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Principles and practices controlling the use of earthmoving equipment

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PRINCIPLES AND PRACTICES CONTROLLING THE USE

OF EARTHMOVING EQUIPMENT

BY

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A

THESIS

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INTRODUCTION

The purpose of this study is to show how the efficiency in earthmoving projects can be increased by the application of engineering principles.

Efficiency can be increased by choosing equipment to fit the given set of conditions; and, having chosen the equipment, operate it at its maximum efficiency.

In order to choose the right equipment it is necessary to study the conditions under which the equipment must operate. It is then necessary to study the machine in order to ascertain its adaptability to the existing conditions.

The adaptability of the equipment is controlled by its design, size and power. The design is important because it determines the method in which the machine operates. Design, size and power limit the working range and the capacity of the machine.

To operate equipment at maximum efficiency entails a study of the variables which affect performance. This study reveals that some of the variables are functions of operating methods and management. To insure maximum efficiency, operation and management must take advantage of the favorable variables and eliminate, or minimize the effect of, the unfavorable variables.

REVIEW OF LITERATURE

Much literature has been, and is being, published about earthmoving projects. Most of this literature consists of descriptions of the different operations. It covers the type of work, type and size of equipment being used, general layout of the project, purpose of the project, production figures and similar data. The Excavating Engineer, The Explosives Engineer, Construction Methods, Western Construction News, and similar monthly periodicals feature these articles.

Most of the literature published by the United States Bureau of Mines on open-cut mining operations has likewise been descriptive. Bureau of Mines Bulletin 433 "Open-cut Metal Mining" by E. D. Gardner and McHenry Mosier and the Bureau of Mines Bulletin 298, "Methods, Costs, and Safety in Stripping and Mining Coal, Copper Ore, Iron Ore, Bauxite, and Pebble Phosphate" by E. R. Cash and M. W. van Bernewitz are good examples of the Bureau of Mines Bulletins.

Besides the Bulletins, the Bureau of Mines has published numerous Information Circulars and Reports of Investigations reporting the various open-cut operations or some particular phase of the operation. Examples of this type of literature are Bureau of Mines Information Circular 1013 "Powershovel and Dragline Placer Mining" by E. D. Gardner and P. T. Allsman, Report of Investigations 3416 "Truck vs Rail Haulage in Bituminous-Coal Strip Mines" by Albert L. Toenges and Frank A. Jones, Information Circular 6959 "Some Aspects of Strip Mining of Bituminous Coal in Central and South Central States" by Albert L. Toenges and Robert L. Anderson, and Information Circular 6248 "Methods and Costs of Stripping and Mining at the United Verde Open-pit Mine, Jerome, Arizona" by E. M. J. Alenius.

Few, if any, of the articles in the technical periodicals analyze the projects to indicate how engineering principles have been, or might be, applied to increase efficiency or production.

Most of the engineering data that has been published has been submitted by the field engineers and the sales engineers of the companies making heavy equipment. For the most part this information is based on sound engineering principles but it must be remembered that the men who submit it are interested in selling some type of equipment. This motive might lead to a misinterpretation of data.

Among the men who have contributed worthwhile information on the power-shovel and the dragline are A. E. Holcomb, Lee DuBois, and T. Davidson.

Foremost among the contributors on bulldozers, scrapers, Tournapulls, and like equipment is Kenneth F. Park. Park's Principles of Modern Excavation and Equipment is published by R. G. LeTourneau, Inc. who are the leading manufacturers of that type of equipment. Park has also contributed many worthwhile articles to The Co-Operator, a bi-monthly publication of the R. G. LeTourneau corporation.

J. R. Thoenen of the United States Bureau of Mines has made many studies of modern equipment and practices. A series of Information Circulars on sand and gravel excavation written by Mr. Thoenen give good analyses of equipment and methods. Mr. Thoenen also co-authored Reports of Investigations numbers 3461, 3467, and 3502. These Reports of Investigations are time study analyses of quarry shovel loading, quarry haulage, and quarry drilling. These time studies are valuable because they suggest methods by means of which delay factors may be evaluated.

The bibliography of this paper is a more complete listing of articles covering various phases of the subject under discussion than space will permit reviewing here.

SOIL CHARACTERISTICS THAT INFLUENCE EQUIPMENT PERFORMANCE

The terms and phrases "yards per hour," "bank yards," "loose yards," "yards content," "2-yard bucket," and others are frequently used in describing equipment capacity and performance. A study of excavating equipment must begin with a definition of these terms. Unless defined, the term "2-yard bucket" could give rise to ambiguity. The bucket might hold two yards only when heaped, or it might hold two yards when level full. The yardage might have been determined when the material was compacted in the bank, or it might have been determined when the material was loosened by the excavating machine or by blasting.

Bank measure is the measurement of the volume occupied by material when in its original state of compaction in place; i.e., "in the bank." Bank measure is a measure of the material at the time the material has greatest density.

When the material in the bank is disturbed by the processes of excavation, voids are created. The void spaces which are created cause the material to occupy a greater volume than it did when in place. The failure to appreciate this swelling of volume of loose material is a common cause of inaccuracies in earth-moving estimates. Material movement is usually based on bank or "solid" cubic yards. When loaded into the hauling units, this material is in a loose state. It is, therefore, necessary to determine the percentage of swell to determine the pay load in the hauling unit in bank cubic yards.

The amount of void space created in the material depends upon

The amount of void space created in the material depends upon the type of material, the degree of disturbance, the moisture content, the shape of the fragments or particles, the size ranges of the particles, and the amount of material in each size range. The number and nature of these variables make it impossible to determine an accurate constant to represent the percent of swell. Rather the amount of swell falls within quite definite ranges for each type of material. From these ranges it is possible to determine an average figure for the different types of material.

Table I gives the observed limits of the percentages of voids and swell for some materials. The voids are the percentage of loose measurement. Swell is given as a percentage of bank volume. The mathematical relationships are

$$V\ (\%) = \frac{LV - BV}{LV} \times 100, \qquad (1)$$

$$S\ (\%) = \frac{LV - BV}{BV} \times 100, \qquad (2)$$

$$S\ (\%) = \frac{W_1 - W_2}{W_2} \times 100, \qquad (3)$$

where V = voids,

LV = loose volume of a unit weight of material,

BV = bank volume of a unit weight of material,

S = swell in percent,

W_1 = weight of a unit volume in place, and

W_2 = weight of a unit volume loose.

Table I gives 26 percent and 41 percent as the limits for the percentages of voids in dry lumpy clay with rock. The average value

TABLE I[1]

SWELL AND VOIDS IN REPRESENTATIVE MATERIALS

Material	Percent Swell	Percent Voids
Clean sand or gravel. . . .	5 to 15%	4.75 to 13 %
Top soil.	10 to 25%	9 to 20 %
Sandy, clayey loam.	10 to 35%	9 to 26 %
Good common earth	20 to 45%	17.7 to 31 %
Clay with sand or gravel. .	25 to 55%	20 to 35.5 %
Clay—friable and light . .	30 to 60%	23 to 37.5 %
Clay—dry, lumpy and tough, with rock	35 to 70%	26 to 41 %
Shale and soft rock	40 to 85%	28.5 to 46 %
Hard rock—well to poorly blasted	50 to 100%	33.3 to 50 %

(1) Park, K.F., Principles of modern excavation and equipment, p. 30, Peoria, Ill., R. G. LeTourneau, Inc., 1942.

is 33.5 percent. To use this average value in calculations could result in an error of 7.5 percent either way. Not to have considered the voids could have resulted in a minimum error of 26 percent and a maximum error of 41 percent.

Table II gives the average percentages of swell and the swell factors for some common materials. The use of the swell factor simplifies calculations. The swell factor is derived from the per-

TABLE II [2]

SWELL FACTORS

Material	Weight in Bank Per Cu. Yd. in Lbs.	% of Swell	Swell Factor	Loose Weight Per Cu. Yd. (Lbs)
Dry sand or gravel	3250	12	.89	2900
Wet sand or gravel	3600	14	.88	3168
Earth with sand or gravel	3100	18	.85	2640
Loam	2700	20	.83	2240
Earth	2800	25	.80	2240
Clay — light.	2800	30	.77	2160
Clay — dense, tough	3000	40	.71	2130
Shale or soft rock — blasted	3000	45	.69	2070
Hard rock — well blasted. .	4000	50	.67	2680
Rubbery clay, hard pan or poorly blasted rock. .		80	.56	
Coal — anthracite	2200	35	.74	1630
Coal — bituminous	1900	35	.74	1400

(2) Euclid Road Machinery Co., Estimating production and costs of material movement with Euclids, Form No. 350-R, p. 3, Cleveland, Ohio, 1946.

cent of swell.

$$SF = \frac{100}{100 + S}. \qquad (4)$$

where SF = swell factor

and S = swell (percent).

To find the bank measure, multiply the loose measure by the swell factor. Multiply the bank cubic yard weight of a material by the swell factor to find the loose cubic yard weight. A simple illustration will demonstrate the usefulness of the swell factor.

Example 1

Five hundred cubic yards (bank measure) of a light clay must be excavated to construct a large swimming pool. A truck is to be used to haul the clay. The capacity of the truck is 5 yards (heaped loose measure). How many loads must the truck carry?

The swell factor of light clay is .77. The 500 yards bank measure divided by .77 gives 650 yards loose. 650 yards loose measure divided by 5 yards per load equals 130 loads. Not to have considered the swell factor would have resulted in estimating 100 loads.

The bank cubic yards pay load permissible on a given hauling unit can be determined by dividing the rated pay load capacity (in pounds) by the bank cubic yard weight of the material.

The loose cubic yards pay load permissible on a given hauling unit can be found by dividing the rated pay load capacity (in pounds) by the loose weight per cubic yard of the material. The results of this calculation can be compared with the heaped pay load capacity of the hauling unit to determine the correct body size for the material.

Because material is always loose when in a bucket, dipper, wagon, or truck, it follows that the rated capacity of each is always in loose measure.

The capacity of dippers and buckets is always given as the struck cubical content. The capacity of railroad cars is also the struck cubical content.

The capacity of trucks is given in three ways. The commonest designation is the loose measure capacity heaped with a 3:1 slope. Usually the struck content of the truck is also given. The third method of designating capacity is in pounds of pay load permissible. Scrapers and carryalls are rated in the same manner in which trucks are.

The figures given in Table II are representative. To determine a more accurate figure for a given type of rock or earth, equation (3) is the most convenient to use. The specific gravity of a small solid sample of rock can be easily determined. From the specific gravity, W_1 can easily be found. A truck can then be weighed when empty and when loaded to its struck capacity to find W_2.

<u>Example 2</u>

The specific gravity of the rock is 2.3. The weight of the truck empty is 16,000 lbs. The weight of the truck loaded with 5 yards struck loose measure is 29,250 lbs. What is the swell factor of the rock?

Five yards of solid rock would weigh

$$(2.3)\ (62.5)\ (27)\ (5) = 19,450 \text{ lbs.}$$

Five yards of rock loose weigh

$$(29,250 \text{ lbs.} - 16,000 \text{ lbs.} = 13,250 \text{ lbs.}$$

The percent of swell is

$$\frac{19,450 - 13,250}{13,250} \times 100 = 46.8.$$

The swell factor is

$$\frac{100}{100 + 46.8} = .68.$$

The swell of broken rock is a limiting factor on the performance of shovels and draglines when these machines are casting as in stripping overburden. In some types of mining the depth and width of the cut that the excavating machine can make are determined by the dumping height of the machine, by the amount of swell in the material which is being excavated, and by the angle of repose assumed by the loosened material.

The angle of repose of a material is a function of the type of material, the moisture content, the shape of the fragments or particles, the size ranges of the particles, the amount of material in each size range, the superimposed load (static load of height and impact load of material being added), and the velocity of added material. Appendix A gives the angle of repose for some common materials.

Figures 1 and 2 illustrate how the volume of a cone-shaped pile is determined by the dumping height and the angle or repose of the material. In terms of the angle of repose "R", and the height "h", the volume becomes

$$V = \frac{(1.0472)\ (h^3)}{\tan^2 R} \qquad (5)$$

The volume in terms of the bank measurement is the swell factor times equation (5).

The volume of the pile in terms of the angle of repose and the radius is

$$V = (1.0472)\ (r^3)\ (\tan R) \qquad (6)$$

In casting, as in stripping, the pile may be considered to be a wedge-shaped windrow. The cross-sectional area of this windrow becomes

$$A = \frac{h^2}{\tan R} \qquad (7)$$

The cross-sectional area of the cut (the product of the depth and width of the cut) cannot exceed the swell factor of the material times the cross-sectional area of the windrow. If the cross-sectional area of the cut exceeded the area of the windrow, there would be no place in which to dispose some of the spoil. When the cross-sectional area of the cut is small, the excavating machine spends more of its time moving. Frequent moving decreases the capacity of the machine. Figures A3 and A4 illustrate this.

In Figure 30 another way in which the angle of repose can limit performance of a dragline is illustrated. If the angle of repose of the material on which the dragline is standing is too small, the

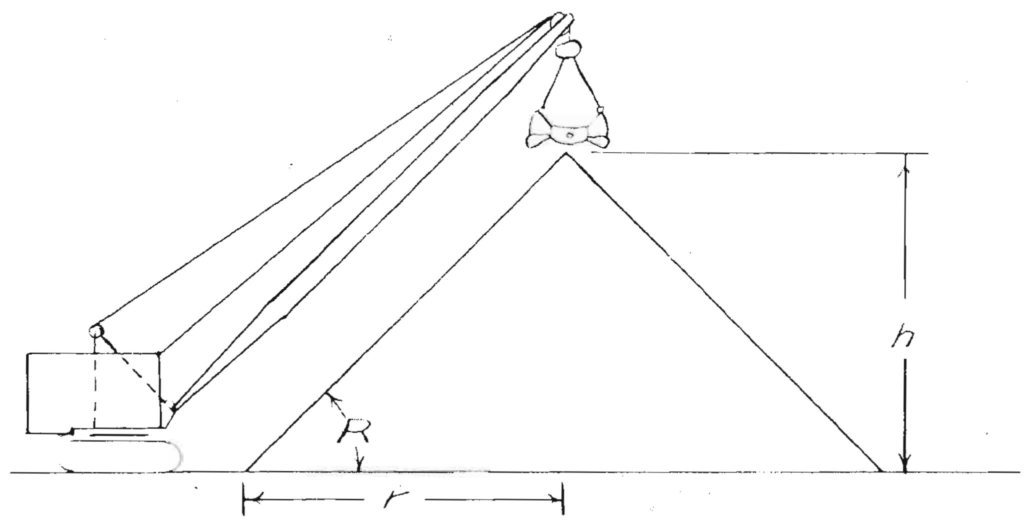


Figure 1. Effect of angle of repose and swell on a spoil pile.

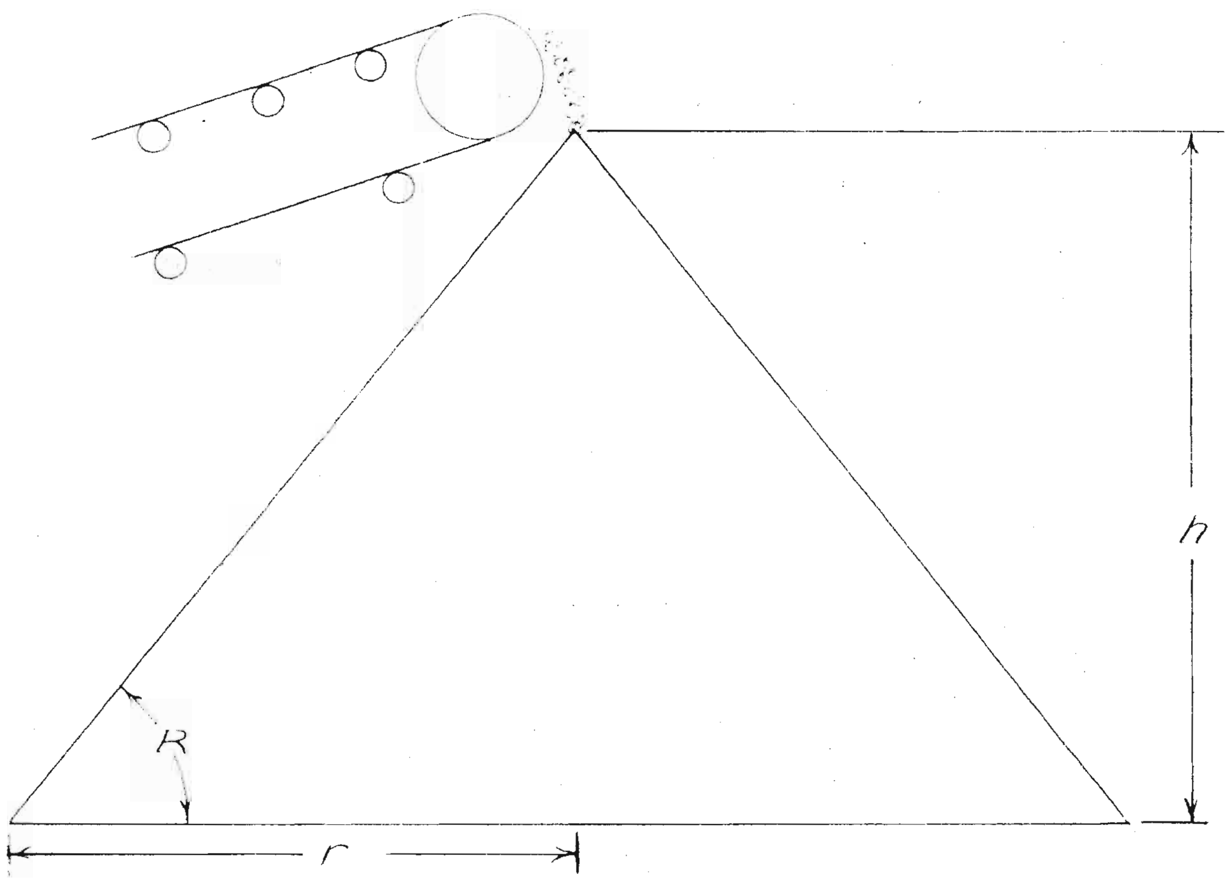


Figure 2. Effect of angle of repose and swell on the volume of a surge pile.

depth to which the dragline can dig will be limited; because as the depth increases, the face of the bank advances toward the machine, and it could undermine the machine.

The angle of repose limits scraper performance. If the angle of repose is small, it is difficult for the scraper to pick up a full load. This is especially true when the scraper is loading downhill.

By limiting the height of surcharge, the angle of repose is also a limiting factor in bucket, dipper, and truck performance.

The bearing capacity of soils is the third characteristic that influences equipment performance. The bearing capacity also influences the type of equipment that can be used at times. Table III gives some allowable soil pressures.

The large stripping shovels exert pressures of from 3 tons to 4 tons per square foot of bearing area. The large walking-type draglines exert less than one short tons per square foot of bearing area. It is therefore necessary to use walking draglines on soft soils because power shovels would sink into the soil. With the smaller shovels it is often necessary to use mats or floats (see Figure 6) to increase the bearing area of the shovel.

Figure 57 illustrates the relationship between bearing capacity of soils and tire size and pressure. The effects of tire penetration will be discussed later. The increase of rolling resistance due to tire penetration will also be discussed in detail.

The permeability and porosity of soils and rock are important because they determine or control the water content. The moisture

TABLE III [3]

ALLOWABLE SOIL PRESSURE IN SHORT TONS PER SQUARE FOOT

Kind of Material	Minimum	Maximum
Quicksand; alluvial soil	0.5	1
Soft clay	1	2
Wet clay; soft wet sand.	1	2
Moderately dry sand; fine sand, clean and dry	2	3
Compact coarse sand; stiff gravel. . .	3	6
Clay and sand in alternate layers. . .	2	3
Firm and dry loam or clay; hard dry clay or fine sand	2	5
Coarse gravel; stratified stone and clay; rock inferior to best brick masonry	5	8
Gravel and sand well cemented.	6	10
Good hardpan or hard shale	6	10
Good hardpan or hard shale unexposed to air, frost or water.	10	15
Very hard native bedrock	15	25
Very hard native bedrock, in thick layers, under caisson	30	

(3) Merriman, Thaddeus and Wiggin, T. H., American civil engineers' handbook, p. 711, New York, Wiley, 1946.

content also affects the cohesive and adhesive properties of soils. A wet clay can be troublesome by hanging or sticking to the shovel dipper, the dragline bucket, or the truck body. The same clay when dry might be as easy to handle as hard rock.

Table XII and a later discussion will show the affect of moisture on the coefficient of friction of the drive wheels and the surface.

Other properties of soils are also worthy of some consideration. Unit weights determine work and equipment size and power. The hardness and sharpness of fragments and particles influence the wear of teeth, blades, scraper bowls, crawler pads, tires, and truck beds. The toughness and brittleness of rock influence blasting results. Tough rock usually gives large sizes of fragments upon blasting and these large fragments are more difficult for excavating equipment to handle.

Other effects of the already mentioned characteristics of soils and rocks will be brought out in the ensuing chapters.

THE POWER SHOVEL

The power shovel is the most used of the different types of excavating machines; it is also the most misused. The misuse arises from the failure to realize that, though the power shovel is ruggedly built and powerful, it is not indestructable. The misuse and the abuse that the machine can stand is definitely limited by the overall design of the machine, the design and metallurgical properties of its members, and by the power available. Each machine is designed within a set of maximum allowable stresses. To constantly operate the machine at these maximum stresses results in exceeding these maximum stresses because of impact loads. This results in quick failure of the most overstressed members, fatigue or progressive fracture in other members, and rapid wear of shafts, gears, bearings, drives, clutches, and brakes.

Most of the power shovels now being used are powered by internal-combustion engines, Diesel engines, or by electricity. The bucket capacities (struck) vary from 3/8 of a cubic yard through 40 cubic yards. The small machines are driven by internal-combustion engines or by Diesels. Most of the small machines are crawler-mounted. A few are mounted on truck chassis for greater mobility and speed of travel. Those machines in the middle range are usually Diesel or Electric. All of the large machines are electric. Almost all of the machines in the last two classes are crawler-mounted.

The maximum range of operation of a shovel is controlled by the size and the design. Figure 3 gives the maximum working range dimen-

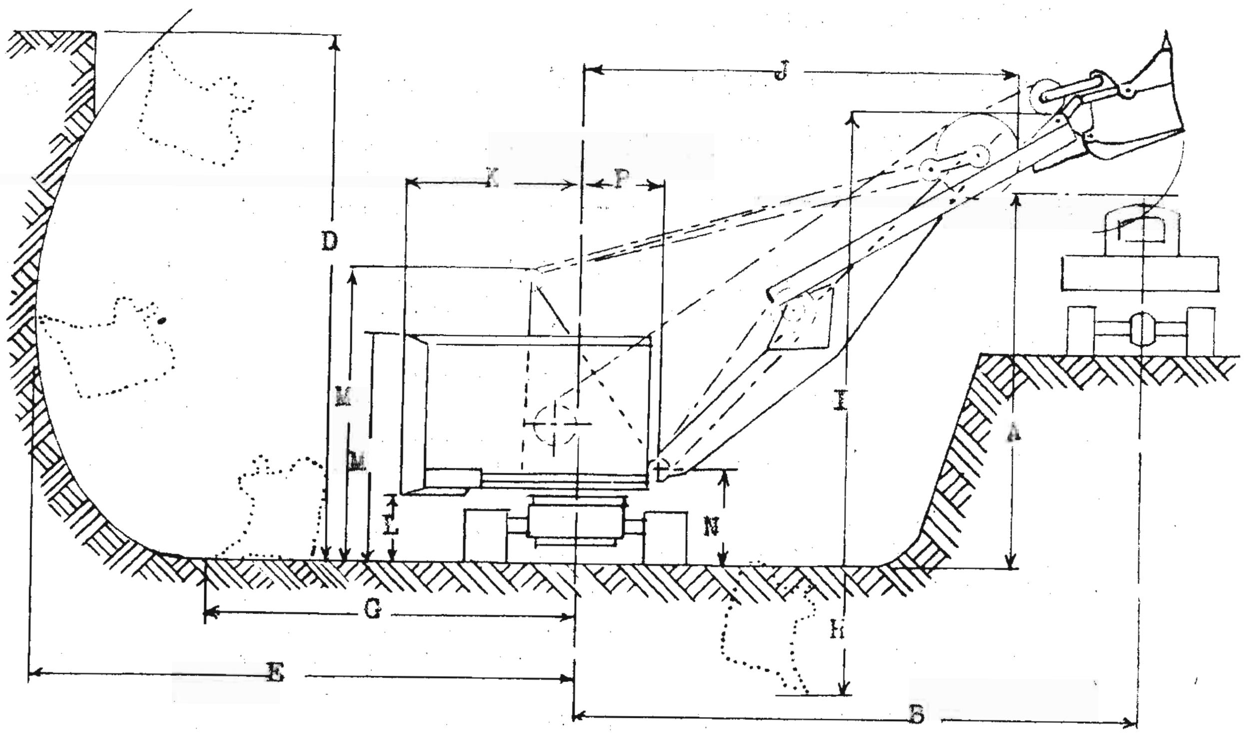


Figure 3. Shovel Working Range Dimensions

Capacity of dipper..................	Dipper—2½ cu. yd.
Length of boom—center foot pin to center of sheave...........	25'-0"—(Effective length is 27'-9")
Length of handle....................	18'-0"

Angle of boom, degrees.............	40	45	50	55	60
A—Dumping height—Max..............	17'-6"	19'-3"	21'-3"	23'-0"	[illegible]
B—Dumping radius—at max. dumping height....................	32'-6"	31'-0"	29'-6"	27'-6"	25'-6"
B—Dumping radius—Max.............	34'-3"	33'-6"	32'-9"	31'-9"	31'-0"
D—Cutting height—Max..............	27'-6"	28'-9"	31'-6"	33'-9"	34'-6"
E—Cutting radius—Max..............	38'-3"	37'-6"	36'-9"	35'-9"	35'-0"
G—Radius of level floor—Max.....	23'-9"	23'-3"	22'-9"	22'-3"	21'-9"
H—Digging depth below ground level—Max.......................	9'-9"	9'-0"	8'-3"	7'-9"	7'-3"
I—Clearance height of boom point.	25'-0"	26'-6"	28'-0"	29'-6"	30'-9"
J—Clearance radius of boom point.	27'-0"	25'-6"	24'-0"	22'-0"	20'-3"

K —Clearance radius of revolving frame, standard cwt..	12'-9"
L —Clearance under frame to ground level.............	3'-9"
M —Clearance height—boom and A frame lowered.........	12'-3"
M —Clearance height—std. A frame.....................	17'-6"
N —Height of boom foot pin above ground level.........	6'-3"
P —Distance boom foot pin to center of rotation.......	5'-3"

ensions of a Bucyrus-Erris 54-B $2\frac{1}{2}$-cubic yard shovel. The dimensions given here are the maximum limits as determined by the structural limitations of the machine. A further limitation on this shovel is the maximum bail pull which it can develop. For this machine the maximum pull with a 2-part rope and a 25-inch hoist drum is 74,200 lbs. If the weight of the dipper, the dipper sticks and the running block is approximated at 15,200 lbs., there would be 59,200 lbs. pull available for digging under the most favorable conditions; i.e., when bailing vertically. This would limit digging to banks where 59,200 lbs. of pull would be enough to force the dipper teeth through the bank to fill the dipper.

Figure 4 illustrates a more efficient range of the same shovel. Generally, the efficient range for a shovel is to dig from the level surface on which the shovel is traveling upward into a high well-prepared bank, swing less than 90 degrees and dump into a haul unit operating on the same level as the shovel. The height of the saddle blocks is an efficient limit to the dumping height.

Grades up to 30 percent can be negotiated by shovels if the ground surface is of such a nature as to permit the available tractive force to be applied. However, the power shovel operates best on a level surface. It is essential in large machines that the upper and lower turntables be in a horizontal plane when the machine is swinging. To achieve this on uneven ground, these machines are equipped with hydraulic equalizing and leveling jacks between the crawlers and the lower frame at each corner.

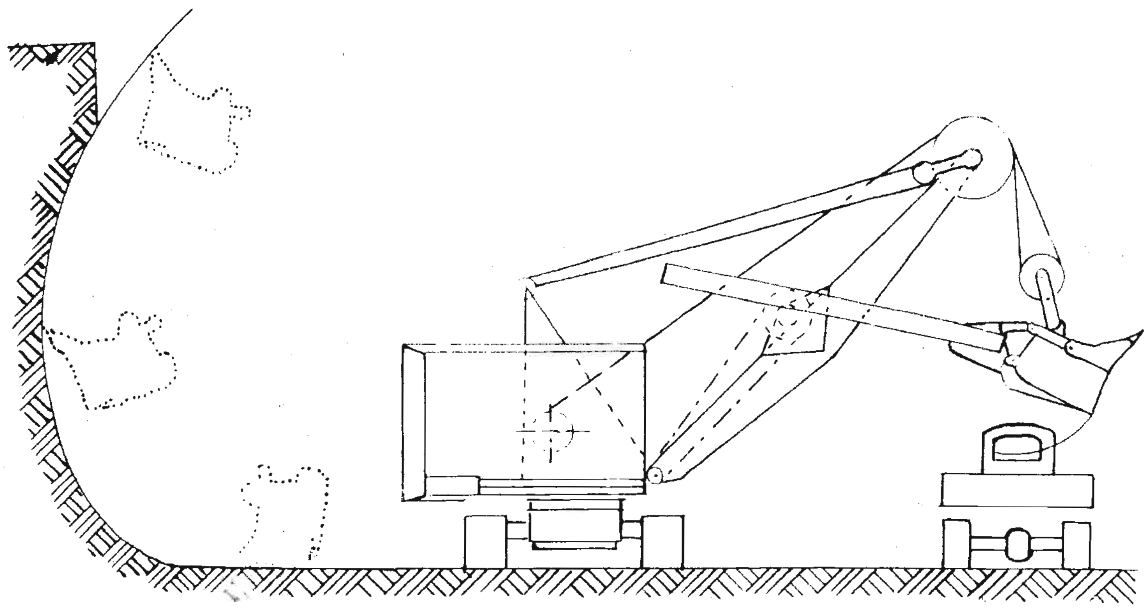


Figure 4. An efficient dumping height.

The smaller shovels can operate when the turntable is not horizontal but not efficiently. The shovel is not balanced about the center pin at all times in the vertical plane through the center pin, the boom, and the dipper stick. When the machine is working on a level surface, these unbalanced loads do not cause rotation. When the machine is not on a level surface, the previously mentioned plane is inclined. Unbalanced loads in this plane cause a rotating moment. These unbalanced loads consume more power in swinging and necessitate greater use and wear of brakes, clutches, rollers, center pin and the other swing mechanisms involved.

Figure 5 illustrates another hazard that may arise when digging a grade. In the upper part of the figure, distance "a" represents the moment arm of the loaded dipper about the tipping fulcrum. If the machine is now swung 180 degrees (as it would be if loading trucks to the rear) the moment arm is increased to distance "b". This increased moment could overturn the machine. Further, if the shovel was digging upgrade as in the upper part of the figure and applying digging effort into the bank, the crawlers would skid and the machine would slip downgrade.

The surface upon which the shovel is operating should, if possible, be smooth enough to give more than three points of support for the crawlers. If there are only three points of support, digging, dumping, crowding, or shifting the center of gravity in some other manner, will cause the shovel to rock on two points. Rocking the shovel causes undue wear on the crawler pads and links, pins, rollers, sprockets, and other parts of the track. This rocking also

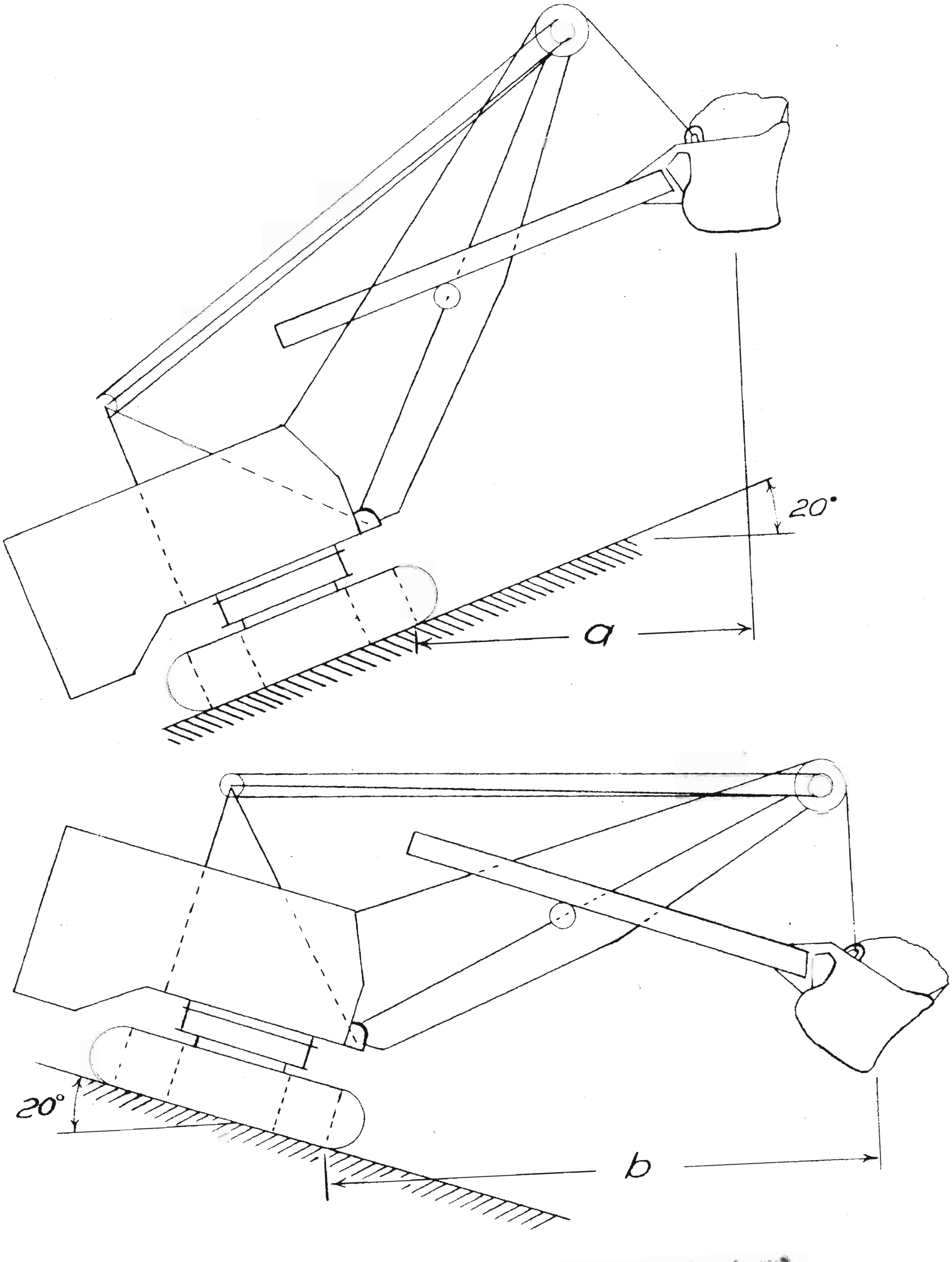


Figure 5. The arm of the tipping moment on the shovel.

sets up impact loads. The impact loads when frequently repeated will flatten or roll out the pins and bearing surfaces in the crawlers.

When the weight of the machine is supported by too few points of support, the stresses set up in these points may exceed safe limits and cause fracture of the parts involved.

In addition to the above, the rocking of the machine is a cause of increased operator fatigue.

It will, of course, be impracticable to have a perfectly smooth floor at all times to provide an even bearing surface for all the crawler pads. It is practicable to have the shovel level the surface as it digs forward in soft materials. If the floor is of sharp blasted rock, the operator can sort out some finer rock and dump it in the path of the crawlers to make a smoother bearing surface for the crawlers. The operator or the oiler can build up the low points with a hand shovel to provide additional points of support. In any case, the least that should be done is to remove all large, loose stones so the shovel will not travel over them, or, worse yet, rock on them as it is excavating.

Figure 15 illustrates another way in which the shovel may be caused to rock. If the bed in which the dipper teeth are buried is sufficiently resistant, the shovel will exert its maximum bail pull when digging through. In exerting its maximum bail pull, the moment of the digging effort will cause the back of the shovel to rise and the shovel will then have the front pads of each crawler as a tipping fulcrum. When the dipper breaks through, or the engine stalls, the tipping moment will decrease rapidly and the shovel would settle back with an impact.

Rocking of the shovel is a source of accidents. Large rocks that are balanced on the outer edge of the dipper can be jarred off when the shovel rocks as it swings. These rocks may fall on the hood or cab of the truck or on personnel that are too near. Operators and oilers can be hurt when thrown about in the machine by this violent rocking.

The surface from which the shovel is working must be dry enough and firm enough to support the shovel and to allow it to travel. The bearing pressures of shovels vary from near 20 lbs. per sq. in. for small machines up through 51.7 lbs. per sq. in. for such machines as the Marion Type 5561 40 cubic yard coal stripping shovel.

Bearing area can be increased by using floats or mats. The use of mats increases the shovel moving time. Mats are illustrated in Figure 6. The shovel rests on at least two mats at all times. Before moving forward, the shovel is swung 180 degrees to the rear. The wire rope or chain bridle of the mat is hooked over a dipper tooth. The dipper is then swung back dragging the mat across in front of the crawlers. The shovel then moves forward onto the newly placed mat and off one of the mats upon which it had been standing. The cycle is repeated as the shovel continues to move forward over the soft ground.

Bearing capacity of the soil can be increased by proper drainage. It can also be increased somewhat by placing some dry material from high on the working face in front of the shovel to form a dry roadbed.

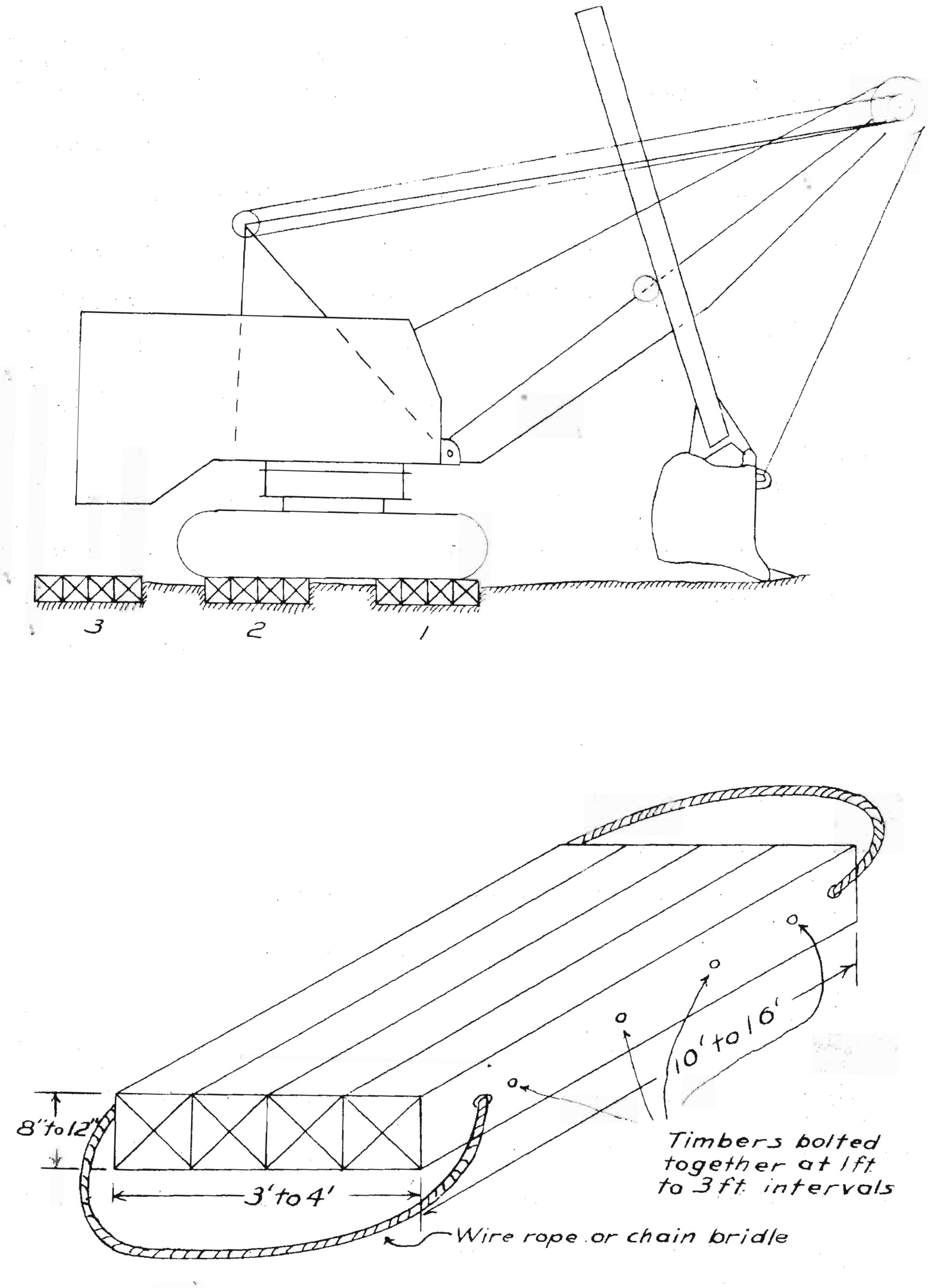


Figure 6. Mats for operation on soft ground.

Figure 3 shows the depth to which a shovel can dig below its level. This depth is the maximum and not the efficient limit. To dig below the surface on which it is traveling, the shovel must straddle the ditch and use mats for safety. The shovel cycle is slowed down because it takes more time to place the bucket at this depth. The shovel cannot swing until the bucket is hoisted clear of the ditch. Fewer buckets are removed per shovel move, and moves are more frequent than under other conditions of use. The power shovel is an inefficient machine for this type of ditching.

To dig larger ditches, such as road cuts or drainage cnals, the shovel must work in the ditch. In cases such as these it may be necessary to overexcavate to allow the shovel to swing to load haul units to the rear. The angle of repose of the material might also cause overexcavation but this would be true for any other machine, also.

If the shovel is loading into haul units on the bank above, the depth to which the cut can be taken is limited by the dumping height of the shovel as illustrated in Figure 3. If the shovel is casting to the bank above, the depth to which the cut can be taken is limited by the dumping height of the machine, by the swell and the angle of repose of the spoiled material, and by the angle of repose of the bank.

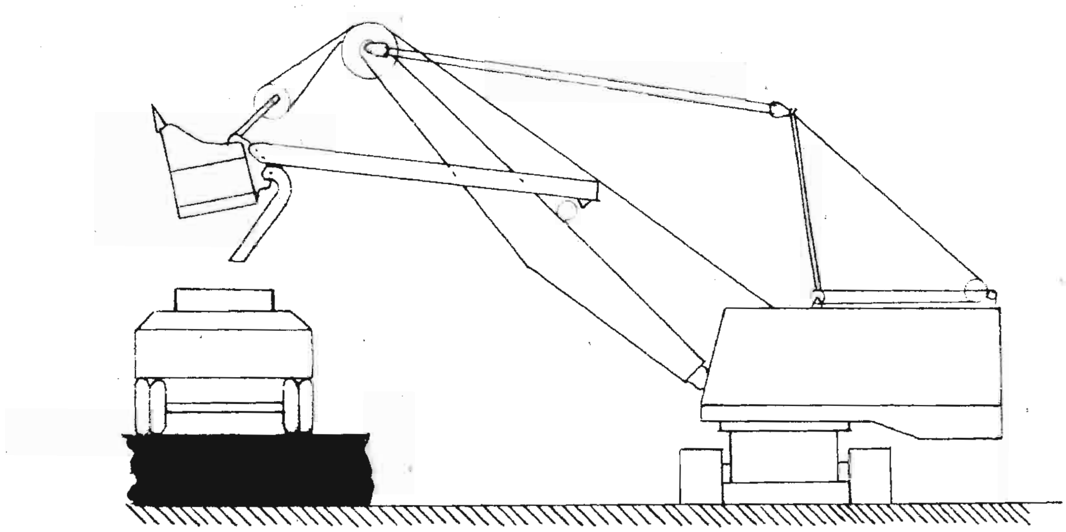
Ditches are frequently wet because they are below the surface. They readily catch rain water and seepage. This decreases shovel efficiency because it may necessitate the use of mats. In any case, the shovel cannot excavate below the level of the water table.

Ditching is often an auxiliary operation in mining and in other earth-moving projects. If the ditching is a small part of the work or if an emergency necessitates the ditching, it is economical to use the power shovel if it is the only machine immediately available. If there is much ditching to do and no emergency, it would be more economical to use a more suitable machine, such as a trench hoe, clamshell, dragline, or a scraper. Other than the large stripping shovels, most shovels can easily be converted into clamshells, draglines, and trench hoes.

Figure 3 shows the maximum dumping height of the power shovel. A study of the picture reveals that for the dipper to be raised to the height shown, the hoist must work against the crowd. This is not only time and power consuming, but also a cause of destructive wear.

As stated before, Figure 4 shows a more efficient dumping height. Generally, if the truck operates on the same surface as the shovel, hoisting distance is a negligible factor since the difference between the distance through which the dipper is hoisted to fill it and the distance it must be hoisted to dump is so small that the shovel can accomplish it while swinging through a short arc. Further, when the trucks are operating on the same level as the shovel, the swinging angle can be cut to the minimum as can the crowding distance.

An exception to this principle is shown in Figure 7. It is standard practice in coal mining to have the trucks operate on the upper bedding plane of the coal and the shovel on the lower bedding plane. The hard, smooth upper bedding plane provides an excellent road for the trucks without added cost. In some cases, such as that



Marion Type 111-M Coal Loader

Figure 7. Standard practice in coal-loading.

shown in Figure 7, the extra hoisting time is more than compensated for by the decreased swing time. Further, since coal loaders have long booms and dipper sticks, the dumping height is still near the height of the saddle blocks.

In sand and gravel excavation shovels are frequently used to excavate down to the water table. The trucks operate on a level above the shovel where the drier sand and gravel forms a better road bed. The same is true in digging shallow ditches. The trucks operate on the bank above rather than on the disturbed and wet soil in the ditch.

Loading to units on the bench above is also practiced occasionally to aid in gaining altitude rapidly. By gaining elevation at the shovel, a decrease in the grade of the road may be possible. This is especially important when track haulage is used.

In some mining and other earth-moving systems, hoppers which feed onto conveyor belts take the place of hauling units. Figure 8 shows the system used for stripping overburden at the Claycraft Company Shale Pit, Taylor Station, Columbus, Ohio. The hopper and the belt are mounted on skids and the assemblage is moved by the shovel. This short conveyor belt makes it possible for one machine and two men to handle the entire stripping operation.

Figure 9 shows another system that has been used. Here the hopper straddles the conveyor belt and travels on rails. The shovel makes a cut parallel to the conveyor belt. As the shovel advances it moves the hopper. After a cut has been completed the conveyor belt, which is built in sections, is shifted close to the long

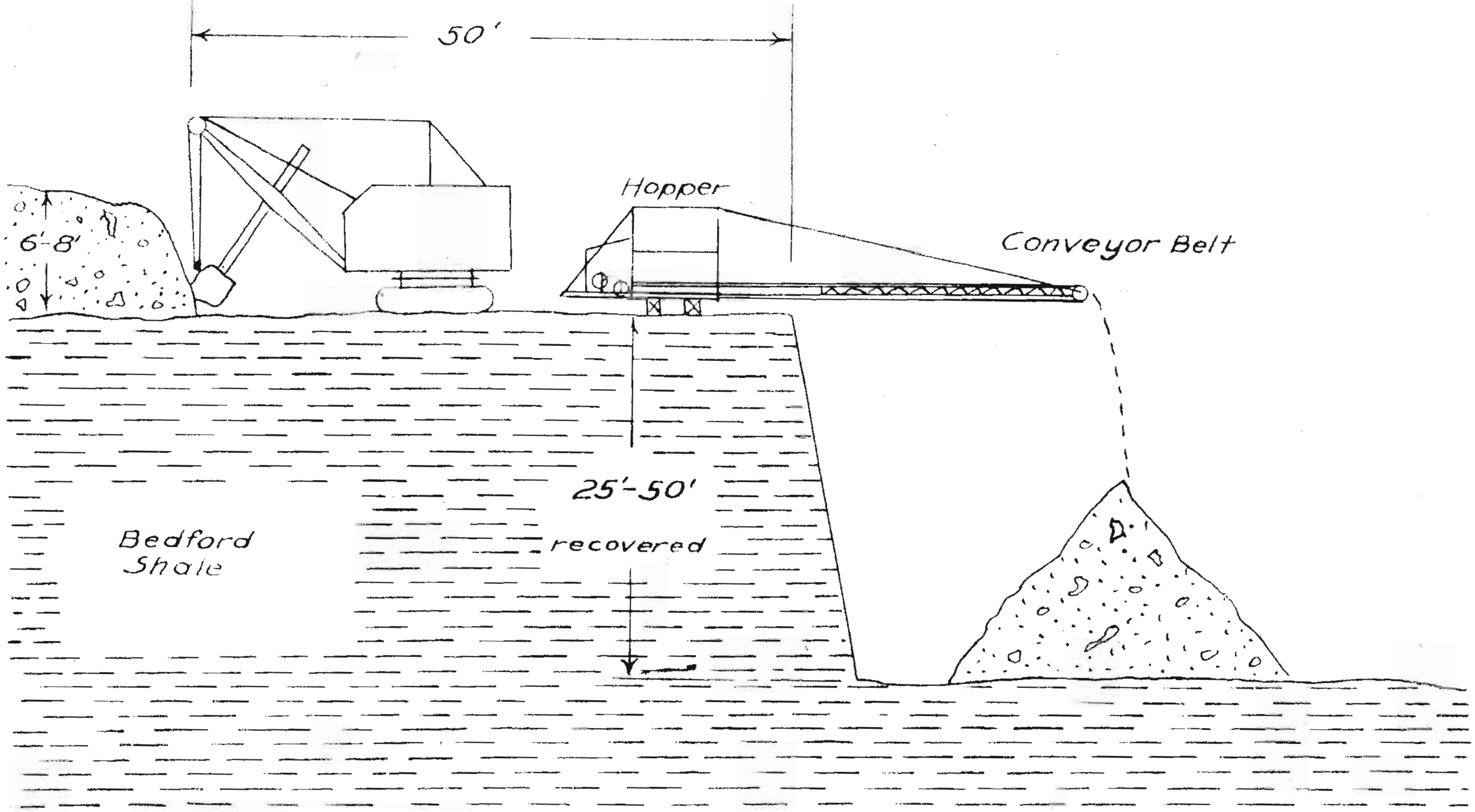
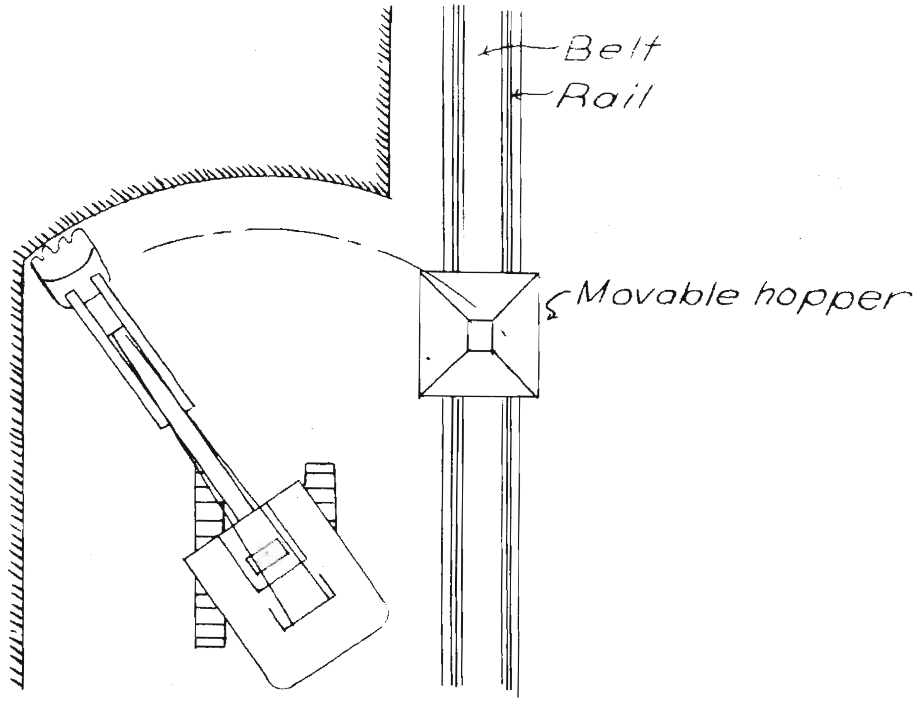
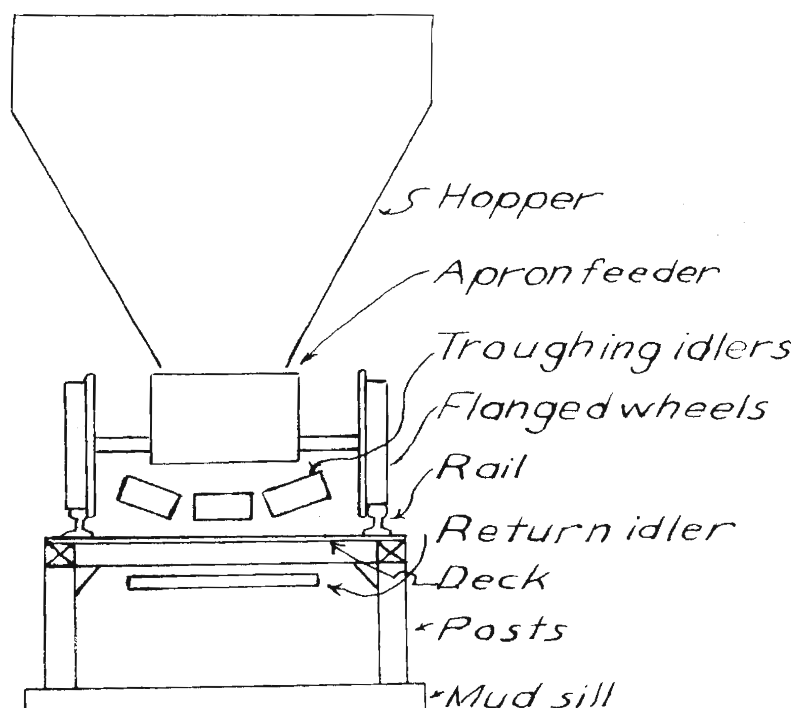


Figure 6.[4] A power shovel — hopper — conveyor belt system

(4) After Linter, E. J., Mining and grinding methods and costs at the Claycraft Co. shale pit, Taylor Station, Columbus, Ohio; U.S. Bureau of Mines, I.C. 6885, p. 5, 1936.



PLAN



SECTION THROUGH HOPPER

Figure 9. Shovel-hopper-conveyor belt system.

face and another cut is begun. An overhead tractor loader can be used to widen the cut before shifting the belt.

In still another system a short portable conveyor has a hopper and feeder at one moveable end. The other end of the conveyor is the pivot end. At the pivot end the portable conveyor has a transfer chute to a second conveyor. By swinging the hopper end of the conveyor, it is possible to work a large area.

These conveyor systems have shown startling economies of operation and first cost. Conveyor systems minimize production delays due to such things as haulage delays due to truck breakdown or truck slow downs in bad weather. Conveyors insure an even, continuous production with a minimum cost and labor.

Thorough bank preparation is essential for efficient shovel operation. Though a power shovel can dig a tighter bank and handle larger rock than any of the other excavating machines commonly used, it is not a substitute for explosives. Much wear, many break-downs, and increased maintenance costs are caused by repeatedly putting a stalling load on the machine when digging into a poorly-prepared bank.

Digging in an ill-prepared bank greatly increases the shovel cycle time. A tight bank decreases the hoisting speed and often necessitates two or three passes of the dipper to get a full load. Time is also lost in balancing large boulders on the dipper to load them. In one series of time studies (Table IV) the time lost in moving large boulders to the side for later blockholing varied from 1.1 percent to 12.3 percent of the total test time.

Destructive wear is caused by the practice of some operators in attempting to dislodge boulders in the bank by placing the side of the dipper against the boulder and swinging the machine. The shovel is not designed to take a stress of this magnitude acting perpendicular to the vertical plane of the dipper sticks and the boom. This stress is also put on the machine when the operator uses the dipper to sweep the pit floor in front of the machine or, again, when he begins to swing the machine before the dipper is clear of the bank.

A worse practice is that in which the operator drops the dipper on large boulders to break them so that the fragments will fit the dipper. A power shovel is not a substitute for a rock-crusher.

The number of dippers full that can be taken from a bank without moving the shovel increases as the height of the bank increases. A high bank is also insurance that each dipper will be filled in one pass upward through the bank. In low banks it is impossible to fill the dipper. The shovel piles the material in a windrow as it advances into low banks. As bank height decreases the time used moving the shovel increases.

The height of the bank that a shovel can efficiently and safely work is limited by the way the bank overhangs as excavation progresses at the toe of the bank. All material, even unconsolidated sand and gravel, will overhang. Even though the caving of the overhanging bank does not always injure the personnel or machine, it almost always slows down production because as the bank slumps against the crawlers of the machine it makes it necessary to back the shovel out to clean up the pit floor. The large boulders will roll farther causing more road clean up.

Personnel should never go between the shovel and the bank. If repairs or servicing are necessary on the bank side of the shovel, move the shovel out before beginning to work.

For maximum efficiency and minimum wear the shovel should work as close to the bank as is consistent with safety. A study of the structure of the shovel and the stresses set up in digging shows that the digging effort is much greater with a short handle than it is with a long handle and the resulting reactions in the machine are much less.

In Figure 10 the shipper-shaft is the fulcrum about which the moments of the stresses are taken. Length "a" is the moment arm of the digging effort. Length "b" is the moment arm of the center of gravity of the dipper, the load in the dipper, and the dipper sticks. Length "c" is the moment arm of the bailing effort. When taking the summation of the moments about the shipper-shaft, arm "c" times the bailing effort should equal arm "b" times the weight acting at the center of gravity plus the product of arm "a" and the resistance to digging. Solving for the digging resistance will give the digging effort possible. In the cases that follow the bail pull is a constant. The weight of the dipper, the dipper sticks and the load in the dipper is also constant.

Using a long dipper stick as shown in Figure 11, the possible digging effort is 41,000 lbs. The reactions in the machine are of interest. The reaction of the dipper sticks against the shipper

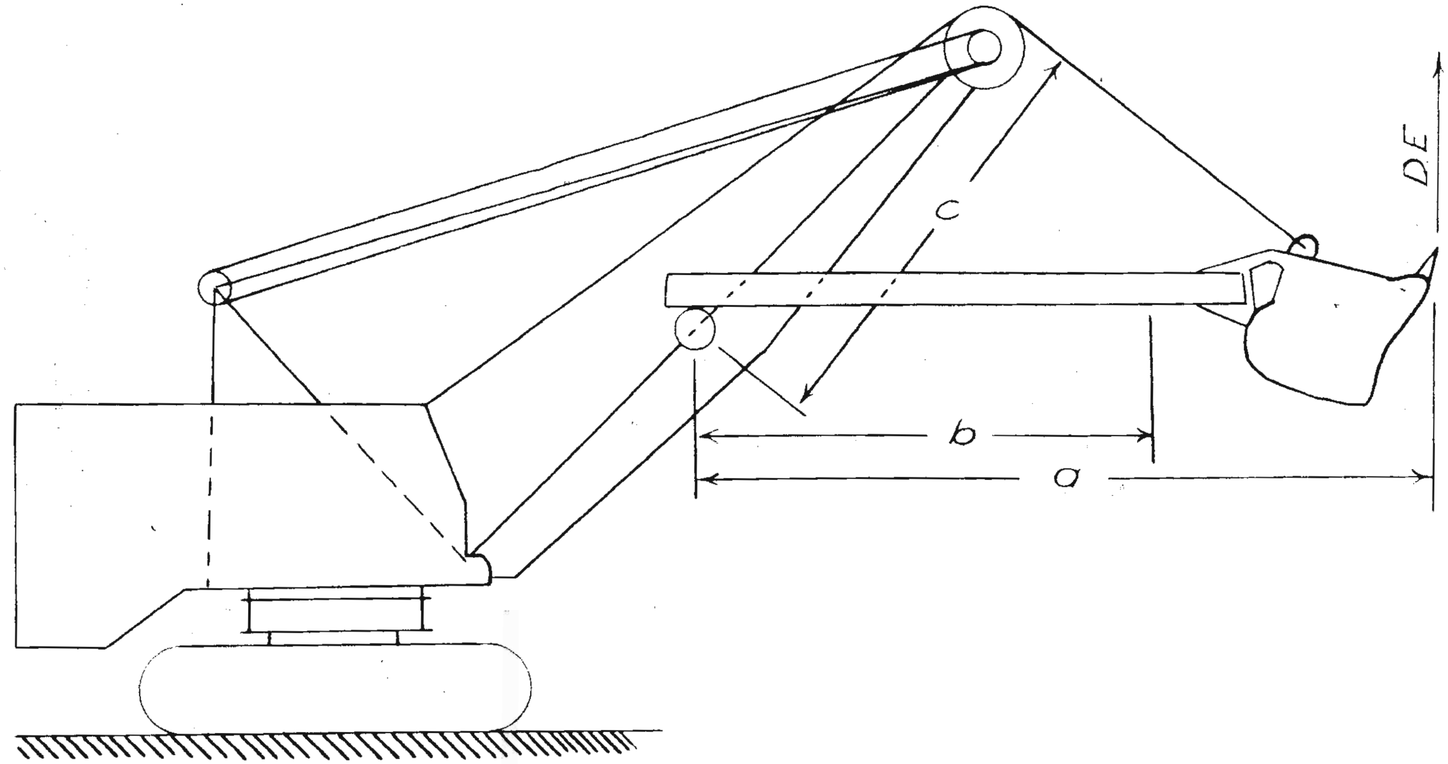


Figure 10. Moment arms of forces about shipper-shaft.

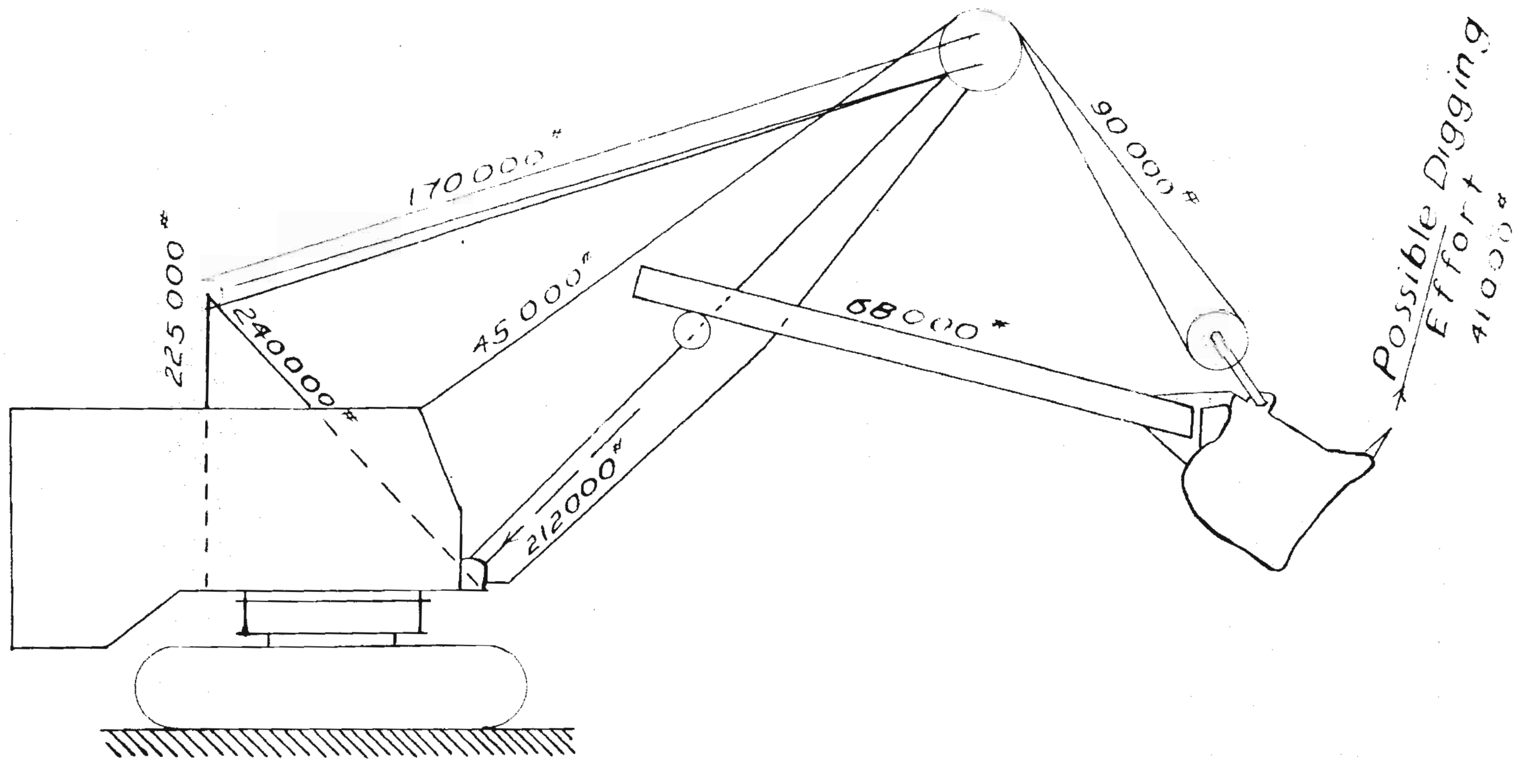


Figure 11. Stresses and possible digging caused by a constant bail pull and extended dipper sticks.

shaft is 66,000 lbs. The boom foot reaction is 212,000 lbs. The suspension tackle stress is 170,000 lbs. The reaction in the front legs of the A-frame is 240,000 lbs., and the reaction in the back legs of the A-frame is 225,000 lbs.

Contrasting the preceding results with those obtained with the same machine using a short dipper stick, as in Figure 12, shows amazing differences. With the short stick, the possible digging effort is 55,800 lbs., an increase of 36.1%. The stress in the suspension tackle is 114,000 lbs., a decrease of 33%. The boom-foot reaction is 210,000 lbs., a decrease of 1%. The resultant stress in the front legs of the A-frame is 160,000 lbs., a decrease of 33.3%. The stress in the back legs of the A-frame is 151,000 lbs., a decrease of 32.9%.

In these calculations the weight of the boom was not considered because its effect on the stresses would be a constant in both cases. The center of gravity of the dipper, dipper sticks, and the load was assumed to be at the same point in both cases. Actually, the center of gravity changes as the length of the dipper stick changes. The effect of this would be to lighten the load acting through the center of gravity since some of the weight of the dipper and its load would be counterbalanced by the weight of the dipper sticks extending to the left beyond the shipper-shaft. The net results of this would make the differences greater than shown above.

Another interesting feature is brought out by the study of Figures 11 and 12. In Figure 11, the bail pull has a large resultant stress acting in line with the dipper sticks. To hold the

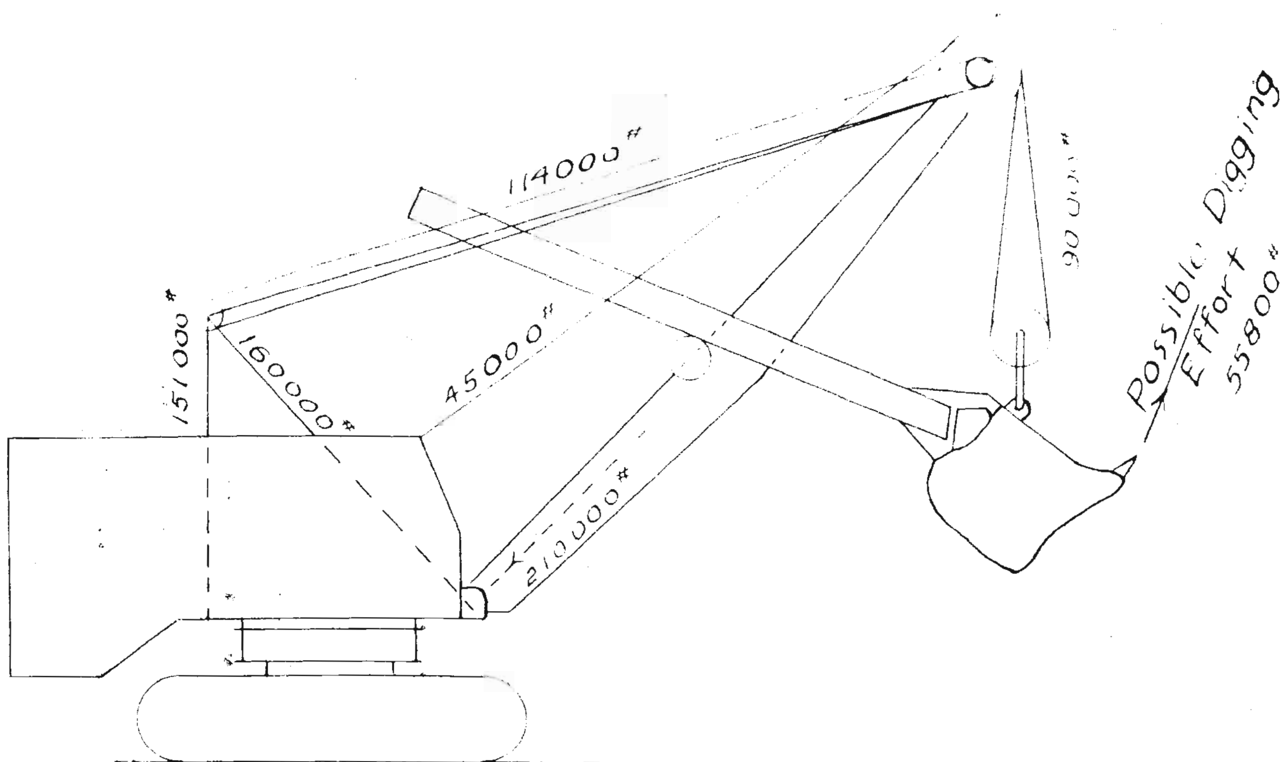


Figure 12. Stresses and possible digging effort with a constant bail pull and short dipper sticks.

dipper in the bank, the crowd machinery must work against this resultant of the bail pull. Thus there are two parts of the shovel working against each other. The same condition can be observed in Figure 3 when the shovel is dumping at its maximum height. In Figure 12 the resultant of the bail pull acting along the dipper sticks is almost negligible and the crowd machinery is not overworked.

Figures 13 and 14 illustrate another aspect of the importance of working with a short dipper stick. In this case an assumption was made that the shovel was digging through a bed that offered 35,000 lbs. of resistance to digging. The summation of moments was then made about the shipper-shaft to find the needed bail pull to overcome the digging resistance. With a long handle, as in Figure 13, 75,200 lbs. of bail pull were required to break through the bed. The reacting stresses in the shovel are as shown on the sketch. Using a short handle, as in Figure 14, the required bail pull was only 67,300 lbs. With the short handle, the boom-foot reaction was 22.9% less; the suspension tackle reaction was 46.9% less; the reaction in the back legs of the A-frame was 69% less; and the reaction in the front legs of the A-frame was 68.5% less. The economies possible with the short handle are not only those of lessened stresses and resultant wear but also of lessened bail pull which allows greater hoisting speed and less consumption of power.

In the last mentioned case, further wear and tear on the shovel may be saved by changing the boom angle. Shovels normally operate with the boom at a 45 degree angle. If the shovel encounters tough

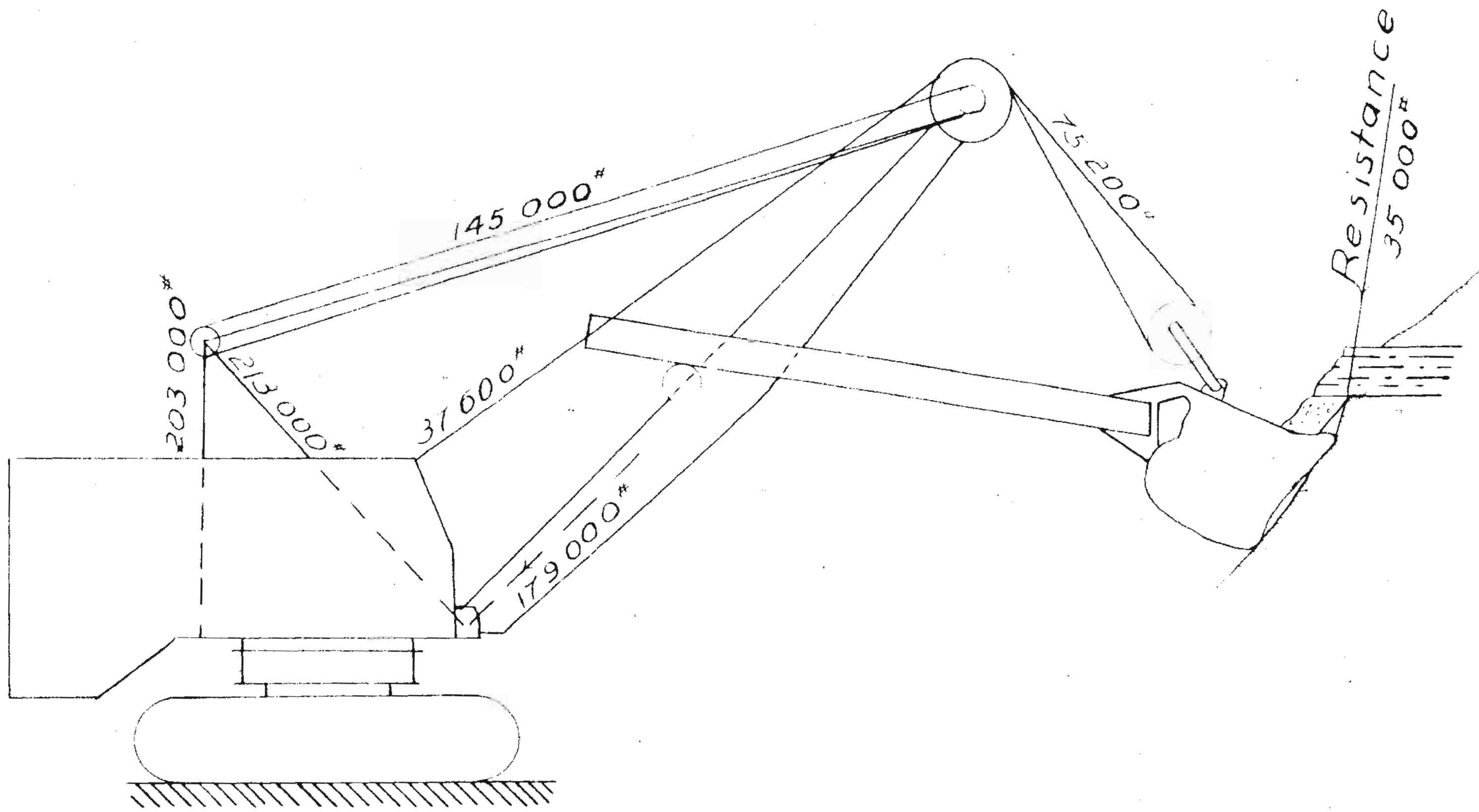


Figure 13. Stresses and bail pull with constant resistance and extended dipper sticks.

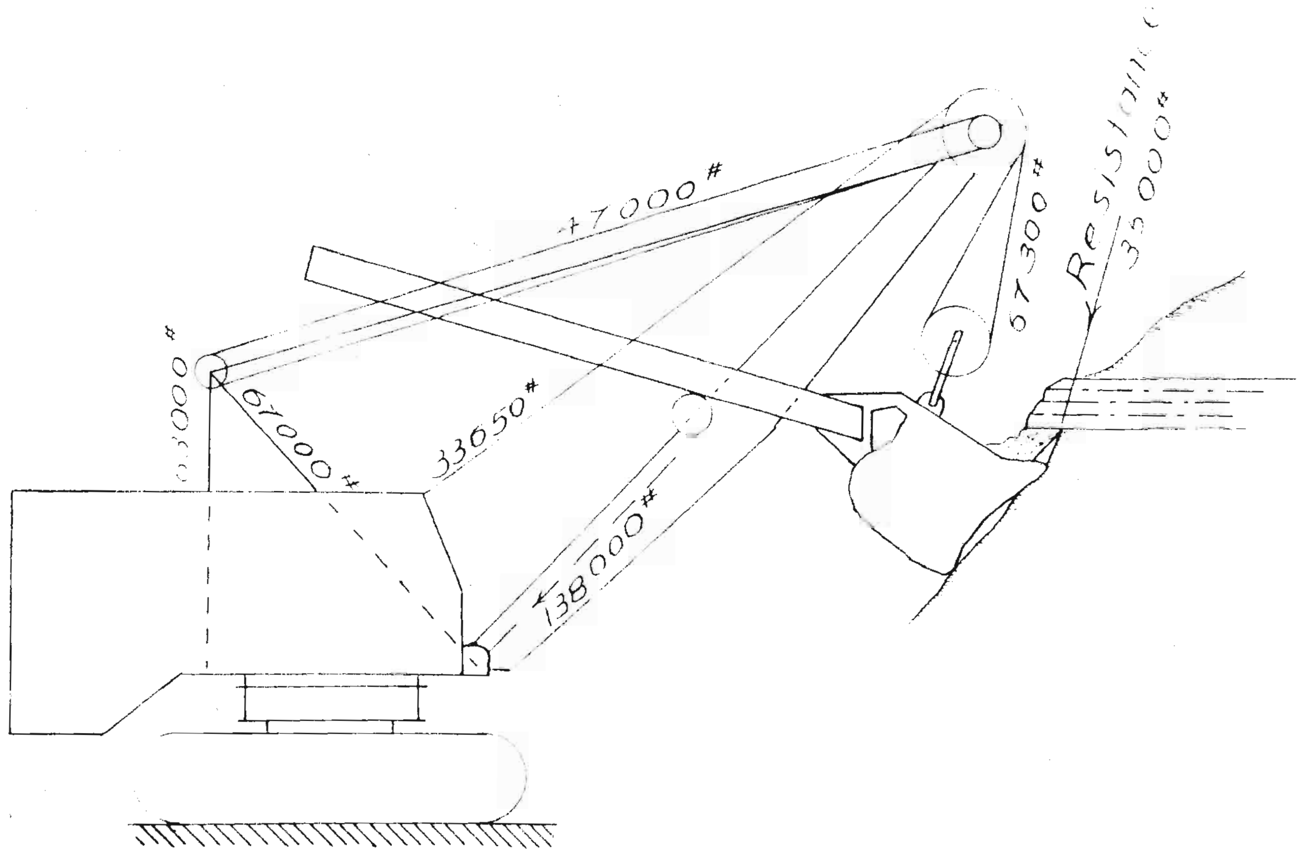


Figure 14. Stresses and bail pull with constant resistance and short dipper sticks.

digging near the top of the bank, as in Figure 15, it is good practice to use a short handle and to increase the boom angle. To raise the boom and use a short handle shortens the tipping moment arm ("a" in Figure 15.) To shorten the tipping moment arm decreases the magnitude of the tipping moment which moment causes the rear of the shovel to raise and thus causes rocking of the machine. To rock the machine will cause the ill-effects previously mentioned.

Frequently, too, the boom angle is increased to increase the dumping height as when loading into haul units on the bench above the shovel. Or, again, the boom angle may be increased to decrease the needed clear radius for swinging as in ditching.

If tough digging is encountered at the toe of the cut, it may prove helpful to decrease the boom angle as shown in Figure 16. In this case, decreasing the boom angle makes it possible for the crowd to transmit more of the weight of the boom to the dipper teeth to hold the teeth into the bank. Further, there will be a horizontal component of the bail pull which will help to force the teeth into the bank.

Another adjustment of angles that does not receive the attention it should is adjustment of that angle which the dipper makes with the dipper sticks. This angle is controlled through the dipper pitch braces as shown in Figure 17. If the pitch braces are not properly set, the digging force will not act in line with the dipper teeth and direction of motion as it should (see "B", Figure 17). If the pitch of the dipper is too great, the digging force will act through the flat upper surface of the teeth as in "A", Figure 17. This increases digging resistance and breaks the dipper teeth. If

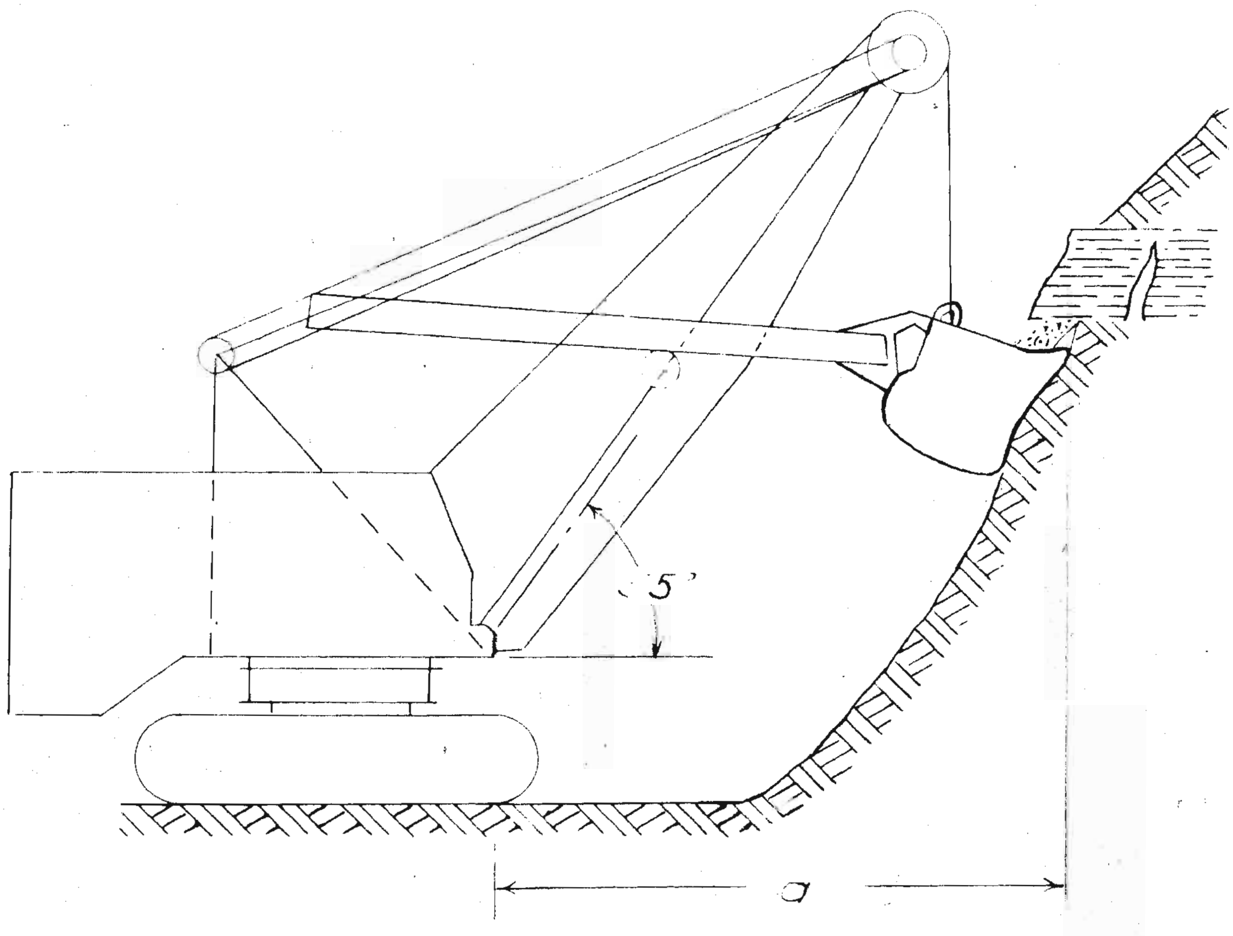


Figure 15. The correct use of the high boom angle.

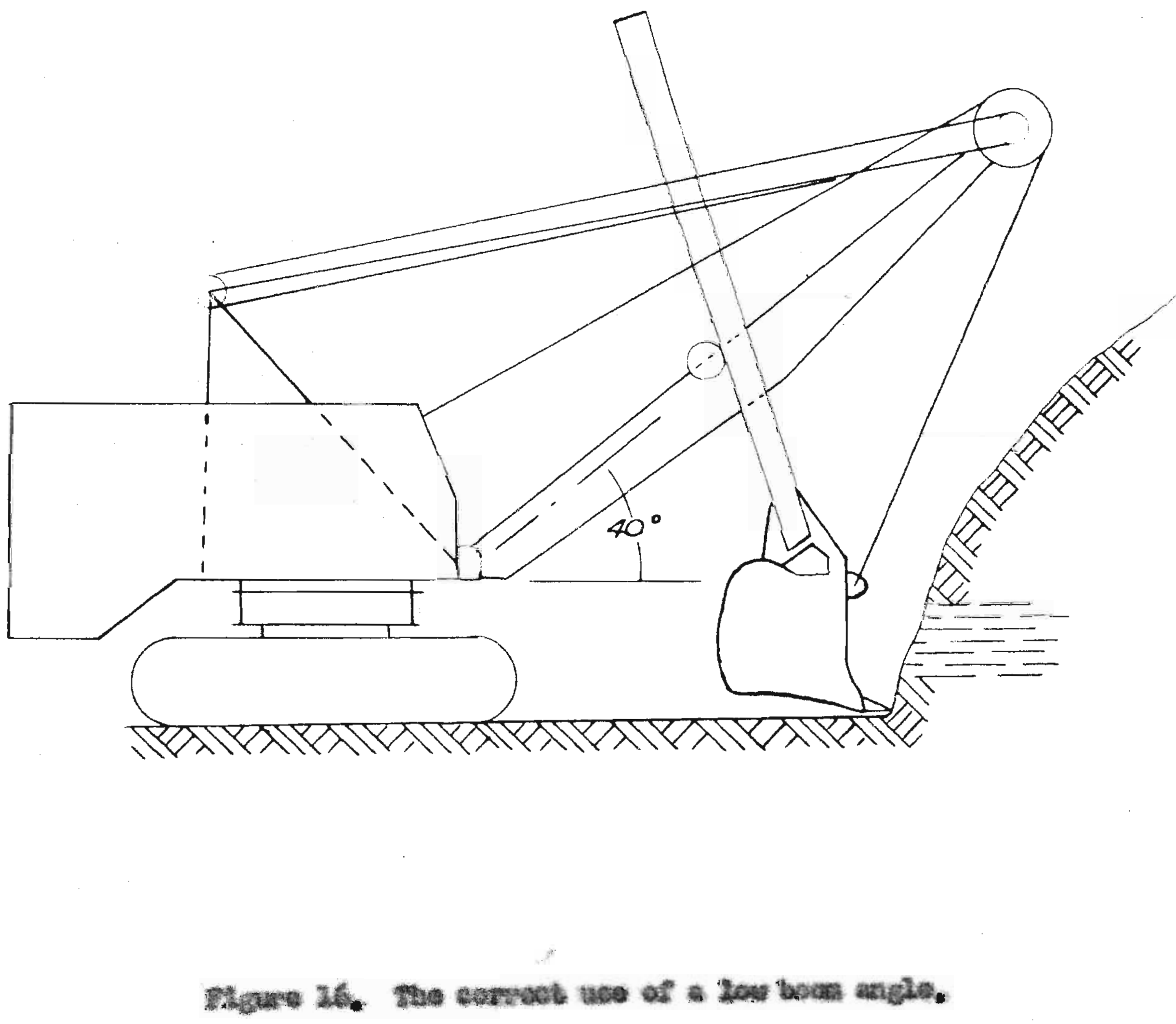


Figure 16. The correct use of a low boom angle.

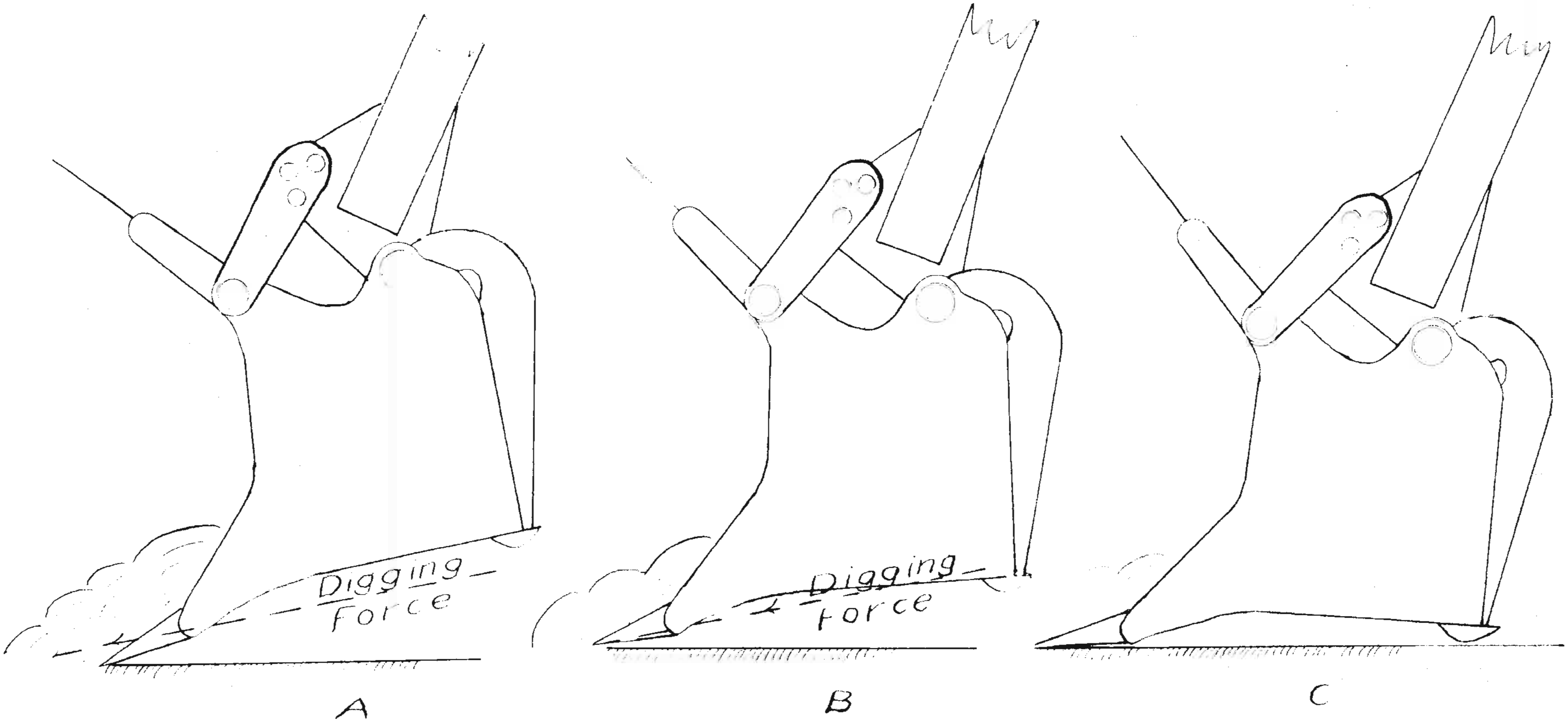


Figure 17. Adjustment of dipper pitch brace.

the pitch is too low, the bottom of the dipper is subject to undue wear as in "C," Figure 17.

To increase shovel capacity every effort should be made to decrease the angle of swing. A study of Table IV shows that over 50 percent of the shovel cycle time is used in swinging. The following equation may be used to compute the swing time of a shovel per cycle.

$$\text{Swing time (secs.)} = \frac{(2)\ (\text{swing angle})(60)}{(360)\ (\text{shovel RPM})} \qquad (8)$$

Example 3 demonstrates the use of equation (8) and the increase in production made possible through a decrease of average swing angle.

<u>Example 3</u>

A small shovel is making 2.5 cycles per minute with an average swing angle of 100 degrees. The swing speed of the shovel is 3 revolutions per minute. What would be the increase in production if the swing angle were reduced to 60 degrees?

The decrease in the swing angle equals 100 degrees minus 60 degrees equals 40 degrees.

Using equation (8), the time saved by not swinging 40 degrees is

$$\frac{(2)(40)(60)}{(360)(3)} = 4\text{-}4/9 \text{ seconds saved per cycle.}$$

The time for the old cycle is

$$\frac{60}{2.5} = 24 \text{ seconds}$$

The time for the new cycle becomes

$$24 \text{ seconds} - 4\text{-}4/9 \text{ seconds} = 19\text{-}5/9 \text{ seconds.}$$

The shovel can make 11 cycles in the same time in which it previously could make but 9 cycles if 40 degrees are cut from the average angle of swing. This decrease in cycle time amounts to a 22.2 percent increase in production.

In this example it is not necessary to make allowance for acceleration and deceleration since they would be the same for 60 degree swing as they would for a 100 degree swing.

The relationship between the average swing angle and the depth of cut is illustrated in Figure 18. With the narrow cut shown here, the average angle of swing is 59 degrees. To work a narrow cut such as this the shovel must spend more time moving than it would if working a wider cut. In Figure 19 the shovel has an average angle of swing of 95 degrees. In this case the shovel can excavate a greater amount from the wider cut per move than it can in Figure 18.

The advantages of the cases pictured in Figures 18 and 19 can be combined by double spotting of trucks as shown in Figure 20. Here the average angle of swing has been cut to 67 degrees for one truck and to 72 degrees for the second truck. The trucks are loaded alternately to each side. This system has the further advantage of minimizing delays caused by waiting for one truck to pull into the loading position as the other is pulling out. In Figure 20, one truck can be drawing into the loading position at one side while another truck is being loaded at the other side.

Figures 18, 19 and show show the correct method of placing the truck in the loading position. The hauling unit should always be

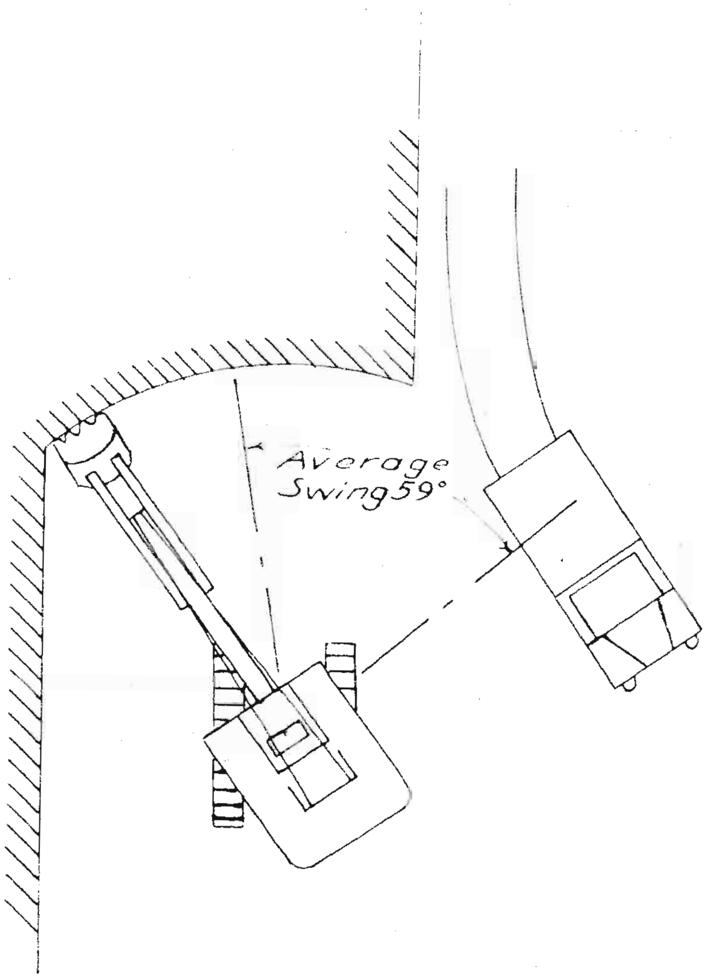


Figure 18. Narrow cut and small swing angle.

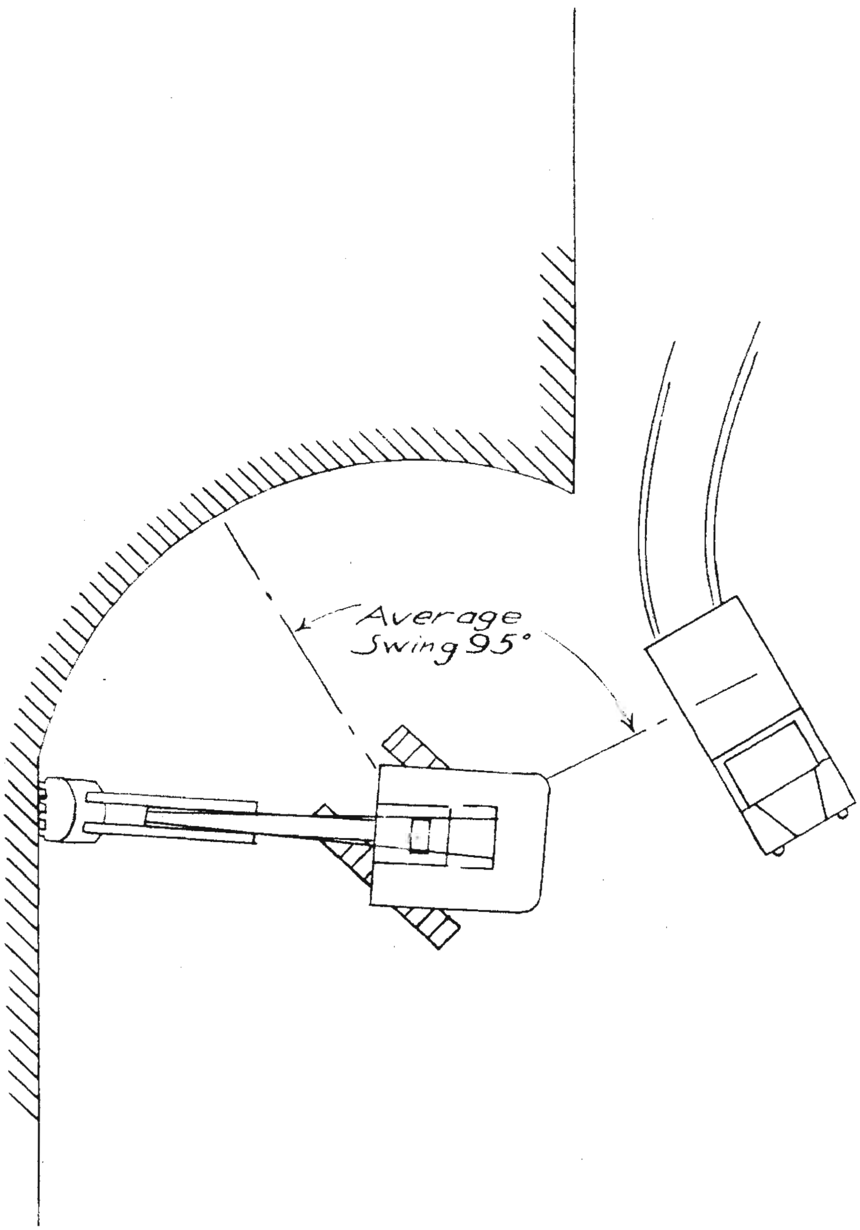


Figure 19. Wide cut and large swing angle.

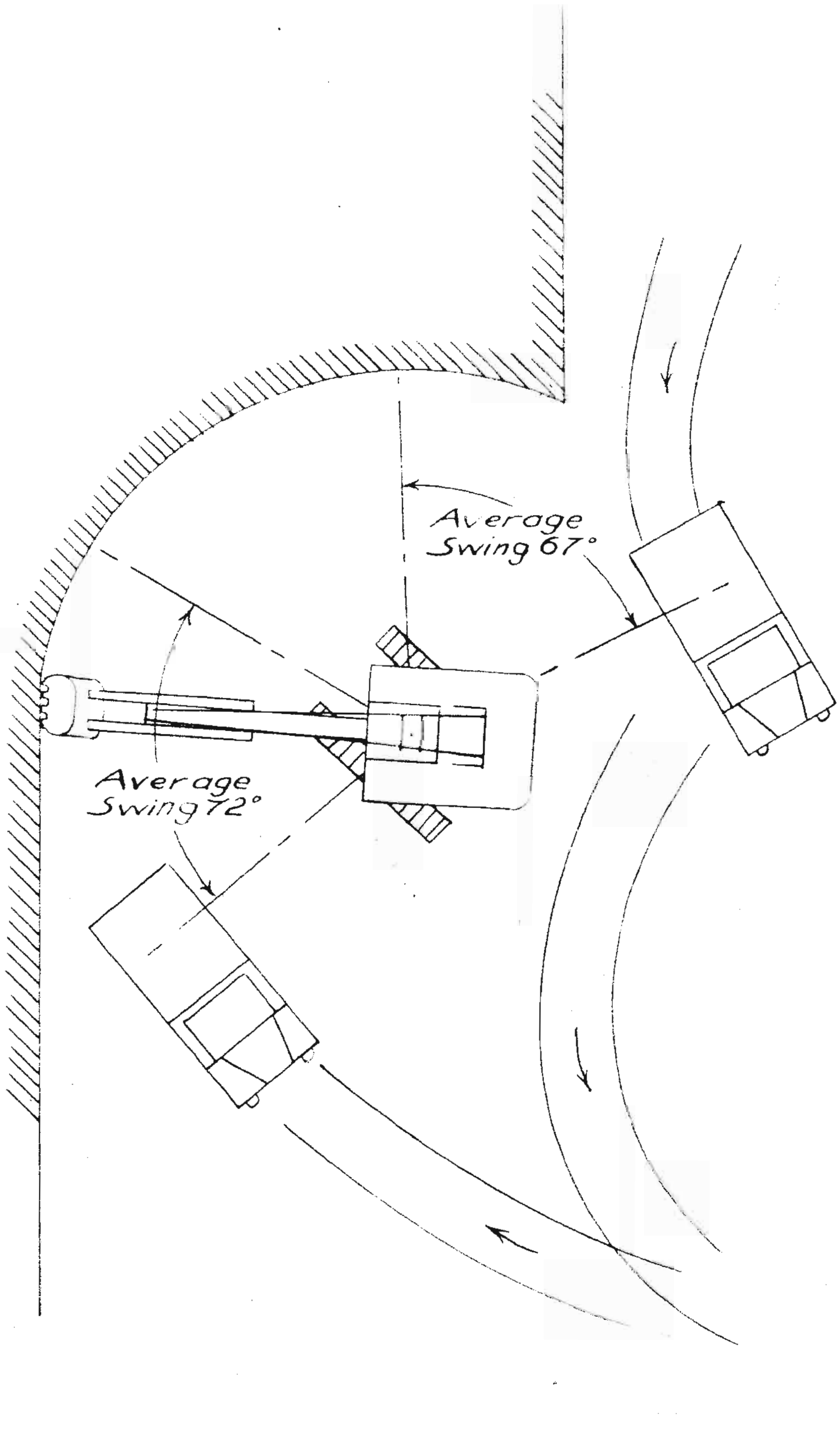


Figure 20. Double spotting to decrease swing angle.

placed so that the long dimension is along the arc that the dipper describes. Care should be taken in spotting the trucks so that the dipper need not be fully extended to reach the truck. When the haul units are properly placed as shown here, the shovel operator can determine how much his dipper sticks must be extended on the first pass. This distance will be the same for succeeding passes. Having determined this distance on the first pass, the operator can then either rack in or crowd as he is swinging his loaded dipper to have it in the right position at the end of the swing. This results in a saving in the shovel cycle and less fatigue to the operator.

Figures 18, 19 and 20 also illustrate an axiom of safety. The dipper of the shovel should never be swung over the cab of a vehicle.

If the bank is not too high and if it does not overhand dangerously, the best way to move the shovel into the face is illustrated in Figure 18. Here the crawlers are parallel to the cut. The shovel advances into the cut in the direction in which the crawlers are headed. This is the fastest method since it is not necessary to back, turn and then advance.

However, if the banks are high and dangerous, the method shown in Figure 18 is dangerous. If the bank into which the shovel is digging were to cave, the shovel could back out readily. If the side bank were to cave, the shovel could be moved out only with difficulty.

In dangerous banks the method shown in Figures 19 and 20 should be used. Here the shovel could back out quickly if there was a warning to either bank caving. Even if the bank did cave against the crawlers, the machine could readily extract itself. To have the crawlers at an angle to the direction of travel as shown here has the disadvantage of taking more time for each move into the cut. Here the shovel must turn as it backs out and then turn as it heads back in.

Except under a few special conditions, the small shovel is not an efficient machine for casting. It is severely restricted by the low dumping height and short working radius as well as by the swell and angle of repose of the excavated material. An efficient casting application for the small shovel is making a cut in the side of a hill and spoiling the material on the downhill side.

The large stripping shovels are exclusively casting machines. The height of the bank which they can strip is definitely limited by the dumping height of the machine. In every case (see Figure 21) the dumping height must be greater than the height of the bank. The cross-sectional area of the cut cannot exceed the cross-sectional area of the spoil pile times the swell factor of the material. This is illustrated in Figure 21.

The cycle of these large stripping shovels will depend on the average angle of swing. The average angle of swing will depend on the width of the cut as illustrated in Figures 18 and 19. Generally, the cut is made as wide as the spoil area will permit to minimize moving. In almost every case these large machines get at least one cycle per minute.

The operations of the Fairview Collieries Corporation at the

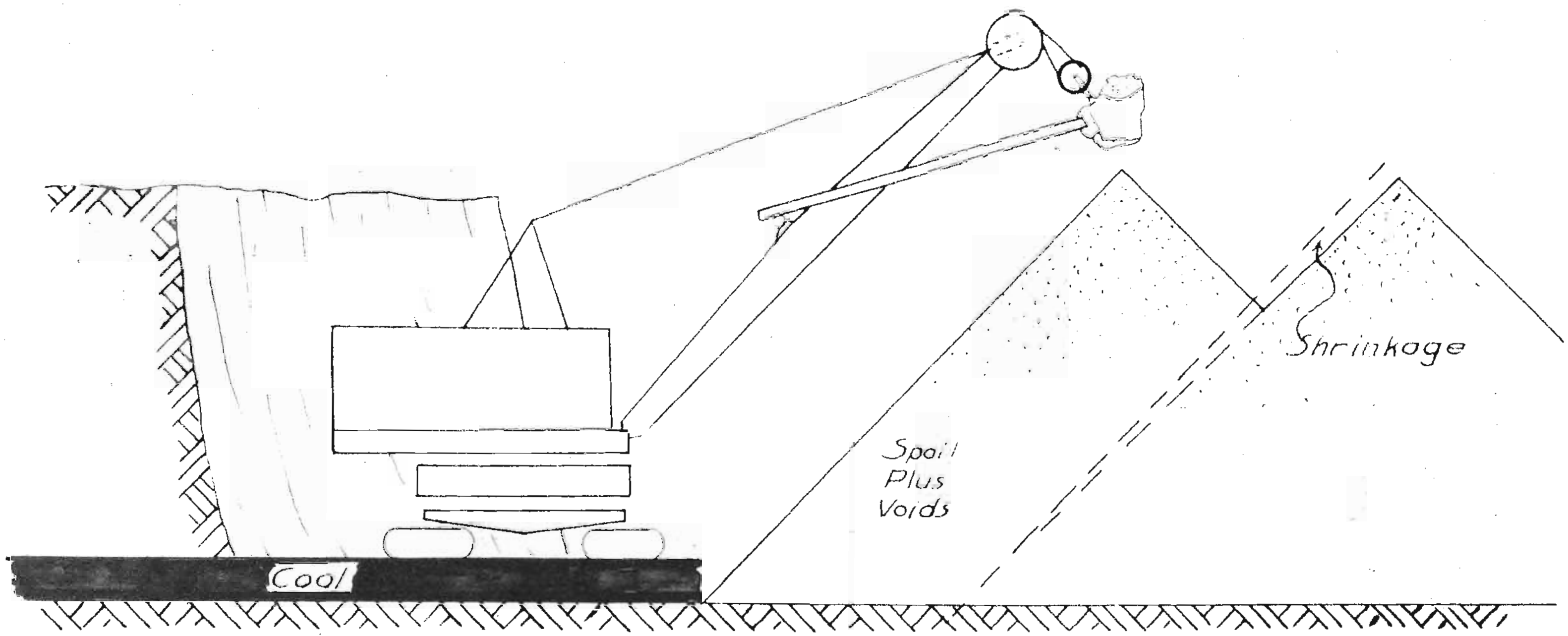


Figure 21. Casting with a large stripping shovel.

Flamingo Mine in Illinois is an excellent example of an efficient coal stripping operation.

The machine used here is a Bucyrus-Erie 1050-B stripping shovel. This machine has a 113-foot boom, a 70-foot tubular steel dipper handle, and a 33-cubic yard dipper. The weight of the machine is 1,400 tons. The machine has automatic hydraulic levelling jacks at each corner and a moving counterweight to counterbalance the dead weight of the dipper and handle.

Other characteristics of the machine are: dumping height, 73 ft., 3 ins.; dumping radius, 114 ft.; height of cut, 98 ft.; cutting radius, 124 ft.; and level floor radius, 78 feet.

In the mining system used at the Flamingo Mine a 40-foot berm of coal is left between the high wall and the spoil. The shovel clears a strip of coal about 50 feet wide spoiling the material across 40 feet of the previous cut leaving a strip of coal 90 feet wide. A 7½-yard Marion coal loader follows the stripping shovel. The coal loader takes out the outside 50 feet of coal leaving a 40-foot berm. The coal is loaded into trucks.

The shale immediately above the coal is placed adjacent to the coal rib on the fire clay of the mined-out cut. This windrow is built up ahead of the stripping machine to the maximum dumping radius of the machine. The softer material from the top of the bank is cast behind this windrow. This is practiced to prevent the soft material from sliding in on the coal berm. The practice is illustrated in Figure 22.

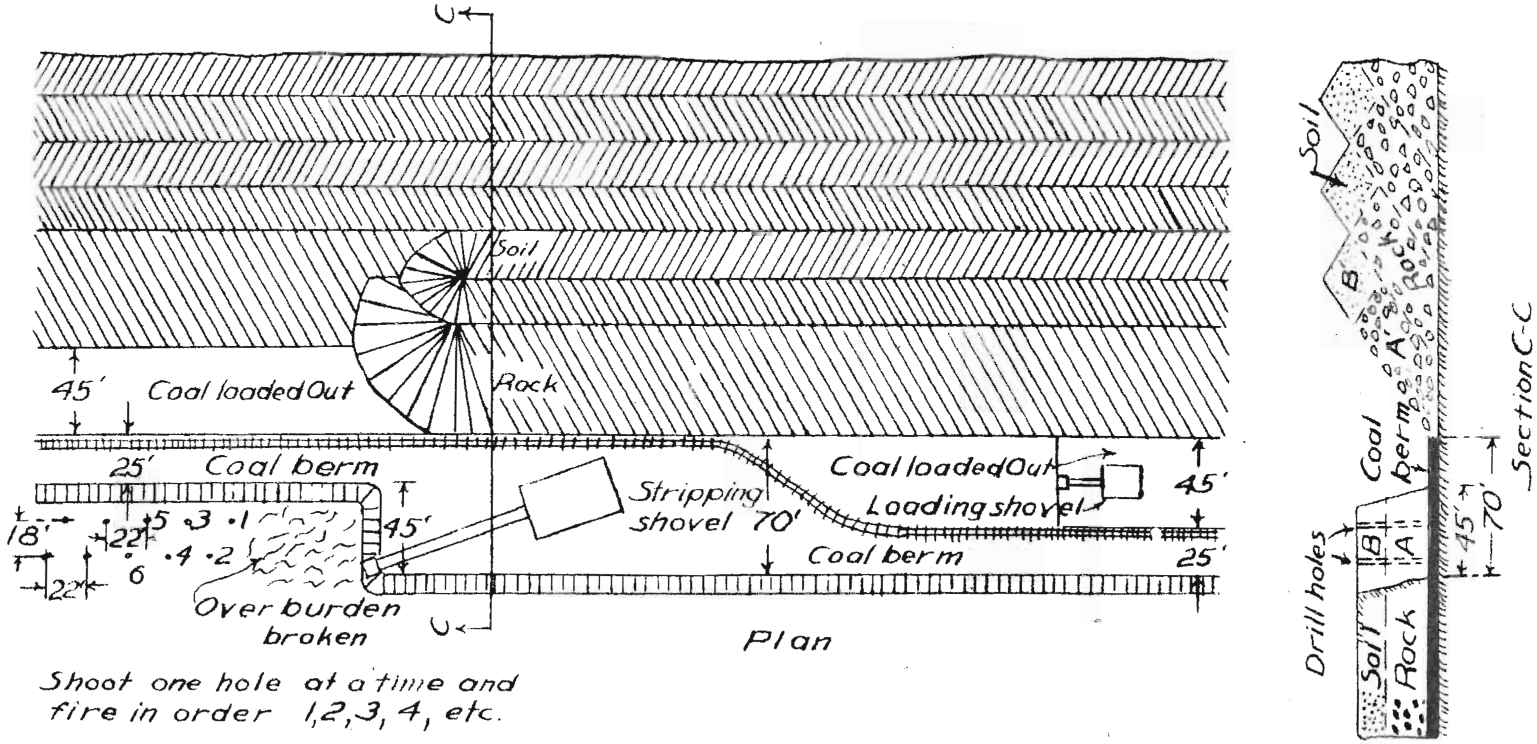


Figure 22.[5] Stripping coal with a power shovel.

(5) After Toenges, A. L. and Anderson, R. L., Some aspects of strip mining of bituminous coal in Central and South Central States: U.S. Bureau of Mines, I.C. 6959, p. 47, 1937.

About six inches of the material on top of the coal is left in place. This blanket protects the coal from deterioration by the weather and from breaking by trucks and other machinery. Bulldozers clean the coal ahead of the coal loader.

To prepare the bank for the stripping shovel, 5-inch holes are drilled from 45 feet to 50 feet into the high wall by a Parmanco drill. The holes are spaced about 30 feet apart. Five or six holes are shot at a time. From 150 lbs. to 200 lbs. of gelatine dynamite are used in each hole.

The cycle of the stripping shovel varies between 50 and 55 seconds. The stripping shovel normally travels between 140 and 150 feet per 8-hour shift. Where overburden was less than the average 45 feet in depth, the stripper sometimes dug and traveled 250 feet in eight hours.

Generally, the system is somewhat like that shown in Figure 22 with the exception that track haulage is seldom used. At the present time trucks are used almost exclusively in strip coal mines.

To shorten the truck haul in some mining systems, coal berms are left at intervals in the spoil piles as in Figure 23. Another advantage of this system is that if the stripping shovel stays far enough ahead of the coal-loader there is no dangerous necessity of the large shovel casting over the trucks.

The disadvantage of leaving intermediate berms for haulage is that it may take up too much spoil area. If the bank is high, the shovel may encounter difficulty in spoiling the material in the

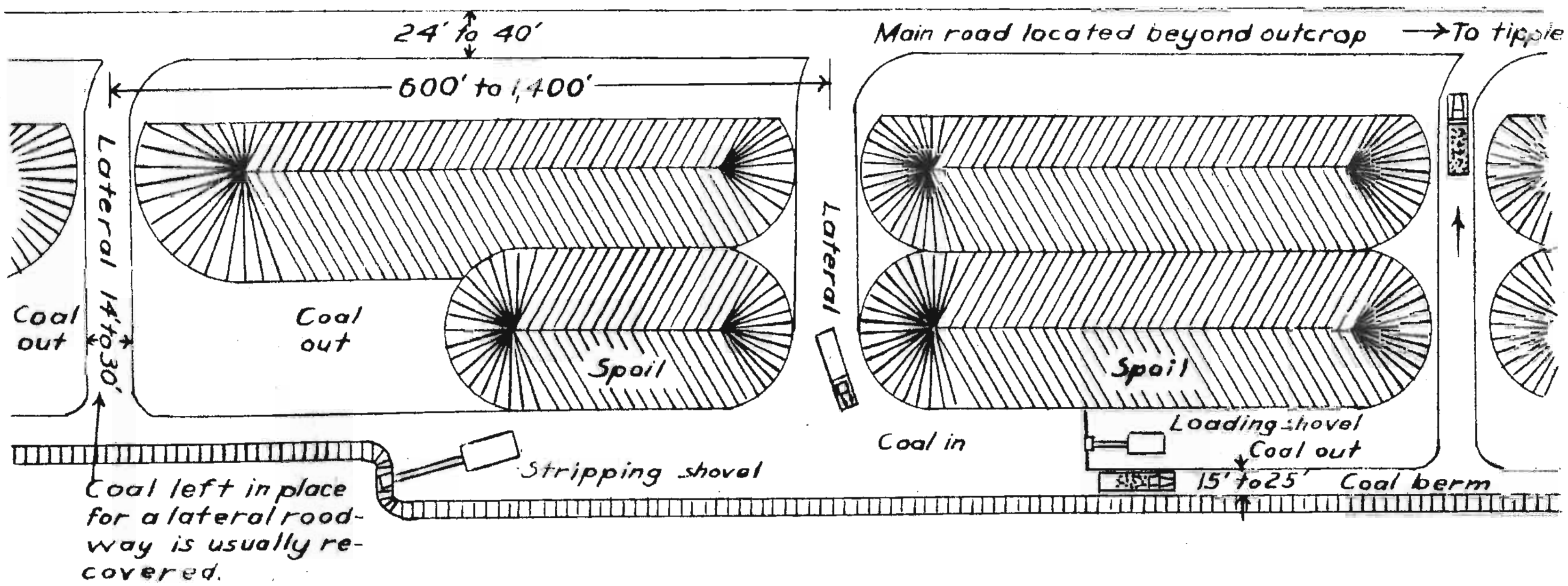


Figure 23.(6) An efficient road system in coal stripping.

(6) After Toenges, A. L. and Jones, F. A., Truck vs. rail haulage in bituminous-coal strip mines: U. S. Bureau of Mines, Report of Investigations 3416, p. 5, 1938.

bank at the end of the intermediate berms into the restricted area at either side of these berms.

An alternative to the methods mentioned is to have the trucks come into the pit on the berm between the coal loader and the high wall, turn around on the wide berm between the coal loader and the stripping shovel, and then go out on the same road on which they came in.

In strip mining coal or other deposits the starting cut is often a difficult problem and a cause in almost every case of extra work. The difficulty arises from the lack of spoil space. When the shovel is in the first cut it must spoil to the bank above. If the overburden is thick there is little space between the dumping height of the machine and the surface of the ground.

If the coal outcrops, stripping will begin at the outcrop and no problem will exist. To minimize rehandling of overburden, at pits where coal does not outcrop, the first cut is usually started where the overburden is the shallowest.

Figure 24 illustrates one method of securing more spoil space for the starting box cut. Here a dragline excavates a borrow pit at "A" and casts the material into the spoil pile "A". The additional spoil space provided by the dragline enables the stripping shovel to take the first cut down to the coal.

Figure 25 illustrates another method of securing spoil space for the material from the first cut. This method entails costly rehandling of the material but has the advantage of not requiring the use of an additional machine.

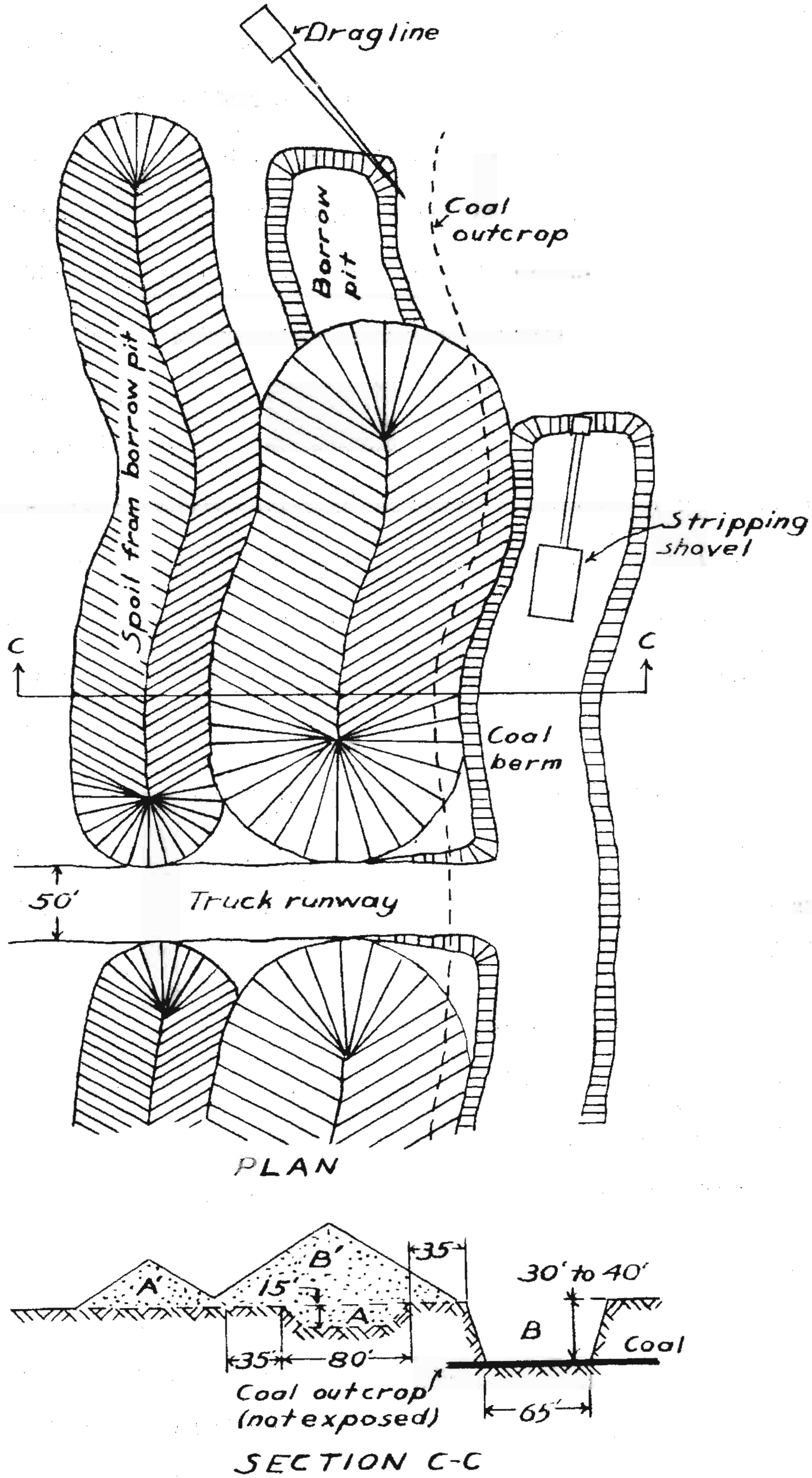


Figure 24.[7] A starting cut in coal stripping.

(7) After Toenges, A. L. and Anderson, R. L., Some aspects of strip mining of bituminous coal in Central and South Central States; U.S. Bur. Mines, I.C. 6959, p. 29, 1937.

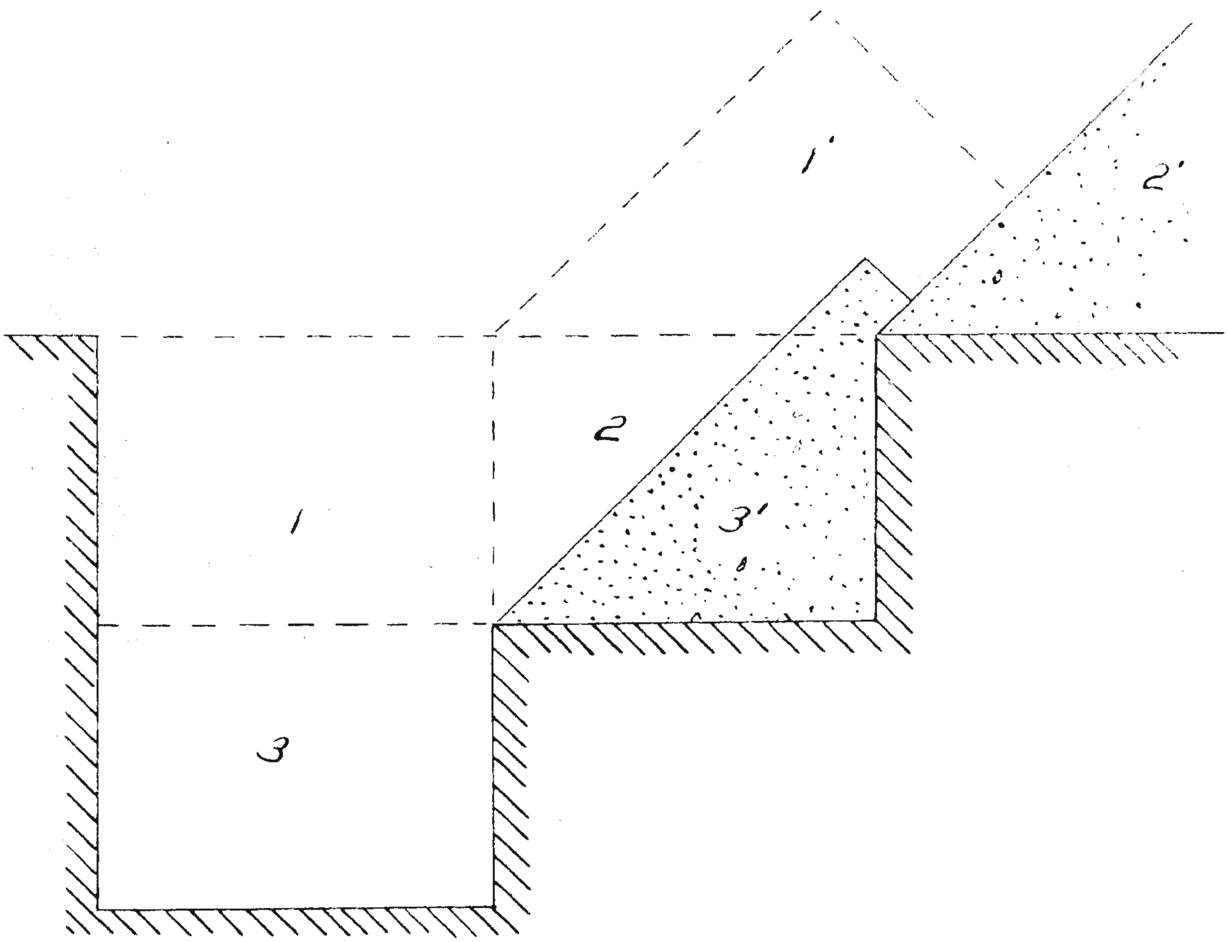


Figure 25. Starting box cut.

Figure 26 illustrates a method of making a turn on the completion of a cut. This or some similar method is necessary because the uncovered coal must be mined before the area can be used for spoil piles. An alternative would be to deadhead the stripping shovel to the other end of the cut and excavate one way only.

As bank height increases beyond the ability of the shovel to spoil the material, it becomes necessary to change the mining system or the type of equipment used. Figure 27 illustrates a mining system which employs both a shovel and a dragline. In this system the shovel excavates the hard material in the lower bank. A dragline follows the shovel and excavates the soft soil in the upper part of the high wall and casts it beyond the spoil pile built by the shovel. This is made possible by the greater working radius and dumping height of the dragline.

This is an inefficient digging position for the dragline but at times it is the only method that can be used. The system does have an advantage in that the broken rock "A'" forms a retaining wall to hold the softer soil "B'". Placing this soft soil at the top of the spoil area makes the land easier to recovery for possible agricultural uses. It also facilitates vertical drilling from the surface of the bench formed by the dragline.

Usually a shovel is not applicable to selective mining. In taking its upward slice through the bank, a shovel mixes the material if it happens to be banded or stratified. In some cases this mixing action is a desirable feature. In excavating material for road fill from borrow pits, it is desired to mix stratified sand,

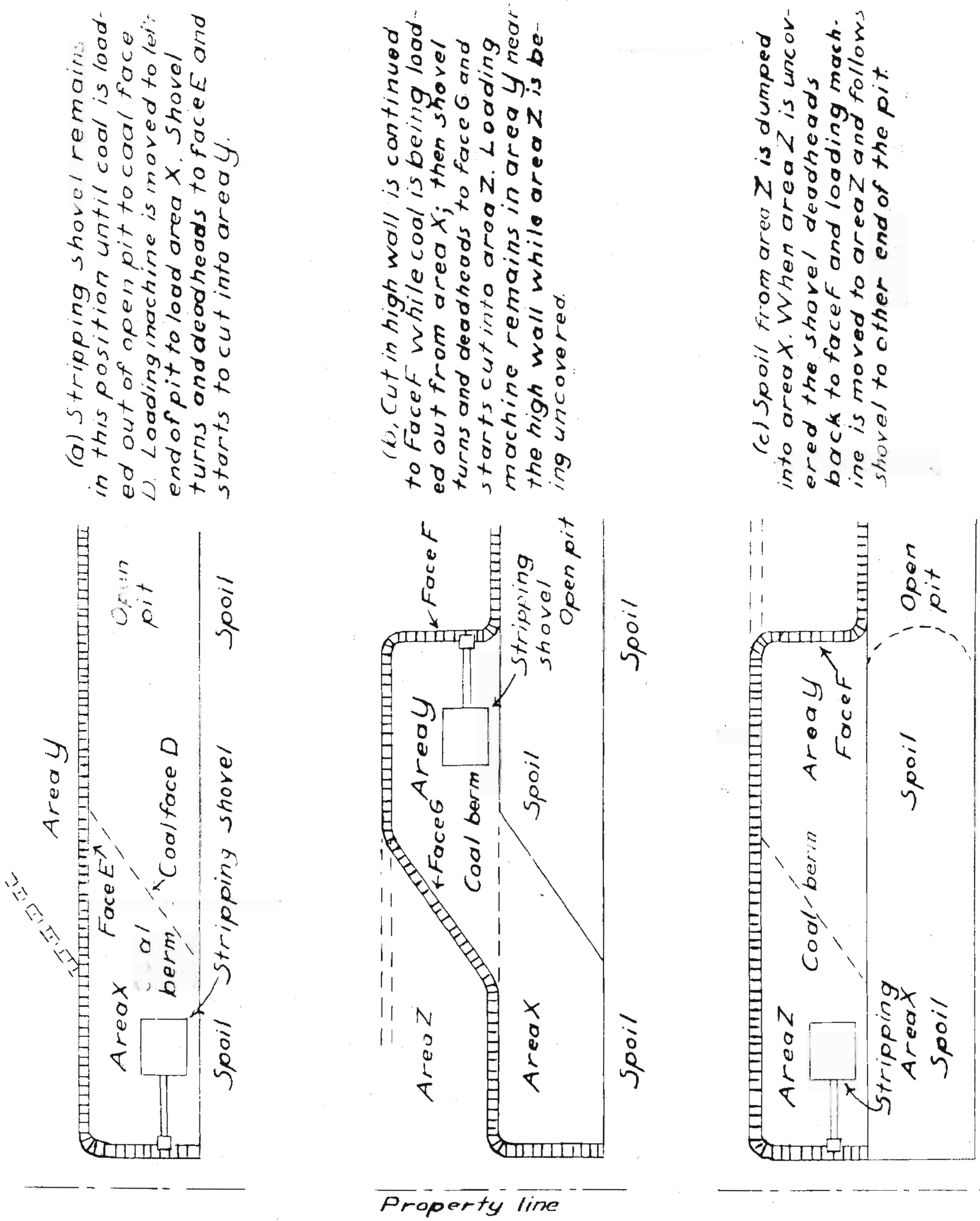


Figure 26.[8] A method of starting a new strip.

(8) After Toenges, A. L. and Anderson, R. L., Some aspects of strip mining of bituminous coal in Central and South Central States: U. S. Bureau of Mines, I.C. 6959, p. 50, 1937.

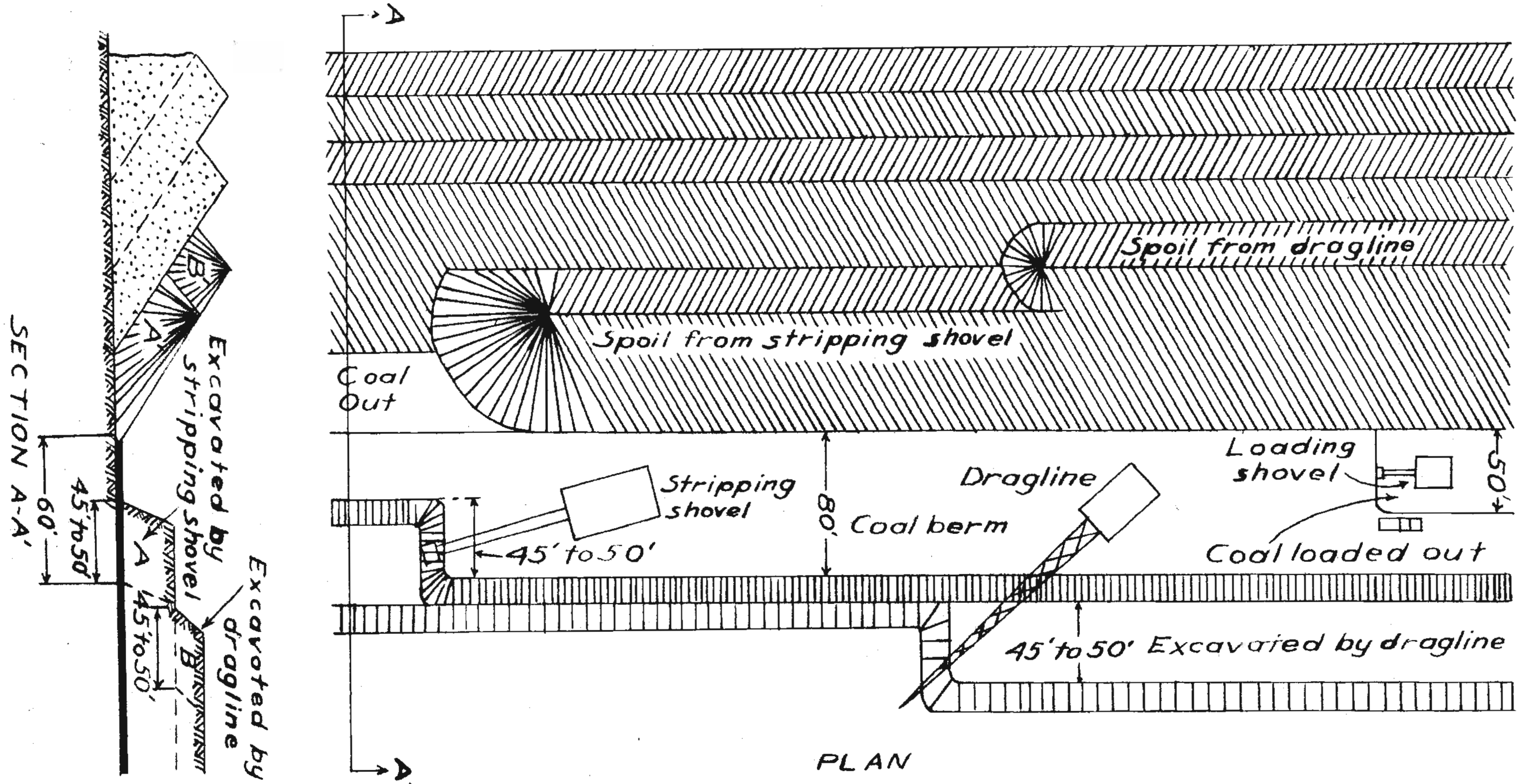


Figure 27.[9] Use of dragline and stripping shovel together.

(9) After Toenges, A. L. and Anderson, R. L., Some aspects of strip mining of bituminous coal in Central and South Central States; U. S. Bureau of Mines, I.C. 6959, p. 39, 1937.

sandy clay, clay, and gravel to insure better binding and compaction when spread. In cases such as this it has at times been specified in contracts that the open dipper be passed through the bank 1, 2, or 3 times for each dipper full removed to insure a better mix. This mixing action is frequently desirable in some industrial clay and shale deposits to aid in getting a more uniform raw material. Planers have been used to accomplish this mixing. In some metallic ore deposits this mixing can be helpful in that it sends a more uniform feed to the mill. In homogeneous continuous materials this mixing action is of no consequence.

In some materials such as the residual phosphates and clays, this mixing action is a definite detriment to the use of the power shovel. Materials of this type frequently enclose lenses of sand. In these cases the shovel cannot dig from the bottom of the bed upward without mixing the sand with the other material. In most cases it would be impracticable, if not impossible, for the shovel to attempt to mine out the clay or phosphate above the sand lense, then drop to the lower level of the sand lense to load out the sand, and then drop still farther to recover the rest of the clay or phosphate. The shovel could dig through the bank mixing sand and clay and load them out as waste. A third possibility would be to have the shovel excavate around the material leaving it in place. In a thick deposit which is being worked in several benches this last mentioned possibility would be exceedingly wasteful.

Proper servicing and preventative maintenance are essential for economic, continuous, and efficient shovel operation. Proper lubrication is a must. At one extreme is the lack of lubrication--at the other extreme is over-lubrication. Too much grease around the machinery eventually results in grease-coated clutch and brake bands. When spotted or coated with grease, the coefficient of friction is reduced and these parts cannot perform their function properly. To attempt to force too much grease into sealed bearings might break the seal.

Besides the usual bearings, the roller rack of the shovel should be greased each shift. The shovel should be swung through 1, 2, or 3 complete swings each shift to change the position of the dollies to distribute the wear evenly.

Wire rope should be lubricated daily to increase its useful life. It should also be checked periodically and replaced if necessary to insure that breakdowns will not occur while operating. The sockets and wire rope fastenings should also be checked periodically. It is a good practice to cut several feet off the end or ends of wire rope occasionally to change the wear points. In addition to all of this, the grooves in the sheaves should be checked periodically and remilled or relined if necessary to prevent undue wear on the wire rope.

It is important to keep dipper teeth built up and sharp for ease of penetration and save wear on the dipper. Dipper teeth can be cast in manganese or alloy steel or forged from high carbon steel. The high carbon teeth can be repointed by forging or welding. A high carbon rod is used to replace the lost metal and then a coat of suitable "hard-facing" material is applied. The manganese steel teeth can be built up by welding or repointed by using a diamond-shaped applicator bar to replace the lost material.

Figure 28 illustrates the use of the alloy steel applicator bar. The upper part of the figure illustrates one method of applying the stringer beads of long-wearing metal. This pattern is used when digging in blasted rock. There is light wear between the beads but this seldom exceeds 1/8 of an inch. After the original stringers begin to wear appreciably, additional beads are run between them.

When small material is being handled, the criss-cross pattern to the stringer beads is the best to use. The dirt will fill up the small depressions and remain there. Only the stringer will wear.

Bucket life can be increased by from three to five times by facing all wearing parts with stringer beads. Stringer beads have several advantages over the use of solid deposits of weld rod. Stringer beads are more economical in material, they are applied with greater speed, and the use of beads also eliminates the danger of overheating manganese castings and thus destroying grain structure.

It is a good practice to build up the cutting edge of the dipper tooth to a slightly greater width than the rest of the tooth as shown in Figure 29. This aids in preventing rocks from becoming wedged between the teeth. Rocks wedged between the teeth increase the digging resistance. To build up the teeth in the manner illustrated will aid in decreasing wear on the dipper itself.

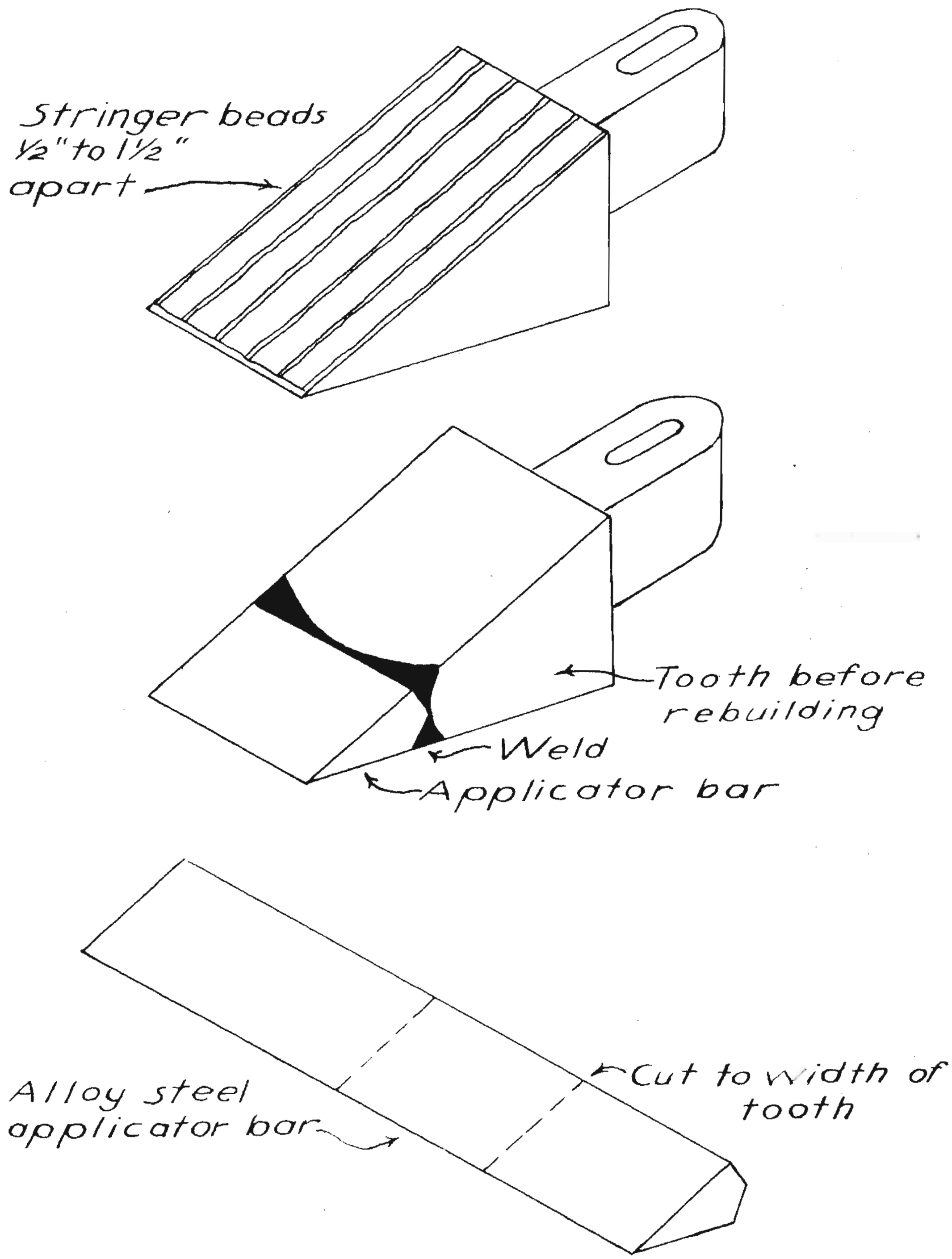


Figure 28. A method of building up dipper tooth.

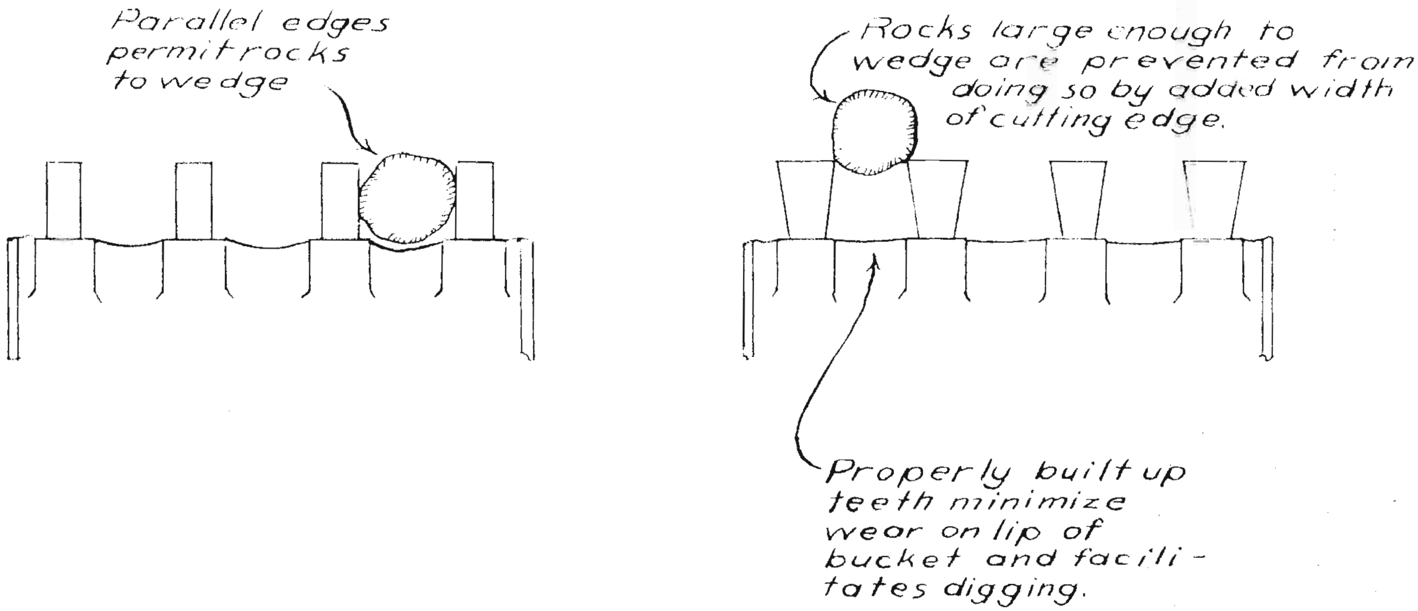


Figure 29. The elimination of rocks wedging between teeth.

The shank of the tooth as well as the cutting edge should receive attention. If the shank fits loosely in the socket, the socket will continue to wear larger.

All parts of the shovel should be checked periodically. This check-up should include the power-plant, all moving parts, take-ups on the cats, drum lagging, all bolts and nuts, and all pins.

One of the biggest factors in influencing shovel capacity is the competence and skill of the operator. Not only is the operator a big factor in the shovel output, but also a big factor in shovel life and maintenance costs. An operator who jerks a machine may appear fast on casual inspection, but a stop watch soon reveals the fallacy of this assumption. A survey of repair records soon shows the difference in those costs. The good operator blends all movements into one smooth cycle. He makes no unnecessary movements of his hands and feet or of the machine.

There are many little things that an operator can do during short haulage delays to increase shovel capacity and efficiency. Among the things that the operator can do are: move the shovel closer to the face, clean the pit floor ahead of the shovel, loosen the bank by passing the open dipper through it, move large boulders aside, load the dipper from the far end of the swing, and place the loaded dipper over the dumping spot to aid the drivers spot their trucks when backing into the shovel. During longer delays the operator should check and service the machine and make necessary adjustments.

The good operator respects the limitations of the machine.

He does not swing until the dipper is clear of the bank. He does not use the dipper as a broom. He gets a full dipper load in one pass through the bank. If the material being excavated sticks to the car bottom or truck bed, an operator may help by placing a dipper full of dry earth in the car or truck first. The same thing may help save time in cold weather when material freezes to the car or truck bed. Operators should be trained to pay attention to these and similar factors.

It is desirable to be able to anticipate the capacity of a shovel or any other piece of machinery to be used. The variables affecting shovel output are the type and size of the shovel, the type and amount of power available, bailing speed and power, swing speed, travel speed, crowd speed, bucket size, type of material being dug, height of bank, degree of bank preparation, weather, coordination with haul units, type of footing, operator efficiency and competence, average angle of swing and the dumping height. From a study of the nature and the number of these variables, it can be appreciated that it is difficult to accurately predict shovel performance. There are, however, several methods of closely approximating the performance.

The most accurate method of forecasting shovel performance would be that method based on accurate records of the performance of like equipment operating under conditions the same as those anticipated. Accurate production records are valuable not only in anticipating future performance but also in estimating costs of production. Besides this there are several methods of closely approximating the performance.

The movements that enter into the shovel cycle are hoisting and lowering the dipper, crowding, racking in, swinging and dumping. A study of the shovel in action will show that the hoisting is being done while digging and sometimes while swinging to the proper dumping height. The crowding is likewise done while digging and while swinging. The dipper is normally stationary while dumping. The lowering and racking-in is done on the return swing. When these motions are properly blended by a good operator, the shovel cycle can be considered to be made up of digging time, swinging time and dumping time.

One method of finding the shovel cycle is based on time studies of digging and dumping time and calculation of the swing time from the rated swing speed of the shovel and the average angle of swing. Combining these factors will give the theoretical cycle. The theoretical cycle combined with the bucket capacity will give the theoretical capacity. The actual capacity is then the swell factor of the material times 65 percent efficiency times the theoretical capacity. This drastic efficiency factor is necessary because no allowance has been made for acceleration and deceleration of swing, because a machine seldom, if ever, operates at maximum efficiency for long periods, and to allow for delays such as moving into the bank.

An example[10] of the results of a time study based on several

(10) Thoenen, J. R., Sand and gravel excavation, Part 1, The power shovel, the dragline excavator, and the excavator crane: U.S. Bur. Mines, Information Circular 6798, p. 19, 1934.

thousand observations under various working conditions in sand and gravel for 3/8 to $1\frac{1}{2}$ cubic-yard shovels follows:

	Load Time (sec.)	Dump Time (sec.)	Combined (sec.)
Minimum average	8.9	2.1	11.0
Maximum average	13.6	4.4	18.0
		Average	14.5

The results of other time studies in quarry loading are given in Table IV. The shovels which these studies cover vary in size from $1\frac{1}{2}$ cu. yd. through 4 cu. yds. The shovels were powered by steam, electricity, or by Diesel engines. The material being loaded was trap, granite, limestone and graywacke. A study of the loading and dumping times given here shows that there is not an appreciable difference between blasted rock and sand and gravel. The dumping times given here are not representative of what they would be in some types of wet earths and clays. In wet earths and clays the dumping time is often much greater because of the clay hanging up in the dipper necessitating jerking of the dipper to swing the door to jar the load out of the dipper.

The swing time may be calculated by using equation (8) and the rated rotating speed as given by the manufacturers. There is a slight difference in swing and return time because of the difference in acceleration time with and without a load. Sometimes the return swing time might be greater than the swing time. This happens on machines that do not drop the dipper by gravity. On the return the dipper must be dropped to the toe of the cut

TABLE IV[11]

TIME DISTRIBUTION IN COMPLETE SHOVEL CYCLES, SECONDS

Quarry	Load	Swing	Dump	Return	Total
1	9.925	6.895	3.043	7.430	27.29
2	5.930	5.790	2.438	4.897	19.06
3	8.840	6.418	2.634	6.987	24.88
4	9.685	5.960	2.876	7.085	25.61
5	9.200	7.740	4.975	8.140	30.06
6	10.685	5.623	3.110	5.740	25.16
7	10.155	7.630	3.070	8.070	28.93
8	13.780	6.575	3.153	7.410	30.92
9	9.720	5.807	2.578	6.493	24.59
10a	12.475	8.400	3.102	9.350	33.33
10b	11.945	7.050	2.514	7.105	28.61
13a	15.980	8.547	3.365	10.290	38.18
13b	10.580	5.645	2.753	6.520	25.50
14	8.113	7.870	2.940	8.237	27.16
15	9.280	5.628	2.123	5.755	22.79
16	9.680	6.372	1.765	6.013	23.86
17a	13.340	7.020	2.228	8.050	30.66
17b	14.570	7.500	2.732	10.310	35.11
18	11.210	6.907	2.895	8.682	29.70
19	8.500	5.800	2.110	7.100	23.51
20	10.445	9.050	3.265	10.210	32.97
21	6.800	4.908	2.794	4.798	19.30

(11) Thoenen, J. R., and Lintner, E. J., Time study analyses, Progress report 1, Quarry shovel loading; U.S. Bureau of Mines Report of Investigations 3461, p. 6, 1939.

whereas on the loaded swing the dipper is swung from the top of the cut. The lowering time for the dipper might be greater than the return swing time. Regardless of which way the difference is, it is almost negligible since the 65 percent efficiency factor will adjust for it.

The theoretical cycle time of the shovel equals the swing time as computed in equation (8) plus the loading time plus the dumping time. The theoretical capacity then becomes

$$\frac{(3,600 \text{ secs})(\text{dipper capacity in cu. yds.})}{\text{Theoretical cycle time in secs.}} \qquad (9)$$

The actual capacity in cubic yards per hour then becomes

$$(.65)\ (\text{swell factor})\ (\text{theoretical capacity}) \qquad (10)$$

The following example will illustrate the use of this method.

Example 4.

A shovel with a 2-cu. yd. dipper is to be used in a sand and gravel pit. The rated swing speed of the shovel is 3 rev. per minute. The work is to be laid out so that the average angle of swing will be 95 degrees. What will be the shovel's capacity?

The swing time (using equation (8)) is:

$$\frac{(2)(95)(60)}{(360)(3)} = 10.6 \text{ seconds}$$

The combined loading and dumping time is 14.5 seconds. The theoretical cycle time is:

$$10.6 \text{ secs.} + 14.5 \text{ secs.} = 25.1 \text{ secs.}$$

Using equation (9), the theoretical capacity is found to be

$$\frac{(3,600 \text{ secs.})(2 \text{ cu. yds.})}{(25.1 \text{ secs.})} = 287 \text{ cu. yds. per hour.}$$

The swell factor (from Table II) is .89. The actual capacity becomes

$$(.89)(.65)(287 \text{ cu. yds/hr.}) = 166 \text{ cu. yds/hr.}$$

This system does not consider some of the important variables such as the height of bank, which influences the time spent moving and the time spent in digging.

If the shovel is casting, omit the dumping time from the theoretical cycle and solve as before.

A more comprehensive system is that given by Holcomb.[12] In this system, coefficients have been established to modify each of the several variables which influence shovel capacity. These multipliers, or coefficients, are given in Tables V, VI, and VII. The multipliers are based on the following shovel speeds:

Swinging . . . . 4 rev. per min.

Hoisting (2-part line) . . . . 85 ft. per min.

Crowding . . . . 130 ft. per min.

Racking-in . . . . 180 ft. per min.

The basis on which the calculations are predicated is that an average operator on the above shovel with a 1-cu. yd. dipper is ex-

(12) Holcomb, A. E., Output factors for excavation and material-handling, Civil Engineering, Vol. 1, No. 1, pp. 26-30, October 1930.

TABLE V [13]

MULTIPLIERS OR OUTPUT COEFFICIENTS FOR SHOVELS

Material	Multiplier
Hard shale and other rocky formations poorly blasted	0.40
Fairly well-blasted rock or hard-pan, and tough, rubbery clay.	0.50
Clay boulders	0.60
Heavy clay (not sticky)	0.70
Clay gravel	0.80
Wet, sandy clay	0.90
Ordinary earth	1.00
Light, dry loam or clay, loose sand and gravel, cinders, or ashes	1.10
Light, moist clay and loam	1.25
Size of Dipper	
3/8 cu. yd.	0.38
1/2 cu. yd.	0.50
5/8 cu. yd.	0.63
3/4 cu. yd.	0.75
1 cu. yd.	1.00
1¼ cu. yd	1.25
1½ cu. yd	1.50
1-3/4 cu. yd.	1.75
2 cu. yd.	2.00

(13) Holcomb, op. cit., p. 29

TABLE VI (14)

MULTIPLIERS OR OUTPUT COEFFICIENTS FOR SHOVELS

Multipliers

Depth of Cut	Size of Dippers Cubic Yards								
	3/8	1/2	5/8	3/4	1	1¼	1½	1-3/4	2
1'-0"	.67	.66	.65	.64	.63	.62	.59	.56	.52
1'-6"	.76	.75	.74	.73	.72	.72	.69	.66	.62
2'-0"	.83	.82	.81	.81	.80	.80	.77	.74	.70
3'-0"	.89	.88	.87	.86	.85	.85	.82	.79	.75
4'-0"	.94	.93	.92	.91	.90	.89	.86	.83	.80
5'-0"	.97	.96	.95	.94	.93	.93	.91	.88	.85
6'-0"	1.00	1.00	.98	.98	.97	.96	.94	.91	.88
7'-0"	.97	.97	1.00	1.00	.98	.98	.96	.93	.90
8'-0"	.94	.94	.97	.97	1.00	1.00	.98	.96	.93
9'-0"	.91	.91	.94	.94	.97	.97	1.00	.98	.96
10'-0"	.88	.88	.91	.91	.94	.94	.97	1.00	.98
11'-0"	.85	.85	.88	.88	.91	.91	.94	.97	1.00
12'-0"	.82	.82	.85	.85	.88	.88	.91	.94	.97
13'-0"	.79	.79	.82	.82	.85	.85	.88	.91	.94
14'-0"	.76	.76	.79	.79	.82	.82	.85	.88	.91
15'-0"	.74	.74	.76	.76	.79	.79	.82	.85	.88
16'-0"	.72	.72	.74	.74	.76	.76	.79	.82	.85
18'-0"	.70	.70	.72	.72	.74	.74	.76	.79	.82
20'-0"			.70	.70	.72	.72	.74	.76	.79
22'-0"					.70	.70	.72	.74	.76
24'-0"							.70	.72	.74
26'-0"								.70	.72

(14) Holcomb, *op. cit.*, p. 29

TABLE VII (15)

MULTIPLIERS OR OUTPUT COEFFICIENTS FOR SHOVELS

<u>Type of Operation</u>	
Side casting	1.25
Loading trucks in rear, 180 degree swing from cut.	0.80
<u>Trimming Slopes</u>*	
For 1:1 slope	0.60
For 1.5:1 slope	0.40

* Where slopes have to be trimmed when computing yardage in cut, take for purposes of estimate 1 ft. 0 inches of material over slopes as separate yardage. Then, for this yardage, use slope multipliers.

(15) Holcomb, op. cit., p. 29.

cavating ordinary earth from an 8-foot cut and loading trucks or cars through an average swing of 90 degrees. Under these conditions the operator should be able to take out an average of 120 cu. yds. per hour, place measurement.

A shovel mounted on crawlers usually loses about 15 seconds in moving up. This has been considered in establishing the original hourly yardage. If mats must be used when moving, the time lost will be approximately 1 minute. The effect on the yardage output of this extra moving time will depend on the ratio of the excavating time to the sum of the excavating time plus the moving time.

The examples which follow illustrate the use of these multipliers.

Example 5.

A shovel with a 2-cu. yd. dipper is to be used to load fairly well-blasted rock from a 13-ft. bank and load into trucks at the rear. What will be the hourly average production?

From Table V, the coefficient for fairly well-blasted rock is 0.50.

From Table V (lower part), the multiplier for a 2-cu. yd. dipper is 2.00

From Table VI, the coefficient for a 13-ft. bank and a 2-cu. yd. shovel is 0.94.

From Table VII, the coefficient for loading to the rear is 0.80.

Combining the above multipliers gives the output coefficient, as follows:

$$(0.50)\ (2.00)(0.94)(0.80) = .752.$$

The hourly output then becomes

(.752)(120 cu yd. per hr.) = 90.24 cu. yds. per hr., place measurement.

The effect of a decreased swing angle can be nicely shown in this example. Had the shovel been loading trucks to the side, the output coefficient would have been

$$(2.00)(0.94)(0.50) = .94.$$

The output then would have been

(.94)(120 cu.yd. per hr.) = 112.8 cu. yds. per hour, place measurement. This is a 25 percent increase in shovel production.

Example 6.

A 1-3/4 cu. yd. shovel is to be used to load trucks to the side from a 7-ft. bank of clay gravel. The shovel must move on mats, using 1 minute per move. The shovel will be able to excavate 30 yards between moves. What hourly production can be anticipated?

From Table V (upper), the coefficient for the type of material is 0.80.

From Table V (lower), the coefficient for the dipper size is 1.75.

From Table VI, the coefficient for the bank height and shovel size is 0.93.

Neglecting the extra moving time, the output coefficient is

$$(0.80)(1.75)(0.93) = 1.302.$$

Under these conditions the hourly output would be (1.302)(120 cu. yds. per hour) = 156.24 cu. yds. per hour.

The time needed to excavate 30 cu. yds. will be

$$\frac{(30)(60 \text{ min.})}{(156.24)} = 11.4 \text{ min.}$$

Allowing 1 minute total moving time on mats minus 15 seconds, already allowed in the original yardage assumption, will give

$$\frac{11.4}{11.4 + (1-0.25)} = .938$$

as the job multiplier because of having to move on mats.

The shovel capacity then becomes

(.938)(1.302)(120 cu. yds. per hr.) = 144.6 cu. yds. per hr.

Mr. Holcomb further suggests that each organization determine its individual organization coefficient. This coefficient will depend upon such things as the overall management of the work; the placing and handling of equipment; maintenance of equipment; the coordination of equipment; and the coordination of the several departments.

Management should, as a matter of policy, keep records of production from all machines to use as a basis for cost and as a basis for future estimates.

It must be remembered, that whatever the system used to predict performance, the results will be long-time averages. Thus, in example 6, one cannot expect the shovel to load out exactly 144.6 cu. yds. per hour every hour. Barring any protracted unforeseen delays, the machine will average very nearly this amount over a period of several weeks or a month when working under the stipulated conditions.

Most shovel makers have devised "slide rules" or tables by means of which the performance of their machines can be estimated. The system devised by Mr. Holcomb is used by the Koering Company.

It is perhaps the most complete and accurate in that it considers a greater number of the variables than do most slide rules.

In choosing a shovel it is a good practice to pick a machine capable of delivering from 15 to 20 percent more than its scheduled production. This is desirable to insure that the shovel will not be operating at its absolute peak at all times. Further, a shovel with some overcapacity will aid in "catching" up should there be a serious delay.

There are many sources of delay and impaired production over which the shovel operator does not have control. A study of Table VIII will show the source and magnitude of the delay. These time studies were made in stone quarries under a variety of operating conditions.

In this classification of delay factors, moving boulders was the time spent is moving large boulders to the side for later secondary blasting. Large boulders are those which are too large for the shovel dipper, haulage equipment, or crusher to handler. The number of large boulders is determined by the effectiveness of the primary blasting.

Moving shovel covered the time lost in advancing the shovel to keep it within working range of the face. In this study this also covered the time required to move the shovel from and return it to the face at the end and beginning of each shift or while blasting. The height of the quarry face and the blasting schedule determine the amount of time in this factor.

Under the heading of shovel delay is included the time lost

TABLE VIII [16]

TIME CONSUMED BY DELAY FACTORS IN PERCENT OF TOTAL TEST PERIOD

Quarry Number	Moving Boulders	Moving Shovel	Shovel Delays	Quarry Delay	Haul-age	Total Delays	Loading Time
1	1.1	0.9	4.9	—	35.3	42.2	57.8
2	2.2	—	—	—	56.6	58.8	41.2
3	2.5	9.5	9.4	4.3	22.7	48.4	51.6
4	—	1.0	5.8	7.6	40.2	54.6	45.4
5	1.9	4.4	2.6	3.5	48.2	60.6	39.4
6	4.5	1.4	5.4	3.8	26.8	41.9	58.1
7	12.3	6.7	3.7	5.6	17.7	46.0	54.0
8	4.4	5.0	6.7	3.0	13.8	32.9	67.1
9	5.2	6.1	4.9	2.6	1 7.4	36.2	63.8
10a	7.4	7.2	2.8	2.1	17.2	36.7	63.3
10b	12.2	9.9	1.6	2.7	15.9	42.3	57.7
13a	3.1	1.9	20.7	7.8	24.9	58.4	41.6
13b	6.0	2.2	6.6	8.2	30.4	53.4	46.6
14	7.1	2.4	—	2.9	53.4	65.8	34.2
15	6.0	2.2	—	6.7	51.9	66.8	33.2
16	4.1	2.2	.5	8.3	40.1	55.2	44.8
17a	4.2	1.3	4.8	4.9	41.8	57.0	43.0
17b	7.0	2.6	6.7	6.7	24.4	47.4	52.6
18	11.5	3.1	3.6	16.3	20.7	55.2	44.8
19	1.1	6.7	4.5	12.8	34.4	59.5	40.5
20	6.3	2.3	17.1	2.4	36.0	64.1	35.9
21	3.3	1.6	5.1	1.7	44.2	55.9	44.1

(16) Thoenen, J. R. and Lintner, E. J., Time study analyses, Progress report 1, Quarry shovel loading: U. S. Bureau Mines, Report of Investigations No. 3461, p. 10, 1939.

directly because of the shovel itself. Making up this figure is the time out for oiling, repairs, relocation of power line, fueling, and time lost by the operator. This figure might be decreased by scheduling service time to periods when the shovel is inactive. Preventative maintenance will cut time lost by breakdowns.

Quarry delay is the time lost because the shovel is engaged in cleaning up scattered rock following a blast, and the time lost while the shovel is waiting for secondary blasting of oversize boulders, while the shovel dipper is being used to dislodge large boulders in the rock pile that interfere with continuous loading, and while the face of the quarry is being trimmed and made safe. This factor can be reduced by improving the primary blasting and by using a bulldozer for cleaning up the quarry floor.

Under haulage is grouped the times when the shovel is idle because of the lack of cars or trucks to load. Included here is the time when the shovel waits between finishing the load on one truck until the next is ready for loading. In this study in several cases the delays due to repair or shifting of tracks or clearing of truck roadways are included, as are delays due to break-down in the haulage system.

In this study there were two instances of crusher delays. Loading was held up due to necessary crusher repairs. These two instances are not noted in the Table.

In mining operations plant delays almost always entail pit delays. Plant delays might be caused by power failure, crusher breakdowns, screen repairs, lack of cars or adequate storage for finished product, conveyor break-downs and similar causes. These are reflected in the pit or quarry since most operations lack adequate surge capacity between the plant and the pit.

Whatever the cause, this time study and others show startling total time losses in shovel operations. A study of the causes show that reductions can be made in the delays through better planning, bank preparation, coordination of equipment, better roads, and more surge capacity.

Further economy and greater efficiency can be attained through standardization of equipment. If more than one shovel is to be used, it is a good practice to have shovels of the same type and size. This cuts down the parts inventory which must be carried. It also makes for greater operator efficiency should it become necessary to switch operators from machine to machine. Servicing and preventative maintenance can be systemitized. Shovels of the same type and size also give more accurate records on which to base comparisons of production and on which to predict performance.

THE DRAGLINE

The crawler-mounted dragline resembles the power shovel in all respects except the front end. The dragline has a longer boom and it does not have a dipper stick. In the smaller sizes the two machines are readily interchangeable. Generally the size range and power plants of draglines correspond with those of the power shovel. In the larger sizes of draglines there is a difference, however, in the mode of locomotion. Most large draglines are walking machines. Draglines are built up through a 35 cu. yd. maximum size.

The maximum range of operation of the dragline is controlled by the design, size, power, boom length and angle, and the size of the hoist and drag cable drums. Figure 30 illustrates the method used by manufacturers in designating the working dimensions of the dragline. The effect of boom length and angle is illustrated by Figure 31. It should be noted that the dragline can work beyond the end of the boom. This distance, "F" in Figure 30, is from 1/3 to 1/2 the dumping height depending on the skill of the operator. The dragline can also cast material beyond the end of the boom as illustrated in Figure 32. This is generally considered to be a poor practice since the bucket begins to swing and time is lost in taking the slack from the drag cable. Cable wear is also increased by this practice.

Generally, size for size, the dragline has a greater working range both vertically and horizontally than a power shovel. The

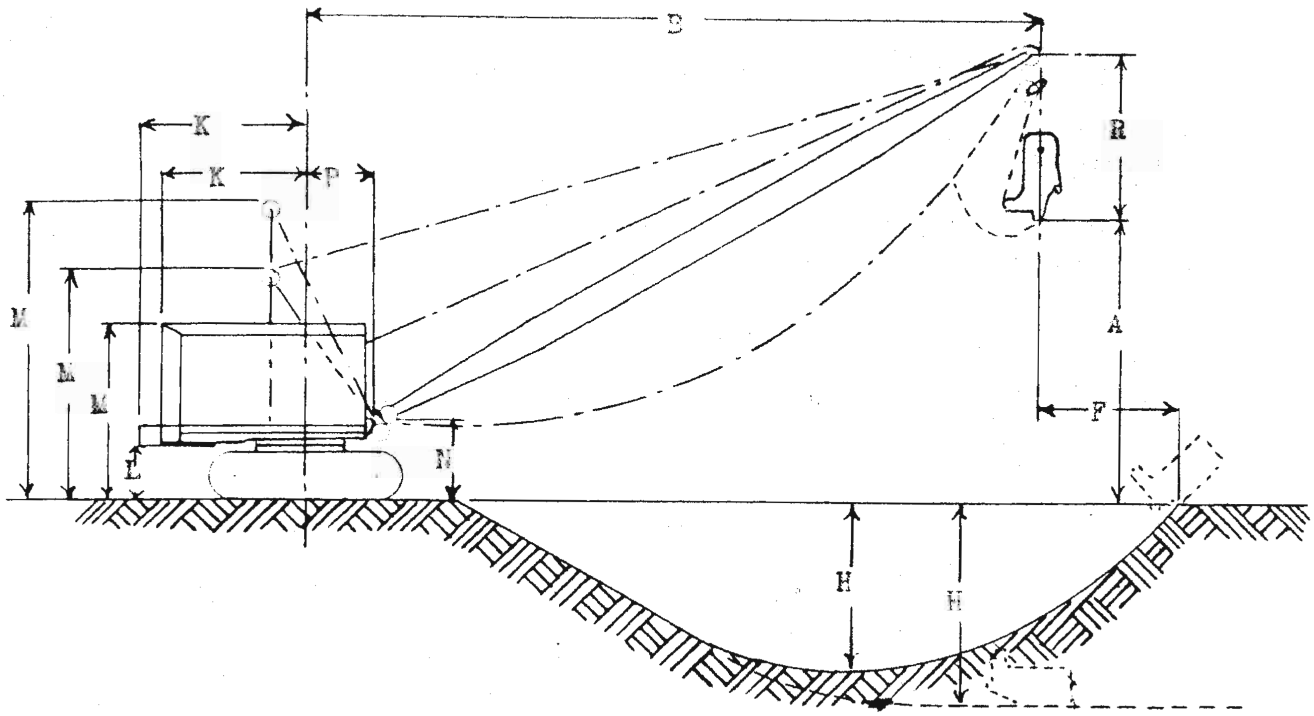


Figure 30.

Dragline Working Range Dimensions

A --Dumping Height (Height Boom Point Sheave Pin Minus R)....

B --Dumping Reach (Operating Radius)........................

F --Throw of Bucket 1/3 to 1/2 Dumping Height (A) Depending on Operator..

H --Digging Depth...

K --Clearance Radius of Revolving Frame (Standard shovel counterweight).......................................12'-9"

K --Clearance Radius of Revolving Frame with Counterweight C, D & E...15'-3"

L --Clearance under Frame to Ground level.................. 3'-9"

M --Clearance Height--High A Frame.........................25'-1"

N --Height of Boom Foot Pin Above Ground Level.............. 6'3½"

P --Distance Boom Foot Pin to Center of Rotation............ 5'-3"

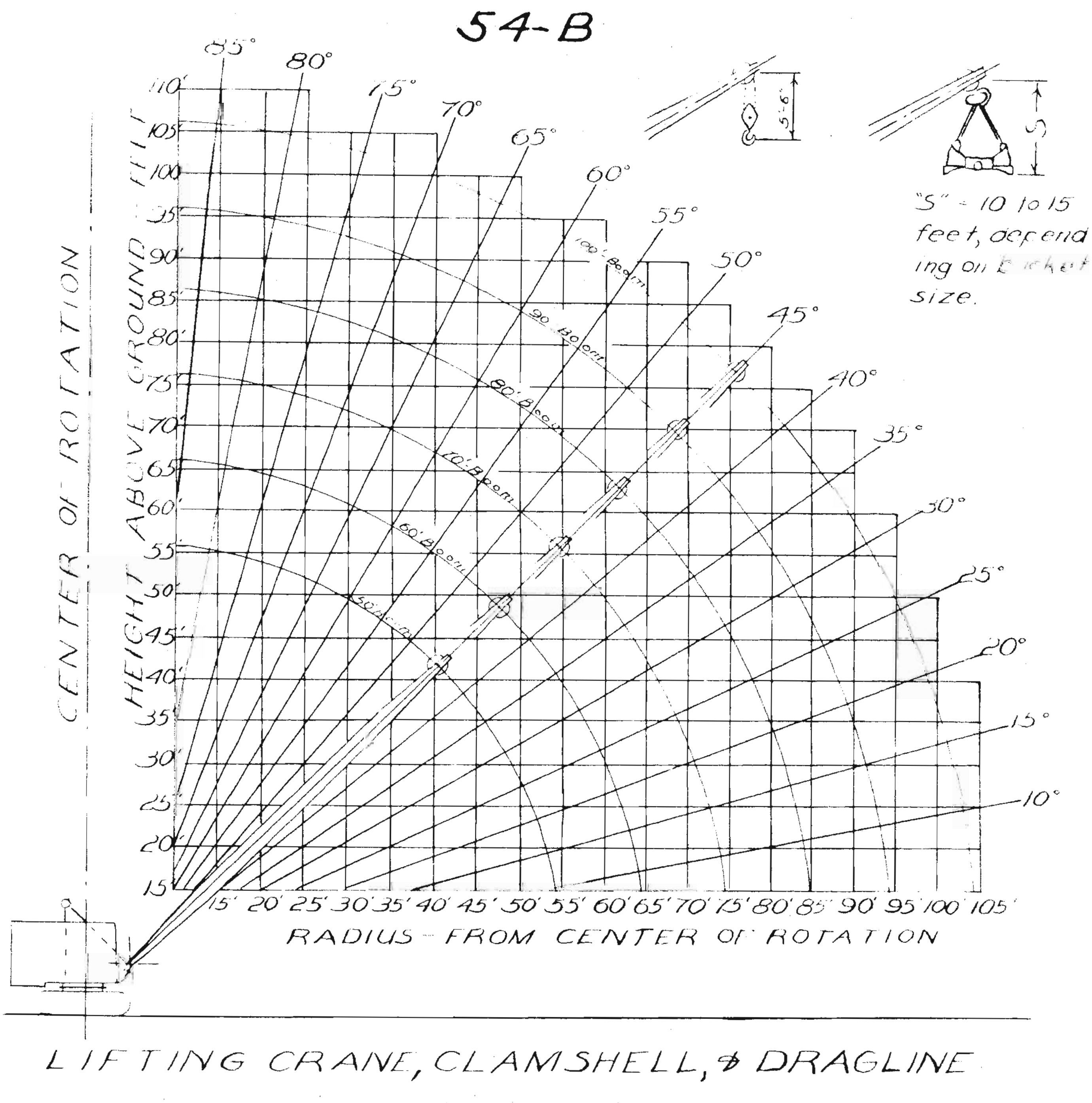


Figure 31. Effect of boom length and angle on working dimensions of dragline.

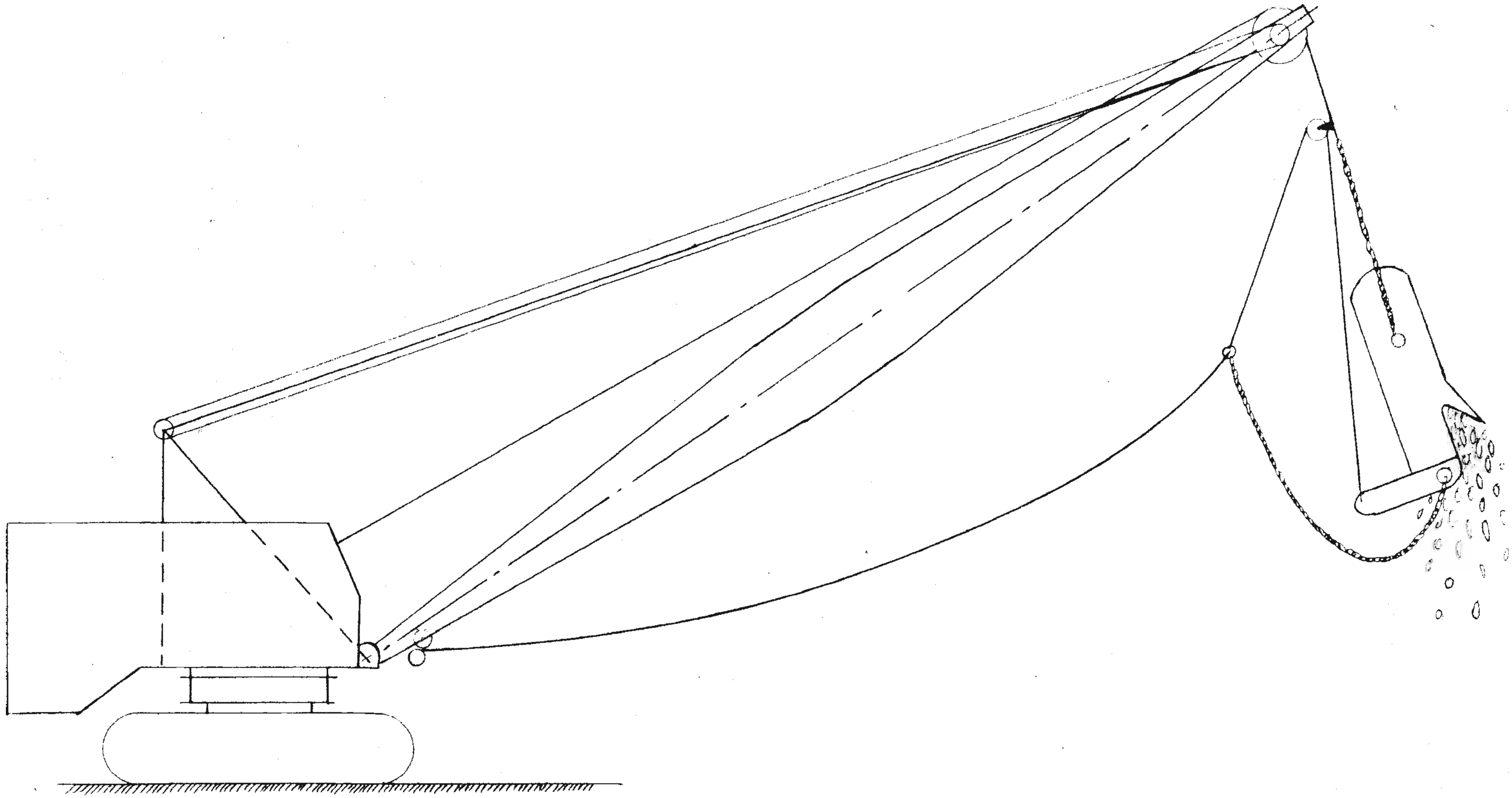


Figure 32. Casting beyond the end of the boom.

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depth to which the dragline can dig is limited by the angle of repose of the material and by the length and angle of the boom and by the amount of hoist and drag cable that can fit the drums. Figure 30 illustrates how the angle of repose of the material can limit digging depth. If the angle of repose of the material on which the dragline is working is low and if the depth "H" becomes too great, the caving bank will undermine the crawlers.

There is one very important difference between the shovel and the dragline which greatly affects the efficiency and applicability of the two machines. The shovel is so constructed that it can apply a positive force behind the bucket when digging. The dragline can only drag the bucket toward the machine and must depend on the weight of bucket and the design of the bucket for penetration.

Bucket design is an important variable in dragline performance. Bucket efficiency depends on the ratios between weight, width, and length. Efficiency is affected by the angle of the teeth, balance about the hoist trunion, position of the dump sheave, length of dump rope, position of loading clevis pin, length of loading chain, length of hoist chain, and almost every other feature of the bucket.

Because of lack of positive control of the bucket, the dragline is less suited to loading into haul units than is the power shovel. Large draglines are seldom, if ever, used to load haul units. Trucks provide a small target for the free swinging dragline buckets. The small draglines, however, are often used to load small trucks. Where excavating conditions indicate that the dragline is the better machine of the two to use, or if the dragline is the only machine available, it is used to load haul units though it is not quite as applicable as the power shovel to loading haul units.

When the dragline is used to load haul units, the ideal conditions of loading are as illustrated in Figure 33. Here the dragline excavates a strip parallel to, and on the same level as, the truck. The dragline then swings through a very small arc to bring the bucket over the truck box. The truck should be spotted under the end of the boom so that there will be some tension in the drag rope giving better control of the bucket. If the truck were in too close to the machine, the dragline would have to take up more on the drag cable. As the bucket was dumped it would swing out too far and batter the truck cab or cab guard. If the truck were beyond the end of the boom, the dragline would have to throw the load at the truck box. Figure 33 illustrates the ideal condition of use which is not met with often.

Sometimes it is advantageous to load haul units operating on the bench above the excavator. This may be the case when it is desired to gain elevation rapidly to reduce grades and increase speed. Or again, this may be necessary because the soil on the lower benches is not firm enough to furnish adequate footing and traction for the trucks. This condition of loading is illustrated in Figure 34.

Large draglines are frequently used to dump into hoppers. The hoppers are usually large to provide a good target for the

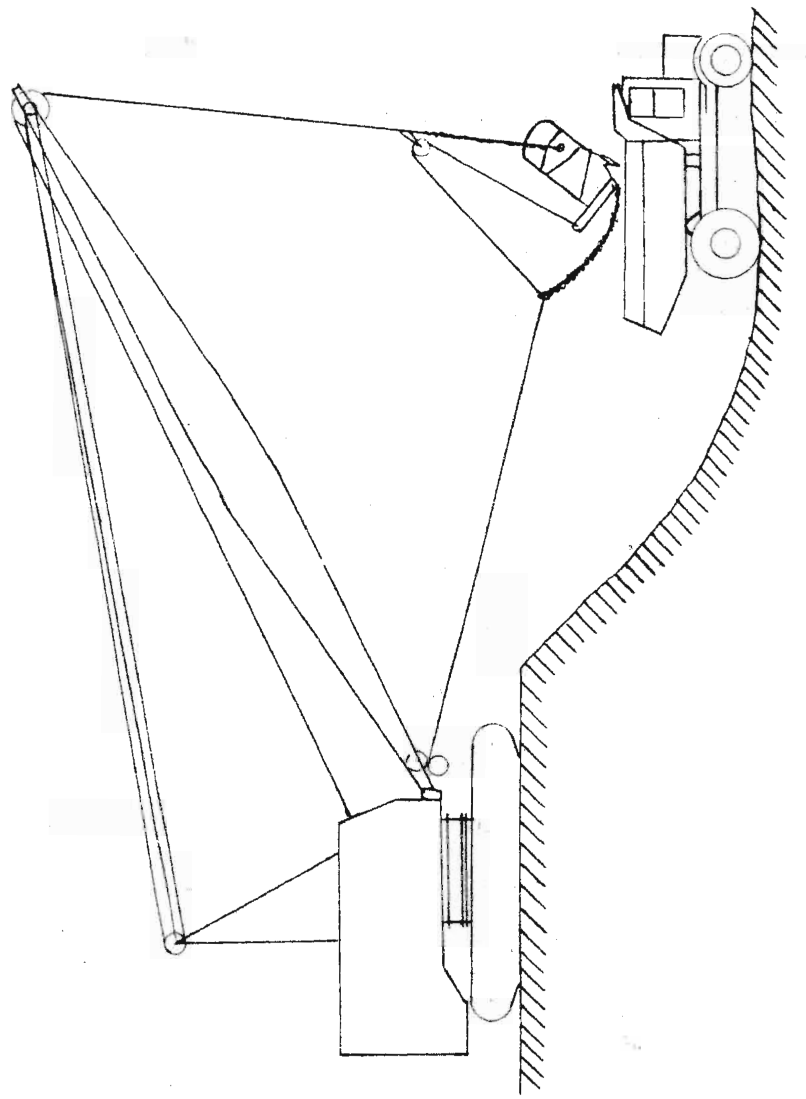

Figure 33. Ideal loading conditions with the dragline.

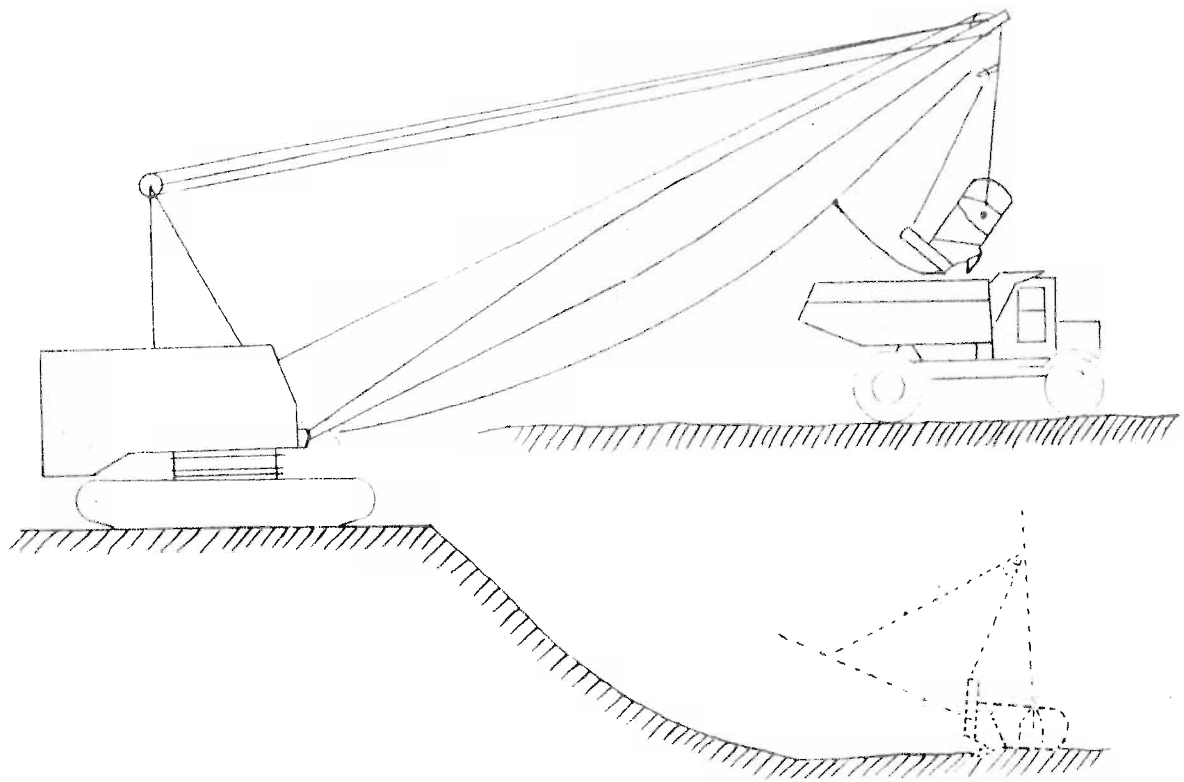

Figure 34. Digging from below dragline and loading to trucks above to gain elevation.

bucket so the operator will not lose time in manipulating the bucket into the dumping position.

A system that is commonly used in sand and gravel deposits is illustrated in Figure 35. Here a dragline loads into a hopper. The hopper feeds the sand and gravel to a portable conveyor belt which portable belt transfers it to a permanent belt and thense to the washer. If the hopper is not too heavy, the dragline can be used to move it. A bulldozer can be used to shift the conveyor.

A variation of the method illustrated in Figure 35 was used by the California Rock and Gravel Company in a pit near Livermore Valley. A 3-cubic yard Bucyrus-Monighan walker with a 90-ft. boom was used to load a hopper. The hopper was equipped with a reciprocating feeder. From the hopper the material was fed to a 200-ft. conveyor section. This conveyor section was portable and could be swung from a pivot point at one end. The dragline excavated material from depths of 50 feet below water level. The normal working depth was 40 feet. The 3-cu. yd. bucket handled about 2½-cu. yd. per trip under water. Production averaged 225 tons per hour. Despite the dirt and water, the drag cables lasted for 45 days and the hoist cables lasted for 120 days.

Another method is illustrated in Figure 36. In this method the dragline loads into a large hopper. Trucks then drive under the hopper and are loaded without delay. This system affords a little surge capacity so that the dragline can continue operating even though there is a haulage delay. The trucks are loaded more rapidly and with less impact than when being loaded by the dragline.

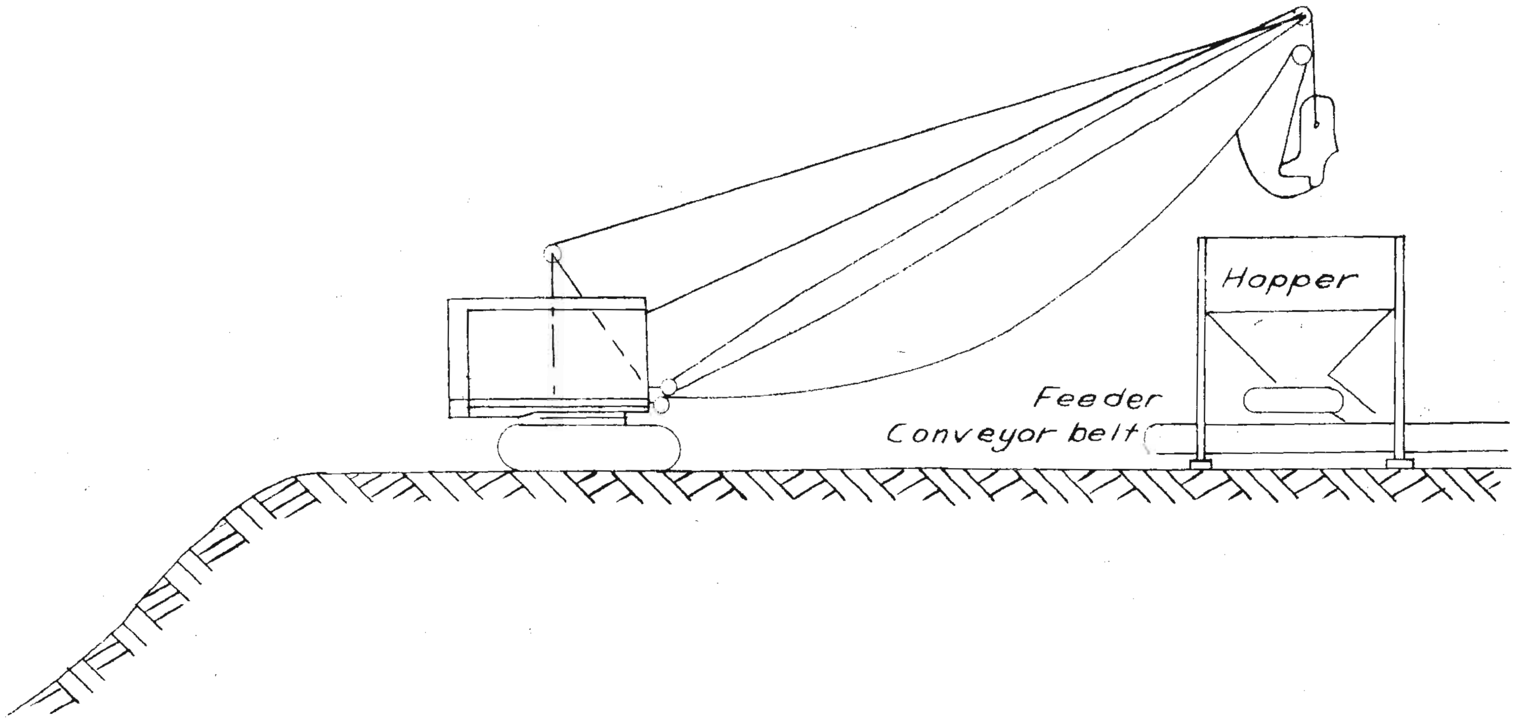

Figure 35. Dragline - hopper - conveyor belt system.

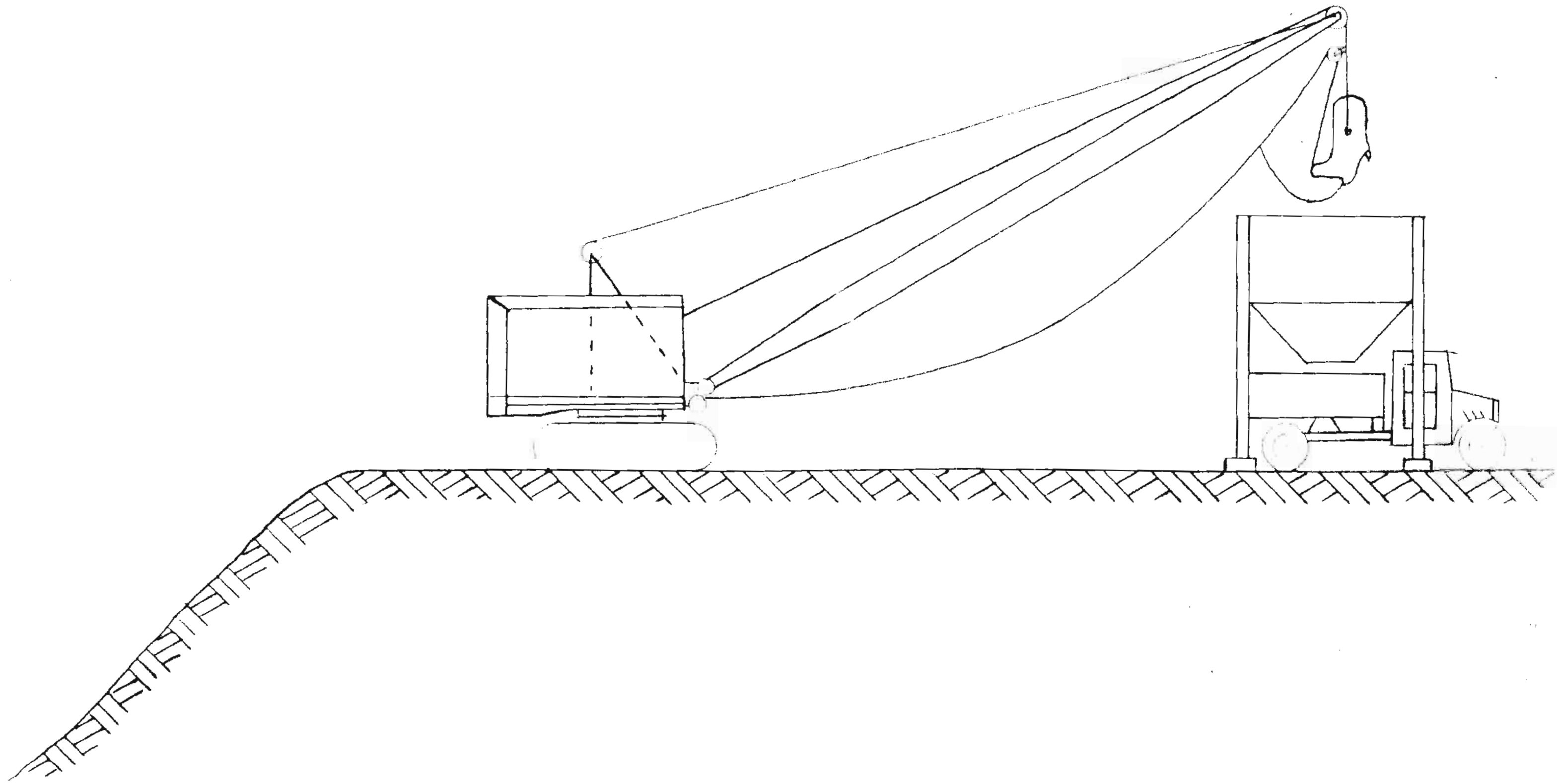

Figure 36. Dragline - hopper - truck system.

The Basalt Rock Company used this last mentioned system at one of their pits on the Russian River at Healdsburg, California. A Bucyrus-Monighan 3-W with a 3½-cu. yd. bucket and an 80-ft. boom loaded gravel into the hopper. The hopper was constructed of steel plate and weighed 20 tons unloaded. The hopper was mounted on pontoons and was moved by dragging with a Euclid truck. The discharge gates were three feet long and two feet wide. The gates rolled on tracks and were operated by compressed air furnished by the compressor on the dragline. The capacity of the hopper was 75 tons. The capacity of the gates which were arranged in series was 5,000 tons per hour.

The use of the hopper cut the truck loading time from three minutes down to 20 seconds. Capacity of the dragline was greatly increased by the elimination of haulage delays.

The average production per shift was 2,300 yds. of material. Digging depth varied between 35 and 40 feet. Working 22 hours out of 24, the 1½-inch preformed drag cables lasted 240 hours. The life of the 1-inch hoist lines varied between 600 and 800 hours. The maximum for the hoist line was 860 hours.

Figure 37 illustrates the latest and largest dragline to hopper system. This system is being used to strip overburden from the South Agnew mine near Hibbing, Minnesota. At this operation a 25 cu-yd. 1150-B Bucyrus-Erie dragline is stripping glacial drift and loading it into a large screen-equipped hopper. The undersize from the hopper is fed to a conveyor belt. The conveyor transports the waste to the stacker on the dump. The over-size is loaded to trucks

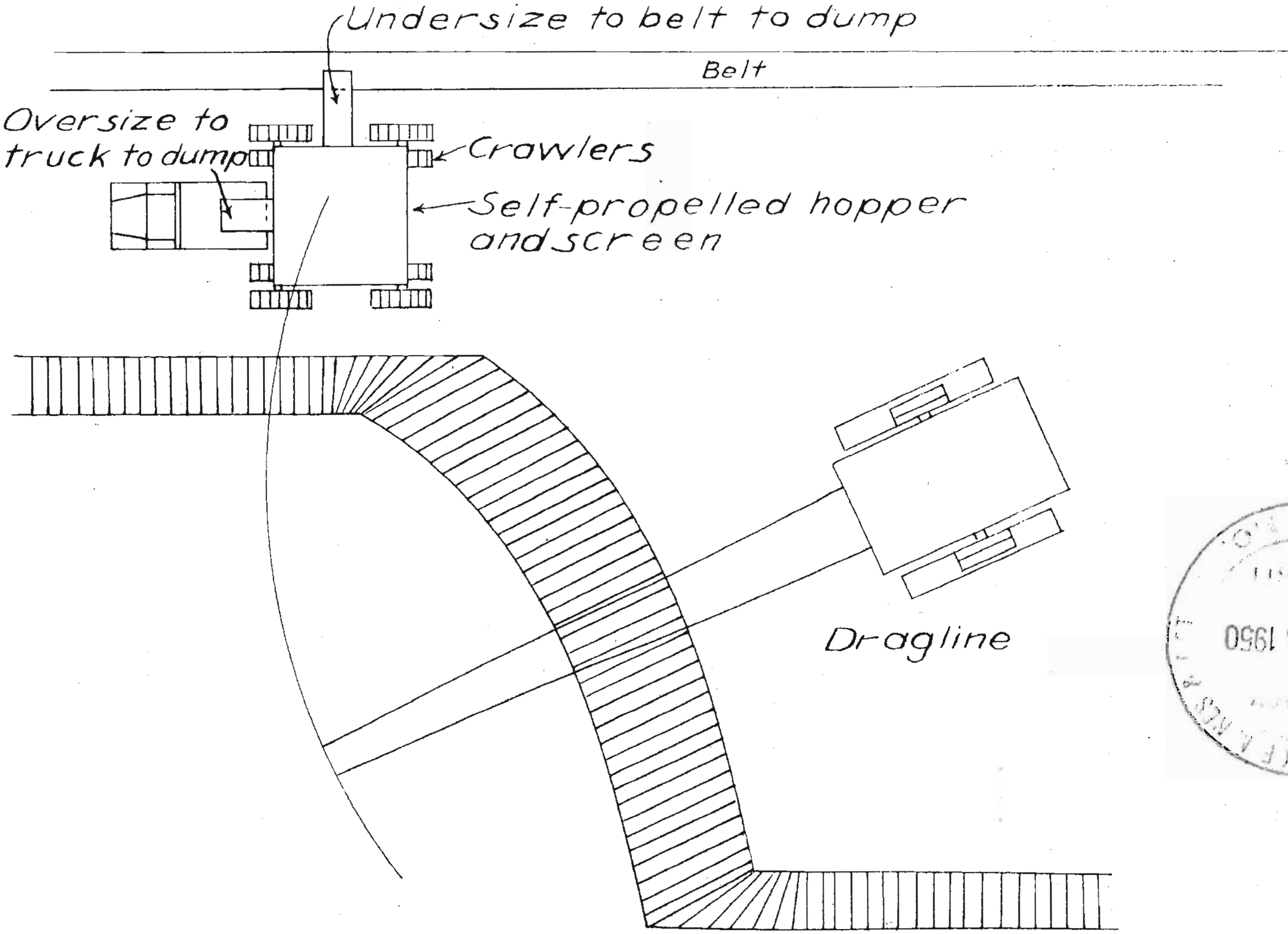

Figure 37. Dragline - self-propelled hopper - conveyor belt system.

which haul to the dump.

The hopper is mounted on a model 550B shovel base which is equipped with hydraulic levelling jacks at each corner. The base is equipped with 4 50-hp. electric motors to make the unit self-propelling. The opening of the hopper is 28 ft. by 30 ft. which dimensions allow the dragline a good target. The sides of the hopper have a 55-deg. slope to a 7 ft. by 18 ft. manganese pan feeder.

The pan feeder feeds a 7-ft. by 12-ft. vibrating screen which operates at 900 rpm. The screen is set at an angle of 20 degrees. The longitudinal skid bars of the screen have an opening of 5 inches at the feed end and 9 inches at the discharge end. The oversize amounts to from 1% to 4% of the total material. It is loaded into trucks through a hydraulically-operated quarter pan gate from the rock hopper into large trucks.

The undersize from the screen falls onto a 60-inch conveyor belt. A boom conveyor 40 ft. long feeds the undersize to a self-propelled, rail-mounted hopper that straddles the movable conveyor. This boom conveyor can be swung through a horizontal arc of 180 degrees and it can also be raised and lowered enough to compensate for minor ground irregularities.

The first movable conveyor discharges to a second movable conveyor. The second movable conveyor discharges to the main conveyor. The main conveyor discharges to a stacker on the waste dump. The movable conveyors are skid-mounted so that they can be moved by caterpillar tractors. The conveyor belts are 48-in. 6 ply 48-oz. duck. The conveyor speed is 550 feet per minute. The conveyor system is designed to handle 1700 cu. yd. per hour. The maximum possible capacity of the dragline is 1700 cu. yd. per hour.

The dragline has a 180-ft. boom giving it a working radius of 170-feet. The width of the cut, however is 230 feet. The overburden is to be removed in two levels.

From the beginning of the full-scale operation on June 1, 1948 through August 31, 1948, 1,595,000 cubic yards of material were removed by this system.

Another dragline to hopper system is illustrated in Figure 36. This system was used by the Universal Placer Mining Corporation at a dry placer mining operation near Santa Fe, New Mexico.

At this operation a Bucyrus-Erie 54-B 1-3/4 cu. yd. dragline excavated a "band" of gravel from 90 feet to 100 feet wide, from 70 feet to 80 feet long and to a depth of 30 feet. The dragline dumped the gravel on an inclined grizzly. The oversized rolled off the lower end of the grizzly. The undersize fell through the grizzly to a hopper below. From the hopper the undersize went to a trommel and thence on through the concentrating plant. The dragline output varied between 100 cu. yds. and 125 cu. yds. per hour.

The lack of positive control over the bucket results in the dragline not being able to dig some of the materials that the power shovel can handle. Banks must be better prepared for dragline excavation than they are for shovel excavation. Even in small boulders the dragline is handicapped at times. When excavating gravel from below water level, the dragline might dig a trough. The larger rocks will roll to bottom of the tough. In time the floor of the

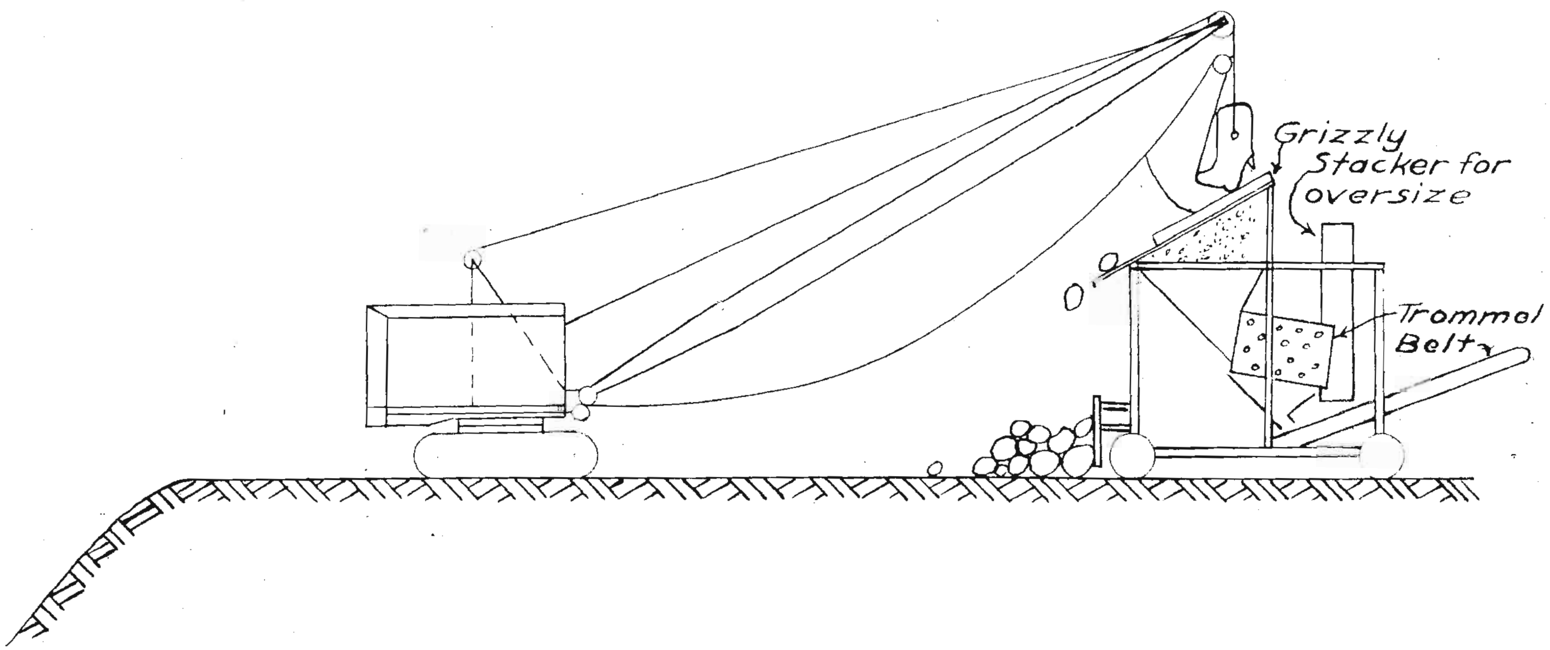

Figure 38. The use of the dragline at a dry placer.

subaqueous pit becomes rock-covered. The dragline bucket will ride over these rocks and thus limit dragline performance.

There is a striking dissimilarity between the ideal digging conditions for the shovel and for the dragline. As mentioned before, the shovel is at its greatest efficiency when working upward from the level of the machine into a well-prepared bank. The shovel advances into the cut as it excavates. The dragline, on the other hand, excavates best from a bank below the level of the machine. The dragline retreats before an end cut or travels parallel to a side-cut. Thus it is possible for a dragline to travel on a firm dry surface and excavate material from below that surface even though that material is below the water level. The dragline can excavate material from under water with the same ease, though with slightly diminished efficiency, that it can material out of the water.

Working as it does from the top surface of the bank which it excavates makes the dragline an ideal machine for many types of excavating and mining. Figure 39 illustrates the use of the dragline in placer mining. Here the dragline works on a dry surface above the water table, or pond level, and excavates from the bedrock upward through the water and casts the gravel into the hopper of the floating concentrator.

This feature of the dragline makes it the ideal machine for mining phosphate rock in Florida. One of the common mining systems used in Florida phosphate is illustrated in Figure 40. Both draglines travel on the surface of the overburden where they are insured proper footing. The water table in the phosphate fields is high and

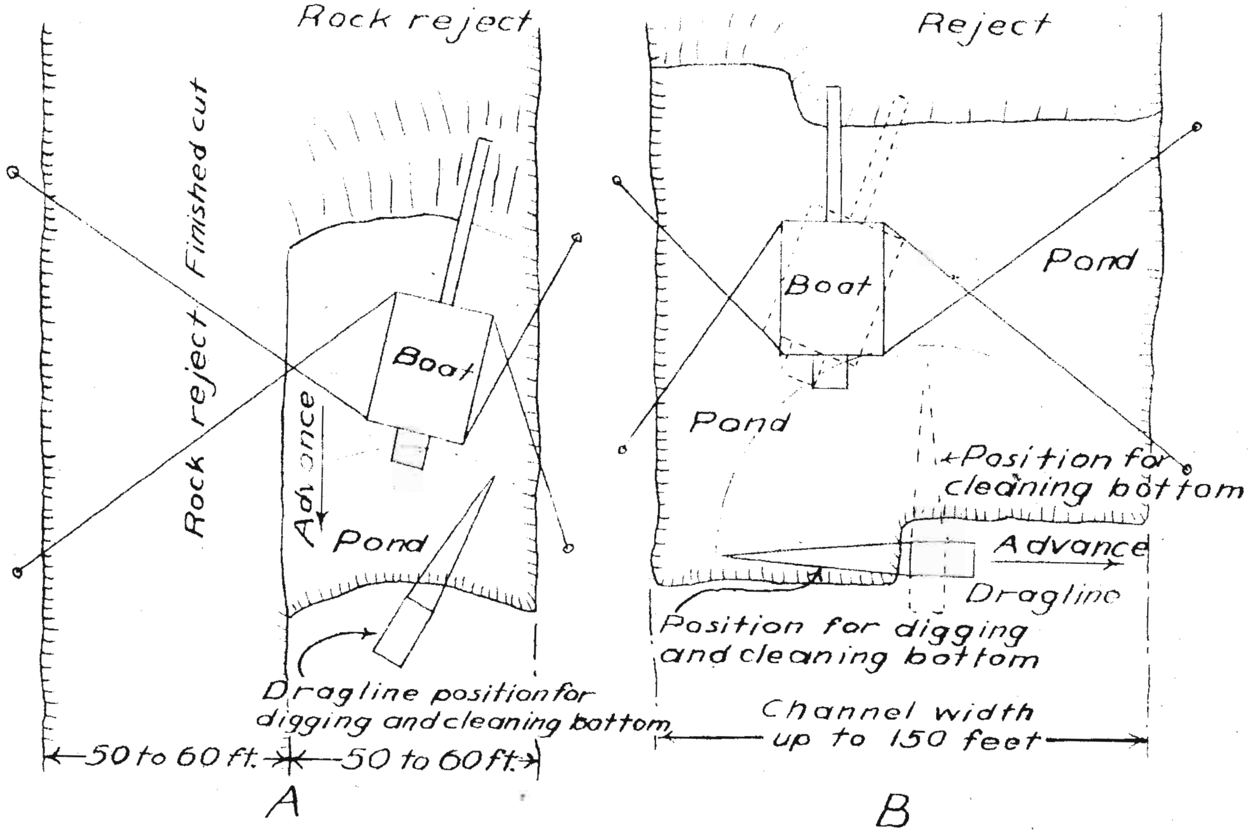

Figure 39.[27] The use of the dragline in gold-dredging.

(27) After Gardner, E. D. and Allsman, P. T., Power-shovel and dragline placer mining: U.S. Bureau of Mines, I.C. 7013, p. 8, 1938.

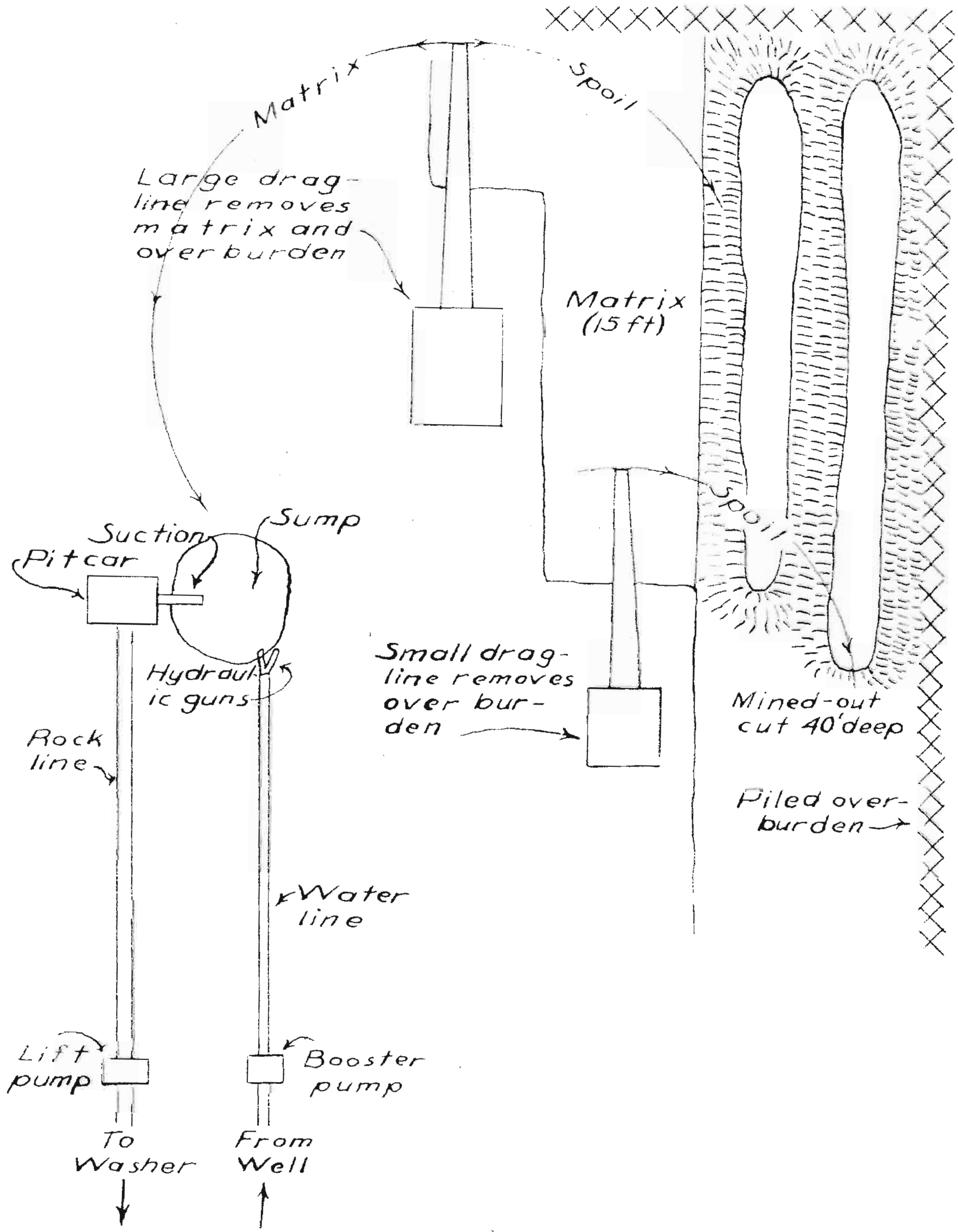

Figure 90. Mining system in Florida land pebble phosphate fields.

the wet matrix is soft and slippery so that it gives poor footing. The small dragline removes overburden only and casts it into the mined-out cut. The larger dragline travels parallel with the smaller machine. The large dragline alternately casts overburden to the mined-out cut and matrix to the sump where large hydraulic giants break it up and wash it to the suction through which it is pumped to the rock line and thence to the plant.

The economic overburden ratio in this system of mining is about 8 to 1. Every effort is made to increase the width of the dragline cut to minimize moving of the pit pump. The pit pump is located in such a way as to give a minimum swing for the draglines and at the same time minimize the relocation of the pumps.

At one operation of the International Minerals and Chemical Corporation, a Bucyrus-Erie 1150-B with a 215-ft. boom and a 21-cu. yd. Red Arch bucket moved dirt at an average rate of 1420 cu. yd. per hour over a period of 29 months. The digging cycle averaged 1.13 swings per min. It is interesting to note that at this operation, one set of hoist cables moved 3,038,000 cu. yd. of dirt and one set of drag cables moved 2,082,000 cu. yd. The average for like cables is 1,650,000 cu. yd. for hoist cables and 1,500,000 cu. yd. for drag cables. This overburden and matrix is easy to dig and not very abrasive as demonstrated by the cable lives.

The ability of the dragline to excavate from below the level of the machine and the large radius of operation make the dragline an efficient and economical machine for systems of mining or excavating similar to that illustrated in Figure 41. Here the drag-

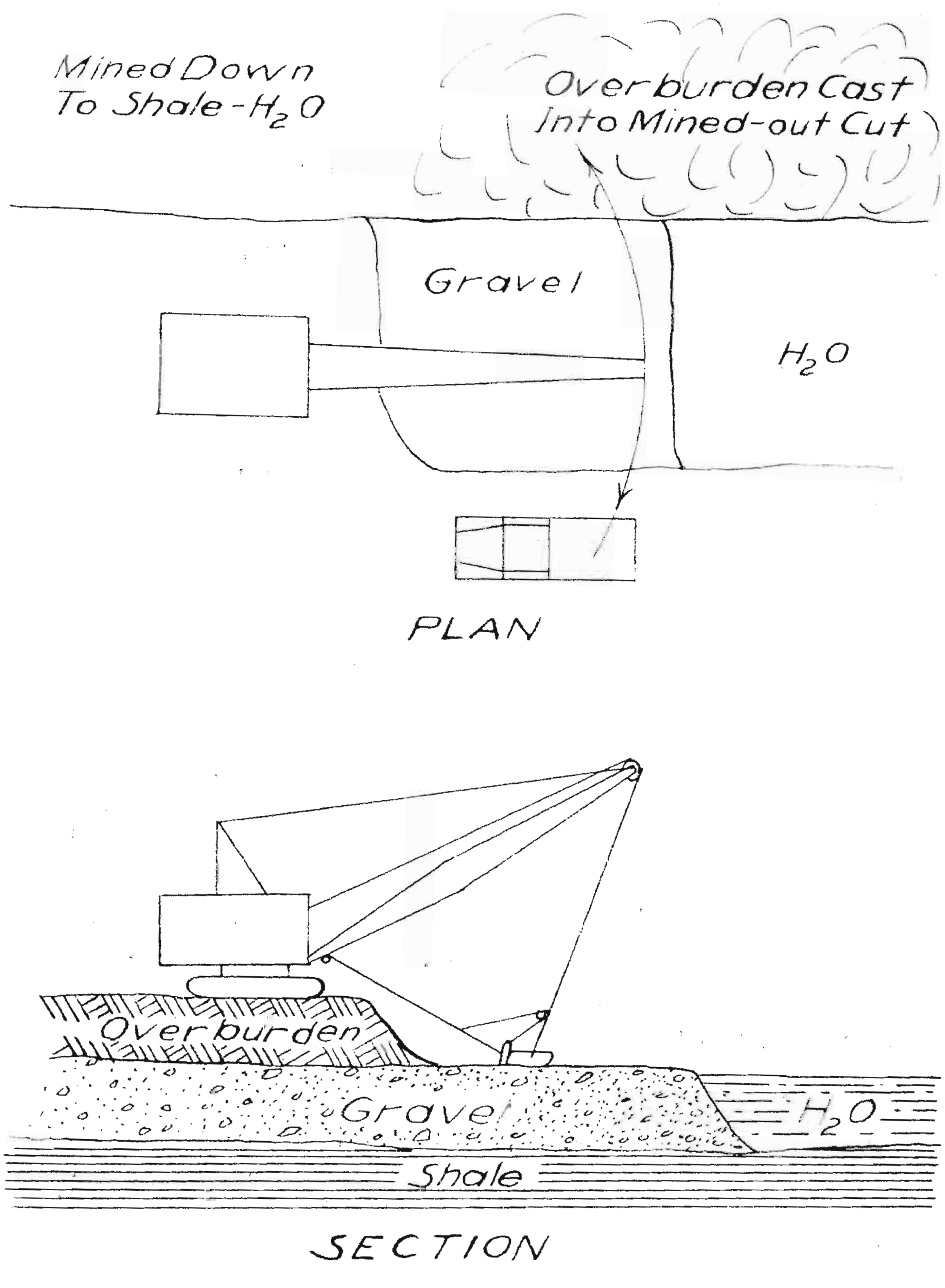

Figure 41. Mining gravel with a dragline.

line operates from a dry surface. It first strips the gravel deposit and casts the overburden into the mined-out cut. After the section of gravel is stripped clean, the dragline begins to excavate the gravel from below the water table and loads the gravel into trucks. Flexibility such as this could never be achieved with a power shovel.

Residual phosphate deposits (as in Tennessee) and residual clay deposits frequently have a very rough lower contact. The Tennessee phosphate deposits are underlain by limestone. Undecomposed ridges and pinnacles of the limestone protrude upward into the phosphate beds forming "cutters" or valleys. Figure 42 illustrates diagrammatically the conditions that exist.

If a power shovel were to be used here, it could excavate down to the level of the tops of the pinnacles and it would leave much of the phosphate in place. The lower surface would be too rough for the shovel to travel on and the cutters would be too narrow for the shovel to travel between. To attempt to carry a level floor at the bottom of the phosphate beds would entail too much blasting and too much sorting to be economically feasible.

In cases such as this the dragline is again the ideal machine. The dragline can work from the level surface and get almost complete extraction of the phosphate without dilution by waste. The dragline bucket can be dropped at any point and dragged around the pinnacles in the cutters. In practice a large dragline is used to dig most of the phosphate and a smaller machine with a small buc-

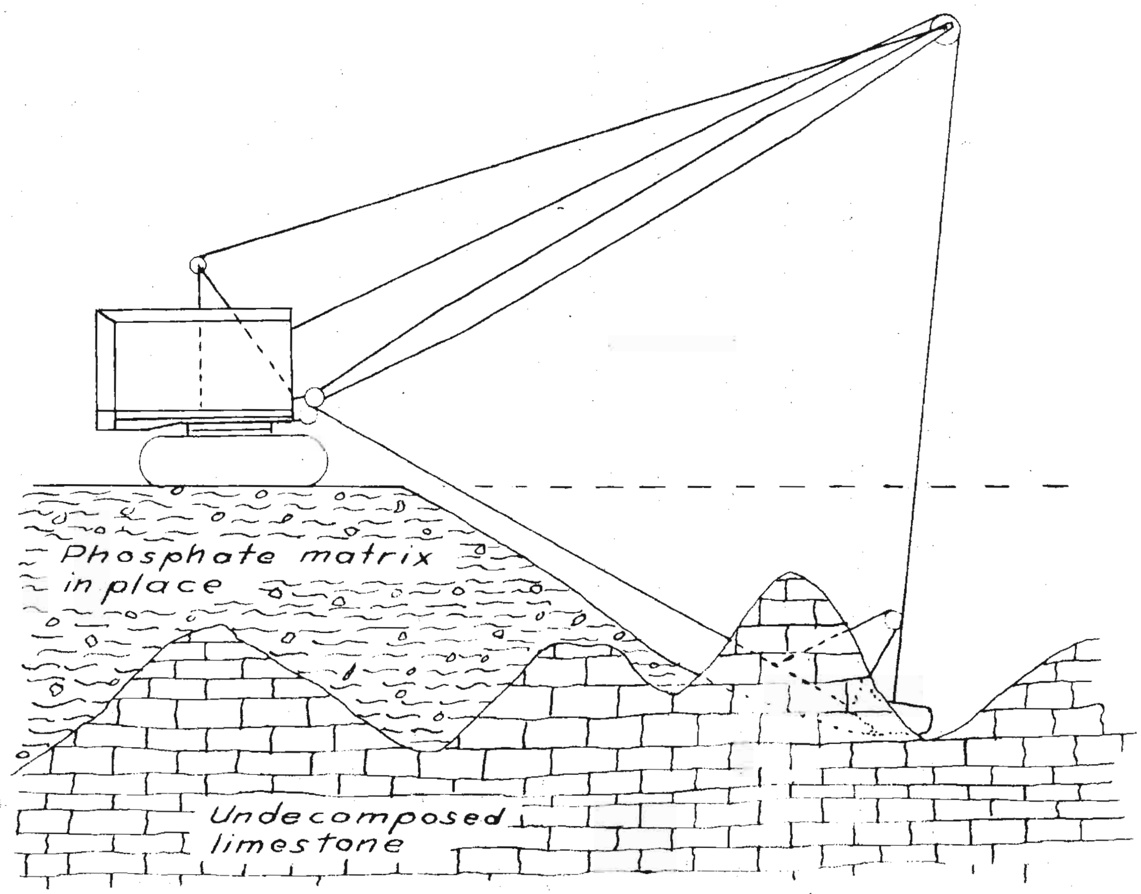

Figure 42. The use of the dragline on deposits with a rough lower contact.

ket is used to excavate the phosphate from the narrow cutters.

Another advantage in favor of the dragline here is that the dragline simplifies the haulage problem. The dragline can load into haul units operating on the same level as the dragline or on a level above. This minimizes road building and road maintenance.

To summarize the advantages of the dragline as illustrated by the above examples, it can be said that the dragline has a greater vertical and horizontal range of operation than does the power shovel. Pit drainage is not a problem in dragline excavation since the machine operates from a dry upper level. Road building and road maintenance are less a problem in dragline systems than they are in power shovel systems. Road grades may be held to the minimum with a dragline since greater elevation may be gained with a dragline than can be gained with the power shovel. The dragline has greater flexibility than does the power shovel.

Because of the features mentioned above, the dragline is an excellent machine for digging and for cleaning drainage ditches, canals, and similar excavations. In work of this type the depth is limited by the length of hoist and drag cable on the machine. Sometimes the angle of repose of the bank is a limiting factor. A high water table does not limit the dragline as it does the power shovel. Dumping height is not a limiting factor with the dragline. The dragline need not over-excavate as the power shovel frequently must in this type of excavation. Further, the great working radius gives the dragline more area over which to spoil the material and level it. Or, if desired, the dragline can build levees on one or both sides of the ditch or load into haul units with equal facility.

On one drainage canal project a 6150 Bucyrus-Monighan with a 12-cu. yd. bucket and a 160-ft. boom was used. This machine excavated a canal with a 46-ft. bottom, 2:1 side slope and a depth of 9.25 feet. The excavated material was used to build a uniform levee with a crown 24 feet wide on both sides of the canal. The dragline moved an average of 11,000 cu. yd. of earth per 24-hour day. A smaller dragline (Bucyrus-Erie 37-B) was used with a Caterpillar bulldozer to dress the final shape of the slope and top levees. The 37-B moved 1,100 yds. in 24 hours in cutting and dressing the slopes.

The great lateral and vertical working limits of a dragline make it applicable in coal-stripping. Large shovels are also frequently used in stripping coal, but even the 40-cu. yd. shovel cannot work some banks that a 25-cu. yd. dragline can handle with ease.

The simplest and ideal system of stripping with a dragline is illustrated in Figure 43. If the depth of the overburden does not exceed the digging depth of the dragline, the dragline can spoil all the overburden into the mined-out cut even though the percentage of swell of the overburden may be great and the angle of repose small. In this system, as in most stripping systems, the dragline first builds a small windrow with the stone taken from just above the coal. The windrow has two purposes. First, the material has a higher angle of repose under the surcharged weight than the finer material would have. Second, it acts as a barricade to prevent the other material from rolling onto the coal

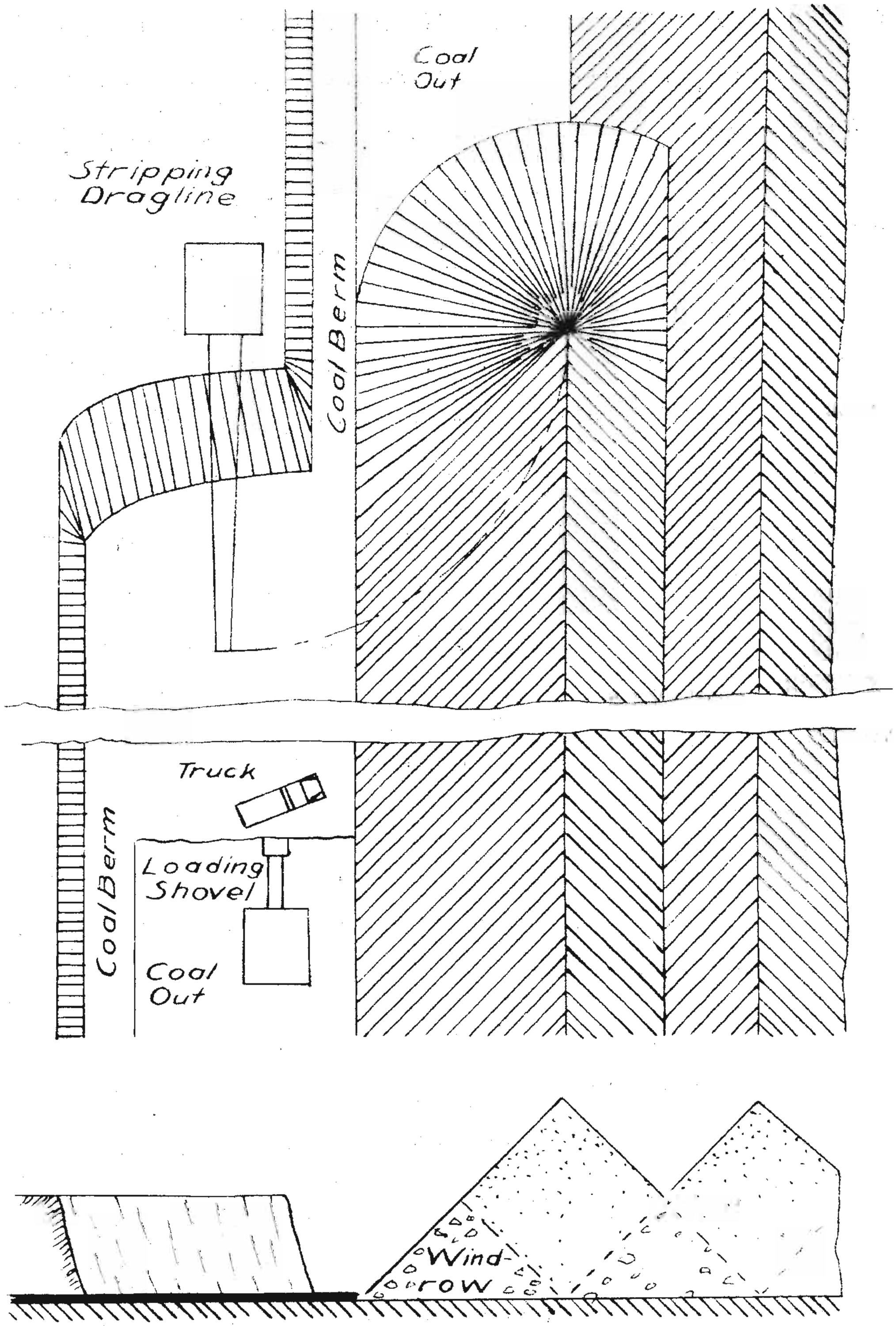

Figure 13. Strip-mining coal with a dragline in one cut.

bed. Also by placing the coarser material at the bottom of the spoil piles, reclamation of the area is made possible.

Sometimes it is necessary for the dragline to excavate material from above the level of the machine. The absolute limit from which it can excavate is the dumping height of the machine. This should be avoided if possible. When the bucket is being pulled down grade toward the dragline, aided by gravity it piles dirt in front of the machine. It is also more difficult to get a full bucket because the angle of repose of the material resists movement into the bucket. When the bucket is being pulled up grade to the gragline, gravity aids in filling the bucket in a shorter distance.

Although an inefficient practice, it is sometimes necessary to have the dragline dig downgrade. In some coal-stripping operations, the overburden is too thick to handle efficiently and conveniently in one cut. To dig as deep as required by the high wall in one cut would necessitate using too long hoist and drag cables. Hoisting time would also become excessive. For these reasons the dragline sometimes works two benches as illustrated in Figure 44.

In the system illustrated in Figure 44, the dragline first operates from position "A" to dig the face "D" and to cast this spoil into spoil pile "D'". After face "D" has been carried back to the machine, the dragline takes two or three steps toward the old high wall to position "B". From "B" the dragline excavates the upper bench "C" and spoils the material to "C'". The depth of face "C" is usually about 1/3 of the total depth of overburden.

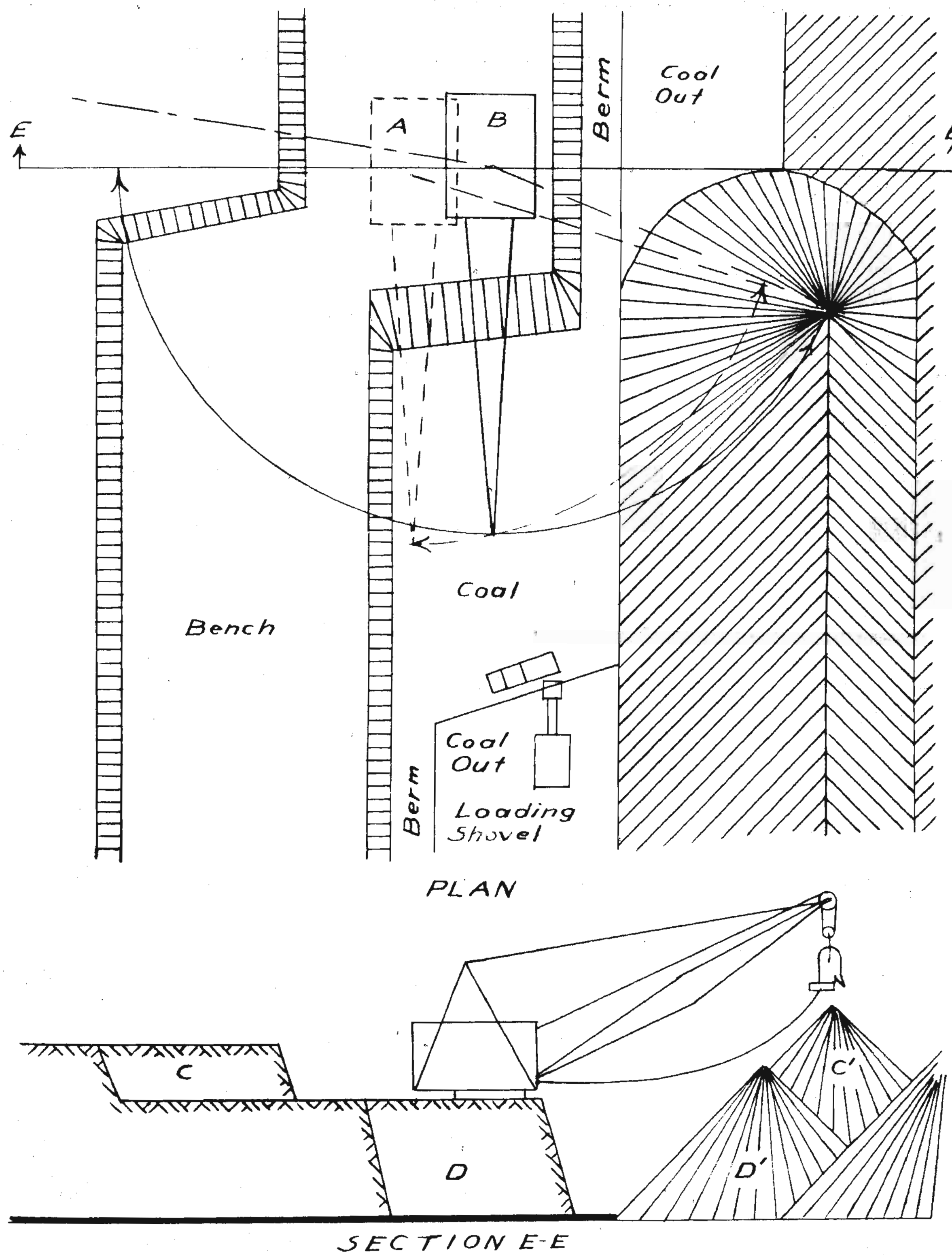

Figure 44. A dragline stripping from a bench.

The disadvantages of this system are apparent. When digging from position "B" the bucket is used to dig down grade. Also in this position the average angle of swing is very nearly 180 degrees. The advantage of the system is also readily apparent. In working face "D", the hoisting distance and time is at a practicable minimum as is the average angle of swing.

A variation of this last mentioned system is used by the Maumee Collieries at their Mine Number 28. A 1150-B Bucyrus-Erie with a 180-ft. boom and 25-cu.yd. bucket is used to strip 75 feet of overburden to get 3 feet, 9 inches of coal.

Here the pit is laid out so that the high wall forms a long sweeping outside curve. This reduces the height of the spoil piles. These spoil piles rarely exceed 15 to 20 feet in height above the original ground level. The dragline makes 50-ft. progress moves on sections which are 85 feet wide.

The dragline operates from a bench 15 feet below the ground level. From its position 85 feet from the highwall, the machine first removes 15 feet of clay overburden from the 50-ft. section in order to save excessive hoisting in later excavation. Next, after moving up along a line 85 feet from the old highwall, the dragline cuts a triangular slot 40 to 50 feet wide at the old face and tapering to the machine. This pilot cut is carried to the coal seam and thus forms the new highwall. The material from the pilot cut is used to form a retaining wall just beyond the usual coal berm.

After the pilot cut and retaining wall are completed, the dragline takes three steps (21 feet) toward the old highwall. From this new position the dragline begins to dig from the lower bench. Using a continuous swing much of the time, the dragline digs and casts the dirt over the retaining wall. The machine swings in a clockwise direction, stopping only to load the bucket. The bucket is dumped "on the fly". The concave spoil area, the new position closer to the bank, and the added throw of the continuous swing gives the operator a wide dumping area. The material is thrown an extra 25 feet by centrifugal force. By not stopping to dump and reversing the swing, power is also saved.

The dragline makes three more moves toward the pit as it removes the balance of the 85-ft. wide by 50-ft. long progress section. On the last move the dragline is standing on the edge of the old highwall. From these new positions it is able to spoil the dirt farther back.

The last operation before moving on to the next section is stripping top soil from the bench in a radius of 180 feet. A layer of shale is then added to form a travel way. The excavator completes a 50-ft. progress section where the overburden is from 65 to 70 feet deep every eight-hour shift.

When working as described above, this dragline has averaged 750,000 cubic yards a month. The high for a single month was 831,400 cubic yards when working 24 hours a day and seven days a week.

A recorder on a swing motor keeps a continuous record of the performance. This indicates that the dragline has averaged 450 bucketfulls per shift for many months. The life of the drag and hoist ropes is about a million and a half cubic yards.

The different positions of the dragline bucket when digging are illustrated in Figure 45. Position "c" is the best position for digging because the bucket at this position has the shortest distance through which to be hoisted. If digging is done in area "b," the dragline has the added distance "d" through which to hoist. When digging in area "a," there is still another added distance "d" through which the bucket must be hoisted. If the fairleads of the dragline are high, the vertical component of the drag pull will reduce digging efficiency. Further, when the bucket is in position "a" or "b" the hoist and drag must work against each other as the bucket is hoisted.

Hoisting distance should be such that when the dragline has been swung into dumping position, the bucket has also been hoisted into dumping position. In coal stripping this can be accomplished in part by casting the material dug from the toe of the cut into the first windrow. Thus hoisting distance is almost negligible here. Then the material that is dug from higher on the face can be cast behind the windrow and so here, too, the hoisting distance is minimized.

Sometimes it is imperative to gain elevation. In these instances not only is no effort made to minimize hoisting distance, but also the hoisting is carried to the maximum. Figures 45 and

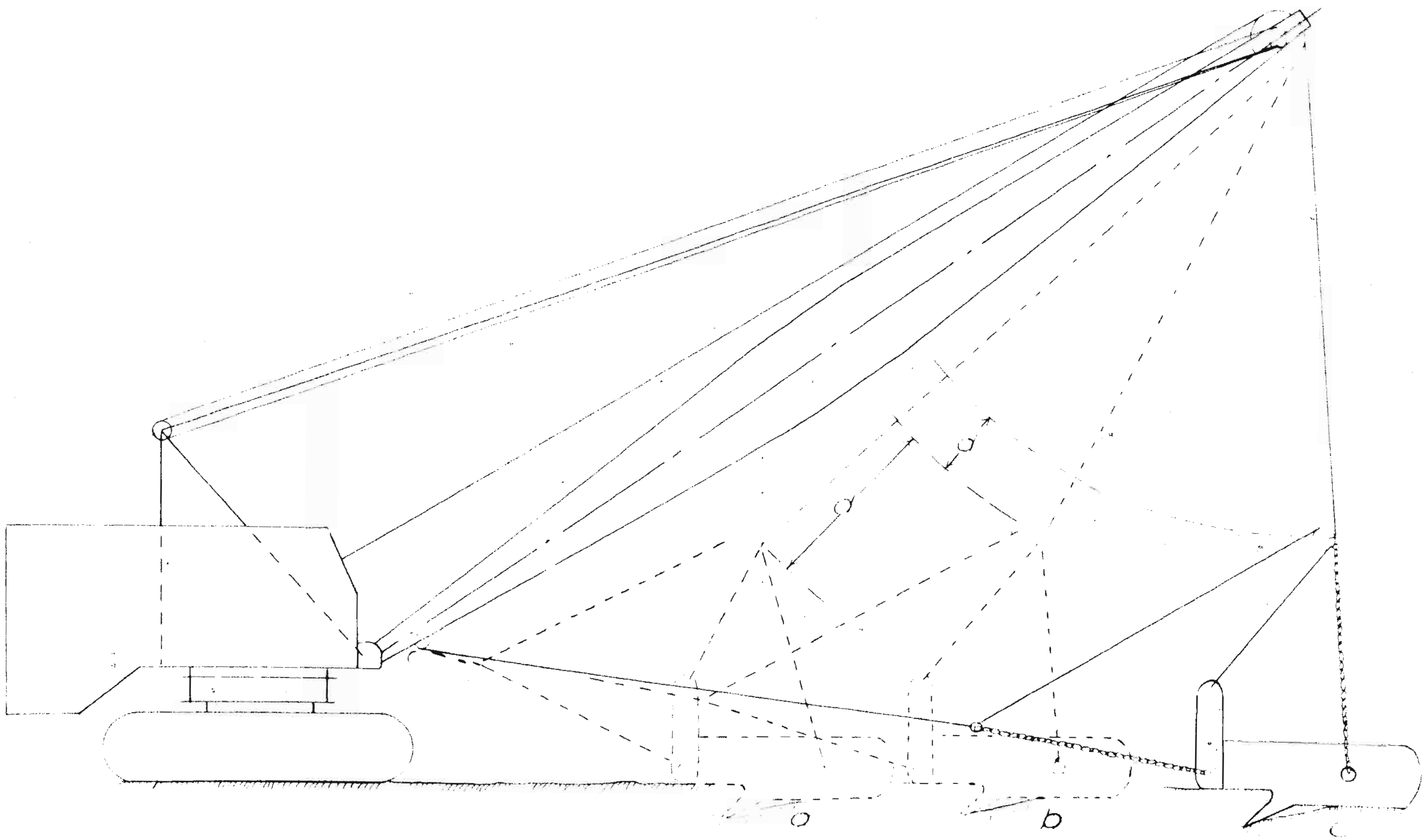

Figure 45. Digging positions of dragline bucket.

and 46 illustrate a novel system of stripping and mining in which advantage was taken of the vertical range of draglines.

In 1942 the War Production Board issued an order that stopped gold mining. The Cooley Brothers of Como, Colorado had been using their two draglines to mine placer gold. Rather than let their equipment lie idle, the Cooley Brothers moved it to Webb City, Missouri, and began to mine a lead-zinc deposit in the manner shown.

Because the area of the lease was limited, it was necessary to gain elevation as rapidly as possible. A 6 percent grade for efficient truck haulage would have entailed a 2000-ft. approach or a like length of benches. Because of these factors and the equipment available, the "stair-step" benches as shown in the illustration were used. This method of excavation allowed almost complete extraction of the ore. Figure 46 shows the sequence in which the benches were excavated. Normally, it is uneconomical to handle material twice or three times as shown here, but in this case it was better to use the equipment in the manner shown at a decreased efficiency and economy rather than let the equipment remain idle.

The draglines were Bucyrus-Monighan 3-W electric powered machines. The booms were 80-feet long and 2½ to 3½-cu. yd. buckets were used. Walk ways for the machines had 20 percent grades.

A greater degree of selective mining is possible with the dragline than is possible with the power shovel. In the discussion on the power shovel, the phosphate and clay deposits which contained sand lenses were used as illustrations. The same illustration will serve to illustrate the efficiency and applicability of the drag-

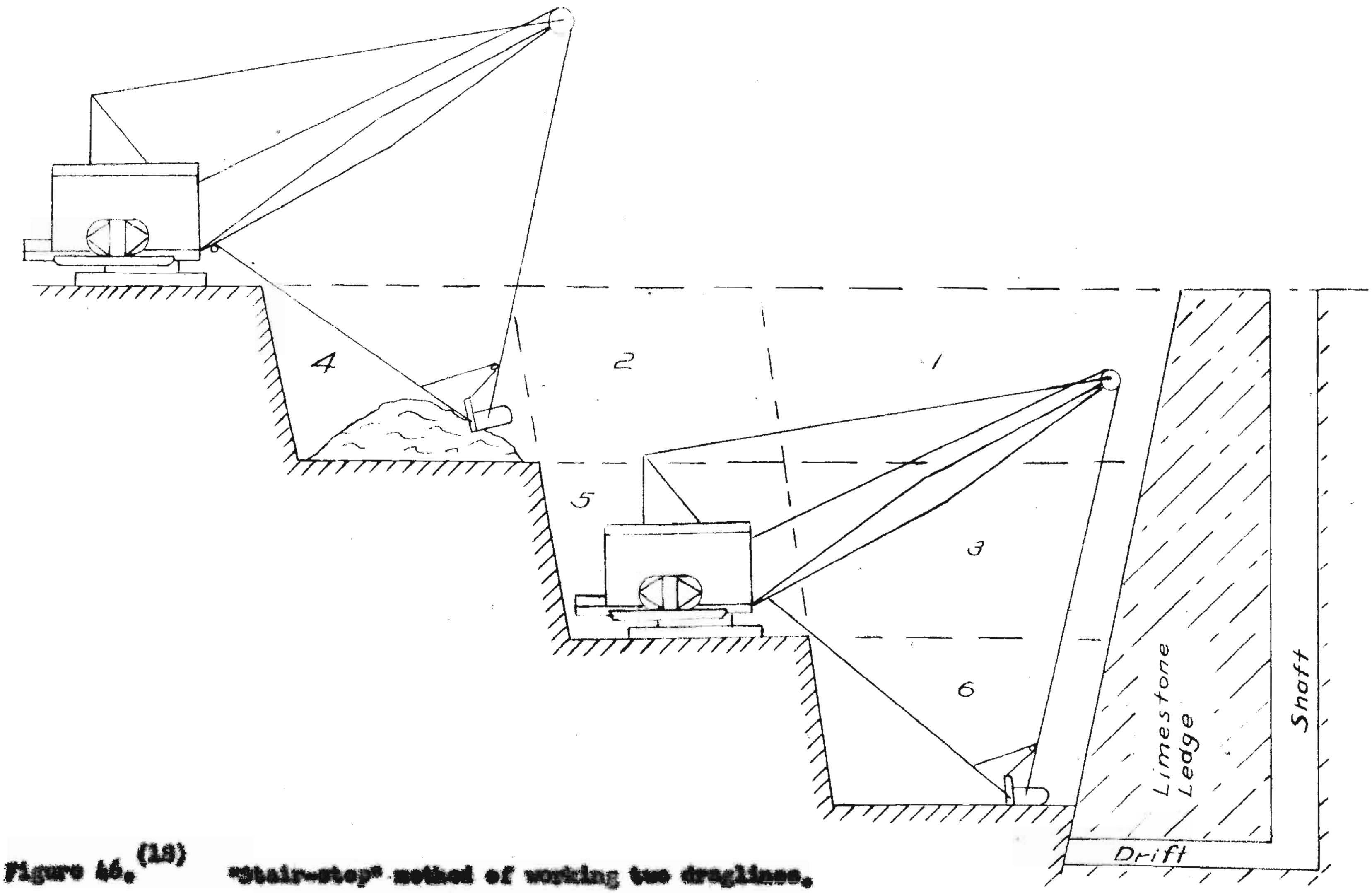

Figure 46.[18] "Stair-step" method of working two draglines.

(18) After Grace, J. D., Sunken Flats; Excavating Engineer, Vol. XXXVIII, No. 3, p. 116, March, 1944.

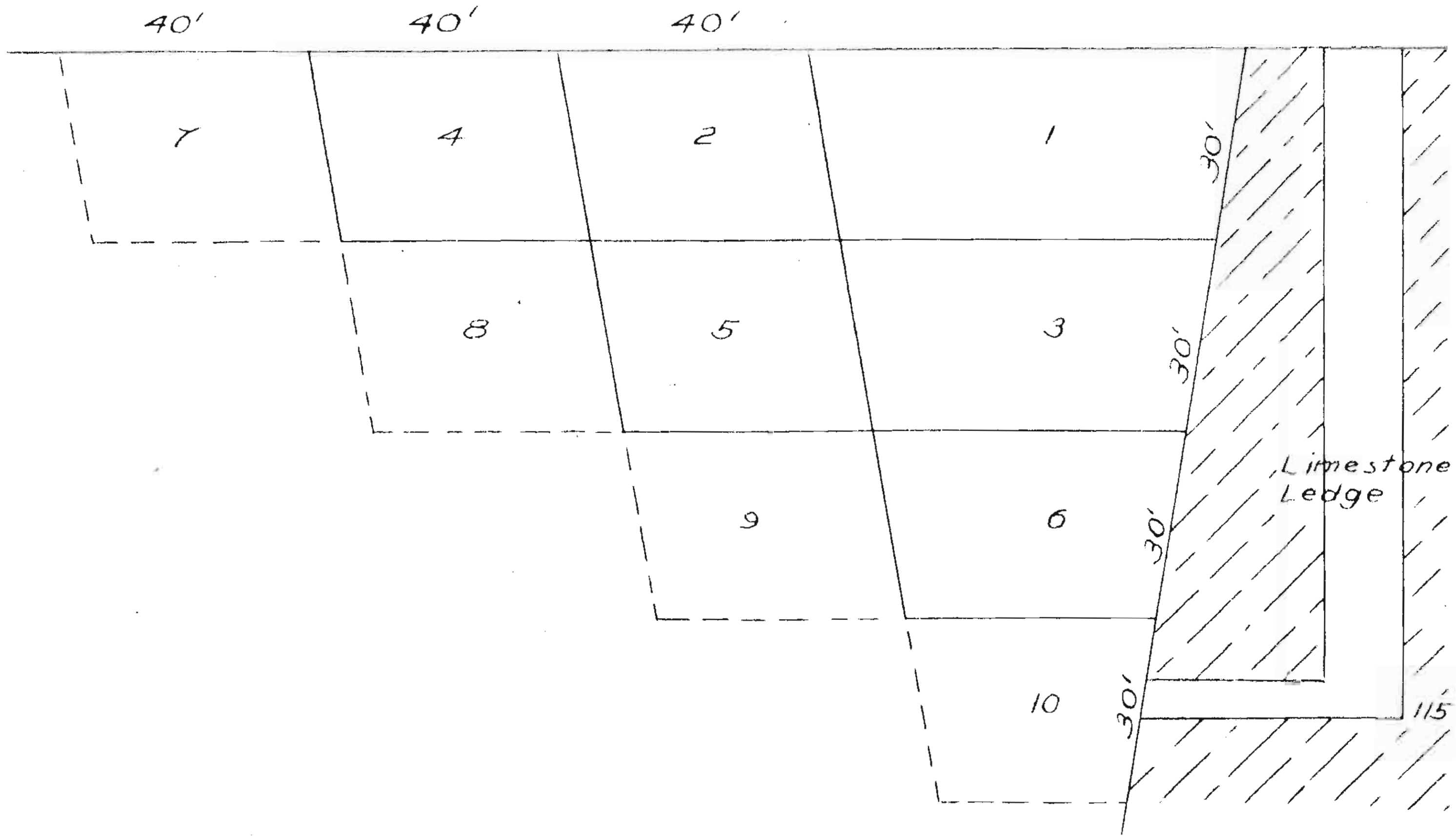

Figure 47.[19] Sequence of excavation in "stair-step" system.

(19) Ibid., p. 117.

line. In these cases the dragline could mine down to the top of the sand lense. The dragline could then load the sand into waste trucks without dilution. After the sand lense has been removed the dragline could continue to mine the phosphate or clay that lay below it. All of this could be done without moving the dragline from its initial position.

Some types of clay deposits are stratified in thin beds and it is desired to mine these without mixing. For a shovel to mine the beds would mean mining out one thin bed (an inefficient procedure with a shovel) for a large area and then dropping to the next thin bed and so on through the deposit. The dragline can mine a thin bed over a small area completely by taking horizontal slices as illustrated in Figure 48. After one bed has been cleaned up, the dragline can mine the next lower bed from the same position without moving and load it into units without dilution or mixing.

If mixing is desired, the dragline can mix as efficiently as the shovel by taking inclined rather than vertical slices from the bank as illustrated in Figure 48. Flexibility such as this is not possible with the power shovel.

Generally the dragline can work under conditions of poorer footing than can the power shovel. The 3-cu. yd. to 6-cu. yd. shovels used in mining have bearing pressures varying from 20 to 25 lbs. per square inch. This could be decreased by increasing the bearing area. The bearing area could be increased by making the crawlers wider and longer. There are structural limitations

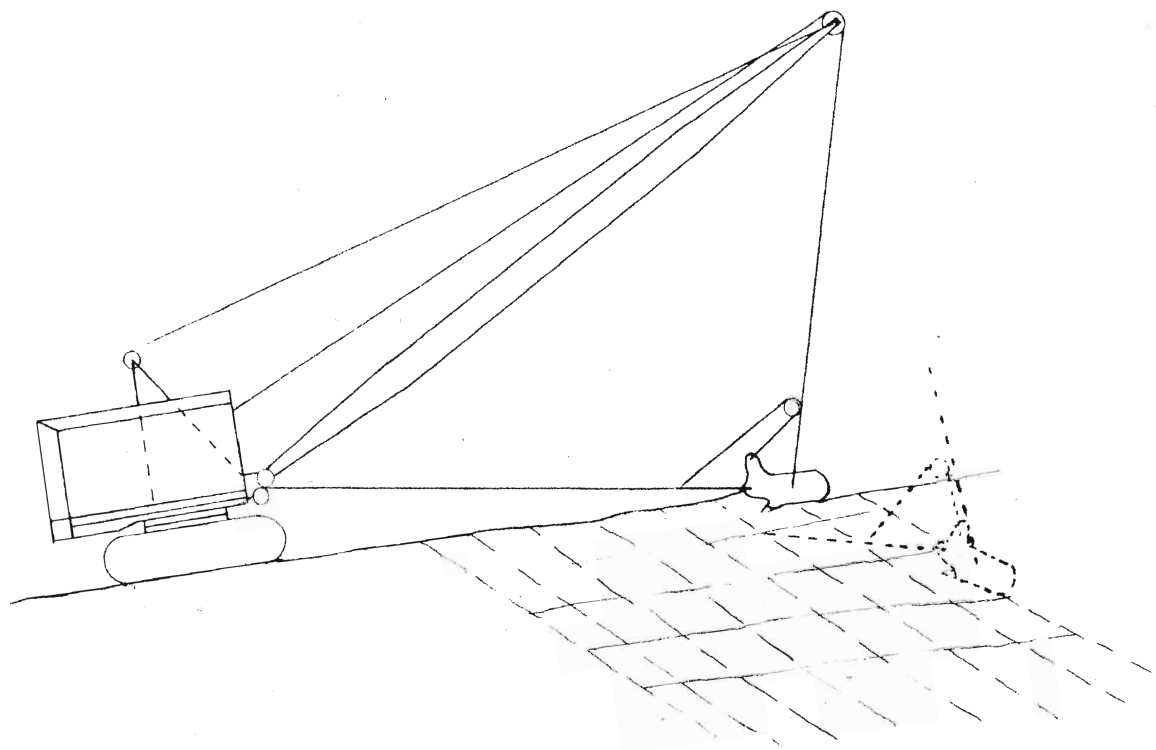

Figure 12. Inclined and horizontal cuts with dragline.

to increased crawler width. Length of crawlers cannot be increased on shovel because if they were made longer they would interfere with the bucket when it was brought back to the toe of the cut. On crawler-mounted draglines crawler length can be increased to increase bearing area and stability without interfering with digging. Further, tappered-end crawlers (see Figure 34) can be used on draglines to give them greater climbing ability in soft ground. As for the large machines, the huge walking shoes and base of the walking draglines give them very adequate support, whereas, the large shovels have bearing pressures of about 50 lbs. per square inch.

The small crawler-mounted draglines can use mats or floats to further increase bearing area just as the small shovels do.

Yet another consideration is that of method of operation. The shovel must work at the bottom of the cut. Here rain water and ground water collects to make footing conditions worse. The dragline works above the cut and drainage is into the cut. This improves soil conditions for support of the dragline.

What has been said for the operator on a power shovel also applies to the operator on a dragline. If anything, the effect of operator efficiency and competance on the performance and capacity of the dragline is greater than it is on the power shovel.

The dragline cycle is made up of fewer motions than is the cycle of the power shovel. The dragline drags, hoists, swings and dumps. In operation, if work is properly laid out, the hoist time need not be considered. The cycle breaks down to two elements. The dragline drags the bucket to pick up a load. After it has been loaded, the dragline hoist the bucket while it swings. If loading into haul units there is a dumping time. If casting, there is no dumping time since the dragline dumps as it reverses the direction of swing. The bucket is then lowered as the machine swings back and the cycle is repeated.

The length of time needed for loading depends upon the drag speed and the distance through which the bucket must be dragged to pick up a load. The distance through which the bucket must be dragged depends, in turn, upon the type of material being dug, the length of the bucket and other features of the bucket design. The following drag distance are based upon observations by J. R. Thoenen:[20]

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(20) Thoenen, J. R., Sand and gravel excavation; Part 1: The power shovel, the dragline excavator, and the excavator crane; U. S. Bureau of Mines, I.C. 6798, 1934.

Bucket Size - cubic yards	Minimum (3 x av. bucket length)	Maximum (6 x av. bucket length)
3/8, 1/2, 3/4.	10	20
1, 1-¼, 1-½, 1-3/4, 2, 2-½, 3, . .	15	30
3-½, 4, 5, 6, 7, 8, ...	20	20

These averages can be taken for preliminary estimates. Some improvements in bucket design have resulted in buckets which can load in shorter distances. The makers of the Page Automatic Bucket advertise their bucket as being able to load in from 1 to 4 bucket lengths.

By taking the drag speed from the machine specifications, the loading time can be computed readily by using the equation which follows.

$$\text{Loading tim in secs.} = \frac{\text{60 secs. times drag distance, ft.}}{\text{speed of drag cable in ft. per min.}} \quad (11)$$

The swinging time of the dragline can be found by using equation (8).

If loading into trucks, cars or hoppers the dumping time is the same as for the power shovel. If casting, the dumping time is not considered, because the bucket is dumped as the swing is reversed.

The theoretical cycle of the dragline is the sum of the loading time, the swinging time, and the dumping time (if not casting.) The theoretical capacity can be found by using equation (9). The actual capacity can be found by using equation (10).

In place of the swell factor of the material in equation (10), it would perhaps be more accurate to use a bucket efficiency factor such as one given in Table IX. These bucket efficiency factors make allowance for a possible heaped load in the bucket.

Example 7

A 2-cu. yd. dragline with a rotating speed of 3 rpm and a drag cable speed of 130 f.p.m. is to be used to load trucks. The average angle of swing will be 90

TABLE IX[21]

Bucket Efficiencies of Draglines in Various Materials*

Easy digging factor, 95-100%	Medium digging factor, 80-90%	Medium-hard digging factor, 65-75%	Hard digging factor, 40-65%
Sand and small gravel, dry or moist	Materials not hard to dig without blasting but breaking with large voids	Materials requiring light blasting; bulky and not easily penetrated by bucket	Blasted rock with large voids, difficult to enter
Loam and loose earth	Clay, wet or dry	Well broken limestone, sandstone, shale, etc.	Hard, tough shale
Muck	Coarse gravel	Ores not massive in character.	All hard rocks
Sandy clay	Clay-gravel, packed	Heavy, wet, sticky clay	"Caliche"
Lose clay-gravel	Packed earth	Gravel with large boulders	Mixtures of coarse and fine broken material
Cinders and ashes	Anthracite	Cemented gravel	Tough, rubbery clay which shaves from bank
Bituminous coal			
Well loosened material			

* % of bucket capacity actually filled

(21) Peele, Robert and Church, J. A., Mining engineers' handbook, Vol. 1, p. 10-455, New York, Wiley, 1944.

degrees. What will be the output of the machine?

$$\text{Swing time} = \frac{(60)(2)(90)}{(360)(3)} = 10 \text{ secs.}$$

$$\text{Loading time} = \frac{(60)(22.5)}{130} = 10.4 \text{ secs.}$$

$$\text{Dumping time} = 3.3 \text{ secs.}$$

$$\text{Theoretical cycle} = 27.7 \text{ sec.}$$

$$\text{Theoretical capacity} = \frac{(3600)(2)}{27.7} = 260 \text{ cu. yd.}$$

If the dragline was excavating sand, the actual capacity would be:

$$(.65)(.95)(260 \text{ cu. yd.}) = 160 \text{ cu. yd. per hour.}$$

Another method of estimating dragline performance is that of Holcomb.(22) The multipliers are similar to those previously given

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(22) Holcomb, A. E., Output factors for excavation and material — handling equipment, Civil Engineering, Vol. 1, No. 1, pp. 26 – 30, October 1930.

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for the power shovel.

The basic assumption of this system is that operating conditions are favorable; that reasonably dry, ordinary earth is to be excavated from an 8-ft. cut; that a 1-cu. yd. bucket is to be used; that the material is to be wasted in a spoil bank at the side of the machine so that the average angle of swing will be 90 degrees. Under such conditions the operator and machine should excavate about 95 cu. yd. per hour, place measurement.

The multipliers are based on the following speeds:

Swinging	3.5 rpm
Hoisting	170 ft. per min.
Dragging	140 ft. per min.

The multipliers given in Table X apply when the dragline is making an end-cut. For a side-cut in ditching, where the bucket must dig down and trim the bank as well as dig coming up, the multiplier would be 0.95 to 0.90 of the end-cut multiplier (depending on the ratio of the depth to the width of cut and to the working radius) until the depth becomes approximately 0.4 of the working radius; after this the yardage, except in easy-digging material, which would diminish so rapidly as to make it desirable to change to a digging type of clamshell bucket. As before, the use of larger machines and buckets would affect the depth multiplier.

Digging rates in sticky material depend on how much the bucket becomes clogged and how often it has to be cleaned.

The time lost in moving back if mats are used is 3/4 of a minute instead of the normal 15 seconds. The 15 second moves have been allowed for in the Tables.

The use of Tables X and XI is illustrated by the examples which follow:

<u>Example 8</u>

Assume a 2-cu. yd. bucket is to be used in tough, rubbery clay making an 8-ft. side cut, and loading trucks through a 90-degree swing. What will be the production?

TABLE X (23)

MULTIPLIERS OR OUTPUT COEFFICIENTS FOR DRAGLINE WORK

Materials	Multiplier
Fairly well-blasted rock or hard-pan.	0.40
Tough, rubbery clay	0.45
Clay and boulders	0.55
Heavy clay (not sticky)	0.70
Clay gravel	0.80
Wet, sandy clay	0.90
Ordinary earth and dry, sandy clay	1.00
Light, dry loam and clay, loose sand and gravel, cinders, or ashes	1.10
Light, moist clay and loam	1.25

Size of Bucket (assuming medium heavy construction)

3/8	cu. yd.	00.38
1/2	cu. yd.	0.50
5/8	cu. yd.	0.63
3/4	cu. yd.	0.75
1	cu. yd.	1.00
1-1/4	cu. yd.	1.25
1-1/2	cu. yd.	1.50
1-3/4	cu. yd.	1.75
2	cu. yd.	2.00
2-1/4	cu. yd.	2.25
2-1/2	cu. yd.	2.50
2-3/4	cu. yd.	2.75
3	cu. yd.	3.00

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(23) Holcomb, *op. cit.*, p. 29.

TABLE XI [24]

MULTIPLIERS OR OUTPUT COEFFICIENTS FOR DRAGLINE WORK

(Casting Empty Bucket Beyond Point of Boom . . . . Multiplier is 0.85 to 0.75 depending on arc of outward swing. 0.80 is an average)

Depth And Cut of Excavation	Multiplier Size Drag Bucket Cubic Yards 3/8 ½ 5/8 3/4	1 1¼ 1½	1-3/4 2 2¼	2½ 2-3/4 3
1'	.90	.82	.75	.60
2'	.95	.90	.82	.75
4'	.98	.95	.90	.82
6'	.99	.98	.95	.90
7'	1.00	.99	.97	.93
8'	.975	1.00	.98	.95
9'	.96	.99	1.00	.97
10'	.95	.975	.99	.98
12'	.925	.95	.975	1.00
14'	.90	.925	.950	.975
16'	.875	.90	.925	.95
18'	.850	.875	.90	.925
20'	.825	.850	.875	.90
22'	.80	.825	.850	.875
24'	.775	.800	.825	.850
26'	.75	.775	.80	.825
28'	.725	.750	.775	.800
30'	.70	.725	.750	.775
32'		.70	.725	.75
34'		.68	.70	.725
36'		.66	.68	.70
38'			.66	.68
40'			.64	.66

Type of Operation

Loading trucks 0.90
Dumping at 180 degree-swing from cut 0.80

Side Cut — to be used where dragline must dig down and trim the bank as well as dig coming up. 0.95

Multiplier varies from 195 for normal conditions down to 0.90 when the working depth exceeds 0.4 of the working radius.

Moving time . . . . 15 seconds.
Moving time if mats are used . . . . 45 seconds.

(24) Holcomb, op. cit., p. 29

From Table X the multiplier for the clay is 0.45. From Table X the multiplier for the bucket size is 2.0. From Table XI the multiplier for depth of the cut is 0.98. From Table XI the multiplier for loading trucks is 0.90. The multiplier for trimming banks is 0.95.

The output coefficient is the product of the above multipliers, as follows:

(2)(0.45)(0.98)(0.90)(0.95) = .754.

The dragline production then becomes:

(0.754)(95 cu. yd. per hr.) = 71.63 cu. yd. per hr., place measurement.

<u>Example 9</u>

Assume a 3-cu. yd. bucket working in wet, sandy clay and digging a 12-ft. side-cut channel. The machine swings through 180 degrees to deposit the material on a spoil bank. The dragline needs mats to move along the ditch. The average yardage excavated per move is 100 cu. yd. The average yardage excavated per move is 100 cu. yd. What production can be expected?

From Table X the multiplier for the clay is 0.90. From Table X the multiplier for the bucket size is 3.0. From Table XI the multiplier for the depth of cut is 1.00. From Table XI the multiplier for a 180-degree swing is 0.80. The multiplier for trimming banks on a side-cut is 0.95.

Disregarding the extra time for moving on mats, the output coefficient becomes:

$$(0.90)(3.0)(1.00)(0.80)(0.95) = 2.052.$$

Since the original assumption was 95 cu. yd. in 60 minutes, the excavating time between moves will be:

$$\frac{(100)(60)}{(95)(2.052)} = 32.5 \text{ minutes}$$

But it requires 45 seconds to move on mats. The 15 seconds for a move without mats was allowed for in the original yardage assumption. Using 0.75 minutes to move, the resulting job coefficient will be:

$$\frac{32.5}{(32.5) + (0.75 - 0.25)} = 0.985.$$

Output then equals:

(2.052)(0.985)(95) cu. yd. per hr.) = 192 cu. yd. per hr., place measurement.

It must be remembered that the results in examples 7, 8 and 9 will very closely approximate long period averages if there are no major delays.

Tables X and XI illustrate some of the variables in dragline production. They also illustrate that there are optimum conditions of operation for each machine.

THE CLAMSHELL, TRENCH HOE, AND SIMILAR EQUIPMENT

The clamshell, trench hoe and the horizontal thrust loader are somewhat similar in design to the power shovel and the dragline. These machines are very unlike the power shovel and the dragline in that their applicability to different types of excavation projects is very limited. In fact, the trench hoe is a one-purpose machine.

Of the machines mentioned in this section, the clamshell has the greatest flexibility and the widest range of efficient application.

The working radius and dumping height of the clamshell are functions of the boom length and angle. The digging depth of the clamshell is a function of the length of wire rope carried by the machine as well as by the boom length and angle. Both the digging depth and height to which material can be piled might be restricted by the angle of repose of the material. Figure 31 illustrates the working dimensions of a clamshell.

The clamshell, like the dragline, can excavate material from below water level with the same facility but decreased efficiency as it can excavate material above the water table. One of the common and efficient uses of the clamshell is in dredging.

The clamshell is restricted to digging unconsolidated small-sized material. The ideal materials are sand, gravel, cinders, crushed stone, and screened coal.

In poorly blasted rock the clamshell is inefficient. The efficiency in poorly blasted rock can be increased by the use of an orange-peel bucket.

The clamshell can be used to load haul units but it has the same limitations that a dragline has in this type of work. The efficiency can be increased in this type of work by providing a larger target as is done in using a hopper. A hopper also provides some surge capacity and thus makes more continuous operation possible. Trucks can be loaded more rapidly from a hopper than by the machine directly and so there is a saving in time here also. Charging hoppers is one of the most common and efficient uses for the clamshell. This use is illustrated in Figure 49.

The clamshell reaches its peak efficiency under conditions where relatively great elevations must be reached in restricted working areas as shown in Figures 49 and 50. Figure 50 illustrates an efficient use of the clamshell as an excavator. Here elevation can be gained easily and rapidly without over-excavation. Excavations such as foundation pits and shallow shafts can be efficiently and economically dug by a clamshell.

Handling material to or from stockpiles is a common and efficient clamshell application. When used in stockpiles precautions are necessary to prevent segregation of the various sizes of material. To prevent this segregation the stockpile should be built up in layers. First the clamshell should place a thin layer of material over the entire base area of the pile. This is followed by successive layers each placed over the available area in

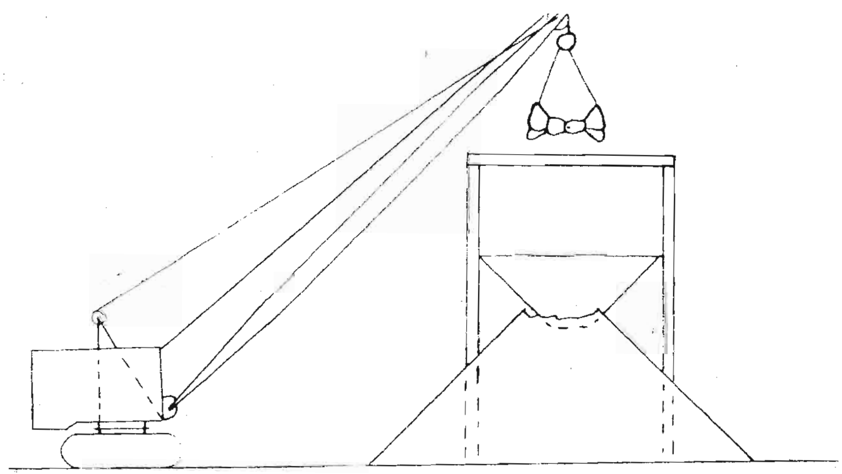


Figure 49. Charging a hopper with a crane excavator.

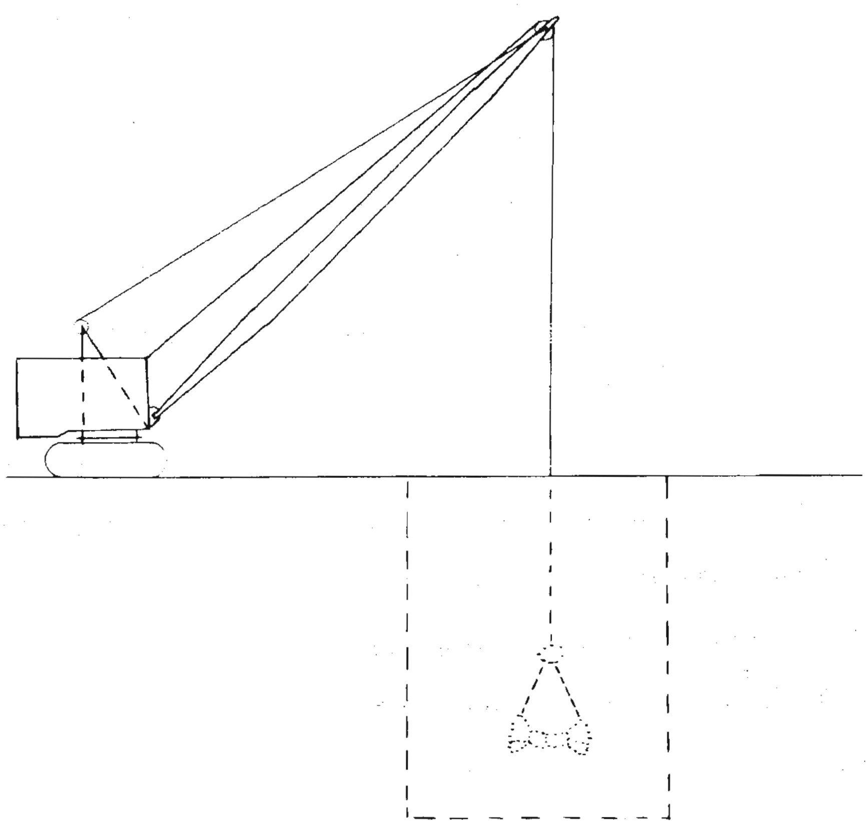


Figure 50. Shaft and foundation excavation with a clamshell.

the same manner in which the first was placed. In this way the material is prevented from rolling down the sides of the pile as it would if the material was dumped at the apex of a cone-shaped pile. Rolling down the sides of the pile will cause segregation of the sizes with the large pieces at the bottom and the fine fragments or particles at the top.

To prevent segregation of sizes when removing material from a pile, excavation should begin at the top of the pile as in Figure 49. If excavation were by a machine such as a power shovel, the material would roll down into the cut and cause some segregation of sizes.

A clamshell is the only machine discussed here that can be used to unload railroad gondola cars. This is a large field of application for the clamshell.

The clamshell has found some use as a ditching machine. In this type of work it operates in much the same manner as a dragline. The clamshell can work from either the end or the side of a cut. The material can be loaded into haul units, used to build levees, or just spoiled to either side or both sides. In one case fast progress was made on a 4,700 ft. canal. The canal was 70 ft. wide and 8 ft. deep. The water table was close to the surface. A Bucyrus-Monighan 2-T with a two-yard Owen clamshell bucket was used.

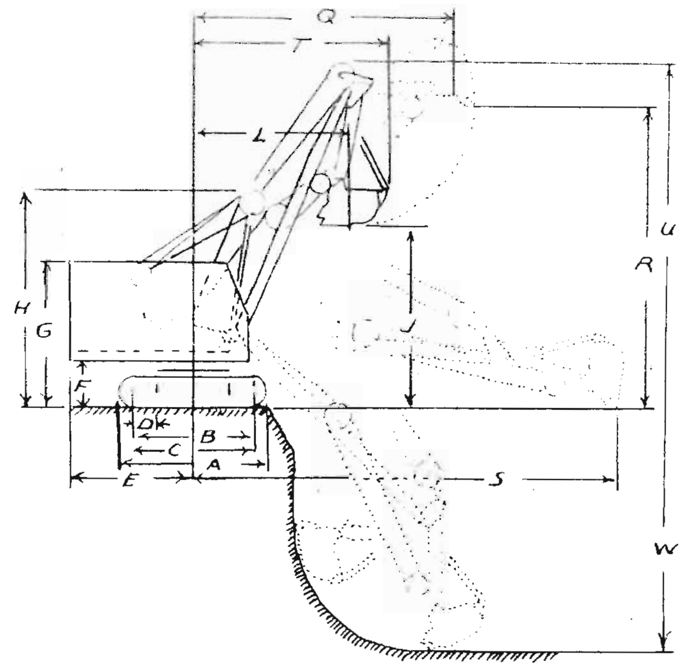
Another efficient application of the clamshell is that of placing riprap on levees, earth dams and similar construction.

The clamshell has found a few applications in mining. One such application is that at the North Shore Sand and Gravel Pit at Flushing, Long Island. Here a bank of sand 75 feet high is being mined. To approach this bank with a power shovel would be dangerous because of slides. At this operation a crane with a 45-ft. boom at a 45 degree angle and a 2¼ cu. yd. clamshell bucket digs the sand and loads it into a hopper that feeds the sand to a portable belt conveyor. The machine has a 37-ft. working radius and so it can operate at a safe distance from the bank. This machine handles 1,200 buckets full (over 2,500 cu. yd.) per eight-hour shift.

The clamshell is also used to mine pockets of clay, iron ore (Alabama), and a few other minerals.

The trench hoe or pull shovel is a one-purpose machine. Figure 51 illustrates the maximum working ranges of a trench hoe. A study of this diagram also shows why the machine is restricted to ditching or very closely allied operations. By attaching a backfiller blade, the machine can be used to back fill ditches.

The horizontal thrust loader is the least commonly used of the machines under discussion. Its commonest application is in loading coal in the strip mining of coal. The machine can probably excavate thin cuts more efficiently than can the power shovel.



Boom Length....................	24'-0"
Dipper Arm....................	11'-8"
Dipper........................	1 Yd.

A--Overall Length of Standard Crawlers	13'-8"
C--Overall Width of Crawlers with--	
24" wide shoes................	10'-8"
30" wide shoes................	11'-2"
Overall Width of Cab...............	10'-7 3/8"
E--Tail Swing Clearance...............	11'-7"
With Cat Walk.....................	13'-3"
Clearance Heights--	
G--Over Cab.........................	12'-9"
H--Over Jib Frame.....................	19'-3"
J--Clearance Height at Beginning of Dump..........................	15'-9"
Q--Radius at End of Dump, Highest Position........................	25'-6"
R--Clearance Height at End of Dump.....	25'-6"
S--Max. Digging Reach.................	40'-6"
T--Clearance Radius of Boom and Bucket at Max. Dumping Height.....	19'-7"
U--Height of Boom at Maximum Dumping Height........................	32'-3"
W--Digging Depth.....................	23'-0"

Figure 51. Working dimensions of a pull shovel.

DRAWBAR PULL, RIM PULL AND THE COEFFICIENT OF TRACTIVE EFFICIENCY

There are several factors that affect the performance of rolling equipment. These factors are drawbar pull, rim pull, the coefficient of friction, rolling resistance, grade resistance and acceleration. These factors are dependent wholly or in part on the available power in the engine of the machine, gear ratios, rolling radius, weight on the drive wheels, total weight, properties of roadway surface, type of driver (wheel or crawler), tire size, pressure and tread, and on the percent grade.

Drawbar pull is the actual pulling force available at the drawbar of a tractor. It is usually measured in pounds of force. Drawbar pull is a function of engine torque, total gear reduction, rolling radius, and the coefficient of friction. If drawbar pull is measured when the tractor is in motion, the pull would also be a function of the rolling resistance.

Theoretical rimpull is the driving force supplied by the engine and measured at the point of contact between the tires and the ground surface. Theoretical rimpull is a function of the same variables as the drawbar pull less the coefficient of friction. For a given machine, rimpull will have a maximum assigned value that can be developed in each gear range. However, there may be times when not enough friction will be present between the tire and the surface to allow all the rimpull that the machine is capable of developing to be transmitted to the ground to move the machine and its load. The rimpull actually used — the actual rimpull or tractive effort — is a function also of the coefficient of friction. If enough friction can be developed between the wheel and the ground so that all the available rimpull can be transmitted through to the ground, the theoretical rimpull and the actual rimpull will be equal. This is usually the case with heavy equipment.

The coefficient of friction (coefficient of tractive efficiency) is the relationship between the weight on the tracks or drive wheels, and the friction developed between the road surface and the tires or tracks. This relationship is expressed as follows:

$$\text{Coefficient of friction} = \frac{\text{Drawbar pull in pounds}}{\text{weight on the track (pounds)}} \qquad (12)$$

For rubber-tired units the relationship is expressed as:

$$\text{C. F.} = \frac{\text{Rimpull (lbs.) at point of slippage}}{\text{Weight imposed on the tires of drive wheels}} \qquad (13)$$

The maximum drawbar pull of a Caterpillar D-7 tractor in first gear is 20,100 pounds at the rated engine speed. The weight of the machine is about 23,000 pounds. If slippage is on the point of occurring when 20,100 pounds is reached, the coefficient of friction is 20,100 pounds divided by 23,000 pounds which equals 0.875 or 87.5% of the weight on the tracks. This means that this machine, or a machine of like weight and track design, could pull 87.5% of its weight on the above road if the engine were capable of developing that much power.

Using the same machine, assume that road conditions were such that the coefficient of friction were .55. In this case the force available for pulling (as a scraper, rooter, or cat wagon) would be .55 x 23,000 = 12,650 lbs.

In the case of a rubber-tired unit the solution is similar. Assume that a given truck has a theoretical rimpull of 9,500 lbs, and that there are 20,000 lbs. on the drive wheels. If working on wet clay loam (see Table XII) the coefficient of friction would be .45. The maximum force available for pulling would be:

(.45)(20,000) = 9,000 lbs.

this means that 500 lbs. of the rimpull that the truck is capable of delivering cannot be used; to apply more than 9,000 lbs. of rimpull would cause the wheels to spin.

On the other hand, assume the truck was operating on dry loam which has a coefficient of friction of .55. The maximum force available for pulling would be:

(.55)(20,000) = 11,000 lbs.

This means that 11,000 lbs. of rimpull could be used before the wheels began to spin, and it also means that the engine will "lug down" under full load since the machine can only develop 9,500 pounds. All the rimpull that can be developed will be used and there will be no slippage.

Values for the coefficient of friction for rubber tires on different surfaces are given in Table XII. These factors, though approximate, are of practical importance. To increase these factors results in increases in equipment efficiency. The coefficients can be increased by better road maintenance, change in tread design, increasing bearing surface through larger tires and lower pressure, and by having more weight on the drive wheels.

The coefficient of friction for trucks on ordinary dirt roads is usually considered to be .55. The coefficient of friction for crawler tractors on dirt is from .60 upward. Crawler tractors develop more friction because the long lugs penetrate further into the road bed giving a greater friction bond.

Another term frequently used is drawbar horsepower. This is the time rate at which drawbar pull is used. One drawbar horsepower equals 375 lb. miles per hour. Mathematically:

$$DHP = \frac{D \times S}{375} \qquad (14)$$

where DHP = drawbar horsepower

D = speed in miles per hour

and 375 = the conversion factor.

A study of the equipment characteristics in Appendix G shows that the drawbar pull varies with the speed. This makes the consideration of drawbar horsepower necessary.

TABLE XII [25]

COEFFICIENTS OF FRICTION

Type of Roadway	Factor
Rough concrete	.88 to 1.00
Clay loam; dry	.50 to .58
Clay loam; wet	.40 to .49
Rutted clay loam	.40 to .44
Gravel road	.36
Rutted sandy loam	.20 to .35
Loose sand	.20 to .35
Packed snow	.20
Ice	.12

(25) Park, K. E., Principles of modern excavation and equipment, p. 32, Peoria, Ill., R. G. LeTourneau, Inc., 1942.

ROLLING RESISTANCE AND TIRE PENETRATION

Rolling resistance is an important, though frequently overlooked, factor in equipment performance. Rolling resistance, sometimes called "rolling friction" is that resistance which the surface of the ground offers to the movement of a wheel or tire.

The three cases that contribute to rolling resistance are illustrated in Figure 52. In "A" the surface has been depressed, in "B" the tire has been deformed, and in "C" a small obstruction offers resistance. Any combination of these three cases can exist.

If "P" in Figure 52 is a horizontal force which causes the center of the wheel to move with a constant velocity, the reaction "R" of the surface will pass through some point "B." Since the velocity of the wheel is constant, the three forces, "P," "R," and "W" are in equilibrium and so the reaction, "R," of the surface on the wheel must pass through point "O," the center of the wheel. Taking moments about "B" gives

$$M_B = (W)(b) - (P)(h) = 0 \qquad (15)$$

Solving equation (15) for "P", it becomes

$$P = \frac{Wb}{h} \qquad (16)$$

Since the distance "h" is very nearly equal to the radius of the object, the radius "r" will be substituted for "h". Equation (16) becomes:

$$P = \frac{Wb}{r} \qquad (17)$$

The horizontal component of the reaction "R" is equal to "P"

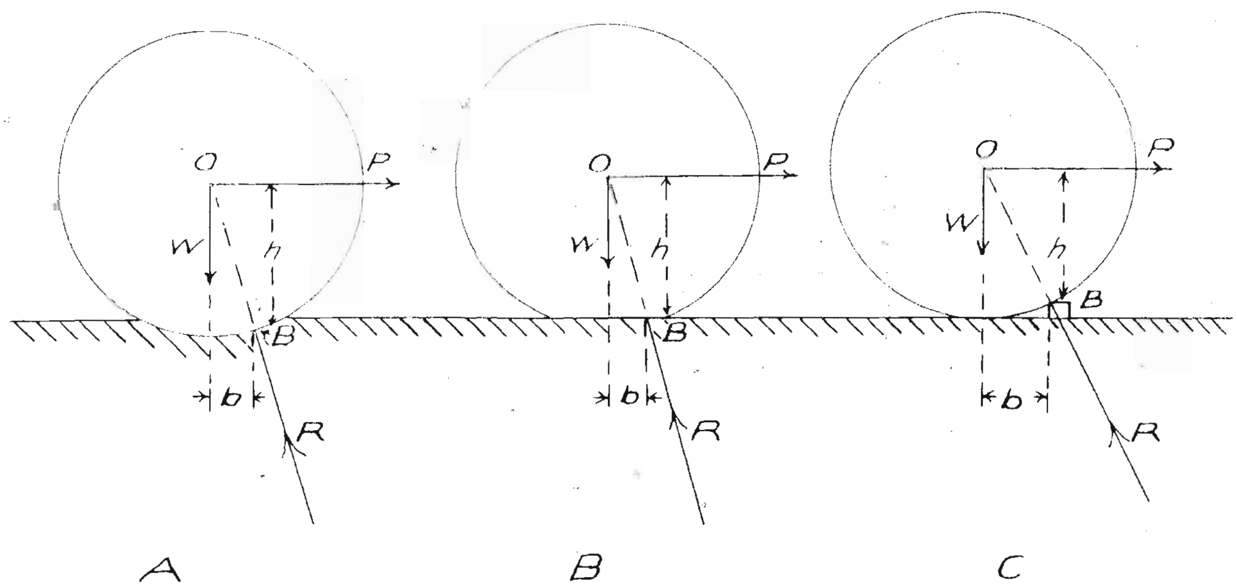


Figure 52. Causes of rolling resistance.

and it is called the "rolling friction" or rolling resistance. The distance "b" is called the coefficient of rolling resistance. It is a linear quantity and generally expressed in inches. Some coefficients of rolling resistance are given in Table XIII. This matter has not been thoroughly investigated and so the coefficients should be used with caution.

TABLE XIII (26)

COEFFICIENTS OF ROLLING RESISTANCE

(Due to Coulomb and Goodman)

Steel on steel	0.007 to 0.015
Steel on wood	0.06 to 0.10
Steel on macadam road	0.05 to 0.20
Steel on soft ground	3.0 to 5.0
Pneumatic tires on good road	0.02 to 0.22
Pneumatic tires on mud road	0.04 to 0.06
Solid rubber tire on good road	0.04
Solid rubber tire on mud road	0.90 to 0.11

Rolling resistance is dependent on the firmness and freedom from undulations of the roadbed. Resistance is increased by tire penetration, compaction and displacement of material making up the road, by added weight, and by sidewall tire action.

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(26) Seely, F. B., and Ensign, N. E., Analytical mechanics for engineers, p. 141, New York, Wiley, 1946.

Rolling resistance is most commonly designated in pounds per gross ton of vehicle weight. Less commonly rolling resistance is designated pounds per thousand pounds of gross weight, as a percentage of grade resistance per ton (20 lb. of road resistance per ton is equivalent to 1% of adverse grade per gross ton), or as a percent of the gross vehicle weight.

Table XIV gives a list of representative rolling resistances. It must be remembered that these resistances are not fixed. The values would vary somewhat depending upon such things as the size and type of tire used. However, these values are accurate enough to use in making working calculations.

Rolling resistance for crawler-mounted machines is about 150 pounds per gross ton of vehicle weight. This large resistance is due in part to the many hinges in the crawlers and to the penetration of the lugs.

To put a knowledge of rolling resistance into practice, the basic engineering formulas that follow will be helpful:

$$TGR = (TR)(OAR) \qquad (18)$$

$$MPH = \frac{(\text{Engine RPM})(RR)}{(168)(TGR)} \qquad (19)$$

$$ET = \frac{(HP)(5252)(12)}{(RPM)} \qquad (20)$$

$$RP = \frac{(.90)(ET)(TGR)(ME)}{(RR)} \qquad (21)$$

$$DBP = \frac{(.90)(ET)(TGR)(ME)}{(RR)} - RoRi \text{ (in lbs.)} \qquad (22)$$

$$GA = \frac{(.90)(ET)(TGR)(ME)(100)}{(RR)(GVW)} - RoRi \text{ (in \% of GVW)} \qquad (23)$$

TABLE XIV [27]

ROLLING RESISTANCES

Type of Road Surface	Rolling Resistance as Pounds per Ton of Gross Vehicle Weight
1. Smooth, hard, dry dirt and gravel. Well maintained. Free of loose material	40 lbs.
2. Dry dirt and gravel. Not firmly packed. Some loose material	60 lbs.
3. Soft unplowed dirt or poorly maintained dry dirt, rutted surface	80 lbs.
4. Wet muddy surface on firm base	80 lbs.
5. Soft, plowed dirt or unpacked dirt fills	160 lbs.
6. Loose sand or gravel	200 lbs.
7. Deeply rutted, sticky or muddy, soft spongy base	320 lbs.

(27) Euclid Road Machinery Company, Estimating production and costs of material movement with Euclids, Form No. 350-R, p. 5, Cleveland, Ohio, 1946.

In the foregoing formulae;

TGR is total gear reduction,

TR is the transmission ratio,

OAR is the overall axle ratio,

MPH is the vehicle speed in miles per hour,

RPM is engine revolutions per minute,

RR is tire rolling radius,

ET is engine torque in inch-pounds,

HP is engine horsepower,

RP is theoretical rimpull,

ME is mechanical efficiency (85% to 90%),

DBP is drawbar pull in pounds,

RoRi is rolling resistance in the designated units, and

GVW is gross vehicle weight in tons.

In the above, the (.90) is a correction factor to give an average value to engine torque since high speed truck engines are not always running at the rated speed and, therefore, the maximum torque. Figures 53 and 54 show the torque characteristics of high speed gasoline and Diesel engines. Figure 55 gives the characteristics for a low speed Diesel engine such as is used in crawler tractors. On the low speed Diesel the torque curve is flatter and so the constant is neglected.

Solving equation (19) for TGR gives

$$TGR = \frac{(RPM)(RR)}{(168)(MPH)} \qquad (24)$$

Substituting equation (24) for TGR in equation (21) gives

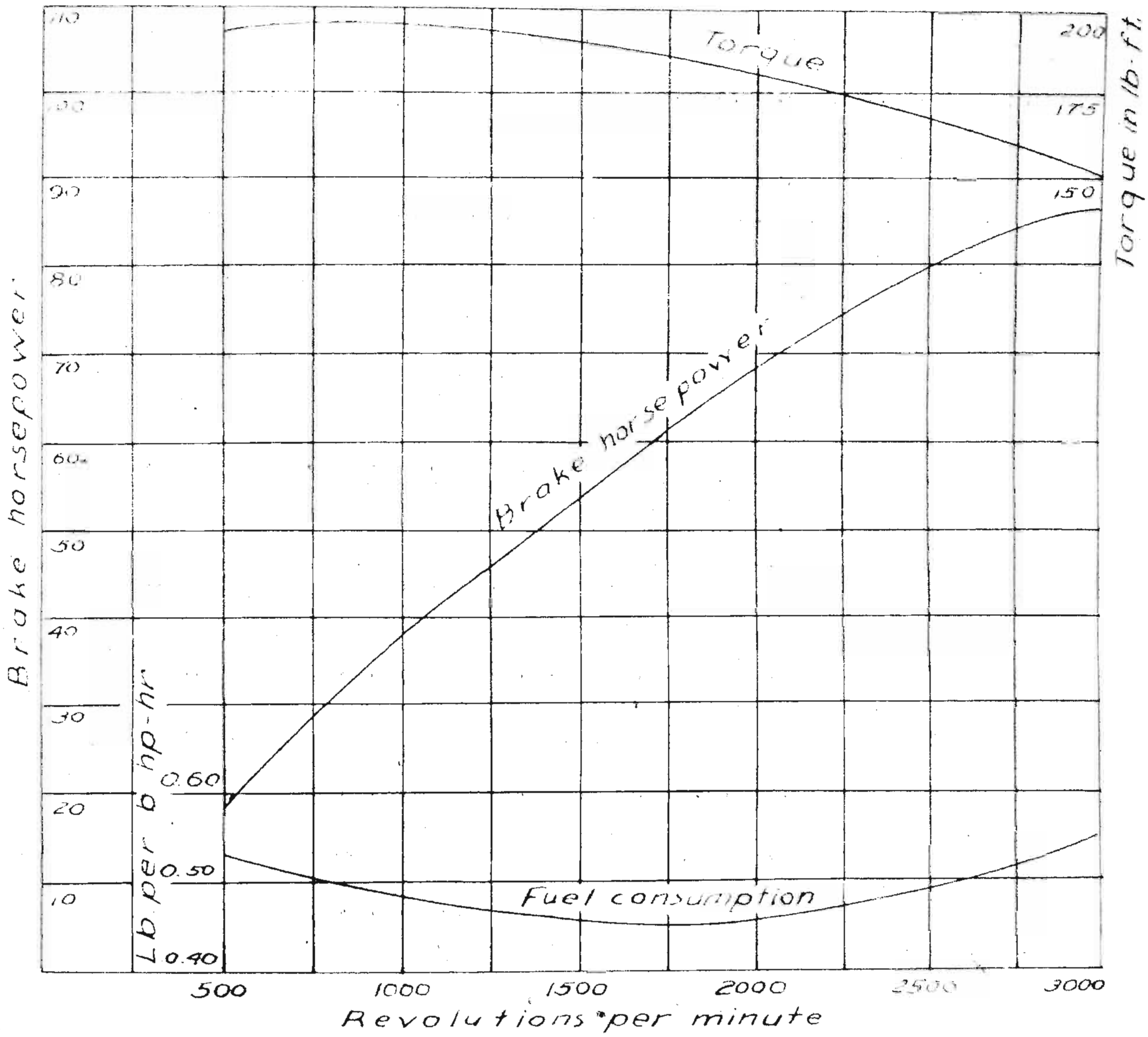

Figure 53.(28) Typical performance curves of a six-cylinder four-stroke cycle high-speed Diesel engine, 278 cu. in. piston displacement, variable torque and speed.

(28) Severns, W. H., and Degler, H. E., Steam, air, and gas power. p. 470, New York, Wiley, 1947.

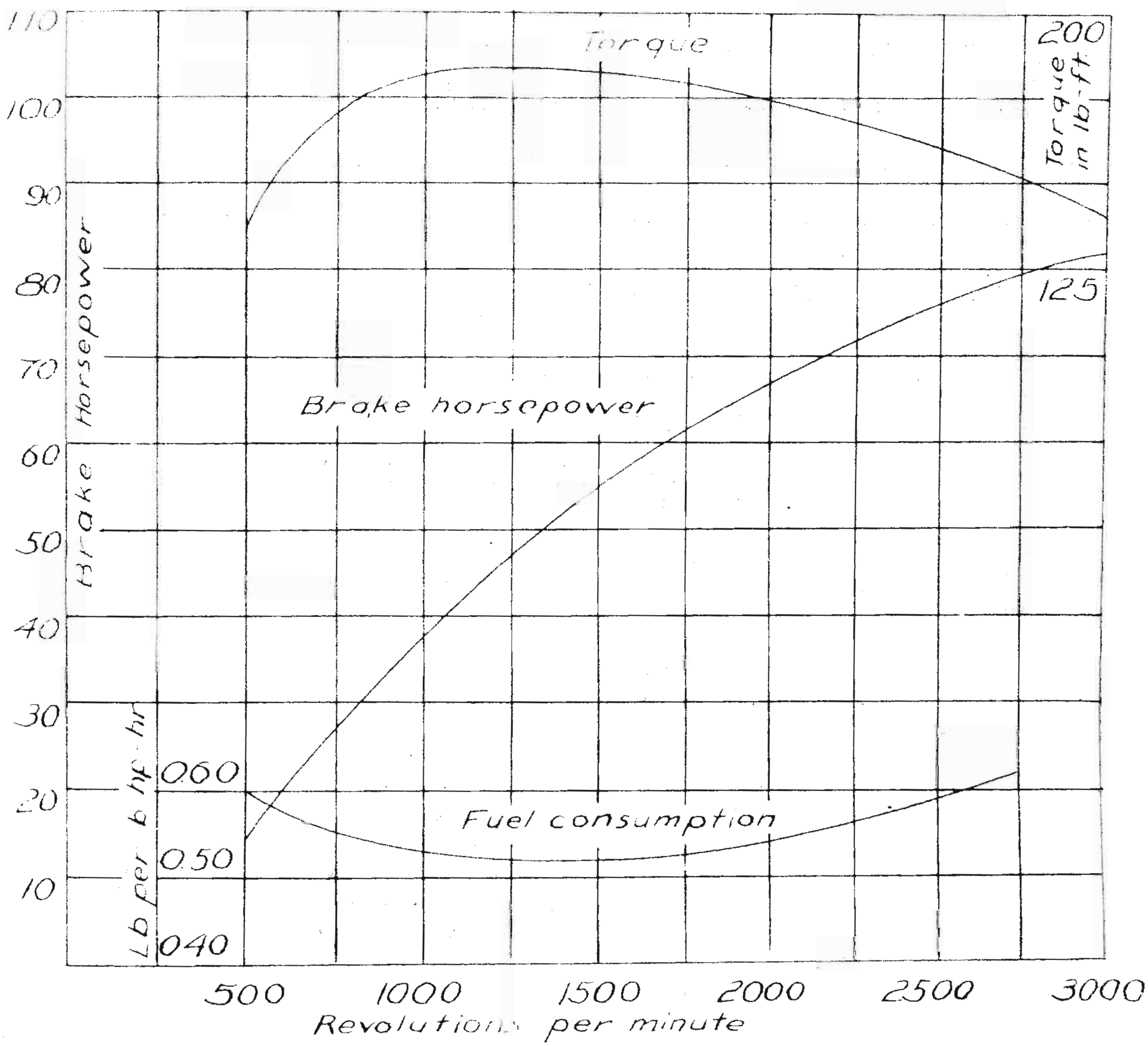

Figure 9A.(29) Typical performance curves of a six-cylinder gasoline engine, 282 cu. in. piston displacement, variable torque and speed.

(29) Severns, W. H., and Degler, H. E., Steam, air, and gas power, p. 468, New York, Wiley, 1947.

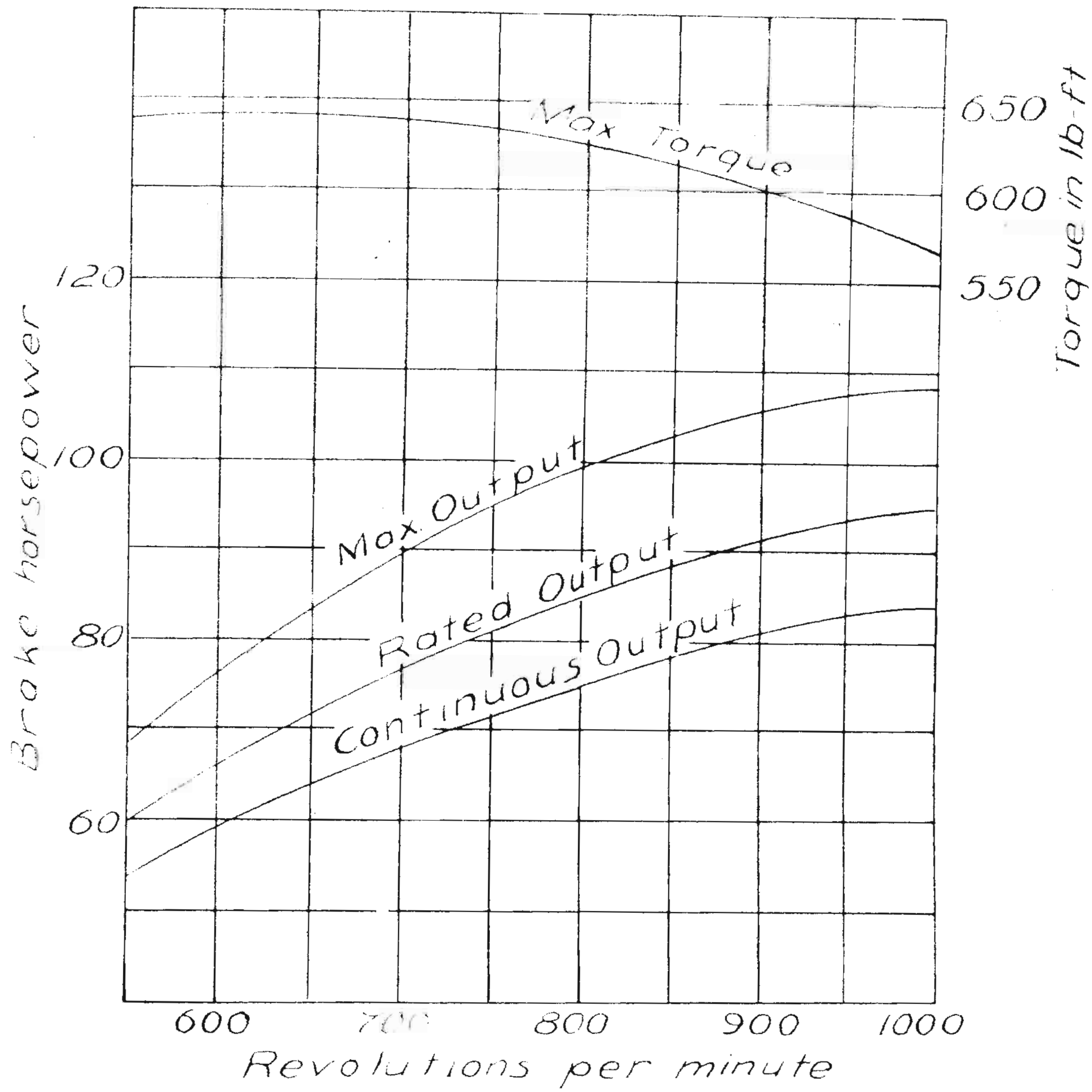

Figure 55.[30] Characteristics of a low-speed Diesel.

(30) After Caterpillar Tractor Catalogue-Ch, Form 5350, p. 21, Peoria, Caterpillar Tractor Company.

$$RP = \frac{(.90)(ET)(RPM)(RR)(ME)}{(268)(MPH)(RR)} \quad (25)$$

which reduces to

$$RP = \frac{(.90)(ET)(RPM)(ME)}{(168)(MPH)} \quad (26)$$

Substituting equation (20) for ET in equation (26) gives

$$RP = \frac{(.90)(12)(HP)(5252)(RPM)(ME)}{(RPM)(168)(MPH)} \quad (27)$$

which reduces to

$$RP = \frac{(.90)(HP)(375)(ME)}{(MPH)} \quad (28)$$

The more useful form of equation (28) is found by solving for MPH to give

$$MPH = \frac{(.90)(HP)(375)(ME)}{(RP)} \quad (29)$$

To illustrate the use of equation (29) and to illustrate the importance of lowering the rolling resistance of a road through proper road construction and maintenance, a typical case will be shown with different rolling resistances assumed.

Example 10

The conditions of the problem are as follows:

Truck HP = 150. When loaded the truck weighs 66,000 lbs., or 33 tons. When empty the truck weighs 33,000 lbs. Payload is 33,000 lbs. Payload @ 3,000 lbs. per bank cu. yd. = $\frac{33,000}{3,000}$ = 11 cu. yd.

Cost of owning and operating a truck per hour is $5.00. Coefficient of friction is .6. Weight on drive tires when loaded is 50,000 lbs. There are no grades to consider.

A 1-mile haul loaded. Return at top speed of truck which equals 37 MPH. Dump time is .5 min. The loading time at the shovel is 2 minutes. The costs and performances are to be analyzed for rolling resistances of 40 lbs. per ton, 60 lbs. per ton, 80 lbs. per ton, 100 lbs. per ton, and 160 lbs. per ton.

Analysis:

Case (1), Rolling resistance equals 40 lbs. per ton.

(33 tons, gross)(40 lbs./ton) = 1320 lbs. rimpull required to overcome rolling resistance.

Case (2), Rolling resistance equals 60 lbs. per ton.

(33 tons)(60 lbs./ton) = 1980 lbs. rimpull required.

Case (3), Rolling resistance equals 80 lbs. per ton.

(33 tons)(80 lbs./ton) = 2640 lbs. rimpull required.

Case (4), Rolling resistance equals 100 lbs. per ton.

(33 tons)(100 lbs./ton) = 3300 lbs. rimpull required.

Case (5), Rolling resistance equals 160 lbs. per ton.

(33 tons)(160 lbs./ton) = 5280 lbs. rimpull required.

None of these rimpull requirements are in excess of what the machine can develop or transmit through the ground since (.6)(50,000) = 30,000 lbs. rimpull that could be transmitted.

The haul speed can be found by using equation (29).

Case (1)

$$MPH = \frac{(.90)(150)(375)(.85)}{1320} = 32.6.$$

Case (2)

$$MPH = \frac{(.90)(150)(375)(.85)}{1980} = 21.8.$$

Case (3)

$$MPH = \frac{(.90)(150)(375)(.85)}{2640} = 16.6.$$

Case (4)

$$MPH = \frac{(.90)(150)(375)(.85)}{3300} = 13.05$$

Case (5)

$$MPH = \frac{(.90)(150)(375)(.85)}{5280} = 8.16$$

On the basis of the above speeds the complete haul cycle can be found. No allowances are made for acceleration.

Case (1)

$$\frac{(1\ mi.)(hr.)(60\ min.)}{(32.6\ mi.)(hr.)} = 1.84\ min.,\ \text{haul time, loaded.}$$

$$\frac{(1\ mi.)(hr.)(60\ min.)}{(37\ mi.)(hr.)} = 1.62\ min.,\ \text{return time.}$$

Cycle = 1.84 min. + 0.5 min. + 2 min. + 1.62 min. = 5.96 min.

Case (2)

$$\frac{(1\ mi.)(hr.)(60\ min.)}{(21.8\ mi.)(hr.)} = 2.75\ min.,\ \text{haul time.}$$

Cycle = 2.75 + 1.62 + 0.5 + 2 = 6.87 min.

Case (3)

$$\frac{(1 \text{ mi.})(\text{hr.})(60 \text{ min.})}{(16.6 \text{ mi.})(\text{hr.})} = 3.61 \text{ min., haul time.}$$

Cycle = 3.61 + 1.62 + 0.5 + 2 = 7.73 min.

Case (4)

$$\frac{(1 \text{ mi.})(\text{hr.})(60 \text{ min.})}{(13.05 \text{ mi.})(\text{hr.})} = 4.6 \text{ min., haul time.}$$

Cycle = 4.6 - 1.62 - 0.5 - 2 = 8.72 min.

Case (5)

$$\frac{(1 \text{ mi.})(\text{hr.})(60 \text{ min.})}{(8.16)(\text{hr.})} = 7.35 \text{ min., haul time.}$$

Cycle = 7.35 + 1.62 + 0.5 + 2 = 11.47 min.

Yards per 50-minute hour can now be calculated. A 50-minute hour is usually used in haulage calculations to compensate for unavoidable delays.

Case (1)

$$\frac{(50 \text{ min.})(\text{load})(11 \text{ cu. yd})}{(\text{hr.})(5.96 \text{ min.})(\text{load})} = 92.2 \text{ cu. yd/hr.}$$

Case (2)

$$\frac{(550 \text{ cu. yd})}{(6.87 \text{ hr.})} = 80 \text{ cu. yd./hr.}$$

Case (3)

$$\frac{(550 \text{ cu. yd.})}{(7.73 \text{ hr.})} = 71.1 \text{ cu. yd./hr.}$$

Case (4)

$$\frac{(550 \text{ cu. yd.})}{(8.72 \text{ hr.})} = 63.1 \text{ cu. yd./hr.}$$

Case (5)

$$\frac{(550 \text{ cu. yd.})}{(11.4 \text{ hr.})} = 47 \text{ cu. yd./hr.}$$

The cost per cubic yard for hauling then equals the cost of owning and operating the truck per hour divided by the number of cubic yards the truck can haul per hour.

Case (1)

$$\frac{(500¢)(\text{hr.})}{(\text{hr.})(92.2\text{ cu.yd.})} = \frac{500¢}{92.2\text{ cu. yd.}} = 5.42¢/\text{cu. yd.}$$

Case (2)

$$\frac{(500¢)}{(80\text{ cu. yd.})} = 6.25¢/\text{cu.yd.}$$

Case (3)

$$\frac{(500¢)}{(71.1\text{ cu. yd.})} = 7.03¢/\text{cu.yd.}$$

Case (4)

$$\frac{(500¢)}{(63.1\text{ cu.yd.})} = 7.92¢/\text{cu.yd.}$$

Case (5)

$$\frac{(500¢)}{(47\text{ cu. yd.})} = 10.6¢/\text{cu.yd.}$$

Assume that it was necessary to move 800 cu. yds. each hour. The number of trucks necessary in each case would be 800 cu. yds. an hour divided by the capacity per hour per truck.

Case (1)

$$\frac{(800\text{ cu. yd.})(\text{hr.})(\text{truck})}{(\text{hr.})(92.2\text{ cu.yd.})} = \frac{(800\text{ trucks})}{(92.2)} =$$

8.7 trucks = 9 trucks.

Case (2)

$$\frac{(800\text{ trucks})}{(80)} = 10\text{ trucks}$$

Case (3)

$$\frac{(800\text{ trucks})}{(71.1)} = 11.2\text{ trucks} = 12\text{ trucks.}$$

Case (4)

$$\frac{(800\text{ trucks})}{(63.1)} = 12.7\text{ trucks} = 13\text{ trucks.}$$

Case (5)

$$\frac{(800\text{ trucks})}{(47)} = 17\text{ trucks}$$

If each truck cost \$13,000.00, the original investment would be the number of trucks necessary times the cost per truck.

Case (1)

$$\frac{(9\text{ trucks})(\$13,000)}{(\text{truck})} = \$117,000.00$$

Case (2)

$$(10)(\$13,000) = \$130,000.00$$

Case (3)

$$(12)(\$13,000) = \$156,000.00$$

Case (4)

$$(13)(\$13,000) = \$169,000.00$$

Case (5)

$$(17)(\$13,000) = \$221,000.00$$

To summarize the above information:

	Case 1	Case 2	Case 3	Case 4	Case 5
Rolling resistance. . . (Lbs. per ton)	40	60	80	100	160
Haul speed . . . . (Miles per hour)	32.6	21.8	16.6	13.05	8.16
Cost per cu. yd . . . . (Cents)	5.42	6.25	7.03	7.92	10.6
No. of trucks to haul 800 cu. yd. per hour. .	9.0	10.0	12.0	13.0	17.0
Investment in trucks to haul 800 cu.yd. per hour.	117.0	130.0	156.0	169.0	221.0

Actually in the above example the costs would become even greater proportionally as the rolling resistance increased since there would be greater maintenance costs due to hauling on poorer roads.

In the next chapter the effect of the rolling resistance on the grade ability of the truck will be illustrated.

An expression for MPH in terms of DBP will also prove useful. Since the term to the right of the equal sign in equation (21) is identical to the first term to the right of the equal sign in equation (22), it follows that equation (28) can be substituted for the first term to the right in equation (22). This substitution gives

$$DBP = \frac{(.90)(HP)(375)(ME)}{MPH} \; \underline{6} \; RoRi(lbs.). \qquad (30)$$

Solving equation (30) for MPH gives

$$MPH = \frac{(.90)(HP)(375)(ME)}{(DBP) - (RoRi)}, \qquad (31)$$

where (RoRi) is the total rolling resistance of the tractor in pounds.

The minimum rolling resistance found in practice is 40 pounds per ton. This is found only under the most favorable conditions which do not commonly exist. The increase in rolling resistance beyond this amount is due to the penetration or displacement of the ground by the tire, to irregularities of the surface, and to deformation of the tire.

It is evident that if a tire under a load deflected so as to carry a load of 45 pounds per square inch of bearing surface on a soil that had a bearing capacity of over 45 pounds per square inch, there would be no penetration of the soil by the tire. The same tire and load would penetrate into a soil that had a bearing capacity under 45 pounds per square inch. In the latter case the tire would penetrate until enough bearing area had been formed along the sidewalls and circumference of the tire. Also in the latter case, if pressure in the tire had been lower, the tire would have deflected more to create a greater bearing area which enlarged area would have caused a decrease in bearing pressure. The decrease in bearing pressure might result in no penetration.

In Figure 56 it can be seen that if the sustaining capacity of sidewalls is neglected, the weight "W" will all be transmitted to the ground through the confined fluid. The pressure of a confined fluid is equal on all parts of the interior surface of the confining vessel which is, in this case, the tube. It follows then that the area of contact between the tire and the ground should be equal to the weight of tire and load divided by the tire pressure. To allow for the sustaining value of the sidewalls of the tire,

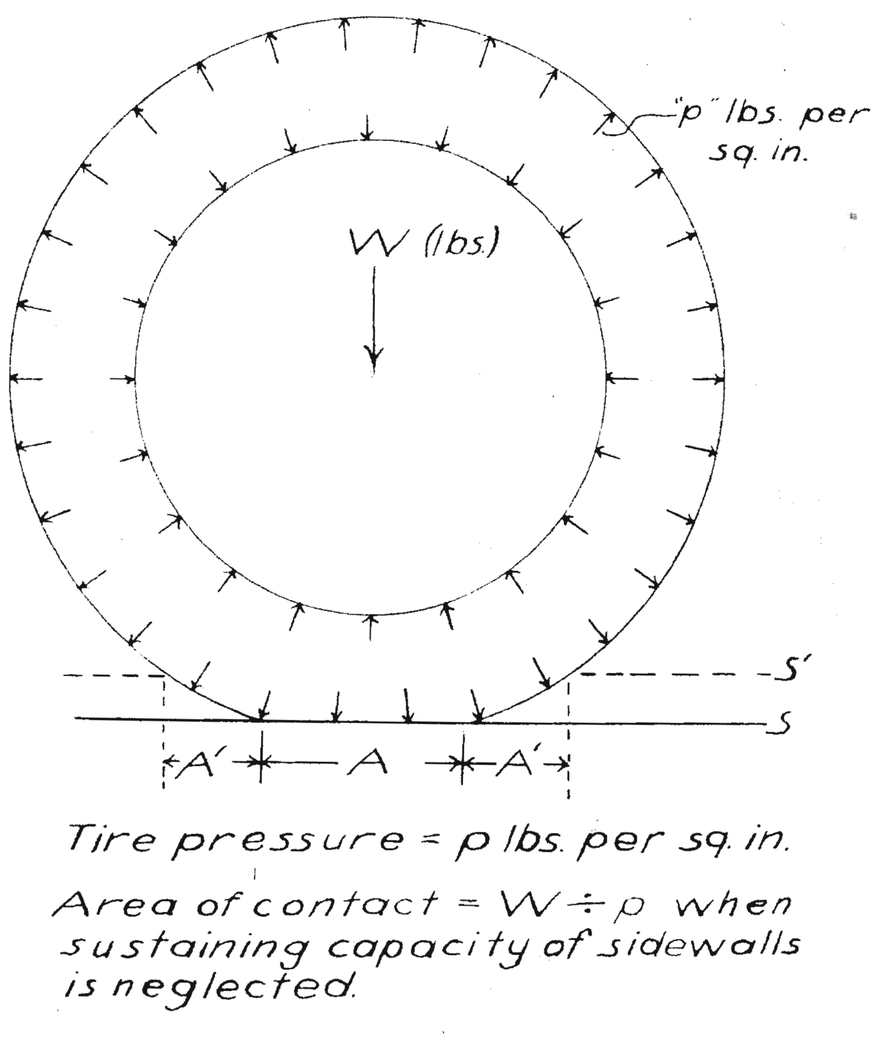

Figure 34. The effect of tire pressure on area of contact.

Park[31] has modified this so that the bearing area will be 0.90

(31) Park, K. E., Principles of modern excavation and equipment, p. 45, Peoria, Ill., R. G. LeTourneau, Inc., 1942.

of the theoretical area as follows:

$$\text{Bearing area in square inches} = \frac{(.90)(\text{weight on tire, lbs.})}{(\text{tire pressure, lbs/sq.in.})} \quad (32)$$

<u>Example 11</u>

If a tire had 40 pounds of pressure per square inch and carried a 10,000 lb. load, would it penetrate into a roadbed having a bearing capacity of 35 pounds per square inch?

If the tire pressure were reduced to 30 pounds per square inch, would there be penetration?

$$\text{Bearing area} = \frac{(.90)(10,000\text{ lb.})(\text{sq. in.})}{40\text{ lb.}} = 225\text{ sq. in.}$$

$$\frac{(10,000\text{ lb.})}{(225\text{ sq. in.})} = 44.4\text{ lb. per sq. in.}$$

Since the bearing pressure is greater than the bearing capacity, there would be penetration, if the tire had 40 pounds of pressure per square inch.

$$\text{Bearing area} = \frac{(.90)(10,000\text{ lb})(\text{sq. in.})}{(30\text{ lb.})} = 300\text{ sq. in.}$$

$$\frac{(10,000\text{ lb.})}{(300\text{ sq. in.})} = 33\text{-}1/3\text{ lb. per sq. in.}$$

Since the bearing pressure is less than the bearing capacity of the soil, there would be no penetration if the tire had 30 pounds of pressure per square inch.

It has been estimated[32] that for every inch of tire pene-

(32) Park, Op. Cit., p. 46.

tration, rolling resistance is increased by 30 pounds per gross ton. Thus if the load per wheel is 2 tons, and if the tire penetrates 1 inch, 60 lbs. of rolling resistance are set up. If the tire penetrates two inches, 120 pounds of rolling resistance are set up. Three inches of penetration result in 180 pounds of resistance.

To decrease tire penetration by increasing tire size and decreasing tire pressure to set up a greater bearing area and thus lessen penetration results in a net decrease in rolling resistance. There is an additional rolling resistance set up through tire deflection and sidewall action, but this additional resistance is almost negligible when compared to the 30 pounds per gross ton per inch of penetration.

The effects of impact loads on the truck and on penetration are less with low pressure tires than with high pressure tires. However, low pressure tires do have a somewhat shorter life than do high pressure tires.

Figure 57 illustrates the results of increasing tire size and decreasing pressure to minimize tire penetration. This illustration is based on a load of 12,000 pounds per tire on a surface which has a bearing capacity of 27 pounds per square inch. Here 90 horsepower gave a speed of 8.1 miles per hour with a small tire and high pressure. With the same load, a large tire and low pres-

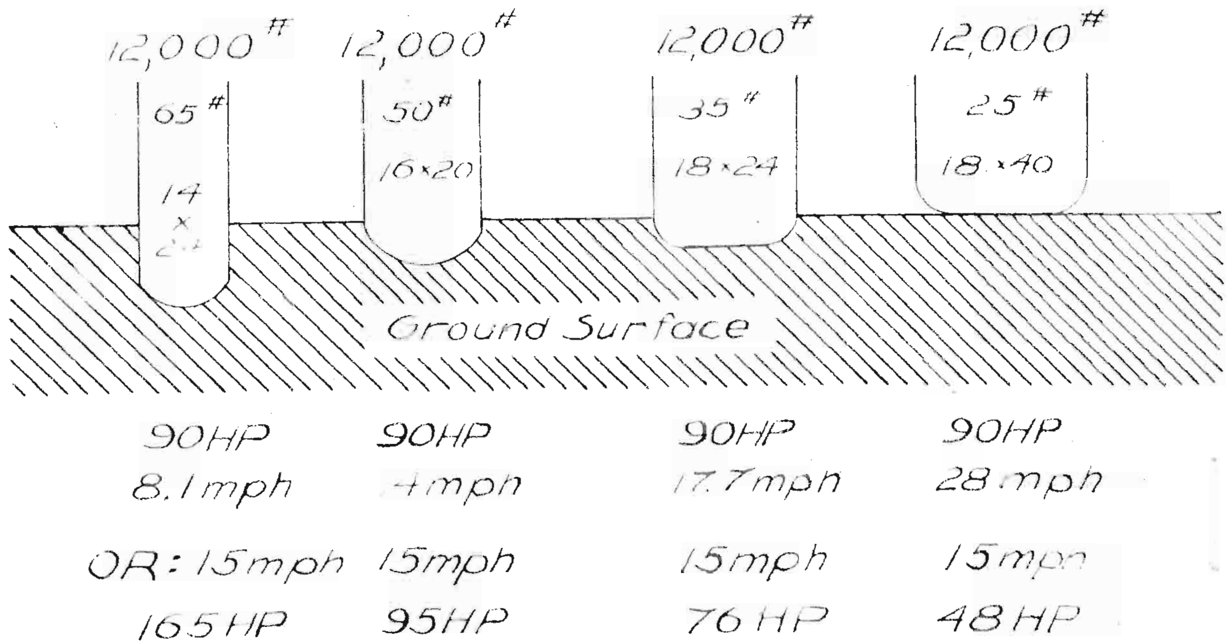

Figure 57. Effect of tire penetration on speed and power consumption.(33)

(33) Park, K. E., and Evans, R. D., How to make full use of Tournapull speed: The Co-operator, Vol. 5, No. 3, p. 7, April, 1943.

sure would result in 28 miles per hour with the same horsepower. Or, to attain 15 miles per hour, 165 horsepower is required for the small tire and 48 horsepower is required for the large tire.

On well-surfaced roads there is little danger of tire penetration. On such roads it is advisable to use high pressure tires since the deflection of low pressure tires is a cause of rolling resistance. Also, flexing of low pressure tires results in heating and consequent deterioration of the tire.

The best rule is to suit the tire pressure to the conditions that prevail. If only short stretches of bad road are present on otherwise good roads, a tire of from 40 to 55 pounds pressure would be indicated. If the reverse conditions prevail, a low pressure tire would be indicated. The low pressure tires vary from 25 to 40 pounds of pressure per square inch. Or, again, a compromise pressure might prove economical in tire wear and vehicle performance.

To aid in keeping rolling resistance to a minimum, equipment should have wheels that track. Older model scrapers had a narrower guage axle for the front wheels than for the back. With this arrangement each wheel must cut its own path. Many machines still have smaller tires on the front than they do on the back. Here the front wheels cut a narrow path and the back wheels must widen the path.

The present trend, and wisely so, is to have the same size tires on front and back with an even or nearly even weight distribution, and both front and back have the same gauge. All

All roads for permanent hauls should be constructed and maintained to keep rolling resistance at a minimum. Road maintenance is not necessary for crawler-mounted machines. In most cases crawler-mounted machines will improve roads over which only they can move by compaction. Original rolling resistance of these machines is so high and the power available so great, that it is not economical to improve roads for minor decreases in rolling resistance.

GRADE RESISTANCE AND ECONOMICAL GRADES

Grade resistance is another important factor in equipment performance.

Assume that the load "T," in Figure 58, is mounted on wheels that are frictionless. This assumption can safely be made since the friction in the wheels and between the wheels and the roadway has already been taken into consideration as rolling resistance.

Since the legs of the angles are mutually perpendicular, the angles "S" are equal. The compnent of the weight "T" acting parallel to the inclined plane is "P." "P" is equal to (T)(sin S). P is the force acting parallel to the plane and will produce movement down the plane unless acted upon by and equal and opposite force.

For small angles the sine of the angle is very nearly equal to the tangent of the angle. If angle "S" is expressed as being 1% grade, the tangent of the angle is 1/100 or 0.01. If "T" equals 1 ton, P then equals (.01) (2,000 lb./ton) equals 20 lb. per ton per percent grade. So for most calculations grade resistance can be considered as being equal to 20 lb. per ton per percent grade.

The coefficient of tractive efficiency is a function of component "V". "V" in turn is equal to (T)(cos S). For the grades used in haulage, cos S is very nearly equal to one. This being true, the effect of a grade on the coefficient of friction need

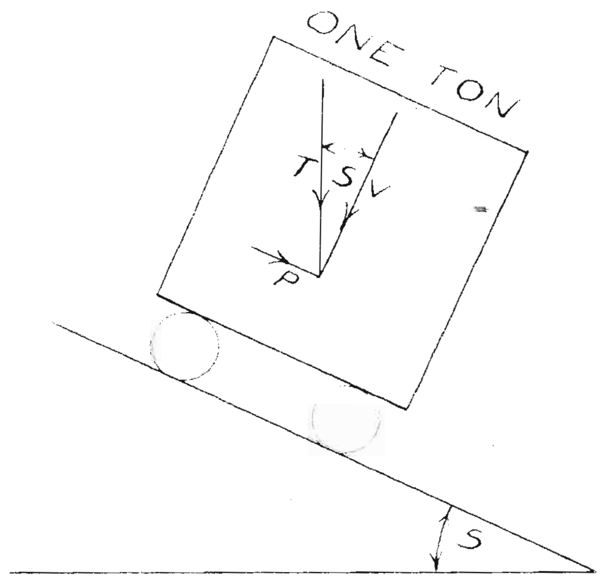

Figure 58. Derivation of value of grade resistance.

not be considered.

If movement impends upgrade, grade resistance is a consumer of drawbar pull and rimpull. If movement is down grade, this parallel component adds to the drawbar pull and rimpull. In the latter case it is actually a grade pull rather than a grade resistance and it may be used to do work.

Example 12

What is the grade resistance of 5 tons on a 10 percent grade?

$$\frac{(5 \text{ tons})(10 \text{ percent})(20 \text{ lb.})}{(\text{ton})(\text{percent})} = 1000 \text{ lb. of grade resistance.}$$

Equation (23) is one expression of the grade ability of equipment. To arrive at a more usable form equation (21) is solved for the rolling radius as follows:

$$RR = \frac{(.90)(ET)(TGR)(ME)}{RP} \qquad (33)$$

Equation (33) is substituted for RR in equation (23) to give

$$GA = \frac{(.90)(ET)(TGR)(ME)(100)(RP)}{(.90)(ET)(TGR)(ME)(GVW)} - (\text{RoRi in \% of GVW}) \qquad (34)$$

Equation (34) reduces to

$$GA = \frac{(100)(RP)}{(GVW)} - (\text{RoRi in \% of GVW}). \qquad (35)$$

But

$$\text{RoRi in \% GVW} = \frac{\dfrac{(\text{GVW in lb.})(\text{ton})}{(2000 \text{ lb.})} \dfrac{(\text{RoRi in lb.})}{(\text{ton})}}{(\text{GVW in lbs.})}$$

which reduces to

$$\text{RoRi in \% GVW} = \frac{(100)(\text{RoRi})}{(2000)} . \qquad (36)$$

Substituting equation (36) for the last term in equation (35) gives

$$\text{GA} = \frac{(100)(\text{RP})}{(\text{GVW})} - \frac{(\text{RoRi})(100)}{(2000)} \qquad (37)$$

Equation (37) reduces to

$$\text{Ga} = (100)\left[\frac{(\text{RP})}{(\text{GVW})} - \frac{(\text{RoRi})}{(2000)}\right] \qquad (38)$$

To put equation (38) to use, it is necessary to know the rimpull. This can be obtained from the manufacturers specifications or computed by using equation (21).

In the example which follows, the same data will be used which was used in example 10 on rolling resistances. In addition to those data will be the rimpull in each gear as specified by the manufacturer.

<u>Example 13</u>

Truck weight = 33,000 lb.

Truck load = 33,000 lb.

Gross weight = 66,000 lb.

Rimpull in 1st gear @ 3.4 mph. = 13,700 lb.

Rimpull in 2nd gear @ 6.84 mph. = 6,850 lb.

Rimpull in 3rd gear @ 12.7 mph. = 3,700 lb.

Rimpull in 4th gear @ 22.3 mph. = 2,100 lb.

Rimpull in 5th gear @ 35.2 mph. = 1,360 lb.

None of these rimpulls are in excess of the tractive ability.

Using the above data, find the grade ability of the truck in each gear range for rolling resistances of 40, 60, 80, 100 and 160 pounds per ton.

Case (1), Rolling resistance equals 40 pounds per ton.

In 1st gear

$$GA = (100)\left(\frac{13,700}{66,000} - \frac{40}{2000}\right) = 18.8\%$$

In 2nd gear

$$GA = (100)(6,850/66,000 - 40/2,000) = 8.4\%$$

In 3rd gear

$$GA = (100)(3,700/66,000 - 40/2,000) = 3.6\%$$

In 4th gear

$$GA = (100)(2,100/66,000 - 40/2,000) = 1.18\%$$

In 5th gear

$$GA = (100)(1,360/66,000 - 40/2,000) = 0.06\%$$

Case (2), Rolling resistance equals 60 pounds per ton.

In 1st gear

$$GA = (100)(13,700/66,000 - 60/2,000) = 17.8\%$$

In 2nd gear

$$GA = (100)(6,850/66,000 - 60/2,000) = 7.4\%$$

In 3rd gear

$$GA = (100)(3,700/66,000 - 60/2,000) = 2.6\%$$

In 4th gear

$$GA = (100)(2,100/66,000 - 60/2,000) = 0.18\%$$

In 5th gear

$$GA = (100)(1,360/66,000 - 60/2,000) = -0.94\%$$

A negative grade as immediately above indicates that the vehicle must be favored by the grade.

Case (3). Rolling resistance equals 80 lb. per ton.

In 1st gear

$$GA = (100)(13,700/66,000 - 80/2,000) = 16.8\%$$

In 2nd gear

$$GA = (100)(6,850/66,000 - 80/2,000) = 6.4\%$$

In 3rd gear

$$GA = (100)(3,700/66,000 - 80/2,000) = 1.6\%.$$

In 4th gear

$$GA = (100)(2,100/66,000 - 80/2,000) = -0.82\%.$$

In 5th gear

$$GA = (100)(1,360/66,000 - 80/2,000) = -1.94\%$$

Case (4). Rolling resistance equals 100 lb. per ton.

In 1st gear

$$GA = (100)(13,700/66,000 - 100/2,000) = 15.8\%$$

In 2nd gear

$$GA = (100)(6,850/66,000 - 100/2,000) = 5.4\%$$

In 3rd gear

$$GA = (100)(3,700/66,000 - 100/2,000) = 0.6\%$$

In 4th gear

$$GA = (100)(2,100/66,000 - 100/2,000) = -1.82\%$$

In 5th gear

$$GA = (100)(1,360/66,000 - 100/2,000) = -2.94\%$$

Case (5). Rolling resistance equals 160 lb. per ton.

In 1st gear

$$GA = (100)(13,700/66,000 - 160/2,000) = 12.8\%$$

In 2nd gear

$$GA = (100)(6,850/66,000 - 160/2,000) = 2.4\%$$

In 3rd gear

$$GA = (100)(3,700/66,000 - 160/2,000) = -2.4\%$$

In 4th gear

$$GA = (100)(2,100/66,000 - 160/2,000) = -4.82\%$$

In 5th gear

$$GA = (100)(1,360/66,000 - 160/2,000) = -5.94\%$$

To summarize the above results:

	Case 1	Case 2	Case 3	Case 4	Case 5
RoRi	40	60	80	100	160
1st gear	18.8 %	17.8 %	16.8 %	15.8 %	12.8 %
2nd gear	8.4 %	7.4 %	6.4 %	5.4 %	2.4 %
3rd gear	3.6 %	2.6 %	1.6 %	0.6 %	- 2.4 %
4th gear	1.18%	.18%	-0.82%	-1.82%	- 4.82%
5th gear	0.06%	-0.94%	-1.94%	-2.94%	- 5.94%

This summary clearly illustrates another benefits to be gained by decreasing rolling resistance: i.e., the increases possible in the grade abilities.

Equation (38) when solved for RoRi yields

$$RoRi = (2000)(RP/GVW - GA/100). \qquad (39)$$

In this form the equation can be used to find the rolling resistance of a road if the other variables are known.

Example 14

Super "C" Turnapull, weight empty = 31,000 lb. Load, 13yds. @ 2,600 lb. = 33,800 lb. Gross weight = 64,800 lb.

This machine lugs down to third gear at 8.5 mph. on a 1% grade. The rimpull of the machine is 5,295 lb. in third gear at 8.5 mph. What is the rolling resistance?

RoRi = (2000)(5,295/64,800 - 1/100) = 144 lb.

The rolling resistance of the road is very close to 144 pounds per gross ton.

Equation (38) when solved for RP yields

$$RP = \left(\frac{GA}{100} + \frac{RoRi}{2000}\right) GVW. \qquad (40)$$

In this form the equation can be used to find the necessary rimpull to meet a given set of conditions.

Example 15

Estimated tractor weight	=	24,000 lbs.
Estimated trailer weight	=	27,400 lbs.
40 cu. yd. coal @ 1,600 lb.	=	64,000 lbs.
Total weight	=	115,400 lbs.
Grade	=	10%
Rolling resistance	=	60 lb. per ton.

Substituting the above information in equation (40) gives

RP = (10/100 + 60/2000)(115,400) = 15,000 lb.

A Euclid Model 39 FDT-95W will meet this rimpull requirement.

Since drawbar pull is obviously equal to RP less the rolling resistance of the machine, the drawbar pull plus the rolling resistance of the tractor can be substituted for RP in equation (38) so that for machines rated by drawbar pull

$$GA = \left(\frac{DBP + RoRi \text{ of tractor}}{GVW}\right) - \frac{RoRi}{2000} \qquad (41)$$

Solving equation (41) for the drawbar pull gives

$$DBP = \left(\frac{GA}{100} + \frac{RoRi}{2000}\right) GVW - RoRi, \text{ of tractor.} \qquad (42)$$

<u>Example 16</u>

Estimated tractor weight	=	30,000 lbs.
Estimated scraper weight	=	23,400 lbs.
15 cu. yd. load at 2,900 lbs. per cu. yd.	=	43,500 lbs.
Total weight	=	96,900 lbs.
Grade	=	16 %
RoRi	=	100 lb. per ton.

Find what size tractor will be needed to operate under the above conditions.

$$DBP = \left(\frac{16}{100} + \frac{100}{2000}\right)(96,900) - (15)(150)$$

$$DBP = 20,400 - 2,250 = 18,150 \text{ lbs.}$$

A caterpillar D8 can supply 19,537 lbs. of drawbar pull in second gear. A caterpillar D7 can supply 21,351 lbs. of drawbar pull in 1st gear. An Allis Chalmers HD14 operating in second gear, or an Allis Chalmers HD10 in first gear can also furnish the needed drawbar pull.

To find the maximum economical grade in highway construction for ascending vehicles, the equations which follow have been used. (34)

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(34) Agg, T. R., The construction of roads and pavements, 5th ed., p. 113, New York, McGraw-Hill, 1940.

$$P_p = \frac{T}{20} - \frac{R}{20} \quad \text{and} \qquad (43)$$

$$P_p = \frac{T}{20} - \frac{R}{20} + \frac{3.65}{L}\left(S_1^2 - S_2^2\right), \text{ where} \qquad (44)$$

P_p = maximum economical plus grade (vehicle ascending), in feet per 100 ft. of road,

T = the maximum tractive effort of the engine at the desired speed in high gear, in pounds per ton of weight of vehicle and load,

R = tractive resistance of the vehicle on the proposed type of road at the desired speed, in pounds per ton of weight of vehicle and load,

S_1 = speed in miles per hour at beginning of ascent,

S_2 = speed in miles per hour at the top of the hill, and

L = length (of profile stations) in feet.

The foregoing equations are economical from the standpoint of gasoline consumption. The same expressions will not hold true for Diesel fuels since fuel economy increases with the load in Diesel engines. Further, in excavation fuel economy is of minor importance as contrasted to economy of time and capital investment.

Since time is a function of speed rate and speed rate is a function of available power, total load, and total resistance, these variables will be combined into one equation. Substituting the value of RP in equation (40) for RP in equation (29) and collecting terms will give

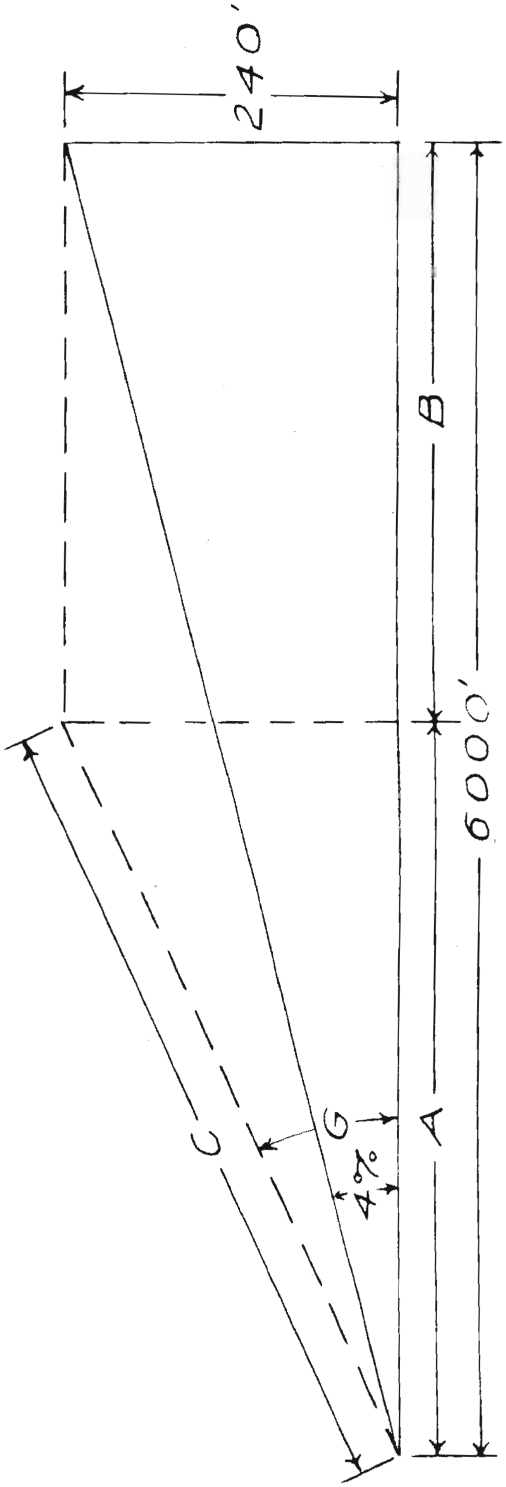
$$MPH = \frac{(750,000)(.9)(HP)(ME)}{(20\ GA + RoRi)(GVW)} . \tag{45}$$

The use of equation (45) is demonstrated in examples 17 and 18. These examples also suggest a method of finding economical grades.

Example 17

Assume that 240 feet of elevation must be gained in 6,000 ft. of horizontal distance. For the trucks being used, the gross vehicle weight is 66,000 lbs., the horsepower is 150, the mechanical efficiency is 0.9, and the rolling resistance will be 40 lbs. per ton of gross vehicle weight. Find the speed of the truck for the different profiles of the road as illustrated in Figure 59.

Using equation (45) to find the speed on grades which are increments of 2% gives the following tabulated values as shown on p. 181.



$$A = 240 \tan G$$
$$C = \sqrt{A^2 + 240^2}$$
$$B = 6000 - A$$

Figure 59. Grade diagram.

% Grade	MPH	Ft. Per Min.
0	34.5	3040
2	17.3	1520
4	11.5	1013
6	8.6	755
8	6.91	607
10	5.76	507
12	4.94	436
14	4.32	380
16	3.84	338
18	3.46	304
20	3.14	276

The profile through the road can be broken down into slope distances for various grades and horizontal distances with the relationships expressed in Figure 59. Using grades which are increments of 2%, the various values become:

% Grade	Distance C	Distance B	Time on C	Time on B	Total Time
4	6005	0	5.93	0.00	5.93
6	4007	2000	5.31	0.65	5.96
8	3009	3000	4.96	0.99	5.95
10	2416	3600	4.76	1.18	5.94
12	2014	4000	4.62	1.32	5.94
16	1519	4500	4.48	1.48	5.96
18	1354	4667	4.45	1.53	5.98
19	1224	4800	4.43	1.58	6.01

The slight differences in total time are due to using the tangent of the angle instead of the sine when finding the value for grade resistance. For all practical purposes, it can be said that there is no difference in total times. There would be economy of fuel with the gentle grade. On the other hand, there might be economy of road construction with the steep grades.

Example 18

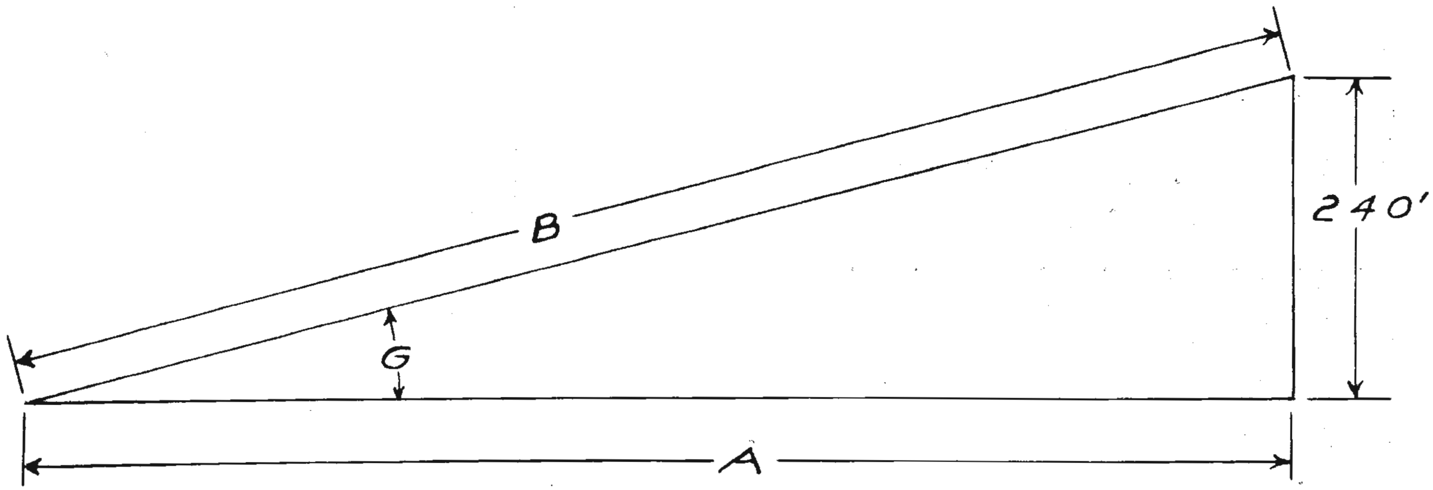
Using the same data given in Example 17, find the shortest time and the shortest distance in which the truck can gain 240 feet of elevation. The relationship between grade and distance is given in Figure 60.

Using equation (45) gives the following results:

% Grade	A Feet	B Feet	Speed Rate On A in Ft./Min.	Time on B in Minutes
20	1200	1224	276	4.43
16	1500	1519	338	4.5
12	2000	2014	436	4.61
10	2400	2416	507	4.75
8	3000	3009	607	4.96
6	4000	4007	755	5.3
4	6000	6005	1,013	5.92

A study of the above tabulated data shows that if it is necessary to gain elevation in the shortest possible distance, the fastest way is through using the steepest grades which the truck

Figure 6. Grade diagram.



$$A = 240 \tan G$$

$$B = \sqrt{240^2 + A^2}$$

can climb. Further large economies are possible because of less road to build and less road to maintain.

The obvious conclusion is to allow for worst possible rolling resistance and friction and allow for deterioration of equipment and then construct the roads with as steep a grade as lies within the ability of the equipment.

ACCELERATION

Thus far in this study no allowances have been made for acceleration time or for the rimpull requirements for acceleration time or for the rimpull requirements for acceleration. The various equations, such as (29), (38), and (45), equating rimpull, speed, grade ability, rolling resistance and other factors, are for conditions of constant speed. To attain any velocity, whatever, its magnitude, means that added rimpull is necessary. In truck-operation this added rimpull is gained by starting in low gears (see Example 13) in which gears large rimpulls can be developed. As higher velocities are reached, gear ratios are changed and rimpull decreases.

The relationship between force and acceleration is

$$F = m\,a = \frac{w}{g}\,a, \qquad (46)$$

where F = force in pounds,

m = mass,

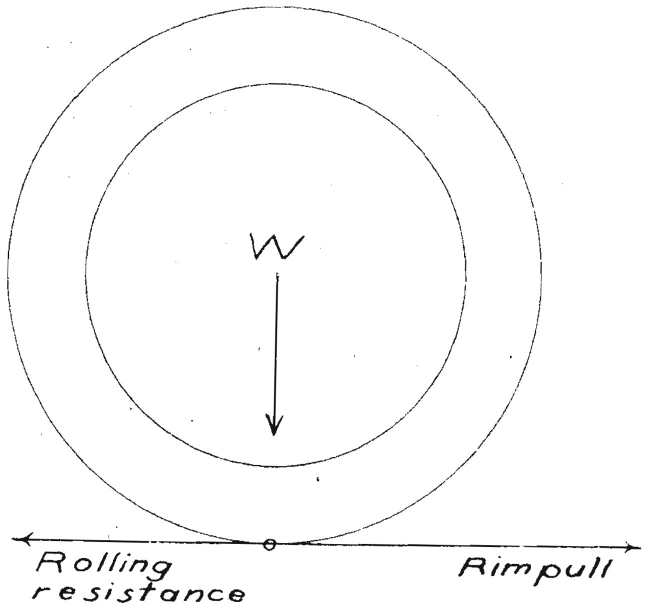
a = acceleration in feet per sec. per sec.,

w = weight in pounds, and

g = acceleration of gravity in ft. per sec. per sec.

In a vehicle, the force available for acceleration will be that rimpull which is in excess of the rolling resistance as shown in Figure 61.

If the vehicle is operating up a grade, the grade resistance will also reduce the rimpull available for acceleration as shown in Figure 62.



Vector Diagram

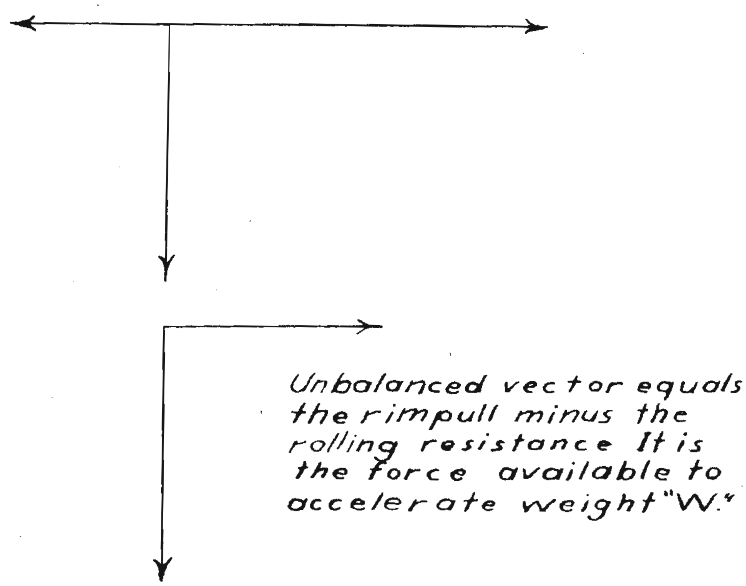


Figure 61. Rolling resistance and rimpull.

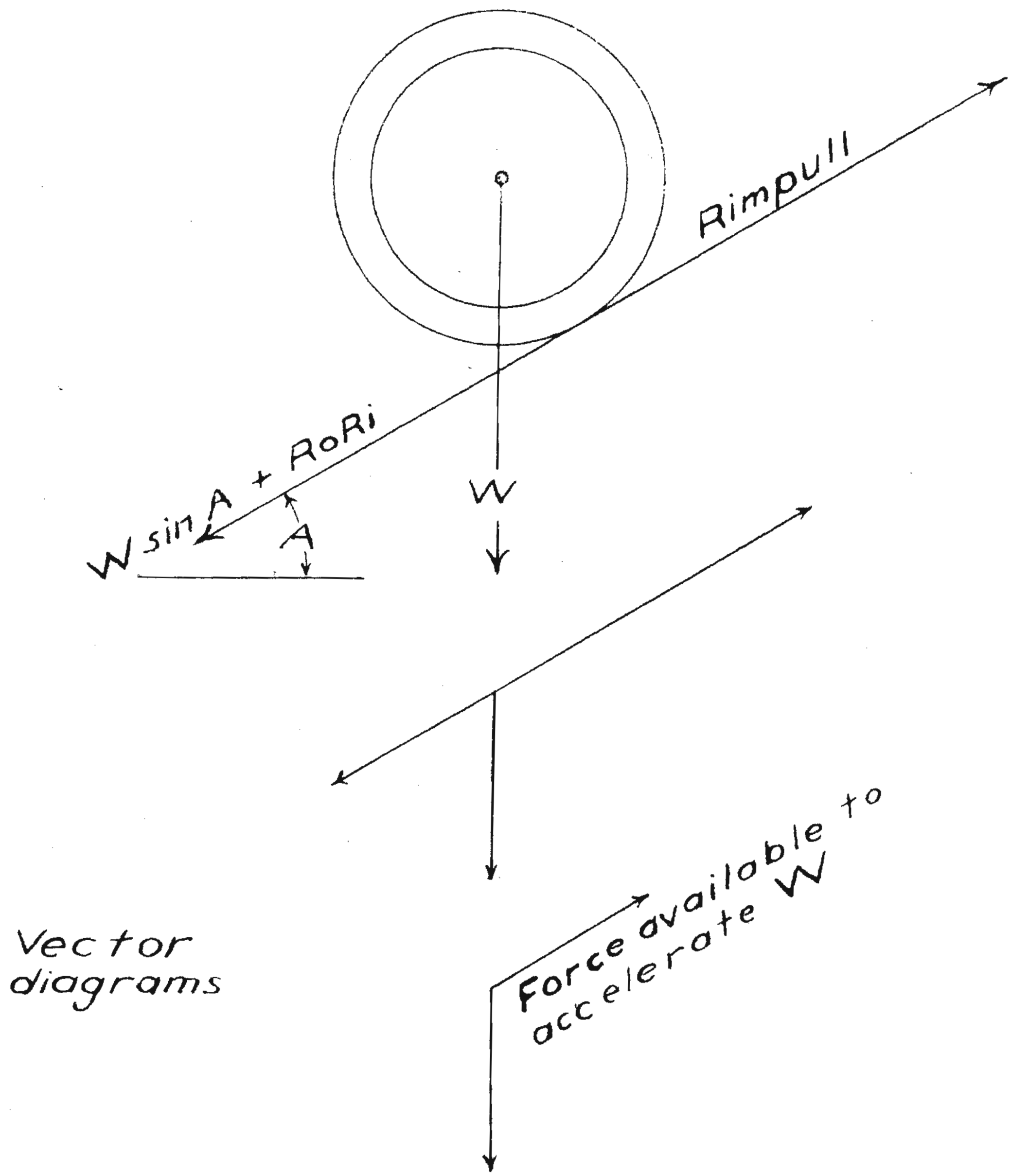


Figure 62. Rolling resistance, grade resistance, and rimpull.

It follows that

$$F = RP - RoRi - GR, \tag{47}$$

where F is the force available for acceleration.

Substituting the value of F in equation (46) for F in Equa- (47) and designating "W" as "GVW" gives

$$\frac{(GVW)a}{g} = RP - RoRi - GR. \tag{48}$$

Solving equation (48) for RP gives

$$RP = \frac{(GVW)a}{g} + RoRi + GR. \tag{49}$$

The value of RP in equation (49) will be the total rimpull requirements to give a rate of acceleration "A".

In terms of the basic engineering formula

$$HP = \frac{Fs}{550\,t}. \tag{50}$$

where F = force in pounds,

s = distance in feet through which force acts, and

t = time in seconds.

Also

$$s = vt, \tag{51}$$

where v = velocity in feet per second.

Substituting the value of s in equation (51) for s in equation (50) gives

$$HP = \frac{Fv}{550}. \tag{52}$$

Since RP in equation (47) is the total pulling force, it may be substituted for F in equation (52) to give

$$HP = \frac{(wa/g) + RoRi + GR}{550} v, \quad (53)$$

where HP will be the horsepower requirement for the condition in the equation.

Solving equation (53) for a gives

$$a = \frac{g}{w} \frac{(550)(HP)}{v} - RoRi - GR \quad (54)$$

Equation (54), when modified by the mechanical efficiency and a constant for the average value of the torque, becomes

$$a = \frac{g}{w} \frac{(.9)(550)(HP)(ME)}{v} - RoRi - GR . \quad (55)$$

Equation (55) is difficult to use, because it is almost impossible to assign a value to the horsepower. From Figure 53, 54 and 55 it can be seen that the horsepower of a motor is a variable of the speed. When gears are shifted, the engine is slowed down. As speed is gained, horsepower is increased. There is no easy method of determining this engine pick-up.

A more practicable approach would be to decide on the desired acceleration rate and velocity, and from this find the rimpull requirements to meet that rate.

The relationship between acceleration, velocity and time is

$$a = \frac{v^2}{2s} . \quad (56)$$

Substituting the value of "a" in equation (56) for "a" in equation (46) gives

$$F = \frac{wv^2}{2gs} , \quad (57)$$

where F = force in pounds,

V = final velocity in feet per second,

g = the acceleration of gravity = 32.18 ft. per sec. per sec. , and

s = acceleration distance in feet.

When velocity in feet per second is changed to velocity in miles per hour, equation (57) becomes

$$F = \frac{wV^2}{29.92\ s}, \tag{58}$$

where V is velocity in miles per hour and the other values are as before. This equation can be used to find the rimpull needed to give a desired velocity in a given distance.

An expression of "F" in terms of pounds per ton would be convenient since rolling resistance and grade resistance have been so expressed. Also, acceleration can be conveniently expressed as miles per hour per second.

$$1 \text{ mi./hr./sec.} = \frac{5280}{60 \times 60} = 1.46 \text{ ft/sec.}^2$$

$$F = \frac{(2000)(\text{lb})(1.46)}{(32.2)(\text{ton})(\text{hour})(\text{sec./mi.})} = 90.7 \text{ pounds per ton}$$

per mile per hour per second.

Figure 63 is the graphical solution of equation (59) giving the rimpull in pounds per ton per miles per hour per second acceleration.

The acceleration rate and distance will affect the average velocity. The average velocity, in turn, will affect the time cycle of a machine. Average velocity is equal to one-half the

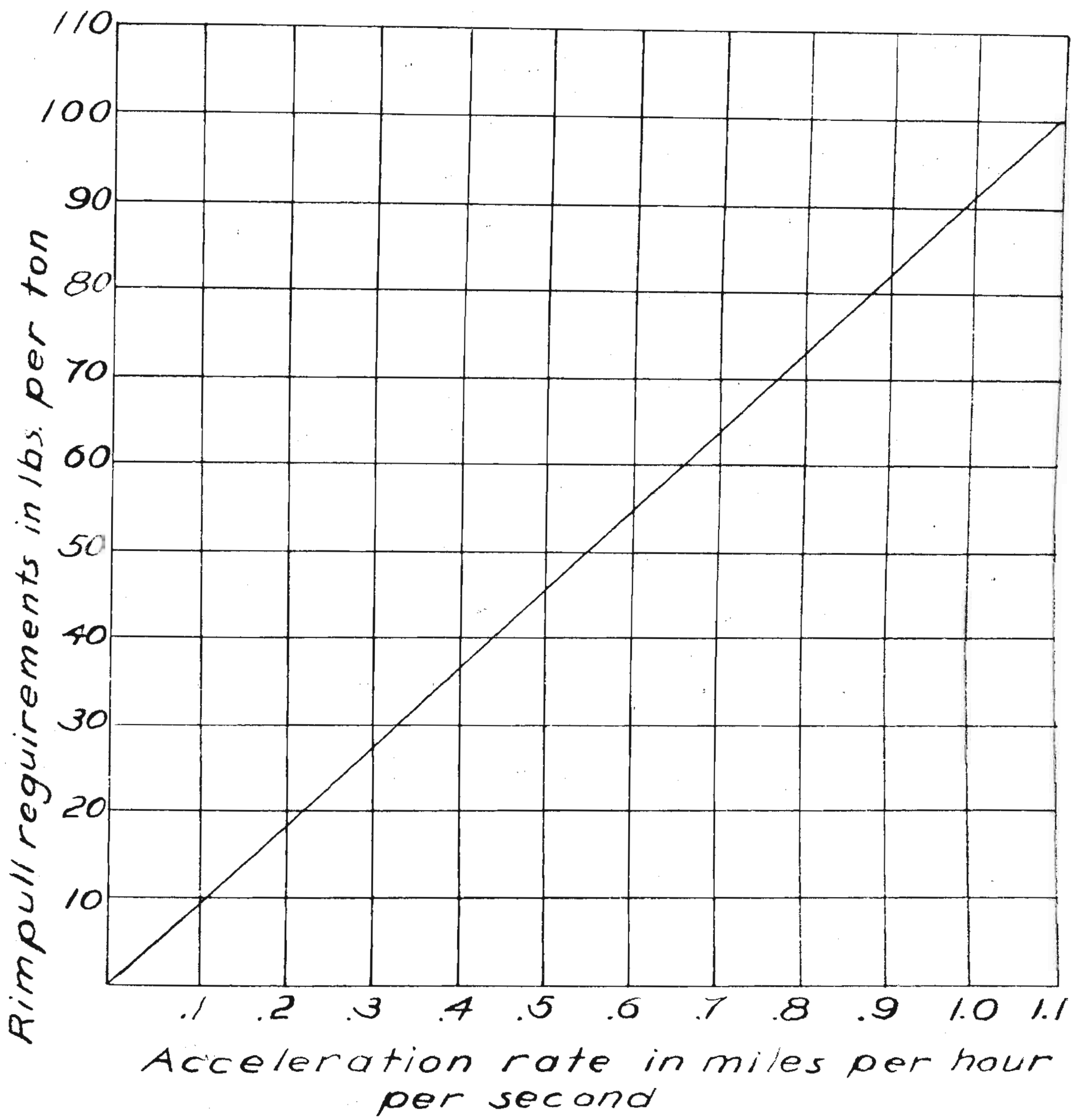


Figure 63. Rimpull requirements for acceleration rates.

sum of the initial and final velocities. Equations (29), (38), and (45) give final velocities.

To find haul time it is necessary to know the distance over which the vehicle accelerates as well as the average velocity. As previously noted, it is difficult to find the exact acceleration distance because of changing horsepower and shifting gears. These distances can be approximated as in Table XV. Table XV is for Euclid trucks.

In using Table XV to determine the average travel speed, multiply the maximum attainable speed as computed using equations (29) and (45), by the factor opposite the correct haul road section length. Return speed can be taken as being from 80% to 85% of the top attainable speed.

Example 19

A given haul unit can attain 8.8 miles per hour on a road the grade of which is 6% and the rolling resistance of which is 40 pounds per ton. What will be the average speed over a 3000 foot haul road if the unit starts from a stop.

The factor from Table XV is .75.

(.75)(8.8 miles per hour) = 6.6 miles per hour, average speed.

TABLE XV [35]

CORRECTION FACTORS FOR ACCELERATION

Length of Haul Road Section	Unit Starting From Stop	Unit Entering Haul Road Section After Accelerating
500 Ft.	.50	.70
1000 Ft.	.60	.80
2000 Ft.	.70	.80
3000 Ft.	.75	.80
4000 Ft. and up	.80 - .85	.80 - .85

(35) Euclid Road Machinery Company, Estimating production and costs of material movement with Euclids, Form No. 350 - R p. 6, Cleveland, Ohio, 1946.

TRACK-TYPE TRACTORS AND BULLDOZERS

The crawler tractor, with its various attachments, is the most versatile of the different pieces of excavating equipment in use today. It is an important auxiliary in almost every excavating operation, and, at times, it is the main piece of equipment in the operation. Appendix C gives the characteristics of some of the more common tractors.

Without attachments to the machine, the crawler-tractor can be efficiently used as a towing machine for work such as skidding timber, pulling stone boats, and moving machinery and other heavy objects. It is also used for stump pulling in land-clearing, and frequently for pulling rocks.

When towing a cat-wagon or a multi-wheeled trailer, the tractor becomes a haul unit. Because of the low speed rates, the economic limit to haul distance is short. The great field of use of the crawler wagon is in those places where bearing capacity of soils is so low as to preclude the use of wheel-mounted equipment. This is the case in sand and gravel excavation along river banks, levee construction, channel excavation, road construction through swamps, mining wet clay deposits and similar projects.

Frequently, too, the tractor and crawler wagon are used where grades restrict the use of trucks. Because of greater power, lower speeds, and greater coefficient of tractive efficiency, the crawler tractor has greater grade ability than does the truck. In some types of excavation it is more economical to use the slower speed tractor and steep grades than it is to use high-speed trucks and low grades. Low grades entail a greater expenditure for road construction and road maintenance. Low grades can be costly in mining in that to construct low grades on roads it may be necessary to tie up ore in the benches.

The crawler tractor and wagon have the advantage over other equipment in that grades, bearing capacity of soils, and weather are factors which limit their performance less. Bad weather stops trucks but not tractors. The chief disadvantage of the tractor-wagon combination is that of low speed.

To increase the capacity of the tractor-wagon combination and to compensate somewhat for the lack of speed, the tractor frequently tows two crawler wagons in tandem.

As a towing machine the crawler tractor is frequently used to pull a sheepsfoot roller. The sheepsfoot roller is used to compact and solidify loose fills. The weight of the tractor and the design of the crawlers is such as to aid in compaction.

The rooter is a common attachment to the crawler tractor. Rooters are commonly either 3-tooth or 5-tooth, but can be used with 1, 2, 3, 4, or 5 teeth depending on what the tractor can pull in a given set of conditions. Closer furrows are possible with the 5-tooth.

The rooter is used to prepare material for excavation by the bulldozer, scraper or elevating grader. The rooter is used to break up root systems as its name suggests. It is also used to tear out stumps and buried rocks. The rooter is used to break up frozen earth, sand, and gravel. It is also used to break up rocky material and hard-packed clay. Figure 64 illustrates the breaking action of the rooter.

If more weight is needed to help the rooter penetrate hard material, rocks or sand bags can be placed on top of the rooter.

Compared to the cost of most earth-moving equipment, the rooter is inexpensive. The use of the rooter can greatly increase the capacity of elevating graders and scrapers. The use of the rooter can make scraper excavating and loading by elevating graders possible where otherwise a power-shovel might be needed.

Great economies are possible with the use of a bulldozer-rooter combination such as that illustrated in Figure 64. In using this combination, the tractor roots in one direction, turns and then digs and pushes in the opposite direction as illustrated in Figure 65. If being used as a pusher in scraper loading, the same combination and cycle can be economically applied.

The most common attachment for the crawler tractor is the bulldozer blade. The blades, the size of which varies as the machine size, are either operated by cables or by hydraulic lifts. The cable control gives a higher blade lift and a lower blade drop than does the hydraulic control. The cable controlled blade is also faster and more powerful. The cable controlled blade depends on the dead weight for penetration. With the hydraulic control some of the weight of the tractor can be put on the blade for greater

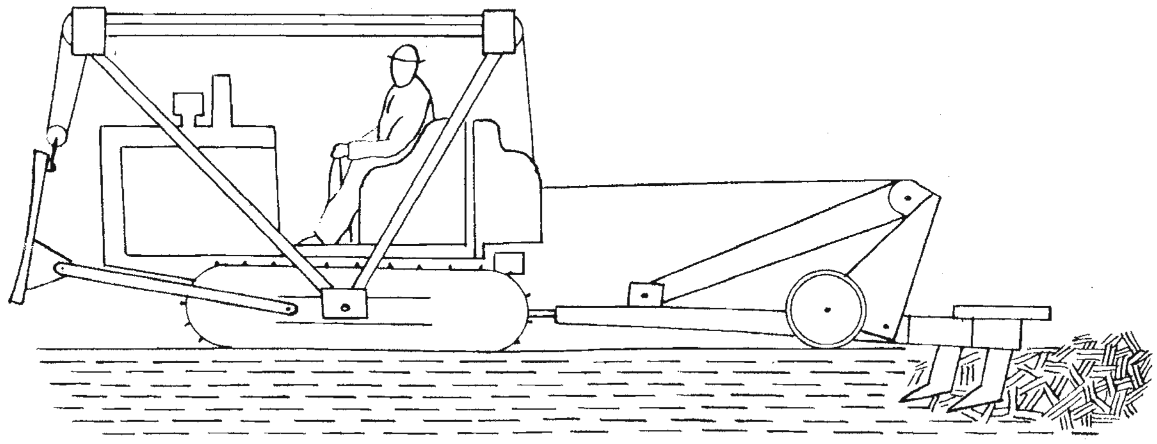
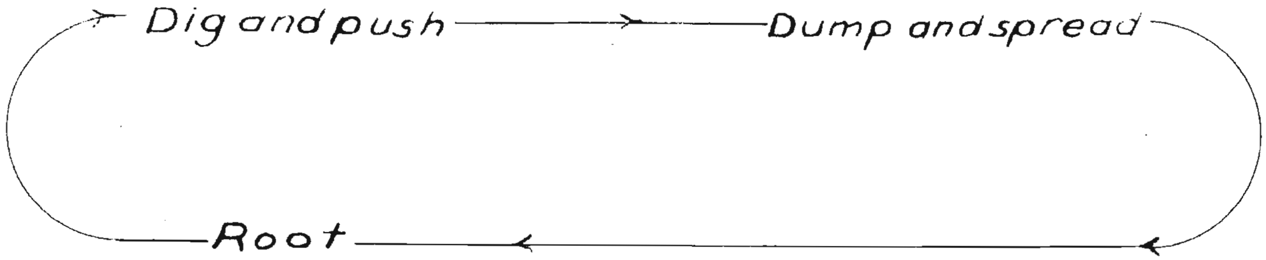


Figure 64. Bulldozer and rooter.

Figure 65. Bulldozer and rooter cycle.



down-pressure. This last mentioned feature of the hydraulically operated blade is not always an advantage. To add the weight of the machine to the blade removes it from the tracks and decreases tractive efficiency.

Variations on the bulldozer blade are the tiltdozer, the bullgrader, the treedozer, and the angledozer.

The tiltdozer is a blade that can move through a small vertical angle about an axis which is parallel to the long axis of the machine. Thus a tiltdozer can drop one end of the blade to more readily make V-cuts. Some bulldozer blades have adjustments for tilt. V-cuts can be made without tilting the blade.

The bullgrader differs from the bulldozer only in size. The bullgrader blade is longer than the bulldozer blade. The bullgrader blade is used for spreading fill and for running preliminary grades.

The treedozer is a cable-operated bulldozer that has a higher possible blade lift. The high blade lift makes it possible to get more leverage on a tree when using the machine to clear land.

The angledozer differs from the bulldozer in that the blade is set at an angle to the machine. This makes the machine more efficient in grading, in making V-cuts, and in making side-hill cuts. Many bulldozer blades are constructed so that they may be angled if the work demands it.

As an auxiliary tool, the bulldozer has many uses on earthmoving projects. It is used to spread fill in road construction and to make the preliminary grades. In like manner it is used to keep dumps level in stripping operations in mining.

The use of a bulldozer in a pit can greatly increase the capacity of the power-shovel or dragline by relieving these machines of slow clean-up work. The bulldozer can also efficiently clean up the stone scattered by blasting and by spill in loading. The bulldozer can also keep roads clear of material that spills from the haul units.

As an auxiliary the bulldozer is also used as a "pusher" to aid in scraper loading. Some crawlers are equipped with a pusher front for this one purpose. The bulldozer is preferred, however, since when not being used as a pusher it can be used as a bulldozer. When using a bulldozer as a pusher, care must be taken so as not to injure the rear tires of the scraper.

The bulldozer is frequently used to increase the capacity of surge piles. In Figure 2 it can be seen that the capacity of the surge pile is limited by the angle of repose of the material. In practice a bulldozer is frequently used to spread the material over a greater area and thus increase the capacity.

The bulldozer is unexcelled by any other common machine for land-clearing. Land-clearing is the necessary first step in most earth-moving projects whether these projects are in mining or in construction. Bulldozers can fall trees up to three feet in diameter. In falling large trees the bulldozer first moves some dirt and cuts the roots on the fall side of the tree. Then the roots on either side of the tree are cut with the blade or with a rooter.

Then the bulldozer pushed the tree over by placing the blade as high as possible up the trunk for good leverage. Frequently the final push is made by placing the blade under the root ball and lifting while pushing.

The rooter is a useful auxiliary to the bulldozer in land-clearing. By cutting the root systems with the rooter, bulldozer capacity and efficiency are increased.

The dozer is an ideal tool for making the opening-cuts in road-building. For initial side-hill cuts, the angledozer is more efficient than the bulldozer since the angledozer sidecasts the material as it moves forward. With the dozer it would be necessary to swing to dump.

As an excavating machine the bulldozer has some efficient applications. Generally, the bulldozer should be limited to hauls less than 200 feet long. Under some conditions the economic limit can be increased to about 300 feet. For efficient operation, the dozer should also be restricted to broken and soft material. As previously mentioned, the applicability to many soils is increased by the use of the rooter.

As an earth-mover the capacity of the bulldozer can be increased by dozing downhill whenever possible, by using shuttle-like movements, by dozing in trenches or troughs, by using highest possible speeds, and by using two machines traveling abreast of, and parallel to, each other.

As indicated in the discussion on grade-resistance, to work downgrade makes it possible to take advantage of the pull of gravity on a machine to increase drawbar pull. To increase drawbar pull will increase possible speed and the possible load. Likewise, in pushing downhill it is possible to take advantage of the angle of repose of the material and thus increase the bowl load. This possibility is suggested by Figure 69.

When turns are eliminated, as in Figure 66, capacity is increased by decreasing the cycle time. In the cycle pictured in Figure 66, the bulldozer digs and pushes, dumps at the end of the push, shifts to reverse, backs up in highest attainable speed, and then shifts to a forward gear to repeat the cycle. There might be an additional shift to a higher forward gear after the bulldozer has filled the bowl. If it is necessary to swing and dump as in Figure 67, about 15 seconds will be added to the cycle time.

If the dozer is operated over the same path so as to form a trench or trough, capacity of the bowl will be increased because spill around the ends of the blade will be prevented by the sides of the trench. This same increase may be accomplished to a minor degree by operating two bulldozers abreast of each other. In this latter case spill is prevented around the adjacent ends of the two bulldozer blades.

Every effort should be made to use the highest possible speeds at all times. If the dozer has a low reverse speed it should turn at the end of the cut and return in the highest possible forward speed. If the bulldozer has a high reverse speed, it should return

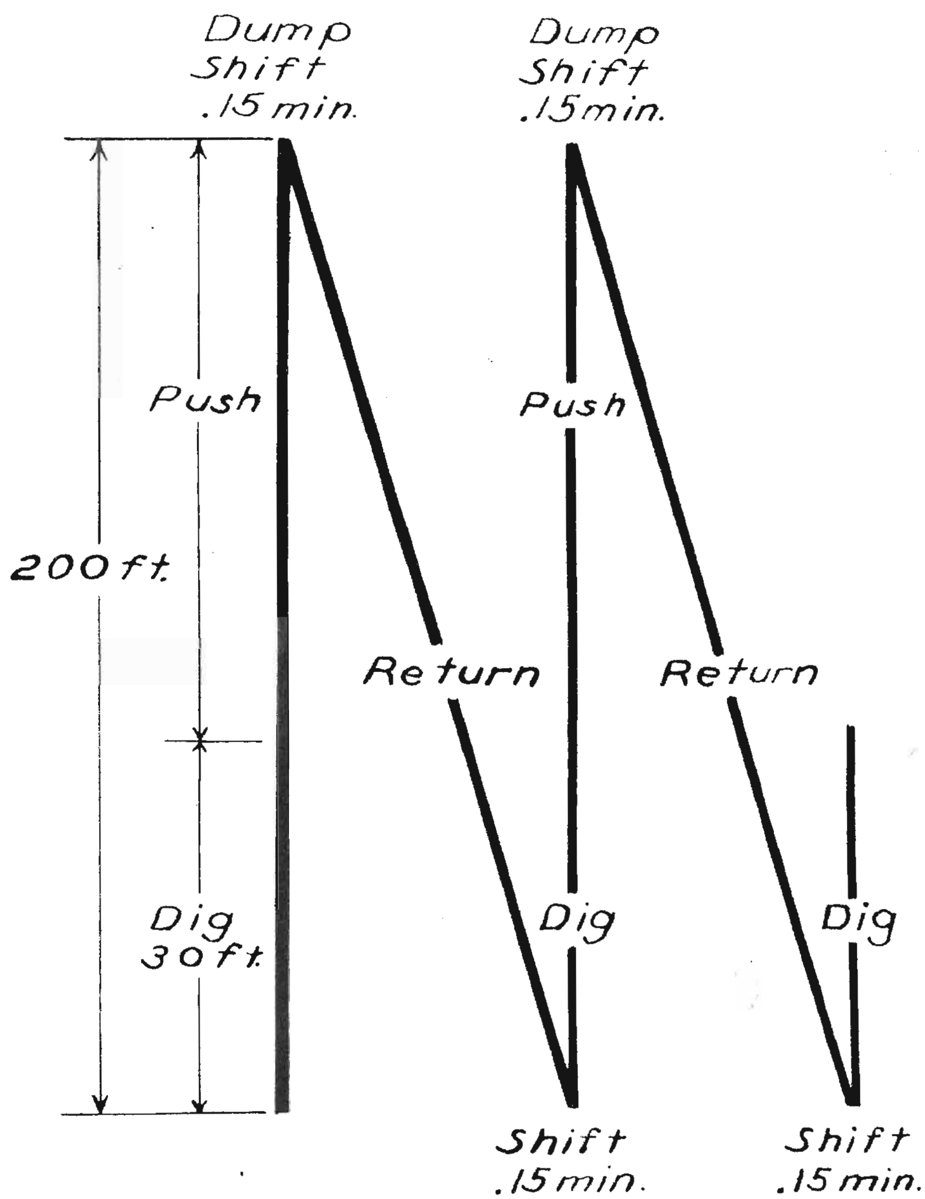


Figure 66. Bulldozer cycle using shuttle movement.

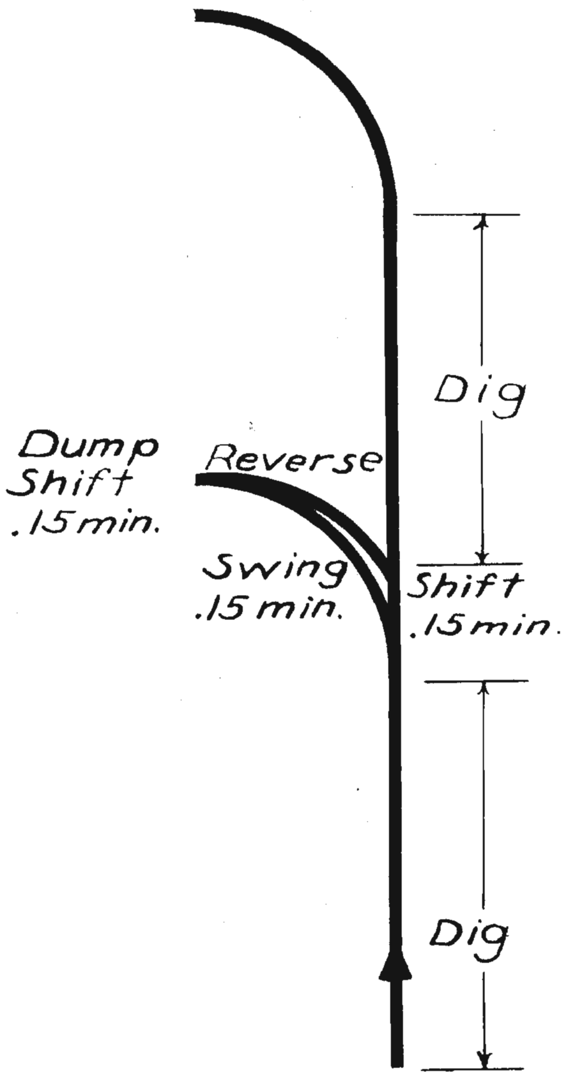


Figure 67. Bulldozer cycle when turning and dumping.

in that speed and eliminate turning around. Thus when using a Caterpillar D8 on a long enough haul the unit should return by turning at the end of the cut and using the high forward speed which is 4.9 miles per hour. The highest possible reverse speed of this unit is 2.6 miles per hour. On the other hand, if a Caterpillar D7 is being used, it should return in the high reverse speed which is 5.4 miles per hour. The highest forward speed of the D7 is 6.0 miles per hour. The difference between these two speeds is not great enough to justify the added time entailed in making two extra turns.

The ideal conditions of use for the bulldozer in a stripping operation are illustrated in Figure 68. In this case the bulldozer is dozing soft material downhill and dumping without turning. Though not illustrated here, trench-dozing is possible under these conditions. When beds outcrop at the side of a hill in this manner, the bulldozer is the ideal tool to use for making the initial cut.

The bulldozer is sometimes used to strip very shallow mineral deposits of limited areal extent even though it is not possible to waste the overburden downhill. In these cases the bulldozer pushes the overburden to the sides of the deposit to keep pushing distance to a practicable minimum. An effort is made to keep the grades as flat as is possible and economical.

The bulldozer has some efficient applications as a loading machine. Figures 69 and 70 illustrate how this can be accomplished.

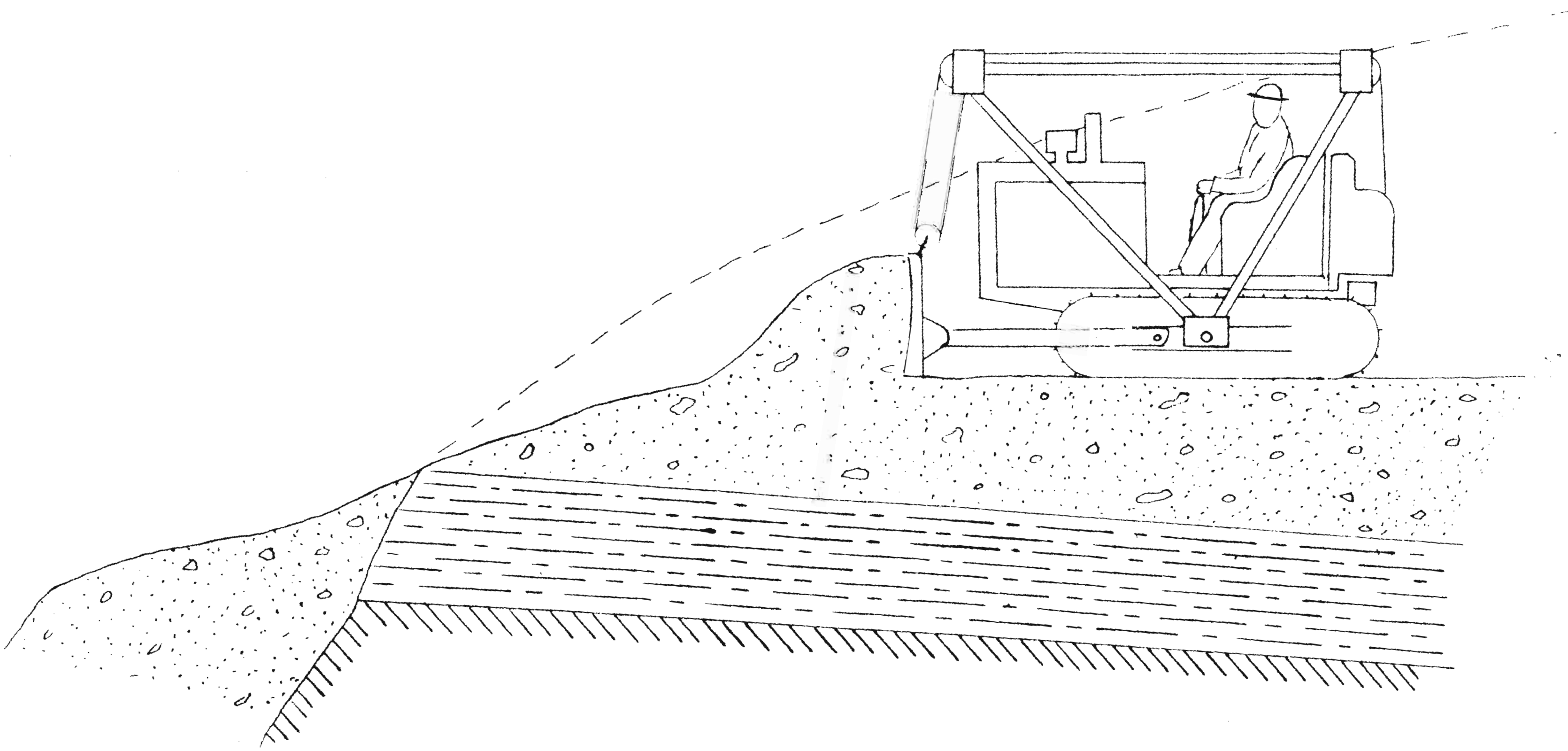


Figure 68. Ideal conditions for stripping with a bulldozer.

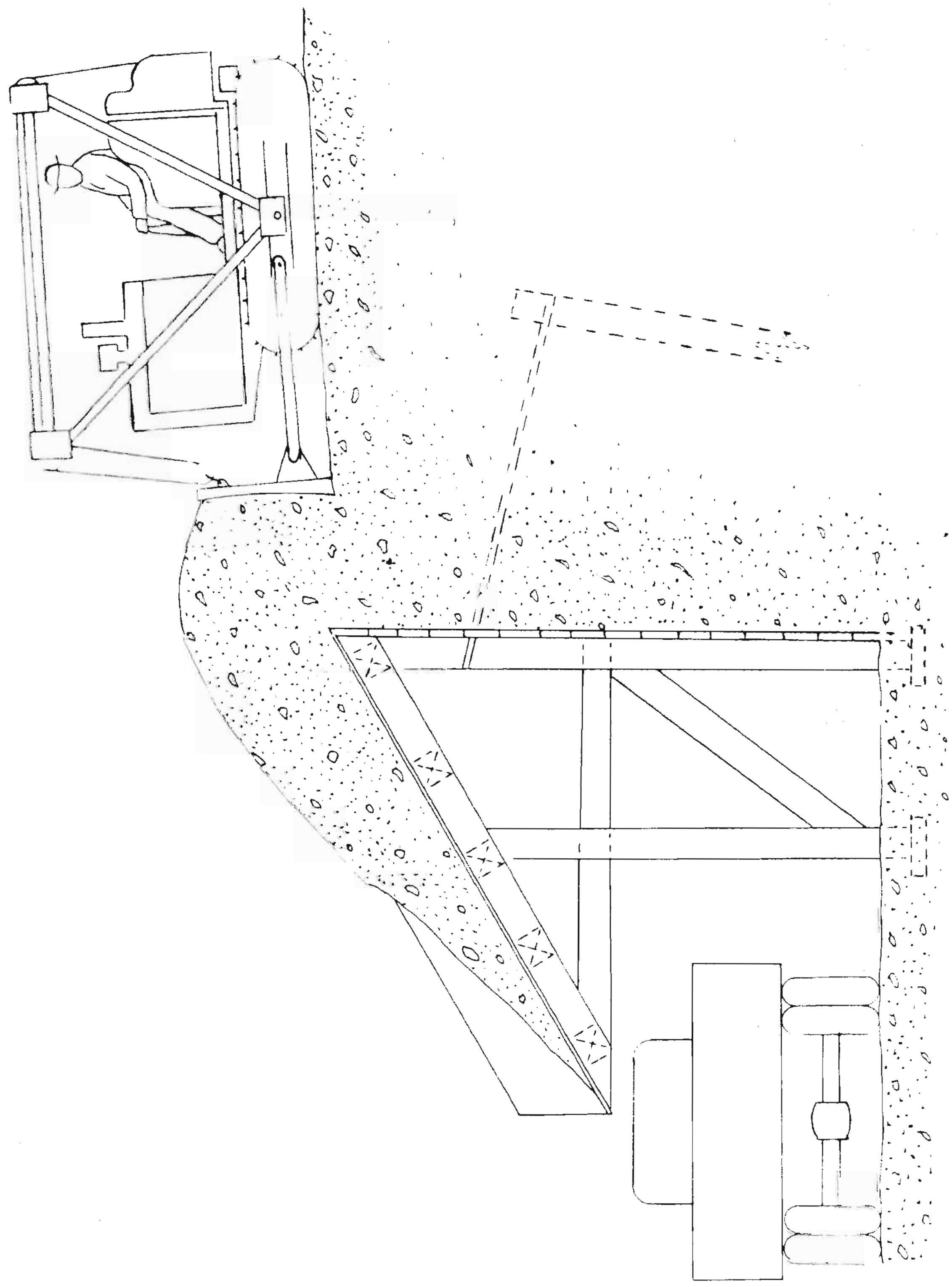


Figure 69. Bulldozer loading trucks from a ramp.

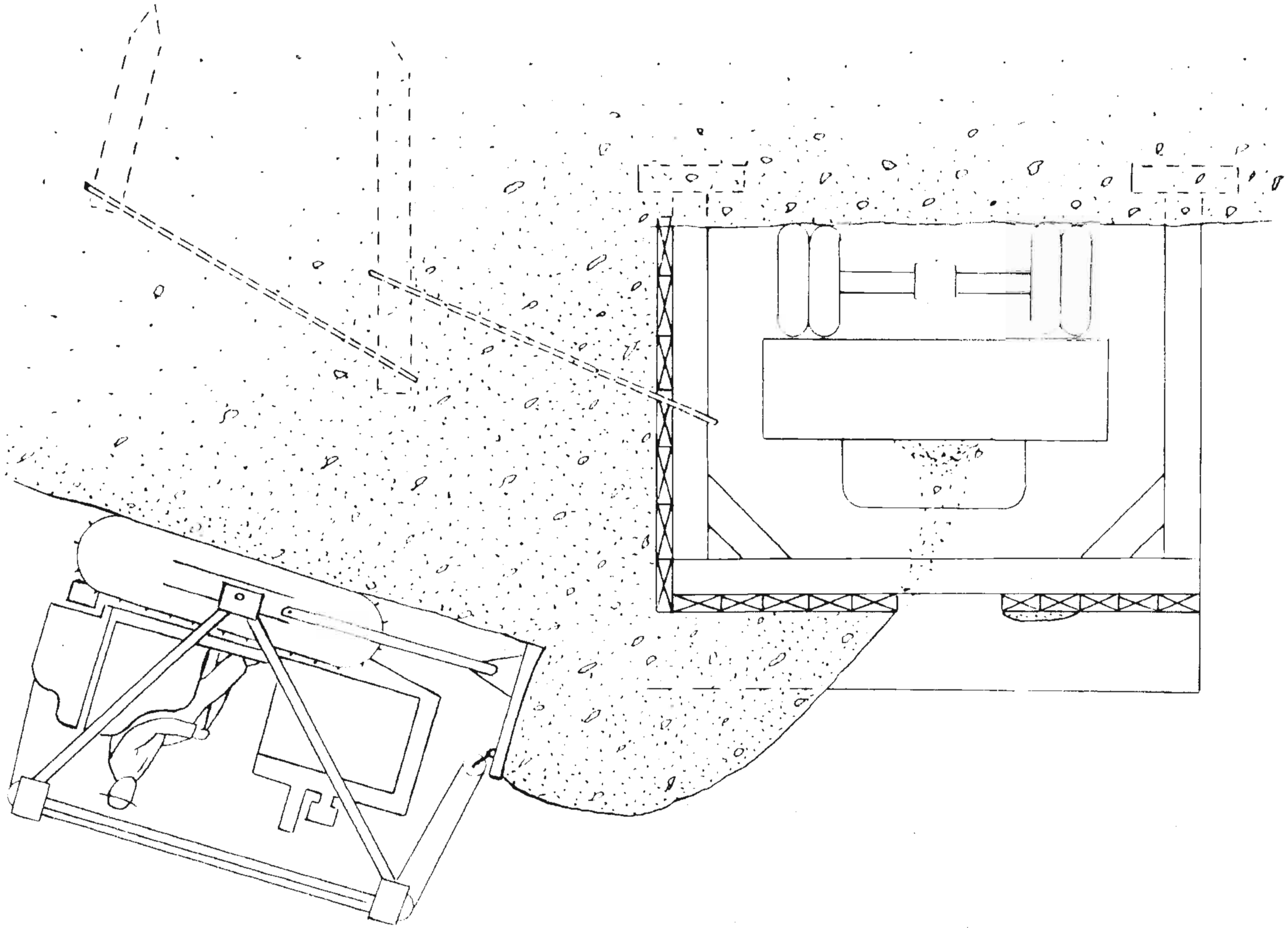


Figure 70. Bulldozer loading trucks from a trapped bridge.

These arrangements have proved to be economical at many sand and gravel pits.

Usually a rough timber ramp and crib, or trapped bridge and crib, is built at a low point. The bulldozer then builds an earth ramp. The sand and gravel is then pushed by the bulldozer over the ramp or bridge to load the truck below. If possible the ramp or bridge is constructed so that it will not be necessary for the dozer to push a load upgrade.

When there are no trucks present the bulldozer is occupied in digging material from the further reaches of the deposit and transporting it to a position near the ramp where it will be available for quick loading of the trucks. At other times the machine can be used to clear the land and strip the deposit. In many gravel pits only one machine and one man are needed.

In at least one instance this method was used to mine a vein deposit of gold-bearing ore. This vertical vein outcropped in a hill. One bullgrader was used to push the ore down from the apex of the outcrop and the other bullgrader working below pushed it over a ramp to load trucks. This operation was run by the North Moccasin Mines Syndicate near Lewistown, Montana.

On the same general principle the bulldozer has been used to feed crushers from a stock pile and to feed conveyor belts from surge piles. Figure 71 illustrates the conditions that exist at stock or surge piles. After the pile has funneled to the angle of repose from the feeder outward, a bulldozer is necessary to clean up the rest of the material.

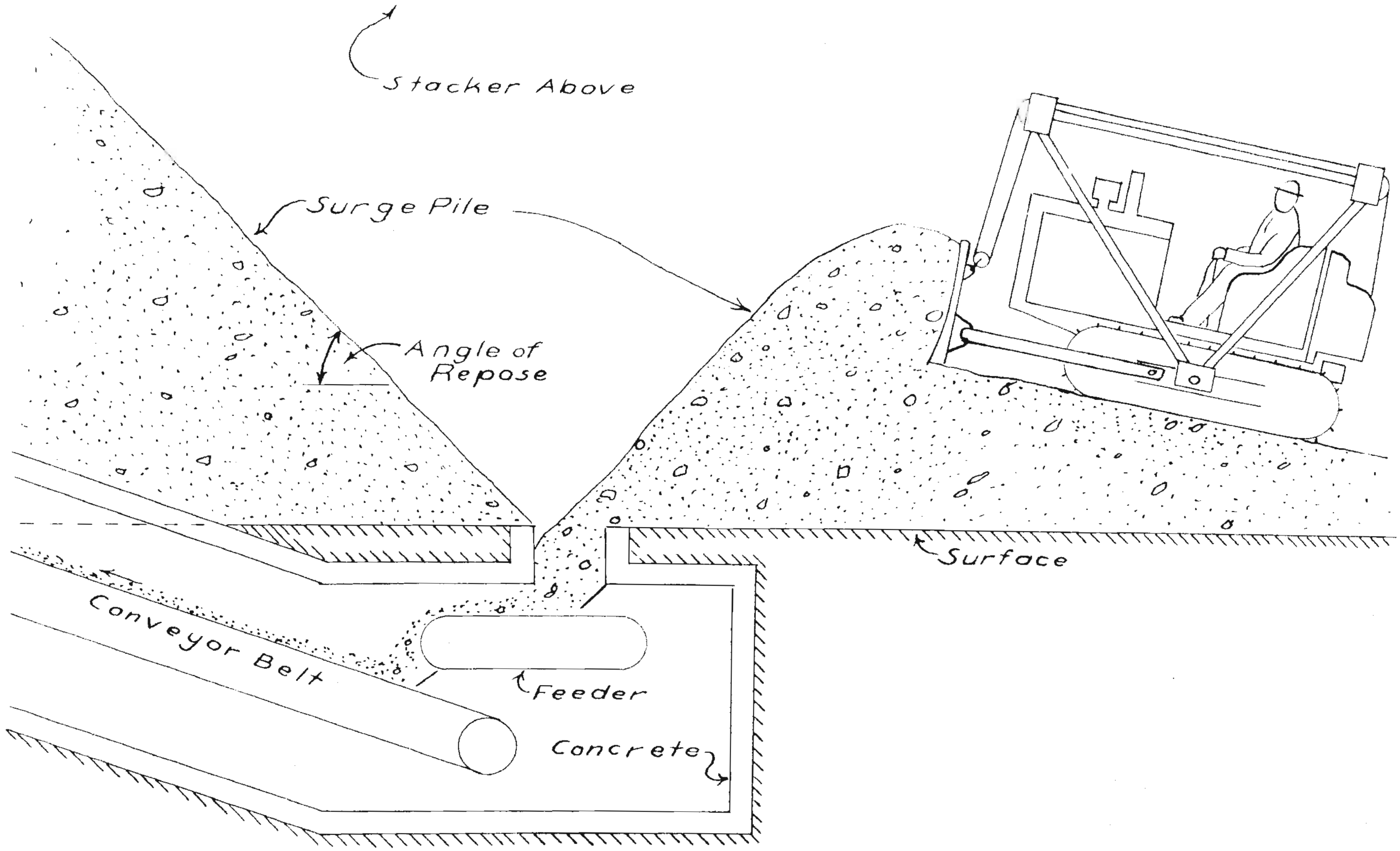


Figure 71. Bulldozer feeding conveyor belt from surge pile.

A method of approximating the capacity of the bulldozer follows.

The blade load of the bulldozer may be found by the equation

$$BL = \frac{(L)(H)(2/3\ H)}{27}\ (SF), \tag{60}$$

where BL = blade load in bank cubic yards,

L = length of blade in feet,

H = blade height in feet,

2/3 H = 2/3 of blade height in feet which is equal to average thickness of load, and

SF = swell factor of the material being excavated.

The capacity of the bulldozer may then be found by the equation

$$C = (60/T)(BL)(E), \tag{61}$$

where C = bulldozer capacity in cubic yards, place measurement, per hour,

T = total time in minutes per round trip, and

E = working efficiency.

Total time is found by adding .15 minutes for each shift of the gears to the digging and traveling time. Digging distance will average about 30 feet. If a 90 degree turn is necessary to dump, as in making side-hill road-cuts, an additional .15 minutes is added. If gears must be shifted to make a 180 degree turn at each end of the cut, 0.30 minutes must be added to the cycle time. If wide radius curves are made at each end of the cut, this extra distance must be included to find the travel time, but no shift-time is added at the curves.

Working efficiencies may be found from the following table;

TABLE XVI (36)

TRACTOR EFFICIENCIES

Conditions	Excellent	Average	Poor
For tractor equipment	90%	83%	75%
Rubber tired equipment	85%	75%	65%

(36) Park, K. E., Principles of modern excavation and equipment, p. 61, Peoria, Illinois, R. G. LeTourneau, Inc., 1942.

If material is being pushed down grade the capacity is increased by 40% for each 5% of grade. If material is being pushed up grade, the capacity is reduced by 20% for each 5% of adverse grade.

Example 20

What is the hourly capacity of a Caterpillar D7 digging and pushing common earth 300 feet? The machine digs in first gear, pushes in third gear, dumps without turning, and returns in high reverse gear. The bulldozer blade is 10 feet long by three feet high. It is operating under average working conditions.

$$\text{Blade load} = \frac{(3)(2/3)(3)(10)}{(27)} (.80) = 1.8 \text{ cu. yd.}$$

Speed in first gear = 1.4 mi./hr. = 123.3 ft./min.

Speed in third gear = 3.2 mi./hr. = 281.6 ft./min.

Speed in reverse = 5.4 mi./hr. = 475.2 ft./min.

Cycle:

Shift time	= 0.15 min.
Dig 30 ft. at 123.3 ft./min	= 0.24 min.
Shift to third	= 0.15 min.
Push (300 ft. - 30 ft) at 281.6 ft./min	= 0.96 min.
Shift to reverse	= 0.15 min.
Return 300 ft. at 475.2 ft/min.	= 0.63 min.
Total time	= 2.28 min.

Capacity = (60/2.28)(1.8)(.83) = 29.3 cu. yd. per hour, place measurement.

Example 21

A Caterpillar D8 is being used to push dirt 300 ft. It digs in first gear and pushes in fourth. It can either turn at each end and return in the sixth gear forward, or it can return in the high reverse speed without turning. Which system gives the shortest cycle?

First gear forward = 1.6 mi./hr. = 140.8 ft./min.

Fourth gear forward = 3.0 mi./hr. = 264 ft./min.

Sixth gear forward = 4.9 mi./hr. = 431.2 ft./min.

High reverse speed = 2.6 mi./hr. = 228.8 ft./min.

Cycle with turns at each end:

30 feet at 140.8 ft./min.	= 0.213 min.
Shift to fourth	= 0.15 min.
(300 ft. - 30 ft.) at 264 ft./min.	= 0.023 min.

Turn and shift	= 0.30 min.
300 ft. at 431.2 ft./min.	= 0.695 min.
Turn and shift	= 0.30 min.
Total time	= 2.68 min.

Cycle when returning in reverse:

30 ft. at 140.8 ft./min.	= 0.213 min.
Shift to fourth	= 0.15 min.
(300 ft. - 30 ft.) at 264 ft./min.	= 1.023 min.
Shift to reverse	= 0.15 min.
300 ft. at 228.8 ft./min.	= 1.315 min.
Shift to first	= 0.15 min.
Total time	= 3.00 min.

In this case time would be saved by turning at the end of the cut and returning in a high forward speed.

Buckets and scoops have been attached to the crawler-tractor to form machines commonly known as front-end loaders, dozer-shovels, high-lifts, and overhead-loaders. These machines vary in capacity from one-half to three cubic yards.

These machines can efficiently be used for making small excavations such as basements and foundations. They are also useful for loading material from stock piles. A common use is in charging cement mixers with aggregate. Machines of this type have been used in sand and gravel excavation. Generally these machines are restricted to unconsolidated material and to small-scale projects.

The overhead loader has the greatest capacity of the machines of this type. An interesting application of the overhead loader was in connection with the system pictured in Figure 9. After the first cut was made with the power shovel, a large overhead loader was used to make another cut before the belt was moved. This overhead loader was mounted on a Caterpillar D8. It used a shuttle movement as in Figure 66. The machine loaded as it moved forward into the bank. The scoop was raised overhead as it returned in reverse to the hopper.

The latest development in the tractor field has been the rubber-tired bulldozer. Foremost among these machines is the Tournadozer which is made by the LeTourneau Company. These machines have several advantages over the crawler-type tractor. First, they have greater possible speed. Second, shifting time is only .03 minutes, whereas the crawler tractor uses .15 minutes. Third, rubber-tired tractors can travel over streets and highways since they do not have lugs that injure the surface.

In the writer's opinion, the principle disadvantage of these machines is that they do not have the tractive ability in soft soils that the crawler tractor has. They also lack the grade ability of the crawler tractor. An ideal application for a machine of this type is cleaning the coal bed ahead of the coal-loading shovel in strip mining coal.

THE SCRAPER

The scraper is a comparatively recent development in earth-moving equipment. The scraper is finding increasingly wide and varied applications. It is a versatile, flexible, self-contained unit. When properly applied, the scraper makes possible startling economies over the shovel-truck system.

The tractor-drawn scraper has two limitations. The first limitation is in the type of material that can be effectively and efficiently handled. This material must be unconsolidated and small-sized. The second limitation is that of speed. Average speed under excellent working conditions will not greatly exceed 400 feet per minute.

The first limitation can be extended somewhat by loosening the material with a rooter and by adding drawbar pull by using a "pusher" or a "snatch-cat." Even then the machine is restricted. The size of stone the machine can handle is limited to the size of stone the tractor and scraper can straddle. Even with small blasted stone the efficiency is decreased by the difficulty in picking up a full load.

The slow speed of the crawler tractor limits the economical length of haul for the scraper to 1000 feet. This limitation can be extended somewhat by increasing scraper size or by using two scrapers in tandem. To do this will necessitate the use of a pusher or snatch-cat to aid in loading. By these methods the haul limit might be extended to about 2000 feet.

Foremost among the favorable features of the scraper is that the one machine operated by one man can load, haul, dump and spread. The machine operates independently of all other units. If one scraper breaks down it does not affect other units as is the case if a power shovel breaks down. When a power shovel breaks down, it idles all the trucks serving it.

The digging action of the scraper makes it possible to carry uniform grades. The scraper loads by taking a thin slice of material as it passes over the ground. This slice can be horizontal or inclined. A horizontal slice is desirable in making road cuts, and land-levelling for irrigation or for air-port construction. In mining thinly stratified deposits, a horizontal cut is desired. A scraper can efficiently make thin cuts on which it would be uneconomical to use a power shovel.

The inclined slice makes it possible for the scraper to mix material as it excavates it much the same as the power shovel does. This mixing action is desirable when excavating material for road-fill.

The scraper is a haul unit. Against the disadvantage of the short economical haul distance is the advantage possessed by the tractor-drawn scraper in being able to operate on road conditions that exclude the use of trucks. Crawler-tractors can develop greater tractive effort under almost any conditions than a truck can. By being able to operate on poorer roads than trucks can, the tractor-drawn scraper makes possible a saving on road building and road maintenance. Crawlers improve most earth road beds by travel; trucks destroy road beds. With greater tractive effort is greater grade ability. Tractor-drawn scrapers can gain elevation more rapidly than trucks can. Still another advantage given to the scraper by its greater tractive efficiency is that the scraper is less affected by weather than is the truck.

The scraper needs no help in dumping and spreading. The machine dumps its load while traveling without the loss of time. The dumping is by a positive action that effectively cleans the bowl. The spreading of the load while dumping means that no auxiliary equipment, such as the bulldozer or the grader, is needed. Further, by spreading in a thin blanket and compacting the previous load, better road compaction is possible than is possible with trucks. This is a highly desirable feature in road-building.

If desired, the scraper can dump from a standing position into a bin or pocket. This is necessary in mining. Figure 72 illustrates how this can be accomplished. The bars of the grizzly are spaced close enough together so that they (tractor and scraper) can safely and easily travel over them. The scraper is stopped over the grizzly and dumps from a standing position. The scraper load passes through the grizzly to the bin or pocket below. This system has been used in excavating sand and gravel, clay, iron ore, manganese and other ores.

Because the scraper is self-contained, there are fewer delays in scraper excavation than there are in power-shovel and truck excavation. There are no waiting periods at the shovel

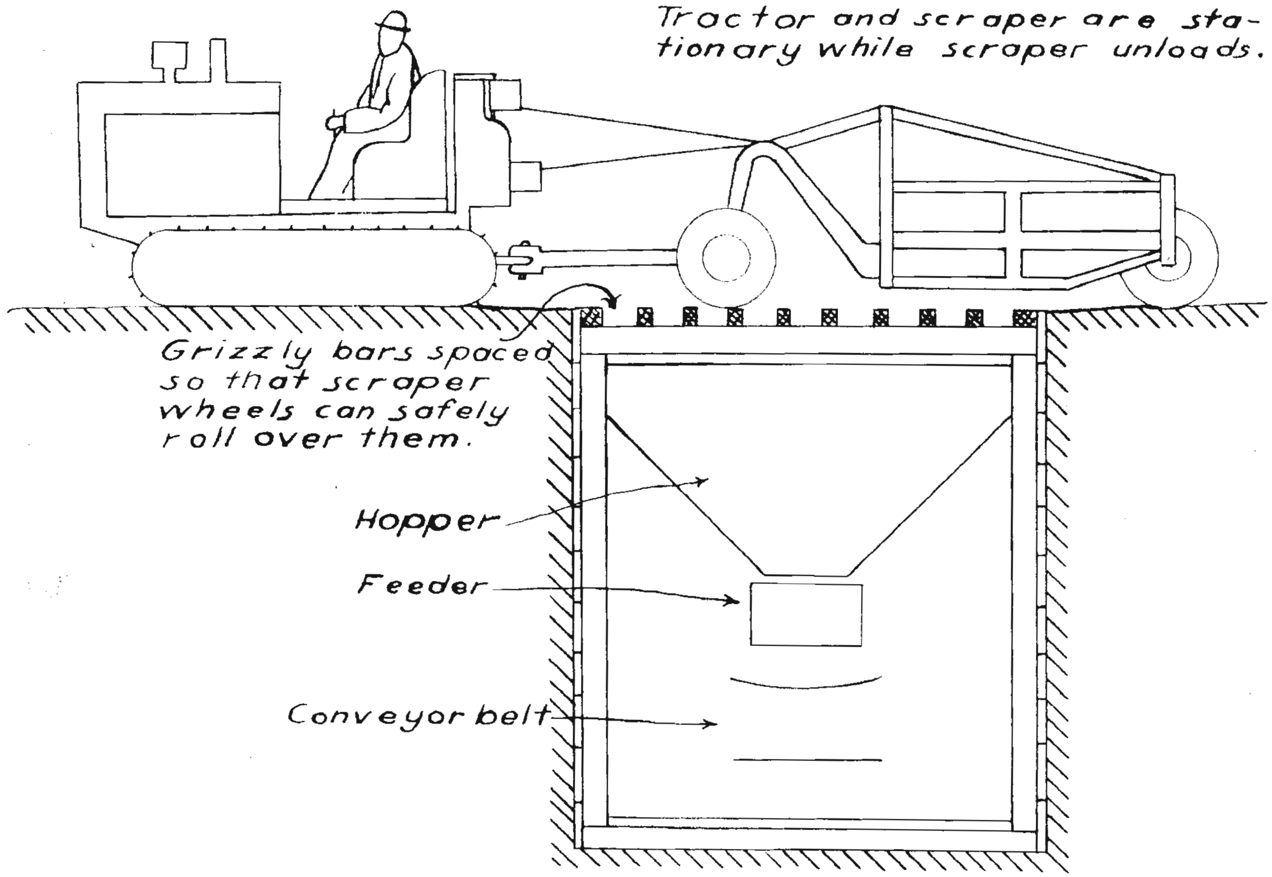


Figure 72. Scraper unloading through grizzly.

or at the dump when scrapers are being used.

Because it is a self-contained unit, the tractor-drawn scraper possesses a flexibility that is lacking in other earth-moving equipment. The scraper can go any place at any time to make almost any type of excavation. This is an especially useful characteristic in mining deposits of ores that occur as scattered pockets.

If a shovel were to be used to mine these pockets, much preliminary work is entailed. First it is necessary to build roads for trucks. Then the slow task of moving the shovel must be undertaken. The pocket must then be stripped, before the ore can be loaded out. Grades and roads must be maintained for the trucks.

When the scraper is used, a bulldozer can rough out the roads and clear the trees. Then the scraper can strip the deposit, mine, and haul the ore.

In mining the deposits of greater lateral extent, stripping and mining can be carried on together. The scraper can load ore and haul it in one direction and strip and haul overburden in the other direction to make a smooth balanced cycle.

There are many possibilities for balancing the scraper cycle. Every effort should be made to eliminate unnecessary turns and unproductive travel. When individual balances are used as in the upper part of Figure 73, there are two turns for each load. The distance traveled without production is equal to the distance traveled while working. If the "Figure Eight" is used, there are two

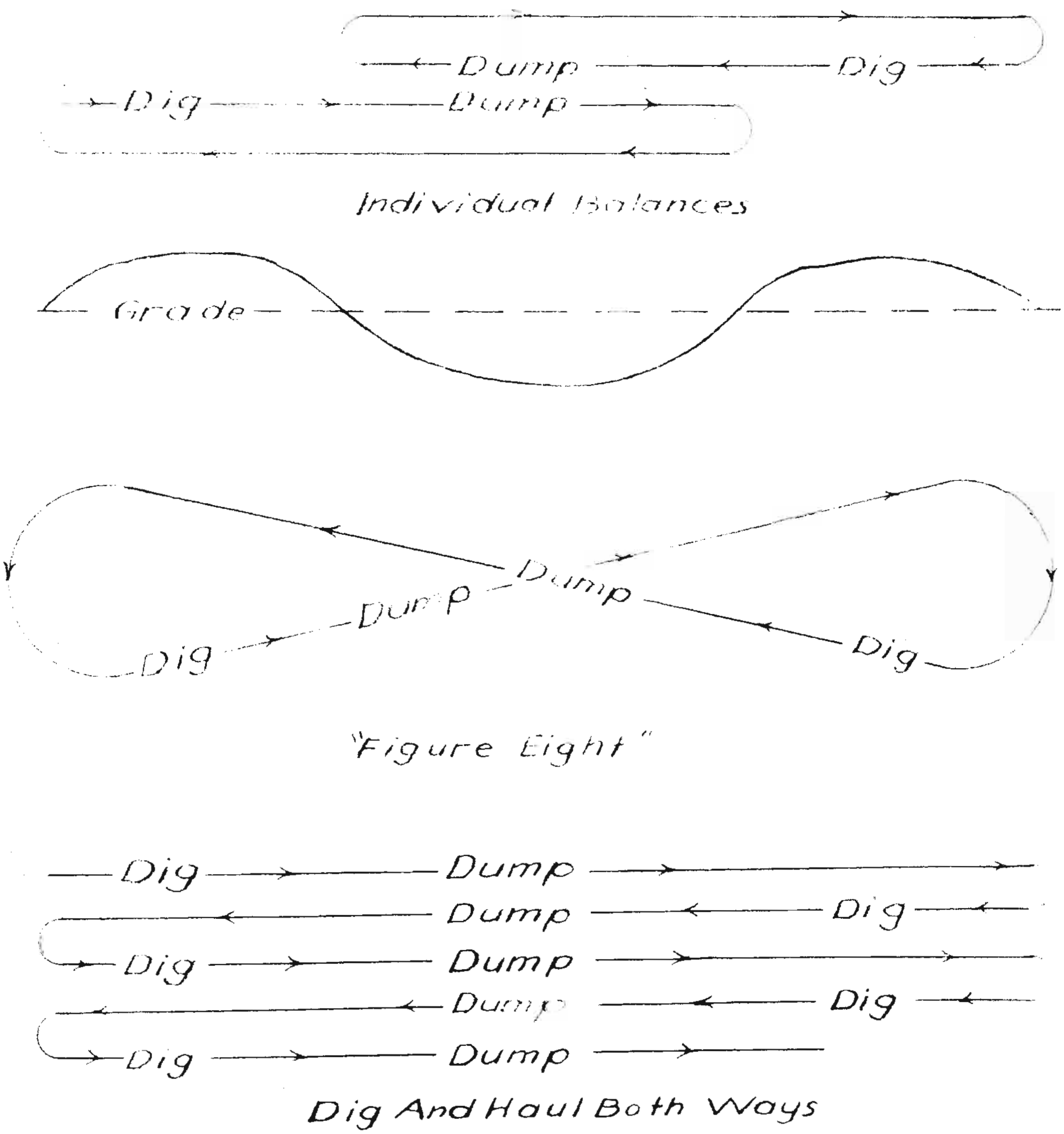


Figure 73.[37] Tractor-drawn scraper cycles.

(37) After Park, E. F., Principles of modern excavation and equipment, p. 12, Peoria, Ill., R. G. LeTourneau, Inc., 1942.

turns for two loads and the distance traveled empty is reduced appreciably. The same reductions are made by digging and hauling both ways as shown as the bottom of Figure 73.

Figure 74 shows how it might be possible to make further reductions in turns and wasted travel distance by lengthening the distance being worked. Using individual balances there are two turns per load and unproductive distance equals productive distance. By lengthening the cycle, as in the bottom part of the figure, the number of turns is reduced to one for four loads and unproductive travel distance is almost completely eliminated. Figures 76 and 77 also illustrate these gains in efficiency.

Figure 75 illustrates another method of increasing the scraper efficiency by improving the cycle. The illustration shows how a scraper might be efficiently used in digging drainage ditches or canals. By leaving earth ramps in place at close intervals, the round trip distance for the scraper can be reduced. Reducing the distance will keep the scraper operating within economical distance. This economical distance is less than 1000 feet for a tractor-drawn scraper. After the scraper has dug the canal to grade, the earth ramps can be excavated.

In scraper loading the resistance of the cutting edge of the scraper when loading is the greatest consumer of drawbar pull. While loading the rolling resistance is a negligible factor. After the scraper is loaded, the load becomes a trailing load and rolling resistance is again the controlling factor along with

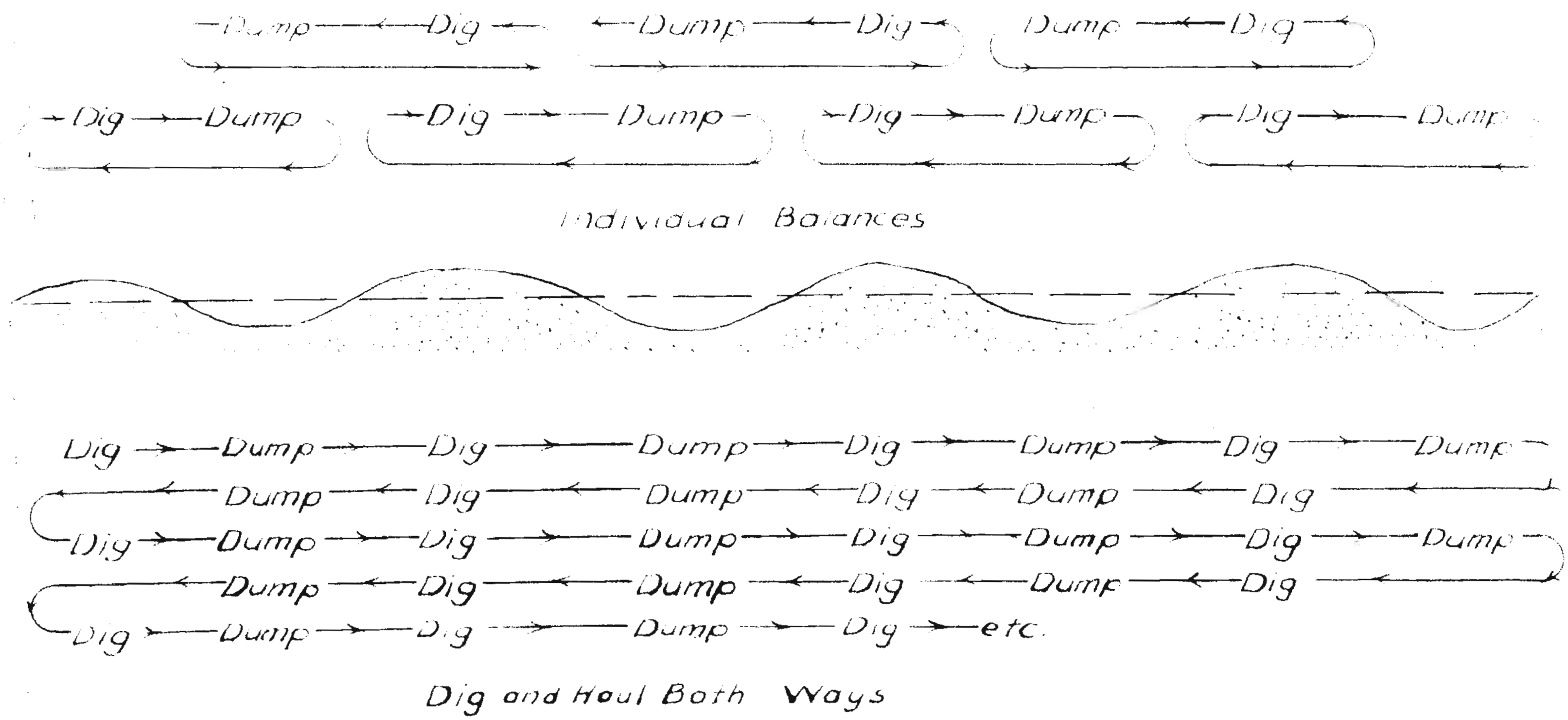


Figure 74.[38] Scraper cycles.

(38) After Build Sao Paulo Highway: The Co-operator, Vol. 5, No. 10, p. 13, January, 1944.

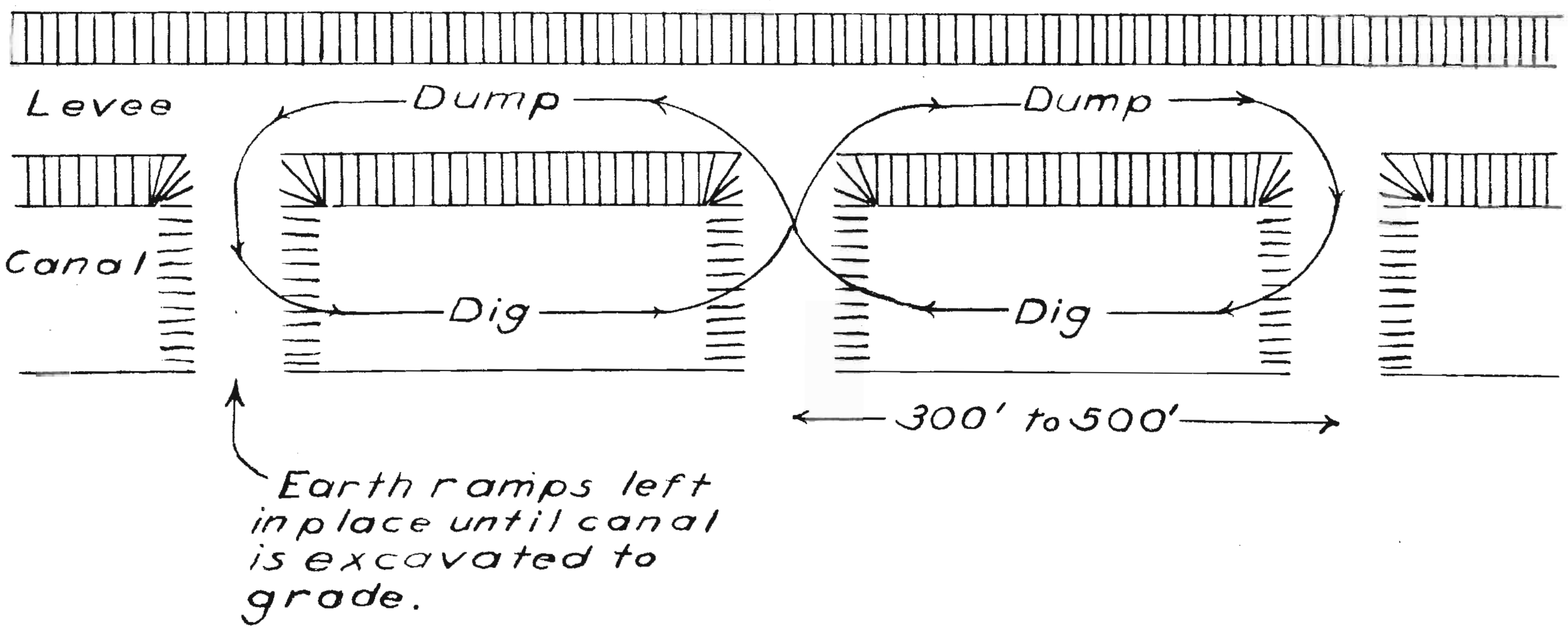


Figure 75.[39] Scraper cycle in canal excavation.

(39) After Bixby, F. S., Draining cotton fields the easy way: Excavating engineer, Vol. XXXVI, No. 4, pp. 196-198, April, 1942.

grade resistance.

The tractor-drawn scraper's capacity for self-loading and its drawbar pull are very nearly equal, i.e., it requires one pound of drawbar pull to self-load one pound of dirt. Thus a Caterpillar D8 tractor has a drawbar pull of 26,200 lbs. and so can self-load very nearly 26,200 lbs. of dirt into the scraper while operating on a level surface. A D8 tractor is capable, however, of pulling a much greater trailing load than this and to do so will increase the economical length of haul. To get a load larger than 26,200 lbs., the scraper must have assistance in loading. This assistance is commonly obtained by using pushers. Example 22 will illustrate the above relationships.

Example 22

A Caterpillar D8 is to be used for pulling a scraper. The haul will be up a 10% grade. To maintain the desired production it will be necessary to haul in fourth gear. The maximum rolling resistance will be 100 lbs. per ton. The material being moved weighs 3000 lbs. per loose cubic yard. What size scraper can be used? What size tractor will be necessary to aid in loading?

Solving equation (42) for GVW gives

$$GVW = \frac{\text{DBP} + \text{RoRi of tractor}}{\left(\frac{GA}{100} + \frac{RoRi}{2000}\right)} \qquad (62)$$

The drawbar pull of a Caterpillar D8 in fourth gear is 13,707 lbs. The D8 weighs 18 tons. Substituting these values in equation (62) gives

$$GVW = \frac{(13,707) + (18)(100)}{\frac{10}{100} + \frac{100}{2000}} = 103,380 \text{ lbs.} = 51.7 \text{ tons}$$

The maximum possible trailing load is then

51.7 tons − 18 tons = 33.7 tons.

This trailing load will be made up of the scraper and its load. A LaPlant Choate C84 weighs 22,000 lbs. or 11 tons.

33.7 tons − 11 tons = 22.7 tons, the allowable load.

$$\frac{22.7 \text{ tons}}{1.5 \text{ tons per cu. yd.}} = 15 \text{ cu. yd., permissible load.}$$

The capacity of the LaPlant Choate C84 varies from 13.6 cu. yd. struck to 16 cu. yd. heaped. It will be used.

The drawbar pull of the Caterpillar D8 in first gear is 26,208 lbs. The unit can then self-load very nearly 26,208 lbs.

45,400 lbs. − 26,208 lbs. = 19,292 lbs. additional scraper capacity which will require additional drawbar pull for loading.

A Caterpillar D7 has 21,351 lbs. of drawbar pull in first gear and so it will be used as a pusher to assist loading.

In practise the pusher will serve several scrapers. An efficient pusher cycle with one or with several scrapers is illustrated

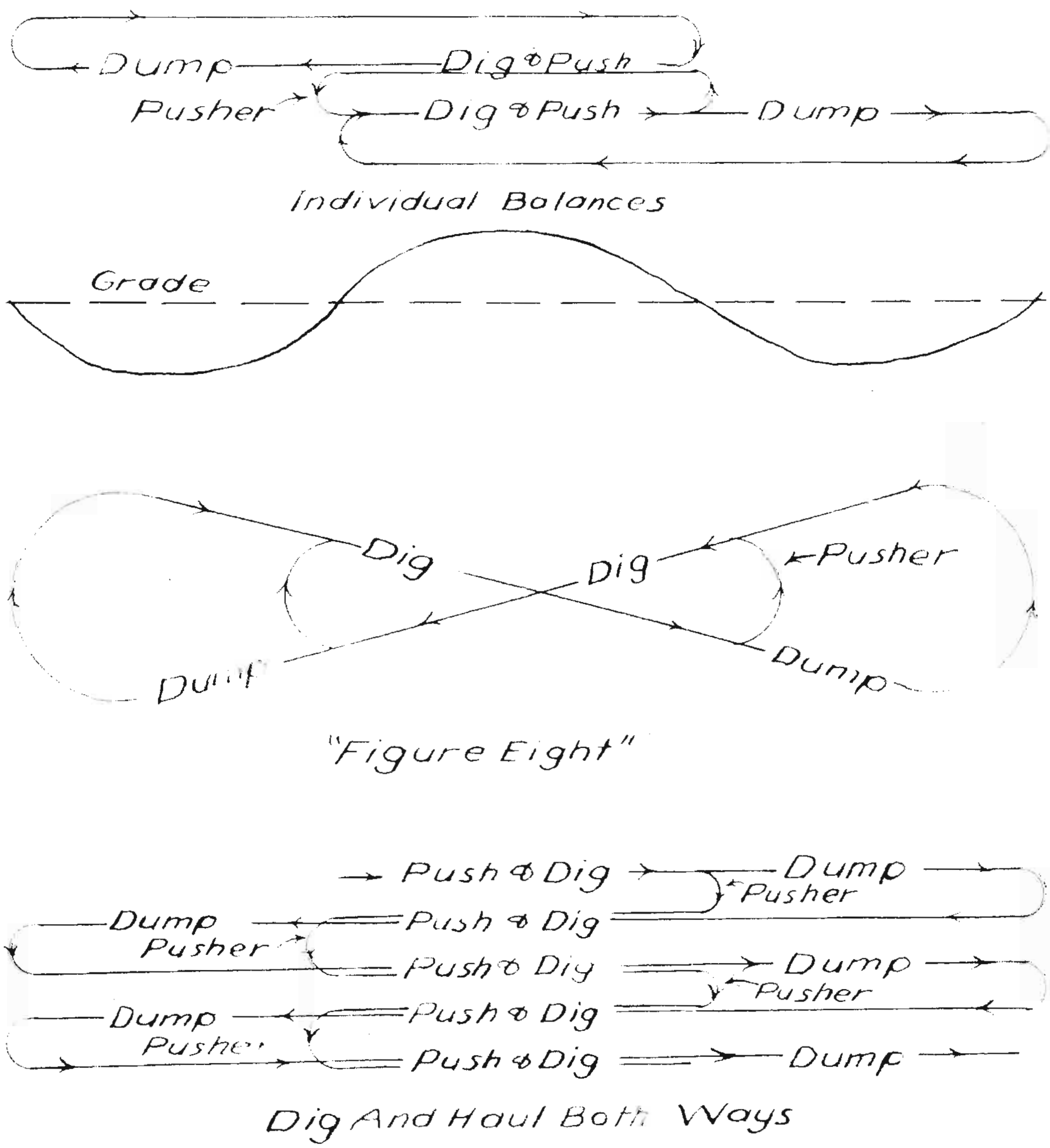


Figure 76. Scraper and pusher cycles.

in Figure 76. By eliminating unnecessary turns and travel distance, it is often possible to use only one pusher where two might otherwise be needed. Good coordination between pushers and scrapers require good management.

Figure 77 illustrates the efficient use of a pusher. Scraper No. 1 self-loads to the limit of its ability. As scraper No. 1 reaches its limit, the pusher comes up behind it to help complete the load. As the pusher is helping scraper No. 1, scraper No. 2 is self-loading parallel with the pusher. After scraper No. 1 is loaded, the pusher swings over to aid in loading scraper No. 2. This system is carried on through the operation. When carefully coordinated no time is lost in loading. If the hill shown in Figure 77 was a borrow-pit or a stripping area, the same system could be used.

If the pusher is not kept busy in loading, the idle time could be used in rooting. The cycle shown in Figure 65 could be used to an advantage.

Scrapers are now being built in sizes up through 40 cu. yds. These large scrapers require the assistance of two additional machines if they are to be loaded to capacity. In this case one machine pushes and the other machine pulls. The pulling machine is referred to as a "snatch-cat." Cycles similar to that shown in Figure 77 can be used.

Pushers can often be eliminated if the work is laid out to take advantage of grades. If movement is downgrade the 20 lbs. of grade resistance per ton per percent grade becomes 20 lbs. of

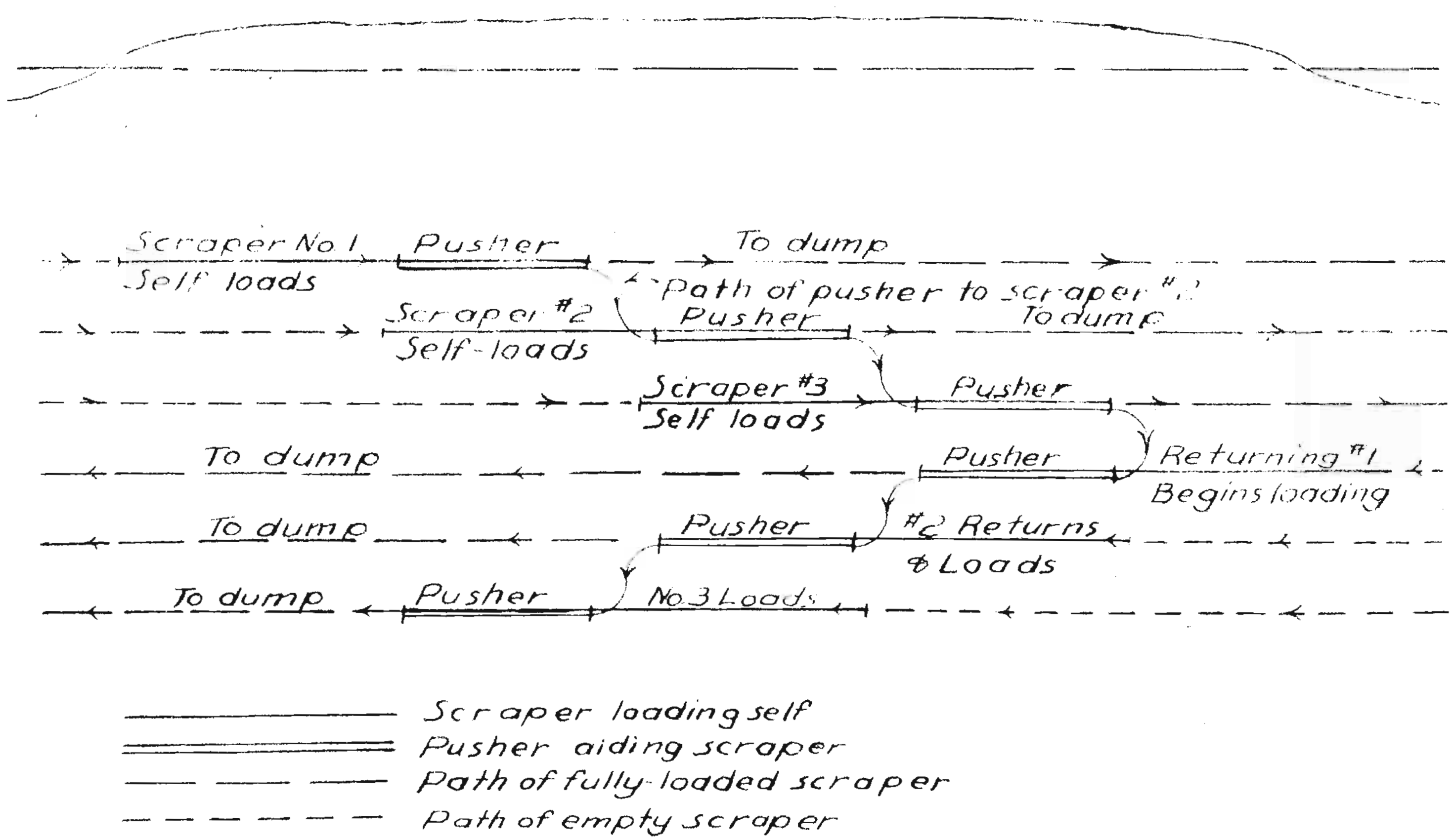


Figure 77. Scraper and Pusher cycles.

grade pull per ton per percent grade.

Example 23 will illustrate how a favorable grade aids scraper loading.

Example 23.

How large a load can a Caterpillar D7 tractor pulling a Bucyrus Erie S152 scraper self-load working down a 5% grade? The dirt weighs 2,700 lbs. per loose cubic yard.

D6 weight is		25,300 lbs.
Bucyrus Erie S152 scraper empty weighs	-	23,400 lbs.
Total weight of machinery	=	48,700 lbs.
Self-load on level surface (since D7 has 21,351 lbs. pull in low gear)	=	21,000 lbs.
Weight of machinery and load	=	69,700 lbs.

69,700 lbs. = 34.85 tons.

Grade pull due to working down the 5% grades =

(5%)(20 lbs/percent grade ton) = 100 lbs. per ton.

(34.85 tons)(100 lbs./ton) = 3485 lbs. additional pull set up by the 5% grade due to the weight of the machinery and the material which could be self-loaded.

3485 lbs. additional drawbar pull will load its equivalent in earth; i.e., 3485 more poundsof dirt.

$$\frac{3485 \text{ lbs. ton}}{2000 \text{ lbs.}} = 1.743 \text{ tons.}$$

(1.743 tons)(100 lbs./ton) = 174.3 lbs. pull set up by the added 3485 lbs. of dirt. This will load an equivalent of 174.3 lbs. or earth.

$$\frac{174.3 \text{ lbs. ton}}{2000 \text{ lbs.}} = .087 \text{ tons additional load.}$$

(.087 tons)(100 lbs./ton) = 8.7 lbs. of additional pull set up which will load an equivalent amount of dirt.

This process can be continued down to an infinitesmal amount of material.

The total amount of material that can be self-loaded operating down a 5% grade equals

$$21{,}000 \text{ lbs.} + 3485 \text{ lbs.} + 174.3 \text{ lbs.} + 8.7 \text{ lbs} = 24{,}668.0 \text{ lbs.}$$

$$\frac{24{,}668 \text{ lbs. cu. yd.}}{2{,}700 \text{ lbs.}} = 9.15 \text{ cu. yds., the load}$$

that the scraper can pick up unassisted.

The capacity of the Bucyrus Erie S152 scraper is 17 cu. yds. heaped or 13.5 cu. yds. struck. A pusher or a steeper grade will be needed to pick up a full load.

In the above example the total weight of the tractor, scraper, and the material self-loaded equals:

$$69{,}700 \text{ lbs.} + 3485 \text{ lbs.} + 174.3 \text{ lbs.} + 8.7 \text{ lbs.} = 73{,}262 \text{ lbs.} \tag{63}$$

A study of equation (63) shows that

3485 = (69,700)(C/2,000), where C = (% grade)(20). In equation

(63) C = (5)(20) = 100. (64)

In equation (63)

174.3 = (3485)(C/2,000) = (69,700)(C/2,000)(C/2,000). (65)

$$8.7 = (174.3)(C/2,000) = (69,700)(C/2,000)(C/2,000)(C/2,000). \quad (66)$$

Substituting the values given in equations (64), (65), and (66) into equation (63) gives:

$$\text{Total weight} = (69,700) + (60,700)(C/2,000) + (60,700)(C/2,000)^2 + (60,700)(C/2,000)^3. \quad (67)$$

Equation (67) is a geometric progression of the type

$$a + ar + ar^2 + ar^3 + ar^4 \, . \, . \, . \, . \, . \, . \, . \, . \quad (68)$$

The progression in (68) is convergent is $-1 < r < 1$, and if convergent the sum is found by

$$S = \frac{a}{1 - r} . \quad (69)$$

where S = the sum,

a = the first term, and

r = a ratio which is greater than - 1 but less than + 1.

To use equation (69) to find scraper capacity the terms will be changed so that:

S = TW = weight of tractor + weight of scraper + total weight of material that can be loaded.

a = SLW = self-loaded weight = weight of tractor + weight of scraper + weight of material that can be self-loaded on level ground, and

r = C/2000, where C = (% grade)(20 lbs. per ton per percent grade).

Substituting the above values into equation (69) gives

$$TW = \frac{SLW}{1 - (C/2000)} . \quad (70)$$

Checking the results of Example 23 by using equation (70) gives

$$TW = \frac{69,700}{1 - (100/2000)} = 73,368.4 \text{ lbs.}$$

Pounds of payload are then equal to total weight minus the weight of tractor and empty scraper.

Pounds of payload = 73,368.4 - 48,700 = 24,668.4 lbs.

$$\text{Volume of load} = \frac{24,668.4 \text{ lbs. cu. yd.}}{2,700 \text{ lbs.}} = 9.15 \text{ cu. yd.}$$

Example 24

Using the same data given in Example 23 and equation (70) find the favorable grade down which the scraper can load completely without the use of a pusher.

When loading down a 10% grade,

$$G = (10)(20) = 200.$$

$$TW = \frac{69,700}{1 - (200/2000)} = 77,500 \text{ lbs.}$$

$$\text{Vol.} = \frac{77,500 - 48,700}{2,700} = 10.7 \text{ cu. yds.}$$

A ten percent grade is not steep enough.

When loading down a 15% grade,

$$G = (10)(20) = 300.$$

$$TW = \frac{69,700}{1 - (300/2000)} = 82,000 \text{ lbs.}$$

$$\text{Vol.} = \frac{82,000 - 48,700}{2,700} = 12.33 \text{ cu. yds.}$$

A 15 percent grade is also inadequate.

When loading down a 20% grade,

$$G = (20)(20) = 400.$$

$$TW = \frac{69,700}{1 - (400/2000)} = 87,200 \text{ lbs.}$$

$$Vol. = \frac{87,200 - 48,700}{2,700} = 14.25 \text{ cu. yds.}$$

A 20 percent grade will give a struck load.

When loading down a 25% grade,

$$G = (25)(20) = 500.$$

$$TW = \frac{69,700}{1 - (500/2000)} = 93,000 \text{ lbs.}$$

$$Vol. = \frac{93,000 - 48,700}{2,700} = 15.68 \text{ cu. yd.}$$

A 25 percent grade will probably be adequate.

In practice scrapers can be loaded down grades as steep as 50 percent. Almost every project can be laid out so that scrapers can load down hill. The economy of the practice is readily apparent.

Troughing as in Figure 78 is another method by means of which scraper efficiency can be increased. In the Figure, cuts 1, 2, and 3 are carried lower than 5 and 4. This practice aids in carrying a better grade. This also prevents the tractor, scraper, and pusher from skidding to either side. Digging in the troughs prevents end spill. Sections 5 and 4 are loosened considerably as troughs 1, 2, and 3 are being excavated. This loosening enables the scraper to pick up a load more easily when digging from 5 and 4. Cuts 1 and 2 must not be carried so deep that the machinery will be unable to straddle the mound at 5.

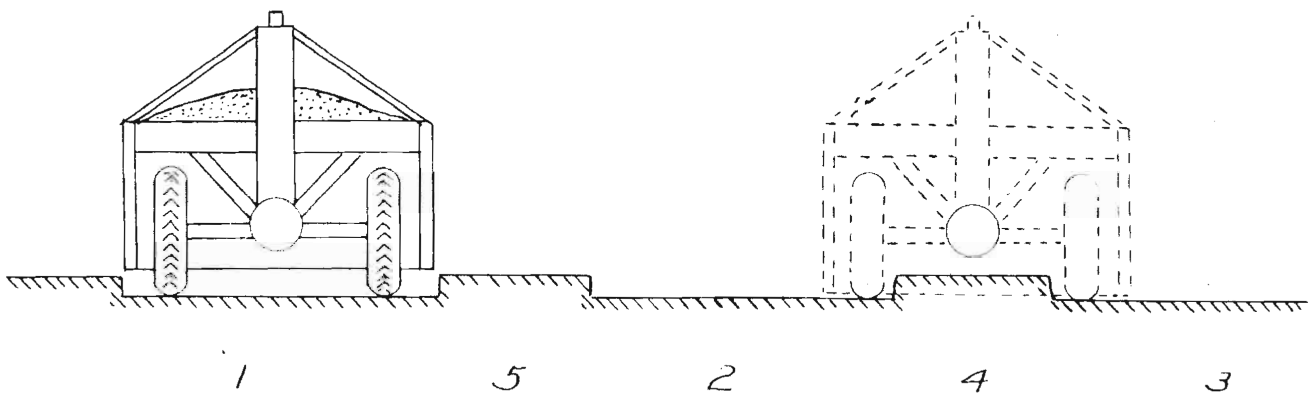


Figure 78. Scraper loading.

Another method used to increase the size of the scraper load is known as "pumping." In this practice the operator drops the cutting edge of the scraper and begins to load. When the scraper approaches its self-loading capacity the tractor crawlers begin to slip or the motor might lug down. At this point the cutting edge is raised and the machine is allowed to pick up speed. After enough speed has been picked up, the cutting edge is dropped and more material is forced into the bowl. This method takes advantage of the force of the inertia possessed by the total weight.

Figure 79 illustrates scraper loading in rooted areas. In the Figure, the first load has been taken from the designated area. About two-thirds of the second load is picked up from a freshly rooted area. The scraper finishes the second load in an area over which the first scraper has already worked. The reason for this that the material in the overlapped area is harder and so it will force more material into the scraper bowl. There is less tendency for this harder material to spill around the ends of the cutting edge of the scraper.

The distance over which the scraper loads varies as the dimensions of the scraper and the depth of cut. Generally this distance will be about 100 feet. Loading speed is determined by the speed in low gear of the tractor. Haul and return speeds likewise depend on the tractor characteristics. Turning takes very nearly 20 seconds. Dumping time can be found by the following equations: (41)

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(41) Park, K. F., Principles of modern excavation and equipment. p. 60, Peoria, Ill., R. G. LeTourneau, Inc., 1942.

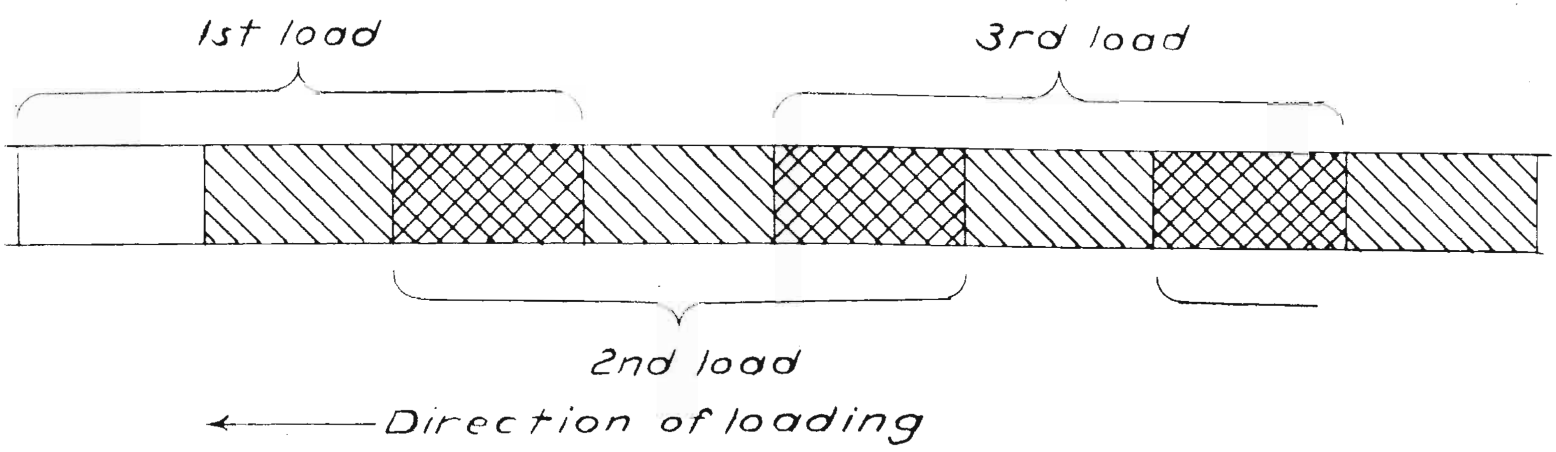

Figure 79.[40] Scraper loading procedure in a rooted area.

(40) After Park, K. F., Use of rooters to make scrapers more effective; The Co-operator, Vol. 5, No. 1, p. 5, February, 1943.

$$DT = \frac{\text{Cu. ft. of material in bowl}}{\text{(Travel speed, ft/min)(Depth of spread in ft)(Bowl width in ft)}}$$

where DT = dumping time.

Average figures that can be used to estimate scraper cycle time are given by T. R. Paulsen.[42] Including allowances for

(42) Paulsen, T. R., Post-war application of tractor equipment: Excavating engineer, Vol. XXXVIII, No. 1, pp. 38-40, January, 1944.

loading, dumping and returning, round trip speed will average 440 feet per minute with excellent conditions, 330 feet per minute under fair conditions and 275 feet per minute under poor conditions. These figures are applicable on hauls of from 500 feet to 1500 feet.

The latest development in scrapers is the Tournapull. This machine has a large advantage over the crawler drawn scraper in that much higher haul speeds can be realized. It has the disadvantage of decreased tractive efficiency. The Tournapull does not have the self-loading ability and the grade ability of the crawler tractor.

The data in Table XVII which follows will illustrate the advantage that higher speeds gives the Tournapull. It also illustrates the importance of speed as haul distances increase.

TABLE XVII (43)

SCRAPER PERFORMANCE

Super C Tournapull with LP Carryall

Haul Distance	Yards per Hour*	Cost per Hour**	Cost per Cu Yd. (Cts.)
800 feet	140	$ 6.75	4.8
1,000 feet	129	6.75	5.3
2,000 feet	96	6.00	6.3
3,000 feet	75	5.62	7.5
6,000 feet	45	5.40	12.0

* Corrected to job efficiency
** Includes pusher cost pro-rated.

113 HP Tractor with LP Carryall

800 feet	80	$ 5.00	6.3
1,000 feet	70	5.00	7.2
2,000 feet	43	5.00	11.2
3,000 feet	30	5.00	16.6

(43) Evans, R. D., Earthmoving economy - Part III, The Co-operator, Vol. 8, No. 2, p. 14, March, 1946.

HAULAGE

Several of the factors pertaining to haulage have been discussed in the preceding sections. Among these factors are the coefficient of friction, rimpull, rolling resistance, tire penetration, grade resistance, grade ability, economical grades, speed, and acceleration.

The applications of the above factors have been brought out in the examples.

Example 1 illustrates the effect of swell of material on the payload. Example 10 illustrates the effect of rolling resistance on haulage speed and cost. The effect of grades on speed is illustrated in Example 13. The choice of proper grades is illustrated in Example 17 and 18. Example 19 illustrates the correction of attainable speed rate to find an average rate.

The haulage cycle is the sum of the spotting time at the shovel, the loading time, the haul time loaded, the turning and dumping time, and the return time.

To allow for fixed times in haulage cycles, rough assumptions were used in Example 13. Closer average values are given in Table XVIII. These values are for Euclid trucks but can be used for any other comparable type.

A study of Table XVIII shows that the rear dump truck uses the most time in dumping. The reason for this is that the rear dump truck must stop, shift, and back up to the edge of the dump or to the bin as illustrated in Figures 80 and 81.

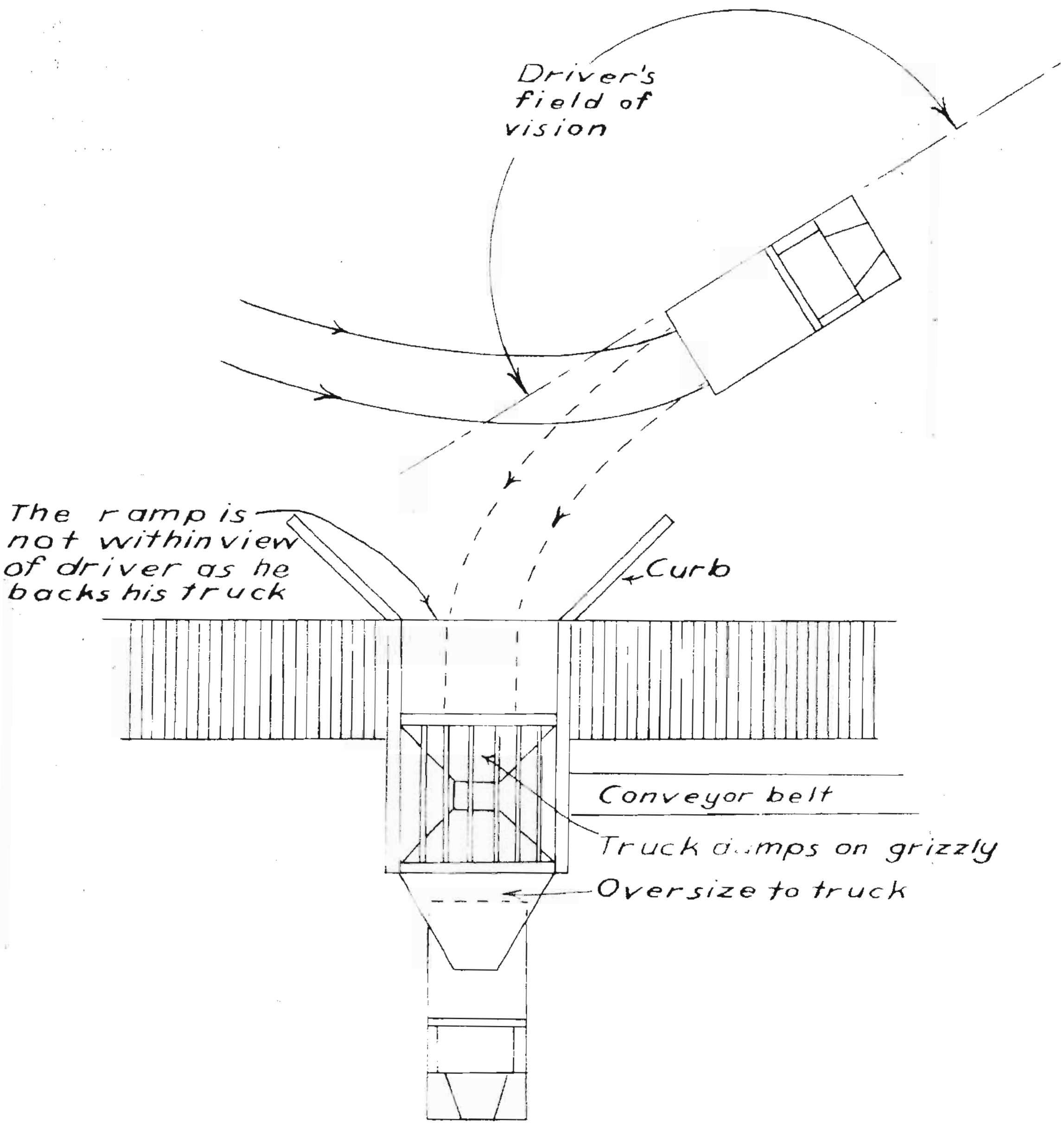

Figure 60. The incorrect road and arrangement at the dumping point.

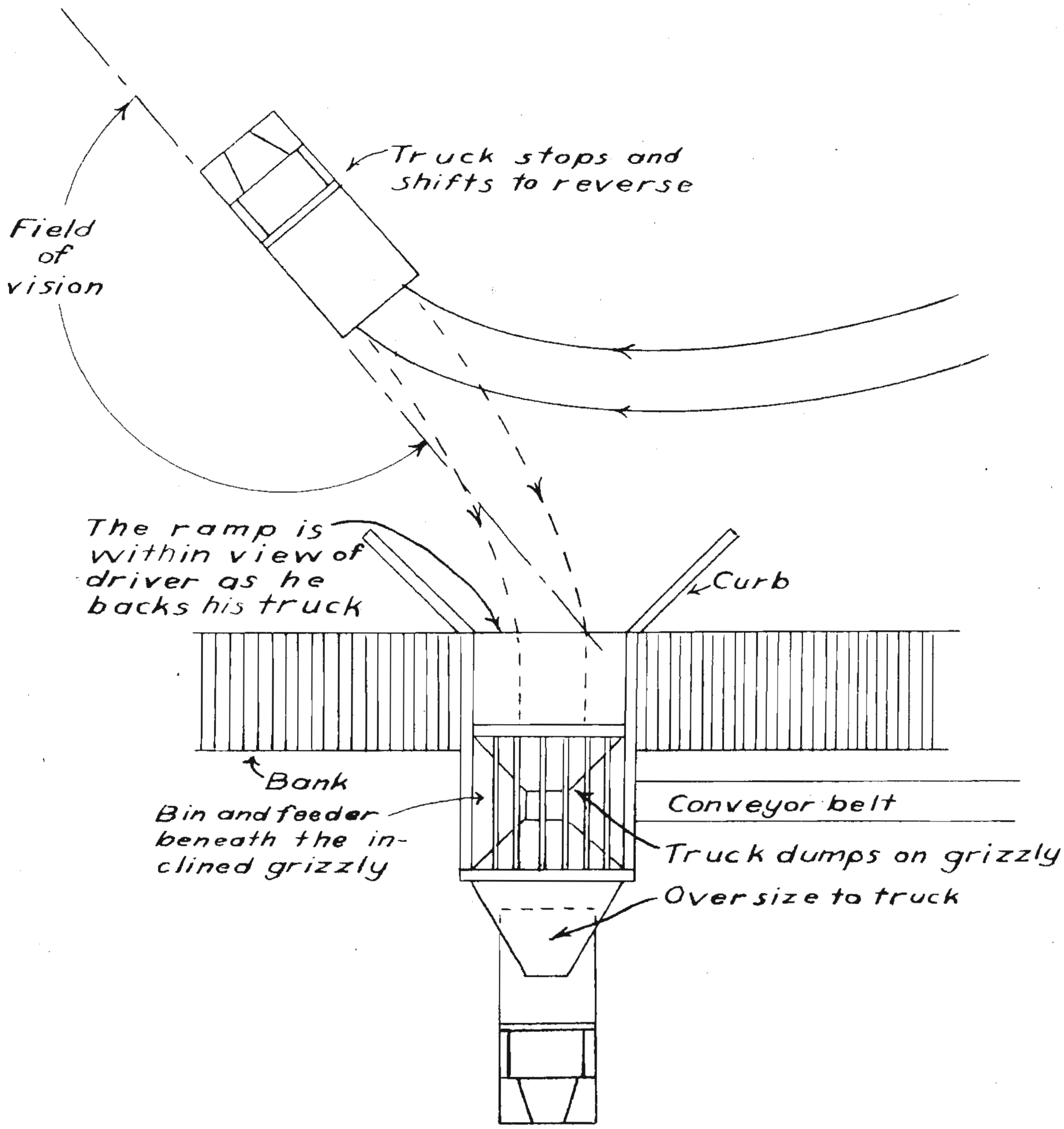

Figure 21. The correct road arrangement at the dumping point.

TABLE XVIII [44]

TOTAL TURNING AND DUMPING TIME PER HAULING CYCLE

Operating Conditions	Bottom Dump Tractor-Semi-Trailer	Euclid Rear Dump	Euclid Side Dump Tractor-Semi-Trailer
Favorable	0.7 Min.	1.5 Min.	1.0 Min.
Average	1.0 Min.	2.0 Min.	1.5 Min.
Unfavorable	2.0 Min.	2.5 Min.	2.0 Min.

(44) Euclid Road Machinery Company., Estimating production and costs of material movement with Euclids, Form No. 350-R, p. 7, Cleveland, Ohio, 1946.

The bottom dump tractor--semi-trailer, on the other hand, drives over the bin or pocket in a forward gear. The tractor and trailer straddle a narrow opening over the pocket. The load is discharged through this opening and the unit moves on in a forward gear to form a loop in the road. Some of the newer installations have long bins and automatic dumping arrangements so that the truck need not stop but merely slows down somewhat while discharging its load.

The side dump tractor--semi-trailer likewise does not back into a dumping position. It proceeds in forward to the edge of the dump or bin, dumps over the side, and continues on in forward.

To speed up the truck cycle of rear dump trucks, several types of drive-over pockets have been designed and installed. On one type the truck drives over the pocket and stops in the dumping position. The deck over which the truck has driven is

Then operated by hydraulic or pneumatic pistons and cylinders so that it comes apart and leaves an opening through which the truck can dump. After discharging its load the truck moves on in a forward gear. The deck is closed so the next truck can drive over.

A variation on these types of pockets is to have a permanent opening between the ramp treads. The truck straddles the opening as it drives over the pocket. The truck stops in the dumping position, dumps, and moves on in forward. Some of the material remains on the treads. To remove this, sheet steel skirts which form part of the curb and which are operated by hydraulic or pneumatic rams, push the dirt through the opening as they are forced inward. The skirts return to their position and the tread is clear of obstructions so the next truck can drive over.

The more elaborate of these drive-over pockets have slight depressions for the rear wheels at the dumping position and slight humps for the front wheels of the truck. The depressions aid the driver in spotting his truck quickly and accurately. The difference in elevation between front and rear wheels gives a slope to the truck body and makes dumping more rapid and easier.

By using these drive-over ramps or pockets, valuable seconds can be saved while dumping. There is also a decrease in driver fatigue since this method eliminates shifting once and, more important, it eliminates the necessity of the driver to turn and stretch as he must while backing up as in Figures 80 and 81.

If it is necessary to back up to dump at a pocket or at a dump, roads should be laid out as in Figure 81. When roads are laid out in this manner, the driver has a better field of vision and can back up more rapidly and more safely.

In Figure 80 the driver's field of vision does not include the entrance to the pocket. In this case the driver must back up slowly until the entrance of the pocket comes within his field of vision. This method is not as safe as that illustrated in Figure 81.

If using a shuttle-truck such as a Koehring Dumptor, no turns are necessary. Spotting at the shovel or at the dumping point is rapidly and safely done. These trucks, however, are not economical on long hauls because of small capacity.

Table XIX gives average values for time used spotting the truck at the loading machine.

TABLE XIX (45)

SPOTTING TIME AT LOADING MACHINE

Operating Conditions	Euclid Bottom Dump Tractor-Semi-Trailer	Euclid Rear Dump	Euclid Side Dump Tractor-Semi-Trailer
Favorable	0.15 Min.	0.15 Min.	0.15 Min.
Average	0.50 Min.	0.30 Min.	0.50 Min.
Unfavorable	1.00 Min.	0.50 Min.	1.00 Min.

(45) Euclid Road Machinery Co., Estimating production and costs of material movement with Euclids, Form No. 350-R, p. 7, Cleveland, Ohio, 1946.

A study of Table XIV shows that generally the rear dump truck is more maneuverable than the tractor--semi-trailer type. Under favorable conditions none of the types need back into dumping position so the spotting times are the same. This condition is illustrated in Figure 18. Where conditions are not favorable, it is necessary to back in. When this is necessary the less maneuverable tractor--semi-trailers use more time than the rear dump truck.

If double-spotting, as in Figure 20, is used, the spotting time need not be considered in the truck cycle since one truck can spot while another truck is being loaded.

If it is necessary to back up in spotting at the loading machine, the trucks should back in so the driver has a clear field of vision.

The loading time for the haul unit may be computed in either of two ways. The shovel output in cubic yards, loose measurement, per hour can be divided by 60 to find the shovel output per minute. The truck capacity in cubic yards, loose measure, can be divided by the shovel capacity per minute, to find the loading time in minutes. The other way is to determine the average cycle time for the shovel and the number of passes necessary to load the shovel. The product of these two factors will give the loading time in minutes.

To compensate for delays haulage computations are usually based on a 50-minute hour. The truck cycle time divided into 50 minutes will give the number of trips per hour. The number of trips per hour times the truck capacity in cubic yards gives the number of cubic yards moved per hour. The number of trucks required equals the hourly production desired divided by truck capacity per hour.

Example 13

A 150 hp rear dump truck of 10 cu. yd. capacity is to be used to haul blasted rock from a 5-cu. yd. power shovel. The shovel can produce 450 cu. yd., bank measure, per hour. The swell factor of the rock is .67.

The mechanical efficiency of the truck is .90. The gross vehicle weight is 66,000 lbs. Conditions are favorable both at the shovel and at the back-in pocket.

Road conditions are as follows:

Road	Length	Grade	RoRi
Shovel to haul road	300 ft.	2%	160 lbs.
Haul road	600 ft.	5%	60 lbs.
Haul road	1000 ft.	0%	60 lbs.

The truck can return in high speed of 37 miles per hour.

The travel speed over the different sections of the road is found by substituting in equation (45).

Speed over the first section equals

$$\frac{(750{,}000)(.9)(150)(.9)}{(20 \times 2 + 160)(66{,}000)} = 6.9 \text{ miles per hour.}$$

Average speed factor from Table XV is 0.5. The average speed is

$$(.5)(6.9) = 3.45 \text{ miles per hour.}$$

$$(3.45)(88 \text{ ft./min.}) = 303.6 \text{ ft. per min.}$$

The time over the first section of road is then

$$\frac{300 \text{ ft. min.}}{303.6 \text{ ft.}} = .99 \text{ minutes.}$$

The speed over the second section of road equals

$$\frac{(750,000)(.9)(150)(.9)}{(20 \times 5 + 60)(66,000)} = 8.63 \text{ miles per hour.}$$

The average speed factor from Table XV is .7. The average speed is then

$$(.7)(8.63) = 6.041 \text{ miles per hour.}$$

$$(6.041)(88 \text{ ft. per min.}) = 531.6 \text{ ft. per min.}$$

The time over the second section of road is then

$$\frac{(600 \text{ ft.})(\text{min})}{(531.6 \text{ ft.})} = 1.13 \text{ minutes.}$$

The speed over the third section equals

$$\frac{(750,000)(.9)(150)(.9)}{(20 \times 0 + 60)(66,000)} = 23 \text{ miles per hour.}$$

The average speed factor from Table XV is .8. The average speed is

$$(.80)(23 \text{ miles per hour}) = 18.4 \text{ miler per hour.}$$

$$(18.4)(88) \text{ ft. per min.}) = 1619.2 \text{ ft. per min.}$$

The time over the third section equals

$$\frac{(1000 \text{ ft.})(\text{min.})}{(1619.2 \text{ ft.})} = .67 \text{ minutes.}$$

The average speed factor from Table XV for the return trip is .85. The average return speed is then (.85)(37 miles per hour) = 31.45 miles per hour. (31.45)(88 ft. per min.) = 2767.6 ft. per min.

The return time is then

$$\frac{(1900 \text{ ft.})(\text{min.})}{(2767.6 \text{ ft.})} = .69 \text{ minutes.}$$

The shovel loading capacity in loose cubic yards per minute is

$$\frac{450 \text{ cu. yds.}}{.67} = 670 \text{ cu. yd.}$$

$$\frac{670 \text{ cu. yd.}}{60 \text{ min.}} = 11.2 \text{ cu. yd. per minute.}$$

The average time to load one truck equals

$$\frac{(10 \text{ cu. yd.})(\text{min.})}{(11.2 \text{ cu. yd.})} = .9 \text{ minutes.}$$

The truck cycle becomes the total of the individual times

Spot at loading machine.	= 0.15 min.
Loading time	= 0.9 min.
Time on first road section	= 0.99 min.
Time on second section	= 1.13 min.
Time on third section	= 0.67 min.
Turn and dump time	= 1.5 min.
Return time	= 0.69 min.
Total cycle time	= 6.03 min.

The number of trucks needed to keep the shovel operating continuously assuming perfect timing and coordination

$$\frac{6.03}{.9} = 6.7 = 7 \text{ trucks}$$

The number of trips per truck per 50-minute hour equals:

$$\frac{(50 \text{ min.})(\text{trip})}{(6.03 \text{ min.})(\text{hour})} = 8.3 \text{ trips per hour per truck.}$$

The actual production is equal to the number of trucks times the capacity per truck times the number of trips per hour, as follows:

(8.3)(7)(10 cu. yd., loose measure) = 580 cu. yd., loose measure, per hour.

(.67)(580 cu. yd.) = 390 cu. yd., bank measure, per hour.

The choice of the size of hauling units is a problem of both technology and economics. The size of the haulage units should be such that it takes at least two, preferably three or more, passes of the excavating unit to load it. The reason is readily apparent. A truck may be rated as being able to haul ten tons. This means it can haul a ten ton load; it does not mean that the truck can stand up under the impact of receiving this ten tons at one time as would be the case if it was loaded with one dipper full.

The grade of the haul roads is a limiting factor on the size of trucks. The large tractor—semi-trailer units do not have the grade ability possessed by the six or ten wheel dump trucks. Of the different haul units the crawler-tractor and trailer unit has the greatest grade ability.

The size of haul units becomes increasingly important as the length of haul increases. If the speed rate is constant, cost of haulage per cubic yard decreases as the size of the truck increases.

Speed of haul units also becomes increasingly important as the length of haul increases. A higher speed decreases the cost of transport per cubic yard as length of haul increases. This is illustrated by the comparisons in Table XVII. As speeds increase the importanse of road construction insreases. High speeds demand smooth hard surfaces.

COSTS

The method of computing costs has almost become standardized to the method given here. With reference to earthmoving equipment, costs consist of the cost of ownership and the cost of operation.

Cost of ownership is a fixed cost. It is made up of the first cost of the machine plus interest on the first cost plus taxes, storage, and insurance.

The first cost is the sum of the price of the machine F.O.B. factory, freight, erection or installation charge, and testing cost.

To find the yearly cost of ownership it is necessary to determine the life of the machine. The life of trucks, crawler-tractors, scrapers, small shovels, and similar equipment is commonly taken as 10,000 operating hours. Schedules have been set up by the Associated General Contractors of America, Inc., the Power Crane and Shovel Association, and others. The life of equipment as given in these various schedules agrees closely. The life of the equipment in years is then the life in hours divided by the number of operating hours per year.

The depreciation cost per year is the first cost divided by the life in years. The depreciation cost per hour is the depreciation cost per year divided by the number of operating hours per year.

Interest, taxes, and insurance are commonly taken as ten percent of the average yearly investment. This ten percent consists of interest, six percent; taxes, two percent; and insurance, two percent.

The average yearly investment as a percent of the first cost can be taken from Table XX. In general, the average yearly investment in dollars will be

$$\frac{(n + 1)}{(2n)} \text{ (First cost)},$$

where "n" is the life of the machine in years.

Cost of operation is made up of the operator's wages, social security, compensation, power and fuel cost, maintenance, tire costs, wire rope costs, repairs and replacements, oil and grease, and similar items.

Fuel costs are usually computed by the cost per horsepower-hour. Fuel consumption per horsepower-hour is best estimated by fuel consumption for similar equipment over a period of time.

Tire costs per hour are usually arrived at by estimating tire life in hours, and less commonly tire life in miles. A tire life of 3,000 hours is average. Wire rope costs can be estimated from life of rope in hours or in tons or yards of material moved.

Maintenance, repairs, and replacements are taken as a percent of first cost per year or as a percent of the average yearly investment per year. The schedules published by the Associated General Contractors of America and the Power Crane and Shovel Association give average figures for these costs as a percent of the first cost.

TABLE XX

DEPRECIATION

Depreciation Period in Years	Percent Yearly Depreciation	Average Yearly Investment is Equal to:
1 year	100	100% of first cost
2 years	50	75% of first cost
3 years	33.33	66-2/3% of first cost
4 years	25	62.5% of first cost
5 years	20	60.0% of first cost
6 years	16-2/3	58-1/3% of first cost
7 years	14-2/7	57-1/7% of first cost
8 years	12.5	56¼% of first cost
9 years	11-1/9	55-4/9% of first cost
10 years	10	55.0% of first cost
11 years	9-1/11	54-6/11% of first cost
12 years	8-1/3	54-1/6% of first cost
13 years	7-9/13	53-11/13% of first cost
14 years	7-1/7	53-4/7% of first cost
15 years	6-2/3	53-1/3% of first cost

Oil, grease, and greasing labor are usually estimated at a fixed amount per operating hour.

The best basis for estimating costs is complete and accurate costs records and performance data of like equipment operating under conditions similar to those anticipated.

Example 14 is given to show how hourly costs can be computed.

<u>Example 14</u>

The costs of ownership and operation of a 1½ cu. yd. Diesel shovel:

Cost of the machine f.o.b. factory. . . .	$30,000
Freight.	650
Assembly and testing	400
Total first cost	$31,050
Depreciation at 20% (5 years),	
(.60)($31,050)	6,210
Annual interest charge at six percent	
of average yearly investment,	
(.06)(0.6)($31,050)	1,117.80
Annual taxes, and insurance,	
(.04)(0.6)($31,050)	745.20
Maintenance and repairs at 15% of	
first cost	4,657.50
Total annual fixed charge	$12,730.50

Fixed charges per hour based on two shifts per day, five days per week, and 47 weeks a year for a total of 3760 hours.

Cost of ownership per hour,	
($12,730.50)/(3760)(hours)	$ 3.39
Labor, operator, per hour.	1.90
Labor, oiler, per hour	1.12

Fuel, six gallons @ 14¢, per hour. . . . $ 0.84

Oil and grease per hour 0.20

Total hourly cost of owning and operating . . $ 7.45

The cost per yard or per ton of material handled will be the capacity of the machine per hour divided by the cost of owning and operating per hour.

The cost of moving material per unit is a good index of the efficiency of operation and management.

CONCLUSIONS

There is no all-purpose earthmoving machine. Each earthmoving unit operates at maximum efficiency under definitely restricted working conditions. It is management's duty to ascertain the conditions of maximum efficiency for each machine and, as far as is possible, to match the existing conditions with the proper machine.

The fundamental principles of physics and mechanics can be applied to the study of earthmoving equipment to determine its working capacity and applicability. The various assumptions of values for friction and resistances are no more arbitrary when applied to earthmoving equipment than they are when applied to the flow of fluids or problems in mechanics.

The efficiency and production of earthmoving equipment can be increased through correct application, good operation, proper co-ordination among the different pieces of equipment, elimination of nonproductive movements, time studies, preventative maintenance, and correct sizing of units.

APPENDIX A

Angle of Repose of Some Common Materials

Material	Condition	Angle Degrees
Anthracite	Broken, loose	27
Ashes	Dry	40-45
Cement	Dry	40
Clay	Dry, loose	37
Clay	Dry, natural	4 45
Clay	Damp	27-45
Clay	Wet	16
Coal, bituminous	63% through 10-mesh	34.5
Coal, bituminous	98% through 100-mesh	16
Coal, bituminous	Broken, loose 7	35-45
Coal, bituminous	Slack	37.5-45
Copper ores	As blasted in bank	34-45
Earth	Dry	29
Earth	Moist	45
Earth	Mud	17
Gravel	Clean, loose	37
Gravel and clay	Loose	37
Iron ore, soft	Broken	35
Rock	Hard riprap	45
Sand, clean, loose	Clean, loose	34
Sand and clay	Loose	37
Sand	Wet	22

Material	Condition	Angle Degrees
Sand, clay, gravel	Suction-dredged	26
Stone	Crushed, fines screened out	37
River mud	Suction-dredged	18

APPENDIX B

Weight of Materials

Material	Condition	Lbs. Per Cu. Yd.
Ashes		810
Barite	Broken, loose	4860
Cement, Portland		2540
Clay	Dry, loose	1890
Clay	In place	3140
Clay	Wet excavated	2970
Clayey earth	Rolled dry	2970
Coal	Broken Pennsylvania Anthracite	1540
Coal	Broken bituminous	1400
Coke	Blast furnace	810
Concrete	Wet mixed	3650
Dolomite	Fine or lumps	2700
Earth	Excavated common loam, dry	2160
Earth	Excavated common loam, moist	2430
Earth	Excavated common loam, wet	2970
Granite	Broken	2600
Gravel	Screen ¼ inch to 2 inches	2840
Gravel and sand	Pit run	3240
Gravel	Dry	3020
Gravel	Wet	3375
Gypsum rock	Crushed	2700
Iron ore (Hematite)	Loose	4050

Material	Condition	Lbs. Per Cu. Yd.
Lignite	Broken	1404
Lime		1730
Limestone	Block	4860
Limestone	Broken	2700
Magnesite	Broken	2840
Marl	Wet excavated	3780
Masonry	Debris	2430
Mud	Fluid	2970
Peat	Moist	1350
Peat	Wet	1890
Phosphate rock	Broken	2970
Plaster	Ground	1620
Quartz	Pulverized	2025
Salt	Loose	2080
Sand	Dry	2700
Sand	Slightly damp	2835
Sand	Packed	2970
Sand	Wet	3300
Slag	Broken furnace	2970
Slate		2600
Stone, average	Crushed	1215-2700
Stone, heavy	Crushed, loose	2890
Sulphur		3375
Trap rock	Broken	2840

APPENDIX C

Characteristics of Some Equipment

Model A Tournadozer

Engine . . . . Butane-Propane
Maximum Brake HP 750 at 1800 RPM at Sea Level

Speed in MPH at 1800 Engine RPM
1st. 1.5
2nd. 3.1
3rd. 6.9
4th Gear 13.8

Tires . . . . 4 (30.00 x 33)28

Blade
Width of Blade . . . 16' 6"
Height of Bowl . . . 4' 8"

Overall measurements
Length . . . . 21' 9"
Width . . . . 16' 6"
Wheelbase . . . . . . 9' 0"

Approximate shipping weight, lbs. . . . . 93,000.

Model B Tournadozer

Engine
1125 cu. in. displacement, 4 cycle, supercharged 8 cylinder maximum brake HP—300 at 1800 RPM at Sea Level

Speed in MPH at 1800 engine RPM
1st 1.6
2nd 3.3
3rd 6.8
4th 13.6
The same four speeds available in reverse.

Steering
Two wheels on each side controlled by air actuated disc clutches and brakes.

Tires . . . . 4(24.00 x 29)24

Model B Tournadozer, cont'd

Blade
Width of blade . . . . 13' 10"
Height of bowl . . . . 4' 8"

Overall measurements . . . .
Length . . . . 19' 9"
Width . . . . 13' 10"

Wheelbase . . . . 7' 6¼"

Approx. shipping weight, lbs . . . 50,700.

Model C Tournadozer

Engine . . . . (Diesel)
844 cu. in. displacement 4-cycle, 6 cylinder Diesel.
Maximum brake HP - 180 at 1800 RPM at sea level.

Speed in MPH at 2,000 engine RPM
1st Gear 1.91
2nd Gear 3.81
3rd Gear 7.62
4th Gear 15.24
The same 4 speeds available in reverse.

Steering
Two wheels on each side controlled by air actuated disc clutches and brakes.

Tires . . . . 4(21.00 x 25)16

Blade
Width of blade . . . . 11' 2"
Height of bowl . . . . 43"

Overall measurements
Length . . . . 16' 1"
Width . . . . 11' 3"
Wheelbase . . . . 5' 11-5/8"

Approximate shipping weight, lbs. . . . . .31,000.

D Roadster Tournapull with E-9 Carryall

Overall measurements
- Length . . . 27' 10-3/4"
- Width . . . 8' 0"
- Wheelbase . . . 15' 7-3/8"
- Height (without cab) . . . 8' 2"

Tires
- Tournapull . . . 2(14.00 x 32)12
- Carryall . . . 2(14.00 x 32)12

Turning radius, min . . . 19'

Approximate shipping weight . . . 20,700 lbs.

Engine . . . (Diesel)
448 cu. in Displacement, 4 cycle,
4 cylinder, 1800 RPM, governed speed,
maximum brake HP 100 at 1800 RPM at sea level.

Speed in MPH at 1800 engine RPM
- 1st Gear Fwd . . . 2.77
- 2nd Gear Fwd . . . 5.19
- 3rd Gear Fwd . . . 10.17
- 4th Gear Fwd . . . 17.69
- 5th Gear Fwd . . . 25.27
- Low Reverse . . . 2.76
- High Reverse . . . 5.28

Speed based on ring gear and pinion ratio of 2.58 to 1.

Capacity
- Tons . . . 9
- Cubic yards . . . 7

Cutting edge
- Length . . . 7'

Maximum depth of spread . . . 26-1/2"

B Tournapull with E-35 Carryall

Overall Measurements
- Length . . . 38' 7-3/8"
- Width . . . 11' 7"
- Height . . . 11' 3"
- Wheelbase . . . 24'

B Tournapull with E-35 Carryall, cont'd

Tires
Tournapull . . . . 2(24.00 x 29)36 or 24
Carryall . . . . 2(24.00 x 29)36 or 24

Turning radius, minimum 16'6"

Shipping weights, approximate
Tournapull 25,800
Carryall 21,000

Engine . . . . (Diesel)
1125 cu. in. Displacement, 8 cylinder Engine, Max. Brake H.P., 240 at 1800 RPM at sea level.

Speed in MPH at 1900 RPM
1st Gear, Fwd. and Reverse 2
2nd Gear, Fwd. and Reverse 4
3rd Gear, Fwd. 8
4th Gear, Fwd. 16
Speed based on ring gear and pinion ratio of 3.857 to 1.

Differential
Tournamatic, torque-proportioning.

Carryall specifications

Capacity
Tons 35
Cubic yards 30

Cutting edge
Length . . . . 9' 6"

Maximum depth of spread . . . . 18"

C Tournapull with E-26 Carryall

Overall Measurements
Length . . . . 30'
Width . . . . 12' 4"
Height . . . . 9' 3"
Wheelbase . . . . 18' 2"

Tires
Tournapull . . . . 2(21 x 25)20
Carryall . . . . 2(21.00 x 25)20

C Tournapull with E-16 Carryall, cont'd

Turning Radius	Minimum 14' 6"
Shipping weight, approximate	
Tournapull	14,720
Scraper	14,500
Engine	(Diesel)
844 cu. in. Displacement 6 cylinder engine	
Max. brake HP -- 150 at 1800 RPM at Sea Level	
Speed In MPH	
1st Gear, Fwd. and Rev	2.19
2nd Gear, Fwd. and Rev	4.38
3rd Gear, Forward	8.65
4th Gear, Forward	17.3
Speed based on 1800 governed Engine RPM and 4.5:1 Ring Gear and Pinion Ratio.	
Capacity	
Tons	16
Cubic yards	13.3
Cutting edge.	8' 6"
Maximum depth of spread	21-3/4"

C Tournapull with E-C Tournarocker

Overall Measurements	
Length	26' 1"
Width	10' 3-½"
Height (with cab)	9' 8-½"
Wheelbase	15' 3"
Tires	
Tournapull.	2(21.00 x 25)20
Tournarocker.	2(21.00 x 25)20
Turning radius, minimum	12' 6"
Approximate Shipping Weight (Combined).	28,500 lbs.
Capacity of Tournarocker	
Tons	16
Cubic yards	14.5

C Tournapull with E-C Tournarocker, cont'd

Bowl

Width	10' $3\frac{1}{2}$"
Height (From ground to top)	6' 6"
Height (When dumping) . . . . (dumps to the rear)	13' 7"
Length	11' 2"

C Tournapull with W-10 Tournatrailer

Tournatrailer Struck Capacity	9.5 yds
Heap capacity	12 yds
Tires, tournatrailer	2- 21.00 x 24
Tires, tournapull	
Weight	25,000 lbs.

B Tournapull with W-20 Tournatrailer

Struck capacity	14.0 yds.
Heap capacity	19 yds.
Tires, trailer	2 - 30 x 40
Tires, tractor	same

B Tournapull with W-30 Tournatrailer

Struck	21.1 yds.
Heap	30 yds.
Tires	2 - 30 x 40

B Tournapull with W-40 Tournatrailer

Struck	26.1 yds.
Heaped	35 yds.
Tires	2- 30 x 40

C Tournapull with "LS" Carryall

Engine Horsepower	90 yds.
Carryall Struck measure	8.2 yds.
Carryall Heap measure	11 yds.

C Tournapull with "LS" Carryall, cont'd

Cutting width	8' 6"
Tire equipment — carryall (a)	4 - 14.00 x 20
or (b)	2 - 16.00 x 20
Tire equipment — Tournapull	2 - 21.00 x 24
Approximate shipping weight	24,950 lbs.

Super "C" Tournapull with "LP" Scraper

Engine horsepower	130 or 150
Carryall Struck measure	12.1 yds.
Carryall Heap measure	15 yds.
Cutting width	8' 6"
Tire equipment — carryall	2 — 18.00 x 24
Tire equipment — Tournapull	2 — 21.00 x 24
Approximate Shipping Weight	30,500 lbs.

B Tournapull with "LU" Carryall

Engine horsepower	120 or 200
Carryall Struck measure	15.0 yds.
Carryall Heap measure	19 yds.
Cutting width	8' 6"
Tires — Carryall	2 — 24.00 x 32
Tires — Tournapull	2 — 24.00 x 32
	or 2 — 30.00 x 40

B Tournapull with "FU" Carryall

Engine horsepower	200
Carryall Struck measure	17.7 yds.
Carryall Heap measure	23 yds.
Cutting width	10 ft.
Tires — Carryall	2 — 24 x 32
Tires — Tournapull	2 — 24 x 32

A Tournapull with "NU" Carryall

Engine horsepower (2 engines)	400
Struck measure	33.3 yds.
Heap measure	45 yds.
Cutting width	10 ft.
Tires — Carryall	30 x 40 (2)
Tires — Tournapull	36 x 40 (2)

Caterpillar D8 Tractor

Approximate working weight is 35,500 pounds.

Belt horsepower is 131.

Gear	Speed (mph)	Drawbar pull (lbs)
1st, forward	1.6	26,208
2nd, forward	2.2	19,537
3rd, forward	2.6	15,973
4th forward	3.0	13,707
5th, forward	3.6	11,266
6th, forward	4.9	7,995
1st, reverse	1.6	
2nd, reverse	2.6	

Catepillar D7 Tractor

Approximate working weight is 25,300 pounds.

Belt horsepower is 92.8.

Gear	Speed (mph)	Drawbar pull (lbs)
1st, forward	1.4	21,351
2nd, forward	2.2	13,454
3rd, forward	3.2	9,090
4th, forward	4.6	5,994
5th, forward	6.0	4,550
1st, reverse	1.6	
2nd, reverse	2.6	
3rd, reverse	3.8	
4th, reverse	5.4	

Caterpillar D6 Tractor

Approximate working weight is 17,500 pounds.

Belt horsepower is 65.0.

Gear	Speed (mph)	Drawbar pull (lbs.)
1st, forward	1.4	14,300
2nd, forward	2.3	9,100
3rd, forward	3.2	6,200
4th, forward	4.4	4,000
5th, forward	5.8	2,650
1st, reverse	1.8	
2nd, reverse	2.8	
3rd, reverse	3.9	
4th, reverse	5.4	

Allis-Chalmers HD-19

Approximate working weight is 40,000 pounds.

Belt horsepower is 163.

Gear	Speed (mph)	Drawbar pull (pounds)
Low, forward	0 to 3.0	36,000
High, forward	0 to 7.0	28,000
Reverse	0 to 5.5	36,000

This machine is equipped with a torque converter.

Allis-Chalmers HD-14

Approximate working weights is 29,000 pounds.

Belt horsepower is 150.48.

Allis-Chalmers HD-14, continued

Gear	Speed (mph)	Drawbar pull (lbs.)
1st, forward	1.7	28,019
2nd, forward	2.2	22,699
3rd, forward	2.8	17,265
4th, forward	3.5	13,769
5th, forward	4.4	10,074
6th, forward	7.0	5,579
1st, reverse	2.0	
2nd, reverse	3.2	

Allis-Chalmers HD-10

Approximate working weight is 21,000 pounds.

Belt horsepower is 101.62.

Gear	Speed (mph)	Drawbar pull (lbs.)
1st, forward	1.7	19,002
2nd, forward	2.0	15,507
3rd, forward	2.7	11,421
4th, forward	3.8	7,867
5th, forward	4.6	6,190
6th, forward	6.0	4,157
1st, reverse	1.8	
2nd, reverse	4.2	

International Harvester TD-18 Tractor

Approximate working weight is 23,200 pounds.

Belt horsepower is 84.66.

Gear	Speed (mph)	Drawbar pull (lbs.)
1st, forward	1.5	18,973
2nd, forward	2.0	13,357
3rd, forward	2.5	10,561
4th, forward	3.3	7,827
5th, forward	4.6	5,157
6th, forward	5.7	3,839
1st, reverse	1.5	
2nd, reverse	3.4	

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VITA

Woodrow John Latvala, son of John Arthur Latvala and Ida Elaine Latvala, was born at Great Falls, Montana, 10 April 1917. He attended public schools at Nashwauk, Minnesota, and was graduated from Nashwauk High School in 1935.

From 1936 to 1942 he worked as a carpenter in heavy construction, a heavy equipment operator, and a plantman in the construction and mining industries. Among his employers were the Great Northern Railroad, Johnson Coal Company, Streater Lumber Company, Butler Brothers Mining Company, Evergreen Mines Company, M. A. Hanna Mining Company, and Nick F. Helmers and McWilliams Dredging Company, Inc.

He enlisted in the United States Naval Construction Battalions in 1942 and served until April 1945.

He enrolled at the New Mexico School of Mines in August 1945 and was graduated in May 1948 with the degree of Bachelor of Science in Mining Engineering.

He became Instructor in Mining Engineering at the Missouri School of Mines and Metallurgy in June 1948.