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THE UNIVERSITY OF MICHIGAN

COLLEGE OF ENGINEERING
DEPARTMENT OF NAVAL ARCHITECTURE AND MARINE ENGINEERING

Systematic Investigation of Marsh Screw Rotor Water Propulsion

J. L. MOSS

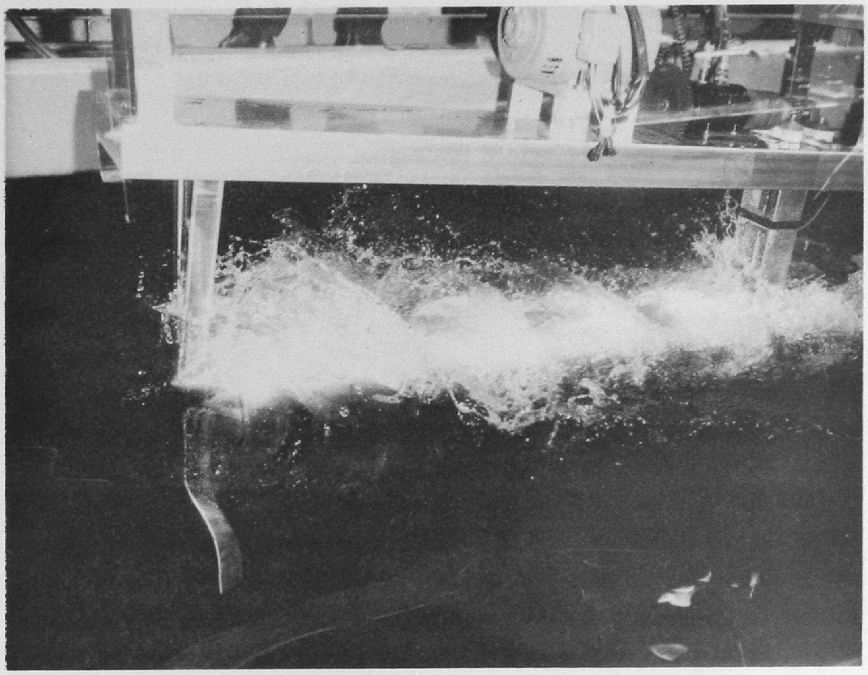
Under contract with:

Chrysler Corporation
Defense Engineering
Order No. DEX-39005
Contract No. NObs-4633

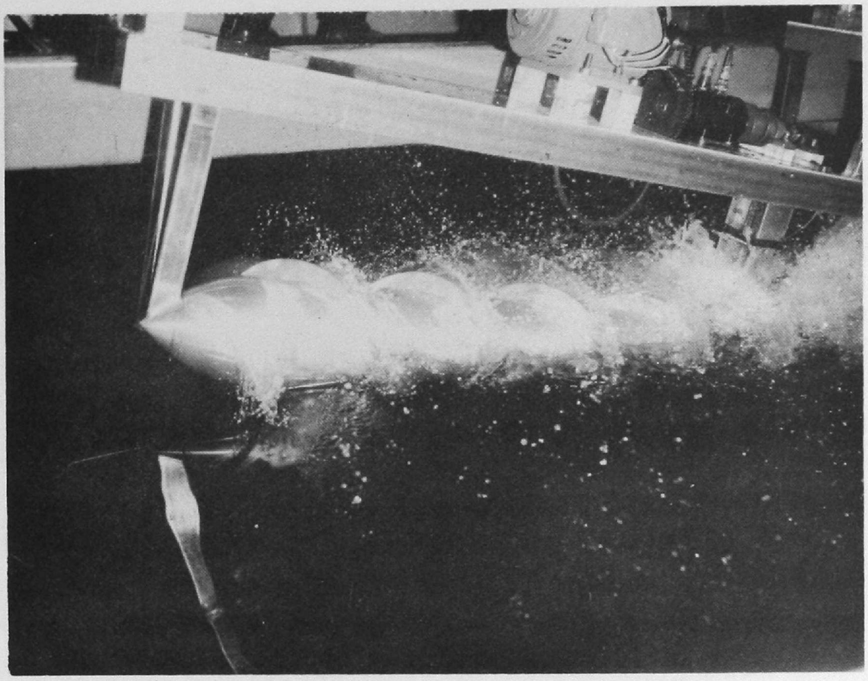
Administered through:

March 1965

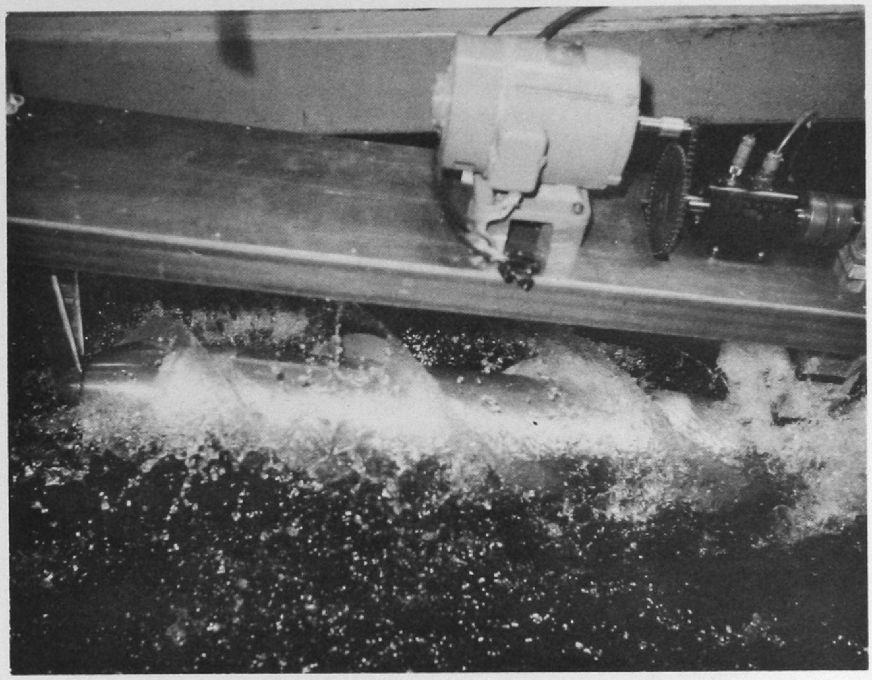
OFFICE OF RESEARCH ADMINISTRATION • ANN ARBOR



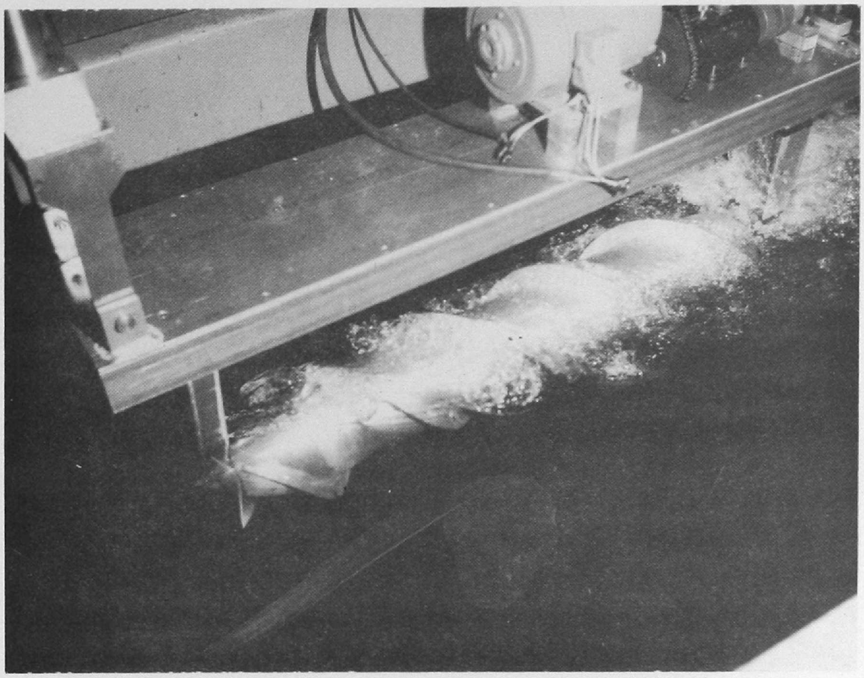
Helix angle	40 deg.
Full scale blade height	4.68 inches
Full scale hub diameter	24.0 inches
Full scale displacement	1145.3 lbs.@59°F.S.W.
% displacement	50% of rotor hub
Full scale velocity	9.5 mph
Initial trimming angle	2.2 deg. bow up
Running trimming angle	8.2 deg. bow up
Running full scale sinkage	0.71 inches down
Full scale RPM	216



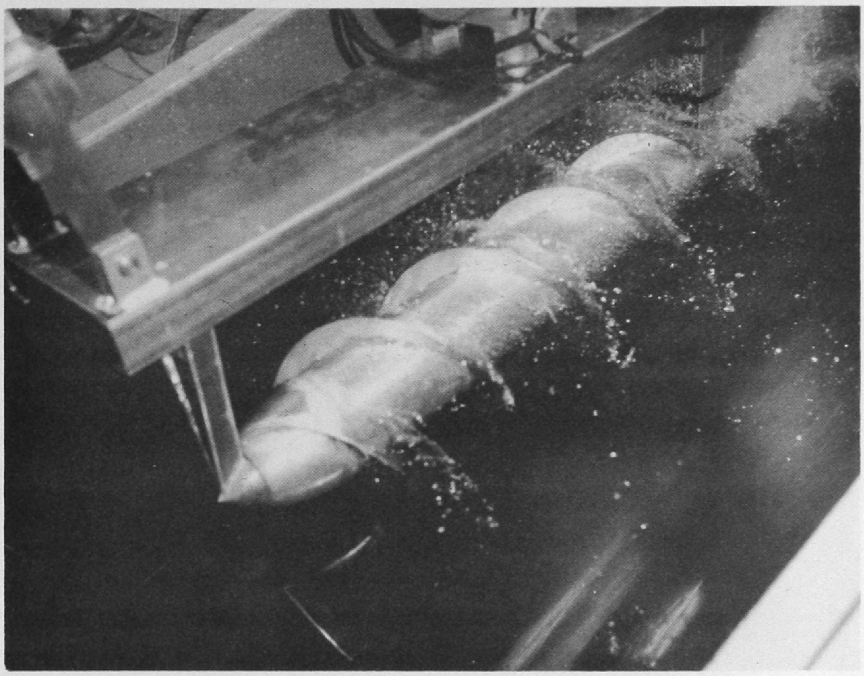
Helix angle	40 deg.
Full scale blade height	4.68 inches
Full scale hub diameter	24.0 inches
Full scale displacement	1145.3 lbs.@ 59°F.S.W.
% displacement	50% of rotor hub
Full scale velocity	9.5 mph
Initial trimming angle	4.0 deg. bow up
Running trimming angle	9.3 deg. bow up
Running full scale sinkage	0.95 inches down
Full scale RPM	225



Helix angle	50 deg.
Full scale blade height	5.63 inches
Full scale hub diameter	24.0 inches
Full scale displacement	1145.3 lbs.@59^{o}F.S.W.
% displacement	50% of rotor hub
Full scale velocity	9.5 mph
Initial trimming angle	2.2 deg. bow up
Running trimming angle	11.4 deg. bow up
Running full scale sinkage	0.59 inches up
Full scale RPM	159



Helix angle	40 deg.
Full scale balde height	4.68 inches
Full scale hub diameter	24.0 inches
Full scale displacement	1718.0 lbs.@59° F.S.W.
% displacement	75% of rotor hub
Full scale velocity	6.5 mph
Initial trimming angle	0
Running trimming angle	0
Running full scale sinkage	5.68 inches down
Full scale RPM	104



Helix angle	40 deg.
Full scale balde height	5.63 inches
Full scale hub diameter	24.0 inches
Full scale displacement	572.7 lbs.@ 59^{o} F.S.W.
% displacement	25% of rotor hub
Full scale velocity	9.5 mph
Initial trimming angle	2.3 deg. bow up
Running trimming angle	10.8 deg. bow up
Running full scale sinkage	0.47 inches down
Full scale RPM	188

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SYSTEMATIC INVESTIGATION OF MARSH SCREW ROTOR WATER PROPULSION

ORA Project 06779

J. L. MOSS

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TABLE OF CONTENTS

INTRODUCTION

A. Scope

Under contract with Chrysler Corporation Defense Engineering, The University of Michigan Ship Hydrodynamics Laboratory undertook water tests in connection with Marsh Screw means of propulsion. The areas of investigation fell into two main categories:

a) a systematic approach to determine the effects of displacement, blade height and helical angle over a wide range of these variables

b) tests and analysis of data of various other variables which were considered either less basic than those under a) or where relatively few tests were needed to investigate them.

Those items under b) became known as Phase I and under a) as Phase II.

In late 1963 and early 1964 a limited investigation of Marsh Screw water propulsion was undertaken and reported to Chrysler Defense Engineering.*

In the current report previous results are drawn upon with regard to effects of blade cross-section shape variations and number of blades. Although some of the testing for these items was

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*J. L. Moss, "Marsh Screw Vehicle Rotor Model Test Results," The University of Michigan Ship Hydrodynamics Laboratory Report 06100-1-F, January 1964.

accomplished under the previous contract results are reported herein for the sake of completeness where necessary. A total of results of six tests from the earlier testing program are included. For an outline of further conditions under Phase I, the reader is referred to Section I in the Table of Contents of this report.

Under Phase II rotors which filled out a matrix of blade height and helix angle variations were tested. Ten rotors were tested, each at three displacements, within a matrix of four helix angles and four blade heights. All models under Phase II were eight inches hub diameter and 48 inches long. Section II gives the results which are presented in non-dimensional coefficient form for use in extrapolation to any size. Methods of using the results of Section II are given in a design example in Section IV.

Typical photographs of Phase II tests are included as frontispieces in order to familiarize the reader with the running attitude, spray patterns, etc. observed.

In all cases, Phase II results and individual items under Phase I, the results were analyzed largely by means of crossplotting with a view towards pointing out possible optimum configurations or effects of variations as well as towards providing design information. In fact, Section IV contains a brief design example which is mainly intended to illustrate the procedures to be used in employing the results as reported.

Section III contains analysis of data with regard to scaling from model to full scale. Initially the Phase II tests were to be run on 24 inch long models two of which were run. The results, when compared to those on 48 inch rotors of identical configuration, showed lack of correlation. In summary, it is felt that reasonably good full scale estimates of torque and RPM can be made by adhering to Froude similitude only, and if the model lengths of the configurations tested are about 48 inches or more in length. The results from 24 inch rotors exhibited appreciable scale effects. The reasoning which leads to these statements will be found in Section III.

It is strongly emphasized that all tests under Phase II were of rotors and the rest of any supposed vehicle was not introduced. The effects of shrouding, chain housings, etc. can only be estimated from the results under Phase I. That is, accuracy of prediction of RPM and horsepower from Phase II will suffer, but optimum configurations of rotors can still be determined. In ship design the interaction effects are thought of in terms of wake and thrust deduction fractions. The naval architect has a reasonably good estimate of these factors from many years' backlog of experience with ships similar to those which may be of current interest. Therefore, with the aid of systematic screw propeller data it is possible to efficiently combine hull form and propeller, model tests being utilized for refinement and confirmation of the design. In the case of Marsh Screw propulsion, accurate prediction of exact speed and power relationships for an arbitrary design is somewhat elusive owing to the reasons cited here.

B. Testing Method and Apparatus

Most models were supplied by Chrysler and were built of aluminum. Some model construction and changes were accomplished in the University of Michigan shops. Blades were tack-welded to the hubs as helical surfaces and the heights were diminished near the ends to nearly zero. The hubs were tapered at the ends to fair into a cone at the leading end and a chain housing at the stern. The dynamometer and rotor assembly are shown schematically in Fig. 1.

Two degrees of freedom, trim and sinkage, were allowed by the necessary hardware. That is, in the running condition the rotors were allowed to seek their equilibrium angle of attack and vertical position. Measurements of torque, RPM, trim, sinkage and speed of advance (carriage speed) were made. Test runs were made at the model propulsion point. With the towing carriage at constant speed the drive motor speed was varied until the thrust output balanced the hydrodynamic drag at which time the force transducer had zero output.

The maximum speed at which any given rotor was run was limited by one of two factors. The first was the maximum load which the electric motor would absorb. The second was with regard to the degree of freedom in trim.

As shown in Fig. 1, in order to allow trim the horizontal distance between the two sliding rods was allowed to vary by means of a small sub-carriage. Due to the mechanical arrangement the carriage travel was limited such that about 14 degrees bow up was the maximum trim possible. In the design of the test rig allowance for higher angles of trim was not provided in that such angles seemed unrealistic for any proposed vehicle operation. There were cases where, as speed increased, trim would have exceeded 14 degrees within the normally run speed range.

Generally, except for these two possibilities, all tests were run up to about 16 feet per second.

C. General Observations

There were, during testing and subsequent calculations, a number of observations regarding the flow and general running behavior of the models which point out basic characteristics which should be kept in mind in any design application.

There is reasonable analogy between a Marsh Screw rotor and either a ship or propeller which is pointed out in Section III of this report. In the scale effect analysis of that section it was assumed that all the energy is expended in the form of either viscous flow, waves, or spray. It is pointed out that spray and frictional drags are considered the largest single contributors. Wave drag is the least predominant. These considerations are important in that because the large amount of spray flow was uninhibited in the testing program, the same may not necessarily be expected in an actual vehicle. The proximity of the hull and/or shrouds could alter the spray characteristics. In fact, such a consideration is the main cause of the differences found in the shroud tests reported in Section I, I.

Another observation pertinent to running attitude is the similarity between a rotor and a planing boat, particularly with regard to trim and sinkage. At slow speeds, before the dynamic forces on the bottom of a planing surface begin to develop, the boat sinks slightly and usually trims bow-down or zero, but only a small amount. With increasing speed the boat begins to trim bow-up to as much as several degrees and the sinkage becomes more and more positive (up) until when after planing is fully developed the trim decreases slightly with increasing speed and the sinkage becomes nearly constant at a positive value. Curves of trim and sinkage for the rotors exhibit similar tendencies although the forces involved would not be directly analogous to those on a planing surface. Unlike most planing surfaces the rotors trimmed to maximum angles of up to 14 degrees bow-up. In some cases, the speeds were not sufficiently high to determine if the trim would decrease at the highest speeds. Also, generally sinkage did not reach a nearly constant positive value but was still increasing with speed.

These observations are stated so that the reader will have a good feeling for the general running characteristics when reading the remaining sections of this report.

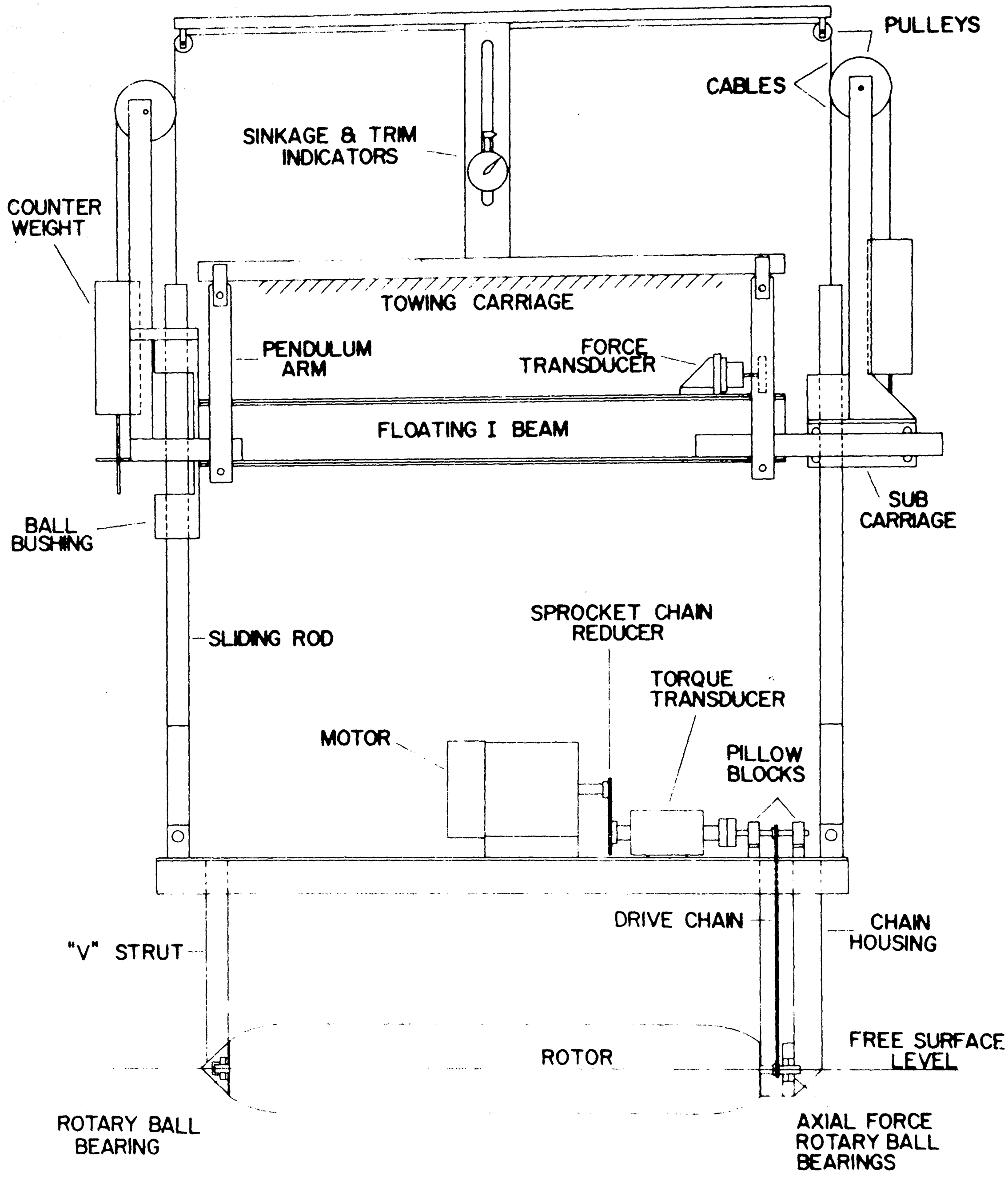


FIG. I MARSH SCREW ROTOR TEST FIXTURE

Section I: Phase 1

The test results reported in this section are of effects of a series of limited investigations of configuration changes and features a designer would expect to be incorporated into a specific vehicle. Within this group no set of tests was intended to thoroughly explore any particular aspect of rotor propulsion, but merely to indicate trends in horsepower absorption. Some tests were designed to study the interaction between rotors and other parts of a vehicle, e.g. chain housings, shrouds, and between the rotors themselves. In each of these groups of tests, variations in geometry were done one at a time. For instance, dual rotor tests were run without shrouds since the shroud tests were run on a single rotor. Therefore, it should not be expected that had dual rotors with shrouds been run that the results would be entirely in agreement with either of the individually run variations.

It should be emphasized that each test of the entire investigation reported in this report was run on a single rotor of constant 8 inches hub diameter, 48 inches in length, constant helix angle, constant blade height, symmetric end for end, with only the chain housing and rotor, and two blades except the variations in this section and the additional 24 inch long rotors used for the scale effect study of Section III.

Power and revs reported in this section are presented either as percentage or full scale values depending upon which seemed appropriate in the individual group of tests.

A. Effect of Initial Trimming Angle

Near the beginning of the testing program, a series of tests was conducted in order to determine the optimum zero speed trimming attitude in that it was anticipated that torque and RPM would be influenced by the initial longitudinal center of gravity position. The sequence of tests was made on the following rotors in the percentage of hub displacement indicated and over a range of from about two degrees bow down to four degrees bow up.

	Helix Angle	Blade Ht.	% Displ.
1	40 deg.	1.56 in.	25
2	"	"	50
3	"	"	75
4	"	1.875 in.	25
5	"	"	50
6	"	"	75
7	50 deg.	1.56 in.	25
8	"	"	50
9	"	"	75
10	"	1.875 in.	25
11	"	"	50
12	"	"	75

The results are presented as torque and RPM percentages, based on values at one degree trim, and as contours of speed for 5 and 10 mph using a linear scale ratio of three. See Figs. I-A-1 and I-A-2. The figures show that the optimum angle is somewhat a function of the percentage displacement and speed as well as the geometry of the individual rotors, but generally one degree bow up initial trim gives the least horsepower absorbed. This value was adopted for all tests under Phase II and corresponds to a vehicle with LCG somewhat aft of the mid-length of the rotors. For vehicles with center of gravity positions different than that corresponding to one degree bow up initial trim, estimates of power increases can be made from the graphs in this section.

B. Effect of Length-Diameter Ratio

Most of the model configurations were of rotors of L/D of six, i.e. 48 inch long rotors with a hub diameter of 8 inches. However one set of configurations was designed to investigate the effect of varying the diameter, holding the length of the models constant. The "parent" model in this series was the 50 deg. helix angle, 1.875 inch blade height rotor with an 8 inch hub diameter. Based upon a linear scale ratio of three, the latter figure corresponds to 24 inches full scale. Three other models were tested with full scale hub diameters of 14.25, 18 and 36 inches. Blade heights and hub diameters in the tapered ends were in the ratio of hub diameters. For instance for the 18 inch diameter rotor the blade height was

$$(3 \times 1.875)\left(\frac{18}{24}\right) = 4.22 \text{ inches full scale}$$

All the rotors were tested in two displacement conditions, 40% and 60% of the total hub volume in each case, except the 18 inch full scale diameter rotor which was not tested in the 60% condition owing to a shaft failure on the model. Adoption of constant percentage displacement meant that actual volume displacement varied from one rotor to another.

Figs. I-B-1 and I-B-2 give full scale horsepower and RPM, respectively, for the rotors in the 40% displacement condition, and similarly in the 60% condition in Figs. I-B-3 and I-B-4.

C. Effect of Number of Blades

In all cases rotors had two blades, or leads, except those exceptions reported in this section.

In the previous program one rotor had three blades, uniformly spaced, and was run fixed at zero sinkage and trim. The geometric properties are delineated on Fig. I-C-3 where the horsepower versus speed curves are given for two-bladed and three-bladed, but otherwise identical, rotors based on a linear scale ratio of three. The corresponding full scale hub diameter was 26 inches. Evidently the presence of three blades is deleterious to the lift developed on each even though total blade area is increased. In the opposite direction would be a one-bladed rotor which was tested in the current program in two displacements and with other geometric properties as given in Figs. I-C-1 and I-C-2. Here the rotors were unrestricted in trim and sinkage. From the curves, it appears that by reducing the number of blades to one reduces blade area to an insufficient level so that the best trade between unimpediment of flow and blade area is accomplished by the use of two blades.

D. Effect of Rotor Stern Shape

Upon observation of the planing hull analogy, as pointed out elsewhere in this report, the effect of the shape of the hub was investigated. One rotor was built up in wax on the stern end only to complete the cylindrical shape rather than tapering the end as on the bow. This was done in order to have the stern of the rotor ventilate much in the same manner as that of the transom on a high speed boat. The chain housing drag also was reduced somewhat in that the underside of the housing was no longer wetted. The horsepower results are presented in Fig. I-D-1 and RPM in Fig. I-D-2. The reduction in horsepower is rather dramatic and is probably attributable to the lower trimming angle and better dynamic lift, the latter as indicated by reduced sinkage, as shown in Fig. I-D-3.

This is a good opportunity to stress that in not thorougly investigating those configuration changes reported in Section I the danger of arriving at misleading conclusions is introduced. While the paragraph above, and accompanying figures, conclusively shows the desirability of the cylindrical stern the amount of horsepower reduction will vary according to the rest of the geometry. On the other hand in other sections a change in configuration may give adverse results, but complete abandonment of the original idea may not be warranted. Perhaps by a slightly different selection of design change beneficial results may have been obtained. This may be true of the rotor with varying helix angle. In any actual design model tests of the complete configuration are recommended.

E. Effect of a Length-Wise Varying Helix Angle

Following the reasoning that the blades probably tend to accelerate the flow towards the stern of the rotor one rotor was designed to compensate for this effect by incorporating a higher pitch at the stern than at the bow. The helix angle varied linearly from 35 degrees at the bow to 50 degrees at the stern. The full scale blade height was 4.68 inches based on a scale ratio of three. Three displacements were run.

It is difficult to say what rotor would be most nearly comparable. The average helix angle is 42.5 degrees but it cannot be assumed that the power absorption should correspond to that of a rotor with constant helix angle of 42.5 degrees. Figs. I-E-1 through I-E-3 show the horsepower and RPM at each of the three displacement percentages, respectively. At high speed only adverse effects of the varying helix angle are shown, but at low speeds the effects are not easily concluded. Perhaps an angle distribution other than linear would be best, but in any case owing to the marginal reduction in power at low speeds other helix angle variations may be recommendable in any specific design.

F. Effect of a Varying Hub Diameter

Another configuration which incorporated a hub geometry change was run on a model which had a linearly varying diameter. That is, but for the shaped ends, the hub was in the shape of a truncated right cone. The conical shape was continuous over the middle 36 inches (6 inches on each end for the shaped ends) with a 7 inch diameter near the bow and 9 inches near the stern. The blade height was 1.56 inches with a constant helix angle at the surface of 50 degrees. The tests were run at two percentage displacements, 40% and 60%. A nearly equivalent rotor with a cylindrical hub is the 50 degree, 1.56 inch, 8 inch diameter rotor.

The following table makes a comparison of horsepower per pound displacement for a scale ratio of three.

100 HP/Δ

V_{mph}	Conical Hub 40%	Conical Hub 60%	Cylindrical Hub 50%
5	.50	.47	.18
6	1.00	.90	.38
7	1.89	2.41	.65
8	3.65	--	.99

The table clearly indicates that the conical hub rotor of the configuration used offers no advantage over the cylindrical hub rotor.

G. Effect of Blade Cross-Section

The results presented in this section are drawn entirely from previous work already referenced. They are presented here again for completeness. The earlier tests were carried out on rotors 48 inches long and 8.67 inches hub diameter. As in all cases in the presently reported investigation the blade cross-sections were rectangular except the earlier investigation included a set of tests where a "cupped" blade was used. For details of the blade cross-section as well as the remaining rotor geometry refer to Figs. I-G-1 and I-G-2. The dimensions in the latter figure are full scale, based on a linear scale ratio of three, as are power and speed in the former.

The original test apparatus did not allow for freedom in heave and trim. For one test the sinkage and trim were fixed at zero. A second test was run with sinkage at zero and trim at four degrees bow up throughout the entire speed range. The results in Fig. I-G-1 show that with four degrees of trim, which is more realistic than zero but is generally less than the angles of attitude found in the present investigation, that the "cupped" blades were beneficial at speeds corresponding to in excess of 8 mph. However, both blades absorbed more power in the trimmed attitude than in the level attitude in which case the "cupped" blades were highly detrimental.

H. Interaction Effects Between Two Rotors

In order to investigate the interference between two rotors running side-by-side the testing apparatus shown in Fig. 1 was modified to allow two rotors to be run on parallel courses at various distances apart. The 50 deg. helix angle, 1.56 inch blade height model was chosen for the right-handed helix and a left-handed mate was constructed to be used for the second rotor. The longitudinal centerline spacing was selected as numbers of hub diameters. The actual model spacings are given below:

℄ Spacing in Diameters	Model spacing in inches
2	16
4	32
6	48

The left-handed model was run alone to ensure comparable results with the right-handed rotor. The small differences found have been averaged for computing RPM and torque for two rotors, or infinite spacing. All configurations were run inboard turning (i.e. starboard left-handed and port right-handed) except at the two diameter spacing where both inboard and outboard turning were run. Also, all tests were run in the 50% hub displacement condition.

The results are shown in Fig. I-H-1 as percentages of horsepower and RPM compared to two rotors at infinite spacing. Generally, small spacings are to be avoided owing to the adverse effects on power. Interestingly, there exists a narrow range of spacings, near four diameters, where the horsepower for two rotors is less than that for twice a single rotor. This may be due to cancelling effects on energies lost in spray, waves, etc. Such a phenomenon is possible to design into catamaran hulls in that the wave resistance of the twin hulls can be made to be less than twice that for one hull when the spacing is correct. Also analogous to the catamaran case is the fact that the cancelling effect is a function of rotor speed and might be expected to be a function of rotor geometry, hence spray patterns, as well. In the figure the percentages are plotted as a family of curves of speed in mph based on a linear scale ratio of three. For use with vehicles of a different scale ratio the percentages will remain the same when the speeds are converted by Froude's Law.

It should be kept in mind that the tests were run with no shrouds or hull simulated. In an actual vehicle, according to its geometry, the results shown in Fig. I-H-1 will vary.

The differences found between inboard and outboard turning were never more than one or two percent, outboard turning being slightly favorable.

I. Effects of Shrouds

As pointed out elsewhere in this report interaction phenomena are to be seriously considered. One attempt at determination of changes in horsepower required owing to the proximity of the rest of the vehicle to the rotor was designed as a limited investigation of the effects of shrouds. The rotor configuration chosen is given in the table on Fig. I-I-1.

Two shrouds were tested in various positions relative to the rotor, which always rotated clockwise looking from the stern. The shroud configurations were as follows

a. 180 deg. (i.e. from water surface, over the rotor, to water surface in the static condition), 1 inch space between the blade tips and the inside of the shroud

b. 180 deg., 2 inch space

c. 90 deg. port (i.e. from the water surface on the port side to top-center to simulate an outboard turning rotor), 2 inch space

d. 90 deg. starboard (i.e. from top-center to the water surface on the starboard side to simulate an inboard turning rotor), 2 inch space.

All shrouds were longitudinally taken segments of cylinders and were of the same length as the rotor. The spacings given above, as well as the horsepower versus speed results of Fig. I-I-1, are for a linear scale ratio of three.

The plotted results show that generally the lesser spacing is desirable, as well as shrouds of smaller included angle. The inboard turning rotation is desirable compared to outboard turning.

J. Effect of Chain Housing Drag

During testing it became apparent that the impingement of spray on the leading surface of the drive chain housing represented a significant amount of drag which the rotor had to overcome. A series of tests was designed which lead to an estimate of the power required to balance the drag of the housing.

The frontal area was about 4.5 inches wide and the height upon which the spray impinged was up to about 10.5 inches above the interface of the rotors and housing, dependent upon speed. Flat plates which extended the width, but held the height constant, were attached to increase the frontal area. One plate was 16.5 inches wide, or 3.67 times the area of the housing itself, and the other was 27.5 inches wide, or 6.11 times the area. This range of widths was chosen as it was estimated that about the same proportion of area of each would be affected by the spray over the speed range of interest. The resulting full scale horsepower curves using a linear scale ratio of three are shown in Fig. I-J-1. Fig. I-J-2 shows the resulting full scale RPM required.

From Fig. I-J-1 differences in horsepower between tests with plate extensions and the curve labelled "1 x Area" were plotted as a function of change in frontal area on Fig. I-J-3 and then extrapolated to zero area. The resulting ordinate intersects, although negative, represent an estimate of the induced horsepower for the chain housing as normally run. Subtraction of the zero area horsepower values in Fig. I-J-3 from the "1 x Area" curve in Fig. I-J-1 gives a curve of horsepower required to overcome the rotor drag only.

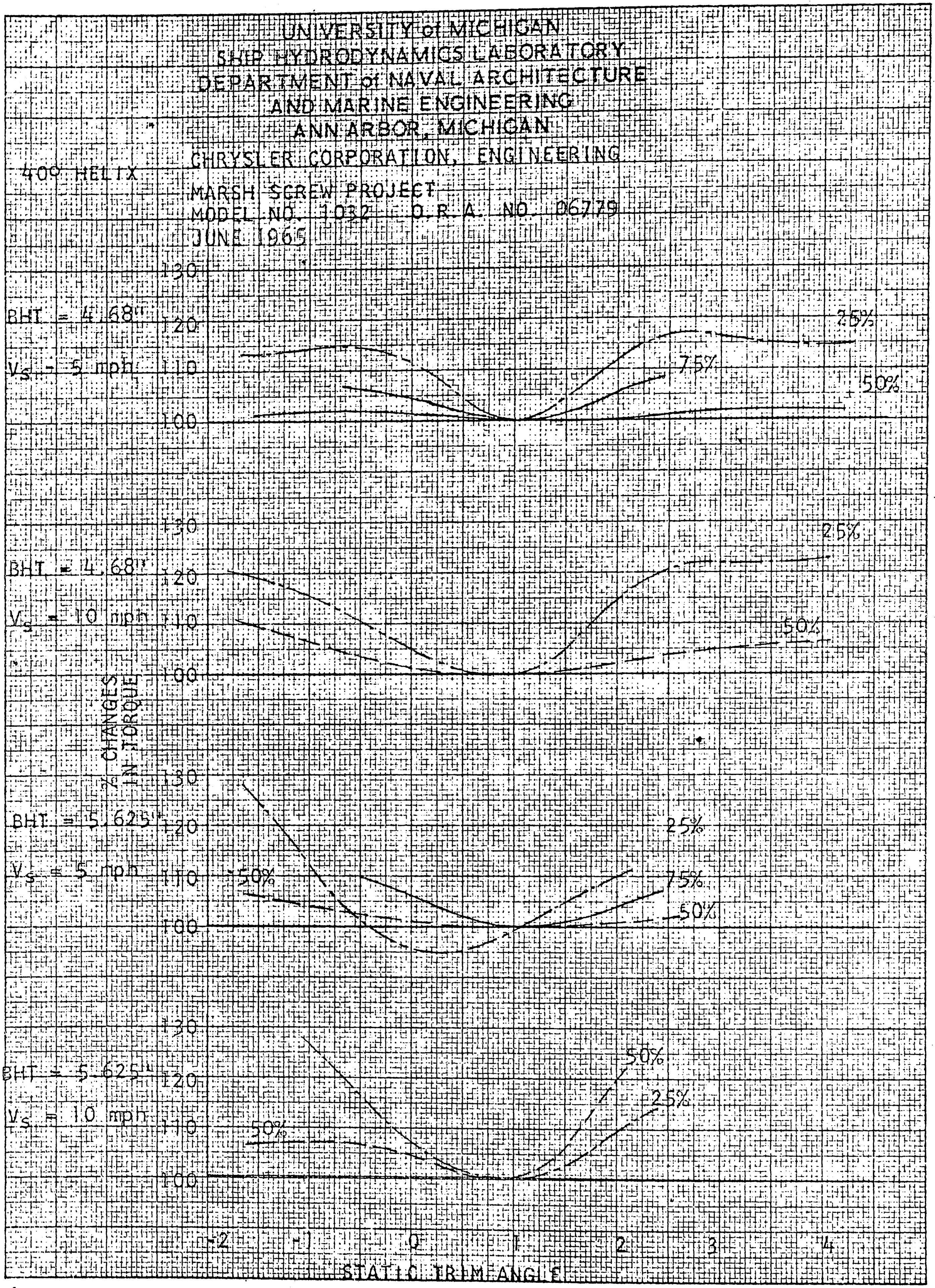
This method ignores the interaction between the plates and rotor, but does point out the desirability of maintaining the chain housing frontal area as small as practicable.

Interaction effects should be exhibited in changes in sinkage and trim between the various tests. Although these differences would affect the comparative horsepowers, it seems unlikely that the differences in Fig. I-J-3 were seriously affected.

K. Draw Bar Pull Tests

In one case a simple draw bar pull test was run on the model of the 50 deg. helix angle, 4.68 inch blade height rotor with displacement of 50% of the hub volume and initial trim of one degree bow up. The results are presented in Fig. I-K-1 as curves of tow bar force and RPM versus horsepower absorbed. Entering the graph with horsepower available, one can read out the revs and force for one rotor as indicated by the broken line. The results can be converted to any size rotor by adhering to Froude similitude.

FIGURE I-A-1



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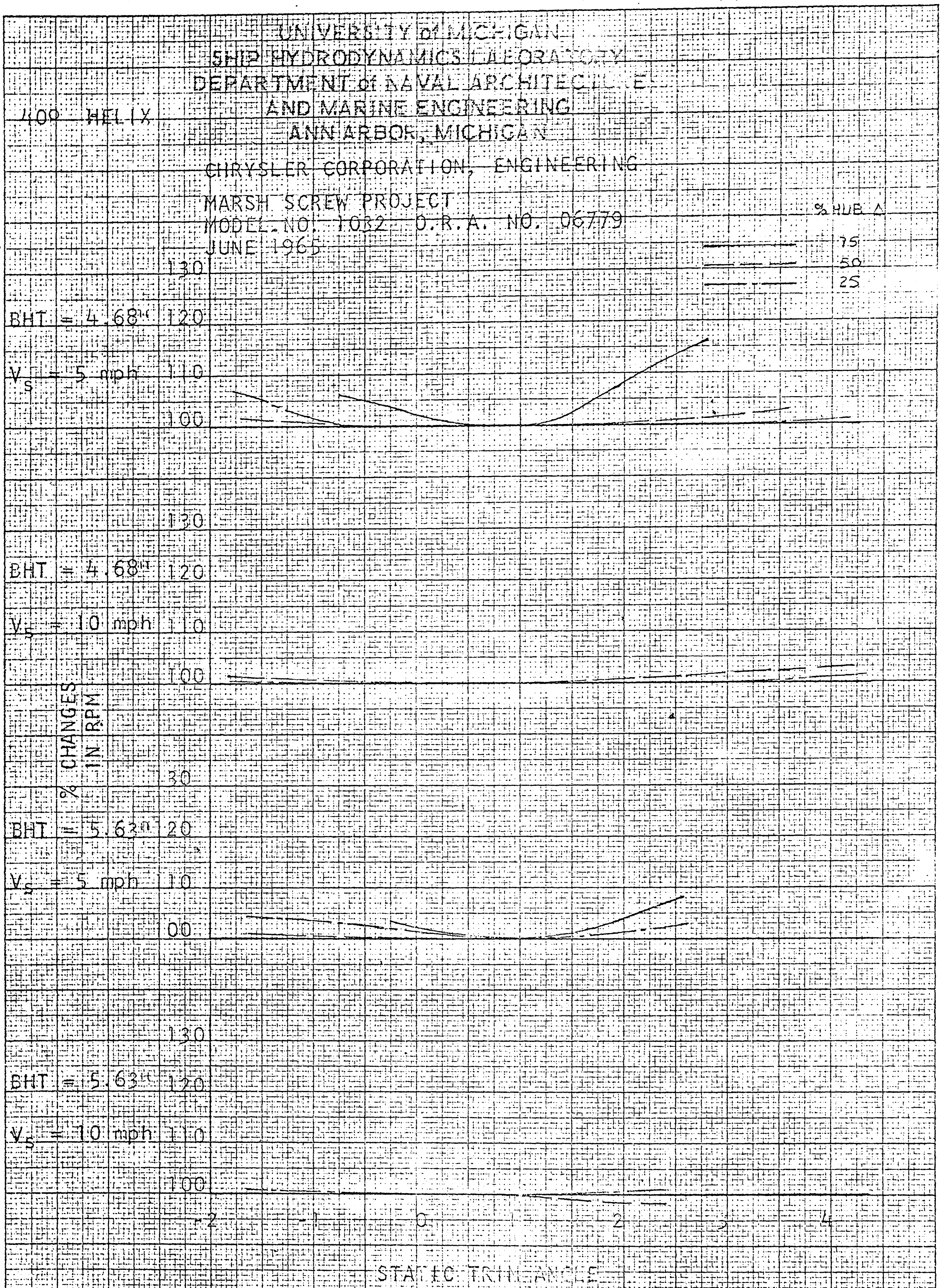
ANN ARBOR, MICHIGAN
CHRYSLER CORPORATION, ENGINEERING
40° HELIX
MARSH SCREW PROJECT
MODEL NO. 1032 O.R.A. NO. 06779
JUNE 1965
BHT = 4.68"
Vs = 5 mph
25%
75%
50%
BHT = 4.68"
Vs = 10 mph
25%
50%
% CHANGES IN TORQUE
BHT = 5.625"
Vs = 5 mph
50%
25%
75%
50%
BHT = 5.625"
Vs = 10 mph
50%
50%
25%
130
120
110
100
-2
-1
0
1
2
3
4
STATIC TRIM ANGLE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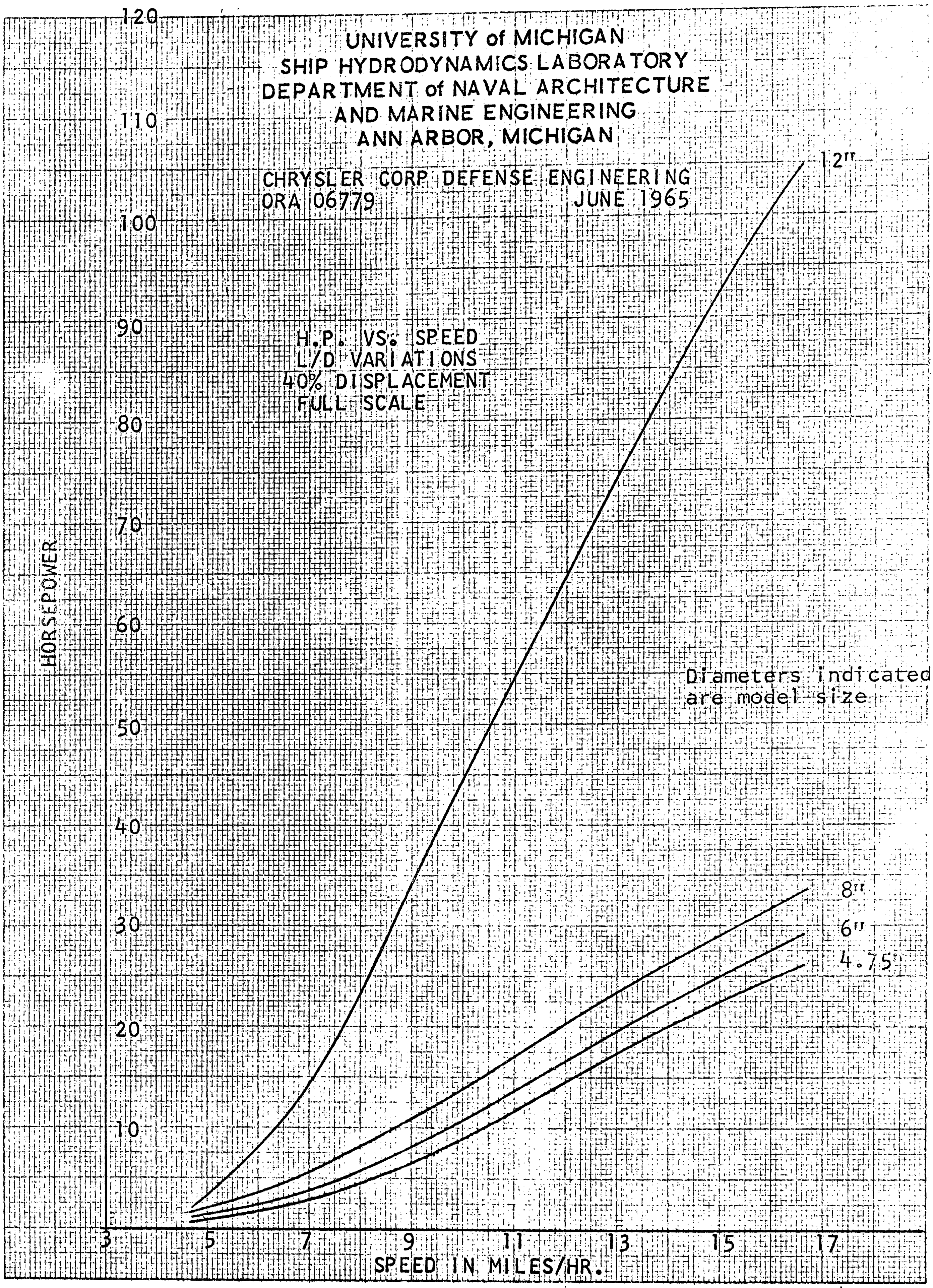
ANN ARBOR, MICHIGAN
400 HELIX
CHRYSLER CORPORATION, ENGINEERING
MARSH SCREW PROJECT
MODEL NO. 1032 O.R.A. NO. 06779
JUNE 1965
% HUB △
75
50
25
BHT = 4.68"
V_s = 5 mph
BHT = 4.68"
V_s = 10 mph
% CHANGES IN RPM
BHT = 5.63"
V_s = 5 mph
BHT = 5.63"
V_s = 10 mph
130
120
110
100
-2
-1
0
1
2
3
4
STATIC TRIM ANGLE

FIGURE I-B-1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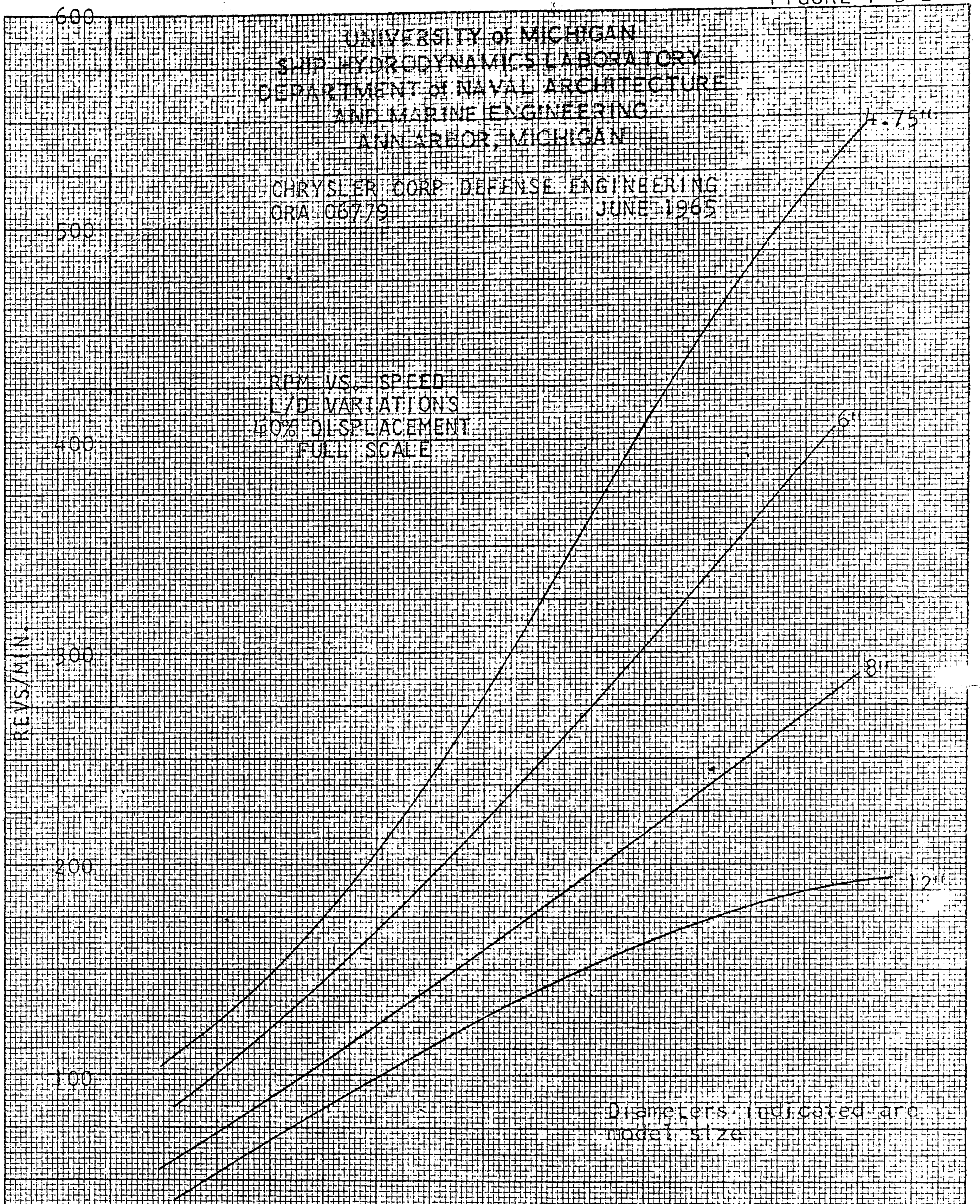
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H.P. VS. SPEED
L/D VARIATIONS
40% DISPLACEMENT
FULL SCALE
Diameters indicated are model size
12"
8"
6"
4.75"
HORSEPOWER
SPEED IN MILES/HR.
120
110
100
90
80
70
60
50
40
30
20
10
3
5
7
9
11
13
15
17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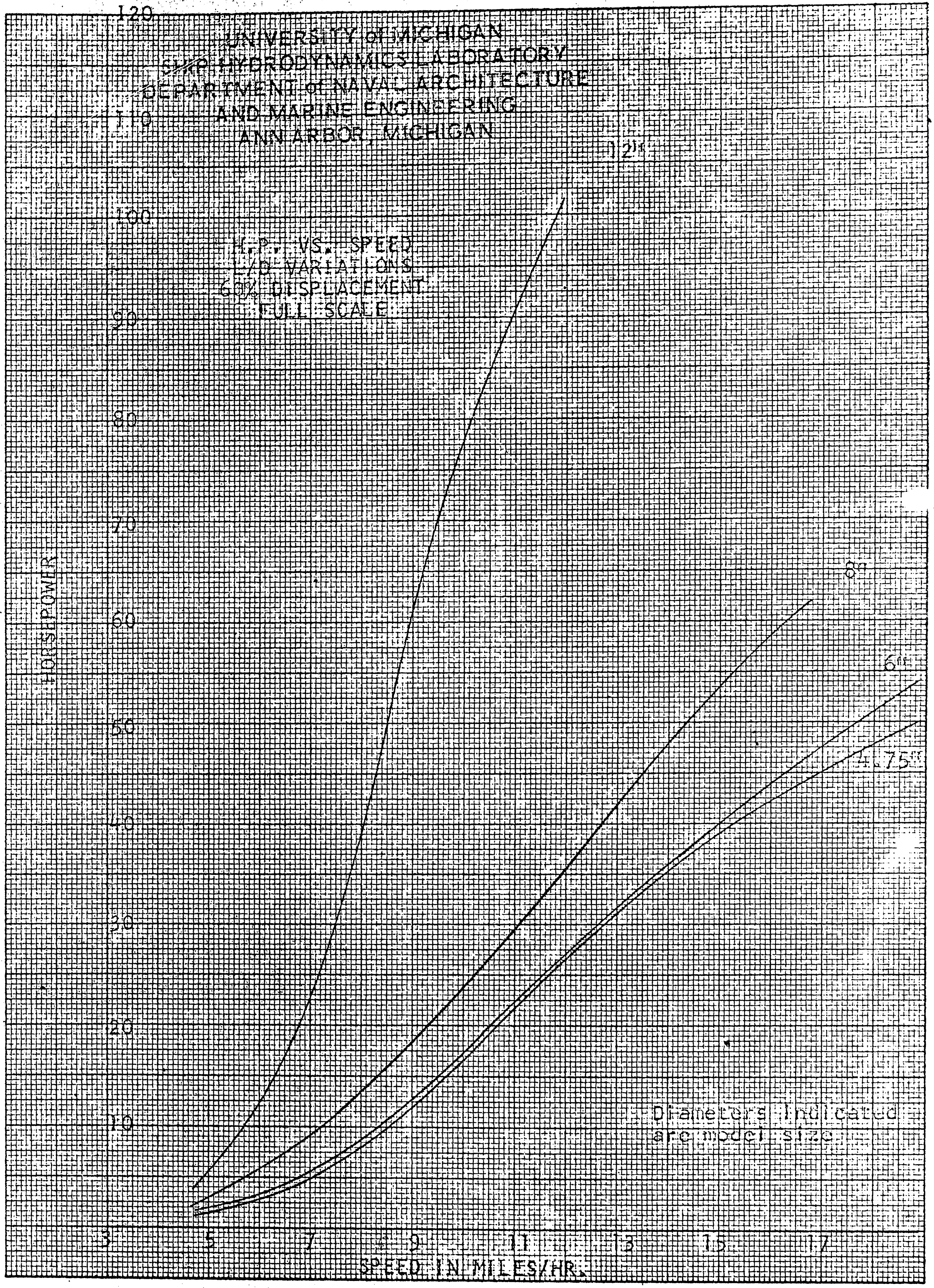
ORA 06779
JUNE 1965
RPM VS. SPEED
L/D VARIATIONS
40% DISPLACEMENT
FULL SCALE
4.75"
6"
8"
12"
Diameters indicated are model size
REVS/MIN.
SPEED IN MILES/HR.
600
500
400
300
200
100
3
5
7
9
11
13
15
17

FIGURE I-B-3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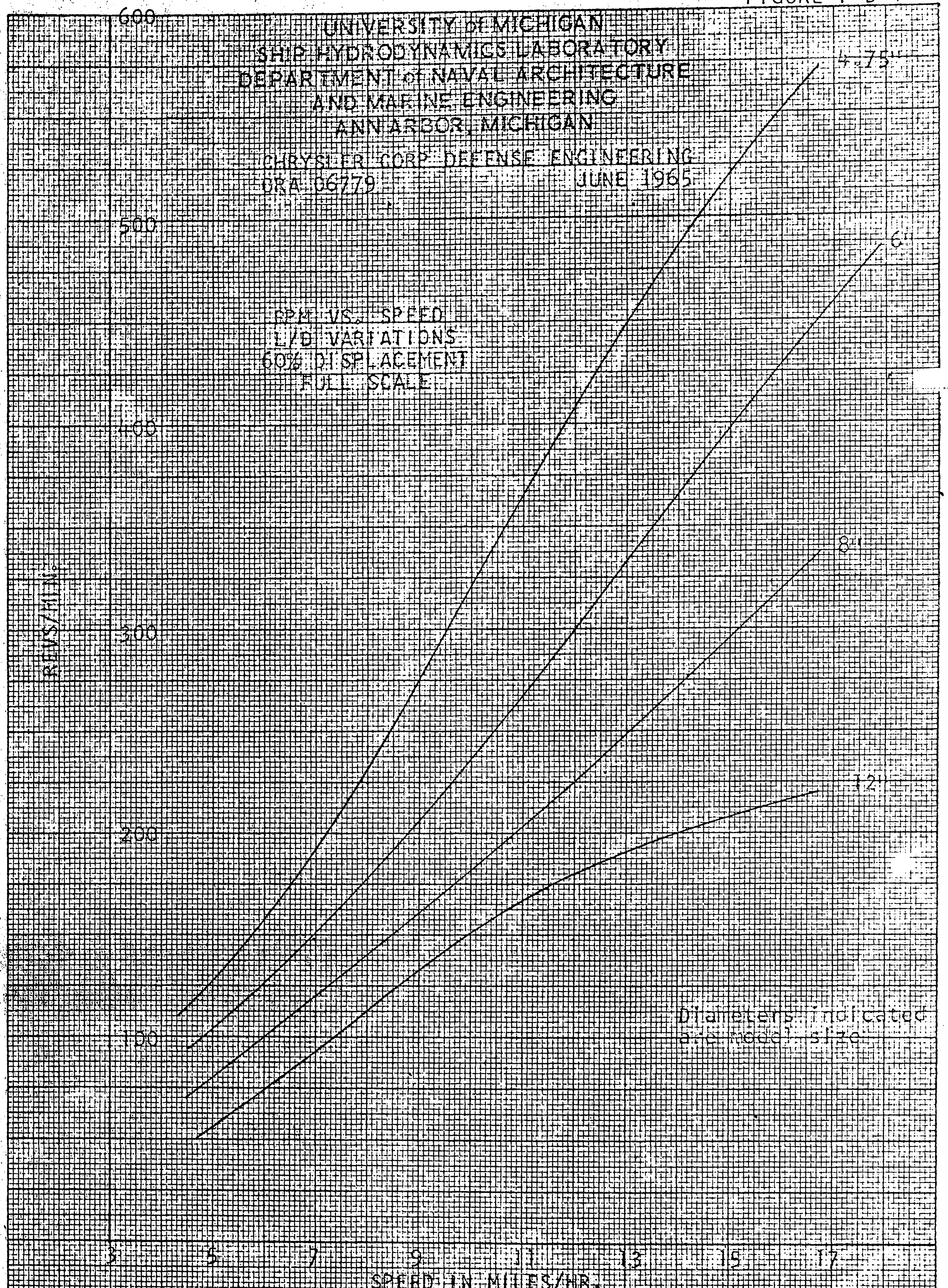
ANN ARBOR, MICHIGAN
H.P. VS. SPEED
L/D VARIATIONS
60% DISPLACEMENT
FULL SCALE
HORSEPOWER
120
110
100
90
80
70
60
50
40
30
20
10
12"
8"
6"
4.75"
Diameters Indicated
are model size
3
5
7
9
11
13
15
17
SPEED IN MILES/HR.

FIGURE 1-B-4



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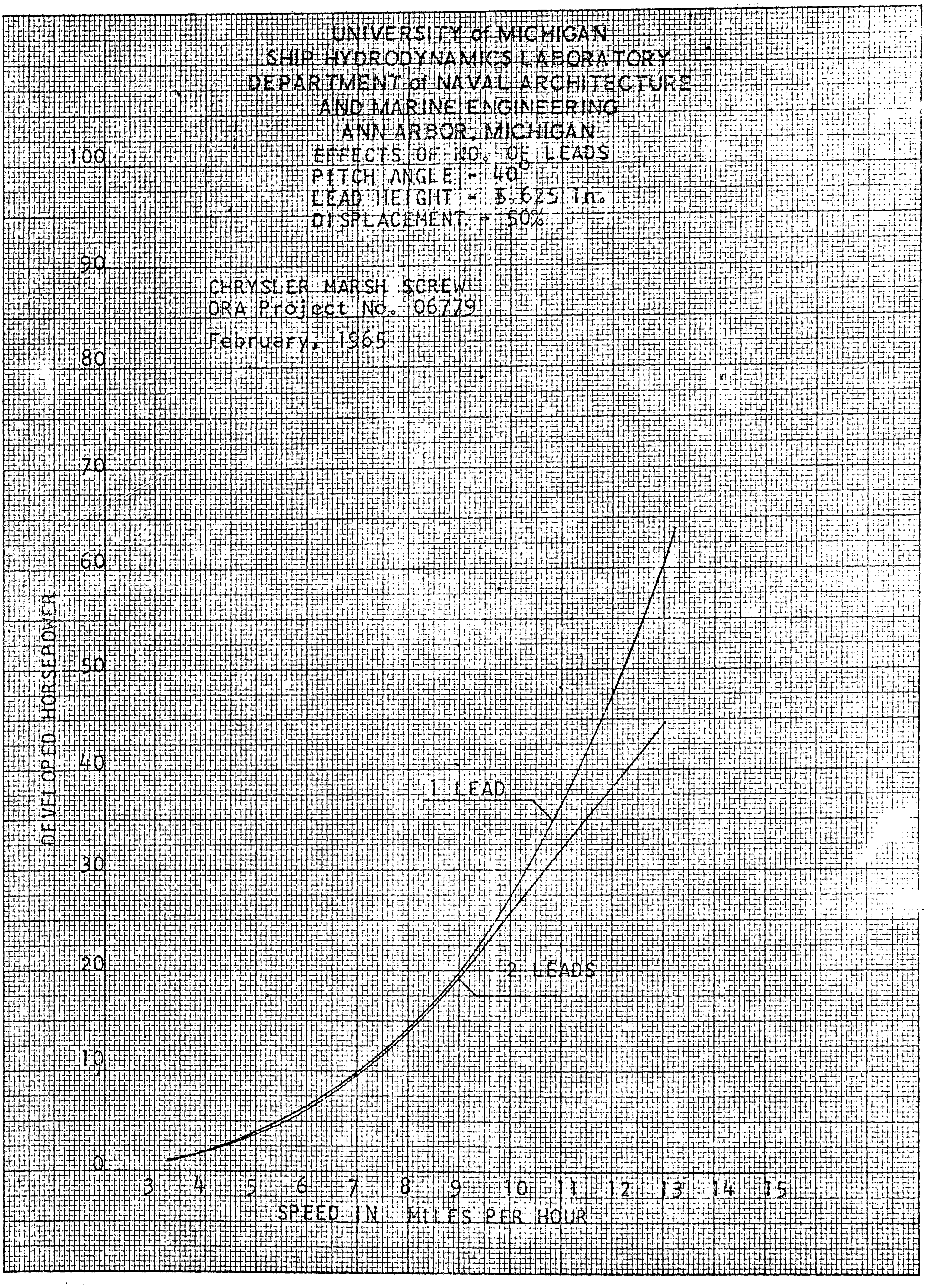
ORA 06779
JUNE 1965
RPM VS. SPEED
L/D VARIATIONS
60% DISPLACEMENT
FULL SCALE
REVS/MIN.
600
500
400
300
200
100
4.75"
6"
8"
12"
Diameters indicated
are model size
3
5
7
9
11
13
15
17
SPEED IN MILES/HR.

FIGURE I-C-1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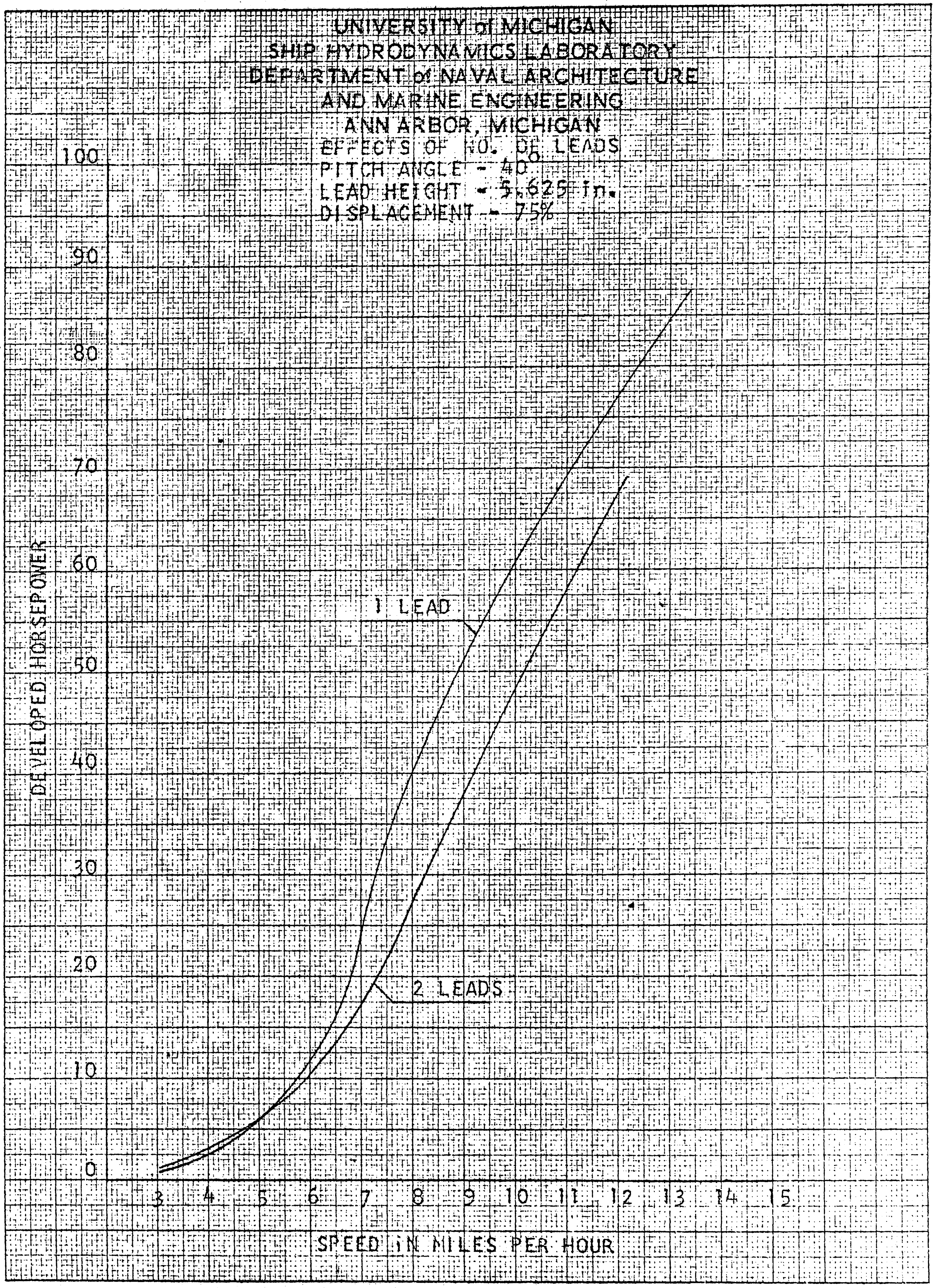
EFFECTS OF NO. OF LEADS
PITCH ANGLE = 40°
LEAD HEIGHT = 5.625 In.
DISPLACEMENT = 50%
CHRYSLER MARSH SCREW
ORA Project No. 06779
February, 1965
DEVELOPED HORSEPOWER
SPEED IN MILES PER HOUR
1 LEAD
2 LEADS
0
10
20
30
40
50
60
70
80
90
100
3
4
5
6
7
8
9
10
11
12
13
14
15

FIGURE I-C-2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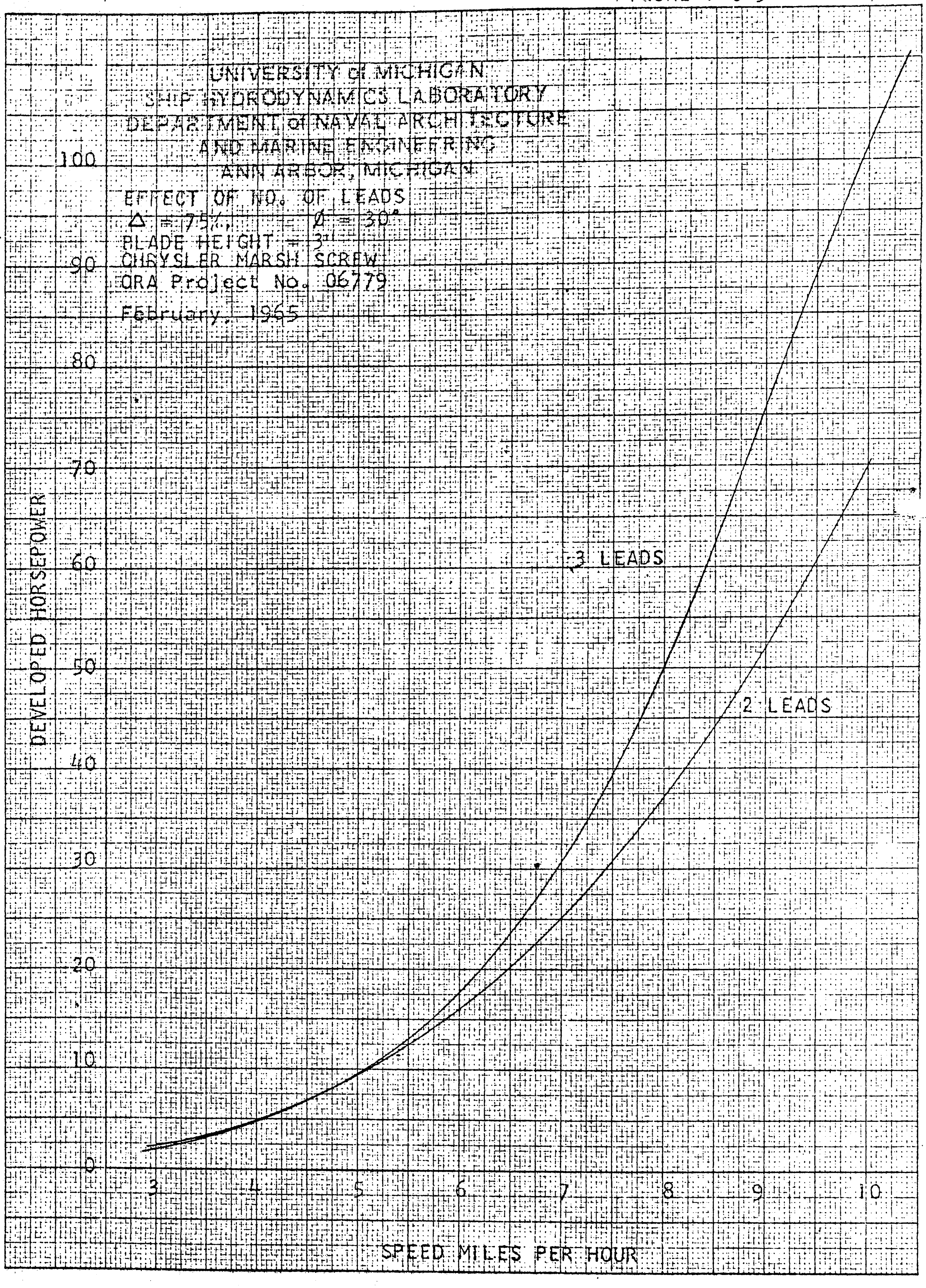
EFFECTS OF NO. OF LEADS
PITCH ANGLE - 40
LEAD HEIGHT - 5.625 in.
DISPLACEMENT - 75%
DEVELOPED HORSEPOWER
0
10
20
30
40
50
60
70
80
90
100
1 LEAD
2 LEADS
3
4
5
6
7
8
9
10
11
12
13
14
15
SPEED IN MILES PER HOUR

FIGURE I-C-3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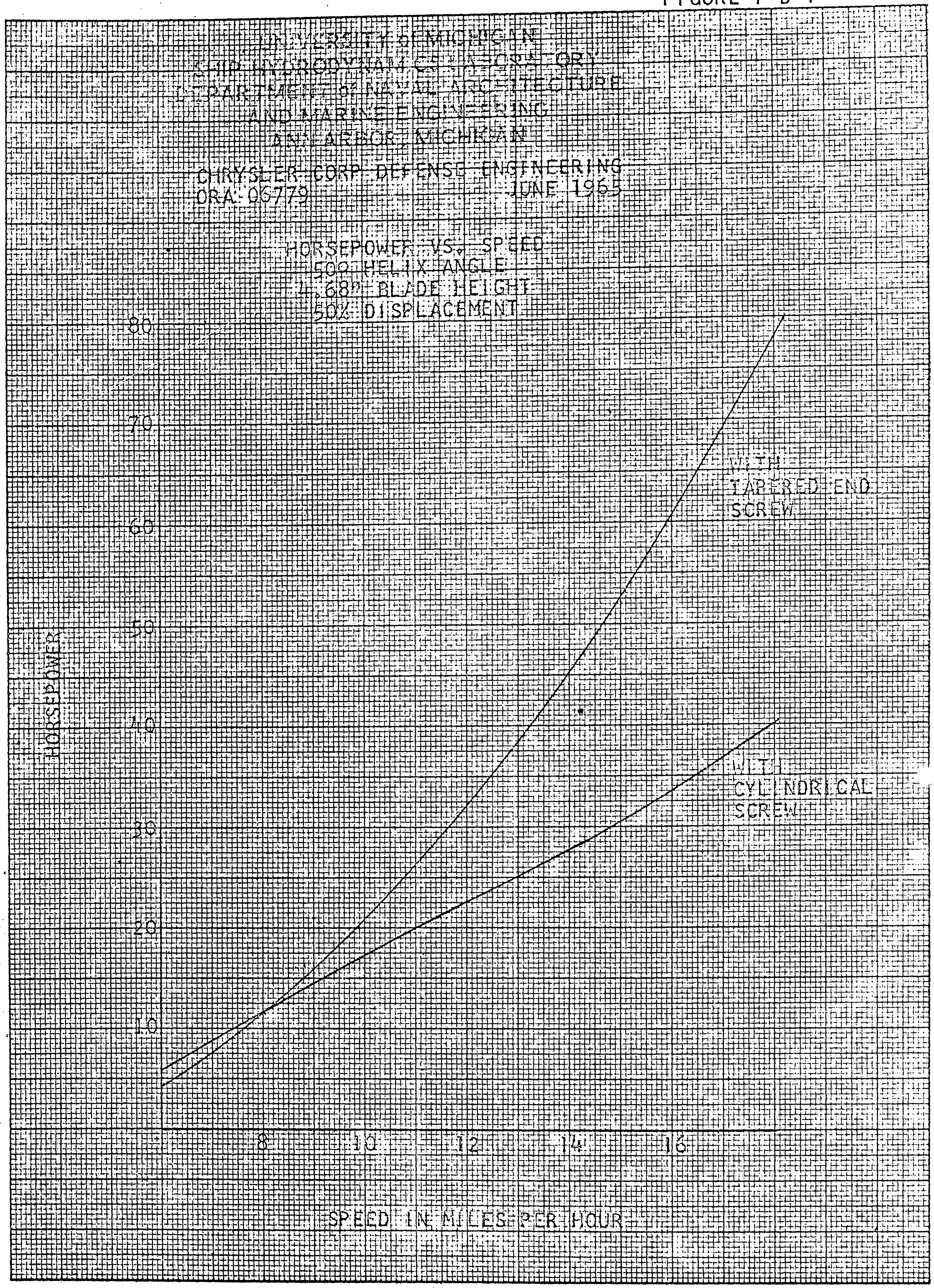
EFFECT OF NO. OF LEADS
Δ = 75%, Ø = 30°
BLADE HEIGHT = 3"
CHRYSLER MARSH SCREW
ORA Project No. 06779
February, 1965
DEVELOPED HORSEPOWER
SPEED MILES PER HOUR
3 LEADS
2 LEADS
0
10
20
30
40
50
60
70
80
90
100
3
4
5
6
7
8
9
10

FIGURE I-D-1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AND MARINE ENGINEERING
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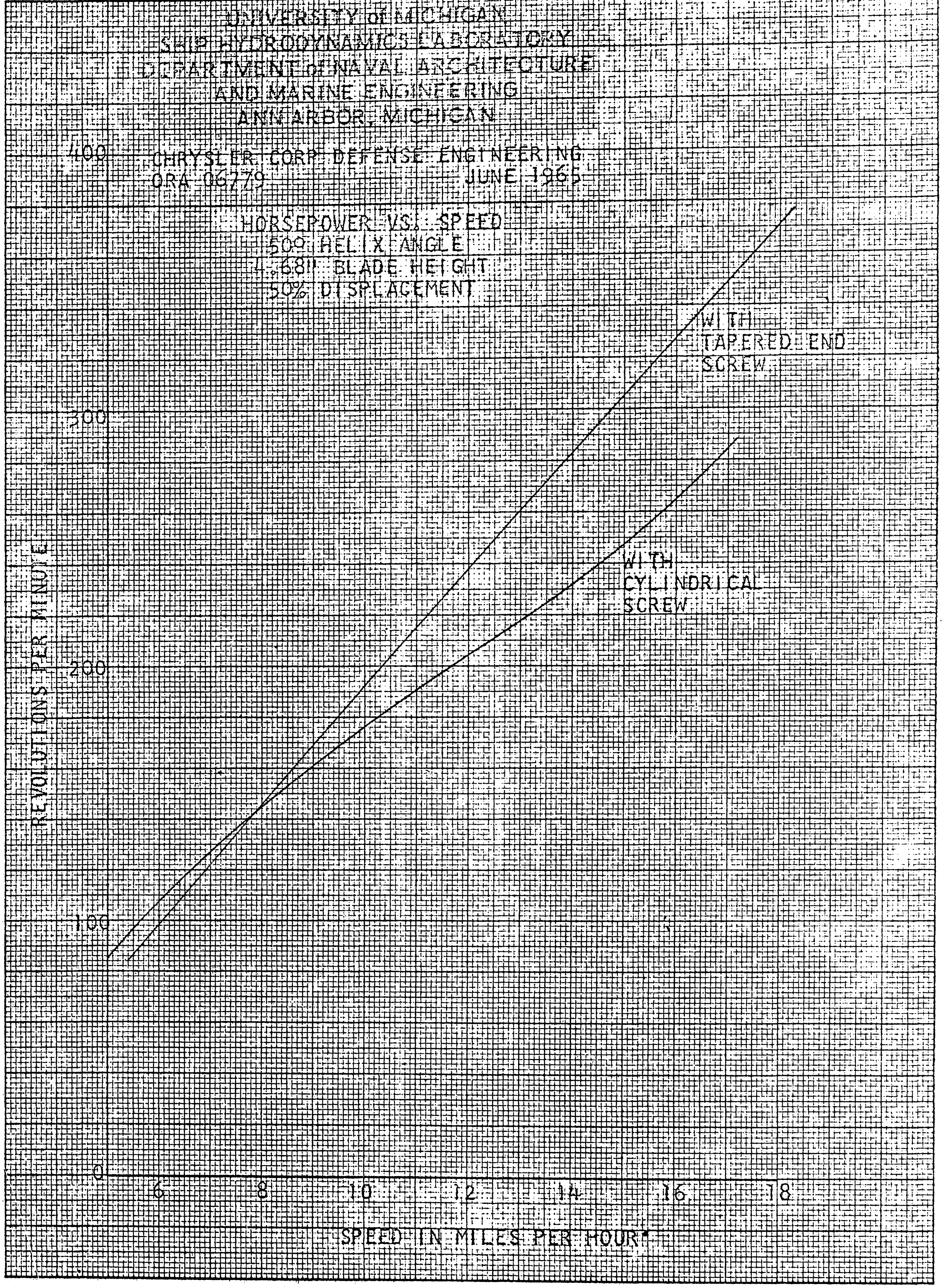
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
HORSEPOWER VS. SPEED
50° HELIX ANGLE
4.68" BLADE HEIGHT
50% DISPLACEMENT
HORSEPOWER
80
70
60
50
40
30
20
10
WITH TAPERED END SCREW
WITH CYLINDRICAL SCREW
8
10
12
14
16
SPEED IN MILES PER HOUR

FIGURE I-D-2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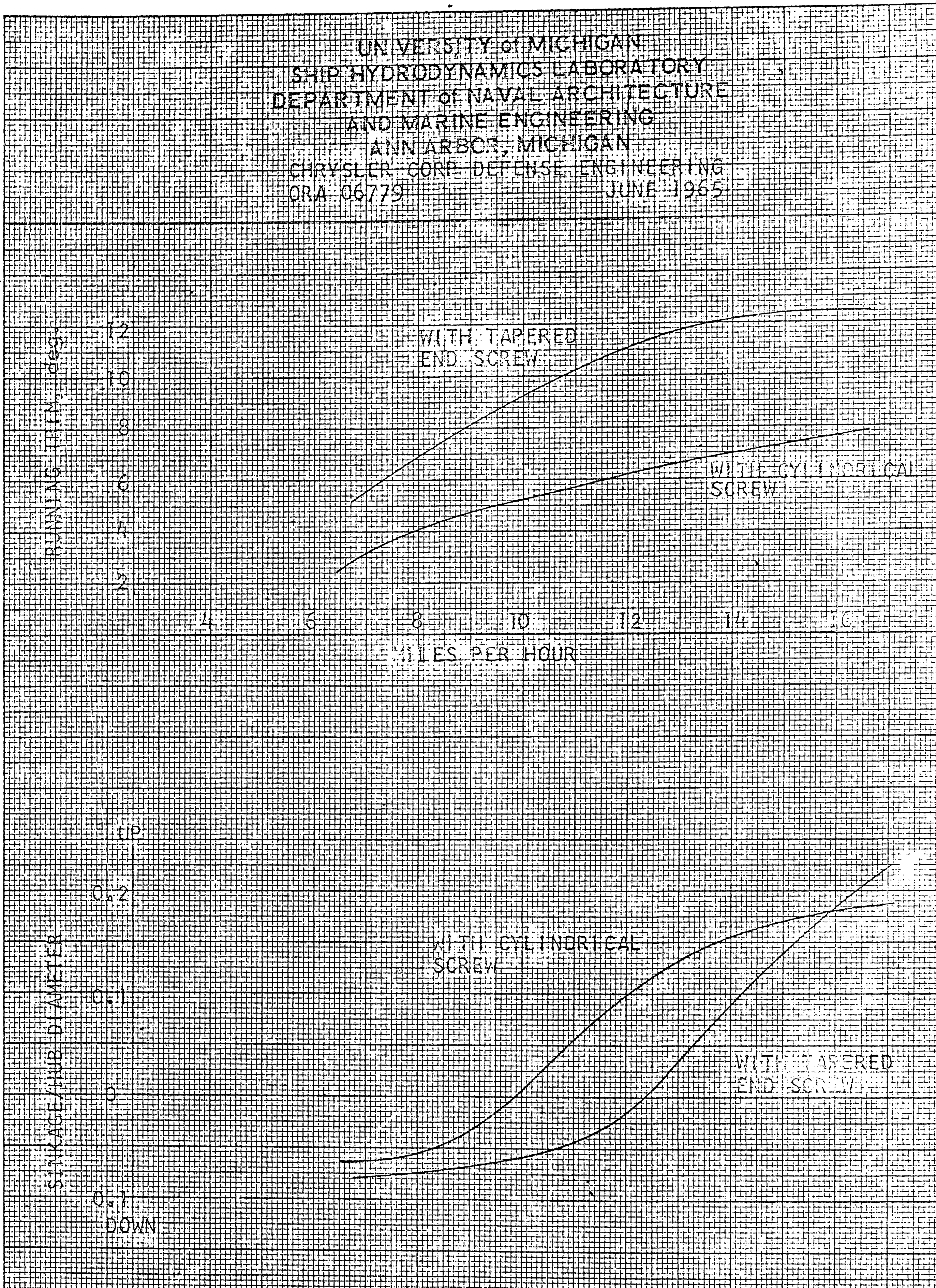
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
HORSEPOWER VS. SPEED
50° HELIX ANGLE
4.68" BLADE HEIGHT
50% DISPLACEMENT
WITH
TAPERED END
SCREW
WITH
CYLINDRICAL
SCREW
400
300
200
100
0
6
8
10
12
14
16
18
REVOLUTIONS PER MINUTE
SPEED IN MILES PER HOUR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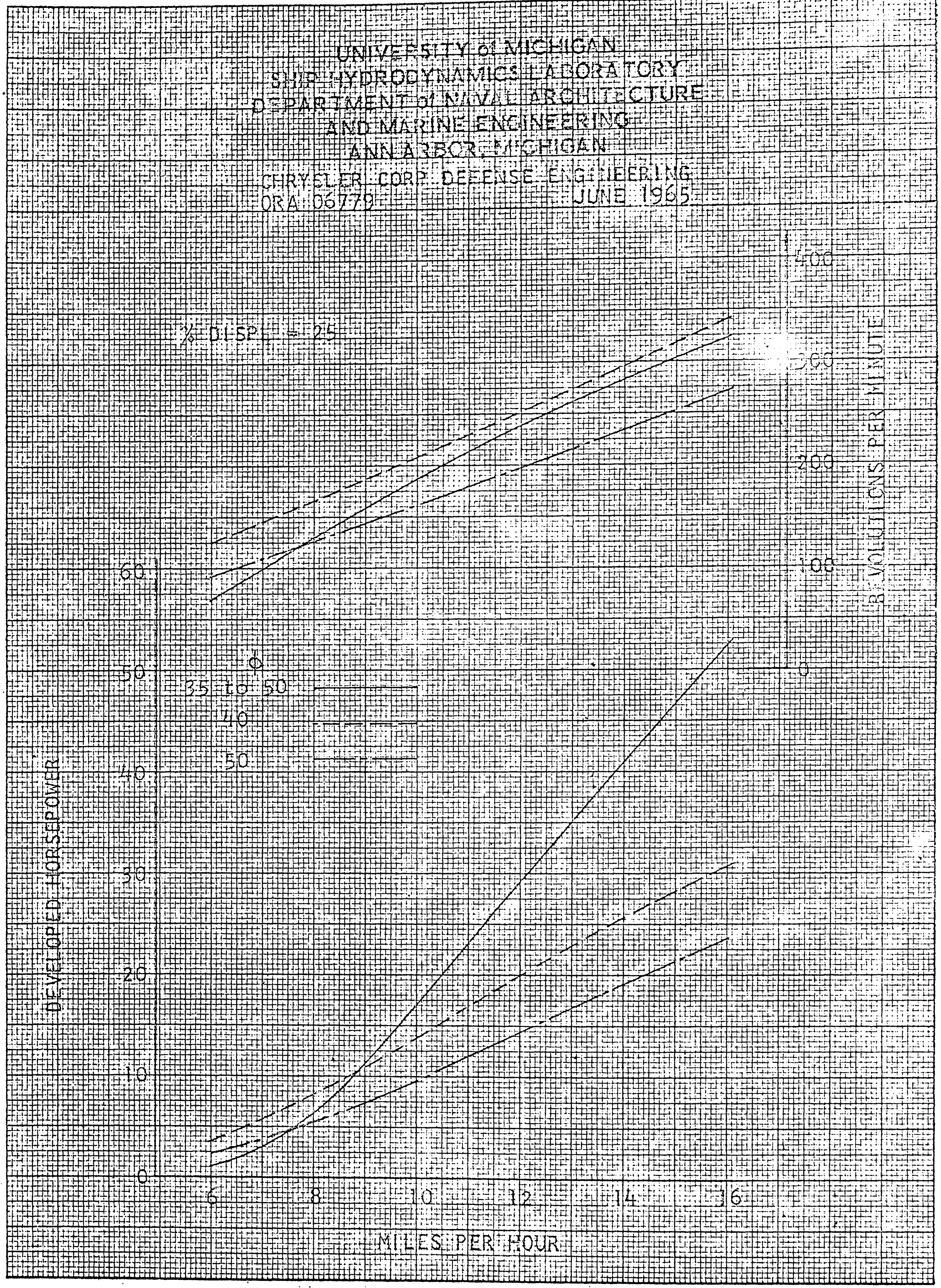
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
RUNNING TRIM, deg.
12
10
8
6
4
2
WITH TAPERED
END SCREW
WITH CYLINDRICAL
SCREW
4
6
8
10
12
14
16
MILES PER HOUR
SINKAGE/HUB DIAMETER
UP
0.2
0.1
0
0.1
DOWN
WITH CYLINDRICAL
SCREW
WITH TAPERED
END SCREW

FIGURE 1-E-1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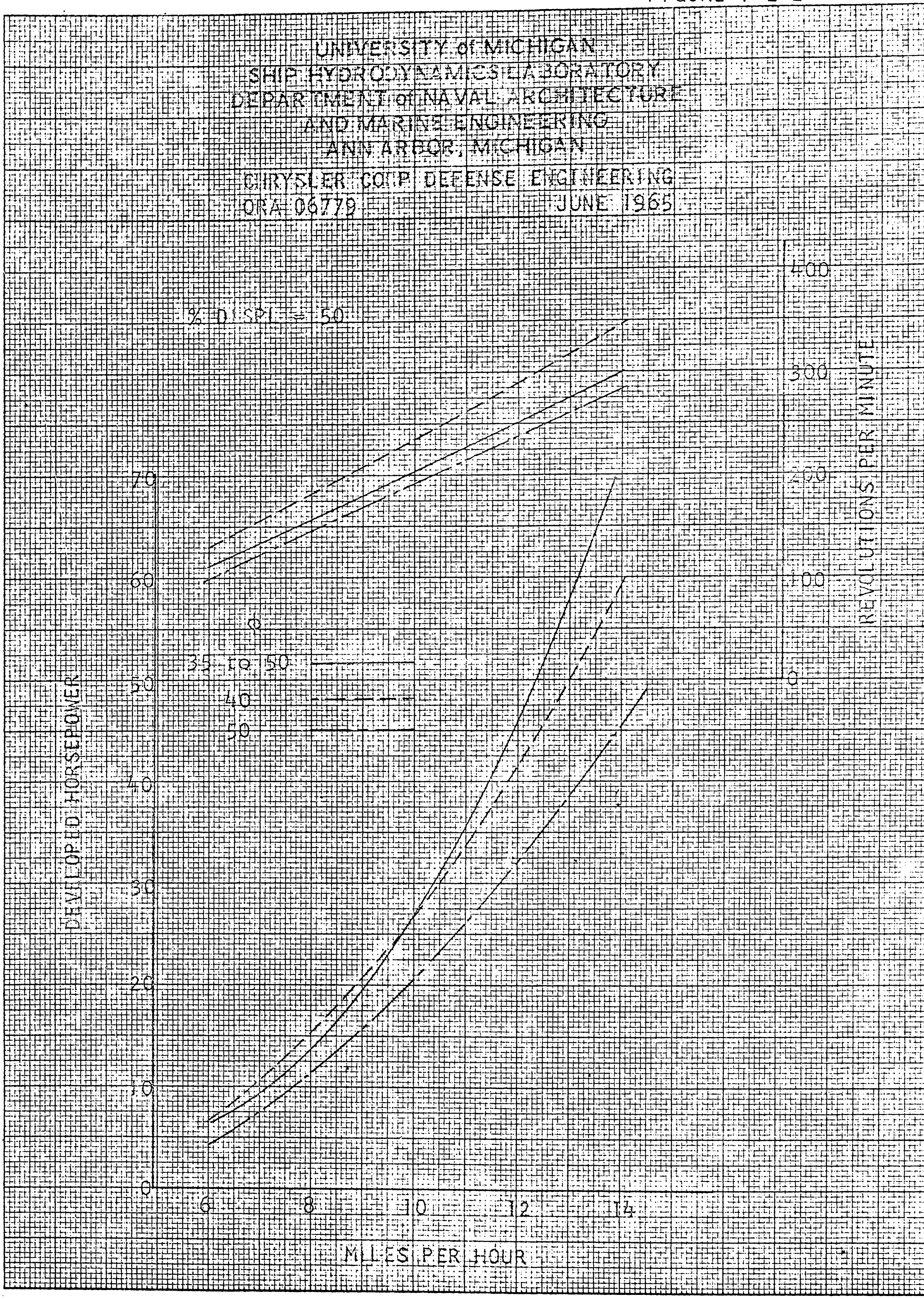
CHRYSLER CORP. DEFENSE ENGINEERING
ORA 06779
JUNE 1965
% DISPL. = 25
Φ
35 to 50
40
50
DEVELOPED HORSEPOWER
0
10
20
30
40
50
60
REVOLUTIONS PER MINUTE
0
100
200
300
400
6
8
10
12
14
16
MILES PER HOUR

FIGURE I-E-2



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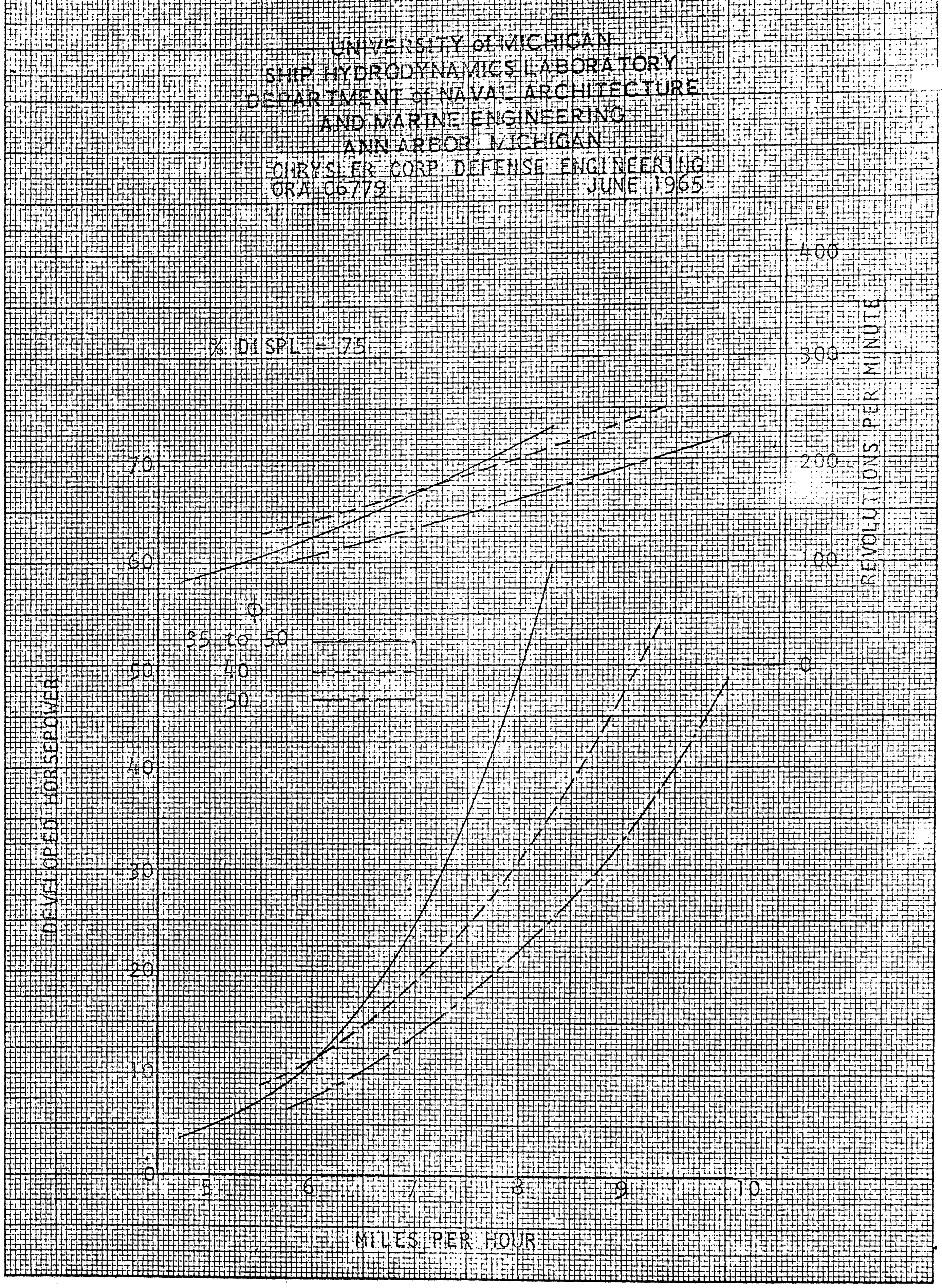
ORA 06779
JUNE 1965
% DISPL = 50
%
35 TO 50
40
50
DEVELOPED HORSEPOWER
REVOLUTIONS PER MINUTE
MILES PER HOUR
0
10
20
30
40
50
60
70
100
200
300
400
6
8
10
12
14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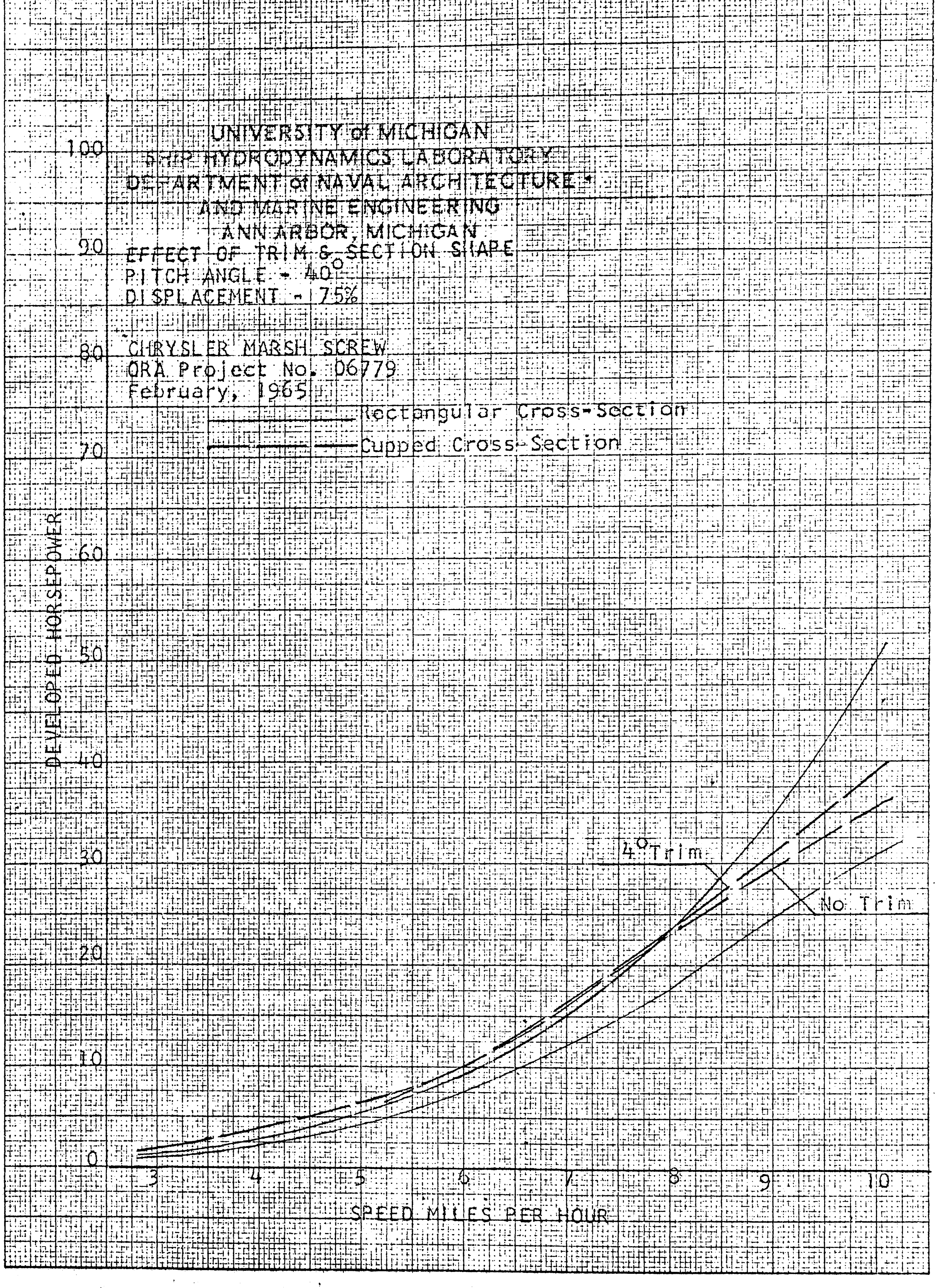
CHRYSLER CORP. DEFENSE ENGINEERING
ORA 06779
JUNE 1965
% DISPL = 75
Φ
35 to 50
40
50
DEVELOPED HORSEPOWER
REVOLUTIONS PER MINUTE
MILES PER HOUR
0
10
20
30
40
50
60
70
100
200
300
400
5
6
7
8
9
10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EFFECT OF TRIM & SECTION SHAPE
PITCH ANGLE - 40°
DISPLACEMENT - 175%
CHRYSLER MARSH SCREW
ORA Project No. 06779
February, 1965
Rectangular Cross-Section
Cupped Cross-Section
4° Trim
No Trim
DEVELOPED HORSEPOWER
SPEED MILES PER HOUR
0
10
20
30
40
50
60
70
80
90
100
3
4
5
6
7
8
9
10

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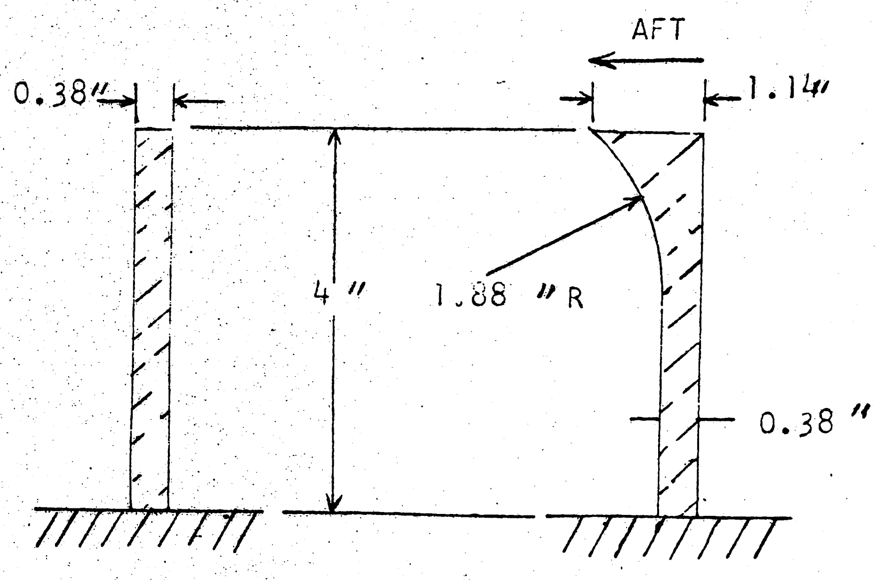
ANN ARBOR, MICHIGAN

MARSH SCREW ROTOR BLADE
CROSS-SECTION SHAPES

February 25, 1965



RECTANGULAR SECTION

CUPPED SECTION

FIGURE NO. 1-H-1

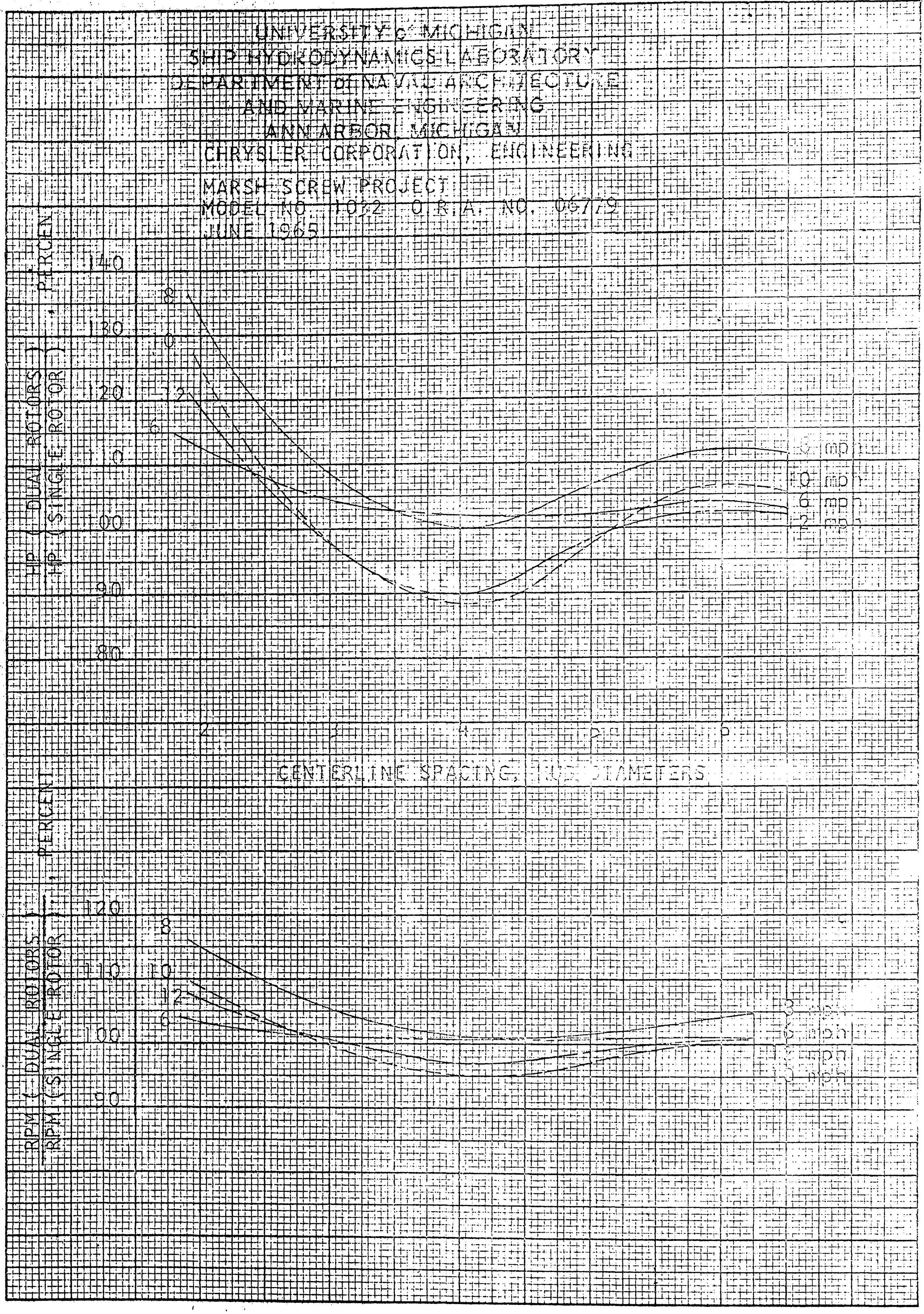
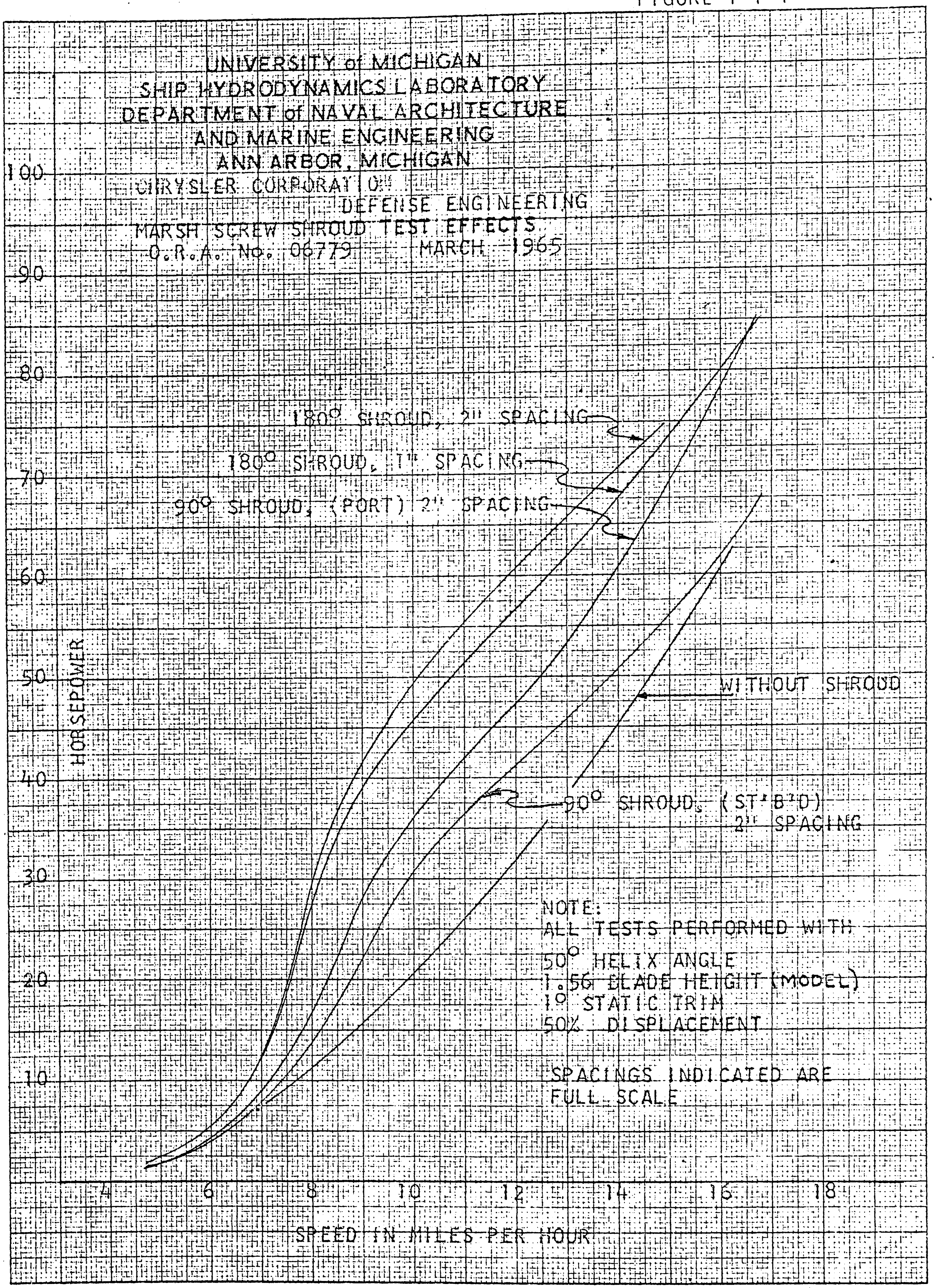


FIGURE I-I-I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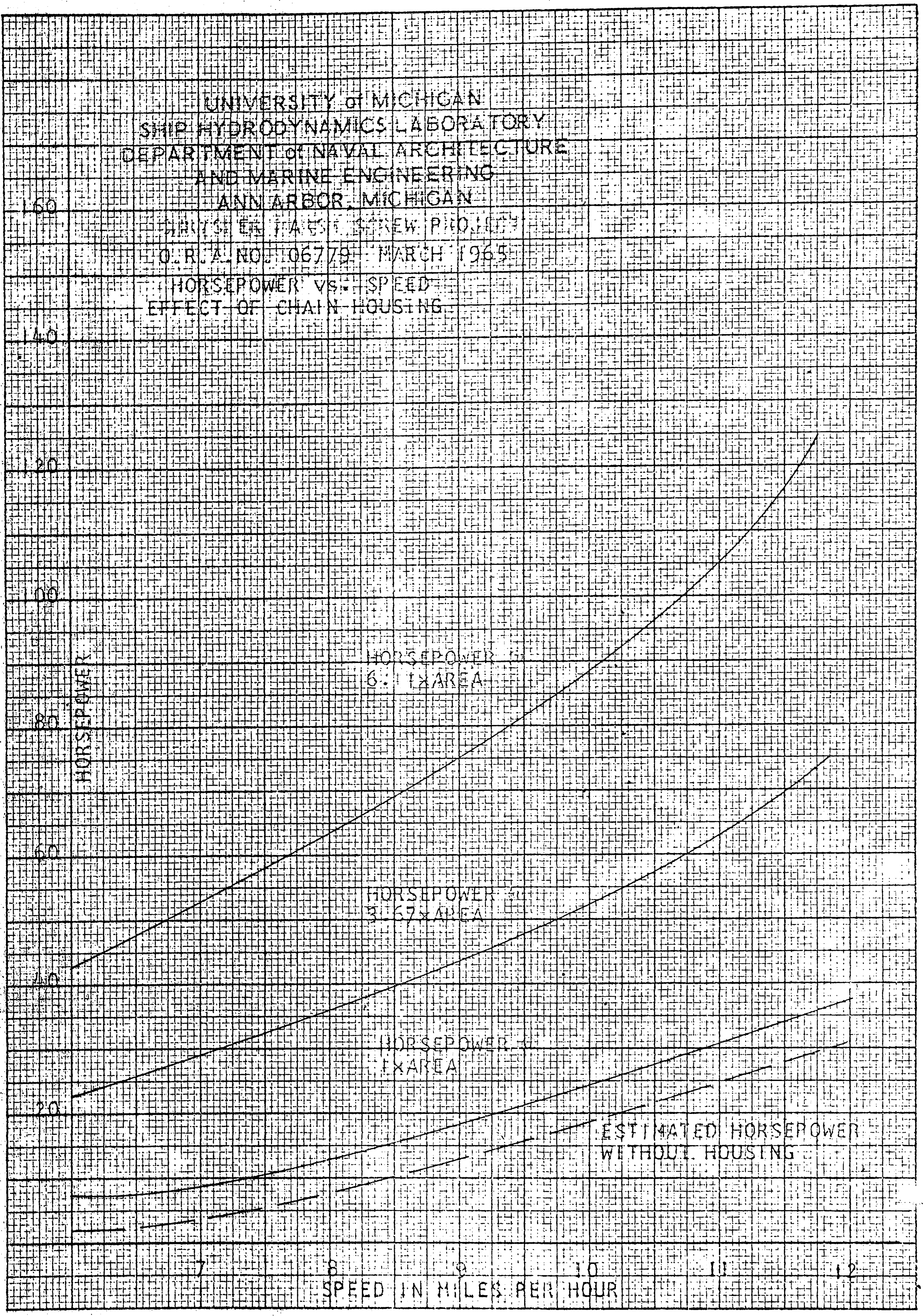
ANN ARBOR, MICHIGAN
CHRYSLER CORPORATION
DEFENSE ENGINEERING
MARSH SCREW SHROUD TEST EFFECTS
O.R.A. No. 06779 MARCH 1965
180° SHROUD, 2" SPACING
180° SHROUD, 1" SPACING
90° SHROUD, (PORT) 2" SPACING
WITHOUT SHROUD
90° SHROUD, (ST'B'D)
2" SPACING
NOTE:
ALL TESTS PERFORMED WITH
50° HELIX ANGLE
1.56 BLADE HEIGHT (MODEL)
1° STATIC TRIM
50% DISPLACEMENT
SPACINGS INDICATED ARE
FULL SCALE
HORSEPOWER
100
90
80
70
60
50
40
30
20
10
4
6
8
10
12
14
16
18
SPEED IN MILES PER HOUR

FIGURE I-J-1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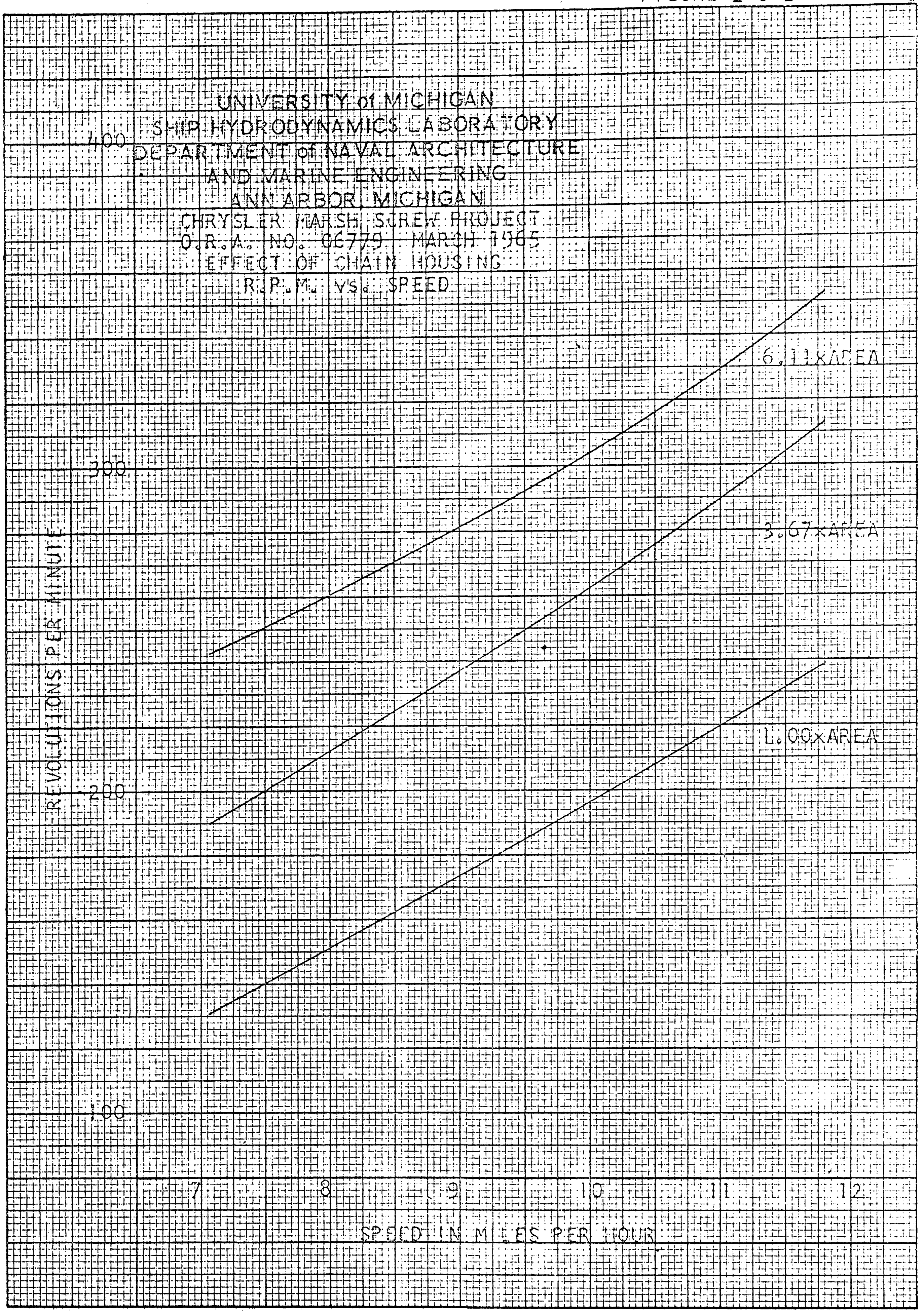
O.R.A. NO. 06779 MARCH 1965
HORSEPOWER VS. SPEED
EFFECT OF CHAIN HOUSING
160
140
120
100
80
60
40
20
HORSEPOWER
6.11xAREA
3.67xAREA
1xAREA
ESTIMATED HORSEPOWER
WITHOUT HOUSING
7
8
9
10
11
12
SPEED IN MILES PER HOUR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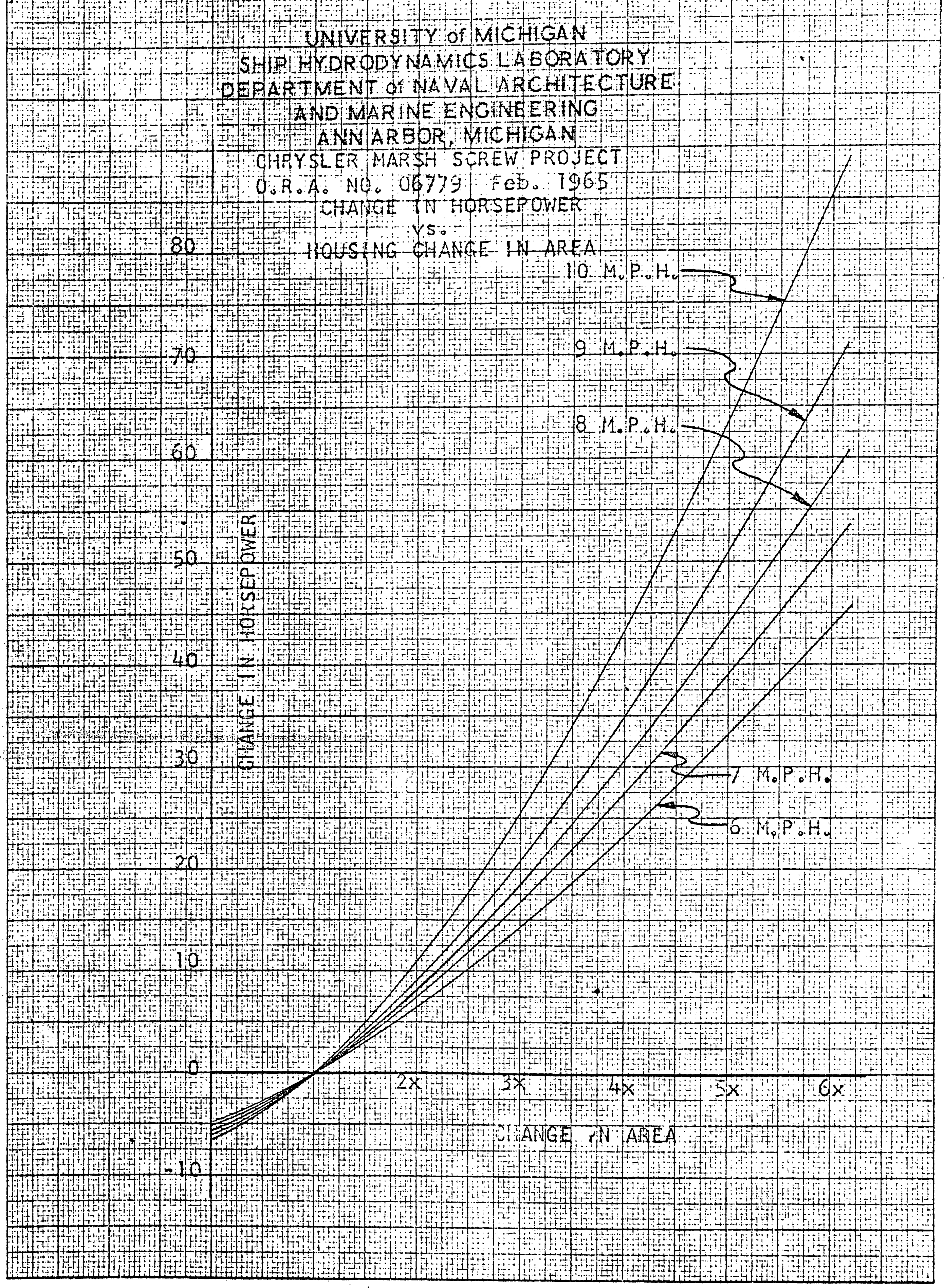
ANN ARBOR, MICHIGAN
CHRYSLER MARSH SCREW PROJECT
O.R.A. NO. 06779 MARCH 1965
EFFECT OF CHAIN HOUSING
R.P.M. VS. SPEED
400
300
200
100
REVOLUTIONS PER MINUTE
6.11xAREA
3.67xAREA
1.00xAREA
7
8
9
10
11
12
SPEED IN MILES PER HOUR

FIGURE I-J-3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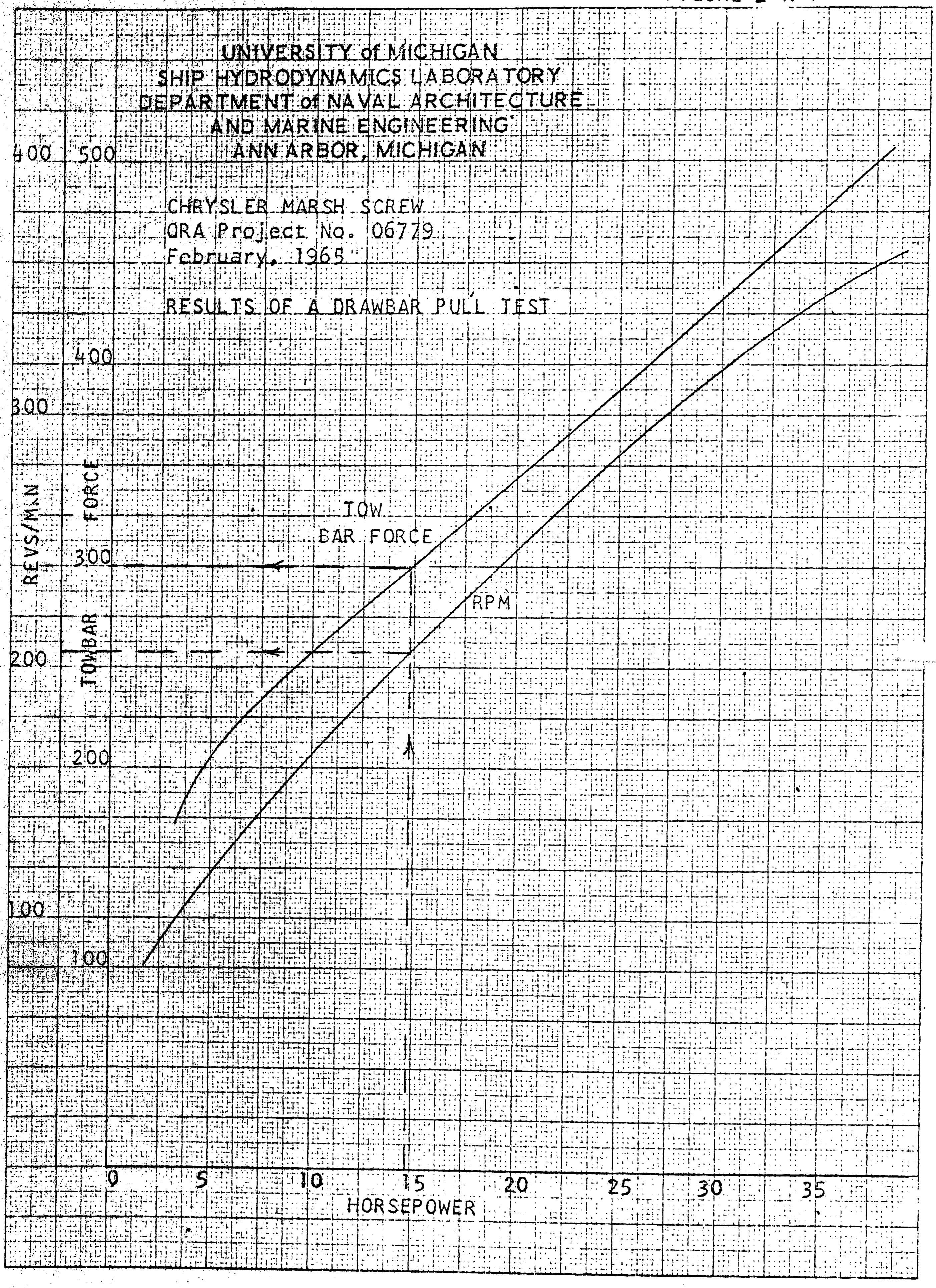
ANN ARBOR, MICHIGAN
CHRYSLER MARSH SCREW PROJECT
O.R.A. NO. 06779 Feb. 1965
CHANGE IN HORSEPOWER
vs.
HOUSING CHANGE IN AREA
10 M.P.H.
9 M.P.H.
8 M.P.H.
7 M.P.H.
6 M.P.H.
CHANGE IN HORSEPOWER
80
70
60
50
40
30
20
10
0
-10
2x
3x
4x
5x
6x
CHANGE IN AREA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ANN ARBOR, MICHIGAN
CHRYSLER MARSH SCREW
ORA Project No. 06779
February, 1965
RESULTS OF A DRAWBAR PULL TEST
400
500
400
300
REVS/MIN
TOWBAR FORCE
TOW
BAR FORCE
300
RPM
200
200
100
100
0
5
10
15
20
25
30
35
HORSEPOWER

Section II: Phase II, Systematically Related Rotors

A rather thorough investigation of torque and rotary speed variations as a function of geometry was undertaken on a group of models all systematically related through blade height and helix angle. Viewed as a matrix of heights and angles the rotors indicated in the following table were tested.

Blade Height, inches	Helix Angle, degrees			
	30	40	50	60
0.920	x		x	
1.250		x	x	
1.560	x	x	x	x
1.875		x	x	

Blade heights are given as model dimensions. Other model characteristics were:

Length	48.0 inches
Hub Diameter	8.0 inches
Total Hub Volume	1.3247 cu. ft.

Each rotor was tested in the 25, 50 and 75 percent of total hub volume displacement conditions. Buoyancy of the blades was neglected. Generally, all rotors were tested with one degree bow up initial trim but exceptions were those rotors where the initial trim was varied, as reported in Section I-A. In these cases model torque and rpm at one degree initial trim were determined through the use of cross plots against trim. The highest speeds possible were run according to that which the test apparatus would allow.

With all data in hand, a convenient non-dimensional form for presentation of data was sought. The IBM 7090 computer was used to compute several different speed of advance, rotary advance, and torque coefficients, all of which conformed to Froude similitude as well as being non-dimensional. From the hundreds of sets of numbers trial plots of torque coefficient versus speed-length ratio, etc., under various definitions of these parameters, were prepared in an effort to seek a set of coefficients which would be most amenable to cross-plotting in order to determine their values throughout the entire matrix for configurations where no models were tested. Upon inspection of the computer output many sets of numbers were readily discarded and the following definitions were finally selected as fulfilling the requirements most satisfactorily, but without making powering calculations too unwieldy:

Speed-length ratio = $V/\nabla^{1/6}$

Advance coefficient = $J = \dfrac{V}{n\nabla^{1/3}}$

Torque coefficient = $K_q = \dfrac{Q}{\rho V^2 \nabla}$

where

V = advance speed, ft/sec.

∇ = initial immersed hub volume, cu. ft.

n = revolutions per second

Q = torque, ft.-lbs.

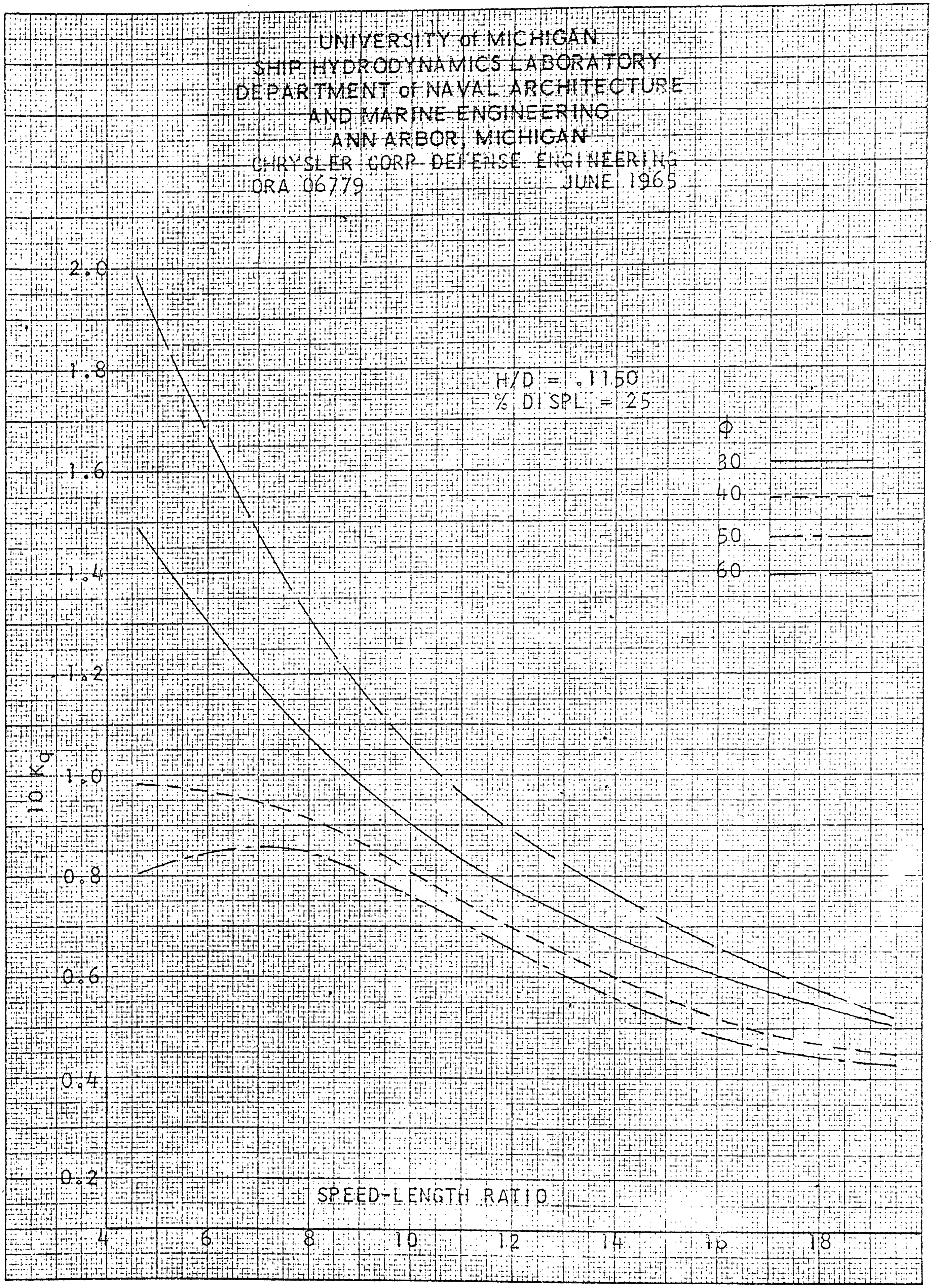
ρ = density of water, lb.-sec.^{2}/ft.^{4}
(1.936 for fresh water at 70°F and
1.991 for sea water at 59°F)

When all cross-plotting had been accomplished final figures were prepared, Figs. II-1 through II-12 for K_q versus speed-length ratio and II-13 through II-24 for J versus speed-length ratio.

Some observations from the cross-fairing can be made. While none of the original data had to be violated to evolve the curves presented, except in the most minor way, it would have been useful to have had test data for one or two configurations not tested, particularly at the highest helix angle. Secondly, the use of coefficient curves enabled the data analysis and extrapolation to be done more accurately owing to the systematic and gradual transition from one curve to another, more so than with raw data curves. For this reason there are J values plotted at higher speed-length ratios than at which torque coefficients are presented for some configurations. This resulted from the relative ease in cross-fairing the rpm parameter. However, where torque coefficients are unavailable the rpm's which might be computed should be considered as widely extrapolated. It may be possible, for instance, to find a rotor configuration which would be incapable of supporting the weight of the power plant needed to drive it at the given speed in these extreme cases.

Finally, Figs. II-25 and II-26 show some typical running sinkage and trim curves.

FIGURE 11-1



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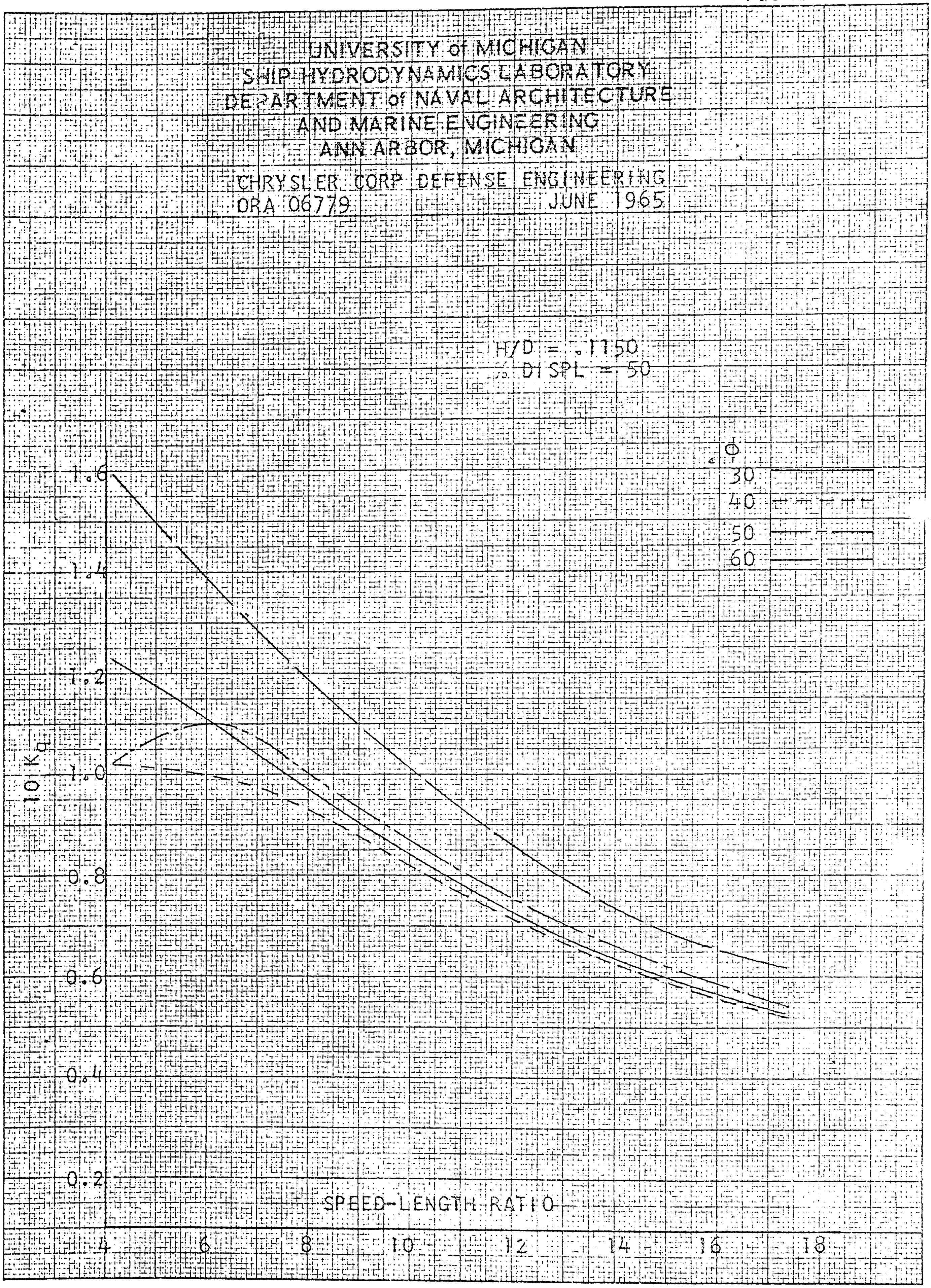
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .1150
% DISPL = 25
Φ
30
40
50
60
10 Kq
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18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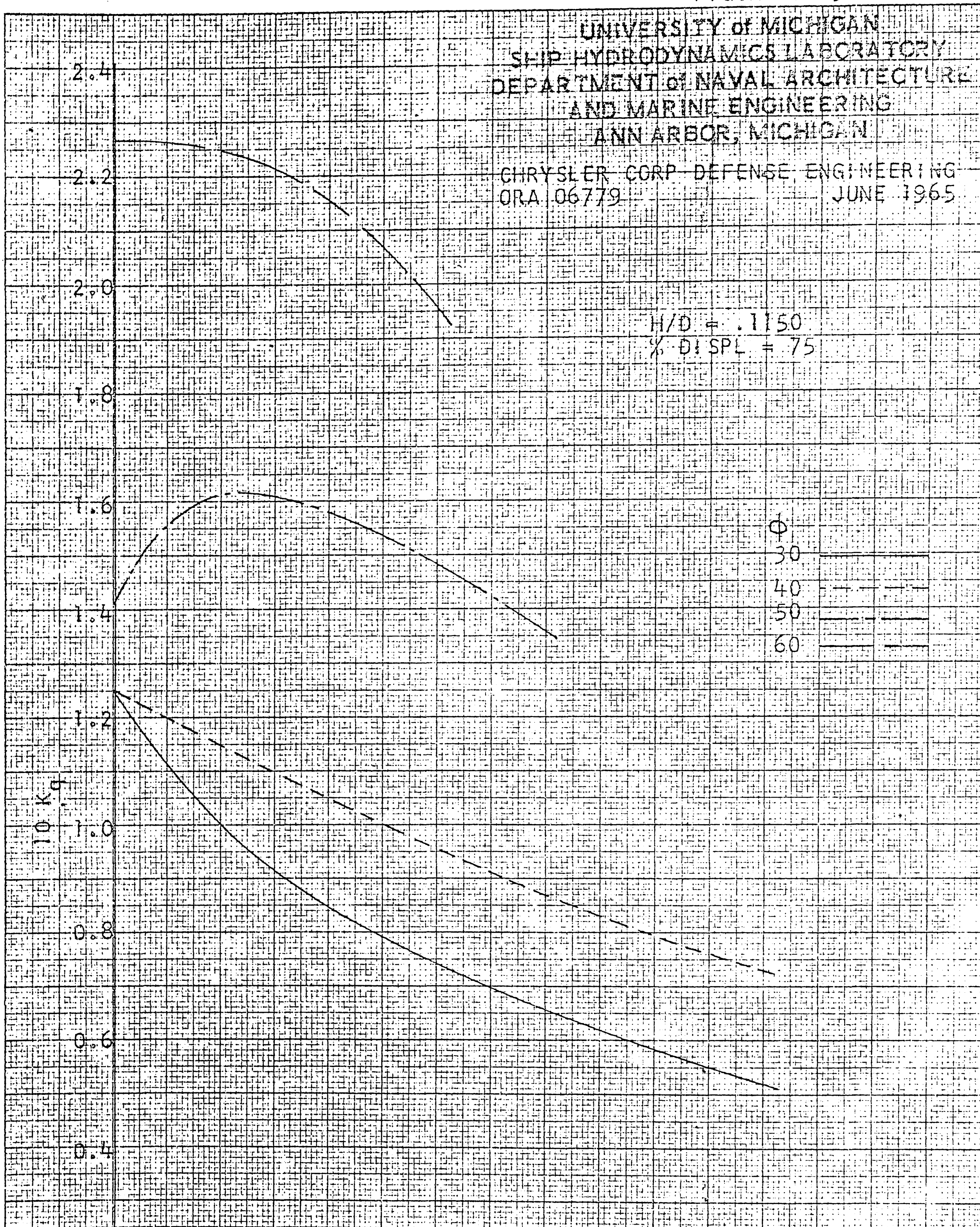
ORA 06779
JUNE 1965
H/D = .1150
% DISPL = 50
Φ
30
40
50
60
10 K_q
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

FIGURE 11-3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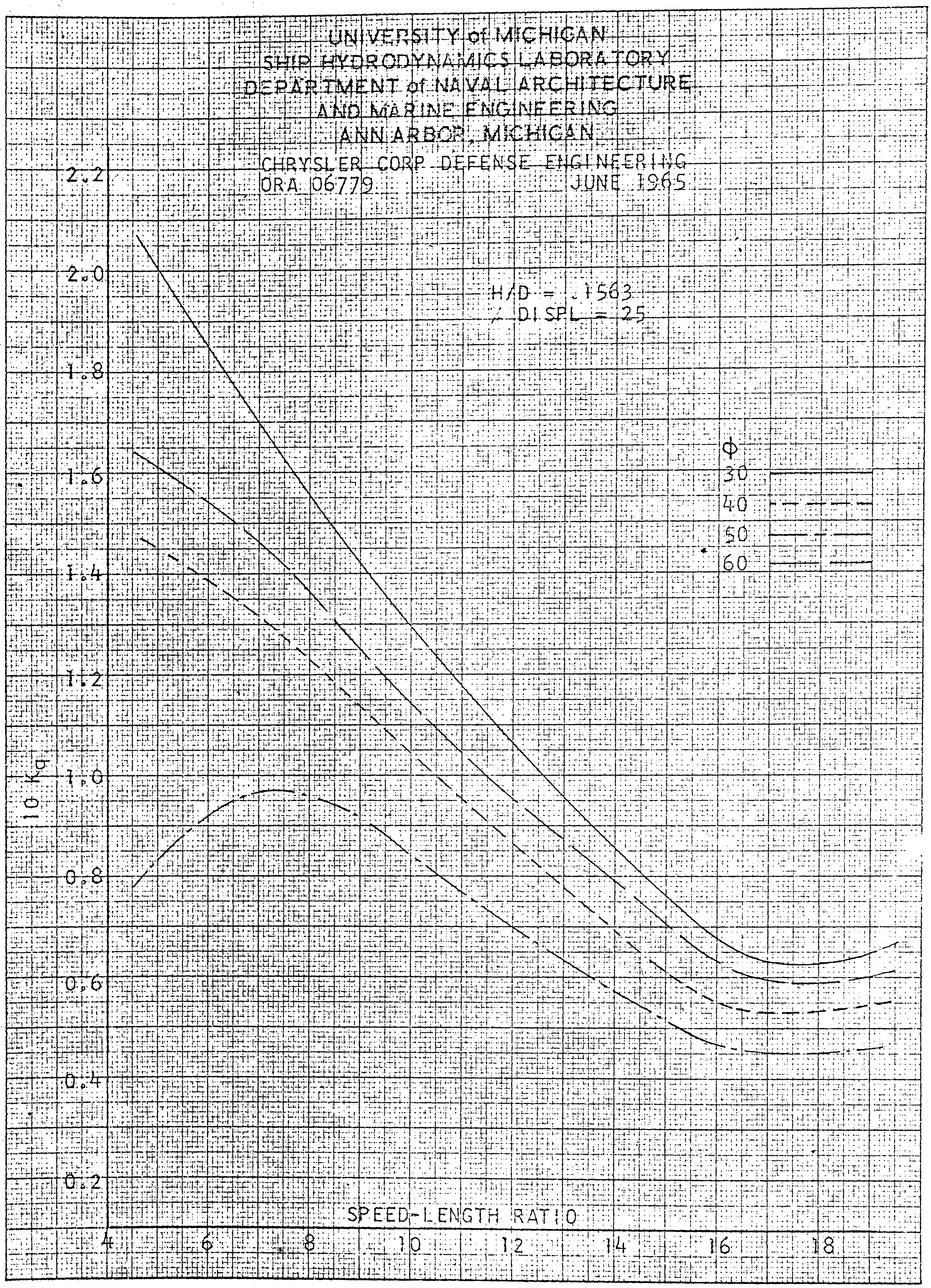
ORA 06779
JUNE 1965
H/D = .1150
% DISPL = 75
Φ
30
40
50
60
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
10 Kq
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

FIGURE 11-4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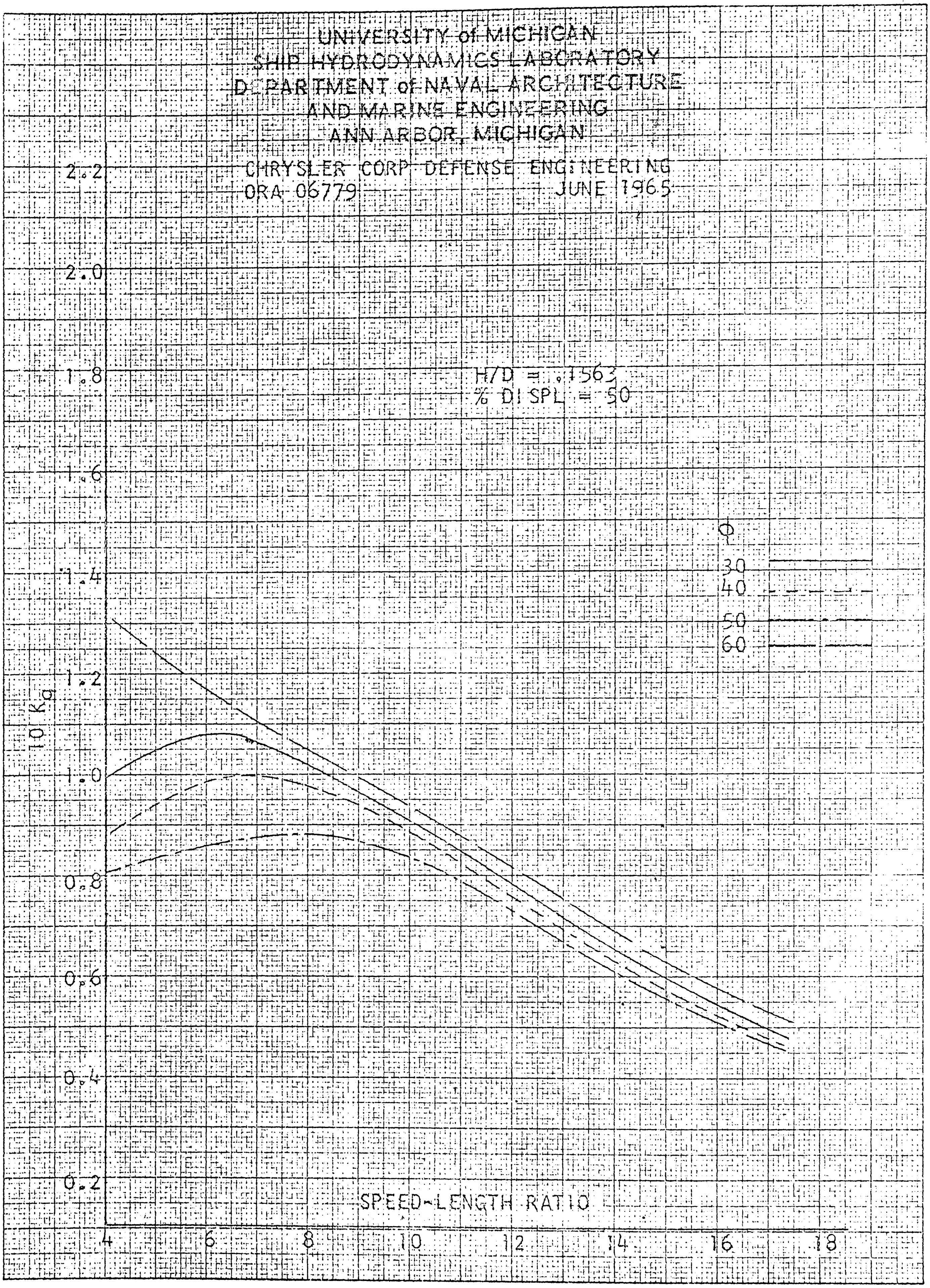
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .1563
% DISPL = 25
Φ
30
40
50
60
10 Kq
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

FIGURE 11-5



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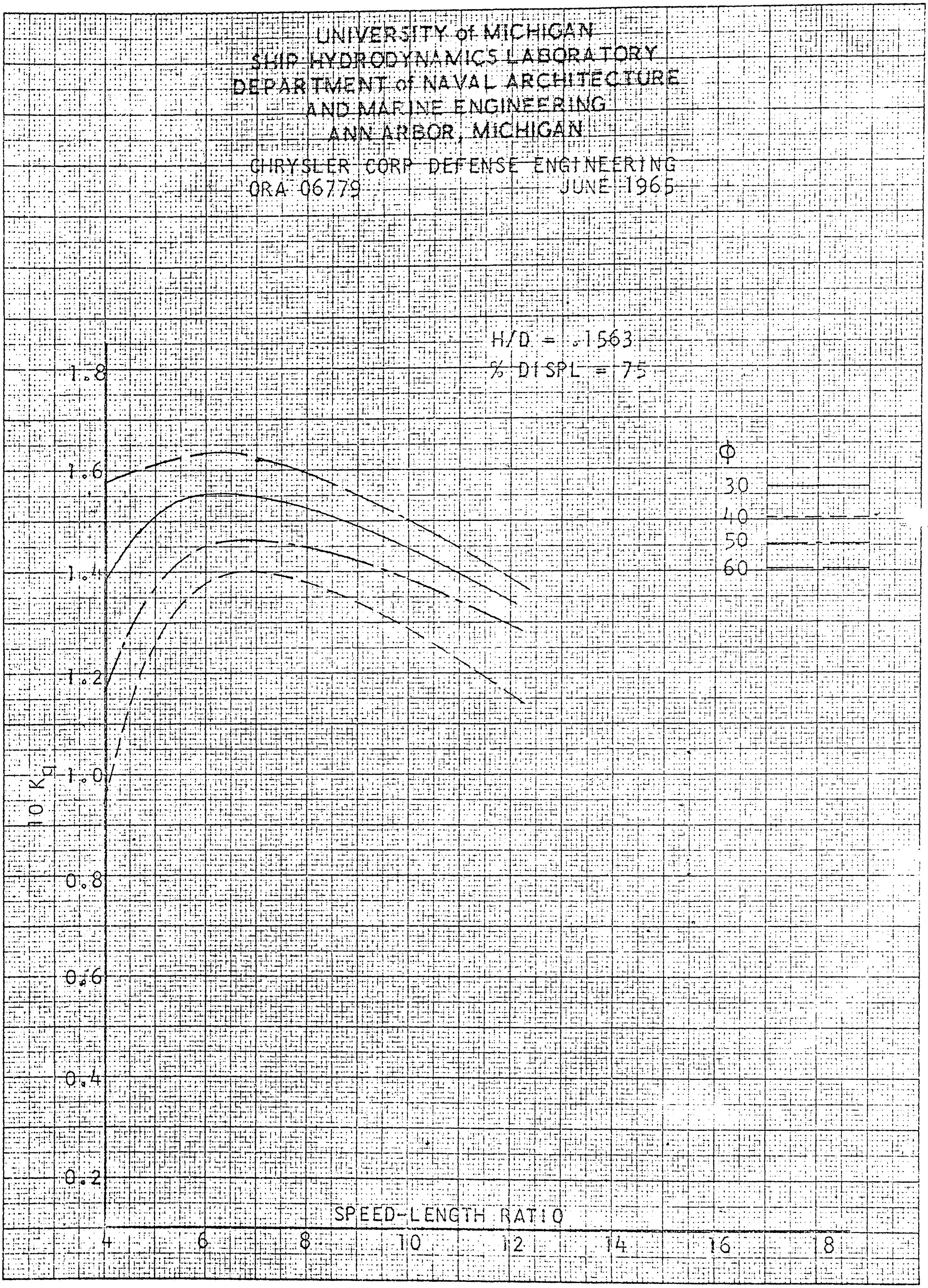
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .1563
% DISPL = 50
φ
30
40
50
60
10 Ka
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18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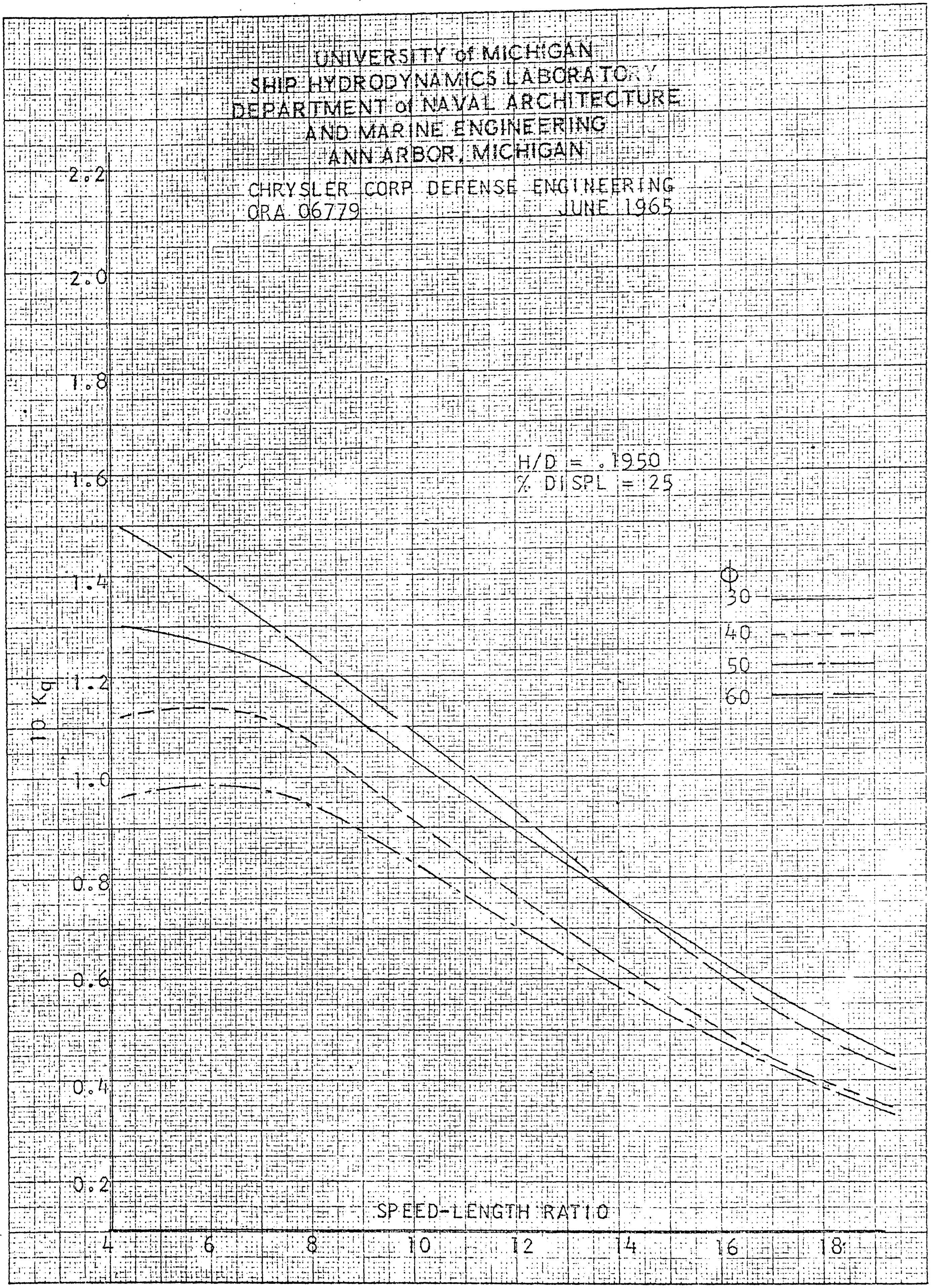
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .1563
% DISPL = 75
Φ
30
40
50
60
10 Kq
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18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CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .1950
% DISPL = 25
Φ
30
40
50
60
10 Kq
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

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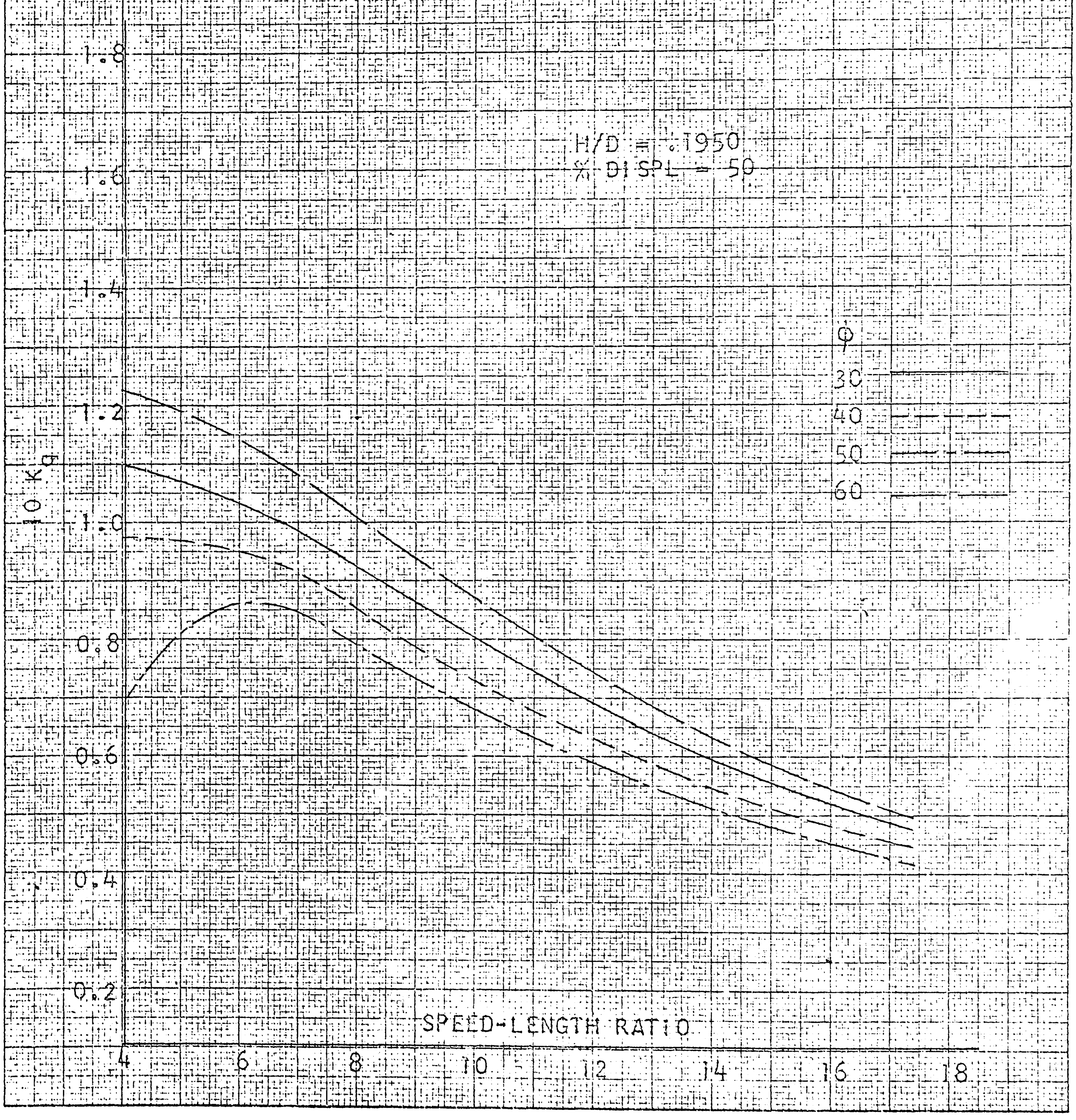
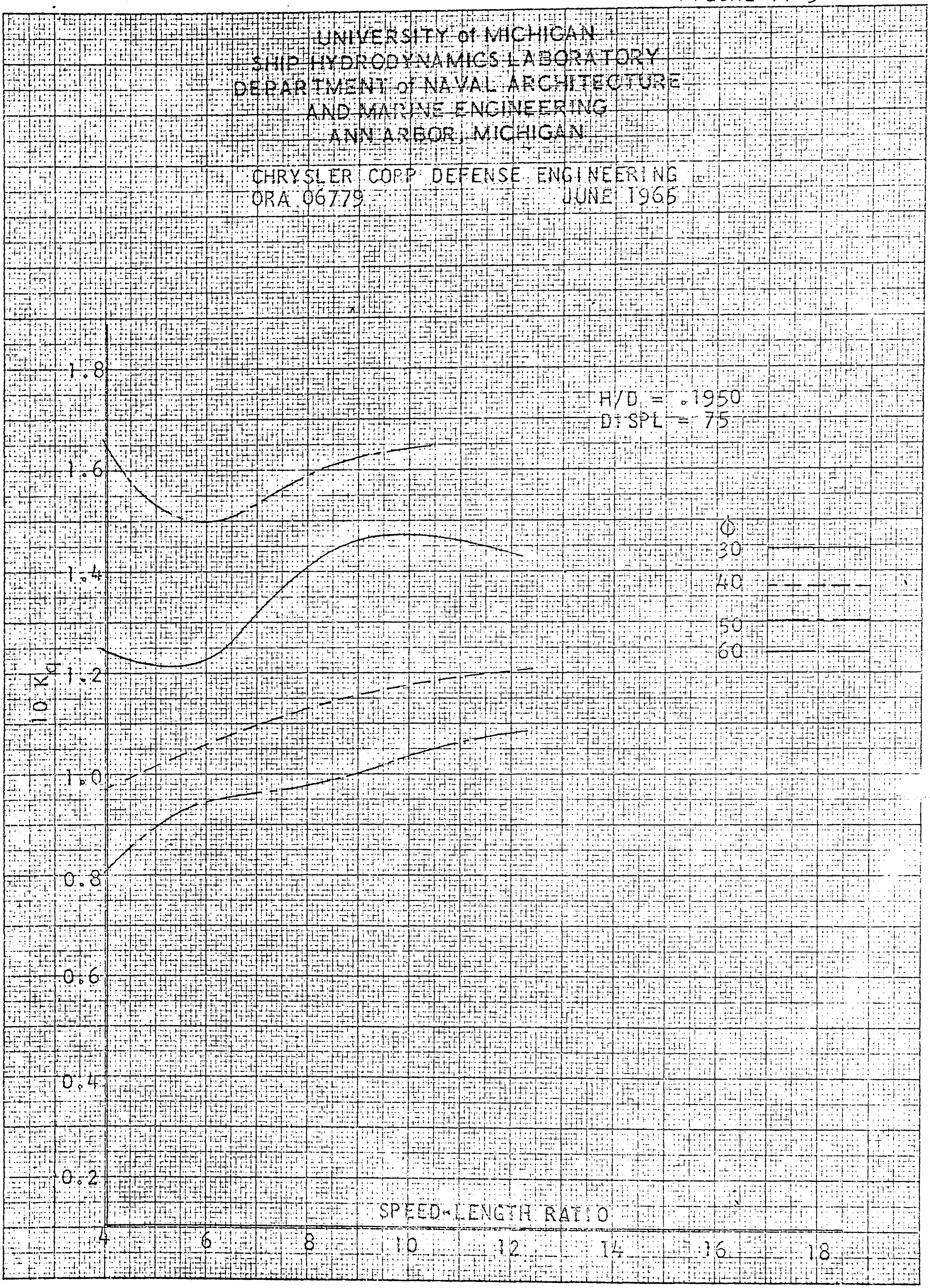


FIGURE 11-9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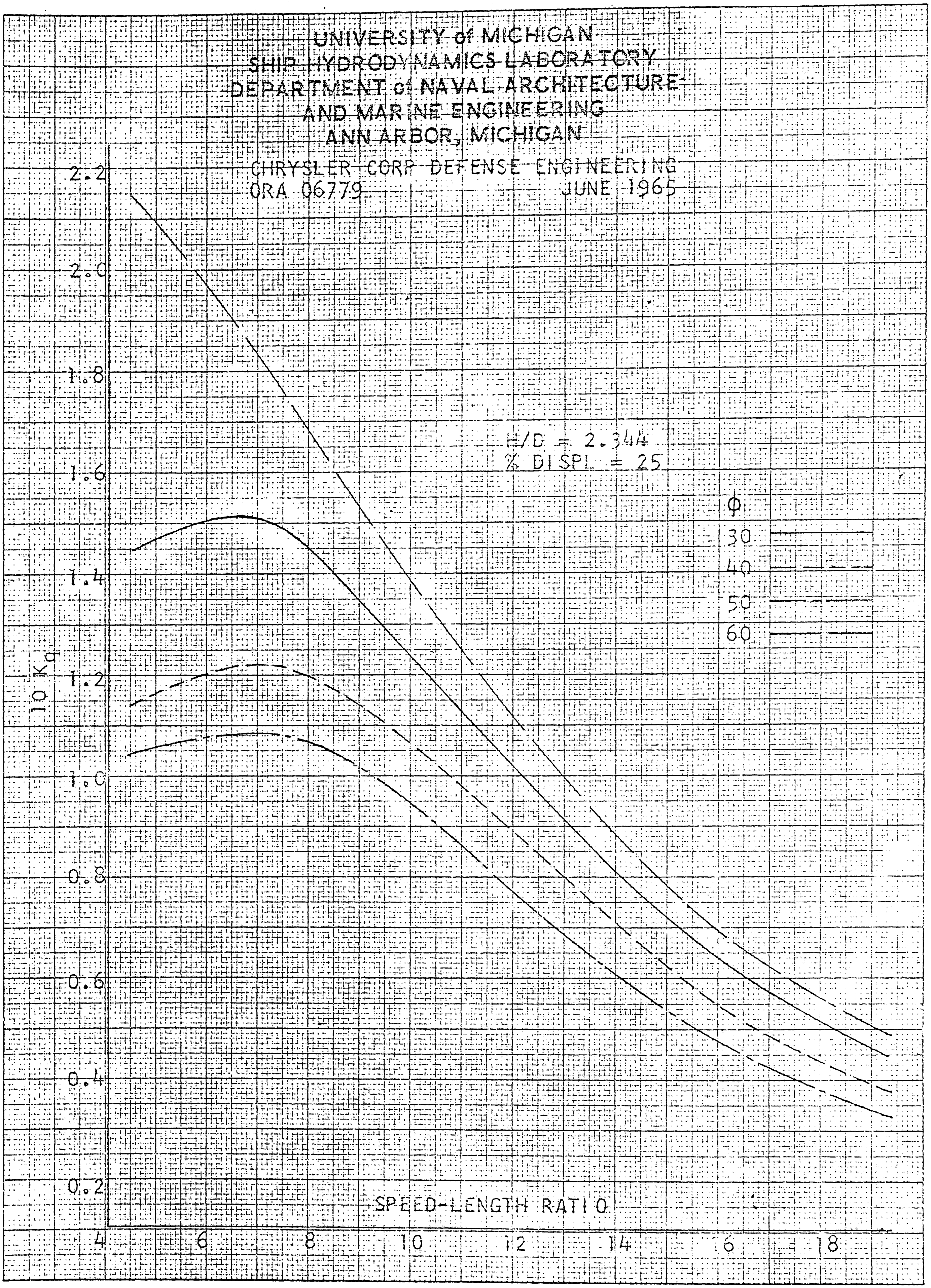
ORA 06779
JUNE 1966
H/D = .1950
DISPL = 75
Φ
30
40
50
60
10 Kq
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

FIGURE 11-10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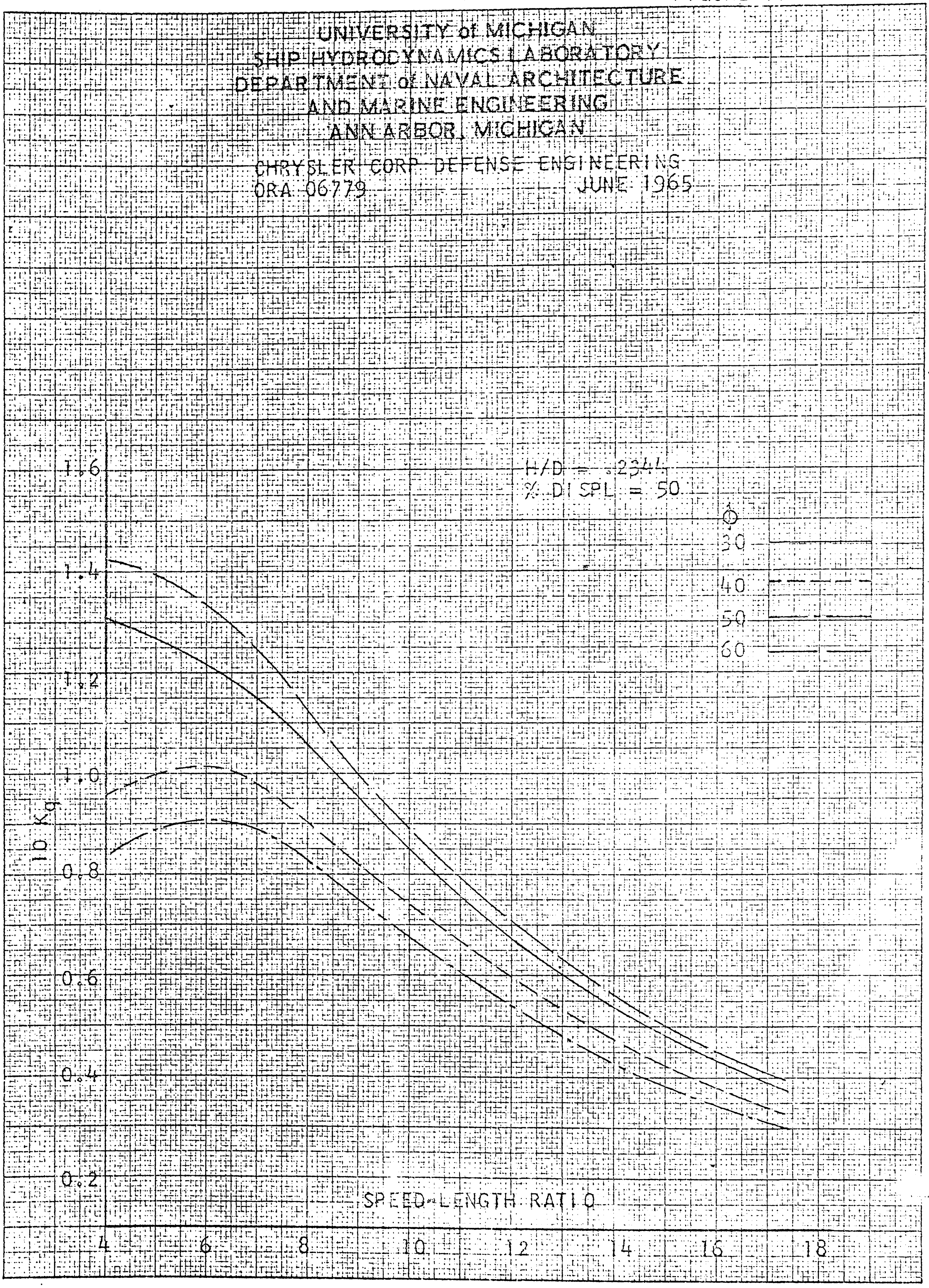
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = 2.344
% DISPL = 25
φ
30
40
50
60
10 Kq
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
4
6
8
10
12
14
16
18
SPEED-LENGTH RATIO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ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D = .2344
% DISPL = 50
θ
30
40
50
60
lb Kg
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
SPEED-LENGTH RATIO
4
6
8
10
12
14
16
18

FIGURE 11-12

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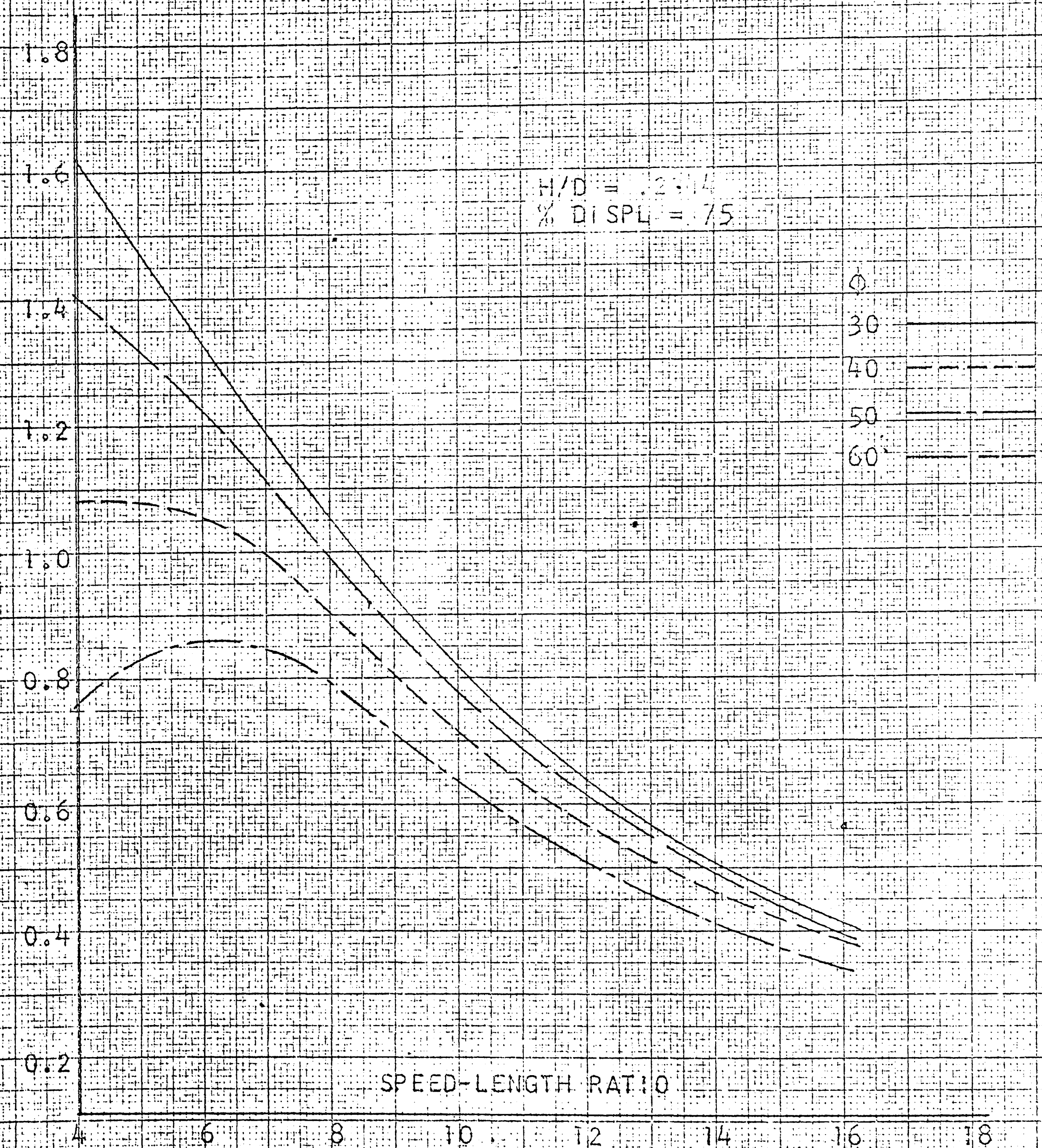
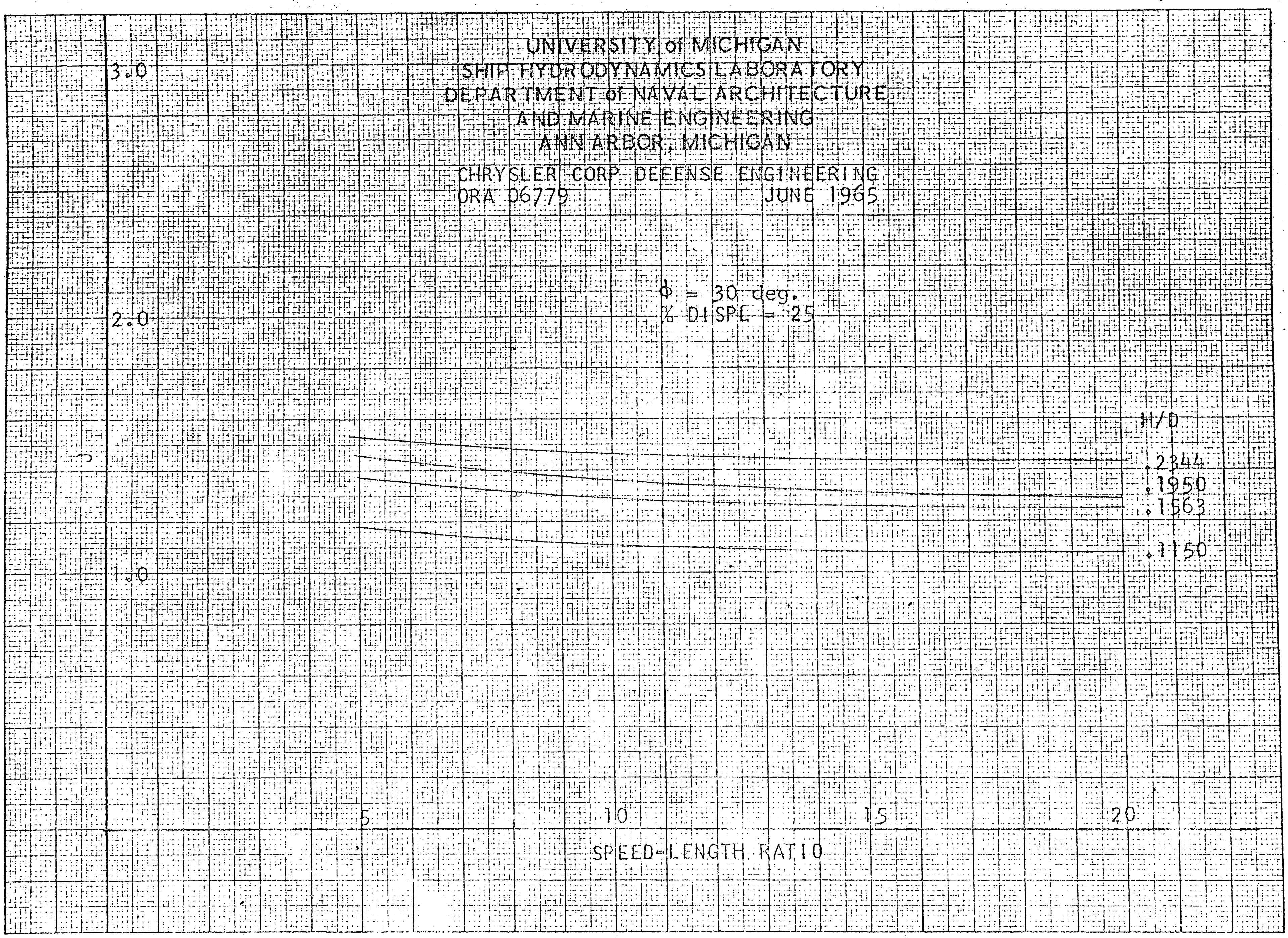
ORA 06779 JUNE 1965





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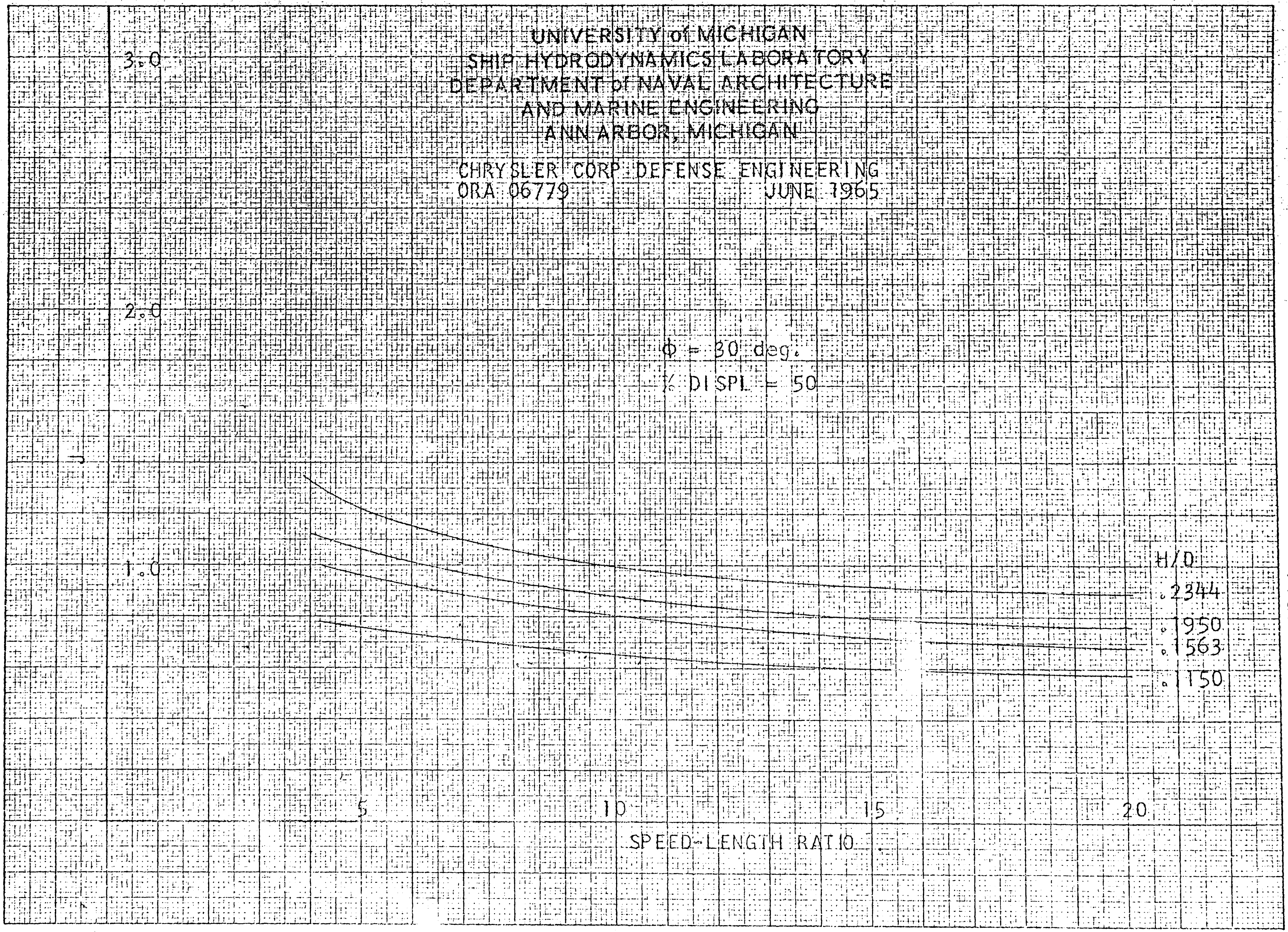
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
φ = 30 deg.
% DISPL = 25
3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-13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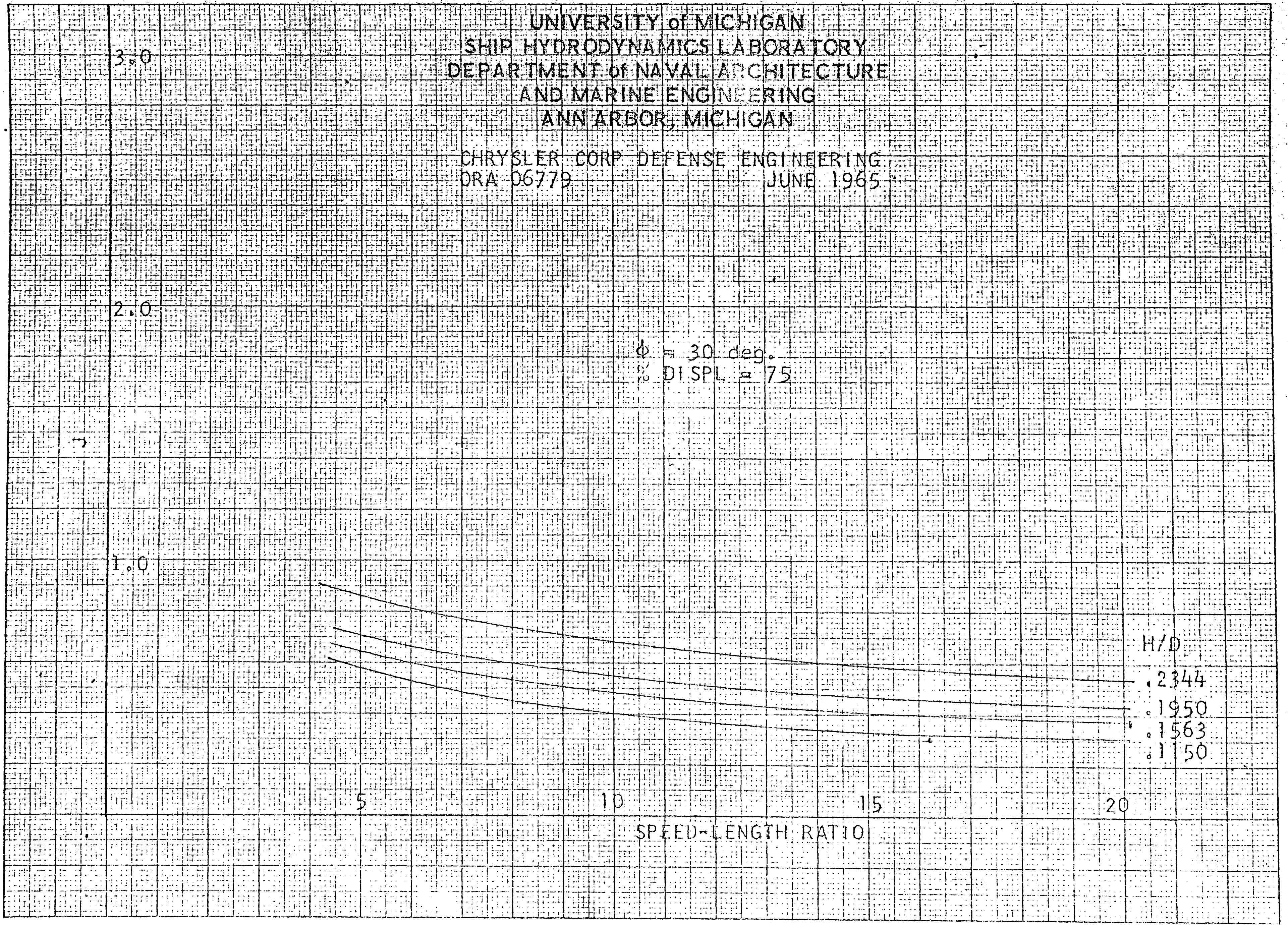
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
φ = 30 deg.
% DISPL = 50
3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-14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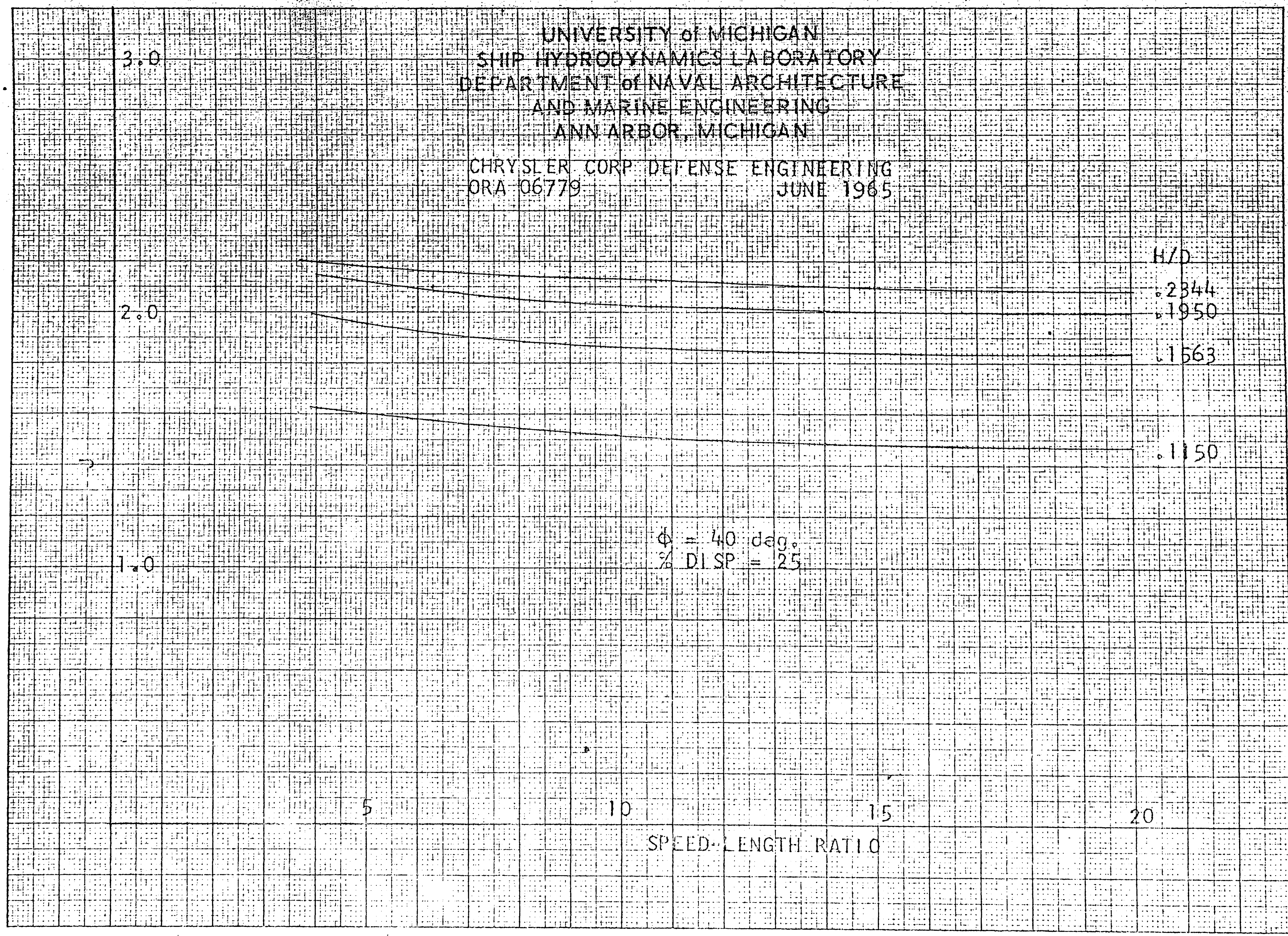
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
ϕ = 30 deg.
% DISPL = 75
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE II-15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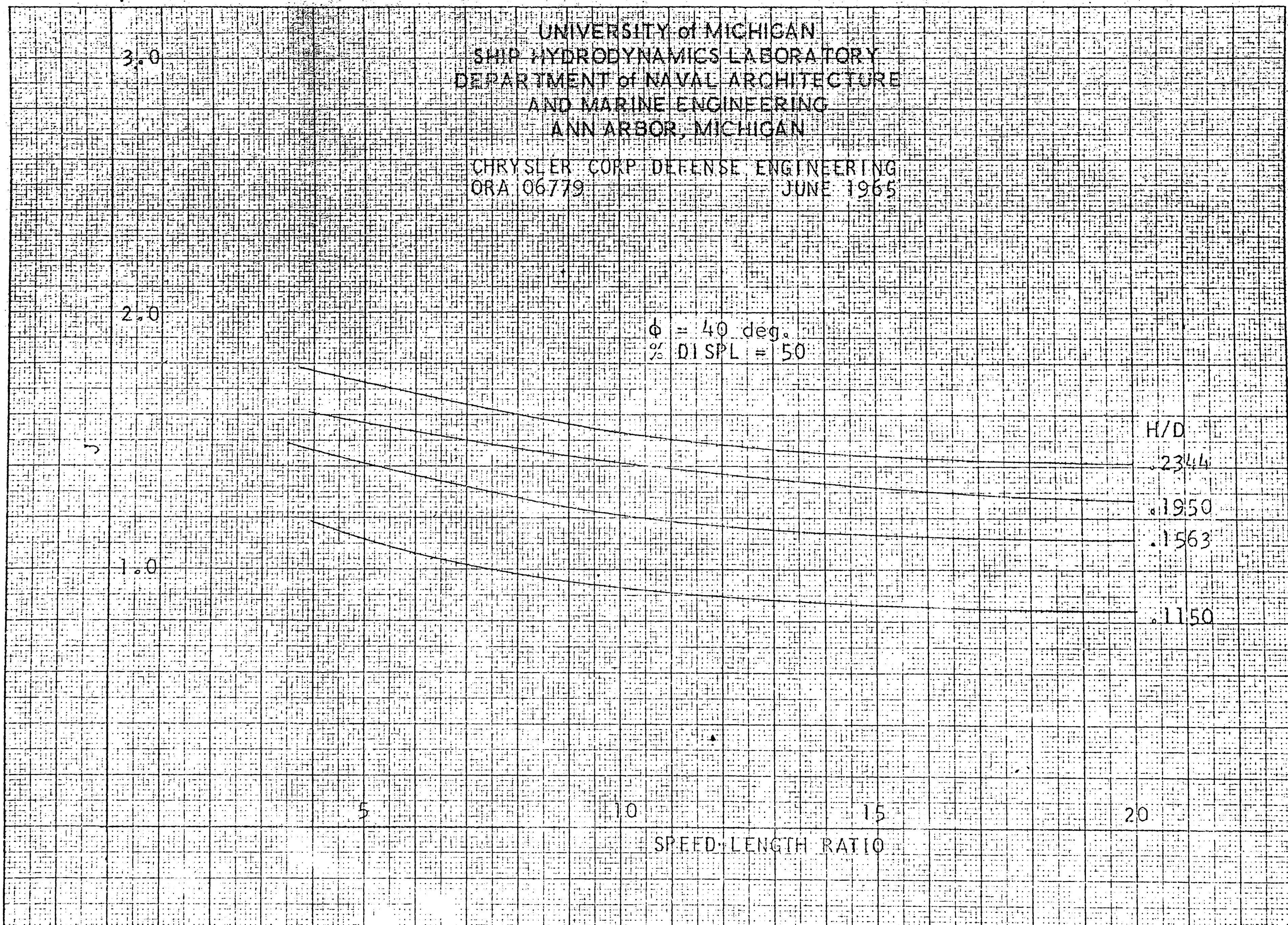
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
H/D
.2344
.1950
.1563
.1150
3.0
2.0
1.0
ϕ = 40 deg.
% DISP = 25
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-16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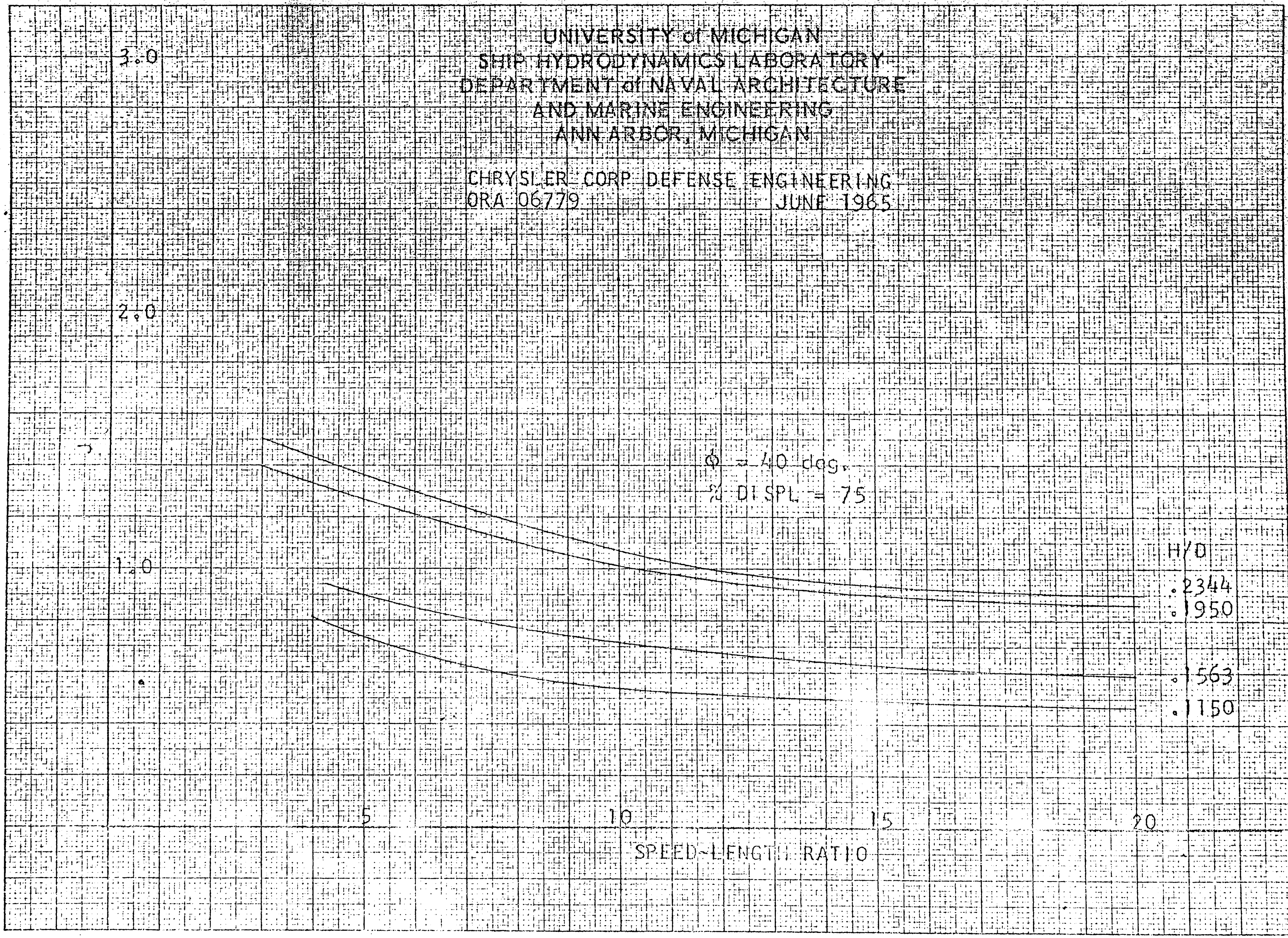
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
φ = 40 deg.
% DISPL = 50
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-17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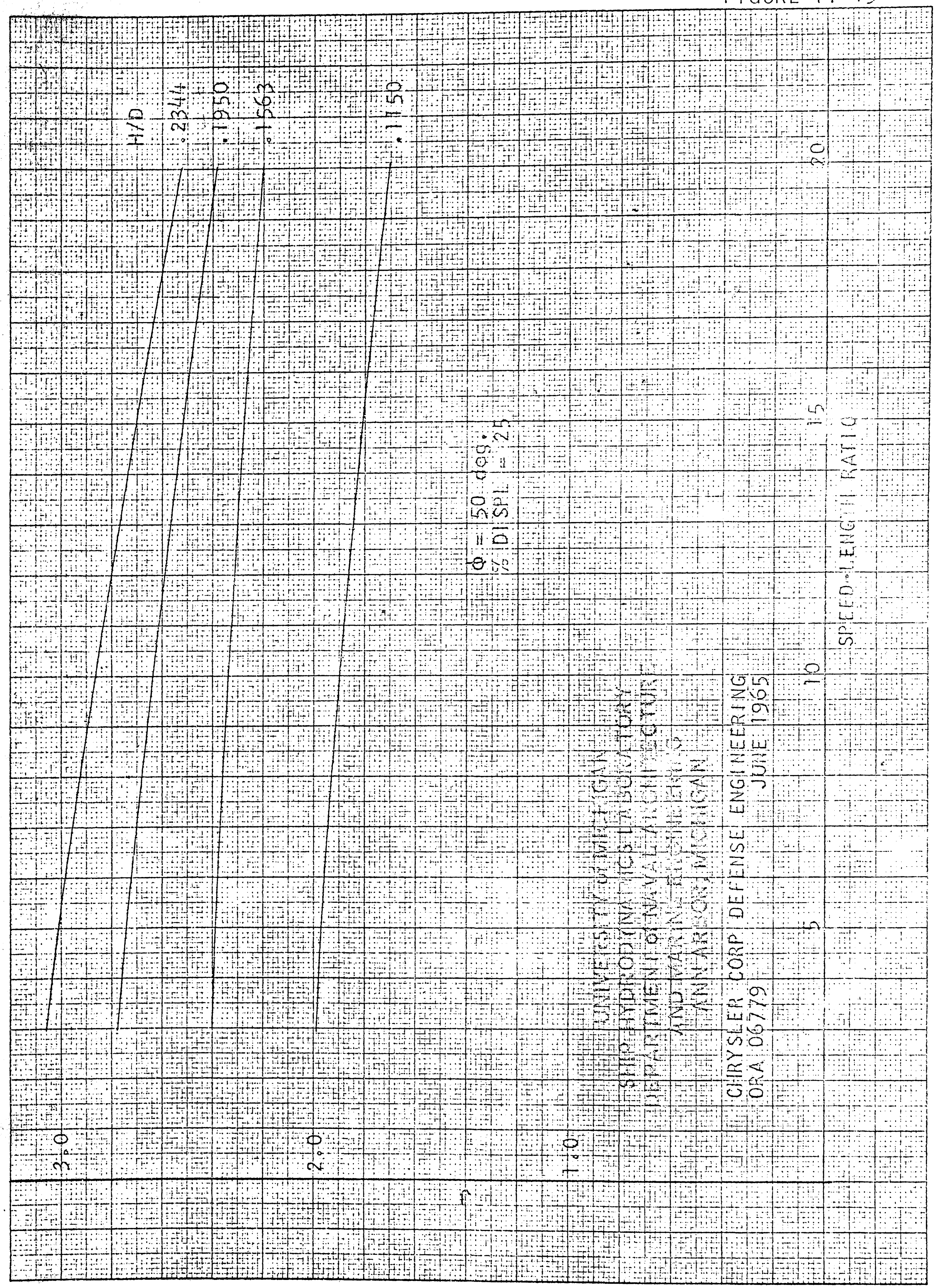
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
ϕ = 40 deg.
% DISPL = 75
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-18

FIGURE II-19



H/D
.2344
.1950
.1563
.1150
Φ = 50 deg.
% DISPL = 25
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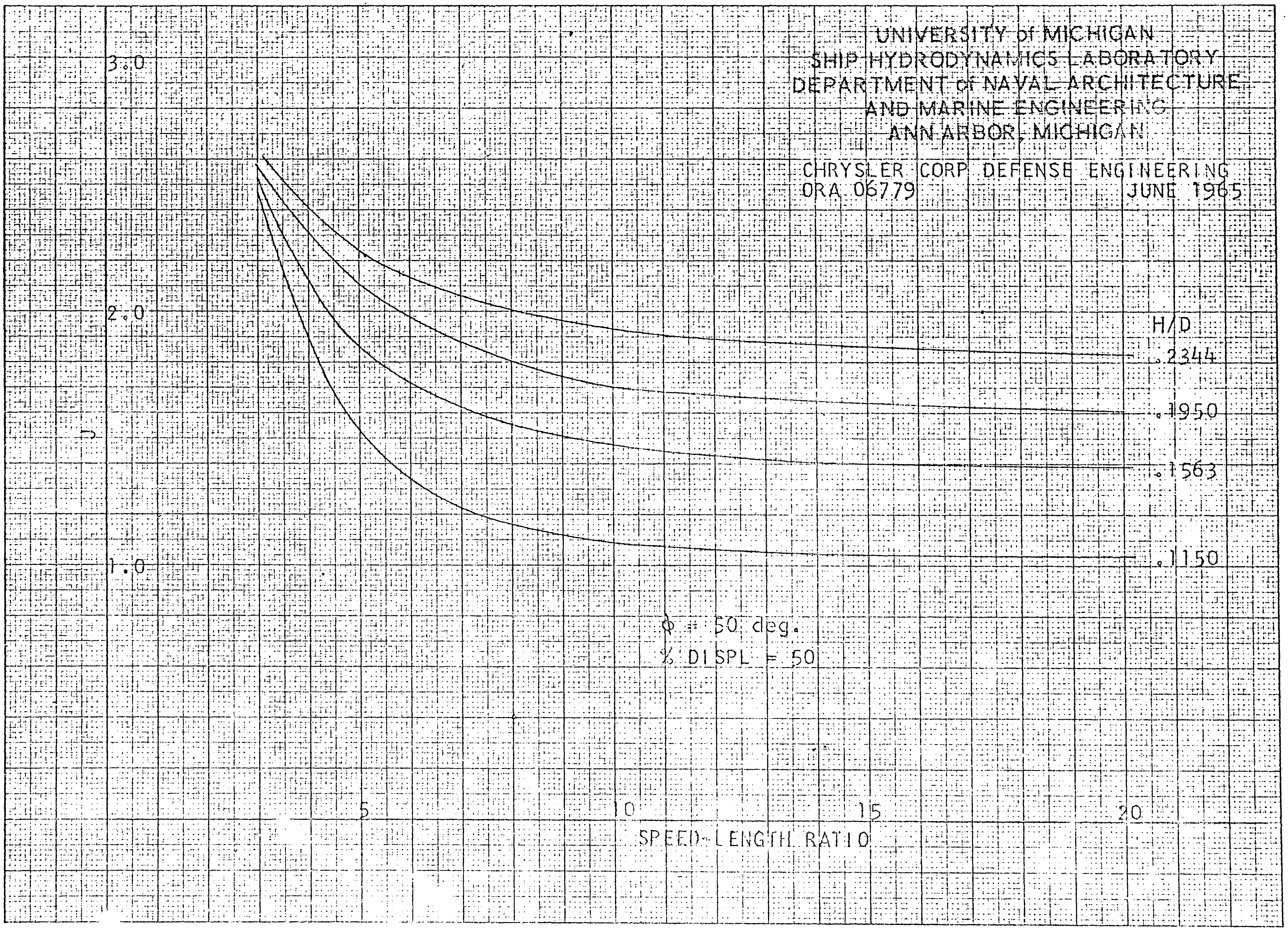
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779 JUNE 1965
3.0
2.0
1.0
5
10
15
20
SPEED-LENGTH RATIO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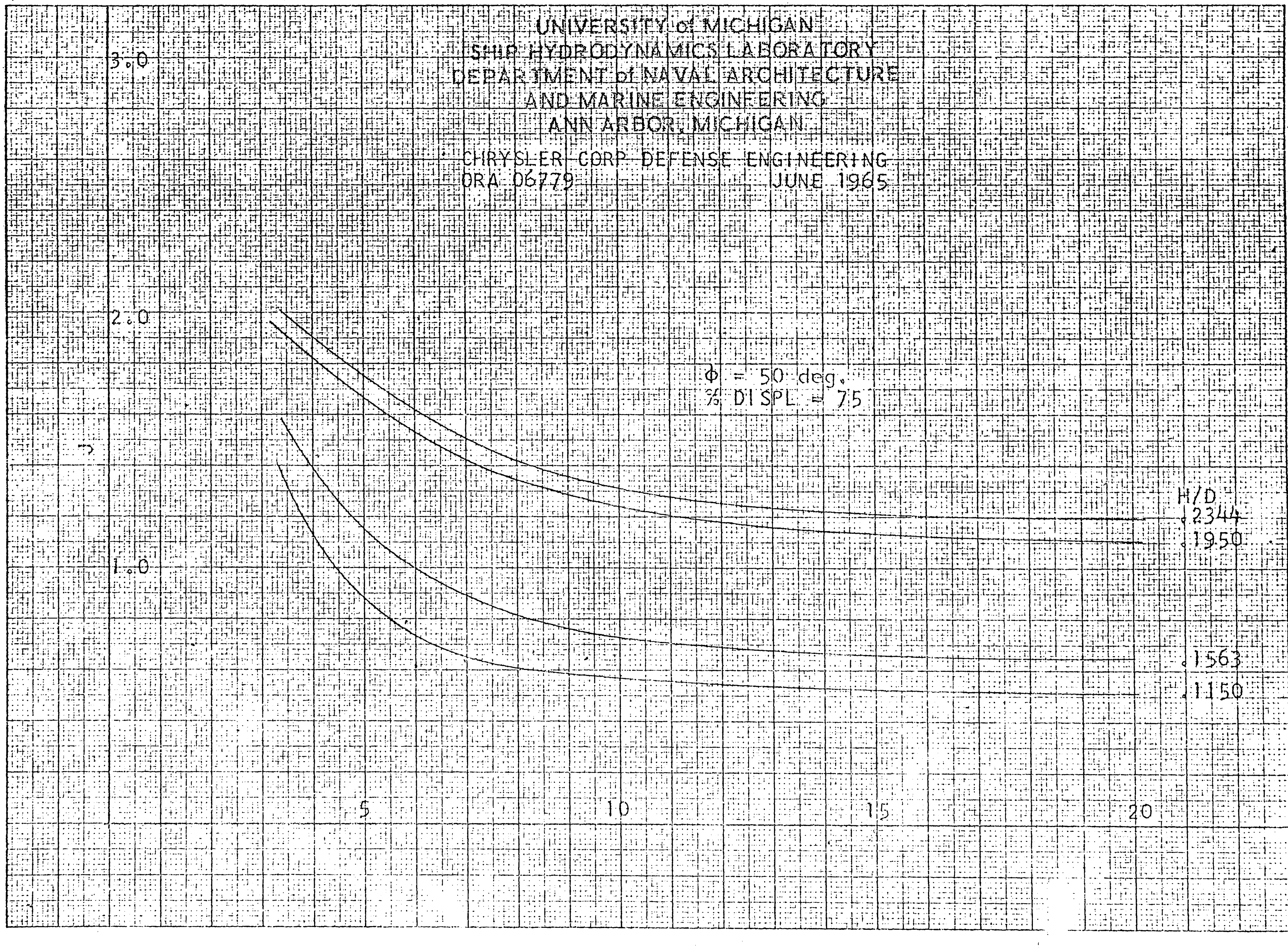
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
φ = 50 deg.
% DISPL = 50
5
10
15
20
SPEED-LENGTH RATIO

FIGURE II-20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AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP. DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
φ = 50 deg.
% DISPL = 75
H/D
.2344
.1950
.1563
.1150
5
10
15
20

FIGURE II-21

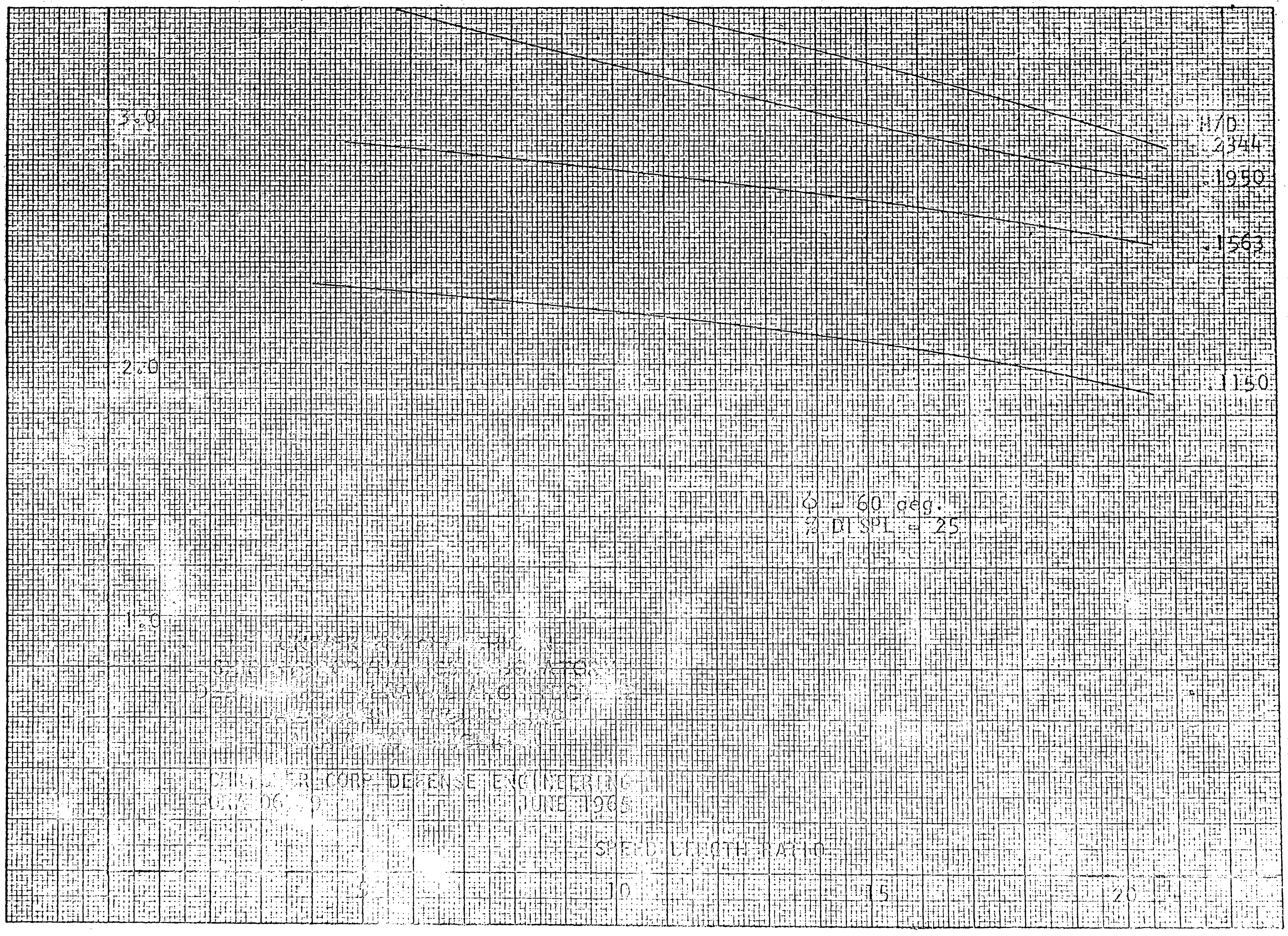
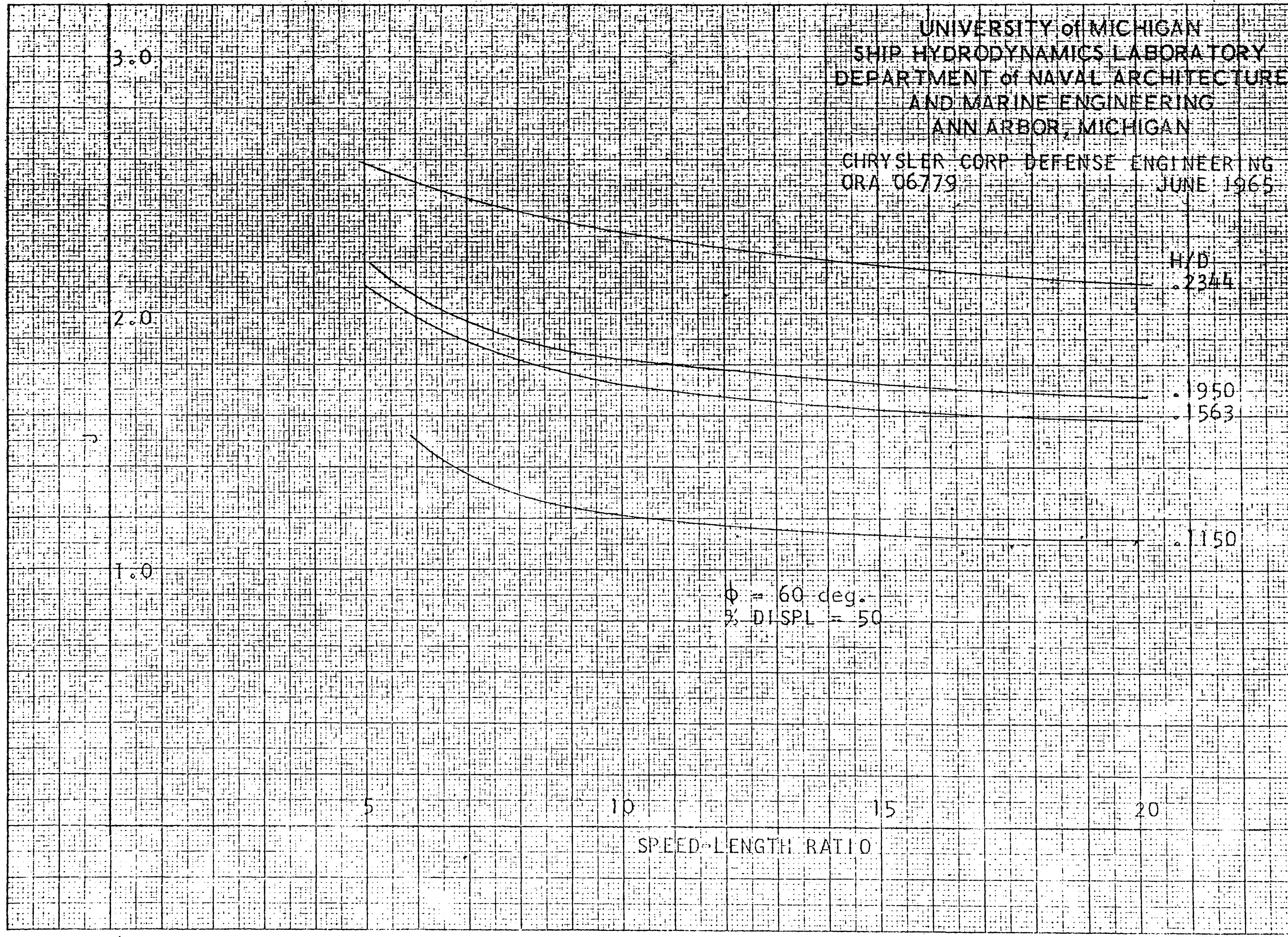

3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
Φ = 60 deg.
% DISPL = 25
SPEED LENGTH RATIO
5
10
15
20

FIGURE 11-22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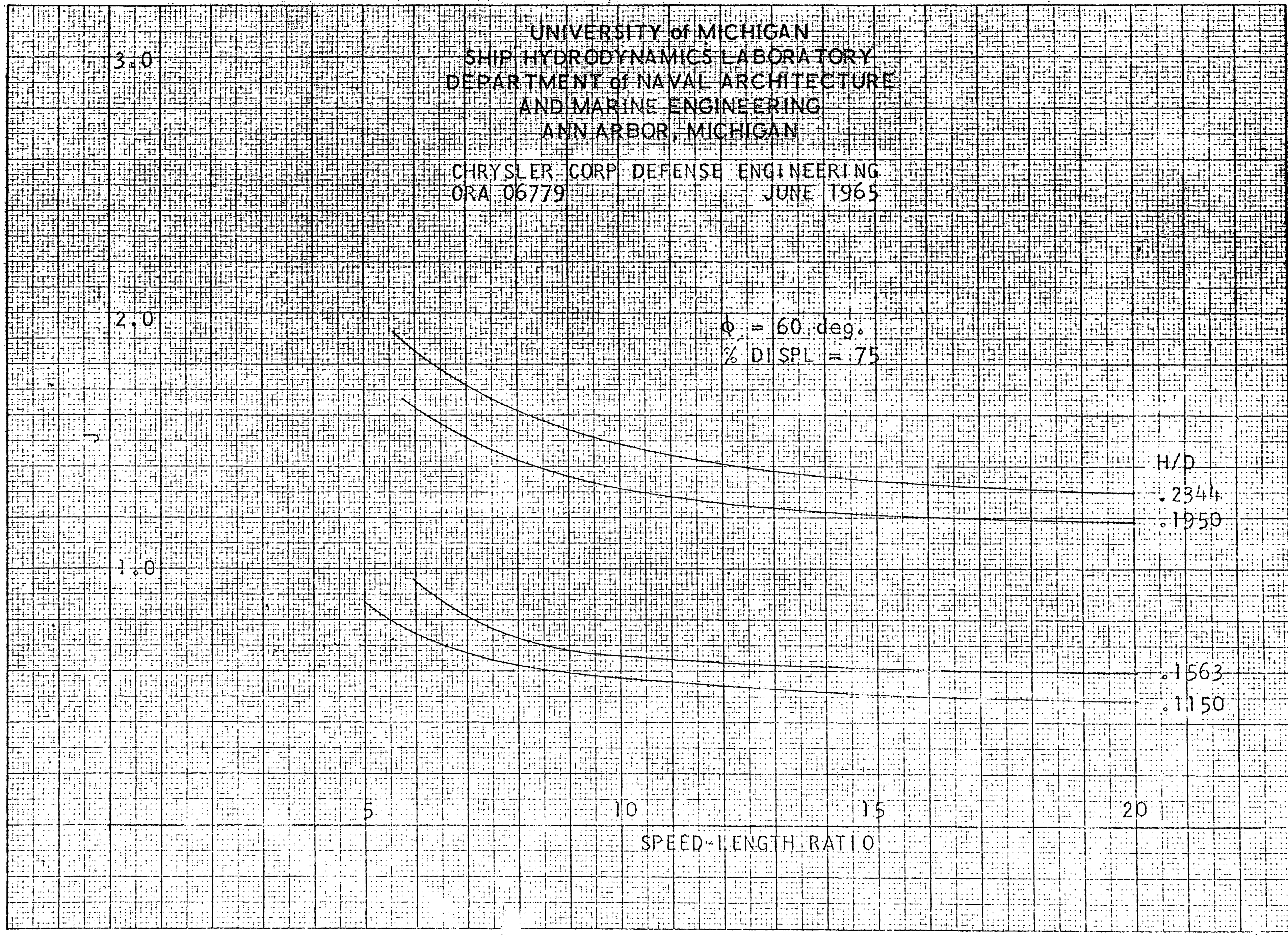
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
Φ = 60 deg.
% DISPL = 50
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-23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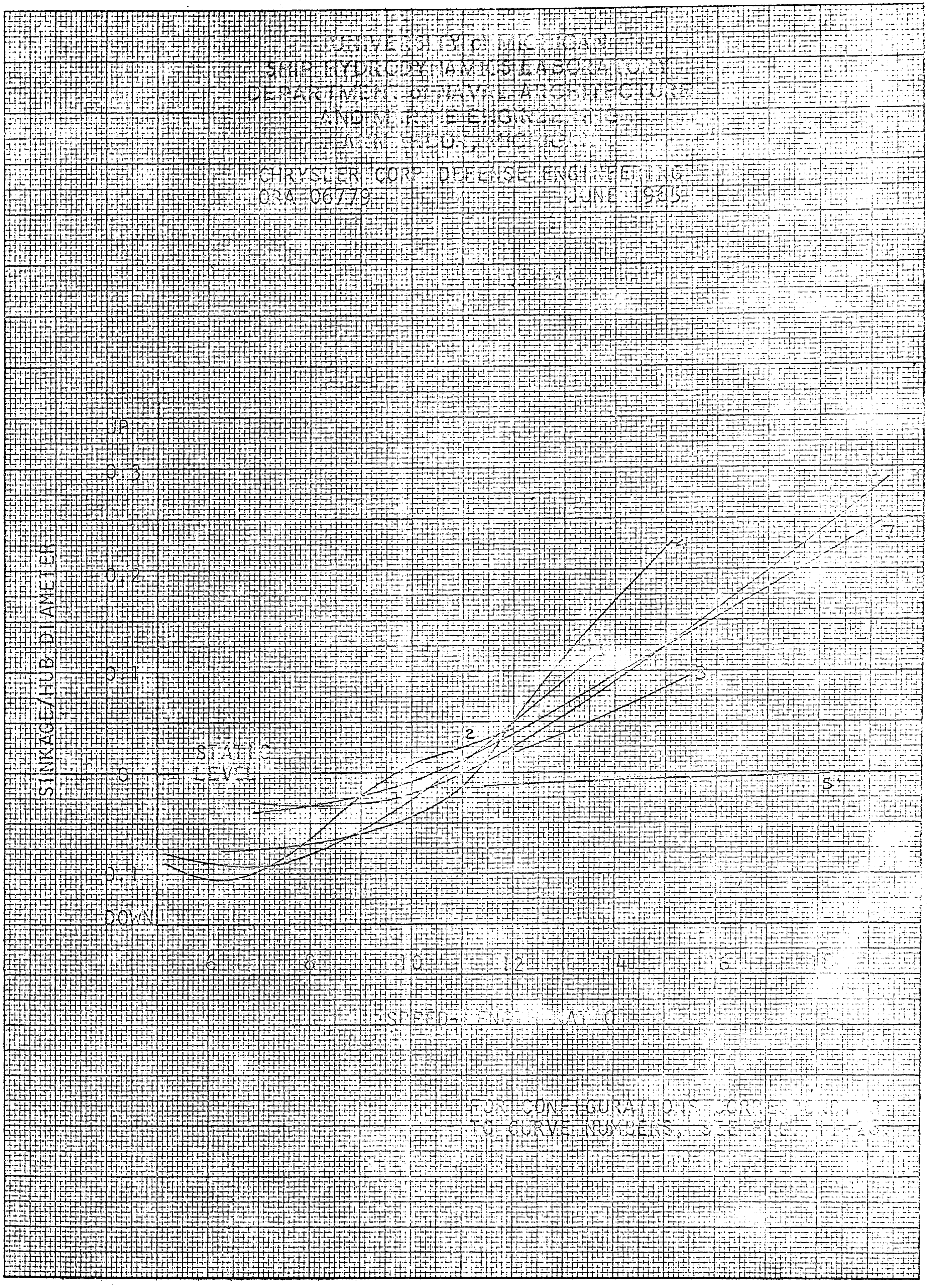
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
φ = 60 deg.
% DISPL = 75
3.0
2.0
1.0
H/D
.2344
.1950
.1563
.1150
5
10
15
20
SPEED-LENGTH RATIO

FIGURE 11-24

FIGURE 11-25



CHRYSLER CORP
ORA 06779
JUNE
UP
0.3
0.2
0.1
0
0.1
DOWN
SINKAGE/HUB DIAMETER
STATIC
6
8
10
12
14
16
2
3
5
7

FIGURE 11-26

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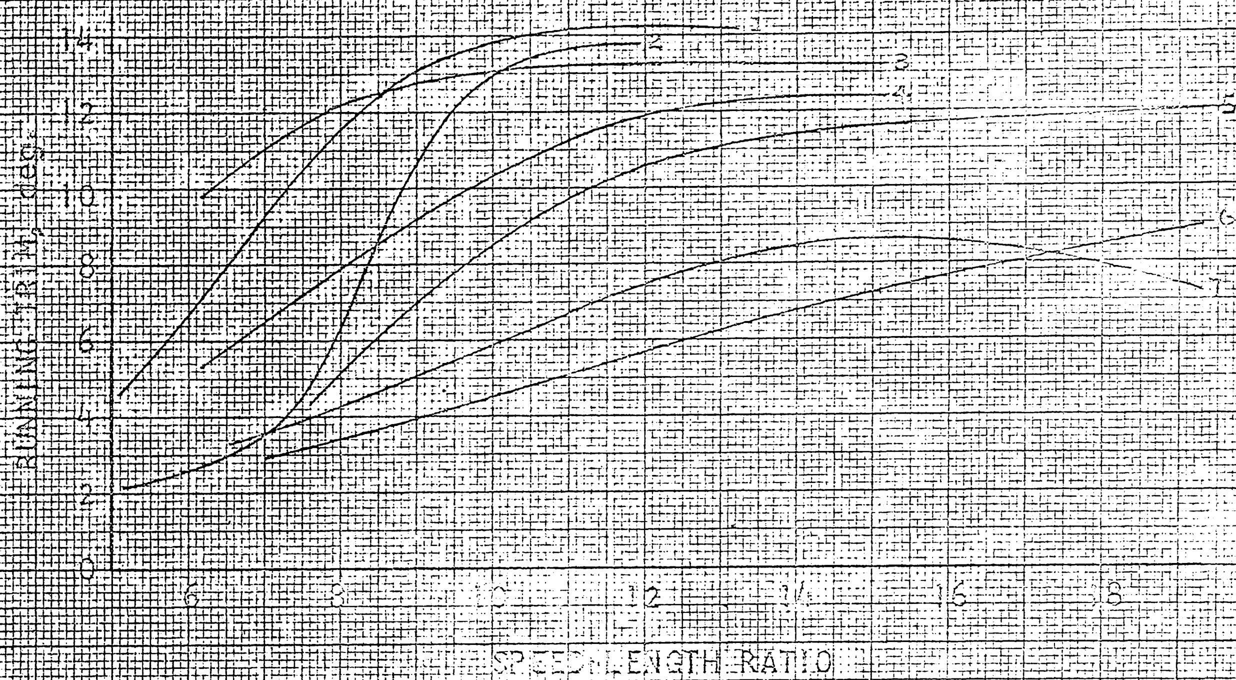
CHRYSLER CORP. DEFENSE ENGINEERING
ORA 06779 JUNE 1965



CURVE	Φ	[illegible]	% DISPL
1	50	.1950	75
2	40	.1563	50
3	60	.1950	50
4	50	.1950	50
5	30	.1150	25
6	50	.1563	25
7	50	.2950	25

Section III: Scaling and Scale Effects

Explanation of Symbols:

D	rotor hub diameter, ft.
F	force in direction of helix, lbs.
L	rotor length, either 2 ft. or 4 ft.
n	rotary speed, revolutions/sec.
N	force normal to blade, lbs.
Q	torque absorbed, ft.-lbs.
v	vector summation of rotary and advanced speeds $v^2 = (\pi nD)^2 + (V_m)^2$, ft./sec.
V_m	advance (towing carriage) speed, ft./sec.
α	angle of attack, degrees
ν	kinematic viscosity, 1.0552×10^{-5} ft.^{2}/sec. @ 70° F.F.W.
ρ	density, $1.9362 \frac{\text{lb. sec.}^2}{\text{ft.}^4}$ @ 70° F.F.W.
σ	surface tension, .00497 lb./ft. @ 70° F.F.W.
ϕ	helix angle, degrees
K_q	total torque coefficient, $Q / \frac{\rho n^2}{2} D^5$
K_{qf}	frictional torque coefficient $Q_f / \frac{\rho}{2} n^2 D^5$
R_e	Reynolds number, vD/ν
$\bar{\sigma}$	"spray number," $\frac{\rho v^2 D}{2\sigma}$
V_m/nD	advance coefficient
$V_m/\sqrt{L}$	speed-length ratio

Please notice that the nomenclature in this section differs slightly from that of Sections II and IV, specifically for v, torque coefficient, advance coefficient and speed-length ratio.

Note: The speed-length ratio is Froude number times $g^{\frac{1}{2}}$ and differs from speed-length ratio used in ship resistance work in that the latter has speed expressed in knots.

As pointed out in the University of Michigan report 06100-1-F dated January 1964 to Chrysler Corporation, Defense Engineering: "The Marsh Screw rotor can be likened to neither a ship nor propeller in that while it contributes to the buoyany of the vehicle it has drag force associated with it as in the case of the ship hull. Concurrently, however, the rotor provides propulsive thrust as does a propeller." That is, in the case of the ship and propeller combination means have been developed to treat each as a separate system utilizing concepts related to the interaction between the two systems. Unfortunately, the Marsh Screw does not lend itself to the same treatment, but certain analogies may apply.

The purpose of the present Section is to exploit as far as possible data obtained which may shed some light on the basic phenomena occurring. Particular emphasis is directed toward problems of scaling from model to prototype size the basic quantities involved.

A series of eight tests were run. Four rotors 4 ft. long of two different configurations were run as well as four rotors 2 ft. long of the same configurations. The test conditions are set forth in the following table.

TEST CONDITIONS FOR INVESTIGATION OF SCALING

Test No.	Helix Angle	Blade Ht.	% Displ.	Initial Trimming Angle
3	40 deg.	1.56 in.	25	2.14 deg.
3 A	40 deg.	0.78 in.	25	2.14 deg.
8	40 deg.	1.56 in.	50	4.01 deg.
8 A	40 deg.	0.78 in.	50	4.01 deg.
29	50 deg.	1.56 in.	50	0 deg.
29 A	50 deg.	0.78 in.	50	0 deg.
32	50 deg.	1.56 in.	25	0 deg.
32 A	50 deg.	0.78 in.	25	0 deg.

Tests designated with the suffix "A" were of 2 ft. long rotors, and the remainder of 4 ft. rotors. The sinkage and trim of the smaller rotors were pre-set to represent the same attitude with respect to the water as the 4 ft. rotors at corresponding speed-length ratios.

A. Dimensional Analysis

Using a dimensional analysis approach for torque on a rotor $Q = f(D \text{ or } L, n, V_m, \nu, \rho, \sigma, g)$ where for the rotors of interest either diameter or length may be used since the ratio of the two dimensions is constant. Torque would also vary according to blade height, helix angle and displacement but since only tests of two sizes of rotors with these items fixed are being investigated they may be left out. That is, comparison between pairs of tests (Test Nos. 3 and 3 A, etc.) is being made, not between tests of varying configurations. Surface tension, σ, is included because of visible dissimilarities in the spray patterns between small and large models.

In non-dimensional parameter form $K_q = g(R_e, V_m/\sqrt{L}, \bar{\sigma})$, where the velocity term in the Reynolds number is the vector summation of the rotary and advance velocities the use of which is completely analogous to the treatment normally given marine propellers. The surface tension parameter, $\bar{\sigma}$, is called "spray number" here because of the descriptiveness of the term.

A similar procedure may be taken for the rotary speed parameter, V_m/nD, which is called advance coefficient.

Both torque coefficient and advance coefficient are plotted against each of the dimensionless numbers, Reynolds number, speed-length ratio and spray number in Fig. III-1, III-2, and III-3 respectively. Speed-length ratio differs from being dimensionless by the constant $g^{\frac{1}{2}}$.

Fig. III-1 lends itself to analogy with the ship model case where total resistance coefficients are plotted against Reynolds number as has been done for K_q here. Several observations and conclusions with regard to the K_q plot follow.

a) with increasing R_e corresponding pairs of tests give converging results such that one might expect identical K_q at the same R_e for large and small rotors with sufficiently high R_e.

b) again for high enough R_e, K_q is fairly constant under any given test configuration.

c) at low R_e and for the small rotors there is a noticeable drop-off in the value of K_q. This might be taken as analogous to the ship model case when there is lack of fully developed turbulent flow. Coincidentally, perhaps, above a R_e of roughly $2x10^5$ this problem does not exist and that R_e is generally accepted in model propeller open water tests as being the minimum desirable in order to avoid Reynolds scale effect. In each case the definitions for R_e are completely analogous.

d) for ship models of varying size one can connect points of equal speed-length ratio on the resistance coefficient curves as has been done in Fig. III-1 for Tests 29 and 29A. For the ship model one expects decreasing negative slopes of the resulting lines with increasing speed-length ratio. This occurs because of the decreasing negative slope of the flat plate friction coefficient line with increasing R_e. A similar effect can be observed in Fig. III-1 except for the lowest speed-length ratio. However, had K_q for Test 29A not decreased below $R_e = 2x10^5$ all these lines would exhibit decreasing slope with increasing R_e.

e) for the ship model case total resistance coefficients are plotted and the difference between the resulting line and the flat plate line is expanded by Froude's Law. The skin friction coefficient is added back in at the appropriate R_e. For the marsh screw an analogous procedure can be adopted if the appropriate friction-torque coefficient line can be found. However, one can easily imagine large acceleration and deceleration of the flow on the immersed portion of the rotor as well as other flow phenomena dissimilar between shearing flow on a flat plate and that on a rotor. No doubt these would detract from the accuracy of extrapolation in a manner analogous to the ship case. Further discussion on equivalent flat plate frictional torque lines will be found in subsequent paragraphs.

In summary, Fig. III-1 does not provide an extrapolation method although it does suggest that a procedure analogous to ship model resistance expansion to full scale could be sought with the addition of considerably more data than that presently in hand. In any event data below a R_e of $3x10^5$ should be regarded with suspicion because of the possibility of at least partially laminar flow.

Fig. III-3, that is K_q plotted against $\overline{\sigma}$, exhibits similar tendencies to those observed in Fig. III-1. Here the analogy with the ship case previously explained becomes somewhat confused in that the anomalies in correlating large and small rotor results at both low and R_e may be due to either surface tension effects or shearing flow effects, or probably a combination of both. At high $\overline{\sigma}$, however, K_q is essentially constant (even more so than when plotted against R_e) showing a lack of torque dependency upon surface tension at sufficiently high $\overline{\sigma}$. However, below a $\overline{\sigma}$ of about 0.5 correlation is poor.

In both Figs. III-1 and III-3 the RPM parameter, advance coefficient, is not discussed. The plots are shown only to display lack of correlation.

Also little can be said of the results in Fig. III-2 alone except that these give a more vivid picture of the differences found between corresponding pairs of tests. If in Figs. III-1 and III-3 it had been possible to find a method of subtracting from the total torque those amounts due to frictional flow and surface tension, the remainder, when plotted as in Fig. III-2, might show good correlation. That is, the differences shown in Fig. III-2 are due to varying R_e and $\overline{\sigma}$. However, in Figs. III-1 and III-3 it has been shown that these effects which are large at low values of R_e and $\overline{\sigma}$ are diminished at sufficiently high values of these non-dimensional parameters. Therefore, expansion of data on the basis of Froude's Law alone becomes increasingly more accurate with increasing model size. Further, the results of expansion of data from 2 ft. rotors by Froude's Law are inaccurate even to the extent that relative merits of one rotor compared to another may not necessarily be in the correct order. The latter should not be expected of results from 4 ft. rotors.

B. Frictional Torque Extrapolation Line

As stated in Section III-A, if it were possible to define a frictional torque line analogous to a flat plate friction line as used for extrapolation in the ship model case, as well as properly determining the spray scale effect, expansion of rotor torque might be handled in the same manner as ship model resistance expansion. Flat plate friction lines are characteristically of high negative slope at low R_e with decreasing slope as R_e increases. At high enough R_e, the change in friction coefficient becomes a small percentage of the total resistance coefficient. The slope of the friction line, rather than the magnitude, is the predominant factor. With this in mind simplifying assumptions were made in order to estimate an equivalent frictional torque line. The assumptions were:

a) The flow follows a helical path, guided by the blades.

b) In the case chosen, 50 percent displacement, the effective length for computing R_e is half the helical distance from one end of the rotor to the other.

c) The velocity for computing R_e is the vector summation of the rotary and advance velocities.

d) At the R_e computed by assumptions b) and c) the flat plate friction coefficient, using the 1947 A.T.T.C. line, may be multiplied by the hub radius and the cosine of the helix-angle to obtain the equivalent frictional torque coefficient.

The computation was made for Test No. 32 and the resulting line is shown in Fig. III-1, designated K_{qf}. The slope of the line is extremely flat at the higher R_e's so that, under the above assumptions, one can assume very little frictional scale effect in going from 48 inch long rotors to full scale.

C. Proportion of the Total Drag of the Various Forces--Viscous, Spray, and Wave-Making

Sections III-A and B pointed out lack of correlation between corresponding pairs of 24 inch and 48 inch long rotors and insignificant frictional scale effect in extrapolating from 48 inch rotors to full scale. Since the possibility of scale effect on the spray component still exists an analysis of the scaling of the spray drag is desirable. Unfortunately, there is no direct method analogous to the ship case for such an analysis as there was for the frictional component. However, an estimate of the proportions of the various forces can be made.

Figure III-4 shows a vector diagram of the velocities and forces on an element of a rotor blade. The forces parallel to the blades are represented by F which is the frictional component. Normal forces are represented by N. For the model propulsion point condition

$$F \sin \phi = N \cos \phi$$

where ϕ is the helix angle and the left hand side is the overall hydrodynamic drag which is overcome by the term on the right hand side. For the torque on the rotor

$$Q = \frac{D}{2}(F \cos \phi + N \sin \phi)$$

where $F \cos \phi$ represents frictional torque force and $N \sin \phi$ is the induced torque force which is represented in the wave and spray energies. Under any given configuration, where ϕ is known, the two equations may be solved for $F \cos \phi$ and $N \sin \phi$ which may then be expressed as fractions of the total. For a 50 degree helix angle rotor

$$N \sin \phi = .59\left(\frac{2Q}{D}\right)$$

$$\text{and } F \cos \phi = .41\left(\frac{2Q}{D}\right)$$

and so forth for other helix angles. Fig. III-5 shows the percentage split between frictional and induced torque versus helix angle. It should be kept in mind that this method is only an estimate since the effects of trim are not accounted for in Fig. III-4 and it seems unlikely that the percentages shown in Fig. III-5 are independent of speed, blade height, etc. However, the trends of the curves in Fig. III-5 should be valid in which case less frictional scale effect and more spray scale effect at high helix angles and the reverse at low angles may be expected.

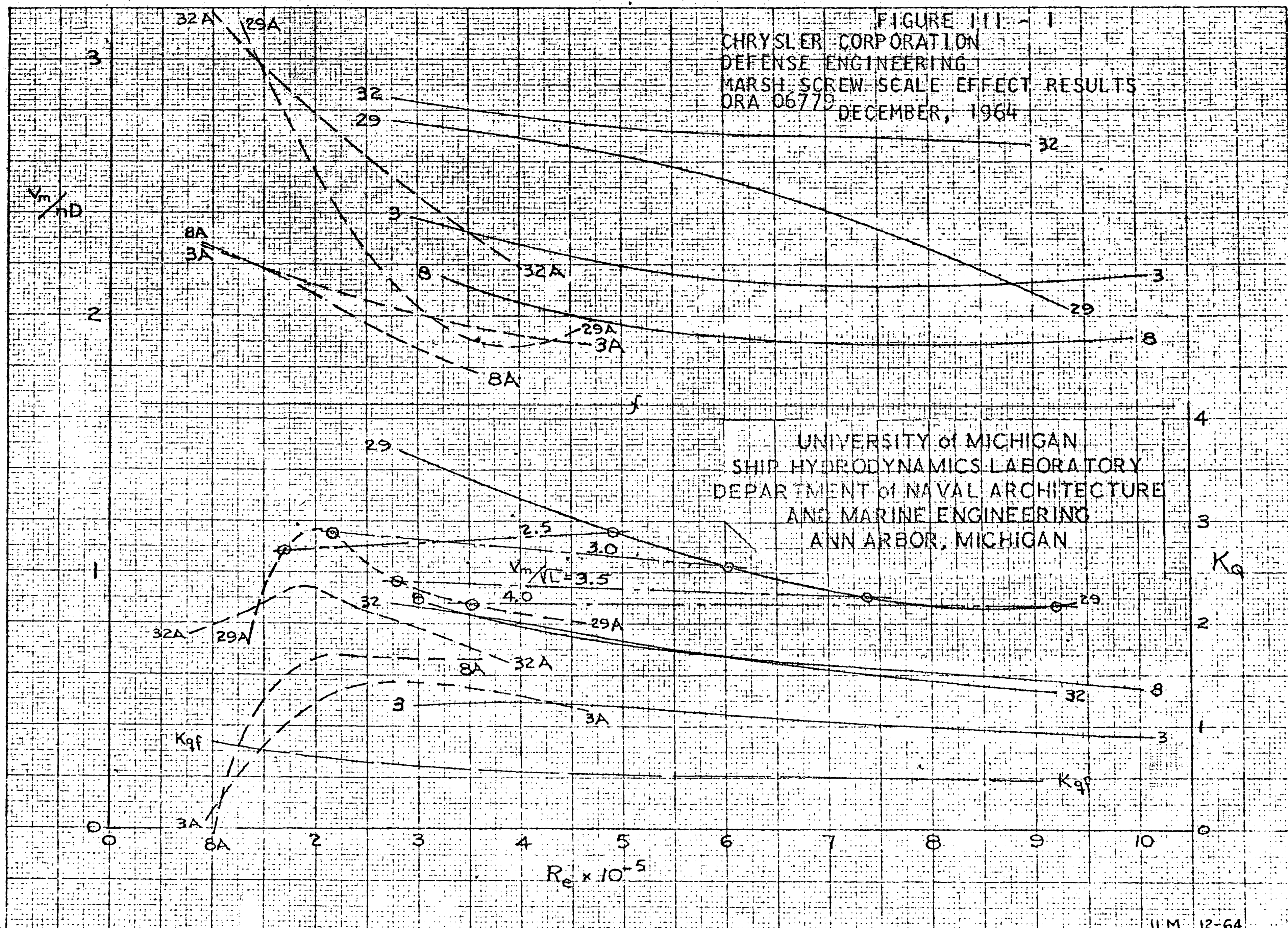
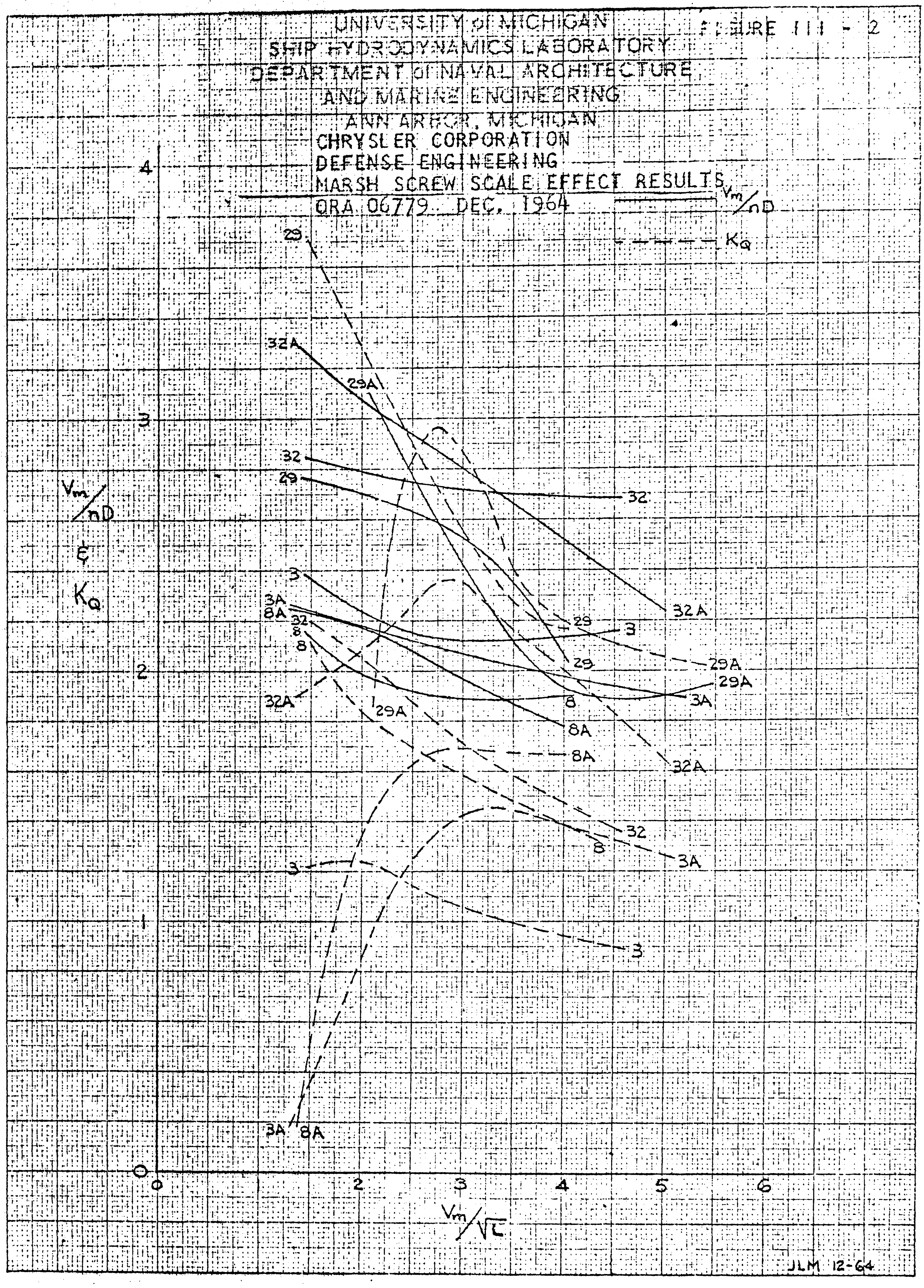


FIGURE III - 1
CHRYSLER CORPORATION
DEFENSE ENGINEERING
MARSH SCREW SCALE EFFECT RESULTS
ORA 06779 DECEMBER, 1964
UNIVERSITY of MICHIGAN
SHIP HYDRODYNAMICS LABORATORY
DEPARTMENT of NAVAL ARCHITECTURE
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
Vm/nD
KQ
f
Kqf
Vm/√L = 3.5
2.5
3.0
4.0
Re x 10^-5
JLM 12-64



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CHRYSLER CORPORATION
DEFENSE ENGINEERING
MARSH SCREW SCALE EFFECT RESULTS
ORA 06779 DEC. 1964
FIGURE III - 2
Vm/nD
KQ
Vm/nD & KQ
Vm/√L
0
2
3
4
5
6
1
29
32A
29A
32
29
3
3A
8A
32
8
8
32A
29A
3A
8A
29
3
29A
29A
3A
8
8A
8A
32A
32
8
3A
3
3
32A
JLM 12-64

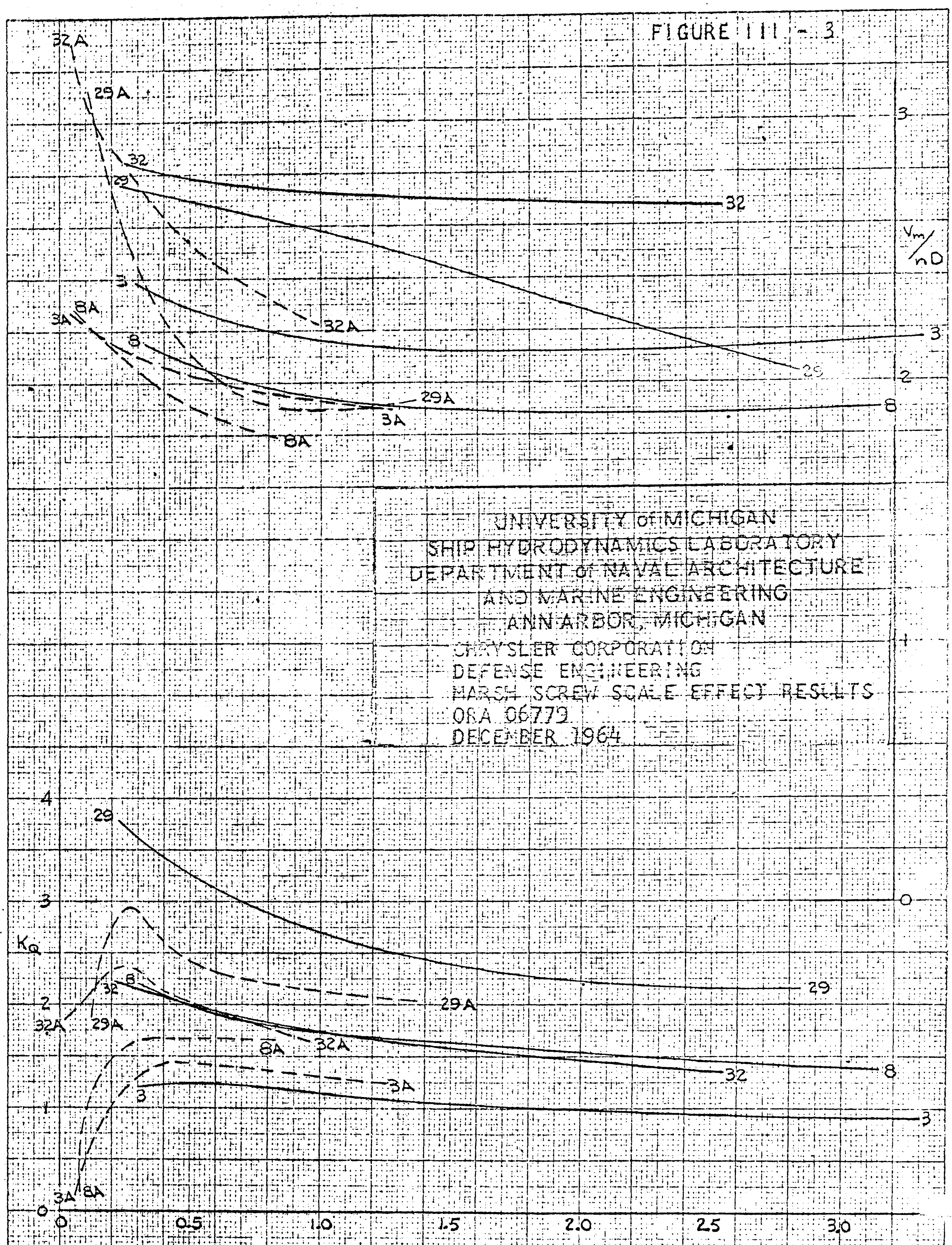

FIGURE III - 3
32A
29A
32
29
32
3
3A
8A
32A
8
3
29
2
29A
3A
8
8A
V_m/nD
3
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CHRYSLER CORPORATION
DEFENSE ENGINEERING
MARSH SCREW SCALE EFFECT RESULTS
ORA 06779
DECEMBER 1964
4
29
3
0
K_Q
8
32
2
32A
29A
29
29A
8A
32A
32
8
3A
3
1
3
3A
8A
0
0
0.5
1.0
1.5
2.0
2.5
3.0
σ
JLM 12-64

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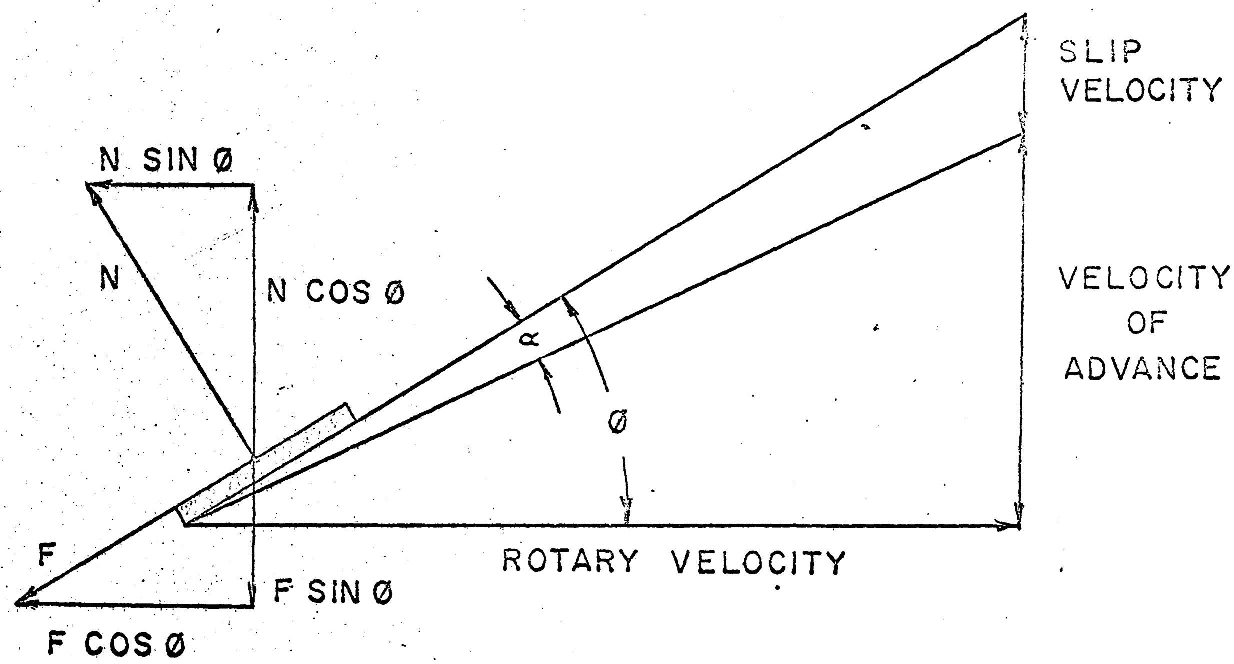

FIGURE III-4 ROTOR ELEMENT VELOCITY AND FORCE VECTOR DIAGRAM

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Figure III-5

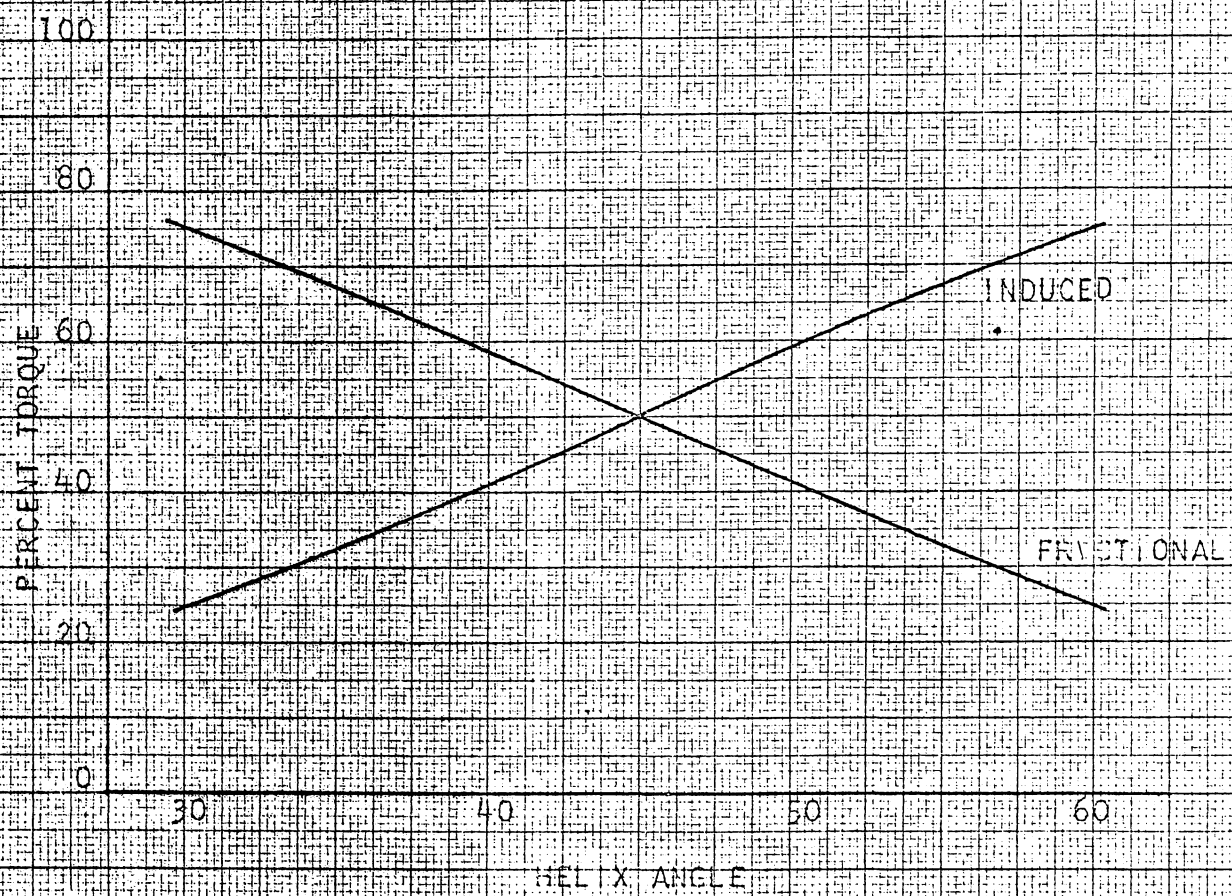
PERCENTAGE INDUCED AND FRICTIONAL TORQUE

vs HELIX ANGLE

CHRYSLER MARSH SCREW

ORA PROJECT No. 06779

MARCH 1965



Section IV: Design Example

As agreed between laboratory and Chrysler personnel this section contains a step-by-step design procedure which is intended to illustrate the use of the material presented in previous sections. The systematically related set of test results of Section II will be utilized to select the optimum rotor configuration of L/D = 6.0. Remarks will be made concerning interaction and other effects from the results of Section I insofar as they pertain to the design example.

Statement of the Problem:

It is desired to determine optimum rotor length, diameter, blade height, helix angle, longitudinal center of gravity location, and RPM and horsepower relationships for a hypothetical vehicle with an overall length of 25 ft., an overall width of 12 ft., a payload of 3,000 lbs. and a light weight of 8,000 lbs.

It is assumed that there are to be two rotors running parallel to each other but that they need not be as long as the vehicle. In fact, it may be desirable to have the vehicle of smaller size than the stated limiting dimensions. Three rotor lengths are assumed, 24 ft., 21 ft. and 18 ft., each slightly shorter than the intended vehicles, respectively, to allow for chain housings, etc. Length-diameter ratio is set at six. Three speeds are assumed, 12, 16 and 20 mph. The weight displacement of each rotor is half the vehicle weight plus payload, or 5500 lbs., assuming that all the buoyancy is contributed by the rotors.

The accompanying tables show calculations of revolutions per second and torque in ft.-lbs. as well as the product of the two which is directly proportional to horsepower. The values of J and K_q are obtained from the figures in Section II and by means of crossplots against displacement where the needed percent hub displacements are as shown in the table.

Initially for the 21 ft. rotor all blade heights are used in the calculation. The product of Q and n shows the highest blades to be the best and for the remaining two rotor lengths only values for H/D = .2344 are calculated. That is, it is assumed that there is no limitation on blade height, but for land and marginal terrain use blade height may have to be limited in an actual design.

By inspection of the tabulated Qn values for H/D = .2344 for all three rotor lengths the 24 ft. rotor will yield the lowest power, but only slightly lower than that for the 21 ft. rotor. In an actual design the weight displacement would probably be less for smaller vehicles since light ship components would not be as heavy. In such a case the optimum configuration would be a vehicle of length less than 25 ft., but for illustrative purposes here it is assumed that a rotor length of 24 ft. is considered best.

Finally, the best helix angle is to be determined. Fig. IV-1 shows cross-plots against helix angle at three speeds for the 24 ft. rotor. The optimum angle is a function of speed as the broken line on Fig. IV-1 indicates, but $\phi = 52$ deg. is selected since it is assumed that sufficient power is available to propel the vehicle at 20 mph. As the last step, the developed horsepower and RPM values are computed and plotted as in Fig. IV-2. The designer would probably use more than three speeds in this last computation.

The curves shown in Fig. IV-2 are an estimate only and adjustments have to be made for chain housings, shrouds, spacing between rotors, etc., all dependent on the actual geometry of the rest of the vehicle. For instance, for the rotors selected the centerline spacing is 1.53 diameters to keep the overall width, from port bladetip to starboard tip, within 12 ft. According to Fig. I-H-1, an approximate 30 percent increase in power will result from the narrow spacing if the flow between the rotors is totally unimpeded. The shrouds and hull may limit the adverse condition but it may also be necessary to reconsider the shorter rotors since the vehicle width need not be reduced simultaneously with the length. A greater spacing would result with less adverse interference. If shrouds are resorted to in order to limit the interference flow, Fig. I-I-1 indicates that shrouds with small enclosed angles are best, but if the angle is too small the interference flow will be largely uninhibited. Carefully designed shroud size and placement will be required. In short, as with ships, further model tests may be recommended to refine the design.

$L = 21.00'$ $\% \Delta = 44.80$ $\lambda = 5.250$ $J = \dfrac{V}{n \nabla^{1/3}}$

$\nabla = 65.88 \text{ FT}^3$ $\rho = 1.991 \dfrac{\text{LB-SEC}^2}{\text{FT}^4}$ $K_Q = \dfrac{Q}{\rho V^2 \nabla}$

$\nabla^{1/3} = 4.412$ $\rho \nabla = 171.0$

$\nabla^{1/6} = 2.100$

$H/D = .1150$ $D = 3.500'$ $H = .4025'$ $H^3 =$

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				10 K_Q				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°
12	17.60	8.38	309.8	.76	1.08	1.29	1.46	.96	.90	.92	1.13	5.25	3.69	3.10	2.74	509	477	487	599	267	176	151	162
16	23.47	11.18	550.8	.72	1.02	1.22	1.36	.78	.75	.76	.89	7.37	5.22	4.37	3.91	735	706	716	838	542	369	312	328
20	29.33	13.97	860.2	.67	.96	1.18	1.30	.65	.62	.59	.71	9.92	6.93	5.66	5.11	956	912	868	1044	948	632	491	533

$H/D = .1563$ $D = 3.500'$ $H = .5471'$ $H^3 =$

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				10 K_Q				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°
12	17.60	8.38	309.8	.93	1.37	1.70	1.99	1.02	.97	.86	1.04	4.29	2.92	2.35	2.01	540	514	456	551	232	150	107	111
16	23.47	11.18	550.8	.88	1.31	1.60	1.89	.84	.80	.75	.87	6.04	4.08	3.32	2.82	791	754	706	819	478	308	234	231
20	29.33	13.97	860.2	.82	1.26	1.57	1.87	.66	.62	.59	.68	8.13	5.26	4.24	3.56	971	912	868	1000	789	480	368	356

$H/D = .1950$ $D = 3.500'$ $H = .6825'$ $H^3 =$

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				10 K_Q				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°
12	17.60	8.38	309.8	1.00	1.54	1.94	2.09	1.10	1.01	.91	1.15	3.99	2.59	2.06	1.91	582	535	482	609	232	139	99	116
16	23.47	11.18	550.8	.94	1.50	1.83	1.99	.88	.78	.70	.92	5.69	3.54	2.92	2.67	829	735	659	867	472	260	192	231
20	29.33	13.97	860.2	.90	1.46	1.81	1.95	.70	.59	.57	.71	7.39	4.56	3.68	3.42	1030	868	838	1044	761	396	308	357

$H/D = .2344$ $D = 3.500'$ $H = .8204$ $H^3 =$

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				10 K_Q				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°	ϕ=30°	40°	50°	60°
12	17.60	8.38	309.8	1.12	1.69	2.15	2.57	1.06	.91	.83	1.13	3.56	2.36	1.86	1.55	561	482	440	599	200	114	82	93
16	23.47	11.18	550.8	1.08	1.62	2.07	2.49	.79	.69	.62	.83	4.93	3.29	2.57	2.14	744	650	584	782	367	214	150	167
20	29.33	13.97	860.2	1.02	1.57	2.04	2.41	.57	.51	.45	.60	6.53	4.24	3.27	2.76	838	750	662	883	547	318	216	244

L = 18.00′

$\% \Delta = 71.15$

$\nabla = 85.88\ FT^3$

$\nabla^{1/3} = 4.412$

$\nabla^{1/6} = 2.100$

$\lambda = 4.500$

$\rho = 1.991\ \frac{LB\text{-}SEC^2}{FT^4}$

$\rho\nabla = 171.0$

$J = \frac{V}{\eta \nabla^{1/3}}$

$K_Q = \frac{Q}{\rho V^2 \nabla}$

H/D = .2344 D = 3.600′ H = .7032′ H^3 =

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				$10 K_Q$				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°
12	17.60	8.38	309.8	.76	1.20	1.46	1.67	.99	.86	.76	.95	5.25	3.32	2.74	2.39	525	456	403	503	276	151	110	120
16	23.47	11.18	550.8	.69	1.08	1.32	1.55	.70	.61	.56	.67	7.77	4.92	4.02	3.44	659	575	527	631	512	283	212	217
20	29.33	13.97	860.2	.64	1.01	1.30	1.47	.51	.46	.41	.49	10.47	6.58	5.11	4.53	750	677	603	721	785	445	308	327

L = 24.00′

$\% \Delta$

$\nabla = 85.88\ FT^3$

$\nabla^{1/3} = 4.412$

$\nabla^{1/6} = 2.100$

$\lambda = 6.000$

$\rho = 1.991\ \frac{LB\text{-}SEC^2}{FT^4}$

$\rho\nabla = 171.0$

H/D = .2344 D = 4.000′ H = .9376 H^3 =

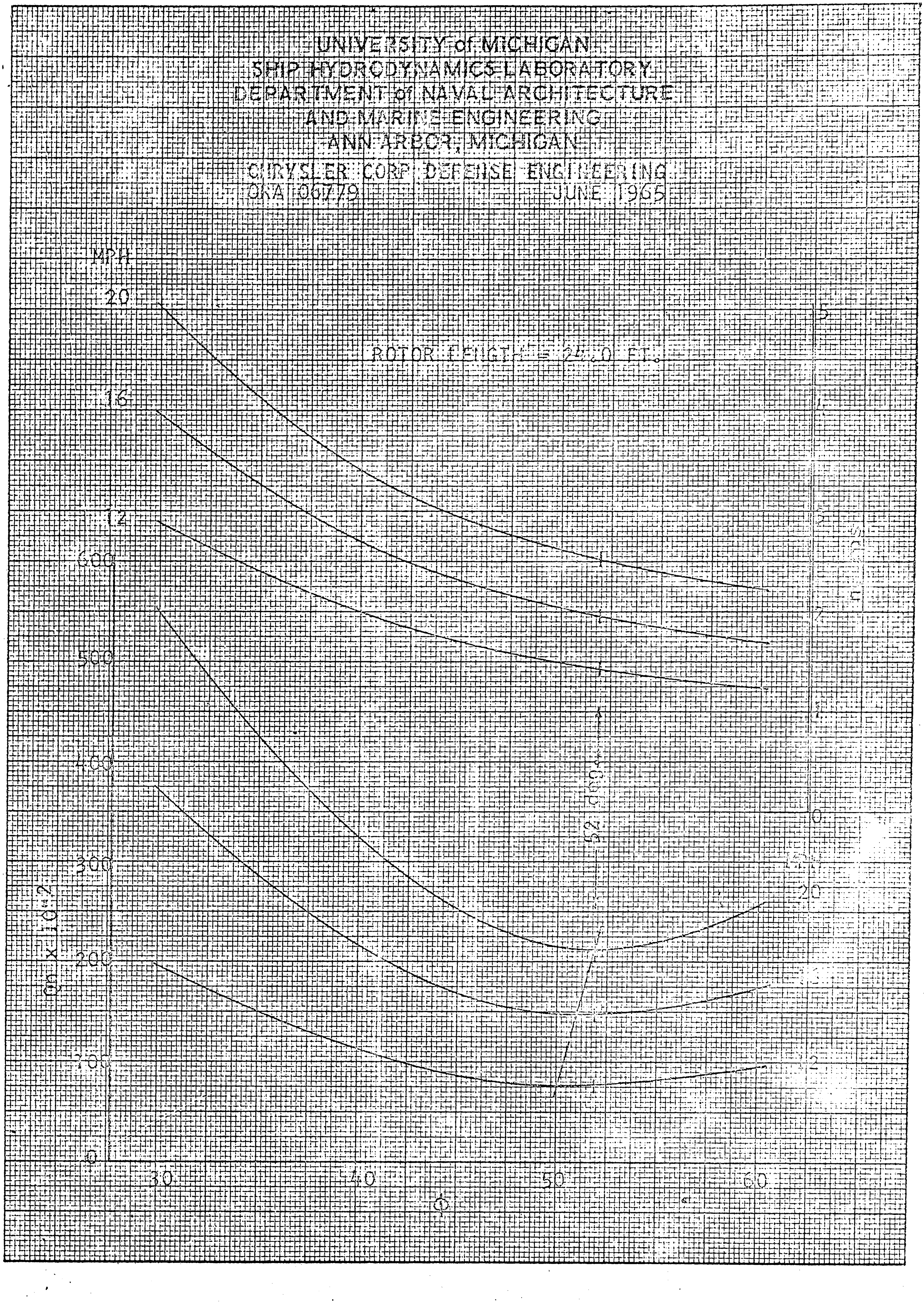
V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				$10 K_Q$				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°
12	17.60	8.38	309.8	1.38	2.02	2.71	3.21	1.29	1.09	.98	1.45	2.88	1.98	1.48	1.24	683	577	519	768	197	114	77	95
16	23.47	11.18	550.8	1.34	1.99	2.61	3.14	1.00	.87	.77	1.10	3.96	2.68	2.04	1.69	942	819	725	1036	373	219	148	175
20	29.33	13.97	860.2	1.32	1.96	2.55	3.01	.74	.65	.56	.79	5.04	3.40	2.61	2.21	1088	956	824	1162	548	325	215	257

H/D = D = H = H^3 =

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				$10 K_Q$				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°

H/D = D = H = H^3 =

V_{MPH}	$V_{FT/SEC}$	$\frac{V_{FT/SEC}}{\nabla^{1/6}}$	V^2	J				$10 K_Q$				η				$Q \times 10^{-1}$				$Q\eta \times 10^{-2}$			
				Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°	Φ=30°	40°	50°	60°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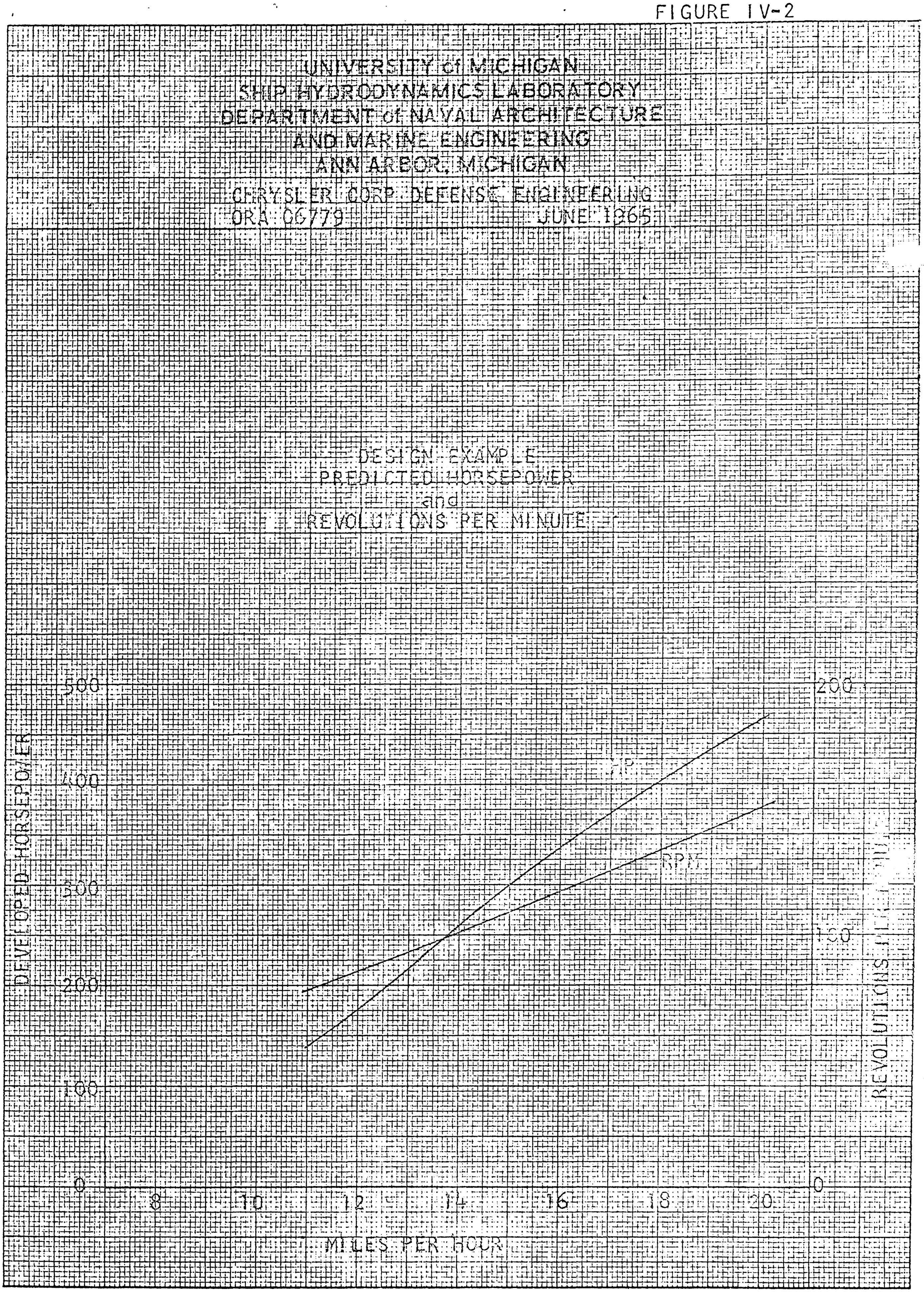
DEPARTMENT of NAVAL ARCHITECTURE
AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
ROTOR LENGTH = 24.0 FT.
MPH
20
16
12
600
500
400
300
200
100
0
30
40
50
60
52 000
20
φ

FIGURE IV-2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AND MARINE ENGINEERING
ANN ARBOR, MICHIGAN
CHRYSLER CORP DEFENSE ENGINEERING
ORA 06779
JUNE 1965
DESIGN EXAMPLE
PREDICTED HORSEPOWER
and
REVOLUTIONS PER MINUTE
DEVELOPED HORSEPOWER
500
400
300
200
100
0
HP
RPM
200
100
0
REVOLUTIONS
8
10
12
14
16
18
20
MILES PER HOUR