The design and development of a rockcutting machine for gold mining

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SUMMARY

This paper describes the design and construction of the first experimental underground rockcutting machine from measurements of cutting forces determined in tests on small rock specimens in the workshop. The deficiencies of this design, revealed by stoping with almost identical rockcutting machines, are described. An account is given of the design and construction of further experimental machines incorporating substantial improvements which made them technically feasible for use in production.

INTRODUCTION

One of the most serious problems in South African gold mining is that during rockbreaking by means of explosives the goldbearing portion of the reefs, which is often less than 15 cm thick, becomes intimately mixed with the adjacent waste rock. Other major problems also result from the use of explosives. Most mining takes place at great depths, and the severe rock mechanics and thermal control problems which arise can be shown to be proportional to the effective stoping width[1, 2]. With explosive rockbreaking, the effective stoping width or milling width is almost identical to the actual stoping width of about 1 m, since so little of the waste rock can be sorted and packed. In addition, the use of explosives precludes continuous mining because virtually all personnel have to be evacuated from the mine before blasting and are not permitted to re-enter until a few hours after the blast.

Economic assessments[3] have shown that substantial benefits would result from the use of an alternative method of rockbreaking which would permit the reef to be mined more selectively. With the view to the development of a continuous, selective mining process an investigation was made of most known methods of rockbreaking[4]. These included ultra high pressure water jets, mechanical wedging with or without impact, diamond sawing, roller bits, percussive tools, drag bits, various thermal methods such as jet piercing, lasers, electron beams and electrical techniques. A drag bit method with a tungsten carbide tipped tool for cutting the rock was selected as the most feasible of the techniques applicable within the near future for developing a machine which could be made to mine selectively under the conditions encountered in deep-level, hard rock mining.

Rockcutting investigations were carried out in a shaping machine and in a lathe with the aid of instrumented tool shanks to measure the cutting forces and tool wear on several types of rock for various feeds, depths of cut and cutting speeds[4]. In order to simulate the cutting of a long slot a large drill core 15 cm in diameter was set up in a lathe and a rectangular section groove was cut using a feed, as in thread cutting, of more than twice the width of the cutting edge.

These experiments showed that the force acting on the cutting tool consisted of two roughly equal components, namely, the cutting force parallel to the direction of the movement of the tool relative to the rock and the penetrating force acting at right angles to the rock surface. The cutting and penetrating forces on a tool were estimated to be of the order of 40 kN when making a cut 20 mm wide and 9 mm deep in moderately hard rock. Wear on the tool was found to be moderate provided the cutting speed was less than 1.5 m/sec. Specific tool wear was less than 1 part of tungsten carbide in 25 000 parts of quartzite by weight when deep cuts, 2 mm or more, were made at speeds of less than 0.5 m/sec.

These results indicated that it was possible to design a machine capable of cutting at a rate of 450 cm^2/min using a cutting speed of 15 cm/sec and a depth of cut of 5 mm, which would result in a rate of mining of 8 centares per six-hour shift, cutting for, say, half of the time.

This paper discusses the various configurations for such cutting machines and describes the design of the first prototype, underground rockcutting machine using a linear, reciprocating cutting action. The testing of this and other similar rockcutting machines showed that this design formed the basis of a feasible system of continuous selective mining, and revealed deficiencies in detail which had to be overcome to yield a machine capable of mining at an economic rate. The design of a second prototype, incorporating the necessary improvements, is described and the development of each of the sub-assemblies towards a reliable rockcutting machine suitable for use in production is detailed.

CONCEPTUAL DESIGN

Selective mining of the substantially plane gold-bearing reef could be achieved by cutting parallel slots in the waste rock immediately above and below the reef. These slots would separate the reef from the waste rock and relieve the vertical stress acting on the reef, which otherwise prevents its easy removal from the stope face. At depth the high vertical stress in the face would cause the reef between the slots to become detached from the face in the form of slabs much in the same manner as discing of diamond drill cores occurs when drilling in rock subjected to high stress transverse to the drill axis[5]. In the event of the stresses in the face being

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insufficient to cause slabbing, the reef between the slots could be detached from the rock ahead of the slots by wedging or jacking. The waste rock on either side of the slots would then be removed to restore the face shape and stoping width. Breaking of this waste rock would also be greatly facilitated by the removal of the vertical stress. The reef would be transported from the stope and the waste rock would be packed in the mined-out area immediately behind the face.

The gold-bearing portion of the most important reefs in South African gold mines is 15 cm or less in thickness[4] and deviates less than 10 cm from flat over distances of about 3 m. If the slots could be made to follow the undulations of the reef, a separation between them of 20 cm would be sufficient to embrace the gold-bearing portion but, if the slots had to be straight over a distance of about 3 m due to machine configuration, a separation of about 25 cm would be needed to embrace the gold-bearing portion.

Whenever an excavation is extended, the weight of the overlying strata causes substantial and immediate convergence of the roof and floor. Removal of the vertical stress by slotting gives rise to elastic closure of the slots at the rate of about 2 mm for every 100 mm depth of slotting. This closure had to be allowed for in the design of a slot-cutting machine and imposed a practical limit of about 30 cm on the depth of slot which would be cut without removing the surrounding rock.

In principle, it was decided to use a tungsten carbide drag bit mounted at the end of a thin blade which would be used to deepen the slot by up to half the width of the drag bit at each pass. The bit could be made to generate the required slot while moving in a path which was curved, linear, or a combination of curved and linear motions. Accordingly, three main cutting configurations were considered and these are shown in Fig. 1.

A machine using a rotary action, Fig, 1a, was rejected because it suffered from three major drawbacks. Firstly, the varying depth of cut inherent in this configuration would cause excessive tool wear by abrasion during the shallow portion of the cut. Secondly, the need for the tool to re-enter the slot at each revolution would cause wear on the sides of the bit as a result of the continuous closure of the slot as it is deepened. Thirdly, the considerable engineering difficulties in building a rotary machine of acceptable weight and bulk capable of exerting a torque of about 70 kN/m to drive the cutter were evident at an early stage of the design study.

A machine using both rotary and linear cutting actions in a manner of a chain saw, Fig. 1b, appeared to offer many advantages over the rotary cutter.

Used with a plunge feed normal to the stope face, as shown in the figure, this machine would make possible the attainment of near optimum drag bit operating conditions. Cutting depth along the base of the slot would be constant and the variation in cutting depth during the curved portions of the cut would not necessarily be excessive. The use of two tools as shown would result in almost continuous cutting and a greater number of tools could be used to increase the rate of cutting if needed. Furthermore, this machine would be able to cut into a straight face.

Unfortunately the engineering difficulties involved in the design and manufacture of such a machine are formidable. The mechanism for driving and guiding the tools together with a means for feeding them into the face are likely to make this machine bulky and expensive, and it would certainly take longer to develop than a machine based on a simpler configuration. This concept was, therefore, also rejected.

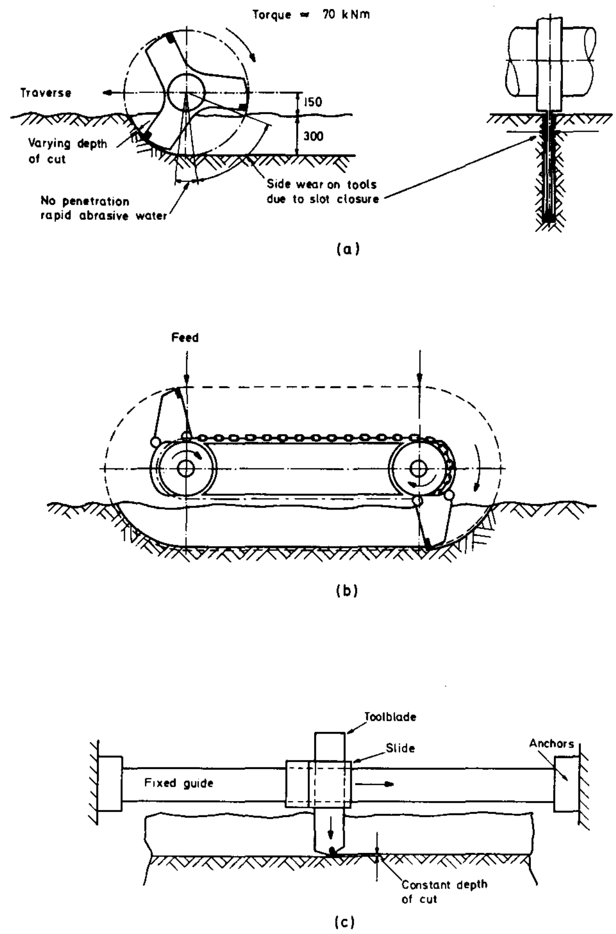


Fig. 1—Alternative methods of applying a drag bit for cutting rock.
(a) Rotary cutter
(b) Combined rotary and straight line cutter
(c) Simple straight line cutter

A simpler configuration, based on the action of a shaping machine (Fig. 1c) was then considered. A machine of this type offers several important advantages over the previous type in ease of design and construction. Important among these is the ease with which reciprocating linear motion of adequate force can be generated using a screw, a hydraulic cylinder, or a rack and pinion. The problem of guiding the drag bits to follow a straight line can be solved easily by mounting them on a simple slide or saddle guided on a rail or bed. The linear motion produces the constant depth of cut found to be important in the preliminary laboratory experiments.

There are three main objections to this simplified machine. Firstly, the drag bit requires a free face from which to commence cutting; a free face is also required at the end of the slot to prevent the bit from bumping repeatedly against solid rock at the end of each stroke. Secondly, the reciprocating motion entails a time consuming return stroke between cutting strokes. Finally, in its return down the slot the drag bit is exposed to excessive wear to its sides and to the danger of damage from jamming against fragments of rock in the slot, unless the additional complication of completely withdrawing the tool from the slot on the return stroke is added.

In spite of these disadvantages, the shaping machine type of cutter was chosen because of its simplicity and the constant depth of cut which it gives. This choice was influenced by the need to establish the practicability of a rockcutting machine using tungsten carbide drag bits to cut a pair of parallel linear slots in the rock face of a gold mine with the least possible delay and expense.

DESIGN OF FIRST PROTOTYPE

The layout of the first prototype rockcutting machine is shown in Fig. 2. The reciprocating head consisted of a saddle or slide A on which were mounted the cutting tools B, the manual tool feed mechanism and the clapper mechanism to relieve the forces on the tool during the return stroke C. The saddle fitted over, and was guided by, the bed D which was of circular section. A web E was welded to the bed to stiffen it in a transverse direction and to restrain the saddle from rotating. It also provided mounting points for the three hydraulic mine props F to which the machine would be clamped in the stope.

The cutting force was generated by a double-acting hydraulic cylinder, with a stroke of 3 m, housed within the bed. Hydraulic pressure was chosen as the means for generating the cutting force because of the ease with which a light and simple cylinder can provide large forces and the convenience of generating hydraulic power in a remote power pack connected to the machine by flexible hoses. The cylinder was anchored by a nut and threaded spigot to a diaphragm G which was welded into the end of the bed. The piston rod was connected by a hemispherical joint H to the saddle through the split cylindrical column I. Hydraulic fluid from the remote high pressure, electrically powered pump unit was piped to the cylinder through flexible hoses entering the web at J after passing through a control unit K, which incorporated a control valve, a pressure gauge, and a pressure relief valve.

A disadvantage of the hydraulic cylinder compared with the alternatives of a screw or rack and pinion type linear actuator is the long reciprocating piston rod and connecting column. It is necessary to provide a clearance in the stope, at the one end of the machine, equal to the working stroke of the piston. A machine using a hydraulic cylinder coupled direct in this manner has an overall length varying between a minimum of its stroke plus the saddle length, and a maximum of twice the stroke plus the saddle length. Using other types of actuator it is possible to design a machine of constant length equal to one stroke plus the saddle length. To these lengths must be added a few centimetres of dead length for items such as connections and attachments. In practice, difficulties due to the reciprocating piston rod and connecting column have been minor.

Because of the large forces imposed on the machine during cutting, anchoring of the machine in the cutting position is of primary importance. Two methods of anchoring the machine were considered:

(i) The machine could be anchored to bolts and hooks set into holes pre-drilled into the rock

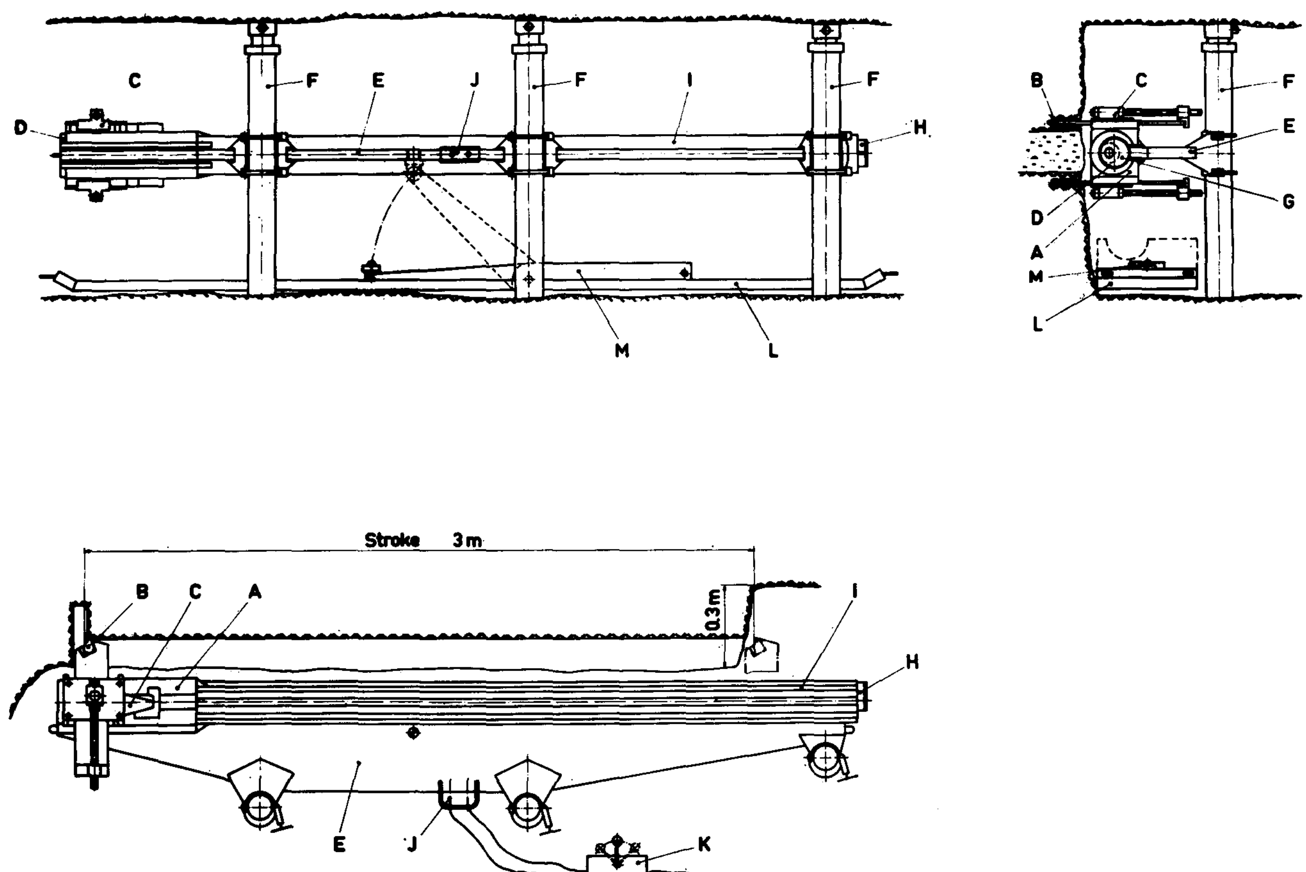


Fig. 2—Elevation, section and plan of the first prototype rockcutting machine, installed at a stope face. A is the saddle with tools B and tool feed C. D and E are the bed and web of the frame mounted on hydraulic props F.

footwall, hangingwall or stope face. The problems of accurate drilling for this purpose made this idea very unattractive.

(ii) Anchoring the machine between the hanging- and footwall by friction forces generated by jacks. This method was considered most likely to be successful. Tests have shown that a prop set at about 200 kN is capable of sustaining a transverse force of about 40 kN at either or both ends. As props required no pre-drilled holes and could be re-used indefinitely, it was decided that the machine should be anchored in the stope by clamping it to three 200 kN hydraulic props which would form an abutment capable of restraining a total reaction force of about 120 kN in a plane substantially parallel to that of the reef.

The machine was designed to rest on removable cradles on a sled L, so that it could be moved in the stope. To the sled was fixed a lever-type hydraulic jack M for lifting the machine into the cutting position.

Handling heavy equipment in the narrow confines of a stope is usually difficult and the first machine was built to be light so that final positioning could be done manually. The procedure for moving in the stope was to be as follows: With the machine resting on removable cradles of the sled and the collapsed props lightly clamped with their lower ends clear of the footwall, the sled was to be pulled parallel to the face by means of a chain block fastened to one of the pulling eyes and anchored to a prop some distance up the stope near the face. When the machine was adjacent to the section of face which it was intended to cut, the sled was to be manhandled close up to the face with the help of pinch bars.

To set the machine up in its cutting position, first the prop clamps were to be slackened off allowing the props to rest on the footwall, though still in an upright position guided by the clamps. Next, the machine would be raised clear off the cradles which would then be removed. The lifting socket on the machine was located at the centre of gravity so that the machine could be positioned manually while the height was controlled by the elevating jack. At this stage it was intended that the hydraulic props be set and clamped tightly one at a time, securing the machine in position. Lowering of the jack arm would then allow the cutting operaton to commence. If the clearance remaining between the collapsed jack and the machine were insufficient for the reciprocating saddle to clear it would be necessary to remove the sled. But it was preferred that the sled remain in position beneath the machine during cutting. After cutting of the slots had been completed, the weight of the machine was to be taken by the jack, allowing the props to be released, after which the cradles would be replaced on the sled and the machine would be lowered into them before moving to the next position.

DEVELOPMENT OF THE ROCKCUTTING MACHINE

The first machine was intended mainly to test the feasibility of cutting rock underground. Nevertheless, it was designed to be the basis of a mining machine. Critical to the success of this new method of mining were the answers to a number of important questions:

(i) Would an economic cutting tool life be attainable?

(ii) Would it be possible to advance the face and maintain the desired stoping width?

(iii) Would the handling of a machine of this size and weight in the stope prove too difficult?

(iv) Would it be possible to achieve an economic mining rate?

The success of the first prototype was such that, for the initial development, the basic machine concept was retained and design changes were made only where necessary to facilitate continued testing. At an early stage it became apparent that the first basic concept could, in fact, form the basis of a workable mining machine and that there was no need to deviate from this concept, though there was no shortage of ideas for alternative machine designs or for alternative methods of mining. The development process was followed through step by step, the worst defects remaining at each stage being eliminated.

To hasten the ultimate manufacture of production prototypes, the Chamber of Mines decided to involve manufacturing companies in the development of machines at the earliest possible stages, so that they could derive the benefit of experience in testing these machines right from the start. Three companies, namely, Anderson Mavor (South Africa) (Pty) Limited, Delfos and Atlas Copco (Pty) Limited, and Klöckner-Ferromatik (S.A.) (Pty) Limited, have collaborated actively with the Chamber of Mines in the development of machines, and two others, Boart and Hard Metal Products S.A., Limited and Sandvik (Pty) Limited, have collaborated in the development of cutting tools.

A brief history of the machines which were built may assist the reader to keep the developments referred to in perspective.

The first machine described above was designed and built by the Chamber of Mines and was tested underground at Luipaardsvlei Estate and Gold Mining Company, Limited.

Thereafter, four machines were built by two of the companies to the order of the Chamber of Mines. The machines were designed in collaboration with the Chamber of Mines, and the most significant changes were in the shape of the bed, the tool feed mechanism, the provision of increased cutting force, and the addition of a fourth prop. One machine built by Anderson Mavor was installed at West Driefontein Gold Mining Company, Limited, but testing was curtailed by the flooding of that mine. The other three machines were built by Delfos and Atlas Copco. These were used at Doornfontein Gold Mining Company, Limited and underwent many modifications in the course of testing. Mining with these machines showed that it was feasible to obtain a satisfactory tool life and that the face could be advanced over large distances.

Early in 1969, another machine was designed and built by the Chamber of Mines. This machine, shown in Fig. 3, incorporated significant changes in the design of the tool holding arrangement, the hydraulic drive control, and the anchoring and moving of the machine. Klöckner-Ferromatik collaborated in the design of the new props while the saddle and tool feed mechanism from a Delfos and Atlas Copco machine was used. Experience with this prototype showed that it could be handled with much greater ease and speed in a stope despite its greater weight, and that a satisfactory rate of mining could be achieved. In the meantime, continued improvements in cutting tool design and performance were being made.

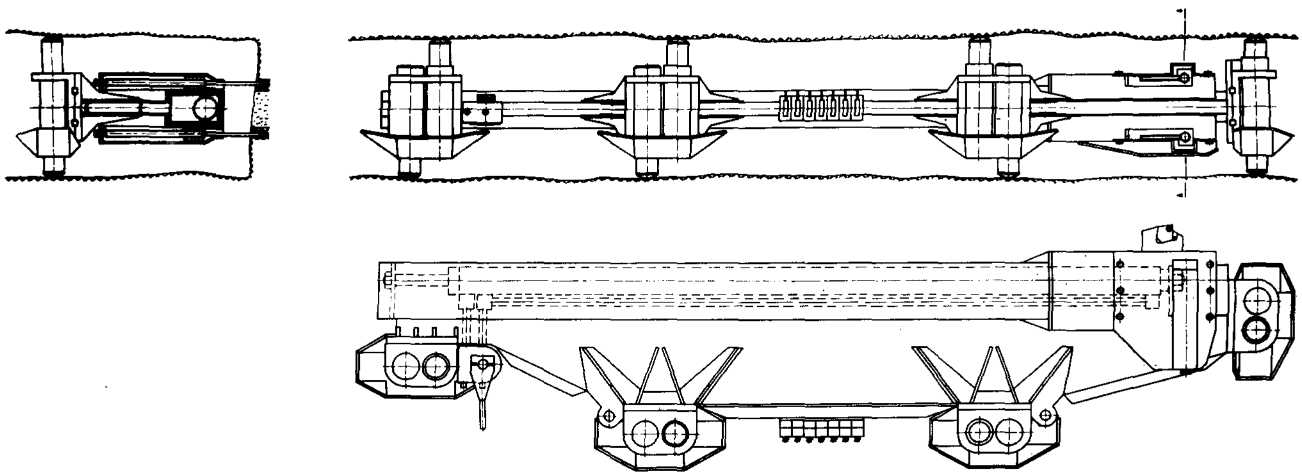


Fig. 3—Section, elevation and plan of the second prototype rockcutting machine, showing the offset twin anchoring props and fixed sleds for moving.

The Chamber of Mines then entered into development contracts with Anderson Mavor and Klöckner-Ferromatik for the design and construction of a series of six machines from each of them, suitably spaced in time to allow the incorporation of improvements, as indicated by the underground trials being carried out at Doornfontein Gold Mining Co., Ltd. The emphasis in this series of machines has been on improving machine reliability. The first of the machines delivered under these contracts is shown in Fig. 11.

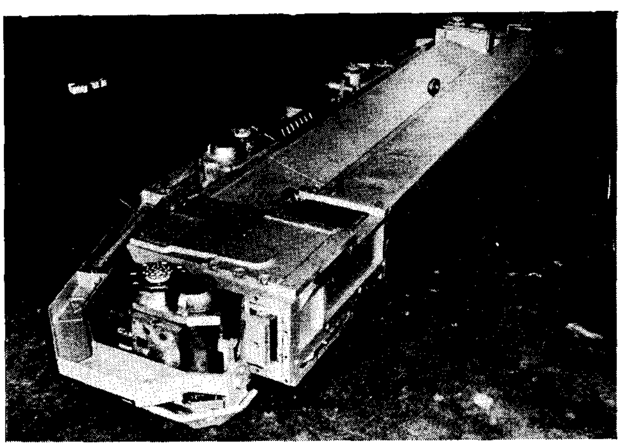


Fig. 11—One of the first machines supplied by Anderson Mavor.

The initial design and subsequent development of the many parts or sub-assemblies of the machines will now be described in detail.

Cutting tool bit

The tool bit (Fig. 4a) as designed originally, was similar to those used in metal and coal cutting. It consisted of a nearly rectangular tungsten carbide bit brazed into a notch in a hard steel body. The bit was made wider than the tool body and blade so that they could be advanced into the slot without becoming pinched between the closing edges of the deepening slot. The front and side faces were given clearance angles of 5° and 3° respectively and the leading face had zero rake angle, that is, it was perpendicular to the direction of tool travel.

Many types of brazing material were investigated and copper-based materials containing a high proportion of copper proved most successful. An important factor in selecting braze materials is that the brazing temperatures must be sufficiently high to cause hardening of the steel of the tool body near the bit which must be hard enough not to yield when cutting. The high melting

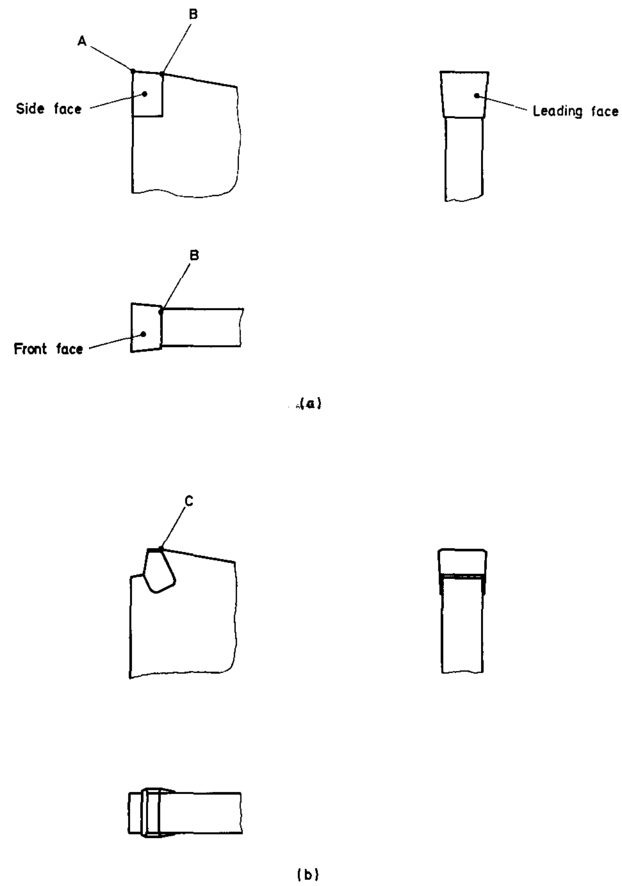


Fig. 4—Plan, front and end views of cutting tool bodies.
(a) Original drag bit shape.
(b) Latest drag bit shape showing reduced front face width, bevelled corners and improved method of inserting bit into holder.

temperature of 1 000°C to 1 100°C of the copper-based brazing alloys results in hardening of the EN30B steel body near the bit to RHC50.

The cutting action in hard, brittle rocks is very different from that in metal or coal. The front face bears on the rock and is subjected to heavy wear while the leading face of the bit virtually never comes into contact with the rock. As a result a flat surface, almost parallel to the base of the slot forms on the front face and increases in area as wear progresses. The area of this worn surface is a measure of the bluntness of the tool. Experience obtained in cutting underground has shown that the corner at A, Fig. 4a, remains rectangular. With tungsten carbide bits containing less than about 9 per cent cobalt thin flakes tended to break away from the leading face, whereas when the bits contained more than 9 per cent cobalt, plastic flow of the tungsten carbide occurred in the forward cutting direction so that a small projecting lip was formed on the leading face at A.

Of the grades of tungsten carbide used the types containing about 9 per cent cobalt, as used in percussive drilling, proved best. Grades containing less cobalt tended to fracture and grades containing more cobalt wore rapidly. Even with the optimum grades, fracture tended to develop at B when the flat worn surface on the front spread towards B. This fracture pattern was possibly due to the trailing part of the edge of the bit at B coming under load and, since this edge was relatively poorly supported by the steel, flaking away as the tool was dragged over the rock.

In the initial design of tool, the most common type of failure was over the brazed joint where the entire bit broke away from the body. There were two major causes of this type of failure: (i) the brazed joint was put into tension during the return stroke when the bit frequently snagged on irregularities in the slot or when small chips of rock wedged between the bit and the base of the slot. (ii) Because of irregularities in the rock and especially inclined rock faces at the commencement of cutting, side loads developed on the bit and, under extreme conditions, pushed it out by shearing the brazing material.

Many difficulties were overcome in a series of developments which resulted in the design shown in Fig. 4b which incorporates all the successful changes. Essentially the bit has been put into a socket thus improving resistance to side loads and to damage occurring during the return stroke. The leading face has a negative rake of 10° which reduces the plastic flow and the fracturing of this face. The long axis of the bit is approximately in the direction of the resultant of the forces acting on it. The trailing edge, C, of the front face is well supported, so that the bit itself is more resistant to fracture. The tool, in effect, remains sharp for longer since the front face is smaller and becomes equal to the area of the front face of the initial design only after considerable wear has taken place. Sharp edges have been removed from the front face to minimize stress concentrations in the bit.

Blade and tool body

Tool performance was unpredictable and it was decided to provide means for fitting and replacing cutting tools easily in the stope. Clamping and wedging devices used to hold the tool body in position in the blade had, like the blade itself, to be narrower than the tungsten carbide bit by at least 6 mm, which was the closure expected when a single slot was cut to a depth of 0.3 m.

A 12.5 mm thick by 150 mm wide blade was originally considered sufficiently strong to carry the cutting forces generated in a tool bit 18 mm wide, but experience with the prototype indicated that the cutting forces acting on the bit were about twice those expected by linear scaling of the forces measured in the laboratory experiments. These greater forces made it necessary to increase the dimensions of the blade to 14.5 mm by 250 mm for the machines which were built by Delfos and Atlas Copco. At this stage, the blade material was changed from EN9 to 'Abrasalloy', a proprietary high-strength abrasion-resistant steel. The early trials had proved that, under conditions of high vertical stress, slabs of rock separate from the face during cutting and the effective slot depth remains small so that this thicker blade would not be pinched by closure in deep slots.

The original tool body shown in Fig. 5a was retained in the blade by a wedge drawn in by an Allen screw. The large penetrating forces tended to rotate the tool body forward, generating a high reaction force on the finger retaining the wedge; this resulted in distortion of the finger and loosening of the tool body in the blade. When this weakness became apparent, the design in Fig. 5b was adopted. The acute angle between the two seating faces of the tool body ensured that the tool body was seated securely in the blade by the cutting forces, and the forces on the finger retaining the wedge were relieved considerably.

In all the designs, the tool body was seated in the blade on V-shaped edges so as to resist forces in a direction transverse to the plane of the blade. These

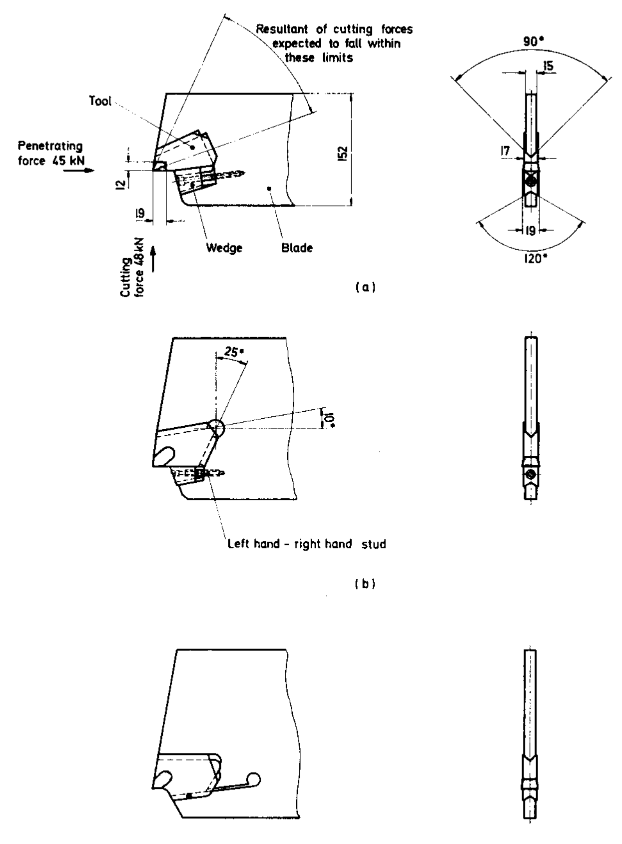


Fig. 5—Methods of fastening tool body into blade:
(a) Original design.
(b) Improved design with reduced loading on wedge.
(c) Present wedgeless design with resilient retaining finger.

transverse forces were caused by uneven rock surfaces and sometimes were responsible for a high proportion of tool and blade failures.

Retaining the wedge by means of an Allen screw was not satisfactory, as the hexagon socket was often damaged or became packed with dirt, making its tightening or removal difficult. The use of double-ended, left and right-hand threaded studs resulted in only a slight improvement (Fig. 5b). Finally, the screw was eliminated by providing a resilient finger and drawing the wedge into place where it was held by a soft transverse rivet. However, this modification required higher standards of manufacturing accuracy. Also, the flat contact face between the wedge and the tool body allowed transverse movement between these parts.

A design without a wedge was the next logical step (Fig. 5c). To reduce the degree of accuracy required in manufacture, the resilient finger was made more flexible by increasing its length. The wedge angle was calculated to give an adequate pre-load when the tool body was driven into the blade. The tool body was secured in position by means of a soft rivet through the V-shaped joint on the resilient finger.

The wedgeless design is still in use but may be modified to provide greater strength against transverse forces. A tongue-and-groove seating would provide greater resistance to bending than does the V-shape.

Tool feed mechanism

The tool is fed into the slot manually by means of a screw. Initially, the screw engaged with a nut fastened to the back end of the blade. The blade was mounted in a clapper or pivoted member which was free to rotate by 5° to provide automatic withdrawal of the tool from the base of the slot of about 5 mm during the return stroke (Fig. 6a). The penetrating force on the tool bit, acting through the blade, was taken in tension by the screw. The screw was anchored at its other end to a swivel, co-axial with and above the clapper pivot, which allowed the screw to swing with the clapper. The cutting force on the tool bit was balanced by reactions at the clapper pivot and the clapper stop.

The high penetrating force encountered in the underground trials resulted in severe bending of the blade and screw. This was due mainly to the eccentricity between the axis of the screw and the plane of the blade. Also, the tool did not withdraw sufficiently on the return stroke, especially in the early stages of cutting a slot when the radius of rotation of the bit was smallest. Furthermore, this mechanism was completely exposed and quickly became covered with abrasive grit.

The first modifications to the tool feed mechanism were made in collaboration with Delfos and Atlas Copco (Fig. 6b). The thicker and wider blade was housed in a robust clapper and the whole mechanism was protected by a cover plate. The feed screw was positioned in the plane of the blade and engaged with a rack machined into the side of the blade. The screw was anchored to the clapper and took the penetrating force in compression, transferring it ultimately to the clapper pivot. The other reactions were taken up by the machine frame at A and by the clapper pivot. This mechanism worked better than the previous one but the components deteriorated rapidly, and cutting the rack on the blades was an expensive operation. The cutting force caused high engagement loads between the blade rack and the screw, which increased as the blade was fed into the slot, with the result that the screw became indented. Rock fragments sometimes lodged between the stop at A and

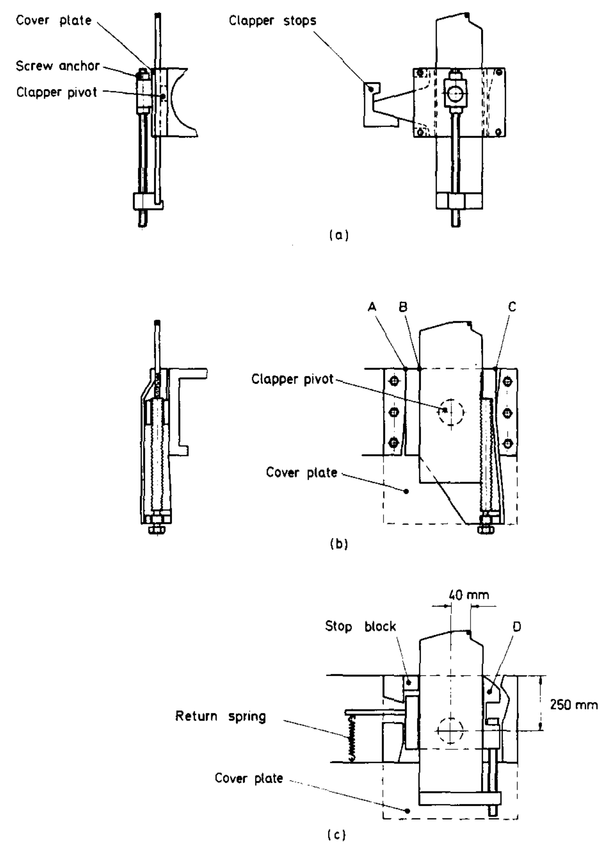


Fig. 6—Clapper layout.

(a) Original design with offset feed screw and narrow blade.
(b) Improved design with feed screw in the plane of the blade.
(c) Present design with simplified blade.

the clapper box with the result that the large unsupported reaction at B permanently deformed the clapper. When rock fragments lodged in front of the leading edge of the clapper at C, they prevented rotation of the clapper and tool withdrawal, resulting in tool damage during the return stroke.

An optimum position for the clapper pivot was determined after much experimentation. The withdrawal of the tool bit during the return stroke is determined by the position of the pivot and the angle of rotation of the clapper. A withdrawal of approximately 13 mm can be obtained with a rotation of 10°, when the pivot is 250 mm from the front face of the machine and 40 mm behind the bit, as shown in Fig. 6c. The withdrawal can be increased by increasing the 40 mm separating the bit and the pivot centre line. However, the penetrating force acting on the bit can then cause the clapper to rotate and the tool to withdraw on the forward stroke. This was found to leave a lump at the base of the slot, which invariably caused damage to the bit on the subsequent return stroke. Experience has shown that, with the present position of the pivot, the tool never withdraws on the forward stroke and a withdrawal of 13 mm on the return stroke is adequate.

Fully automatic feed mechanisms have been considered, but very robust and elaborate arrangements would be required to make them foolproof. Furthermore, due to variations in rock stress and hardness, the rock cuttability on a stope face varies from place to place and a continuously variable feed increment is desirable. Consequently, further developments have been continued with the clapper and normal screw feed mechanism.

The latest arrangement, developed in collaboration with Anderson Mavor and Klöckner-Ferromatik, is illustrated in Fig. 6c. A return spring has been added to act on the clapper to ensure that the tool bit is not in the withdrawn position at the commencement of the cutting stroke. Replacement of damaged tools is sometimes difficult at the stope face, and the mechanism has been designed so that the blade can be removed quickly and replaced by a spare blade fitted with a new tool. The assembly of the lower feed mechanism was difficult because its weight had to be supported in the stope. The components have now been arranged so that each is held in position separately and the lower mechanism can be assembled one part at a time. The screw engages with the blade, as in the original design, through a nut, but this is now arranged so that the screw lies in the plane of the blade. The nut is allowed a slight degree of rotation inside a block which is fixed to the back of the blade, and the screw pulls on the clapper through a spherical washer. These arrangements eliminate the danger of bending the screw. A lip, D, has been added to the clapper to prevent the ingress of rock fragments during the cutting stroke. The main reaction to the cutting force is taken through the blade by a replaceable hardened stop block. The other reaction is taken by the clapper at a block which is also subjected to the direct pull of the screw. The design of this block, as part of the clapper, has been the main subject of the latest modifications.

Bed and saddle

During slotting the cutting tools are guided in their linear path by the bed along which the saddle reciprocates.

The bed on the prototype consisted of an unmachined 165 mm diameter circular steel tube of 6 mm wall thickness. To stiffen this tube, a 50 mm thick web fabricated from 6 mm steel plate was welded to it (Fig. 7a). The saddle was made from thickwalled mechanical tubing. For high stiffness, consistent with lightness, it was bored eccentrically before slitting. Two flat plates were gussetted on either side of the slot to enclose a portion of the web. Brass bearing material was fused directly onto the internal surfaces of this saddle, over an axial distance of 15 cm at each end. The axial separation between the two end bearings was formed to provide and maintain a moment arm and allow the saddle to follow the uneven, unmachined bed. To reduce the weight of the saddle further, the tube was bored out between the bearings, concentric with the outer surface.

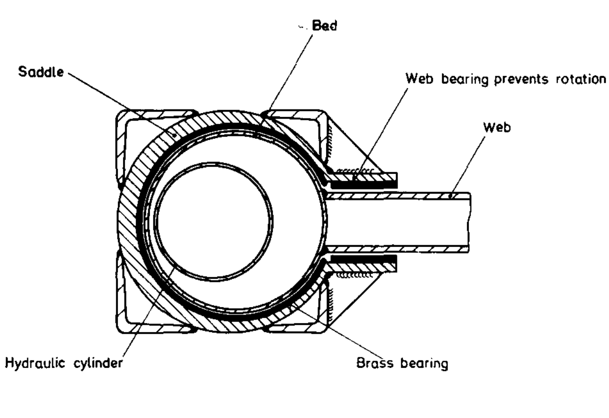


Fig. 7—Bed and saddle bearings.
(a) Prototype, unmachined bed.

To facilitate the mounting of the tool feed mechanism the outer section of the saddle was made square by welding on 50 $\times$ 50 $\times$ 8 mm steel angles.

Tool guidance with this arrangement was inadequate because of large clearances and forces greater than had been expected. To improve tool guidance the bed on the first Delfos and Atlas Copco machine was of rectangular section and was machined on the bearing surfaces (Fig. 7 b). This bed consisted of a hollow rectangular steel section 250 $\times$ 154 $\times$ 9.5 mm thick to which was welded a 62 mm fabricated web. The improved tool guidance obtained with this beam was satisfactory and the rectangular form remained basically unaltered until recently.

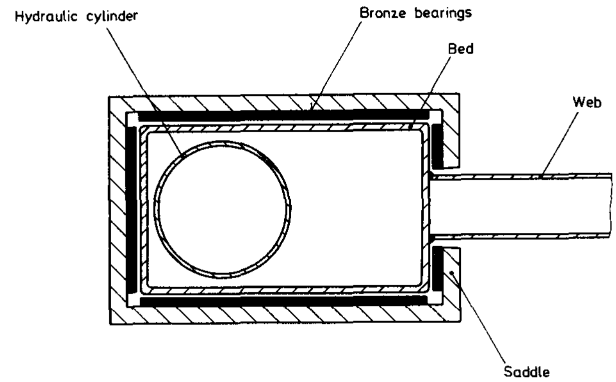


Fig. 7(b) Improved machine with rectangular machined bed.

Because of its open C-shape, the saddle is quite flexible, particularly when transverse forces act on the tool tips and apply torsion about the axis of the bed. The rectangular saddles of the first three Atlas Copco machines were fabricated from 13 mm plates and stiffened by three encircling ribs made by welding on 38 $\times$ 76 mm rectangular bar. This appeared adequate at the time because the overall lack of rigidity of the machine was due mostly to the method of supporting the machine in the stope on standard mine props. However, with the introduction of the fixed twin-props, which gave much improved support to the frame, the excessive flexibility of the saddle became evident. Furthermore, no allowance had been made for taking up wear of the phosphor bronze liners between the saddle and the bed (Fig. 7b). As these wore a considerable amount of slack developed and the direction of the cutting tools became controlled by the rock rather than by the machine.

In the current series of machines built under contract by Anderson Mavor and Klöckner-Ferromatik the saddle has been stiffened considerably by fabricating it from 50 mm thick steel plate or by using a casting of equivalent section. Means for taking up liner wear have been provided as follows: Wear in the horizontal direction is taken by means of screws and locknuts pushing the two narrow liners at the back of the bed (Fig. 7c). These screws are in a relatively protected position. A different arrangement had to be used for taking up wear in the vertical direction because any screws extending above or below the saddle would be damaged very quickly during operation and the working loads on the bearings would be concentrated on the adjusting screw ends which would be overstressed. A system of wedges pulled together by means of screws in the body of the saddle was used instead.

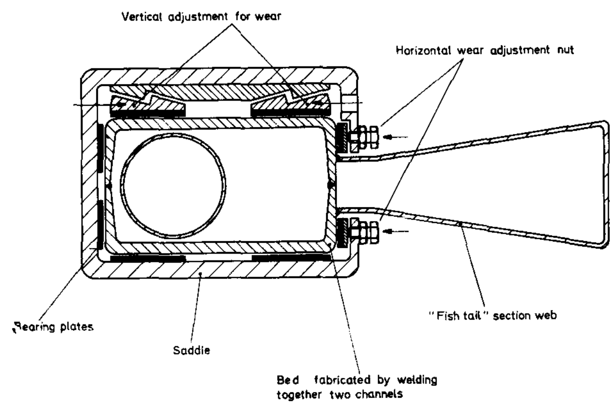


Fig. 7(c) Later version of (b) with adjustable bearing clearance.

The wedges operate quite well but the screws at the back of the beam are not easily accessible. Also, the cost of manufacture and internal machining of the saddle is high. New methods of achieving adequate rigidity of the saddle and of taking up linear wear were therefore sought.

The solution proposed by Klöckner-Ferromatik (Fig. 7d) and incorporated in their latest machine is promising. The bearing surfaces are arranged in the form of three V's and adjustment for wear is effected by pushing the pair of V's at the back of the bed. This is done by means of wedges, as is the vertical adjustment shown in Fig. 7c, except that these wedges are pushed in a direction parallel to the axis of the bed by screws which are readily accessible at the open ends of the saddle. It is intended in the next machine to replace this manual-mechanical method of adjustment by an automatic method.

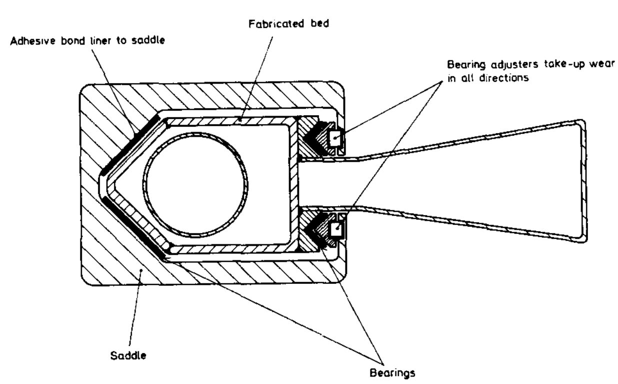


Fig. 7(d) Latest type with simplified bearing adjustment.

If an automatic method of taking up wear proves reliable, it will be possible to rely on the re-entrant V's at the back of the bed to increase the stiffness of the saddle, allowing its section to be reduced. Meanwhile the manufacture of the saddle had been simplified by using two heavy forged steel channel sections which are machined on their internal surfaces before being welded together along their two edges. After stress-relieving, one of the welds is cut away to form the opening of the C-shape. No further machining is required on the internal surfaces.

Frame structure

The machine frame, of which the bed forms a major part, transmits the reactions due to the cutting forces to the supporting hydraulic props. The frame should be very rigid so that the bits are deflected as little as possible from their intended linear path during cutting.

The props should be disposed so that support is provided as close as possible to the cutting tools. It is necessary, however, to allow a clearance of 0.7 m for the blades in their fully withdrawn position and for the feed screws which must pass between the rock face and the supporting props. Fortunately the end supports can be placed much closer to the face as they are beyond the travel of the saddle. The disposition of the props out of a straight line increases the rigidity of the frame. It also provides stability during positioning, as the centre of gravity is within the boundary of the base formed by the four downward pointing props on which the machine is raised.

The weakest link in the transmission of forces from bed to props is the relatively thin straight line junction between the bed and the web. This is unavoidable because of the need to accommodate the bearing strips in the lips on the C-shaped saddle. When the twin-prop system was introduced it became necessary to reinforce the web because the eccentricity of the twin-props introduced large couples when the machine was set in position in the stope. The web was then fabricated from parallel flat plates separated by a zig-zag spacer while the gussets supporting the prop mounting plates were external to the web (Fig. 7b). In all subsequent machines a fish-tail section has been used (Fig. 7c, d) in which all bracing and gussetting is internal. In addition to providing greater strength and rigidity this arrangement provides an area in which hydraulic tubing can be protected. However, the clearance between the web and the reciprocating blades and feed-screws has been reduced. This may cause difficulties when fallen pieces of rock accumulate in the trough between the bed and the upper slope of the web.

Anchoring and moving of the rockcutter

Moving the prototype rockcutter on its sled parallel to the face with a chain block proved to be surprisingly easy, but moving it towards the face was difficult. To facilitate positioning of the machine close to the face the sled was modified to permit the jack to slide transversely on the sled after removal of a holding pin. This rendered the operation dangerous because, after lifting the machine off the sled and removing the pin holding the jack, it was very difficult to control the movement of the machine on top of the sliding jack.

The original concept of the setting-up procedure for the machine was not successful as the balance of the machine, while it was being lifted at its centre of gravity, was easily disturbed by the weight of a prop. It was intended that the props should slide easily in their clamps so that their weight would not affect the balance, but this could not be achieved. To overcome this difficulty, the props had to be removed completely from the clamps before the machine was set up.

To improve control of the machine, the prop clamp at the tail end was modified so that it would swivel. The setting-up procedure was then to lift the machine, position the tail end by hand and insert and set the tail end prop. Since the tail end prop clamp could swivel, it was possible to manoeuvre the other end of the machine while the tail end was held in position. The worst feature of this modification was that the swivel at the clamp was a mechanically weak point.

The use of ordinary props for anchoring the machine resulted in much wasted labour since the props had to be put in and taken out of the clamps every time the machine was set in a new position. On occasion, when the machine was set up near the hangingwall the props had to be inverted since the clamps could grip only on the external cylinder of the props. It was necessary to move out the sled and jack from under the machine while it was cutting because rock cut from the face would fall onto the sled and could not be removed easily. From the foregoing it will be evident that the process of moving the machine from one cutting position to the next was both lengthy and arduous. In practice the operation was seldom achieved in less than an hour.

This system of anchoring proved to be too flexible because the props were not designed to withstand side loads and had excessively large bearing clearances. Also, there was some clearance between the clamps and the props which could never be set up exactly parallel to one another. On the prototype only three props were used and were arranged nearly in line. Consequently transverse forces on the tools caused large vertical movements of the machine. The introduction of a fourth prop, offset from the line of the other three, reduced the extent of this movement considerably.

When it was realized that the machine would have to be supported more rigidly and that moving it in the stope took more time than cutting, a complete re-design of the moving and anchoring system was undertaken.

The standard props were replaced by twin props keyed and bolted permanently to the machine. These twin props consisted of two parallel adjacent prop cylinders rigidly connected together, with one ram extending upwards and the other downwards. This eccentric twin prop arrangement made it possible to have large bearings in the props to carry side forces and still have a good expansion ratio. The closed height of the assembly is 0.48 m and the fully expanded height 0.99 m. The expansion can be increased when necessary by the addition of extension pieces which fit into female conical surfaces in the rams providing bending stiffness so that side forces can be transmitted through the extension pieces. When the extension pieces are not required they are replaced by short caps which fit into the same female conical recesses.

Separate controls are provided in a central control valve block for each of the eight rams in the four twin props. The lower rams are used for raising, lowering and tilting the machine. To ensure stability the props are disposed so that the centre of gravity of the machine falls well within the area enclosed by the lower rams. When the machine is in position the upper arms are expanded to anchor it. Each pair of cylinders in each twin prop is connected hydraulically through a non-return valve which makes it possible to set the load by extending the top ram without disturbing the position of the lower ram and, therefore, that of the machine. When pressure is applied to set the prop the non-return valve opens and the same pressure is applied automatically to both rams. The twin props are activated by an air powered hydraulic pump supplying a two per cent oil-in-water emulsion at a pressure of 250 bars.

A small skid plate is fixed under the body of each of the twin props and the machine rests on these plates when the lower rams are retracted. The machine is towed parallel to the face and positioned with two double acting hydraulic cylinders towards or away from the face. These two hydraulic cylinders are pin-jointed loosely to each end of the machine and are activated through the central control valve block. They react against props pre-set in the stope at suitable positions.

A very rugged wrap-around bumper guides the movement of the machine and protects the hydraulic controls and hoses. Standard hydraulic props are used in the stope for face support and are placed in a straight line about 1.5 m from the face. The machine is trapped between the props and the face, and the bumper allows it to slide against the props.

This arrangement of twin props, fixed skid plates, permanently attached positioning jacks and wrap-around bumper, has resulted in a machine of greatly improved mobility which can be moved and set up in less than ten minutes. The resultant saving of nearly one hour in the cutting cycle time has proved to be the most significant step towards the development of a reliable production rockcutting machine.

The development of the twin props, carried out in collaboration with Klöckner-Ferromatik, has progressed in a number of small stages. The bearings and seals in the props have been improved. The two cylinders have been combined into a single forged rectangular block and all hydraulic conduits and connections are drilled into this block. The number of hose connections has been reduced to a minimum. They are of a patented type consisting of a spud with an O-ring and a simple connecting staple.

Hydraulic drive

The hydraulic cylinder used to drive the saddle in the prototype machine consisted of a cylinder of 76 mm dia, 3 m long, with a 38 mm dia piston rod. It was originally operated at a maximum pressure of 200 bars giving a maximum cutting force of 68 kN. The slender piston rod had to be protected against buckling in case the saddle struck an obstruction during the return stroke. A by-pass valve was therefore incorporated into the piston to limit the pressure difference between the two sides of the piston during that stroke (Fig. 8a). This valve was also used to unload the hydraulic system at the end of the cutting stroke when the valve was opened by a probe, causing the oil to flow past the piston.

Very early in the testing programme it became obvious that the maximum cutting force would have to be increased. The cylinder was reinforced by winding it with wire under tension and the maximum pressure was increased to 250 bars. However, the piston rod could not be strengthened and its factor of safety was reduced to a low value.

All machines built after the prototype have had cylinders of 110 mm dia with 48 mm piston rods. A maximum operating pressure of 170 bars provides a maximum cutting force of 130 kN. The by-pass valve in the piston has been dispensed with and hydraulic cushions have been introduced to obviate impact at the end of each stroke. The danger of buckling the piston rods was reduced by the adoption of a lower operating pressure but protection against this danger was later achieved by the introduction of a fast return stroke device which reduces the maximum force automatically during that stroke.

In the earliest machines a simple hydraulic circuit with a separate control unit was used which was connected to the machine and to the power pack by flexible hoses provided with self sealing, snap-on quick couplers. The control unit consisted of a lever operated, closed centre, two way control valve, a pressure relief valve set at the maximum working pressure and a pressure gauge, all mounted in a tubular steel protective frame (Fig. 8a).

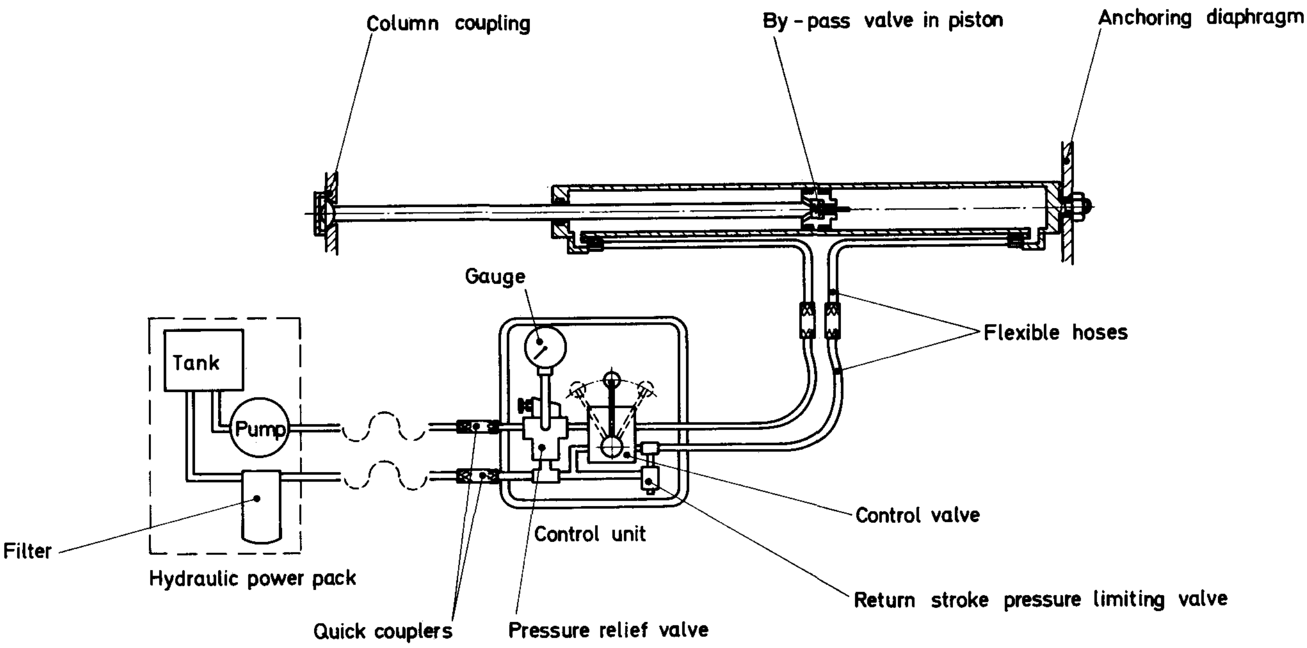


Fig. 8—Cutting cylinder hydraulic system.
(a) Prototype without rapid return stroke.

These early machines were slower on the return stroke than on the cutting stroke because fluid was pumped into the crown side of the cylinder for the return stroke. A quick return stroke was achieved by connecting both ends of the cylinder to the pump. The difference between the areas of the crown and annulus sides of the piston providing the return force. The supply from the pump had only to displace the piston rod, the remaining flow into the crown side coming from the annulus side of the piston. This system suffered from the resulting high rates of flow in the valves. The 68 litre/min supplied by the pump combined with 292 l/min from the annulus side of the cylinder to provide a total flow of 360 l/min into the crown side. Commercial high pressure valves are not made for such high rates of flow and excessive pressure loss and heating of the fluid occurred.

A special rotary valve with large ports was then designed at the Mining Research Laboratory which made it possible to obtain an efficient quick return motion by the method described above[6]. The operation of this valve is illustrated in Fig. 8b. The annular side of the cylinder is always connected to the pump and:

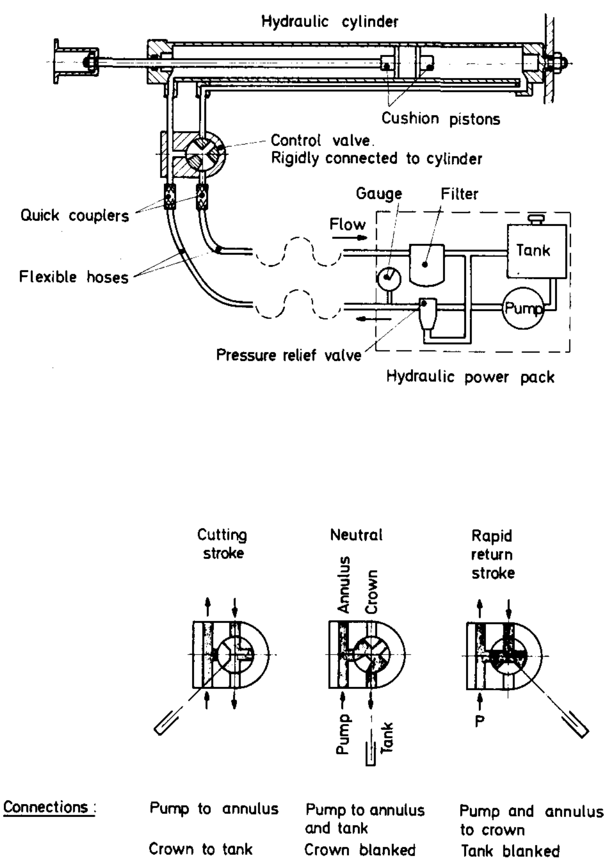
(i) for cutting, the crown side is connected to the tank

(ii) in neutral, the crown side of the cylinder is shut off and the pump is connected to the tank and fluid flows freely through the valve

(iii) for the return stroke, the crown side is connected to the pump and fluid is then transferred from the annulus to the crown side as the piston rod is displaced.

On the return stroke the speed is increased and the stalling force decreased by a factor of 4.3.

The rotary valve was mounted direct onto the gland end of the cylinder and was connected to it with large bore steel tubes without any restrictions. The



(b) Present self-contained design with rapid return stroke and simplified connection.

pressure relief valve, the pressure gauge and the filter are all placed on the power pack and hydraulic hoses from the power pack are taken direct to the machine. The cylinder and control valve are fitted to the machine and removed from it as a single unit. This arrangement has caused difficulties regarding the strength of the frame but from an operational point of view it has proved reliable over more than a year of underground testing.

The cylinder has normally been anchored to the bed by means of a threaded spigot at the closed end of the cylinder which passes through a diaphragm in the hollow bed and is secured by a nut. In machines supplied by Anderson Mavor a transverse pin passing through an eye in the end of the cylinder has been used. In future machines different means of securing the cylinder to the bed will possibly be used which will incorporate the hydraulic connections to the valve. These connections will be built into the frame with the tubing, rather than being in unit with the cylinder as in current machines. It will then be possible to place the valve in the most advantageous position from an operating point of view and without affecting the strength of the frame.

Drill jig

Initially the blade tended to become damaged at the end of the stroke because, as the slot became deeper, the finger in the front bumped against the end of the slot and took the impact and full cutting force generated by the cylinder. A 62 mm clearance hole is therefore drilled into the face before cutting each slot, so that the tool and blade reach the end of the stroke in the hole. With the first machine this hole was located by eye, but on subsequent machines a drill jig, detachable from the machine, was used to locate the hole accurately.

The first design of jig weighed 180 N and carried two drill rods for drilling a hole for each of the two slots. While this jig was relatively easy to instal in a workshop it was very cumbersome underground. It was re-designed to carry only one drill steel and had to be set up separately on top and then on the bottom of the machine frame. Setting up on the bottom side was awkward and time consuming.

The latest jig consists of a guide fixed permanently to the frame on the far side from the face and a small detachable guide which is fitted into a hole in the machine bed. The drill steel carries one small bush which fits into the detachable guide while the other end of the steel passes through a slot in the fixed guide. With this arrangement setting up is quick and easy.

Water sprays

Water sprays are needed for cooling the tools, cleaning the slots and for dust suppression, although very little dust is produced. Experience in tunnelling applications of drag bits has indicated that, in order to minimize thermal shock loading of the tungsten carbide, the water should not be sprayed directly onto the bits. Consequently, two water jets are used in line with each tool and are directed into the slot immediately in front of and behind the blade.

On the earlier machines, the water was carried in pipes to the nozzles projecting in front of the machine. These pipes and nozzles were very vulnerable to damage and on later machines the water was carried through internal ducts in the saddle to counter sunk nozzles which are removable.

FUTURE TRENDS

Underground tests at the Doornfontein Gold Mining Company, Limited have shown that the basic concept behind the rockcutting machines can result in a workable mining system. Much effort is being devoted to improving the mechanical reliability of the machine and ancillary equipment, and it is feasible that within the period of about a year satisfactory reliability will be achieved. However, the efficiency of mining would be greatly improved if the machines could work on a straight face[7]. In deep mines where there are high stresses on the faces the rock fracture pattern is such that it tends to straighten the faces and it is very difficult to maintain the benches from which a rockcutting machine can start its cutting. Machines with different configurations such as illustrated in Fig. 1b could effectively work a straight face. But it is possible to adapt a machine with the linear action to work a straight face if oblate holes, large enough to accommodate the blades at the beginning of each cut, could be made in the face. The addition of devices to the machine to generate these openings is feasible and is receiving attention.

Some difficulty has been experienced in cutting very hard rock and possibilities exist for supplementing the drag bit cutting action with other methods of cutting rock. It has been found that rapid heating of the rock in the path of the bit can cause sufficient damage to it to reduce the forces required to cut it by about a half[8]. Fortunately, thermal damage is most effective in those rocks which are hardest to cut. Also, the addition of a vibratory force might assist in the cutting of the harder rocks.

CONCLUSION

An experimental rockcutting machine was designed primarily to test the feasibility of cutting rock underground. It was also designed to form the basis of a machine for continuous, selective mining.

Extensive testing of similar machines has shown them to be capable of cutting rock underground and, after a series of developments, to be capable of initiating mining by this method. The initial concept was sufficiently workable and further developments did not depart radically from it. These developments have resulted in a machine which is highly maneouverable and able to cut rock effectively, but further minor developments are still necessary before the machine can be regarded as sufficiently reliable for use in production.

Two major developments are being considered which would improve the efficiency of rockcutting machines. The first is a modification to the present machines or a new design which would permit the machine to work on a straight face. The second is a hybrid design, incorporating heat or vibration, which would enable the machine to cut the hardest rocks.

ACKNOWLEDGEMENTS

The work described in this paper forms part of the activities of the Engineering and Simulation Divisions of the Mining Research Laboratory of the Chamber of Mines of South Africa.

The authors are indebted to many of their colleagues in the Laboratory, among the Manufacturers and on the Doornfontein Gold Mine for their help, but especially to Dr N. G. W. Cook and Mr J. H. Roux for their invaluable support and assistance.

REFERENCES

1. Cook, N. G. W. 'The design of underground excavations.' Eighth Rock Mechanics Symposium, University of Minnesota, 1966, in "Failure and breakage of rock," C. Fairhurst, Ed.

2. Hodgson, K. and Joughin, N. C. 'The relationship between energy release rate, damage and seismicity in deep mines.' Eighth Rock Mechanics Symposium, University of Minnesota, 1966, in "Failure and breakage of rock," C. Fairhurst, Ed.

3. Cook, N. G. W., Immelman, D. A. and Mrost, M. 'Economic factors affecting planning research and development for large South African gold mines.' Ninth Commonwealth Mining and Metallurgical Congress, London, May, 1969.

4. Cook, N. G. W., Joughin, N. C. and Wiebols, G. A. 'Rock cutting and its potentialities as a new method of mining.' *Jl. S.Afr. Inst. Min. Metall.*, 68 (10), 1969, 435.

5. Obert, L. and Stephenson, D. E. 'Stress conditions under which core discing occurs.' *Trans. Soc. Min. Engrs.*, 232, 1965, 227-34.

6. Pallister, G. F. and Cook, N. G. W. 'A note on a quick-return valve for an hydraulic actuator.' *S.Afr. Mech. Engr.*, *20* (7), 1970, 226-28.

7. Joughin, N. C., *et al.* 'Progress towards a method of mining gold reef selectively and continuously.' To be submitted to *Jl. S.Afr. Inst. Min. Metall.*

8. Wagner, H., *et al.* 'Laboratory experiments on cutting hard rock.' To be submitted to *Rock Mechanics.*

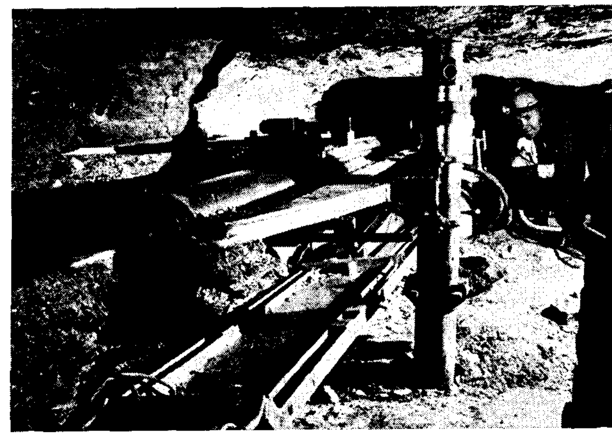


Fig. 9—Prototype machine operating in a stope.

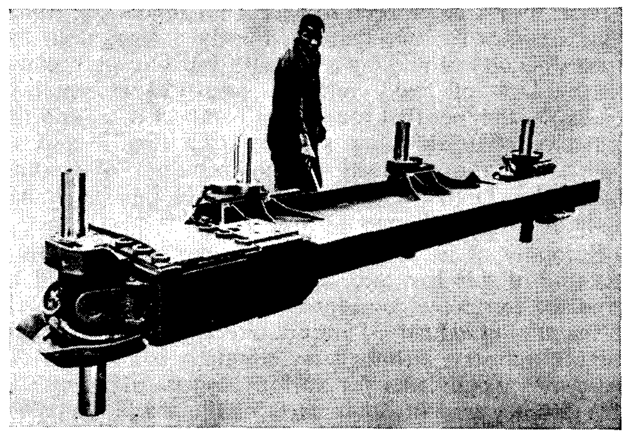


Fig. 10—Improved twin-prop machine.