full of gas suspended on an umbilical 100 ft above one's base.

The field of diving from submerged bases is as yet young. A relatively small amount of experience has been accumulated. It is too early to be able to say that one concept of life-support equipment is superior and another is unsatisfactory. Much more research, development, and field trial is needed. Each concept and procedure must be analyzed by the diving team with reference to situations existing at the time and place of the dive. Safety in the developmental phase can be enhanced if three basic guidelines are followed:

1) Each member of the crew is competent in his job and has a thorough working knowledge of the equipment being used.
2) The equipment is, within the limits of the state-of-the-art, conceptually and mechanically sound.
3) Means are included in the wet life support-system which warn the diver and supporting crew of hazards before they reach a critical level. An example would be the monitoring of oxygen partial pressures, which in themselves give little warning of dangerous variations prior to the onset of unconsciousness.

Almost all existing government standards and regulations, with respect to diving, apply to shallow water situations. Those which relate to operational procedures are usually inappropriate to commercial operations. Often they have been conceived and written by persons oriented to military or amateur diving who are unacquainted with commercial problems and procedures. In general, they are loosely, if at all, enforced, and serve primarily as a weapon to condemn a violator after an accident has occurred.

What can be done in the future to establish guidelines which will be workable and will effectively enhance the safety of deep water diving? The mere publication of a list of rules is unlikely to solve many problems. A significant question exists as to whether the industry is yet sufficiently stabilized to allow the establishment of intelligent guidelines. However, there are two areas where worthwhile results might be achieved. Most important is the establishment of adequate training programs for divers and supporting personnel. The author is convinced that efforts in this direction will be far more productive than fixed regulations. No amount of regulation can compensate for ignorant or foolish actions. Licensing of divers and other personnel according to grades of capability might be considered. The second area lies in the certification of equipment sold on the open market. Although the safe use of equipment is the responsibility of the diver himself, he has the right to expect that gear acquired on the open market is as safe and well built as we know how to make it. The establishment of a competent certifying authority would do much to provide this.

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Performance of Waterjet Propulsion Systems—A Review of the State-of-the-Art

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Waterjets may solve some problems facing high-speed marine propulsion. To judge their potential, a literature study was conducted to determine the state-of-the-art of waterjet technology with emphasis on 1) performance criteria and data, and 2) performance evaluation and experimental techniques. A general lack of definitive experimental data was noted, with the greatest need for information on the design of efficient, cavitation-free, high-speed inlets. Work also is needed on lightweight pumps capable of sustained high performance under relatively severe cavitation. Thrust efficiency was usually confused with propulsive efficiency, the product of thrust efficiency and the hull/waterjet interaction efficiency, which is a more definitive parameter but inherently more difficult to obtain. Model experiments are required to separate resistance and propulsive forces in determining this efficiency. A review of model experimental techniques and facilities shows the capability for carrying out the necessary experiments.

Nomenclature

C_p = $(p - p_o)/\frac{1}{2}\rho V^2$
d = diameter, L
e = mean roughness height, L
E = Euler number, $V/(\rho/2\Delta p)^{1/2}$
F_n = Froude number, $V/(lg)^{1/2}$
g = gravitational acceleration, ft/sec^2, L/T^2
H = pump head rise, ft, L
H_e = exit nozzle head, ft, L
H_i = inlet head, ft, L
H_1 = absolute pressure at shaft centerline—vapor pressure, ft, L
H_L = system head loss
H_{sv} = net positive suction head *npsh*, ft, L
H_s = static head, atmospheric + depth, L
J = advance ratio, V/Nd
K_H = head rise coefficient, gH/N^2d^2
K_L = system loss coefficient, $H_L/V_J{}^2/2g$
K_q = torque coefficient, $q/N^2d^5\rho$
K_t = thrust coefficient, $T/N^2d^4\rho$
k = jet velocity ratio, V_j/V
m = inlet velocity ratio, V_j/V_i
l = length, ft, L
l_s = ship length and model length, respectively, ft, L
M = Mach number, $(\rho V^2/E)^{1/2}$
N = rmp or rps, $1/T$
N_s = pump specific speed, $NQ^{1/2}/(gH)^{3/4}$
OPC = P_E/P_B
p = static pressure, lb/ft^2, M/LT^2
P_E = effective horsepower ($R \times V/550$), ft-lb/sec, ML^2/T^3
P_B = brake horsepower ($2\pi Nq_{\text{brake}}/550$), ft-lb/sec, ML^2/T^3
P_D = propeller horsepower (delivered), ML^2/T^3

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P_l, P_o = power input and output, respectively, ft-lb/sec, ML^2/T^3
P_p = pump horsepower, $\rho gQH/550$, ft-lb/sec, ML^2/T^3
P_s = shaft horsepower, ft-lb/sec, ML^2/T^3
P_T = thrust horsepower $(T \times V/550)$, ft-lb/sec, ML^2/T^3
p_v = vapor pressure, lb/ft²
p_o = freestream static pressure, lb/ft²
Q = volume flow rate, ft³/sec, L^3/T
q = shaft torque, lb-ft, ML^2/T^2
R = resistance, lb, ML/T^2
Re = Reynolds number, Vl/ν
r = impeller radius, ft, L
T = thrust, lb, ML/T^2
t = thrust deduction, $1 - R/T$
u_a, u_t = axial and tangential velocity, respectively, fps, L/T
V = ship velocity, fps, L/T
V_i = inlet velocity, fps, L/T
V_j = jet velocity, fps, L/T
V_p = velocity through pump, fps, L/T
W = Weber number, $\rho l V^2/\xi$
w = wake fraction, V_a/V
w_1 = weight rate of flow, lb/sec, ML/T^3
V_o = freestream velocity, fps, L/T
V_a = propeller speed of advance, fps, L/T
β = advance angle, deg
β_i = hydrodynamic pitch angle, deg
ΔH = $(H_e - H_i)$, ft, L
ΔV = $(V_j - V_i)$, fps, L/T
ΔY = elevation (inlet to pump), ft, L
η_A = external efficiency
η_{hull} = hull efficiency, $1 - t/1 - w$
η_j = jet efficiency, $V/(V + \Delta V/2)$
η_{jpr} = real jet propulsive efficiency
η_p = propeller efficiency, $TV_a/550\ P_D$
η_{pump} = pump efficiency, $\rho gQH/550\ P_D)$
η_s = system efficiency
γ = specific weight, lb/ft³, M/L^2T^2
τ = linear ratio, l_s/l_m
μ = coefficient, dynamic viscosity, lb-sec/ft², M/LT
ν = coefficient, kinematic viscosity, ft²/sec, L^2T
ω = angular velocity, rad/sec, $1/T$
φ = flow coefficient, Q/Nd^3
ρ = mass density, slugs/ft³, M/L^3
α = cavitation index, $H_1/V^2/2g$
σ_{Thoma} = Thoma cavitation number, H_{sv}/H
τ_o = shear stress, lb/ft², M/LT^2
τ_o = surface tension, lb/ft, M/T^2

Introduction

WATERJET propulsion of marine craft is not new. As pointed out by Taggart,[1] evidence exists of experimental evaluation of a jet-propelled craft in England in 1661; by 1900 inherent disadvantages in ducting losses and weight were recognized. Improvements within the succeeding 60 years have not fully realized the attractive advantages of waterjets. Extensive historical surveys have been made by Papir[2] and Schuster et al.[3] A brief history of waterjets[4] was published in 1962 by Engel et al.

A waterjet is a marine propulsor in which water is fed to internal pumps, which add energy and expel the water aft through a nozzle at a higher velocity than the incoming stream, with thrust achieved through the resulting momentum exchange. It can be classed as an internal ducted propeller with a long duct. In this study, a waterjet uses water alone, as opposed to water-gas mixtures.

A waterjet system, as distinguished from all other types of shipboard jet propulsion, is located mostly within the ship hull. Therefore, it has 1) an intake duct that inducts fluid from outside the hull, 2) a pump that transmits energy to this fluid, and 3) an exhaust duct and nozzle that guide the fluid jet out of the hull.

It could be expected that a waterjet-propelled ship, having no elements protruding beyond the limits of the ship's hull, would have less drag than a ship with a conventional propeller. However, the internal flow is accompanied by hydraulic losses. Resistance may change significantly when the ship is propelled, since the propulsion system changes the distribution of pressures on the hull. The intake opening in the stern can strip off the boundary layer, which could decrease frictional resistance. On the other hand, an opposing force to the intake induction momentum can suck the hull down with possible increase in trim drag and, thus, form drag. Also, the ejection of the jet in the vicinity of the ship's wake can change the magnitude of the useful thrust. These are examples of interaction effects of hull flow and waterjet flow. The determination of the reaction coefficients under such conditions is a prime need in research on waterjets.

The importance of waterjets can be ascertained by considering what this form of propulsion can offer. The propulsor itself is generally more expensive, less efficient, heavier, and more complicated than a propeller. However, for special purposes, such as shallow-draft operation or high-power, high-speed operation, waterjet propulsion may eliminate, or diminish, unavoidable disadvantages of a propeller.

A number of significant development problems remain to be solved before the full use of waterjet propulsors can be realized. These include 1) proper design of hull inlet to prevent separation and cavitation, including during yaw and heave conditions, 2) optimization of system ducting and pump elevation losses, 3) design of high-speed axial-flow waterjet pumps capable of meeting the cavitation, efficiency, and off-design performance required for vehicle takeoff and cruise conditions, and 4) adequate methods and techniques for both model-scale and prototype performance evaluations.

This paper is a critical survey of current technical literature on waterjet propulsion of marine craft. Problems are discussed in the light of the advantages and disadvantages of this type of propulsion. Theoretical treatments and experimental techniques for evaluating performance of system components, and of completely installed systems, plus information on performance evaluation found in the literature are presented. A list of unsolved problems existing in the light of the conclusions reached by theory and experiment, and recommendations for performance parameters and modeling techniques complete the paper.

General Considerations

Existing Literature

The literature varies in its treatments of aspects of waterjet propulsion. Some are feasibility studies; some are general studies proposing procedures for system design; and others are theoretical and/or experimental investigations. The variables considered and measurements made in the experimental work are given in Table 1. Information from the literature pertains to theoretical and experimental treatment of 1) waterjet propulsion system components, including pumps, inlets, and ducting; 2) complete waterjet propulsion systems; and 3) waterjet propulsion system installed in a hull, including efficiency, model, and prototype testing.

Applications

Current U.S. waterjet propulsion applications include small pleasure planning craft, small military river patrol boats,[5] and hydrofoil craft. Seriously proposed are larger military patrol planning boats as well as high-speed hydrofoil and captured-air-bubble craft.

The majority of the small pleasure boats are of 200–300 hp; several thousand are in service, with propulsors of the Buehler,[6] Berkeley, or Jacuzzi design. For the most part, European manufacturers have comparable units. Qualitative data on operational performance of complete waterjet-propelled hulls have been obtained, taking into consideration the mission of the craft, by studying reports of owners and operators. Efficiency is not a prime requirement for private water-ski or sport runabouts at present. Instead, speed, maneuverability, boat handling, noise and vibration,

Table 1 Summary of experimental waterjet work

	Characteristics \ Reference No.	1	2	3	4-10	11	12	13	14	15	16	17	18	19	20	21	22	23 28	29	30
	Pump Impeller		X		*	*	*	*	X	X					X		X		X	
	Pump Cavitation		X						X	X							X		X	
	Static Inlet																X			
	Dynamic Inlet								X								X			
Model Boat	Propulsive Efficiency, η_D																			
Model Boat	Thrust Efficiency, η_T																			
Model Boat	Pump Characteristics																			
Model Boat	Inlet Flow			X																
Model Boat	Nozzle Flow			X																
Model Boat	Thrust & Drag			X																
Model Boat	Lift																			
Model Boat	RPM																			
Model Boat	SHP																			
Model Boat	Trim & Heave			X																
Prototype Boat	Static Inlet			X					X	X	X	X	X							
Prototype Boat	Dynamic Inlet			X					X	X	X	X	X							
Prototype Boat	Propulsive Efficiency, η_D										X									
Prototype Boat	Thrust Efficiency, η_T			X					X		X	X	X							
Prototype Boat	Pump Characteristics			X					X	X	X	X	X							
Prototype Boat	Inlet Flow			X					X	X	X		X							
Prototype Boat	Nozzle Flow			X					X	X	X	X	X							
Prototype Boat	RPM			X					X	X	X	X	X							
Prototype Boat	SHP			X					X	X	X	X	X							
Prototype Boat	Trim & Heave			X					X		X	X	X							
Prototype Boat	Boat Velocity			X					X	X	X	X	X							

	Characteristics \ Reference No.	31	32	33	34	35	36	37	38	39	40	41	42	43	44	47	48	49	50	51	52	53	54	55	56	57	58	59 70
	Pump Impeller	X	*	*	*	*	X	X	X		X	X	X	X	*												X	
	Pump Cavitation	X					X	X	X		X	X	X	X														
	Static Inlet																			X°	X°	X°	X°	X†	X†	X		
	Dynamic Inlet																											
Model Boat	Propulsive Efficiency, η_D																											
Model Boat	Thrust Efficiency, η_T																											
Model Boat	Pump Characteristics																											
Model Boat	Inlet Flow																											
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Prototype Boat	Static Inlet																										X	
Prototype Boat	Dynamic Inlet																											
Prototype Boat	Propulsive Efficiency, η_D																											
Prototype Boat	Thrust Efficiency, η_T																										X	
Prototype Boat	Pump Characteristics																											
Prototype Boat	Inlet Flow																										X	
Prototype Boat	Nozzle Flow																											
Prototype Boat	RPM																											
Prototype Boat	SHP																										X	
Prototype Boat	Trim & Heave																											
Prototype Boat	Boat Velocity																											

*Textbook, °Marine Condenser Scoops, †Aircraft-type Inlets.

and elimination of the external propeller for safety are stressed. At present, some in-service designs are good with respect to maneuvering ahead and handling, but efficiency is lower than comparable conventional propulsion installations.

Recent attention has been given to waterjets for high-speed oceangoing ships[7,8] of the nondisplacement type. Serious consideration is mainly limited to hydrofoil craft and, more recently, air-cushion vehicles of the captured-air-bubble (CAB) type. Certain major design tradeoffs for these ships are highly dependent on mission cruising range. For the smaller high-speed but short-range (duration of a few hours) craft, the weight of propulsion machinery must be kept low; however, high system efficiency may not be critical. Efficiency becomes increasingly important as the ratio of fuel weight to gross weight increases and fuel weight is significant for a long cruise-range oceangoing craft. In this respect, the CAB sidewall configuration achieves a more efficient water ducting system than a hydrofoil, "up-the-strut" craft.

Advantages and Disadvantages

The selection of any propulsion system is based on over-all value judgments. Following is a list of advantages and disadvantages in waterjets. The final application determines how these are weighed and compared.

Practically every publication on waterjet propulsion mentions certain inherent advantages, but very few unqualified statements can be made because possible advantages depend on the application and must be compared with all other existing propulsor types. Potential advantages are 1) elimination of external underwater appendages; 2) free choice of location of the propulsion machinery; 3) elimination of complex transmission machinery where right-angle drives are required, such as in hydrofoil craft strut-propeller applications; 4) possible alleviation of underwater radiated propeller cavitation noise through more control of static pressure and removal of the propeller from the main body of water; 5) control of steering and maneuvering directly from the propulsor† (note, however, that this is done by attaching control surfaces to the waterjet nozzle, which, in effect, calls the rudder part of the propulsor); 6) possible alleviation of detrimental effects of propeller vibration because of better control of the impeller inflow characteristics; and 7) on towing boats and icebreakers, the production of greater tow-rope pulls than an open propeller, although a Kort nozzle will be more efficient.

Potential disadvantages must also be qualified in the light of application and competitive propulsors. Considering only the propulsor, 1) waterjet systems are heavier than most other practical propulsor systems (note that the weight of the water in the system must be included); 2) in general, a waterjet system will not be as efficient as a propeller system; 3) cavitation in the inlet system can adversely affect performance, which means that new sources of cavitation are to be considered; and 4) impeller access is poor compared to conventional propeller designs, making inspection, repair, or removal of debris difficult.

† Right-angle drive units have been built successfully for 360° directional control.

System Component Performance

Efficiency

The over-all propulsor performance of special-mission, high-speed craft is dependent on a number of factors. For a specific mission and design speed, consideration should be given to payload, horsepower, specific fuel consumption (SFC), range, dash capability, gross weight, reliability, etc.

A number of special performance coefficients or merit factors have been proposed to lump these variables together into meaningful parameters. Examples are the so-called Karman-Gabrielli factor[9] and Telfer's merit factor.[10] The Karman-Gabrielli factor is

$$MKG_1 = \frac{1}{8800} + \left(\text{equivalent } \frac{L}{D} \text{ ratio}\right) \times \frac{W_{\text{payload}}}{W_{\text{initial}}} \times V \tag{1}$$

A portion of this factor pertains to a hydrodynamic propulsion parameter. On normal, conventional propeller-hull installations, over-all powering performance of the craft is based on[10,11]

$$\text{OPC} = \frac{\text{effective horsepower}}{\text{brake horsepower}} = \frac{P_E}{P_B} = \frac{RV}{P_B} = \eta_p \times \eta_{\text{hull}} \tag{2}$$

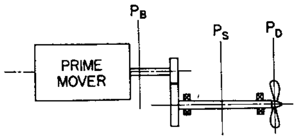

Fig. 1 Schematic of power transmission.

where P_B is measured at the output of the prime mover; P_E is effective power to overcome resistance; R is towed hull drag (sometimes utilizing bare hull drag and at other times using appended hull drag); and η_p is for the "behind ship" condition.

For purposes of discussing efficiency of propulsion, the term over-all propulsive coefficient (OPC) will not be used in this paper; a quasi-propulsive coefficient, called propulsive efficiency η_D, will be used instead:

$$\eta_D = P_E/P_D = (\text{OPC})P_B/P_D \tag{3}$$

P_D is delivered to propeller, and P_D/P_B is transmission efficiency and reflects the power losses between the prime mover coupling and the propeller due to shafting, bearings, etc.

Model testing usually involves measurement of P_D. In full-scale trial work, P_S is normally measured by a torsionmeter on the inboard shafting intermediate between propeller and prime mover (see Fig. 1). Because of the variation in locations for measuring P_S, and since P_B would not correlate directly from model to full-scale, it is sensible to utilize P_D.

Propulsive efficiency η_D is, by definition,[71] equal to

$$\eta_D = RV/P_D \tag{4}$$

where R = resistance, in general. This is not the "propulsive efficiency" used frequently in the waterjet literature, TV/P_S. The latter can be referred to as thrust efficiency η_T, since it is clearly not identical to propulsive efficiency;

$$\eta_T = TV/P_S \tag{5}$$

η_T has been widely used because it is relatively easy to measure in sea trials and its form corresponds to the expression for jet efficiency η_j. η_j is sometimes also known as ideal propulsive efficiency or Froude efficiency.[12,13]

In attempting to improve the propulsive performance of a system, it is necessary to relate over-all propulsive efficiency of a waterjet-propelled craft to the aggregate of the propulsion component efficiencies. In conventional propulsion

$$\eta_D = \frac{RV}{P_D} = \frac{T(1-t)V}{P_D} = \eta_{\text{propeller "behind"}} \times \eta_{\text{hull}} \tag{6}$$

where R = bare hull resistance (since R_{bare} is the base drag); $\eta_{\text{propeller "behind"}} = TV(1-w)/P_D$; $\eta_{\text{hull}} = 1 - t/1 - w$, by definition; and $1 - t$ = thrust deduction. The term $(1 - w)$, which is called wake gain, for waterjet pumps is tied in with the pump efficiency; the propeller does not come in contact directly with the hull wake due to the long duct. Therefore, η_{hull} is not directly applicable to waterjet systems. Instead, for a waterjet, $\eta_{\text{hull/inlet}}$ must be defined.

A reasonable choice of form for relating the system component efficiencies to the over-all propulsive efficiency is the following. Equate

$$\eta_D = \frac{RV}{P_D} = \frac{T(1-t)V/550}{P_D} \text{ to } \eta_{\text{pump}} \times \eta_{\text{jet}} \times \eta_{\text{system}} \times \eta_{\text{hull/inlet}}$$

where $\eta_{\text{pump}} \times \eta_{\text{jet}}$ is essentially equal to $\eta_{\text{propeller}}$, depending on the way the flow velocities are chosen; η_{system} (or η_S) by definition is proportional to the ratio of internal ingestion and ducting pressure losses to the freestream pressure; and $\eta_{\text{hull/inlet}}$ (or $\eta_{h/i}$) characterizes the effect on powering performance of the interaction of the inlet and hull and will be defined as $(1 - t)$, where $(1 - t) = R/T$, since the propulsor produces drag over and above the bare hull drag. This additional drag is an effective reduction in output of the waterjet.

The equivalence of terms on either side of the following equality can now be demonstrated;

$$\frac{T(1-t)V/550}{P_D} = \eta_{\text{pump}} \times \eta_{\text{jet}} \times \eta_S \times \eta_{h/i} \tag{7}$$

$$\frac{T(1-t)V/550}{P_D} = \frac{\rho gQH/550}{P_D} \times \frac{V}{V + \Delta V/2} \times \eta_S \times (1-t) \tag{8}$$

Rewriting pump efficiency as $(T_pV_p/550)/P_D$ instead of $(\rho gQH/550)/P_D$ and noting for a simple pump (ducted propeller, see Fig. 2)

$$V_p = V + (V_j - V)/2 = V/2 + V_j/2 \qquad T_p = T \tag{9}$$

all terms across the equality cancel, and

$$\eta_{D_{\text{waterjet}}} = \eta_{\text{pump}} \times \eta_{\text{jet}} \times \eta_{\text{system}} \times (1-t) \tag{10}$$

Experimental determination of η_D for a waterjet-propelled craft requires tests of bare hull, appended, and self-propelled conditions, as discussed later. References pertaining to experimental evaluation (boat tests)[3,14–18] provide data on η_T. Reference 16 attempts to correct jet thrust with the reduction in thrust due to inlet drag.

η_T does not completely describe the performance of a marine craft. Conceivably, the magnitude of propulsor thrust can be large in comparison to the thrust of a second propulsor, and, consequently, P_T would be high. However, a significant percentage of the propulsor thrust may be required to overcome the hull resistance that the propulsor itself adds. Thus it is possible to have a high P_T but, at the same time, a high P_S to propel the craft at design speed. Although the resulting η_T might be comparable with that of another propulsor or propulsion configuration, P_S required could be considerably larger. This was borne out in system tradeoff studies by the Boeing Company.[19] For comparison of various propulsion system installations on a particular craft, the most meaningful parameter is P_S.

In several reports, attempts are made to calculate a propulsive efficiency for a waterjet installation, either as a product of pump, jet, system and hull interaction efficiencies, or as a product of external, internal, and jet efficiencies utilizing momentum considerations.[3,4,14,19,21–28]

Ideal Propulsive Efficiency (η_j)

η_j can provide insight into the ideal performance of a particular waterjet system in stages of preliminary design. The derivation of η_j, based on momentum theory for one-dimensional flow and referring to Fig. 3, is as follows. Assuming $V = V_{\text{craft}} = V_{\text{inlet water}}$,

$$\eta_j = \frac{\text{power out}}{\text{power in}} = \frac{\text{work per unit time by thrust}}{\text{work per unit time by ideal pump}} =$$

$$\frac{P_o}{P_i} = \frac{(T \times V)/550}{P_i} = \frac{\rho QV_i(V_j - V_i)}{\frac{1}{2}\rho Q\{[V_i + (V_j - V_i)]^2 - V_i^2\}} =$$

$$\frac{\rho QV\Delta V}{\frac{1}{2}\rho Q(V^2 + 2V\Delta V + \Delta V^2 - V^2)} = \frac{V\Delta V}{V\Delta V + \Delta V^2/2} \tag{11}$$

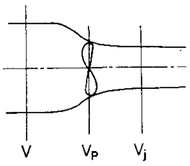

Fig. 2 Pump velocity schematic.

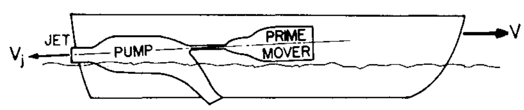

Fig. 3 Schematic of waterjet installation.

Ideal or Jet Efficiency

$$\eta_j = \frac{2\,V\Delta V}{2\,V\Delta V + \Delta V^2} = \frac{1}{1 + \frac{1}{2}\Delta V/V} \tag{12}$$

The ideal efficiency η_j can also be expressed conveniently in terms of the propeller thrust coefficient

$$C_{T_p} = T_2/\tfrac{1}{2}\rho\, V^2 A_p$$

Since $T_p = A_p \times \frac{1}{2}\rho(V_j^2 - V^2)$, then

$$\eta_j = \frac{TV}{P_i} = \frac{2}{1 + V_j/V} = \frac{2}{1 + (1 + C_{T_p})^{1/2}} \tag{13}$$

which indicates that η_j depends only on the propeller load coefficient.

The expression for ideal efficiency η_j, Eq. (12), indicates that a maximum efficiency for a specified system loss exists at a particular value of pump head or of velocity ratio. If for high propulsion efficiency a low $\Delta V/V$ is adhered to, one must pump a large Q because $T = \rho Q \Delta V$. High Q through a pump requires either high flow velocity or large flow passages. The latter reduces pump losses; however, large dimensions mean high fluid weight in the pump. Thus, adjusting to optimum conditions for specific pump application is required.

Real System Propulsive Efficiency

Equation (12) indicates that η_j is dependent on the ratio of jet velocity (nozzle exit velocity) V_j to ship velocity V. For jet efficiency calculated where thrust and velocity developed are a measure of the power out, but where power in is increased to overcome real system loss, and with no assumption made about internal system head loss, the following approach can be taken:

Total loss = inlet duct loss + exit duct loss + elevation loss + nozzle loss. The pump output power P_{pump} is

$$P = \tfrac{1}{2}(W_1/g)V^2 = \tfrac{1}{2}(W_1/g)(V_j^2 - V^2) + W_{hi} + W_{xe}$$

where w_i = weight of water/unit time. Let $h = h_i + h_e + \Delta y$; then

$$\eta_{jpr} = \frac{(w_1/g)(V_j - V)V}{\frac{1}{2}(w_1/g)(V_j^2 - V^2 + 2gh)} = \eta_j \times \eta_{\text{system}} =$$

$$\frac{2(V_j - V)V}{V_j^2 - V^2 + 2gh} = \frac{2(V_j - V)/V}{V_j^2/V + 2gh/V^2 - 1} =$$

$$\frac{2(k-1)}{k^2 + 2gh/V^2 - 1} \tag{14}$$

Estimated or assumed system losses should be verified experimentally. However, initial estimates of waterjet efficiency can be determined by making preliminary estimates of component losses. Toward this end, it is interesting to compare expressions for waterjet system efficiency of real systems as proposed by three authors as follows:

1) Joseph Levy[21]

Assuming $H_L = K_A V_j^2/2g$

$$\eta_{jpr\text{Levy}} = 2(k-1)/[k^2(1 + K_A) - 1] \tag{15}$$

where

$$K_A = H_L/(V_j^2/2g)$$

2) V. Johnson[24]

Assuming $H_L = K_B V^2/2g$

$$\eta_{jpr\text{Johnson}} = 2(k-1)/[k^2 - 1 + K_B] \tag{16}$$

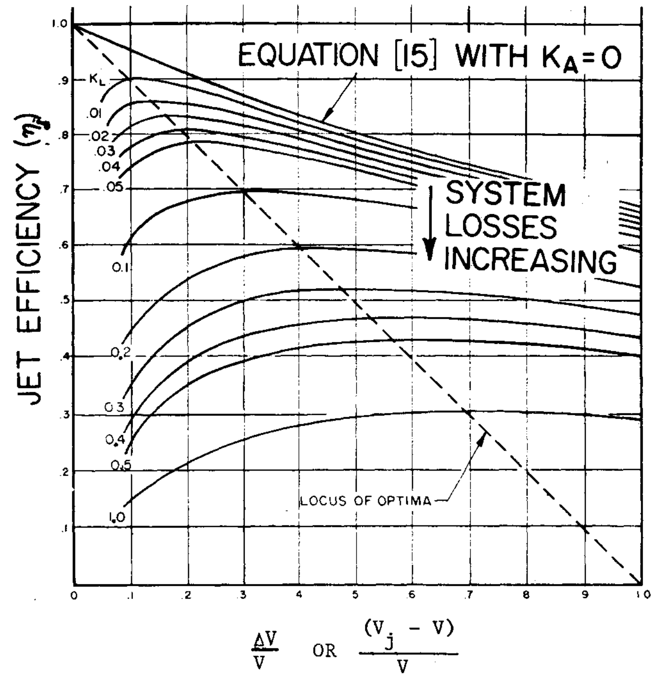

Fig. 4 Effect of system losses on optimum $V\Delta/V$.

3) Hatte and Davis[19]

$$H_L = [K_2 k^2 + [1/(k-1)]^2(\text{DLF})]V^2/2g$$

$$\eta_{jpr\text{Hatte-Davis}} = 2(k-1)/\{k^2(1 + K_2) - 1 + [1/(k-1)]^2\}\text{DLF} \tag{17}$$

where K_2 = nozzle loss coefficient, and DLF = duct loss factor. In the Hatte and Davis expression for η_{jpr}, the inboard losses are not lumped together and are not made proportional to either jet velocity or inlet velocity, as is the case of the Levy or Johnson expressions.

A familiar plot of jet efficiency vs velocity ratio for various values of the system loss coefficient K_L (see Ref. 26) is shown in Fig. 4. η_{jpr} must be multiplied by η_{pump} to obtain the efficiency of the over-all propulsion system. The accuracy of prediction of efficiency by the preceding methods can be no better than the accuracy of the loss coefficients used.

Comparison of Ideal Propulsive and Propeller Efficiencies

The uppermost curve of Fig. 4 corresponds to ideal propeller efficiency (propeller without viscous and rotative losses). By momentum theory given in Eq. (12) and referring to Fig. 5, this can be written

$$\eta_j = \frac{1}{1 + \frac{1}{2}\Delta V/V} = \frac{1}{1 + \frac{1}{2}W_a/V}$$

or ideal propeller efficiency. $\eta_{j\text{propeller}}$ can also be derived from a consideration of the velocity vector diagram for a blade element.[70]

Confusion exists in referring to the magnitude of comparative hydrodynamic performance of propellers and waterjets for high-speed hulls. It has been implied (e.g., Ref. 4) that propeller efficiency drops at high speed while waterjet efficiency increases. However, supercavitating propellers whose hydrodynamic efficiencies do not degrade with increased speed should be utilized in the higher speed range, where the performance of subcavitating propellers falls off.

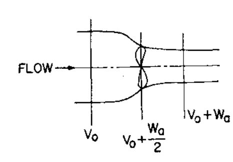

Fig. 5 Propeller velocity schematic.

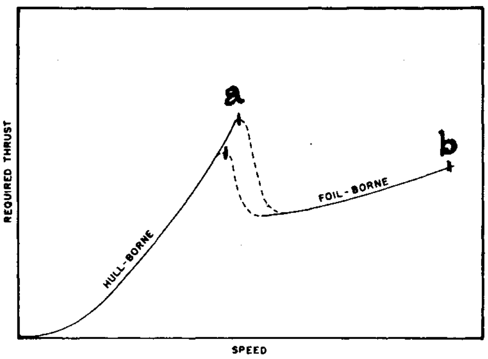

Fig. 6. Typical thrust-speed relation.

The reason that performance of some supercavitating propellers reduces as speed increases is that optimum blade geometry must be sacrificed because of structural problems. There is no positive indication that "long-life" high-speed pump impellers will not face the same problems.

Inherent limitations in the use of thrust efficiency, even when measurements of variables included in η_T are very accurate, should be recognized. Meaningful efficiency evaluation for waterjets should, as in the case of more conventional propulsion, involve determination of $\eta_D \cdot \eta_{D_{\text{waterjet}}}$ as given by Eq. (10), and $\eta_{\text{hull/inlet interaction}}$ by Eq. (9). When circumstances require determination of propulsion system component efficiencies (e.g., where performance improvement or research is the goal), η_D broken down in this way is a satisfactory parameter. For comparison of waterjet with a conventional propeller, if R is the bare hull towed resistance (i.e., without propeller system appendages or, on the other hand, waterjet inlet or exhaust ports), then the ratio of η_D's is the ratio of P_S's at a particular speed. A comparison of η_T for the two systems should not be a valid indication of relative propulsion efficiencies.

Pump

The waterjet propulsion pump design must be compatible with all other components of the propulsion system. Pump requirements and characteristics are treated in Refs. 2 and 21–24, with respect to research and application. The function of the pump in a waterjet system (as in the case of a propeller) is to accelerate the surrounding fluid medium or to increase the energy of the flow and thereby produce thrust. As discussed, the optimum jet velocity ratios for practical waterjet system efficiencies are relatively low. Thus, a pump with characteristically high mass flow rate and low head performance is a major requirement for efficiency. However, a high-speed pump will keep duct size and waterjet weight small.

The types of pumps (broadly classified under turbomachinery) which are suitable for waterjet systems include 1) centrifugal (radial flow)—high head, 2) mixed flow—intermediate head, and 3) axial flow (propeller)—low head. One or more pumps may be employed for each separate ingestlon and discharge duct. Several pumps in series or in a multistage unit cause the same mass flow to be handled by each stage with the head increased with each stage and with rpm lower than required by a single stage. In parallel pumps, each pump develops the same head and the flow is shared. These pumps could be of the same type as a single large pump but with higher rotative speed. Parallel pump arrangements usually require complex plumbing systems.

The pump type can be based on the required head and flow rate determined from the thrust-speed characteristics of the craft. However, innovations are usually required to adapt a design to conform to a specific waterjet application. One major consideration is the establishment of the off-design conditions that affect the cavitation performance of the pump. Pump cavitation becomes a major problem at the two operating conditions for hydrofoil and planing-type hulls shown in Fig. 6, a) at or near takeoff when the pump impeller is turning at a high speed and the total inlet static head is low due to a low ram head and b) at cruise or very high planing speed when the impeller tip speed is highest. The second condition corresponds to the limiting speed of a conventional subcavitating propeller, but in the waterjet case, control of the local pressure at the pump can delay cavitation inception.

The requirements of waterjet pumps for high-speed marine vehicle applications are such that their cavitation parameters fall in the same range as those of turbopumps for space rockets. Consequently, work has been done in developing waterjet pump impellers of the latter design. Rocket pumps, however, have limited lives. Not all existing axial or mixed flow pumps now used in waterjet systems employ inducer-type impellers. Buehler "turbopower" units and Hamilton waterjets (from which both the Buehler and the British version Dowty-Hamilton designs were evolved) are axial flow pumps employing multiblade impellers with single- and multistaging. Inducer-type rotors are being developed by Curtiss-Wright[29] and by Pratt & Whitney[30] Division of United Aircraft Corporation. Other rocket pump manufacturers are experimenting with either subcavitating or fully cavitating inducer designs for high-speed waterjet propulsion. The Aerojet-General Corporation has developed a pump in which the casing or volute functions as the jet nozzle with the purpose of directly converting mechanical energy to jet kinetic energy with minimum prior conversion to potential energy.[31]

Experimental research on pumps is complemented by theoretical approaches. The latter follow either a classical approach in which momentum relations[2,22,31] (and textbooks[32–35]) are utilized, or a lifting-line[39] or lifting-surface theory. Simple cavitation analyses can be made by applying cascade techniques[30,36–38] to a classical solution. For a more detailed analysis of blading and internal flow, however, lifting-line[37] or lifting-surface[20] approaches are useful.

With regard to experimental work on pumps for waterjets as indicated in Table 1, hydraulic performance is reported in Refs. 14, 15, 20, 29, 31, 36–38, and 40–43; certain of these also contain cavitation data.[14,15,29,37,38,40–43] Most testing has been done in cavitation loop facilities in which impeller scaled models are used. Such facilities usually provide visual observation of cavitation. Several large-scale static pump test stands capable of supporting fairly high-power prototype waterjet pumps are in operation, e.g., at Aerojet-General Corporation,[31] Azusa, Calif., and at Pratt & Whitney Division, West Palm Beach, Fla.

Just as good waterjet propulsion pump design differs from stationary-type pump design,[32,35,44] the terminology and performance coefficients of pump technology may not be ideal for waterjet pumps.[33] In addition, published performance limits,[45] should be examined in light of new technology. An analysis based on pump and waterjet pump performance parameters, dimensional analysis, and comparison with coefficients used in propeller technology follows.

At corresponding points in full-scale (or large-scale) and model flows, the flow should be geometrically and dynamically similar.[46] Relating the pressure differences through a simple pump to the pertinent variables and assuming that significant pressure differences are produced by mass forces of the fluid,[32] $P/\rho V^2 = Nd^3/Q$. Thus, a relationship between the rate of flow must exist if dynamic similitude of the hydrodynamic pump action is maintained from prototype to model. In pumps, pressure rise is usually given as head.

Viscosity cannot be neglected in pumps as it can in open propellers. A pump develops thrust T through the pump-head rise H and the volume flow rate Q. H depends on the density ρ and viscosity μ of the fluid, the volume flow Q, the impeller diameter d, and rotation speed N. Treating the product gH as a dependent variable and assuming[13] $gH = f_1(\rho,\mu,Q,d,N)$, dimensional analysis provides the simpler relationship,

$$gH/N^2d^2 = f_2[Q/Nd^3, \rho Nd^2/\mu]$$

These three dimensionless groups are head coefficient K_v, flow coefficient φ, and Reynolds number Re, respectively. Choosing performance parameters in propeller and pump work rests mainly on convenience. In propellers, V and T are easy to measure, whereas in pumps, Q and H are more readily determined.

A brief review of the performance parameters usually used for a propeller follows:

$$T = K_t\, \rho N^2 d^4 \tag{18}$$

and

$$\begin{aligned} P_S &= 2\pi N K_{\text{torque}}\, \rho N^2 d^5/550 \\ \eta_{\text{propeller}} &= P_T/P_S = (K_t/K_{\text{torque}})\cdot(J/2\pi) \end{aligned} \tag{19}$$

where $J = V(1 - w)/Nd$. Slightly different performance coefficients are useful for pump evaluation,

$$\begin{aligned} \eta_{\text{pump}} &= \frac{\text{pump hp}}{P_S} = \frac{P_p}{P_s} \qquad P_p = \frac{\rho g Q H}{550} \\ P_S &= \frac{2\pi N \text{ torque}}{550} \end{aligned} \tag{20}$$

where torque $= K_{\text{torque}}\, \rho N^2 d^5$.

As in the case of a propeller, P_P and P_S can be related to the dimensionless head, flow (as derived from dimensional analysis), and torque coefficients, respectively. Thus $P_P = \rho g Q H/550 = (\rho Q\cdot qH)/550$ where $gH = K_H N^2 d^2$ and $Q = \varphi N d^3$. Therefore

$$P_P = (\rho\varphi N d^3\cdot K_H N^2 d^2)/550 = \rho\varphi K_H N^3 d^5/550$$

Likewise

$$P_S = (2\pi N\cdot K_{\text{torque}}\rho N^2 d^5)/550$$

Dividing P_P by P_S,

$$\begin{aligned} \eta_{\text{pump}} &= \frac{P_P}{P_S} = \frac{\rho\varphi K_H N^3 d^5}{550} \times \frac{550}{2\pi N\; K_{\text{torque}}\rho N^2 d^5} \\ \eta_{\text{pump}} &= \frac{\varphi K_H}{2\pi K_{\text{torque}}} = \frac{K_H}{K_{\text{torque}}}\cdot\frac{\varphi}{2\pi} \end{aligned} \tag{21}$$

where $\varphi = Q/Nd^3$, $K_H = qH/N^2d^2$. Comparing (19) and (21)

$$\eta_{\text{propeller}} = \frac{K_T}{K_{\text{torque}}}\cdot\frac{J}{2\pi}$$

$$\eta_{\text{waterjet pump}} = \frac{K_H}{K_{\text{torque}}}\cdot\frac{\varphi}{2\pi}$$

η_{pump} multiplied by the jet efficiency η_j can be written in a form that is equivalent to $\eta_{\text{propeller}}$ as explained later. For propeller performance evaluation, K_T and K_Q are usually plotted against J. For a waterjet pump, K_H and K_Q could be plotted against φ or against V_j/V the jet velocity ratio, if the pump is operating in a moving craft.

Pump designers use "specific speed," originally introduced in 1915 by a German, R. Camerer, for describing the hydraulic type of water turbines. Each type of pump mentioned previously covers a range of specific speeds N_S, and its maximum efficiency drops on both sides of a particular N_S. Geometrically similar pumps have similar head flow performance characteristics at the same specific speed (assuming viscous effects are small). N_S is proportional to impeller rotative speed and rate of flow, and inversely proportional to pump head rise. Unfortunately, it is usually used in dimensionless form, which causes confusion in pump research work. Despite this, it is useful for comparing performance of different pumps or of pumps with their models.

Specific speed involves only the pump operating conditions by eliminating impeller diameter between the head and flow coefficients. By the provision that rotational speed can be obtained[32] as

$$N_S = NQ^{1/2}/H^{3/4} \tag{22}$$

This is made dimensionless by utilizing gH in the denominator, instead of H:

$$\text{dimensionless } N_S = NQ^{1/2}/(gH)^{3/4} \tag{23}$$

Addison[33] suggests that (23) be called characteristic shape number, with the word shape used because modelling requires geometric similarity or retention of shape. Substituting the coefficients φ and K_H into (23),

$$N_S = \varphi^{1/2}/K_H{}^{3/4} \tag{24}$$

A characteristic shape number could likewise be defined for propellers as

$$N_S = J^{1/2}/K_t{}^{3/4} \tag{25}$$

A point of interest in comparing propeller and pump performance efficiencies is exemplified by the case of a ducted propeller for which "efficiency" has been calculated by, first, a propeller efficiency philosophy and, second, by a pump efficiency philosophy. As a result, different velocities are used in

$$\eta_p = \frac{T\cdot V}{550} \div P_D \tag{26}$$

For the propeller $V = V_a$, speed of propeller advance; for the axial flow pump, $V = V_d$, velocity inside the duct. Since the induced velocity inside the duct causes the pump flow velocity to exceed freestream velocity, the "apparent" efficiency of the pump will be higher. Figure 7 shows a set of comparative efficiency curves for aducted propeller,[47] calculated as pump and as propeller and showing a significant difference.

Note that $\eta_{\text{propeller}}$ and η_{pump} differ because of the difference in V_a (speed of advance) and V_d (speed of flow at propeller disk)

$$V_d = V_a + w_a/2$$

where w_a = total axial induced velocity component or ΔV. Thus,

$$\eta_{\text{propeller}} = TV_a/(550\cdot P_S) \tag{27}$$

$$\eta_{\text{pump}} = \frac{TV_d}{550\cdot P_S} = \frac{T[V_a + \Delta V/2]}{550\cdot P_S} \tag{28}$$

and

$$\frac{\eta_{\text{propeller}}}{\eta_{\text{pump}}} = \frac{V_a}{V_a + \Delta V/2} \tag{29}$$

where

$$\frac{V_a}{V_a + \Delta V/2} = \eta_{\text{jet}}$$

Severe cavitation may affect the pump by causing erosion and degradation of performance. Because waterjet propulsion pumps will, at times, be required to work at relatively low suction heads, cavitation of the impeller will be a problem.[37] An advantage of the axial flow pump is that multistaging of impellers can reduce the tendency for the main load-carrying impeller to cavitate. With an inducer stage in which the static pressure is increased with little kinetic energy increase, it is possible to provide a sufficiently high suction pressure to the main flow accelerating impeller stage to prevent blade cavitation.

A comparison of cavitation inception criteria for pumps[35] and propellers follows. For pumps usually used to produce

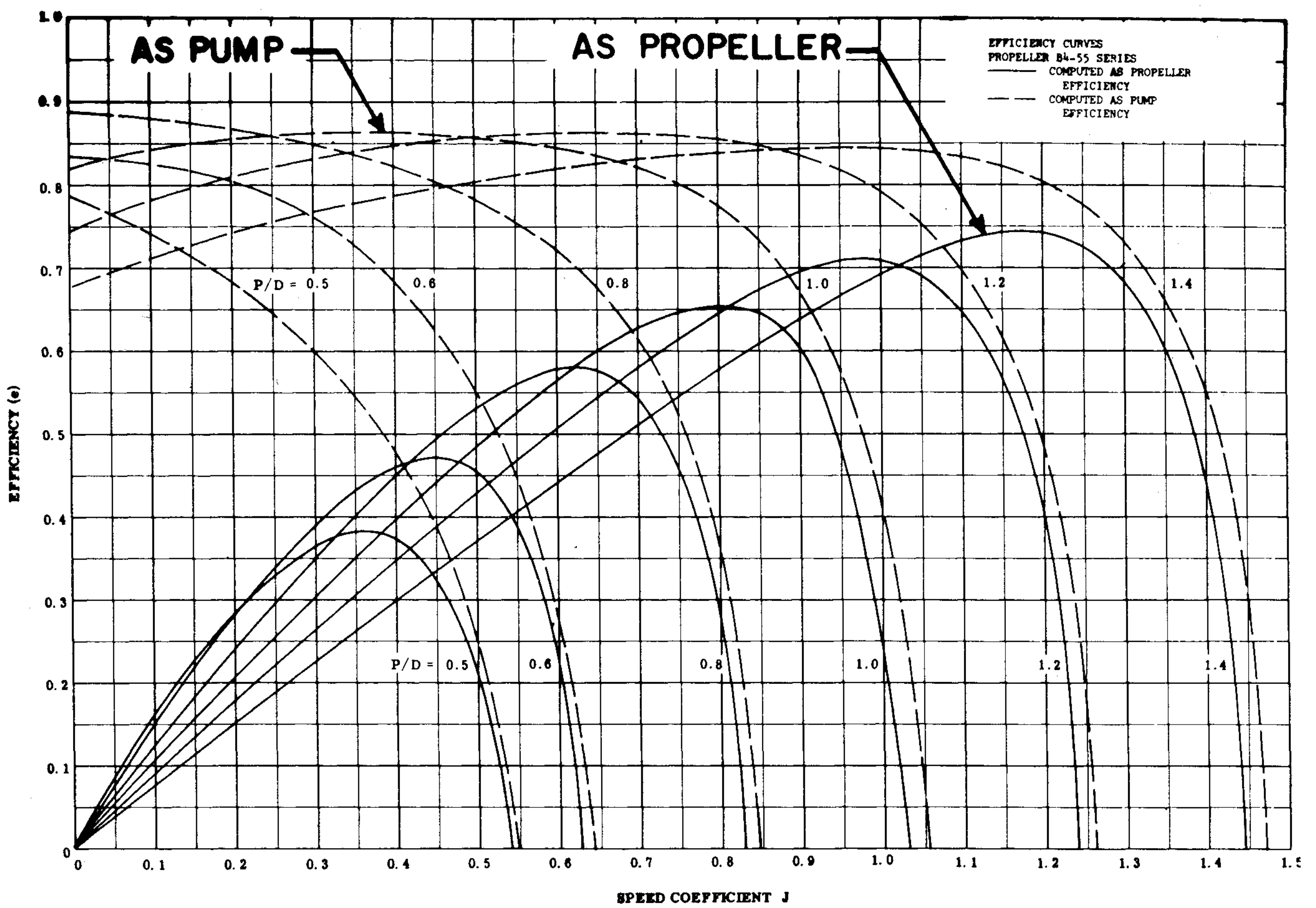

Fig. 7 Efficiency of a ducted propeller calculated by two methods.

a pressure rise through the impeller, the Thoma cavitation parameter

$$\sigma_{\text{Thoma}} = \frac{\text{net suction head}}{\text{pump head produced}} \tag{30}$$

is in general usage. Here, net suction head

$$H_{sv} \text{ (or } npsh) = h_{\text{atmospheric}} + h_{\text{depth}} + (V^2/2g) \quad \text{in ft of water} \tag{31}$$

where V is velocity in the flow approaching the pump. The Thoma parameter is dimensionless, and if its constancy indicates cavitation similarity, then increased pump head requires increased suction head.

For propellers where increase in kinetic energy is desired, the cavitation index is normally used

$$\sigma = \frac{H_1}{V^2/2g} \tag{32}$$

where H_1 = absolute pressure at shaft centerline minus vapor pressure of water and V can be either freestream or inflow velocity at the center section of the propeller.

Effects of cavitation on propeller performance are normally presented through plots of K_T, K_q (torque), and efficiency vs J obtained at various values of σ from tests conducted in a cavitation tunnel, following open-water tests. In pump procedures, a cavitation parameter that relates inlet flow conditions to pump speed and head rise produces a number of large magnitude. It is a direct extension of the specific speed N_S, and is

$$S = NQ^{1/2}/(gH_{sv})^{3/4} \tag{33}$$

where H_{sv} has replaced H. S is termed suction specific speed and appears to be a valid dimensionless parameter.

The Thoma cavitation index σ_t is dependent on the pump itself, being the ratio of the net suction static head to the pump and the pumping head developed. Thus, $\sigma_t = (N_S/S)^{4/3}$. In contrast, the propeller cavitation index σ is dependent on the condition of the flow, as the ratio of static head to dynamic head of the freestream flow. σ can, however, by expressed in terms of V_A (resultant). The use of V_A makes

$$\sigma_A = (p_o - p_v)/\tfrac{1}{2}\rho V_R^2 \tag{34}$$

analogous to pump σ_t.

In summary, pump efficiency must be multiplied by η_j before direct comparison can be made with conventional propeller efficiency. Pertinent performance parameters for waterjet pumps will include flow rate φ, head coefficient K_H, torque coefficient K_q, characteristic shape number (nondimensional specific speed) and a nondimensional form of suction specific speed. Flow rate and head are measured, since they are more directly attainable than in propeller testing. Cavitation number can be defined in various ways (this variation usually arises for propellers because of freedom in where velocity is measured), but for propulsion pumps it can also be based on the ratio of suction head H_{sv} to head developed by the pump H_p.

Waterjet Inlet

Waterjet propulsors require optimization from inlet to exhaust nozzle. As pointed out previously, it does not pay to strive for a high system efficiency through low jet velocity unless system losses are also minimized (see Fig 4). Component losses consist of inlet-diffuser from 10 to 30%, internal ducting from 2 to 10%, and nozzle from 1 to 3%. The losses stated are estimates for a range covering typical planning hull, CAB, and hydrofoil waterjet systems. A sharp bend just behind the inlet and long vertical strut duct, as required for a hydrofoil craft, will produce higher losses. These losses are usually expressed as percent of the ram head $V^2/2g$.

The inlet or scoop location depends on the hull and waterjet configuration. Most hydrofoil waterjet foilborne systems incorporate ram-type inlets in the pod at the bottom of the strut.[24] Planning boat and displacement craft usually employ flush or scoop intakes in the hull bottom. Schemes have been proposed for side or bow located inlets.[72,73] Major location considerations include keeping short the vertical distances that produce elevation losses, keeping the inlet in "green" water to prevent aeration, providing sufficient water flow to the pumps to produce required thrust, and preventing or delaying inception of cavitation. Good inlet design requires low internal losses and high resistance to cavitation during takeoff and low external drag during cruise. From takeoff-mode cavitation consideration, the critical inlet parts are scoop (water entrance) and transition from inlet to diffuser duct. The scoop must capture the freestream flow efficiently between takeoff and cruise.

During takeoff, the ratio of inlet to freestream velocity is high (pressure levels in the scoop are low), and internal cavitation is a primary concern. During cruise, on the other hand, this velocity ratio is relatively low (pressure levels in the scoop are thus relatively high), and resistance to internal cavitation is therefore high. In the cruise mode, susceptibility to external lip cavitation may be significant.

The net effect of cavitation in and around the inlet is twofold. Cavitation can produce significant erosion of the inlet material. However, a more important effect on pump performance is excessive inlet/diffuser head loss and, ultimately, complete choking of the flow (which starves the pump). Variable geometry inlets could assure flow to the pump at low speeds (takeoff) and allow the pump to work at design point over a fairly broad range. However, mechanical complexity in seawater of such an inlet and cavitation problems of the fairing appear significant.

Definitive evaluation of inlet performance involves determination of 1) how efficiently the water can be "captured" or over what range of velocities inlet cavitation will not significantly degrade this "capturing," and 2) how much the hull drag is affected by the presence of the inlet. An analysis of the parameters in model-scaling hydrodynamic forces on submerged bodies is given later.

The literature contains examples of both a review of older published empirical information for application to waterjet inlet, and some original analytical approaches. The latter, for the most part, attempts to predict effects on the inlet of impinging flowfields by calculating the pressure distribution in external and internal areas adjacent to the lip. These analyses do not provide the inlet loss coefficient, but aid in the design of lip and entrance angles, which vitally affect pressure recovery.

Methods currently in use are based on solutions to the potential flow problem, using both linearized and nonlinearized techniques. A two-dimensional method utilizes a distribution of sources on the mean camberline of the lip (see Ref. 48). A more general result that applies to the nonlinear problem can be achieved with a distribution of sources along the surface of the model, for example, the Douglas-Neumann approach to arbitrarily shaped inlets.[22,49,50]

Pressure distributions obtained from the preceding techniques affirm the character of the streamline flow and show whether smooth or sharp peaks exist which could indicate separating flow. Viscosity effects, not included as part of a potential flow solution, require empirical corrections.

It may be possible to utilize existing empirical data on marine condenser scoops[23,24,51-54] and aircraft oil cooler scoop,[55,56] especially in preliminary inlet designs. Some inlet designs (when considered integrally with a particular hull flowfield) could be low in internal loss but have high

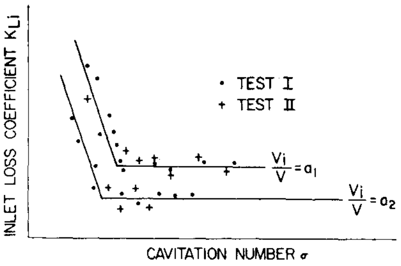

Fig. 8 Inlet internal performance.

external drag. Experimental work is required to determine net performance of such designs. References 14, 19, and 22–24 are valuable for predictions of system performance, using existing empirical data.

Practically all of the inlet studies carried out so far have been slanted toward hydrofoil inlets. The biggest present need is for comparable studies for flush planning hull scoops and ram or flush CAB sidewall-type inlets. The ram, or captured freestream dynamic pressure, recovery through the inlet can be determined from the characteristics of the approaching and leaving water. An inlet with poor flow will experience a large energy or head loss, $V_1^2/2g + H_s$. This could be measured before and after the flow passes the inlet lip and turn/vane (if one exists).

Experimental data (Table 1) from relatively recent testing of waterjet propulsion system inlets are in Refs. 14, 16, 18, 22, and 57. The system of Ref. 14 employs dual wake intakes to provide more uniform inflow. The work reported in Ref. 22 was obtained from hydrofoil craft nacelle-type model tests in the Lockheed Underwater Missile Facility (LUMF) at Sunnyvale, Calif. (variable-pressure towing basin). The data of Ref. 57 were for scale-model inlet tests of a strut-type inlet in the Hughes Aircraft (Tool Division) freejet facility at Los Angeles.

The quantities normally measured in inlet studies include craft velocity V, inlet velocity V_i, inlet geometry (lip, angle, size), guide vane geometry, static pressure, pressure distribution, and velocity profiles. Typical parameters determined from these data include pressure loss coefficient, velocity ratio (V_i/V), geometry, drag coefficient, pressure distribution, velocity profile, critical cavitation number (σ_c), and net positive suction head (*npsh*).

One method of displaying inlet performance is to plot inlet loss coefficient K_{Li} (as percent of freestream velocity head) vs cavitation number for a family of inlet velocity ratios $V_i/V_{\text{freestream}}$. This produces characteristic "break" curves of the type described in Ref. 19 and shown in Fig. 8.

Some inlet instrumentation was employed in boat testing reported in Refs. 3, 14–16, 18, and 58. The Hydronautics Inc. data[18] are from stationary model tests with moving flow.

Summarizing, the waterjet hull inlet is critical with regard to both external and internal flow performance. Analytical methods can aid in the design of lips, inlet angles, etc., in predicting pressure distribution. Inlets for craft which operate from displacement to planing conditions may require variable geometry designs to accommodate both takeoff and cruise. In takeoff, the internal inlet can choke the pump because of cavitation, whereas in cruise the external inlet can cavitate, thereby increasing drag. A good inlet should provide adequate flow, have low K_L, and be cavitation-resistant.

Waterjet Ducting

In a waterjet system, ducting is necessary to move the water from inlet to pump and from pump to exhaust nozzle. Waterjet hydrofoil boats utilizing strut-pod inlets and deck-mounted pumps require relatively extensive ducting. Small planing boats can have very short ducting, and most commercial waterjets incorporate inlet, diffuser, pump casing, and exhaust nozzle in only two or three castings. Captured-

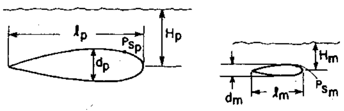

Fig. 9 Schematic of body flow.

air-bubble craft will require waterjet ducting systems of less complexity than for hydrofoil craft.

Layout studies to optimize system designs have been made for specific type hull waterjet installations by Hydronautics Inc., for CABS[59] and Pratt & Whitney Division for hydrofoil craft, air-cushion craft, and large displacement-type ships (destroyers).[60] In addition, Lockheed has published some system layout work[22] and Ref. 24 by Johnson of Hydronautics contains some hydrofoil craft ducing-system layouts. Kim[23] treats the tradeoff of benefits of single ducting vs multiple ducting systems, and the relative importance of bends, straight lengths, vanes, etc.

The duct should deliver the correct quantity of water to the entrance of the impeller efficiently (with minimum pressure loss) and with reasonably uniform velocity distribution, and then exhaust the water efficiently. The efficiency of the ducting, as in any piping system, is dependent on the length of the pipe, number and type of transitions, and pipe roughness. A duct system should be lightweight (including weight of water) and have low hydrodynamic loss. Considerations of low weight and vibration (structural and internal hydrodynamic) are not covered here. Hydrodynamic losses, which will be discussed, involve friction at internal walls, mixing (pressure loss and friction loss) in areas where transitions (enlargements, contractions, and turns) exist, and velocity profiles at sections approaching the impellers and stator vanes. An example that might cause significant problems is an impeller shaft of an axial-type pump that passes through the supply duct and causes nonuniform impeller inflow.

Table 2 Requirements for performance evaluations

Type of study	Requirements
Performance	Determination of power to propel a craft at certain speeds, or comparison of propulsion system designs
Design	Determination of performance of components in order that improvement in their design may effect better over-all system performance
Research	Determination of performance characteristics of components in a systematic experimental program in which the components are not related to a particular design or system

The flow of an incompressible fluid inside a filled duct produces significant inertial, gravity, friction (viscous), and pressure forces. Surface tension, as characterized by Weber number, and compressibility, by Mach number, can be ignored in waterjets. It is common practice to use the following dimensionless parameters

$$Re = \frac{\text{inertial}}{\text{viscous}} = \frac{\rho V l}{\nu} \quad \text{Reynolds number} \tag{35}$$

$$F_n = \frac{\text{inertial}}{\text{gravity}} = \frac{V^2}{gl} \quad \text{Froude number} \tag{36}$$

$$\frac{\text{pressure}}{\text{inertial}} = \frac{\Delta p}{\frac{1}{2}\rho V^2} = \frac{p - p_o}{\frac{1}{2}\rho V^2} = \text{pressure coefficient} \tag{37}$$

where l may be pipe diameter.

These parameters establish dynamic similitude of flow in models of ducting systems. The geometric similarity be-

Table 3 Performance measurements and parameters

Type testing	Measurement	Range	Derived performance parameters	Function
Pump static	Net positive suction head $(H_{sv})^{*a}$ static pump head, flow rate, rpm, shaft torque, impeller inflow, static thrust, velocity distribution; also, if pump intake pressure is controllable, cavitation effect on performance, cavitation breakdown	Design range	Flow coefficient ϕ, head coefficient K_H, torque coefficient K_q, η_{impeller}, η_{pump}, specific speed N_S, suction specific speed S, Thoma cavitation number σ_t	Static pump performance and mechanical reliability, cavitation breakdown
Pump dynamic	H_{sv}, pumphead, flow rate, rpm, cavitation observations under transient intake flow conditions	Design rpm range, range of inlet velocity ratio (V_i/V), range of exhaust nozzle areas	Same as above	Over-all system performance and reliability under real environmental conditions
Water hull-inlet dynamic	Head loss (pressure drop), cavitation inception speed, inlet velocity ratio (V_i/V), inlet drag, inlet pressure distribution, visual cavitation observations, prediffusion and boundary-layer ingestion	Hi-suction/low head at take-off to maximum ship speed or cruise and dash condition	Inlet velocity ratio V_i/V, cavitation number σ, inlet loss coefficient K_{Li}	Optimization of inlet geometry to provide required flow, low internal losses, low drag, high cavitation resistance
Internal intake ducting	Head loss, flow velocity profiles, cavitation observations	Over speed range	Duct loss coefficient, K_{Ld}	Optimization of size and geometry for pump inflow velocity profile and minimum duct losses
Internal exhaust ducting	Head loss	Over speed range	Duct loss coefficient K_{Ld}	Optimization of size and geometry for minimum duct losses
Nozzle	Head loss	Over pump speed range or jet velocity ratio (V/V_j), over range of nozzle exit areas	η_j	Optimization of size and geometry for optimal nozzle-pump matching

[a] Net positive suction head H_{sv} is equivalent to the total static head at the pump entrance.

tween prototype and model should extend to roughness of the internal duct. Nondimensionally, this would make relative pipe roughness (e/d) equivalent for model and prototype. Relative to viscous and pressure forces, gravity forces are minor in the flow through a filled pipe. Consequently, Froude number can usually be ignored in model studies. However, Froude number may be of significance in certain instances. Although it is often associated with free surface gravity effects on flow about bodies, to provide the correct pressure coefficient or cavitation number over the vertical extent of a model, Froude scaling must be maintained. Thus, where the vertical dimension of the body is large relative to the depth of immersion, it may be necessary to consider Froude number in order to properly scale pressure coefficient. Considering Fig. 9, and setting C_p of model and prototype equal, at corresponding points (P_{sp} and P_{sm}),

$$C_{p_{\text{prototype}}} = \frac{\rho g H_p - p_0 - \rho g \tau}{\frac{1}{2}\rho V_p{}^2} = C_{p_{\text{model}}} = \frac{\rho g H_m - p_o - \rho g}{\frac{1}{2}\rho V_m{}^2} \tag{38}$$

If $\rho_p = \rho_m$ and $\tau = l_p/l_m$, $V_p/V_m = \tau^{1/2}[(1 - \sigma_m)/(1 - \sigma_p)]^{1/2}$ where

$$\sigma_m = (p_o - p_v)/\tfrac{1}{2}\rho V_m{}^2 \qquad \sigma_p = (p_o - p_v)/\tfrac{1}{2}\rho V_p{}^2$$

and if $\sigma_m = \sigma_p$ (equality of freestream cavitation numbers), then $V_p/V_m = (\tau)^{1/2}$, which corresponds to Froude scaling.

In preliminary design of waterjet systems, use can be made of published empirical data on pressure loss coefficients in straight ducting, turns or elbows, vanes, nozzles, and other transitions. References 13, 22–24, and 34 are good sources of such data. Waterjet ducting losses are included in the system performance measurements obtained during the boat tests of Ref. 14 (British). References 50 and 61 from Gibbs and Cox, and Ref. 60 by Arcand of Pratt & Whitney utilize published experimental ducting loss data in arriving at predictions of system efficiency. Johnson[24] suggests ranges of magnitude for system loss coefficients, apparently based on published empirical data. The data available in these references are summarized in Table 1. They show that frictional resistance is more significant in internal waterjet systems than in the open or even conventional ducted propeller. The ducting performance problem appears less critical than that of the inlet.

Waterjet Testing

Performance Considerations

Experimental determination of the performance of components or of entire waterjet propulsion systems can be accomplished by applying proper techniques to prototype or model tests. Where prototype testing is impractical, model experiments will generally produce satisfactory results if geometric and dynamic similitude are maintained. The type and scope of the testing will depend on the intended use of the data. Requirements for most performance evaluations are given in Table 2.

Testing of complete low-horsepower waterjet propulsion units under full-scale speed and load finds wide application. High-powered units, requiring high efficiency and reliability, necessitate the expense of constructing models and of using specialized test facilities.[31] Performance parameters presented in the discussion on pumps, inlets, and ducting, are required to characterize performance. Table 3 notes parameters that characterize hydraulic performance and lists measurements required in their evaluation for components of a waterjet system.

In general, experimental data are more difficult to obtain in dynamic than in static testing. All the suggested data have been obtained successfully in tests of prototypes and models of waterjet craft. In addition, photographic data on cavitation phenomena at the hull inlet has been obtained.

Calculation of waterjet thrust from measurements of mass flow rate is generally a satisfactory method. Thrust will have to exceed towed hull drag by an amount equal to the additional drag induced by the propulsion system flow.

Model Testing

Model testing is based on the satisfactory prediction of prototype performance from models by application of scaling parameters. Experience in pump testing indicates that scale effects can produce significant correlation problems if the physical size of a model is very small relative to the prototype.[33]

In model-testing inlets, the full-scale boundary layer of the inlet should be modelled. The similarity of the pressure coefficient and thus cavitation number at all points of the inlet flow will require Froude-scaling, in addition to maintaining the freestream cavitation number of the prototype. This is important when the vertical dimension of the inlet is large relative to the depth of submergence, or when it is desired to scale the pressure gradient in a diffusing inlet. The internal inlet flow depends primarily on testing above a critical Reynolds number, as does frictional stress of the duct wall.

In general, model testing allows closer control of test conditions, but requires proper scaling procedures. Considerable model testing should be performed when the propulsion system is relatively large, and must meet stringent performance requirements.

Testing a Waterjet System

For definitive propulsion performance determination of a propelled craft, resistance and propulsion forces must be separated. This usually requires a towing tank facility.[3] Testing of the prototype craft is ideal, if all pertinent parameters are measured. However, self-propulsion testing of a prototype craft in open-sea conditions (which is done for a great proportion of actual installed waterjet systems[3,14,16,17]) usually does not furnish sufficient information because it is difficult to separate resistance and propulsion. A definitive performance criterion is given in Eq. (10). Any good prediction of performance must include a reasonable estimate of $(1 - t)$. References 3, 4, 14, 19, and 21–26, which contain performance data, include calculations of η_T but not η_D. Experimental work performed to date is listed in Table 1.

Experimental Procedures

Examples, along with recommended applications, of methods of determining hydrodynamic propulsion performance characteristics of installed waterjet systems are given below.

1) Test a hull model with proper external propulsion system inlet characteristics. This is recommended for propulsion tests of the hull inlet/diffuser component. It requires a "sucker" pump at the waterjet inlet, and means to maintain full-scale cavitation number and to simulate full-scale boundary layer. Tests of LITTLE SQUIRT[19] and of inlets of the hydrofoil craft type in the LUMF facility utilized this technique.

2) Test a prototype craft at full-scale speeds in a towing tank and measure towed and propelled performance separately.

3) Test a prototype waterjet propulsion system mounted on a "mockup" of a portion of the prototype hull which is fixed in trim and heave. Such a test could be run in a towing basin or a circulating free-surface channel. An example of the latter is reported in Ref. 18. This procedure requires that the hull-inlet flow conditions be known and simulated on the "mockup."‡

‡ In testing waterjet ingestion systems in hull models or mockups mounted stationary in flow channels, it is doubtful that meaningful magnitudes of the added hull drag will be obtained from intersection of the induced flow by the waterjet system, unless the transverse hull dimension is small compared to channel width.

4) Test a prototype installation in open sea and measure P_S and P_T (by jet flow) for performance, or test a prototype craft in open-sea conditions and measure thrust reaction and inlet drag, in addition to the other usual pertinent quantities.

5) Test a prototype system or component on a hull on which auxiliary propulsors, as propellers or aircraft jet engines, are used to help propel the hull, so that the water pump can be tested at various combinations of thrust and ship speed.

In summary, the inlet and critical hull section can be model-tested independently of the rest of the waterjet propulsion system if an auxiliary sucker pump is used. For installed waterjet propulsion systems, a definitive test of performance requires separation of resistance and propulsion forces.

A type of "open-water" propulsion test for a waterjet propulsor could be run with a suitable hull simulation inlet. It requires testing a prototype waterjet over a range of trims, boat speeds, inlet velocity ratios, and yaw conditions in order to provide adequate performance data from which extrapolation can be made when actual hull dynamic characteristics become known.

Literature of a General Nature

References 22–24, and 61 relate to applying waterjet propulsion to certain mission craft. A much larger number of the references are on general considerations in applying waterjet propulsion to various craft. These include Refs. 1, 3, 4, 7–9, 19, 21–25, 50, 58, and 60–69. Apparently, the availability of high-speed computers has encouraged many feasibility studies which consider mission, structures, propulsion, etc.

Tradeoff analyses become critical to good system application studies. Systematic theoretical and experimental work are now needed on system design and performance evaluation. Feasibility and general studies require factual, verified scientific evidence on performance factors, such as loss coefficients, component efficiencies, and distortion at the interfaces between components (e.g., inlet diffuser to pump). In a final analysis, a propulsion system may or may not be feasible for a particular application because of actual system efficiency, and it is unlikely that this can be determined by guess work.

Summary

1) Waterjet propulsors are special marine propulsive devices whose advantages over propellers include elimination of underwater appendages and complex right-angled transmission systems in certain craft, and, in general, provide for a freer choice of propulsor location.

2) Disadvantages of waterjets include generally higher weight, more power to perform a particular function, and a hull inlet that provides additional cavitation problems.

3) Hydrodynamic performance of waterjet-propelled craft should be based on propulsive efficiency,

$$\eta_D = P_E/P_D = RV/P_D = \eta_{\text{pump}} \times \eta_{\text{jet}} \times \eta_{\text{system}} \times \eta_{\text{hull/inlet}}$$

where $\eta_{\text{hull/inlet}} = (1 - t) = R/T$. Precise determination of η_D is difficult; it is not necessary in comparing two or more propulsors on the same hull, in which case shaft horsepowers P_S should be used.

4) High η_D is required in designs where the ratio of fuel weight to gross weight is high, as in long-range ocean vehicles.

5) Momentum theory indicates that high jet efficiency requires a low ratio of jet velocity to craft velocity and low inlet and ducting losses. Calculation of real jet system efficiency by momentum theory is not an accurate approach to determining performance of an actual propulsion system.

6) The use of various efficiency parameters has confused performance comparisons. Propulsive efficiency, $\eta_D = RV/P_D$, has been confused with thrust efficiency, $\eta_T = TV/P_S$, and pump efficiency η_{pump} with propeller efficiency η_p, whereas

$$\eta_p = \eta_{\text{pump}} \times \eta_{\text{jet}}$$

7) The choice of performance parameters often depends greatly on the relative ease with which these terms can be obtained. Dimensionless waterjet parameters should include flow coefficient φ, head coefficient K_H, torque coefficient K_q, specific speed or shape number $Ns_{\text{dimensionless}}$, suction specific speed $Ss_{\text{dimensionless}}$, and for pumps, Thoma cavitation number σ_t.

8) Research is needed on waterjet inlet technology. Especially needed are hard (experimental) data for flush and semiflush/inlets designed for high-speed ships, including effects of yaw and sideslip.

Two basic shortcomings prevail within waterjet literature, first, a lack of experimental data and synthesized design methods and, second, loose definition of a number of design parameters. Progress in improving waterjets requires research and development work in propulsion system and hull design. This should be built on the use of adequate performance parameters and evaluation techniques.

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