Water-Jet Propulsion for High-Speed Hydrofoil Craft

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Techniques are developed for determining the approximate optimum water-jet propulsion system for high-speed hydrofoil craft. An example system utilizing existing pumps for the propulsion of an 80-knot 500-ton craft is presented. It is suggested that a water-jet system utilizing future light-weight multiple impeller double suction pumps may be competitive with a supercavitating propeller propulsion system.

Introduction

THE propulsion of water craft by passing water through on board pumps and ejecting it aft with an increase in momentum is not a very new idea. It has been considered as a propulsion method since the invention of mechanical pumps, but the simpler, lighter, and more efficient propeller has monopolized the water propulsion field.

When propeller cavitation or vulnerability to damage in shallow water is not a problem, it is clear that the simple screw propeller generally will be superior because there are no inlet, casing, and nozzle losses to contend with. Furthermore, the weight of a propeller is insignificant, whereas the weight of a water-jet system usually represents an appreciable percentage of the gross weight of the craft.

Nevertheless, in the past decade, increasing numbers of water sport craft are being propelled with water jets. No doubt, some of the initial attraction is because of the glamour of the name jet propulsion. However, it seems likely that this method of propulsion of light, relatively high-speed craft, will grow in popularity for more practical reasons. Such craft are capable of operating in shallower water than propellers; they are much more weed proof than propellers; they are safer for nearby swimmers; and several of the current designs incorporate 360° directional control of the nozzle, so that increased maneuverability is achieved.

In the speed range around 35 knots, where most of these sport craft operate, some modern water-jet propulsion systems have propulsive efficiencies comparable to screw propellers, because the cavitation problem begins to deteriorate the propeller efficiency. Furthermore, manufacturers have lightened the weight of water-jet systems by using lighter materials and more compact designs. Water-jet propulsion for the sport run-about already is well established and will probably propel increasing numbers of such craft in the future.

The development of the supercavitating propeller has enabled the screw propeller to continue to be the primary propulsor at speeds above 45 knots. However, the propulsive efficiency of these propellers is no greater, and sometimes less, than that of well designed water-jet systems, but the weight of the water-jet system currently penalizes it, in favor of the supercavitating propeller. In spite of this weight disadvantage, which inventive development effort will surely diminish in the future, the advantages of water-jet propulsion for high-speed hydrofoil craft are so great that much attention now is directed in that area.

The designer of a large 80-knot hydrofoil craft using supercavitating hydrofoils and propellers must face two immediate propulsion problems, which are formidable. These are: 1) Where do you locate a supercavitating propeller with respect to the foil and base vented strut so that mutual cavity wake interference is not intolerable? and 2) Can the bevel gears required to reliably drive the propellers be developed in a small enough package to be acceptable? No doubt tolerable solutions to these problems associated with supercavitating propellers can be found. However, a water-jet system solves them by elimination.

This paper outlines the principles of water-jet propulsion, including a discussion of the various types of pumps, and presents several preliminary designs for high-speed hydrofoil craft. Emphasis is placed on the problem of diminishing the weight of water-jet propulsion systems.

Presented as Preprint 64-306 at the 1st AIAA Annual Meeting, Washington, D. C., June 29–July 2, 1964; submitted June 22, 1964; revision received August 23, 1965.

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Analysis of the Hydrodynamic Performance of Water-Jet Propulsive Systems

Propulsive Efficiency

A schematic sketch of a typical water-jet propulsion system is shown in Fig. 1. In the following analysis it is assumed that the external friction and profile drag of the strut and scoop are zero or at least are not considered as influencing the thrust of the system. In a more refined analysis, any increase in drag, caused by an increase in strut dimensions required specificially for a water-jet system, must be charged as a negative thrust. Furthermore, an additional scoop drag may be produced because of separation near the leading edge of the scoop caused by prediffusion or acceleration, and this drag also should be accounted as a negative thrust.

With the assumption of no scoop or increased strut drag, the net thrust on the system shown in Fig. 1 may be expressed as

$$T_m = \rho Q(V_j - V_0) \tag{1}$$

where

T_m = thrust obtained by the momentum analysis
ρ = mass density of water
Q = flow rate through the system
V_j = velocity of the jet leaving the nozzle
V_0 = forward speed of the craft

The power supplied to the pump to produce this thrust may be expressed as

$$P = (\rho g Q H)/(\eta_p) \tag{2}$$

where

P = power supplied by the prime mover
g = acceleration caused by gravity
H = head produced by the pump
η_p = pump efficiency

The propulsive efficiency η of the system then may be expressed as

$$\eta = (T_m V_0)/P = [(V_j - V_0)V_0\eta_p]/gH \tag{3}$$

Now, if we express the total inlet and internal energy losses (exclusive of elevation changes) in the system as $k(V_0^2/2g)$

and apply the Bernoulli equation across the system, the jet velocity V_j may be obtained from the following equation

$$V_j^2/2g = (1 - k)(V_0^2/2g) + H - h \qquad (4)$$

where h is the difference in elevation between the nozzle exit and the water surface. Thus,

$$V_j = V_0[(1 - k) + (2gH/V_0^2) - (2gh/V_0^2)]^{1/2} \qquad (5)$$

Substituting Eq. (5) into Eq. (3) gives the following relationship

$$\eta/\eta_p = \bar{\eta} = 2\{[1 - k - (2gh/V_0^2) + (2gH/V_0^2)]^{1/2} - 1\}V_0^2/2gH \qquad (6)$$

Now, we define new variables K and H^* such that

$$K = k + (2gh/V_0^2) \qquad H^* = 2gH/V_0^2 \qquad (7)$$

Equation (6) may be written as

$$\bar{\eta} = 2[(1 - K + H^*)^{1/2} - 1]V_0^2/2gH \qquad (8)$$

The relationship given by Eq. (8) is plotted in Fig. 2. An important point revealed by Eq. (8) is that the efficiency of the system depends on the parameter K, which is a single definite number for a fixed geometry, and upon the head selected for the pump. An even more important fact revealed by Eq. (8) is that a maximum efficiency exists for a specified K and thus an optimum value of pump head. This line of optimum efficiency is indicated by the dashed line in Fig. 2 and may be expressed analytically as

$$H_{\text{opt}}^* = 2K + 2(K)^{1/2} \qquad (9)$$

The results shown in Fig. 2 are extremely important and useful in the design of an optimum water-jet system, and this figure will be referred back to many times during the subsequent discussion.

Internal Energy Losses

In order to evaluate propulsive efficiencies by the methods presented in the previous discussion, it is essential to determine the factors that influence the parameter K in Eq. (8) and Fig. 2. Now, K has been defined as $K = k + (2gh/V_0^2)$, and it is desired that K be as small as possible; that is, the head loss coefficient k and the elevation coefficient should be as small as possible.

An immediate conclusion is that in so far as practicable the nozzles should be located as low on the craft as possible. This, of course, could be done by ducting water from the pump back down to nozzles located at some lower level, say on the strut, at or below the waterline, so that $h = 0$. This could probably be accomplished with only small effect on k by using sufficiently large ducts. The weight of these ducts and the water in them obviously would be so great as to make such a

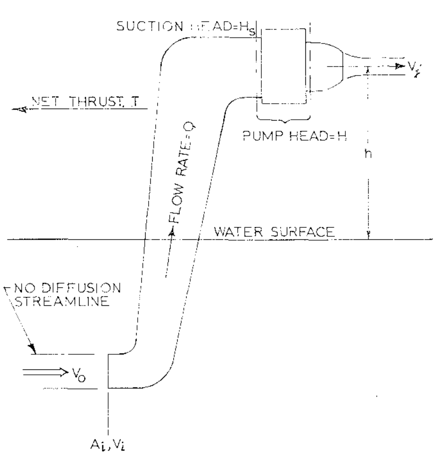

Fig. 1 Definition sketch.

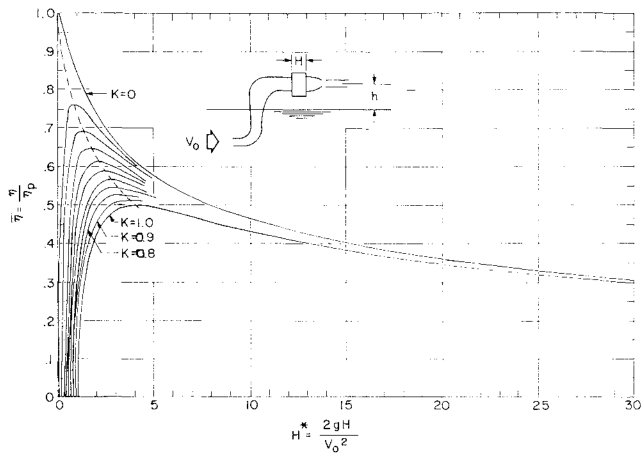

Fig. 2 Hydrodynamic characteristics of water-jet propulsion systems.

scheme absurd. However, it is important to note that the pumps should be located as low as possible so that h can be minimized consistent with minimizing the duct and water weights between the pump and nozzles.

The head loss coefficient, $k = 2gh_l/V_0^2$, for a water-jet system may be broken down into three major components: 1) inlet, diffusion and turning losses occurring between the scoop inlet and the lower part of the strut; 2) friction, diffusion and turning losses between the lower part of the strut and the entrance to the nozzle; and 3) friction losses in the nozzle. The majority of the magnitude of k will be contributed by 1). Methods of determining the actual magnitudes of k for each of these components are outlined in the following paragraphs.

The efficiencies of various types of nozzles as reported in the literature vary from 0.97 to 0.99 with the higher value attained at higher Reynolds number. It therefore seems adequate to state that the best possible nozzle that can be achieved for a water-jet system will have an efficiency of 0.99 so that its head loss will be

$$h_l = 0.01V_j^2/2g = (1 - k)(V_0^2/2g) + H - h \qquad (10)$$

or

$$k_3 = 0.01[1 - k_1 - k_2 - (2gh/V_0^2) + (2gH/V_0^2)] \qquad (11)$$

Since the first four terms in the parenthetical term of Eq. (11) are fixed for a given system, it is clear that the magnitude of k_3 increases with increase in the value of H^*; that is, at conditions immediately after takeoff. Fortunately, however, Fig. 2 shows that even large variations in K have only a small influence on the efficiency at large values of H^*.

The magnitudes of k_1 and k_2 are made up of an inlet loss and a diffusion zone followed by a turn (k_2 may be somewhat more complicated but for the present purpose it is treated as stated). For the case of a hydrofoil craft the turn at the downstream end of the scoop probably will be approximately 90°. It would be desirable to rake the strut forward so as to minimize this angle, but the forward rake would lead to increased divergence tendencies of the strut. For a 90° turn the losses in the turn may be minimized by using turning vanes. Wind tunnel and water tunnel data on such vaned turns reveal that the local loss coefficient for well designed turns based on the velocity through the turn is about 0.2. Similar data on conical diffusers reveal that diffusion losses may be minimized by increasing the area equivalent to a 6° expanding cone. The loss through such a diffuser is given approximately by the equation

$$h_l = 0.2[(V_1^2 - V_2^2)/2g] \qquad (12)$$

Referring to the schematic sketch in Fig. 3, and using the loss coefficients given in the preceding paragraph, the total

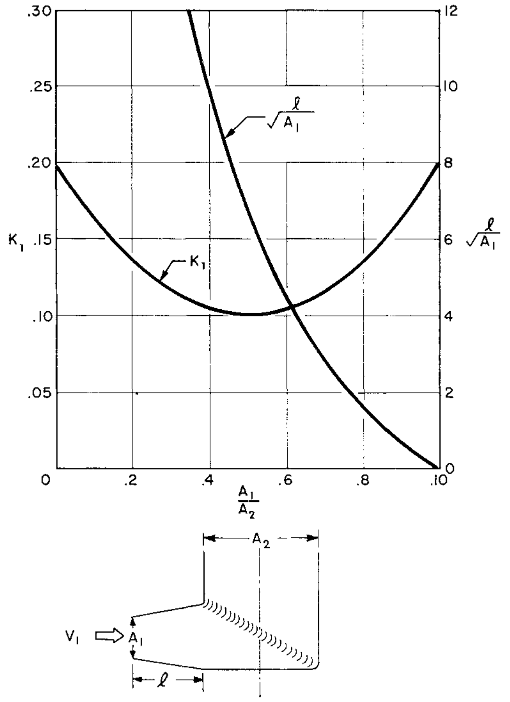

Fig. 3 Diffusor turn loss coefficients.

loss of the diffuser vane combination shown may be expressed as

$$h_l/V_1^2 = 0.2[1 - (A_1/A_2)^2] + 0.2(A_1/A_2)^2 \qquad (13)$$

Since for the case of zero prediffusion or acceleration $V_1 = V_0$ and

$$h_l/(V_0^2/2g) = k_1$$

then,

$$k_1 = 0.2[1 - 2(A_1/A_2) + (A_1/A_2)^2] \qquad (14)$$

Assuming a similar diffuser-turn for the strut loss, it follows from the continuity equation that

$$k_2 \cong (A_1/A_2)^2 k_1 \qquad (15)$$

Equation (14) is plotted in Fig. 3. The important point to note is that k_1 has a minimum value of 0.1 at $A_1/A_2 = 0.5$. Since, for a 6° conical diffuser, this value of $A_1/A_2 = 0.5$ would require an impractical length, it is fortunate that the k_1 curve is quite flat in the vicinity of the minimum. In order to establish a method of determining practical values of A_1/A_2, the relationship between diffuser length l and A_1 and A_2, for a 6° conical diffuser, was derived and is given by the following equation

$$(A_2/A_1) = 1 + 0.15l/(A_1)^{1/2} \qquad (16)$$

Equation (16) also is plotted in Fig. 3. Thus, the designer selects a maximum practicable value of l, determines A_2/A_1 from Eq. (16) or Fig. 3, and then finds k_1 from Eq. (14) or Fig. 3. For the case of the strut losses, l would be taken as the length of the strut and k_2 determined by Eq. (15).

Maximum Possible Efficiency

For a given hydrofoil craft and speed, the relationships presented in the preceding section along with Fig. 2 enables the designer to determine the near maximum efficiency. For example, the minimum possible value of $k_1 + k_2$ for a hydrofoil craft utilizing 2-bend diffusers is 0.125 so that

$$K_{\min} = 0.125 + 0.01\left(1 - 0.125 - \frac{2gh}{V_0^2/2g} + \frac{2gH}{V_0^2/2g}\right) + \frac{2gh}{V_0^2/2g}$$

or

$$K_{\min} \cong 0.135 + 0.01(2gH/V_0^2) + 0.99(2gh/V_0^2) \qquad (17)$$

For a small high-speed boat where h is say 3 ft and $V_0 =$ 100 knots, the elevation term is about 0.006 so that $K_{\min}$ is about 0.15. From Fig. 2, $\bar{\eta}_{\text{opt}}$ is 0.72. Using a pump efficiency of 0.9 the approximate maximum propulsive efficiency of this craft using water-jet propulsion is about 0.65. As given by Fig. 2, the optimum pump head to drive this craft should be just equal to the forward velocity head.

For a large low-speed hydrofoil boat where h is say about 21 ft and V_0 is 50 knots, the elevation term in Eq. (17) is about 0.18 so that $K_{\min}$ is about 0.34. From Fig. 2, $\bar{\eta}_{\text{opt}}$ is 0.63 and using $\eta_p = 0.9$, the approximate maximum over-all propulsive efficiency is 0.567. For this craft the optimum pump head as given by Fig. 2 is about twice the forward velocity head.

In order to illustrate how the scoop inlet area, pump discharge, nozzle areas, and power requirements are determined, the last example, shown previously, is now continued with the further specification that the thrust required at 50 knots is 100,000 lb. The procedure is as follows: 1) Using Eq. (3) and the value of $\eta = 0.567$; $P = [100{,}000(84.5)]/([567)(550)] =$ 27,100 hp; 2) From Fig. 2, as pointed out in the preceding paragraph, $H^* = 2$ so that $H = 225$; 3) From Eq. (2) using $H = 225$, $\eta_p = 0.9$, $P = 27{,}100$, $Q = [(0.9)(27{,}100)(550)]/[(1.96)(32.2)(225)] = 948$ cfs; 4) Assuming 2 scoops and zero prediffusion or acceleration, area/scoop = 948/2(84.5) = 56.2 ft^2; 5) From Eq. (5), $V_j = V_0(1 - K + H^*)^{1/2}$ so that $V_j = 84.5(0.63 + 2)^{1/2} = 137$ fps; and 6) Assuming 2 nozzles, area/nozzle = 948/2(137) = 34.6 ft^2.

The previous results for this large low-speed boat reveal the compromise that will have to be made in each design between propulsive efficiency and weight. The large scoop and nozzle areas which result at optimum efficiency obviously are too large and would result in increased strut size and weight, and water weight aboard the craft which may approach the absurd. The design should therefore proceed exactly as outlined previously following the $K = 0.34$ line in Fig. 2 in the direction of increasing H^*. It will be noted that the value of $\bar{\eta}$ decreases below the optimum initial value obtained, but the decrease (although significant) is not extremely great. The lower discharges that will result from the increased heads will permit smaller scoop and jet areas and consequently reduced strut drag and weight and weight of water on board. A more detailed example of the approach to the hydrodynamic design of a water-jet system is presented in a subsequent section that describes the preliminary design of a 500-ton 80-knot craft.

Pump Selection for Water-Jet Propulsion

It is clear that both the propulsive efficiency and weight of a water-jet propulsion system is dependent on the pump or pumps selected to produce the optimum head and discharge determined by the methods outlined in the previous section. These pumps should operate cavitation free over the entire speed range at as high a rotational speed as possible in order to minimize their weight, the weight of the contained water and the weight of reduction gear requirements.

Pump Types

A useful parameter defining the cavitation characteristics of pumps is the suction specific speed S given by the following equation:

$$S = [n(Q)^{1/2}]/(H_{sv}^{3/4}) \qquad (18)$$

where

n = rotational speed
Q = discharge
H_{sv} = total head above vapor pressure at the impeller; referred to as the net positive suction head

For preliminary design purposes, experience has shown that if n, Q, and H are prescribed in revolutions per minute, gal/min, and feet, respectively, the approximate maximum value of S for cavitation free performance is 12,000. Since high pump rotational speeds will tend to decrease the pump weight for prescribed Q and H, the maximum possible value of S is desirable.

Another important parameter, which is related to the pump type (that is, centrifugal, mixed flow, or axial flow) is the specific speed n_s defined as

$$n_s = [n(Q)^{1/2}]/H^{3/4} \tag{19}$$

where H is the head developed by the pump.

If the suction specific speed is taken as 12,000 and Eqs. (18) and (19) are combined, the specific speed can be written as

$$n_s = 12{,}000(H_{sv}/H)^{3/4} \tag{20}$$

The value of H_{sv} in a water-jet propulsion system depends on the forward speed, the static lift h, and the internal head losses. In a hydrofoil craft installation the maximum thrust requirement is generally demanded at takeoff speed; furthermore, the power plant is usually capable of producing a considerable increase in power during the short duration of the craft takeoff and acceleration. Consequently, if the cruise head and discharge are maintained constant at takeoff, the increased thrust requirement can be produced in spite of the decrease in efficiency at the increased value of H^*. One method of maintaining the cruise discharge at the reduced takeoff speed is to open an auxiliary takeoff scoop. The advantage of this constant specific speed schedule of operation is that it insures cavitation free operation of the pump and maximum pump efficiency at both cruise and takeoff. Although experience may alter this schedule somewhat, it simplifies the present analysis without any important compromise.

With the preceding assumption of equal takeoff and cruise head, it is clear that the lower value of H_{sv} at takeoff will control the maximum specific speed of the pump. If the takeoff speed is assumed to be equal to one-half the cruise speed, and

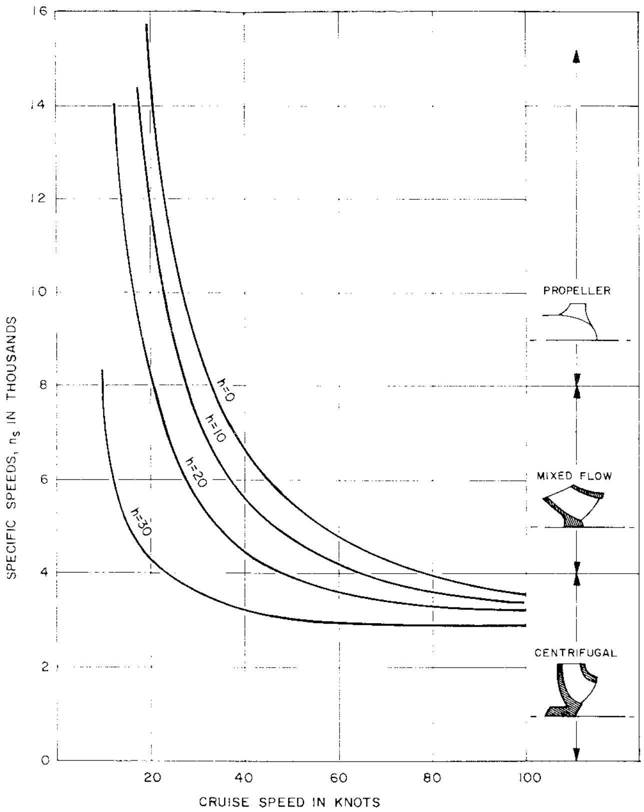

Fig. 4 Single-stage pump types for water-jet propulsion (H^* assumed equal to 1.5).

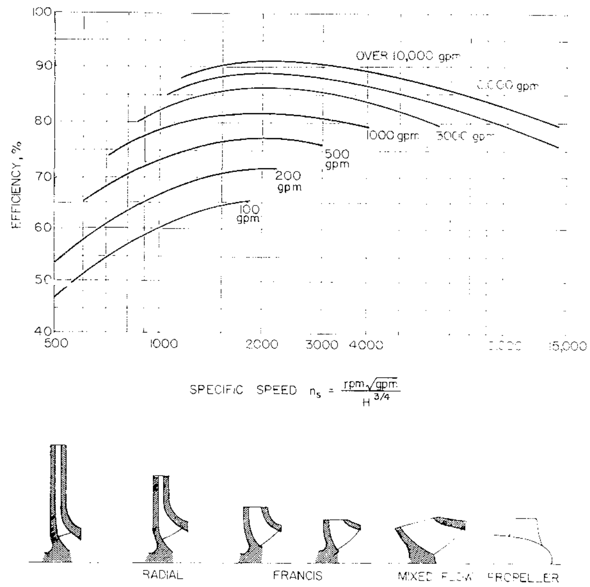

Fig. 5 The influence of capacity and specific speed on pump efficiency (Stepanoff).

the head is developed across a single pump stage, the following expression for pump specific speed may be derived

$$n_s = 12{,}000\left[\frac{33 + (1 - k_1 - k_2)\frac{1}{4}(V_0^2/2g) - h}{(H^*)(V_0^2/2g)}\right]^{3/4} \tag{21}$$

where V_0 is the cruise speed.

Using a typical optimum value of $H^* = 1.5$ and $k_1 + k_2 = 0.125$, Eq. (21) becomes

$$n_s = 12{,}000\ [(33 + 0.22V_0^2/2g - h)/(1.5V_0^2/2g)]^{3/4} \tag{22}$$

Figure 4 shows the pump specific speed at various craft forward speeds for several values of static lift as determined from Eq. (22). The types of pumps associated with these specific speeds also are shown in Fig. 4. It should be noted that the higher specific speeds in each of the ranges of pump types also could be achieved by multistaging a higher specific speed type. For example, a single stage, 4700-specific speed, mixed flow pump, could be replaced by two stages (series) of axial flow pump, each having a specific speed of 8000. Although, in some cases, series multistaging may be used to simplify the arrangement of a water-jet system, the pump itself generally has greater weight than a single stage unit of lower specific speed. Furthermore, the efficiency of single stage pumps is generally higher than multistaged pumps particularly if the single stage unit has a specific speed in the vicinity of 2500.

The efficiency of pumps for various specific speeds and sizes is shown in Fig. 5. Figure 5 illustrates the advantages of maintaining the discharge greater than 10,000 gal/min and of operating at specific speeds near 2500.

After determining the optimum pump specific speed by the methods outlined in the preceding discussion, it is necessary to select the number of such pumps, operating in parallel, required to produce the required thrust. The total discharge required may be determined from Eqs. (2) and (3) as

$$Q_{\text{total}}(\text{cfs}) = 2T/(\rho H^* V_0 \tilde{n}) \tag{23}$$

where T is the total thrust required.

Since the product of the rotational speed and (discharge)$^{1/2}$ are fixed for a prescribed head and specific speed, it is clear that the development of the entire thrust by a single pump may lead to unsatisfactorily low values of impeller rotational speed. To produce the required head and discharge at these low revolutions per minute may require very large reduction gearing and impeller dimensions. Figure 6 presents typical parameters related to the impeller dimensions in terms of the head, discharge, and rotational speed. The parameters pre-

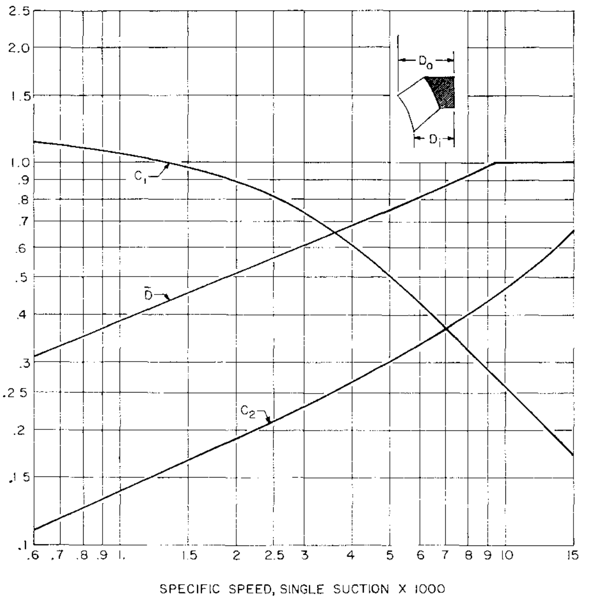

Fig. 6 Impeller constants (Stepanoff).

sented are defined as follows

$$C_1 = 2gH/U_2^2 = (8gH)/(\omega^2 D_0^2) \qquad \bar{D} = D_i/D_0$$

$$C_2 = Q/[A_i(2gH)^{1/2}]$$

where

u_2 = impeller peripheral velocity
D_i = impeller inlet diameter
D_0 = impeller outlet diameter
A_i = impeller annular inlet area
ω = rotational speed in rad/sec

Since C_1 is approximately defined for a given specific speed, it is clear that a prescribed H will be developed at smaller impeller diameters as the rotational speed is increased. Furthermore, since the internal volume and weight of a pump vary as D_0^3, it is clearly advantageous to use as high a rotational speed as practicable. Consequently, multiple parallel pumps are essential to the problem of minimizing the weight of a water-jet propulsion system required to produce large thrust. The optimum number of these pumps depends primarily on using sufficiently large pumps ($Q > 10{,}000$ gal/min) so as not to compromise efficiency and the selection of a rotational speed compatible with the prime mover, which minimizes gearing weight. Thus, for a prescribed rotational speed, the number of pumps required N is determined from Eqs. (18) and (19) as

$$N = [n^2 Q_{\text{tot}}(\text{gal/min})]/(n_s^2 H^{3/2}) \quad \text{(single suction)} \qquad (24)$$

If the discharge/pump, Q_{tot}/N, is less than 10,000 gpm, then consideration must be given to altering the rotational speed so as to reduce the number of pumps and so increase their capacity and efficiency.

One method of paralleling centrifugal or low specific speed, mixed flow pumps so as to achieve higher rotational speeds, for a specific discharge and head, has been in popular use for many years in the form of the double-suction pump. This paralleling is accomplished on a single impeller by allowing inflow to both sides of the impeller. Such a pump is clearly advantageous in a water-jet system since it effectively halves the number of impellers required, that is Eq. (24) becomes

$$N = [n_2 Q_{\text{tot}}(\text{gal/min})]/(2n_s^2 H^{3/2}) \quad \text{(double suction)} \qquad (25)$$

One additional scheme for increasing the rotational speed of the pumps required for high-speed craft utilizing water-jet propulsion is to utilize a lightly loaded axial flow stage upstream of the centrifugal pump inlet and so effectively increase the suction head of the centrifugal stage. The design of such axial flow inducers has received much attention recently in the development of light-weight turbopumps for space craft.

Preliminary Design of an 80-Knot 500-Ton Hydrofoil Craft

Preliminary designs of water-jet propulsion systems for large hydrofoil craft operating in the 80-knot speed range which utilize "off the shelf" double-suction centrifugal pumps have tended to be impractically heavy. For example, a 500-ton 80-knot craft design, utilizing 12 double-suction centrifugal pumps driven by 6 Pratt and Whitney FT-4A gas turbines (operating rotational speed range of 3500 to 4000 rpm, each capable of 30,000 hp maximum and 20,000 hp continuous), resulted in a propulsion system weight, exclusive of the en-

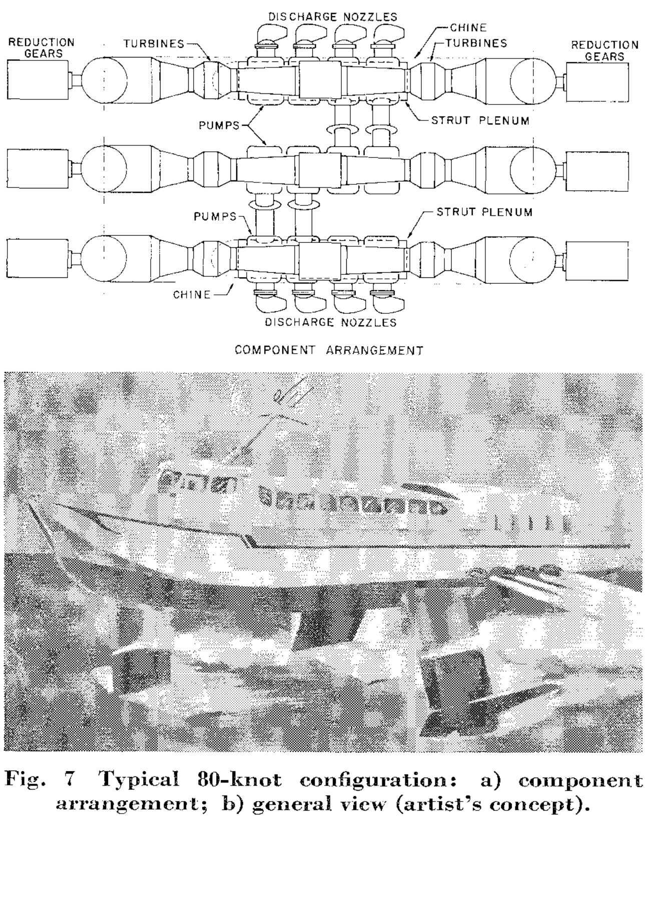

Fig. 7 Typical 80-knot configuration: a) component arrangement; b) general view (artist's concept).

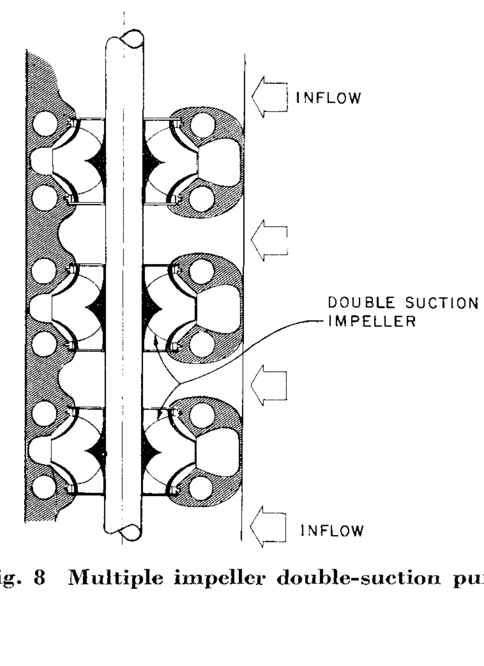

Fig. 8 Multiple impeller double-suction pump.

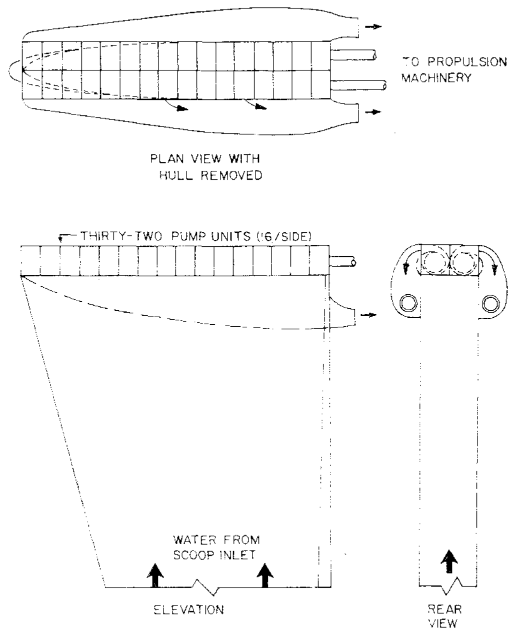

Fig. 9 Multiple impeller pump arrangement with pumps in hull.

gines, of 150 tons. The thrust requirements of this particular craft were 400,000 lb at takeoff and 300,000 lb at the 80-knot cruise speed. Clearly, a propulsion system weight that is 30% of the craft total weight is not attractive. Figure 7 is an illustration of the scoop and pump arrangement for this 12-pump example of a water-jet system for a 500-ton 80-knot craft.

The estimated 150-ton weight of the previous example may be approximately broken down into 1) pumps with water, 80 tons; 2) reduction gears, 50 tons; and 3) water in the strut and other ducts, 20 tons. Thus, in order to reduce weight, 1) the pumps must be made smaller, 2) the gear box reduced in weight or eliminated, and 3) the pumps located lower in the strut, if possible. The first two items can be accomplished by increasing the pump rotational speed and utilizing a larger number of smaller pumps.

For example, if the methods outlined earlier are used, it may be concluded that 64 double-suction centrifugal pumps operating at 3600 rpm will produce the required head and discharge to meet the specifications of the design example. Off the shelf pumps could of course be utilized, but a further savings in weight will be accomplished by installing some fraction of the total pumps within a single casing with the impellers on the same shaft; for example, 16 impellers per pump. Such a pump is illustrated in Fig. 8.

These proposed pumps would utilize impellers, approximately 1 ft in diameter, have an inlet diameter of approximately 9 in., and the impellers would be spaced approximately 1 ft apart on the shaft. Fabricated of aluminum alloy, such a pump (with water) would weigh only about 2.5 tons. Four such pumps would weigh 10 tons as compared with 80 tons estimated in the off the shelf 12-pump system.

Equally important is that these new pumps require no reduction gears. Possibly, for arrangement purposes, some bevel gearing will be required, but these gears will be relatively light, probably less than 10 tons. Furthermore, these gears are installed in the hull and not at the lower end of the strut as is required in a supercavitating propeller system.

A schematic diagram of a typical arrangement that utilizes such multiple impeller pumps is shown in Fig. 9. Again estimating the weight of water in the struts as 20 tons, the total weight of this water-jet system is approximately 40 tons. Since a supercavitating propeller system requires both a reduction gear in the hull and bevel gears in the propulsion pod, the over-all weights of the two systems may not be greatly different.

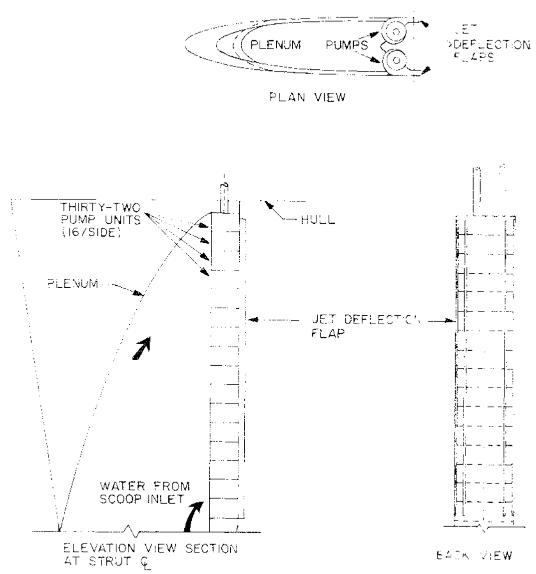

Fig. 10 Multiple impeller pump arrangement with pumps in strut.

Once the multiple impeller pump idea is grasped, many promising arrangements are suggested. One of these is illustrated in Fig. 10. In this scheme the pumps are located vertically in the base of the strut, and so another 10 tons of weight of water in the strut can be reduced. This idea is particularly attractive, because little or no ducting is required to the exhaust nozzle; the jets are exhausted immediately out the strut base. Deflection of the jet for maneuvering and backing becomes very simple. The efficiency of the system is improved because of the reduced static lift. Such a system does face the problem of supplying large power down a shaft with a flexible support structure. Furthermore, the drive arrangements within the hull may be somewhat complex, but this doesn't seem to be a particularly difficult problem to solve. The arrangement will be considerably improved, for example, if a three-strut system were used, with a single-20-impeller pump, or two 10-impeller pumps in each strut.

Conclusions

Water-jet propulsion of high-speed hydrofoil craft can be a strong competitor of the supercavitating propeller at high speeds. Although, water-jet propulsion eliminates the problem of locating the propeller, strut, and wing so as to avoid mutual cavity interference and the problem of supplying large power through right angle drives at the propulsion pod, new problems associated with scoop inlet design and the development of light-weight pumps are introduced. Neither of these problems are believed to be insurmountable.

The propulsive efficiency of water-jet systems for craft in the 80-knot speed range is estimated to be approximately 55%. Utilizing the concept of multiple impeller, double-suction centrifugal pumps, it seems possible to limit the weight of the water-jet system to approximately 5 to 10% of the craft gross weight. Such a propulsion system is very attractive. With adequate research and development effort, it is believed that these estimates can be achieved.