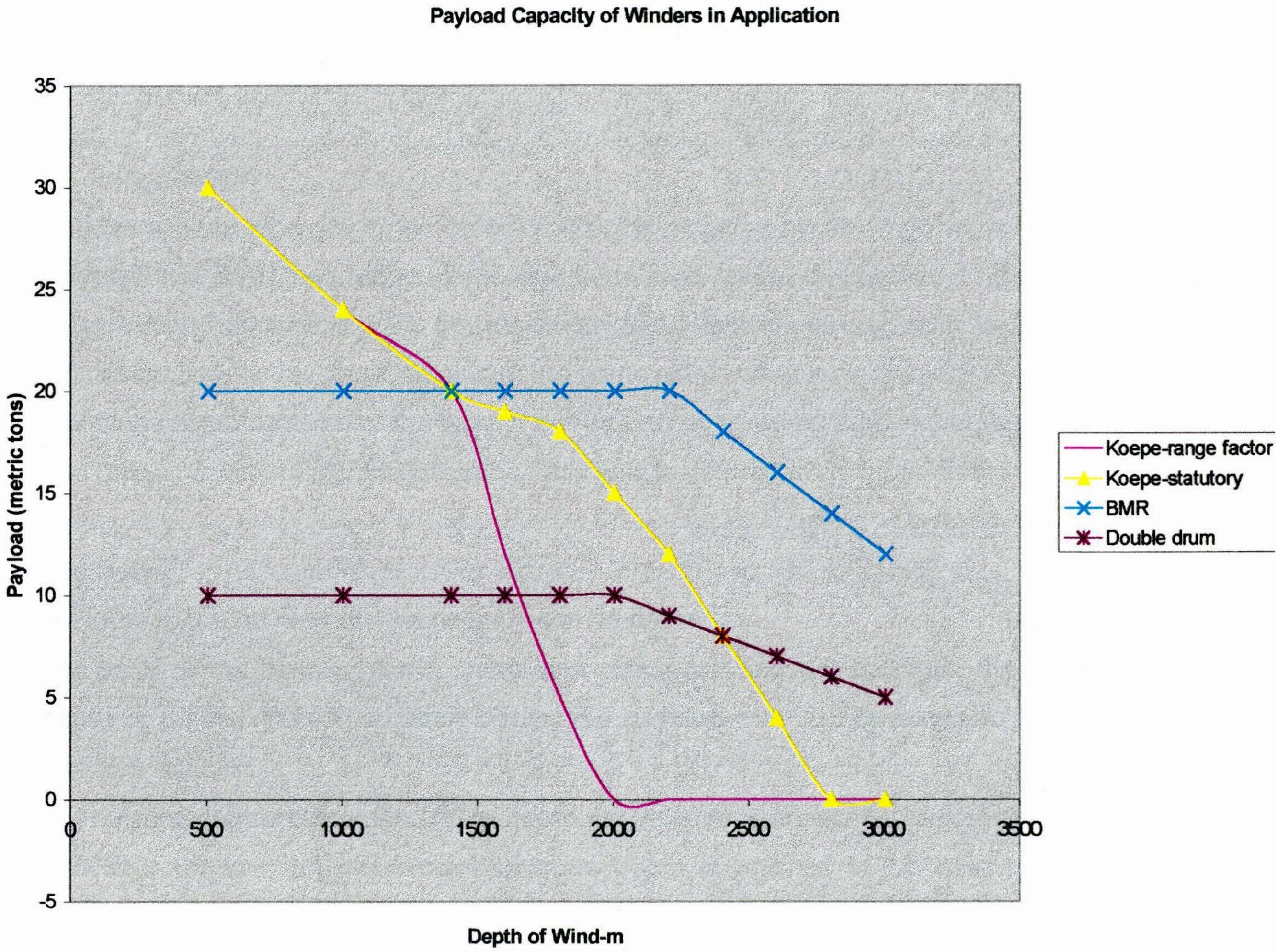


Figure 6.1

grade increases hence attainment of increased performance by using such ropes may attract penalties in the form of reduced rope life or increased drum/rope diameter ratios.

3. Ropes for Koepe

All Koepe system performance figures are based on the use of South African manufactured ropes of fishback construction as well as round strand and non-spin rope all from 1800Mpa steel.

Rope deterioration factor

Normally it is prudent to choose ropes that are initially stronger than required by statute so that some deterioration in breaking force can be allowed during the life of the rope without contravening of regulations. Therefore all statutory safety factors are increased by a factor of 1.07 to allow for some deterioration.

Koepe 'range factor'

To achieve a good rope life in the order of 100 000 hoisting cycles, rope manufacturers recommend that static load range in a Koepe head rope should not exceed 11.5 % of the breaking force. This means that, the maximum static tension at any point in any rope should not exceed the minimum tension at the same point in that rope by more than 11.5 % of the breaking force of the rope. It will be unwise to select a hoist that will operate beyond this limit without first investigating. This limit is referred to as range factor

Comparisons:

From Figure 6.1, a number of observations can be drawn up;

- For each of the three systems, there is a critical winding depth. When the hoisting depth is greater than this critical depth, the payload that can be elevated starts to diminish rapidly.
- For hoisting depths below the critical depth the payload is constant in the case of drum and Blair winders but increases rapidly as depth diminishes in the case of koepe winder.
- For each system, the critical depth is constant for any given grade of steel rope, regardless of rope size. As the strength to mass ratio of the rope increases so does the critical depth. This relation could support the use of tapered mass rope because of the high strength to weight ratio that is expected from them.
- The critical depths as stated in the statute for the drum and koepe are practically the same at approximately 1.9km. The critical depth for the Blair system occurs at about 2km for 1800Mpa rope. The point at which the range factor begins to affect the koepe performance is also constant around 1.4km, this is independent of rope size.

Figure A 3.0 in appendix 3 illustrates these observations.

Depth/ payload characteristics for hydraulic hoisting

In contrast hydraulic hoisting depth limit is determined by the pump capacity and the pressure it generates. The current limit on hydraulic hoisting depth is around 2000m which can be extended by the use of intermediate pump stations. Pressure of up to 250 bar is generated by each pump and this can sustain coarse grained material of up to 80mm with a density of not more than $4.5t/m^3$ and a maximum capacity of 500t/h.

For hydraulic hoisting payload to dead weight ratio can be derived from the material density and volumetric concentration of the material in relation to the volume and density of the carrier fluid since there is no other 'dead-weight' carried. To avoid blockage in the uplift pipe, it is generally accepted that volumetric concentration of the material being hoisted must be between 20 to 40 %. Such a restriction will give a payload to 'dead-weight' ratio of around 3:7. However it must be borne in mind that this system has a bigger advantage over any other because it is continuous and as such has a superior discharge rate for the same depth over rope hoisting once it is in steady state. The implication is that under steady state flow conditions, the output per unit time and the payload are constant irrespective of the depth of hoisting.

In rope hoisting the output is a product of cycles per unit time and payload and these two factors diminish with depth. High output can be obtained by increasing the number of cycles. This implies high hoisting speed but these will bring in a heavy penalty in the form of peak and RMS power requirement. Furthermore reliability decreases as hoisting speed increases so does the frequency of rope replacement.

Another interesting option would be to avoid the excessive speed of hoisting (i.e. less cycles per unit time) and increase payload capacity of the hoist so as to maintain output (cycle x payload). This, however, has practical limitations such as:

- The size of conveyance which will be accommodated in the shaft,
- Rope strength limitations
- Availability of an appropriate winder.

Comparison of the productivity of a hydraulic system to a rope hoisting system will clearly show that technically hydraulic hoisting can be 'limitless' in relation to rope hoisting but the reliability is a definite drawback. This is because of the inevitable high wear rate of the pipeline- especially bends if coarse material is being hoisted.

6.2 Power Requirements

The comparison considers at the three basic winding systems common in Southern Africa namely; Double Drum, Koepe and Blair Multi-Rope winding systems. These are in turn compared to the option of hydraulic hoisting. Figure 6.2 is a comparison of the RMS power/ torque requirements of the rope hoisting systems.

The pattern for RMS power requirement is similar to that for peak torque requirement. The power requirement of the BMR system is in excess of that required for the Koepe throughout the entire range of depths. The curious inverse shape of the BMR curve below critical depth of about 1990m is due to the fact that the same diameter hoist and same size of ropes are required for any depth in that range.

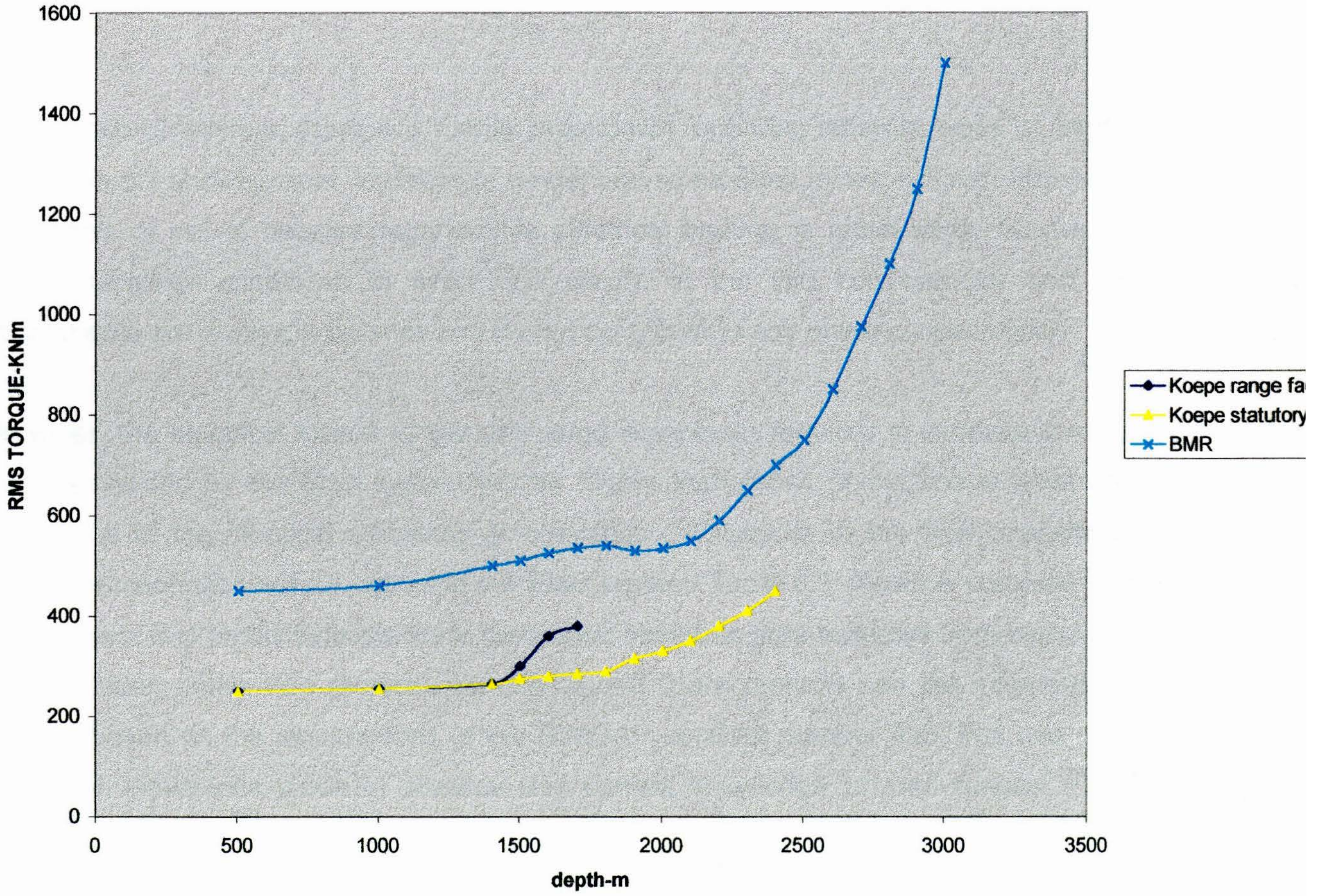


Figure 6.2-RMS torque requirements of BMR and Koepe hoists to handle constant Payload of 15t from various depths.

If it were possible to use diminishing sizes of ropes as depth diminishes (as is the case in Koepe) it is possible that BMR RMS requirements would be lower than in Figure 6.2.

In a comparison of the overall cost (ie. Equipment installation and running) of hydraulic versus winding, it is likely that the cost and installation of a winder would be much higher than hydraulic although replacement cost of wearing components in hydraulic may be the same, even more.

6.2.1 Energy consumption-Winding.

What is also of importance is the running costs such as labour :

- Personnel for maintenance and supervision of pumps and ancilliaries
- Personnel for maintenance of winders plus driver, banksman, onsetter etc.

The major running cost however is the electric energy cost.

In the case of winders, there is a further breakdown regarding different types of winders and Table 6.1 shows actual statistics of winders on some deep mines in South Africa.
No details of out of balance regenerative electrical braking is given partly because all double winding- conditions in which the weight of the two conveyances and their attachments are always in balance and do not contribute to any energy consumption

However, the electrical losses in the very large equipment required to produce the high torques needed by the Blair system will be higher than those of the koepe hoist drive regardless of the identical efficiency of conversion. Therefore in the final analysis the energy consumption will be higher in the Blair system. Table 6.1 shows a comparison of some power and productivity statistics calculated based on data collected from some deep South African mines and projects. Number 2 and 3 are projects and the information is derived based on the assumptions of the SIMGAP contract number 418. Full use of the proposed regulations, code of practice and current knowledge of load ranges[14,15,17] is done.

Table 6.1 -

Productivity and Energy Consumption Analysis For Some Deep Mine and Projects

	Winder Type	Depth m	Payload tons	Drive Motor Power KW (RMS)	Energy Cons. KW/trip	Productivity t/hr	Energy/ton kWhr/t
1	BMR-elect. Coupled	3150	13.5	7319 x2 (forced vent.) 17001x 2 peak motor	285	123.5	21
2	Double drum	4000	14	6500 x2	270	122.5	19
3	Double drum	4000	16	7750 x2	320	140	20
4	BMR	2670	17.2	6626 x2	269	160	15.6
5	Double drum	2816	15	4802 x2	198	120	13.2

Due to this fact, the productivity is much higher than would be expected if the present regulations were applied for such a depth. The energy consumption per ton of material hoisted would also be higher than indicated. It should follow that since a BMR system is more suited to application in deeper level mining than other rope hoisting means, the combination of such a hoist and the tapered mass rope would be attractive.

If the new proposed regulations are put into effect, significant improvements in payload and hence productivity could be realised.

6.2.2. Hydraulic Ore Hoisting

Power requirements Analysis

Hydraulic transportation, just like any other forms of moving material, involves the expending of energy. The emphasis in this section is to make a comparative analysis of the energy requirements of the alternatives in hydraulic hoisting and winding.

Analysis of the pump pressure requirement shows a direct relation to the daily power requirement. It is also clear that hoisting of the fine sized-particles demands less pressure from the pump and consequently lower power than for the coarse ones.

Figure 6.3 illustrates the daily power requirements for the fine option. From these graphs, the conclusion could be made that the fine option is favourable. However, hoisting the fine particles requires an underground crushing circuit, significantly increasing the investments for developing such a hoisting system.

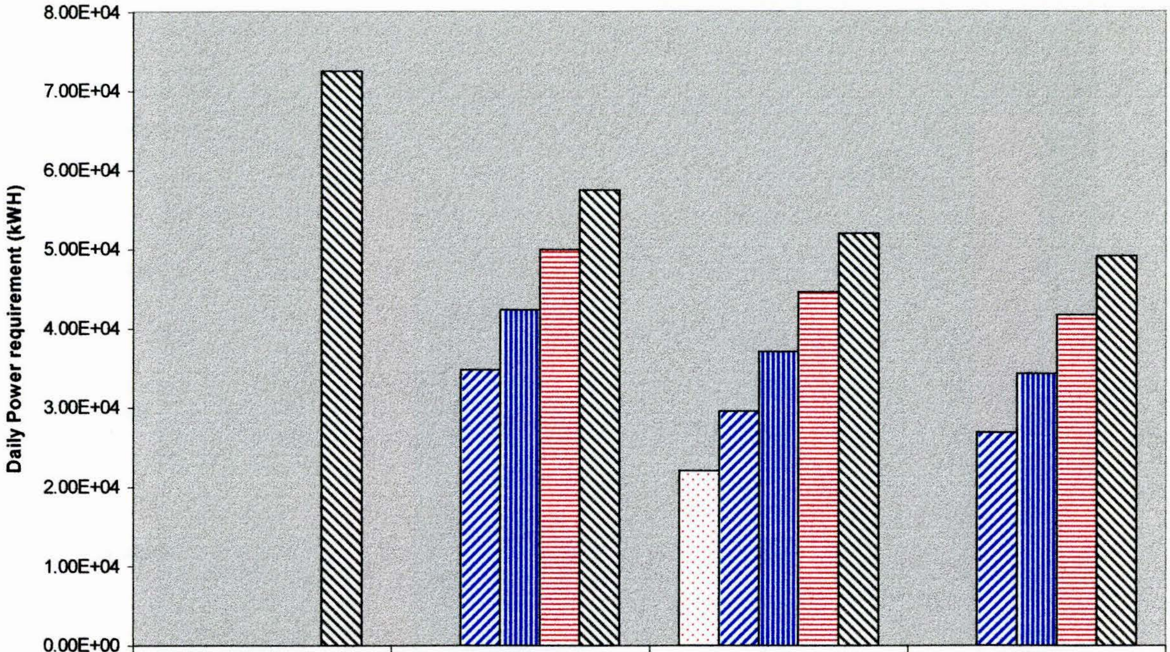
A look at the total hydraulic power requirement will show that for bigger particle size mostly and the fine option to some extent, the daily total power requirement is very high if compared to winding. However the power consumption per ton of material pumped to surface is comparable to that hoisted by winding especially over shorter hoisting periods like 12 hours to 4 hours per day. Table 6.2 below illustrates this observation.

Table 6.2- Energy Consumption Analysis for Hydraulic hoisting- KWHr/ton

Particle Size (inches)	4	6	8	10	***24***	***26***
Operating Duration hours/day						
4					***308***	***352***
8			22			
12		34	29	26	***913***	***1047***
16		42	37	34		
20		49	44.5	41		
24	72	57.45	51	49	***1802***	***2088***

The energy consumption per unit mass hoisted shown in Table 6.1 (Winding) is only based on energy loss due to heating and other losses while energy expended doing useful work is assumed to be minimal as explained under that section. This explains disparity between the unit mass energy consumption of the winding system compared to the hydro-hoisting (Table 6.2). It must be noted in Table 6.2 that for coarse material where large pipe diameters are used, power demand per unit mass hoisted is much greater when compared to fine material using smaller diameter pipes.

However, new developments in hydraulic pumping prove that energy recovery is possible in solid and clear-water volume streams and also in reverse osmosis streams. Such a development, if applied, will undoubtedly boost the energy efficiency of hydro-hoisting. The energy consumption per unit mass in such a scenario will be due to pressure loss at the pipe feeder, pump or intermediate pump station (for deeper levels)



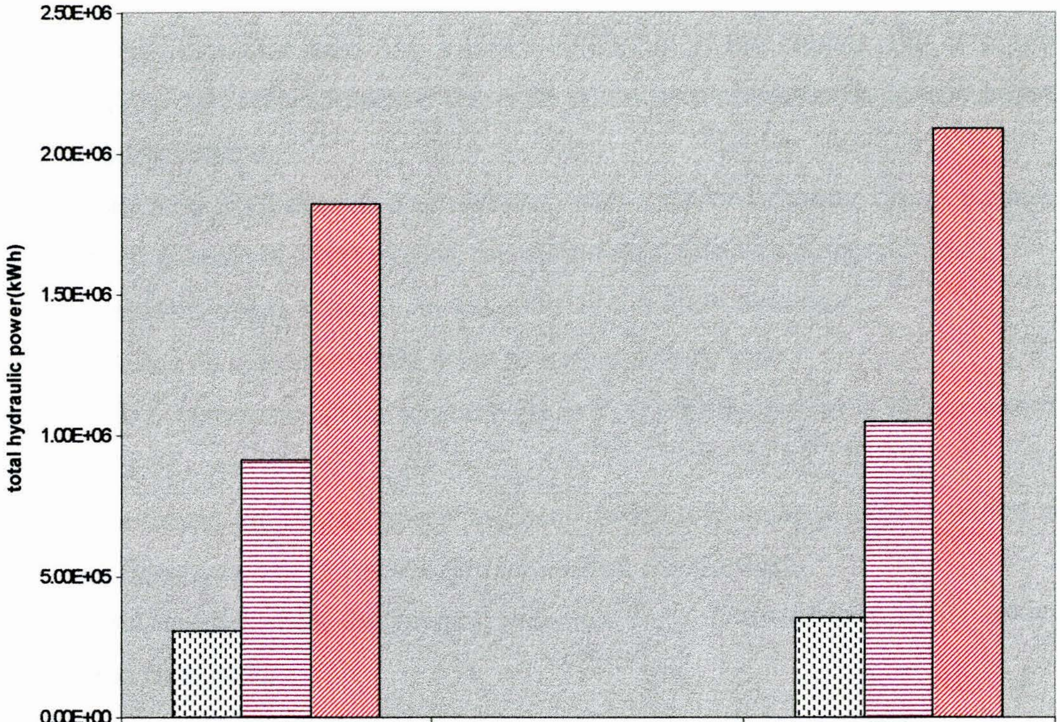
	4inch	6inch	8inch	10inch
8hrs/day	0.00E+00	0.00E+00	2.20E+04	0.00E+00
12hrs/day	0.00E+00	3.47E+04	2.95E+04	2.68E+04
16hrs/day	0.00E+00	4.23E+04	3.70E+04	3.43E+04
20hrs/day	0.00E+00	4.99E+04	4.45E+04	4.17E+04
24hrs/day	7.25E+04	5.75E+04	5.20E+04	4.91E+04

pipe size- inches

Figure 6.3-Daily Power Requirement/ demand- Fine particle size option

The next illustration (Figure 6.4) is a representation of the power demand in the situation where hoisting is largely for coarse material with fraction size greater than 4inches.

It will be clear that the energy requirements in this instance are higher than the option where the fine material is predominant. It is however equally true that less underground crushing is required for the coarse option. This also implies that less development work is required.



	24inches		26inches
4hrs /day	3.09E+05		3.53E+05
12hrs/day	9.13E+05		1.05E+06
24hrs/day	1.82E+06		2.09E+06

standard pipe size-inches

Figure 6.4- Daily power required(demand)- coarse material

The next step therefore is determination total costs for hoisting one tonne of ore, over the life of the project or mine, using the determined figures for the daily power requirement.

The hydraulic pumping requirements of pressure and power produces a trend which shows a diminishing effect as the pipe size increases from 4 inches to about 10 inches and starts increasing steeply as the pipe diameter increases beyond about 12 inches for a given operating period per day.

Appendix 5 illustrates these observations with power and pressure curves.

6.3 Cost Analysis

6.3.1 Rope Winding

The drum to rope diameter ratio has a direct influence on the capital cost of a winder. There is a definite tendency to reduce the ratio in modern installations. This is based on the following observations:

- A 4MW direct coupled double drum winder, with 4,877m diameter drum, suitable for hoisting from a depth of 1700m costs six million rand (1M USD) approximately
- Winder costs vary approximately as the cube of the drum diameter
- Rope life varies as the cube of the drum to rope diameter ratio
- Rope life of a 49mm triangular strand rope is 100 000 cycles on a winder exampled above.
- Rope costs based on current prices in South Africa and savings over a period of 30 years assuming no price increases and neglecting discounting.

The Figures 6.5 and 6.6 show the relation between drum diameter to rope diameter ratio and winder and rope costs.

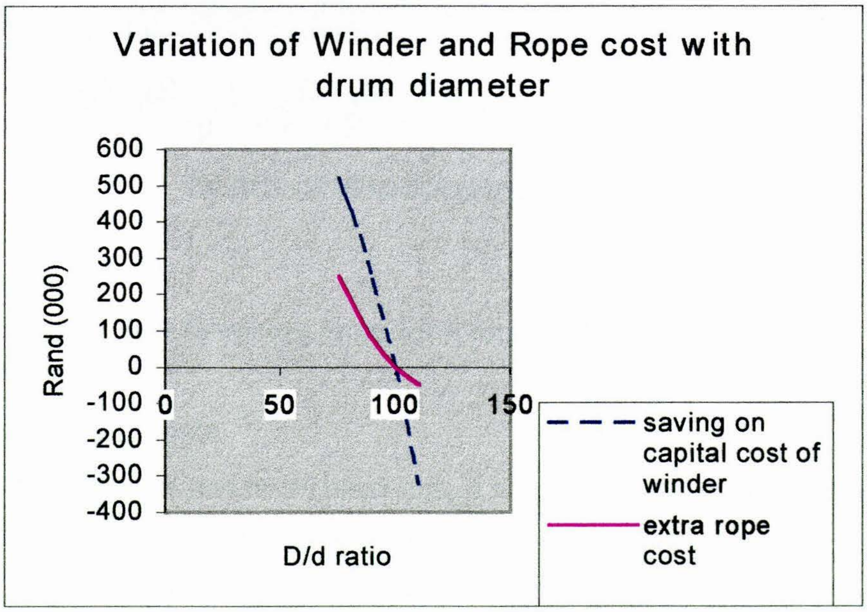


Figure6.5- Winder and rope cost behaviour vs. drum diameter

1 The extra cost of rope due to reduced rope life is more than off-set by the capital savings in the winder within certain limits.- This leads to the consideration of `optimum rope performance'

2 The cost of achieving very long rope life is uneconomical. Caution still has to be exercised when designing a winder to optimise the saving in capital with respect to increased rope cost, that rope life does not become so short as to be a financial embarrassment.

3 A satisfactory rope life is of the order of 18 months for a drum winder operating with standard types of triangular strand ropes. The proportion of the time of an operating shaft dedicated to rope maintenance and changing is restricted.

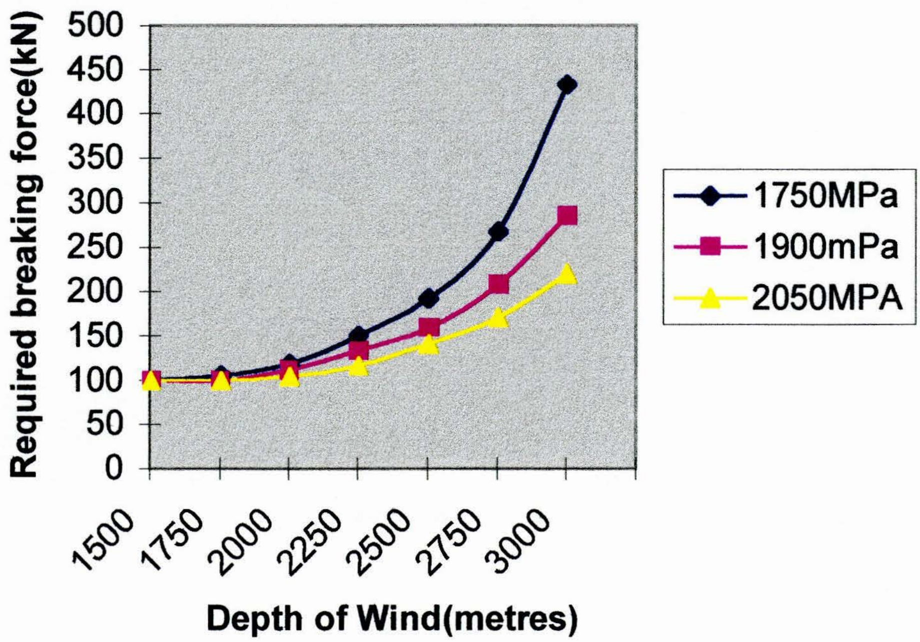


Figure. 6.6- Breaking force for 1 ton end load

Comparison of winder costs

Many factors that influence cost might have little or no relation with the design and function of the equipment, whilst customer preference may result in quite dramatic differences in cost for the same basic hoist. Figure 6.7 is a relative comparison of the cost of these winders which have been discussed. The costs are historic, and as such, time value of money has not been incorporated.

The intention is therefore to show on a comparative basis rather than true cost of any specific winder. The double drum has the same cost curve as the BMR but the absolute cost is between that of BMR and Koepe.

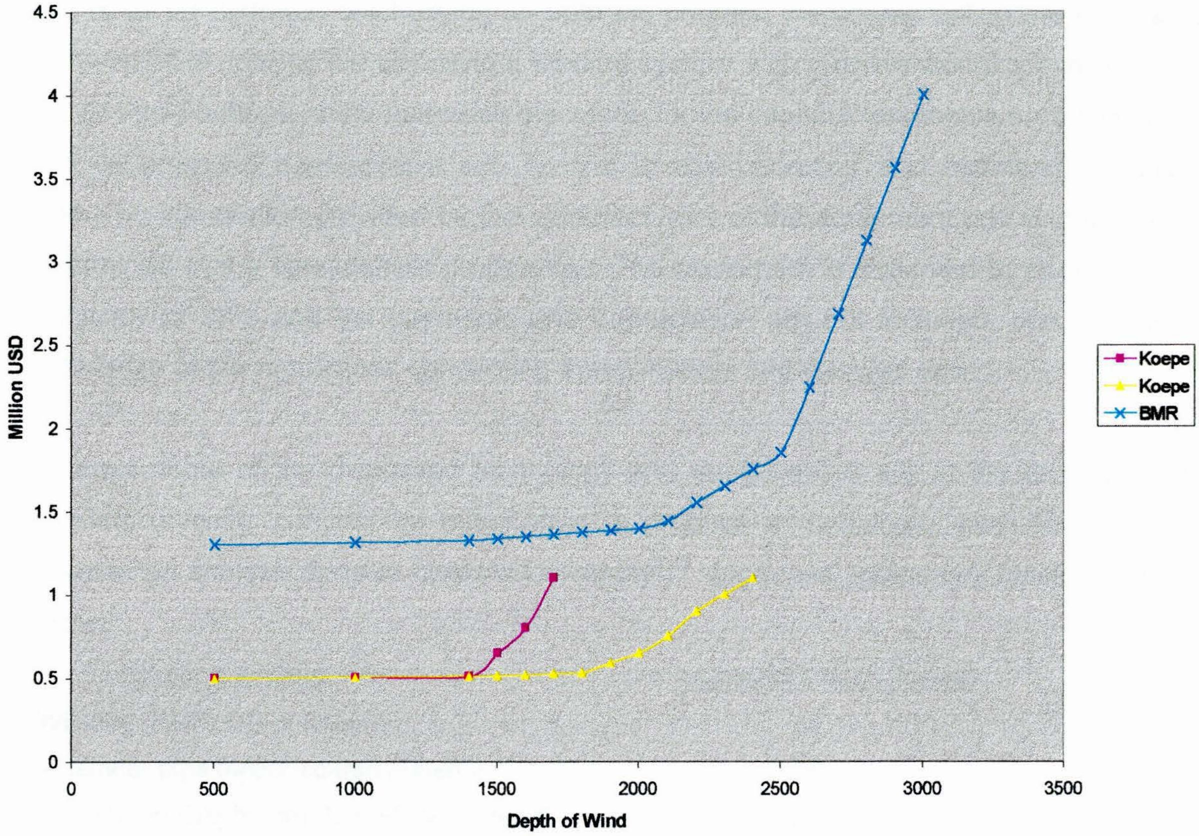


Figure 6.7- Estimate Cost of BMR and Koepe Hoist Capable of Handling 15t from various depths.

Financial Analysis-Hydro-hoisting

According to the information concerning the case study reported earlier in this document, the dimensions and contents of the QK-Zone made available in early 1997, the projected life of the mine was estimated at a little more than 6 years, at a daily production of 1000 tonnes (25% of projected annual production was planned to be achieved in the first year, during development-in-ore). The hydraulic ore hoisting installation was estimated to operate 335 operating days per year.

To determine the total costs per tonne for hydraulic ore hoisting over the life of the project, a financial analysis was conducted using discounted cash-flows, with an estimated cost of capital of 12%.

The cost of capital is quite likely to change over the life of the project, in an unpredictable fashion, but 12% as an average is a suitable assumption at this stage.

The costs for hydraulic ore hoisting for both the fine and the coarse option are compared to the costs of hoisting the ore using a winding system, with a conventional conveyance.

For all three options, costs estimates are made for the capital investments for acquisitions and underground development and for the annual operating and maintenance costs, based on information provided by the manufacturers of the equipment and on the figures determined in the spreadsheet programme. The equipment is assumed to operate at an efficiency of 93% and tax payments and depreciation are not included, due to lack of knowledge of the regulations concerning these matters and relevant data.

Because some of the discussed technology and equipment is still at the design stage, estimate of costs can not be established to produce a meaningful cash-flow analysis however an analysis done at Queens University[8,22] produced preliminary hoisting cost as below:

Hoisting system:	Cost/tonne (R/tonne)
Winding (BMR; DD or Koepe)	54
3-chamber pipe feeder-coarse material	35
Geho Piston Diaphragm Pump-Fine material	25

This was based on assumptions that might not hold for all mining environments, some of them being:

- Cost of capital at 12%
- Production capacity used might be too low but this will mean probably reduced unit cost for higher production scale as a result of economies of scale especially on operating and production costs
- Life of the project or mine is assumed at 6 years but this also might actually be shorter or longer.

The proposed mine design for the QK-Zone with the skip hoist had an estimated hoisting costs of R54 per tonne. For the hydraulic ore hoisting options the financial analysis determines the costs for the coarse option, using the Siemag 3-chamber pipe feeder to be R35 per tonne and for the fine option, when the ore size is reduced to 100% < ¼"(6mm) and hoisted by a GeHo Piston Diaphragm Pump, the resulting costs per tonne were R28.
According to the financial analysis, the most favourable option for hoisting the ore from underground to the processing plant is by installing an underground 3-step crushing circuit plus a hydraulic ore hoisting system using a GeHo Piston Diaphragm Pump. The estimated hoisting costs per tonne of ore are significantly lower than when the ore is hoisted by skip.
In conclusion however it must be borne in mind that whether hydraulic hoisting is used or not, there will still be a need for an investment in man transportation to and from underground. This means that there will still be need for the shaft to be sunk to accommodate the man winding operation. This is a cost which has not been included in the foregoing analysis. So this requirement, for extra winding capacity and its related requirements, is a serious drawback which needs to be evaluated and included as a cost of hydraulic hoisting.

The use of complete underground mineral processing concept, as espoused at the 20th International Mineral Processing Congress in Germany 1997 by Professor Varbanov, will provide substantial reductions in hoisting capacity requirements.[23] The concept can be further developed into a system in which the milling can be achieved on a modular basis either near the production end or at every producing panel. Concentration is performed at the same scale and the product is then hoisted to surface. In such an instance there is no vertical hydraulic transportation necessary. Figure A4.0 in appendix 4 illustrates this concept as proposed in the case of radioactive ore.
Another proposition is the in-situ leaching of low grade ultra-deep deposits. The leachate is collected at the lower boundary of the deposit for further processing on surface by pumping.

CHAPTER 7 CONCLUSION AND RECOMMENDATION

7.1 Conclusion

7.1.1 Skip Hoisting

The mining of minerals at great depths necessitates investments that can only be compensated for by some or all of the following.:

1. By concentration of the winding operations in a few shafts of high output. If the need is to increase the output figures per shift, it is of little use to increase the winding speed. This aim can only be attained by increasing the payloads, minimising the loading and unloading times and eliminating all interruptions in product winding due to other winding tasks. There are, however, horizontal distance limitations which will introduce low productivity and uneconomic hoisting.
   The single-purpose shaft with fully automatic winding operations offers optimum conditions for maximum output figures. Winding installations for payloads of up to 50t are tried and tested for some operations especially those using Koepe winders.
2. With the control systems developed jointly by Cegelec Electrical and Anglo American Technical Services to combat rope loading, rope safety factor as low as 3.17 has been targeted. To operate at this factor and retain the same margins of safety as are enforced by the present regulations, it will be necessary to:
   1. Satisfy the criteria contained in the Code of Practice and
   2. Limit peak rope loads and minimise induced oscillations during winder operation. A system of closed loop brake control has been developed to implement the necessary brake control see figure 7.1 which illustrates the impact of the revised factor of safety on hoisting capability and related efficiency factor using two skip sizes.
3. A new design of the rope to include the positive benefits of the tapered mass rope seems to be a very viable proposition if it can be perfected. Rope manufacturers have considered carbon fibre ropes but because of the number of problems associated with these ropes it was found to be unsuitable for winding. The main problems being:
   a. Treatment of the end terminations of the rope and joining of the rope

b. Lack of a visible or detectable indication of the condition or deterioration of the rope.

If answers could be found to some or all the mentioned problems, then a combination of the benefits of a light Tapered Mass rope with a high strength to weight ratio will be a breakthrough in single stage hoisting from ultra deep mining operations.

The depth of 4000m and over can be tackled with a reasonable payload being hoisted as illustrated by Figure 7.1, constructed with the help of data generated when simulating for hoisting with revised new factors of safety and other regulations which already can be found the regulations Chapter 16 and SABS 0293/0294 .[14]

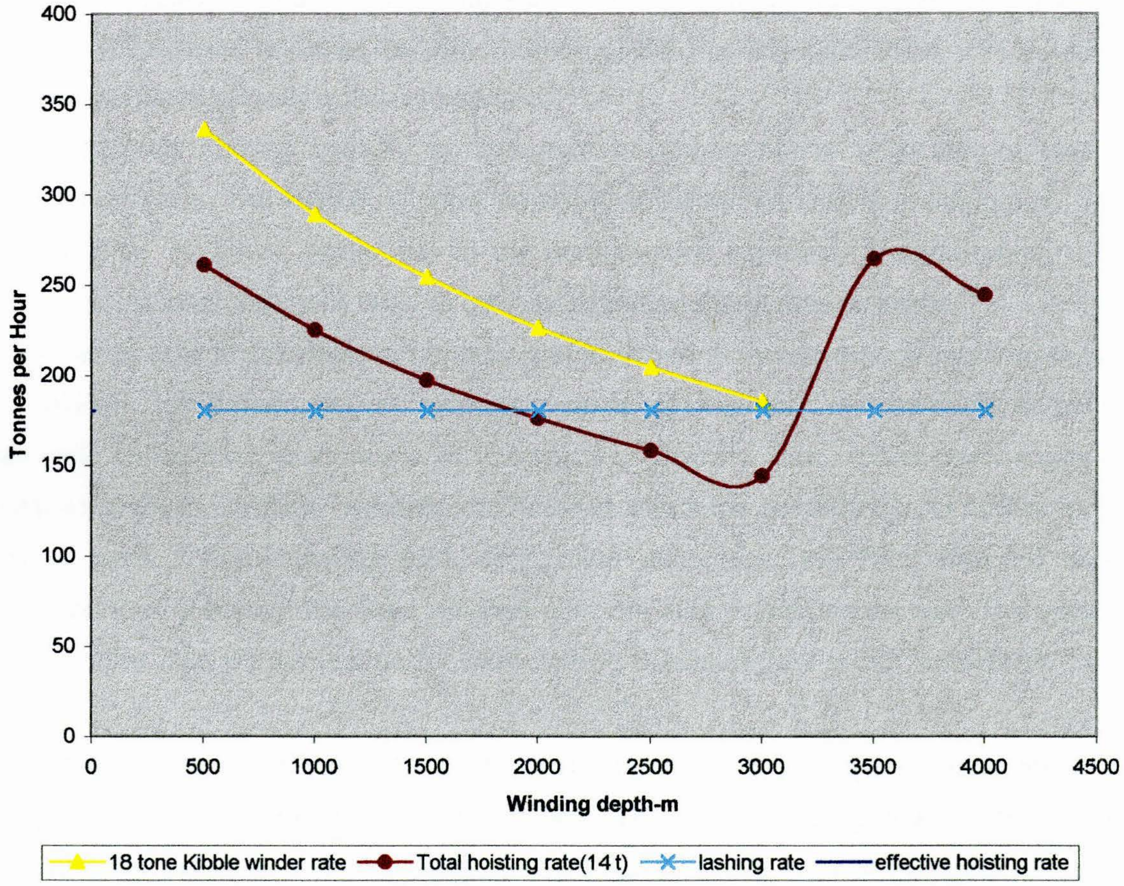


Figure 7.1 : Ultra Deep level Project- Expected production rate

Note: The lashing rate and the effective hoisting rate are equal. The above illustration gives the expected benefit of the revised regulations. This performance can be improved on if new light material is finally introduced in rope manufacture. Already some existing mining projects are expected to benefit from this development, Western Deep Levels are expecting to be at 4000m in 2002 and many others are looking at the same range of depths and more.

7.1.2 Hydraulic Hoisting Conclusions

Before concluding the option on hydraulic transportation, it is essential to note once more that all data retrieved from the calculations in the spreadsheet programme based on formulas from published literature and a further number of assumptions are made.

For clarity, the assumptions are listed below, including their possible implications:

1)Most importantly, the terminal settling velocity v_t is based on calculations assuming a spherical particle. A more irregular shape of the particle will decrease the settling velocity and thus will actually improve the performance of the hoisting installation. The v_t used in the calculations is actually quite conservative.

2)The hindered settling velocity v_t' assumes a homogeneous collection of identical spherical particles. The velocity v_t' must be adjusted for the particle size distribution, when this is known. However, again due to the conservative approach of establishing v_t, the pump requirements are more likely to be overestimated than underestimated.

3)The carrier fluid is assumed to be clear water. When the performance requirements for the hoisting installation rise to extreme values, is may be recommendable to add chemicals to the fluid, to increase its transporting capacity. Also, it has been suggested (Sellgren,1985) to combine hydraulic ore hoisting with mine de-watering, to further reduce hoisting costs. When hoisting with 'dirty' mine water (i.e. the fines are not settled underground in a sump), the fines will also alter the fluid properties as well. The situation for the carrier fluid will be unique for each mining operation and is not considered in this design.

4)The friction factor f depends on the relative roughness of the pipe wall ε/D, where ε is the symbol for the irregularities of the pipe wall. The factor f changes with different pipe materials, but for pipes that are considered suitable for this type of transport, f can be considered to be fairly constant.

5)The specific gravity of the solids and the slurry are considered constant, although in real situations changing constantly during transport, especially the value for the slurry due to the movement of the particles in the fluid. The density for the ore is the maximum value and again implies a conservative approach in this case. The change in slurry concentration is position-dependent throughout the pipeline and a general approach for the entire pipeline is sufficient in this stage of designing a hydraulic ore hoisting system.

6)Depending on the exact set-up of a hydraulic ore hoisting installation, a number of bends will be part of the pipeline. At such bends and where erosion also takes place, a loss of pressure occurs that can be significant. This loss again is unique for each mining operation and must definitely be considered in a later stage of a design.

The above assumptions are likely to change the pumping requirements calculated in the spread- sheet program, but the calculations suffice at this stage of the design.

Hydraulic ore hoisting appears to be a feasible and very attractive option to winding. The hoisting costs per tonne of ore are expected to be significantly lower than when hoisting mechanically. The overall costs per tonne for producing the ore will decrease as well, as crushing in the processing procedure can be avoided. Further reductions in overall mining costs can be realised, especially in ultra deep level, if the hydraulic hoisting of ore is in tandem with mine cooling system. The 3- chamber pipe system, which operates on the principle of pressure lock, offers as part of a mine cooling system great advantages for the transportation of the cooling fluid. The fluid is fed at high static pressure, with the possible addition of ice, into the 3- chamber pipe feeder and from there at low pressure to the underground heat exchangers at a reported efficiency of 98%[22].

This example shows that with the applied spreadsheet program it is fairly simple to find the optimal operating conditions for a hydraulic ore hoisting installation at a mining operation, after which an understanding is created of the financial investment required. After combining the estimations for the investments with predicted operating and maintenance costs in a financial analysis, an approximation of the total costs for hoisting each tonne of ore results. This allows for a comparison with the hoisting costs for conventional mechanical hoisting to determine the economical feasibility of a hydraulic ore hoisting installation.

In general it can be concluded that the above method can practically and easily be applied to roughly estimate the overall feasibility of hydraulic ore hoisting at any mining operation.

Regardless of the analysis on hydraulic hoisting and its positive attributes, very few development projects have tried it and this implies that there is very little practical experience on the application of this method of hoisting.

The fact that a shaft is still required, though with smaller dimensions, is a drawback. The lack of enough knowledge on the wear in pipes also contributes to the number of disadvantages against this means of hoisting. On the other hand, the advantages of rope

hoisting over hydraulic hoisting are numerous, the most important being the advances already made in design and equipping of the winding system. It is against such a background that whatever benefits hydraulic hoisting has over winding , it will be very difficult to be accepted by the industry at large. Hydraulic hoisting is largely considered as a grey area where there is no experience to back-up ventures into its use.
Underground milling and the subsequent transportation to surface require extra space than normal. There is also the extra requirement in needs for ventilation due to extra heat load generated by all these processes especially when small mobile modular milling units are used.

7.2 Recommendations

7.2.1 Hoisting through winding.

Rope hoisting is the traditional method of vertical transportation for underground mining operations. From observation, it has emerged that rope hoisting with all the stated disadvantages imposed by financial, regulation and technical requirements, still remains as the dominant means of vertical transportation now and into the near future. Whilst that remains the situation, ultra deep level mining can be accessed in one stage by use of technologies which have been developed such as:

- Closed loop brake control system to reduce shock loading of the rope and thus enable lower safety factors to be used thus enabling higher payloads to be attained at these depths.
- Lightweight materials be used for rope conveyance and attachment manufacture where applicable and without jeopardising technical and safety specifications.
- Rope-less technologies- these are still futuristic but could be vital if they prove to be implementable.
- Complete Underground Milling-'power down gold up'- This is a very useful innovation in the sense that less emphasis is placed on high speed and capacity hoisting. Less tailings find their way to surface and hence there are less waste dumps and environmental degradation.

A combination of the different innovations and improvements will yield definite savings in the vertical transportation cost structure. For example, in the case of the investigation by US Bureau of Mines on light synthetic rope for hoisting, there might be a worthwhile

benefit if the polyutherane is covered on the outside by steel wire strands. This would result in composite rope with high strength to weight ratio. The limitation however could be unequal stretch of the outer strands and the synthetic core, i.e. different coefficient of elasticity of steel versus plastic as such elasticity of steel might be insufficient to allow equal sharing of the load. This limitation could be investigated further.
This could be proved in tension testing and the possible "bird caging" as a result.
The rope might be useful as a guide rope in deep shafts as well as rubbing ropes (i.e. those that separate two conveyances at the passing point in the mine shaft).[24]

7.2.2 Hydraulic Hoisting

It is necessary to note that the data resulting from the spread sheet program is largely based on theory and literature of previous research. When seriously considering the design of a hydraulic ore hoisting system, it is essential to test the predicted pumping requirements and to monitor the actual delivered slurry concentration. This can be established for example by installing a pipeline of the selected diameter, but of shorter length in an existing shaft, to pump small amounts of ore at the calculated pressure.
Furthermore it has been suggested before (Sellgren, 1987) that it would be ideal to combine hydraulic ore hoisting with mine dewatering. Despite the obvious economic benefits of such combination, the likelihood of impracticalities to occur is high, especially when hoisting with unsettled mine water, demanding the slurry to be cleaned before further processing so as to prevent dilution.
Definite decision making on starting a detailed hydraulic ore hoisting system design requires more practical testing, under the specific circumstances of the considered operation. In the early stage of mine design, the applied method to consider hydraulic ore hoisting is both fast, easy and fairly reliable.
The combination of hydraulic hoisting and the transportation of cooling water to underground heat exchangers should prove to be very energy efficient. There is also the energy recovery through reverse osmosis in this set-up.
The most inefficient part of the hydraulic hoisting after frictional losses, is the pressure loss at the feeder as the material is fed into the system regardless of the existence of the pressure lock as in the 3 chamber pipe feeder on which it has been experimented. This implies huge energy losses comparable to the losses encountered in the winder motor.

Further research is required in this area as well as particle sizes range determination. The following are points that militate against the use of hydraulic hoisting:

1.Conventional shafts equipped with a conveyance must be available for the transport of men and equipment; it is therefore reasonable that the same shaft will continue to be used for ore hoisting.

2.Shafts are still required for ventilation purposes and in deep mines this factor rather than the hoisting capacity often determines their size. A hydraulic hoisting system may therefore unlikely influence the shaft size.

3.The overall efficiency of present hydraulic hoisting systems is low and is only of the order of 50% but could be higher using reciprocating (positive displacement pumps). Additional handling of ore underground is necessary and this requires special underground chambers. However with the improvement in technology, if continuous mining becomes feasible re-handling of material may not pose any problems.[2,11] However special provision may also have to be made to dissipate the heat generated by such a process or alternatively the milling can be located near the up-cast ventilation.

4.Depending on the material to be hoisted, de-watering facilities will have to be provided on surface.

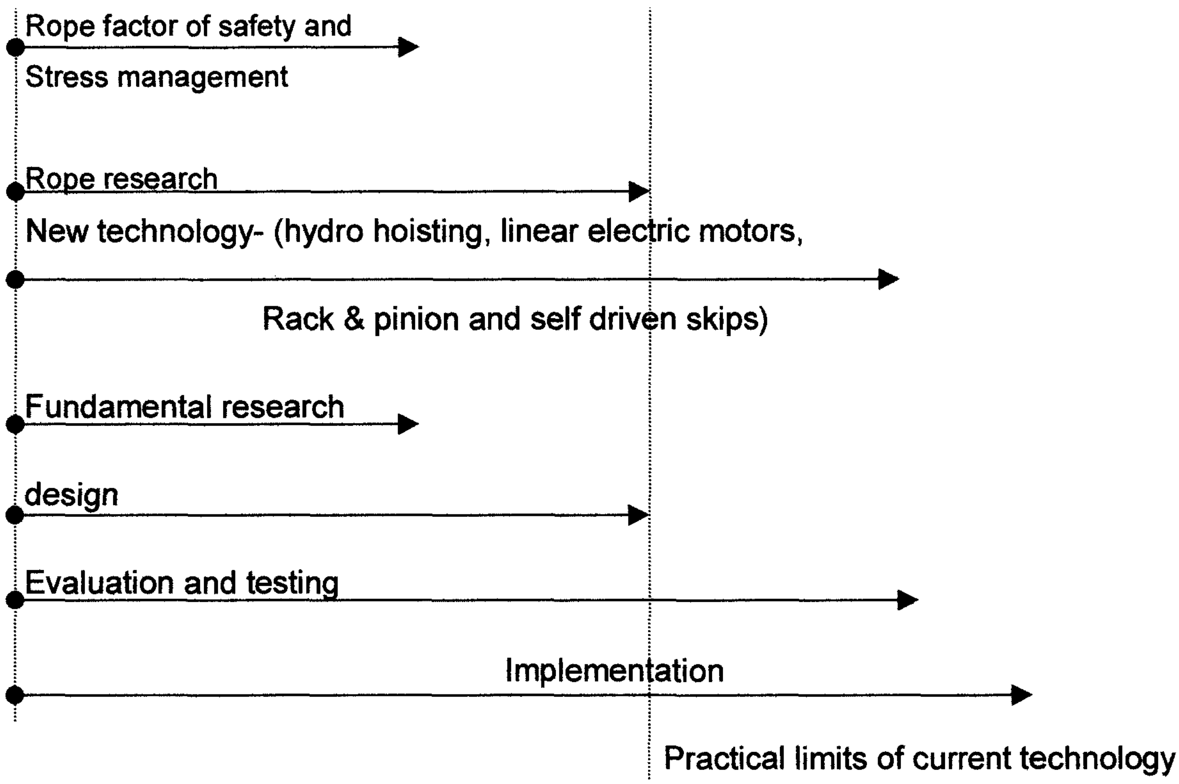
5.Wear in pipes and equipment needs further research.

7.2.3 Other hoisting Options

New technologies like the linear motor applications, rack pinion and other self driven system can not be overlooked but there is need for more fundamental research into the subject including cleanliness of the shaft before there can be a realistic time-scale for their applications in the field. The same can be said of the possibility of adapting the deep-sea manganese and diamond mining technology to underground mining conditions.

7.3 General

The practical limits of present technology are clear and fairly defined. Substantial research has been done and is still progressing, both on current and new technology. A parallel process of the search can be espoused as below:



Basing on the above schedule, rope research (factor of safety and stress management) and new technologies can all be investigated concurrently. The present situation however indicates that efforts and work put into skip hoisting research has resulted in winding as the technology which will most likely continue to be dominant for the foreseeable future, certainly until the research in other areas produce options which are cost effective and practical.

APPENDICES

Appendix 1

Rope power and size calculations

If L is the actual suspended length of rope with an end-load, M_e then the equivalent length of rope is $L_s - L$ ... (1)

And the mass/unit length of the rope is $\frac{m_e}{(L_s - L)}$... (2)

Which reduces to $m'_R = \frac{m_e(1+f)}{c * \sigma_t * f^{-1} - L}$ (3)

m_e is the payload in kg,

L is the suspended length in m,

m'_R is the rope density in kg/m.

Empirical formulae can be derived relating rope density and diameter.

For triangular strand ropes,

$M'_R = C_2 * d_R^{1.983} + C_0$

$m'_R = 0.004\,6\ x d_R^{\ 1.983} + 0.15$ (triangular strand) ... (4)

Where D_R is the rope diameter in millimetres.

For non-spin the formulae below applies

$m'_R = 0.004045\ x d_R^{\ 2}$ ... (5)

Simplifying equations 3,4,and5 for m'_R produces a payload:

$$P = \frac{1}{1+s}\left\{\left(\frac{9.61\sigma_t}{f}\right) - L\right\}\left(0.0046 d_R^{1.983} + 0.15\right) \ldots \quad (6)$$

where 's' is the skip factor.

Capacity Factor Limitations

Equation 9 does not hold over the full range of winding depths because of the limitations imposed by the capacity factor, c.

If the capacity is c, then the equivalent end load, m_e, in terms of rope length, L_{eq} must not exceed.

$$L_{eq} = c_0 * \frac{\sigma_t}{c} \quad \text{where } C_0 \text{ is a constant= 9.61} \qquad (7)$$

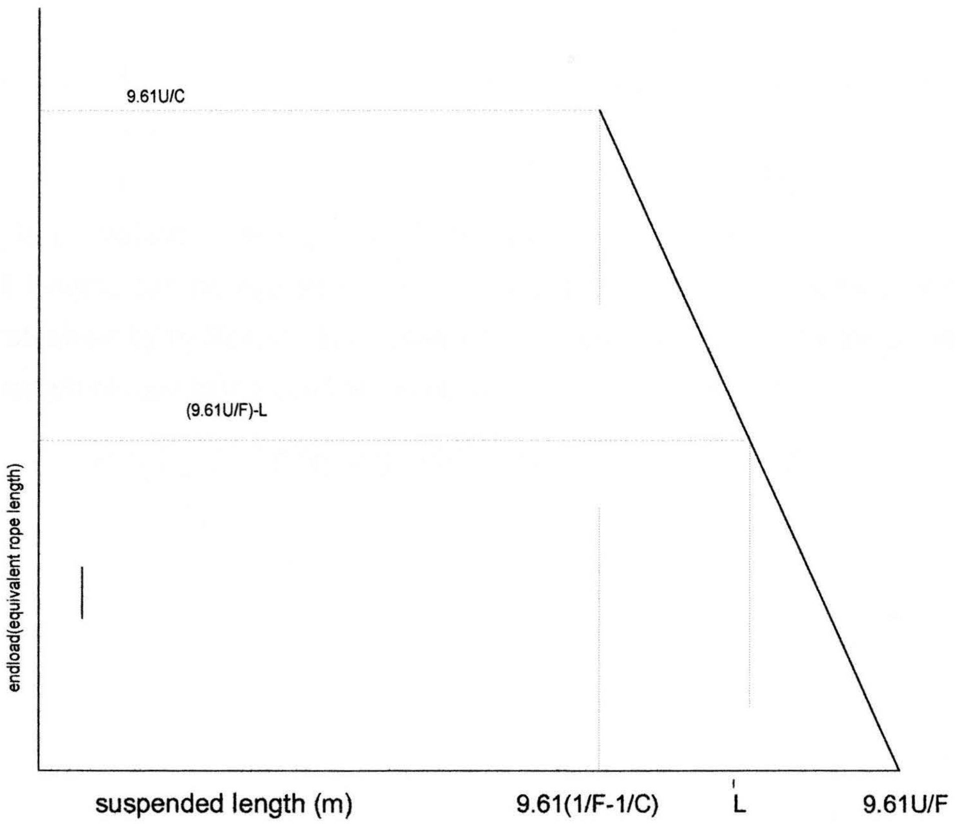


Figure A.1.0- **Variation of End load versus Depth**

As it has already been established that: $L_{eq} = L_u * f^{-1} - L = c_0 * \frac{\sigma_t}{f} - L$

Then at some depth where L=x (x is a " transition" depth)

$$c_0 * \frac{\sigma_t}{c} = (c_0 * \frac{\sigma_t}{f}) - x \qquad (L_u c^{-1} = L_u . f^{-1} - x) \qquad \ldots \qquad (8)$$

Or

$$x = L_u (f^{-1} - c^{-1}) = \frac{L_u}{f - c} \Rightarrow \frac{L_u}{x} = f - c \qquad \ldots \qquad (9)$$

Provided that L is not less than X, equation 9 holds. If L>X, equation9 will hold provided X is substituted for L. Figure A1.0 illustrates this graphically.

Power Requirements

If a rope of length L metres is moving upwards at a speed of v (m/s) and it has a mass of m'_R (kg/m), the power required is

$$P = m'_R L v g \quad kW \qquad \ldots \qquad (10)$$

Where g is acceleration due to gravity ($9.81 m/s^2$).

Since all lengths can be expressed as per units, the power at the various points of the cycle is obtained by multiplying the corresponding equivalent length by the power to raise the unit length of rope being used at the velocity under consideration.

That is, $$P = L_u . f^{-1} . m'_R . v . g / 10^3 \quad [kW] \ldots \qquad (11)$$

Appendix 2

Hydraulic transportation

Flow properties and characteristics

Typical examples of slurries which give rise to homogeneous flow at normal pipeline velocities-1.2 to 2.3 m/s-are drilling muds, and concentrated suspensions of finely ground limestone and mineral concentrates.

Homogeneous flow occurs above the velocity (v_m) given as: $v_m = (1800gD\omega)^{\frac{1}{3}}$

Where g= acceleration due to gravity-9.807m/s ^{2}

D= pipe diameter-m

ω =settling velocity of particle in quiescent liquid –m/s

Design Parameters

The design of a slurry pipeline system requires the establishment of the following variables:

Type of liquid-usually water

Solids concentration

Flow velocity

Pipe diameter

Frictional losses

Solids Concentration All of these variables are governed by the size distribution and specific gravity of the solids. If the solids concentration is fixed, the slurry characteristics can be easily determined. The solids concentration should be maintained as high as possible so as to reduce the amount of liquid required. A knowledge of the relationship between slurry rheology and concentration may be used to select a pumpable concentration. It is advantageous to select a maximum slurry concentration below the point at which the rate of viscosity increase with slurry concentration becomes very high.

The slurry volume fraction soilds, weight fraction soilds, and slurry specific gravity are related as follows:

$$\rho_m = \Phi(\rho_s - \rho_l) + \rho_l$$

$$\rho_m = 1/[W/\rho_s + (1-W)/\rho_l]$$

$$\Phi = W\rho_m/\rho_s$$

$$\Phi = [W/\rho_s]/[W/\rho_s + (1-W)/\rho_l]$$

where ρ_m= slurry specific gravity

Φ= volume fraction solids

ρ_s= solids specific gravity

ρ_l = liquid specific gravity

w = weight fraction solids

Slurry Rheology The presence of solids in a liquid increases the viscosity of the liquid. Theoretical formulas for determining this viscosity change have been derived for uniform-size particle at very low concentration. In most practical application the size of the solids will not be uniform and in most cases the solids will be in high concentration.

Flow Velocity The flow velocity should be such that solid particles do not form a bed on the bottom of a horizontal pipe. Since the particles in slurry are kept in suspension by turbulence in the liquid, it is evident that the flow in a slurry pipeline should be turbulent.

Pipe Diameter: For a given solids through-put and slurry concentration, the slurry flow rate is fixed. A pipe diameter is selected such that the flow velocity is greater than both the laminar turbulent transition velocity and the deposition velocity.

Frictional Losses for Mixed Sizes Most of the formulas for heterogeneous flow are based on the assumption that particles are of a uniform size. When a mixture of two or more size fractions is pumped, a weighted average particle size diameter is derived. Use of a weighted average diameter, or drag coefficient, has been suggested in some research but a better approach has been to divide the solid particles into two fractions: one that is carried in homogeneous flow and one that is carried in heterogeneous and saltation flow.

The frictional losses for each fraction are computed separately using appropriate formula and then added together to obtain the total slurry frictional loss.

Frictional Loss in Vertical Pipe In vertical pipe flow, there is no concentration gradient, and the slurry flow may be treated as homogeneous flow. For coarse particles, the frictional loss for the slurry has been found to be the same as that for water at the same velocity. For fine particles, the viscosity of the slurry should be considered when computing.

Appendix 3- Permissible winding depths vs. payloads and output

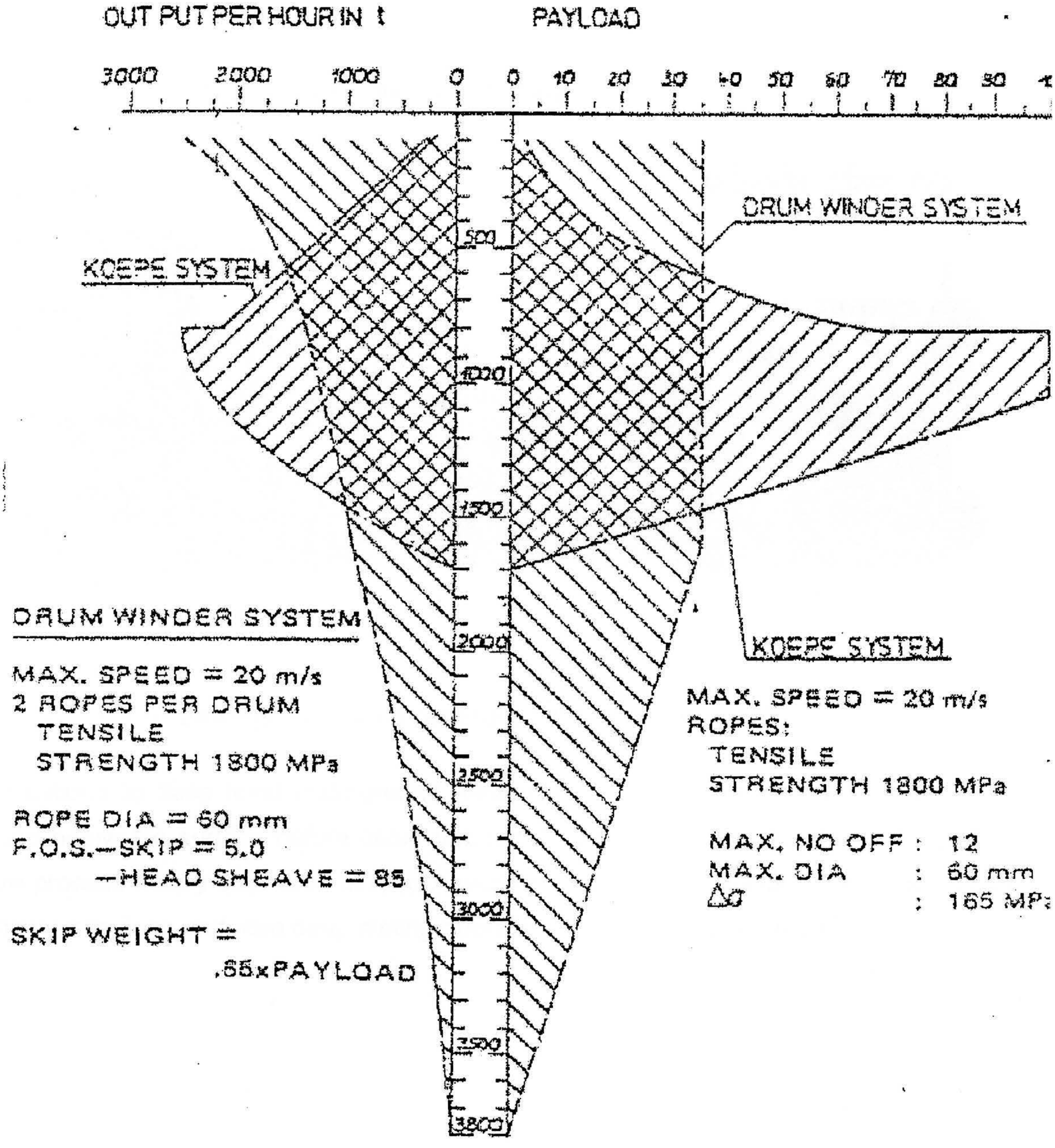


Figure A 3.0- **Permissible payloads and output in relation to shaft depth**

The depth limitation of the Koepe winder as compared to the drum winder. Since BMR is much more superior at depth than drum winders, the above profile shows that economic quantities of material can be hoisted at deeper levels with the BMR winder.

Appendix 4- Underground ore processing

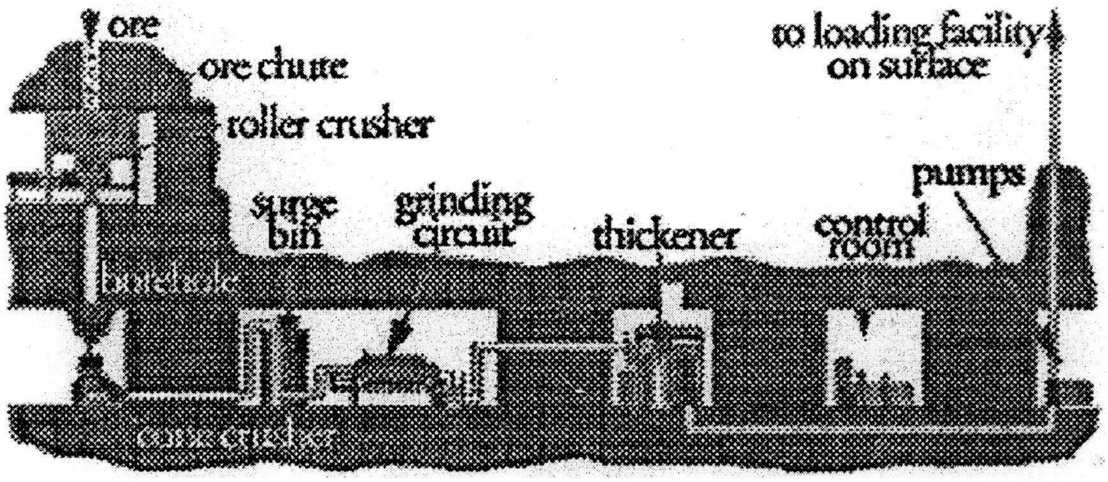


Figure A 4.0- **Underground ore processing of radioactive ore[27].**

Applications to deep level underground mineral processing- Ore will be crushed and ground below the surface before being pumped to the surface in a mud-like slurry. The entire process is fully contained to reduce worker exposure to radiation. This concept can be applied to deep and ultra deep mining operations like gold and platinum.

Appendix 5

Hydraulic hoisting pressure and power requirements curves

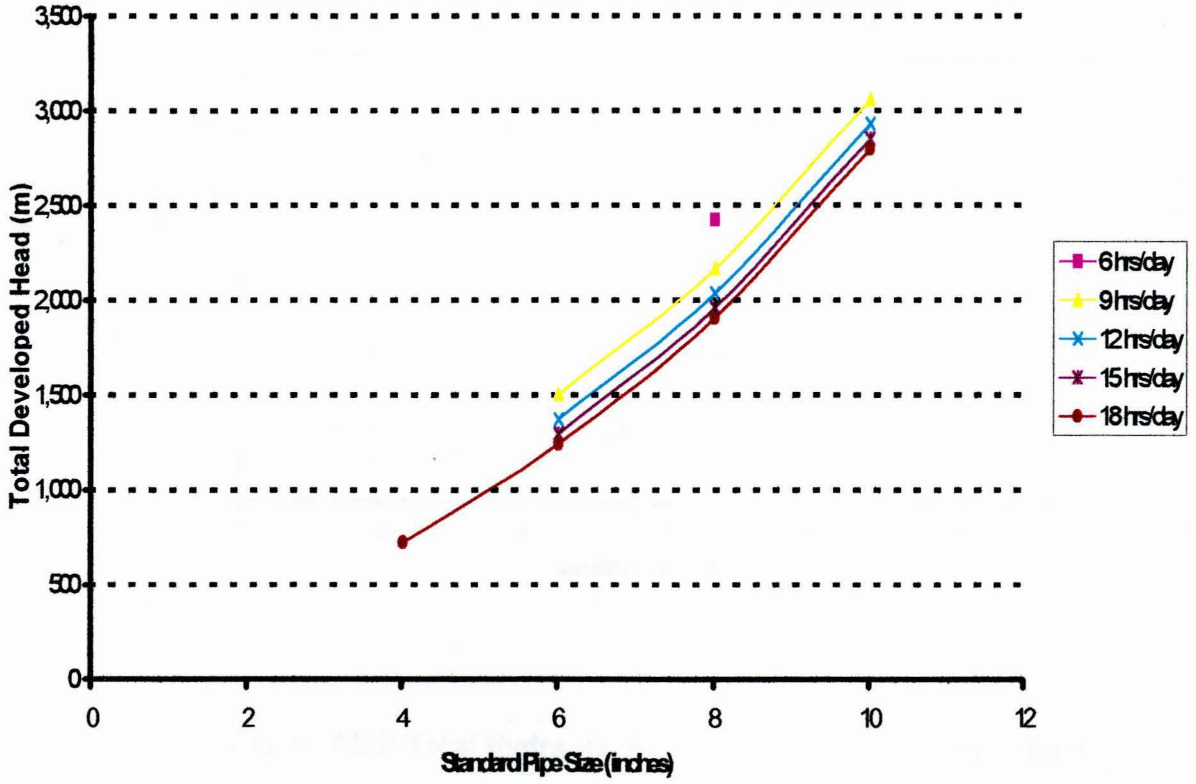


Figure A5.1- **Pump pressure requirement- fine material**

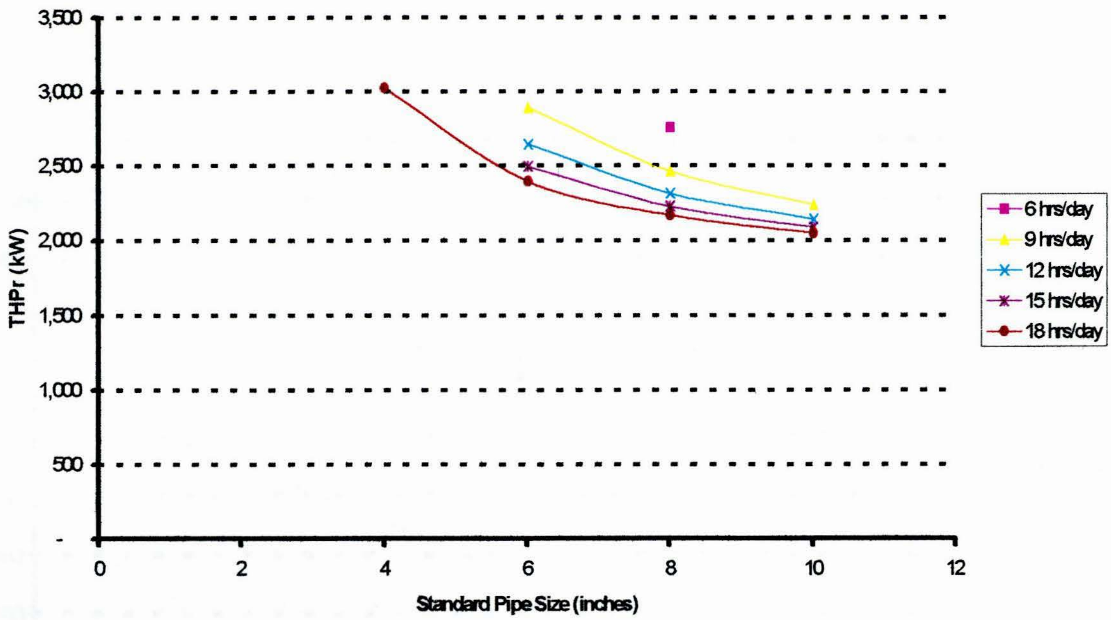


Figure A5.2-**Total Hydraulic Power Requirement-fine material**

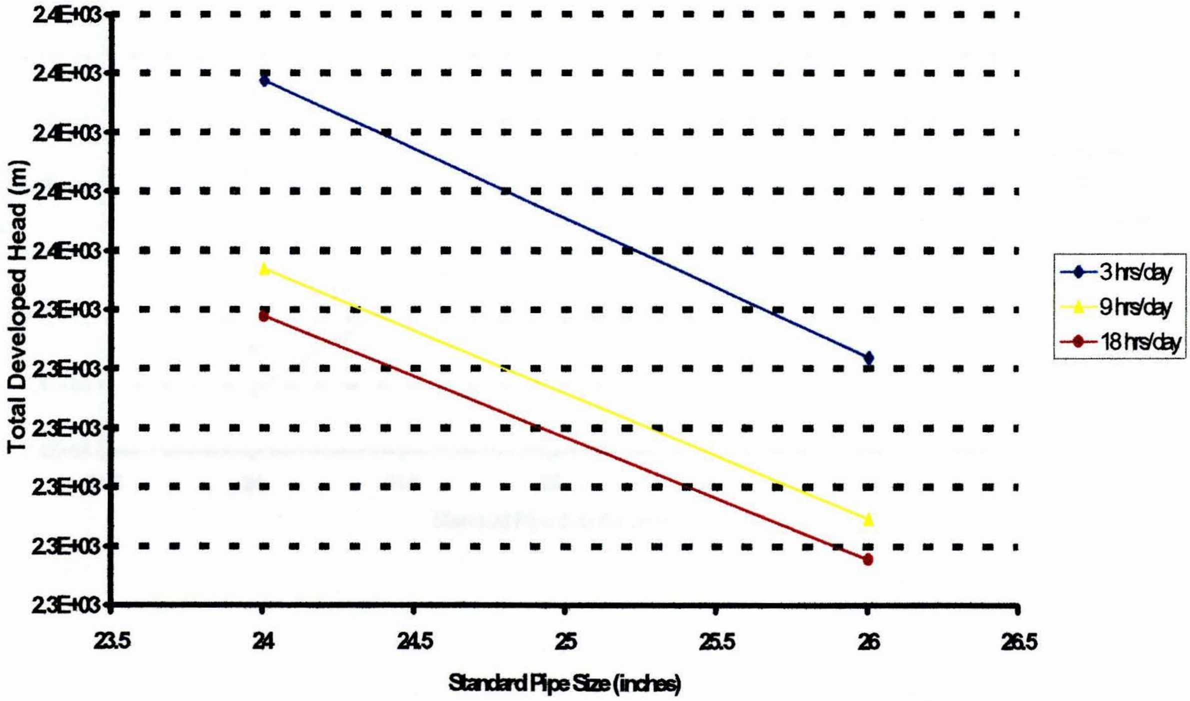


Figure A5.3- **Pump pressure requirement for coarse solids**

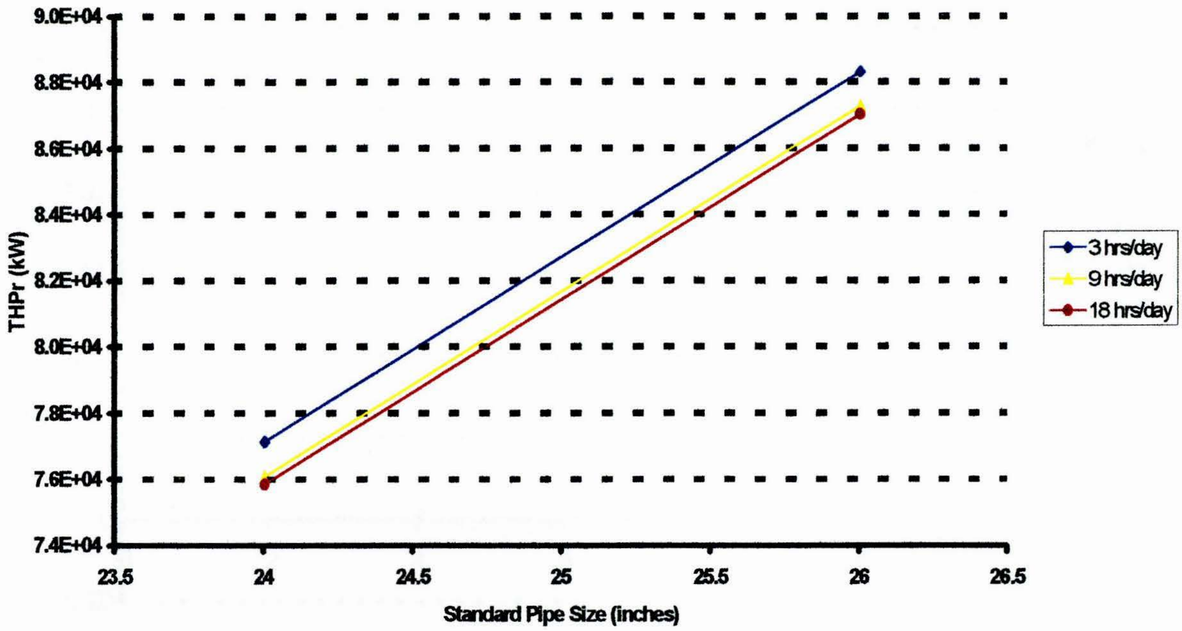


Figure A5.4- **Hydraulic power requirement- Coarse material**

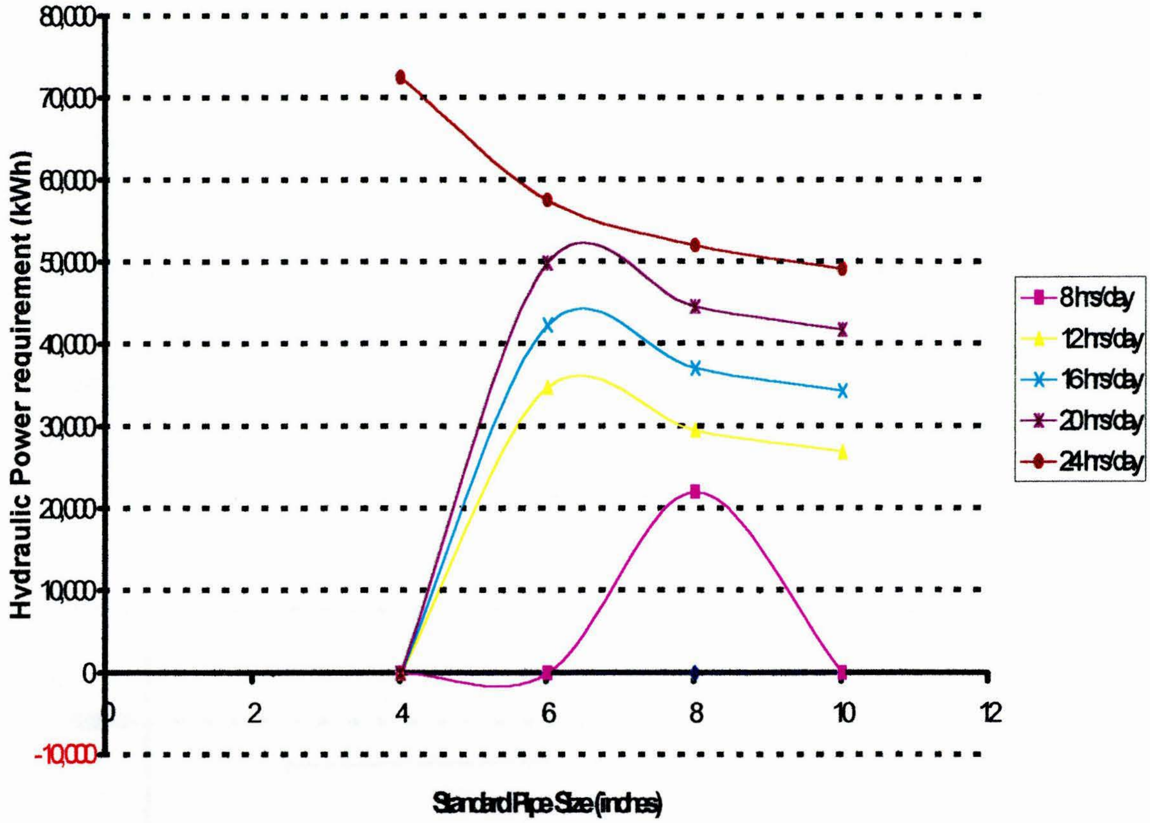


Figure A5.5-**Total hydraulic power requirement –Fine material**

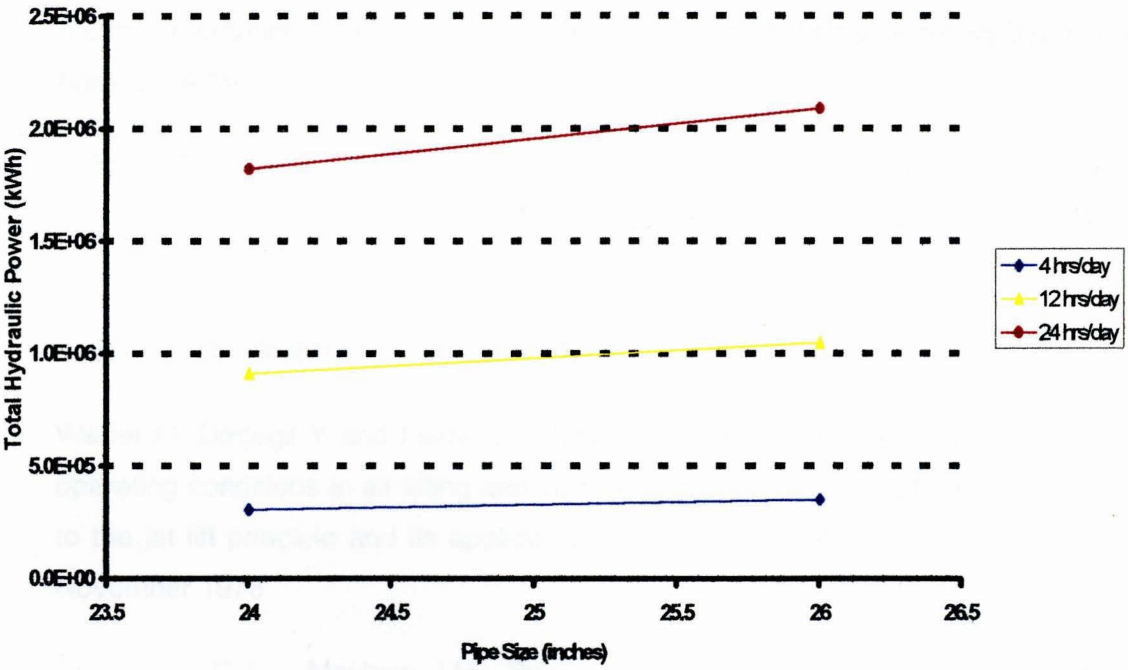


Figure A5.6- **Daily power requirement- coarse material**

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