

~~THE~~ SELECTION OF OPEN PIT EXCAVATING EQUIPMENT— A SYSTEMS APPROACH

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1. INTRODUCTION

The ever increasing world demand for minerals over the past twenty years has caused mining engineers to increasingly turn their attention to the economic exploitation of near surface deposits of low grade and high overburden ratio. This has resulted in larger outputs from individual mines because of the increased volumes of waste and the need to achieve economies of scale. Open pit mining machinery has shown a marked tendency to increase in size to enable these deposits to be mined economically at high outputs. Open pit mining has therefore tended to become a problem of materials handling on a massive scale.

The economies that can be achieved by the introduction of large machines must however be set against their disadvantages. Breakdowns assume much greater importance. The following difficulties arise:

Component parts are larger and repairs take longer.

Because of their size, spare parts are costly to hold in stock.

The custom built nature of the machinery means that spare parts are more difficult to obtain quickly.

Loss of production due to breakdowns is greater and more costly.

The need for more detailed investigation into the selection of machinery for open pit mining is readily apparent and a better understanding of machine performance is essential. Additionally greater reliability is essential. Four major lines of action are available to obtain increased reliability:

1. Improved operating characteristics
2. Improved mechanical and electrical design
3. The correct selection of materials
4. The use of high reliability components.

Most of these items follow from a better knowledge of machine performance.

Systems approach to the selection of open pit mining machinery

The action of an excavator must be related through its mechanical drive, its electrical system and structure to a cost figure which can provide a quantitative basis for selection. It is obvious that mining, mechanical, electrical and structural engineers cannot work in isolation and a systems approach to the selection of open pit machinery is essential. Similarly although the selection of the excavating machinery is of considerable importance it is also essential to remember that the various operations within an open pit e.g.

ground preparation,

loading,

transport and

mineral treatment,

are interdependent. As an illustration the increased use of explosives may result in reduced loading, transport and treatment costs which amply repay the additional ground preparation costs. As the various operations are inter-related, the optimum cost per tonne cannot necessarily be obtained by attempting to minimise all costs. The ground preparation method and crusher size must be compatible, while loading and transport equipment must be appropriately matched. The primary crusher size must also be related to the dipper or bucket size of the loading machine. The production requirements have a significant bearing on all operations and the costs of all operations are interlinked to some degree. The logic diagram for a simple shovel/trucking operation (Fig 1.1) shows the sequence of various operations, inter-related items and feedback. It is within this framework that

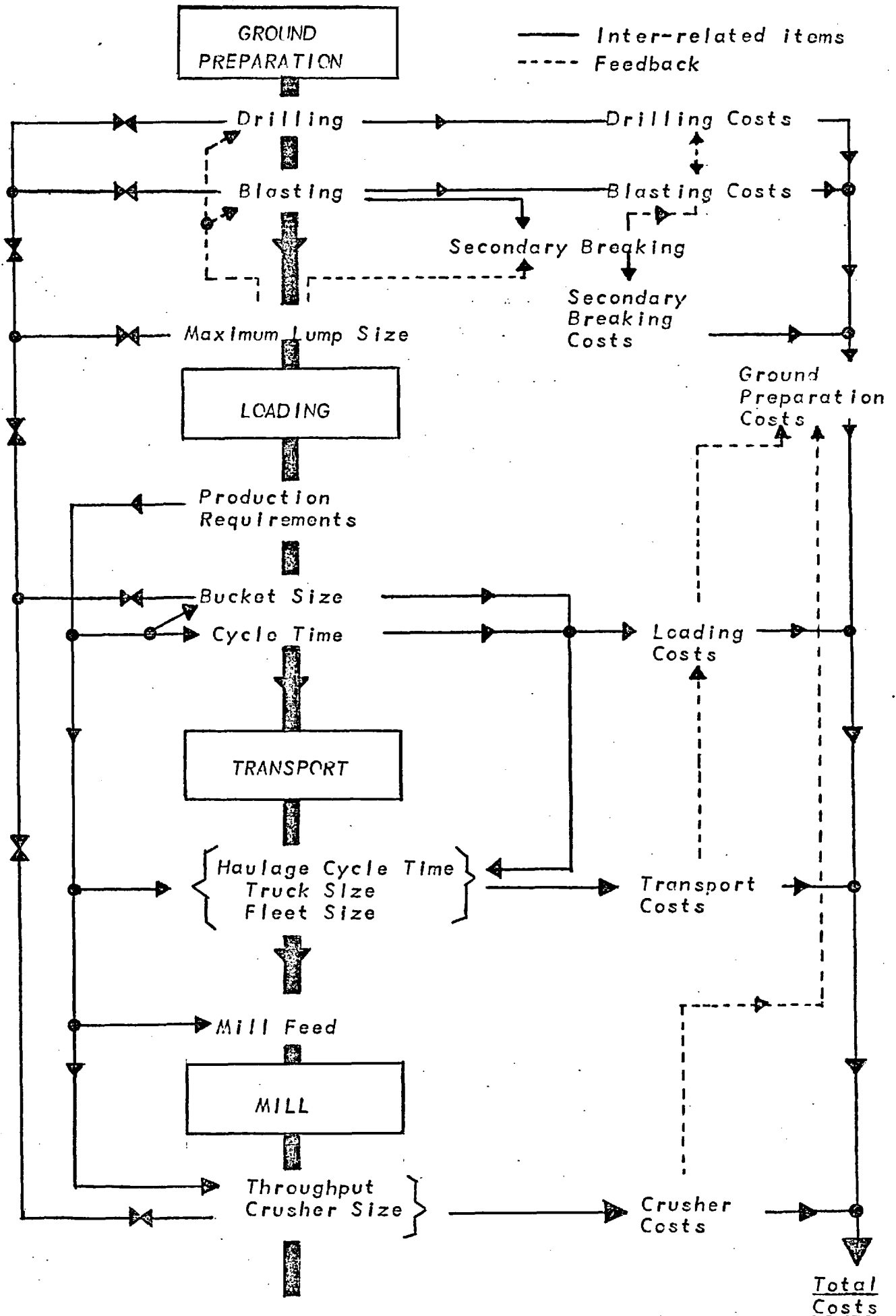


FIG.1.1 LOGIC DIAGRAM SHOVEL-TRUCKING OPERATION

the selection of open pit mining equipment must be selected.

To further illustrate this, reference is made to an extensive HOCUS Manual Simulation exercise, which although primarily carried out by the author to prove the method for teaching purposes, gives an excellent indication of the interdependent nature of open pit mining operations and the importance of the systems approach. A mine model was prepared (Fig 1.2) using from one to five shovels and with up to 18 trucks in service. Cumulative probability distributions for poorly and well fragmented ground were prepared from data compiled at Sherman Mine, Tergami, Canada, supplemented by discussions with the mine staff to establish any limiting conditions (Fig 1.3 and 1.4). The data was not originally prepared for this purpose and was further refined by discussions with several experienced open pit mining engineers. This in itself is a valid statistical process since it increased the sample size and covered many man-years of experience. The actual loading times were read from Figs 1.3 and 1.4 using random number tables. The travel and dumping times were fixed for simplicity and a single crusher station assumed. The simulation model was run over 3 days (simulated time) for each case (the single shovel was simulated for 4 days, but little change in the results was observed from a 3 day period). Fig 1.5 summarises the average results of the simulation for badly fragmented ground and Fig 1.6 for well fragmented ground. These figures clearly indicate that for each number of shovels, the production increases linearly as the number of trucks increase up to a point where rate of increase in production starts to fall away. If the number of trucks is further increased production becomes constant i.e. the system becomes "saturated". It will be noted for badly fragmented ground

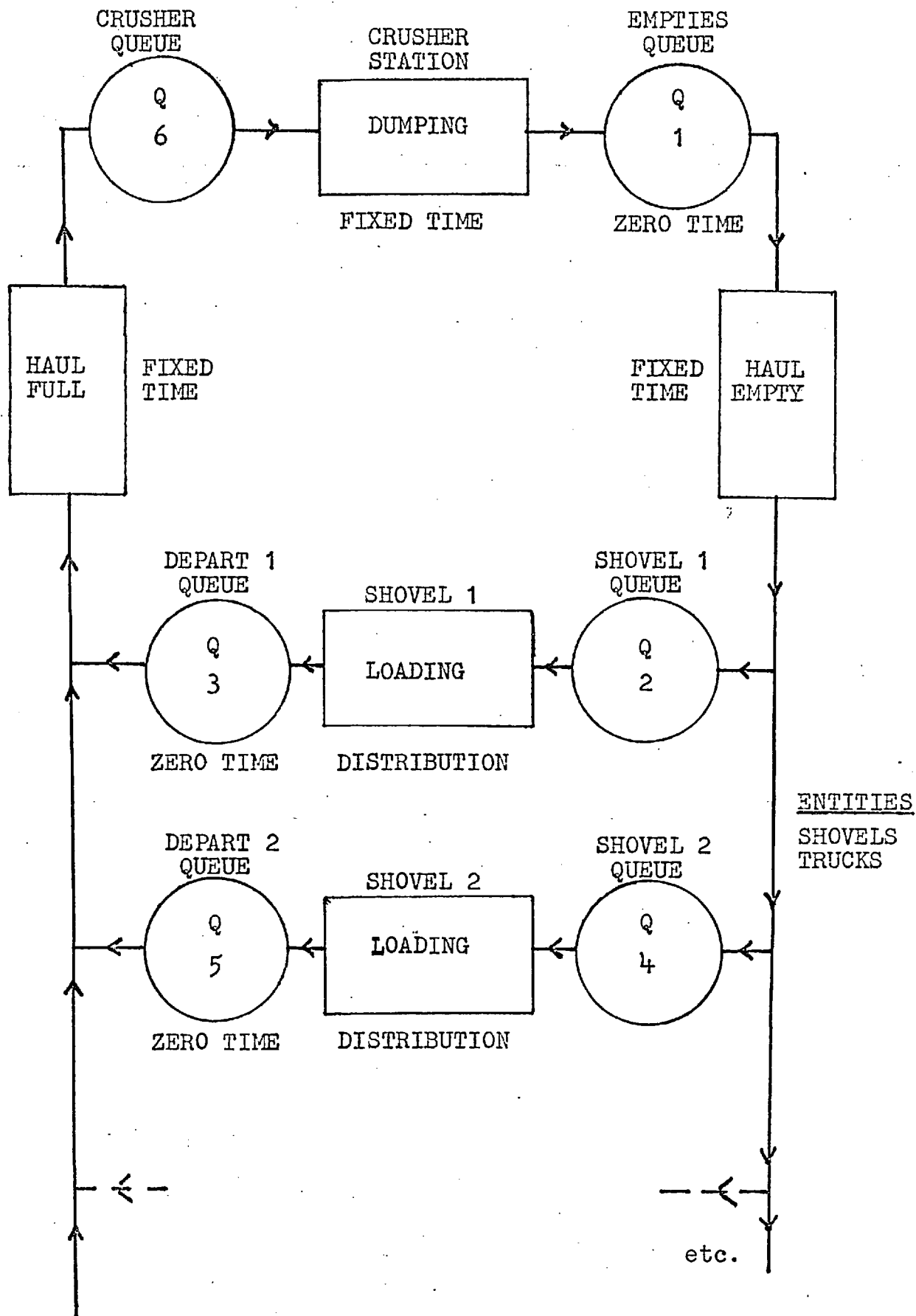
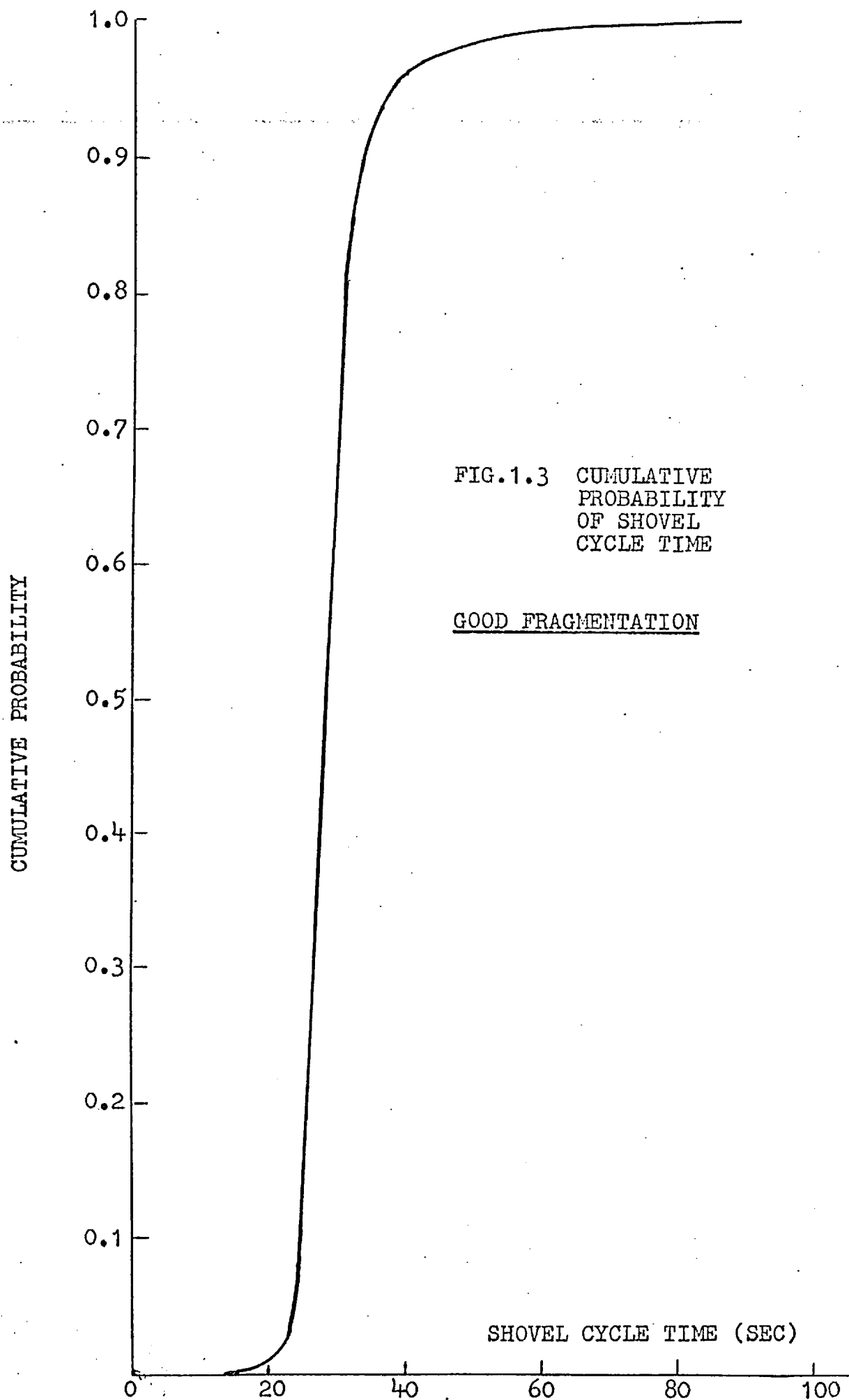
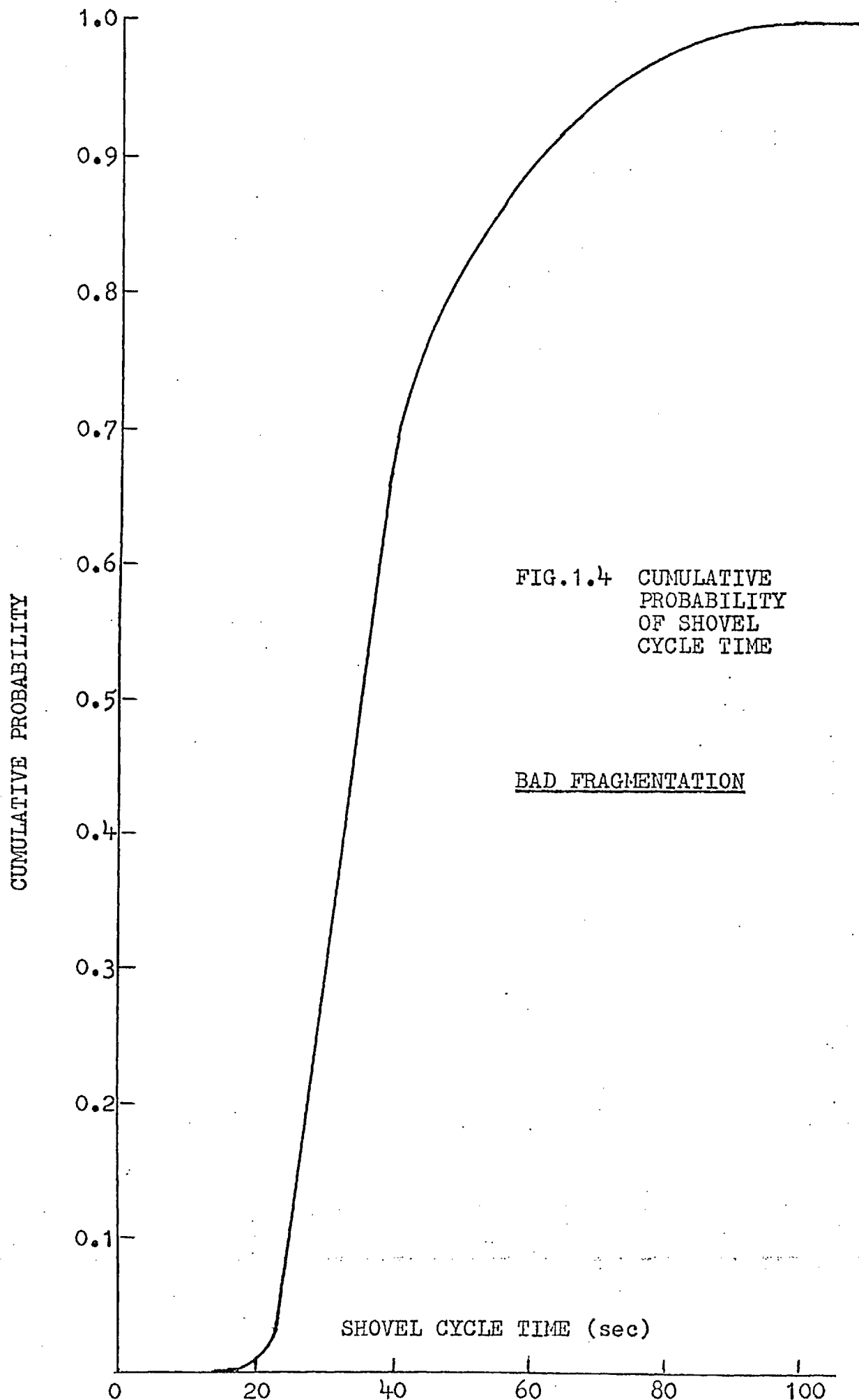


FIG.1.2 BASIC MANUAL SIMULATION MODEL OF
SHOVEL/TRUCK SYSTEM





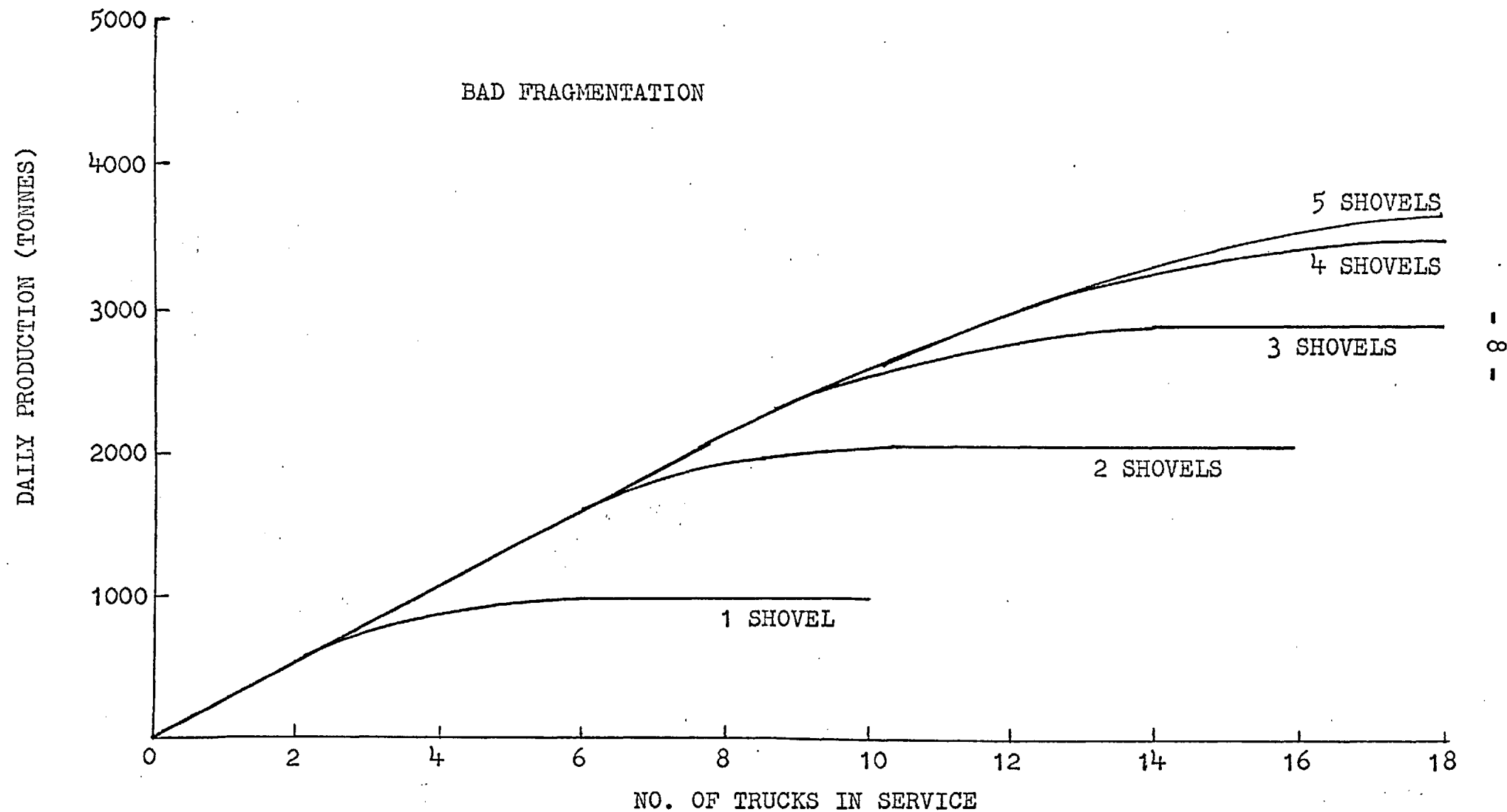


FIG.1.5 SHOVEL/TRUCK SYSTEM - PRODUCTION WITH BAD FRAGMENTATION

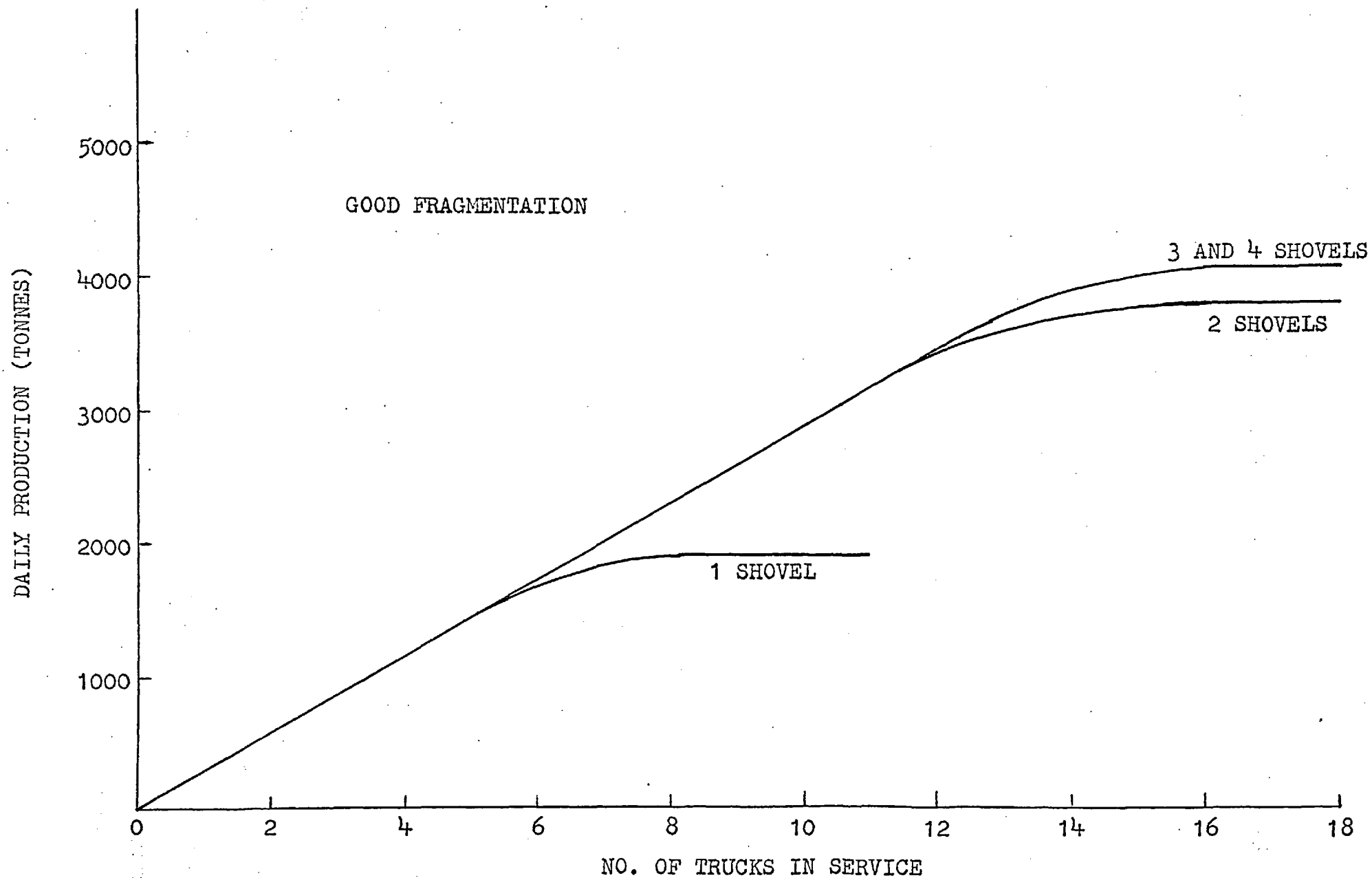


FIG.1.6 SHOVEL/TRUCK SYSTEM - PRODUCTION WITH GOOD FRAGMENTATION

(Fig 1.5) that the addition of the fifth shovel only increases production by a very small amount over 4 shovels.

Fig 1.6 for well fragmented ground shows much better performance, the maximum production being 4050 tonnes/day with 3 shovels and 16 trucks. It will be noted that the addition of a further shovel i.e. 4 shovels, does not increase production.

The maximum production for the system appears therefore to be 4050 tonnes/day. The reason for the failure of additional shovels and trucks to increase production is almost entirely due to queues at the crusher.

Figs 1.7 and 1.8 show the waiting times per day recorded during the simulation for a single shovel against the number of trucks in service. These curves can be used to calculate optimum truck fleet sizes to obtain the minimum overall cost per tonne where the ground preparation costs are known.

Purpose of Thesis

The purpose of this thesis is to suggest more comprehensive procedures for the analysis of the critical data used in selecting open pit excavating and loading equipment. The thesis is restricted to excavating and loading equipment since there is an abundance of literature on general open pit design and the selection of equipment systems rather than examining the equipment in detail. Extensive literature is also available on matching loading to transport equipment. It must be clearly appreciated however that it is not good practice to select excavating and loading equipment in isolation (or any other open pit equipment) but to consider all the aspects involved is beyond the range of a single

FIG.1.7 PRODUCTION AND WAITING TIMES PER SHOVEL

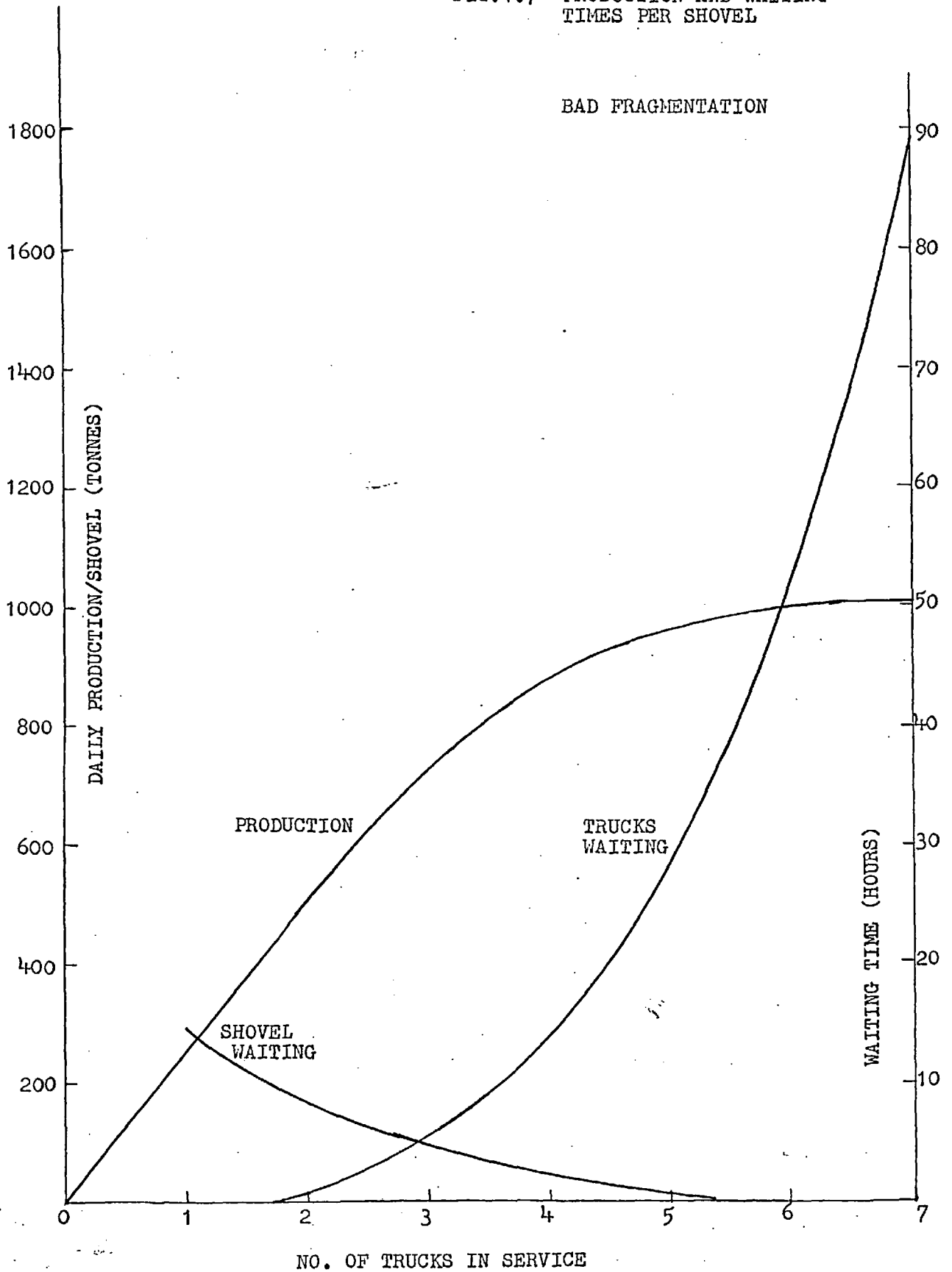
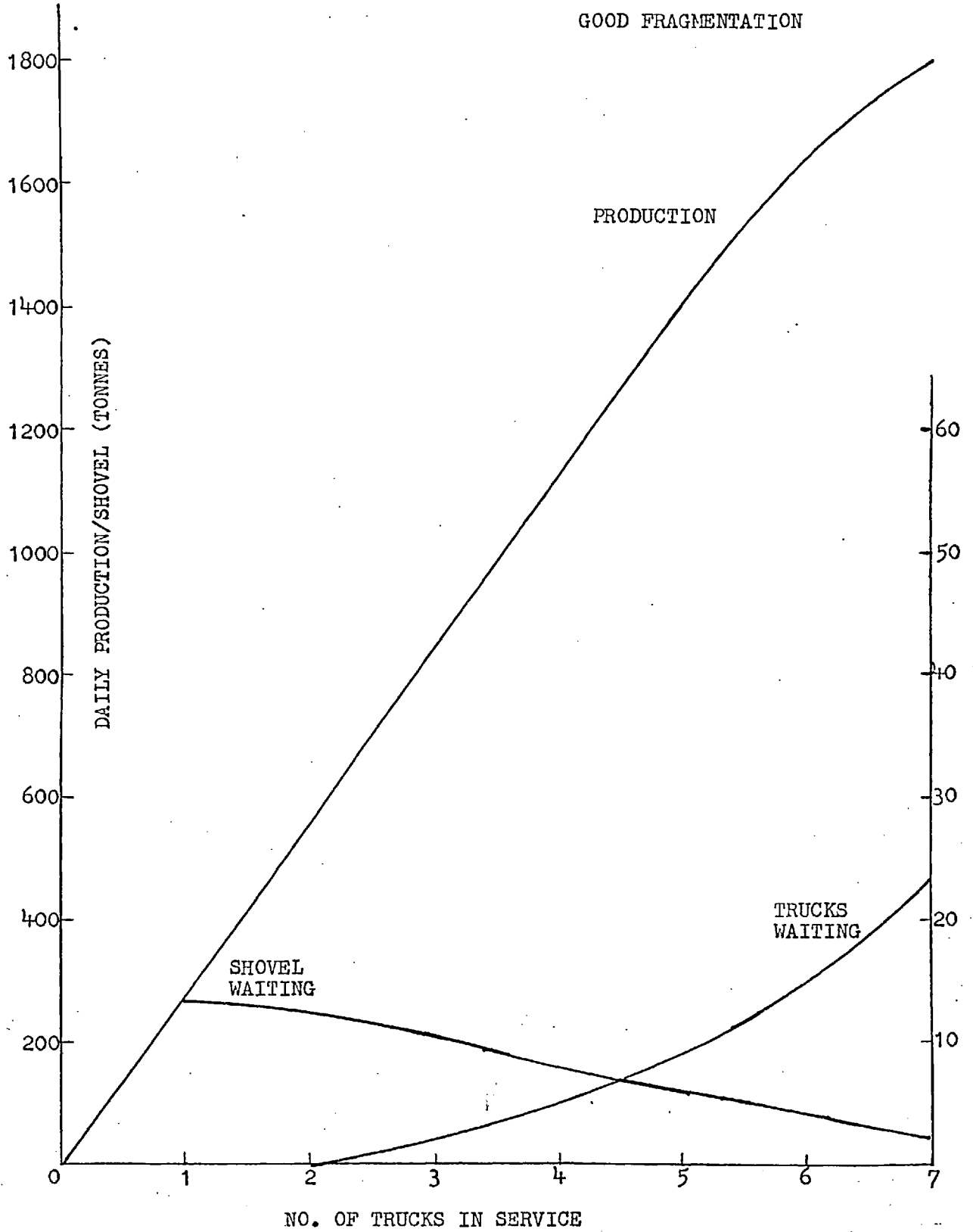


FIG.1.8 PRODUCTION AND WAITING
TIMES PER SHOVEL



thesis and would require the efforts of a team of workers. The selection of the excavator or loader is of prime importance because it largely determines the other equipment required, the mode of operation and it is the key to low-cost production. It is not necessarily the starting point in mine planning and this thesis should be considered in conjunction with other works on open pit optimisation, ground preparation, transport systems, etc.

Because of the breadth of treatment required for the subject, a considerable fund of known work has had to be summarised to enable a continuous theme to be maintained throughout the thesis. The author has attempted to integrate the original work presented in the thesis with this previous work in an attempt to provide a document which will be readily usable by mining engineers and others concerned with open pit equipment selection. This has resulted in a somewhat voluminous but hopefully more useful document.

The breadth of treatment, embracing several disciplines, e.g. industrial engineering, rock properties, electrical engineering, mining practices, etc., perhaps requires some explanation but no apology. The subject of the thesis is mining engineering, prepared in a Department concerned with mining engineering by a mining engineer. As such it must be broadly based if it is to be of value to mining engineers. Criticism by industry of an academic system which all too often produces graduates and undertakes research which are unsuited to industry's needs is endemic. Academics counter with the accusation that industry makes little effort to inform universities of its true requirements. Both viewpoints are unfortunately justified.

The author having experienced both industrial and academic life has set out to provide a document covering inter-disciplinary investigations which are intended to be of immediate use to mining engineers. The Joint Science Research Council and Social Science Research Council Committee ¹¹ has taken the view that research in breadth is as challenging and demanding as specialist research, and is in no way superficial or shallow, requires able people to pursue its aims and is a proper activity for a university to undertake. This cannot be more true than for the subject of mining engineering, which embraces an extremely wide spectrum, combining technical expertise, commercial and social considerations.

Acknowledgements

The work described in this thesis has been carried out over the past four years at the Royal School of Mines and is based on information collected and investigations made over that period plus approximately the previous twelve years spent in industry. The author is indebted to all his friends and professional colleagues for their patience and help. Acknowledgement is especially due to J. F. Weis and Dwight Wilcox of Marion Power Shovel, Inc., Marion, Ohio, U.S.A.; A. Major-Stevenson of Ruston-Bucyrus Ltd., Lincoln; A. Krumrey of Fried. Krupp GbmH Maschinen-und Stahlbau, Rheinhausen, West Germany; the late M. A. Neslin of General Electric, Schenectady, U.S.A.; F. Lamb of Caterpillar Tractor Co. Ltd., London; F. Weiser, Rheinische Braunkohlenwerke; S. C. Brealey and R. Dow of Powell Duffryn Technical Services Ltd., London.

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Finally thanks are due to my wife for the laborious business of typing the document.

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2. GENERAL - EXCAVATION

ASSEMBLING THE CRITICAL DATA

Before selection procedure can be commenced, all the necessary input data must be collected. Method and work study, materials testing, sampling and cost investigations are usually necessary. These preliminary studies are of prime importance since no matter how sophisticated the procedure, the results obtained will not exceed the quality of the input data i.e. the Critical Data. A considerable degree of overlap exists between the many types of excavating and loading equipment available. The final choice must be based on an unbiased analysis of their operations and costs. With accurate data a realistic result can be expected but mine planners should continuously record and process figures of equipment performance to confirm the accuracy of their selection or correct the original critical data on which selection is based.

The competence and "diggability" of the ground is of major importance. It depends on many factors, e.g.

1. The intact strength of the ground.
2. The competence of the ground as a whole.
3. The abrasive properties of the mineral constituents.
4. The bulk density of the ground, both in situ (bank) or broken (loose).
5. The moisture content.
6. The flow properties of the broken ground, possible stickiness, etc.
7. The degree of ground preparation contemplated; fragmentation.

Diggability

At present there is no generally accepted quantitative measure of diggability ²¹, but a fairly reliable indication can be obtained from:

- a) similar excavations in the area;
- b) the behaviour of ground excavated in trial pits, etc.
- c) physical tests on core samples recovered by drilling: some caution must be exercised as exposed beds may have very different characteristics from the cores - the most widely used tests are uniaxial and triaxial compressive strength, shear box and hardness tests, and a range of simple field tests to augment laboratory tests is available ²¹;
- d) refraction seismology tests. These are easily carried out but their interpretation can be difficult in disturbed geological conditions. Fig 2.1 shows the excavation possibilities without blasting. (See Appendix 2.A).

Generally, several of the above methods are used in conjunction with one another.

Digging Conditions The following brief descriptions are widely used.

- E - Easy digging, loose, free running material
e.g. sand, small gravel.
- M - Medium digging, partially consolidated
materials, e.g. clayey gravel, packed
earth, clay, anthracite, etc.

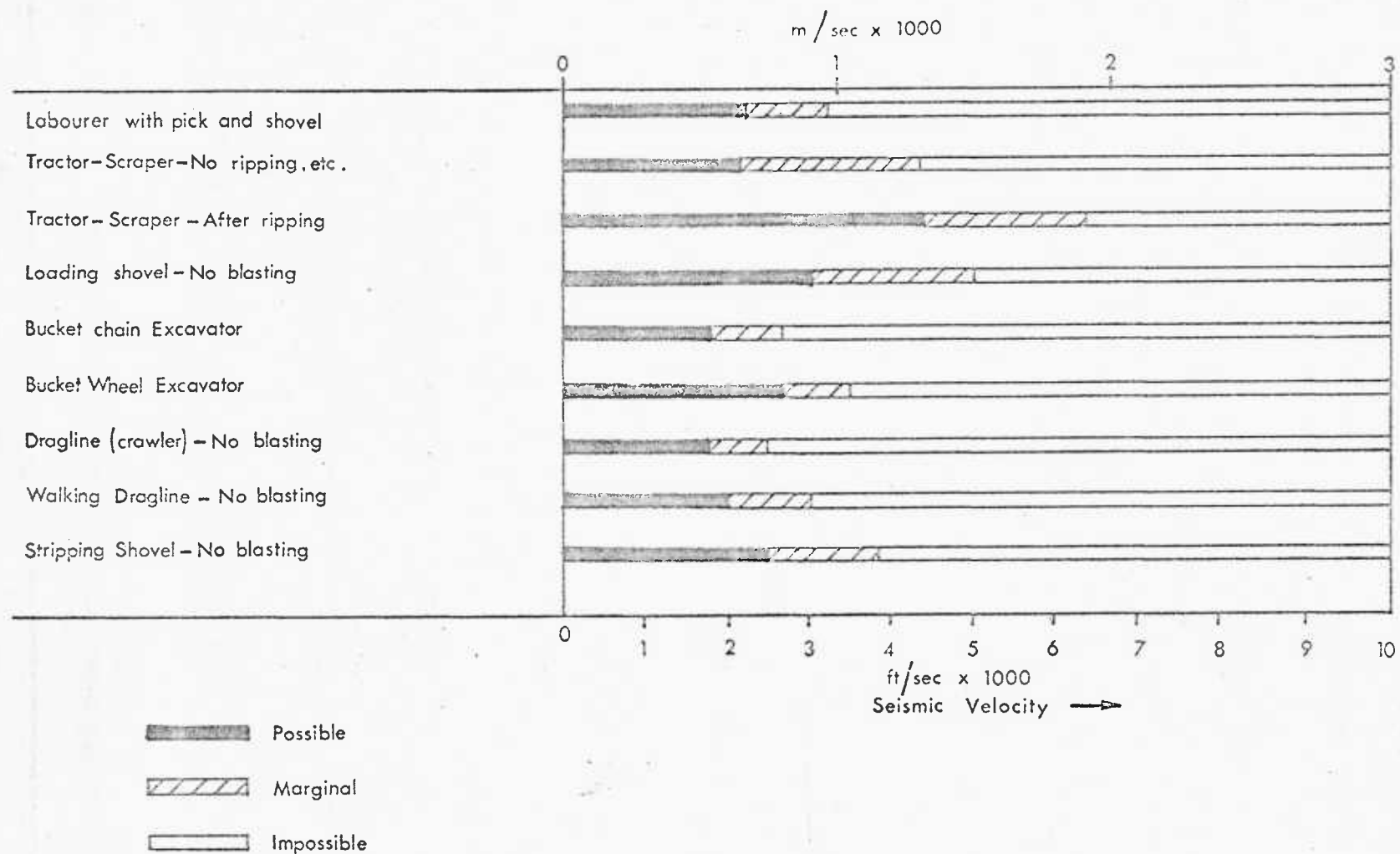


FIG.2.1 SEISMIC VELOCITY METHOD FOR DETERMINATION OF EXCAVATION POSSIBILITIES

- M-H - Medium-hard digging, e.g. well blasted limestones, heavy wet clay, weaker ores, gravel with large boulders, etc.
- H - Hard digging - materials that require heavy blasting and tough plastic clays, e.g. granite, strong limestone, taconite, strong ores, etc.

Other data

Other factors which have to be taken into account are the reach of the excavator, both horizontal and vertical, above and below grade, both digging and discharging, the bearing strength of the working bench, manoeuvrability, power supply, capacity, lead time for supply, etc. Where the mineral grade varies considerably and stockpiling is uneconomic or not possible because of the nature of the mineral, a larger number of smaller capacity excavators may be required to meet the mill feed requirements than if the grade is reasonably uniform or if stockpiling is possible.

CYCLIC AND CONTINUOUS EXCAVATORS

Numerous operational systems and machines are available and it is necessary to summarise the factors which influence their selection. Fig 2.2 the Transport System Classification Diagram ²², is based on the method of working, i.e. continuous or cyclic, and shows the transport possibilities.

Where it can be applied, continuous operation is preferable since it provides fuller plant utilisation and reduces margins added to plant capacity, as well as peaks of all forms, e.g. material flow, mechanical stresses and electrical maximum demands, and hence reduces overall costs. In open

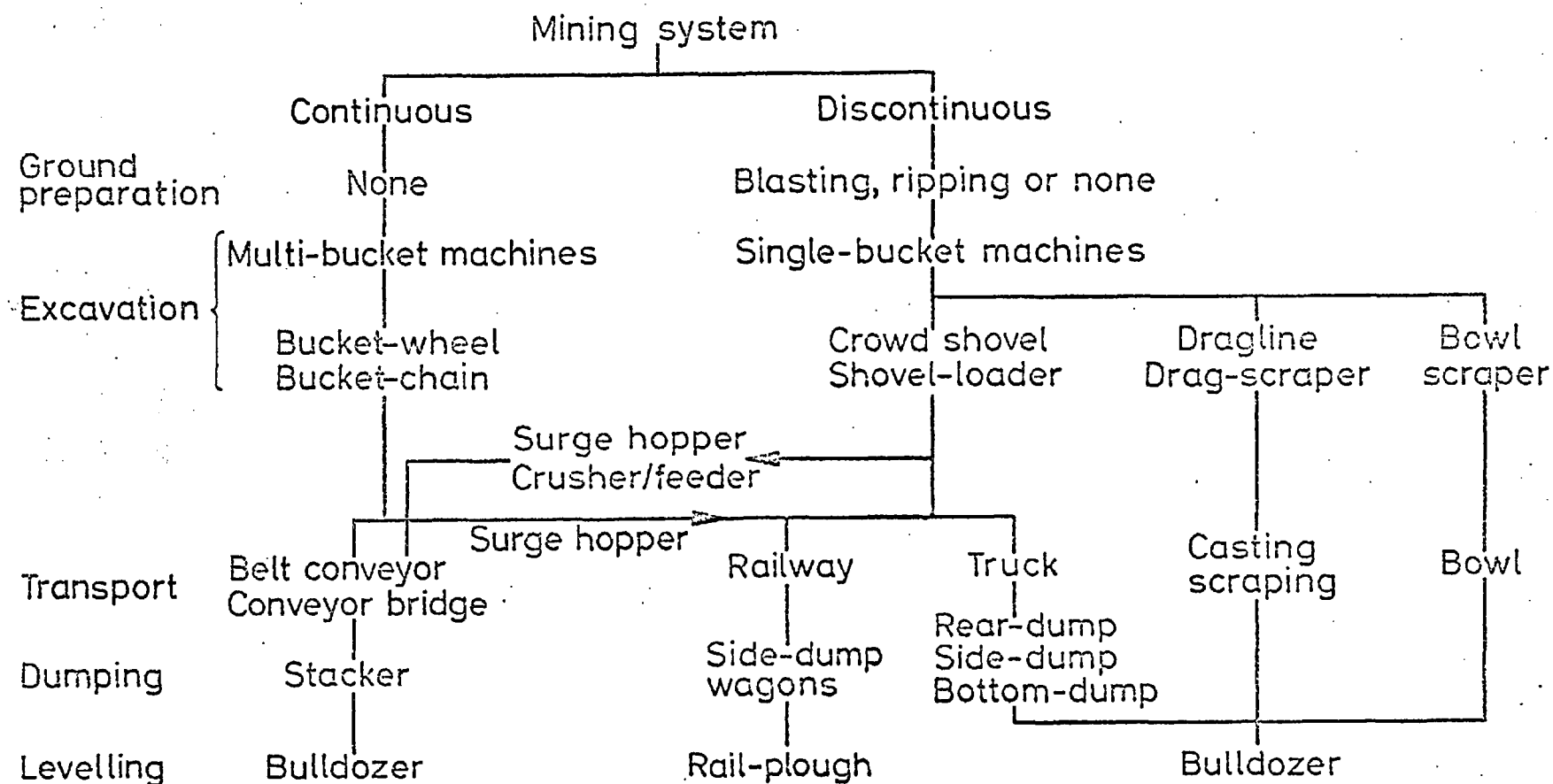


FIG.2.2 CLASSIFICATION OF OPERATIONAL SYSTEM

pit mining the continuous system is typified by the multi-bucket excavator (bucket wheel and bucket chain excavators). The cyclic system is represented by shovels, draglines, wheel loaders (front-end loaders), scrapers, rippers, bulldozers, etc.

DEFINITIONS

A number of definitions are common to a wide range of excavating and loading equipment and it is convenient to list these here. Other definitions specific to particular equipment are described in the appropriate chapters.

Bank The bank volume (or bank density) of ground is the "in situ" measure, as found in its natural, undisturbed condition. Mine planners invariably calculate volumes in "solid" or bank measures.

Loose Loose measure applies to material in the disturbed state after excavation when it occupies a greater volume than in bank.

Density The mass of material per unit volume, either bank or loose. Sometimes described as the "weight" in U.S.A. terminology. Very often inadequate density tests are carried out.

Several are available.

- a) Cylinder method
- b) Balloon method
- c) Oil method
- d) Sand Cone method
- e) Nuclear Density Moisture Gauge.

All the methods adopt the following procedure except for the nuclear method:

- 1) Remove sample from bank state
- 2) Determine volume of hole
- 3) Weigh sample
- 4) Calculate density (kg/m^3 bank or lb/yd^3 bank)

The nuclear density moisture gauge is usually restricted to soil like materials. A common radiation channel emits either neutrons or gamma rays into the material. In determining soil density, the number of gamma rays absorbed and back scattered by the particles of the material is INDIRECTLY proportional to the density of the soil. When measuring moisture content, the number of moderated neutrons reflected back to the detector after colliding with the hydrogen particles in the soil is DIRECTLY proportional to the moisture content of the soil. A number of density investigations are essential for statistical accuracy.

Load Weighing The most accurate method of determining the actual load carried is by weighing. For wheel mounted units this may be done by weighing one wheel or axle at a time using portable scales. Load cell or hydraulic scales of "wafer" construction used on level ground give reasonable accuracy. Machine mass is the sum of the individual wheel or axle weights. Again a number of loads must be weighed for statistical accuracy. The loose volume can be obtained by detailed survey and the loose density calculated.

$$\text{Swell Factor} = \frac{\text{Weight/unit volume (bank)}}{\text{Weight/unit volume (loose)}}$$

This is felt to be a preferable method of defining swell factor as it indicates the degree of swell. Most U.S.A. literature describes this figure as "percentage swell" and its reciprocal as "swell factor".

Some caution is needed in placing a value on swell factor - especially with less consolidated materials, which can reduce in volume with repeated handling. This must be taken into account when field measurements are being planned. Table 2.1 provides a guide for common materials.

TABLE 2.I
BULK DENSITY, SWELL FACTOR AND DIGGABILITY

FOR COMMON MATERIALS.*

Rock	Density t/m ³	(bank) lb/yd ³	Swell factor	Fill- ability ⁺	Diggability
Asbestos ore	1.9	3200	1.4	0.85	M
Basalt	2.95	5000	1.6	0.80	H
Bauxite	1.9	3200	1.35	0.90	M
Chalk	1.85	3100	1.3	0.90	M
Clay (dry)	1.4	2400	1.25	0.85	M
Clay (light)	1.65	2800	1.3	0.85	M
Clay (heavy)	2.1	3600	1.35	0.80	M-H
Clay and gravel (dry)	1.5	2500	1.3	0.85	M
Clay and gravel (wet)	1.8	3000	1.35	0.80	M-H
Coal (anthracite)	1.6	2700	1.35	0.9	M
Coal (bituminous)	1.25	2100	1.35	0.9	M
Coal (lignite)	1.0	1700	1.3	0.9	M
Copper ores (low-grade)	2.55	4300	1.5	0.85	M-H
Copper ores (high-grade)	3.2	5400	1.6	0.80	H
Earth (dry)	1.65	2800	1.3	0.95	E
Earth (wet)	2.0	3400	1.3	0.9	M
Granite	2.41	4000	1.55	0.8	H
Gravel (dry)	1.8	3000	1.25	1.0	E
Gravel (wet)	2.1	3600	1.25	1.0	E
Gypsum	2.8	4700	1.5	0.85	M-H
Ilmenite	3.2	5400	1.4	0.85	M
Iron ore 40% Fe	2.65	4500	1.4	0.8	M-H
Iron ore + 40% Fe	2.95	5000	1.45	0.8	M-H
Iron ore + 60% Fe	3.85	6500	1.55	0.75	H
Iron ore (taconite)	4.75	8000	1.65	0.75	H
Limestone (hard)	2.6	4400	1.6	0.80	M-H
Limestone (soft)	2.2	3700	1.5	0.85	M-H
Manganese ore	3.1	5200	1.45	0.85	M-H
Phosphate rock	2.0	3400	1.5	0.85	M-H
Sand (dry)	1.7	2900	1.15	1.00	E
Sand (wet)	2.0	3400	1.15	1.00	E
Sand and gravel (dry)	1.95	3300	1.15	1.00	E
Sand and gravel (wet)	2.25	3800	1.15	1.00	E
Sandstone (porous)	2.5	4200	1.6	0.8	M
Sandstone (cemented)	2.65	4500	1.6	0.8	M-H
Shales	2.35	4000	1.45	0.8	M-H

* These figures vary from location to location and tests should be made where possible. Allowance should be made for operation in wet conditions as density varies with moisture content.

⁺ Based on shovel dippers.

OPERATING EFFICIENCY

Although allowances are made for swing factor, fillability, "deadheading", etc. the theoretical production of an excavator invariably exceeds the actual production. The factor relating the two is defined as the

Operating Efficiency which is dependent on:

- a) human element
- b) job layout
- c) matching of loading and transport equipment
- d) number and duration of machine failures
- e) climate
- f) availability of spares and service facilities.

Operating efficiency is difficult to estimate except from records of similar operations and is often defined as the product AQ

where A is the Availability of the machine for work during the manned hours. It is generally defined as the mechanical availability during scheduled hours.

Q is the Job Operational Factor. An excavating or loading machine is almost always part of a system and is subject to delays due to management, supervision and labour deficiencies, job conditions, climate, etc. The machine capacity must be adjusted to compensate for these losses in production time.

Both A and O can be determined from plant records or by time studies and some mine planners prefer to continuously compile and up-date such data but there are obvious difficulties in a single organisation obtaining an adequate volume of data for statistical significance. Collection of these data is also difficult since many companies do not keep adequate records, whilst others use some form of logging device, e.g. Servis Recorder, but fail to usefully analyse the records provided. There are almost as many definitions of A as there are mines and O is not often fully understood. A simple statement using time worked and time standing but available for work does not adequately describe O, as shown in Fig 2.3, since there are times when the machine is working but certainly not at full capacity.

In an attempt to overcome the difficulties, use has been made of collected data dating back to 1954, by making time studies and snap studies or activity samples since 1965, by the issue of a questionnaire based on market research techniques to over 70 operators since 1969 and by the use of published data.

The collected data was obtained from a number of operations listed in Appendix 2.B. Some adjustment has been made to the earlier data to allow for the general improvement in mechanical availability which has occurred over the time period covered. The time studies and activity samples were also made at a number of the operations listed in Appendix 2.B. The methods used are also described in the Appendix 2.B.

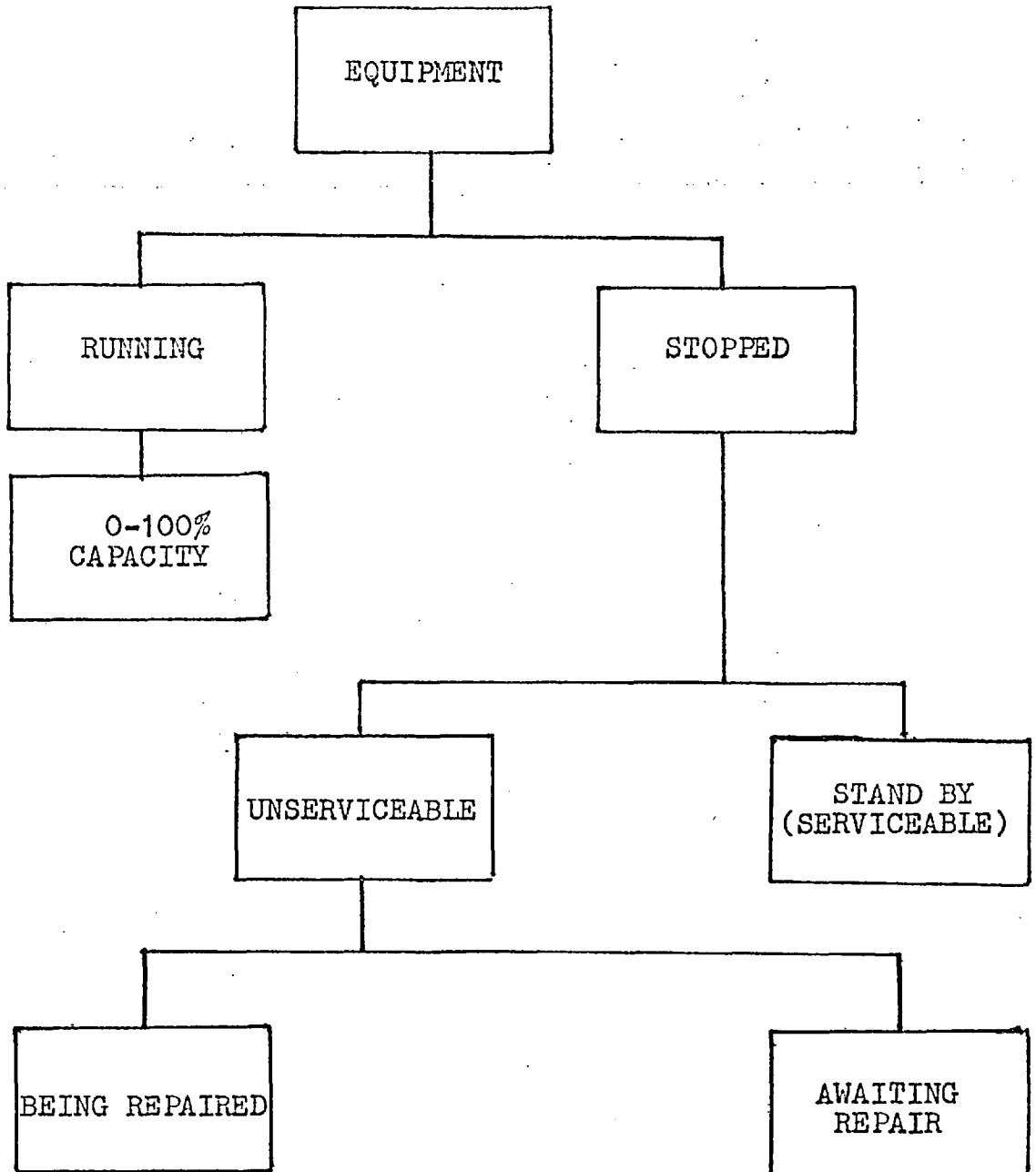


FIG.2.3 POSSIBLE CONDITIONS FOR AN ITEM OF EQUIPMENT

The questionnaire was partially completed by the operators listed in Appendix 2.C. Much of the information received was incomplete and was useful only because it could be used in conjunction with other information.

The published data used is listed in Appendix 2.D.

All the data was analysed by adopting the mode of the data collected, since the majority of the information was partially qualitative. The process of seeking as large a sample as possible, from a wide spectrum of industry is of course conducive to achieving some statistical significance. The processed data was then assembled in the following form, which is normally accepted for industrial engineering purposes, the results being:-

Management Conditions. If management and supervision are excellent with good workshops, planned maintenance programme, minimum delays in the systems, high availability, etc. the time spent producing will be high. Conversely, poor management and supervision due to incompetence, lack of continuity, poor economic or political conditions, etc. will reduce production time.

TABLE 2.II

MANAGEMENT CONDITIONS

	<u>Factor</u>
Good	0.95
Average	0.84
Poor	0.63

Job Conditions. Given that a climate is extreme, in a dusty environment, where the ground is dense and abrasive, if the quality of the labour is poor, performance will be adversely affected due to "poor" job conditions. If labour is excellent, the ground fragments well, flows freely, is not dense, climate and environment are not unpleasant then the time spent producing will be high.

TABLE 2.III
JOB CONDITIONS

	<u>Factor</u>
Good	0.89
Average	0.79
Poor	0.65

If these tables are combined to give the Operating Efficiency the following results are obtained

TABLE 2.IV
OPERATING EFFICIENCY

<u>Job</u> <u>Conditions</u>	<u>Management Conditions</u>		
	<u>Good</u>	<u>Average</u>	<u>Poor</u>
Good	0.845	0.747	0.561
Average	0.750	0.664	0.498
Poor	0.617	0.546	0.41

The limitations of this table are readily apparent. The coarse divisions of "Good, Average and Poor" are imprecise. The table is similar to a well accepted table (Table 2.V) used for industrial engineering purposes except that the collected values have a wider range. Because of possible bias in the results processed from the

collected data and because the use of Table 2.V is well established in earthmoving practice its use was advocated by the author ²³.

TABLE 2.V
OPERATING EFFICIENCY
(Ref. 24, 25, 26, 27, 28)

Job Conditions	Management Conditions			
	Excellent	Good	Fair	Poor
Excellent	0.83	0.80	0.77	0.70
Good	0.76	0.73	0.70	0.64
Fair	0.72	0.69	0.66	0.60
Poor	0.63	0.61	0.59	0.54

Following publication of Ref.23 several North American operators, notably in taconites and a shovel electric's manufacturer provided further information which partially substantiated Table 2.V but indicated that further processing was necessary to take account of lower values of Job Conditions. Most of the operators concerned felt that their job conditions were worse than poor and the term "severe" was used repeatedly. As Table 2.V is well accepted and does cover the majority of the range it is convenient to retain it with an addition for "severe" job conditions. The following table is submitted as a guide to planners where no equivalent experience and records are available.

TABLE 2.VI
OPERATING EFFICIENCIES OF OPEN PIT
EXCAVATING AND LOADING EQUIPMENT

Job Conditions	Management Conditions			
	Excellent	Good	Fair	Poor
Excellent	0.83	0.80	0.77	0.70
Good	0.76	0.73	0.70	0.64
Fair	0.72	0.69	0.66	0.60
Poor	0.63	0.61	0.57	0.52
Severe	0.53	0.45	0.40	0.37*

*estimated value.

A much more detailed analysis is available ²⁹ but is intuitive. No other table based on actual data has been found in an intensive literature search and it is believed that Table 2.VI represents a close factual approximation of real life values. The main limitation in the use of the table is in defining "Excellent, Good, Fair, Poor and Severe", but hopefully those concerned with the use of the table will have sufficient overall experience and judgement to be cautious where they have no direct experience of a machine or the conditions in which it is to work.

An attempt was made to use standard scheduling formulae but the data available did not lend itself to this method. An attempt was also made to differentiate between single, two and three shift operation but in fact the number of shifts worked had no identifiable effect on operational

efficiency. As an example large stripping machines in the U.S.A. working on three shifts covering 22.5 scheduled hours per day had operating efficiencies equal to the best operations working one shift per day. The same situation occurred for bucket wheel excavators scheduled for 19 hours per day in West Germany and Greece.

EFFECT OF ALTITUDE AND TEMPERATURE ²¹⁰

High elevations and temperatures reduce the density and therefore the oxygen content per unit volume of air. This reduces the efficiency of diesel engines normally rated at sea level and 15°C (60°F). Usually for altitudes above 300m (1000 ft) the following reductions are made

4 cycle naturally aspirated

3% in power per 300m (1000 ft) above the first 900m (3000 ft)

1% in power for each 5.5°C (10°F) temperature rise above 15°C (60°F)

2 cycle

1% in power per 300m (1000 ft) above the first 900m (3000 ft)

1% in power for each 5.5°C (10°F) temperature rise above 15°C (60°F)

An additional 1% increase in power should be added for each 5.5°C (10°F) temperature drop below 15°C (60°F).

For supercharged engines the first 1500m (5000 ft) of elevation instead of the first 300m (1000 ft) can be neglected for correcting the power available.

REFERENCES

- 2.1 FRANKLIN, J.A., BROCH, E. and WALTON, G. "Logging the mechanical character of rock". Trans. Instn. Min. Metall. (Sect. A; Min. Industry) 80, 1971, A1-9.
- 2.2 BREALEY, S.C., and ATKINSON, T. "Opencast Mining" The Mining Engineer Dec. 1968, pp 147-163.
- 2.3 ATKINSON, T. "Selection of open pit excavating and loading equipment". Trans/Sect.A. IMM, Vol. 80, 1971, pp A101-129.
- 2.4 Caterpillar Tractor Co. "Caterpillar Performance Handbook". Edition 1, Dec. 1970, pp 19.5.
- 2.5 Power Shovel and Crane Association. "Power Cranes-Shovels-Draglines" Technical Bulletin No.4 1953.
- 2.6 WEIS, J.F. "Application Data". Marion Power Shovel Co. Inc., Marion, Ohio, 1970.
- 2.7 WEIS, J.F. "Large Machines Application Data Book" Marion Power Shovel Co. Inc., Marion, Ohio 1969.
- 2.8 MAYNARD, H.F. "Industrial Engineering Handbook". 2nd Edition, McGraw-Hill, New York, 1963.
- 2.9 DREVDAHL, E.R. "Estimation of shovel and dragline outputs for Systems Analysis". Symp. Surf. Min. Practices 1960. Krumlauf, H.E. Ed. Tucson, Ariz. (Univ. Ariz. 1960), pp 94-107.
- 2.10 DREVDAHL, E.R. "Profitable use of excavating machinery" Technical Publications Desert Laboratories Inc., Tucson, Ariz. 1961.

APPENDIX 2.A

REFRACTION SEISMOLOGY TESTING*

This well-known, geophysical method has the great advantage of being quantitative and makes use of the principle that seismic (sound) waves travel through subsurface materials at different speeds, dependent upon the hardness, degree of brokenness and the orientation, openness, filling, continuity and surface texture of the planes of the breaks. Seismic waves travel through loose top soil at about only 300m/sec (1,000ft/sec) but at 6,000m/sec (20,000ft/sec) through a hard, intact rock. The technique generally provides a much quicker indication and is more economic than core drilling but of course the two are often used in conjunction with each other.

Seismographs

There are a number of seismographs available, the most commonly used being the R-150 Terra-Scout Portable Refraction Seismograph, the Soiltest MD₃ and the MD₁ Engineering Seismograph of the Geophysical Specialities Co. The seismograph consists of a recorder connected to a geophone which is set into the surface. A steel plate which is also connected to the recorder is placed a pre-determined distance, usually about 3m (10ft), from the geophone and struck with a sledge-hammer. The procedure is repeated at the same distance intervals along a straight line from the geophone (Fig 2A1).

*From "Ground Preparation by Ripping in Open Pit Mining" by T. Atkinson, Min. Mag. Vol. 122, No.6, June 1970. For a more detailed study see "Applied Geophysics for Engineers and Geologists" by D.H. Griffiths and R.F. King, Pergamon 1969, and Gough, D.I. "A new instrument for Seismic Exploration at very short ranges". Geophysics, 17, pp 311-333, 1952.

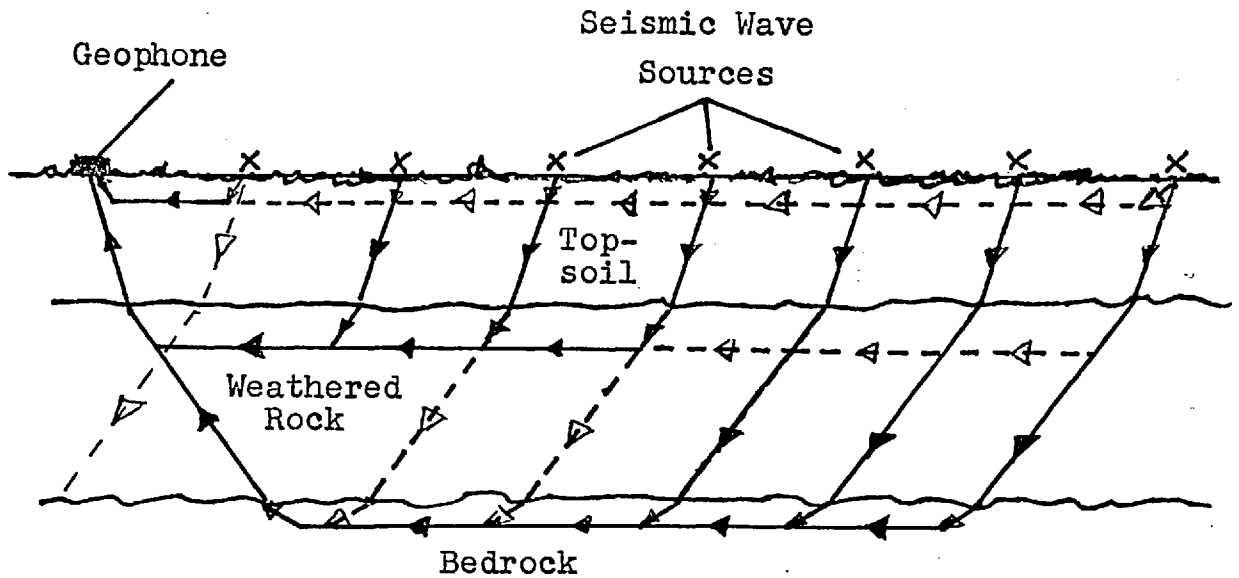


FIG.2A1 MEASUREMENT OF SEISMIC WAVE VELOCITIES

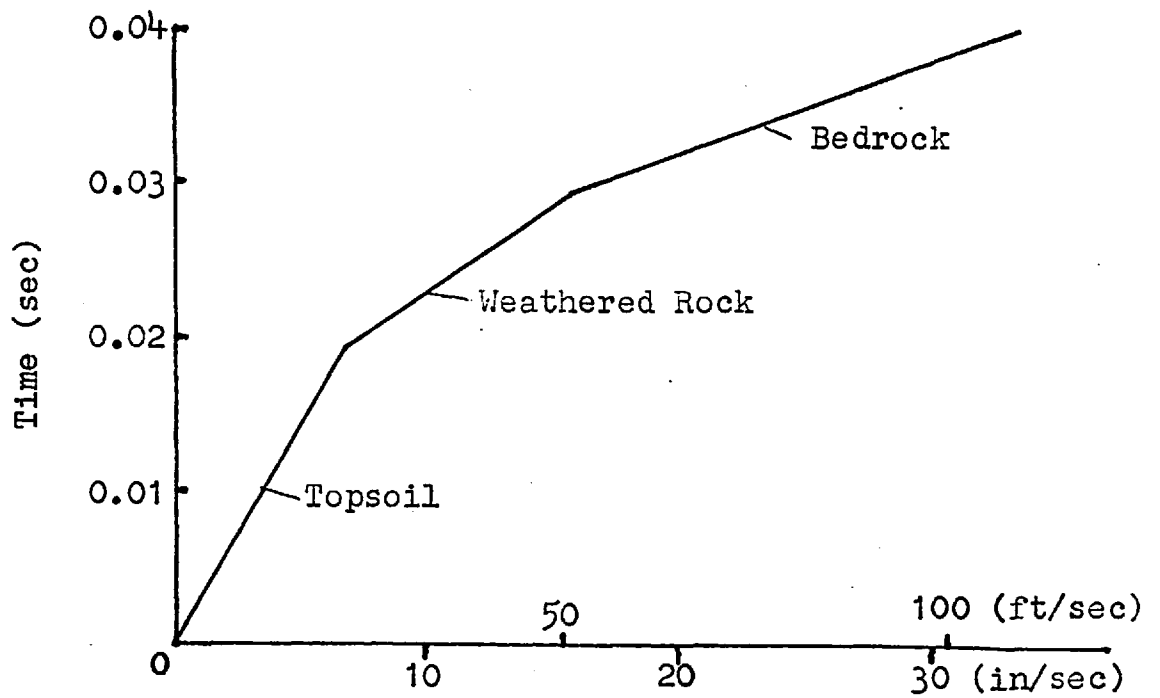


FIG.2A2 COMPARISON OF SEISMIC WAVE VELOCITIES

From the first position the first wave to reach the geophone travels through the topsoil. Other waves from the first blow travel down to the strata below and are reflected, reaching the geophone after the wave which travelled the much shorter distance through the topsoil.

As the distance is increased to 6m-9m (20ft-30ft), however, the first waves detected by the geophone are those which travel through the surface rock (Fig 2A1), because of its higher seismic velocity characteristics. After 15m (50ft) the first waves to arrive at the geophone are those which travel through the lower, harder and less broken bedrock (Fig 2A1).

By plotting the recorder results as shown in Fig 2A2 it is possible to determine:

- (a) the seismic velocity characteristics of the subsurface materials, and
- (b) the depth of the individual strata; the slope of the graph indicating velocity and the point at which the slope changes, representing the depth of that material below the surface.

The interpretation of the results of seismograph tests is a relatively simple procedure and can provide an extremely accurate guide to diggability. In certain conditions however, e.g. where a bed of hard rock with a high seismic velocity characteristic overlies a weaker bed of low seismic velocity characteristic, or in disturbed geological conditions or where the rocks have poor reflection and refraction properties; there can be difficulties in interpreting the recorder results.

APPENDIX 2.B

OPERATIONS AT WHICH METHOD STUDY TECHNIQUES WERE APPLIED.

1. Snowy Mountains Hydro Electric Authority, Australia.
2. State Electricity Commission of Victoria, Australia, Yallorn Operations.
3. Neyveli Lignite Corporation (Private) Ltd., Neyveli, Madras, S. India.
4. Dai Han Coal Corpn. Korea, Sam Chok open pit operations.
5. Kelmac Ltd., Carnforth, Lancs.
6. Penfold Quarries Ltd., South Wales.
7. Abu Zabal Basalt Quarries, General Organisation for Exploitation of Mineral Wealth, Cairo, U.A.R.
8. Ptolemias Mining and Industrial Corpn., Greece.
9. National Coal Board Opencast Executive, various sites.
10. British Steel Corporation, various sites.
11. British Gypsum Ltd., Grantham.
12. Fundidora Steel Corpn., Monteray, Mexico.
13. Mission Pit, ASARCO, USA.
14. Bingham Canyon Mine, Kennecott Copper Corpn., Bingham, Utah.
15. Sherman Mine, Tergami, Ont., Canada.
16. Broken Aro Mine, Peabody Coal Co., Ohio.
17. Warrior Mine, Peabody Coal Co., Ohio.
18. Thompson Aggregates, Thompson, Man., Canada.
19. Pipe Mine, Thompson, Canada.
20. Konin Lignite Mine, Poland.
21. Fortuna Nord, R.B.W., West Germany.
22. Frenchen Mine, R.B.W., West Germany.
23. L'Ouenza Iron Ore Mine, Algeria.

CONVENTIONAL ACTIVITY TIME STUDY

A time study is "a work measurement technique for recording the times and rates of working for the elements of a specified job carried out under specified conditions, and, for analysing the data so as to obtain the time necessary for carrying out the job at a defined level of performance" (B.S. 34001). An activity time study is the application of time study to machine activities and is carried out to determine the Time at Standard per Cycle where:

Standard Performance (Machine) - "The rate of output achieved, in specified conditions with due regard to safety, by a properly maintained and correctly operated machine when all allied manual work is carried out at Standard Performance".

Standard Performance (Man) - "The rate of output that qualified workers will naturally achieve without over-exertion as an average over the working day or shift provided they know and adhere to the specified method and provided they are motivated to apply themselves to their work". (B.S. 34001)

The Technique of Activity Time Study

The work of the machine or job cycle is broken down into elements and timed with a stop watch. Each element is timed on a number of occasions. The object is to determine a representative value. Elements which include a delay or a reason for a reduction in output are excluded. The mean times for each element are calculated and added together to give the mean cycle time.

Observed times for each element are recorded on a time study sheet. The times are then transferred to the time study abstract sheet, separating element times which include a delay or loss of output.

Representative Values

It is essential that sufficient observations be taken to give a representative value for each element and this proved to be the most difficult part of the study since the purpose of the visit to many mines was not primarily to conduct an activity time study. Where possible the number of observations was determined from a method based on the following references:

Ref. No.

2.B.101 MUNRO, H. "The accuracy of work values".

Productivity Measurement Review No.17, May, 1959

2.B.102 MUNRO, H. "The accuracy of work values". Work Study and Industrial Engineering, Sept. 1958.

2.B.103 HANSEN, B.L. "A graphic method for finding the required number of time study readings". Journal of Industrial Engineering, May-June 1957.

When a number of basic times obtained from observations of an element are examined, it will be found generally that they vary. Apart from variations caused by rating errors the principal causes of this effect are slight changes in method or conditions which the method study engineer cannot define or measure and to which he cannot assign a compensation factor. The conventional method of calculating a single time from a number of differing basic times is to find their arithmetic mean, but the question arises of how many values are necessary in order to have confidence that the mean does represent a fair value and is not reduced or inflated

by the extreme reading. A fair value, by convention, is a value that has a 95% probability of being within $\pm 5\%$ of the mean of an infinitely large number of readings.

Consider the following two groups of times.

<u>Group 1</u>	<u>Group 2</u>
(minutes)	(minutes)
0.50	0.90
1.00	1.00
<u>1.50</u>	<u>1.10</u>
Mean 1.00 minutes	Mean 1.00 minutes

Each group has the same mean value, but it can be seen that there would be less confidence in Group 1, the range of which is 1.00 minutes than in Group 2 the range of which is 0.20 minutes.

This leads to the conclusion that the ratio 'range/mean' can be used to give an indication of how many observations are required and that the higher the range/mean ratio the greater will be the number of observations that are required for the result to be within the prescribed limits of accuracy.

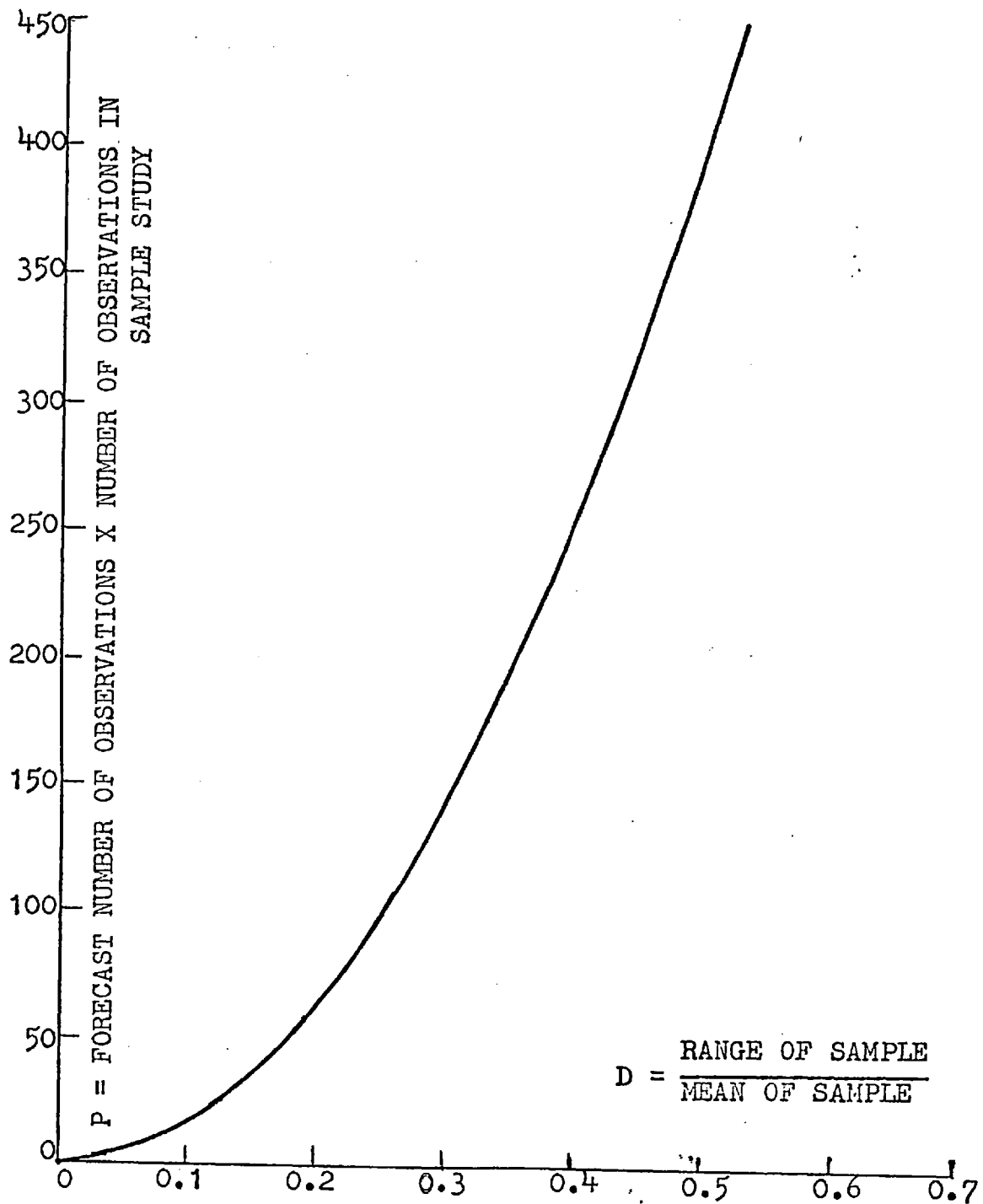
If this ratio is applied after a study, it is possible to check the validity of the result, but this may show that the study has been continued for too long, involving unnecessary expense. Therefore it is more profitable to apply the range/mean ratio after a preliminary study of a few observations of the element, and then to forecast the number of observations that will be needed to obtain the required degree of accuracy, provided it is possible to check that this degree of accuracy has been achieved after the completion of the full time study.

Although it has been shown that the range/mean ratio and the number of observations required are related, this relationship has not been defined and to do so would be extremely laborious. To simplify this a curve can be drawn using statistical techniques from which the forecast number of observations required to achieve an accuracy of $\pm 5\%$ may be derived. The curve is shown in Fig 2B1.

The number of observations in the sample study must influence the forecast figure since a study of ten observations will be more reliable than a study of, say, three observations. A table of compensating factors L for the number of observations in the sample study is shown in Fig. 2B1.

The procedure for using the curve and the compensating factors is as follows:-

1. Make a short study of between four and ten basic times of the element. Let this number of observations be n .
2. Calculate the arithmetical mean of these basic times. Let the mean be M .
3. Calculate the range of the basic times, that is the difference between the highest and lowest values. Let the range be R .
4. Calculate the ratio of $\frac{\text{Range}}{\text{Mean}}$ and let it be equal to D , i.e. $D = \frac{R}{M}$
5. Let P be the product of the forecast number of observations to achieve $\pm 5\%$ accuracy and the number of observations in the sample study.
6. From the graph of $D \sim P$ (Fig. 2B1), read off P for the calculated value of D .



n	3	4	5	6	8	10
L	4.00	2.15	1.70	1.5	1.4	1.4

FIG.2.B.1 P - D CURVE

7. Let S equal the forecast number of observations to give the required accuracy of $\pm 5\%$ without compensating for the number of observations in the sample study.

$$\text{Then } S = \frac{P}{n}$$

8. From the Table in Fig 2B1 find the factor L for the appropriate value n. Then $N = SL$ where N is the number of observations which will give the required accuracy of $\pm 5\%$ or better.
9. Increase N to the next highest multiple of 10.

Example

- 1 A preliminary study is made of an element and four basic times obtained. These are:-

0.38 Basic minutes

0.30 " "

0.33 " "

0.35 " "

- 2 The mean basic time M is the sum of the sample basic times divided by four, then,

$$M = \frac{1.36}{4} = 0.34 \text{ basic minutes}$$

- 3 The range R = $0.38 - 0.30 = 0.08$

4 $\frac{\text{Range}}{\text{Mean}}$ D = $\frac{0.08}{0.34} = 0.235$

- 5 From Fig. 2B1, when D = 0.235, P = 88

- 6 The forecast number of observations S is given by:-

$$S = \frac{88}{4} = 22$$

- 7 From Fig.2B1 when $n = 4$
the compensating factor $L = 2.15$
Then $N = SL$
 $N = 22 \times 2.15$
 $= 47.3$ observations

- 8 Increase N to the next higher multiple of 10
 $= 50$ observations.

The use of Fig.2B1 cannot correct for timing or rating errors or assessing relaxation allowances. It assumes that the variation from their mean of all possible basic times is a normal distribution and that the observations made, both during the preliminary study and the main study work, are representative of this distribution.

By the very nature of this exercise observations were therefore made in a random way this being beneficial since a number of short studies are better than a smaller number of long studies for the same total number of observations.

For example, if the preliminary study of four to ten observations is made on one operator, the range is likely to be smaller than if the preliminary study is made on a number of operators. The predicted number of observations to be taken in the main study is therefore smaller than in fact is required and when a check is made additional studies will be necessary to complete the required number of observations.

By means of the previously described procedure, N , the total number of observations required was predicted.

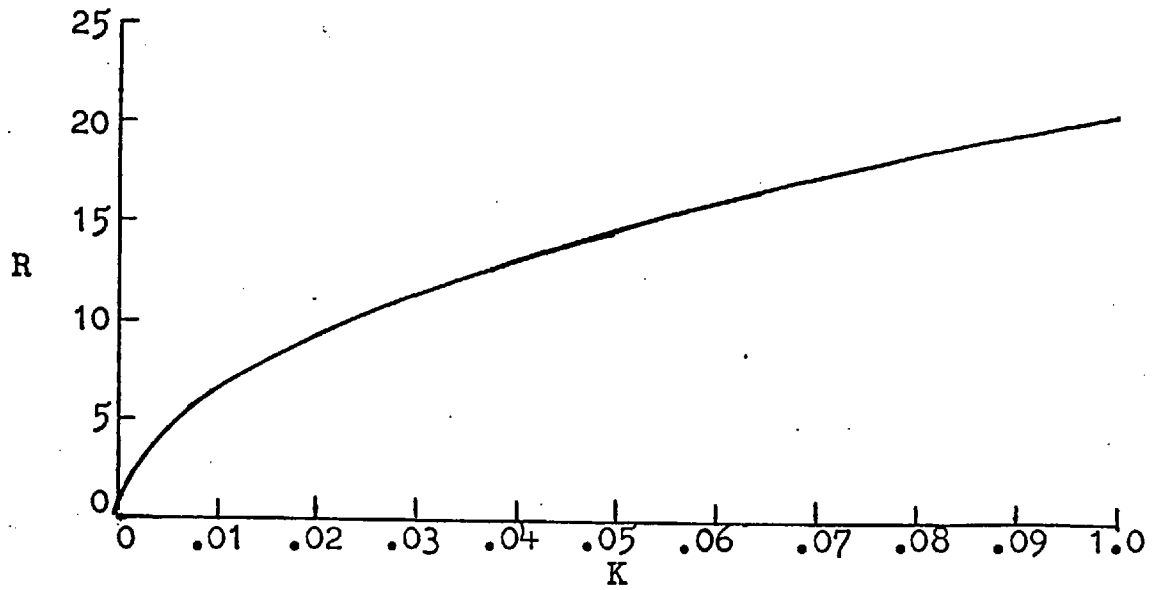
In practice N is subject to error and will not give the stated accuracy. It is therefore necessary to estimate the accuracy achieved. The best method is to calculate the Standard Deviation of the observations made in the main study, but this involves considerable arithmetic calculation which is not normally justified. A simpler method which provides accurate results for a sample size over 50 is as follows:

1. Divide the observations into groups of 10.
2. Calculate the range of each group.
3. Calculate the mean of these ranges.
4. Calculate the mean of the whole sample.

5. Calculate $\frac{\text{Mean Range}}{\text{Sample Range}}$

6. Calculate
$$K = \frac{\left(\frac{\text{Mean Range}}{\text{Sample Range}} \right)^2}{N}$$

7. Read off the relative error from Fig.2B2.



R = RELATIVE ERROR
(95% PROBABILITY)

$$\left[\text{ie } \frac{\text{TRUE MEAN-SAMPLE MEAN}}{\text{TRUE MEAN}} \right]$$

FIG.2 B 2 ACCURACY OF TIME STUDIES

ACTIVITY SAMPLING

The technique of Activity Sampling is well known and was probably due to L.H.C. Tippet ("A snap-reading method of making time studies of machines and operatives in factories". Journal of Textile Institute, No.36, 1935, pp 51-70) and was developed by R.L. Morrow ("Time Study and Motion Economy". Ronald Press 1946. Second Edition 1957) for general use, with particular reference to the determination of delays incidental to the performance of work. Since then other literature has appeared and many applications of the technique have been described. The technique is known variously as "activity sampling", "work sampling", "random observations", "activity ratio" and "ratio delay". The general purpose of activity sampling is to determine the percentage of the working period spent on a series of specified activities e.g. machines "running", "idle" or "unserviceable". It can be defined as:

A technique in which a large number of instantaneous observations are made over a period of time of a group of machines, processes or workers. Each observation records what is happening at that instant and the percentage of observations recorded for a particular activity or delay is a measure of the percentage of time during which that activity or delay occurs.

Apart from analysing an activity (as used by the author for this exercise) the technique can be used:

1. To indicate when improvements in methods and equipment would be beneficial
2. To analyse sources of delay and causes of stopping in large installations.
3. To investigate causes of reduced performance.

Several of the exercises were undertaken while the author was employed by P.D.T.S. and the data collected used for this thesis.

Activity sampling requires advance preparation of a time schedule for observations or the start of observation "tours". No rating is carried out when observations are made, so no conversion to basic times is necessary, the recording of activities being ultimately expressed as a percentage of the whole working period.

Derivation of Formulae for Accuracy and Number of Observations Required in Activity Sampling Study

Assuming the distribution is 'normal', a 95% probability is attained when limits of $\pm$ two standard deviations are imposed. This is illustrated in Fig.2B3.

For a binomial distribution

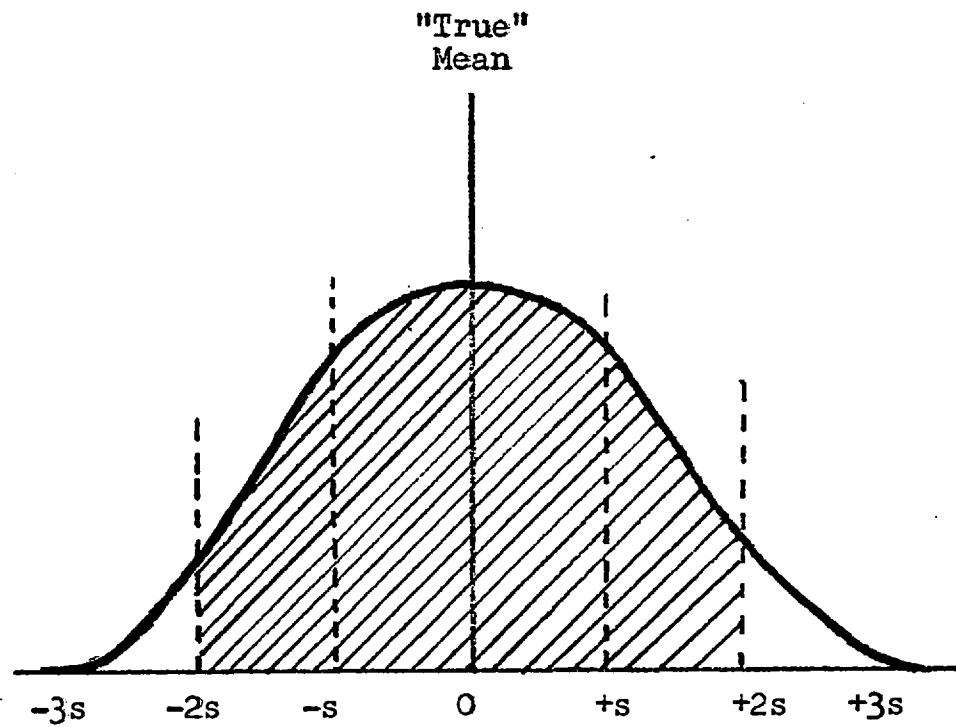
$$s \text{ (Standard deviation of } N p) = \sqrt{Npq}$$

Where N = Number of observations

p = Probability of an activity occurring

q = Probability of an activity not occurring

$$\begin{aligned} \therefore s.p \text{ (Standard deviation of } p) &= \frac{\sqrt{Npq}}{N} \\ &= \sqrt{\frac{pq}{N}} \end{aligned}$$



Area of shaded portion - 95% probability
 s = One standard deviation

FIG. 2B3 - NORMAL DISTRIBUTION- 95% PROBABILITY AREA

Using 95% probability limits

$$E_A \text{ (Absolute error of } p) = \pm 2 \sqrt{\frac{p q}{N}}$$

If p and q are expressed as percentages, the value found for E_A will also be a percentage.

$$\text{Also } q = 100 - p$$

$$\therefore E_A \% = \pm 2 \sqrt{\frac{p(100 - p)}{N}} \quad \dots (2B1)$$

and $E_R \%$ (Relative error of p)

$$= \pm \frac{2 \times 100}{p} \sqrt{\frac{p(100 - p)}{N}} \quad \dots (2B2)$$

Number of Observations Required

The number of observations required to achieve any accuracy may be calculated from the formulae (2B1) and (2B2)

Therefore

$$N = \frac{4p(100 - p)}{E_A^2} \quad \dots (2B3)$$

Therefore

$$N = \frac{40,000(100 - p)}{p \times E_R^2} \quad \dots (2B4)$$

Calculations from these formulae will only be valid when the following conditions apply:-

1. The population is large and only two possible circumstances are considered at one time, i.e. the possibility of an activity occurring or of it not occurring.
2. The observations must be taken at random. Each item must have an equal probability of being selected from all the possible items in the population.

Example

400 observations have been taken on three activities which take about (a) 50%, (b) 30%, (c) 20% of the total study period. Determine the absolute error (E_A) and the relative error (E_R) in each case.

(a) 50% Activity

From (2B1)

$$E_A\% = \pm 2 \sqrt{\frac{50(100) - 50}{400}} = \pm 5\%$$

From (2B2)

$$E_R\% = \pm \frac{2 \times 100}{50} \sqrt{\frac{50(100 - 50)}{400}} = \pm 10\%$$

(b) 30% Activity

From (2B1)

$$E_A\% = \pm 2 \sqrt{\frac{30(100 - 30)}{400}} = \pm 4.6\%$$

From (2B2)

$$E_R\% = \pm \frac{2 \times 100}{30} \sqrt{\frac{30(100 - 30)}{400}} = \pm 15.3\%$$

(c) 20% Activity

From (2B1)

$$E_A\% = \pm 2 \sqrt{\frac{20(100 - 20)}{400}} = \pm 4\%$$

From (2B2)

$$E_R\% = \pm \frac{2 \times 100}{20} \sqrt{\frac{20(100 - 20)}{400}} = \pm 20\%$$

Example

A working period is divided between three activities taking (a) 60%, (b) 25%, (c) 15% of the whole period. Find in each case the number of observations to be taken during the whole working period to achieve (1) an absolute error of $\pm 5\%$. (2) a relative error of $\pm 5\%$.

(a) 60% Activity

From (2B3)

$$N = \frac{4 \times 60(100 - 60)}{5^2} = 384 \text{ observations. } (E_A = \pm 5\%)$$

From (2B4)

$$N = \frac{40,000(100 - 60)}{60 \times 5^2} = 1067 \text{ observations } (E_R = \pm 5\%)$$

(b) 25% Activity

From (2B3)

$$N = \frac{4 \times 25(100 - 25)}{5^2} = 300 \text{ observations } (E_A = \pm 5\%)$$

From (2B4)

$$N = \frac{40,000 (100 - 25)}{25 \times 5^2} = 4800 \text{ observations } (E_R = \pm 5\%)$$

(c) 15% Activity

From (2B3)

$$N = \frac{4 \times 15(100 - 15)}{5^2} = 204 \text{ observations } (E_A = \pm 5\%)$$

From (2B4)

$$N = \frac{40,000(100 - 15)}{15 \times 5^2} = 9067 \text{ observations } (E_R = \pm 5\%)$$

Table 2BI shows the number of observations required for absolute and relative errors of $\pm 5\%$ for activities taking between 5% and 95% of the working period.

Before making the calculation of the number of observations required, it must be decided whether the results will describe the activities of the whole system or whether the activities of each item of equipment is wanted.

TABLE 2BI

NUMBER OF OBSERVATIONS FOR 5% ABSOLUTE AND
RELATIVE ERRORS

p as % of whole working period	N for $E_A = \pm 5\%$	N for $E_R = \pm 5\%$
5	76	30,400
10	144	14,400
15	204	9,067
20	256	6,400
25	300	4,800
30	336	3,734
35	364	2,971
40	384	2,400
45	396	1,955
50	400	1,600
55	396	1,309
60	384	1,067
65	364	862
70	336	686
75	300	534
80	256	400
85	204	283
90	144	178
95	76	85

Representative Period for Study

The next step is to decide over what period the observations are to be collected. This will depend to some extent on the nature of the equipment being studied, and the study period will have to be long enough to take in any variations in equipment performance that are likely to occur.

As the study proceeds two types of control chart should be constructed to indicate the progress and the performance of the plant.

The first chart is used, as the study progresses, to determine how the values of 'p' or the proportions of time spent on particular activities are levelling out. (See Table 2BII)

TABLE 2BII
CUMULATIVE PLOT CONTROL CHART -
ACTIVITY "RUNNING LOADED"

DAY	RUNNING (loaded) ON THAT DAY	POSSIBLE ON THAT DAY	RUNNING (loaded) TO DATE	POSSIBLE TO DATE	% ACTIVITY
	A	B	CUMULATIVE A	CUMULATIVE B	$\frac{\text{CUM A}}{\text{CUM B}} \times 100$
1st	21	28	21	28	75.0
2nd	24	28	45	56	80.4
3rd	19	28	64	84	76.3
4th	16	28	80	112	71.5
5th	17	28	97	140	69.3
6th	19	28	116	168	69.0
7th	20	28	136	196	69.4
8th	22	28	158	224	70.5
9th	19	28	177	252	70.3
10th	23	28	200	280	71.4
11th	19	28	219	308	71.1
12th	19	28	238	336	70.9
13th	17	28	255	364	70.0
14th	23	28	278	392	71.0
15th	16	28	294	420	70.0

The second chart, similar to the quality control chart, indicates whether on any particular day the plant has produced a variation in performance due to an assignable cause, as distinct from chance variations due to the study method employed. Using this chart it is possible to determine whether extra days of study are required to compensate for any particular days when conditions have been abnormal.

Procedure for construction of second control chart:

1. Estimate p for "equipment running loaded" as soon as possible after start of study, from Cumulative Control Table 2BII.

$$p = \frac{\text{actual observations}}{\text{possible observations}}$$

$$\text{Say after 5th day} = \frac{97}{140} = 69.3\%$$

2. Set "Upper and Lower Control Limits" based on $p = 69.3\%$ and 28 possible observations per day.

$$\begin{aligned}\text{Control Limits are: } &= p \pm 2 \sqrt{\frac{p(100 - p)}{N}} \\ &= 69.3\% \pm 2 \sqrt{\frac{69.3 \times 30.7}{28}} \\ &= 86.7\% \text{ and } 51.9\%\end{aligned}$$

where

N = number of daily observations

3. Construct daily chart for, p values, by plotting percentage values of p for "equipment running loaded" for each day of study, e.g. 5th Day (See Table 2BII) where on 17 out of 28 occasions the plant was running loaded.

$$\text{i.e. } p = \frac{17}{28} = 60.7\%$$

These values should lie within the control limits (86.7% and 51.9%) unless there is an assignable cause to the variation within the operation itself.

APPENDIX 2.C

LIST OF MINES WHICH PROVIDED INFORMATION

ON FACTORS A AND O

1. State Electricity Commission of Victoria, Australia, Latrobe Valley Brown Coal Operations.
2. Hammersley Iron Pty. Ltd., Mount Tom Price Mine.
3. Mount Lyell, Tasmania, Australia.
4. Palabora, South Africa.
5. Demerara Bauxite Ltd. (Now Gyana Bauxite Ltd.)
6. N.C.D.C., India, Ranchi Area Mines.
7. N.C.C.M. Ltd., Zambia.
8. R.C.M. Ltd., Zambia
9. Mount Goldsworthy Mine, Western Australia.
10. Mount Newman Mine, Western Australia.
11. Twin Buttes Mine, U.S.A.
12. Boliden's Aitik Copper Mine, Sweden.
13. Phelps Dodge, various U.S.A. mines.
14. Sherman Mine, Tergami, Ontario, Canada.
15. Cleveland Cliffs Iron Co., Empire Mine, Mich., U.S.A.
16. A. Teichert and Sons Inc., Aggregates Divsn., Sacramento, U.S.A.
17. LAMCO, Liberia.
18. Granges AB, Sweden.
19. Hanna Mining Co., U.S.A. various mines.
20. Highmont Mining Corp., B.C., Canada.
21. Inspiration Consolidated Copper Co., Ariz., U.S.A.
22. A.O. Nigeria Ltd., various mines
23. International Nickel Co., of Canada, various mines.
24. Iron Ore Company of Canada, various mines.

25. Sherbro Minerals Ltd., Sierre Leone. (Now Sierre Rutile Ltd)
26. Reynolds Jamaica Mines Ltd., Jamaica.
27. Reynolds Mining Corp., Ark., U.S.A.
28. Savage River Mines, Tasmania, Australia.
29. Pipe Mine, Thompson, Canada.
30. Mount Isa Mines Ltd., Australia.
31. Kaiser Aluminium & Chemical Corp., Jamaica.
32. Bingham Canyon Mine, Kennecott Copper Corp., U.S.A.
33. Kilembe Mines, Uganda.
34. Denison Ltd., Tasmania, Australia.
35. New York and Honduras Rosario Mining Co., U.S.A.
36. Bougainville Copper Pty., Ltd., Bougainville.
37. Brenda Mines, Peachville, Canada.
38. Biliton International Metals, Brazil Operations
(Now National Lead)
39. Brunswick Mining and Smelting Corp., Canada.
40. Canadian Johns-Manville Co. Ltd., Canada.
41. Gaspe Copper Mines Ltd., Canada.
42. Williamson Diamonds Ltd., Tanzania.
43. Tsumeb Corp. Ltd., S.W. Africa.
44. Union Carbide Corp., U.S.A.
45. Highmont Mining Corp., Canada.
46. Lornex Copper, Canada.

APPENDIX 2.D

SOURCES OF PUBLISHED DATA USED TO DETERMINE
OPERATING EFFICIENCY

1. American Metal Climax Co.
2. Marion Power Shovel Co. Inc.
3. Bucyrus-Erie Co.
4. Ruston-Bucyrus Ltd.
5. Harnischfeger Corp.
6. Caterpillar Tractor Co.
7. Clark International Ltd.
8. Orenstein-Koppel LMG GmbH.
9. Fried. Krupps GmbH, Rheinhausen.
10. LAMCO.
11. Barber-Green Co.
12. International Harvester Co.
13. Anaconda Copper Corp.
14. Kennecott Copper Corp.
15. Marinduque Mining and Industrial Corp.
16. Cleveland Cliffs Iron Co.
17. Power Shovel and Crane Association.
18. State Electricity Commission of Victoria.
19. Rheinische Braunkohlenwerke, G.F.R.
20. Utah Mining Corp., Navajo Mine.
21. A.B. Gullhogöns Bruk, Sweden.

3. LOADING SHOVELS

INTRODUCTION

The crawler-mounted loading shovel is the machine most capable of handling hard, dense, abrasive, badly fragmented ground - by virtue of its positive crowd action and the possibility of applying a high breakout force. It can accurately spot for loading into dump trucks, rail wagons, loading hoppers, etc. Because of its robust construction and simple action it can have relatively high availability, but it is not very mobile and has a poor sub-grade digging capability. A competent floor is essential unless oversize crawlers of low bearing pressure are used.

The operation of the loading shovel has the following cyclic pattern:

1. Loading by crowding the dipper into the rock pile and then hoisting it through the rock pile, (digging)
2. Swinging the superstructure till the dipper is above the dumping point in the transport system (swinging or slewing)
3. Dumping the dipper load (dumping)
4. Swinging the superstructure, while lowering the dipper, until the dipper is repositioned for digging (return swing).

Loading shovels can be divided into three main categories:

- (1) heavy-duty mine loading shovels - for dense, abrasive, badly fragmented ground, as is found in most metalliferous mining operations:

(2) general purpose loading shovels - for lighter, well fragmented materials, e.g. sand and gravel, coal, bauxite, etc.: and

(3) hydraulic shovels (see later).

Loading shovels with 25-yd³ (19-m³) dippers have been in service since 1968.

DIPPER SIZE

The first major step in shovel selection is the determination of dipper size. Since mine planners are mostly concerned with in situ volumes (bank volumes), these are generally used in calculations. Dipper size can be expressed as

$$D_c = \frac{Q}{C \times S \times A \times O \times B_f \times P} \quad \dots (3.1)$$

where

D_c = dipper capacity (volume)

Q = production required (bank volume/h)

C = theoretical cycles per hour for a

90° swing = $\frac{60}{t_c}$, and

t_c = shovel cycle time for a 90° swing (min)

A = mechanical availability during the scheduled hours of work

O = job operational factor

D_f = dipper factor

S = swing factor

P = propel time factor

C - Theoretical cycles per hour

C may be obtained from manufacturers' literature or from time studies.

Alternatively, the approximate times in Table 3.I can be used in association with an approximate dipper size. The skill of the operator has some effect on the cycle time and without the benefit of time studies or previous experience in similar conditions precise figures cannot be justified initially.

TABLE 3.1

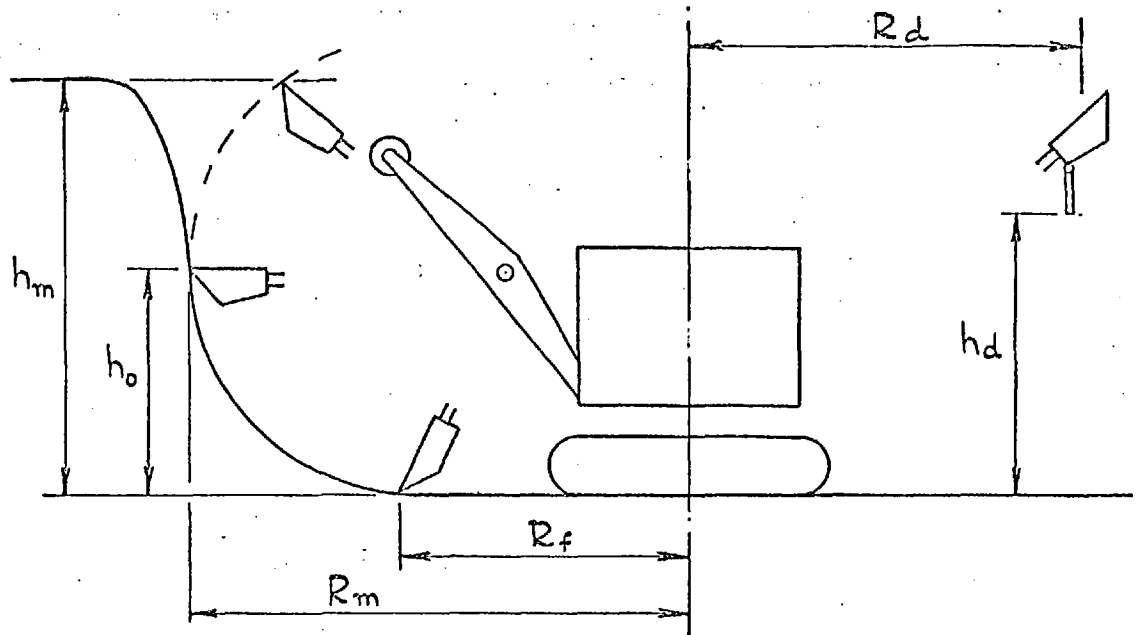
LOADING SHOVEL CYCLE TIMES (sec) **

(90° swing)					
yd ³	D _c	Digging conditions*			
	m ³	E	M	M-H	H
4	3	18	23	28	32
5	4	20	25	29	33
6	5	21	26	30	34
7	5.5	21	26	30	34
8	6	22	27	31	35
10	8	23	28	32	36
12	9	24	29	32	37
15	11.5	26	30	33	38
20	15	27	32	35	40
25	19	29	34	37	42

*See Chapter 2

**Compiled mainly from manufacturers' literature but confirmed by complete time studies and time studies of digging and dumping plus swing times taken from manufacturers' literature.

For most open-pit mining operations shovels have no difficulty in loading up to their optimum digging depth. In strip mines, however, the mineral bed may be relatively thin: hence, the crowd-hoist time in the shovel cycle will be increased to obtain a full dipper load. This may be corrected from Table 3.II. The optimum digging depth for a range of shovel sizes is shown in Fig.3.1.



h_m = MAXIMUM CUTTING HEIGHT
 h_o = OPTIMUM CUTTING HEIGHT
 h_d = MAXIMUM DUMPING HEIGHT
 R_m = MAXIMUM CUTTING RADIUS
 R_f = LEVEL FLOOR RADIUS ("CLEAN UP")
 R_d = MAXIMUM DUMPING RADIUS

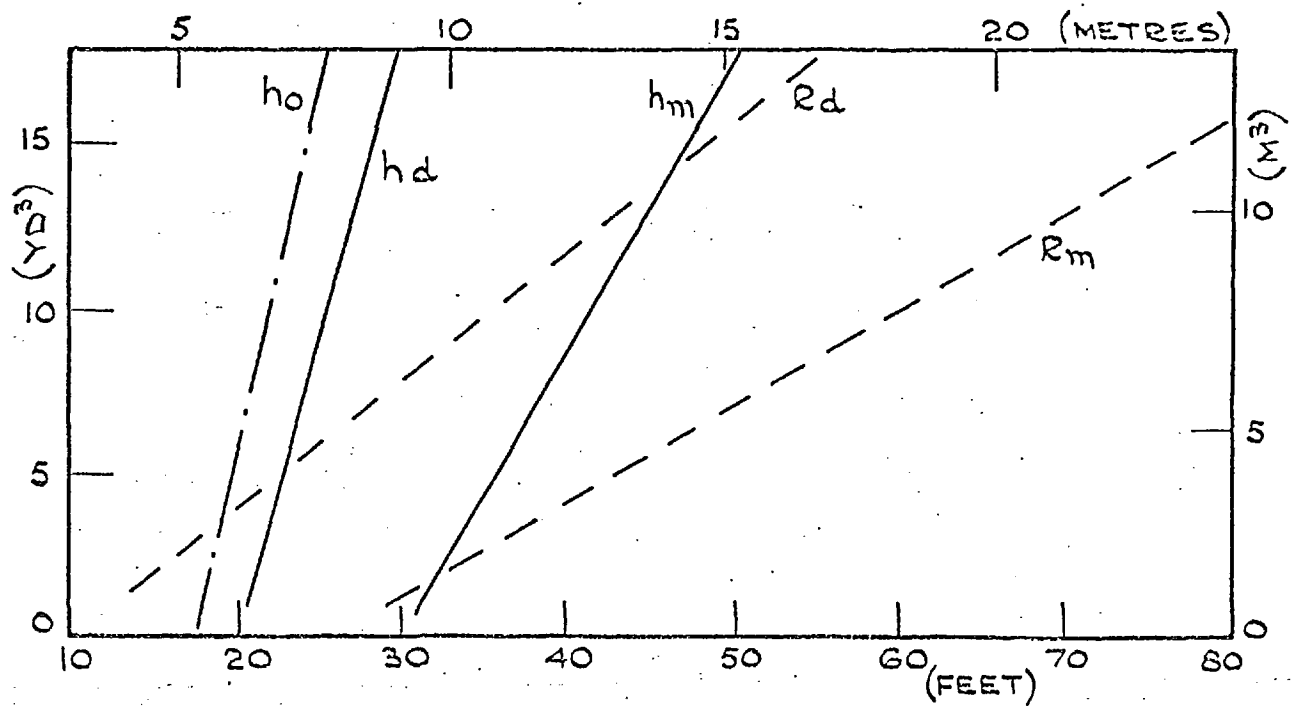


FIG.3.1 APPROXIMATE DIMENSIONS OF LOADING SHOVELS FOR INITIAL INVESTIGATIONS

TABLE 3.II

CORRECTION FACTOR FOR SHOVEL CYCLE
TIME WHERE DIGGING DEPTH IS LESS
THAN OPTIMUM*

Optimum digging depth, per cent	40	60	80	100	120	140	160
Cycle time correction factor	1.25	1.10	1.02	1.00	1.03	1.1	1.75

*Little opportunity was available to make time studies for these figures and they are based on published figures 31, 32, 33, 34.

In some strip mining operations the bed underlying the mineral may be insufficiently competent, e.g. fire-clays, for the passage of rubber-tyred trucks. The haul roads are then located on top of the mineral bed and the loading shovel may be fitted with a longer than standard boom and dipper handle to provide sufficient reach to load the trucks. In these circumstances the loading shovel cycle times will be greater than those shown in Table 3.I, and manufacturers literature should be consulted, if available. An approximate allowance of 7 - 12 per cent may be added to the cycle times shown in Table 3.I.

S - Swing Factor

Loading shovel cycle times are normally based on a 90° swing. Variations in angle of swing necessitate correction of the shovel cycle time: most manufacturers provide charts to determine this correction. Table 3.III is based on manufacturers' literature plus time study figures used in conjunction with manufacturers' swing time curves.

TABLE 3.III
SWING FACTOR

Angle of Swing degrees	45	60	75	90	120	150	180
Swing factor	1.2	1.1	1.05	1.0	0.91	0.83	0.73

A - Availability

A may be determined from plant records by industrial engineering methods, time studies, etc.

O - Job Operational Factor

The job operational factor may also be determined by industrial engineering methods. Care must be taken to ensure that if the propel time factor P is included in O that it is eliminated from formula (3.1)

A O

Where no experience is available to enable A and O to be determined their product A O, the operating efficiency may be determined from Table 2.VI.

D_f - Dipper factor

$$D_f = \frac{\text{Fillability}}{\text{Swell Factor}}$$

Fillability is the loose volume of material excavated in an average load as a ratio of the dipper capacity D_c : this is best determined by field measurements but Table 2.I provides a guide.

P - Propel time factor

P depends upon the time required to propel the shovel during manoeuvring. In strip mining, where a relatively thin narrow bed is exposed, the shovel must move more often than where a high rock pile is being loaded. Again industrial engineering methods can be used to determine P. It was impractical to make time studies to determine P and insufficient of the shovels included in the data available were fitted with data loggers to obtain statistically significant figures. Table 3.IV is therefore compiled from as many sources as possible without recourse to any sophisticated processing of the data. The figures are partially confirmed by a manufacturer's literature ³⁵.

TABLE 3.IV
PROPEL TIME FACTOR

Strip mines	0.75
Multi-bench mines	0.85
Sand and gravel pits	0.90
High face quarries	0.95

In many cases the propel time factor in the data compiled was combined with O or A0 and allowances have been made for this. When using collected data care must be taken to avoid double compensation.

The approximate dipper size having been fixed, a standard dipper size can be chosen and a range of shovels can be selected from manufacturers' information. The actual production capacity of each model must then be calculated, manufacturers' data being used to determine the theoretical

cycle times and swing factors to ensure that the production requirements will be met.

LOADING SHOVEL GEOMETRY

Detailed studies are required to determine the space requirements for a loading shovel. Ample bench room is necessary for operational efficiency, but in general the slope angle for a single bench, with double spot loading, is too flat for most multi-bench, conical pits (Fig 3.2) and several bench levels forming a machine loading group must be excavated in sequence. Space must also be available for drilling and blasting and to allow for the passage of trucks. For pits with rail haulage, clearance must be provided for rail track, trolley lines and shovel, the whole sequence of drilling, blasting, loading, trolley and track shifting being planned in detail. The approximate leading dimensions of loading shovels are shown in Fig.3.1 and the clearance heights of rear dump trucks in Fig.3.3. These may be used for early planning, but the actual dimensions of the selected equipment must be used in subsequent mine plans.

Bench heights must be established, most mine operators preferring to scale the working face using the loading shovel, for cost reasons and increasingly because of statutory requirements. Many larger modern shovels will scale up to 15m (50ft) and the present trend is to increase bench heights from 10m (30ft) to 15m (50ft). With this configuration, economies can be effected in drilling and blasting costs, especially for overburden where fragmentation may not be important.. These economies must of course be balanced against larger loading shovels. The type of rock and degree

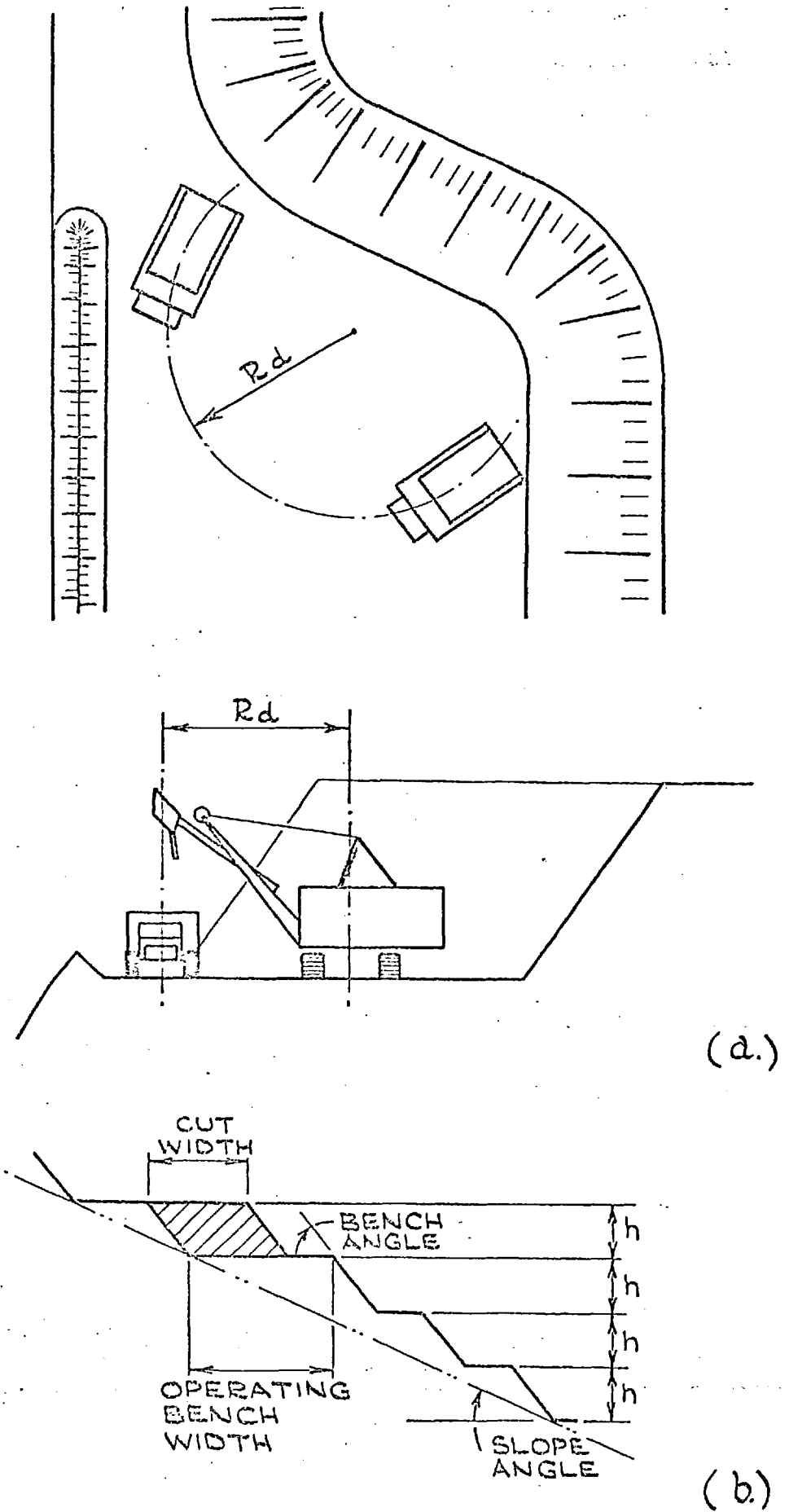


FIG.3.2 DOUBLE SPOT LOADING AND MACHINE EXCAVATION GROUP

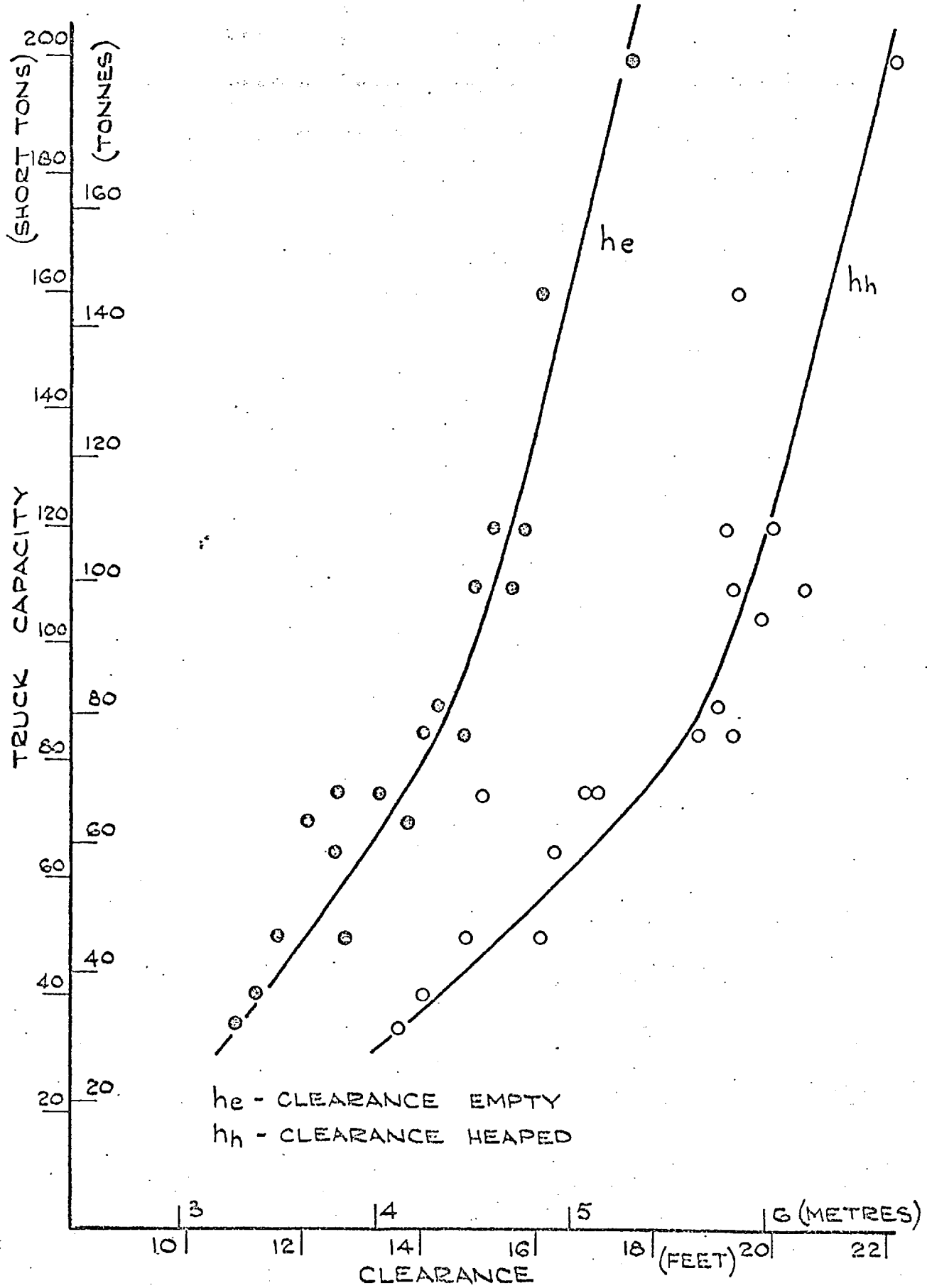


FIG.3.3 CLEARANCE HEIGHTS FOR REAR DUMP TRUCKS
FOR INITIAL INVESTIGATIONS

of fragmentation obtainable determines the suitability of a particular size and type of bucket to be used with a loading shovel. The limiting factor on dipper size is often the duty cycle of the hoist motor. Thus when a shovel digs the full height of the face in a soil-like material or a gravel-clay material subjected to blasting by horizontal holes ("toe holes"), the duty is more severe than digging a fragmented rock pile.

LOADING SHOVEL DRIVES

Diesel Drives

For small loading shovels engaged in small scale operations the diesel drive is almost universal since these operations are usually characterised by a lack of detailed planning so that the diesel drive with its relative mobility and flexibility places no restrictions on operations. The cost of cable handling is also eliminated; this being of some importance for small shovels where the capital cost of mechanical handling of the small diameter trailing cables is not justified.

The main disadvantages of the diesel drive are:

1. Poor availability compared to electric drives
2. Relatively high fuel costs
3. Relatively high maintenance costs
4. Atmospheric pollution due to exhaust fumes
5. Need to service shovel with diesel fuel.

Diesel - Electric Drive

The precision and convenience of electrical control has considerable advantages in larger shovels. In the following situations, electricity supply problems can occur and the diesel-electric drive which combines the

flexibility of the diesel drive with the control characteristics of the electric drive, may be selected:

- 1 During the early life of a remote mine before an adequate power system is established e.g. the Western Australian iron ore mines.
- 2 In some short life operations where the establishment of a large electricity system is not justified e.g. some short life operations of the National Coal Board Opencast Executive.

Electric Drives

Electric drives have the following advantages:

1. Small physical dimensions
2. Motors can be located to eliminate complex gear trains and chain drives
3. Maintenance and fuelling of the diesel engine are eliminated
4. Exhaust gases are eliminated
5. Operation is smooth and quiet
6. Readily controllable

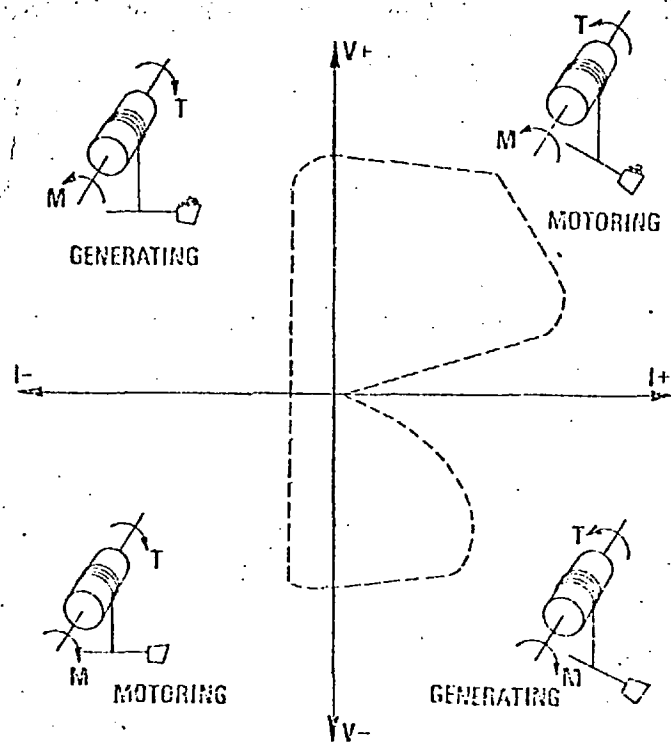
The required operating characteristics of the various drives of the loading shovel are relatively severe. Safe, efficient but simple electrics are therefore probably not feasible. If the simple example of an electric shovel's swing drive is examined, it is apparent that when starting to swing a torque must be applied in an arbitrarily positive direction. The drive then acts as a MOTOR. To bring the shovel to rest an opposite torque (negative) must be applied.

The rotational energy of the super structure and the drive are converted to electrical energy and the drive acts as a GENERATOR, i.e. regenerative braking occurs.

To return the dipper to its original position a negative torque must again be applied to rotate the shovel superstructure in a negative direction, the drive again acting as a MOTOR. To decelerate the shovel a positive torque must be applied while the direction of swing is still negative. Again the stored energy is converted into electrical energy and the drive acts as a GENERATOR. It is easy to visualise similar cycles for crowd and hoist motions. Fig. 3.4 illustrates the theoretical cycle for a hoist motion drive.

Assuming constant flux and negligible armature voltage drop, the hoist drum speed is directly proportional to the hoist motor armature voltage, V , and the hoist drum torque is directly proportional to the armature current, I . During hoisting the voltage is positive, during lowering negative. For a direct hoisting torque the current is positive and the reverse torque negative. Where voltage and current have the same sign, the drive is MOTORING. Where the signs are different to each other, the drive is GENERATING.

Drive characteristics are further complicated by friction which opposes acceleration but assists deceleration, by back-lash in gears, rope stretch, etc. If the dipper strikes a solid obstruction during loading, the stall current overshoot must be immediately restricted to prevent electrical and mechanical damage. These requirements make some sophistication in the electrical system essential.



M - direction of motion
 T - direction of torque
 V - armature voltage
 I - armature current

FIG.3.4 FOUR QUADRANT OPERATION OF
 HOIST MOTION DRIVE OF AN
 ELECTRIC SHOVEL

Two types of electric drive are in common use:

1. Ward Leonard control
2. Eddy current coupling.

Thyristor fed dc drives and hydrostatic crowd drives are also available (see later).

Most leading manufacturers use Ward Leonard systems for swing and crowd motions because, to avoid reversing clutches and brakes, it is virtually essential to select variable speed dc motors with their inherent advantages of flexibility, excellent electrical "tuning" characteristics and potential for absorbing sudden and severe reversals. The majority of manufacturers also use the Ward Leonard drive for the hoist motion, but one prefers the eddy-current coupling.

The Ward Leonard Drive

Fig.3.5 shows the basic schematic diagram of the Ward Leonard drive, the principles of which are well known. The voltage supplied to the driving motor is varied by varying the generator field current. Relatively constant motor flux is thus obtained and the motor field is not weakened for speed control. This results in an acceptable "stiffness ratio" of field ampere turns to armature reaction ampere turns throughout the full range of motor speed, eliminating any possible instability.

A representative duty cycle for a typical loading shovel with Ward Leonard drive is shown in Fig.3.6 and a cycle recorded by a Wattmeter chart recorder is also shown for comparison. It will be noted that the cycles are characterised by the peaks of load and regeneration. The

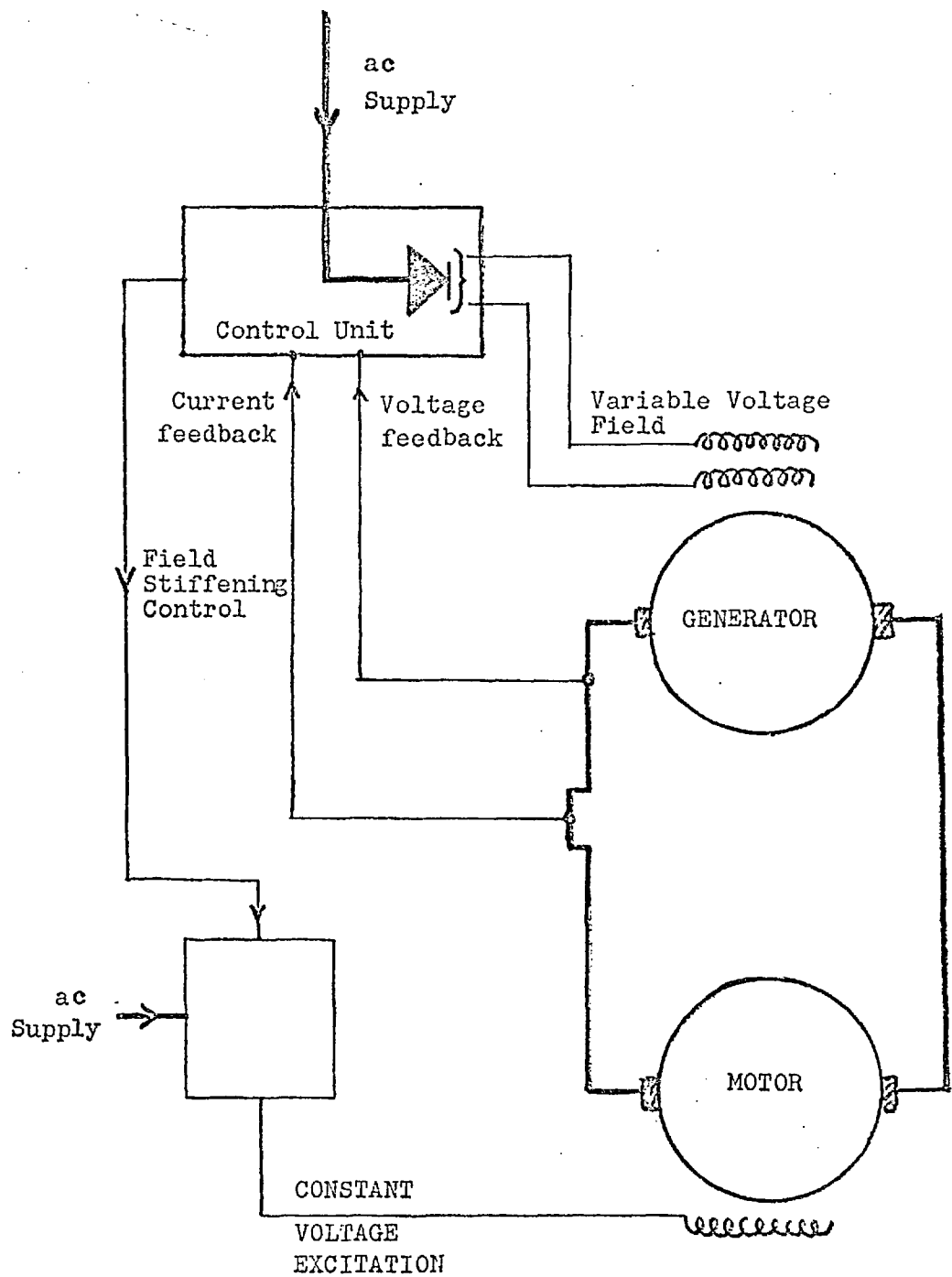


FIG. 3.5 BASIC WARD LEONARD SYSTEM FOR EACH SHOVEL MOTION - HOIST, CROWD, SWING

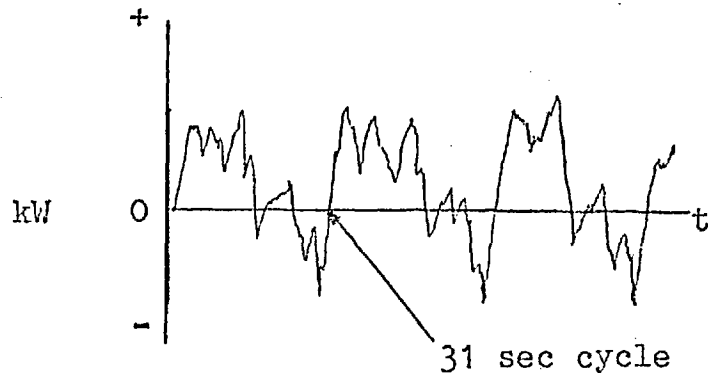


FIG. 3.6a ELECTRICAL LOAD CYCLE OF A
LARGE LOADING SHOVEL

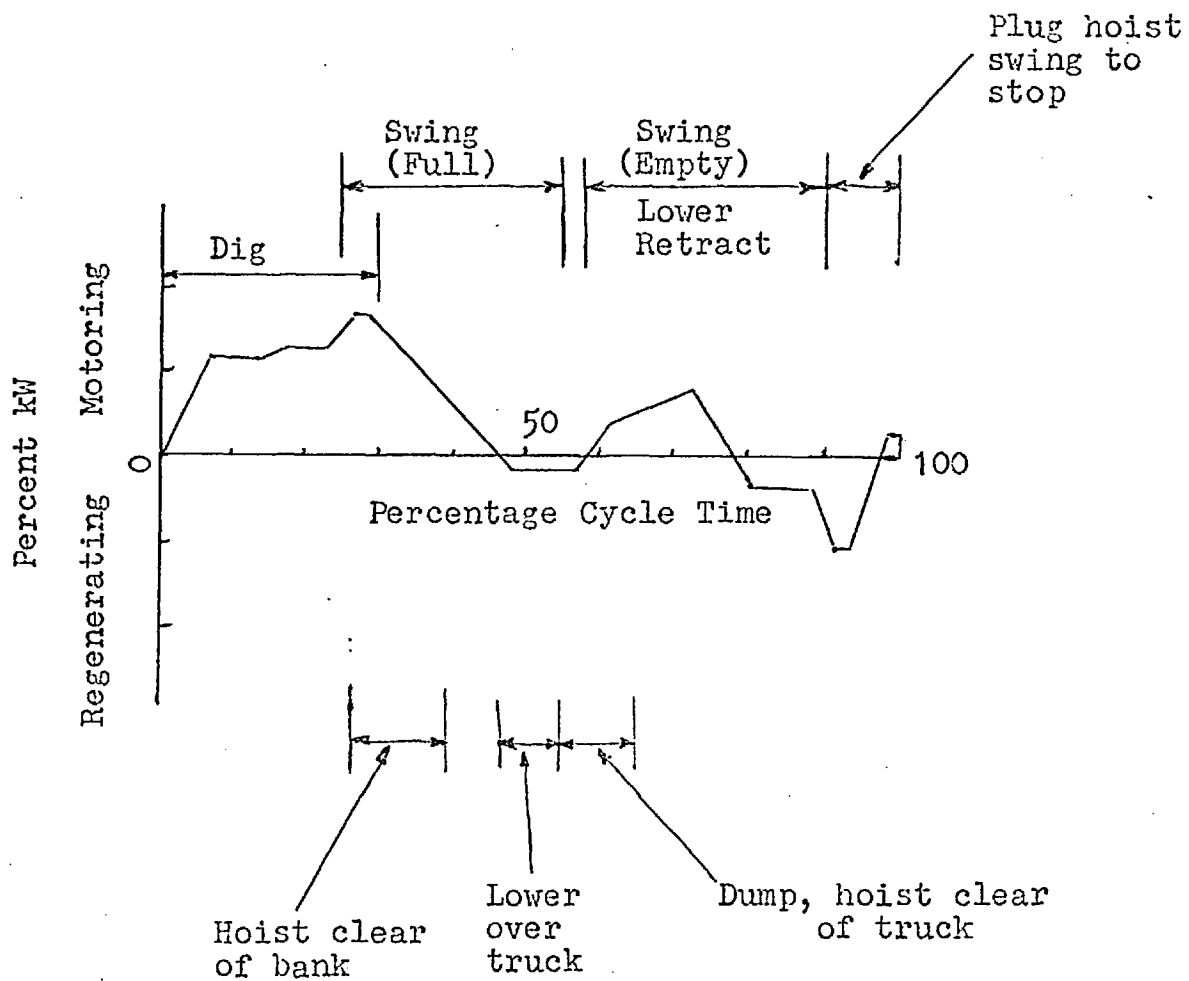


FIG. 3.6b IDEALISED SHOVEL ELECTRICAL LOAD CYCLE
FOR SIMULATION STUDY

various operations in the shovel cycle and the periods of operation of the crowd hoist and swing drives are shown as they occur. The effect of this fluctuating load on electric power consumption and maximum demand level has some relevance regarding electricity charges and because of its importance in the selection process it is examined later.

The Eddy Current Coupling (Magnetorque electromagnetic clutch manufactured by Harnischfeger Corporation) is used for hoist motor drive applications, and provides a driving torque in the hoisting direction only, i.e. it is non-reversing. It can provide no energy to accelerate the dipper stick assembly when initially lowering the dipper. When the hoist drum is released by the eddy current coupling, the dipper falls, being accelerated by the potential energy of the dipper stick assembly mass only. As the dipper approaches the ground, the potential energy has largely been converted into velocity energy. During this part of the lowering motion the eddy current coupling is de-energised, i.e. no energy is being transmitted in either direction. To decelerate the dipper the eddy current coupling must supply hoisting energy and this is the greatest effort required of the coupling since:

- a) it must transmit hoisting energy from the ac driving motor to decelerate the dipper stick assembly, and
- b) dissipate within the coupling the heat energy equivalent to the original potential energy of the dipper stick assembly.

Therefore energy is transmitted from the ac driving motor during the later part of the lowering motion.

The eddy current coupling is simpler in operation, lower in first cost and is inherently easier to maintain than the Ward Leonard drive. Its characteristics are of course mainly suited to the hoist motion drive although it is also used in the swing motion drive of larger diesel drive shovels.

A typical load cycle recorded by field measurement, for a loading shovel with an eddy-current hoist motion drive and Ward Leonard drives for the crowd and swing motions is shown in Fig.3.7. It will be noted that the regenerative peaks of the full Ward Leonard drive are absent in the eddy-current coupling drive.

Because of the absence of regeneration in the load cycle of the eddy current coupling drive, it appears that the power consumption of this drive would be greater than that of the conventional Ward Leonard set, because:

- a) a single Ward Leonard drive shovel connected to a metering point would not have any electrical power consumption recorded against it during regeneration, nor would it be credited with the regenerated power, since the normal mode of metering does not operate in this manner. The eddy-current coupling drive shovel would have zero power recorded when the dipper fell up to about 80% of the dipper travel. It must then decelerate the dipper by applying a reverse torque and some power consumption would be recorded.
- b) in most situations, several shovels or other forms of electrical load are connected to a metering point and a Ward Leonard drive

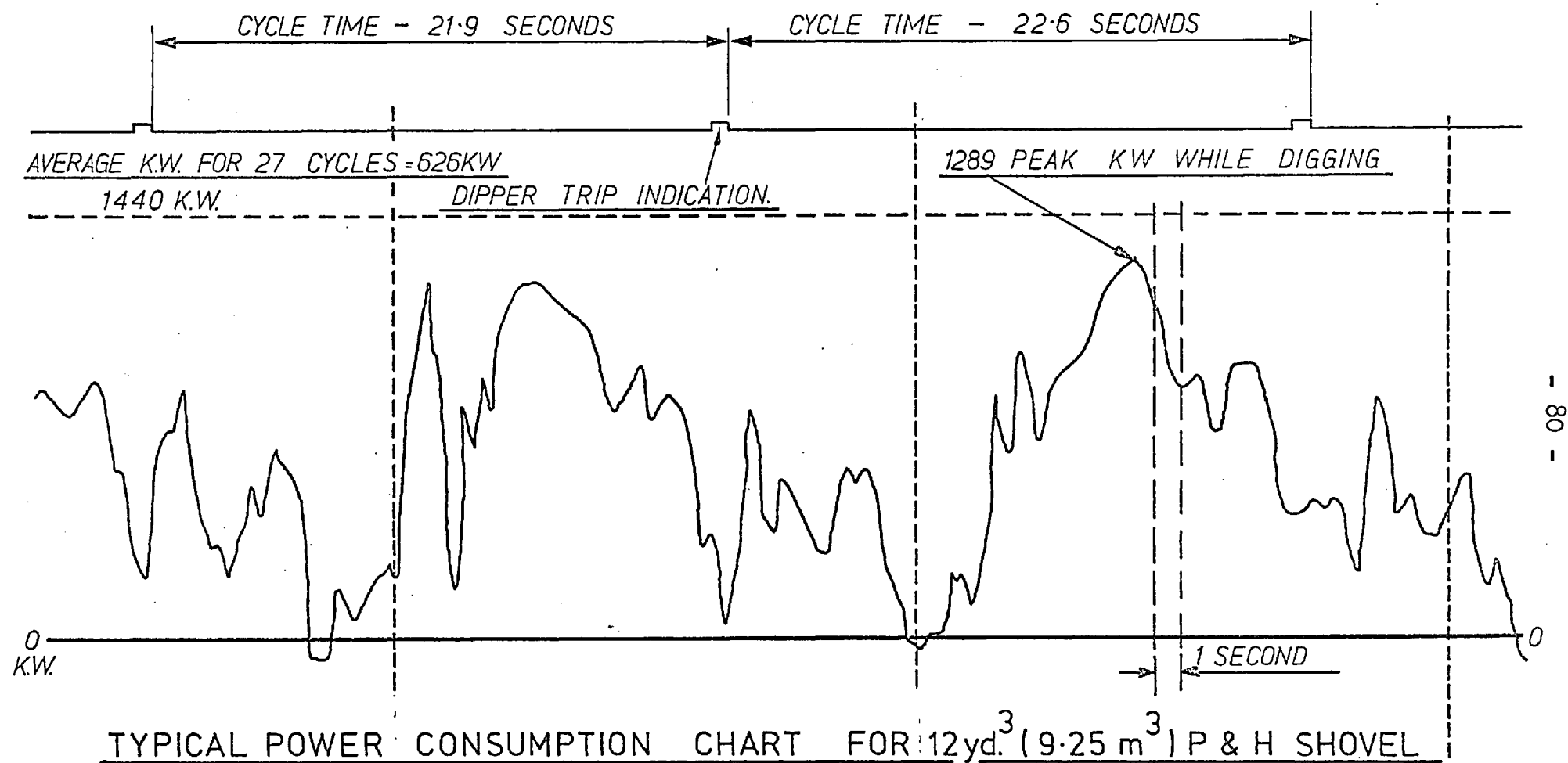


FIG.3.7 RECORDED LOAD CYCLE OF EDDY CURRENT COUPLING HOIST DRIVE SHOVEL

shovel may be expected to supply the other loads during periods of regeneration and in effect receives a credit for this regenerated power. The eddy current drive shovel cannot receive such a credit since no regeneration occurs.

Tests carried out by the author on 6yd³ shovels in 1967 using surveyed bank volumes and a standard kWh meter produced the following results.

TABLE 3.V

POWER CONSUMPTION OF 4.6m³ (6yd³)
LOADING SHOVELS

Type of Drive (Hoist)	kWh consumed	m ³ (bank) moved	yd ³ (bank) moved	kWh /m ³	kWh /yd ³
Eddy Current Coupling	55,650	50,500	65,508	1.10	0.85
Ward Leonard	21,950	49,550	64,427	0.443	0.34
Ward Leonard	22,100	37,500	48,673	0.589	0.453

The average power consumption for the Ward Leonard drive shovels was 0.506 kWh/m³ compared with 1.10 kWh/m³ for the eddy current coupling drive shovel, thus the power consumption of the eddy current coupling drive shovel was 2.17 times that of the Ward Leonard drive for shovels operating in the same mine on the same type of operation.

This single test appears to be validated by other figures:

TABLE 3.VI

FIELD MEASUREMENT OF SHOVEL POWER CONSUMPTIONS 36, 37

Location	Type of Drive		$\frac{a}{b}$
	Eddy Current Coupling(a)	Ward Leonard(b)	
Eastern USA (Iron)	0.42 *	0.21 *	2.00
Mid-West USA (Iron)	0.856**	0.392**	2.19
Peru (Copper)	0.34 *	0.123 *	2.76

* kWh/short ton

**kWh/yd³

This indicates that a considerable saving in kWh charges can be made by selecting a complete Ward Leonard drive shovel in preference to an eddy current drive shovel. The method of charging for electrical energy is not usually based on power consumption alone however and some form of maximum demand (kVA) charge is usually levied which must be taken into account.

Electrical Load Characteristics

Fig.3.8 shows a typical dc machine characteristic (G.E. "armoured" dc machine). The solid line indicates the "static limit" characteristic and represents the maximum cyclic limit that the machine can commutate. It follows that at high voltages (approaching 600 V dc) and low currents (approaching 0A) the voltage per segment of the commutator is highest. As the current increases the volt/seg and consequently machine volts must be gradually decreased to maintain good commutation up to approximately 62.5% of the machine's rated stall current. For larger currents the voltage must be greatly reduced otherwise commutation deteriorates and becomes unacceptable. For normal operation,

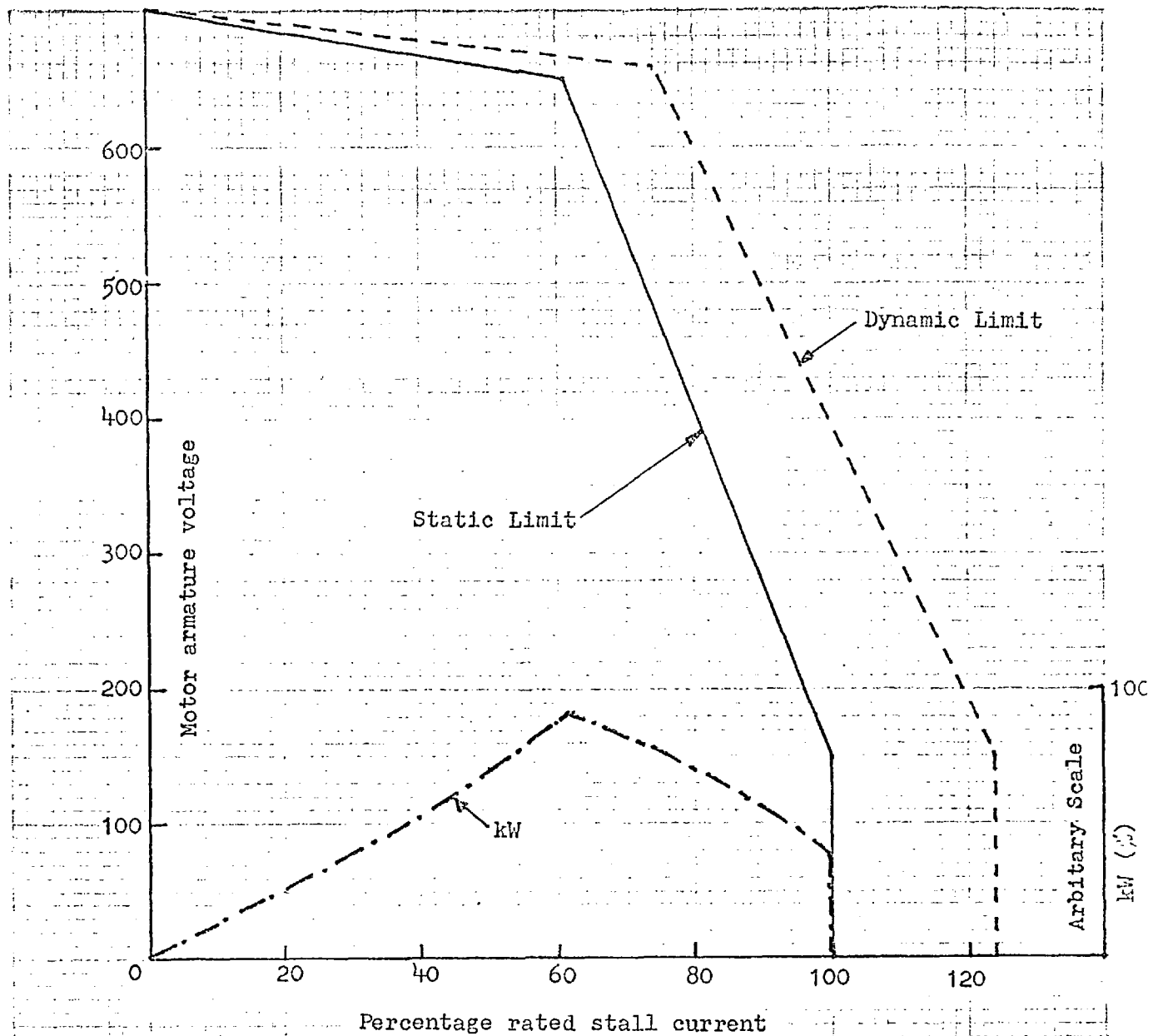


FIG. 3.8 COMMUTATION LIMITS OF A TYPICAL dc SHOVEL MOTOR

therefore it is essential to prevent the dc machines working beyond this static limit characteristic by suitably tuning the control system i.e. the static limit is a desirable and not an inherent characteristic. The dotted line in Fig.3.8 is the "dynamic limit" characteristic, and is the maximum permissible condition under which the machine can operate under transient conditions e.g. severe digging approaching a stall.

The dynamic limit for excavating machines is normally 25% greater in current for a given voltage than for the static limit.

It is normal to operate right up to the static limit of a dc machine during the severest part of the cycle in order to reduce the cycle time. The combination of this mode of operation for the crowd, hoist and swing drives plus the machine losses, auxiliaries, etc. produce the load cycle diagram for a Ward Leonard drive shovel, shown in Fig.3.6a, i.e. the shovel motor-generator set load plus auxiliaries. This ac line load is of considerable significance to the successful operation of the shovel and is of great importance in the selection procedure. The greater the maximum load the larger the voltage drop at the m/g set driving motor terminals, which in turn reduces the capability of the motor to handle the maximum load, since the maximum torque capacity of an ac induction motor varies as the square of applied voltage ($T \propto V^2$).

The NEMA standard, which is universally accepted, requires that ac induction motors deliver their rated power continuously at ac line voltages plus or minus 10% of rated voltage.

The voltage should not exceed 110% rated voltage because of:

1. Greatly increased magnetising currents.
2. Excessive magnetic forces on the motor windings during starting.
3. Increased electrical potential stress on the insulation.
4. Electronic control gear is sensitive to excessive voltage.

For pit electrical supply systems where loading shovels are used, experience shows that it is desirable to limit the voltage drop in order to maintain 95% of motor rated voltage at the m/g set driving motor at times of maximum power because:

- 1 The greater the voltage drop the larger the line current to provide maximum power. Forces on the motor windings are also proportional to the square of the current.
- 2 The possibility of the ac driving motor "pulling down" i.e. stalling, increases with increasing voltage drop, since maximum (stalling) torque is proportional to the square of the line voltage. This results in excessive motor currents and usually the over-current protection stops the driving motor causing operations to cease.
- 3 The fall in m-g set speed can cause overloading of the dc generator field regulators.

These factors indicate the need for the electric drive and its electricity supply system to be considered as an integral part of the selection process.

The general basis for selection of the various machines forming the Ward Leonard drives for the crowd, hoist and swing motions requires some explanation (Fig.3.9):

The dc motor must develop the necessary torque and have the capacity to operate at the speed set by the shovel driver. Thus the dc motor is generally selected on the basis of meeting the thermal requirements of the load.

The dc generator is driven at constant speed. A high speed is invariably chosen to obtain equipment which is physically small so it can be accommodated within the shovel. As a result of the high generator speed, generator commutation is the limiting factor rather than thermal capacity. In general if a generator can meet the static limit characteristic of Fig.3.8 as it approaches stall current it will have adequate thermal capacity to supply the needs of the dc motor.

The m-g set driving motor, usually an ac induction motor, must drive all the dc generators throughout the duty cycle and therefore is subjected to a fluctuating load (Fig.3.6). The auxiliary loads, blowers, compressor, lighting, etc. amount to about 10% of the m-g set driving motor rated power. At maximum load (Fig.3.6) the driving motor must be capable of operating without pulling down even though the supply voltage is reduced to 95% of rated motor voltage (see Fig.3.10) i.e. the stalling torque at 95% rated voltage is $0.95 \times 0.95 = 90\%$ of the stalling torque at 100% motor rated voltage. If the stalling

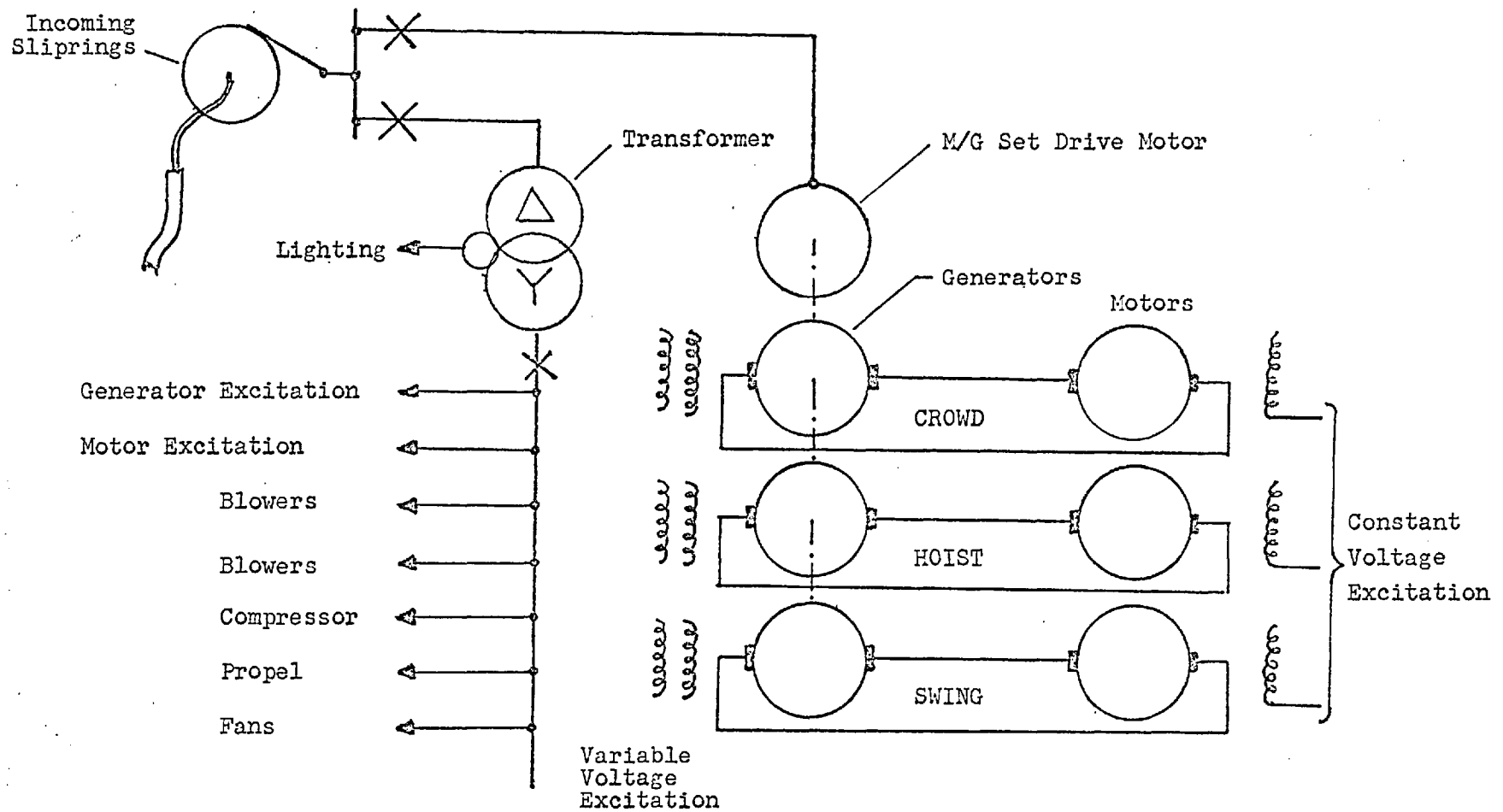


FIG. 3.9 SHOVEL ELECTRICS - SINGLE LINE DIAGRAM

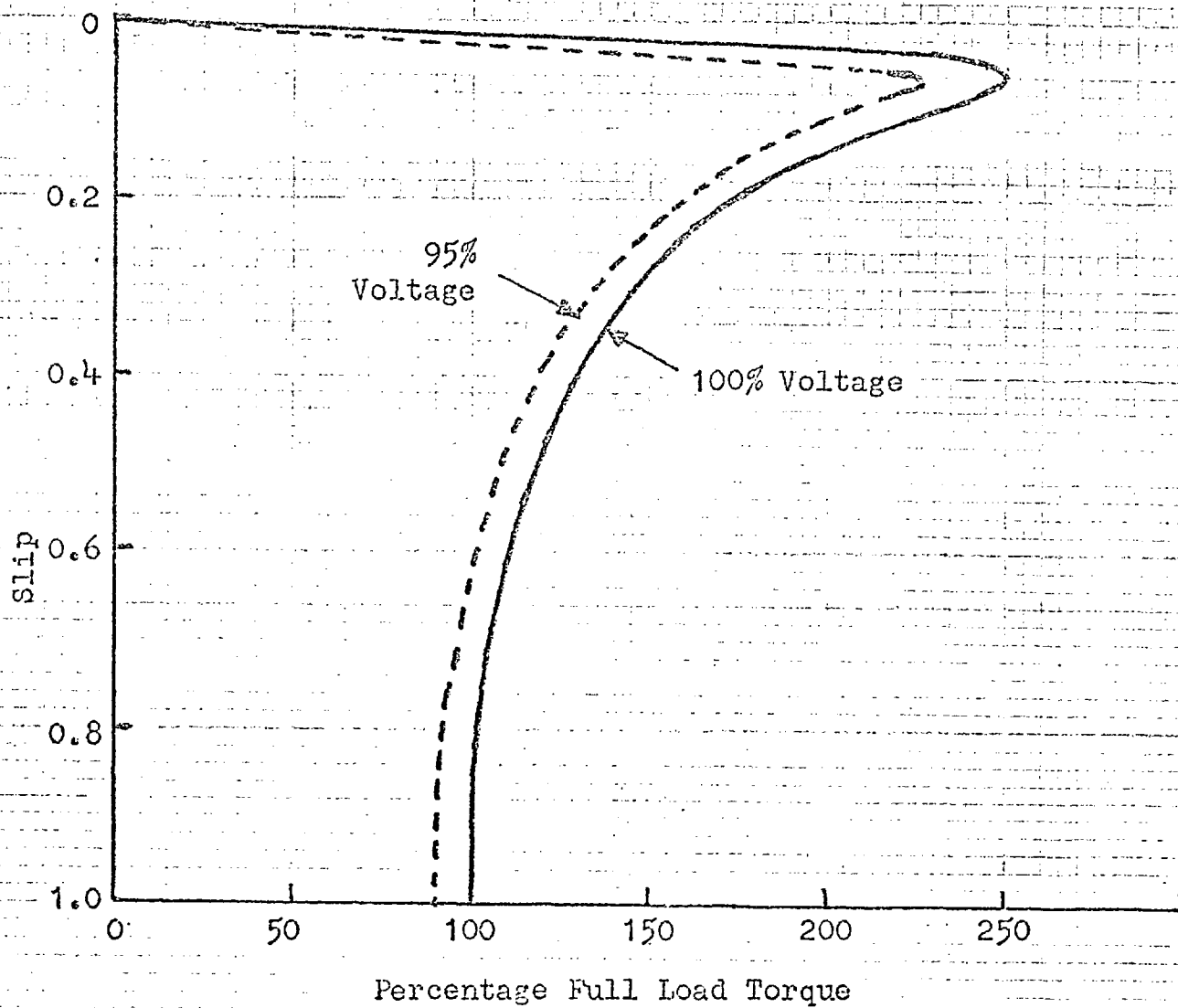


FIG. 3.10 TORQUE-SLIP CHARACTERISTICS OF STANDARD INDUCTION MOTOR USED FOR SHOVEL M/G SET DRIVE

torque of the driving motor is exceeded the loads on the dc generators must be reduced quickly to allow the m-g set driving motor to accelerate back to its operating speed range. It is therefore necessary to ensure that the stalling torque of the m-g set driving motor is not exceeded during normal operation. The practical result of this is that the m-g set driving motor does not operate at the rms power of the nameplate rating. Measurements taken on a large number of shovels confirm this point.

It would of course be possible to design a motor with the required stalling torque but with a reduced power rating. Such a motor would however be a "special", would probably be expensive and the interests of the shovel user are best served by the use of standard motor designs, e.g. N.E.M.A. or equivalent ratings.

Trailing Cables. The importance of maintaining at least 95% motor rated voltage at the m-g set driving terminals is clearly apparent. Because of this the size of trailing cable must not only be selected because of thermal capacity and short circuit rating but also on voltage drop considerations. For normal lengths of trailing cables used in open pits, the thermal capacity and short circuit ratings generally are adequate if the trailing cable is selected on the basis of voltage drop.

Transformer. In many mine electrical systems the transformer supplying a single shovel load is based on its thermal capacity being equal to that of the nameplate power rating of the m-g set driving motor (which approximates the rms value of Fig.3.6). The cyclic maximum load however approaches 200% of full load. Under this

condition the transformer windings are subjected to impulse loads four times the designed full load forces (since $F \propto I^2$), leading to shortened transformer life. Specially designed transformers can be manufactured to withstand these cyclic impulse loads but these are expensive and in the interests of standardisation, standard industrial transformers are preferred. As an alternative a larger standard transformer can be used. The NEMA standards suggest that for a transformer subjected to a cyclic load, with a frequency between 100 and 200 cycles per hour, the transformer rated kVA should be 72% of the pulse kVA magnitude, i.e.

$$\begin{aligned} & 500 \text{ hp m-g set} \\ & \quad 90\% \text{ efficiency} \\ & \quad 0.9 \text{ full load power factor} \\ \equiv & \quad 460 \text{ kVA line load} \end{aligned}$$

$$\begin{aligned} \text{Maximum cyclic load} &= 2 \times 460 = 920 \text{ kVA} \\ \text{Auxiliaries} &= \frac{40}{960} \text{ kVA} \end{aligned}$$

(The power factor at maximum cyclic load would be greater than 0.9 but its effect can be ignored)

$$960 \times 72\% = 690 \text{ kVA}$$

The nearest larger standard transformer is 750 kVA and this is the recommended size. Such a transformer would have the added advantage of reduced voltage drop compared with a 500 kVA transformer which would have the required thermal capacity but would have shortened life.

Transformers supplying single shovels should be selected on the basis of peak cyclic loading.

Transformers supplying more than one shovel. In many open pit electrical systems, a transformer may be required to supply several shovels and there will be some diversity effect between their electrical loads since the load peaks of one shovel will on occasion coincide with the regenerative peaks of another shovel. The shovel cycles will not always be optimum as shown in Fig.3.6b, cycle times will vary, etc. This diversity effect for Ward Leonard drive shovels was measured by field trials ³⁸ by the author but the length of time available (1 week) was probably inadequate. In addition chart examination for longer periods is tedious and tends to become inaccurate. A more recent approach ³⁹ using Monte Carlo simulation techniques produced the diversity curve shown in Fig.3.11, again for Ward Leonard drive shovels. It will be seen that with up to 3 shovels, there is little diversity and that the combined peak load is:

no. of shovels x peak cyclic load for 1 shovel

For 4 shovels or more diversity occurs.

Transformers supplying up to 3 shovels should therefore be rated on the basis of cyclic loading (but at reduced frequency) and transformers supplying 4 or more shovels should be rated on the basis of voltage drop considerations.

Using the same simulation method for shovels with eddy-current coupling hoist drive and Ward Leonard crowd and swing drives and the generalised version of the load curve the author obtained the diversity curves shown in Fig.3.12. These may be compared with the results obtained by field measurements carried out by D. Houlton, of Selection Trust Ltd. on behalf of the author.

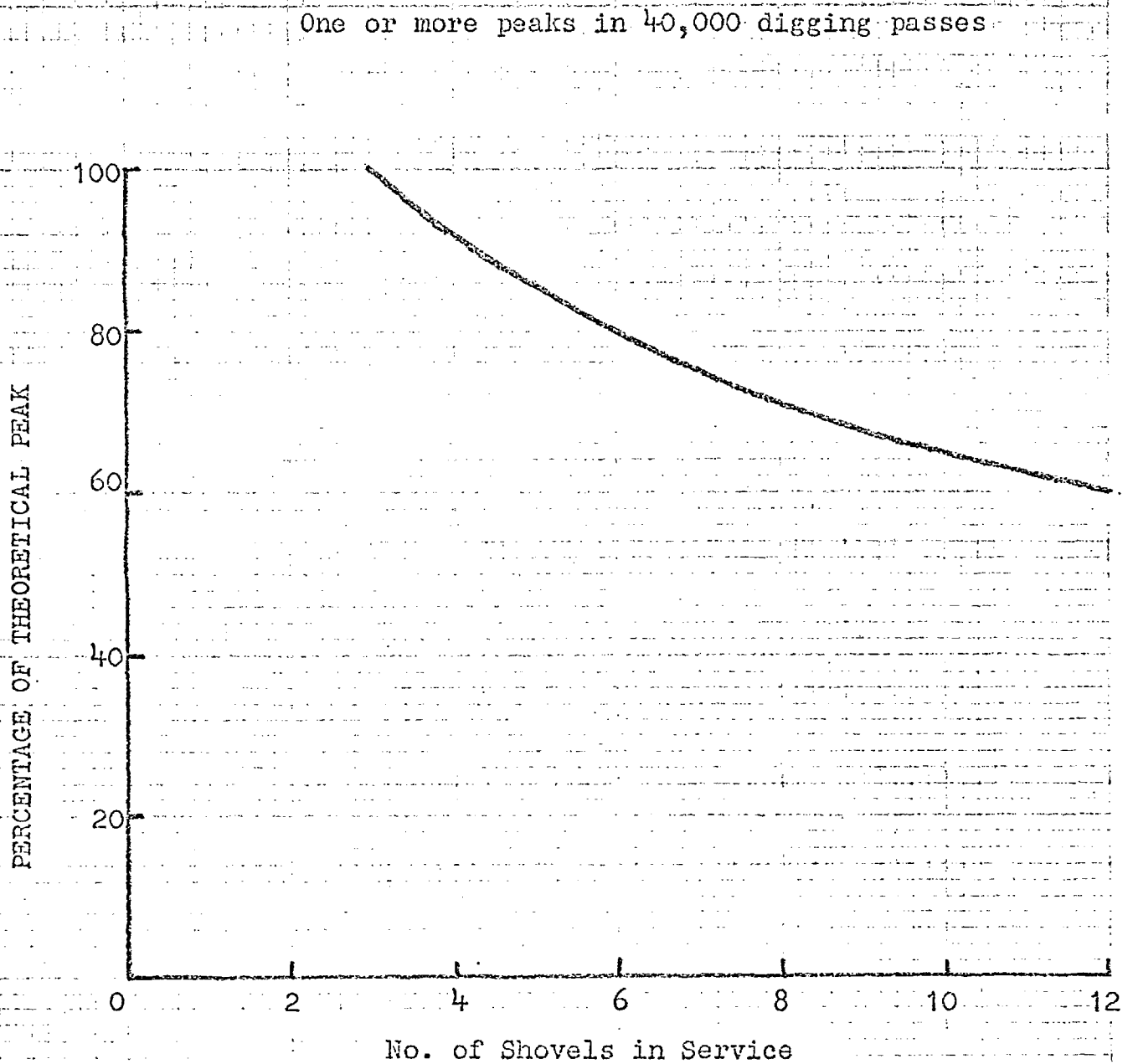


FIG. 3.11 DIVERSITY FACTOR FOR WARD LEONARD
DRIVE SHOVELS

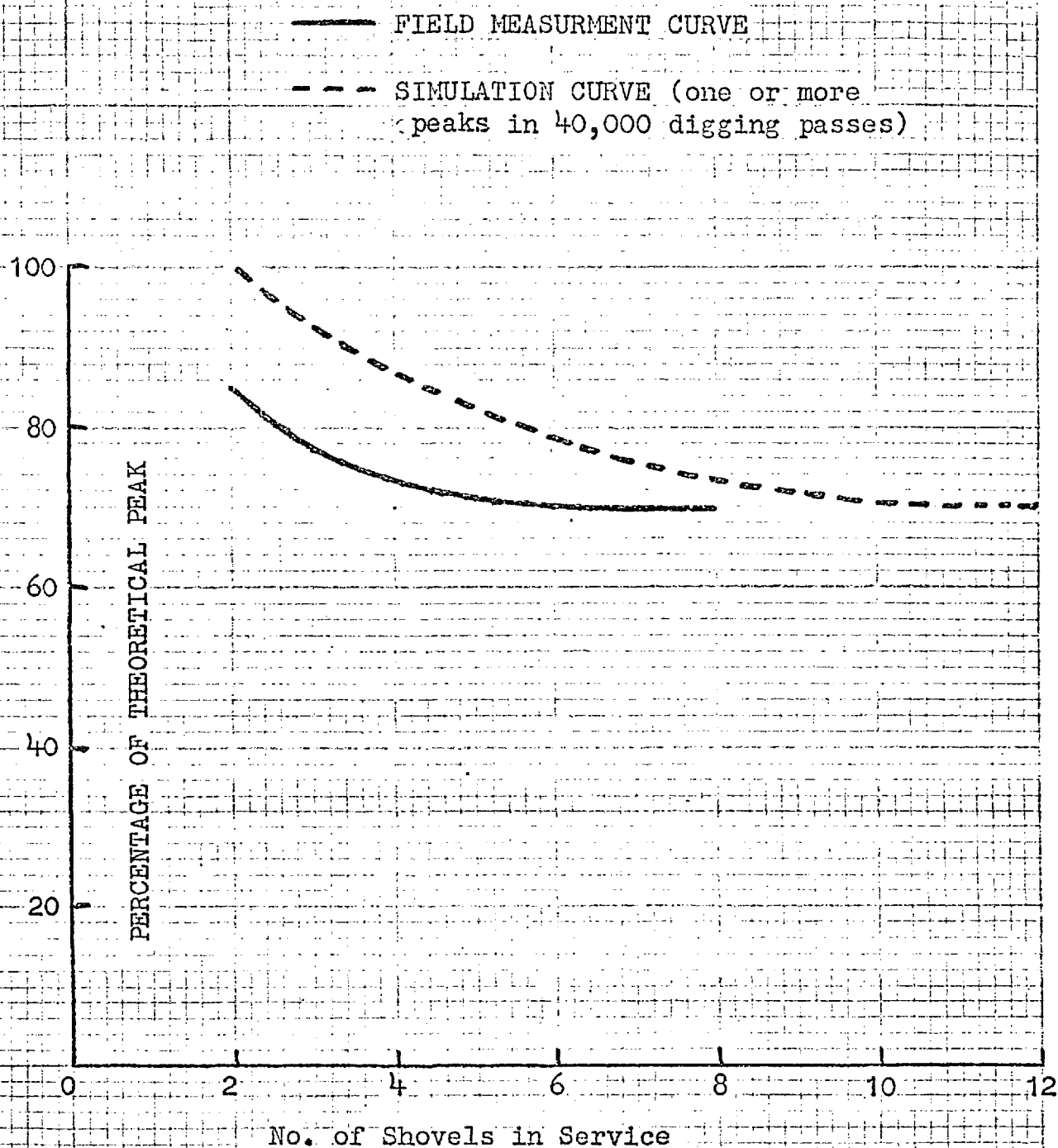


FIG. 3.12 DIVERSITY CURVES FOR EDDY CURRENT COUPLING HOIST DRIVE SHOVELS

Summarising the basis for selection for the electrical equipment for loading shovels:

TABLE 3.VII
BASIS FOR SELECTION
OF LOADING SHOVEL
ELECTRIC DRIVE EQUIPMENT

<u>Item</u>	<u>Basis for Selection</u>
dc motor	thermal capacity
dc generator	commutation limits
ac induction motor	stalling torque
trailing cable	voltage drop
Transformer	
up to 3 shovels	cyclic loading
4 or more shovels	voltage drop

ELECTRICAL SYSTEM DESIGN

Although the electricity supply system does not usually form part of the loading shovel specification, as previously shown it has a powerful influence on shovel selection and must be considered as part of the selection procedure using the systems approach. It is necessary to determine:

- 1 the motor terminal voltage at peak load, and
- 2 the stalling torque of the m-g set driving motor under these conditions to ensure it will not be exceeded.

The following method is suggested for selection purposes. Final electrical system design may follow the same procedure but more precise information should then be available.

- 1 Draw the system as a single line diagram
(Fig.3.13)

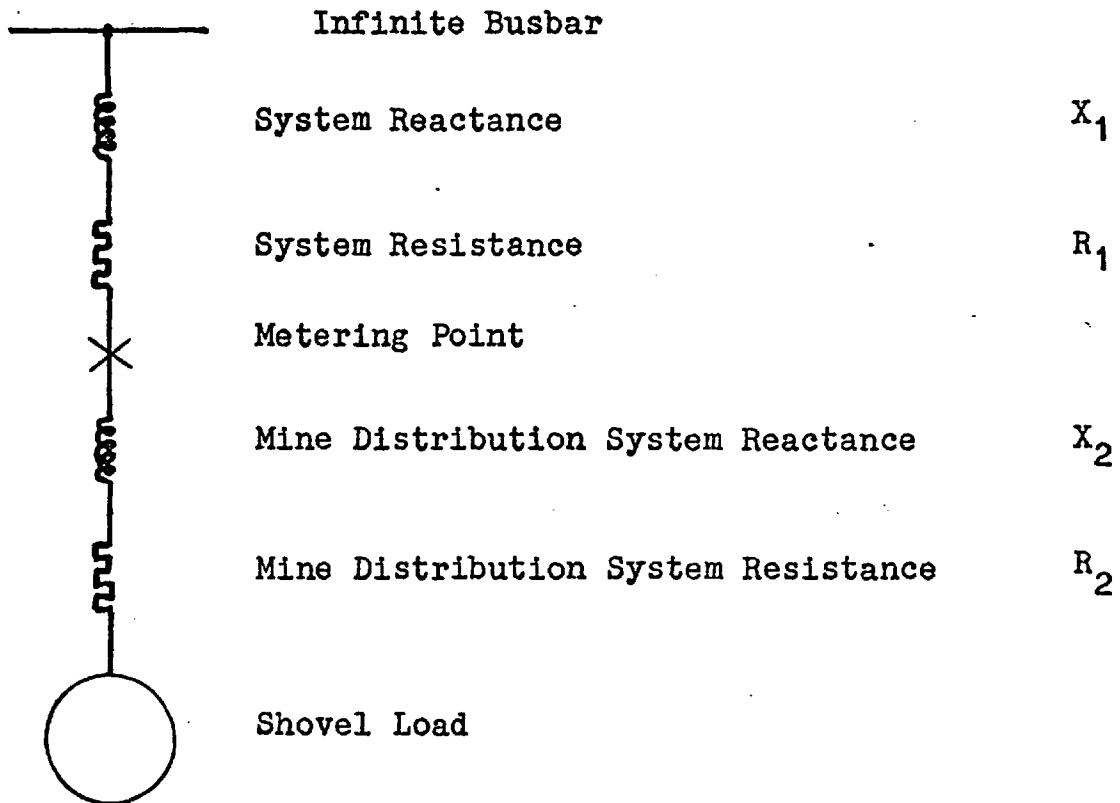


FIG 3.13 SINGLE LINE DIAGRAM OF SHOVEL ELECTRICAL SYSTEM

- 2 Select a BASE MVA for the system this may be any convenient figure.
- 3 Using the Per Unit System (see Appendix 3.A) determine the NO LOAD VOLTS, BASE VOLTS and PER UNIT VOLTS for the various voltage levels throughout the system.
- 4 Convert the ohmic values of resistance and reactance into per unit values (see Appendix 3A)
- 5 Summate the total resistance and the total reactance. The total impedance is:

$$Z = \sqrt{R^2 + X^2}$$

- 6 Assuming that the infinite busbar voltage is 1.1V determine I, the per unit current from the ratio of actual MVA to Base MVA.
- 7 The system voltage drop is IZ .
- 8 Calculate the motor terminal voltage.
- 9 From the m-g set driving motor slip-torque characteristic determine the value of torque at which the motor will stall. Adjust this value for the motor terminal voltage ($T \propto V^2$)
- 10 Determine whether this value is adequate for the shovel peak load under the electrical system conditions.

If the slip-torque characteristic for the m-g set driving motor is not available, the motor impedances may be used to calculate the motor output power and hence the torque developed by the motor may be calculated for a range of slip values. This calculation is shown in detail in Appendix 3.A. It is time consuming and requires arithmetic precision and is best carried out by computer. A suitable programme for use with a Hewlet-Packard 91008B computer is shown in Appendix 3.B.

Appendix 3.A illustrates clearly the need to consider the total electrical system in conjunction with the shovel peak load when selecting a loading shovel.

THYRISTOR DRIVE SHOVELS 310

Manufacturers have long made a plea for simpler excavator electrics claiming that the improved operating characteristics of expensive, sophisticated systems have minor effects on efficiency when allowance is made for mechanical maintenance. The ac motor driven eddy-current

coupling is a simpler alternative to the Ward Leonard system, but because of the field coils of the eddy current coupling, it is inherently slow in response, and it has high power consumption. The successful application of thyristor-controlled, adjustable speed drives in other industries, and their relative simplicity has attracted the attention of the mining engineer towards their application to open pit excavators. The advantages of thyristor controlled drives are:

1. Reduced maintenance. The use of thyristors in place of motor-generator sets greatly reduces bearing, commutator and brush maintenance.
2. Reduced power consumption. The overall efficiency of motor-generator sets is 80-87% against 96-98% for thyristor controls, resulting in a significant reduction in kWh charge.
3. Shovel construction. Modern thyristor drives are light, compact and do not require special bedplate structures, eliminating space and mounting problems associated with motor-generator sets.
4. Control Characteristics. Because of the absence of field coils, thyristor drives have improved performance over Ward Leonard or eddy current coupling drives, resulting in faster response to the operator's controls and tighter control of current overshoot during bank stalls, thus reducing peak stresses on ropes, gears, motors, structures, etc.

5. Spares. Spare parts holding is reduced and spares are more readily available than for motor-generator set drives.
6. Noise and Vibration. Reductions in these result in greater operator comfort, better communication and improved conditions for maintenance and "trouble-shooting".
7. Trouble-shooting. Replaceable pull-out trays of modular design and fault indication panels assist trouble-shooting.
8. Voltage fluctuations. Thyristors do not create the large current inrush caused by starting motor-generator sets and avoid the associated voltage drop. It is possible to build-in additional thyristor capacity (at extra cost) to cater for the usual voltage fluctuations experienced in open pit mining operations (usually about 15%). For severe voltage drops, including complete failure, regenerative operation of a drive is not possible. It is essential to use dynamic braking or a fast acting mechanical brake on hoist drives.

Early thyristor drives in open pit mining were applied to the comparatively easy duty of blasthole drills and have proved highly successful. Initially however the cost for a shovel was higher than for equivalent conventional drives, since thyristors have little thermal storage capacity compared with rotating machines and must be liberally rated

to cater for the fluctuating nature of an excavator load. Recent developments, however in high power thyristors have improved their cost competitiveness, and the first four thyristor shovels, manufactured by Harnischfeger Corp. (P and H) went into service in late 1968. A further shovel with G.E. electrics went into service in Arizona in 1972.

Perhaps the greatest problem is in the control of power factor. At low speeds and high torques the dc current is high but the dc power is relatively low. On the ac side of the thyristor a high current is drawn at full voltage as there is a relatively fixed ratio between dc current and ac current. Since the power demand is not high, the wattless component of the ac current is very large, resulting in a poor instantaneous power factor. The power factor will swing throughout the cycle from near unity to very low values. The trailing cable R.M.S. heating effect will be more than doubled compared with a motor-generator set drive, so a larger cable cross-section and a stiffer power system is required. It is necessary to adopt a systems approach to determine relative costs in comparing the thyristor drive with other drives.

The average power factor can be improved by addition of permanently connected capacitors, but this could increase voltage regulation problems. Installation of an automatically regulated synchronous condenser would solve the problem but would also remove much of the advantage gained by eliminating the motor-generator set. Thyristor-controlled capacitors would provide coarse power factor correction, but the cost of installation could render it uneconomic.

It is important to recognise that the cost of electrical energy for shovel applications is usually based on a charge for kWh and for kVA maximum 15 minute demand or a kW maximum 15 minute demand, linked with power factor. The savings in kWh charge can possibly be eliminated by additional power factor charges.

Special attention must be paid to the design of the thyristor system and the dc drive motors to reduce the possibility of sparking due to the ripple content of the thyristor dc output; otherwise greatly increased brush and commutator maintenance can be incurred.

Mining engineers are concerned with cost/tonne of their product and it is essential to view the operation of the thyristor drive with cost in mind. The assessment must include all the savings/additional costs in:

- 1 Capital cost variations
- 2 Maintenance and downtime
- 3 Power costs
- 4 Spare parts holding
- 5 Trailing cable rating
- 6 Need to reinforce the mine power system.

The Hydrostatic Crowd Drive

A major shovel manufacturer recently introduced hydrostatic crowd and propel drives. The compactness of the hydrostatic drive is well known and requires little comment. Its application in propelling has obvious advantages. For crowding duties it has the advantage that rope reeving or chain drives are eliminated, being replaced by a pair of hydraulic hoses; the crowd drive being located at the saddle block.

The hydrostatic drive can also be "tuned" to avoid excessive overloads and to smooth out the system peaks. The system is cheaper in initial cost but of course is much less efficient due to energy losses in the hydraulic system. Crowding however takes place during a relatively short part of a shovel cycle and the loss of efficiency is not so serious.

To date no opportunities for field trials have been available to the author to forecast the eventual role of the hydrostatic crowd drive shovel.

LOADING SHOVEL COSTS

In normal circumstances the truck costs of a shovel/truck operation are by far the major part of the total system costs (see Figs. 3.14 and 3.15). The shovel must therefore be selected to ensure that the truck system operates at minimum cost, i.e. the shovel cannot be considered in isolation. This can be best achieved by Monte Carlo simulation methods which although do not optimise the situation, can be used to select an optimum solution.

The minimum cost solution can be partially accomplished as follows:

- a) by correctly matching the size of the trucks and the shovel. This involves costing a range of shovel/truck combinations. This will minimise truck waiting time. Normally for maximum efficiency the truck should be loaded in 3 or 4 passes of the shovel;

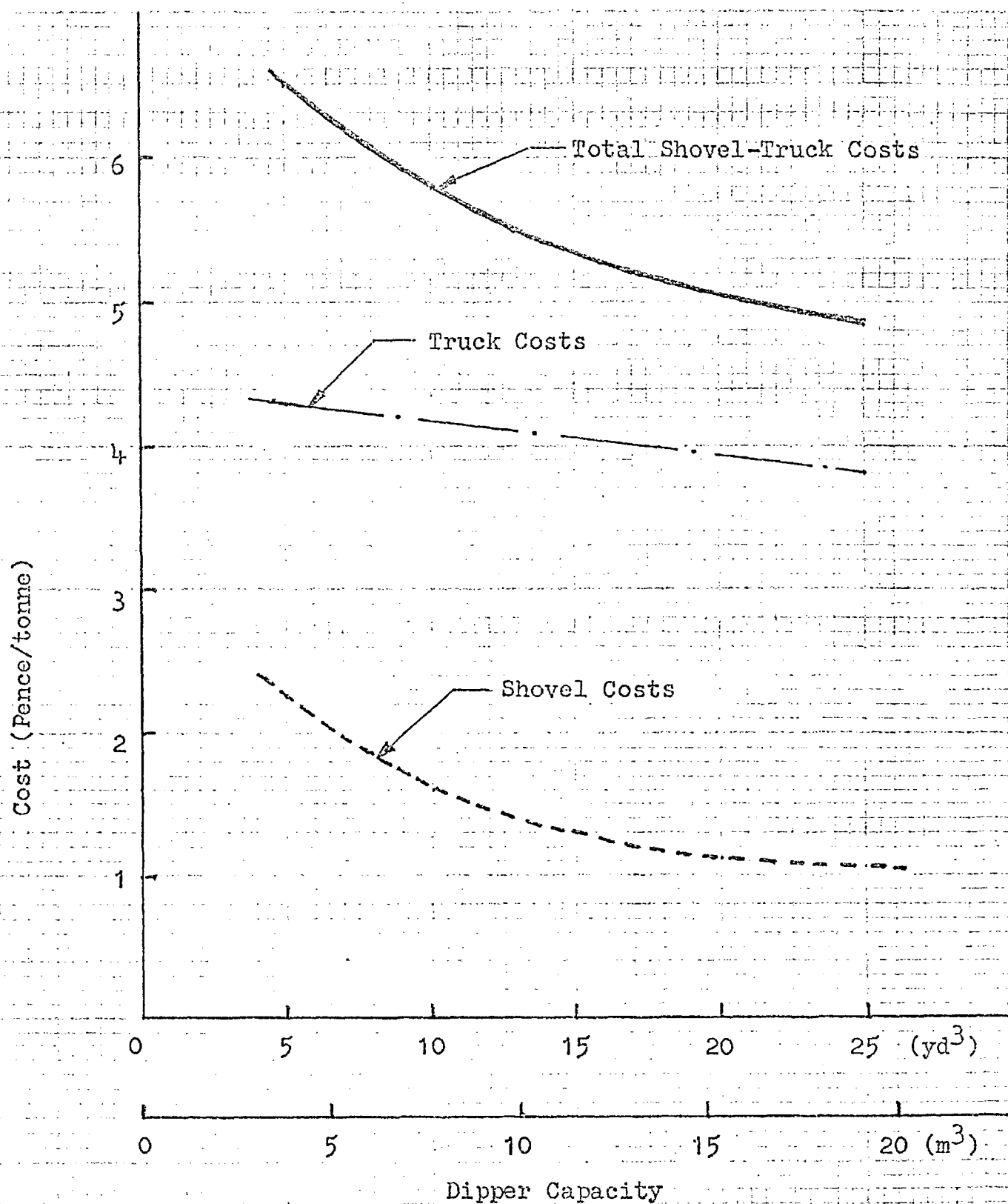


FIG. 3.14 SHOVEL-TRUCK COSTS BASED ON 100 TON TRUCKS

(Data collected mainly from Canadian sources,
corrected for haul distance)

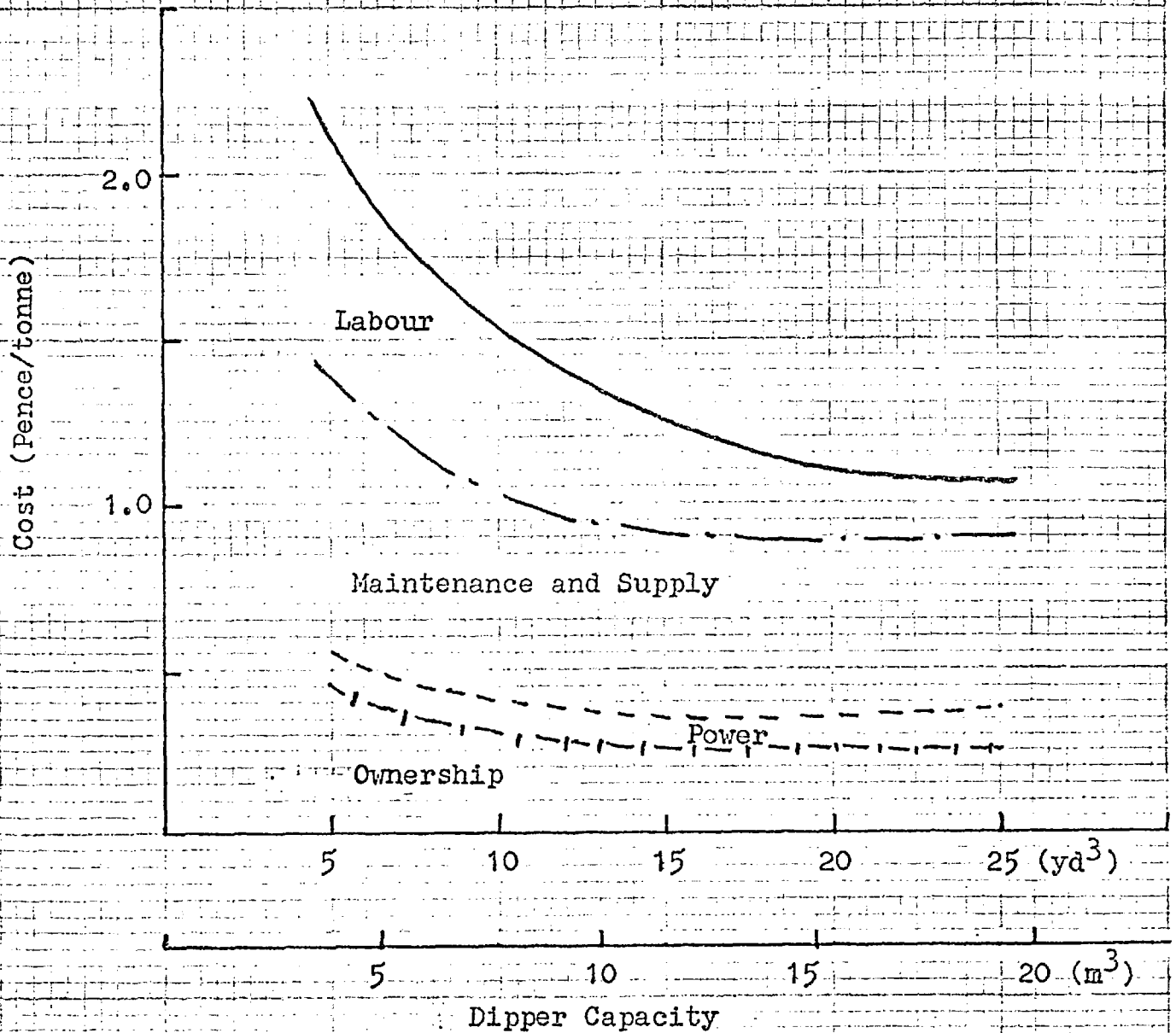


FIG. 3,15 SHOVEL COSTS

(Data collected mainly from Canadian sources)

- b) by using a shovel with adequate range to distribute the load throughout the truck body. Here the loading shovel with the boom and dipper handle arrangement has a distinct advantage over the wheel loader (front end loader) arrangement; and
- c) by reducing the shock loads imposed on the trucks. Here again the shovel has the distinct advantage over the wheel loader, in that the shovel dipper can be lowered well into the truck container before emptying, thereby greatly reducing the shock load.

The relationship between shovel/truck match, fragmentation and waiting times is shown in Chapter 1.

Additionally to a), b) and c) a high shovel availability is essential to secure maximum working of the truck fleet. To obtain this it is necessary to select designs which facilitate disassembly and reassembly, e.g. pin-connected dippers, saddle liners designed for quick adjustment or complete sub-assembly replacement, etc.

Ownership costs

Many shovels have a useful life in excess of 20 years, but it is unusual for this figure to be assumed in any assessment of depreciation. Individual company depreciation and replacement policies may determine the depreciation period, and advantage may be taken of tax concessions to write off a machine over a shorter period.

In addition to the FOB price, freight and insurance costs, erection costs, insurance during erection and interest charges up to the time of going into production must be included.

The following list is intended as a guide for the calculation of ownership costs.

- (1) FOB machine price, including optional extras,
sales taxes, etc. £....
- (2) Freight and insurance (to site) £....
- (3) Import duty £....
- (4) Sub-total £....
- (5) Ballast, if locally manufactured
(allow £40/t) £....
- (6) Erection costs (see Table 3.VIII) £....
- (7) Insurance during erection £....
- (8) Sub-total £....
- (9) Interest up to start of production
(allow interest on 30% of sub-total 8) £....
- (10) Sub-total £....
- (11) Shovel write-off period
n = ...years (11a)
...hours/year (11b)
...total hours (11c)
- (12) Machine depreciation and amortization

$$\text{Cost/h} = \frac{(\text{Sub-total 10})}{(\text{Item 11c})} = \text{£....}$$
- (13) By use of average investment formula,
assuming depreciation charges replace
original investment
Average machine investment

$$= \frac{(\text{Sub-total 10}) \times (n + 1)}{2n} = \text{£....}$$

(14) Interest rate%

(15) Insurance%

(16) Taxes, etc.
(if any)%

(17) Total%

(18) Interest, taxes, insurance, etc.
Costs/h

$$\text{Machine} = \frac{(\text{Total}\%) \times (\text{Item 13})}{(\text{Item 11b})} = \text{£}....$$

(19) Trailing cable costs
Capital cost + import duty
+ insurance up to start = £....
of production (see Table 3.IX)

(20) Trailing cable life
.....years (20a)
.....h/year (20b)
.....total h (20c)

(21) Trailing cable depreciation

$$\text{Cost/h} = \frac{(\text{Item 19})}{(\text{Item 20c})} = \text{£}....$$

(22) Trailing cable, average investment
= $\frac{(\text{Item 19}) \times (\text{Item 20a} + 1)}{2 \times (\text{Item 20a})}$ = £....

(23) Trailing cable interest, taxes,
insurance, etc.
Cost/h = $\frac{(\text{Item 22}) \times (\text{Item 17})}{(\text{Item 20b})}$ = £....

(24) Total ownership costs/h
(Item 12) + (Item 18) +
(Item 21) + (Item 23) = £....

TABLE 3.VIII
LOADING SHOVEL ERECTION COSTS
(1973 Projections)

Dipper Capacity		Erection Costs
yd ³	m ³	£
5	4	2500
8	6	3800
10	8	5400
15	12	6500
25	20	*

*Inadequate data: manufacturers figures must be obtained.

The above figures include the cost of supervision of erection by the manufacturer, i.e. time, subsistence, local transport, etc., skilled and unskilled labour, erection equipment, i.e. cranes, tools, slings, etc., workshop costs, etc., but not the travel expenses of the manufacturer's erectors. The costs should be adjusted to take account of:

- (a) distance from port of entry, railhead, road, etc., from erection site; off-loading facilities, availability of heavy transport, etc.;
- (b) availability of skilled labour
- (c) quality of mine supervisory staff; and
- (d) availability of cranes, tools, workshop services, etc.

Trailing cable costs may conveniently be costed on a linear basis, Table 3.IX however is intended for preliminary estimating purposes only and final costs must be based on the actual cable prices.

TABLE 3.IX

TRAILING CABLE PRICES - LOADING SHOVELS (£/m)

Dipper Capacity		Voltage, kV		
yd ³	m ³	3.3	6.6	11
5 - 10	4 - 8	4.5	3.5	3.6
10 - 15	8 - 12	8.4	6.8	5.5
15 - 25	12 - 20	-	9.3	7.6

Operating Costs

(25) Maintenance and supply costs/h.

$$= \frac{10\% \times (\text{Sub-total 4}) \times (\text{H}(\text{Table 3.X}) \times (\text{M}(\text{Table 3.XI}))}{(\text{Item 11b})}$$

= £....

(26) Electrical power consumption/h x
cost/kWH
(see Table 3.XII)

= £....

(27) Labour rate/h (to include social
benefits, taxes, insurance, etc.)

= £....

Note where there are no figures
available from other mining operations
in the area, rates based on local wage
rates are invariably too low. Opening
a new mine usually stimulates the local
economy and local wage rates often
double within one year. The social
benefits must be carefully investigated
as these can be as high as 250% of the
monetary wages.

(28) Total operating costs/h

$$(\text{Item 25}) + (\text{Item 26}) + (\text{Item 27})$$

= £....

Total Ownership and Operating costs

(29) Total ownership + Operating Costs

$$= (\text{Item 24}) + (\text{Item 28})$$

= £.... /h

$$\text{cost/t} = \frac{(\text{Item 29})}{\text{t/h}}$$

= £.... /t

TABLE 3.X

CORRECTION FACTOR H FOR HOURS
OF OPERATION

<u>h/year</u>	<u>H</u>
Up to 3400	1.0
3401 - 4500	1.15
4501 - 5000	1.20
5001 - 5500	1.30
5501 - 6000	1.35
6001 - 6500	1.40
6501 - 7000	1.45
Above 7000	1.50

Values of H have been derived from a large volume of figures collected by the methods described in Chapter 2 and from figures provided by manufacturers (mainly Marion Power Shovel Co.)

TABLE 3.XI

CORRECTION FACTOR M FOR DIFFERENT
MATERIALS

<u>Material</u>	<u>M</u>
Coal	0.70
Sand	0.85
Clay	0.90
Limestone	1.00
Copper Ores	1.05
Hematite	1.00
Magnetite	1.05
Granites	1.20
Taconite	1.35

The abrasive properties and density of the materials must be carefully considered.

Compiling this table presented some difficulty since the descriptions "copper ores", "limestone", "granite", etc. are imprecise making analysis of operators figures prone to inaccuracies.

TABLE 3.XII

ELECTRIC POWER CONSUMPTION - LOADING SHOVELS

Dipper Size		Consumption	(kWh/h)*
yd ³	m ³	(a)	(b)
5	4	110	85
8	6	260	160
10	8	355	200
15	12	450	250
20	15	670	370
25	19	900	500

(a) Good conditions, high operating efficiency

(b) Bad conditions, poor operating efficiency

* Multiply by 2.2 for Eddy-current coupling hoist drive.

FURTHER STUDIES

The foregoing work has certain gaps that can be filled by further practical and theoretical studies, which although beyond the scope of this thesis, are of importance in the systems approach to excavator selection. Perhaps the most obvious of these concerns electricity consumption.

Electricity Consumption.

Contrary to generally held opinion high electricity consumption is associated with well fragmented ground. High production is obtained and a high kW developed when the operator is able to dig at voltages (speeds) in the peak power range (see Fig. 3.8) i.e.

- 60% stall current and 550V to
- 70% stall current and 550V.

Bad fragmentation requires low speeds (low voltage) and a very high torque (high current, approaching stall conditions), with a low kW developed. This has been allowed for in the Chapter (see Table 3.XII). The other work described indicates how the electricity supply arrangements must be fitted into the systems approach to ensure satisfactory and economic performance. It does not however fully cover the true power consumption costs of the loading shovel.

Power costs are a relatively small part of total shovel costs (see Fig 3.15), but of course have an important role in electrical system design, which lies in the realm of the electrical engineer. The treatment provided here is adequate for initial selection purposes.

To cover this gap it is recommended that further simulation studies based on a loading shovel of standard configuration, be carried out to determine:

- a) Maximum demand (15 minute period)
- b) Maximum demand (30 minute period)
- c) kWh consumed, and
- d) Power swing frequency charges,

for conventional Ward Leonard, eddy current coupling and thyristor drives and possibly for hydrostatic crowd drives. It is proposed to include this work in a "Handbook of Open Pit Electrical System Design" which will cover most electrical design aspects of open pit operations.

Electrical System Reinforcement

It has been indicated that the RMS heating effect in cables supplying thyristor drive shovels is much greater than for other electric drive shovels, and that some electrical system reinforcement is required. At present this is done on a fairly localised basis with little reference to the overall electrical system, and the true costs of this reinforcement are not precisely determined. This should also be the subject of further theoretical studies and it is proposed to include these in the "Handbook of Open Pit Electrical System Design".

Shovel Systems

In 1969 the author suggested that "fundamental investigations into the actions and performance of excavating machinery to provide a more basic analysis appear necessary"³¹¹, and pinpointed the work of Druce and Andrews³¹² on walking draglines as an example. This need is still apparent. Much of the work must be based on field studies and can probably only be performed by mine operators. The processing of the data obtained from field studies however may be beyond the resources of the mine operators and co-operation with university departments or similar research organisations is indicated. Additionally the author stated in 1969 that "much of this information may have already been determined by individual manufacturers but little appears to

have been published or made available to those concerned with mining operations". A recent study undertaken by Bucyrus Erie provides an interesting example.

The tubular dipper handle and rope crowd mechanism of Bucyrus Erie are well known. The freedom of the dipper handle to partially rotate eliminates serious torsional stresses during the digging motion caused by one side of the dipper lip entering hard ground. More recently they have added a cushioned sheave assembly at the rear of the dipper handle for a single rope used for crowding, to equalise the crowd rope loads and reduce impact loads. The cushion is formed of high-energy rubber pads, used normally for truck rear suspensions.

Operating data was collected in extensive field investigations. The data were formulated so that the measured (true) damping characteristics of the driving and rope systems could be integrated into computer simulation programmes. This provided a working model for the development and refinement of the various systems, enabling the various parameters, in the crowd drive system, e.g. gear ratio, gear and shaft inertias, component deflections, spring constants and damping constants to be optimised.

Fig 3.16 ³¹³ shows the correlation between field tests and simulated crowd drives. The results indicate that with a cushioned sheave an increase in crowd rope life will be gained. The impact energy damping characteristics of the rubber cushions should also increase service life due to reduced mechanical and electrical impact loading.

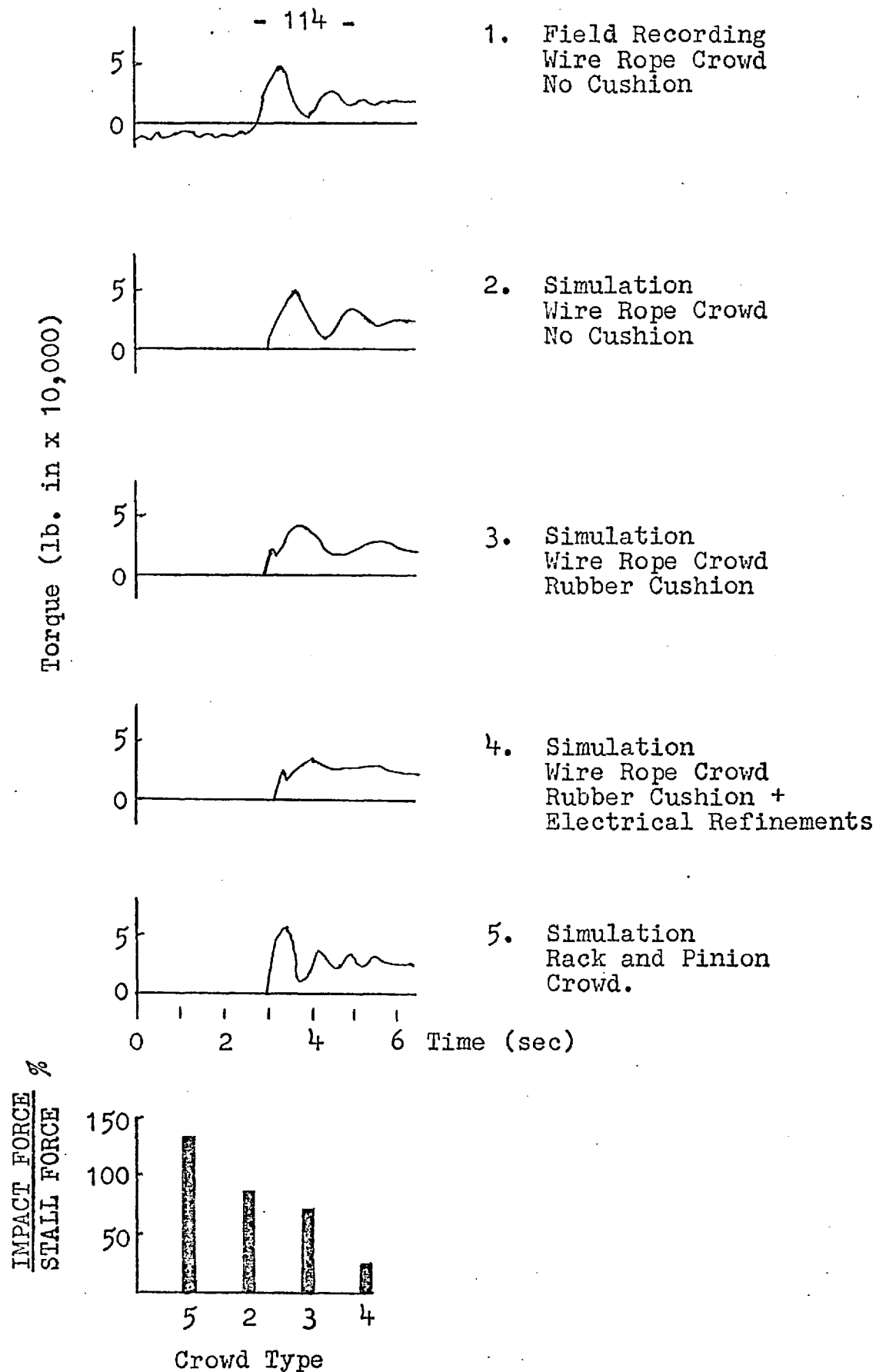


FIG. 3.16 SIMULATED CROWD DRIVES AND IMPACT OVERLOADS

The provision of a simulation model of a shovel would allow those involved in selection to compare the various mechanical characteristics of individual shovels. The model would have to be based on extensive field studies if it is to be realistic. It is also possible that an analogue simulation model would be more suitable than the use of digital computers. Such a model could verify many opinions held by mining engineers which are difficult to quantify and could also expose some well advertised shovel features as "gimmicks".

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APPENDIX 3.A

THE PER UNIT SYSTEM

The performance of a whole system of electrical apparatus, regardless of size, can often be expressed by a single set of constants when those constants are expressed as percentages. By this it is meant that the loss will be a certain percentage of its kW rating, its voltage drop (or more correctly voltage regulation) a certain percentage of voltage rating, etc. The advantage of this method of representation is that it provides a better comparison of performance of apparatus of different rating. A 100V drop in a transmission line has no significance until the voltage base is given, whereas, a percentage drop would have much significance.

A disadvantage of the percentage system is the confusion that results from the multiplication of percentage quantities. Thus a 20% current flowing through a 40% reactance would by simple multiplication give 800% voltage drop, whereas the correct answer is an 8% voltage drop. The PER UNIT SYSTEM (see Westinghouse Electrical Transmission and Distribution Reference Book, Pittsburgh, 1950) has all the advantages of the percentage system but avoids this disadvantage. In this system the rating quantity is regarded as unity. Any other quantity is expressed e.g. 1.05 or 0.92, etc.

Both the Per Unit and the percentage systems have the advantage of eliminating troublesome coefficients. This is a mixed blessing however as a definite disadvantage of both systems lies in the loss of the dimensional check.

VOLTAGE

Per Unit volts = O/1V, put all voltages on a COMMON base so that a per unit (O/1) change can be converted to volts easily from the O/1 value and the BASE quantity

$$O/1 V = \frac{\text{Volts}}{\text{Base Volts}}$$

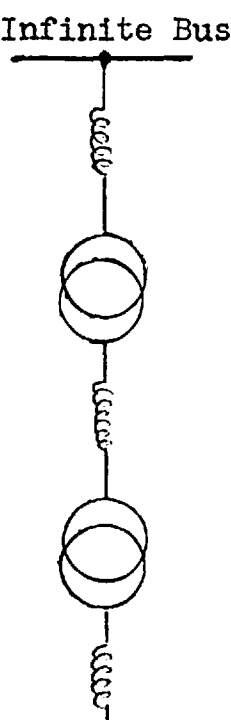
A system will have a different base volts whenever there is a transformer ratio change that provides different voltage levels.

$$O/1 V = \frac{E_1}{E_1 \text{Base}} = \frac{E_2}{E_2 \text{ Base}} = \frac{E_3}{E_3 \text{ Base}} \text{ etc.}$$

$$\text{therefore } E_{2\text{Base}} = \frac{E_2}{E_1} \cdot E_1 \text{Base}$$

$$\text{and } \frac{E_2}{E_1} = \text{Transformer Ratio} = t$$

AT NO-LOAD the O/1V is the same throughout the system

	NO-LOAD VOLTS	BASE VOLTS	$\frac{O/1V = \text{NO-LOAD VOLTS}}{\text{BASE VOLTS}}$
	E_o	E_o	$\frac{E_o}{E_o} = 1.0$
E_1	$t_1 \cdot E_o$	E_1	$\frac{t_1 \cdot E_o}{E_1} = \frac{E_1}{E_1} = 1.0$
E_2	$t_2 \cdot E_1$	E_2	$\frac{t_2 \cdot E_1}{E_2} = \frac{E_2}{E_2} = 1.0$

Given the transformer ratios, t_1 , t_2 , t_3 , etc. and $E_{1\text{Base}}$ (or any other E_{Base}) in the system then all E_{Base} can be fixed.

If the transformers have tap settings these must be included, if used in the transformer ratios, t_1 , t_2 , t_3 , etc.

Let $E_{\text{Base}} = 6.9 \text{ kV}$

If volts = 6.9kV, then $O/1V = \frac{6.9}{6.9} = 1.0$

If volts = 6.75kV, then $O/1V = \frac{6.75}{6.9} = 0.978$

If volts = 7.09kV, then $O/1V = \frac{7.09}{6.9} = 1.028$
, etc.

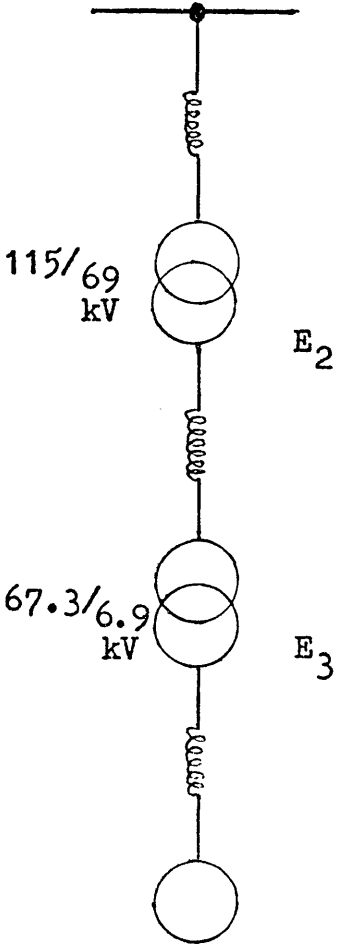
In any system, to compute no-load volts it is necessary to calculate from the source.

To compute each BASE VOLTS, it is necessary to work from the VOLTAGE SELECTED AS BASE VOLTS.

For open pit mining systems it is usually convenient to select the M/G Set DRIVING MOTOR VOLTAGE of the loading shovels (or other excavators) as BASE VOLTS.

Example 1

see over

		NO-LOAD VOLTS	BASE VOLTS	O/1V
	$E_1 = 119\text{kV}$	119kV	107.2kV	$\frac{119}{107.2} = 1.11$
		↓	↑	
	E_2	71.4kV	64.4kV	$\frac{71.4}{64.4} = 1.11$
		↓	↑	
	E_3	7.32kV	6.6kV	$\frac{7.32}{6.6} = 1.11$
		↓	↑	
	$E_m = 6.6\text{kV}$	7.32kV	6.6kV	$\frac{7.32}{6.6} = 1.11$

For No-Load Volts - START FROM SOURCE

$$E_2 = \frac{69}{115} \times 119 = 71.4 \text{ kV}$$

$$E_3 = \frac{6.9}{67.3} \times 71.4 = 7.32 \text{ kV}$$

$$\text{and } E_m = 7.32 \text{ kV}$$

For BASE VOLTS - use MOTOR RATED VOLTAGE

$$E_m \text{ Base} = 6.6 \text{ kV}$$

$$E_3 \text{ Base} = 6.6 \text{ kV}$$

$$E_2 \text{ Base} = \frac{67.3}{6.9} \times 6.6 = 64.4 \text{ kV}$$

$$E_1 \text{ Base} = \frac{115}{69} \times 64.4 = 107.2 \text{ kV}$$

CURRENT

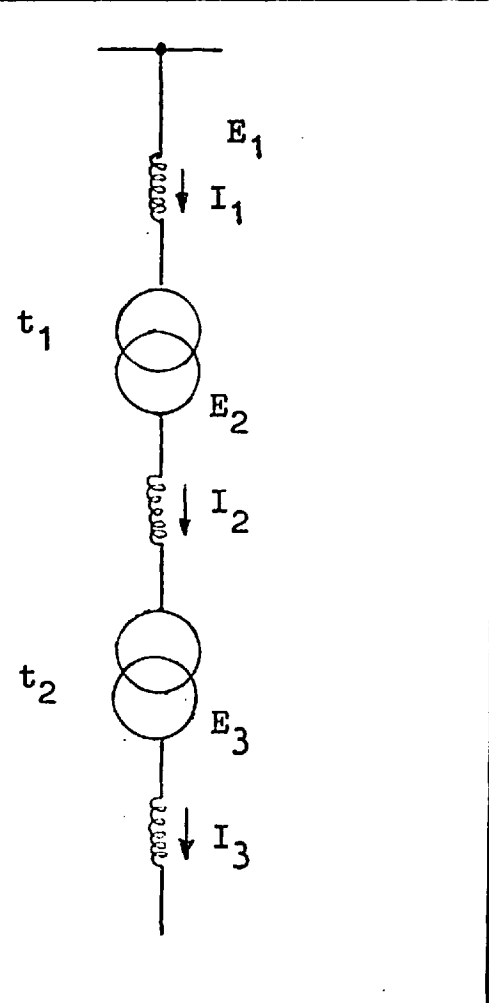
If a BASE CURRENT is selected, a different base current must be selected for each transformer ratio change that produces a voltage change.

A simpler approach is to use a MVA BASE. This allows simple calculation of amperes and eliminates the need to allow for transformer ratios.

BASE MVA is the same in all parts of the system even though base volts may be different throughout the system.

$$\text{Base MVA} = (\text{Base kV}) (\text{Base I}) \times 10^{-3} \times \sqrt{3}$$

If I_1 is selected to be Base I in that part of the circuit, then $I_{2\text{Base}} = t_1 \cdot I_1 = I_2$

	Base I	Base kV	Base MVA
	I_1	E_1	$\sqrt{3} E_1 I_1 \cdot 10^{-3}$
	$\frac{I_1}{t_1}$	$t_1 \cdot E_1$	$\frac{t_1 \cdot I_1}{t_1} E_1 \cdot \sqrt{3} \cdot 10^{-3}$ $= \sqrt{3} E_1 I_1 \cdot 10^{-3}$
	$\frac{I_2}{t_2}$	$t_2 \cdot E_2$	$\frac{t_2 \cdot t_1}{t_2} \frac{E_1}{t_1} I_1 \sqrt{3} \cdot 10^{-3}$ $= \sqrt{3} E_1 I_1 \cdot 10^{-3}$

$$I_{3\text{Base}} = \frac{I_2}{t_2} = \frac{I_1}{t_2 \cdot t_1}$$

If E_1 is selected to be the BASE kV in that part of the system,

then

$$E_{2\text{Base}} = t_1 \cdot E_1 = E_2$$

and

$$E_{3\text{Base}} = t_2 \cdot E_2 = t_1 \cdot t_2 E_1$$

OHMS

If Base Volts and Base Current are selected this defines

BASE OHMS

$$\text{Base Volts} = \sqrt{3} (\text{Base Current})(\text{Base Ohms})$$

$$\text{Base Ohms} = \frac{\text{Base Volts}}{\sqrt{3} (\text{Base Current})}$$

Also

$$\text{Base Ohms} = \frac{\text{Base Volts}}{\sqrt{3} (\text{Base Current})} \times \frac{\text{Base kV}}{\text{Base kV}}$$

$$= \frac{(\text{Base Volts}) (\text{Base kV})}{\text{Base kVA}}$$

therefore

$$\text{BASE OHMS} = \frac{(\text{BASE kV})^2}{\text{BASE MVA}}$$

$$\text{PER UNIT OHMS} = \frac{\text{Ohms}}{\text{Base Ohms}} = 0/1 \text{ Ohms}$$

$$= \frac{\text{Ohms}}{\frac{(\text{Base kV})^2}{\text{Base MVA}}}$$

therefore:

$$0/1 \text{ Ohms} = \frac{(\text{Ohms})(\text{Base MVA})}{(\text{Base kV})^2}$$

The system single line diagram must show transformer ratios.

- a) Select Base kV (motor rated voltage)
- b) Select Base MVA (motor rated MVA or any other convenient figure)
- c) Solve for O/1 OHMS in each part of the system.

NB For 3 phase systems use ohms/phase values

Some equipment, e.g. transformers, have PER UNIT or Percentage values expressed to their own base. These may be converted to the selected base as follows:

It must be remembered that the Ohmic Value of any component in the system does not change, regardless of Base MVA or Base kV

$$\text{O/1 Ohms} = \frac{(\text{Ohms})(\text{Base MVA})}{(\text{Base kV})^2}$$

$$\therefore \text{Ohms} = \frac{(\text{O/1 Ohms})(\text{Base kV})^2}{\text{Base MVA}}$$

Since the Ohmic Value does not change:

$$\frac{(\text{O/1 Ohms}_1)(\text{Base kV}_1)^2}{\text{Base MVA}_1} = \frac{(\text{O/1 Ohms}_2)(\text{Base kV}_2)^2}{\text{Base MVA}_2}$$

Example 2

Transformer - 7.5 MVA, 6.9 kV with impedance of 0.07 O/1 on 7.5 MVA, 6.9 kV base (its own rating)

Find O/1 Ohms on Base MVA = 10.0

Base kV = 6.6

Then

$$\frac{0.07 (6.9)^2}{7.5} = \frac{(\text{O/1 ohms})(6.6)^2}{10.0}$$

$$\text{O/1 ohms} = \underline{0.102 \text{ O/1}} \rightarrow$$

If the impedance is expressed as a percentage e.g. 11%, the per unit value is simply $\frac{11}{100} = 0.11$ O/1

Often the impedances from source to the metering point are not known, but the FAULT LEVEL (MVA_{sc}), at the metering point is generally known. To calculate the total impedance.

$$\begin{aligned} \text{O/1 ohms} &= \frac{\text{O/1V}}{\text{O/1I}} = \frac{(\text{O/1V})(\text{O/1 kV})}{(\text{O/1I})(\text{O/1 kV})} \\ &= \frac{(\text{O/1 V})(\text{O/1 kV})}{\text{O/1 kVA}} = \frac{(\text{O/1 kV})^2}{(\text{O/1 MVA})} \end{aligned}$$

∴ O/1 Ohms of System from source to metering point

$$\begin{aligned} &= \frac{(\text{O/1 kV})^2}{\frac{MVA_{sc}}{\text{Base MVA}}} \\ &= \frac{(\text{O/1 kV})^2 \times (\text{Base MVA})}{MVA_{sc}} \end{aligned}$$

Example 3

Fault Level at metering point = 150 MVA

Base MVA = 10 MVA

$$\text{O/1 ohms (System)} = (1.0)^2 \times \frac{10}{150}$$

$$= \underline{0.066 \text{ O/1}}$$

The foregoing is valid using the following assumptions

- a) Infinite Busbar or Source Volts = 1.0 O/1 for the duration of the fault. This is not always true but is acceptable.

- b) Use the MINIMUM fault level to obtain the MINIMUM impedance. Some systems have varying impedances depending on which circuits are switched in. For voltage drop calculations, the MAXIMUM impedance is required.
- c) Zero power factor (no resistance) is assumed in the fault. The impedance is therefore entirely Reactance. If R and X values are available it is of course preferable to use them.

A further example is where a motor is connected to a system.

Example 4

System based on 0/1 V at Source

Motor rating = 6.0 MVA

Starting current inrush = 450% full load current
= 4.5 0/1 I

On motor base

$$0/1 \text{ ohms} = \frac{(1.0)^2}{4.5} \times \frac{6.0}{6.0}$$

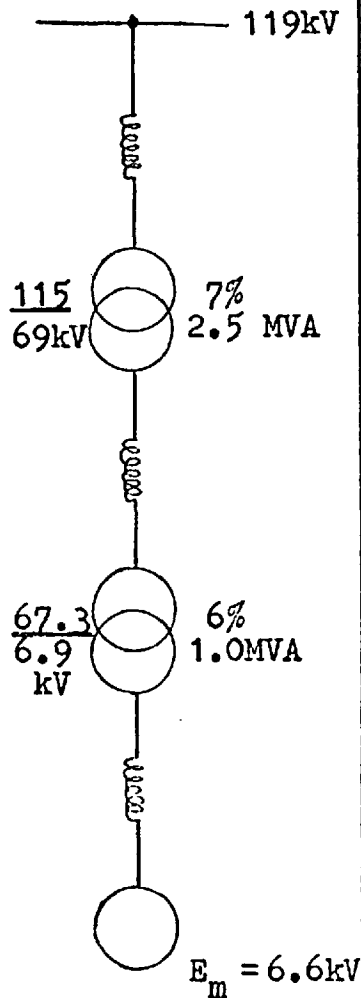
$$\underline{0/1 \text{ ohms} = 0.222 \text{ 0/1 on motor base}}$$

On 10 MVA base

$$\frac{0.222}{6.0} = \frac{0/1 \text{ Ohms}_2}{10.0}$$

$$\underline{0/1 \text{ Ohms} = 0.37 \text{ 0/1 on 10 MVA Base}}$$

Using Example 1



R Ohms/phase	O/1 R	X Ohms/phase	O/1 X
		Fault level = 250 MVA	0.0028
			0.0226
0.05	0.000084	0.008	0.0000135
			0.042
0.25	0.004	0.07	0.0011
Total	0.0041		0.0685

$$O/1 Z = 0.0041 + j 0.0685 = 0.0686 \quad O/1$$

For a typical shovel load the cyclic peak load is about 2.0 x motor rated load (the actual figure would normally be used).

$$O/1 \text{ Voltage drop} = I Z$$

$$O/1 I = 2.0$$

$$\therefore O/1 \text{ Volt drop} = 2 \times 0.0686 = 0.1372 \quad O/1$$

$$O/1 \text{ Motor Voltage at no-load} = 1.11 \quad O/1$$

$$O/1 \text{ Motor Voltage at peak load} = 1.11 - 0.1372$$

$$= 0.9728 \quad O/1$$

NEMA motors are required to provide their full load power at 95% motor rated voltage.

The m/g set driving motor must however be able to provide the maximum cyclic driving torque i.e. at peak load. To check this it is necessary to know the MOTOR STALLING TORQUE or to have the SLIP~TORQUE characteristic of the motor.

Typically, m-g set driving motors are selected to have a stalling torque of 2.5 x full load torque.

In the above example the motor stalling torque at peak load is:

$$0.9728^2 \times 2.5 = 2.37 \text{ O/1 Torque}$$

In this case the motor would operate satisfactorily. It must be remembered however that if the cyclic peak power is 2.0 x full load power, the peak torque is > 2.0 , since the slip increases with increased torque. For a positive check it is necessary to refer to the Slip~Torque Characteristic.

The peak torque is then determined as follows:

$$\text{O/1 Peak Torque} = \text{O/1 Peak Power} \times \frac{(1 - s_{f1})}{(1 - s_p)}$$

where s_{f1} = Slip at full load

s_p = Slip at peak load

The slip at peak load can only be determined approximately but the calculation can be repeated and a reasonably accurate figure for O/1 Peak Torque determined.

EQUIVALENT CIRCUIT METHOD

Where the m-g set driving motor parameters are available it is possible to use an iterative method of solution.

Fig 3.A1 shows the usual equivalent circuit of the induction motor.

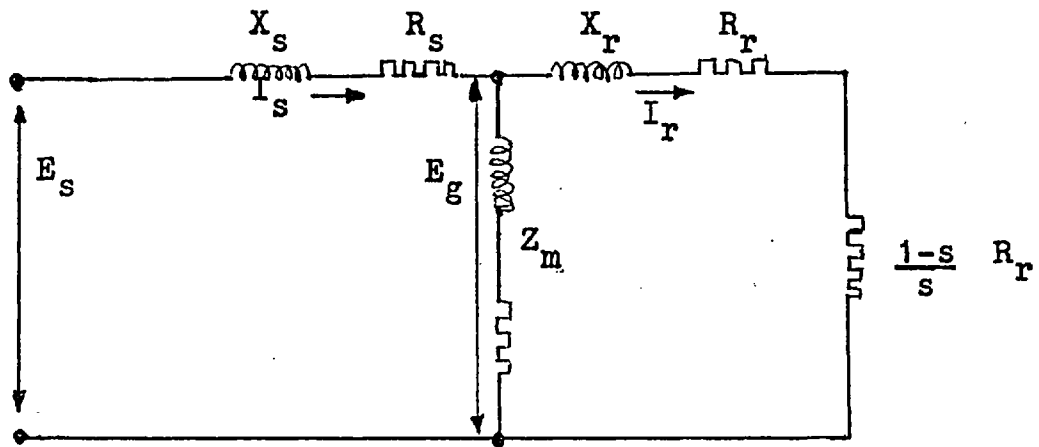


FIG 3.A.I EQUIVALENT CIRCUIT OF THE INDUCTION MOTOR

where

- R_s = stator resistance
- X_s = stator leakage reactance at rated frequency
- R_r = rotor resistance
- X_r = rotor leakage reactance at rated frequency
- Z_m = shunt impedance to include the effect of magnetising current and no-load losses
- E_s = applied voltage
- I_s = stator current
- I_r = rotor current
- s = slip

The justification for this diagram is briefly explained as follows: the air gap flux created by the currents I_s and I_r induces the voltage E_g in the stator and sE_g in the rotor. In the rotor the voltage drop due to the impedance is:

$$I_r R_r + js I_r X_r ,$$

since the reactance varies with the frequency of the rotor currents. The rotor current can therefore be determined from:

$$E_g = \frac{R_r}{s} I_r + jX_r I_r$$

It follows from this that the rotor circuit can be completely represented by placing a circuit of impedance $\frac{R_r}{s} + jX_r$ across voltage E_g . The total power absorbed by $\frac{R_r}{s}$ must be the sum of the rotor losses plus the useful shaft power, so that, resolving $\frac{R_r}{s}$ into resistance R_r and $\frac{1-s}{s} R_r$, the power absorbed by R_r represents the rotor copper loss. The power absorbed by $\frac{1-s}{s} R_r$ represents the useful shaft power.

The complete system can be represented by the following equivalent circuit:

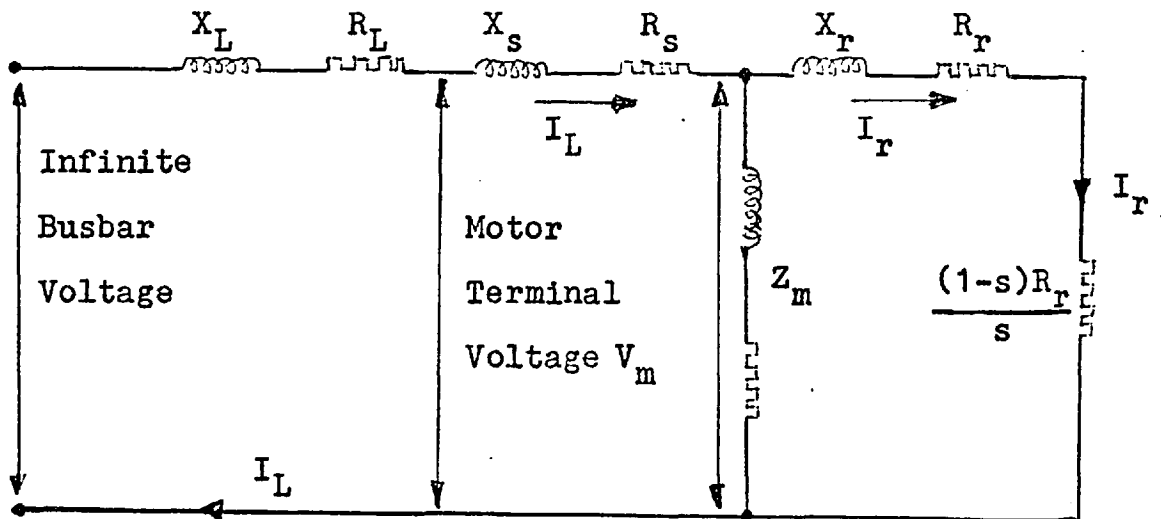


FIG 3.A2 SYSTEM EQUIVALENT CIRCUIT

where

X_L = Line reactance

R_L = Line resistance

I_r = Rotor current

I_L = Line current

For practical calculations Z_m is usually considered to a reactance X_m .

If the Busbar Voltage is known it is possible to solve for I_L using any value of slip.

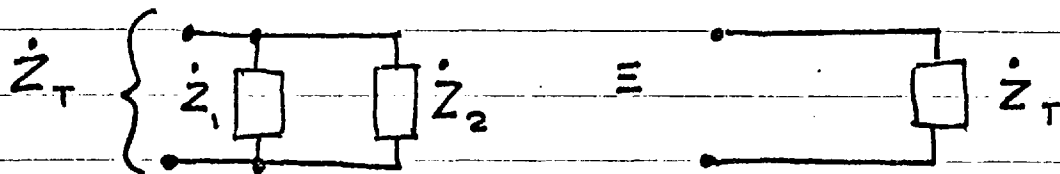
The useful shaft power =

$$I_r^2 \cdot R_r \cdot \frac{(1-s)}{s}$$

By iteration it is therefore possible to calculate I_r , motor output power, torque, line power, motor terminal volts, etc., by assuming a range of values for slip.

SOLUTION OF THE EQUIVALENT CIRCUIT

The equivalent circuit cannot be directly solved as the RHS (See Fig. 3A2) involves a parallel branch. This branch can be represented as a series circuit. By Kirchhoff's Laws:



$$\text{and } \frac{1}{\dot{Z}_T} = \frac{1}{\dot{Z}_1} + \frac{1}{\dot{Z}_2} + \frac{1}{\dot{Z}_3} + \dots + \frac{1}{\dot{Z}_n}$$

$\therefore$ for two parallel branches

$$\dot{Z}_T = \frac{\dot{Z}_1 \cdot \dot{Z}_2}{\dot{Z}_1 + \dot{Z}_2}$$

$\therefore$ For the system equivalent circuit (Fig. 3A2)

$$\dot{Z}_T = \frac{(0 + jX_m) \left(\frac{R_r}{s} + jX_r \right)}{(0 + jX_m) + \left(\frac{R_r}{s} + jX_r \right)} \quad \begin{array}{l} \text{since } X_m \gg R_m \\ \text{and } R_r + R_r \frac{(1-s)}{s} = \frac{R_r}{s} \end{array}$$

$$\dot{Z}_T = \frac{jX_m \left(\frac{R_r}{s} \right) - X_m X_r}{\frac{R_r}{s} + j(X_m + X_r)}$$

$$= \frac{(jX_m \left(\frac{R_r}{s} \right) - X_m X_r) \left(\frac{R_r}{s} - j(X_m + X_r) \right)}{\left(\frac{R_r}{s} \right)^2 + (X_m + X_r)^2}$$

$$= \frac{j X_m \left(\frac{R_r}{s}\right)^2 + X_m \frac{R_r}{s} (X_m + X_r) - \frac{R_r}{s} X_m X_r + j X_m X_r (X_m + X_r)}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

$$= \frac{\frac{R_r}{s} X_m (X_m + X_r) - \frac{R_r}{s} X_m X_r + j \left[X_m \left(\frac{R_r}{s}\right)^2 + X_m X_r (X_m + X_r) \right]}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

$$\therefore Z_T = \frac{\frac{R_r}{s} X_m (X_m + X_r) - \frac{R_r}{s} X_m X_r}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

$$+ j \frac{X_m \left(\frac{R_r}{s}\right)^2 + X_m X_r (X_m + X_r)}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

$$= \frac{\frac{R_r}{s} \cdot X_m^2}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2} + j \frac{X_m \left[\left(\frac{R_r}{s}\right)^2 + X_r (X_m + X_r) \right]}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

$$\therefore R_T = \frac{\frac{R_r}{s} \cdot X_m^2}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2} \quad \text{and}$$

$$X_T = \frac{X_m \left[\left(\frac{R_r}{s}\right)^2 + X_r (X_m + X_r) \right]}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

and $Z_T = \sqrt{R_T^2 + X_T^2}$

Z_T is required to solve for I_L

Z_T is required to solve for V_1

For Fig. 3A2 the total circuit resistance:

$$R = R_L + R_s + \frac{R_r \cdot X_m^2}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2}$$

and the total circuit reactance:

$$X = X_L + X_s + X_m \left[\frac{\left(\frac{R_r}{s}\right)^2 + X_r(X_m + X_r)}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2} \right]$$

Total Circuit Impedance, $Z = \sqrt{R^2 + X^2}$

$$Z = \left[\left(R_L + R_s + \frac{R_r \cdot X_m^2}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2} \right)^2 + \left(X_L + X_s + X_m \left[\frac{\left(\frac{R_r}{s}\right)^2 + X_r(X_m + X_r)}{\left(\frac{R_r}{s}\right)^2 + (X_m + X_r)^2} \right] \right)^2 \right]^{1/2}$$

The per unit voltage at the infinite busbars is assumed to be 1.1

$$\therefore I_L = \frac{1.1}{Z}$$

$$I_r = \frac{V_1}{Z_r} \quad \text{and} \quad V_1 = I_L Z_T$$

$$\therefore I_r = \frac{I_L Z_T}{Z_r} \quad \text{and} \quad Z_r = \sqrt{\left(\frac{R_r}{s}\right)^2 + X_r^2}$$

$$\therefore I_r = \frac{I_L Z_T}{\sqrt{\left(\frac{R_r}{s}\right)^2 + X_r^2}}$$

$$\text{Motor Output Power} = P_m = I^2 R_r \frac{(1-s)}{s}$$

$$\text{Motor Torque} = \frac{P_m}{1-s}$$

Motor Terminal Voltage

Motor resistance as viewed from the motor terminals =

$$R_i = R_T + R_s$$

Motor reactance as viewed from the motor terminals =

$$X_i = X_T + X_s$$

$$Z_i = \sqrt{R_i^2 + X_i^2}$$

$$\therefore \underline{V_m = I_L Z_i} \rightarrow (\text{per unit value})$$

$$\underline{\text{Line kVA} = 1.1 I_L}$$

This procedure is best illustrated by a practical example.

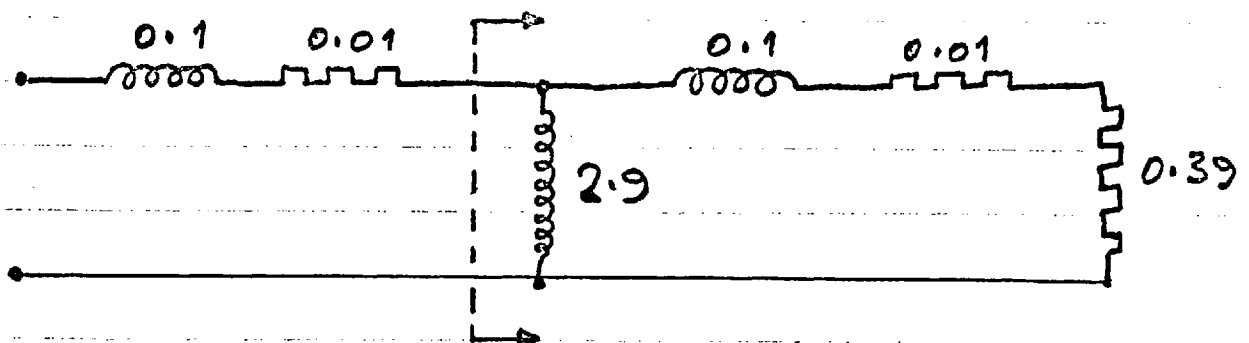
Example 2

Let

$$\begin{aligned} X_s &= 0.1 \\ R_s &= 0.01 \\ X_r &= 0.1 \\ R_r &= 0.01 \\ X_m &= 2.9 \\ R_m &\text{ is ignored} \\ s &= 0.025 \quad (2.5\% \text{ slip assumed}) \end{aligned}$$

all are per unit values typical of a 15 yd³ shovel

$$\text{Then } \frac{1-s}{s} R_r = \frac{0.975}{0.025} \times 0.01 = 0.39$$



Resolving the RH part of the circuit.

$$\underline{Z} = \frac{(0 + j2.9)(0.40 + j0.1)}{(0 + j2.9) + (0.40 + j0.1)}$$

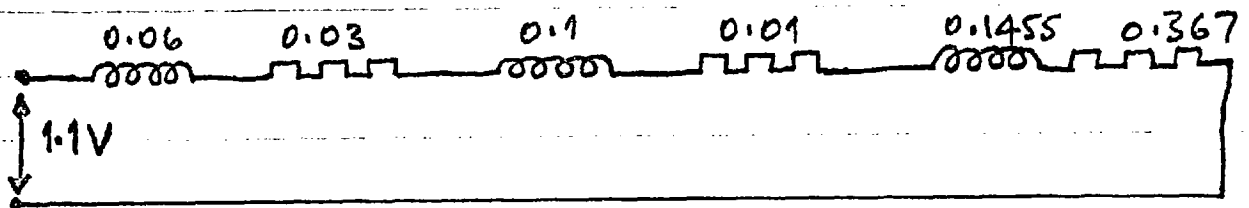
$$= \frac{j1.16 + j^2 0.29}{0.4 + j3.0} = \frac{j1.16 - 0.29}{0.4 + j3.0}$$

$$= \frac{(j1.16 - 0.29)(0.4 - j3.0)}{0.4^2 + 3.0^2}$$

$$= \frac{j0.464 - j^2 3.48 - 0.116 + j0.87}{9.16}$$

$$= \frac{3.364 + j1.334}{9.16} = 0.367 + j0.1455$$

If the external system impedance is added to the circuit it becomes:



where

external system reactance = 0.06 per unit

external system resistance = 0.03 per unit

from the machine terminals to infinite busbar

$$\text{Total } R = 0.03 + 0.01 + 0.0367 = 0.407$$

$$\text{Total } X = 0.06 + 0.1 + 0.1455 = 0.3055$$

$$\text{Total } Z = 0.407 + j0.3055$$

$$\therefore Z = \sqrt{0.407^2 + 0.3055^2} = 0.508$$

$$I = \frac{1.1}{0.508} = 2.17$$

Considering the original RH part of circuit.

$$Z_{r+m} = \sqrt{0.367^2 + 0.1455^2} = 0.395$$

$$V_{r+m} = I Z_{r+m} = 2.17 \times 0.395 = 0.857$$

$$I_r : \text{Rotor Current} = \frac{V_{r+m}}{Z_r}$$

$$Z_r = \sqrt{0.4^2 + 0.1^2} = 0.412$$

$$\therefore I_r = \frac{0.857}{0.412} = 2.08$$

$$\text{Motor Output Power} = \frac{I_r^2 R_r (1-s)}{s}$$

$$= 2.08^2 \times 0.39$$

$$= 1.69 : \text{Motor output power} \rightarrow$$

$$\text{Motor Output Torque} = \frac{\text{Power}}{\text{Speed}} = \frac{\text{Power}}{1-s}$$

$$= \frac{1.69}{0.975} = 1.73 \rightarrow$$

$$\text{Motor Terminal Voltage} = I Z_d$$

where Z_d is the driving motor equivalent impedance

$$\begin{aligned}\dot{Z}_d &= 0.01 + 0.367 + j 0.1 + j 0.1455 \\ &= 0.377 + j 0.2455\end{aligned}$$

$$\therefore Z_d = \sqrt{0.377^2 + 0.2455^2} = 0.45$$

$$\therefore V_d = 2.17 \times 0.45 = \underline{0.976 \text{ V}} \rightarrow$$

$$\text{Line kVA} = 1.1 \times 2.17 = \underline{2.39} \rightarrow$$

Summarising conditions at 2.5 % slip

Motor Output Power	=	1.69
Motor Output Torque	=	1.73
Motor Terminal Voltage	=	0.976
Line kVA	=	2.39

These values can be calculated for a range of values of slip and a TORQUE ~ SLIP curve plotted. From this curve it can be determined whether the shovel m/g set driving motor will stall at peak power.

Table 3A I shows the results of the calculations for a range of slips for this particular system.

TABLE 3 A I

M - G SET DRIVING MOTOR PARAMETERS

CALCULATED FROM EQUIVALENT CIRCUIT

DATA

Slip s	$1 - s$	$R_2 \frac{(1-s)}{s}$	$\frac{R_2}{s}$	Motor Power	Motor Torque	Motor Terminal Volts	Line kVA
0.005	0.995	1.99	2.0	0.485	0.488	1.062	0.69
0.01	0.99	0.99	1.0	0.940	0.948	1.040	1.17
0.025	0.975	0.39	0.4	1.690	1.730	0.976	2.39
0.03	0.97	0.323	0.33	1.755	1.810	0.955	2.68
0.04	0.96	0.24	0.25	1.800	1.875	0.920	3.11
0.05	0.95	0.19	0.20	1.720	1.810	0.897	3.42
0.075	0.925	0.1235	0.134	1.485	1.605	0.862	3.90
0.1	0.90	0.09	0.1	1.100	1.220	0.843	4.21

Graphically it can be shown (Fig. 3 A 1) that the maximum torque developed by the m/g set driving motor is about 1.875 (per unit value).

If the shovel load cycle imposes a torque in excess of 1.875 on the m-g set driving motor it will stall, i.e. it will "pull down," causing a production delay.

If the maximum torque imposed on the m-g set is 1.9 then with the external electricity supply conditions, the shovel would not operate satisfactorily since

the m-g set driving motor would repeatedly "pull down".

If the external electricity system is ignored an erroneous result can occur. It is common practice to assume that the motor terminal voltage is 1.0 (per unit value). In this case the following maximum torque would have been calculated:

Assuming slip at maximum torque = 0.04
(from Fig. 3A1)

$$Z \text{ for motor} = 0.325$$

$$\therefore I = \frac{1.00}{0.325} = 3.08$$

$$V_r = I Z_{r+m} = 3.08 \times 0.26 = 0.805$$

$$I_r = \frac{V_r}{Z_r} = \frac{0.805}{0.269} = 2.99$$

$$\text{Power} = 2.99^2 \times 0.24 = 2.15$$

$$T_{\text{torque}} = \frac{2.15}{0.96} = 2.24$$

If the peak torque imposed is again 1.9 and a 10% allowance is added to ensure satisfactory operation the required torque is $1.9 \times 1.1 = 2.09$

The above calculation indicates that the m-g

set driving motor would operate satisfactorily but with the electricity supply conditions previously outlined it would repeatedly pull down.

This situation clearly indicates the need for the systems approach to excavator selection.

APPENDIX 3.B

HEWLET-PACKARD 9100B COMPUTER PROGRAMME FOR SOLUTION OF THE EQUIVALENT CIRCUIT OF THE INDUCTION MOTOR WITH SYSTEM LINE IMPEDANCE FED FROM INFINITE BUSBARS.

Rearranging the formulae of Appendix 3A

$$R_T = \frac{R_r}{s} \cdot X_m^2 \quad ; \quad X_T = \frac{X_m \left[\left(\frac{R_r}{s} \right) + X_r (X_m + X_r) \right]}{\left(\frac{R_r}{s} \right)^2 + (X_m + X_r)^2}$$

$$R_i = R_T + R_s \quad ; \quad X_i = X_T + X_s$$

$$R = R_i + R_L \quad ; \quad X = X_i + X_L$$

$$Z_T = \sqrt{R_T^2 + X_T^2} \quad ; \quad Z_i = \sqrt{R_i^2 + X_i^2}$$

$$Z = \sqrt{R^2 + X^2} \quad ; \quad Z_r = \sqrt{\left(\frac{R_T}{s} \right)^2 + X_r^2}$$

$$I_L = \frac{1.1}{Z} \quad ; \quad V_m = I_L \cdot Z_i \quad ; \quad \text{Line kVA} = 1.1 I_L$$

$$T_m = \left(\frac{I_L Z_T}{Z_r} \right)^2 \frac{R_s}{s} \quad ; \quad P_m = (1-s) T_m$$

Procedure

1. Set machine power switch : ON
2. Set printer : ON, x, y.

3. Set PROG/RUN switch to RUN
4. Set decimal to give required form of output (4 dec. places)
5. Press END
6. Insert card in slot, arrow A down:
Press ENTER
7. Press END, CONT. Display should be $\left. \begin{matrix} 0 \\ 0 \\ 0 \end{matrix} \right\}$

8. Key in data as follows :-

$X_L \uparrow R_L \text{ CONT}$
 $X_S \uparrow R_S \text{ CONT}$
 $X_r \uparrow R_r \text{ CONT}$
 $X_m \text{ CONT}$
 $S \text{ CONT}$

Data are automatically printed out followed by print out of:

V_m , Line kVA, T_m , P_m

9. To continue with new value of S (other parameters remain the same) operate the following Keys :-

Go To 2 4 "S" CONT

The author is indebted to Dr. J. Bray for instruction in the use of the Hewlett-Packard computers and assistance with programming.

4. DRAGLINES AND CLAMSHELLS

DRAGLINE OPERATIONS

Draglines can be conveniently divided into:

1. Truck-mounted
2. Crawler-mounted
3. Walking draglines

The truck-mounted machine is generally used in civil construction or as a service tool in open pit operations and not as a primary excavator. The walking dragline is a stripping machine and is dealt with in 5. Large Stripping Machines.

The crawler-mounted dragline (Fig 4.1) finds only limited application in open pit mining because:

- a) it has a less positive digging action, particularly when "chopping down", than the shovel, and the smaller sizes are not suitable for loading dense, badly fragmented rocks,
- b) the cycle time is longer than that of an equivalent shovel, and
- c) it has poor "spotting" ability when loading.

The excellent deep-digging capability of the dragline results in reduced gradients for truck and rail haulage operations, where the bucket excavates below grade. It can operate from any horizon within the excavation and, unlike the shovel can stand on a selected competent bed. Because of its deep-digging capability, the crawler-mounted dragline particularly finds application in wet pit operations, e.g. sand and gravel, chalk, etc., excavating box cuts, recovering remnants from pit floors, etc., and in general work, e.g.

digging sumps, extending inclines, etc.

The dragline is used primarily for digging below the working level of the machine but can excavate above this level ("chopping down"). In this mode of operation, bucket filling is less positive and the swing angle is usually greater than that for below grade digging, resulting in increased cycle time.

The cycle pattern of a dragline is:

- a) dragging and filling the bucket,
- b) hoisting clear and swinging the bucket to the dumping point,
- c) "spotting" over the truck, hopper or other means of transport
- d) dumping
- e) lowering and swinging back the empty bucket
- f) positioning or casting the bucket for the next load.

The major factor in keeping the cycle time to a minimum is the skill of the operator and his ability to correctly position the bucket at the start of the cycle. This is all important since the effective digging power is dependent on that portion of the total bucket mass transmitted to the teeth of the bucket⁴¹. The following factors also affect cycle time:

1. Depth of cut
2. Topography, since the dragline may have to provide its own working surface
3. Material to be excavated
4. Weather conditions
5. Quality of operation, maintenance and supervision

6. Job layout, drainage, etc.

Summarised the advantages of the dragline are:

1. Will dig much deeper than an equivalent loading shovel, bucket wheel excavator, front-end loader or tractor-scraper.
2. Excluding a dredge, the dragline will handle material with higher water content and poorer stability than other excavators.
3. Since draglines usually work on the surface they are less restricted by pit dimensions than other excavators. They are therefore more flexible in operation
4. The operation is usually safer since the dragline is not subject to the hazards of pit flooding or highwall slides
5. The dragline can select any competent horizon as a working bench and is not therefore dependent on the safe bearing load of the pit floor.

The bucket size of a dragline can be expressed as:

$$B_c = \frac{Q}{C \times A \times O \times B_f \times P} \dots\dots\dots 4.1$$

where

- B_c = bucket capacity (volume)
 Q = production required (bank volume/h)
 C = theoretical cycles/h = $\frac{60}{t_c}$
 t_c = theoretical cycle time (min)
 A = mechanical availability during scheduled hours of work

O = job operational factor

B_f = bucket factor

P = propel time factor

C - Theoretical cycles per hour

The cycle time of a loading dragline is extremely variable for several reasons and is particularly dependent on the ability of the operator. Because of this it is inappropriate to take account of all the variables and the following average times have been established from a large number of operations as outlined in Chapter 2.

TABLE 4.1

CRAWLER-MOUNTED DRAGLINES

HOIST TIMES (sec)*

<u>Bucket Size</u>		<u>Lower Limit</u>	<u>Higher Limit</u>
<u>m³</u>	<u>yd³</u>		
1.5 - 2.5	2 - 3	12	24
3 - 4	4 - 5	12	24
4 - 4.5	5 - 6	12	24
5.5 - 6	7 - 8	14	28
8 - 9	10 - 12	16	32
10.5 - 11.5	14 - 15	18	36 x

x inadequate data, use with caution

* The above table includes digging time, hoist time and lost time in spotting the bucket. The higher limits include the extra time required for digging high banks within the limits of the machine and for hard rocks where the hoist speed is generally slow. The lower limit applies to easy - medium digging conditions.

The hoist time selected for a given condition added to the time of swing to the truck and back to the pit again establishes the theoretical cycle time. These times may be taken from manufacturers' literature or from the following approximate figures which have been established from a large number of operations as outlined in Chapter 2.

TABLE 4.II
CRAWLER-MOUNTED DRAGLINES
SWING TIMES (sec)
(one way only)

<u>Bucket Size</u>		<u>Degrees</u>							
m ³	yd ³	60	80	100	120	140	160	180	
1.5 - 2.5	2 - 3	5	6	7	8	9	10	11	
3 - 4	4 - 5	6	7	8	9	10	11	12	
4 - 4.5	5 - 6	7	8	9	10	10.5	11.5	12.5	
5.5 - 6	7 - 8	7	8	9	10	10.5	11.5	12.5	
8 - 9	10 - 12	7	8	9	10	10.5	11.5	12.5	
10.5 - 11.5	14 - 15	7	8	9	10	11	12	13	

A number of authorities ^{42, 43, 44} suggest the use of a correction factor where the dragline is not digging at optimum depth. In view of the variability of the cycle time due to the factors previously described, it is considered that this precision is not justified and is allowed for in Table 4.I.

A - Availability

A is the availability of the dragline for work during the manned hours: it is generally defined as the mechanical

availability during the scheduled hours. This can be determined from plant records by industrial engineering methods, time studies, etc.

O - Job operational factor

A crawler dragline serviced by some form of transport is part of a system and is subject to delays due to management, supervision and labour deficiencies, job conditions, climate, etc. The bucket capacity must be increased to compensate for these losses in production time. The job operational factor can also be determined by industrial engineering methods. Care must be taken to ensure that if the propel time P is included in O, it is eliminated from formula 4.1.

A0

Where no experience is available to enable A and O to be determined their product A0, the operating efficiency may be determined from Table 2.VI.

B_f - Bucket factor

$$B_f = \frac{\text{Fillability}}{\text{Swell factor}}$$

The values of fillability given in Table 2.I are mainly based on shovel dipper results since most observations were made on shovel operations. As a guide however the following approximate figures were determined for loading draglines:

TABLE 4.III

LOADING DRAGLINE FILLABILITY FACTORS

Digging Condition	Fillability
Easy	0.95 - 1.0
Medium	0.8 - 0.9
Medium-Hard	0.65 - 0.75
Hard	0.40 - 0.65

P - Propel time factor

This is the most difficult item to determine due to the wide range of duties performed by the draglines observed. In wet pit sand and gravel operations averaged it 0.90. The approximate bucket size having been fixed, a standard bucket can be chosen and a range of draglines can be selected from manufacturers literature. The actual production capacity of each model must then be calculated, manufacturers' data being used to determine the theoretical cycle times, to ensure that production requirements will be met.

Table 4.IV provides a summary of the outputs of crawler-mounted draglines.

TABLE 4.IV

OUTPUTS OF CRAWLER-MOUNTED DRAGLINES

(Bank volumes)*

Bucket capacity		Boom length		Production, yd ³ /h (bank)		
yd ³	m ³	ft	m	Digging conditions**		
				E	M	M-H
3	2.5	65	20	220	135	70
4	3	70	21	250	160	85
5	4	80	24	315	200	105
6	4.5	100	30	330	215	115
7	5.5	140	43	341	224	125
10	8	160	49	435	290	160

*Based on a job efficiency of 0.8, a fillability of 0.75, a swell factor of 1.35 and a swing angle of 110°. No propel time factor is included

**For explanation see footnote to Table 2.I.

DRAGLINE DRIVES

Because the duties of crawler-mounted draglines are usually varied, or they are employed in smaller scale operations such as sand and gravel extraction where detailed planning of operations is not usual the diesel drive is generally preferred to give maximum flexibility. Conventional brake and clutch operated gear trains are employed. Some of the larger sizes employ electric drives similar to loading shovels.

CRAWLER-MOUNTED DRAGLINE COSTS

Ownership Costs

Draglines can have lives well in excess of 20 years but it is unusual for these figures to be used in assessing the depreciation costs. Individual company depreciation and replacement policies usually determine the depreciation period and taxation may be the deciding factor.

The following format provides an adequate estimation of ownership costs for loading draglines.

(1) FOB machine price, including optional extras, sales taxes, etc.	£.....
(2) Freight and insurance (to site)	£.....
(3) Import duty	£.....
(4) Sub-total	£.....
(5) Ballast, if manufactured locally (allow £40/t)	£.....
(6) Erection costs (See Table 4.V)	£.....
(7) Insurance during erection	£.....
(8) Sub-total	£.....
(9) Interest up to start of production (allow interest on 30% of sub total 8)	£.....
(10) Sub-total	£.....

(11) Dragline write-off period

n = ...years (11a)

.....hours/year (11b)

.....total hours (11c)

(12) Machine depreciation and amortisation

cost/h = $\frac{(\text{Sub-total 10})}{(\text{Item 11c})}$

(13) By use of average investment formula, assuming depreciation charges replace original investment

Average machine investment

= $\frac{(\text{Sub-total 10}) \times (n+1)}{2n}$ = £.....

(14) Interest rate%

(15) Insurance%

(16) Taxes etc (if any)%

(17) Total%

(18) Interest, taxes, insurances, etc.

Costs/h = $\frac{(\text{Total \%}) \times (\text{Item 13})}{(\text{Item 11b})}$ = £.....

(19) Total ownership costs/h

(Item 12)+(Item 18) = £.....

TABLE 4.V

LOADING-DRAGLINE ERECTION COSTS
(1973 projections)

Dipper Capacity		Erection costs
yd ³	m ³	£
5	4	2500
8	6	3800
10	8	5500
15	12	6600

The above figures include the cost of supervision by the manufacturer, i.e. time, subsistence, local transport skilled and unskilled labour, erection equipment, workshop costs, etc., but not the travel expenses of the manufacturer's erectors. The costs should be adjusted in the same way as for loading shovels. (See Chapter 3).

Operating Costs

(20) Maintenance and supply costs/h

$$= \frac{9\% \times (\text{Sub-total 4}) \times (H(\text{Table 4.VI}) \times (M(\text{Table 4.VII})))}{\text{Item 11b}}$$

= £.....

(21) Labour rate/h (to include social benefits, taxes, insurances, etc.

= £.....

(22) Fuel consumption

$$\frac{\text{Engine hp or ps}^*}{40} = \text{fuel consumption (gal/scheduled h)}$$

$$(\text{cost/gal}) \times (\text{gall/h}) = \text{£.....}$$

*most engines have horsepower or ps ratings rather than kW ratings and fuel will be measured in gallons for some time to come.

(23) Lubricants, etc.

Lubricant costs collected vary considerably but the following percentages of fuel costs per scheduled hour have been compiled.

Lubricants (gall/h) - 7%

Special oils, hydraulic, etc (gall/h) - 2%

Greases - 3%

Filter costs (3.0 p/h)

= £.....

(24) Total Operating Costs

$$= \text{Items (20 + 21 + 22 + 23)} = \text{£.....}$$

Total Ownership and Operating Costs

(25) Total ownership and operating Costs

= Item (19) + Item (24) = £.....

Cost/tonne

(26) $\frac{\text{Item 25}}{\text{t/h}}$ = £.....

CLAMSHELL OPERATIONS

The Clamshell (Fig 4.2) is essentially an adaptation of the power crane. Its unique feature is the clamshell bucket, which makes it especially applicable to vertical excavations of limited area. It is more suitable for loose materials since the only force that can be applied for digging is the clamshell weight and has limited application as a primary excavator except for wet pit operations in sand and gravel, etc., for stockpiling and other materials handling operations.

The clamshell is opened and closed by a separate line (see Fig 4.2), a "tag line" is also provided to prevent rotation of the clamshell bucket.

There are three types of clamshell buckets:

- 1 - light duty
- 2 - general purpose
- 3 - heavy duty

All are usually provided with bottom, corner and side digging teeth for mining duties. The operating cycle is

- a) lowering the opened clamshell on to the digging area,
- b) closing the clamshell onto the load, c) raising the loaded clamshell, d) simulataneously swinging the boom to the dump point as the clamshell clears the excavation,
- e) positioning the clamshell over the dump point,
- f) opening the clamshell to unload, g) swinging the boom

back to the excavation as the clamshell is being lowered.

Because of its sensitivity to the type of material being excavated and the angle of swing the following output formula must be used with some judgement

$$C_c = \frac{Q_L}{C \times A_0 \times \text{Fillability} \times S} \dots\dots\dots 4.2$$

where

C_c = clamshell capacity (volume)

Q_L = production required (loose volume/h)

A_0 = job operational efficiency
(see Table 2.VI)

S = swing factor (see Table 4.VI)

C = theoretical cycles /h = $\frac{60}{t_c}$

t_c = theoretical cycle time

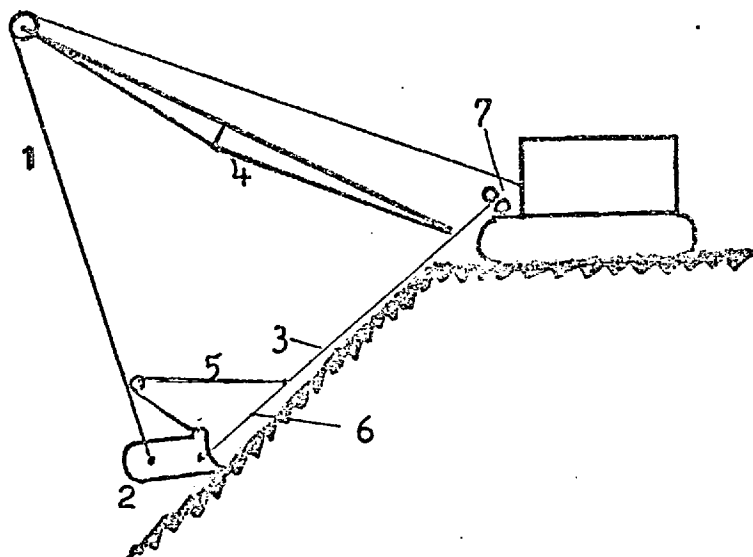
S - Swing Factor

TABLE 4.VI

SWING FACTOR FOR CLAMSHELL OPERATIONS

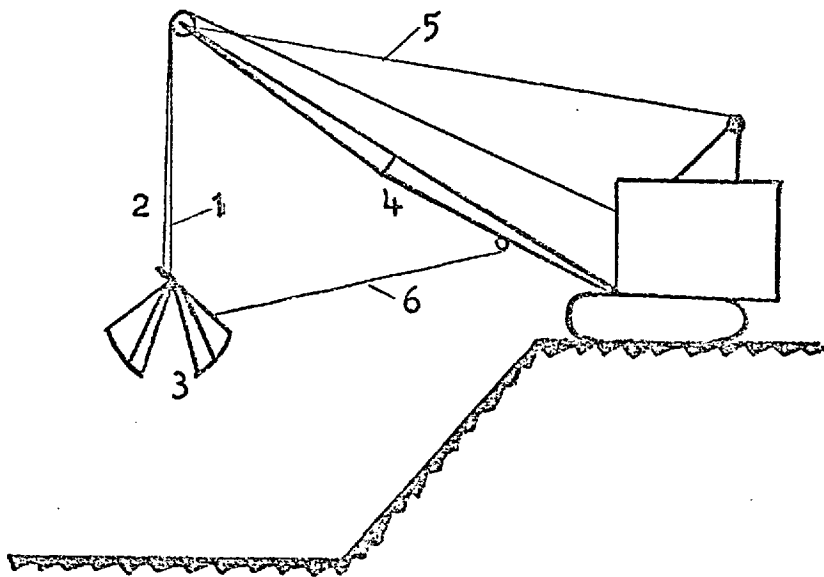
<u>Angle of Swing-degrees</u>	<u>Factor</u>
30 -----	0.75
45 -----	0.85
60 -----	0.90
75 -----	0.95
90 -----	1.00
120 -----	1.10
150 -----	1.20
180 -----	1.30

In general the Ownership and Operating Cost Format used for crawler-mounted draglines can be used for clamshells. Rope maintenance costs for draglines are higher than for clamshells but bucket maintenance for draglines costs less than clamshell maintenance. The limited use of the clamshell does not warrant detailed investigation here.



- 1 Hoist Rope
- 2 Bucket
- 3 Drag Rope
- 4 Boom
- 5 Dump Rope
- 6 Dump Chain
- 7 Fairlead

FIG 4.1 CRAWLER-MOUNTED DRAGLINE TERMINOLOGY



- 1 Digging Line
- 2 Hoist Line
- 3 Clamshell Bucket
- 4 Boom
- 5 Boom Line
- 6 Tag Line

FIG 4.2 CLAMSHELL TERMINOLOGY

REFERENCES

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mines. Vol.3, Pergamon Press, New York, 1966.

5. LARGE STRIPPING MACHINES

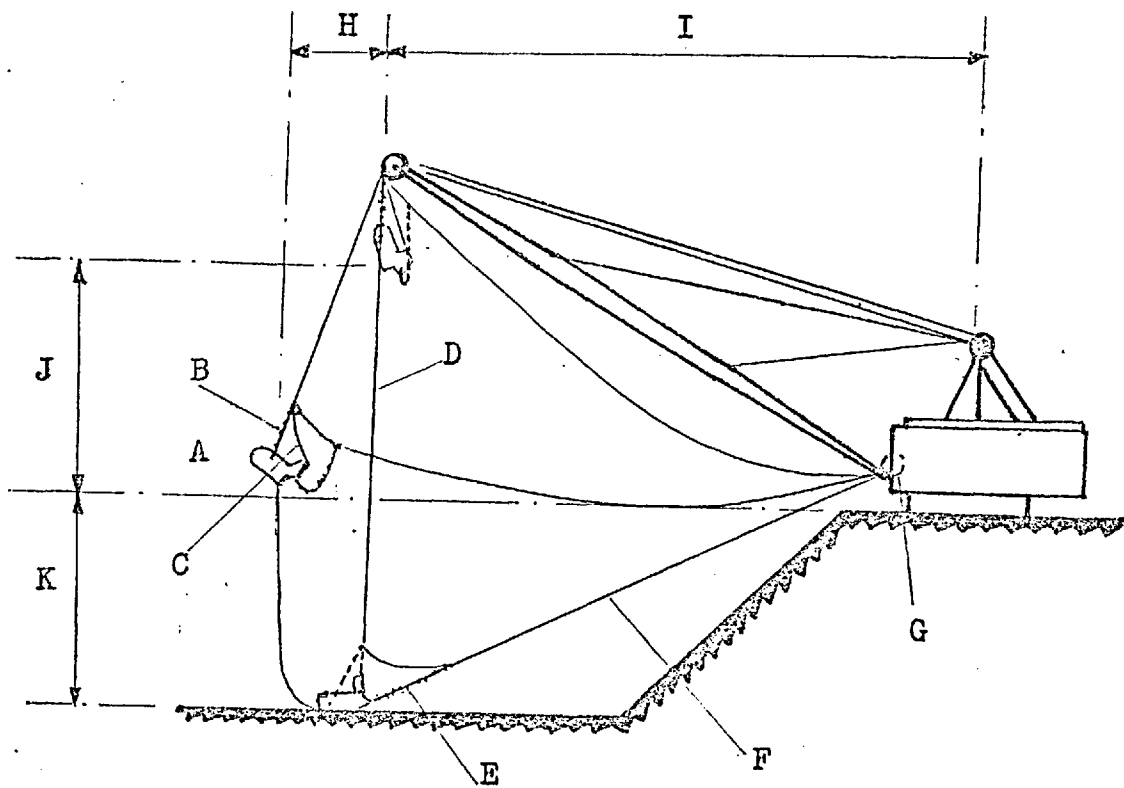
Large single bucket stripping machines were mainly developed in the U.S.A. to expose relatively near surface coal deposits, whereas the major developments in multi-bucket machines took place in German brown coal mines. The application of these machines has now extended into all strip mining operations. The most important machines in this group are walking draglines, stripping shovels and bucket wheel excavators (Fig 5.1, 5.2 and 5.3).

Because of the huge capital investments involved, and because the investment decisions are usually irrevocable, the selection procedure for large stripping machines must be sufficiently comprehensive to cover all eventualities throughout the stripping of a deposit.

The first stage in the selection procedure is to fix the production requirements of the mineral loading machines: from this the output of the stripping machine is determined. This is not usually a single calculation since some optimisation procedure within the market constraints is often essential.

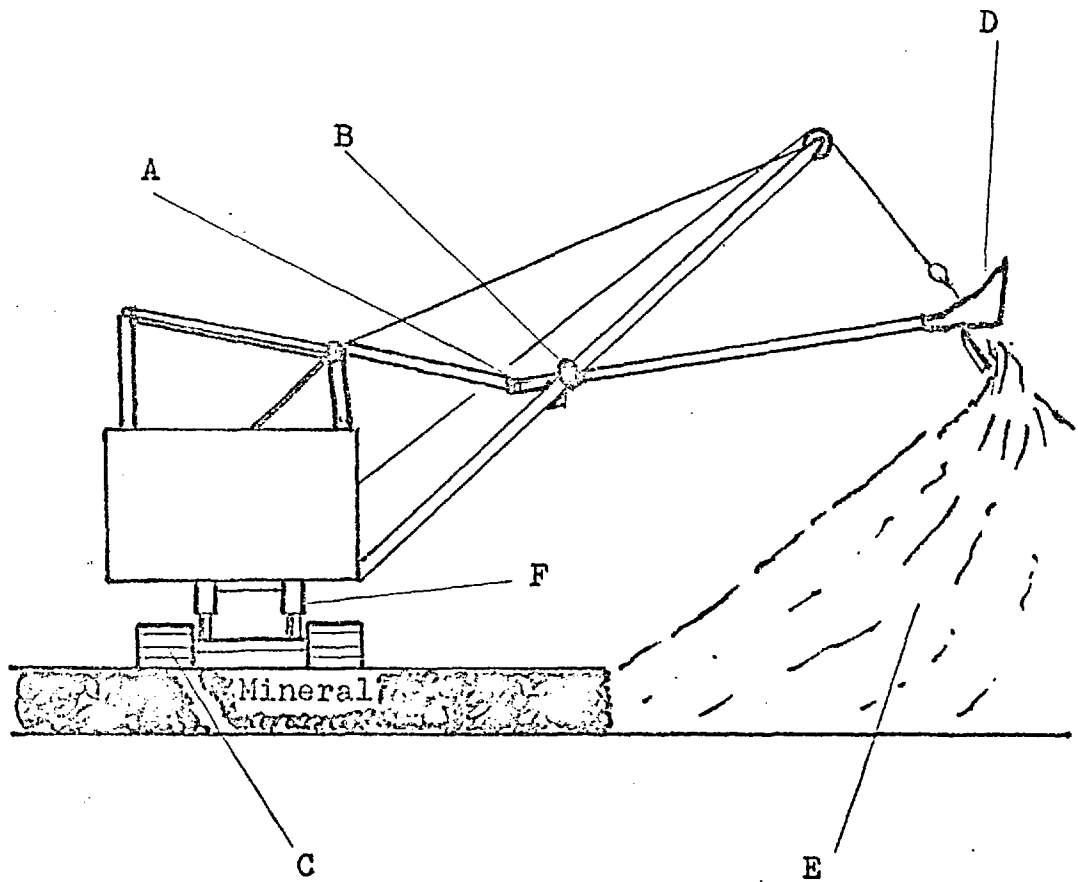
The maximum volume to be stripped is the main factor in the determination of the output of the stripping machine, and the maximum depth decides the reach.

The geometry of most strip mining operations allows fairly limited reserves only to be exposed: stripping and loading facilities must therefore be correctly matched and machine availabilities must be accurately forecast.



- A - Bucket
- B - Hoist Chain
- C - Dump Rope
- D - Hoist Rope
- E - Drag Chain
- F - Drag Rope
- G - Fair Lead
- H - Bucket Throw
- I - Dumping Radius
- J - Dumping Height
- K - Depth

FIG 5.1 WALKING DRAGLINE NOMENCLATURE



- A - Knee Action Crowd Arrangement
- B - Saddle Block
- C - Self-cleaning Crawlers
- D - Dipper
- E - Spoilbank
- F - Hydraulic Levelling Jacks

FIG 5.2 STRIPPING SHOVEL NOMENCLATURE

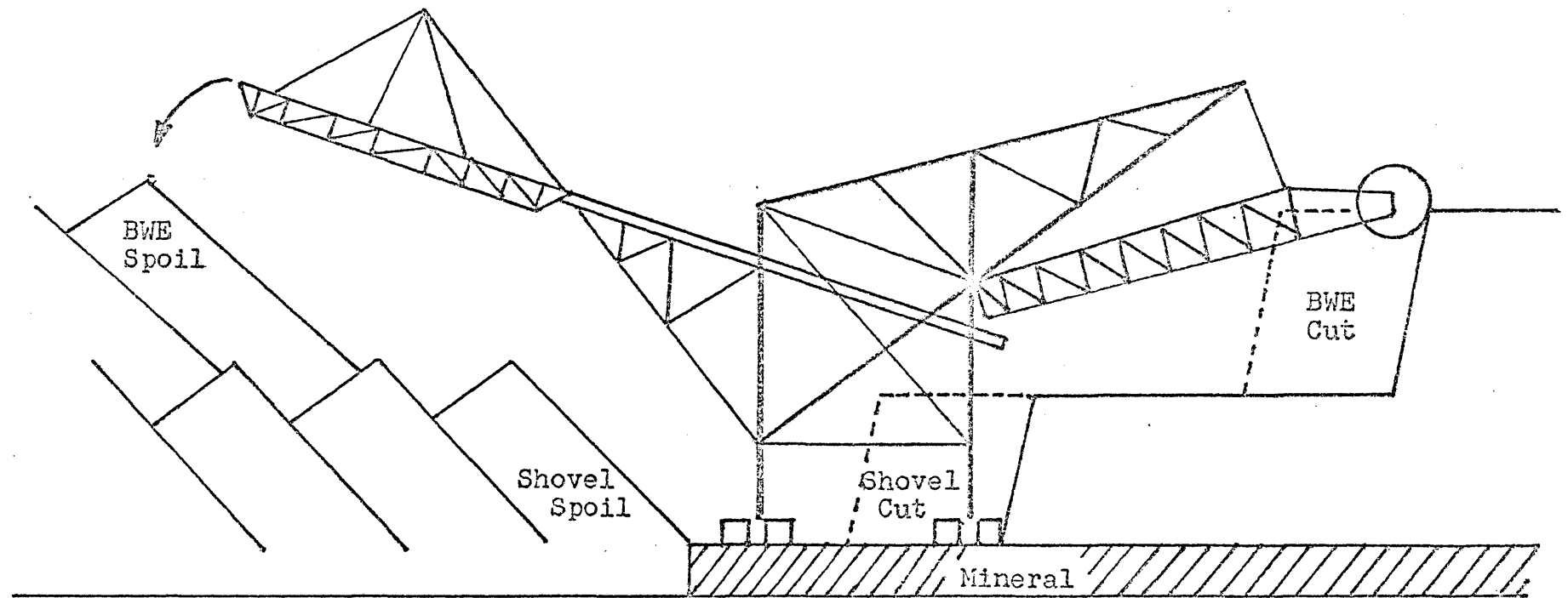


FIG 5.3 BUCKET WHEEL EXCAVATOR DIRECT CASTING IN TANDEM WITH A STRIPPING SHOVEL

- (8) The shovel has more wear points, e.g. hydraulic levelling jacks, crawlers and their drives, etc. and incurs greater maintenance costs than draglines especially in hot, dusty conditions.
- (9) Where the overburden is strong and dense, the shovel is preferred.
- (10) If bad fragmentation occurs due to hard beds, fissures, open joints and other discontinuities, a shovel is more suitable.
- (11) With ample reserves, with overburden up to medium thickness, and where high production is required, a shovel is generally more economic.
- (12) The crawlers of large shovels may sufficiently fragment brittle mineral beds, e.g. coal, gypsum, etc. to eliminate the need to blast the mineral.

The choice should be made on the basis of economic factors but some considerations are difficult to quantify and prejudice may also be involved. The advent of inexpensive explosives, such as AN-FO and metallized slurries has resulted in considerable reductions in blasting costs, and this has tended to favour the dragline, which requires better fragmentation of the overburden. Large modern shovels have relatively light dippers and also need well fragmented overburden: hence some of their advantages are reduced. In general, overall costs of any excavating or loading machine

Where the demand for a product is seasonal e.g. building materials, fuel minerals, agricultural fertilisers, etc., and stockpiling is either not possible or uneconomic, the viability of the operation must be based on the use of machines that can meet the maximum production requirements. If significant variations in grade occur, and again if stockpiling is impracticable, a number of smaller stripping operations may be needed to exploit a deposit. This entails the use of a greater number of smaller-capacity machines, but since the reach of the stripping machine is determined by the depth of overburden, the reach will remain substantially the same, i.e. as for a larger capacity single machine. This invariably results in increased overall capital cost.

SINGLE BUCKET MACHINES

Because they are used for similar duties, it is convenient to consider walking draglines and stripping shovels together. Multi-bucket machines are considered separately in Chapter 6.

Walking Draglines

Walking draglines are extensively used for direct casting operations since they have a better (capacity x reach) to service weight ratio than any other single bucket machine. They can be designed to create relatively low bearing pressures by the use of a large tub but this of course results in the centre of rotation being farther back from the edge of the cut, thus reducing the reach of the machine. Cam operated walking mechanisms were universally adopted, but because of weight distribution problems, hydraulic systems have been adopted for the Bucyrus Erie 4280W machine ("Big Muskie") and for some large machines built in

the U.S.S.R. The American machine experienced considerable troubles during its early operation due to metallurgical problems with the tracks of the hydraulic "feet". The outcome of this has yet to be published but the author understands that greater than 80% availability is now being obtained. Little has been made known of the U.S.S.R. machines. Walking draglines have been extensively developed over the past twenty years (Fig 5.4) and Big Muskie has a 170m^3 (220yd^3) bucket, a 95m (310ft) boom, a machine mass (service weight) of 12700t (14000 short tons) and a connected ac load of 36000 kW (48500 hp).

Although small machines work more efficiently in less consolidated, well fragmented rocks, large draglines are extensively used in hard but well blasted ground. To prepare a highwall for dragline working blasting must result in sufficient fragmentation to ensure efficient bucket loading, while the blasted burden must remain sufficiently stable to allow the dragline to dig and travel without danger of collapse. This may entail buffer blasting and decking of charges where some strata are likely to yield large lumps.

The bottom plates of the dragline tub can become punctured by point loading, and it is usual for a track to be prepared on which the machine will "walk". The costing of the operation must take into account the bulldozer or other units used for this purpose.

Stripping Shovels

The stripping shovel has found application because it is usually more productive than the walking dragline by virtue of its positive dipper loading action, shorter swing time and its ability to handle strong, dense rocks. The largest machine reported in service to date has a 138m^3

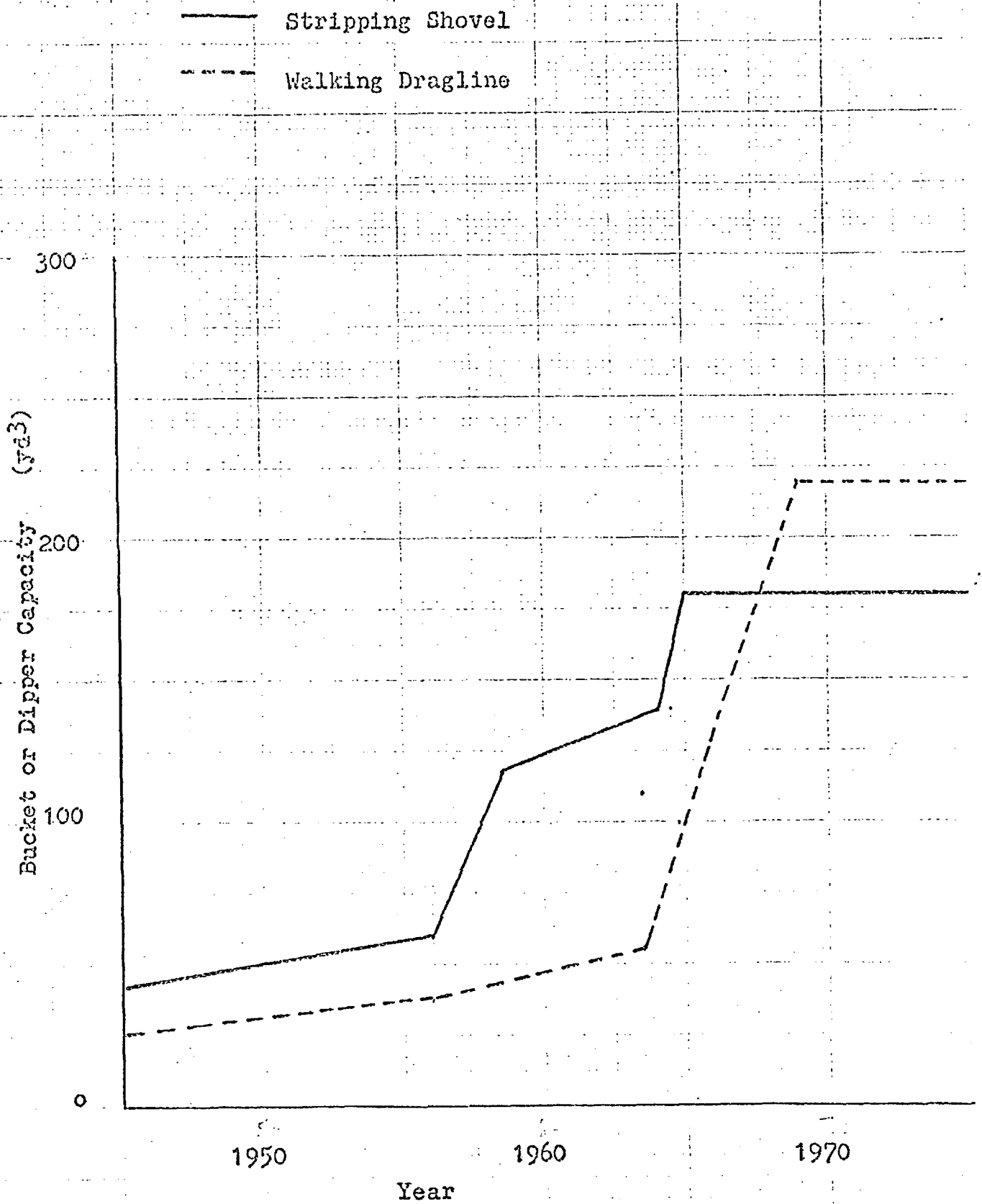


FIG 5,4 SINGLE BUCKET STRIPPING MACHINE GROWTH
1945 - PRESENT

(180 yd³) dipper on a 65m (215ft) boom with an ac installed load of 22500 kW (30000hp). Self cleaning crawlers and hydraulic levelling jacks are universal.

A competent floor is essential and for large machines some preparation of the floor may be necessary to reduce excessive local bearing pressures and to obtain satisfactory crawler life. The cost of this operation must be accounted for.

Shovel versus Dragline

The fields of application of shovels and draglines greatly overlap and the following points must be considered:

- (1) The shovel is more productive as it casts approximately an average of 42m³ (bank) per cubic metre of dipper capacity per hour, whereas a dragline casts approximately 35m³ (bank) per cubic metre per hour.

On the other hand for the same pit width and bucket or dipper capacity, the dragline usually has the lower mass (service weight) and involves a smaller capital investment. The inherently greater reach of the dragline is a distinct advantage in thick overburden.

- (2) Shovels must stand on top of the mineral bed and subject it to average bearing pressures of 350-420 kN/m² (50-60 lbf/in²). If the mineral cannot withstand this load a dragline must be used. A competent working horizon can be selected for a dragline anywhere within the overburden.

- (3) If the mineral bed surface is irregular, subject to faulting, etc., and the surface reasonably regular (or can be "chopped down" to form a level bench) a dragline is more suitable.
- (4) If the mineral lies in lenses, or if substantial lateral variations in grade occur, the economically recoverable mineral lying in patches, the stripping machine must "deadhead" over the surface relatively often. These conditions favour the dragline: because
 - (a) the tub has a low bearing pressure: more easily prepared routes are available, and,
 - (b) a shovel must either dig itself out of the pit, or have a suitable ramp prepared by auxiliary equipment.
- (5) Draglines are more suited to the excavation of the initial box cut.
- (6) Where the spoil has a flat angle of repose, a dragline is more suited to provide the necessary reach.
- (7) Large stripping machines represent substantial capital investments, they are relatively immobile and in wet conditions or where there are flooding hazards the dragline has the advantage that it does not stand on the pit floor.

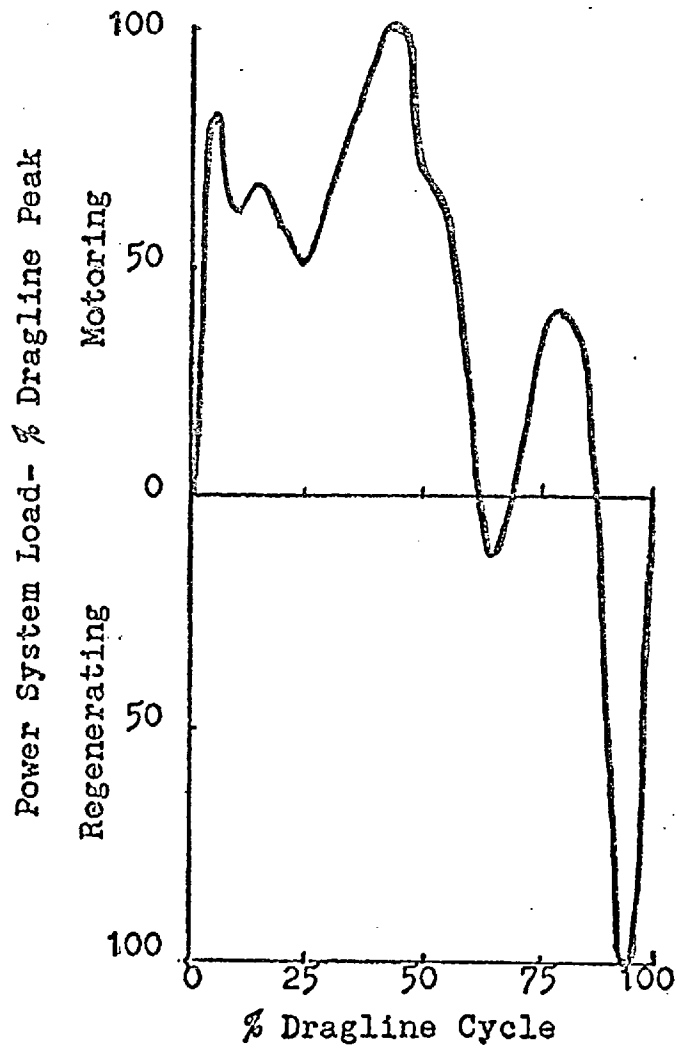
are reduced by good fragmentation because of increased availability, reduced maintenance costs, reduced bucket or dipper filling time and improved fillability.

SINGLE BUCKET STRIPPING MACHINE DRIVES

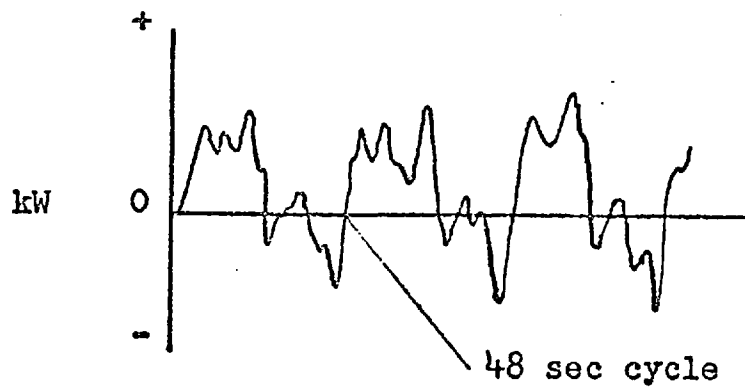
Fig 5.5a shows Power~Time for a single cycle of a dragline excavator. The cycle is representative of a large number taken from a series of charts recorded by the author on three draglines using a Record portable recording kW meter. Fig 5.5b shows a recording wattmeter trace of a large stripping shovel recorded by the same Record kW meter. It will be noted that the cyclic nature of the operation results in large load peaks and periods of regeneration when load is fed back into the system.

The characteristics of the Ward-Leonard system (previously described in Chapter 3) have led to one of the forms of this drive being almost exclusively adopted for single bucket stripping machines. The prime mover is almost invariably a synchronous machine as control of line VAR's is usually essential for power system stability and such control cannot be achieved simply with an induction machine subjected to a single bucket stripping machine cycle. Thyristor controlled drives, previously described in Chapter 3 are not yet adequately developed for this duty and the problem of power system stability would be worsened by the high reactive current drawn when low-speed/high-torque loads are demanded.

The major part of the operating cycle of draglines (and shovels to a lesser extent) is taken up in swing time. To obtain the fastest safe swing time and optimal performance, the whole of this operation must consist of acceleration and deceleration. Every one second saved in 20 - 60 second cycle



(a) TYPICAL DRAGLINE LOAD~TIME CYCLE (kW~sec).

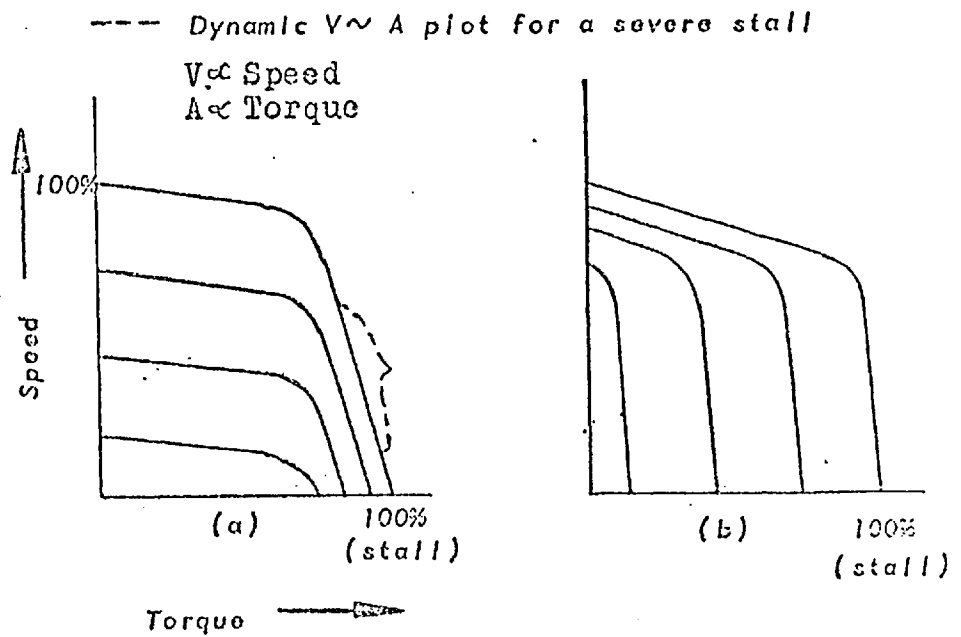


(b) LOAD~TIME CYCLE - STRIPPING SHOVEL

FIG 5.5 STRIPPING MACHINE LOAD CYCLES

times can produce significant production improvements. It is well known that precise controls can be achieved with solid state control systems but speed of response is of equal importance. Precise controls are also highly desirable to allow the operator to reduce speed quickly when the dipper or bucket enters the bank, to reduce mechanical and electrical shock loadings and consequently reduce maintenance down-time and costs. A further problem occurs where a dipper or a bucket enters badly fragmented ground, causing severe mechanical and electrical shock loading and high motor stall currents. To avoid this a high dynamic response is essential, both to reduce the stalled current and to improve the motor speed-torque characteristic. Fig 5.6 shows the motor characteristics obtainable and the stalling current effect of striking hard ground with solid state controls⁵¹, recorded by the author on a large dragline. These characteristics allow the selected motor speed to be maintained up to the maximum safe torque after which any increase in imposed torque reduces the speed automatically without excessive current overshoot.

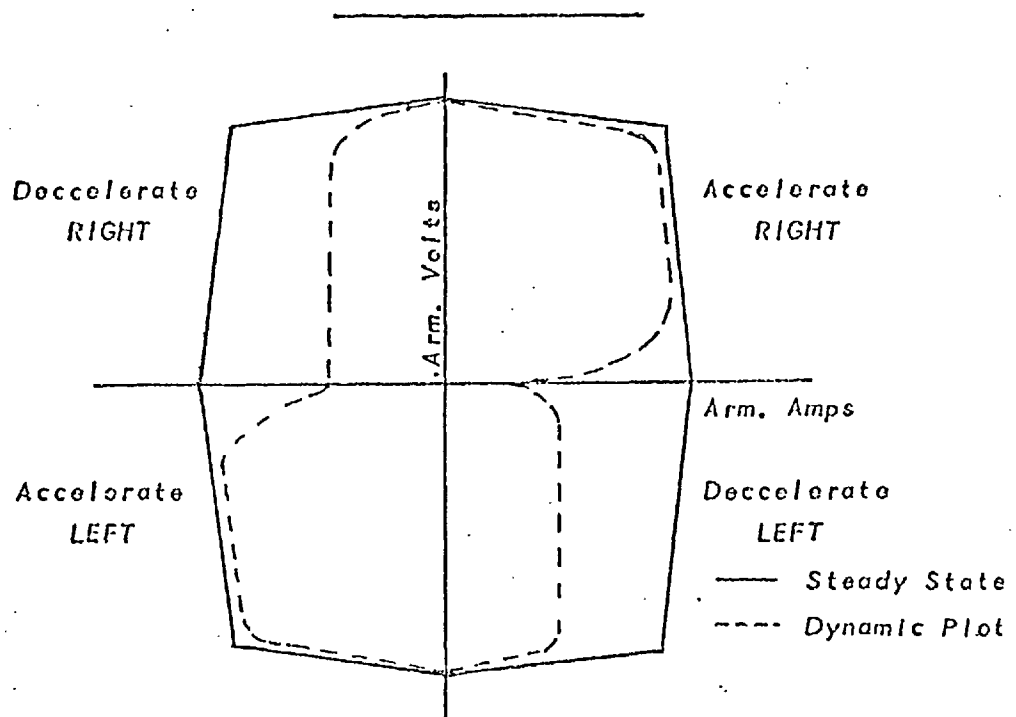
To obtain these advantages the dynamic performance of a single bucket excavator as a complete system must be considered. The generators, motors, gear boxes, rope systems, excavating and swing motions, as well as control systems response, must be considered as a whole to obtain optimum performance⁵¹. For example, to obtain maximum safe swing acceleration and deceleration, the mechanical transmission losses which oppose acceleration but assist deceleration must be accounted for. It is also essential to ensure that



Typical Steady State Speed $\sim$ Torque Characteristics
with Static/Solid State Controls

- (a) Holst, Crowd or Drag Motions
(b) Swing Motion (Dragline or Shovel)

Each unbroken line represents the steady state characteristic for a position of the control lever.



Hypothetical Dynamic $V \sim A$ Characteristic
for dragline swing motion

FIG 5.6 SINGLE BUCKET STRIPPING MACHINE - DESIRABLE DRIVE CHARACTERISTICS

clearances, wear, flexibility, etc., in the swing motion be taken up at low speeds to reduce shock loading (Fig 5.6). Fig 5.7 illustrates the walking cycle of a walking dragline excavator with a good dynamic characteristic. These features can be provided in a solid state control system with its inherently high speed of response due to the absence of mechanical movements and magnetic components, to ensure the necessary dynamic performances ⁵².

The first objective in seeking good dynamic performance is reduced cycle time and consequently higher production, but other and perhaps equally important benefits are to be gained:

- a) Because of reduced mechanical and electrical shock loading, failure of components and maintenance costs are reduced.
- b) The reduction of excessive over-currents due to stalling avoids electrical overloading and reduces commutation difficulties, again reducing maintenance costs and downtime.
- c) The faster responses and finer controls associated with improved dynamic performance systems greatly assists the operator to synchronise the motions of a large excavator, resulting in reduced operator fatigue and increased production particularly during the latter half of the shift.

Single Bucket Stripping Machines on Small Power Systems

Up to a few years ago stripping machines with buckets or dippers larger than 40-m^3 (50-yd^3) had not been used outside the U.S.A. with only a few greater than 20-m^3 (25-yd^3) in service. When large electric drive single bucket machines

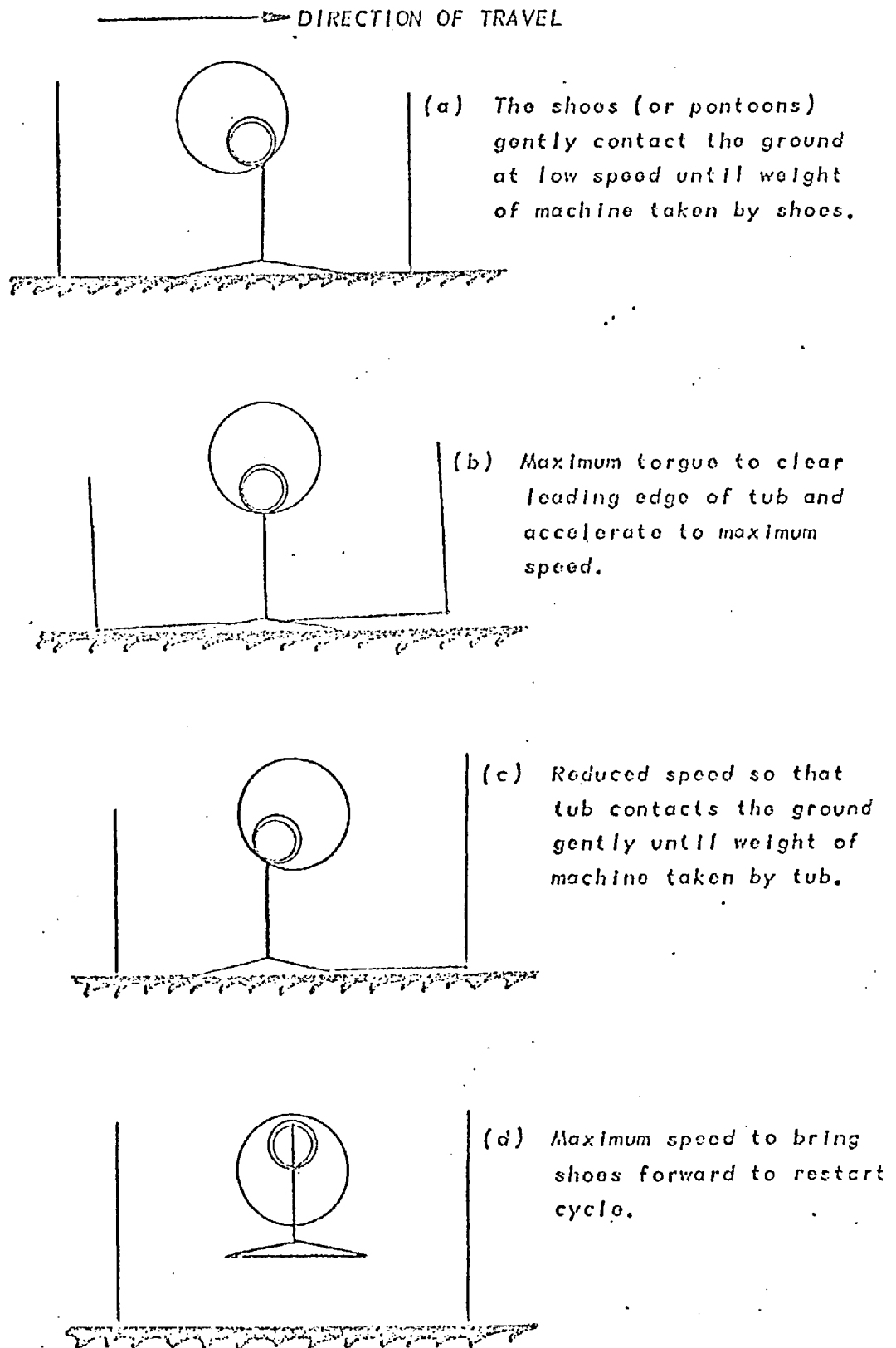


FIG 5.7 WALKING DRAGLINE -DESIRABLE CHARACTERISTICS FOR PROPEL MECHANISM

are supplied by small or relatively "weak" power systems they can cause troublesome voltage fluctuations and frequency disturbances because of the large and sometimes almost periodic load swings which Fig 5.5 shows to be inherent in the operation of single bucket excavators. These problems were not readily apparent in the U.S.A. with relatively "stiff" power systems, nor have they been a major argument against the use of the machines in the U.K.: but they are of real concern on the small power systems of developing nations and in remote locations with relatively weak power systems, where they may inhibit the selection and economic use of large single bucket stripping excavators. Although the power system may be considered by mining engineers to be outside the scope of open pit excavator and loading equipment selection, it has a powerful influence on equipment selection when considering stripping shovels and draglines and must be taken into account when adopting a systems approach to selection. This subject is fully discussed in Appendix 5.A.

SELECTION PROCEDURE

The main steps in selecting a stripping machine are:

1. Determination of the approximate dipper or bucket capacity.
2. Determination of the machine geometry.
3. Selection within a standard range, where applicable, and re-assessment of the model selected.

The approximate dipper capacity may be obtained from:

$$B_c = \frac{Q}{C \times S \times A_0 \times B_f \times P} \dots\dots\dots (5.1)$$

Q - Production or stripping rate

The stripping rate is determined from the mineral production requirements and the overburden ratios. Because of the huge capital investments involved, stripping machines are usually scheduled to work 22.5h per day up to 350 days per year. Detailed investigation and planning are essential to ensure that adequate exposed reserves are available during periods allocated for dead-heading, overhaul, etc.

C - Theoretical cycles per hour

C may be obtained from time studies or from Table 5.1.

TABLE 5.1
APPROXIMATE CYCLES PER HOUR FOR
STRIPPING SHOVELS AND DRAGLINES*

Bucket or dipper size		Dragline	Shovel
yd ³	m ³		
8-35	6-27	58	69
36-59	28-45	56	68
60-200	46-150	53	64

*These figures are based on a 90° swing for a shovel and a 120° swing for a dragline, which approximates most field conditions, and have been assembled from time studies and manufacturers literature.

S - Swing factor

No correction for swing factor need be made in the preliminary calculations unless, in the case of a dragline, a large proportion of chopping down is proposed. That portion of the operation where chopping down is used should have the number of cycles per hour reduced by 20 per cent.

Example A dragline performs 60 theoretical cycles per hour for a 120° swing; 30 per cent of the overburden is cast by chopping down.

The cycles per hour when chopping down are

$$60 \times 0.8 = 48$$

The total approximate theoretical cycles per hour are

$$0.3 \times 48 = 14.4$$

$$0.7 \times 60 = 42.0$$
$$\underline{\underline{56.4}}$$

OA - Operating efficiency

Where no experience is available to determine O and A, their product, OA, may be determined from Table 2.VI.

B_f - Dipper or bucket factor

If no data are available, the fillability and swell factor may be obtained from Table 5.II.

TABLE 5.II

STRIPPING MACHINES SWELL FACTOR AND FILLABILITY*

Overburden conditions	Swell Factor	Shovel dipper fillability	Dragline bucket fillability
Light blasting	1.23	0.90-0.95	0.85-0.90
Medium blasting	1.33	0.85-0.95	0.80-0.90
Heavy blasting	1.40	0.80-0.90	0.75-0.85
Bad fragmentation	1.45	0.75-0.85	0.70-0.75

*For fillability the lower figures refer to small machines and the higher figures to large machines. This table is based on the author's studies and limited publications.

P - Propel time factor

The author was unable to make adequate time studies of propel time but in the absence of time studies^{53, 54} the following propel time factors can be used .

Shovels 0.96

Draglines 0.94

These figures are based on normal operations and include the deadheading associated with such operations. Should the mineral deposit be such that greater than normal deadheading is necessary, allowance must be made for this. Time studies often include the propel time in the operating efficiency, and care must be taken to avoid over-compensation.

The approximate dipper size having been determined the next step is to determine the machine geometry.

MACHINE GEOMETRY

The two main dimensions to be determined are the dumping radius and the dumping height. Fig 5.8 illustrates an idealized dragline stripping operation. Usually, to avoid an excessively long boom, no berm is used in dragline operations, the cut width being equal to the pit width and all exposed mineral being loaded out. If the dragline is located on the surface, as in Fig 5.8, the minimum width of cut can be determined by the mineral loading and transport equipment requirements (see Figs 3.1, 2 and 3).

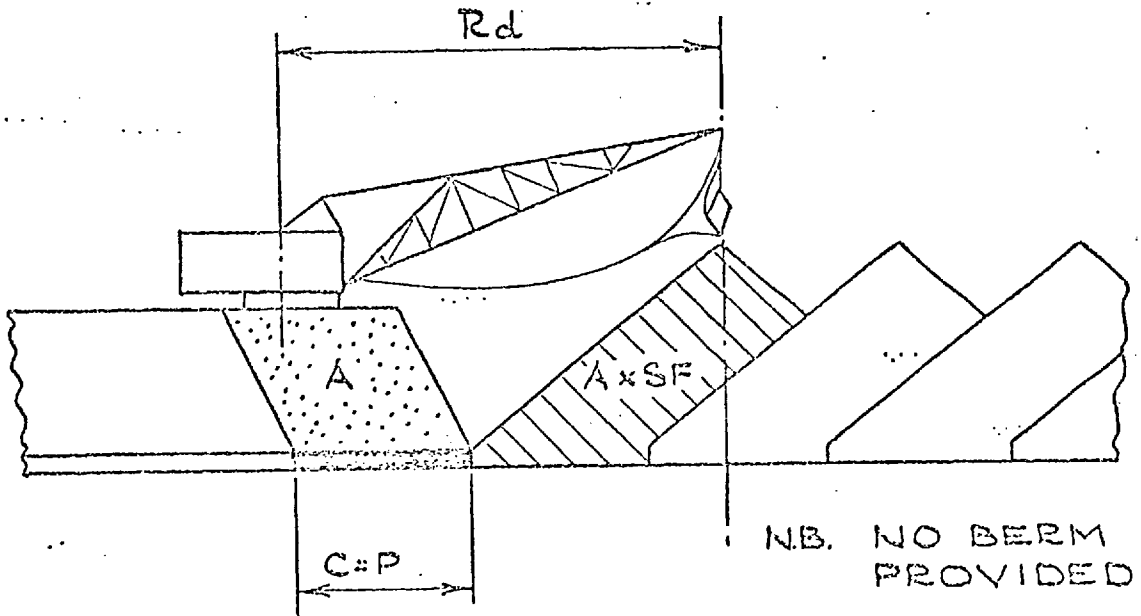


FIG 5.8 IDEALISED DRAGLINE OPERATION

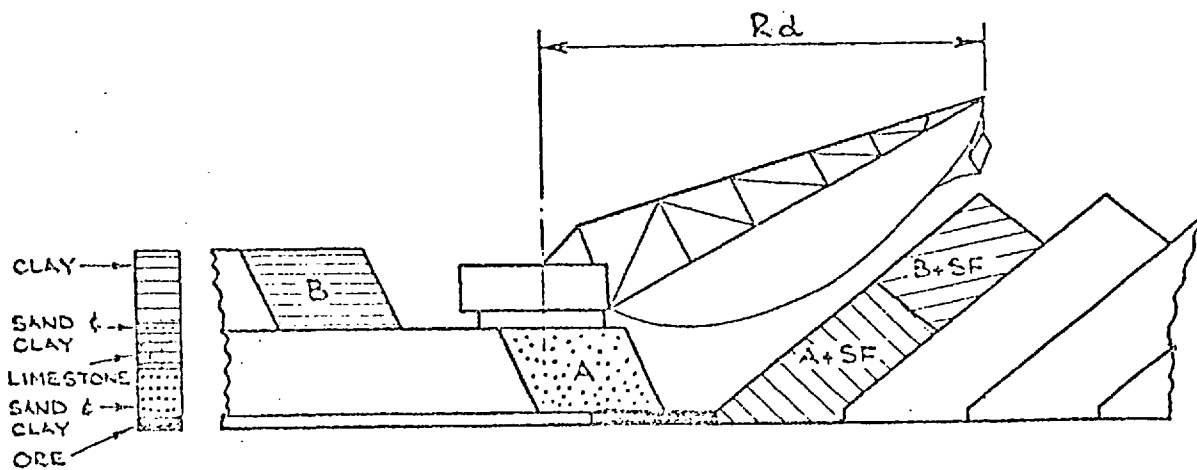


FIG 5.9 DRAGLINE OPERATION FROM COMPETENT INTERMEDIATE BENCH

For small loading shovels pits can be as narrow as 15-18 m (50-60 ft); for large shovels - up to 11.5 m³ (15 yd³) - pit widths of 24-30 m (80-100 ft) are required. Apart from a reduction of the boom length required, and hence of the capital cost, a narrow cut (and in this case pit) width provides more efficient use of the spoil space since it reduces the valleys between the spoil peaks. Short boom lengths reduce the dragline cycle time, some increase in stripping capacity being achieved. Other factors affect pit width, e.g. an increase may be desirable to allow greater flexibility of the loading operation, or safety of personnel and equipment may require a wider pit than the minimum. Fig 5.9 illustrates the practice adopted where the ground bearing strength of the surface is too low to support the dragline. A competent bed is selected within the overburden as the horizon of the working bench. In this case the minimum pit width must be sufficient to allow the dragline boom to swing through 90° when in the operating position nearest the edge of the highwall.

Fig 5.10 illustrates an idealized shovel stripping operation. Obviously, the pit must be wide enough to allow the shovel to swing through 90° when it is in the operating position nearest the spoil bank. The stripping cut width may be less than the pit width and a berm of mineral remains. The stripping cut width is dependent on the width required for the operation of the mineral loading and transport equipment. It is therefore generally equal to the mineral cut width or a multiple of it.

The overburden thickness may vary considerably due to working a hillside outcrop, to relatively steeply inclined

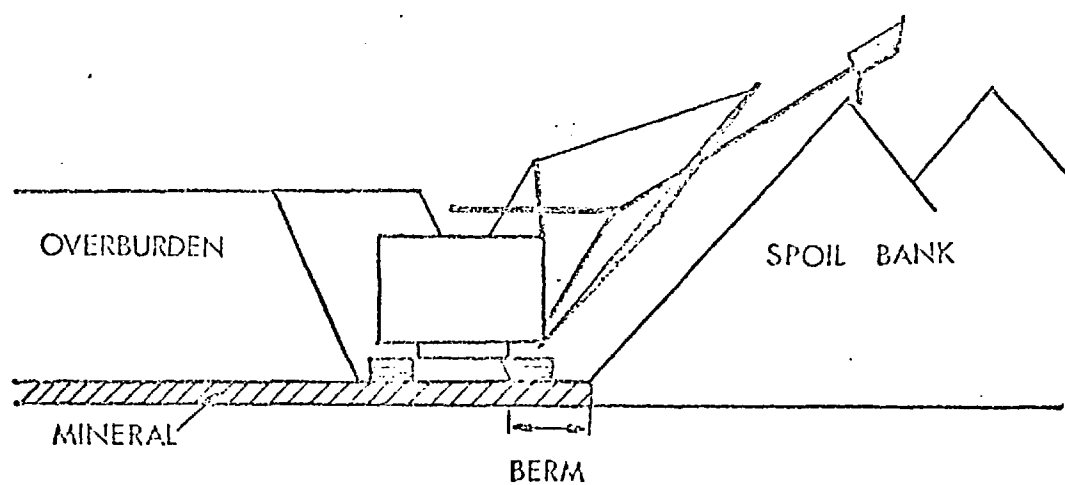


FIG 5.10 IDEALISED STRIPPING SHOVEL OPERATION

mineral beds or to areas of undulating surface topography. If in these circumstances the reach of the dragline is determined by the average overburden thickness it will be inadequate for the thicker overburden. To avoid over-investment where occasional areas of thick overburden occur the following short term measures can be adopted:

1. Spoil the overburden on an outside curve to create additional spoil space: in contour mining operations with short irregular strips this can often be achieved.
2. Temporarily reduce the pit width to the minimum possible.
3. Make a forecut, outside contractors possibly being employed.
4. In the case of a shovel where the spoil is broken rock, operate with the spoil bank side crawler off the mineral bed.

If the thick overburden is extensive it may be necessary to fix the reach of the stripping machine to cater for the maximum overburden thickness or some form of compromise may be possible (see later). Alternatively, rehandle the spoil by one of the following methods:

Haymaking (See Fig 5.11). The overburden is "pulled back" by an auxiliary dragline to create additional spoil space in the vicinity of the main stripping machine. Generally DCF investigations indicate that the thinner overburden should be excavated first, the stripping unit advancing into thicker overburden until it reaches the limit of its reach. It is also usually economic to fix the

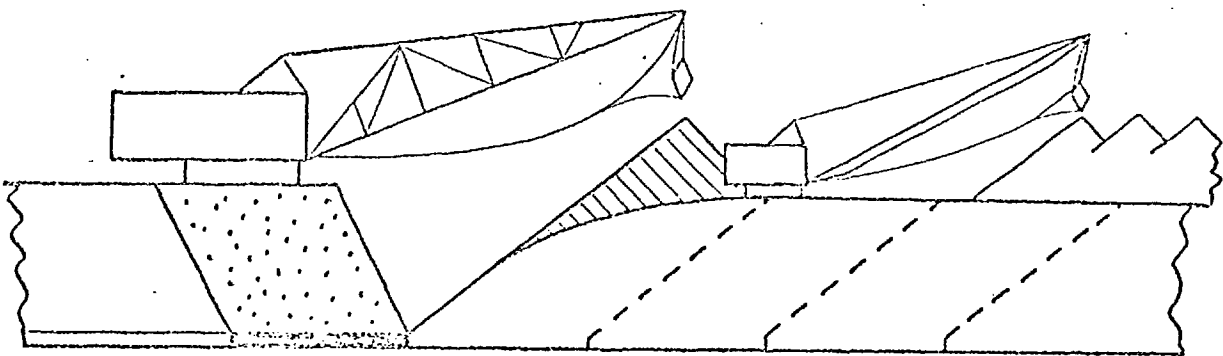


FIG 5.11 "HAYMAKING OPERATION" USING A SECOND
SMALLER DRAGLINE ON THE SPOILBANK

productive capacity of the stripping unit at this limit so that it has no additional capacity for rehandling spoil. At this stage, providing the discharged spoil is sufficiently stable, an auxiliary dragline is located on the spoil to "up-drag" a portion of the spoil "pulling it back" to form a higher spoil bank behind itself, thereby creating additional spoil space for the main stripping unit. This provides the most flexible rehandling method. It has the operational disadvantage of requiring trailing cables on the spoil heap. This method has the advantage that capital investment can be delayed by later purchase of the "pull-back" dragline when advancing into deeper overburden, but, both machines must work in synchronism, since they must work reasonably close to each other.

Extended Bench (See Fig 5.12). Where the distribution of overburden thickness is irregular it may be necessary to provide sufficient productive capacity and dumping height in a dragline so that it can rehandle the spoil. A suitable horizon is selected in the overburden for the dragline(s) and a spoil bridge is formed by dumping a portion of the spoil in the Vee formed by the highwall and the spoilbank, establishing a spoil "bridge" on which the machine can subsequently stand. From this "bridge" or extended bench the dragline not only handles the remaining in-bank overburden but also rehandles the bridge material on which the dragline previously stood. The level for the machine's final location must be such that it permits efficient

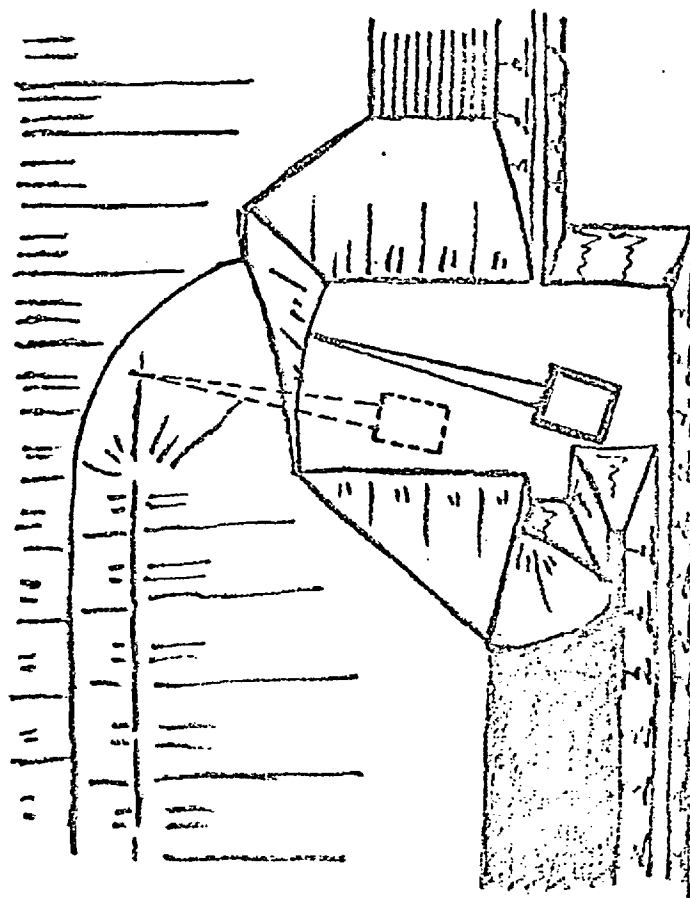
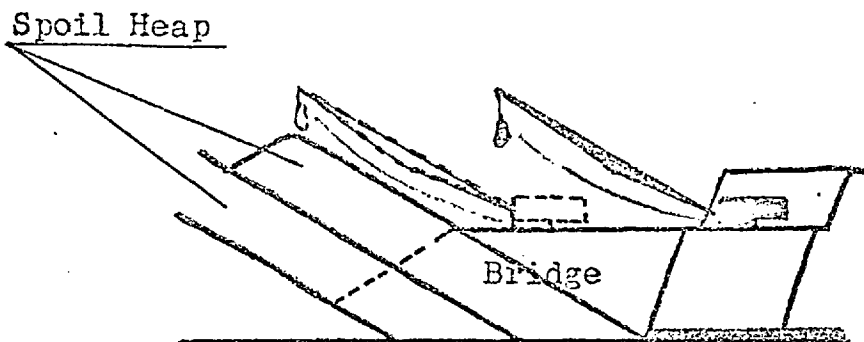


FIG 5.12 THE EXTENDED BENCH (overburden bridge) METHOD

excavation, adequate dumping height and the vee wedge, which is formed between highwall and spoil bank, to be so sized as to hold the volume of spoil to be rehandled. This system can be varied using:

- (a) a single dragline on one bench horizon,
e.g. Mercure Mine, S. Italy.
- (b) a single dragline on two bench horizons,
e.g. Reynolds Mining Co., Bauxite, Ark.,
U.S.A.
- (c) two draglines in tandem on the same
bench horizon
- (d) two draglines in tandem on two bench
horizons, e.g. Goonyella Mine, Queensland,
Australia.

Method (a) has the advantage of simplicity while (b) involves lost time due to "deadheading" but method (c) finds favour when advancing into deeper overburden. For very deep overburden method (d) has the advantage of increased safety due to the berms providing increased highwall and spoilbank stability. The method is described in Appendix 5D. This method has the disadvantage that when advancing into deeper overburden the dragline must have sufficient capacity to rehandle the spoil. The dragline then has excess capacity during the early life of the mine, i.e. there is some over-investment. The method is however attractive for generally thick overburden where the provision of a sufficiently long boom is either impractical or uneconomical.

Terrace or Forecut Method ⁵⁵ A forecut is excavated by a smaller dragline which dumps its spoil in reach of the larger, main dragline. The main dragline rehandles this spoil, swinging through 180° , in addition to excavating the main bench and direct casting its spoil. This method is simple to apply and has obvious advantages in unstable spoil. When advancing into deeper overburden it has the advantage that capital investment for the smaller, forecut dragline can be delayed, but of course there must be some over-investment since the main dragline will have excess capacity before rehandling starts. A typical example of this operation at Texas Gulf Sulphur Company's Lee Creek Mine, Aurora, N.C., U.S.A. is described in Appendix 5D.

The geometry of the stripping machine is best determined by drawing average and extreme pit sections and plans showing both stripping and loading operations. This must involve some trial and error. Fig 5.13 illustrates a pit in uniform conditions for which the following relationships can be derived, where:

P = Pit width

C = Cut width

R_d = Dumping radius (dragline)

R_s = Operating radius (shovel)

H_d = Dumping height (dragline)

H_s = Dumping height (shovel)

B = Haulage berm width

H = Overburden thickness

M = Mineral bed thickness

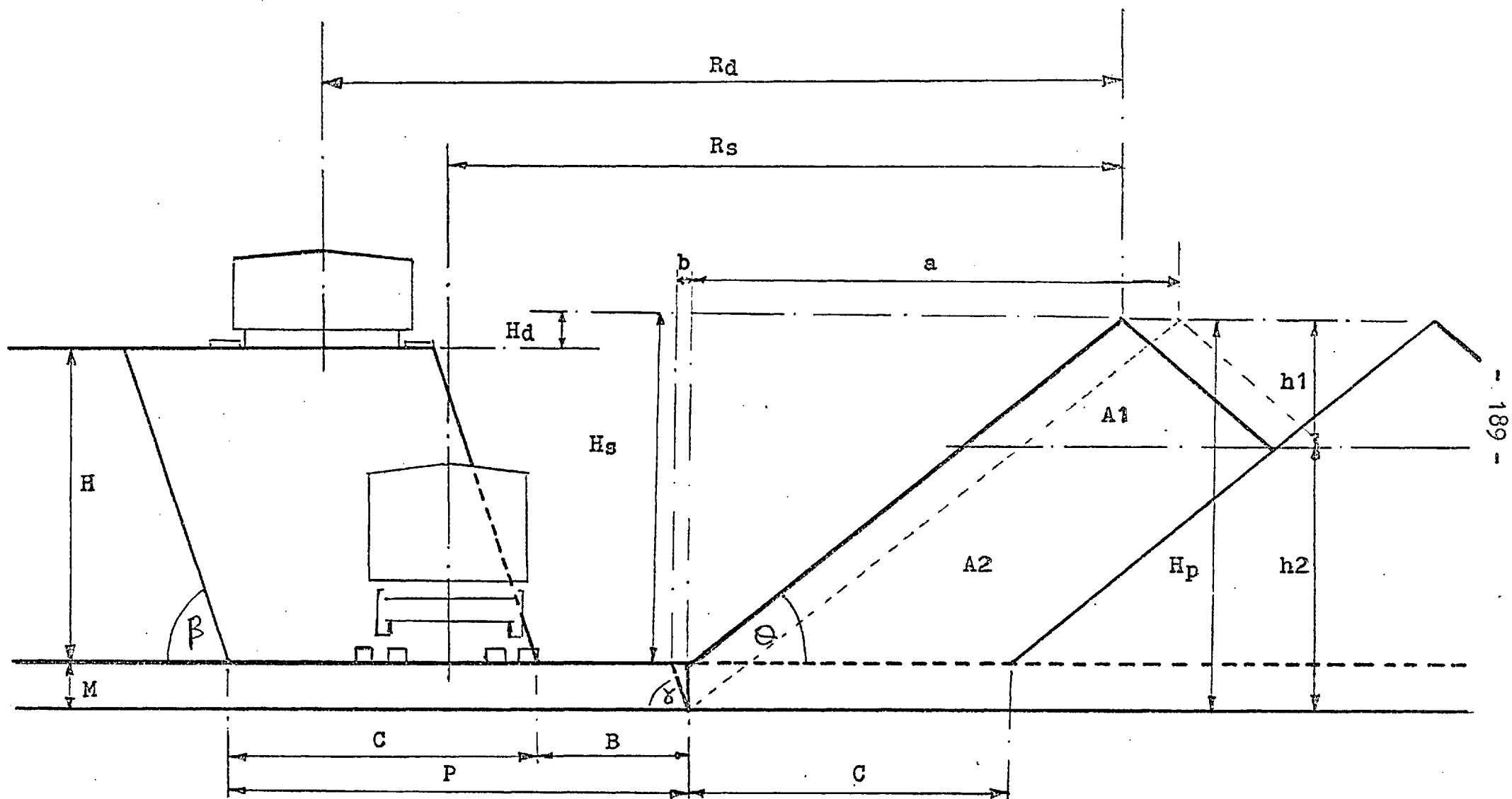


FIG 5.13 PIT SECTION PROPORTIONS

SF = Swell Factor

H_p = Spoil heap peak height

Θ = Angle of repose of spoil

γ = Slope angle of mineral

β = Slope angle of highwall

By inspection $h_1 = \frac{C}{2} \tan \Theta$

$$A1 = \frac{C^2}{4} \tan \Theta$$

But $CH \times SF = A1 + A2$ and $A2 = Ch_2$

$A2$ is shown made up of a parallelogram and a rectangle but is of equal area to a parallelogram of height h_2 and length C

therefore

$$Ch_2 + \frac{C^2}{4} \tan \Theta = CH \times SF$$

$$h_2 = H \times SF - \frac{C}{4} \tan \Theta$$

and

$$H_p = h_1 + h_2$$

$$= H \times SF - \frac{C}{4} \tan \Theta + \frac{C}{2} \tan \Theta$$

$$H_p = H \times SF + \frac{C}{4} \tan \Theta \dots\dots\dots (5.2)$$

therefore

$$H_s = H \times SF + \frac{C}{4} \tan \Theta - M \dots\dots\dots (5.3)$$

$$H_d = H \times SF + \frac{C}{4} \tan \Theta - M - H$$

$$H_d = H(SF-1) + \frac{C}{4} \tan \Theta - M \dots\dots\dots (5.4)$$

If a dragline is operating from an intermediate bench of height H_1 , then:

$$H_d = H \times SF + \frac{C}{4} \tan \theta - M - H_1 \dots\dots\dots (5.5)$$

To determine R_s and R_d it is necessary to locate the stripping machine. Formulae (5.6) and (5.7) are based on the following assumptions:

1. The shovel crawlers are located at the edge of the exposed mineral bed.
2. The dragline is located so that the shoes come to the top edge of the highwall.
3. The spoil is dumped so that the toe of the spoil heap reaches the top of the mineral bed. This means that H is much greater than M , which is usually the case.
4. It is also assumed that spoil is dumped between the spoil peaks to obtain more efficient use of the spoil space. This requires a slightly longer boom (when M is relatively thin). The spoil area then available is slightly greater than $CH \times SF$ but this is ignored and results in greater flexibility in operation.

Then $R_s = a + b + B + \frac{1}{2}$ crawler width

$$a = \frac{H_p}{\tan \theta} = \frac{H \times SF}{\tan \theta} + \frac{C}{4}$$

$$b = \frac{M}{\tan \delta}$$

$$\therefore R_s = \frac{H \times SF}{\tan \theta} + \frac{C}{4} + \frac{M}{\tan \gamma} + B + \frac{1}{2} \text{ crawlerwidth} \dots\dots\dots (5.6)$$

$$\text{and } R_d = a + b + c + B + \frac{1}{2} \text{ width over shoes}$$

$$C = \frac{H}{\tan \beta}$$

$$\therefore R_d = \frac{H \times SF}{\tan \theta} + \frac{C}{4} + \frac{M}{\tan \gamma} + \frac{H}{\tan \beta} + B + \frac{1}{2} \text{ width overshoes} \dots\dots\dots (5.7)$$

(See Table 5.III for crawler and shoe widths)

For thin mineral deposits it is usual to assume vertical seam sides and in many operations no berm is required. In these cases the B and M terms are eliminated from (5.6) and (5.7).

If a dragline is operating from an intermediate bench the R_d becomes:

$$R_d = \frac{H \times SF}{\tan \theta} + \frac{C}{4} + \frac{M}{\tan \gamma} + \frac{H_1}{\tan \beta} + B + \frac{1}{2} \text{ width overshoes} \dots\dots\dots (5.8)$$

Where the mineral is relatively thick it is usual for the toe of the spoil heap to coincide with the edge of the base of the mineral bed (Fig 5.14) for stability when the mineral is removed. In this case the dumping radii are:

$$R_s = \frac{H \times SF}{\tan \theta} + \frac{C}{4} + \frac{M}{\tan \gamma} + \frac{1}{2} \text{ crawler width} + d \dots\dots\dots (5.9)$$

where d = additional radii to allow spoil space to be more efficiently used.

d depends on the parameters governing pit geometry.

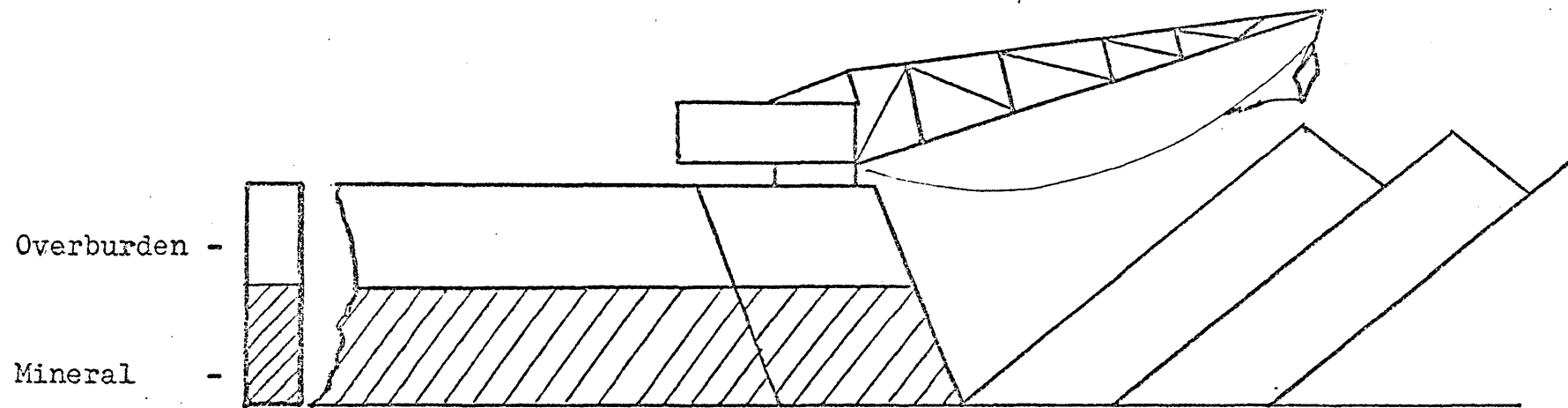


FIG 5.14 THICK MINERAL GEOMETRY

$$\text{and } R_d = \frac{H \times SF}{\tan \theta} + \frac{C}{4} + \frac{M}{\tan \gamma} + \frac{H}{\tan \beta} + \frac{1}{2} \text{ width overshoots} + d$$

..... (5.10)

Since in these conditions a berm is not usually required, the B term is eliminated.

In thick mineral operations it may be advantageous to load out the mineral with the stripping dragline, (see Fig 5.15). The pit plans and sections must be drawn in detail to determine the machine geometry, and account must be taken of the increased cycle time when loading as the bucket must be accurately spotted.

One manufacturer ⁵⁴ uses computer-generated tables covering a range of pit geometry variables but these are limited to values of θ and β found in U.S.A. coal mining.

Pit sections for uniform conditions can however be quickly prepared by the use of the simple graphical method shown in Fig 5.16. This method has considerable advantage in that the user is actually preparing sections rather than simply referring to formula or tables and obtains a "feel" for the situation. The following are known:

- overburden and mineral thicknesses
- safe slope of the highwall
- angle of repose of the spoil
- cut width (from loading and transport requirements)

The width of berm (if any) required for transport, horizontal blast hole drilling or other purposes can be decided and the pit width determined. The following procedure is then adopted

- (a) The section is partially drawn, the slope of the pit side of the spoil heaps only being indicated.

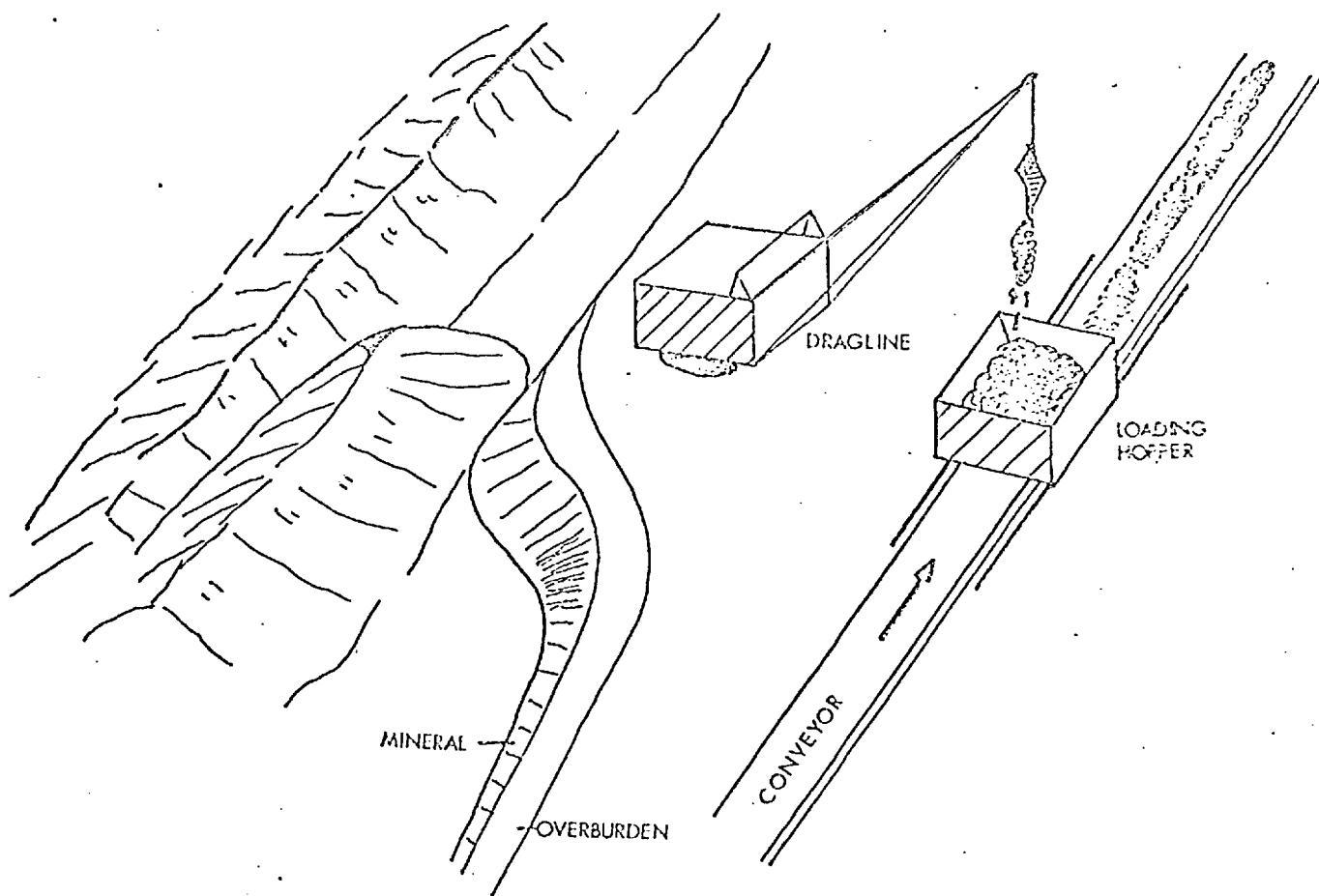


FIG 5.15 THICK MINERAL EXCAVATION METHOD

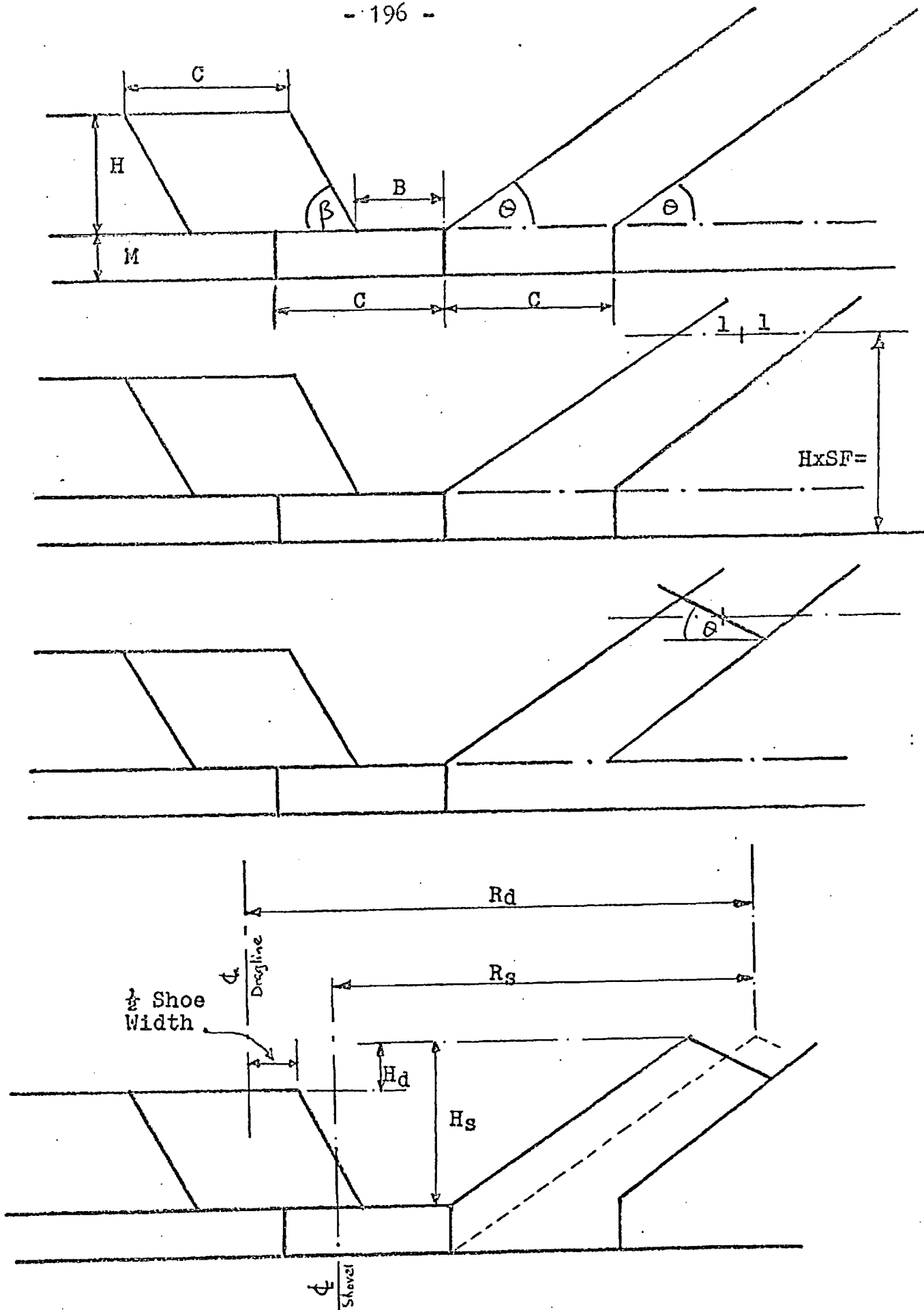


FIG 5.16 GRAPHICAL DETERMINATION OF CUT GEOMETRY

- (b) The chain line 1-1 is drawn, as shown, at distance $H \times$ swell factor parallel to the floor of the mine
- (c) 1-1 is bisected as shown and the peaks of the spoil heaps determined by drawing in the other face of the spoil heaps at the angle of repose of the spoil.
- (d) The dumping heights and dumping radii of shovels or draglines can then be determined knowing the widths over the crawlers or shoes. (See Table 5.III).

This provides a method of quickly determining the machine geometry for relatively flat deposits.

For inclined deposits the following general analysis is proposed:

Considering Fig 5.17

$$SF \times CH = \text{Area of Spoil} = SF \times C'H' + H_p C' - \text{Area A4}$$

$$\text{Area A4} = \frac{C'}{2} e \cdot \sin(\theta + \alpha)$$

$$\text{where } \frac{e}{\sin(\theta - \alpha)} = \frac{C'}{\sin(180^\circ - 2\theta)} = \frac{C'}{\sin 2\theta}$$

$$\therefore e = \frac{\sin(\theta - \alpha)}{\sin 2\theta} C'$$

$$\therefore \text{Area A4} = \frac{C'^2 \sin(\theta + \alpha) \cdot \sin(\theta - \alpha)}{2 \sin 2\theta}$$

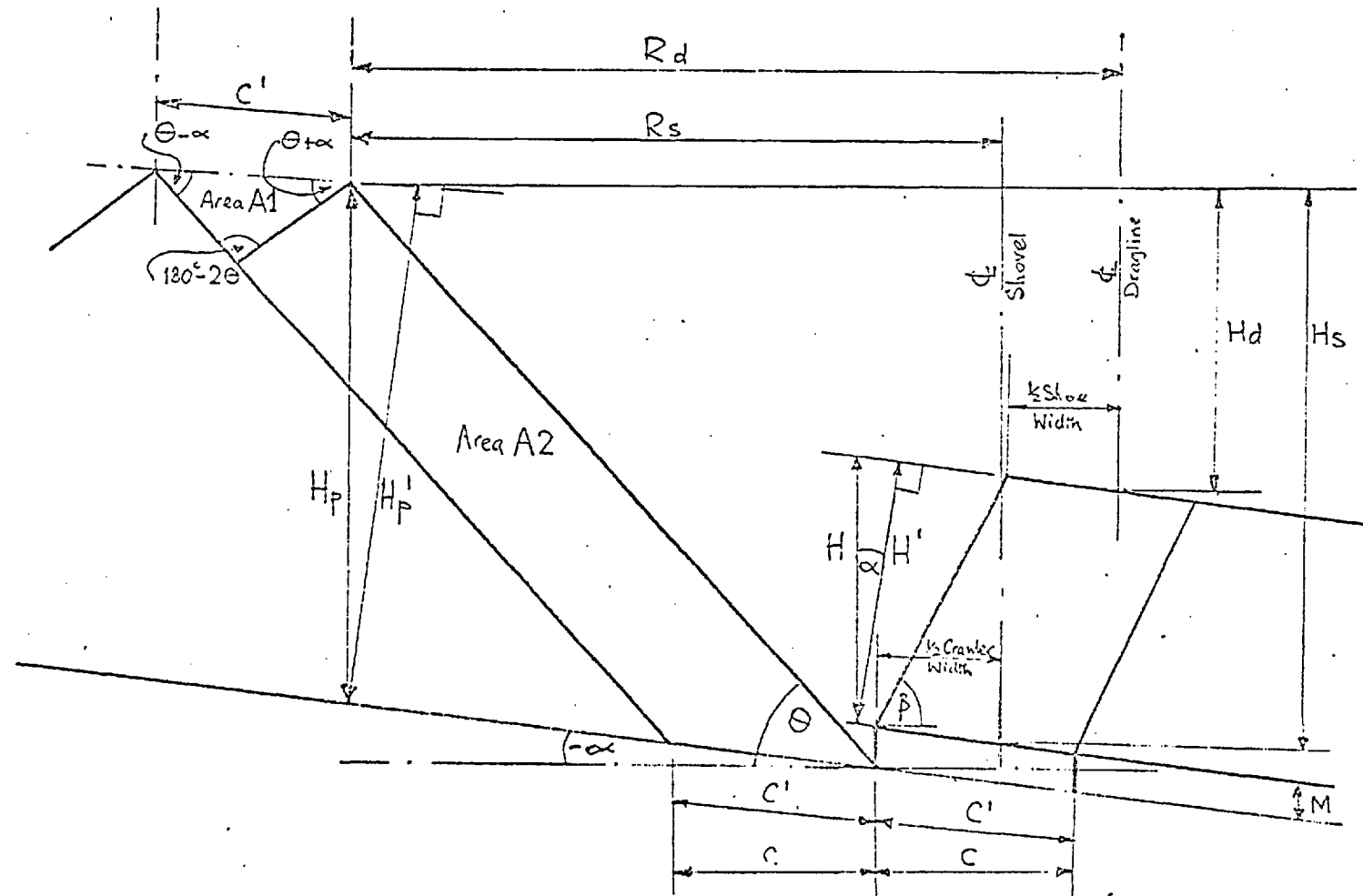


FIG 5.17 PIT SECTION FOR INCLINED DEPOSIT

$$\text{Area } A2 = H_p' C' = \frac{C'^2 \sin(\theta + \alpha) \cdot \sin(\theta - \alpha)}{2 \sin 2\theta}$$

$$= SF \times H' C' = H_p' C' = \frac{C'^2 \sin(\theta + \alpha) \cdot \sin(\theta - \alpha)}{4 \sin \theta \cos \theta}$$

$$= SF \times H' = H_p' = \frac{C' \sin(\theta + \alpha) \cdot \sin(\theta - \alpha)}{4 \sin \theta \cos \theta}$$

but $C' = \frac{C}{\cos \alpha}$; $H' = H \cos \alpha$; $H_p' = H_p \cos \alpha$

Substituting

$$SF \times H \cos \alpha = H_p \cos \alpha = \frac{C \sin(\theta + \alpha) \sin(\theta - \alpha)}{4 \cos \alpha \sin \theta \cos \theta}$$

$$H_p = SF \times H + \frac{C \sin(\theta + \alpha) \sin(\theta - \alpha)}{4 \cos^2 \alpha \sin \theta \cos \theta}$$

$$H_p = SF \times H + \frac{C (\sin \theta \cos \alpha + \cos \theta \sin \alpha) (\sin \theta \cos \alpha - \cos \theta \sin \alpha)}{4 \cos^2 \alpha \sin \theta \cos \theta}$$

$$H_p = SF \times H + \frac{C (\sin \theta \cos \alpha)^2 - (\cos \theta \sin \alpha)^2}{4 \cos^2 \alpha \sin \theta}$$

Dividing through RH term

$$H_p = SF \times H + \frac{C}{4} \left(\tan \theta - \frac{\tan^2 \alpha}{\tan \theta} \right)$$

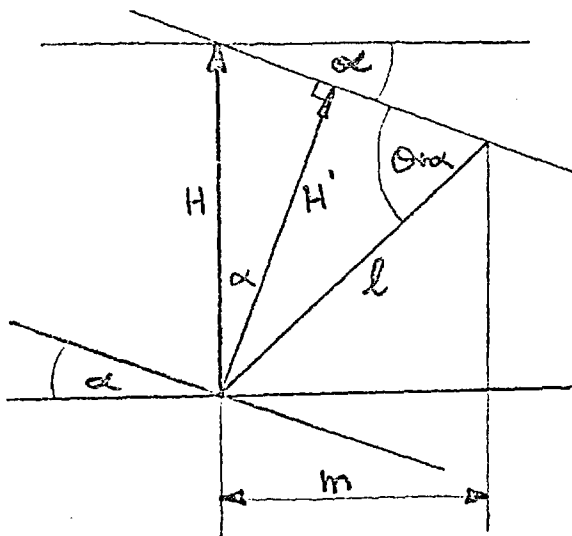
$$H_p = SF \times H + \frac{C}{4} \tan \theta \left(1 - \frac{\tan^2 \alpha}{\tan^2 \theta} \right) \text{----- (5.11)}$$

i.e. H_p is a linear function of H and C and a parabolic function of $\tan \alpha$

To determine the MACHINE GEOMETRY
i.e. R_d, R_s, H_d, H_s

Referring To Fig 5.17

H' is known - To find l



$$\sin(\theta + \alpha) = \frac{H'}{l}$$

$$\therefore l = \frac{H'}{\sin(\theta + \alpha)}$$

$$\text{and } m = l \cos \theta$$

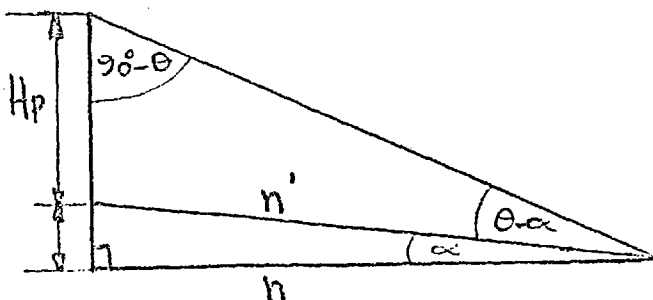
$$\therefore m = \frac{H' \cos \theta}{\sin(\theta + \alpha)}$$

$$H' = H \cos \alpha$$

$$\therefore m = \frac{H \cos \alpha \cdot \cos \theta}{\sin(\theta + \alpha)}$$

$$= \frac{H \cos \alpha \cdot \cos \theta}{\sin \theta \cos \alpha + \cos \theta \sin \alpha}$$

$$m = \frac{H}{\tan \theta + \tan \alpha}$$



$$\frac{n'}{\sin(90^\circ - \theta)} = \frac{H_p}{\sin(\theta - \alpha)}$$

$$\frac{n'}{\cos \theta} = \frac{H_p}{\sin(\theta - \alpha)}$$

$$\therefore n' = \frac{H_p \cos \theta}{\sin(\theta - \alpha)}$$

$$n' = \frac{H_p \cos \theta}{\sin \theta \cos \alpha - \cos \theta \sin \alpha}$$

$$n' = \frac{n}{\cos \alpha}$$

$$\therefore n = \frac{H_p \cos \alpha \cos \theta}{\sin \theta \cos \alpha - \cos \theta \sin \alpha}$$

$$n = \frac{H_p}{\tan \theta - \tan \alpha}$$

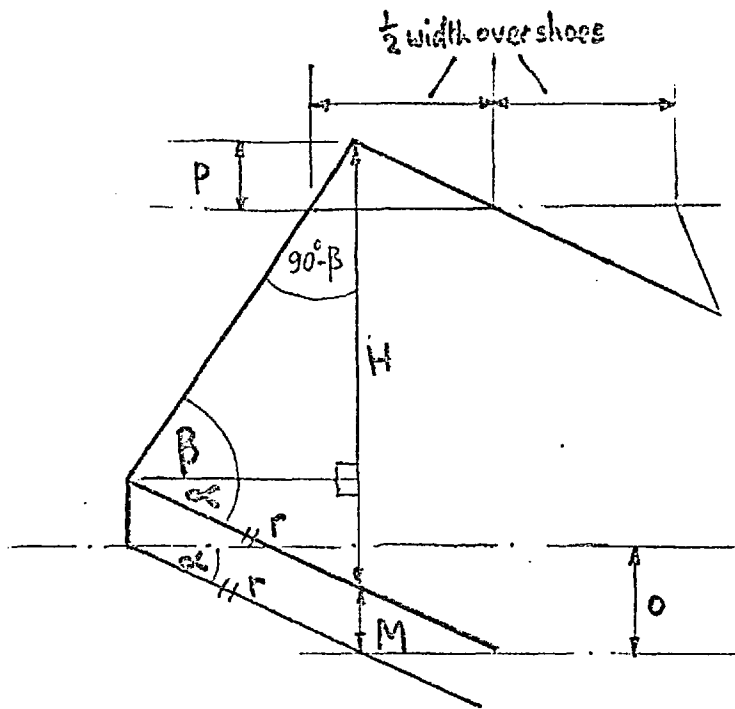
$$\text{Now } R_d : m + n + \frac{1}{2} \text{ width over shoes} \dots\dots\dots (5.12)$$

By inspection

$$R_s : n + \frac{1}{2} \text{ crawler width}$$

$$R_s : \frac{H_p}{\tan \theta - \tan \alpha} + \frac{1}{2} \text{ crawler width} \dots\dots\dots (5.13)$$

N.B. Both (5.12) and (5.13) assume that the stripping unit stands on a horizontal surface. This is a valid assumption since a horizontal track would normally be prepared for the operation of, a dragline and for any differences beyond the limits of the levelling jacks for a shovel.



To find H_d

Height of dragline tub
above base line

$$= H + M - O - P$$

and

$$\frac{H}{\sin(\beta + \alpha)} = \frac{r}{\sin(90^\circ - \beta)}$$

$$\therefore r = \frac{H \cos \beta}{\sin(\beta + \alpha)}$$

but $O = r \sin \alpha$ $\therefore \frac{H \cos \beta \sin \alpha}{\sin(\beta + \alpha)}$

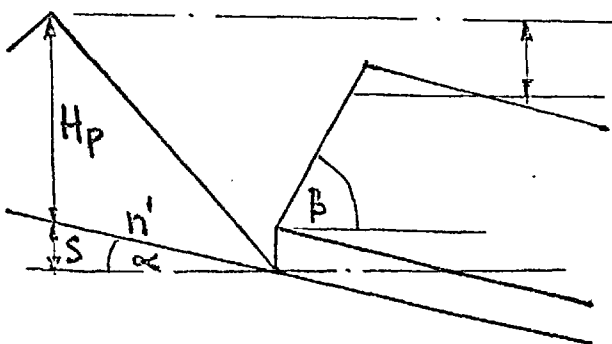
$$= \frac{H}{\tan \beta \cot \alpha + 1}$$

and $P = \frac{1}{2} \text{ width overshoots } \cdot \tan \alpha$

$\therefore$ Height of Tub above baseline =

$$= H + M - \frac{H}{\tan \beta \cot \alpha + 1} - \frac{1}{2} \text{ width overshoots } \cdot \tan \alpha$$

Height of spoil peak above base line = $H_p + S$



$$n' = \frac{H_p \cos \theta}{\sin(\theta - \alpha)}$$

and $S = n' \sin \alpha$

$$S = \frac{H_p \cos \theta \sin \alpha}{\sin(\theta - \alpha)}$$

∴ height of spoil bank peak above base line

$$= \frac{H_p \cos \theta \sin \alpha}{\sin \theta \cos \alpha - \cos \theta \sin \alpha} + H_p$$

$$= \frac{H_p}{\tan \theta \cot \alpha - 1} + H_p$$

$$= H_p \left(\frac{1}{\tan \theta \cot \alpha - 1} + 1 \right)$$

$$\therefore H_d = H_p \left(\frac{1}{\tan \theta \cot \alpha - 1} + 1 \right) - H - M + \frac{H}{\tan \beta \cot \alpha + 1} +$$

$\frac{1}{2}$ width over shoes. $\tan \alpha$

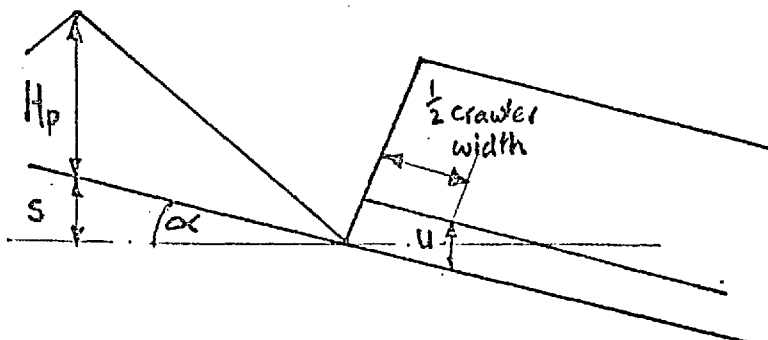
$$\therefore H_d = H_p \left(\frac{1}{\tan \theta \cot \alpha - 1} + 1 \right) - H \left(1 - \frac{1}{\tan \beta \cot \alpha + 1} \right)$$

$$- M + \frac{1}{2} \text{ width over shoes } \tan \alpha \text{ ----- (5.14)}$$

To determine H_s

By inspection

$$H_s = H_p + s - u$$



$$V = \frac{1}{2} \text{ crawler width, } \sin \alpha$$

$$\therefore U = M \cos^2 \alpha - \frac{1}{2} \text{ crawler width. Since}$$

$$\therefore H_s = H_p + \frac{H_p \cos \theta \sin \alpha}{\sin(\theta - \alpha)} - M \cos^2 \alpha + \frac{1}{2} \text{ crawler width} \cdot \sin \alpha$$

$$\therefore H_s = H_p \left(1 + \frac{\cos \theta \sin \alpha}{\sin(\theta - \alpha)} \right) - M \cos^2 \alpha + \frac{1}{2} \text{ crawler width} \cdot \sin \alpha$$

----- (5.15)

Triangle XYZ represents the volume which must be rehanded. This is normally represented as a percentage of the total overburden and results in additional materials handling. It is important therefore to reduce rehandling to a minimum.

The height H_0 of the spoil bench horizon above the top of the mineral bed must therefore be a minimum. The dragline bench surface may be assumed to be horizontal (in practice this is so) as α is small.

The volume of material to be rehandled is from geometric considerations:

$$V_r = \frac{(H_o + M)^2}{2} \cdot (\cot \alpha + \cot \beta)$$

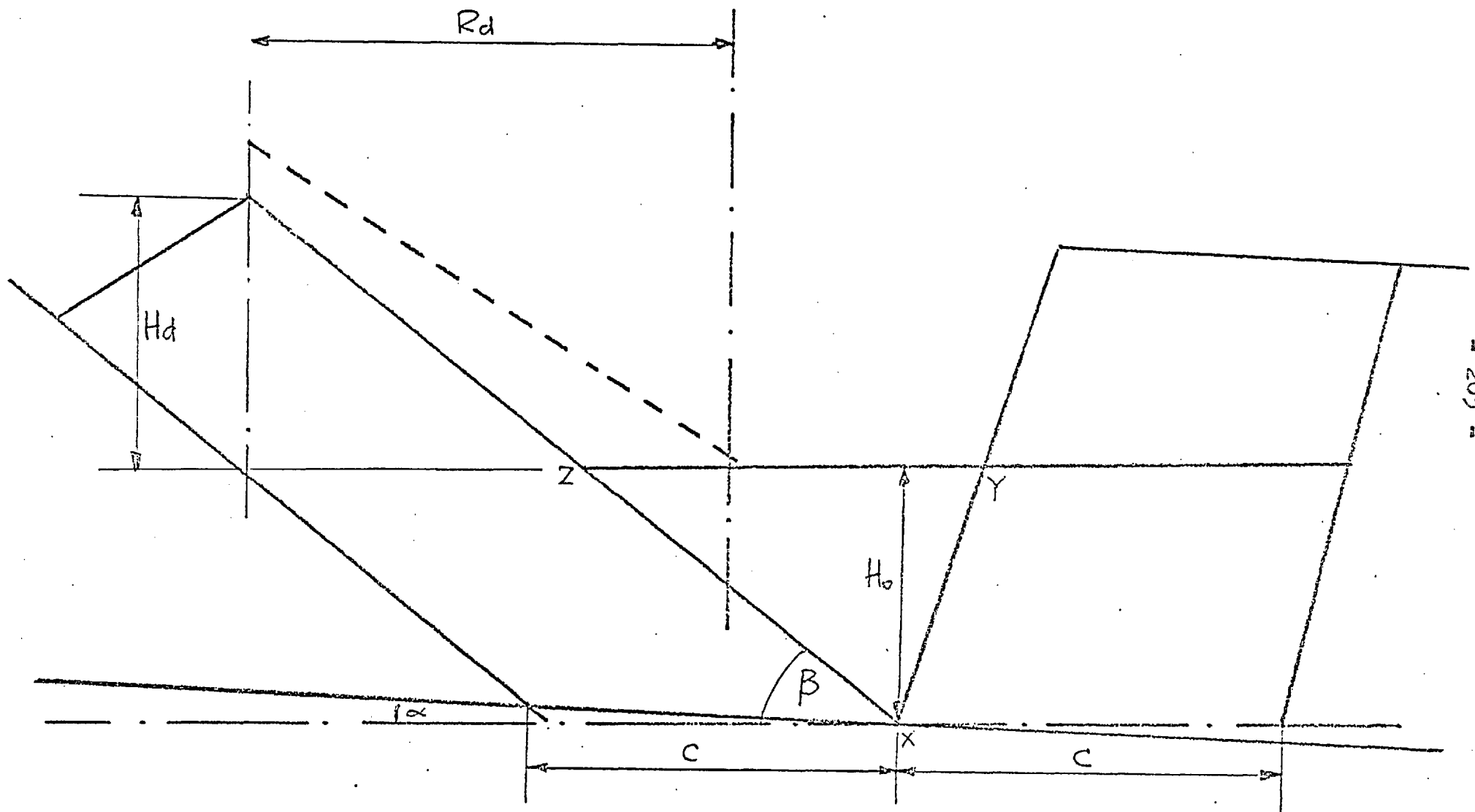


FIG 5.18 OPTIMUM SPOIL BRIDGE HEIGHT FOR MINIMUM REHANDLING

The volume of overburden to be removed is:

$$V = CH$$

∴ the percentage rehandling is

$$= \frac{(H_o + M)^2 \cdot (\cot \alpha + \cot \beta)}{2 CH}$$

By inspection for H_o to be a minimum the following two formulae must be simultaneously satisfied:

$$(H_p - H_o - M) \cot \beta + \frac{1}{2} \text{ width over shoes} = R_d$$

and

$$H_p - H_o = H_d$$

$$\text{Hence } H_o = \frac{SF \times H + \frac{1}{4} \tan \beta \left(1 - \frac{\tan^2 \alpha}{\tan^2 \beta}\right) - H_d}{1 + \cot \beta \tan \alpha} \text{----- (5.16)}$$

for a minimum

∴ the percentage rehandling is :

$$\frac{\cot \alpha + \cot \beta}{2 CH} \left[\frac{SF \times H + \frac{1}{4} \tan \beta \left(1 - \frac{\tan^2 \alpha}{\tan^2 \beta}\right) - H_d}{1 - \cot \beta \tan \alpha} \right] \text{----- (5.17)}$$

In practice the percentage rehandling is usually determined by trial and error graphical methods. Using formulae (5.16 and 5.17) it becomes possible to arrive at an accurate figure. This must then be followed by drawing several pit sections, to determine any practical limitations of the machine geometry.

STRIKE ADVANCE - INCREASING OVERBURDEN THICKNESS TO DIP

A further problem occurs when for blending or other reasons it is decided to advance along the strike, with a dip-wise cut, in a deposit where the overburden thickness increases to the dip. In this situation the stripping machine must have a long reach and a high capacity, i.e. an expensive machine, in the thicker overburden while a short reach and a low capacity i.e. a low cost machine, would meet the stripping requirements in the thinner overburden. In these circumstances some form of compromise is necessary to avoid (a) over-capitalisation or (b) production shortfall.

An actual example first used by the author when devising the method is the best means of illustrating the selection procedure.

A deposit in flat topography dips at 1 in 120. The mineral deposit is 4.1m thick and the overburden starts at 9m thick, increasing in thickness to 40m where 50m high sand dunes are encountered. The technical investigations indicate clearly that a walking dragline is most suitable for stripping. The loading operation requires a cut width of 18m, no berm is needed i.e. $P = C$. A cut diagram in 40m thick overburden indicates that a reach of 86.5m is necessary, a well-tried machine with this reach but with inadequate stripping capacity in 40m thick overburden has only 60% of the capital cost of a machine with both adequate reach and able to meet the maximum stripping requirement. If the cheaper machine can expose sufficient reserves when working in thinner overburden to compensate for the lack

of capacity in thick overburden, it becomes the obvious choice. This can be checked using the method devised in Fig 5.19.

The following curves are plotted in Fig 5.19

Pit Advance/day v Overburden Depth
(for the dragline selected)

Pit Advance/day v Annual Mineral Production

Spot Overburden Ratio v Overburden Depth

Average Overburden Ratio* v Overburden depth

*Mean starting at 9m up to the spot overburden depth.

Using the procedure shown in Fig 5.19 for an annual production requirement of 2.5 million m^3 of mineral:

- | | |
|---------------------------|---|
| AB → BC → CD | 1) Pit advance 94m/day |
| AB → BC → CE | 2) Average overburden thickness
= 21.25m |
| AB → BC → CF | 3) Spot Overburden Ratio = 5.2 |
| AB → BC → CF ¹ | 4) Average Overburden Ratio = 3.8 |
| AB → BC → FG → GH | 5) Maximum overburden thickness
= 33m |

i.e. The dragline cannot strip overburden deeper than 33m if it is to expose 2.5 million m^3 of mineral per annum.

For cut-off at 40m thick overburden

Using Fig 5.19, projecting from D = 39m

(1m cleared by bulldozer)

- | | |
|------------------------|---|
| IJ | 1) Average Overburden Ratio = 5.93 |
| IJ → JK → KL → LM | 2) Average Overburden Depth = 24.25m |
| IJ → JK → KL → LN → NO | 3) Pit Advance = 83m/day |
| IJ → JK → KL → LN → NP | 4) Mineral Production = 2.2 million
m^3 /year. |

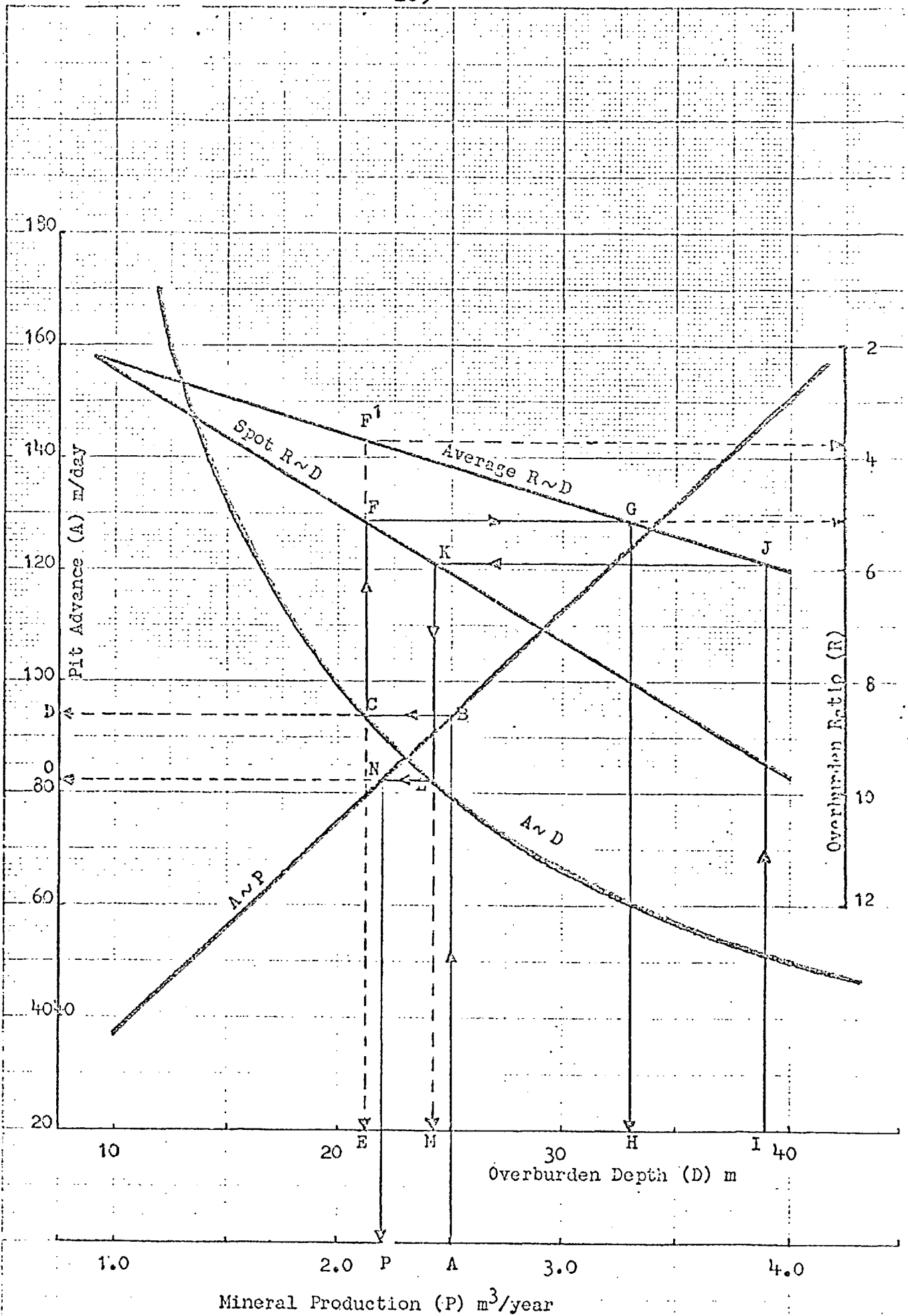


FIG 5.19 GRAPHICAL METHOD OF DETERMINING PRODUCTION AND OVERBURDEN REQUIREMENTS

i.e. there will be a production short fall of 300,000 m³/year.

In this particular case it was decided to select this machine despite the production shortfall because:

- a) the lower capital cost
- b) an increase in production up to 4.0 million m³/year was planned for a future date by the purchase of a second machine. Two machines can strip for a production of 4.4 million tonnes/year, and
- c) a forecut operation in the deeper overburden using mobile equipment proved a less expensive short term expedient to produce 2.5 million m³/year.

The method allows rapid analysis for any particular machine. The A v D curve requires 3 points to plot but all the other curves (straight lines) only require two points.

Preliminary Dimensions

For initial analysis it is necessary to know the major dimensions of suitable machines before proceeding to final selection. These may be obtained from manufacturers literature. Table 5.III are a summary of approximate stripping machine specification's compiled from leading manufacturers models.

TABLE 5.III

APPROXIMATE SPECIFICATIONS OF STRIPPING MACHINES

Walking draglines										
Bucket size		Dumping radius		Dumping height		Width over shoes		Machine mass,t	Ballast, t	FOB price,
yd ³	m ³	m	ft	m	ft	m	ft			£ x 1000*
15	11.5	50	165	21	70	14	45	470	115	385
20	15	58	190	23	75	16	52	550	160	540
40	31	67	220	26	85	22	72	1250	200	1130
50	38	79	260	30	100	23	76	1950	250	1600
60	46	84	275	37	120	27	90	2900	320	2500
90	70	92	300	41	135	32	105	4200	340	2900
110	85	92	300	44	145	35	115	5700	370	4300
Stripping shovels										
Dipper size		Dumping radius		Dumping height		Width over crawlers		Machine mass,t	Ballast, t	FOB price
yd ³	m ³	m	ft	m	ft	m	ft			£ x 1000*
40	31	38	126	27	90	13.5	44	1300	350	1200
60	46	43	140	30	100	14.5	48	2000	600	1950
80	62	49	160	37	120	17.5	58	3500	870	3000
90	70	55	180	43	140	18.5	60	4100	950	3350
100	77	60	195	44	145	20	65	4900	1050	4100
120	92	63	205	46	150	23	75	7200	1350	5100
180	140	66	215	46	150	27	88	11000	1650	8500

*1973 projections : freight, insurance, erection or ballast not included.

Final Selection

Having determined the bucket or dipper size, the dumping radius and dumping height, it is not always possible to select a machine with perfect dimensions and the nearest model may have to be adopted. Usually an increase in dumping radius can be achieved by a reduction in dumping height, and vice versa. Careful investigation is essential to avoid selecting an oversize machine, and, hence, incurring over-investment. Often large stripping machines are not catered for by standard models, and investigations and discussions with manufacturers are essential before the final design is settled.

When the machine has been selected it is necessary to recalculate its production using the manufacturer's cycle times and swing factors. If a single manufacturer has not been selected, Table 5.IV may be used for estimating cycle times.

TABLE 5.IV

THEORETICAL CYCLE TIMES FOR STRIPPING MACHINES (sec)

Dipper or bucket size		Draglines				Shovels		
yd ³	m ³	90°	120°	150°	180°	90°	120°	150°
Up to 19	Up to 15	55	62	69	77	51	57	63
20-34	16-26	56	63	70	78	52	58	64
35-59	27-44	57	64	71	79	53	59	66
60-74	45-57	59	65	72	80	54	61	67
175-120	58-92	60	66	73	81	55	62	68
120-200	93-150	62	69	76	84	57	63	70

The figures given in Table 5.IV have been compiled from a wide range of manufacturers' literature, and are based on normal geometry machines i.e. for stripping shovels.

Average overburden thickness = $0.4 \times \text{maximum dumping radius}$

for walking draglines

Average overburden thickness = $0.3 \times \text{maximum dumping radius}$

For extra-long boom machines add 0.75 sec to the cycle time for each 1m that the above reduced radii dimensions exceed the overburden thickness.

OWNERSHIP COSTS-STRIPPING MACHINES

The format previously used for loading shovels and crawler draglines may also be used for single-bucket stripping machines. A stripping machine can have a useful life in excess of 30 years, but company policy, tax concessions, etc., influence the depreciation period. It is usual to schedule large stripping machines to operate up to 350 days/year on a three shift basis of 22.5 scheduled hours.

It is not possible to define erection costs as simply as for loading shovels, for both the dumping radius and the dumping height, as well as the dipper or bucket size influence the erection cost of stripping machines. The mass (service weight) however provides a reasonable guide in the absence of manufacturers' figures. The mass (service weight) may be obtained from manufacturers' literature or from Rumfelt's MUF ~ service weight curves ⁵⁶. It should be noted however that these curves use the reach of the machine and not the dumping radius. The erection costs based on 1973 projections are:

$$\text{Erection costs } £(\text{stg}) = cW \dots\dots\dots (5.18)$$

where

Imperial Units

W = machine mass (service weight), lb

c = 0.036 for stripping shovels

c = 0.049 for walking draglines

S.I. Units

W = machine mass (service weight) tonnes

c = 16 for stripping shovels

c = 22 for walking draglines

The figures are obtained on the same basis as those for loading shovels (Table 3.VIII): they should be adjusted as necessary where conditions warrant.

Some care is necessary with very large machines as erection costs may include on-site fabrication and machining of large components. In these circumstances very large capital sums are involved and detailed discussion and negotiation with the manufacturer is essential.

The trailing cable should form part of an integrally engineered electrical system, but for preliminary purposes, the cost figures in Table 5.V may be used.

TABLE 5.V

TRAILING CABLE PRICES - STRIPPING MACHINES

(£stg/m*)

Dipper or Bucket Capacity		Voltage kV		
yd ³	m ³	5-10	10-15	22-25
Up to 19	Up to 15	5.25	5.7	-
20 - 34	16 - 26	9.0	9.5	-
35 - 59	27 - 44	18.5	14.5	-
60 - 74	45 - 57	-	23.0	13.5
75 - 120	58 - 92	-	33.5	24.0
120 - 200	93 - 150	-	43.0	28.0

*Based on a copper price of £790/ton: 1973 projected prices.

OPERATING COSTS - STRIPPING MACHINES

The format used for loading shovels may also be used for calculating the operating costs of single bucket stripping machines. Maintenance and Supply Costs are calculated in the same way except that the following figures should be used in item 25:

Walking draglines 7% *

Stripping shovels 8% *

*based on figures obtained from collected data plus data from Marion Power Shovel Co. and Bucyrus Erie.

The correction factor H does not generally apply as it is usual to operate the maximum possible number of hours and the above figures are based on an annual operation of at least 7000 hours.

The correction factor M for different types of overburden is shown in Table 5.VI. Some account should also be taken of the abrasiveness of the overburden, e.g. if the overburden is highly abrasive and badly fragmented, then M would be taken as 1.25.

TABLE 5.VI

CORRECTION FACTOR FOR DIFFERENT OVERBURDEN MATERIALS

Overburden Conditions	M
Light blasting	0.9
Medium blasting	1.0
Heavy blasting	1.1
Bad fragmentation	1.2

TOTAL OWNERSHIP AND OPERATING COSTS - STRIPPING MACHINES

The total ownership and operating costs, the cost per unit volume (bank) and the stripping cost per tonne of mineral can be calculated in the same general way as for loading shovels. No administrative costs or amortization charges are included. On this basis, the costs calculated from the operations previously listed vary from 2.2 to 4.6 p/m³ (bank) (1.7-3.5p/yd³) (bank), long reach machines with small buckets being at the higher end of the scale.

SELECTION OF BUCKETS AND DIPPERS

Abrasion and wear of buckets and dippers cannot be eliminated, but liner plates, shrouds, etc. can be used to obtain longer life. The increased weight involved however means that longer life can only be provided by a reduction in capacity. By correct location of liner plates, etc., and by careful selection of the materials used, wear can be economically counteracted.

For walking draglines lightweight buckets can provide maximum capacity, but the actual drag pull per unit volume excavated is reduced, thus it can only be used for relatively light, easily loaded materials. Their selection should be restricted to conditions where wear life and maintenance costs are not critical factors. If a spare bucket is available or if maintenance can be scheduled to avoid downtime, a lightweight bucket may be economic.

Medium weight buckets are generally selected and have the widest application.

For severe duties, or operations where it is uneconomic to hold a spare bucket, e.g. small mines or short life operations which require at least 14,000 hours of operation

before major repairs are carried out, heavy-duty buckets of lower capacity are needed.

Bucket capacity is primarily dependent on the maximum allowable suspended load of the dragline and the mass of the bucket plus rigging. Thus for abrasive conditions where an oversize drag chain is used to combat wear, the bucket capacity would have to be reduced.

Although stripping shovels are often selected to handle dense, poorly fragmented overburden, there has been a recent trend towards lighter, higher capacity, fabricated dippers and considerations similar to those used for draglines buckets apply.

All dippers whether for stripping or for loading, should have smooth, correctly profiled interiors to promote flow of rock through them and a wide variety of designs is available. Equally important is the provision of special shroud, lip and teeth fastenings which allow them to be replaced quickly.

Considerable investigation of the mechanical design of buckets and dippers should precede their selection.

CONCLUSIONS

Stripping machines represent huge, long term investments and warrant considerable initial investigation. In many parts of the world deposits with thin to medium overburden are no longer available for exploitation and unless huge machines with excessively long booms are adopted, rehandling of spoil will be essential. Most informed opinion considers that a plateau has been reached in machine size with the development of "Big Muskie" in 1969, because its initial

performance caused some difficulties. It is evident therefore that further work on rehandling methods is required. A number of examples are described in Appendix 5D.

Chapter 3 indicates the need for a better understanding of the operation of the loading shovel as a system and advocates some form of analogue computer for these studies. The same arguments can be even more forcibly applied to large stripping machines which represent much larger capital investments.

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APPENDIX 5.A

THE APPLICATION OF LARGE SINGLE BUCKET STRIPPING
MACHINES ON WEAK ELECTRIC POWER SYSTEMS

In Chapter 5 the effect of large single bucket stripping machines on relatively weak power systems is briefly discussed in qualitative terms and in certain conditions power system instability i.e. voltage and frequency disturbances, can occur which interfere with the normal operation of all equipment connected to the system, e.g. repeated m-g set driving motor "pull out" causing loss of production, low and high voltage problems for other equipment, etc.

Referring to Fig 5.5 the fluctuating load imposed by the dragline on the power system causes the system voltage to vary. As the electricity supply company attempts to hold the sending voltage constant, the receiving voltage must vary. In bad cases this can cause equipment malfunctions. One partial solution is to vary the reactive power flow (VAR's) as load power conditions change by varying the field strength of the synchronous motors driving the dc generators used to supply drag, hoist and swing motions. With positive (motoring loads) the power factor is forced leading, and during regeneration it is allowed to lag. The result is an approximately constant receiving voltage for all conditions of load. (See Fig 5A1). Unfortunately this cannot be done without limit: the pull-out torque of the synchronous motors cannot be exceeded, hence limiting the amount by which the power factor can lag. Similarly, saturation of the synchronous motor iron circuit and the thermal capabilities of the field restrict the amount of leading reactive power that can be caused to flow. Within these limitations however reactive power control is effective

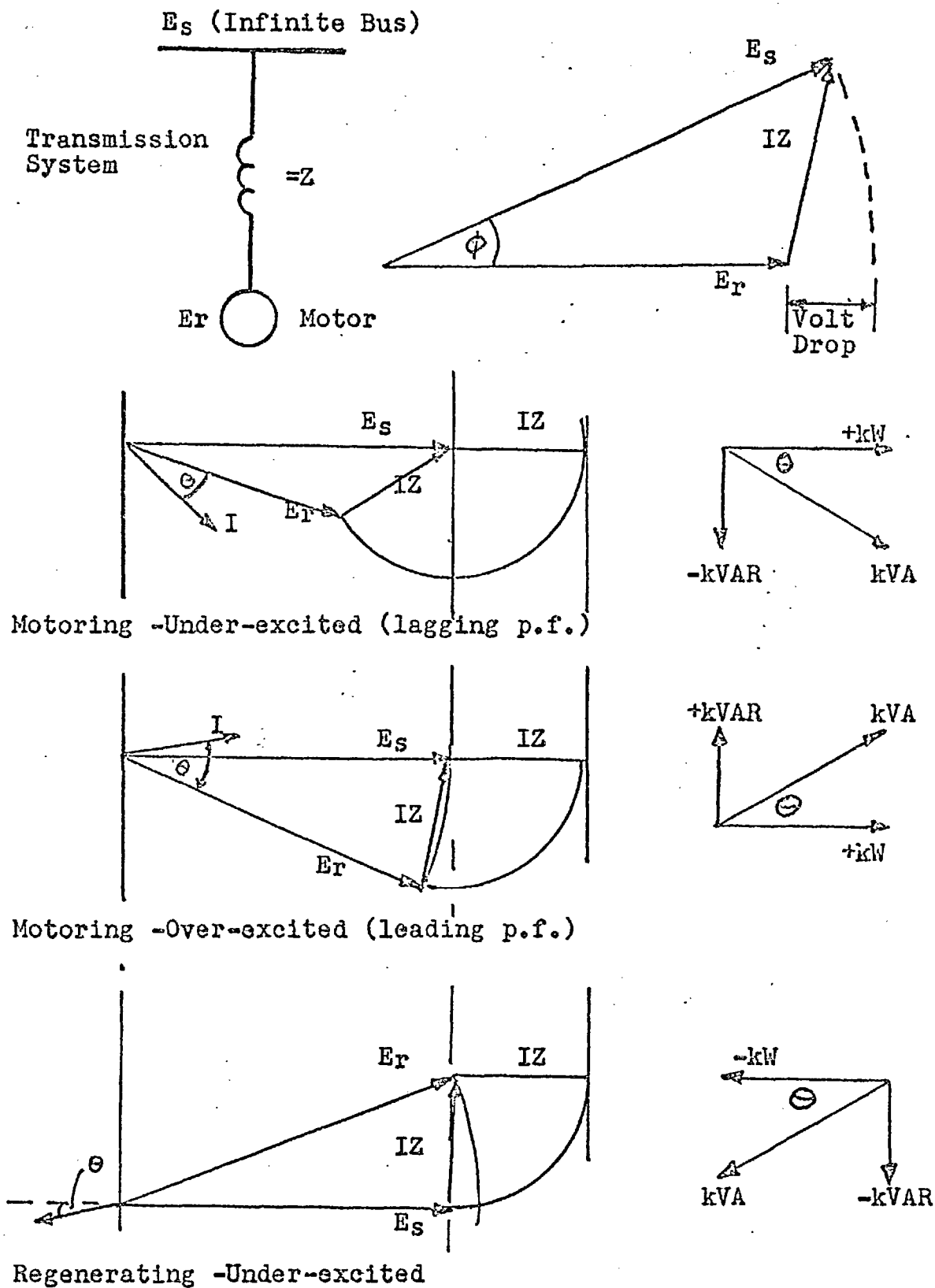


FIG 5A1 MAINTENANCE OF APPROXIMATELY CONSTANT RECEIVING VOLTAGE BY M-G SET EXCITATION CONTROL

in handling voltage fluctuations. The real power flow (kW) however remains unchanged.

The variation in kW load (Fig 5.5) is reflected into energy flow changes at the power system generating stations. Brief energy flow changes are accommodated by small changes in alternator speed but as the duration of each power disturbance increases, governor action occurs. In a steam-powered station the governor changes the steam flow in an attempt to keep the speed (and consequently frequency) deviations small. In practice the effectiveness of minimising frequency change may be a compromise between the rapidity of the governor operation and the increased maintenance of the regulating equipment resulting from increasing the duty on the governor.

Often the governor may be capable of following and compensating for the approximately one minute load cycle of the dragline, but the generating station boilers may not be. The one minute cycle tends to disturb boiler firing and boiler water-level control. As steam is drawn from the boiler, the steam pressure tends to drop. As the pressure lowers, the water in the boiler drum begins flashing to steam and thus the liquid becomes filled with vapour bubbles. Because the mixture of vapour and liquid is less dense than liquid alone and since there has been no appreciable change in mass, the volume occupied by the mixture is suddenly greater than that of the liquid alone. This phenomena ("swelling") may be sufficient to cause liquid to be carried over into the turbine, causing catastrophic damage.

There is also a possibility of cumulative action.

As swelling occurs, boiler level controls reduce the feedwater inflow and the swelling also reduces heat transfer. As the load drops so do the steam requirements and the liquid contracts to a now lower and cooler level. Firing and boiler feedwater flow are increased and a more violent fluctuation occurs during the next cycle.

Frequency variations may also affect other equipment, though the average frequency is constant e.g. clock timing, electronic equipment, etc., so indirect problems can also occur.

Voltage or frequency variations are unlikely to be problems on large interconnected power systems such as in the U.S.A. and U.K. where it is possible to tolerate cyclic power fluctuations of 5 - 10% of the connected, running, steam-generating capacity and where most power lines have the capacity to keep voltage swings within commonly accepted flicker limits. On small systems however, the magnitude of the fluctuating loads can be serious.

Initial Investigations This situation first came to the notice of the author while investigating the use of up to three draglines for a re-casting operation for the Neyveli lignite deposit in 1956, where an adequate power supply would not be available during the early life of the mine. Comparable situations where field measurements could be taken were not readily available at that time, nor were computing facilities for the use of "Monte Carlo" simulation techniques which were then relatively unknown. An attempt was made to determine the resultant load for two draglines using the following "brute force" simulation method:

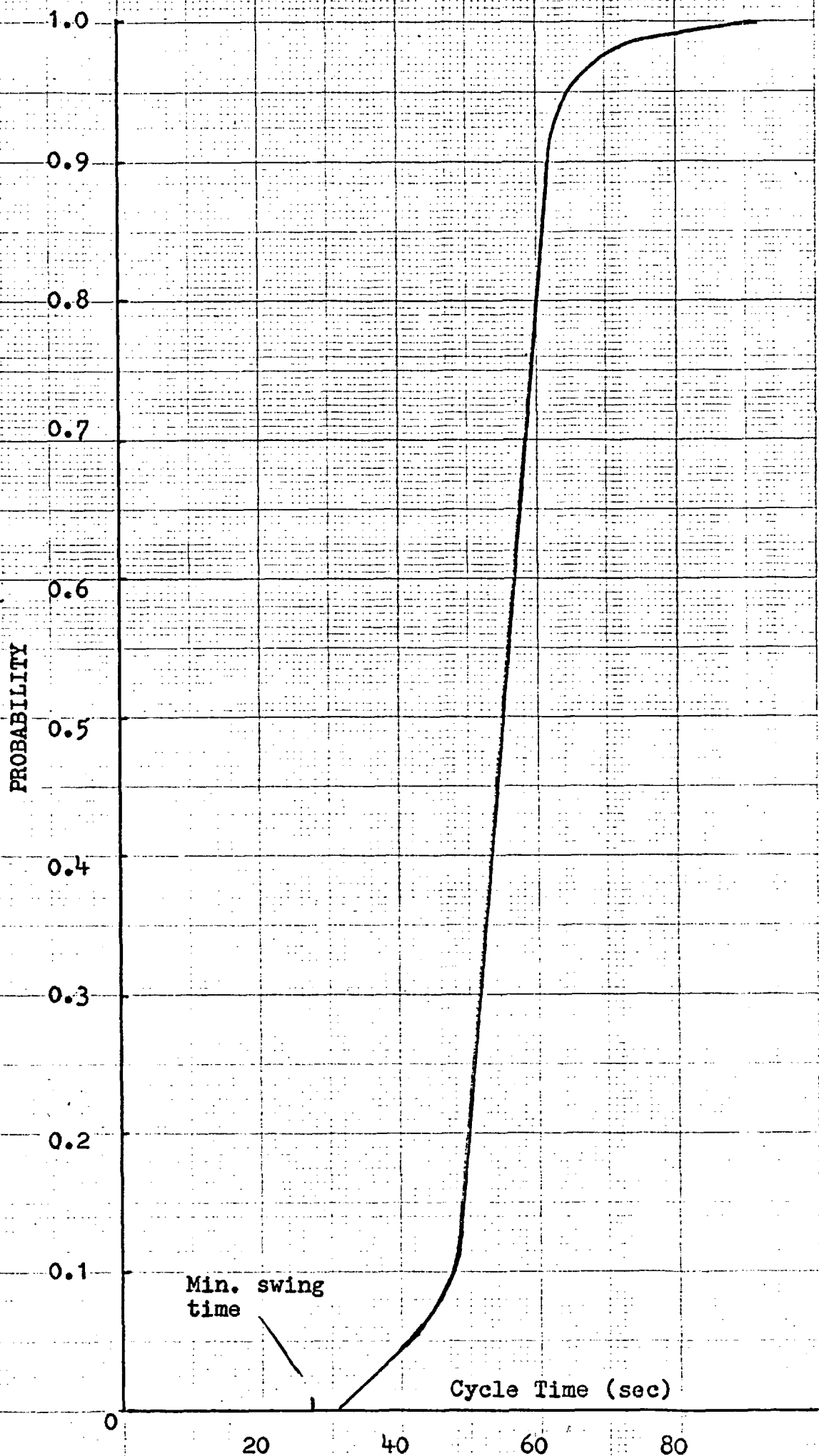


FIG 5A2 CUMULATIVE PROBABILITY - DRAGLINE CYCLE TIME

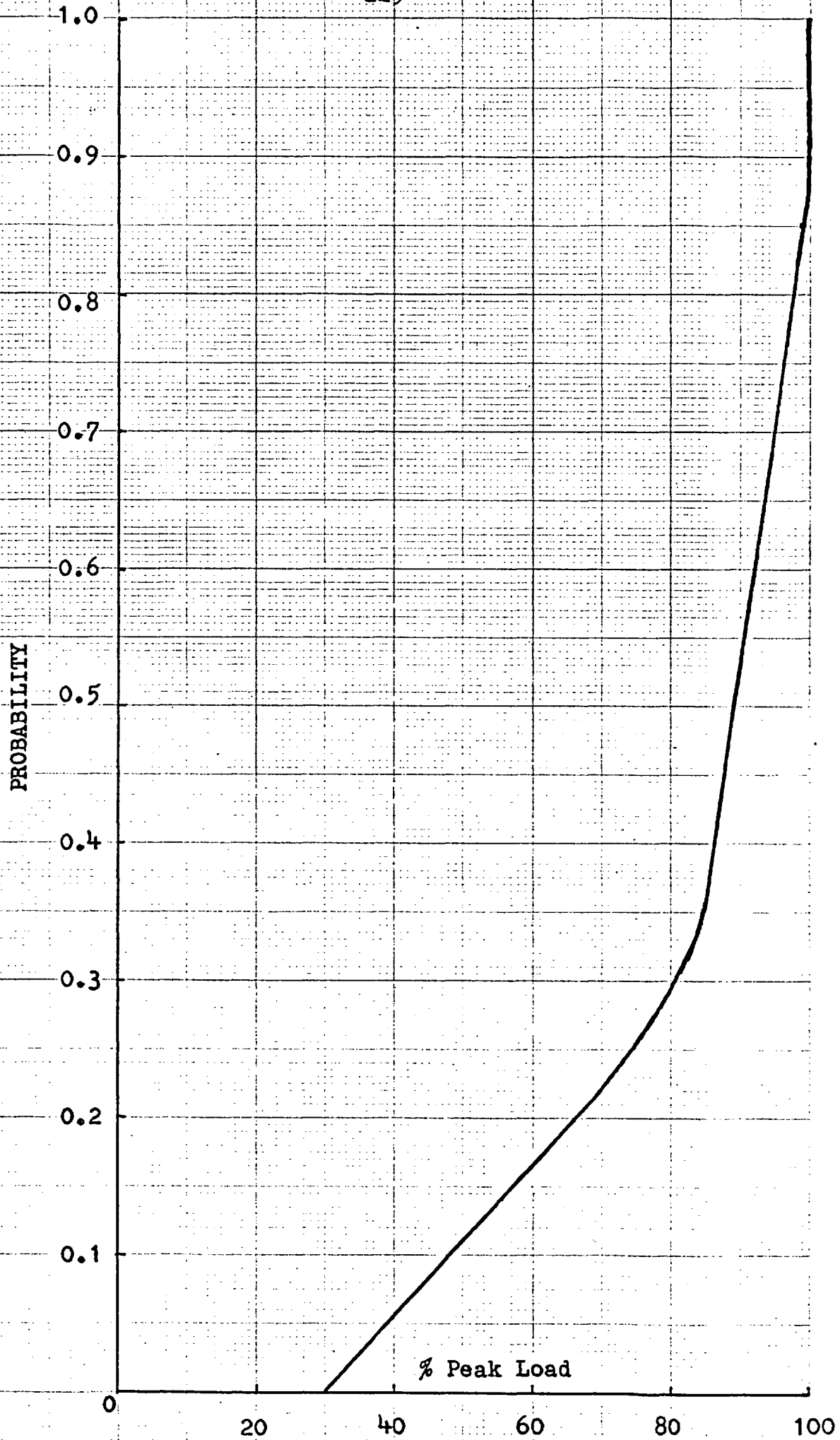


FIG 5A3 CUMULATIVE PROBABILITY - % DRAGLINE PEAK LOAD

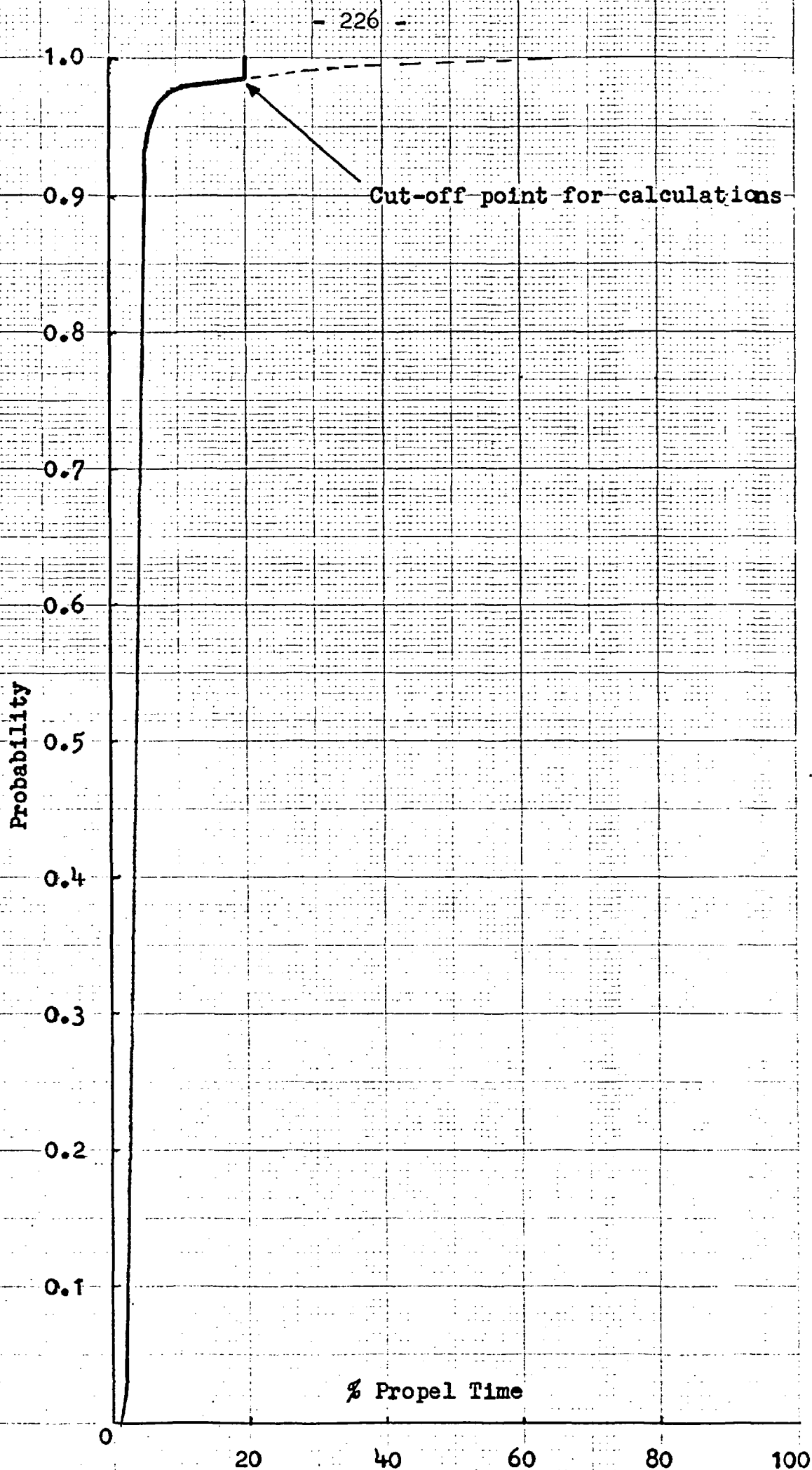


FIG 5A4 CUMULATIVE PROBABILITY - % DRAGLINE PROPEL TIME

1. Cumulative probability curves of cycle time and cycle amplitude were prepared (Figs 5A2 and 5A3). It was assumed that the regenerative peak for lowering the hoist motion (empty bucket) would remain constant.
2. The starting point in each dragline cycle was determined from a random number. The remaining time in each cycle was calculated.
3. The simulation time clock was started at ZERO.
4. The cycle times for draglines A and B were determined from the cumulative distribution curve (Fig 5A2) using two random numbers.
5. The cycle amplitudes for draglines A and B were determined from the cumulative curve (Fig 5A3) using two random numbers.
6. The power demand cycle for draglines A and B was calculated from 4 and 5.
7. The power demand curve was summated from 6.
8. Steps 4 to 7 were repeated for approximately 120 cycles.

It will be noted that the ~~cummulative~~ cycle time curve included a portion (6%) for propel time to cater for dragline manoeuvring. When this occurred a period of light load, 15% motoring was assumed, this being based on actual machine operational data. The length of propel time was also determined from a ~~cummulative~~ probability curve (Fig 5A4).

No allowance was made for down-time for repairs, etc. This could easily have been catered for by extending the propel time curve and using zero load for all times above 360 seconds (an approximate limit for manoeuvring) but the

times when the draglines operated in this simulated manner represent the most severe conditions. As the load demands of the two machines working together are the subject of the investigation it was decided not to include down-time for repairs.

A sample of the simulated power demand curve is shown in Fig 5A5. This appeared to be a considerable improvement on the peaky cycle of a single dragline and the indications were obviously that a greater improvement would result if a third dragline should be added. At this stage however it was decided to abandon the dragline scheme in favour of bucket wheel excavators and because of the tedious volume of calculation involved the exercise was not completed for the third dragline. It was hoped that it would be possible to check the findings by field measurements at a later date but a comparable situation never became available to the author. Engineers of the Westinghouse Electrical Corporation of the U.S.A. were able to do this in 1968, Fig 5A6 shows a typical section of the power trace obtained. This is very similar to the power demand curve computed by the author.

The problem of application of single large draglines on small power systems was again considered by the author in 1967 and 1968 for applications in North Africa and Mexico. Both power systems were particularly weak and problems could be expected to occur. A survey of the literature at that time indicated that apart from some work in Australia, (Waldie, R.D., Beaton K.A. and Booth R.R. "Investigation of Supply to an unusually Large Dragline from a Small Power System". Elect. Eng. Trans, Inst. Eng. Aus., Mar. 1968) little experience had been gained and further investigations were required. The Australian investigations were mainly

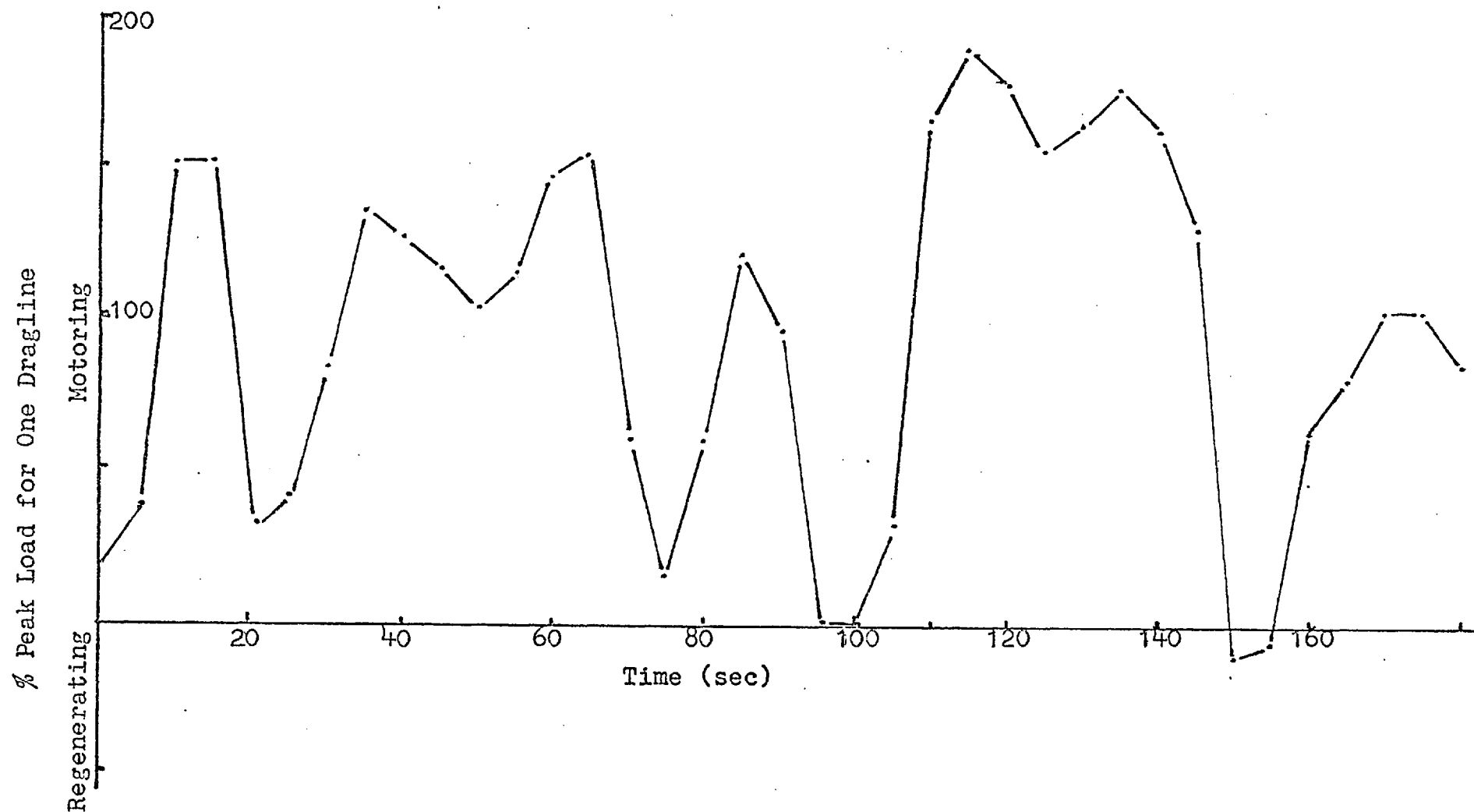


FIG 5A5 POWER TRACE OF TWO DRAGLINES CONNECTED TO THE SAME POWER SOURCE
Prepared by manual simulation

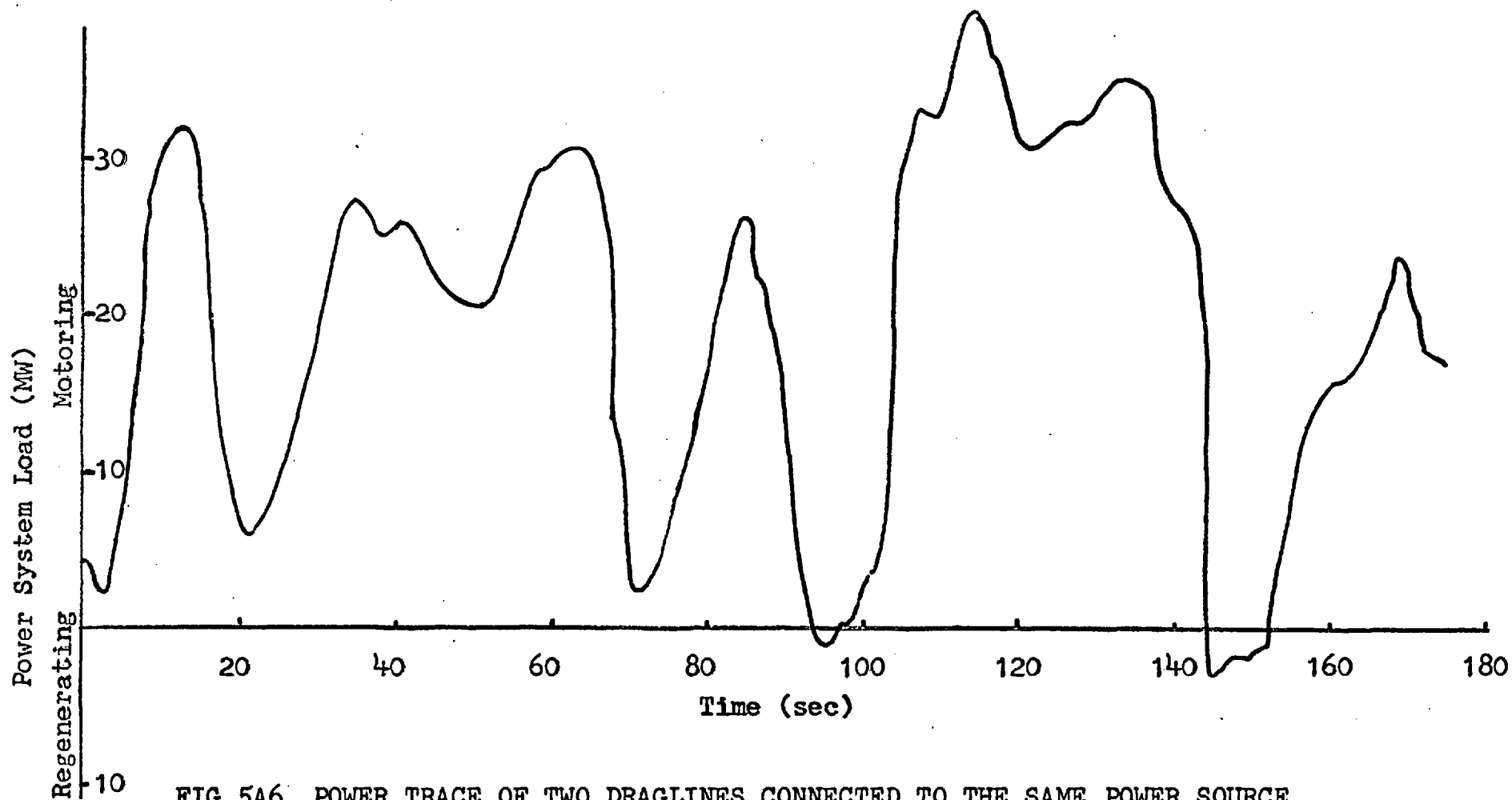


FIG 5A6 POWER TRACE OF TWO DRAGLINES CONNECTED TO THE SAME POWER SOURCE
Plotted from Westinghouse chart - Instrumentation - Westinghouse
Class A Recording (kW) Power Meter.

concerned with the use of an ac network analyser for assessing the effects of a single dragline excavator on a relatively small power system and the results of field measurements. Their findings may be summarised as follows:

1. The generating capacity on the line must be greater than the peak demand of the excavator, particularly as regards boiler capacity.
2. Adequate transformer and feeder capacity is required.
3. The dragline m-g set, synchronous driving motors should be fitted with high-speed, stepless, automatic regulators to minimise voltage fluctuations. This may require an increase in the size of the synchronous motors to be provide additional kVAR capacity.
4. The peak of the dragline load can be limited, both in amplitude and rate of rise. This increases the excavator cycle time and should be avoided if possible but may be essential at times of light system load, e.g. at night.
5. The use of on-load tap changer transformers at the mine.
6. The regenerative peak can be reduced by the connection of a resistive load to the m-g set driving motor terminals at appropriate times during the cycle.

The power system was "stiffer" than either considered by the author and hence would be more stable. In neither of the cases to be considered were network analysers or any form of computing facility available. A simplified approach using

the Per Unit system covering the power system calculations is suggested by the author in Appendix 5C. This simple approach though approximate indicates whether more sophisticated analysis is required.

The first and most obvious approach to the problem of large stripping machines on weak power systems was to use some form of peak load compensator such as an energy-storing flywheel coupled to the generator shaft. This is a simple well known method of reducing the load peaks of mine winder drives and the use of an induction motor drive with a slip regulator was investigated. Unfortunately a dragline duty is much more severe than that of a mine winder and the lack of ability to control VAR's could result in voltage instability. A synchronous condenser was specified to overcome this problem but this was finally ruled out because:

- a) greatly increased capital cost
- b) additional maintenance requirements
- c) difficulties experienced by the manufacturers in accommodating the additional machine on the machinery deck of the dragline.

Discussions with electrical equipment manufacturers resulted in the Westinghouse Electrical Corporation suggesting the use of the "Cycloconvertor", a device previously used for variable, low-speed mill drives, in a peak load compensating arrangement.

Flywheel Peak Load Compensator

To use a flywheel for energy storage it is necessary to control the speed of the generator set, so that energy is stored and released into and out of the flywheel. Several techniques have been used in the past for speed control of

the rotating parts of such drives, almost all of which involves items of electrical rotating machinery in addition to the main flywheel set. All these systems are more complicated than the induction-motor plus slip regulator previously mentioned. Generally these systems supply variable-frequency voltage to the slip rings of the wound-rotor machine (Fig 5.A7) and so force the angular velocity of the rotor to change. Such systems generally are limited to sub-synchronous speeds.

The stator of the motor produces a synchronously rotating magnetic field at supply frequency, while the rotor, excited from a variable-frequency voltage supply through its slip rings, establishes a rotating magnetic field in the rotor that rotates in synchronism with the variable frequency supply. As the rotor is free to rotate mechanically it will move in such a direction as to lock the rotor and stator fields together, i.e.

$$\omega_s = \omega_r + \omega \quad \dots\dots\dots (5A1)$$

where

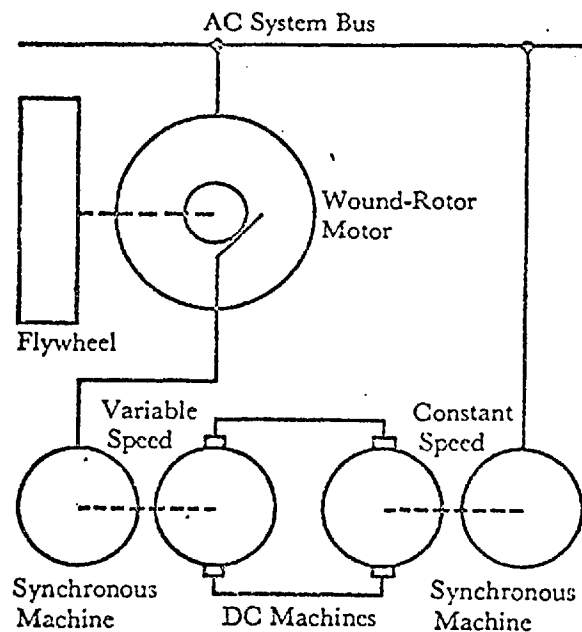
ω_s = angular velocity of stator field
with respect to stator

ω_r = angular velocity of rotor field
with respect to rotor

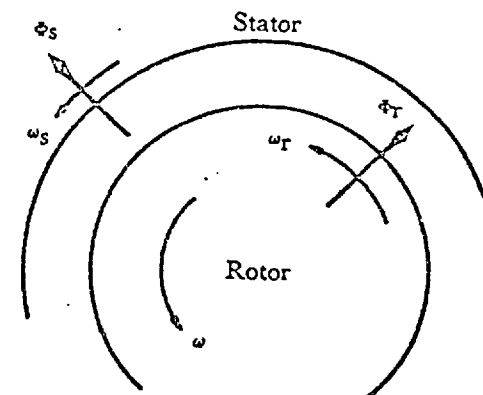
ω = angular mechanical velocity of rotor.

With this condition satisfied the wound-rotor motor operates as a synchronous machine.

By advancing or retarding the phase of the rotor supply, the rotor field can be made to advance or retard slightly from its normal steady state condition. The resulting interaction with the stator field causes an accelerating or



a) Rotating Machines used for Variable Frequency Supply



b) Synchronously Rotating Fields and Mechanical Rotation of rotor fed Slip Ring Induction Motor

FIG 5A7 SYNCHRONOUS OPERATION OF SLIP RING INDUCTION MOTOR - ROTOR FED WITH VARIABLE FREQUENCY SUPPLY FROM EXTERNAL SOURCE

decelerating torque to be developed and the rotor to change speed. For simplified analysis power losses can be ignored as they are relatively small. The power flow of the system is then P_g , the power crossing the air gap. This divides into P_m , the mechanical power applied to the rotating masses and the electrical power, P_e , flowing in the rotor circuit. The division of power is related to the slip, s :

and

$$P_g = P_m + P_e \quad \dots\dots\dots (5.2)$$

$$P_m = P_g (1 - s) \quad \dots\dots\dots (5.3)$$

$$P_e = P_g s \quad \dots\dots\dots (5.4)$$

where

$$s = \frac{\omega_s - \omega}{\omega} \quad \dots\dots\dots (5.5)$$

When s approaches zero, i.e. near synchronous speed, a relatively small amount of power flows into the motor circuit compared with that being transferred into the flywheel. If the set could operate through synchronous speed, the rotor electrical power requirement is minimised, while the flywheel size could be reduced since the higher average speed would require less mass to store a given amount of energy.

The cycloconverter system is designed to permit operation of the flywheel motor-generator set through synchronous speed. The cycloconverter provides a source of variable-frequency power for the rotor (Fig 5A8). The three phase cycloconverter can generate low frequency alternating currents of either phase rotation, or direct current. It is then possible for ω_r to be positive, negative or zero, and thus ω can be

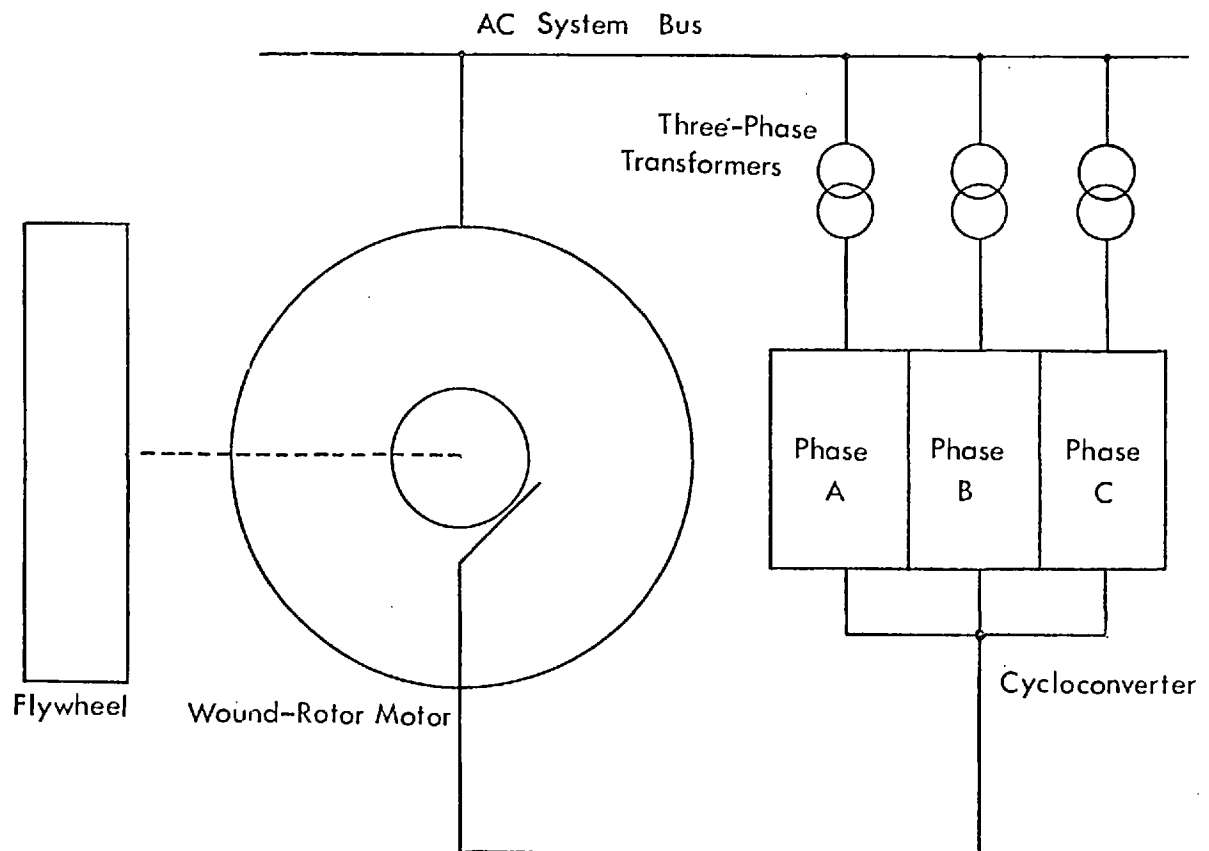


FIG 5A8 THE CYCLOCONVERTER ROTOR FED MACHINE

sub-synchronous, synchronous or super-synchronous.

Power can thus be made to flow from the flywheel during motoring load peaks and into the flywheel during low motoring or regenerative loads by appropriately controlling the cycloconverter output frequency (Fig 5A9). The exact size of the shaded areas of Fig 5A9 is established by the requirement that the net energy flow in and out of the flywheel over the whole cycle must be zero.

For a single machine however the rotating equipment would have to be capable of handling the power flow into the flywheel during the period of maximum generation which is more than twice that during maximum motoring. Economies can be achieved if this power transfer requirement is reduced by dissipating some of the regenerative power in resistors (Fig 5A10) thus reducing the size and cost of the flywheel motor generator set. Where more than one dragline is connected to the system, simultaneous peak regeneration of all machines would occur only infrequently (Figs. 5A5 and 5A6) and the requirement for resistor dissipation of power is small.

For a single machine, it is necessary for flow of energy into and out of the flywheel to be zero and the positive peak power requirements should approximately equal the negative peak requirements to minimise flywheel motor-generator set size (Fig 5A11) The acceptable power swing must be determined from the penalties imposed by the supplier versus the capital and operating costs of the compensating system. Used in conjunction with VAR control, the cycloconverter system should enable an electrical supply system of adequate power rating to supply a single bucket stripping machine with minimum system disturbance.

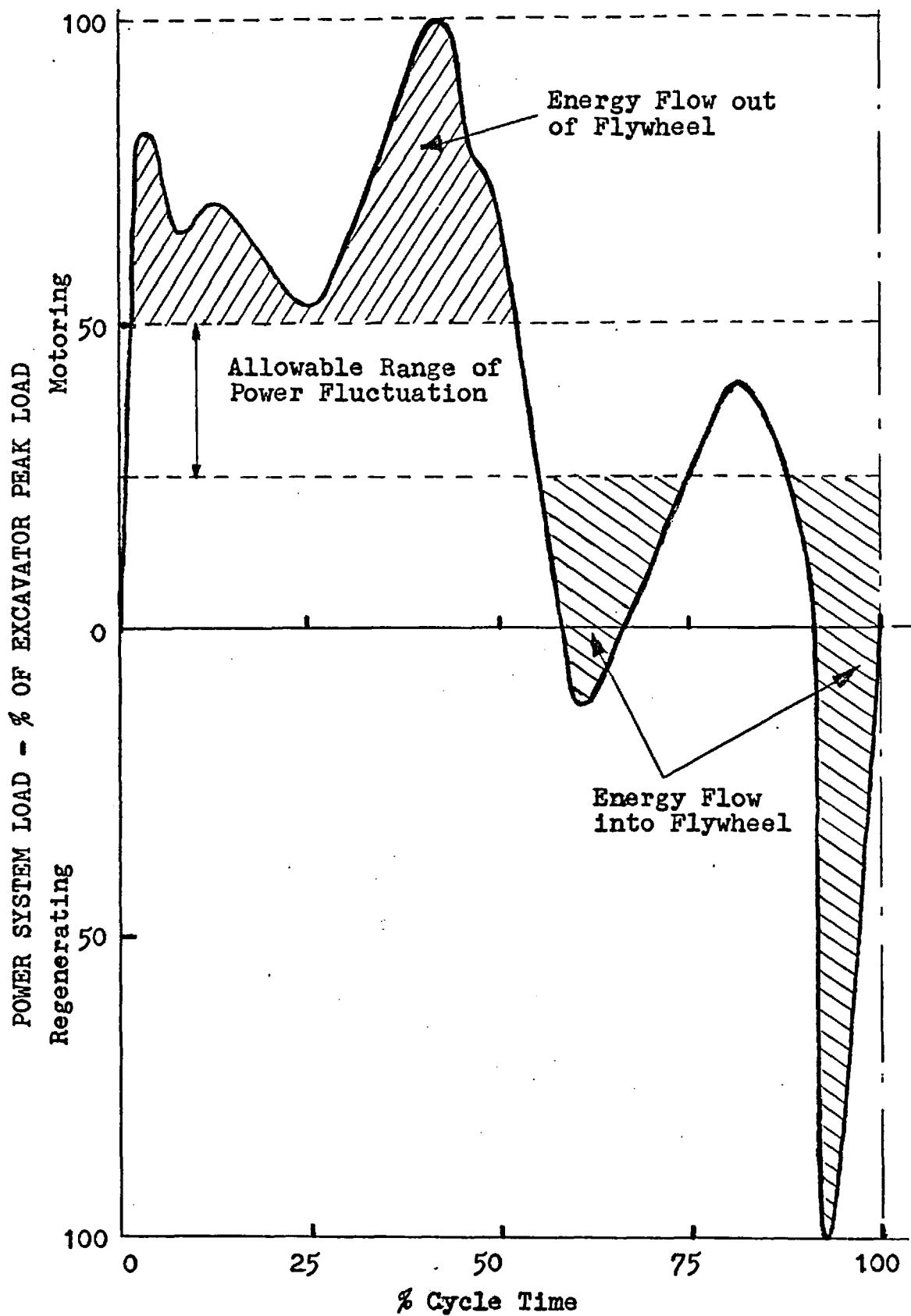


FIG 5A9 PEAK "LOPPING" WITH FLYWHEEL ENERGY COMPENSATION

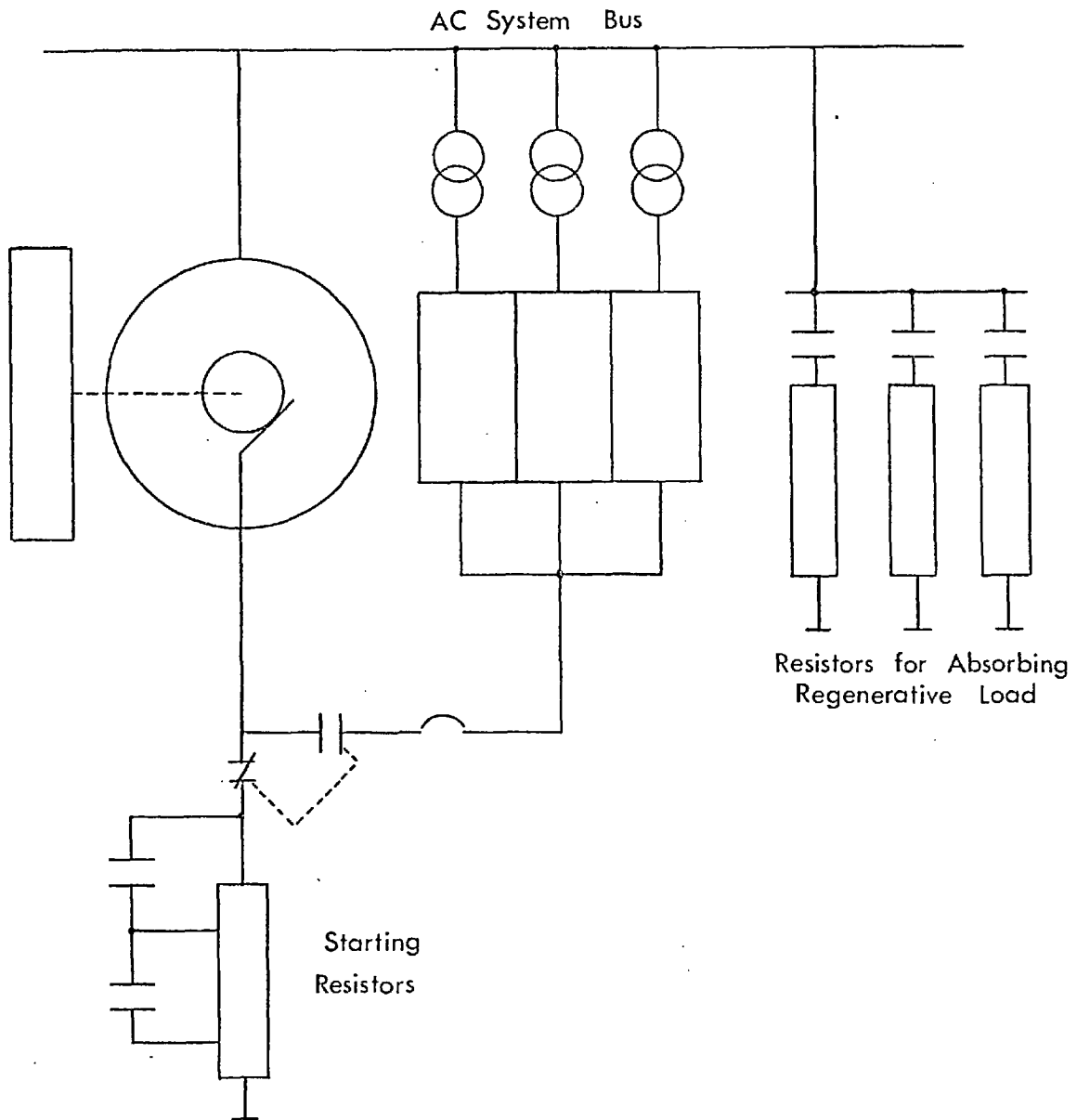


FIG 5A10 RESISTOR DISSIPATION OF REGENERATED POWER

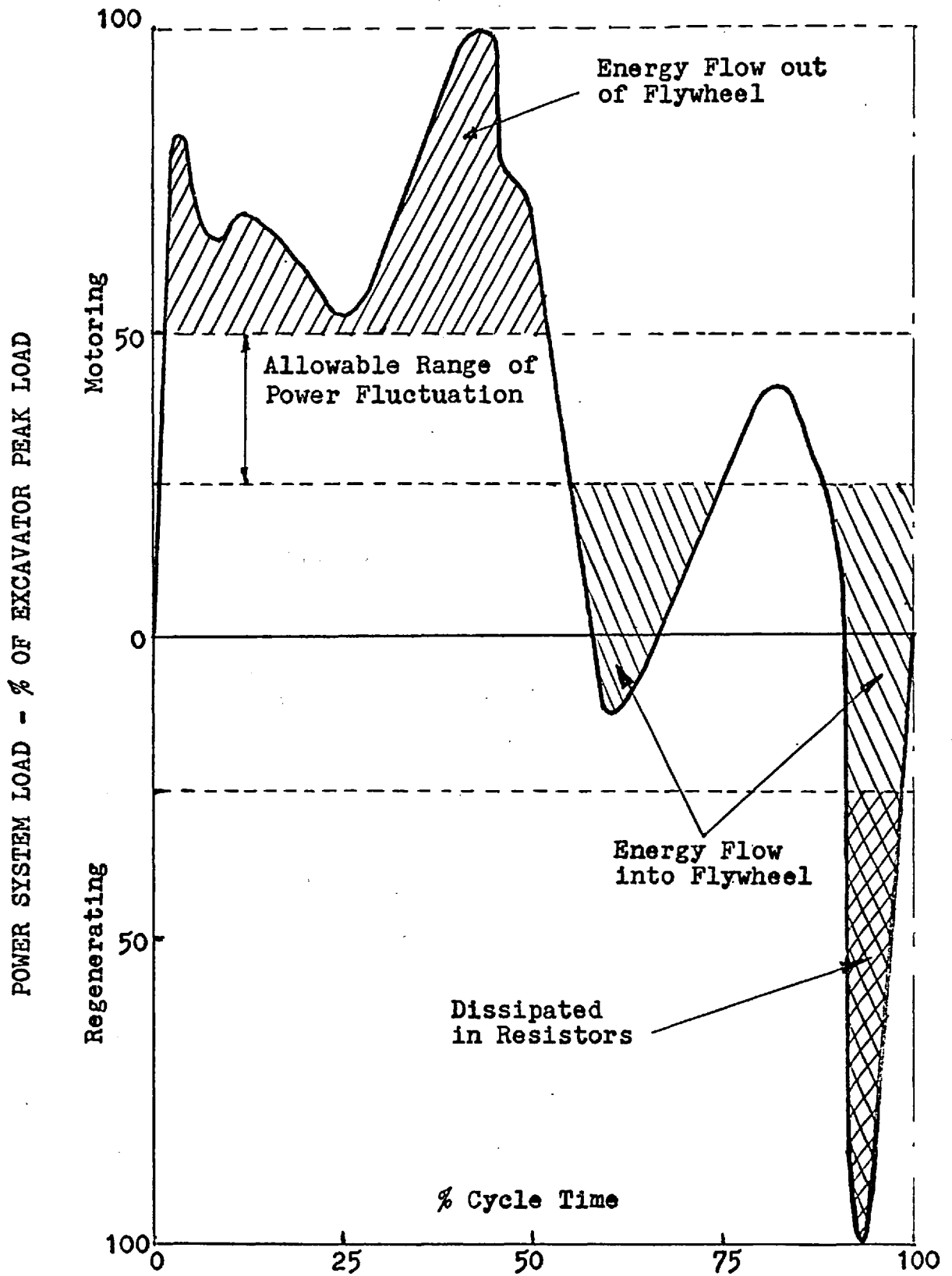


FIG 5A11 REDUCTION OF EQUIPMENT SIZE BY RESISTOR DISSIPATION OF REGENERATED POWER

2 Hz OSCILLATION PHENOMENA

A further problem observed by the author where large stripping machines are connected to weak power systems is that of a 2Hz oscillation of system voltage. At first it was thought that this was caused by wire rope elasticity which produced similar frequency vibrations. This was eliminated and the most obvious cause appeared to be "hunting" of the automatic excitation regulator. This was also eliminated. Further investigations were carried out by General Electric who discovered that the phenomena was not uncommon but in most cases the oscillations go by unnoticed. The phenomena appears generally to occur when

1. Where the total power swing of the excavator from peak motoring load to peak regeneration exceeds 5 - 10% of the installed generating capacity.
2. Where the system impedances from excavator to infinite busbar are such that the ratio of reactance to resistance exceeds 3.5 to 1.
3. High distribution system impedance.
If most of the impedance occurs between the infinite bus and the mine substation further study is indicated.
4. Where adjacent loads are sensitive to flicker, e.g. television transmitters, computer facilities, etc., since a possible

effect of the oscillation is voltage flicker, which if it exceeds 1% (preferably 0.5%) can result in difficulties for users of electronic equipment and irritation to other users.

To overcome these difficulties it is usually possible to reinforce the supply system with only modest expenditure. This requires close co-operation between the electricity authorities, the electrical equipment manufacturer and the mine operator. The possibility of using some form of peak load compensating drive for the excavator m/g set must of course also be investigated.

APPENDIX 5B

DIPPER AND BUCKET CONTROL

A further problem which became apparent to the author during these investigations was the lack of trained operators, and the facility for training them, in the developing countries. Where production is largely dependent on keeping a single stripping machine in constant operation at maximum output it is essential that:

- a) the bucket or dipper must be filled as quickly as possible, and
- b) bank stalls due to inexperienced operation must be avoided.

The author also considered that these inexperienced operators would lack the motivation to keep their machines at optimum output since they lack the traditional background of U.S.A. operators. A further complication would be bad fragmentation, since blasting control could not be expected to be so effective, and bank stalls resulting in m-g set motor pullout, plus severe mechanical and electrical shock loading could be expected to reduce production.

After discussions with many mining companies and the major excavator manufacturers the author considered that some form of dipper or bucket control was essential but none of the electrical equipment manufacturers were able to offer such a control system.

The author considered that a much more basic knowledge of excavator operation was needed and in a paper (see Ref 51) stated:

"Fundamental investigations into the actions and performance of large excavating machinery to provide a more basic analysis appear necessary. One of the problems that can be envisaged with such investigations is to determine what may be described as "typical" or "average" operations, since these are dependent on digging conditions, the swing angle of the machine, the dumping height and the geometry of the excavation. The skill of the operator also has some significance. Any investigations should therefore be planned on a statistical basis to obtain meaningful results".

General Electric expressed interest and using test facilities provided by Peabody Coal Company at Elm Mine, Trivoli, Illinois, a series of tests was conducted involving a skilled operator using manual and automatic control. Tests also involving a partially skilled driver would have had great validity but were not possible. The conditions for the tests were:

1. A large number of digging cycles had to be performed.
2. The overburden to be homogeneous.
3. The fragmentation to be uniform.

To ensure 2 and 3 were fulfilled a long test was not possible and it was decided to perform 200 digging cycles as long similar conditions prevailed throughout the test.

The mechanics of the tests were based on the following simplifying assumptions:

Fig 5B1 shows the typical hoist speed ~ pull characteristic for the stripping shovel.

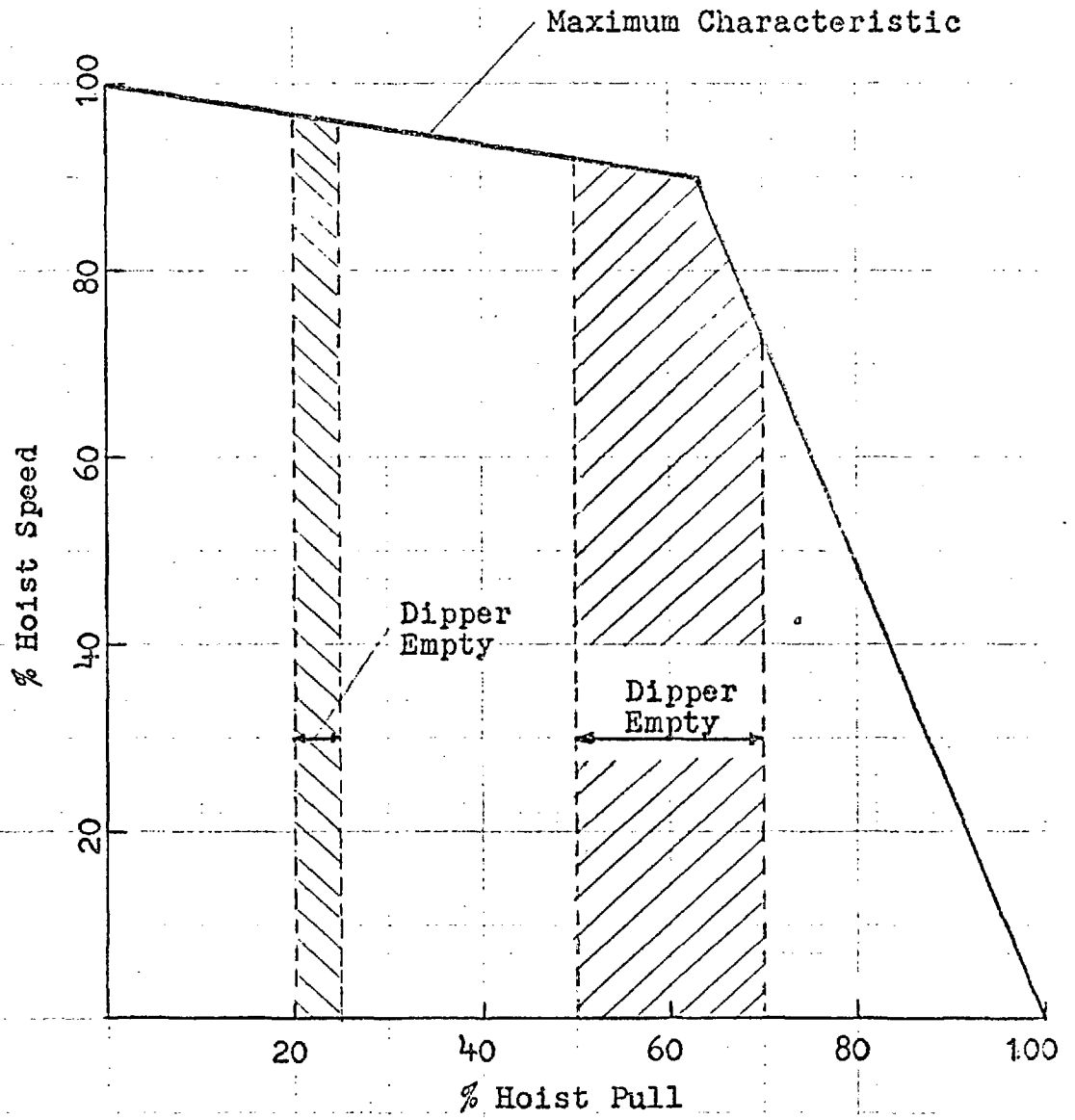


FIG 5B1 HOIST SPEED ~ PULL CHARACTERISTIC

Fig 5B2 shows a simplification of the forces and velocities involved in a dipper being hoisted through overburden, where the crowd horizontal speed and acceleration are zero, and the dipper vertical speed is constant during the interval considered.

The overburden is a homogeneous clay like material.

Summating the vertical forces:

$$H_h = W + A + F_s$$

For a normal shovel hoist operation A is negligible and:

$$H_h = W + F_s$$

where

W = Vertical force due to dipper mass + vertical force due to overburden

The vertical force due to the overburden will vary from zero at the start of digging up to a maximum when the dipper is filled. Penetration of the dipper into the overburden will not be uniform since crowding and hoisting must take place simultaneously at the start of digging, but if the average figures are used the error is negligible.

The tests also showed that the hoist rope pull forces were very sensitive to bank penetration and confirmed that F_s was a function of dipper penetration.

The following difficulties are experienced by the operator when controlling dipper penetration:

1. Lighting conditions
2. Varying geometry of dipper handle and crowd machinery.
3. Degree of fragmentation.
4. Colour contrast.

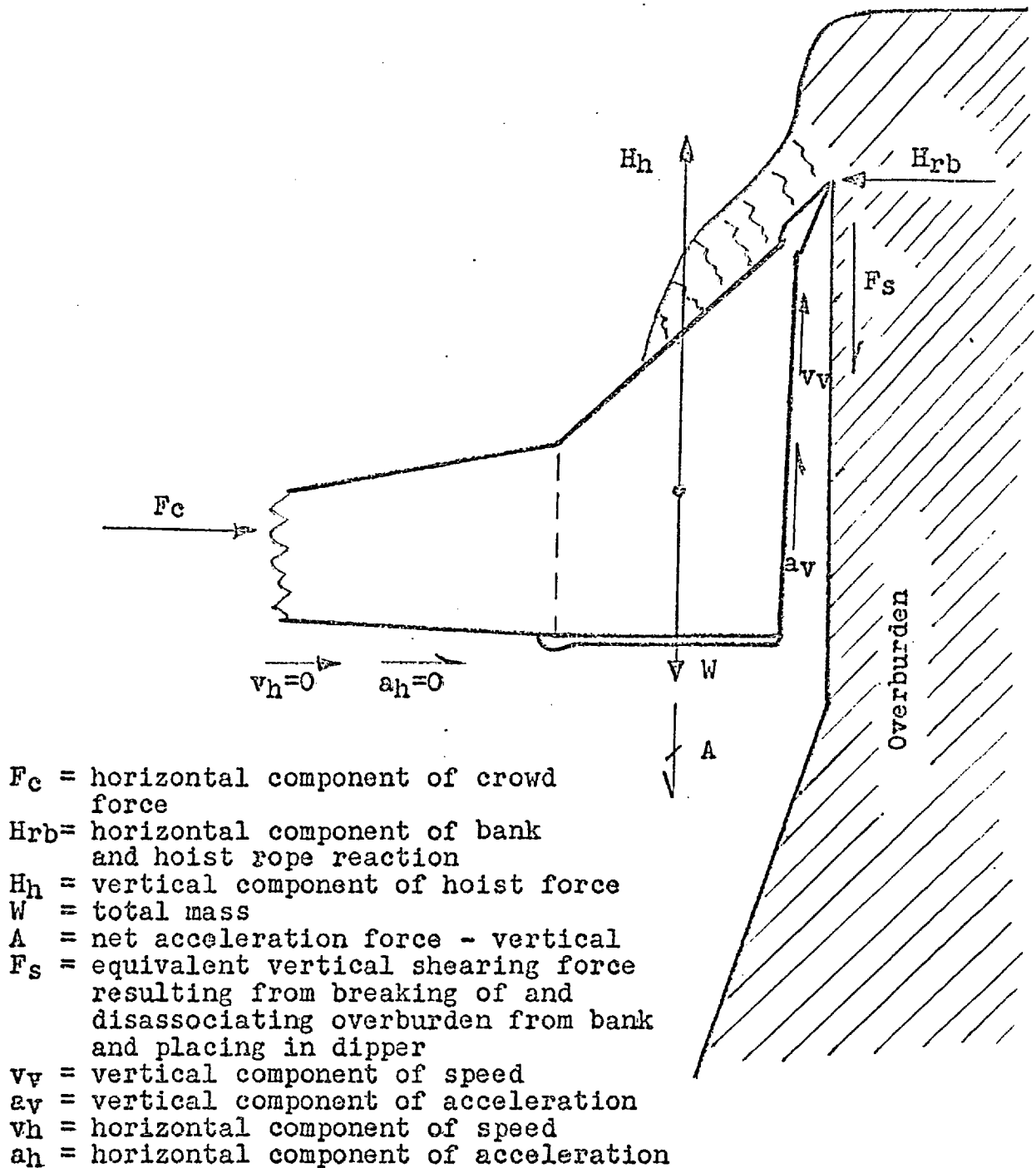


FIG 5B2 DIPPER SHEARING ACTION IN HOMOGENEOUS GROUND

5. Position of operator's cabin.
6. Shovel accelerations on all three axes.
7. Noise.
8. Operator's physical condition.

An experienced operator can generally judge penetration within 100mm (4in) and at the beginning of the shift perhaps within 50mm (2in).

The tests, the thinking behind them and the results are briefly reported by Neslin, M.A. and Smith, J.H. "Automatic Dipper Control for Stripping Shovels". Coal Age, Nov. 1971, but for completeness additional information is provided here.

The tests were carried out using a first class operator over 305 digging cycles, 173 with automatic and 132 with manual control, this being a large enough sample for statistical analysis. It was also important that the number of cycles be limited so that no serious change occurred in the test conditions. In conducting the tests a six channel, direct-writing oscillograph recorder was used. The channels recorded:

- a) hoist motor volts (hoist speed)
- b) hoist motor current (hoist pull)
- c) crowd motor volts (crowd speed)
- d) crowd motor current (crowd pull)
- e) crowd master switch volts, and
- f) a voltage to illustrate when the shovel operated in MANUAL, SEMI-AUTOMATIC and AUTOMATIC modes.

Because of variation in overburden quality the three modes of shovel digging function were adopted. SEMI-AUTOMATIC controls a constant hoist pull established by the position of the crowd master switch and AUTOMATIC controls a varying

hoist pull with respect to time, increasing from an initial value of 60% of stall pull to a final value of 80% as the dipper becomes completely filled.

There are some constraints on automatic operation, these also usually being constraints on manual operation.

1. Bad fragmentation. Automatic operation does not improve this situation, but does in fact provide a very reliable indication of whether or not fragmentation is adequate.
2. Low face height. Automatic control needs at least 3 seconds in the bank to ensure a full dipper load.
3. Hard bands particularly just above the top of the mineral.
4. The presence of large isolated boulders.
5. Long clean up times.

All these constraints reduce the time that can be spent in automatic operation.

The practical actions required need some explanation. With the selector switch in one of the automatic modes, the operator positions the dipper at low speed, with the teeth about to enter the bank. Manual control is in effect. The master switch handles are then moved to the full hoist position and appropriate crowd "out" position. Automatic regulation occurs as the dipper penetration loads the hoist drive. Thereafter the crowd moves smoothly to satisfy the automatic control. In the meantime the operator may remove his hands from the controllers, while the regulators fill the bucket automatically. When the operator observes the dipper is full, the movement of either master switch restores

normal manual control for spoiling.

The simple concept of regulating crowd motion penetration as a function of hoist pull required extensive system design work. Gains had to be established for individual motions so that automatic feedback gain could be established. This was achieved by electrical measurement and by discussions with stripping shovel operators. Analysis showed that the correct gain in the automatic feedback would be for the crowd motion to respond from 100% "crowd out" to 100% "crowd in" for a 10% increase in the hoist pull being regulated.

The transfer functions and parameters established by test and analysis were used to build an analogue computer model of the entire hoist and crowd mechanical and electrical systems, based on operational amplifiers. It was necessary to include the constraints imposed by shovel geometry e.g. automatic operation could not be permitted where crowd "out" could lift ("jack") the boom. Additionally the regulator had to observe those changes in the system which occur when the crowd movement approaches either end of the crowd handle.

The object of the test was to compare production rates rather than times to fill the dipper. Measurement of the latter was impractical because:

- a) variation in the dipper filling,
- b) determination of the start and finish of dipper filling.

The dipper weight and overall cycle time are discrete easily identified parameters for calculating production. It was decided to regard any cycle as "automatic" where the regulators took over for a period of one second. The weight of the dipper was determined from the hoist motor armature

current when the swing motor armature voltage was at full speed and the hoist motor voltage were at zero. The recorder chart reading was expressed as an index rather than an actual weight.

Histograms of cycle time against frequency of occurrence showed a range of 24 sec to 72 seconds and indicated that the range of the automatic cycle times was slightly less than the manual cycle times.

Cumulative probability plots of cycle time were similar in shape for both modes of operation and also close together. The slope of the plot for automatic operation was slightly less than for manual indicating that cycle times were more consistent when operating automatically. The mean cycle times were:

Automatic	50.04 sec
Manual	50.18 sec

This is not statistically significant. The standard deviations were:

Automatic	8.01 sec
Manual	9.19 sec

again indicating that cycle times were more consistent in automatic mode than manual mode.

Histograms and cumulative probability plots were drawn for net weight per cycle against frequency of occurrence. These again indicated that the operator's performance was similar for both modes of operation, except that the net weight in automatic operation was consistently above that of manual operation. The means were:

<u>Operation</u>	<u>Index</u>
Automatic	7.07
Manual	6.56

The standard deviations were:

Automatic 1.77

Manual 1.88

again indicating greater consistency in automatic operation.

Histograms of production rate against frequency of occurrence (Figs 5.B3 and 5B4) and cumulative probability Fig 5.B5 were drawn. Again the operator's performance is similar but automatic operation is consistently more productive than manual operation. Statistical analysis showed that for 95% confidence automatic operation provides higher production than manual. The mean productivity indexes were:

Automatic 5.675

Manual 5.225

$$\frac{\text{Automatic}}{\text{Manual}} = \frac{5.675}{5.225} = 1.086$$

i.e. automatic production was 8.6% greater than manual production during the tests.

The conclusions derived from these tests are:

1. A 7.8% higher net average weight per dipper was obtained by an experienced operator in a statistically designed test.
2. The average cycle time using an experienced operator was practically the same for both modes of operation.
3. A statistically significant, higher production rate, i.e. 8.6% was obtained using automatic control compared to manual control

(Based on 173 digging cycles)

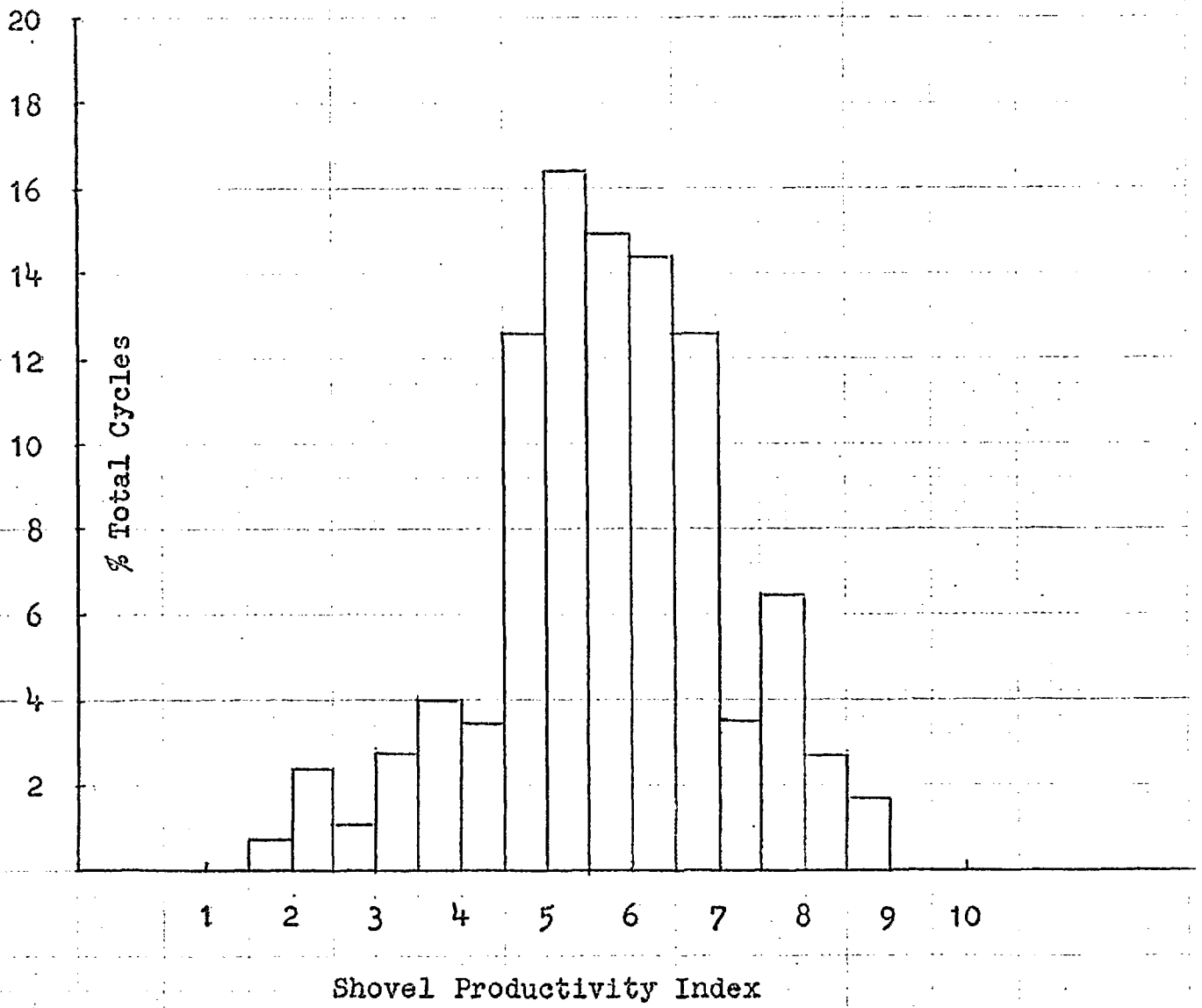


FIG 5B3 HISTOGRAM OF SHOVEL PRODUCTIVITY INDEX
- AUTOMATIC OPERATION -

(Based on 132 digging cycles)

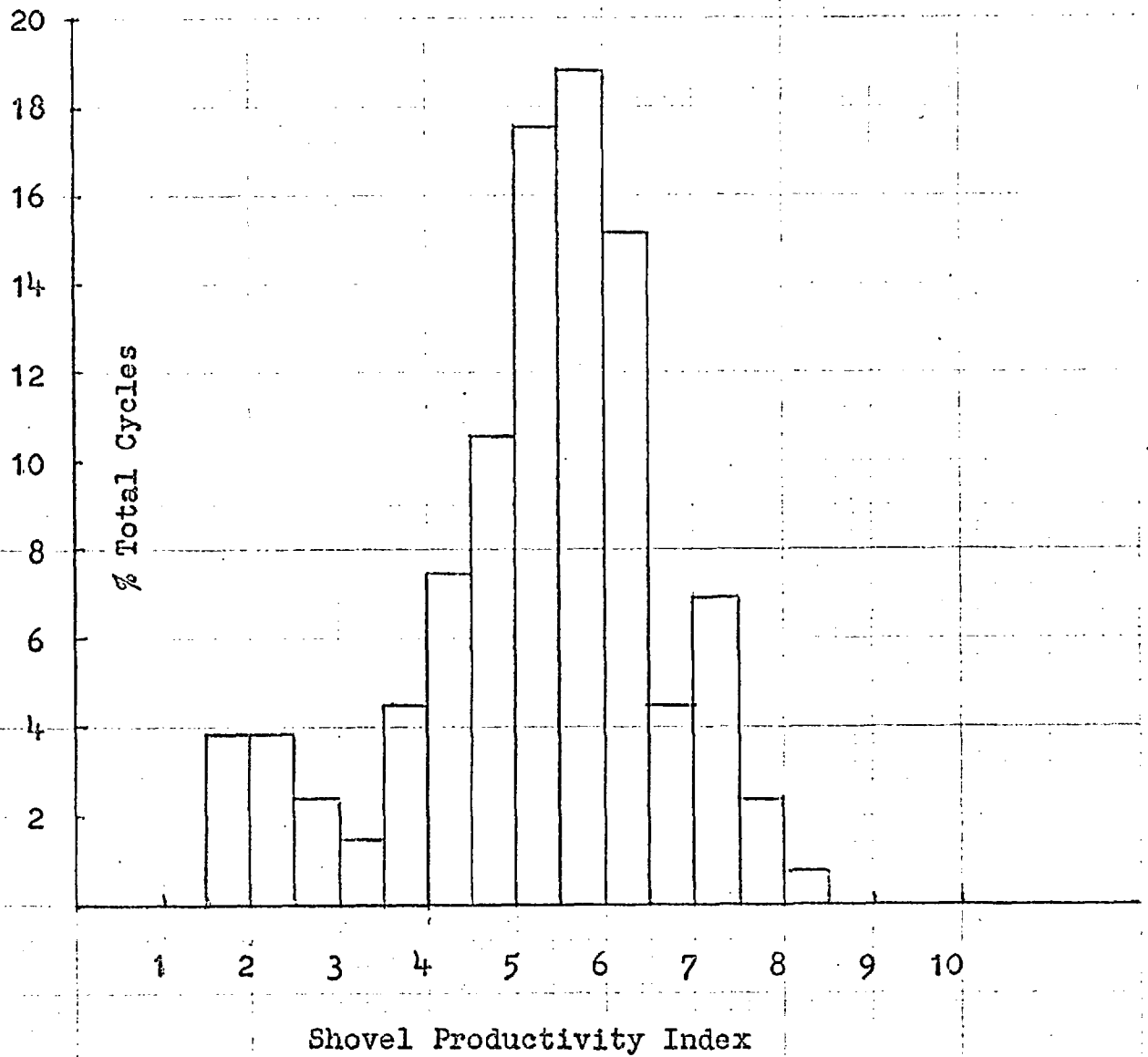


FIG 5B4 HISTOGRAM OF SHOVEL PRODUCTIVITY INDEX
- MANUAL OPERATION -

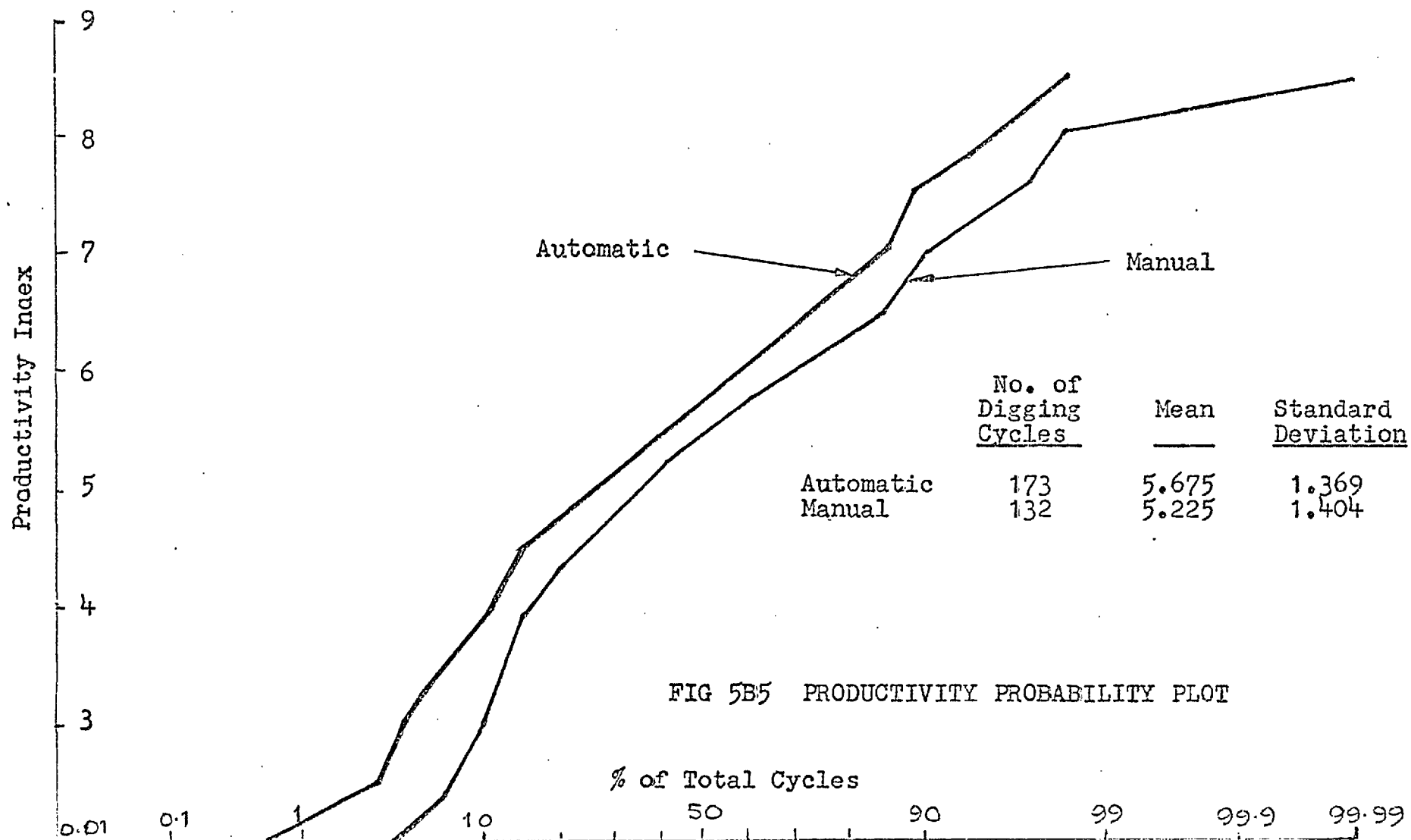


FIG 5B5 PRODUCTIVITY PROBABILITY PLOT

4. The hoist motion cannot be stalled by an operation when in the automatic mode.

The tests were carried out in shallow (less than optimum thickness) overburden over a limited period. Further long term tests are now indicated in deeper overburden with operators of varying experience to indicate or confirm:

- a) more consistent and therefore more predictable production by using automatic control,
- b) achievement of reduced cycle times as the operator applies his attention almost solely to determining when the dipper is full,
- c) that spoiling will be improved because the operator has more time to consider his next action,
- d) less operator fatigue and therefore improved manual control when required,
- e) improved blasting, as automatic operation gives an almost quantitative measure of "diggability",
- f) evaluation of dipper tooth sharpness and dipper pitch for the same reasons as e) above, and
- g) reduced maintenance due to reduced mechanical shock.

The tests and their results seem to justify the statistical approach and give additional emphasis to the simulation of stripping machine systems by the use of analogue computers.

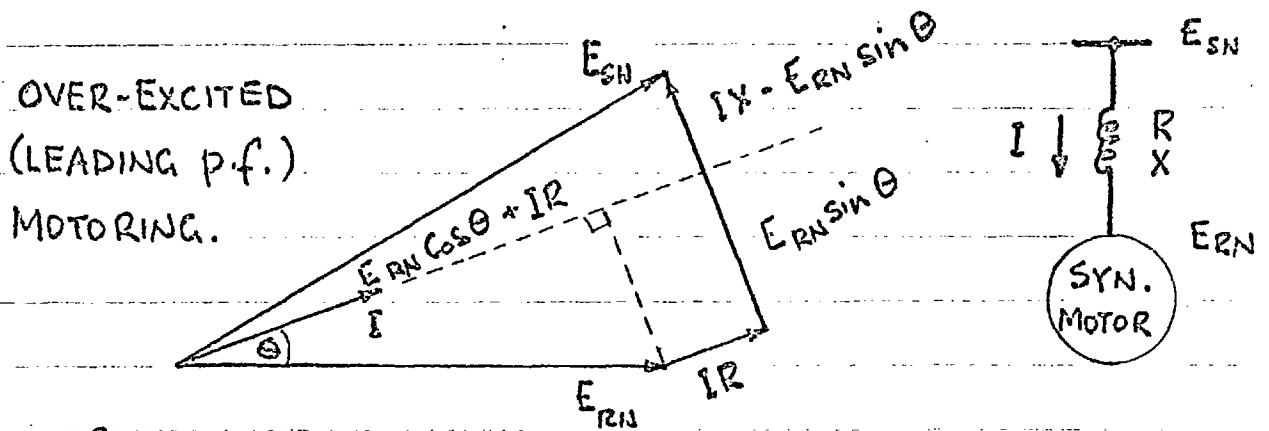
APPENDIX 5C

VOLTAGE CALCULATIONS FOR LARGE STRIPPING MACHINES

CONNECTED TO WEAK POWER SYSTEMS (A simplified method).

ASSUMPTIONS

- a) The stripping machine m-g set is driven by a synchronous motor.
- b) Power factor control can be effected by varying the excitation of the driving motor field.
- c) A high speed stepless field regulator is used for automatic power control
- d) Any phasor voltage diagram where IMPEDANCE values are LINE-TO-NEUTRAL is expressed in LINE-TO-NEUTRAL VOLTAGES.



$$E_{SN}^2 = E_{RN}^2 (\cos^2 \theta + \sin^2 \theta) + 2(R E_{RN} I \cos \theta - X E_{RN} I \sin \theta) + I^2 (R^2 + X^2) \quad (a)$$

Since by Pythagorus

$$E_{Sn}^2 = (E_{RN} \cos \theta + IR)^2 + (-E_{RN} \sin \theta + IX)^2$$

where

E_{RN} = Receiving Voltage L - N

E_{SN} = Sending Voltage L-N

θ : Load power angle

R = Resistance L-N

X = Reactance L-N

$$RW_{\phi} = \text{LOAD power/phase}$$

11/10/21. Last working night before...

Now $\cos^2 \theta + \sin^2 \theta = 1$

$$E_{SN} = \frac{E_S}{\sqrt{3}}, \quad E_{SN}^2 = \frac{E_S^2}{3} \quad - (b)$$

$$E_{RN} = \frac{E_R}{\sqrt{3}}, \quad E_{RN}^2 = \frac{E_R^2}{3} \quad - (c)$$

$$E_{RN} I \cos \theta = \text{Watts/phase} = 1000 \overline{\text{kW}} \phi \quad - (d)$$

$$E_{RN} I \cos \theta = \frac{1000 \overline{\text{kW}}}{3} \quad \text{from} \quad - (e)$$

$$1000 \overline{\text{kW}} = \sqrt{3} E_R I \cos \theta$$

$$E_R = \sqrt{3} E_{RN} \therefore 1000 \overline{\text{kW}} = 3 E_{RN} I \cos \theta$$

$$\text{also } E_{RN} I \sin \theta = \frac{1000 \overline{\text{kVAR}}}{3} \quad - (f)$$

$$I^2 = \frac{(1000)^2 \overline{\text{kW}}^2}{3 E_R^2 \cos^2 \theta} \quad - (g)$$

Combine (b), (c), (d), (e), (f) and (g) with (a)

$$\frac{E_S^2}{3} = \frac{E_R^2}{3} (1) + 2 \left(\frac{1000 \overline{\text{kW}} R}{3} - \frac{1000 \overline{\text{kVAR}} X}{3} \right) + \frac{(1000)^2 (\overline{\text{kW}})^2 (R^2 + X^2)}{3 E_R^2 \cos^2 \theta}$$

$$E_S^2 = E_R^2 + 2000(R \cdot \overline{\text{kW}} - X \cdot \overline{\text{kVAR}}) + \frac{(1000)^2 (\overline{\text{kW}})^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta} \quad - (h)$$

Where all the VOLTAGES are used LINE-TO-LINE
and kW, kVAR are 3 phase quantities and
R and X are LINE-TO-NEUTRAL imp.

E_R : Receiving Voltage L-L

E_S : Sending Voltage L-L

$\overline{KW}$: Load Power, 3 phase

$\overline{KVAR}$: Load Reactive Flow, 3 phase

To convert to PER UNIT values, divide by base quantities.

$$\text{BASE VOLTS} : V_B = (\text{Base ohms})(\text{Base Amperes})(\sqrt{3}) \quad - (i)$$

$$V_B = (1000)(kV_B) \quad - (j)$$

$$V_B^2 = (V_B)(\text{Ohms}_B)(I_B)(\sqrt{3})$$

$$= 1000(kV_B)(\text{Ohms}_B)(I_B)(\sqrt{3})$$

$$V_B^2 = (\text{Ohms}_B)^2(I_B)^2(3) \quad - (k)$$

Divide (h) by (i), (j), (k) as appropriate

$$\frac{E_S^2}{V_B^2} = \frac{E_R^2}{V_B^2} + \frac{(2000)(R \overline{KW} - X \overline{KVAR})}{(1000)(kV_B)(\text{Ohms}_B)(I_B)(\sqrt{3})} + \frac{(1000^2)(\overline{KW})^2(R^2 + X^2)}{(E_R^2 \cos^2 \theta)(\text{Ohms}_B)^2(I_B^2)(3)}$$

$$0/1 E_S^2 = 0/1 E_R^2 + (2) \left(\frac{R}{\text{Ohms}_B} \frac{\overline{KW}}{0/1} - \frac{X}{\text{Ohms}_B} \frac{\overline{KVAR}}{0/1} \right) + \frac{\left(\frac{\overline{KW}}{\text{Ohms}_B} \right)^2 (R_{0/1}^2 + X_{0/1}^2)}{E_{R_{0/1}} \cos^2 \theta}$$

$$\text{where } \frac{E_S^2}{V_B^2} = 0/1 E_S^2, \quad \frac{E_R^2}{V_B^2} = 0/1 E_R^2$$

$$\frac{R}{\text{Ohms}_B} = 0/1 R, \quad \frac{X}{\text{Ohms}_B} = 0/1 X$$

$$\frac{\overline{kW}}{(kV_B)(I_B)(\sqrt{3})} = \frac{\overline{kW}}{\overline{kVA}_B} = 0/1 \overline{kW}$$

$$\frac{\overline{kVAR}}{(kV_B)(I_B)(\sqrt{3})} = \frac{\overline{kVAR}}{\overline{kVA}_B} = 0/1 \overline{kVAR}$$

$$\frac{(1000)^2 (\overline{kW})^2 (R^2 + X^2)}{(E_R^2) (\cos^2 \theta) (\text{Ohms}_B)^2 (I_B)^2 (3) \frac{(1000 kV_B)^2}{(1000 kV_B)^2}}$$

$$\frac{E_R^2}{(1000 kV_B)^2} = E_R^2 0/1$$

$$\frac{R^2 + X^2}{(\text{Ohms}_B)^2} = R_{0/1}^2 + X_{0/1}^2$$

$$(kV_B)^2 (I_B)^2 (3) = \overline{kVA}_B^2$$

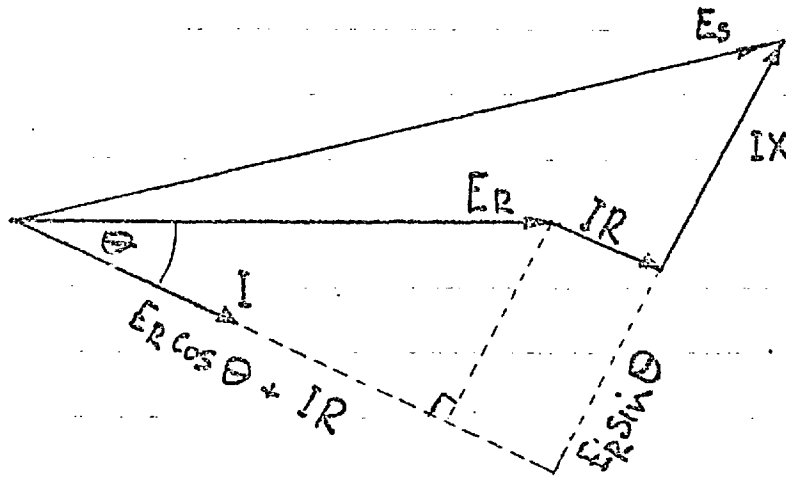
$$\frac{\overline{kW}^2}{(kV_B)^2 (I_B)^2 (3)} = \overline{kW}_{0/1}^2$$

Hence.

$$E_s^2 = E_R^2 + 2(R \overline{kW} + X \overline{kVAR}) + \frac{\overline{kW}^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

in PER UNIT VALUES using 3 PHASE $\overline{kVA}$, $\overline{kW}$, $\overline{kVAR}$,
LINE-TO-LINE VOLTS, LINE-TO-NEUTRAL IMPEDANCES.

UNDER-EXCITED (LAGGING p.f.) - MOTORING

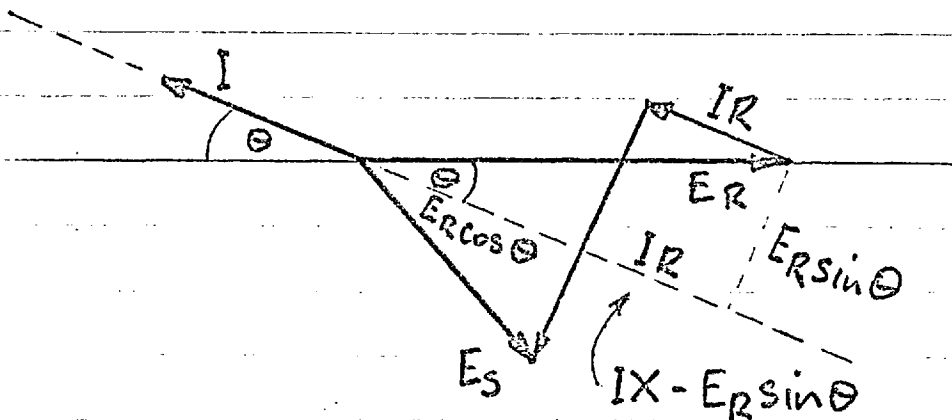


$$E_s^2 = (E_R \cos \theta + IR)^2 + (E_R \sin \theta + IX)^2$$

The only change from the previous case is the sign of " $E_R \sin \theta$ ". This changes the final expression to:

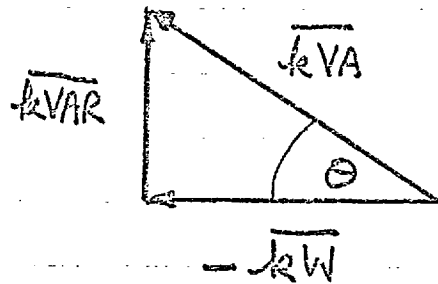
$$E_s^2 = E_R^2 + 2(R \overline{kW} + X \overline{kVAR}) + \frac{\overline{kW}^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

OVER-EXCITED - GENERATING



$$E_s^2 = (E_R \cos \theta - IR)^2 + (-E_R \sin \theta + IX)^2$$

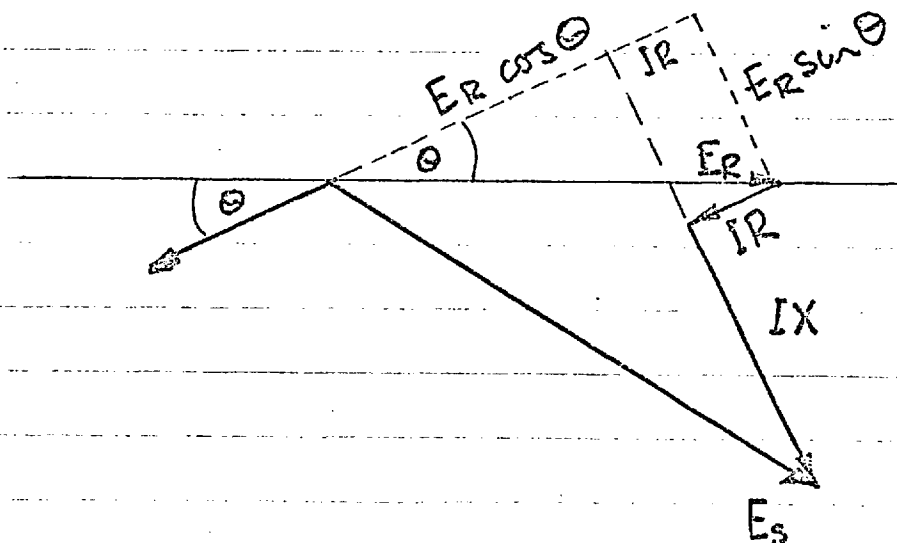
care must be exercised in establishing θ .
using $\overline{kW}$ and $\overline{kVAR}$



The equation becomes:

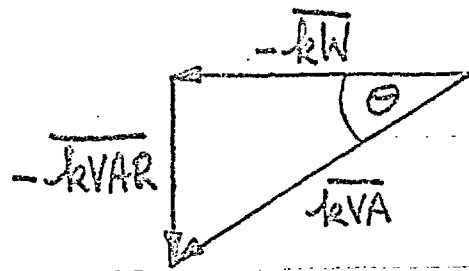
$$E_s^2 = E_R^2 - 2(R\overline{kW} + X\overline{kVAR}) + \frac{(R^2 + X^2)\overline{kW}^2}{E_R^2 \cos^2 \theta}$$

UNDER-EXCITED - GENERATING



$$E_s^2 = (E_R \cos \theta - IR)^2 + (E_R \sin \theta + IX)^2$$

Again care must be exercised in establishing θ . Using $\overline{kW}$ and $\overline{kVAR}$



The final equation becomes:

$$E_s^2 = E_R^2 + 2(-R \overline{kW} + X \overline{kVAR}) + \frac{\overline{kW}^2(R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

EXAMPLE 1

BASE MVA : 4.0

θ : 36.9° (LAGGING UNDER-EXCITED)

$\overline{kW}$: 20,000 (Motoring)

$\cos \theta$: 0.8

$\sin \theta$: 0.6

$\overline{kVAR}$: 15,000 (UNDER-EXCITED VAR's into motor)

E_s o/p : 1.044

R : 0.0033 o/p on a 4.0 MVA BASE

X : 0.0062 o/p " " " " "

o/p $\overline{kW}$: 5.00

o/p $\overline{kVAR}$: 3.75

$$E_s^2 = E_R^2 + 2(R \overline{kW} + X \overline{kVAR}) + \frac{\overline{kW}^2(R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

$$E_s^2 = 1.044^2 = 1.09$$

$$R \overline{kW} = (0.0033)(5.0) = 0.0165$$

$$X \overline{kVAR} = (0.0062)(3.75) = 0.0233$$

$$0.0398$$

$$R^2 + X^2 = 1.09 \times 10^{-6} + 38.5 \times 10^{-6} = 49.4 \times 10^{-6}$$

$$\frac{(\overline{kW})^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta} = \frac{(25)(49.4)(10^{-6})}{(E_R^2)(0.64)} = \frac{1.93 \times 10^{-3}}{E_R^2}$$

$$1.090 = E_R^2 + 0.08 + \frac{0.00193}{E_R^2}$$

$$1.01 = E_R^2 + \frac{0.00193}{E_R^2}$$

Hence solve by iteration or by completing the square

$$\underline{E_R = 1.048 \quad 0/1}$$

EXAMPLE 2

Let $\overline{kW} = 40,000$ MOTORING

$\overline{kVAR} = 30,000$

$\cos \theta = 0.8$

$\sin \theta = 0.6$

$$E_s^2 = E_R^2 + 2(R\overline{kW} + X\overline{kVAR}) + \frac{\overline{kW}^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

$$1009 = E_R^2 + 0.16 + \frac{0.00722}{E_R^2}$$

$$0.93 = E_R^2 + \frac{0.0072}{E_R^2}$$

$$E_R^4 - 0.93 E_R^2 = -0.0077$$

Complete the Square

$$\left(E_R^2 - \frac{0.93}{2}\right)^2 = -0.0077 + \left(\frac{0.93}{2}\right)^2$$

$$\left(E_R^2 - \frac{0.93}{2}\right)^2 = -0.0077 + 0.216 = 0.208$$

$$E_R^2 - 0.465 = \pm \sqrt{0.28} = \pm 0.457$$

$$E_R^2 = 0.465 \pm 0.457 = 0.922 \quad \text{-ve inadmissible}$$

$$E_R = 0.961 \quad \text{per unit values}$$

$$1.0 \text{ pu} = 66 \text{ kV}$$

$$E_R = 66 \times 0.961 = \underline{63,500 \text{ Volts}} \quad \text{L-L}$$

CHECK BY V and I

$$P_{3\phi} = \sqrt{3} E_{L-L} I_{L-L} \cos \theta = 1.732 \times 63,500 \times I_L \times 0.8$$

$$\therefore I_L = 455 \text{ A}$$

$$I_L R_{L-N} = 455 \times 3.55 = 1615$$

$$I_L R_{L-L} = 455 \times 3.55 \times \sqrt{3} = 2800$$

$$I_L X_{L-N} \sqrt{3} = 455 \times 6.68 \times \sqrt{3} = 5270$$

Check by calculation

$$V_{D \text{ L-L}} = \sqrt{(E_R \cos \theta + I_{L-L} R_{L-L})^2 + (E_R \sin \theta + I_{L-L} X_{L-L})^2} - E_R$$

$$\begin{aligned} E_R \cos \theta &= 63500 \times 0.8 = 50,800 \\ I R_{L-L} &= \underline{2,800} \\ &53,600 \end{aligned}$$

$$\begin{aligned} E_R \sin \theta &= 63500 \times 0.6 = 38,100 \\ I X_{L-L} &= \underline{5,270} \\ &43,370 \end{aligned}$$

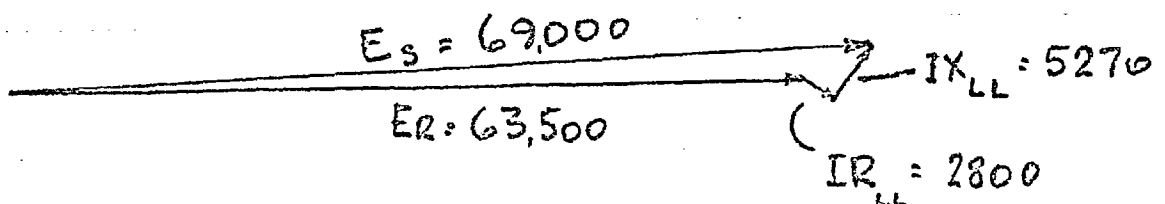
$$\begin{aligned} (5.36)^2 \times 10^8 &= 28.75 \times 10^8 \\ (4.34)^2 \times 10^8 &= \underline{18.80 \times 10^8} \\ &47.55 \times 10^8 \end{aligned}$$

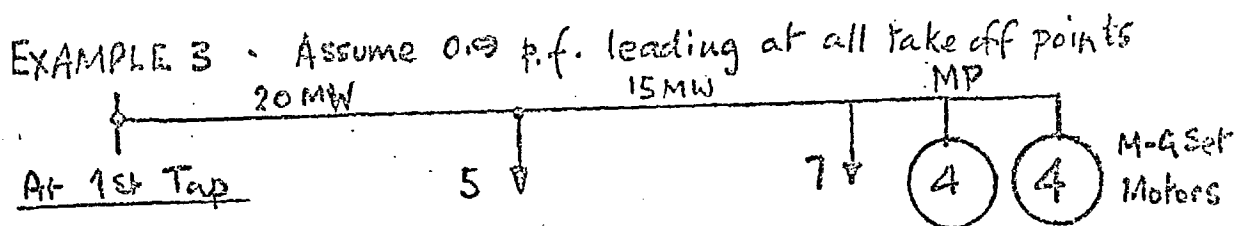
$$\sqrt{47.55 \times 10^8} = 69,000$$

$$V D_{L-L} = 69,000 - 63,500 = \underline{5,500} \text{ Q.E.D.}$$

Check by Graphical method

1. Draw the E_R vector (reference)
2. $I R_{L-L}$ is known in direction and magnitude
3. Draw $I X_{L-L}$ in quadrature with $I R_{L-L}$
4. Complete the phasor diagram by drawing vector E_s
5. Measure E_s , it should equal 69000.





$$E_B^2 = E_R^2 + 2(R \overline{kW} - X \overline{kVAR}) + \frac{\overline{kW}^2 (R^2 + X^2)}{E_R^2 \cos^2 \theta}$$

$$\theta = 25.8^\circ$$

$$\cos \theta = 0.900$$

$$\sin \theta = 0.435$$

$$\tan \theta = 0.483$$

$$\cos^2 \theta = 0.810$$

$$1.044^2 = E_R^2 + 2 \left[(0.0033 \times 5) - (0.0062 \times 2.42) \right] + \frac{(25)(0.492)(10^{-4})}{0.81 E_R^2}$$

$$\text{Base MVA} = 4.00$$

$$0/1 \overline{kW} = \frac{20}{4} = 5.0 \text{ o/a}$$

$$0/1 \overline{kVAR} = \frac{9.66}{4} = 2.42 \text{ o/a}$$

$$0/1 R = 0.0033 \quad 0/1 X = 0.0062$$

$$R^2 + X^2 = Z^2 = 1.09 \times 10^{-6} + 38.5 \times 10^{-6} = 49.4 \times 10^{-6}$$

$$1.09 = E_R^2 + 2(0.0165 - 0.015) + \frac{0.00152}{E_R^2}$$

$$1.09 = E_R^2 + 2(0.0015) + \frac{0.00152}{E_R^2}$$

$$1.087 = E_R^2 + \frac{0.00152}{E_R^2}$$

$$\text{By iteration } E_R = 1.042 \text{ o/a Volts}$$

At the SECOND TAP

$$E_s = 1.042$$

$$E_s^2 = 1.0855$$

$$0/1 \overline{kW} = \frac{15}{4} = 3.75$$

$$\%1 \overline{kW}^2 = 14.10$$

$$0/1 \overline{kVAR} = \frac{7.25}{4} = 1.81$$

$$1.0855 = E_R^2 - 2 \left[(0.0037 \times 3.75) - (0.0069 \times 1.81) \right] + \frac{14.1 \times 61.3 \times 10^{-6}}{0.81 E_R^2}$$

$$R^2 + X^2 = 13.7 \times 10^{-6} + 47.6 \times 10^{-6} = 61.3 \times 10^{-6}$$

$$1.085 = E_R^2 + 2(0.00138 + \frac{0.001068}{E_R^2})$$

$$E_R^4 - 1.0855 E_R^2 + 0.00276 E_R^2 = -0.001068$$

$$\left(E_R^2 - \frac{1.0827}{2} \right)^2 = 0.293$$

$$E_R^2 = 1.083$$

$$E_R = \underline{1.0406}$$

At the MP (metering point)

$$E_s = 1.0406 \quad E_s^2 = 1.083$$

$$0/1 \overline{kW} = \frac{8}{4} = 2.0$$

$$0/1 \overline{kVAR} = \frac{3.87}{4} = 0.967$$

$$R = 0.0012 \%$$

$$X = 0.0023 \%$$

$$1.083 = E_R^2 + 2 \left[(0.0012 \times 2.0) - (0.0023 \times 0.967) \right] + \frac{4.0 \times 7.04 \times 10^{-6}}{0.81 E_R^2}$$

$$R^2 + X^2 = 1.44 \times 10^{-6} + 5.6 \times 10^{-6} = 7.04 \times 10^{-6}$$

$$1.083 = E_R^2 + 2(0.0002) + \frac{0.0000347}{E_R^2}$$

By iterative process

$$E_R^2 = 1.083$$

$$E_R = 1.0406 \text{ O/1 value}$$

At the STRIPPING MACHINE

$$E_s = 1.0406, \quad E_s^2 = 1.083$$

$$\text{O/1 kW} = \frac{4}{4} = 1.0, \quad \text{O/1 kVAR} = \frac{1.935}{4} = 0.484$$

$$R = 0.0284 \text{ O/1}, \quad X = 0.1076 \text{ O/1}$$

$$1.083 = E_R^2 + 2 \left[(0.0284 \times 1.0) - (0.1076 \times 0.484) \right] + \frac{1.0 \times 124 \times 10^{-4}}{0.81 E_R^2}$$

$$R^2 + X^2 = (8.08 \times 10^{-4}) + (116 \times 10^{-4}) = 124 \times 10^{-4}$$

$$1.083 = E_R^2 + 2(-0.0236) + \frac{0.0153}{E_R^2}$$

$$E_R^2 - 1.13 = -\frac{0.0153}{E_R^2}$$

$$\text{By iterative process } E_R^2 = 1.119$$

$$\underline{E_R = 1.057 \text{ O/1 Volts} \rightarrow}$$

CONSIDER WORST STARTING CONDITION

1ST Motor on line without field $\approx 2000 \text{ kVAR}$

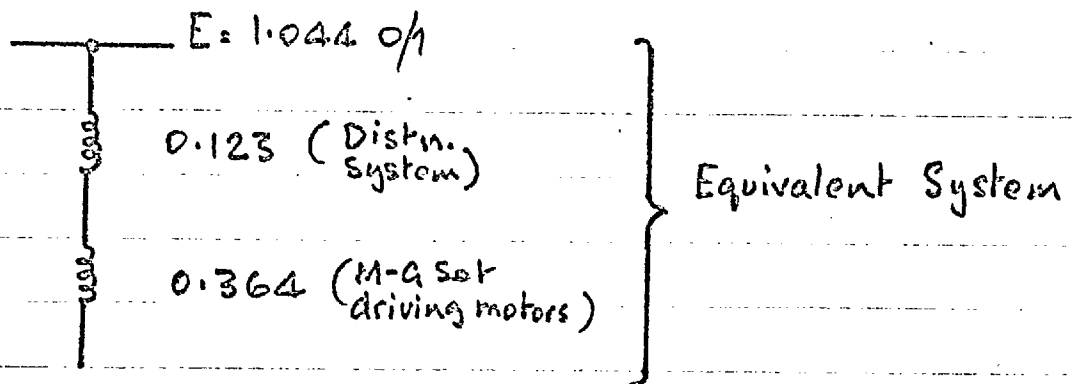
2ND Motor started up

ASSUME starting p.f. so low that only reactances are significant. - neglect Resistance effects.

ASSUME 450% current inrush on starting on motor's own base $\approx 9000 \text{ kVAR}$ at p.f. < 0.30

$$0/1 \text{ kVAR} = \frac{11000}{4000} = 2.75 \text{ o/1}$$

$$\text{The 2 motors are } \approx \frac{1.0}{2.75} = 0.364 \text{ o/1 ohms}$$



$$E_{\text{MOTORS}} = \frac{0.364}{0.487} E = \frac{0.364}{0.487} 1.044$$

$$\underline{E_{\text{MOTORS}} = 0.784 \text{ o/1}}$$

Note that this would be adequate to bring the M-G set driving motor up to full (synchronous) speed.

It will be seen that the majority of the system impedance is at the mine and not in the system itself

This simplified approach allows engineers to make initial calculations before precise electrical system parameters are established without recourse to network analysers or computers.

By inspection it will be seen that if the ratio of REACTANCE to RESISTANCE is high, that excessive VD (voltage drop) can occur. Calculations indicate that the System Reactance should be < 3.5 System Resistance

for satisfactory operation.

Additionally if the system impedance is high i.e. if most of the impedance occurs between the mine substation and the infinite bus, then further study and possible reinforcement of the system is necessary.

From total system stability considerations, the stripping machine total power swing from peak load to peak re-generation should not exceed 10% of the power system connected spinning capacity.

APPENDIX 5D

SOME EXAMPLES OF SINGLE BUCKET STRIPPING MACHINES
USING REHANDLING METHODS IN THICK OVERBURDEN

Direct casting of overburden across the pit ("side casting") is desirable because it has:

- a) shortest possible transport distance
- b) lowest possible lift, and
- c) permits economic reclamation of the mined out area.

Where the overburden is relatively deep, the length of the boom of a single bucket stripping machine becomes impracticable and some other method becomes necessary. The following methods are in use:

1. Long discharge boom BWE and BCE (Bucket Chain and Bucket Wheel Excavators) as developed in Germany (Rasper, L. "Continuous Direct Overthrow in Opencuts using Bucket Wheel Excavators". ["]Fördern und Heben, Vol 14, 1967). These machines are treated in more detail in Chapter 6.
2. Combinations of a long discharge boom BWE on an upper bench, discharging directly across the pit and a direct casting dragline or shovel on a lower bench.
3. Combinations of a BWE loading a "round-the-pit" conveyor system on an upper bench and a direct-casting dragline or shovel on a lower bench.
4. "Haymaking".
5. Extended Bench Method.
6. The Forecut Method.

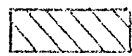
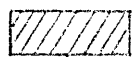
The first traceable deep overburden stripping operation was a direct, across-the-pit conveyor boom BCE machine manufactured by L.M.G., Lubeck, in 1897, for the clay pit of Ludovice in the Palatinate of the Rhine. In 1905 the same company built a similar machine with a 22m (72 ft) discharge boom. These machines appear to have established the principle of direct, across-the-pit conveyor boom BWE machines eventually adopted by Kolbe in the 1940's, in the U.S.A. (see ref.⁶⁸) for thicker overburden removal.

Combinations of BWE "round-the-pit" operations using side slewable belt conveyors, with direct casting on the bottom benches using large single bucket stripping machines have since found application in thick overburden e.g. Demerara Bauxite, Gyana; Radar North, U.K.: etc. Additionally "round-the-pit" overburden transport systems are used in conjunction with stacker systems directly casting across the bottom benches e.g. Neyveli, South India, Ptolemias, Greece, etc.

Where strong overburden occurs, where it fragments badly or where boulders may be encountered, rehandling methods using single bucket stripping machines, usually draglines, offer marked advantages in thick overburden, due to their ability to handle large lump sizes.

U.S. Gypsum's Haymaking Operation

This mine at Alabaster, Mich., U.S.A. progressively moved into deeper overburden up to 20m (70 ft) thick, comprising glacial till composed of a clayey material. It is stable when relatively dry but unstable when water is left on the exposed floor of the mine or allowed to collect on the spoil bank surface, and spoil slides can occur. Also when snow is mixed with the overburden in winter instability

-  - Rehandled Spoil
-  - Mineral

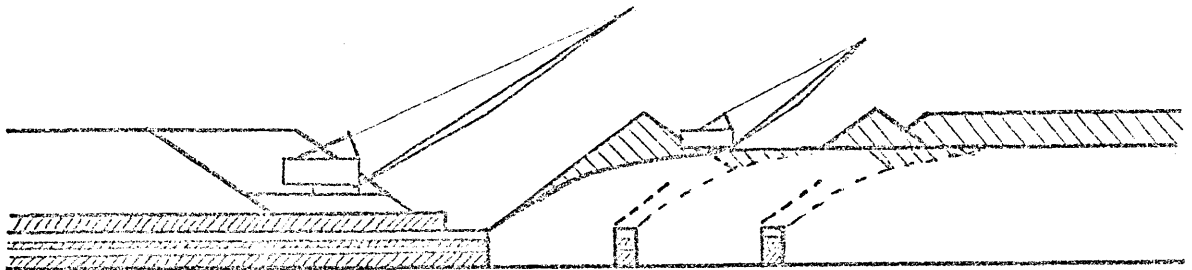


FIG 5D1 "HAYMAKING" METHOD - ALABASTER, MICH., U.S.A.

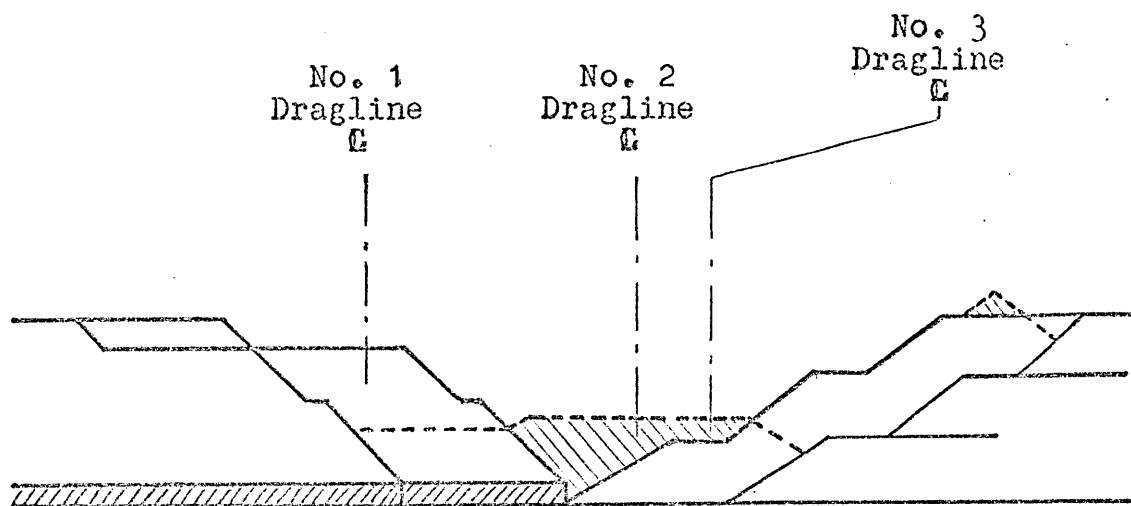


FIG 5D2 EXTENDED BENCH METHOD - BAUXITE, ARK., U.S.A.

occurs when thawing occurs. Experience indicated that it was impractical to build a spoilback without rehandling. Because of this a smaller "pull-back" dragline was introduced.

The highwall slope has to be maintained at 37° for stability and the main dragline stands on an intermediate bench to obtain a competent base, chopping down approximately 75% of the highwall, normally dragging the remainder. Re-handling is limited to between 20 - 40%, averaging 35%. Pulling back the spoil enables a stable spoil bank 12 - 15m (40 - 50 ft) high to be formed next to the strip, the remaining spoil being dumped well back from the strip, flattening the overall spoil slope and promoting stability. A stable spoilbank is therefore built in very difficult conditions by flattening of the spoil slope and creating more spoil space in the immediate vicinity of the strip. A further measure, leaving a pillar of mineral, can be adopted when conditions deteriorate (Fig 5D1)

Renolds Mining Co., Bauxite, Ark., U.S.A.

This operation strips over 40m (140 ft) of overburden from a bauxite deposit. The overburden comprises mostly unconsolidated materials with some boulders, and is not sufficiently consistent to allow a fixed pattern operation, but some principles are followed.

The topography is rolling with trees and dense undergrowth. The area has a well defined drainage pattern but because of high rainfall can become flooded. This has a marked effect on bank stability.

The dragline chops down Cut A1 because the surface is not competent to support the dragline (Fig 5D2). A1 spoil is saturated and unsuitable as a base material for the spoil bank. It is therefore temporarily dumped on top of the

surface of the next cut for drying or placed in the filled pit section for rehandling to the top of the final spoilbank. Cut A2 is normally dragged from below the tub of the dragline on the same traverse of the pit and placed in the bottom of the fill section as the foundation for the spoilbank. After A1 and A2 are removed for a length of strip, the fill is levelled to provide a second dragline bench. The dragline then "walks" to this bench and removes Cut A3 dumping it on top of the spoilbank being constructed. This is a modification of the normal extended bench method. About 20 - 30% rehandling of spoil is necessary.

The top and bench surfaces are dozed level to make them less permeable to water, and this greatly improves spoilbank stability.

The company plans to mine down to 60m (200 ft) deep overburden. Again Cut A1 would be taken. During the same advance Cut A2 would be dragged to about 27m (90 ft) below the dragline tub, leaving about 24m (80 ft) to be removed from Cut A3. The dragline then moves to the top of the spoil bridge and casts as much as possible of the rehandled material to the spoil bank. The dragline next moves to where the remainder of the rehandled material can be re-cast.

The rehandling is expected to be 50 - 60% but is still more economic than any other analysed method. A small crawler-mounted dragline will be used on the spoilbank for further rehandling and to establish stable slopes.

Goonyella Mine, Australia

This operation of Utah International in Queensland is an adaptation of the extended bench method. The selection of this method was influenced by the thick overburden up to 60m (200 ft) and thick coal seam 7.5m (24 ft). The method is

shown in Fig 5D3. Stage A shows the first dragline operating from a level bench about 6m from the surface. It chops down ahead of itself (cut A) and dumps the material in the adjacent mined out area as shown.

Stage B shows the same machine digging below itself in a conventional manner for cut B. The spoil is dumped so a crawler bulldozer can easily form a working surface for the second dragline.

In stage C the second dragline uses a cross chopping method in removing cut C. This action plus the relatively large swing angle reduces the stripping rate but in order to obtain the maximum reach possible from the dragline in thick overburden, this method of working is essential. With cut C removed the dragline moves along the strike and rehandles a portion of the spoil from cuts A and B, dumping it on the spoil bank as shown.

The overall slope of the highwall is flattened by the berm and the spoil bank is also greatly stabilised by the presence of the flat working bench.

Texas Gulf Sulphur Co., Lee Creek, N.C., U.S.A.

This phosphate deposit is 27m below sea level, the surrounding ground level being 4m (14 ft) ASL. Fig 5D4 shows the operation using the Forecut Method previously described, with the following modifications.

The smaller dragline in addition to excavating the forecut, also excavates the shell bed, a source of phosphate, dumping it into a well from which it is pumped to a stockpile. The main dragline chops down a bench to give the smaller machine space to dump the forecut spoil, rehandles the forecut spoil to the spoilbank, normally drags below its tub level and dumps this material on the spoil bank. It also

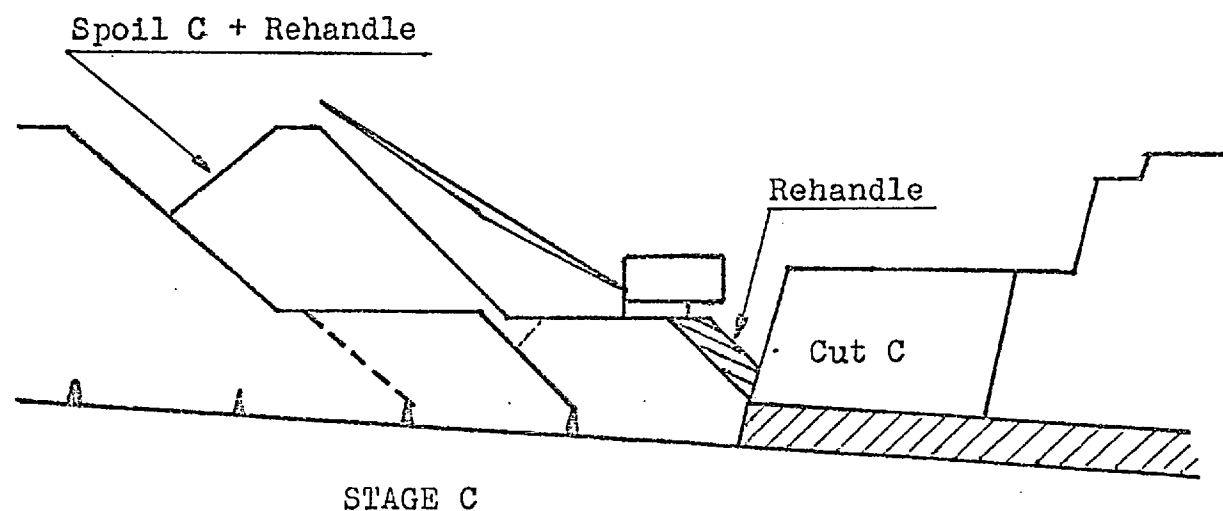
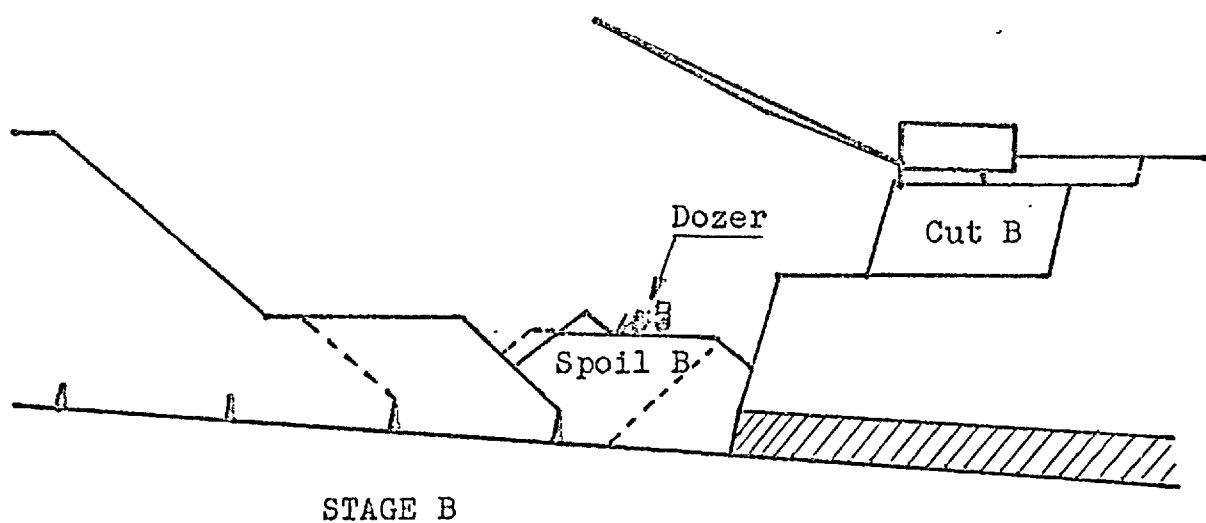
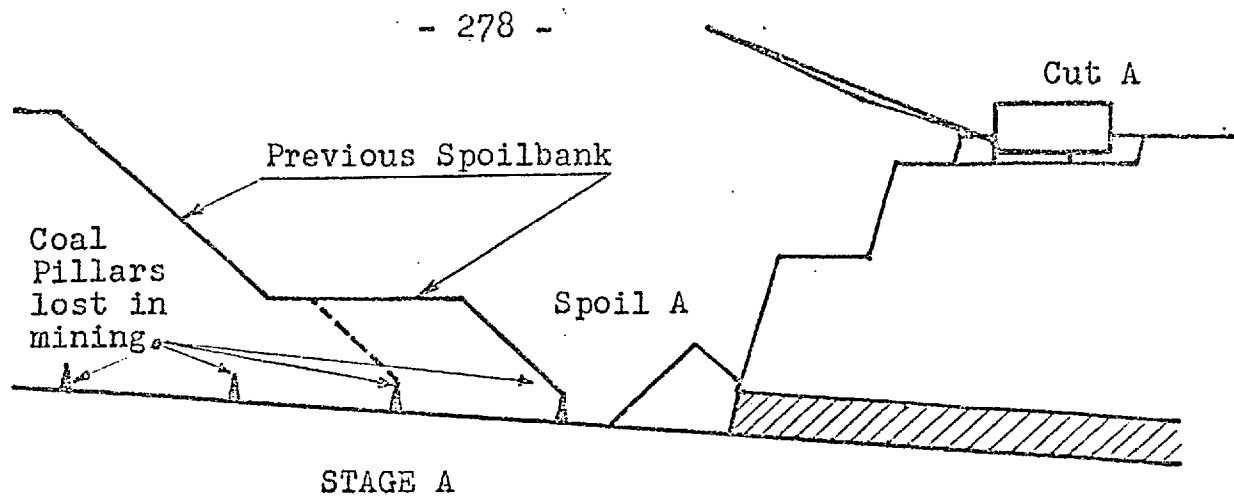


FIG 5D3 GOONYELLA MINE OPERATIONS, AUSTRALIA.

 — Mineral

 — Rehandled Spoil

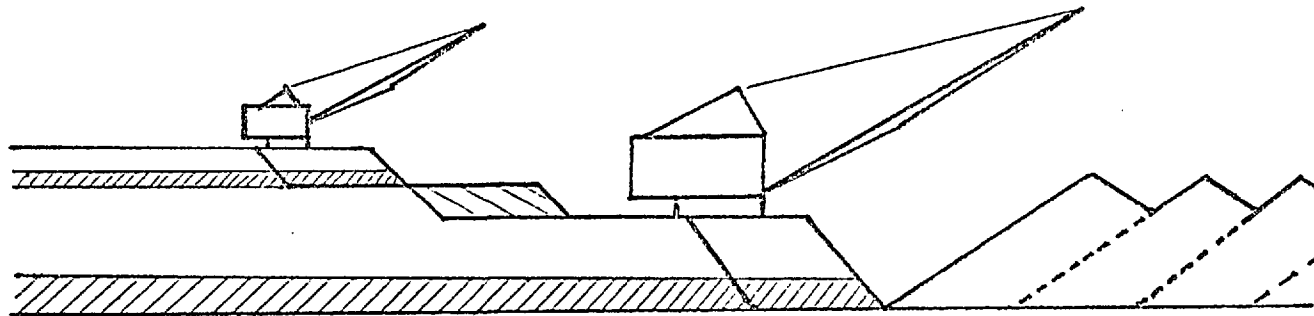


FIG 5D4 FORECUT OPERATIONS - LEE CREEK MINE, AURORA, N.C., U.S.A

excavates the main phosphate bed and dumps it into a trench at the toe of the new highwall of the main dragline bench. The phosphate is then sluiced along the trench by hydraulic monitors to the processing plant.

6. CONTINUOUS EXCAVATORS

The two most important types of continuous excavator - the bucket-chain and the bucket-wheel excavator were mainly developed in the German brown coal fields. Some important developments have also taken place in Czechoslovakia, the U.S.A. and the U.S.S.R.

The Bucket Chain Excavator (B.C.E.)

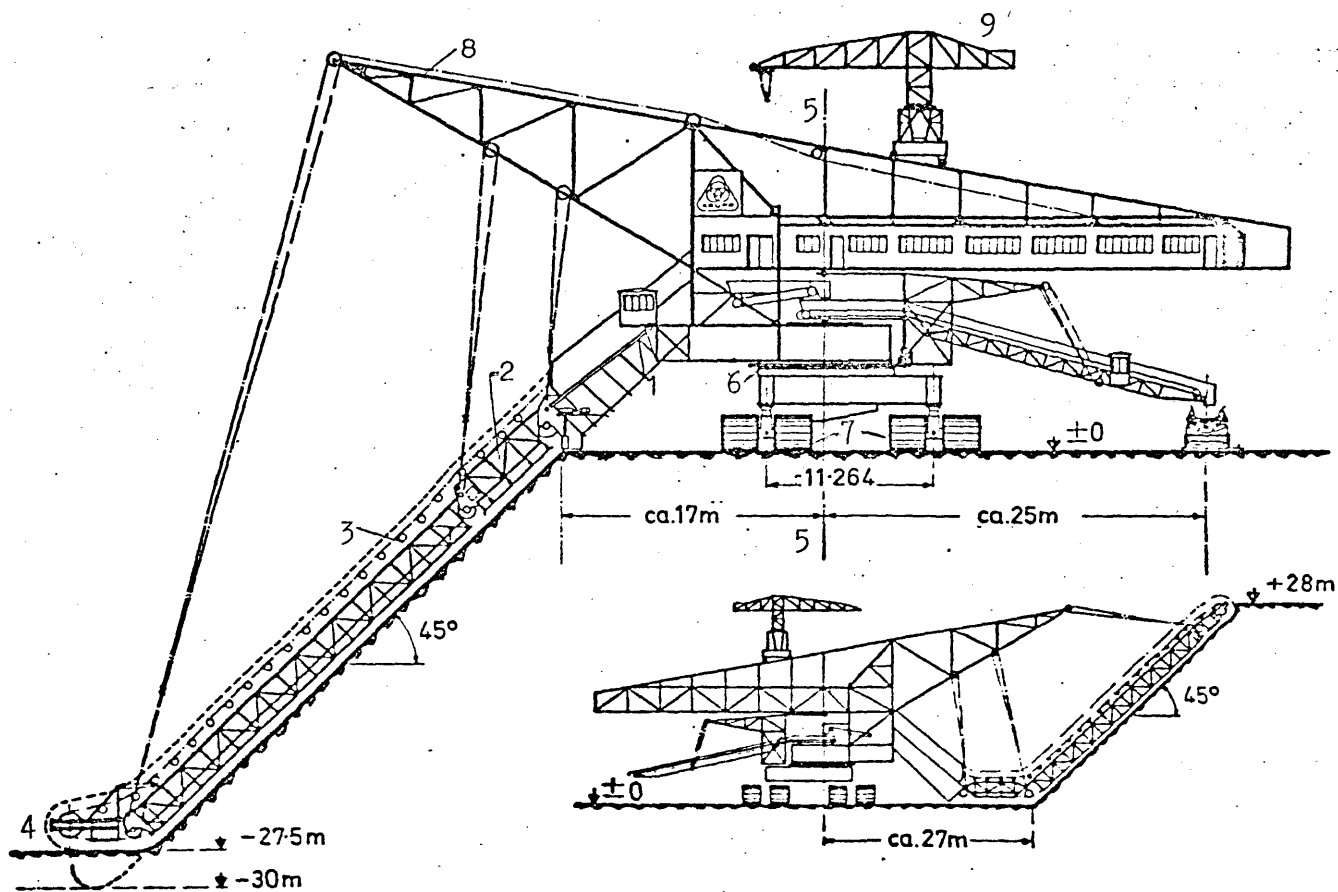
Earlier machines were rail-track-mounted, but this involved frequent track shifting due to the inherently narrow width excavated at each pass of the excavator and modern machines are invariably of the slewing type mounted on crawlers, capable of performing block excavating operations. Fig 6.1 shows a large BCE which is in service in an Italian mine: it has a theoretical output of 1470 m^3 (1920 yd^3) bank per hour, a machine mass (service weight) of 1495 tonnes (1640 short tons) and an installed ac power of 2285 kW (3036 hp).

The BCE can mine high outputs in weak unconsolidated ground. It cannot dig hard ground and is not selective in operation. Its most advantageous feature is its excellent down-digging capability, combined with reasonable upward-digging capability (Fig 6.1).

The terminology used for the BCE is illustrated in Fig 6.1.

The Bucket Wheel Excavator (BWE)

The BWE is the most effective machine for mining large outputs in weakly consolidated ground, although its application is extending into harder formations. It can selectively mine faulted or intercalated ground (Fig 6.2). It can accurately cut bands as thin as 100mm (4 in), but with



- 1 - Chute
- 2 - Levelling Piece
- 3 - Ladder
- 4 - Planing Piece
- 5 - Slew Axis
- 6 - Slewing Ring
- 7 - Crawlers
- 8 - Jib
- 9 - Service Crane

FIG. 6.1 THE BUCKET CHAIN EXCAVATOR

much reduced output. It has poor downward-digging capability, but this can be improved by a boom sandwich belt, or a special long boom design which reduces the boom angle sufficiently to allow successful conveyor operation when digging below grade. The sandwich belt does not give satisfactory service when boulders or large frozen lumps are being handled.

The BWE is fitted with evenly spaced buckets on the periphery of the wheel. Excavated material is fed via a transfer point inside the wheel, e.g. a plough, rotating disc, etc., to the belt conveyor system of the excavator for discharge. Machines are in service with theoretical outputs of up to 10000 m^3 (13000 yd^3) bank per hour and with machine masses (service weights) exceeding 7000 tonnes (7700 short tons). A BWE of 13000 tonnes (14300 short tons) machine mass and a theoretical output of the order of 20000 m^3 (26000 yd^3) bank per hour is scheduled for service in West Germany in 1976. This is undoubtedly an expression of confidence by West German mining engineers to build and operate these huge machines for economic mining.

Advantages of Continuous Excavators

Continuous excavators have lower impact loadings than comparable single-bucket machines, which reduce dynamic stresses, machine mass (service weight), maintenance costs and power consumption. The slower slewing (swing) speed and reduced digging impacts result in gradual load transfer across the crawlers, with a consequent reduction in ground bearing pressure: operation is therefore possible in conditions in which a single-bucket machine may work with difficulty.

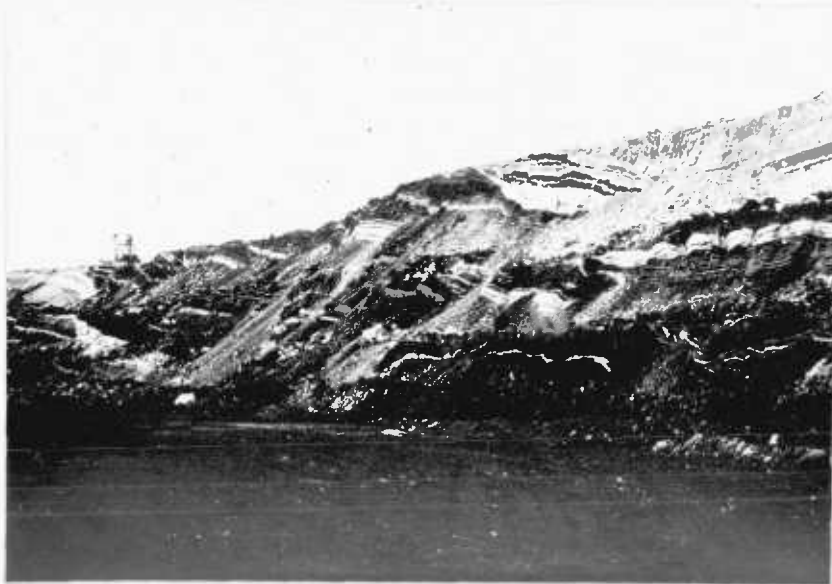


FIG. 6.2a INTERCALATED
STERILES - PTOLEMAIS
LIGNITE (Selective
Mining)

- 1 - Bucket Wheel
- 2 - Bucket Wheel Boom
- 3 - Discharge Boom
- 4 - Counterweight
- 5 - Tower
- 6 - Slew Axis
- 7 - Slewing Ring
- 8 - Crawlers

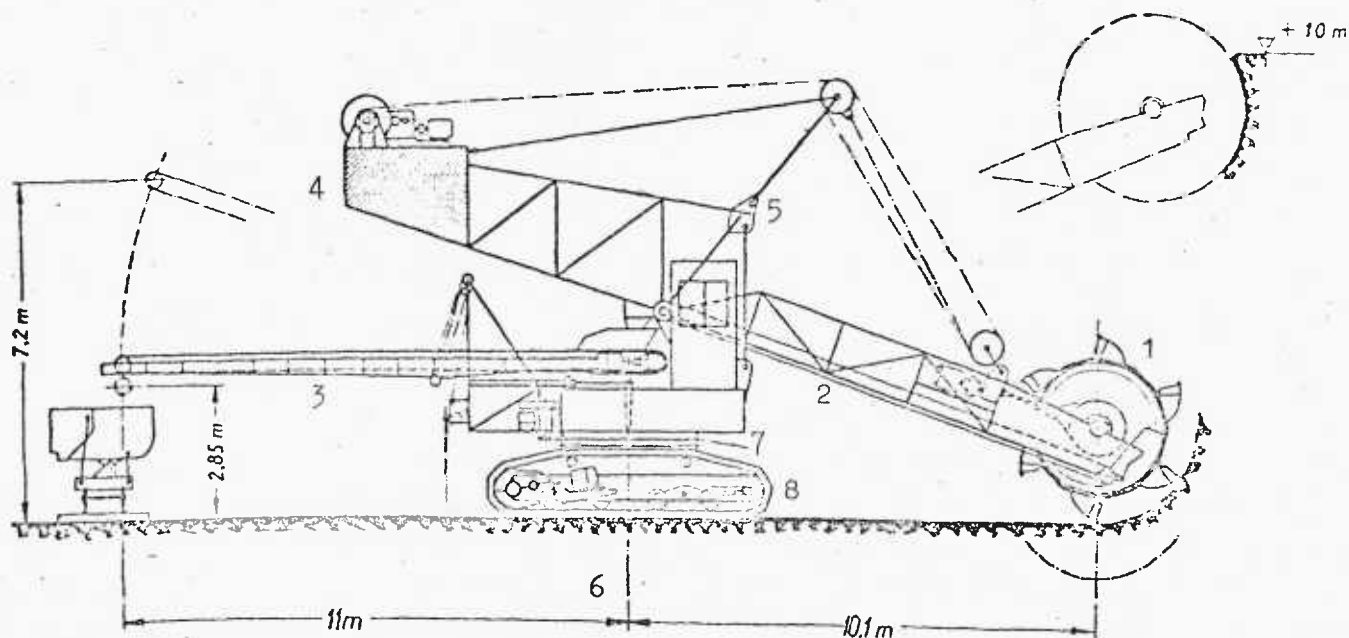


FIG. 6.2b THE BUCKET WHEEL EXCAVATOR

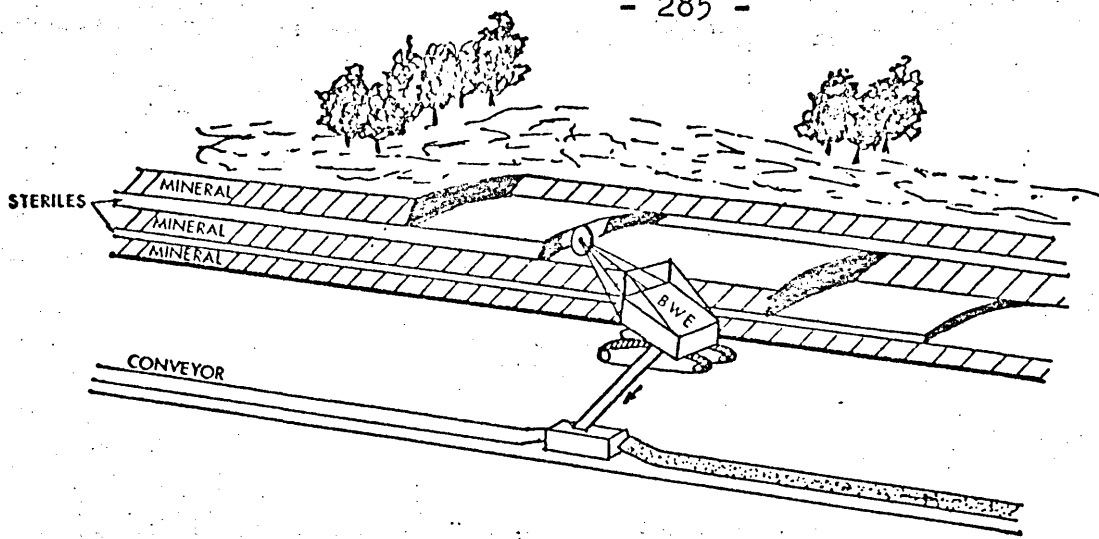


FIG. 6.3 BWE PART BLOCK OPERATION

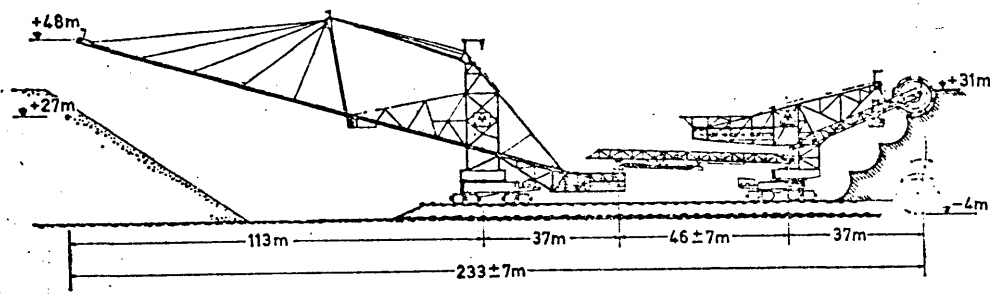


FIG. 6.4 BWE AND MOBILE BOOM STACKER (Across the pit dumping)

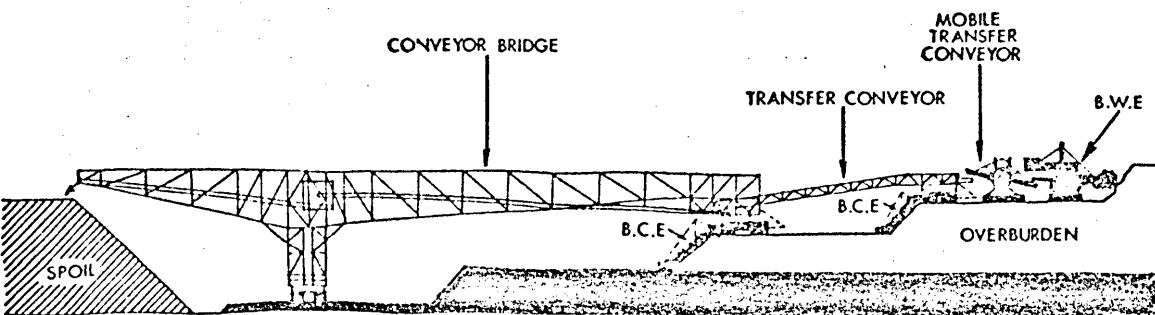


FIG. 6.5 THE OVERBURDEN BRIDGE METHOD

TABLE 6.1

APPROXIMATE POWER CONSUMPTION OF EXCAVATORS*

Machine	Power consumption kWh/m ³	
Loading shovel	0.45	- 0.71
Stripping shovel	0.52	- 0.91
Walking dragline	0.88	- 1.21
BCE	0.41	- 0.60
BWE	0.30	- 0.50

*Collected from wide variety of sources,
see Ref ⁶¹.

Land reclamation is generally easier with continuous excavators - many fine examples of reclamation can be seen in Germany and Czechoslovakia.

BCE versus BWE

Since the second world war the BWE has, to a large extent, superseded the BCE. The BCE continued to be build in East Germany, where ground conditions are comparatively easy and where designs and operational experience with the BWE were not available. The present trend in East Germany, however, is towards the BWE.

In selecting a continuous excavator the following points must be considered.

- (1) The BWE is selective in operation because of its horizontal action in part block or lateral operation (see Fig 6.3). The BCE cannot selectively mine a deposit.
- (2) The BWE can, in certain conditions, excavate relatively hard ground (sandstone, shales, etc.). The BCE cannot dig hard ground or handle occasional boulders.

(3) Because it has a much greater number of wear points, the maintenance costs of the BCE are higher than those of the BWE. Consequently, the availability of the BCE is less than that of the BWE, when working in the same ground conditions.

(4) The BWE has excellent high-digging capability, whereas the BCE is not as efficient owing to bucket filling difficulties in high digging. The BCE has excellent deep-digging capability and the BWE has relatively poor deep-digging capability.

(5) The bucket clearing action of the BWE is superior to that of the BCE and it has less difficulty in handling sticky materials.

The following conditions favour the BCE, provided that boulders or hard bands are not present and selective mining is not necessary:

- (a) soft, nonabrasive rocks where high and deep cuts must be performed by one machine;
- (b) situations in which the initial box cut is opened in these conditions;
- (c) wet pit operations;
- (d) where it is necessary to reduce transport gradients;
- (e) overburden bridge operations where cuts must be taken below grade level of the bridge crawlers ⁶²,
- (f) where exact slope profiles are required; and
- (g) where there are large undulations in the surface of the mineral bed.

Most of these advantages are due to the better downward-digging capability of the BCE. Most other applications, how-

ever, favour the BWE.

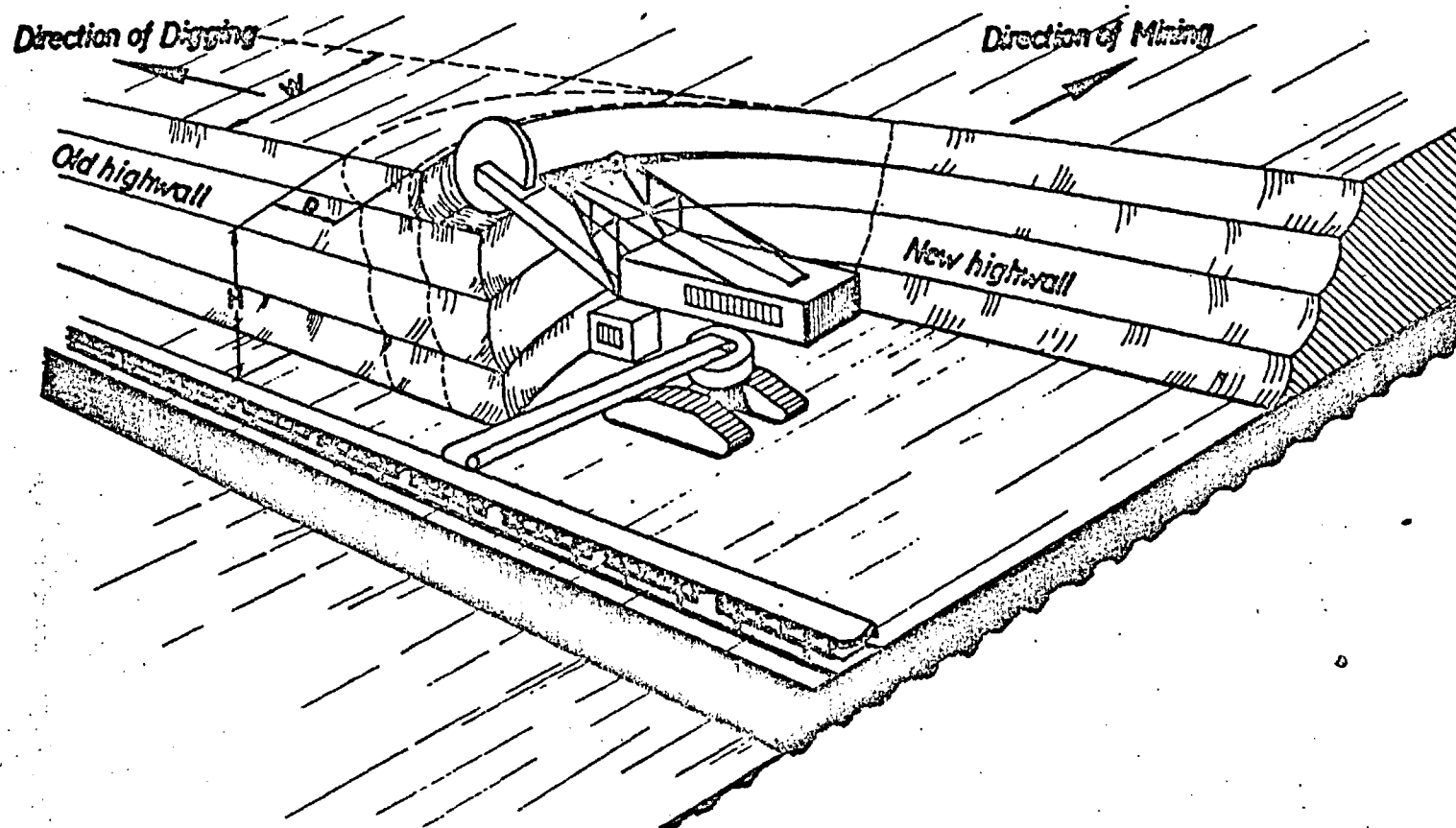
OPERATIONS

Continuous excavators are generally limited to the following gradients: excavating, 1 in 20: travelling, 1 in 10.

The BWE can be used in thin mineral/thin over-burden strip mining operations in place of direct casting and can strip deeper overburden or provide a greater pit width than single-bucket stripping machines. The BWEs in service at the River King and Northern Illinois mines of Peabody Coal Co. in the U.S.A. are typical examples of this form of machine.

To obtain the advantage of the shortest possible lift and transport distance and lowest stripping costs in thicker overburden and/or thicker mineral, the direct across-the-pit range of the BWE and the BCE can be extended by use of (a) the mobile stacker boom (Fig 6.4) and (b) the overburden bridge (Fig 6.5). Because of their limited operational flexibility these units impose a strict discipline on planning, and the following conditions should exist:

- (1) sufficient proved reserves to justify the capital cost of the relatively expensive machine complex.
- (2) relatively horizontal stratification over a wide area of the deposit and
- (3) uniform geological conditions, i.e. absence of major faulting, severe undulations, large variations in overburden thickness, etc.



Block-digging operation with wheel excavator feeding onto a movable conveyor belt.

H=Height of face; D=Depth of block; W=Width of block.

FIG. 6.6 BWE FULL BLOCK OPERATION (Fixed Boom Terrace Cut)

The most important application of continuous excavators is in weak sedimentary deposits of relatively great thickness where direct transport of the overburden across the pit is impracticable because of excessive pit width. Most of these operations employ several benches, and overburden must be transported round the pit by a conveyor or rail transport system and dumped in the spoil space provided by the advancing pit.

The full block method with BWEs (Fig 6.6) is normally adopted where selective mining is not necessary. The part block method has lower volumetric efficiency since the thickness of the ground being mined is usually less than the optimum, it involves greater travelling time, with increased crawler wear and reduced digging time, and is therefore restricted to selective mining operations. Short booms are not suitable for part block operations. If part block operation is proposed at a horizon high up the bench face, an oversize bucket wheel is necessary to obtain clearance for the boom if a reasonable block width is to be maintained.

Blasting for continuous excavators is not normally required. Recently BWE's capable of digging hard ground without blasting have been developed, e.g. at Neyveli, South India for an abrasive sandstone; in a hard, pottery-clay pit in the Rhineland where a high bucket wheel speed and small chip size have replaced blasting and shovel operation⁶³; and in the U.S.A. for digging shales previously blasted for shovel loading⁶⁴.

The productivity of an excavator is primarily determined by:

- a) the digging rate at which material can be cut and collected from the exposed working face, and
- b) the discharge rate at which material can be discharged and the cutters can resume the digging process

Although some continuous excavators have cutting "bows" leading the bucket lip, most machines perform the cutting operation with the bucket lip only.

A major limitation of shovel and dragline outputs is cutting speed. In these machines the bucket is much greater than those of continuous excavators of comparative outputs. The correspondingly higher cutting and lifting forces necessitate low cutting speeds which in large capacity shovels and draglines range between about 0.75 - 1.1 m/sec (150 - 220 ft/min).

Maximum cutting speeds of continuous excavators are as follows:

BCE	-	1.4 m/sec	(275 ft/min)
BWE	-	5 m/sec	(1000 ft/min)

Consequently continuous excavators have a higher productivity potential than single bucket, i.e. cyclic, excavators.

Continuous excavators should discharge without interruption and simultaneously with excavation. If however bucket emptying at the discharge point is incomplete, material is carried over and taken back to the working face or on to the incoming buckets in BCE's with a consequent reduction in output. The emptying of buckets is of even greater importance in continuous excavators than in cyclic (single bucket)

excavators, particularly in materials that do not run freely. Restricted bucket discharge is one of the major limiting factors in BCE performance. Previously the productive capacity of BWE's was also restricted by slow discharge but the introduction of the cell-less bucket wheel greatly improved discharging and consequently improved productivity⁶⁴.

Transport Systems

Clearly considerable planning is essential before a major, almost irrevocable capital decision can be made to select a continuous excavator. As with any continuous system, the excavators and their transport systems must be engineered initially as a complete system.

The two main systems used with continuous excavators for round-the-pit transport are:

- a) high-speed belt conveyors, and
- b) locomotive rail haulage

The side-slewable, high-speed belt conveyor is gaining preference over rail transport because operation on steep slopes (1 in 3) is possible, large areas used for wide curves, inclines, sidings, etc., by rail track especially in deep pits, being eliminated and the time spent by the excavator in waiting for the transport system is greatly reduced, i.e. the whole system becomes truly continuous.

The main disadvantages of high-speed belt conveyors are their high capital cost and high power consumption.

Rail transport can still prove economic where long distances on relatively horizontal grades have to be covered, e.g. from benches to treatment plant, pit to outside spoil bank, etc.

As each block width is mined (Fig 6.6) the transport system must be advanced. To reduce the frequency of this operation and hence the production lost, a mobile, crawler-mounted, transfer conveyor can be used between the excavator and the transport system to increase the range of the excavator.

The "three bench" system (see later) is also adopted to increase the range of large BWE's and reduce conveyor shifting.

OUTPUT OF CONTINUOUS EXCAVATORS

Continuous excavators are usually rated in terms of theoretical output, where

$$Q_{th} = \frac{60.F.s}{\text{Swell factor}} \text{ m}^3/\text{h (bank)} \quad \dots\dots\dots(6.1)$$

where Q_{th} is the theoretical output, F is the capacity of a single bucket and s is the number of bucket discharges per minute, and the swell factor is that of the material being excavated.

Some caution is needed in comparing the theoretical outputs of similar BWEs as manufacturers of different nationality may define F, the bucket capacity, differently. In addition to the actual bucket capacity, some may add, for cell-less buckets, 50% or 100% of the associated ring volume and for bucket wheels with cells the volume of the individual cell. It is suggested that the best basis for comparison is to use the bucket volume only. For selection purposes however on a systems basis a much more fundamental understanding of continuous excavator operations is essential. The following analyses are intended to provide this.

BCE Output

Although rail-mounted BCE's have been commissioned recently in East Germany it is unlikely that further machines will be built and crawler-mounted machines will undoubtedly be selected in the future. It is also unlikely that non-slewing BCE's will be built in the future. The following types of BCE's are in service

1. Downward digging, rail-mounted, non-slewing.
2. Non-slewing, crawler-mounted. Some of these machines have been converted from rail-mounted machines. All are of course suited to downwards digging only.
3. Upward-downward-digging, rail-mounted. To achieve upward and downward digging the BCE must be able to slew its superstructure 360° relative to the underframe (or sub-structure). This machine cannot perform block excavating operations, as the distance between the track and the highwall is too small.
4. Upward-downward-digging, crawler-mounted. This machine is also capable of 360° slewing and can perform block excavating operations.

Type 4 is probably the only form of BCE that will be built in the next decade. The operations of types 1, 2 and 3 are also performed by type 4, so analysis of type 4 only is necessary. Fig 6.7 illustrates its operation. A typical digging cycle commences with the chute mouth a distance S_a from the top edge of the highwall and the ladder horizontal (Fig 6.7). The ladder, which is pivotted at the chute mouth, is lowered a small amount corresponding to the bucket depth of cut T and the necessary sideways movement is made by the

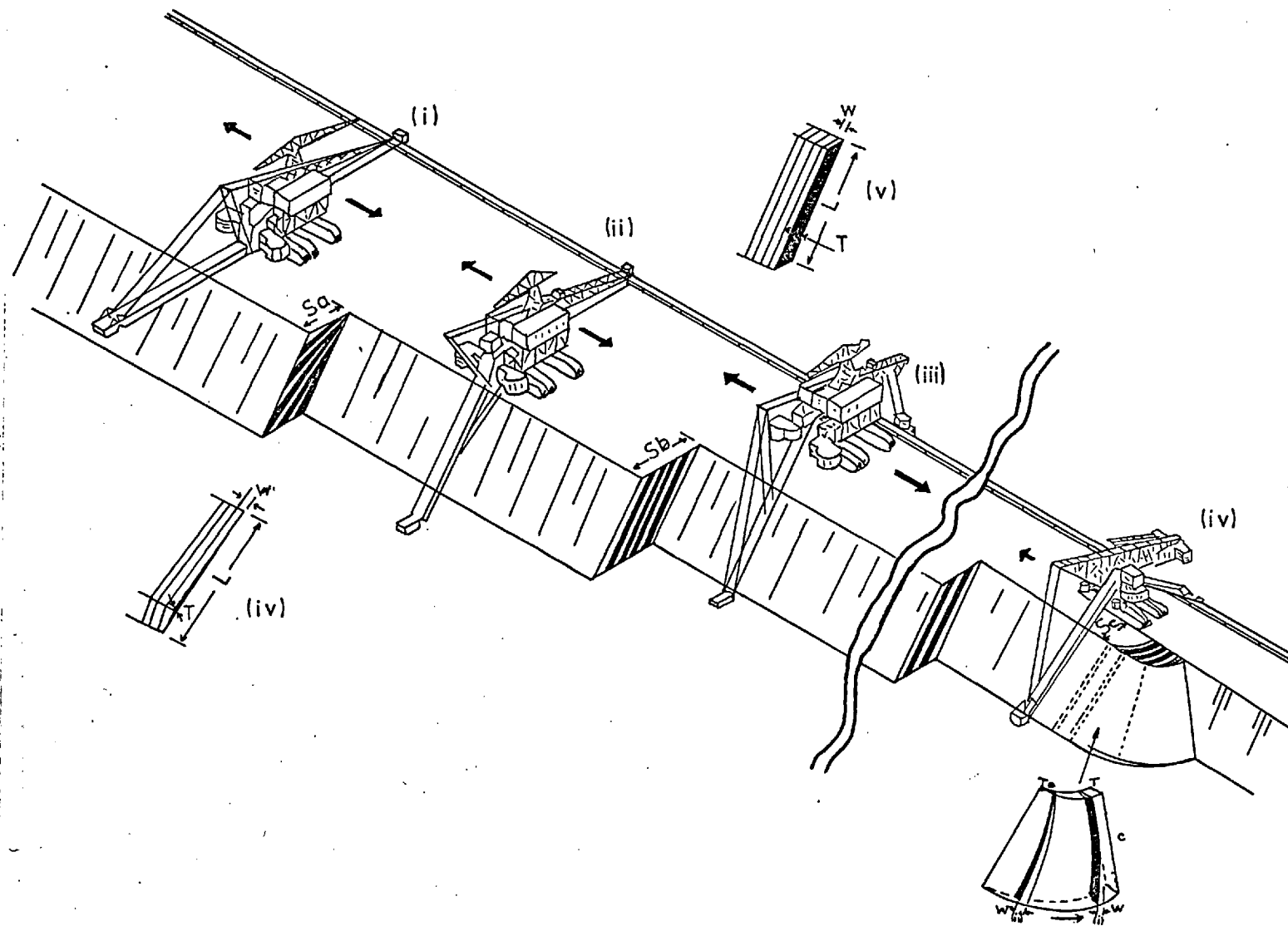


FIG. 6.7 CRAWLER MOUNTED BCE DOWNWARDS DIGGING

BCE travelling on its crawlers. At the end of each pass the ladder is again lowered the requisite amount and crawler travel is reversed. This process is repeated until the ladder has been lowered to full face depth, the planing piece becoming horizontal. (Fig 6.7).

Part block width S_b is then dug by parallel cuts (or slice cut excavation), the ladder inclination remaining fixed. (In operations, surface irregularities and steering variations require minor ladder adjustments to vary the length of the cut). The process continues until S_b is excavated (Fig 6.7).

Part block width S_c may then be excavated by block excavating method, i.e. using a slewing cut to remove the segment shown in Fig 6.7, this ensuring the maximum BLOCK WIDTH, $S = S_a + S_b + S_c$, for each advance of the transport system. Depending on conditions however, the transport system may be advanced after each time S_b has been excavated since

- a) the parallel cut is more productive than the slewing cut, and
- b) the crawlers must travel a second time over the same ground to excavate part block width S_c . If the bench surface is likely to break up, the transport system would be advanced and the cycle recommended.

During the slewing cut the inclination of chute, ladder and planing piece remain unaltered except for minor adjustments for surface irregularities. Each cycle consists of slewing the ladder from 90° to the conveyor, in a direction towards the conveyor to where its projection angle to the

line of advance is equal to the required standing slope of the highwall (usually approximately 60°). Slewing then ceases and the BCE is advanced distance T away from the working face. Slewing is then recommenced in the reverse direction until the ladder has returned to a position 90° to the conveyor. Fig 6.7 shows the segment shape of cut for one swing. It can be seen that T decreases to zero at the outer side of the segment. In practice to compensate for this the slewing speed can be varied over a range of about 2 - 6 m/sec (7 - 21 ft/sec), but even so bucket filling does decrease after reaching a slew angle of 76° . To reduce this effect, slewing can be reversed when T has decreased to about $\frac{1}{3}$ of its maximum value and the material left to be dug during the third outward swing.

A wider part block width than S_c may be excavated to obtain a wider block width, increasing the track shifting range, but this requires the excavation of an intermediate section between the working face and the new highwall, involving considerable boom hoisting movements and inefficient digging. Consequently the "semi-wide" block (Fig 6.7) is adopted.

Upward digging is shown in Fig 6.8. First part block width S_a is excavated. After cutting a working face, the ladder remains at the same inclination as the highwall, the levelling piece remains horizontal and for each segment the BCE moves towards the working face distance T . The segment shape is shown in Fig 6.8. As with the deep cut, full slewing may be deferred till every third cut. On completion of S_a , part block width S_b is excavated using a parallel cut.

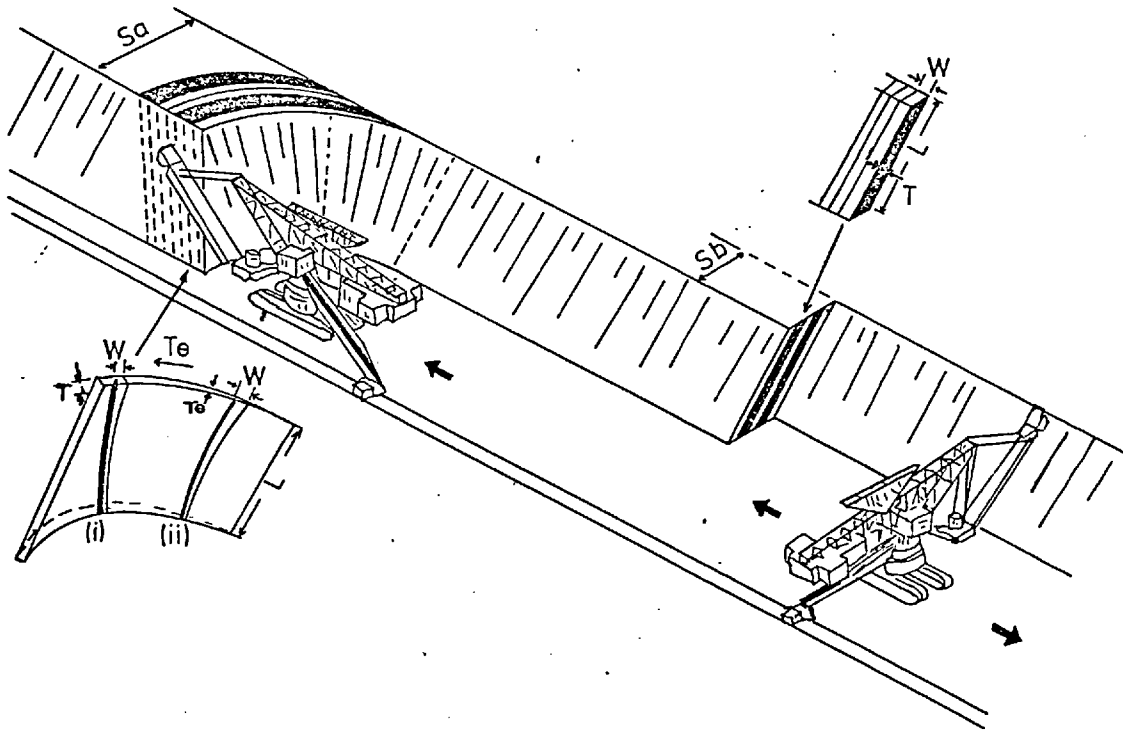


FIG 6.8 B.C.E. UPWARDS DIGGING

Fig 6.7 shows the shape of slices made by successive cuts for part block width S_a . The bucket filling for each slice is derived as follows:

Symbols

f_p = bucket filling factor for each pass
 f = average bucket filling factor
 t' = travelling time per pass (minutes)
 t = total travelling time for a number of passes
 (minutes)

W = width of cut for each bucket
 T = depth of cut normal to the ladder
 L = length of cut parallel to the ladder
 V_s = sideways (travel) speed per minute
 SF = swell factor
 J = bucket nominal contents
 s = bucket discharges per minute
 u = utilization factor, i.e.

Period while chain is digging

Total BCE operating period for excavation of complete block, but excluding transport delays

Q = effective hourly production (bank)
 Q' = hourly output rate per pass (bank)

Then

$$f_p \times J = 1/2 T.L.W.SF$$

$$\text{i.e. } f_p = \frac{T.L.V_s.SF}{2 J.s} \quad \dots\dots\dots (6.2)$$

and the production per hour, i.e. for a number of passes, may be expressed as:

$$Q = \frac{u(f.J.s.60)}{SF} \quad \dots\dots\dots (6.3)$$

Although in service slight adjustment in travel speed may be necessary to compensate for face irregularities, the travel speed V_s for each pass may be regarded as constant for the purpose of expressing output rates. Thus the hourly output during a pass is:

$$Q' = \frac{1}{2} \cdot T \cdot L \cdot V_s \cdot 60 \quad \dots\dots\dots (6.4)$$

and the effective hourly output for a number of passes may be expressed as:

$$Q = \frac{u}{t} \sum Q' t'$$

$$\text{i.e. } Q = \frac{30 \cdot u}{t} \sum (T \cdot L \cdot V_s \cdot t') \quad \dots\dots\dots (6.4a)$$

It is apparent from Fig 6.7 and formulae (6.2) and (6.4) that the travel speed V_s must be varied between passes to maintain production. Referring to formulae (6.2), (6.3) and (6.4) for any given BCE, J and s are constant, SF varies very slightly and the maximum value of T is limited by bucket geometry. Therefore the full production can only be attained when the ladder has been lowered enough to give an adequate length of slice L . Additionally creating the new working face requires near maximum travel speed and depth of cut. This tends to produce larger lumps which impose greater impacts on loading equipment and can cause blockages at transfer points.

The slice section during excavation of part block S_b (Fig 6.7) is of uniform cross section and the bucket filling is:

$$f_p = \frac{T \cdot L \cdot V_s \cdot SF}{J \cdot s} \quad \dots\dots\dots (6.5)$$

Since the cross section is uniform for each pass of the excavator, then V_s will be constant and production is:

$$Q = u (T \cdot L \cdot V_s \cdot 60) \quad \dots\dots\dots (6.6)$$

From formulae (6.5) and (6.6) it is clear that parallel slice operation is much more productive than operations

during the excavation of Sa (Fig 6.7. Also since L remains at its maximum, for a given output T.W is a minimum. Although theoretically the most efficient excavation is obtained when $T = W$, in West Germany it has been found preferable with BCE's performing parallel cuts, to keep T as large as practicable. This reduces the number of passes for a given block to a minimum, thereby reducing crawler and steering gear wear. Additionally steering adjustments can be coarser than if T is small and Vs may also be lower. In practice however Vs cannot be kept too low without special crawler motor cooling arrangements. Generally, T is determined by the method of reversal at the end of each pass, the load on the bucket chain drive and the minimum travel speed.

The excavation of part block width S_c , is very complex and depends to a large extent on whether only every third cut makes a complete arc. For about $\frac{2}{3}$ of the cut it approximates the excavation of part block width S_b . Field tests ⁶³ show that the ratio of slewing cut production to parallel cut production to be approximately 0.7.

Bucket Chain Speeds

BCE chain speeds are limited to the range
1.0 - 1.4 m/sec (200-275 ft/min) since:

1. In addition to providing the digging force to the buckets, the chain lifts the excavated material to the discharge point at the driving tumbler. The power for digging and lifting is determined by the digging resistance (see later) and the density of the material being excavated, and is directly proportional to chain speed.

2. The bucket chain must be supported on guide rails within the ladder and chute, or on the working face with typical friction coefficients:

steel and lignite - 0.15

steel and steel - 0.25 - 0.30

steel and sandy overburden - up to 0.55,

requiring a large power to overcome frictional resistance.

At higher speeds wear becomes excessive especially in overburden.

Conventionally BCE's use 8 sided drive tumblers, although 10 and 12 sided tumblers have been used but have greater weight and the "jerky" action of the 8 sided tumbler improves bucket discharge. As each link engages and disengages with the tumbler, the chain is subjected to periodic acceleration and deceleration. Also, severe load variations occur as each full bucket is changed at the tumbler from linear to rotational motion, and then as the empty bucket changes back to linear motion. A sharp impact occurs at each link as each full bucket engages the drive tumbler. These variations impose pulsations on the tumbler drive and BCE frame. Devices to minimise their effect give satisfactory service, but their frequencies and amplitudes become excessive if higher chain speeds were used ⁶⁵.

Chain speed has another important bearing on bucket discharge, since a high chain speed could result in insufficient discharge time plus the increased centrifugal effect which tends to prevent discharge.

The use of bottomless buckets can give an improved discharge rate ⁶⁵ but practical difficulties have been

experienced particularly in sticky materials.

The various forces acting on the bucket chain greatly influence chain size and weight, and tumbler drive power, these largely determining BCE size and machine mass (service weight).

BWE Output

The BWE was originally developed about 1930 in Germany when it became necessary to excavate lenticular lignite deposits with considerable intercalated sterile inclusions. BCE's with multiple jointed ladders proved fundamentally unsuitable whereas the BWE is inherently suited to selective mining.

Many early BWE's were fitted with crowd action booms, the inner end of the bucket wheel boom being mounted on guide rails within the superstructure, enabling the bucket wheel to be advanced or retracted with a crowding action. Despite the complications of the telescopic conveyor arrangement, travelling counterweight and thrust drive, this type of machine proved more suitable for selective mining and part block operation (Fig 6.3) than early fixed boom BWE's.

Considerable mechanical and structural difficulties however have been experienced with crowd-action booms. Many have been converted to rigid booms and some experienced observers had considered that crowd-action booms would become obsolete. A recently announced 1972 Czechoslovakian machine however incorporates a crowd-action boom. It has a theoretical output of $6600 \text{ m}^3/\text{h}$ ($8500 \text{ yd}^3/\text{h}$), a high-digging capability of 32 m (105 ft) and a deep-digging capability of 6.3 m (21 ft). The excavator is mounted on a hydraulic walking mechanism which reduces ground bearing pressure to half that of conventional crawlers. Some discussion of crowd-action

boom BWE's is therefore indicated.

The Crowd-Action BWE - Upwards-digging

Fig 6.9 shows the upwards-digging, "terrace" cut of a crowd-action BWE, where selective mining is not required i.e. full block operation. The block is excavated in a number of layers of equal height, $h = \text{approx. } 66.7\%$ of the bucket wheel diameter, D , depending on the total face height. At the commencement of the top terrace the bucket wheel is in a retracted position and at the end of each pass it is extended a distance T , the hoist drive being operated to maintain horizontal wheel advance and then slewing reversed. No crawler travel is involved and the axis of slewing remains fixed, therefore the segments excavated are concentric. Fig 6.9 shows a typical segment, with slice thickness T , uniform except at the ends. Although $+ 90^\circ$ slewing is possible, resulting in a greater block width and therefore reduced transport system advancement, it is advantageous to reduce the angle of slew and avoid the reduction in cross section of each segment⁶⁶. It is possible to eliminate these losses however by the use of a variable slew speed drive.

On completion of the top terrace the BWE travels back from the working face, the bucket wheel is retracted and lowered into position for the next terrace. The extent of crawler travel and bucket wheel travel depends upon:

- a) the block width, S
- b) the total face height, and
- c) the dimensions and clearances
of the BWE.

It is apparent therefore that for high faces or when in part block operation (Fig 6.3), the crowd length must be great enough to avoid a mixutre of crowd-action and crawler travel.

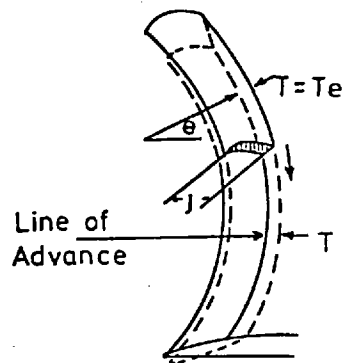
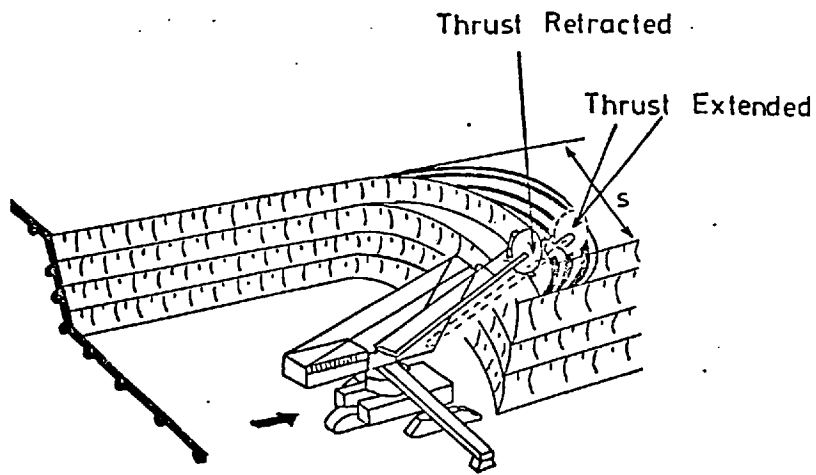


FIG. 6.9 BWE CROWD ACTION BOOM (Terrace Cut)

The Fixed Boom BWE - Upwards-digging

Fig 6.10 shows the terrace cut of a fixed boom (crowdless) BWE digging upwards, normally used where selective mining is not necessary. As in Fig 6.9 a block width S , is being excavated in 4 terraces. The major difference from the crowd-action BWE is that with crowd-action the slew angle may be 90° for each terrace, the highwall slope angle being formed by adjustment of crowd length, while the fixed boom BWE may only slew 90° for the top terrace, the highwall slope angle being formed by reducing the slew angle on the lower terraces. The permissible highwall angle has therefore an important bearing on the block width which can be excavated by a fixed boom BWE.

Fig 6.10 shows that the thickness of cut T , is adjusted by crawler travel of the BWE at the end of each pass of the face, but that adjustment of the boom hoist is not required. Since the slew axis progresses along the line of advance of the BWE by successive distances T , the edges of the segment are not concentric but are "sickle" shaped (Fig 6.10); therefore the thickness of slice is not constant and at an angle of slew θ a close approximation is :*

$$T_\theta = T(\cos \theta + \frac{\sin^2 \theta}{2p}) \quad \dots\dots\dots (6.7)$$

where $p = \frac{R_m}{T}$

and $R_m =$ mean radius from the slew axis to the outer edge of the segment.

*See Appendix 3.A for proof.

Since p is greater than 40, then for practical purposes:

$$T_\theta = T \cos \theta \quad \dots\dots\dots (6.8)$$

Since production may be expressed as the product $T.V_s$,

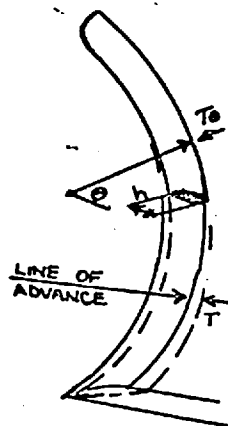
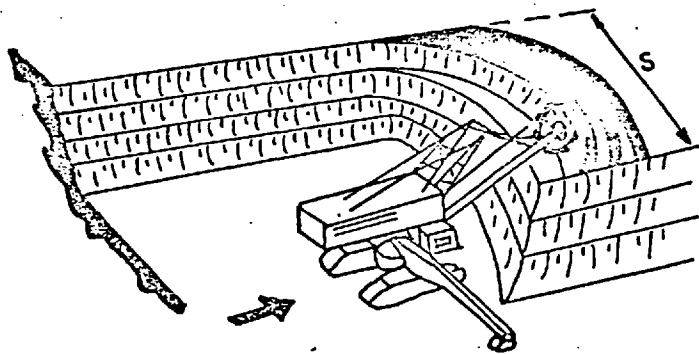


FIG.6.10 FIXED BOOM BWE.
UPWARDS DIGGING,
TERRACE CUT

then for constant output the slewing speed V_s must be gradually increased so that at a slew angle θ from the line of advance, the slewing speed $V_{s\theta}$ should be:

$$V_{s\theta} = \frac{V_s}{\cos \theta} \dots\dots\dots (6.9)**$$

when V_s = slewing speed at the line of advance
i.e. when $\theta = 0^\circ$

Some recent German machines provide automatic speed control of the slewing motion ⁶⁷.

**See formula (6.15) for a more complete treatment, also (6.16) for drop cut output calculations.

Because fixed boom BWE's must adjust the slice thickness T by crawler travel, it follows that they are subject to much greater crawler movement than crowd-action BWE's. When working in softer material therefore, fixed boom BWE's require lower specific ground bearing pressures than crowd-action BWE's.

Part Block (or Lateral) Operation

This method is adopted for selective mining of deposits containing intercalated steriles (Fig 6.3), since it permits separate excavation of bands of material of relatively horizontal banding, and loading out of one class of material for much longer periods than is possible with full block operation (Fig 6.9 and 6.10).

The excavator stands at the side of, and not behind the working face. Since any number of working faces can be prepared (Fig 6.3), changes in the type of material excavated can be made quickly by the BWE travelling between working faces.

The part block method has lower volumetric efficiency than the full block method since the thickness of ground being mined is usually less than optimum, it involves greater

travelling time, with increased crawler wear and reduced digging time, and is therefore restricted to selective mining operations only.

The crowd-action boom is inherently suited to part block operation although long, fixed boom machines are used. Short booms are not suitable for part block operations. If part block operations are proposed for high faces an oversized bucket wheel is necessary to obtain clearance for the boom, if a reasonable block width is to be maintained.

Part block operations may also be used to lower the height of a face to that workable by strip mining ⁶⁸.

Downwards-digging - Terrace Cut

It should be noted that all BWE's including those specifically designed for upwards-digging, have some degree of downwards-digging capability for correction of grade and ramping purposes. Fig 6.11 shows the downwards-digging cut of a fixed boom BWE. The method shown provides maximum terrace height h , and therefore a minimum number of terraces. This requires a side trench to be cut at the inside of each terrace to provide the necessary clearance for the wheel and boom. The boom is set at a fixed slew angle and the BWE travels backwards and forwards parallel to the line of advance, and the wheel lowered after each pass until the required terrace height, is cut. Excavation then proceeds in a similar manner to that for upwards digging. The top terrace height is determined by the BWE dimensions and the clearances required to excavate the lower terrace. After the top terrace is excavated the BWE travels forward, cuts the lower side trench and then excavates the lower terrace in segments as shown. Fig 6.11 shows the segment shape and slice cross section obtained. Slewing up to 90° is not possible when

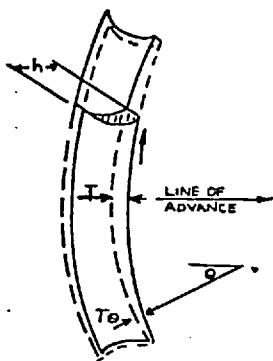
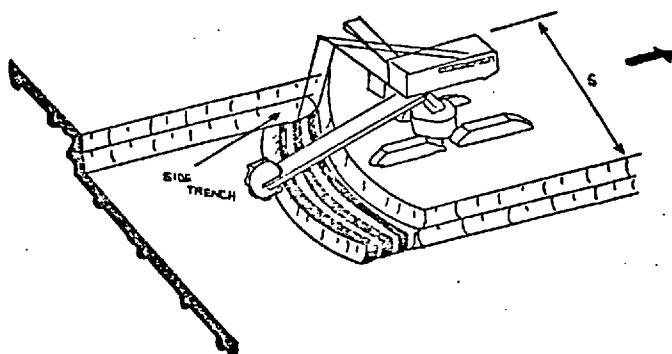


FIG 6.11 FIXED BOOM B.W.E. DOWNWARD
DIGGING TERRACE CUT

more than one terrace is excavated as the slewing radius of the lower terrace is less than the top terrace. Where upwards and downwards digging must work in synchronism to reduce transport system shifting, this is not a disadvantage because the proportions of the downwards terraces and their relation to the transport system are similar to the lower and not the higher, terraces of upwards digging. The block width S, may be kept the same for upwards and downwards digging⁶⁹, though of course the largest block width which may be dug is influenced by the relative lengths and slew angles of the bucket wheel boom and the discharge boom as well as the excavating depth.

This method has the major disadvantage of the need to excavate the side trenches. This can involve up to 10 passes for each terrace, involving complicated crawler travel and reduced output. Although efficient digging is possible after the trench is cut it is often better to avoid this method and use more terraces of reduced height. This alternative is preferable where a soft bench surface requires minimum crawler travel. In this type of operation the slew angle is determined by several factors including the permissible highwall slopes and the BWE proportions, the vertical and horizontal clearances of the bucket wheel being the major influence on terrace height.

The Drop Cut - Upwards-digging

Terrace cut methods are normally used in Europe and the U.S.A., as they are the most efficient for free cutting materials. In the Latrobe Valley lignite deposits of Australia and also in the Athabasca Tar Sand deposits of Canada, conditions are such that the "drop" cut is more satisfactory. The drop cut has some advantages in hard,

abrasive ground, the hoist controls must however be of a more sophisticated nature than for conventional operation, since precise control of the thickness of slice is essential, and this must be engineered during the selection procedure.

At the beginning of the drop cut, a top cut is taken by slewing the boom at maximum face height, so that the terrace formed is about 50% of the bucket wheel diameter (Fig 6.12) to prevent overhang. The BWE then retreats a short distance and the remainder of the block is dug in segments by lowering the wheel to cut slice thickness T , which also involves retreating a distance appropriate to the frontal face slope. Reduction of the slew angle θ , as the bucket wheel is lowered forms the new highwall face. (Note Fig 6.12 shows the highwall face smooth but in fact it has a "saw toothed" appearance).

On completion of the block, a small toe ridge is left. This may be cut by the BWE or by a supporting bulldozer. In practice it is preferable to move this forward a short distance to maintain a toe trough to trap debris from the working face.

Where the face height permits, the top terrace excavation is not taken and the whole block excavated with the drop cut.

The sickle depth variation in the drop cut may be expressed by re-writing (6.7):

$$A_{he} = A_h \left(\cos \theta + \frac{\sin^2 \theta}{2\pi} \right) \dots\dots\dots (6.10)$$

where

A_h = depth of slice normal to the working face at 0° slew angle (Fig 6.13)

$$\pi = \frac{R}{A}$$

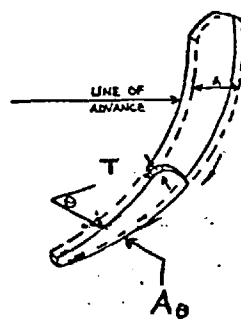
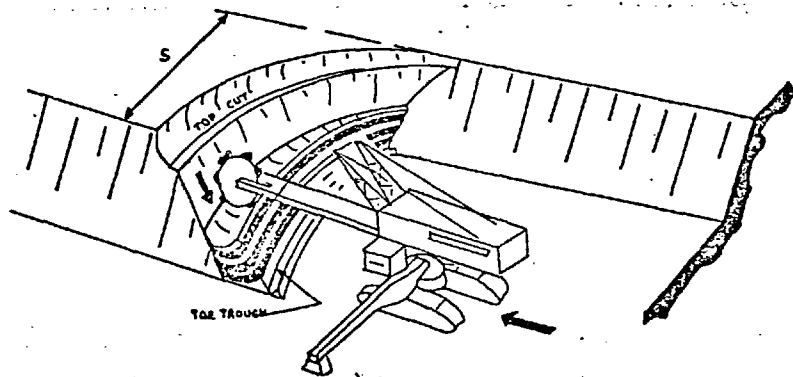


FIG 6.12 FIXED BOOM B.W.E. DROP CUT

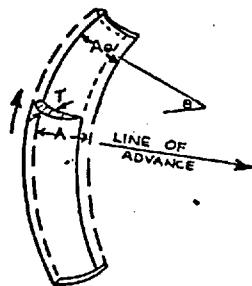
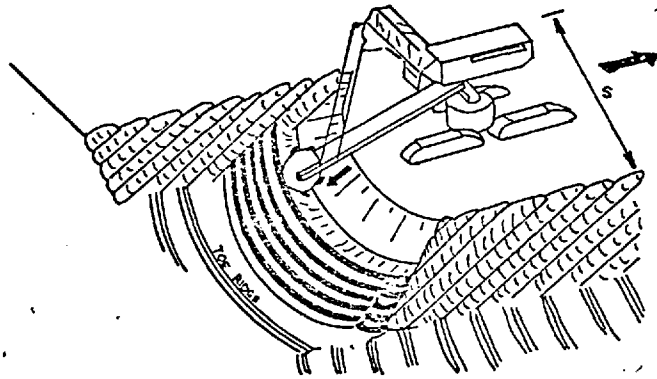


FIG 6.13 B.W.E. DROP CUT DOWNWARDS
DIGGING

R = Radius from slew axis to outer edge of segment

A = Depth of advance = $A_h \operatorname{cosec} \theta$

In (6.10) p_1 is more important than p in (6.7) because its value is approximately 6-8, and a slice depth does not become zero until θ is greater than 90° . The slewing speed range for a fixed boom BWE is effective over a greater angle for the drop cut than for the terrace cut, though an adequate speed range is needed for removal of the top terrace of the drop cut. For BWE's with speed range of 5:1 or more, the cross sectional area $T_\theta . W_\theta$ can be kept constant and A_h decreases as θ increases. Thus if production is to be maintained, $T . W_\theta$ must increase with θ and this increasing slice cross section can lead to the excavation of large lumps.

The Drop Cut - Downwards-digging

Downwards-digging may be carried out using the drop cut (Fig 6.13). As in upwards-digging the bucket wheel centre is moved T, parallel to the working slope for each pass, this being achieved by crawler travel and lowering the wheel. The swing towards the inside of the block is progressively reduced to provide adequate wheel clearance and to establish the new highwall slope. The block depth A is selected so that A_h normal to the working face at the line of advance is about $\frac{2}{3} D$. The new highwall shows a ridged appearance and as in upwards-digging a ridge is left at the toe of the face. If the lower bench is not required for operations and no drainage problems are present, the toe ridges can be left behind. If they must be removed the BWE can excavate them but with loss of output.

Where a cover-belt is used over the boom conveyor for

downwards-digging (see later), the drop cut is probably more suitable as it does not produce large lumps.

The segment shape (Fig 6.13), like downwards-digging with a terrace cut tends to be more symmetrical than in the upper segments of upwards-digging. Since the bucket wheel is above the face, a top terrace is not necessary as in upwards-digging and the whole block can be excavated using a drop cut.

Terrace versus Drop Cut

Fig 6.14 shows the terrace cut, the radial thickness of slice starting at zero, increasing to T at half the bucket wheel height, then reducing slightly. The horizontal thickness however remains T (Fig 6.14) and the vertical cross section of slice is $h.T$

$$\text{The width of slice, } W = \frac{V_s}{s}$$

where V_s = slew speed

s = discharges per minute

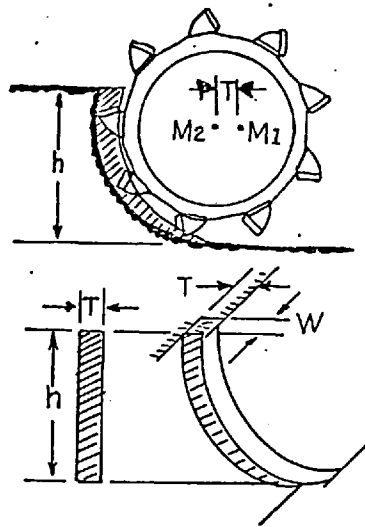
therefore the solid volume cut by each bucket is:

$$C_t = h.T \frac{V_s}{s} \dots\dots\dots (6.11)$$

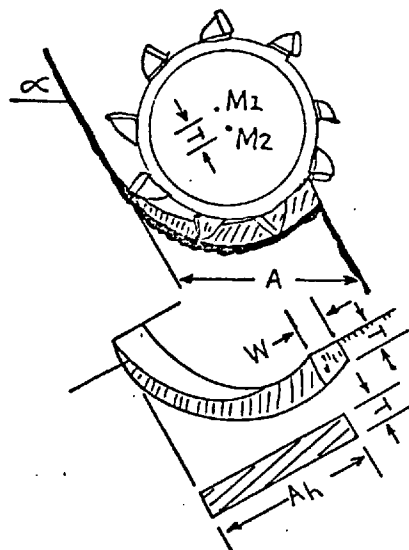
Fig 6.14 shows a drop cut excavating a highwall of slope α . This involves moving the bucket wheel centre successively by distance T i.e. M_1 to M_2 , etc., parallel to the face of the highwall, all the excavation taking place below the bucket wheel centre. The block depth A is usually determined by $A_h = \frac{2}{3}$ bucket wheel diameter

$$\text{where } A_h = A \sin \alpha.$$

Fig 6.13 shows the bucket slice shape and since T is constant when measured parallel to the slope, the cross sectional area is $A_h.T$.



TERRACE CUT



DROP CUT

FIG.6.14 BUCKET SLICE CROSS SECTIONS
AT LINE OF ADVANCE

The solid volume cut by each bucket is then:

$$C_d = A_h \cdot T \cdot \frac{V_s}{s} \quad \dots\dots\dots (6.12)$$

Since A_h and h would normally be $\frac{2}{3}$ bucket wheel diameter, then:

$$C_t = C_d = \frac{2 \cdot D \cdot T \cdot V_s}{3 \cdot s} \quad \dots\dots\dots (6.13)$$

The bucket wheel cutting action is more complex than indicated above e.g. it will be apparent that the distance of each bucket from the slew axis varies as the bucket wheel rotates and the instantaneous slewing speed of each bucket is directly proportional to its distance from the slew axis. Hence neither for terrace or drop cuts is $W(\text{ie. } \frac{V_s}{s})$ uniform. Also formulae (6.8) and Fig 6.10 show that as the bucket wheel moves away from the line of advance, the slice thickness T , decreases for a terrace cut. As previously described, for the drop cut the slice depth A_h , decreases as ϕ increases. For comparison purposes however the simplification used is sufficiently accurate. For a fuller discussion of slice and segment details see Appendix 6.A.

Formula (6.13) indicates that for the same slew speed and cut thickness, the volume per bucket and therefore output is equal for terrace and drop cuts. Output however also depends on utilisation of the bucket wheel drive power which may be derived as follows:

- a) transmission losses
- b) separation of material from working face
- c) elevate material to discharge point.

For a given bucket wheel speed and power input it is obvious that transmission losses are independent of the type of operation. It is also obvious that the power to elevate the material is lower with the terrace cut. Hence more cutting power will be available with the terrace cut and in homogeneous material higher output may be expected. e.g. German Rhineland brown coal. In many weakly consolidated deposits however, suitable for BWE operation, there are relatively horizontal bedding planes and the horizontal digging resistance is usually much lower than the vertical digging resistance (i.e. that normal to the bedding planes) e.g. Latrobe Valley brown coal, Ptolemais lignite, Athabasca tar sands. Also the horizontal bedding may be widely spaced so that the terrace cut can produce large blocky lumps (up to 1 tonne at Athabasca and Latrobe Valley, and 0.75 tonne at Ptolemais) due to the tendency to "lever" off large blocks at the top of the working face. The occurrence of large lumps results in chute blockage, impact damage to conveyor idlers and belting. A further serious effect is the severe fluctuation of force which occurs as the bucket clears the face, the force being approximately normal to the bucket wheel boom and at maximum distance from the slew axis, generating vibration and causing a fluctuation in bucket wheel load.

In the drop cut, the main cutting force acts almost parallel to the bucket wheel boom and therefore the vibration effect is not so serious as for the terrace cut.

It should be noted that crawler travel using the drop cut is much less than when using the terrace cut (Fig 6.12)

BWE Output

The output of a BWE can be expressed by (6.3) but as the slice cross section does not remain uniform bucket filling is more liable to variation than for a BCE. It is preferable therefore to express output as the product of the segment cross section and the speed of sideways movement. Hence the instantaneous output of a BWE using a terrace cut is:

$$Q' = h.T_{\theta}.V_{s\theta}.60 \quad \dots\dots\dots (6.14)$$

With a fixed boom BWE, T_{θ} is variable and $V_{s\theta}$ may also be variable, hence the hourly output over a number of segments is:

$$Q = u \left(\frac{\sum V.60}{\sum t'} \right) \quad \dots\dots\dots (6.14a)$$

where V = segment volume

t' = cutting time in minutes
for each segment

For a crowd-action BWE, h and T_{θ} remain constant except at the ends of the segment and consequently output can be maintained with V_s constant (and of relatively low value).

As shown in Fig 6.10, T_{θ} and hence the segment volume reduces rapidly as the slew angle approaches 90° resulting in a marked decrease in output. This may be shown as follows:

The block width S is:

$$S = R(\sin \theta_L + \sin \theta_R) \quad \dots\dots\dots (6.15)*$$

where

R = radius of segment from slew axis
to top of terrace

θ_L = angle of slew to left of line of
advance

θ_R = angle of slew to right of angle of
advance

*Block width remains constant throughout the whole height of the block although θ_L and θ_R vary. This variation is caused by the highwall slopes as well as variation in distance from slew axis to various parts of the cutting circle. The slewing radius is also altered when the boom is lowered for the next terrace.

The volume per segment is:

$$V = h.T.S. \quad \dots\dots\dots (6.16)$$

Referring to Fig 6.10 when θ_L for the top terrace is increased from 80° to 90° , the block width increases by:

$$R(\sin 90^\circ - \sin 80^\circ) = 0.015R$$

and the length of segment arc increases by:

$$\frac{\pi R(90-80)}{180} = 0.175R$$

i.e over this range the length of arc and consequently slewing time is nearly 12 times the increase in block width, which represents volume.

The proportion of arc length to block width from 0° to 80° is only 1.42, and since V_s is limited by practical considerations, output tends to drop as θ approaches 90° . In practice the slewing angle for the top terrace is usually limited to 80° on the new highwall side and 45° - 50° on the old highwall side.

For the drop cut, output in the top section is the same as for the terrace cut. In the remainder of the block, whether the BWE has a crowd-action or fixed boom, the segment is sickle shaped and hence V_s must be varied to maintain constant output. For the drop cut the hourly output may be expressed as in (6.14) and (6.14a) i.e.

Instantaneous output is:

$$Q' = A_h \theta . T . V_{se} . 60 \quad \dots\dots\dots (6.17)$$

and the effective hourly output is:

$$Q = u \frac{\sum V_d \cdot 60}{\sum t'} \dots\dots\dots (6.17a)$$

where

V_d = segment volume of drop cut.

It is apparent from formula (6.10) that for the drop cut, output does not diminish after maximum slewing speed has been reached to the same extent as it does for the terrace cut. It is still advantageous however to slew to less than 90° . Also as the slice length decreases as θ increases, this can result in the excavation of large lumps.

MECHANICAL DESIGN FEATURES

Bucket Wheel Design can have the following construction:

1. With cells i.e. with individual chutes from the buckets to the transfer point supplying the boom conveyor
2. with half cells
3. cell-less

Fig 6.15 shows the trend in bucket wheel speeds since 1940. Most modern excavators have two or more bucket wheel speeds so that the "high" and "low" speeds have been plotted. Up till 1950 it will be noted that the high speeds of German BWE's were fairly constant at about 2 m/sec (400 ft/min). Earlier bucket wheel designs appear to have been based on BCE practice⁶¹⁰, which is subject to high frictional, elevating and pulsating loads not present in the BWE. Following this a marked increase in speeds occurred, probably stimulated by the Kolbe wheel developments in the 1940's⁶⁸. German bucket wheels now appear to have approached a practical limit of 5 m/sec (1000 ft/min) due to centrifugal force on

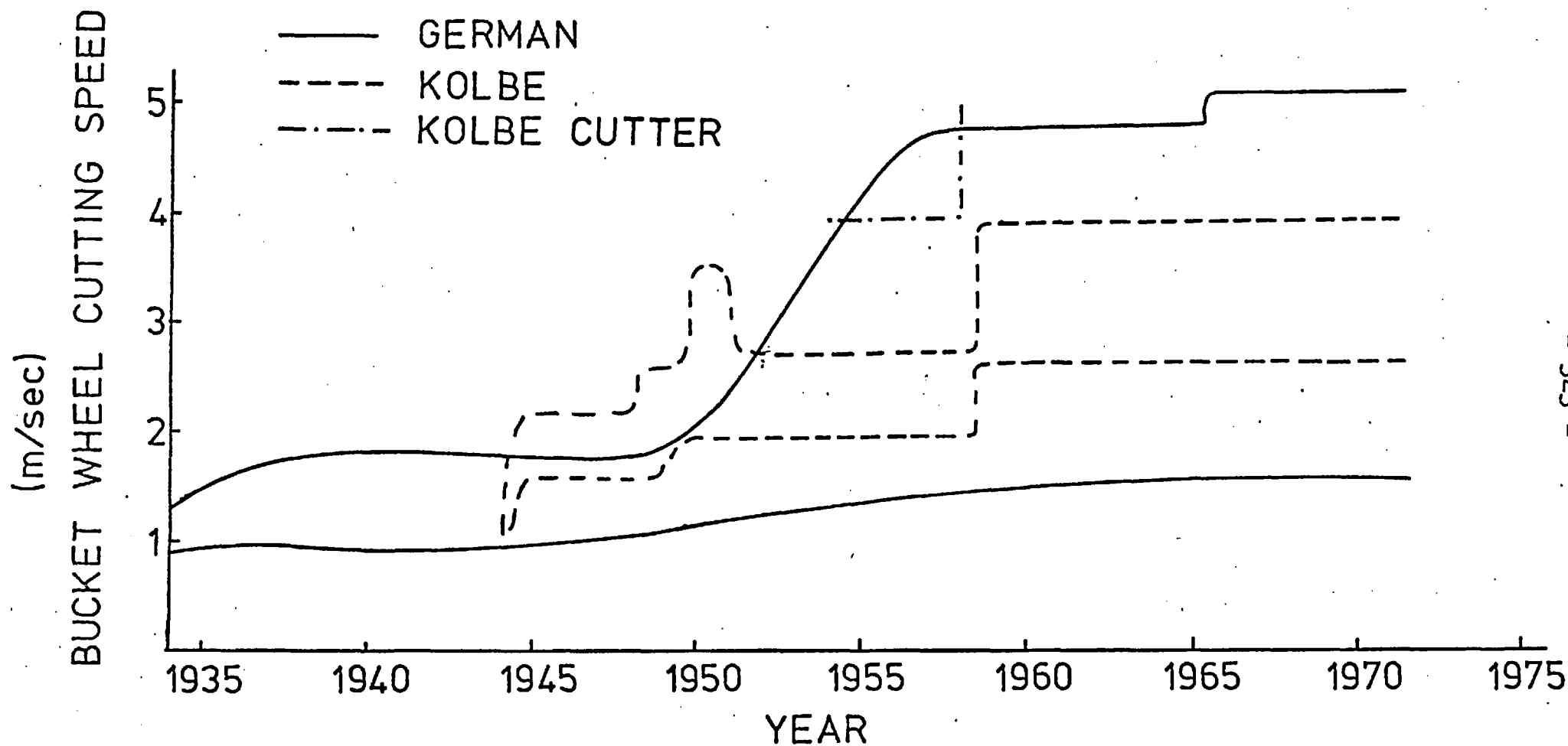
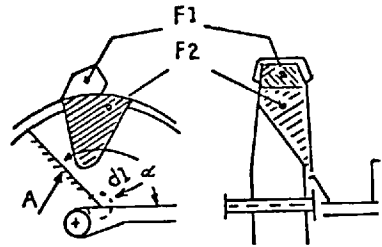


FIG 6.15 BUCKET WHEEL CUTTING SPEEDS

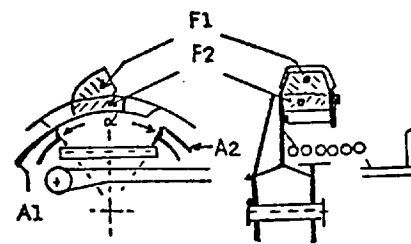
the bucket contents and wear on the cutting edges (see later).

The development of the Kolbe machine is also shown on Fig 6.15. These BWE's were developed for United Electric's coal operations in the U.S.A. The first machine, with a celled wheel, was commissioned in 1943 and had a bucket wheel speed of only 1.05 m/sec (207 ft/min). A second BWE was commissioned in 1948, with a 7.6 m (25ft) diameter bucket wheel and a bucket wheel cutting speed of 2.5 m/sec (500 ft/min). In 1950, this bucket wheel was replaced by a 6 m (20ft) diameter wheel which operated at a wheel cutting speed of 3.5 m/sec (700 ft/min) for a short period, but as satisfactory performance was possible at a lower speed with much reduced shock loading the speed was reduced to 2.7 m/sec (530 ft/min). In 1959 Kolbe BWE (W.4) was commissioned with a 8.25m (27 ft) diameter bucket wheel with a speed range of 2.6 - 3.85 m/sec (510-760 ft/min). In 1954 United Electric also introduced their W3 BWE with a "bucketless" wheel (Kolbe cutter) (see later). Also a BWE with a bucket wheel speed range of 2.2 - 3 m/sec (435 - 600 ft/min) was commissioned in 1956 by Truax Traer Coal Co. in the U.S.A. ⁶¹¹.

Excavator productivity is primarily dependent on excavating rate and discharging rate. With the BWE the latter has been the limiting factor, except in granular and other free-flowing materials, up till the introduction of the cell-less wheel, which was able to effectively discharge sticky materials at high bucket wheel cutting speeds. The significance of this will be more apparent from an analysis of the characteristics of bucket wheels with and without cells. Fig 6.16i, shows the celled bucket wheel with a chute down which excavated material slides onto the boom conveyor belt. Material



(i) Celled wheel



(ii) Celless wheel

FIG 6.16 BUCKET WHEEL TYPES

collected on the up-run of the bucket is prevented from falling by cut-off plate A, until the bucket and cell approach the vertical. Discharge must occur within α . Consequently the bucket wheel speed must be slow enough to allow adequate discharge time. To improve discharge the backs of the cells are not radial but tangential to circle, diameter d_1 .

When digging free flowing material discharge is satisfactory, with a high bucket filling. With sticky material however, the material can adhere to the inner surface of buckets and cells, greatly reducing flow of material through the wheel. This can result in a considerable volume of excavated material being "carried over" and dumped on the working bench.

Fig 6.16ii shows the cell-less wheel, allowing free fall of the excavated material when discharging onto a roller table, rotating disc or other positive transfer method to the boom conveyor belt. The cut off plate A for the celled wheel is replaced by ring chutes A1 and A2.

It will be noted (Fig 6.16) that although the buckets, F1, may be identical, the volume of the cell F2, is much greater than the volume of the ring chute F2. Some caution is therefore necessary when comparing theoretical (and guaranteed) outputs of similar machines as manufacturers of different nationality may define F, the bucket capacity differently. Where only the bucket volume F1 is used a celled wheel has no difficulty in achieving its output in free flowing material. For cell-less bucket wheels however F2 is much less than for celled wheels, and if the bucket capacity F is given as F1, much closer attention is required to the filling and utilisation factors of cell-less buckets.

In addition, for cell-less buckets, some manufacturers may add 50% or 100% of the associated ring volume; or for celled wheels, the cell volume. The method of specifying the bucket capacity must therefore be clearly defined for selection purposes.

It appears that high cutting speeds are possible with cell-less wheels because of the direct discharge method. The limiting bucket wheel cutting speed is that at which centrifugal force is equal to the gravitational force on the bucket contents, i.e.:

$$V_{lim} = \sqrt{g \frac{D}{2}} \quad \dots\dots\dots (6.18)$$

In practice speeds approaching V_{lim} are not possible, because in some cell-less bucket wheels the discharge time would be too short and because excessive wear would occur. At present therefore cutting speeds are limited to about $0.7 V_{lim}$ or 5 m/sec (1000 ft/min) whichever be the smaller.

The introduction of cell-less bucket wheels and also very large wheels appears to have resulted in an increase in permissible cutting speeds for celled bucket wheels. Current literature ^{63, 66, 69} indicates that the maximum cutting speed for celled wheels should be 0.4 to 0.5 V_{lim} , when excavating medium-heavy, non-sticky materials. The celled wheel is not normally selected for sticky materials.

The Kolbe "bucket-less" or cutter wheel uses the centrifugal force to assist the discharge process. The wheel is 5m (16.5 ft) diameter and operates at 49 m/sec (960 ft/min). The spoil is carried up a small curved chute as it leaves the face and is thrown onto the bucket wheel conveyor. The slice dug by the cutter wheel is a downward terrace cut (reverse cutting direction).

The celled bucket wheel must be of sufficiently large diameter to accommodate the cells, but this cell space is not required for cell-less bucket wheels, so the same number and size of buckets may be fitted to a smaller wheel. This can reduce excavator weight but it must be kept in mind (see formulae (6.14 and 6.17)) that a reduction in wheel diameter D , reduces h or A_h , and this would have to be compensated for by an increase in slewing speed V_s , if output is to be maintained constant. It then follows that an increase in discharges per minute s , cannot lead to an increase in output unless V_s is also increased.

The Kolbe BWE's were first to adopt higher wheel speeds. These had crowd-action booms in a terrace cut operation. As previously stated this operation requires a comparatively slow slewing speed and consequently, an increase in bucket wheel cutting speed readily gave better use of the slewing speed range and hence increased output. Also modern fixed boom BWE's working in terrace or drop cut and designed for high wheel speeds are able to use a higher slewing speed in the thicker part of the segment and, although bucket filling tends to be poor towards the ends of the segment, the time to complete a pass is reduced and consequently production is higher than with lower bucket wheel cutting speeds.

Another advantage of the higher speeds of the Kolbe bucket wheels is that the period between cutting and discharge is small and this together with the low filling factor, combined with flexible, chain-backed buckets, enables sticky

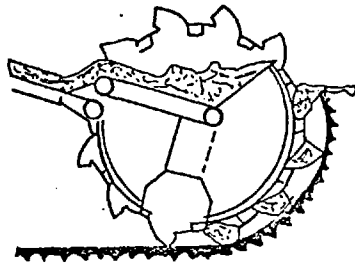
overburden to "float" rather than clog the wheel.

Higher bucket wheel cutting speeds also allow the same output to be maintained with a given wheel speed and slewing range, but with a reduced chip size. This can be useful for excavating hard deposits (see later). Higher bucket wheel cutting speeds may also provide the same output with smaller buckets but a number of other factors; e.g. digging height or depth and side clearances affect wheel diameter (see later).

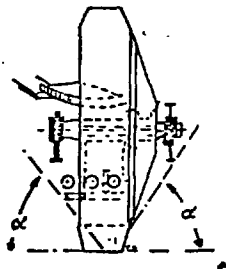
Transfer of material from the cell-less bucket wheel may be achieved by one of the following devices:

1. Inclined steel plate. This may be heated, to improve the flow of sticky materials. Also a large diameter roller may be located at its lower edge to reduce impacts on the boom conveyor idlers and to centralise loading (small machines).
2. Roller table (Fig 6.16) (medium machines)
3. Short feeder belt (Fig 6.17) (medium-large machines)
4. Rotary disc feeder (small-medium machines).

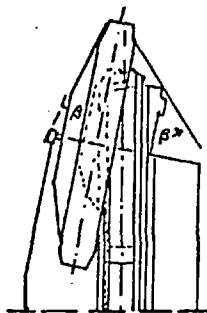
Since in the cell-less wheel material may fall directly on to the transfer device the dumping height for the buckets can be kept to a minimum. This allows both the transfer device and the boom conveyor to be mounted higher than would be possible with a celled bucket wheel (Fig 6.17). This in turn increases the vertical clearance of the bucket wheel boom and may increase the digging height for the same block width and boom diameter. Another feature of this arrangement is the possibility to mount the bucket wheel



(i)



(ii)



(iii)

FIG 6.17 ARRANGEMENT OF FEEDER BELT
AND BUCKET WHEEL HEAD GEOMETRY

drive within the bucket wheel, improving the vertical free cutting angle α , as shown in Fig 6.17ii. It should be noted that the higher the mounting the greater the improvement in α ⁶¹². Fig 6.17iii shows the plan view of the offset bucket wheel, facilitating the entry of the feeder belt. Together with the location of the drive within the bucket wheel, providing an improvement in the lateral fall cutting angle β . High values of α and small values of β are essential where highwall slope angles must be kept relatively flat for stability.

Where cohesive material is handled there is a tendency for this to compact in the ring volume of cell-less bucket wheels. For large bucket wheels with ample space, the half cell arrangement has been found most suitable and spillage due to "carry-over" has been found to be negligible handling cohesive overburden in the German Rhineland⁶⁶.

The BWE has an inherently poor downwards-digging capability, but this can be improved by a boom sandwich belt or a special long boom design which reduces the boom angle sufficiently to allow successful conveyor operation when digging below grade with reversed buckets (Fig 6.18).

For a sandwich belt the forces resisting slipping on the top surface of the material are:

$$(P + G_1)(\cos \alpha) \mu$$

and on the lower surface

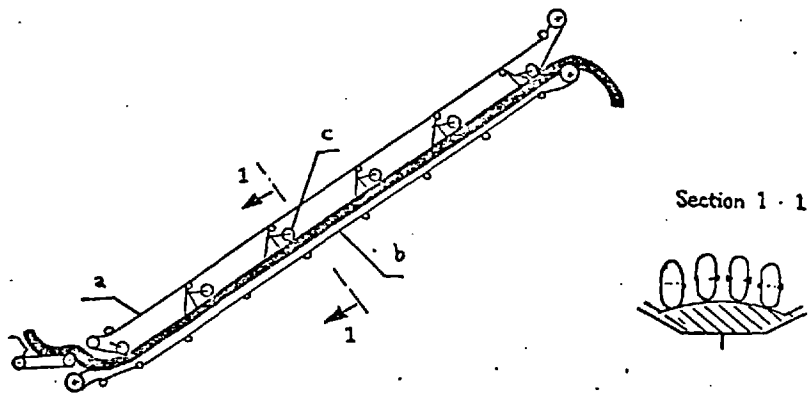
$$(P + G_1)(\cos \alpha + Q \cos \alpha) \mu$$

and the force tending to cause slipping is:

$$Q \sin \alpha$$

Slippage would then occur when:

$$Q \sin \alpha > (2(P + G_1) - Q) \mu \cos \alpha$$



- a- cover belt
- b- conveyor belt
- c- loaded idlers

FIG. 6.18 COVER BELT CONVEYOR FOR STEEP ANGLE BUCKET WHEEL BOOM

Hence the maximum conveyor inclination should be limited to:

$$\alpha < \tan^{-1} \frac{(2(P + G_1) + 1)\mu}{Q} \dots\dots\dots(6.19)$$

where

P = average vertical load per unit
length imposed on the cover belt
by the loaded rollers

Q = mass per unit length of the material
being conveyed

G₁ = mass per unit length of the cover
belt

μ = coefficient of friction between the
material being conveyed and the
belting during operation

In (6.19) the same coefficient of friction is assumed between the material being conveyed and both cover belt and conveyor belt. Where different materials are used for the two belts the maximum value of α would be calculated using the two values of the coefficients of friction. Modern machines are usually fitted with identical belting materials, chain woven cover belts being relatively obsolete.

Sandwich belts do not usually give satisfactory service where boulders or large frozen lumps are being handled.

Although values given in the literature vary, recent experience with 30° conveyor troughing shows that reliable conveying without a cover belt is possible at 23°. This has led to the design of BWE's with high-deep digging ratios of 3 : 1 for service in the Rhineland brown coal mines. These have a greater machine mass (service weight) than equivalent cover-belt BWE's due to the longer boom and larger

counterweight arrangement but this is compensated by increased block width possibilities, simplicity and better performance where lumpy or sticky materials are excavated.

A number of approaches have been used to the handling of sticky materials. Chain backed buckets tend to flex in operation and have a superior clearing action to solid buckets, but in the Latrobe Valley "Linatex" rubber linings have proved more successful ⁶²⁰. Cementing the Linatex directly to the bucket proved ineffective but by simply clamping it at its edges so that it can flex proved successful. The Linatex is reinforced by cementing canvas to the back. At one bauxite operation with very sticky overburden, material flow has been greatly improved in those parts of the buckets and wheel not subject to scouring action by the fitting of magnesium liner plates ⁶²¹. Attempts have been made to heat the buckets electrically without marked success but Kolbe ⁶⁸, reports improvements, using flame jets fired by LPG or fuel oil, for both frozen and sticky ground.

There are numerous bucket teeth configurations and some trial and error may be necessary before a satisfactory arrangement is found. In strong, highly abrasive material, such as the Cuddalore sandstone at Neyveli, South India, tungsten-carbide inserts with welded, hard-facing of the parent metal may be necessary ⁶²².

Bucket Wheel Drives

In variable ground it is necessary on occasions to adjust the bucket wheel speed (and slewing speed) to meet the digging requirements and to avoid excess vibration in hard ground.

This may be achieved as follows:

- (1) Gearbox drive with selector dog clutch giving
two forward and two reverse (for downwards digging)

speeds (ac induction motor drive). Because of the physical space requirements this drive is usually restricted to small- medium machines.

- (2) Epicyclic gearing with a pony motor giving two forward and two reverse speeds (ac induction motor drive).
- (3) Variable-speed dc drive with a Ward Leonard or a thyristor system. This method provides stepless control, but the maintenance problems of the dc machines are inherently more difficult than those of ac machines.
- (4) Hydrostatic drives - these offer stepless control and have versatile speed ~ torque characteristics. They are less efficient than electrical drives and have been restricted to smaller machines.

All bucket wheel drives are fitted with some mechanical form of overload protection which allows the motor and preferably the gears to continue to rotate if the bucket wheel is suddenly stalled due to the presence of boulders or other obstructions. It is essential that the rotor of the motor be disconnected since its inertia when referred to the bucket wheel is very high, since it is increased by:

$$\left(\frac{\text{rotor speed}}{\text{Bucket wheel speed}} \right)^2 .$$

Among the types of protection used are:

- (a) Friction clutches used to find general application, but are inconsistent in performance.

- (b) Hydraulic couplings are finding greater application. The duration of slip must be limited to avoid over-heating. Normally, they must be fitted on the high-speed (low-torque side) of the drive gears because of space requirements and the high cost of slow-speed, high torque couplings. The gears are not therefore fully protected from damage by the kinetic energy stored in them, and this form of protection is not ideally suited to ground where boulders are present. The fluid coupling effectively protects the drive against shock loads and provides smooth starting but its speed~torque characteristics are such that it cannot accurately cut-off at overload. Because of this the drive may be fitted with some device e.g. a load cell, which accurately measures the motor torque and shuts down the motor at a pre-determined overload. Alternatively the fluid coupling may be used in conjunction with (a) the friction clutch.
- (c) Mechanical couplings, such as the 'Metalluk' unit - a centrifugal coupling with steel balls.
- (d) Electromagnetic couplings also suffer from overheating problems for long periods of slip.
- (e) The use of dc motor drives with specially designed speed~torque characteristics, as used for large single-bucket excavator drives.

Crawlers

As continuous excavators usually operate in less consolidated formations, it is often necessary to limit the ground bearing pressure to a low value, and rigid crawlers such as those used on shovels should only be used for very small machines up to about 50 t (55 short tons) machine mass (Fig 6.19(a)). With this form of construction it is economically possible to limit the average ground bearing pressure to 60 kN/m^2 (8.5 lbf/in^2).

Up to 150 t (165 short tons) machine mass, crawler tracks with two wheel bogies (Fig 6.19(b)) should be used so that the crawlers adapt at least partially to uneven terrain. Larger machines generally require the type of crawler design shown in Fig 6.19(c).

The larger the excavator and the crawler bearing area, the greater may be the imposed ground bearing pressure since a large pressure cone is developed beneath the crawlers and average ground bearing pressures between 150 and 160 kN/m^2 (21 and 23 lbf/in^2) can be used. Large spreaders with the crawler arrangement in Fig 6.19 (c) working on unconsolidated spoil in the German Rhineland impose bearing pressures as high as 120 kN/m^2 (17 lbf/in^2). With crawlers of this construction tests show that the maximum ground bearing pressure is about 1.4 times the average figure. For the rigid construction shown in Fig 6.19(a) the maximum figure can be up to three times the average figure.

Twin crawler machines are steered by running the inside crawler at reduced speed or by braking it. Triple crawler drives are steered by one or more of the crawlers being turned by a steering gear mechanism. In either case each crawler must slew against the ground while the BWE travels in a curve and

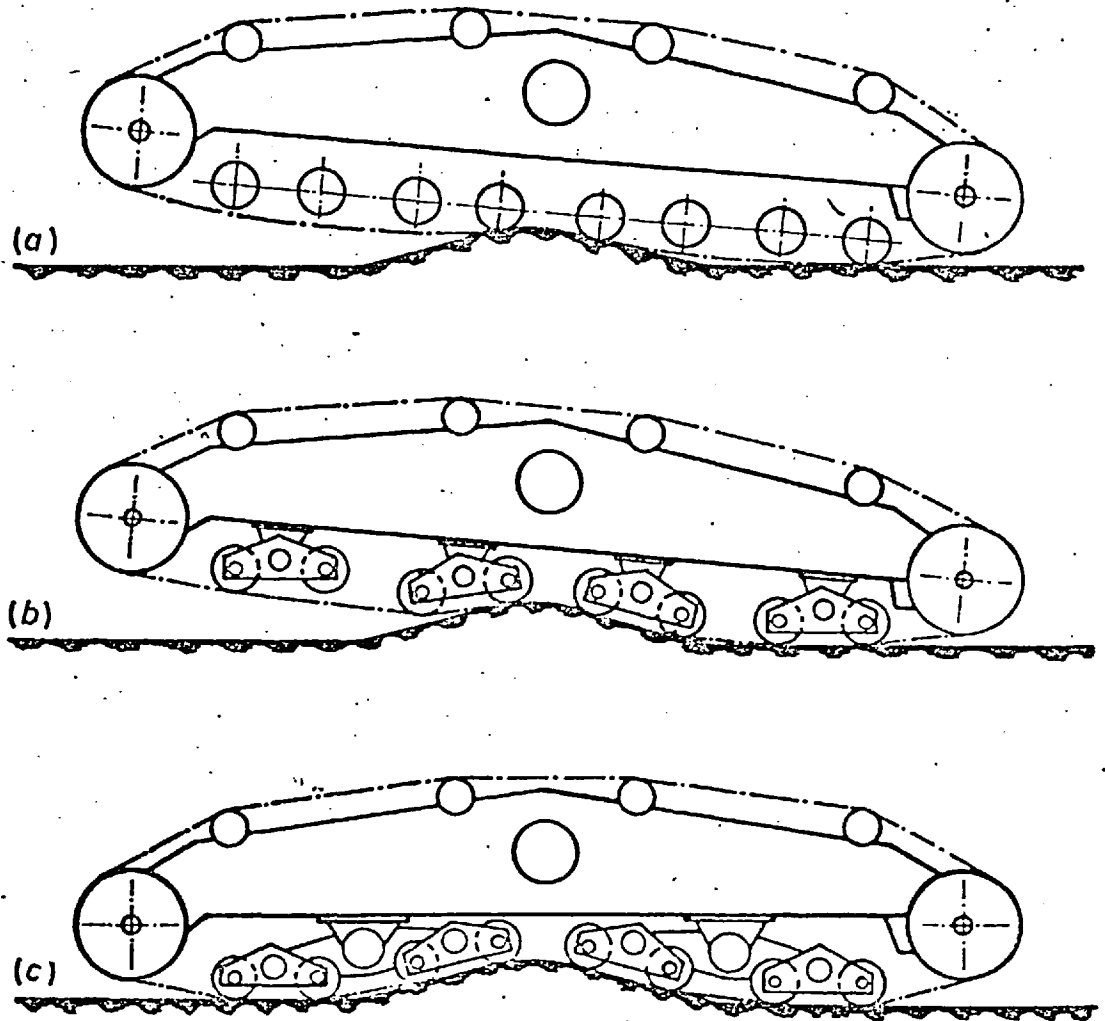


FIG. 6.19 CONTINUOUS EXCAVATOR CRAWLERS

the crawler must be constructed to withstand the heavy transverse forces involved.

SPECIFIC CUTTING FORCE (BWE)

The bucket-wheel drive power, the mechanical strength of the bucket and machine mass (service weight) depend on the digging resistance of the material being excavated. Investigations in Germany, Czechoslovakia and the U.S.S.R. have so far failed to produce a uniform quantitative approach to digging resistance. The relationships between digging resistance, intact rock strength, jointing, bedding, etc., tooth shape and sharpness, angle of attack and wedge angle of the bucket cutting lip have yet to be determined.

A specific cutting force (k) (expressed in kp/cm of bucket cutting lip edge) is widely adopted. For machine design, however most engineers prefer to use the 'specific digging force' (expressed in kp/cm^2 of bucket slice area).

Spade Analogy. If a slice 10cm x 10cm is dug with a spade, the cutting length L is $10 + 10 = 20\text{cm}$, and the area F is $10 \times 10 = 100 \text{ cm}^2$. For 1000 kp cutting force the specific cutting force k is:

$$\frac{1000}{20} = 50 \text{ kp/cm}$$

and the specific digging force k^1 is:

$$\frac{1000}{10 \times 10} = 10 \text{ kp/cm}^2$$

If the slice size is increased so that the length of each edge is 20 cm, then the cutting length L is $20 + 20 = 40 \text{ cm}$ and the slice area is $20 \times 20 = 400 \text{ cm}^2$.

For 4000 kp cutting force then:

$$k = \frac{4000}{40} = 100 \text{ kp/cm and}$$

$$k^1 = \frac{4000}{400} = 10 \text{ kp/cm}^2$$

With the same ratio of cutting force to cutting area as above, the specific cutting force k , has theoretically doubled. Practical measurements show of course that this is not so and for this reason most engineers prefer to use specific digging force k^1 , referred to the area of slice.

Table 6.II suggests values for specific cutting and digging forces.

TABLE 6.II

SPECIFIC CUTTING AND DIGGING FORCE (BWE)

Ref- erence	Material	Kp/cm cutting lip	Kp/cm ² slice area
613	Light ground-soft, loose, soil of low cohesion which can be parted with a hand shovel, e.g. unconsolidated sand or fine gravel	15-50	2-5
613	Medium ground-soil of medium to strong cohesion which can be parted with a spade e.g. sandy clay, soft and plastic clay, coarse gravel, loam and loess	30-100	5-8
613	Heavy ground-strong to very strong cohesion, which must be parted with a pick	60-120	8-10
614	Compact sands	33-103	
614	Chalk	26-113	
614	Clay (soft)	9-13	
614	Clay (hard)	40-160	
614	Sandstone	14-142	
614	Slate clay	200+	
615	Sandy clay	20	
615	Sand and clay	29	
615	Sand and loam	33	
615	Greasy clay	56	
616	Argillaceous shale	156	23.8
616	Compact clay	86	7.82
616	10% sand + 90% clay	86	5.91
616	Sandy clay and loam	72	5.67
617	Light ground		1.8-2.5
617	Medium ground		3.0-3.5
617	Heavy ground		7-18

There is still much to be done in this field in relating conventional rock properties to specific digging force. Careful investigation is essential. If an insufficiently robust machine is designed, structural and drive failures will occur, but overdesign can, of course be costly. The

large capital investments required for continuous excavators warrant detailed study of similar operations and comprehensive discussions with manufacturers before the limits of specific digging force are specified.

HARD GROUND OPERATION

Large modern BWE's with high bucket wheel cutting speeds and slower slewing speed, producing a smaller chip size can mine materials that cannot be mined by shovels without blasting, e.g. hard clays, shales, phosphates, sandstones and frozen oil sands. This is only possible with machines of sufficient mass to transfer the cutting forces through the crawlers to the ground. Provided that the rock does not contain large boulders which will not pass through the bucket wheel, ground with a uniaxial compressive strength of $1.5 - 1.8 \text{ MN/m}^2$ ($2150-2550 \text{ lbf/in}^2$) can be excavated without ground preparation (e.g. Neyveli, South India; Athabasca Tar Sands, Canada; and several U.S.A. coal stripping operations). Fig 6.20 shows a large BWE digging frozen oil sands at Athabasca.

The production rate of the BWE operating in hard ground is usually much less than the possible theoretical output because of loading difficulties and flow difficulties through the bucket wheel. Some attempts have been made to excavate stronger rocks subjected to blasting but with mixed success. The sandstone overburden at Neyveli gave promising results at first, but it does not appear that the proposed blasting development programme was completed and results are inconclusive.



FIG.20a BWE FOR HARD GROUND OPERATION,
ATHABASCA TAR SANDS



FIG.20b BWE HANDLING LARGE LUMPS IN HARD
GROUND, ATHABASCA

The major requirement is to obtain uniformly fragmented ground otherwise considerable difficulty is experienced owing to the breaking out of the face and spilling of large lumps.

The drop cut has some advantages in hard, abrasive ground. The buckets excavate the full slice width at the beginning of each cut. In horizontally stratified, hard ground, digging produces a smaller chip size which parts more easily as previously discussed, also the teeth enter the bank at approximately 90° . It is necessary however to be able to adjust the thickness of slice with some precision since too thin a slice reduces output and too thick a slice overloads the bucket wheel drive as well as producing large lumps. For a fixed boom machine using a terrace cut this must be done by adjustment of crawler travel. For a drop cut the thickness of slice is adjusted by lowering the bucket wheel boom, which lends itself to a much higher degree of precision (Fig 6.21).

Because of the desirable features of continuous excavators, attempts will undoubtedly be made to extend their field of use in hard ground. Present opinion in West Germany is that blasting does not yet provide the answer to the problems involved but the author believes that based on the experience gained at Neyveli and the advent of low cost explosives that this is a fruitful field for further development.

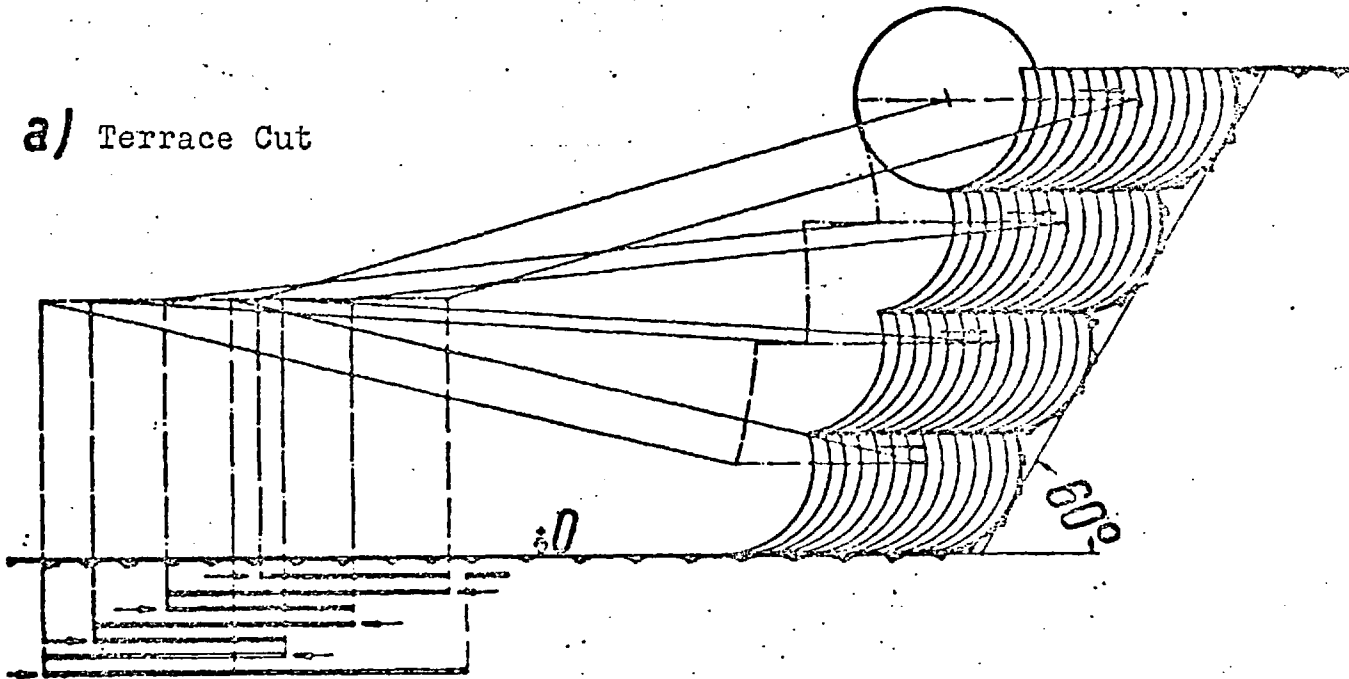
EXCAVATOR QUALITY COEFFICIENT

The following index is used for comparing different classes of excavator:

$$G_v = \frac{\text{Excavator Mass}}{\text{Maximum cutting height} \times \text{hourly output} \times \text{overall length}}$$

..... (6.20)

a) Terrace Cut



b) Drop Cut

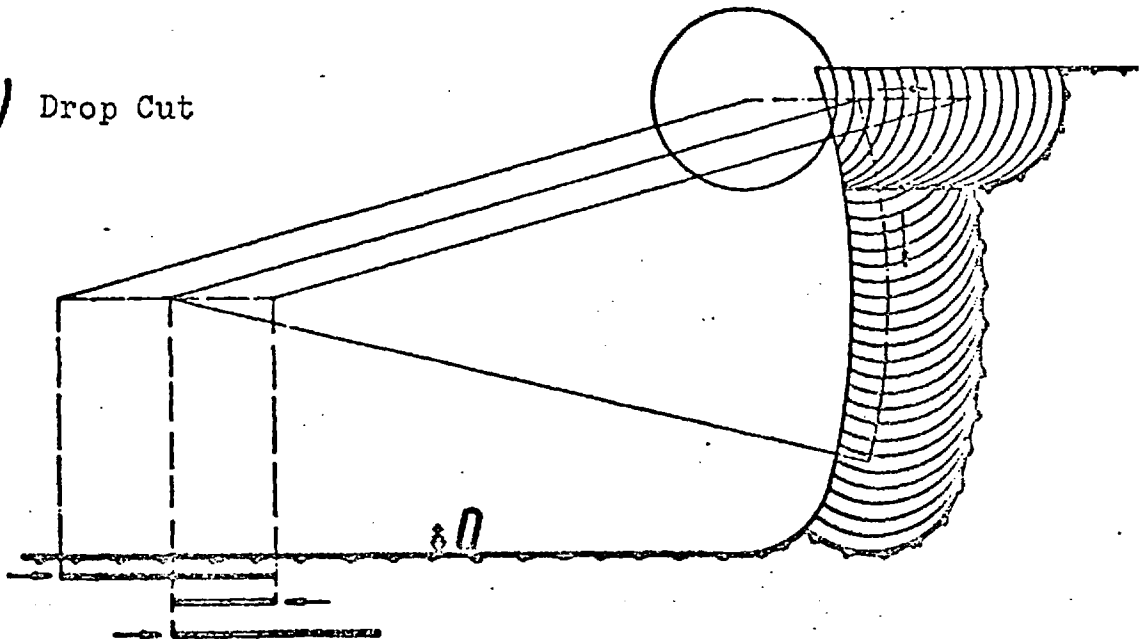


FIG.6.21 BWE DIGGING METHODS

and is described as the excavator "quality coefficient". A number of authorities have shown inconsistencies in its use 63, 69, 618, 619 and it appears that the excavator mass G_v , should be the service weight minus the ballast, since although the ballast may be 5-15% of the service weight, its value is usually less than 1% of the total cost of a BWE.

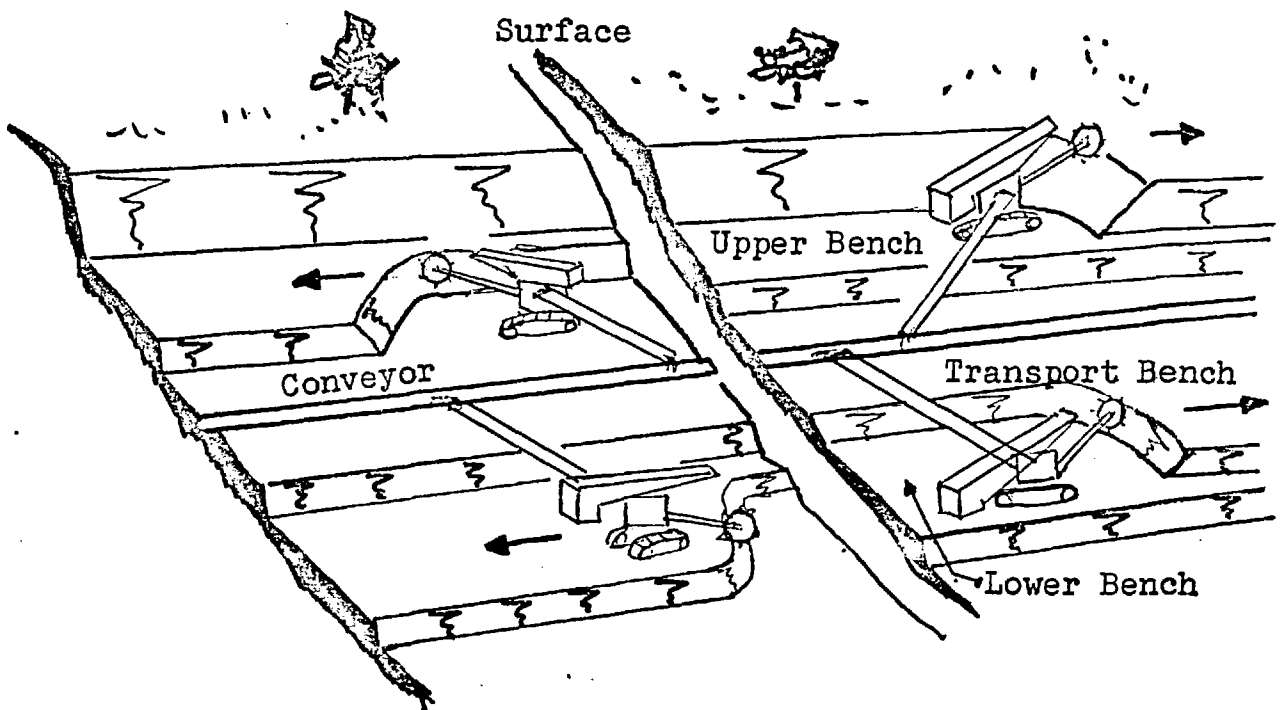
Maximum cutting height has been taken as the sum of upwards and downwards digging, which although satisfactory for "high-deep" BWE's, does not fully account for the quality of multi-bench operations (see Fig 6.22).

Hourly output is usually taken as the theoretical output i.e. bucket capacity x discharges per hour, although Gdertner ⁶⁹, uses the average net hourly output over a full days operation. He also uses the overall length with the BWE discharge boom and bucket wheelboom fully extended.

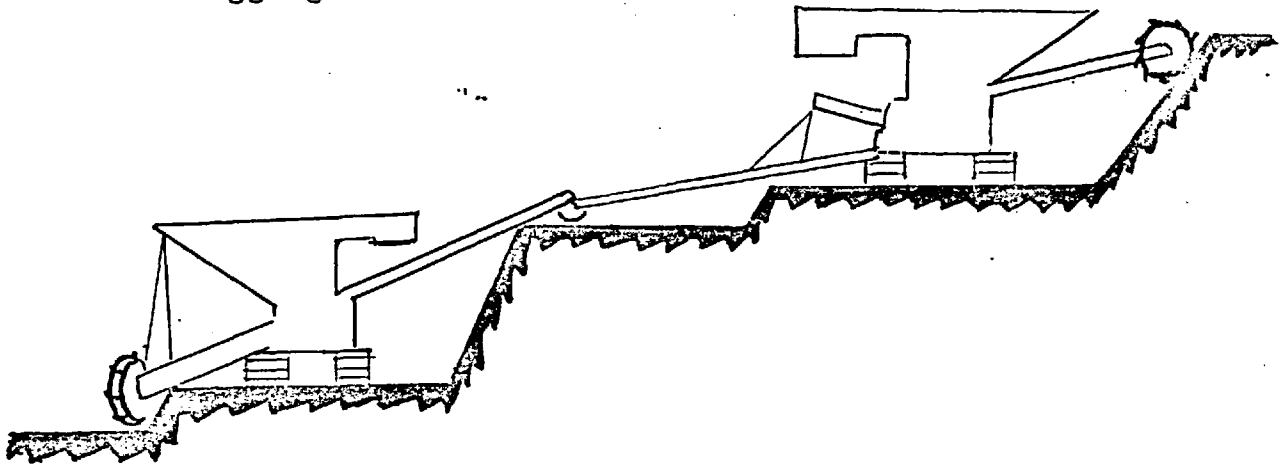
The excavator mass is not only a function of the hourly output but is also dependent on the digging height and overall length. An increase in digging height and overall length has a greater influence on excavator mass than an increase in output, but however an increase in output inturn involves an increase in digging height and overall length. X

Fig 6.23 is a range of the indices collected from a large number of BWE's. Owing to different structural designs it will be noted that the indices vary within the limits shown.

The use of coefficient G_v is not entirely satisfactory for excavator selection purposes, as it does not directly consider the nature of the ground and the operating conditions. It should therefore only be used for comparing BWE's of

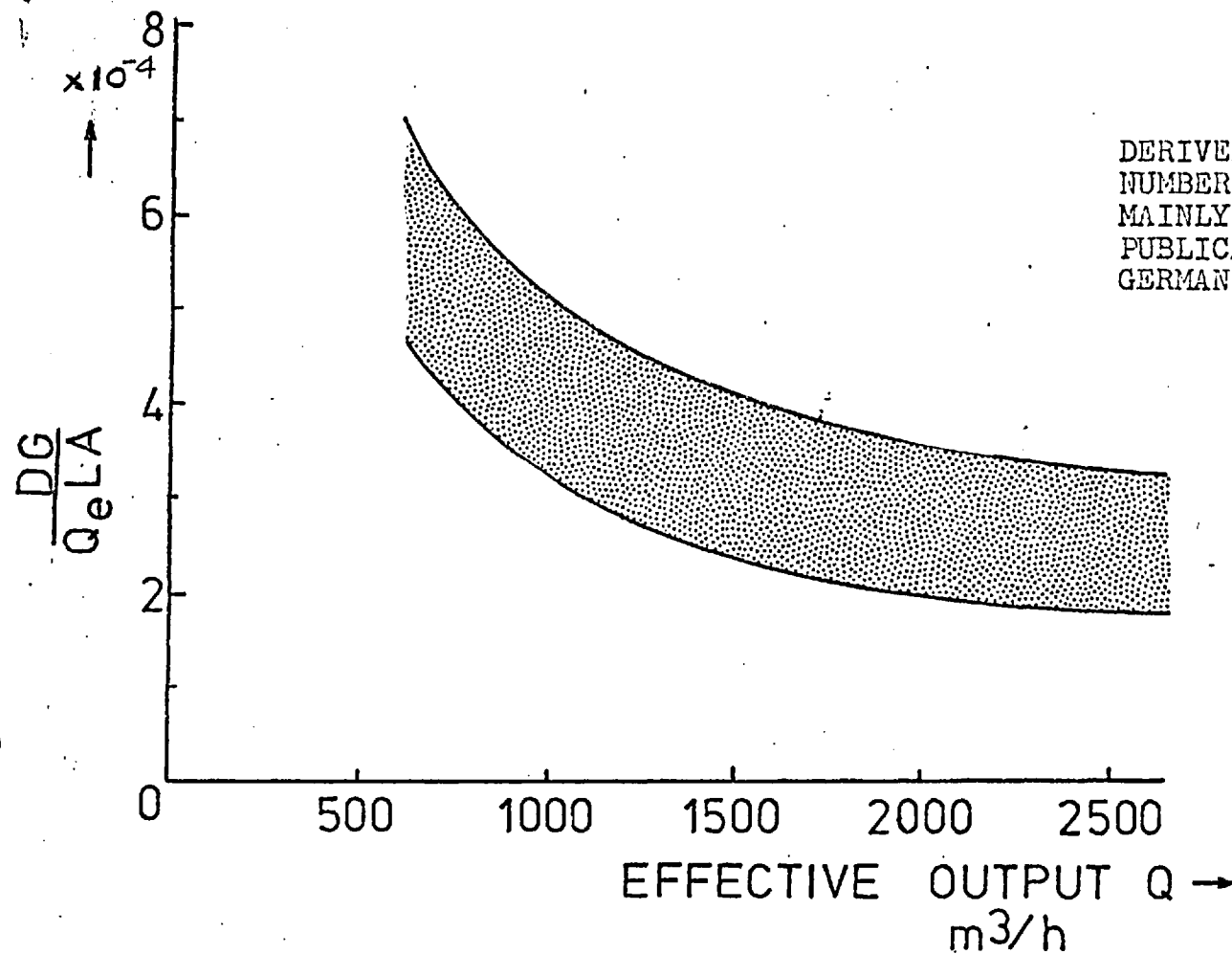


a) Upwards Digging on Three Faces, Downwards Digging on One Face



b) Upwards Digging on Two Faces, Downwards Digging on Two Faces

FIG 6.22 THREE BENCH SYSTEMS



DERIVED FROM A LARGE
NUMBER OF SOURCES.
MAINLY MANUFACTURERS'
PUBLICATIONS AND WEST
GERMAN LITERATURE

FIG 6.23 RANGE OF B.W.E. QUALITY COEFFICIENTS

different classes and types where the digging and transport conditions are comparable.

CONTINUOUS EXCAVATORS - ELECTRICAL REQUIREMENTS

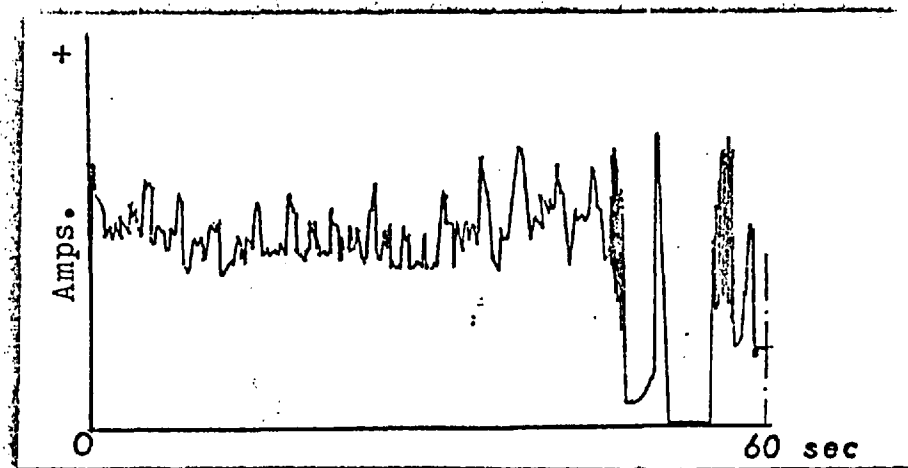
A typical load ~ time curve for a BWE is shown in Fig 6.24. It will be noted that the type of load is one much more acceptable to electricity supply authorities than those of cyclic (single bucket) excavators and therefore has less effect than for cyclic excavators on the systems approach to excavator selection. The electrical requirements however must be considered in detail. This subject is covered elsewhere ^{623, 624}.

SELECTION PROCEDURE

Machine Output

The theoretical output of a continuous excavator provides a convenient means of comparing similar machines, but the actual production rate may vary considerably from the theoretical. The author's field observations indicate that the bucket filling can vary from 0.4 for difficult conditions, e.g. hard or sticky ground, ground containing boulders, etc. to 1.32 for free flowing granular materials such as sands and gravels, when down-digging with a BCE. Some care must be exercised to use these values correctly since methods of calculating theoretical outputs vary. Most mine planners usually prefer to use a 'field factor' which allows for

- (a) bucket filling
- (b) planned maintenance stoppages
- (c) stoppages due to breakdowns
- (d) conveyor or rail track shifting and
- (e) non-working days.



2mm = 1sec

FIG 6.24 BWE LOAD~TIME TRACE (I~t)

Because of the large capital investments involved, continuous excavators are usually operated on a three-shift basis. In Germany and elsewhere statistics show that a daily scheduled operating time of 19.2 h is normally attained if the excavator is correctly matched to operating conditions. If Sundays and holidays are available for maintenance (including annual overhaul) 5000 scheduled h per year is the average operating time. In difficult operating conditions e.g. hard, abrasive or sticky ground, severe climate, etc., this will be reduced.

The annual production is then

$$Q/\text{year} = Q_{th} \times 5000 \times \text{Service factor, m}^3/\text{year} \dots (6.21)$$

where the service factor covers periods when the excavator is not working to full capacity because of poor bucket-filling manoeuvring, transport stoppages, etc. This service factor can vary from 0.5 to 0.8, depending on ground conditions, management efficiency, climate, etc., and may be estimated from plant records.

For initial planning, however, it is usual to base calculations on a field factor since the scheduled hours, management efficiency, climate, ground conditions etc., are not independent of one another.

The annual production is then

$$Q/\text{year} = Q_{th} \times \text{days/year} \times 24 \times \text{Field factor} \dots (6.22)$$

where Q_{th} is based on the volume of the bucket only and the cell or ring volumes are not included.

If studies from similar operations are not available the values in Table 6.III and 6.IV may be used, where

$$\text{Field factor} = OE \times JC \dots \dots \dots (6.23)$$

TABLE 6.III*

OPERATING EFFICIENCY

- CONTINUOUS EXCAVATORS (OE)

Ground conditions	Management conditions			
	Excellent	Good	Fair	Poor
Light	0.70	0.63	0.55	0.47
Medium-heavy	0.65	0.58	0.50	0.42
Heavy	0.57	0.50	0.43	0.36
Hard ground	0.42	0.37	0.32	0.27

TABLE 6.IV*

JOB CONDITIONS FACTOR

- CONTINUOUS EXCAVATORS (JC)

Excellent	0.92
Good	0.83
Fair	0.73
Poor	0.62

The same comments apply for Table 6.III and 6.IV as for Table 2.VI.

*See Chapter 2 for method of derivation.

Machine Geometry

The machine geometry of continuous excavators is largely determined by the pit layout. The overall slope angles involved are not usually steep because

- a) the rocks involved are relatively weak and
- b) the deposits mined may be relatively flat, advance being lateral rather than vertical, so that advance stripping is not quite so economically undesirable as for a deposit of mainly vertical aspect.

Because of this the benches are usually wide and space restrictions are not often a major problem. If an excavator must perform upward and downward digging it requires adequate reach to work on both sides of the transport system. In a rotating pit operation the excavator must be able to work up to the pivot point, where space is restricted, however whereas in parallel advancing pit, where end-bench lift conveyors may be used, the excavator must be capable of digging up to the corner of the pit. The machine geometry may impose some limitations on pit layout, and considerable planning is required before the final machine geometry can be decided. The solutions to these problems are best achieved by drawing pit plans and sections. In some cases it may be necessary to use a mobile, crawler-mounted transfer conveyor; for large machines a telescopic discharge boom conveyor may have to be incorporated into the excavator design.

It is simple geometry that for a BWE, the steeper the highwall slope, the greater the block width and both field tests and model experiments show the relationship between bucket wheel head geometry and the highwall slope.

Clearance for the bucket wheel head as it approaches the highwall slope is essential to prevent it fouling. These two parameters therefore have a vital bearing on continuous excavator geometry. The construction of a scale model for this purpose is a vital part of the selection procedure (see Appendix 6B)

The so-called "compact design" BWE (Kompaktbauweise), typically with wheel diameters of 9-10m (30-33 ft) mounted on booms 12 - 15m (40-55 ft) long, discharge booms 20 - 27m

(65-90 ft) long, and machine masses less than 1000 tonnes appear attractive since they can be mounted on two crawlers. This eliminates the extra weight required for one crawler, its associated steering gear and electrics, since a two crawler machine is simply steered by differential speed between crawlers. Moreover the two crawler machine can turn in a much tighter radius. Compact designs therefore have lower capital cost than long boom machines. The digging heights achieved by compact designs are however limited to about 12m (40 ft) at highwall slope angles of 60° . Also for downwards-digging compact designs require a cover belt on the bucket wheel boom conveyor, which makes it unsuitable for material that is sticky, frozen or contains boulders. Short booms are not suitable for part block excavation in selective mining, since an oversize bucket wheel and a long boom is needed for clearance high up on the bench face.

The long boom machine provides greater digging height, greater block width and as slewing speed for both compact design and long boom machines will be similar, the long machine will mine less passes per hour, therefore the possibility of instantaneous flooding of belt conveyor transport is reduced.

In general therefore:

1. Long boom BWEs are more suited to deep deposits with high bench heights, where lumpy or sticky material is dug downwards and where conveyor transport is used.

2. For deposits where fairly low benches are used, for across the pit (direct) dumping, where rail (or rubber tyred) transport is used, since it is not affected by instantaneous flooding, compact design BWE's are suitable.

Fig 6.22 shows the "Three Bench" operation whereby four working faces can be mined from three benches with the use of a crawler-mounted, interconnecting, bridge conveyor, using upwards digging with a long boom BWE, on faces A, B, C and downwards digging on face D.

The major advantage of the three bench system is the possibility of mining a much greater overall height than with compact designs. Ramps must of course be cut for BWE movement between benches. Multiple bench operation is also possible for medium size machines where it is still economic to completely suspend the discharge boom from the BWE (Treue. Brown Coal Mine, West Germany, LMG BWE No.1134). Similar operations are possible by incorporating a mobile, crawler-mounted conveyor in the layout.

CONTINUOUS EXCAVATOR COSTS

Ownership Costs

Continuous excavators have service lives in excess of 30 years and machines commissioned in 1935 are still in daily operation in East Germany, but tax concessions, company depreciation policies, etc., can influence the write-off period. It is difficult to compare single-bucket excavators with continuous excavators because they usually perform different duties, but for the same (production rate x reach), continuous machines are usually more expensive in capital cost than single-bucket machines. Because of the large capital

sums involved and the irrevocable nature of the investment decision, comprehensive investigations and detailed discussions with manufacturers are essential before the selection of a continuous excavator can be finalized. As they also form part of a continuous system which in a multi-bench mine must be synchronized with other excavator-transport systems, detailed planning of mine operations is imperative.

The following can be used as a guide in calculating ownership costs.

1. FOB machine cost including sales taxes, etc. £.....
2. Freight and insurance (to site) £.....
3. Import duty £.....
4. Sub-total £.....
5. Erection costs (formula (6.24)) £.....
6. Insurance during erection £.....
7. Sub-total £.....
8. Interest up to start of production (allow interest on 30% of sub-total 7) £.....
9. Sub-total £.....
10. Excavator write-off period
 - n = ...years (10a)
 - h/year (10b)
 - total hours (10c)
11. Machine depreciation cost/h = $\frac{(\text{sub-total 9})}{(\text{Item 10c})} = \text{£}.....$
12. With average investment formula, assuming depreciation charges replace original investment
 - Average machine investment
 - = $\frac{(\text{Sub-total 9}) \times (n + 1)}{2n} = \text{£}.....$

13. Interest rate =....%
14. Insurance =....%
15. Taxes, etc (if any) =....%
16. Total =....%
17. Interest, taxes, insurance etc. cost/h =
 $(\text{Total } \%) \times (\text{Item 12}) \div (\text{Item 10b})$ £.....
18. Trailing cable costs
capital costs (CIF) + import duty + insurance up to start of production (see Table 6.VI) = £.....
19. Trailing cable life
.....years (19a)
.....h/year (19b)
.....Total h (19c)
20. Trailing cable depreciation
 $\text{cost/h} = \frac{\text{Item 18}}{(\text{Item 19c})}$ = £.....
21. Trailing cable - average investment
 $= \frac{(\text{Item 18}) \times (\text{Item 19a} + 1)}{2 (\text{Item 19a})}$ = £.....
22. Trailing cable, interest, taxes, Insurance etc.
 $\text{cost/h} = \frac{(\text{Item 21}) \times (\text{Item 16})}{(\text{Item 19b})}$ = £.....
23. Total ownership costs/h
 $(\text{Item 11}) + (\text{Item 17}) + (\text{Item 20}) + (\text{Item 22}) =$ £.....

If manufacturers' figures are not available, erection costs based on 1973 projections may be calculated from

$$\text{Erection costs} = aM \dots\dots\dots (6.24)$$

where M is machine mass and a is constant (Table 6.V)

TABLE 6.V
CONSTANT FOR CALCULATING ERECTION
COSTS OF CONTINUOUS EXCAVATORS

Machine mass, t	a
Below 1000	120
1000—2000	100
2000—3000	85
3000—4000	72
Above 4000	60*

*Some discretion is needed with very large machines as increased erection costs may be necessary because of increased site fabrication due to the size of component parts, but a reduction in works cost may be expected.

The above figures include cost of supervision of erection by the manufacturer, i.e. time, subsistence, local transport, etc., skilled and unskilled labour, erection equipment, i.e. cranes, tools, slings, etc., workshop costs, etc., but not travel expenses of the manufacturer's erectors. The costs should be adjusted to take account of

- a) distance from the port of entry, railhead, road, etc. from the erection site: off-loading facilities, availability of heavy transport etc.,
- b) availability of skilled labour;
- c) quality of mine supervisory staff; and
- d) availability of cranes, tools, workshop services, etc.

Operating Costs

24. Maintenance and supply costs/h

$$= \frac{(\text{Sub-total 9} + \text{Item 18}) \times (M) (\text{see Table 6.VII})}{(\text{Item 10b})} = \text{£} \dots$$

25. Electrical power consumption/h x cost/kWh (see Table 6.I) (the method of charging for electricity must be carefully investigated) = £.....

26. Labour rate/h, to include social benefits, taxes, insurance, etc. (a BWE crew usually numbers from 3 to 6) = £.....
27. Total operating costs/h
(Item 24) + (Item 25) + (Item 26) = £.....

TABLE 6.VI

TRAILING CABLE PRICES - CONTINUOUS EXCAVATORS *

Theoretical output m ³ /h	Voltage, kV		
	5 - 10	10 - 15	22 - 25
0 - 1000	4.0	5.0	—
1000 - 2000	7.5	8.0	—
2000 - 3000	10.5	10.5	—
3000 - 5000	13.5	13.0	13.5
5000 - 7500	—	—	16.5

*Based on fstg 700/ton. 1973 projected prices.

Total Ownership and Operating Costs

28. Total ownership + operating costs
=(Item 23) + (Item 27) = £.....
- cost/t $\frac{\text{Item 28}}{\text{t/h}}$ = £.....

No administrative or amortization charges are included in these figures.

TABLE 6.VII

MAINTENANCE AND SUPPLY FACTOR (M)

Material	M
Light ground (free-flowing)	0.01
Medium ground	0.02
Heavy ground	0.03
Hard or frozen ground	0.05
Ground containing boulders	0.04-0.05
Abrasive sandstones, shales, etc.	0.05

FURTHER STUDIES

The past decade has produced considerable improvements in BWE heads and improvements in excavating techniques. The continuous excavator is much more susceptible to analysis than other excavators and even greater improvements can come about by a deeper understanding of the fundamental operation of the BWE. Although Appendix 6A goes some way in providing this understanding, the author believes that further work can bring about simplifications in design resulting in lower capital costs, further improved efficiency and because of a better fundamental knowledge, considerably reduced maintenance costs.

Additionally the application of the BWE in hard ground merits much greater attention. At present there is no basic research into this subject. The nature of the industry is such that this is difficult to set up. No single mine has a sufficiently wide range of conditions to carry out this work and competition between manufacturers has prevented co-operation between them. Such a project would be best carried out by a university department backed by industry as a whole. The first step must be a deeper investigation into Specific Digging Forces, which is not fully understood at present. Perhaps the most important contributions in this field have come from Lubrich ⁶²⁵ and Himmel ⁶²⁶ but both have mainly investigated relatively weak materials. Again this work would require the co-operation of industry as a whole.

It is difficult to envisage any major breakthrough in the field of hard rock excavation in the near future but the considerable advantages to be derived from continuous excavation methods clearly indicate that this could be a fruitful area of research.

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APPENDIX 6.A

A RIGOROUS ANALYSIS OF BUCKET WHEEL EXCAVATOR OPERATION

The formulae defined in Chapter 6 for BCE operation are straight forward and this appendix is limited to BWE operations. The analysis used is derived from the cut geometry developed by Härtig, H and Ciesielski, R. "Grundlagen für die Berechnung von Braunkohlentage bauen", VEB Deutscher Verlag für Grundstoffind industrie, Leipzig, 1966.

Fig 6.14 indicates a continuous variation in the width of slice W, while the radial thickness of slice relative to the bucket wheel centre also varies continuously. Additionally the slice thickness (slice depth for the drop cut) radial to the slew axis also varies with angle of slew. It is obvious that these variations are due to rotation in a vertical plane of the bucket wheel about its axis and simultaneous rotation in a horizontal plane about the slew axis.

Because of the continuous variation in bucket slice shape, output formulae based on the summation of volumes cut by each bucket throughout each segment are complex, particularly for the drop cut. It is therefore proposed to derive formulae on the basis that production is a function of segment cross section and slewing speed. This is valid for bucket wheels where the contents excavated by each bucket does not exceed the heaped volume of the bucket and its associated ring volume. This is the general case in modern BWEs and even for celled bucket wheels filling in excess of bucket plus cell capacity is uncommon. This approach should eliminate the empiricism necessary with the discharges per minute approach. Some of the derivations used employ approx-

imations for simplicity but these are well within the practical requirements for BWE production formulae.

THE TERRACE CUT

a) Slice Cross Section Area

Slice cross section area was previously given as hT , but since the bucket wheel slews sideways when excavating, each bucket does not cut a slice with vertical sides or with uniform width. The area referred to therefore is a vertical cross section of the segment in a plane radial to the slew axis (Fig 6A1).

The segment cross sectional Area A, is then: referring to Fig 6A1, assuming the full terrace height is established and the wheel centre has moved forward horizontal distance T, and the slice cross section is sickle shaped, $E_2 O_2 Q E_1$. Note that area $O_2 Q O_1$ is not dug by the bucket wheel. However T is less than $0.1 D$, and $h = \text{approx. } 66.7\% D$, therefore $O_2 Q O_1$ is less than 0.3% of A and can be neglected, hence the slice arc becomes $E_2 O_2 O_1 E_1$.

The cutting circle engages only arc $O_1 Q E_2$, i.e. $O_2 E_2$ when digging. Considering then portion $O_1 Q E_1 K_1$, it is clear that when this portion moves forward the area traced out in the vertical plane by the leading edge $O_1 Q E_1$ is equal to the area traced out by the trailing edge $O_1 K_1$, when $O_1 Q E_1$ moves distance T to $O_2 E_2$, then $O_1 K_1$ moves distance T to $O_2 K_2$. Hence area $E_2 O_2 O_1 E_1 = \text{area } K_1 O_2 O_1 K_1$ which is equal to hT .

An alternative proof is the area of element LNPQ at angle ϕ to the vertical $M_2 O_2$ is given by:

$$dA = LN \cdot dy$$

$$LN = T$$

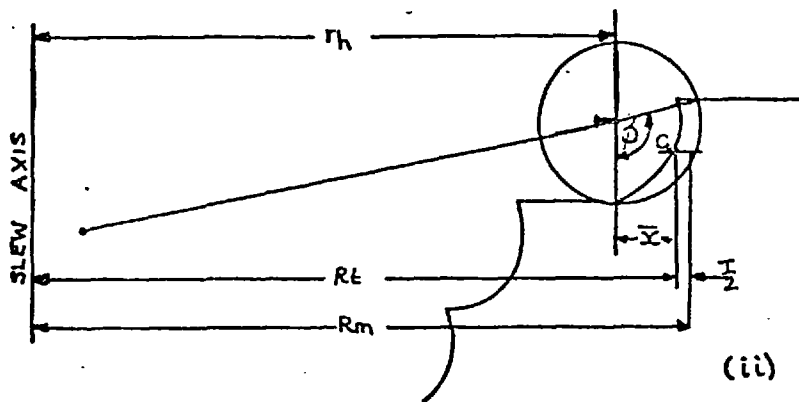
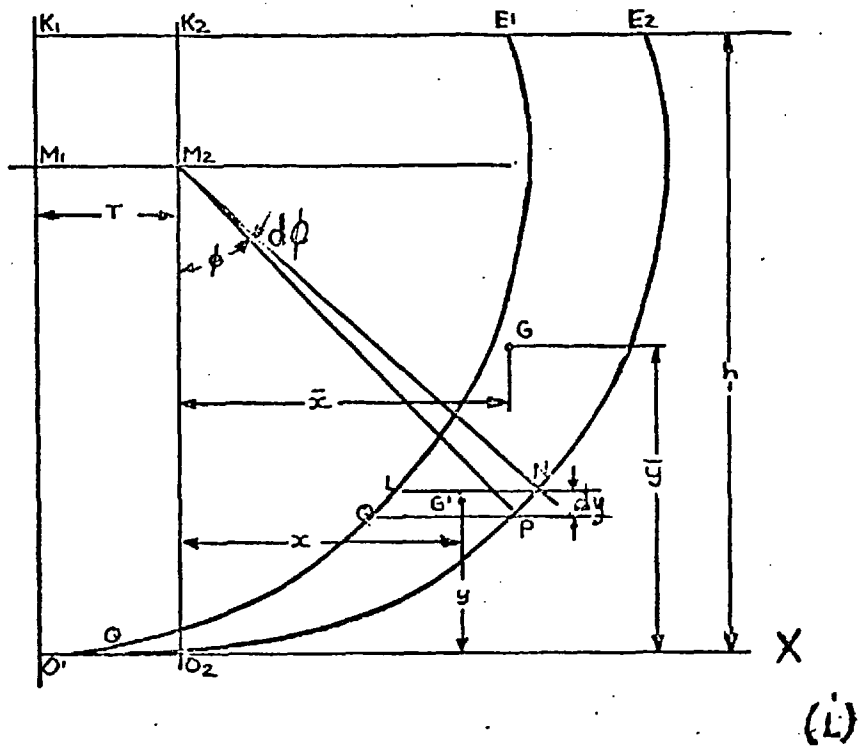


FIG 6.A1 THE TERRACE CUT, SEGMENT CROSS SECTION AND SLEWING RADIUS DETAILS.

and $dy = NP \sin \phi$
 now $NP = M_2 N \cdot d\phi = \frac{D}{2} d\phi$
 i.e. $dA = \frac{TD}{2} \sin \phi \cdot d\phi$

Then between angles ϕ_1 and ϕ_2

$$A = \frac{TD}{2} \int_{\phi_1}^{\phi_2} \sin \phi \cdot d\phi$$

and $A = \frac{TD}{2} [-\cos \phi]_{\phi_1}^{\phi_2}$

At the start of cutting a terrace, and at the thin end of the segment; ϕ_1 is usually greater than zero, so the cross section and slew speed requirements may be obtained by integration between known limits of ϕ_2 and ϕ_1 .

If the full terrace height is established i.e. $\phi_1 = 0$, and neglecting the variation at the thin end of the segment i.e. where slew angle θ approaches 90° ,

ϕ_2 becomes ϕ and...

$$A = \frac{TD}{2} (1 - \cos \phi)$$

but $\frac{D}{2} (1 - \cos \phi) = h$, therefore

$$A = h.T \quad \dots\dots\dots (6.A1)$$

Slice Centre of Area

If y is the distance from the centroid G^1 of element L N P Q to the axis $O_2 X$, then $\bar{y}$ the centroid of the whole segment cross section may be expressed as:

$$A \bar{y} = T \int_0^h y \cdot dy$$

$$= \frac{1}{2} Th^2$$

but $A = hT$ and

$$\bar{y} = \frac{h}{2} \quad \dots\dots\dots (6.A2)$$

b) Referring to Fig.6.A1, to find $\bar{x}$ of the centre of area of the cross section

Taking moments about axis $O_2 M_2$

$$x = \frac{D}{2} \sin\left(\phi + \frac{d\phi}{2}\right) - \frac{T}{2}$$

$$\text{but } dA = \frac{TD}{2} \sin\phi \cdot d\phi \quad \text{and}$$

$$A\bar{x} = \frac{TD^2}{4} \int_0^\phi \sin^2\phi \cdot d\phi - \frac{T^2D}{4} \int_0^\phi \sin\phi \cdot d\phi$$

$$= \frac{TD^2}{4} \left[\frac{\phi}{2} - \frac{\sin 2\phi}{4} \right]_0^\phi - \frac{AT}{2}$$

$$\text{but } A = hT, \text{ and } h = \frac{D}{2} (1 - \cos\phi)$$

$$\therefore \bar{x} = \frac{D}{2(1 - \cos\phi)} \left(\frac{\phi}{2} - \frac{\sin 2\phi}{4} \right) - \frac{T}{2} \dots\dots\dots (6.A3)$$

Volumes of Segment and Block

Fig 6.A1 shows G lying within the segment. It is obvious however that this only occurs for low values of h and/or for high values of T. With higher values of h and normal values of T, G lies outside the segment, see Fig 6.A2 which indicates that a segment cross section of height h and thickness T, with its centroid at G, can be represented by an equivalent rectangle hT, also having its centroid at G. The whole segment may thus be represented as a sickle of width S and vertical sides, rather than the correct shape (Fig 6.10).

The limits of Θ_L and Θ_R of the equivalent segment on either side of the line of advance are determined by the slope at the junction of the working face and the highwall slopes. From formula (6.15):

$$S = R_m (\sin \Theta_L + \sin \Theta_R)$$

where

R_m = radius to the outer edge of the "equivalent" segment.

This has the same value as that given above for R_m , i.e. the mean radius to the outer edge of the slice.

Formula (6.16) gives the volume of the segment as:

$$V = h.T.S$$

By applying the methods used to derive (6.A1) it can be shown that the area of any horizontal cross section (actual) may be expressed as T.S. If the distance between each such horizontal cross section is dh then:

$$V = \int_0^h T.S.dh$$

$$\text{i.e., } V = h.T.S.$$

The same result is obtained by evaluating the volume of the "equivalent" segment.

If A is the depth of advance to excavate a terrace of n segments of thickness T,

then

$$A = nT$$

The terrace volume, V_t , is

$$V_t = n.H.T.S.$$

$$\text{i.e., } V_t = A.h.S.$$

..... (6.A4)

Similarly if the block is excavated in n terraces having heights, $h_1, h_2, h_3, \dots, h_n$ then the block height H is

$$H = h_1 + h_2 + h_3 + \dots + h_n$$

From formula (6.A4), the total block volume V_B is:

$$V_B = A \cdot H \cdot S \quad \dots\dots\dots (6.A5)$$

THE DROP CUT

The slice cut area of the drop cut at the line of advance was previously given as:

$$T A_h$$

Fig 6A2 shows a bucket wheel digging a face of inclination α , the segment cross section having thickness T parallel to the face and depth A_h normal to the face. This segment is formed by moving the bucket wheel centre from M_1 to M_2 .

Considering $O_2 X$ and $O_2 M$ as axes for the co-ordinates of the cross section, then the methods used to derive formula (6.A1) show that the cross sectional area A_d is:

$$A_d = T A_h \quad \dots\dots\dots (6.A6)$$

Slice Centre of Area

Again taking $O_2 X$ and $O_2 M$ as axes then the co-ordinates of centroid G are:

$$\bar{y} = \frac{A_h}{2}$$

and $\bar{x}$ again is given by (6.A3), i.e.:

$$\bar{x} = \frac{D}{2(1 - \cos \phi)} \left(\frac{\phi}{2} - \frac{\sin 2\phi}{4} \right) - T_2$$

(i)

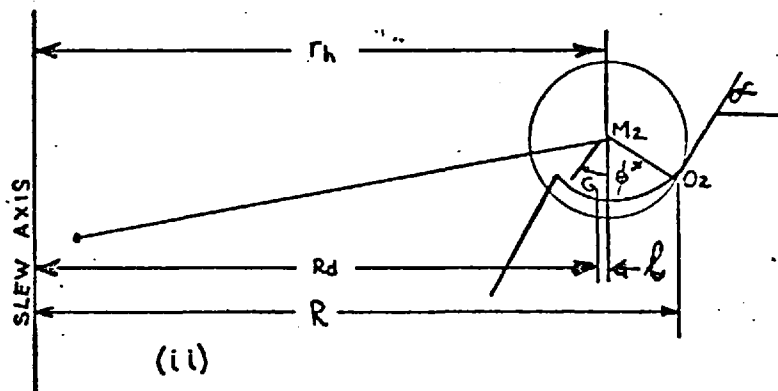
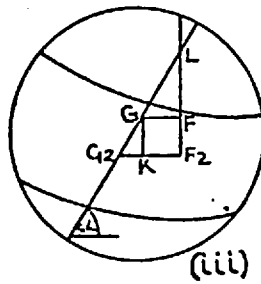
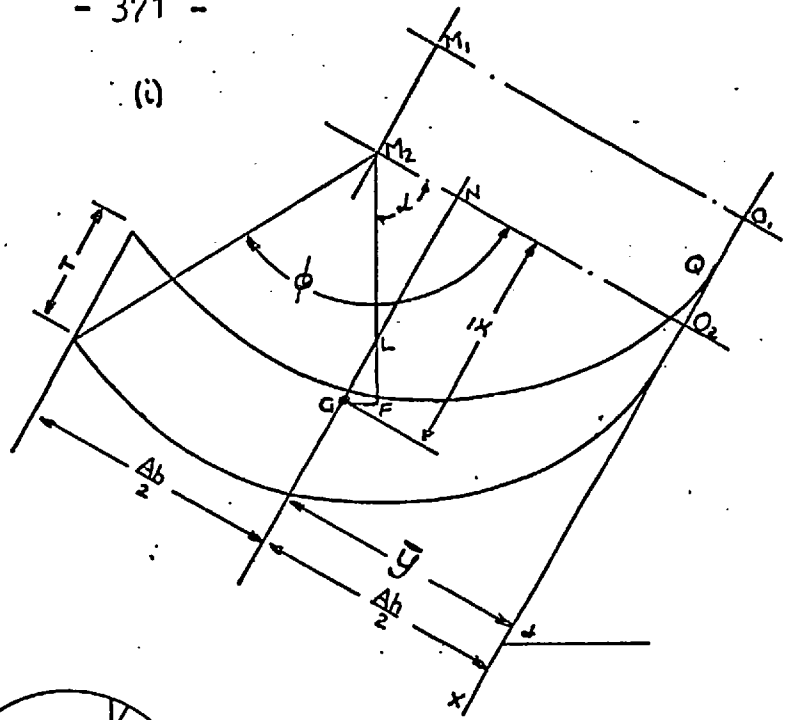


FIG. 6.A2 THE DROP CUT, SEGMENT
CROSS SECTION AND SLEWING
RADIUS DETAILS.

Mean Slewing Radius

Referring to Fig 6A2, R_d , the horizontal distance from the slew axis to the cross section centroid is:

$$R_d = r_h - l \quad \dots\dots\dots (6.A7)$$

Again r_h is the horizontal distance from the slew axis to the wheel centre and in this case l is the horizontal displacement of the centroid G from the wheel centre M_2 .

Also R is the radius from the slew axis to the outer edge of the segment. This radius is of importance for determining the variation of A_h with slew angle θ (see later). For a given boom inclination, i.e. a given value of r_h , and a fixed value of face angle α , R does not vary but R_h does. By inspection from Fig 6A2:

$$R = r_h + \frac{D}{2} \sin \alpha$$

Again referring to Fig 6A2, line GN is parallel to the face i.e. $O_2 X$, and $GN = \bar{x}$. FM_2 is vertical through the wheel centre and L is the intersection point of FM_2 and GN.

$$\text{Now } l = GF$$

$$\text{and } GF = GL \cos \alpha$$

$$\text{and } GL = \bar{x} - LN$$

$$\text{Since } LN = NM_2 \tan \alpha$$

$$\text{and } NM_2 = \frac{1}{2}(D - A_h)$$

Thus

$$l = (\bar{x} - \frac{1}{2}(D - A_h) \tan \alpha) \cos \alpha$$

$$\text{i.e. } l = \bar{x} \cos \alpha - \frac{1}{2}(D - A_h) \sin \alpha$$

Substituting $\bar{x}$ as formula (6.A3) and since

$$A_h = \frac{D}{2} (1 - \cos \phi) \quad \text{then}$$

$$l = \frac{D \cos \alpha}{2(1 - \cos \phi)} \left(\frac{\phi}{2} - \frac{\sin 2\phi}{4} \right) - \frac{T}{2} \cos \alpha$$

$$- \frac{D}{4} \sin \alpha (1 + \cos \phi) \dots\dots\dots (6.A8)$$

In practice extreme accuracy may not be necessary and an approximation may be made with the centroid half way between the two arcs. Since $\bar{y} = A_h$ this gives

$$\bar{x}_2 = \sqrt{\left(\frac{D}{2}\right)^2 - \left(\frac{D}{2} - \frac{A_h}{2}\right)^2} - \frac{T}{2} \dots\dots\dots (6.A9)$$

This may be written

$$\bar{x}_2 = \frac{1}{2} \sqrt{2 D A_h - A_h^2} - \frac{T}{2} \dots\dots\dots (6.A10)$$

The effect of this simplification may be shown in Fig 6.A2 (iii) with new centroid position G_2 , and hence l becomes $G_2 F_2$ which becomes l_2 . If GK is parallel to the vertical through M_2 , then:

$$l_2 = K F_2 + K G_2$$

$$= G F + G G_2 \cos \alpha$$

$$= l + (\bar{x}_2 - \bar{x}) \cos \alpha$$

$$\text{hence } l - l_2 = (\bar{x}_2 - \bar{x}) \cos \alpha, \text{ which}$$

although can be large in relation to l , its effect on R_d is not usually significant and an accuracy better than 99% is unusual for BWE's of normal geometry.

VOLUMES OF SEGMENT AND BLOCK

The conditions applicable to the terrace cut which allow the development of an "equivalent" segment do not apply to the drop cut since:

- (a) the segment does not lie between two horizontal planes spaced at uniform distance h ,
- (b) the cross section centroid does not move in a horizontal plane which bisects the segment,
- (c) even if α remains constant, a large variation in Rd can occur.

However in a segment starting at $\theta_L = 0^\circ$, only 4% of its volume is excavated between 80° and 90° and between $\theta_L = 0^\circ - 80^\circ$ it may be shown that the average slope of the path traversed by G around the segment does not exceed 1° .

These statements apply to crowd action BWE's using a drop cut, i.e. the bucket wheel is positioned by hoist and crowd motions only and the slew axis does not move, therefore α remains constant. With a fixed boom BWE the bucket wheel must be positioned by hoist and crawler movement i.e. the slew axis moves along the line of advance between successive segments. This causes the face slope angle to increase as the wheel slews away from the line of advance. Also the rate of increase of α varies as r_h varies between successive segments. Consequently in the formula derived here α is the instantaneous value for fixed boom BWE's. From the geometry of the fixed boom BWE and the value of α at the line of advance it is possible to express α as a function of θ .

The formulae derived for the crowd action BWE are somewhat complex and it is readily apparent that rigorous volume and output formula for fixed boom BWEs using the drop cut will be more complex. To demonstrate the complexity of the

drop cut, the determination of segment volume is shown here.

By Pappus' Theorem, the drop cut volume is expressed:

$$V_d = \int_{\theta_L}^{\theta_R} A_{h\theta} \cdot T \cdot R d\theta \cdot d\theta$$

since

$$R d\theta = r_h - l_\theta$$

then

$$V_d = \int_{\theta_L}^{\theta_R} A_{h\theta} \cdot T \cdot (r_h - l_\theta) \cdot d\theta \quad \dots\dots\dots (6.A11)$$

θ_L is taken as the angle to the mid point of the junction between the segment and the standing high wall.

From formulae (6.10)

$$A_{h\theta} = A_h \left(\cos \theta - \frac{\sin^2 \theta}{2 p_1} \right)$$

Also at any position, the angle of contact ϕ of the circle and the segment may be expressed as:

$$A_{h\theta} = \frac{D}{2} (1 - \cos \phi) \quad \text{and from}$$

(6.10) for $A_{h\theta}$, ϕ may be expressed as a function of θ , thus:

$$\phi = \cos^{-1} \left(1 - \frac{2A_h}{D} \left(\cos \theta + \frac{\sin^2 \theta}{2 p_1} \right) \right) \quad \dots\dots\dots (6.A12)$$

By substituting this value of ϕ in formula (6.A8), l

may be expressed as a function of θ , then (6.A11) may be developed:-

$$\text{Let } \frac{2A_h}{D} \left(\cos \theta + \frac{\sin^2 \theta}{2p_1} \right) = f$$

$$\text{and } \sin 2\phi = 2 \cos \phi \sqrt{1 - \cos 2\phi}$$

then (6.A11) becomes

$$V_d = \frac{D}{2} \cdot T \cdot r_h \int_{\theta_L}^{\theta_R} f \cdot d\theta -$$

$$\frac{D \cdot T \cdot \cos \alpha}{2} \int_{\theta_L}^{\theta_R} f \left[\frac{D}{4F} \left(\cos^{-1}(1-f) - (1-f) \right) \sqrt{2f - f^2} \right.$$

$$\left. + \frac{Df}{4} \tan \alpha - \frac{D}{2} \tan \alpha - \frac{T}{2} \right] d\theta$$

..... (6.A13)

It would normally be sufficiently accurate to use h_2 for f .

$$\text{For simplicity let } \cos \theta + \frac{\sin^2 \theta}{2p_1} = m$$

and A_h be a constant fraction k of diameter D , then (6.A11) becomes

$$V_d = k \cdot D \cdot T \cdot r_h \int_{\theta_L}^{\theta_R} m \cdot d\theta -$$

$$\frac{1}{2} (k \cdot D^2 \cdot T \cdot \cos \alpha) \int_{\theta_L}^{\theta_R} m \left(\sqrt{2km - k^2 m^2} \right. \\ \left. - \frac{T}{D} - \tan \alpha (1 - km) \right) \cdot d\theta$$

..... (6.A14)

In any given segment, A_h , r_h , p_1 , T and k are constant so it is shown that it is possible to express segment volume in terms of the slew angle θ . A simple geometric method however is obviously preferable to the complex integration used having established that V_d can be expressed in terms of a single variable. The geometric method may also be applied to the drop cut with a fixed boom BWE, the mathematical form of which is obviously more complex than formulae (6.A13) and (6.A14). Simple geometric methods are derived in the output section.

BWE OUTPUT

Terrace Cut

Formula (6.14) gives the instantaneous output for the terrace cut as

$$Q^1 = h \cdot T_\theta \cdot V_{s\theta} \cdot 60$$

The formula is obviously valid, but further derivation can illustrate the effect of slewing speed on output and the problems associated with the slewing system of a BWE.

Fig 6A3 shows a segment of a terrace cut having a thickness T at the line of advance and height h through its range of θ_L to θ_R . The slewing centres for this and the previous segment are C_1 and C respectively. i.e. $CC_1 = T$. OEE_1O_1 is an element of the segment at slew angle θ and extending over the small angle $d\theta$.

From Pappus Theorem, the volume dV of this element is (area of its cross section in a radial plane) $\times$ (distance travelled by the centroid of the cross section). The centroid moves from C to C_1 and its radius R_t is $r_h + \bar{x}$ (see previously). Then

$$dV = h \cdot T_\theta \cdot R_t \cdot d\theta$$

$$\text{Now also } Q^1 = \frac{dV}{dt}$$

therefore from (6.A15)

$$Q^1 = h \cdot T_\theta \cdot R_t \cdot \frac{d\theta}{dt} \quad \dots\dots\dots (6.A15)$$

The angular slewing velocity ω_θ at θ is:

$$\omega_\theta : \frac{d\theta}{dt} \quad \text{rad/min} \quad \text{and the slewing speed is}$$

$$V_{s\theta} = R_t \omega_\theta \quad \text{m/min}$$

Thus the instantaneous value of hourly output at slew angle θ is:

$$Q^1 = h \cdot T_\theta \cdot V_{s\theta} \cdot 60 \quad \dots\dots\dots (6.14)$$

Then the average hourly output, Q_a , throughout the segment is:

$$Q_a = \frac{1}{t} \int Q^1 \cdot dt$$

where t' is the time in minutes to traverse the whole

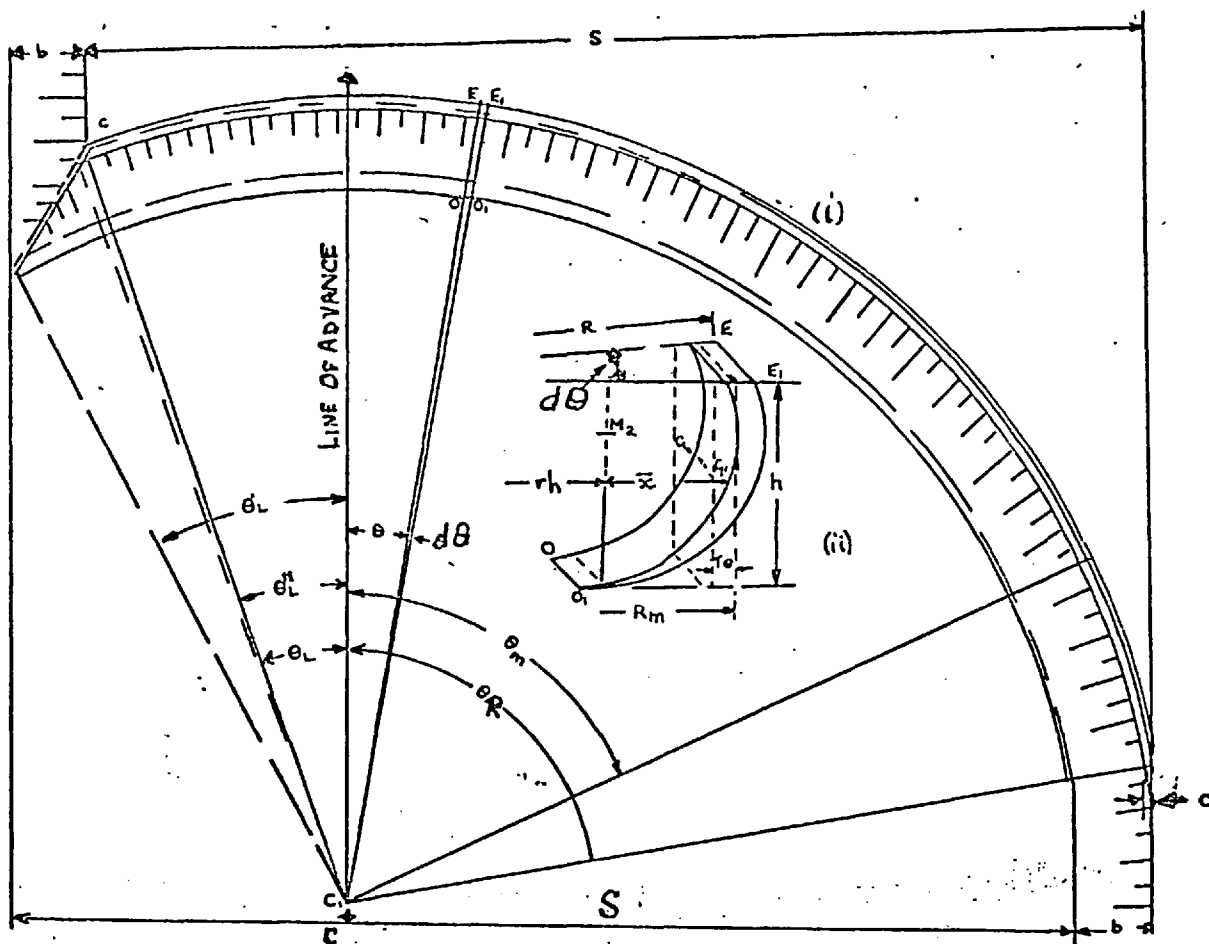


FIG 6.A3. THE TERRACE CUT. TYPICAL SEGMENT DETAILS.

segment.

This is obviously the same as:

$$Q_a = \frac{V \cdot 60}{t'}$$

The time lost at the end of each segment may be estimated from the characteristics of the BWE and the block and terrace dimensions, which together with the total digging period enables the utilisation factor, u , to be determined.

The effective hourly output over a number of segments is then:

$$Q = u \frac{(\sum V \cdot 60)}{(\sum t')} \quad \dots\dots\dots (6.A16)$$

This formula represents the actual output from a BWE. The segment volume may be determined accurately from r_h and D , and from selected values of h , T and the slew limits, θ_L and θ_R . The slewing speed variations possible with a modern BWE however make it impossible to precisely determine t' . (It is shown later however that satisfactory estimates may be made for t').

The left and right slew limits for the top of the segment ($E.E_1$) are θ_L and θ_R , respectively. By inspection the radius R at the top of the segment is:

$$R = r_h + \sqrt{D_h^2 - h^2}$$

The radius at the bottom of a segment of full height h is r_h and the angle of slew θ_R is the same for both top and bottom of the segment. The difference in width between the top and bottom of the segment, to the right of the line of advance, is b , and by inspection

$$b = \sin \theta_R \cdot (R - r_h)$$

i.e.

$$b = \sin \theta_R \sqrt{Dh - h^2} \dots\dots\dots (6.A17)$$

The width of the block is constant throughout the segment so that to the left of the line of advance, the slew angle θ_L' at the bottom of the segment is greater than that for the top of the segment θ_L . Also by inspection:

$$r_h \cdot \sin \theta_L' = R \sin \theta_L + b$$

$$= (r_h + \sqrt{Dh + h^2}) \sin \theta_L + \sin \theta_R \sqrt{Dh - h^2}$$

i.e. $\theta_L' = \sin^{-1} \left(\sin \theta_L + \frac{1}{r_h} \sqrt{Dh - h^2} (\sin \theta_R + \sin \theta_L) \right) \dots\dots (6.A18)$

(6.A18) is valid up to $\theta_L' = 90^\circ$

As shown in Fig 6.A3 the "equivalent" segment drawn about the locus of G is nearer to the centre than E and hence its width to the right of the line of advance, $R_m \sin \theta_R$, is less than $R \sin \theta_R$. Therefore the width to the left of the line of advance $R_m \sin \theta_L$ is greater than $R \sin \theta_L$.

The outer edge of the equivalent segment is shown dotted and by inspection:

$$c = \sin \theta_R (R - R_m) = R_m \sin \theta_L'' - R \sin \theta_L$$

Then

$$\theta_L'' = \sin^{-1} \left[\frac{R}{R_m} \sin \theta_L + \sin \theta_R \left(\frac{R}{R_m} - 1 \right) \right] \dots\dots (6.A19)$$

θ_L'' is usually about 1° or more than θ_L and for most cases the angles can be considered equal.

To determine t' , three intervals are considered:

t_1 = time to slew from θ_L' to θ_L'' or

more usually θ_L' to θ_L

t_2 = time to slew from θ_L'' (or θ_L) to θ_m (see later)

t_3 = time to slew from θ_m to θ_R

The length of these intervals will depend on:

a) BWE size

b) bucket wheel drive power

c) type of slewing speed control

e.g. (i) manual

(ii) automatic to maintain constant

bucket wheel drive load and

(iii) proportional to $\frac{1}{\cos \theta}$

For (i) t' is largely dependent on the skill of the driver and cannot be accurately defined in mathematical terms.

For (ii) the slewing speed is inversely proportional to cutting resistance to maintain constant cutting force. Cutting resistances may vary widely throughout the segment and precise definition of t' is not possible. (See later - drop cut, which gives a close approximation for t' and provides reliable results for system (ii))

System (iii) is suitable for the terrace cut in materials of relatively uniform cutting resistance. Reasonably accurate estimates of t_1 , t_2 and t_3 can be made.

The slewing speed at the line of advance is V_s and the corresponding angular velocity is $\omega = \frac{V_s}{R_m}$

The segment height increases from zero at θ_L' to h at θ_L and a faster speed would be used than $\frac{1}{\cos \theta}$ speed. On faces where steep highwalls are possible, $\theta_L' - \theta_L$, may be $5^\circ - 8^\circ$, and an average slewing speed of $2V_s$ is indicated for this

zone. In the lower terraces where $\theta_L' - \theta_L$ may be $10^\circ - 20^\circ$, an average slewing speed of $3.5 - 4.0 V_s$ would be appropriate.

For constant output

$$V_{s\theta} = \frac{V_s}{\cos \theta} \quad \text{and} \quad \omega_\theta = \frac{\omega}{\cos \theta}$$

At slew angle θ , the time dt_2 to traverse $d\theta$ is: $dt_2 = \frac{d\theta}{\omega_\theta}$

and
$$t_2 = \frac{1}{\omega} \int_{\theta_L}^{\theta_m} \cos \theta d\theta$$

The slewing speed cannot be increased indefinitely and the angle at which maximum slewing speed occurs is designated θ_m , the corresponding angular velocity being ω_m .

θ_m is to the right of the line of advance, then:

$$t_2 = \frac{1}{\omega} (\sin \theta_m + \sin \theta_L) \quad \dots\dots\dots (6.A20)$$

t_3 occurs from θ_m to θ_R and obviously:

$$t_3 = \frac{\theta_R - \theta_m}{\omega_m} \quad \dots\dots\dots (6.A21)$$

If the slewing speed range for this segment is:

$$n = \frac{\omega_m}{\omega}$$

and since $t' = t_1 + t_2 + t_3$, then

$$t' = \frac{1}{\omega} \left[\frac{\theta_L' - \theta_L}{2} + \sin \theta_m + \sin \theta_L + \frac{\theta_R - \theta_m}{n} \right] \dots\dots\dots (6.A22)$$

Note n is not necessarily the overall slewing speed range of the BWE.

If the overall slewing speed range is large and θ_R is 80° or less, then θ_m may be greater than θ_R . In this case:

$$t' = \frac{1}{2V_s} \left[R_m (\theta_L' - \theta_L'') + 2S \right] \dots\dots\dots (6.A23)$$

The Drop Cut

The instantaneous output for a BWE operating in drop cut is:

$$Q' = A_{h\theta} \cdot T \cdot V_{s\theta} \cdot 60 \quad \dots\dots\dots (6.17)$$

and the effective hourly output is:

$$Q = u \left(\frac{\sum V_d \cdot 60}{\sum t} \right)$$

These formulae may be proved by methods similar to those used for the terrace cut, but as previously indicated the variations in segment cross section and volume for the drop cut are more complex than for the terrace cut. Fig 6. A4 is a scale drawing of a typical drop cut segment and block. Referring to Fig 6A2 it is sufficiently accurate to take the edge of the segment as O_2 rather than Q , and this line is represented by line $O-O'$, in Fig 6 A4. This line is horizontal and has radius R from slewing centre C_1 . The bucket wheel centre M , remains at distance r_h from C_1 and moves round the horizontal arc $M-M'$. As previously shown and from Fig 6A2 the radius R_d from the centroid G of the segment cross section to the slew axis increases with the angle of slew. The locus of G is shown by $G-G'$.

From formula (6.10) and by inspection of the drop cut diagrams, the distance from the lower edge of the segment to the slew axis increases with θ and this edge is represented by $E-E'$.

Considering element $O_2 O_2' E_2' E_2$ at slew angle θ , extending over small angle $d\theta$, its volume is:

$$dV_d = A_{h\theta} \cdot T \cdot R_{d\theta} \cdot d\theta$$

and the corresponding output rate is:

$$Q' = A_{h\theta} \cdot T \cdot R_{d\theta} \cdot \frac{d\theta}{dt}$$

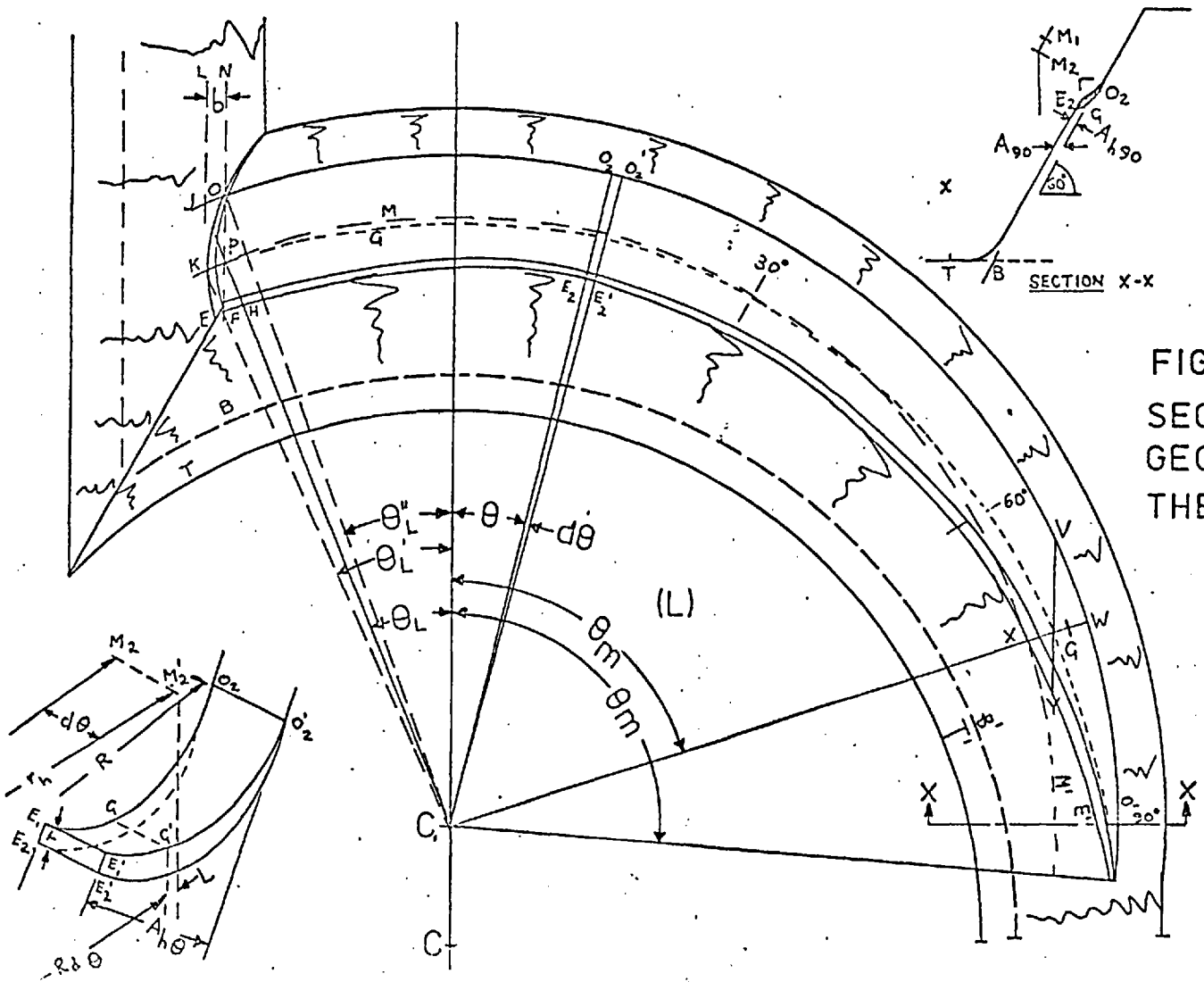


FIG 6.A4
SEGMENT
GEOMETRY.
THE DROP CUT

i.e.

$$Q' = A_{h\theta} \cdot T \cdot R_{d\theta} \cdot \omega_{\theta} \cdot 60 \quad \dots\dots\dots(6.A24)$$

Now if

r = overall ratio between slewing motors
and super structure

N_{θ} = slewing motor rev/min at slew angle θ

then

$$\omega_{\theta} = \frac{2\pi N_{\theta}}{r} \quad \text{rad/min}$$

Thus

$$Q' = \frac{2\pi \cdot T \cdot 60}{r} (A_{h\theta} \cdot R_{d\theta} \cdot N_{\theta}) \quad \dots\dots\dots(6.A25)$$

and $N_{\theta} = \frac{Q' r}{2\pi \cdot T \cdot 60 \cdot A_{h\theta} \cdot R_{d\theta}}$

The time dt to traverse $d\theta$ is

$$dt = \frac{d\theta}{\omega_{\theta}}$$

i.e. $dt = \frac{r \cdot d\theta}{2\pi \cdot N_{\theta}}$

and

$$t = \frac{r}{2\pi} \cdot \int_{\theta_L}^{\theta_R} \frac{d\theta}{N_{\theta}} \quad \dots\dots\dots(6.A26)$$

The value of Q' from (6.A25) is the same as that from (6.17) but the terms used here emphasise the influence of slewing speed on output.

As for the terrace cut, the average hourly output throughout the segment is:

$$Q_a = \frac{1}{t} \sum Q' dt$$

For a given segment once A_h, T, α and r_h are selected, $A_{h\theta}$ and $R_{h\theta}$ will vary as previously shown, (6.10), etc.

Since r is a fixed ratio throughout the segment, it follows that apart from the geometry of the BWE, actual output depends upon the variation of slewing speed. The same methods (i), (ii) and (iii) are adopted as for the terrace cut. For the drop cut the constant load system b), may be expected to give higher outputs (see later).

Fig 6 A4 shows O as the intersection between the top of the segment and the highwall and FON as a horizontal line along the highwall, parallel to the line of advance. P is the intersection of ON and the vertical through M and:

$$R \sin \theta_L'' = r_h \sin \theta_L$$

For normal values of A_h and α , the difference between $R_{d\theta}$ and r_h is less than 1.5% for normal values of θ_L and hence lines M-M' and G-G' would be close together at P.

Then:

$$r_h \cdot \sin \theta_L = R_{d\theta_L} \sin \theta_L$$

OKE projects the junction between the highwall and the segment, the junction approximating towards an ellipse, the ovality depending on D , α and θ_L . The extreme angle of slew θ_L' is that between the line of advance and the tangent drawn from C_1 to OKE. Calculation of this tangent point is complex and adequate accuracy is provided by taking it as K, the point where the bottom of the wheel cuts through the highwall.

If horizontal line KL is drawn along the highwall, the vertical distance of K below O is

$$d' = \frac{D}{2} - \frac{D}{2} \cos \alpha$$

and the distance L to N normal to the line of advance is:

$$b = d' \cot \alpha$$

$$\text{and } b = \frac{D \cot \alpha}{2} (1 - \cos \alpha)$$

$$\text{also } r_h \sin \Theta_L' = r_h \sin \Theta_L + b$$

$$\Theta_L = \sin^{-1} \left(\sin \Theta_L + \frac{b}{r_h} \right) \dots\dots\dots (6.A27)$$

Fig 6 A4 shows line OF as a vertical plane passing through the segment at slew angle Θ_L , VY being a similar plane through Θ_m .

By inspection, the segment width S' is the distance normal to the line of advance between the two vertical planes. Then if the values of $R_{d\Theta}$ at Θ_L and Θ_m are R_{dl} and R_{dm} respectively, S' becomes:

$$S' = S_L + S_m$$

where

$$S_L = R_{dl} \sin \Theta_L \quad \text{and}$$

$$S_m = R_{dm} \sin \Theta_m$$

The volume of this portion of the segment is then:

$$V_d = A.S'.T \sin \alpha \dots\dots\dots (6.A28)$$

To the left of OF the volume is less than 1% of the full segment volume and is neglected here.

Formula (6.17) is based on a segment volume delineated by radial planes, and here these are represented by $C_1 J$ at Θ_L , and $C_1 W$ at Θ_m . As an alternative to the complex integrations, formulae (6.A13) and (6.A14), the following close approximations can be used for calculating the segment

volumes between radial planes.

Fig 6.A4 shows lines M-M' and G-G' at Θ_L almost in coincidence and the volume of segment between planes PH and PF is almost equal to that between planes PO and PJ. Similarly at Θ_m , the volume between G'X and G'Y is almost equal to the volume between G'W and G'V.

The volumes between radial planes may thus be taken as the same as that between planes parallel to the line of advance. Then:

$$V_m = A_h \cdot T(R_{dl} \sin \Theta_L + R_{dm} \sin \Theta_m) \dots\dots\dots(6.A29)$$

V_m and Θ_m are also used to designate the volume of the segment and slew angle at which maximum slewing speed occurs.

The time to traverse the segment is estimated in a similar manner to that used for the terrace cut:

$$t_1 = \text{time to slew from } \Theta'_L \text{ to } \Theta_L$$

$$t_2 = \text{time to slew from } \Theta_L \text{ to } \Theta_m$$

$$t_3 = \text{time to slew from } \Theta_m \text{ to } \Theta_R$$

As for the terrace cut

$$t_1 = \frac{1}{a \omega} (\Theta'_L - \Theta_L)$$

For angles up to 25° , the average slew speed $a\omega$ may be taken as 1.5ω . For values of Θ_L above 25° , 2ω is appropriate.

If the slewing motor speed is directly proportional to $\sec \Theta$ then as for the terrace cut:

$$t_2 = \frac{1}{\omega} \int_{\Theta_L}^{\Theta_m} \cos \Theta \cdot d\Theta$$

and

$$t_2 = \frac{1}{\omega} (\sin \theta_m + \sin \theta_L) \dots\dots\dots(6.A30)$$

Again as in the terrace cut

$$t_3 = \frac{\theta_R - \theta_m}{\omega_m}$$

If $n = \frac{\omega_m}{\omega}$ then $t_1 + t_2 + t_3$ is :

$$t' = \frac{1}{\omega} \left[\frac{\theta'_L - \theta_L}{a} + \sin \theta_m + \sin \theta_L + \frac{\theta_R - \theta_m}{n} \right] \dots\dots(6.A31)$$

This indicates that for the same slewing limits θ_L and θ_R , the slewing speed being controlled by the "cosine" system, that the cutting time would be the same as for the terrace cut. This is theoretically correct but the output rates are not the same and using the drop cut where θ_m is greater than 60° , the large increase in instantaneous output and the corresponding increase in bucket wheel drive load, requires restrictions in slewing speed. These increases are obvious from Fig 6 A4 where G-G₁ diverges from M-M₁ after 50° .

Fig 6 A4 also shows the extent to which G lies beyond the vertical through M at higher values of θ (section X-X, $\theta = 90^\circ$, where $A_{h\theta}$ has an appreciable value) and also how the distance of E₂ below O₂ causes the horizontal projection of the segment to be greater than A_Q.

FURTHER WORK

Alternative empirical solutions are available due to W.D. Scott and Co., "Consultant's Report to the State Electricity Commission of Victoria, Analysis of Bucket Wheel Dredger" March, 1958, with modifications by H.C.G. Rodgers, "Developments in Equipment used for Overburden Removal and

Coal Winning in the Brown Coal Industry" Proc. Aus. I.M.M., No.19⁴, 1960. In addition to the foregoing analyses however further mathematical investigation is indicated, for although the output of a BWE is largely dependent on the power of the bucket wheel drive, the effective hourly output is clearly influenced by the geometry of the BWE and its bucket wheel, and slewing speed is of vital importance.

The rotation of the bucket wheel about its own axis coupled with the slewing motion generates segments of complex shapes. The simplified formulae derived here are generally of sufficient accuracy, within the limits given, while the rigorous solutions are too complex and unwieldy for general application. Much depends on the utilisation factor, u . A reliable estimate of u is difficult, requiring a detailed examination of the block being dug, the BWE characteristics and an assessment of driver skill. Field observation appears to be the most readily accessible method of determining values of u . The author's experience indicates that it seldom exceeds 0.7.

The foregoing derivations applied to BWE's in service indicate that where a range of cutting speeds are required for varying conditions, that slewing speed ranges of up to 10 to 1 are required.

The constant load system of automatic slewing speed control appears to give a higher potential output than the cosine system for most types of operation.

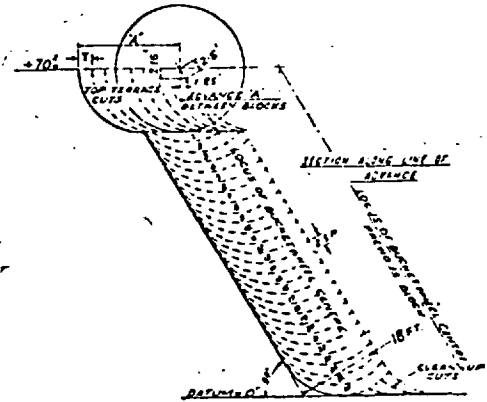
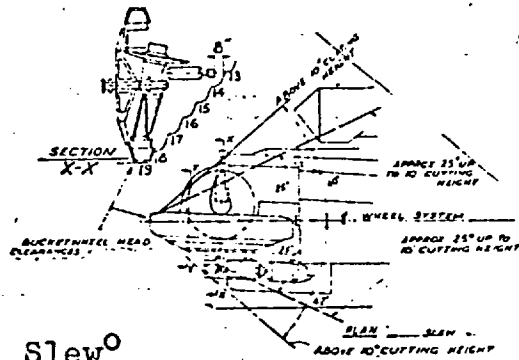
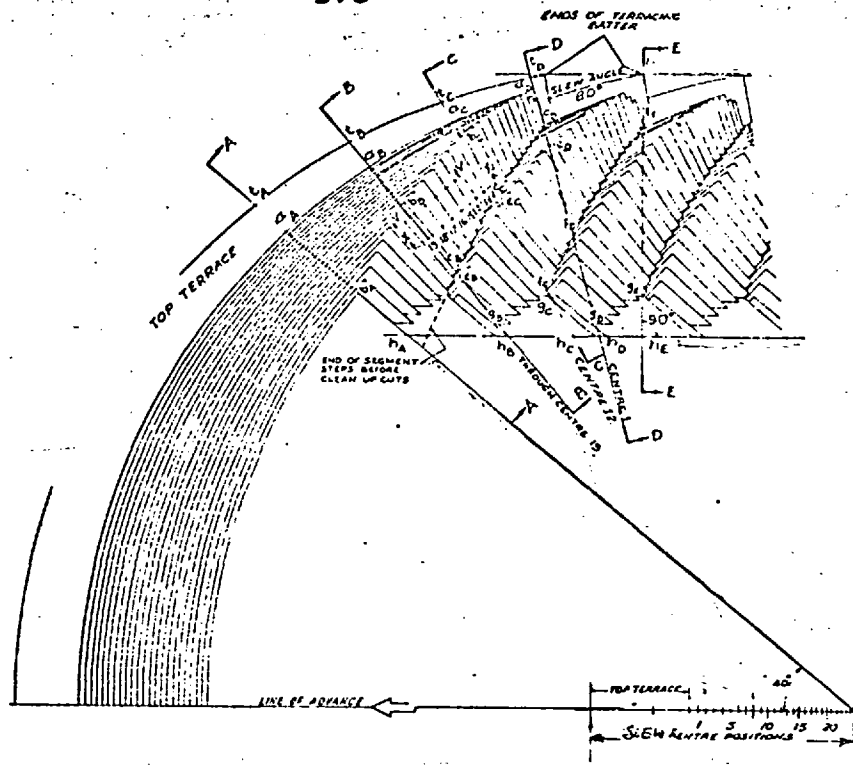
A clearer understanding of the operation, based on mathematical analysis could result in improved output, due to more efficient operation.

APPENDIX 6.B

BUCKET WHEEL HEAD GEOMETRY

Where a bucket wheel must cut a relatively flat highwall, the bucket wheel head clearances become all important. These may be determined by the production of scale drawings, but as the bucket wheel head geometry is not a simple configuration and the two dimensional aspect of the scale drawings must be related to three dimensions, considerable draughting skill as well as patience is required. Fig 6B1 shows typical cross sections for a drop cut for a BWE of LMG manufacture. The interpretation of such cross-sections also requires considerable skill and a deep practical knowledge of BWE operations. A further problem is that the cross sections do little to help BWE drivers to understand a proposed operation.

One major problem is that most BWEs are manufactured in Germany where steep highwalls are possible and the problems of bucket wheel head clearance were not fully understood. Outside Germany it became evident that the development of wheel head clearances and the interpretation of complex face cross sections would be greatly aided by the use of scale models. Fig 6B2 shows diagrammatically a wooden model with a true-to-scale bucket wheel head mounted on an adjustable boom, describing a terrace cut. The bench surfaces are covered with a thick layer of plasticene which is indented by the bucket wheel periphery. In this way a three dimensional scale model of the operation can be prepared.



Segment	Slew ⁰
1	75
2	74
3	73
4	72
5	70.5
6	69
7	68
8	66.5
9	65
10	64
11	62.5
12	61
13	59.5
14	58
15	57
16	55.5
17	54
18	52
19	50.5
20	49
21	47.5
22	46
23	44.5
24	43
25	41.5
26	40

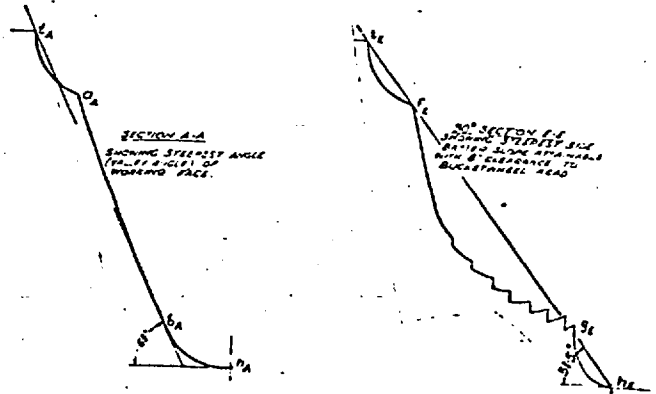


FIG 6B1 CROSS SECTIONS FOR A DROP CUT

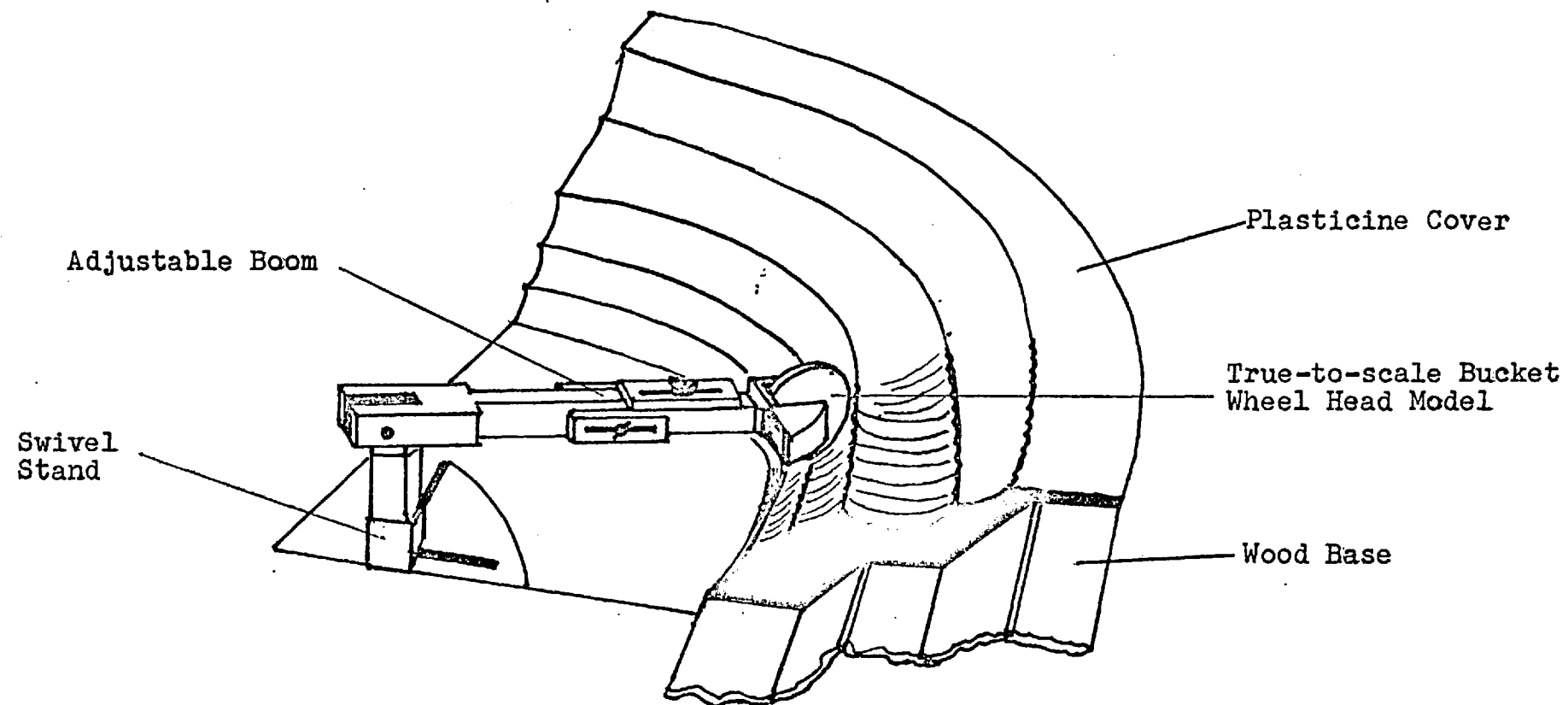


FIG 6B2 SCALE MODEL FOR DETERMINATION OF BUCKET WHEEL HEAD GEOMETRY

It is evident from Fig 6B1 that the bucket wheel diameter has a major influence on bucket wheel head clearances. In addition however the bucket wheel boom must be considered since the arrangement of the gearbox and walkways can affect clearances. The end of the bucket wheel boom conveyor can also cause problems. When the boom is lowered and the conveyor angle is increased, the vertical inclination of the chute plate across the end of the conveyor is reduced. This can cause congestion in the chute and a reduction in excavation rate becomes necessary. This can be overcome by extending the conveyor tail drum out beyond the bucket wheel centre, but this obviously increases the bucket wheel head clearance angle. This difficulty can be overcome to some extent by tilting the bucket wheel through a small angle, a point which would be difficult to establish without the use of a scale model.

7. MOBILE EQUIPMENT

Mobile earthmoving equipment originally developed for the construction industry has found considerable application in mining operations during the past ten years.

The major types of equipment are:

front-end loaders

tractor-scrapers

bulldozers

hoe excavators

rippers

compactors, etc.

In general their use has been limited to soils and the weaker, less consolidated rocks, but the introduction of inexpensive explosives, e.g. ANFO has greatly extended their range and there is some overlap between their application and that of conventional loading equipment.

DEFINITIONS

The following definitions refer to mobile equipment.

GVW Gross vehicle weight

RR - Rolling resistance - is a measure of the force that must be overcome to pull or roll a wheel over a surface. It is affected by ground conditions and vehicle load - the deeper the vehicle sinks into the ground the higher the rolling resistance. Experience has shown that a minimum resistance of 20 kgf/tonne (40 lbf/ton) on the wheels must be overcome to move a rubber tyred machine. It has also been found that for every 25 mm of tyre penetration an additional 15 kgf/tonne (30 lbf/ton/in) must be overcome 71, 72, 73.

These values combined give:

Rolling Resistance Factor (RRF) =

$$20 \text{ kgf/tonne} + \left[(15 \text{ kgf/tonne/25 mm}) \times (\text{tyre penetration in mm}) \right]$$

$$(40 \text{ lbf/ton} + \left[(30 \text{ lbf/ton/in}) \times (\text{tyre penetration in inches}) \right])$$

Then the Rolling Resistance =

$$RR = RRF \times GVW \quad \dots\dots\dots (7.1)$$

expressed in kgf or lbf.

Another method is to express RR as a percentage of GVW i.e.

$$RR = 2\% \text{ GVW} + 0.06\% \text{ per mm tyre penetration}$$

$$(RR = 2\% \text{ GVW} + 1.5\% \text{ per inch tyre penetration}).$$

It should be noted that it is not necessary for the tyres to actually penetrate the road surface for RR to increase above the minimum. If the road surface flexes under load, the effect is nearly the same i.e. the tyre is running "uphill". Only on very hard, smooth surfaces with a well compacted base will RR approach the minimum.

TABLE 7.1

ROLLING RESISTANCE FOR RUBBER-TYRED TRACTORS AS
A PERCENTAGE OF GROSS VEHICLE WEIGHT

(Ref. 71, 72, 73)

Description	% GVW*
Hard pavement, no penetration (concrete or blacktop)	2
Flexible pavement (Macadam or pack gravel construction)	3 - 3.5
Dirt road, flexing considerably (maintained)	5
Unmaintained dirt road (no compaction)	7.5
Loose sand or gravel	10
Soft muddy rutted road	15 - 20

*Based on low pressure (short haul) tyres and anti-friction bearings as used for loading equipment.

N.B. Crawler-tracked machines utilise steel wheels rolling on a "prepared steel road". The term "rolling resistance" is not usually applied since it is relatively (but not absolutely) constant and is accounted for in fixing the "Draw-bar Pull" rating of the machine.

Grade Resistance is the force that must be overcome to move a machine over adverse gradients (uphill).

Grade Assistance is the force that assists machine movement on favourable gradients (downhill). Gradients (or grades) are expressed as, 1 in 10 or 10% (% grade) i.e. a 10m rise or fall for every 100m of horizontal distance. Then:

$$1 \text{ in } 100 = 1\% \text{ grade}$$

$$1 \text{ in } 50 = 2\% \text{ grade}$$

$$1 \text{ in } 20 = 5\% \text{ grade}$$

$$1 \text{ in } 10 = 10\% \text{ grade}$$

For a 1 in 20 gradient

The grade resistance or assistance is:

$$= \frac{GVW}{20} \dots\dots\dots (7.2)$$

$$\text{or} = \frac{GVW}{100} \times \% \text{ grade} \dots\dots\dots (7.3)$$

Total Resistance is the combined effect of rolling and Grade resistance i.e.

$$\begin{aligned} \text{Total resistance} &= \text{Rolling resistant} + \\ &\quad \text{Grade resistance} \dots\dots (7.4) \end{aligned}$$

Traction is the driving force developed by a wheel or crawler track as it acts on a surface. It is expressed as usable drawbar pull for a crawler-tracked tractor unit or rimpull for a rubber tyred wheel. The following factors affect traction:

- a) weight on driving wheels or crawler tracks.
- b) ground conditions.
- c) gripping action of wheel or crawler track.

The coefficient of traction is the ratio of the maximum pull developed by the machine to the force due to the machine mass on the drivers (wheels or crawler track).

Therefore the tractive effort (TE) for a given machine is

TE = Coeff. of Traction x weight on drivers.

Approximate coefficients of traction are shown in Table 7.II.

TABLE 7.II
APPROXIMATE COEFFICIENTS OF TRACTION
(Ref. 74, 75, 76, 77.)

Surface Description	<u>Coefft. of Traction</u>	
	<u>Rubber Tyres</u>	<u>Crawler Tracks</u>
Concrete	0.90	0.45
Clay, loam, dry	0.50-0.58	0.90
Clay, loam, wet	0.40-0.49	0.50-0.80
Rutted clay loam	0.40	0.70
Dry sand	0.20	0.30
Wet sand	0.40	0.50
Gravel road (loose)	0.35	0.40-0.50
Open pit rock floor	0.60-0.70	0.45-0.55
Packed snow	0.20	0.25
Ice	0.12	0.12*
Firm earth	0.50-0.60	0.90
Loose earth, dry	0.40-0.50	0.60
Loose earth, wet	0.40	0.60
Macadam, dry	0.70	-
Macadam, wet	0.65	-

*Semi-skeleton shoes 0.27

Drawbar pull (dbp) is the pull exerted by a crawler-tracked machine. It is the maximum effective pull that can be exerted under a given set of conditions. It should be noted that:

dbp = Traction - effort to drive tractor ,

Manufacturers usually rate the drawbar pull on effort to drive the tractor equivalent to 5.5% RR, which is conservative for a crawler-tracked vehicle.

Most open pit mining applications use "rocker" type, rubber-tyred tractor units in tractor-scraper operations and the use of crawler-tracked tractors for towing duties is now uncommon. The term "drawbar pull" is not therefore in common use in open pit mining operations.

Rimpull (R_p) is the maximum force which the rubber tyres of a unit can exert on a surface on which they are travelling. It is a function of the output of the power unit, the efficiency of the transmission system and the operating speed.

$$R_p = \frac{3600 P \eta_g}{v} \quad \text{Newtons} \quad \dots\dots\dots (7.5)$$

$$R_p = \frac{368 P \eta_g}{v} \cdot \text{kgf} \quad \dots\dots\dots (7.6)$$

where

p = engine power output in kW

η_g = transmission system efficiency

v = speed in km/h

$$R_p = \frac{375 P \eta_g}{v} \cdot \text{lb} \cdot \text{f} \quad \dots\dots\dots (7.7.)$$

where

p = engine power output in hp

η_g = transmission system efficiency

v = speed in mile/h

Although a rubber tyred machine is capable of producing a certain R_p it may be limited by the "coefficient of traction" between the tyres and the ground surface.

(See Appendix 7A for graphical treatment).

Capacity

Struck Capacity is that volume contained in a bucket or bowl after the load is levelled by drawing a straight edge, resting on the cutting edge and the top of the spill plate across the bucket or the equivalent surface of the bowl of a tractor-scraper.

Heaped Capacity is the struck capacity plus that additional material that would heap on the struck load at a 2 : 1 angle of repose with the struck line parallel to the ground.

AIR RESISTANCE

This is only significant for units travelling in excess of 60 km/h (35 mile/h). Such speeds are seldom achieved by mobile equipment, except possibly tractor-scrappers on long hauls over good surfaces. The effect of wind resistance is ignored here but is dealt with in Appendix 7A.

CRAWLERS V. RUBBER TYRES

The following summarises the features of crawler propelled and rubber-tyred mobile equipment.

Crawler-mounted

Crawler-mounted units have the following features.

- (1) A strong digging capability as a front-end loader or as a tractor-scraper.
- (2) Relatively low ground bearing pressure; excellent tractive effort in bad ground conditions.
- (3) Performance on steep inclines is good.
- (4) Degree of manoeuvrability is high.
- (5) Travel speeds are relatively slow - can only be economically used where the distances between digging and discharge points are short.

- (6) Maintenance costs in abrasive materials are relatively high.

Rubber-tyred

Rubber-tyred units have the following characteristics.

- (1) They are very mobile.
- (2) Maintenance costs in easy to medium conditions are relatively low.
- (3) Tyre costs are high where sharp, broken or abrasive rocks are encountered, e.g. basalt, taconite, angularly broken schists, etc.
- (4) They are less manoeuvrable than crawler-mounted units, but centre-articulated units are still highly manoeuvrable in comparison with conventional excavating and loading equipment. They are well suited for operation on narrow benches. Four-wheel-steered units have reasonable manoeuvrability, whereas two-wheel-steered units have poor manoeuvrability.
- (5) Travel speeds are relatively high: they can therefore economically dig and transport excavated material over longer distances than crawler-mounted machines. This feature has been responsible for the development of rubber-tyred units as 'load-haul-dump' machines. e.g. front-end loader and tractor-scraper.
- (6) Ground bearing pressures are relatively high. The machines are not efficient in bad conditions where rolling resistance is high.

TYRE SELECTION

Considerable research has been undertaken by tyre manufacturers, particularly in the U.S.A., and a very wide range of tyre types is available. In view of the importance of tyre maintenance and replacement costs, it is surprising that many mine operators fail to keep accurate records of, or make detailed investigations into, tyre life. Undoubtedly, the best selection and costing procedures are based on past records. The factors to be considered are surface conditions, loading, speeds, curves, grades, temperature, maintenance standards, etc.

In general, the following tyre types are used.

Long haul - Earthmoving tyres: heating is a problem and, to reduce heat generation, suitable rubber compounds are selected with standard tread and wall thickness.

Short haul with medium - difficult rock conditions -

Deep tread tyres: tread depth increased by 50 per cent, undertread and side thicknesses increased by 100 per cent over earthmoving tyres. Wear-resistant rubber compound selected for rock conditions to reduce tyre damage, i.e. to make tyre denser, tougher and more abrasion-resistant.

Short haul with difficult - extreme rock conditions -

Extra-deep tread tyres. Tread depth increased by 150 per cent, undertread and side-wall thicknesses increased by 100 per cent over earthmoving tyres. Wear-resistant rubber compound selected to reduce tyre damage.

For very abrasive rocks in dry conditions a high solids to voids ratio of the tyre tread should be used to

provide more wearing material. In extreme cases an entirely solid tread may be necessary, the only void being a wear gauging slot on the edge of the tyre side wall.

Table 7.III shows the average life of tyres used in their correct applications. Remould tyres, have about 80% of the life of new tyres, and usually three remoulds are possible during the life of one tyre.

TABLE 7.III

TYRE LIFE

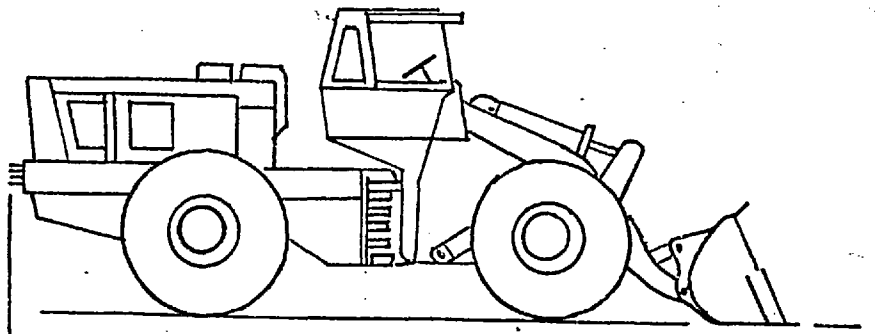
	Job conditions			
	A	B	C	D
Average life, operating hours	4000- 5000	3000- 3500	2000- 2500	400- 1500

- A, Good haul roads with operation over soft, non-abrasive rocks.
- b, Clays, marls, soft shales, wet gravels, etc., with average haul roads.
- C, Medium hard rocks, e.g. hematite, abrasive shales, etc., with short transport distances in average bench conditions.
- D, Severe conditions, taconite, angularly fragmented basalt, etc., with poor bench surfaces.

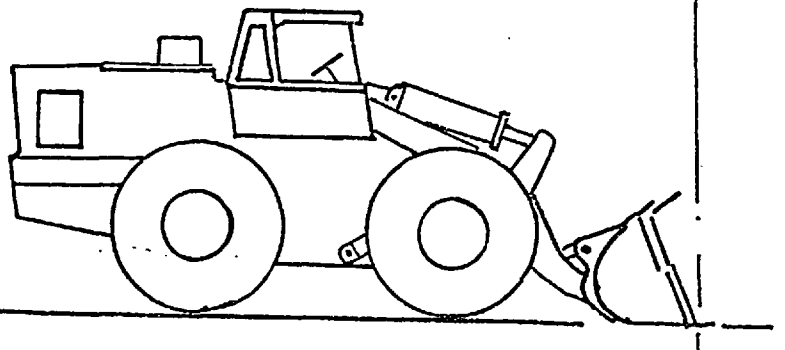
Further details of tyre nomenclature, loading and construction are given in Appendix 7.B.

THE WHEEL LOADER OR FRONT END LOADER (Fig 7.1)

During the past ten years rubber-tyred wheel loader (front-end loader) applications have greatly extended in open pit mining. Units with rock buckets up to 11 m^3 (15 yd^3) are in service and a design with a 27 m^3 (35 yd^3) bucket is



a) The Articulated
Wheel Loader



b) The Rear-wheel Steer
Wheel Loader

FIG 7.1 ARTICULATED AND REAR-WHEEL STEER
WHEEL LOADERS (Front-end Loader)
WITH SAME BUCKET CAPACITY.

under development, but in many metalliferous mines the wheel loader has been mainly restricted to overburden stripping and auxiliary duties. Compared with loading shovels they are short life machines not greatly affected by obsolescence. Many observers forecast increased acceptance of wheel loaders in face loading ⁷⁹, ⁷¹⁰, and larger machines capable of handling rock will undoubtedly be developed to take advantage of the following features:

1. Mobility - the wheel loader can quickly travel from one part of a pit to another, move out of blasting zones, etc.
2. Comparatively low capital costs - the correct selection procedure is to compare different machines by using a discounting method, but the wheel loader becomes attractive in situations of uncertainty, e.g. political, economic, etc. where capital expenditure must be restricted.
3. Operational flexibility - wheel loaders can perform many auxiliary operations, e.g. haul road construction and maintenance, drainage and bunding operations, ripping, truck "boosting" on inclines, etc.

There has been a marked trend to larger machines in the past 10 years, mainly due to the introduction of the centre-articulated unit. Most early front end loaders were rear-wheel steered, having a shorter wheel base than the articulated unit but despite this the articulated unit has a smaller turning radius (see Table 7.IV). Additionally stability is improved, tyre life is increased due to absolute wheel tracking and spotting the bucket for dumping

is easier since the front part of the machine is moved.

TABLE 7.IV

WHEEL LOADERS - TYPICAL DATA

Bucket	m ³	0.77	1.5	2.5	3.85	4.6	7.7
Size (SAE)	yd ³	1	2	3½	5	6	10
Operating	R	5-6	9-11	13-15	20-22	-	-
Weight (t)	A	-	-	13-16	28-31	38-42	50-56
Engine	R	65 - 70	100-120	140-160	260-280	-	-
Power (hp)	A	-	-	150-180	280-300	350-400	400-500
Turning	R	5.5-6.0	6.0-7.0	7.0-8.0	8.0-10.0	-	-
Radius (m)	A	-	-	6.5-7.0	7.5-9.5	8.0-11.0	8.5-12.0
Wheel Base	R	1.9-2.0	2.0-2.5	2.5-2.6	2.8-3.0	-	-
(m)	A	-	-	2.5-3.4	3.5-3.8	4.2-4.5	4.5-4.8
Operating weight	R	6 -7.5	10-11	12-15	16-18	-	-
Bucket width (ton/cm)	A	-	-	12-15	23-25	25-30	30-36

R - rear wheel steering

A - articulated steering

In Table 7.IV, the specific digging force

$\frac{(\text{crowd force})}{(\text{bucket width})}$ is represented

by $\frac{(\text{operating weight})}{(\text{bucket width})}$. As the crowd force is

approximately equal to the operating weight of the machine multiplied by the coefficient of traction, this more readily stated comparison is valid. It should be noted that the specific digging force increases with machine size. This indicates that the larger the machine the greater the

digging capability and accounts for the increasing use of wheel loaders in rock conditions, once considered the exclusive field of the loading shovel.

In the field of open pit mining the front-dumping bucket has found almost universal application. Where bench widths are restricted to obtain steep slopes in deep pits, e.g. Pennsylvania anthracite field, the side-dumping bucket is more economic in space. Fig 7.2 shows the width required for loading with a side-dumping bucket. The side-dumping bucket is heavier, more expensive and has a narrower dumping width, than the front-dumping bucket. Because of its narrow dumping width the side-dumping bucket can be spotted very accurately and speedily. The side-dumping bucket has usually a fillability of only 0.75 - 0.85 of that of the equivalent front-dumping bucket, due to the absence of the bucket sides.

Wheel Loaders can travel on level ground at up to 26 km/h (16 mile/h). For distances up to 200m (650 ft) it can therefore be an economic "load-haul-dump" unit, eliminating the need for haulage units.

Production Rate

Because of its mobility, the most important factor in determining the production rate of a wheel loader is the operating load (bucket load) rather than bucket capacity, since if a wheel loader operates at maximum load when handling a light material, it will be under-powered and unstable if transferred to handling a dense material. Bucket capacity has been defined by the Society of Automotive Engineers ⁷¹¹. In determining bucket capacity, the bucket load being known, the struck volume of the bucket should be used, as most materials do not heap during the loading process.

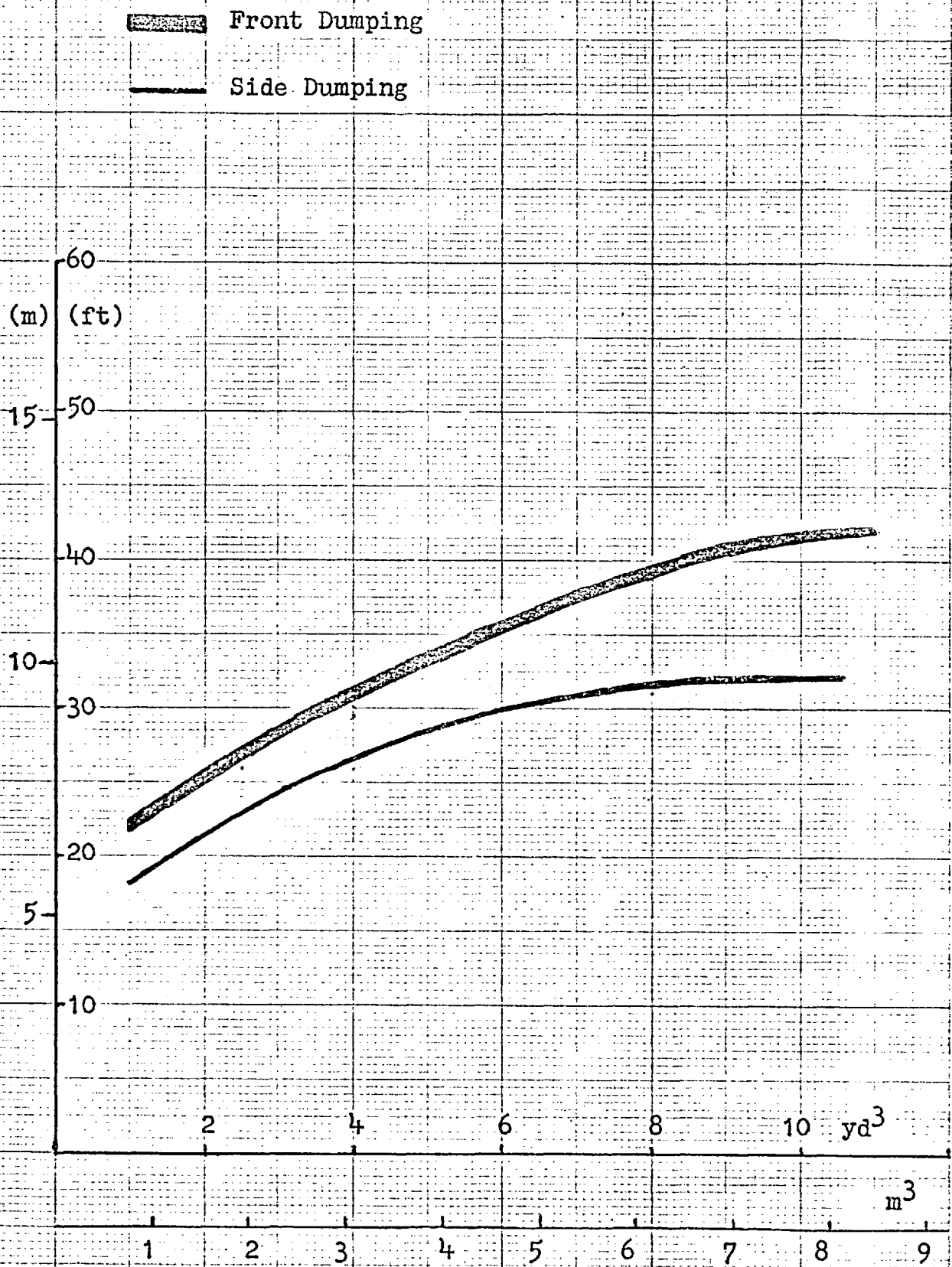


FIG 7.2 TOTAL WIDTH REQUIRED FOR WHEEL LOADER OPERATIONS
(Loader + Truck. Based on truck filled in 5
bucket loads. NB Widths must be checked during
tender analysis.)

For granular materials, e.g. sand, gravel, etc., a fillability approaching unity is normal, but for other materials the fillability factors in Table 2.I should be used.

The bucket capacity is:

$$B_c = \frac{\text{Operating Load}}{\text{Loose density} \times \text{Fillability}} \dots\dots\dots (7.8)$$

A further important factor is the stability of the wheel loader when travelling due to the combination of suspension movement and rough ground. The static tipping load of a wheel loader is defined as ⁷¹¹:

"The minimum weight at the centre of gravity of 'SAE Rated' load in bucket which will rotate rear of machine to a point where,----- on wheel loaders, rear wheels are clear of the ground under the following conditions:

- a) loader on hard level surface and stationary
- b) unit at standard operating weight
- c) bucket tilted back
- d) load at maximum forward position during raising cycle
- e) Unit with standard equipment as described in specifications unless otherwise noted under the heading."

In order to comply with SAE standards, the operating load of wheel loaders should not exceed 50% of the static tipping load.

The operating load should then be:

$$\text{Operating load} \leq \text{Tipping load} \times s \dots\dots\dots (7.9)$$

where s = stability factor

Table 7.V indicates typical values for s , for various conditions, based on vehicle stability calculations⁷¹² and tests carried out by C.V.M. Brown and the author.

TABLE 7.V

STABILITY FACTOR " s " FOR WHEEL LOADERS

Surface Conditions	Travel Speed			
	mile/h	0	8	16
	km/h	0	13	26
Smooth level hard surface		0.50	0.40	0.35
Rough ground mostly level; or up to 20% grade smooth hard surface		0.40	0.30	0.25
Rough ground up to 20% grade: or up to 40% grade smooth hard surface		0.25	0.15	0.0

The operating load should not exceed these values unless the tyres are ballasted, by the addition of counterweights or additional attachments, e.g. ripper, etc., in accordance with the manufacturer's specification.

The following steps are followed in estimating the production rate.

1. Fix the production rate
2. Determine the loader cycle time and the theoretical cycles per hour.
3. Allow for the Operating Efficiency
4. Determine the operating load per cycle in m^3 (yd^3) and kg (lb).
5. Determine the bucket size required for the loose density of the excavated material.

6. Select the machine that meets the bucket size and stability criteria.

The operating cycle is made up of:

- a) loading
- b) manoeuvring (four reversals of direction, full hydraulic cycle and minimum travel)
- c) hauling
- d) dumping, and
- e) return

The loading, dumping and manoeuvring times are combined as the "fixed time", and hauling and return as the "variable time".

The fixed time may be based on time studies or, in the absence of experience, values may be taken from Table 7.VI for a variety of digging conditions. (See Appendix 7C for a further method)

TABLE 7.VI

FIXED TIME - WHEEL LOADERS*

Bucket capacity		Fixed time, sec			
yd ³	m ³	E	M	M-II	H*
5	4	32	33	41	-
6	4.5	33	34	42	-
7	5.5	33	35	44	-
10	7.5	37	39	51	-
12	9	39	42	56	-
15	11.5	41	44	60	-

*The application of wheel loaders in 'hard' digging conditions is marginal and requires comprehensive investigation.

*Based on observations made at a large number of operations, published data, etc., with normally competent operators.

The variable time may be determined from time studies or from manufacturers figures taking into account the gradients, underfoot conditions, etc. Most manufacturers provide estimated haul time curves for various gear ratios. It should be noted that the top gear of most wheel loaders is primarily used for propelling the unloaded machine on a firm surface and is not usually operated for loading duties. A typical manufacturers curve is shown in Fig 7.3. In the absence of experience the variable time may be obtained from Fig 7.4 which is compiled from a large number of operations.

The operating load of a wheel loader can be calculated from:

$$\text{Operating load} = \frac{\text{Production Rate}}{C \times A \times O} \dots\dots\dots (7.10)$$

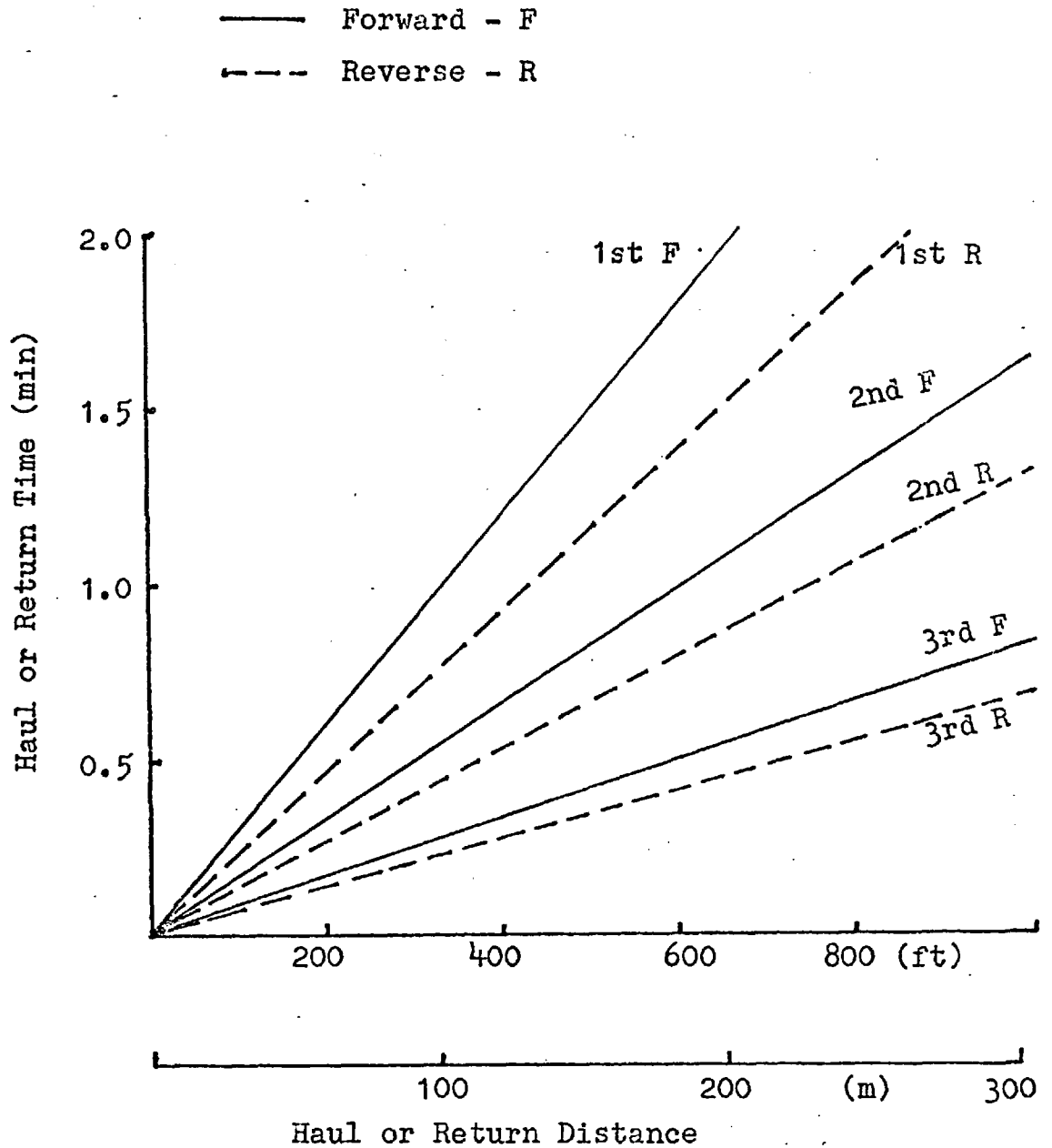
C - Theoretical Cycles per hour

$$C = \frac{60}{\text{Fixed time(min)} + \text{Variable time(min)}} \dots\dots (7.11)$$

A - Availability The mechanical availability during scheduled hours may be obtained from plant records or by industrial engineering methods.

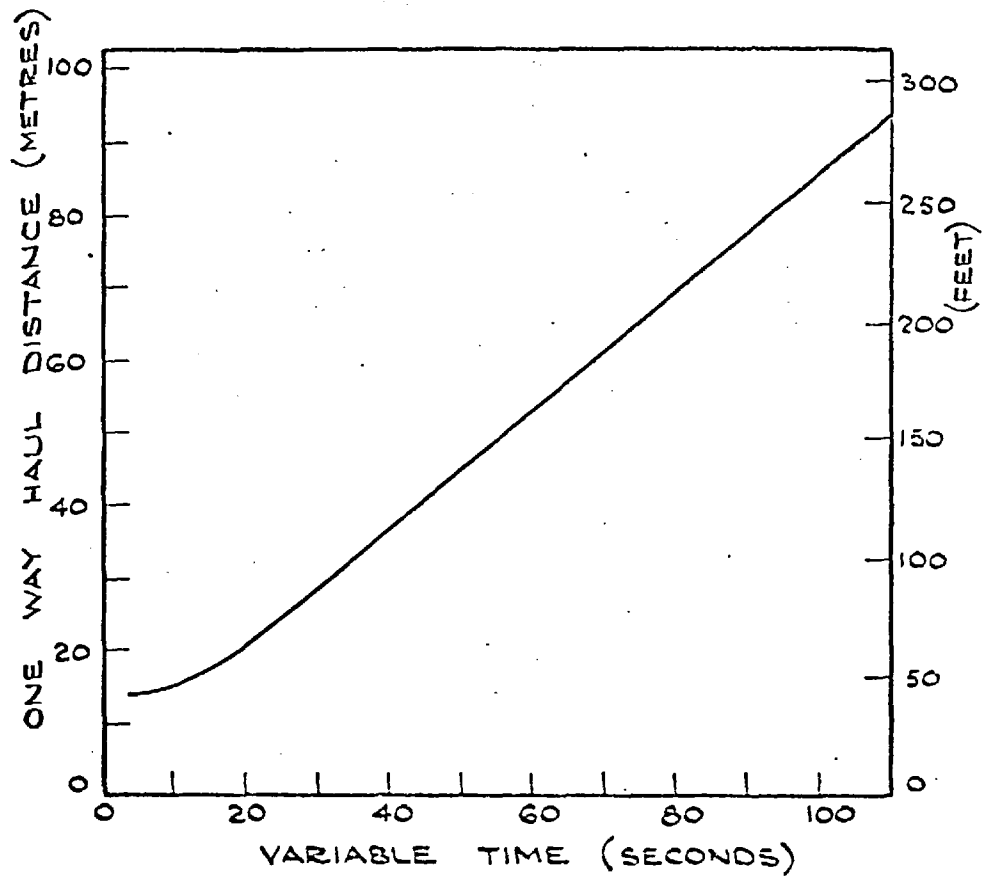
O - Job Operation factor This may be obtained by similar methods.

AO - Where no previous experience is available to determine A and O, their product AO may be obtained from Table 2.VI.



N.B. Load, manoeuvre and dump times must be added to travel time to obtain cycle time
4th gear only used for transporting machine.

FIG 7.3 TYPICAL ESTIMATED HAUL OR RETURN TIMES FOR
A WHEEL LOADER (980B Cat)



(Based on large number of time studies)

FIG 7.4 VARIABLE TIME - WHEEL LOADERS

Wheel Loader Costs

Ownership Costs The economic life of a wheel loader depends on the conditions of service, the skill of the operator, the standards of maintenance and management supervision.

Table 7.VII prepared on the same basis as Table 7.VI provides a guide to the life in scheduled hours where no previous experience is available.

TABLE 7.VII
ECONOMIC LIFE - WHEEL LOADER
(Scheduled Hours)

Job conditions	Management conditions			
	Excellent	Good	Fair	Poor
Excellent	12000	11500	11000	10000
Good	11000	10500	10000	9000
Fair	10500	10000	9000	8000
Poor	9000	8500	8000	7000

If the duty involves medium long transport distances over good surfaces on non-abrasive materials on relatively flat grades and easy digging conditions, with highly skilled operators, the job conditions are classed as 'excellent'. For very short transport distances over abrasive rocks, where dense, badly fragmented material must be dug and the quality of the operating labour is poor, the job conditions are classed as 'poor'.

If management and supervision are excellent and maintenance is good, management conditions are classed as 'excellent'; conversely, poor management and supervision combined with bad maintenance would require the management conditions to be classed as 'poor'.

Because of the relatively short life of wheel-loader tyres, their cost is deducted from the machine cost and the tyre costs are treated as an operating cost item.

The following format may be used to calculate ownership and operating costs.

- (1) FOB machine price, including optional extras, sales taxes, etc. £.....
- (2) Freight and insurance to site £.....
- (3) Import duty £.....
- (4) Sub-total £.....
- (5) Minus tyre replacement costs £.....
- (6) Sub-total (basic machine costs) £.....
- (7) Wheel-loader life

n = ... years (7a)

.... scheduled h/year (7b)

.... total hours (7c)

(see Table 7.VII: 7a is obtained from 7b and 7c)

- (8) Machine depreciation

$$\text{cost/h} = \frac{\text{sub-total 6}}{\text{Item 7c}} \quad \text{£.....}$$

- (9) By use of average investment formula

Average investment =

$$\frac{(\text{sub-total 4}) \times (n + 1)}{2n} = \text{£.....}$$

- (10) Interest rate =.....%

- (11) Insurance =.....%

- (12) Taxes, etc. (if any) =.....%

- (13) Total =.....%

- (14) (Total %) x (Average investment)
÷ (item 7b) = £.....

- (15) Total ownership costs/h
= (Item 8) + (Item 14) = £.....

Operating costs

- (16) Fuel costs (see Table 7.VIII)
 (Fuel consumption/h)x(cost/gallon) = £.....
- (17) Lubricants (see Table 7.IX)
 Crankcase (hourly consumption) x
 (cost/gallon) = £.....
 Transmission (hourly consumption) x
 (cost/gallon) = £.....
 Filter costs/h (see Table 7.X) = £.....
 Grease costs/h = 0.2 x Cost/lb = £.....
 Starter 1.2 p/h = £.....
- (18) Hydraulic oil = 4p/h
- (19) Maintenance: repair parts and labour
 M x (Item 8) = £.....
- where M is 0.9 for excellent job
 conditions, 1.0 for good conditions,
 1.2 for fair job conditions and 1.3
 for poor job conditions. Some judg-
 ment must be used in fixing a value
 for M. If high import duties are
 paid on spares, high freight charges
 are involved, etc., M must be suit-
 ably adjusted. Cost histories of
 similar machines provide the most
 reliable guide.
- (20) Tyre costs
 Tyre replacement costs = £.....
 Tyre life (see Table 7.III)
 (Mine operators are usually able to
 negotiate fleet prices for tyres;
 25-40% reduction from list prices)
- (21) Operator's wages/h, including social
 benefits, taxes, insurance etc. = £.....
- (22) Total operating costs
 = Items (16+17+18+19+20+21) = £.....

Total ownership and operating costs*

(23) Total ownership + operating costs
 = (Item 15) + (Item 22) = £.....
 Cost/tonne

(24) Item 23
 t/h = £.....

*No administrative, development or other charges are included in these figures.

TABLE 7.VIII

APPROXIMATE FUEL CONSUMPTION OF
RUBBER-TYRED EQUIPMENT DIESEL ENGINES

	Operating conditions**		
	Light	Normal	Heavy
Gal/horsepower h*	0.02	0.04	0.065

*At present most engines have horsepower ratings or PS ratings rather than kW ratings and fuel will probably be sold in gallons for some years to come.

**Light light-duty cycle without severe adverse grades and good digging conditions; normal, short transport distance, digging well fragmented medium rocks; heavy, short transport distance digging dense, poorly fragmented rocks, with no transport system delays.

TABLE 7.IX

CRANK CASE LUBRICATING OIL CONSUMPTION -
MOBILE EQUIPMENT DIESEL ENGINES (gal/h)

Engine hp	Operating conditions		
	Light	Normal	Heavy
150	0.1	0.11	0.13
250	0.17	0.18	0.2
350	0.2	0.22	0.27
500	0.3	0.33	0.39
750	0.38	0.42	0.49

TABLE 7.X

FILTER COSTS - MOBILE EQUIPMENT

DIESEL ENGINES (p/h) *

Engine hp	Dusty conditions	Normal conditions
150	2.5	1.7
250	3.0	2.2
350	6.2	4.4
500	7.5	6.0
750	8.0	6.2

*If high import duties or transport charges must be paid on filter elements, these costs must be suitably adjusted.

Wheel Loaders versus Loading Shovels

Since the introduction of the 10-yd³ wheel loader in 1967, controversy has abounded regarding the relative merits of wheel loaders and loading shovels as primary loading equipment. A number of somewhat naive economic analyses proving the cost advantages of both forms of equipment have been published, but none has properly defined the limits of operation of either machine. Briefly restated the features of both machines are as follows.

Wheel Loaders

Advantages

1. Excellent mobility - can travel quickly to any part of the pit.
2. Good versatility - can be used for a wide variety of duties, including face loading, cleaning up, stockpile operations, haul road construction and maintenance, etc. This must not be over-emphasized, however as in one mine in which a large wheel loader was used the

following applied: primary loading, not sufficiently robust; cleaning up, satisfactory with bucket teeth fitted: difficulties experienced with continuous bucket lip; stockpile duties, excessive degradation of iron pellets when bucket teeth fitted, but satisfactory with continuous bucket lip. Changing duties may not therefore simply mean travelling to a different location.

3. Can operate on moderate grades without difficulty.
4. Lower capital cost: has obvious advantages in situations of capital rationing, overseas investments where there is political and economic uncertainty, etc.
5. Not affected by obsolescence.
6. Only one operator needed per machine.
7. Large lumps are not trapped in the bucket as can happen with a shovel dipper.

Disadvantages

1. Unsuitable for hard, dense rocks which fragment badly due to open fissures, joints, bedding planes and other discontinuities.
2. Ground preparation must be extremely thorough. There are of course overall savings to be made by increased blasting with most forms of loading machines.
3. Tyres require special attention to avoid excessive costs.
4. High operating costs.

5. Greater operator fatigue and proneness to operator abuse.
6. Relatively high ground bearing pressures, poor performance with bad floor conditions.

Shovels

Advantages

1. Well proven in the field of digging hard, dense, badly-fragmented rocks.
2. Low operating costs.
3. Less sensitive to poor quality maintenance than wheel loaders.
4. Reduced operator fatigue and less prone to careless operation.
5. Lower ground bearing pressures and not so badly affected by poor floor conditions.

Disadvantages

1. Lack of mobility : for multi-bench, sequence operations in open pits with a number of shovels this need not be a major disadvantage.
2. Single purpose loading machine.
3. Higher capital cost.
4. Cannot travel on steep grades.
5. Affected by obsolescence, but many 30 year old shovels are still performing economically in severe conditions.
6. Support equipment may be needed for bench clean-up, etc.

Operational Experience

Of about twenty mine operators with experience of both wheel loaders and loading shovels, those handling hard, dense rocks, particularly metalliferous ores, considered that the wheel loaders presently available are not sufficiently robust for regular primary loading duties. Almost all agreed their wheel loaders were indispensable for general duties and emergency primary loading. All were appreciative of their mobility and versatility ⁷¹³.

Some operators engaged in mining weaker rocks felt that the wheel loader had some application, but the majority preferred the loading shovel because of its long-term reliability. Prejudice against the wheel loader possibly exists as some operators complained of excessive operating costs towards the end of its life when, probably it should have been scrapped.

Those engaged in mining weaker materials, or soft overburden, were slightly in favour of wheel loaders. Some who were producing materials for the construction industry felt that they could not realistically plan ahead beyond four years and they therefore favoured the wheel loader because of its lower capital costs.

CRAWLER-TYPE TRACTOR LOADERS

Recent developments of the wheel loader have resulted in an extension of its use into job applications previously considered suitable only for crawler-mounted machines. The main duty of the crawler-type, front-end loader in open-pit mining is as a support machine.

A comparison between the relative mobility of the loading shovel, the crawler-mounted tractor loader and the wheel loader

is shown in Table 7.XI.

TABLE 7.XI
OPERATING SPEED OF LOADING MACHINES

Machine	Maxn. travel Speed km/h (mile/h)	Operating Speed (level grade)	
		loaded	empty
Loading Shovel	1 - 3 (0.6 - 2)	-	-
Crawler-mounted Tractor Loader	10 - 12 (6 - 7.5)	3 - 5 (2 - 3)	5 - 7 (3 - 4)
Wheel Loader	25 - 35 (15 - 21)	12 - 15 (7.5 - 9.5)	15 - 20 (9.5 - 12.5)

Although the crawler-mounted, tractor shovel has not the mobility of the wheel loader, it is much more mobile than the loading shovel and is particularly useful for excavation of inclines, sumps, etc.

Production Rate

For face loading the machine follows a V-shaped path with overall cycle times between 0.7 and 1.0 min (Fig 7.5). The distance per leg travelled is approximately 5m (16.5 ft), the maximum turn being less than 90°. For excavating sumps, inclines, a fixed time of 0.65 min (load, manoeuvre, dump) is normal and an overall haul speed of 4 km/h (2.5 mile/h) can be used to calculate the variable time (Fig 7.6).

The calculation of production rate can follow the same method as proposed for wheel loaders, but it should be borne in mind that the operations involved are not usually repetitive, as for a wheel loader, and the same standards of efficiency are seldom achieved.

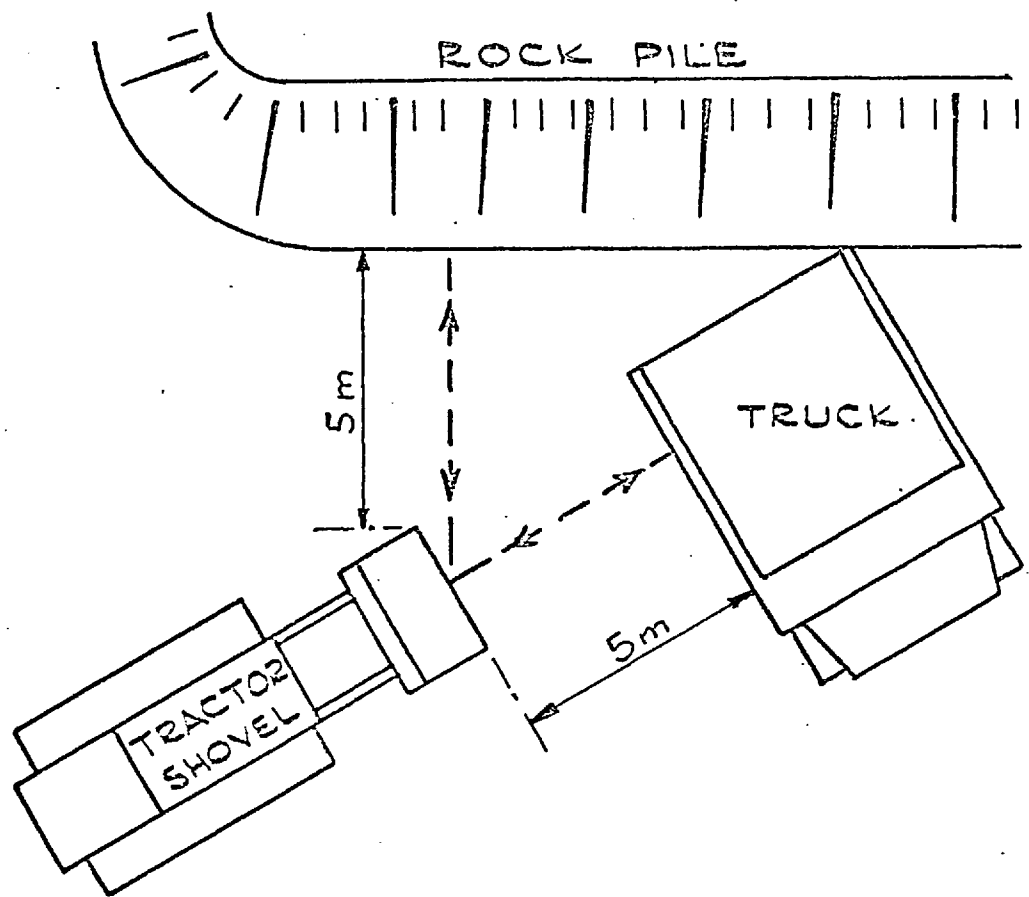


FIG 7.5 "V" SHAPED PATH FOR TRACTOR LOADER OPERATION

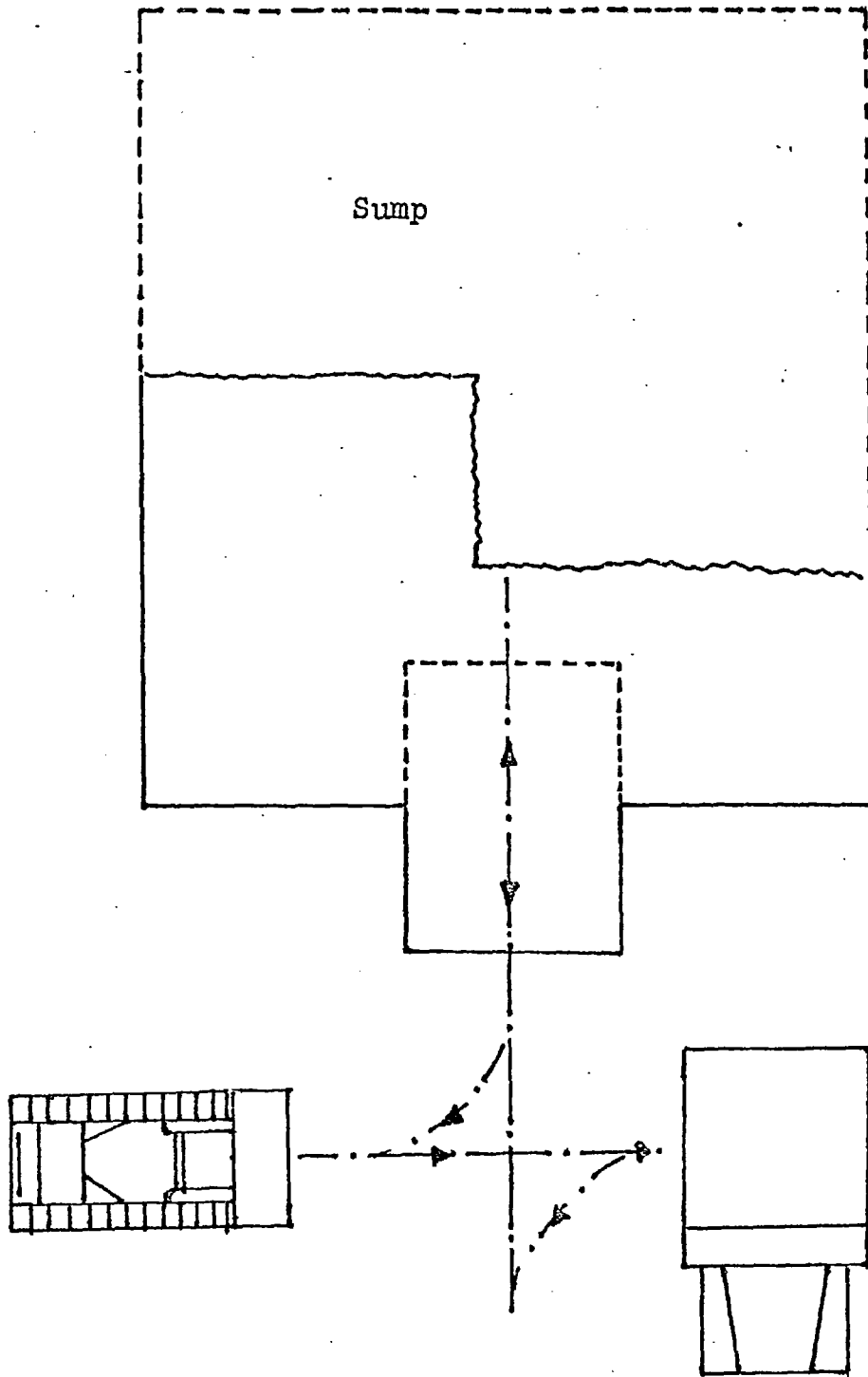


FIG 7.6 SUMP EXCAVATION USING CRAWLER-TYPE TRACTOR LOADER

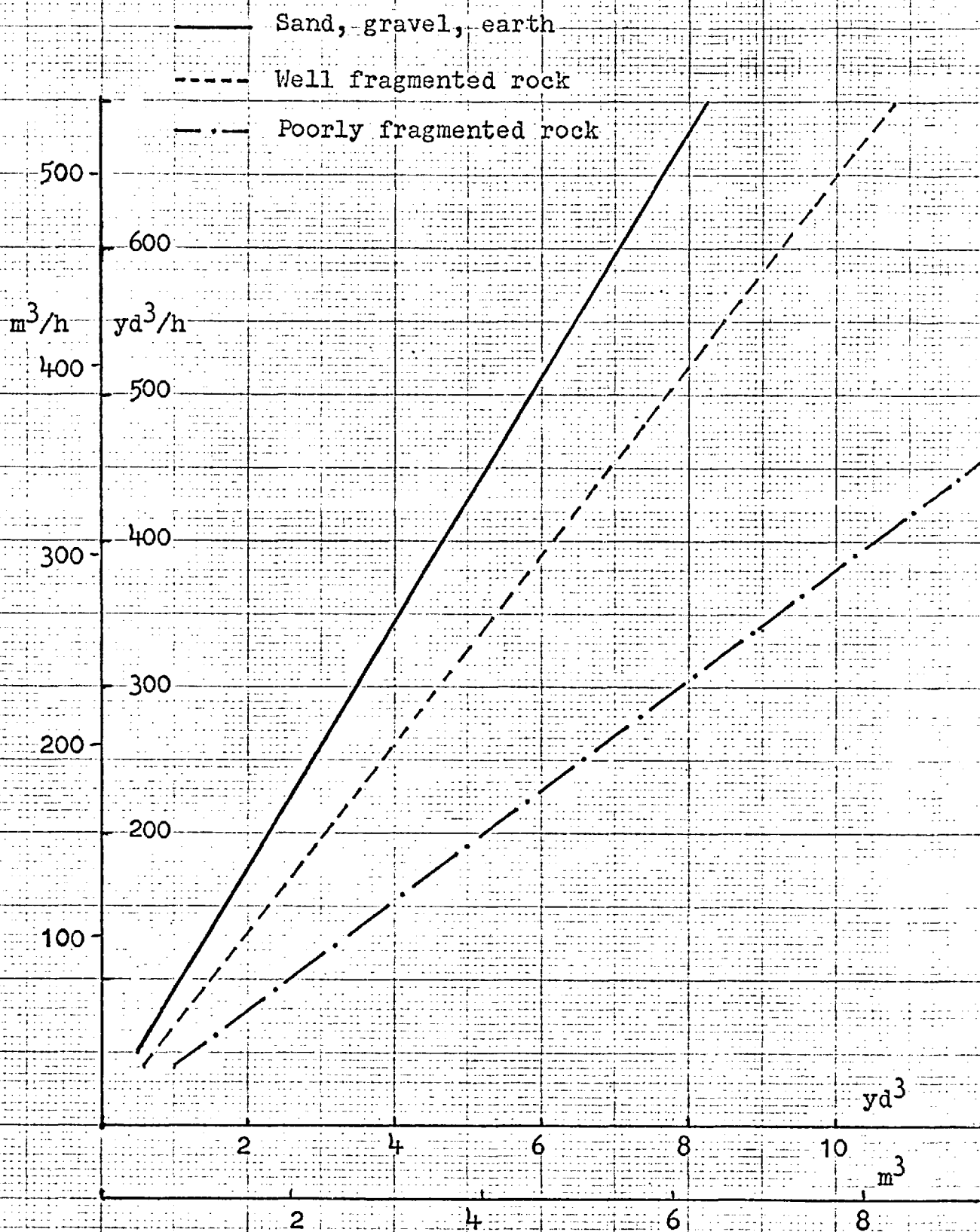


FIG 7.7 PRODUCTIVITY OF TRACTOR LOADERS
(NB Curves are a guide only.
Operational figures should be
used where available)

Fig 7.7 shows the approximate production of both wheel loaders and crawler-mounted tractor loaders for face loading with a variety of materials. The curves are for a 60 min hour i.e. no relaxation is included for availability or job operational factor. It should be noted that the figures compiled by the author for mobile machines are consistently pessimistic compared with those provided by manufacturers.

Crawler-mounted, Tractor Loader Costs

Ownership Costs

The method used for wheel loaders may be used for calculating ownership costs, except that no deduction is made for tyre costs.

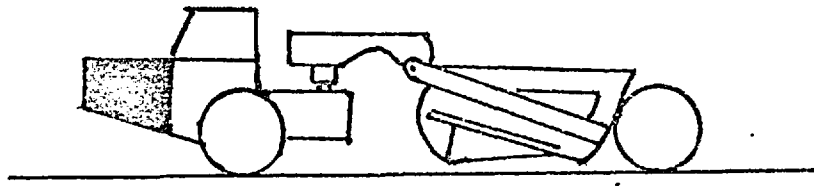
Operating Costs

The fuel and lubricant costs are generally 1.2 times greater than shown in Table 7.VIII and 7.IX. Maintenance costs are about 1.2 greater than those for wheel loaders. There are of course no tyre costs.

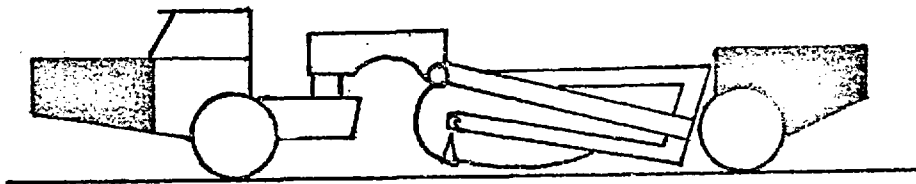
THE TRACTOR-SCRAPER

Many medium to strong rocks tend to blocky fracture after ground preparation and are unsuitable for excavation by tractor-scrapers. Their open-pit mining applications are mainly confined to overburden stripping, where their productive capacity and mobility are of considerable value, although they are also used to some extent in limestones and other medium rocks which fragment well after blasting or ripping.

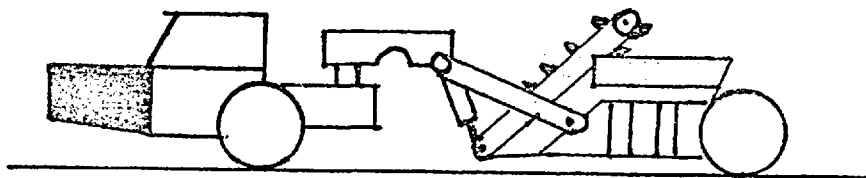
Many models are available, some of 70-m³ (90-yd³) capacity, with 92-m³ (120-yd³) designs on the drawing-board, but the most popular sizes in mining are in the



a) The Single-engined, Conventional Scraper



b) The Tandem Powered Scraper



c) The Elevating Scraper

FIG 7.8 TRACTOR SCRAPER TYPES

23-38 m³ (30-50 yd³) struck capacity range. Although here they are being considered as excavators, they are 'load-haul-dump' machines, and their complete cycle is amenable to investigation by simulation techniques.

There are three main types of tractor-scraper:

1. The single-engine, conventional scraper
2. The tandem power scraper
3. The elevating scraper.

The single-engine, conventional scraper (Fig 7.8a) has the widest range of economic application. Except in very easy, downhill conditions, a "pusher"-tractor is needed for loading. The tractor then 'boosts' the scraper to near haul speed. The conventional scraper does not perform well where the duty cycle includes steep adverse grades, high rolling resistances and poor floor conditions. Where the haul distances are short queues ("bunching") can occur while waiting for the pusher-tractor to start the loading cycle.

The tandem power machine (Fig 7.8b) has an all-wheel drive from front and rear engines which provides a high power/weight ratio, tractive effort and retarding ability, suitable for conditions with high rolling resistances and adverse grades.

It can therefore work where other scrapers cannot, especially in mud or when dumping in wet conditions on uncompacted spoil heaps. Normally, the tandem power scraper operates with the assistance of a push-tractor, but it can partially self-load or even fully load on downhill runs.

Elevating scrapers (Fig 7.8c) are self-loading and are ideally suited for short to medium haul distances where a queue would otherwise form while waiting for the pusher-tractor. They do not perform well on adverse grades or with high rolling resistances due to the single engine drive.

The elevating scraper cannot efficiently handle sticky materials or material containing rock or boulders larger than 200 mm (8 in). Because of its ability to work alone the elevating scraper is ideally suited for selective mining or for small-fleet operations. The action of the elevating flights also breaks down and helps to blend the material being loaded. It is easier to spot over dumping hoppers than other scrapers.

Section of tractor-scraper type

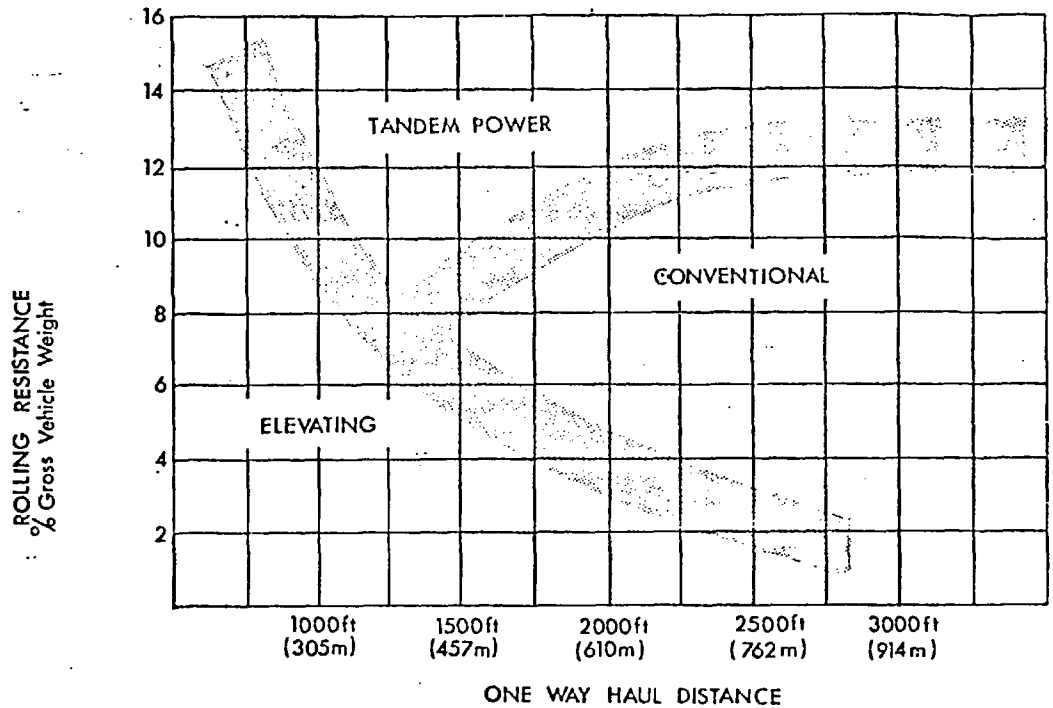
Fig 7.9 indicates that the application zones of the three tractor-scraper types are not sharply defined and that some overlap exists as indicated by the shaded areas.

This figure is based on the results of a computer investigation using Caterpillar's vehicle simulation programme. The three types of tractor-scraper were simulated over three types of courses consisting of hauls with rolling resistances from 2% to 10% on level, 8% adverse and 8% favourable grades 714.

Scraper Cycle Times

The fixed time is made up of decelerating, spotting the scraper by the pusher vehicle, loading, accelerating ('boosting'), dumping and spreading plus the variable time, travelling between cut and dump. The fixed time is best determined by time studies, but, in the absence of experience in similar conditions, the figures given in Table 7.XII may be used.

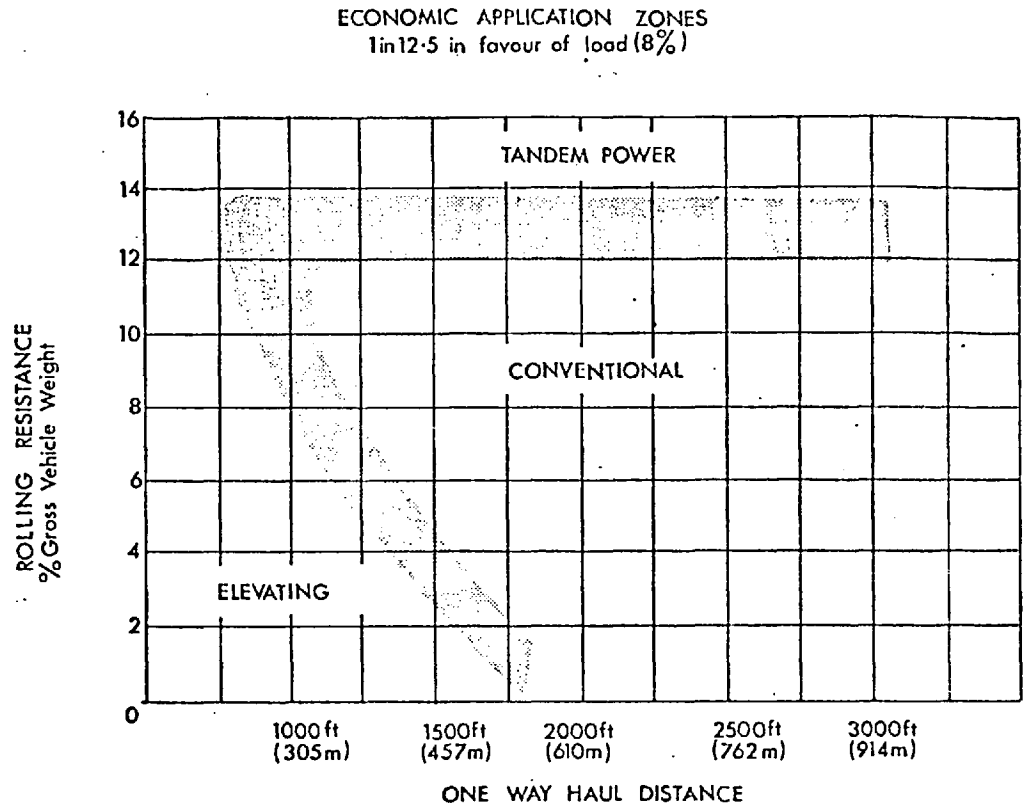
ECONOMIC APPLICATION ZONES
FOR LEVEL HAULS



For level haul roads the advantage of the elevating scraper is apparent on shorter hauls, but as rolling resistance increases, the maximum economic haul distance of the elevating machine decreases. This results from the elevating scraper's low power/weight ratio.

On level hauls, tandem powered scrapers are most economical on high rolling resistances. This type has greater power and consequently costs more to operate than the conventional scraper. The tandem powered machine is the most economic only when its additional power can yield enough extra production to offset the increased cost. On level hauls, conventional scrapers have the widest application range.

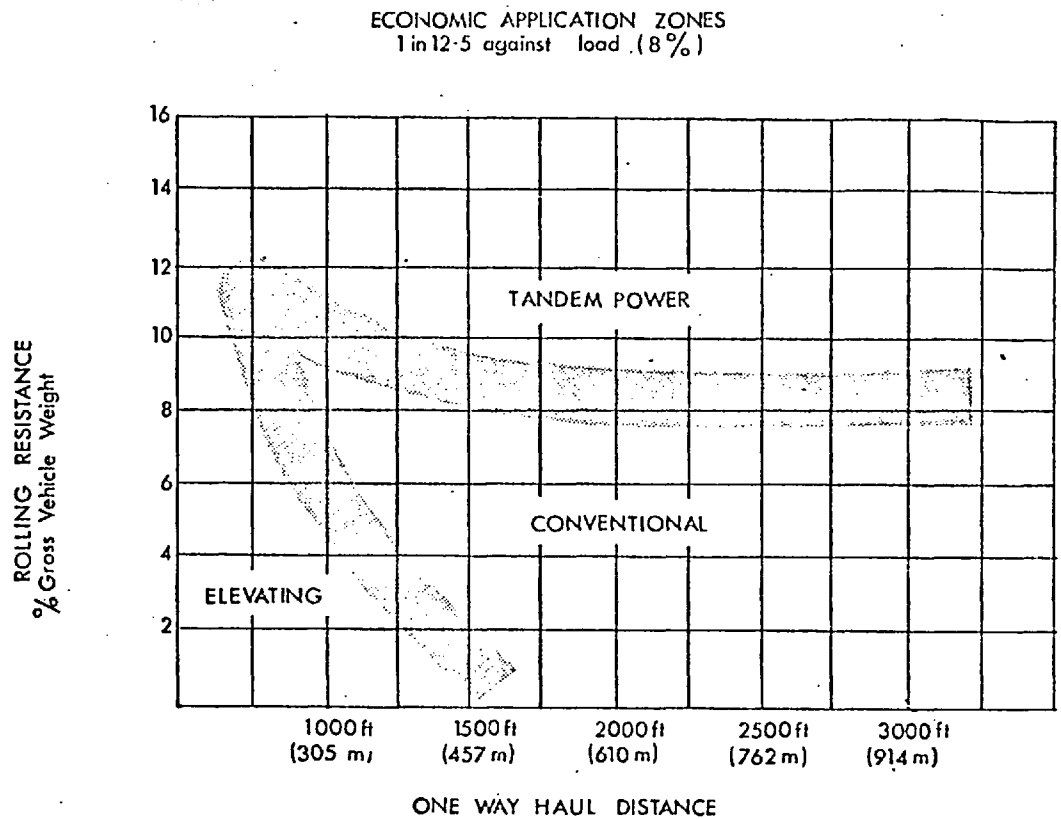
FIG 7.9a THE TRACTOR SCRAPER - LEVEL HAULS



On 8% favourable grades the significant comparison is between conventional and elevating scrapers. Favourable grades do not allow the tandem powered scraper to use its greater power, retarding ability and tractive effort any great advantage. Only when very high rolling resistances and steeper than 8% grades are encountered does the tandem powered machine achieve an economic advantage.

The elevating scraper has again the economic advantage on short haul distances, but as rolling resistance increases, the economic haul distance falls considerably. On an 8% favourable grade and 10% rolling resistance, the elevating scraper is most economical up to about 150 m (500 ft). Again the conventional scraper has the widest economic application zone.

FIG 7.9b THE TRACTOR SCRAPER - 8% FAVOURABLE GRADE



On 8% adverse grades, the tandem powered scraper's superior power, tractive effort and retarding capability provide it with much wider application. When the rolling resistance approaches 8 - 9% the tandem powered machine becomes most economic.

Again the elevating scraper is limited to relatively short haul distances. The conventional unit is most favourable only when rolling resistances in all parts of the haul are low.

FIG 7.9c THE TRACTOR SCRAPER - 8% ADVERSE GRADE

TABLE 7.XII

STANDARD FIXED - TIME FOR TRACTOR-SCRAPERS

Type	min	
Conventional	1.45	(Push-loaded)
Tandem power	1.25	(Push-loaded)
Elevating	1.65*	(Self-loaded)

*No spotting time included.

Table 7.XII is based on a large number of observations. It is interesting to note the comparison between the author's figures and those of the Caterpillar Tractor Co.

<u>Scraper Type</u>	<u>Conventional</u>	<u>Tandem</u>	<u>Elevating</u>
Fixed time (min) A	1.45	1.25	1.65
C1	1.35	1.10	1.50
C2	1.35	1.25	1.60
Type of loading	Pusher	Pusher	Self

A - Author's figures.

C1- Ref 714.

C2- Caterpillar 1973, Performance Handbook, average figures.

This appears to confirm that manufacturers original figures tend to be optimistic.

The variable time is made up of loaded and empty journey times.

Variable time =

$$\frac{\text{Haul distance}}{\text{Haul speed (loaded)}} + \frac{\text{Haul distance}}{\text{Haul speed (empty)}} \dots\dots(7.12)$$

The haul speeds loaded and empty are preferably determined from time studies of similar operations, or from manufacturers performance charts, the gradient and rolling resistance of the haul road being taken into account (see Appendix 7.A)

The scraper cycle time is then:

$$t_c = \text{Fixed time} + \text{Variable time} \dots\dots\dots (7.13)$$

Scraper Production

The load hauled is best expressed in bank volume: it is based on scraper heaped capacity, fillability and swell factor.

$$\begin{aligned} &\text{Scraper production (bank vol/h)} \\ &= \frac{\text{Scraper heaped capacity} \times \text{Fillability} \times 60 \times AO}{\text{Swell factor} \times t_c} \end{aligned} \dots\dots\dots (7.14)$$

O - For both conventional and tandem powered scrapers, the maximum attainable production must be reduced to account for queue times resulting from changing haul lengths, etc. On short to medium hauls, where bunching is greatest, the author's time studies indicate factors in the 0.88 - 0.92 range for bunching only. This compares with 0.90 suggested by Caterpillar ⁷⁶.

AO - Operational efficiency. Where no previous experience is available to determine A and O, their product AO can be obtained from Table 2.VI. If the job conditions are bad owing to excessive rain, ice and snow, dust, bad haul roads, severe adverse grades, dense, abrasive poorly fragmented rock, poor-quality operating labour, etc., then it should be given a 'poor' rating. If management and supervision are poor, with bad maintenance, inefficient workshops, poor availability, bad job layout and organization, then it should also be given a 'poor' rating. Conversely, with first-class management and very good job conditions 'excellent ratings' are used.

Fillability

The fillability of scraper bowls is generally higher than that of other excavators because of the limited range of materials handled and the loading method. Table 7.XIII provides an approximate guide.

Swell factor

Because of its mode of loading, the swell factors of scraper-handled material can be different from those for other machines. Table 7.XIII gives approximate figures.

TABLE 7.XIII

FILLABILITY AND SWELL FACTOR - TRACTOR SCRAPERS

Digging conditions	Fillability	Swell factor
Easy, dry	0.95-1.00	1.15-1.25
Medium, common earth	0.90-0.95	1.2 -1.3
Medium-hard, clays	0.85-0.9	1.35
weak rock	0.85-0.95	1.35-1.45
Hard, wet clay	0.70-0.80	1.4
well broken rock	0.75-0.80	1.5
Extreme, broken basalt, etc.	0.50	1.5-1.6

In determining production rate it is important to precisely define the tractor-scraper capacity. Some manufacturers refer only to the struck capacity as the pay load (the product of the width between the scraper side plates and the area of one side plate; if the top of the apron in the closed position is lower than the side plates, this capacity is reduced by a 45° plane extending inwards from the top of the apron). Others use the heaped capacity. A few define both.

In the past with narrow bowl designs operating in soil/clay materials, the struck capacity approximately equalled

the payload, since with these designs:

$$\frac{\text{heaped capacity}}{\text{struck capacity}} = 1.335 \text{ approx.,}$$

the fillability was taken as 0.75

therefore:

$$\begin{aligned} \text{pay-load} &= \text{struck capacity} \times 0.75 \times 1.335 \\ &= \text{struck capacity} \times 1.0 \end{aligned}$$

Design trends in recent years for some tractor-scraper have tended towards lower, wider bowls, which carry a greater proportion of their load in the heap due to the larger base area and lower side plates. It is therefore important to define struck and heaped volumes, i.e. a tractor-scraper with a 24 yd³ struck volume and a 34 yd³ heaped volume should be described as a "24/34" machine.

The following indicate the variations in a number of manufacturers designs of similar size:

<u>Struck Capacity</u> <u>yd³</u>	<u>Heaped Capacity</u> <u>yd³</u>	<u>Heaped Ratio</u> <u>Struck</u>
21	30	1.43
22.5	30	1.335
24	32	1.335
24	34	1.415
25	32	1.28

The heaped/struck ratio is additionally a useful indication of the loadability, since a tractor-scraper carrying most of its capacity as struck volume necessarily has a high, narrow bowl that requires more time and power to load. The loadability improves as the tractor-scraper bowl is widened and lowered

Pusher tractors

To avoid queuing ('bunching') of scrapers at the start of each cut the number of scrapers must be correctly matched to the number of pushers. Because of variations in cycle time, a perfect match is never possible in an actual mining operation and some waiting time is usually unavoidable.

The maximum number of scrapers served by each pusher tractor should not exceed

$$\text{Scrapers (max n served)} = \frac{\text{Scraper cycle time}}{\text{Pusher cycle time}} \dots\dots\dots (7.15)$$

where

$$\begin{aligned} &\text{Pusher cycle time (min)} \\ &= 1.4 (\text{Scraper load time}) + 0.25 \dots\dots\dots (7.16) \end{aligned}$$

The pusher cycle time is usually in the range of
1.2 - 2.5 min.

In determining the number of pushers required it is necessary to take the fleet size into account especially with conventional scrapers, since the added costs of an extra pusher shared between a number of units may be justified by obtaining maximum production. Where haul distances are short, 'bunching' also can become a serious problem, and an adequate number of pushers is essential. The problem with tandem-power scrapers is not so acute as they can partially self-load.

Push-pull operations

Pairs of tandem-power scrapers are used in the push-pull method. The two machines link up through push blocks and a hook and bail assembly and enter the cut: the second scraper push-loads the first, the first then pull-loads the second; they disconnect and drive individually to the dump.

Cycle times are slightly greater than for single push-loading, but of course the pusher-tractor costs are not incurred.

Tractor-scraper costs

The economic life of a tractor-scraper depends on the job and management conditions. Where no previous records are available, Table 7.XIV, which is based on collected data may be used to provide a guide.

TABLE 7.XIV
ECONOMIC LIFE - TRACTOR SCRAPERS
(Scheduled hours)

Job conditions	Management conditions			
	Excellent	Good	Fair	Poor
Excellent	13000	12500	11500	10500
Good	12500	11500	10500	9500
Fair	11000	10500	9500	8500
Poor	10000	9000	8500	8000

Easy digging, long level hauls over good roads of non-abrasive materials with highly skilled operators would rate as 'excellent' job conditions, whereas hard-digging, short hauls over steep adverse grades, with bad, rutted, muddy or abrasive road surfaces would rate as 'poor' job conditions. Similarly, with good supervision and maintenance the management conditions are 'excellent', whereas bad supervision, together with inadequate maintenance, would rate as 'poor'.

Tyre costs are normally deducted from the machine cost and treated as an operating cost item.

The following format can be used as a basis for calculating scraper ownership and operating costs.

Ownership costs

- (1) FOB machine price, including optional extras, sales taxes etc. £.....
- (2) Freight and insurance to site £.....
- (3) Import duty £.....
- (4) Sub-total £.....
- (5) Minus tyre replacement costs £.....
- (6) Sub-total (basic machine cost) £.....
- (7) Tractor-scraper life
 $n = \dots\dots \text{years (7a)}$
 $\dots\dots \text{scheduled h/year (7b)}$
 $\dots\dots \text{total hours (7c)}$
 (see Table 7.XIV: 7a is obtained from 7b and 7c)
- (8) Machine depreciation

$$\text{cost/h} = \frac{(\text{Sub-total 6})}{\text{Item 7c}}$$
- (9) By use of average investment formula
 Average investment

$$= \frac{(\text{Sub-total 4}) \times (n + 1)}{2n} = \text{£} \dots\dots$$
- (10) Interest rate =.....%
- (11) Insurance =.....%
- (12) Taxes, etc (if any) =.....%
- (13) Total =.....%
- (14) $(\text{Total } \%) \times (\text{Average investment}) \div (\text{Item 7b}) = \text{£} \dots\dots$
- (15) Total ownership costs/h
 $= (\text{Item 8}) + (\text{Item 14}) = \text{£} \dots\dots$

Operating costs

- (16) Fuel costs (see Table 7.VIII)
 $(\text{Fuel consumption/h}) \times (\text{Cost/gal}) = \text{£} \dots\dots$

- (17) Lubricants (see Table 7.IX)
- Crankcase (hourly consumption) x
(Cost/gal) = £.....
- Transmission (hourly consumption)x
(Cost/gal) = £.....
- Filter costs (see Table 7.X) = £.....
- Grease costs = 0.2 x Cost/lb = £.....
- Starter (1.2 p/h) = £.....
- (18) Hydraulic oil (4.0 p/h) = £.....
- (19) Maintenance: repair parts and labour
- M x (Item 8) (see Table 7.XV for
values of M) = £.....
- (20) Cutting edge cost
- $\frac{\text{Cutting edge replacement cost}}{\text{Cutting edge life}}$ = £.....
- Cutting edge life will vary
between 350 and 700 h (costs
from 20-50 p/h, the highest
figures for large machines
working in severe ground)
- (21) Tyre costs
- $\frac{\text{Tyre replacement costs}}{\text{Tyre life (see Table 7.III)}}$ = £.....
- (Fleet prices usually apply)
- (22) Operator's wages/h, including social
benefits, taxes, insurance etc. = £.....
- (23) Total operating costs
- Items (16 + 17 + 18 + 19 + 20 +
21 + 22) = £.....
- Total ownership and operating costs
- (24) Total ownership and operating costs
- = (Item 15) + (Item 23) = £.....
- (For push loading operations with
conventional and tandem-power
machines the partial cost of the
pusher vehicle must be added to
the tractor-scraper costs. The
pusher vehicle costs can be
estimated by the method used for
bulldozers, except that the blade
costs are eliminated)

Cost/t

$$(25) \quad \frac{(\text{Item 23}) + (\text{Pusher cost, if applicable})}{t/h} = \text{£} \dots$$

(No overheads for administration, development, etc., are included)

TABLE 7.XV

MAINTENANCE FACTOR, M - TRACTOR-SCRAPERS*

Scraper type	Job conditions			
	Excellent	Good	Fair	Poor
Conventional	0.65	0.80	1.00	1.2
Tandem power	0.66	0.82	1.02	1.23
Elevating	0.68	0.84	1.05	1.28

*Some judgment must be used in fixing a value for M to allow for excessively high import duties, high freight costs, etc. Cost histories provide the best guide. Table 7.XV based on collected data, see Chapter 2.

Tractor-Scraper Operations in Rock

Push-scraper loading of rock initially appalls many mechanical engineers. Their three major objections are:

1. Scraping an adequate load.
2. Excessive mechanical maintenance costs.
3. Excessive tyre costs.

In loading, lumps should be less than 650 mm (2 ft), as larger lumps tend to slow down production, but of course shovel dippers have similar problems.

Ground preparation is all important, a higher powder ratio for explosives may be necessary or close spacing of ripping passes may be needed to increase fragmentation. Cross ripping can be helpful but many engineers believe that only ripping in the direction of scraping should be used in harder materials.

The fragmentation of the rock is an important feature.

If well graded, there are sufficient fines to act as a "lubricant" in promoting material flow. If fragmentation is blocky, with little fines, "bridging" can occur, making loading difficult and maintenance costs high.

From the mechanical maintenance standpoint, tractor scrapers are prone to operator abuse. This is especially so for scrapers in rock, where close field supervision is essential. The wearing parts are usually increased in weight by 50 - 60%. There is usually a fall off in availability as conditions worsen, down-time becoming greater than for a shovel-truck combination.

With regard to tyre costs, close field supervision is needed, and avoidance of wheel spin, colliding with large lumps, etc., is imperative. The tyre-life curves provided by manufacturers are similar to the author's figures (Table 7.III), except that the range of Table 7.III is wider.

To sum up, tractor scrapers have a shorter life loading rock, but usually have lower capital cost than the equivalent shovel-truck combination. The shovel-truck combination may be 50% higher on a £/ unit volume of hourly production basis, so that tractor-scraper operations in rock can be competitive.

Some of the additional advantages of tractor-scrapers are:

- flexibility in operation
- dumping capability (a bulldozer may not be needed) and,
- tractor-scrapers can operate on moderately steep grades.

THE BULLDOZER

Although not largely employed as a primary excavator the bulldozer has considerable application in open-pit mining. The crawler-mounted machine is generally more accepted and 67 t units are at present in operation, but because of its mobility, the rubber-tyred unit is increasingly finding application and designs up to 85 t are under development.

These units, with "super-wide" blades, are particularly expected to find application in land reclamation in strip mining operations. The following are some of the bulldozer applications in open pit mining:

1. Top soil stripping and land clearing.
2. Short haul overburden stripping in strip mining.
3. Excavation of ripped rock, especially in the quarrying industry.
4. Construction and maintenance of haul roads.
5. Clean up duties for other excavators.
6. Profiling, etc., for land reclamation.
7. Ripping.
8. Truck "boosting".
9. Bund construction.
10. Move equipment e.g. cable skids, pumps, pipes, etc.

The general conditions suited to crawler-mounted and rubber-tyred bulldozers are:

Conditions favourable to RUBBER-TYRED <u>Bulldozer</u>	Conditions favourable to CRAWLER-MOUNTED <u>Bulldozer</u>
1. Moderate slopes	1. Steep slopes
2. Long dozing distance	2. Short dozing distance
3. Mobility required.	3. Mobility not important.

The other items described earlier in the chapter also apply. For short hauls where tractor-scraper operations would be difficult dozing can usually be adopted.

Blade Selection

To obtain efficient production the blade type must be matched to the duty.

Materials

Most soils and fragmented rocks can be successfully bulldozed, but performance is affected by:

1. Fragment or Particle Size and Shape. The larger the size the more difficult it is for the bulldozer blade cutting edge to penetrate. Angular fragments tend to resist the natural rolling action of the bulldozing and require more power to move than more regular, rounded fragments.
2. Voids. A material with low voids ratio is more compacted, so that the individual fragments have more of their surface areas in contact with each other. This additional bonding must be broken up when bulldozing. A well graded material with few voids requires more force to doze in the bank state.
3. Water Content. In most materials other than clay, lack of moisture increases the bond between fragments or particles and makes the material more difficult to remove in the bank state. A high water content makes material more difficult to doze because the material is heavy and requires more force to move. The presence of some moisture is beneficial to dozing, as well as reducing dust and increasing operator comfort.

The effect of freezing depends on moisture content, the bonding strengthening as the material's moisture content increases and the temperature decreases.

Freezing does not affect dry materials.

Specific Digging Power is an indication of a bulldozer blade's ability to penetrate a material and obtain an economic blade load. It is suggested that it be expressed as:

Power/linear unit of blade length

e.g. $\frac{\text{Engine Power}}{\text{blade length}}$

Because most bulldozers are based on U.S.A. designs, specific digging force is usually expressed as hp/ft.

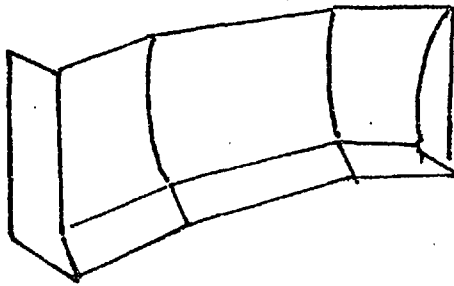
The higher the specific digging power the greater the ability of the bulldozer blade to penetrate a material, subject of course to the constraints of traction.

Specific Dozing Power is a measure of a bulldozer's ability to push its blade load. It is suggested that it be expressed as:

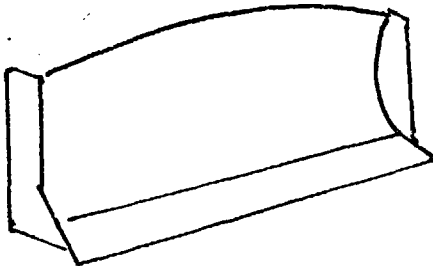
$\frac{\text{Engine Power (hp)}}{\text{Blade Load (loose yd}^3\text{)}}$

The greater the specific dozing power the greater the ability of the bulldozer to doze a material at higher speed.

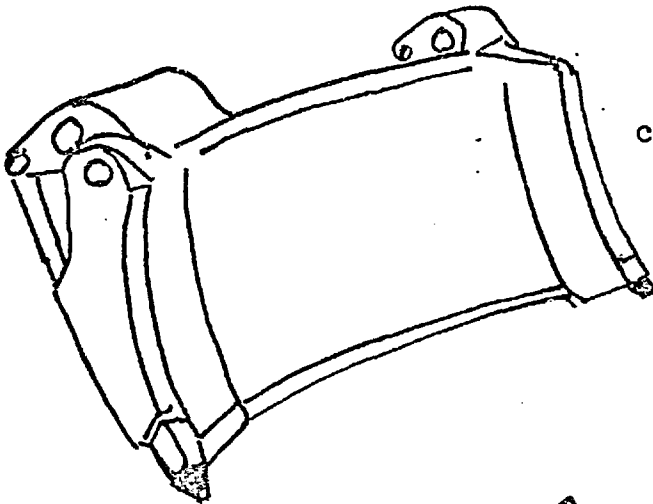
The "U" Blade (Universal Blade) ⁷⁶ - the large wings make this blade (Fig 7.10) efficient for moving big loads over long distances, e.g. land reclamation, stockpiling, etc. It has a lower specific digging power than a straight (S) blade and is not ideally suited to applications where strong penetration is essential. It also has a low specific dozing power and is therefore best suited to lighter or relatively easily dozed materials. If the blade is fitted with tilt cylinders it becomes more versatile, as its ability to ditch,



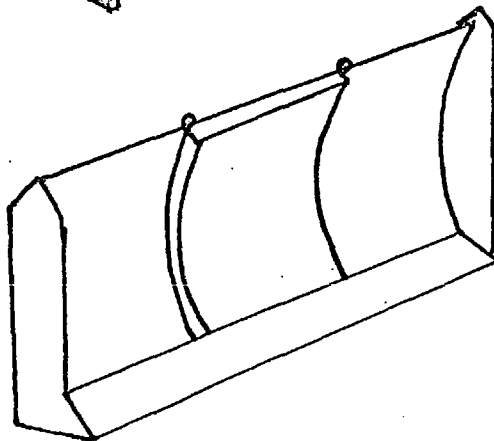
a) Universal 'U' Blade



b) Straight 'S' Blade



c) Ripdozer 'R' Blade



d) Cushion 'C' Blade

FIG 7.10 BULLDOZER BLADE TYPES

level and pry out obstructions is greatly increased.

The "S" Blade (Straight Blade) is the most versatile bulldozer blade. It is physically smaller than the U blade, easier to manoeuvre and can handle a wider range of materials. It has higher specific digging and dozing powers than the equivalent U blade and is therefore more suited to difficult, denser materials. Its versatility can also be improved by the use of a blade tilting device. It can be used for push-loading tractor-scrapers when fitted with a push plate.

The "R" Blade (Ripdozer) is basically a straight blade with a heavy duty adjustable ripper shank at each end of the moldboard (Fig 7.10). The R blade has the same capabilities as the S blade, plus the ability to penetrate very difficult material. Each shank extends up to 150 mm (12") below the blade cutting edge for additional penetration. The additional mass plus the very high digging power of the ripper shank allows it to obtain a large load in a short distance. It is normally fitted with a tilting device and is able to perform a variety of jobs, e.g. side hill digging, maintaining the toe of rock slopes, removing material beyond the reach of conventional rippers, etc. It can also be fitted with a push plate to push-load tractor-scrapers.

The "C" Blade (Cushion Blade) is primarily used for push-loading tractor-scrapers "on-the-go". The impact of contacting the tractor-scraper push block is absorbed by rubber cushions. When not used for push loading it can be used for other dozing jobs associated with tractor-scraper operations. The narrow width of the C blade is designed to increase machine manoeuvrability in congested cuts and reduce the possibility of cutting tyres associated with S and U blades.

The "A"Blade (Angling Blade) can be positioned straight or angled 25° to either side. It is used for side casting, pioneering roads, backfilling, cutting ditches, etc. and can reduce the amount of manoeuvring required for these jobs due to the side dozing action. It has a cushion frame that makes it suitable for fitting push loading attachments.

The "Bowldozer" has high side walls to move large volumes of light, preferably granulated materials over longer distances (Balderson manufacture).

The Light Material U Blade (Balderson) is designed for stockpiling, coal and reclamation work. It provides high volume movement of light, non-cohesive materials.

Note A and C blades are not considered to be primary production bulldozing tools. In severe conditions they have only 50% of the performance of S blades.

Production Rate

The blade capacity (loose volume) can usually be obtained from manufacturers' literature. If this is not available, Table 7.XVI can be used.

TABLE 7.XVI

BULLDOZER BLADE CAPACITY

m³ loose measure (loose yd³)

Tractor		Blade Type			
tonnes	sh.tons	A		S	U
11	12	2.3	(3.0)	3.2	(4.2)
15	16.5	3.0	(3.9)	4.3	(5.6)
22	24	4.2	(5.5)	6.0	(7.8) 7.7 (10)
29	32	5.7	(7.4)	8.0	(10.5) 10.5 (13.5)

The cycle time of a bulldozing operation has an insignificant fixed time element (about 3 sec for power shift). The gears to be adopted for various materials, gradients, etc., and their speeds are normally taken from manufacturers literature but the following figures may be used:

First gear dozing speed - 1.1 m/sec (3.6 ft/sec)

Third gear reverse speed - 2.8 m/sec (9.3 ft/sec)

The cycle time =

$$t_c = \frac{\text{Haul distance}}{\text{Dozing speed}} + \frac{\text{Haul distance}}{\text{Reverse speed}} \quad (\text{min})$$

The production is then

$$\text{Loose volume/h} = \frac{\text{Blade capacity} \times 60 \times A_0}{t_c} \quad \dots\dots(7.17)$$

A₀ may be obtained from Table 2.VI

A further method that avoids the use of individual judgement needed in interpreting Table 2.VI is:

$$\text{Bank volume/h} = \frac{\text{Blade capacity} \times \text{Min/h}}{\text{Swell factor} \times t_c} \quad \dots\dots(7.17a)$$

where

Min/h is the minutes per hour worked as an average throughout the shift. For bulldozing, the operation is not usually part of a system involving other items of equipment e.g. a shovel-trucking operation, and the minutes worked per hour can be expressed with reasonable precision. The following are acceptable standards:

	<u>Tractor Type</u>	<u>Min/h</u>
Day operation	Crawler-mounted	50
	Rubber-tyred	45
Night operation	Crawler-mounted	45
	Rubber-tyred	40

The swell factor for bulldozed materials is not the same as for other forms of excavation and the figures in Table 7.XVII may be used:

TABLE 7.XVII

SWELL FACTOR - BULLDOZED MATERIALS

<u>Material</u>	<u>Swell factor</u>
Broken rock	1.65
Heavy clay (wet)	1.45
Earth with boulders	1.35
Earth	1.25
Sand and small gravel	1.1

A further quick method ⁷⁶ is available:

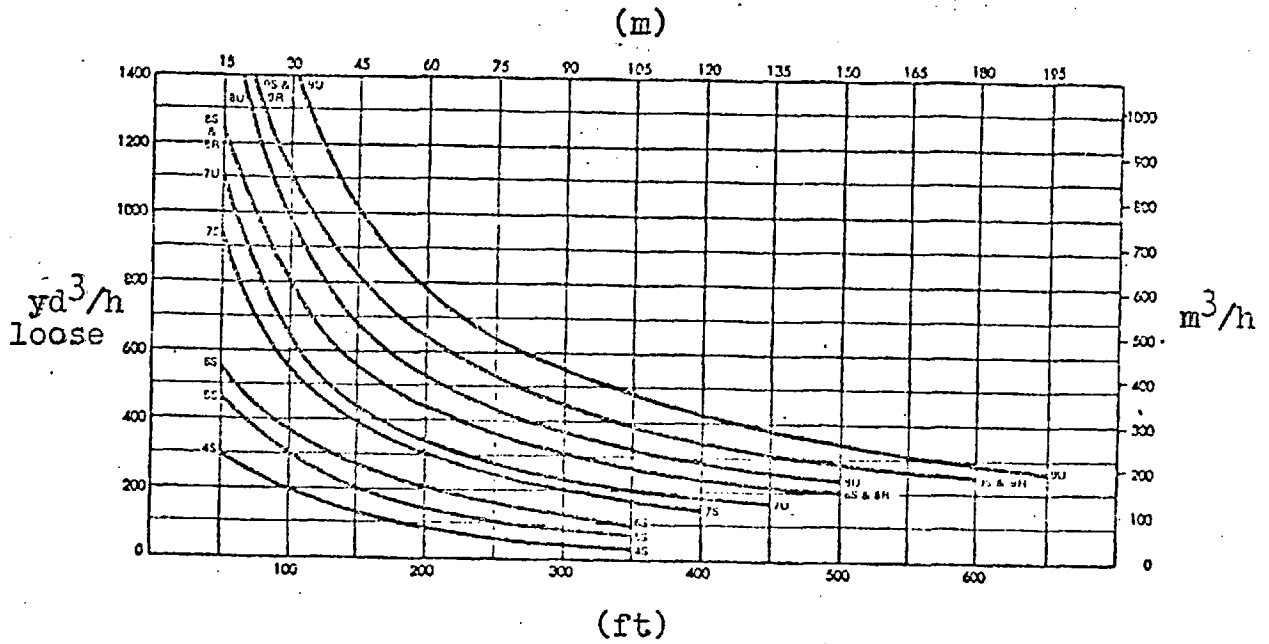
Production Rate =

Loose volume/h = maximum production x correction factors

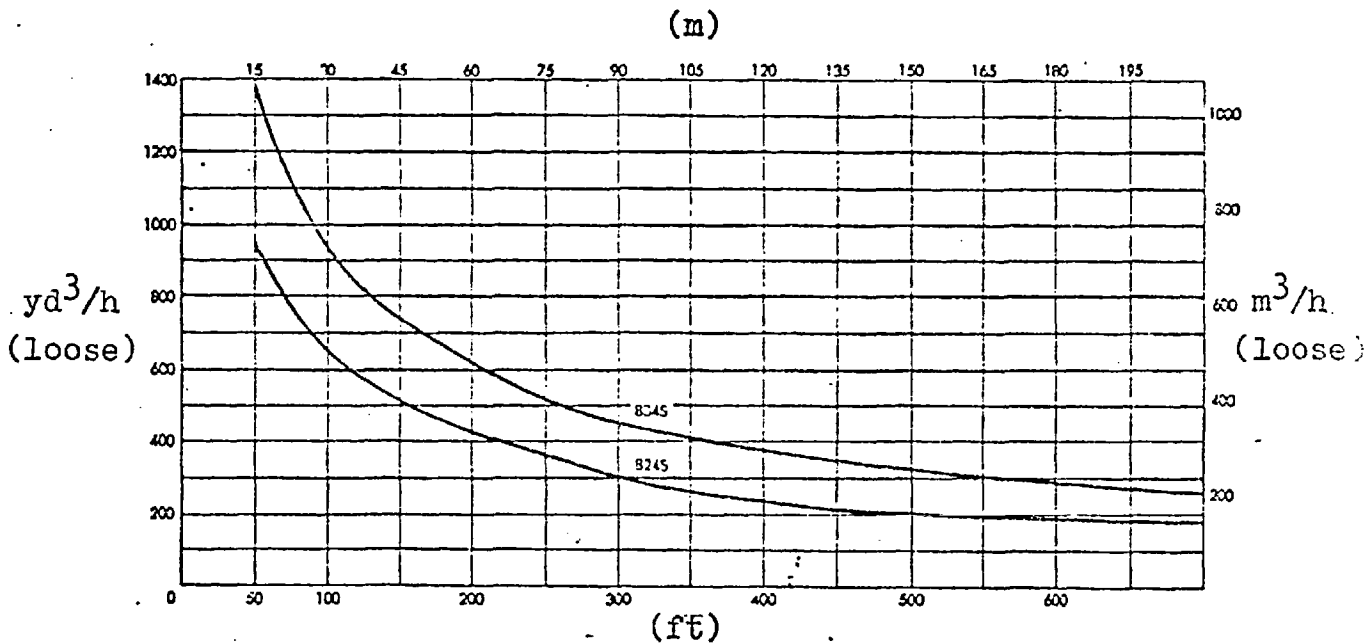
The maximum production is obtained from a family of manufacturer's curves. Fig 7.11 shows a typical example for S, U and R blades against average dozing distance. It will be readily appreciated that the bulldozer is most economic for short haul applications.

Correction Factors

<u>Job Condition Corrections</u>	<u>Crawler-Mounted</u>	<u>Rubber-Tyred</u>
Operator - Excellent	1.0	1.0
Average	0.75	0.6
Poor	0.6	0.5
Material Type - Loose stockpile	1.2	1.2
Hard to cut; frozen - with tilt	0.8	0.75
without tilt	0.7	-
cable control	0.6	-



Average Dozing Distance
a) Crawler-mounted Bulldozer 76.
U, S, and R Blades



Average Dozing Distance
b) Rubber Tyred Bulldozer.

FIG 7.11 ESTIMATED BULLDOZER PRODUCTION 76.
(Caterpillar Tractor Co.)

<u>Job Condition Corrections</u> cont'd	<u>Crawler-Mounted</u>	<u>Rubber-Tyred</u>
Hard to drift; "dead" (dry, non-cohesive material or very sticky material)	0.8	0.8
Rock, ripped or blasted	0.6-0.8	-
Slot Dozing	1.2	1.2
Side-by-side Dozing	1.15-1.25	1.15-1.25
Visibility - Dust, rain, snow, fog or darkness	0.8	0.7
Job Efficiency - 50 min/h	0.84	0.84
45 min/h	0.75	0.75
40 min/h	0.67	0.67
Direct Drive Transmission (0.1 min fixed time)	0.80	-
Angling Blade A	0.5-0.75	-
Cushioned Blade C	0.5-0.75	0.5-0.75
Rip Blade R	1.0-1.2	-
Light material U	1.2	1.2
Bowldozer Blade (stockpiles)	1.3	1.3
Gradients	see Fig 7.11	

Bulldozer Costs

The ownership and operating costs may be obtained by using the format for tractor-scrapers, with the following modifications. The economic life should be taken from Table 7.VII; tyre costs do not of course apply to crawler-mounted machines. For Maintenance: repair, parts and labour costs, see Table 7.XVIII.

TABLE 7.XVIII

MAINTENANCE FACTOR, M - BULLDOZERS*

Type	Job conditions			
	Excellent	Good	Fair	Poor
Crawler	0.8	0.95	1.1	1.3
Rubber-tyred	0.7	0.85	1.0	1.2

*Includes blade maintenance

For push-tractor operations where relatively little dozing is carried out multiply M by 0.8 to allow for reduced blade wear.

RIPPING*

The major advantages of ripping in some open pit applications have been the considerable economies in ground preparation costs compared with drilling and blasting. Because of the development of more powerful and efficient machines and improvements in techniques, ripping has moved into areas where previously only explosives were considered.

In addition to cost, ripping also has the following advantages:

Safety. Whilst good procedures eliminate most of the dangers of blasting, it is inherently more hazardous than ripping. This also results in some less obvious cost advantages for ripping since men and machines must be moved out of blasting areas, reducing labour utilisation and machine availability; also insurance premiums are increased where blasting is practised, etc. Ripping can also often be carried out where blasting is forbidden.

Public relations. Ripping is preferable to blasting near residential or industrial areas. Ground and air vibrations from blasting invariably lead to bad public relations, occupying management time which could be spent more productively.

Slope stability. The seismic waves produced by blasting can trigger off slope failures where pit slope angles are near to equilibrium.

*From "Ground Preparation by Ripping in Open Pit Mining"

Flexibility. The tractor-ripper is a relatively inexpensive versatile unit which simply by raising the ripper beam can be used for a variety of other tasks, e.g. dozing, haul road maintenance, stockpile operations, 'pushing', moving equipment, etc.

Increased crusher capacity. Better fragmentation and a drier product due to ripping can result in much higher crusher throughputs. One operator reports a sustained 40 per cent increase ⁷¹⁵.

Improved blending Where a tractor-scraper is used for loading, it has the ability to dump on a stockpile in thin horizontal layers. By reclaiming normally in vertical slices the desired blend can be obtained.

Ripper types. There are two basic ripper types, (a) the towed ripper and (b) the tractor-mounted ripper. The towed ripper finds little application in modern mining since to obtain adequate penetration in harder rocks the weight of the ripper becomes excessive and a huge tractive effort is needed. There are five basic types of tractor-mounted rippers where the ripper beam (or tool beam) is pivoted from the rear of, and forms part of, the tractor assembly:

Straight bar rippers can mount up to three shanks (or tines). They are of lighter construction than most other rippers and are unsuitable for hard ground or where boulders occur.

Parallelogram rippers maintain a constant tip angle of penetration of the shank as the ripper beam is lowered and provide positive control over ripping depth. Where the

angle of penetration must be altered at frequent intervals, e.g. due to changes in rock type when working at different horizons in a mine, this can be a disadvantage but a variety of ripper shanks is available for different rock conditions (Fig 7.12)

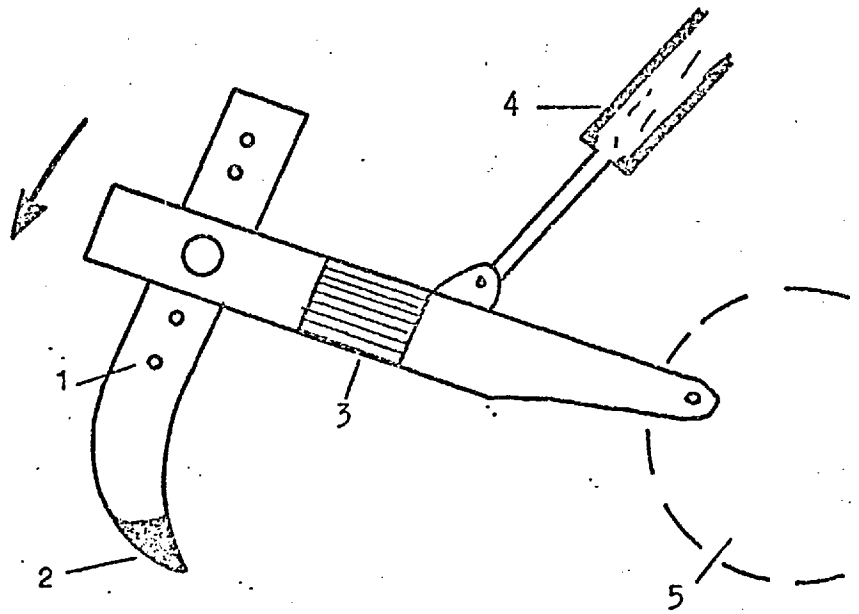
Radial rippers are of extremely simple robust construction, the ripper beam being raised or lowered in an arc by hydraulic cylinders (Fig 7.12). The tip angle of penetration can be varied by altering the angle of the ripper beam, but control over the depth of penetration is lost. This disadvantage can be overcome by raising or lowering the shank in its pocket.

Vibratory rippers are not suitable for dense materials but give additional impact in loose, porous or broken materials and improve production. This type loses the main advantage of the ripper i.e. simplicity, and has the disadvantage of increased capital and maintenance costs. They are not widely used.

Controlled angle rippers are equipped with hydraulic cylinders to vary the shank angle allowing control of both the tip angle of penetration and the depth of penetration, so the ripper shank can always be set to give optimum breakage of the rock. This type of ripper is a relatively new development but has not so far found wide use because of its greatly increased maintenance costs.

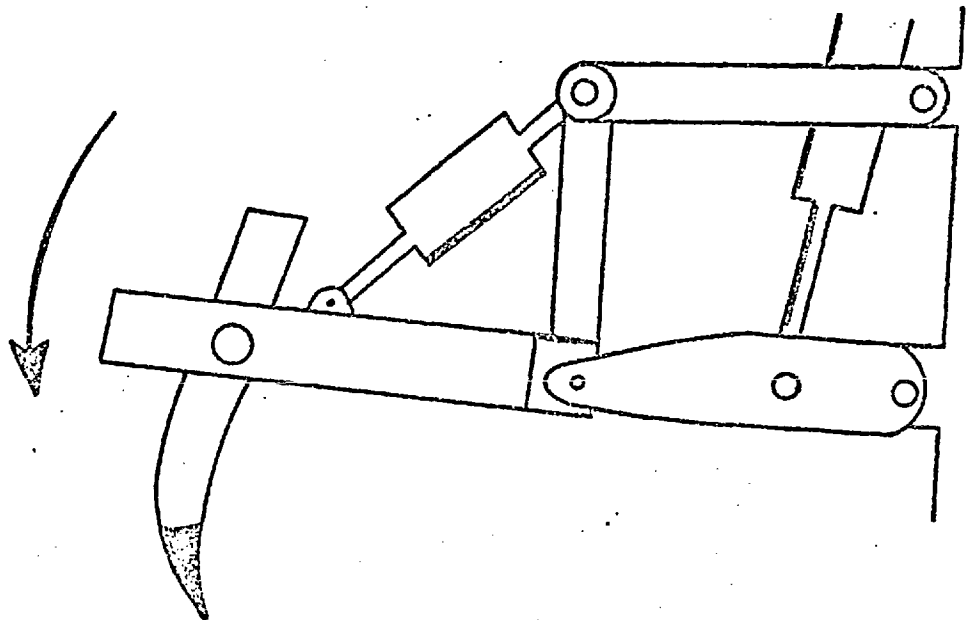
Determination of rippability Whether or not a rock can be economically ripped is dependent on the following factors:

- (a) Natural brokenness, i.e. the degree of jointing, bedding, fissuring, fracture, etc.



1. Shank or Tine
2. Point
3. Ripper Boom
4. Hydraulic Cylinders
5. Crawlers of Tractor

Radial Arm Ripper



Parallel Action Ripper

FIG 7.12 RIPPER TYPES

- (b) Intact rock strength.
- (c) Rock texture, i.e. abrasiveness, porosity, brittleness, etc.

Some of these factors show a degree of correlation, for instance the porosity of a rock is a fair measure of its intact strength and porous, non-abrasive rocks are usually excellent ripping propositions. The most obvious method is to determine the rippability of a rock by trial and error, but this is seldom possible in openpit mining due to the size or depth of a proposed operation.

The most common materials and their ripping characteristics are:

Topsoil - loose, surface material which does not usually require any ground preparation, except tree roots which may require ripping.

Glacial till - granular, non-cohesive materials, ranging in grain size from sand to boulders. This material is readily prepared by ripping.

Clays - cohesive, fine grain size material. Moist clays require little or no ground preparation and in some circumstances loading is not improved by ripping. Dry hard clays can be easily ripped but there are a few notable exceptions.

Sedimentary rocks - these include chalk, sandstones, limestones, coal, etc. Because of the presence of joints, bedding planes, cleavage, fissures, fractures, etc., they are generally easy to rip, and to load with tractor-scrapers but some strong carboniferous limestones are not suitable for ripping.

Metamorphic rocks - schists, slate, quartzite, etc.

These are rocks which have been changed in character by tectonic action, e.g. heat, pressure, chemicals, etc.

Usually stronger than sedimentary rocks, their rippability depends largely on their degree of brokenness. Metamorphic rocks tend towards blocky fracturing and can be unsuitable for tractor-scraper loading.

Igneous rocks - these are formed by the solidification of magma and are dense; usually without any form of stratification. They are generally too strong to be economically ripped although there are some exceptions, particularly if fractured or jointed.

In general if a material is of low intact strength, has a high degree of brokenness, or a combination of both, it can be economically ripped.

Rock conditions which are not conducive to ripping are:

- (1) Massive, monolithic structures with a very low degree of brokenness.
- (2) Fine-grained, materials with strong, non-porous cementing agents.
- (3) Non-crystalline, non-brittle materials.
- (4) Plastic materials, e.g. wet clays.

By far the most common procedure for determining rippability is seismographic testing. This well-known, geophysical method has the great advantage of being quantitative and makes use of the principle that seismic waves travel through subsurface materials at different speeds, dependent upon the hardness, degree of brokenness and the orientation, openness, filling, continuity and surface texture of the planes of the breaks. Seismic waves travel through loose top soil at about only 300m/sec (1,000 ft/sec) but at 6,000 m/sec (20,000 ft/sec) X

through a hard, intact rock. The technique generally provides a much quicker indication and is more economic than core drilling but of course the two are often used in conjunction with each other.

Rock classification

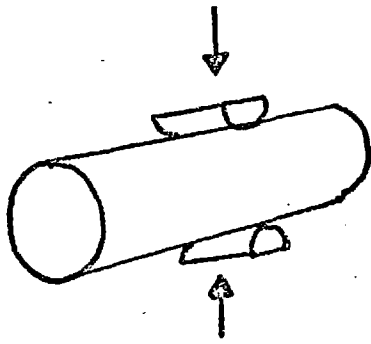
An interesting approach has been suggested by Franklin 716, for the classification of rocks. The method avoids the difficulties which may be experienced with seismograph test interpretation. It makes use of simple rapid methods of measurement which do not require special skills and employ simple robust apparatus; plus the fact that in most mining operations cores are available from exploratory drilling.

Rippability is largely dependent on the intact strength and the brokenness of a rock. Intact rock strength is commonly defined by the uniaxial compressive strength, but its determination requires laboratory apparatus and specimens must be machined prior to test, a difficult problem with weak or friable rocks. Most rock strength estimates, e.g. compressive and tensile, correlate closely with one another so that a reasonable strength indication can be determined from one test. Depending on the samples available Franklin recommends three tests:

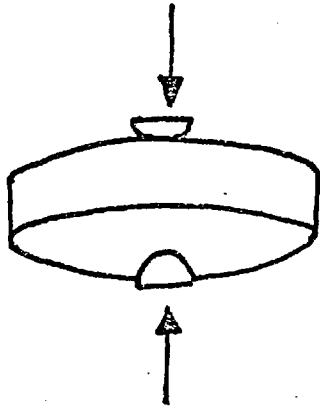
- (1) For a long core, a diametral point load test (Fig 7.13a)
- (2) For short length cores, i.e. 'discs' an axial point load test (Fig 7.13b)

Where core splitting must be carried out (1) and (2) avoid the objections of the geologists since the cores are not destroyed.

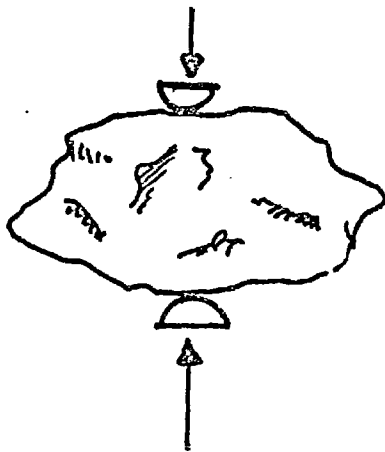
- (3) For irregular lumps, where rock aggregate or outcrop samples only are available, an



a) Diametral Point Load Test on a Long Core



b) Axial Point Load Test on a Short Core (disc)



c) Axial Point Load Test on an Irregular Lump

FIG 7.13 RAPID METHODS OF DETERMINING STRENGTH INDEX, I_s

axial point load test (Fig 7.13c)

A common point load strength index for all three tests is defined as:

$$\frac{\text{applied load}}{(\text{distance between loaded points})^2}$$

This index is proportional to the direct tensile strength of the rock specimens used, so that the index itself may be used as a direct indication of the intact strength of the rock. The apparatus for such tests is light and portable since only small forces are involved.

Fracture index

The definition of brokenness presents more complications since to be fully descriptive it should include the orientation, roughness, openness, continuity, filling and alteration of the joints, planes or fissures. As a simple field measure, however, a fracture index defined as the average linear size of blocks that constitute the rock mass is used. This can define a closely-fractured, soil-like material at one end of the scale to a monolithic mass such as an igneous rock at the other. Such an index can be easily visualised and can be readily mapped. The fracture index can be calculated from the average lump size in a heap of excavated rock or from the average spacing of fractures in a core or along an outcrop.

Fig 7.14 shows the logs of fracture index and point load strength index for the horizons A to F of a borehole. These are plotted on a classification diagram (Fig 7.14) to indicate the rippability of each horizon. Rippability plans and sections can be plotted from the classification diagram.

The method at present suffers from a lack of field performance data but has been successfully applied in several

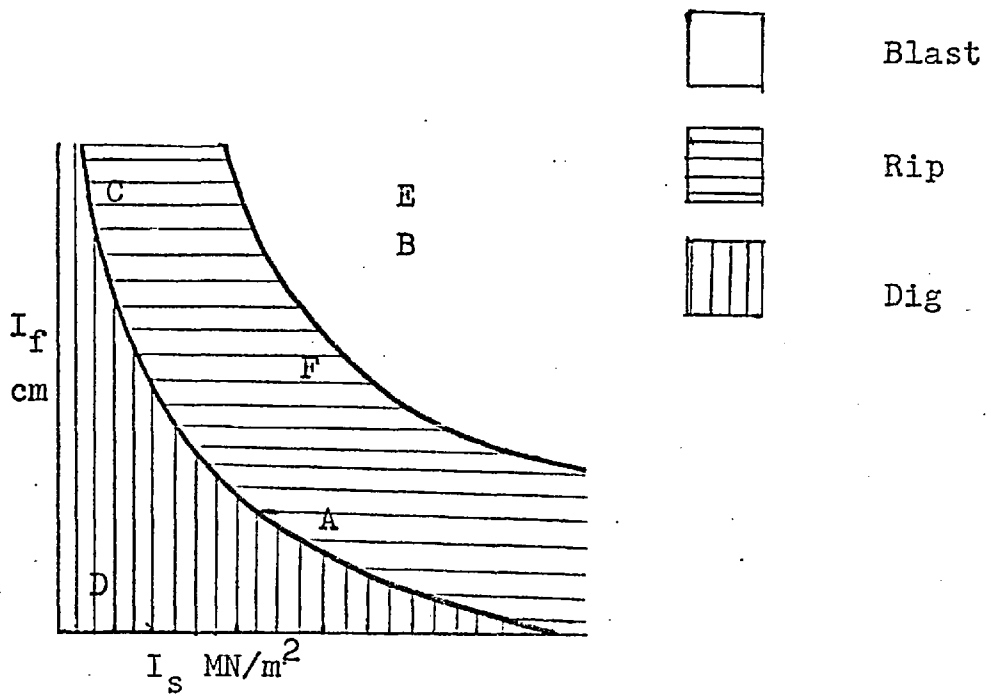
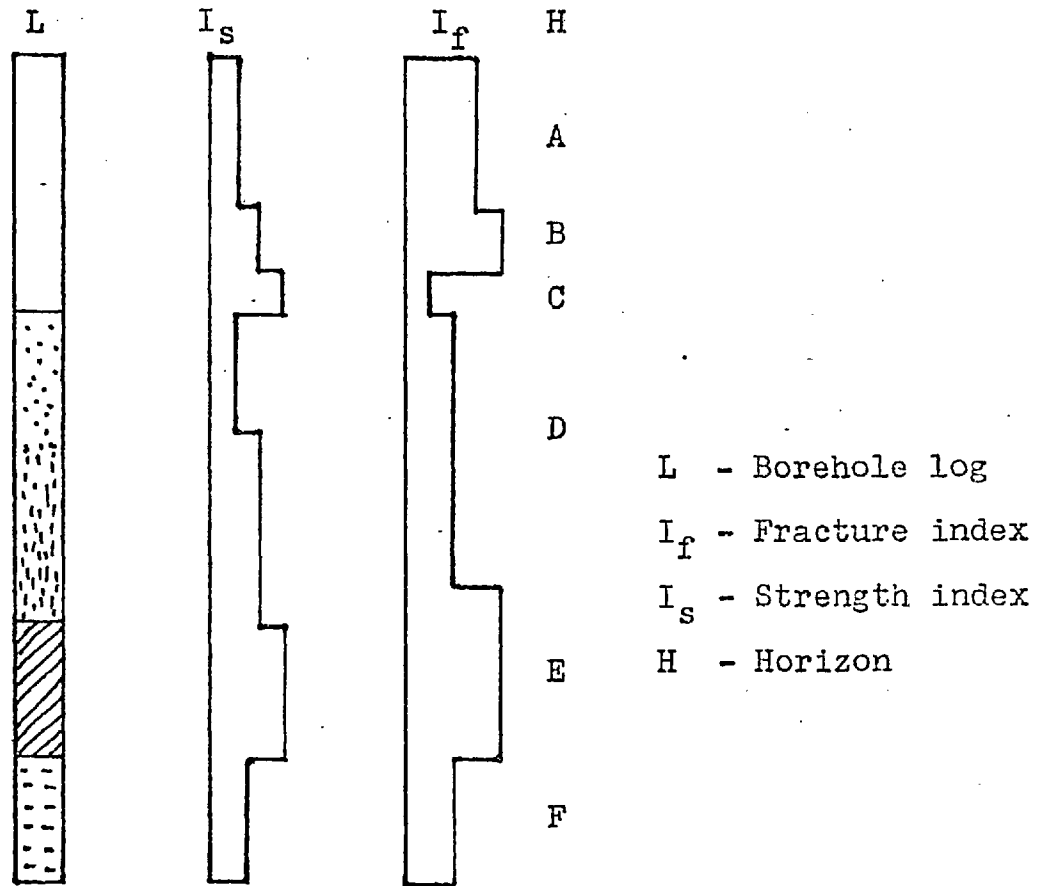


FIG 7.14

DETERMINATION OF RIPPABILITY FROM
 STRENGTH AND FRACTURE

locations and shows particular promise for mining applications.

Ripper selection

Individual preferences (and prejudices) enter into ripper selection but experience gained over the past ten years can act as a guide to selecting the right type of ripper for the prevailing rock conditions. For construction jobs a very wide range of applications may be necessary but in mining the duties of a ripper will generally be more specific. The key to economic production is to use the following combination:

- (a) Select the correct tractor size for the rock. A 30 ton tractor will rip material that a 20 ton tractor cannot (Fig 7.15)
- (b) Match the production required to the machine size. The larger the tractor the higher the hourly operating costs, so if a smaller tractor will do the job do not select a larger unit. Equally do not select a unit for a duty outside its range.
- (c) Select the ripper beam and ripper shank which is suited to the duty.

The important characteristics of the main types of ripper beam assemblies can be summarised as follows:

Straight bar

Multi-shank. Suitable for relatively easy ripping i.e. top soils, glacial till (without boulders), chalk, weak sandstones, etc. Produces relative high volumes in good conditions but is unsuitable for slabby material or occasional boulders as the distance between tractor and shank is usually less than 900mm (36 in) and large lumps

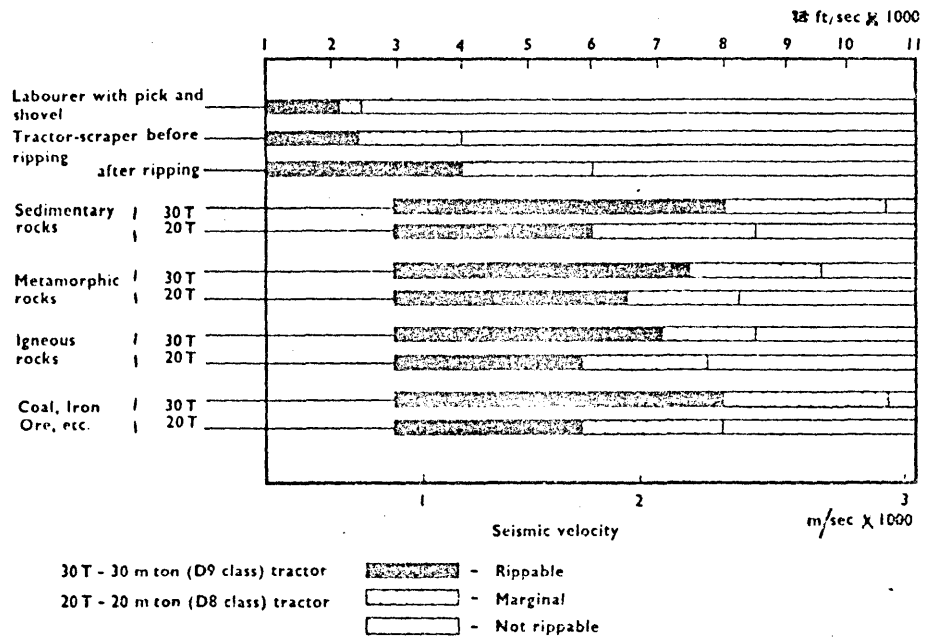


FIG 7.15 RIPPING CAPABILITY
(30t and 20t tractors)

become trapped between the shank and the rear of the tractor tracks.

Single shank Suitable for up to medium strength or broken limestones, glacial till with small boulders, etc.

Parallelogram

Multi-shank Widely used for glacial till (without large boulders), medium or broken limestones, coal, etc., but does not perform well in slabby materials as 'raking' can occur.

Single shank Suitable for the most difficult materials but if the tip angle of penetration must be altered at frequent intervals, minor delays due to altering the shank angle are experienced.

Radial

Multi-shank As for multi-shank parallelogram rippers but due to the shank configuration this design is not quite so susceptible to 'raking'.

Single shank Suitable for the most difficult materials, including some igneous rocks, e.g. basalt.

The 'narrow gauge' ripper has found some application in mining particularly for hard rocks. Two closely spaced shanks are fitted to a ripper beam within the tractor tracks. This considerably reduces 'lugging' action by the tractor and greatly reduces track wear. Only shallow penetration is possible but fragmentation is excellent and good production rates can be achieved. This method is entirely unsuitable for slabby or blocky materials as the tracks suffer excessive damage and wear.

Multi-shank rippers perform best in easy conditions and with three shanks good fragmentation over a wide path is obtained. With two shanks, one in each of the outer pockets of the ripper beam, a wide path is ripped, the broken material being relatively coarse. Shallow penetration depths are achieved with multi-shank rippers. Single shank ripping is essential in difficult conditions if maximum penetration is to be achieved.

Matching shank to duty

There are a wide range of ripper shanks available and most manufacturers will advise on the right shank for a specific application. Matching the shank design to the duty is of vital importance for economic operation since greater production, better control of fragmentation, reduced tractive effort and longer component life are obtained.

Straight shanks are particularly suitable for slabby and blocky materials and although not so popular with construction contractors they find a wide range of application in mining and quarrying.

A number of curved shanks are available. Their shape gives a lifting action that results in good fracture characteristics particularly in fine grain, unbroken material. Of these the single-offset shank produces a wide fracture area and is best suited to easily penetrated materials. The double offset shank is ideal for non-abrasive conditions. A self clearing design is available for easily penetrated materials such as coal. In abrasive conditions wear plates are essential. Apart from protecting the shank from excessive wear, they also reduce tractive effort due to their self-sharpening characteristics. Because of its versatility the double-offset shank fitted with wear plates

and mounted on the low capital-cost radial ripper beam is probably the most widely used ripper configuration.

The correct shank tip is vital to good performance and a sharp tip is essential to achieve penetration. In harder rocks the tip should be changed regularly to maintain production and a low-cost, short tip which is highly resistant to breakage but has less metal for wear resistance is usually most economic. Medium length tips have better wear resistance, good resistance to breakage and find wide application. For weaker but highly abrasive rocks where resistance to breakage is not so important, e.g. quartz sandstones with a weak matrix, a long tip with adequate metal for wear is most suitable.

For the most economic production, the longest tip which gives maximum wear without excessive breakage should be selected.

Ripping operations

Ripping has not reached the same general level of acceptance in mining as in the construction industry. Whilst it is widely used for the mining of industrial minerals and in overburden stripping, the harder, stronger rocks encountered in metalliferous mining, coupled with some degree of conservatism, have impeded the application of ripping in this field, even where the rocks have a high measure of brokenness.

There are three basic operations practiced in mining and quarrying:

(a) 'Rip-Scrape-Haul-Dump'

The rock is ripped to provide the right product size. The broken rock is loaded by tractor-scraper, often assisted by a 'pusher' vehicle, usually a bulldozer. The pusher

vehicle then 'boosts' the tractor-scraper up to speed for its haul journey to the dumping point. After dumping the scraper returns to recommence the cycle. This method may be unsuitable in blocky or very hard rock conditions.

(b) The rock is ripped, loaded into trucks, usually by a wheel loader for mobility, and hauled to the dumping point; or directly transported by the wheel loader as in contour mining stripping operations.

(c) The rock is ripped along the top of a bench, dozed over the bench edge to the bench below where it is loaded into trucks by shovel or wheel loader. This method has advantages where a good haul road surface over the ripped bench surface is difficult to maintain. In contour mining operations where a short haul is involved, overburden may be stripped by dozing the ripped material from the highbank side to the spoilheap side of the pit.

Pit design

In optimising pit design for an irregular or stock-work deposit where there are significant grade variations it is usual to assign a value to each mineable block. The configuration of these blocks must be fixed to suit the mining method so that the block dimensions for a ripping operation will be quite different from those for a conventional blasting operation. Similarly the method of assigning a value to a block must be influenced by the thin horizontal slices obtained by ripping.

Although modern rippers can penetrate right up to the toe of a bench face and there are no serious difficulties experienced in maintaining the correct pit slope angles, the ripper must usually penetrate the bench toe at 90 degrees to

the normal direction of ripping. This usually means that the operation must be performed separately to the main operation. An alternative is to use the rip dozer, (see previously).

With a single shank extended and the dozer blade tilted the rip dozer can rip away the toe of the slope to the correct profile in a single forward movement.

Where tractor-scrapers are used, some attention to detail is necessary when determining bench width to ensure adequate space for manoeuvring. Experience has shown that where this method is adopted in deep, multi-bench, roughly conical shaped pits, there is some advantage in mining each bench to its full pit limits before starting the next lower bench.

Fragmentation

In construction work the major objective is to break the rock just sufficiently so that it may be economically loaded. In mining, however, a small sized mill feed is desirable for most ores and minerals and there may be advantage in fragmenting the ore to a smaller lump size than is really necessary for loading. This can be accurately achieved by multiple passes, control of ripping depth and cross-ripping, although cross-ripping in rocks is deprecated by many practising engineers, who prefer to rip in the direction of scraping or loading only. Where tractor-scrapers are used for loading, the additional costs of ripping down to a small lump size are well covered by the savings in scraper operating costs, due to reduced scraper cutting edge wear, tractive effort and impact loads, as well as ensuring better filling of the scraper bowl. Ground preparation costs cannot therefore be considered in isolation because it is the total cost which

is of importance (Fig 7.16). Because of better control of, and the increased fragmentation that can be achieved by ripping it can often show savings in total cost when the ground preparation costs are the same as those for blasting.

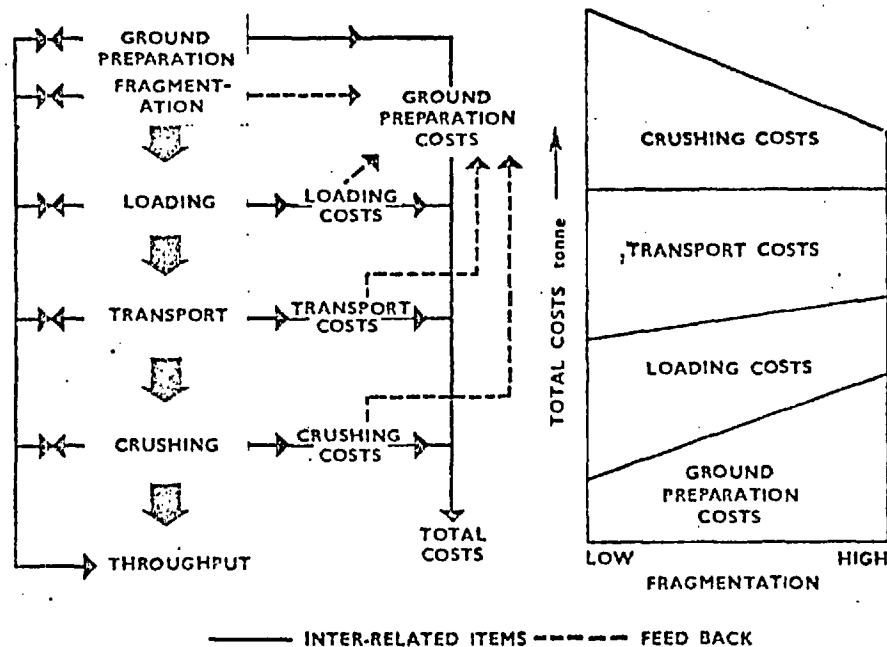
In overburden it is only necessary to fragment the rock sufficiently for efficient loading. For face shovel loading the maximum lump size should not exceed half the dipper dimensions so that the broken rock will flow freely through the dipper door. The bucket of a wheel loader is not subjected to the same restrictions, but a well fragmented material will give reduced loader operating costs due to the reduction in mechanical shock, wheel spin and because of better bucket filling. If bulldozing is employed for overburden transport it is only necessary to rip the material sufficiently to allow the dozer to operate to its full capacity.

Ripping sequence

The layout and sequence of ripping runs must be carefully planned to ensure the lowest cost of production. The direction of the run should take advantage of joints, stratification, etc. The ripping direction should be:

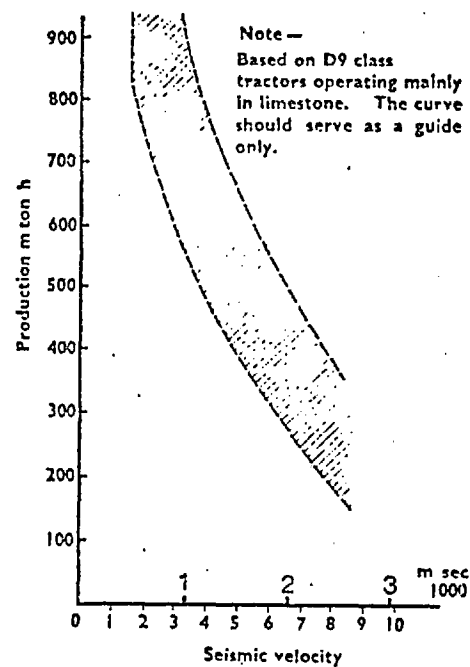
- (a) At 90 degrees to any vertical joint planes, and
- (b) down-dip of any inclined stratification planes.
- (c) The ripping depth should take advantage of any suitable horizontal stratification plane, and
- (d) downhill ripping should be arranged where possible.

Ripping runs of 70m (200 ft) to 90m (300 ft) usually give best results. Careful consideration must be given to the planning of the whole ripping, loading and transport operation to ensure that one individual part does not



(a) Logic Diagram
Cost relationship in a mining operation

FIG 7.16



Approximate production
by ripping with D9 class tractor

FIG 7.17

interfere with another. Adequate breakage can usually be obtained by spacing ripping runs about half a tractor width apart but the spacing can be closed up to obtain smaller fragmentation.

When ripping virgin ground, in wet conditions or where a stratification plane leaves a hard smooth surface, greatly improved performance can be obtained by lightly scarifying the surface before commencing ripping to improve traction. Where possible the ripping depth should be adjusted to allow a slow forward speed of approximately 3 km/h (2 mile/h) to be maintained since this is usually found to be most productive, greatly reduces track wear and avoids impact shocks.

Many mine operators, probably because of difficult ripping conditions, prefer single shank to multi-shank operation, as they claim it gives less shank wear, better operator control, less track wear due to reduced slipping and 'lugging', less mechanical shock and usually greater production. A shallow ripping depth is generally preferred, again probably because of the relatively difficult conditions experienced in mining and greater production is claimed due to reduced track wear, better traction conditions for 'pusher' tractors and improved travelling surfaces for loading and transport equipment. All mine operators prefer a sophisticated transmission system, e.g. 'power-shift', to direct-drive tractors.

Close attention to all the foregoing points invariably improves efficiency and reduces production costs.

Sun-drying of chalk, gypsum, etc., is much more efficient if the material is ripped several days before excavation.

This form of aeration is also useful to prevent frost penetration which would impede excavation. The East German openpit brown coal mines rip the bench surfaces in winter time, part of the ripped material being dozed over the edge of the bench onto the slope to form an insulating layer ⁷¹⁷. When ripping frozen ground, the ripping shank can become brittle and it often pays to heat the shank by a small flame fed by a butane gas cylinder mounted on the ripper beam.

Ripping v. Blasting

A general comparison of costs between ripping and blasting is extremely difficult to make. Both methods are seldom employed in the same circumstances in the same mine so that calculated costs must generally be used for comparison purposes. Calculations for an Oregon operation ⁷¹⁸ showed that for a fairly difficult rock condition the total costs were in favour of a 'rip-scrape-haul-dump' operation. As conditions moved towards solid rock a break-even point was reached and beyond this ripping was not economic, or feasible.

For modern blasting operations in medium rocks the following ground preparation costs which include depreciation and interest, are shown in Table 7.XIX.

TABLE 7.XIX

COMPARISON OF GROUND PREPARATION
COSTS IN MEDIUM ROCKS (FOR A MEDIUM
LIMESTONE, 4,000 to 5,000 ft/sec)*

	Ripping	Drilling and Blasting
	pence	pence
Large scale		
Cost/m ³	2.8	6.6
Cost/yd ³	2.1	5.0
Small scale		
Cost/m ³	21.7	25.3
Cost/yd ³	16.7	19.3

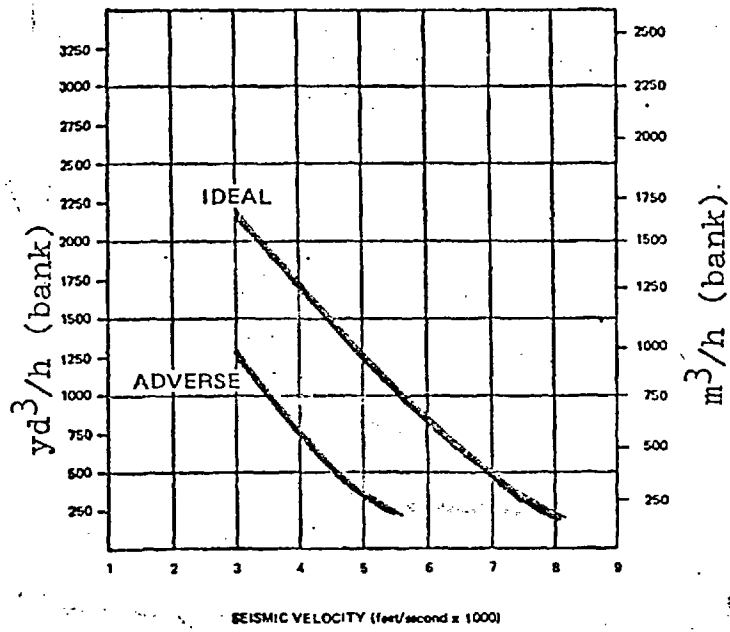
*These figures are compiled from a wide range of sources and in many cases allowances have had to be made for fixed charges, e.g. depreciation, interest, etc. Reported figures which vary markedly from the general range of values have been omitted.

The cost per ton of ground prepared by ripping is:

$$\frac{\text{hourly ownership} + \text{operating costs}}{\text{hourly production}}$$

For both blasting and ripping hourly production is somewhat difficult to forecast as it depends to a considerable extent on different rock properties and the desired degree of fragmentation. A number of works are available on blasting calculations ^{719, 720} but in the final outcome some form of 'site factor' based on judgement and inspection must be used. There are no equivalent comprehensive figures available for ripping but Fig 7.17 which is based on a number of quarry operations using D9 class tractors, mostly in limestone, can be used as a very approximate guide. Unfortunately the decision to rip in many successful operations

ESTIMATED RIPPER PRODUCTION D8H with 8D Single Shank



ESTIMATED RIPPER PRODUCTION D9G with 9D Single Shank

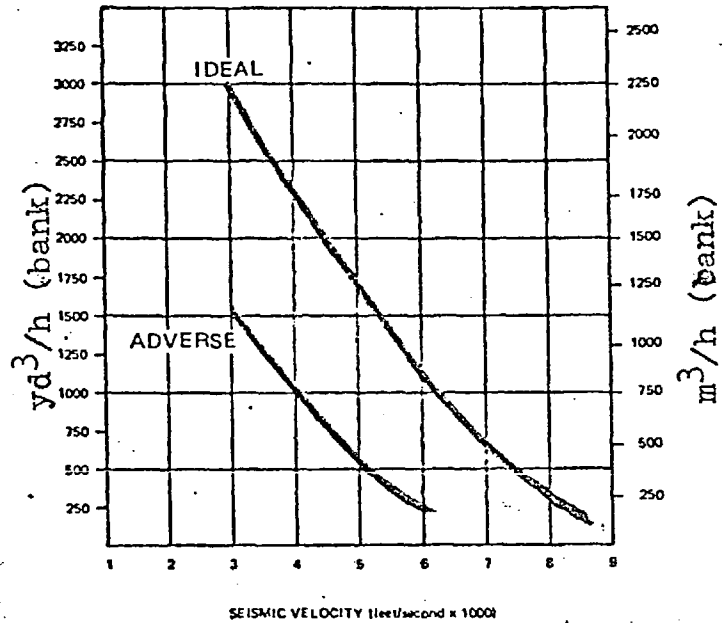


FIG 7.18 RIPPER PRODUCTION 76.

was taken after visual inspection by experienced personnel and no quantitative measures of rippability were made. Because of this lack of field performance data it is difficult to assemble a reliable quantitative guide.

Fig 7.15 shows for seismic velocities greater than 2,000 m/sec (7,000 ft/sec) to 2,500 m/sec (8,000 ft/sec) that only low outputs can be ripped. This performance can be improved by tandem ripping using a 'pusher' tractor behind the tractor-ripper. The hourly ownership and operating costs are doubled but usually the output is more than doubled, resulting in a reduced operating cost per ton. There are some applications, however, e.g. rocks of high intact strength and a very small degree of brokenness, where only explosives can be used successfully. Manufacturers literature have adopted similar curves to those used in Fig 7.17. A typical example ⁷⁶ is shown in Fig 7.18. As a general rule, ground preparation using good ripping techniques is more economic than blasting in materials with seismic velocities less than 3,500 m/sec, for up to medium-large outputs.

THE COMPACTOR

Although not a loading or excavating machine the compactor is an important mobile machine used by open pit mining engineers for slimes dam construction and is included here for the sake of completeness.

Consolidation and Compaction

Consolidation is usually defined as the natural settlement of loose materials while Compaction is the process of physically densifying or packing soil-like materials and fragmented rock, resulting in an increase in mass per unit volume. It is generally accepted that the strength of a

soil can be increased by densification. The following three factors affect compaction:

1. Size Distribution
2. Moisture Content
3. Compactive Effort

Size distribution is the percentage distribution by weight of the different sizes of particles within a sample of material. A sample is described as well-graded if it contains an even distribution of particle sizes. If a sample contains predominantly one size particle it is described as poorly graded. In well graded material the smaller particles tend to fill the voids between the larger particles, allowing better compaction.

Moisture content is extremely important to compaction.

Water "lubricates" the particles, aiding them to slide into the most dense position. Water also assists clay particle bonding, giving cohesive materials their sticky qualities.

Experience shows that it is very difficult to achieve proper compaction in materials that are too dry or too wet. There is an optimum moisture content at which it is possible to obtain maximum density for a given compactive effort. The Proctor compaction curve (Fig 7.19) shows the relationship between dry density and moisture content.

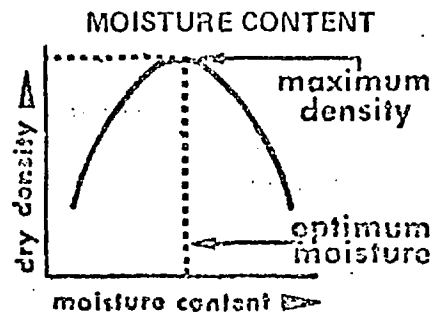


FIG 7.19 COMPACTION - MOISTURE-DENSITY OR PROCTOR CURVE

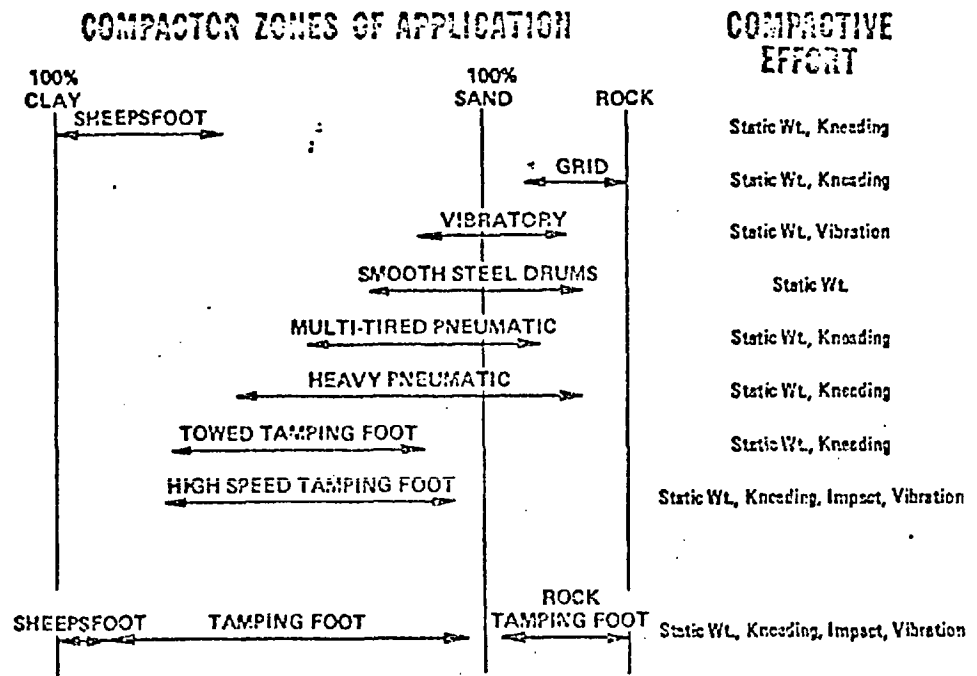


FIG 7.20 ZONES OF APPLICATION FOR COMPACTORS

Compactive Effort is the method used by the compactor to impart energy into the material to achieve compaction. Compactors usually employ one or a combination of the following types of compactive effort:

- a) Static weight (mass)
- b) Kneading action
- c) Impact
- d) Vibration

Compactor Application

The following types of compactors are used:

- 1. Sheepsfoot roller
- 2. Grid or Mesh
- 3. Vibratory
- 4. Smooth steel drum
- 5. Multi-rubber-tyred pneumatic rollers
- 6. Heavy pneumatic
- 7. Towed tamping foot
- 8. High speed tamping foot.

Combinations are available, e.g. the vibrating smooth steel drum.

The economic application zones of these compactor types are shown in Fig 7.20 ⁷⁶, for a range of materials from 100% clay to 100% sand, plus a rock zone.

Compactor Production ⁷⁶

Compactor production is expressed in compacted unit volume per hour. In the bank or solid state, material is measured in bank unit volume, after removal and when placed as fill it is measured in loose unit volume, and when loose fill is compacted it is measured in compacted unit volume. The relationship of compacted material to bank material is the Shrinkage Factor:

$$\text{Shrinkage Factor} = \frac{\text{Compacted Volume}}{\text{Bank Volume}}$$

The construction industry has developed the following formula for estimating compactor production:

$$\text{Compacted m}^3/\text{h} = \frac{\text{W.S.L} \times 1000}{P} \dots\dots\dots (7.18)$$

For a 60 min hour

where

W = Compacted width per pass (m)

S = Average Speed (km/h)

L = Compacted thickness of lift (mm)

P = No. of machine passes to achieve compaction.

P is of considerable importance and the Roads Research Laboratory standards are generally accepted in the U.K.

EXCAVATION USING A COMBINATION OF MOBILE MACHINES

The application range of most mobile machines is limited and can often be most economically used in combination. Perhaps the best illustration of this is in the Contour Mining of coal where mobile machines are particularly suited to the difficult terrain. Assuming first for comparison purposes that the terrain is level with a horizontal coal seam, and an overburden thickness not greater than about 14m (45 ft), three main mining methods are available.

1. Small dragline (say up to 15 m³ (20 yd³))
2. Bulldozer (crawler-mounted)
3. Wheel loader (front-end loader)

These methods are shown diagrammatically in Fig 7.21. It will be noted that the dragline cuts a steep highwall and forms a relatively steep spoil heap i.e. the minimum transport distance is used therefore costs should be low.

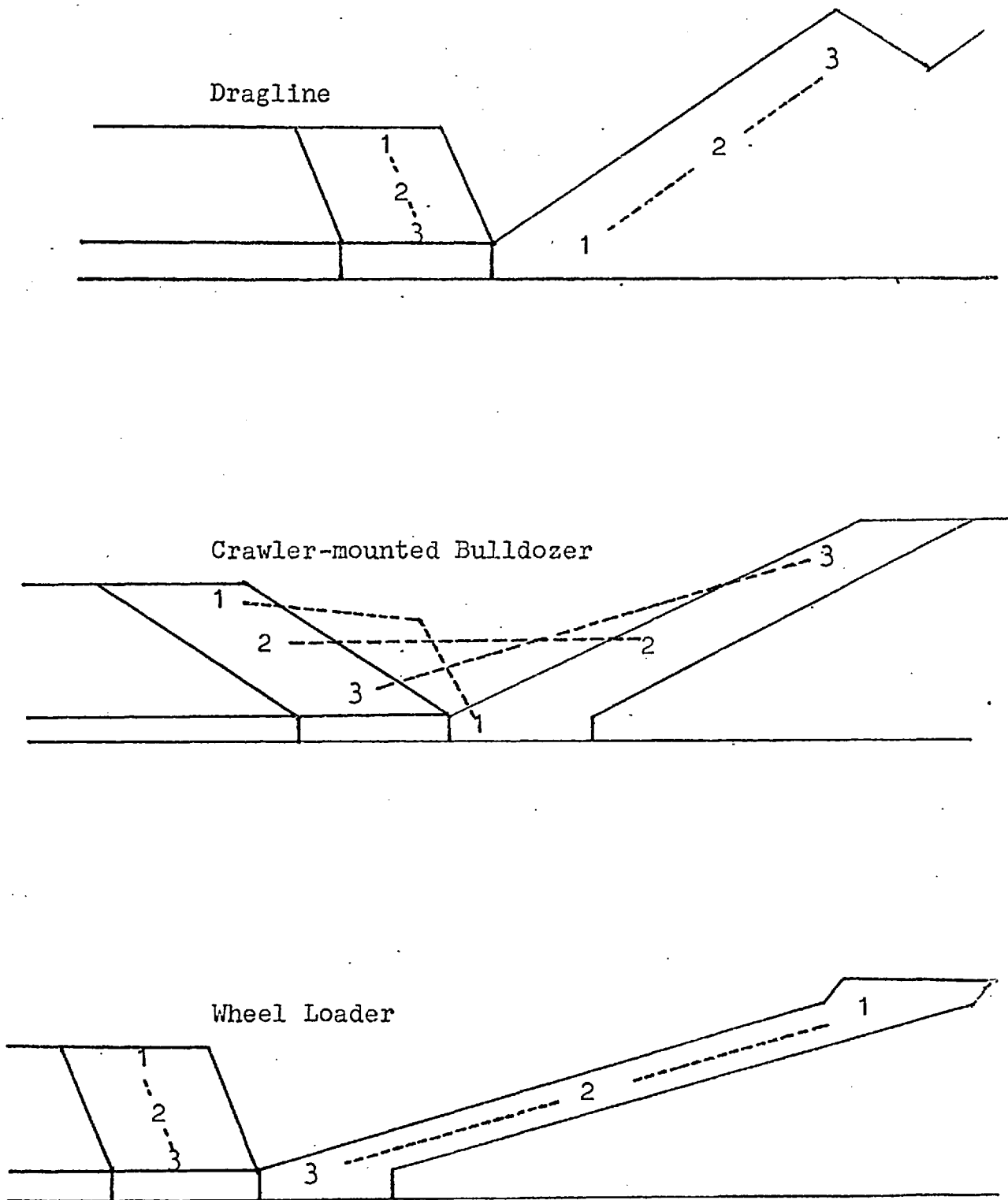


FIG 7.21 COMPARISON OF OVERBURDEN HANDLING METHOD
IN STRIP MINING OPERATIONS

The dragline has the following features:

- a) high capital costs
- b) long service life
- c) low operating cost
- d) relatively inflexible and immobile in operation.

It is therefore suitable for mining larger, more uniform deposits.

Mobile equipment such as bulldozers, wheel loaders, tractor scrapers, etc., are:

- a) highly mobile and flexible in operation
- b) relatively short service life
- c) low capital cost
- d) high operating cost
- e) not subject to obsolescence.

They are therefore suitable for mining smaller deposits, deposits with difficult topography and/or geology, changing excavation conditions, etc. Fig 7.22 shows typical costs for:

12 yd³ dragline

30 yd³ dragline

10 yd³ wheel loader

385 hp U blade dozer

for moving 1 yd³ of overburden vertically at the gradients shown in Fig 7.21. Because of the different spoil slopes for dozer and wheel loader methods the horizontal distances required to obtain these vertical lifts vary.

Fig 7.23 shows typical hourly production rates for the configurations shown in Fig 7.21 for:

10 yd³ wheel loader

385 hp U blade dozer

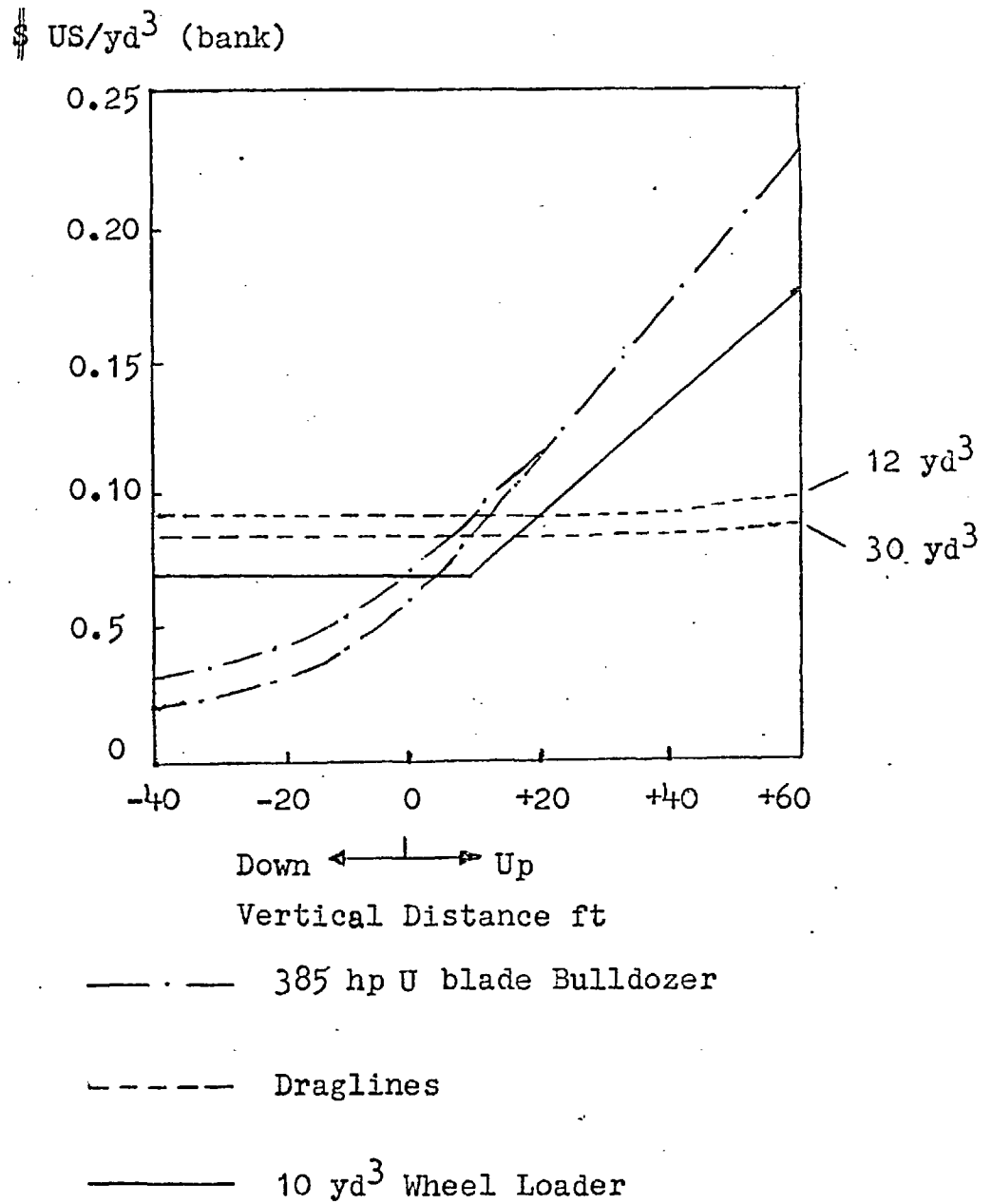


FIG 7.22 STRIPPING COSTS ~ VERTICAL
DISTANCE FOR DRAGLINES,
WHEEL LOADER AND BULLDOZER

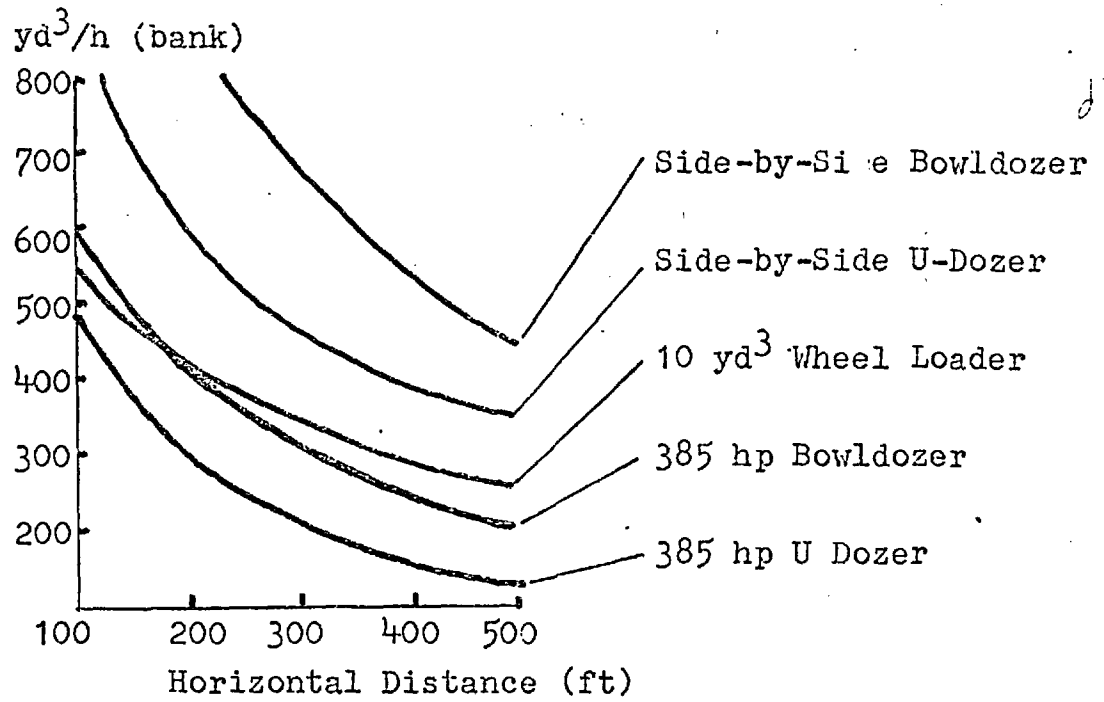


FIG 7.23 STRIPPING RATE - BULLDOZERS AND WHEEL LOADERS

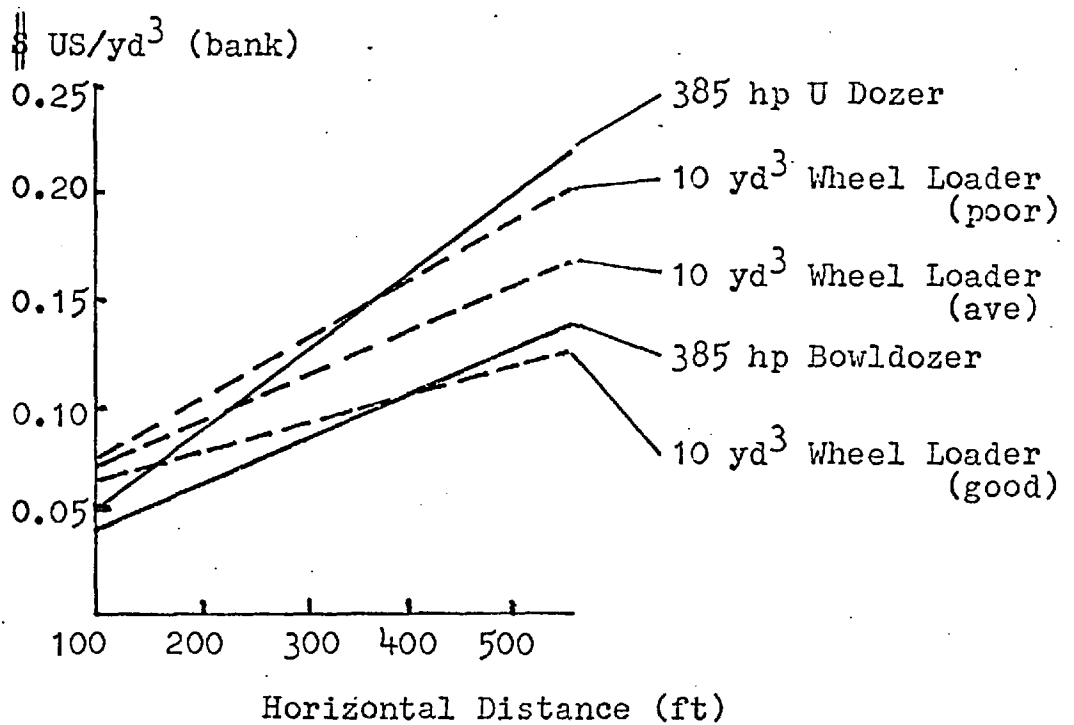


FIG 7.24 STRIPPING COSTS - BULLDOZERS AND WHEEL LOADERS

and Fig 7.24 shows the cost per yd^3 of overburden striped for these machines and illustrates how an increase in horizontal distance increases costs. Both Fig 7.23 and 7.24 are based on published figures from contour strip mining operations in the Appalachian coalfields, U.S.A. 721, 722.

For short haul distances it can be seen that the bulldozer is most economic, while for longer haul distances the wheel loader becomes more economic.

Costs are also affected by the degree of fragmentation achieved due to ground preparation. A high degree of fragmentation increases drilling and blasting costs (or ripping costs) but may result in an reduction in overall costs. Fig 7.25 shows the result of a controlled investigation carried out by the author for overburden stripping of a coal seam in Mexico.

Fig 7.26 shows how a combination of mobile machines can be used to provide the most economical method of stripping a flat, stratified deposit in a contour mining operation. After land clearance, the drill bench is excavated into the hillside. After blasting a crawler-mounted bulldozer is used to remove the top 3 - 6 m (10 - 20 ft) of overburden and transport it to the spoil bank. When the dozing distance exceeds 60 m (200 ft), a large wheel loader is used to load and carry. Dumping time can almost be eliminated by using a "bumper" pile at the far side of the spoil bank. The bucket is dumped while simultaneously reversing the wheel loader on contact with the bumper pile.

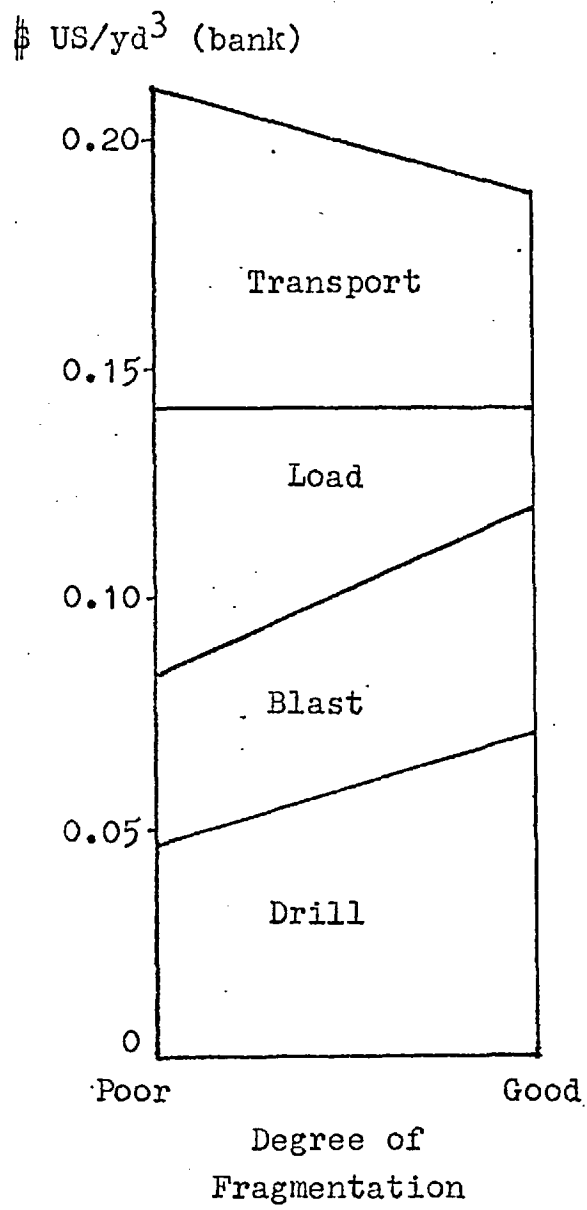


FIG 7.25 RELATIONSHIP OF FRAGMENTATION
TO PRODUCTION COSTS

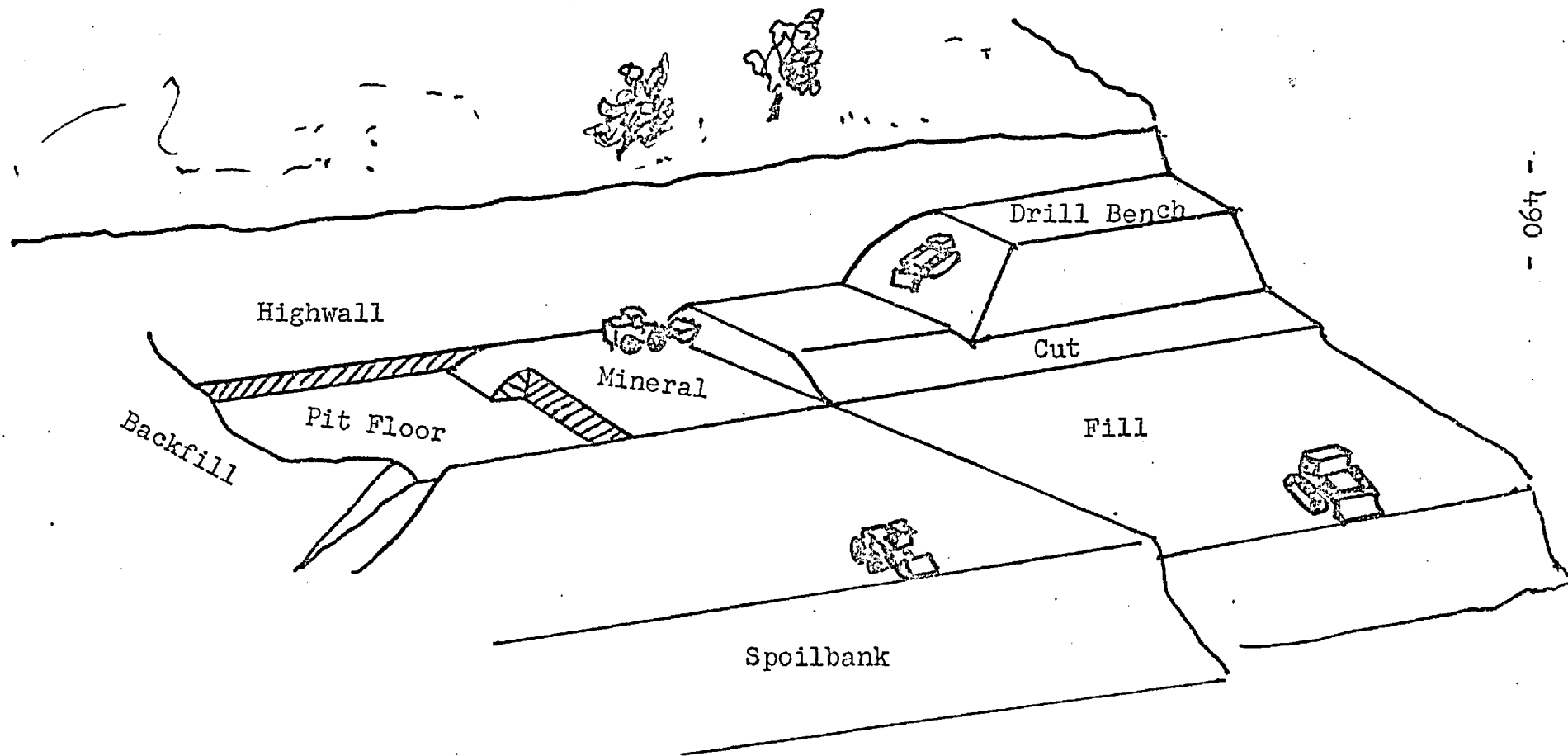


FIG 7.26 CONTOUR STRIP MINING USING COMBINATION OF WHEEL LOADERS
AND CRAWLER-MOUNTED BULLDOZERS

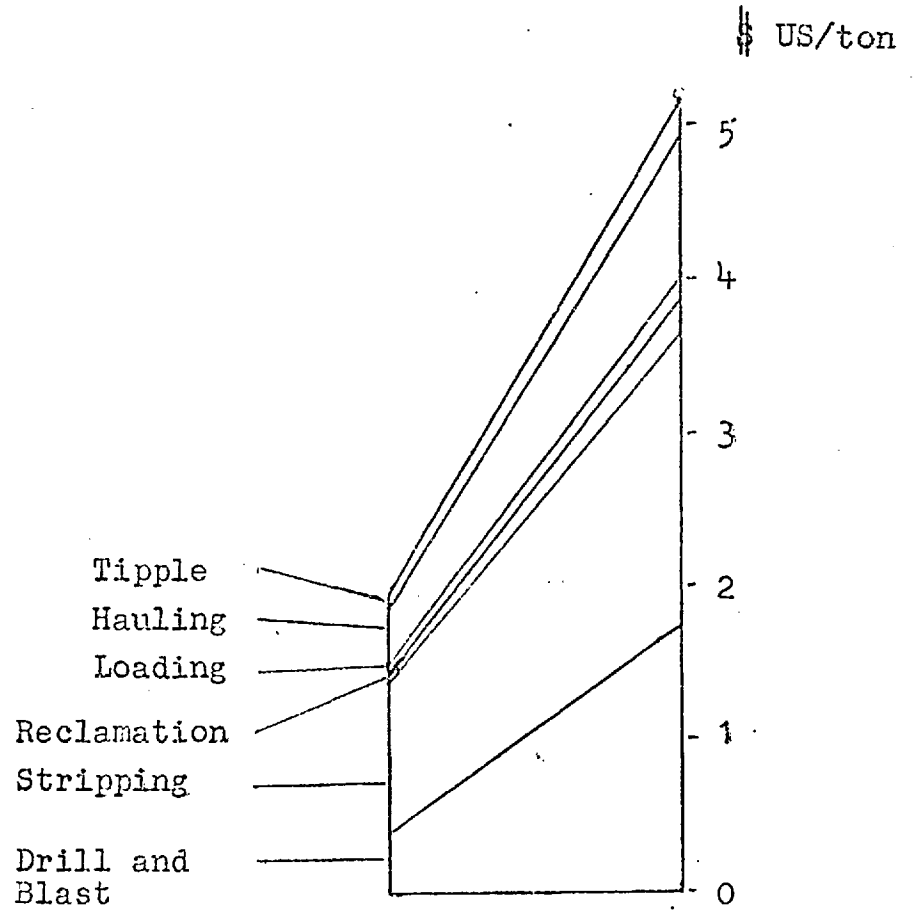


FIG 7.26a CONTOUR MINED COAL - RANGE OF COST ELEMENTS, U.S.A. (Ref 721 and 722)

An additional advantage of this method is that the total area of land disturbed at anyone time will be significantly less than with conventional strip mining, since reclamation is readily integrated with stripping.

Economically, because of lower capital cost the possible advantages are:

1. A higher rate of return on investment (ROI). The basic return on investment formula is
$$\frac{(\text{Profit gained})}{(\text{Capital invested})}$$
2. Less capital tied up in production equipment
3. Reduced loan capital, consequently less interest payable.
4. Additional capital available for other requirements.

It must be again stressed however that all the possible schemes should be compared using discounting methods. Here it is apparent that the bulldozer has cost and production advantages when dozing downhill for relatively short distances. Wheel loaders are more economic over longer distances up gentle grades. Stripping shovels and draglines perform best moving rock over short distances and forming higher spoilbanks. A combination of two or more systems often gives the most economic performace.

HYDRAULIC EXCAVATORS

Hydraulic excavators are not strictly mobile machines but they have been developed from construction industry equipment for mining purposes. They are usually more mobile than the loading shovel or crawler-mounted dragline however. Their features are:

- 1 Not particularly mobile, 3.2 km/h
(2 miles/h) is common, but a recent
machine using tank type tracks can
travel at 11 km/h (7 mile/h)
- 2 Mining type machines are crawler-
mounted and create low ground bearing
pressures, 0.3 - 1.0 kgf/cm² (5 - 15 lbf/in²)
- 3 Short turning space - highly manoeuvrable
- 4 Travel on 30 - 40% grades (steep grades)
- 5 360° swing
- 6 Short life 3 - 6 years.

Hydraulic excavators can be divided into:

- a) Hydraulic Hoe
- b) Hydraulic Dragline
- c) Hydraulic Shovel

Hydraulic Hoe

This machine is best suited for digging below grade where the machine remains above the excavated area. It can also pull material down from a working face within its dumping range (Fig 7.27). The hydraulic hoe has been used mainly for auxiliary duties in mining, in bucket sizes up to 2.5 m³ (3 yd³), for trenching, sump preparation, cleaning up ore pockets, removing thin sterile bands etc., but recently the Poclain EC 1000 model has been introduced into operations of the Opencast Executive of the National Coal Board, U.K. ⁷²³. A larger edition of other hoes, the unit has a 9 m³ (11½ yd³) bucket and a production rate of the order of 1000 tonne/h and more. It stands on the coal seam and excavates below grade. The seat earth beneath the coal seam is a soft fireclay which is unsuitable for the

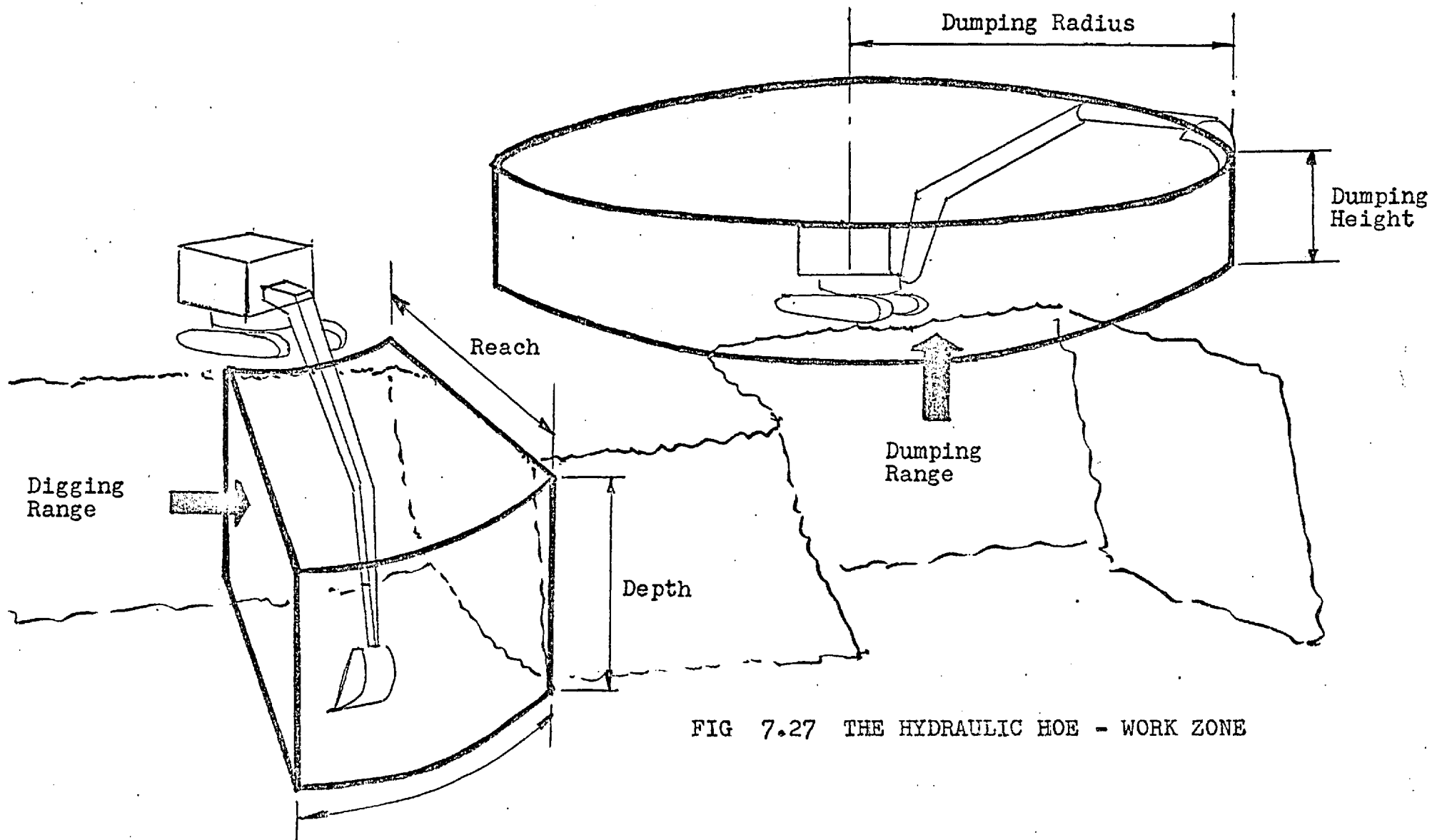


FIG 7.27 THE HYDRAULIC HOE - WORK ZONE

passage of trucks. The use of a bucket arm hydraulic cylinder can provide extremely high break-out forces which have enabled successful operation to be achieved in coal. A typical digging range envelope for a hydraulic hoe is shown in Fig 7.28.

Hydraulic Dragline

The long boom hydraulic hoe is often referred to as the "hydraulic dragline". It can perform operations similar to those performed by the crawler-mounted dragline and has the following advantages:

- 1 Precise control in positioning the bucket both for loading and for dumping.
- 2 Far better break-out force available than for the dragline
- 3 Can excavate a level base
- 4 Is selective in excavating sterile bands, etc.

To determine the suitability of a hydraulic hoe for performing as a dragline reference must be made to the digging range envelope of the machine.

Hydraulic Shovel

This form of hydraulic excavator has found greater favour in Europe than in the U.S.A. e.g. those manufactured by NCK Rapier, U.K. and Menck and Hambrock, German F.R. Additionally a unit called the "Skooper" is manufactured by Koehring in the U.S.A. Its front end horizontally crowds out hydraulically a distance of 2.5 - 3m (8 - 10 ft) depending on the model. It has a high crowd force and the tilt back action of the bucket provides a high break-out force from a working face above grade. These type of machines can handle dense rocks provided fragmentation is

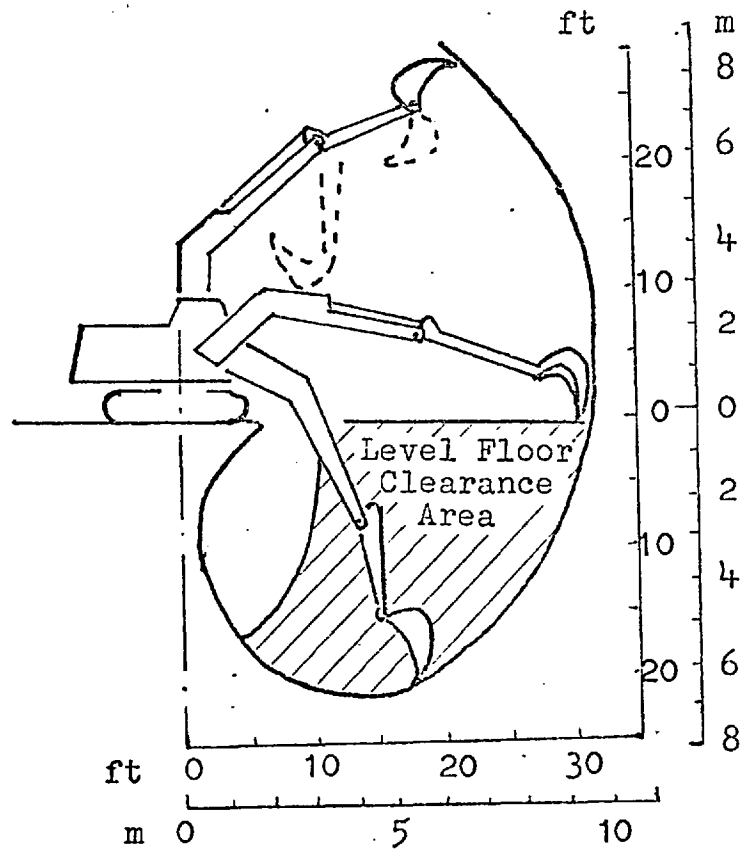


FIG 7.28 HYDRAULIC HOE EXCAVATION ENVELOPE
(Caterpillar 225)

not excessively bad.

Production Rate

Most hydraulic excavators are used for auxiliary duties in open pit mining and it is not usually possible to estimate the production rate with precision. Fig 7.29 shows the hourly capacity for common materials. These figures may be used in conjunction with the following correction factors:

Production Rate =

$$m^3/h = (\text{Rate: Fig 7.29}) \times d \times s \times m \times e \dots\dots(7.19)$$

where

d = depth factor (Table 7.XX)

s = swing factor (Table 7.XXI)

m = diggability factor (Table 7.XXII)

e = job efficiency (Table 7.XXIII)

TABLE 7.XX

d - DEPTH FACTOR - HYDRAULIC HOE *

Maxn. Depth		Ave Depth		Depth Factor
m	ft	m	ft	d
1.5	5	0.75	2.5	0.97
3.0	10	1.5	5	1.15
4.5	15	2.25	7.5	1.00
6.0	20	3.0	10	0.95
7.5	25	3.75	12.5	0.85
9.0	30	4.5	15	0.75

*Based on information from Hymac (Gt.B) Ltd.

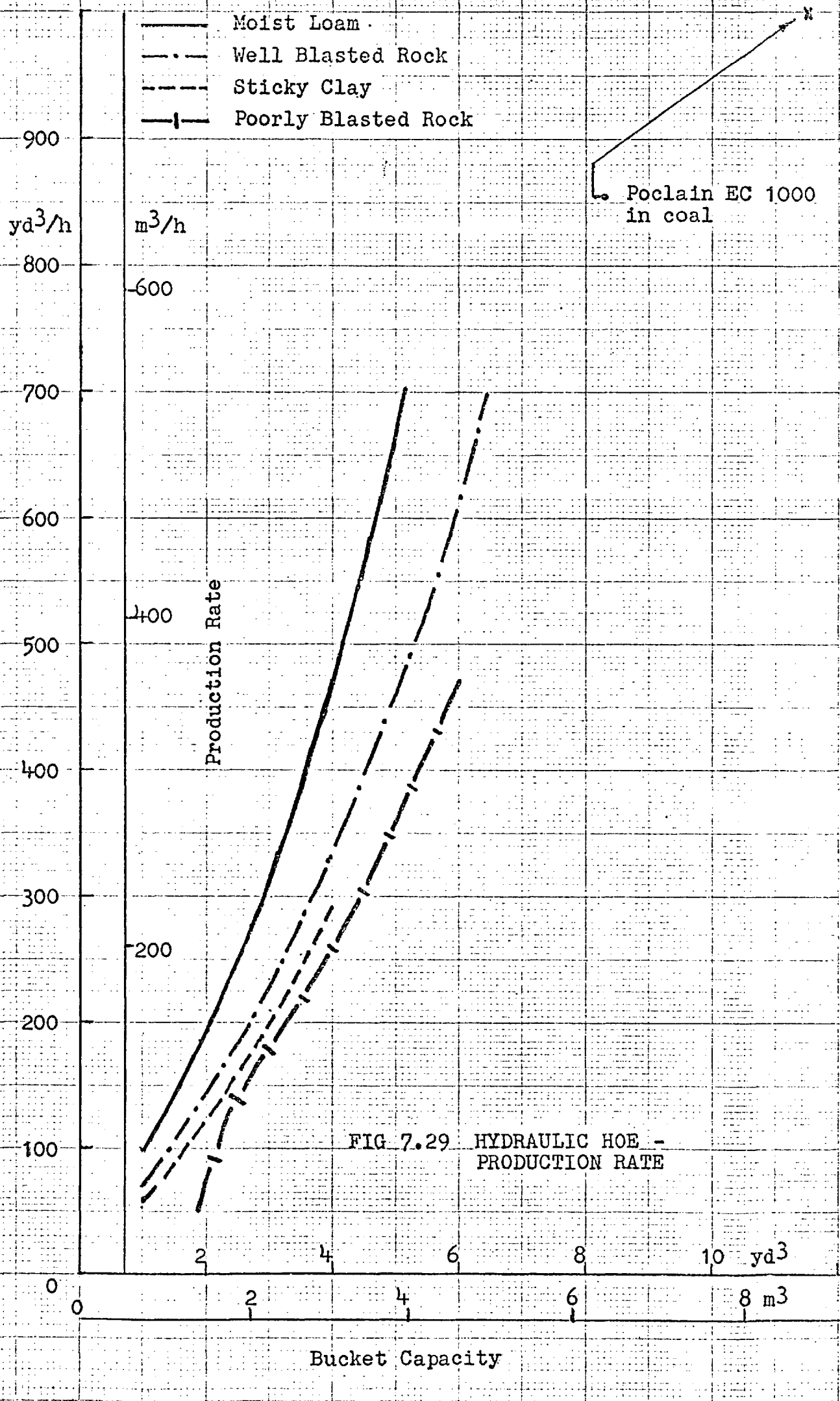


FIG 7.29 HYDRAULIC HOE -
PRODUCTION RATE

TABLE 7.XXI

s - SWING FACTOR - HYDRAULIC HOE *

<u>Swing Angle</u> Degrees	<u>Swing Factor</u> s
45	1.05
60	1.00
75	0.93
90	0.86
120	0.76
180	0.61

*Based on Hymac information

TABLE 7.XXII

m - DIGGABILITY FACTOR - HYDRAULIC HOE *

<u>Diggability</u>	<u>Factor m</u>
E	0.90 - 1.00
M	0.80 - 0.90
M-H	0.65 - 0.75
H	0.40 - 0.65

*Based on Hymac information

TABLE 7.XXIII

e - JOB EFFICIENCY FACTOR *

<u>Job Efficiency</u>	<u>Factor e</u>
Excellent	1.10
Good	1.00
Fair	0.90
Poor	0.80

*Based on Hymac information

Hydraulic Excavator Application Zones

The specific digging force of a hydraulic excavator is generally higher than those of comparable wheel loaders (front-end loaders) and crawler-mounted draglines. Fig 7.30, prepared from a wide range of collected data, indicates the order of the forces involved. It should be noted that the figures only cover up to 3 m^3 (4 yd^3) hydraulic excavators, as inadequate data was available above this size. These figures indicate that the hydraulic excavator is better suited to handle difficult materials than the wheel loader or crawler-mounted dragline.

Ownership Costs

The economic life of hydraulic excavators depends on the conditions of service, operator skill, standards of maintenance and management supervision. Unfortunately little data was available for sizes greater than 3 m^3 (4 yd^3) mainly because of the short time these machines have been in service. Table 7.XXIV has been compiled from the data collected and provides a guide in the absence of experience. The U.S. Treasury Department recognised a wide variation in economic life in 1971 and changed its depreciation rate for mining operations from 10% to 20% to bring it into line with the construction industry. The user however must eventually make the depreciation allowance consistent with his replacement policy, although he can use established guidelines during the early life of a machine.

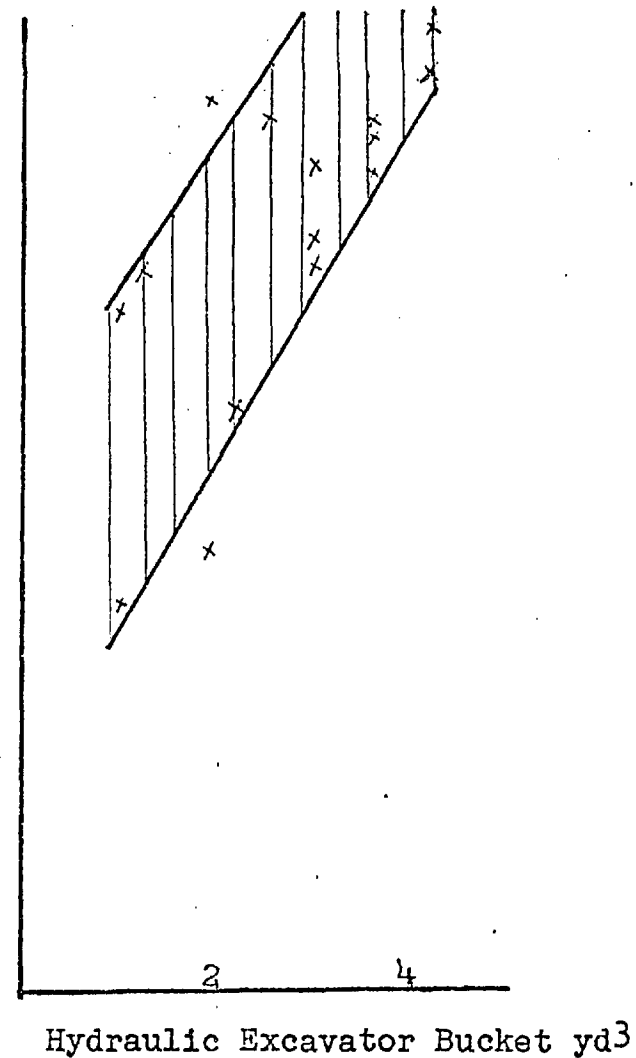
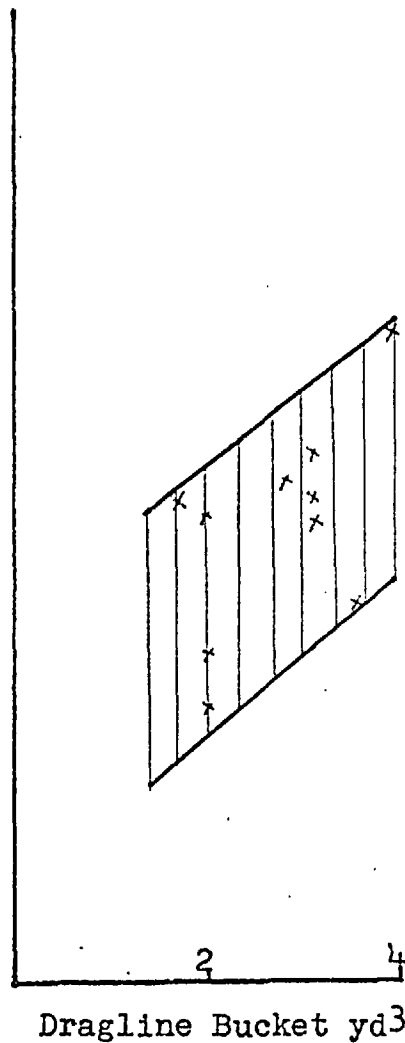
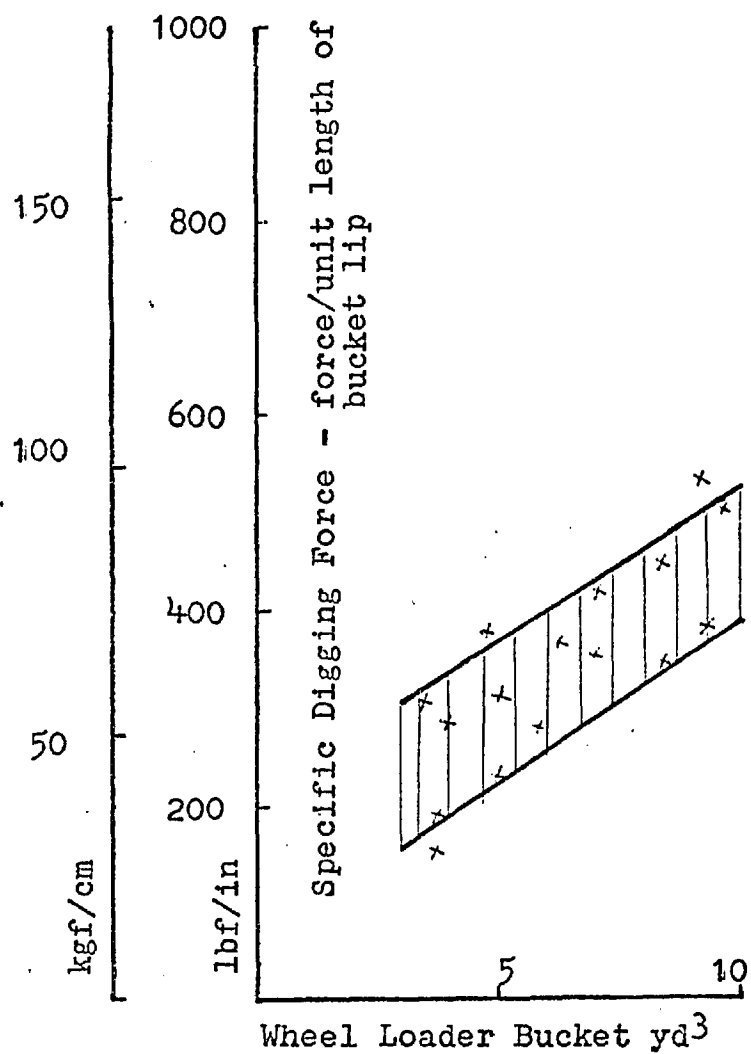


FIG 7.30 SPECIFIC DIGGING FORCES - COMPARABLE EXCAVATING EQUIPMENT

TABLE 7.XXIV

ECONOMIC LIFE - HYDRAULIC HOES

(Scheduled Operating Hours)

<u>Bucket Size</u>		<u>Hours</u>
<u>m³</u>	<u>yd³</u>	
Less than 1.0		9000 - 12000.
1.5	2.0	11000 - 14000
2.5	3.0	11000 - 18000
4.0	5.0	12000 - 19000

The format for calculating ownership costs for wheel loaders may be used for calculating hydraulic excavator ownership costs except of course that no deduction is made for tyre costs.

Operating Costs

(1) Fuel consumption
see Chapter 4, Item (22) = £.....

(2) Lubricants, etc
see Chapter 4, Item (23)
but an allowance must be included
for hydraulic oil = £.....

(3) Maintenance: repair parts and labour
M x depreciation

<u>Job Conditions</u>	<u>M *</u>
Excellent	0.8
Good	0.9
Fair	1.0
Poor	1.2

*Based mainly on data collected +
Hymac information = £.....

(4) Operator's Wages/h, including social
benefits, taxes, insurance, etc. = £.....

- (5) Total operating costs
= Items (1 + 2 + 3 + 4)

Total Ownership and Operating Costs

(6) Ownership Costs + Item (5) = £.....

(7) Cost/tonne

$\frac{\text{Item (6)}}{\text{t/h}}$ = £.....

It should be noted that no administrative, amortisation or other charges are included in these costs.

The hydraulic excavator has been in service for over 10 years but it is still a relatively recent development. Further data is needed so that a systems approach to selection procedures can be used. The next few years should make this data available.

CONCLUSIONS

Bigger more robust machines have helped mobile equipment play a larger role in open pit mining. Bigger machines prove their value because:

1. Larger boulders can be handled more easily, and fragmentation in ground preparation becomes less critical.
2. Larger tyres give better traction and have less rolling resistance.
3. Loading times are reduced, truck waiting times reduced and fleet sizes can be reduced, leading to less congestion, greater efficiency and improved safety.
4. Power drive efficiency is increased since auxiliaries are a smaller percentage of the total power consumed.

6. Mobile units are essentially one man machines, with resulting savings in wages, social benefits, taxes infrastructure costs, etc.

To obtain the maximum benefit from mobile equipment, as previously shown new excavation methods are essential. Most open pits are designed round loading shovels and other relatively static equipment. Bench heights are usually a compromise between shovel size and grade control requirements. Drilling equipment capability is based on bench height while workshops services are designed to handle the range of shovel and drill sizes. To take maximum advantage of mobile equipment, mine design should be influenced by:

- a) lower bench heights to suit wheel loaders (front-end loaders), primary design criteria being based on grade control requirements;
- b) better fragmentation to suit mobile equipment;
- c) maintenance facilities based on mobile equipment, e.g. "drive-in" shops rather than field teams; the heavier, more complicated maintenance being carried out under better conditions. Greater use of replacement sub-assemblies to minimise down-time;
- d) planning to eliminate electrical power systems to reduce capital cost; and
- e) improved operator training and a change of personnel attitude to regard mobile equipment as a primary mover rather than a utility tool.

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APPENDIX 7.A

VEHICLE MECHANICS

AIR RESISTANCE Ra *

$$Ra = \frac{C_D \cdot A \cdot d \cdot V^2}{2g} \dots\dots\dots (7.A1)$$

where

V = vehicle speed

A = projected area of front of vehicle

d = air density

C_D = drag coefficient = 0.40 for "saloon"
body to 0.8 for a truck type profile
0.7 for a tractor-scraper

*"Mechanics of Vehicles" Taborek, J.J. Machine Design
May 30 - Dec 26, 1957.

RIMPULL CURVES

The traction between rubber-tyred wheels and the surface varies according to the mass on the driving wheels and the type of surface over which the vehicle operates. If "wheel spin" occurs, it can be remedied by:

1. Adding more mass on the driving wheels
2. Improving the underfoot conditions

The basic limitation on traction is the machine mass e.g. if a vehicle has only 40% of its mass supported by the driving wheels the maximum force it can exert is the equivalent of that mass. The underfoot conditions can make it much less.

The second limitation on traction is the engine power combined with the operating gear, since a vehicle is designed to provide combinations of speeds and rimpulls. Since most mobile machines have some sophisticated form of transmission, e.g. torque converter, the rimpull-speed

combinations are infinite and are best shown in the form of Rimpull Curves (See Fig 7.A1).

The performance of a mobile machine can be determined from manufacturers' Rimpull-Speed-Gradeability Curves. A typical example is shown in Fig 7.A2 ⁷⁶ for a tractor-scraper.

The maximum speed attainable, gear range and available rimpull can be determined from the curves knowing the gross vehicle weight (including payload) and total effective grade.

RETARDER CURVES

The speed at which a vehicle can be maintained (without the use of the service brake) when the vehicle is descending a gradient with the retarder fully on, knowing the gross vehicle weight and total effective grade can be determined from manufacturers' Retarder Curves. A typical example is shown in Fig 7.A3 ⁷⁶ for a tractor-scraper.

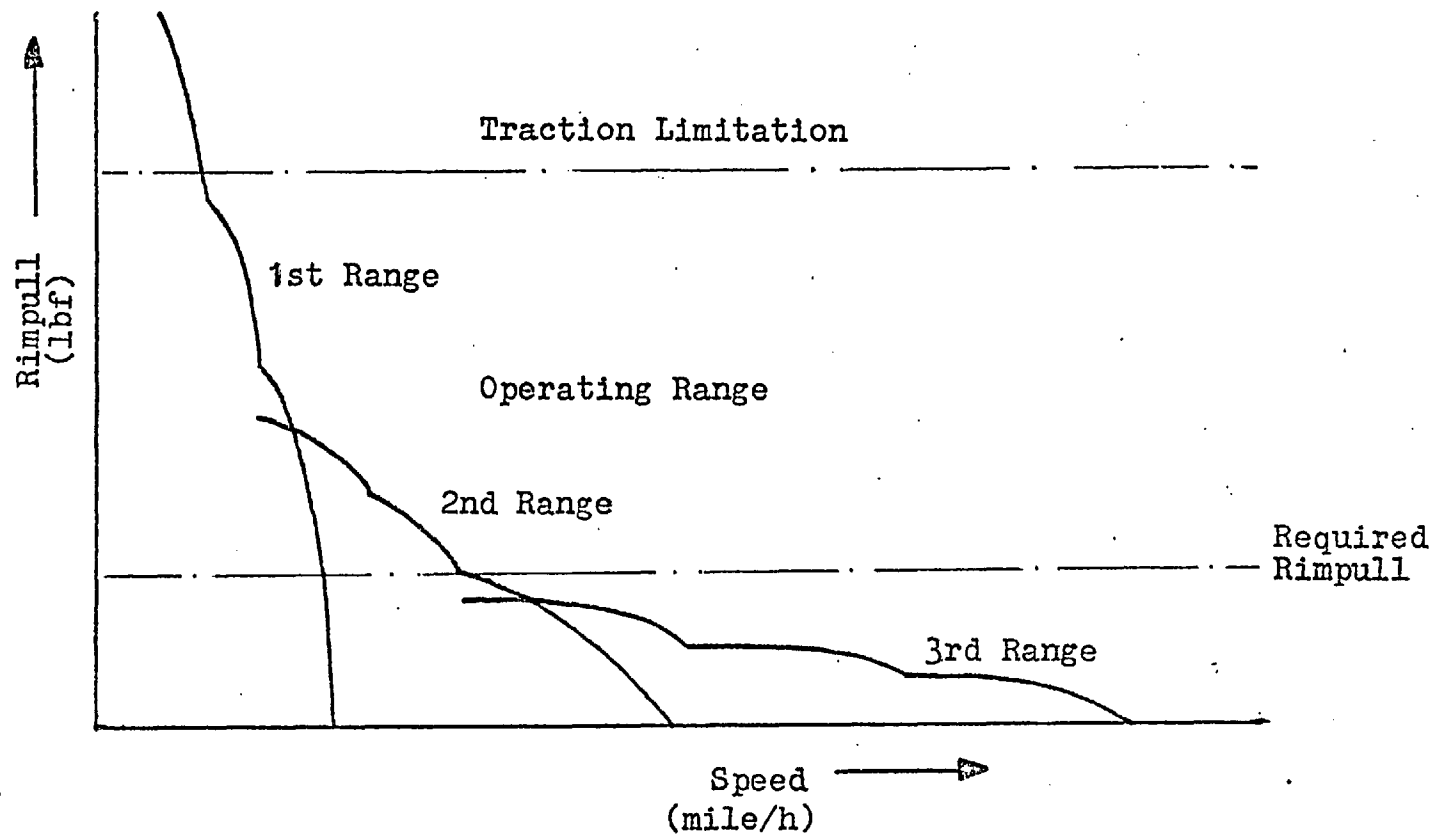


FIG 7A1 RIMPULL ~ SPEED CURVES

GRADEABILITY - SPEED - RIMPULL CURVES

Use of Curves

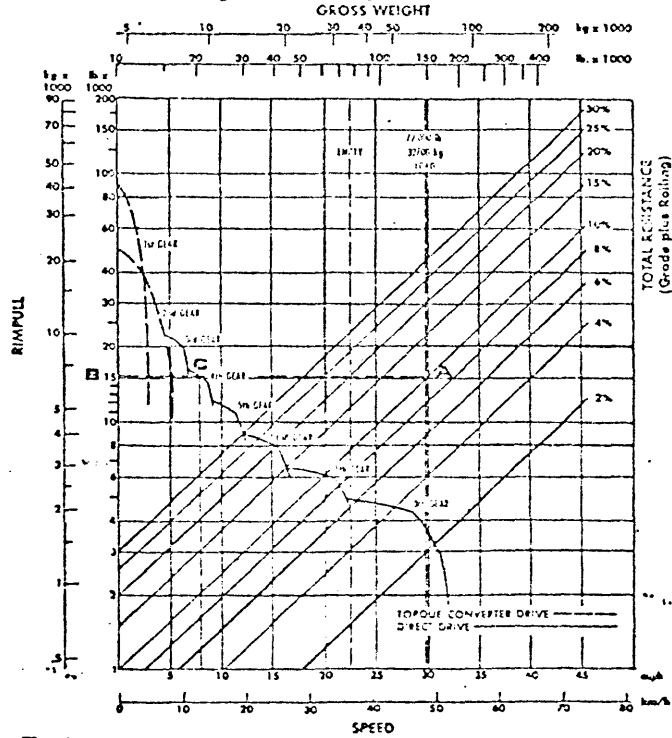
Example-

631C with an estimated payload of 72,000 lbs. (32700 kg) is operating on a total effective grade of 10%. Find the available rimpull and maximum attainable speed.

Empty weight & payload = Gross Weight

$$76,700 \text{ lbs.} + 72,000 \text{ lbs.} = 148,700 \text{ lbs.}$$

$$(34800 \text{ kg}) + (32700 \text{ kg}) = (67500 \text{ kg})$$



To determine gradeability performance:

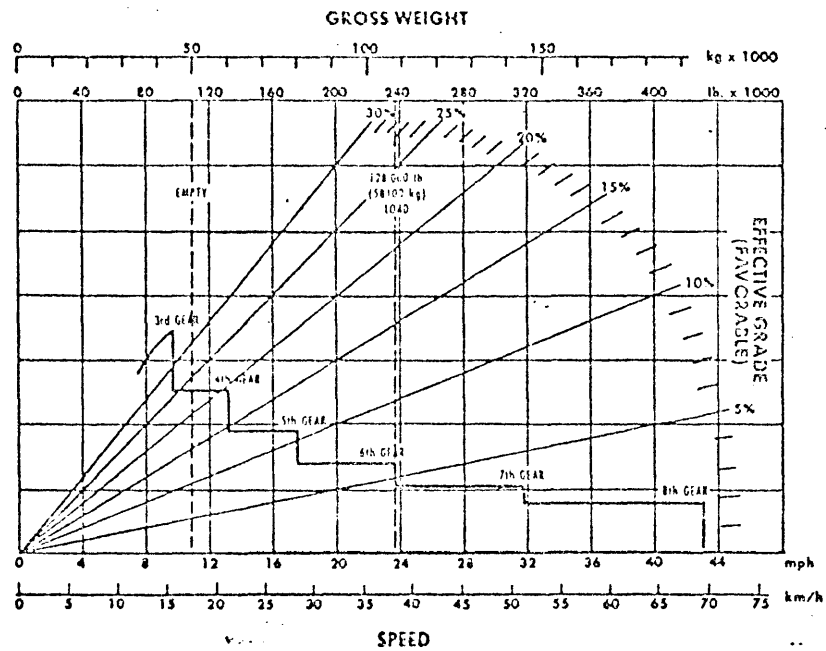
Read from 148,700 lbs. (67500 kg) on top of the gross weight scale down to the intersection of the 10% total effective grade line (point A).

Go across horizontally from A to the Rimpull Scale on the left (point B). This gives the available rimpull - 15,000 lb. (6800 kg).

Where the line A-B cuts the speed curve (point C), read down vertically to obtain the maximum speed attainable for the 10% effective grade: 8.5 mph (13.7 km/h).

Answer: The vehicle will climb the 10% effective grade at a maximum speed of 8.5 mph (13.7 km/h) in 4th gear. Available rimpull is 15,000 lbs. (6800 kg).

FIG 7A2 RIMPULL GRADIENT SPEED CURVES
(Caterpillar Tractor Company)



Retarder Curve

The speed that can be maintained (without use of service brake) when a vehicle travels downhill with retarder fully on can be determined from Retarder Curve knowing total effective grade and gross vehicle weight

Total Effective Grade is grade assistance minus

rolling resistance

15% favourable grade (grade assistance)

- 5% rolling resistance

10% Effective Grade

Gross Vehicle weight = 100700 kg (53500 kg empty)

Interest GVW 10% Effective Grade Line

Read Intersect Retarder Curve off speed =

21.1 km/h in 5th Gear.

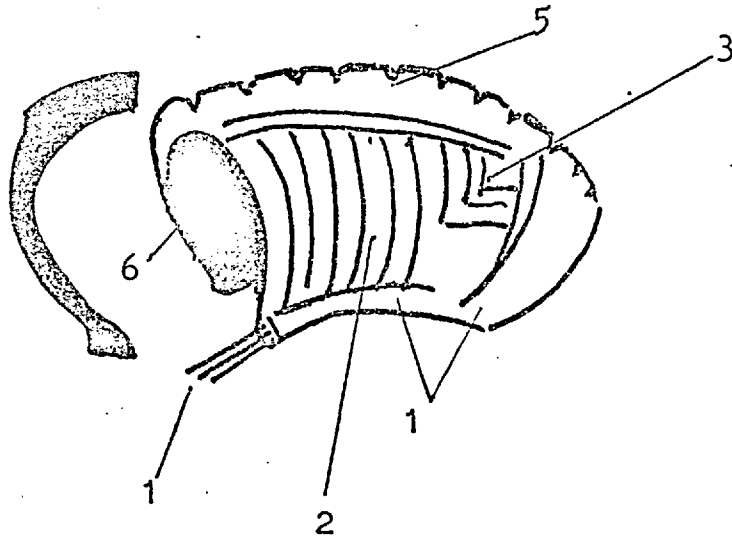
FIG 7A3 USE OF RETARDER CURVES

(Caterpillar Tractor Company)

APPENDIX 7.B

TYRE NOMENCLATURE

BIAS PLY TYPES (Fig 7.B1)

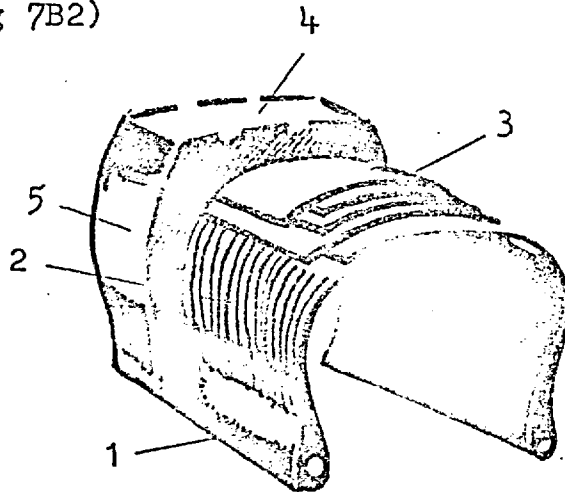


1. Beads - these steel wire bundles anchor the tyre to the rim. All the plies are tied into the beads to prevent any change of shape or fit on the rim. One or more bead stacks may be used.
2. Cord Body - layers of rubber cushioned cord (or fabric) to confine the inflation pressure. Each cord in each ply is completely covered in a resilient rubber compound, each ply being insulated from each other. The term "ply rating" as defined by The Tire and Rim Association is an index of the tyre strength and does not necessarily indicate the number of chord plies in the tyre.
3. Breaker or Tread Plies are normally confined to the crown area of the tyre and give additional impact resistance to the tyre, providing greater protection to the cord body.

4. The Sidewalls are protective rubber coverings on both outer sides of the chord body. They are designed to flex and "squirm" without cracking when subjected to normal continuous deflection or impact loads.
5. The Tread is the part of the tyre in contact with the road. It has to provide traction, long wear and be cut resistant.
6. The Tubeless Inner Liner covers the inside of the tyre from bead to bead, retaining the air in the tyre.

RADIAL PLY TYRES (Fig 7B2)

1. Tyre Bead
2. Radial Cords
3. Breaker Plies
4. Tread
5. Sidewalls



The carcass of the radial ply tyre is constructed of (2) steel chords running radially from bead to bead, arranged in single ply construction. Across the crown, steel breaker plies stabilise the tread area. The construction is referred to as "belted".

TYRE SIZE

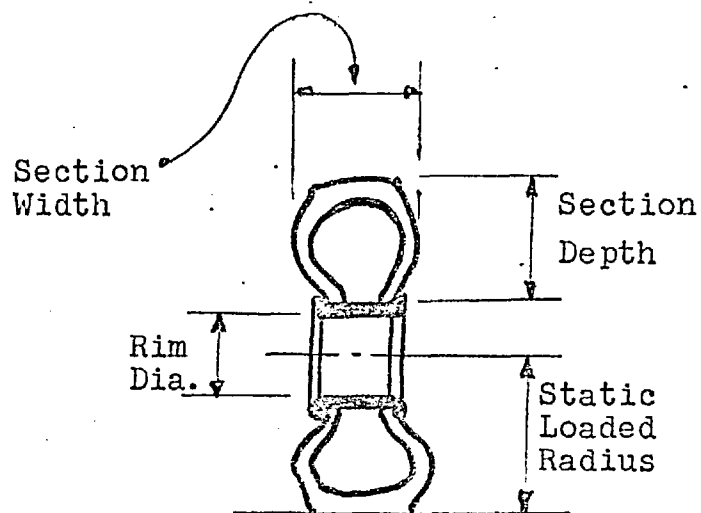
Tyre size is expressed as follows:

26.5 - 29

Section width - 26.5"

Rim diameter - 29"

TYRE GEOMETRY



$$\text{Aspect Ratio} = \frac{\text{Section Depth}}{\text{Section Width}}$$

For earthmover tyres

<u>Tyre Type</u>	<u>Aspect Ratio</u>
Standard	1.0
Wide Base	0.8
Low Profile	0.65

A wide base tyre with the same rim size as a standard tyre does not necessarily have a larger overall diameter, e.g. the 18.00 - 25 conventional tyre has a larger overall diameter than the 20.5 - 25 wide base tyre.

CODE IDENTIFICATION

E - <u>Earthmover</u>	L - <u>Loader and Dozer</u>
E - 1 Rib	L - 2 Traction
E - 2 Traction	L - 3 Rock
E - 3 Rock	L - 4 Rock Deep Tread
E - 4 Rock Deep Tread	L - 5 Rock Extra Deep Tread
E - 5 Rock Intermediate HR*	ML - <u>Mining and Logging</u>
E - 6 Rock Maximum HR	ML - 1 Rib
E - 7 Floatation	ML - 2 Traction
G - <u>Grader</u>	ML - 3 Rock
G - 1 Rib	ML - 4 Rock Deep Tread
G - 2 Traction	C - <u>Compactor</u>
G - 3 Rock	C - 1 Smooth
*HR - Heat Resistant	C - 2 Grooved

Michelin Tyres

Four designations - XRA, XRB, XKA, XKB

All Michelin X tyres have "X" construction, i.e. steel radial ply with steel breakers.

R or K designates the tread design

R - traditional Michelin tread pattern.

(lugs and bars) similar to other makes except for the bar spacing. They are described as "high traction" tread pattern, designed to give additional resistance to impacts and cutting.

K - this tread is designed to give maximum protection to the carcass, and is recommended for use in rock, slate, shales, gravels, etc. which normally cause excessive tread wear. The tread is wide flat and massive and the

tread blocks are extended to the shoulder to provide additional protection to the side walls. It is unsuitable where maximum penetration (and traction) is needed or where TMPH requirements are above its rating.

A - Added sidewall reinforcement and cut resistant compound

B - Standard Construction

D1 - Deep Tread (L4 tyres)

D2 - Extra Deep Tread (L5 tyres)

Ply Rating - Michelin Earthmover tyres are marked *, **, or *** to designate the strength of the casing, the asterisk replacing the Tire and Rim Association's ply rating.

TMPH (Ton -Miles/h)

The major cause of premature tyre failure is internal heat caused by rolling, flexing and "squirming" of the tyre in service. When sufficient heat is generated, the tyre can reach a sufficiently high temperature to reverse the vulcanisation process and "revert" the rubber. Ply separation and tyre failure results. Even at temperatures just below reversion level the strength of the rubber is greatly reduced and the tyre becomes highly susceptible to impact damage.

The Ton-Mile per hour formula was developed by the SAE (SAE Handbook. Society of Automotive Engineers, New York, 1967) to predict tyre temperature rise, by using the product (load x speed) to provide an index of temperature rise.

The maximum allowable tyre temperatures are as follows:

Fabric cord tyres - 107°C (225°F)

Steel cord tyres - 93°C (200°F)

Heat generation within a tyre depends on:

1. The load on the tyre
2. Speed
3. Ambient temperature

During a working cycle the tyre is subjected to the load of the empty and the loaded machine.

The Mean Tyre Load is the average of the empty and loaded machine masses, therefore:

$$\text{Mean Tyre Load (tons)} = \frac{\text{Tyre Load (empty)} + \text{Tyre Load (loaded)}}{2}$$

The Workday Average Speed is the total miles travelled in a working day divided by the total hours in the working day, including lunch breaks, shift changes, rest periods, etc., therefore:

$$\text{Workday Average Speed (mile/h)} = \frac{\text{Total distance travelled}}{\text{Total workday hours}}$$

For the Workday Average Speed to be valid, it is essential that no significant change from the average speed occurs from hour to hour.

To select a tyre, it is necessary to check that the tyre size and type at the ambient temperature has a rating equal to, or higher than, that calculated from:

$$\text{TMPH} = \text{Mean Tyre Load} \times \text{Workday Average Speed}$$

Manufacturers occasionally change the construction of their tyres and it is usually advisable to obtain their TMPH figures before finally selecting a manufacturer's product.

Load-Haul-Dump TMPH

When a wheel loader performs load-haul-dump duties, its tyres are subjected to similar operating conditions to tractor-scrappers. Again the TMPH Rating of the tyres should be checked against the duty. As a guide for loader tyres compared with E3 tyres (Table 7.BI):

L3 - 80% E3 Rating

L4 - 70% E3 Rating

L5 - 60% E3 Rating

Before final selection the duty should be checked against the manufacturer's rating.

Tyre overheating can be a major problem during delivery of new wheel loaders (and similar mobile machines) and during movement from one job to another, due to the tendency for them to be driven at high speeds. The TMPH ratings should not be exceeded. L5, Extra-Deep Tread tyres are an extreme example and require special care when transporting machines.

Tire and Rim Association Ratings

These provide a guide to the structural capacity of a tyre. They are in the form of Tyre Loading Tables for standard ply tyres at various cold inflation pressures. If heavy loads are anticipated, the tables indicate whether a higher ply rating is required.

The tables do not apply to steel cord radial tyres, as these tyres are of single ply construction, and use *, **, or *** designation. Radial steel cord tyres of the same size and "star rating" are used in different applications in different machines. Adjustment of the load carrying capacity is made by adjusting the inflation pressure.

APPENDIX 7.C

WHEEL LOADER CYCLE TIMES

Wheel loader (front-end loader) cycle times for truck loading, for bucket sizes up to 5.5 m^3 (7 yd^3) can be calculated from the following method, based on field measurements.

It is assumed that competent operator and centre-articulated steering are used. The basic tests were carried out on granular material and well graded, small size coal on a, solid, flat surface with further observations made elsewhere.

The cycle time is made up as follows:

- a) load
- b) manoeuvre (four reversals)
- c) dump
- d) minimum travel
- e) full cycle of hydraulic operation

Basic Cycle Time 0.5 - 0.65 min.

The lower figure refers to smaller loaders and the higher to $5 - 7 \text{ yd}^3$ machines.

The following times should be added to the basic cycle time for varying conditions

<u>Material</u>	<u>Min</u>
Slack (up to 25 mm sizes)	+ 0.03
20 mm to 150 mm sizes	+ 0.02
+ 150 mm coal	+ 0.03
+ 150 mm rock	+ 0.05
In bank material	+ 0.06*

*where within economic capability of loader.

OTHER

Company trucks	<u>Min</u> 0.00
Contractors' trucks	+ 0.06
Occasional loading	+ 0.08
Regular loading (bonus system)	- 0.02
Small trucks	+ 0.06
Fog, rain, dust, snow, bad light	+ 0.08
Stockpile, 3m high	+ 0.00
Stockpile, less than 3m	+ 0.02
Stockpile, irregular dumping	+ 0.04

8. HYDRAULICKING

Hydraulicking consists of directing a high pressure stream of water from a nozzle against a bank of loose or in-situ material. The material is first undercut by the water jet and then it caves. The broken material is transported away from the face by the water either in ground channels or in sluices to the cleaning or disposal area. In gold bearing deposits the sluices form the preparation system as well as the disposal system.

Hydraulicking has been used for a wide variety of materials, e.g. sand, clay, gold bearing gravel, tin, coal, etc. The suitability of a material for hydraulicking depends upon:

1. Disintegration under a water jet.
2. Adequate water supply, preferably under a high natural head .
3. Bedrock contours should allow water transport of the material. If concentration in sluices is desired, the material must do so under variable conditions of pulp density and rate of flow. Boulders always reduce the efficiency of the operation.
4. Location of suitable disposal areas; attention to swell and flow characteristics is essential.
5. Existence of stream pollution regulations.

Legal limitations may make the operation uneconomic.

OPERATIONS

Monitors or hydraulic giants with pressure heads between 30 and 200m (100-600 ft) are extensively used to mine unconsolidated materials.

The volume of water required is:

$$Q = c a \sqrt{h}$$

where

Q = rate of flow (ft³/min)

c = coefficient of discharge
(2.68 - 3.14 depending on
design and finish of monitor)

a = cross sectional area of nozzle (in²)

h = head of water (ft)

The productivity of a monitor depends on bank height, character of material and/or bedrock, and water pressure. Generally productivity is expressed in yd³/MID.

MID = "miner's inch day" which is represented by
a flow rate of 1½ ft³/min or 11½ U.S. gallons/min
of water flowing continuously for 24 hours.

Typical figures - 4½ - 6 yd³/MID in well-rounded gravel and under favourable conditions. Extremes of 1-10 yd³/MID have been reported.

In the U.S.A. monitors were developed specifically for hydraulicking. In the U.K. firefighting monitors have been adapted for mining the Cornish china clays and similar equipment is used in South-East Asian tin mining. Remote control of monitors is common especially where high banks can cave suddenly, but in S.E. Asia, low banks are normal and manual control is used.

The monitor (Fig 8.1) consists of a base for attachment to the end of the service pipe, having a ball race fitted to afford lateral movement. Above the ball ring is a 90° bend making a ball and socket joint with the barrel to give vertical movement. A cast iron nozzle is fitted to the end of the barrel to provide the water jet. Control is exercised

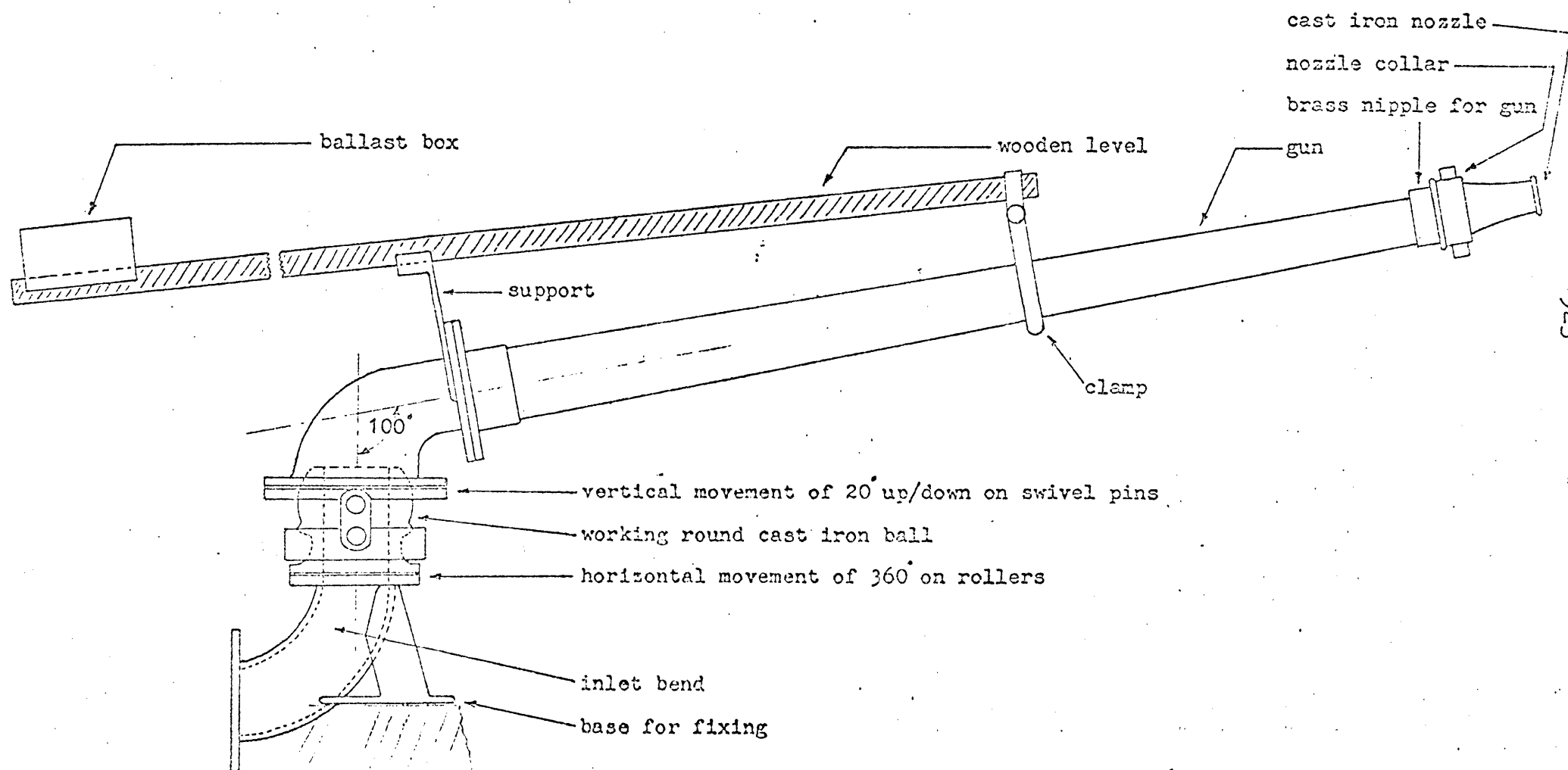


FIG 8.1 HYDRAULIC MONITOR FOR ALLUVIAL MINING OF TIN

by means of a level, fitted to the barrel and base piece, having a ballast box at the handling end to provide balance. Nozzles from 25 - 250 mm (1" - 10") diameter are used, depending on water supply, pressure and mining conditions.

In the Cornish China Clay workings ceramic lined nozzles are found to give best service ⁸¹.

Plots of jet impact against nozzle diameter almost exactly follow a square law suggesting that the maximum possible nozzle diameter is desirable. The desirability of a sharp nozzle discharge edge and a careful blending of the parallel section (if any) into the tapered section of the nozzle has been demonstrated and is of increased importance at elevated working pressures ⁸¹. The advantages of a high polish on the inside surface of nozzles has been similarly shown and an improvement in Relative Roughness (Centre-line Average Method) from 15 micro-inches to 1 micro-inch (an attainable figure) can yield a 10% increase in impact between 500 - 575 ft head ⁸¹. To date better finishes are possible with metal nozzles than with ceramic lined ones (Sintox, Roydalox, Hilox 961 and Purox have been tried); however they do not hold their finish as well as ceramic linings. Chromium plating of "Ajax" nozzles has been tried but was rejected after field tests because of the expense of producing a low relief finish (less than 10 or 15 micro-inches - C.L.A.) on a convergent - paralalled profile nozzles. Straight taper ceramic lined nozzles have been proved out of economic necessity and it is expected that for washing pressures in excess of 460 ft head, plating and polishing of straight tapered brass nozzles will be reconsidered.

The position of the monitor depends on mining conditions and changes as work progresses. Generally it will commence at the lowest point of the bedrock to obtain maximum gradient. In tin-mining the spoil is transported to a sump via ground sluices and pumped to the surface by gravel pump.

From time to time the top of the bank is inspected for cracks and signs of impending collapse. If a heavy fall is expected, the monitors are turned away from the vicinity so that a possible cave-in is not assisted in its travel by a flow of water. Working faces are normally kept square and side banks free from overhang. For this purpose cutting is directed from right to left, having a buttock of ground immediately ahead of the monitor so that slides are directed away from the monitor.

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9. ROPE HAULED SCRAPERS

Rope hauled scrapers fall into two main categories:

1. The Slackline Excavator (Fig 9.1)
2. The Drag Scraper (Fig 9.2)

They are extensively used for waste and refuse handling but also are particularly useful for below-grade, wet pit operations, as the only moving part in contact with the excavation is the scraper itself.

The slackline excavator has a digging capability similar to that of an dragline and its ability to handle hard ground improves with bucket size. The carrier rope prevents bucket "bounce" present in dragline operations, but the inability to change direction makes it difficult to handle boulders or bands of hard material sloping across the digging line. It cannot therefore be used for selective mining. If the excavated material caves or is free flowing, digging can be continued on one line over a long period, otherwise the tail anchor must be moved at shorter intervals. If no caving occurs and the bucket does not "wobble", it can become jammed in its own slot and operations become uneconomic.

The operation of the slackline is based on the suspension of a scraper bucket on a close hitched dragchain from a trolley running on a track rope, which is anchored as shown in Fig 9.1. The drag rope hauls the scraper bucket to the dump point where it is tripped. The bucket then returns by gravity. Drag speeds of 1 m/sec (200 ft/min) for digging and 3 m/sec (600 ft/min) for hauling are typical.

With the drag scraper no track rope is used and the bucket does not leave the ground in a normal cycle. The

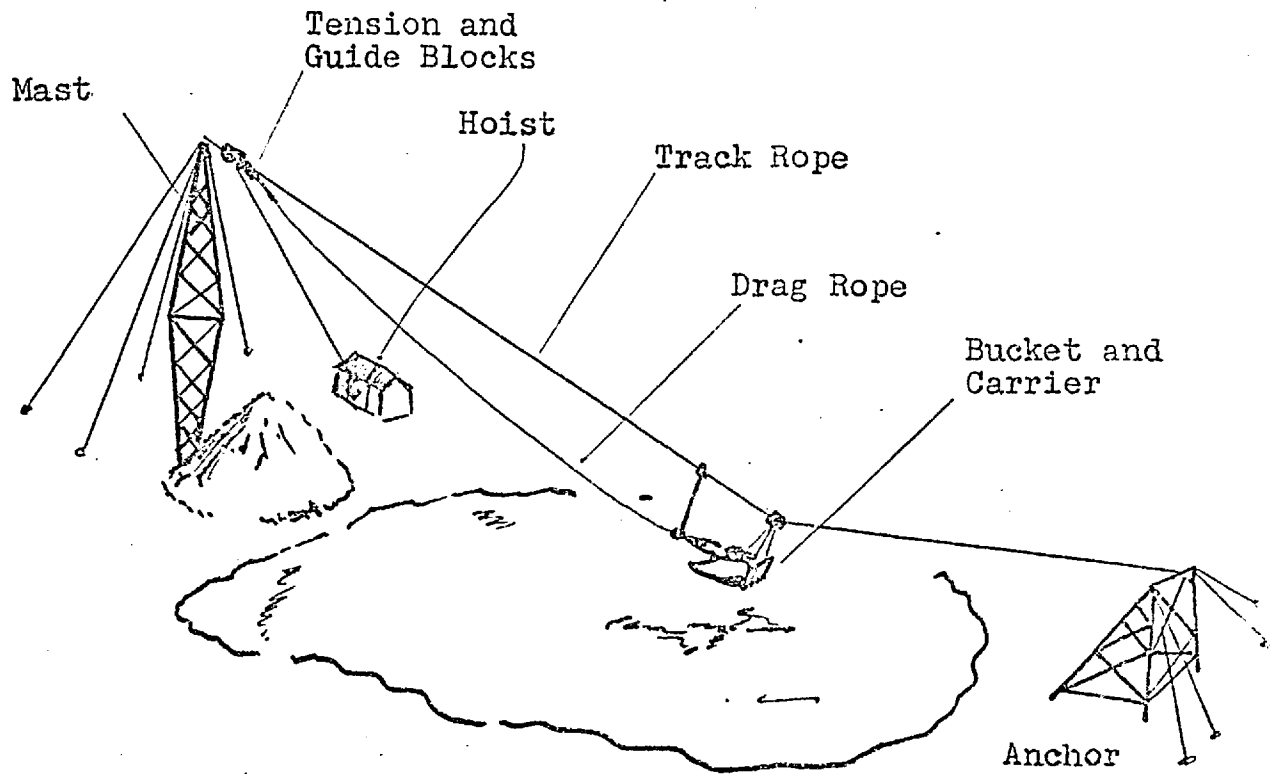


FIG 9.1 THE SLACKLINE EXCAVATOR

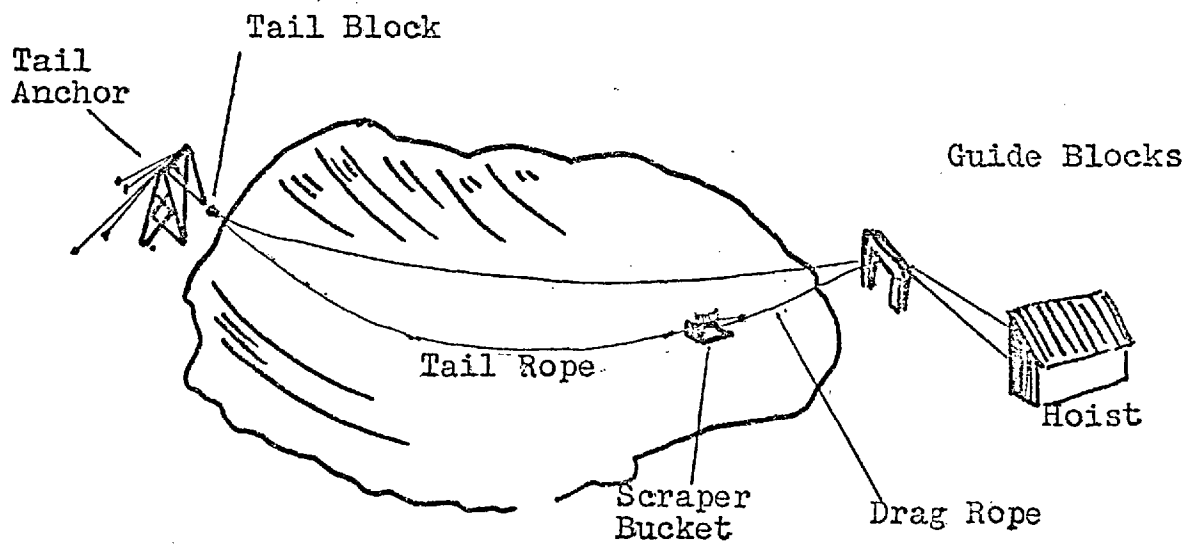


FIG 9.2 THE DRAG SCRAPER

bucket is dragged over a hopper usually covered by grizzly bars. The bucket is then hauled back by the tail rope. Again operation can be uneconomic if the material does not cave or flow freely. The drag scraper is unsuitable for selective mining.

Both slacklines and drag scrapers have their best applications in digging extensive areas of uniform material e.g. sand, gravel, clay, etc., such as pre-stripping at Marinduque, Philippines, and Guthega Riverbed, Snowy Mountains, Australia. In the right application they are very efficient, have low maintenance and operating costs, and low capital costs because of the small number and light weight of the moving parts.

The drag scraper costs less than a similar slackline excavator installation and is equally satisfactory for most operations, but for long haul distances, or digging deep under water, the higher haul speed of the slackline makes it more productive.

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10. CONCLUSIONS

The purpose of this thesis has been to present in a simple fashion a set of rules and data, based on a critical appraisal of present techniques, supplemented by the author's original contributions, to enable practising mining engineers to form a systems approach to the selection of open pit excavating and loading machinery. It should be realised however that one man's experience is a poor statistical sample and the reader is advised to adopt a cheerful independence in the acceptance of any data and should compile his own figures to supplement and refine any published figures, including those provided here.

This work seems to clearly indicate that more use should be made of industrial engineering techniques, which can greatly improve the validity of any data collected. The quality of this data is all important for the validity of any system study involving simulation or other sophisticated techniques which must necessarily follow excavator selection, is only as good as the data upon which it is based.

FURTHER STUDIES

The most obvious step which should follow is the preparation of a "Handbook on Electrical System Design for Open Pit Mines". This work can be proceeded with immediately as almost all the basic parameters are known.

There are considerable advantages to be derived from continuous excavation methods but it is difficult to envisage any major break-through in the field of hard rock mining in the near future. A much better knowledge of rock

properties and particularly rock machining, must be gained before any revolutionary developments can be forecast. At present no mechanical device can match the huge quantities of energy that can be so compactly stored in chemical form in an explosive charge and the drill-blast-load cycle is likely to remain the major excavation method for some time.

Much has been made of the future use of wheel loaders, and undoubtedly their application will widen greatly: but because of tyre costs, their use in dense, badly fragmented rocks will not expand at the rate that has been forecast by some of their supporters. A new approach to tyre application, perhaps based on compressible silicone liquid filling with special soft spring-rate, liquid filled suspensions which may reduce impact damage while still avoiding excessive side wall flexing and squirming appears relevant. X

The whole field of excavator design and selection is an exciting one that has long been neglected by mining engineers. The work of Bunting and Rodgers (State Electricity Commission of Victoria) 620, etc., illustrates the improvements that can be achieved by careful analysis. If this work focusses greater attention by mining engineers on the subject, the author will have achieved one of the objects of the thesis.