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Discussion on "latest dredging practice" by ole p. Erickson, Proc. ASCE, 87 (WW3), p. 142, (August 1961), Reprint No. 181 (62-24)

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DISCUSSION

MEAN DIRECTION OF WAVES AND WAVE ENERGY[a]

Closure by Omar J. Lillevang

OMAR J. LILLEVANG,[6] M. ASCE.—Dunham related the alignment and history of natural shoal forms and raises the point that shoals resulting from the influence of artificial structures may eventually develop similarities to the natural. Sandy Hook, at the north end of the New Jersey Coast, may well be nature's analogy of the breakwater tip shoal, particularly at a location at which the energy flux of major river flows is an element compounding the complications of wave action. Sandy Hook's recurved end may exist to a considerable extent because of the river effect and also due to the local chop of waves generated in Lower New York Bay by offshore winds. Neither complicating effect is a possibility at the locations discussed in the paper.

The Santa Barbara record appears to support Dunham's suggestion that as inshore contours move eastward in an accretion area the outline of deposits, or the alignment of shoal deposits, is forced seaward. Fig. 6 and the preceding remarks concerning it, bear on this concept. The bottom contours paralleling the Santa Barbara breakwater did move seaward quite consistently after the structure was built, about 1930 to 32. Whatever trend lines one might elect to draw through the individual survey data plotted on Fig. 6 would show a swing to the seaward of the shoal limits. It is planned that expected future shoal developments at Del Mar Harbor will be removed frequently to maintain navigation, as they have been at Santa Barbara. Thus, except for the different wave exposure, the Oceanside project may be comparable to the Santa Barbara one.

Bruun correctly takes the writer to task for referring to energy as a directed phenomenon. Here simplification may have been overdone to avoid a title for the paper that might otherwise have read "Mean Direction of Waves and of the Flux of their Energy at Coasts and Barriers."

It is encouraging that the problems discussed in the paper have brought out discussions from the heads of two of the world's exceptional coastal engineering laboratories. Their independent references to model study for development of sound theory is impressive. The hope exists that such work may be recognized as a very useful thing for allocation of research funds. The successful outcome of such research would be a reduction of the empirical and an increase in the rational practice of coastal engineering.

Jordaan's interest in the uniform longshore current is not intensely shared by the writer. However, probably this is because none of his experience has encountered such a beach-paralleling current swift enough to move bed load, as would be done by a stream or by swift tidal flow at an estuary. On many occasions along the California coast when waves were breaking obliquely to a

[a] March, 1960, by Omar J. Lillevang (Proc. Paper 2423).

[6] Vice Pres., Leeds, Hill and Jewett, Inc., Cons. Engrs., Los Angeles, Calif.

long, straight shore, the writer has noted pronounced longshore currents in the surf zone that were set in the opposite direction one might expect from the oblique incidence of the waves.

Jordaan has suggested that "the Q-factor equation might be improved by utilizing w $\sin^2$ I rather than w Sin I Cos I." As derived by the Los Angeles District of the Army Engineers, the latter resulted from the following reasoning:

1. Energy content per unit length of wave crest offshore is representable as w;
2. At the shoreline the energy content per unit length of wave crest has been modified by refraction, diffraction, shoaling depths, island screening, and so forth and is, thus, represented by w E;
3. The wave crests at shore may not be parallel to shore. Thus, the energy content per unit length of shoreline, if I is the angle between crest and shore, is computed as w E Cos I;
4. The flux of the above unit energy per foot of shoreline has an incidence angle at the littoral zone, I: Thus, the component of that flux longshore is the product of the energy flux and the sine of the incidence angle, or w E Cos I Sin I, that was confusingly written in the paper as w E Sin I Cos I, and properly reduced to $\frac{1}{2}$ w E Sin (2 I).

The writer closes this discussion with sincere appreciation for the criticisms and additions that Dunham, Bruun and Jordaan have contributed and with an expression of hope that correspondence or presentations in the literature of their contributions may follow.

DESIGN CONSIDERATIONS FOR CALIFORNIA MARINAS[a]

Discussion by Omar J. Lillevang

OMAR J. LILLEVANG,[21] M. ASCE.—The author has emphasized that the prediction of direction and magnitude of waves by rational refraction and diffraction computations can be made and that harbor designers should learn and use these procedures, or commission those already skilled to perform them. Another wave-induced phenomenon is more difficult to predict mathematically, unless the harbor channels and basins approximate simple geometrical forms separated by sharp constrictions or other simplifying limits. The problem referred to is resonance. At one California small craft harbor, waves of certain critical frequencies, which come in the entrance channel from the ocean, pass a constricted side entrance to a large mooring basin. They induce a resonating surge in the basin which has torn fittings from boats, broken mooring hardware on slips, severed lines and otherwise made the basin unattractive to owners of boats who would willingly rent moorings there. With hindsight, it is clear that rather simple model studies of the harbor in the design phase might have avoided what may now prove expensive, and certainly will be inconvenient remedial measures. Not the least expense, by any means, is the bad reputation the surges have made for the harbor among boat owners.

At most harbors on an open coast, wave approaches from many directions are to be expected, because the waves are propagated in storm centers which may occur anywhere in the oceans. Thus, in general, there is no alignment of an entrance which will not at some time have waves moving to the inner areas with only slight attenuation of their offshore characteristics. It follows that some type of energy absorptive works at the harbor end of the entrance is desirable. A gently sloping beach directly across the direction of the channel is superior to other devices, but often cannot be provided because of land limitations or location of continuing channels to the inner harbor areas. Vertical barriers should be avoided at all costs, because they almost totally reflect the waves back on themselves and the result is a series of standing waves in the vicinity which have twice the amplitude of the waves prior to reflection. Care must be taken with sloped boundaries, when they cannot be as gentle as a beach, that they be built of rough, porous rubble or otherwise be absorptive, lest they reflect the incident waves with results nearly as drastic as those of the vertical barrier.

Providing maneuvering room for sailing craft without auxiliaries can be an expensive luxury. The number of such boats is diminishing rapidly and the limited number of slips needed to provide for these severe space-consuming considerations should be located along main channels or basins. Likewise, to

[a] November, 1960, by James W. Dunham (Proc. Paper 2658).

[21] Vice Pres., Leeds, Hill, and Jewett, Inc., Cons. Engrs., Los Angeles, Calif.

adopt a lenght-to-width ratio of slips to allow for the occasional "square" hull imposes a cost in terms of water area consumed which is unjustified. It seems reasonable to expect the owner of the abnormally wide boat to meet the greater rental expense of a longer slip and thus have the required width.

Fig. 2 is particularly interesting when compared with the record of nearly 9,000 boats owned in Orange County, the area in which the famed Newport Bay is located in southern California. In planning for two new harbors, each to provide for more than 500 boats in initial development, the writer analyzed punch card data for every boat more than 15 ft long registered in Orange County in 1960. Fig. 14 presents those data by percentage distribution of lengths, and the San Francisco Bay Area data are drawn as a dotted line for comparison.

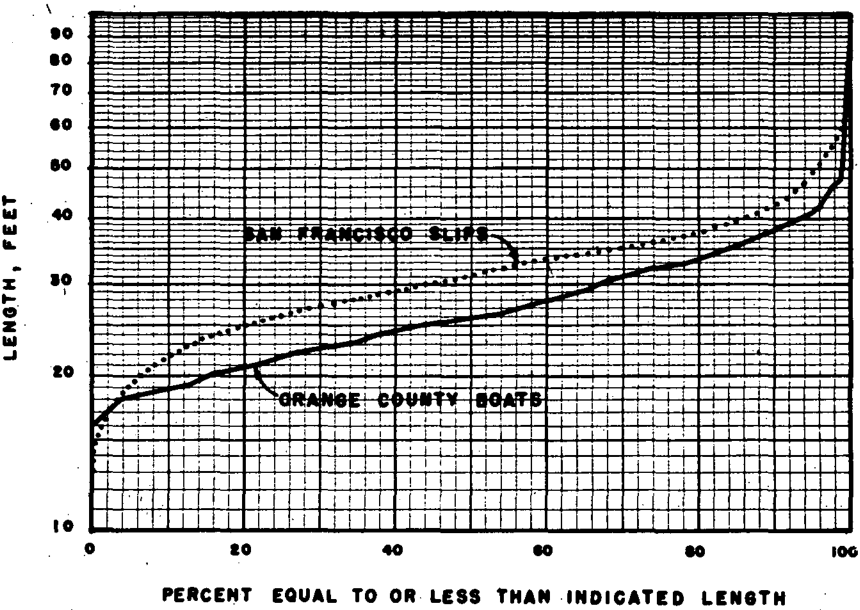

FIG. 14.—PLEASURE CRAFT REGISTERED IN ORANGE COUNTY IN 1960

Apparently a marina designed for southern California size distributions would not fit the demand in San Francisco Bay.

Analysis of the same punch card record for Orange County boat ownership reveals that virtually all boats more than 27 ft long are moored in marinas the year around, and practically none less than 19 ft are kept in slips at all, but are trailered from dry storage to the water. Marina operators report that the expense of managing slip rentals for boats less than 25 ft long is disproportionately high because of the readiness of their owners to terminate rental agreements and remove their boats to dry storage. The situation is perhaps comparable to the difficulties of managing small furnished apartments where occupants are highly transient. A difference lies in the insistence by owners of the small boats that they pay no more rent per foot of slip than do the owners of larger boats. One compensating factor is that the cost of land, whether already water-covered or dredged to make installation of mooring feasible, is porportionately greater for the larger slips than for the smaller, because broader expanses of maneuvering areas per slip are necessary for the large boats. Whatever distribution of slip sizes one may adopt after studying the local statistics, the majority of boats will be in slips longer than the individual needs, if the usual rule is enforced that no boat may extend into the approach channel. Unless a large marina is built "from scratch," it is not economical to break the slip sizes down to as small as 5 ft increments of length. A current design for 575 slips in a southern California harbor is distributed to fit the Orange County record as follows:

Length of Slip, in feet	Number	Percentage Equal or Less than Length Tabulated
25	250	44
33	205	79
43	90	95
51	20	98
More than 51	10	100

In a marina, the floating facilities are more or less open to the walking public, which includes the unsteady on foot who are young, or old, or wearing spike heels, or full of "good cheer." Perhaps none of these should be on the floats unassisted, but they are. Thus the stability of the floats is important and quickness of response by floats to wave action, or boat impact, or any other moving load, deserves careful consideration. Also, working of joints in highly flexible structures often becomes a maintenance problem. Flexibility of deck systems and light weight, small displacement flotation elements should be avoided to the extent that cost and connection stresses will allow. With framed decks bridging from pontoon to pontoon, it is reasonably easy to design for stability against horizontally applied loads on the slip fingers and walks. The deck system can be cross-braced and act as a deep truss laid on its side. However, it is difficult to design a system so shallow, vertically, to resist torsional displacement with eccentric vertical loading. Assemblies of concrete boxes, tied together with plank facias and with their upper surface serving as the walking deck, have been used for slips and develop great assistance to torsional displacement. This is principally because their monolithic performance as a deep girder resists distortion under any eccentric loading which would not otherwise sink the assembly.

Recently the writer went to their established marinas, two of which the slips were wood frames on lightweight floats and the third was a continuous concrete float system. The more reactive wood frame fingers, 40 in. wide, deflected 2 1/8 in. under a load of 165 lb, applied vertically 6 in. in from one edge. The stiffer wood frame, 44 in. wide, deflected only 3/4 in. under the same load, but for an all-concrete float system slip finger only 34 in. wide, it required 280 lb 4 in. in from one edge to develop a 1/8 in. deflection. The relative stability was even more dramatic under quick load, applied as a rapid shifting of weight from one side to the other or by jumping from the decks of boats in the slips to the walking surface of the floating slip enclosures.

LATEST DREDGING PRACTICE[a]

Discussion by Charles E. Behlke, John B. Herbich, and Alf. H. Sorensen

CHARLES E. BEHLKE,[2] M. ASCE.—The writer would like to offer a few statistics and practices of dredging in The Netherlands that are interesting to compare with those of American dredging given by the author. Most of the information provided was given to the writer by Ir. H. T. den Breejen.

Some of the most striking differences between Dutch and American dredging practices are illustrated by the following figures on Dutch dredges for the year 1957.

Bucket Dredges	265
Suction Dredges	204
Hopper Dredges	18
Dripper Dredges	1
Barges to haul dredged material	1231

The Dutch use a relatively large number of bucket dredges. This is in sharp contrast with American practice where a bucket dredge is seldom used except, perhaps, in mining operations. Suction dredges in Holland are sharply on the increase. The figures also indicate the relatively large number of barges used.

Generally, Dutch dredges are smaller than those in America and the crews are frequently housed on the dredges of any size.

Dutch bucket dredges are usually rated by the size of the buckets. Hence, on a 400 l dredge, each bucket would have a capaicty of 400 l. On the average, Dutch bucket dredges cost approximately 3,000 Dutch Guilders per l of capacity (abbreviated f. 3,000) (Approximately f. 3.7 = $1.00). On the basis of f. 3,000 per l per bucket, the following figures would apply approximately to a 400 l bucket dredge.

Initial cost of dredge = 400 (3,000) = f. 1,200,000

1. Maintenance including dry dockage at 8% of initial cost per yr	= f.	96,000
2. Insurance at 3% per yr	=	36,000
3. Sinking fund at 6% interest and 25 yr life	=	21,900
4. Interest on initial cost at 6%	=	72,000
5. Overhead at 3% of initial cost	=	36,000
Total fixed cost per yr	= f.	261,000

[a] February 1961, by Ole P. Erickson (Proc. Paper 2729).

[2] Assoc. Prof. of Civ. Engrg., Oregon State Unov., Corvallis, Oreg.

Operating costs per week of operation

1. Wages		= f.	1,000
2. Social insurance			500
3. Coal			2,000
or diesel	f. 500 per week		
4. Store sundries			800
Total weekly operating costs			
for a coal operated dredge		= f.	4,300
for a diesel dredge		= f.	2,800

The average Dutch dredge works approximately 30 weeks per yr, so the weekly charge of operation is the direct operating cost plus the fixed costs prorated more than 30 months. This amounts to f. 4,300 + (f. 261,000/30)

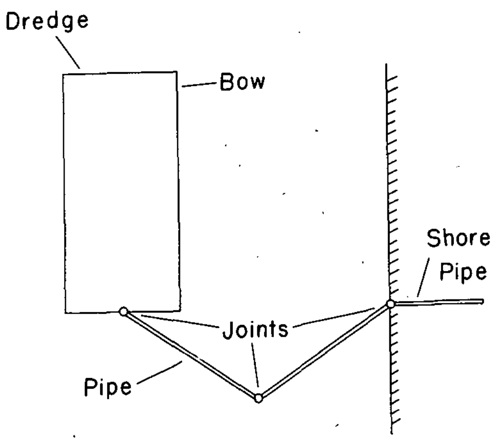

FIG. 10.—THREE JOINT PIPE CONNECTING METHOD

= f. 13,000 per week of operation. Of this, manpower costs only f. 1,500, or approximately 11.5%. The best manpower can be sought because there is little difference in cost between good and poor crews. However, earnings are almost directly a function of how well the dredge is operated. The relatively small cost of labor also explains the fact that Dutch dredges are usually less automated than American dredges.

Some of the Dutch methods of operation are of considerable interest. The writer observed that no anchors were handled by booms on Dutch dredging operations. Auxiliary boats usually lift and place the mooring anchors.

Many Dutch dredging firms like to use the three joint method of connecting their pipe between the shore pipeline and the dredge. This is shown in Fig. 10. This method only works well in calm water, but it allows the dredge to cover a great area without handling any piping.

Occasionally, the Dutch use cylindrical pontoons to support the floating pipeline. These cylinders are concentric with the supported pipe as shown in

Fig. 11. This type of pontoon works especially well in rough weather, but it has a glaring disadvantage because leaks in the pontoon are quite difficult to find.

Another type of pontoon is shown schematically in Fig. 12. This type of pontoon has a removable top on each segment allowing several to be stacked like pans when the dredge is moved.

Few of the Dutch dredges are self-propelled. They feel that, because during operation the dredge is only moved a few times a day, it is cheaper to have a small tug provide the necessary propulsion, thus saving space and capital investment.

Some of the bucket dredges have hoppers into which the dredged material is dumped and from which it is picked up by a suction line and put into the discharge pipe. This is essentially a dredge within a dredge. It would seem better to make the dredge a suction dredge to begin with, but the Dutch feel that for many materials such as clay, the bucket dredge works more efficiently than a suction or suction cutter dredge.

The writer has recently received information that IHC Holland, a combine of six Dutch companies that builds and repairs ships and dredges, is presently constructing two cutter suction dredges of the stationary type, each having

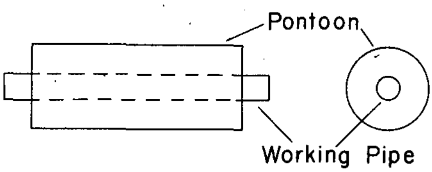

FIG. 11.—CIRCULAR PONTOON

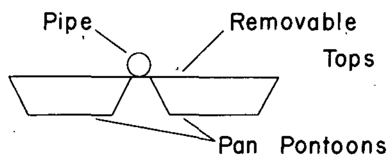

FIG. 12.—PAN TYPE PONTOON WITH REMOVABLE TOPS

5,300 hp. They will be the largest dredges ever constructed by these Dutch contractors. The dimensions of these dredges are 190 ft by 44 ft by 14 ft.

It is also interesting to note that IHC Holland operates a testing laboratory in Delft that seeks improvements in dredging methods. Here models are constructed and tested under controlled conditions. Much valuable information has been obtained in this laboratory. The writer knows of no other laboratory of this type in the world.

The Dutch construct many dredges and own and operate so many that during the depression of the 1930's, there was not enought room in home harbors to moor all of their unused dredges that were working throughout the world and at home at the beginning of the depression.

While in Holland, the writer witnessed the aftermath of a serious dike break at Tuindorp-Oostzaan in January, 1960. Immediately following the dike break, five large suction dredges that were in the vicinity were moved to the afflicted area. As soon as the dike repair was affected, the dredges placed their suction lines in the flooded area and proceeded to pump out the flood water, dredging only water and no soil.

The new subway for the City of Rotterdam will pass through the center of the business district. The Dutch will not tunnel to accomplish this construction, but will work in a large open trench that will be created by dredging.

When the Maas tunnel was constructed under the Maas River in Rotterdam, the construction was performed without tunnel. A trench was dredged across the river and precast tunnel sections 200 mi long were floated into position and sunk in place. These sections were then connected in a watertight manner and the water removed, thus creating a tunnel with dredges performing the necessary excavation.

The preceding examples illustrate the fact that the Dutch, besides being a trading nation, are also a dredging nation. The writer does not wish to imply that Dutch methods are better or worse than American methods. The two cannot be compared because the financial structure in the two countries with regard to labor is so completely different. This information has been offered to indicate some of the practices in another country that are quite important in the world of dredging.

JOHN B. HERBICH,[3] M. ASCE.—The principal intention of the writer is to supplement the paper and to briefly summarize the current research program at Lehigh University aimed at improving the efficiency of dredge pumps, particularly for pumping silt-clay-water mixtures.

In analyzing the history of hydraulic dredging (the principle of dredging by means of a centrifugal pump), mention should be made of the existence of a hydraulic hopper dredge General Moultrie[4] in the United States in 1855. The dredge pump with an impeller of approximately 6-ft diam, was revolving on a vertical axis, its 19-in. diam suction pipe with a bell-mouth lower end resting on the channel bottom, while it discharged the dredged material directly into a "hopper" in the vessel. The pump was moved by the steam engine which was also used to propel the ship. The records indicate that on the average, 328 cu yd of material was dredged per working day.

It appears, therefore, that the hydraulic or suction principle was first used for dredging in the United States. The General Moultrie became a casualty of the Civil War, and dredging by hydraulic means was not tried again in the United States until 1871, when a steamer, Henry Burden, was converted for suction dredging and used in improving the mouth of the St. John River, Fla. The dredging equipment of the Burden consisted of a 9-in. centrifugal pump, a 6-in. suction pipe on each side, tee-connected to the single pump, and two 6-in. pipes tee-connected to the 9-in. pump discharge.

A great number of hopper dredges were either purchased or built by the Corps of Engineers between 1891 and the presnet time, culminating with construction of hopper dredge Essayons in 1949. The Essayons is 525 ft long, has a hopper capacity of 8,000 cu yd, maximum dredging depth of 60 ft, and is equipped with two 36-in. suciton - 32-in. discharge dredge pumps, 1,850 hp each. Construction of new hopper dredges by the Corps of Engineers since 1936 was aimed to replace the older dredges. Because of the improved efficiency of the modern dredges with their larger hopper capacity, greater speed, and better maneuverability, the number of dredges operated by the Government

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[4] Journal, Franklin Inst., Vol. 32, 3rd Series, No. 6, December, 1856.

has decreased. The complete history, development, and operation of the Corps of Engineers dredges are described elsewhere.[5]

In describing various hydraulic dredges, mention might also be made of the "portable" type dredges. These are hydraulic pipeline cutterhead type with main dredge pump driven by a high-speed Diesel engine through a reduction gear. The hull is approximately 52 ft by 20 ft by 4 ft, and they can be disassembled and transported overland to another location. The pump has a 13 1/4 in. suction and 12-in. discharge, and is operated by a 260 hp motor.[6] It has a maximum digging depth of 26 ft and is capable of pumping distances up to 3,000 ft, with outputs varying from 100 to 300 yd per hr in normal materials. The protable dredges which were built for the Indonesian Government may be used to great advantage by local authorities or the contractors.

The trend is away from crew's quarters on dredges, however, the current practice on large dredges is to provide quarters sufficient to permit the dredge to operate on a 24-hr schedule when necessary. Also, the dredges operating in remote areas, such as the two dredges recently built for the Brazilian Government, contain living quarters for forty-five officers and men.

The author mentions that the discharge vane angles at tip vary between 20° and 30°, and entrance angles vary 16° to 24°. The writer finds that the discharge vane angles varied anywhere from 22 1/2° to 35°, and even 67° in the older dredge pump. However, the trend seems to be to reduce the discharge vane angles; for example, the recently built dredge S. S. Zulia in Japan has a discharge angle of 22 1/2°. Such low angle is usually recommended for pumps handling water,[6] and it has not been used on dredge pumps until recently.

The writer finds that the entrance vane angles vary from 37° to 40°; the S. S. Zulia and Essayons having an angle of 45°.

The author mentions a number of empirical formulas for computing friction in pipelines and rightly states that unless reasonably correct allowances are made, the computations may be misleading. There appears to be a great research need to determine the effect of concentration of the solids in water, the grain size, and distribution on the friction factor f. Observations in the laboratory indicate that when the concentration of solids is low (up to 1,200 g/l), the resulting mixture is essentially water with solids in suspension, and the solids settle readily. However, when the concentration is high (up to 1,400 g/l), the mixture appears to be homogeneous. It has properties of a non-Newtonian fluid and the solids do not settle readily. The author mentions that the dredge may pump up to 40% solids. This is misleading unless further clarified whether the percentage solids is by weight or by volume. This, too, may be misleading unless the "solids" are defined. "Solids" as they are sometimes referred to in dredging practice, are actually comprised of the dry voidless grains plus the water which occupies the void spaces between grains. To avoid confusion, these could be called "solids" as "in situ" material, or "bottom material." The density of material expressed in grams per liter with its percentage by volume equivalent will be compared. The total weight may be expressed as

$$x\ (S.G.) + (1000 - x)\ (1) = 1400 \qquad (1)$$

5 "The Hopper Dredge," by F. C. Scheffauer, Editor-in-Chf., U. S. Govt. Printing Office, Washington, D. C., 1954.

6 "Centrifugal and Axial Flow Pumps," by A. J. Stepanoff, John Wiley and Sons, Inc., New York, 1948.

in grams per liter, in which x = cubic centimeter of true solids and S. G. = specific gravity of solids.

Assuming the S. G. of solids = 2.60, x = 250 cu cm of true solids, leaving 750 cu cm of water. Thus the mixture contains 25% by volume of true solids.

A study was initiated by Lehigh University, under the sponsorship of U. S. Army Engineers, Philadelphia District, in 1958, with an object of improving the design of a dredge pump primarily for pumping silt-clay-water mixtures.[7] The affect of impeller design on pump efficiency was studied in some detail, particularly for the silt-clay-water mixtures. The discharge vane angle was varied between 22 1/2° and 35°, as well as the vane shape in the 1:8 scale model pump of Essayons dredge pump. It is not intended to present the model study results here, however, it should be indicated that the pump efficiency can be increased for 71% to 76% for 1,380 g per l, by changing the vane shape from a radial to an involute curve. Also, a change in the discharge vane angle can produce an increase of efficiency from 80% to 82%, and 76% to 83% for the prior-mentioned densities, respectively.

ALF H. SORENSEN,[8] A. M. ASCE.—The author is correct in stating that there is quite a bit of disagreement among dredge designers and builders as to the exact methods and procedures in this field both as to equipment design and its job application. A large majority of dredge people today (1961) do agree on the certain facts and some of these deserve the following comments.

Standard equipment built today by United States and European manufacturers feature 10-in. hydraulic pipeline dredges with hull sizes as small as 40 ft long, 14 ft wide, and 4 ft deep.

A large majority of modern dredge pumps presently built and used by major United States and Canadian contractors are single suction, volute type pumps with a one-piece pump case (or cast in two halves on larger dredges). The engine side head and the suction side head are lined with either Ni-hard or diamond alloy or abrasion resistant steel liners. Usually the pump case and the impeller is made from steel alloy castings from either one of the three general groups of steel such as (a) Abrasion resistant carbon alloy steel, (b) Chrome-nickel-molybdenum alloy steel, heat treated to a high Brinell hardness for abrasion resistant properties, and (c) Manganese steel (manganese steel is used when pumping gravel where the gravel causes impact hardening and increases the abrasion resistant qualities).

The pump heads are made from either cast steel or fabricated steel and do not require abrasion resistant qualities.

Fully lined pumps with fabricated cases are mostly used in applications where extremely abrasive sand and gravels are handled. When abrasion qualities become the principal design criteria, the pump parts which are in contact with slurry mixture is usually made of Ni-hard or of diamond alloy. These alloys, however, have little tensile strength and require an outer fabricated casing to obtain the necessary structural or tensile strength in the pump as a whole.

A check with the major contractors and pump manufacturers in the United States shows that a substantial majority of these disagree with the author in

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7 "Characteristics of a Model Dredge Pump," by J. B. Herbich, Fritz Lab. Report No. 277-PR. 31, Lehigh Univ., 1959.

8 Civ. Engr., Ellieott Machine Corp., Baltimore, Md.

the statement that fully lined pumps are more economical than conventional pumps.

In almost no practical case can a 10-in. dredge pump, pumping through 10-in. I. D. pipeline absorb as much as 600 hp. The majority of 10-in. pumps built today (1961) are designed for power applications between 100 hp and 400 hp. Most manufacturers have standardized on power range between 200 hp and 300 hp.

While the first cost of "built-up cutters" (or cutters welded together from castings or structural steels) is low, the long-term operation economy still

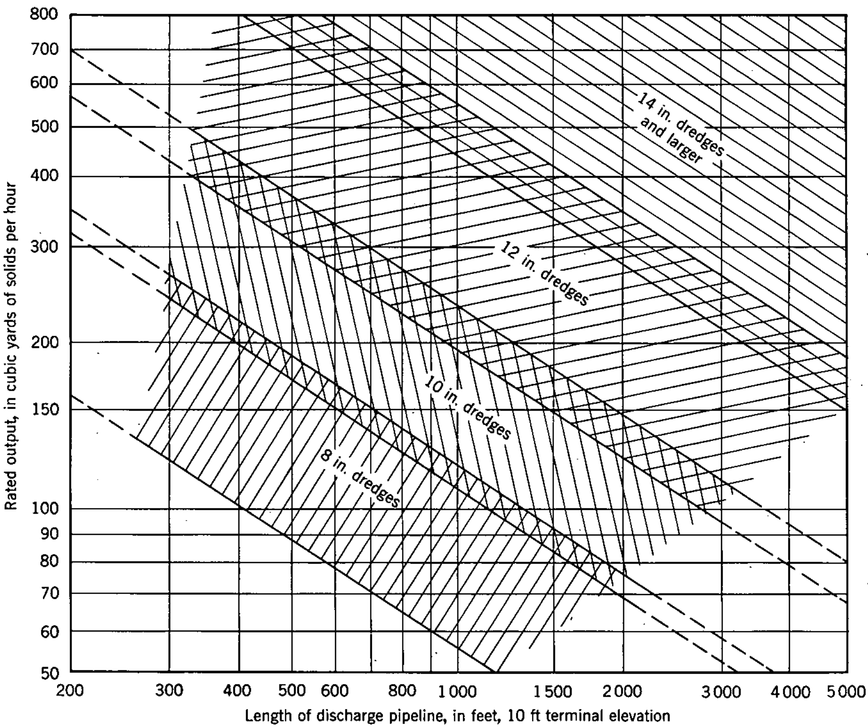

FIG. 1

favors the one-piece casting type cutter or with bolted-on blades. The one-piece cutter is still common.

Many designers recommend a cutter-shaft-thrust bearing to be installed as a separate unit outside and forward of the reduction gear, instead of as an integral part of the reduction gear itself. The reason is that when a bearing failure occurs, particularly on large dredges, it is more economical and timesaving to repair a separate thrust bearing instead of dismantling the entire reduction gear.

The direct suction pipe cutter drive was first used when hydraulic dredges came into common use in the United States in the beginning of the 1900's and already at this time was shown as an impractical design. The majority of contractors agree that this system is impractical from a maintenance point of view.

It should also be mentioned that recent years have brought into existence hydraulic drives for winches and cutters on dredges.

Mention is made of a 40% solids content in dredge pipeline slurry. It should be emphasized that this is a 40% solids by weight. It is, however, more common to use percentage by volume since dredge material is always mentioned by cubic yards or cubic meters and in this case the 40% figure would correspond with approximately 20% by volume.

It still remains to be proven theoretically as well as empirically that a direct suction pipe cutter drive gives a less water vacuum than a conventional cutter suction design.

The maximum production in a hydraulic dredge system is basically the function of the pipeline velocities, but this is only true where the dredge pump has a positive feed. That is, where the solids are mixed with the water at a predetermined ratio in front and above the suction inlet of the pump. On hydraulic dredges, however, the production is a function of the suction velocity and the ability of the cutter and suction head to feed the suction end of the ladder.

Fig. 1 shows how one manufacturer in the United States qualifies dredge capacities for smaller hydraulic pipeline dredges.

To base the capital cost of the new dredge on the horsepower alone may be quite misleading without qualifying whether it is a diesel, diesel partial electric, diesel electric or all electric dredge. A recent investigation of dredges built in the United States shows that dredges in the sizes 8-in. up to 16-in. vary in price from $180.00 to $350.00 per hp. Larger dredges in the sizes 20-in. through 36-in. vary from $275.00 and as high up as $600.00 per hp.

MARINE OIL TERMINAL FOR RIO DE JANEIRO, BRAZIL[a]

Discussion by Glenn B. Woodruff, Richard S. Winkler, and Joseph H. Finger

GLENN B. WOODRUFF,[10] F. ASCE.—The determination of berthing and mooring forces and the most efficient means of providing plays an important part in the economics of the design of a fuel handling pier. The author has given an excellent example of such a design. In this particular case, the location was such that wind currents and waves were a minor consideration. In the general case, the mooring rather than the berthing forces may be controlling.

Referring to Fig. 3, many designers prefer to set the breasting dolphins ahead of the hose handling platform so that the tanker does not come into contact with the latter. This eliminates any movement of the platform in reference to the submarine lines and reduces the amount of fendering required.

While the location is well-sheltered, the tanker may be subject to winds of 33 knots, currents of 1 knot and waves 5 ft high. Precise data for computing the forces resulting from these causes are not available. Especially with no more flexibility than is provided, the mooring forces against the fenders may well be at least in the same general order as those computed during berthing.

Fig. 6 gives an excellent picture of various conditions during berthing. The reduction coefficient allows for the distance between the center of gravity of the tanker and the point of impact. The author has neglected the hydrodynamic mass that may be considerably greater than the mass of the vessel, the division of the impact energy between the tanker and the structure and wind, wave, and current forces during berthing. This entire matter is complicated; the designer has the option of selecting such approach velocities and angles that will permit of great variation in the potential energy in the fender system. An analysis of the available literature leads to the conclusion that average and presumably satisfactory practice be expressed by

$$E = \Delta\ (0.004 - \Delta \times 10^{-8} \quad \ldots\ldots\ldots\ldots\ldots \quad (6)$$

in which E is the potential energy transmitted to the fender, in ft tons, and Δ represents the displacement of vessel, in tons.

The assumptions for various designs range from 0.50 times to 1.50 times those given by Eq. 6. The smaller value may be used when winds, currents, and waves are negligible; the larger ones when such conditions are severe. For the author's case of a vessel of 137,000 tons, E becomes 375 ft tons as against the 249 ft tons used.

In detail design of the fender system, the writer prefers to secure greater flexibility than the author proposes. This is especially important, because the

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[a] February, 1961, by H. W. Reeves (Proc. Paper 2733).
[10] Cons. Engr., San Francisco, Calif.

wave forces fall off rapidly with increased flexibility. To achieve such results the writer combines flexible breasting platforms with the fendering. While for this particular case the assumption of 100 ton bollards may be sufficient, the writer suggests that this should not be taken as a precedent by others. The designer has no control over the number and strength of the lines the tankers master will use. The U. S. Navy has adopted as a standard for aircraft carriers that are considerably lighter than the supertankers, bollards of 200 ton capacity.

None of the preceding should be considered a criticism of the author's design but rather as suggestions.

RICHARD S. WINKLER,[11] A. M. ASCE.—The terminal, while small, considering the number of berths available, is of interest in that a number of modern advances have incorporated in its design which give it much operational flexibility. The use of positive displacement flow meters, telemetering, and the successive use of pipe lines for differnet products without causing contamination are all of particular interest. Although the writer is more familiar with the structural aspects of such a terminal, a paper on these operation features would also be greatly appreciated.

In calling attention to the problem of choosing an approach velocity for a berthing ship, the author has emphasized one aspect of structural design subject to the most arbitrary sort of personal opinion. Perhaps this is due to a difficulty in discriminating between a reasonable design condition and an accident. At any rate, as tankers grow larger and as terminals must be placed in more exposed locations, these design problems assume greater importance.

Maximum Tanker Size.—The rate at which the size of the newest tankers has grown since World War II is awesome. No other class of ship has shown such rapid increase in size in so short a period, and it is likely that this rate of growth shall continue. At the present time (1961) two 130,000 DWT tankers are on order in Japan. These ships will cost $14,000,000 and will have a capacity of 900,000 bbls. Their proposed dimensions are: length overall 955 ft, beam 141 ft, depth 73 ft and draft 54 ft. These vessels will be driven by steam turbines rated at 28,000 shp at speeds up to 16 knots. Although the present surplus of shipping tonnage has lead a number of oil companies to suggest that perhaps the limit in the size of tankers has been reached, it can be seen from Table 2 that cost advantages exist for tankers as large as 200,000 DWT and possibly larger. There are a number of technical problems foreseeable in designing and constructing tankers of this size but these problems will undoubtedly be solved.

As seen from Table 2, in comparison with a 50,000 DWT tanker, a 150,000 DWT tanker is expected to reduce transport costs by one third. The increments of savings become proportionately less as size increases, but not to the extent that a 200,000 DWT tanker would appear to be a poor investment compared to a 150,000 tonner if the quantity of oil to be transported is sufficient to insure its full utilization and terminal facilities are available to avoid inordinately long port time. The larger the vessel, the greater the penalty for any idle time, probably the principal reason why larger tankers than 105,000 DWT have not yet been built.

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It is also interesting to note that for the 200,000 DWT tanker the cost of transporting a barrel of crude oil is 10 to 20 times the direct operating and capitalization cost for a typical terminal. It is thus unlikely that the cost of such a terminal will be any deterrent to the future use of such a tanker. The estimated dimensions for the 150,000 DWT and 200,000 DWT tankers are as follows:

Dimension	150,000 DWT	200,000 DWT
Length overall	1,000 ft - 1,100 ft	1,000 ft - 1,200 ft
Beam	150 ft - 170 ft	170 ft - 180 ft
Draft	51 ft - 55 ft	55 ft - 60 ft

Since the water depth at this new island wharf is 59 ft at mean low water, it should be possible to berth tankers as large as 150,000 DWT and still provide allowance for heave, squat, and some navigable clearance.

Ship Handling Flexibility.—The author is correct in stating that a modern oil terminal need not necessarily be designed as a single continuous pier or dock structure, and that an island wharf with local strong points has probably the lowest possible first cost. On the other hand, however, a continuous face wharf with regularly spaced bollards does have a certain flexibility not obtainable with a design in which separate structures are provided.

TABLE 2.—ESTIMATED COST OF CARRYING CRUDE OIL FROM THE PERSIAN GULF TO RIO DE JANEIRO, BRAZIL

Ship DWT Classification	50,000	100,000	150,000	200,000
Round Trips per Year	7	7	7	7
Total Fuel Cost per Year (M$)	580	1160	1570	1910
Operating Costs per Year (excluding fuel oil - M$)	840	1000	1160	1290
Interest and Amortization per Year (M$)	980	1420	1860	2300
Total Cost per Year (M$)	2400	3580	4590	5500
Total Crude Oil Capacity per Year (MBbls.)	2500	4900	7200	9700
Transportation Cost ($ per Bbl.)	0.96	0.73	0.64	0.57

Unfortunately there is still a lack of uniformity among tankers regarding fittings, mooring provisions, and cargo loading and discharging facilities. Such lack of uniformity causes considerable expense and difficulty to terminal operators. It would seem a worthy goal for the industry to achieve as much uniformity as possible in tanker design, particularly on the question of cargo unloading and discharge facilities. There is a normal tendency to increase manifold capacity in the larger ships. In the newer vessels the cargo handling system is generally divided into four sections, each with its own line to the manifold location. The main headers and crossovers are usually of 14 in. or 16 in. pipe, however, there may be a wide range in pipe sizes as shown by the five 16-in. lines of the Esso Gettysburg (47,400 DWT) and the four 12-in. lines of the Universe Leader (85,500 DWT). Tankers built by oil companies and their affiliates usually have a greater capacity in the basic cargo handling facilities as well as more special cargo handling equipment than do those built for charter or speculation. The normal location for the manifold connections is from 45% to 50% of the ships length from the bow. All of the present tankers except those of the Universe Leader class have a single manifold. These have two manifolds located approximately 46% and 57% of the length from the bow. The connections vary from 2 ft-6 in. to 5 ft-0 in. above the deck and from 10 ft to 20 ft back from the ships side. The principle purpose of having the two manifold locations appears to be only to offer more flexibility in making connection to the wharf's pipe lines.

There is also some variation regarding the number and types of mooring lines. The newer ships are equipped with a variety of chocks and winches for wire or manila rope, or both. Power winches are located on the bow and stern and some ships have constant tension winches. The newest Esso tankers carry $1\frac{1}{2}$ in. steel cable with a breaking strength of 50 tons, while the braking power of the winches is 45 tons. The W. Alton Jones has six mooring winches with $1\frac{1}{2}$ in. wire rope, and the 105,000 DWT tanker has ten mooring winches. In the case of a wharf with isolated strong points it is difficult to plan efficient mooring arrangements for the whole range of vessels from coastal tankers of 2,000 DWT to super tankers of 100,000 DWT. If mooring diagrams for the various ships were made based on the known locations of mooring fittings and manifolds, it is likely that the flexibility of a continuous face wharf would be apparent. In addition it is likely that in the case of the super tankers, the mooring dolphins which are opposite the breast lines will be heavily loaded, especially if the ship's master should double up on his lines in event of a storm. The use of limit load bollards for these dolphins might prove wise. The author suggests that in the present case this load would be 100 tons per bollard. The long leads required to run the mooring lines for some of the ships might indicate a desirability for power capstans on the mooring dolphins. In the case of tankers of 130,000 DWT or more, the overhang of the bow of the vessel also becomes rather large when isolated breasting structures are used.

In addition to the preceding points regarding the flexibility of such a continuous face wharf to handle a large range of ships, there is a definite advantage to such a wharf when the number of berths to be made available is greater. If the required time in port can be reduced by improving the ease of berthing, deberthing or mooring, it may be possible to not only reduce the transport cost, but also to reduce the number of berths required for a given throughput. In this connection, most ship owners have indicated a desire that a ship be turned around in less than 24 hr regardless of size.

Berthing Forces.—The author makes a strong argument in favor of his opinion that large tankers do not necessarily produce impact forces exeeding those of smaller size ships. His reasoning follows closely that of P. Leimdorfer, F. ASCE, as presented at the 1957 congress of the P.I.A.N.C. in London. This opinion is also held by a number of other persons, especially those who have tried to observe actual ship velocities. Leimdorfer noted in his report, however that the Stockholm harbor is most sheltered, being located 27 miles from the sea with hardly any currents, maximum wave heights of $2\frac{1}{2}$ ft, and moderate winds. He also stated that all vessels are accompanied by tug boats within the harbor waters, and he warned that conditions existing in the Stockholm harbor can hardly be generalized for use in other ports.

The Bureau of Yards and Docks has made field measurements of mooring forces of full sized ships in correlation with the effects of winds and currents. These tests were being made with various vessels but especially with aircraft carriers at the Navy Yard piers in Bremerton, Washington and Terminal Island, California. The results of the tests indicated low line pull and in some cases, no pressures were registered against the wharf structure. Due to the surprisingly low values measured, the Bureau of Yards and Docks considered these values to be unrealistic. They had also intended to measure impact velocities of ships berthing at wharfs but these tests have not yet been made to the writer's best knowledge.

One method of determining the ship's approach velocity is to assume that this is due solely to wind on the vessel acting over a certain period of time and being in turn resisted by the drag of the water on the moving ship. P. Callet in his report to the 1953 conference of the P.I.A.N.C. has presented this analysis quite well. From his analysis, several reasons are apparent why a large vessel might not have as high an approach velocity as a smaller ship. First, the force of a gust on a large surface is not as great as on a smaller one. Second, the drag of the water which must pass under the bottom of the ship drifting broadside towards the pier varies inversely as the square of the clearance of the ship above the bottom and directly with the length of the vessel. In addition, a ship approaching a pier directly on its beam would have a much greater drag in this respect than one approaching at a considerable angle to the pier. This could explain some of Leimdorfer's observations. Trapping of water between a solid face wharf and the ship could also produce results similar to those observed by the Bureau of Yards and Docks. Third, the maximum wind velocity to be considered is the value beyond which the master deems it wise to put off the maneuver. This velocity obviously depends on the ship and on the characteristic of the harbor, especially on the position of the mooring structure with respect to the wind. The master's knowledge of the fender system's effectiveness may also have a bearing on his handling of the ship.

A few large vessels are constructed these days without model tests concerning the propulsion and power requirements, it would seem to be possible to make some of these models self-propelled and to conduct tests of the approach velocities and impact forces for simulated berthing operations. One objection which might be raised in this connection is that since the time scale ratio must be proportional to the square root of the length scale ratio, in order to conform to Froudes law, a ship model must be made to react rather quickly. This has proven to be no real handicap, however, and the Wageningen Scheepsbouw Proefstation in The Netherlands for example is performing such sea handling tests in their Sea Keeping Laboratory. The test conditions simulate a ship in the open sea and may include both waves and wind from any direction, but as far as the writer knows, no tests are to be made simulating shallow water or the berthing maneuvers in which the structural engineer as a wharf designer is interested. Other laboratories have run navigational tests on self-propelled model ships in harbor or river models, but here again, so far as is known, no attempt has been made to determine berthing forces.

This subject of the ship's velocity at impact is an item of study at the twentieth International Navigation Congress to be convened in Baltimore on September 11, 1961. It is expected that additional light will be shed on this subject at that time.

Fender Systems.—The author emphasizes that the rubber sandwich buffer is perhaps the most efficient type of spring for absorbing energy. A pretensioned steel spring, however, could be used to yield a lower reaction than a comparable rubber sandwich buffer for a given energy absorption and deflection. Steel springs on the other hand are subject to breakage of the structural guides since whenever energy must be absorbed by a rigid object, the reaction approaches a large value.

A fender system which is satisfactory for a 100,000 DWT tanker tends to be too stiff for a 2,000 DWT coastal tanker. This might call for a two element system such as a combination of rubber rolls and rubber sandwich buffers, or pehraps rubber sandwich buffers with a design deflection of 24 in. to 30 in. might be used. The English licensee of these fenders has completed tests on models representing fenders with as much as 30 in. of deflection without observing any tendency toward instability. They have also made overload tests indicating that these fenders can be subjected to at least a limited number of overload deflections without signs of distress. It would seem wise to incorporate the advantages of both high deflection and provision for overload deflection in future designs even though at some additional cost.

JOSEPH H. FINGER.[12]—Having carefully followed through the author's computations on the fender system designed for a 105,000 DWT tanker and the resultant reactions and deflections for each buffer shown in the load diagram (Fig. 13), it was difficult to understand how he could reach the conclusion that approximately 30 ft of the bow of the vessel will be in actual contact with the fender system at the time of maximum deflection.

Referring to Fig. 13, it is noted that, with the acting forces (13 kips per lin ft) located at about the center of the breasting platform, the end buffers deflect about 1/2 in. with a reaction of 2.78 kips and the adjacent buffers deflect about 7 3/8 in. with a reaction of 38.82 kips, successive buffers having respectively greater deflections and reactions toward the center.

Consider the steel wale which will be capable of transmitting these loads to the buffers. The section modulus of such a steel member, computed on the basis of an elastic limit of 33 kips per sq in., should not be less than 170 sq in. A steel member having a section modulus of this magnitude would be close to a 18 WF 96, with a moment of inertia equal to 1674. Considering such a steel member to be a continuous wale in the fender structure, and recognizing the stiffness of such a steel member, it is most difficult to understand how the beam could ever deflect 18 in. from the first to the fourth buffer without exceeding its elastic limit and resulting in a permanent deformation. It must, therefore, follow that if a number of buffers are connected by a continuous steel wale, the stiffness of the wale becomes the limiting factor of the resiliency of the buffers, and consequently of the fender system as a whole.

Another approach to the case shown in Fig. 13, assuming that a 18 WF 96 steel member were used as a continuous wale, is to accept the fact that the two center buffers were deflected approximately 18 in. Adjacent buffers, considered away from the center, would be deflected 17 in., and 16 in., and 15 in., respectively, if the elastic limit of the steel wale were not to be exceeded. To obtain such deflections on the buffers would require an acting force far greater than that shown (13 kips per lin ft), with a contact surface greater in length

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than the 30 ft determined by the author. Under such conditions the fender cannot assume the shape of the bow of the super tanker.

In fact, due to the stiffness of the wale, the contact of the vessel with the fender, under the berthing conditions given for Case II, will be restricted to much less than the 30 ft shown in Fig. 13. Consequently the magnitude of the acting force, per lineal foot, will be much higher than the 13 kips used by the author for his load diagram. The result of this situation will be that the buffers, applied as shown, will never reach the deflections obtained when tested singly in the laboratory.

Despite the discrepancies previously cited in the design of the fender system selected as the most suitable for the berthing conditions described, the author is to be congratulated on the thoroughness of his analysis of site conditions. His determination of the kinetic energy which a super tanker may impart to a mooring platform, considering wind direction and velocity, wave action, tides and currents, as well as velocity and angle of approach, were clearly presented. Having carefully compiled the basic data on which an adequate fender system must be based, the author then briefly analyzed the gravity, spring, and certain types of rubber fender systems, discarding them all in favor of the rubber-sandwich (Raykin) system. Although in actual operation the system selected may be affording satisfactory service, its adequacy to fullfill the design criteria shown, is questioned.

To go one step further, since the author states in his conclusion that, "it is hoped this paper will lead to discussion . . . . . so necessary for the proper design of modern marine terminals," attention is directed to another type of resilient fender system which the author has apparently overlooked. This is the Retractable Fender System, in use for the past 7 yr on the most diversified types of pier structures with excellent results, but without the massiveness or complicated suspension system which the author considers objectionable in his analysis of gravity and inertia fenders. The retractable system is based on absorbing kinetic energy by utilizing the gravity of the frame, the frictional resistance obtained from the travel of the fender on an inclined plane, and the friction between the upward moving fender and the hull of the vessel. Depending on site conditions and the amount of kinetic energy to be absorbed, the retraction (or length of travel) may vary from 8 in. to 48 in. Various aspects of this system have been more thoroughly described elsewhere.[13,14,15]

[13] "New Retractable Marine Fenders System," by Palmer W. Roberts and Virgil Blancato, Proceedings, ASCE, Vol. 84, No. WW1, January, 1958.

[14] "A Bresting Dolphin for Berthing Supertankers," by John M. Weis and Virgil Blancato, Proceedings, ASCE, Vol. 85, No. WW3, September, 1959.

[15] Dock and Harbor Authority, December, 1960.

BEHAVIOR OF BEACH FILLS IN NEW ENGLAND[a]

Discussion by A. C. Rayner

A. C. RAYNER.[11]—The author is to be commended for the work he has accomplished in compiling useful data regarding the use of beach fills as a shore protection measure. As he has stated, the periods of study since initial placement of the beach fill are generally short. During the initial period losses are frequently difficult to distinguish from changes due to profile adjustments. Longer periods of observation should furnish more realistic rates of losses in many cases. Additional data to June, 1960 for Prospect Beach made available to the Beach Erosion Board since writing of the original paper indicate an average annual loss of approximately 13,000 cu yd from between the planes of mean high and mean low water. This is an indication of the probable annual nourishment requirement, rather than the indication of no nourishment requirement based on data to June, 1959. It is hoped that surveys will be continued to provide data on behavior of beach fills over longer periods.

[a] February, 1961, by Harry S. Perdikis (Proc. Paper 2744).
[11] Chf., Proj. Development Div., Beach Erosion Bd., Washington, D. C.