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THE TRANSPORTATION OF TUNNEL MUCK BY PIPELINE

by

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16. Abstract

The objective of this study was to advance the technology of tunnel excavation by increasing the rate of muck removal from the tunnel face. If the advancement of the technology of muck removal does not keep pace with advances in tunneling machine technology, muck removal can become the limiting constraint on the forward movement of the tunnel face, and hence, on the growth of tunneling.

A previous study for the U.S. Dept. of Transportation examined a pneumatic and hydraulic pipeline muck haulage system. It was apparent that some areas of the analysis of the muck transportation system could be improved.

This report updates muck quantities and to some extent, muck quality (in terms of its hardness and geology). Crushing equipment is examined more thoroughly as is extensible conveyor equipment. A survey of extensible equipment is made to aid in suggesting approaches for their application in tunnels to pipeline muck haulage. Recent headloss data for coarse slurries are presented for the hydraulic muck haulage system. Consideration is given to a jet pump eductor for feeding a centrifugal pump from a mixing tank. A more compact and less expensive dewatering system in analyzed.

17. Key Words
Tunneling, muck, pneumatic pipeline, hydraulic pipeline, crushing, extensibility, dewatering, jet pumps

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PREFACE

This study was sponsored by the Office of Rail Technology, Office of Technology Development and Deployment of the Urban Mass Transportation Administration of the U.S. Department of Transportation. The effort was conducted under contract with the Transportation Systems Center for the Urban Rail Supporting Technology Program.

The objective of this study was to advance the technology of tunnel excavation by increasing the rate of muck removal from the tunnel face. If the advancement of the technology of muck removal does not keep pace with advances in tunneling machine technology, muck removal can become the limiting constraint on the forward movement of the tunnel face, and hence on the growth of tunneling.

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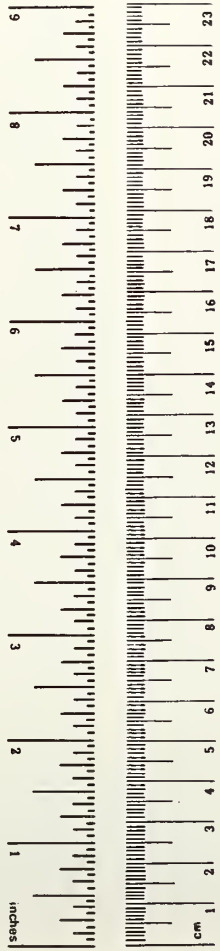
Special thanks go to Ted Miller Associates of Denver, Colorado for the consultation on dewatering systems and to the Colorado School of Mines Research Institute of Golden, Colorado for consultation on crushing technology.

METRIC CONVERSION FACTORS

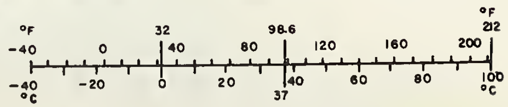
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yerds	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yerds	0.8	square meters	m^2
mi^2	square milas	2.6	squara kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gel	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Calsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	squere centimeters	0.16	squere inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	ecres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilogrems	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fehrenheit temperature	°F



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1. INTRODUCTION

One of the major problems in the rapid excavation of tunnels is muck haulage. Present day muck haulage systems such as rail, truck, and belt have not always been able to remove the muck generated by tunnel boring machines.

Tunneling is a complex process and the muck haulage method, as a sub-system, is also complex. Problems of confined space, wide variations in mucking rates, wear of equipment, noise, and dust all contribute to hostile environment.

In a previous study entitled, "Pneumatic-Hydraulic Material Transport System for the Rapid Excavation of Tunnels", August, 1974,(1) the authors suggested a transportation system for muck haulage consisting of either a pneumatic pipeline or a slurry pipeline. Since pipeline haulage systems were uncommon to American tunneling practice, a total haulage system evaluation was made both technically and economically. Both technical and economic feasibilities of the pipeline haulage systems were hampered by lack of information in several areas:

1. Muck preparation
2. Extensible conveyor systems
3. Pipeline haulage, including pressure drop correlations, special mixing and pumping capabilities.
4. Dewatering of slurries.

The initial evaluations of muck haulage systems by pipeline looked promising, to the extent that the present study was undertaken to investigate further the areas of interest just mentioned. Emphasis was placed on investigating better techniques and technology, rather than costs. The premise was adopted that any improvements in methods and technology were apt to reduce both fixed and operating costs. Component-entry cost data were included wherever readily available but no total haulage system costs were compiled as in the previous study.

Appendixes A through D are provided as background material for the systems and concepts developed in this report.

2. SYSTEM INPUT

For further analysis and refinement of the muck transport system it is necessary to establish the characteristics of the muck which constitute the system input. These are both critical and difficult to quantify adequately for a detailed transport system design. The muck generated reflects geological conditions of the tunnel environment as well as the cutting action of the tunneling machine.

Muck characteristics of primary concern for detail design are volumes, particle size, rock hardness and abrasiveness. Moisture content of the muck is not of immediate concern in the overall design of the transport system. It is assumed that most tunneling will be conducted below the ground water table. The rock which is penetrated will be saturated so that some moisture will always be present in the muck as mined. Moisture may be added at one or more stages for dust suppression. It is assumed that the muck will not be corrosive to the system and that ambient temperatures at the working face are not abnormal.

Available information on prior tunneling projects reveals a great variation in average rates of advance (volume of muck generated) dependent on the tunneling procedures, job efficiency, and geological formations encountered. Detailed data correlating the specific geological formation and tunneling machine performance with muck volumes are limited. By reducing the data to simple straight line relationships, it is possible to project what appear to be reasonable families of curves, correlated with hardness of the rock formation and the year of construction. The year of construction reflects estimates of the state of the tunneling technology. The capabilities and performance (mechanical availability) of tunneling machines have been improving significantly in recent years.

For purposes of establishing the anticipated volumes of muck to be handled in the future, current trends for typical tunnel diameters have been extrapolated into the future to provide better insight into muck system requirements. (See Fig. 2-1 through 2-3). It must be emphasized that these curves were based on prior construction experience with tunneling machines and that they represent a range of data with a significant deviation. These curves are computed from average rates of muck production. Because tunnel boring machines tend to deliver material from the face in surges, peak production rates are a vital consideration for systems design. In this study, a surge rate 25% higher than average muck rates has been assumed for detail component design. While these data might be challenged when related to specific jobs, there appears to be no better basis for establishing system criteria available at this time.

As a frame of reference, it is assumed for this study that the projected muck rates for 1985 can be used as a basis for muck system design.

PROJECTED MUCK RATES (tph) FOR 1985

	Bore Diameters		
Muck Hardness	12'	20'	35'
Soft	353	573	875
Medium	128	213	370
Hard	93	145	250
Very Hard	68	95	165

Direct correlation of muck characteristics from different job sites is possible only in general terms. The range of geologic conditions which may be encountered in the average tunnel is difficult to categorize in simple form. Incomplete or inadequate data on the tunnel route geological conditions often exist for a variety of reasons, among them the physical cost of collecting it, and second, the radical changes that can occur in relatively short distances. Higher rates of tunnel advance are keyed to the desire to construct longer

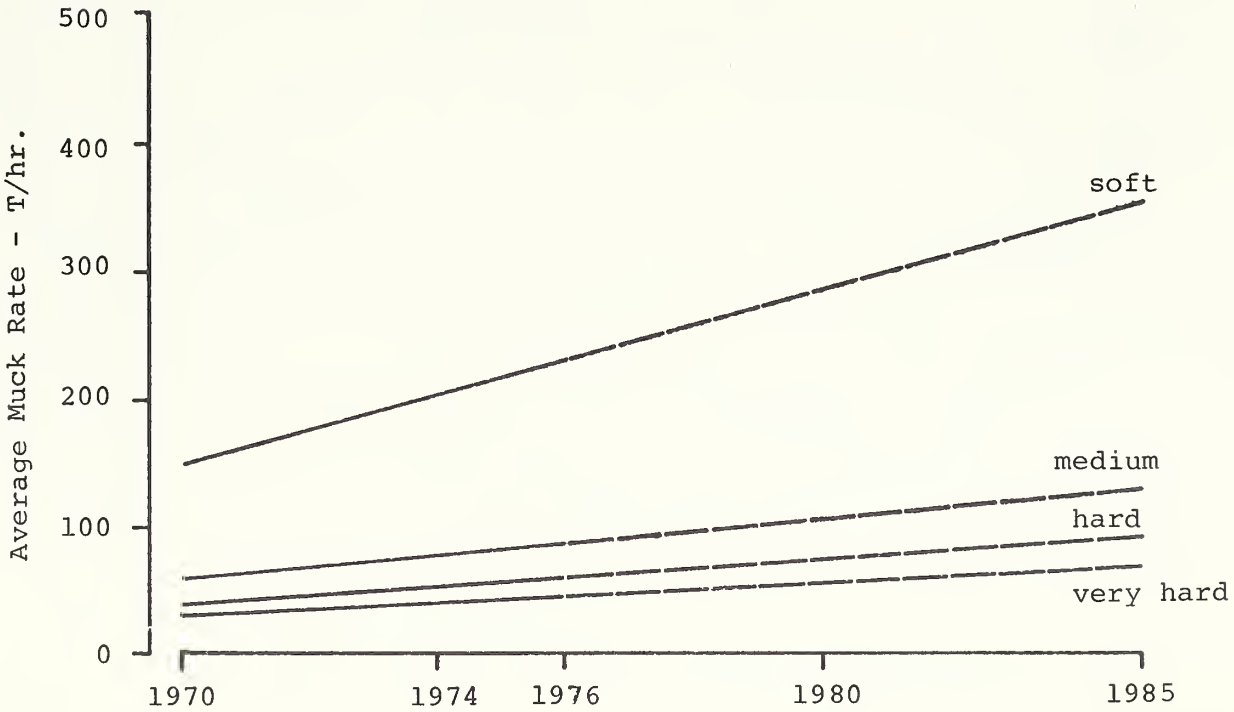


FIG. 2-1 Recent Trends in Tunnel Muck Production: 12 Ft. Bore

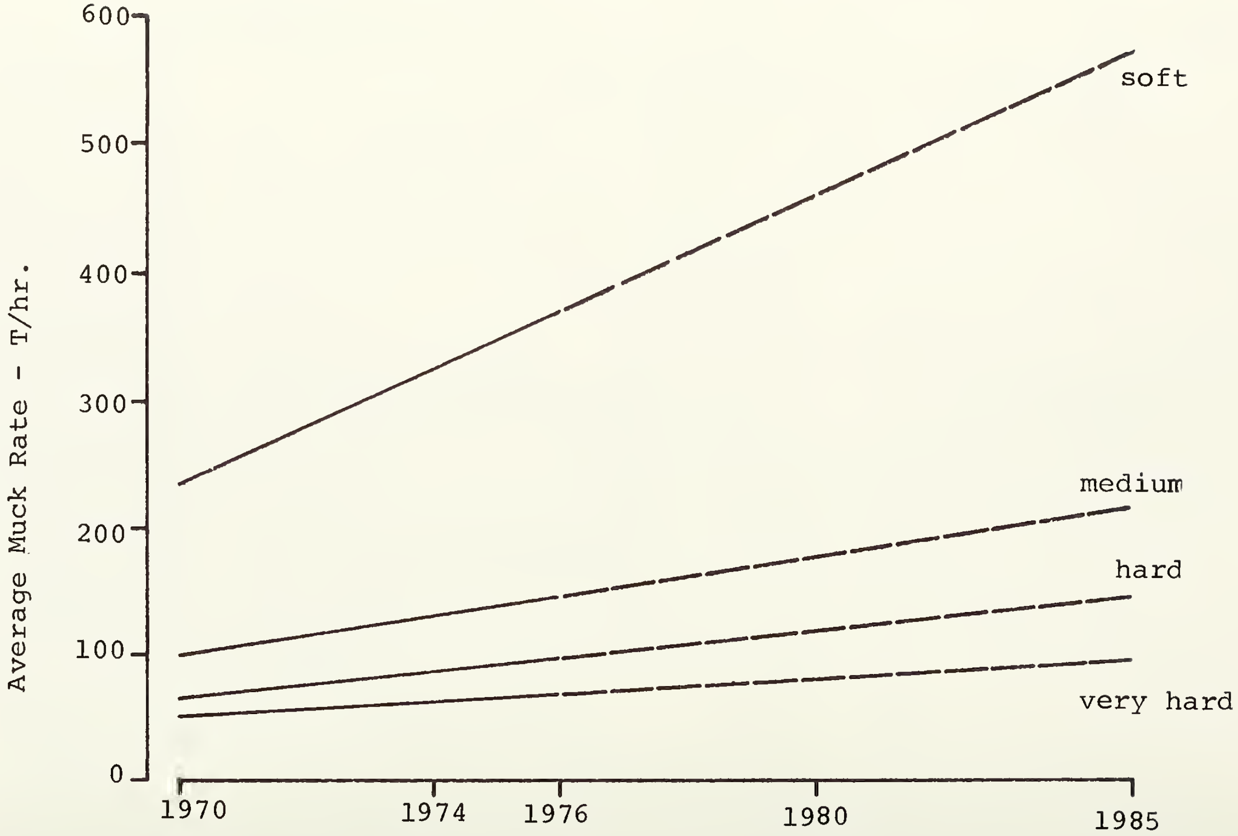


FIG. 2-2. Tunnel Muck Production: 20 Ft. Bore

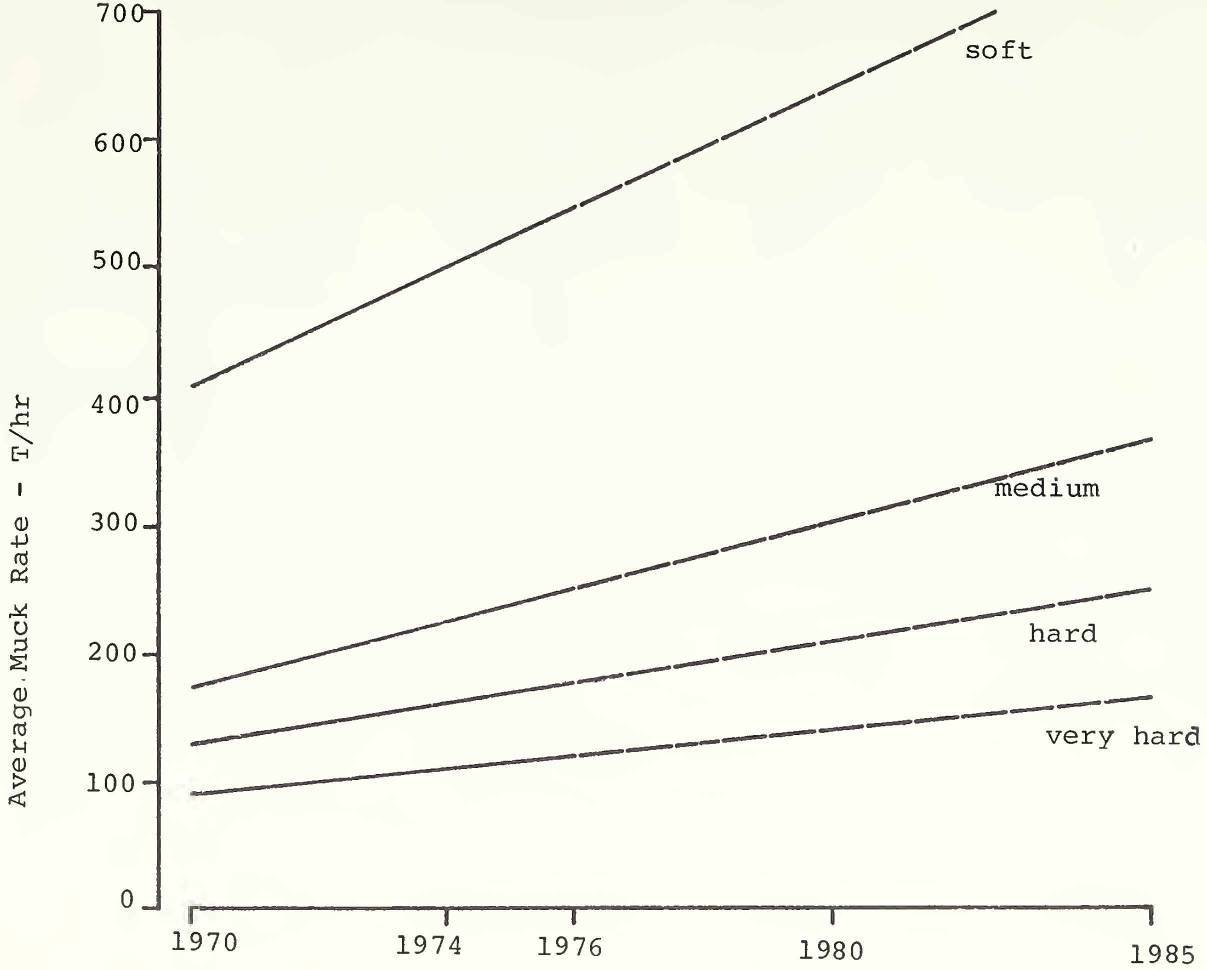


FIG. 2-3. Tunnel Muck Production: 35 Ft. Bore

tunnels, which implies that the range of formations encountered can be extremely great - essentially forcing the muck transport system to be designed with capabilities to handle broad ranges of muck characteristics.

The Holmes and Narver (2) study provides the most comprehensive data available on muck size distribution. They developed a Muck Designation Number (MDN) system to describe the muck characteristics measured at a broad variety of tunneling sites. These data included tunneling operations performed with conventional drill and blast techniques as well as tunneling machines.

The size distribution data included in that study have been reworked to eliminate all but measurements made on the muck produced by tunneling machines. Fig. 2-4 shows the resulting MDN classifications and their corresponding size distributions. Table 2-1 summarizes in outline form general observations for each classification. As the number increases, generally smaller particle sizes are indicated and size distributions tend to become more uniform.

The Holmes and Narver study shows that a relatively minor percentage of total muck volumes is greater than 16" in size. In this study, the +16" fraction is assumed to represent about 1% of the muck total under conditions of normal operation. Some cases may be envisioned where the +16" fraction may be increased disproportionately, but these cases have not been considered in the present study.

Coarse material (+1") is found in all MDN size distributions, indicating that sizing of the muck will be required in all expected working conditions. Excavation in unconsolidated sand, clay, and excessively sticky or free-flowing materials is not normally undertaken with tunnel boring machines and is beyond the scope of this study.

The projected muck rates were based both on estimates of daily tunnel advance and hardness of the rock to be penetrated. There is no specific correlation between advance rate and rock hardness other than the recorded experience from a large number of tunneling projects around the world. The

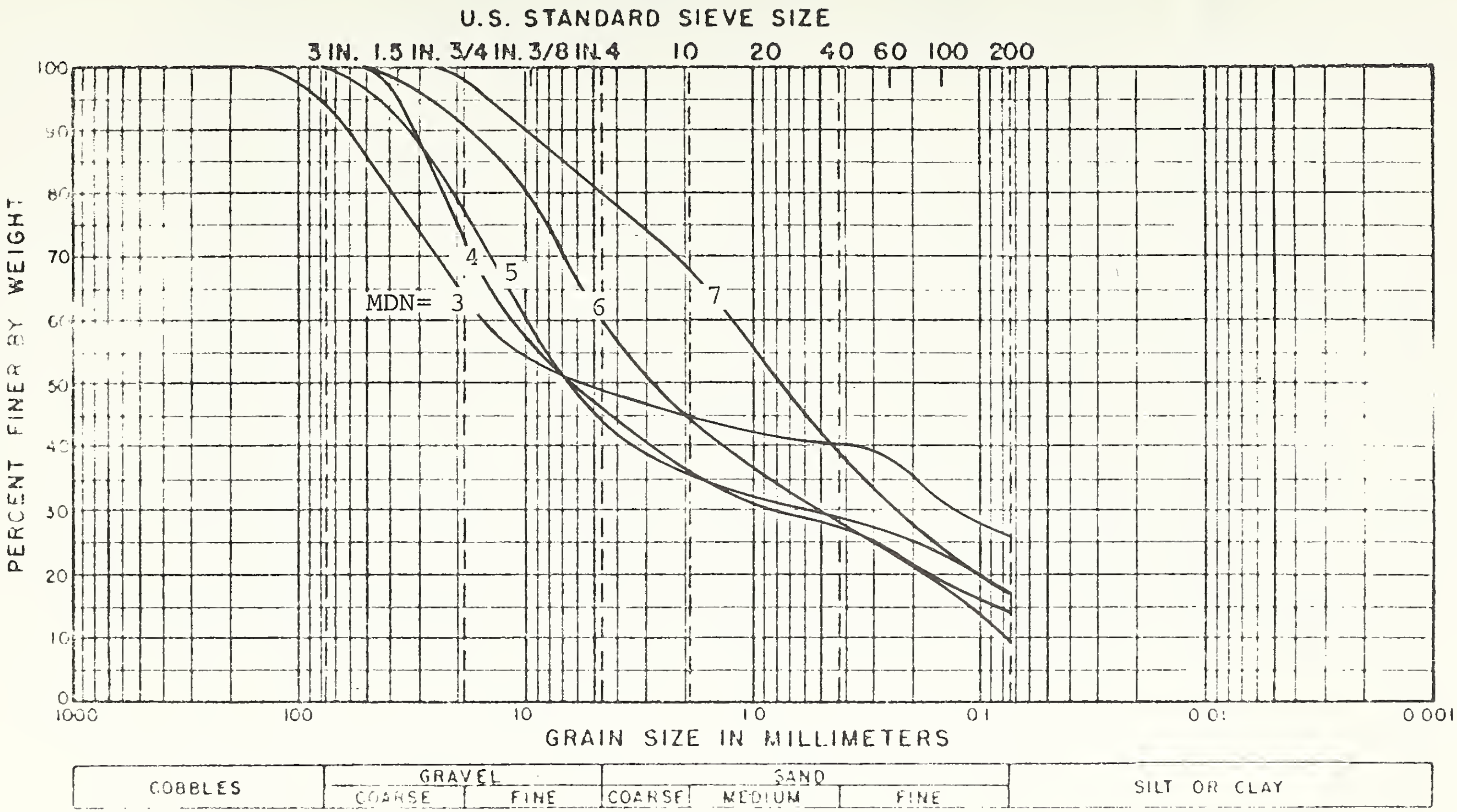


FIG. 2-4. MDN GRADATION CURVES

Source: Holmes and Norver (Ref. 2)

TABLE 2-1

General Observations

Muck Designation Number (MDN) vs. Size Distributions

MDN-3	1.	Obtained in tunnels bored through sedimentary rock of soft to medium hardness.
	2.	Size distribution is "skip-graded": -heavy concentration of +½" material -heavy concentration of -40M material -few fines in the coarse sand range.
	3.	The distorted size distribution is attributable to the excessive volume of -40M generated.
MDN-4&5	1.	Obtained penetrating medium to hard sedimentary rock (sandstones, limestones) and some metamorphic rocks of medium hardness.
	2.	Size distribution is "skip-graded" as seen in MDN-3, but excess of fines (-40M) is not as significant.
MDN-6	1.	Obtained in tunnels penetrating generally hard rock (sedimentary, metamorphic, igneous)
	2.	Muck shows uniform size gradation with small (-2") top size.
MDN-7	1.	Obtained in penetrating fine grained (possibly weakly jointed) rock of soft to medium hardness
	2.	Muck size distribution is very uniform with small (-1") top size.

charts presented earlier have attempted to draw on this experience. Further consideration of rock hardness requires that the categories selected as a reference base be identified in terms of some common test criteria and related, if possible, to typical geological rock types. Table 2-2 summarizes the data in the referenced Holmes and Narver study relating the MDN to the measured physical strength characteristics. It is apparent that the size distribution of the tunneling machine muck does not bear any recognizable relationship with rock hardness. Literature reviewing tunneling machine cutter performance, however, provides further insight into broad categories of rock hardness that can be identified as significant. Table 2-3 incorporating the data from the above table and other published sources would appear to be a current concensus of what might be classified as soft, medium, hard, and very hard rock in tunneling applications.

Rock abrasiveness is an important consideration if any subsequent sizing operations are required on the muck and in the selection and design of the haulage system. Unfortunately this rock characteristic cannot be correlated with any of the other identified muck properties. While it might seem logical to find a consistent relationship between hardness and abrasiveness this is not possible in the variety of rocks encountered in tunneling. It is only possible to rank various common formations, based on laboratory tests, with respect to their "work" or "abrasive" indices. See Table 2-4. These indices provide a reference framework and in some specific cases, can be employed to calculate wear rates. For our purposes in considering the various aspects of muck transport and processing, the indices serve primarily as a warning of potential excessive wear problems if rock with the high indices is anticipated.

TABLE 2-2. MDN Size Distributions vs. Rock Strength

	ARPA Site Code (Ref. 1.)	Rock Type	Unconfined Compressive Strength (KSI) (Ref. 2)	Classi- fication *
MDN-3	NAV-1	siltstone	2	S
	LAY-1	sandstone	10	S
	7-2	sandstone	22	M
	WNG-1	sandstone	<0.5	S
MDN-4	LAW-2	limestone	29	H
	LAW-3	limestone	29	H
	LAW-4	limestone	20	M
	RO-1	sandstone	11	S
	KM-1	mudstone	11	S
	72-1	siltstone	22	M
	EVG-1	limestone	26	H
	EVG-2	limestone	30	H
	NAV-2	sandstone	<1.0	S
MDN-5	5-1	sandstone	22	M
	MIL-1	limestone	36	V
	MIL-2	limestone	36	V
	QL-1	schist	11	S
	MIL-3	limestone	22	M
	LAY-2	conglomerate	22	M
	NY-1	schist	15	M
	NY-2	schist	13	S
MDN-6	CL-1	granite gneiss	9	S
	LK-5	quartz monzon- ite	32	H
	LK-6	quartz monzon- ite	30	H
	CNT-1	conglomerate	28	H
MDN-7	NAV-2	sandstone	<1.0	S
	WNG-2	sandstone	<1.0	S
	NAST-1	granite	18	M
	NAST-2	granite	18	M
	NAST-3	granite	24	M

*S=soft, M=medium, H=hard, V=very hard

TABLE 2-3. Assumed Muck Hardness Classification (Ref. 2)

	Strength Range	Typical Rock Types
Soft	0-15,000 psi	(S) - Salt (S) - Coarse grained, weakly cemented sandstones (S) - Fossiliferous limestones (I) - Altered igneous rocks (S) - Claystones,shales (S) - Coal
Medium	15-25,000 psi	(S) - Marlstones, limestones (M) - Marble (S) - Shales, siltstones, sandstones (M) - Phyllites (M) - Highly micaceous shists (I) - Altered intrusive igneous rocks (M) - Altered metamorphic rocks
Hard	25-35,000 psi	(M) - Slates (S) - Crystalline limestones (I) - Diabase (S) - Silicious, cemented sandstones (M) - Gneisses and schists (M) - Fine grained marble (M) - Pyroxenites (I) - Coarse grained granites
Very Hard	Over 35,000 psi	(M) - Quartzites (M) - Amphibolites (S) - Dolomites (I) - Fine grained granites (I) - Basalt, diabase (I) - Syenites (I) - Gabbros (M) - Iron ores

Rock Classes:
(S) Sedimentary
(I) Igneous Rocks
(M) Metamorphic

TABLE 2-4. Average Abrasion (Ref. 3)

Material	Specific Gravity	Work Index	Abrasive Index
Dolomite	2.7		.0160
Shale	2.62	9.9	.0209
L.S. for Cement	2.7	12.7	.0238
Limestone	2.7	11.7	.0320
Cement Clinker	3.15	13.5	.0713
Magnesite	3.0		.0783
Heavy Sulfides	3.56	11.4	.1284
Copper Ore	2.95	11.7	.1472
Hematite	4.17	8.5	.1647
Magnetite	3.7	13.0	.2217
Gravel	2.68	15.4	.2879
Trap Rock	2.80	17.8	.3640
Granite	2.72	16.6	.3880
Taconite	3.37	16.3	.6237
Quartzite	2.7	17.4	.7751
Alumina	3.9	17.5	.8911

3. MUCK PREPARATION

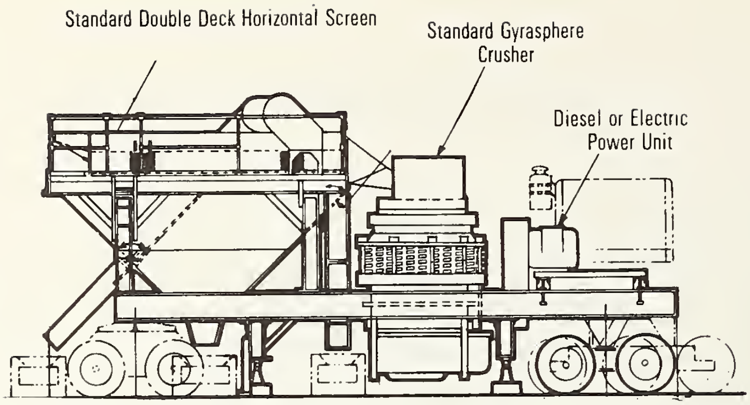
3.1 INTRODUCTION

For pneumatic and/or slurry pipeline transport of the muck from the tunneling machine (characterized in the previous section) consideration must be given to additional preparation of the muck prior to introduction into the transport system. While nothing can be done about the hardness and abrasive characteristics of the muck, it can be reduced in size, and particle shape can be modified to some degree to optimize the pipeline performance. The unit which would physically perform this function can be called a muck preparation unit and would be a skid-mounted assembly pulled along immediately behind the tunneling machine. The muck from the mole would be the unit input and the prepared muck would be discharged directly into the haulage system.

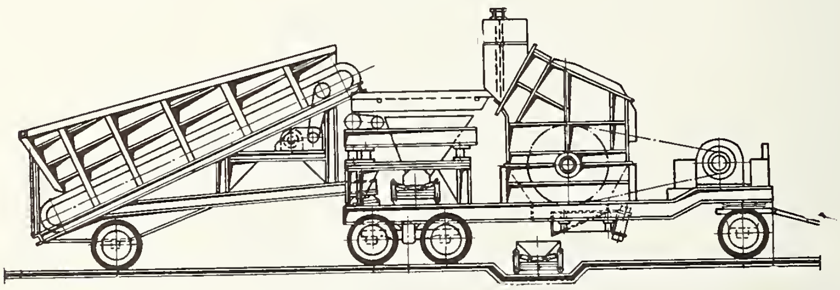
Several commercial portable crushing plants are shown in Figure 3-1 which perform the functions of the muck preparation unit as integrated systems. Sizing may be accomplished with one crusher (single-stage) or two crushers (two-stage), which operate either in open or closed circuit with one or more screens. In closed circuit operations, oversize material is recycled through the plant to obtain closer control of product size.

Commercial units range in capacity from 75 to 400 tons per hour. These units are about 10 to 12 ft. wide, 30 to 50 ft. in length and up to 16 ft. high. Weights may exceed 75 tons for a few plants. In general, the muck preparation unit would be similar in appearance, but simpler in design and more rugged and compact in construction. The muck preparation unit would not require closed circuit operation, and wheels would be eliminated in favor of skid mounting.

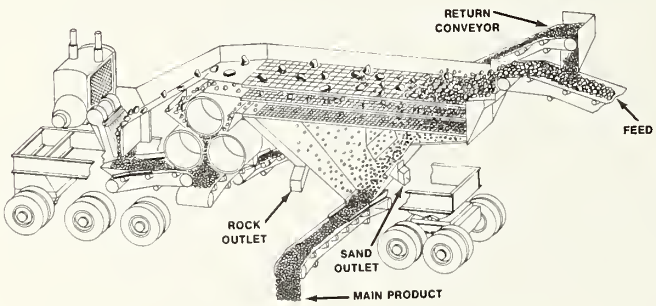
A few portable crushing units are already in use in restricted underground mining applications. The Stamler Corporation produces a line of "feeder-breakers" (Fig. 3-2)



A. Plant with double-deck screen and gyrasphere (cone) crusher. (courtesy Smith Engineering Works.)



B. Plant with apron feeder, scalping screen and impactor crusher. (courtesy Hazemag USA).



C. Plant operating inclosed circuit with triple-deck screen and two-stage (jaw and rolls) crushing. (courtesy Iowa Mfg. Co.)

FIG. 3-1. Examples of Several Commercially Available Portable Crushing Plants.

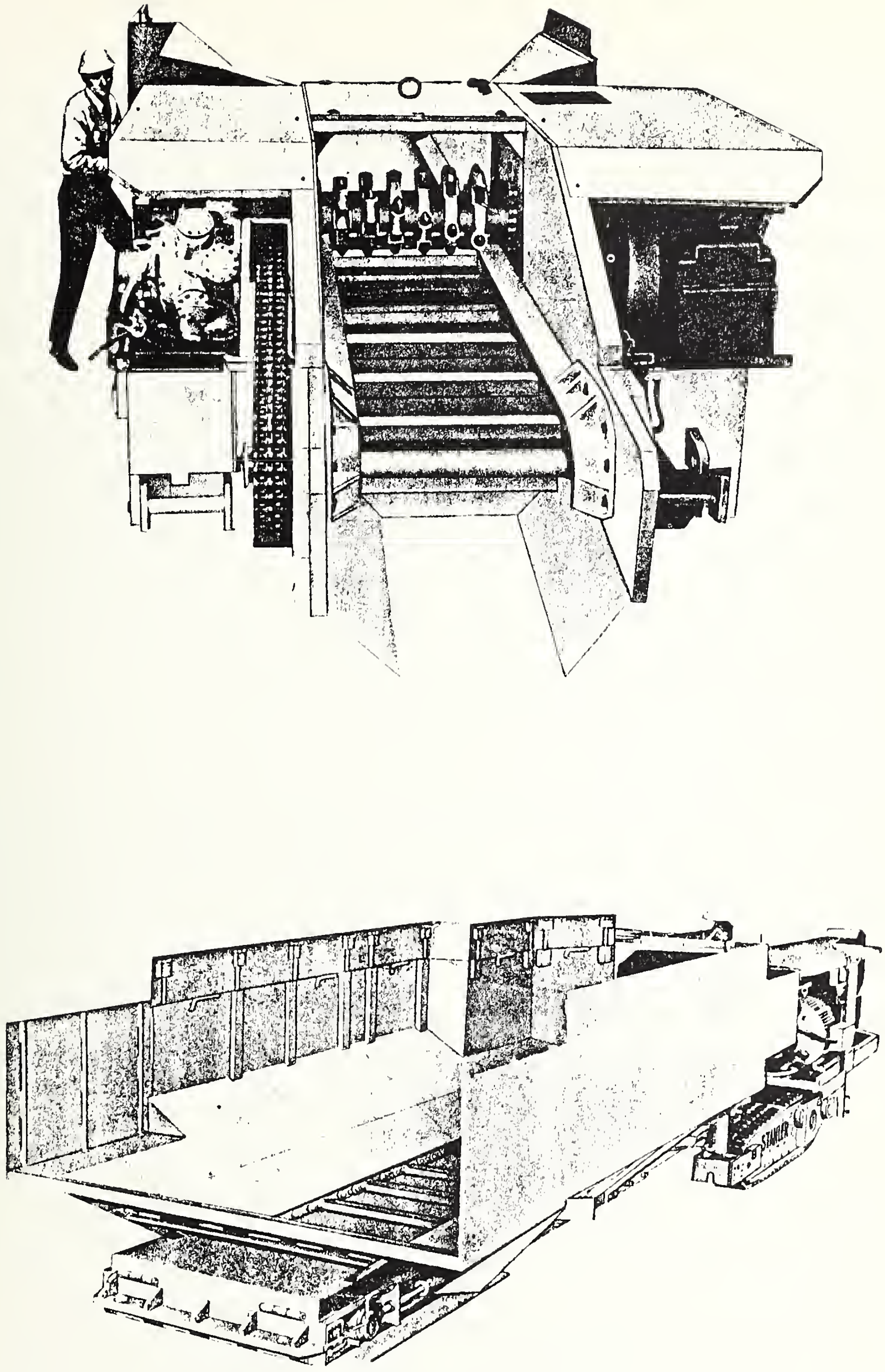


FIG. 3-2 Front and rear views of Stamler feeder-breakers used in underground mining crushing applications. (courtesy Stammler)

which provides the capability of primary size reduction of run-of-mine material at the transfer point to the haulage system (typically a fixed belt conveyor). These units have much simpler configurations than the crushing plants shown in Fig. 3-1. Feeder-breakers are essentially modified belt feeders, incorporating a single, toothed crushing roll mounted in-line.

Feeder-breakers are designed to perform coarse (primary) sizing only, and are presently used in applications where material to be crushed is not excessively hard or abrasive. Although these units are not directly suitable for performing size reduction as required in the proposed muck preparation unit, they include a number of interesting features which could be incorporated into the design of the preparation unit.

(1) The method of breakage results in lower power requirements per ton than are obtained with other types of crushers.

(2) Because feeding and crushing operations are combined, these units are capable of high throughputs (to 3000 tph) but require less head room than conventional crushers.

(3) Capital first costs and operating costs are generally lower than can be obtained with other types of crushers.

(4) Feeder-breakers are specifically designed to assist in evening out surges delivered to continuous haulage systems. This is accomplished with a chain conveyor with built-up sides which functions as a temporary storage hopper.

(5) These units are ruggedly built and are relatively easy to maintain. Worn pick points are quickly and easily replaced; automatic lubrication systems are available as an option.

The detail construction of the muck preparation unit would be customized for a particular tunnel application, and the tunneling machine and haulage system with which it would be mated. For simplicity (reliability) and physical size it would consist essentially of a vibrating screen, intermediate conveyor, crusher(s), a discharge conveyor and mounting frame. This would permit a simple open circuit configuration with the screen providing a means for segregating out the smaller size product to bypass the crusher. There is considerable flexibility in the design of the unit dependent primarily on size limitations imposed by the tunnel diameter. The essential configuration of the proposed muck preparation unit is shown in Figure 3- 3.

All of the components for such units are commercially available and have been used in heavy construction and mining. Crusher selection is the prime consideration dictating the effectiveness of the unit.

A number of fundamentally different concepts have been used in designing crushers which are offered by different manufacturers. For most models, a number of optional features are available to suit specific sizing applications. A detailed examination of crushing equipment and features should be undertaken before crusher selection is made. Manufacturers' representatives can provide valuable assistance in selecting the optimum crusher package to handle given feed and product requirements.

Before discussing the proper crushing equipment one must establish what is required. Ideally the crusher must:

1. be of small physical size adaptable to a portable installation.
2. be safe to work around with acceptable noise and dust levels.
3. require a minimal initial investment and minimal operating cost.
4. be reliable and permit simple maintenance procedures compatible with anticipated operating hours.
5. be capable of handling the anticipated volumes with allowance for surges.

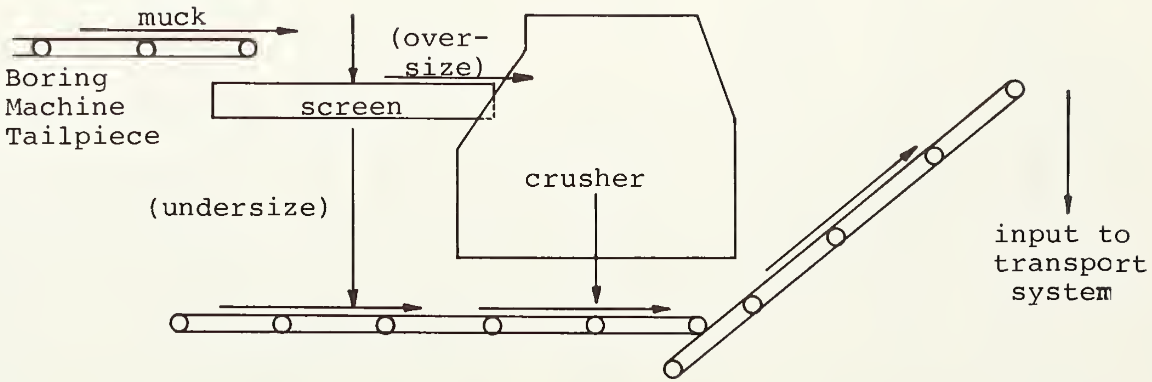


FIG. 3- 3. Diagram of proposed muck preparation unit.

6. be capable of processing rock of the hardness and abrasiveness anticipated in the muck.
7. be capable of providing the necessary size reduction for the input from the tunneling machine.
8. produce an even graded product with a minimum of fines.
9. provide a cubical shaped product.
10. feed and discharge requirements must not increase the vertical profile of the unit excessively.

No hopper is envisioned as being incorporated into the unit, which means that there is no surge capacity built into the system for load leveling other than the limited capabilities inherent in a screen. For study purposes it will be assumed that rocks larger than 16" would occur very infrequently and require special handling procedures.

A high proportion of fines (-200 mesh) if fed into the transport system imposes dust problems at the pneumatic system discharge and/or difficulties in solids separation in the dewatering of slurry systems for water reuse. There are no specific limitations that can be established for fines. Fundamentally, however, crushing systems which produce few fines are preferred over others.

While it is difficult to quantify, experience indicates that a cubical shaped particle is more desirable for pneumatic and/or slurry transport than one with a flat or slab shape. The product from tunneling machines tends to be flat because of the nature of the cutting action. Again, it is difficult to establish how much emphasis should be placed on this consideration in crusher selection.

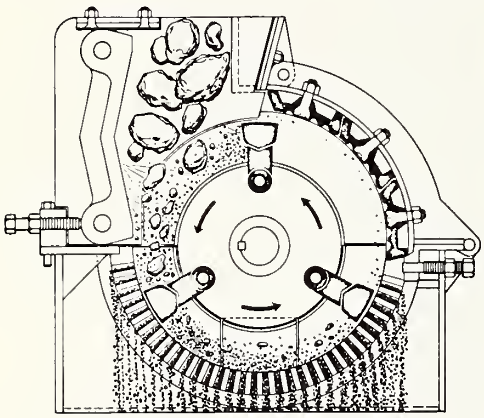
The prior discussion indicates that a crusher for the preparation unit has unique requirements which dictate its selection criteria. With these special needs in mind, all of the crusher types currently available were analyzed in an attempt to establish the preferred configuration. The analysis of each type which follows has been broken down into subcategories.

3.2 BASIC CRUSHER DESIGNS

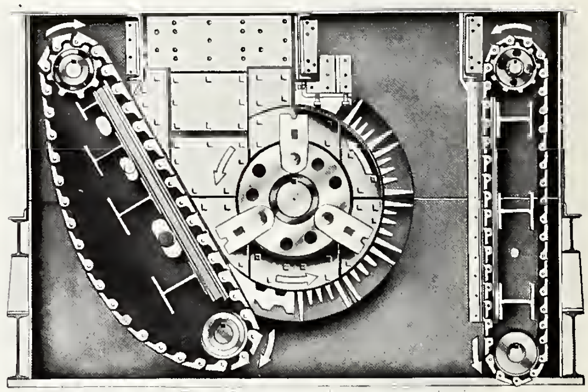
3.2.1 Hammermills

1. Nature of Crushing Action: Hammermills consist of a set of hammers mounted on a heavy central rotor which is rotated at very high speed. Hammers may be either fixed or free swinging on the rotor. Fixed-position hammers are easier to replace. Higher moments of inertia can be obtained with fixed hammers which allow larger feed to be crushed in units with hammers of this type (See Fig.3-4).

    In the upper portion of the crushing chamber, rock is broken against a series of breaker bars or plates (a single large anvil in some cases). A series of grates are arranged around the lower portion of the chamber. Rock entering the hammermill is struck as it falls by hammers moving at velocities of 10 to 12,000 ft/min., causing the rock to collide with breaker plates, bars, etc. in the upper part of the chamber. As the rock reaches the lower part of the chamber after repeated impact cycles



a. Non-reversible hammermill (courtesy Missouri-Rogers)



b. Non-reversible hammermill with traveling breaker plate (courtesy Hammermills,Inc.)

FIGURE 3-4. FIXED HAMMERMILLS

it is ground by the hammers against the grates.

2. Crushing Ratio: Crushing ratio and product gradation are both affected by rotor speed and rock friability (hardness). In general, very high ratios of reduction can be achieved with hammermills - as much as 35:1.
3. Particle Shape: Hammermills (like all impact-type crushers) provide the advantage of a cubic-shaped product, regardless of the shape of particles in the feed.
4. Rock Hardness Range: Restricted to use in crushing rock of soft-medium hardness.
5. Resistance to Abrasive Materials: Size reduction by attrition is an essential feature of the crushing action in hammermills. Manufacturers generally recommend this type of crusher for material with silica content less than 15% (relatively non-abrasive material).
6. Ability to Handle Plastic Fines: Wet or sticky feed does not generally affect the performance of hammermills when specially equipped with moving breaker plates (See Fig. 3- 4b.
7. Wear and Maintenance: Wearing surfaces include hammers, breaker bars, plates and grates. Worn hammers can be replaced on some models, resurfaced on others. Grates are replacement items. Because crushing is accomplished mainly by impact in the upper part of the crushing chamber, hammers normally wear out about five times as fast as grates, and require correspondingly greater attention in maintenance. Many hammermills may be rotated in both directions, doubling the wear life of hammers and breaker parts in the upper crushing chamber.
8. Feed Requirements: Height of fall is a critical consideration for efficient crushing in hammermills. Particles must attain sufficient velocity to enter the crushing chamber to a depth where full impact can be delivered by the hammers. Material can be fed with too much or too

little velocity for efficient crushing to take place.

9. Foundation and Mobility: Need not be very elaborate; most installations employ simple box frame of deep section I-beams. The box section is welded to the assembly prior to shipment by some manufacturers. Hammermills appear to be ideally suited to portable installation, and are often used in portable surface plants.

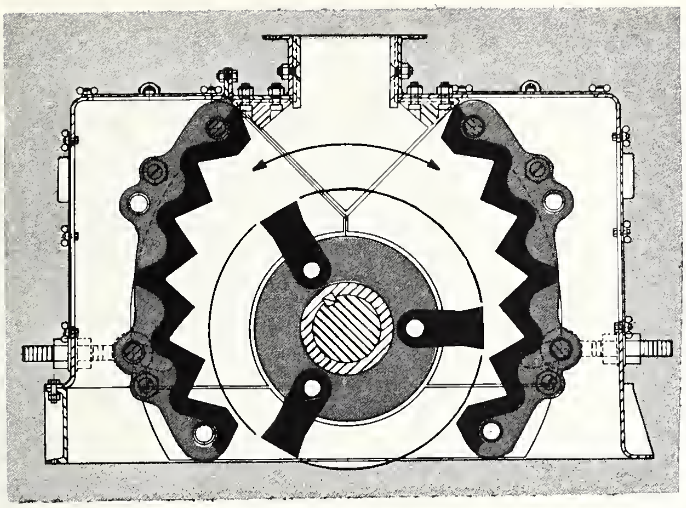
10. Size and Capacity: (Averages based on models available from several manufacturers assuming heavy duty service and maximum product size of 2")

Capacity (tph)	100	300	600
Weight (lb.)	9400	32,200	56,200
Horsepower	110	330	660
Length (ft.)	7-3/4	7-1/4	10-3/4
Width (ft.)	7-1/2	10-1/2	10-1/2
Height (ft.)	9-3/4	12	10-1/2

Single and Double Impactor Crushers

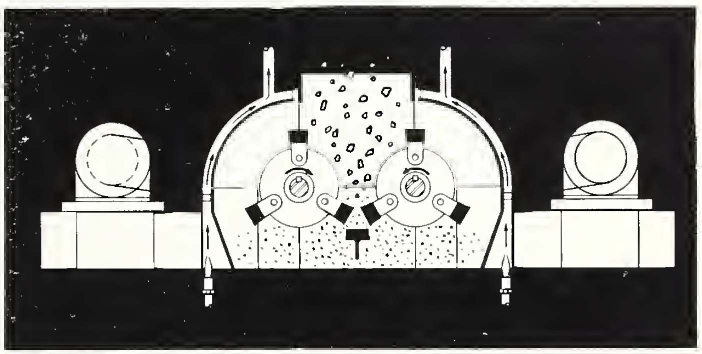
1. Nature of Crushing Action: Impactor crushers and hammermills are similar in both function and appearance. Hammers are mounted on a rotor (two rotors in double impactor types). Rock entering the crushing chamber is struck by the hammers moving at very high velocity. Crushing is accomplished by (1) direct impact from hammers (2) secondary impact of particles against breaker bars and plates (3) collisions between rebounding particles inside the chamber. Unlike hammermills, impactor crushers have no grates, and do not perform fine crushing (See Figure 3-5).

2. Crushing Ratio: As in hammermills, the crushing ratio varies with rock hardness, and may be altered by varying the rotor speed. High reduction ratios are obtained with soft materials,greater than 20:1, and 6 to 10:1 for harder rock.



a. Single Impactor Crusher (Reversible)
(courtesy Williams Patent Crusher & Pulverizer Co.)

b. Double Impactor Crusher



(courtesy Pennsylvania Crusher Corp)

FIGURE 3-5. IMPACTOR CRUSHERS

3. Particle Shape: Provides a cubic-shaped product regardless of characteristics of the feed.
4. Rock Hardness Range: Because attrition is not a significant factor in the crushing mechanism, impactors will handle hard rock successfully.
5. Resistance to Abrasive Materials: Hammer wear tends to occur at excessive rates when abrasive material is crushed. Some manufacturers recommend that impactor crushers be used only for material containing less than 15% free silica.

6. Ability to Handle Plastic Fines: Although some buildup of fines may occur, the performance of impactors is not adversely affected by the presence of fines.
7. Wear and Maintenance: Many impactor crusher models are designed for operation of rotors in either direction to extend life of major wear surfaces. Hammers can generally be resurfaced when worn excessively. Worn breaker bars must be replaced in most cases (not reconditioned). Liner plates on inside of chamber should wear at slower rates than hammers and breaker bars, and may be hardsurfaced. Hammers and liners will work-harden during operation.
8. Feed Requirements: The height of fall must be adjusted so that particles attain the proper trajectory into the path of the rotating hammers for most effective crushing.
9. Foundation and Mobility: Impactors do not require heavy foundations as a rule; even large-capacity impactors may be used in portable plants. Although impactors are not especially bulky, they tend to have tall profiles.
10. Size and Capacity: (Averages based on models available from several manufacturers assuming heavy duty service and maximum product size of 2".)

Capacity(tph)	100	300	600
Weight(lb.)	14,000	28,600	52,500
Horsepower	125	300	600
Length(ft.)	5-1/4	6-1/4	6-1/4
Width(ft.)	4	5-3/4	11-1/2
Height(ft.)	4	5	5

Cage Mills

1. Nature of Crushing Action: A cage mill (See Fig. 3-6). crushes purely by impact. Material is fed into the side of the mill and falls directly into a spinning cage. Material is thrown out of the cage at a very high speed

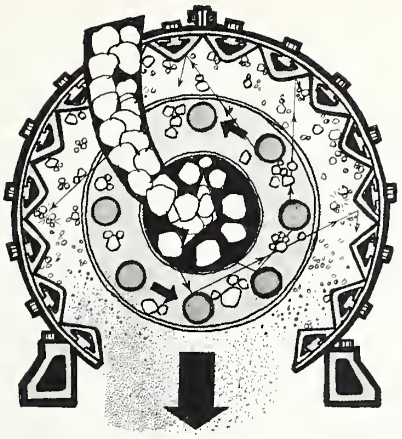


FIG. 3- 6. Single-Cage Mill (courtesy Stedman Foundary & Machine Co.)

to impact against a series of breaker plates located along the perimeter of the mill. A portion of the crushing action is achieved in secondary collisions of particles rebounding back into the stream of rock being ejected from the cage. Because of the high velocities of particles leaving the crushing chamber, a baffle must be installed below the crusher to protect the conveyor belt from impact damage.

2. Crushing Ratio: The largest feed size is limited to 10-12". Size reduction is a function of rock hardness but can be modified by changing cage rotation speed. Maximum ratios of 24:1 can be achieved in soft rock.
3. Particle Shape: Tends to be cubic, regardless of shape of feed.
4. Rock Hardness Range: Will crush material of any hardness. Very hard rock will lead to accelerated wear, but because of ease of maintenance, this is probably a lesser consideration for cage mills than for other types of crushers.
5. Resistance to Abrasive Materials: Because attrition is not an essential part of the crushing action, abrasion is somewhat less critical in producing wear of cage mill parts than is true for other types of crushers.

Abrasive materials still produce greater wear rates than non-abrasive materials. Because of ease of maintenance, this is probably not a vital consideration in judging the suitability of cage mills for a tunneling application.

6. Ability to Handle Plastic Fines: The performance of cage mills is not affected by the moisture content of feed or the presence of plastic fines.

7. Wear and Maintenance: Wearing surfaces include breaker plates (located around circumference of mill), liner plates (sides of mill housing) and cage bars. All interior parts and surfaces are made of work-hardening manganese steel. Breaker plates are slip-fit mounted on T-bars for ease of replacement (no bolts); liner plates are interchangeable, doubling wear life. Cage mills are reversible for maximum wear of cages and breaker plates. Breaker plates are made of work hardening alloy. No design provision is made for hard surfacing of worn parts. All worn parts are designed to be replaced - probably a plus for the design application in tunnels.

8. Feed Requirements: Surge feeding is not a specific problem, but uniform feeding is desirable for optimum performance. Cage mills cannot be choke fed, which may be a problem if very heavy surges are delivered to a mill of limited capacity.

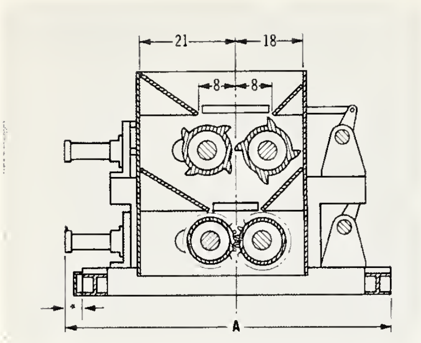
9. Foundation and Mobility: Foundation must be sufficiently massive to damp out vibrations. Cage mills should be level when operating. Because of low silhouette and ease of maintenance, unit would be readily portable.

10. Size and Capacity: (Averages based on models available from one manufacturer assuming heavy duty service and maximum product size of 2".)

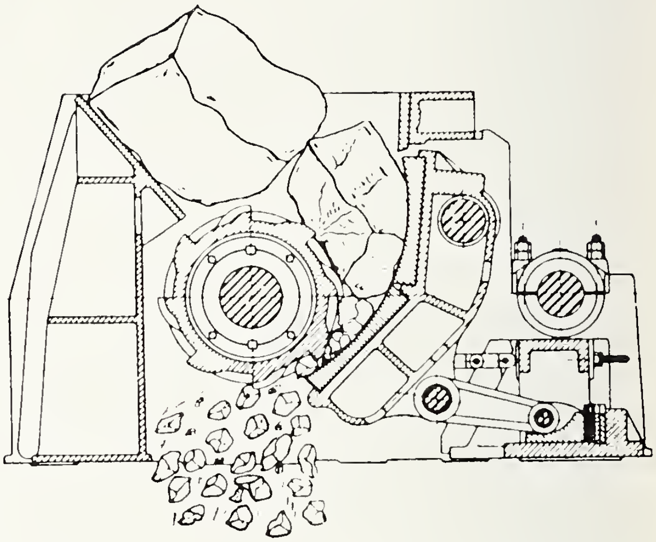
Capacity(tph)	100	300	600
Weight(lb.)	12,400	36,500	45,000
Horsepower	100	350	800
Length(ft.)	7-1/4	10	11
Width(ft.)	6-1/4	8-1/2	10-1/2
Height(ft.)	6	8	10

Roll Crushers

1. Nature of Crushing Action: Roll crushers (See Fig. 3-7) are available in several configurations, of which the single stage (double-roll) and two-stage (quad roll) types are probably most important. Material is fed through a pair of counter-rotating rolls set at a desired spacing for a given product size. Material is broken by a combination of compression, impact and shearing (attrition). In two-stage roll crushers, the feed from the first pair of rolls falls directly through a second pair, with coarser reduction occurring in the first stage. Because material is pulled between the rolls, gravity feed is not a critical consideration affecting capacity, and the height of fall does not affect the efficiency of the crushing action.
2. Crushing Ratio: A ratio of about 4:1 is the maximum reduction obtained in single stage roll crushers. A ratio of 6 to 7:1 may be obtained in two-stage units.
3. Particle Shape: Roll crushers will give a slabby product if the feed is slabby.
4. Rock Hardness Range: Very effective crushing rock of soft to medium hardness such as limestone, dolomite, and shale. Heavy duty models will crush hard rock and may be used in such applications if abrasion (related to rock silica content) is not too severe for adequate maintenance. Large, shrouded fly-wheels must be added for heavy-duty (hard rock) crushing service.



Two-Stage Quad Roll Crusher
(courtesy T.J. Gundlach
Machine Co.)



Single Roll Crusher
(courtesy Pennsylvania
Crusher Div., Bath
Iron Works Corp.)

FIG. 3- 7. ROLL CRUSHERS

5. <u>Resistance to Abrasive Materials</u>: Attrition plays a significant part in the crushing action of roll crushers, particularly under conditions of choke feeding or operation at high speeds. Rolls show excessive wear by developing grooves around the circumference. Grooving or bellying tends to occur if rolls are not properly fed, and may develop rapidly if hard material constitutes the feed.

6. <u>Ability to Handle Plastic Fines</u>: Reports are contradictory on this point. Passage of plastic fines is probably dependent to a degree on the type of roll surface in use. Spring loaded roll scrapers may be used to couter the buildup of moist fines on rolls.

7. Wear and Maintenance: Maintenance of rolls will vary with type of roll surface used. Hard surfacing may be added to worn surfaces of some rolls. Toothed rolls must be replaced in some models: individual teeth can be replaced on others.

8. Feed Requirements: Mechanical feeding is almost essential. Uniform feed distribution should be delivered to the crusher for even wear along the length of the roll. Crusher should be of sufficient capacity to minimize choke feeding to the rolls.

9. Foundation and Mobility: Compact design configurations require minimal head room for operation. Heavy supporting structures are not required.

10. Size and Capacity: (Averages based on models available from one manufacturer assuming heavy duty service and maximum product size of 2".)

Capacity(tph)	100	300	600*
Weight(lb.)	6000	11,500	-
Horsepower	40	75	-
Length(ft.)	6	6	-
Width(ft.)	6-3/4	9-1/4	-
Height(ft.)	4-1/4	4-1/2	-

*Available up to 500 tph

Jaw Crusher

1. Nature of Crushing Action: There are several types of jaw crusher designs, of which the Blake jaw (See Fig. 3-8) is used almost exclusively in high capacity hard rock applications. In this type, the moving (swing) jaw pivots on an overhead shaft, and the fixed jaw is the end of the crusher frame itself. As the rotating eccentric shaft turns, it alternately raises and lowers an arm (the pitman), which translates motion through a double-toggle linkage to the swing jaw.

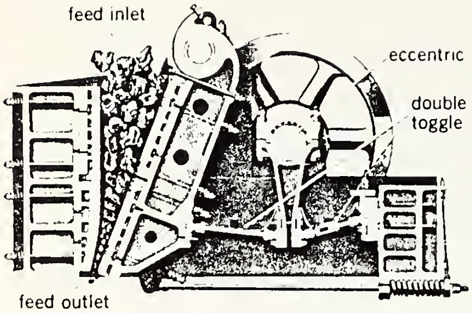


FIG. 3-8. Double-Toggle (Blake) Jaw Crusher (courtesy Allis Chalmers Co.)

Single-toggle jaws (See Fig. 3-9) differ from the Blake (double-toggle) type in that the swing jaw pivots on the eccentric shaft itself. Because of this feature, motion of the swing jaw exists at both the upper (feed) side and lower (discharge) ends imparting an elliptical path of travel to the swing jaw over one cycle. This type of action provides for higher ratios of reduction, but increases wear due to the shearing action introduced in crushing.

2. Crushing Ratio: Maximum crushing ratios obtained with Blake jaws in secondary crushing installations is about 10:1.
3. Particle Shape: Feeding rock with pronounced planar orientation (finely jointed, laminated, foliated) there is a pronounced tendency for a slabby feed to produce a slabby product, owing largely to the parallel orientation

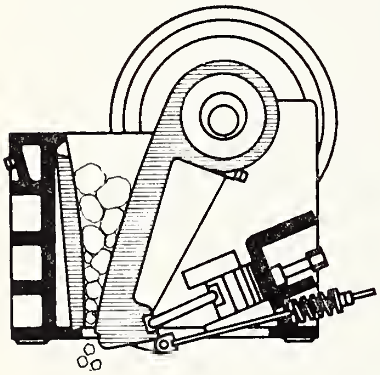


FIG. 3-9. Single-Toggle Jaw Crusher. (courtesy Pit and Quarry Magazine.)

of jaws and feed opening. This tendency can sometimes be reduced by feeding a portion of the scalped fines through the crusher along with the oversize.

4. Rock Hardness Range: Best suited for hard-rock crushing. Can crush rock up to 50,000 psi compressive strength.
5. Resistance to Abrasive Materials: Jaws are made of work hardening manganese steel. The swing jaw tends to wear unevenly (more at the discharge end, less at the feed end) and the problem is intensified if extremely abrasive material (greater than 40% silica) is crushed.
6. Ability to Handle Plastic Fines: Very poorly suited to crushing moist, soft, clayey plastic material. This type of feed packs inside the jaws, impeding material flow.
7. Wear and Maintenance: There are few wearing parts. Jaws may be hard surfaced, but are relatively inaccessable for major maintenance. If jaws are to be removed, a crane is generally necessary. The swing jaw is often reversible for extended wear life.
8. Feed Requirements: Feed must be scalped of fines ahead of the crusher. Mechanical feeders are generally used with jaw crushers. Slabby feed may require diversion of some scalped material through the crusher to improve product shape, but capacity is significantly reduced when this practice is followed.
9. Foundation and Mobility: Blake jaws are frequently used in underground installations, but are seldom found in portable plants so far as is known. This would suggest that while maintenance can be performed well enough in a tunnel environment the foundation requirements are possibly excessive for mobile systems. Jaws, because they are not internally counterbalanced, would transmit considerable vibrations to the foundation.

   Smaller single-toggle jaws are sometimes applied in portable plants, although bulky foundations may be necessary.

10. Size and Capacity: Because the crushing angle cannot be modified with other changes of scale, there is a noticeable increase in minimum discharge setting of jaw crushers as size and capacity are increased. The Blake jaws studied all give coarser products than are desired for the proposed application. The single-toggle jaws (Telsmith) can provide an acceptable sized product, but at very restricted capacities. (Averages based on models available from one manufacturer [Telesmith] assuming heavy duty service and maximum product size indicated.)

For 2" product top size:
(Largest available capacity is about 80 tph)
For 8" product top size:

Capacity(tph)	100	300	600
Weight(lb.)	26,600	35,500	105,800
Horsepower	100	125	200
Length(ft.)	7-1/2	9	12-1/4
Width(ft.)	7-1/4	7-3/4	9-1/2
Height(ft.)	7-3/4	8-1/2	13-1/2

Gyratory and Cone Types

1. Nature of Crushing Action: The gyratory crusher (See Figure 3-10) is basically a jaw crusher with the fixed and moving jaws formed into a conic geometry. The outer shell ("concave") is the fixed jaw. The crushing head is the moving jaw, which is eccentrically mounted on a vertical shaft. As the crushing head rotates within the outer shell, one edge is always approaching the shell (closed side) while the opposite edge is retreating from it (open side). A cone crusher (See Fig. 3-11) is a gyratory designed specifically for secondary product sizing. A long parallel aperture is formed along the discharge edge between the concave and the crushing head when the head is rotated to the closed side setting. This geometry provides close control of product top size.

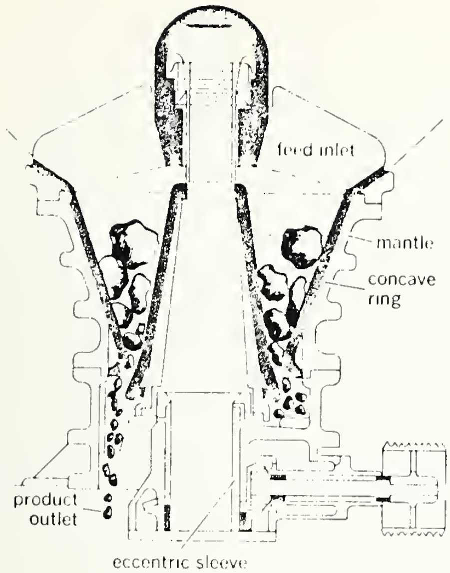
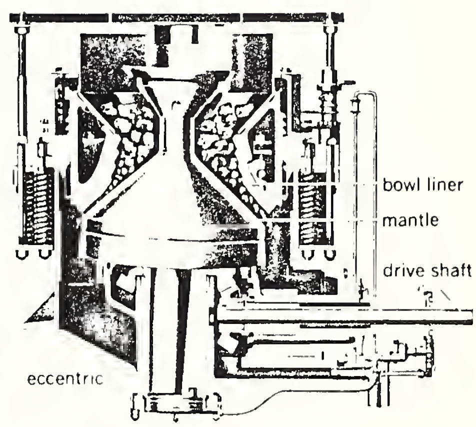


FIG. 3-10. Gyratory Crusher (courtesy Allis-Chalmers Co.)

FIG.3-11. Symons Cone Crusher (courtesy Nordberg Co.)



Cones are available with a variety of liners and head designs. A short head design is used for very fine crushing. A standard head and coarse liners would be preferred for the proposed application.

2. Crushing Ratio: 8 to 1 is about the maximum reduction possible in a cone unless material is very soft. Maximum feed size is about 12" (in a 5 ft. cone).
3. Particle Shape: Tends to give a cubic particle, even with hard slabby feed, because of geometry of crushing surfaces.

4. Rock Hardness Range: Can crush any type of rock, but will permit material to pass (such as tramp iron) which cannot be crushed.
5. Resistance to Abrasive Materials: Crushing action consists of compression with some attrition action as well. As a result, rock abrasive properties are a crucial consideration in selecting this type of crusher. It is at the same time true that cones and gyratories are used in the most rigorous crushing applications to be found. Concaves and mantles can be selected of various alloys to resist wear for a given work situation.
6. Ability to Handle Plastic Fines: Cones do not work well handling very moist or soft plastic material. Where hard rock is to be crushed containing some clayey fines, cones are somewhat better suited than jaws to the application.
7. Wear and Maintenance: Shell liners (concaves) and crushing head liners (mantles) are the only parts normally replaced as a consequence of wear. For the larger installations, it is necessary to have a crane to replace these parts. The installation must also be mounted with sufficient clearance to remove the eccentric shaft for major repairs (underneath).
8. Feed Requirements: Surge feeding is not a problem; choke feeding cannot occur in gyratories or cones due to the concave shape of the liners. Product removal from the lower side does not impose any special requirements.
9. Foundation and Mobility: Gyratories (even small secondary types) are heavy and require substantial foundations. Cones have a lower profile, and can be placed on smaller steel cage foundations. One reason for the difference appears to be that cones are better counterbalanced internally, so that the foundation does not have to handle as much vibration. Cones are frequently applied in portable surface plants.

10. <u>Size and Capacity</u>: (Averages based on models from several manufacturers assuming heavy duty service and a maximum product size of 2".)

Capacity(tph)	100	300	600
Weight(lb.)	22,000	91,600	148,500
Horsepower	60	200	300
Diameter(ft)	5-3/4	8-1/2	10-1/2
Height*(ft.)	7-1/2	12	14-3/4

*Excluding height required for feed and discharge.

3.3 DISCUSSION OF CRUSHER ALTERNATIVES

Having considered the operating features of basic crusher types available, it is necessary to compare their capabilities, considering the requirements of the tunneling application. Impactors and hammermills are similar in design as well as crushing action. Cage mills and roll crushers incorporate different means of rock breakage, but share several common application requirements and restrictions. Jaws and cones employ the same method of crushing and are designed to handle similar problems of size reduction.

Impactors and hammermills offer several distinct advantages over compression-type crushers (jaws, cones) in soft to moderately hard rock crushing applications.

1. Low capital cost.
2. Large throughput capacities.
3. Large ratios of reduction.
4. Single stage crushing capability.
5. Ability to handle wet and plastic material without clogging.
6. Compact size.

Impactors are variously used in single stage crushing operations, as well as two-stage configurations where the impactor may do either the primary or secondary crushing. Hammermills generally do the secondary crushing in two-stage plants, but they are also often used in single stage applications.

Impactors and hammermills are readily adaptable to applications crushing rock with compressive strength up to about 30,000 psi, provided that the silica content does not exceed about 15%. The crushing mechanism is not affected by the hardness or abrasion of the feed per se, but performance characteristics of these crushers in terms of wear become noncompetitive with jaws and cones when hard, abrasive rock is to be handled.

Product size is controlled primarily by varying the rotor speed of impact crushers. Gradation is also affected by breaker plate setting, the number of hammers (or rows of hammers) mounted on the rotor, and in hammermills, by the grate configuration. Because hard rock requires increased speed (and horsepower) for adequate product sizing in an open circuit, rotor speed and power are the actual limiting factors of product gradation. In general, impactors require less power (1.0 HP/tph) than hammermills (1.1 HP/tph), and therefore offer a modest savings in operating costs.

Hammermills are chosen when a small product top size is desired from an open circuit operation. Fine crushing is accomplished by the hammer tips abrading rock against the grate system in the lower area of the crushing chamber. Because attrition crushing produces accelerated wear, maintenance costs of hammermills are 50 to 75% higher than for impactors of equivalent capacity. For this reason, impactors are selected when product top size is not of critical concern.

Impactors are inherently well suited to crushing run-of-mine rock containing significant amounts of plastic fines. Hammermills may be modified to incorporate a traveling breaker plate to eliminate the accumulation of caking fines, but with this feature the capital cost of a hammermill is roughly doubled.

Impactors and hammermills are sometimes used to crush hard, abrasive rock when the requirement for a portable crushing plant with high capacity is sufficient to justify the high maintenance costs. One example is known where a portable impactor is used to crush river gravel for aggregate stone. In this installation, abrasion is so severe that hammers must be resurfaced every 4 hours. In another "high-wear" application, where a large hammer mill is used to crush automobile engine blocks, hammers are rotated from the center of the rotor toward the outside until they are worn so severely that they are discarded.

One particularly interesting impactor is a design offered by Hazemag, a German manufacturer. This unit is a compound crusher in which two rotors are positioned in sequence, with the second rotor working at higher rpm than the first. Increased size reduction and finer product gradation can be achieved with this design than are normally obtainable. The Hazemag line of impactor crushers is notable for extremely robust construction and for providing extra heavy-duty operating service.

Head room is sometimes a consideration in choosing a hammermill over an impactor. For impactors, the feed must achieve a sufficient fall velocity to enter the hammer impact zone to the proper depth for effective crushing to take place. Impactors have higher physical profiles as a consequence. In some models, the housing is built up to assure that the feed will enter the chamber at the correct drop height. In other models, the feed must be dropped into the receiving opening from an elevated position.

The cage mill is another type of impact crusher, and is similar in some respects to the impactor and hammermill. Cage mills are about the same size as hammermills for a given capacity. They provide the same operating costs and benefits as impactors. Maintenance of cage mills is much simpler to perform than for the other types of impact crushers.

Although these advantages are considerable, cage mills are not, in general, suitable for the proposed application. Cage mills are not designed for single stage crushing. The largest feed size acceptable is about 10 to 12 in. Cage mills are utilized for secondary crushing in closed circuit to produce a uniform product with a small (-1/2") top size.

In open circuit crushing, the top size of the product gradation is difficult to control. For a feed with maximum size of 12 in., the product top size would be about 5 in. from a single-cage unit, which is not acceptable for the proposed application. To some degree, the product top size may be reduced by installation of a multi-cage unit but the throughput capacities of multicage mills are less than 40-50% of single cage units of equal size and power consumption.

Roll crushers work by a combination of compression and attrition. Like cage mills, rolls offer notable practical advantages in crushing certain types of material but are not broadly applicable for tunnel-muck preparation for several reasons. Roll crushers are attractive because of:

1. Large throughput capacities.
2. Low capital cost.
3. Low-profile dimensions.
4. Good control of product size in open circuit.

Rolls probably require the least capital investment per unit capacity of all crusher types manufactured. However, they suffer the following disadvantages:

1. Limited ratio of reduction.
2. Limited range of rock hardnesses crushed efficiently.
3. High maintenance costs.
4. Poor control of product shape.

Like cage mills, roll crushers are not readily adaptable for single stage crushing. A few large, toothed rolls, called sledging rolls, are used as primary crushers of coal, shale and limestone. Stamler produces a very rugged "feeder/breaker" roll crusher which has been used to crush hard limestone and copper ore. This unit can reduce 4 ft. rocks to a size of minus 10 in. The following comparison of operating costs has been made:

Crusher Type	Material Crushed	Operating Cost/Ton
Jaw	Medium limestone	Less than 1¢
	Hard sandstone	8¢
	Scoria	1-3/4¢
Impactor	Medium limestone	1¢
	Hard sandstone	10-15¢
Stamler feeder-breaker	Hard sandstone	13-15¢
	Coal	Less than 1¢

The majority of roll crushers are designed to perform secondary size reduction of materials of soft to medium hardness. The most rugged roll crushers currently available can only handle material with a maximum compressive strength of about 28,000 psi.

High maintenance costs incurred in operating roll crushers stem from wear of roll surfaces. Secondary crushing rolls are poorly suited for crushing abrasive materials in view of high roll wear rates. Crushing rolls must be hard surfaced regularly, and maintenance time is greater for the larger roll diameters used in high-capacity units. Pickpoints must be replaced regularly on the larger sledging rolls.

Rolls do not generally offer the flexibility required from a crusher in the proposed application. In large tunnels in soft rock, where large muck volumes (rates) may require two-stage crushing, a roll crusher could be considered for primary or secondary size reduction.

In industrial applications, gyratory and jaw crushers are almost always selected where efficient crushing of hard, abrasive rock is required. In multi-stage crushing plants, the primary breaker is usually a jaw or gyratory. The secondary crusher may be a jaw or a cone.

These machines crush rock by compression, and are extremely massive and rugged in order to tolerate the very high loads which are generated in crushing cycles.

The following advantages may be cited favoring selection of cones and jaws for the proposed application:

1. Ability to crush very hard and abrasive rock, as well as softer materials.
2. Rugged design provides reliable service in operation.

3. Very low maintenance costs compared with other types of crushers.

Because compression crushers are large in physical size relative to their throughput capacities, it is difficult to obtain large capacities from machines which are at the same time small enough to fit inside a tunnel. At the same time it is difficult to select a cone or jaw which can handle large capacities while producing a small (-1 1/2") product size.

While it is possible to adjust cone and jaw crushers so that the closed setting is smaller than the rated discharge setting, the results of this practice are undesirable. Too high a ratio of reduction leads to localized increased wear around the discharge opening, leading to an overall decrease in mechanical efficiency of the unit. At the same time power consumption is disproportionately higher. Both conditions contribute to increased down time and operating costs.

The disadvantages of selecting a cone or jaw crusher for the proposed application can be summarized in the following statements:

1. High capital cost relative to other types of crushing equipment.
2. High profile requirement - large dimensions.
3. Require heavy foundations.
4. Replacement of worn parts may require an overhead crane for hoisting.
5. Small capacity.

Capital costs of cones and jaws are not competitive with rolls and impact crushers of similar capacity. Some of the additional expense can be recovered over a period of operation in the form of reduced maintenance costs. However, when required, maintenance of jaws and cones would be more difficult to perform inside a tunnel.

Cones are somewhat better suited to installation in single-stage portable crushing plants than jaws, due to the shape of the cone and its reduced foundation requirements. An assortment of liner shapes is available for achieving a desired product gradation. Liners may be selected of different alloys to achieve the best possible balance between wear and replacement costs. In general, standard head cones with coarse-crushing liners would appear to be best suited for the proposed application.

Whereas a mechanical feeder must be used with a jaw in order to minimize the effects of surge volumes of feed, cones operate most efficiently when the receiving opening is kept partly full. As a result, cones are desirable in crushing plants which are normally subjected to surge conditions. Proper distribution of the feed around the top of the crushing chamber is essential for best performance, which is enhanced by a surcharge volume in the receiving opening.

The rated capacity of the crusher selected for the muck preparation unit should exceed the expected throughput by a factor of 50 to 75% to accomodate surge input to the system, and to avoid "overworking" the crushing unit in general. A surge factor of 25% has been considered in system design. The effects of surge feeding on crusher performance are difficult to assess without conducting specific tests of various models. The performance of continuous transport systems may be adversely affected in several respects by muck input to the system in the form of large surges. In some cases in soft ground, it may even be necessary for the mole operator to adjust the rate of advance to avoid excessive muck surges.

A comparison of maximum horsepower requirements for various crusher types at different throughputs shows that the variation of power with capacity is not equivalent for different designs. Compression-type crushers (jaws, cones,rolls) are less consumptive of power than impact-type crushers (impactors,hammermills, cage mills) in achieving a given capacity of an equivalent product size. Of all crusher designs, rolls are generally the most efficient. Impact-types require roughly twice as much horsepower per tph as compression-type crushers, comparing units of equal capacity.

A provision for passing pieces of tramp steel is made in the design of most crushers. In impactors and hammermills, uncrushable material is swept into a trap where it remains until it is removed during maintenance. Cage mills do not have this feature.

One roll of a roll crusher is usually spring-loaded. Iron or uncrushable material which is small enough to be drawn into the crushing zone is passed by forcing the spring-loaded roll to spread. A similar design is incorporated in cone crushers, in which the entire shell is spring-mounted.

Several different designs are used in jaws to minimize damage from tramp iron. Safety relief plates are present in some models which buckle when uncrushable material is encountered, permitting the swing jaw to yield and material to pass.

3.4 CRUSHER SELECTION

The average size of rock fragments in muck produced by tunnel boring machines is small (minus 1/2 in. to minus 20 mesh). Probably no more than 1 to 3% of the muck will be large enough to be difficult to feed to the crusher. Impactors and hammermills are capable of accepting large (plus 20 in.) material. Other types of crushers require smaller (10 to 15 in.) feed top sizes. In certain cases, where an excess volume of large fragments is generated at the face, some alternate means of crushing or transport may be necessary.

Table 3-1 shows the distribution of muck sizes for each of the MDN categories discussed in Chapter 2. From a description of muck in terms of the appropriate MDN, the percentage of the total muck volume requiring size reduction may be estimated.

TABLE 3-1

Description of Muck Size Distributions by MDN

	MDN-3	MDN-4	MDN-5	MDN-6	MDN-7
-6"	91	100	-	-	-
-3"	87	97	100	100	-
-2"	69	84	87	94	100
-1"	56	67	66	85	93
-1/2"	48	44	50	49	80

For muck scalped of all minus 1 in. material ahead of the crusher, the percentage to be crushed is approximately

	MDN-3	MDN-4	MDN-5	MDN-6	MDN-7
+1"	44%	33%	34%	15%	7%

Feeding the oversize (plus 1 in.) material to the crusher, different size distributions of muck are produced depending on the type of crusher in service, operating specifications of the crusher, and rock hardness. Table 3-2 shows typical product gradations produced by different types of crushers at specified operating conditions.

From Table 3-2, several additional remarks may be made concerning crusher performance. Impactors produce a wide range of sizes, with a heavy proportion of fines (minus 1/4 in.). Hammermills are designed to perform fine crushing, while providing close control of product top size. Cones and jaws produce a uniform "pea gravel" size product without excessive fines. Cones and jaws can be set to give almost identical product size distributions. Cage mills do not perform well when operated in open circuit. They provide about the same product as a jaw, but require much finer feed to achieve a similar product gradation.

From the foregoing discussion, several tables have been devised to assist in crusher selection for the muck preparation unit. No single crusher can be recommended for general service at all tunneling sites. Tables 3-3, 3-4 and 3-5, were drawn up considering various requirements imposed by rock hardness, throughput capacities, and tunnel diameters.

TABLE 3-2.

Product Gradation for Nominal 1" Setting

Open Circuit Operation

	Jaw[1]	Standard Head Cone[2]	Impactor[3]	Hammer-mill[4]	Single Cage Mill[5]
	(Percent Passing)				
-2"		100	100		
-1-3/4		97	98		
-1-1/2		91	94		
-1-1/4	100	79	87		
-1	85	60	79	100	70
-7/8	78	49	74	-	-
-3/4	70	40	69	97	60
-5/8	60	30	63	92	-
-1/2	50	22	57	86	40
-3/8	40	16	50	71	30
-1/4	29	11	40	57	25

1. For 1" setting.
2. Coarse crushing liner; 1" closed side setting.
3. at 600-850 rpm.
4. at 800-1200 rpm.
5. For 6" feed; unspecified rpm.

TABLE 3-3.

CRUSHER THROUGHPUT CAPACITY (TPH)*

	0–200	200–400	400–600
Soft	Hammermill Impactor Roll-1	Hammermill Impactor-5 Roll/Roll	Roll/Roll Roll/Hammer-mill Roll/Impactor-5
Medium	Impactor Hammermill Roll-1,3	Hammermill Impactor-5 Roll/Roll-3	Roll/Hammer-mill Roll/Roll-3 Roll/Impactor-5
Hard	Cone-1,2 Jaw-1,2 Jaw/Cone Impactor-3	Jaw/Cone Impactor-3,5	
Very Hard	Cone-2 Jaw-2 Jaw/Cone Impactor-4	Jaw/Cone Impactor-4,5	

*85-90% Crushed to minus 1-1/2 in.

12 - FT. TUNNEL BORE

1. Marginal reduction ratio.
2. Few commercial units available.
3. High maintenance.
4. Excessive maintenance.
5. Head room limitations.

TABLE 3-4

CRUSHER THROUGHPUT CAPACITY (TPH)*

	0–200	200–400	400–600	600–800
Soft	Impactor Hammermill Roll-1	Impactor Hammermill Roll-1	Impactor Hammermill Roll-1	Roll/Roll Impactor/Impactor Roll/Impactor
Medium	Impactor Hammermill Roll-1,3	Impactor Hammermill Roll-1,3		
Hard	Cone-1 Impactor-3	Cone-1 Impactor-3		
Very Hard	Cone-1 Impactor-4 Jaw-1,2			

*85-90% Crushed to Minus 1-1/2 in.

20 - FT. TUNNEL BORE

1. Marginal reduction ratio.
2. Few commercial units available.
3. High maintenance.
4. Excessive maintenance.

Table 3-5

CRUSHER THROUGHPUT CAPACITY (TPH)*

	0–200	200–400	400–600	600–800	800–1000
Soft	Impactor Hammer-mill Roll-1	Impactor Hammermill Roll-1	Impactor Hammermill Roll-1	Impactor Hammermill-2	Impactor Roll/Roll Impactor/Impactor Roll/Impactor
Medium	Impactor Hammer-mill Roll-1,3	Impactor Hammermill Roll-1,3	Impactor Hammermill Roll-1,3		
Hard	Cone-1 Impactor-3 Jaw-1,2	Cone-1 Impactor-3			
Very Hard	Cone-1 Impactor-4 Jaw-1,2	Jaw/Cone Cone-1 Impactor-4			

*85-90% crushed to minus 1-1/2 in.

35 - FT. TUNNEL BORE

1. Marginal reduction ratio.
2. Few commercial units available.
3. High maintenance.
4. Excessive maintenance.

Because open circuit operation is envisioned in all cases, cage mills were not evaluated in the tables, as they provide reliable performance in closed circuit operation only. Indicated throughput capacities are volumes fed to the crusher (not volumes excavated). Ordinarily, muck would be scalped of undersize material (minus 1 in. to 1-1/2 in.) ahead of the crusher, so that the crusher would receive only a portion of the total muck volume. In some cases (as noted), single-stage crushing does not appear to be feasible with commercial units presently available. In these situations two-stage crushing is indicated, with the first unit noted performing primary size reduction and the second unit doing the finer crushing.

Two-stage crushing should be considered for tunneling in hard or very hard rock, and in cases where high muck volumes are generated in soft rock. Because hard-rock crushers (cones and jaws) are bulky units, only low-capacity models can be installed in a small tunnel bore. The top size of hard rock muck tends to be small (less than 3 in.), but the occasional blocks of material which slough from the face without being subjected to muck grinding could be difficult to process in jaws or cones because of their limited ratios of reduction. Two-stage crushing would increase both effective capacity and reduction ratio.

For high-volume, soft-rock applications, two-stage crushing provides not only increased capacity but also an opportunity for reduced operating costs attendant with the use of roll crushers in either the primary or secondary stages. Roll crushers cost less to install than most other types of crushers and require less power in operation then impactors or hammermills.

In general, single-stage crushing is preferable whereever possible. Two crushers cost at least twice as much as one crusher to purchase and install. Two-stage crushing also requires additional screening units ahead and behind the primary unit in order to achieve the total throughput capacities from small primary and secondary crushers. The additional screens and belts add to the cost of the installation and would increase the complexity of the overall design of the muck preparation unit.

Some consideration has been paid to various methods which could be employed to reduce the impact of surge feeding on the transport system. A reduction of the feed size during a period of surge would increase the effective capacity of the system, while conserving power. A mechanical screen bank could be designed for the preparation unit which would insert a smaller screen cloth during surges. More muck would be diverted to the crusher during surge periods. Most crushers, in turn, could be regulated in a feedback loop to crush coarser or finer in response to the system demand. Impactors and hammermills with variable speed drive units could produce a finer product simply by increasing rpm. The crusher spacing on rolls and jaws is frequently adjusted with hydraulic rams and shims. The hydraulics of these units could be tied into the feedback system. Several cone crushers are also available with hydraulic setting controls, and may be similarly adaptable to feedback system control.

4. EXTENSIBLE CONVEYOR SYSTEMS

4.1 INTRODUCTION

Muck from the tunneling machine, after being reduced in size in the preparation unit, is ready for transport to the disposal area. Anticipated muck volumes were established in Chapter 2 - System Input, together with information on rock hardness and abrasiveness. The expected size distribution of muck leaving the preparation unit was defined in Chapter 3. These considerations provide the basic information for selection of the muck transport system.

Ideally, the transport system should be continuous in nature in order to complement the capabilities inherent in tunneling machines. Because of its efficiency, a continuous transport system would provide options of handling higher volumes and/or compaction of the system design.

Elaborate moving platform and switching techniques have been developed for sequence loading of rail cars. These have reduced the delays inherent in this haulage system, and provided a means for advancing the loading station behind the tunneling machine. The effectiveness of these approaches through the years had been progressively improving, but at the expense of increased complexity and cost. The rail systems common to tunneling can, by their very nature, never achieve the ideal of continuous haulage.

The slurry pipeline offers the potential for long distance, high volume, low cost muck transport, if a practical means can be provided to continuously extend the system to match the advance of the tunneling machine. While a number of schemes have been attempted for direct operational extension of a slurry pipeline, these have proven to be excessively complex and impractical. Successful application of the pipeline appears to depend on finding a simple reliable means for providing a short intermediate extensible link between the preparation unit and the pipeline installation.

Several basic considerations must be recognized at the onset. The conceptual difference between what is termed an intermittent and a continuous transport system is essentially one of degree. Continuous transport for every schedule hour of tunnel construction is probably impossible. Equipment maintenance requirements and repairs generally classified as mechanical availibility significantly reduce actual transport activities. The tunneling machine, while generally considered to be continuous in operation, is intermittent in so far as the jacks used for advancing the unit must be periodically reset.

By its very nature, a continuous system does not provide for transport of any materials to the face, except possibly with very special designs. Any secondary material to be handled would have to be similar in nature to the muck. Continuous operation does not allow time for the loading of irregular objects which must be individually placed. This means that personnel, maintenance and operating supplies, and lining material must be transported by an auxiliary system. Continuous systems tend to be relatively compact in their cross-sectional configuration, so that space would exist in most tunnels for a dual system. The cost of the auxiliary system, because of its light duty requirement, would be substantially less than a similar primary system capable of transporting muck, but is still an important factor in considering overall systems economics.

With a dual transport system, questions arise concerning what portion of the tunnel should be assigned to each. The most complex and space consuming interval of the muck transport system is the forward portion, immediately behind the tunneling machine, where the preparation unit and the extensible section must be installed. This also is the area where unloading of supplies, etc., is preferred for the auxiliary transport system. There appears to be no single answer to this problem, since it depends on the requirements for the specific project.

It is worthy of note, however, that the use of a monorail for one of the systems has been shown to be an effective approach to space utilization.

The "extensible link" must be a muck transport unit which can be continuously extended over a significant period of time to some maximum length at which point haulage is interrupted, the unit retracted and the main haulage system advanced proportionately. The operational characteristics of the extensible portion must be matched to the main haulage system, and the delays for retraction and extension of each must be compatable for optimum performance. The ultimate goal is minimum interruption of continuous muck transport. While an extensible link is applicable in concept to any haulage system, its application in conjunction with a slurry pipeline has some unique requirements.

Conveyor systems would appear to offer a low cost, proven approach to providing the extensible link for a slurry pipeline. They are employed as the basic means for removing the muck from the cutting face to the rear of the tunnel boring machine for disposal. Similarly, they have been utilized in unit lengths of from 150 to 450 feet, gantry mounted on rail or wheels, in a variety of rail car loading configurations. Conveyors have been used extensively in underground coal mining operations.

Before investigating the conveyor options available, it is appropriate to define the system application criteria.

1. Capable of handling the muck volumes with the size characteristics outlined under "System Input", as modified by the preparation unit.
   Volumes: up to 900 tph
   Sizes: 100%-3", 95%-1"
2. Capable of handling dry or wet rock ranging from soft to very hard, and slightly to very abrasive.

3. Capable of continuous operation while being extended (front end dragged forward by the tunneling machine) providing for at least 20 hours of uninterrupted operation.
4. Capable of being readily retracted to its compressed length under its own power and in a time span less than that required to extend the slurry pipeline.
5. Minimum compressed length.
6. Capable of operating with variations in vertical and/or horizontal alignment of ± 10%.
7. Fit within the tunnel with sufficient additional space available for transport of personnel, service and maintenance supplies.
8. Minimal manpower for operation.
9. Maximum safety for personnel adjacent to unit with an effective emergency shut-down capability.

This can be summarized briefly in terms of general performance requirements:

Continuous
Extensible
Bulk handling (muck)
Horizontal or slightly inclined operation
Straight line-occasional large radius curves
Compact
Reliable

Omitting for a moment the consideration of "extensibility", the available conveyor designs can be reviewed to identify those best suited to this application. Tables 4-1, 4-2, and 4-3 provide a general overview of the important characteristics of current systems designed for bulk handling of relatively coarse materials. A great deal of literature is available describing each of those configurations and specific applications. Based on the application being considered, it appears that chain (flight or drag) and troughed belt conveyors should be given primary consideration. Chain conveyors of the apron, pan, plate, etc. types would not function well

TABLE. 4-1. CONVEYORS - GENERAL CHARACTERISTICS

General Type	Bulk Mtls Handling	High Capacity	Horizontal or Slightly Inclined*	Straight Line	Compactness
Chain:					
Apron & Pan	G	G	E	E	E
Plate & Slat	U	G	E	E	E
Drag Link	E	G	E	E	E
Flight & Scraper	E	G	E	E	E
Belt:					
Flat	G	G	E	E	E
Troughed	E	E	E	E	E
Vibratory:					
Vibrating / Oscillating	G	G	G	E	G
Screw	G	P	E	E	E

* + or - 10^{o} Maximum

E=Excellent
G=Good
P=Poor
U=Unsuitable

TABLE 4-2. CONVEYORS - ADAPTABILITY TO HANDLING MUCK

General Type	Muck Size Distribution				Flowability (Material Type)			
	Very Fine	Pebbles (¼" to 2")	Gravel (2" to 4")	Large Cobble (6" to 12")	Very Free	Free	Sluggish	Abrasive Resistance
Chain:								
Apron & Pan	P	G	E	G	P	E	G	E
Plate & Slat	U	U	P	P	U	U	U	E
Drag Link	G	E	E	P	P	E	G	E
Flight & Scraper	P	G	E	E		E	G	
Belt								
Flat	U	G	G	U	P	U	G	G
Troughed	G	E	E	E	E	E	G	G
Vibrating								
Oscillating	P	G	E	E	P	E	P	E
Screw	G	G	E	P	P	E	P	P

E=Excellent
G=Good
P=Poor
U=Unsuitable

TABLE 4-3. CONVEYORS - COMPARATIVE PERFORMANCE

General Type	Horizontal Carry: Up to 100'	Horizontal Carry: up to 250'	Horizontal Carry: over 250'	Relative Initial Cost	Relative Operating Cost	Past Applications to Muck Handling
Chain:						
Apron & Pan	E	G	P	G	G	Feeder systems only
Drag Link	E	G	P	G	G	Common
Flight & Scraper	E	G	P	G	G	Occasionally
Belt:						
Flat	E	E	E	E	E	Infrequent
Troughed	E	E	E	E	E	Common
Vibrating						
Oscillating	E	G	G	G	E	Feeder systems only
Screw	E	G	P	P	P	None

E=Excellent (Low)
G=Good (Medium)
P=Poor (High)
U=Unsuitable

with the high percentage of "fines" anticipated. The flat belt suffers from carrying capacity in comparison with the troughed belt. The vibratory and oscillating conveyors cannot handle sticky materials, cannot transport up a gradient and are normally applied for only short conveying distances.

The above conclusions must next be considered with respect to the special requirement of extensibility. Information on high capacity continuous haulage systems which can be extended while in operation is extremely limited. Special ship loading and "in plant" situations have been customized. Elaborate systems have been developed for handling overburden and coal in overseas surface mines. Few have a product to be transported similar to tunnel muck, and the necessity for compactness to fit the tunnel environment.

The literature (4) contains one specific reference to a tunneling application of extensible conveyor systems. In 1972 and 1973, a tunnel boring machine (TBM) was used in construction of one section of the Eastern Suburbs Tunnels, an extension of the rapid transit system serving the metropolitan area of Sydney, Austrailia. The project consisted of two sections of twin tunnels: one section was 2,640 ft. long and one section was 5,280 ft. long. The tunnels were driven through a formation known as Sydney sandstone. Rolling kerf cutters with tungsten carbide inserts, used on the TBM cutter-head, resulted in uniformly sized cuttings with a maximum ranging up to 6 in. Due to the known characteristics of muck size, the contractor chose a conveyor system to remove the muck from the TBM to the tunnel portal. Access from the surface to the tunnel portal was by inclined ramp. The muck was loaded into trucks at the portal for removal from the site.

Sufficient conveyor section modules were provided to extend a total of 5,280 ft. Each module consisted of a conveyor 3 ft. wide by 98 ft. long on a traveling gantry mounted on rubber tired wheels, four wheels under the support at each end of the gantry. The conveyor was driven by electric motors powered from the tunnel electrical power supply. The conveyor behind the TBM acted as a telescoping section, moving forward with the TBM and discharging material onto the first conveyor module positioned behind the TBM. Additional sections were placed in the system with each 98 ft. advance of the tunnel. The conveyor modules were assembled at the portal area and trammed into position in the tunnel by a crawler mounted front-end loader. The wheel assemblies at each end of the module were independently steerable to facilitate transportation and positioning in the tunnel.

The closest parallel is found in U.S. underground coal mines where extensible conveying systems are employed to interconnect continuous coal miners with permanent belt conveyor systems. The following is a brief discussion of some of the applicable systems known to have been applied in coal mining (5). Basically they fall into two categories: systems that provide extensibility by overlapping or cascading a series of conveyor unit assemblies, and those that utilize a reel to provide additional length within a storage unit.

4.2 PRIOR EXPERIENCE-EXTENSIBLE CONVEYORS

Beam Stage Loaders

The stage loader is the simplest form of the cascading conveyor approach used underground. It was developed in the early 1950's and provides a simple conveyor unit link between the end of a longwall face conveyor and the main conveyor system. In its most common form the skid-mounted tail piece is attached to the face conveyor and advances with that unit. The conveyor is mounted on a structural beam which is supported at the discharge end with dollies riding on a track on a special rigid frame tail piece section of the main conveyer. A 30 to 45 ft. in-line overlap between the two conveyors provides the extension. The dolly supports the conveyor drive head and chute for smooth transfer of the discharge. Most commonly the conveyer is of a chain type. (See Figure 4-1).

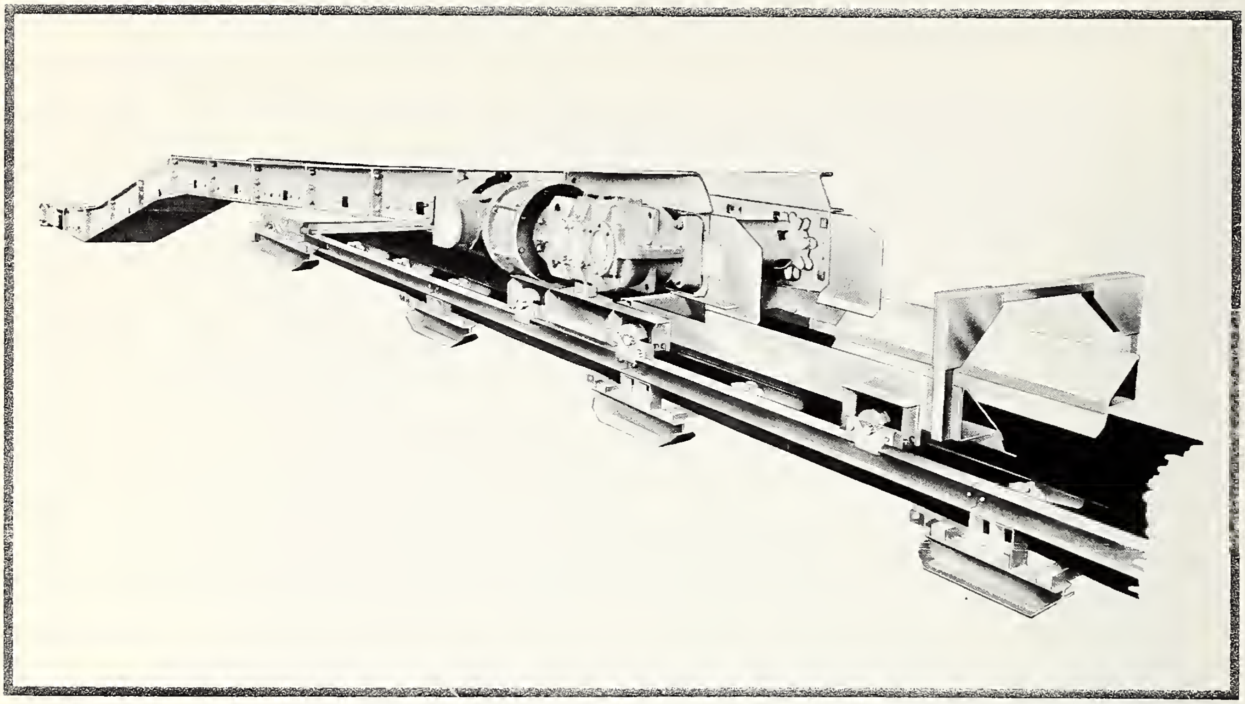


FIG. 4-1 Stage Loader
(Courtesy of Dowby Meco Limited)

There were no provisions in this basic design for mechanically connecting these conveyor units in a train. Two systems, each using about 15 conveyors, were built and saw limited underground use. None of these units is operating at the present time. The field operational problems included high labor required for manhandling units, storage of the unused units, reliability because of the multiplicity of units, and spillage and dust at the transfer points.

Portable Conveyer

The early portable conveyor systems comprised a series of narrow, lightweight belt conveyors that were designed to be manhandled into place behind the miner as it advanced. Aluminum frame elements were mounted on a pair of rubber-tired wheels and were balanced so that they stood in place with the inby end of the conveyor on the ground and the outby end elevated. This allowed the coal flow to cascade from one unit to the next as it was carried back to the permanent belt conveyor.

The unpowered wheels on each portable conveyor were normally positioned for longitudinal movement of the unit, but they could be unlocked from this position and swung through 90° and locked to permit moving the unit sideways into the conveyor line.

The proportions of the portable conveyors were designed to allow nesting of the units, so that they could be telescoped to produce an overlap of 8 to 12 feet, depending on the model. A movable hopper was fitted to each conveyor which could be adjusted so that coal coming from the next inby unit would fall onto the hopper regardless of the overlap between units.

Each conveyor belt unit had its own electric drive, and a long permissible cable connection allowed them to be interconnected electrically as they were added to the train.

Following are some general specifications:

Overall length	19 ft.
Maximum width	4 ft.
Height (depending on wheel size)	34 to 48 in.
Belt width	18 in.
Belt speed	360 fpm
Capacity	60 to 90 tph

Articulate Self-Tracking Conveyor

Consolidation Coal Company, in the 1950's, built a self-tracking conveying system. This machine was assembled and operated for a period of time in one of their mines, but development and use of the machine were eventually discontinued.

The complete conveying apparatus consisted of a receiving section, a number of intermediate sections, and a discharge section, each with an independent conveyor belt, interconnected to form a train of cascading conveyors that were pulled behind a continuous miner.

The train was advanced by a cable winch that was fastened to the rear of the miner. This gave the miner freedom to make small moves without having to move the conveyor train. For moving the conveyor in an outby direction, a winch was used part of the time, and a loading machine was used on other occasions.

All units of the system were mounted on two wheels and were interconnected by a ball-joint connection that provided the flexibility required for the train to turn a corner and to adapt to uneven bottom conditions. None of the wheels was equipped with steering means, but since the axle on each unit was located midway between the connecting pivot pins at each end, the train units were supposed to track each other when they were pulled forward by the miner or backward by the shuttle car.

The prime mover for powering all of the conveyors on the various units was an electric motor mounted on the discharge section. This connected into a longitudinal drive shaft, interconnected between sections with double universal joints Gear belt drives from these shafts to right angle gear boxes powered the conveyor belt drive pulleys at the inby end of each conveyor.

Mineveyor

Hewitt-Robins' "Mineveyor" was designed in 1959. Exact information on how many were built is not available, but two are known to be operating at the present time. (See Fig. 4-2).

The Mineveyor was a mobile bridge conveyor that was conceived as a belt conveyor device to provide continuous haul-

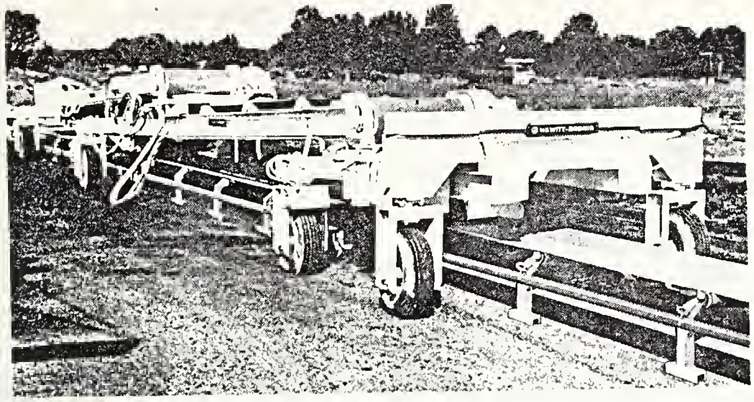


FIG. 4-2 Mineveyor (courtesy Hewitt-Robins)

age with maximum flexibility, and toward this end it was designed so that no mechanical connection would be required between either the miner and the bridge system or between the bridge system and the main belt.

The mobile bridge conveyor consisted of a series of self-powered two-wheeled bridge conveyors (usually three to five) and a self-powered four-wheeled discharge unit. All propel and steering functions on these units were hydraulic powered. The inby ends of the bridge conveyors were supported on two wheels, and the outby ends rested on a carriage on the next outby unit. The carriages on the bridges rode on small wheels, which allowed one bridge to telescope over the next outby bridge a distance of about 15 feet.

The connections between bridges, and between the outby bridge and the discharge unit, were designed to allow steering articulation 90° each side of center, as well as to provide joint flexibility in other planes so that the train could adapt to rough floor conditions.

The four-wheeled discharge cart was designed to straddle a low extensible main belt that was specially designed for this application. In operation, the discharge cart always remained above the main belt. The two-wheeled bridges were designed so that the entire train could be pulled back to a position where it was parallel to and above the main belt, with all wheels of the train straddling it.

The hydraulic power pack for the complete unit was located on the discharge cart. Duplicate sets of propel and steering controls were arranged on each side of this cart, and the controls allowed each set of wheels to be steered or propelled independently, or collectively. When the wheels were all propelled together, the bridge units moved in unison. When it was desired to telescope the bridge sections, the wheel units were powered individually or in combination to give the desired telescopic effect. By turning one or more pairs of wheels through 90°, a part of the bridge could be moved laterally.

In addition to the propel and steer controls of the discharge cart, other sets of controls for propel and steer were located at the inby end of the train. In practice, two men were needed to propel and steer the unit, one inby and one outby. The man on the discharge cart also controlled a hydraulically operated chute that directed the coal onto the main belt.

Following are some general specifications:

Bridge belt width	36 in.
Length of each bridge	32 ft.
Overall height	4 ft. 9 in.
Loading height at tail	3 ft.
Width overall	8 ft. 3 in.
Width of frame	4 ft. 6 in.
Conveying capacity	450 tph
Propel speed	0 to 50 fpm
Horsepower (total for unit)	40 hp

The Mineveyor apparently performed well. It is understood that there is a possibility for further refinement and development of this design.

Push Button Miner Train

The Push Button Miner, designed by the Joy Manufacturing Company in 1960, incorporated a 1000 foot conveyor train.

This machine was operated for a period of time in an Ohio strip mine.

In operation, the conveyor train followed a multi-head boring type miner into an opening in the strip mine highwall. Coal coming out on the conveyor was discharged to a transfer conveyor, which was part of a large, mobile structure that was used to store and launch the conveyor train.

The conveyor train consisted of a series of sixty 2-wheeled cars, which were coupled at intervals of 16.5 feet. Each car mounted a 64 inch wide, double-chain cross-flight conveyor that was 18.5 feet long and was side-boarded to a width of 90 inches. Each conveyor was driven by a 7.5 hp motor-reducer unit at a conveyor chain speed of 105 fpm.

The conveyor unit of each car was pivotally mounted at the inby end of the car, in such a way that the outby end was raised so that the discharge could cascade onto the next outby unit. In addition, the conveyor on each car was designed so that the outby end could be lifted from a latched, inline position on the car element and swung to the side to allow discharge to the transfer conveyor. With this arrangement, the conveyor units beyond the discharge point were always empty, which facilitated their storage on a helical ramp which was part of the launching structure.

In operation the train was coupled at the inby end to the miner, and coal was discharged to it directly from the miner. Initially, alternate conveyor cars were powered for traction with a 4 hp drive synchronized to the speed of the miner. Traction problems with this arrangement were experienced on the storage ramp.

The conveyor system on the Push Button Miner had a conservative capacity of 350 tph. Overall height was 34 inches. The complex design experienced excessive maintenance and serious dust and spillage problems due, in part, to the unmanned conveyor operation in the tunnel.

Moleveyor

"Moleveyor" was developed by Jeffrey Mining Machinery Company in the early 1950's. Two with capacities of 250 tph were placed in service; none are currently operating. A Bureau of Mines research and development contract is under way for building a redesigned unit ("Multiunit Conveyor") for further testing and evaluation.

The Moleveyor (See Figure 4-3) consisted of a 4-wheeled receiving end, a train of 4-wheeled intermediate cars, and a 4-wheeled discharge end. Each unit of the train had an individually-powered conveyor running the full length, with the discharge end elevated to transfer coal to the next unit in line. Steering articulation between each pair of cars was 45^{o} each side of center.

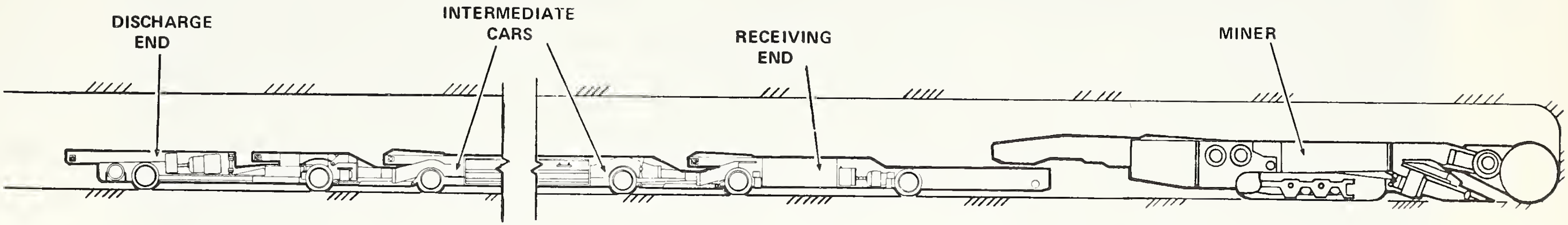
The end cars had 4-wheel drive and 4-wheel steer, and were arranged so that crab steering was available if required. The intermediate cars all had 4-wheel drive, and had steering linkages that were interconnected between cars so that the total train had very good tracking capabilities.

Early versions of this design had belt-type conveyors, but later designs used chain-type conveyors. The discharge end of the train was fitted with a short conveyor section that could be swung or elevated hydraulically. This was used to discharge coal laterally onto a main belt.

Bridge Conveyor-Bridge Carrier System

Bridge conveyor-bridge carrier systems are the most prevalent continuous face haulage systems to be found in U.S. mines at the present time. Approximately 150 such systems are operating, split roughly equally between belt-type and chain-type conveyor designs.

This type of system consists in its modern version, either of two bridge conveyor units and a mobile bridge carrier unit, or of three bridge conveyor units and two mobile bridge carrier units.



COURTESY JEFFREY MINING MACHINERY CO.

Fig. 4- 3 Jeffrey Moleveyor

The first combination is arranged with one of the bridge conveyors attached to the tail conveyor of a mining or a loading machine with the outby end of the conveyor resting on a dolly that travels several feet along the inby end of the carrier. The outby end of the carrier supports the second bridge unit, whose outby end rests on a dolly that rides on the side frames of the main conveyor.

The bridge carriers have a conveyor running their full length, so that in operation coal is received from the inby bridge and carried back and transferred to the outby bridge.

Bridge conveyors range in length from 30 to 45 feet, and are independently driven with electric motors. The belt-type generally are equipped with 36 inch wide belts running at about 400 to 450 fpm, giving a capacity of 500 to 600 tph. The chain-type bridge conveyors carry chain widths up through 28 inches, running at speeds up to 300 fpm, giving a capacity up to 500 tph, with peaks to 650 tph.

With 40 feet of length on the bridge conveyor, and 30 feet of length on the carrier, a three-unit system provides approximately 130 feet of extensibility which becomes approximately 200 feet with a 5-unit system. Total horsepower required ranges from 100 to 180.

Bridge carrier units are currently crawler mounted, although some earlier designs were rubber tired. Crawler drive is generally hydraulic, and the carrier conveyor drive is electric. See Fig. 4-4. Tramming speeds range from 0-90 fpm.

These systems have been marketed for approximately fifteen years. Mechanical problems were quite prevalent with the early units, which appeared to discourage further use of the concept until improved designs appeared a few years ago.

In the form described above, these systems are manufactured by the following companies:

Jeffrey Mining Machinery Company
Long-Airdox Company
West Virginia Armature Company

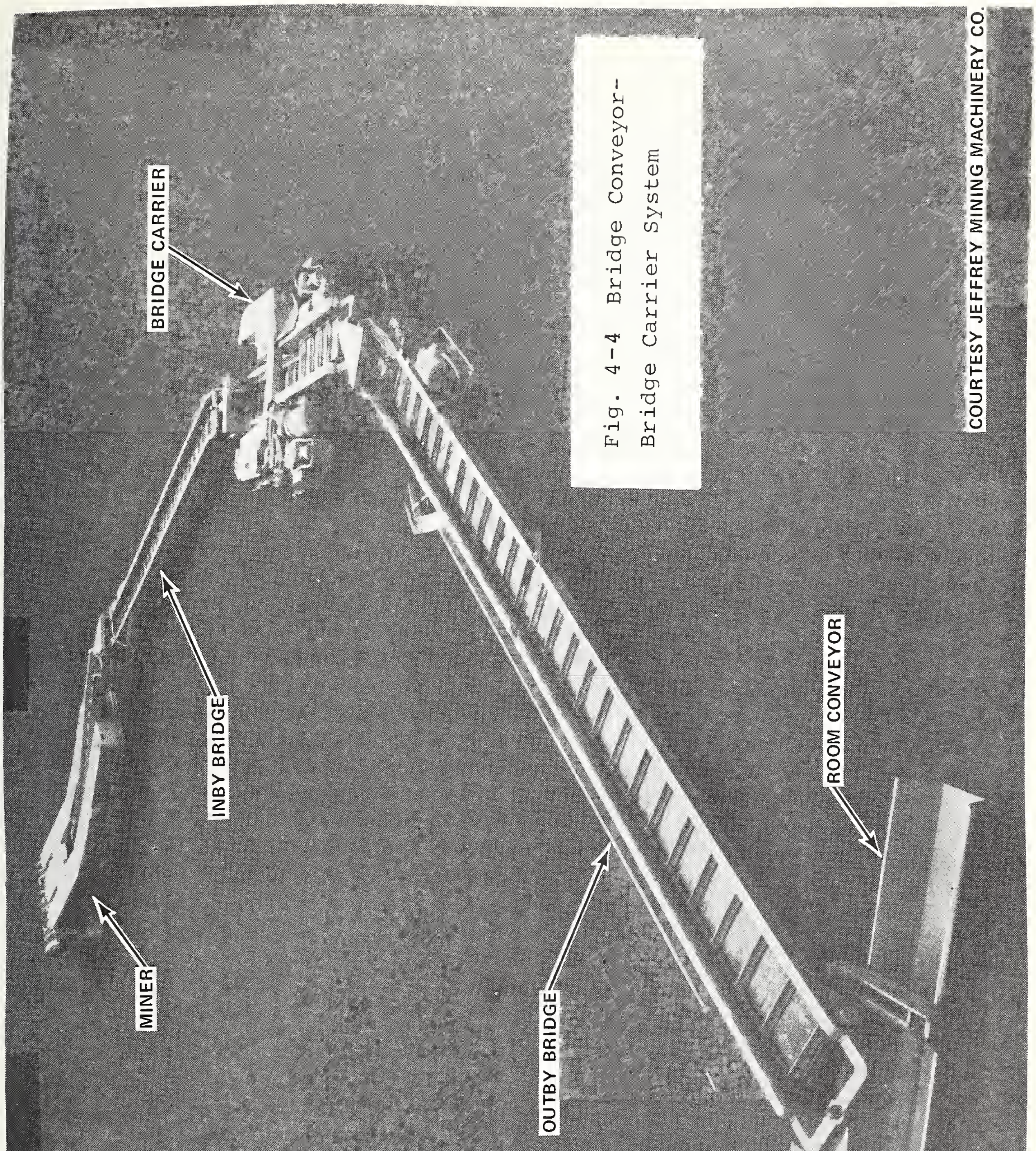


Fig. 4-4 Bridge Conveyor-Bridge Carrier System

The Jeffrey units are chain-type conveyors, and the Long-Airdox and West Virginia Armature units are belt-type conveyors.

Connecting joints between the various units of the bridge conveyor-bridge carrier system allow the units to pivot to negotiate corners, and also allow the joints to flex to accomodate undulations in the floor. Additional flexibility is provided by having the carrier so designed that the inby and outby ends of the carrier conveyor may be raised or lowered hydraulically. To reduce height, idlers have been replaced with slide plates of stainless steel. Noise levels of below 90db have been claimed in some applications.

Severe operating problems have developed in service when very wet conditions exist. Traction becomes a problem, and build-up of material underneath the load carrying side of the belts results in periodic shut-downs for cleaning, which is a slow and difficult task.

Flexible Conveyor Belt Systems

Two Flexible Conveyor Trains, or FCT's were built jointly by the B.F. Goodrich Company and Joy Manufacturing Company in 1974. They are a mobile conveyor system that consists of a special endless elastomeric belt (capable of horizontal bends) supported on a chain of interconnected wheeled carriages, with a self-propelled tractor connected at each end of the chain. The belt is molded with embedded wire rope reinforcements, which limit belt stretching and also provide some stiffness to the belt cross-section. A considerable number of problems developed with componentry but the prime difficulty was with severe cracking of the belt, and with the carryback of fines in belt convolutions (see Fig. 4-5).

At the present time there is only one flexible conveyor installation (200 ft. long) operating in a U.S. underground coal mine. This installation is supported on a roof mounted monorail system. The system requires 7 ft. of clear height,

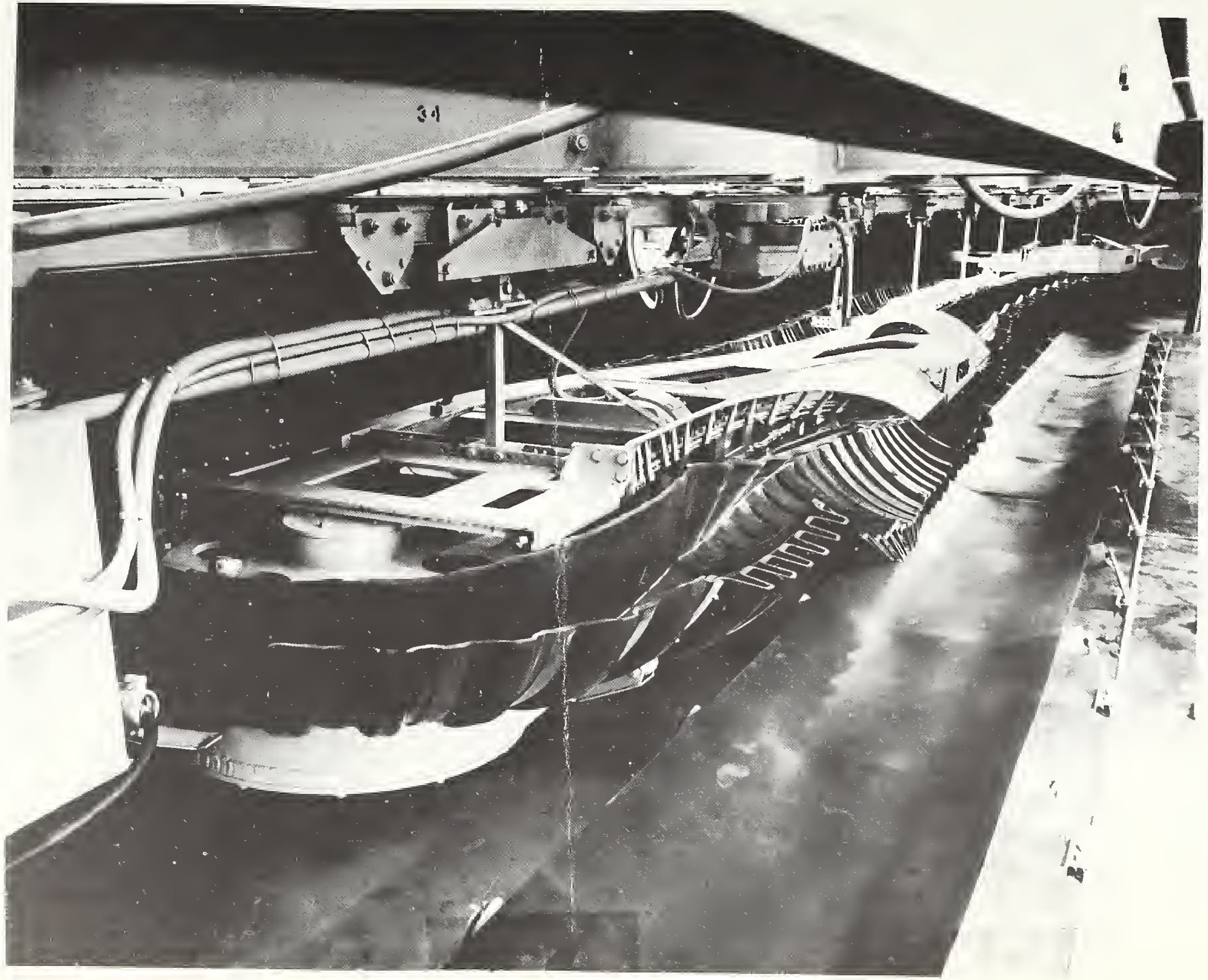


Fig. 4- 5 Flexible Belt System
(Courtesy Joy Manufacturing Co.)

negotiates 90° turns with 15 ft. radius bends, and employs a 32 in. wide belt running at 400 fpm which results in a carrying capacity of approximately 500 tph. This unit is at present relatively noisy, with recorded sound levels in excess of 100 dBA. A newer design incorporating a 40 in. wide belt running at 360 fpm with a capacity of 600-700 tph has recently been offered for sale.

Shaker Belt Conveyor

An experimental shaker belt conveyor was built by Joy Manufacturing Company in the 1950's, but was never put into production. The unit consisted of a 30 inch flexible stainless steel belt preformed to a trough cross section. The belt was stored on a reel at the outby end with reciprocating motion applied by oscillating the reel or the reel carriage. The inby end of the 300 foot system was designed to be anchored to a continuous miner. The belt was to operate over low idlers powered by a spring device on the miner. Material transported was to be discharged outby, either by moving it over the storage reel, or by removing it from the shaker by a plow device. The system had a high power requirement, imposed high loads on the continuous miner, and muck flow characteristics were sluggish.

Extensible Belt Conveyor Systems

Extensible belt conveyors appeared on the U.S. market in the early 1950's. Their use in underground mine applications apparently peaked in the 1960's. It is estimated that approximately fifty commercially manufactured extensible belt systems are presently operating in U.S. underground coal mines. (See Figure 4-6.)

An extensible belt system consists of a drive-storage unit, a number of light support frames for the extended belt, and a tail piece. The drive-storage units are generally crawler mounted, with belt drive, belt storage, and belt takeup all in one frame.

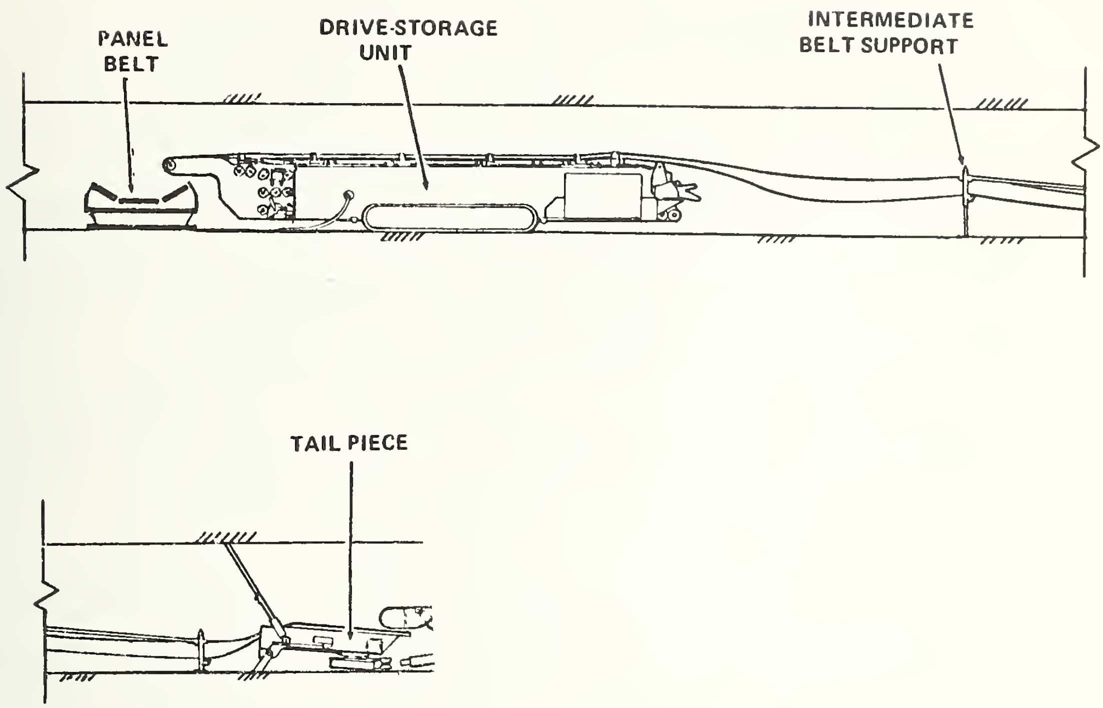


FIGURE 4-6. EXTENSIBLE BELT SYSTEM
(Courtesy Lee-Norse Co.)

In operation the drive-storage unit of an extensible belt conveyor is positioned to discharge to a conventional belt system. As the tail section (crawler or skid mounted) is advanced, intermediate supporting frames are put in place to support the extended belt. When all the stored belt is pulled from the drive-storage section (generally after a 50 to 80 foot advance of the tail section) belt is added in 100 to 250 foot lengths at the inby end of the drive. This is done by splicing in the new length of belt and returning the storage unit pulleys to their original position, which pulls the new length of belt into the drive-storage unit.

Like any conventional belt conveyor, the extensible belt conveyor must be operated in a straight line. The tail section must be kept in alignment with the drive-storage units, and the intermediate belt supports must be maintained in line with the drive unit and the tailpiece.

Maximum overall extended length for an extensible belt conveyor is usually in the 1000 foot range, but heavy duty units have been built to provide up to 2000 feet of extension.

The height of the various units for extensible belt conveyor systems, including drive-storage units and tail sections, varies from 30 inches to 54 inches, depending on make, model, belt width, belt storage capacity, and ground clearance desired. Increases in belt width, storage capacity, and ground clearance result in a higher machinery profile.

Conveying capacity for a 36 inch wide belt running at 500 fpm ranges from 350 to 500 tph. The spillage problem can be serious if the alignment of the belts is not maintained and if the intermediate belt supports are poorly placed.

It should be noted that in the coal mine application these various extensible systems have occasionally been applied in series. The extensible belt conveyor system (storage reel) has been used as a rapid means of extending the main haulage conveyor while bridge conveyor-bridge carrier units have been used to provide the link between the continuous miner and the main conveyor.

4.3 OPTIMUM SYSTEM FOR TUNNELING

While there are similarities between the continuous underground miner and the tunneling machine in terms of their cutting action and the belt discharge of the muck, the extensible conveyor system operation requirements for coal mining and rock tunneling applications are significantly different.

a. Regulatory safety standards for operations and equipment design are significantly more restrictive in coal mining.

b. The size and shape of the muck produced differs. Coal is a uniform, substantially softer, less abrasive material to handle.

c. Equipment height is much more restricted in mining of thin-seam U.S. coal.
d. The mine operational requirements necessitate relatively high flexibility in a vertical plane to follow undulations of the seam.
e. The mine operational requirements necessitate a number of sharp horizontal bends in the extensible transport system.
f. The mine operational requirements necessitate frequent retracting of the extensible transport system to permit roof bolting and/or relocation of the miner to another mining face.
g. The propel power required for cutting and frequent maneuvering of the miner prevent towing of the transport system by the miner, necessitating self-propel devices in the extensible system.
h. Mining operations have a lesser need for lining material flow up to the face past the transport system. (Temporary vs. permanent for a tunnel).
i. Severe degradation of the coal by the transport system is generally undesirable.
j. The mine floor is more obstructed with debris, primarily from rock falls from the roof.

Table 4-4 summarizes the previously described extensible conveyor installations. It is apparent that while considerable effort has been applied for over 25 years to the design of a number of innovative approaches, their actual success has been limited. Even those designs that reached a manufacturing production status are in reality currently applied in a relatively small number of installations.

The use of those systems however, is closely tied to the acceptance of the continuous miner (plus the growing use of longwall techniques) and continuous haulage systems, the availability of equipment with satisfactory reliability, the intoduction of new mine planning techniques, opening of new mines

TABLE 4-4.

Extensible Conveyors

Applied in U.S Underground Coal Mines

Type	Year of Introduction	Conveyor	Max. Capacity (tons/hr)	Max. Extensibility (ft.)	Common Widths (inches)	Comments
Overlapping or Cascading Units:						
Beam Stage Loader	1950	Belt or Chain	-	30-45*		
Portable	1949	Belt	60-90	8-12*	18	Experimental
Articulate Self-Tracking	1950	Belt				Experimental
Moleveyor	1950	Belt-Chain	250			Experimental
Mineveyor	1959	Belt	450	75*	36	Limited Service
Push Button Miner Train	1960	Chain	350	1000	90	Limited Service
Bridge Conveyor Bridge Carrier	1960	Belt Chain	500-600 500-650	130-200	36 28	Production Design
Flexible Belt	1970	Belt	600-700	400	40	New Product
Reel or Storage Units:						
Shaker Belt	1950	Flexible Steel			30	Experimental
Extensible Belt	1950	Belt	350-500	1000	36	Production

*Multiple units can be used to increase extension.

with adequate capital, approval of related regulatory requirements, etc. In short, the applications have been curtailed by a variety of business performance considerations.

Equally restrictive are the severe technical requirements of coal mining applications. In particular the very low profile (generally less than 5'), the negotiation of sharp turns, and the frequent retraction and relocation of the system impose costly design additions in terms of complexity, and reliability. These do not exist in tunneling. Minimal height profiles for typical projects range from 8' to 30'. Tunnels are generally straight for considerable lengths with gradual turns accomodated by the inherent flexibility of the components and sharper turns negotiated with a series of short extensions and the introduction of curved or flexible insert sections. Relocation is infrequent; associated with completion of a project phase and generally involving dismantling and reassembly at a different site.

The muck rates (tph) currently transported in mining are compatible with present tunneling needs but must be progressively increased to meet anticipated future tunneling demands. The required length of extensibility tends to be greater in tunneling.

How much extensibility should be provided is dependent on the type of main haulage system employed and the optimum frequency for shut-down to extend this semi-permanent portion of the system. Factors additionally influencing this decision include:

- Maintenance requirements for the tunneling machine
- Maintenance requirements for the extensible conveyor
- Maintenance requirements for the main haulage system.
- Schedule work hours.
- Available crew size.

The tunneling muck rate charts included in Chapter 2 have been redrawn (See Figures 4-7, 4-8, 4-9) to illustrate the extensibility required if it were assumed that two ten-hour production shifts were scheduled each day. The remaining 4 hours are to be utilized for maintenance, retraction of the extensible conveyor, and the corresponding extension of the main haulage system. Projecting into the future, and recognizing differences in tunnel diameter and rock hardness, the anticipated advance rates per day range from 20 to 800 feet. These assumptions represent the extreme in terms of the extension capability required. Obviously, any significant down-time for other reasons should be utilized whenever possible to restore the extensible conveyor to its compressed length in preparation for the next period of sustained operation.

While underground applications have included both drag chain and belt conveyors in extensible units, they have been primarily for the transport of coal. The drag chain used for other than the special armored face conveyors (practical limit, 600 ft. long) have all been relatively short--less than 200 feet. These conveyors have high noise levels, are expensive and have had limited prior application in handling abrasive materials. The general opinion of those who have worked with chain conveyors is that they would not hold up under the severe abrasive service that would be encountered in handling tunnel muck. Current experience with the newer belt constructions suggests, on the other hand, that they will perform well in this service, have no length limitations, and are the most logical choice for tunneling.

Underground experience suggests that the overlapping or cascading design approach is more attractive that the reel-storage type, because of the design simplicity. Storage units are costly and would not provide the desired capacity for many tunneling applications. The extensible systems with belt storage incur delays when belting is added and can experience problems with spillage if the intermediate belt supports

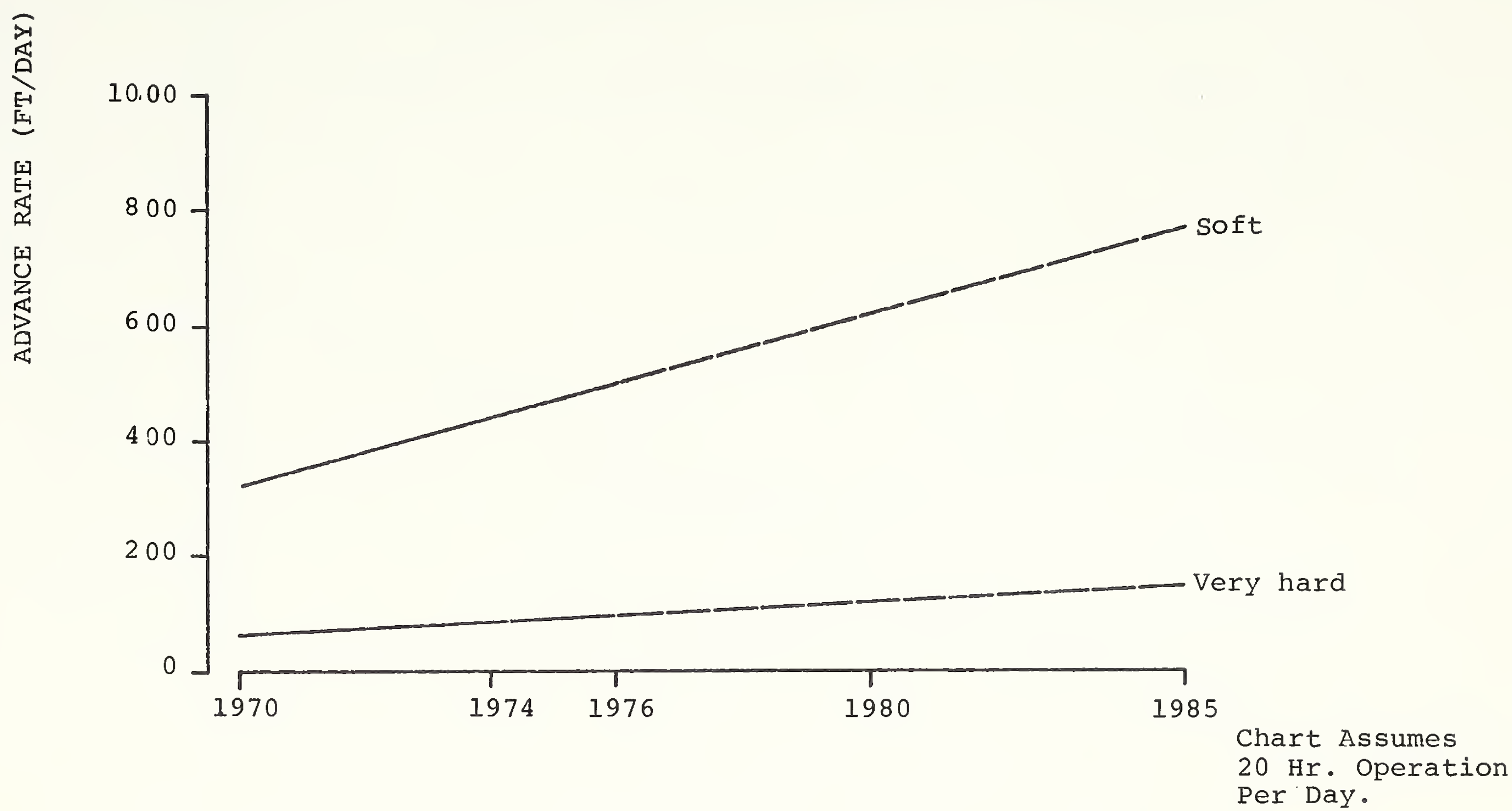


FIGURE 4-7. PROJECTED DAILY TUNNEL ADVANCE RATES: 12 ft. BORE

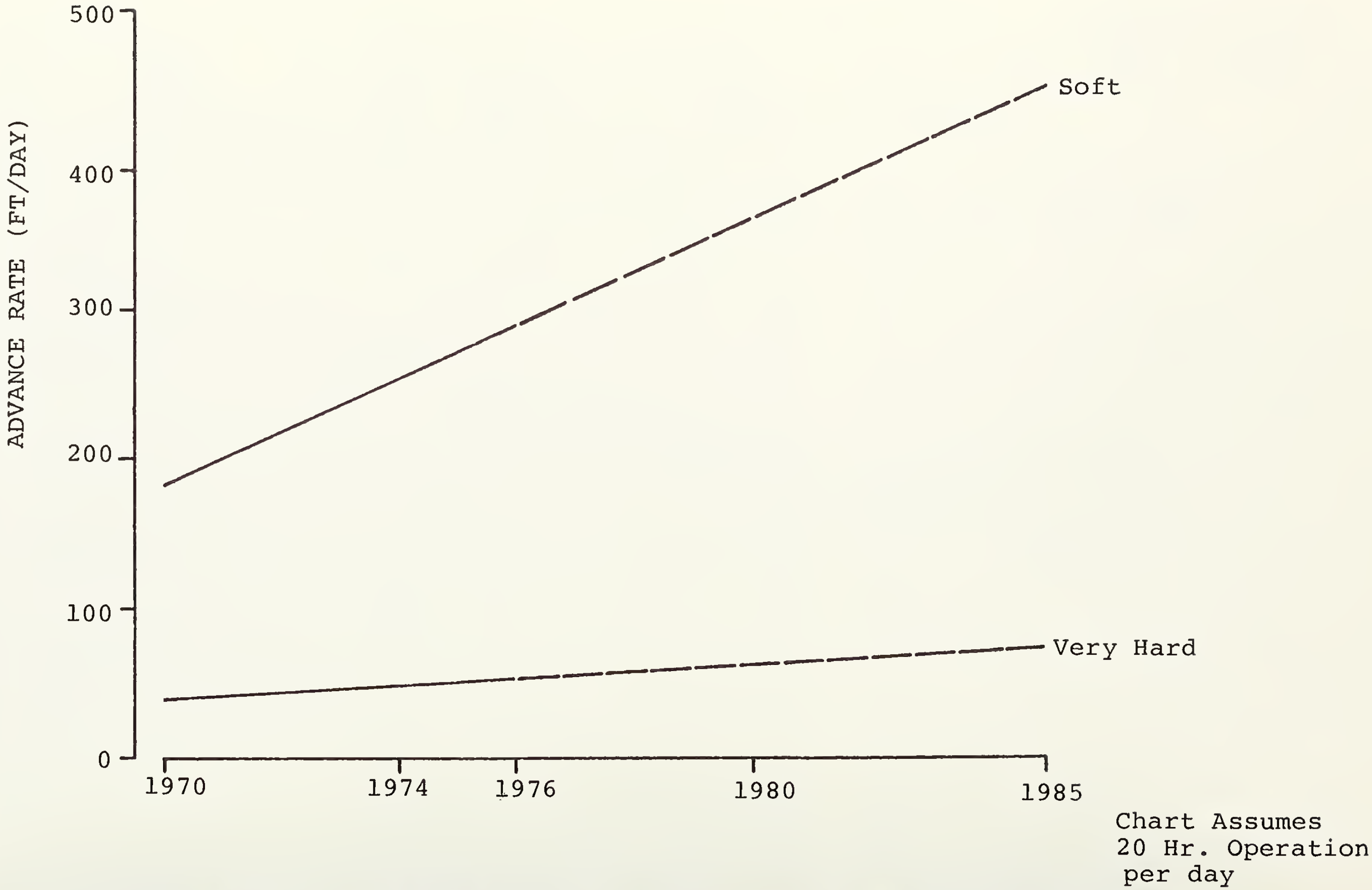


FIGURE 4-8. PROJECTED DAILY TUNNEL ADVANCE RATES: 20 ft. BORE

FIGURE 4-9. PROJECTED DAILY TUNNEL ADVANCE RATES: 35 ft. Bore

are poorly placed, but their prime disadvantage is the complexity of the operational sequence. If the extension requirements are greater than the storage capacity, 15 to 20 minutes with a trained crew are required to insert 100 ft. of belting and the necessary framework for a 50 ft. extension. To retract, the drive storage unit must be advanced, the intermediate belt supports removed and stored, and belt sections removed.

The cascading designs to date have employed multiple modular units 30 to 60 feet in length, with the total extensible length equal to the initial overlap between succeeding units times the number of units. Most units have been wheel mounted to facilitate advancing and retraction of the extensible system. Earlier versions assumed that the units were stored and added into the system as additional length was required. More recent designs made the units "stackable", so that they were continuously in the system with the overlap between units successively reduced as the inby end of the system was advanced. This latter design eliminated the storage problem of the extra units but the number of transfer points continuously in the system is maximized.

Designs of this basic type should work effectively in a tunneling operation. The modular units involved, however, could be much simpler and longer than the past mining versions because of the greater height normally available in a tunnel, the large radius turns encountered and the possibility of advancing the inby end by a direct connection to the preparation unit. A hydraulic or electric winch at the rear of the unit could be utilized for retraction (compression) of the units to their overlapped configuration. The only power required on the modules would be that for the belt drives.

While wheel mounted cascading units have been the most popular in the past, considerable interest is developing in the use of monorail suspensions. The monorail reduces the congestion on the floor and has proven to be efficient and trouble-free. Power requirements to move the suspended conveyor are on the order of 25-30 lbs. of force per ton of suspended weight, compared to forces of 250-300 lbs. to move a ground-supported system with wheels or crawlers. The monorail installation generally requires a roof-bolting device (common to underground construction), bolts, and relatively easy to handle rail sections.

Wheel mounting, if desired, is straight forward because the loads are low (approximately 5 tons for a 40 ft. loaded section) and there is no powered drive required. Caster-type wheels (restricted rotation) with some guide frames on the succeeding units would probably provide adequate tracking and no steering mechanism would be needed.

In the cascading configuration, each conveyor unit essentially rides above and discharges to a conveyor below. If the units are negotiating a turn so that they are not fully aligned, considerable care is required to provide some form of transfer chute that properly directs the flow from one belt to another. Final discharge is to the permanent haulage system that transports the muck from the tunnel. This main belt and/or slurry pipe line would normally be located on the floor or hung on brackets on the side of the tunnel, in a position which provided an acceptable compromise between ease (and cost) of installation vs. providing space for the transport of men and supplies to the face. A short transfer conveyor at the discharge end of the cascading system may be required to direct the flow to a side mounted haulage system.

The complexity of the overlapping or cascading design approach is a function of the length of uninterrupted extensibility desired. The potential extensible length required in tunneling is greater than in mining, which suggests a large number of transport points. Each transfer point increases the amount of belt abuse, possible spillage, dust generation and degradation of the transported muck. The design objective is ideally to use the cascading technique and minimize the number of transfer points.

The simplest approach would be to utilize two wheel-mounted conveyor modules with an overlap length equal to the maximum extension anticipated. The upper conveyor, connected to and fed directly by the preparation unit, would be straddle mounted above and discharge to the second which in turn would discharge to the main transport system. The two conveyor units could be of single piece construction. During extension the upper conveyor would travel forward until the overlap was eliminated. Retraction would necessitate simple uncoupling of the lower conveyor from the main transport line and use of a winch to pull the unit forward to the full overlap position. The main conveyor would then be extended in a conventional manner. This configuration would appear to be the cheapest and most reliable, but does require two complete independent conveyor assemblies and would be limited to essentially straight tunnels.

A variation of the two fully overlapping conveyors described above would be the monorail suspension of the upper unit. This approach is also quite straight forward, with minimal congestion of the floor, but requires progressive addition of modular lengths of monorail immediately before or following the preparation unit, so that the upper conveyor can be continuously advanced. It would be generally similar to the above proposal but the excess monorail length would have to be disassembled when the unit was retracted to start a new cycle. The removed monorail modules would in this arrangement be recycled to the front for further advance of the system. A monorail suspension might, however, occupy space required for the secondary transport system mentioned earlier, for materials and supplies. The same monorail beams could not be used for both purposes because supplies could only be brought forward to the point where the overlapped extensible conveyor ended, which would be a substantial distance behind the tunneling machine.

The use of only two overlapped conveyors means that each could be quite long - up to 800 ft., dependent on advance rates and projected periods of sustained continuous excavating. With lengths of this magnitude, there may be a requirement to negotiate a gradual curve or bend beyond the capabilities of a standard belt conveyor installation. In that event, consideration must be given to the use of flexible belts or reverting to shorter cascading modules, which could provide the necessary horizontal radius.

While we have been considering only the requirements for the extensible conveyor, it is important to recognize a related problem associated with the main haulage system. Maximizing the time that the transport system operates continuously implies that the extensible system be retractable in a short period of time, and that the main system be extended ideally in equal or less time. Actually, the retraction-extension cycle may be paced by the main system and unless present installation procedures can be modified, this operation would seriously reduce the operational time.

Four-man crews in underground coal mining can add 100 feet to a standard floor-mounted conveyor in from 2 to 8 hours. While these times would probably be reduced in the tunnel environment, delays for main conveyor extension of this magnitude would appear to be unacceptable. However, some thought has been given to the possibility of mechanizing conveyor installation procedures to reduce the time required. Preliminary studies for the U.S. Bureau of Mines suggest that with a specially designed and properly equiped mobile assembly vehicle, 100 ft. of conveyor might be installed by 4 men in less than half an hour. The reduced delay is offset by the cost of the equipment and the congestion that the assembly vehicle adds to the tunnel.

Less is known about the time required to advance other potential main haulage systems, such as a slurry pipeline. The problems are similar to the conveyor, however, and it is apparent that careful consideration must be given to detail procedures and special mobile storage-assembly vehicles to minimize installation time.

If the main haulage system can be rapidly advanced, the door is open to another variation of the earlier thinking on extensible systems when used in conjunction with main belt haulage. The preparation unit could feed and pull along a monorail (continuously advanced) suspended conveyor section of a length equal to the required extensibility. This conveyor would initially fully overlap and discharge directly to the main conveyor, floor-mounted below. The open space beneath the suspended conveyor, created as the system advanced, could be utilized to prepare for a permanent extension of the main conveyor. Work could progress to the point of splicing in a new belt section to minimize the shut-down when the initial overlap had been used up. The practicality of this approach is dependent on the adequacy of the working space and time required to complete the tie to make the main conveyor system operational.

It is of interest to note that for tunneling it would appear logical, as indicated earlier, to advance any extensible system by direct connection to the tunnel boring machine. If the capacity of the machine to provide this added forward thrust is marginal or inadequate, then another device must be introduced. Unit modules attachable to the conveyor train have been applied in underground mining which duplicate the propel action of the tunnel boring machine. Similar units combining vertical hydraulic roof jacks and horizontal push rams, skid mounted, could be designed to synchronize the advance of the extensible system with the TBM.

The major components of an extensible belt conveyor system module are:

a) Belt
b) Idlers: carrying and return
c) Pulleys: head and tail
d) Drive unit
e) Supporting structural frame
f) Suspension (wheel or monorail)

g) Belt tension unit

h) Feed accessories

i) Discharge accessories

j) Belt cleaners (optional)

The available design data covering each of these in belt conveyor installations is too extensive to be discussed in detail. However, it is helpful to review briefly a few of the more important considerations that provide added background to the tunneling application. Conveyors are very compact, high capacity systems, capable of handling muck but with some limitations in respect to the lump sizes that can be handled. Tables 4-5, 4-6, 4-7, illustrate the relationships between belt width, speed, lump size, and capacity that might be expected in muck handling.

Some of the broad design factors are as follows:

1) If at all possible provide some means to minimize the material surges prior to placement on the belt. Figure 4-10 shows a Cowlishaw Walker & Co. (England) development for use in coal mines, employing a hopper-type bunker with an apron feeder on the bottom to move coal into and out of the bunker on demand with automated controls.

2) The number of load transfers from one belt to another must be kept to as low as possible, to minimize dust conditions and potential belt damage. Transfers are high maintenance areas subject to clogging.

3) Belt width can be minimized if maximum rock size can be reduced (preparation unit), with an attendant increase in belt life and a reduction in cost.

4) Consideration should be given to cable belt conveyor designs (idlers supported on cables) and/or flexible cable idler suspension configurations because of their low cost, simplicity, and speed of installation. Their capability to adapt to the conveyor load distribution results in slightly higher capacities.

TABLE 4-5. Suggested Minimum Belt Width in Relation to Lump Size Distribution

Type of Material	Ratio Belt Width to Lump Size	Belt Width (in.) 18	24	30	36	42	48
		(Largest Lump Dimension-in.)					
Mine Run Material							
Not more than 1% max. lump; balance 1/2 max. lump, or less.	2-1/4	7	9	11	14	16	18
Not more than 6% max. lump; balance 1/2 max. lump, or less	3	6	8	10	12	14	16
Crusher product	3-1/2	5	7	9	10	12	14

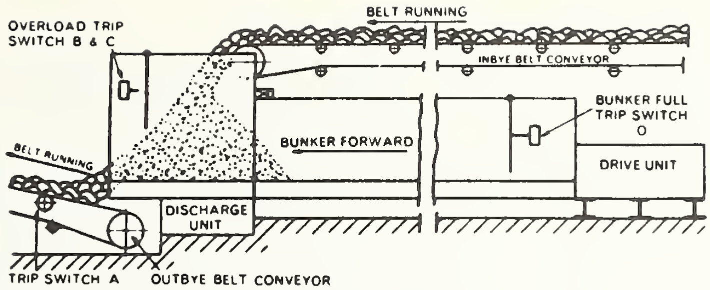
TABLE 4-6. Recommended Maximum Belt Speeds (fpm)

	Belt Width (in.) 18	24	30	36	42	48
Sand	400	500	600	600	650	700
Gravel, Crushed Stone	300	400	450	550	550	600
Small Abrasive Material	300	400	450	500	500	550
Large Abrasive Material	-	300	300	350	350	400

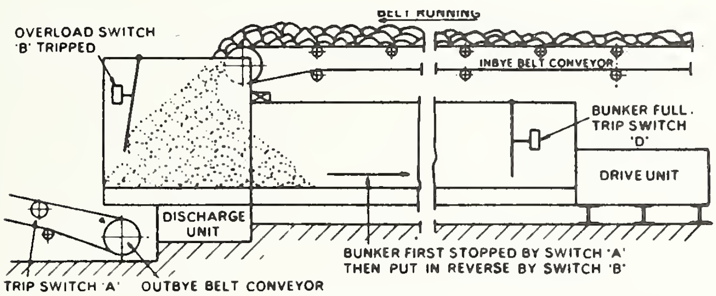
TABLE 4-7. Muck Handling Capabilities TPH

Belt Width (in.)	Belt Speeds in Feet Per Minute 100	200	300	400	500
18	70	140	210	280	350
24	132	264	396	528	660
30	215	430	645	860	1075
36	318	636	954	1272	1590
42	442	884	1326	1768	2210
48	575	1170	1755	2340	2925

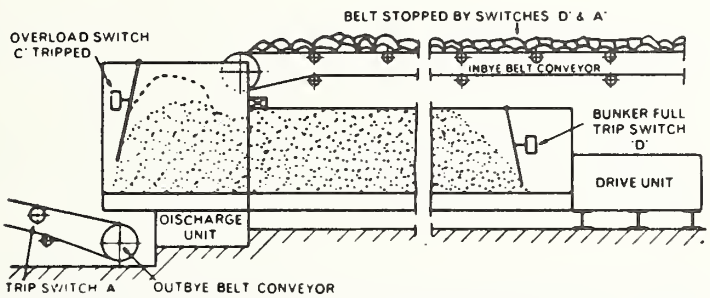
(35° Three equal roll troughing idlers-25° surcharge angle - 100#/cu.ft. material)



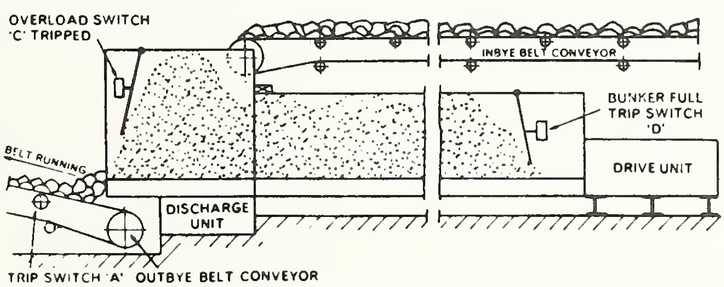
In the Cowlishaw Walker system of automatic control for bunker conveyors, the normal condition has both inby and outby belts running. Should the flow from inby increase, an overload trip switch starts the bunker chain in reverse, thereby relieving the overload on the outby belt. As soon as inby flow returns to normal, forward loading resumes.



When the outby belt is stopped, the bunker chain is reversed to shunt incoming coal to the bunker. This moves the load away from the switch plate. Should the outby belt remain stopped, the sequence is repeated as the overload switch is again tripped. This permits the bunker to be filled gradually to its maximum capacity.



When the bunker is full and the outby belt stopped, the combination of existing signals, together with the trip from the overrun plate, stops the inby belt and prevents further loading.



When the outby belt restarts, its signal activates the bunker chain to move forward and unload the bunker. The inby belt does not restart until all overload is cleared from behind the forward sensing plate.

FIG. 4-10 Cowlishaw Walker System for Automatic Control of Bunker Conveyors (Courtesy Cowlishaw Walker)

5) Almost all the wear damage imposed on a belt occurs at the feed point. While many detail design features are generally aimed at this consideration, the primary need is to load whenever possible in the center of the belt, in the direction of belt travel, and employ means to introduce the fines first, so that they form a bed to support the lumps.
6) Hydraulic drives are common because of their compactness and ability to provide broad speed ranges. Fireproof emulsions are available for the hydraulic fluids.
7) The best combination of belt width and speed is the maximum speed and minimum width at which the material can be handled without creating operating or maintenence problems, such as spillage, degradation, excessive belt wear, and ineffective material transfer at conveyor junctions.
8) Rectangular tubing can be advantageously utilized in truss-type side frame supports for the conveyor, reducing weight and improving accessibility for inspection of the top and bottom of the belt.
9) Crawler mountings for conveyor module units are practical when the bottom is soft and low ground bearing pressures are necessary.
10) Deep-troughed belts will transport more material on narrower belts operating at slower speeds. With modern belt construction, the use of deep trough conveyors does not reduce the expected belt life.
11) Large diameter carrying idlers reduce belt damage and take less power.
12) Reduced voltage starters on higher-horsepower motors minimize line surges and belt splice maintenance.
13) Consideration should be given to sequential starting arrangements with outby units initiated first.
14) For repair purposes, each belt must be capable of being operated independently.

The critical need for ruggedness and reliability in the design requires little discussion. The extensible unit is a vital link in the total tunnel excavation process. System availability for operation is the product of the mechanical availability of all of the modular components. There are no alternate or back-up systems if the unit is down for service or repairs.

Most conveyor units are customized in terms of their overall configuration but the individual components such as idlers, pulleys, drives, etc., are proven, relatively standardized and available from numerous manufacturers. A substantial amount of test data and field operating experience have been incorporated into the production designs. Broad application experience is available from the manufacturers, and their technical staffs are available for consultation on special problems.

The proposed system contains moving belts which can present a safety hazard but the conventional drive and take-up systems are well enclosed and the idlers and frame structure are of a simple, clean design. Most importantly, the extension and retraction operations which present the greatest hazard are performed at very low speeds. Underground safety experience has been good.

Safety considerations have led to the broad use of the following:

1) Pull cords extending the conveyor length, which can stop or restart the unit in the event of an emergency.
2) Alignment switches at appropriate locations which detect serious misalignment of the belt and stop the conveyor.
3) Over-and-under-speed centrifugal devices which shut off the conveyor if the belt is not running at a safe operating speed.
4) Electrical interlocks so that any conveyor which is stopped will automatically stop all conveyors loading onto it.

5) Drives equipped with automatic fire protection, such as deluge sprinklers.
6) Prestart warning devices.
7) Holdbacks or brakes on inclined conveyors where slopes exceed 2%.

Conveyors lend themselves to automated controls and a number of units are in service with a high degree of sophistications. As an illustration, computer control systems have been employed in England for monitoring conveyor operations which provide feedback on:

Load lockout	Slipping belt
Overheating	Impending overload
Smoke	Belt misalignment
Brake overheating	Safety line used
Torn belt	Maintenance run

Little specific data can be presented on anticipated cost of an extensible conveyor system. The commercial units marketed for use in underground coal mines are much more complex than those required for tunneling. Based on 1975 prices for a 12 TPM system, the prices for these systems ranged from $800 to $1100/ft length, with flexible belt units at the upper end of the range. The simple two unit cascading modules applicable to tunneling, custom designed (with standard components) for the more severe muck handling, would be expected to fall in the $300 to $500/ft (retracted length) range.

The only other continuously extensible types of systems which have been considered in any detail for tunneling in recent years have been pneumatic and slurry. Experimental installations for both have been operated but insufficient cost and operational data are available at this time to evaluate their relative competitiveness.

In summary, it can be concluded that belt conveyor systems can be very effectively applied as an extensible link between a tunnel boring machine and a permanent main haulage system, to permit continuous muck transport for prolonged operational periods. While considerable development effort has been directed towards extensible systems, particularly for underground coal mines, the current units are unnecessarily complex for tunneling applications. Simplified configurations of cascading conveyor models would appear adequate to meet anticipated applications. The technology and the componentry exist for design and manufacture of such systems. The projected extensible lengths required and the muck carrying capacities required pose no serious problems. An extensible belt conveyor system can be characterized as follows:

1) High capacity
2) Compact
3) Relatively simple proven componentry
4) High reliability
5) Low operating cost
6) Low labor intensity
7) Operation can be readily controlled mechanically and electrically to minimize dependence on operating personnel.
8) Relatively safe to operate.

5. PIPELINE TRANSPORT

5.1 HYDRAULIC PIPELINE

A slurry pipeline system for muck haulage will necessitate two pipelines, an outgoing pipeline to transport the muck slurry and an incoming pipeline to carry the water supply. If the tunnel is wet, seepage water can be discharged along with the slurry, albeit the solids concentration will be reduced. It is unlikely that a wet tunnel can consistently supply the water requirements for a slurry pipeline.

5.2 CLEAR WATER SYSTEM

Figure 5-1 is a plot of specific power in kilowatt-hours per ton-1000 ft. for discharging water in gal. per min. (gpm, or tons per hour) in pipe diameters of 4 in. through 14 in. Two pipewall roughnesses (ε) are shown, smooth (ε=0 ft.), and new commercial steel (ε=0.00015 ft.). Plastic pipe for an incoming water pipeline can be regarded as smooth hydraulically, whereas steel pipe wall roughness can vary from smooth to very rough. In the former case, turbid water containing abrasive solids may polish a steel pipe to a smooth finish. For the latter case, an intermittent flow operation may produce a pipewall roughness somewhat rougher than new commercial steel as a result of oxygen corrosion. Note that the graph is for pipe wall friction only and does not include power requirements for resistance of fittings and minor losses, elevation differences, or pump-motor efficiencies. Variation in water temperature from 20°C will not effect significantly the specific power requirements. Flow in all cases is turbulent.

To use the graph, enter the abscissa with the desired flowrate in gpm, or tons of water per hour, and select the pipe diameter of the nearest commercial size (rough or smooth) to give the lowest specific power. For example, a flow of 400

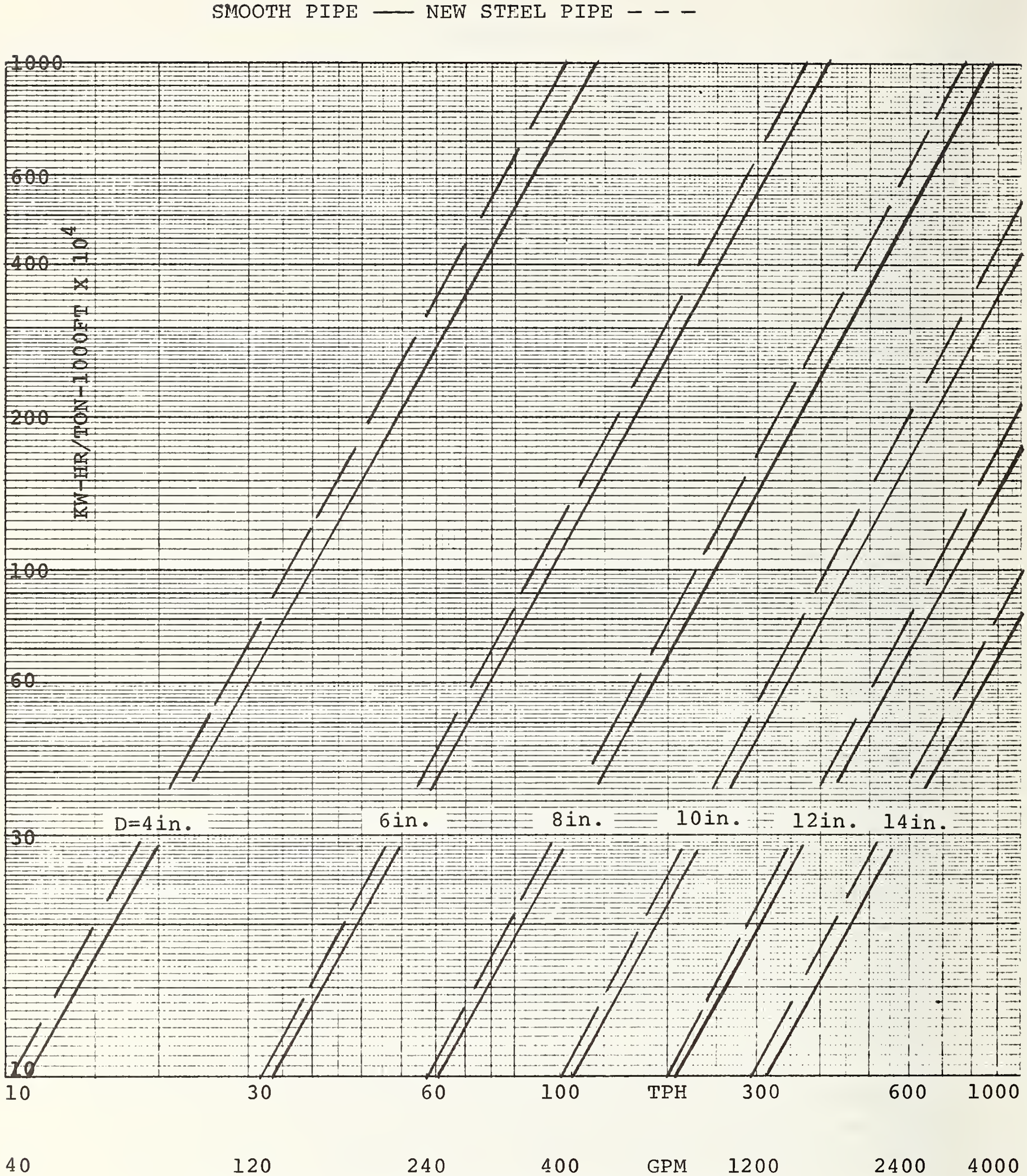


FIGURE 5-1. ENERGY REQUIREMENTS TO OVERCOME WALL FRICTION IN PIPELINING WATER AT 20°C

gpm of water in a new steel pipe of 8-in. diameter gives a velocity of 2.6 fps and a specific power of 0.0019 kw-hr/ton-1000 ft. or 0.0019x100 tph x 2000/1000=0.38 kw for a 2000 ft. length of pipe.

Since $0.38 \text{ kw} \times 1.341 \text{ hp/kw} = Q_{cfs} \cdot \Delta P_{psf}/550$,

then $\Delta P_{psi}/2000 \text{ ft} = \dfrac{0.38 \times 1.341 \times 550 \times 448.83}{400 \text{gpm} \times 144} = 2.184$

where ΔP is the pressure drop to overcome pipewall friction.

It is desirable to perform an exercise in economics to determine if there is some benefit to be obtained by using identical diameters for both slurry pipe and water pipe. For abrasive slurries, sections of slurry pipe can be interchanged with water pipe to extend the wear life of the slurry pipeline system.

Slurry System

Nomographs were prepared for various size distributions of coarse slurries from pipeline data available in the CSM slurry data bank. Figs. 5-2, 5-3, and 5-4 show specific power in kw-hr/dry ton-1000 ft. versus throughput capacity in dry short tons of solids per hour, for particle top sizes of 1 inch, 1/4 inch and 4 mesh (0.186 in). The corresponding weighted mean particle sizes were about 1/2 in. (13 mm), 2 mm, and 0.28 mm respectively.

The first two data sets were developed from raw oil shale slurries of several size distributions pumped through a 6-in. diameter and an 8-in. diameter pipeloop. Headloss correlations were established for these slurries and then scaled to tunnel muck slurries by adjusting the specific gravity of the raw oil shale from 2.19 to 2.65 for "average" tunnel muck. Shape factors for the oil shale were measured from 0.25 to 0.50. A value of 0.5 was assumed for the tunnel muck. The shape factor measures the deviation from sphericity according to drag coefficients measured for particles falling in quiescent water. Spheres have a shape factor of unity.

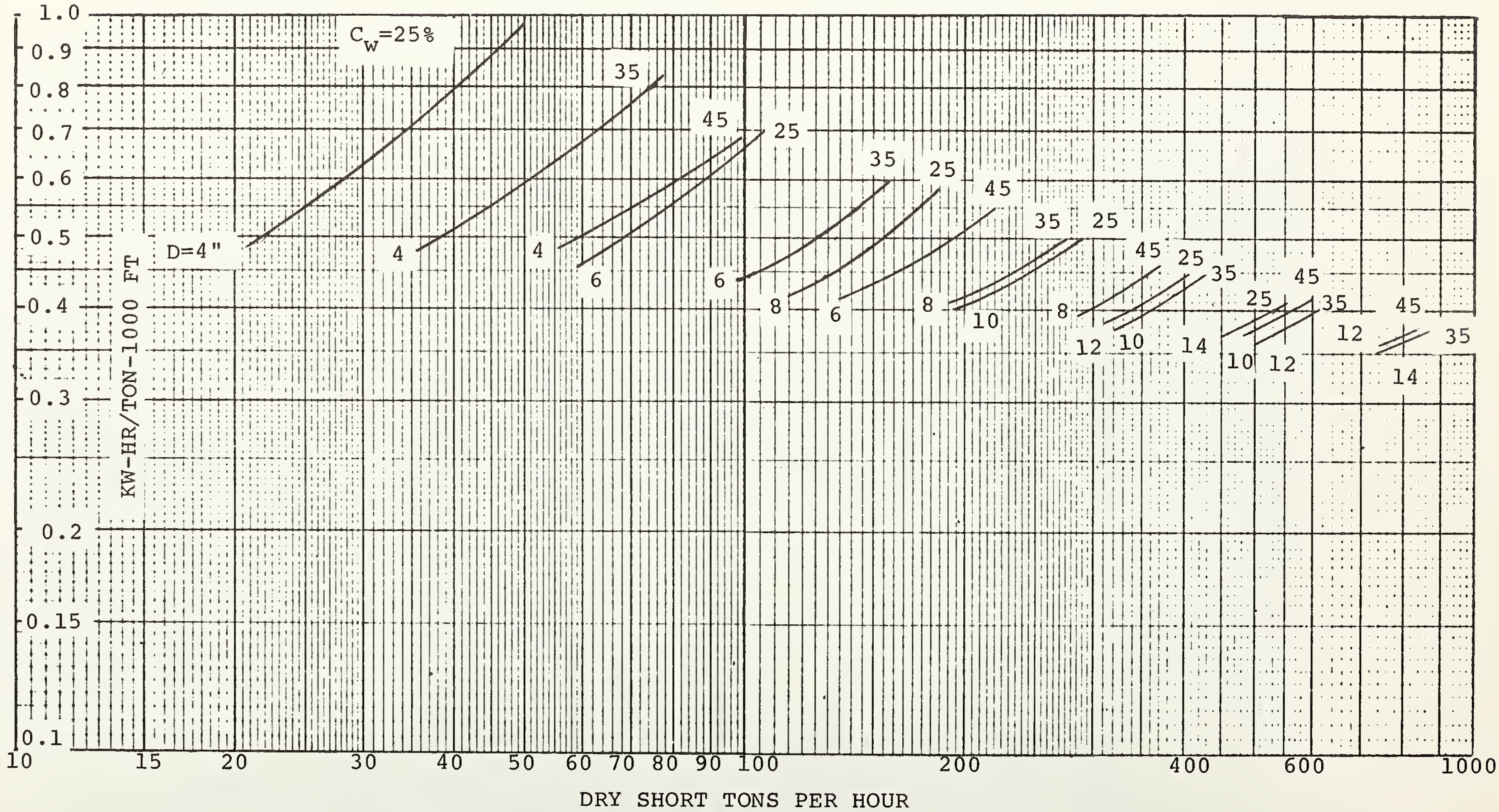


FIG. 5-2. SPECIFIC POWER VS THROUGHPUT FOR PIPELINE TRANSPORTATION OF 1 INCH TOP SIZE TUNNEL MUCK.

(New Steel Pipe, 20°C Slurry, Solids Concentration by Weight.)

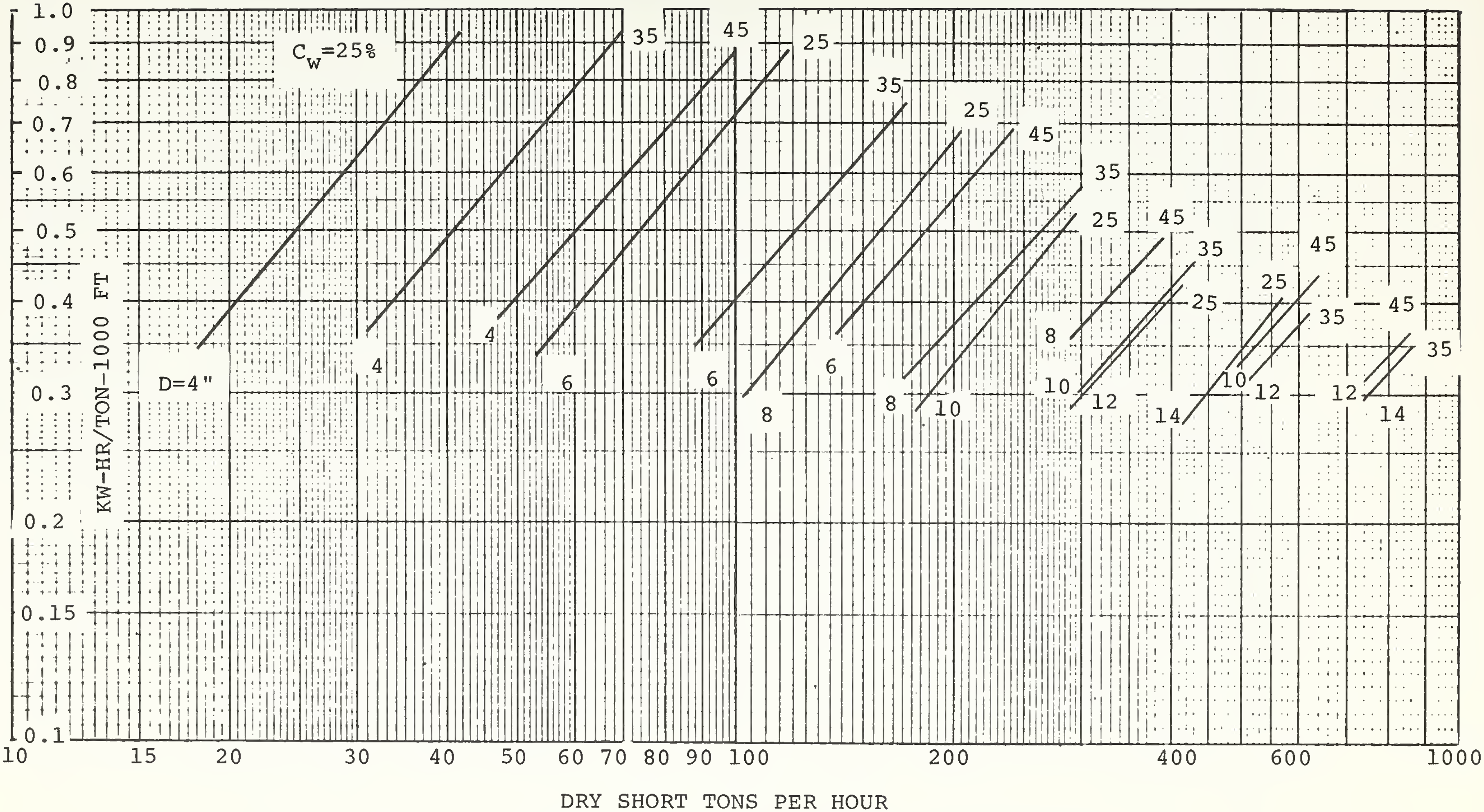
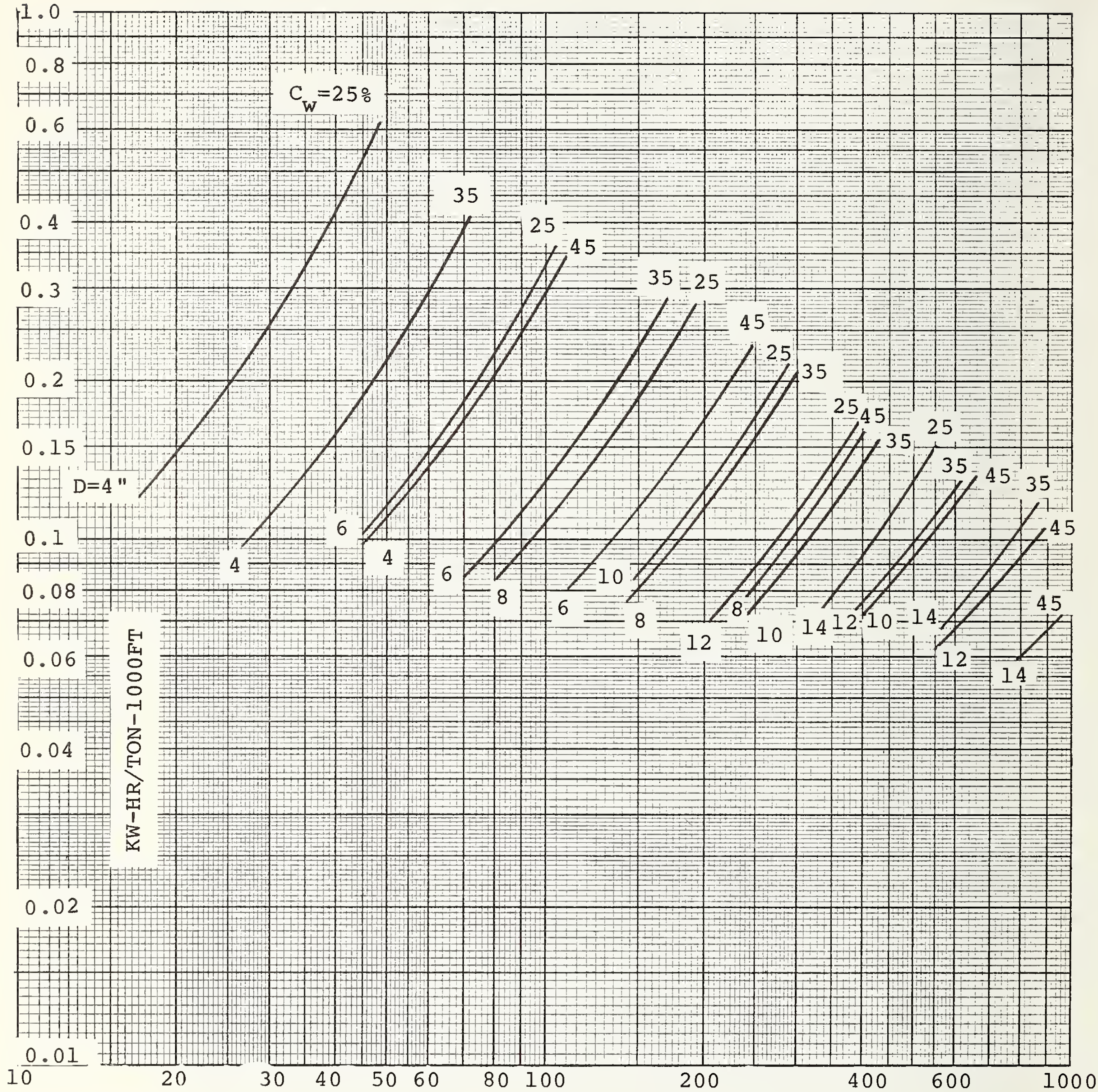


FIG. 5-3. SPECIFIC POWER VS THROUGHPUT FOR PIPELINE TRANSPORTATION OF 1/4 INCH TOP SIZE TUNNEL MUCK. (New Steel Pipe, 20°C Slurry, solids Concentration by Weight)



SHORT DRY TONS PER HOUR

FIG.5-4. SPECIFIC POWER VS. THROUGHPUT FOR PIPELINE TRANSPORT OF 4 MESH(0.185 l.c.) TOPSIZE MUCK
(New Steel Pipe, 20C Slurry, Solids Concentration by Weight)

The third data set was taken from pumping tests performed by the Saskatchewan Research Council (6) and used by the authors in an earlier report (1).

Table 5-1 lists the properties of the slurries and the headloss correlations developed from the pipeloop data.

The purpose of the nomographs is to give a range of power requirements recessary to transport various sizes of muck by pipeline. New commercial steel pipe with a wall roughness of 0.00015 ft. was used. Slurry temperature was set at 20°C. Note that the power requirements are for overcoming pipewall resistance only.

The slurry nomographs are used as follows: Suppose the desired peak muck haulage rate is 350 dry short tons per hour of one inch top size tunnel muck. From Fig. 5-2, select the solids concentration and pipe diameter curve which gives the lowest specific power for overcoming wall friction. In this case a 10 inch pipe carrying 35% solids by weight gives a specific power of 0.39 kw-hr/ton-1000ft or

$$0.39 \times 1.341 \times 350 = Q_{fcs} \cdot \Delta P_{psf} / 550$$

Now throughput $T(tph) = V_{fps} A_{ft^2} \, C_w \cdot S_m \times 62.4 \times 3600/2000$

where S_m = specific gravity of the slurry for solids of specific gravity 2.65.

C_w	S_m
0.25	1.184
0.35	1.219
0.45	1.389

In the selected case the flow rate (Q=VA) is

$$\frac{350 \times 2000}{0.35 \times 1.279 \times 62.4 \times 3600} = 6.961 \text{ ft}^3/\text{sec}$$

TABLE 5-1. PROPERTIES OF SIMULATED TUNNEL MUCK USED IN HYDROTRANSPORT HEADLOSS CORRELATIONS

<u>1" Top Size</u>

Size	%
1" / 3/4	16.34
3/4"/ 1/2"	33.21
1/2"/ 1/4"	21.88
1/4"/4M	4.58
4/8	6.92
8/14	4.06
14/35	3.09
35/65	1.30
65/150	1.51
150/pan	7.12
Total	100.00

<u>1/4" Top Size</u>

Size	%
1/4"/4M	3.31
4/6	12.13
6/10	46.32
10/20	19.09
20/35	8.03
35/65	3.81
65/150	1.66
150/pan	6.65
Total	100.00

1" Top Size		1/4" Top Size
11.6	Weighted Mean Diameter, mm	2.1
2.65	Solids Specific Gravity	2.65
46	No. of Pipeline Data Points	45
$\phi=55.84\psi^{-0.884}$	Headloss Correlation	$\phi=35.79\psi^{-0.512}$
0.98	Coefficient of Correlation	0.86

ϕ and ψ are defined in Ref.(1) p. 6-5.

TABLE 5-1. (Cont'd.)

4 Mesh Top Size

Size	%
4/8	0.23
8/10	0.22
10/14	1.31
14/20	2.07
20/28	8.66
28/35	12.00
35/48	8.12
48/65	8.22
65/100	17.38
100/150	9.09
150/200	2.67
200/325	3.89
325/pan	26.14
Total	100.0

Weighted Mean Diameter, mm	0.28
Solids Specific Gravity	2.65
No. of Pipeline Data Points	178
Headloss Correlation	$\phi=13.50\psi^{-0.946}$
Coefficient of Correlation	0.82

and the mean slurry velocity is $\dfrac{6.961}{\frac{\pi}{4}\cdot\frac{10^2}{12^2}}$ = 12.76 ft/sec.

As long as one stays on the specific power-throughput curves, the mean velocity of flow will be high enough to prevent the solids from depositing on the invert of the pipe.

The pressure required to overcome pipewall friction is

$$\text{Psi/1000 ft.} = \frac{0.39 \times 1.341 \times 350 \times 550}{6.961 \times 144}$$

$$\doteq 100 \cdot 4 \text{psi/1000 ft.}$$

It is interesting to note that if the tunnel muck were crushed to a minus 1/4-inch top size, Fig. 5-3 gives a specific power of 0.36 kw-hr/ton-1000 ft. for 350 tph in a 10-in. diameter pipe at 35% solids by weight.

The approximate locus of minimum operating points is discernable on the nomographs. At these flowrates, deposition of solids is imminent leading to unstable slurry pipeline operation. This locus is considered approximate because of the difficulty in scaling deposition velocities from pipeloop test data to larger pipe diameters. Futhermore, the wide variation in solids loading to the slurry pipeline system will require a selection of fairly high flow velocities.

Fig. 5-2, the nomograph for the coarsest size distribution (nominal 1" top size) is considered valid for all muck sizes greater than about 3/8" top size. The reason for this is that theoretically, an increase in particle size above some minimum value does not increase the power requirements for a constant concentration. For solids of specific gravity 2.65, this minimum size is about 1/8". This is a mean size. The corresponding top size is larger. Within the experimental accuracy experienced in coarse slurry pumping this rule seems to hold as long as the amount of fines passing 200 mesh is not sufficient to produce a "heavy medium" carrier liquid whose viscosity is several times that of clear water.

Obviously, this does not mean that the coarsest size distribution should be chosen. Other factors such as geometric clearances in pump passages and pipelines will limit the top size that can be pumped. Wear is probably the most important aspect. Abrasive wear increases with the kinetic energy imparted to the solid particles by the water. For a stable pipeline operation the velocity of flow must be maintained high enough to keep most of the solids in suspension. Thus the high velocity required to transport coarse solids promotes greater wear on an exponential basis.

On the other hand, a coarse slurry is desirable to minimize crushing, grinding, and de-watering efforts. Thus an optimum size distribution must exist for each slurry pipeline system, but this is beyond the scope of this study. A discussion of a technique for optimization of a size distribution for coal slurry pipelining is given in Ref. 7.

5.3 PUMPS

The maximum spacing and minimum number of centrifugal booster pumps required along the pipeline can be attained by using wear-resistant pumps with high pressure casings to accomodate large abrasive solids at fairly high pressures. This was discussed in detail in the earlier report (1). In the costing of the several case studies sited therein, the pump discharge pressure was limited to 75 psi. Recent discussions with several pump manufacturers suggest that the discharge pressure can be increased to 125 psi or even 150 psi with finer particle slurries. Thus, fewer booster pumps would be required, operating and maintenance costs would be reduced, and presumably, greater reliability would be enjoyed.

Pumps in series can be close-coupled on a single skid or individually connected in modular style. The former design imposes high pressure on the downstream pump and the downstream piping. The advantage of grouping the pumps is the convenience of operation, maintenance, and connection of utilities. The latter arrangement consisting of slurry pipeline modules is more conducive to the progressive construction of a tunnel but separate controls and utilities are necessary.

5.4 JET PUMPS FOR SLURRY TRANSPORT

The jet pump is a device for discharging solids in a liquid under increasing pressure. Jet pump technology is relatively new, most of the technical literature emerging after 1930.

A jet pump is a device in which a jet of fluid (the driving fluid) is used to entrain more fluid. It consists of a nozzle, a suction box, a mixing tube (the throat), and usually, a diffuser on the downstream side. The jet pump operates on fluid dynamic and energy principles. It has no moving parts which accounts for its simplicty and low efficiency.

The jet pump is known by various names depending on its application. When operation as an eductor, the driving fluid, water, is used to entrain additional water to obtain a greater mass flow, but at a lower pressure than that of the driving liquid.

The application under consideration is to induce solids into a pipeline from a low-profile mix tank. The specific application will be discussed in detail later but is mentioned now to confine the discussion on jet pumps to a solids-liquid operation whereby a slurry of excavated muck and water is induced into a pipeline by a jet pump driven either by clear water or a low concentration fine slurry (heavy medium).

5.4.1 Principle of Operation

As the drive jet discharges through a stagnant or slowly moving liquid, mixing results between the driving and entrained liquids. A transfer of momentum accelerates the entrained liquid in the direction of flow of the driving liquid jet. In this process the drive jet transfers energy to the suction liquid. The liquid entrainment occurs in the suction box immediately downstream of the nozzle and the acceleration of the flow through the nozzle results in a high-velocity, low-pressure jet. As the two liquids flow downstream, they spread into the mixing tube.

At the entrance to the mixing tube, the entrained liquid fills the annular space between the driving liquid jet and the wall of the mixing tube. At the mixing tube exit, mixing is complete and both liquids flow forward at the same velocity. The diffuser serves as a head recovery device by converting kinetic energy to pressure energy.

For design purposes the flow and pressure of the drive jet are required to produce a specific suction flow rate and discharge pressure.

5.4.2 Geometry

There are three basic configurations of jet pumps as shown in Fig. 5-5. These are the center-drive type, the side nozzle type, and the annular type. While each differs in the manner in which the drive liquid enters the mixing chamber, the pumping operation remains essentially the same.

Because of its simplicity and more common usage, only the center-drive jet pump will be discussed. (See Fig. 5-6.) Fig. 5-7 shows the head ratio characteristics of liquid-liquid jet pumps versus mass flow ratio. These characteristics have been shown to be substantially the same when solids are entrained up to 25% by weight concentration delivered for sand of specific gravity 2.6. (8). As with normal pump operations a steep characteristic curve is desirable to ensure system stability. Thus, in Fig. 5-7 a reasonably large d/D ratio is recommended which in turn suggests a mass flow ratio of 2 to 3. Fig. 5-8 is a graph of jet pump efficiencies versus head ratio. Note the low order of efficiency ($\not>$38%) and the limited range of d/D for maximum efficiencies.

5.4.3 Design Criteria

Design considerations for jet pumps desire a high mass flow ratio, minimum wear and cavitation possibilities in the mixing tube, high jet pump efficiency, and small pipe size and weight. The mixing tube diameter must also be larger than the largest particle size capable of being entrained (9).

5.4.4 Theory

As with any pump, the characteristics are a head ratio and mass flow ratio. These are defined below.

$$H = \frac{U-V-W}{W-U+I} \qquad (5.1)$$

$$\text{head ratio} = \frac{\text{head gained by suction flow}}{\text{head lost by driving liquid}}$$

where $U = 2B + 2\frac{(\phi B)^2}{1-B} \quad \gamma_w \frac{\left[C_v(\gamma s-\gamma w)+\gamma w\right]}{\gamma^2 su}$

$$\frac{-B(1+\phi)\gamma_w}{\gamma_d} \quad \frac{\left[2B+2\phi B\left[C_v(\gamma_s-\gamma_w)+\gamma_w\right]\right]}{\gamma_{su}} \quad \frac{-B^2(1+\phi)^2\gamma_w k_{mc}}{\gamma_d} \qquad (5.2)$$

$$V = \frac{\phi B^2}{1-B} \cdot \frac{1+k_{su}}{\gamma_{su}} \qquad (5.3)$$

$$W = \frac{B^2(H\phi)^2(k_{dif}-1)}{S_d} \qquad (5.4)$$

$$I = 1 + k_j \qquad (5.5)$$

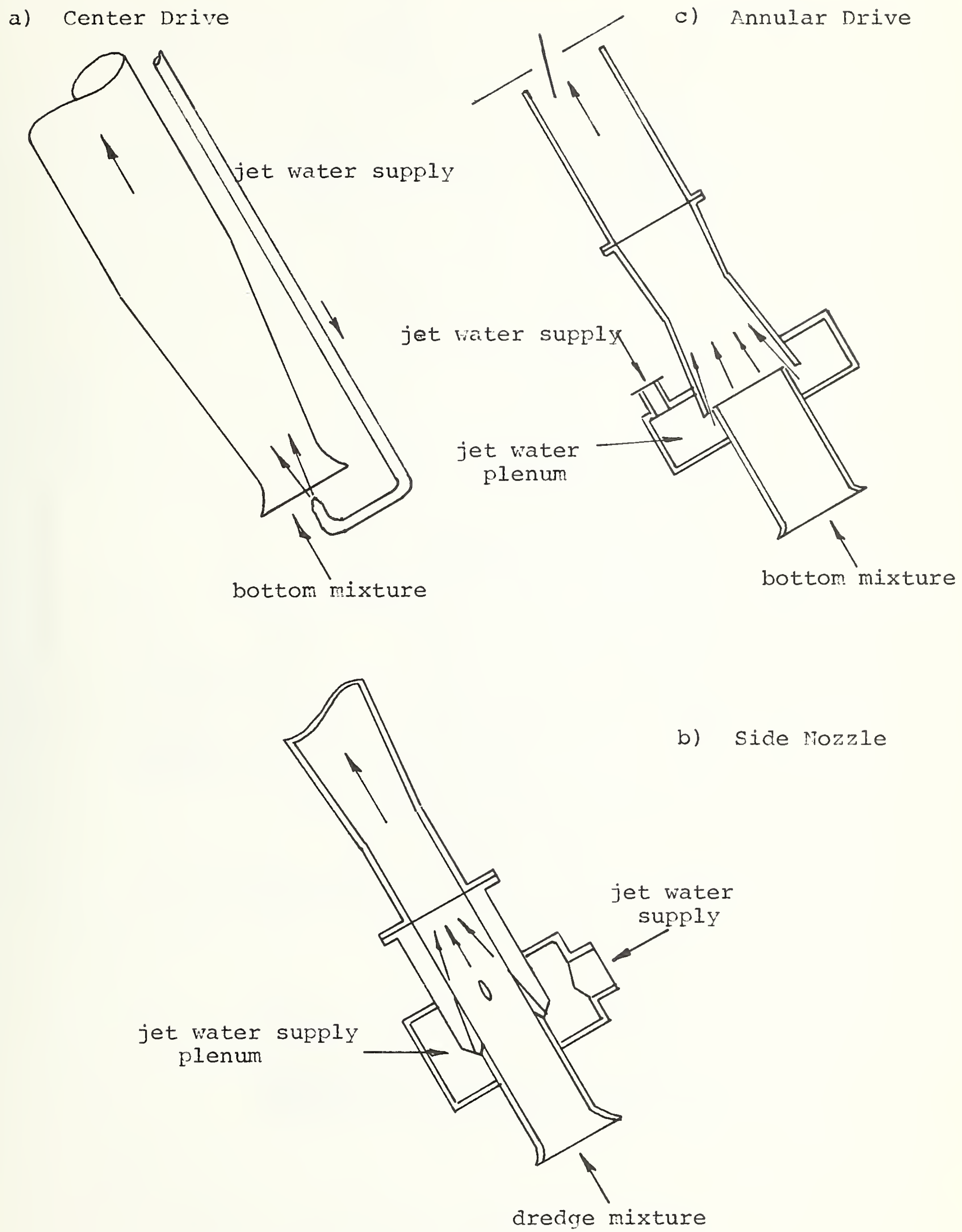


FIG. 5-5. Variations of Jet Pumps in Common Use
Ref. 10.

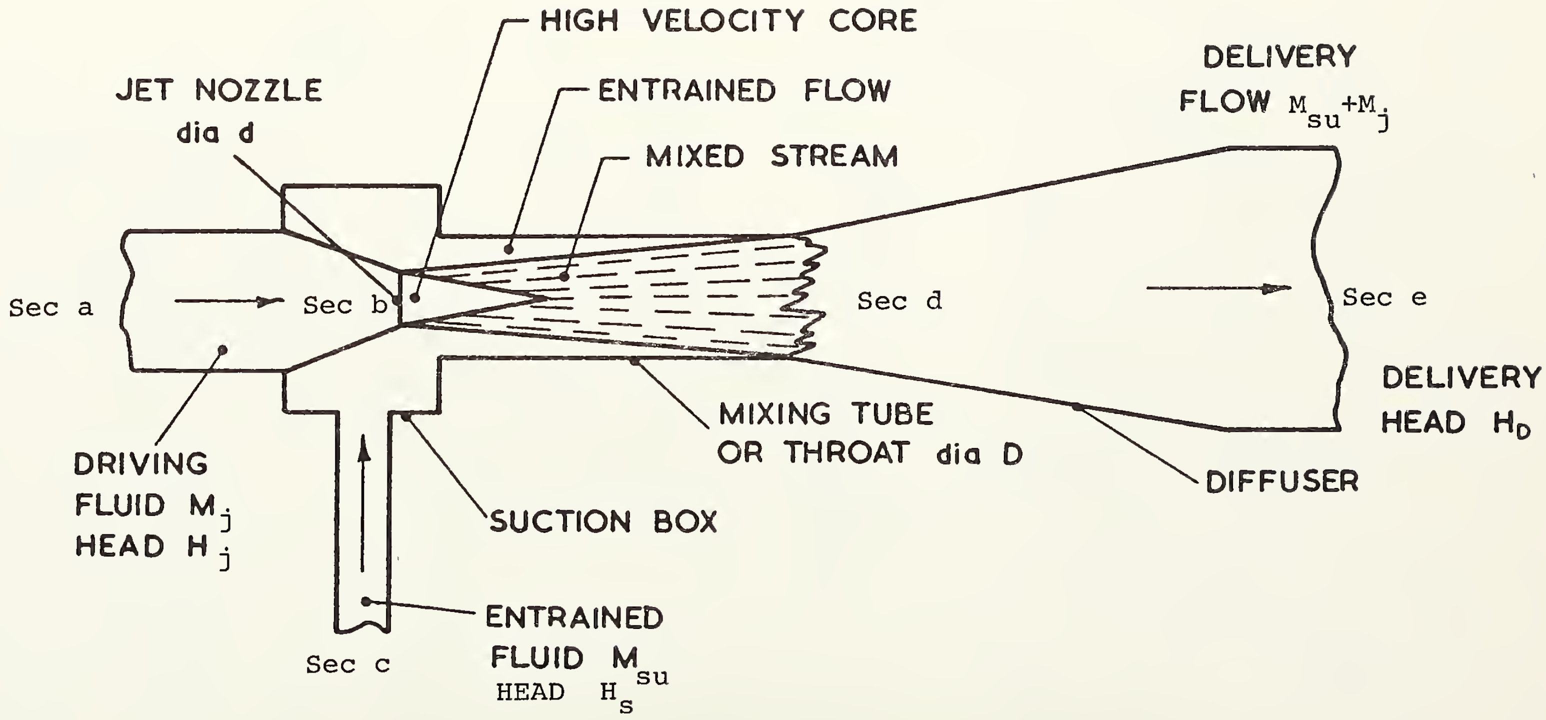


FIG. 5-6. Typical Jet Pump (Ref. 8).

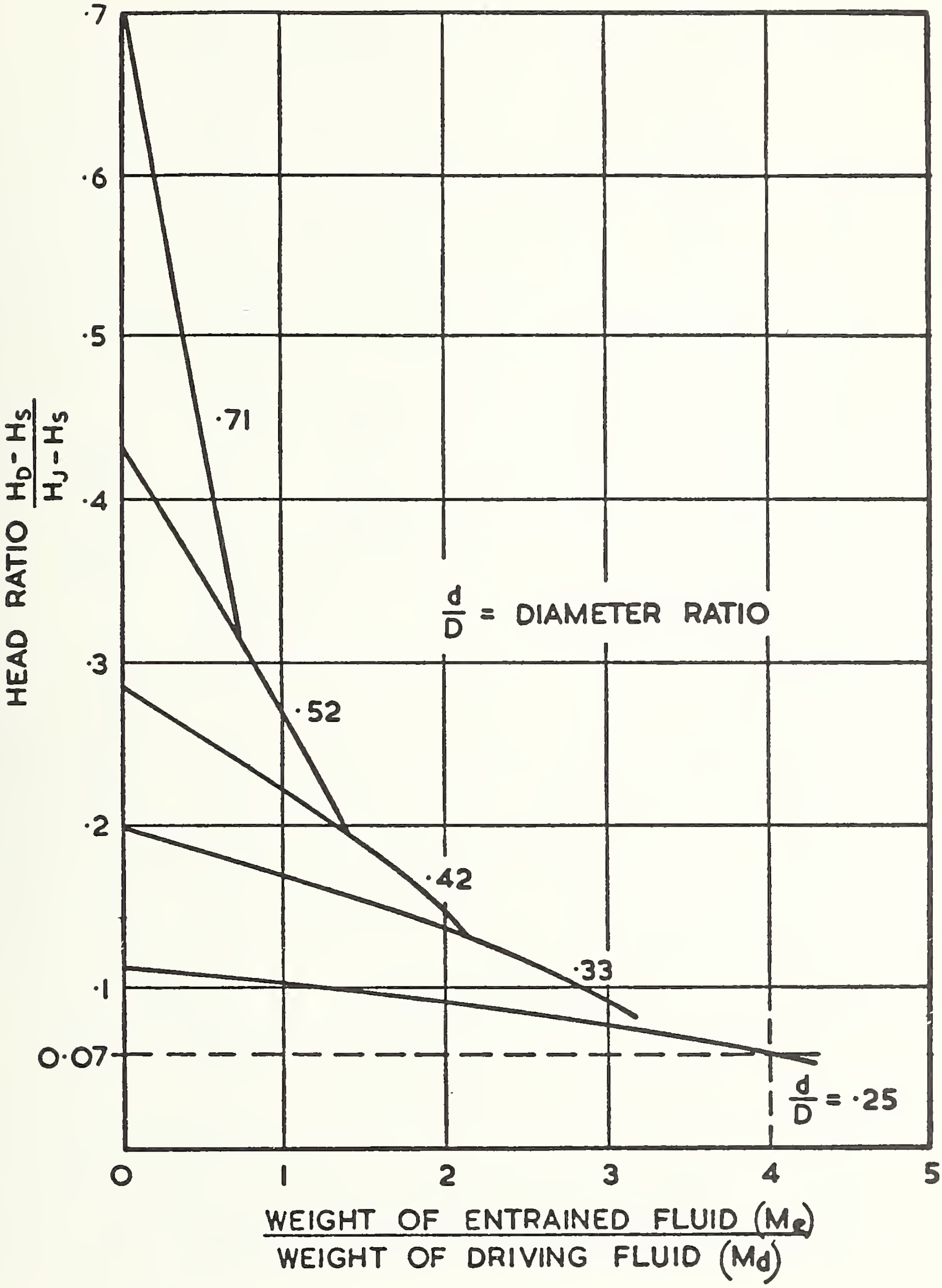


FIG. 5-7. Head Flow Characteristics of a Jet Pump

Ref. 8.

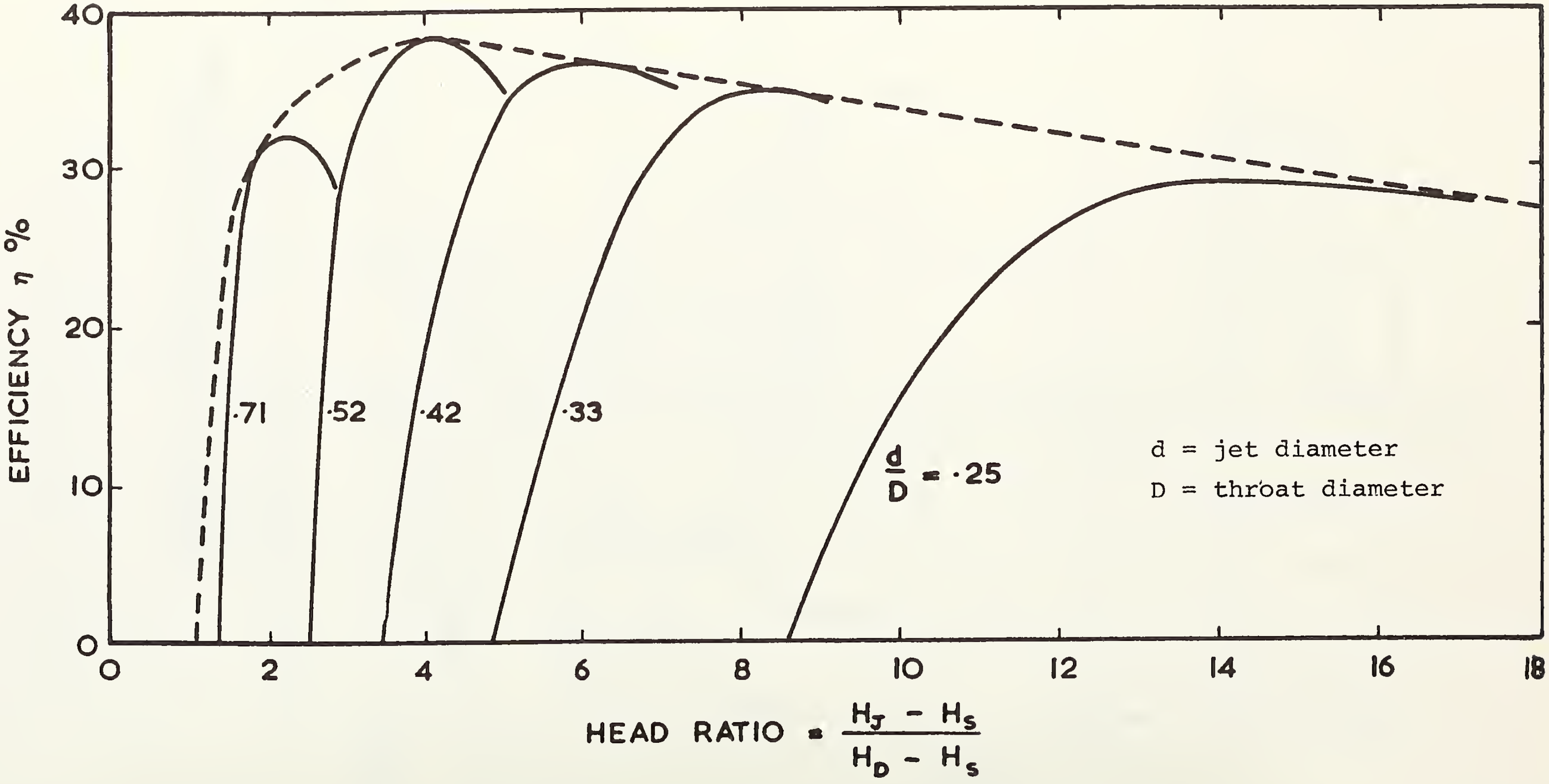


FIG. 5-8. Efficiency vs. Head Ratio for Various Diameter Ratios

Ref. 8.

where B = area ratio = jet area/mixing chamber(tube) area.

C_v = volumetric concentration (fraction) in suction line, volume of solids to volume of slurry

k_{dif} = loss coefficient for diffuser

k_j = loss coefficient for driving nozzle

k_{mc} = loss coefficient for mixing chamber

k_{su} = loss coefficient for suction nozzle

S_d = specific gravity of slurry discharge

γ_w = specific weight of water, lb_f/ft^3

γ_s = specific weight of solids, lb_f/ft^3

γ_d = specific weight of discharge slurry, lb_f/ft^3

γ_{su} = specific weight of suction slurry, lb_f/ft^3

ϕ = flow ratio = M_{su}/M_j where generally, $M = Q\gamma$ lb/sec

Q = total flow rate at suction (su) or at driving jet (j), ft^3/sec.

Experimental data (8,9) suggest that best results are obtained for an area ratio $B = 0.06$ to 0.11 which gives the following loss coefficients:

driving jet or nozzle loss	$= k_j$	$= 0.12$
suction nozzle loss	$= k_{su}$	$= 0.05$
mix chamber loss	$= k_{mc}$	$= 0.094$
diffuser loss	$= k_{dif}$	$= 0.25$

The diffuser angle should be about 5.5° and the mass flow ratio about 3.

To prevent cavitation in the mixing chamber, a critical value of mass flow ratio must not be exceeded. It is given by

$$\phi_c = \sqrt{\frac{\Delta P_b}{\Delta P_a}\left[\frac{1-B}{B}\right]^2\left[\frac{\gamma_{su}}{\gamma_w}\right]^2 \frac{1+k_j-D}{\alpha_s(1+k_{su})}} \qquad (5.6)$$

where ϕ_c = mass flow ratio for incipient cavitation

ΔP_b = pressure head differential $= \left[P_b - P_c\right] / \gamma_w \cdot S_{su}$ (5.7)

ΔP_a $= \left[P_a - P_c\right] / \gamma_w$ (5.8)

P = pressure at cross section denoted: a = jet pipe, b = jet nozzle, c = entrance to mixing chamber.

$$D = \left[\frac{\alpha_j}{\alpha_a}\right]^2 \left[\frac{d_j}{d_a}\right]^4 \qquad (5.9)$$

where α = velocity deflect at section denoted.

A rough estimation of the value of ϕ_c may be obtained from the equation

$$\phi_c = 0.2B^{-1.32} \qquad (5.10)$$

formulated from experimental data produced by Govatos and Zandi (10).

5.5 JET ASSISTED SLURRY PUMPING

As mentioned earlier, there are wide variations between average and peak mucking rates, the ratio going as high as two. Since the maximum efficiency range of a pipeline operation is quite narrow, wide fluctuations in muck output are undesirable. Furthermore, to mix the muck with water and to put it through a centrifugal pump is a difficult task when headroom is limited. Slurry pumps, especially for coarse solids, require a substantial suction head which in turn requires a high-profile mix tank. Without adequate suction head, a centrifugal pump will pull air from vortices set up in a mix tank which leads to reduced pumping capacity and possibly cavitational damage to the pump.

One possible solution to the muck surge problem is to install a jet pump (eductor) in the suction elbow between the mix tank and centrifugal pump as shown in Fig. 5-9. The jet

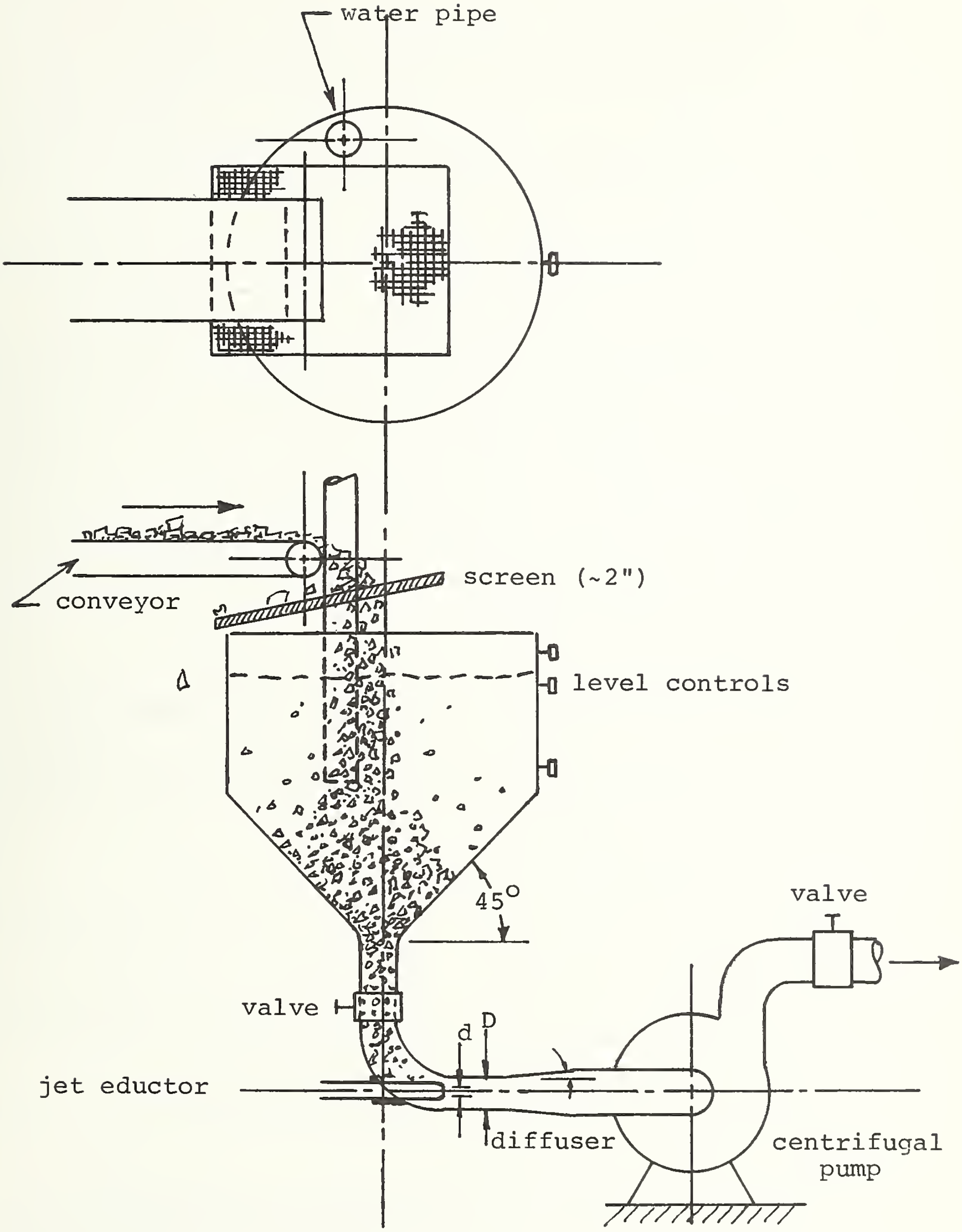


FIG.5-9. Slurry Mix Tank for Muck Haulage by Pipeline

eductor would serve as an auxiliary pump when increased muck loadings developed. It would require only a small water pump and motor, small diameter piping from the main water line and a throttling valve. It would be put into operation manually when the slurry pipeline operator observed high muck throughputs on the conveyor belts from the tunnel boring machine and the crusher. Later, with experience gained, the jet eductor could be controlled by electronic or fluidic sensors on the conveyor belt. The jet eductor would be installed in a flanged large radius suction bend such that it could be easily replaced during routine maintenance periods. Because of the configuration of the installation, the jet eductor would be subject to abrasive wear. While tunneling through hard rock at low advance rates, the jet eductor could be removed to prevent its wearing. While tunneling through softer rock at higher speeds, the jet eductor could be re-installed to assist pumping the higher muck loads.

The jet pump introduces the solids into the pump in a controlled manner thereby reducing throughput surges and maintains a higher level of muck throughput in the slurry pipeline without the necessity of a high-profile mix tank and mechanical agitation.

5.6 PNEUMATIC PIPELINE

Blowing solids through a pipeline can be accomplished in several ways:

1. as a dilute suspension of solids in which the mass flow to air flow ratio (M*) is less than 20 to 1,
2. as a dense suspension for which M*>100, and
3. vertically as a packed bed.

Tunnel muck, being coarse and heavy, permits only a dilute suspension to develop in penumatic pipelines. Thus, only dilute suspensions will be considered in this discussion.

As solids are blown through a pipeline the pressure loss and hence power requirements are the sum of

1. the pressure drop required to accelerate the solids to a uniform velocity somewhat less than the air stream velocity,
2. the pressure drop required for established flow to overcome wall friction in inclined and horizontal pipes and,
3. the pressure drop required to flow through fittings such as bends.

A literature survey was made to determine the availability and reliability of correlations for estimating pressure drops in pipes and fittings and estimating minimum transport velocities. The latest and most extensive reference is a series of course notes entitled, "The Priniciples and Practice of Pneumatic Transport", authored by R. A. Duckworth and presented at a short course at the University of Kentucky in June 1977. His technique is described here and because of its complexity, an example is illustrated.

The restrictions are:

1. dilute phase, mass flow ratio, $M^*<20$,
2. particles relatively spherical in shape, and
3. particle size >600 μm or 0.025 inch

The equations to be used are listed below and followed by a nomenclature:

I. Acceleration Length

$$L_a/D=6\left[(M_s/\gamma_f\sqrt{g}\ D^{2.5})\cdot\sqrt{D/d}\ \cdot\sqrt{\rho^*}\right]^{1/3} \tag{5-11}$$

II. Acceleration Pressure

$$\Delta P_{sa}/\rho_a V_a^2/2 = M^*\cdot A\phi_4(V_a^2/gd\rho^{*2})\cdot A\phi_5(\Theta) \tag{5-12}$$

III. Velocity of the Solids

$$V_s/V_a = (1/2)A\phi_4(V_a^2/gd\rho^{*2}) \cdot A\phi_5(\Theta) \quad (5\text{-}13)$$

IV. Minimum Transport Velocity

$$V_{min}/V_\infty = V_m\phi(d/D) \cdot V_m\phi(\Theta) \cdot M^{*0.3} \quad (5\text{-}14)$$

V. Pressure Drop for Established Flow

$$\frac{\Delta P_m}{(\rho_a V_a{}^2/2)(L/D)} = f_a + f_s + 2(gD/V_a^2)\sin\Theta\left[1 + M^*(V_a/V_s)(1 - 1/\rho^*)\right] \quad (5\text{-}15)$$

where $f_s = F\phi_1(M^*) . F\phi_2(d/D) . F\phi_3(\varepsilon) . F\phi_4(\rho^*) . F\phi_5(\Theta) . F\phi_6(V_a^2/gD)$ (5-16)

VI. Pressure Drop in Bends (Same as Acceleration Pressure)

$$\Delta P_{sb} / \rho_a V_a^2/2 = M^* . A\phi_4(V_a^2/gd\rho^{*2}) . A\phi_5(\Theta) \quad (5\text{-}17)$$

Nomenclature

L_a = Length of pipe in feet required to accelerate solids from zero velocity at the feeder to terminal velocity.

D = inside diameter of pipe in feet

M_s = mass flow rate of solids, lb/sec

γ_f = specific weight of air (0.075 lb/ft^3 at sea level, 0.06 lb/ft^3 at CSM test site).

g = gravitational acceleration, 32.2 ft/sec^2

d = representative particle size, feet. A weighted mean particle size can be used.

ρ^* = density of solids, ρ_s(lb$_m$/ft^3), density of air ρ_a(lb$_m$/ft^3)

ΔP_{sa} = pressure in lb/ft^2 required to accelerate solids to terminal velocity.

V_a = mean velocity of airstream, ft/sec

M^* = mass flow rate of solids, M_s(lb/sec) mass flow rate of air, M_a(lb/sec)

$A\phi_4$ = acceleration function relating pressure drop to densimetric Froude No. $V_a^2/gd\rho^*$ Note: all functions are dimensionless.

$A\phi_5$ = acceleration function relating pressure drop to pipe inclination measured upward from horizontal,

V_s = mean velocity of solids, ft/sec

V_{min} = minimum transport velocity, ft/sec

V_∞ = terminal settling velocity of mean representative particle in an infinite fluid, ft/sec

$V_m\phi$ = function relating minimum transport velocity with representative particle size to pipe diameter ratio (d/D) or to pipe inclination Θ, measured upward from horizontal.

ΔP_m = total pressure of solids-gaseous mixture required for established flow, lb/ft^2

L = length of pipeline over which pressure drop is measured, ft

f_a = Darcy-Weisbach friction factor for airflow
f_s = Darcy-Weisbach friction factor for solids phase
$F\phi$ = function relating friction factor to parameters M^*, d/D, etc.
ΔP_{sb} = pressure drop in lb/ft^2 for re-accelerating solids around bends

There are eleven graphs accompanying the analysis; two functional relationships for the acceleration pressure, two functional relationships for the minimum transport velocity and six functional relationships for the solids friction factor. These are tabulated below and are shown at the end of the chapter as Figures 5-10 through 5-20.

Functions	Symbol	Fig.No.
Acceleration Pressure	$A\phi_4$	5-10
	$A\phi_5$	5-11
Minimum Transport Velocity	$V_m\phi(d/D)$	5-12
	$V_m\phi(\Theta)$	5-13
Solids Friction Factor	$F\phi_1(M^*)$	5-14
	$F\phi_2(d/D)$	5-15
	$F\phi_3(\varepsilon)$	5-16
	$F\phi_4(\rho^*)$	5-17
	$F\phi_5(\Theta)$	5-18
	$F\phi_6(V_a^2/gD$	5-19
Terminal Settling Velocity	V_∞	5-20

Example

An illustrative example is provided here based upon some nominal values of variables encountered in the CSM pneumatic pipeline loop. These will be expanded in a later report.

Assume the following data:

1. M^* = 10 which is approximately 118 tons per hour in a

horizontal 10-inch diameter pipe carrying 6545 cfm at a **mean velocity** of 200 ft/sec.

$$\therefore M^* = \frac{118 \times 2000}{0.06 \times 6545 \times 60} = 10$$

2. Solids specific gravity = 2.65

$\therefore\ \rho s = 2.65 \times 62.4/32.2\ \ lb_m/ft^3$

$\rho a \simeq 2 \times 10^{-3}\ lb_m/ft^3$ (adjusted for summer temperatures at 6000 ft elevation)

$\rho^* = \rho s/\rho a = 2568$

3. Assume particle size (weighted mean) is 1 in.

   hence $d/D = 1/10$

4. Assume pipe length is 565 ft. with four horizontal bends.

I. Acceleration Length (Eq. 5-11)

$$L_a = 6 \times \frac{10}{12}\left[\frac{118 \times 2000 \times 12^{2.5}}{3600 \times 0.06\sqrt{32.2}\ \ 10^{2.5}}\sqrt{10 \times 2568}\right]^{1/3} = 182 \text{ ft.}$$

For half-loading (59tph) L_a=145 ft.

For 118tph of 2 in. particles, L_a=162 ft.

For coarse solids, the acceleration length is independent of air velocity and viscosity.

II. Acceleration Pressure (Eq. 5-12)

Assuming horizontal pipe, $\Theta=0$ and $A\phi_5(\Theta)=1.0$

$V_a^2/gd\rho^{*2}=200^2/(32.2 \times \frac{1}{12} \times 2568^2) = 2.26 \times 10^{-3}$

Fig. 5-10 gives $A\phi_4(V_a^2/gd\rho^*) = 0.65$ and

$\Delta P_{sa} = 10 \times 0.65\ 0\ 1.0 \times 2 \times 10^{-3} \times 200^2/2/144 = 1.81$psi

For half loading $\Delta P_{sa} = 0.91$psi.

III. Velocity of the Solids (Eq. 5-13)

$A\phi_5(\Theta) = 1.0$ for horizontal pipe

$A\phi_4(V_a^2/gd\rho^{*2}) = 0.65$ as above

$\therefore V_s = \frac{1}{2} \times 200 \times 0.65 \times 1.0 = 65$ ft/sec

For a 2" particle, $Va^2/gd\rho*^2 = 1.13 \times 10^{-3}$

and $A\phi_4(Va^2/gd\rho*^2) = 0.50$ from Fig. 5-10

hence $V_s = 100 \times 0.5 \times 1.0 = 50$ ft/sec.

Note that these velocities are 1/3 to 1/4 of the mean air velocity and are independent of solids loading.

IV. Minimum Transport Velocity (Eq. 5-14)

for d/D = 0.1 $V_m\phi(d/D)$ from Fig. 5-12 gives 0.5

for $\Theta = 0^\circ$, $V_m\phi(\Theta) = 1.0$ from Fig. 5-13

The terminal settling velocity V_∞ is required. Assume kinematic viscosity of air ν_a, adjusted for site conditions = 1.8×10^{-4} ft^2/sec. The conventional drag coefficient-particle Reynolds number relationship for spheres is used. See Fig. 5-20. While adjustments can be made here for non-sphericity, for expediency, none are made here. Non-spherical particles would be expected to have lower terminal velocities.

$$\frac{d}{\left[\dfrac{3\nu_a^2}{4g(\rho^*-1)}\right]^{1/3}} = \frac{1/12}{\left[\dfrac{3 \times 1.8^2 \times 10^{-8}}{4 \times 32.2 \times 2567}\right]^{1/3}} = 1253$$

From Fig. 5-20, $$\frac{V_\infty}{\left[4g\nu_a(\rho^*-1)/3\right]^{1/3}} = 45$$

and $V_\infty = 45(4 \times 32.2 \times 1.8 \times 10^{-4} \times 2567/3)^{1/3} = 122$ ft/sec

$V_{min} = 122 \times 0.5 \times 1.0 \times 10^{0.3} = 122$ fps

For half loading $V_{min} = 122 \times 0.5 \times 5^{0.3} = 99$ fps.

V. Pressure Drop for Established Flow (Eq. 5-15)

for $M^* = 10$	$F\phi_1(M^*) = 7.7$	from Fig. 5-14
5	4.6	
$d/D = 1/10$	$F\phi_2(d/D) = 1.0$	from Fig. 5-15

Note: $F\phi_2 = 1.0$ for all particle sizes>3/4 in.

$\varepsilon = 0.75$(est.);	$F\phi_3(\varepsilon) = 1.2$	from Fig. 5-16
$\rho^* = 2568$;	$F\phi_4(\rho^*) = 1.0$	from Fig. 5-17
$\Theta = 0$;	$F\phi_5(\Theta) = 1.0$	from Fig. 5-18

$$\frac{V_a^2}{gD} = \frac{200^2 \times 12}{32.2 \times 10}\ ; \qquad F\phi_6(V_a^2/gD) = 0.002 \quad \text{from Fig. 5-19}$$

Hence $f_s = 7.7 \times 1.0 \times 1.2 \times 1.0 \times 1.0 \times 0.002 = 0.01848$

For half loading ($M^* = 5$), $f_s = 0.0185 \times \frac{4.6}{7.7} = 0.01104$

For friction factor of airflow, assume new commercial steel pipe with wall roughness (k) = 0.00015 ft.

Hence, relative roughness = $k/D = 0.00015 \div (10/12) = 0.00018$

Reynolds No. = $VD/\nu = 200 \times \frac{10}{12} \times 10^4/1.8 = 9.26 \times 10^5$

From the Moody-Stanton diagram (see any Fluid Mechanics text) $f_a = 0.0145$

For horizontal flow

$$\frac{P_m}{(\rho a Va^2/2)(L/D)} = fa + fs \text{ and}$$

$$P_m = (0.0145 + 0.0185)\ 2 \times 10^{-3} \times \frac{200^2}{2} \times \frac{565}{10} \times \frac{12}{144}$$

$$= 2.73 + 3.48 = 6.21 \text{ psi}$$

For half loading ($M^* = 5$, 59 tph)

$$\Delta P_m = 2.73 + (3.48 \times \frac{0.01104}{0.01845}) = 2.73 + 2.08 = 4.81 \text{ psi}$$

VI. Pressure Drop in Bends (Eq. 5-17)

Assume that 4 bends exist in the horizontal plane , two @ 90^o, one @ 30^o, one @ 60^o. The pressure drop correlation presented here does not take the bend geometry into consideration, only that an elbow impedes the velocity of the particles. By assuming that the bend stops the particles completely, Eq. 5-12 can then be used to compute the pressure to re-accelerate the solids. Since the particles do not come to rest, this analysis should overestimate the pressure drop in the bend. The bends are combined as three 90^o bends.

$A\phi_5(\Theta) = 1.0$ for $\Theta = 0$

$A\phi_4(V_a^2/gd\rho^{*2}) = 0.65$ as before

$$\therefore \Delta P_{sb} = 2 \times 10^{-3} \times \frac{200^2}{2} \times 10 \times 0.65 \times 1.0 \times \frac{1}{144} \times 3(\text{bends})$$

$= 5.42$ psi

Total Pressure Drop in the Pipeline (psi/565 ft.)

Acceleration Pressure	= 1.81
Pressure drop due to air flow	= 2.73
Pressure drop due to solids flow	= 3.48
Pressure drop in 3 bends	= 5.42
Total pressure drop	13.44 psi

Total horsepower requirement assuming a combined blower-motor efficiency of 70% is

$$\frac{Q.\Delta P}{550(\text{effic})} = \frac{6545 \times 13.44 \times 144}{60 \times 550 \times 0.7} = 548 \text{ hp}$$

It is evident that the high power consumption of a pneumatic pipeline is the high flowrate necessary to keep the solids in suspension.

A few comments on the foregoing theory are in order. The velocity of the solids is computed as less than the minimum transport velocity. Actual data points are not shown on the eleven functional curves given here. Had they been included, Fig. 5-12 would show a substantial scattering which suggests inaccuracies either in the data or in the experimental procedures. Also, the pressure drops computed for the bends appear to be too high.

In conclusion, it is apparent that the theory of pneumatic flow of solids is as least as complex as that of slurry flow.

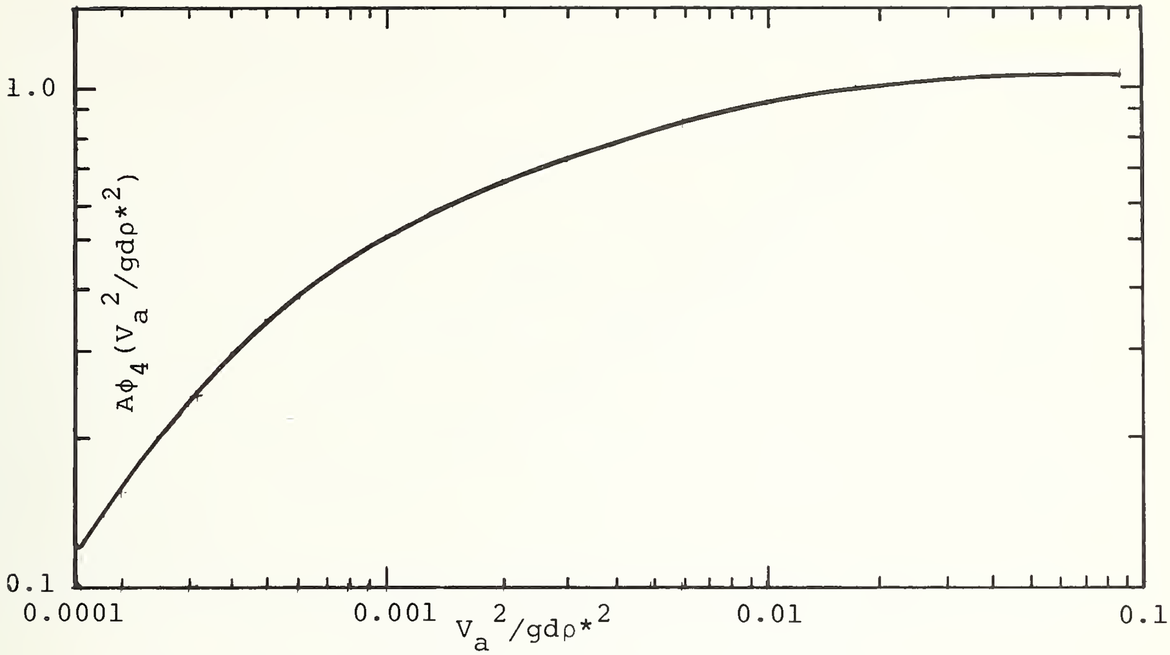


FIG. 5-10 Acceleration Pressure Function-Froude Number

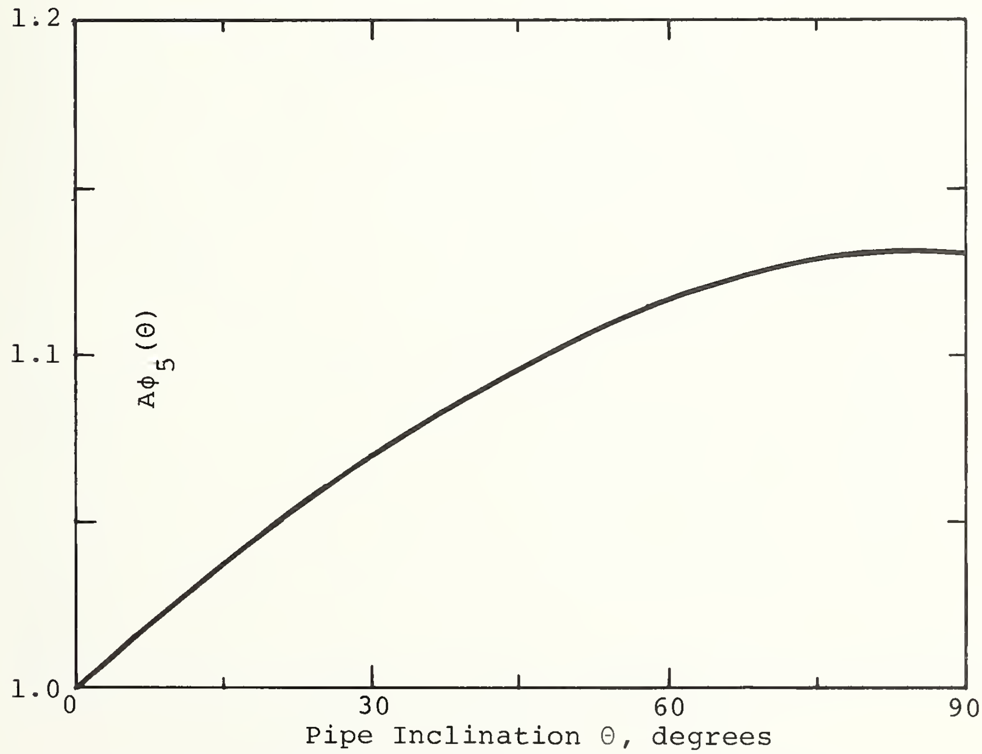


FIG. 5-11 Acceleration Pressure Function-Pipe Slope

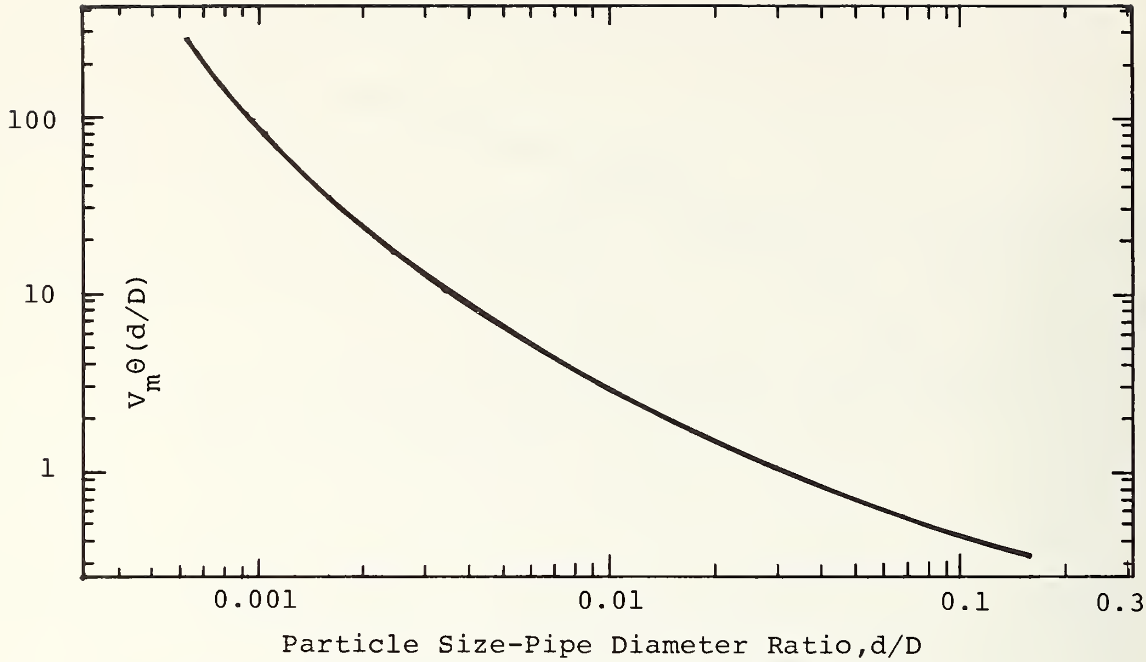


FIG. 5-12 Minimum Velocity Function-Size Ratio

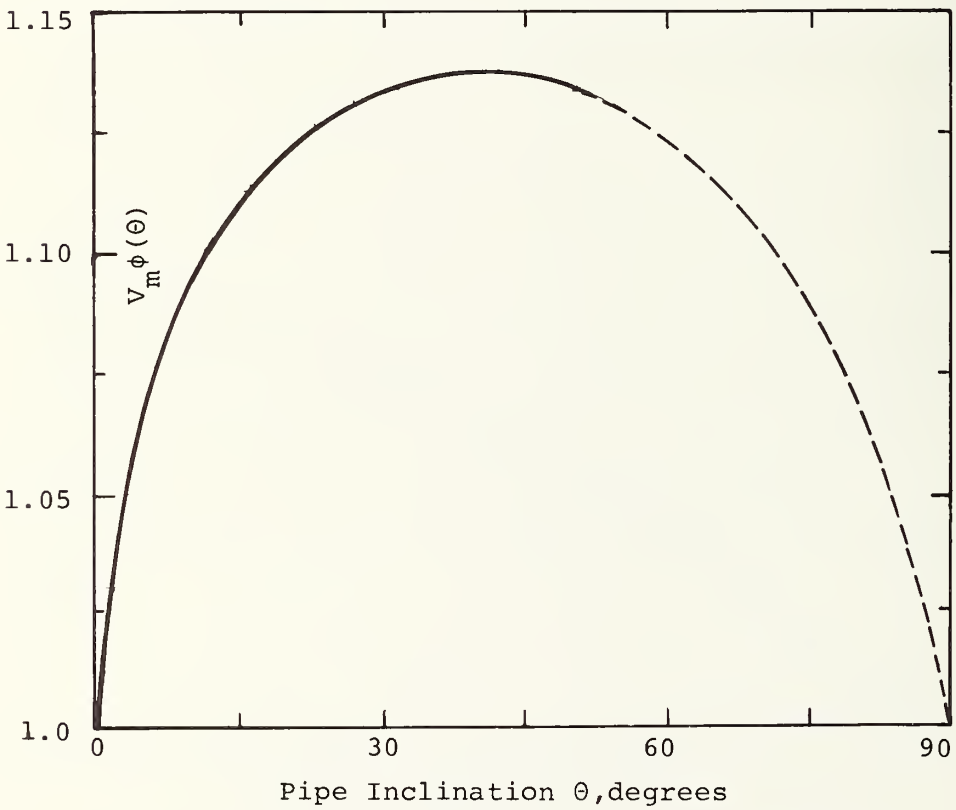


FIG. 5-13 Minimum Velocity Function-Pipe Slope

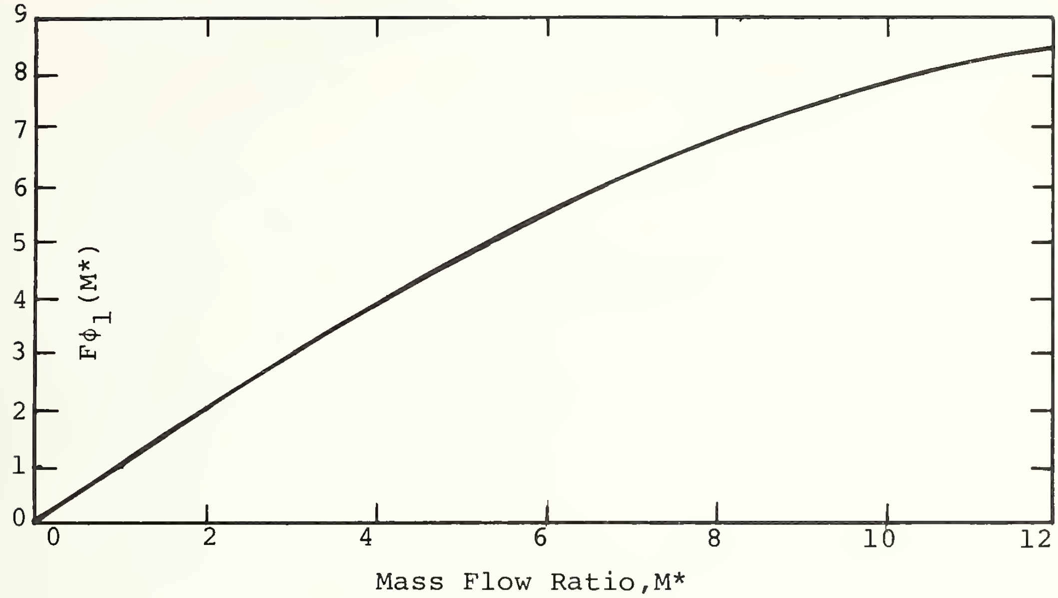


FIG. 5-14 Friction Factor Function-Mass Flow Rate

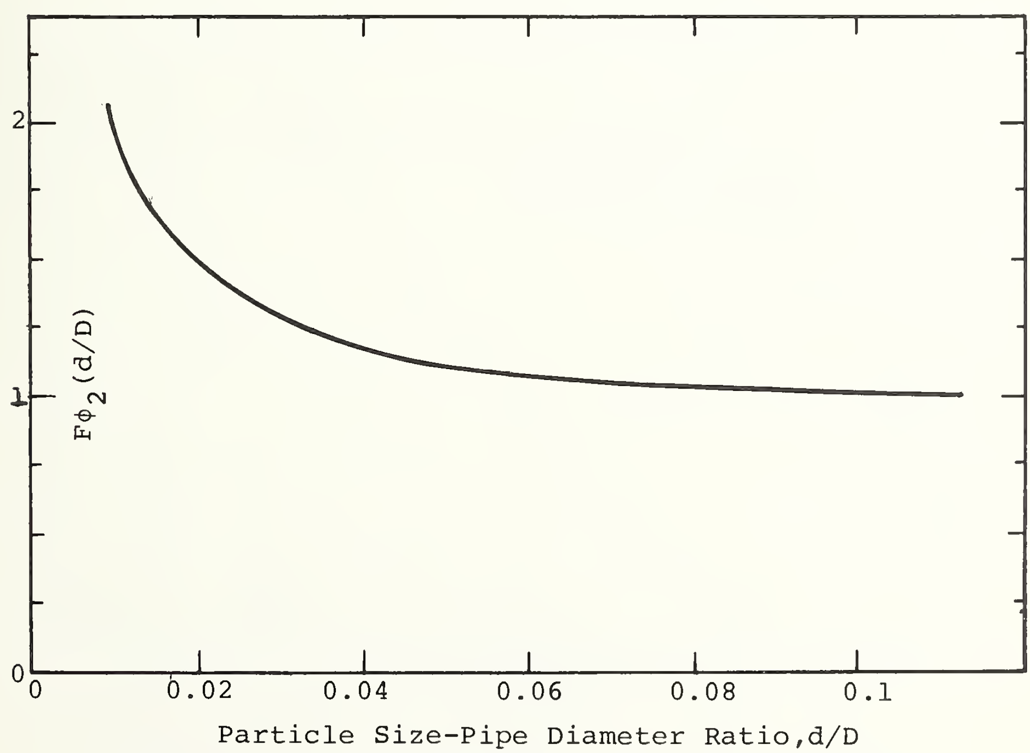


FIG. 5-15 Friction Factor Function-Size Ratio

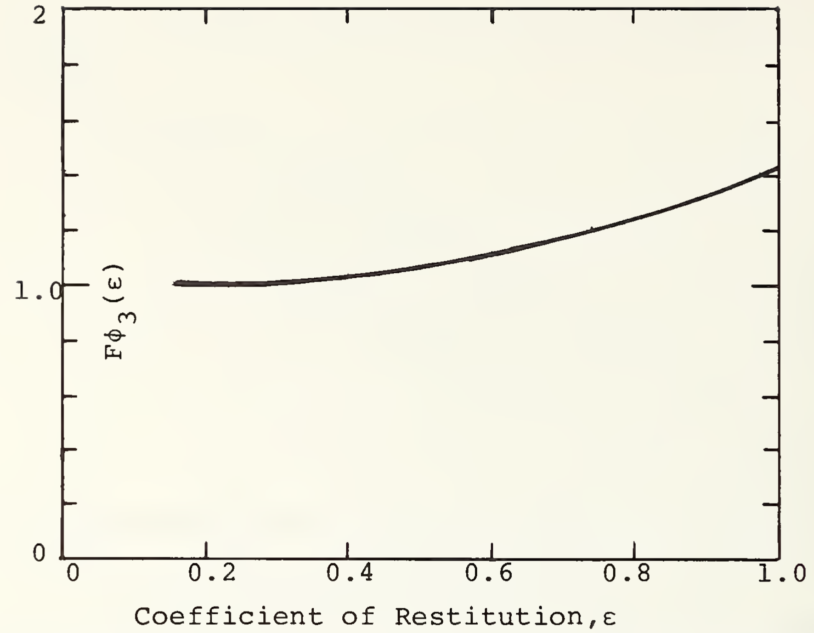


FIG. 5-16 Friction Factor Function-Coefficient of Restitution

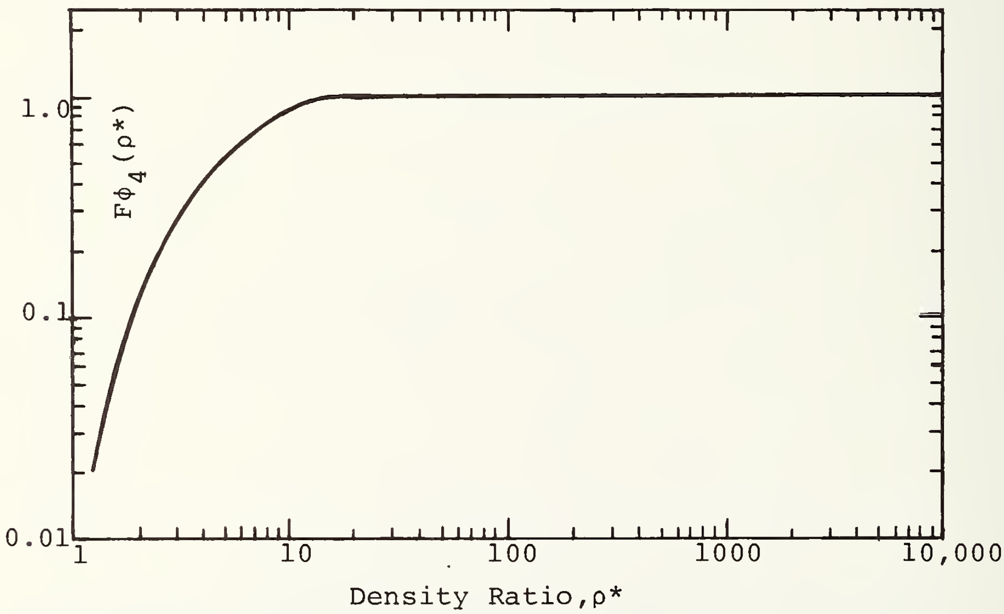


FIG. 5-17 Friction Factor Function-Density Ratio

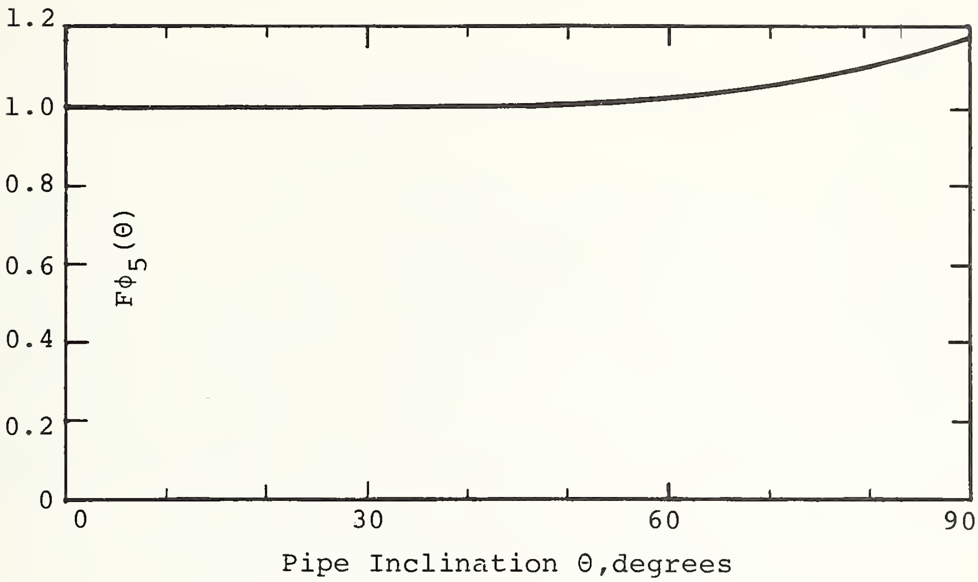


FIG. 5-18 Friction Factor Function-Pipe Slope

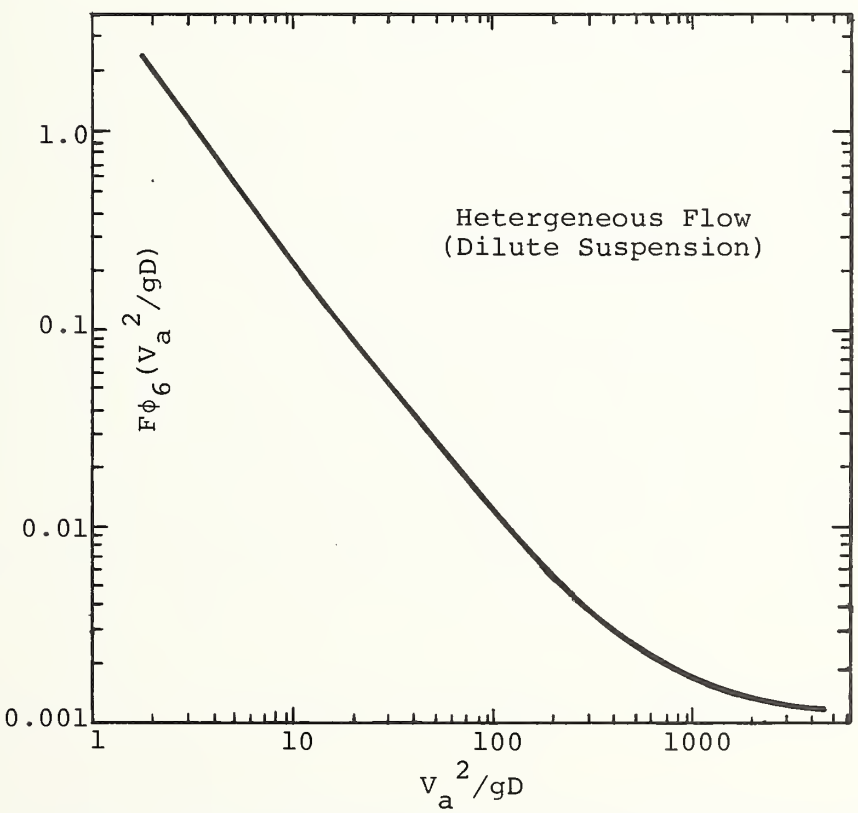


FIG. 5-19 Friction Factor Function-Froude Number

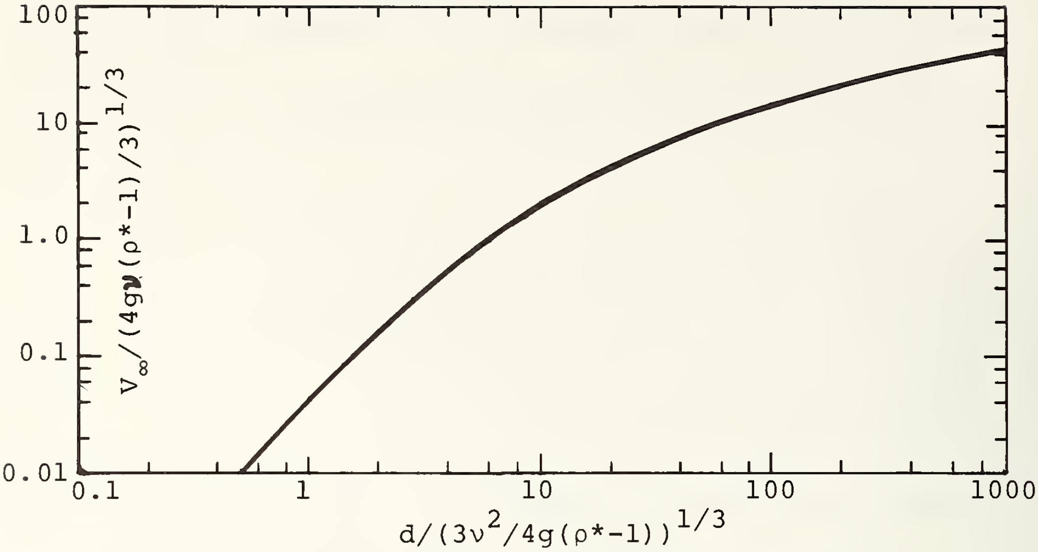


FIG. 5-20 Settling Velocity Function

6. DEWATERING TUNNEL MUCK SLURRIES

The advantages of a slurry pipeline for muck haulage are principally high volume throughputs for small space requirements. The principal disadvantage is the carrier vehicle, water which may have to be purchased, and in all probability, treated before final discharge. Eventually, it is usually mandatory that the solids be separated from the liquid phase. For coarse particle slurries, this presents no problem. For fine particle slurries, dewatering is costly. Normally, tunnel muck is considered to have no inherent worth, thus dewatering is an undesirable cost, and of a very low priority. Recent work (11) suggests this thinking may change.

Regardless of how coarse the muck may be generated, fines are produced by virtue of handling on conveyor belts, mixing in slurry tanks, pumping, pipelining, and dewatering.

In the August, 1974 report by the authors (1), emphasis was placed on dewatering componentry consisting of hydrocyclones, screens (vibrating and sieve bends) and thickeners (conventional and tray). In the present study, emphasis was directed toward rubber-lined screens for increased life, inclined gravity thickeners with flocculant systems and high surface areas for reduced operating costs and space requirements, and hydrocyclones with rubber wedge valves in the apex to handle wider ranges of feed conditions. Details on these components are given in Appendix C.

Both studies examined the dewatering of a coarse slurry and a fine slurry with the following screen size distributions discharging from the slurry pipeline:

Coarse Slurry		Fine Slurry	
Mesh	% between	Mesh	% between
1"/ 1/2"	22.5		
1/2"/4M	22.0		
4/8	12.0	4/8	9.0
8/14	6.5	8/14	11.5
14/28	3.5	14/28	20.0
28/48	4.0	28/48	15.0
48/100	4.0	48/100	13.0
100/200	7.0	100/200	12.5
200/pan	18.5	200/pan	19.0
	100.0		100.0

These approximate size distributions were examined for three throughputs. The six cases are listed on the following pages as

Fig. 6-1. General Flow Sheet for Dewatering Tunnel Muck

Table 6-1. Case 1, 425 tph, 33% solids by weight, coarse muck.

Table 6-2. Case 2, 382 tph, 45% solids by weight, coarse muck.

Table 6-3. Case 3, 200 tph, 45% solids by weight, coarse muck.

Table 6-4. Summary of Dewatering Equipment and Price, coarse muck.

Table 6-5. Case 1A, 425 tph, 33% solids by weight, fine muck.

Table 6-6. Case 2A, 382 tph, 45% solids by weight, fine muck.

Table 6-7. Case 3A, 200 tph, 45% solids by weight, fine muck.

Table 6-8. Summary of Dewatering Equipment and Price, fine muck.

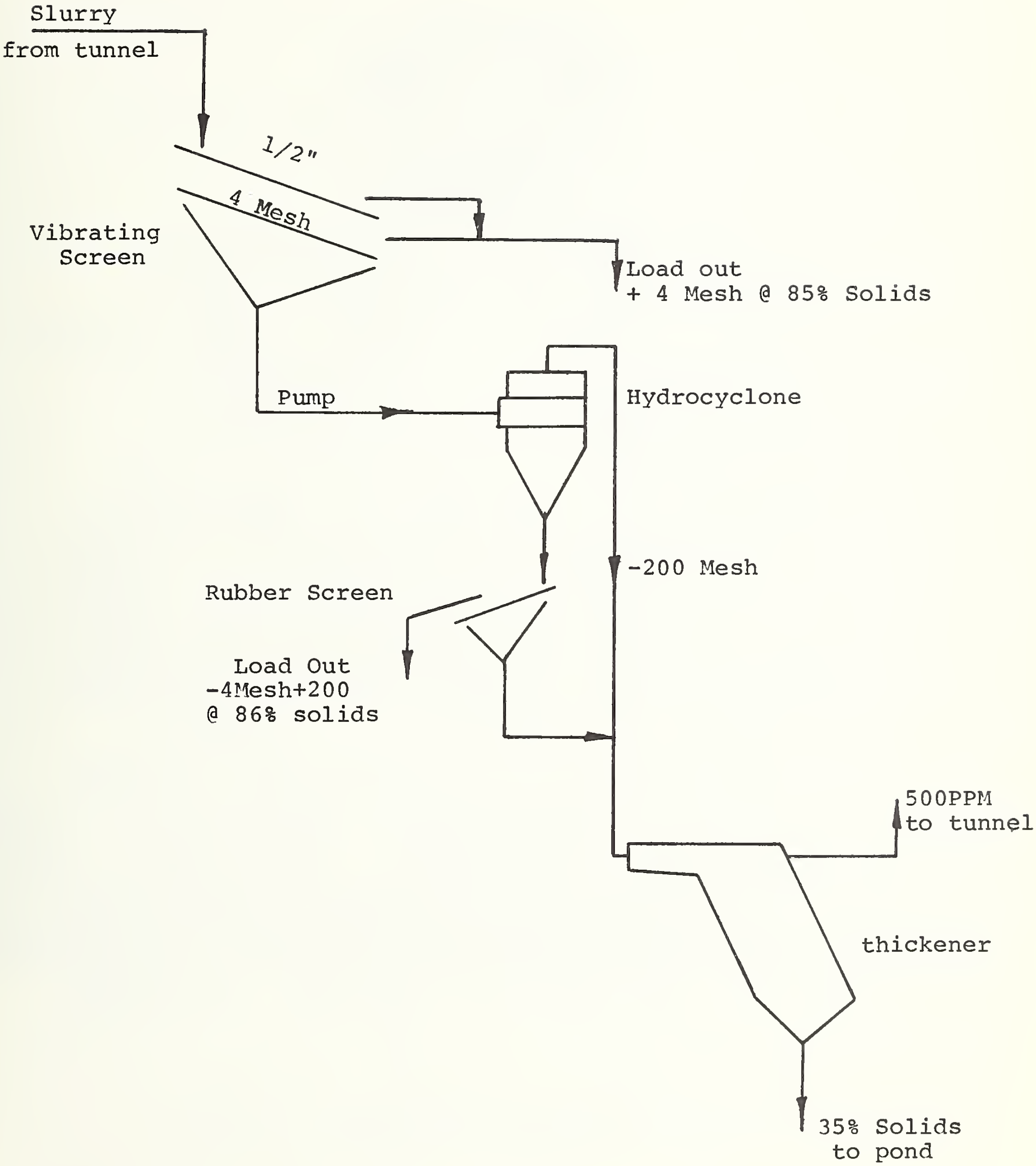


FIG. 6-1. General Flow Sheet for Dewatering Tunnel Muck

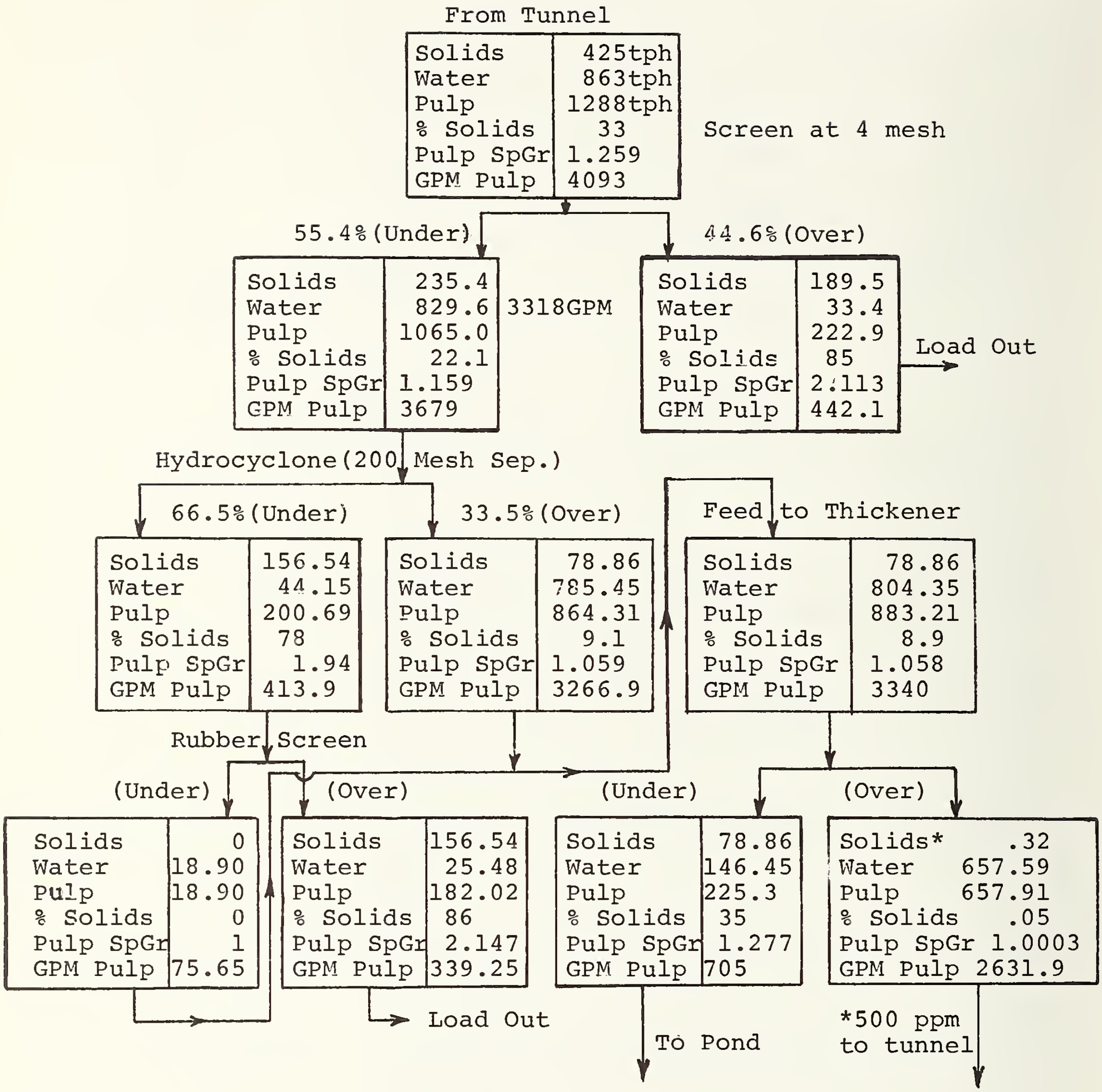


TABLE 6-1. CASE NO. 1, 425TPH, 33% SOLIDS

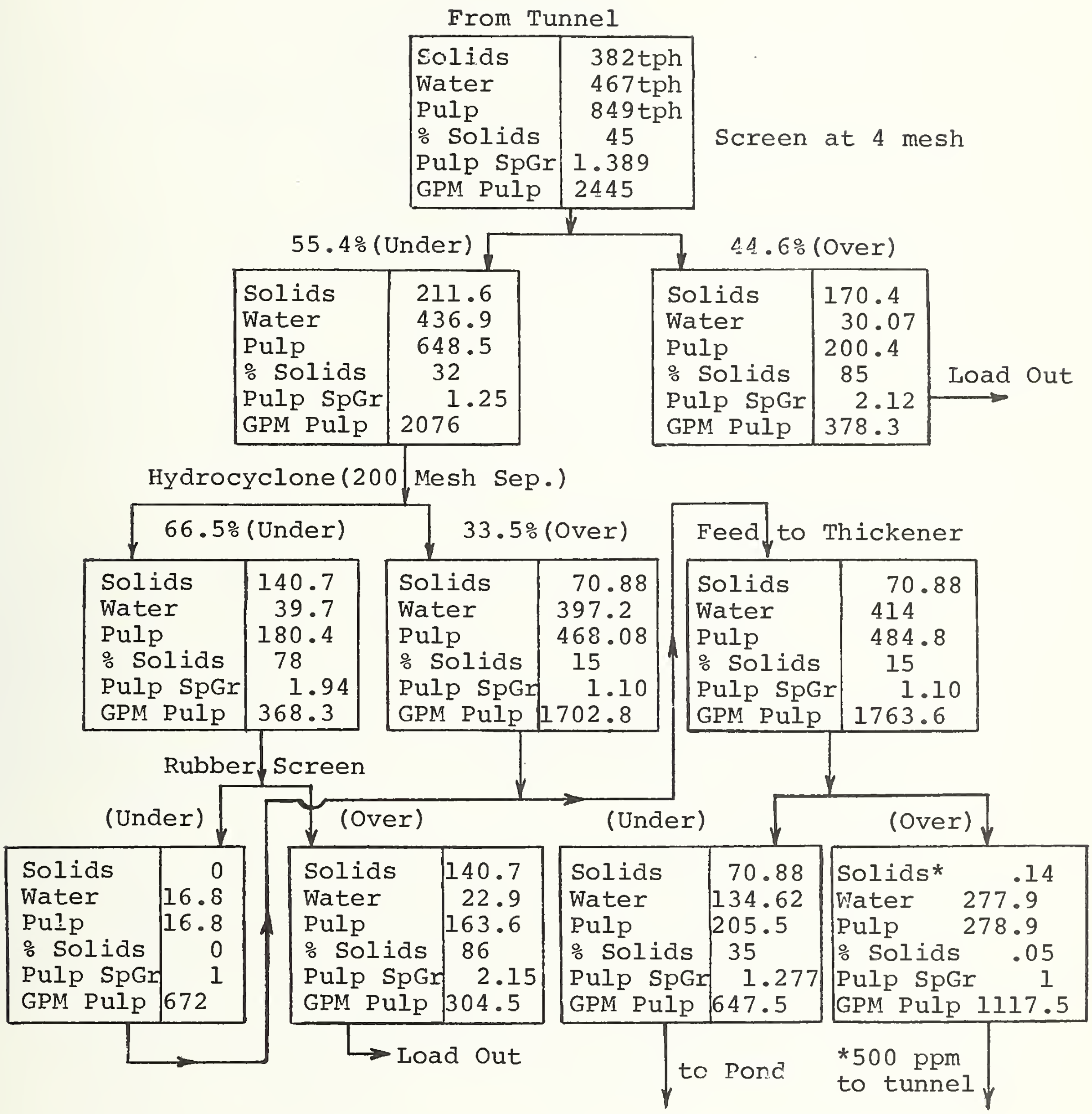


TABLE 6-2. CASE NO. 2, 382TPH, 45% SOLIDS

From Tunnel

Solids	200tph
Water	245tph
Pulp	445tph
% Solids	45
Pulp SpGr	1.389
GMP Pulp	1280

Screen at 4 mesh

55.4%(Under)

Solids	110.8
Water	231.9
Pulp	342.7
% Solids	32.3
Pulp SpGr	1.25
GPM Pulp	1097

44.6%(Over)

Solids	89.2
Water	15.71
Pulp	104.9
% Solids	85
Pulp SpGr	2.12

Load Out

Hydrocyclone(200 Mesh Sep.)

66.5%(Under)

Solids	73.68
Water	20.78
Pulp	94.46
% Solids	78
Pulp SpGr	1.94
GPM Pulp	190.9

33.5%(Over)

Solids	37.11
Water	211.12
Pulp	248.23
% Solids	15
Pulp SpGr	1.10
GPM Pulp	903

Feed to Thickener

Solids	37.11
Water	219.91
Pulp	257.02
% Solids	14
Pulp SpGr	1.10
GPM Pulp	935

Rubber Screen

(Under)

Solids	0
Water	8.79
Pulp	8.79
% Solids	0
Pulp SpGr	1
GPM Pulp	35.17

(Over)

Solids	73.68
Water	11.99
Pulp	85.67
% Solids	86
Pulp SpGr	2.15
GPM Pulp	159.44

Load Out

(Under)

Solids	37.11
Water	68.9
Pulp	106.03
% Solids	35
Pulp SpGr	1.277
GPM Pulp	334.08

tc Pond

(Over)

Solids*	.07
Water	151.01
Pulp	151.01
% Solids	.05
Pulp SpGr	1
GPM Pulp	

*500 ppm
to tunnel

TABLE 6-3. CASE NO. 3, 200TPH, 45% SOLIDS

TABLE 6-4. EQUIPMENT SELECTION & PRICE

Coarse Muck Slurry Transportation and Dewatering

	Case 1	Case 2	Case 3
	425tph, C_w=33%	382tph, C_w=45%	200tph, C_w=45%
Vibrating Screen	1-6x16ft.Ripl-Flo 20 HP $18,050	1-6x14ft.DD Ripl-Flo, 15HP $16,500	1-4x14ft.DD Ripl-Flo,10HP $10,500
Hydrocyclone	1-Model 1036A $7,650	2-S830A,could use 1, 2@$4850 $9,700	1-S836A $7,230
Rubber Screen	1-4x10ft,150tph 5HP $19,000	1-4x10ft, 5HP $19,000	1-4x8ft, 5HP $18,000
Thickener	2-Model 2500/45 10HP,2@$63,000 $126,000	1-2500/45 3HP $63,000	1-1700/45 3HP $52,500
Pump - Motor	1-12x10x17in.CW 75HP motor $4819	1-12x10x17in.CW 60HP motor $4620	8x6x17in.CW 30HP motor $3156
Total Cost	$175,519	$112,820	$91,386
Unit Cost	$413/tph	$295/tph	$457/tph
Installed Horsepower	110	83	48

From Tunnel

Solids	425tph
Water	863tph
Pulp	1288tph
% Solids	33
Pulp SpGr	1.26
GPM Pulp	4093

Hydrocyclone(200 Mesh sep.)

81%(Under)

Solids	344.3
Water	97.1
Pulp	441.3
% Solids	78
Pulp SpGr	1.94
GPM Pulp	9103

19%(Over)

Solids	80.8
Water	765.9
Pulp	846.7
% Solids	9.5
Pulp SpGr	1.06
GPM Pulp	3196

Rubber Screen

(Under)

Solids	0
Water	41.1
Pulp	41.1
% Solids	0
Pulp SpGr	1
GPM Pulp	164.3

(Over)

Solids	344.3
Water	56.1
Pulp	400.3
% Solids	86
Pulp SpGr	2.15
GPM Pulp	745

Load Out

Thickener

Solids	86.8
Water	807.0
Pulp	893.7
% Solids	9.7
Pulp SpGr	1.06
GPM Pulp	3373

(Under)

Solids	86.8
Water	161.2
Pulp	248.0
% Solids	35
Pulp SpGr	1.277
GPM Pulp	777.1

to Pond

(Over)

Solids*	0.3
Water	645.8
Pulp	646.0
% Solids	.05
Pulp SpGr	1.00
GPM Pulp	2596

*500 ppm
to tunnel

TABLE 6-5. CASE NO. 1A, 425TPH, 33% SOLIDS

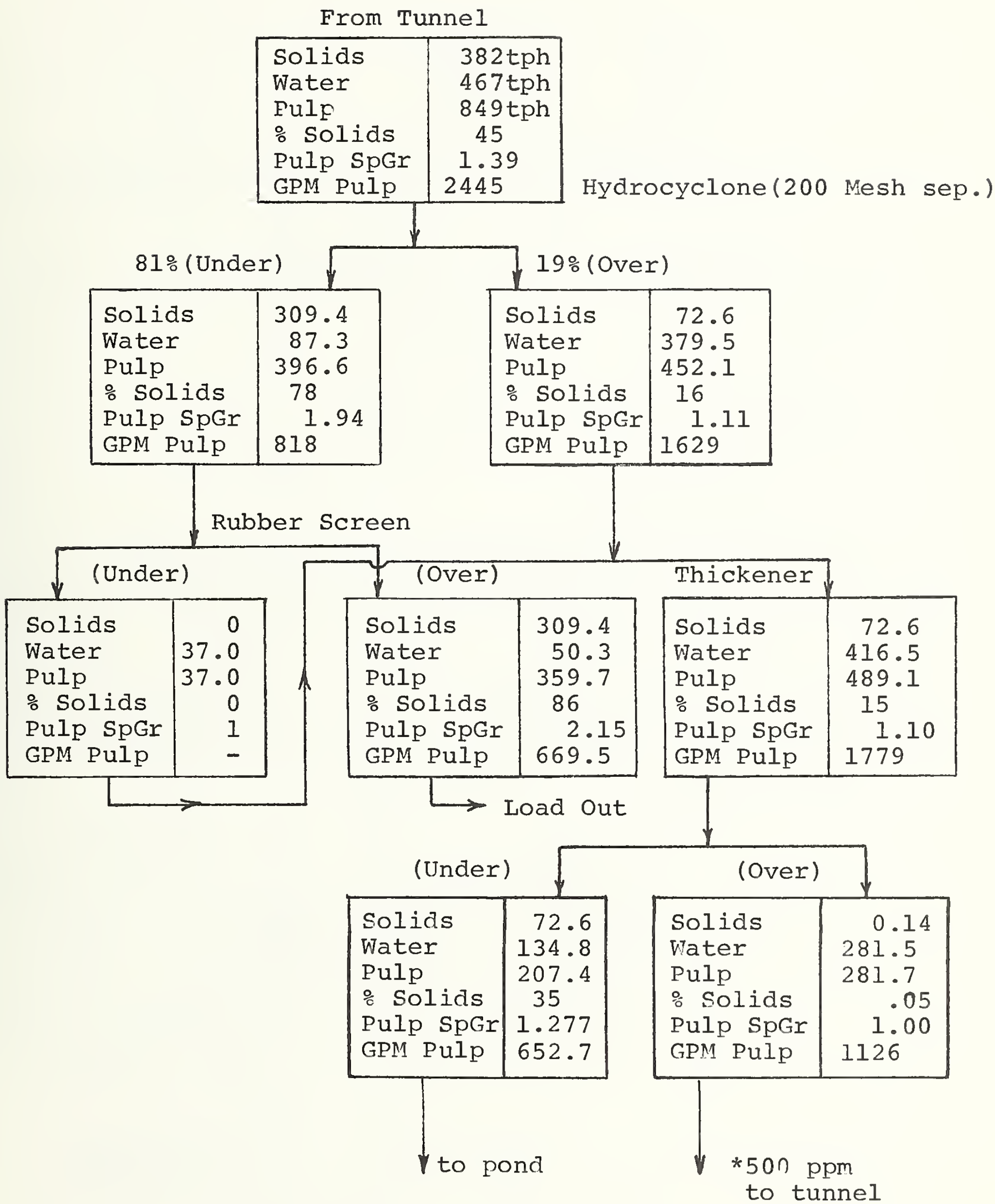


TABLE 6-6. CASE NO.2A, 382TPH, 45%, SOLIDS

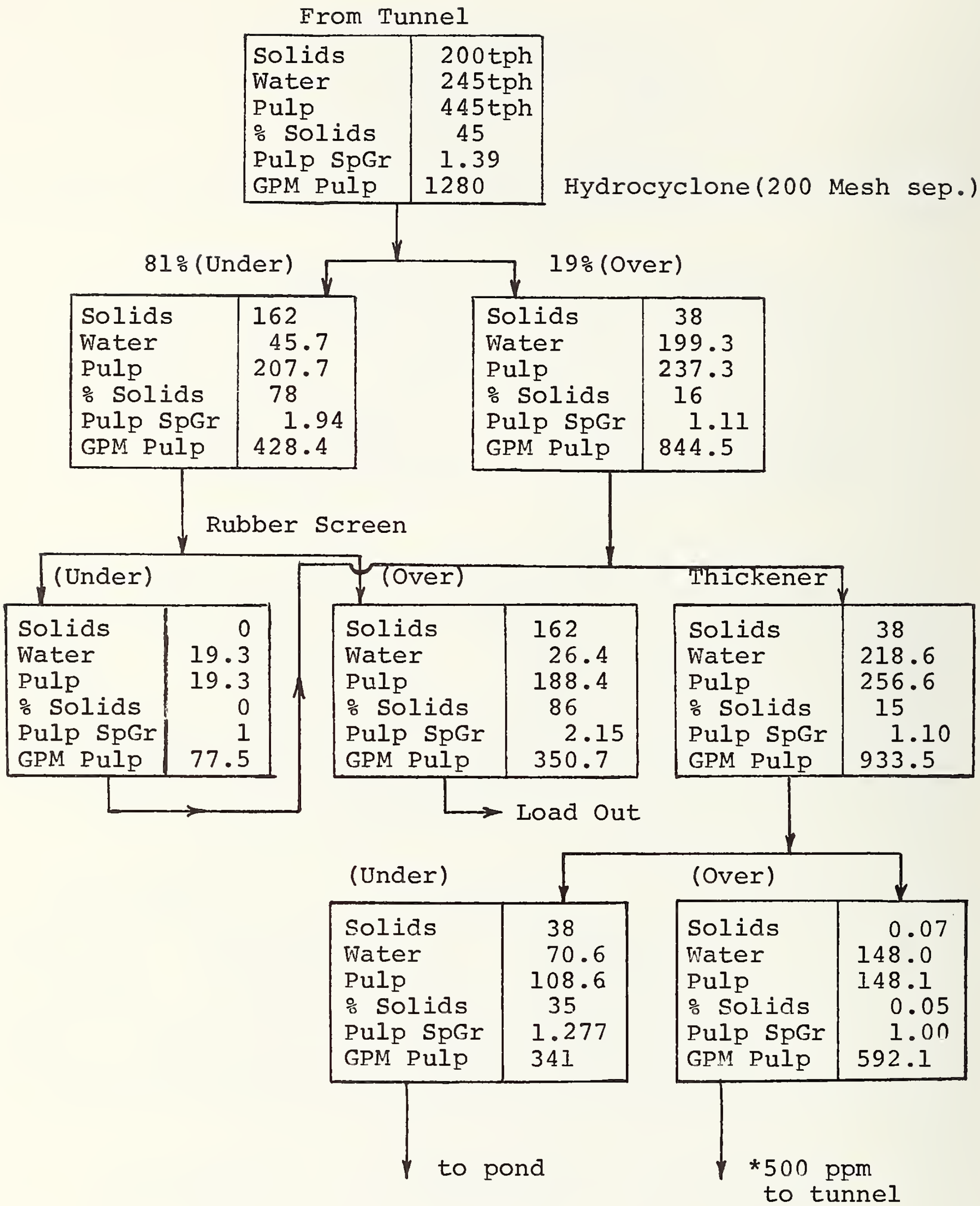


TABLE 6-7. CASE NO. 3A, 200TPH, 45% SOLIDS

TABLE 6-8. EQUIPMENT SELECTION & PRICE

Fine Muck Slurry Transportation and Dewatering

	Case 1A	Case 2A	Case 3A
	425 tph, C_W=33%	382tph, C_W=45%	200tph, C_W=45%
Hydrocyclone	3-Model S-830A $4850 ea. $14,550	2-Model 1036A $7650 ea. $15,300	2-Model 830A $5385 ea. $10,770
Rubber Screen	3-Model 4x8ft. 15HP $18,000 ea. $54,000	2-4x10ft. 10HP $19,000 ea. $38,000	2-4x8ft. 10HP $18,000 ea. $36,000
Thickener	2-2500/45 5 HP $63,000 ea. $126,000	1-2500/45 3HP $63,000	1-1700/45 3HP $52,500
Pump-Motor	12x10x17in.CW 100HP motor $5039	12x10x17in.CW 75HP motor $4819	8x6x17in. 40HP motor $3397
Total Cost	$199,589	$121,119	$102,667
Unit Cost	$470/tph	$317/tph	$513/tph
Installed Horsepower	120	88	53

The proposed dewatering scheme is examined in some detail in this study because of its importance economically in the overall cost evaluation of a slurry pipeline muck haulage system. The earlier study produced dewatering capital costs as follows:

Case 1	425 tph	$282,000	135 hp installed
Case 2	382 tph	$259,600	145 hp installed
Case 3	200 tph	$137,100	80 hp installed

These costs were not modified for the coarse and fine slurries but were developed on the basis of the amount of fine solids passing a 200 mesh (74 micron) screen. Both slurries described previously had approximately the same amount of minus 200 mesh solids (~19%). These costs were adjusted to December 31, 1973.

In comparing Tables 6-4 and 6-8 for the February 1977 dewatering costs in this present study, the earlier costs without being adjusted for inflation, are higher by approximately 35% to more than double. Because the installed horsepower for the dewatering system is so much greater in the earlier study, the operating costs for the present dewatering system should be appreciably less.

Thus the present dewatering scheme is better than the earlier version in several categories. It

a. is cheaper in capital costs.

b. is cheaper in operating costs

c. is expected to last longer, possibly allowing amortization over more than one job.

d. occupies less space

e. is more flexible in terms of surge loading

f. is more mobile.

7. REFERENCES AND BIBLIOGRAPHY

7.1 REFERENCES

1. Faddick, R.R. and J.W. Martin, "Pneumatic-Hydraulic Material Transport System for Rapid Excavation of Tunnels", U.S. Dept. of Transportation, DOT-TSC-75-17, August, 1974.
2. Haller, H.F., H.C. Pattison, and O.C. Baldonaldo, "Interrelationship of In-Situ Rock Properties, Excavation Method, and Muck Properties", Holmes and Narver, Inc., Report to Advanced Research Projects Agency (ARPA), U.S. Dept. of Defense, July, 1973
3. Bond, Fred C., "Lab Equipment and Tests Help Predict Metal Consumption in Crushing and Grinding Units", Engineering and Mining Journal, V. 165, N. 6, pp. 169-175, 1964.
4. Fillip, Lester M., "Material Handling Consideration in Bored Tunnels", Proceedings Vol.I, Rapid Excavation and Tunneling Conference, San Francisco, California, June 24-27, 1974, published by the Society of Mining Engineers.
5. Cowan, David, "Continuous Face Haulage Systems", U.S. Bureau of Mines, Study Contract HO 242025, July, 1975.
6. Schriek, W., L.G. Smith, D.B. Haas, and W.W. Husband, "Experimental Studies on the Transportation of Two Different Sands in Water in 2,4,6,8,10, and 12 inch Pipelines", Report VII, Saskatchewan Research Council, Nov. 1973
7. Faddick, R.R., and G.S. DaBai, "Optimization of Particle Size Distribution for Coal Slurry Pipelines", Slurry Transport Association, 2nd International Technical Conference, Las Vegas, NV, Mar. 2-4, 1977.

8. Bain, A.G., and S.T. Bonnington, "The Hydraulic Transport of Solids by Pipeline", Pergamon Press, Oxford, UK, 1970, p. 88-91.

9. Bonnington, S.T., and A.L. King, "Jet Pumps and Ejectors", Engineering, 123, May 3, 1968.

10. G. Govatos, and I. Zandi, "Pumping Solid-Liquid Mixtures by Jet Pump", A.I.Ch.E 7th National Meeting, Atlantic City, N.J., Aug. 29-Sept. 1, 1971.

11. Liu, T.K., "Handbook of Muck Utilization in the Urban Rapid Transit Tunneling Process", U.S. Dept. of Transportation, October, 1975.

12. McCain, D.L., "Future Slurry Transport of Large Particles Based on Operations with Coarse-Coal," AIME Trans. Vol. 258, pp. 363-366, Dec. 1975.

13. McCain, D.L., and R.W. Umphrey, "Continuous Underground Coal Haulage by Hydraulic Transport", Hydrotransport 3, Paper A6, BHRA, CSM, Golden, CO. May 1974.

14. Dahl, H.D. and D.L. McCain, "Continuous Underground Slurry Transport of Coal", Mining Congress Jl., May 1974.

15. Poundstone, W.N., "Hydraulic Transportation of Coal from Face to Preparation Plant", SME Fall Mtg., Acapulco, Mexico, Sept. 22-25, 1974.

16. Anon. "Strategies and Outlook", Continental Oil Co., A Presentation to Security Analysts, May 12, 1977.

7.2 BIBLIOGRAPHY

7.2.1 Muck Preparation

Note: A considerable number of manufacturers' catalogs were examined in this study. Because the material is too voluminous for inclusion here, the following bibliography lists only the more important literature.

Anon., "Role of Crushing in the Minerals Processing Industry", Minerals Processing, V.14, N.3, p.7-9, Mar. 1973.

Anon., "Crushing by Dynamic Impact", World Construction, V.18, N.11, p. 30-31, Nov. 1965.

Anon., "Let's Look at Hammermills", World Construction, V.18, N.10, p.30-32,&37, Oct. 1965.

Anon., "Impact Breakers for Primary Reduction", Mine and Quarry Eng., V.29, N.6, p.234-246, June 1963.

Anon., "A Comparative Analysis of Secondary Crusher Types-I & II", Pit and Quarry, V.63, N.2,3., Part I, p.83-84, 89-94, Aug. 1970, Part II, p.96-101, 109., Sept. 1970.

Boszhardt, A.K., and M.P. Hahn, "Preventive Maintenance for Crushers & Screens", Pit and Quarry, V.52, Part I., p.102-107, Part II, p.122-124, 129, Feb. 1960.

Brownell, "Crushing Practice and Theory", Rock Products, McGraw-Hill, V.55, N.12, Dec. 1952.

Burkhardt, S., "Efficient Crushing Improves Reduction Costs", Allis Chalmers Engineering Review, V.40, N.2, p.25-27, 1975.

Currie, John M., "Unit Operations in Mineral Processing", Curriculum Development Branch, Dept. of Education, Prov. British Columbia, 1973.

Day, D., "How to Select a Crusher", Engineering and Contract Record, V.88, N.3, p.18-19, Mar. 1975.

Kochanosky, B.J., "Portable Crusher for Open Pit and Quarry Operations", Mining Engineering, V.12, N.12, p.1271-1274, Dec. 1960.

Lewis, F.M., James L. Coburn, Roshan B. Bhappu, "Comminution: A Guide to Size-Reduction System Design." Mining Engineering, V.28, N.9, September 76, pp. 29-34.

Mabson, L.R., "Primary Crusher Selection", Mine and Quarry Eng., V.27, N.3, p.114-121, Mar. 1961.

Mitchell, R.J., "Crushing Roundup", Rock Products, V.67, N.3, P.59-92, Mar. 1964.

Rundquist, W.A., "Some Simple Basics in Aggregates Plant Design", Pit and Quarry, V.66, N.7, p.124-129, Jan. 1974.

Stern, A.L., "A Guide to Crushing and Grinding Practice", Chemical Engineering, V.69, p.129-146, Dec. 10, 1962.

7.2.2 Conveyors

Anon., "Belt Haulage Economics", Coal Age, V.65, No.10, Oct. 1960.

Anon., "Retractable Conveyors", Colliery Guardian, V.210, No. 5429, May 7, 1965.

"Belt Conveyors for Bulk Materials", Conveyor Equipment Manufacturers Association, Cahners Publishing Company, Inc., Boston, Massachusetts, 02110.

"Continuous Haulage Methods Designed to Keep Continuous Haulage Continuous", Coal Age, July, 1976.

Faddick, R.R., and J.W. Martin, "Material Handling for Tunnel Construction Workshop" held at Keystone, Colorado, for the U.S. Department of Transportation, August, 1977.

Flowers, A.E., "New Continuous Haulage System Paces Production Gains at B&Z", Coal Age, May 1965.

Grierson, A., "Economics of Belt Conveying", Colliery Guardian, V.209, No. 5391, Aug. 14, 1964.

Hurley, V.L., "Thin Seam Continuous Mining", Mining Congress Journal, V.45, No.11, Nov. 1959.

Peabody's Camoy System, "True Continuous Coal Mining", presented at 1976 SME-AIME Fall Meeting, Denver, Colorado, September 1-3, 1976.

Schweitzer, F.W., and L.G. Dykers, "Belt Conveyors vs Truck Haulage: Capital & Expense", Presented at 1976 SME-AIME Fall Meeting, Denver, Colorado, September 1-3, 1976.

Torre, D.C., "Continuous Belt Haulage Makes Mark Underground", Coal Mining and Processing, March 1974.

Woodley, Douglas R., "Encyclopedia of Material Handling, Vol. I", Pergamon Press Ltd., Headington Hill Hall, Oxford, 4&5 Fitzroy Square, London, W.1.

APPENDIX A

PNEUMATIC PIPELINE COST UPDATE

In the earlier report, "Pneumatic-Hydraulic Material Transport System for the Rapid Excavation of Tunnels", August, 1974, Appendix A dealt with cost formula derivations. Nine separate cost items were evaluated. These were checked again in February 1977 and the revisions are shown below:

C1 - Operating Cost

55 cents/ton Radmark Engineering, Vancouver B.C.

1 operator at $15/hr

C2 - Equipment Costs

Hardened steel pipe $40/ft for 10" diameter

Stower and Power Pack

Capacity	100 tph	200 tph	300 tph
Stower	$33,500	$45,000	$52,500
Power Pack	$10,500	$12,000	$14,000

Blower Assembly (Includes Motor and Accessories)

3,200 cfm - $30,000

7,800 cfm - $44,000

12,000 cfm- $48,000

Efficiency of blower ~0.8

APPENDIX B

CONOCO-CONSOL SYSTEM

Consolidation Coal Co., now owned by Continental Oil, has probably the best example of a materials handling pipeline system which comes closest to duplicating muck haulage by pipeline in a rapid transit tunnel. The Conoco-Consol system pipelines coarse coal slurries from the face of an underground coal mine to the preparation plant.

A coal mine differs from a subway tunnel in two major ways: coal is softer than hard rock and eastern coal seams are much smaller in height than subway tunnels. These two factors are somewhat self-compensating in tunnel excavation. There is more headroom in subway tunnels but the rock may be harder.

The Conoco-Consol system is described in detail because it is a working materials handling systems and because some of its concepts may be applicable to tunneling. The following paragraphs list the specifications of the past and future Conoco-Consol systems (Ref. 11,12,13, and 14.) Many features of the system are undoubtedly conducive to hard-rock tunneling. Possible exceptions may be the hose hauler which has greater maneuverability in room and pillar mining than in subway tunneling, and the roll crusher which may not be rugged enough for hard rock.

Past System

Equipment Train: continuous miner (drum head), crusher surge vehicle (Fig. B-1), mix (fluidizing) hopper (Fig. B-2), pump (Fig. B-3), hose hauler, booster pump, pipeline to preparation plant.

Location: Consol's Robinson Run mine no. 95 near Shinnston, W.V.

Details:

Mix hopper: 300 gal., water jet equipped, baffles to reduce waves, accepts 5.3 ton/min. of crushed coal.

Injection pump: 12x36" suction, 10" discharge, 350 hp, 3000 gpm, 60 psi, vertical shaft, right angle gear reducer, variable speed drive, top horizontal suction, centrifugal, handles 4 in. top size.

Booster pump: Horizontal shaft, centrifugal

Pipeline: 10 in. diameter, flexible hose, and steel pipe, horizontal length = 2950 ft., lift = 115 ft., V = 12 to 14 fps., hose length = 500-750 ft., water and slurry pipe identical for interchangeability.

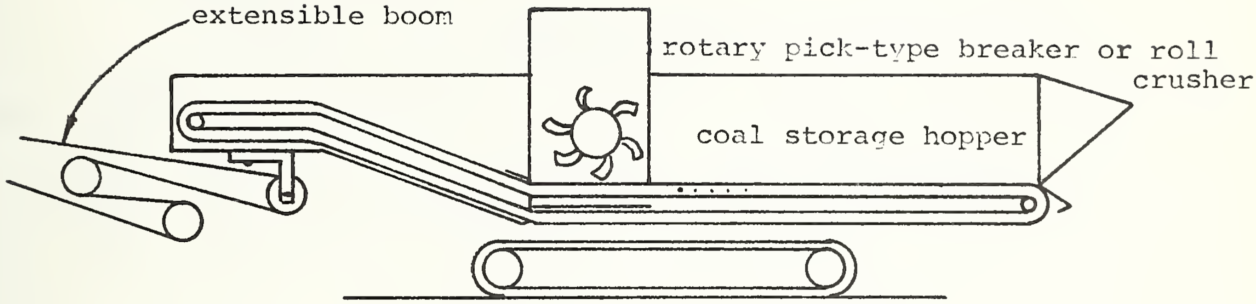


FIG. B-1. Crusher-Surge Vehicle

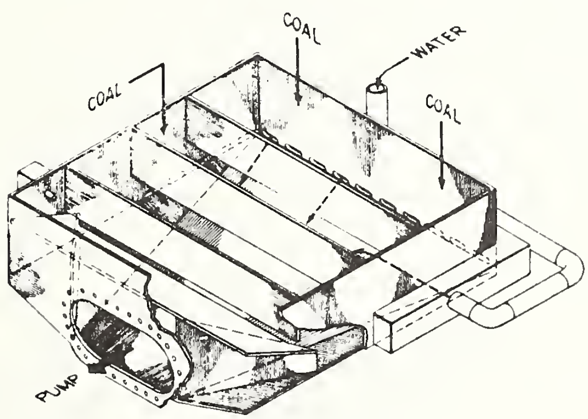


FIG. B-2. Mix Hopper

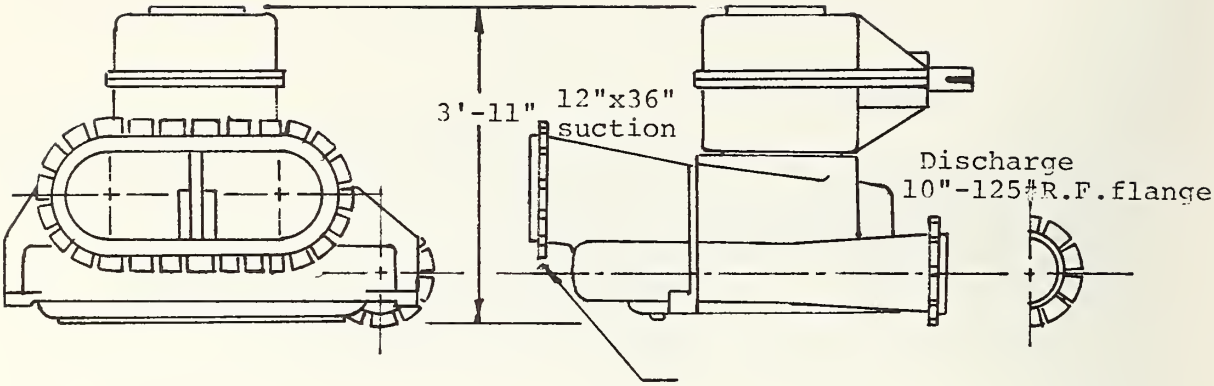


FIG. B-3. Coal Slurry Pump

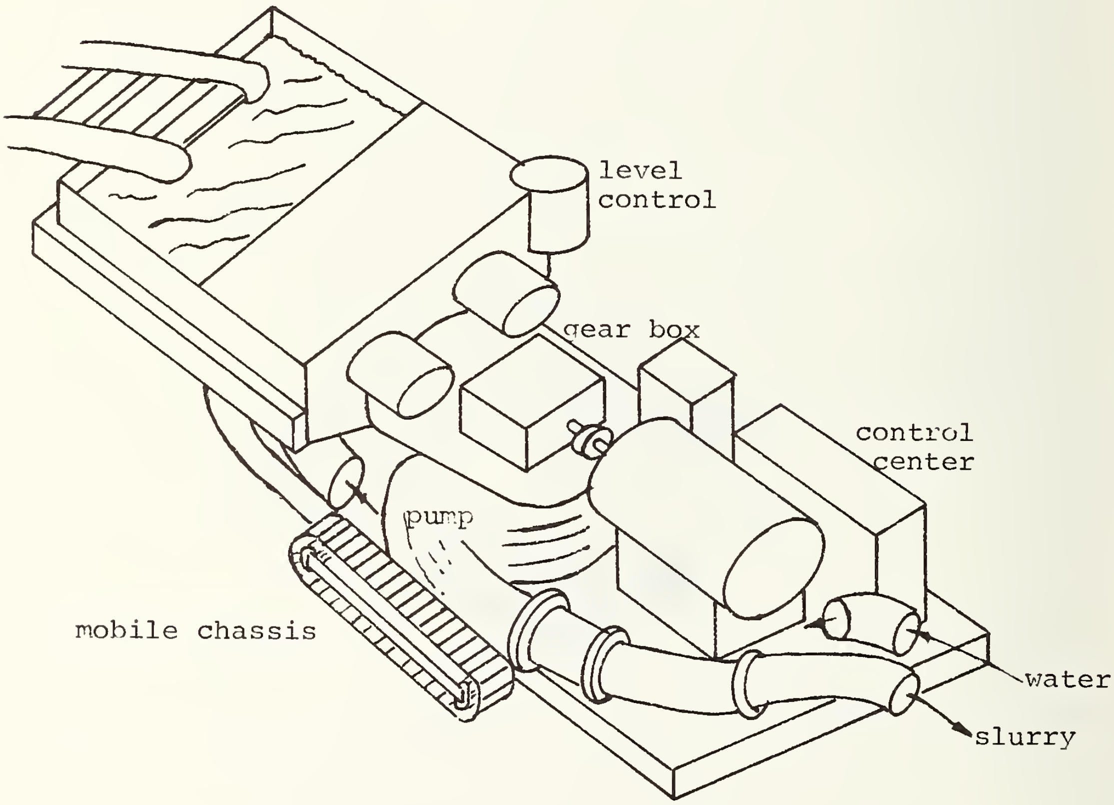


FIG. B-4. Pump Vehicle with Wet Crusher

Future System

Equipment train: Mobile pump vehicle (Fig. B-4) will replace mix hopper, injection pump, and crusher surge vehicle. Contains submerged two-stage (4 roll) Gundlach crusher. Somewhat less surge capacity compromised for equipment compactness and centralization.

Location: Loveridge Mine, W.V.

Anticipated Operation: May 1978

Details: The Loveridge installation (16), as now planned, will service two continuous miners and one longwall section. Peak capacity is 20 tons per minute. Eight-inch diameter lines will carry slurry from the two continuous miner sections to the collection point, and a fourteen-inch diameter line will serve the longwall section. All three lines will terminate in a multiple feed sump. From there, the combined slurry will be pumped vertically 850 feet to the surface, and then 2.4 miles overland through a twelve-inch line to a water separation facility at the preparation plant. A 450-ft hill must be negotiated by the pipeline. Seven centrifugal pumps in series with a variable speed drive will be used. Rated horsepower is 2000. Top particle size will be 4 in. for coal and 1-1/2 in. for rock. Solids will be left in the plant, and the water will be clarified and then returned to the mine. See Figures B-5 and B-6.

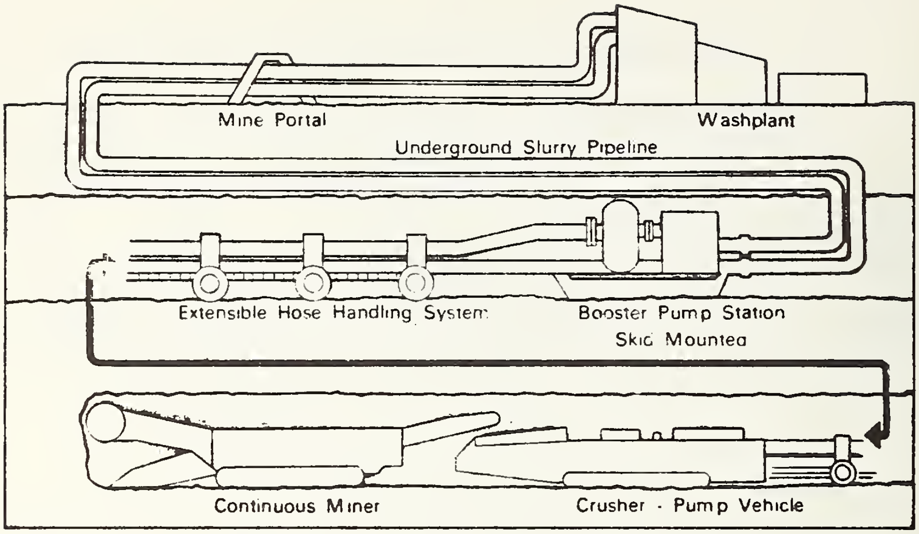


FIGURE B-5. HYDRAULIC HAULAGE

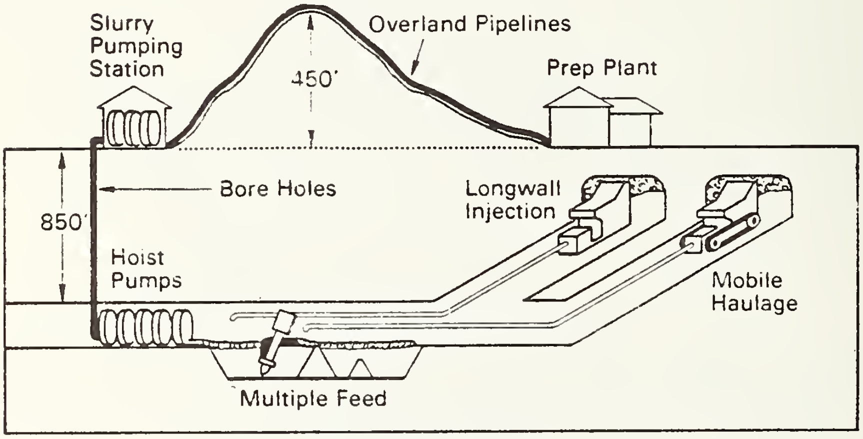


FIGURE B-6. LOVERIDGE SLURRY TRANSPORT HAULAGE PROJECT

APPENDIX C

DEWATERING EQUIPMENT

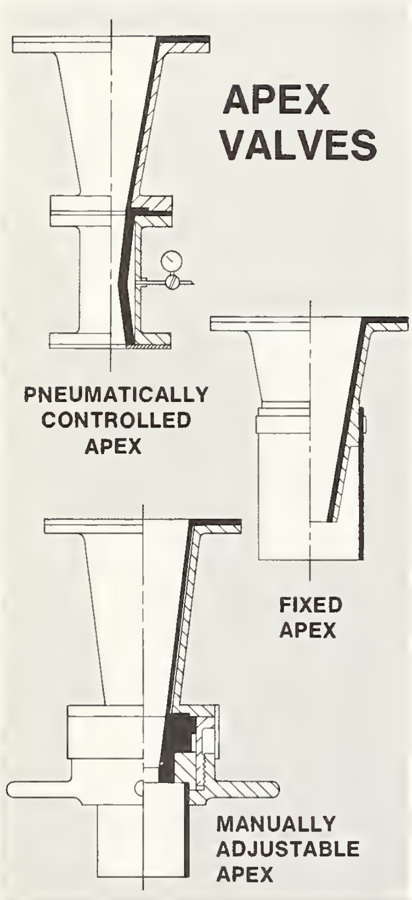
The components discussed here in detail are the hydrocyclone and gravity thickener. The inclusion of manufacturers' names does not constitute endorsement by the authors or the U.S. Department of Transportation. The names are intended only as examples of dewatering equipment.

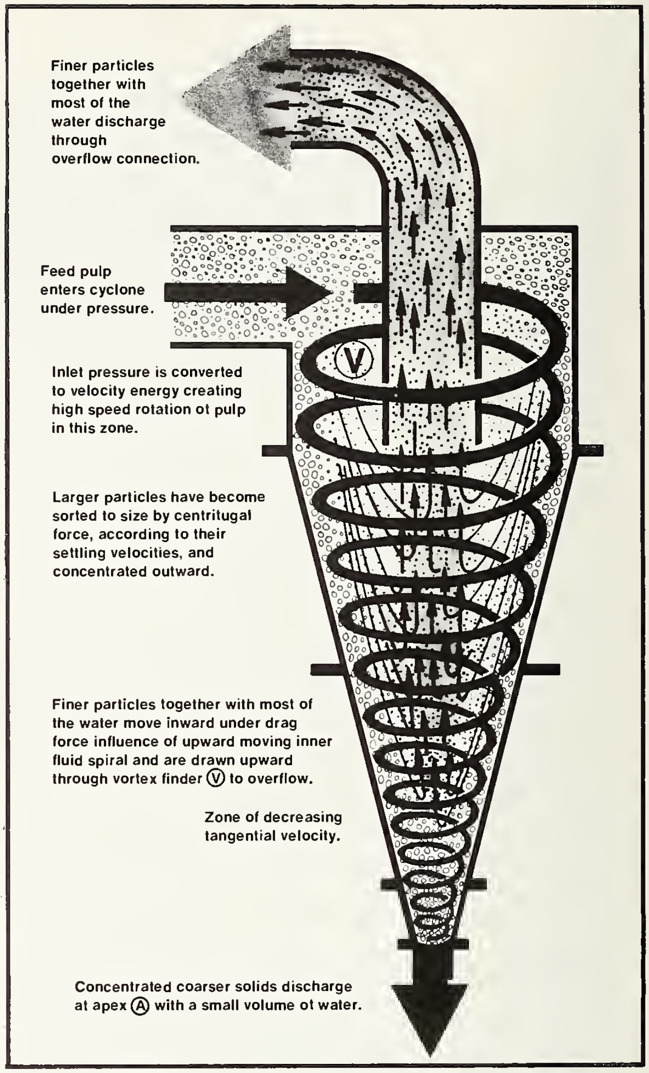
Hydrocyclone

Hydrocyclones were discussed in detail in the earlier report (1). The Linatex separator (hydrocyclone equipped with automatic underflow regulator) discussed in the present study has two features of interest. First, it is lined with natural rubber to give a greater wear life, and second, it has a rubber wedge valve in the apex which is used to regulate the underflow or discharge of thickened solids. See Fig. C-1. These two features are seen as special assets for dewatering tunnel muck because of its relative coarseness and hence abrasive qualities and because of the wide variation in muck throughput.

Consider a tunnel muck of solids specific gravity of 2.6 and a size of minus 16 mesh (~1mm). A conventional hydrocyclone will give an underflow discharge of 60 to 65% solids. The discharge spurts out of the apex in an "umbrella" shape. With a surge overload, the discharge concentration increases to 70 to 75% solids by weight but the apex discharge tends to choke. Simultaneously, the overflow contains coarse solids which should have passed through the underflow. These losses result from the solids being short circuited to the overflow because of inadequate underflow apex capacity.

The manufacturer of the Linatex separator claims that a relatively constant discharge concentration of 72 to 76% solids by weight is maintained by the automatic underflow regulator. The rubber wedge valve is throttled according to variations in the muck throughput. For surge loadings, the





FLOW PATTERN IN CYCLONE

FIGURE C-1. RUBBER-LINED HYDROCYCLONE
(Courtesy Linotex)

valve is opened wider, and for below design feed rates, the valve is closed more to allow the underflow solids concentration to increase. The end result is a more constant high density discharge regardless of the muck feed rate.

Gravity Thickener

The Lamella gravity settler is an inclined thickener combined with a flocculating tank and a vibrator. The principle of operation is shown in Fig. C-2. Slurry is pumped through the flocculating tank where a chemical (solid or liquid) can be added in controlled amounts and agitated with the slurry. The slurry enters a bottomless feed box and flows under side plates, then upwards on inclined plates. The clarified effluent exits at the top of the tank. The solids settle out by gravity onto the plate surfaces and slide downward into the sludge hopper. Thickening of the settled solids is accomplished with a low-amplitude vibrator pack located in the sludge hopper.

Since thickening by gravity is independent of slurry depth the inclined thickening trays compact the vast surface area required to settle large slurry flows. There are ten square feet of settling area for each square foot of surface (or floor space area). Being so much smaller than the conventional circular thickener tank, the unit becomes portable and is easily moved about by truck. It lends itself to a pre-assembled package which reduces field erection costs.

Typically the gravity thickener will handle an overflow rate of 0.6 to 1.5 gpm/ft^2. For sand-like tunnel muck, a rate of 0.6 to 0.8 gpm/ft^2 is usually considered. When flocculants are used on fine solids, about 1 ppm of polymer is used. The design overflow is said to be 30 ppm with flocculation and about 550 ppm without flocculation. A flash mixer and chemical preparation equipment are required with flocculation. In this study the flocculation tank and the flash mixer are both included in the equipment cost. Flocculation

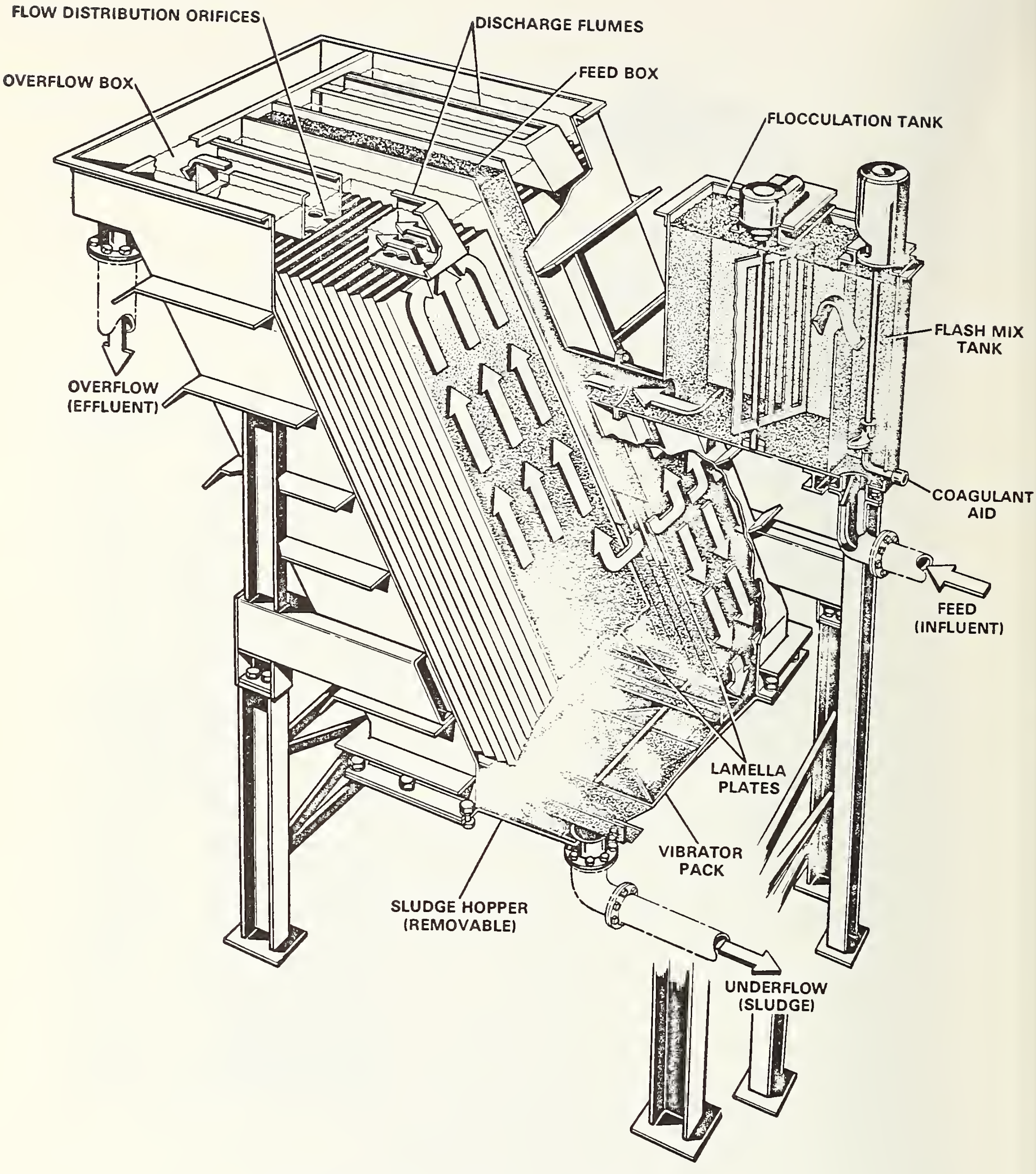


FIGURE C-2. LAMELLA THICKENER
(Courtesy Ted Miller Associates)

would probably be used in the thickener if the water were not returned to the tunnel for pipeline use.

For an extra $5000 the hopper can be enlarged to hold more solids. The prices quoted in this study do not include the enlarged hopper. As a rough guideline the gravity thickener is said to cost two-thirds of the total installed cost of a comparable capacity conventional thicknener and to occupy 10% of the space. Erection costs of the gravity thickener include a concrete pad, crane rental, bolting and piping. They are said to comprise about 1% of the purchase price of the gravity thickener.

APPENDIX D

COAL HOISTING IN THE U.K

1. National Coal Board, Shirebrook Colliery
North Derbyshire Area, England

Now in Production

SYSTEM SPECIFICATIONS

Material	=	Coal 60%; Rejects 40%(shale)
Size	=	-1/2", pieces up to 2 in. in length
Capacity	=	50 ltph, Max: 61 tph
Horizontal Distance (Underground)	=	327'-6"
Vertical Distance	=	1035'
Horizontal Distance (Surface)	=	166'
Elbows	=	2 @ 90°, 1 @ 60°, 1 @ 45°
Pipe Diameter	=	12" (14" at surface) Supplier: Esser Werke
Discharge Device	=	Cyclone
Blower H.P.(Connect)	=	700
System Pressure	=	<5psi(air); 11.5psi @ 60-70tph
Radmark Feeder	=	RTL 205
Power Package	=	25HP @ 2000 psi
Hydraulic Fluid	=	Fire Resistant
Blower	=	Waller 1435 MK $\overline{\underline{V}}$

CONTROLS: See Fryston

NOTE: No dust problem. No water required at feeder or in water ring at discharge. Shale and sandstone mildly abrasive. Water at 0.8gpm used for cooling blower. No noise at surface pipe discharge or at shaft bottom. 110db at blower underground.

2. National Coal Board, Fryston Colliery
North Yorkshire Area, England

Start-up due October, 1977

SYSTEM SPECIFICATIONS

Material	=	run of mine coal
Size	=	-1-1/2"
Capacity	=	50 ltph
Horizontal Distance (Underground)	=	41 ft.
Vertical Distance	=	1635 ft.
Horizontal Distance (Surface)	=	40 ft.
Elbows	=	2 @ 90°
Pipe Diameter	=	14"
Discharge Device	=	Cyclone
Blower H.P.	=	900 (Connect)
Pressure	=	14psi
Radmark Feeder	=	RTL205
Power Package	=	25HP @ 2000psi
Hydraulic Fluid	=	Fire Resistant to N.C.B. Spec.
Blower	=	Waller 1640 MK $\overline{\underline{V}}$

CONTROLS: Mercoid switches preventing overpressures-sequenced start-up and shut-down. Remote control from surface.

NOTE: The pipe line will be supplied by Esser Werke, with the vertical section comprising approximately 9 sections of Esser patented rotatable pipe each at 187 ft. Each section is supported on a console plate and ball bearing for ease in turning, and the sections are interconnected through a gland and stuffing box arrangement.

APPENDIX E

REPORT OF INVENTIONS

No inventions or discoveries were made or conceived during the course of this contract. However, several types of equipment not currently in use as components of muck handling equipment in tunneling were analyzed in detail to determine their potential for improvement of tunnel muck transportation. In the future, use of selected equipment may advance the technology of tunnel excavation.

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