THE NATHAN DANIEL SUPEROUTRIGGER

AN INVESTIGATION OF THE POTENTIAL OF THE CONCEPT

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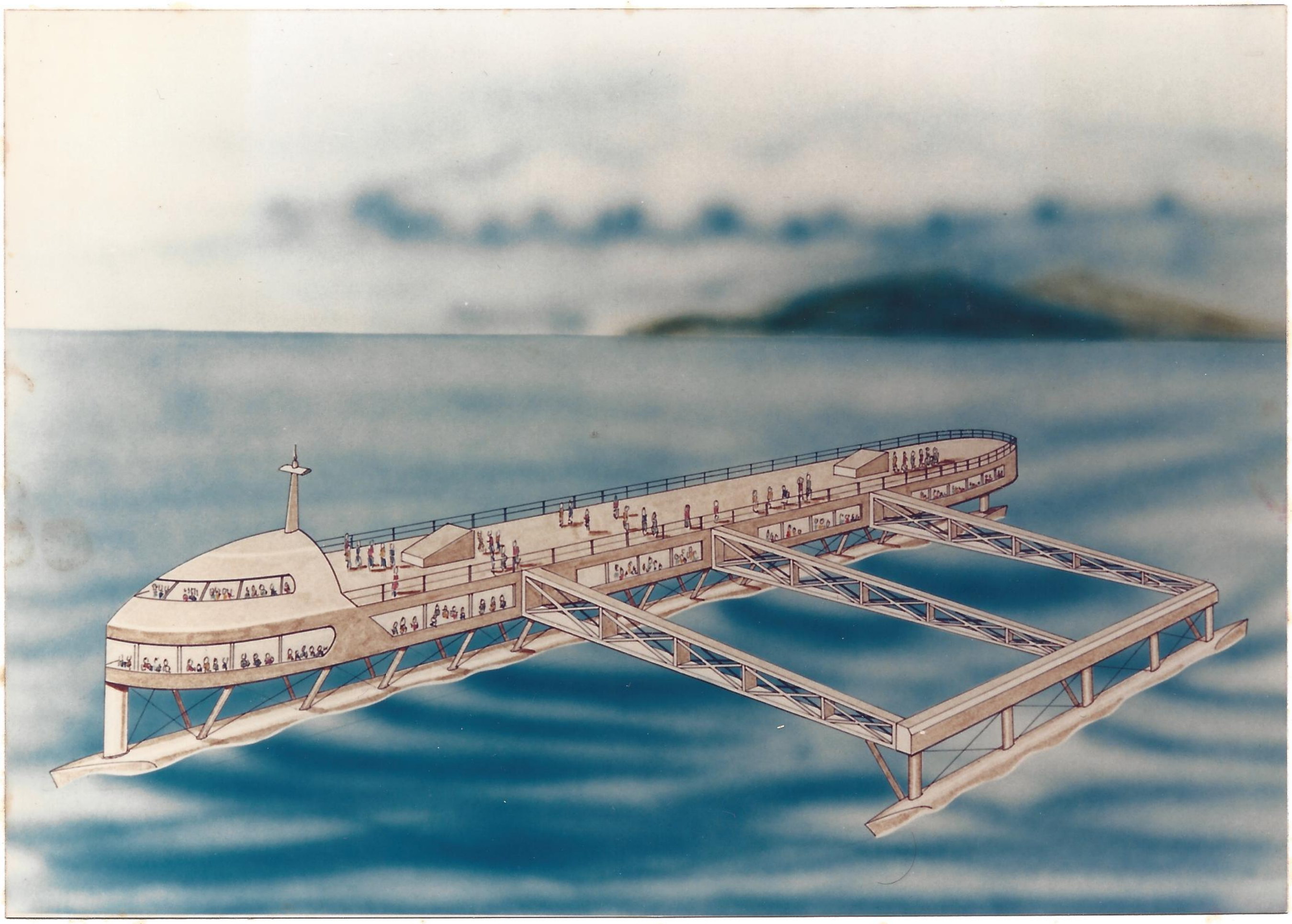
January 1987.



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Definition of Symbols

l	Length, hull waterline	ft
d	Diameter, of cylindrical hull	ft
r	Radius, of cylindrical hull	ft
C_f	Skin friction coefficient	non-dimensional
RN	Reynolds Number $= \frac{V.l}{\nu}$	" "
ν	Kinematic viscosity of water	ft^2/s
b	Beam of single hull	ft
F_N	Froude No. $= \frac{V}{\sqrt{g\,l}}$	non-dimensional
g	Acceleration due to gravity	ft/s^2
W	Weight of vessel	lb or l.tons
V	Speed of vessel relative to water	ft/s or knots
∇	Displacement	ft^3
h_W	Depth of water	ft
$C_{R_{TOT}}$	Total resistance coefficient based on wetted area and $= \frac{R_{TOT}}{\frac{1}{2}\rho V^2 S_W}$	non-dimensional
R_{TOT}	Total resistance	lb
ρ	Mass density of water	$slugs/ft^3$ $\left(= \frac{lb/ft^3}{g}\right)$
S_W	Wetted area	ft^2
D	Drag = Resistance = R	lb
λ	Scale $= l_{f.s} \div l_m$	non-dimensional
k	Surface roughness height	ft
shp	Shaft horse power of propulsion engine(s)	non-dimensional
η	Propulsive efficiency	" "
C_B	Block coefficient $= \frac{\nabla}{l_{W.L.} \times b \times d'}$	" "
d'	Draught	ft
C_W	Waterplane area coefficient = area of waterplane $\div \left(l_{W.L.} \times b\right)$	non-dimensional
C_r	residuary resistance coefficient	non-dimensional
R_r	residuary resistance	lb

Subscripts

f.s.	Full scale
m	Model scale
W.L.	Water line
M	Main hull
S	Secondary hull
TOT	Total
o/a	Overall

Figures

1. The Principle

The fundamental reason why vessels can have uncomfortable motions when at sea is that they are physically small in relation to the waves they encounter. To lessen this problem, auto-stabilization systems can be fitted, as on the Boeing Jetfoil hydrofoil and, more recently, the Marinteknik Marinjet monohull craft, two examples of small craft to be so equipped. Alternatively, a vessel can be designed to have as little as possible of its structure intersecting the surface of the sea, the SWATH (Small-waterplane-area twin hull) type achieves this with fully submerged hulls and only the struts, supporting the main payload-carrying structure, actually passing through the sea surface. The fully submerged hydrofoil system also achieves this result, again the Boeing Jetfoil benefits from this arrangement. Both these solutions involve relatively deep draughts and not particularly attractive economics.

The essence of the inventions of Nathan Daniel is the provision of a craft with "extended dimensions" in its supporting elements in order to achieve stability through greater size but without incurring the penalty of greater resistance. That is, the length and effective beam of these elements in relation to the payload-carrying area is very substantially greater than that which can be achieved (with acceptable economics) for hydrofoils, hovercraft, conventional displacement vessels, planing and semi-planing vessels. The SuperOutrigger is in fact an exploitation of the basic principles that lie behind the evolution of the Pacific outrigger boats which have evolved to make it possible to navigate in the rough Pacific waters.

It has been recognised for a considerable time that a long, slender hull offers reduced craft motions and reduced wave-making resistance. However, in terms of monohull vessels such an approach leads to insufficient volume for machinery and payload and insufficient transverse stability. To overcome this limitation, a catamaran craft form may be considered employing two slender hulls and with the added advantage of improved roll stability and less wave-making resistance. If now a single hull could be substituted for the twin hulls, then an even more slender hull could be employed, reducing wave drag still further and extending the effective waterline length of the vessel still further, thereby increasing its scale relative to sea state. Such a slender hull then must be stabilised in roll. This can be done either by a hydrofoil system or by a secondary hull, outrigger fashion. Nathan Daniel has considered both these options, as covered by his patents, but this report will be addressed to his SuperOutrigger concept.

The relatively high frictional resistance of a very long, slender hull is offset by the very much reduced wave-making resistance. It can be shown by way of illustration that if the two hulls of a catamaran are replaced by a single hull of twice the length, then (assuming for this illustration that a cylindrical hull is being considered) for the same displacement the wave drag will be halved. In addition, the frictional resistance will be less since the flow Reynolds Number for a given speed is doubled ($RN = \frac{V.l}{\nu}$) giving a frictional resistance drop of some 10%.

The wave drag of a typical conventional semi-planing high-speed catamaran is about 50% to 60% of its total drag at cruising speed, so it can be seen that a halving of this component is a very significant saving and, coupled with the frictional resistance saving, allows quite a considerable margin for increases of weight and drag of the outrigger stabilising element.

More fundamentally, it can be argued that by doubling hull length the Froude Number (the parameter which governs the wave-making characteristic of a hull) is reduced by 30% and, for the type of displacement hull being considered, this can only mean a substantial reduction in wave-making resistance.

2. The Nathan Daniel Patents

The patents covering the concept outlined in section 1. are as follows:

United States Patent 4,304,190. Dec. 8, 1981 (filed June 5, 1978)

Ferry Boat

United States Patent 4,452,166. June 5, 1984 (filed Nov. 20, 1981)

Foil Stabilized Monohull Vessel

U.K. Patent 2022 519 B. Sept. 8, 1982 (filed May 21, 1979)

Surface Vessel Capable of Operating in Rough Seas.

- also registered: Hong Kong Registration No.462 of 1983
  Singapore Registration No.529/82

(Copies of these patents are included in the Appendix to this report and the reader is referred to these for an understanding of the essential elements of the inventions. As is often the case, the patent documents do not, however, describe all the characteristics and advantages of the inventions and it is therefore the intention of this report to investigate some of these and to highlight the potential offered by the SuperOutrigger concept as covered by the first and third patents listed above.)

Additional SuperOutrigger patents:

Australian Patent 524854. May 11, 1979 (filed June 5, 1978)
Ferry Boat

Italian Patent 1116224. Feb. 10, 1986 (filed in 1979)
Ferry Boat

New Zealand Patent 190445. May 14, 1979 (filed June 5, 1978)
Ferry Boat

Canadian Patent 1194730. Oct. 8, 1985 (filed Nov. 18, 1982)
Foil Stabilized Monohull Vessel

European Patent 0 080 308. March 19, 1986 (filed Nov. 12, 1982)
Foil Stabilized Monohull Vessel:

Euro-Belgian Patent 0 080 308 (notification on April 7, 1986)
European (Dutch) Patent 0 080 308
European (French) Patent 0 080 308
European (German) Patent P 32 70 017.2-08 (notification on Feb. 20, 1986)
European (Italy) Patent 0 080 308
European (Sweden) Patent 0 080 308
European (U.K.) Patent 0 080 308 - also being registered in Hong Kong and Singapore

Japanese Patent Application 202232/82 is pending.

3.0 Review of Towed Model Resistance Measurements

3.1 Interpretation of Makai Ocean Engineering Resistance Tests.

A 28.33 ft model of the SuperOutrigger concept was constructed with hull-to-hull centreline distance of 8.85 ft and resistance tests carried out in 1978. The model was run at weights of 900 lb and 1438 lb. Ref. 1 gives details of this work and Fig. 1 shows the form of the model. The hulls were of cylindrical form with vertical wedge-shaped bows and sterns. See Appendix.

3.1.1. Model Details

Main hull length: 28.33 ft (main hull)

Main hull diam.: 1.063 ft

Secondary hull length: 17.70 ft

Secondary hull diam.: 0.50 ft

Hull-to-hull centreline: 8.85 ft

Weight of model: 900 lb "A" condition, 1433 lb "B" condition

Draught: 0.532 ft "A" condition

Wetted area, at rest, 50% of diam. immersion, main hull: 45 ft^2

" " " " " " " " , secondary hull: 14.7 ft^2

Main hull length/beam ratio: 26.65

3.1.2. Tests Results

The model was towed from a vehicle running along a pier over a range of speeds with the force on the model being read from a scale mounted on the model.

In the table on p.5 these results are reduced to D/W values versus main hull Froude Number.

From test points of p. 2 of Appx. of Ref. 1, where time to cover 30 ft distance versus drag force is given:

"A" condition, W = 900 lb.

(1)	t secs.	5.0	3.8	3.10	2.75	2.41
(2)	drag lb	15.0	29.0	39.0	56.5	72.0
(3)	V ft/s	6.0	7.89	9.74	10.91	12.45
(4)	$\frac{D}{W}$	0.0167	0.0322	0.0433	0.0628	0.0800
(5)	$F_{N_{\text{Main hull}}}$	0.199	0.261	0.323	0.361	0.412

"B" condition, W = 1433 lb

(1)	t secs.	5.6	5.1	4.35	3.80	3.18	2.80
(2)	drag lb	24.0	32.0	37.0	50.0	69.0	82.8
(3)	V ft/s	5.25	5.88	6.90	7.89	9.43	10.71
(4)	$\frac{D}{W}$	0.0167	0.0223	0.0258	0.0349	0.0482	0.0578
(5)	$F_{N_{\text{Main hull}}}$	0.177	0.195	0.228	0.261	0.312	0.355

These results are shown plotted in Fig. 2. It is interesting to note that the drag/weight ratio does not increase uniformly with model weight. With a 58.2% increase in displacement (900 lb to 1433 lb) there is (assuming good accuracy in the test results) only a 41% increase of $\frac{D}{W}$ at $F_N = 0.2$, 12.3% increase at $F_N = 0.3$ and no increase at $F_N = 0.345$.

The drag analysis given in Ref. 1 is not believed to be appropriate since the test model did not have polished smooth surface hulls so that the skin friction coefficients cannot be assumed (as in Ref. 1) to be those for such surfaces. The following investigation has therefore been undertaken.

3.1.3. Coefficient of Resistance for Complete Model Craft

Condition "A"

Wetted area of the two hulls at 50% submergence is given in Ref. 1.

Main hull: $S_{W_M} = 45.0 \text{ ft}^2$

Secondary hull: $S_{W_S} = 17.1 \text{ ft}^2 = 0.38\ S_{W_M}$

A total resistance coefficient, $C_{R_{TOT}}$ can be calculated from the test results but not the resistance coefficient of the hulls separately. A $C_{R_{TOT}}$ value will therefore be calculated based on total wetted area (at rest) = $S_{W_{TOT}} = S_{W_M} + S_{W_S} = 62.1\ ft^2$ and will be considered with RN and F_N based on a mean length $(l_m + l_S) \div 2.0 = (28.33 + 17.70) \div 2.0 = 23.02\ ft = \bar{l}$

$$\text{Now } C_{R_{TOT}} = \frac{R_{TOT}}{\frac{1}{2}.\ \rho\ V^2.\ S_{W_{TOT}}}$$

Assume $\rho = 1.98\ slugs/ft^3$ for sea water

$$\text{Then } C_{R_{TOT}} = \frac{R_{TOT}}{61.48\ V^2}$$

From model test:

V ft/s	6.00	7.89	9.74	10.91	12.45
R_{TOT}	15.0	29.0	39.0	56.5	72.0
$C_{R_{TOT}}$	0.00678	0.00758	0.00669	0.00772	0.00756

Assuming that the major part of the resistance is skin friction resistance (since hulls have very high length/beam) it is of value to examine these $C_{R_{TOT}}$ values against a plot of skin friction coefficient, C_f , versus Reynolds Number.

V ft/s	6.00	7.89	9.74	10.91	12.45

$\bar{l} = 23.02$ ft

$\nu = \frac{1.23}{10^5}\ ft^2/s$

RN million	11.229	14.766	18.228	20.418	23.300

In these figures, no clear variation of $C_{R_{TOT}}$ with RN is seen. If these figures are plotted on a skin friction coefficient C_f versus RN diagram for flat surfaces, Fig. 3, (extracted from Ref. 2) it is seen that they fall into a surface roughness area, defined as length ÷ surface roughness $\left(\frac{l}{k}\right)$ of between 10^3 and 3×10^3. For the main hull length of 28.33 ft this gives a roughness height of between 0.34 in and 0.11 in. The analysis however neglects the component of induced wave making drag caused by the passage of the finite width hulls through the water surface. If this is arbitrarily put at 15% of total drag (see later examination of this component) then the $C_{R_{TOT}}$ component due to skin friction would be as follows (neglecting change of wave-making drag with F_N):

V ft/s	6.0	7.89	9.74	10.91	12.45
0.85 $C_{R_{TOT}}$	0.00576	0.00644	0.00569	0.00656	0.00643

The values of $\frac{l}{k}$ in Fig.3 are now seen to be higher at about 6.5×10^3 to 3×10^3, giving roughness heights, k, on a length of 28.33ft of between 0.052in and 0.133in. These levels are probably typical of the levels achieved in the test model hulls but the wedge-shaped bows would also have contributed to substantial increases in frictional resistance.

Now it can also be seen from Fig.3 that if $\frac{l}{k}$ were to be reduced to, say, 3×10^4, then C_f reduces to about 0.004 which is 30% lower than the lowest ($C_{R_{TOT}}$ - 15%) value as calculated above. To do this requires a surface with irregularities no higher than $\frac{28.33 \times 12}{30000} = 0.0113$in, i.e. approaching a racing yacht finish.

A more modest target would still pay very useful dividends. In addition, it should be mentioned again that the bow form of the test model hull is a major turbulence generator (and wave generator) so a very significant reduction of $C_{R_{TOT}}$ should be possible for a refined hull design.

It should also be noted from Fig.3 that RN effects may possibly be ignored for the range which we are considering, i.e. the model test speed range (RN 1.0×10^7 to 2.3×10^7) provides results of C_f directly applicable to full-scale craft length, the curve is flat with RN but this is only true if a fair level of hull surface roughness is accepted.

The above results suggest that some attention should be given to examining two aspects of the vessel hull, the surface finish and the longitudinal waterline shape with a view to reducing resistance. The idea of cylindrical hulls is very attractive from production and cost considerations and in providing a high level of displacement for a given length of hull. The minimisation of wave-making resistance is not so easy to achieve, however, with a rather full bow hull form*. To see how this aspect could perhaps be improved, a search was conducted for tank test results on very slender hull forms. Very little indeed appears to exist for length/beam ratios greater than 10. Some resistance test results have, however, been made available to the writer for "Rowing 8" hull forms with length/beam ratios up to 31 and these can be used to see how far the SuperOutrigger concept can be carried as far as resistance minimisation is concerned. These hull forms have of course been designed for relatively calm waters and are unlikely to have precisely the characteristics ideally suited to cutting through waves. They do though permit some assessment to be made of the best transport efficiency that can be expected from a slender hull concept, knowledge of which is of vital importance if the craft is to be considered for commercial service.

3.2 Resistance characteristics of a "Rowing 8" hull

3.2.1. Model tests

A "Rowing 8" hull with the following characteristics is selected for consideration of its resistance characteristics:

Length, waterline:	56.16ft
Beam, waterline:	1.805ft
Length/beam ratio:	31.11
Block coefficient, C_B:	0.423
Displacement as tested:	1910 lb
Wetted surface area:	99.64ft^2
Waterplane area coefficient, C_W:	0.681
Draught, midships:	0.687ft

* See Appendix for photographs of SuperOutrigger demonstration craft hull forms and wave generation.

The total resistance of this model is shown in Fig.4 together with the results from the SuperOutrigger model. A drag/weight versus speed curve is shown in Fig.5 for the "Rowing 8" hull, corresponding to similar curves in Fig.2 for the SuperOutrigger model. The difference in drag/weight ratio for the two types of model hull is very large. The difference is due to hull form difference causing relatively greater wave drag to be generated by the SuperOutrigger model (though still not the largest percentage of the total) and surface roughness and bow shape of the SuperOutrigger hulls. If these SuperOutrigger test values were to be scaled for a full-scale vessel, then very pessimistic values would result.

By examining the "Rowing 8" results, it is contended that it is possible to predict full-scale performance for the SuperOutrigger based on near optimum (hydrodynamically) hull shape.

3.2.2. <u>"Rowing 8" drag versus RN and F_N</u>

From Ref.3:

$W = 1910$ lb $\quad l_{W.L.} = 56.166$ft $\quad S_{W_m} = 99.64\text{ft}^2$

V ft/s	6	8	10	12	14	16	18	20
R_{TOT} lb	9.5	16.5	25.0	35.5	48.0	62.0	79.5	98.4
$\frac{R}{W}_{TOT}$	0.0050	0.0086	0.0131	0.0186	0.0251	0.0325	0.0416	0.0515
F_N	0.141	0.188	0.235	0.282	0.329	0.376	0.423	0.470

Tests at 60^oF = 1.938 slugs/ft^3

$$C_{R_{TOT}} = \frac{R_{TOT}}{\frac{1}{2}\rho S_W . V^2} = \frac{R_{TOT}}{0.5 \times 1.938 \times 98.64\ V^2}$$

$$= \frac{R_{TOT}}{96.55\ V^2}$$

$C_{R_{TOT}}$	0.00273	0.00267	0.00259	0.00255	0.00254	0.00251	0.00254	0.00255
$RN_{million}$	27.4	36.53	45.67	54.8	63.93	73.06	82.2	91.34

These results are shown plotted in Fig.6 together with those for the 900lb SuperCutrigger model. Over the speed ranges tested, the variation with Reynolds Number is indeterminate, perhaps the "Rowing 8" model showing some small trend in line with the roughness line trends in Fig.3.

The hull results in Fig.6 do include wave drag (and aerodynamic drag) with the consequent differences between models, as discussed above.

Plotting against Froude Number (Fig.7) shows the smooth nature of the "Rowing 8" model $C_{R_{TOT}}$ variation with F_N with its very small increment of wave drag. In neither sets of results is there a perceptible variation with F_N which can be attributed with confidence to a variation of wave drag.* The scatter in the SuperCutrigger results is probably due primarily to waves breaking over the bows of the hulls.

Extrapolating the $C_{R_{TOT}}$ values to higher RN full scale is unlikely to give reduced values. As shown in Fig.3, when there is a surface roughness present, even of a very small degree, the skin friction coefficient remains nearly constant at RN values above the model test values. As discussed earlier, surface finish improvement can lead to useful resistance reductions and improved shaping can lead to a reduction in wave resistance; these are the areas to tackle for overall improvement.

Ref.3 contains an estimate of wave resistance for the "Rowing 8" hull tested and reported on here. Fig.5 shows this estimate in terms of D_{wave}/W versus F_N and it is seen to be a small quantity, 10.8% of D/W at $F_N = 0.475$ to 4.8% of D/W at $F_N = 0.30$.

*The difference between the measured total drag and calculated skin friction drag is normally referred to as residuary resistance, the major part of this increment being wave drag.

This wave resistance will be a larger quantity for the SuperOutrigger hulls since wave-making theory indicates that:

$$\frac{\text{wave drag}}{\text{displacement}} \text{ is proportional to: } \frac{\text{displacement}}{\text{length}^3_{\text{W.L.}}}$$

and this latter parameter is obviously larger for a cylindrical hull than for a finely shaped "Rowing 8" hull, such as is illustrated in Fig.8.

4. Power requirements for a full-scale SuperOutrigger vessel

Using $C_{R_{TOT}}$ values from the "Rowing 8" tests, it is now possible to estimate full-scale power requirements for a SuperOutrigger vessel employing "minimum resistance" hull forms.

The following calculation is for a 20-knot vessel with a 320ft main hull length, as previously considered in Ref.1. For scaling laws see p.67 of Ref.5.

$$\nu = \frac{1.23}{10^5} \text{ ft}^2/\text{s}$$

$$\rho = 1.992 \text{ slugs/ft}^3 \text{ at } 80^\circ\text{F}$$

$V_{f.s.}$ = 20 knots

	Main hull 320 ft		Secondary hull 200 ft	
	Model*	Full-scale	Model*	Full-scale
Length_{WL} ft.	56.16	320.00	56.16	200.00
Scale,	5.70:1.0		3.56:1.0	
Beam_{WL} ft.	1.805	10.29	1.805	6.43
S_W Wetted area, ft^2	99.64	3237	99.64	1263
Displacement, lb	1910	353719	1910	86175
" " " , l tons	0.853	157.9	0.853	38.47
Full-scale speed, knots	-	20	-	20
Full-scale speed, ft/s	-	33.78	-	33.78

* "Rowing 8" hull form

	Main hull 320 ft		Secondary hull 200 ft	
	Model	Full-scale	Model	Full-scale
Froude No. F_N	0.332 ←	0.332	0.421 ←	0.421
Model speed $= F_N\sqrt{gl}$ ft/s	14.12	-	17.90	-
$RN = \frac{V.l_{WL}}{\nu}$	6.45×10^7	8.79×10^8	8.17×10^7	5.49×10^8
$C_{R_{TOT_m}}$ (Fig.6)	0.00255	-	0.00255	-
C_{f_m} (Fig.6)	0.00220	-	0.00213	-
$C_{r_m} = C_{R_{TOT_m}} - C_{f_m}$	0.00030	-	0.00042	-
$R_{f_m} = \frac{1}{2}\rho V^2.C_{f_m}.S_{W_m}$ lb	43.53	-	67.73	-
R_{TOT_m} (Fig.4) lb	48.90		78.30	
R_{r_m} lb $= R_{TOT_m} - R_{f_m}$	5.37		10.57	
Full-scale, $R_r = \lambda^3 R_{r_m} = R_{r_{f.s.}}$ lb	-	994	-	477
$C_{f_{f.s.}}$ (Fig.6)	-	0.00159	-	0.00168
$R_{f_{f.s.}} = \frac{1}{2}\rho V^2 C_f.S_{W_{f.s.}}$ lb	-	5850	-	2412
$R_{r_{f.s.}} + R_{f_{f.s.}}$ lb	-	6844	-	2889

$R_{TOT_{f.s.}}$ = 9733 lb

$W_{f.s.}$ = 439894 lb

$\frac{R_{TOT}}{W}$ = 0.022

For 25 knots the following results are obtained:

	Main hull 320 ft		Secondary hull 200 ft	
	Model	Full-scale	Model	Full-scale
Full scale speed, knots	-	25	-	25
" " " , ft/s	-	42.23	-	42.23
Froude No.	-	0.415	-	0.526
Model speed ft/s $= F_N\sqrt{gl}$	17.65	-	22.38	-
$RN = \dfrac{V.l_{WL}}{\nu}$	8.06×10^7	10.99×10^8	1.021×10^7	6.86×10^8
$C_{R_{TOT_m}}$ (Fig.6)	0.00255	-	0.00255	-
C_{f_m} (Fig.6)	0.00213	-	0.00206	-
$C_{r_m} = C_{R_{TOT_m}} - C_{f_m}$	0.00038	-	0.00049	-
$R_{f_m} = \frac{1}{2}\rho.V^2C_{f_m}.S_{W_m}$ lb	65.85	-	102.4	-
R_{TOT_m} (Fig.4) lb	76.00	-	125.00 (extrapolated)	-
R_{r_m} lb $= R_{TOT_m} - R_{f_m}$	10.15	-	22.6	-
Full scale, $R_{r_{f.s.}} = \lambda^3 . R_{r_m}$ lb	-	1880	-	1020
$C_{f_{f.s.}}$ (Fig.6)	-	0.00155	-	0.00164
$R_{f_{f.s.}} = \frac{1}{2}\rho V^2 C_f.S_{W_{f.s}}$ lb	-	8912	-	3679
$R_{r_{f.s.}} + R_{f_{f.s.}}$ lb	-	10792	-	4699

$R_{TOT_{f.s.}}$ $\underline{15491 \text{ lb}}$

$W_{f.s.} = 439894$ lb

$\dfrac{R_{TOT}}{W}$ $\underline{0.0352}$

If propulsive efficiency is assumed = 60%, then for speed = 20 knots

$$\text{shp required} = \frac{9733 \times 20 \times 1.689}{0.60 \times 550}$$

$$= \underline{996}.$$

If payload is assumed = 20% of displacement, then

shp/payload l.ton knot (an index of transport efficiency)

$$= \frac{996 \times 2240}{0.2 \times 439894 \times 20}$$

$$= \underline{1.27}.$$

Then, as before, if η = 60%, speed = 25 knots

$$\text{shp required} = \frac{15491 \times 25 \times 1.689}{0.60 \times 550}$$

$$= \underline{1982}.$$

If payload is assumed = 20% of displacement, then

shp/payload l.ton knot

$$= \frac{1982 \times 2240}{0.20 \times 439894 \times 25}$$

$$= \underline{2.019}.$$

Hull towing test results are not available for Froude Numbers above 0.5 so only extrapolated estimates could be made for 30 knots. Assuming shp is proportional to V^3 (a slightly pessimistic assumption) the shp for 30 knots would be 1982 x $(\frac{30}{25})^3$ = 3425. The shp/payload l.ton knot then becomes $\underline{3.49}$.

Fig.9 shows these estimated values in relation to those for current high-speed ferries. The basis for the values shown is given in the following table: