The design of a stoping scraper scoop

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SYNOPSIS

Improvements in the drilling and blasting techniques used in the stopes of narrow reefs have caused the clearing of broken rock to become the major cause of delay in the achievement of a regular cyclic operation. This paper discusses the development of a scoop in accordance with a predetermined set of performance parameters.

The test results obtained to date have shown that the basic objectives have been achieved within tolerable limits of variation.

SAMEVATTING

In afbouplekke met 'n nou rifband het verbeterde afboutegnieke veroorsaak dat die verwydering van gebreekte rots die belangrikste faktor geword wat verhinder dat 'n gereelde daaglikse siklus in die afbouproses behaal word. Hierdie artikel bespreek 'n metode wat gevolg is om 'n skraper te ontwikkel wat voldoen aan 'n stel voorafbepaalde prestasieveranderlikes.

Die resultate tans beskikbaar toon dat die gedefineerde doelwitte binne aanvaarbare afwykings bereik is.

Introduction

This paper describes the design and development of a scraper scoop that could improve the stope-cleaning activities in narrow reef stopes.

Improvements in drilling and blasting techniques, coupled with more effective blasting barricades and less labour-intensive support systems, have increased the average monthly stope-face advance by 80 per cent on Stilfontein Gold Mining Company Limited. One of the major factors inhibiting the attainment of a blast per panel per day on a cyclic basis is the ability to clear the broken rock in the time available. As the length of the stope strike gully becomes progressively greater than the length of the panel, an imbalance in scraping capacity begins to occur. Even though higher-capacity winches are used for the strike gully, scraping capacity is still the major cause of delay. For this reason, the clearing of broken rock from the strike gully by increasing the scoop capacity formed the initial objective of the work described in this paper. The aim was therefore to obtain a scraper scoop that, in balance with the available winch power and ancillary scraping equipment, would provide optimum cleaning characteristics. Secondary objectives were to minimize any maintenance work, and to balance the economic life of the scoop with the life of the stope panel.

All the commercially available scraper scoops were evaluated, and none was found to combine all the required properties in one unit. Because of this, it was decided to start at the beginning by designing a scoop that would embody the necessary characteristics.

The methodology described can be applied to various other mining methods with different design parameters, and it is not suggested that the scraper scoop described in the following pages is the ultimate design for all situations.

Mining Conditions

The following are the conditions under which the scraper scoop was required to operate:

Capacity of scraper winches	37 kW 55 kW
Maximum length of stope strike gully	120 m
Average panel length	20 m
Average stope width	1,30 m
Type of rock	Quartzitic
Average tons per blast	85 t
Size of stope strike gully	1,5 m wide by 3,00 m high.

Approach Adopted

At the start of any project, it is necessary to design a set of preference conditions that should be achieved in terms of clearly understood parameters. For this purpose, a set of objective parameters subdivided into Performance, Engineering, and Economics parameters were defined.

In addition, the following group of design parameters could be manipulated to attain the desired results: *Physical dimensions, Basic layout, Blade angle, Blade design, Mass, Material, Method of assembly, Wear attachments, and Attachment lugs.*

A matrix was constructed from the two groups of parameters (Table I) so that the interrelationship between the various factors could be evaluated.

The objective parameters are first discussed in detail, followed by a description of how the design parameters were incorporated to meet the defined requirements.

Performance Parameters

(a) *Fragmentation Handling*

Fragmentation handling is mostly affected by the basic shape and blade design of a scoop. The common configurations into which practically all scoops can be classified are shown in Figs. 1 to 3.

Of these three basic designs, the shovel type would handle broken rock best. Because of its totally exposed blade and the absence of any side plates, the shovel type of scoop is not sensitive to the size of rock in its path. In contrast to this, the box type can be considered to be a compromise. With the correct blade design on a hoe type of scoop, its rock-handling capability could closely approach that of a shovel type of scoop. The

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TABLE I

INTERRELATIONSHIP BETWEEN THE DESIGN AND THE OBJECTIVE PARAMETERS

Design parameters	Objective parameters														
	Fragmentation handling	Penetration	Rock retainment	Tipping action	Fouling action	Corner digging	Stability	Face digging	Capacity	Winch power	Ease of manufacture	Durability	Structural strength	Maintenance	Production cost
Physical dimensions					×	×	×		×	×					×
Basic layout	×	×	×	×	×	×	×	×	×			×	×	×	×
Blade angle	×	×	×	×			×	×		×					×
Blade design	×		×		×	×	×	×				×	×	×	×
Mass	×	×					×	×		×			×		×
Material												×	×	×	×
Method of assembly											×	×	×	×	×
Wear attachment												×	×	×	×
Attachment lugs							×	×			×				×

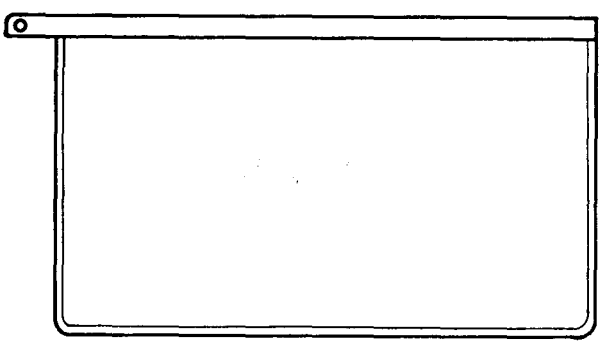

Fig. 1—Box type of blade

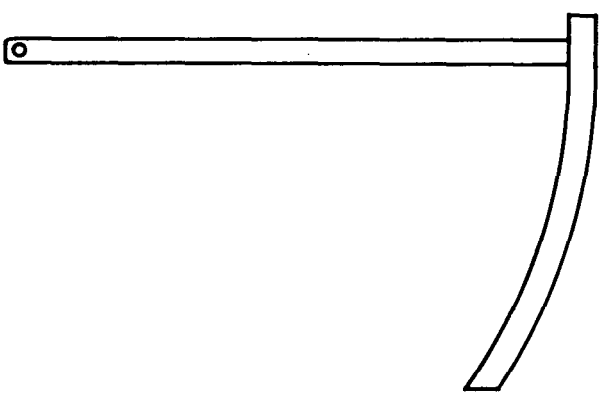

Fig. 2—Shovel type of blade

mass of rock that can be moved by a scoop increases as the mass of the scoop increases.

(b) *Penetration of Broken Rock*

The penetration of a scoop blade depends on the stress on the blade edge in contact with the medium. In the parallel-sided blade shown in Fig. 4, the resultant force R acting on the blade is a composite of the weight

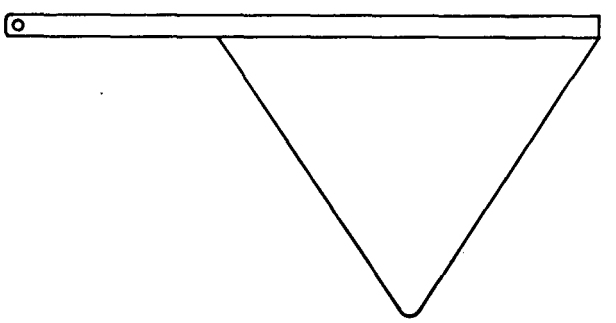

Fig. 3—Hoe type of blade

W and the thrust T of the winch rope. The blade will move in a direction that offers the least resistance to the resultant force R. No matter how unmeasurable, the resistant force to the blade could be represented by a polar diagram to indicate its magnitude and direction as shown in Fig. 5.

If this polar diagram of the resistant force applies to the parallel-sided blade, the force A may be in the same direction as, and of equal magnitude to, R, but B may be less than R's component in the direction of B. Thus, the blade will move in the direction of B. If R could point in the direction of B, the resultant penetrating force would be greatest.

The predicted polar diagram is an instantaneous representation of the resistance to the blade and varies with each infinite distance travelled by the blade. It is very difficult to predict the specific polar diagram for a specific medium that has to be penetrated, but, if the medium is homogeneous in structure and resists a similarly shaped obstacle equally well in all directions, the polar diagram of resistance would be mainly determined by the shape of the obstacle.

For the blade shown in Fig. 4, the direction of initial least resistance would be $\theta/2$ from the horizontal and,

when the point of the blade had penetrated, it would approach θ from the horizontal. The reason for this is that the vector representing the infinitesimal increase in projected area of the blade is initially smallest in the direction $\theta/2$ and, once the point has penetrated, it would be the smallest in the direction θ.

In a homogeneous medium, the direction of least resistance would be in the direction of maximum stress, which can be represented by the ratio R/A, where R is the force vector and A the projected area vector in the same direction as R. If the blade moves in the direction where R/A is largest in Fig. 4, the blade will move in direction $\theta/2$ if $\epsilon=\theta/2$, and will eventually move in the direction θ if $\epsilon=\theta$.

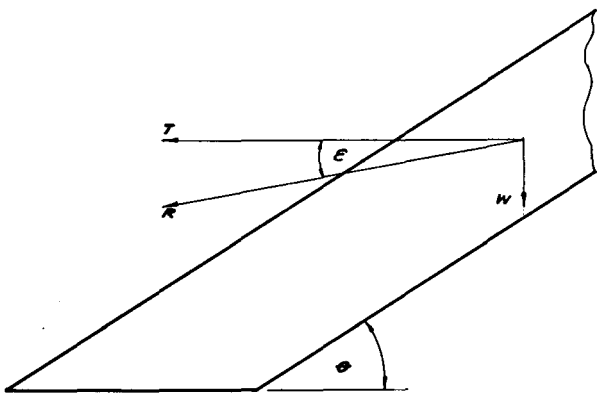

Fig. 4—Parallel-sided blade

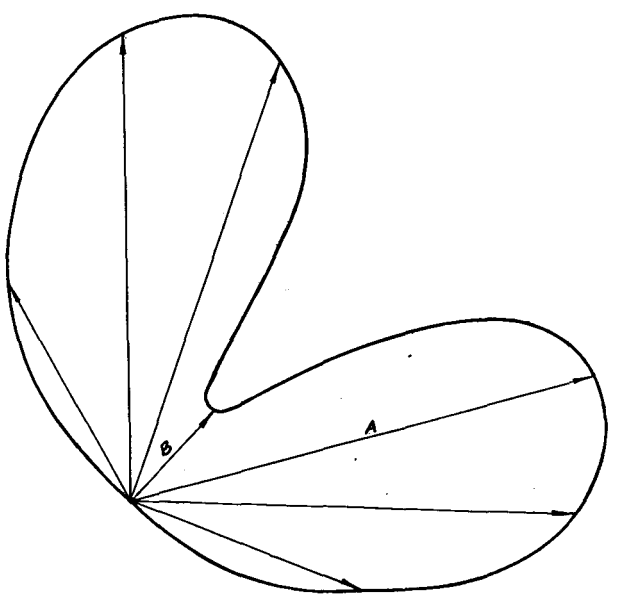

Fig. 5—Instantaneous polar diagram of resistant forces

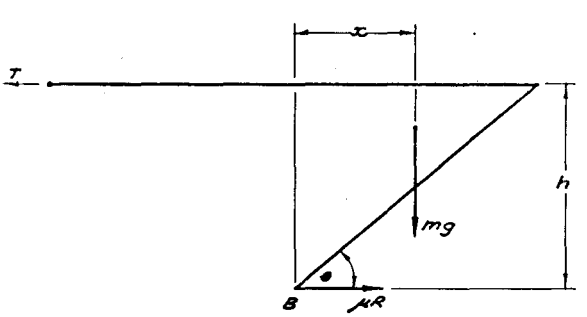

Fig. 6—Schematic sketch of a scoop in equilibrium

If $\epsilon < \theta/2$ initially and the blade tilts (i.e., θ increases), it will have an adverse effect on the penetration because the direction of maximum R and minimum A will move further apart. If $\epsilon > \theta/2$ initially, the tilting of the blade will have a favourable effect because the angle between maximum R and minimum A will decrease, thereby moving them closer to one another. The direction of A is fixed to the particular shape of the blade, but R varies as the force T increases.

Thus, it can be concluded that, with the parallel-sided blade shown in Fig. 4, the point of minimum resistance will be reached if the scraper blade is initially tilted backwards (i.e., $\theta \rightarrow 0$) and tilts forward (i.e., θ increases) whenever rope thrust T is applied. If the blade has penetrated partially into the medium, further tilting of the blade will be unfavourable and the best condition will prevail if R points in the direction θ (because $\Delta A=0$ in this direction for a parallel-sided blade). In other words, this means that the best penetration will be achieved if the resultant force of the blade points in the direction of the blade. It should also be recognized that, as a scoop fills, the contents assist the weight W, resulting in a favourable condition of ϵ being increased.

(c) *Retainment of Broken Rock and Stability*

The box type of scoop with its side plates providing a large running surface gives the best stability, but it discharges much of its payload when gliding over large rocks in its path. The hoe type of scoop, once stabilized, provides a good compromise. If the blade angle of a scoop is beyond the dynamic angle of repose of the broken rock, it will tend to dump its contents. The smaller the blade angle from the horizontal, the better the scoop can carry its load, but this in turn offsets the penetration and tipping action.

It is thus necessary to determine the optimum combination of these two functions.

In (b) it was determined that the centre of mass of a scoop should lie behind the blade contact point. To determine the requirements to impose a scoop's stability, the equilibrium conditions of the scoop shown in Fig. 6 were analysed.

For a condition of dynamic equilibrium,

$T=\mu R$ and $R=mg$.

$\therefore T=\mu mg$.

Taking moments about point B,

$Th=mgx$ or $\mu mgh=mgx$.

$\therefore h=x/\mu$.

Therefore, if $h > x/\mu$, the scoop will tilt forward and θ will increase. Conversely, if $h < x/\mu$, θ will decrease.

To achieve the condition for optimum stability, h should always be smaller or equal to x/μ, i.e. $h \leqslant x/\mu$.

As the scoop fills, μ will increase, making the ratio x/μ smaller and more difficult for h to remain smaller. The distance x to the centre of mass is also affected when the scoop fills, but could even increase to beyond the value of an empty scoop. The best solution to improve

stability is therefore to lower the bridle (making h small).

(c) *Tipping Action*

As mentioned above, the tipping action of a steeply inclined blade will be superior to that of a shallow-angled blade. A blade angle close to the natural angle of repose of the rock was selected as a compromise. This was assessed subjectively and set to 24° from the horizontal.

(d) *Fouling Action*

This requirement refers to the ability of the scoop to avoid 'hooking' onto any projecting edges of a gully. If an even action can be achieved, shedding of the load can be avoided and a greater overall width can be achieved. The hoe type of scoop lends itself best to a rounded-contour design to eliminate all sharp edges, whereas, for the box type of scoop, this requirement is very difficult and expensive to achieve.

(e) *Corner Digging*

Broken rock in the gullies always forms a tapered trench as it piles into the corners. This rock will inevitably be lost if a scoop cannot dig into the corners to remove it. A scoop the same width as the gully would be ideal, but in practice it would continuously foul onto the sidewall. When a scoop is narrower than the gully, it always tends to drift to the middle.

Large rocks in the gully deflect a scoop towards the sidewall and consequently clean the gully edges. The major factor contributing to a high corner-digging performance lies with the design of the blade edge, as will be described later.

(f) *Face Digging*

The ability of a scoop to penetrate into the broken rock pile as close as possible to the face where the return snatchblock is anchored can best be achieved with the shovel type of scoop, because the exposed blade without side plates enables it to penetrate more quickly into the rock pile. The box type of scoop is very poor in this regard; a hoe scoop with an extended blade could closely simulate the action of a shovel type of scoop.

(g) *Capacity*

The amount of rock a scoop can move in a single run depends on its size and retainment ability. This parameter should be matched with the scraper winches in use.

It was established that a capacity of 1½ to 2 tons for the gully scoops would be the optimum range to match the scraper winches used. The winches used for the purpose of this study are 22 kW on the face, 37 kW in the gullies, and 37 or 55 kW in the centre gully. A 22 kW winch can cope with 1 gully scoop, a 50 h.p. with 2 gully scoops, and a 55 kW winch can just cope with 3 gully scoops of the abovementioned size.

(h) *Power Requirements*

The power requirements have already been discussed.

Engineering Parameters

(a) *Ease of Manufacture*

Ease of manufacture is of vital interest to the persons concerned with the manufacturing process. A scoop must be designed so that it can be manufactured with the minimum of special facilities. The less sophisticated the equipment required, the greater the benefit. Because of high labour costs, it is advisable to avoid labour-intensive designs. A design that utilizes only a few components is an objective worth striving for. The material used plays a very important role, especially when welding is to be used as an assembly technique. Very hard steels should be carefully selected because they are more difficult to weld.

The box type of scoop has the most components, and the shovel type probably has the least. Hinged blades as found on certain box types of scoops should be avoided.

(b) *Durability*

The economics parameters are closely interrelated with durability. The optimum life of a scoop depends on the initial costs, the maintenance costs, and the probbility of re-use. A scoop that lasts longer than the life of one panel has to be moved to a new panel for re-use if optimum advantage is to be gained from its durability. The probability that the equipment will not be re-used to the full decreases the advantage obtained from a more durable scoop. If a scoop could be designed with an expected life expressed as a multiple of panel lives, it would also limit the transporting activity on a gold mine.

The probability of re-use of a scoop affects the allowable cost of a more durable one. Factors were derived to determine, for different probabilities of re-use (p), the cost increases that could be allowed in the manufacture of a scoop with a longer life. Table II indicates these factors for scoops that could outlast one, two, or three stope panels.

As shown in Table II, if the probability of re-use is 60 per cent, a scoop that could outlast two panels should cost only 1,61 times as much as a scoop that lasts the life of one panel.

The cost factor may not be the limiting constraint in the manufacture of a more durable scoop; there may be

TABLE II

FACTORS BY WHICH SCOOP COST CAN BE INCREASED FOR A HIGHER DURABILITY

Scoop life as a multiple of panel life	Probability of re-use					
	0,8	0,7	0,6	0,5	0,4	0,3
1	1	1	1	1	1	1
2	1,79	1,72	1,61	1,52	1,41	1,32
3	2,87	2,55	2,25	1,98	1,75	1,55

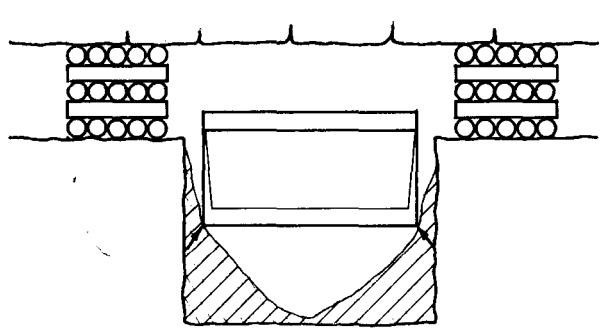

Fig. 7—Profile of broken rock lying in gullies

other practical reasons. It should be borne in mind that a piece of equipment is only as durable as its least durable component, which demands a good balance between the components. The higher the objective durability, the more difficult it is to achieve this balance.

The most vulnerable component on a scoop is the corner section of the blade. This is due to the tapered shape of the broken rock lying in the gullies, which is indicated in Fig. 7. This explains why most scoop blades wear down in a curved shape, greatly reducing the overall performance of the scoop.

(c) *Structural Strength*

Many scoops have failed because of their inability to resist shocks and strains. A survey of scoops has indicated that this problem is mainly caused by sharp corners and edges. In order to overcome this problem, certain designs utilize reinforcing plates. This practice is not favoured because the plates tend to be added as an afterthought and increase the manufacturing time and cost.

Scoops with hinged blades are particularly subject to breakages. A design that was recently tested used component plates to build up the scoop body, but also failed as a result of fractures at the seams.

Economics Parameters

(a) *Maintenance*

Maintenance is very difficult to carry out on scoops in the stopes because it is a very tedious task to transport tools and equipment to and from a stope, and the maintenance is often neglected. It is therefore recommended that a scoop should require as little maintenance as possible. Certain designs incorporate bolts or locking pins and wedges to hold the components together. These practices are not favoured because, if these components are not protected against wear, they can be removed later only by use of a cutting torch. Welding-seams in particular should be well protected, because, if they wear down, premature fractures will result.

(b) *Manufacturing Costs*

The labour cost constitutes a large proportion of a scoop's manufacturing cost. Scoop prices vary from approximately R280 to R850, the box types of scoops tending to be more expensive than the other types. The reason is obvious if the number of separate components is considered. A comparison of manufacturing costs should be based on standard material and labour costs for all scoops.

Summary of Objective Parameters

Table III gives a summary of the preceding discussion, and indicates the advantages and disadvantages of the three basic scoop designs.

Design and Development of a Scraper Scoop

Based on the evaluation of the interrelationships between the objective and design parameters in the foregoing discussions, an actual design was established.

Scoop Size

The maximum width of the scoop is governed by the width of the stope strike gully, which is 1,5 m. In fact, the final size was influenced by the dimensions of a standard mild-steel plate, taking into consideration minimum off-cut wastage.

Basic Layout

The hoe type of scoop layout was selected because, as shown in Table III, it has the potential to achieve the best balance between the requirements of the objective parameters. The body of the scoop was designed as shown in Figs. 8 and 9, and consists of the following five components:

(a) a body shell, which is pressed or rolled from a profiled plate,
(b) a bottom plate, also profiled to fit the bottom of the body shell,
(c) a bridle consisting of two rolled or pressed semi-order flat bars welded to the body shell,
(d) rope-attachment lugs welded onto the front and back,
(e) the blade (which will be discussed later).

This design was selected because it has the following advantages.

(a) The shape is totally rounded, which will prevent the scoop from fouling onto sharp edges in the gully.
(b) There are very few components, which will assist in the manufacture of the scoop.
(c) Because of the relatively few welding seams and round shape, the structural strength of the scoop should be approaching optimum.

The Blade Angle

The three blade angles most commonly used at

TABLE III

COMPARISON OF THREE SCOOP DESIGNS

	Fragmentation handling	Penetration	Rock retained	Tipping action	Fouling action	Corner digging	Stability	Face digging	Capacity	Ease of manufacture	Durability	Structural strength	Maintenance	Manufacturing cost	Ease of handling
Box type	Fair	Poor	Good	Fair	Poor	Fair	Good	Poor	Fair	Poor	Fair	Fair	Poor	Poor	Poor
Hoe type	Good	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair	Fair
Shovel type	Good	Good	Poor	Good	Poor	Fair	Poor	Good	Fair	Fair	Fair	Fair	Fair-Poor	Fair	Fair

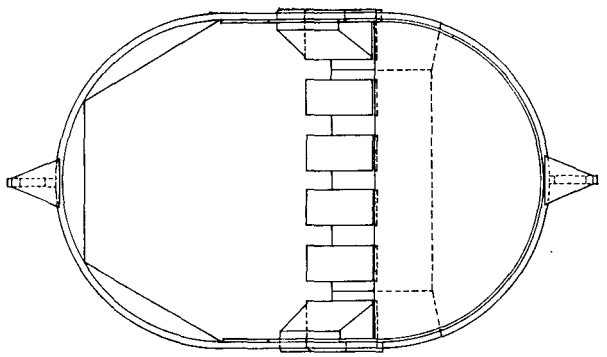

Fig. 8—Body of the scoop developed — in plan

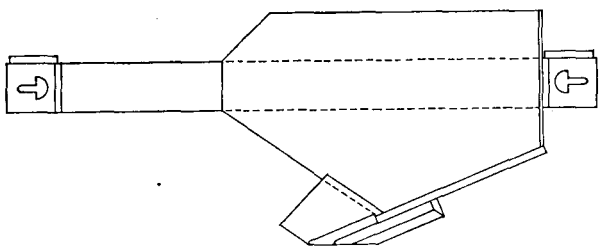

Fig. 9—Body of the scoop developed — in section

present are 20°, 24°, and 35° from the horizontal. Three experimental scoops with these blade angles were manufactured and tested. Of these, the 24° blade proved to give the best overall results. The 20° blade retained too much broken rock during its return travel, and the 35° blade retained too little broken rock during its forward travel. The scoop with the 24° blade retained its contents well, tipped itself clear, and gave good performance in penetration and broken-rock handling.

Blade Design

The blade design was one of the most contentious elements of the scoop. Certain designs used a very sharp edge pointing downwards as shown in Fig. 10. This was intended to improve the penetrating ability. The blade was also equipped with a double edge, which can be rotated once the edge of one side has worn down. This proved unpractical because the blades were too often left to wear down to a state beyond repair. The use of bolts or rivets is also very troublesome because they usually need to be cut with a gas flame. The double-edged blade was attached as shown in Fig. 10, which gives no wearing surface on the footwall. A single-edged blade as attached in Fig. 11 utilizes the bottom area of the blade as a wearing surface on the footwall, except that the area of the wearing surface decreases as it wears down.

For the proposed design, it was therefore decided to retain a blade with parallel sides as shown in Fig. 12. This design also provides for well-protected welding seams. The most vulnerable point on all scoops has proved to be the corners of the blade, which implies that a scoop can last only as long as the corner section. During the investigation, it was found that many scoops had worn right down into the body shell at the corners, causing premature fractures. The box type of design is favoured here because of its heavy side runners. Since the hoe type of scoop was adopted for this exercise, an experimental corner section was designed as shown in Fig. 13 and cast from a hard-wearing steel.

As indicated, this corner section has a larger wearing surface than the blade. The 'body' of the corner section was turned outwards to present a wearing surface in the direction of maximum wear. This design also incorporated a sharp edge to improve corner digging. Although this design performed well during test runs, the forward-pointing edge wore down very rapidly. Consequently, this sharp edge was replaced by a wearing surface as shown in Fig. 14.

This design considerably improved the wear resistance, but some corner-digging efficiency was sacrificed. The ridge on the side was eventually abandoned because the actual measurements of wear patterns showed that the advantages did not warrant the expense. The inner corner was filled by a metal fillet as shown in Fig. 15 to maintain a large wearing surface on the corner as the wear progressed. This design has proved successful and is used on the production model.

After 6 months of operation, 90 per cent of the blade material has been worn away and is in balance with the condition and expected life of the body of the scoop.

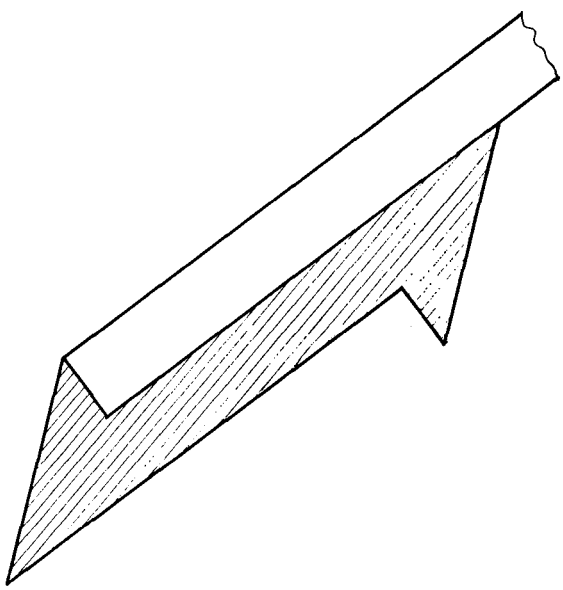

Fig. 10—Sharp-edged blade pointing downwards

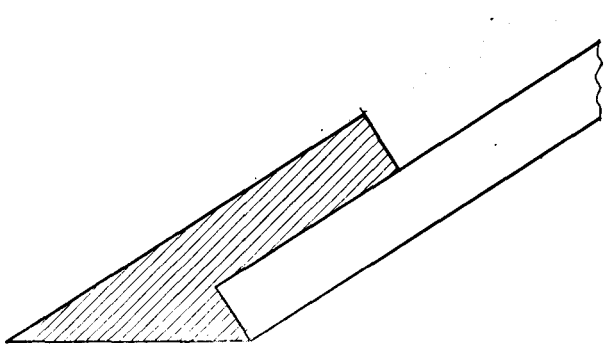

Fig. 11—A single-edged blade utilizing the bottom area of the blade as a wearing surface on the footwall

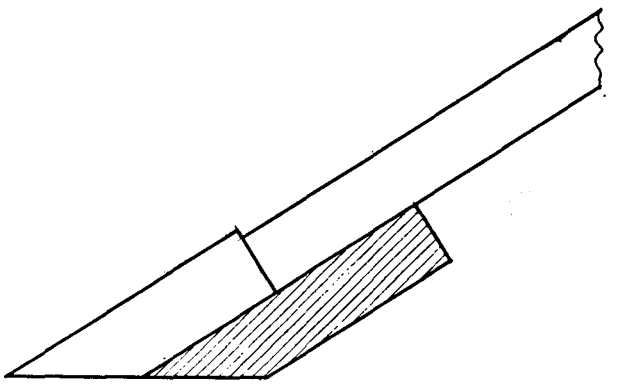

Fig. 12—The blade with parallel sides that was used in the scoop developed

Mass of the Scoop

The mass of the scoop in its production form is 410 kg. The blade sections contribute 170 kg, and the body 240 kg of the total mass. With this relatively large mass, the scoop penetrates into the broken rock and picks up large rocks very well.

Material

The material originally used for the scoop blade was R00-LAST AH400, which is an Iscor trade name. This steel has a high initial hardness of 400 Rockwell C, and does not rely on work-hardening. However, this plate has the disadvantage of being very difficult to weld. It requires preheating and skilful welding to prevent cracks. It was subsequently rejected in favour of a highly wear-resistant cast steel bearing the trade name

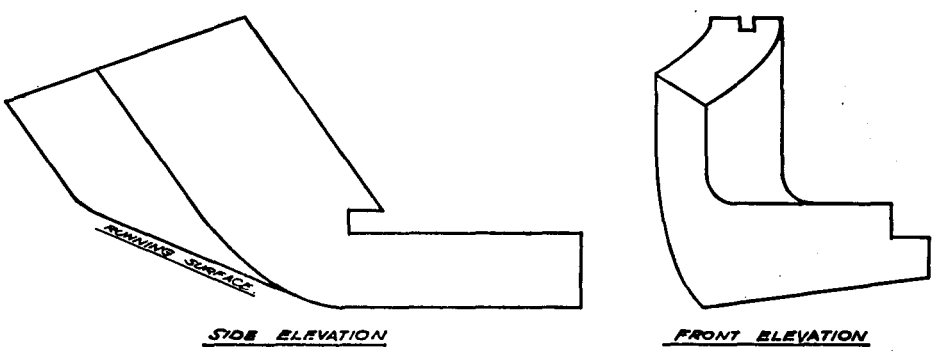

Fig. 13—Corner section of the initial design

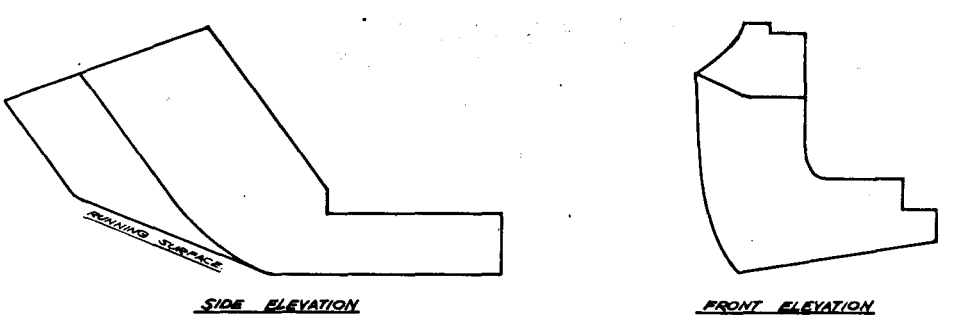

Fig. 14—Corner section of the second design

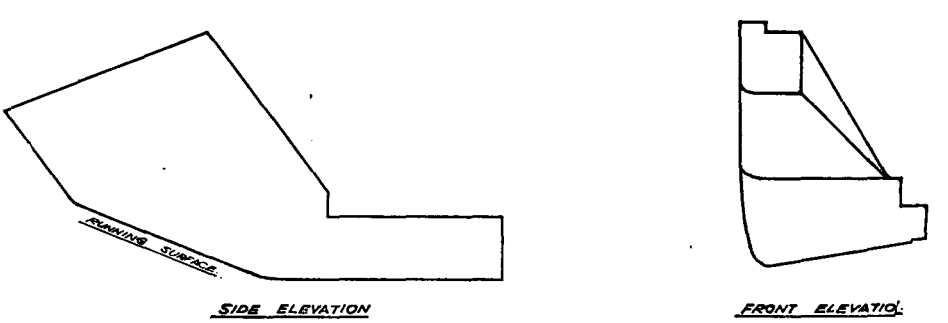

Fig. 15—Corner section of the final design

APEX-6, which is relatively easy to weld and requires little preheating. The fast fluxcore welding process was used to weld the corner pieces onto the body.

If the corner sections last well, they will protect the centre section of the blade, which means that a hard-wearing material is not required for the latter. The centre section was therefore cast from BS 592 steel, which is inexpensive but is slightly harder than mild steel. The body is made of 10 mm and 20 mm thick mild-steel plate, and the bridle is made of 150 mm by 20 mm flat bar.

Method of Assembly

Welding was selected as the assembly technique because it is relatively inexpensive and quick. The technique also conforms to the requirements of low maintenance and structural strength. However, care must be taken that the welding seams are not exposed to wear. The complete body design shown in Fig. 16 indicates that these requirements have been fully met. Details of the blade and inner corner section are shown in Figs. 17 and 18.

Wear Attachments

The use of a heavy corner section to retard corner wear has already been discussed. Another point that requires protection from the scoop rope is the top edge of the body. Used hexagonal drill-steel stems are welded onto the edge to prevent the scraper rope cutting into the body. A hard-steel section is also welded onto the leading edges of the body to prevent tearing.

Rope-attachment Lugs

The attachment lugs are shown in Fig. 19. The design uses a button-hole principle to accept a 16 mm chain

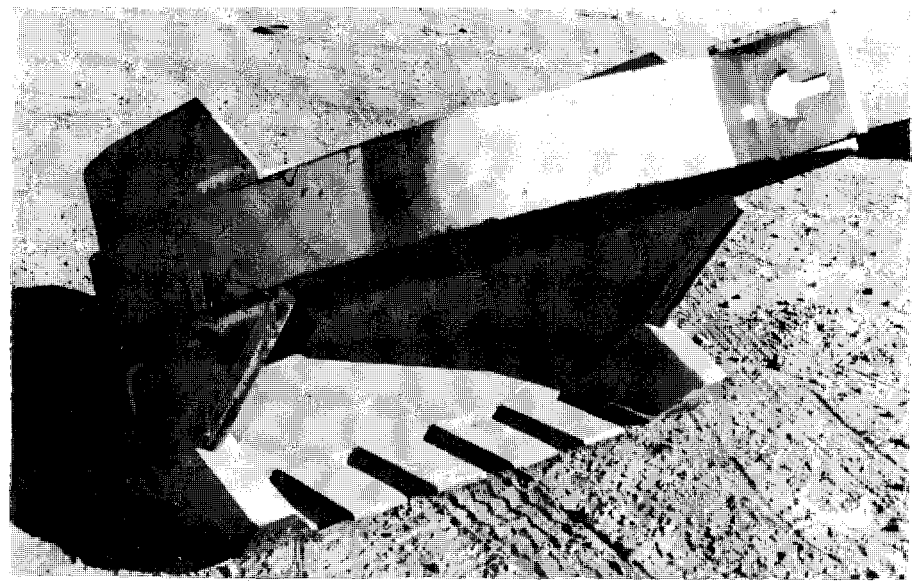

Fig. 16—Hoe type of gully scoop

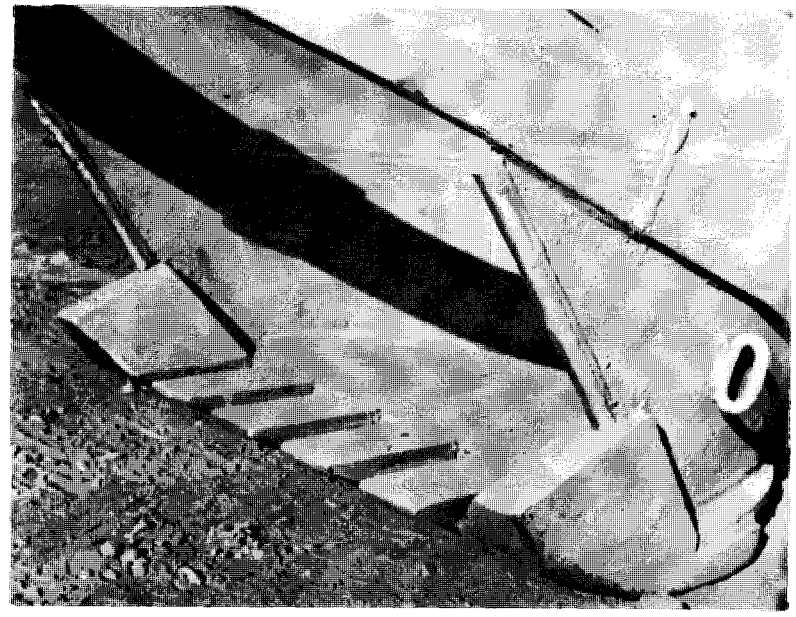

Fig. 17—Detail of gully-scoop blade

Fig. 18—Detail of inner corner section of gully-scoop

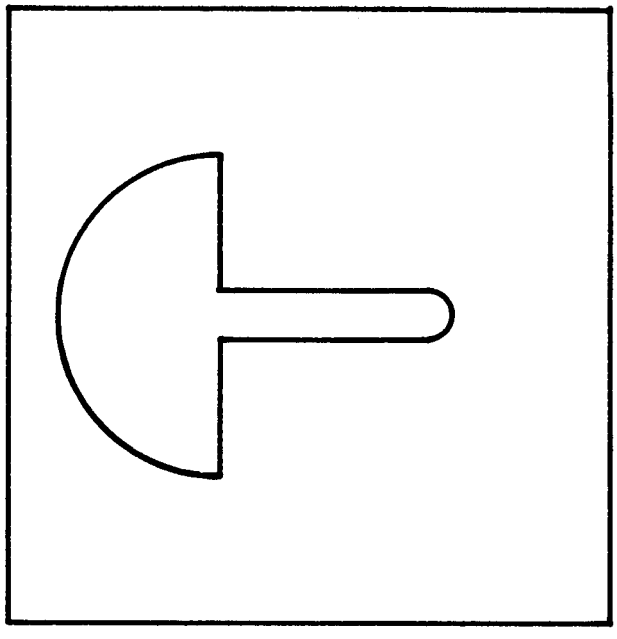

Fig. 19—Design of attachment lug

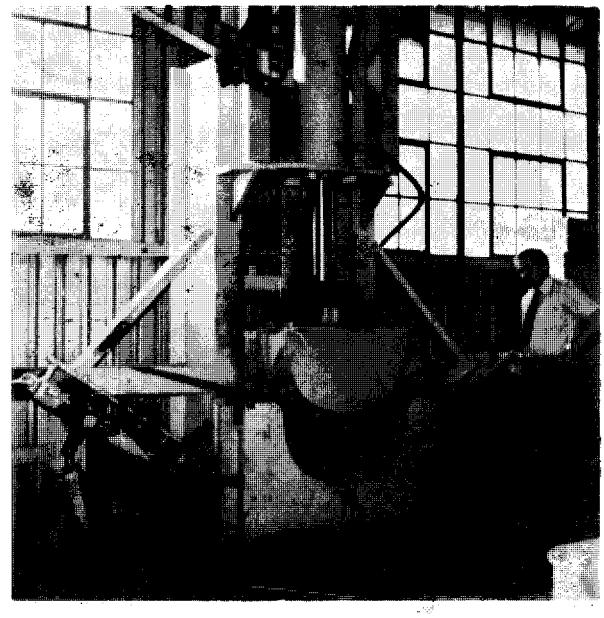

Fig. 20—Hydraulic press for the hoe type of gully scoop

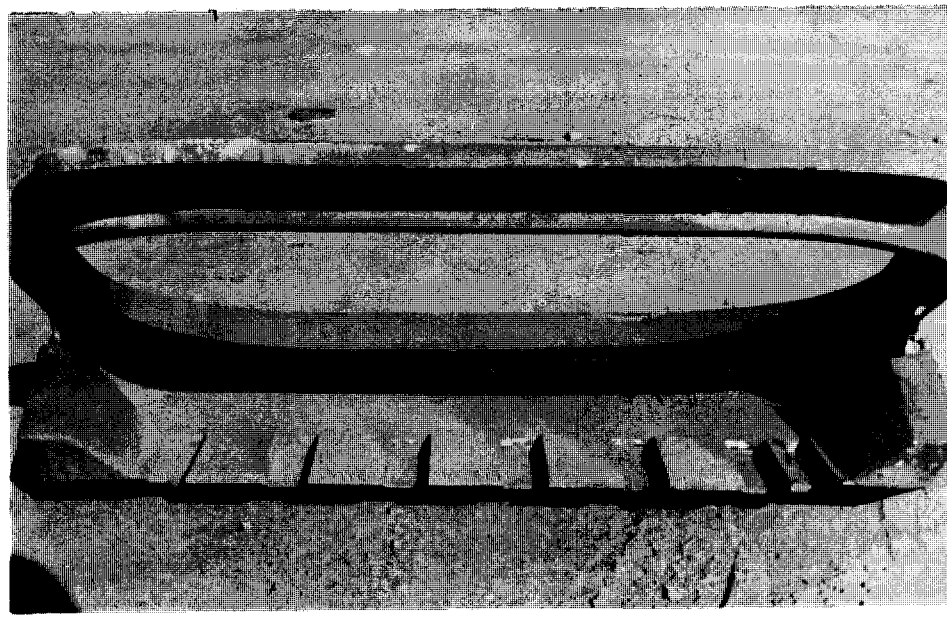

Fig. 21—Hoe type of face scoop

link. In this way, the chain carries the full pull load and eliminates the distortion of bolts and shackles, which with present use become impossible to loosen.

Manufacturing Considerations

As previously mentioned, labour is the major cost-contributing factor of the present scoop designs. To roll the scoop body into the semi-round shape is very time consuming, and, in order to reduce this, a hydraulic press was designed as shown in Fig. 20 to simultaneously press the body and bridle into the required shape. After the body is pressed, it is swung over on a jig and tacked together. It is then suspended on a chain block and moved to the welders for completion. With this process, only 7 man-hours are required for the manufacture of a gully scoop.

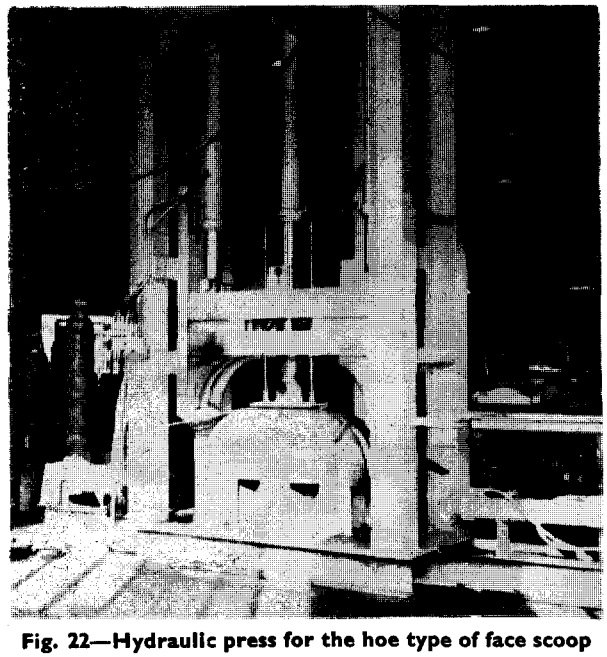

Fig. 22—Hydraulic press for the hoe type of face scoop

Development of a Face Scoop

The same development pattern as described above was followed in the design of a face scoop. Figs. 21 and 22 show a face scoop and a press for the body shell. This scoop has a non-fouling advantage, particularly on panels where self-advancing barricade systems are used.

Conclusion

Close monitoring of the prototype scoops has proved that, to a large degree, most of the objectives have been achieved. The designed load-carrying capacity of the scoop matches the winch power available, which has been proved in practical on-the-job tests. One of the major objectives, which was to produce a maintenance-free scoop capable of lasting the life of a stope panel, has been met. In the working places monitored, the number of blasts lost as a result of incomplete cleaning was significantly reduced.

The gully scoops are now in general use, and the overall actual effects will be measured in approximately 6 months' time by a comparison of the present consumption of scoops with that previously, and by an evaluation of stope efficiencies and average face advance.

Although the design evolved is not the ultimate, it is hoped that further improvements, to suit other stoping situations in narrow reef, can be made by application of some of the basic theories presented in this paper.

Acknowledgements

The authors express their appreciation to all who were associated with this project for the time and effort they spent on the practical testing and implementation of the theory that was developed. In addition, they thank General Mining & Finance Corp. Ltd for permission to publish this paper.

NIM reports

The following reports are available free of charge from the National Institute for Metallurgy, Private Bag X3015, Randburg, 2125 South Africa.

Report no. 1969

The interaction of silicon monoxide gas with carbonaceous reducing agents. (30th May, 1978).

The rate of reaction between different carbonaceous reducing agents and silicon monoxide gas was studied by the use of a technique developed in Norway by Tuset and Raaness. Argon carrier gas was used to pass a known fixed concentration of silicon monoxide gas through an 11 cm^3 sample of precalcined reducing agent at 1650 °C. Silicon monoxide gas was generated by the heating of a mixture of silica and silicon carbide, and the progress of the reaction was followed by monitoring the carbon monoxide in the off-gas with an infrared spectrometer.

The reactivity of silicon monoxide gas towards a number of carbonaceous reducing agents (charcoal, Iscor coke, Lurgi char, and petroleum coke) was shown to be similar to results obtained with the carbon dioxide reactivity test. The silicon monoxide reactivity test was able also to indicate the extent to which fines were generated when the reducing agent was converted to silicon monoxide. Of the reducing agents studied, Lurgi char appeared to be the most suitable for the production of ferrosilicon because it has a high reactivity towards silicon monoxide gas and high strength when converted to silicon carbide.

So that industry could be given some indication of the suitability of certain reducing agents for the production of ferrosilicon and silicon metal, a reactivity scale similar to that of Tuset and Raaness was constructed that compares the volume of silicon monoxide gas required by a reducing agent to reach a carbon monoxide content of 10 per cent in the off-gas during the conversion of the reducing agent to silicon carbide. This scale assists in the selection of those reducing agents that require less silicon monoxide gas for conversion to silicon carbide, and that give savings in material and power costs because less silicon monoxide gas need be generated in the furnace.

Report no. 1975

Reactions in the production of high-carbon ferromanganese from Mamatwan manganese ore. (18th Aug., 1978).

The influence of the reducing agents Iscor coke and Delmas coal on the reduction of Mamatwan manganese ore to ferromanganese was examined by use of a variation of the 'stationary charge in controlled environment' (SCICE) technique.

The charges of Mamatwan ore and the appropriate reducing agent were heated to temperatures between 1300 and 1600 °C and either cooled immediately the required temperature had been reached or held at that particular temperature for periods of up to 4 hours. The heating rate was 350 °C per hour, the particle size of the ore was between 2,83 and 12,7 mm, and the particle size of the reducing agent was between 2,83 and 6,35 mm.

After being allowed to cool in air, the reacted charges were examined by optical microscopy, X-ray-diffraction analysis, scanning electron microscopy, and electron-microprobe analysis. No significant difference was found between the qualitative changes observed during the reduction of Mamatwan manganese ore by Iscor coke and by Delmas coal. A tentative hypothesis is advanced to explain the mechanism of formation of the Fe–Mn–C alloy.

Report no. 1985

The determination of boron in high-purity aluminium metal by spark-source mass spectrometry. (15th Aug., 1978).

A method is described for the determination of boron in high-purity aluminium metal. Both isotopic boron lines ($^{10}B^+$ and $^{11}B^+$) are used for the analysis. As there are no low-abundance isotopic lines for aluminium, measurements were made direct without reference to aluminium as an internal standard.

The boron concentration values of eight aluminium samples analysed by this method compared favourably with results obtained from other techniques.

World copper

Western World consumption of refined copper is expected just to tip the seven million mark for the first time this year, before turning down again slightly in 1979. Buoyant fabricator activity in the U.S.A. and Japan in the first half of this year helped to maintain the demand for copper at a remarkably brisk level, but a marked slowdown in the American economy next year will be reflected in a fall in copper consumption there. Meanwhile, little growth is expected in the Japanese and European demand for copper.

This is the picture painted in CRU's latest quarterly review of the copper industry. However, CRU's outlook for copper prices is not quite as gloomy as might be expected, since a small decline in mine output this year is pushing the industry into supply deficit, and the resultant rundown in metal stocks will have a firming effect on LME prices in the next few months. The wirebar quote could pass the £800 mark by the end of the year.

The tighter supply situation currently being experienced is the result mainly of the familiar problem of operational and transport difficulties in central Africa. Cost pressures and imports are continuing to affect North American production, and labour problems may reduce shipments from Peru.

By the second quarter of next year, the slackening in world demand for copper, combined with the reappearance of a supply surplus as new mines open in Mexico, Iran, and the Philippines, will once again hit prices, and CRU expects the LME wirebar quote by the summer of 1979 to be only modestly up on the levels prevailing at present.

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