SECOND INTERNATIONAL CONFERENCE ON
PORT AND OCEAN ENGINEERING UNDER ARCTIC CONDITIONS
UNIVERSITY OF ICELAND
DEPARTMENT OF ENGINEERING AND SCIENCE

# DEVELOPMENT OF A CRAFT CAPABLE OF PREPARING AN ICE FREE CHANNEL THROUGH SOLID ICE COVER

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## INTRODUCTION

The U. S. Coast Guard is the governmental agency responsible for icebreaking in the United States. There are numerous lakes, rivers, and harbors where shipping is hampered during the winter by the formation of solid ice cover. With the help of conventional icebreakers, this shipping can be kept moving for several weeks beyond the normal closing of the navigation season and about one or two weeks at the beginning of the navigation season.

A major problem associated with using conventional icebreakers to clear channels is that, although they break up the ice cover, they do not leave an ice free channel. As a consequence, many ships find it difficult to transit the channel left by an icebreaker even when the ship is following close astern. The U. S. Coast Guard embarked on a development program in 1970 to increase the channel clearing ability of icebreakers and thereby increase their capacity to further extend the navigation season. One concept which looks particularly promising that has evolved from that program is referred to as the Mechanical Ice Cutter, or MIC. This paper provides a review of the progress made during three years of effort on the development of this craft.

## DESCRIPTION OF THE MECHANICAL ICE CUTTER

The requirements for the MIC were to clear a fifty-foot wide channel in fresh water ice two feet thick at speeds of 4 to 5 knots. The emphasis throughout the development program has been on clearing the channel so that it is free of ice.

The concept of the MIC is to utilize mechanical ice cutting equipment and a special shape of the underwater portions of the hull to clear the channel. Three circular saws are mounted on runners in front of the craft and are used to cut narrow slots through the ice. The cutters form two cantilevered ice slabs which are periodically broken by the hull. As the craft moves forward, the ice slabs are forced down and under the hull by the gently sloping hull form. The skeg separates the ice slabs and forces them to the side of the channel. The blocks of ice come to rest under the ice sheet on each side, leaving the channel free of ice.

To test the concept, an experimental 1/6th scale model of the MIC was constructed and tested in 2 to 5 inches of ice on a fresh water lake. Figure 1 is a drawing of the model showing the arrangement of the ice cutters and the skeg. The model has an overall length of 24 feet 6 inches, a beam of 7 feet 6 inches, and a draft of 1 foot 6 inches.

Field tests were conducted during January and February 1973. Figure 2 is a photograph of the model in ice during these tests. The field tests demonstrated that the MIC is a viable and effective system for clearing a channel in ice. Figures 3 and 4 show the open water channel cut by the model. It was also demonstrated that the circular saws were efficient machines for cutting the ice. They did not unduly restrict the maneuverability of the vessel, and in all respects circular saws will be suitable for a full scale prototype.

## ICE CUTTERS

The ice cutters for the experimental model of the MIC were 18 inch diameter, circular saws with twenty-four teeth and six side cutters. The teeth and side cutters are carbide blanks brazed to a heat treated steel disc. Figures 5 and 6 are photographs of the saws mounted on the boat.

The saws were driven by hydraulic motors through a chain drive. Hydraulic power was supplied by a spark-ignition engine driving a variable displacement, axial piston pump.

### Theory of Ice Cutting

An ice cutting system, in general, requires power to overcome four types of loads as it cuts a slot through infinite sheets of plate ice floating on water. These loads are caused by:

1. the power required to disaggregate the ice (the so called clevage or cutting power),
2. the power required to remove the ice chips from the slot,
3. the power required to pull the cutter tools through the water, and
4. the power required to overcome tool-ice sliding friction.

With efficient cutting, the power to overcome sliding friction is small and can be neglected as a variable independent of the disaggregation forces. However, this will only be the case provided that the motion of the MIC does

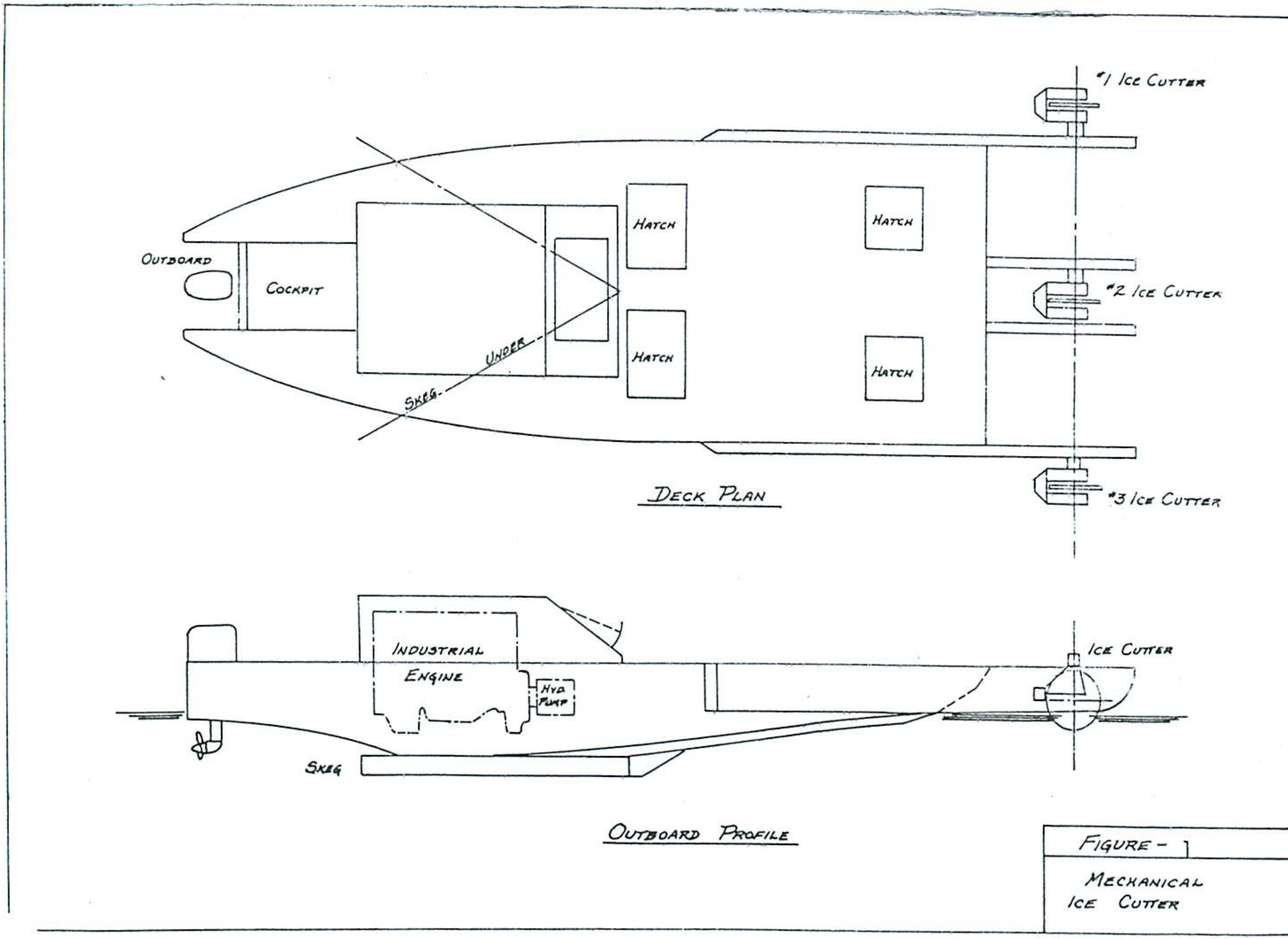



Figure - 1
Mechanical Ice Cutter

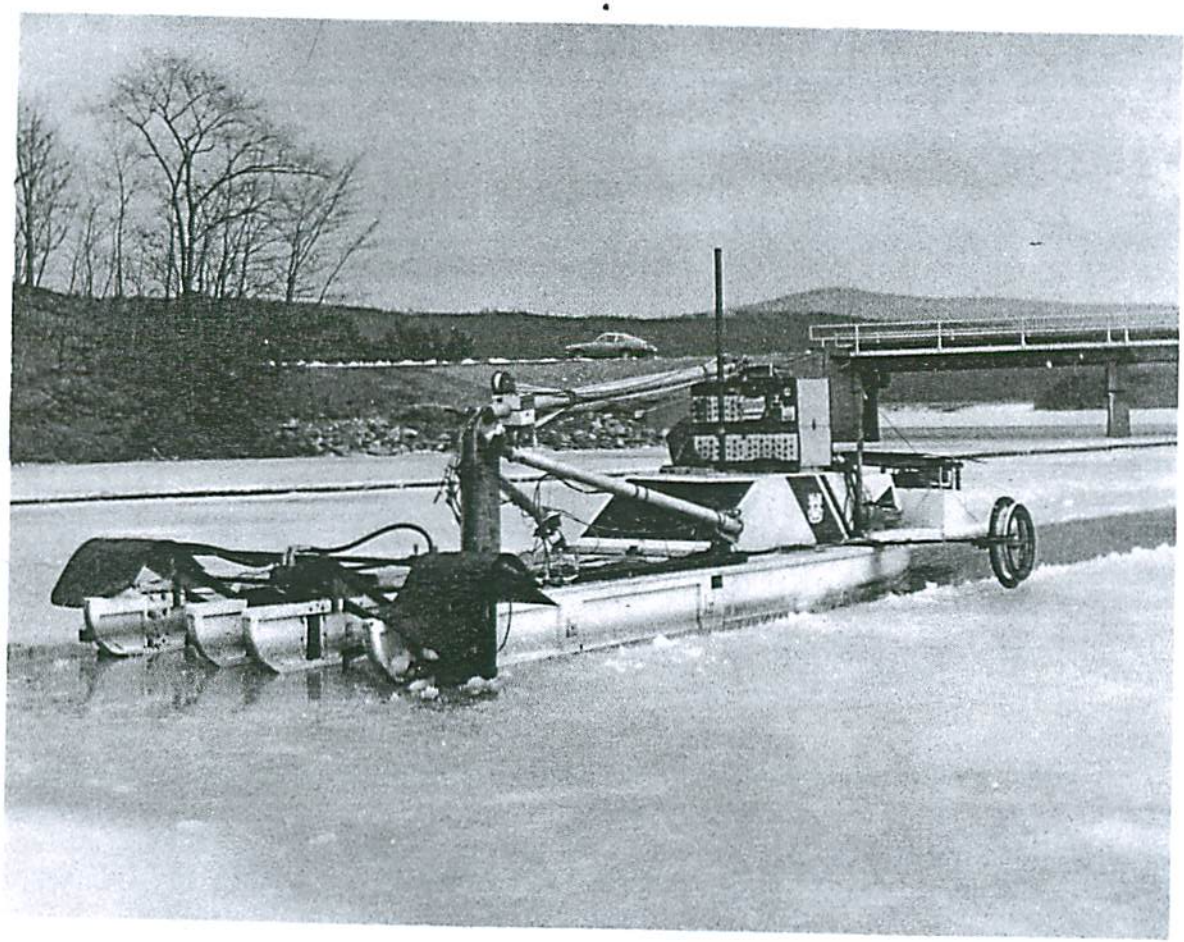


FIGURE 2  Mechanical Ice Cutter in Ice

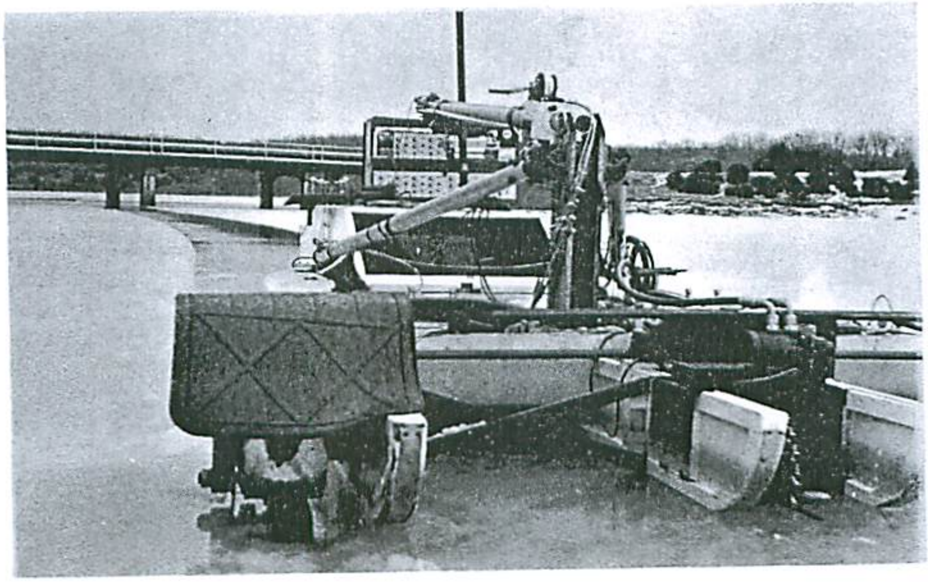


FIGURE 3 Mechanical Ice Cutter (MIC)

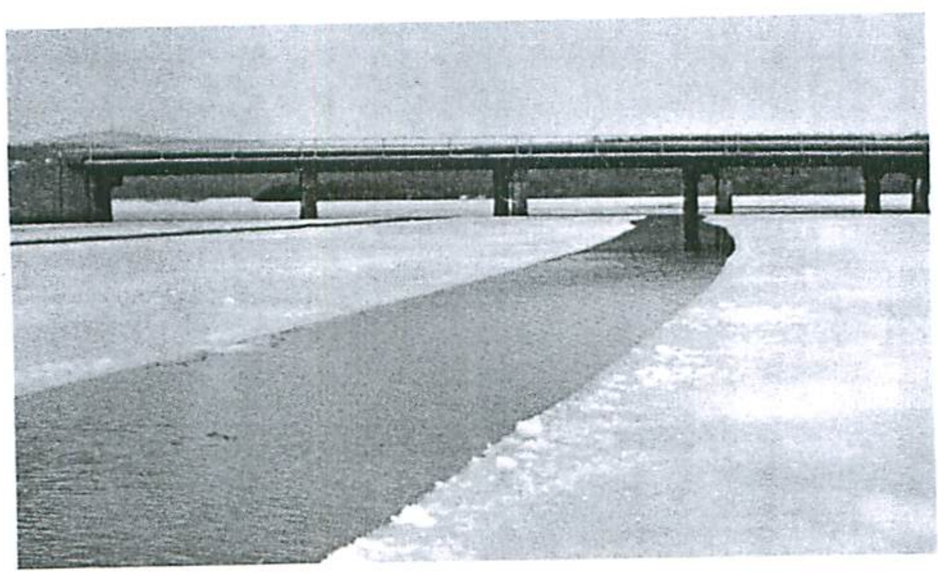


FIGURE 4 Ice Free Channel Cleared By The MIC

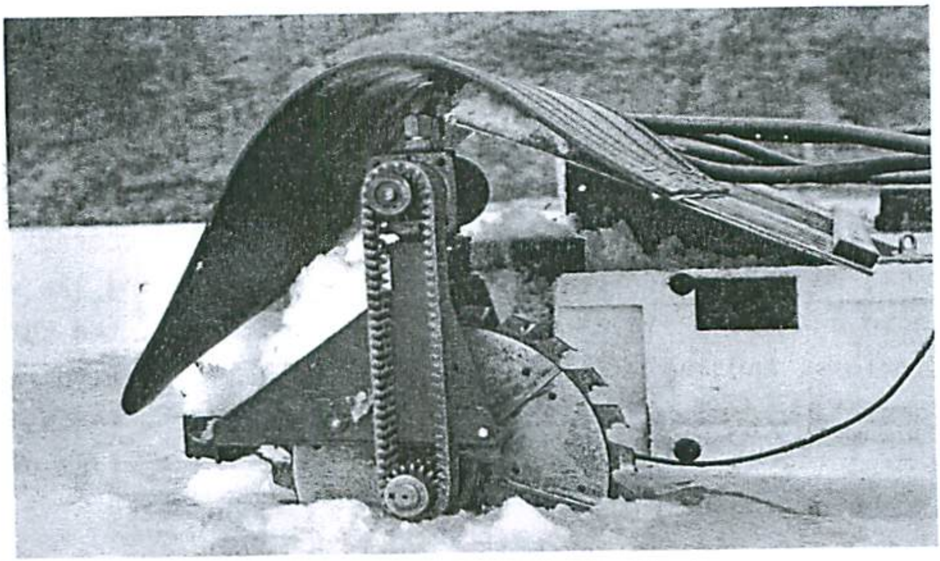


FIGURE 5 Circular Saw Mounted on the Port Side

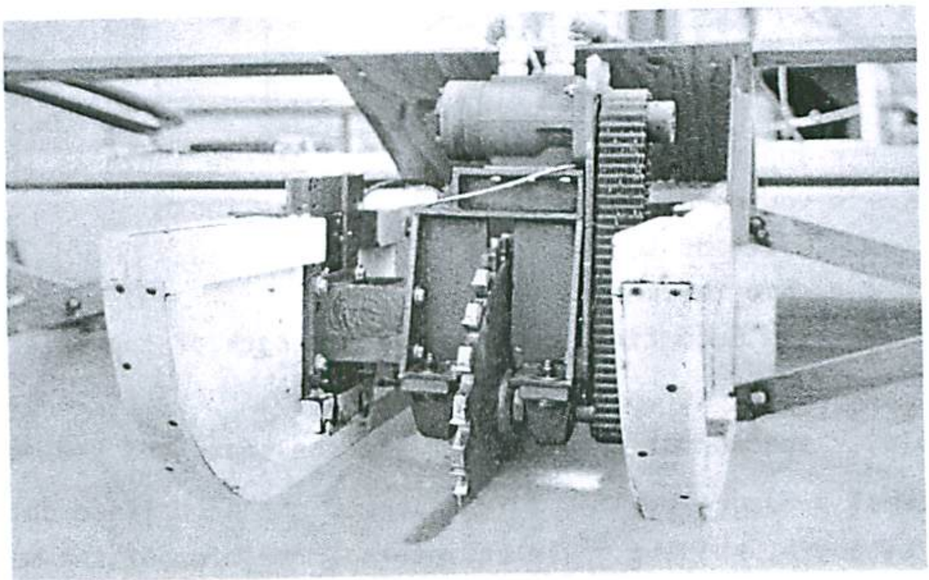


FIGURE 6 Center Saw

not cause the tool to bind in its slot, that the required depth of cut per revolution is less than the depth of the cutting teeth, and that the cuttings are removed from the teeth and do not clog. The kinematic design consideration must ensure that all of the above constraints are met.

Power to Disaggregate Ice. - The mechanics of disaggregation of any solid material by mechanical tools is extremely complicated, and there are no proven ways by which the required power to disaggregate at a given rate can be predicted. In Reference [1] Nalezny proposes a method whereby disaggregation power can be obtained from first principles. Assuming the material to be cut to be an ideal, rigid-plastic, Coulomb material, the cutting teeth to be frictionless, and the width of the tooth to be large with respect to the chip thickness (i.e., plane strain), Nalezny computes the forces on the cutting teeth by integrating the normal pressure over the surface of the tooth.

Referring to Figure 7 which shows the geometry of a cutting tooth, the normal force $F_N$ and the tangential force $F_T$ are given in Reference [1] as

$$\frac{F_N}{Wrk} = \int_0^{\frac{\pi}{2}} \sigma(\theta) \cos \theta . d\theta \tag{1}$$

$$\frac{F_T}{Wrk} = \int_0^{\frac{\pi}{2}} \sigma(\theta) \sin \theta \, d\theta + \left(\frac{d}{r} - 1\right) \frac{q_u}{k} \tag{2}$$

where

$\theta$ = angle around the cutting edge of the tooth as shown in Figure 7

$r$ = radius of the cutting edge of the tooth

$W$ = width of tooth

$d$ = chip thickness

$q_u$ = the unconfined compressive strength of the material

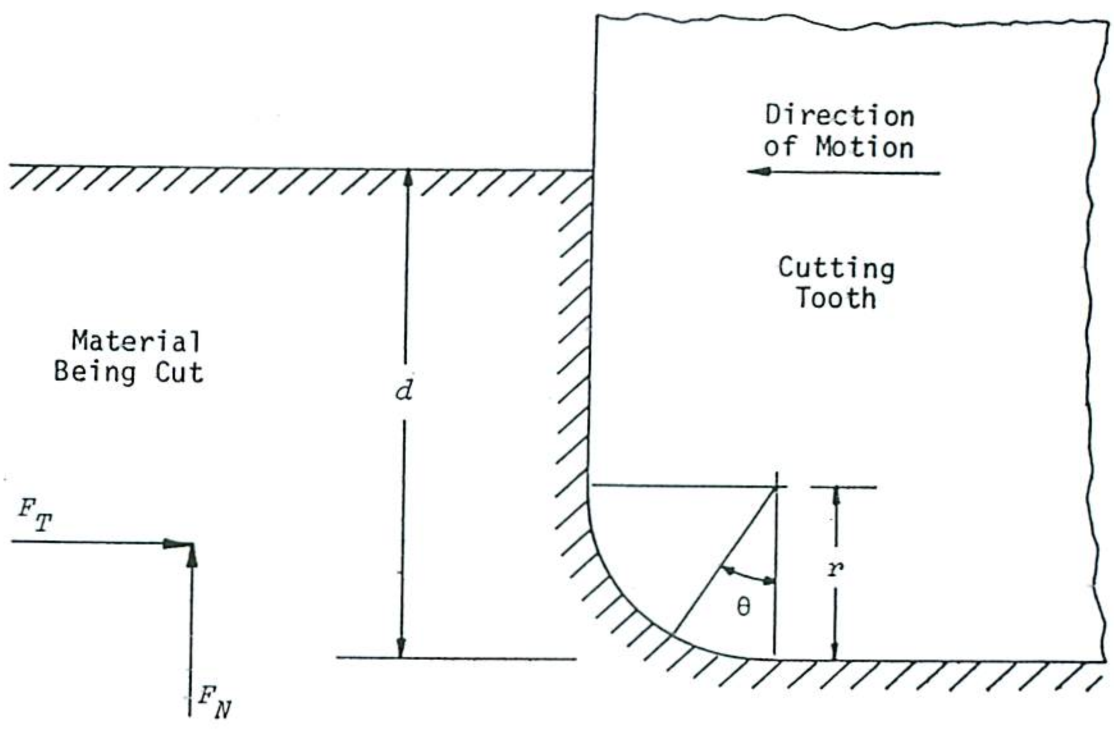
$k$ = shear strength of the material

$\sigma(\theta)$ = the normal pressure between the tooth and the material

The normal pressure $\sigma(\theta)$ is a function of the unconfined compressive strength, shear strength, and the angle of internal friction of the material as well as being a function of $\theta$. Unfortunately, these properties of ice are not known.

Nalezny's theory can be simplified if it is assumed that the cutting teeth are infinitely sharp: that is, $r = 0$.

Figure 7. Geometry of Cutting Tooth



Under this assumption, equation (1) reduces to

$$F_N = 0 \tag{3}$$

and equation (2) reduces to

$$F_T = Wdq_u \tag{4}$$

Equation (4) states simply that the resistance of the cutter to forward motion is equal to the unconfined compressive strength of the material times the area of the face of the tooth in contact with the material. This assumption of $r = 0$ is not unreasonable providing the depth of the cut is large compared with the radius of the cutting edge. This will be true in the case of the ice cutter operating at its designed speed of advance.

Accepting this theory provides a direct method for predicting the force on the tooth. The forces on all the teeth in contact with the ice can be summed to compute the vertical and horizontal forces as well as torque and power required for the saw.

For a circular saw the torque ($Q_d$), and hence the power ($P_d$) can be calculated by summing the tangential forces on the teeth in contact with the ice

$$Q_d = R \sum_{i=1}^{m} F_i \tag{5}$$

where

$F_i$ = $F_t$ on the $i$th tooth

$R$ = radius of the circular saw

$i$ = 1, 2, 3, - - - - -, $m$ are the individual teeth in the ice

Substituting equation (4) for $F_i$ in (5),

$$Q_d = R\, q_u\, W \sum_{i=1}^{m} d_i \tag{6}$$

Reference [1] gives the chip thickness as

$$d_i = \frac{V}{nN} \sin \Omega_i \tag{7}$$

where

$V$ = speed of advance

$n$ = rotary speed of cutter (rev/sec)

$N$ = number of teeth on the cutter

$\Omega$ = angle from the vertical (Figure 8) and, therefore,

$$Q_d = \frac{R\, q_u\, W\, V}{n\, N} \sum_{i=1}^{m} \sin \Omega_i \tag{8}$$

Averaging this expression over the time for rotation through the angle

$$\Omega = \frac{2\pi}{N} \tag{9}$$

gives

$$Q_d = \frac{R\, q_u\, W\, V}{n\, N} \times \frac{N}{2\pi} \int_0^{\Omega_m} \sin \Omega \, d\, \Omega \tag{10}$$

where $\Omega_m$ = the total angle of travel in the ice (dependent on the thickness of the ice).

Integrating equation (10),

$$Q_d = \frac{R\, q_u\, W\, V}{2\pi n} (1 - \cos \Omega_m) \tag{11}$$

From Figure 8, the ice thickness $h$ and the saw radius $R$ are related by

$$h = R\,(1 - \cos \Omega_m). \tag{12}$$

Substituting in equation (11), the torque is

$$Q_d = \frac{q_u\, W\, h\, V}{2\pi n}. \tag{13}$$

The power for disaggregation is

$$P_d = 2\, \pi\, n\, Q_d. \tag{14}$$

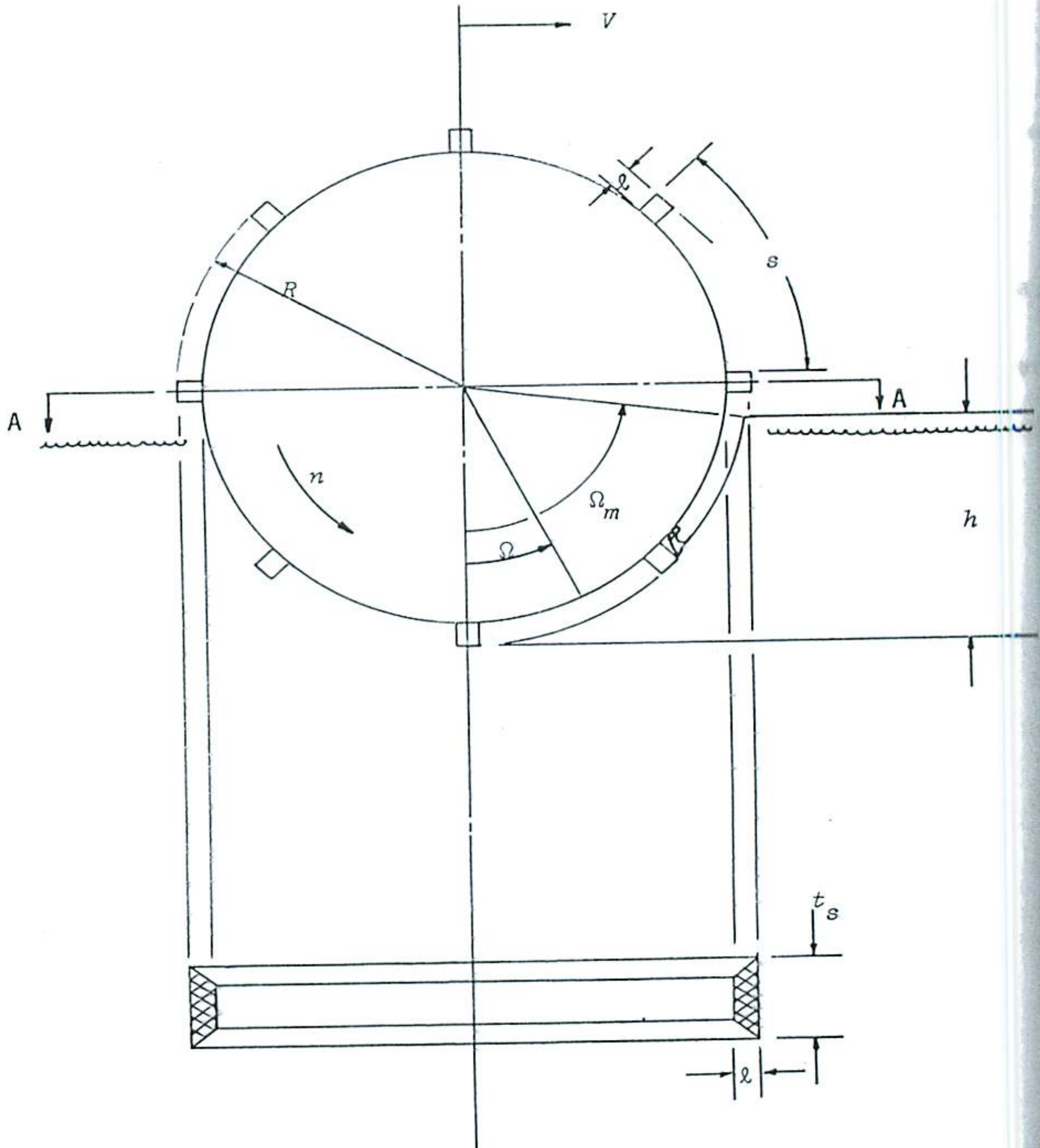
Combining equations (13) and (14) results in the following expression for disaggregation power:

$$P_d = q_u\, W\, h\, V \tag{15}$$

If the width of the slot $t_s$ equals the width of the tooth $W$, equation (15) becomes

$$P_d = q_u\, t_s\, h\, V \tag{16}$$

Figure 8
Sketch of a Circular Saw Type Ice Cutter




SECTION A A

Equation (16) is in the same form as that proposed by Mellor in Reference [2] using "specific energy", that is, the power of disaggregation is proportional to the volumetric rate at which the material is removed. Nalezny's theory, however, provides more information in that forces can be calculated directly. His theory also allows for a detailed analysis as more is learned of the properties of ice in relation to the cutting process. For these reasons Nalezny's theory relating forces, torque, and power to the unconfined compressive strength of ice was adopted for analysis of the ice cutters.

Chip Removal Power. - The power required to remove chips from the slot once they have been broken out can be estimated using energy methods. The energy required to accelerate each chip up to the speed of the tooth is given by:

$$KE_{\text{chip}} = \frac{1}{2} m_c v_c^{\,2} \tag{17}$$

where

$m_c$ = the mass of the chip

$v_c$ = the speed of the tooth

The power is given by:

$$P_c = \frac{1}{2} t_s h V \rho_i v_c^{\,2} \tag{18}$$

where

$t_s$, $h$, and $V$ are as defined in the previous section

$\rho_i$ is the ice density

For these calculations it has been assumed that the cuttings are cleaned from the teeth by their own energy; that is, by centrifugal force.

Energy Required To Overcome Fluid Drag. - When using cutters to cut slots in floating ice sheets, power must be expended in overcoming the drag forces and displacing water lying in the slot. The power required to overcome drag of the cutters through the water, $P_w$, is given by

$$P_w = \text{drag} \times v_c \times N_w$$

$$P_w = 1/2\ C_D\ \rho_w\ v_c^{\,2}\ A\ v_c\ N_w$$

$$P_w = 1/2\ C_D\ \rho_w\ v_c^{\,3}\ A\ N_w \tag{19}$$

where

$C_D$ = the drag coefficient

$A$ = area of the face of the tooth

$\rho_w$ = density of water

$N_w$ = number of cutter teeth in the water

MATHEMATICAL MODEL OF HULL RESISTANCE

Predictions of hull resistance were made by developing a mathematical model. The field test data were compared with the math model, giving confidence that the resistance of the full scale prototype could be predicted. The total resistance may be separated into three major components

$$R_t = R_w + R_c + R_i \tag{20}$$

where

$R_t$ = total forward motion resistance

$R_w$ = resistance caused by water drag

$R_c$ = resistance caused by the ice cutter system

$R_i$ = resistance caused by breaking ice and moving the broken ice pieces (hull resistance)

The magnitude of the first component ($R_w$) caused by water drag was determined from open water tests conducted during the 1971-72 winter season. From these tests, this component of resistance was found to be a function of speed expressed by

$$R_w = 0.0367\ V^2 \tag{21}$$

where $R_w$ and $V$ have units of pounds and inches per second respectively.

The second component ($R_c$) caused by the ice cutter system can be predicted using Nalezny's theory from Reference [1].

The objective of the math model, then, is to describe analytically the third major component of resistance ($R_i$). To aid in the development of this model $R_i$ was separated into two additional components:

$$R_i = R_{ism} + R_{icb} \tag{22}$$

where

$R_{ism}$ = component due to Ice Slab Movement

$R_{icb}$ = component due to Ice Cantilever Breaking

The first component, $R_{ism}$, is due to the previously broken ice slabs being submerged by the hull and then being pushed clear of the channel by the skeg as the craft proceeds. To adequately describe this portion of the model, the slabs were treated as rigid bodies acted upon by

- hull forces (normal and frictional)
- buoyancy forces
- hydrodynamic forces
- forces caused by adjacent ice pieces

The slabs experience both linear and angular accelerations; therefore, the problem is a dynamic one. The forces were determined by writing the linear and angular momentum equations for each ice slab in contact with the hull. Once these appropriate forces were determined, resistance was obtained by summing the components of the hull forces in the direction of the forward motion of the craft.

In addition to the resistance which arises from moving broken ice slabs, the hull resistance also arises from the second component, $R_{icb}$, due to breaking of the cantilevers. The approach used in developing this portion of the math model was to analyze the response of the cantilever as it follows the contour of the hull. Included in the analysis were the following considerations:

- ice cantilever dynamic response (both inertia and elastic)
- ice cantilever buoyancy
- ice - hull friction
- dynamic response of supporting water
- effect of in-line (horizontal) forces

The horizontal and vertical forces to break the cantilevers are large; however, the time required for failure to occur is very small. The average of these forces is therefore small, and they can be neglected. As a result,

$$R_i \approx R_{ism} \tag{23}$$

The results of the math model are presented in Figures 9, 10, and 11. In the figures the non-dimensional form for the horizontal resistance ($R_i / \rho_w gbhl$) and the vertical force ($F_v / \rho_w gbhl$) are plotted versus $V / \sqrt{gl}$ for varying hull angle $\theta$, skeg angle $\beta$, and coefficient of friction $\mu$, where

$\rho_w$ = water density

$g$ = acceleration due to gravity

$b$ = width of the ice slabs

$l$ = length of the ice slabs

$h$ = ice thickness

$V$ = velocity of the craft

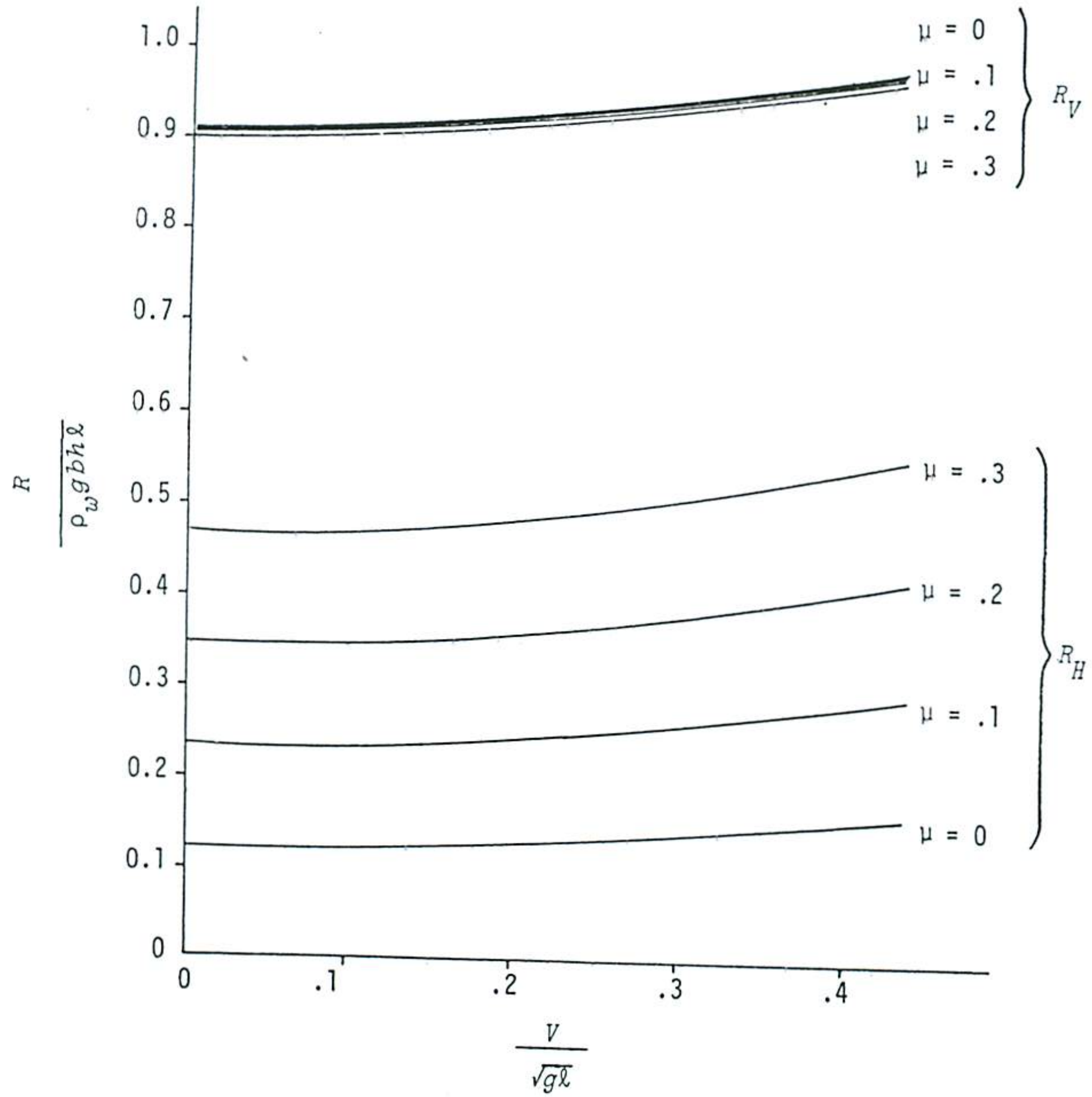



Figure 9. Influence of Friction on the Resistance

$\theta = 7°$

$\beta = 25°$

$\mu = 0, .1, .2, .3$



15

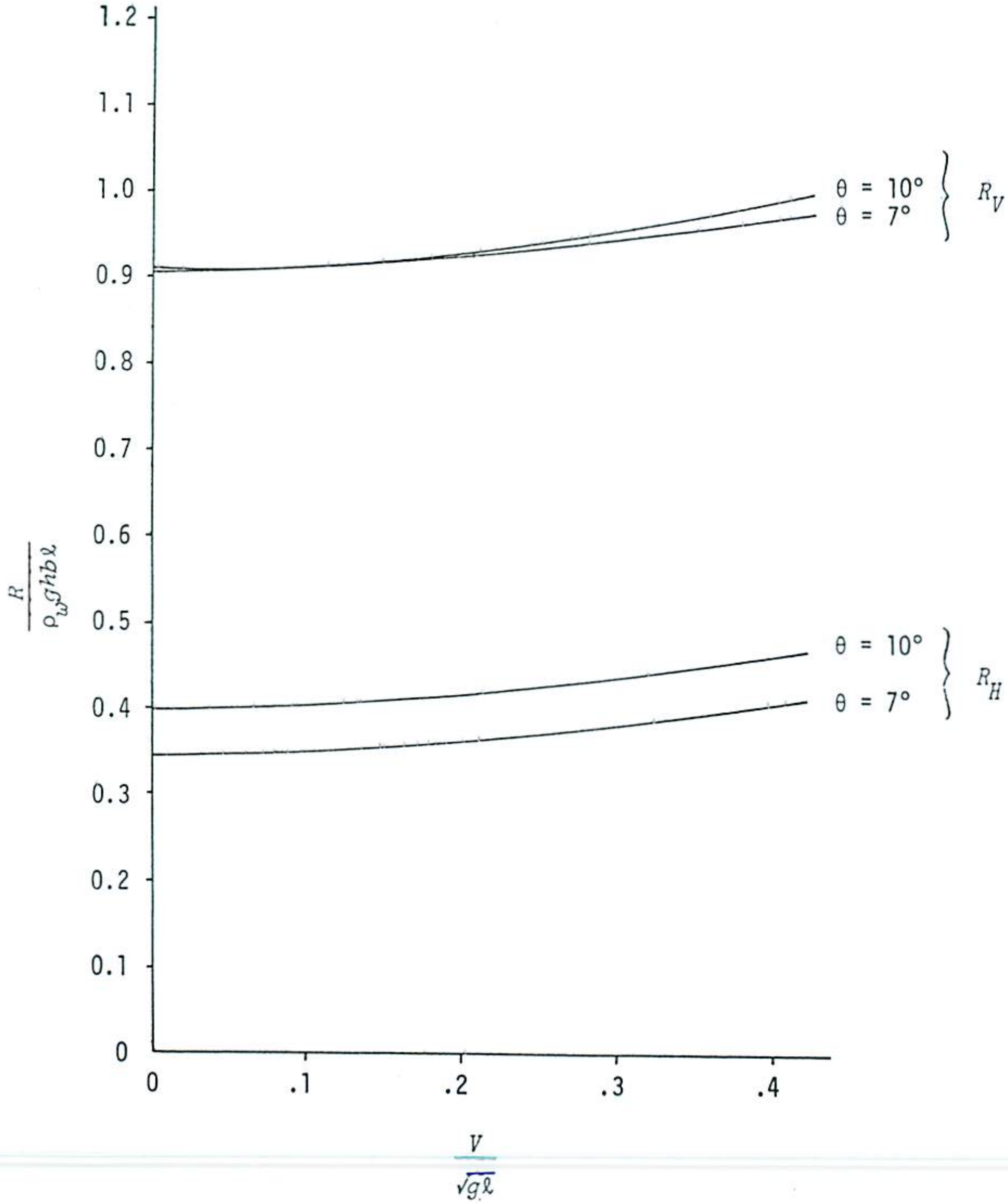



Figure 10. Non-Dimensional Force Versus Non-Dimensional Velocity

$\beta = 25°$

$\mu = 2$

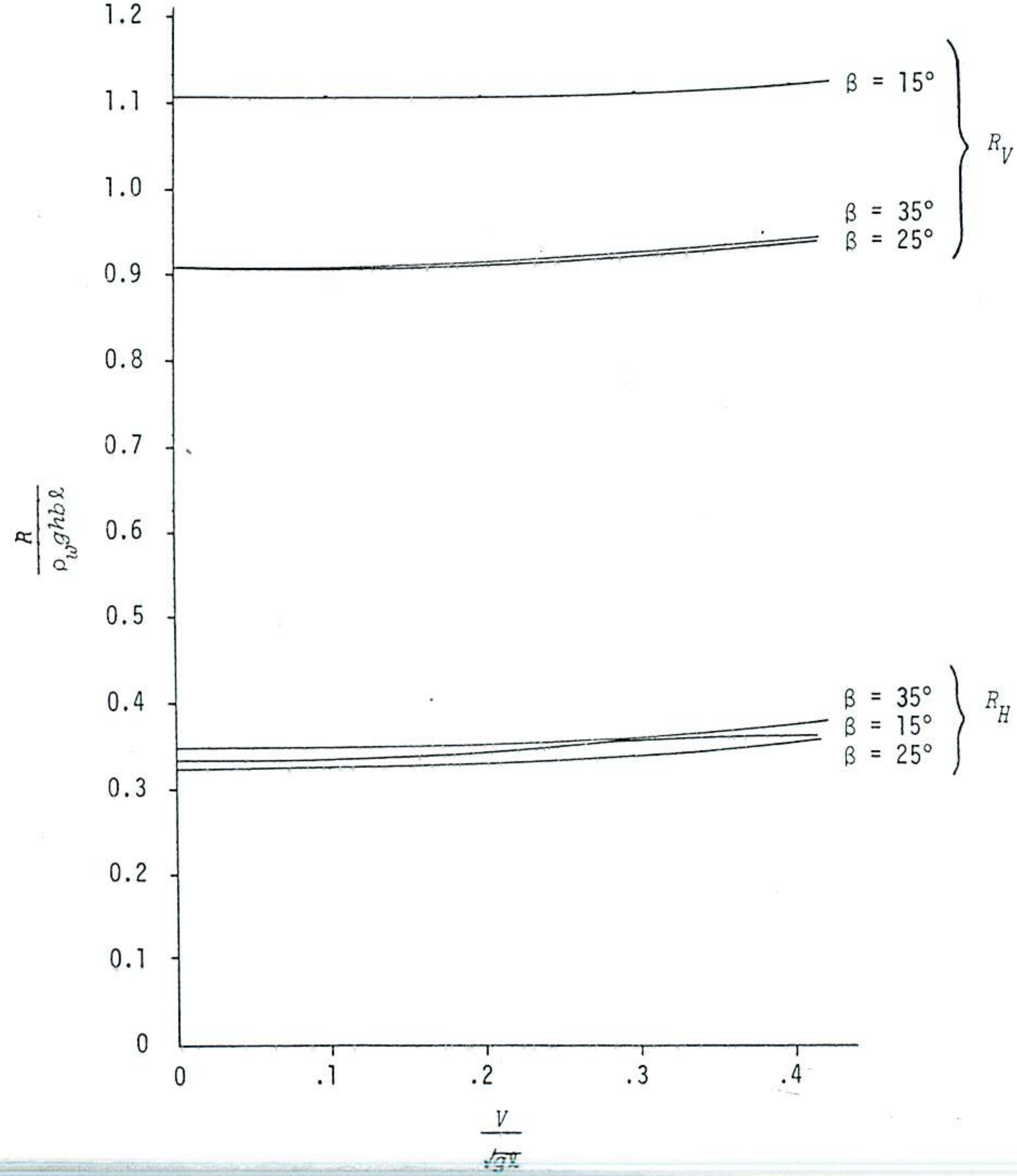



Figure 11. Influence of Skeg Angle on Resistance

$\theta = 7°$

$\mu = .2$

$\beta = 15°, 25°, 35°$



154

## FIELD TEST DATA

The hull resistance data from the field tests are presented in Figures 12 and 13. The data center around two distinct thicknesses -- 2.6 and 4.75 inches. To compare the math model with these data, the total resistance predicted for these ice thicknesses is ploted, and the corresponding field test data have been superimposed on these plots. The ice resistance $R_i$, the cutter resistance $R_c$, and the hydrodynamic resistance $R_w$, which are summed to obtain the predicted total resistance $R_t$, are also shown in the two figures.

The scatter in the data is due primarily to the fact that the motion of the boat was not in a straight line. Slight turns were unavoidable, causing the forces on the saws to vary. This variation has been indicated in Figures 12 and 13 by the high and low estimates shown by the dashed lines.

In full scale, the forces acting on the saws are much less significant since they will scale by $\lambda^2$ while the other resistance forces will scale by $\lambda^3$. This is readily seen by comparing the model predictions in Figures 12 and 13 with the full scale predictions in Figure 14.

The ice cutter data from the field tests are presented in Figure 15. From the theory the total power consists of power for disaggregation, power for water drag, and power for chip acceleration. The power for chip acceleration is negligible for the 18-inch saws, so the power for disaggregation was obtained by subtracting the power for water drag from the total measured power.

The average value for the unconfined compressive strength of the ice was 529 psi with a deviation from the mean of 129 psi.

## FULL SCALE DESIGN

The successful completion of the field tests has demonstrated that the mechanical ice cutter concept is indeed a viable system for cutting and clearing a channel in ice. The field tests have provided considerable operating experience in ice, as well as providing the experimental data necessary to accomplish the full scale design. To illustrate the application of the design theories, the following calculations have been made to predict the power requirements for a full scale prototype of the MIC.

### Ice Cutter Design

The full scale circular saws will be designed with the following dimensions and specifications:

| | |
|---|---|
| saw radius | $R$ = 54 inches |
| number of teeth | $N$ = 24 |
| rotational speed | $n$ = 5.1 rev/sec |
| slot width | $t_s$ = 3 inches |
| ice thickness | $h$ = 24 inches |
| speed of advance | $V$ = 100 inches/sec |

Figure 12. Resistance of $\frac{1}{6}$th Scale, Field Test Model Mechanical Ice Cutter

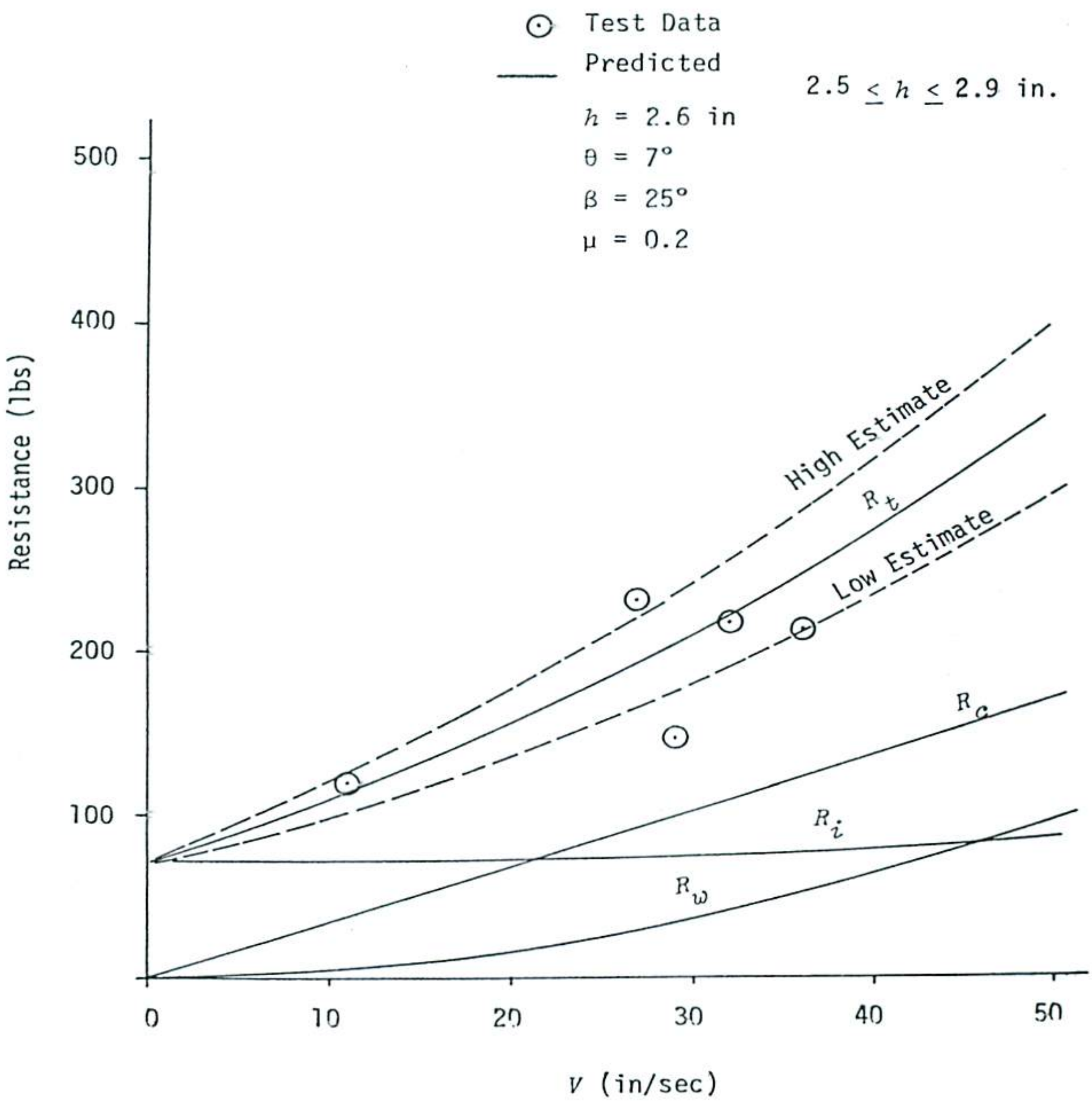


Figure 13. Resistance $\frac{1}{6}$th Scale

Field Test Model Mechanical Ice Cutter

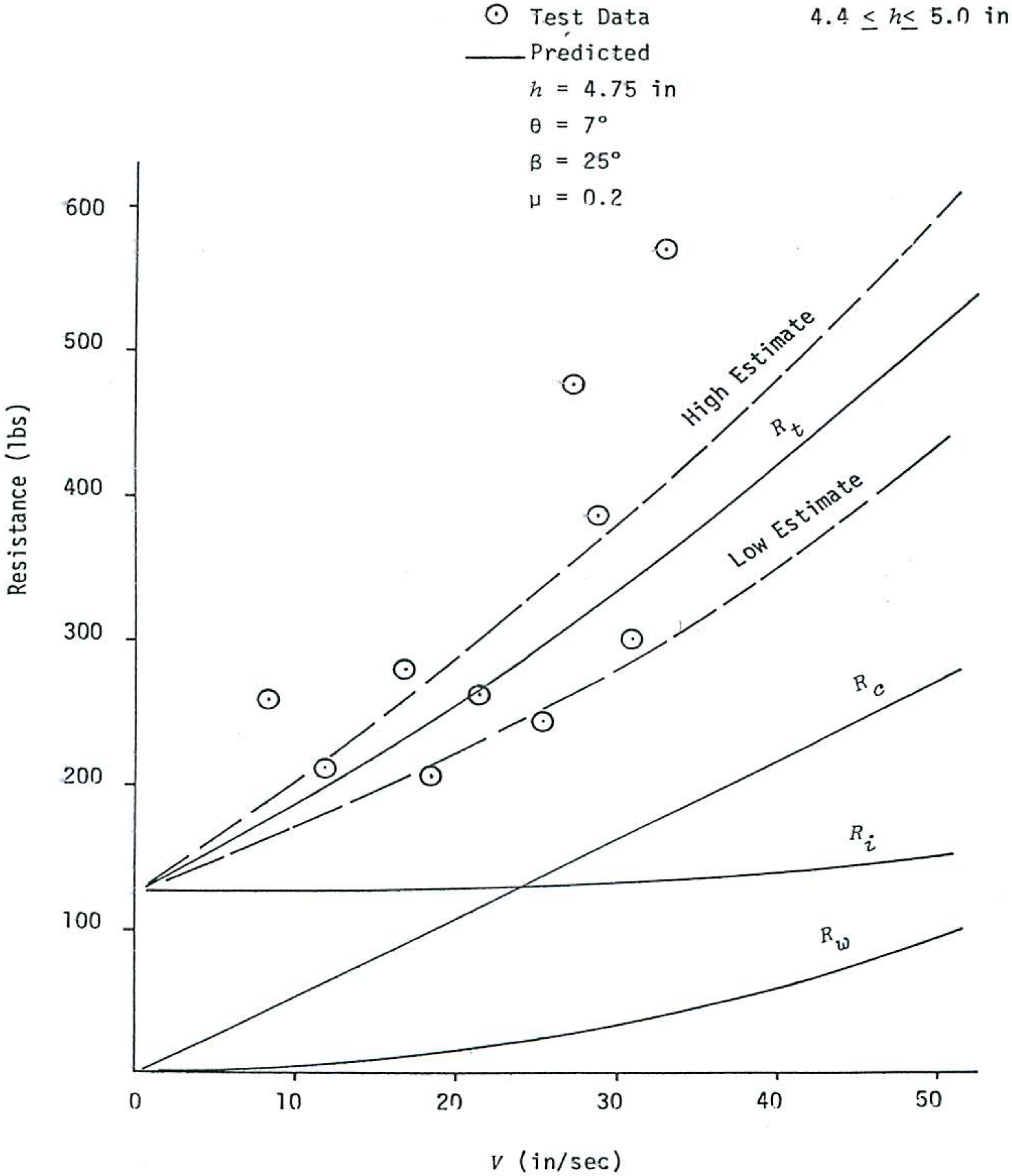


Figure 14. Full Scale Resistance
Mechanical Ice Cutter

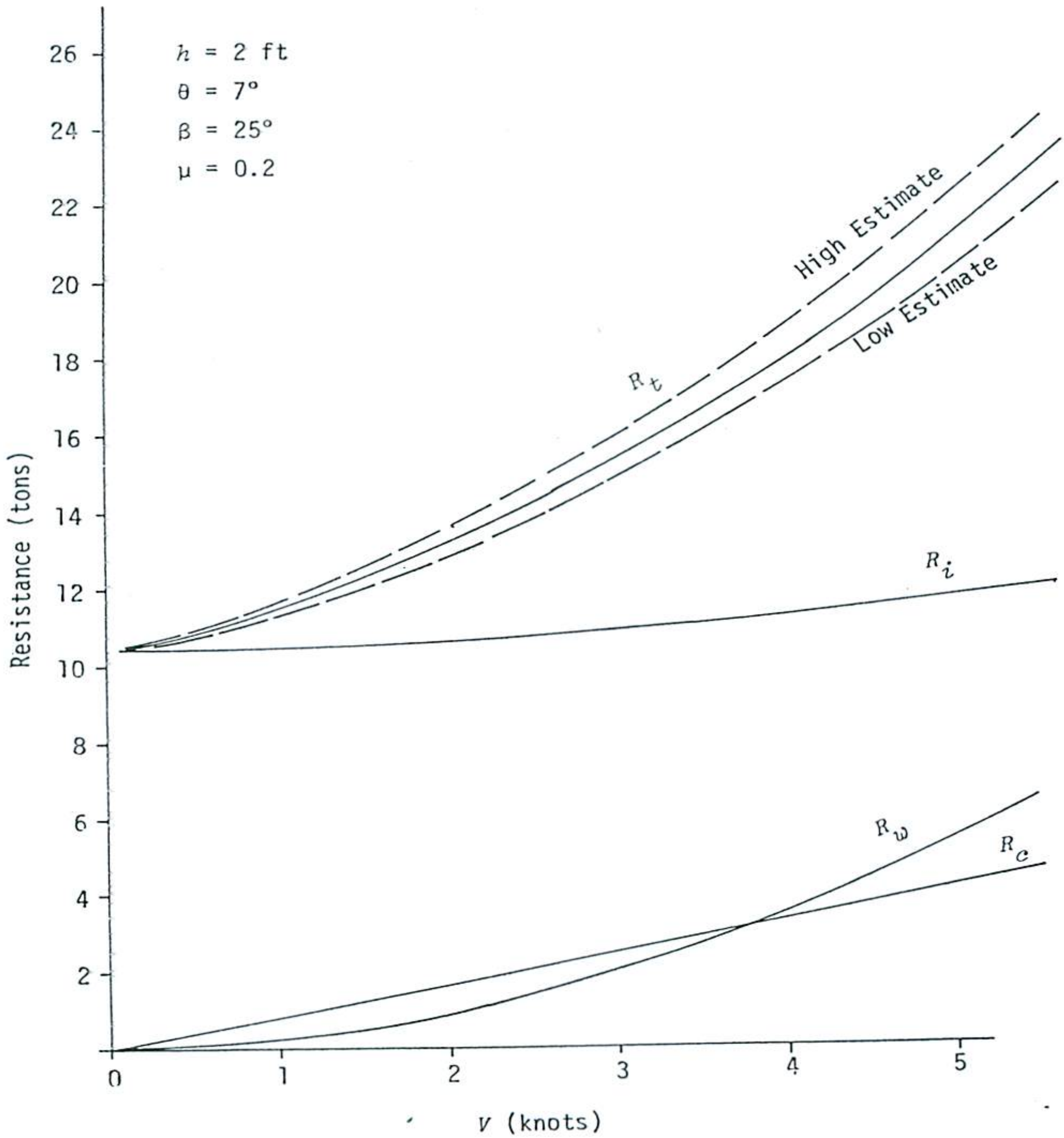
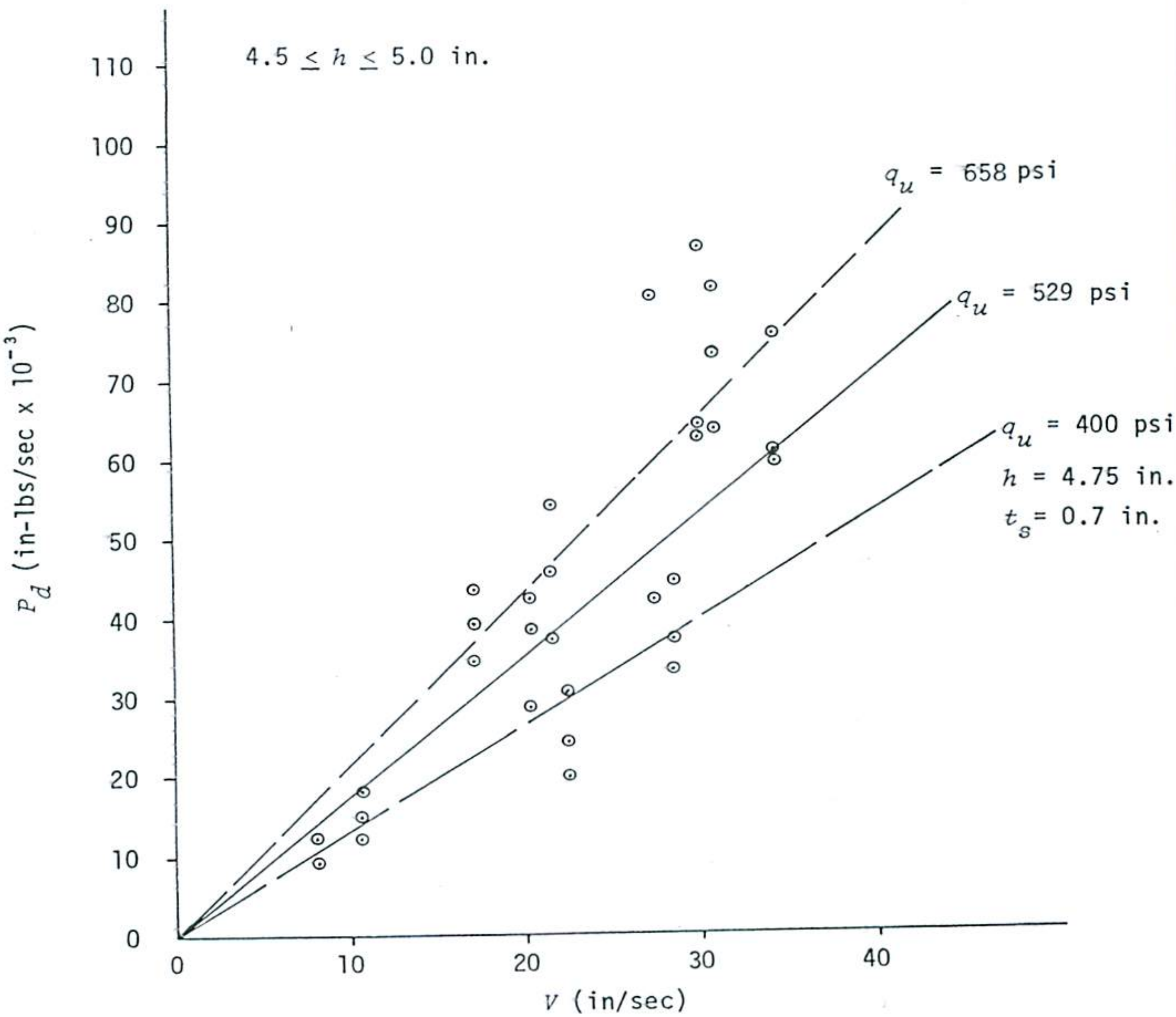


Figure 15. Disaggregation Power Data from Field Tests



Using these dimensions the power required for one saw has been calculated to be:

power for disaggregation, $P_d$ = 580 HP

power for chip removal, $P_c$ = 140 HP

power for water drag, $P_w$ = 120 HP

The total power for one saw is:

$$P_T = P_d + P_c + P_w$$

$$P_T = 840 \text{ HP}$$

Hull Resistance

The full scale prototype will cut a channel fifty-four feet wide in ice two feet thick at a speed of 5 knots (8.4 ft/sec.). The ice will be cut and broken into slabs twenty-seven feet by twenty feet. The ice resistance is determined from Figure 14 as

$$R_t = 22 \text{ tons}$$

Assuming that a marine propulsion system designed to provide the required thrust under these conditions with an efficiency of 0.33, the power required for propulsion will be 2300 HP. The power requirements for the MIC, including the three saws and the propulsion system, totals 4820 HP.

Comparison with Conventional Icebreakers

In order to compare the mechanical ice cutter with the conventional icebreakers, the power will be estimated for an icebreaker similar to the USCGC MACKINAW but with a fifty-four foot beam. The resistance can be estimated using the following equation from Reference [3]:

$$R = 3.455\ \rho_w g B h^2 + 4.680\ \rho_w g^{1/2}\ B h^{3/2} v + 0.019\ B h \sigma \quad (24)$$

where

$\rho_w$ = 1.94 lb-sec$^2$/ft$^4$

$g$ = 32.2 ft/sec$^2$

$B$ = 54 ft.

$h$ = 2 ft.

$v$ = 8.4 ft/sec

$\sigma$ = 128 psi = 18,400 lbs/ft$^2$

The resistance is calculated to be

$R$ = 66 tons.

Using the same propulsion system efficiency, the power requirements for the conventional icebreaker is 6850 HP. Comparing this to the total power of the MIC, it is seen that the MIC required 30% less power than the conventional icebreaker.

As well as requiring less power, another significant advantage of the MIC is that the cut channel is free of ice. Following ships can proceed unhindered in the open water channel; this alone makes the MIC attractive as an icebreaker.

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2. Mellor, Malcolm, "Mechanics of Trasverse-Rotation Cutting Devices", CRREL, U.S. Army Corps of Engineers, June 1972.

3. Edwards, Roderick Y., Jr., Lewis, Jack W., Wheaton, James W., and Coburn, Joseph, Jr., "Full-Scale and Model Tests of a Great Lakes Icebreaker", Transactions, SNAME, Vol. 80, 1972.