The Current State of Disc Cutter Design and Development Directions

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ABSTRACT: Rolling disc cutters are the business end of hard rock and mixed face tunnel boring machines (TBMs). Since their first successful employment on a TBM over fifty years ago, disc cutter technology has constantly evolved, allowing modern TBMs to excavate very hard and abrasive rock efficiently. Disc cutters have also been employed successfully on earth pressure balance (EPB) and slurry machines, cutting rock under water and ground pressure. The range of materials excavated by machines today is broader, excavation rates higher and cutter costs lower than ever before, proving the value of investment in cutter development.

This paper describes recent improvements in disc cutter components including cutter ring materials and processing, as well as improvements in lubricants, bearings, seals and cutter condition monitoring. These developments have resulted in more reliable cutters capable of operating in a wide range of geological conditions. Ultimately these improvements result in a better value for both contractors and project owners.

INTRODUCTION

Disc cutters are used on a wide range of tunneling equipment, from pipe jacked slurry micro-TBMs less than a meter in diameter to 15 m diameter, hard rock boring TBMs. The geological conditions under which they are employed range from sands and gravels with several bar of water pressure to extremely hard, massive rock with UCS up to 420 MPa.

Regardless of the type of machine or geology, one thing remains constant: changing worn out cutters is costly. When cutters must be changed in the middle of a tunnel, the contractor incurs the cost of downtime as well as the cost of refurbishing or replacing the cutters. In the case of catastrophic cutter failures, the project can be stopped completely for long periods, and the costs mount up rapidly while tunnel production is at a stand still. Catastrophic cutter failures include, for example, the occasion when hard rock disc cutters are failed in groups (called a "wipeout" phenomenon) and the operator fails to stop the machine, which results in severe damage to the cutterhead. The cause of this can be either an undetected failed cutter propagating damage to surrounding cutters, or operator error in steering. Regardless of the cause, the resulting damage can take days or even weeks to repair. When cutters fail prematurely on EPB or Slurry machines in certain geological conditions, it is impossible to evacuate the chamber and therefore impossible to get into the chamber to change the cutters. The solution is frequently an unplanned intervention shaft which must be sunk at great cost. The importance of being able to predict cutter life accurately, and in all geological conditions, cannot be overemphasized.

For these reasons, cutters have been developed for specific machine types and sizes, as well as for specific geological operating conditions. Clearly, different size cutters are required for different size machines. For example, while 19 and 20-inch (483 and 508 mm) cutters are used on large diameter TBMs, it is impossible to employ cutters so large on a small micro-TBM. Also, the cutter must be designed for the specific geological conditions under which it will be operating. The disc cutter rings required to bore extremely hard rock are the most expensive of cutter rings; however, they provide little advantage in weaker rock formations where far less expensive rings will do the job as well. It is important to choose the correct cutter for the machine and the geological conditions in order to get the best balance of cost and risk.

LARGE DIAMETER, HARD ROCK DISC CUTTER DEVELOPMENT

In the early days of hard rock tunnel boring, TBMs were being employed primarily in weak to moderate strength sedimentary formations. The first successful use of disc cutters was on the Robbins main beam TBM 910-101 in 1952 on the Oahe Dam project in South Dakota where the machine excavated faulted, jointed shale at only 1 to 3 MPa. The cutters were small and looked little like modern hard rock disc cutters.

Because the rock could be excavated with low cutter loads, the cutter bearings didn't need to be very large and the cutters were kept small which made them very easy to handle and change. However, soon TBMs were forced into ever harder rocks, which resulted in an unacceptable rate of cutter wear on the small cutters, along with a rising number of cutter failures due to catastrophic bearing failure. As a result, over the succeeding years, cutter size and bearing capacity increased (see Table 1).

Development of the 19-inch Cutter and Its Application

The story of the development of a successful 19-inch cutter and its application is one of incremental improvement in component design. As one component was improved, another component became the weak link and required further development. However, it was not just the cutter assembly that required continual design improvement, but the application; cutterhead design, cutter management and cutter lubricants all received investigation and improvement over the years.

During development of the 19-inch cutters, Robbins engineers collected massive amounts of data from job site cutter shops, examining and counting failed 17-inch cutter components. When Robbins introduced the first 19-inch cutter in 1989, many things were new in the design, and much improved compared to the existing 17-inch cutter design:

- The cutter ring wear volume of 19 inch cutter was increased by 38% (see Figure 1).
- Ratio of Cutter Load Rating to Cutter Bearing Load Capacity was reduced. Robbins 19-inch cutters full load rating is only 84% of the bearing's rated capacity (i.e., 32 t/38 t) whereas the 17-inch cutter's full load rating is 93% of the bearing's rated capacity (i.e., 27 t/29 t) (see Figure 2).
- The new Wedge-Lock cutter mounting system proved a vast improvement in reliability versus the previous V-block mounting system.

Table 1. Cutter diameters and load rating vs. year

Diameter (inch)	Load (kN)	Year Introduced
11	85	1961
12	125	1969
13	145	1980
14	165	1976
15.5	200	1973
16.25	200	1987
17	215	1983
19	312	1989
20	312	2006

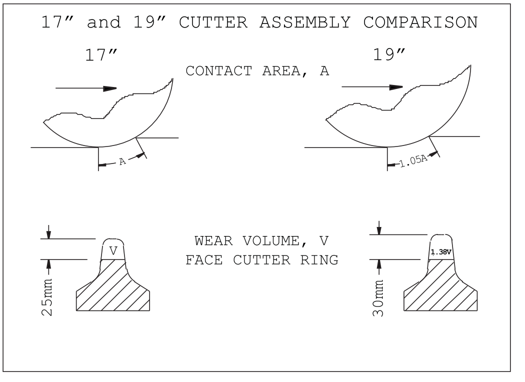
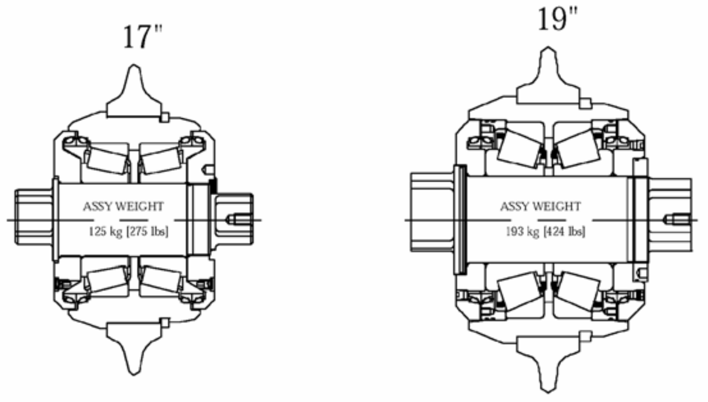


Figure 1. Cutter ring wear volume: 17-inch vs. 19-inch

- The face seal torics were much larger in cross-sectional diameter, allowing greater misalignment. Seal torics, the silicone or nitrile o-ring which support the metal face seal, in the 17-inch V-block cutter had a small cross-section diameter (6.35 mm) which allowed only slight misalignment of the shaft to the hub. Finite element analysis revealed that, when the cutter is severely impact loaded, deflection of the shaft can cause misalignment exceeding that allowed by the small cross-section toric thus allowing ingress of foreign materials. The original toric material also tended to become non-elastic quickly if the cutter temperature was too high. The larger, silicone torics used on the 19-inch Wedge-Lock cutters eliminated both of these problems.
- Cutter hub life was increased dramatically through improved materials and processing to give a very wear resistant hub surface at the hub-to-ring interface.
- The new Wedge-Lock cutter mounting system proved a vast improvement in reliability versus



TONNES	POUNDS		TONNES	POUNDS
27	60,000	TRC RATED CAPACITY	32	70,000
29	63,900	**TIMKEN BASIC RADIAL RATED CAPACITY ***	38	84,500

* 500 RPM 3000 HRS L10

Figure 2. Bearing load rating and assembly weight: 17-inch vs. 19-inch cutter

the previous V-block mounting system. Cutter housing life was also dramatically improved.

Initially, cutter ring life on the 19-inch cutter was not as long as had been expected. Too many rings failed prematurely due to spalling rather than slow wear, and some were fracturing. While the larger bearings could provide good support at the higher loads, clearly the existing cutter ring materials were not capable of withstanding the increased Herzian contact stress. Increasing the cutter ring tip width would reduce the contact stress, but also resulted in reduced penetration into the rock for a given cutter load, moving the problem from the cutter ring to the bearing. Increasing ring tip width was the short term solution; however continuing metallurgical research eventually resulted in rings made from tool steel and later, from proprietary modified tool steels. With proper heat treatment, these rings have much higher hardness as well as increased fracture toughness compared to the previous materials. In addition, these steels retain their strength at the elevated temperatures incurred when boring very hard rock. Today's 19-inch cutter rings can be used in most geology at the same tip widths as 17-inch cutters, allowing 19-inch cutters to penetrate at the same rate with only a very slightly increased load.

Another problem with the early deployment of the 19-inch cutter was a tendency to be susceptible to multiple cutter, wipeout failures. In effect, if one cutter failed catastrophically (i.e., broken ring, failed bearing) there was a tendency for the cutters in the paths next to that cutter to fail also, and the failure pattern might repeat until 5 to 10 cutters were failed in a group. This was eventually identified to be a result of the cutter spacing and was corrected over the following years. As cutterhead design evolved, 19-inch cutter data was collected from the field. With improvements made over the years in cutter rings, 19-inch ring life became very good and the weak link in the cutter seemed to be the bearing, once the strongest part of the design.

When cutters are worn and removed from the TBM, they are subject to one of two treatments before being returned to the TBM:

- Re-ring: Remove and replace the cutter ring, and change the lubricant
- Rebuild: Completely disassemble and replace cutter rings, bearings, seals, other small parts, and/or lubricant.

Obviously, rebuilding is far more expensive than re-ringing. Routine monitoring of the cutter re-ring-to-rebuild ratio is necessary to maintain the lowest consumable cost on projects. A high cutter rering-to-rebuild ratio is indicative of a high-quality cutter body assembly, and always results in lower total cutter cost for the project. For the 19-inch cutter, Robbins wanted to increase the cutter re-ring-to-rebuild ratio. The solution was two-fold: improved lubricants and precise record keeping. The lubricants recommended today are quite expensive, but their return on investment is substantial. Precise record keeping allows the cutter manager to be constantly aware of each cutter's time in service, providing the

Table 2. Improvements in 19-inch cutter ring life

Project Location	Svartisen, Norway 1990	Atlanta, Georgia 2005
Rock Types	Micaschist, granite, chalkstone	Very fine-grained medium grade metamorphic rocks
UCS Range (MPa)	49 to 196 MPa	Average UCS of 255 MPa and localized maximum of 530 MPa
Cutter Life (m^3/ring)	146 m^3	187 m^3

cutter manager the information necessary to predict the probable end of life of the cutter assembly and rebuild it in advance of failure.

Ten years after its introduction, the 19-inch cutter proved itself a clear choice for excavating hard and mixed face rock. TBMs employing the 19-inch cutter were excavating the hardest rock worldwide and setting impressive production records while doing so. See Table 2 for a comparison of cutter ring life at the introduction of the 19-inch cutter with that from more recent projects.

Having increased the cutter re-ring-to-rebuild ratio, the next step was for engineers to once again extend the life of the cutter ring.

Development of the 20-inch Cutter

There are two methods by which one can increase disc cutter ring life:

- Increase cutter ring strength and abrasion resistance.
- Increase the wear volume available on the cutter ring.

Research of new disc metallurgy and processing to increase cutter ring life in hard and abrasive rock has yielded incremental improvements, at high cost for the research. Currently 19-inch cutter rings are being manufactured using three different cutter ring metallurgies and heat treatment processes (Standard, Heavy Duty, and Extra Heavy Duty). Some research test rings have shown marked improvement in ring life, but only at an unacceptable cost for the rings. While metallurgical research continued, in the short term there was no economical way to improve cutter ring life through increases in material strength and/or abrasion resistance. It became clear that an increase in wear volume was the most cost effective way to improve cutter ring life for the 19-inch cutter.

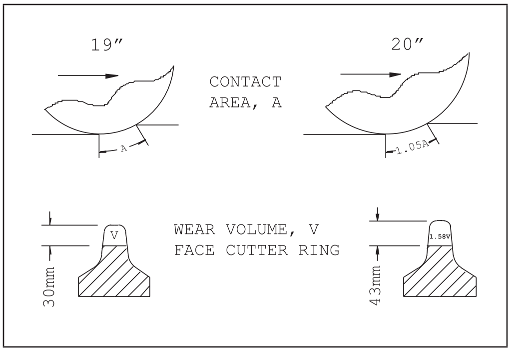


Figure 3. Wear volume: 19-inch vs. 20-inch ring

To increase the wear volume one can increase the cutter ring tip width, the cutter ring diameter or both. Obviously, increases in cutter ring tip width have an adverse effect on cutter ring penetration into the rock. To achieve the same penetration, a cutter ring with an increased tip width would require an increase in thrust, which would have an adverse affect on the cutter bearing. However, an increase in diameter would have only a negligible effect on cutter load to achieve the same penetration. Essentially, one can increase the "tip length" to provide more wear volume in the ring. The risk of a longer cutter ring tip is the potential for fracturing of the long tip.

This story is somewhat similar to the advent of the 17-inch cutter, which was simply a change in the cutter ring diameter mounted on the original 15.5-inch cutter assembly. And, the same logic was used when the extended tip 17-inch cutter was employed successfully on many sedimentary rock jobs. The cutter provided deeper penetration capability and larger wear volume compared to a standard 17-inch cutter ring. This experience provided empirical field data to support the concept that an extended tip 19-inch cutter might prove successful.

Using a conservative approach, new 20-inch cutters were developed for use on TBMs which could use either 19 or 20-inch disc rings. The 19-inch cutter hub/bearing/shaft assembly is used for both 19- and 20-inch cutter rings, so either cutter ring can be employed. Cutter housings and cutterheads were initially designed to accept either the 19- or 20-inch ring cutters for the following three TBMs:

- 14.5 m, Hard Rock Open, High Performance TBM (HP-TBM) for the Niagara, Canada hydropower expansion.
- Two 10.0 m, Hard Rock Double Shield, HP-TBMs for the AMR water transfer project in India.

The 20-inch cutter rings produced offer a 58% increase in wear volume compared to the 19-inch cutter ring (see Figure 3).

Figure 4. The cutter on the left quit turning and wore flat. The cutter on the right experienced a massive bearing failure that resulted in a damaged seal retainer and shaft.

Initial reports from the Niagara project are good, with the 20-inch cutter rings in continual use in the sedimentary rock, giving good penetration and good life to date. The AMR project requires boring through hard rock from 160 to 190 MPa UCS. This will be the first hard rock test of the new 20-inch cutter ring. The machines will start boring in mid-2008 and cutter ring performance will be reported after data has been collected from the project.

CUTTER BEARING DEVELOPMENT

Cutter Bearings

Most rolling disc cutters use two tapered roller bearings that are arranged in what the bearing industry calls "indirect mounting." These bearings are asked to cope with extremely dynamic loading and vibration in a very harsh environment. As cutter rings have grown, so have the bearings and the loads they must handle. In this section we will discuss bearing life and failure modes.

Catastrophic Failures—Post Mortem Inspection

When a cutter assembly fails catastrophically (see Figure 4), it is never easy to determine how the failure was initiated. A post mortem inspection is the only way to diagnose and remedy the problems. There is always some amount of doubt as to why the cutter assembly is full of tunnel muck, which results in some classic questions. Did a bearing(s) fail, allowing misalignment and letting muck pass the seals? Or did the seals fail and allow contamination of the bearing? Fortunately most cutter/bearing failures are not this severe and are usually detected and repaired/replaced before they get to this point.

Seal Inspection

Seal failure is a death sentence for a mechanical assembly in an underground environment and nowhere is this more certain than with TBM cutters. Post mortem inspections of catastrophically failed cutters generally start with the seals. If for no other reason they are the first part to be removed during disassembly. In some instances post mortem seal inspection will immediately reveal the "smoking gun" but quite often the mode of failure isn't so obvious. Complicating things further, it is not always easy to determine the cause of failure once the failure mode is isolated.

Modes of seal failure:

- Damaged torics
- Fused faces
- Abrasive wear

Causes of seal failure:

- Assembly error
  - Too much drag
  - Face pressure too low
- Rust from long periods of no use
- Packing—Seal filled with clay or mud that is allowed to harden while cutter is not turning
- Use past service life

Bearing Inspection

In most hard rock tunneling conditions the cutter bearings will remain serviceable through multiple cutter rings if maintained properly. Proper maintenance requires frequent inspections and these inspections often reveal impending problems. If failing bearings are not detected, adjacent cutters will become overloaded and prematurely fail and lead to a chain reaction of multiple cutter failures. The TBM industry calls this phenomenon a "wipeout," which can cost hundreds of thousands of dollars in equipment and downtime (see Figure 5). Much like seal failures, this mode of failure does not always leave a clear path to the original cause of the failure.

Modes of bearing failure:

- Spalled or grooved raceways (see Figure 6)
- Worn or broken roller cage
- Damaged rollers
- Brinelling/false brinelling

Causes of bearing failure:

- Extreme dynamic loading
- Overloading

Figure 5. Example of a cutter wipeout phenomenon

Figure 6. Close up of spalling in the load zone of the inner race

- Loss of lubrication
- Overheating
- Assembly error—Too much or too little pre-load
- Contamination—ingress of tunnel muck—usually a seal failure

Bearing Life—Theoretical vs. Actual

There is a wealth of information and many studies have been done on calculating and estimating bearing life in mechanical assemblies. Bearing manufacturers and 3rd party associations such as ISO, SAE and the ASME have done comprehensive scientific research and have developed mathematical models that predict serviceable duration under controlled conditions. ISO 281 is generally accepted as the industry standard that provides an estimated number of cycles or hours that the bearing can be expected to perform. Unfortunately, experience tells us that these models are not able to accurately predict bearing life in disc cutters. Problems with predicting bearing life in disc cutters are two-fold. First and foremost, the industry has not been able to accurately define and quantify the extreme dynamic loads to which these bearings are subject and secondly geology dictates that conditions are always far from controlled or consistent.

Table 3. Comparison of calculated vs. actual bearing life

	17" Cutters	19" Cutters
Calculated Bearing Life (Hours)	43	2165
Actual Field Data:	Manapouri, NZ	Cobb County, Atlanta
TBM Working Time (Hours)	7950	3117
Actual Bearing Sets Used	1612	191
Average Bearing Set Life (Hours)	5	16
Ratio of Actual/Standard L10	11.47%	0.75%

Theoretical Life

The following definition is from the ISO's website (ISO 281, 2007). "[ISO 281:2007] specifies methods of calculating the basic rating life, which is the life associated with 90% reliability, with commonly used high quality material, good manufacturing quality and with conventional operating conditions. In addition, it specifies methods of calculating the modified rating life, in which various reliabilities, lubrication condition, contaminated lubricant and fatigue load of the bearing are taken into account. ISO 281:2007 does not cover the influence of wear, corrosion and electrical erosion on bearing life." Table 3 provides examples from two long term tunnel projects. One of the jobs used 17" cutters for a duration of 7,950 total operating hours. The other job utilized 19" cutters for 3,117 hours. In both cases the rock was generally considered to be hard to extremely hard (100 MPa UCS and higher). The calculated bearing set life per ISO 281 should have been 43 hours for the 17" cutters and a very impressive sounding 2,165 hours for the 19" cutters.

Actual Life

In practice, actual service life varies drastically from theoretical life estimate. In addition to extreme dynamic loads on bearings, cutters can also be damaged due to service factors such as ingress of debris

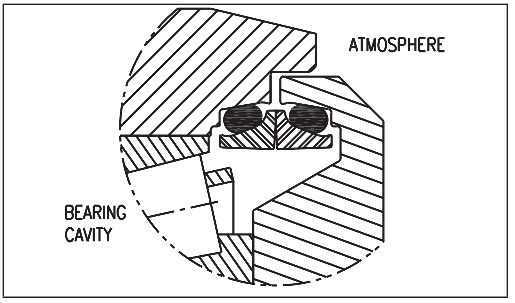


Figure 7. Cross-sectional diagram showing cutter seal

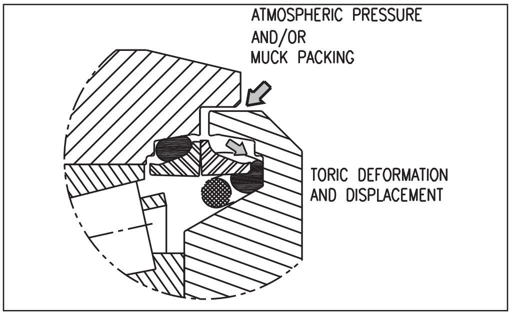


Figure 8. Seal toric displaced by fluidized slurry at high pressures

and water (seal failure). The damaged cutters typically show bearings with severe spalling in the load zone, damaged rollers, broken cages and or overheating. Sometimes the normal inspection or servicing of cutters may show early impending bearing damage, and in these cases bearing components are replaced much more frequently than their theoretical lifespan. Table 3 shows that actual bearing life in the aforementioned projects was orders of magnitude shorter than the ISO 281 calculation predicted.

CUTTER SEAL DEVELOPMENT

One of the most significant developments in disc cutter technology came with the application of the Caterpillar metal face seal in the early 1970s (Handbook of Mining and Tunnelling Machinery, 1982). This seal type uses two metal rings that are loaded axially such that they ride against each other with a film of lubricant between them, creating a dynamic face-seal interface. Each ring, one on the rotating hub, and one on the stationary shaft, is sealed to the mounting gland by a rubber ring (toric, or o-ring), which also acts to allow a relatively generous hub-to-shaft misalignment tolerance, and as a spring to load one seal ring against the other (see Figure 7). Although proven effective over 30 years of hard rock disc cutter application, there are problems associated with the seals in some specific applications.

One of the problems encountered with this seal is when a slurry of certain rock types pushes through the labyrinth created by the hub and seal retainers and then, given time, dries, effectively cementing one metal face seal ring to the other. When the cutter starts rolling again, the cemented metal rings may momentarily spin with each other and stretch the rubber toric holding the metal seal rings. This can rip the toric, allowing leakage of lubricant out of the cutter, as well as ingress of dirt and abrasives into the bearing cavity. To alleviate this problem, the toric should minimize the amount of metal ring exposed to the slurry, hopefully allowing the rings to break free and rotate against each other as intended.

Another problem arises when the cutters are used in high pressure applications, such as on EPB and slurry machines. The standard seal is rated, per the manufacturers' recommendations, for about 3 bar of pressure differential between the atmosphere and the bearing cavity. Three bar of pressure is reached at 30 meters below the water table and many tunnels are far deeper. The problem that arises with higher pressure is the fluidized slurry pushes against the toric, forcing it down the seal gland ramps. This forces the metal rings tighter against each other, increasing face contact pressure of the seal surfaces, causing the seal rings to act as a brake. This has the effect of 'locking up' the cutter, preventing it from rolling in softer material. In extreme cases, with the smaller diameter torics used in many older cutter designs of the 1970s and 1980s, a portion of the toric on the non-rotating side of the seal can push past the back of the seal, causing a major leak (see Figure 8). This problem is lessened considerably by use of larger diameter torics, such as those used today.

Various methods have been tried over the years to try to compensate the internal pressure in cutters to match the exterior pressure, in order to overcome the pressure differential problems noted in the paragraph above. The following is a list of some of the methods by which pressure compensation has been attempted:

1. A pressure compensating piston in the center of the shaft, which moves with the increase in exterior pressure, causing the interior pressure to rise to an equal level. The problem with this method is plugging of the exterior side of the pressure compensator with muck, preventing the piston from traveling in the opposite direction, which can over-pressurize the interior, destroying the seals.
2. A diaphragm, in the shape of an annular ring mounted in the seal retainer, which moves as the atmosphere side pressure rises, causing the internal pressure to rise. This, too, has problems with

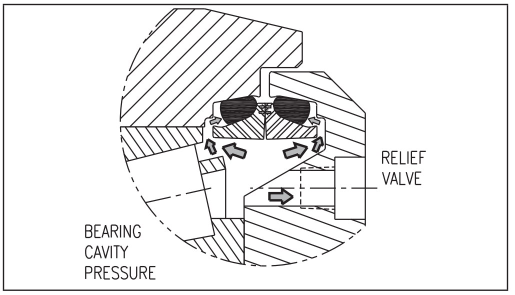


Figure 9. Locking cutter rotation through internal pressure

the breather holes in the diaphragm's protective cover becoming easily clogged.

3. A spring-loaded piston, which can supply between 2 and 3 bar of pre-load pressure internally, causing a maximum differential of 3 bar when the exterior pressure is up to 6 bar. This method was left open to atmosphere (through a breather) on one early version; however, it was found that closing the atmosphere side and allowing a partial vacuum to develop still allowed the devise to work satisfactorily.

Methods 1 and 2 have been successful in applications where the amount of fluid greatly exceeds the amount of muck, thus having a flushing affect on the exterior side of the pressure compensator. Where the muck is in more of a paste form, plugging of the exterior side of the devices is a major problem for these two methods.

All of these types of cutters generally use grease as the lubricant, and in most cases the cutters have a pressure relief valve. It has been shown that without a pressure relief valve, heat from normal operation can cause internal pressure to increase sufficiently to push the torics hard toward each other, locking the cutter rotation as the seal rings act as a brake (see Figure 9).

An obvious, but as yet untried method of pressure compensation would be to have all of the cutters plumbed together to a common lubricant supply. Pressure in the cutterhead chamber would be sensed and the cutter hydraulic system would automatically apply the proper internal pressure to the cutters through the plumbing lines. The drawbacks to this method would be isolation of the lubricant during cutter change and the danger of a leak anywhere in a common lube system.

The overall goal is to develop either an effective pressure compensation devise, or a modified/new seal design that uses the robust characteristics of the metal-to-metal seal currently in widespread use, but make modifications that prevent pressure differential from affecting seal operation.

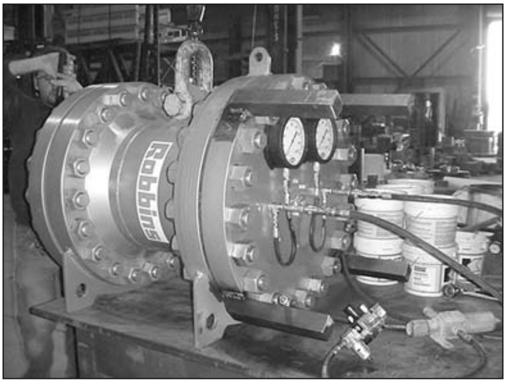


Figure 10. Test fixture to examine cutter rolling torque at various pressure differentials and temperatures

To provide rapid development of an improved EPB cutter, a test fixture was built that allows mounting either a single or twin disc cutter inside a pressure vessel (see Figure 10). Both the atmospheric side and cutter bearing cavity can have their pressure adjusted and monitored independently. The cutter can be rotated during testing at any chosen rpm. The rolling torque of the cutter can be measured at various pressure differentials or temperatures. The temperature of the lubricant adjacent to the seal can be monitored. The pressure (atmosphere and/or bearing cavity) can be raised to as much as 60 bar.

A variety of tests have already been run on both standard and modified seal sets. These tests have shown the effect of various pressure compensation devises. While new designs have not yet been finalized, a better understanding of standard seal performance has been realized. Tests on slightly modified seals (material and surface finish changes) have shown promising improvements are possible (at some cost trade-off, naturally). While initial major redesigns of seal configurations have been less promising so far, the test fixture has proven quite valuable in efficiently reaching conclusions at a speed which cannot be duplicated in the field.

CUTTER RING DEVELOPMENT

The Goal: Increase Advance Rates, Decrease Downtime

Cutter rings have frequently proven to be the limiting component for further improved cutter performance. Recent advancements in cutter technology have continually increased production while lowering overall cutter costs. Continual refinement in proprietary alloys and metallurgical processes has improved the hardness, abrasion resistance and most importantly the toughness of today's high performance Heavy Duty (HD) and Extra Heavy Duty (XHD) cutter rings. Cutter ring development has improved production in two ways:

- Higher penetration per revolution is possible because the Heavy Duty cutter ring stays sharper longer than standard rings. Furthermore, because the HD cutter ring is stronger than standard material, narrower tip width rings can be used without breaking.
- Longer cutter life means less down time for cutter changing and more time available for boring.

Hardness

The high contact stress seen by the highly loaded 19-inch cutters has pushed metallurgical research to new highs. Engineers quickly moved from commercial steels to proprietary steels in the search for higher strength and toughness. In 1995, a Main Beam TBM was put on to the Midmar project in South Africa. This project included dolerites and sandstones with strengths to 350 MPa. The demands on rings were severe and resulted in another round of materials and processing R&D. The result was state-of-the-art Heavy Duty rings, in use on more hard rock projects today than any other ring. This new material improves the cutter life because it has high "hot strength." The material has a reduced wear rate compared to the standard cutter when boring hard rock.

Toughness

It is intuitive to think that the key to creating a high performance disc cutter is extreme hardness. This is true to some degree but it is not the sole factor. Hardness is needed to retard deformation when the ring is pressed against rock but the hardness is useless if it is too brittle, therefore toughness and hardness are the most important properties for the cutter designer to manage. The standard toughness test for steels is the Charpy Impact test which consists of one blow from a swinging pendulum under defined, standard conditions. To increase toughness, sophisticated analysis and post processing methods have been used to monitor and control grain size, microstructure and chemical composition. Continual refinement of the heat treating and tempering processes has proven to be of equal importance to specifying the proper alloy if the goal is to increase toughness and hardness simultaneously. The toughness obtained in proprietary steel rings has allowed the manufacture of rings that are even harder than was considered possible just a few years ago.

DEVELOPING TECHNOLOGY: REMOTE CUTTER STATUS MONITORING

The current focus in disc cutter design and application is finding the optimum cross point of disc cutter cost vs. boring system performance, measured in penetration rate and availability. There are three basic components to TBM downtime related to cutters:

1. Cutter inspection: Hard rock TBMs must be routinely stopped to allow workers to inspect the condition of all the cutters, to insure that there are no failed cutters or severely worn cutter rings, to insure that the TBM can continue to be operated until the regularly scheduled cutter change.
2. Cutter change: Routine maintenance encompassing: Inspections for mechanical fitness and replacement of cutters with worn rings or damaged bearing/seal assemblies, as well as moving cutters for even ring wear.
3. Cutterhead repairs: This is related to either extreme geological conditions (abrasive rock in fault zones or blocky rock) or the failure of a series of cutters, either of which can lead to cutterhead damage. The work required may include repair of cutter housings, cutterhead wear plating, muck buckets and/or conveyor components.

Clearly, a system which constantly monitors the status of every cutter on the cutterhead would provide a tremendous advantage. If one could monitor all the cutters at all times, it would not be necessary to make routine physical inspections (#1 above), which would reduce TBM downtime. Furthermore, if it was possible to constantly know the status of every cutter, then the operator would know immediately when a cutter failed, or was beginning to fail, and stop the machine thereby preventing a wipeout cutter failure before its occurrence and preventing damage to the cutterhead.

Historically, it was entirely up to the TBM operator to detect a failure by noting an increase in cutterhead torque requirements, which is very difficult to detect given a single cutter failure on a machine which may be fitted with 20 to 80 cutters. Some TBM operators have reported being able to detect the aroma of overheating cutter lubricant in the tunnel from a very hot cutter—not a desirable or dependable failure detection system. TBM users have long sought a cutter monitoring system whereby the operator would get more precise information on the status of each cutter. In the past, the only way to get this data from the rotating cutters and cutterhead to the operator was via a hard-wired system with multiple contact rotating union, which proved wholly unreliable in the harsh environment.

New Developments in Remote Cutter Status Monitoring

Today, engineers are developing real-time cutter monitoring systems with the following specifications:

- Monitoring of every cutter on the TBM.
- Indication of smooth rolling of all cutters on an human-machine interface (HMI) display in the TBM.
- Cutter rolling alarm, with cutter number, when any cutter stops rolling
- Indication of cutter wear, for every cutter, on an HMI display
- Indication of cutter temperature, for every cutter, on an HMI display
- Cutter temperature alarm, with cutter number, when any cutter temperature exceeds the preset limit
- Wireless transmission of all data

Rotational Speed of the Cutter

Cutters are not individually powered. They roll only because the cutterhead in which they are mounted presses the cutters against the rock face as the cutterhead rotates. The diameter of their circular, concentric path is determined by how far they are mounted from the rotating cutterhead center. The rotation speed of the cutter is a function of the radius of their mounting position on the cutterhead, the rotational speed of the cutterhead and the diameter of the cutter ring, which changes as the cutter wears. Knowing the rotating speed of each cutter will provide two pieces of information:

1. Whether the cutter is rolling or not rolling. This information helps us to detect catastrophically failed cutters before the failure propagates to other cutters or causes cutterhead damage.
2. The diameter of the cutter, which gives the cutter ring wear. This information eliminates the need for downtime for routine cutter inspections, and helps to plan in advance for cutter changes.

Temperature of the Cutter

The temperature of the cutter is also an indicator of mechanical condition of a cutter. Unusually high temperature can indicate slipping seals, failing bearings, loss of lubricant or ingress of foreign materials. The temperature of the entire cutterhead increases during a boring stroke. Other TBM mechanical problems might be indicated by the temperature rising above the average normal increase during a boring stroke. Results obtained from the first machines fitted with this system will be presented in a future paper.

CONCLUSIONS

Industry investment in cutter development has resulted in cutters today which efficiently bore through an ever wider range of materials at ever lower cost in real, inflation-adjusted dollars. Improvements in pressure compensation devices will improve the survivability of cutters operating under pressure. Continuous, real-time cutter condition monitoring will become a standard in a very short time and will result in saving scores of times the system cost.

Substantial challenges, however, remain. We must continue to fund metallurgical and materials processing research to find the next cutter ring material, allowing rings to be loaded ever higher, increasing TBM advance rates in hard rock. We must search for more abrasion resistant materials and improved sealing systems to improve cutter life on EPB and slurry machines.

Improvements in cutter technology have resulted in lower total project costs for contractors and owners worldwide. However, we must all remain aware of the need to continually fund research and development in this area if we hope to continue the trend of improved systems reliability and performance with reduced cost.

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