THE UNIVERSITY OF MICHIGAN

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Ship Hydrodynamics Laboratory

DEVELOPMENT OF HIGH-SPEED FULL-FORM HULL

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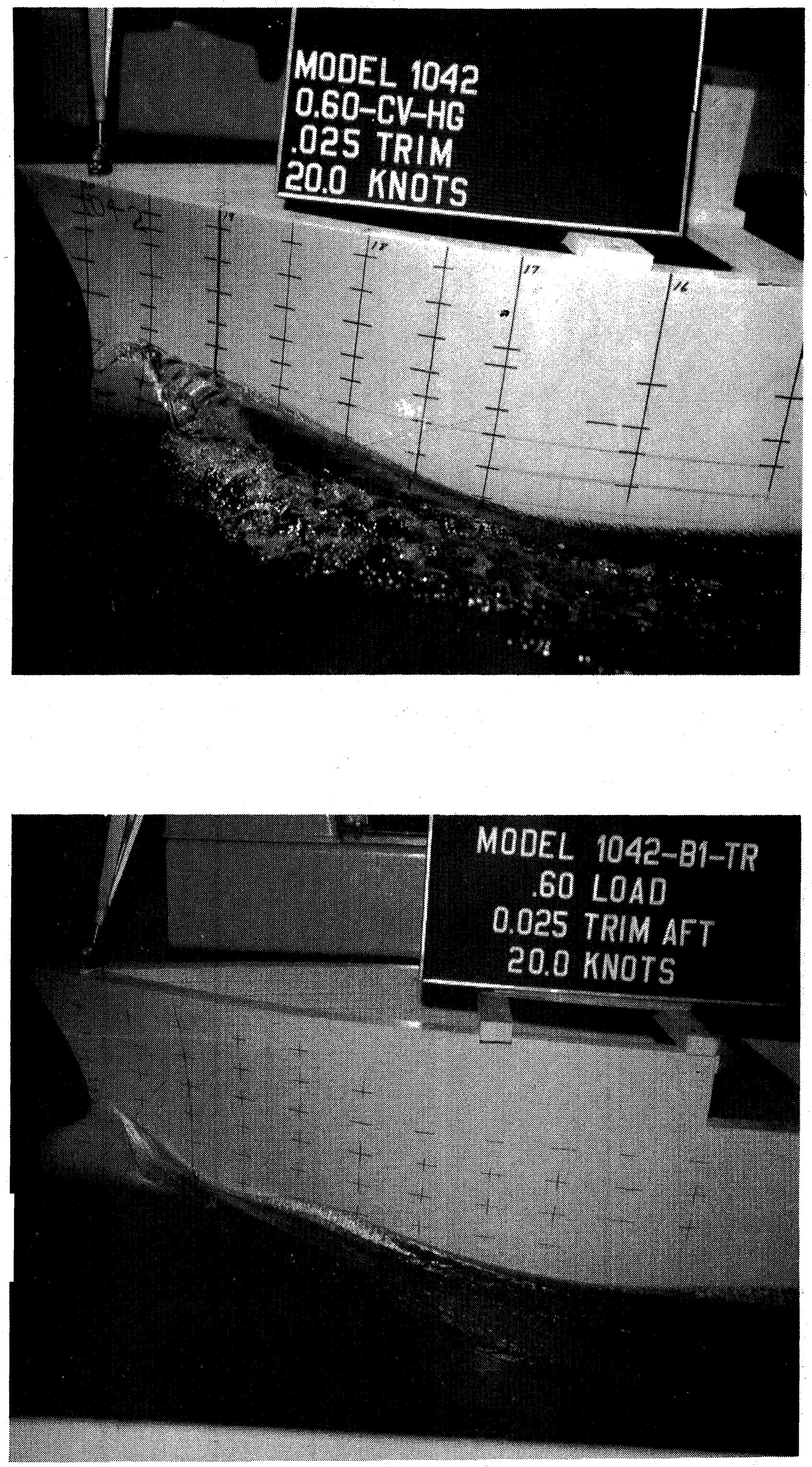
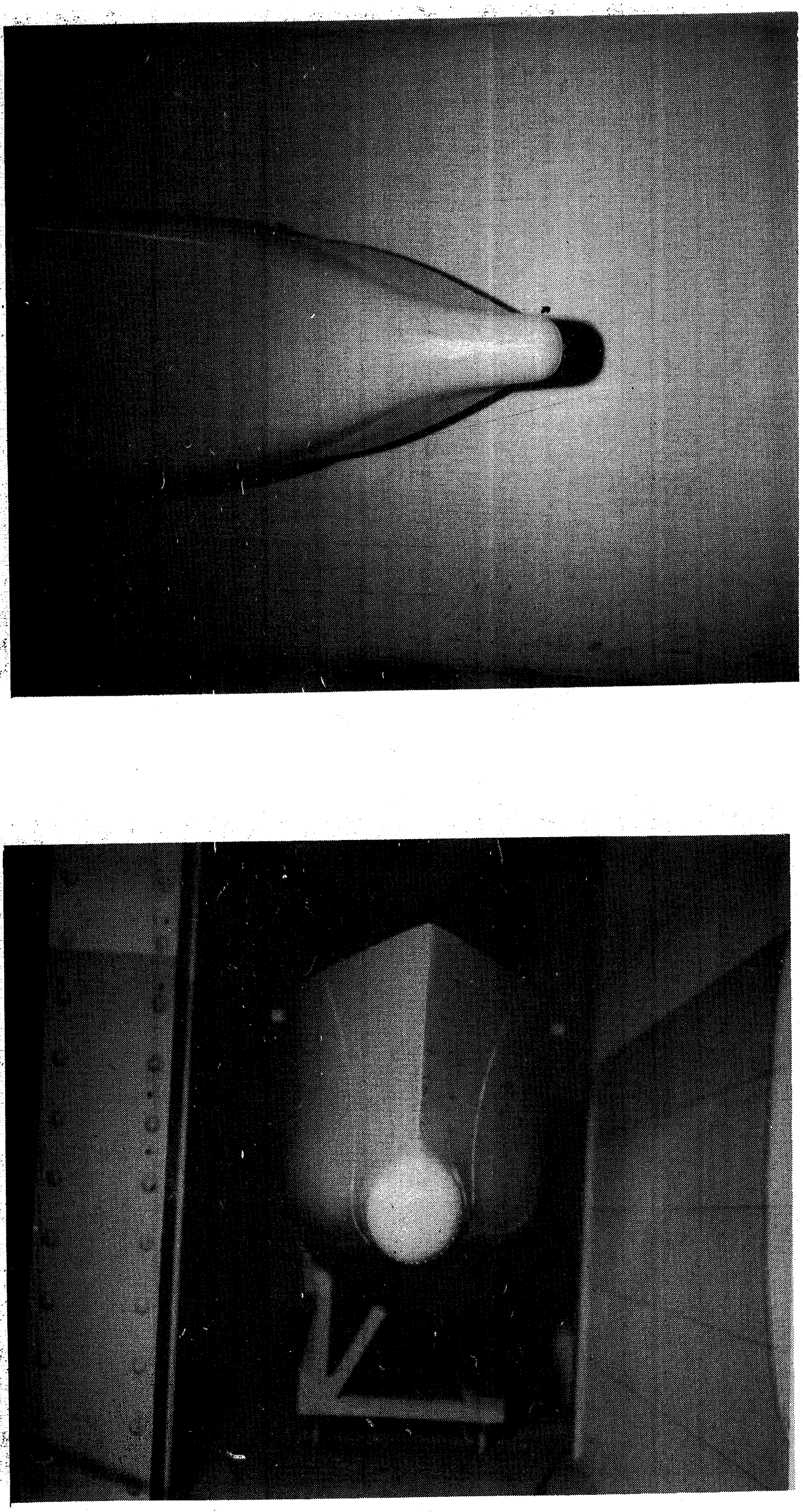
PREFACE

The following pictures of the models which were tested under the program described in this report are included here for the purpose of assisting the reader, at the outset, in developing a clearer idea of what types of hull forms have been investigated.

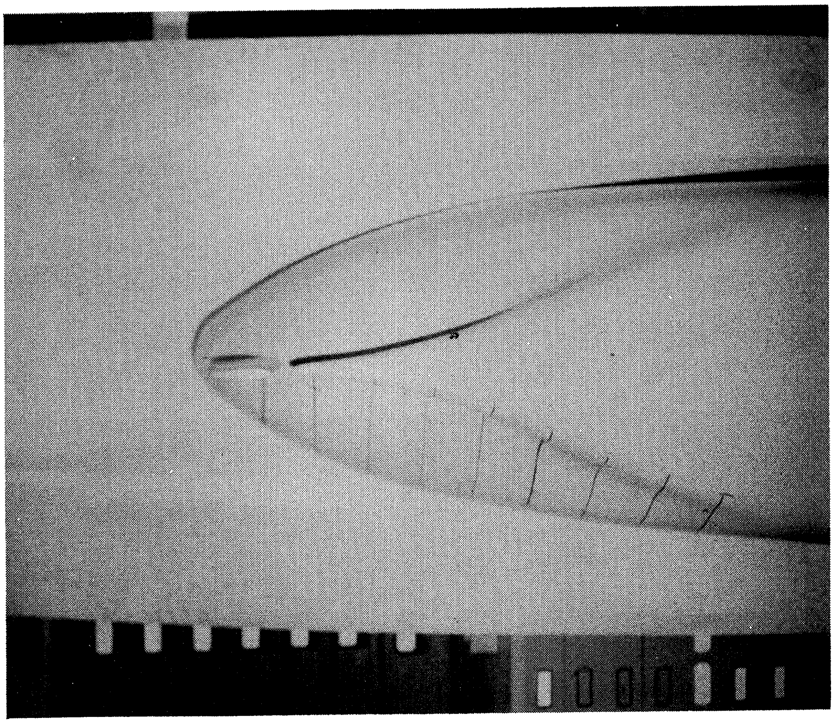


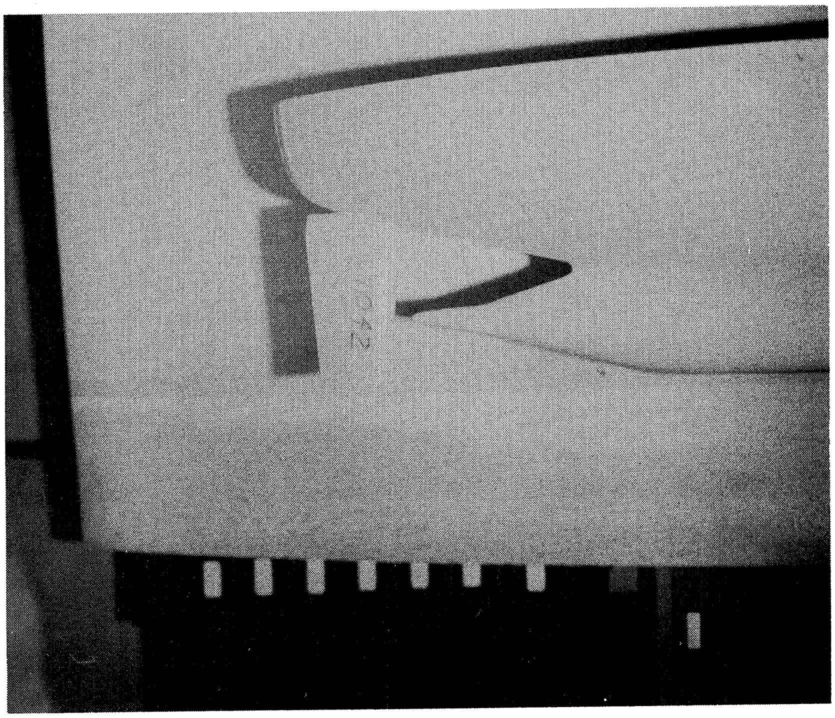
Comparison of Bow Waves
at $V/\sqrt{L} = 0.87$ (60% displ., $2\frac{1}{2}$% trim)

Top: Conventional Bow
Bottom: Bulbous Bow No. 1

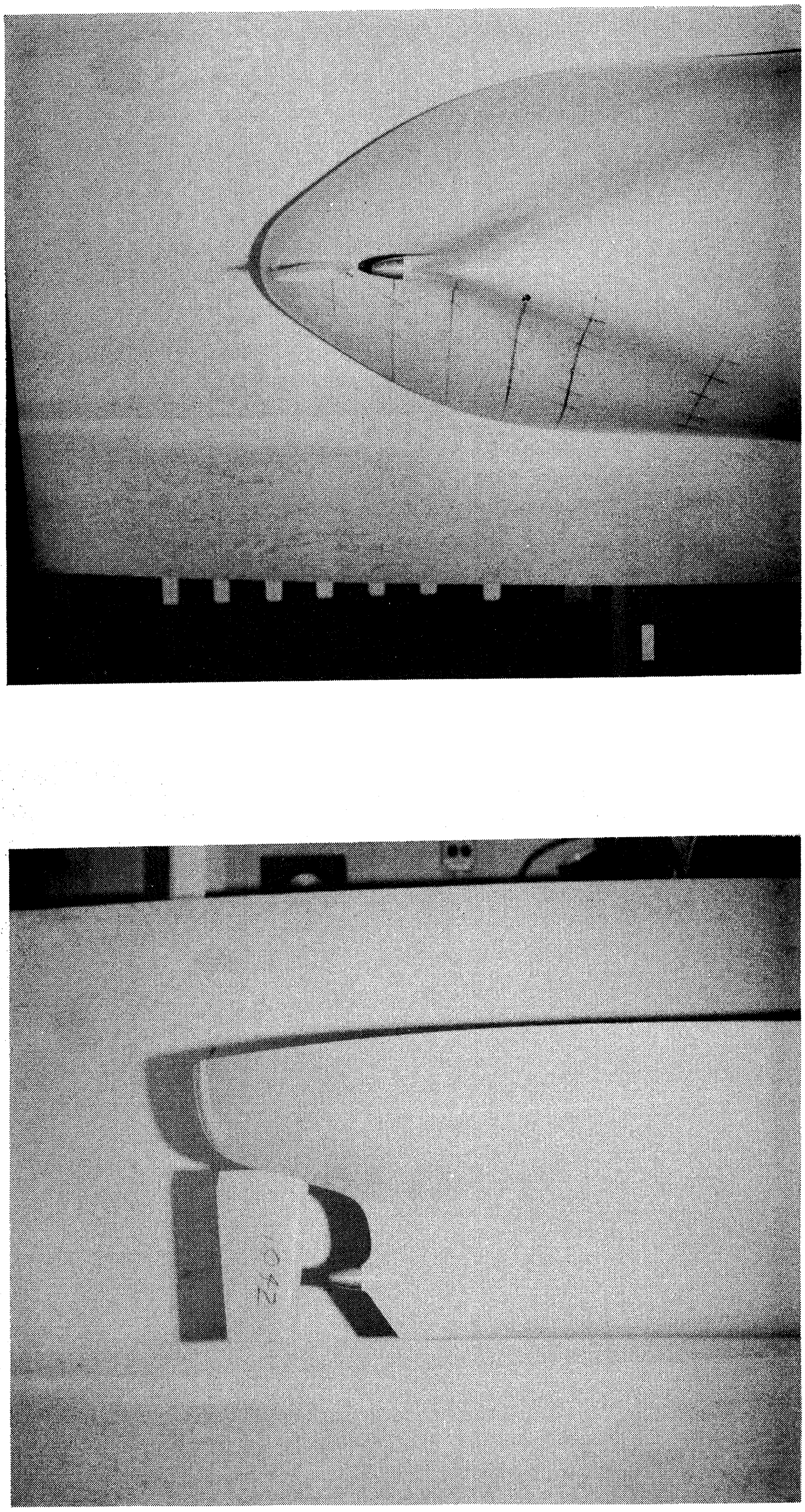


Modified Bulbous Bow No. 1

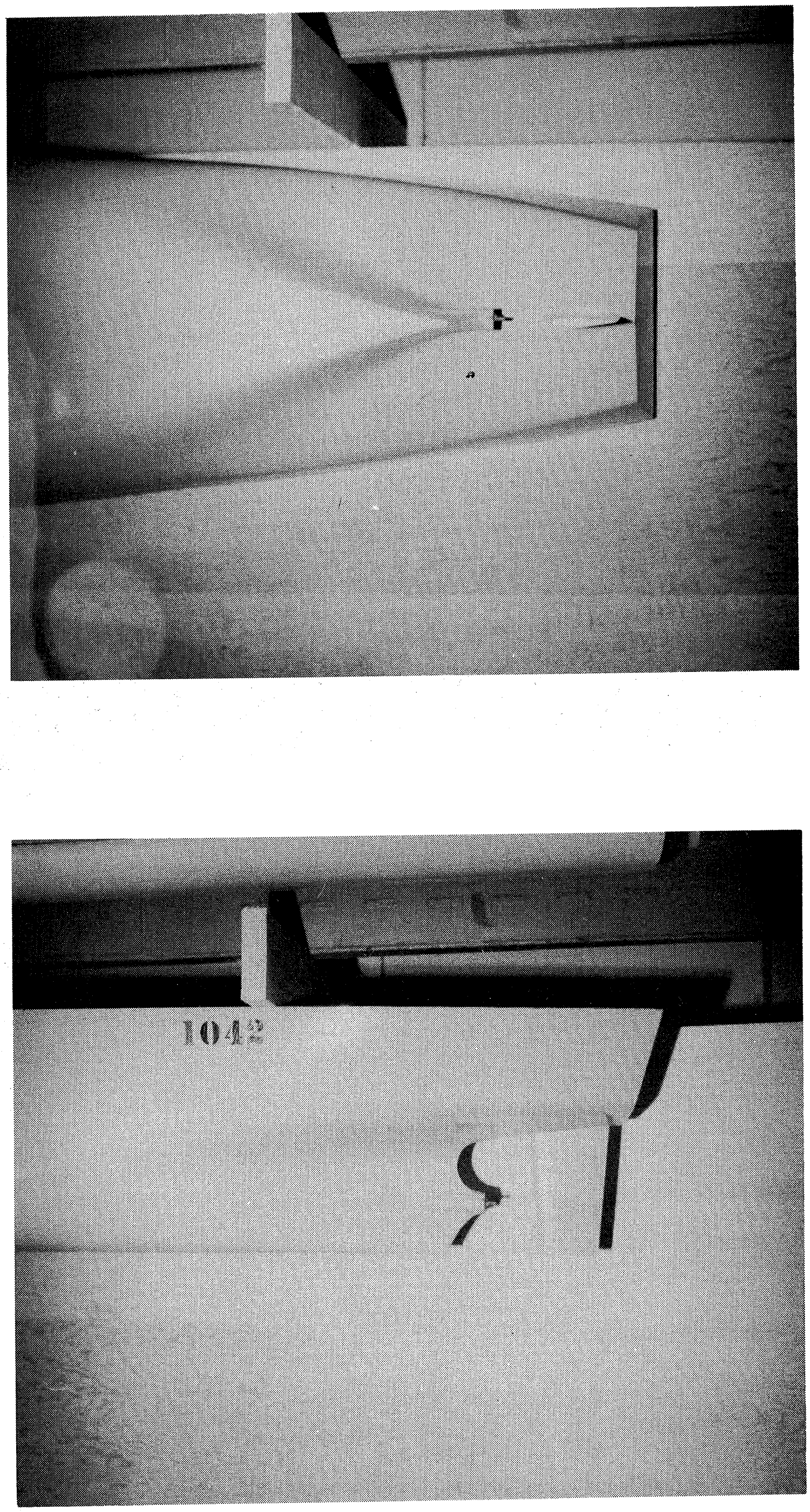




Modified Hogner Stern



Conventional Stern



Transom Stern

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DEVELOPMENT OF HIGH-SPEED FULL-FORM HULL

The purpose of the research reported herein is the development of a high-speed full-form hull configuration for the U.S. Maritime Administration. The work has been supported entirely by the Administration, and was initiated upon its request in September, 1964.

The course of this work has been to develop the configuration so that resistance is minimized at a chosen speed and design condition. The natural extension of this project should be the continuation of this development at the chosen speed for a small range of loading conditions, thus simulating actual operating practices more closely.

Although the design condition chosen has been that of full displacement, attention has also been given to the resistance characteristics at other displacements.

A brief summary of the sequence of development is given by listing the section titles:

Principal Characteristics
Forebody Design
Afterbody Design
Resistance Tests on Stern Configurations
Bulbous Bow Design
Resistance Tests on Bulbous Bows
Simulation of Change of Bow Configuration
Self-Propulsion Tests
Comparisons with Standard Series

The resistance tests were carried out on five different hull configurations of 14'-0" LBP. The first three consist of a conventional bow (i.e., without bulb) and different stern configurations: conventional; Hogner-type; and transom. The fourth configuration consists of the transom stern with a bulbous bow, and the fifth is the same except for a modification of the bulbous bow.

The self-propulsion tests were conducted on two configurations: the transom stern with the bulb-less bow and the transom stern with the modified bulbous bow.

PRINCIPAL CHARACTERISTICS

This development has been carried out on an experimental basis only, and only indirect attempt has been made to aid the development through the use of analytical solutions utilizing the concept of singularities in fluid mechanics.

The models may be considered to represent ships that have been "shortened" without sacrifice of displacement. This was initially suggested by P. C. Pien, and is fully discussed by him in his paper presented at the Fifth Symposium on Naval Hydrodynamics.[1]

It was initially thought that any full-form hulls would involve designs with high block coefficients. Investigations into the performance characteristics of a number of models tested at The University of Michigan's Ship Hydrodynamics Laboratory, and of the reports of model tests at other facilities both domestic and foreign, clearly indicated that high block coefficients (greater than, say, 0.80) would most likely lead to poor performance at the high speeds with which this project is concerned. (It should be noted that "high speeds" for ships as full as we are considering constitutes speed-length ratios in excess of 0.80.) The result of those investigations was to choose a block coefficient near to 0.75: a value which was considered the best compromise between the demands of speed and the demands of the desired fullness.

From the viewpoint of providing a large degree of freedom to vary the longitudinal distribution of displacement, a low block coefficient is desirable.

Fullness, as indicated by the displacement-length ratio, $\Delta/(L/100)^3$, was selected to be substantially higher than that for most full ships of comparable size. It was not the intent to make it extreme, however, with a value of about 190 being chosen. To keep the draft within reasonable limits, a beam/draft ratio of about 2.8 was adopted.

In order to minimize the effect of a wide beam, special attention was given to the section area curve in the forebody so that a pronounced "shoulder" was avoided. This is also reflected in the waterplane form.

The choice of the ship's length was based on the present high-speed cargo ship designs. Considering a length of 615' and a block coefficient of 0.60, the characteristics of a fictitious ship were calculated based on the parent form of Series-60 (C_B=0.60). Then that ship was shortened without sacrifice of displacement to a length approximating that of the Mariner class. A final length of 530' was chosen, which also gave a satisfactory displacement-length ratio. Listed below are the dimensions of the fictitious Series-60 ship and of the ship form used in this development.

	Series-60	UM Model 1042
LBP	615'-0"	530'-0"
LWL	625'-4"	530'-0"*
Beam	82'-0"	83'-9"
Draft	32'-10"	30'-0"
Displ. (mld., L.T.)	28,400	28,400
Block Coeff.	.600	.747
Prismatic Coeff.	.614	.762
Midship Coeff.	.977	.980
$\Delta/(L/100)^3$	122	191
Length/Beam	7.50	6.33
Beam/Draft	2.50	2.79
1/2 angle of entrance	7°	25°

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*546'-4" with transom stern.

FOREBODY DESIGN

Inherent in the decisions determining the general shape of both the forebody and the afterbody is the amount of parallel midbody that is to be allowed. In this case, seeking to have the minimum amount of parallel midbody for reasons called, by Inui, a "condition for the 'waveless' state,"[2] a length of 20% of the ship's length had to be allowed. Also, the amount of parallel load waterline has been kept to a minimum.

It should be noted that most ships with displacement-length ratios above 170 usually have in excess of 40% parallel midbody. The lines of the forebody are given in Figures 1 and 2. (Complete, large scale line drawings accompany this report to MarAd.) Special attention was given to the hollowness of the waterlines, and as mentioned before; the "shoulder" in the forebody was eliminated as much as possible. The desirability of the hollowness was found in a detailed study of a group of unrelated full form ships, upon which bulbous bow experimentation was carried out at The University of Michigan. The conclusions of that study are given in Appendix A.

The design waterline (30'-0") was made as hollow as possible, but owing to the extreme fullness of the ship, it resulted in only a very slight hollow. In view of the anticipated addition of a bulbous bow, the lower waterlines were given noticeable hollowness. This allows easier fairing of the bulb into the forebody, and also helps establish the bulb as a separate body, with its own wave-generating properties, instead of merely being an addition to the forebody. This is discussed in greater detail in a later section, "Bulbous Bow Design." Nondimensional curves of section area and design waterline are given in Figures 3 and 4.

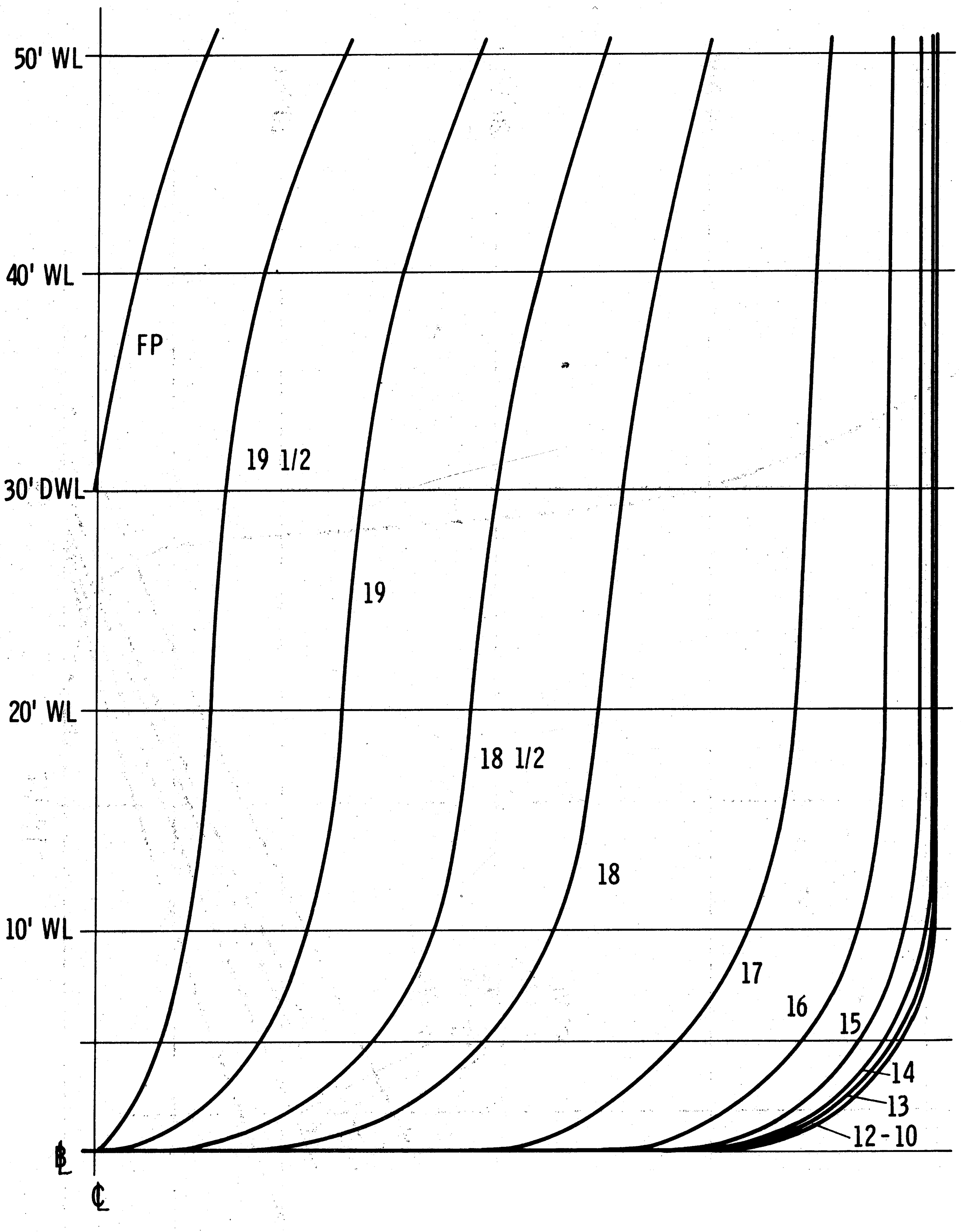

Figure 1. Conventional forebody.

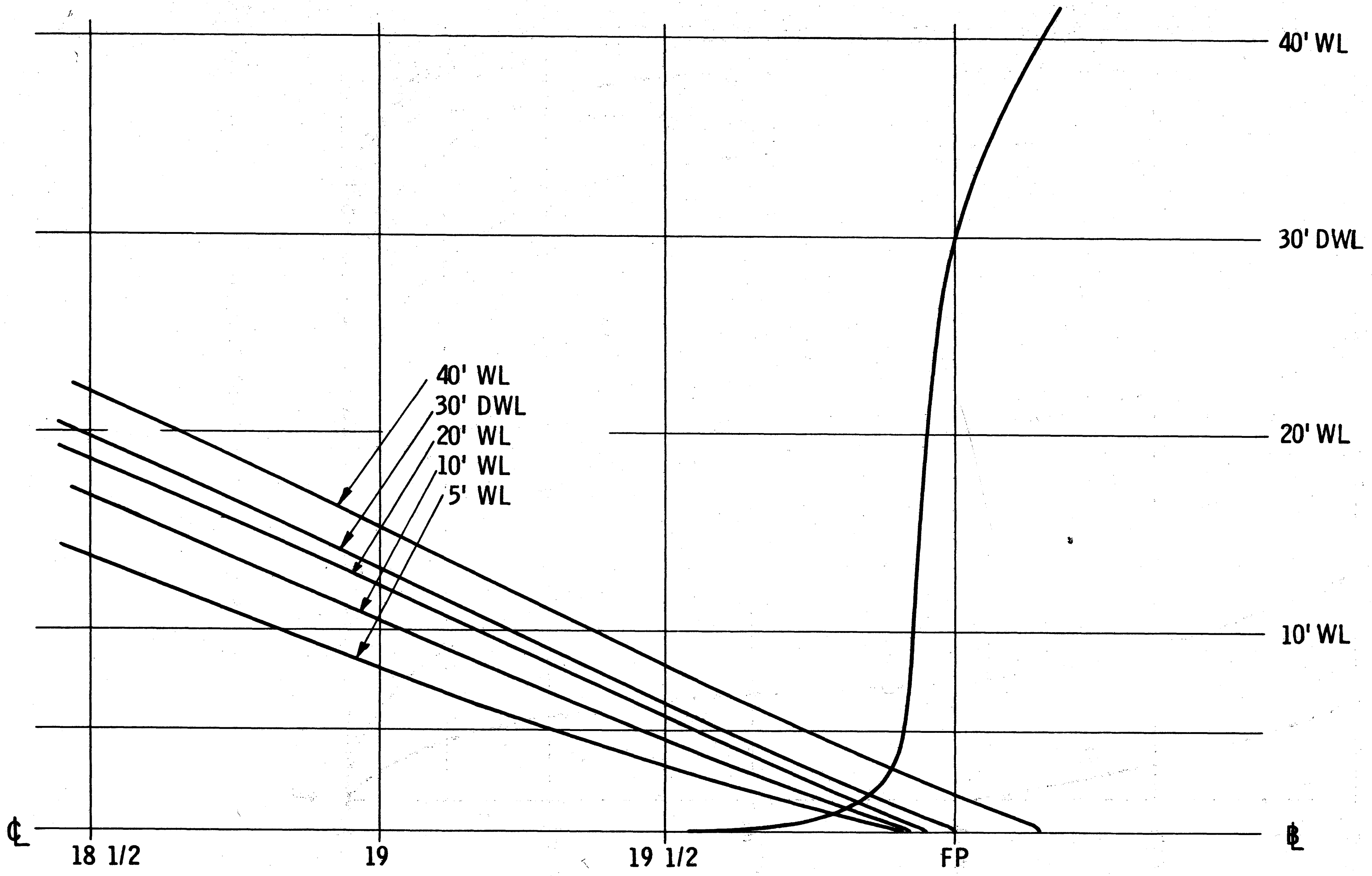

Figure 2. Conventional bow.

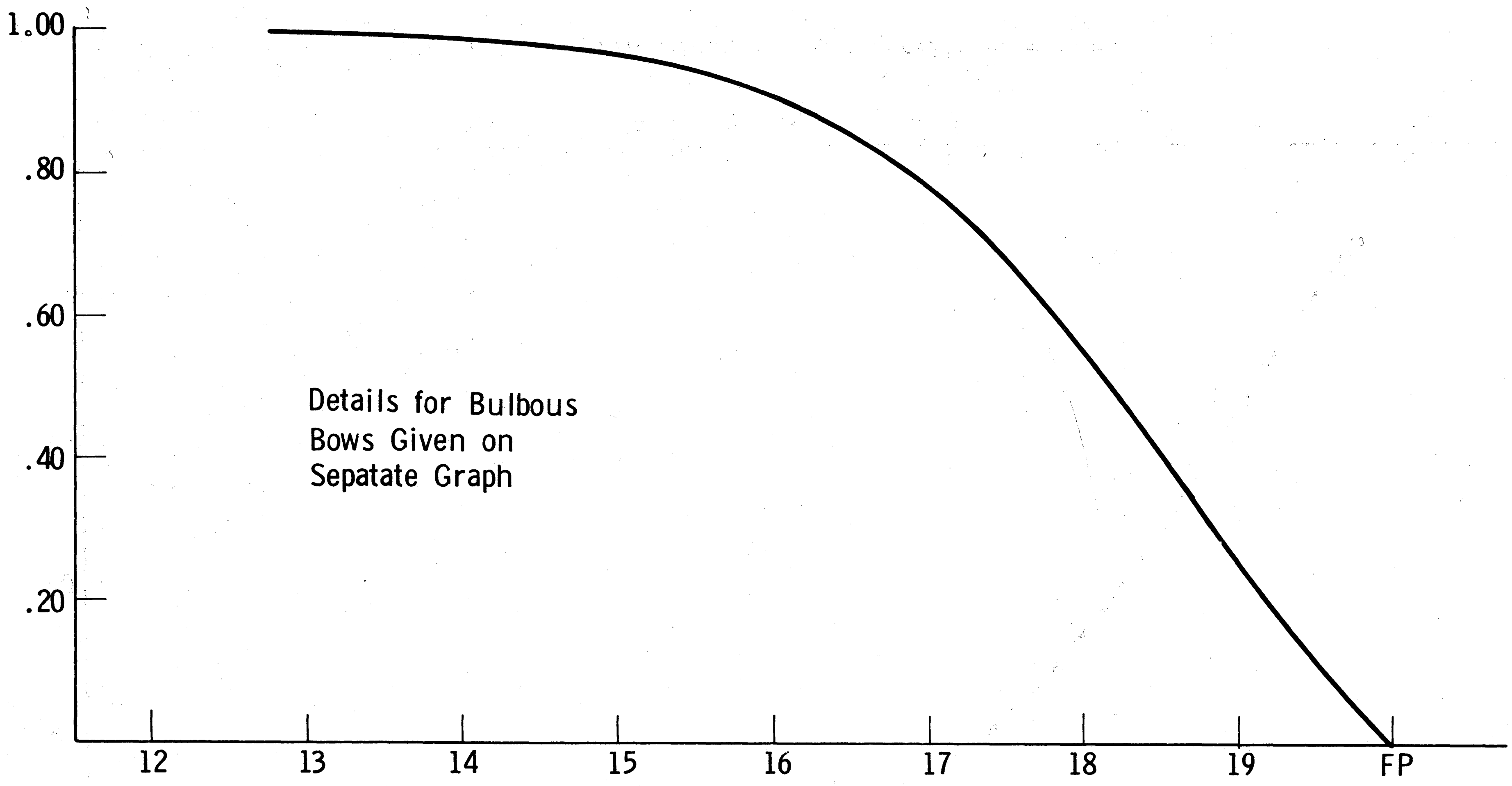

Figure 3. Nondimensional section area of forebody.

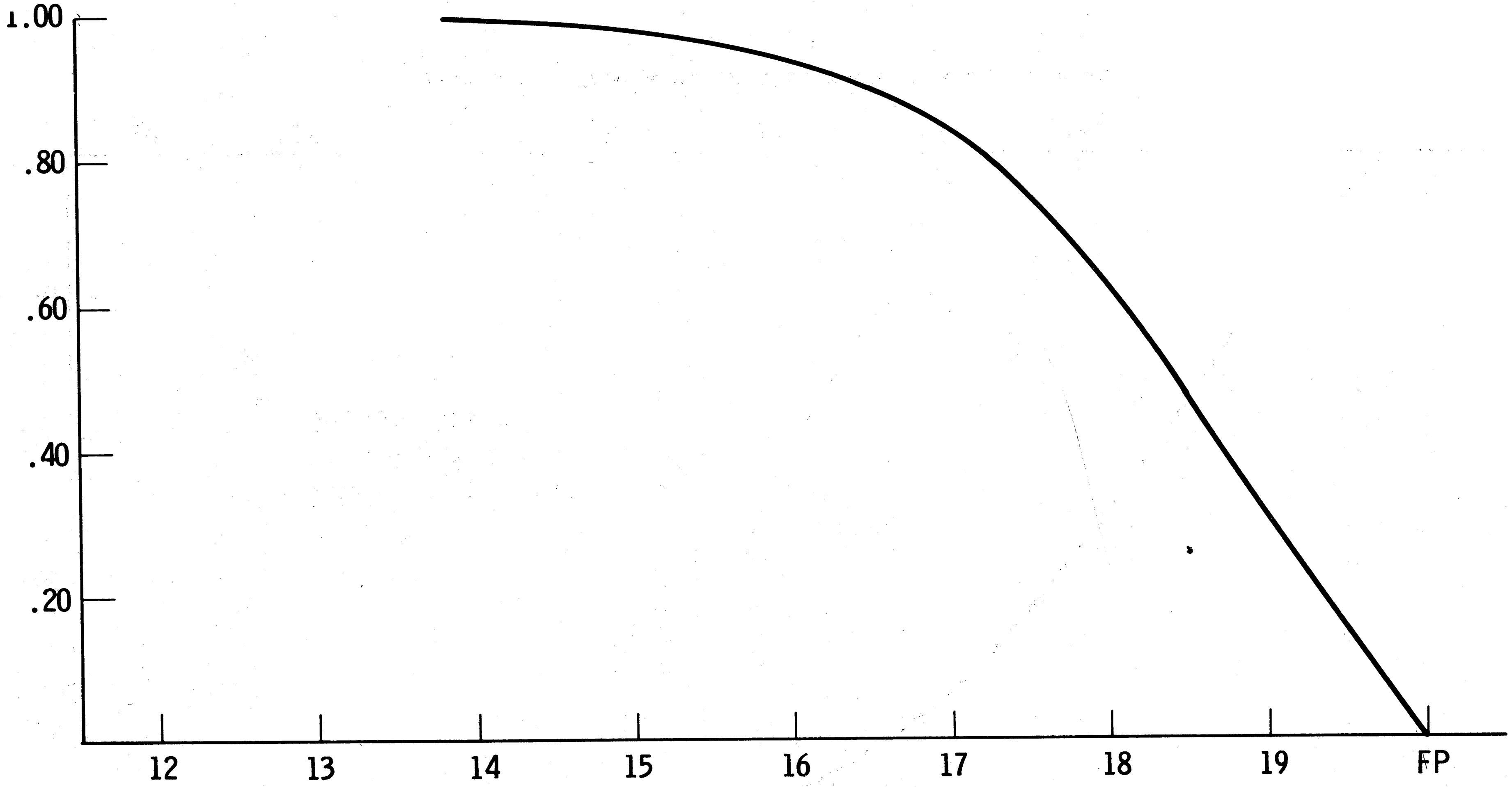

Figure 4. Nondimensional forebody design waterline.

AFTERBODY DESIGN

Three stern configurations were designed and resistance-tested; each with the same conventional (CV) bow. In view of the extreme fullness and beam, careful consideration had to be given to the flow into the propeller disc. Although this general form was expected to give a high overall wake fraction, it was realized that there would be a difference in wake distribution between the upper part of the propeller disc and the lower part of the disc that would be more severe than that found in ships with a finer hull form and a smaller beam/draft ratio. For propulsion, it is desirable to have the wake as uniform as possible. This was accomplished by opening up the area forward of the upper part of the propeller disc, resulting in a modified Hogner-type stern configuration (HG), shown in Figures 5 and 6.

In order to reasonably evaluate the performance of 1042-CV-HG (conventional forebody, Hogner-type afterbody), it was considered necessary to construct, for that same model, a second stern configuration with as many of the characteristic parameters identical to the original as possible, but of a more conventional configuration. This resulted in 1042-CV-CV (conventional forebody, conventional afterbody), for which the same forebody was used as in 1042-CV-HG. The lines for the conventional stern are shown in Figures 7 and 8.

The third stern configuration, a transom stern (TR) was considered because it would complete the variation as far as the flow characteristics are concerned. In the case of the Hogner-type stern, it is expected that the flow into the upper part of the propeller disc would have a large horizontal component (that is, inward toward the centerline on waterplanes). For the conventional stern, the flow into the propeller disc would have a smaller horizontal component, since it would be, at that point, primarily along a diagonal. For the transom stern, since the lower part of the stations in the vicinity forward of the propeller are closer to being vertical than in the conventional stern, it is expected that the flow into the propeller disc would have a negligible horizontal component. The lines for the transom stern are shown in Figures 9 and 10. Certain considerations given to the design of the transom stern should be noted.

At the design speed-length ratio of 0.85, for a 530' length, the speed of the ship is 19-1/2 knots. At that speed, it is expected that the flow from under the transom will not break clean (as it will on high-speed destroyers at, say, 30 knots). For this reason it was desirable to keep the depth of transom immersion low. But conflicting with such a design feature, the small transom immersion depth meant that the width of the transom at the DWL would be too small since certain minimum slopes of the stations are desirable from a seakeeping point of view. (The slope of the stations is most

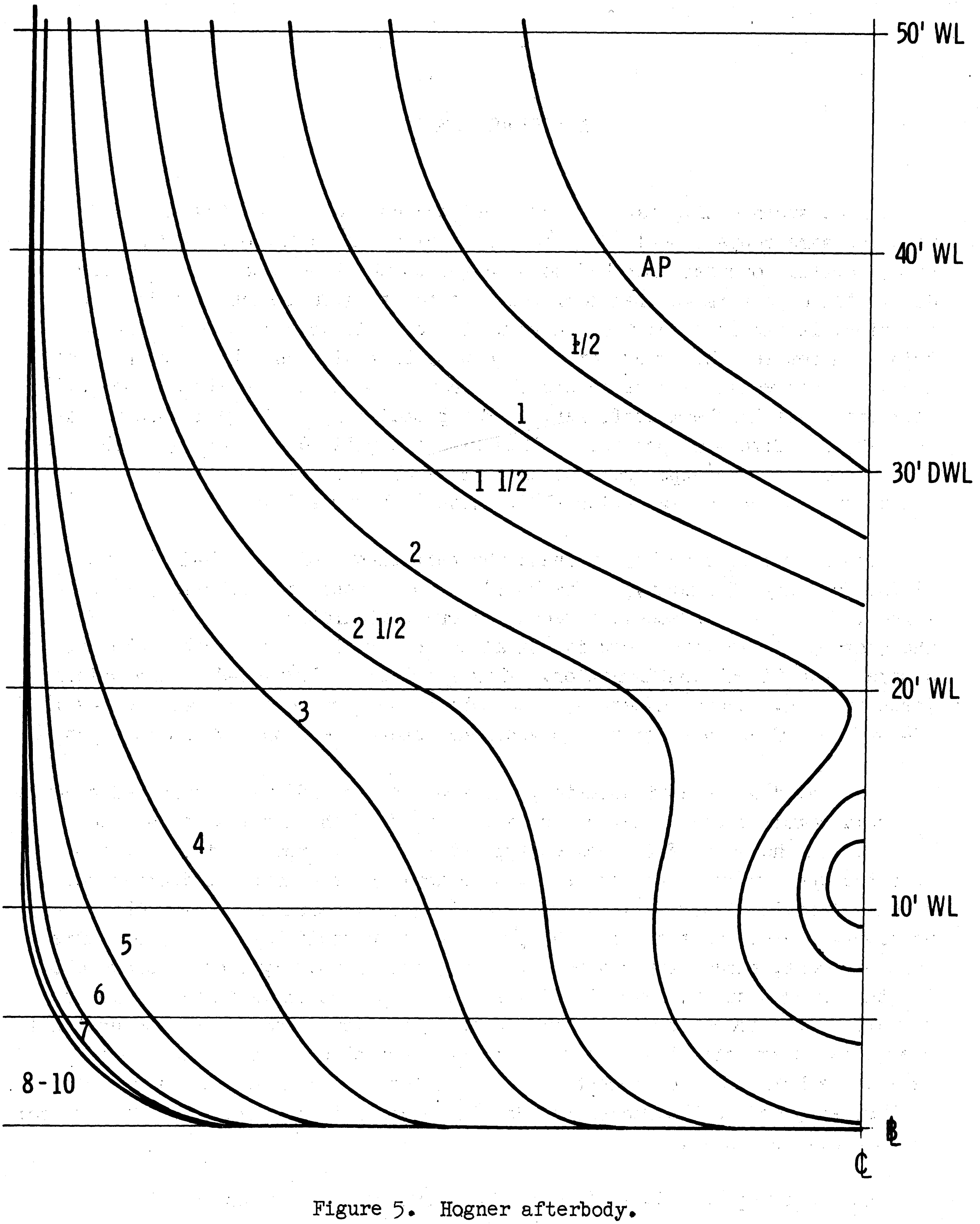

Figure 5. Hogner afterbody.

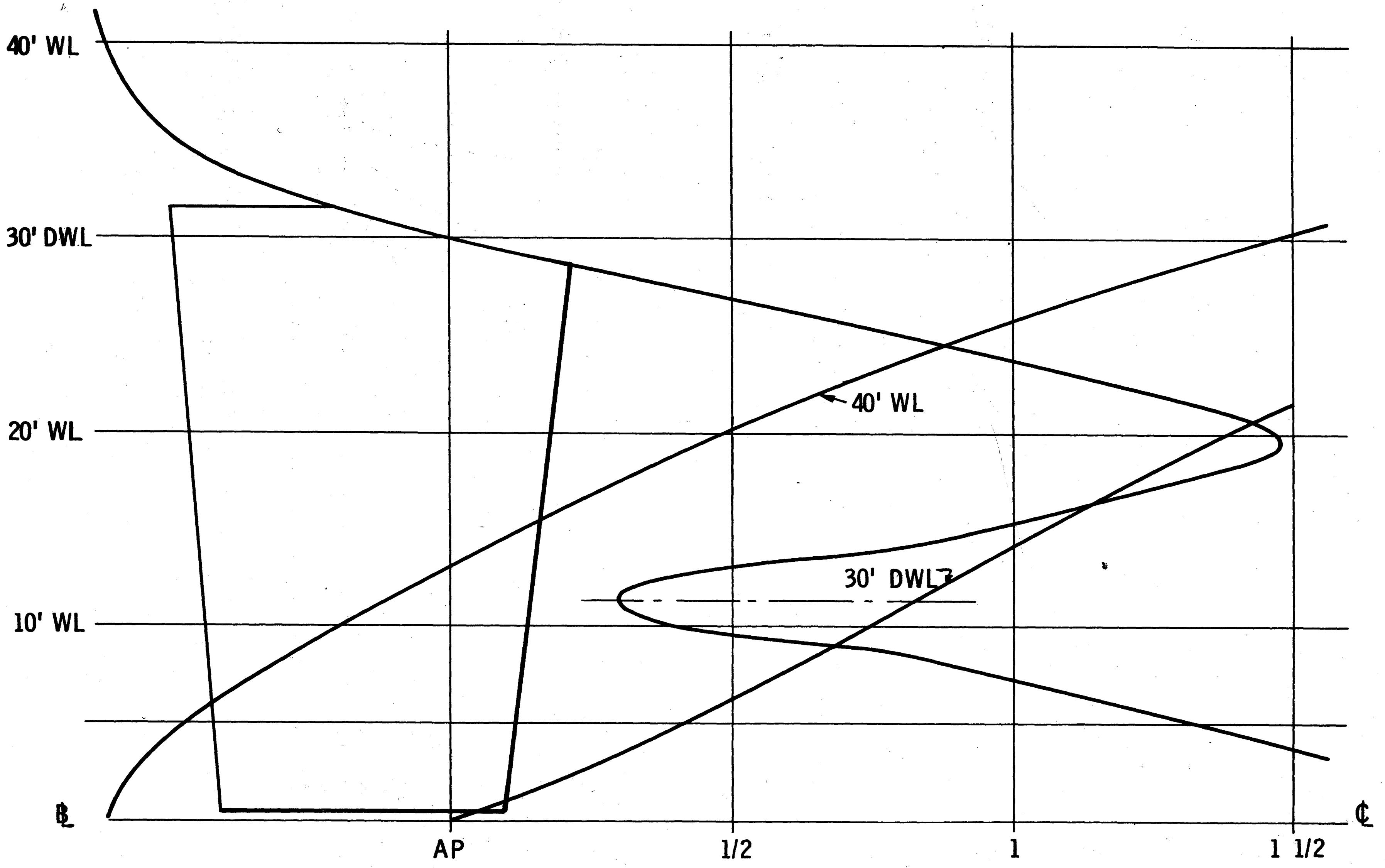

Figure 6. Hogner stern profile.

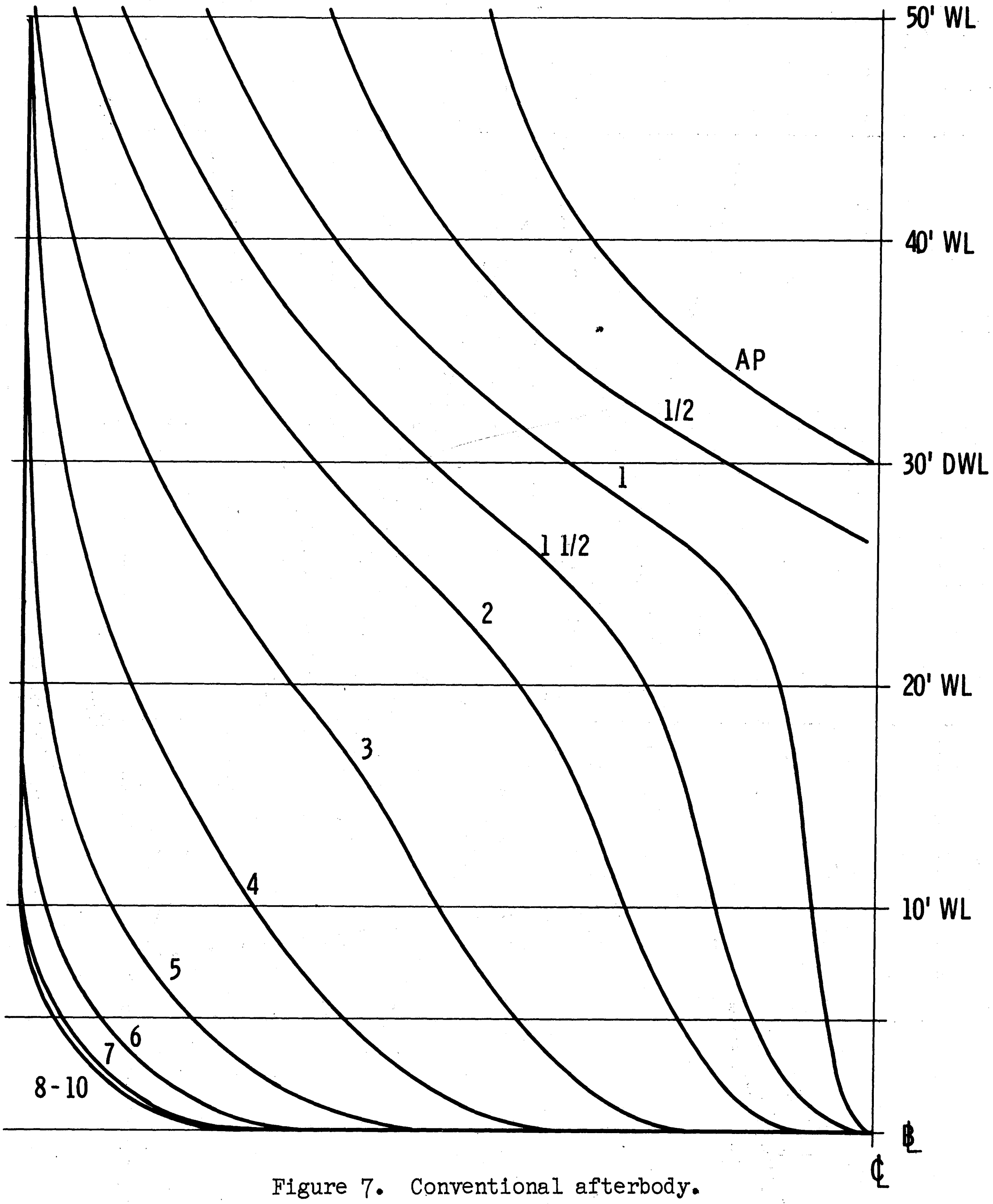

Figure 7. Conventional afterbody.

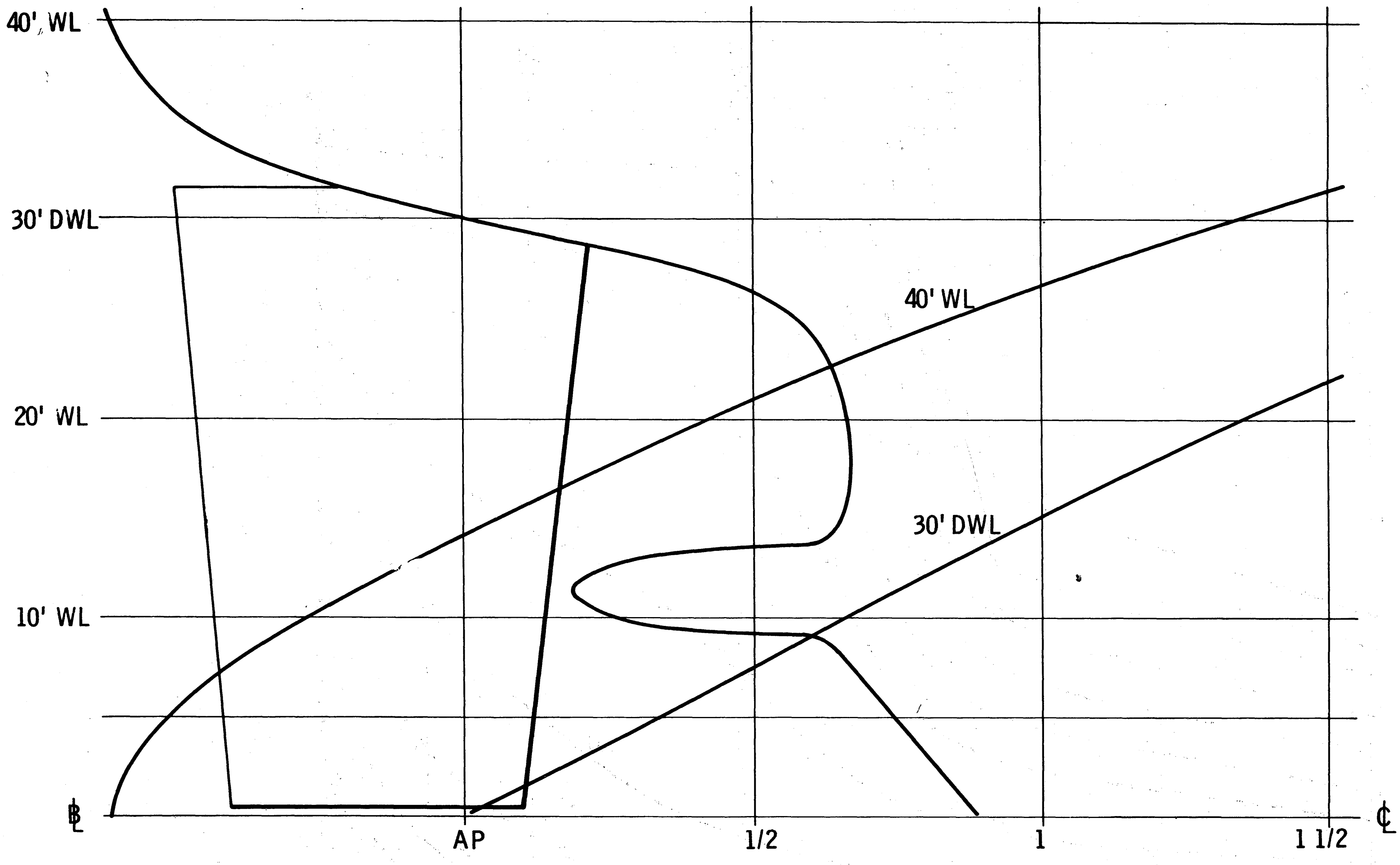

Figure 8. Conventional stern profile.

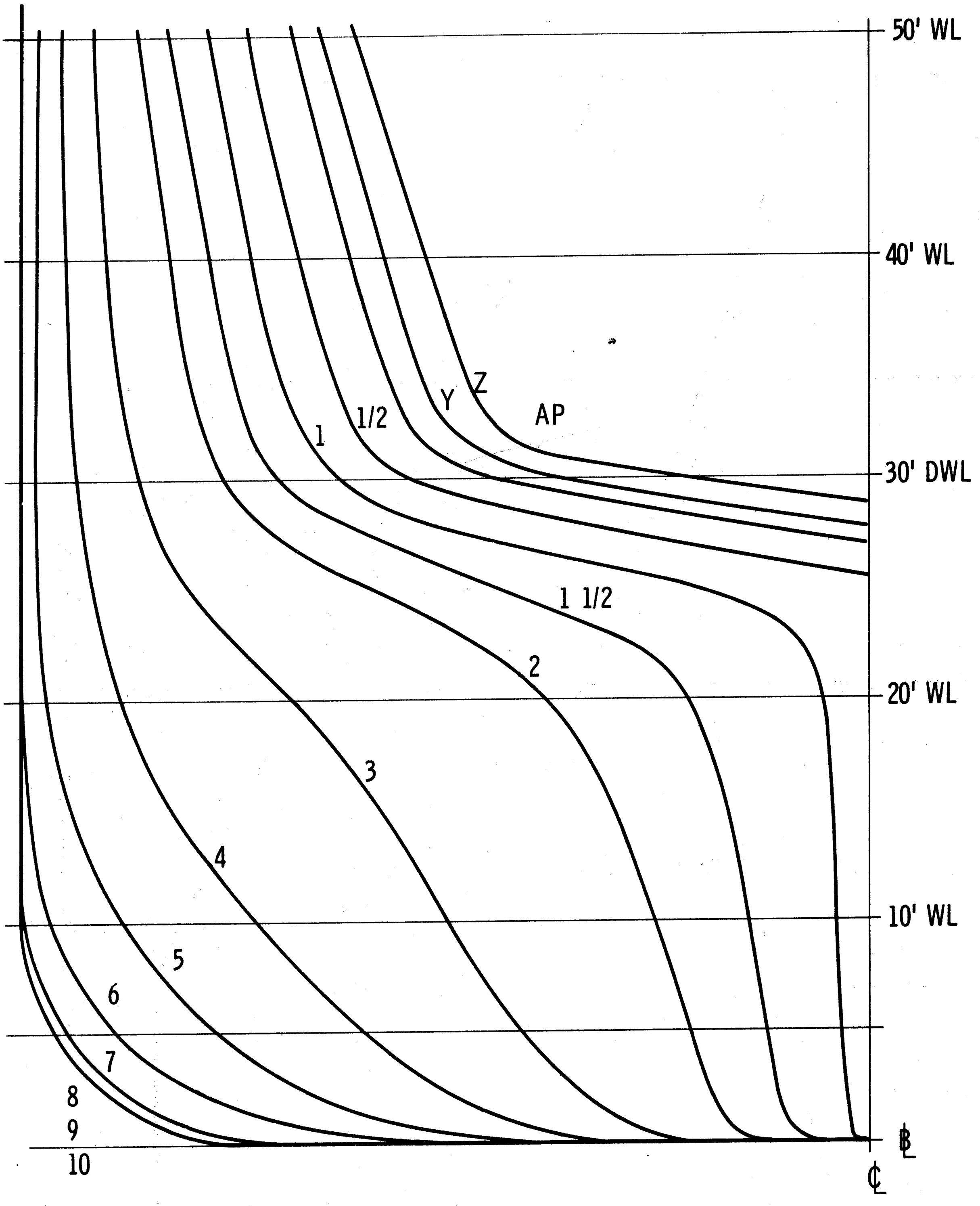

Figure 9. Transom afterbody.

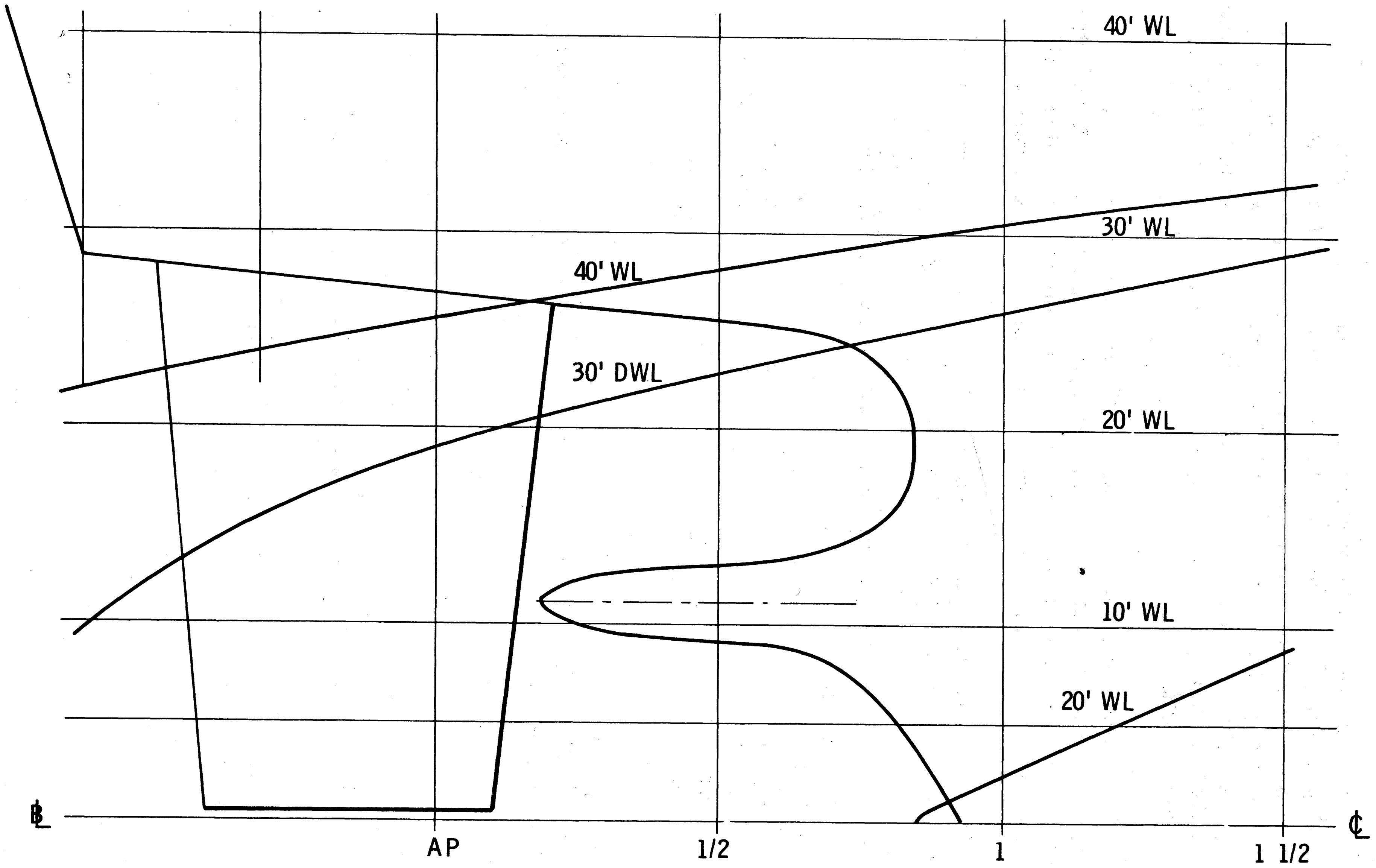

Figure 10. Transom stern profile.

appreciated when the ship is pitching in less-than-full load condition, at which time a broad, flat underside would experience slamming.) The advantage of the transom stern, i.e., the absence of any steep-sloping waterlines at the surface, is lost when the width of the transom is less than about 35% of the beam. In this case, at the 30' DWL, the width is only 22% of the beam. But as the ship squats at moderate speeds, the effective transom width at the surface is 45%.

A knuckle was considered but was not incorporated into the design, again due to the fact that the 19-1/2 knot speed would not be sufficient for the water to break clear of the hull on the underside of the knuckle; and flow around a knuckle would very likely be detrimental to performance.

The variations of parameters and coefficients among the three sterns when incorporated with the conventional forebody, are given below.

	1042-CV-HG	1042-CV-CV	1042-CV-TR
LCB/LBP (fwd.)	1.83%	1.73%	1.68%
L_X/LBP	.200	.215	.200
L_E/LBP	.400	.400	.400
L_R/LBP	.400	.385	.400
C_{P_E}	.747	.747	.747
C_{P_R}	.658	.644	.658
W.S. (sq. ft.)	60,950	60,310	62,010

The nondimensional section area curves for the three afterbodies are shown in Figure 11.

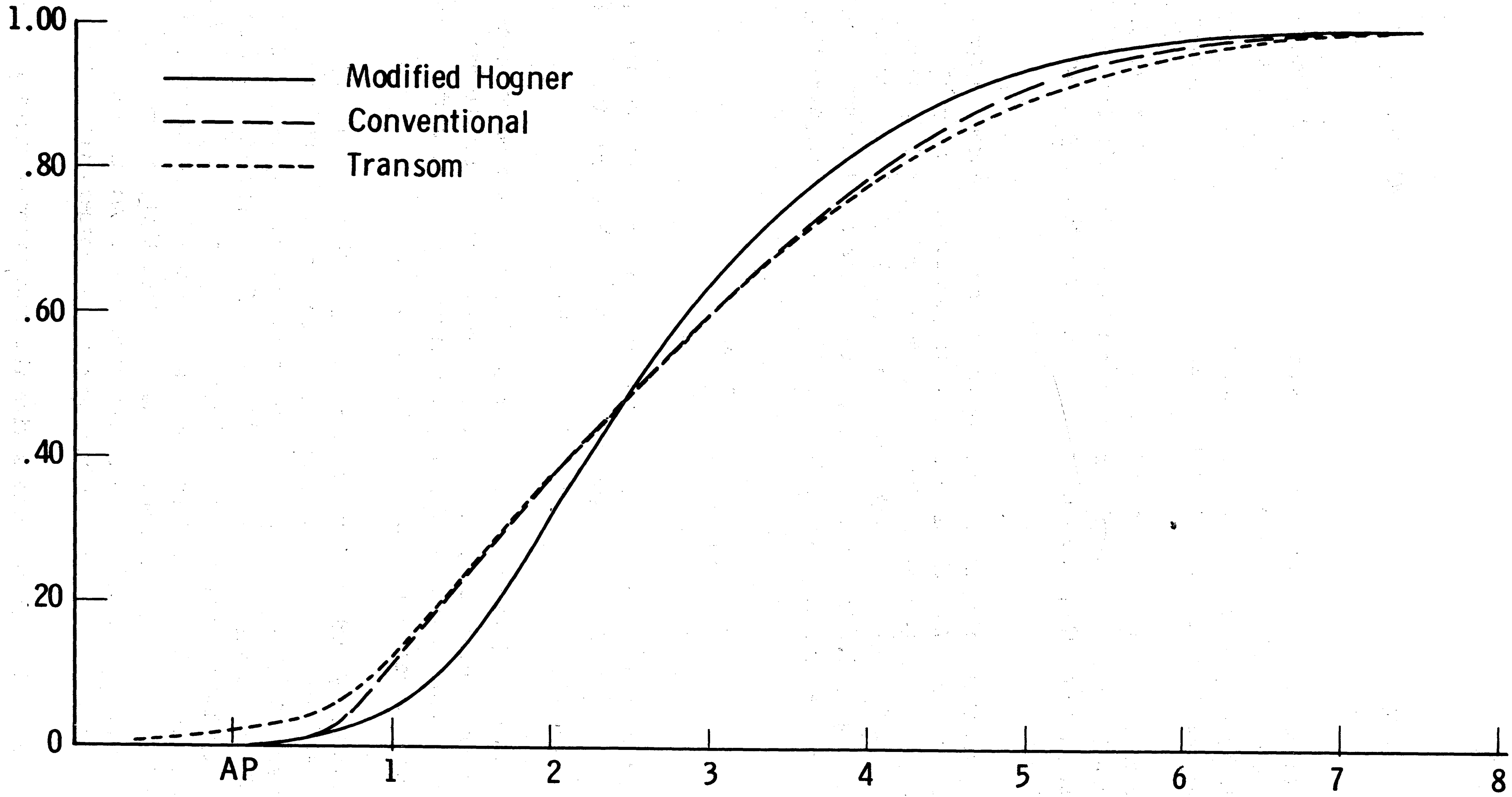

Figure 11. Nondimensional section areas for sterns.

RESULTS OF RESISTANCE TESTS ON STERN CONFIGURATIONS

Resistance tests were conducted on models consisting of the conventional bow and each of the sterns. The flow in the vicinity of the propeller disc was studied by the use of wool tufts attached to the surface or suspended small distances from the surface of the hull.

For each model, measurements of towing resistance were made at three standard conditions: 100% displacement with no trim; 80% displacement with 1% trim (i.e., 1% of LBP, trimmed by the stern); and 60% displacement with 2-1/2% trim.

Since, at this point in the development we are interested in comparing the models for resistance characteristics, the resistances have been reduced to nondimensional resistance coefficients. The usual coefficients are $C_T = R_T/(\rho/2\ SV^2)$ where R_T is taken to be the total resistance (or the residual resistance, R_R). But in this case, because the displacements are identical and the wetted surfaces are different, the tests were compared by:

$$C_T^{\circ} = R_T/\left(\frac{\rho}{2}\ V^2\ \nabla^{2/3}\right)$$

Using the values obtained for the conventional stern as the basis, the changes of C_T° and C_R° for each of the displacements are given in Figures 12, 13, and 14 (ΔC_F = .0004, Schoenherr friction).

Because this hull-form is being developed for optimum performance at the full displacement condition, the transom stern was chosen as the best from a resistance point of view. At $V/\sqrt{L} = 0.85$ it has a total resistance 4% less than the conventional stern, as shown in Figure 12.

The extremely high resistance of Hogner-type stern in all displacement conditions is primarily due to certain flow conditions observed during the flow studies. On that stern an early point of separation was observed that began at about station 1-3/4 and the 12' W.L. This separation, when further acted upon by the unusual flow in the vicinity of the "cone" of the stern, induced inboard-rotating vortices, the result of which was to cause severe down-flow in the propeller plane. It was realized that only major modifications to the stern configuration could correct the flow; and those modifications would remove the supposed advantages of the Hogner-type stern.

The flow studies on the models with the conventional stern and the transom stern showed, in the vicinity of the propeller disc and the stations immediately forward of it, that flow into the disc was characterized by some

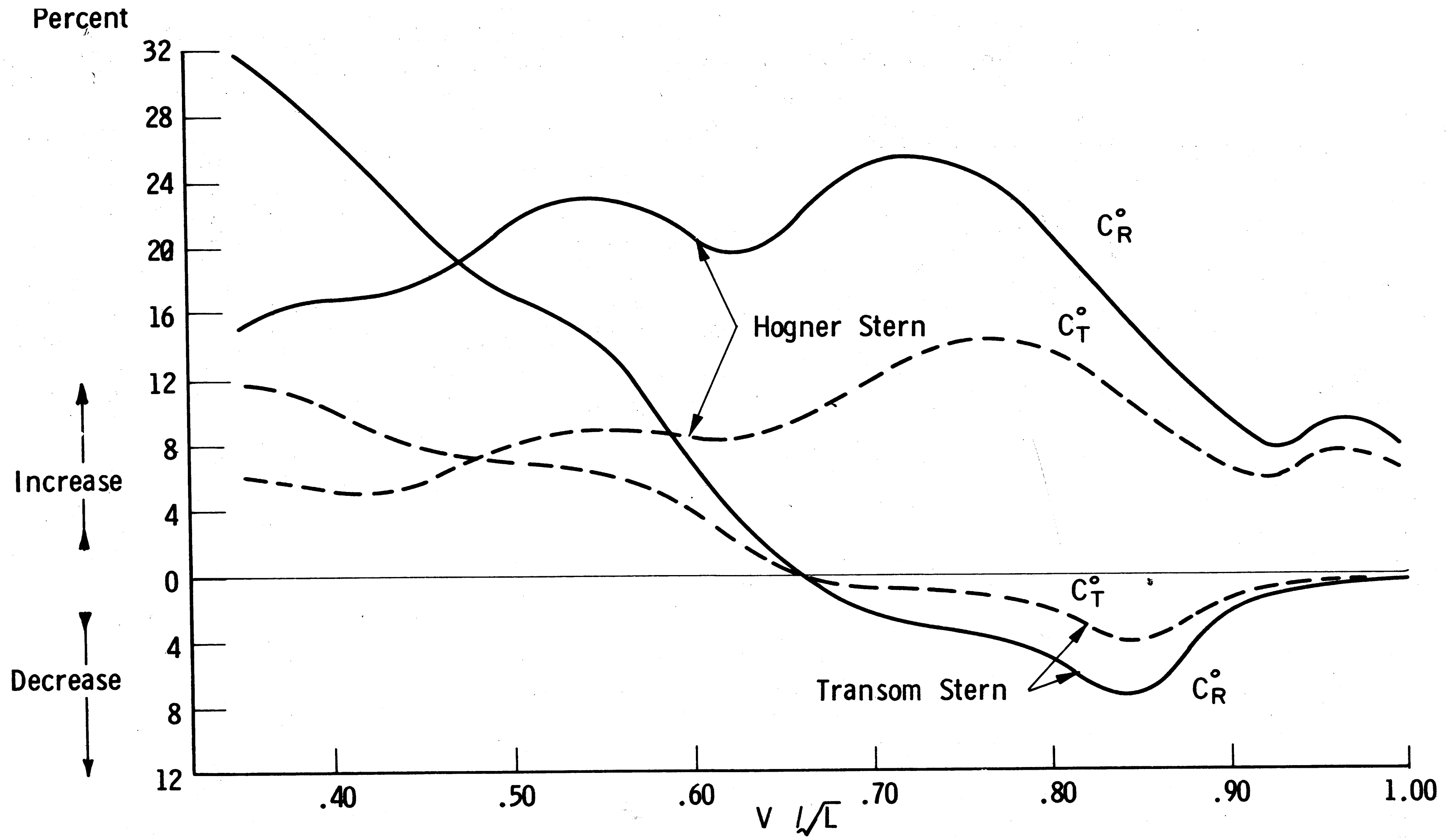

Figure 12. Change of C_R° and C_T° due to variation of stern configuration—100% displ.

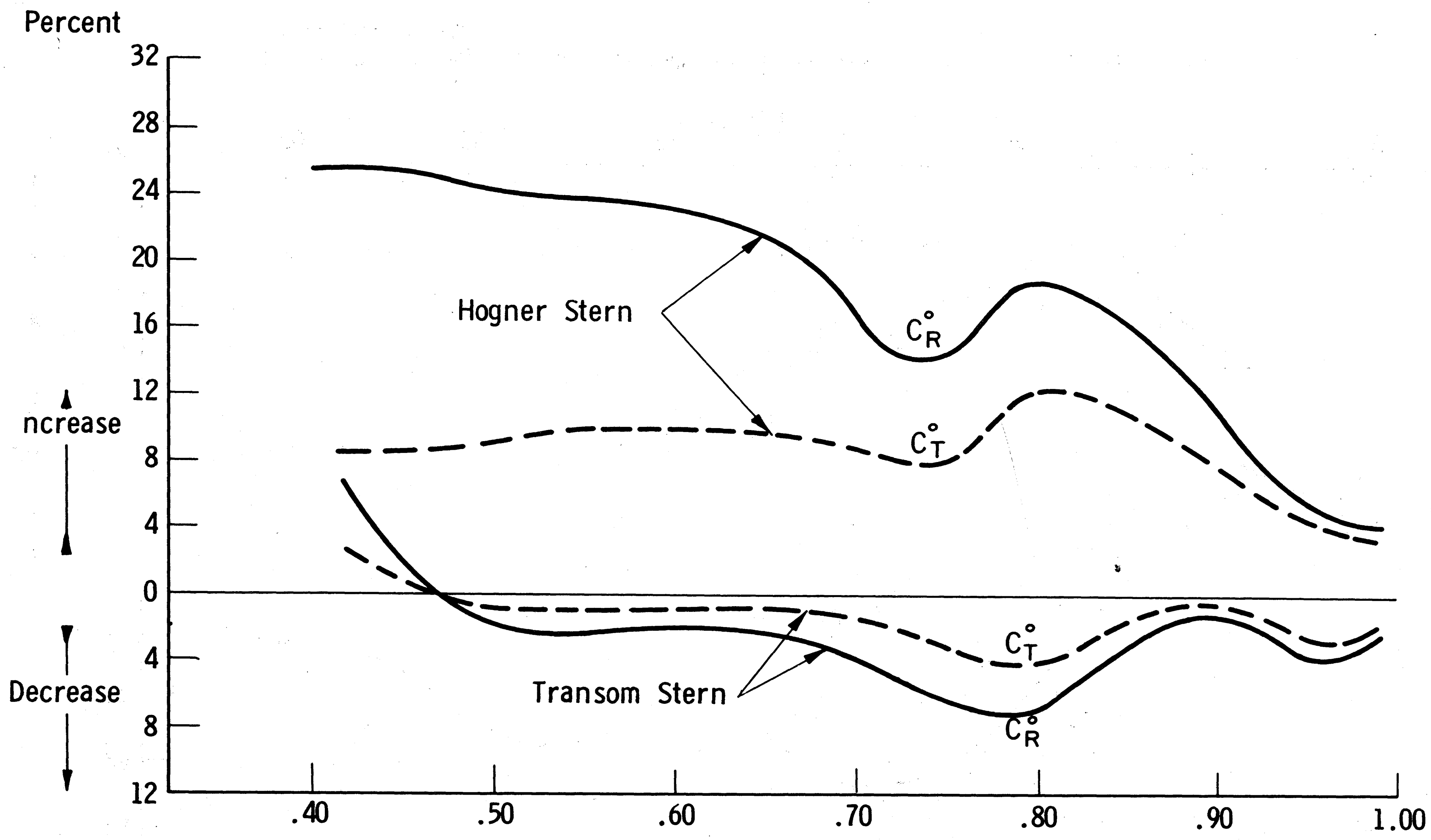

Figure 13. Change of $C_R°$ and $C_T°$ due to variation of stern configuration—80% displ.

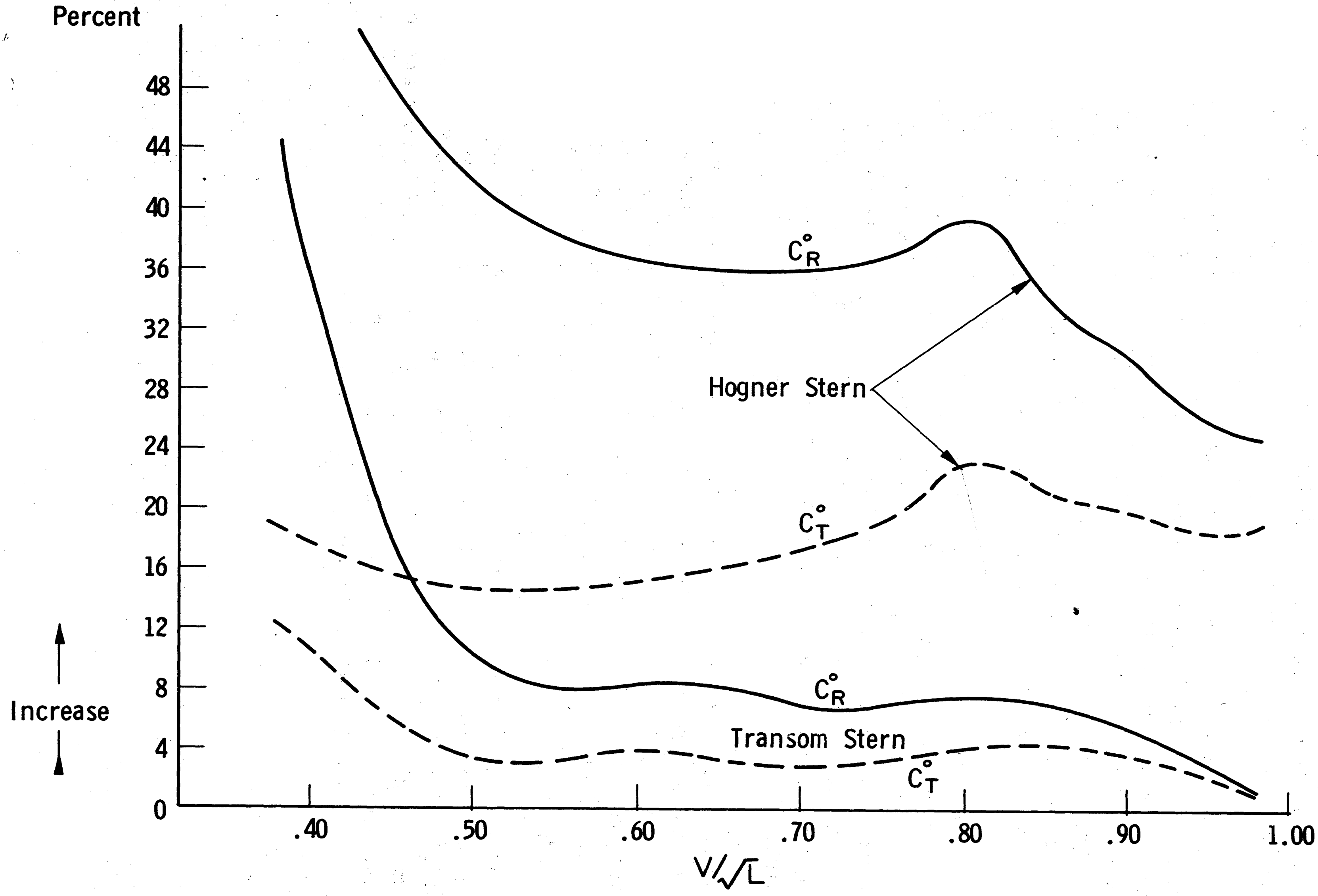

Figure 14. Change of C_R° and C_T° due to variation of stern configuration—60% displ.

down-flow in a thin sheet on the hull surface, but otherwise nearly-horizontal flow outside of that thin sheet (i.e., horizontal when viewed from the side of the model). In a small region immediately above the propeller disc, there was a distinct lack of streamlines.

The flow studies, as mentioned earlier, were conducted by using wool tufts pinned to the surface or suspended small distances above the hull surface; and these were visually recorded by 16mm movie cameras, situated so they viewed the underwater portions of the stern from a horizontal position. (The enlargement of a few frames of those films, that were to be published here, proved unsuccessful.) The complete films, as part of this study, have been forwarded to the Maritime Administration.

The flow study on the conventional stern was conducted both with and without a propeller turning at an appropriate rpm. On the transom stern, it was conducted without a propeller. In most areas, the tufts were not steady, but did oscillate over a small arc (about 10°). Above the propeller disc the tufts "fluttered" irregularly over a total arc of about 90°. On the lower trailing edge of the "skeg" there were large oscillations which are not at all unusual.

The irregular oscillatory motion above the propeller disc is not expected to effect the powering requirements of the ship, although it may prove to be a cause of vibration. Further minor modifications of the lines at the stern may reduce the unsteadiness to some degree. The effect of such modification on the unsteadiness of the wake should be investigated in future studies of this hull form.

The flow into the propeller discs of the conventional stern and of the transom stern appeared to be similar—and both are what would normally be considered quite satisfactory. The irregular oscillations mentioned above are common to single-screw ships. On the basis of measured resistance at the full displacement condition (Figure 12), the transom stern was chosen as for the hull used in the development of the bulbous bow.

BULBOUS BOW DESIGN

It is presently generally agreed that the most effective means of reducing a ship's total resistance is to minimize the wave-making resistance. According to their proponent's claims, many proposals have been made which would reduce the frictional resistance. But as none have yet been made applicable for ships, attempts to utilize these concepts have not been made. Special bulbous sterns have been proposed and in some instances proven slightly effective.[3] But their applicability has not yet been sufficiently developed.

A most readily applicable method of minimizing the wave-making resistance, which is sufficiently well developed is that of adopting a bulbous bow. Inui, Kajitani, and Kasahara,[4] and Pien and Moore[5] have begun investigating the second generation of low-resistance nonbulbous hull forms. But these have not yet been made sufficiently practicable.

Prior to the actual design of the bulb, extensive investigations into the literature and into the results of other designs were carried out. A group of unrelated, relatively full-form ships, upon which bulbous bow experimentation had been carried out at The University of Michigan, was studied in detail. The conclusions of this study are given in Appendix A.

The general result of these investigations was to learn that the bulb must be considered an integral part of the hull form, and cannot be utilized to the greatest extent if designed as an appendage, or considered as an afterthought. As cited in an earlier section, the bulb-less (conventional) forebody was designed with the forthcoming addition of a bulb in mind.

The problem of utilizing a bulbous bow design for this model proved unique and exceptional in many respects. Most ships having displacement-length ratios greater than 160 are also slow or moderate-speed ships. This model, having a displacement-length ratio of 191, is a high-speed ship. Also, ships operating at comparable speeds are usually found to have ratios less than 130. Thus, any bulbous bow information gleaned from existing ships or models must be used carefully because none of it is directly applicable, as we have noted. If the speeds are comparable to this model, then the hull forms are too fine. On the other hand, if the hull forms are comparable, then the design speeds are too low. Of course this situation merely substantiates the necessity for the model tests conducted in this development.

One basic difference between bulbs designed for finer, moderate-speed ships and those designed for fuller, slow-speed ships is that, in the former case, total resistance reduction is achieved primarily through a decrease of wave-making resistance, and in the latter, through a reduction of underside eddying and general reduction of unsteady flow around the forebody as well.[1]

It is generally agreed that choosing the relative size of some representative cross-sectional area of the bulb is the first step to a good bulb design. Usually either the area at the fore-perpendicular or at the longitudinal center of the bulb is nondimensionalized by dividing it by the midship-section area. (Since most midship-section coefficients are comparable, this is considered a consistant method.)

In this case, the area at the longitudinal center of the bulb is considered to be the area which will most effectively govern the resistance characteristics of the ship with the bulbous bow. The longitudinal center of the bulb is some distance aft of the foremost protrusion of the bulb, taken here to be the mean radius of curvature of the forward part of the bulb.

This longitudinal location should not, however, be considered the "effective" center of the bulb. The "effective wave-generating center" of the bulb can be considered as the fore-aft origin which enables us to view the bulb wave mathematically almost completely as a negative cosine wave. A similar "effective wave-generating center" of the ship enables us to regard the ship wave almost completely as a positive cosine wave. Maximum cancellation of waves occurs when the effective centers coincide and the wave amplitudes are equal. Because the waves are not exactly of cosine form and because the viscous interaction of the bulb and the ship upon each other's wave generating characteristics is presently not well known, it is extremely difficult to define the effective centers.

Of all the important parameters associated with the bulb, the desired longitudinal center is the most evasive. Although there may be consistency between similar ships in other parameters, generally the longitudinal center of the best bulb for each has been found to be different. For this reason it would have been informative to test several bulbs that were identical in most respects except for the longitudinal center.* The limitations of funds and time, and the desire to look at the variation of a different parameter (as discussed later) prevented tests of this nature from being conducted.

The problem of determining the proper longitudinal center arises from the difficulty encountered in defining the effective wave-generating center of both the bulb and the ship. Since almost all the waves generated by the ship arise at the forebody, it is best to consider the problem of matching the bulb to the forebody, and not to the entire ship. (This does not rule out the possibility that the bulb effects the wake of the ship at the stern; but as investigations into that matter have not been conducted, it would only be speculation to discuss it at this time.) The effective center of the bulb,

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*Recent tests at The University of Michigan Ship Hydrodynamics Laboratory of a Series 60 (0.80 block) form indicate that some range of longitudinal center is permitted with little effect on resistance.

which is understood to include the fairing into the bulb-less hull, will depend on the shape of the main body of the bulb, and of the fairing. The effective wave-generating center of the forebody will depend on the relative fullness of waterlines near the stem, the shape of the waterlines in the forebody, and the rake of the stem. In view of this, it is quite easy to understand why the longitudinal center of the bulb, which is of prime importance to the effective center of the bulb, is very difficult, if not impossible, to define in simple terms.

In order to investigate, to a small extent, the relative importance of the fairing on the effective center of the bulb, a bulbous bow was tested with two fairings, one considerably finer than the other. The longitudinal center and the bulb shape forward of that center were kept the same.

There are several possible shapes that the section through a bulb may take. The three most distinctive ones are shown in Figure 15, which illustrates the relative widths necessary for the same section areas. Not much is known about the effects on resistance that the various shapes of bulb sections will have. Thus, the circular shaped bulb designed for model 1042 was chosen for certain other considerations.

The vertically-oriented elliptical shape was rejected on the basis that the shape of the leading edge of the waterplane would be too blunt in lightly ballasted conditions,* and also the relative submerged bulb volume in ballast condition would be too small. Although this hull form is presently being developed for one load condition (full displacement) it was decided that consideration for other loading conditions could be taken into account if it resulted in no sacrifice of performance at that load condition.

The best solution to the problems that might be encountered with the elliptical shape would be the use of the "tear-drop" shape, which would preserve suitable waterplane shapes in ballasted condition and also keep the submerged bulb volume a maximum in all conditions. But if the required section area is large, since the width of that bulb would be greater than either the elliptical or circular shaped bulbs, the extra width of the bulb would prevent the smooth downward flow of water, as illustrated on the left side of Figure 16. (This direction of flow was noted on subsequent flow studies using wool tufts.) The interference of the flow from the side of the hull to the bottom in the region of the wide bulb could be eliminated if we accepted a narrowing of the waterline, as shown by the solid line in the right part of Figure 16. This shape might lead to a point of separation, however, somewhere just aft of the FP (as illustrated) because of an adverse pressure gradient.

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*As we shall see later, it is no longer believed that bluntness of the waterplane is a valid criterion for rejecting a bulb shape. As a matter of fact, it may prove to be desirable. But at the time of the first bulb design for model 1042, this insight had not been gained.

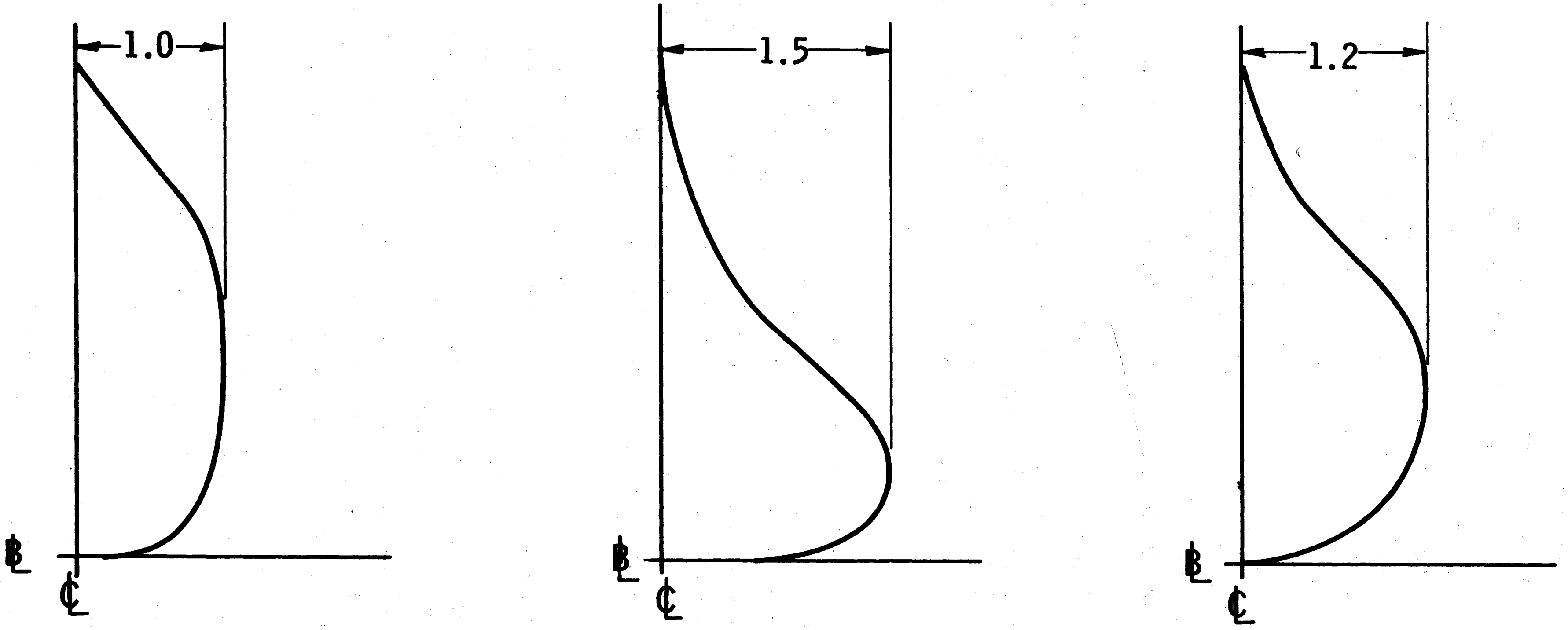

Figure 15. Typical variations of bulbous-bow sections.

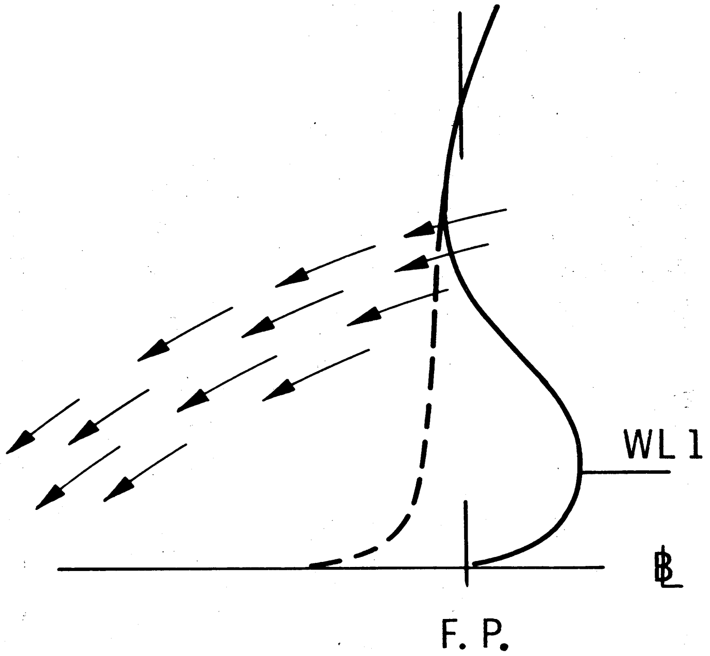

Figure 16a. Typical flow in vicinity of bulbous bow.

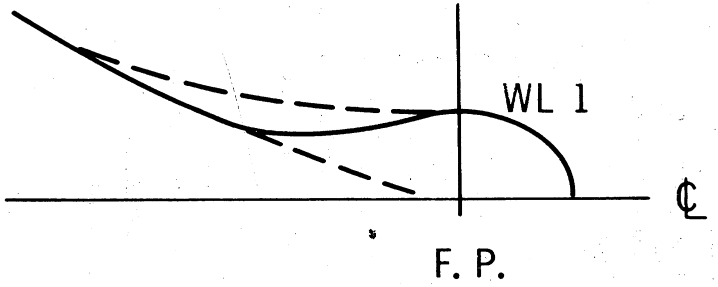

Figure 16b. Variations of waterline through bulbous bow.

The best solution appeared to be the circular section which preserves suitable waterplanes at the surface for shallower drafts than the elliptical section, and which permits easier downward flow aft of the bulb than the tear-drop section would allow.

The profile of the bulb is, by itself, not considered important in the design. However, for a given section shape and for a required bulb volume the profile will almost be completely determined. The profile could be cut shorter if part of the bulb volume is added to the fairing behind the main body of the bulb. But it is thought that such a volume distribution would not be as effective for wave generation as at the forward part of the bulb. Special attention should be given to the section on Bulb Volume in Appendix A.

The designed bulb is shown by the solid lines in Figures 17 and 18. The dotted lines in Figure 17 show the corresponding stations for the alteration of the fairing mentioned previously. Deadrise at the forward part of the bulb has been included to prevent excessive slamming in heavy seas. The section area curves for the bulb-less bow and for the bulbous bow, with both fairings, are shown in Figure 19.

The characteristics of the bulb with the fuller fairing are given below:

Bulb volume:	0.82% of nonbulbous volume
Area at long. center:	11.4% of midship area
Longitudinal center:	0.5% LBP forward of FP
Mean radius of bulb:	1.66% LBP
Protrusion of bulb:	2.16% LBP forward of FP
Bulb depth below DWL:	66.9% of draft

The bulb with the finer fairing had the same characteristics except for its bulb volume: 0.71% of nonbulbous ship volume.

In this development the only change has been the addition of the bulb and fairing. This results in a slight change of displacement, and so comparisons of resistance are made on a semi-dimensionless basis: ship resistance in pounds divided by ship displacement in tons.

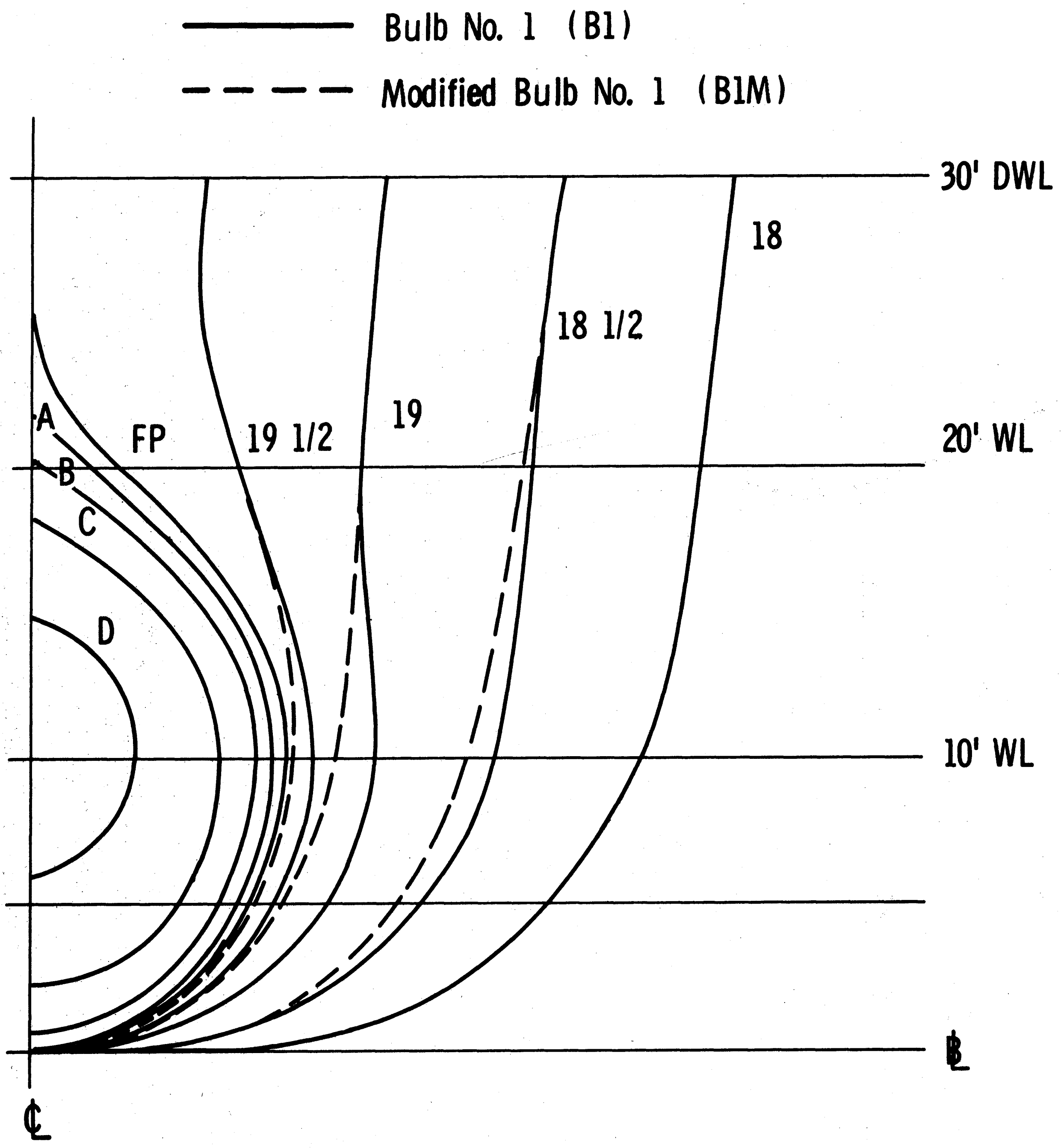

Figure 17. Sections of bulbous bows.

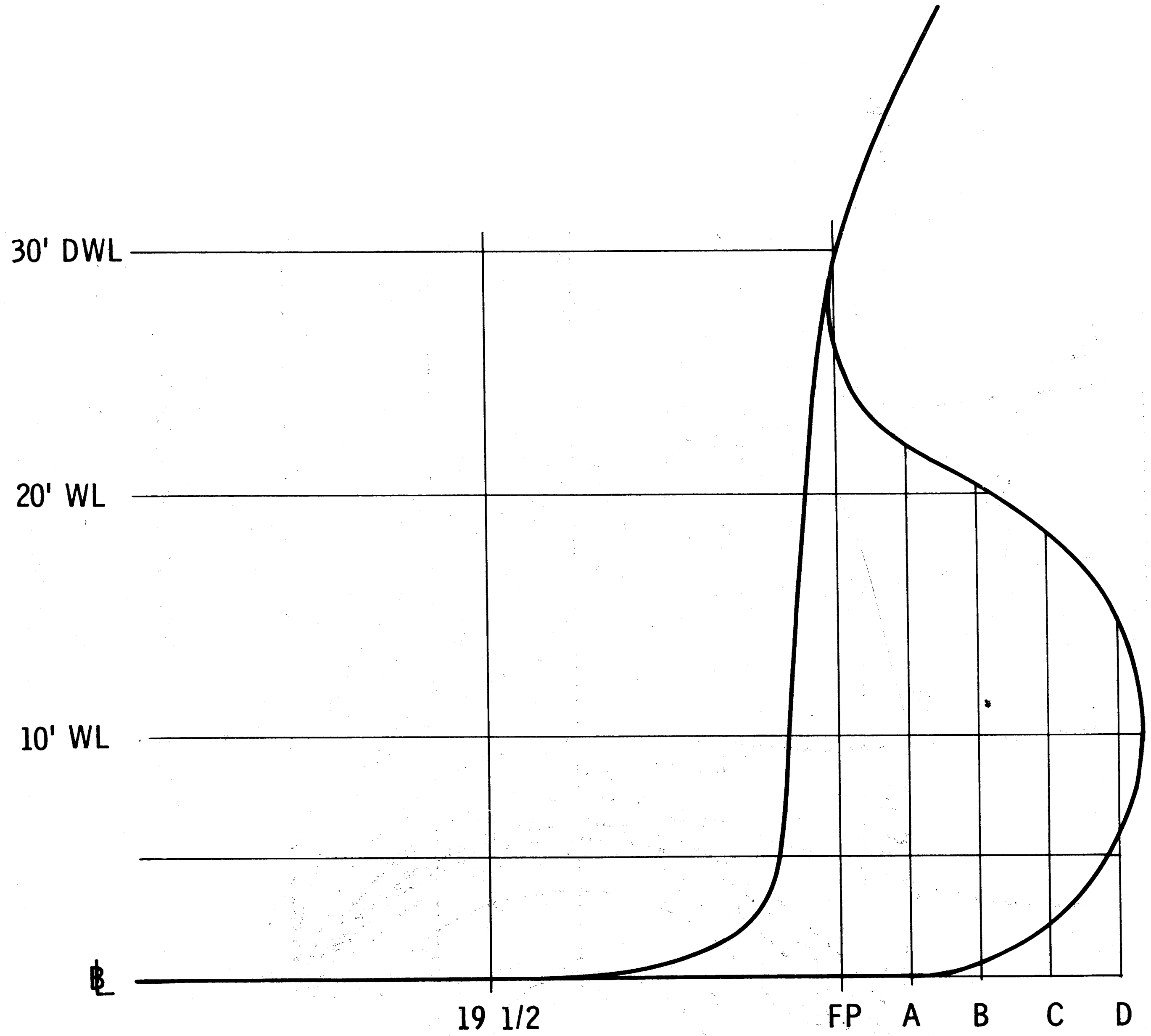

Figure 18. Profile of bulbous and conventional bows.

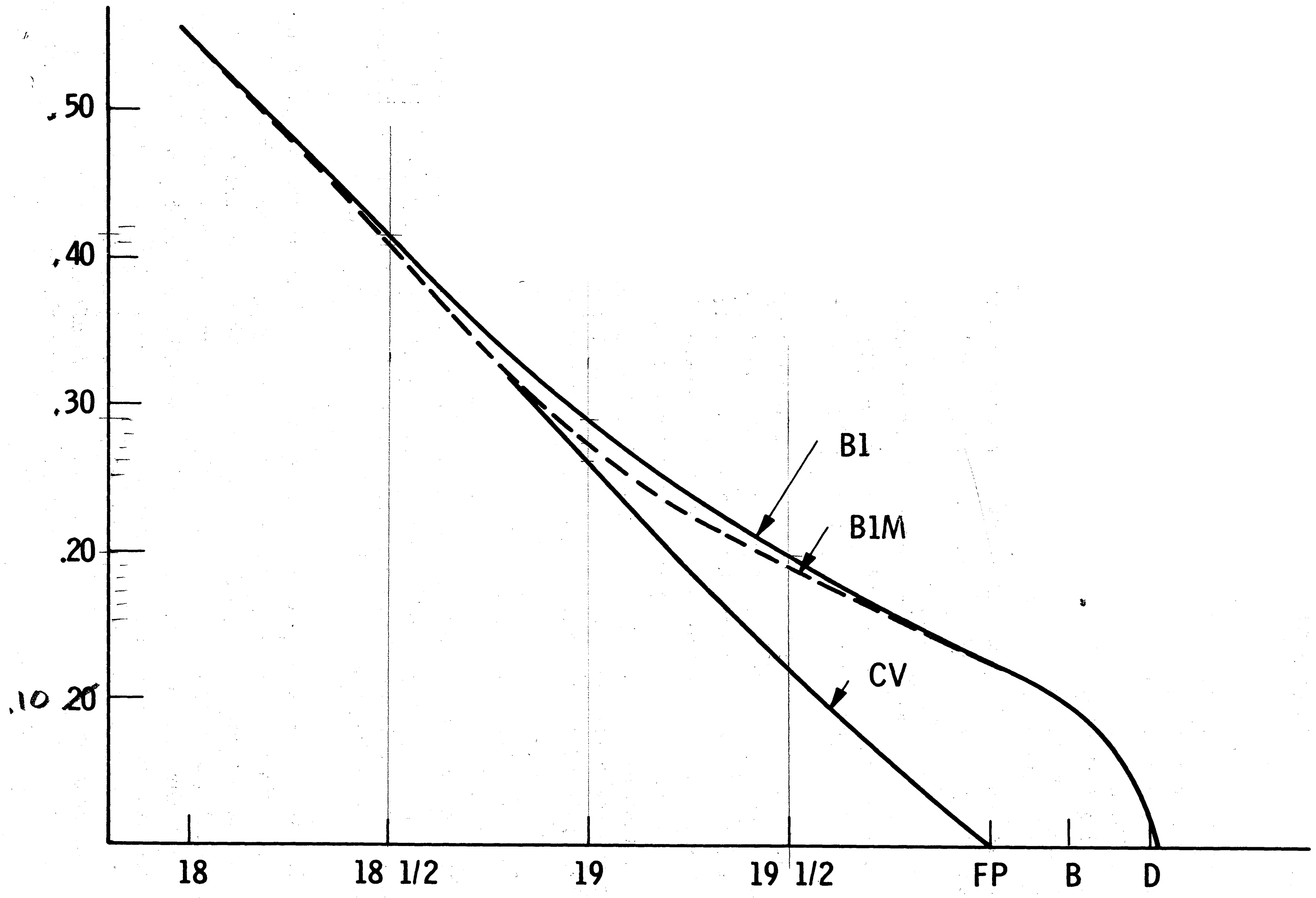

Figure 19. Nondimensional section areas of bulbous bows.

RESULTS OF RESISTANCE TESTS ON BULBOUS BOWS

The model with the first bulbous bow and the transom stern (1042-BI-TR) was tested over the same speed range as was done for the bulb-less hull with the transom stern. The resistance of each on the basis of R/Δ is shown in Figures 20, 21, and 22 for the 100% displacement (no trim), 80% displacement (1% trim) and the 60% displacement (2-1/2% trim) conditions, respectively. Curves for both R_T and R_R are shown. Figure 23, for the 100% displacement (3% trim) condition will be discussed later, but is inserted with the others for convenience. The comparisons by decreases or increases of resistance per ton are shown in Figure 24 (R_T) and Figure 25 (R_R).

As can be noted from the graphs, at a speed-length ratio of 0.85 (19-1/2 knots), decreases in resistance for the 100%, 80%, and 60% displacement conditions are, respectively, 8%, 13%, and 22%.

Later, similar tests were conducted using the modified bulb with the transom stern (1042-BIM-TR). The results of those tests on the same bulb with a finer fairing are not shown because they would only confuse the graphs. But it should be noted here the comparison between BI and BIM showed that the total resistance of BIM was greater than that of BI for some speeds, and less for other speeds. The curves of resistance crossed several times, and at no point was the variation greater than 2%. For that reason, no further work has been carried out in investigating the effect of changing the fairing.

The changes of C_R and C_T were also calculated, and are shown on Figures 26, 27, and 28. Whether such a comparison is valid is questionable, since both displacement and wetted surface are different. Thus resistance coefficients based on wetted surfaces (C_T) or on displacement ($C_T{}^\circ$)* will not simultaneously eliminate both differences. This could be eliminated if both models (with and without bulb) had exactly the same displacement.

It now becomes a matter of determining how the performance of the model could be further improved by altering the bulb configurations. In order to investigate the correctness of the longitudinal position of the bulb, the wave profiles along the hull for the forward quarter of the length were recorded for the hull both with and without the bulb. The wave profiles for one speed at two displacement conditions are given in Figures 29 and 30. From the graph for the 100% displacement condition (Figure 29) we can see that the trough of the bulb-less hull wave occurs aft of the crest of the effective bulb wave. Therefore, any alteration in the longitudinal direction

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*$C_T{}^\circ$ is defined in the section on Resistance Tests on Stern Configurations.

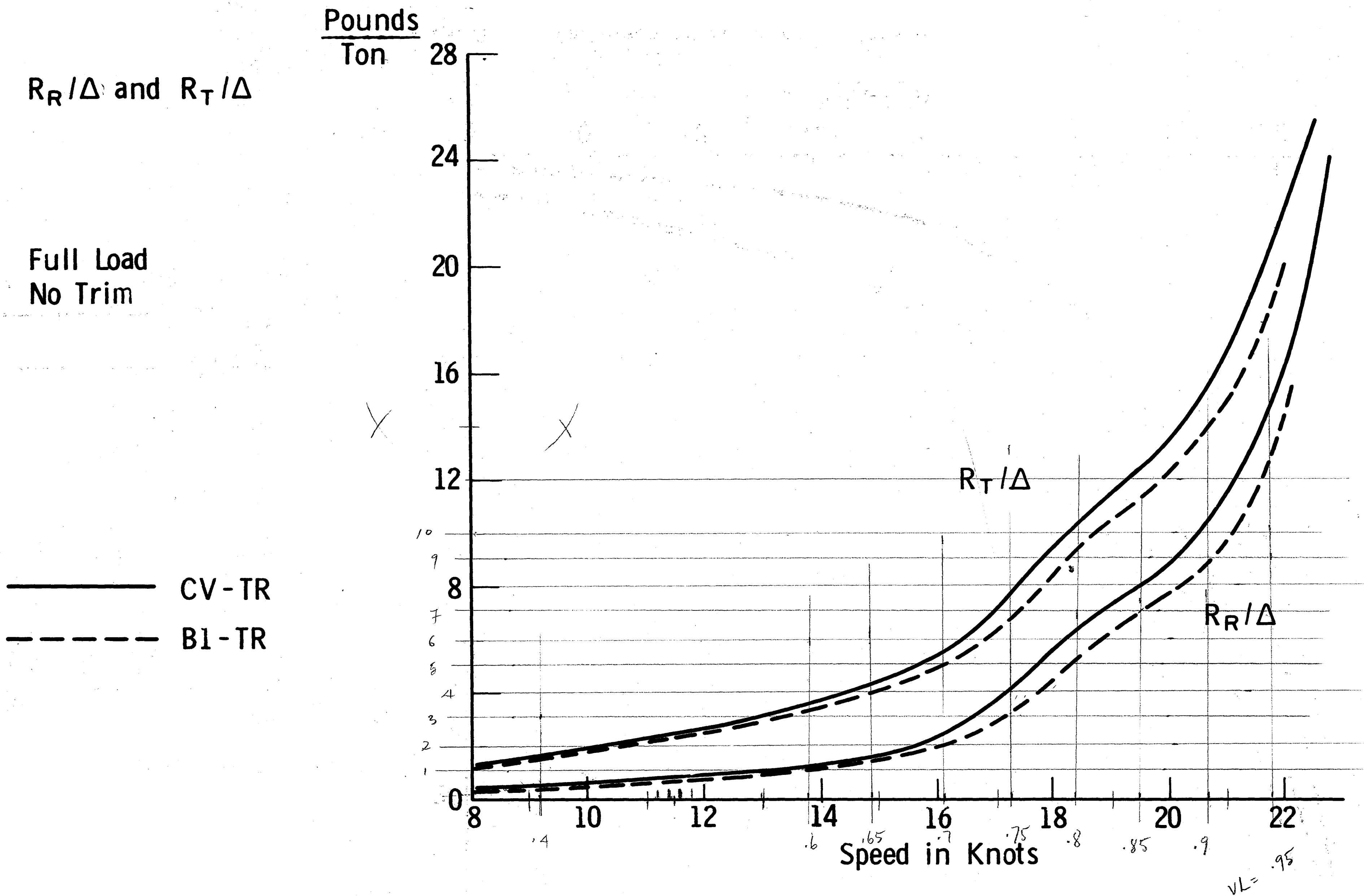

Figure 20. Comparison of resistances (100% displ.).

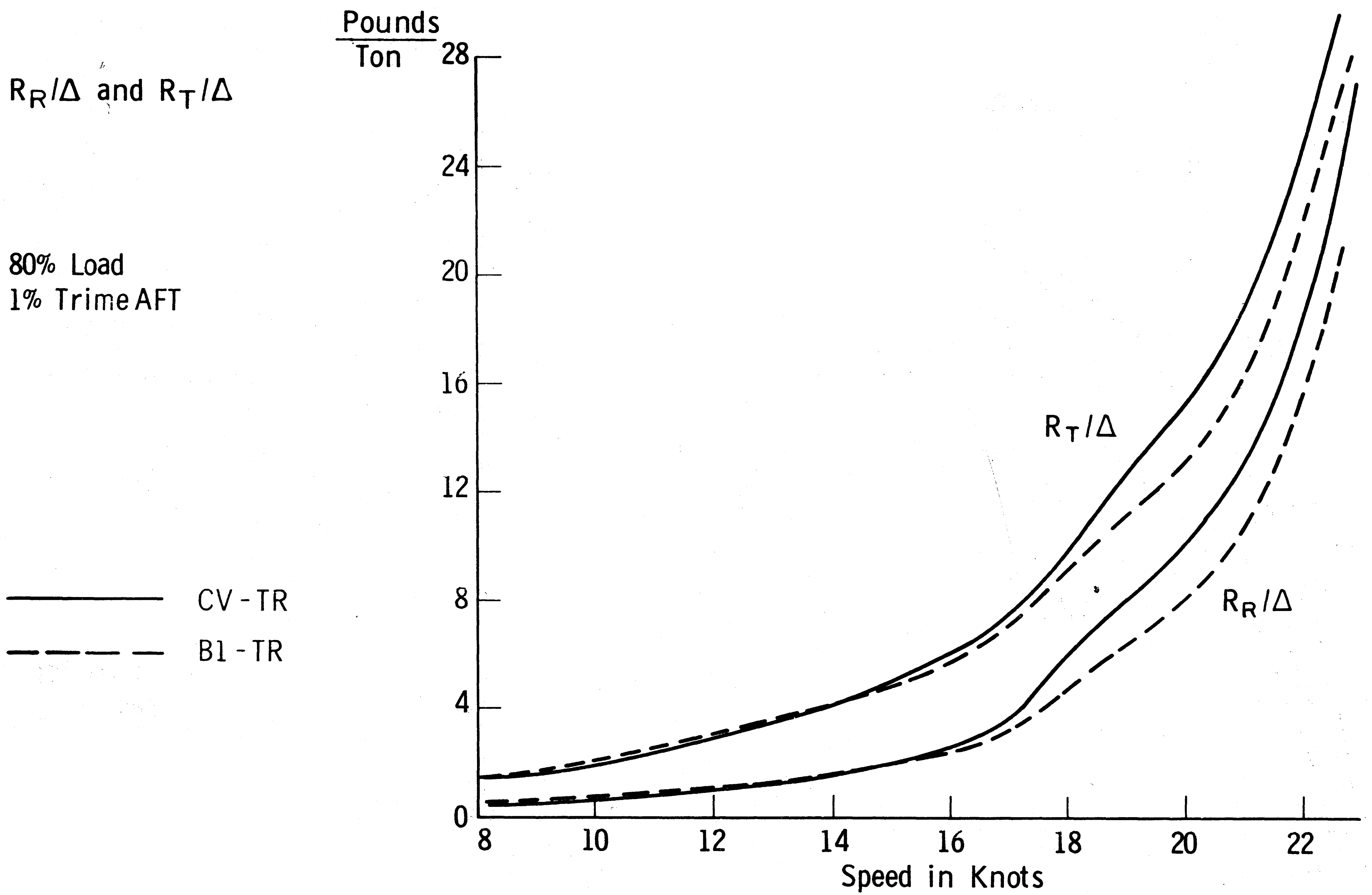

Figure 21. Comparison of resistances (80% displ.).

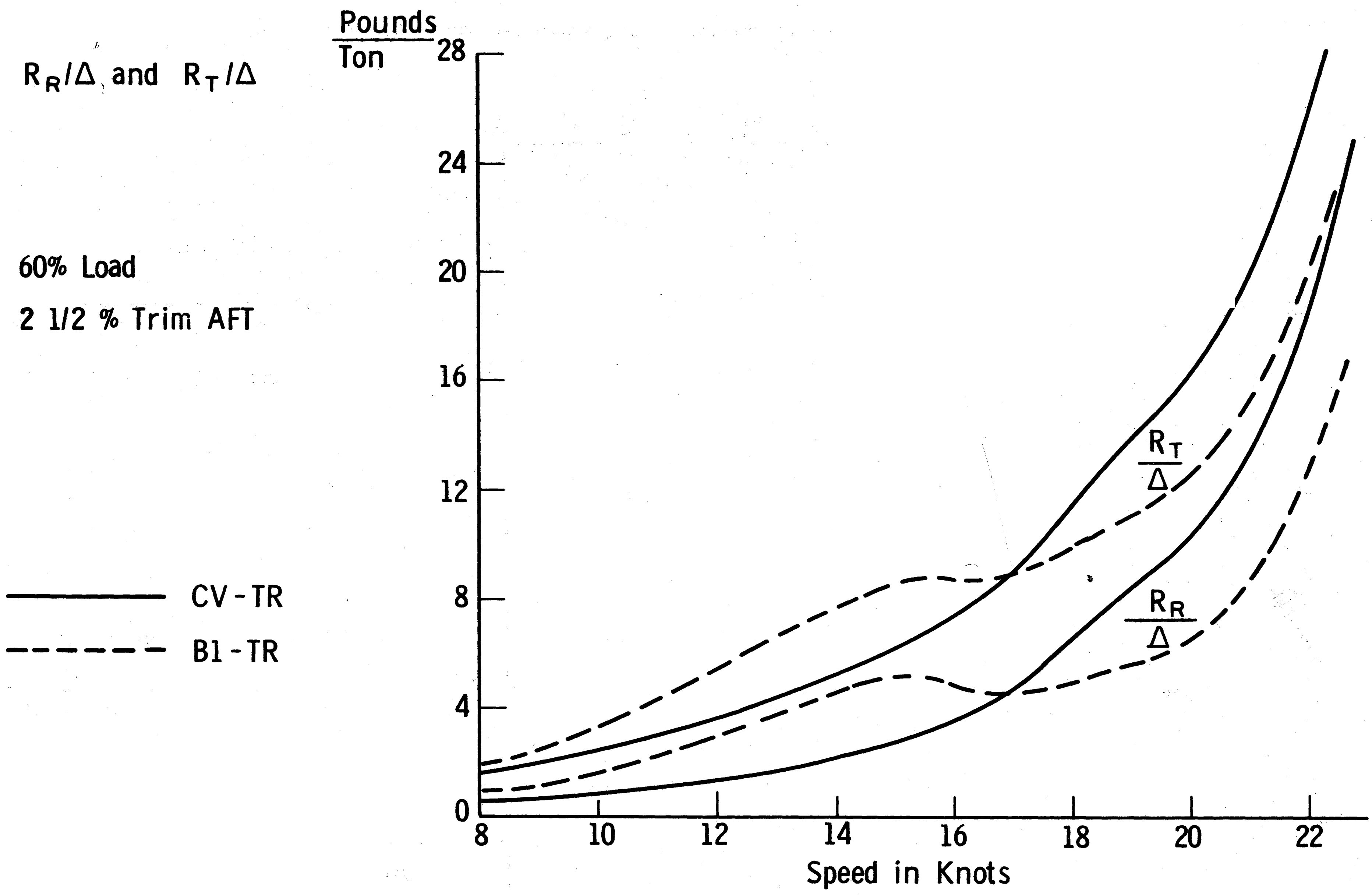

Figure 22. Comparison of resistances (60% displ.).

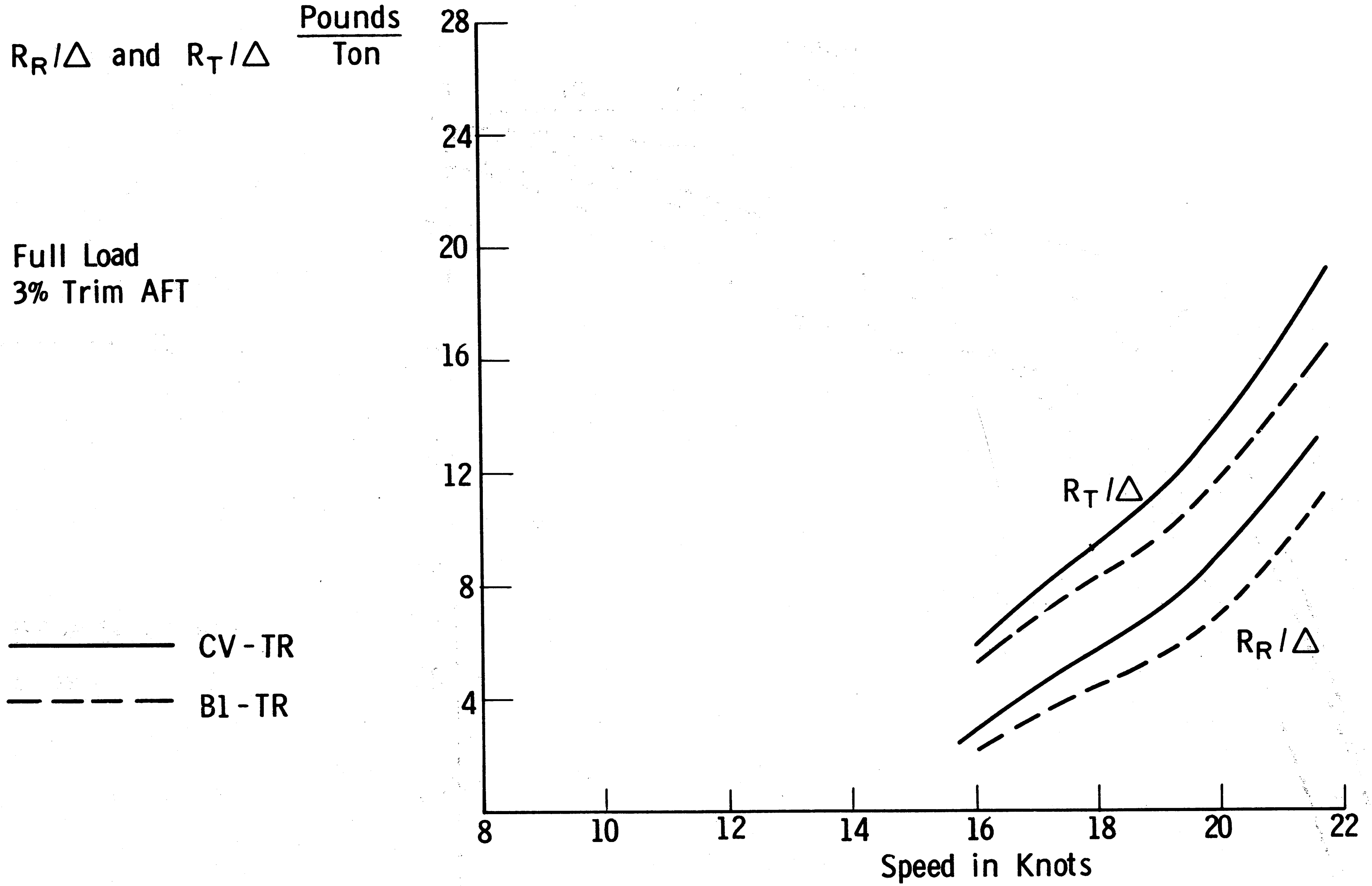

Figure 23. Comparison of resistances (100% displ. with trim).

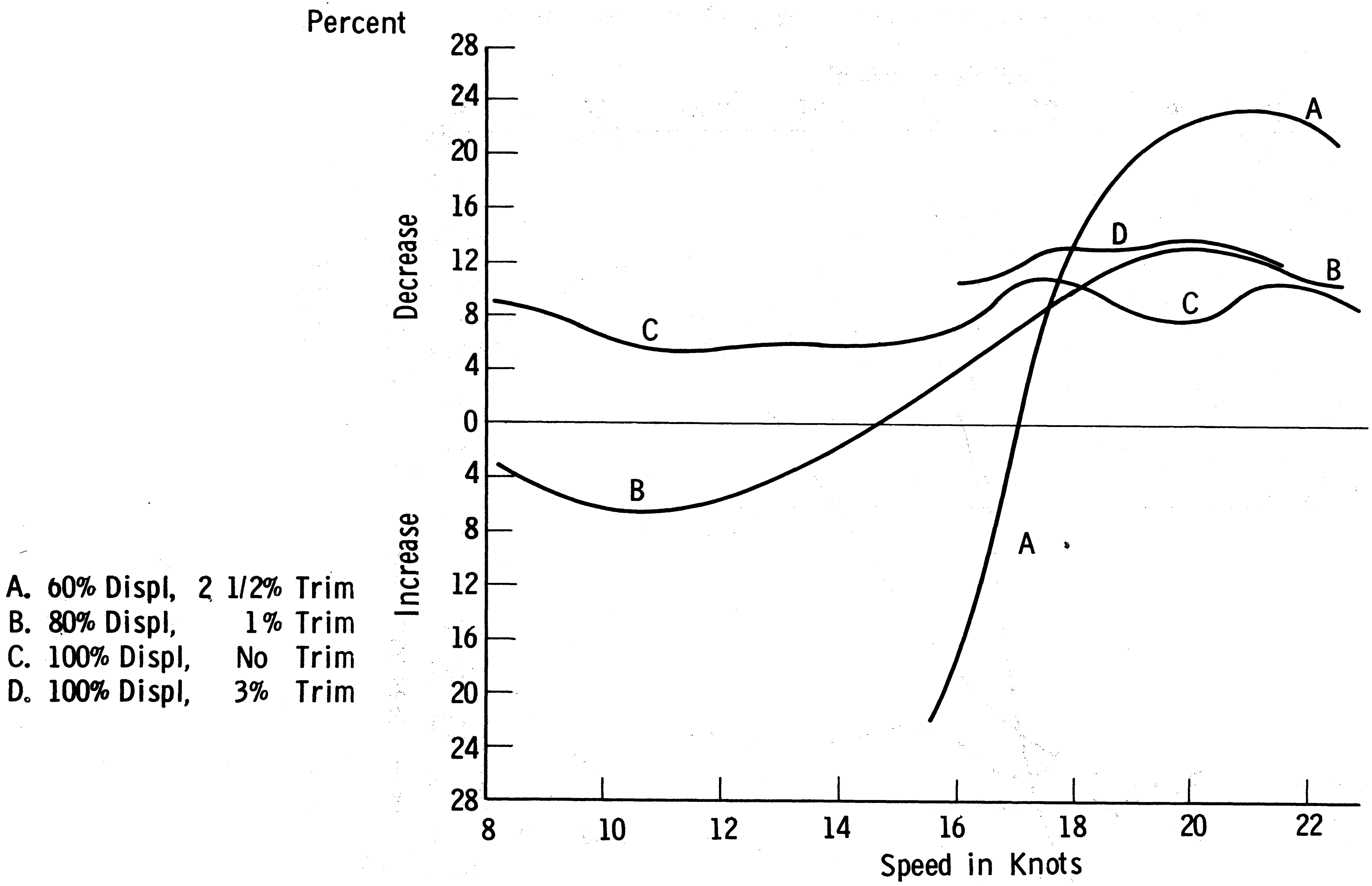

Figure 24. Change of R_T/Δ due to addition of bulb No. 1.

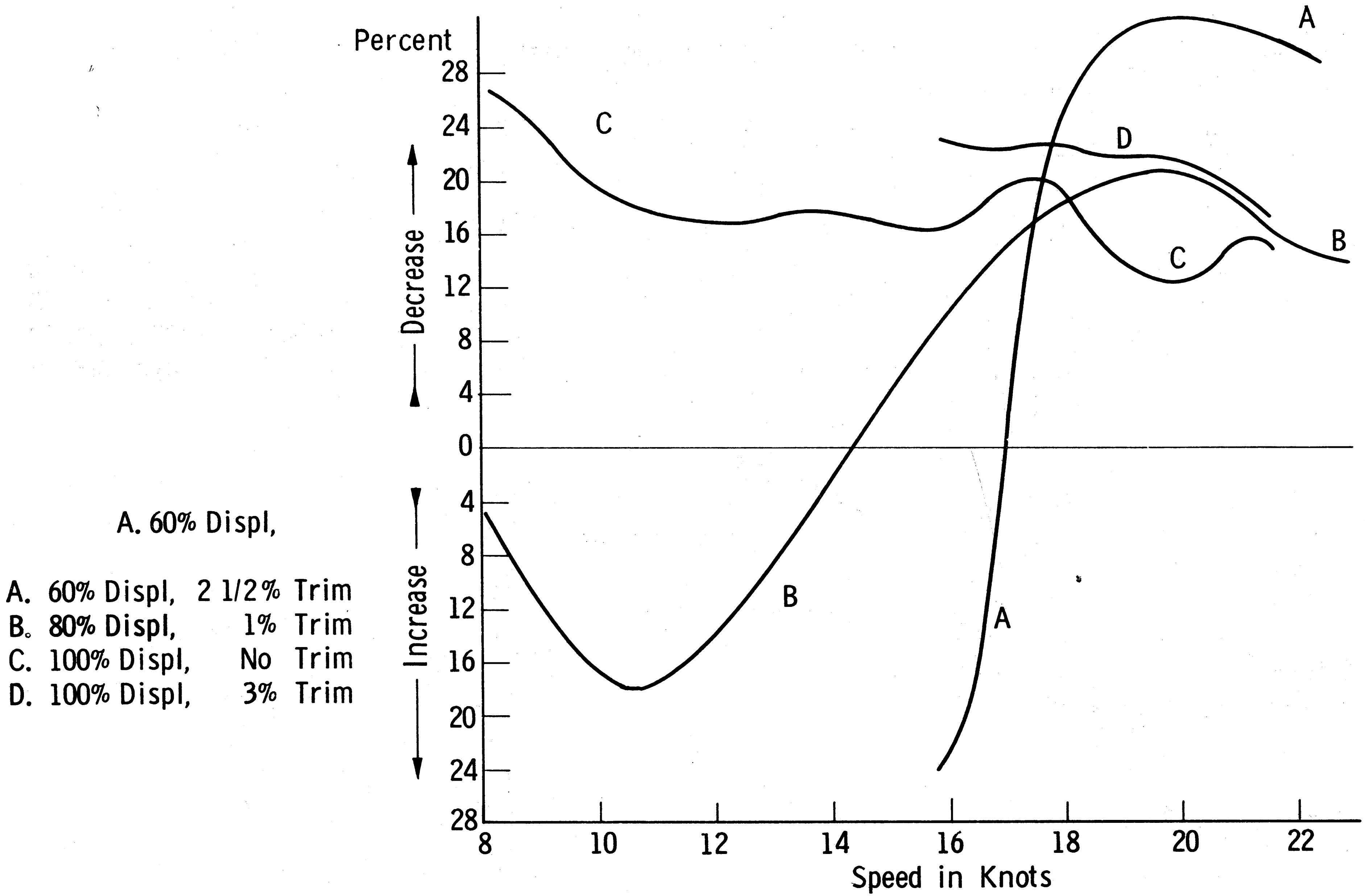

Figure 25. Change of R_R/Δ due to addition of bulb No. 1.

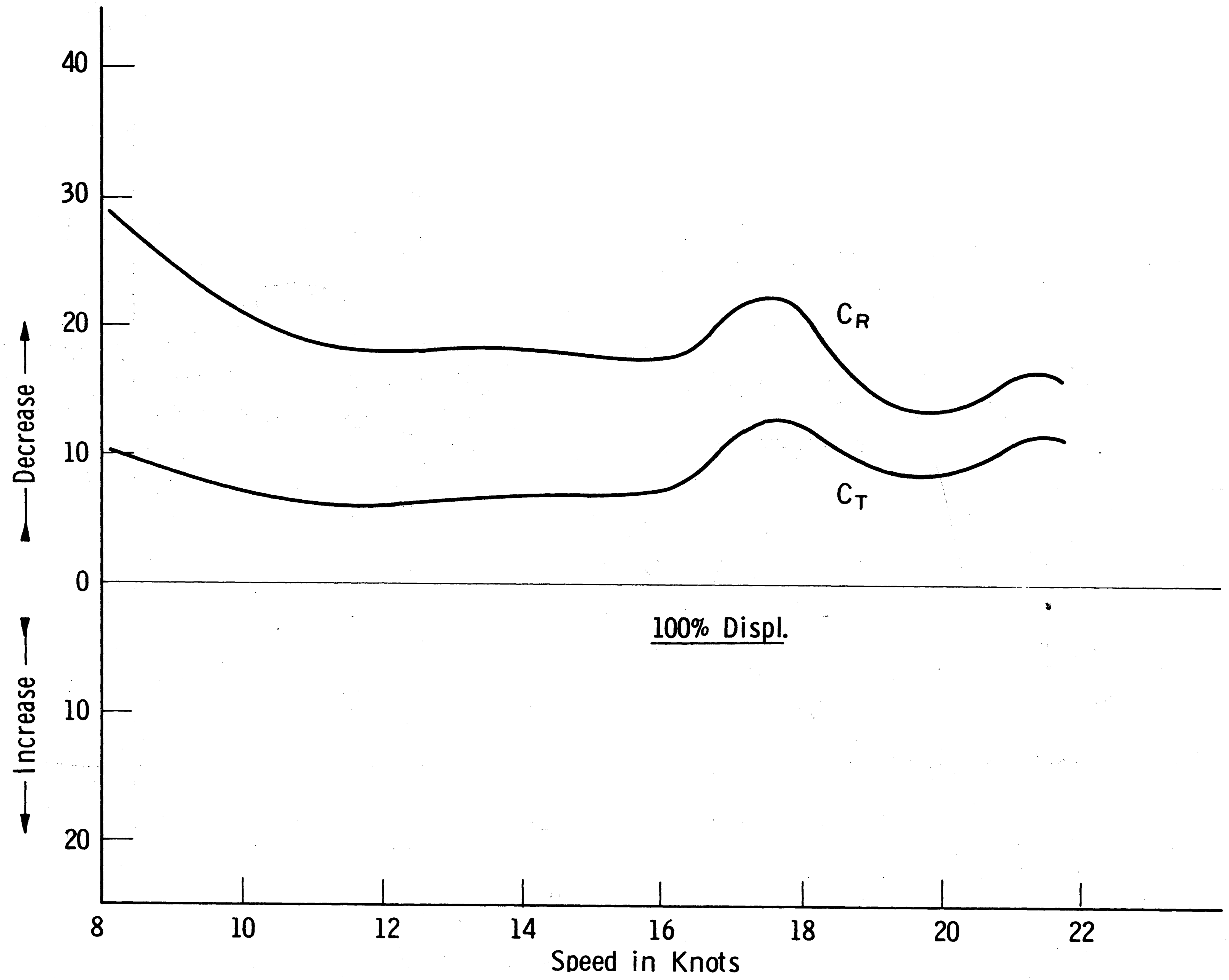

Figure 26. Percent change of C_R and C_T due to addition of bulb (100% displ.).

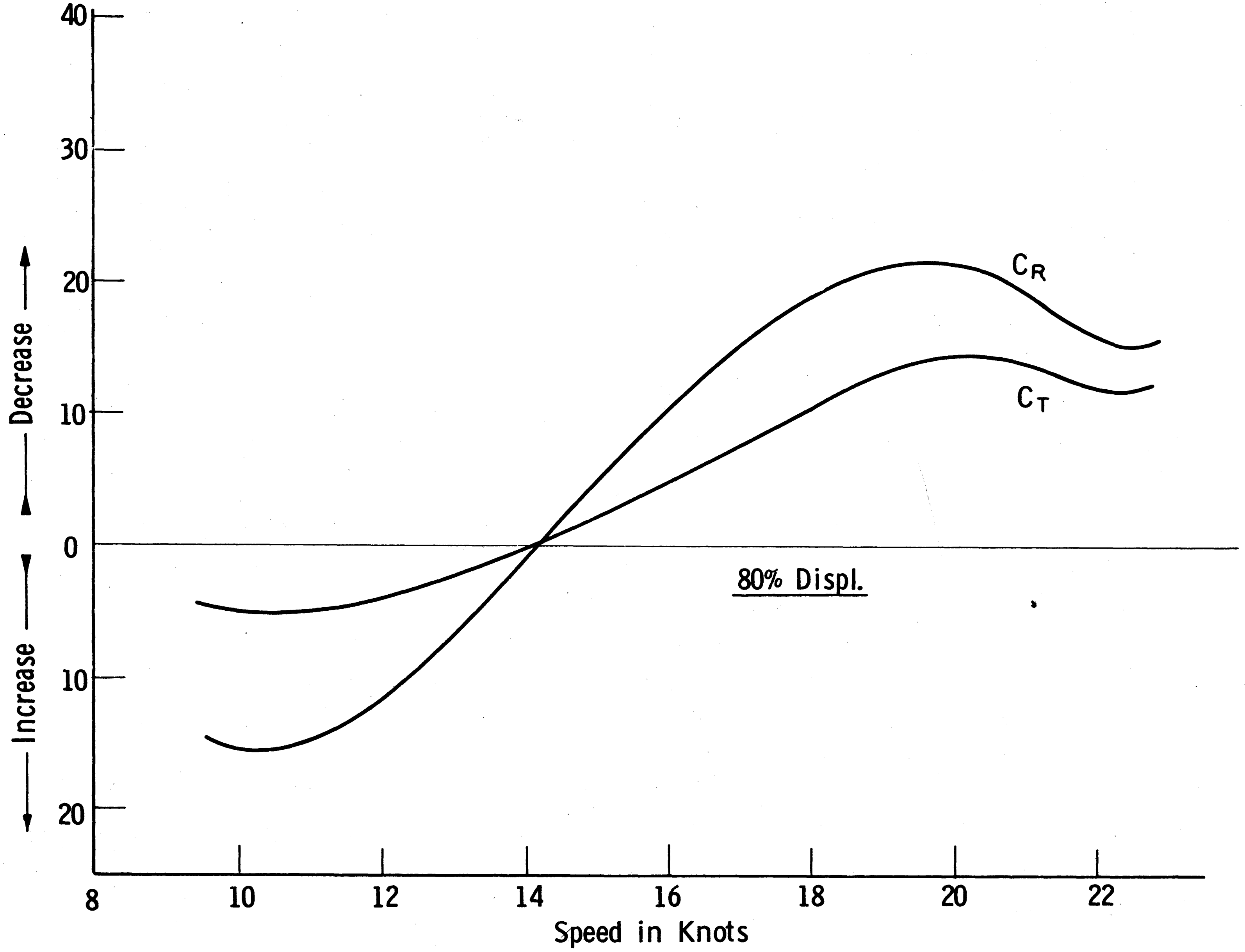

Figure 27. Percent change of C_R and C_T due to addition of bulb (80% displ.).

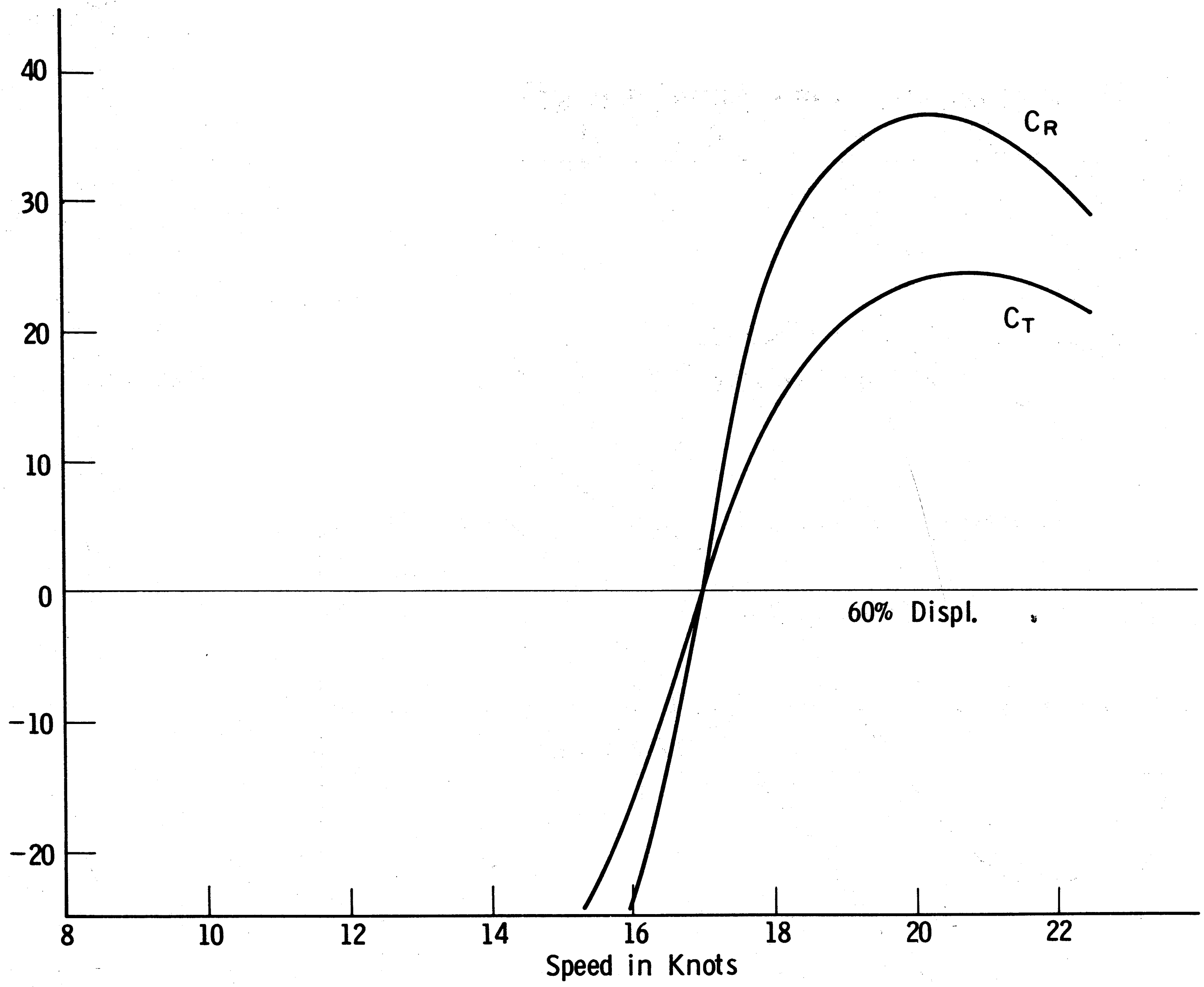

Figure 28. Percent change of C_R and C_T due to addition of bulb (60% displ.).

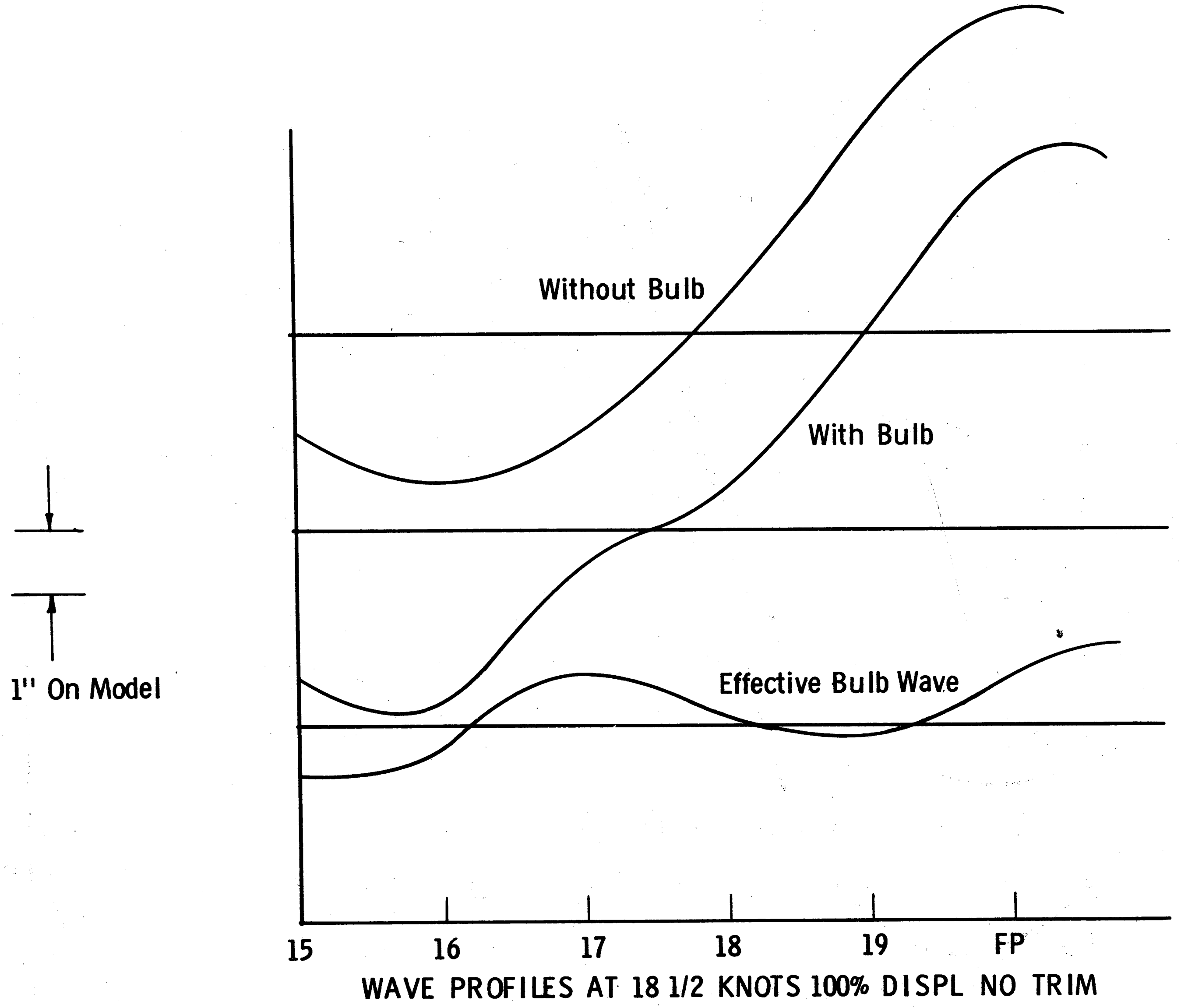

Figure 29. Wave profiles at 18-1/2 knots 100% displ. no trim.

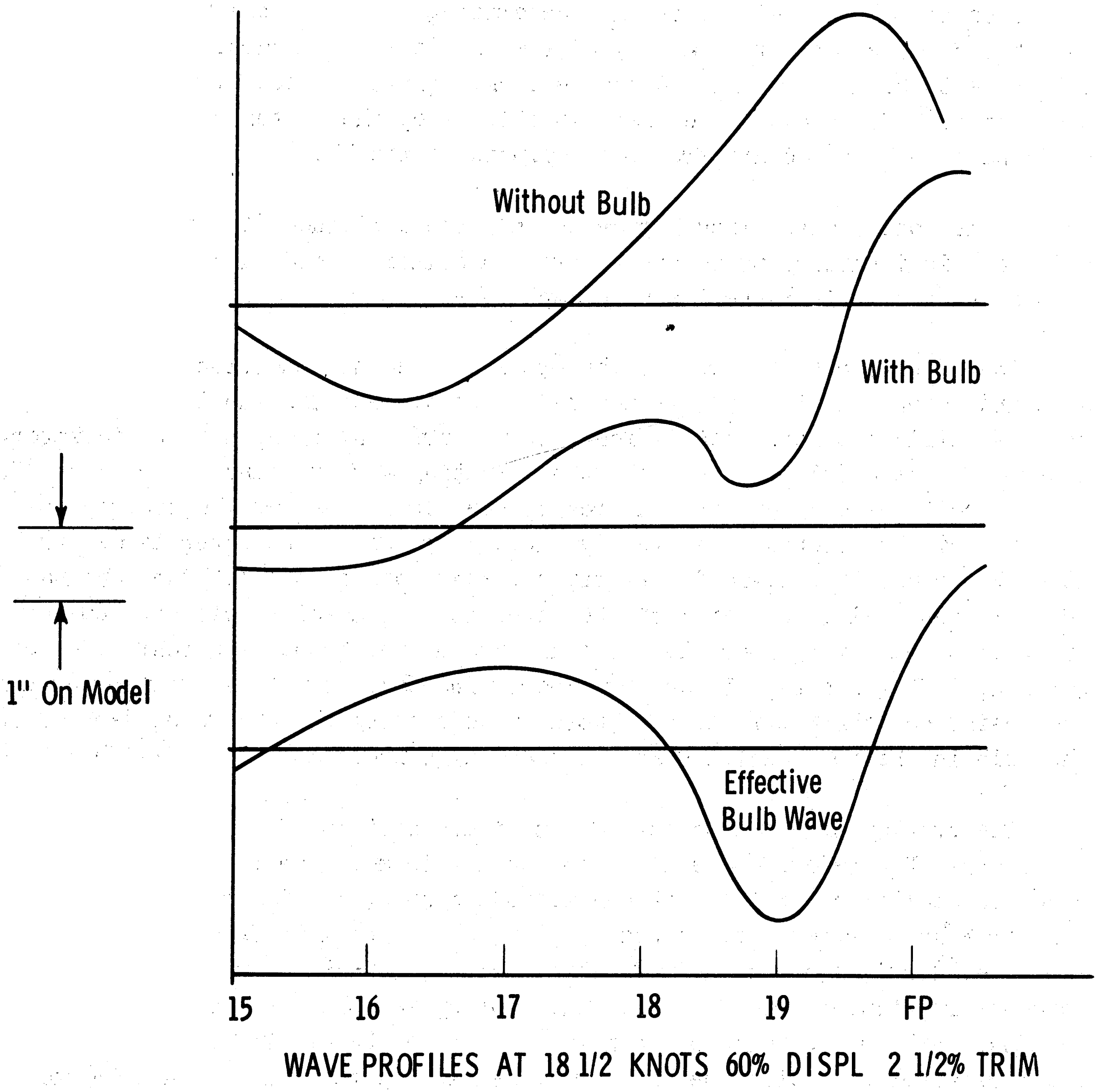

Figure 30. Wave profiles at 18-1/2 knots 60% displ. 2-1/2% trim.

should be to move the effective wave-generating center of the bulb aft. Although the trough and crest are 4% of the ship's length apart, the bulb will only have to be moved a small distance back. (This has been learned from experience.) It is estimated that, in this case, the center of the bulb would have to be moved aft from 0.5% forward to the FP.

It is noted, also, from Figure 29 that the amplitude of the effective bulb wave is insufficient to cause wave cancellation, and therefore something should be done to increase the wave amplitude.

The effective bulb wave for the 60% displacement condition (Figure 30) also indicates that the bulb is too far forward. But the amplitude of that wave is sufficient for cancellation. At the 60% condition, the static waterline is at the middle of the bulb, thus creating a very blunt waterplane. It was noticed, while conducting the resistance tests, that at low speeds, the flow around the forebody was greatly impeded by that blunt waterplane. But at high speeds, the stagnation pressure forward of the bulb's blunt end was sufficient to raise the level of the water in the immediate vicinity so that the water flowed very smoothly above and around the bulb. The resistance comparisons in Figures 24, 25, and 28 clearly indicate that this greatly influenced the resistance characteristics. Photographs showing the change of flow around the bulb in the 60% condition at a low and high speed appear in Figure 31.

The problem of changing the longitudinal location of the bulb is not a major one. But before proceeding any further in model tests, it has been decided to deal first with the wave-amplitude problem. Prior to explaining the further tests carried out, we note that when the obtained data is expanded from the 14.00' model to the 530' ship, using a $\Delta C_F = .0004$ based on the Schoenherr friction line, the effective horsepower obtained at 19-1/2 knots is 19,400 EHP in the full load condition with the bulb. The EHP curves for the three sterns and the one bulb are given in Figures, 32, 33, and 34 for the 100%, 80%, and 60% displacement conditions, respectively.

The tests were conducted on a 14.00' model using a trip wire for turbulence stimulation of 0.036-inch diameter at 5% LBP aft of the FP; and on the bulb studs were used following normal practice for large bulbous bows. The rudder was in place for the resistance tests.

Figure 31a. 1042-BI-TR at $V/\sqrt{L} = 0.70$ (60% displ., 2-1/2% trim).

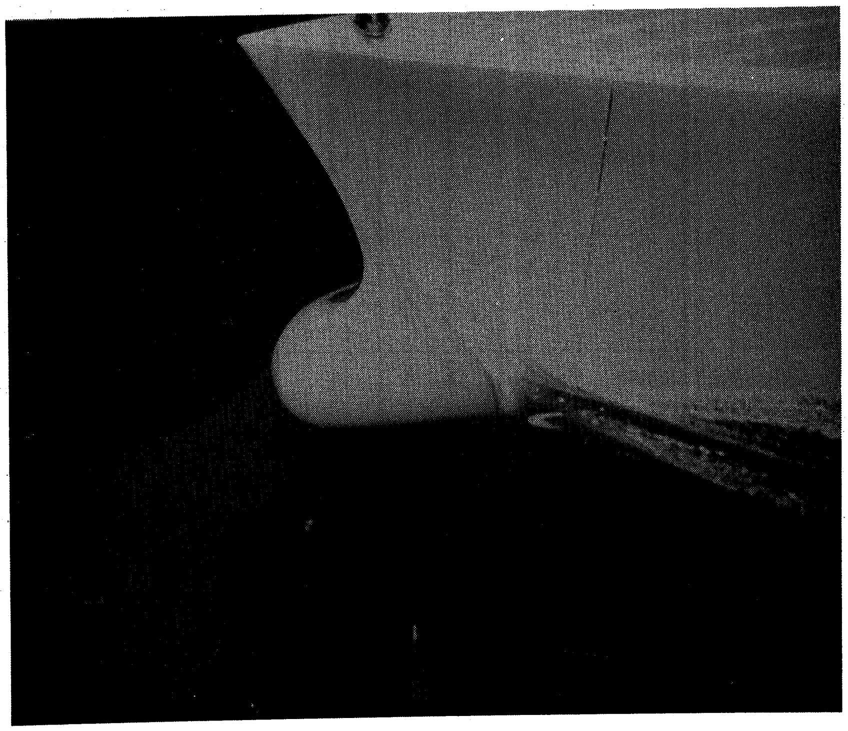

Figure 31b. 1042-BI-TR at $V/\sqrt{L} = 0.85$ (60% displ., 2-1/2% trim).

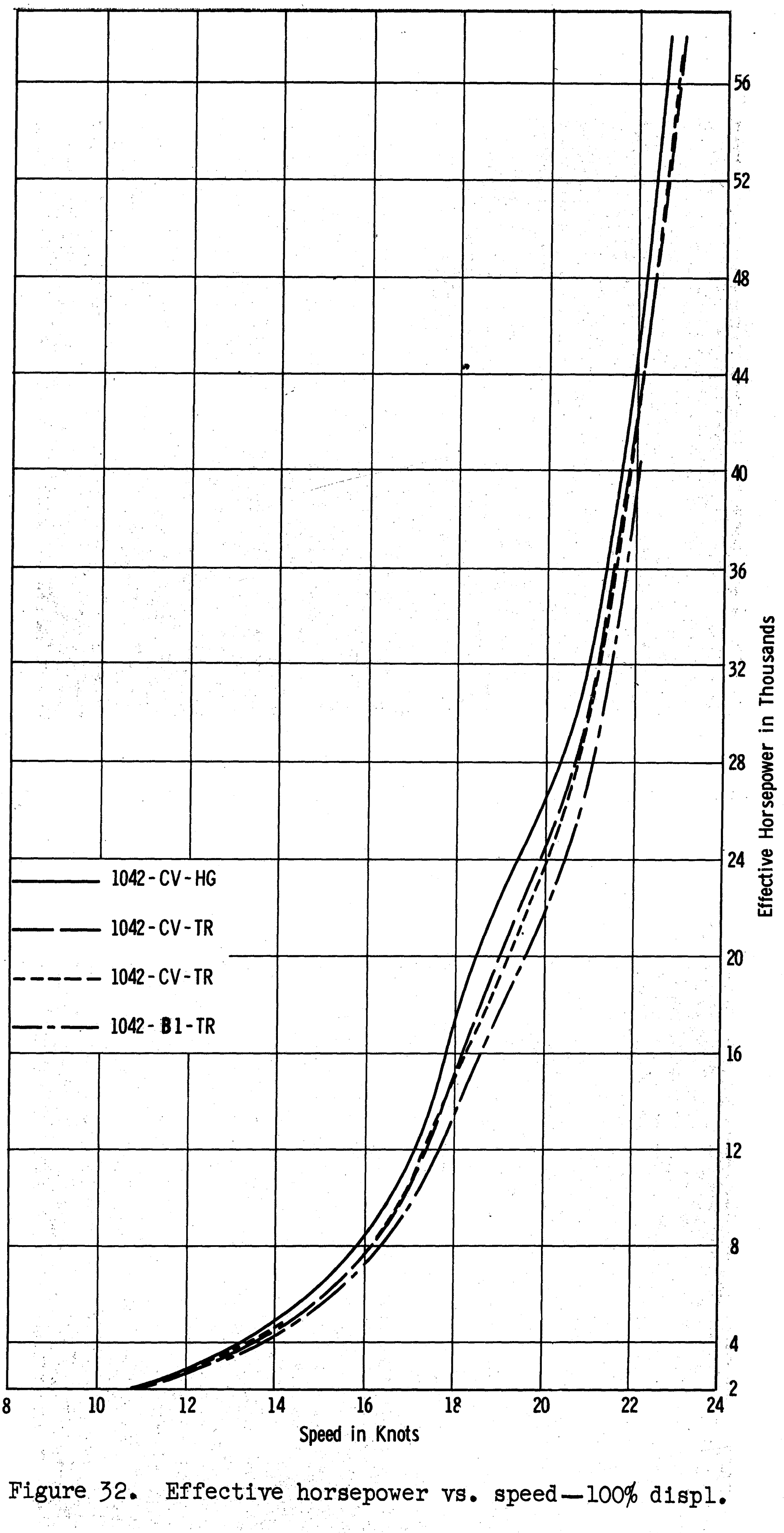

Figure 32. Effective horsepower vs. speed—100% displ.

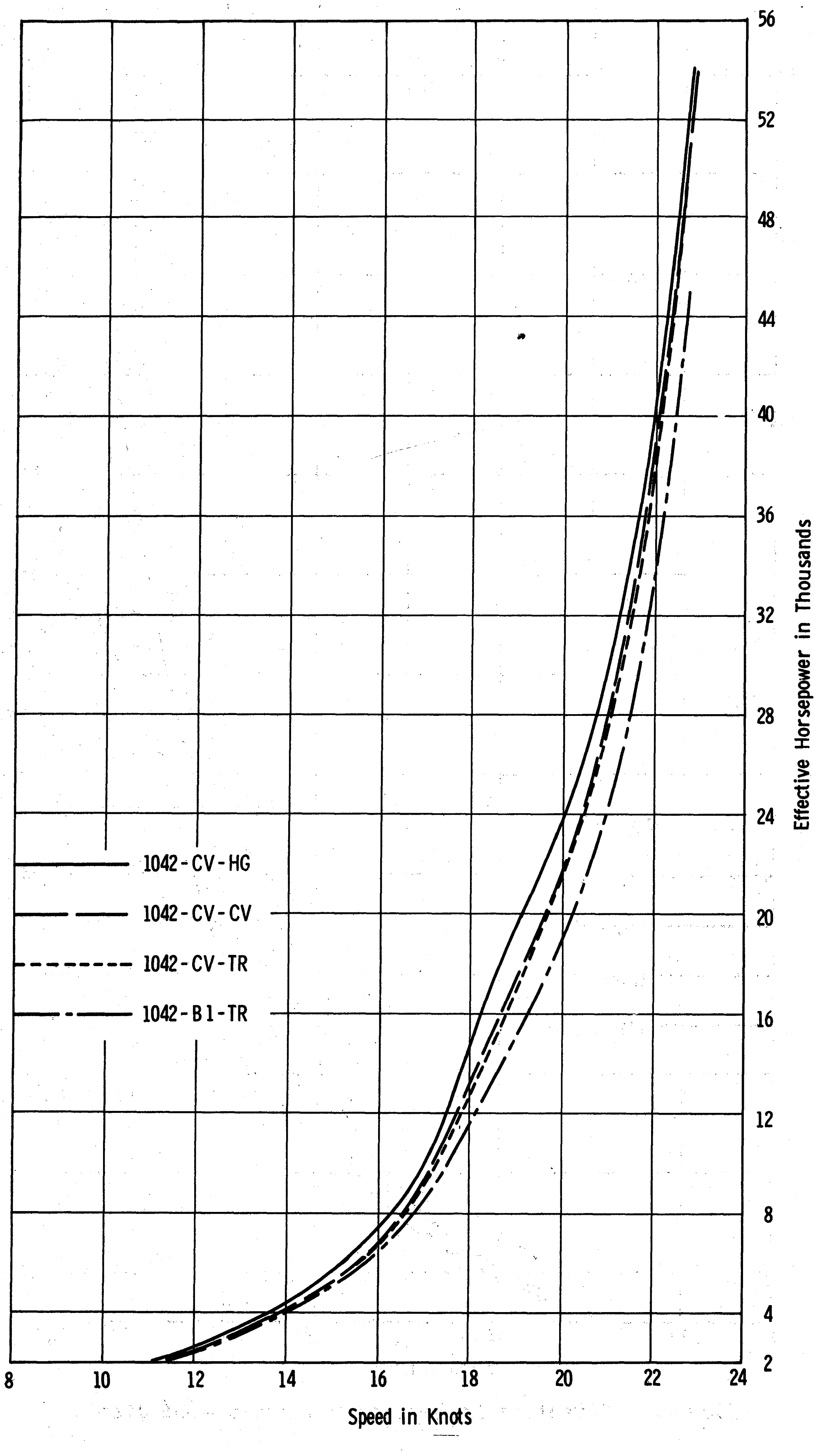

Figure 33. Effective horsepower vs. speed—80% displ.

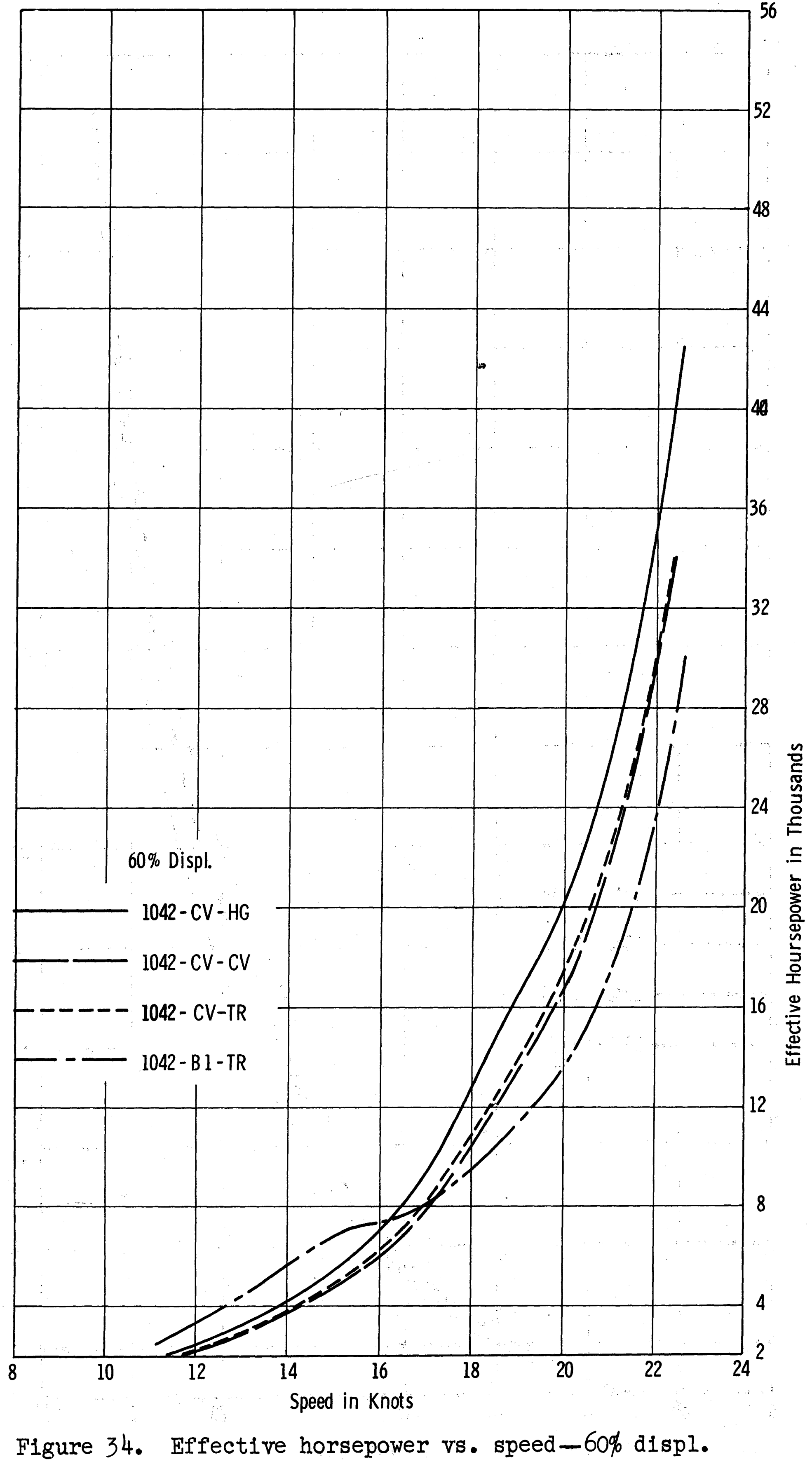

Figure 34. Effective horsepower vs. speed—60% displ.

SIMULATION OF CHANGE OF BOW CONFIGURATION

In pursuing the problem of wave amplitude, it was noted that the amplitude of the effective bulb wave in the 60% condition was substantially greater than that for the 100% load condition (Figures 29 and 30). Believing that this was due to depth of immersion of the bulb, a series of tests were devised that would quickly simulate the variation of the depth of the bulb.

Resistance tests were carried out on the model without the bulb and with the transom stern (1042-CV-TR) in full displacement with several different trim conditions. The trim ranged from 1% forward to 3% aft. Then with the bulb in place the same tests were run, thus varying the depth of the bulb from 76% to 41% of the mean draft. Hence, the comparisons were made for identical conditions. The results showed that at the greatest trims (i.e., with the bulb closest to the surface) the most significant reductions were obtained. Figure 23 shows the values of R_T/Δ and R_R/Δ with and without the bulb at the 100% displacement (3% trim) condition. Curve "D" in Figures 24 and 25 show the changes of R_T/Δ and R_R/Δ due to the addition of the bulb. We note that the total resistance decreased by 14%, whereas in level trim the total resistance decreased by only 8%.

For the purpose of reporting on all obtained results in this report, Figures 35-38 are included. They show the changes of C_T and C_R due to trimming in the full load, as compared to the no-trim case, for the model with and without the bulb.

It appears obvious that the next step on the development should then be to design and test a forebody having the bulb center closer to the surface. The design has been completed and is shown in Figures 39 and 40. Note from them that the forefoot has been completely removed. It is thought that this will further reduce any underside eddying. When a ship of such a configuration is to be constructed, it may be found that a forefoot below the bulb will be necessary to maintain steerage when moving through cross winds and seas.

The construction of this model must await further appropriation of funds from the Maritime Administration. However, once it is constructed, besides testing it in the usual conditions of 100% displacement (no trim), 80% (1% trim), and 60% (2-1/2% trim), it will also be tested in full load at several trims, so that in at least one of them the static waterline will be in the middle of the bulb (as in the present 60% condition), thus creating a very blunt static waterplane.

Although it is expected that the resistance reductions in the 80% and 60% conditions might not be as great with the raised bulb as they are now, it is

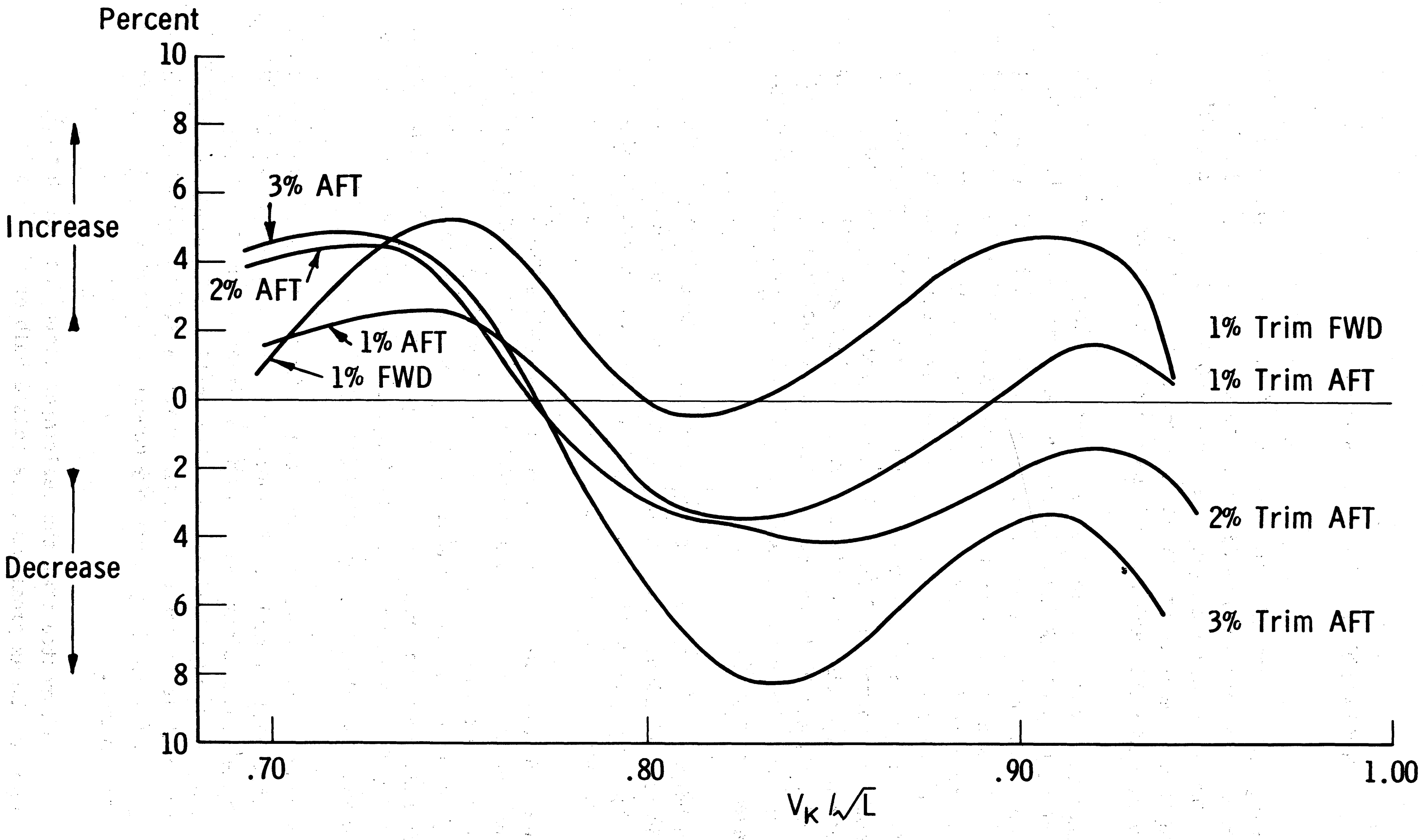

Figure 35. Change of $C_T°$ due to trim—full displ.—1042-BI-TR.

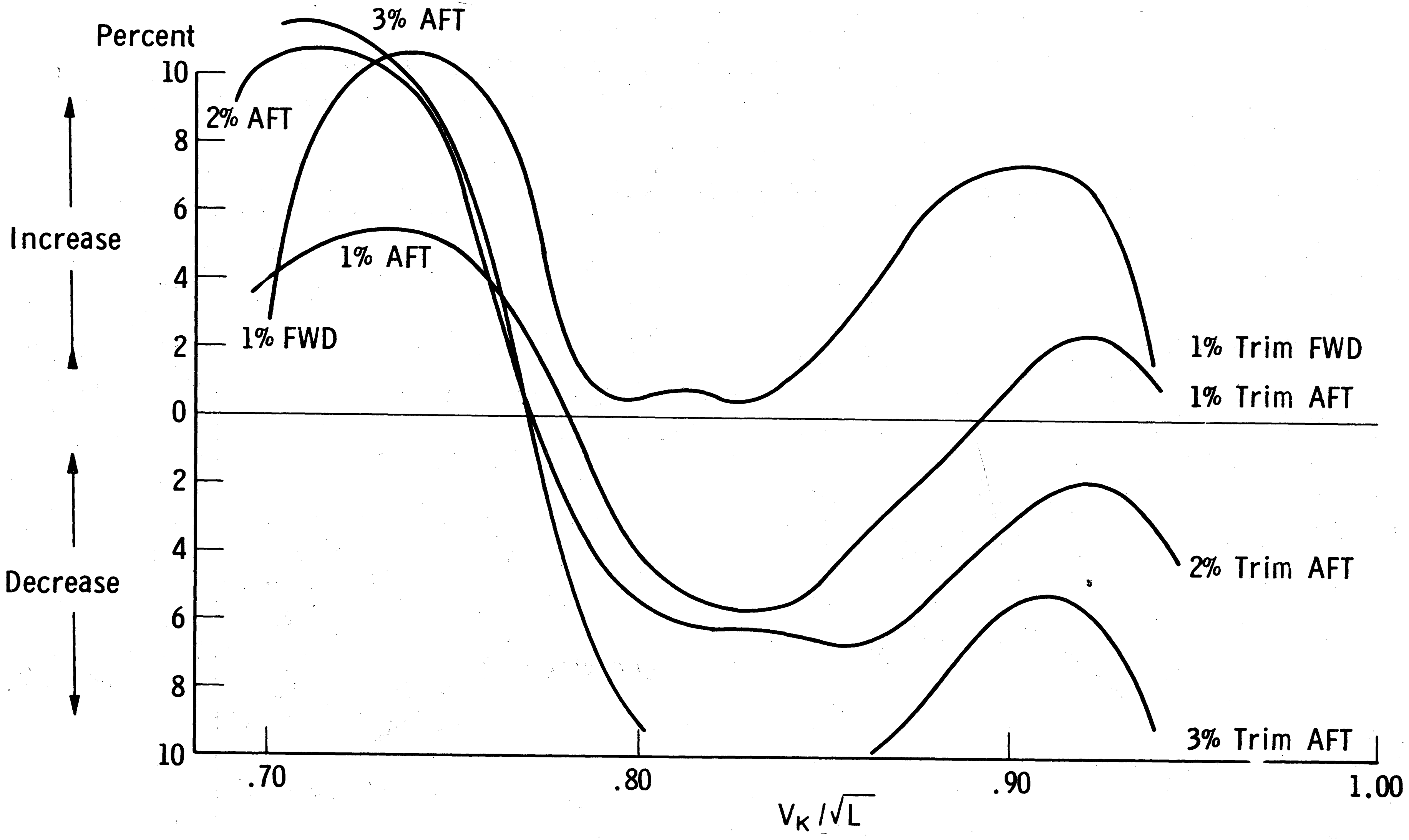

Figure 36. Change of C_R° due to trim—full displ.—1042-BI-TR.

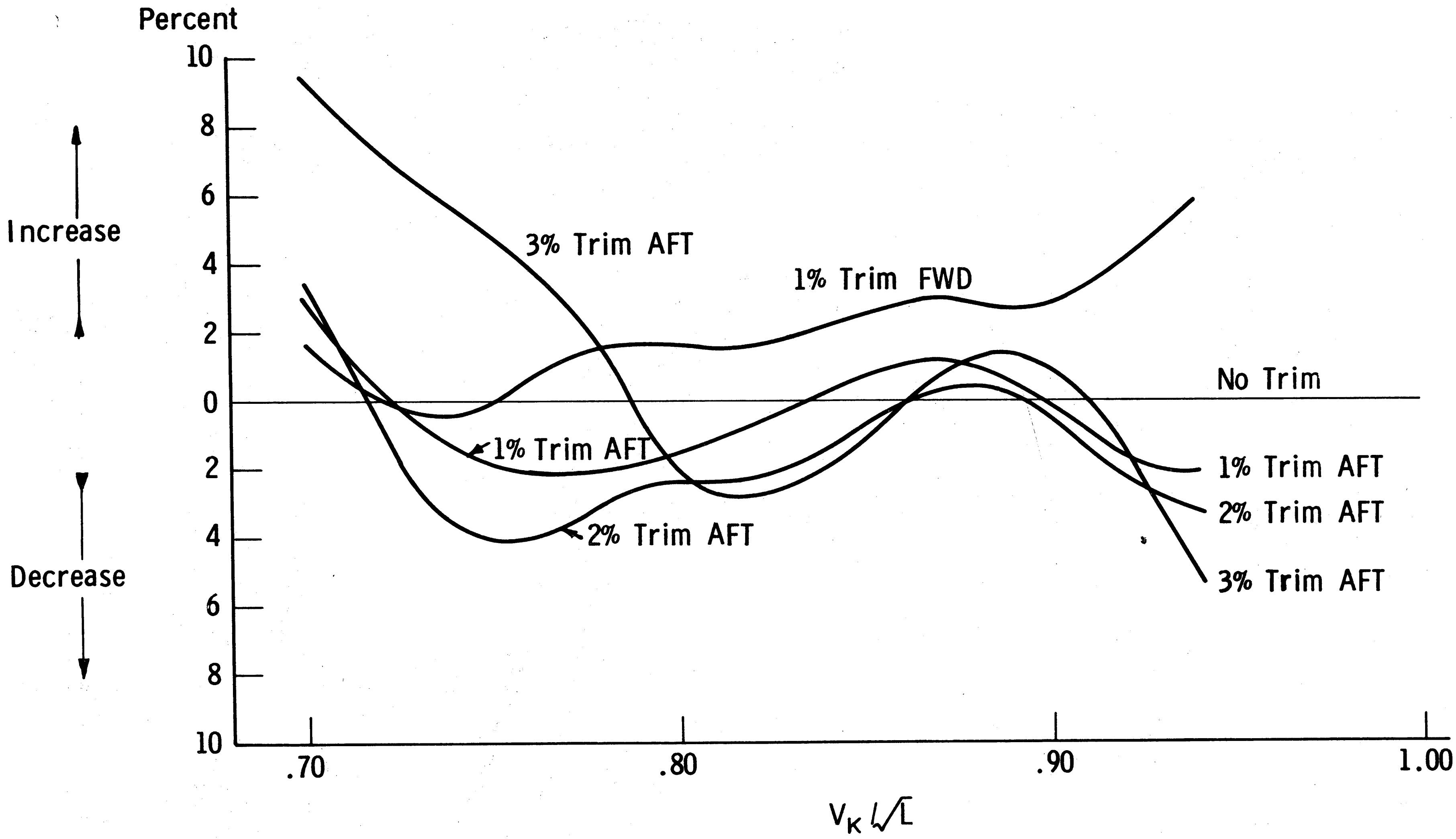

Figure 37. Change of $C_T°$ due to trim—full displ.—1042-CV-TR.

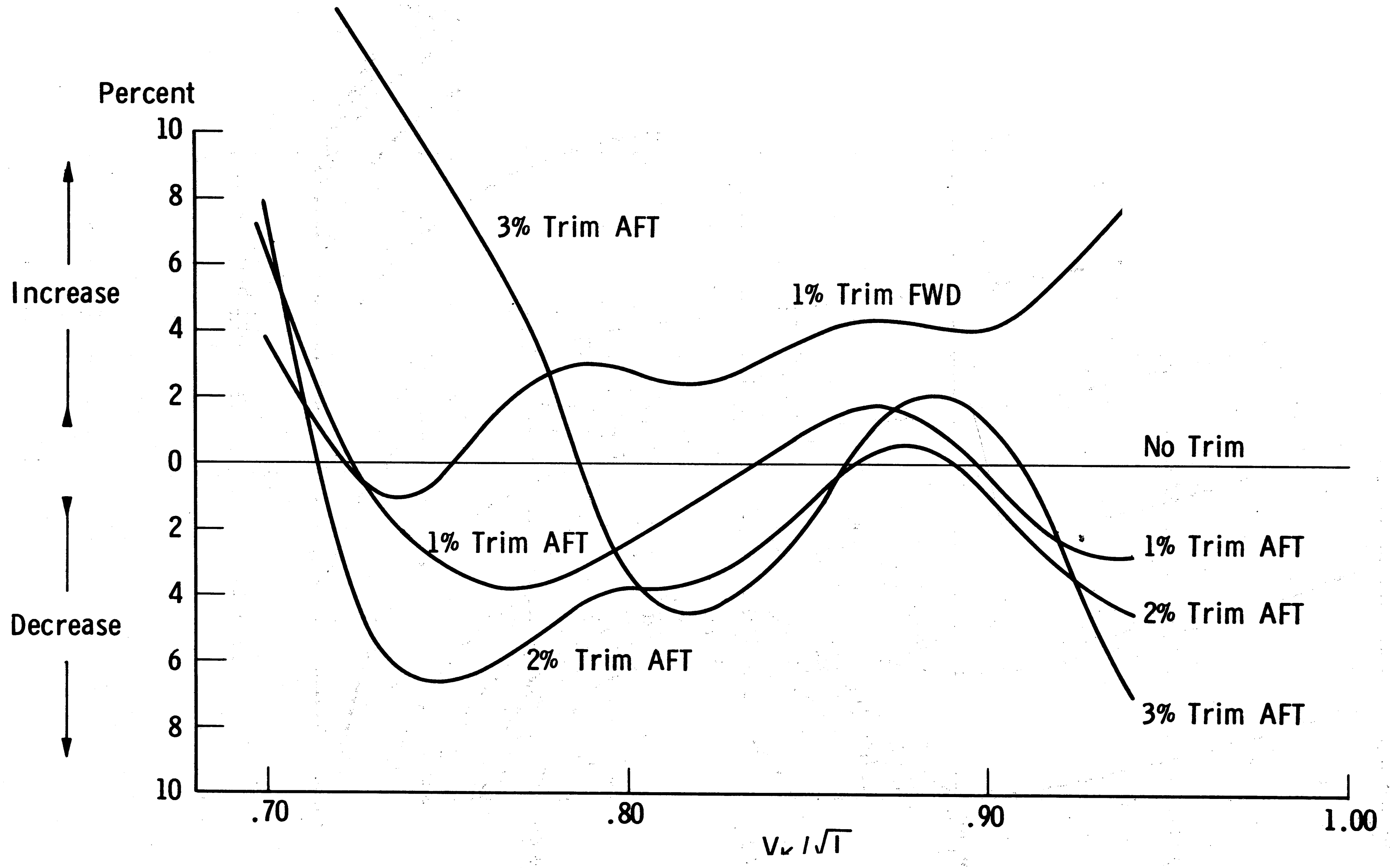

Figure 38. Change of $C_R°$ due to trim—full displ.—1042-CV-TR.

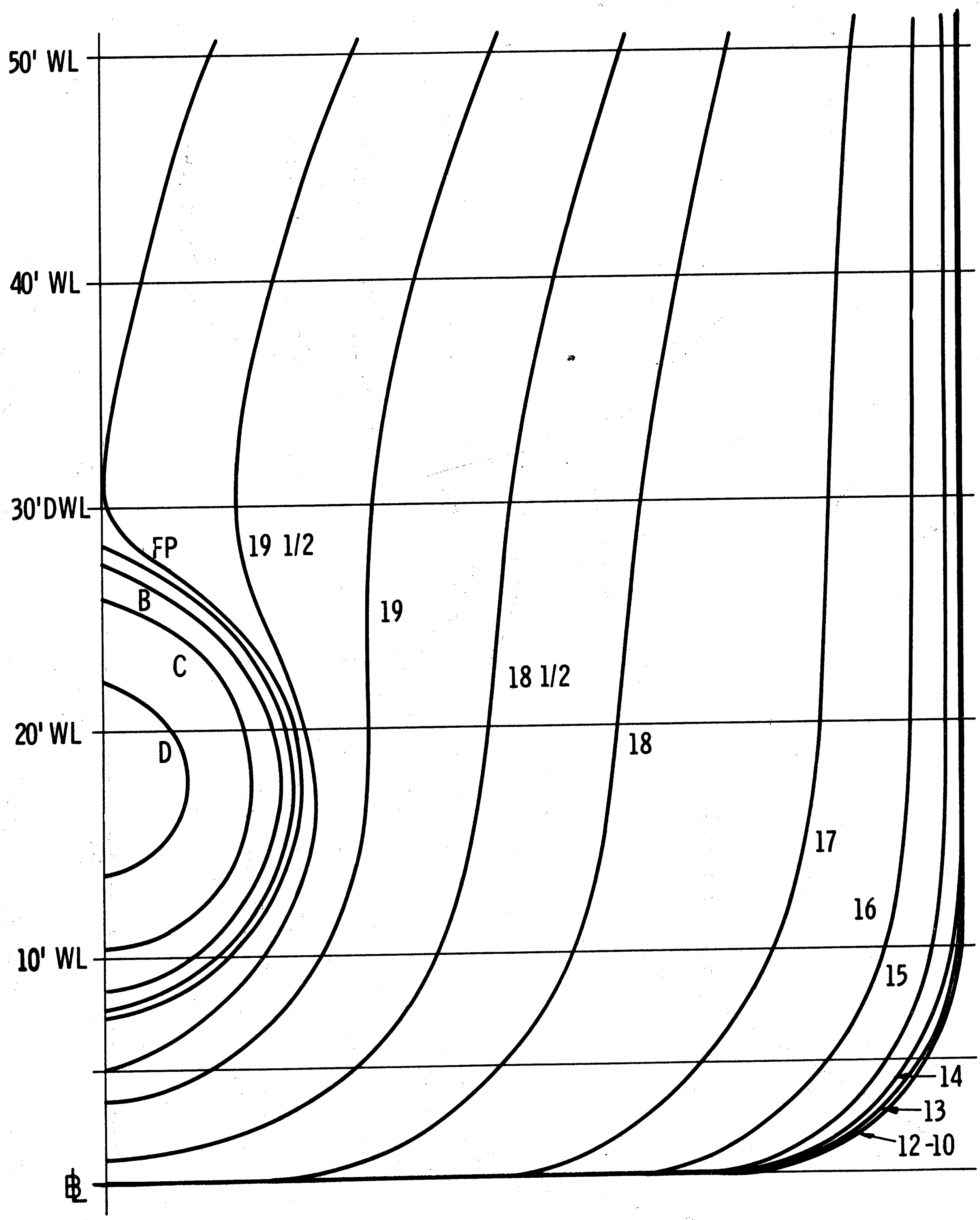

Figure 39. Proposed raised-bulb forebody.

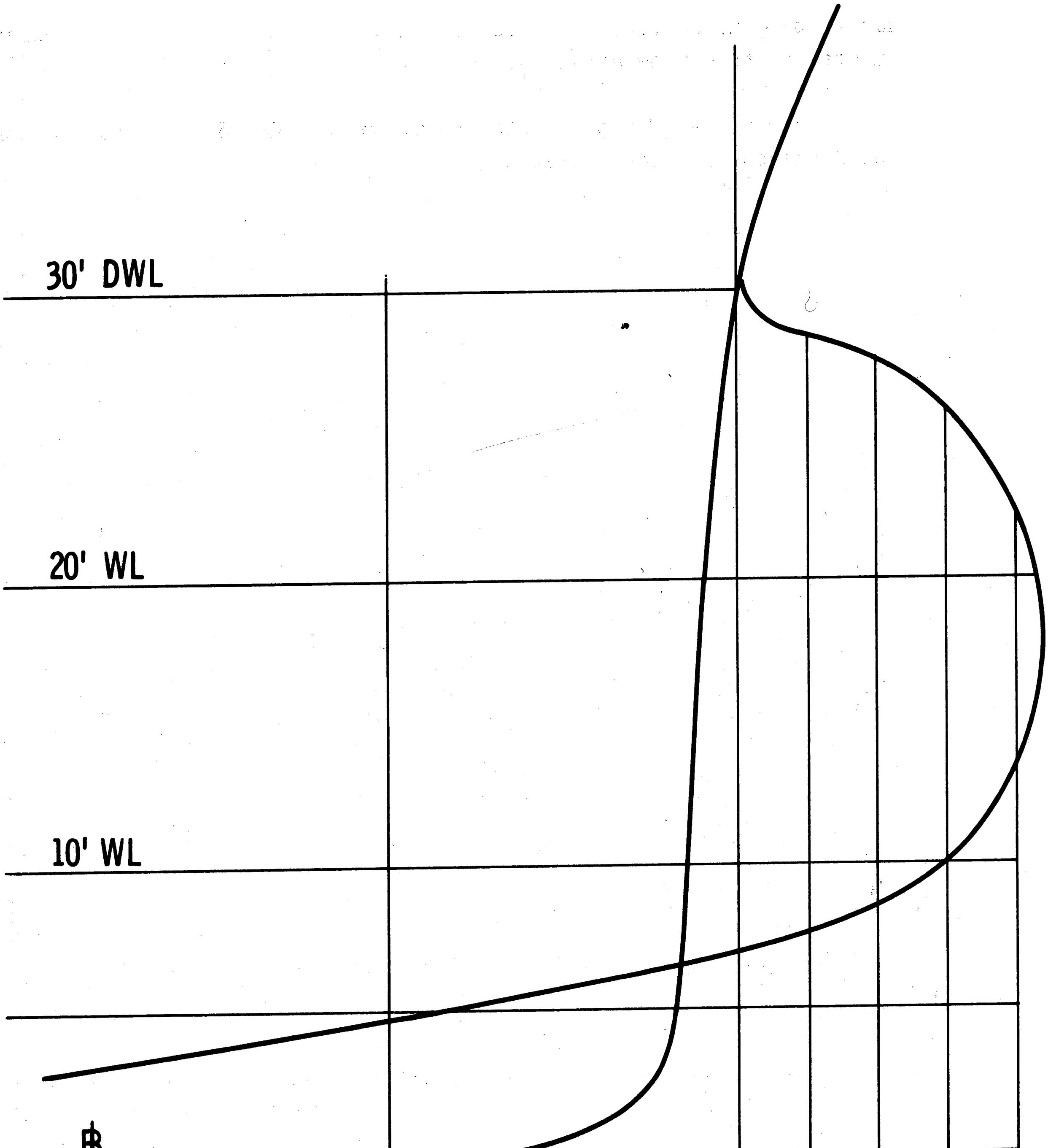

Figure 40. Profile of proposed raised bulb.

thought that reductions of resistance in full load greater than the 14% achieved in the simulated tests may be attained.

Based on the results of those tests it may be considered desirable to construct a third bulbous bow configuration.

SELF-PROPELLED TESTS

Self-propelled tests were conducted in the full displacement condition on models 1042-BIM-TR and 1042-CV-TR using University of Michigan propeller number 2. The propeller is a Troost propeller having a rake of 6° instead of the 15° on the original Troost series. The ship was designed for a 22'-1" diameter propeller, and the scale size of UM propeller No. 2 is 22'-0". The characteristics of the propeller and the open-water test curves of the propeller are given in Figure 41. The diameter of the model propeller is seven inches.

The results of the two tests are given in Figures 42 and 43. At the design speed of 19.5 knots the bulbous hull form requires 29,230 SHP: 10.0% less than the bulb-less form. At the same time the EHP is only 6.5% less. Thus it is apparent that the bulbous bow has some overall beneficial effect on the propulsive characteristics.

For purposes of a detailed comparison between the two SHP tests, listed below are the calculated summaries at the design speed of 19.5 knots ($V/\sqrt{L} = 0.85$).

	1042-CV-TR	1042-BIM-TR
t	.178	.169
w_Q	.242	.267
w_T	.248	.256
$(1-t)/(1-w)$*	1.089	1.123
e_p	.595	.600
e_{rr}	1.008	1.007
J_T	.594	.609
J_Q	.599	.600
EHP/SHP	.653	.679

---

*In which w is the average of w_T and w_Q.

It is seen from this summary that the immediate effect of the bulb is to cause a slightly higher wake fraction and a lower thrust deduction; thus a higher hull efficiency. Due to the reduced EHP the propeller is operating with a lighter load on the model with the bulb; resulting in a higher propeller efficiency. The change of e_{rr} is negligible.

The lighter load on the propeller means that the momentum change through the propeller disc is less, and this might account for the higher wake fraction.

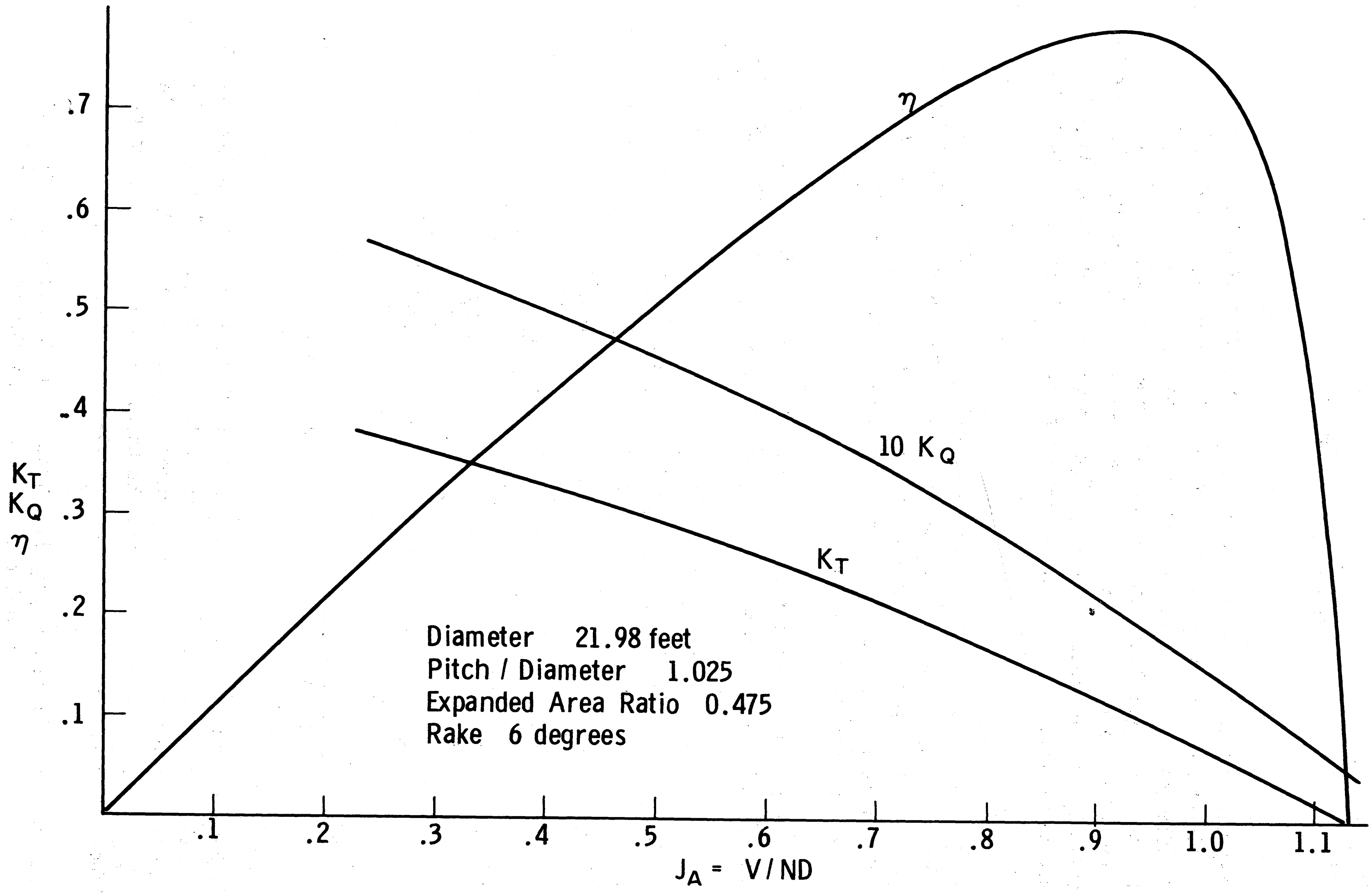

Figure 41. Open water characteristics of UM Prop. No. 2.

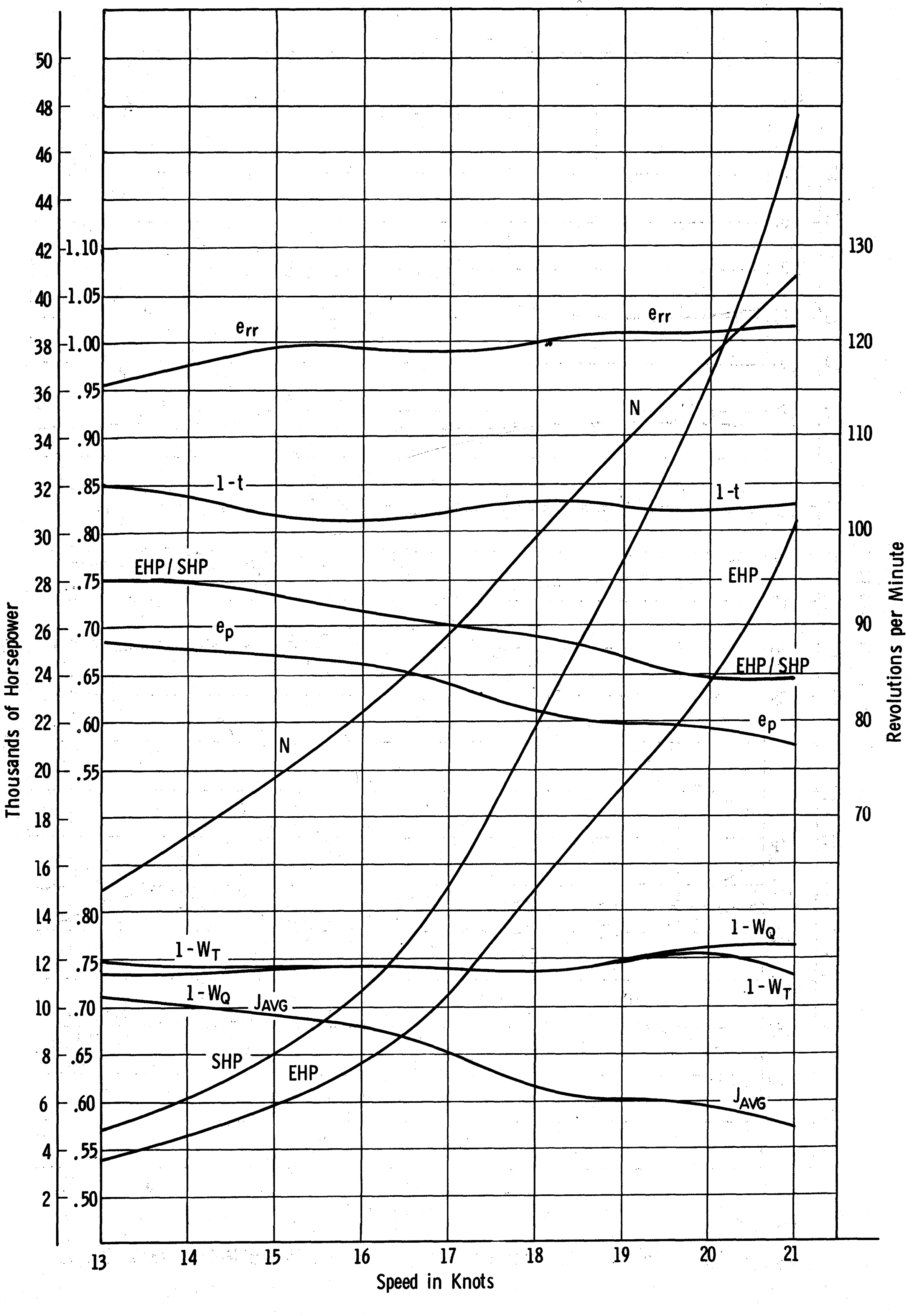

Figure 42. Shaft horsepower test, model 1042-CV-TR 100% displ.

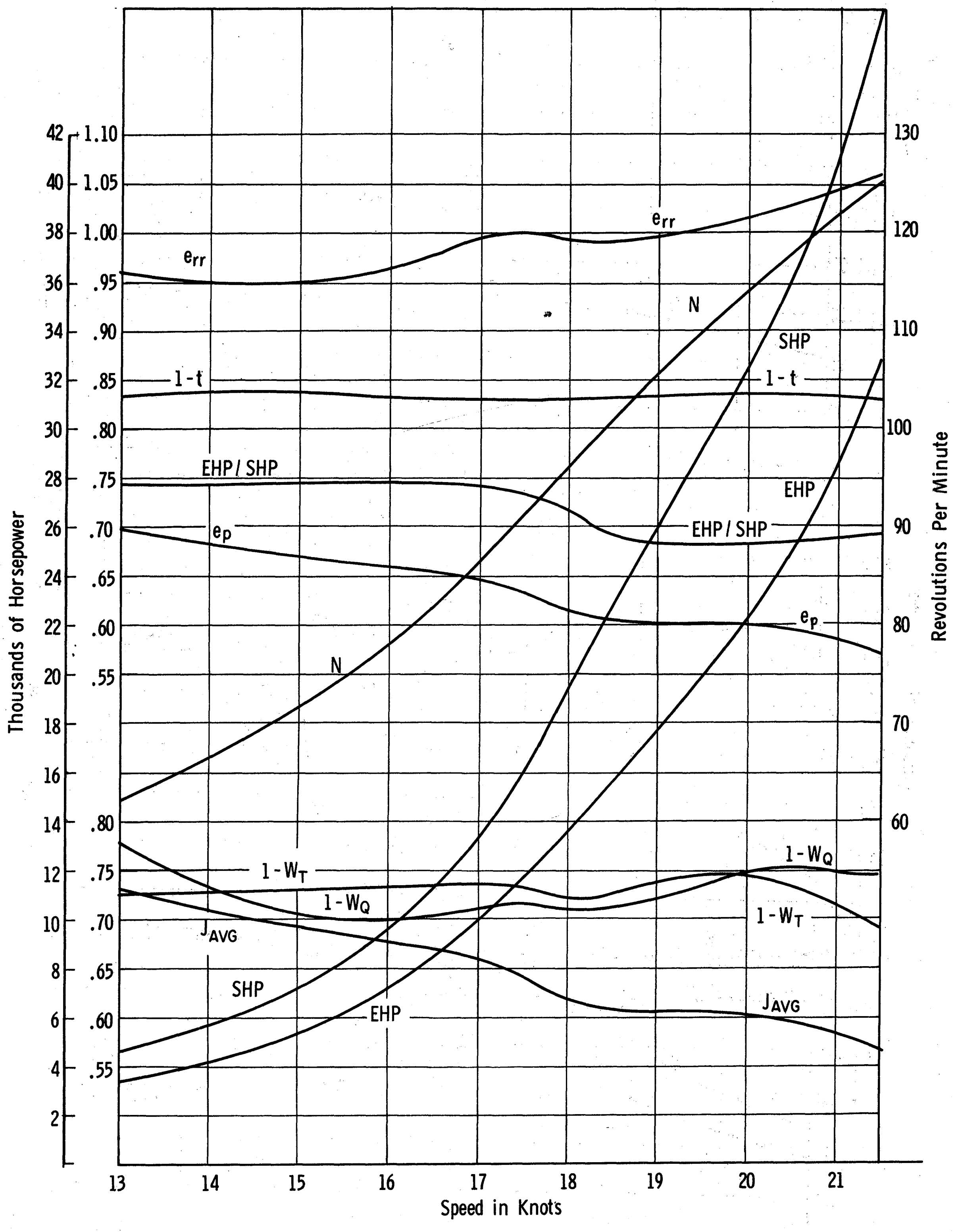

Figure 43. Shaft horsepower test, model 1042-BIM-TR 100% displ.

Based on the derived wake fractions and thrust deductions, the redesign of the propeller was attempted using Troost's charts. For all variations of blade area ratios and pitch/diameter ratios, the calculations showed that no propeller designed from the charts gave a higher propeller efficiency than the values obtained in the tests. Hence the model propeller used is considered satisfactory.

COMPARISONS WITH STANDARD SERIES

Comparison with a Series-60 ship is not possible at the design speed-length (0.85) ratio due to the fact that models of comparable fullness have been limited to ratios less than 0.80. However at that speed-length ratio model 1042-CV-TR has, for all practical purposes, a resistance equal to that of the equivalent Series-60 model; and model 1042-BIM-TR has 9.5% less EHP than the Series-60. A comparison of EHP versus speed is shown in Figure 44.

For volumetric coefficient (∇/L^3) and beam/draft ratio the same as model 1042, Taylor's Series gives, by a small extrapolation, a C_R of 3.0×10^3 while model 1042-BIM-TR has C_R equal to 2.9×10^3. The dimensions and configurations of Taylor's Series are different than those of the model tested, and thus the validity of the comparison is questionable.

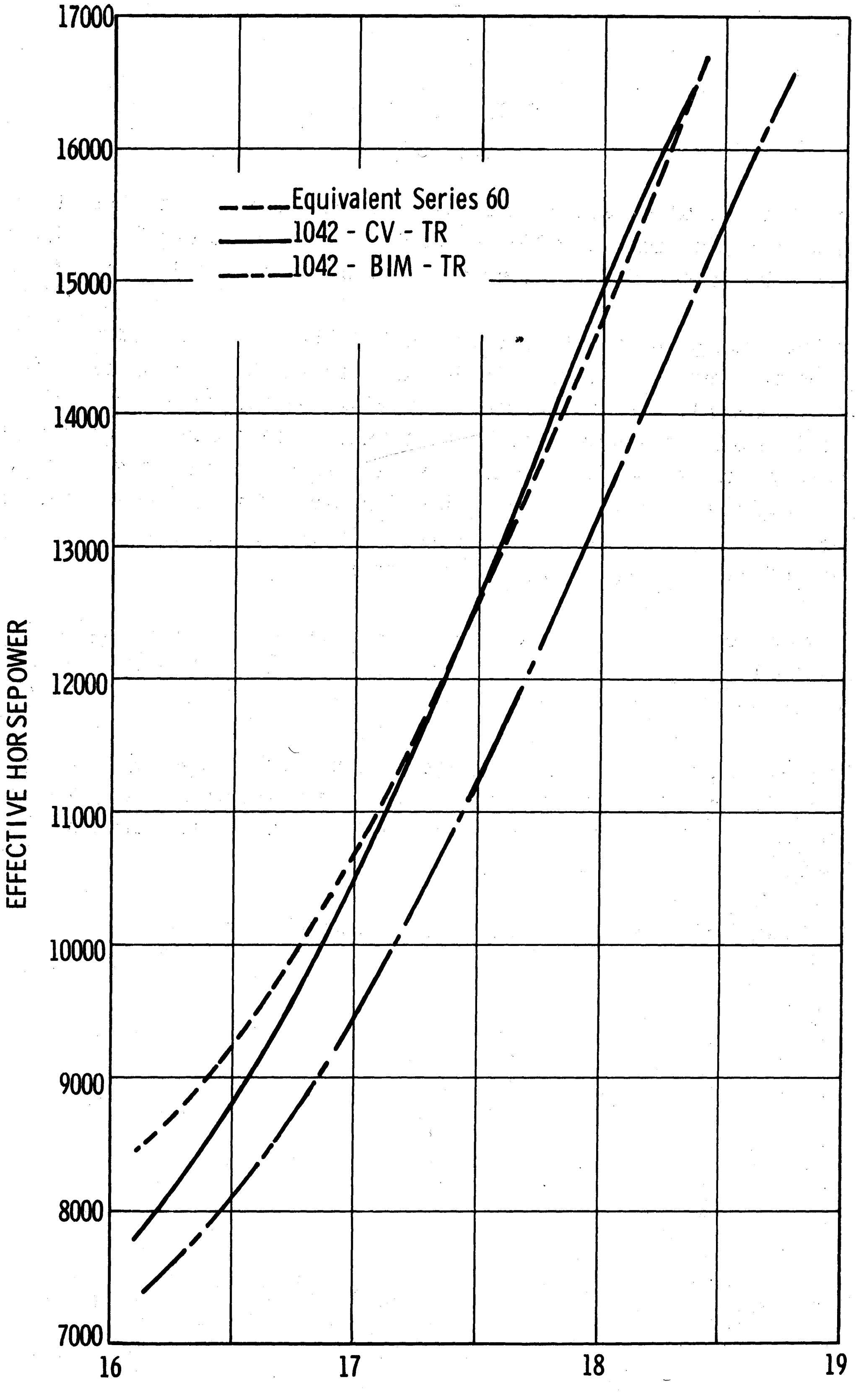

Figure 44. Comparison of EHP with Series-60.

CONCLUDING REMARKS

Now that the bluntness of the static waterplane is not definitely considered to have detrimental effects, certain considerations regarding the waterplanes must be noted that may seem to contradict the conclusions of Appendix A.

It has been pointed out that in the 60% displacement condition the static waterplane was very blunt. But it was also shown that the dynamic waterplane (i.e., the waterplane when the ship is moving) was different than the static one. It is still considered necessary to have a fine leading edge to the dynamic waterplane if the flow is to be at all smooth past the stem. Exactly how far above the static waterplane the dynamic one will be cannot be determined until the vertical position of the bulb is known. And then the most rapid way of determining it may be to test it.

It must be emphasized that we are presently only contemplating a blunt static waterline. Much more will be known once the next bulbous bow as described previously, is constructed and tested.

The sacrifice of performance that is expected in the 80% and 60% displacements with the raised bulb is not considered detrimental to the object of this research, as it is hoped to develop the optimum hull configuration for the full load condition first.

In Appendix B are the hydrostatic curves of form.

ACKNOWLEDGMENTS

The authors wish to acknowledge the extensive aid given by Dr. H. C. Kim and Mr. N. E. Rabe. Mr. Alan Gilbert's aid with the computer programs, and Mr. Bernard Young's many calculations and drawings are greatly appreciated.

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APPENDIX A

COORDINATED ANALYSIS OF UNRELATED BULBOUS BOW EXPERIMENTS

The following conclusions concerning the design of bulbous-bow hull forms are based on the experimental test results of a group of unrelated models tested at the Ship Hydrodynamics Laboratory of The University of Michigan. For each of the models included in this study, several bulbous bows were designed and tested. The models represent ships with tanker proportions. The design speed-length ratio for these models is in the range of 0.62 to 0.72, and unless otherwise specified, it is within that range that the results are valid. Resistance tests were carried out for each model without any bulbous bow also; and comparisons to this "bulb-less" test are made.

BLOCK COEFFICIENT: The models with the generally most undesirable resistance characteristics have C_B greater than 0.80; those with the best characteristics have C_B less than 0.77. All others, having exhibited resistance characteristics between the extremes, have C_B between 0.77 and 0.80.

LENGTH/BEAM: Aside from those ships which were "artificially" lengthened by the addition of a midbody, there is apparently no preferable ratio of length-to-beam. Those models with the best resistance characteristics included those with the extremes in L/B: 6.9 and 7.4. The one lengthened ship (L/B = 8.2) exhibited moderate resistance characteristics in both full displacement and ballasted (60% displacement) condition.

DISPLACEMENT-LENGTH RATIO: There is very little variation in the ratio $\Delta/(.01L)^3$ among all the models (164-179). Several of the best models are in the middle of the range (168-171); some of the poorer have ratios slightly higher. Generally, for normal tanker proportions, it is of small consequence.

BULB VOLUME: The ratio of (bulb volume)/(ship volume), including fairing, appears to be one of importance. In full displacement condition the reduction of C_R due to the addition of the bulb, for all models, is between 5 and 15%. Within this small range there was no significant order relating the reduction of C_R to the bulb volume. The bulb volume ratio in full load may appear to be inconsequential (within a reasonable range) due to the greater immersed depth. In ballasted condition, however, meaningful correlations were observed. Those models exhibiting the greatest reduction of C_R (35-45%) have the largest bulb volume ratios (.007-.008); whereas those models having the least reduction of C_R (10-20%) have the smallest volume ratios (.004-.005).

It should be noted that some of those models having shown the greatest reduction of C_R at higher speeds also show the greatest increase of C_R at lower speeds (as much as 120%). But at the lower speeds, C_R is a small part of C_T, unlike the higher speeds.

BALLASTED TRIM: It is difficult to see that certain trim-length ratios are desirable. Several of the models exhibiting the poorest resistance characteristics have ratios near the extremes (.007 and .018). All others have ratios in between these values. However, it is felt that insufficient information is available to arrive at any definite conclusion, or even to state that the trim is inconsequential.

BULB SECTION AREA: In full displacement condition, there is definitely a correlation between the larger area ratio (A_{FP}/A_M) and the decrease in C_R at the higher speeds. Apparently the larger bulbs (in full load, 11-13%) are very close to optimum, compared to the smaller bulbs (8-9%).

In ballasted condition the same correlation is apparently applicable, where 15-18% of the ballasted midship area is the most favorable. The only major exception is for a model having an area ratio of 20% in ballasted condition. In that specific case the decrease of C_R is not comparable to others having large bulbs. This may result from the fact that it is beyond the optimum size for this particular ship.

FULL LOAD WATERPLANE SHAPE: For all cases included, there are no outstanding hollows in the full load waterlines. The model exhibiting the poorest resistance characteristics (highest R_r/Δ) in full load has the fullest run, i.e., the tangent to the waterplane at the 1/2 station (based on 20 station LBP) forms the greatest angle with the C_L. The models having the lowest R_r/Δ have slight hollows in the waterplane, and have the smallest 1/2 station tangent angles. The decrease in C_R is not entirely consistent with the overall residual resistance. The ship having the greatest R_r/Δ also shows one of the greatest reductions in C_R (compared to the bulb-less hull). This merely indicates that a hull of poor design can be improved more easily than one of originally good design.

The finer entrance is a direct result of the finer prismatic and the finer block; and this then substantiates the observations under "block coefficient" above.

BALLASTED WATERPLANE SHAPE: The two models having the lowest R_r/Δ are characterized by extreme hollows in the ballasted waterplane; the 1/2 station tangent angles are less than 5°. These are the same models as those having slight hollows in full load.

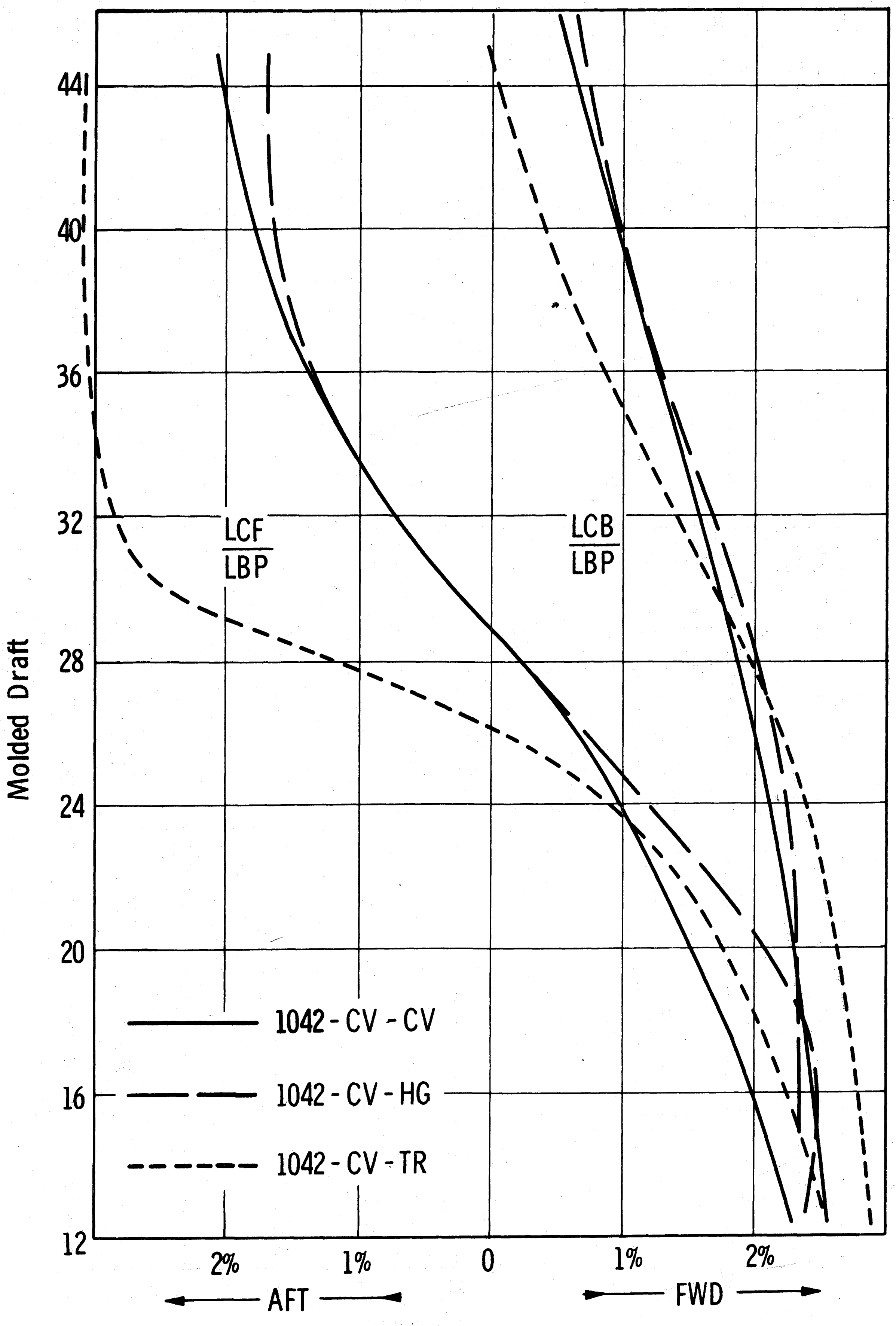
Two of the three models having the greatest reduction in C_R (compared to the bulbless hull) have the extreme hollow; and the third has a noticeable hollow with a tapering waterplane.

Of the two models having the highest R_r/Δ (and the least decrease of C_R), one has almost no hollow in the waterplane, and the other has some hollowness, but a very blunt stem.

Thus, for ships of tanker proportions, the following recommendations are made for the hydrostatic parameters: Block coefficient should be as low as possible, and should definitely be less than 0.77. The length/beam ratio and the displacement-length ratio may be chosen within the range of at least normal proportions. Bulb volume ratio should be as high as feasible (.007-.008). Bulb area ratio should be in the range of 11-13%. The most desirable ballasted trim should be experimentally determined for each model. Fine waterline entrances with hollows in the forebody planes are most desirable for utilization with a bulbous bow.

APPENDIX B

CURVES OF FORM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40
36
32
28
24
20
16
12
LCF/LBP
LCB/LBP
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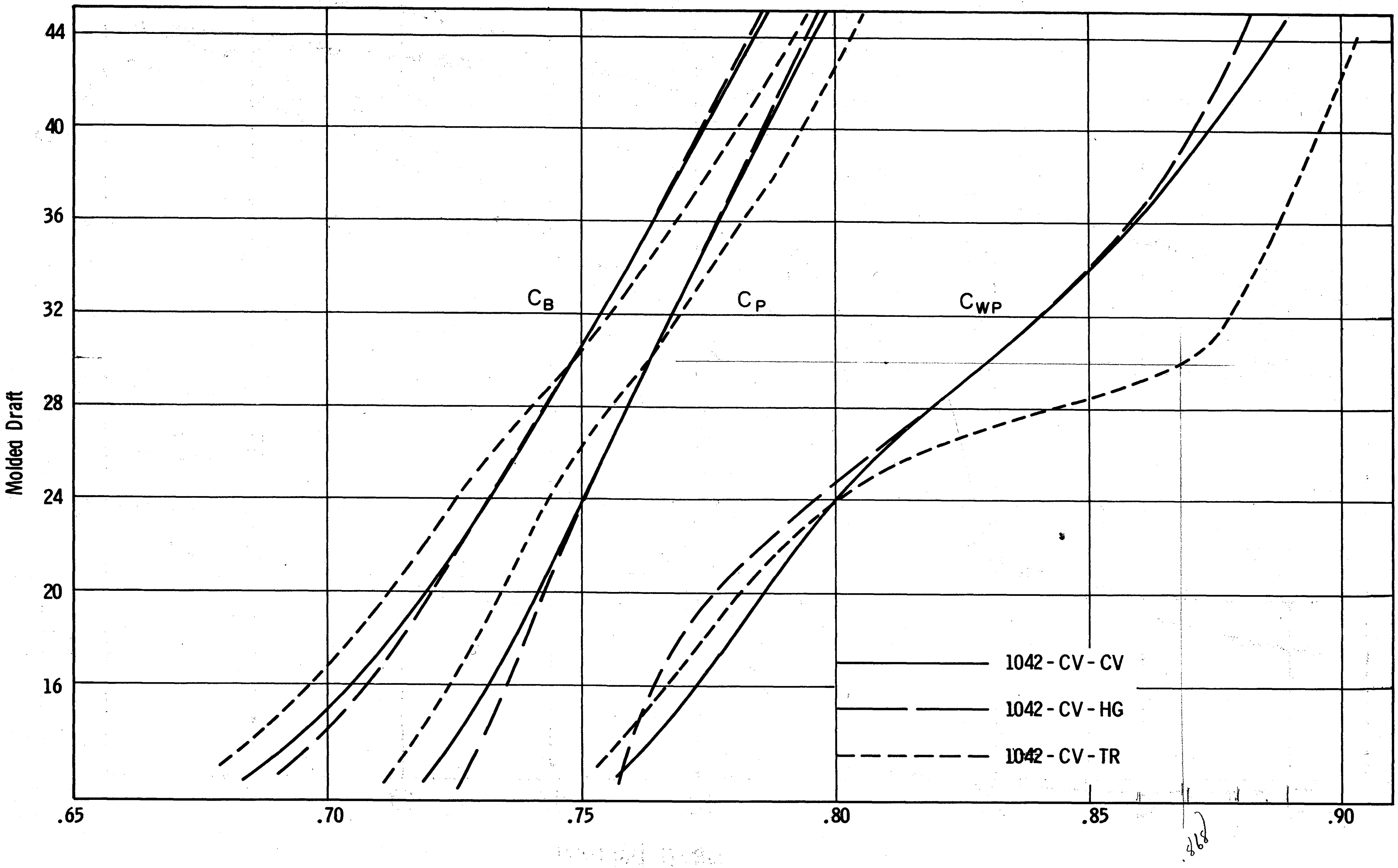
1042 - CV - HG
1042 - CV - TR
2%
1%
0
1%
2%
AFT
FWD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36
32
28
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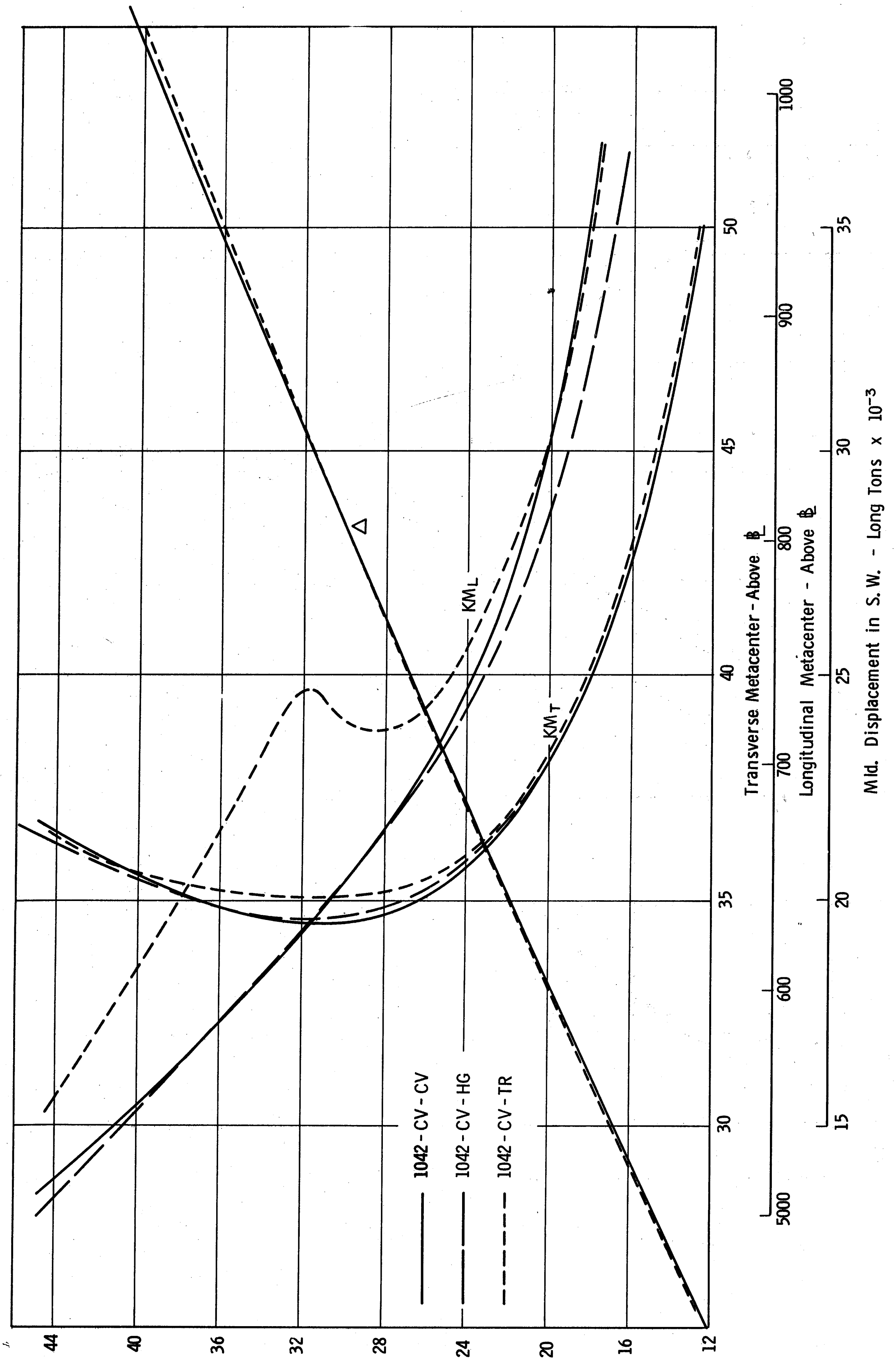
20
16
.65
.70
.75
.80
.85
.90
C_B
C_P
C_{WP}
1042-CV-CV
1042-CV-HG
1042-CV-TR
.898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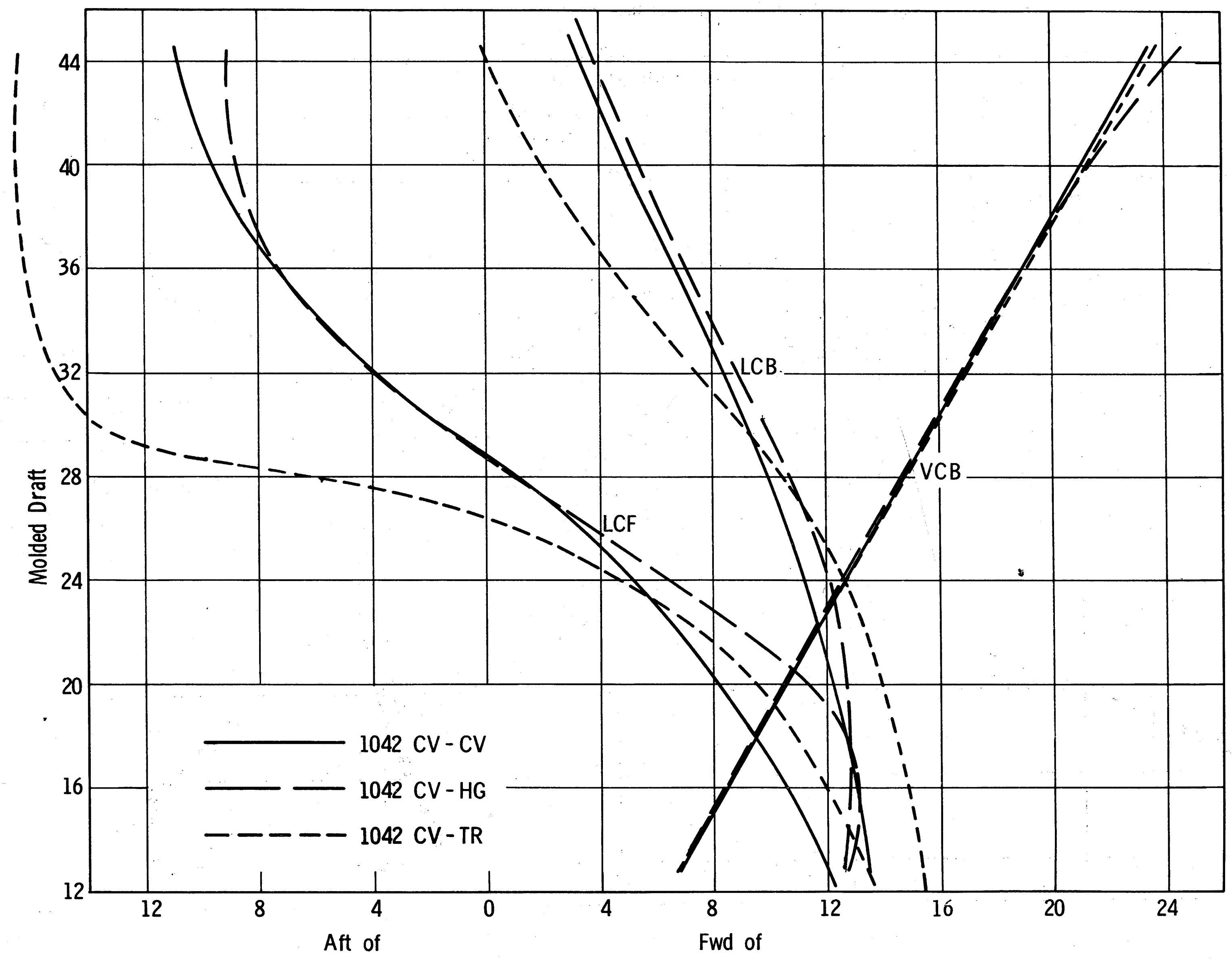
16
20
24
28
32
36
40
44
Transverse Metacenter - Above ℄
30
35
40
45
50
Longitudinal Metacenter - Above ℄
5000
600
700
800
900
1000
Mld. Displacement in S. W. - Long Tons x 10^{-3}
15
20
25
30
35
KM$_L$
KM$_T$
Δ
1042-CV-CV
1042-CV-HG
1042-CV-TR


Molded Draft
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36
32
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16
12
12
8
4
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4
8
12
16
20
24
Aft of
Fwd of
LCB
VCB
LCF
1042 CV - CV
1042 CV - HG
1042 CV - TR