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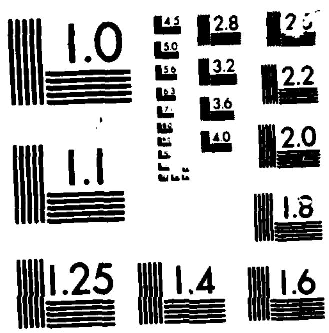

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DESIGN CALCULATIONS

Power requirements

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Tool forces

Bar forces

Traction requirements for cutting

Vertical reaction to cutter forces

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Power Requirements

One way of estimating power demand is by considering the specific energy consumption for drag bit cutting in relation to the strength of the rock. The attached graph gives experimental data for drag bit cutting. Assuming that we need enough power to tackle rock up to 20,000 lb/in^2 compressive strength, and making a slightly conservative interpretation of the data, the specific energy needed is about 3000 lbf/in^2. However, when drag bits are fitted to a mobile machine the performance is often less efficient than in a laboratory test of a single tool. On the basis of practical experience with various machines, we would suggest taking a value of 5000 lbf/in^2 for overall specific energy (including transmission losses).

Specific energy is power divided by volumetric excavation rate. With an advance speed of 3 ft/min, a trench depth of 7 ft, and a kerf width of 6 3/8 in., the volumetric excavation rate is 1.93×10^4 in^3/min. The required power is thus 243 h.p.

Another approach is to see what power levels have proved feasible on previous machines. The attached graphs provide data for a variety of machines -- continuous belt types, transverse rotation machines, and axial rotation machines. Power levels are plotted against a nominal working surface area, so that the graphs give power density, or power per unit area. This is not a straightforward thing, however. Power density tends to decrease with increasing machine size (elephants and dinosaurs aren't as peppy as birds and bees). This trend is easily seen when comparing axial rotation machines. Designers of small diameter drills have the delusion that they can deliver up to 1500 hp/ft^2, while designers of very large tunnel boring machines will settle for 1 hp/ft^2.

For the kind of machine being considered here, 50 hp/ft^2 is probably a maximum practical value. If we take this maximum value and assume a 6 3/8 in. wide bar cutting to a depth of 7 ft at an angle of 60^o to the horizontal, the corresponding power input is 215 h.p.

The 215 h.p. value is a gross cutter power that covers losses in the final drive and in chain friction. The 243 h.p. based on specific energy is also gross cutting power, which includes "windmilling" power. For the saw envisaged here, we estimate a power loss of 30 h.p. in the chain and the final drive, so that the net power for cutting and clearing is about 185 h.p. or 213 h.p. for the power density and specific energy estimates respectively.

It is suggested that 200 h.p. be taken as the gross cutter power for preliminary design purposes. Higher powers might be more than the system could handle safely (this also means that 3 ft/min at 7 ft deep in 20,000 psi rock is a bit too optimistic).

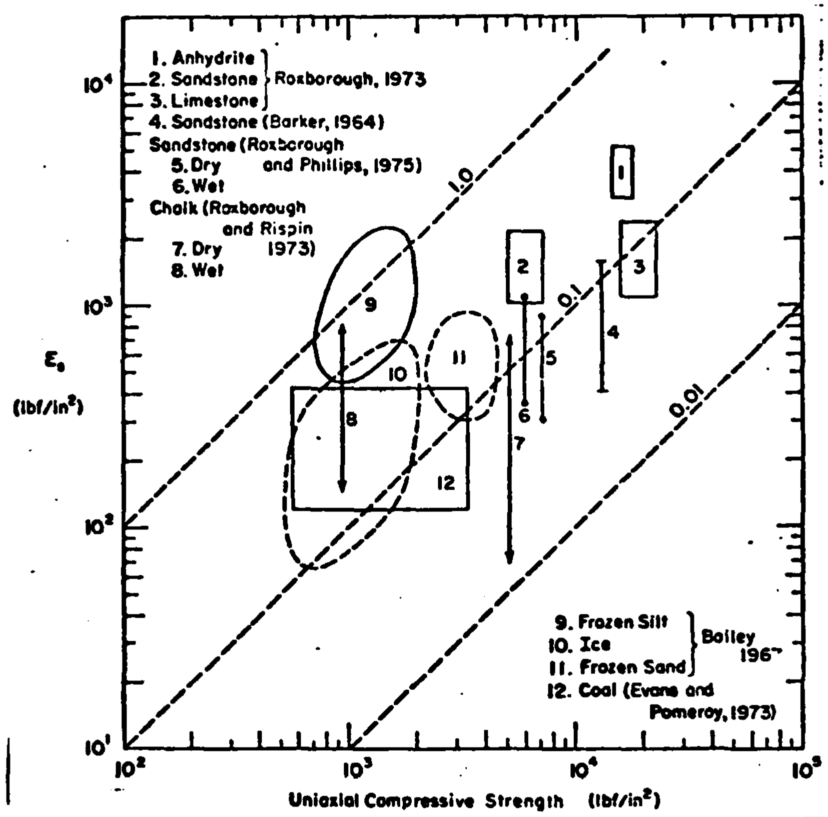
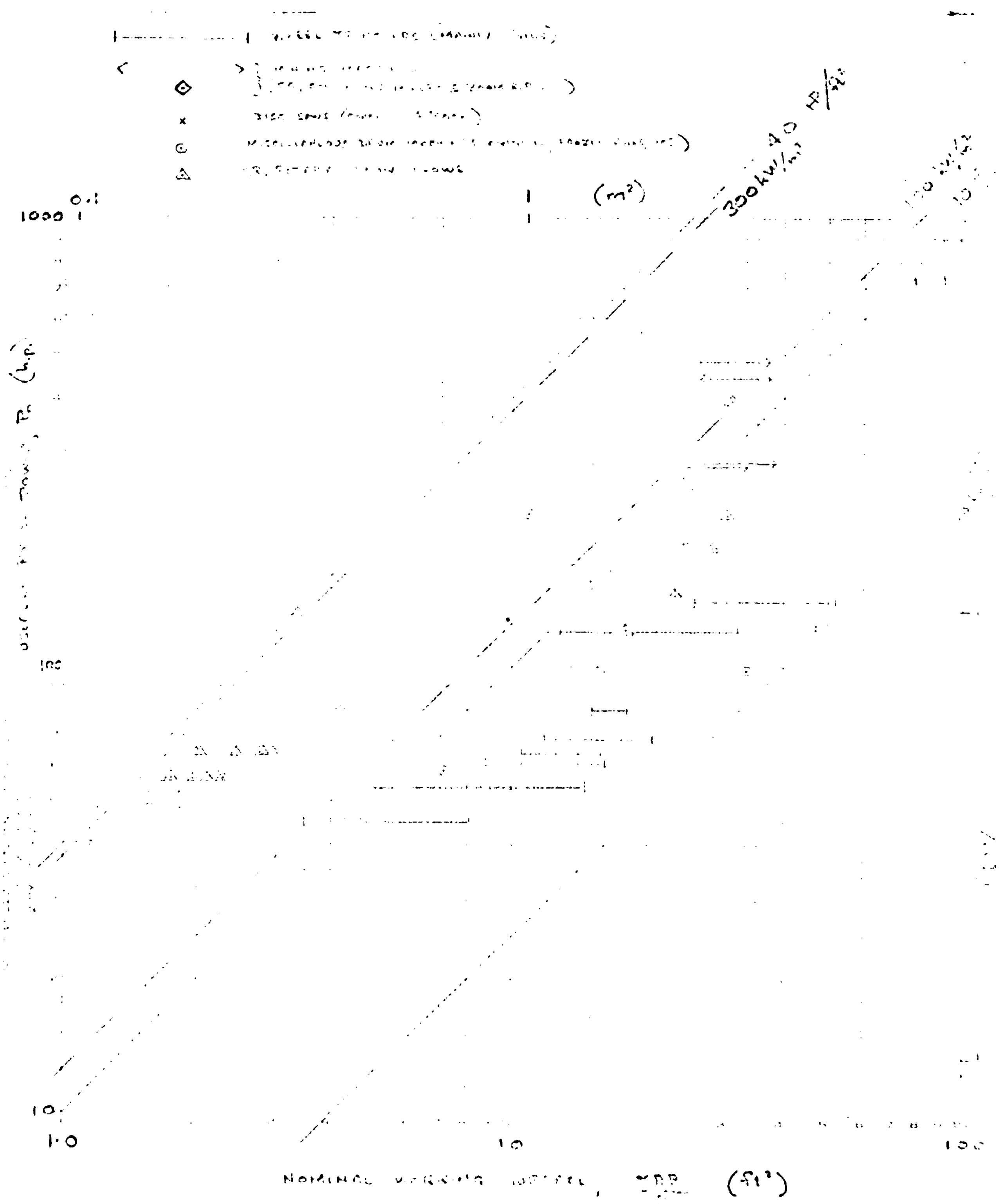
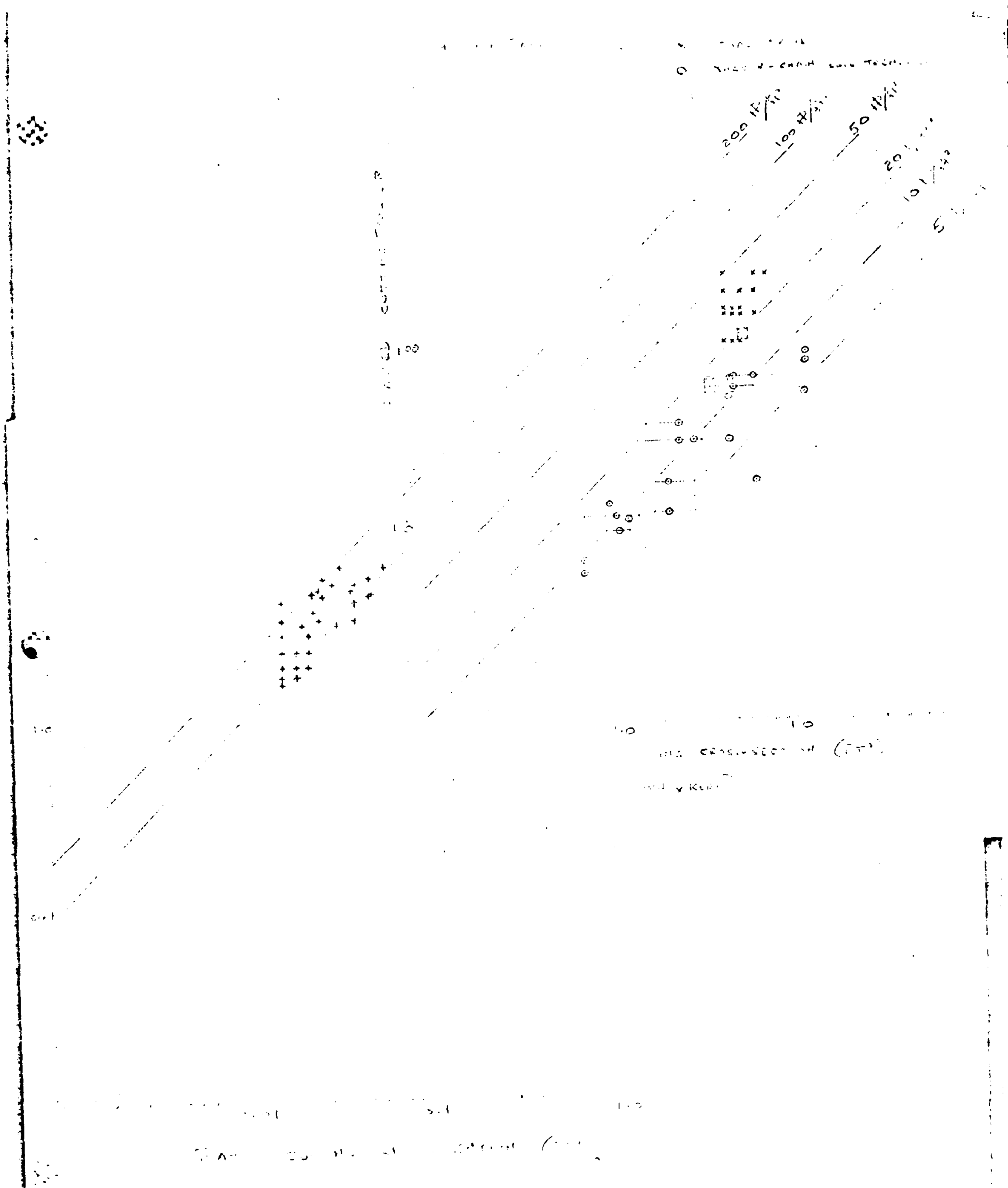


Figure 88. Specific energy for drag bit cutting plotted against uniaxial compressive strength. The lines drawn across the plot denote fixed values of the ratio E_s/σ_c.



(m²)
300kW/m²
1000
100
10
1.0
10
100
(ft²)



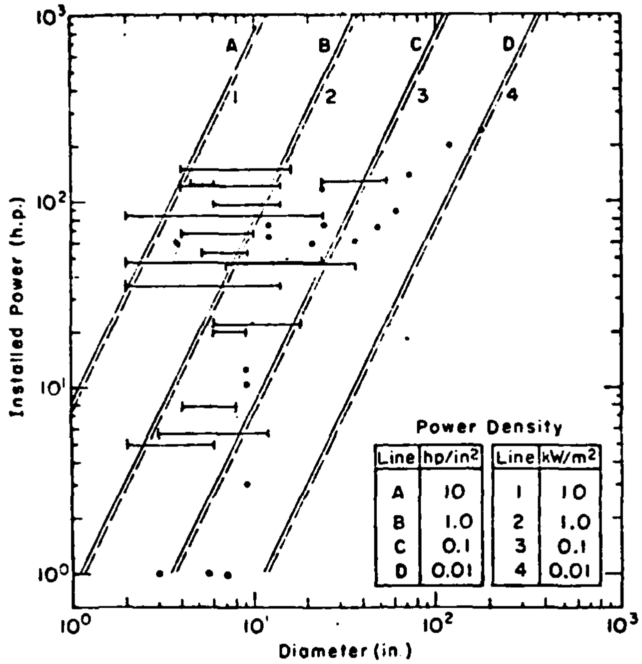


Figure 10. Installed power of existing rotary diggers and auger drills plotted against bit diameter. Lines representing a range of power density levels are superimposed.

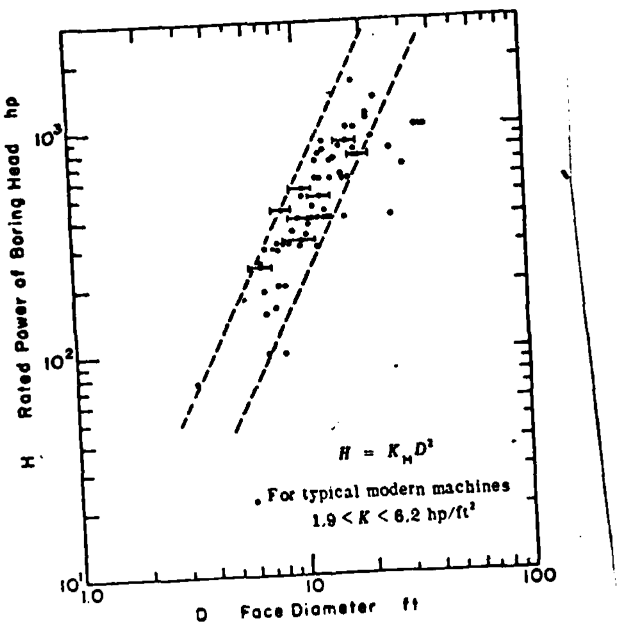


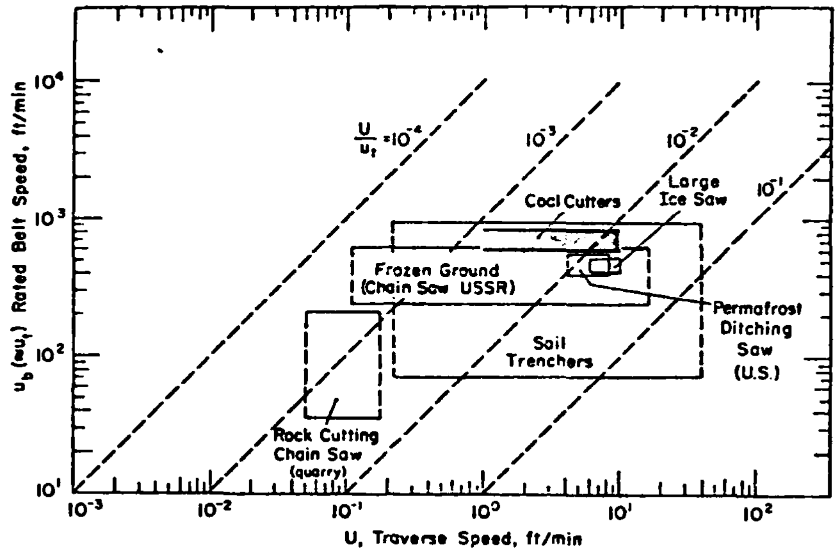
Figure 2. Rated power of boring head versus face diameter for hard rock tunneling machines.

Chain Speed

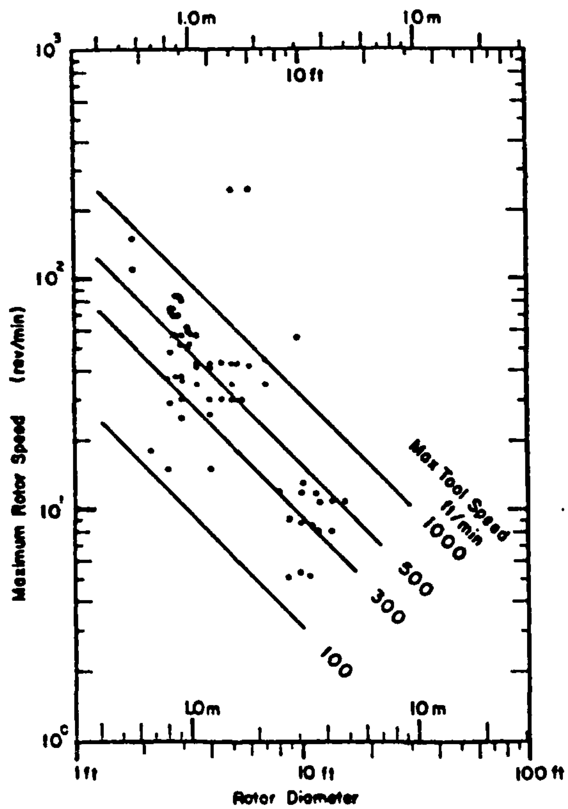
The chain speed, or tool speed, will be kept approximately constant. If tool speed is too low, tool forces become unacceptably high and jerking operation can be felt. For a given power level, chain tension and torque increase as chain speed decreases. In air, very high tool speeds are unacceptable because they lead to high tip temperatures and rapid wear. This should not be much of a problem underwater, but very high chain speeds might lead to high fluid drag forces and high inertial forces for accelerating cuttings.

The attached graphs show the ranges of tool speeds that have been used on various rock cutting machines equipped with drag bits. They cover trenchers, heavy chain saws, large disc saws, wheel ditchers, concrete and asphalt planers, rotary drills, tunneling machines, and mining machines. Overall, the acceptable range for cutting hard non-metallic materials is about 200 to 800 ft/min.

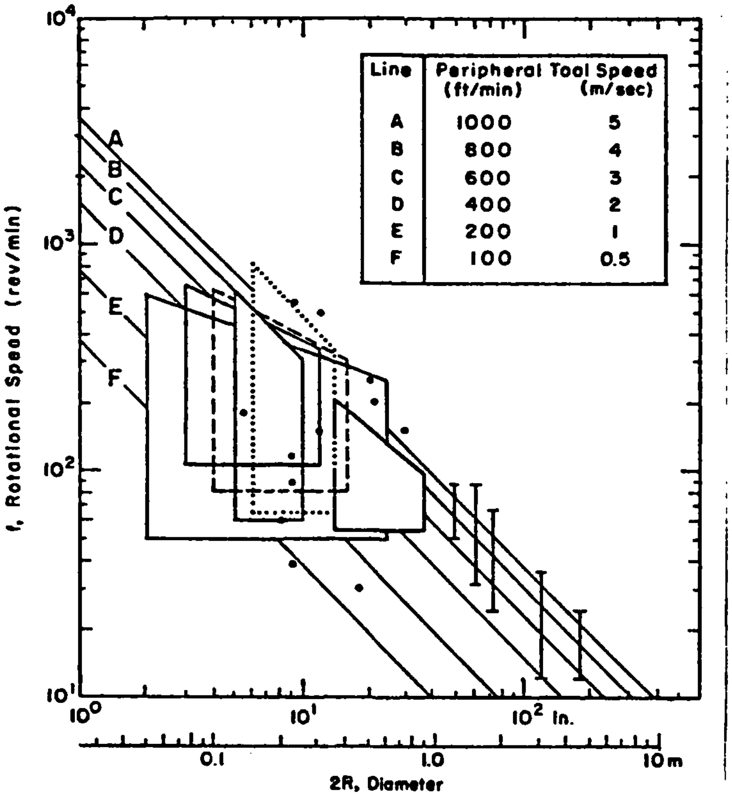
For a start, we suggest a chain speed of 700 ft/min, having already checked some implications with respect to chipping depth, tool forces, bar forces, and traction requirements.



U, Traverse Speed, ft/min
u_b ($\approx u_t$) Rated Belt Speed, ft/min
$\frac{U}{u_t}=10^{-4}$
10^{-3}
10^{-2}
10^{-1}
Coal Cutters
Large Ice Saw
Frozen Ground (Chain Saw USSR)
Permafrost Ditching Saw (U.S.)
Soil Trenchers
Rock Cutting Chain Saw (quarry)



1.0m
10m
10 ft
Maximum Rotor Speed (rev/min)
Max Tool Speed ft/min
1000
500
300
100
1 ft
10 ft
100 ft
Rotor Diameter


Line
Peripheral Tool Speed
(ft/min)
(m/sec)
A 1000 5
B 800 4
C 600 3
D 400 2
E 200 1
F 100 0.5
f, Rotational Speed (rev/min)
2R, Diameter
in.
m
0.1
1.0
10 m

Tool Forces

The dynamics of rock cutting tools is a confused technical area that has been the subject of a major study at CRREL. Results are not yet published, but they have been compiled and digested. Some numbers will be drawn from this data compilation.

We make the assumption that chipping depth in hard material will be about 5 mm, or 0.2 inches. For a start, we will consider that each tool has an effective width of about 0.5 inches at the cutting tip. Looking over available experimental data, the tangential component of tool force f_t for fresh sharp tools is likely to be in the range 200 to 800 lbf, depending on the rock type. These values could double as the tools wear.

The maximum averaged value of f_t is set by the maximum chain force and the number of teeth engaged in the work. With 170 h.p. available for cutting (i.e. 200 h.p. minus 30 h.p. drive and friction loss), and a chain speed of 700 ft/min, the maximum tangential force that can be applied to the teeth is 8000 lbf. With the bar cutting 7 ft deep at a 60° angle, there might be about 14 teeth in the work, giving an average of 570 lbf per tooth. With the bar cutting to a depth of only 3.5 ft, and full power still maintained, the average value of f_t would rise to 1140 lbf per tooth.

The normal component of tool force f_n depends on the design of the tool and the amount of wear. With a sharp tool chipping deeply, the ratio f_n/f_t can be less than unity, but with a blunt or worn tool, f_n/f_t can increase to 2 or more. For design purposes, we can take $f_n/f_t = 1$ for fresh tools working properly and $f_n/f_t = 2$ to represent the tools in a worn condition.

Bar Forces

Forces on the cutter bar can be obtained from the tool forces. For convenience, we resolve the resultant bar force into tangential and normal components F_t and F_n, and take its point of application as the center of the working surface.

Since the maximum value of F_t is set by the drive power and the chain speed, we take as the design value F_t = 8000 lbf.

To obtain design values of F_n, we consider the tool force ratio f_n/f_t. For fresh tools we take $fn/_{ft}$ = 1, so that $F_n = F_t$ = 8000 lbf. For worn tools we take f_n/f_t = 2, so that $F_n = F_t$ = 16,000 lbf.

Traction Requirements for Cutting

The drawbar pull needed to propel the cutter (F_n) is obtained from the bar forces and the bar angle θ. With θ measured from the horizontal and the bar "trailing.":

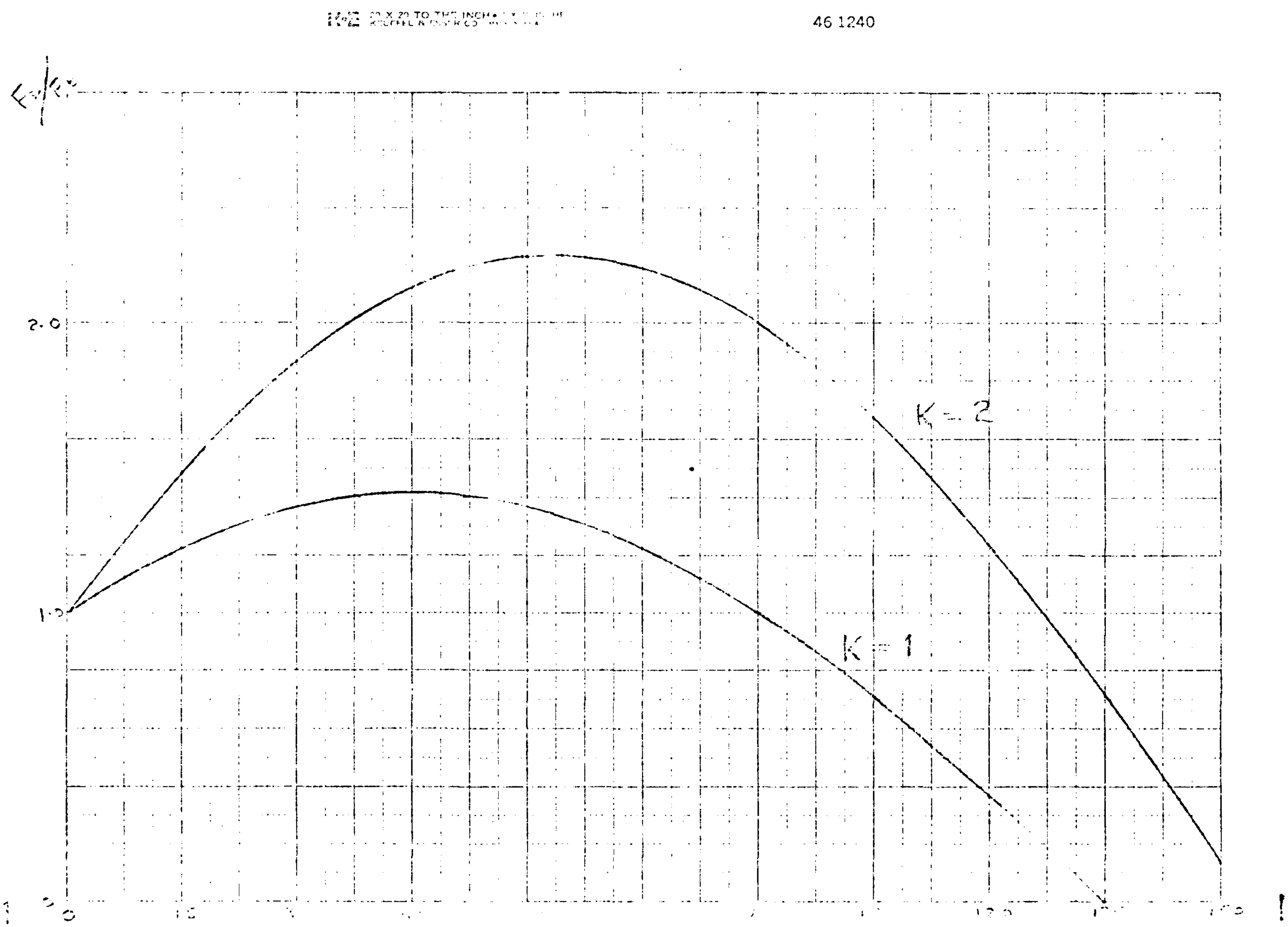
$$F_D = F_t \cos\theta + F_n \sin\theta$$
$$= F_t (\cos\theta + K \sin\theta)$$

where $K = F_n/F_t = f_n/f_t$. The attached graph gives F_D/F_t as a function of θ, with K as parameter.

It is clear from the graph that there are significant advantages when the nose of the bar is leading rather than trailing ($\theta > 90^\circ$). However, there are practical difficulties in trying to work with $\theta > 90^\circ$. One is the design of the feed shoe, although this might not be too much of a problem for cables up to 4 in. diameter or so. Another more serious problem is the difficulty of sumping in with a feed shoe behind the saw.

With a bar angle of 60° and fresh tools, the drawbar force at full power is $F_D = 1.3666 \times 8000 \approx 11{,}000$ lbf. With worn tools it could be as high as 18,000 lbf.

With a bar angle of 90°, F_D is 8000 lbf with fresh tools, and up to 16,000 lbf with worn tools.



K<>E 20 X 20 TO THE INCH
46 1240
2.0
K = 2
K = 1

Vertical Reaction to Cutter Forces

The machine reaction needed to counteract the vertical component of force produced by the cutter bar, F_v, is

$$F_v = F_n \cos\theta - F_t \sin\theta$$

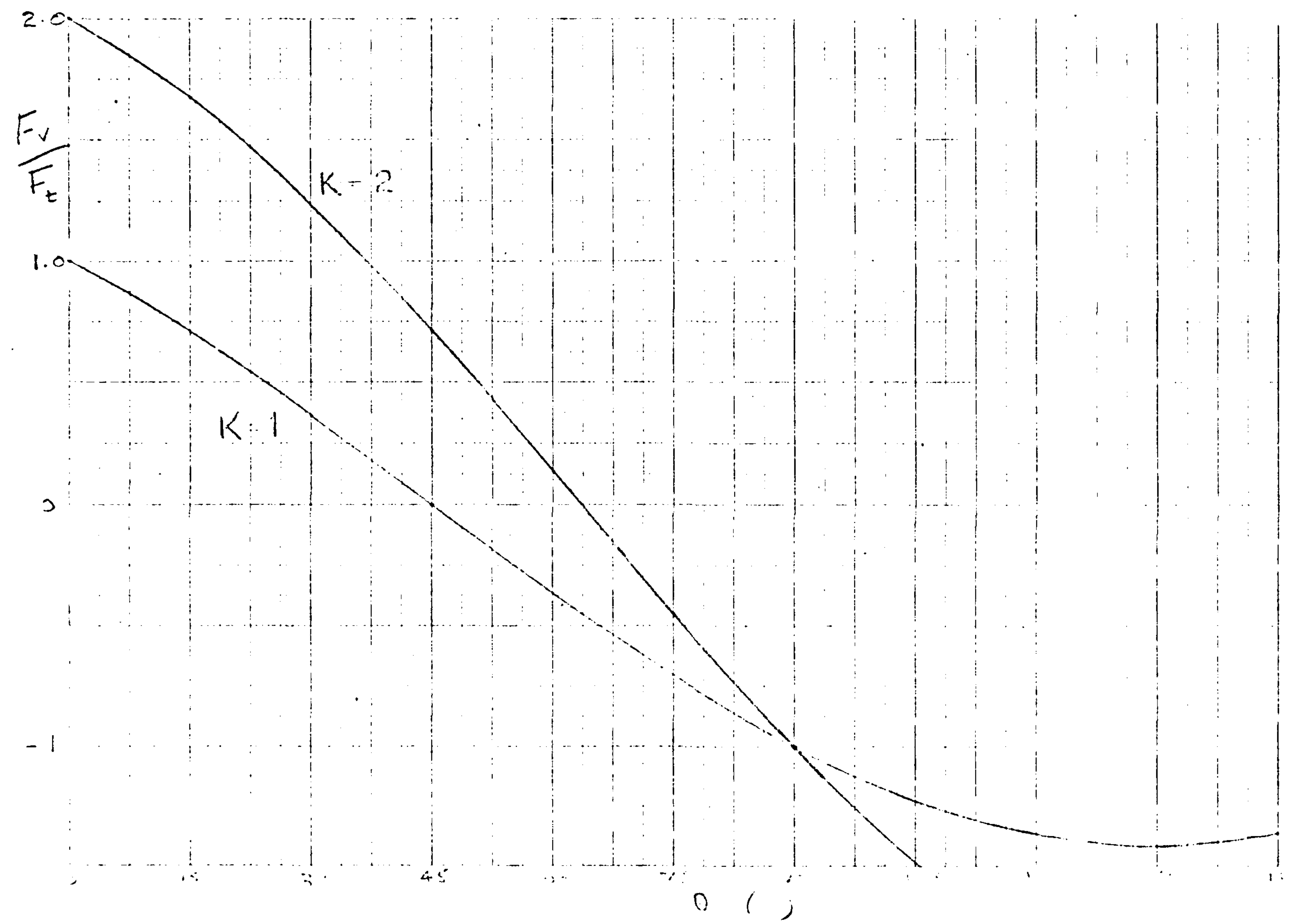
$$= F_t (K \cos\theta - \sin\theta)$$

where positive values of F_v denote a downthrust from the machine (tending to push its tail up). The attached graph gives F_v/F_t as a function of θ, with K as parameter.

From the graph, we see that the cutter is pulling itself down into the work for the range of bar angles that are of most interest. At a bar angle of 90°, the cutter pulls down with a force of 8000 lbf at full power, irrespective of the condition of the teeth.

With a bar angle of 60° and sharp teeth, the cutter pulls down with a force of about 3000 lbf at full power. With worn teeth and a 60° bar angle, the vertical force balance is more or less neutral.

A downpull from the cutter increases the effective weight of the carrier vehicle, thus tending to improve traction on a hard surface.


2.0
1.0
0
-1
F_v / F_t
K = 2
K = 1
45
θ ( )

Cutter Bar Moments

The moment of the cutter bar influences the weight and balance of the carrier vehicle. Let L be the horizontal distance along the ground to the face of the bar from the ground level point about which moments are being taken (e.g. below rear bogey, below front bogey of either carrier unit, center of pressure of track system). If d is the cutting depth (trench depth) and θ is the bar angle, the cutting moment M_c (excluding effects of bar weight) is

$$M_c = -F_t L \sin\theta + F_n \left(L \cos\theta + \frac{d}{2\sin\theta}\right)$$

$$= F_t L\,(K\cos\theta - \sin\theta) + \frac{KF_t d}{2\sin\theta}$$

$$= F_v L + \frac{KF_t d}{2\sin\theta}$$

where a positive moment is one that tends to lift the tail of the vehicle.

Moments arising from the weight, position and angle of the cutter bar are considered under the static weight distribution of the whole machine.

Cutting Teeth

In the U.S., coal cutters are commonly fitted with bullet bits (plumb bob bits), small drag bits, or hardfaced picks. None of these are really suitable for work in tough abrasive materials. In concrete and frozen soils, wear rates with these rather feeble tools are prohibitive for most practical purposes. However, there are more robust forged tools that carry heavy carbide tips in secure mounting pockets, and in some experimental applications they have proved to have a wear life about 100 times greater than the weaker tools.

So far, no U.S. source of supply for heavy duty picks has been found, although there is no reason why they could not be produced by manufacturers such as Kennametal, Carboloy, Carmet, Austin, Bowdil, or others. Suitable tools have been obtained in the U.K. from Austin Hoy, and other makers such as Padley and Venables or Marwin Mining Tools could probably supply similar picks.

For best performance, sharp tools are needed. It may be advisable to make provision for grinding the flank, or relief face, to keep the tools sharp.

It is also very important to mount the cutters correctly. Bullet bits (plumb bob bits) are usually mounted at a 45^{o} angle to the chain surface, but some of them have a 90^{o} cone as the carbide tip. This means that apparent relief angle is zero, and actual relief angle for finite feed rate is negative. This arrangement may create the illusion that the bit is self sharpening, but the fact that the bit retains a point is largely irrelevant when that point is not being applied to the work.

We would require that the cutting teeth should have a relief angle of at least 5° plus the designed "kinematic" relief angle. The included angle will probably have to approach 90° for durability, so that rake angle will be not far from zero.

The grade of carbide will be chosen so as to strike a balance between hardness (for abrasion resistance) and toughness (for shock resistance).

We have a file of design drawings for the bits of several manufacturers. Being proprietary information, these may not be reproduced, but we have available for inspection the drawings of those bits that we regard as suitable for this project.

Tooth relief angle

The kinematic relief angle β_2'

is given by

$$\beta_2' = \theta - \tan^{-1}\left[\frac{\sin\theta}{\cos\theta + (U/u_t)}\right]$$

This is a maximum for $\theta = 90^\circ$, i.e. for a vertical bar. Making allowing for high traverse speed in soft material, we take U = 10 ft/min, u_t = 700 ft/min, and $\theta = 90^\circ$. This gives $\beta_2' \approx 1^\circ$.

To minimize cutting forces, the "dynamic" relief angle should be at least 5°.

Thus the estimate for minimum acceptable relief angle is 6°. We would probably specify a higher value, perhaps 10° to 12°.

Tooth Layout

The exact tooth layout cannot be decided until the bar and chain have been fully identified, but some approximate ideas are useful for planning. We shall assume that only one cutting tooth will be fitted to each chain link, that the chain pitch will be around 6 or 7 in., and the kerf width will probably be 6 3/8 in. (nominal kerf widths of 6, 7 or 8 in. can be provided). The required number of cutting tracks depends to some extent on the type of cutters used, and on the overbreak characteristics of the material to be cut. It has been suggested elsewhere that optimum lateral spacing of kerf is given by

$$(s-w)/\ell \approx \tan \phi$$

where s is center to center spacing of adjacent parallel kerfs, w is effective tool width, ℓ is chipping depth, and ϕ is the overbreak angle of the rock. For design purposes we shall assume w = 0.2 in, ℓ = 0.2 in, and ϕ is from 50^o to 70^o. This gives S = 0.44 in for $\phi = 50^o$ and S = 0.75 in for $\phi = 70^o$.

On machines that have evolved from practical experience, there are usually 7 to 13 cutting tracks over a kerf width of 6 3/8 in. A high number of tracks is used where the saw cuts tough minerals such as trona, while a small number of tracks is used for cutting friable materials like coal.

The calculation gives 8 to 14 spaces across a 6 3/8 in kerf, which means 9 to 15 bit tracks.

For constant chain speed and constant feed rate, chipping depth increases with the longitudinal spacing of cutters, which in this case means that it increases with the number of cutting tracks.

Having already made some exploratory calculations that take into account other parts of the system, we suggest using 9 cutting tracks.

We now have to check whether this gives reasonable values for chipping depth. Assume a traverse speed U of 3 ft/min, a chain speed u_t of 700 ft/min, a chain pitch of 6.5 in, and one tooth on each chain link. This implies a longitudinal tool spacing S of 58.5 in. The chipping depth is $(U/u_t)\, S \sin\theta$, so that with a bar angle of 60° the chipping depth is 0.22 in., and with a bar angle of 90° it is 0.25 in.

These are substantical chipping depths. They are probably very efficient for cutting the weaker materials, but the traverse rate might have to be reduced somewhat when working in the hardest material the system can tackle.

For a tooth lacing, we would probably use a staggered vee arrangement.

Cutting clearance

The conveying capacity of the cutter chain cannot be calculated exactly until the design is fully known, but there should be no difficulty for traverse speeds up to 20 ft/min.

HYDRAULIC SYSTEM CALCULATIONS

This analysis is based on hydraulic system operating pressures of about 3000 lb/in^2, since those pressures can conveniently be accommodated by commercially available large-diameter hydraulic hoses. The procedure followed here is to size hydraulic motors or actuators based on torque, speed, or force requirements, size hoses, size pumps, and finally size the prime mover.

Trencher Chain Drive

The required power at the chain drive sprocket is 200 hp and the design chain speed is 700 ft/min. Assuming a sprocket diameter of 1 ft, the sprocket speed is given by:

$$n = \frac{u_t}{\pi D}$$

Where n = sprocket speed, rpm

u_t = chain speed, ft/min

D = sprocket pitch diameter, ft.

In this case, the sprocket is found to rotate at 223 rev/min. The required torque at the sprocket is:

$$T = \frac{5252 \times hp}{n}$$

$$T = 4710 \text{ ft-lb}$$

This seems to lend itself quite well to a direct drive from a high-torque, low-speed hydraulic motor. For example, a Sundstrand Model MH-187, type JC hydraulic motor operates at speeds up to 250 rpm, 3000 psi continuous pressure, has a flow rate of 80.8 gal/min per 100 rpm, and has a

theoretical torque rating of 2478 ft-lb per 1000 lb/in^2 hydraulic pressure. This motor would require approximately 180 gal/min at a pressure of 1900 lb/in^2. The next smaller size motor, the MH-117, could provide the same power at a flow rate of approximately 113 gal/min at a pressure of 3040 lb/in^2.

Track Drive

The track drive system must develop 40 hp total (20 hp per unit) to propel the vehicle at speeds of 50 ft/min in soft sand. In addition, the drive system must be capable of developing sufficient power to provide 12,000 lb tractive force at vehicle speeds from 0-10 ft/min. The vehicle tentatively selected has a track drive sprocket pitch diameter of 22 in., a final drive ratio of 12.47:1 and a differential ratio of 1.28:1.

A 22 in. pitch diameter sprocket rotates at 8.7 rpm at traverse rates of 50 ft/min. A low-speed high-torque motor (e.g. Sundstrand Model MH-21) coupled directly to the final drive unit on each vehicle would meet the requirements. The requirements could also be met using a high-speed low-torque motor and appropriate gear reductions; however, because of the wide speed range requirement, a manually operated 2-speed transmission may be necessary.

Using the Sundstrand Model MH-21 as an example, this motor would require theoretical flows of approximately 0 to 0.5 gal/min at 2470 lb/in^2 at the low speeds and 12.7 gal/min at 2710 lb/in^2 at the highest speed.

Hydraulic Hose

Assuming separate supply hoses to the track drive motors and to the trencher chain motor, with a common return line for all functions, we can arrive at approximate hose sizes and types based on the calculated flow rates and pressures. A flow velocity of 20 ft/sec will be assumed for all hoses. For a given flow rate, Q, the hose inside diameter can be determined by:

$$d = 0.639 \sqrt{\frac{Q}{v}}$$

where d = hose inside diameter, in.

Q = flow rate, gal/min

v = flow velocity, ft/sec.

For a flow velocity at 20 ft/sec this expression becomes:

$$d = 0.1429 \sqrt{Q}.$$

The trencher chain drive supply system with a flow rate of 113 gal/min would require 1½ in. I.D. hose minimum and withstand working pressures greater than 3040 lb/in^2 (actual hose working pressures are calculated later and are equal to the pump pressure requirements for supply lines). The track drive system,with a total maximum flow rate of 25.4 gal/min requires a 3/4 in. I.D. hose capable of withstanding working pressures greater than 2710 lb/in^2. The return hose must be capable of handling the 113 gal/min from the trencher chain drive, plus 1 gal/min for the slow speed track drive, for a total of 114 gal/min. A 1½ in I.D. hose will barely suffice.

The pressure drop in the lines must be calculated to determine the pump characteristics desired. The pressure drop in a hose is given by:

$$p = 0.000668 \frac{\mu l v}{d^2}$$

where p = pressure drop, lb/in^2

μ = fluid absolute viscosity, centipoise

l = hose length, ft.

v = fluid, velocity, ft/sec

d = hose inside diameter, in.

Thus, for hydraulic oil with an absolute viscosity of 35 cp, flowing in 600 ft of hose, the pressure drop is:

$$p = 14.028 \frac{v}{d^2}$$

From this equation, the pressure drops are found to be 128 lb/in^2 in the trencher supply line, 459 lb/in^2 in the track drive supply line at maximum speed, 17 lb/in^2 in the track drive supply line at 10 ft/min, 129 lb/in^2 in the common return line while trenching, and 29 lb/in^2 in the common return line while mobile.

Hydraulic Pumps

Neglecting pressure effects due to differences in elevation, the numbers above, when added to the motor pressures, may be used directly to determine the theoretical pump horsepower requirements. The horsepower is given by:

$$hp = \frac{pQ}{1714}$$

where hp = horsepower

p = pressure, lb/in^2

Q = flow rate, gal/min

From this we find that the trencher pump should produce 217 horsepower and the track drive pump should produce 47 horsepower. Table 1 summarizes the theoretical motor, hose, and pump requirements. These values are all theoretical and must be corrected for volumetric and mechanical efficiencies when specific hardware is selected. Assuming another 2 gal/min at 3000 lb/in^2 is required for the attitude control system, an additional 3.5 horsepower is required.

Diesel Engine

The maximum engine horsepower is that required to power the trencher, to propel the vehicle at low speeds, and to actuate the attitude control system. The theoretical power required is 222.5 hp (it is assumed that the unbilical hose will be reeled in or payed out with the trencher power shut off).

Table 1. Theoretical Motor, Hose, and Pump Requirements

	Motor Flow gal/min	Pressure lb/in^2	Hose Inside dia. in.	Hose Velocity ft/sec	Hose Pressure drop lb/in^2	Pump Flow gal/min	Pump Pressure lb/in^2	Pump hp
Trencher	113.0	3040	1½	20.5	128	113.0	3297	217
Tracks @ 10 ft/min	1.0	2470	3/4	0.7	17	1.0	2616	2
Tracks @ 50 ft/min	25.4	2710	3/4	18.4	459	25.4	3198	47
Return, trenching	114.0	--	1½	20.7	129	--	--	--
Return, mobile	25.4	--	1½	4.6	29	--	--	--

Kedging

Winch System Calculations

System requirements:

Max. pull: 20,000 lb.

Rate: Variable to 10 feet/min.

Cable length: 250 feet

Design recommendations:

A 1-inch, 6 x 37 construction, stainless steel wire rope has a breaking strength of approximately 77,300 lb. The recommended minimum sheave diameter is 18 inches. An 18.5 inch wide reel will accomodate 250 feet in approximately three layers. The thickness of three orderly wraps is 2.7 inches with a maximum torque arm of 11.1 inches. Maximum and minimum revolutions of the reel will be:

$$\frac{10}{\pi \times 18/12} = 2.01 \text{ rpm max}$$

$$\frac{10}{\pi \times 22.5/12} = 1.70 \text{ rpm min.}$$

The cable should be self level winding over most of the distance. Based on a $1\frac{1}{2}^{\circ}$ fleet angle, the minimum self level winding distance is:

$$\frac{9}{12 \text{ Sin } 1.5^{\circ}} = 29 \text{ ft.}$$

Maximum torque:

$$20{,}000 \times \frac{11}{12} = 18{,}333 \text{ foot-lb.}$$

The low rpm required precludes the use of a low speed high torque (LSHT) hydraulic motor directly without some intermediate reduction. A 3:1 chain drive coupled to a gear reducer (such as a Funk Series 27-000, hydrostatic planetary drive reducer with a reduction of 116.9) will provide an overall reduction of 350:1.

The input power required for this combination assuming a 75% efficiency:

$$\frac{18{,}333 \times 2\pi \times 1.7}{33{,}000 \times .75} = 7.9 \text{ HP}$$

Max torque required at drive motor:

$$\frac{7.9 \times 33{,}000}{2\pi \times 1.7 \times 350} = 69.7 \text{ ft-lb}$$

A Sundstrand Model MH-02 LSHT hydraulic motor has following specifications:

Speed: variable to 750 rpm

Output torque: 30 ft-lb/1000 psi

Max. pressure: 3000 psi

Flow: .97 gpm/100 rpm

Kedging Anchors

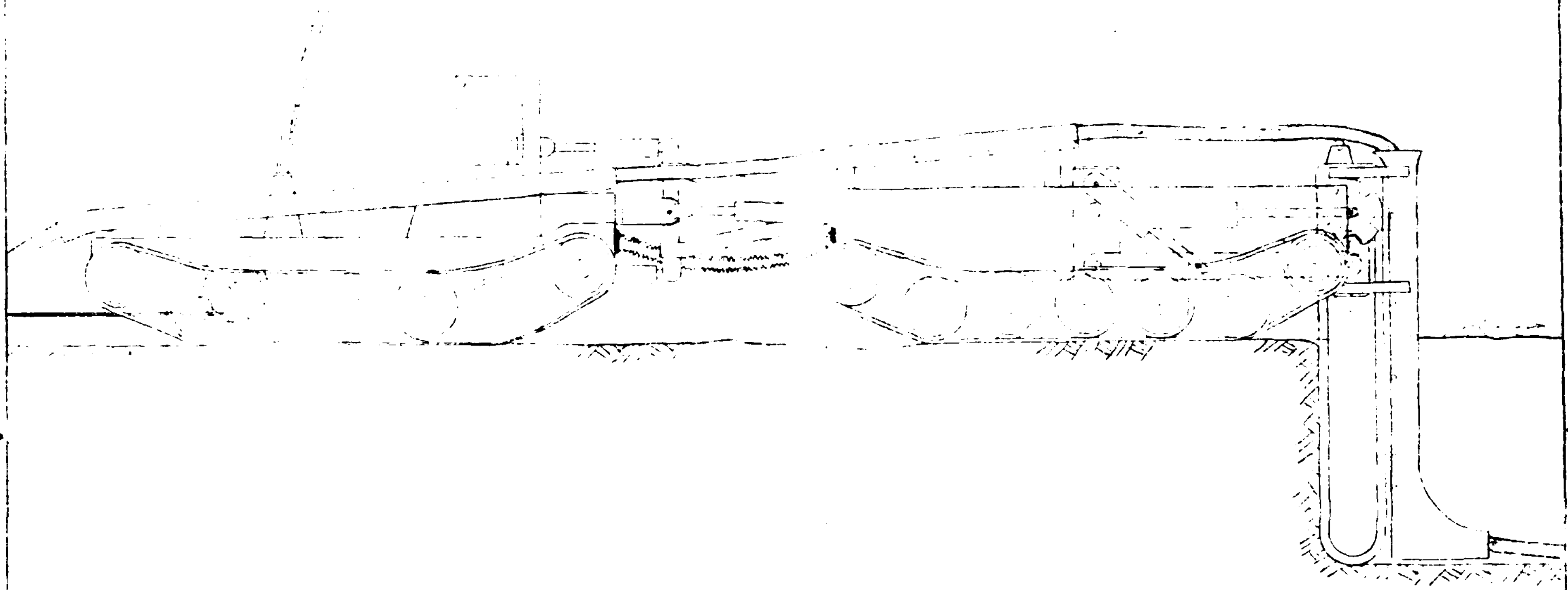
CRREL has no expertise in ship anchor technology, but it is assumed that the necessary guidance on kedging anchors is readily accessible to NAVFAC. However, it appears that the anchor problem is not trivial. If the undersea vehicle "bellies out" in soft sediment, the estimated frictional resistance to towing could be in the range 9000 to 13,000 lbf without buoyancy. If it were towed fast enough to draw full cutter power (an unlikely situation in deep sediment), the horizontal cutting force could add another 8000 lbf. In any event we have to plan for anchors that could resist a horizontal pull of around 12,000 lbf.

Looking at the holding force of Danforth anchors in soft sediment, it appears that resistance can be expressed as $3.2S^2$ lbf, where S is fluke length in inches. For 12,000 lbf resistance, this means a fluke length of about 5 ft! With a navy stockless anchor, we might be talking about a 3 ton anchor! If conventional anchors are the only solution, a separate tender would be required for laying kedge anchors along soft sections of the cable route.

If sections of hard bottom are available for placing anchor points, then drilled or driven ground anchors could be used. Two CRREL reports on ground anchors are attached herewith.

Actually, it seems likely that transverse "deadman" anchors could be set by divers. A long piece of pipe jiggled down into the muck makes a secure yacht mooring when the boat is moored fore and aft.

REVISIONS



GENERAL NOTES

UNLESS OTHERWISE SPECIFIED:

1. ALL DIMENSIONS ARE IN INCHES
2. DO NOT SCALE
3. BREAK ALL CORNERS & EDGES
4. FILLET RADII TO BE .005–.025R
5. TOLERANCES

.X ± .020 XX ± .010

XXX ± .005 FRACTIONS ± 1/32

ANGLES ± 0°-30'

ENGINEER	
DRAWN BY	E PERKINS 27 JAN 77
CHECKED BY	
PROJECT ENG	M MELLOR
SUPERSEDES	
SUPERSEDED BY	
QUANTITY	
MATERIAL	
ASSEMBLY SHEET No	
ITEM No	

U.S. ARMY
COLD REGIONS RESEARCH & ENGINEERING LABORATORY
HANOVER, NEW HAMPSHIRE

UNDERWATER TRENCHER

SIZE	DRAWING NUMBER
B	EE77-13

SCALE | JOB No | SHEET of

DRAFT SPECIFICATIONS

Cutter bar

Cutter bar mounting

Main cutter drive

Cutter chain

Carrier vehicle

Vehicle drive system

Kedging winch system

Hydraulic power source

Hydraulic hoses and hose reels

Cable trough and feed assembly

Controls and instrumentation

Closed circuit TV

Electrical cable for control and monitoring

Cutter bar

The cutter bar shall conform in general design to a standard coal cutter bar, such that the standard cutting chains of various manufacturers can be fitted. The standard nominal kerf width for such cutters is 6 in. (6.375 in. actual), but widths of 7 and 8 in. are available.

The unobstructed length available for cutting shall be 8.1 ft, permitting penetration to a depth of 7 ft at angles of 60° to 90° from the horizontal.

The bar shall have provision for tensioning the cutting chain.

The bar shall be designed to permit removal from the machine as a unit, complete with chain and drive sprocket. The intention is to have several complete bars available for interchange, underwater if necessary.

There shall be provision for lubrication of the cutting chain and its running track.

Cutter Bar Mounting

Alternative #1 - The cutter bar and attached feed shoe shall be carried on a mounting frame that provides the following movements:

1. Bar swing, about the axis of the cutter bar drive sprocket, giving a total range of 90°.

2. Boom swing about a horizontal axis on the carrier vehicle, such that the cutter bar swing axis can be moved through a vertical distance of 4.5 ft.

3. Boom swing about a vertical axis on the carrier vehicle, such that the cutter bar can move a horizontal distance of 1.5 ft to either side of the center line.

4. Boom roll, such that the plane of the cutter bar can deflect 30° to either side of the neutral (vertical) position.

The bar swing (1) actuators shall be capable of resisting a horizontal force of 20,000 lbf on the face of the cutter bar. The vertical motion boom swing (2) actuators shall be capable of resisting the dead weight of the cutter bar and mounting members, plus a cutter downpull of 9000 lbf. They shall also be capable of providing 5000 lbf of downthrust, which may include the deadweight of the cutter bar and mounting frame.

Alternative #2 - As above, deleting functions (3) and (4)

Main Cutter Drive

The main cutter chain shall be powered by a hydraulic motor, with power transmitted either directly using a low-speed, high-torque hydraulic motor, or through appropriate drive shafts and gear reductions using a high-speed low-torque hydraulic motor. The drive system shall be sized to provide 200 horsepower continuous duty to the cutter chain drive sprocket at a full-load chain speed of 700 ft/min. The motor shall be of fixed displacement type. The motor shall be capable of operation in sea water at depths up to 150 ft, with provisions to ~~manually~~ adjust case pressures to 5 psi above ambient water pressure to prevent sea water ingress. The hydraulic motor shall operate with petroleum based hydraulic fluids and shall be compatible with the hydraulic pump and hydraulic transmission lines.

Cutter Chain

The cutter chain shall conform in general design to a standard coal cutter chain. Nominal kerf width is usually 6 in (6.375 in actual), but 7 and 8 in widths are available. Chain pitch shall be compatible with the cutter bar sprockets.

The chain shall be a heavy duty type of the kind rated for 200 to 230 h.p. drive in coal cutting. The normal working load will involve an average tension of almost 10,000 lbf.

All rollers, sliding surfaces, and other surfaces of contact shall be of sufficient hardness to assure high durability, at least equal to the durability of existing coal cutter chains.

Chain speed will be in the range 600 to 800 ft/min.

The cutter chain shall be fitted with mounting boxes for replaceable cutting tools. Cutting tools shall be specified by the design engineers. Positions and angles of pick boxes shall be in accordance with the specifications of the design engineer.

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CARRIER VEHICLE SPECIFICATIONS

Vehicle Type:

Two unit tracked, articulated vehicle with freedom in roll and controlled pitch and yaw between units. The basis for the proposed vehicle concept is the Food Machinery Corporation Model 200 Logging Vehicle which utilizes components which are geometrically similar to those used on the Army's M113 Personnel Carrier also produced by FMC. The logger components are constructed of steel rather than aluminum increasing durability and weight, both of which are important to the carrier.

Vehicle Dimensions:

Front Unit:	195" long
	103" wide
	102" high
	40" bed height
Rear Unit:	195" long
	103" wide
	57" maximum bed height

Distance Between Units: Approximately 70"

Overall Length: 460"

Weight:

Front Unit: Normal GVW: 24,500 lbs.

Maximum GVW: 29,000 lbs. (4500 lbs. ballast)

Rear Unit: GVW: 24,500 lbs.

Track and Suspension:

Type of Suspension: Road wheels suspended by torsion bars, no support rollers, sprocket location either front or rear dependent upon design simplicity.

Type of Track: Forged steel grousers, single pin shoe with rubber bushed hinge pin. No track pad.

Track Shoe Width: 22"

Length of Track On Ground: 113"

Area of Track On Ground: 9944 in^2

Ground Pressure: Nominal 4.93 psi

Maximum, Front Unit: 5.83 psi

Road Wheels: Type: Solid Steel

Size: 24.5" diameter

Number per side: 5

Drive System:

Normal: Each unit driven by a 20 HP hydraulic motor which is fed by means of an umbilical from a surface support barge. The motors drive the track through a speed reducer, a differential, and a final drive.

Extreme Soft Soil Conditions: The tracks are driven by the hydraulic motors to overcome rolling resistance while the vehicle is propelled with an on-board winch.

Steering:

The vehicle is steered by means of a hydraulically actuated articulation joint that can be controlled either on-board at the operator station

or from a remote site. To assist in remote steering, vehicle attitude sensors will allow monitoring of the pitch angle, the roll angle, and the yaw angle between the two units. In addition a television camera system will permit visual monitoring of vehicle course, bottom profile, and vehicle attitude.

Materials:

The fact that the vehicle is to operate undersea requires that care be taken in the selection of materials to avoid or reduce corrosion from sea water minerals. Intimate contact of non-similar metals will be avoided at all costs and materials sensitive to corrosion will either be adequately coated, sealed, or not used at all.

Vehicle Drive System

The vehicle tracks shall be powered by hydraulic motors. The drive system shall be sized to develop 20 hp per unit (40hp total) to propel the vehicle at maximum speeds of 50 ft/min while operating in soft sand. The speed of the vehicle shall be controlled remotely from the main operating console, with provisions for a start-stop override control on the vehicle control panel. The drive system must also be capable of developing 12,000 lb tractive force (6000 lb per unit) with continuously variable vehicle speeds from 0-10 ft/min. These speeds shall also be manually adjusted from the surface platform control panel, with start-stop override controls on the vehicle control panel. In addition, adjustable pressure controls for the track drive system shall be provided on the surface platform to prevent excessive track slippage or stalling the cutter chain. The drive system may consist of a hydraulic motor for each track or one for each vehicle, driving through a differential. Hydraulic motors may be low-speed high-torque types or high-speed low-torque types with appropriate gear reductions. The track drive shall be capable of being operated simultaneously with the Kedging winch when the trencher is operating in very soft bottom materials.

Kedging Winch System

The winch system shall be able to provide a pull of 20,000 pounds at a maximum speed of 10 ft/min. It shall consist of the following components:

1. 250 feet of stainless steel cable 1 inch in diameter and 6 x 37 construction.

2. A cable reel with an 18 inch minimum diameter drum and of sufficient width to provide 18 level wound wraps of cable. (18.5 inches between flanges.) The reel will be mounted on the front of the forward unit. The entire assembly shall be designed for a minimum load of 60,000 lb. No level winding device should be necessary. The material of construction should withstand the corrosive nature of the environment.

3. A 3:1 stainless steel chain drive sufficient to transmit 8 horse-power at 5.1 input rpm. The chain will pass through the vehicle hull. No lubrication should be necessary.

4. An additional 117:1 gear reducer shall be mounted within the vehicle hull and connect to the chain drive. A pressure compensator shall be employed to keep the reducer housing under a constant inner pressure higher than the surrounding hydrostatic pressure. Output torque shall be at least 6100 ft-lb.

5. A low speed high torque hydraulic motor shall be coupled to the gear reducer. The minimum requirements shall be 8 horse-power at 700 rpm. The minimum torque shall be 50 ft-lb at 3000 psi. The motor shall be continuously variable from at least 200 rpm to the maximum rpm by varying the input flow. The motor case shall be kept under a constant internal pressure higher than the surrounding hydrostatic pressure.

The hydraulic motor shall be connected to the main traction drive motor with a flow divider and a control valve located with the other machine controls. The system shall be usable either alone or simultaneously with the main traction drive.

Hydraulic Power Source

The hydraulic power source shall consist of a diesel engine driving one or more hydraulic pumps through a pump drive unit, together with a fuel tank, hydraulic fluid reservoir, filters, and appropriate controls. The engine shall be water cooled. It shall be sized to provide for simultaneous operation of the main cutter drive, vehicle drive, vehicle attitude control systems, vehicle winching system, and hose reels. The main cutter drive requires a constant volume of hydraulic fluid; the vehicle drive, vehicle winching system, and hose reels are variable volume systems; and the vehicle attitude and cutter bar attitude control systems are of the on-off type. The engine shall be designed for stationary operation with appropriate controls, indicators, and safety shut-off devices. The pump or pumps may be either geared or direct coupled; however, an overcenter clutch shall be incorporated to uncouple all pumps from the engine.

Hydraulic controls for all systems shall be mounted on a control panel for convenient operation.

The entire power unit, including controls and hose reels shall be skid-mounted and protected by a weatherproof housing with provisions for 360° operator visibility from the control panel. Doors shall be provided for hose egress, with rollers to prevent damage to the hose or door frames. The complete skid shall be equipped for lifting, with provisions also for tie-downs. Dimensions of the skid shall allow transporting by truck, and in a C-141 aircraft.

Hydraulic Hoses and Hose Reels

Hydraulic hoses shall be selected for operation with petroleum base hydraulic fluids. The hoses shall be of standard construction compatible with the operating pressures and flows involved. Maximum system operating pressure shall not exceed 3000 psi. Hoses shall be sized so that fluid velocities do not exceed 20 ft/sec at maximum flow conditions. The minimum number of hoses compatible with efficient system operation shall be used. Hose lengths shall be such as to allow machine operation 600 ft from the main power unit.

Quick-disconnect hydraulic hose couplings shall be used at the trenching machine connections. These couplings shall be compatible with petroleum base hydraulic fluids. The couplings shall be constructed to provide no oil spill on connecting or disconnecting. A lock shall be provided to prevent accidental disconnection of couplings. The couplings shall be located on the machine to allow connecting and disconnecting by divers while under 150 feet of water. The couplings shall be compatible with system operating pressures and provide minimum pressure drop at maximum flow rates.

Separate hose storage reels shall be provided for each hose type. Each reel shall be sized to store the complete length of hose, and the core diameter shall be compatible with the minimum hose bend radius. The reels shall be hydraulically powered and have continuously variable manual speed control to allow payout and take-up speeds of zero to 50 ft/min, with the hoses dragging through soft sand. Hydraulic swivel connections shall be provided to allow reeling hose while oil is being pumped through it.

Cable Trough and Feed Asserbly

The cable trough and feed assembly shall consist of three sections.

1. The forward trough section shall be mounted on the superstructure of the forward vehicle. The trough shall be of open construction with wall spacing to accomodate at least a 4-inch diameter cable. Nylon rollers shall be appropriately spaced along the bottom of the trough.

2. The rear trough section shall be of identical construction as the forward unit. It shall be mounted on the rear vehicle with adequate spacing to prevent interference with the forward trough and to allow articulation between the vehicles.

3. The feed section shall be mounted on the cutter bar and shall have a maximum width of 6 inches. It shall be able to move with the cutter bar and permit the laying of cable with the cutter bar in a vertical position and at a maximum depth of 7 feet. A removable backfill blade shall be attached to the feed shoe assembly. Hinged retainer rollers shall be employed at appropriate positions along the upper portion of the troughs and feed shoe. This will permit the cable to be laid into the trough and then restrained from climbing out. Construction material should be either aluminum or stainless steel. A minimum bending radius of 2½ feet for the cable shall be maintained throughout the trough and feed assemblies.

A light hoist shall be mounted on each vehicle to assist in raising the cable and placing it in the trough and feed shoe assemblies.

Controls and Instrumentation

The following instruments, in addition to TV monitors, are required on the surface control platform to monitor vehicle attitude and operation:

1. Pitch and roll of both forward and aft halves of the vehicle.
2. Yaw, or angle between the vehicles while turning.
3. Swing angle of cutter bar from vertical.
4. Cutter bar operating depth.
5. Trenching chain speed.
6. Trenching motor torque (hydraulic pressure).
7. Vehicle traverse speed.
8. Vehicle tractive effort (hydraulic pressure).

The following controls are required on the surface control platform to control vehicle attitude and operation:

1. Positive pitch, roll, and yaw control of the vehicle.
2. Cutter bar operating depth.
3. On-off control of trencher chain motor.
4. Torque (hydraulic pressure) control of trencher mtoor.
5. Vehicle traverse speed and direction.
6. Vehicle tractive effort (hydraulic pressure).

The following controls are required on the vehicle itself for diver operation of the vehicle:

1. Positive pitch, roll, and yaw control of the vehicle.
2. Cutter bar operating depth.
3. On-off control of trencher chain motor.
4. Forward, reverse, or stop control of vehicle.

Closed circuit TV

A closed circuit TV system shall be provided for monitoring machine behaviour and local environment when undersea visibility conditions permit. The underwater camera shall be mounted in a waterproof housing capable of performing at depths of 150 ft or more. The camera shall have remote control of pan and tilt movements, focus, zoom and iris. A suitable source of illumination shall be provided for night operation. There shall be a monitor set and control panel for use at the surface.

Electrical cable for control and monitoring

The umbilical shall include an electrical cable for control and monitor functions. The cable shall be an underwater steel jacketed cable with waterproof connectors. A rated strength of approximately 10,000 lbf is required. The cable should have conductors for the following circuits: TV signal (coax), pan, tilt, focus, zoom, iris, cutter position sensors, vehicle attitude sensors, underwater light, 6 reserve circuits.

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