CRREL
REPORT 76-47

Development of large ice saws

Cover: Modified coal saw attachment removing ice from lock wall at Sault Ste. Marie, Michigan. (Photograph by Ben Hanamoto.)

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Development of large ice saws

D.E. Garfield, B. Hanamoto and M. Mellor

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This report describes two mechanical ice-cutting systems for the removal of ice collars at the high pool level on the Poe Lock of the St. Marys Falls Canal at Sault Ste. Marie, Michigan. One system was a narrow-kerf (3¼-in.-wide) coal-cutting chain saw mounted on a bar, driven by a 65-hp wheeled trencher. The other system was a lumber-cutting chain saw mounted on a bar, driven by a 30-hp wheeled soil trencher which cut a 0.56-in.-wide kerf. The lumber-cutting saw's bar was too flexible and the desired cutting traverse speed was not met. The coal-cutting saw cut 6-ft-deep ice collars at traverse speeds of up to 10 ft/min and is acceptable. With a few modifications, the coal-cutting saw would be operational.

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PREFACE

This report was prepared by Donald E. Garfield, Research Mechanical Engineer, of the Engineering Services Branch, Technical Services Division; Ben Hanamoto, Research General Engineer, of the Applied Research Branch, Experimental Engineering Division; and Dr. Malcolm Mellor, Physical Scientist, Experimental Engineering Division, U.S. Army Cold Regions Research and Engineering Laboratory.

This study was conducted under Civil Works Code 030601, CWIS 31334, Program: *Ice Engineering Subprogram, Ice Formation*; Work Unit title: *Preventing and Removing Ice from Adhering to Lock Walls and Gates.*

Technical review of the report was performed by G. Frankenstein and K.L. Carey of CRREL.

The authors acknowledge with thanks the contributions of L. Gould, E. Perkins, and W. Burch, who helped greatly in meeting the project deadlines. The assistance provided by R. Wiinamaki of the Sault Area Office is also gratefully acknowledged.

TERMINOLOGY

Esplanade – The level, unobstructed area next to the lock walls suitable for driving vehicles and equipment along the locks. A fabricated steel curbing approximately 8 in. high borders the lock side of the esplanade.

Included angle – The angle between the leading face and the relief face of the cutter.

High pool elevation – The water level within the lock walls when it is the same level as the water in the canals upstream of the lock.

Overbreak – A cutting tool working in brittle material excavates a groove that extends beyond the limits of the area swept by the cross section of the tool. The difference between the total groove volume and the volume swept out by the tool itself is called overbreak.

Rake angle of cutter – The angle between the leading face of the cutter and a normal to the cut surface at the cutting edge.

Relief angle or *clearance angle* – The angle between the flank or relief face of the cutter and the tangent to the cut surface at the cutting edge.

CONVERSION FACTORS: U.S. CUSTOMARY TO METRIC (SI) UNITS OF MEASUREMENT

These conversion factors include all the significant digits given in the conversion tables in the ASTM *Metric Practice Guide* (E 380), which has been approved for use by the Department of Defense. Converted values should be rounded to have the same precision as the original (see E 380).

Multiply	*By*	*To obtain*
inch	25.4*	millimeter
foot	0.3048*	meter
inch/minute	0.4233333	millimeter/second
foot/minute	0.00508*	meter/second
foot/hour	0.3048*	meter/hour
degree	0.01745329	radian
revolution/minute	0.1047	radian/second
pound force	4.448222	newton
pound force/inch^2	6894.757	pascal
pound force/foot^2	47.88026	pascal
horsepower	745.6999	watt

* Exact.

DEVELOPMENT OF LARGE ICE SAWS

Donald E. Garfield, Ben Hanamoto
and Malcolm Mellor

INTRODUCTION

In order to maintain year-round navigation in the Great Lakes system, the locks at Sault Ste. Marie, Michigan, have to be operated throughout the winter. One of the problems that arises during winter operation of the locks is that ice collars form on the lock walls near high pool elevation, reducing the effective width of the locks. The widest lock, Poe Lock, is 110 ft wide, only 5 ft wider than the widest ships on the lakes, the *Roger Blough* and the *Presque Isle* (Fig. 1). Without remedial action, the total width or thickness of the ice collars can easily exceed 5 ft, thus making navigation of the lock impossible.

Two processes form ice collars: the direct freezing of water on cold walls, and the crushing of floating ice against the walls by ship hulls. In both cases, adhesion to the lock wall depends on the temperature of the wall being below 0°C. Where water level is constant immediately above and below the mitre gates, the ice collars that form at the splash line are small (Fig. 2). However, at Soo Locks the water level in the active lock is always held well below (5 to 17 ft below) high pool elevation; consequently, the ice collars that form on the air-cooled wall are very deep (Fig. 3), commonly 8 ft deep or so.

As part of the Great Lakes Winter Navigation Program, CRREL has been developing methods for clearing ice collars from lock walls. The methods considered for keeping walls clear of ice have included heating, coating with chemicals, covering with inflatable deicing boots, cleaning with high-pressure water jets, and mechanical cutting. This report deals with the development of large chain saws for cutting off ice collars.

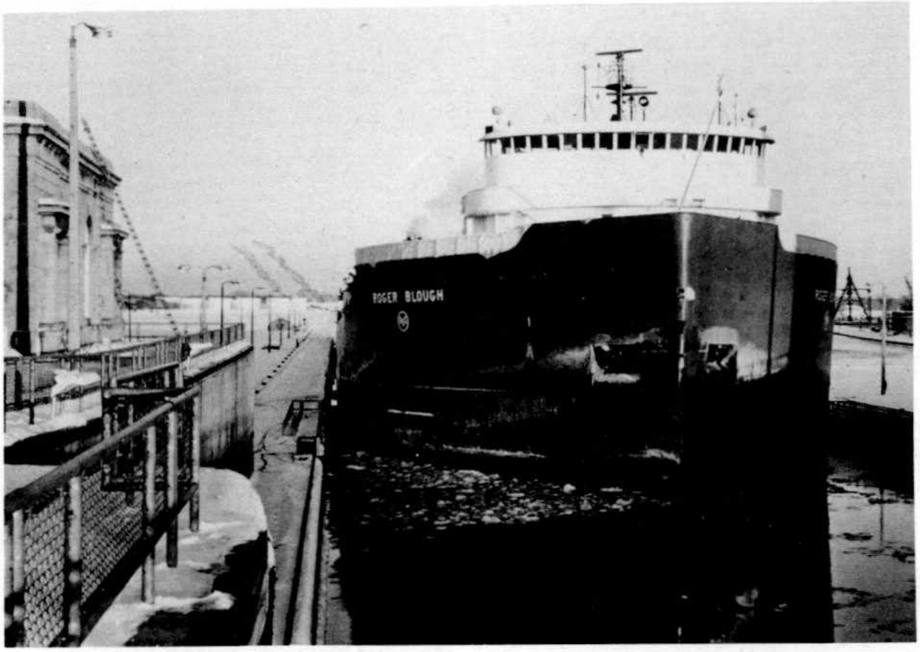
PERFORMANCE SPECIFICATIONS

Performance specifications of chain saws were developed at CRREL through consultations with people familiar with the overall problem. The chief difficulty was in deciding the depth of ice collar that had to be tackled. Direct observations at the site gave the impression that the thickness of ice collars could perhaps be reduced to about 2 ft by maintaining water level near high-pool elevation during the very long periods between ship passages. However, such a procedure was not in operation at the time of the study; therefore, the saws were designed to cut an average thickness of 6 ft of ice, with provision for cutting through 8 ft or so at reduced travel speed.

The specification adopted for the saw's rate of working was highly arbitrary. It was assumed that a single machine ought to be capable of clearing ice from both walls of the lock each day. This was taken to mean that the saw had to cut at least 2000 linear feet in 8 hours. (The complication of how to transfer a machine from one side of the lock to the other was ignored at this stage.) However, from preliminary design calculations, this requirement seemed too lax in that it could probably be met without much effort. It was therefore changed to one for cutting 1000 linear feet of 6-ft-deep ice in 1½ hours, which seemed to be a realistic goal for efficient design.

DESIGN CONCEPT

The general idea was to use a chain saw moving parallel to the lock wall to slice through the ice within a few inches of the ice/wall interface. The working section of the chain saw bar had to be at least



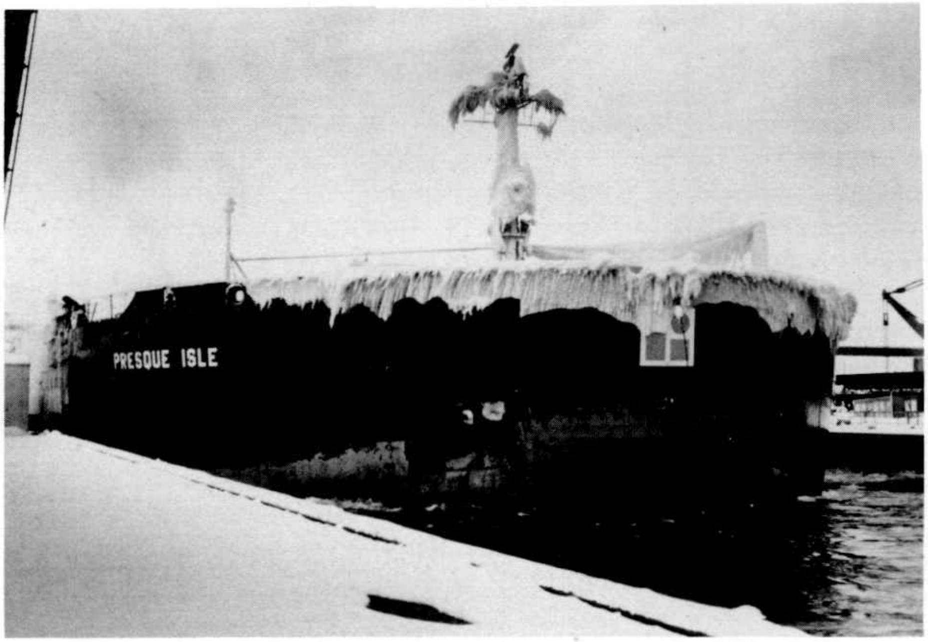


Figure 1. Ore-carriers Roger Blough *and* Presque Isle *at Soo Locks.*

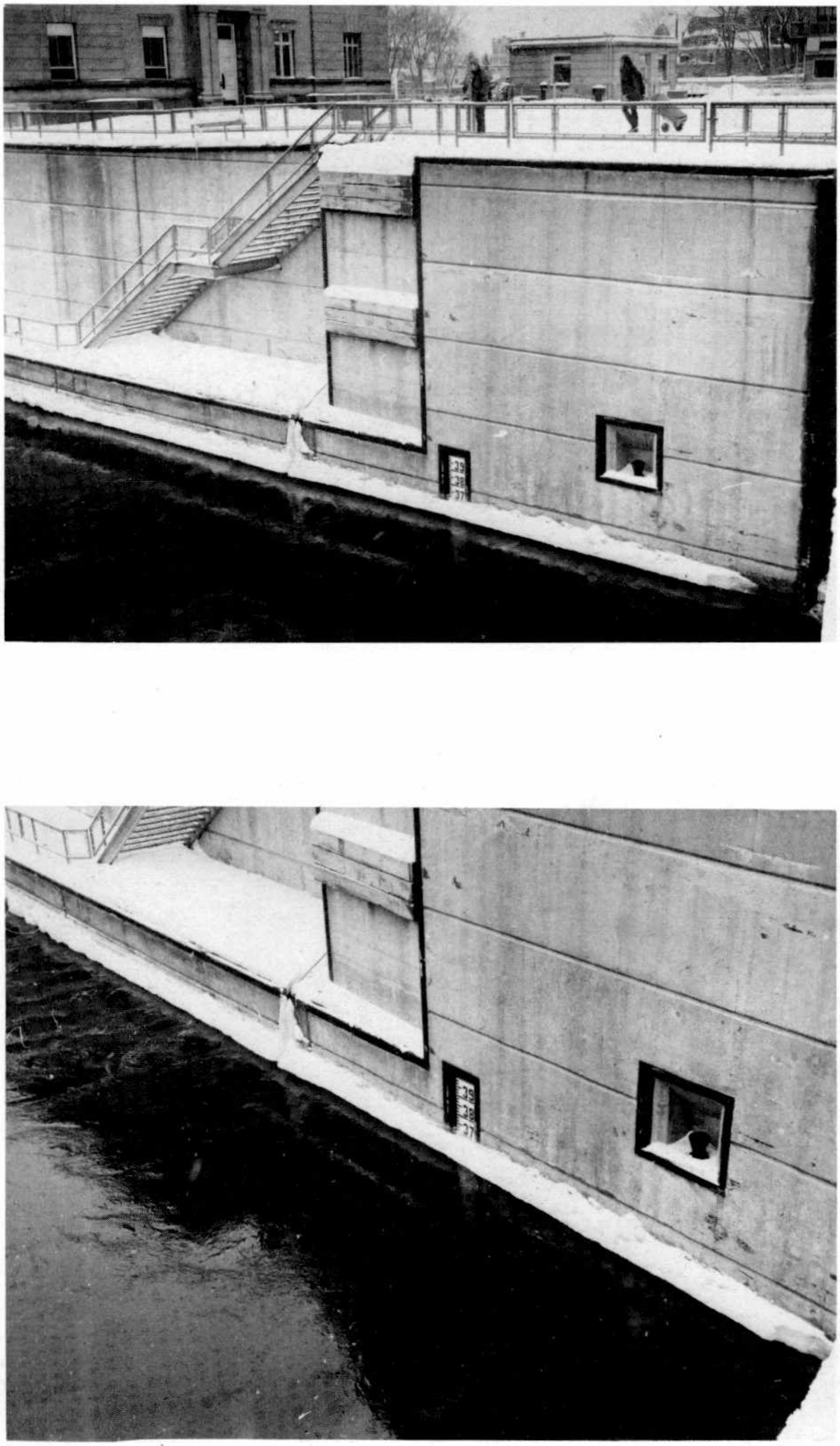


Figure 2. Thin ice collars formed downstream of lower mitre gate, where water level does not fluctuate much.

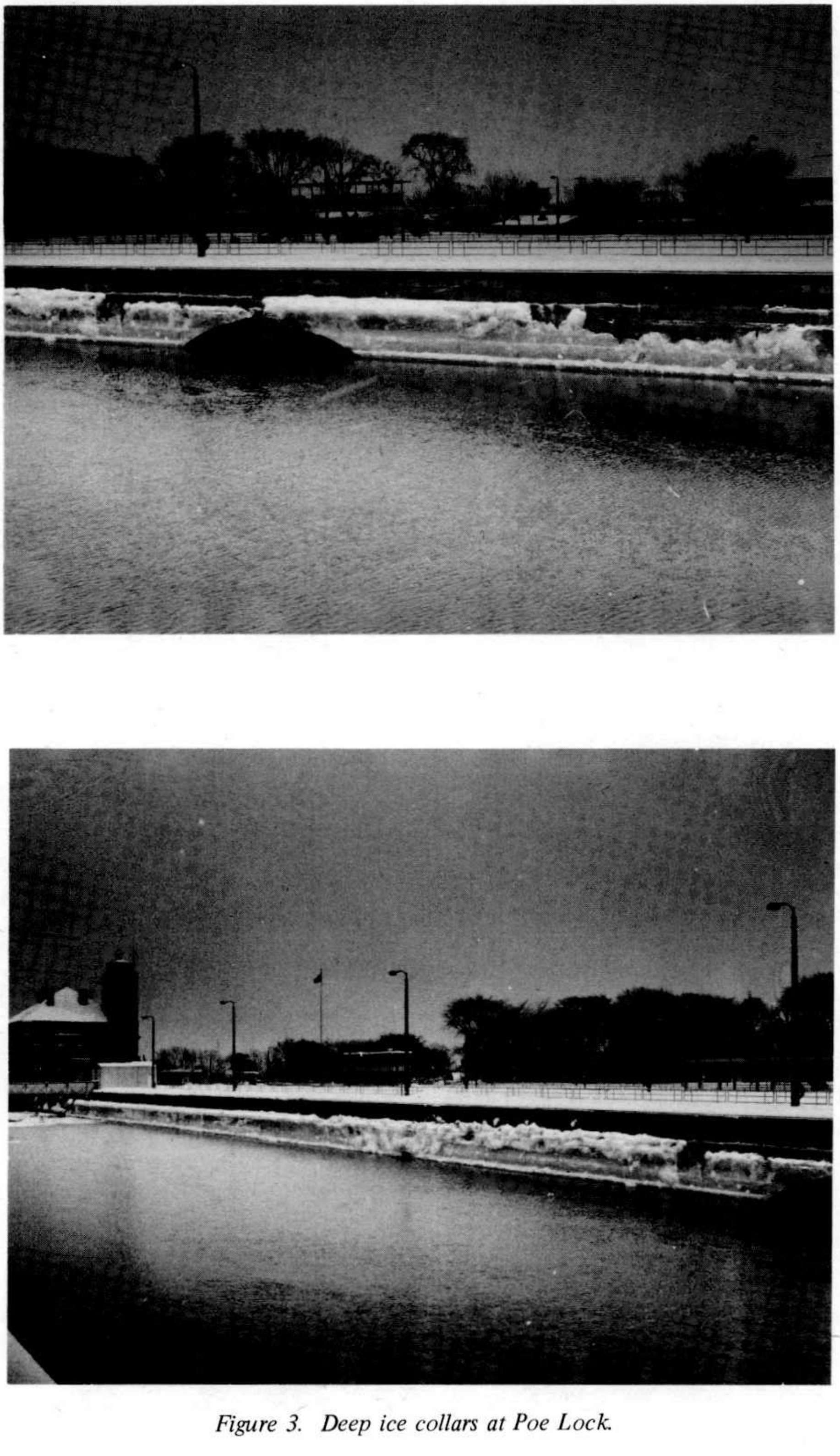


Figure 3. Deep ice collars at Poe Lock.

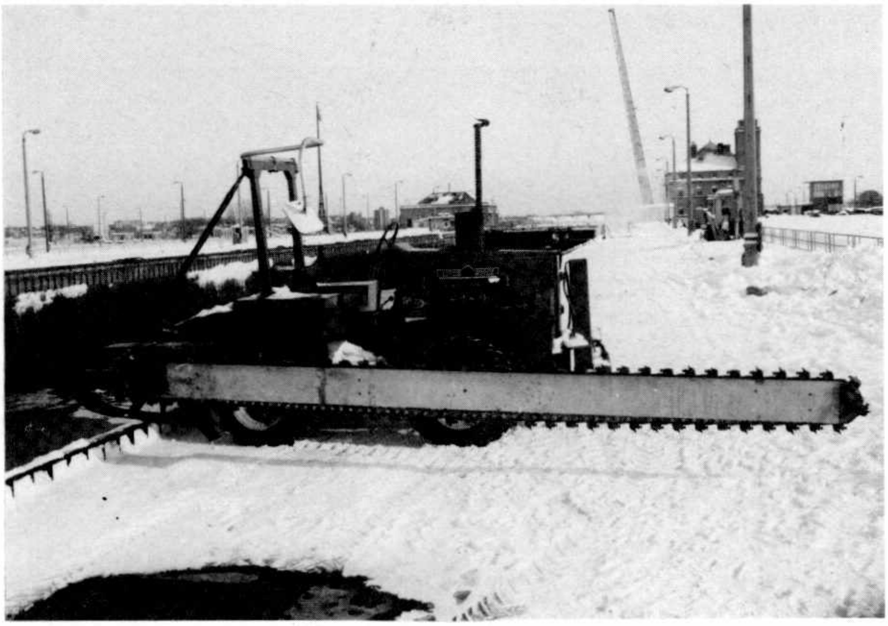


Figure 4. Thin-kerf coal saw mounted on offset-drive soil trencher.

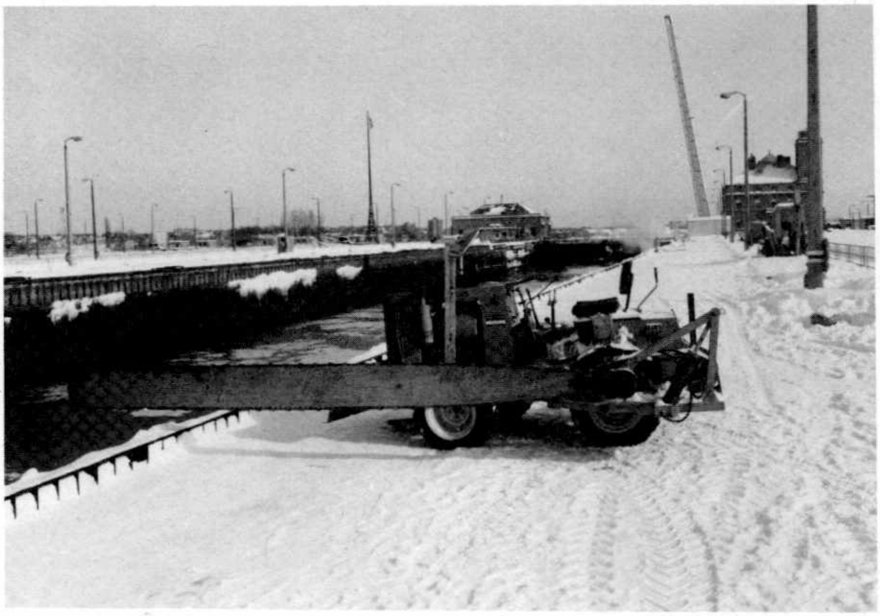
8 ft long to slice through 8 ft of ice at a single pass, and additional length was needed to accommodate the drive mechanism. There were two possible arrangements: 1) to have the shortest possible bar held into the work by a boom, with electric or hydraulic power supply; or 2) to have a long bar reaching down from the carrier vehicle, with direct drive from that vehicle. The latter seemed the more practical arrangement, even though it required a very long bar (at least 8 ft of cutting length, about 5 ft of reach from the top of the ice collar to the level of the esplanade pavement next to the wall, and additional reach from the pavement level to the chain saw drive shaft on the carrier).

Ideally, the chain saw should cut a very narrow kerf in order to minimize force levels and power requirements. On the other hand, the bar should be reasonably stiff, and the cutting chain should be capable of withstanding occasional abuse. The type of saw selected for the main development effort was a "thin kerf" coal saw that is used in small underground mines. This was mated with a small rubber-tired soil trencher that had an offset drive (Fig. 4). For a secondary effort, a large lumber-cutting chain saw (deck or pond saw) was mounted on a very small soil trencher (Fig. 5). Both machines were intended to travel along the lock esplanade with their saw bars hanging down with a forward inclination of about 20° to the vertical (Fig. 6).

Details of the coal saw machine

The coal saw, the heavier of the two machines, had as its cutting unit a coal saw manufactured by the Bowdil Company. The overall length of the bar was 16 ft, the bar width to the chain guides was $9^1/_2$ in., and the actual thickness of the bar was $1^5/_8$ in. The width of kerf cut by the gage cutters of the chain was $3^1/_4$ in., and the bar width measured to the tips of the cutting teeth was 15 in. The pitch diameter of the chain's drive sprocket was 8.09 in. The reversible and replaceable cutting teeth had rake angles of +10° and relief angles of 50°, with the included angle 30°. The maximum gage (maximum chipping depth) was 0.9 in. for an unworn tool. A link of the chain fitted with a cutting tooth is shown in Figure 7. The pitch of the chain was $2^1/_2$ in., and there was a cutting tooth every 5 in. The cutting teeth were angled out of the plane of the bar to give 7 cutting tracks across the width of the kerf.

The large chain saw was mounted on the tractor of a soil trencher, the Ditch Witch R65. Preliminary design calculations indicated that power requirements for cutting a $3^1/_4$-in. kerf in 6 ft of ice at a traverse



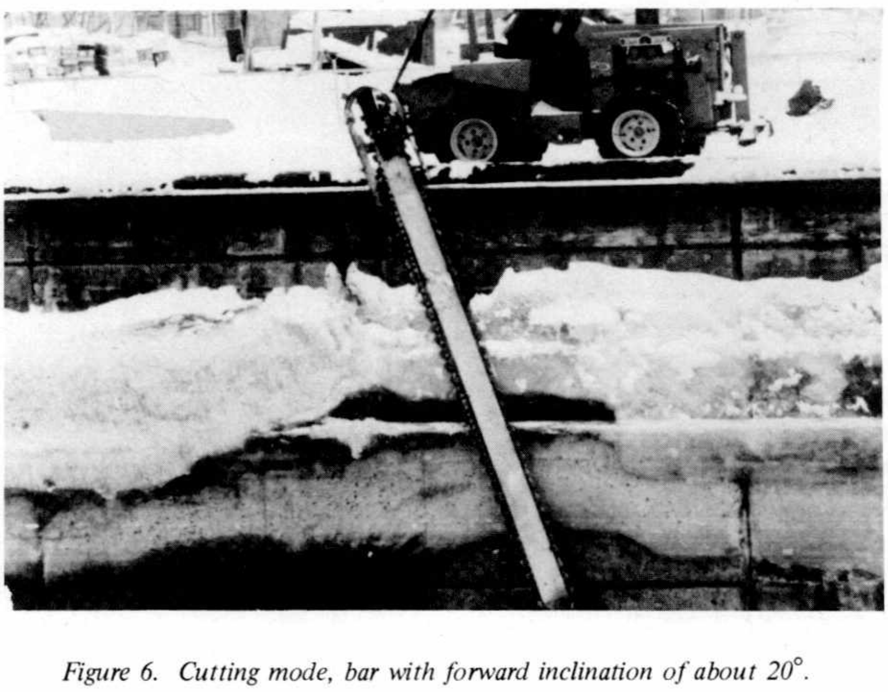


Figure 6. Cutting mode, bar with forward inclination of about 20°.

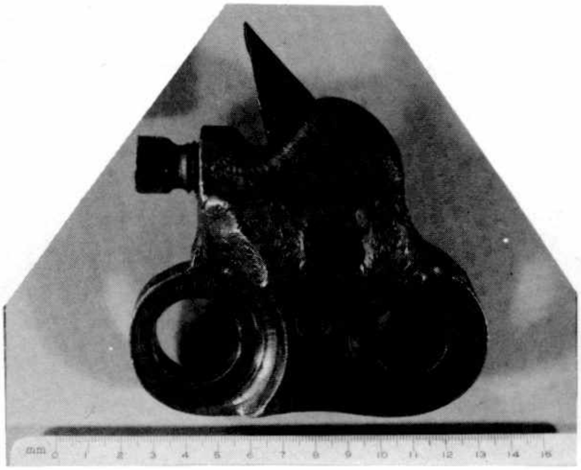


Figure 7. Chain link and replaceable hardfaced cutting tooth for coal saw.

rate of 12 ft/min would call for a minimum of 50 hp, assuming a specific energy of 500 lbf/in.[2] for cutting ice. Action was initiated to purchase a tractor unit, with factory modifications to adapt the Bowdil coal saw. When it became obvious that this could not be accomplished in time to meet the test schedule, it was decided to purchase a tractor without attachments and adapt the saw at CRREL.

A Ditch Witch Model R65 and offset trenching attachment, manufactured by Charles Machine Works, Inc., Perry, Oklahoma, were purchased. This tractor, without attachments, weighs 4100 lb and has a wheelbase of 59 in., an overall length of 138 in., an overall width of 72 in., an overall height of 95 in., and a width of 64 in. at the outside of standard 32×9:00×16 tires. The tractor has a 65-hp Wisconsin air-cooled engine that delivers 63 hp at 2600 rev/min. The trencher is driven through a 4-speed transmission, which rotates the output shaft at no-load speeds of 35 rev/min in first, 77 rev/min in second, 137 rev/min in third, and 241 rev/min in fourth gear. Tractor groundspeed in the trenching mode is continuously variable from 0 to 37.5 ft/min. The tractor is equipped with full-time 4-wheel drive and power steering.

With the 8.090-in. pitch diameter sprocket furnished by Bowdil Company, the maximum chain speed attainable was about 510 ft/min. Bowdil Company had advised that the maximum chain speed should be limited to 200 ft/min, so the 510 ft/min was a compromise between the original 600 ft/min requirement and Bowdil's recommendation.*

The factory-made offset trenching attachment was designed for a bar offset of about 27½ in. from the tractor centerline. For operation on the locks, an additional offset of approximately 28 in. was required. This was accomplished by welding a flange to the existing trencher pivot assembly and cantilevering a torque tube from this flange. This torque tube could be rotated to provide vertical bar movement by using the original boom hydraulics. An outer sleeve bearing was braced to the front of the tractor and to the top of the rollover protective frame to provide additional support near the outer end of the torque tube. A drive shaft extension ran through the center of the torque tube and was supported by a ball bearing at the end of the torque tube. A bell crank was bolted to the end of the torque tube. The chain saw bar was bolted through slots in the bell crank arm, which provided chain tension adjustment.

Preliminary calculations indicated that the tractor hydraulics might not raise the chain saw bar to the horizontal position. When the unit was assembled the evening before it was to be shipped, it was discovered that the bar could not be raised to the horizontal position. Therefore, an extension to the cylinder crank arm on the pivot assembly was fabricated to correct this problem. This modification limited bar travel, but was still within design requirements. Drawings of the fabricated components are available at CRREL.

The stability of the tractor during cutting operations was of some concern, since the weight of the bar and chain (approximately 1200 lb), and the weight of the vertical component of the cutting force, would have a tendency to overturn the tractor into the lock. However, calculations showed that, with the outboard tires loaded with calcium chloride solution and additional wheel weights, overturning would not be a problem. Even if the tractor did begin to tip, the end of the bar would contact the lock wall and prevent further tipping.

** Coal cutter chains are routinely designed to run at speeds up to 850 ft/min.*

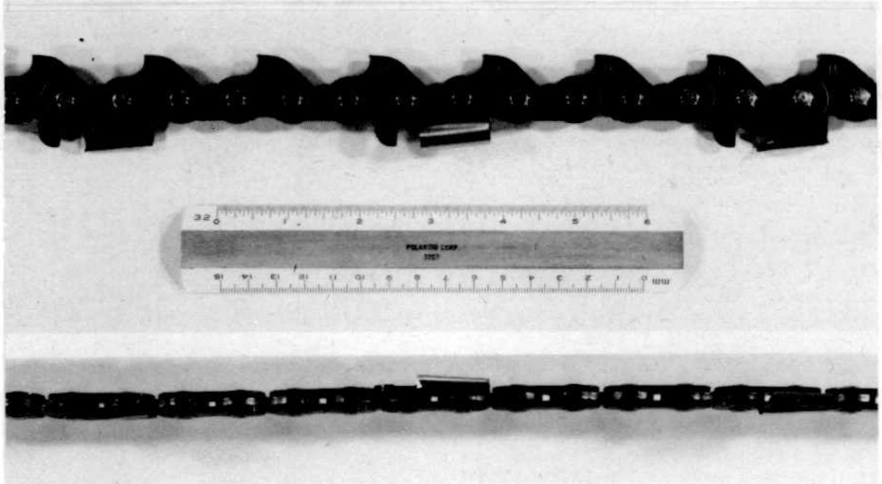


Figure 8. "Skip tooth" modified lumber chain.

Details of the lumber-cutting saw machine

The cutting unit on the small machine was basically a logging industry deck saw. It was supplied by the L-M Equipment Company, Inc., Portland, Oregon, and had a chain made by the Oregon Chain Saw Division of Omark Industries, Portland, Oregon. The chain was a "skip-tooth" modification of an Oregon 11BC chipper chain, which had a 0.78-in. pitch and a tooth every 4.7 in. along the chain (alternately left and right teeth). Most of the teeth had the top parts ground away (Fig. 8) so that they only cut at the gage limits. A pair of original teeth were left intact after each set of 2 reground teeth. The effective rake angle was approximately $+25^\circ$ and the relief angle was 5°. The overall length of the cutter bar was 14 ft 4 in., its width was $11\frac{3}{4}$ in., and its thickness was approximately $^{5}/_{16}$ in. The kerf cut by the chain was 0.56 in. wide, and the maximum chipping depth, determined by the projection of the cutting edge beyond the raker, was 0.06 in.

Preliminary design calculations indicated that approximately 6.5 hp would be required to cut a $^{9}/_{16}$-in. kerf in 6-ft-thick ice while the saw was travelling at the rate of 12 ft/min. Allowing for inefficiencies in the system, the minimum horsepower required was 10 to 12 hp. CRREL had a Ditch Witch Model V30 trencher, which seemed ideal for mounting the pond or deck saw.

The V30 trencher weighs approximately 2800 lb and is powered by a Wisconsin VH4D 30-hp air-cooled engine. The overall tractor width is 64 in. and the overall length is 118 in. without the saw or digging attachments. The wheelbase is $46^{3}/_{8}$ in., and the width to the outside of the $27 \times 8.50 \times 15$ tires is 56 in.

The saw mandrel and V-belt drive were mounted on a frame which was hinged to the side of the tractor. An adjustable top link assembly allowed for fixing the saw bar in a position parallel to the lock wall. This arrangement also would provide a method for automatically keeping the bar parallel to the lock wall by using a servo-controlled hydraulic cylinder in place of the top link assembly. Such an arrangement is desirable if the esplanade pavement next to the wall is not kept completely free of ice and snow. Otherwise, any vertical movement of one vehicle tire with respect to the other results in a greatly amplified horizontal movement of the tip of the saw bar, and may cause the chain to gouge the lock wall, either damaging the chain or the wall. Tractor hydraulics were utilized to raise and lower the bar. A four-bar linkage arrangement was designed to rotate the bar through 110° from approximately 5° above horizontal to 15° beyond vertical. In its original configuration, the trencher drive rotated at a maximum speed of 230 rev/min. To operate the chain at the desired 1200 ft/min, the trencher drive pulleys were interchanged and an additional speed increase was provided with a second V-belt drive.

FIELD TESTS

Tests were made on the Poe Lock at Sault Ste. Marie, Michigan, during early February 1976. Both saws were hoisted onto the esplanade of the south wall (i.e., north-facing, or shaded, wall) from a barge operated by the Sault Area Office. Test runs were made when there was a heavy ice collar extending from about 5 ft below the esplanade pavement to about 13 ft below the esplanade pavement. The ice was the dense, impermeable kind, as distinct from the weaker, and more permeable, material crushed onto the wall after recent ship passage. Air temperatures during the test period were in the range -9° to -22°C.

To start a fresh cut, the saw bar was lowered slowly from its horizontal stowed position, so that the nose and rear side of the bar were cutting into the ice collar. During this starting cut, the untensioned side of the chain was working, and there was a potential for throwing the chain out of its guides. Therefore, it seemed beneficial to inch the tractor forward occasionally during the course of a starting cut. When the bar reached its normal operating position, the tractor began its forward travel, and the side of the chain tensioned by the drive sprocket did the cutting.

Testing was simply a matter of determining the maximum sustainable travel speed of the tractor when the saw was cutting. The limit of speed in these tests was set by available engine power; the saw drive began to lug down as travel speed became too high. Under other circumstances the limit might be set by available traction, by high force levels, or by inadequate clearing of cuttings.

An unsuccessful attempt was made to determine the normal component of force on the large cutter bar. The plan was to tow the carrier vehicle through a dynamometer, both with the saw operating in a vertical position and with the saw withdrawn from the work. However, the small tractor, which was the only tow vehicle available, was unable to pull the larger tractor when its saw was operating (the hydraulic drive motor ports are blocked off, preventing rotation of the drive train).

The heavier saw (coal saw) was able to travel at 9 to 10 ft/min while cutting through ice 4 to 6 ft thick, and at 6.5 to 6.7 ft/min while cutting through ice 7 to 8 ft thick. These rates were achieved with the drive shaft running at its maximum speed of about 235 rev/min in fourth gear.

The small (lumber-cutting) saw could cut through 5 to 6 ft of ice at speeds of 2 to 3 ft/min with the tractor in third gear and the output shaft turning at 610 rev/min. However, the flexible bar of the small saw was easily deflected, and it tended to ride out of the work at transitions from a thin section of ice to thicker ice.

DESIGN CALCULATIONS AND PERFORMANCE ANALYSIS OF COAL SAW

Chipping depth

During the initial design of the heavier saw, a capability for forward travel at 12 ft/min was proposed to permit clearing of a 1000-ft-long wall in 1½ hours, allowing 7 minutes for stoppages. It was expected that a chain speed of 600 ft/min would be available, and it was expected that the cutting teeth of the large saw would be set to give 5 cutting tracks across the width of the kerf, instead of the 7 tracks as furnished. The planned operating position for the saw was at an inclination of 70° to the horizontal.

The theoretical chipping depth ℓ of the teeth is given by

$$\ell = \frac{U}{u_t} S \sin\phi \qquad (1)$$

where U is the traverse speed, u_t is the tool speed (chain speed), S is the distance between tracking cutters, and ϕ is the inclination to the horizontal [see Mellor (1976)]. The Bowdil Company chain has cutters spaced every 5 in.; so with 5 cutting tracks, S is 25 in. Thus, the theoretical chipping depth under the expected conditions is:

$$\ell = \frac{12}{600} \times 25 \times 0.9397 = 0.47 \text{ in.} \qquad (2)$$

The actual chipping depth was expected to be less than this value as a result of overbreak and interaction between adjacent parallel kerfs (see Mellor, in prep.). With an overbreak angle greater than about 41°, adjacent kerfs would have some overlap.

The test machine was prepared hurriedly to meet project schedules, and not until it was finally assembled and run at the test site were its actual characteristics determined. In reality, the cutters on the chain were set to give 7 cutting tracks, and the maximum chain speed reached during working was 492 ft/min. Thus, the theoretical chipping depth was:

$$\ell = \frac{U}{492} \times 35 \times 0.9397 = 0.0668U. \quad (3)$$

At 12 ft/min this would give a theoretical chipping depth of 0.80 in., and at 10 ft/min a theoretical chipping depth of 0.67 in. For interaction between adjacent kerfs, the overbreak angle would have to exceed 19° and 12 ft/min and 22° at 10 ft/min. Although the overbreak angle for the Bowdil Company tool in ice has not been measured, it is expected that it would be at least 50°. With 7 cutting tracks across a 3¼-in. overall kerf, and with an overbreak angle of 50°, the actual chipping depth would not be more than about 0.46 in. With an overbreak angle of 70°, actual chipping depth would not be much over 0.2 in.

Clearance of cuttings

Another consideration is the ability of the chain to convey cuttings out of the kerf without clogging the cutting teeth. Around each tooth there must be enough space to accommodate the cuttings accumulated during a complete sweep through the work. A simple criterion for adequacy of cutting conveyance is given by:

$$(1 - s_t/S)(h_t/d) \geqslant K_b U/u_t \quad (4)$$

where s_t is the equivalent length of the tool, h_t is the equivalent tool height (such that $s_t h_t$ is the volume per unit width of the tool), S is the distance between tracking cutters, d is the depth of cut measured normal to the traverse direction, and K_b is a bulking factor that can be taken as 1.85 [see Mellor (1975) for details].

For the Bowdil Company tool, s_t and h_t are taken as 2.25 and 2.5 in., respectively, and for a 7-track tool layout $S = 35$ in. With these values the condition for adequate conveyance is:

$$(U/u_t)d \leqslant 1.26 \quad (5)$$

where d is in inches. With a 6-ft-deep ice collar and a chain speed of 492 ft/min, the maximum travel speed, according to this criterion, is 8.6 ft/min. With an 8-ft-deep ice collar, the maximum travel speed for adequate cutting clearance is 6.5 ft/min. According to these calculations, the heavy chain saw has cutting clearance arrangements that are barely adequate at the maximum available chain speed, and there is some dependence on spillage and compaction. With the planned chain speed of 600 ft/min and the planned 5-track tool arrangement, cutting clearance would have been just adequate at a traverse speed of 12 ft/min in 6 ft of ice.

Specific energy

During the initial design it had been hoped that the specific energy for the cutting processes could be brought down to approximately 100 to 200 lbf/in.^{2} The test program made no provision for measuring power consumption for the cutting process, but from rough estimates, it appears that energetic efficiency was appreciably worse than had been hoped.

The rated power output of the tractor engine was 63 hp. From this, 5 hp might be subtracted for running the hydraulic system (including the wheel drives), for driving the alternator, etc. Assuming 80% efficiency for the complex mechanical transmission of the power takeoff, this leaves 46 hp for delivery to the drive sprocket of the chain saw. The manufacturer of the coal saw advised that approximately 15 hp would be needed to run the chain without cutting, so that if this is deducted the power available for cutting ice would be 31 hp.

The specific energy E_s is the energy per unit volume of material cut [see Mellor (1975)], which is the same thing as the power P divided by the volumetric cutting rate $\dot{V}$:

$$E_s = \frac{P}{\dot{V}} = \frac{31 \times 3.3 \times 10^4}{(3.5/12) \times d \times U} = \frac{3.51 \times 10^6}{dU} \quad \text{lbf/ft}^2 \quad (6)$$

where actual width of the finished kerf is taken as 3.5 in. and the cutting depth d and the traverse speed U are in feet and feet per minute, respectively. The test results for the thin-kerf coal saw gave values of (dU) from 47 to 60 ft^2/min; the corresponding range for E_s is 7.46×10^4 to 5.85×10^4 lbf/ft^2. In the more familiar units of lbf/in.^{2} (i.e., in-lbf/in.^{3}), the range for E_s is 518 to 406 lbf/in.^{2} These values are higher than had been hoped for, but they are much lower than the specific energy estimated from the performance of a Joy 10RU coal saw cutting glacier ice in Greenland (1740 lbf/in.^{2}, without making allowance for losses of energy due to chain friction).

Tooth forces and bar forces

With the sharp, new cutting teeth used during the tests, it is probably realistic to assume that the

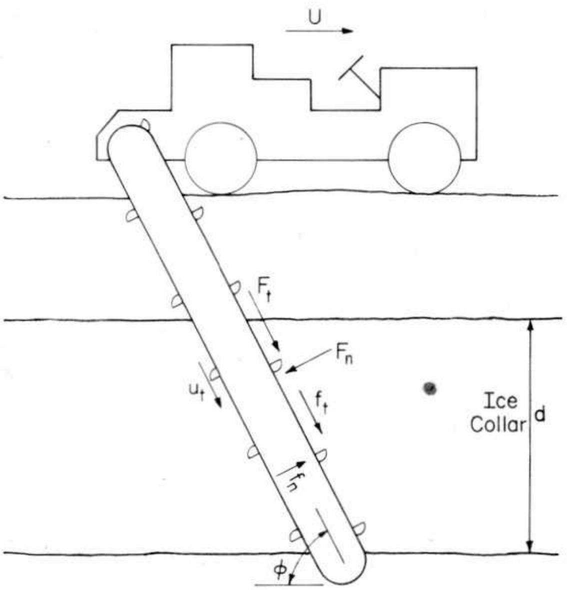


Figure 9. Force diagram.

tangential component of tooth force f_t and the normal component of tooth force f_n are approximately equal [see Mellor (in prep.) for details]. This assumption also implies that the total tangential force on the cutting bar F_t is approximately equal to the total normal force on the bar F_n.

The power needed for cutting is given by the tractive thrust of the tractor multiplied by the tractor speed, plus the tangential force on the cutter bar multiplied by the chain speed (see Fig. 9):

$$P = (F_n \sin\phi - F_t \cos\phi)U + F_t u_t \quad (7)$$

where ϕ is the angle of the cutter bar from the horizontal ($\approx 70°$). Since u_t is about 50 times greater than U, the power consumed in thrusting the cutter bar horizontally $[(F_n \sin\phi - F_t \cos\phi)U]$ can be neglected. If 31 hp is used in the cutting process when operating at maximum speed, and if the chain speed u_t is 492 ft/min, the tangential bar force is:

$$F_t = P/u_t = (31 \times 3.3 \times 10^4)/492 = 2079 \text{ lbf.} \quad (8)$$

As indicated earlier, it is assumed that F_n is equal in magnitude to F_t. Thus, the net tractive thrust H that the tractor has to provide is:

$$H = F_n \sin\phi - F_t \cos\phi = 2079(0.9397 - 0.3420)$$

$$= 1243 \text{ lbf.} \quad (9)$$

The vertical downpull on the tractor V, excluding bar weight, is:

$$V = F_n \cos\phi + F_t \sin\phi = 2079\ (0.3420 + 0.9397)$$

$$= 2665 \text{ lbf.} \quad (10)$$

The mean tangential force on an individual cutting tooth when the saw is being operated at maximum performance is F_t divided by the number of teeth in the work:

$$f_t = f_n = \frac{F_t}{(d/\sin\phi)/(5/12)} = 866 \sin\phi/d \quad \text{lbf} \quad (11)$$

where d is the depth of the ice collar in feet. With $\phi = 70°$, tooth force f_t or f_n is 203 lbf when $d = 4$ ft and 102 lbf when $d = 8$ ft.

DESIGN CALCULATIONS AND PERFORMANCE ANALYSES – LUMBER-CUTTING SAW

Chipping depth

With its output shaft turning at 880 rev/min, the chain speed on the lumber-cutting saw was 1152 ft/min. Regarding the cutting teeth as simply left or right, without taking account of tooth width after grinding, the space between tracking cutters S is 18.75 in. At a bar angle of 70°, the theoretical chipping depth is therefore 0.031 in. at a traverse speed of 2 ft/min and 0.046 in. at a traverse speed of 3 ft/min.

Clearance of cuttings

As previously defined, with an equivalent length of tool, $s_t = 1.0$ in., equivalent tool height $h_t = 0.5$ in., distance between tracking cutters $S = 18.75$, bulking factor $K_b = 1.85$, the condition for adequate conveyance is:

$$(U/u_t)d \leqslant 0.256. \quad (12)$$

When the tractor is operated in third gear with a drive-shaft spread of 610 rev/min, the chain speed u_t is 798 ft/min. With the saw cutting an ice collar with a depth d of 6 ft, the maximum travel speed U is 2.84 ft/min. In fourth gear with a drive-shaft spread of 880 rev/min, the chain speed increases to 1152 ft/min and U is 4.09 ft/min for adequate cutting clearance. The small lumber saw therefore has cutting clearance arrangements which are not adequate to meet the proposed 12-ft/min maximum traverse speed. The maximum recommended chain speed is 1200 ft/min.

Specific energy

The tractor used to power the lumber-cutting chain saw was rated at 30 hp. While the saw is in the cutting mode, other power requirements include the power required for running the hydraulic drive system and that required for running the accessories. Assuming 5 hp is required for the above, an 80% efficient power train to the power takeoff, a 90% efficient saw mandrel and drive pulley system, the available power to the sprockets is about 18 hp. If 3 hp is needed to spin the chain, 15 hp is available for cutting. The volumetric cutting rate $\dot{V}$ is kdU, with the cutting kerf $k = {}^9/_{16}$ in. Then the specific energy E_s is:

$$E_s = \frac{P}{\dot{V}} = \frac{15 \times 3.3 \times 10^4}{(0.5625/12)(dU)} = \frac{1.06 \times 10^7}{dU} \quad \text{lbf/ft}^2. \qquad (13)$$

At the speed where adequate cutting conveyance occurs, 4 ft/min and through 6-ft ice, $E_s = 4.40 \times 10^5$ lbf/ft^2 = 3056 lbf/in.^{2} At a traverse speed of 10 ft/min, E_s becomes 1220 lbf/in.^{2}, a figure much higher than that for the coal saw, and nowhere near the hoped for values of 100 to 200 lbf/in.^{2}

Tooth forces and bar forces

Assuming that the tangential and normal components of the tooth force are equal and total tangential forces F_t = total normal force F_n, and neglecting the power consumed in thrusting the bar horizontally, then

$$F_t = F_n = \frac{P}{u_t} = \frac{(15)(3.3 \times 10^4)}{1152}$$

$$= 429 \text{ lbf in fourth gear} \qquad (14)$$

$$F_t = F_n = 620 \text{ lbf in third gear.}$$

The net tractive thrust H required from the tractor is:

$H = F_n \cos\phi - F_t \cos\phi$ with ϕ = bar angle from horizontal = 70°

$H = 429(0.9397-0.3420) = 256$ lbf in fourth gear

$H = 371$ lbf in third gear.

The vertical downpull on the tractor V is:

$$V = F_n \cos\phi + F_t \sin\phi = 429(0.9397+0.3420)$$

$$= 550 \text{ lbf (fourth gear)} \qquad (16)$$

$$V = 795 \text{ lbf (third gear).}$$

The mean force components on an individual cutting tooth f_n and f_t at maximum chain speed are:

$$f_t = f_n = \frac{F_t}{(d/\sin\phi)(4.7/12)} = 26.3 \text{ lbf for a collar depth of 6 ft.} \qquad (17)$$

CONCLUSIONS AND RECOMMENDATIONS

The use of large chain saws for cutting ice is far from being a new idea. During tunneling operations at the edge of the Greenland Ice Cap in 1957, a Joy 10RU coal cutter with a 9-ft-long bar was used (Abel 1961). The average cutting rate was 2 ft/min, with 7.5 to 9 ft of bar engaged in the work (kerf width was 6.5 in.). More recently, the University of Alaska, and later, the U.S. Naval Civil Engineering Laboratory (USN CEL), adapted two small soil trenchers (both crawler track types) for cutting ice. These machines were a Davis TF-700 and a Davis TF-1000 (J.I. Case Co.), both fitted with 8-in.-wide "frost chains" and re-equipped with sharp 30° conical steel teeth to replace the carbide-tipped teeth normally used in frozen soils. The TF-700 machine, with a 30-hp engine, achieved a cutting rate of 13.9 ft/min in 31-in.-deep artificial sea ice, while the TF-1000 machine (Fig. 10), with a 60-hp engine, reached 10 ft/min in 72-in.-thick sea ice (Vaudrey 1975, Brier and Vaudrey 1975). Small ladder trenchers were also used during the 1975/76 winter at Prudhoe Bay to cut 6-ft-thick sea ice.

There is no longer any doubt that large chain saws can be very effective tools for cutting ice. However, efficiency can vary within wide limits depending on the design and layout of teeth, and on the mode of operation of the saw (chain speed, traverse speed, thrust force, etc.). The coal saw used for tunneling in

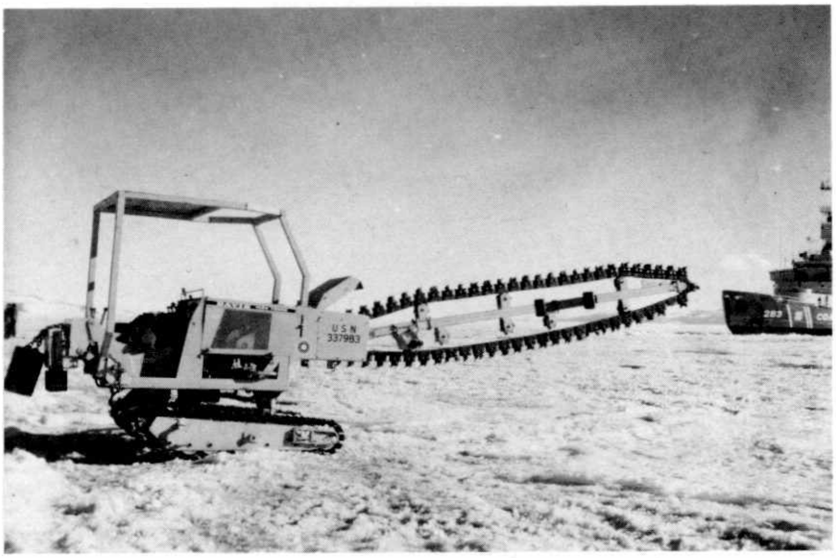


Figure 10. USN CEL ice saw (developed from soil trencher).

Greenland was obviously very inefficient, even when the most favorable assumptions about power consumption are made. At the other extreme, the ladder trenchers adapted by USN CEL were very efficient ice cutters according to the reasonable power assumptions that were made – both machines achieved process specific energy values of approximately 200 lbf/in.^{2} (compared with 400 to 500 lbf/in.^{2} for the lock wall coal saw working in cold freshwater ice). Note that the USN CEL saws were symmetrically mounted on relatively heavy crawler tractors that were capable of providing substantial force reactions; low chain speeds were used (90 ft/min on the TF-700), so that the tangential cutting forces must have been comparatively high. On the lock wall saws, it was necessary to run the offset bars at high chain speeds to minimize overturning moments and to avoid control problems with the light rubber-tire tractors.

The modified lumber-cutting chain saw unit is unacceptable for removing ice from the lock walls because of two major deficiencies: 1) The bar is too flexible, and we believe that additional stiffeners would not help substantially, since they could only extend approximately 6 ft from the driven end of the bar; 2) The chain design allows too little clearance for cuttings to meet the desired 12-ft/min traverse speed.

The Bowdil Company coal saw unit was judged conditionally acceptable for further consideration as a lock wall deicing machine. The major problem was the very short life of the main chain drive sprocket. This would have to be corrected before the improvements suggested in the following paragraphs are considered.

The chain tightening mechanism used was taken from a suggested design by Bowdil Company. The long bolt intended to tighten the chain stripped in the field during the first attempt to tighten the chain. The chain was tightened with the bar in the horizontal position, which created a high tensile force in the chain due to the catenary effect (Fig. 11). If the bar is 15 ft long, $x = 90$ in., the maximum desired sag $y = 1$ in., and chain weight w is assumed to be 1 lbf/in., then the tension T can be calculated as:

$$T = wa \cosh z \tag{18}$$

where $a = x/z$ = distance from lowest point of catenary to the directrix

z = auxiliary variable

$y/x = (\cosh z - 1)/z$.

For this case, $z = 0.02$, so $a = 4500$ and $T = 4500$ lbf. The force on the bar required to achieve this tension is double the chain tension, or 9000 lbf. The tightening mechanism must overcome this force plus any additional friction forces between the bar and the crank arm. These frictional forces are unknown, but may be quite large due to large reaction forces between the keys on the bar and the crank arm, which are necessary to hold the bar horizontal. Some friction force can be eliminated by propping up the end of the bar when tightening the chain.

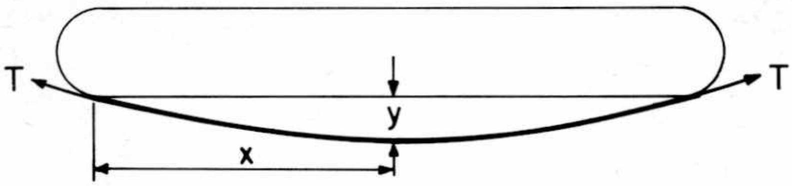


Figure 11. Chain tightening mechanism.

Chain forces of the magnitude required for 1-in. chain sag may also present problems in other areas. The outboard bearing has a radial load rating of 3370 lbf at 500 rev/min. Under static loading conditions, this loading can be increased to about 10,000 lbf without indenting the bearing raceway. The saw should not be operated for extended periods of time with the bar in the horizontal position or bearing damage may occur and life of the wearing shoe on the end of the bar would be limited.

When the bar is in the normal operating position of 20° off vertical the catenary effects are greatly diminished. The tension in the chain due to its own weight is then only slightly over 500 lbf, and this presents no problem.

One method of tightening the chain is to hang the bar vertically over the edge of the lock wall and tighten the bolts before raising the bar. This would require a modification to the lifting mechanism, to allow the bar to be moved into a vertical position. However, there is a possibility of getting the chain too tight and destroying the outboard shaft bearing.

Another method of tightening the chain, which appears to have more merit, is to hydraulically tension the chain with the bar hanging over the edge of the lock wall 20° off vertical. The tensioning mechanism would allow chain tension to be decreased when the bar is horizontal.

A dolly wheel to guide on the lock wall curbing, thus increasing vehicle stability, could possibly be incorporated into the jack mechanism for tensioning the chain. The adjustable feature of the dolly wheel would be quite attractive. However, the mounting position for the two purposes may be incompatible.

The desirability of higher chain speed is obvious. Cutting forces decrease for the same power input, and traverse rates increase accordingly. There may be overriding requirements with regard to chain, bar or sprocket design that would preclude further increasing chain speeds, and chain speeds may have to be decreased. This should be coordinated with Bowdil Company. Changing chain speed is a simple matter of changing one of the drive sprockets.

A guide system for steering the tractor is desirable, since it is difficult for the operator to maintain the tractor at a given distance from the lock wall. This may not be as simple as it seems at first, since a positive guide would have to withstand high tire scuffing forces to maintain proper tractor attitude. Possibly a simple hydromechanical servo system could be incorporated to automatically steer the tractor through its power steering system. It may also be possible that strategically located pointers would provide adequate guidance for the operator to steer the tractor.

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